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Wall heating effects on aeroacoustic fields radiated by rigid bodies at different flow regimes

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1 Abstract

This study presents several direct numerical computations concerning wall heating 2 effects on aeroacoustic fields derived from flow-rigid body interaction. Both laminar 3 and turbulent flow configurations involving isolated and multibody arrangements 4 have been addressed. Some insights into the physical mechanisms concerning these 5 problems are addressed and discussed. In particular, we observed that the aeroacous-6 tic fields produced by laminar flows can be more easily controlled and practically 7 suppressed in terms of acoustic emission by wall heating. By contrast, in turbu-8 lent flows, the effectiveness of the analysed technique is more limited. Indeed, wall 9 heating produces a slight increase in overall emissions. However, at the tonal peak 10 frequency derived from adiabatic configuration, the acoustic contribution is consid-11 erably reduced. 12

13 Key words:

14 Wall heating, OpenFOAM, Aeroacoustic fields, Direct Numerical Simulation

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15 1 Introduction

Self-noise produced by aerodynamic bodies is an important issue in many applications ranging from wind turbines and helicopter rotors to fan blades and
airframes, [1]. Similar problems can also be found in broad sense for bluff bodies with reference to built environment noise pollution, [2].

The flow fields derived from aforementioned application areas share unsteadiness in the boundary layer and/or in the wake region which are responsible for the creation of acoustic perturbations, [1, 3]. In this context, it is easy to understand that an aeroacoustic field can be controlled by altering the wake geometry as well as boundary layer (if present) detachment features.

Passive control techniques have been widely studied and reported. These ap-25 proaches use additional objects near the reference body to mitigate acoustic 26 disturbance, [4, 5]. Active control techniques have also received significant at-27 tention by several authors, mainly in relation to flow-field modification. Specif-28 ically, we can find some studies treating flow-field control through the impo-29 sition of line, transverse, or rotary oscillations, [6, 7]. Other techniques have 30 even considered synthetic jets, [8]. Nevertheless, the effect of forced transverse 31 oscillations on the sound generation and propagation from a circular cylinder 32 immersed in uniform flow was only addressed very recently, [9, 10]. 33

It is worth emphasising that wake geometry can also be controlled by heating 34 the body's surface. This phenomenon was experimentally observed in a circu-35 lar cylinder by Lecordier et al. [11, 12]. However, the acoustic effects related to 36 wall heating were investigated preliminarily by D'Alessandro et al. [13], who 37 extended their analyses to non-circular geometry sections as well. It was re-38 vealed that the wall temperature increment can reduce the aerodynamic force 39 pulsations generated by the Karman vortex street, which is shed over bluff 40 bodies in laminar flows. In this context, any reduction in lift pulsations is 41 crucial because it leads to aeroacoustic perturbations decay. With the present 42 study, we want to advance the state-of-art concerning the wall heating conse-43

quences on the aeroacoustic sound generation and propagation. In particular, the role of dynamic viscosity and flow/sound problems involving more than one body are investigated. In addition, we investigated the thermal effect on the aeroacoustic sound emitted using an NACA 0012 airfoil under turbulent flow. This is a particular aspect of the novelty of this study. In fact, at the time of writing, the impact of wall heating on aeroacoustic fields had not yet been studied in a similar flow regime.

⁵¹ Direct computation of the flow and acoustic fields was adopted as an analysis ⁵² tool. In particular, an open-source solver called **caafoam**, developed by the ⁵³ authors themselves, [13], was adopted.

The remainder of this paper is organised as follows, the governing equations are presented in Section 2, while the adopted numerical discretisation techniques are briefly discussed in Section 3. Section 4 is devoted to numerical results. The conclusions are presented in Section 5.

58 2 Flow model

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The flow model equations, based on the conservative variables $\mathbf{u} = (\rho, \rho u_i, \rho E)^T$, are as follows:

$$\frac{\partial \mathbf{u}}{\partial t} + \boldsymbol{\nabla} \cdot \mathbf{F}_{c} \left(\mathbf{u} \right) - \boldsymbol{\nabla} \cdot \mathbf{F}_{v} \left(\mathbf{u}, \boldsymbol{\nabla} \mathbf{u} \right) = \sigma \left(\mathbf{u}_{ref} - \mathbf{u} \right), \tag{1}$$

where $\mathbf{F}_{c}(\mathbf{u})$ and $\mathbf{F}_{v}(\mathbf{u}, \nabla \mathbf{u})$ are convective and viscous fluxes, respectively, given by:

$$\mathbf{F}_{\mathbf{c},\mathbf{j}} = \begin{pmatrix} \rho u_{j} \\ \rho u_{1}u_{j} + p\delta_{1j} \\ \rho u_{2}u_{j} + p\delta_{2j} \\ \rho u_{3}u_{j} + p\delta_{3j} \\ \rho u_{j}H \end{pmatrix}, \quad \mathbf{F}_{\mathbf{v},\mathbf{j}} = \begin{pmatrix} 0 \\ \tau_{1j} \\ \tau_{2j} \\ \tau_{3j} \\ \tau_{ji}u_{i} - q_{j} \end{pmatrix}.$$
(2)

The viscous stress tensor is computed using the constitutive relation for Newtonian fluids, whereas for heat flux vector components, the Fourier postulate
is used

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$$\tau_{ij} = \mu \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3} \frac{\partial u_k}{\partial x_k} \delta_{ij} \right), \quad q_j = -\lambda \frac{\partial T}{\partial x_j}.$$
 (3)

In this study, the dynamic viscosity μ was handled in two different ways. In the first case, a constant value was imposed, whereas the Sutherland law [14] was adopted in the second approach. However, the thermal conductivity was obtained using the Prandtl number.

