Int. J. Therm. Sci. (2000) 39, 74–84

S1290-0729(00)00194-X/FLA

Blade cooling improvement for heavy duty gas turbine: the air coolant temperature reduction and the introduction of steam and mixed steam/air cooling

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(Received 4 March 1999, accepted 2 August 1999)

Abstract — This paper proposes a theoretical study of some alternative solutions to improve the blade cooling in the heavy-duty gas turbine. The study moves to the evaluations of the air coolant reduction temperature effects, considering two different methods: a water surface exchanger (WSE) and a cold water injection (CWI). A logical development of these possible cooling system improvements is the steam cooling application, particularly suitable for mixed or combined gas–steam cycles; the steam cooling is evaluated using open and closed loop configurations; the possible interaction of steam and air cooling is also studied. All the simulation is realized with a family of modular codes developed by authors and the study is conducted with the analysis of the characteristic cooling parameters (efficiency, effectiveness) and by the evaluation of blade temperature distribution. The study is related to a typical configuration of heavy-duty rotor blade with a standard air cooling scheme and the possible variations are related to coolant characteristics only. The results show the interesting possibility due to air coolant temperature reductions, particularly for the CWI method, but the steam cooling turns out to be more incisive. All of the considered techniques show the possibility of a mass coolant reduction and/or the possibility of a maximum cycle temperature increase in comparison to the standard air-cooling. The best results are obtained for an innovative closed–open/steam–air cooling system. © 2000 Éditions scientifiques et médicales Elsevier SAS

gas turbine / air cooling / steam cooling / high efficiency system / water injection / blade cooling systems

Nomenclature

A	cross section area	m^2
c_p	specific heat at constant pressure	$J \cdot kg^{-1} \cdot K^{-1}$
m	mass flow rate	$kg \cdot s^{-1}$
St	Stanton number	
s	bending length	m
T	absolute temperature	K
T_{firing}	total gas temperature at rotor row inlet	K
T_{flame}	total gas temperature at stator row inlet	K
ΔT_{nozzle}	hot gases total temperature drop during nozzle expansion	K
ΔP	pressure drop	Pa

Greek symbols

ε_h	cooling efficiency
ϕ	cooling effectiveness

Subscripts

average	average value
air	referred to the air cooling
b	blade
cool	referred to the cooling
ext	external
g	gas
H_2O	water
in	referred to the inlet
out	referred to the output
max	maximum value
nom	referred to the nominal conditions
steam	referred to the steam cooling
tot	total

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1. INTRODUCTION

In order to improve the gas turbine cycle performances, the cycle maximum temperature has to be increased. Moreover, to obtain an increase in thermodynamic efficiency and specific work too, the temperature growth has to be coupled with the compression ratio increase; this also implies a greater air coolant temperature from the compressor and more coolant flow to assure an adequate blade cooling.

Therefore, every performance improvement cannot be taken without an adequate development of the blade cooling system. In order to increase the cooling efficiency, two possible ways can be followed: either directly operate on the cooling system geometry or, with nonstructural interventions, suitably modify the thermodynamics features of the coolant air; for example, reducing the air temperature or replacing it with a fluid like steam, having a better heat capacity.

Concerning air coolant temperature reduction, several gas turbines manufacturers make provision for these possibilities, particularly when the cooling system performance is not sufficient to guarantee the operating life and reliability of the machine.

As far as the use of alternative coolants is concerned, since heavy duty gas turbines technology partially depends on aeronautical turbines production, for a long time steam was not seriously taken into account as a coolant, although its heat capacity is better than air. The 1980s showed a certain interest of research towards steam [1–3]: they essentially deal with results of theoretical studies or of some experience carried out in order to test steam heat properties. At present, the favorable results obtained and the increase of the heavy duty gas turbine production encouraged some of the biggest manufacturers, such as General Electric [4], to produce prototypes which employ steam as a blade coolant. Similar industrial plant realization has been announced for the present year.

At the Energy Department of the University of Florence, a research group has been studying turbines blade cooling in its various thermodynamic aspects [5, 6] and the possible solutions to improve the blade cooling system. In 1989, Facchini [7] carried out a thermodynamic analysis on cooled expansion supposing the use of fluids alternative to air, such as water and steam. A detailed evaluation of the cycle performance effects of coolant reduction temperature was conducted in 1995 by Bettagli [8]. More recently a procedure has been developed in order to study the temperature distribution of a cooled blade. The procedure is based on a complete study of heat transfer on the external side, on the internal side and through the thickness of the stator [5] and rotor blade [10].

Using these studies, this paper will evaluate the consequences of air coolant reduction temperature and blade steam cooling in different configurations.

2. RESEARCH AND SIMULATION CRITERIA

In order to determine the blade temperature distribution we consider [9–11]:

- the external heat exchanges between the blade and the hot gas,
- the heat flow across the blade thickness,
- the internal heat exchanges between the blade and the coolant.

The problem was then solved with the use of the following computational codes:

- TRAF2D, which studies the flow and the heat exchange outside the blade solving the Navier–Stokes equations in the blade to blade section,
- HT, which studies heat conduction through the blade wall by means of the two-dimensional solution of the Fourier equation,
- RBC, which studies the coolant flow and heat transfer inside the blade solving the one-dimensional flow equations throughout the cooling loop developing.

This family of codes already developed [9, 10] to estimate the effects of the use of steam blade cooling have been modified. On the calculation procedure, the TRAF2D and HT codes were used to repeat the simulation at different blade radial sections, whereas the RBC code was used to study the coolant flow in radial direction: the codes combined use, suitably interfaced, allowed to carry out an “almost 3D” analysis of the phenomenon.

