Case Studies in Thermal Engineering 35 (2022) 102125

Contents lists available at ScienceDirect

Case Studies in Thermal Engineering

journal homepage: www.elsevier.com/locate/csite

Numerical evaluation of heat transfer effects on the improvement of efficiency of a spark ignition engine characterized by cylinder variability

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HIGHLIGHTS

- Heat transfer affects indicated efficiency of spark ignition engines.
- Cylinder-by-cylinder variation penalizes engine thermal efficiency.
- 1D model reproduces overall performance and thermal behavior of cylinders.
- Heat transfer plays a relevant role in efficiency optimization at low speed/load.
- Numerical procedure supports a first-attempt thermal optimization of SI engine.

ARTICLE INFO

Keywords:
SI engine
Heat transfer
1D modelling
Efficiency
Cylinder variability

ABSTRACT

In this work, the effects of in-cylinder heat transfer on indicated thermal efficiency of a spark ignition engine showing a cylinder-to-cylinder variation are numerically analyzed. A 1D CFD model of engine is developed and integrated with a turbulent combustion sub-model and with a refined thermal sub-model for cylinders and exhaust pipes. The model is validated against the engine measurements. Thermal sub-model includes a Finite Element (FE) approach to predict the temperatures of cylinders and of exhaust pipes. The model correctly reproduces the thermodynamic behavior of cylinders at varying the operating condition. Simulations at low load and speed indicate that in-cylinder heat transfer represents a relevant percentage on total fuel energy entering the cylinder. Therefore, heat transfer exerts an important influence on the improvement of engine indicated thermal efficiency when considering the sole combustion phasing optimization of cylinders and the suppression of cylinder-to-cylinder variation.

Acronyms

0D/1D/3D	Zero/One/three dimensional
A/F	Air/Fuel
AFTDC	After firing top dead centre
CAD	Crank angle degree
CFD	Computational Fluid dynamic
DoE	Design of Experiments

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https://doi.org/10.1016/j.csite.2022.102125
Received 8 April 2021; Received in revised form 3 May 2022; Accepted 13 May 2022
Available online 20 May 2022

EGR	Exhaust gas recirculation
FE	Finite Element
HT	Heat transfer
ICE	Internal Combustion engine
IMEP	Indicated mean effective pressure
ITE	Indicated Thermal Efficiency
LP	Low pressure
MBT	Maximum Brake Torque
MFB_{50}	Angle of 50% of mass fraction burned
$MFB_{10\text{-}90}$	Angular duration between 10% and 90% of mass fraction burned
PFI	Ported fuel injection
SI	Spark Ignition
TDC	Top Dead Centre
TIT	Turbine inlet temperature
TWC	Three way Catalyst

Symbols

A	Area
dx	Discretization length
D	Equivalent Diameter
e	Internal energy per unit mass
h	Enthalpy per unit mass/Convective heat transfer coefficient
L_{max}, L_{min}	Maximum and minimum flame wrinkling scale
m	Mass
p	Pressure
Q	Exchanged heat
S_L	Laminar Flame speed
t	Time
T	Temperature/Tumble vortex
u	Velocity
V	Volume

