Cooling Model Calibration in a Collaborative Turbine Preliminary Design Process Using the NASA Energy Efficient Engine Part II: 1D Turbine Modeling

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ABSTRACT

Accurately estimating turbine cooling requirements at a preliminary design stage is crucial for modeling the overall propulsion system. The operating conditions of compressor, combustor and turbine are significantly influenced by these requirements. Empirical cooling models have thus far provided reliable initial estimations. However, for next-generation aero-engines this empirical data becomes outdated. An alternative semi-empirical approach based on an established cooling model is built into a collaborative turbine preliminary tool chain to address this challenge. This cooling model is calibrated for the two cooled high-pressure turbines developed by P&W and GE within the NASA Energy Efficient Engine (E3) program and the resulting data is discussed. Sensitivity studies provide a better understanding of how turbine preliminary design parameters, material and cooling technologies affect cooling requirements. This work focuses on the turbine level studies with an accompanying paper at propulsive system level.

INTRODUCTION

Decisions made at a preliminary turbine design stage can significantly affect cooling requirements and the overall thermodynamic cycle. In the absence of detailed information on blade geometry and internal cooling design, estimating these requirements becomes a challenge. Thus far, the 1D turbine pre-design tool `PrEDiCT` [1], combined with empirical data from MTU aero-engines [2], provided reliable initial estimations for cooling requirements. For next generation aero-engines, operating conditions may significantly change. With higher turbine inlet temperatures as well as new materials and cooling technologies, this empirical data becomes inadequate. To address this challenge, a semi-empirical approach has been introduced in `PrEDiCT`.

This updated approach is based on the well established *Holland and Thake* cooling model [3] already extensively adopted in the literature [4, 5]. Nonetheless, there is a lack of information regarding the empirical values attributed to some model parameters. In the present work, these parameters are either calculated or calibrated for two cooled high-pressure turbines developed within the NASA Energy Efficient Engine (E3) program [6, 7] for comparison with the literature. This provides for once, a better understanding of these parameters and how they can be influence by design decisions. It additionally provides data on both of the modeled HPT performance and cooling systems as a source for future studies with higher fidelity cooling design methods such as `PICCOOLO` [8].

This work starts with an introduction of the preliminary design process in `PrEDiCT` followed by a brief overview of the implemented cooling model. The modeling of the two HPTs is briefly discussed before providing a preliminary analysis on the calibrated cooling parameters. Detailed information on the model integration in `PrEDiCT` is also provided and result data is to be discussed in greater detail.

METHODS

TURBINE MODELING IN PrEDiCT

The 1D meanline program `PrEDiCT` is directed at generating a preliminary design of a multi-spool turbine as part of the process chain introduced in [9]. As the first step in the tool chain, minimal input is needed: power requirements, shaft speed, and inlet flow conditions for both mainstream and coolant flows, at two operating points. Firstly, at the aerodynamic design point that defines the performance optimized turbine geometry and secondly, at the cooling design point that dimensions the cooling system. Additional parameters such as reaction, flow coefficient or Zweifel number can be manually adjusted by the user in an iterative process. The tool then delivers meanline thermodynamic quantities of the flow and cooling requirements for each blade row. The reader is referred to [1] for detailed information.

COOLING MODELING IN PrEDiCT

The ratio of coolant mass flow to mainstream blade row inlet mass flow $\dot{m}_{c,rel}$ calculated at the cooling design point is assumed constant throughout the operating range of the turbine, enabling a direct calculation of the absolute cooling mass flow at the aerodynamic design point.

