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# Effects of Rotation and Buoyancy Forces on the Flow Field Behavior Inside a Triangular Rib Roughened Channel

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#### 1 ABSTRACT

2 The flow field inside a triangular cooling channel for the leading edge of a gas turbine blade has 3 been investigated. The efforts were focused on the investigation of the interaction between effects of 4 rotation, of buoyancy forces, and those induced by turbulence promoters, i.e. perpendicular square 5 ribs placed on both leading and trailing sides of the duct. PIV and Stereo-PIV measurements have been performed for  $\text{Re}_{Dh}=10^4$ , rotation number of 0, 0.2, and 0.6, and buoyancy parameter equal to 0, 6 7 0.08, and 0.7. Coriolis secondary flows are detected in the duct cross section, but contrary to the 8 smooth case, they are characterized by a single main vortex and are less affected by an increase of 9 the rotation parameter. Moreover, their main topology is only marginally sensitive to the buoyancy 10 forces. Conversely, the features of the recirculation structure downstream the ribs turned out to be 11 more sensitive to both the buoyancy forces and to the stabilizing/de-stabilizing effect on the separated 12 shear layer induced by rotation.

13 Keywords: blade cooling, internal cooling, rotating channel, buoyancy, rotational effects, PIV

14

#### 15 NOMENCLATURE

16Achannel cross sectional area without ribs17Bobuoyancy parameter18Boxbuoyancy parameter at a defined x (radial) position19 $D_h = 4A/p$ hydraulic diameter20LSleading side

21	$Nu_0$	Nusselt number computed with Dittus-Boelter correlation
22	Nus	Nusselt number for the static case of [12]
23	р	test section perimeter without ribs
24	R <sub>x</sub>	radius of rotation
25	Re <sub>Dh</sub>	Reynolds number
26	Ro	rotation parameter
27	T <sub>b</sub>	bulk flow temperature
28	$T_w$	wall temperature
29	TS	trailing side
30	Tu	Turbulence intensity computed on u,v,w
31	Tu <sub>xy</sub> '	Turbulence intensity computed on u,v'
32	u,v,v',w	velocity fluctuations on x, y, y', z directions
33	U <sub>b</sub>	bulk flow velocity
34	U,V,V',W	mean velocity components on x, y, y', z directions
35	x,y,z	radial, peripheral and normal coordinates
36	у'	wall normal direction
37	Ω	rotational speed
38		

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#### 40 **1. INTRODUCTION**

The study about the effects introduced by rotation on the thermal and flow fields inside internal cooling passages adopted in the rotor blades of gas turbines has recently seen a growing interest in the research community. Indeed, dedicated test rigs and experimental methodologies have been developed in the last years with the specific aim to investigate these complex flow and thermal fields. When a specific cooling passage is subjected to rotation, two are the main physical phenomena that can have an impact on the coolant path: the rise of Coriolis forces and the appearance of buoyancy 47 effects [1, 2]. The former effect is usually characterized by the rotation parameter defined as  $Ro=\Omega D_h/U_b$  and which takes into account the relative importance between peripheral velocity and 48 49 bulk flow velocity. More precisely, the Coriolis acceleration ( $a_c=-2\Omega xC$ , where C is the relative 50 velocity vector) that acts on the relative flow inside the duct causes the establishment of a spanwise 51 pressure gradient that counterbalances its effects (Fig.1). If a confined flow with non-uniform velocity 52 distribution is considered, a local unbalance between pressure gradient and Coriolis forces is found 53 inside the regions of lower momentum fluid, hence a local flow deflection is induced and a secondary 54 vortical flow establishes in the duct cross section, as sketched in Fig. 1. In the rather common case of 55 a radial outward flow inside a rectangular channel in orthogonal rotation (i.e. with the rotation axis 56 parallel to the duct height, the situation sketched in Fig. 1), the slower near wall flow is pushed towards the leading side (LS) of the duct by the Coriolis induced pressure gradient. Consequently, for 57 58 continuity, the core fluid is displaced towards the trailing side (TS) of the duct and a symmetrical pair 59 of vortex cells rises (Fig.1) [3].

