25th ABCM International Congress of Mechanical Engineering
October 20-25, 2019, Uberlândia, MG, Brazil

COB-2019-0400
AN OVERVIEW OF BLADE COOLING EFFECTS ON TURBINE PERFORMANCE

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Abstract. *The development of gas turbine has focused on more powerful and efficient engines. A way to improve engine performance is by increasing the Turbine Inlet Temperature (TIT). A limit in the performance improvement is imposed by the permissible metal temperature. Cooling techniques have been used to allow the engine running above the melting point of the material. Generally, a portion of air is bled from the compressor for cooling purposes. The cooling air flows through the internal blade channels extracting heat from the metal, being discharged into the hot gases through the holes on the blade surface, the blade tip, and the trailing edge. However, turbine blade cooling decreases engine performance when compared with the uncooled one. This adverse effect is mainly due to pressure losses and the reduction of the mainstream temperature and mass flow through the cooled turbine section. In this paper, a single cooled turbine stage and turbofan engine are simulated. A marked difference between uncooled and cooled results was found for performance and flow parameters. The results also demonstrated that the cooling technique substantially affects predicting cooling airflow. It highlights the importance of considering the cooling on performance estimation since preliminary phases.*

Keywords: *Gas turbine, Engine performance, Simulation, Blade cooling, Cooling technique*

1. INTRODUCTION

There are many ways to improve engine performance, such as increasing component efficiencies, reducing the component pressure loss, employing intercooling and regeneration. Among these methods increasing the Turbine Inlet Temperature (TIT) is the most immediate means of realizing substantial performance improvements for both aeronautical and land-based gas turbines (Silva *et al.*, 2017; Glezer, 2003). For example, consider the performance results of a small subsonic turbojet engine shown in Fig. 1. For a constant pressure ratio, the highest possible temperature is desirable to achieve a higher specific thrust. For a given thrust, this reduces the weight and frontal area of the engine, at the expense of increased specific fuel consumption (SFC). While for a shaft power cycle, increasing TIT improves both specific power and SFC (Cohen *et al.*, 2001).

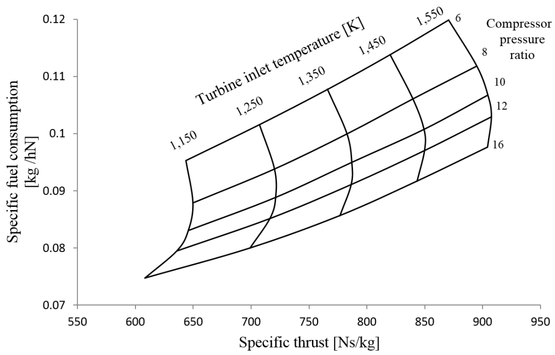


Figure 1: Turbojet cycle performance of uncooled engine (Silva, 2014).

Modern engines run above the allowable temperature of the material used in the turbine section by cooling their components. Nevertheless, blade cooling has a negative effect on engine performance as demonstrated by Horlock (2007); Silva *et al.* (2017); Sanjay *et al.* (2008); caused mainly by mixing losses and the reduction of mainstream gas temperature and mass flow through the cooled turbine section (Consonni, 1992).

In the present paper, an overview of blade cooling effects on turbine performance is performed simulating a cooled turbofan engine. Besides, simulations for a cooled stage were performed to analyze the effect of cooling techniques on cooling parameters.

2. INTRODUCTION TO BLADE COOLING

Operating gas temperature of the current engines can be as high as 2,000 K, exceeding the allowed temperature of the blade materials which is around 1,100 K (Singh and Singh, 2017). Then, the cooling of components becomes necessary to keep the blade temperature at a safe level, while satisfying engine performance and service time requirements, such as thrust/power, a long creep life, low oxidation rates, and low thermal stresses (Han and Wright, 2006; Young and Wilcock, 2002a; Sanjay *et al.*, 2008).

The most common method to cool the turbine components is to bleed a portion of the compressor airflow and inject it on the blades and disks. In the open-loop cooling system, the coolant after extracting heat from the blades mixes with the hot expanding gas, as shown in Fig. 2. While for the closed-loop cooling system, Fig. 3, there is no mixture between the coolant and the mainstream gas flow. Generally, this configuration applies to the combined cycle with steam as the cooling medium. The coolant passes through the internal circuit in the blade absorbing heat from the metal; the heated coolant is either used to increase the air temperature at the outlet of the compressor or mixed with the steam generated in the heat recovery steam generator (*HRSG*), expanding in the intermediate pressure steam turbine (Sanjay *et al.*, 2008).