The object of our study was a rotor blade profile similar to one produced by Nuovo Pignone S.p.A. [10], equipped with a multipass cooling system (*figure 1*) made up of three serpentine sections, the last of which is provided with pin fins, and a series of small axial tubes near the trailing edge. The choice of a quite simple configuration, which has already been employed in the heavy duty turbines field also for medium–small power size, is due to the goal of assessing the effects of coolant temperature reduction and introduction of steam at the present state of the art of technology. Finally, the analysis

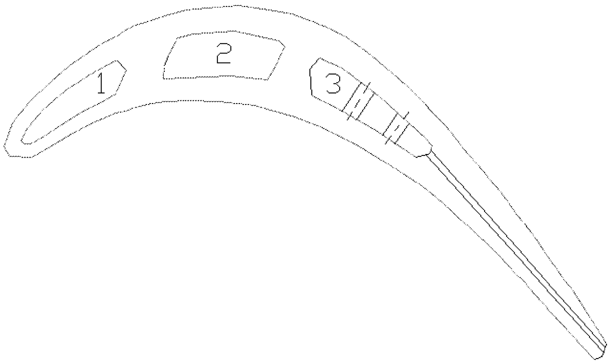

Figure 1. Multipass cooling system of the examined rotor blade.

that has been carried out is referred only to rotor cooling because of its higher complexity.

3. MAIN CHARACTERISTICS OF MODIFIED BLADE COOLING SYSTEMS

In this preliminary work the goal is the simple evaluation of some blade cooling improvements on a standard rotor cooled blade without solid geometrical modification. The two possible solutions here considered require different approaches. With this hypothesis some different ways can be pursued:

- a reduction in the coolant mass flowrate with a constant cooling performance,
- an increase of cooling performance with a constant coolant mass flowrate,
- an increase of cycle maximum temperature.

4. AIR COOLANT REDUCTION TEMPERATURE

We have two choices [8]:

- the use of classic surface exchanger cooled by water (WSE, Water Surface Exchanger),
- the water injection in the air coolant (CWI, Cold Water Injection).

In both cases the water is at the ambient condition, but for the CWI it is necessary an accurate treatment process. The modifications can be realized downstream to the coolant compressor bleeding like it is shown in the *figure 2*. This solution can be easily brought in the combined or mixed gas–steam power plants because, in

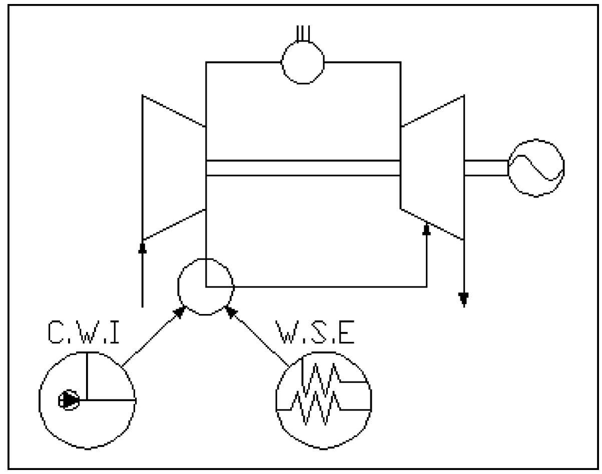

Figure 2. WSE and CWI approximate scheme.

this case, it is very simple to find high quality water; nevertheless the water consumption in the CWI is very limited and the WSE does not require treated water. For WSE technique the difficult is related to the air coolant extraction by the machine and, after pre-cooling, its reintroduction in the machine; on the contrary, for the CWI, we can use water spray atomizers in the air coolant ducts with more compact solutions.

5. STEAM COOLING

The high thermodynamic features of steam allow to obtain high cooling efficiencies with a lower mass flow rate than air cooling. But for economic reasons, the use of steam cooling is possible only in presence of steam production and use for other purposes. Therefore, we consider the steam coolant employment in the case of Gas/steam Turbine Combined Cycle (GTCC) or Steam Injected Gas turbine cycle (STIG). Because of the different use of steam in the two above mentioned types of plants and in order to explain completely its cooling capacities, we would better consider two different blade cooling system configurations:

- the conventional cooling system (OCL, Open Cooling Loop),
- a closed or semi-closed cooling system (CCL or SCL, Closed or Semi-closed Cooling Loop).

In particular, concerning turbines inserted in a STIG, it would be better to use steam with the OCL solution, whereas in the case of GTCC, the use of steam with a CCL or SCL scheme could be particularly advantageous [4].

Actually it will be necessary to introduce some cooling system modifications such as the reduction of the duct section, when high pressure steam is employed or other mass flow rate control is necessary. In order to realize a steam cooling system able to assure a certain operative life and good reliability, it is not enough to take into account this aspect relating only to cooling loop design. It is also essential to face other problems, from material technology to fluid dynamics [4]. Some of these are related to the compatibility of steam with the materials of turbine cooled components [4], others are related to the impurity of the coolant [4] and finally the steam heat transfer characteristic [4, 12].

6. EFFECTS OF THE MODIFIED COOLING USE ON THE PLANT PERFORMANCES

The introduction effects of the various cooling techniques and improvements already described are particularly interesting in terms of global plant performances.

The coolant temperature reduction is thermodynamically not positive for the plant efficiency when the WSE is adopted and a power increase can be achieved only in case of a reduction of the coolant mass flow rate [8]. On the contrary, the gas turbine mass flowrate increase determined by CWI introduction allows a small efficiency advantage and a power increase, proving that the coolant mass flowrate is reduced [8].

The use of steam cooling allows more incisive influences on power plant performance. The steam cooling always let all the compressed air to be used by the combustion chamber and consequently, the turbine inlet mass flowrate increases with the plant power. Moreover, this fact makes easier the employment of advanced premixed combustion chambers to reduce pollutant emissions [4].

In the OCL solution the cooling system implies the mixing of the exhaust coolant with the main stream and this determines an increase of the turbine mass flow rate and of the flue gas specific heat. So, as in all mixed gas–steam cycles, an increase in the produced power and a positive effect on the heat recovery steam generator performances are also determined. Nevertheless, the irreversibility of mixing process allows global performances lower than the high efficiency GTCC ones [7].

The use of CCL cooling techniques requires the use of fluids with high heat transfer features, like steam; adopting these cooling techniques in combined plants, it would be possible to recover, through regeneration, the heat removed during the blade cooling. This particular plant solution is being studied by GE for combined plants using H class turbines [4] (*figure 3*). This recovery of heat energy, such as all regenerative measures, allows improvement of the cycle global thermodynamic efficiency, expecting an average efficiency up to 60 %.

