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Experimental investigation of airflow and heat transfer by natural convection in an insulated box with a Phase Change Material using a Particle Image Velocimetry technique

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Credit Author Statement

Author's name	Contributions
Tanathep Leungtongkum	Conceptualization, Methodology, Investigation, Validation, Formal analysis, Software, Writing - Original Draft Preparation, Visualization.
Onrawee Laguerre	Validation, Formal analysis, Writing - Review & Editing, Supervision, Project Administration, Funding acquisition.
Denis Flick	Methodology, Validation, Formal analysis, Writing - Review & Editing, Supervision.
Steven Duret	Validation, Formal analysis, Writing - Review & Editing, Supervision.
Alain Denis	Investigation, Software.
Nattawut Chaomuang	Conceptualization, Methodology, Investigation, Validation, Formal analysis, Software, Writing - Original Draft Preparation, Visualization, Funding acquisition.

23 empty case, downward flow was unstable, and with the load, it followed preferential pathways.
 24 The type of load exerted little effect on flow patterns at thermal steady state. Thus, a simpler
 25 load (extruded polystyrene) can be used in the first approach. The maximum velocity was about
 26 $0.1 \text{ m}\cdot\text{s}^{-1}$, so free convection cannot be neglected compared with conduction. Regarding
 27 temperature performance, PCM on the side and on the lid showed no substantial difference if
 28 gaps were left between the load and the walls or PCM.

29 **Keywords:** Airflow, Heat Transfer, Natural convection, Insulated box, Phase Change Material

30 Nomenclature

31	A	Aspect ratio [-]
32	c_p	Specific heat [$\text{J}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}$]
33	g	Acceleration due to gravitation [$\text{m}\cdot\text{s}^{-2}$]
34	Gr	Grashof number [-]
35	H	Height [m]
36	L	Length [m]
37	Ra	Rayleigh number [-]
38	Pr	Prandtl number [-]
39	t	Time [s]
40	T	Temperature [$^{\circ}\text{C}$ or K]
41	ΔT	Temperature difference [$^{\circ}\text{C}$ or K]
42	U	Overall heat transfer coefficient [$\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$]
43	v	Velocity magnitude [$\text{m}\cdot\text{s}^{-1}$]
44	W	Width [m]
45	x, y, z	Coordinates [m]
46		Greek letters

47	α	Thermal diffusivity [$\text{m}^2 \cdot \text{s}^{-1}$]
48	β	Thermal expansion coefficient [K^{-1}]
49	λ	Thermal conductivity [$\text{W} \cdot \text{m}^{-1} \cdot \text{K}^{-1}$]
50	ε	Emissivity [-]
51	ρ	Density [$\text{kg} \cdot \text{m}^{-3}$]
52	μ	Dynamic viscosity [$\text{kg} \cdot \text{m}^{-1} \cdot \text{s}^{-1}$]
53	ν	Kinematic viscosity [$\text{m}^2 \cdot \text{s}^{-1}$]

54 Subscripts

55	a	air
56	c	cold
57	g	glass
58	h	hot
59	s	surface
60	th	thermal
61	w	wall
62	∞	free stream

63 Abbreviations

64	CFD	Computational Fluid Dynamics
65	PCM	Phase Change Material
66	PIV	Particle Image Velocimetry
67	TYL	Tylose
68	XPS	Extruded Polystyrene

69 1. Introduction

70 Insulated boxes equipped with Phase Change Material (PCM) have been widely used in the
71 transport of various temperature-sensitive products such as meat and fishery products (Paquette
72 et al., 2017), fruit and vegetables (Zhao et al., 2019), and pharmaceutical products (Robertson
73 et al., 2017). The advantages of insulated boxes with PCM include low investment and
74 operating costs, flexibility in storage temperatures and volumes, and ease of maintenance (Zhao
75 et al., 2020). However, two drawbacks emerge: the temperature in an insulated box is difficult
76 to control, and temperature heterogeneity is often observed, with a low temperature in the
77 vicinity of the PCM and a high temperature further away from the PCM (Laguerre et al., 2008a;
78 Margeirsson et al., 2012; Navaranjan et al., 2013). This problem can result in the degradation
79 of product quality, for instance chilling/freezing injury (a temperature that is too low leads to
80 cell damage) and pose a safety risk (a temperature that is too high leads to bacterial growth)
81 (Laguerre et al., 2019). As reviewed by Leungtonkum et al. (2021), numerous experimental
82 and numerical studies have been conducted to evaluate the thermal performance of insulated
83 boxes equipped with PCM. Choi and Burgess (2007) developed simplified heat transfer models
84 based on an ice melting test for the estimation of heat flow resistance (R-value) which is a factor
85 generally used to determine the insulation performance of the box. Later, Singh et al. (2008)
86 applied this method and reported the R-values of insulated boxes made of various insulating
87 materials with different wall thicknesses and box dimensions. Various 2D and 3D
88 Computational Fluid Dynamics (CFD) models of insulated boxes with PCM were developed to
89 study the influences of different factors on the evolution of air/product temperatures (Du et al.,
90 2020; Laguerre et al., 2019; Laguerre et al., 2018; Margeirsson et al., 2012; Paquette et al.,
91 2017; Xiaofeng and Xuelai, 2021). These factors included box characteristics (dimensions,
92 shape of inner corners, types and thicknesses of insulating materials, internal surface
93 emissivity), PCMs (types, mass, and position in a box), products (types, initial temperature,
94 mass, and arrangement in a box), and operating conditions (ambient temperature and transport

95 duration) (Leungtongkum et al., 2021). To reduce computational time, analytical and zonal
96 models were also developed as a complementary approach (East et al., 2009; Laguerre et al.,
97 2019; Laguerre et al., 2018). These models enable useful information (e.g., PCM melting
98 duration and product temperature change during shipment) to be acquired as a function of the
99 box design and usage conditions. Xiaofeng and Xuelai (2021) developed an insulated box with
100 multiple partitions for delivery of various products, which require different preservation
101 temperatures (ambient, chilled, and frozen temperatures) in the same box. Despite these efforts,
102 there is still no universal optimal condition that can be applied to control the air/product
103 temperatures in an insulated box.

104 All heat transfer modes can occur simultaneously in an insulated box with PCM: conduction
105 (inside the product, PCM and the walls of the box), natural convection (between air and
106 product/PCM in the box), natural/forced convection (between external air and the box), and
107 radiation (between the walls and product/PCM in the box) (Leungtongkum et al., 2021). For
108 model simplification, most numerical studies consider heat transfer by conduction, and
109 sometimes also by radiation, while natural convection inside the box tends to be neglected (Du
110 et al., 2020). In fact, all heat transfer modes can be of the same order of magnitude, and natural
111 convection can exert significant impacts, especially on temperature heterogeneity in the
112 insulated box (Laguerre and Flick, 2010).

113 Natural convection in closed cavities occurs in a wide range of engineering applications and
114 has been extensively studied with both experimental and numerical approaches
115 (Miroshnichenko and Sheremet, 2018; Pandey et al., 2019). The configurations commonly
116 studied are air or water in a rectangular cavity of which two opposite (horizontal/vertical) walls
117 are maintained at different constant and uniform temperatures (hot and cold) while the other
118 walls are perfectly insulated (adiabatic). Many studies also introduce porous medium or solid

119 objects (flat plate, cylinder and sphere) in the cavity to observe their effects on heat transfer and
120 airflow (Ataei-Dadavi et al., 2019; Lee et al., 2016). The knowledge acquired using a cavity
121 filled with porous media can be applied to the case of small products such as cereal grains (ratio
122 between characteristic lengths of product and cavity ≤ 0.02). However, the models are limited
123 in the case of larger products (e.g., meat, fruit and vegetables) where the ratio is much higher
124 (> 0.1) (Laguerre et al., 2008b). Moreover, the configuration of a product-loaded insulated box
125 with PCM is more complex because all walls are subjected to heat loss and the wall equipped
126 with PCM (cold wall) has a non-uniform temperature as PCM melts more rapidly at some
127 positions (Jevnikar and Siddiqui, 2019). Several experimental and numerical studies were
128 conducted to investigate heat transfer and airflow in air-filled cavities with PCM at one wall.
129 Aitlahbib and Chehouani (2015) developed a two-dimensional numerical model to evaluate the
130 thermal performance of an insulated container with one vertical wall equipped with PCM, the
131 opposite one kept at higher temperature while other walls were well-insulated. The results
132 demonstrated that the container with PCM wall achieved superior thermal performance than
133 the one without. The same cavity configuration was numerically studied by Labihi et al. (2017).
134 The effect of volume expansion of the PCM during solidification was considered in the model.
135 Better predictions were obtained when this effect was taken into account. Moreno et al. (2020)
136 developed a transient 2D-CFD model to describe the airflow and the heat transfer in an air-
137 filled cavity with a PCM wall. The numerical results revealed flow patterns similar to ones
138 observed in experiment during melting process. The temperature stratification was intensified
139 after the liquid fraction of the PCM started to predominate ($> 50\%$). Orozco et al. (2021) further
140 performed a numerical study with the same cavity configuration but this time the PCM wall
141 was segmented into small volumes. The results showed that the segmentation of the PCM wall
142 had no significant effect on the flow patterns and the temperature distributions in the air-filled
143 cavity. Both flow topology and temperature stratification in the cavity with the segmented PCM

144 wall were similar to those in the cavity with the non-segmented PCM wall. However, for the
145 same requirement of thermal retention duration, the amount of the PCM could be reduced by
146 dividing the PCM container into segments. These mentioned studies essentially dedicated to
147 the use of PCM to enhance the energy efficiency of buildings. Only the temperature
148 measurements were used for the model validation (no air velocity measurement). To our best
149 knowledge, no experimental studies combining temperature and air velocity measurements
150 inside an empty and loaded insulated box with PCM have been conducted.

