

The Application of Fluidic Sealing in Shrouded Gas Turbine Blades

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32 **ABSTRACT**

33 This paper presents a study conducted on a new gas turbine, designed to limit
34 leakage in the labyrinth seal. The slots in the fin are used to generate a bypass flow, which
35 obstructs the flow in the gap above the fin. The method was tested numerically and
36 experimentally beforehand using a simplified model without rotation or blade passages.
37 In this paper, the validation of the method using a model of a turbine stage is shown.
38 RANS simulations using two turbulence models – Spalart-Allmaras (SA) and k- ω EARSM
39 were conducted. Comparisons of leakage flow and stage efficiency for reference and
40 fluidic sealing configurations are presented. Fluidic sealing configuration is effective and
41 reduces the leakage flow by 13-18.5% (depending on the turbulence model). The analysis
42 of the flow structure in the seal region revealed, that the use of fluidic sealing resulted in
43 significant circumferential flow anisotropy.

44 Keywords: Labyrinth Seal, Flow Control, Leakage reduction

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46 **1. INTRODUCTION**

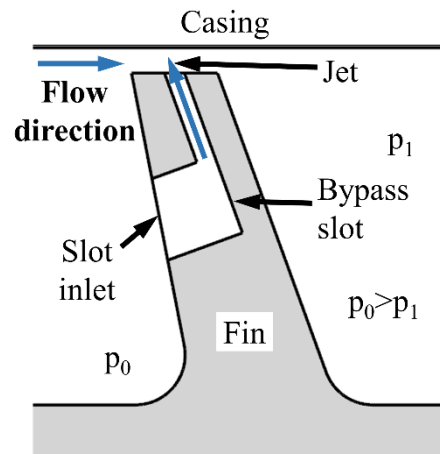
47 Pollution from air travel can be reduced by increasing engine efficiency, which also
48 improves operation profitability. There are many aspects of engine operation that can be
49 investigated in order to reduce loss. However, this study is mainly focused on losses
50 resulting from leakage, which contribute up to 25% of stage losses [1]. Seal leakage
51 impacts three aspects of turbine performance. Firstly, it reduces working potential, as
52 some of the air that could work in the blade passage passes through the seal. Secondly,

53 the flow's kinetic energy is dissipated as mixing occurs in the seal. Lastly, the fluid passing
54 through the seal has a different velocity to the fluid in the main channel, thus mixing
55 occurs, leading to decreased performance in further turbine stages [2]. In the shrouded
56 blades, which are investigated in this work, labyrinth seals are used to limit the leakage
57 [3]. Labyrinth seals consist of several fins, which engage in a series of contractions that
58 obstruct the flow. The shape of those fins is optimized to reduce the leakage [4].
59 Introducing flow control in the seal region can lead to a further reduction, as shown in [5-
60 8]. Considering the widespread use of the gas turbines in aircraft, even a modest
61 reduction in leakage would have a considerable impact globally.

62 Fluidic sealing in turbine labyrinth seals was first presented in the 1950s patent by
63 Auyer [9], which used pressurized air from a compressor and injected it into the tip gap
64 of an unshrouded turbine. Smile and Paulson [10] implemented the same concept with
65 high pressure air introduced in the cavern between the fins' labyrinth seal of a shrouded
66 blade. Hilfer [11] optimized the configuration, which resulted in a 28% leakage reduction.
67 Placing the fluidic seal in the gap above the fin of the labyrinth seal was first proposed by
68 Rushton [12].

69 Subsequent research shows passive configurations of the fluidic sealing, where no
70 additional air supply is needed. In this case, the source of pressurized air is usually
71 upstream of the seal [7], where the pressure is greater than in the gap. Ghaffari [13,14]
72 presented a similar idea, but used the elevated pressure from the stagnation point at the
73 leading edge of the turbine blade.

74 The solution presented in this paper combines two important features of the
 75 solutions presented in the literature review, which, to the best of the authors' knowledge,
 76 has not been done before. Namely, the bypass slot is entirely incorporated within the fin
 77 of the seal while at the same time being passive. The modus operandi is shown in Fig. 1.
 78 In the labyrinth seal, the pressure upstream of the fin is higher than downstream of the
 79 fin. Therefore, the pressure is higher at the slot inlet than the slot outlet, which drives the
 80 flow through the slot. The proposed method is passive, which is an advantage and may
 81 lead to an overall improvement in performance in the gas turbine stage.



82
 83 Fig. 1 Operation principle of the fluidic seal [15]

84 The experiment was conducted on a simplified, static (non-rotating), linear
 85 labyrinth seal configuration of a low-pressure turbine stage [5,6,15], where various
 86 configurations of the seal were placed in a wind tunnel. The pressure ratio between the
 87 section's inlet and outlet was set using valves, after which the stagnation parameters,
 88 mass flow at the inlet and pressure distribution in the seal were measured. A pressure
 89 ratio ranging from 1.05 to 2.05 was tested to verify the design in a wide range of

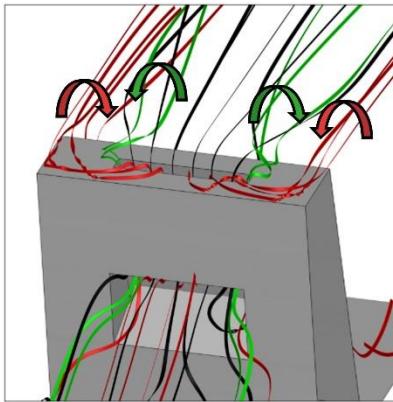
90 conditions. One of the selected conditions corresponds to the pressure ratio in the design
91 point of the labyrinth seal in a turbine. However, due to experimental restrictions, the
92 ambient temperature was used instead of the elevated one. Additionally, the test section
93 allowed for the investigation of configurations with different gap heights, which reflects
94 the changing gap height present during turbine operation. However, in this paper only
95 two gap heights are presented: $h=0.85s$ and $1.25s$, where s is the thickness of the
96 reference fin.

97 The experimental results, comparing the reference case and the case with fluidic
98 sealing [5,6] show the leakage reduction of up to 16% was achieved in the latter
99 configuration.

100 As the purpose of this paper is to present turbine stage simulations, the
101 experimental data is shown only as a validation tool for the numerical model, thus not
102 many details are presented. A more inquisitive reader should refer to [5,6,15], where
103 more information, including the equipment used and measurement errors are presented.

104 The numerical simulations were carried out using the $k-\omega$ EARSM [16] and Spalart-
105 Allmaras [17] turbulence models. Both models showed satisfactory qualitative agreement
106 with the experimental data, with small quantitative differences [5,6]. The efficiency of the
107 fluidic sealing in reducing the leakage flow was proven in simulations using both
108 turbulence models. The CFD approach also showed that fluidic sealing may reduce the
109 leakage flow by up to 22%.