The introduced cooling model assumes a blade with infinite thermal conductivity and thus a uniform temperature. The model, initially designed for convective and film cooled blades, has been extended to consider the effect of an insulating coating [4]. This follows the diagram depicted in Fig. 1. Unlike the blade itself, the coating is thermally modelled by its Biot number Bi_{coat}, requiring input on coating properties and on the averaged heat transfer coefficient between mainstream flow and coating surface h_g. The relative cooling requirements are then calculated according to Eq. 1. A brief overview of the model implementation is given below.

$$\dot{m}_{c,rel} = \frac{\dot{m}_c}{\dot{m}_g} = \frac{c_{p,g}}{c_{p,c}} \cdot \frac{A_b}{A_g} \cdot St_g \cdot HLP \cdot SF \tag{1}$$

$$HLP = \frac{\varepsilon_f \cdot (1-\eta_c) + \varepsilon_0 \cdot (\varepsilon_f \cdot \eta_c - 1)}{\eta_c \cdot (\varepsilon_0 - 1)} \cdot \frac{1}{1+Bi_{coat}} \tag{2}$$

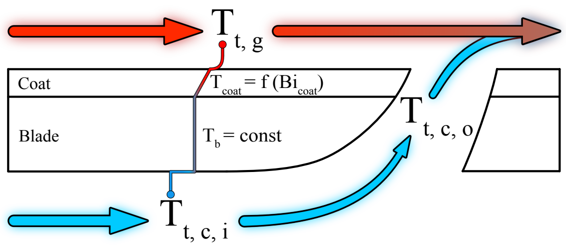

Fig.1 Notation for the heat transfer model, adapted from [4]

As input, the model requires specific heat capacity for both coolant and mainstream flow at the blade row inlet, $c_{p,g}/c_{p,c}$. A_b/A_g is the ratio between blade surface area and mainstream flow cross-sectional area at the throat [4]. Blade surface area is estimated through the blade chord and the correction parameter F_{sa} [10]. The throat area is a function of pitch and blade outlet angle [11]. The averaged mainstream flow Nusselt number Nu_g is estimated from the blade outlet Reynolds number Re_g using empirical data from *Louis* [12], yielding both h_g and the averaged Stanton number St_g. Finally, the heat load parameter HLP is a dimensionless representation of the coolant mass flow and a function of the blade cooling technology level. The internal cooling efficiency η_c models the heating of the cooling air inside the blade and the film cooling effectiveness ε_f the protective effect of the film [3]. Both parameters are a function of averaged mainstream flow inlet temperature $T_{t,g}$, coolant air inlet and outlet temperature $T_{t,c,i}$, $T_{t,c,o}$ and blade maximum allowable temperature T_b from Fig. 1. Both are also user-input, turning this into a semi-empirical approach.

This model is limited to airfoil cooling. Platform or disc cooling are added through a scaling factor SF. Furthermore, it is limited to averaged quantities. Aspects such as mainstream flow temperature profiles or blade temperature distribution cannot be modelled.

NASA E3 TURBINE MODELING

The Energy Efficiency Engine (E3) program was conducted under parallel National Aeronautics and Space Administration (NASA) contracts by Pratt & Whitney (P&W) and the General Electric Company (GE). The E3 program was created to develop fuel saving technology for future transport engines through new aerodynamic, mechanic and system technologies.

Both manufacturers provide detailed reporting at system and component design level [6, 7]. The published data includes turbine aerodynamic analysis as well as structural mechanical and thermal management analysis for the cooled turbine blades. While this does provide valuable validation data for select studies, it occasionally lacks information on essential parameters to model the high-pressure turbine and it's cooling system. To address this problem, both cooled high-pressure turbines are modeled in a collaborative process within the framework GTlab [13]. Turbine level is studied in this work, while analysis at propulsive system level is presented in an accompanying paper [14]. A brief discussion on some aspects of the modeling process is provided below.

The P&W HPT is designed for cruise conditions with a cooling system dimensioned for takeoff. Data from P&W's HPT design report [7] is used as the main source for modeling in `PrEDiCT`. There is generally a good agreement between the results of the modeled turbine and the literature, particularly in terms of the dimensionless turbine design parameters such as loading coefficient, flow coefficient, reaction and pressure ratio. GE's HPT is modeled according to the data provided in [6, 14]. In contrast to the P&W HPT, it is designed for maximum climb with a cooling system scaled for end-of-field hot day conditions. Again, there is a good agreement between the results of the modeled turbine and the literature.