151 The present study attempted to bridge this research gap. The objective was to investigate
152 experimentally the airflow and heat transfer due to natural convection in an insulated box with
153 PCM. The influence of the PCM position and that of the presence of a load on flow patterns
154 and temperature distribution in the insulated box were examined in the study. A non-intrusive
155 technique, Particle Image Velocimetry (PIV), was implemented to characterize the air velocity
156 field. The originality of this study lies in a) the implementation of PIV measurement for low air
157 velocity and b) the analysis of both velocity and temperature fields in different configurations
158 of an insulated box with PCM under a real use condition of food transport i.e., all walls of the
159 box subjected to heat losses and the presence of loads in the box. The obtained knowledge
160 would be useful to suggest the optimal practices for handling insulated boxes used for food
161 transport. The experimental results obtained in this study enable deep knowledge in the
162 exchange phenomena. They will be used to develop CFD and simplified thermal models, which
163 will be presented in the future. These numerical approaches enable the prediction of product
164 temperature change with time at different positions in an insulated box exposed to variable
165 ambient temperatures as in a real shipment.

166 **2. Materials and methods**

167 **2.1 Experimental device**

168 Two insulated boxes were used. Box A was a commercially manufactured box which was used
169 for the thermal study (temperature measurement by thermocouples). Box B was the same box
170 in which two walls were replaced with triple-glazed windows ensuring almost the same
171 insulation, and it was used for the momentum study (air velocity measurement by a PIV
172 system). As shown in **Fig. 1a**, the wall structure of Box A is composed of an expanded
173 polystyrene foam layer (thickness: 25 mm, $\lambda = 0.029 \text{ W}\cdot\text{m}^{-1}\cdot\text{K}^{-1}$) sandwiched between two
174 polypropylene plastic layers (thickness: 3.5 mm, $\lambda = 0.12 \text{ W}\cdot\text{m}^{-1}\cdot\text{K}^{-1}$, $\varepsilon = 0.97$) (Cengel and
175 Ghajar, 2020). The inner gap between the polypropylene layers is 35 mm so that an air layer
176 (thickness: 5 mm, $\lambda = 0.025 \text{ W}\cdot\text{m}^{-1}\cdot\text{K}^{-1}$) is also present. The internal dimensions of the box
177 were 310 mm (W) \times 500 mm (L) \times 300 mm (H), corresponding to about 0.05 m³ in volume.
178 Box B had the same dimensions and wall structure, but two vertical walls were replaced by
179 triple-glazed windows. They were composed of three glass layers ($\lambda = 1.4 \text{ W}\cdot\text{m}^{-1}\cdot\text{K}^{-1}$) each
180 with a thickness of 4 mm and two 10-mm argon gaps ($\lambda = 0.018 \text{ W}\cdot\text{m}^{-1}\cdot\text{K}^{-1}$), as shown in **Fig.**
181 **1b**. In addition, the panes were coated with a low-emissivity material ($\varepsilon = 0.03$, manufacture
182 data) to avoid the transmission of infrared radiation. In this manner, the overall heat transfer
183 coefficient of the glass wall was almost identical to that of the unmodified commercially
184 manufactured box: $U_g \cong U_w = 0.9 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$. This makes it possible to consider that heat losses
185 occurring in Boxes A and B were almost the same; thus, the momentum results obtained
186 experimentally from Box B can be compared with the thermal results obtained from Box A.

187 Tap water filled in a polypropylene recipient (280 mm \times 460 mm \times 50 mm) was used as a PCM
188 for both thermal and momentum studies. The melting point and the latent heat of water is
189 respectively -0.2°C (measured data) and 334 kJ/kg (Cengel and Ghajar, 2020) while the other
190 thermophysical properties of water are given in **Table 1**. Although tap water was used, its
191 properties are close to the ones of pure water since the amount of solute corresponding to the
192 measured melting temperature (-0.2°C) is very low (around 0.05 mol/kg). The PCM was
193 prepared (solidification) by placing horizontally in a freezer (-2.0°C set point) for at least 48 h
194 before being used in each experiment to ensure that it was completely frozen at a temperature
195 close to the melting point. Depending on the experimental conditions, the PCM was placed
196 either on the top wall (lid) or on a vertical (side) wall of the box. (melting point of -0.2°C)

197 Two types of loads were used in the experiment: 4 inert blocks made of extruded polystyrene
198 (XPS, dimensions 200 mm × 400 mm × 50 mm) and 16 packs of test product made of tylose
199 (TYL, dimensions of a pack 200 mm × 100 mm × 50 mm). The presence of XPS blocks made
200 it possible to study the influence of obstruction (without heat exchange with air) on the airflow
201 pattern, while the presence of TYL packs made it possible to study the combined influence of
202 obstruction and heat exchange with air. The use of XPS blocks made it possible to reach a
203 thermal steady state (low thermal inertia) rapidly. Both types of loads were arranged so they
204 had the same stack dimensions: 200 mm × 400 mm × 200 mm. The TYL packs were previously
205 stored in a laboratory refrigerator and their initial core temperature was $5.0 \pm 1.0^\circ\text{C}$ before being
206 used in the experiment. The 1.0°C temperature variation is related to the position of the pack in
207 the refrigerator during preparation. It is to be emphasized that due to the large variety of food
208 products and their thermophysical properties, the test product employed in the standard tests of
209 refrigeration equipment was used. The properties of this test product (called Tylose) are close
210 to those of meat product as shown in **Table 1**.

211 To facilitate interpretation of the results, the boxes (for both the thermal and momentum studies)
212 were placed on small supports (with a height of 50 mm) placed on a table (with a height of 700
213 mm), thereby ensuring homogeneous airflow around the box (including underneath it). The
214 experimental device was placed in a test chamber (dimensions: 340 cm × 340 cm × 250 cm) in
215 which the ambient temperature was controlled at $20.0 \pm 1.0^\circ\text{C}$ throughout the experiments. The
216 humidity in the room was not controlled. The relative humidity measured by using a hygrometer
217 (Testo 174H, accuracy $\pm 3\%$ rh) lied in the range of 45-65%, corresponding to the humidity ratio
218 of 0.006–0.009 kg of water vapor/kg dry air.

219 **2.2 Thermal study**

220 Temperature measurements were performed in Box A using calibrated T-type thermocouples
221 (200 μm diameter, $\pm 0.2^\circ\text{C}$ accuracy) connected to a data logger (Agilent 34972A). The
222 protocols for the measurements under empty and loaded conditions are described hereafter.

223 *2.2.1 Experimental protocol under empty conditions*

224 The temperature measurements were conducted on a middle plane ($x = 250\text{ mm}$) of the box as
225 shown in **Fig. 2a**. Twelve thermocouples were installed on a portable stand at intervals varying
226 from 5 mm (near the top and bottom) to 35 mm (mid-height) and were used to measure the air
227 temperatures across the cavity. Another twelve thermocouples were fixed on the surfaces of the
228 box walls and the PCM in order to measure their surface temperatures. Three thermocouples
229 were also installed at three positions inside the PCM.

230 To establish the temperature profiles, several temperature measurements were carried out by
231 moving the stand from one position to another across the plane. The stand was positioned at a
232 distance of 5 mm from the walls for the initial measurement. The measurement began at least
233 90 min. after the box closure (steady state had been reached) and lasted for a duration of 5 min.
234 with recording intervals of 15 s. The box was re-opened in order to move the stand to the next
235 position and then was closed rapidly to limit the effects of external air ingress (total duration of
236 this operation: $< 1\text{ min}$). Then, temperature measurement was undertaken 15 min. after closure
237 (steady state had been reached again). To address the temperature profile in the boundary layer,
238 fine incremental steps of 5 mm were applied for the first five positions near the wall, then
239 coarser incremental steps (up to 50 mm) were used (eighteen y-positions). The temperature at
240 each position was averaged over 5 min. and the reported temperature profiles were based on
241 these average values.

242 It needs to be emphasized that the phase change process of the PCM is a transient phenomenon
243 and the real steady state never exists. The "pseudo" steady state stipulated in this study indicate

244 the duration in which the air temperatures at different positions in the cavity were relatively
245 constant. Preliminary experiments showed that "pseudo" steady state was reached after 90 min.
246 Then, the standard deviations over 3 h of the air temperatures at eight positions in the box never
247 exceeded 0.3°C.

248 **2.2.2 Experimental protocol under loaded conditions**

249 The stack of TYL packs was carefully placed in the center of the box, which corresponded to a
250 loading percentage (including the PCM) of almost 50% by volume. In this case, all air and load
251 temperatures (core and surface) were simultaneously measured throughout the experiment
252 without intermittent openings. **Fig. 2b** shows the positions of the thermocouples used to
253 measure the air temperatures on the middle ($x = 250$ mm) and the lateral ($x = 15$ mm) planes of
254 the box as well as the core and surface temperatures of four TYL packs. Thermocouples were
255 also installed at three positions inside the PCM. Measurements were undertaken 90 min. after
256 the closure of the box and lasted until the PCM was completely melted (i.e. all measured PCM
257 temperatures started to increase). A recording interval of 30 s was set for the experiment. The
258 air and product (core and surface) temperatures at each point were averaged over 200 min.
259 during which the PCM was melting. These time-averaged values were used to establish the
260 temperature field in the loaded box.

261 **2.3 Momentum study**

262 **Fig. 3** shows the overall view of experimental setup for the air velocity measurements in the
263 insulated box (Box B) using a PIV system which requires tracer particles (oil smoke in our
264 case) for the measurement. The box was connected to a smoke container using a flexible duct
265 in which four small PCM packs were used to precool the smoke before entering the box. A
266 small fan was used to assist the introduction of precooled smoke ($\sim 10^\circ\text{C}$) into the box and its
267 flow rate was controlled by a valve on the connecting duct.

