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Conjugate heat transfer of a rib-roughened internal turbine blade cooling channel using large eddy simulation

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ABSTRACT

This contribution focuses on the conjugate heat transfer computation of a rib-roughened internal turbine blade cooling channel with a blockage ratio of 0.3 and a Reynolds number of 40,000. The work considers the coupling between two numerical tools: a Large Eddy Simulation flow solver and a solid conduction solver. While interested in the thermal steady states of such problems, both solvers provide time dependent de-synchronized solutions. A novel weak coupling strategy, the hFFB method, is applied to solve the challenge of varying fluid and solid time scales. This method first computes the fluid domain until it reaches a statistically steady state, from which the mean temporal solution is obtained. Then, it solves the solid domain with the mean boundary conditions imposed from the fluid leading to an updated fluid boundary condition. This process is repeated until convergence of the coupled problem. The coupled convective-conductive solution shows that conjugate heat transfer is important for rib-roughened internal cooling ducts and has a large impact on the heat transfer at the solid-fluid interface. The results were validated against experimental data provided by the von Karman Institute for Fluid Dynamics.

1. Introduction

In anticipation of claims for environmental awareness and the rising use of energy, the turbomachinery research targets improvements of the gas turbine efficiency, driving new gas turbine designs to higher pressure ratios and increasing turbine inlet temperatures. Since the life of a turbine blade is reported to be reduced by half with an increased temperature of 15–30 K (Han et al., 2000), the prediction of local heat transfer coefficients and temperatures is more than ever a crucial step towards the design of reliable and efficient turbine blades.

Internal cooling channels have largely contributed to the increase of the turbine inlet temperature over the past decades. Rough surfaces have been introduced, investigated and developed to enhance the heat transfer by turbulating the flow, as reviewed by Han et al. (2000); Han (2010); Iacovides and Launder (1995); 2007); Weigand et al. (2001) and Ligrani (2013). To date, ribs have one of the highest potential among roughness elements to turbulate the flow and increase the heat transfer, therefore, a large effort has been put to investigate the flow and the heat transfer of these channels. Experimental investigations of the flow field in rib-roughened channels were done, e.g., by Rau et al. (1998) and Casarsa and Arts (2005). Studies on the heat transfer were done by Cakan (2000); Coletti et al. (2011) and Cukurel and Arts (2013).

Cooling channels were numerically investigated with RANS by, e.g., Ooi et al. (2002) and more recently, e.g., by Keshmiri (2012). Recent research targets and investigates further possible improvements on the RANS modeling, e.g., with non-linear eddy-viscosity modelsRaisee et al. (2009). With LES, first studies were done by Ciofalo and Collins (1992) and Yang and Ferziger (1993). To name a few, more recent studies were done by Cui et al. (2003); Fransen et al. (2012); Leonardi et al. (2004); Lohasz et al. (2006); Miyake et al. (2002) and Tyacke and Tucker (2015).

Despite the vast amount of studies on roughened cooling channels, except for few experimental (Coletti et al. (2012) and Cukurel and Arts (2013)), or two-dimensional, steady numerical investigations (Iaccarino et al. (2002)), the prevailing research neglects the influence of the thermal boundary condition on the heat transfer, even though the impact of the wall temperature distribution can be significant, as indicated by the experiment of Cukurel and Arts (2013).

On the numerical side, current numerical methods call for new numerical strategies. Two reasons explain the current status: first, the complex geometries with high asperities challenge the numerical models. Second, the generally neglected conjugate heat transfer needs to be taken into account for reliable thermal predictions.

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Nomenclature

Bi	Biot number $[ms^{-1}]$
c	speed of sound $[ms^{-1}]$
D_h	Hydraulic diameter $[m]$
D	Thermal diffusivity $[m^2s^{-1}]$
EF	enhancement factor
F	flux tensor
f	friction factor
H	rib height $[m]$
h	heat transfer coefficient $[Wm^{-2}K]$
L	length of channel model $[m]$
Ma	Mach number
Nu	Nusselt number
P	pitch length in the ribbed channel $[m]$
p	pressure $[Nm^{-2}]$
Pr	Prandtl number
q	heat flux $[Wm^{-2}]$
R	gas constant $[J^1K^{-1}mol^{-1}]$
Re	Reynolds number
T	temperature $[K]$
t	time $[s]$
u, v, w	velocity components $[ms^{-1}]$
W	vector of primary variables
w	quantity at the wall
$y+, x+, z+$	non-dimensional measures of the first cell to the wall
Δx	grid cell size $[m]$
Δt	time step $[s]$
ν	kinematic viscosity $[m^2s^{-1}]$
μ	dynamic viscosity $[Pa^1s^1]$
ρ	density $[kg^1m^{-3}]$
λ	thermal conductivity $[Wm^{-1}K^{-1}]$
τ_w	wall shear stress $[Wm^{-1}K^{-1}]$
CFL	Courant-Friedrich-Levy number
LES	Large Eddy Simulation
NSCBC	Navier-Stokes Characteristic Boundary Condition
PIV	Particle Image Velocimetry
SGS	Sub-grid Scale
RANS	Reynolds-averaged Navier-Stokes
∞	free stream quantity
i	iteration step in coupling cycle
f	fluid
s	solid