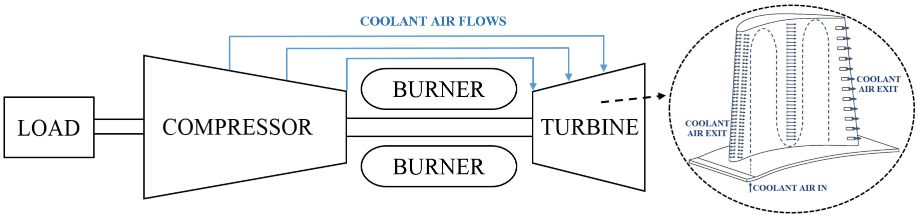


Figure 2: Gas turbine with open-loop cooling system.

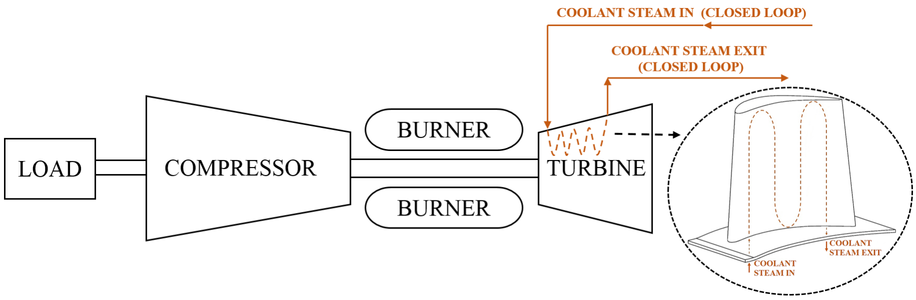


Figure 3: Gas turbine with closed-loop cooling system.

The cooling fluids may be air, steam, water, carbon dioxide CO_2, or air enriched with CO_2 (Sanjay *et al.*, 2008; Carcasci *et al.*, 2002); air being the most common coolant (Lakshminarayana, 1996). For this reason, only the air-cooling techniques will be described. Besides, only the cooling methods applied to the nozzle and rotor blades will be treated in the present paper.

The cooling technique can be classified as internal and external cooling as follows (LeGrives, 1986):

• *Internal cooling*: the coolant flows through the blade internal passages removing heat from the metal. As described by Singh and Singh (2017), the main aim of this method is to obtain the highest cooling possible with the smallest negative impact on engine performance. Examples of this technique are convection and impingement cooling. The technique applies to the mainstream gas temperature below 1,600 K (LeGrives, 1986).

• *External cooling*: a fraction of the internal coolant is ejected out of the blade forming a film layer, that decreases the heat flux to the external blade surface. Local film cooling, full-coverage film cooling, and transpiration cooling are examples. The technique applies to a mainstream gas temperature higher than 1,600 K (LeGrives, 1986).

Convective cooling is the simplest and the most used technique because it always coexists with some cooling method (Consonni, 1992). In general, the air is drawn from the engine compressor; the coolant passes through the internal circuit absorbing heat from the metal; after, it is ejected at the trailing edge and the tip of the blade. Several cooling concepts are used to enhance the heat transfer in the internal passages based on heat flux distribution around the blade and manufacturing constraints. Generally, the internal cooling methods are applied to specific parts of the airfoil. Jet impingement cooling is mostly used near the leading edge of the rotor blade and the stator vane, where the heat load is very high and a more uniform metal temperature distribution is required (Consonni, 1992; Tiemstra, 2014). Impingement cooling is a variation of the convective cooling, the cooling air at high-velocity impacts the internal blade wall increasing the local heat transferred from the metal surface to the coolant. Impingement can also be applied to the mid-chord of the stator vane. For rotor blades, the middle part consists of serpentine cooling passages roughened with some form of turbulence promoters (e.g., ribs, trip-strips). Rib turbulators are the most common technique to augment heat transfer coefficient in the cooling channels because they increase the surface area to heat transfer; induce the coolant flow to swirl after passing the rib, and create turbulence in the areas of flow separation (Han and Wright, 2006; Tiemstra, 2014). The main drawback is the enhancement of heat transfer is accompanied by an increase in coolant pressure drop. Finally, pin-fin cooling is commonly used in the trailing edge due to manufacturing constraints because jet impingement and ribs cannot be applied in this very narrow region (Tiemstra, 2014). The mechanism to enhance heat transfer is the same as ribbed channels. Dimple cooling is an alternative to pin-fin cooling with a reduced loss penalty and moderate heat transfer augmentation (Han and Wright, 2006). All these internal cooling techniques contribute to reducing the quantity of cooling air used because of the enhancement of internal heat transfer. Figure 4 is a schematic representation of the internal cooling methods described above.