The pressure was computed by assuming the ideal gas equation of state as
a thermodynamic model. By contrast, the fluid temperature was evaluated
employing the internal energy equation according to:

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$$p = \rho (\gamma - 1) \left(E - \frac{1}{2} u_k u_k \right). \quad c_v T = E - \frac{1}{2} u_k u_k.$$
 (4)

The last term, in the right-hand side of eq. 1 is not physical and represents the expression of the artificial sponge layer [15, 16]; this term avoids spurious wave reflections at the external boundaries, producing damping of the flow variables to a fixed reference solution, \mathbf{u}_{ref} . In our approach the spatially varying coefficient σ was defined as follows:

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$$\sigma = \sigma_0 \left(\frac{L_{sp} - d}{L_{sp}}\right)^n \tag{5}$$

where L_{sp} denotes the layer width, d denotes the minimum distance from the inflow/outflow boundaries, σ_0 is a constant value, and n is an integer parameter controlling the shape of the sponge profile.

The optimal sponge layer design is a complex problem, [17]. The two main parameters to consider in order to avoid sponge failure are $-\eta_{target}$, which defines the sponge strength, and the sponge width. The first one is defined as follows:

$$-\eta_{target} = 20 \frac{2\log_{10} e}{1 - M_{\infty}^2} \int_{L_{sp}} \sigma d\mathbf{x}.$$
 (6)

and it is expressed in dB. By contrast, the sponge width must be set with the
following constraint for its dimensionless expression:

$$0.5 \le \frac{L_{sp} \cdot f}{a_{\infty}} \le 2 \tag{7}$$

where f is the sound disturbance frequency and a_{∞} is the speed of sound, [17]. 94 For all the laminar flow computations presented in this study, we fixed n = 295 in eq. 5, while the dimensionless parameter $(L_{sp} \cdot f) / a_{\infty}$ was 0.5 in order to 96 limit the computational load. It is important to point out that the sponge layer 97 dimensionless width is based on the characteristic frequency derived from the 98 spectral analysis of the lift coefficient time-history obtained using the adiabatic 99 wall condition. The sponge strength is provided for every configuration in their 100 respective sections. Finally, the sponge layer size for the turbulent flow case is 101 described in detail in Sec. 4.4. 102

¹⁰³ 3 Numerical approximation

The flow governing equations were solved using a collocated cell-centred finite volume method, available within the OpenFOAM v2.3.x library [18]. The code's object-oriented structure enables users to implement their own models and solvers in the baseline codes with relatively little effort (for example [19, 20, 21]), and that is why it has been receiving significant interest from the CFD community in recent years.

In particular, our solution strategy relies on a solver, called caafoam, developed by our research group [13], and has already proven to be reliable for direct computations of the aeroacoustic sound.

The convective part of the Eulerian flux was computed by following Piroz-113 zoli's energy-conserving scheme [22], whereas standard central schemes for 114 diffusive contributions were adopted. For time integration, 2N storage explicit 115 Runge–Kutta (ERK) was employed. In particular, we considered a five-stage, 116 fourth-order accurate ERK scheme available in the open literature [23]. In our 117 computational experience, this scheme was appealing showed very good par-118 allel performance and it allowed for the use of a maximum Courant number 119 equal to 1. 120

121 4 Results

In this section we present several low Mach number cases involving uniform velocity inlet boundary condition. The cases of a single circular and two square cylinders placed side by side, as well as in a tandem configuration, were analysed. In all the aforementioned cases, the Mach number of the undisturbed flow was $M_{\infty} = 0.2$, $\gamma = 1.4$, and the Prandtl number Pr was 0.75. The Reynolds number, based on a reference length L_{ref} , is defined in its standard form $Re = \rho_{\infty} u_{\infty} L_{ref}/\mu_{\infty}$. In addition, the flow/noise problem correlated with the NACA 0012 airfoil at a chord-based Reynolds number equal to $5 \cdot 10^4$, $M_{\infty} = 0.4$ and $\alpha = 5^{\circ}$ was investigated to assess the impact of wall heating in turbulent flow regime. Undisturbed flow parameters for NACA problem are discussed in Sec. 4.4. The acoustic field was analysed in terms of the dimensionless fluctuating pressure, defined as

$$p' = \frac{p - \overline{p}}{\rho_{\infty} u_{ref}^2} \tag{8}$$

where \overline{p} denotes average pressure field. For comparison with literature data, $u_{ref} = a_{\infty}$, for laminar flow cases. By contrast, for NACA airfoil we set $u_{ref} = u_{\infty}$. Polar plots containing the root mean square of p' are shown to provide evidence of sound features in the far field. The dilatation rate field $\partial u_j / \partial x_j$ was also used to visualise the acoustic waves.