The cooling system was the main focus of this modeling process. This includes parameters such as relative cooling mass flow, blade temperature as well as thermodynamic conditions of mainstream and cooling air. The NASA publications provide cooling requirements relative to the compressor inlet mass flow. This is converted to values relative to the blade row inlet mass flow for compatibility with the cooling model. Additionally, the distinction is made between airfoil and total blade row cooling mass flow to calculate the scaling factor SF from Eq. 1. Cooling and mainstream mass flow averaged temperature and pressure data is available, but blade temperature is provided only as a peak value. While this is relevant at a later design stage, it cannot be considered in this initial averaged approach. To estimate the blade averaged temperature required by the cooling model, the cooling effectiveness ε_0 is assumed constant between peak and averaged conditions. On a final note, all blade rows are coated only with a high thermal conductivity Ni-based oxidation resistant coating [7, 6]. It's thermal effect is therefore considered negligible and the coating Biot number $Biot_{coat}$ is set to zero.

RESULTS AND DISCUSSION

The parameters from Eq. 1 and Eq. 2 have been calculated or calibrated to match the $\dot{m}_{c,rel}$ values from the NASA publications.

Literature data on the ratio A_b/A_g is scarce. While [15, 16] estimate a ratio of $10-20$ for a generic blade, this is dependent on a series of geometrical parameters such as chord, stagger angle or pitch. Using the geometrical data available in `PrEDiCT`, this ratio is directly calculated and listed in Tab. 1. A large variation is perceived, in the range $12-21$ for stator rows and $5-11$ for rotor rows. Clearly, A_b/A_g tends to be higher for stator rows than for rotor rows by an averaged factor of ≈ 2. To estimate the blade surface area A_b, the correction parameter F_{sa} is calculated for the blade meanline profiles provided by the manufacturers. F_{sa} appears to be correlated with the flow deflection $|\beta_1-\beta_2|$ and blade maximum thickness to chord t_{max}/c, here represented by the parameter $S=|\beta_1-\beta_2|\cdot t_{max}/c$. This dependency should be analysed for a larger dataset and can prove useful to estimate blade surface area at a pre-design stage.

Table 1 Correction factor and area ratio for NASA E3 HPTs

Component		S [rad]	F_{sa} [−]	A_b/A_g [−]	Source
P&W	S1	0.26	1.13	21.1	[7, p. 13]
	R1	0.63	1.32	11.1	[7, p. 17]
GE	S1	0.22	1.13	11.9	[6, p. 14]
	R1	0.39	1.29	7.1	[6, p. 14]
	S2	0.22	1.19	14.0	[6, p. 14]
	R2	0.27	1.12	5.1	[6, p. 14]

The internal cooling efficiency η_c and the film cooling effectiveness ε_f are input parameters for the cooling model. In the presence of film cooling, ε_f is kept constant while η_c is calibrated. This is the case for the first stage of both HPTs. The range for ε_f is set between $0.15-0.30$. GE's second stage is not film cooled and thus $\varepsilon_f=0$. Calibrated data are provided in Fig. 2. Both parameters are largely dependent on the blade cooling design. For instance, the high η_c in GE's second stage stator S2 is notable. This may be justified on the one side by the combination of impingement cooling with the lack of film cooling, forcing the cooling air to travel the distance to the blade trailing edge. On the other side, this blade exhibits a low HLP i.e. a low cooling mass flow rate. Thus, the cooling air has more time to heat up before exiting the blade, leading to the higher η_c.

The corresponding heat load parameter HLP and cooling effectiveness ε_0 are depicted in Fig. 3. The first stators and rotors S1s and R1s display similar HLP values, indicative of the similar cooling mass flow requirements. With the exception of GE's S2, all blade rows present a combined cooling efficiency η_c^* [14] in the range $0.45-0.73$.