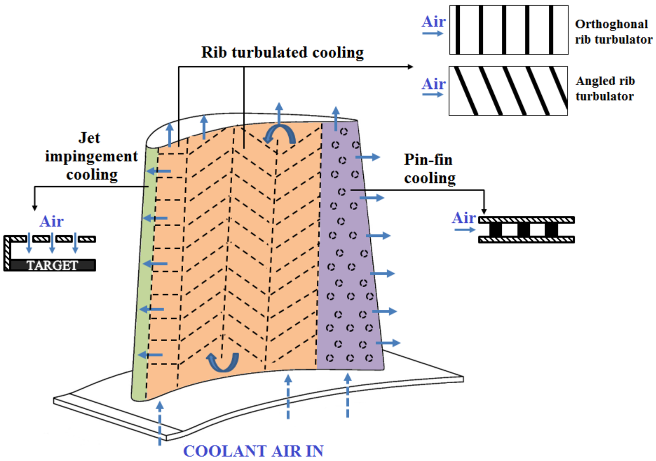


Figure 4: Schematic representation of internal cooling techniques to enhance the heat transfer.

In film cooling, the air used for internal blade cooling is ejected out of airfoil forming a film barrier on the external blade surface protecting it from the hot gas stream, as shown in Fig. 5. The coolant air is at a lower temperature than the mainstream hot gas reducing the heat flux going into the external blade surface. This protective layer protects not only the immediate injection local but also extends to a given downstream region. However, as the air flows downstream of the coolant holes, the film effectiveness decreases rapidly due to the mixture between the coolant air and hot gas increasing coolant air temperature (Tiemstra, 2014; Bogard, 2006). Then, additional hole rows could be required. If a very high number of small closely spaced holes are employed, the technique is called full-coverage film cooling (Lakshminarayana, 1996). It is an intermediate technique between local film and transpiration cooling. In the transpiration cooling, the coolant flow passes through a porous wall of the blade material, forming a layer of cooling air on the surface (Lakshminarayana, 1996). As a result, the heat transfer is directly between the coolant and the hot gas (Boyce, 2006).

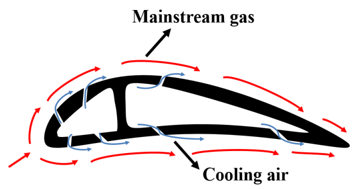


Figure 5: Schematic representation of film cooling concept.

Another method of decreasing the heat flux to blade metal material is through Thermal Barrier Coating (*TBC*), an additional thermal resistance, which is a refractory-oxide ceramic coating applied to the blade surface. The decreasing in the heat transfer is caused by the low conductivity of multi-layers of compost, nearly one-tenth that of conventional superalloys (Padture *et al.*, 2002; Clarke *et al.*, 2012). The use of TBC and internal cooling allows the airfoil to operate with a lower amount of cooling air at a given gas temperature or, a higher gas temperature at a given level of cooling flow as compared to uncoated airfoils (Glezer, 2003).

Modern gas turbines employ a combination of internal and external cooling and TBC to increase the cycle operating temperature; allowing the engine to run at gas temperatures nearly 813 K above the maximum material temperature (Glezer, 2003).

The selection of the cooling technique to be applied depends on many factors, such as the mainstream temperature, the component material, the engine configuration, emission, cost, and manufacturing complexity (Glezer, 2003). In addition, the selected cooling system would require a minimal amount of cooling air to keep the component temperature at an acceptable level, while it produces the smallest negative impact on cycle performance.

According to Cohen *et al.* (2001), the blade temperature can be reduced by 573 K using 1.5-2% of the total cooling flow per blade row. Considering the blade temperature on calculations is also important to satisfy the engine durability requirements. As reported by Han *et al.* (2000), an increase of 10 K at blade temperature may reduce the blade life by half. In this paper, a single cooled turbine blade is simulated to demonstrate the effect of cooling techniques on cooling parameters, such as the amount of cooling air and blade temperature.

3. MODELING OF GAS TURBINE BLADE COOLING

Turbine blade cooling has been the subject of work since the 1940's (Ellerbrock, 1948). There is a substantial literature at the topic comprising different areas, such as heat transfer; design; manufacture; testing; engine performance ((Lakshminarayana, 1996; Han *et al.*, 2000; Silva *et al.*, 2018), to name a few). In this paper, only a limited range of works are quoted, describing the cooling flow and loss estimations.

Holland and Thake (1980) derived a semi-empirical model that calculates the coolant mass flow ratio ($\dot{m}_c/\dot{m}_g$) from known data of cycle calculations (i.e. TIT and $T_{0c,i}$ - compressor outlet air temperature), maximum allowed blade temperature ($T_{b,max}$), and assumed values of cooling efficiency (η_c) and film cooling effectiveness (ϵ_f). Young and Wilcock (2002b) modified the Holland and Thake (1980) work's including the Thermal Barrier Coatings (TBC) and allowing the variation of temperature in the cross direction through the blade wall and coating. However, the authors kept the assumption that blade temperature along span is constant. The continuous expansion model developed by El-Masri (1988) evaluate the effects of different cooling techniques on the cycle performance.