110 Results from the RANS simulations of simplified configuration (described in detail in
111 section 2) served as a source for a brief description of operating principles of the fluidic
112 sealing presented below. The air at the slot outlet has significant velocity, perpendicular
113 to the main flow, which constitutes an obstacle for the main flow. In addition, the
114 difference in the magnitude and direction of the velocity creates additional vortex
115 structures (Fig. 2), which are described in sections 4 and 6. Additionally, since the slot
116 does not cover the entire fin in the circumferential direction, the main flow is impacted
117 by the flow exiting the slot differently at various sections of the gap. This introduces an
118 additional non-uniformity of the flow in the gap, which then propagates into the cavern
119 between the fins. The non-uniformity causes additional mixing in this zone, which leads
120 to a higher dissipation of kinetic energy in the flow. The comparison of the flow structure
121 with and without the fluidic sealing is analyzed in detail in [6].



122
123 Fig. 2 The vortex structure generated by the fluidic seal, direction of vortex rotation
124 marked with arrows. Example representation based on data obtained in RANS
125 simulations [6].
126

127 **2. NUMERICAL MODEL VALIDATION**

128 Before fluidic sealing was implemented in the simulations of the turbine stage
 129 model, an extensive study was conducted on the simplified model, which included
 130 experimental campaign. The comparison of experimental and numerical results was used
 131 to validate the numerical model. The definition of the configuration is presented in this
 132 section, along the validation results.

133 Fine/Turbo Numeca code was used to simulate the flow through the labyrinth seal.
 134 Steady RANS calculations were conducted, with the $k-\omega$ EARSM and Spalart-Allmaras
 135 turbulence models. Perfect gas was assumed with viscosity calculated according to
 136 Sutherland's law. The code uses the second order central difference spatial discretization
 137 scheme with artificial dissipation, as well as the explicit Runge-Kutta numerical scheme.
 138 Additionally, the Full Approximation Storage multigrid strategy, with coarse grid
 139 initialization was utilized. Numerical settings are presented in Tab. 1.

140 Tab. 1 Numerical settings for performed simulations.

Parameter	Setting
Turbulence model	S-A, EARSM
Spatial discretization	2 nd order central difference with artificial dissipation
Numerical scheme	Explicit Runge-Kutta
Multigrid strategy	Full Approximation Storage (3 levels)
Medium	Perfect gas
Viscosity	According to Sutherland law

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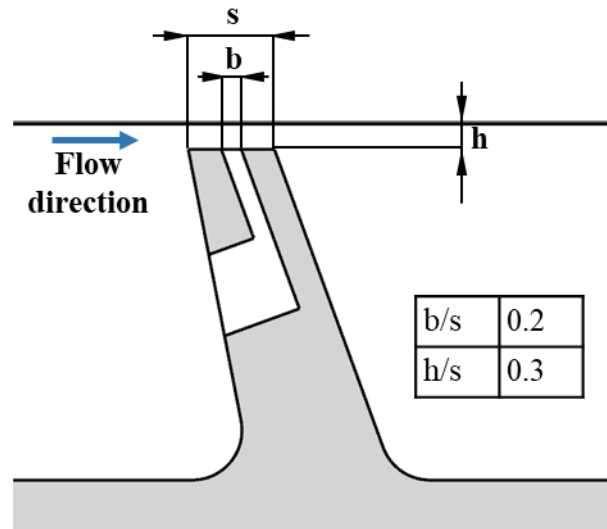
142 The boundary conditions were set according to the measurement set up (described
 143 in section 1). The stagnation pressure and temperature were set at the inlet and the static



144 pressure at the outlet. Walls were set as adiabatic. The turbulence at the inlet was not
145 measured in the experiment, so it was assumed. The turbulent viscosity ratio was set to
146 5 and the turbulence intensity to 5%. The configuration has significant contraction (10:1)
147 between the inlet and the channel leading to the labyrinth seal (which is the focus of the
148 study), followed by yet another contraction (about 10:1, depending on gap height)
149 between the channel and the gap above the fin. Thus the turbulence at the inlet does not
150 impact the flow in the seal in meaningful manner.

151 A block-structured hexahedral mesh was used, refined close to the wall to keep y^+
152 at less than 2 around the seal. The selection of the final grid was preceded by a grid
153 convergence study, so that the results are grid-independent [6]. The final grid size is equal
154 to 5.8 million cells.

155 The focus of the research presented in this paper is the fluidic sealing, which is
156 created using the slots drilled in the labyrinth seal fins. The crucial geometrical features
157 of the slot such as its outlet dimensions are the same for simplified configuration and
158 stage configuration (shown in sections 4-6). The shape of the slot is a compromise
159 between its effectiveness and the manufacturing feasibility. However, in the future,
160 additive manufacturing technology will allow for more complex and optimal slot shapes.
161 The dimensions of the slot, including the inlet and outlet dimensions as well as the slot
162 inclination angle were determined using a parametric study, presented in [8], while the
163 relative dimensions of the slot are shown in Fig. 3.



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Fig. 3. Geometry of the fluidic sealing slot.

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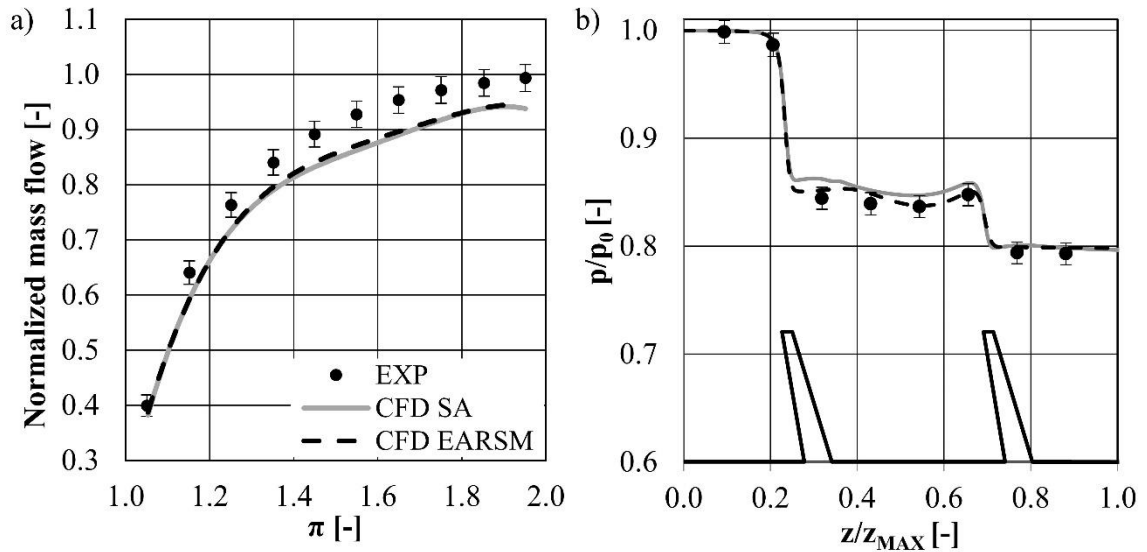
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The labyrinth seal configuration was modified to allow for the design of a relatively simple test section, without sacrificing the key features of the seal flow. The simplifications neglect some secondary geometrical features, reducing the model domain to the seal only (without the blade channel) as well as disregarding rotational effects. The analysis of the impact of those simplifications is described extensively in [5]. Overall, it can be concluded, that while some of the simplifications have a non-negligible impact on the level of leakage flow, the main features of the labyrinth seal flow are conserved in the simplified model. Thus, the simplified model was deemed sufficient for proving the concept and assessing the effectiveness of proposed modifications. Nevertheless, testing of the fluidic sealing in the actual turbine configuration before it can be implemented is essential.