60 The second main effect of rotation is described by the buoyancy parameter defined as:

$$61 \qquad Bo = Ro^2 \frac{R_x}{D_h} \frac{T_w - T_b}{T_b}$$

62 and which expresses the ratio between buoyancy forces and bulk flow inertial forces. The Bo parameter is proportional to  $Ro^2$  and depends on the ratio between the local radial position ( $R_x$ ) and 63 64 the duct hydraulic diameter (D<sub>h</sub>). The effect is triggered by a non-uniform fluid temperature, hence density, distribution that can be characterized by the temperature difference existing between the walls 65 (T<sub>w</sub>, usually hotter than the fluid) and the bulk flow (T<sub>b</sub>). More precisely, the hotter (hence with lower 66 67 density) near wall fluid experiences a lower centrifugal force than the colder (hence with higher density) core fluid. This, can alter substantially the flow topology and consequently the heat transfer 68 field with respect to an isothermal situation  $(T_w=T_b)$ . 69

The most recent contributions in literature that deepen the comprehension about these phenomena, with specific reference to gas turbine cooling channels, deal with the study of the interaction between rotation effects and those induced by the adoption of turbulence promoters on the channel walls [4]. 73 Complex geometries have also been investigated, such as triangular [5], trapezoidal shaped [6, 7] and 74 multipass channels with bends [8, 9]. For a simplified channel configuration (single-side rib-75 roughened rectangular channel with radial outward flow) Coletti et al. [4] showed that the 76 separating/reattaching flow pattern promoted by the ribs is highly affected by rotation. In particular, 77 if the background flow vorticity has the same orientation of the angular velocity of the rotating system (e.g. inside the boundary layer on the LS of a rectangular channel with radial outward flow) a 78 79 stabilizing effect on the boundary layer is induced and a wider recirculation area is found downstream 80 of the obstacles, vice versa occurs at the TS. If non-isothermal conditions are also considered [10], buoyancy effects cause an higher mixing in the near wall flow, with much higher effects on the 81 82 stabilized side wall where the extension of the separated flow region is further increased, conversely less significant differences are found about the extension of the recirculating flow on the opposite 83 84 wall.

85 In the case of rectangular or square channels in orthogonal rotation the basic description about the rotational effects reported above allows a rather good prediction about the main flow features that 86 rise inside the passage. On the other hand, if more complex geometries are considered, the analysis 87 88 is rather less trivial as shown by [6, 7] for a trapezoidal channel used for trailing edge cooling or in 89 case of a triangular leading edge passage investigated in [5]. In particular, the latter contribution 90 showed that inside a smooth triangular channel a non-symmetrical Coriolis vortices pair is found, 91 with separation and reattachment points that are no longer located about the mid span of the lateral 92 walls, but are instead located on the triangle upper apexes. Moreover, a steady flow configuration is 93 never reached, indeed as the flow develops along the channel, more than two Coriolis vortical cells 94 are found inside the channel cross section. The phenomenon is caused by a periodical reversal of the spanwise velocity component (i.e. the velocity component normal to both the rotation axis and the 95 96 radial direction) inside the low momentum region found at the leading apex. The overall behaviour is 97 strongly fastened if Ro is increased.

98 The present work is the first part of a research activity that aims at extending the previous findings

99 about the rotational effects inside triangular shaped ducts by considering also the effects of turbulence 100 promoters and by introducing buoyancy effects. Indeed, concerning leading edge cooling systems, 101 the wide open literature provides an abundance of works focused on thermal investigations and rather 102 complex geometries such as channels with curved walls [11] or systems where cooling is provided 103 by means of jet impingement combined with coolant extraction [12]. Moreover, if rotating channels 104 are considered, the availability of results is further reduced in regards of experimental works and flow 105 field investigations.

For the reasons thoroughly explained in the previous discussion on rotation effects in cooling channels, since information available on simple geometries (i.e. rectangular channel) cannot straightforwardly apply to more complex ones (i.e. triangular channel), this work frames in the logic of understanding the main flow features inside triangular channels. The final goal is to create a background knowledge comparable to the already consolidated on rectangular channels [3, 4, 10] to deepen the comprehension of the thermal studies.

In order to fulfil this purpose, experiments have been conducted on the same rig used by Pascotto et al. [5] but on a modified test section that allows the heating of the lateral walls that have also been equipped with square orthogonal ribs. As in Pascotto et al. [5], the choice for this geometry (i.e. a single passage duct with an equilateral triangle cross section and without extraction holes for film cooling), beside the previous discussion, is motivated by the availability of a wide thermal analysis provided in [13, 14] and that can be complemented by the present findings.