Consonni (1992) presented an analytical model to predict the cooling mass flow as a function of the maximum blade temperature and the cooling configuration. Consonni's method has formed the basis of the subsequent works of Jordal *et al.* (2001); Chiesa and Macchi (2004); Torbidoni and Horlock (2005).

Tiemstra (2014) developed a method to estimate the cooling air mass flow from detailed modeling of various internal and external cooling schemes; allowing film cooling, rib turbulated cooling, pin-fin cooling, and jet impingement cooling to be simulated. The cooling air mass flow is found by an iterative process. The solution is obtained as the difference between the heat absorbed by the coolant air and the heat going into the blade reaches a given convergence tolerance.

In the method proposed by Masci and Sciubba (2018), the coolant mass flow rate is calculated from known values of the internal and external temperature profile of the blade surface along the chord, the maximum allowable metal temperature, and external convective heat transfer coefficient. In addition, the hot gas temperature profile along the blade channel must be provided for external surface temperature estimation.

In some approaches, the cooling mass flow rate is estimated by linear functions, which only depend on the turbine inlet temperature and/or blade metal temperature (Nada, 2014; Kostyuk and Karpunin, 2014).

Generally, the cooling-induced loss is calculated in terms of total pressure losses caused by the coolant friction inside the channels or the mixture between the cooling air and gas flow. Many works use empirical relationships based on Hartsel (1972) and Shapiro and Hawthorne (1947) methods to predict pressure loss, such as in Ainley (1955); Consonni (1992); Horlock *et al.* (2001); Song *et al.* (2015). In other works, the cooling-induced loss is calculated in terms of entropy creation, such as in El-Masri (1986); Young and Wilcock (2002b); Wei (2002); Uysal (2017). An interesting characteristic of Young and Wilcock (2002b) method is the total loss is sub-divided according to each loss mechanism, allowing to identify the magnitude of each component of loss. The negative effect of cooling can also be accounted for the turbine efficiency, such as in Horlock (1966); Kurzke (2002); Young and Horlock (2006); Horlock and Tordidoni (2008).

4. TURBINE COOLING EFFECT ON ENGINE PERFORMANCE

The amount of airflow used for cooling purposes can represent about 25% of the total inlet compressor flow (Glezer, 2003). This large quantity of air can significantly affect engine cycle performance in many manners. The first is the reduction of gas mass flow along the cooled turbine section as compared to the uncooled turbine, as shown in Fig. 6. A fraction of the total cooling air is discharged within each cooled stage; only the coolant mass flow ejected before the rotor is considered to generate work at that stage. Then, the work required to compress cooling air is not completely recovered in the turbine expansion. As the number of cooled rows $n_{row,c}$ increases, the effect of reduced mass flow becomes more pronounced.

The second effect is the reduction of hot gas temperature. After performing the cooling in the internal passages, the coolant air is discharged into the mainstream flow. The cooling air discharged enters the turbine at a lower temperature than

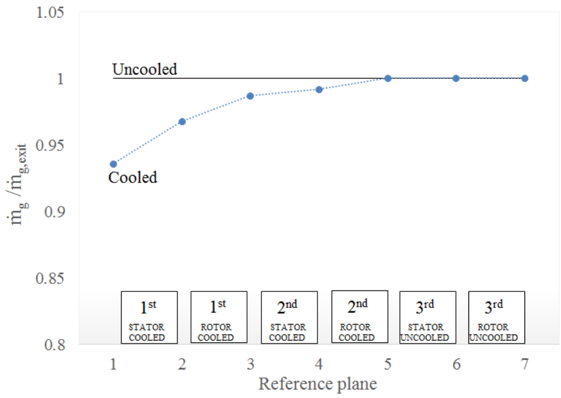


Figure 6: Variation of gas mass flow along a three-stage turbine (Silva *et al.*, 2017).

the gas temperature, resulting in a reduction of mainstream gas temperature after the mixture $T_{0g,mix}$, as demonstrated by the balance of energy given by Eq. (1).

$$(\dot{m}_g^{'} + \dot{m}_c)c_{p,mix}T_{0g,mix} = \dot{m}_g^{'}c_{p,g}T_{0g} + \dot{m}_c c_{p,c}T_{0cx}. \tag{1}$$

where: $T_{0g,mix}$ is the mixed flow temperature; T_{0cx} is the temperature of the discharged cooling air; $\dot{m}_g^{'}$ is the gas mass flow rate before the mixture; $\dot{m}_c$ is the coolant mass flow injected into the mainstream; $c_{p,mix}$, $c_{p,g}$, and $c_{p,c}$ are the specific heat at constant pressure of the mixed flow, hot gas, and coolant, respectively.