268 **2.3.1 PIV system**

269 A 2D-PIV system (LaVision, FlowMaster 2D) was used to visualize the flow pattern and to
270 measure air velocity in the box. The system is composed of three main components: a double-
271 pulsed Nd:YLF laser (527 nm wavelength, 10 mJ pulse energy), a high-speed 12-bit CMOS
272 video camera (Photron, FASTCAM SA3; 1024 × 1024 pixels in resolution) mounted with a lens
273 (Sigma; 105 mm, f/1:2.8), and a programmable timing unit (PTU-X) for the synchronization of
274 the device. For light scattering, a smoke machine (Antari, F-80Z) was used to generate oil
275 particles (Levenly, Smoke Standard; mean diameter of 0.3 μm). Image acquisition and post-
276 processing were performed using DaVis 10.2 interfaced software. The camera and the laser
277 were installed on a three-dimensional displacement system (displacement precision ± 1 mm)
278 and aligned in such a manner that the field of the camera view was perpendicular to the light
279 sheet (thickness of 1 mm).

280 **2.3.2 Image acquisition**

281 The PIV measurements under both unloaded and loaded conditions were performed on the same
282 plane as the temperature measurement under loaded conditions: the middle plane ($x = 250$ mm)
283 and the lateral plane ($x = 15$ mm). Based on the image calibration using a ruler and the DaVis
284 software, the magnification factor of 0.113 mm/pixel was determined and it corresponded to an
285 image size of approximately 115 mm × 115 mm. For each measurement plane, several measured
286 windows with a partial overlap between them were used to cover the entire area of the plane.
287 **Fig. 4** shows the position and its corresponding number of measured windows on the
288 measurement plane for different experimental conditions (unloaded/loaded and PCM on the
289 side/lid). The position of the measured windows was changed by using a displacement system.
290 For each measured window, 500 pairs of images were recorded every 20 ms with a time interval
291 (Δt) between two paired images (two pulsed laser illuminations) of 900 μs, and the total

292 measurement duration was 10 s. Based on the preliminary experiment, the measurement
 293 duration should not exceed 10 s to avoid heat generation by the laser which caused an increase
 294 in wall temperature, thereby affecting airflow. This time interval was considered as an optimal
 295 value for a velocity estimation in our case, because it allowed a mean particle displacement of
 296 less than a quarter of the smallest width of interrogation window (Keane and Adrian, 1992).

297 **2.3.3 Image post-processing**

298 A multi-pass correlation algorithm was used to process instantaneous vector calculation. The
 299 cross-correlation between individual paired images was performed with decreasing
 300 interrogation window sizes: 64×64 pixels with 50% overlap for the first passes and 32×32
 301 pixels with 75% overlap for the final passes. Given the interrogation dimensions of the final
 302 pass, the spatial resolution of the vector field (distance between two vectors) was 8 pixels (about
 303 0.9 mm) in both vertical and horizontal directions. After 500 instantaneous vector fields were
 304 obtained, the mean velocity field (v : 2D velocity magnitude) was then calculated as follows

$$305 \quad v = \frac{1}{N} \sum_{i=1}^N \sqrt{v_{y,i}^2 + v_{z,i}^2} \quad (1)$$

306 where N is the total number of measured windows ($N = 500$ in our case), v_y and v_z are the
 307 horizontal and vertical velocity components expressed in $\text{m} \cdot \text{s}^{-1}$, respectively.

308 The mean velocity fields of all measured windows were then connected to establish the velocity
 309 field of the entire measurement plane. It should be emphasized that the out-of-plane regions
 310 and the regions near high reflection surfaces in the images (e.g. the surfaces of PCM, wall, and
 311 load) were deleted and excluded prior to the vector calculation.

312 The DaVis software uses the correlation statistics method (a-posteriori approach) to quantify
 313 the uncertainty of the PIV measurement. Based on the statistical analysis (Wieneke, 2015), the

314 uncertainty of the mean air velocity reported in this study was less than 5% for $v > 0.04 \text{ m}\cdot\text{s}^{-1}$
315 and less than 10% for $0.02 < v < 0.04 \text{ m}\cdot\text{s}^{-1}$.

316 **2.3.4 Experimental protocol**

317 Air velocity measurements were carried out under steady state conditions which were achieved
318 90 min. after the PCM was introduced into the box. At steady state, the precooled smoke was
319 introduced into the box until its concentration was sufficient. To ensure flow stabilization, the
320 PIV measurement was performed about 30 min. after the smoke introduction. Nine and five
321 measured windows of 500 paired images were captured for the experiments under empty and
322 loaded conditions, respectively. **Table 2** summarizes all experimental conditions for the
323 temperature and the velocity measurements.

324 **3. Results and discussion**

325 **3.1 Air velocity and temperature profiles under the empty condition**

326 The experiment was firstly performed under empty conditions to gain an understanding of
327 underlying momentum and energy transport phenomena inside the insulated box equipped with
328 PCM. The influence of PCM positions (side wall and lid) on these phenomena were investigated
329 and the results are presented as follows.

330 **3.1.1 PCM on a side wall of the box**

331 When the frozen PCM was placed on a side wall of the box, the apparent width (W') of the box
332 was reduced to 260 mm and accordingly the aspect ratio ($A = H/W'$) of the box was about
333 1.15. Based on the temperature difference between the inner surfaces of the PCM and the walls
334 ($\Delta T = T_h - T_c = 7.9^\circ\text{C}$), the Rayleigh (Ra) number based on the box height as defined in **Eq.**

335 **2** was approximately 2.8×10^7 . Thus, the air flow in the box was in the laminar regime ($< 10^9$)
 336 (Saury et al., 2011).

$$337 \quad Ra = Gr \cdot Pr = \frac{g\beta(T_h - T_c)H^3}{\nu^2} \cdot \frac{c_p \mu}{\lambda} \quad (2)$$

338 The thermal properties of air were calculated at the mean air temperature in the box ($T_{mean} =$
 339 7.7°C) by using the correlations proposed by McQuillan et al. (1984). The calculated values are
 340 summarized in **Table 3**.

341 **Fig. 5** shows the air velocity profiles on the middle ($x = 250$ mm) and the lateral ($x = 15$ mm)
 342 planes of the box equipped with the PCM on the side wall. For comparison purposes, the color
 343 scale was limited to $0.13 \text{ m}\cdot\text{s}^{-1}$ over which the maximum value recorded among all experimental
 344 conditions never exceeded. Large (red) arrows were drawn over the velocity field in order to
 345 provide better visualization of flow patterns represented by velocity vectors originally
 346 generated from the PIV software.

347 As shown in **Fig. 5a**, the airflow on the middle plane exhibits a flow pattern similar to the
 348 simple case, largely documented in the literature, of a rectangular cavity with opposite
 349 isothermally hot and cold vertical walls and well-insulated horizontal walls (Lee and Lin, 1995):
 350 upward and downward flow streams adjacent to the vertical surfaces of the side wall and the
 351 PCM, respectively. These flow streams moved horizontally along the bottom and the top walls,
 352 respectively, thus forming a recirculation cell. However, due to the technical limitations of the
 353 experimental device, the visible height was limited to $z = 260$ mm; thus, the complete flow
 354 recirculating cell could not be visualized in our study (dashed-line arrow in **Fig. 5a**).

355 A secondary flow was also detected (see arrow A on **Fig. 5a**): the air at a distance of about 30
 356 mm from the box wall changes its direction from upward to downward and flows toward the
 357 cold wall. This could result from the heat transfer through the bottom wall of the box. The

358 airflow near the ‘warm’ bottom wall has a slightly ascending slope (this is not the case for an
359 adiabatic bottom wall). The air located about 30 mm from the walls is entrained by the main
360 recirculation flow, but it is relatively colder than the air nearer to the wall; thus, at a given point,
361 its trajectory leaves the main cell and becomes downwards.

362 As illustrated in **Fig. 5a'**, the vertical velocity component (v_z) near the box wall tended to
363 increase from bottom to mid-height ($z = 120$ mm) reaching almost $0.09 \text{ m}\cdot\text{s}^{-1}$. Beyond this
364 height ($z > 120$ mm), the velocity started to decrease. A similar trend was observed near the
365 PCM, but in the opposite manner. The vertical velocity component ($|v_z|$) steadily increased
366 from the top until roughly $z = 90$ mm reaching almost $0.08 \text{ m}\cdot\text{s}^{-1}$, then the velocity began to
367 decrease, conceivably because the flow ‘turned’ at the corner. The flow pattern on the lateral
368 plane (**Fig. 5b**) was almost identical to that on the middle plane, but the maximal velocity
369 magnitude was higher ($0.12 \text{ m}\cdot\text{s}^{-1}$). Note that the velocity field on the lateral plane ($x = 15$ mm)
370 was extended to the region of the PCM because the PCM was symmetrically placed in the box,
371 thereby allowing the PIV measurement in the gap between the lateral wall and the PCM ($x >$
372 20 mm).

373 On both planes, there is a zone of stagnant air in the core region of the box. However, this
374 region on the lateral plane was smaller than that on the middle plane. Three-dimensional flow
375 due to additional heat gain through the side wall could explain this difference.