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#### 119 **2. TEST RIG AND EXPERIMENTAL METHODOLOGY**

#### 120 **2.1 Test rig**

121 The present experimental activity has been conducted on the facility of the Turbomachinery and 122 Energy Systems Laboratory of the University of Udine. A detailed description of the rig can be found 123 in [6]. The facility has been designed to allow detailed flow field investigations inside rotating ducts 124 representative of modern cooling system for gas turbine blades and has been recently upgraded for thermal measurements as well. In particular the system has been provided with a rotary fluidic joint (Fig. 2(a)) equipped with 32 slip rings used for signal and electric power transmission on board of the rotating system.

128 The test section consists of a straight channel 820 mm long with an equilateral triangle cross section 129 of side L=130 mm, (Fig. 2). Square ribs of height h=6.5 mm are placed perpendicularly to the channel main axis on the two lateral walls. The ribs have a pitch of 8h and have a staggered disposition with 130 131 respect to the two channel sides, i.e. with the same geometrical characteristics as in [13, 14]. In order 132 to investigate buoyancy effects, each ribbed wall has been machined out of a single aluminium slab. 133 Heating is provided by powering with direct current a 25 µm thick Inconel sheet attached on the 134 external surface of the wall (Fig. 2(e)). An external layer of insulating material minimizes the thermal losses towards the environment. The temperature of each heated wall is monitored with four 135 thermocouples installed inside the aluminium walls (Figs. 2(c, e)) in the inter-rib regions just 136 137 upstream and downstream the investigated area (planes yz and xy' in Figs. 2(c, d)). The calibration of the temperature sensor (connected to the slip rings) and a data averaging process computed on 138 139 stencils of 20 samples acquired at 2 Hz lead to an overall uncertainty of ±0.75 K. Hot test conditions 140 have been obtained by heating the ribbed walls up to 110 °C. Differences of 1K have been observed between the readings of two thermocouples placed on similar points of the same wall but in the two 141 142 different inter-rib locations, while the maximum difference between the hottest and the coldest 143 readings was equal to 5 K and has been observed for tests at the highest rotational speed.

The test section is fed with air at ambient conditions through a rotating settling chamber connected to the rotary fluidic joint. The settling chamber design (Fig. 2(a)) is such that a uniform flow distribution is ensured at the duct inlet, where a 40mm long honeycomb filter prevents also from flow separation. At the channel outlet, the exhausted flow is collected in a small settling chamber and then leaves the test section through a return passage, so avoiding any flow perturbation near the tip region that could conversely occur under rotation if a free radial outlet was set. The return channel was also adopted by [13, 14], but in their case it was necessary in order to pressurize the channel and, therefore, 151 to reach very high rotation numbers.

All tests were performed with the channel z axis aligned with the angular velocity, as depicted in Fig.
The test section was spun in both sense of rotation, in order to investigate the flow field on both
leading and trailing sides of the channel without changing the PIV measurement set up (see next
section).

#### 156 **2.2 Flow field measurement methodology**

Flow field measurements were performed by means of 2D and Stereo-PIV techniques. A single camera configuration (Fig. 3(a)) for 2D measurements was used on plane xy' (Fig. 2(d)), while a double camera configuration (Fig. 3(b)) for Stereo-PIV measurements was used on plane yz (Fig. 2(c)). The PIV system is custom-made and was already described in [7].

161 The PIV raw data (PIV image pairs) were processed using the commercial software PIVview (from PIVTEC Gmbh). In particular, for the Stereo-PIV data, image back-projection and then stereo 162 reconstruction were performed; furthermore, a disparity correction was also used in order to minimize 163 164 the misalignment errors [15]. Stereo calibration of the measurement chain was achieved inside the channel by means of multiple images of a target placed at first on the image plane and then displaced 165 166 in six adjunctive off-plane positions. The nominal magnification of the PIV images ranges from 17 pixel/mm (Stereo-PIV) to 20 pixel/mm (2D-PIV), which turns in a vector resolution from 1.06 to 1.25 167 168 vectors/mm, respectively.