The lower temperature at the rotor inlet results in lower stage outlet temperature and pressure to provide the same power/thrust as an uncooled stage, as shown in Fig. 7. In other cases, where the stage outlet conditions are specified the engine would run at a higher stage inlet temperature to deliver the same specified power as an uncooled engine; for example, increasing the turbine inlet temperature.

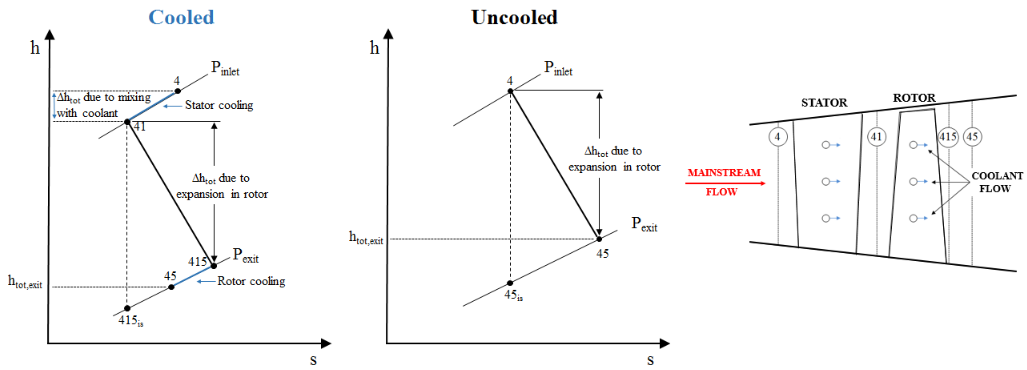


Figure 7: Cooled and uncooled expansion process at one turbine stage (Silva *et al.*, 2017).

The third effect is related to the mixing loss caused by the mixture between the injected cooling air with the gas mainstream flow. This loss is generally related to the total pressure decrease of the mainstream mass flow (Glezer, 2003). Finally, the coolant pumping power requirements also contribute to offset the performance improvement brought about by the higher operating temperature. It represents a decrease in the output power caused by the pumping of the cooling air through the internal passages, increasing the angular moment of the coolant from the entry state to the exit state (injection state) (Young and Wilcock, 2002b; Young and Horlock, 2006). The pumping power is represented by the second right-hand side of Eq. (2), which is used to estimate the cooled stage power (Young and Wilcock, 2002b).

$$P = \dot{m}_g^{'}(h_{0g} - h_{0g,mix}) + \dot{m}_c(h_{0c,i} - h_{0g,mix}) \tag{2}$$

where: $h_{0g,mix}$, h_{0g}, and $h_{0c,i}$ are the enthalpy of mixed flow, gas before mixing, and cooling air at bleed location, respectively.

5. MODEL DESCRIPTION

5.1 Cooling parameter predictions

The cooling predictions are based on the work developed by Young and Wilcock (2002b). The procedure described below is the same for stator and rotor blades except that the latter is carried out concerning the rotating coordinate system.

The cooling airflow rate required to maintain the blade temperature within the safe operating limits is given by Eq. (3), where the coolant flow is expressed as a portion of the mainstream gas flow ($\dot{m}_c/\dot{m}_g$).

$$\dot{m}_c/\dot{m}_g = K_{cool} m_{c+}, \tag{3}$$

where: K_{cool} is the cooling flow factor, a variable based on a constant value of Stanton number ($St_g = 0.0015$) and assumed values of the blade surface to gas area ratio A_{surf}/A_g and the gas to coolant specific heat ratio, $K_{cool} = (A_{surf} c_{p,g} St_g)/(A_g c_{p,c})$. It is treated as a known parameter (typically $K_{cool} = 0.045$, Young and Wilcock (2002b)). m_{c+} is a dimensionless coolant mass flow rate;

$$m_{c+} = \left\{ \frac{\epsilon_b}{\eta_{c,ext}(1-\epsilon_b)} - \epsilon_f \left[\frac{1}{\eta_{c,ext}(1-\epsilon_b)} - 1 \right] \right\} (1 + Bi_{TBC})^{-1}, \tag{4}$$

where: ϵ_b is the blade cooling effectiveness, $\eta_{c,ext}$ is the cooling efficiency defined in terms of the external metal temperature ($T_{m,ext}$), ϵ_f is the film cooling effectiveness, and Bi_{TBC} is the TBC Biot number. The value of ϵ_f is suggested to be within the range of 0.2–0.4, as in Young and Wilcock (2002b). ϵ_b is calculated from known values of the mainstream gas temperature T_{0g}, $T_{0c,i}$, and $T_{b,max}$ by Eq. (5). $\eta_{c,ext}$ can be determined once $\eta_{c,int}$ and metal Biot number Bi_m have been specified, Eq. (6). $\eta_{c,int}$ is the internal flow cooling efficiency defined in terms of internal metal temperature $T_{m,int}$ with assumed value within the range of 0.6–0.8 (Young and Wilcock, 2002b).