Figure 4 presents a comparison of the mass flow from the numerical model with the measurements in the test section, normalized by the maximum value obtained in the

179 experiment. The mass flow for both turbulence models is very similar and is
180 underestimated by about 8% in the entire range. However, the trend of mass flow change
181 with rising pressure ratio is the same for experimental and numerical results. At the same
182 time, the pressure distribution at the platform (upper wall of the model) calculated using
183 both turbulence models, concurs well with the measured values. In previous studies [5,6]
184 it was established, that the leakage flow is sensitive to the turbulence model used. The
185 leakage flow highly depends on the flow area in the gap, which in turn is impacted by
186 separation bubble height. After exiting the gap the flow enters the cavern, where it is also
187 separated. It is well known, that the accuracy of the turbulence models in separated flow
188 is limited. In this case, the proposed model is considered accurate for the purpose of the
189 study. The improvement of accuracy may be obtained with LES simulations, which are
190 planned in the future.

191



192

193 Fig. 4. Comparison of the quantities obtained in the experiment [5] and the numerical
 194 model with two turbulence models (Spalart-Allmaras and $k-\omega$ EARSM).

195 Gap height = 0.85s, pressure ratio = 1.25. a) Mass flow through the seal normalized with
 196 the maximum value in the experiment. b) Static pressure at the casing normalized by
 197 inlet total pressure.

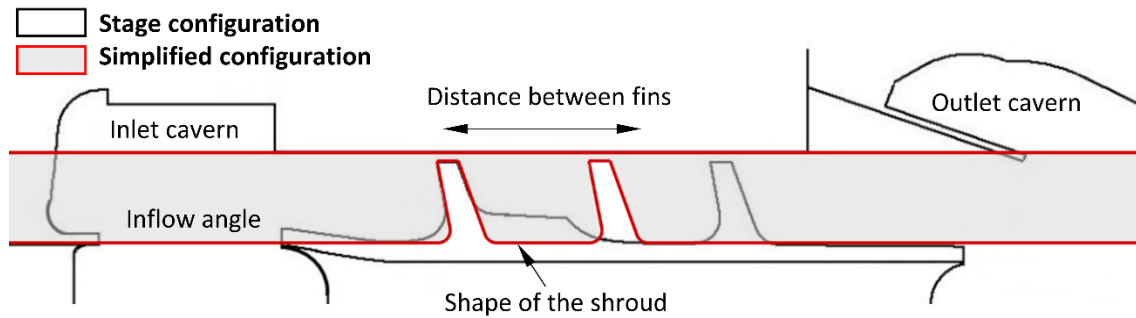
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199 3. TURBINE STAGE MODEL DESCRIPTION

200 Turbine stage simulations used the same numerical settings as the simplified
 201 configurations simulations, while the geometry and boundary conditions were modified.

202 The details of the seal configuration are protected, but key information can be
 203 shared. Similarly to the configuration used in the experimental step, the seal in the stage
 204 configuration has two fins, the clearance used was 0.38 of the fin thickness. However,
 205 there are some geometrical differences between the simplified (experimental) and the
 206 stage configuration, due to the labyrinth seal optimization conducted in the period

207 between the experiment and the stage simulations [18]. Figure 5 presents comparison of
 208 two configurations – simplified and stage. Apart from geometry differences, other
 209 differences between the models that were mentioned in section 2 and discussed in detail
 210 in [5] are listed in the figure.



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212 Fig. 5. Schematic comparison of stage and simplified configurations. Differences

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between configurations are listed in the figure.

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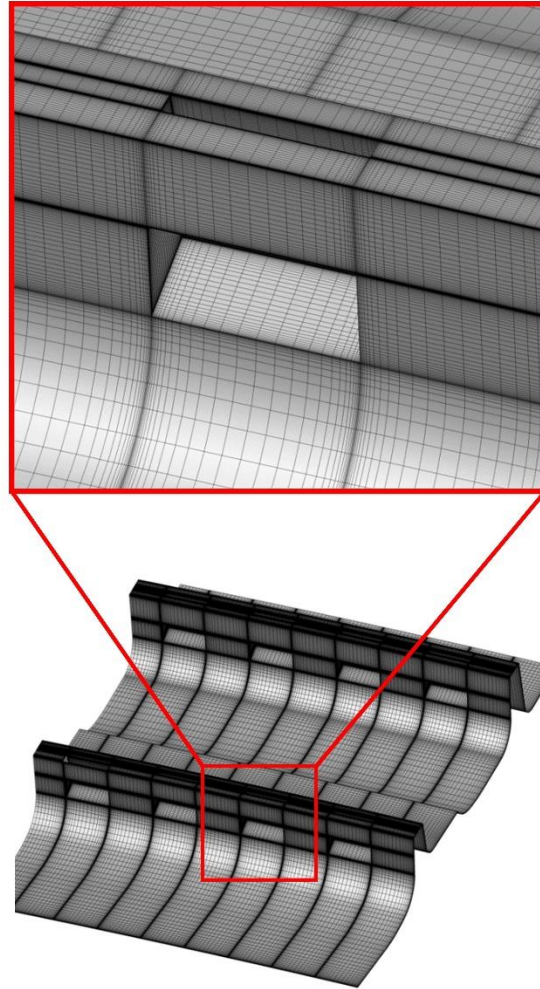
215 Even though the stage configuration is much more faithful representation of a real
 216 turbine than the simplified configuration, there is one major difference between it and
 217 the real engine – namely the honeycomb at the casing was neglected. The experimental
 218 studies on the simplified test section show that the currently investigated fluidic sealing
 219 is not effective when a honeycomb is used. This issue is still being researched, as the
 220 honeycomb adds complexity to the case. Some of the concepts that can make the
 221 configuration with fluidic sealing and the honeycomb combination effective are discussed
 222 in [15]. It is important to note, that the fluidic sealing is investigated as a means to reduce
 223 the leakage flow in a wide range of applications, where the clearance is small and
 unavoidable, not only in LP turbines, where honeycombs are prevalent.