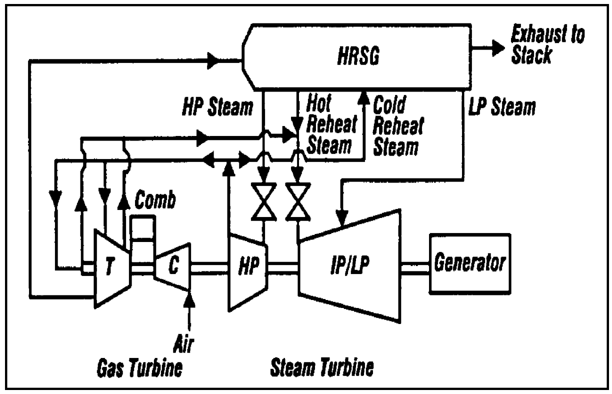

Figure 3. Approximate scheme of a combined plant with gas turbine cooled with steam in closed loop [4].

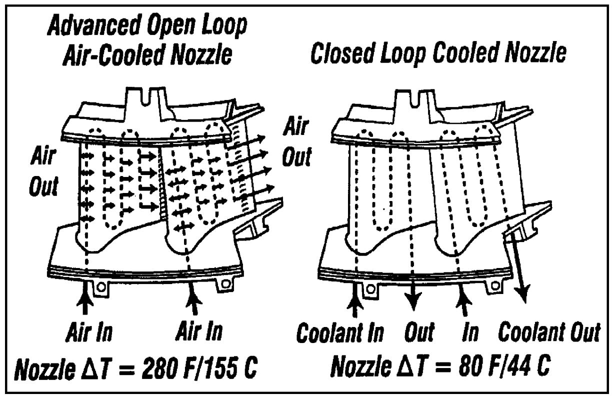

Figure 4. Approximate scheme of the closed loop blade cooling system [4].

In this case we have no more mixing losses. In particular, the total temperature drop during the first nozzle expansion (*figure 4*) is very reduced. This $\Delta T_{\mathrm{nozzle}}$ in cooled turbines of more advanced conception is quite high, around 150 K, whereas in the close loop case it is reduced to about 50 K [4]. Considering that the Temperature Inlet Turbine (TIT) is equal

$$\mathrm{TIT} = T_{\mathrm{firing}} = T_{\mathrm{flame}} - \Delta T_{\mathrm{nozzle}}$$

with the steam closed loop cooling it is possible to increase the rotor inlet temperature of about 100 K, without varying the combustion temperature. This is a very good goal because it would allow an increase in the plant performance without increasing pollutant emissions.

7. SIMULATIONS RESULTS

The codes previously described were used to assess the effects on blade cooling performance of the described cooling systems.

For a global performance evaluation we refer to two classical parameters named cooling efficiency ε_h and effectiveness ϕ, respectively defined as

$$\varepsilon_h = \frac{T_{\text{cool out}} - T_{\text{cool in}}}{T_{\text{b average}} - T_{\text{cool in}}} \tag{1}$$

$$\phi = \frac{T_g - T_{\text{b average}}}{T_g - T_{\text{cool in}}} \tag{2}$$

For the definition of the two parameters, it is necessary to underline that in literature there is often confusion. In our case these parameters allow respectively the evaluation of capacity to exploit the coolant (efficiency) and the effective cooling performance (effectiveness); often, when some possible blade cooling modifications are investigated the two parameter trends can be opposite and the designers must single out a correct balance between the possible alternatives.

To evaluate the reasonable values of ϕ and ε_h we use mean values of $T_{\text{b average}}$ and T_g calculated on three different blade sections (at constant radius), and mean values also for $T_{\text{cool in}}$ and $T_{\text{cool out}}$ considering all the mass flow rate distribution.

7.1. Air coolant reduction temperature

Both WSE and CWI solutions produce a mass flowrate increase if there is not a reduction of the coolant pressure drop across the blade cooling system. For this reason when these techniques are applied it is necessary to reduce opportunely the coolant pressure drop under the nominal value (*figures 5* and *6*). This is obtained simply by a lower pressure at cooling network inlet and this pressure value must be properly selected for any coolant temperature reduction to reduce coolant mass flowrate.

The introduction of WSE (*figure 7*) allows a progressive reduction of $T_{\text{b average}}$ and $T_{\text{b max}}$ but the trend is not the same and the $T_{\text{b max}}$ seems to be less reduced: the lower $T_{\text{b max}}$ decrease suggests a more accurate evaluation of thermal stress due to increased temperature gradient in the metal blade.

The *figures 5* and *7* show that it is possible to maintain the blade at the same nominal average temperature with a mass flowrate saving reducing the coolant temperature and the coolant pressure drop (i.e. the pressure at cooling network inlet): for example, when the coolant temperature is close to $0.9T_{\text{cool nom}}$ and the coolant pressure drop is close to $0.46\Delta p_{\text{cool nom}}$, the average blade temperature remains constant with a coolant mass flowrate close to $0.7m_{\text{cool nom}}$.

Obviously, when the coolant temperature is even more reduced the mass flowrate saving necessary to maintain the blade average temperature at the nominal value might be higher and this reduction would allow a fire temperature increase.

The use of CWI (*figure 6*) allows the same effects of WSE on $T_{\text{b average}}$ and $T_{\text{b max}}$ trends but it needs a little injection of water; in fact only 4 % of water in the mass

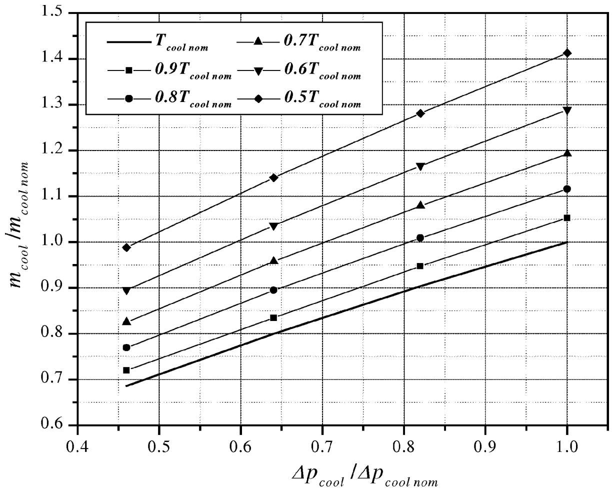

Figure 5. m_{cool} versus Δp_{cool} for WSE technique.