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Aerodynamic performance were evaluated using dimensionless drag and lift
coefficients given by eq. 9:

$$C_D = \frac{2D'}{\rho u_{\infty}^2 A_{ref}}, \quad C_L = \frac{2L'}{\rho u_{\infty}^2 A_{ref}}.$$
(9)

Standard statistics were used to analyse C_D and C_L behaviours, such as the mean drag coefficient $\overline{C_D}$, mean lift coefficient $\overline{C_L}$, root mean square of the lift coefficient $C_{L,rms}$, root mean square of the drag coefficient $C_{D,rms}$, and the oscillation amplitudes of the force coefficients ($\Delta C_D = (C_{D,max} - C_{D,min})/2$ and $\Delta C_L = (C_{L,max} - C_{L,min})/2$). The Strouhal number is defined as:

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$$\operatorname{St} = \frac{f L_{ref}}{u_{\infty}}.$$
 (10)

All the solutions were obtained on distributed-memory parallel machines and the computations requiring a lower load were run on a Linux cluster with 152 16 Intel Xeon E5-2603v3-based nodes for 192 CPU cores operating at 1.6 153 GHz. The code was built using Intel compilers and an MPI library version 154 developed by Intel. Larger cases were run on the MARCONI-100 system hosted ¹⁵⁵ by CINECA. In the PowerPC-based system, OpenFOAM was built using GNU
 ¹⁵⁶ compilers and IBM Spectrum MPI.

157 4.1 Circular cylinder

The first test case considered in this study was the sound generation by the 158 flow past a circular cylinder at Re = 150 where the cylinder diameter D is the 159 reference length. We used a fully-structured O-type grid with far-field bound-160 aries placed at 150 times D, and the height of the first cell next to the wall 161 y_c was set to $y_c/D = 5 \cdot 10^{-3}$. To discretise the radial direction, we used 750 162 cells, whereas 700 were used in the azimuthal direction. Thus, the total num-163 ber of cells N_c was $5.25 \cdot 10^5$. It is also important to note that this grid was 164 already benchmarked by the authors for this kind of flow/sound field, |13|. 165 The sponge's strength was set at 40 dB. 166

Two different settings were considered to investigate the effects of the thermal boundary conditions at the wall on the acoustic field, fixed wall temperatures ($T_w = 2T_\infty$ and $T_w = 3T_\infty$) and a fixed wall temperature gradient ($\partial T/\partial n|_w = 0.5$, $\partial T/\partial n|_w = 1$ and $\partial T/\partial n|_w = 2$). A baseline configuration involving an adiabatic wall was also computed.

Fig. 1 shows the directivity plot based on p'_{rms} evaluated on a circle having 172 a radius r/D = 75. The effectiveness of the proposed grid and solver was 173 demonstrated by comparing our predictions with the DNS data published by 174 Inoue and Hakateyama, [3]. A key element is the clear dominance of lift fluc-175 tuations which yield a typical dipolar acoustic field. Fig. 2 shows p'_{rms} polar 176 plot on a circle having a radius r = 40 D for different fixed wall temperatures. 177 Note that we selected r = 40 D because we wanted to prevent excessive sound 178 wave decay in the far field in order to estimate wall thermal effects on the 179 acoustic field. The impact of the thermophysical model on dynamic viscosity 180 was also assessed; the data reported in Fig. 2, labelled with $\mu(T)$ are related 181

to the Sutherland viscosity model results, whereas the other data refer to the 182 temperature-independent viscosity. Note that the dimensionless Sutherland 183 constant is $S/T_{\infty} = 0.3855$. It is worth noting that the considered flow/sound 184 problems, which relied on adiabatic wall conditions, experienced very limited 185 magnitude of the temperature field gradient. This effect was the result of the 186 wall pressure gradient distribution. Therefore, it is very clear that thermal 187 dependent viscosity produces negligible effects when adiabatic wall boundary 188 conditions are adopted. For non-adiabatic cases, the effect is the opposite. 189