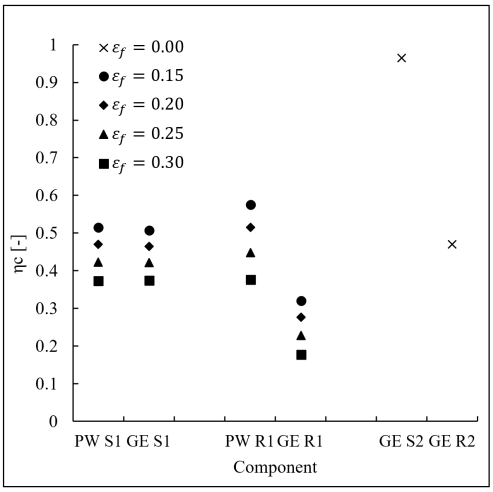

Fig.2 Calibrated internal cooling efficiency η_c for NASA E3 HPTs

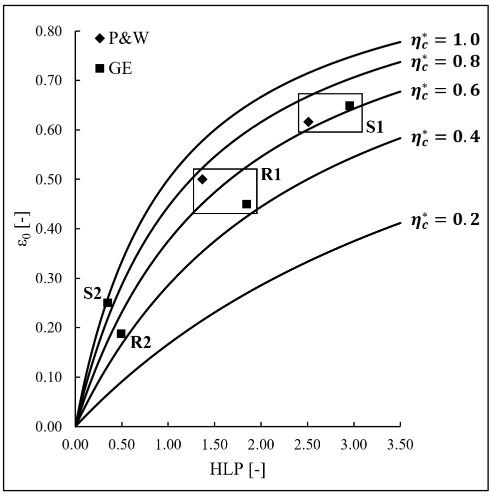

Fig.3 Cooling performance diagram for NASA E3 HPTs

The Stanton number is reported in some studies to assume a value of $St_g \approx 0.005$ [16]. *Torbidoni and Horlock* [5] instead resort to a correlation based on the experimental data of *Louis* [12]. The cooling model in this work uses the same experimental data to calculate the averaged Nusselt number Nu_g. A correction factor is adjusted to calibrate the resulting averaged heat transfer coefficient h_g between `PrEDiCT` and the literature. Both h_g and St_g values, calibrated for the E3 HPTs, are listed in Tab. 2. For a particular stage, variations in flow velocity and density lead to a lower blade outlet Reynolds number Re_g in the rotor, resulting in a consistently higher St_g than for the upstream stator. GE only provides data on the heat transfer coefficient for the first rotor. The calibrated correction factor for this blade row is therefore assumed constant for all remaining GE rows. It is interesting to note that there is a considerable discrepancy between the h_g of both first stage rotors. The different blade profiles result in different Mach number and different h_g distributions. The discussed method serves as a solid starting point, but should be iterated as higher fidelity data for a given blade becomes available.

Turbine inlet conditions affect Re_g and ultimately St_g. Furthermore, temperature and pressure boundary conditions can change considerably throughout the operating range of an aircraft. The effect on St_g is depicted in Fig. 4. A higher inlet pressure results in a higher fluid density and Re_g number, leading to a lower St_g. Similarly, a higher inlet temperature leads to a higher St_g. Looking at this behaviour, the thermal load is expected to rise for operating points

Table 2 Averaged heat transfer coefficient h_g and Stanton number St_g for NASA E3 HPTs

Component		h_g [W/m^{2}K]	St_g [10^{-3}]	Source
P&W	S1	4734	1.25	[7, p. 37]
	R1	3492	1.72	[7, p. 44]
GE	S1	8135	1.94	-
	R1	6541	2.51	[6, p. 42]
	S2	4187	2.10	-
	R2	3461	2.85	-

with a high inlet temperature and a low inlet pressure. Takeoff is typically selected as dimensioning point for the cooling system of civil aero-engines. In the presence of a wider operating range, as is the case of military aero-engines, high altitude operating points may exhibit an increased thermal load.