224 The stage used in simulations is defined based on the last stage of a low-pressure
225 turbine of a GEnx engine. As with the seal geometry, the details cannot be shown, but for
226 reference approximate dimensions and features are specified below. The stator consists
227 of 168 blades, while the rotor of 114 blades. The shroud radius is about 1 m, with blades
228 length of about 300 mm. The gap between the casing and the labyrinth seal roughly 1 mm.
229 The rotation rate of the rotor is 2100 RPM. The ratio of total pressure ratio at stage inlet
230 and static pressure at the outlet is equal to 1.6.

231 Two models were created with two versions of the labyrinth seal mounted on the
232 rotor blade. In the reference configuration a standard, unmodified labyrinth seal was
233 installed, while the seal of the second model included fluidic sealing slots. Apart from the
234 slots, both configurations were identical.

235 The grid for the blades channel was generated by the Autogrid/Numeca code with
236 O4H topology, which provided good quality [15,19]. The grid of the labyrinth seal (Fig. 6)
237 was created in IGG/Numeca and added to the channel grid. The grid of the labyrinth seal
238 was split into two parts, one connected to the stator grid and the one to the rotor grid
239 (see Fig. 7). The periodicity of the seal grid matches the periodicity of the blade passage.
240 Both domains - the seal and the blade passage, were connected with non-matching
241 interfaces. The non-matching connection is created using Alternating Digital Trees (ADT)
242 algorithm [20]. One side of the connection is triangulated and projected on the other side.





243

244

Fig. 6. Grid of the labyrinth seal.

245 The grid resolving the flow through the blades consists of two parts – rotor part and

246 stator part, connected with rotor-stator interface. Table 2 shows the grid resolution in

247 selected directions, such as radial, circumferential, pressure side (PS) and suction side (SS)

248 resolution for the blades, as well as resolution in all three directions for the seal and the

249 gap above the fin.

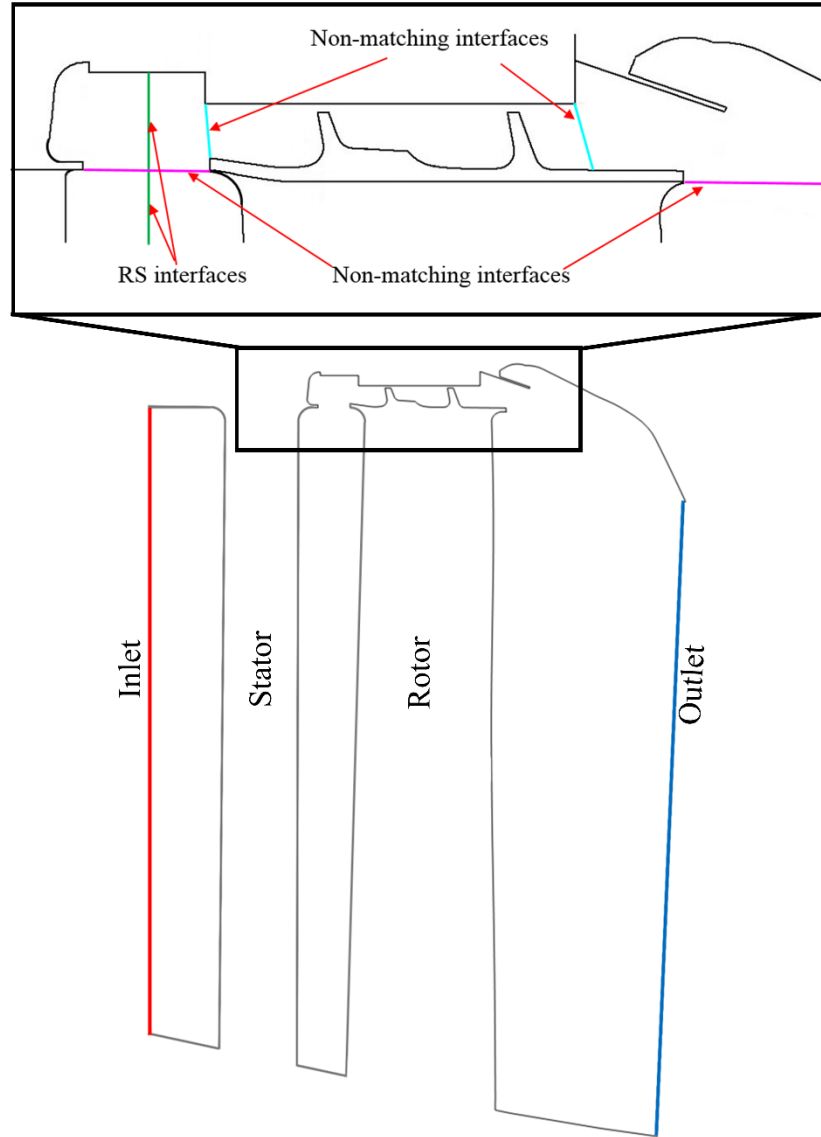
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251 Tab. 2 Resolution of the computational grid in selected directions

	Circumferential (X)	Radial (Y)	Axial (Z)	
Gap	364	36	112	
Seal	364	128	412	
Stator	64	108	112 (PS)	232 (SS)
Rotor	125	108	116 (PS)	300 (SS)

252

253 Appropriate grid resolution on the side walls of the slot leads to a significant number
 254 of elements in the circumferential direction. This causes the grid to be large. Introducing
 255 a non-matching interface (Fig. 7) allows for the reduction of circumferential resolution in
 256 the inlet and outlet caverns of the seal, where refinement is not required. As a result, the
 257 mesh size can be significantly reduced. Still, the final grid consists of 21 million cells, out
 258 of which about 16.5 million are located in the labyrinth seal zone.



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Fig. 7. A numerical model of a turbine stage with the labyrinth seal, with marked boundary conditions and interfaces used.

263

264 The boundary conditions were a result of a 2D, throughflow simulation of the entire
265 engine. The distribution along the turbine radius of total pressure p_1 (average ~ 60 kPa),
266 temperature T_1 (average ~ 700 K) and flow direction was specified at the inlet, upstream
267 of the stator. Turbulence quantities were not known, therefore they were assumed: the
268 turbulent viscosity ratio was at 10 and turbulence intensity at 5%. However, the
269 turbulence quantities at the inlet should not impact the flow in the labyrinth seal, since
270 before reaching it, the flow passes through the turbine stator. At the outlet, a static
271 pressure level of 38 kPa was imposed at the radius of the hub with the hub-to-shroud
272 pressure profile calculated according to the radial equilibrium conditions [21]. The speed
273 of rotation is 2100 RPM.

274 As for the simplified seal configuration the Numeca/FineTurbo code, which is widely
275 used for turbomachinery simulations, was applied. Simulations were carried out for the
276 steady flow and the full non-matching mixing plane (with the conservation of fluxes) for
277 stator/rotor interface. Additionally, the field variables on one side of the interface were
278 circumferentially averaged and applied to the other side.