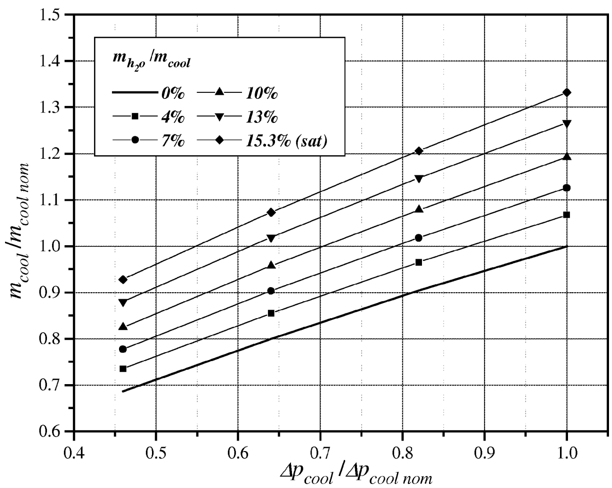

Figure 6. m_{cool} versus Δp_{cool} for CWI technique.

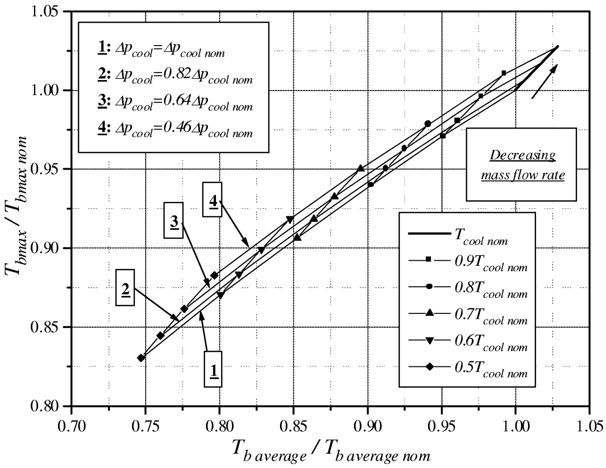

Figure 7. $T_{b\ max}$ versus $T_{b\ average}$ for WSE technique.

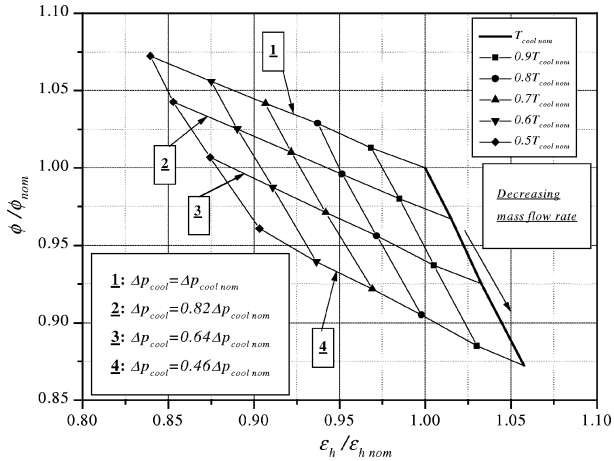

Figure 9. ϕ versus ε_h for WSE technique.

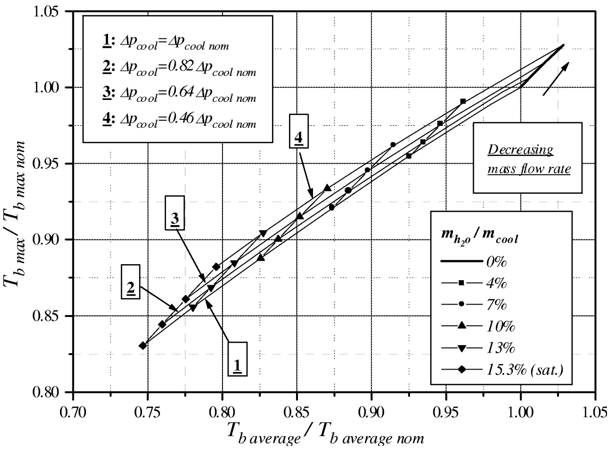

Figure 8. $T_{b\ max}$ versus $T_{b\ average}$ for CWI technique.

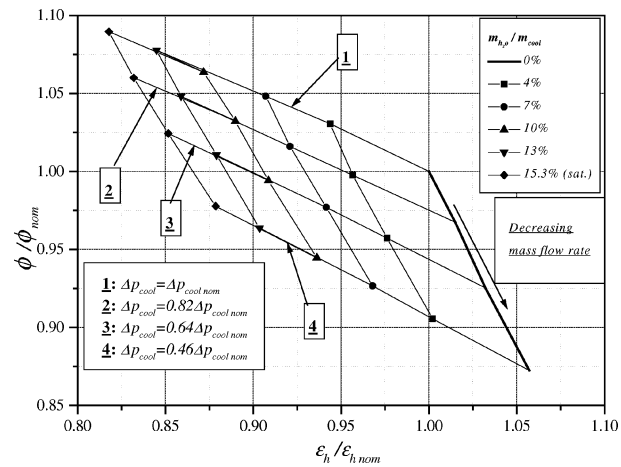

Figure 10. ϕ versus ε_h for CWI technique.