Greeks

λ	Ratio between the actual A/F ratio and the stoichiometric A/F ratio

1. Introduction

The issue of the environmental impact for internal combustion engines (ICEs) in terms of CO_2 and noxious emissions has forced the manufacturer to optimize the performance during the development phase. For Spark Ignition (SI) ICEs, a number of solutions has been proposed to improve the thermal efficiency and pollutants. Main techniques employed in SI ICEs are: innovative valve strategies [1], cooled EGR [2], water injection [3], increased or variable compression ratio [4], alternative fuels and lean/ultra-lean combustions by advanced combustion strategies [5] or turbulent jet ignition [6]. An increasing emphasis is devoted to the ICEs optimization during the development stage, including the thermo-fluid-dynamic processes. An important role is played by heat transfer phenomena, since they have an impact on performance, especially at low speeds and loads. In particular, in-cylinder heat transfer is crucial for the engine efficiency, influencing the occurrence of abnormal combustions. Thermal optimization represents a key step for a regular engine operation, avoiding the risk of overheating, seizure and thermal failures. In addition, some design errors may lead to not negligible cylinder-to-cylinder variation which penalizes heat transfer and efficiency. In this condition, the optimization of individual cylinder operation has effect on the combustion evolution, heat losses and fuel consumption. Generally, both experimental and numerical methods can be adopted to quantify the effects of cylinder operation variability and of heat transfer non-uniformity on engine performance. Experimental methods are often preferred for thermal analysis even if complex, expensive and intrusive instrumentations are required [7]. They are more fitted to actual behavior, but a fast and detailed thermal study is not easily feasible. Numerical models allow to forecast and optimize the engine thermal behavior at varying the operating condition within a reasonable time. Among the numerical approaches, 3D models are characterized by high accuracy and greater calculation time. They are usually adopted to refine thermal analysis of limited engine parts. Conversely, 1D models allow the description of entire engine and if enhanced with advanced sub-models they represent tools for system-level analysis, capable to reproduce the thermal performance with low numerical efforts. Different studies are available in literature on heat transfer processes of engines and relative impact on performance. Literature papers are mainly oriented to thermal analysis of cylinders [8–13]. As an example, Broekaert et al. [9,10] discussed in two papers the in-cylinder heat transfer measurements for SI engines, identifying the driving factors of convective heat transfer. The effects of different engine settings and fuels on heat transfer were studied by a Design of Experiment (DoE) method.

Berni et al. [12] proposed a methodology to refine the prediction of in-cylinder heat flux for SI engines. They developed a heat transfer sub-model for 3D CFD codes aiming to eliminate the overestimation of wall heat flux by standard sub-models. Referring to 1D approach, Irimescu et al. [13] presented an in-cylinder convective heat transfer correlation, validated with optical measurements, and requiring less calibration than Woschni equation. The present study deals with the effects of in-cylinder heat transfer on efficiency and exhaust temperature of a SI engine showing a cylinder-to-cylinder variation. Engine is a twin-cylinder turbocharged PFI SI unit, equipped with a low pressure (LP) cooled EGR circuit (Table 1).

Main contribution of this work is the methodology consisting in a 1D modelling approach enhanced by turbulent combustion and thermal sub-models. Thermal sub-model includes the finite element (FE) approach to forecast thermal characteristics of cylinders and exhaust pipes. Following the methodology, the mechanisms leading to efficiency losses caused by cylinder variability can be consistently simulated and, subsequently, optimized. Indeed, the apparent cylinder-by-cylinder variation of SI engine induces non optimal performance and calibration during experiments. This issue can be preliminary overcome by performing numerical optimizations via 1D model.

First, the engine experimental analysis is performed in various speed/load points. Main performance and in-cylinder pressure traces are recorded, showing a noticeable cylinder-to-cylinder variation. A 1D engine model is developed and validated against the measured data. It is applied to evaluate the efficiency improvement at low load, arising from the sole optimization of combustion process and from the suppression of air/fuel (A/F) ratio unbalance between cylinders. Turbine inlet temperature (TIT) variation deriving from balancing the A/F ratio is also analyzed.

2. Numerical model

Engine is schematized by a network of pipes and volumes. A 1D flow inside the intake/exhaust pipes is implemented, while 0D approach is adopted for cylinders. Fuel injection is realized by standard injector, neglecting spray and liquid film formations on intake walls. Compressor and turbine behaviors are reproduced by the experimental steady maps [14]. The flow inside intake/exhaust pipes is simulated by solving the conservation equations of mass, energy and momentum (1)–(3).

$$\frac{dm}{dt}=\sum_{i=1}^{N}\dot{m} \tag{1}$$

$$\frac{d(me)}{dt}=-p\frac{dV}{dt}+\sum_{i=1}^{N}(\dot{m}h)-\frac{dQ_w}{dt} \tag{2}$$

$$\frac{d\dot{m}}{dt}=\left\{-dpA+\sum_{i=1}^{N}(\dot{m}u)-4C_f\frac{\rho u|u|}{2}\frac{Adx}{D}-AC_p\left(\frac{\rho u|u|}{2}\right)\right\}/\mathrm{dx} \tag{3}$$

where, m, ṁ, e, h, p, V and u are mass, mass flow rate, internal energy per unit mass, enthalpy per unit mass, pressure, volume, and velocity at boundary. A is the flow area and D the equivalent diameter. Q_w is the heat exchanged through walls. C_f and C_p are friction losses and pressure coefficients, while dp and dx indicate the differential pressure and the discretization length. For cylinders, mass and energy equations are solved. During closed valve period, cylinder volume is subdivided in two zones (burned and unburned) and the energy equation is specialized for each zone.