279

280 4. TURBINE STAGE MODEL COMPARED TO SIMPLIFIED CONFIGURATION

281 As mentioned before, the concept of fluidic sealing in a turbine labyrinth seal was
282 proved effective for simplified, non-rotating configurations. However, due to the
283 simplifications used in the model, additional simulations with fluidic sealing applied to a

284 turbine stage configuration were performed. This made it possible to check the concept
285 in target conditions.

286 Ideally, the numerical model for the turbine stage with the labyrinth seal would be
287 validated against the experimental data. However, even conducting the experiment with
288 just the rotating labyrinth seal (without the blade channel) would be too complex a task
289 and currently beyond available options. Therefore, despite the number of differences
290 between the simplified case and the turbine stage configuration, a qualitative comparison
291 is presented to show that the main features of the labyrinth seal and fluidic sealing flow
292 characteristics are maintained. The differences between cases include (Fig. 5):

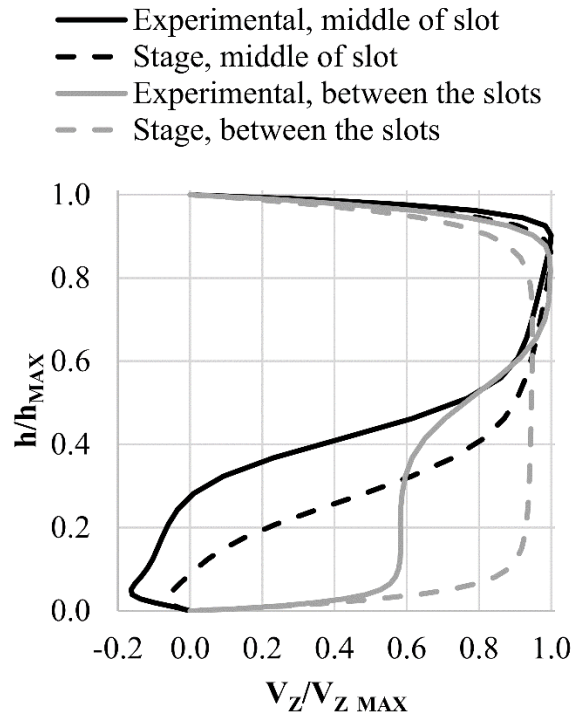
- 293 • Geometry
- 294 • Rotational velocity
- 295 • Boundary conditions

296 Obviously, the simplified case (the static test section) did not include rotational
297 velocity, which is present in stage simulations. While the absolute-to-static pressure ratio
298 differs between the cases (due to the significant dynamic pressure from rotational
299 velocity), the static-to-static pressure ratio in the seal is the same for both cases (the
300 dynamic pressure from axial velocity at the seal inlet is negligible). It is worth noticing,
301 that in the stage simulations, the pressure ratio in the seal is not prescribed, but rather a
302 result of the blade passage solution. The rotational velocity in the seal is significant for
303 the stage simulations, while for the experimental setup it is non-existent. Moreover, in



304 the simplified case, an ambient temperature was set, while in the stage simulations the
305 inlet temperature was derived from the real turbine.

306 Because of these differences, a direct comparison of the mass flows obtained in the
307 experiment and the stage simulations is not valid. Instead, a qualitative comparison of the
308 flow structure created by the jet exiting the fluidic sealing slot, is shown in the following
309 section. The axial velocity profiles (normalized with the maximum value) at two traverses
310 located at the outlet of the gap above the fin are presented in Fig. 8. The first traverse is
311 located in the middle of the slot (black lines) and the second one between the neighboring
312 slots (grey lines). While the flow structure for the stage configuration will be investigated
313 in more detail in section 6, it is worth noting the similarities and differences between the
314 stage and experimental configurations. In both cases, the average velocity is greater in
315 the area between the slots than in the zone where the jet exits the slot, causing a velocity
316 reduction. In the space in the gap close to the slot, the velocity field is very similar in both
317 cases. Downstream of the gap, the flow in the stage configuration is subject to rotational
318 effects (discussed in more detail below), thus differences arise. In both cases, in the area
319 between the slots, the separation in the gap which is present in the reference
320 configurations (not shown here), is reduced when fluidic sealing is used. In the stage
321 configuration, the velocity is almost constant across the gap (excluding the boundary
322 layers), while in the experimental configuration it is variable. This happens, because the
323 complex vortical structure generated by the jet in the simplified configuration is disturbed
324 by significant rotational velocity.

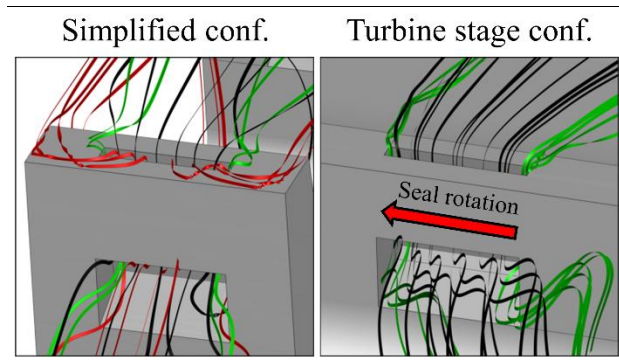


325

326 Fig. 8. Profile of the normalized axial velocity in the first gap for two traverses in the
 327 experimental and stage configurations.

328 Fig. 9 presents the streamlines of relative velocity. The jet exiting the slot (black
 329 streamlines) creates a fluidic seal, that obstructs the main flow. Additionally, the air
 330 exiting the slot in a sideways direction generates stream-wise vortices. They are present
 331 in both of the configurations shown, however in the stage configuration the vortices are
 332 much less developed and more thinly spread in the circumferential direction. In the
 333 simplified configuration one can also notice that the air exiting from the front part of the
 334 slot (red streamlines) enters the separation vortex and later forms a counter-rotating
 335 vortex in the area between the slots. This flow feature is not present in the stage
 336 configurations, as the separation is less pronounced due to the significant rotation

337 velocity of the fin. Moreover, the flow structures generated by mass transport in the
 338 tangential direction are subject to disturbance caused by the rotational velocity of the fin.



339
 340 Fig. 9. Comparison of vortex structure in experimental and stage configurations.