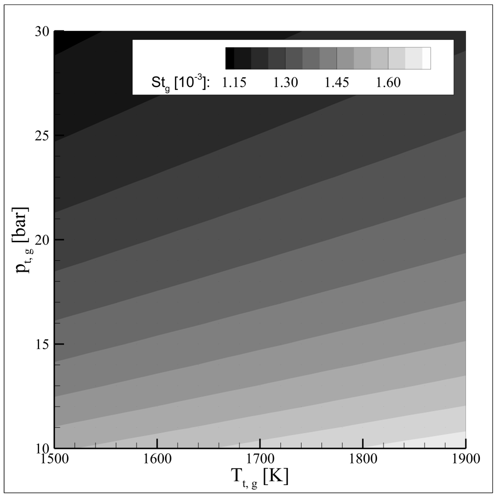

Fig.4 Sensitivity of averaged Stanton number St_g on turbine inlet conditions for NASA E3 P&W HPT S1

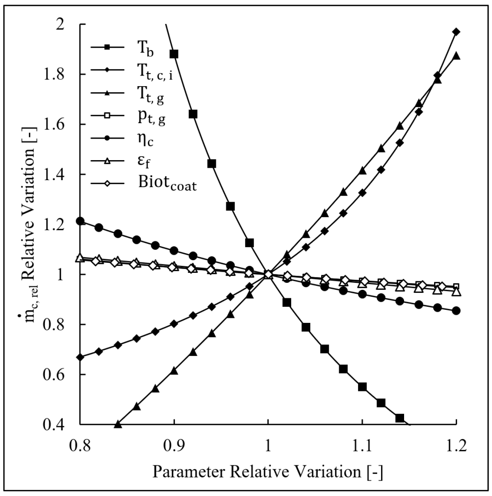

Fig.5 Sensitivity of cooling mass flow on cooling model parameters for NASA E3 P&W HPT S1

The sensitivity of cooling requirements on the cooling model parameters is briefly introduced in Fig. 5. Turbine total inlet temperature $T_{t,g}$ and pressure $p_{t,g}$, defined by the thermodynamic cycle, significantly affect cooling requirements $\dot{m}_{c,rel}$. In terms of new technologies, materials with higher maximum allowable temperatures T_b, such as ceramic matrix composites [17], should provide the biggest reduction in cooling requirements. This is followed by innovative internal cooling configurations η_c facilitated by additive manufacturing [18], advancements in film cooling techniques ε_f and finally lower thermal conductivity coatings Bi_{coat}.

CONCLUSIONS

In the present work an established cooling model is implemented into a turbine preliminary design tool and calibrated for two cooled high-pressure turbines in a collaborative process between turbine and performance modeling. The calibrated cooling model parameters are discussed and preliminary parametric analysis provide insights on possible approaches to reduce cooling requirements. The following conclusions can be taken from the presented results:

- The estimated ratio between blade surface area and mainstream flow cross-sectional area ranges between $12-21$ for stator rows and $5-11$ for rotor rows. Considering the linear dependence of the cooling requirements on this parameter, it is critical to provide an accurate estimation at an early design stage. A discrepancy between stator and rotor cooling requirements is ultimately attenuated by the expected higher averaged mainstream flow Stanton number in rotor rows.
- The ratio between blade surface area and chord length appears to be correlated with flow deflection and blade maximum thickness to chord. Additional data is still required to verify this statement.
- All blade rows exhibit a combined cooling efficiency in the range $0.45-0.73$, with the exception of GE's S2. The low cooling mass flow combined with impingement cooling and no film cooling leads to a higher cooling efficiency in this blade row.
- The averaged mainstream flow Stanton number is dependent on the turbine inlet temperature and pressure conditions. This should be taken into account when dimensioning the turbine cooling requirements within its operating range.
- New high temperature material technologies appear to provide the greatest potential for reducing cooling requirements, followed by new internal cooling configurations, improved film cooling techniques and enhanced thermal barrier coatings.

Future work includes a detailed analysis between the 1D tool `PrEDiCT` and the 2D cooling design tool `PICCOOLO` as well as the implementation of a zooming process between the two.

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REFERENCES

[1] Krumme, A. "Konzeption, Implementierung Und Anwendung Eines Automatisierten Aerothermodynamischen Vorentwurfsprozesses Fuer Axialturbinen". Universitaet Kassel, 2015.

[2] Grieb, H. *Projektierung von Turboflugtriebwerken*. Basel: Springer Basel, 2004. DOI: 10.1007/978-3-0348-7938-5.