169 As can be seen from Fig. 3, the present PIV system is fixed, i.e. not rotating with the test section. 170 Consequently, the measurements at Ro>0 are carried out in phase-locked mode and, therefore, the direct output of the PIV raw data is the absolute flow field. The relative field inside the test section is 171 obtained by subtracting the local peripheral displacement of the system from the PIV data with an ad-172 173 hoc procedure. For the case of PIV measurements conducted on flow planes perpendicular to the rotation axis, Armellini et. al [16] proposed a specific pre-processing methodology to be applied to 174 175 the PIV raw data (image pairs) and that allows to keep the peripheral displacement subtraction error 176 below 1%, i.e. allows to provide a final accuracy level close to the one of standard 2D measurements 177 on static test sections. Unfortunately, the aforementioned methodology cannot be directly used for the present case because the selected measurement planes are not perpendicular to the rotation axis (see 178 179 position of plane xy' in Figs. 2(d), 3(a) and 4). Indeed, since a component of the peripheral 180 displacement lies on the out-of-plane direction (Fig. 4), a correct image reconstruction would need a 181 precise knowledge of the perspective view realized by the optical system, resulting in a rather complex implementation of the methodology. In view of this, a different approach has been chosen, 182 183 namely the second exposure of each PIV image pair has been pre-processed by applying a rigid 184 translation of the image in order to reduce the walls' displacement at less than 1 pixel with respect to 185 the first exposure. This raw peripheral displacement subtraction allows the PIV image processing 186 algorithm to resolve particle displacements of standard magnitude ( $\pm 10$  pixel) and to impose the 187 correct boundary conditions on the processing mesh (i.e. no-slip condition at the solid walls). Successively, the displacement components applied for the rigid image translation are added to the 188 189 resulting data in order to restore the absolute velocity field. The post-processing procedure is then 190 completed by the subtraction of the real peripheral velocity field (the in plane component in Fig. 4), 191 which is accurately measured as described at the end of this section.

192 The processing of the 2D data is further complicated by the need to compensate for the parallax error 193 associated to the out-of-plane component of the peripheral displacement (plane xy' is not 194 perpendicular to the rotation axis, see Figs. 2(d), 3(a) and 4). More in detail, even if rather long focal 195 lenses have been used, it is not possible to assume that the light is collected with parallel rays. 196 Consequently, the out of plane component of the peripheral displacement will be seen by the camera 197 as an additional in plane displacement (i.e. the parallax error) which magnitude and orientation is 198 proportional to the distance between a specific measurement point and the intersection of the camera optical axis with the measurement plane (Fig. 4). Therefore, the peripheral displacement recorded by 199 200 the camera, is different from the real in plane one (which can be calculated as  $v = \Omega R\cos(\alpha)\Delta t$ , where 201 R is the radial position,  $\Omega$  is the measured angular velocity and  $\Delta t$  is the PIV separation time). For the present data on plane xy', the maximum magnitude of the out-of-plane peripheral displacement was 202

14 pixel, hence the resulting in-plane parallax component can be up to 0.9 pixel, which is a not negligible value. A precise parallax error compensation requires the knowledge of two main input data: the lenses focal length and the position of the optical axis projection on the image plane. For sake of accuracy, the actual value of the lens focal length has been directly measured with a dedicated procedure (i.e. to measure the size of the field of view framed with the present optical configuration) and turned out to be equal to 108.5 mm.

209 Optical axis projection position has been determined by an in-situ calibration realized with a target 210 that allows to frame straight lines aligned with the out-of-plane direction on planes either parallel 211 (Fig. 5(a)) or perpendicular (Fig. 5(b)) to the channel wall. An iterative procedure is carried out by 212 moving the target until the lines drawn on its surface are seen as points by the camera (procedure 213 exemplified in figures 5(a, b)), hence defining position of the projection of the camera optical axis on 214 the image plane. Examples of parallax error compensation displacement fields are reported in Figs. 215 5(c, d) for the case Ro=0.6. As can be seen, the projection of the lenses optical axis (highlighted by 216 the intersection of the two red lines in figures 5(c, d) has been placed close to the area where flow 217 recirculation is supposed to be found. This choice guarantees that the most critical area of the analysed 218 flow field is only marginally affected by the parallax error and by the possible uncertainties of its 219 compensation. It is worth to mention that the parallax correction has to be applied to the 2D 220 measurements only, since the Stereo 3D measurements are parallax free by definition.

221 The determination of the peripheral displacement field requires to know the real position of the measurement points with respect to the rotation axis, the test section angular velocity, and to ensure 222 223 a reliable synchronization of the PIV system with the test section angular position. Concerning the 224 centre of rotation position evaluation, a two steps indirect methodology has been used. As a first step, the procedure proposed in Armellini et al. [16] is followed, namely a target with a rectangular grid of 225 226 dots is placed inside the test section on plane z=0 and it is framed by an auxiliary camera with optical axis parallel to the axis of rotation (aux camera in Fig. 3(a)). Two exposures of the target are taken 227 with the test section at two angular positions. The markers centre positions are determined in the two 228

images by means of a cross-correlation with a sample signal and afterwards, the centre of rotation position can be determined applying a least square method on the markers displacement. Providing that the number of dots is sufficiently large, the centre of rotation with respect to a reference point of the target can be located with an accuracy of  $\pm 0.1$  mm. The second step requires to operate the laser and to acquire an image of the laser sheet trace on the calibrated target. The analysis of this realization allows to get an accurate determination of the position of the laser sheet trace with respect to the centre of rotation.