Conjugate heat transfer describes the thermal interaction between the fluid and the solid domain. If the temperature distribution would be known at the interface a priori, the problem would reduce to two single domain uncoupled problems. However, the temperature and heat flux distributions are unknown a priori and are part of the coupled solution of the fluid and solid energy equations. The temperature and heat flux distributions can only be found by solving both domains in a coupled manner, which is the conjugate heat transfer problem as first described and named by (Perelmann, 1961).

The coupling of fluid and solid solvers is a numerically demanding task, as stability problems can occur. Furthermore, the time scales in the fluid and solid domain may vary by orders of magnitude. For ribbed ducts, LES have demonstrated a high accuracy for the turbulent flow field and heat transfer predictions.

However, LES are CPU consuming and amplify the problem of varying time scales in solid and fluid domains, even though recent research also targets a reduction in cost for LES using a zonal wall treatment for ribbed ducts (e.g., Patil and Tafti (2013). Due to the existing challenges, only few computations of conjugate heat transfer (CHT) with LES have been done (e.g., Duchaine et al. (2009, 2013)) and research is still needed for an efficient LES based CHT Strategy.

The present contribution aims at improving the current state of the art thermal predictions for roughened cooling channels and investigates the influence of the thermal boundary condition on the heat transfer inside a rib-toughened cooling channel by comparing LES with iso-flux boundary conditions to coupled convective-conductive computations.

This paper is laid out as follows: the next section describes the used solvers for fluid and solid domain as well as the used coupling approach for the CHT problem. The results have been compared with experimental data by the Von Karman Institute for Fluid Dynamics subsequently for the mean flow field, turbulence statistics and the heat transfer.

Table 1
Channel and flow characteristics Cukurel (2012).

Parameter	Abbr.	Value
Reynolds number	Re_{D_h}	40,000
number of ribs		6
length	L_f	1597.5 mm
hydraulic diameter	D_h	75 mm
rib height	H	22.5 mm
pitch to rib ratio	P/H	10
blockage ratio	H/D_h	0.3

2. Problem definition

The studied geometry models the experimental facility of Cakan (2000); Cukurel (2012) and Cukurel and Arts (2013), which represented an internal rib-roughened aircraft gas turbine blade cooling passage with a squared cross-section, a vertical alignment of the ribs to the flow streamwise direction, a pitch to rib ratio of 10 and a blockage ratio of 0.3 (Fig. 1). The cooling passage was scaled up by a factor of 15 with respect to real engine conditions (Cukurel, 2012), to warrant a high measurement resolution (Cakan, 2000).

The experimental test section with its six ribs was fully represented by the configuration for the numerical simulations (Fig. 2). Only a reduced length of the inlet and outlet sections were simulated with appropriate boundary conditions. The cross section dimensions were 75 × 75 mm and the test section length was 1260 mm (Table 1). Although the rib of the cooling channel repeats itself periodically, the total number of six ribs was kept similar to the experimental model to avoid possible arbitrary effects of periodic boundary conditions that occur as Fransen et al. (2012) have shown. Indeed, the flow field needs more than one rib distance to establish a fully developed pattern independent of the inflow and previous ribs. If assuming that the largest scales inside the channel are of the order of the channel diameter, which corresponds to around one third of the rib to rib distance, the actual axial distance of the ribbed pipe is not enough to establish an uncorrelated flow field as it would be the case if only one pitch with periodic boundary conditions would be used for the simulation domain. Furthermore, the use of periodic boundary conditions would include major complications for CHT computations. Besides technical difficulties for the implementation of periodic boundary conditions for CHT computations, it would not be possible to include temperature dependent material and flow properties. Moreover, the temperature profiles inside the domain are not know a priori and it would be a questionable assumption that the temperature profiles at the inlet and outlet periodic boundaries are identical, both for solid and fluid domain. This assumption would