[3] Holland, M. and Thake, T. "Rotor Blade Cooling in High Pressure Turbines". In: *Journal of Aircraft* 17.6 (1980), pp. 412–418. DOI: 10.2514/3.44668.

[4] Young, J. and Wilcock, R. "Modeling the Air-Cooled Gas Turbine: Part 2 - Coolant Flows and Losses". In: *Journal of Turbomachinery* 124.2 (2002), pp. 214–221. DOI: 10.1115/1.1415038.

[5] Torbidoni, L. and Horlock, J. "A New Method to Calculate the Coolant Requirements of a High-Temperature Gas Turbine Blade". In: *Journal of Turbomachinery* 127.1 (2005), pp. 191–199. DOI: 10.1115/1.1811100.

[6] Halila, E., Lenahan, D., and Thomas, T. *Energy Efficient Engine - High Pressure Turbine - Test Hardware Detailed Design Report*. CR-167955. NASA, 1982.

[7] Thulin, R., Howe, D., and Singer, I. *Energy Efficient Engine - High-Pressure Turbine Detailed Design Report*. CR-165608. NASA, 1982.

[8] Schöffler, R., Grunwitz, C., and Brakmann, R. G. "A Semi-Empirical Model for Conceptual Turbine Vane Cooling Design and Optimization". In: *Proceedings of ASME Turbo Expo 2023*. ASME. Boston, USA, 2023.

[9] Grunwitz, C. et al. "A Comprehensive Multifidelity Design and Analysis Process for Cooled Axial Flow Turbines: From Concept to Component". In: *Proceedings of the International Gas Turbine Congress 2023*. IGTC. Kyoto, Japan, 2023.

[10] Sanjay, Singh, O., and Prasad, B. "Influence of Different Means of Turbine Blade Cooling on the Thermodynamic Performance of Combined Cycle". In: *Applied Thermal Engineering* 28.17-18 (2008), pp. 2315–2326. DOI: 10.1016/j.applthermaleng.2008.01.022.

[11] Aungier, R. *Turbine Aerodynamics: Axial-Flow and Radial-Flow Turbine Design and Analysis*. New York: ASME Press, 2006. DOI: 10.1115/1.802418.

[12] Louis, J. "Systematic Studies of Heat Transfer and Film Cooling Effectiveness". In: *AGARD 229 - High Temperature Problems in Gas Turbine Engines*. 1977.

[13] Reitenbach, S. et al. "Collaborative Aircraft Engine Preliminary Design Using a Virtual Engine Platform, Part A: Architecture and Methodology". In: *AIAA Scitech 2020 Forum*. AIAA Scitech. Orlando, USA: American Institute of Aeronautics and Astronautics, 2020. DOI: 10.2514/6.2020-0867.

[14] Wehrel, P. and Carvalho, F. "Cooling Model Calibration in a Collaborative Turbine Preliminary Design Process Using the NASA Energy Efficient Engine Part I: 0D Performance Modeling". In: *Proceedings of the International Gas Turbine Congress 2023*. IGTC. Kyoto, Japan, 2023.

[15] Horlock, J., Watson, D., and Jones, T. "Limitations on Gas Turbine Performance Imposed by Large Turbine Cooling Flows". In: *Journal of Engineering for Gas Turbines and Power* 123.3 (2001), pp. 487–494. DOI: 10.1115/1.1373398.

[16] Bolland, O. "Gas Turbine Cooling Model for Evaluation of Novel Cycles". In: *Proceedings of ECOS 2005*. ECOS. Trondheim, Norway, 2005.

[17] Wehrel, P. et al. "Performance and Emissions Benefits of Cooled Ceramic Matrix Composite Vanes for High Pressure Turbines". In: *Proceedings of ASME Turbo Expo 2023*. ASME. Boston, USA: American Society of Mechanical Engineers, 2023.

[18] Brakmann, R. G. et al. "A Numerical Analysis of Cross-Flow Reinforced Impingement Cooling with a U-Shaped Flow-Guide on the Hole Plate". In: *Proceedings of ASME Turbo Expo 2023*. ASME. Boston, USA, 2023.