The last technical issues that have to be addressed are the synchronization of the PIV system with the test section rotation and the measurement of the actual peripheral velocity. Both these issues are solved by sampling the TTL signal from a photodiode (see Fig. 3(b)) periodically screened by the test section, all the technical details can be found in Armellini et al. [6]. The mean angular velocity for each revolution,  $\Omega$ , can be computed with a relative error equal to  $\delta\Omega/\Omega=3.5\times10^{-7}$ , while image positioning stability is kept by far smaller than the data spatial resolution (i.e. with image shifts about  $\pm 1$  pixel), so reducing to negligible values the spatial averaging error.

243

#### 244 **2.3 Uncertainty analysis**

245 Concerning PIV data accuracy, only statistical flow quantities will be presented, resulting from the ensemble average of 10<sup>3</sup> instantaneous samples. For a static 2D measurement, the overall upper bound 246 estimate of the mean velocities uncertainty results to be less than 2% with respect to Ub (95% 247 confidence level), while the maximum uncertainty in the estimate of the higher order statistics is 5% 248 of U<sub>b</sub>. For the data acquired under rotation, the pre- and post-processing procedures used for the 249 250 peripheral displacement field subtraction introduce further unavoidable error sources that increase the 251 overall uncertainty on the mean velocities up to 5% of U<sub>b</sub>. A proof of the reliability of this estimation is provided by Fig. 6, where a cross comparison between 2D and Stereo-PIV data is provided for the 252 253 most critical case of Ro=0.6. The plots clearly show a rather good superimposition of the data with 254 maximum differences of about 0.05Ub.

#### 255 **2.4 Test conditions**

All tests have been conducted at  $Re_{Dh}=10^4$  defined on the duct hydraulic diameter ( $D_h = 75.05$  mm), 256 on the flow bulk velocity (U<sub>b</sub>=2.1 m/s), and flow properties measured at the inlet of the test section 257 258 (see T<sub>air,in</sub> in Fig. 2(c)). Rotation numbers of 0.2 and 0.6 have been reached by spinning the test section 259 at about 55 and 160 rpm, respectively. Ribbed walls temperatures of about 110°C corresponded to Bo=0.08 and 0.7 for the two rotating conditions of Ro=0.2 and 0.6, respectively. The given Bo values 260 261 are computed at the radial position where the measurement plane yz is found ( $R_x=709$  mm, Fig. 2(c)). 262 The present working conditions are considered to be representative of current engine conditions and 263 have been selected in order to be consistent with the available studies [5, 13, 14]. 264 For computation of Bo, averaged values of both wall and bulk fluid temperatures have used, namely the average of the 8 thermocouples installed inside the aluminium walls for T<sub>w</sub>, and the average 265

between  $T_{air,in}$  and  $T_{air,out}$  for  $T_b$ . In view of the unavoidable measurement errors and the slightly nonuniform temperature distribution of the walls previously commented, an uncertainty of  $\pm 5\%$  has to be considered for the present definition of Bo. Concerning the computation of Re<sub>Dh</sub> and Ro values, the main error source comes from the measurement of the air flow rate that causes a final uncertainty of about  $\pm 1.5\%$ .

271

#### **3. RESULTS**

#### 273 **3.1 2D data on plane xy'**

The characterization of the separation structure downstream of the ribs is obtained by the data acquired on plane xy', i.e. the plane perpendicular to both the channel wall and the rib axis (Fig. 2(d)). Figure 7 reports the reference conditions measured for the static case. The time-averaged stream tracers' path depicts a separation structure that extends up to about x/h=3.5 and a corner vortex located on the upstream face of the following obstacle. Contour plots of the in-plane turbulence intensity,

279 
$$T_{xy} = \frac{1}{U_b} \sqrt[2]{\frac{u'^2 + v'^2}{2}}$$
, are reported in Fig. 7(b) (the withe areas that surrounds the ribs and close to the

bottom wall are zones of non valid data, which quality has been spoiled by unavoidable laser light reflection). As expected, the detected flow features are in agreement with those observed in a rectangular ribbed channel [17].