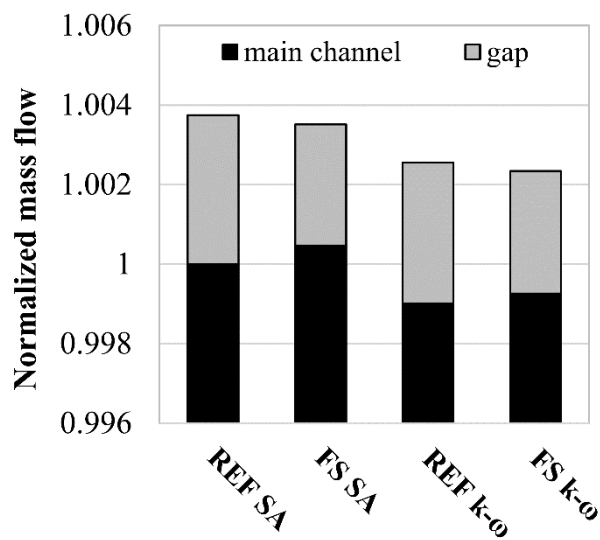
341 Overall, some of the flow features, such as velocity reduction in the area of the slot
 342 and the generation of streamwise vortices at the side edges of the slot, are similar in both
 343 presented configurations. The flow in the area between the slots has a higher average
 344 velocity in both configurations, however the velocity profile in the gap differs.
 345 Additionally, the counter rotating vortices in the area between the slots are not present
 346 in the stage configuration, contrary to the experimental configuration.

347

348 5. TURBINE STAGE MODEL RESULTS

349 The investigation of the impact of fluidic sealing on the flow through a labyrinth seal
 350 in the turbine stage is split into two parts. Firstly, the change in global parameters, such
 351 as leakage flow and stage efficiency is presented. This is followed by a comparison of the
 352 flow structure existing in the reference case and the case with the fluidic seal. This is
 353 helpful in assessing the reasons for the change in the global parameters.

354 The implementation of fluidic sealing in the labyrinth seal of a turbine stage reduces
 355 the leakage flow. The predicted reduction depends on the turbulence model used in the
 356 simulation. For $k-\omega$ EARSM it is equal to 13%, while for Spalart Allmaras it is 18.5%
 357 (Fig. 10). This difference is caused by different separation sizes in the gap above the fin,
 358 predicted by these models, which is discussed in the next paragraph. Interestingly, the
 359 reduction of the leakage flow leads to a slight increase in the blade passage flow.



360

361 Fig. 10. Mass flow for reference and fluidic seal configuration, normalized with the value
 362 of main channel flow for reference Spalart Allmaras case.

363

364 Decreasing leakage leads to an increase in stage isentropic total to total efficiency,
 365 which is defined as:

$$\eta = \frac{1 - \frac{T_2}{T_1}}{1 - \left(\frac{p_2}{p_1}\right)^{\frac{\kappa-1}{\kappa}}} \quad (1)$$

366 Where:

367 p_1, p_2 – total pressure at inlet and outlet

368 T_1, T_2 – total temperature at inlet and outlet

369 κ – heat capacity ratio

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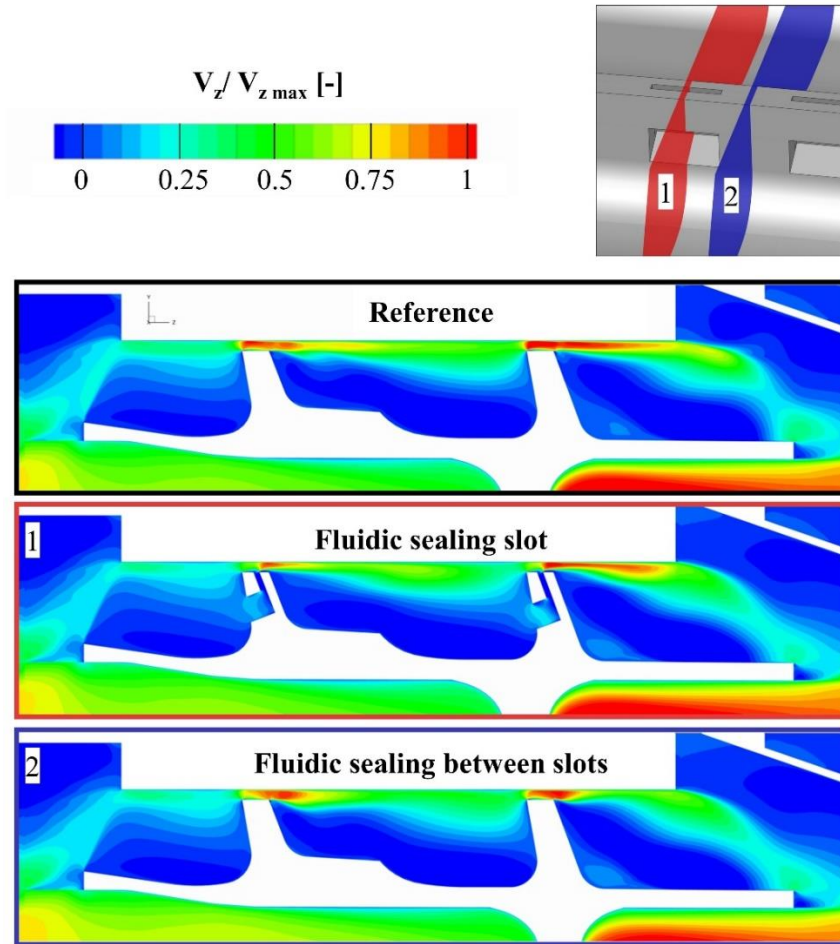
371 Of course, the isentropic efficiency increase is also dependent on the turbulence
372 model and is equal to 0.065% and 0.04% for k- ω EARSM and Spalart-Allmaras respectively.

373 This efficiency increase is not large, however there are several reasons why the concept
374 is worth further investigation. Firstly, in the case of the investigated stage, the blades are
375 very long compared to the gap size, so the leakage flow is only 0.3% of the total mass
376 flow. Therefore, even a significant leakage reduction impacts the stage efficiency only
377 marginally. In the future, the concept will be tested further using blades with different
378 lengths. Additionally, the slots that generate fluidic sealing were not optimized to operate
379 in the rotating framework. In fact, as the aim of the research was to test the viability of
380 the fluidic sealing as a leakage reduction device, the geometry of the slot was not
381 optimized. Last but not least, the presented efficiency is calculated for one stage only,
382 implementing fluidic sealing in more stages would likely bring further gains, since the
383 leakage would be reduced at each stage. Leakage losses are not only connected with loss
384 of working potential, which is could be treated as isolated to the single stage, but also the
385 mixing losses that impact other stages.

386

387 **6. FLOW FIELD ANALYSIS**

388 A flow field analysis was conducted to study the reason behind the leakage
389 reduction when the fluidic sealing was used. A comparison of the flow structure in the
390 reference and the fluidic sealing cases is presented in Figs. 11-12. Figure 11 shows the
391 contour of the axial velocity in the meridional cross-section. Even though there is some
392 circumferential non-uniformity in the reference case, it is minor in comparison to the
393 fluidic sealing case. Therefore, only one cross-section for the reference case is shown,
394 while for the fluidic sealing case, the cross-sections in the middle of the slot as well as in
395 the area between the slots are shown. In Fig. 12, the axial velocity distribution in the
396 constant radius surface is shown. This surface is located in the middle of the tip gap. The
397 location of the fins and slots (in the fluidic sealing case), as well as the rotation direction,
398 is marked in the figure. In the cross section, recirculation zones with very low velocity in
399 the cavities upstream and downstream of the seal can be noticed.