376 **Fig. 6a** shows the air temperature field on the middle plane ($x = 250$ mm) of the box with PCM
377 on the side wall. This temperature field was plotted by interpolation from 252 measurement
378 points over this plane by using MATLAB. The temperature field shows the thermal boundary
379 layers. Due to heat conduction from the exterior, the air temperature increases constantly while
380 it flows along the box walls, and the boundary layer thickness increases. Then, the air is cooled
381 down along the PCM. This is coherent with the flow pattern (recirculation cell) observed by the

382 PIV measurement. Overall stratification was observed: colder air near the bottom, and warmer
383 air near the top. The maximum air temperature was about 10°C at the top corner on the side
384 wall and the minimum air temperature was observed at the bottom corner on the PCM side. The
385 surface temperature of the PCM container varied from 0.5°C at the bottom ($z = 20\text{ mm}$) to
386 around 3.5°C at the top ($z = 230\text{ mm}$). Thermal boundary layers at different heights are
387 illustrated in **Fig. 6b**. As expected, their thickness was close to hydrodynamic thicknesses (**Fig.**
388 **5a'**) since the Prandtl number is relatively close to 1 ($Pr = 0.71$).

389 **3.1.2 PCM on the lid of the box**

390 The apparent height was $H' = 250\text{ mm}$; thus, the aspect ratio of the box with PCM on the lid
391 ($A = H'/W$) was about 0.81 and the Ra number was 2.4×10^7 ($\Delta T = 6.8^{\circ}\text{C}$). As in the case of
392 PCM on the side wall, the airflow was laminar. The values of thermal properties of air are
393 summarized in **Table 3** ($T_{mean} = 6.2^{\circ}\text{C}$).

394 **Fig. 7** shows the air velocity profiles on the middle ($x = 250\text{ mm}$) and the lateral ($x = 15\text{ mm}$)
395 planes of the box with PCM on the lid. It was found from **Fig. 7a** that there were two almost
396 symmetric, counterrotating air-flow cells. This result qualitatively agrees with the numerical
397 solution obtained by Corcione (2003) who also observed such a two-cell flow pattern in a
398 rectangular cavity with one cold top wall, one hot bottom wall and two hot side walls ($A = 0.5$
399 and $Ra = 10^6$ in his study). As shown in **Fig. 7a'**, the positive vertical velocity components
400 were detected along the side walls ($y < 50\text{ mm}$ and $y > 260\text{ mm}$) while the negative ones were
401 mostly in the core region. On the middle plane, the absolute values of the vertical velocity
402 components never exceeded $0.08\text{ m}\cdot\text{s}^{-1}$. The upward flow along the side walls was induced by
403 the relatively high air temperature in these regions as a result of heat conduction through the
404 box walls. Accordingly, these flow streams converged on the top where air was cooled down
405 via the PCM. Becoming heavier, the air then flowed downward in the center region.

406 Near the lateral wall ($x = 15$ mm, **Fig. 7b and b'**), the flow is almost everywhere upwards as is
407 the case near the side wall ($v_z > 0$ for $y = 15$ mm in **Fig. 7a'**). In fact, one would expect a similar
408 2D cell flow pattern in x-z plane ($y = W/2$) as that observed in the y-z plane ($x = L/2$, **Fig.**
409 **7a**). Overall, air flows upwards along the lateral and side walls, whereas it flows downward in
410 the central region (3D flow). Near the corners (junction of lateral and side walls e.g. $x = 15$ mm,
411 $y = 15$ mm) the heat flow by conduction through the walls is the highest, and this explains why
412 the vertical velocity is also the highest in these positions: $0.12 \text{ m}\cdot\text{s}^{-1}$.

413 It should be borne in mind that the presented velocity fields are time-averaged over 10 s and
414 are composed of 9 windows recorded at different times (typically at 2 min. intervals taking into
415 account the time needed to move the camera and save the recorded data). Direct observation of
416 smoke in the middle plane showed that the flow was not stable (it was unsteady) in the central
417 region: the downward flow oscillated in the y direction. This explains why the velocity observed
418 in **Fig. 7a** is not strictly symmetric and that there are some 'jumps' between the 3 parts (3
419 windows in the y direction) of the profiles in **Fig. 7a'**. This type of instability has been observed
420 also for free convection in domestic refrigerators (Laguerre et al., 2005).

421 **Fig. 8** shows the air temperature field on the middle plane of the box. As expected, the cold
422 region was in the center where downward flow was observed, while the warm region was near
423 the side walls where upward flow was observed. The maximum air temperature was about 9°C
424 near the top of the side walls and the minimum air temperature was observed just below the
425 PCM (top of central region); the surface of the PCM container was at a temperature of around
426 $1\text{-}2^\circ\text{C}$.

427 *3.1.3 Comparison between PCM on the lid or on the side*

428 In comparison with the case of PCM on the side wall, the box with PCM on the lid exhibited a
429 lower maximal temperature: 9.3°C (lid)/ 10.5°C (side). This high temperature was observed near

430 the walls. In fact, the product should not touch the walls and should even be placed outside the
 431 boundary layers whose thickness was around 30 mm. If we exclude the boundary layer zone,
 432 the mean temperature (in the middle plane) was lower for PCM on the lid: $T_{mean} = 6.2^{\circ}\text{C}$ (lid)
 433 and 7.7°C (side) and the temperature distribution was also more homogeneous for PCM on the
 434 lid: $T_{max} - T_{min} = 1.7^{\circ}\text{C}$ (lid) and 2.7°C (side).

435 As mentioned previously, flow fluctuations were visually observed in the central region where
 436 the PCM was placed on the lid. To a lesser extent, fluctuations were also observed near the
 437 bottom of the PCM when it was placed on the (right) side. **Fig. 9** presents the instantaneous
 438 velocity evolution during 10 s at two positions (near the center and near the bottom/right
 439 corner). This confirms that the airflow was more stable in the case of PCM on the side. Velocity
 440 variations of up to $0.10 \text{ m}\cdot\text{s}^{-1}$ were detected at the center of the box with PCM on the lid (**Fig.**
 441 **9b**).

442 The convective heat transfer coefficients can be estimated from the measured temperature
 443 profiles. Very near to the wall, the air velocity is close to zero. So, the heat flux ($\text{W}\cdot\text{m}^{-2}$) along
 444 y-direction can be given by

$$445 \quad \lambda_a \frac{\partial T}{\partial y} = h_z (T_w - T_{\infty}) \quad (3)$$

446 where T_w and T_{∞} are the temperatures of the wall and the air outside the boundary layer (free
 447 stream), respectively, λ_a is the thermal conductivity of the air, and h_z is the local convective
 448 heat transfer coefficient at a given height (z) which can be approximately estimated from

$$449 \quad h_z = \frac{\lambda_a (\partial T / \partial y)}{T_w - T_{\infty}} \quad (4)$$

450 For example, when the PCM was at the side wall, at the mid height ($z = 160 \text{ mm}$), the
 451 temperature at the PCM wall, at 5 mm from the wall and outside the boundary layer were 0.8°C ,
 452 5.5°C and 7.2°C , respectively. The slopes ($\partial T / \partial y$) of the tangent line to the temperature profile
 453 near the PCM wall was thus approximately $1^{\circ}\text{C}/\text{mm}$. Accordingly, the local convective heat
 454 transfer coefficient at PCM wall could be estimated around $4 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$. In the same way, the

455 heat transfer coefficient at the vertical internal box walls (warm walls) could be estimated
 456 between 2 and 3 $\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$. It is to emphasize that the temperature fields were measured only
 457 during the 'pseudo' steady state period during which the air temperature was almost invariant
 458 with time. Therefore, the estimation of h_z was given for this period only.

459 Despite low heat transfer coefficients, convection cannot be neglected compared to conduction
 460 in air because the maximum air velocities observed in the box were around $0.1 \text{ m}\cdot\text{s}^{-1}$,
 461 corresponding to the Peclet number (Pe) of more than 100.

462 The Peclet number (Pe) is defined as

$$463 \quad Pe = \frac{v_a L_c}{\alpha_a} \quad (5)$$

464 For the empty box, the value was approximately 1500, given that the characteristic length (L_c)
 465 was 0.3 m (height of the box), the air velocity (v_a) was $0.1 \text{ m}\cdot\text{s}^{-1}$, and the air thermal diffusivity
 466 (α_a) was about $2\times 10^{-5} \text{ m}^2\cdot\text{s}^{-1}$.

467 *3.2 Air velocity and temperature profiles under loaded conditions*

468 The experiments were performed under loaded conditions. The experiment was firstly
 469 conducted with inert blocks (XPS) that made it possible to study the obstacle-effect alone on
 470 the flow pattern. Then, an experiment with the test packs (TYL) was performed to study both
 471 the obstacle-effect and the influence of heat exchange with air. Due to the presence of the
 472 obstacles, the light sheet was restricted; thus, the velocity field behind the load was not
 473 available.

474 *3.2.1 PCM on the side wall of the box*

475 As shown in **Fig. 10**, the airflow in the loaded box, regardless of obstacle types, exhibited
 476 similarities with that in the empty box: upward and downward flow streams close to the vertical

477 surfaces of the side wall and the PCM, respectively. However, under loaded conditions, the
478 upward flow on the middle plane did not result from a two-dimensional recirculation cell
479 (located in this plane) but instead from a three-dimensional flow pattern as illustrated in **Fig.**
480 **11**. At State 1, the air flowed downward (-z direction) in the space between the PCM and the
481 load. Once approaching the bottom wall (State 2), the air flowed rather horizontally (+/- x
482 direction) toward the lateral walls of the box. At the edge of the load (State 3), the air turned
483 and flowed between the lateral wall and the load (-y direction). This is confirmed in **Figs. 10b**
484 and **10d** (lateral plane, $x = 15$ mm) where a strong flow from the right to the left was observed
485 in the bottom region. When it reached the bottom of the side wall (State 4), the air moved both
486 horizontally to occupy the entire gap between the side wall and the load and upwards because
487 it became warmer and warmer (heat exchange with the walls). Finally, starting from State 5, it
488 recirculated to the PCM.