2.1. Combustion and heat transfer sub-models

Advanced in-cylinder sub-models have been adopted and coupled to engine model in the form of user routines. Combustion is reproduced by the fractal formulation [15], according to equation (4):

$$\frac{dm_b}{dt}=\rho_u\cdot A_T\cdot S_L=\rho_u\cdot A_L\cdot\Sigma\cdot S_L=\rho_u\cdot A_L\cdot\left(\frac{L_{max}}{L_{min}}\right)^{D_3-2}\cdot S_L \tag{4}$$

where the burn rate dm_b/dt is proportional to unburned gas density ρ_u, the area of laminar flame front A_L, the flame wrinkling Σ and the laminar flame speed S_L. D_3 is the fractal dimension, L_{max} and L_{min} the scales of macro- and micro-vortices of turbulent flow field, respectively; they are estimated by an advanced K-k-T turbulence sub-model, deeply treated in [16]. S_L of gasoline is specified through a refined kinetically-derived correlation reported in (5):

Table 1
Engine data.

Model	2 cylinders, Turbocharged, PFI
Compression ratio, -	9.9
Displacement, cm^3	875
Valve number, -	4 valves/cylinder
Max Brake Power, kW	64.6@5500 rpm
Max Brake Torque, Nm	146.1@2500 rpm

$$S_L = S_{L0} \cdot \left(\frac{T}{T_{ref}}\right)^{\alpha} \left(\frac{p}{p_{ref}}\right)^{\beta} EGR_{factor} \tag{5}$$

where S_{L0} is flame speed at reference conditions, p the pressure and T the temperature (unburned zone), while EGR_{factor} is a reduction term accounting for the presence of EGR [15]. Referring to thermal modelling, the convective mechanism is adopted for cylinders. In-cylinder heat transfer (gas-to-wall) sub-model resembles the Hohenberg one, (6):

$$h = A \cdot V^{-0.06} p^{0.8} T^{-0.4} \left(k_1 \cdot v_{pm} + k_2 \cdot B\right)^{0.8 \cdot k_3} \tag{6}$$

being V the volume and v_{pm} the mean piston speed. Calibration constants A and B are calculated by Hohenberg and here used as 130 and 1.4, respectively. k_1, k_2 and k_3 are user-defined constants. FE approach is specified for cylinders, defining the material, geometry and initial temperatures of liner, piston, head, valves and ports [17]. A thermal solver is activated for exhaust pipes, where thermal properties (thickness, material and emissivity) and wall boundary conditions (temperature and heat transfer coefficient) are specified. Thermal solver for exhaust pipes combines conductive, convective and radiative heat transfer modes. All sub-models are tuned to match the measured evolutions of in-cylinder pressure signals [18].

3. Model validation

Model ability is proved in 34 acquired points, including different loads (5 ÷ 18 bar IMEP), speeds (2000 ÷ 4000 rpm) and EGR rates. In Fig. 1a–b the numerical/experimental comparisons of Indicated thermal efficiency (ITE) and Turbine inlet temperature (TIT) are plotted. ITE (Fig. 1a) presents an average percent error of 2.4%, demonstrating the model accuracy in capturing the combined effects of turbulent combustion and in-cylinder heat transfer phenomena. TIT (Fig. 1b) shows a low average absolute error of 23.9 K, confirming a refined thermal modeling for exhaust ducts.

Fig. 2 depicts the numerical/experimental comparison of in-cylinder pressure cycles for two cases labelled as 2000@5 (2000 rpm, 5 bar IMEP) and 3000@11-EGR (3000 rpm, 11 bar IMEP and EGR = 19.8%). In both points, good agreements are realized for Cyl#1 and Cyl#2 over a relevant portion of the engine cycle, capturing the cylinder variability induced by A/F ratio non-uniformity (lean mixture for Cyl#1 and rich mixture for Cyl#2). Presented numerical/experimental agreements of pressure traces also confirm the accuracy of in-cylinder heat transfer modeling.