283 A summary of the results obtained under rotation is reported in Figs. 8 and 9 in terms of stream tracers 284 path and Tu<sub>xy</sub> levels, respectively (again, blanked regions are not valid data). When the investigated 285 wall is at the TS (Fig. 8(e-h)), no remarkable effects of both rotation and buoyancy are observed on 286 the flow topology, with the only exception of a small reduction of the recirculation extension found 287 for Ro=0.6. Similar observations are reported also in [10] for the case of a rectangular duct in 288 orthogonal rotation. The explanation of this behaviour is linked to the destabilizing effect that rotation 289 has on the separation structure at the TS, which causes rather strong flow turbulence that reduces the 290 bubble extension and mitigates local density gradients. Also in the present case, higher Tu<sub>xy</sub> are 291 measured at the TS (Fig. 9(e-h)) with respect to both static case (Fig. 7(b)) and LS (Fig. 9(a-d)). More 292 in detail, the Tu<sub>xv</sub> level increases with Ro, and this effect is enhanced by buoyancy.

On the LS, the behaviour is different depending by the Ro value. At Ro=0.2 (Fig. 8(a, b)), the 293 294 separation bubble appears longer than the static case (Fig. 7(a)), in agreement with the stabilizing 295 effect commented in [10]. Conversely, only marginal buoyancy effects are detected on both flow path 296 and Tu<sub>xy</sub> levels (Fig. 9(a, b)). This is also evident by the profiles of <uv'> extracted at y'/h=0.5 (see Fig. 2(e)) and reported in Fig. 10(a). Indeed, the comparison of heated/non-heated cases reveals 297 298 almost identical behaviours, while the destabilizing/stabilizing effects are rather clear as 299 demonstrated by the higher values and peaks closer to the obstacle found for TS cases. A different 300 behaviour is found at Ro=0.6 (Fig. 8(c, d)), where the separation bubble gets only marginally longer 301 with respect to the static case (Fig. 7(a)) and, unexpectedly, the flow agitation is higher for the non-302 heated case than the heated one (Figs. 9(c, d) and 10(b)). Nevertheless, this turned out to be only a 303 local feature, as it will be clarified later on with the plot of Fig. 14. Finally, it is also worth to observe 304 that the stream tracers inside the recirculation regions at the LS show a pronounced centripetal path 305 that is an indication of a relevant 3D character of the flow. This evidence could be at the basis of a

306 possible explanation of the differences commented about the present flow behaviour with respect to

307 the one found for a rectangular channel [10]. This issue is further discussed in the next section.

#### 308 **3.3 3D data on plane yz**

309 Stereo-PIV measurements in plane yz, located at 495 mm from the inlet section (downstream the 310 honeycomb filter, or 709 mm from the rotation axis, see Fig. 2(c)), have been conducted in order to 311 characterize the rotational effects on the overall flow structure. In order to ease the present data 312 analysis, Fig. 11 reports the results found for the smooth, non heated channel configuration obtained 313 on the same rig and reported in [5]. The Coriolis induced secondary flow is characterized by a vortex 314 pair at the lowest rotation condition (see vortices A and B in Fig. 11(a)) and shows a remarkable 315 sensitivity to the change of Ro. Indeed, at Ro=0.4, a third vortex (B2 in Fig. 11(b)) appears close to 316 the LS apex, producing a rather more complex flow distribution. Pascotto et al. [5] provided a 317 justification of this behaviour and shown how it grows in its complexity as rotation is further increased. Moreover, the streamwise velocity distribution turned out to be strongly asymmetric, with 318 319 higher magnitudes on the TS and a pronounced peak located near the lower apex (Fig 11(c, d)). These features were associated to the channel thermal behaviour reported in [13]. 320

321 The secondary flow distribution measured inside the ribbed duct is reported in the plots of Fig. 12. A 322 comparison with the smooth channel case of Fig. 11(a) can be made by looking at the data with the same sense of rotation (i.e. with the LS on the left, Figs. 12(a)-(d)). With respect to the smooth case, 323 324 the clockwise vortex previously found close to the lower apex of the duct (vortex B in Fig. 11(a)) is now rather bigger and extends almost over all the channel cross section (B in Fig. 12(a)). The counter-325 clockwise vortex (A in Fig. 12(a)) is confined close to the upper, non ribbed wall and appears to be 326 327 too small to be well captured at the present data resolution. Moreover, the local flow acceleration on top of the rib (which region is bounded by the 3D separation line that extends all along the rib span 328 at the TS, see Fig 12(a)) pushes the rotation induced vortices closer to the LS, causing an evident flow 329 330 deviation towards the upper wall in the inter rib region at the LS. This is in agreement with the 331 previous observation about the 3D nature of the separation found behind the obstacle (see comments