Tab. 1 and Tab. 2 show that non-adiabatic wall conditions produce contrast-190 ing effects on the $\overline{C_D}$. Conversely, the force coefficient pulsations and St are 191 reduced as a result of the wall heating. It should be noted that similar results 192 agree with those of Lecordier et al. [11, 12], who experimentally observed 193 vortex shedding damping behind a heated circular cylinder. Moreover, vortex 194 shedding damping, related to wall heating was confirmed for both thermo-195 physical models used in this study. This is extremely important to explain 196 why far-field sound abatement occurs at higher wall temperatures. Indeed, 197 the acoustic field derived from the fluid-body interaction is related to the 198 sound sources damping, *i.e.* ΔC_D and ΔC_L . Furthermore, it is evident that 199 the adoption of the Sutherland model amplified sound abatement. In fact, 200 with the temperature-independent viscosity model, we can only account for 201 the modification of the pressure owing to the thermal effect, that is, $p \propto T$. 202 By contrast, when the Sutherland model was enabled, the viscosity increased 203 with temperature; thus, we obtained $\tau_w \propto T$ (where τ_w denotes wall shear 204 stress). Consequently, in the latter case, we increased both the wall shear 205 stresses and the pressure gradient magnitude acting on the wall temperature. 206 When a fixed viscosity was used, only the pressure gradient magnitude was af-207 fected by the temperature field. Owing to the previous reasons we believe that 208 the Sutherland model-based predictions produced more significant dissipation 209 on the flow properties in the wall region, as well as in the near-wall region. 210 Hence, the final effect was the diminishing force oscillations with a consequent 211

212 stronger p'_{rms} average value abatement.

Fig. 3 shows acoustic pressure polar plots obtained using fixed wall temperature gradient on the cylinder surface. Clearly, it can be observed that p'_{rms} exhibits a behaviour similar to that of the fixed temperature case. Nevertheless, the impact of the thermophysical model assumes even greater importance given by the magnitude of the wall temperature gradient.

Therefore, in the following, because of the augmentations reported above, we used the Sutherland model which is more physical than other models. Simultaneously, fixed-gradient boundary conditions for the temperature field were employed because they were simpler to reproduce experimentally (through Joule heating) for the sake of benchmarking.

223 4.1.1 Effect of local wall heating

In order to improve the understanding of the aerodynamic sound active ther-224 mal control we analysed the flow/sound fields performing localised wall heat-225 ing. In other words, we generated three different configurations, namely C1, 226 C2, and C3, and thereafter applied the wall temperature gradient to the up-227 per and lower sides of the cylinder. The aforementioned cases differed from 228 the angular sector extension where the conditions were imposed. In the C1 229 configuration, we assessed two different cases. First, we applied $\partial T/\partial n|_w > 0$ 230 for $0^{\circ} \leq \theta \leq 90^{\circ}$, whereas $\partial T/\partial n|_w = 0$ for $90^{\circ} < \theta \leq 180^{\circ}$. In the latter, we 231 used $\partial T/\partial n|_w = 0$ for $0^\circ \le \theta \le 90^\circ$, whereas $\partial T/\partial n|_w > 0$ for $90^\circ < \theta \le 180^\circ$. 232 Note that θ is clockwise positive and equal to 0° at the cylinder's leading 233 edge (LE), whereas is 180° at the trailing edge (TE). On the bottom side, 234 the boundary conditions for the temperature were symmetric with respect to 235 the upper one. For case C2, we divided both cylinder sides into three equal 236 parts with an angular extension $\Delta \theta = 60^{\circ}$, to obtain three angular sector 237 couples. Successively, we did not apply a null temperature gradient to each of 238 the three couples, and the remaining part of the surface of the cylinder was 239

adiabatic. Finally, for C3, we defined the wall boundary conditions similar to C2 with $\Delta \theta = 30^{\circ}$; hence, we obtained six different cases to study. The results related to the aforementioned configurations, are showed in Fig. 4 – 6. To better estimate the acoustic damping provided by local wall heating, all polar plots also contained p'_{rms} for the fully adiabatic and heated cases, respectively, $\partial T/\partial n|_w = 0$ and $\partial T/\partial n|_w = 1$.

Looking at Fig. 4, it is interesting to note that the application of the heat flux 246 corresponding to $\partial T/\partial n|_w = 1$ only for the cylinder forepart, $0^\circ \leq \theta \leq 90^\circ$, 247 had negligible effects on the sound field. In fact, this configuration produced 248 very similar results to a fully adiabatic condition. By contrast, $\partial T/\partial n|_w = 1$ 249 for $90^{\circ} < \theta \leq 180^{\circ}$ was equivalent to the temperature gradient application on 250 the full surface. Further shrinkage in the thermal flux application produced 251 in the C2 case, provided evidence that almost the entire sound reduction ef-252 fect can be assigned to the $120^{\circ} < \theta \leq 180^{\circ}$ zone, as depicted in Fig. 5. 253 The adoption of an angular step equal to 30° improved our understanding of 254 the discussed phenomenon. Indeed, from Fig. 6, it is clear in the two zones: 255 120° < θ \leq 150° and 150° < θ \leq 180° both contribute to sound emission 256 reduction. However, the latter one has grater weight than the former in the 257 overall context. 258

Therefore, it can be stated that wall heating focused on separated flow region is the key aspect for adequately controlling force oscillations and related sound emission.

262 4.2 Square cylinders arranged side by side

The subject of this subsection is sound generation around two square cylinders placed side by side, as shown in Fig. 7. L is the spacing between the centres of the two cylinders and D is their diameter. In this study, the ratio L/D was set to 3, while the Reynolds number, based on the cylinders' diameter, is 150.