$$\epsilon_b = \frac{(T_{0g} - T_{b,max})}{(T_{0g} - T_{0c,i})}, \tag{5}$$

$$\eta_{c,ext} = \frac{(T_{0c,x} - T_{b,max})}{(T_{m,ext} - T_{0c,i})} = \frac{\eta_{c,int}}{1 + \eta_{c,int} Bi_m m_{c+}}. \tag{6}$$

Equation (4) is applied to convection cooling, film cooling, and TBC thermal protection, separately or a combination of these techniques.

Prediction given by Eq. (3) is based on peak temperature T_{0g}^{max} rather than mainstream gas temperature T_{0g}, ensuring a safety margin to blade metal temperature. T_{0g}^{max} can be estimated from Eq. (7).

$$T_{0g}^{max} = T_{0g} + K_{comb}(TIT - T_{0c,k}). \tag{7}$$

where: K_{comb} is the combustion pattern factor. The value is based on combustor design and the blade row position to the burner outlet (Young and Wilcock, 2002b). $T_{0c,k}$ is the cooling air temperature at the compressor outlet.

Another important parameter is the loss due to cooling. Following Young and Wilcock (2002b), the total rate of entropy creation $\Delta\dot{S}_{TOT}$ is estimated to each row of the cooled stage applying Eq. (8).

$$\Delta\dot{S}_{TOT} = \Delta\dot{S}_{basic} + \Delta\dot{S}_{cool} \tag{8}$$

where: $\Delta\dot{S}_{basic}$ is the loss associated with uncooled operation; it is assumed unchanged in the presence of cooling (Young and Wilcock, 2002a). In the present work, the uncooled pressure drop is based on the method given by Kacker and Okapuu (1982). $\Delta\dot{S}_{cool}$ is loss associated with cooling which might be divided regarding each loss mechanism:

- Heat transfer loss;
- Coolant fluid friction loss;
- Loss due to thermal energy dissipation; and,
- Loss due to the kinetic energy dissipation.

5.2 Flow properties

The two-component semi-perfect gas description is adopted for property estimation (Silva *et al.*, 2019b). In this approach, the working fluid is represented by air and gas (combustion products) (Young and Wilcock, 2002a). The specific heat at constant pressure c_p and the ratio of specific heats γ depend on temperature alone.

A complete mixing between the coolant and gas mass flow is assumed. The gas total enthalpy after the mixture is evaluated from an energy balance. The pressure follows once entropy (Eq. (8)) and mixed gas enthalpy have been determined. The mixed gas flow is given by,

$$\dot{m}_g = \dot{m}_g^{'} + \dot{m}_c \tag{9}$$

6. RESULTS AND DISCUSSIONS

As described in the previous sections, the applied cooling technique affects the amount of cooling air; consequently, the engine performance. A single cooled stage was simulated using CTurb to demonstrate the effects of cooling techniques on results. CTurb is an in-house code at Fortran language developed by Silva (2014) to evaluate the cooling air requirements for a given engine operating condition and allowable metal temperature. The cooling techniques simulated were convective cooling, film cooling, and a combination of TBC ($Bi_{TBC} = 0.3$) and convective blade cooling. The major input data are presented in Tab. 1. $P_{0,g}$ and $P_{0,ck}$ are the total pressure of hot gas and coolant air at the bleed point, respectively. RP_t is the turbine pressure ratio. $\eta_{pol,t}$ is the stage polytropic efficiency.

Table 1: The main input data to simulate a cooled stage with different cooling techniques (Silva, 2014).

Mainstream flow parameters	$T_{0c,k}$	729 K	$P_{0,g}$	1.97 MPa	$P_{0,ck}$	2.03 MPa
	$\dot{m}_g$	8 kg/s				
Turbine parameters	$T_{b,max}$	1,100 K	RP_t	1.95	$\eta_{pol,t}$	0.89

The air mass flow estimated for each cooling technique is shown in Fig. 8. For a given TIT, a substantial reduction in required cooling air results from using film cooling. The decrease is caused by the combined effect of internal and external cooling. A considerable reduction in the amount of cooling air caused by TBC is also demonstrated. It is worthwhile to say that the film was applied to a range of temperatures below the commonly found, only for comparison purposes. A lower cooling airflow could be achieved for convective cooling using rib turbulators and jet impingement. However, the Young and Wilcock (2002b) method does not allow to simulate the internal cooling concepts.