489 The velocity fields obtained with inert blocks (XPS) and with test products (TYL) were very
490 similar. This is because thermal steady state was practically reached in both cases (the test
491 products were introduced practically at the equilibrium temperature and the measurement began
492 after 2 h). In this manner, thermal inertia became negligible. For XPS (with very low thermal
493 conductivity) the air temperature was expected to be relatively homogeneous in the gaps
494 between the load and the lateral wall or the PCM. For TYL, due to conduction, the load surface
495 temperature was lower than the air temperature in the wall-side gap but higher in the PCM-side
496 gap (as shown hereafter in **Fig. 12a**). This could diminish free convection, but the results
497 showed a minor effect ($Pe \sim 150$, L_c (gap) = 30 mm). This means that (steady state) flow
498 characterization can be carried out with inert blocks (XPS), which is much simpler. Certainly,
499 if warm products were introduced in the box initially, the flow pattern would have been
500 substantially altered.

501 **Fig. 12** shows the temperature field on the middle ($x = 250$ mm) and the lateral ($x = 15$ mm)
502 planes of the box with PCM on the side wall and TYL load. The temperature field was coherent
503 with the flow pattern: air was cooled down along the PCM and warmed up along all the box
504 walls. It is obvious that the cold air near the PCM resulted in relatively low load temperatures
505 on this (right) side. Conversely, the warmer air near the wall resulted in relatively high load
506 temperatures on the opposite (left) side. Conduction in the load was not sufficient to
507 homogenize the load temperature. The highest load temperature, 7.8°C , was reached near the
508 side wall (opposite the PCM location). The average load temperature was 6.0°C and the
509 maximum difference was 3.9°C . Air temperature stratification was observed on the lateral plane
510 of the box (**Fig. 12b**).

511 **3.2.2 PCM on the lid of the box**

512 A complex flow pattern was observed in the loaded box with PCM on the lid as shown in **Fig.**
513 **13**. On the middle plane ($x = 250$ mm, **Figs. 13a** and **13c**), regardless of load types, cold air
514 coming from the top (near the PCM) flowed downward in the center of the (left) gap between
515 the side wall and the load. Then, air flowed upwards along the side wall and the load surface.
516 An explanation for the upward flow along the side wall is that the air near this wall is warmed
517 up by conduction through the wall (as in the empty case). The flow direction along the load
518 surface is logically upwards if the surface temperature is higher than the average air temperature
519 in the gap. This was the case for the XPS load because radiation from the wall to the load
520 surface tended to increase the load surface temperature. For the TYL, in addition to radiation,
521 conduction also occurred inside the load. This could explain why upward flow occurred all
522 along the XPS load but occurred only along the upper part of the TYL load. This difference
523 appeared also on the lateral plane ($x = 15$ mm): there were more upward flow regions in the

524 case of the XPS load. However, the easier experiments with the XPS load gave a good
525 approximation of the flow pattern in the presence of load in the box.

526 Due to the limitations of the PIV technique, velocity measurement was not possible in the (right)
527 gap. A similar flow pattern could be expected due to symmetry. However, in fact, flow was not
528 symmetrical above the load: it seemed that a stronger cold air stream flowed down in the right
529 gap than in the left gap. Such dissymmetry was already observed in the empty case (PCM on
530 the lid) where instability was induced by oscillations of downward flow. It seemed that in the
531 loaded case, the flow 'chose' one or another preferential pathway (through the left or right gap).
532 This is related to the non-linear term in the flow equations (Navier-Stokes) which is responsible
533 for a break in symmetry (even before turbulence). This dissymmetry was also confirmed by
534 **Figs. 13b** and **13d** (lateral plane, $x = 15$ mm) where flow from the right to the left was observed
535 in the lower part of the box. The small difference in heat transfer coefficient between the left
536 (insulated wall) and the right (triple-glazed windows) can also induce dissymmetry.

537 **Fig. 14** presents the temperature field on the middle ($x = 250$ mm) and the lateral ($x = 15$ mm)
538 planes of the box with PCM on the lid and TYL load. It shows the thermal boundary layers
539 along the side wall (which explains upward flow along the walls). However, the load surface
540 temperature seemed very close to the adjacent air temperature. At the center of the gap (where
541 downward flow was observed) the air rapidly warmed up (the temperature was well above the
542 PCM surface temperature). This can explain the weak velocity observed in the gap.
543 Dissymmetry was also observed: the right gap had a lower average air temperature than the left
544 gap which in turn affected the load temperature. This also seemed to be due to a stronger cold
545 air stream from the PCM toward the right gap (see the red arrow in **Fig. 14**). For temperature
546 measurements, the dissymmetry cannot be imputed to a difference in insulation (Box A was

547 used). This configuration did not allow stratification since the cold PCM container was placed
548 at the top.

549 In the case of PCM on the lid, the highest load temperature, 7.3°C (7.8°C for PCM on the side)
550 was reached near the bottom (opposite the PCM location). The average load temperature was
551 5.7°C (6.0°C for PCM on the side), and the maximum difference was 3.0°C (3.9°C for PCM
552 on the side). So, there was little apparent difference between the two configurations. These
553 findings suggested that the PCM can be placed either at the side wall or at the lid without
554 compromising the quality and safety of food products if spaces between the PCM and the load
555 and between the side walls and the load are reserved. According to our estimation, the insulated
556 box with ice as a PCM is feasible for the transport of food products in the temperature range of
557 $4\text{--}8^{\circ}\text{C}$ for about 10 h. The experimental results obtained by this study will be used for the
558 development of CFD and simplified thermal models to predict product temperature evolution
559 along a logistic chain. This evolution makes possible the prediction of product quality thanks
560 to a relation with the product temperature. In this manner, the product shelf life under different
561 logistic scenarios can be predicted. The thermal and quality modelling would help the supply
562 chain management to optimize the logistic conditions to reduce food loss and waste.

563 **4. Conclusions**

564 The present study was carried out to characterize the airflow and the heat transfer due to natural
565 convection in an insulated box equipped with PCM by using PIV technique and temperature
566 measurement. The influence of the PCM position on the flow pattern and temperature
567 distribution was investigated. The study was conducted in a progressively more complex
568 manner: empty, loaded with extruded polystyrene (low conductivity and almost no thermal
569 inertia), and loaded with tylose (thermal properties close to those of food). The key findings are
570 summarized as follows:

571 Whatever the configuration, the highest observed air velocities were around $0.1 \text{ m}\cdot\text{s}^{-1}$; therefore,
572 convection cannot be neglected compared to conduction in air (Peclet number > 100).
573 Numerical simulations with either CFD or simplified models should include free convection.

574 When the PCM is on the side wall, the flow pattern is simple to predict. Air flows downwards
575 along the PCM surface and upwards along the side walls. In the empty case, the flow pattern
576 can be approximated by a 2D recirculation cell, but the presence of the load leads to a 3D flow
577 pattern. When the PCM is on the lid, after cooling down in contact with the PCM, a cold air
578 stream detaches from the PCM surface and flows downwards. This cold air stream is unstable
579 in the empty case and shows preferential pathways (symmetry breaking) in the loaded case. The
580 flow pattern is less predictable.

581 In all cases, after cooling down near the PCM, the air temperature increases progressively along
582 the trajectories until returning close to the PCM. The product surface temperatures are close to
583 the temperature of the adjacent air. At steady state, conduction in the load has a minor effect on
584 the flow pattern which can be approximated by replacing the real load with an obstacle of low
585 conductivity.

586 From a practical point of view, it is recommended to leave a space between the PCM and the
587 load (to promote free convection) and between the side walls and the load (to allow evacuation
588 of heat from the ambient via conduction through the walls). The gap should be at least of the
589 order of the boundary layer thicknesses: 2-3 cm. Further experiments without such gaps would
590 be useful.

591 At thermal steady state, there was no significant difference in terms of maximum product
592 temperature and heterogeneity between the PCM on the lid or at the side. The study showed the
593 coldest and warmest zones in both cases, suggesting the best location for products that are
594 sensitive to bacterial growth or chilling injury.

595 Further studies are planned in order to compare these results with CFD simulations and to
 596 develop a simplified model that enables prediction of temperature evolution (at different
 597 locations) as a function of the box, the product and PCM properties, along with ambient
 598 temperature changes.

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Table 1 Thermophysical properties of materials used in the study

Material	Density [kg·m ⁻³]	Specific heat [J·kg ⁻¹ ·°C ⁻¹]	Thermal conductivity [W·m ⁻¹ ·K ⁻¹]	Reference
Extruded polystyrene	35	1210	0.029	Cengel and Ghajar (2020)
Polypropylene	910	1925	0.120	Cengel and Ghajar (2020)
Tylose	1070	3372	0.510	Icier and Ilicali (2005)
Water (liquid)	1000	4217	0.561	Cengel and Ghajar (2020)
Water (solid)	920	2040	1.880	Cengel and Ghajar (2020)

Table 2 Experimental conditions for thermal (temperature measurement) and momentum (air velocity measurement) studies.

Conditions	PCM position	Temperature measurement	Numbers of windows for air velocity measurement [window number]	
			Middle plane	Lateral plane
Unloaded	Side wall	Yes	9 [1-9]	9 [1-9]
	Lid	Yes	9 [1-9]	9 [1-9]
Loaded (XPS)	Side wall	No	5 [1-5]	9 [1-9]
	Lid	No	5 [1-5]	9 [1-9]
Loaded (TYL)	Side wall	Yes	5 [1-5]	9 [1-9]
	Lid	Yes	5 [1-5]	9 [1-9]

XPS is a stack of four extruded polystyrene blocks (block dimensions = 200 mm × 400 mm × 50 mm); TYL is a stack of 16 Tylose packs (pack dimensions = 200 mm × 100 mm × 50 mm); the window number is referred to that in **Fig. 4**.

Table 3 Thermophysical properties* of air used for the Ra estimation.