This flow configuration generates a bifurcation of the wake patterns with dif-267 ferent acoustic responses, [24]. In the following, we refer to a flow field called 268 in literature as in-phase because it exhibits synchronised lift coefficients. It is 269 obtained using a initial field consisting of a clockwise vortex placed behind 270 both the upper and the lower cylinder, [24]. A fully-structured orthogonal 271 computational grid was used, which adopted a sponge layer with a strength 272 of 45 dB. This grid was already successfully tested for similar problems, [13]. 273 The far field was placed 200 D from the midpoint of the two cylinders (see 274 Fig. 7), grid cells were clustered near the cylinder wall with a dimensionless 275 first-cell height of 10^{-2} . The number of cells had a total number of $4.44 \cdot 10^{6}$. 276 The typical run was performed on MARCONI-100 HPC system using 192 277 CPU cores. 278

Fig. 8(a) shows a directivity plot evaluated on a circle having dimensionless radius r/D = 75. Overall, the results showed good agreement with the findings of Inoue et al. [24]. Thus, the usefulness of both the grid and solver was proven.

Fig. 8(b) shows acoustic pressure polar plots obtained with varying wall temperature gradient on the cylinders' surface. It can be clearly observed that in this case, the sound emission was significantly reduced owing to thermal wall flux application. Furthermore, this evidence was in good agreement with force coefficient modifications which are highlighted in Tab. 3.

288 4.3 Square cylinders in tandem

With the same aim as for the side-by-side arrangement, we also considered the flow and sound generation around two square cylinders in a tandem configuration at Re = 150 with L/D = 2, where L and D have the meaning expressed in Fig. 9.

²⁹³ Also in this case we have used a square computational domain with a far field

placed at 200 *D* from the origin. Quadrilateral orthogonal cells were used to discretize the flow domain. The total number of cells, N_c , was about $4.2 \cdot 10^6$, and a grid refinement was performed near the walls of the cylinders adopting $y_c/D = 10^{-2}$ as in our previous work [13]. Acoustic wave reflections at the far boundaries were removed using a configuration derived from the previous test cases; the sponge layer's strength was 45 dB. The typical run was performed on MARCONI-100 HPC system using 192 CPU-core.

Once again, caafoam solver, as well as the selected grid, are in very good agreement with DNS data of Inoue [25], as showed in Fig. 10(a). In Fig. 10(b) can be noted that acoustic pressure is significantly reduced by wall heating also in this configuration. This element is corroborated by force coefficients modifications reported in Tab. 4.

It is very clear that this specific configuration put in evidence a noticeable abatement of the emitted sound if compared with the other ones previously described. In the authors' opinion this result is connected to the aerodynamic coefficients fluctuations magnitude which is the lowest for the present arrangement. Thus, the effect of the wall thermal flux is more marked than in the other flow/sound problems addressed in this work.

312 4.4 Turbulent flow past NACA 0012 at $\text{Re} = 5 \cdot 10^4$, $M_{\infty} = 0.4$ and $\alpha = 5^\circ$

In this subsection, the numerical results of turbulent flow past NACA 0012 313 are presented and discussed. Owing to the strong spanwise coherence of the 314 pressure fluctuation for turbulent flows over airfoils operating at moderate 315 Reynolds numbers [26], the aeroacoustic field was examined through 2-D 316 DNS, [26, 27]. In the following, the undisturbed flow parameters were set for 317 validation, with reference to Sandberg et al. [26] and Jones et al. [28]. In par-318 ticular, we adopted the specific heat ratio, Prandtl number, and dimensionless 319 Sutherland's constant as $\gamma = 1.4$, Pr = 0.72, and $S/T_{\infty} = 0.3686$, respectively. 320

³²¹ The adiabatic wall condition was fixed on the airfoil surface.

The computational domain is a C-type region that extended 7.3 times the 322 airfoil chord length c in the radial direction and 5 c in the wake region. Four 323 fully-structured computational grids were used to achieve grid independence 324 in terms of both aerodynamic coefficients and acoustic pressure. All the afore-325 mentioned grids had a dimensionless height for the first cell next to the wall, 326 y_p/c , equal to 10^{-4} . Considering the complexity of the involved phenomena, it 327 was evident that the correct definition of the grid resolution requirements was 328 mandatory. In this study, an iterative grid generation method was adopted 329 to achieve a suitable mesh configuration. Cell distribution details are listed 330 in Tab. 5. Moreover, the following numerical computations were run on the 331 MARCONI-100 system using 192 CPU cores for the coarsest grid, whereas 332 the finest grid typical run employed 320 CPU cores. 333