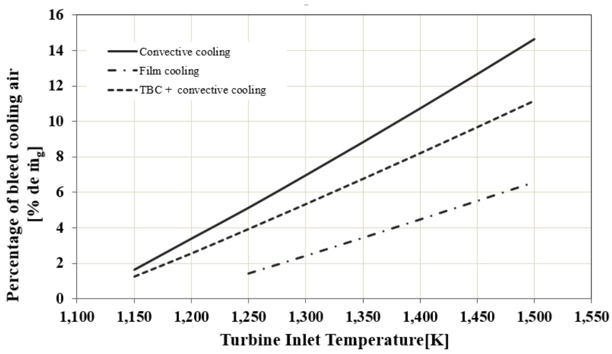


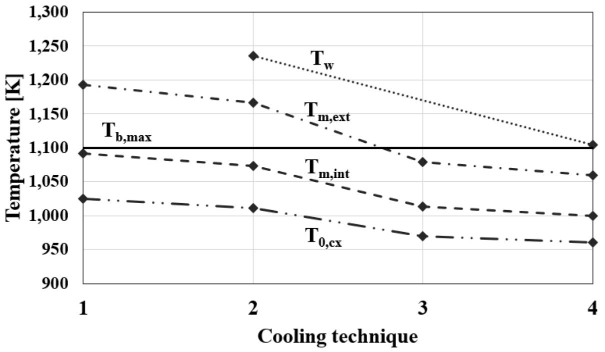
Figure 8: Amount of bleed coolant air as function of cooling technique (Silva, 2014).

Figure 9 shows the variation of blade metal temperature for different cooling techniques and TBC Biot numbers, where cooling techniques are represent by: (1) convective cooling + $Bi_{TBC} = 0.0$, (2) convective cooling + $Bi_{TBC} = 0.15$, (3) film cooling + $Bi_{TBC} = 0.0$, and (4) film cooling + $Bi_{TBC} = 0.15$. The simulations were performed with reported data from Young and Wilcock (2002b) with $TIT = 1,700K$, $T_{0c,k} = 867K$, $Bi_m = 0.2$, $\eta_{c,int} = 0.7$, and $\epsilon_f = 0.4$, keeping constant the coolant to mainstream flow ratio for stator (0.145) and rotor (0.049). The results demonstrate that convective cooling fails to keep the external blade temperature at a safe level for both stator and rotor, even with TBC. The internal metal temperature is below the allowable material temperature for all simulated cooling techniques for stator vane. While for rotor blade, the cooling techniques 1 and 2 present $T_{m,int}$ about 5% and 3% higher than $T_{b,max}$, respectively. Higher values of internal temperature result from lower cooling mass flow rate and higher $T_{m,ext}$ than stator. The results demonstrate the importance of simulating the engine with cooled turbine blade using appropriated cooling technique and cooling predicting method that taking into account the blade metal temperature, avoiding to set the percentage of cooling air in an arbitrary way in the simulations.

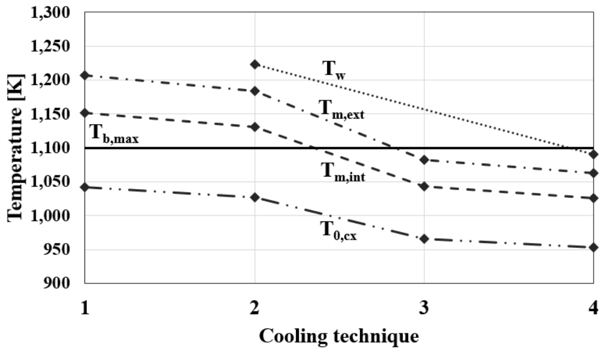
Simulation was carried out for a two-spool turbofan engine using GTAnalysis and CTurb to analyze the cooling effects on engine performance. GTAnalysis is an in-house code at Fortran language developed by Bringhenti (1999) to simulate engine performance at state-steady and transient operation. The main data for engine simulation is given in Tab. 2.

Table 2: Main data to simulate an engine with turbine blade cooling (Silva *et al.*, 2019a).

Mass flow	28.6 kg/s	HPT configuration	One axial stage (cooled)
Bypass ratio (BPR)	1.04	Cooling air flow	4.1 (% engine inlet mass flow)
Overall pressure ratio (OPR)	11.10	$T_{b,max}$	1,100 K
TIT	1,433 K	Cooling technique	Convection



(a) Stator vane



(b) Rotor blade

Figure 9: Temperature as function of cooling technique.