TABLE I

Water injection mass flowrate	Coolant temperature
$0.040m_{cool\ nom}$	$0.85T_{cool\ nom}$
$0.070m_{cool\ nom}$	$0.75T_{cool\ nom}$
$0.100m_{cool\ nom}$	$0.66T_{cool\ nom}$
$0.130m_{cool\ nom}$	$0.57T_{cool\ nom}$
$0.153m_{cool\ nom}$	$0.51T_{cool\ nom}$

coolant is the equivalent of about $0.85T_{cool}$ and in terms of compressor inlet mass flowrate the water flowrate is lower than 1 %, a value comparable with a traditional combustion chamber water injection to NO_x reduction. In the *table I* is compared, for any water injection mass flowrate, the corresponding coolant temperature.

Finally in the CWI solution the mass flowrate saving is greater than in the other technique because in this case water is added to the air coolant.

In both solutions the efficiency ε_h and the effectiveness ϕ have different behavior (*figures 9* and *10*): in fact, for any coolant temperature or for any injected water mass, as the coolant pressure drop decreases, the first of the two parameters increases whereas the second one decreases. In this case too, like for the blade temperature, it is possible to maintain the efficiency at nominal value with a mass flowrate saving, reducing opportunely the coolant temperature together with the coolant pressure drop. In this way there is a slow effectiveness reduction. For example, when the coolant temperature is close to $0.8T_{cool\ nom}$ and the coolant pressure drop is close to

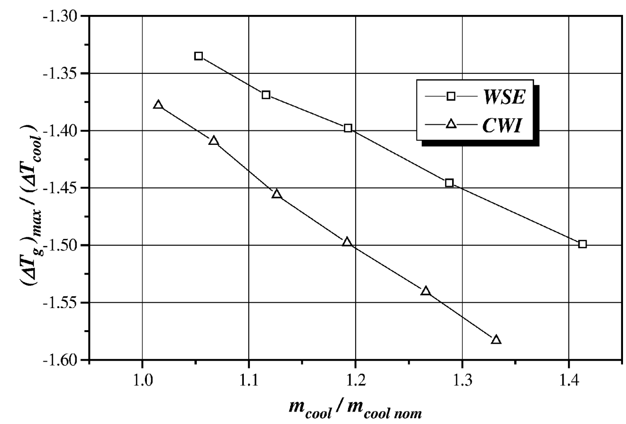

Figure 11. Gas temperature maximum increase/coolant temperature reduction versus mass flowrate for both solutions.

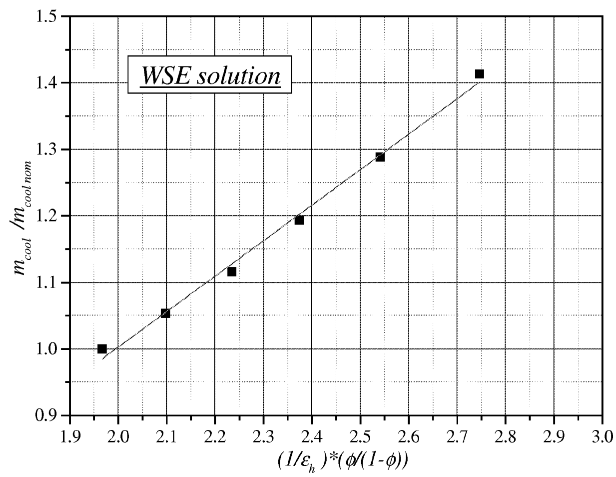

Figure 12. Mass flowrate trend for WSE.

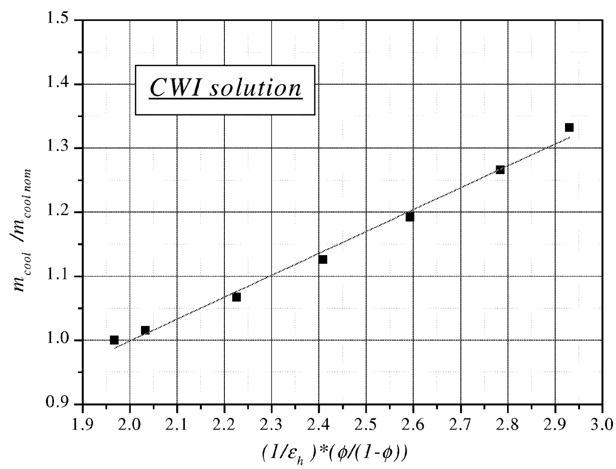

Figure 13. Mass flowrate trend for CWI.

$0.46\Delta p_{\text{cool nom}}$, the efficiency remains constant with an effectiveness reduction close to 10 %.

With the CWI solutions considering the same value of the efficiency and of the coolant mass flowrate a slow increase in effectiveness is shown because of the slow increase in coolant specific heat due to the water injection.

By considering a constant inlet pressure value at inlet cooling system and equal to the nominal value, the coolant temperature reduction produces a coolant mass flowrate increase, so it seems to be possible a TIT increase with consistent performance advantage. To estimate it as gas temperature maximum increase it is possible to apply the following procedure. It is well known that the coolant mass flowrate increase produces an effectiveness increase and an efficiency decrease: maintaining the blade average temperature equal to the nominal value for any coolant temperature variation, from the first parameter variation we can calculate the corresponding gas temperature maximum increase.

This procedure permits to plot the curves in *figure 11*, showing like a more relevant TIT increase is possible in comparison with the corresponding coolant temperature reduction; moreover, this must be considered like a safety margin, because, as previously shown in *figures 5* and *6*, the control on $T_{\text{b max}}$ is less effective.

To validate this procedure it might be interesting to consider global heat transfer relations and to define the coolant–hot gas mass flowrate ratio [13]:

$$\frac{m_{\text{cool}}}{m_{\text{cool nom}}} = \left(\frac{m_{\text{g}}}{m_{\text{g nom}}} \frac{c_{p_{\text{g}}}}{c_{p_{\text{cool}}}} St_{\text{g}} \frac{A_{\text{b}}}{A_{\text{g}}}\right) \frac{1}{\varepsilon_{\text{h}}} \frac{\phi}{1-\phi} \qquad (3)$$

This relation shows like the mass flowrate ratio variation depends on dimensionless parameters if the terms in parenthesis are constant; in the *figures 12* and *13* is reported the relation between $m_{\text{cool}}/m_{\text{cool nom}}$ and the dimensionless group of equation (3); the course of both curves is clearly linear and so the simulation's result confirms the previous hypothesis.