Fig. 3 shows the predicted average wall temperatures for liner, piston and head of cylinders in examined points. Consistently with physical expectations, FE model indicates higher wall temperatures at increasing the load; slightly greater wall temperatures for Cyl#2 are provided according to the cylinder thermodynamic conditions. Computed head temperatures at 3000@11-EGR fall in the measured band for the same operating point referred to a similar SI unit [11], thus proving the FE model consistency. The energy balance of engine is also simulated at 2000@5; numerical outcome, not reported here for brevity, is assessed with a literature-derived result [19] for the case of a similar SI engine at equal speed and at low load region. This comparison highlights a good alignment of the computed heat loss percentage on total fuel energy with respect to the one simulated in [19], thus further confirming the reliability of adopted thermal sub-model.

4. Results and discussion

The effects of cylinder variability suppression on engine heat transfer and efficiency are analyzed at low load point 2000@5. First, combustion phasing (MFB_{50}) of each cylinder is optimized under unbalanced A/F ratio. Then, A/F ratio is balanced between cylinders at stoichiometric level and MBT condition is identified. In both cases, heat transfer and indicated efficiency variations are evaluated. Combustion phasing optimization of cylinders under non-uniform A/F ratio is performed by DoE methodology coupled to 1D model, where MFB_{50} is allowed to vary in the range 0–9 CAD after firing top dead center (AFTDC) with a 1 CAD step. 100 simulations are realized and an improved ITE is identified. Fig. 4a–b shows that the improved ITE corresponds to MFB_{50} equal to 7 CAD AFTDC both for Cyl#1 and Cyl#2.

In Fig. 5, the best DoE solution is compared with the simulation of experimental case (Reference). Fig. 5 highlights that the sole

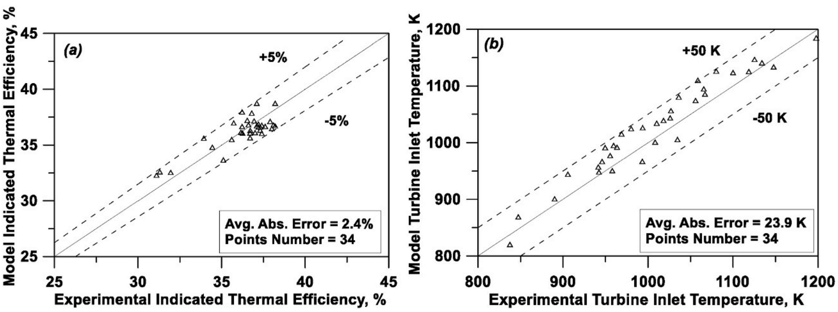

Fig. 1. Numerical/Experimental comparison of Indicated Thermal efficiency (a) and Turbine Inlet Temperature (b).

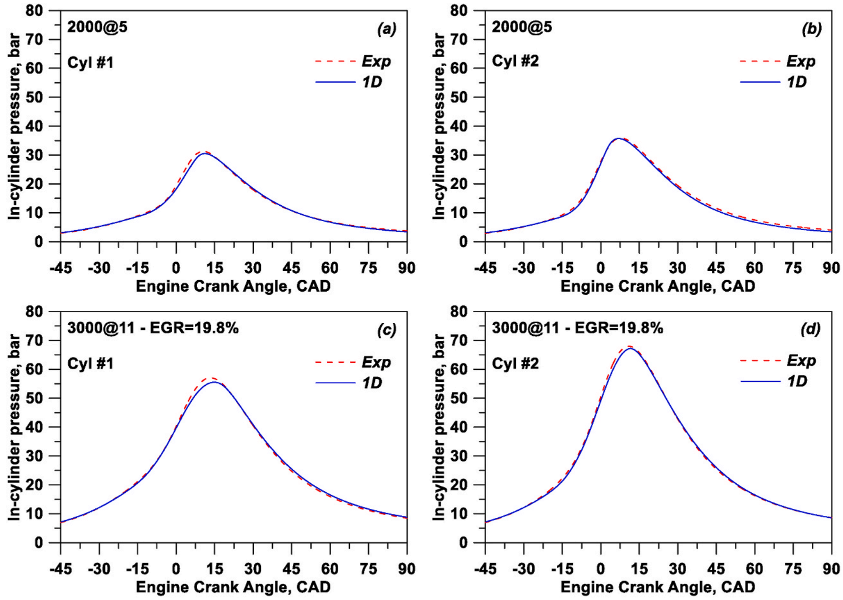

Fig. 2. Numerical/Experimental comparison of in-cylinder pressure cycles for Cyl#1 and Cyl#2 at 2000@5 (a–b) and 3000@11-EGR = 19.8% (c–d).