Parameter	Unit	Value	
		$T_a = 6.2^\circ\text{C}$	$T_a = 7.7^\circ\text{C}$
ρ	kg·m ⁻³	1.264	1.258
ν	m ² ·s ⁻¹	1.387×10^{-5}	1.400×10^{-5}
c_p	J·kg ⁻¹ ·°C ⁻¹	1006	1006
λ	W·m ⁻¹ ·K ⁻¹	0.0245	0.0247
β	K ⁻¹	0.0036	0.0036

*calculated at the average air temperature (T_a) from the correlations proposed by McQuillan et al. (1984)

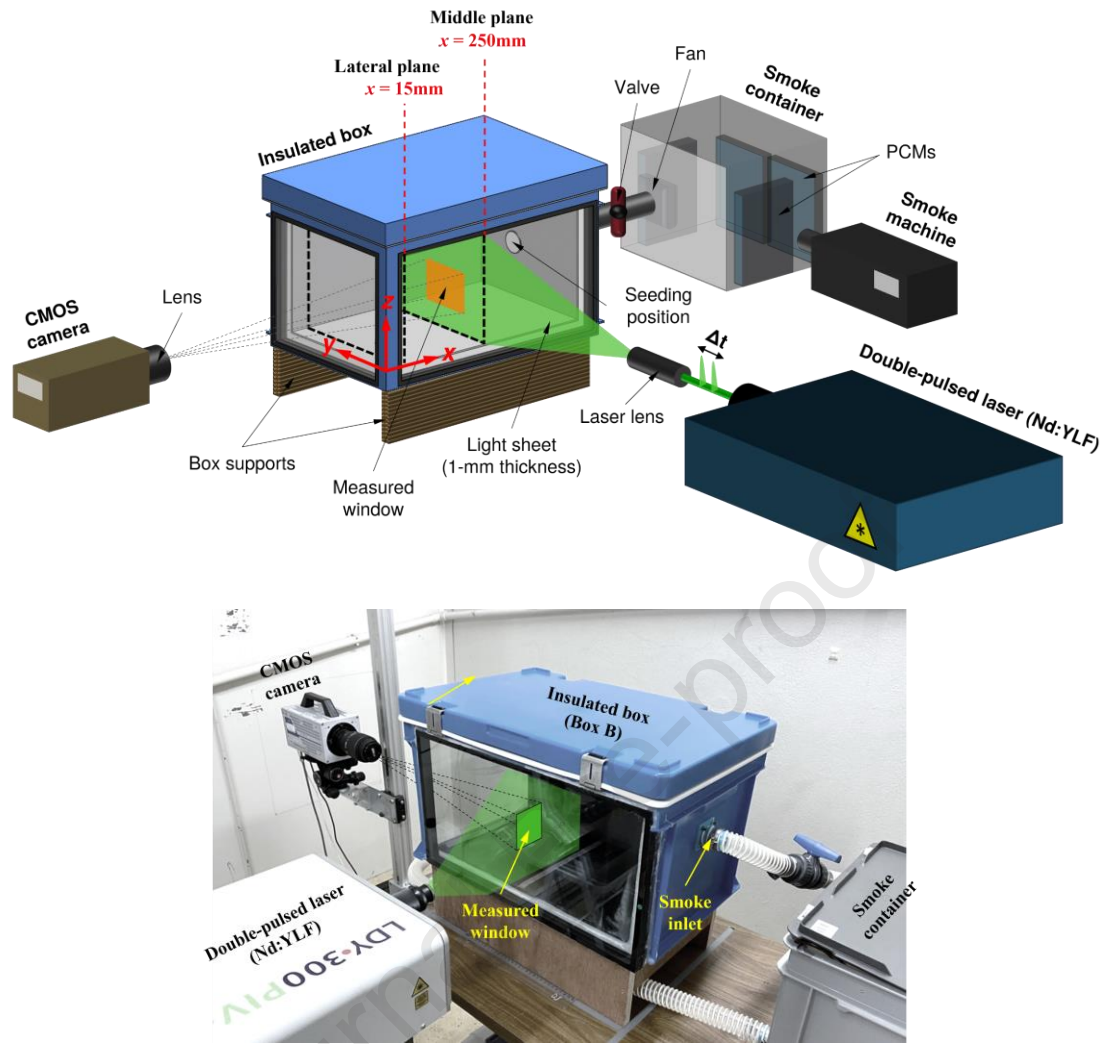


Fig. 3 Diagram (a) and photograph (b) showing the PIV setup.

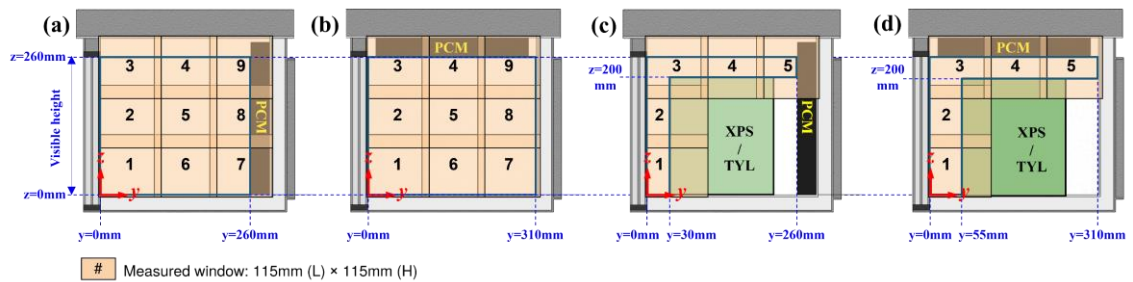


Fig. 4 Position of measured windows for the PIV measurement: (a) empty box/PCM on the side wall, (b) empty box/PCM on the lid, (c) loaded box/PCM on the side wall, and (d) loaded box/PCM on the lid.

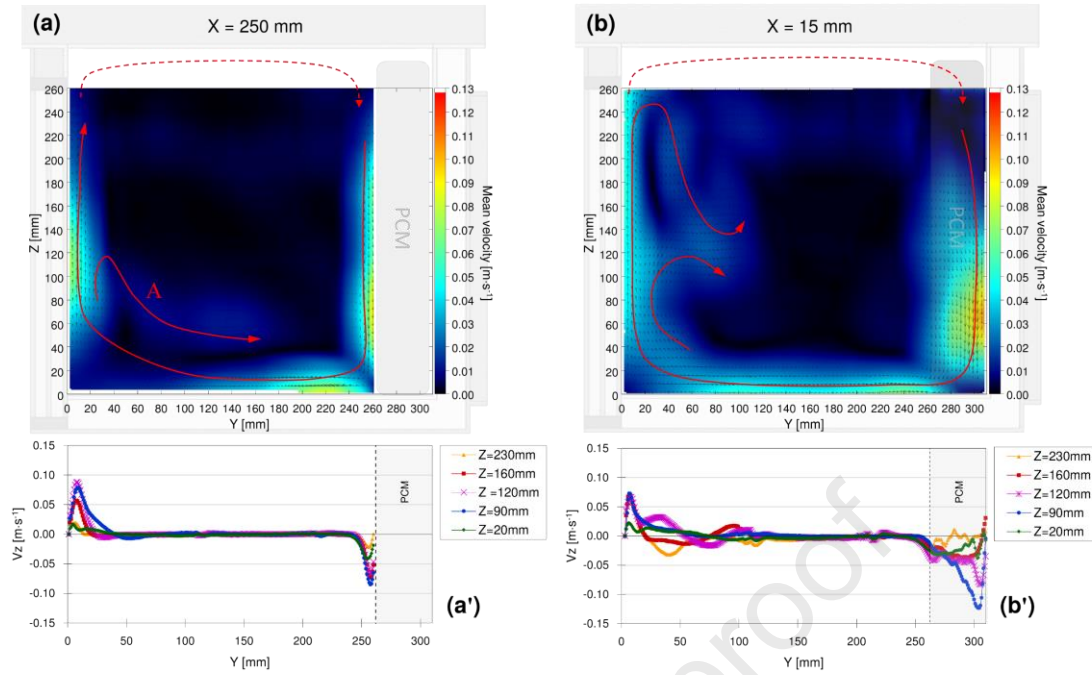


Fig. 5 Air velocity fields on (a) the middle ($x = 250$ mm) and (b) the lateral ($x = 15$ mm) planes of the box with PCM on the side wall. (a') and (b') are the profiles of the vertical velocity component (v_z) at 4 heights on the middle and the lateral planes, respectively.

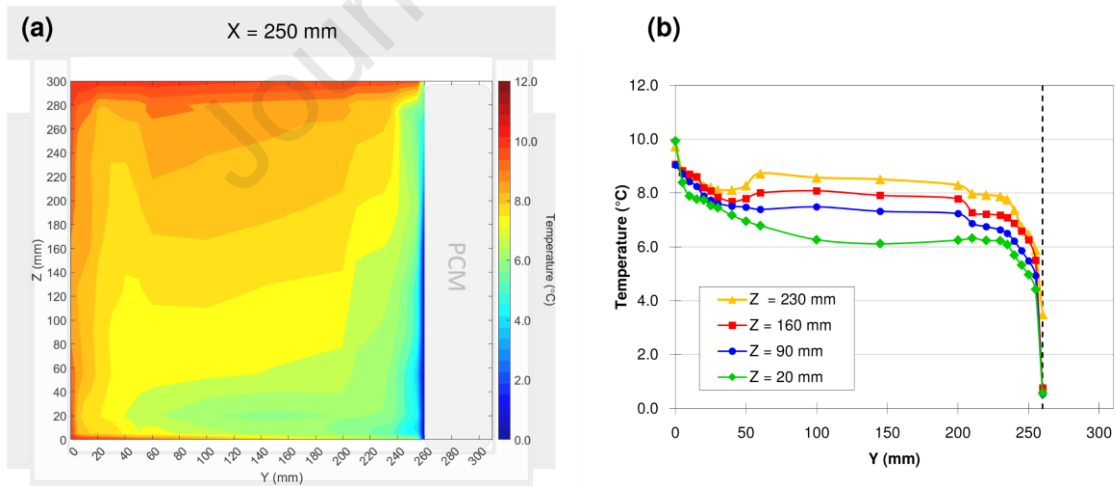


Fig. 6 (a) Air temperature field on the middle plane ($x = 250$ mm) of the box with PCM on the side wall and (b) temperature profiles at four different heights.