Figure 10 shows specific fuel consumption (SFC), thrust, mass flow at the turbine inlet, and turbine outlet temperature (TOT) for uncooled and cooled simulations. The results demonstrate that cooling significantly affects the engine performance parameters. The TIT is the same in all simulations; however, the turbine rotor inlet temperature (TRIT) is lower for the cooled engine, nearly 1,388 K. The work produced by the turbine is fixed by compressor power requirements. Then, the properties at engine outlet change, as demonstrated to TOT, where the temperature reduces about 7% compared to the uncooled case. Consequently, the thrust is reduced. The mass flow at the turbine inlet is also affected because the air extraction occurs before the combustor; the air being reintroduced at the stator and rotor outlet. For a fixed fuel to air ratio (FAR), the decrease in the air at combustor inlet also results in a reduction of the fuel flow, producing a positive effect on SFC; however, with a lower thrust.

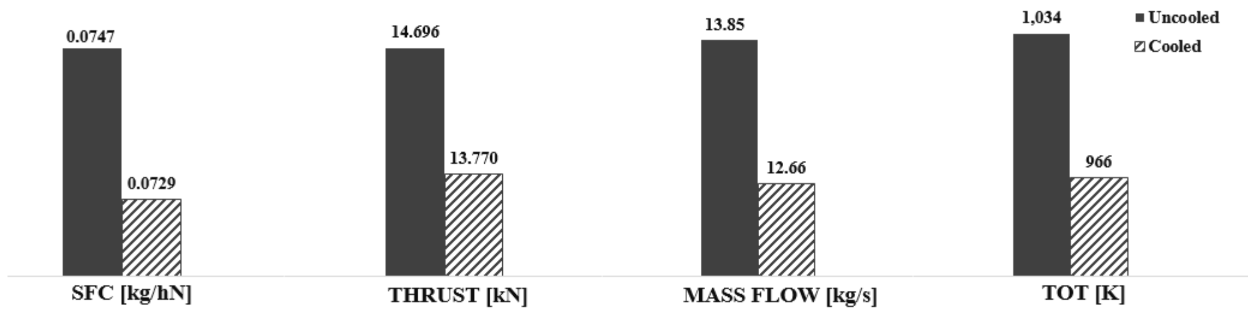


Figure 10: Engine parameters for uncooled and cooled simulations.

The accuracy of cooling results from CTurb can be analyzed by comparison with the data reported by Young and Wilcock (2002b), as given in Tab. 3. The results obtained from CTurb show good agreement with the data reported by Young and Wilcock (2002b) for all cooling parameters and temperatures.

Table 3: Result comparison between CTurb and the method from Young and Wilcock (2002b).

	Estator			Rotor		
	YW	CTurb	$\Delta\%$	YW	CTurb	$\Delta\%$
$\dot{m}_c/\dot{m}_g$	0.145	0.145	0	0.049	0.048	2.04
ε_b	0.75	0.75	0	0.58	0.57	1.72
$T_{0_{c,x}}$ $[K]$	969	969	0	966	972	-0.62
$T_{m_{ext}}$ $[K]$	1,078	1,079	-0.09	1,082	1,082	0
$T_{m_{int}}$ $[K]$	1,013	1,013	0	1,043	1,045	-0,19

7. CONCLUSIONS

In the current work, performance simulations were carried out for a cooled turbine stage and a two-spool turbofan engine using an in-house developed computer program. The purpose of the study was to investigate the effects of cooling on engine parameters. The effect of the cooling technique on results was also analyzed. The cooling parameter estimations were performed using the method proposed by Young and Wilcock.

The results for the cooled stage demonstrate that the cooling technique substantially affects the amount of cooling airflow required. As expected, the convective cooling presented higher values as compared to film and the combination of

TBC and convective cooling. Simulations keeping constant the coolant to mainstream mass flow ratio demonstrated that convective cooling failed to keep the blade temperature at a safe level, highlighting the importance of selecting correctly the cooling technique and the cooling predicting method, taking into account the blade temperature estimations.

Simulations for the turbofan engine show that the blade cooling substantially affects the engine performance parameters, such as thrust and SFC. Once the modern engines run at an operating temperature higher than the melting point of the turbine blade materials, the cooling effects must be considered since design preliminary phase.

8. ACKNOWLEDGEMENTS

CAPES (Coordenação de Aperfeiçoamento de Pessoal de Nível Superior), CNPq (Conselho Nacional de Desenvolvimento Científico e Tecnológico), FINEP (Financiadora de Estudos e Projetos), FAPESP (Fundação de Amparo à Pesquisa do Estado de São Paulo) are acknowledged for their support to the research carried out at the Turbomachines Department at ITA (Instituto Tecnológico de Aeronáutica).

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