7.2. Steam cooling system

Considering the OCL solutions it is not necessary to modify the blade cooling system; it remains the same as in air cooling system. If the same coolant temperature is considered, using steam it is possible to maintain the blade at the nominal temperature obtained through air

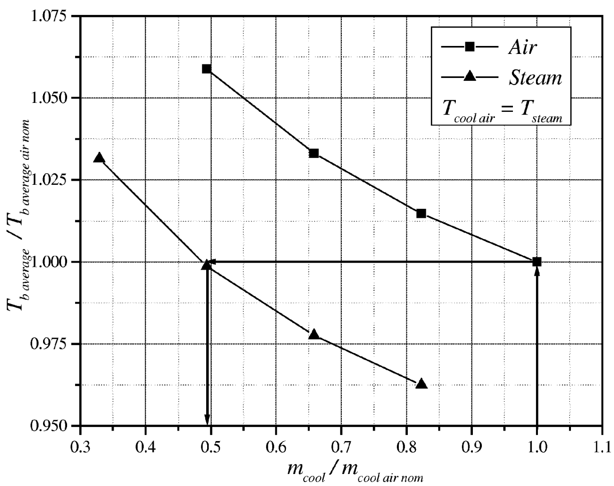

Figure 14. Trends of the average blade temperature with air and steam.

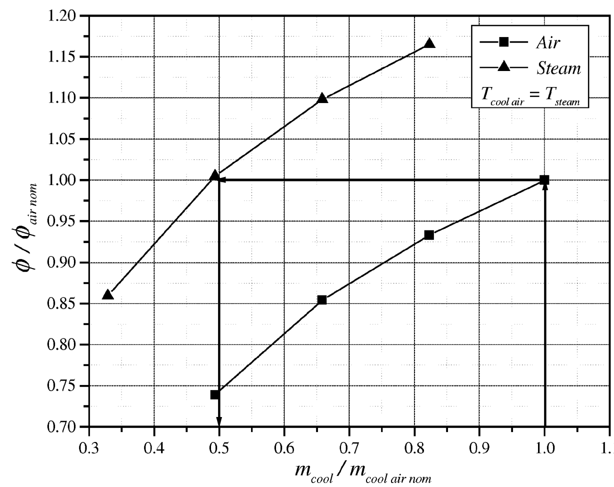

Figure 16. Trends of ϕ obtained with air and steam.

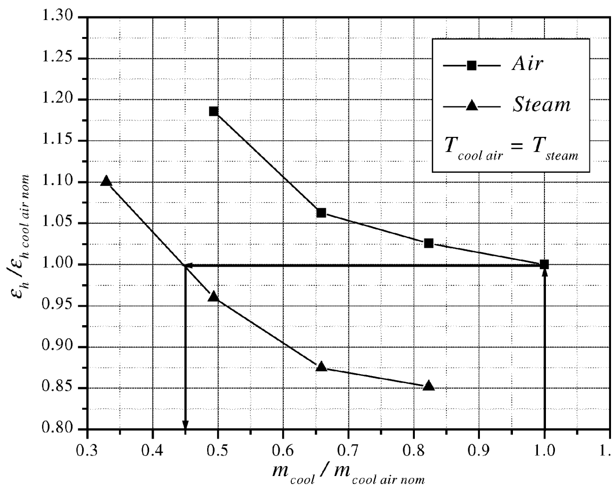

Figure 15. Trends of ε_{h} obtained with air and steam.

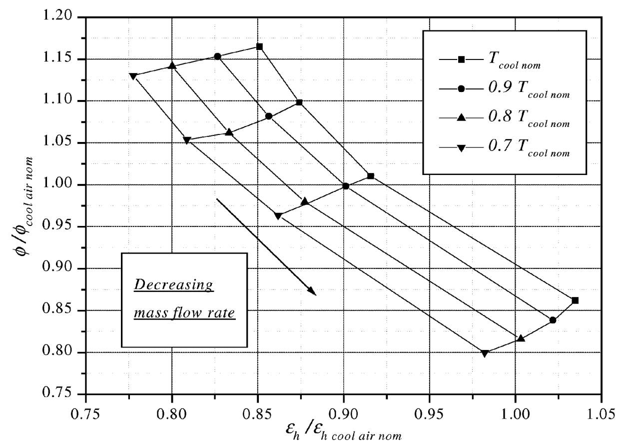

Figure 17. ϕ versus ε_{h} obtained with steam.

cooling, with a flow savings of about 50 % like it is shown in *figure 14*.

As far as efficiency and effectiveness is concerned, in *figure 15* and *figure 16*, respectively, there is a comparison of the trends obtained with air and steam cooling at same coolant temperature: using steam it is possible to obtain ε_{h} and ϕ nominal value with a remarkable mass flow rate reduction.

In *figure 17* we reported the $\varepsilon_{\mathrm{h}}/\phi$ trends obtained using steam at decreasing temperatures, because this opportunity appears realistic using steam coolant. The same blade temperature value is obtained with a steam coolant mass flowrate saving also if the curves show like the steam temperature reduction determines a little effectiveness reduction. In this case it can be interesting to evaluate the effects on the blade distribution temperature of the substitution of air with steam. Still considering the same mass flow rate and coolant temperature, in *figure 18*, the blade appears colder everywhere: this effect is due primarily to the difference in the specific heat between air and steam (the temperature distribution shown in *figure 18* is related to a blade to blade section at the average radius value and s/s_{tot} is a dimensionless parameter that is 0 at leading edge, negative on the pressure side and positive on the suction one).