332 about Figs. 8(a-d)). Finally, if cases at different Ro are compared (see Figs. 12(a) and (c)) the 333 sensitivity of the flow topology on plane yz to the change of Ro is practically absent. Moreover, if 334 heated and non heated cases are compared, no clear effects of the buoyancy are found (see Figs. 12(b, 335 d) and (a, c)). More evident differences can be instead appreciated in the contour plots of the U 336 velocity component reported in Figs. 13(a-d) (blanked areas on top of the rib are regions of not available data due to optical accessibility). Indeed, as Ro and Bo are increased, higher velocity peaks 337 338 are found at the TS as a consequence of the stronger Coriolis induced secondary structures produced 339 at higher rotation. The absolute velocity peak on the lower apex that characterize the flow field inside 340 the smooth channel (Fig. 11(c-d) is instead no longer detected.

341 If the test section is spun in the opposite sense of rotation (Figs. 12(e-h) and 13(e-h)), the Coriolis 342 induced vortex B is correctly found on the right of the plots and with counter-clockwise rotation, i.e. 343 close to the LS and pushing core flow toward the TS. Vortex A is again smaller in size and remains 344 confined close to the upper, non ribbed wall. With respect to the previous comments, the main 345 differences concern the features of the flow in the inter-rib region at the TS that, conversely to what 346 happens at the LS, is by far less 3D (Figs. 12(e-h)) and is characterized by a more uniform U velocity 347 distribution (Figs. 13(e-h)). Again, slight effects caused by an increase of Ro and Bo are found only 348 in the U velocity peaks found close to the TS (Figs. 13(e-h)).

349 A final comment about the effect of rotation and buoyancy can be made by looking at the Tu contour 350 plots of Fig. 14. The stabilizing/destabilizing effect of rotation can be appreciated by comparing the Tu levels between leading and trailing sides at the same working conditions (compare frame (a) and 351 (c) or (b) and (d)), with higher values always observed at the TS. Similarly, the stronger flow mixing 352 promoted by buoyancy is associated to the overall higher Tu levels measured for the heated cases 353 354 (compare frame (a) and (b) or (c) and (d)). This in not in contrast with the comments about Fig. 10(b). 355 Indeed also in Fig. 14(b), at the position of plane xy' (see Fig. 2(d)) the Tu values for Bo=0.7 are smaller than those for non-heated case in Fig. 14(a). Therefore, the commented behaviour about the 356 separated flow length and agitation level at the LS, that differ substantially with respect to the 357

rectangular channel case [10], as to be ascribed to local flow effects that are peculiar of the triangular
geometry, demonstrating once more the need for dedicated analysis.

360

#### 4. THERMAL EFFECTS

361 After providing a description of the time-averaged flow features developing inside the rotating 362 channel, a further discussion can be addressed concerning the combined effects of rotation and buoyancy on the thermal performances of the device. This is possible by exploiting the available data 363 364 from [14]. Figure 15 reports the thermal data from [14] for a set of flow conditions that are comparable 365 to those considered in the present work, including also the values at Ro=0. The plots show the 366 distributions of the Nusselt number enhancement, Nu/Nu<sub>0</sub> and Nu/Nu<sub>s</sub>, along the channel length and 367 for variable flow conditions. The Nu is the result of wall temperature measurements performed in the experiments while Nu<sub>0</sub> is the one obtained from the Dittus-Boelter correlation and Nu<sub>s</sub> refers to the 368 static channel case. In Fig. 15, L1/T1 refer to the half portion of the leading/trailing sides close to the 369 370 bottom apex, while L2/T2 pertain to the upper half of the leading/trailing sides close to the upper apexes. The following analysis will be based only on the 3D data on plane yz, since plane xy' is 371 372 located just in between of the L1/L2 or T1/T2 zones defined in [14].