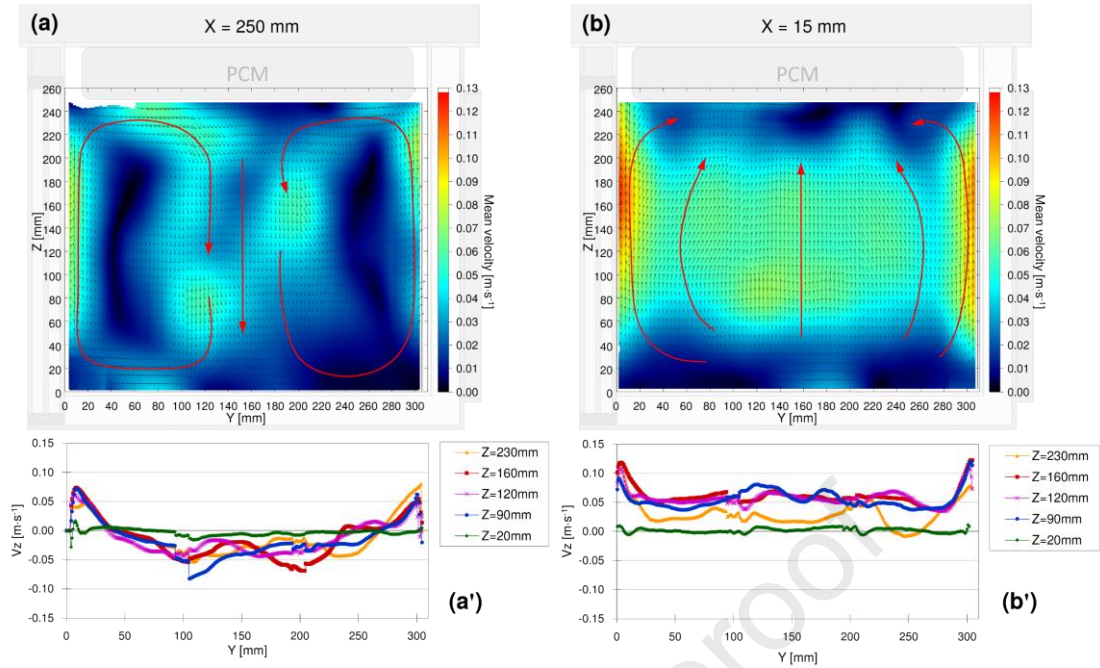


Fig. 7 Air velocity fields at (a) the middle ($x = 250$ mm) and (b) the lateral ($x = 25$ mm) planes of the box equipped with PCM on the lid. The profiles of the z-component of the velocity at 4 heights on (a') the middle and (b') the lateral planes.

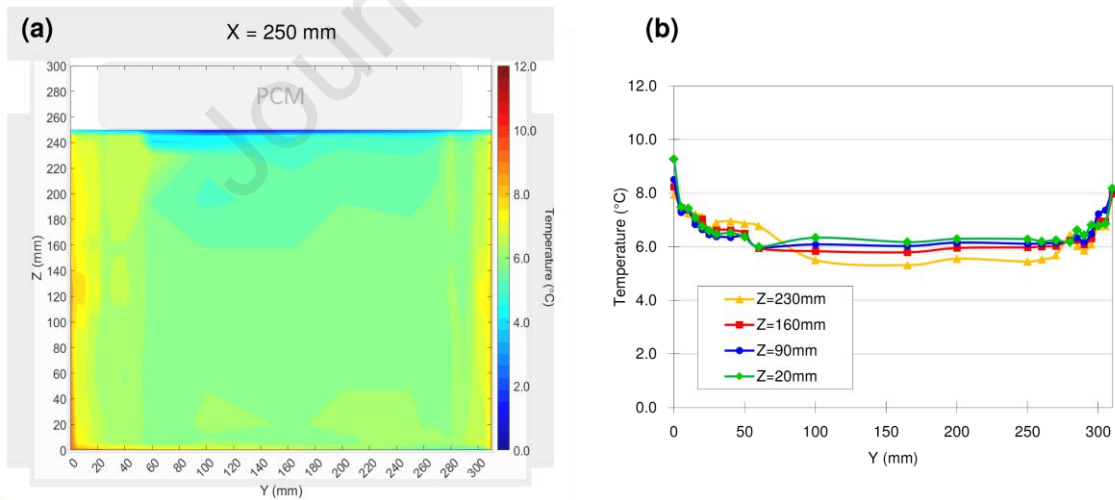


Fig. 8 Air temperature field on the middle plane ($x = 250$ mm) of the box with PCM on the lid.

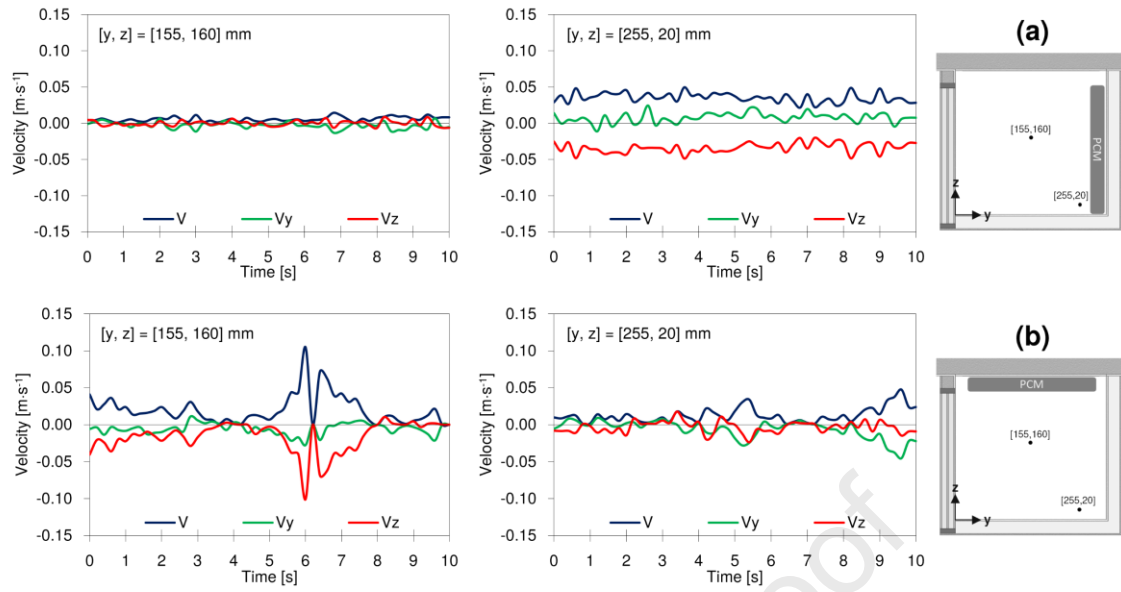


Fig. 9 Velocity variations of the air at the same two positions in the box with PCM on (a) the side wall and (b) the lid.

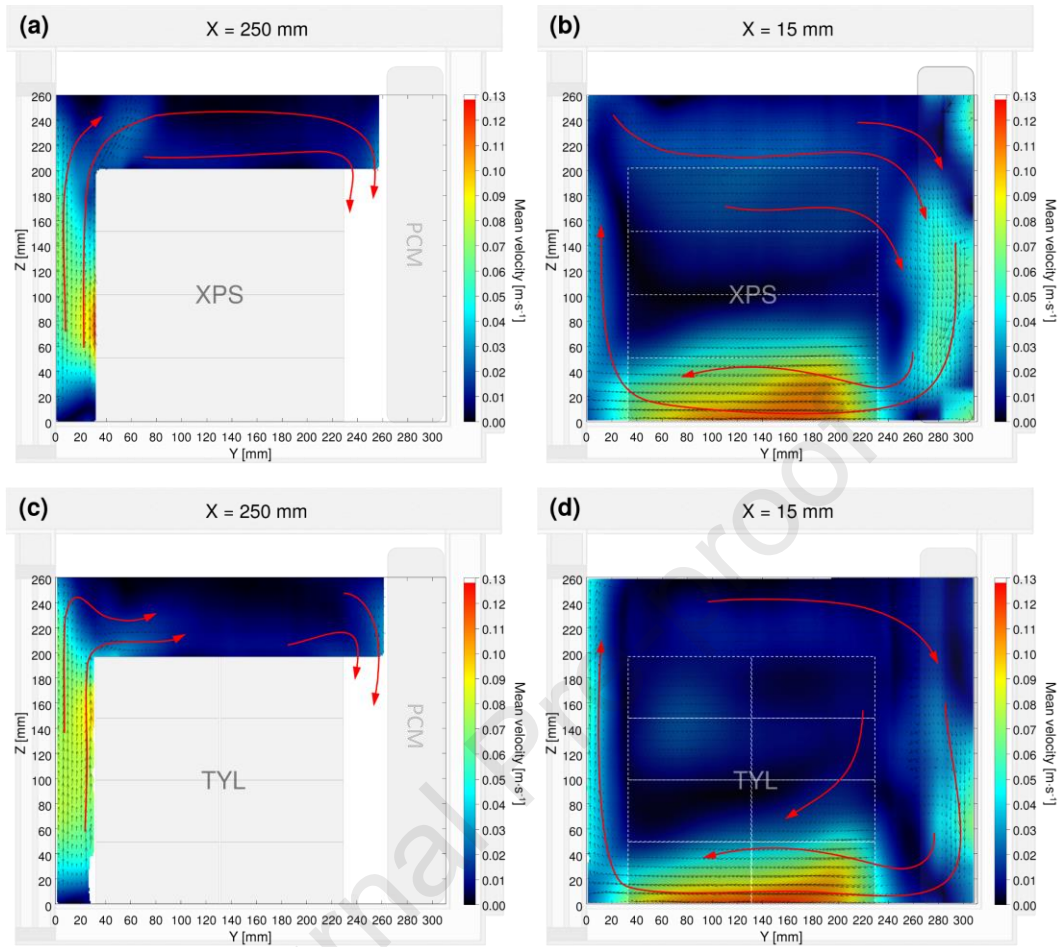


Fig. 10 Air velocity fields at (a and c) the middle ($x = 250$ mm) and (b and d) the lateral ($x = 15$ mm) planes of the box with the PCM on the side wall.