In short, from the analysis of the simulation results, we can notice that the use of steam instead of air implies remarkable advantages in terms of performances of the blade cooling system. In fact, it is possible to maintain the same values of average blade temperature, efficiency

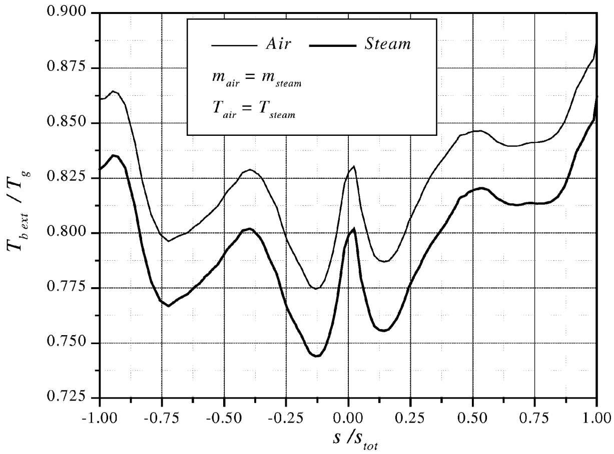

Figure 18. Profiles of the external blade temperature.

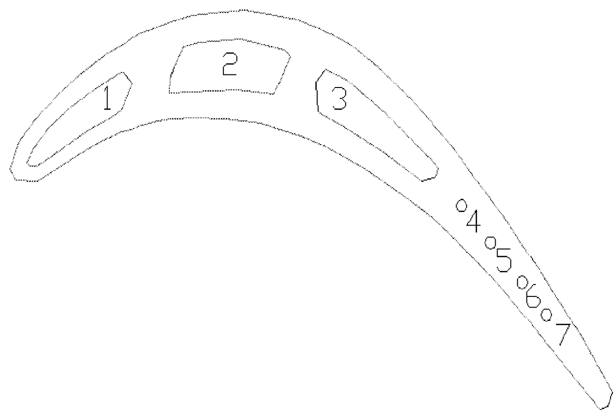

Figure 19. Scheme of the closed loop cooling system.

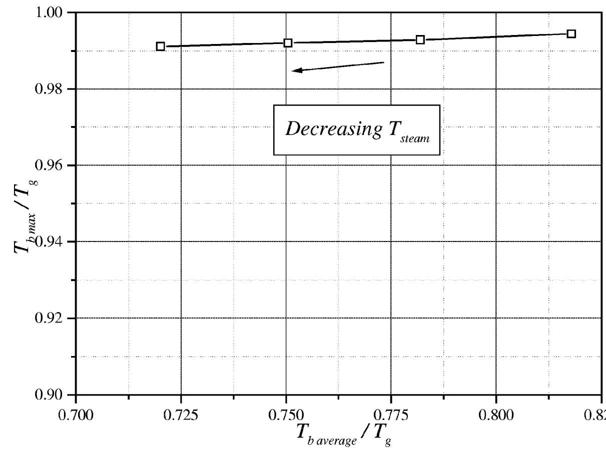

Figure 20. $T_{b\ max}$ versus $T_{b\ average}$ for various steam temperature.

and effectiveness using air cooling in nominal conditions with a coolant flow saving of about 50 %. So, given the limited quantities, the need of cooling steam can be easily satisfied with the use of a heat recovery steam generator. In fact, considering high performance cooled turbines, up to 20 % of the compressed air flow must be used for blade cooling. The quantity of cooling steam necessary to maintain the cooling system efficiency and the average blade temperature constant is about 10 % of gas turbine inlet flow, a compatible value with standard steam injected mass flow rate in the STIG or similar cycles.

Considering CCL solution, the cooling scheme depends on the selected application field. In fact, a heavy-duty application requires, as far as possible, simple modifications of already tested components. This study used a configuration, which could be easily realized starting from the previously analyzed open loop system, and so it would be easily adaptable to the blade profile assumed.

Figure 19 shows such geometry: the duct number 2 is connected with the two adjoining ones (number 1 and 3) by means of two 180° curves at the blade tip. The four radial tubes put on the trailing edge zone have also been alternatively connected by means of another two 180° curves. The proposed model allows steam to flow inside the blade through the ducts 1, 3, 4 and 6 and to get out of it, always in correspondence with the hub, through the ducts number 2, 5 and 7. The steam super-heated by cooling can consequently be conducted, through ducts obtained inside the rotating shaft, to the steam turbine [4].

As far as it is impossible to find results referred to the performances of other closed loop steam cooling systems, we have to limit ourselves to a general analysis of the results.

In this case we consider that the steam temperature varies but the coolant mass flowrate remains constant. *Figure 20* shows where the trends of the average and maximum blade temperature are reported: the first trend is close to those obtained with OCL methods, whereas the second parameter is very high and is practically never influenced by the coolant temperature reduction. This is very important because the low average blade temperature is a good result only if an acceptable blade temperature distribution is obtained.

Concerning effectiveness and efficiency, the values assumed by the first one are close to the typical ones of the open loop multi-pass cooling systems, whereas the values of the second are definitely lower. This depends on the fact that with the two different cooling typologies we obtain almost equal values of the average blade temperature, but in the closed loop the coolant is less heated because it only goes through two segments of

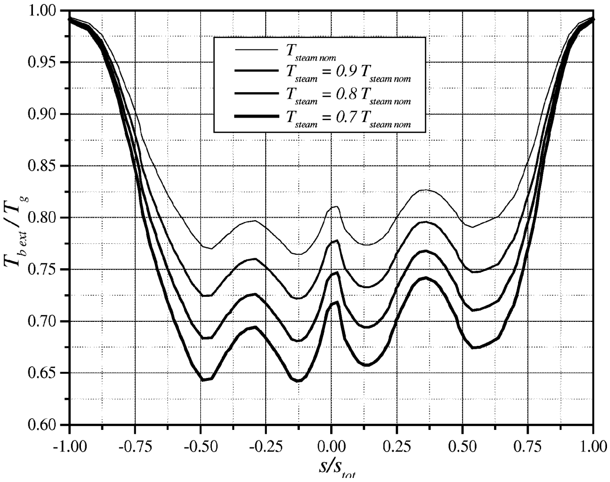

Figure 21. Trends of the external blade temperature profiles.