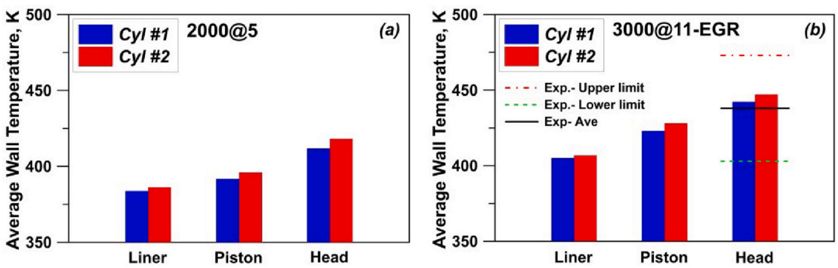

Fig. 3. Average wall temperatures of liner, piston and head for Cyl#1 and Cyl#2 at 2000@5 (a) and 3000@11-EGR (b).

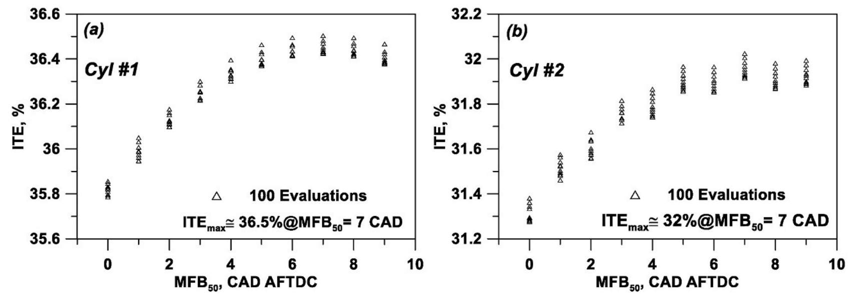

Fig. 4. DoE evaluations: ITE trend with MFB_{50} for Cyl#1 (a) and Cyl#2 (b).

combustion phasing optimization involves a slight ITE increase (~1.3%) compared to numerical-related reference level. The engine ITE advantage is due to the prevailing effect of a reduced engine heat transfer (HT) (gain of ~5.5%) compared to the lengthening of combustion duration ($MFB_{10\text{-}90}$).

In particular, the main driver to the HT power decrease is represented by reduction in the burned gas temperatures. In addition, the optimization of combustion phasing at part load allows to realize a better balance between reduced compression work and increased expansion work [20].

A subsequent optimization is performed at 2000@5, assigning the same A/F ratio (stoichiometric level) to cylinders and considering the MBT. In this case, a value of MFB50 = 7 CAD AFTDC is still confirmed as optimal by the numerical analysis. The outcomes are plotted in Fig. 6, which shows the comparison of numerical results between reference and optimized parameters.

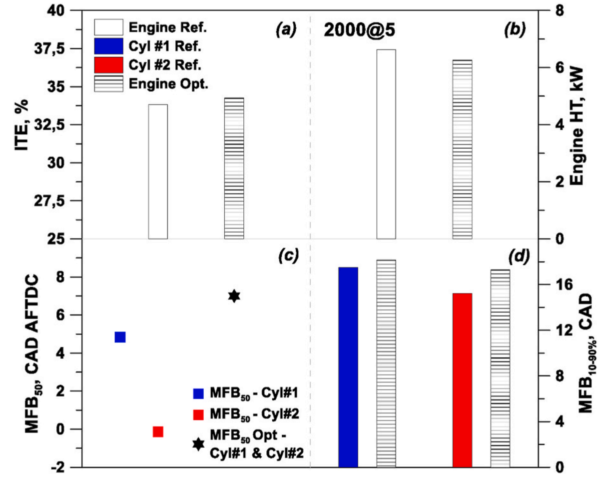

Fig. 5. Comparison of best DoE solution with reference numerical one: ITE (a), Engine HT (b), MFB_{50} (c), $MFB_{10\text{-}90}$ (d).