As briefly introduced above, the configuration considered in this study pro-334 duced a complex variety of fluid phenomena. In particular, a long laminar 335 separation bubble (LSB) with an approximate extension of 0.5 c was present 336 on the suction side. Moreover, LSB reattachment led to vortex shedding that 337 evolved until the TE, where the airfoil-vortices interaction generated a dipo-338 lar tonal noise. The vortical structures that convected on the suction side are 339 shown in Fig. 11(a), whereas Fig. 11(b) provides a view of the acoustic waves 340 through the dilatation rate field $\partial u_i/\partial x_i$. In addition, it can be clearly ob-341 served that the main acoustic source was located immediately after the airfoil 342 TE. Table 6 provides evidence that for the coarsest grid, G1, the time-averaged 343 lift coefficient, $\overline{C_L}$, is underestimated; however, the time-averaged drag coef-344 ficient, $\overline{C_D}$, is slightly overestimated compared with the literature data. The 345 aerodynamic coefficients converged to those in the literature when finer grids 346 were adopted, particularly in terms of $\overline{C_L}$. Globally, a good agreement with 347 Jones [28] was reached for G3 and G4. 348

The local aerodynamic performance was estimated using the pressure coefficient $c_p = 2 \left(p - p_{\infty} \right) / \rho_{\infty} u_{\infty}^2$, and skin friction coefficient, $c_f = 2\tau_w / \rho_{\infty} u_{\infty}^2$. In Fig. 12(a) the mean pressure coefficient, $\overline{c_p}$, distribution is represented. It is confirmed that the G3 grid results provide an appropriate description of the flow around the NACA 0012 airfoil. In particular, the wide extension of the LSB was noticeable when looking at the plateau on the suction side, whereas $\overline{c_p}$ on the pressure side suggested that for the considered angle of attack, the airfoil lower surface had a flat plate-like behaviour.

On the contrary, Fig. 12(b) shows the mean skin friction coefficient, $\overline{c_f}$ distribution, along the airfoil chord. In this framework, it is interesting to note that as the grid resolution increased, the reattachment point moved toward the LE and the secondary separation region was better described.

In order to isolate the vortex shedding characteristic frequency and its related 361 sound emission, a sampling probe was used to collect the pressure data. It was 362 placed at x/c = 0.5 and y/c = 0.6. Note that the origin of the reference frame 363 was placed on the airfoil LE, and the x-axis was aligned with the airfoil chord, 364 as in Sandberg et al. [26]. Pressure data were sampled over a dimensionless 365 time fixed at 120 and Fourier transformed. As shown in Fig. 13(a), a tonal peak 366 was observed corresponding to a reduced frequency $\mu_0 = \pi f c / a_\infty (1 - M_\infty^2)$, 367 equal to 5.036. This holds true for all the considered grids. 368

It is worth emphasising that a critical point of this numerical computation is 369 correlated with proper sponge layer calibration. This flow problem was per-370 formed based on the lowest significant tonal peak exhibited by the spectrum 371 reported in Fig. 13(a). This frequency was selected to obtain $L_{sp} \cdot f/a_{\infty} = 1$, 372 which implies a thicker sponge layer than in laminar configuration. A similar 373 condition is strictly required to dissipate the vortical flow structures shed by 374 airfoil [17]. As $L_{sp} \cdot f/a_{\infty} \geq 2$ can produce sponge-layer auto-reflection issues, 375 we decided to use a compromise value to avoid an excessive overcoming of 376 the sponge thickness upper bound. The efficacy of the previous choice coupled 377 with $-\eta_{target} = 40$ dB is evident from Fig. 11(b), where unphysical pressure 378 waves are not present. 379

³⁸⁰ The aeroacoustic noise produced in this configuration was also investigated

using the polar distribution of acoustic pressure in the mean field. In this spe-381 cific case, the pressure-time history was sampled over 360 equispaced probes 382 placed along a circumference that originated from the TE and had a radius of 383 2 c. Fig. 13(b) highlights the dipolar nature of the acoustic field, related to the 384 lift pulsation dominance. Moreover, it is evident that G1 did not adequately 385 predict the airfoil sound emission, whereas G3 and G4 produced almost indis-386 tinguishable results. Furthermore, the comparison of our finest grid data with 387 the reference ones showed slight differences; in our opinion, this was because 388 of the peculiar LE treatment adopted by Sandberg et al. [26]. 389

In this subsection, we provide an additional insight into the alteration of 390 the aeroacoustic field owing to airfoil suction-side heating. Specifically, we in-391 tended to investigate whether the wall-heating effect, noted in laminar flows, 392 was still effective in the turbulent regime. Two different configurations of 393 wall thermal flux were analysed. In the first configuration, the dimensionless 394 wall temperature gradient was fixed at 1, whereas in the second, we adopted 395 $\partial T/\partial n|_w = 5$. In Fig. 14(a), we show the acoustic pressure polar plot for 396 $\mu_0 = 5.036$, derived from adiabatic computations. It can be clearly observed 397 that the sound emission was effectively reduced for both the wall thermal 398 fluxes. Moreover, the tonal peak frequency moved forward to higher values, as 399 listed in Tab. 7. Looking at Fig. 14(b) and Fig. 14(c) it is possible to observe 400 p'_{rms} polar plots connected to the μ_0 tonal peak derived from the two thermal 401 fluxes considered herein. It can be clearly observed that in these cases, only 402 the configuration producing the μ_0 tonal peak had a significant impact on the 403 sound emission. Nevertheless, the effect on the overall frequencies produced 404 a slight p'_{rms} increase, as shown in Fig. 14(d). This is because of an oppos-405 ing evidence with respect to what was observed for laminar flows. Indeed, 406 for the turbulent configuration studied herein, the wall heating produced lit-407 the amplification in the aerodynamic force pulsation (approximately 2% for 408 $\partial T/\partial n|_w = 5$), as shown in Tab. 7. Therefore, we can conclude that the active 409 thermal control of the aeroacoustic field derived from turbulent flows is not 410