373 At first, Figs. 15(a, b) show that rotation sets a substantial increment of Nusselt number in zone T2. 374 The regions adjacent to the lower apex, L1 and T1, experience also stronger heat transfer although limited with respect to the upper portion of the TS (i.e. T2). Conversely, zone L2 shows no increment 375 or even less Nusselt values with respect to the stationary case. The present aerodynamic data can help 376 to understand this behaviour. The plots in Figs. 12 and 13 show how the L2 region is permanently 377 378 swept by the low momentum Coriolis vortex cell, while the adjacent velocity peak favours heat 379 transfer in region T2. These flow features determine a velocity deficit near the L2 region, hence a 380 heat transfer deficit with respect the whole channel section and a reduction with respect to the static 381 case.

This flow behaviour, as previously commented in section 3.3, becomes even stronger if the Ro increases, and consistently does the enhancement factor from [14] (compare data in Fig. 15(a)-higher 384 Ro, with those in Fig. 15(b)-lower Ro).

Concerning the Bo effects, [14] shows an overall positive effect, with the highest increment of heat transfer again located in region T2, Fig. 15(c). Indeed, as demonstrated by the present data, the effect of buoyancy is an overall augmentation of the turbulent agitation of the flow (Fig. 14), in particular on the TS. On the contrary, in region L2 this effect is much less pronounced and the turbulent quantities remains practically unchanged at varying the Bo, consistently with the existence of the low momentum Coriolis vortex cell above commented.

391

#### 392 CONCLUSIONS

Thanks to the present study, important conclusions are drawn about the buoyancy effects on the flowfield inside a ribbed triangular channel. The main findings are summarized in the following.

• Buoyancy forces have only marginal effects that result in increased velocity peaks and in an overall enhancement of the flow agitation level. No remarkable modifications are found on the rotation induced secondary flows and on the separation downstream the ribs. This behaviour differs from the results in [10] for a rectangular channel, where buoyancy lead to a dramatic increase of the separation length behind the ribs at the LS.

• On the duct cross section, rotation induced secondary flows turned out to be strongly asymmetric and characterized by a single main vortex. Consequently, the present inter-rib flow is characterized by a condition by far different from the one realized in the region of pairing of two symmetric vortices that has been investigated by [10] in a rectangular ribbed duct. It is in the authors' opinion that this evidence is at the basis of the differences here commented.

The present analysis once more confirms the complexity of the flow topology inside this kind of devices, evidence that was already discussed for the smooth case [5]. Therefore, a precise explanation of the aero-thermal behaviour cannot be based on a simplified conjectured flow model but there is the need for a deeper understanding. Indeed, the present aerodynamic data allow a precise justification for the heat transfer performances in rotation that are available in the literature on the same geometry. 410

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## Figure 4







### Figure 6







Figure 8



#### Figure 9



Figure 10











## Figure 13



Figure 14







Figure 1: Illustration of forces and main flows features found in a rotating duct

Figure 2: Experimental facility, nomenclature and positions of the PIV measurement planes (highlighted in red)

Figure 3: Measurement set-up used for 2D-PIV on plane xy' (a) and Stereo-PIV on plane yz

Figure 4: Optical setup sketch and parallax error description

Figure 5: Parallax error compensation for 2D xy' measurements: examples of calibration by means of dedicated target displacement (a, b) and resulting optical axis position (red dashed lines) and displacement correction fields (c, d)

Figure 6: Comparison of 2D and Stereo-PIV data extracted on the intersection line of xy' and yz planes of U and V' profiles from 2D-PIV on xy' and S-PIV on yz

Figure 7: Stream tracers and contour plot of the in-plane turbulence intensity in plane xy' for Ro=0.

Figure 8: Stream tracers in plane xy' for all the rotating cases.

Figure 9: Contour plots of the turbulence intensity in plane xy' for all the rotating cases.

Figure 10:  $\langle uv' \rangle$ Reynolds stresses component extracted at y'/h=0.5 from plane xy', (a) static vs Ro = 0.2 and (b) static vs Ro = 0.6 test cases.

Figure 11: Stream tracers (a, b) and contour plots of non-dimensional streamwise velocity component  $U/U_b$  (c, d) in plane yz for the smooth channel configuration (Pascotto et al., 2014).

Figure 12: Stream tracers in plane yz for all the rotating cases.

Figure 13: Contour plots of the non-dimensional streamwise velocity component  $U/U_b$  in plane yz for all rotating cases.

Figure 14: Tu levels in plane yz for Ro=0.6

Figure 15: Nusselt number ratio distribution in the rotating channel (a - Re = 10000, 400 rpm, b - Re = 30000, 400 rpm) and effect of buoyancy parameter on Nusselt number ratios at x/Dh =4.11 (c) [12]