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Blade cooling optimisation in humid-air and steam-injected gas turbines

J.P.E. Cleeton *, R.M. Kavanagh, G.T. Parks

Department of Engineering, University of Cambridge, Trumpington Street, Cambridge CB2 1PZ, UK

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ABSTRACT

Humidified gas turbine cycles such as the humidified air turbine (HAT) and the steam-injected gas turbine (STIG) present exciting new prospects for industrial gas turbine technology, potentially offering greatly increased work outputs and cycle efficiencies at moderate costs. The availability of humidified air or steam in such cycles also presents new opportunities in blade and disk cooling architecture. Here, the blade cooling optimisation of a HAT cycle and a STIG cycle is considered, first by optimising the choice of coolant bleeds for a reference cycle, then by a full parametric optimisation of the cycle to consider a range of optimised designs. It was found that the coolant demand reductions which can be achieved in the HAT cycle using humidified or post-aftercooled coolant are compromised by the increase in the required compression work. Furthermore, full parametric optimisation showed that higher water flowrates were required to prevent boiling within the system. This corresponded to higher work outputs, but lower cycle efficiencies. When optimising the choice of coolant bleeds in the STIG cycle, it was found that bleeding steam for cooling purposes reduced the steam available for power augmentation and thus compromised work output, but that this could largely be overcome by reducing the steam superheat to give useful cycle efficiency gains.

1. Introduction

Humidified power cycles present one of the key areas of development within the gas turbine industry. Cycles such as the humidified air turbine (HAT) or the steam-injected gas turbine (STIG) promise work outputs and cycle efficiencies comparable to those of a combined-cycle gas turbine, but at a capital cost more comparable to a conventional gas turbine cycle. Gas turbines such as the GE LM2500 STIG have started to be used commercially [1], but the HAT cycle remains at the pilot plant stage, with the 600 kW_e Volvo VT600 derivative at Lund University being the first working pilot plant [2]. Such cycles also present the opportunity for new blade and disk cooling architectures, whereby humidified air or steam, coolants with higher specific heat capacities than air, may be used in turbine blade and disk cooling, currently one of the limiting factors in optimising gas turbine performance.

In the first part of this investigation, the coolant bleed configuration within feasible HAT and STIG cycle layouts is optimised with respect to cycle efficiency, and the effects and consequences of bleeding coolant from the saturator, in the case of the HAT cycle, or the heat recovery steam generator (HRSG) and steam/air mixer for the STIG cycle are considered. The remaining cycle parameters were fixed: the coolant bleed points are the only variables considered. It is important to differentiate between the optimal thermodynamic configuration and a practical configuration: both were investigated. In the optimal thermodynamic configuration no constraint was placed on the number of coolant bleeds that could be used, but for the practical configuration, where cost and complexity must necessarily be considered, the number of bleeds was restricted to three, which is typical of an industrial gas turbine.

In the second half of the investigation, Pareto-optimal sets of designs for the HAT and STIG cycles were produced, and a tabu search (TS) algorithm, developed by Jaeggi et al. [3] and implemented on humid cycles by Kavanagh [4], was used to carry out a parametric optimisation of the cycles. The humidified cycle output parameters were calculated using FORTRAN code developed by Aramayo-Prudencio [5] and Kavanagh in the University of Cambridge Department of Engineering, based on code developed for standard industrial gas turbine cycles by Young and Wilcock [6], in the same department. All of the cycles discussed consisted of two stages (or four blade rows) of blade cooling, and three stages of disk cooling. However, since it is very difficult to model disk cooling accurately, the disk cooling flows are pre-determined, with the focus in this investigation being placed on turbine blade cooling.

2. Analysis and optimisation methods

2.1. The blade cooling model

The blade cooling model used is that used by Wilcock [7], shown in Fig 1: variations of the model may be valid for other

* Corresponding author. Tel.: +44 7743688191; fax: +44 (0) 1223 3 32662.
E-mail address: jpec2@cam.ac.uk (J.P.E. Cleeton).