as effective as that in the laminar regime, in terms of acoustic power level
reduction. In particular, the wall thermal flux acts as a tonal peak frequency
shifter.

414 5 Conclusions

In this study, we investigated the effectiveness of wall heating as an active control technique for the aerodynamically generated sound and/or noise for laminar and turbulent cases. In laminar flows, we investigated both single and multiple body arrangements with reference to well established literature data. By contrast, turbulent flow analyses are limited to NACA 0012 airfoil owing to their high computational resource requirements.

Specifically, we observed that far-field sound abatement at higher wall temper-421 atures was produced for laminar flows over bluff bodies. This is related to the 422 reduction in aerodynamic force pulsations generated by the Karman vortex 423 street, which is shed over bluff bodies in laminar flows. It is worth noting that 424 a similar effect was obtained in several configurations involving more than one 425 single body. For an isolated circular cylinder, we observed that localised wall 426 heating produced the same effect for the overall wall heating. Moreover, ther-427 mal fluxes focused on separated flow regions is the key aspect for adequately 428 controlling force oscillations and related sound emission. 429

In this study, we also addressed for the first time, the impact of wall heating on 430 aeroacoustic field derived from a turbulent flow regime. Thus, we computed 431 the flow and acoustic field developed around the NACA 0012 airfoil. First, 432 we further validated our solver for turbulent flows against well-established 433 literature benchmarks. Therefore, we observed that the sound emission was 434 effectively lower for a reduced frequency with the same maximum tonal contri-435 bution obtained from adiabatic wall condition. However, the overall frequency 436 contribution produced a slight p'_{rms} increase. Based on these results, we con-437

clude that the thermal control of the aeroacoustic field derived from turbulent
flows is not as effective as that in the laminar regime in terms of the acoustic
power level. By contrast, the wall thermal fluxes can be considered effective
as tonal peak frequency shifters. Therefore, the above discussed technique is
promising for specific applications.

443 6 Acknowledgements

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Figure 1. Flow past a circular cylinder. Solver validation.



Figure 2. Flow past a circular cylinder. Fixed wall temperature effect, r/D = 40.



Figure 3. Flow past a circular cylinder. Fixed wall temperature gradient effect, $r/D=40.\,$



Figure 4. Flow past a circular cylinder. C1 case, r/D = 40.



Figure 5. Flow past a circular cylinder. C2 case, r/D = 40.



Figure 6. Flow past a circular cylinder. C3 case, r/D = 40.



Figure 7. Cylinders arranged side by side.



Figure 8. Cylinders arranged side by side. Acoustic pressure results.



Figure 9. Square cylinders in tandem configuration.



Figure 10. Square cylinders in tandem configuration. Acoustic pressure results.



Figure 11. Flow and acoustic field representation, NACA 0012 airfoil. Grid G4.



Figure 12. Time-averaged pressure and skin friction coefficient distribution



Figure 13. Dimensionless acoustic pressure data. NACA 0012 airfoil.



Figure 14. Thermal effect on the acoustic pressure, NACA 0012, Grid G4.

Case	ΔC_L	$C_{L,rms}$	$\Delta C_D \cdot 10^2$	$\overline{C_D}$	St.
$\partial T/\partial n _w = 0$	0.52	0.37	2.56	1.333	0.182
$\partial T/\partial n _w = 0.5$	0.49	0.35	2.45	1.333	0.182
$\partial T/\partial n _w = 1$	0.46	0.33	2.25	1.327	0.181
$\partial T/\partial n _w = 2$	0.41	0.29	1.96	1.323	0.180
$T_w = 2T_\infty$	0.36	0.25	1.61	1.335	0.177
$T_w = 3T_\infty$	0.25	0.18	0.96	1.327	0.170

Table 1 Circular Cylinder - Re = 150, $M_{\infty} = 0.2$, $\mu = cost$ - Force coefficients.