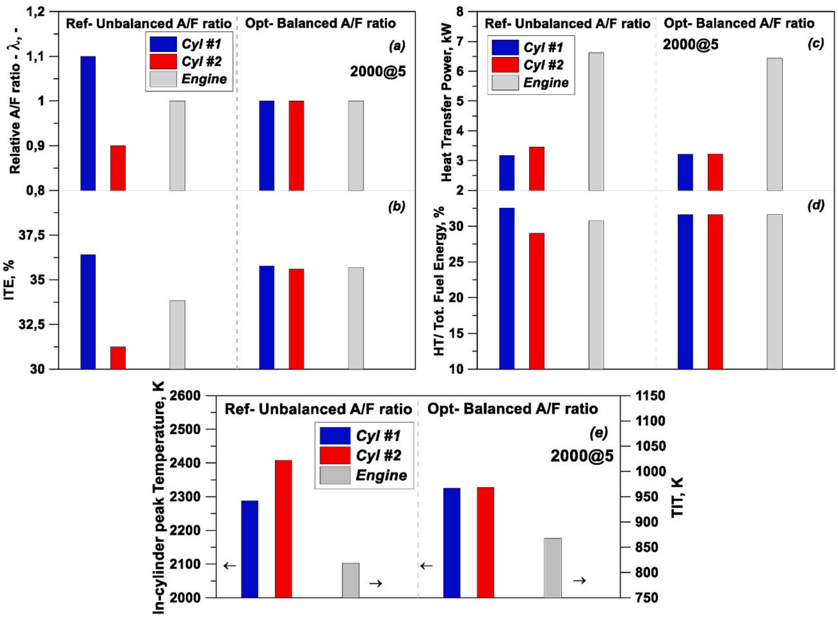

Fig. 6. Comparison between reference and optimized (balanced A/F ratio and MBT) results at 2000@5: relative A/F ratio, λ, (a), ITE (b), HT power (c), HT/Tot. fuel energy (d), In-cylinder peak temperature and TIT (e).

The A/F ratio unbalance between cylinders (Fig. 6a) induces a higher ITE for Cyl #1 (lean mixture) compared to Cyl #2 (rich mixture). The suppression of A/F ratio variability modifies the in-cylinder peak temperatures of gas (Fig. 6e). As a consequence, engine HT is reduced (Fig. 6c), thanks to the predominant effect of a decreased HT for Cyl #2 compared to HT increase for Cyl#1. At 2000@5, a remarkable percent weight is associated to the HT energy when compared to the total energy of fuel entering the single cylinder (~32% in Fig. 6d). Based on last consideration, the improvement in engine HT power (~2.8%) combined with the rise of thermal efficiency for Cyl #2 contribute to ITE increase of engine (~5.5% in Fig. 6b). Simultaneously, the turbine inlet temperature (TIT) shows an increase of about 50 K (Fig. 6e), due to slightly delayed combustions combined with the over-fueling elimination for Cyl#2. However, TIT rise could improve the performance of three way catalyst (TWC) at low load.

5. Conclusion

The effects of in-cylinder heat transfer on efficiency and exhaust temperature of an SI engine showing a cylinder variability are numerically evaluated. A 1D engine model is developed and enhanced with user sub-models, including a FE approach. Model correctly simulates measured engine ITE, TIT and pressure histories of cylinders in different speed/load points. At low load (2000@5), a combustion phasing optimization with unbalanced A/F ratio between cylinders leads to ITE increase of 1.3%. If the balancing of A/F ratio between cylinders is combined with MBT condition, ITE advantage rises to 5.5%. In the latter case, heat transfer plays a relevant role on the energy balance of engine (32%), thus allowing for its efficiency improvement with an HT power reduction of 2.8%. On the other hand, the TIT showed an increase of 50 K, contributing to improve TWC performance. Proposed methodology represents a compromise between results accuracy and computational time for a first-attempt thermal and efficiency optimization of SI engines.

CRediT authorship contribution statement

Luigi Teodosio: Methodology, Software, Validation, Formal analysis, Investigation, Writing – original draft, Writing – review & editing, Supervision. **Cinzia Tornatore:** Conceptualization, Methodology, Validation, Formal analysis, Investigation, Resources, Data curation, Writing – original draft, Writing – review & editing, Supervision. **Luca Marchitto:** Conceptualization, Methodology, Validation, Formal analysis, Investigation, Resources, Data curation, Writing – original draft, Writing – review & editing, Supervision.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Acknowledgements

The authors thank Mr. Alfredo Mazzei for the support to the experiments.

Appendix A. Supplementary data

Supplementary data to this article can be found online at https://doi.org/10.1016/j.csite.2022.102125.

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