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Nomenclature

A_g	blade throat area (m^2)
A_{ij}	Mason and Saxena modification factor
A_{surf}	total external blade surface area (m^2)
c_p	specific heat capacity at constant pressure (kJ/kg K)
f	vector of objective functions
K_{cool}	dimensionless cooling flow factor
M	molar mass (kg/kmol)
m	mass flow-rate (kg/s)
N	number of design variables
n	number of mainstream gas constituents
Nu	Nusselt number
Pr	Prandtl number
Re	Reynolds number
r_p	pressure ratio
St	Stanton number
T	temperature (K)
x	vector of design variables
y	mole fraction
λ	thermal conductivity (W/m K)
μ	dynamic viscosity (kg/m s)
ω	specific humidity (kg_v/kg_{tot})
η_{ov}	overall efficiency (%)

Abbreviations

AC	Aftercooler
ECO	economiser
FPT	free-power turbine
H&J	Hooke and Jeeves
HAT	humidified air turbine
HP	high pressure
HRSG	heat recovery steam generator
IC	intercooler
IM	intensification memory
IP	intermediate pressure
LP	low pressure
LTM	long term memory
MTM	medium term memory
REC	recuperator
SAT	saturator
ST	steam from the HRSG
STM	short term memory
STIG	steam-injected gas turbine
SA	steam/air mixture
TBC	thermal barrier coating
TS	tabu search

Subscripts

aw	adiabatic blade wall
c	coolant
g	mainstream gas
i	inlet
w	blade wall
x	exit
0	stagnation property

applications where film cooling is utilised. Coolant bled at state k is transported to the blade cooling passages, entering the blade at state i (assumed to be identical to k). When only convective cooling is used, the coolant leaves the blade at the trailing edge and tip at states t and b, respectively. When film cooling is also considered, some coolant leaves the blade through various holes across the blade surface, where the coolant is assumed to be at state f everywhere, and at the endwalls at state e. An extended Holland and Thake model [8], as used by Young and Wilcock [6], is then used

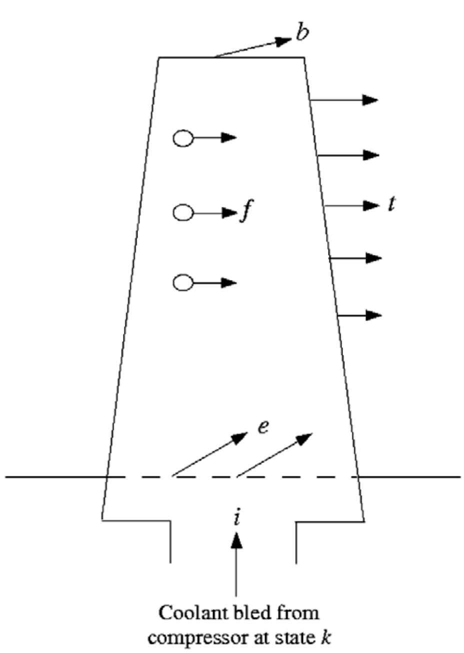

Fig. 1. Schematic diagram of a cooled turbine blade.

to carry out an enthalpy balance between the coolant and mainstream gas, which is equal to the total blade surface heat transfer. This is also equal to the heat transfer via conduction through the blade metal, and any thermal barrier coating.

The coolant mass flow-rate with respect to the compressor inlet mass flow-rate (m_c/m_g) can be defined in terms of a dimensionless coefficient K_{cool}, and a dimensionless coolant mass flow-rate m_{c+} as follows:

$$m_c/m_g = K_{cool} m_{c+} \tag{1}$$

where

$$K_{cool} = \frac{A_{surf}}{A_g}\frac{c_{p,g}}{c_{p,c}} St \tag{2}$$

$$m_{c+} = \frac{T_{aw} - T_w}{T_{0c,x} - T_{0c,i}} \tag{3}$$

The value of K_{cool} is therefore proportional to the mainstream Stanton number, which is itself based on an empirical relationship with the gas Reynolds and Prandtl numbers given by Torbidoni and Horlock [9]

$$St_g = 0.285 Re_g^{-0.37} Pr_g^{-2/3} \tag{4}$$

K_{cool} is also inversely proportional to the coolant specific heat capacity, indicating that a coolant with a higher c_p will reduce coolant mass flow-rate, as one might expect. m_{c+} meanwhile is dependent on the efficiencies of the cooling technology, and on blade material and geometry (and is consequently of less interest for this investigation). Hence the key to minimising cooling demands is minimising K_{cool}.

The focus of this investigation will be blade cooling and not disk cooling. This is because disk cooling flows are very difficult to calculate accurately: wherever disk cooling is necessary, the disk coolant mass flow-rates (relative to the turbine flow-rate) will be pre-specified. This is consistent with the work of Wilcock [7].