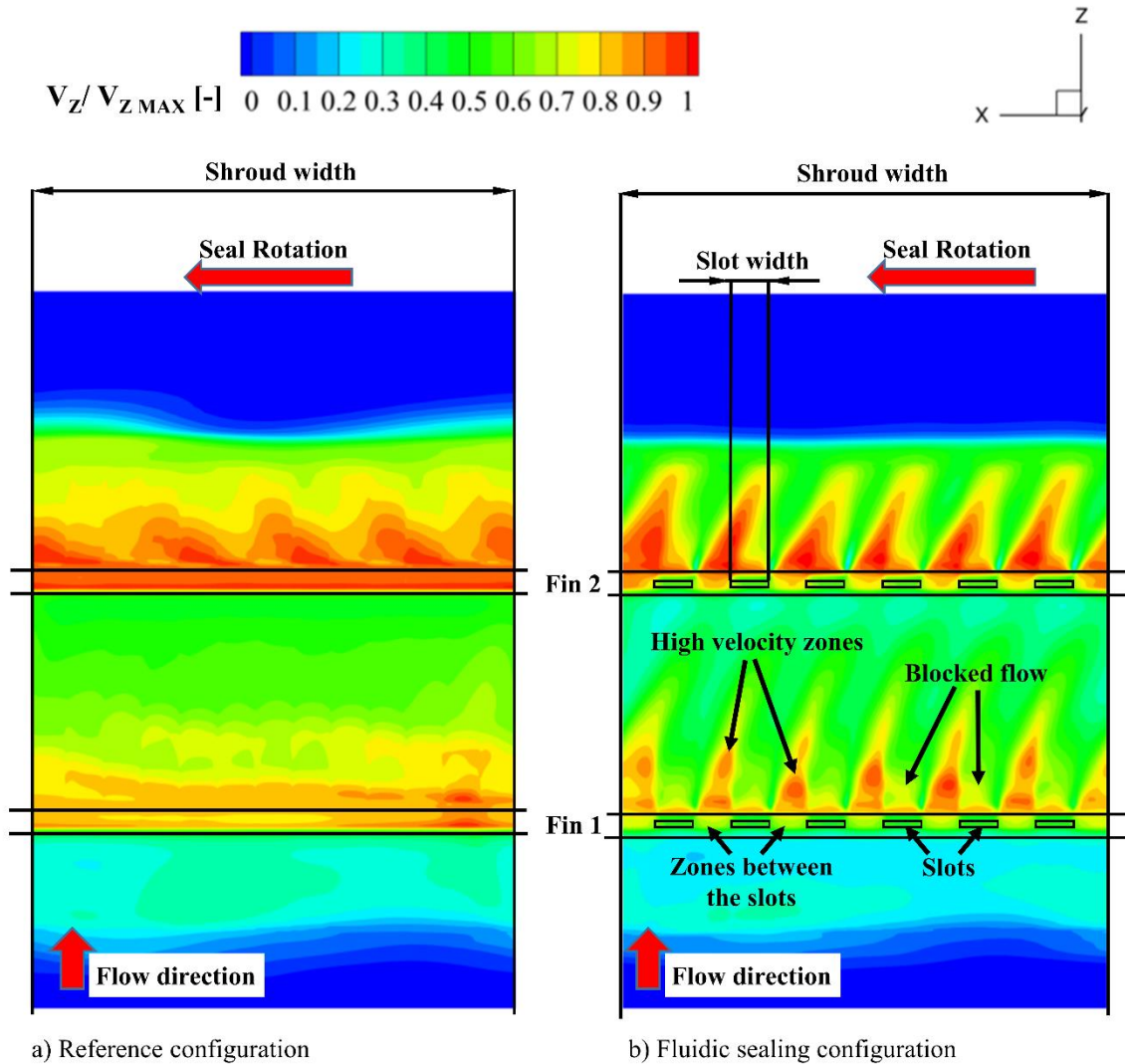


400

401 Fig. 11. Normalized axial velocity for the reference case and the two cross-sections of the
 402 fluidic sealing case.

403 In the reference case, the flow accelerates in the gaps above the fins. This leads to
 404 the creation of the high velocity zone that goes into the area between the fins (which can
 405 be seen in Fig. 11). A similar zone is created above the second fin. In Fig. 12, the non-
 406 uniformity of velocity in the circumferential direction can be noticed. The flow through
 407 the blade passages is not uniform, which impacts the conditions at the inlet and outlet of
 408 the seal. Additionally, the slight impact of the computational grid can be noticed. As
 409 mentioned before, both the reference and fluidic sealing grids are very similar, with the

410 only difference being the presence of slots. In the fluidic sealing case, the grid was created
411 so that the boundary layer inside the slots was properly resolved. This means that the grid
412 refinements which are close to walls propagate in an axial direction which causes visible
413 non-uniformity in the circumferential direction. This effect also exists in the reference
414 grid, since both grids are almost the same. Nevertheless, the non-uniformity of the flow
415 in the reference case, resulting from the grid resolution, is minor and does not
416 significantly affect the results. In fluidic sealing case, the periodic high velocity zones
417 downstream of the second fin have a different periodicity than the grid blocks.



a) Reference configuration

b) Fluidic sealing configuration

419

420 Fig. 12. Normalized axial velocity for the reference and the fluidic sealing case for a
 421 constant radius cross-section.

422 Implementing fluidic sealing significantly impacts the flow in the gap above the fins.
 423 Strong circumferential non-uniformity is present, with two distinct flow patterns. Firstly,
 424 in the zones above the slots, the high velocity is significantly diminished in comparison
 425 with the reference case. The main flow in that region is blocked and pushed upwards by
 426 the jet exiting the slots. Moreover, the very presence of the jet causes the expansion of

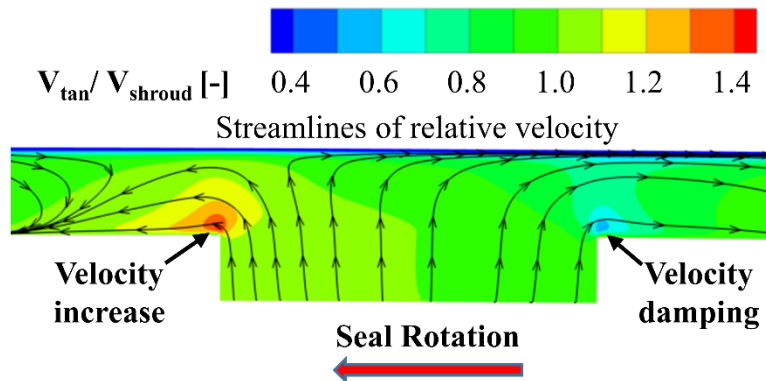
427 the high velocity zone in the cavern to be less prevalent than in the reference case (Fig. 11,
428 cross section 1).