Cylinder - Re = 150, $M_{\infty} = 0.2$, $\mu = \mu(T)$ - Force coefficients.						
Case	ΔC_L	$C_{L,rms}$	$\Delta C_D \cdot 10^2$	$\overline{C_D}$	St	
$\partial T/\partial n _w = 0$	0.52	0.36	2.56	1.333	0.182	
$\partial T/\partial n _w = 0.5$	0.47	0.33	2.21	1.329	0.180	
$\partial T/\partial n _w = 1$	0.41	0.29	1.84	1.325	0.178	
$\partial T/\partial n _w = 2$	0.31	0.22	1.14	1.315	0.173	
$T_w = 2T_\infty$	0.24	0.17	0.67	1.382	0.161	
$T_w = 3T_\infty$	0.08	0.06	0.07	1.402	0.139	

Table 2 Circular Cylinder - Re = 150, $M_{\infty} = 0.2$, $\mu = \mu(T)$ - Force coefficients.

Table 3

Square Cylinder Side by Side, L/D = 3, in phase - Re = 150, $M_{\infty} = 0.2$, $\mu = \mu(T)$ - Force coefficients.

Case	ΔC_L	$\overline{C_L}$	ΔC_D	$\overline{C_D}$	St
$\partial T/\partial n _w = 0$	0.41	± 0.075	0.114	1.592	0.155
$\partial T/\partial n _w = 0.5$	0.39	± 0.096	0.101	1.589	0.151
$\partial T/\partial n _w = 1$	0.35	± 0.119	0.087	1.589	0.149
$\partial T/\partial n _w = 2$	0.26	± 0.163	0.061	1.591	0.146

Table 4

Square cylinders in tandem configuration, L/D = 2, Re = 150, M_{∞} = 0.2, $\mu = \mu(T)$ - Force coefficients.

$\Delta C_L \cdot 10^2$	$\overline{C_L} \cdot 10^4$	$\Delta C_D \cdot 10^4$	$\overline{C_D}$	St
1.951	0	1.05	1.28	0.134
1.118	0.48	0.47	1.27	0.128
0.631	1.10	0.15	1.26	0.122
0.043	0.07	2.90	1.20	0.105
$\Delta C_L \cdot 10^2$	$\overline{C_L} \cdot 10^4$	$\Delta C_D \cdot 10^4$	$\overline{C_D}$	St
5.30	0	7.65	-0.196	0.134
3.80	2.42	3.21	-0.158	0.128
2.59	3.79	1.16	-0.114	0.122
0.26	7.40	0.87	-0.010	0.105
	$\begin{array}{c} \Delta C_L \cdot 10^2 \\ 1.951 \\ 1.118 \\ 0.631 \\ 0.043 \\ \end{array}$ $\begin{array}{c} \Delta C_L \cdot 10^2 \\ 5.30 \\ 3.80 \\ 2.59 \\ 0.26 \end{array}$	$\Delta C_L \cdot 10^2$ $\overline{C_L} \cdot 10^4$ 1.95101.1180.480.6311.100.0430.07 $\Delta C_L \cdot 10^2$ $\overline{C_L} \cdot 10^4$ 5.3003.802.422.593.790.267.40	$\Delta C_L \cdot 10^2$ $\overline{C_L} \cdot 10^4$ $\Delta C_D \cdot 10^4$ 1.951 0 1.05 1.118 0.48 0.47 0.631 1.10 0.15 0.043 0.07 2.90 $\Delta C_L \cdot 10^2$ $\overline{C_L} \cdot 10^4$ $\Delta C_D \cdot 10^4$ 5.30 0 7.65 3.80 2.42 3.21 2.59 3.79 1.16 0.26 7.40 0.87	$\Delta C_L \cdot 10^2$ $\overline{C_L} \cdot 10^4$ $\Delta C_D \cdot 10^4$ $\overline{C_D}$ 1.95101.051.281.1180.480.471.270.6311.100.151.260.0430.072.901.20 $\Delta C_L \cdot 10^2$ $\overline{C_L} \cdot 10^4$ $\Delta C_D \cdot 10^4$ $\overline{C_D}$ 5.3007.65-0.1963.802.423.21-0.1582.593.791.16-0.1140.267.400.87-0.010

Upstream cylinder

Table 5Computational domain discretisation.

Grid	$\rm N_{foil}$	$\rm N_{radial}$	$\mathrm{N}_{\mathrm{wake}}$	$N_{\rm tot}$	
G1	1084	690	755	1789860	
G2	2000	690	755	2421900	
G3	4000	690	755	3801900	
G4	4000	800	1000	4800000	

Table 6 Time-averaged lift and drag coefficients.

	G1	G2	G3	G4	Jones [28]
$\overline{C_L}$	0.439	0.473	0.484	0.484	0.499
$\overline{C_D} \cdot 10^2$	3.13	3.16	3.17	3.17	3.07

Table 7 NACA 0012, Re = $5 \cdot 10^4$, M_{∞} = 0.4. Wall heating impact

	$\partial T/\partial n _w = 0$	$\partial T/\partial n _w = 1$	$\partial T/\partial n _w = 5$
$\overline{C_L}$	0.484	0.484	0.484
$\overline{C_D} \cdot 10^2$	3.17	3.17	3.20
$C_{L_{RMS}} \cdot 10^2$	1.87	1.88	1.91
$C_{D_{RMS}} \cdot 10^3$	2.02	2.05	2.16
μ_0	5.036	5.044	5.061