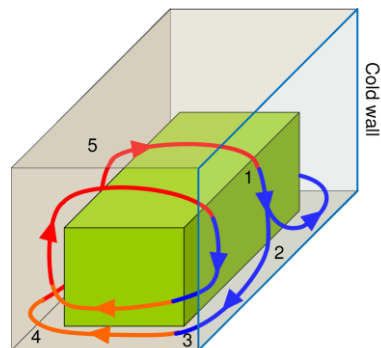


Fig. 11 Illustration of three-dimensional flow in a box with PCM on the side wall. Numbers indicate the states of the flow.

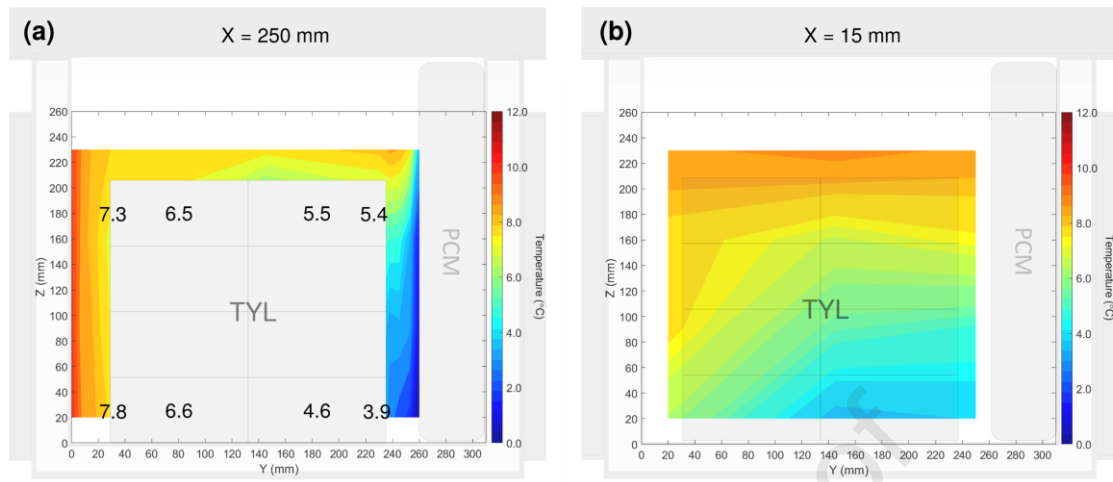


Fig. 12 Air temperature field on (a) the middle ($x = 250$ mm) and (b) the lateral ($x = 15$ mm) planes of the box with PCM on the side wall. Values indicate the time-averaged core and surface temperatures of the test products.

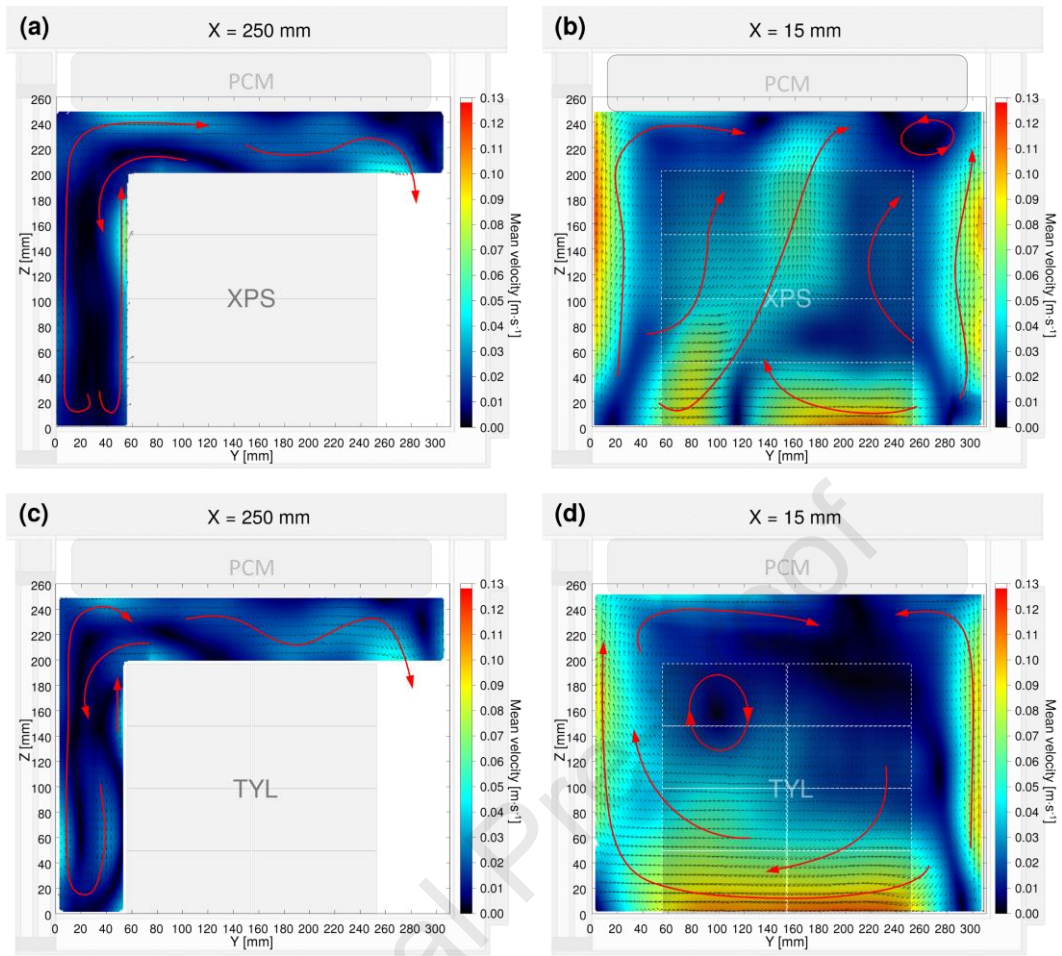


Fig. 13 Air velocity fields on (a and c) the middle ($x = 250 \text{ mm}$) and (b and d) the lateral ($x = 15 \text{ mm}$) planes of the box equipped with PCM on the lid.

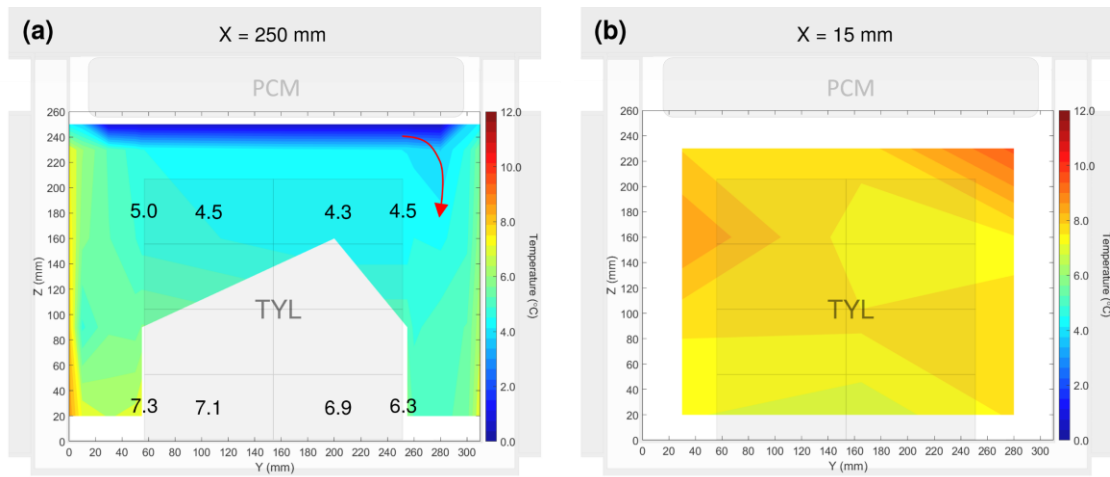


Fig. 14 Air temperature field on (a) the middle ($x = 250$ mm) and (b) the lateral ($x = 15$ mm) planes of the box with PCM on the lid. Values indicate the time-averaged core and surface temperatures of the test packs.

Highlights

- Airflow and heat transfer due to natural convection in an insulated box with PCM were studied.
- Effect of PCM positions on flow and temperature fields was investigated.
- The highest observed air velocities in the box were around $0.1 \text{ m}\cdot\text{s}^{-1}$.
- PCM position exerted no significant effect on maximum product temperature.
- Gaps should be left between the product and the box walls or PCM.

Conflict of Interest and Authorship Conformation Form

Please check the following as appropriate:

All authors have participated in (a) conception and design, or analysis and interpretation of the data; (b) drafting the article or revising it critically for important intellectual content; and (c) approval of the final version.

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