429 On the other hand, a part of the flow blocked by the jets exiting the slots, passes
430 the gap through the zone between the slots. This increases the maximum velocity there
431 and the size of the high velocity zone (Fig. 11, cross section 2). In contrast to the zone
432 above the slots, the flow exiting the gap between the slots expands more in the cavern
433 than in the reference case. One can easily notice in Fig. 12, that the high velocity zones
434 created between the slots, enter the cavern between the fins and bend in the opposite
435 direction to the rotation. This is a result of rotation. The circumferential velocity
436 component in the gap is almost equal to the rotation velocity of the seal, therefore, the
437 shape of the high velocity zones in the gap is not changed. At the same time, the
438 circumferential velocity in the cavern, between the fins and downstream of the second
439 fin is lower than the rotation velocity of the seal. Thus, in the rotating frame of reference,
440 the high velocity zone seemingly bends in the opposite direction to the rotation.

441 In addition, the velocity exiting the seal is slightly more uniform than in the
442 reference case. This is caused by increased mixing due to the high non-uniformity of the
443 flow in the circumferential direction. A more uniform flow at the outlet may be beneficial
444 since mixing occurs between the flow exiting the seal and the blade passage flow.
445 However, this aspect should be further investigated.

446 In the non-rotating framework, the air exits the slots in a direction which is aligned
447 with the main flow, but also with the sides of the slot. When the seal rotates, it has an

448 additional circumferential component. The superposition of those two effects results in
 449 non-symmetric flow in the gap, which can be observed in Fig. 13. On the one side of the
 450 slot, where the direction of the velocity exiting the slot matches that of the
 451 circumferential velocity, there is a local speed-up. On the opposite side of the slot, the
 452 circumferential velocity is reduced by the velocity of the flow exiting the slot. This
 453 promotes additional mixing and may be one of the reasons for more overall flow
 454 uniformity at the exit of the seal.
 455



456
 457 Fig. 13. Contour of circumferential velocity, normalized by the velocity of shroud rotation.
 458 Streamlines (in plane) in rotating frame.

459

460 7. CONCLUSIONS

461 This new concept for fluidic sealing in labyrinth seals, which was previously tested
 462 only in simplified conditions, was implemented in turbine stage simulations. As in the
 463 simplified configurations shown in [5,6], the fluidic sealing proved to be effective in
 464 reducing the leakage flow. The reduction predicted by the simulations differs, depending

465 on the turbulence model used, with 18.5% reduction obtained for Spalart Allmaras and
466 13% for $k-\omega$ EARSM. Evaluation of the stage isentropic efficiency shows improvement by
467 0.065% and 0.04% for $k-\omega$ EARSM and Spalart-Allmaras.

468 The flow structure in the reference and fluidic sealing configurations was compared.
469 The comparison revealed that the jet exiting the slots blocks the flow and leads to
470 significant circumferential flow non-uniformity. This promotes mixing, which in turn
471 increases flow resistance, eventually leading to a decrease in leakage flow.

472

473 **8. FURTHER WORK**

474 Since the fluidic sealing concept has proven to be viable, further steps can be taken
475 in order to assess its full potential, as well as to tackle possible implementation
476 challenges.

477 Firstly, the shape of the fluidic sealing slot used in this study was not optimized with
478 rotating machinery in mind. Therefore, the concept could be more efficient if the slots
479 were a different shape and were inclined in a circumferential direction.

480 The leakage reduction and stage efficiency gain resulting from the use of fluidic
481 sealing should be also assessed for different stages of the turbine and shorter blades. Such
482 a study would show whether the gains achieved in the entire turbine are significant
483 enough to warrant further investigation.

484 Additional analyses, regarding material strength and possible manufacturing
485 techniques should be performed. This would allow for a more precise selection of
486 optimization constraints for the slot. This may also lead to an improvement in efficiency.

487 Last but not least, an experimental campaign using a rotating seal could be
488 conducted to validate the numerical model in a more robust manner. This would
489 definitely illustrate whether or not the concept is viable in rotating seals.

490

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496

497 **11. NOMENCLATURE**

498

b	Size of the slot in axial dimension
h	Gap height
p	Pressure
p_1	total pressure at inlet
p_2	total pressure at outlet
T_1	total temperature at inlet
T_2	total temperature at outlet
s	Thickness of the fin of the seal
y^+	Non dimensional wall distance
V_{tan}	Tangential velocity
V_{shroud}	Velocity of the shroud
V_z	Axial velocity
z	Axial dimension
κ	heat capacity ratio
π	Inlet to outlet pressure ratio
CFD	Computational Fluid Dynamics
FS	Fluidic Sealing
HP	High pressure
$k\text{-}\omega$ EARSM	$k\text{-}\omega$ Explicit Algebraic Reynolds Stress turbulence model
LP	Low pressure
PS	Pressure side
RANS	Reynolds Averaged Navier Stokes
REF	Reference case (without modifications)
RPM	Revolutions per minute

RS	Rotor-Stator
SA	Spalart-Allmaras turbulence model
SS	Suction side

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558 **13. Table caption list**

559 Tab. 1 Numerical settings for performed simulations.

560 Tab. 2 Resolution of the computational grid in selected directions

561

562 **14. Figure caption list**

563 Fig. 1 Operation principle of the fluidic seal [15]

564 Fig. 2 The vortex structure generated by the fluidic seal, direction of vortex rotation
565 marked with arrows. Example representation based on data obtained in RANS simulations
566 [6].

567 Fig. 3. Geometry of the fluidic sealing slot.

568 Fig. 4. Comparison of the quantities obtained in the experiment [5] and the
569 numerical model with two turbulence models (Spalart-Allmaras and $k-\omega$ EARSM). Gap
570 height = 0.85s, pressure ratio = 1.25. a) Mass flow through the seal normalized with the
571 maximum value in the experiment. b) Static pressure at the casing normalized by inlet
572 total pressure.

573 Fig. 5. Schematic comparison of stage and simplified configurations. Differences
574 between configurations are listed in the figure.

575 Fig. 6. Grid of the labyrinth seal.

576 Fig. 7. A numerical model of a turbine stage with the labyrinth seal, with marked
577 boundary conditions and interfaces used.

578 Fig. 8. Profile of the normalized axial velocity in the first gap for two traverses in the
579 experimental and stage configurations.

580 Fig. 9. Comparison of vortex structure in experimental and stage configurations.

581 Fig. 10. Mass flow for reference and fluidic seal configuration, normalized with the
582 value of main channel flow for reference Spalart Allmaras case.

583 Fig. 11. Normalized axial velocity for the reference case and the two cross-sections
584 of the fluidic sealing case.

585 Fig. 12. Normalized axial velocity for the reference and the fluidic sealing case for a
586 constant radius cross-section.

587 Fig. 13. Contour of circumferential velocity, normalized by the velocity of shroud
588 rotation. Streamlines (in plane) in rotating frame.

589