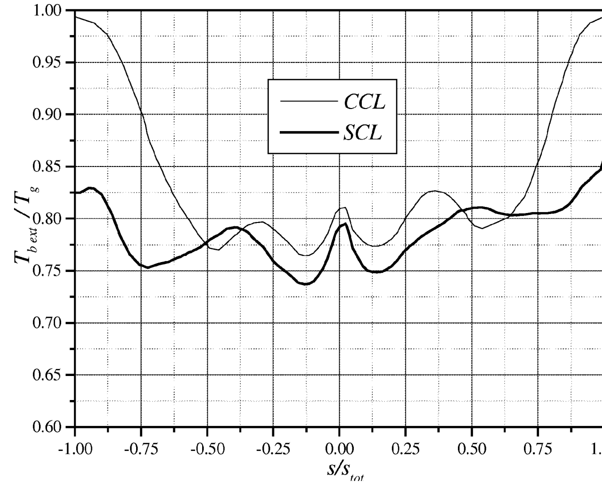

Figure 22. Trends of the external blade temperature obtained with steam in closed loop and in semi-closed loop.

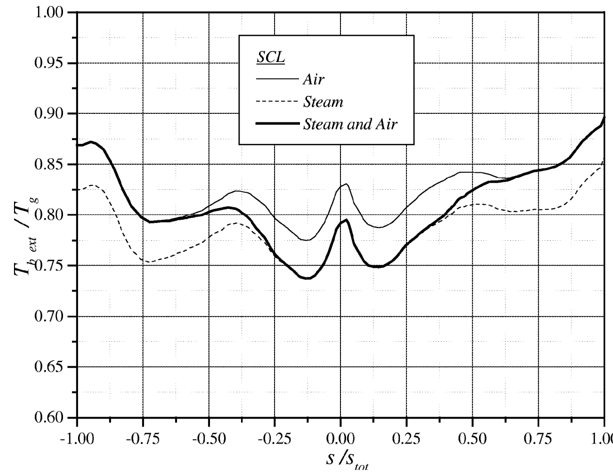

Figure 23. Trends of the external blade temperature obtained with air, steam, air and steam in semi-closed loop.

serpentine before getting out of the blade. In the multi-pass system this happens after it has flown through the pin-fins segment and the axial tubes.

The blade temperature distribution reported in *figure 21* shows that the three rectilinear ducts, derived by the old multi-pass scheme, allow an effective cooling of the front and central part of the blade, but the tubes put close to the trailing edge are inadequate, even for relatively low values of the coolant temperature. This depends on the fact that the four terminal ducts, because of their limited section, do not allow to carry off sufficiently high flows and, in any case, it is very difficult to cool the trailing edge, with its reduced thickness, by a simple closed loop based on the radial tubes.

So it is clear that, in order to have a really effective cooling scheme, the cooling of the last part of the blade has to be improved. For example, the realization of a fourth segment of serpentine would allow to transform the proposed method into an effective closed loop multi-pass system, because an even number of rectilinear segments would guarantee to steam the possibility to enter and get out of the blade always in correspondence of the hub.

This kind of solution is conditioned by the limited thickness of the rear part of the blade, that is we cannot obtain wide section ducts, unless we dangerously weaken it. So it is very difficult to extend the closed loop cooling at this blade part, without considering a radical blade re-design. For this reason we have chosen the SCL configuration; the closed loop section is made up of the serpentine segments 1 and 2, connected by a 180° curve close to the tip of the blade, whereas the segment 3, provided with pin-fins and axial tubes, is the open loop section that discharges the coolant throughout the trailing edge in the main gas flow. So, although this cooling scheme appears the same of *figure 1*, it represents an interesting evolution of the air open loop methods, mainly because they allow the simultaneous use of two different cooling fluids: with the combined plant, for example, the so-called closed loop section could be fed with steam, while the other one with air bled from the compressor. In this way it is possible to cool the terminal part of the blade in an adequate way and, at the same time, to continue to take advantage of the benefits of steam cooling and its regeneration in the bottoming cycle.

The better cooling effectiveness of this last proposed scheme has been confirmed by simulation results (*figure 22*). Using steam at the same temperature in both solutions, the SCL guarantees a better global cooling of the blade and, above all, an important reduction of temperature in its terminal part. Finally in *figure 23* is reported a comparison between the same blade cooled with air, steam and air with steam in the SCL. In this last case the steam is utilized only for the leading edge cooling, whereas the trailing edge is cooled with air.

It is clear that the curves relating to steam and air with steam coincide in correspondence of the leading edge, whereas in correspondence of the trailing edge the curve related to the air with steam is superposed to the one obtained by steam cooling. Obviously, the option with only steam allows a better cooling performance than the solution related to air and steam, but the final choice between the two alternatives depends on the evaluation of global plant efficiency and power.

8. CONCLUSIONS

The present paper evaluated the possibility to enhance the performance of traditional air blade cooling scheme.

In the first part two alternatives for the temperature reduction of air coolant are evaluated: a water surface exchanger (WSE) and a cold water injection (CWI). The second solution seems to be the better one to increase the cooling performance; moreover, its more simple introduction and very small quantity of water required can be very interesting, even if a very careful attention must be dedicated to obtain a very high purity of water.

In the second part the steam coolant introduction without radical blade re-design is evaluated. In particular, a traditional open cooling loop (OCL) has been considered and air and steam cooling have been compared. Steam allows the same cooling performances parameters with a remarkable mass flow rate saving (about 50 %). Consequently, the possibility of using steam in a closed cooling loop (CCL) is considered, where heat removed during cooling could be regenerated in other plant zones. The results show the difficulty to allow an acceptable uniform distribution of blade temperature, while the use of the other semi-closed cooling loop (SCL) option with steam and air allows a more uniform blade cooling and the possibility to use steam and air in the same cooling blade scheme.

Globally, the air coolant temperature reduction and the steam coolant introduction represent two sequential steps in the progressive upgrading process of gas turbine blade cooling towards the two main goals; the reduction of coolant mass flowrate and further TIT increases.

Acknowledgements

The authors would like to thank Ing. U. Corradini from Nuovo Pignone S.p.A. for the valuable advice and useful discussions and Dr. R. Lugli for the linguistic help.

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