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1 **Convective heat transfer characteristics within a multi-layer packaging: application on a**
2 **pallet of strawberries**

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7 **Abstract**

8 Convective heat transfer coefficients (CHTC) on the external sides of a strawberry airtight
9 clamshell (AC) in a pallet were characterized. The CHTCs were measured on the five sides of
10 one instrumented AC. The instrumented AC was then displaced at different positions in two
11 trays representing a half level of a pallet to characterize the corresponding CHTCs. These
12 measurements were performed on four tray designs. The experimental results showed
13 different levels of heterogeneity: along the airflow direction, in the width positions, and
14 between the different sides of the AC. By comparing the performance of different designs, it
15 was found that adding vent holes to the trays improves the average CHTC within the trays but
16 increases the heterogeneity between the different sides of one AC. By contrast, increasing the
17 air headspace height above the AC leads to lower but more uniform CHTC values between
18 the different positions. Based on the effect of mixed convection on the heat transfer,
19 dimensionless correlations related to CHTC were proposed between Nusselt, Reynolds and
20 Rayleigh numbers. Similarly, a 3D CFD model was developed on ANSYS-Fluent code to
21 predict CHTC on the different packaging sides within the trays. A good agreement was found
22 between numerical and experimental values (mean relative error < 20%).

23 **Keywords:** Cold chain, Modified atmosphere package, Multi-layer package, Convective heat
24 transfer coefficient, Experiments, CFD

25 Nomenclature

q	heat transfer rate, W
A	area, m ²
h	convective heat transfer coefficient, W.m ⁻² .K ⁻¹
T	temperature, K
D	diameter, m
u	velocity, m.s ⁻¹
g	gravity acceleration, m.s ⁻²

Dimensionless groups

Nu	Nusselt number
Re	Reynolds number
Pr	Prandtl number
Gr	Grashof number
Ra	Rayleigh number

Superscripts and subscripts

avg	average
max	maximum
min	minimum
a	air
s	surface
num	numerical
exp	experimental
o	orifice
f	flux-meter

Acronyms and abbreviations

CFD	computational fluid dynamics
MAP	modified atmosphere packaging
AC	airtight clamshell
CHTC	convective heat transfer coefficient, W.m ⁻² .K ⁻¹
TD	tray design
TOA	total opening area
RMSE	root-mean-square error

Greek letter

ν	kinematic viscosity, m ² .s ⁻¹
λ	thermal conductivity, W.m ⁻¹ .K ⁻¹
ε	turbulent dissipation rate, m ² .s ⁻³

k	turbulent kinetic energy, $\text{m}^2.\text{s}^{-2}$
β	thermal expansion, K^{-1}
α	thermal diffusivity, $\text{m}^2.\text{s}^{-1}$

26 **1. Introduction**

27 Strawberries represent one of the most popular and desirable fruits due to their appearance
 28 and their nutritional content (Almenar et al., 2007). However, they are highly perishable as
 29 they are susceptible to mechanical damage, water loss, decay, and physiological deterioration
 30 (Anderson et al., 2004).

31 Packaging plays an important role into ensuring functions like containment, protection,
 32 storage, and distribution of horticultural products (Mukama et al., 2020). Moreover, it helps in
 33 maintaining optimal conditions (temperature and humidity) for products in order to preserve
 34 their quality and extend their shelf life. Strawberries are usually packed into multi-layer of
 35 packaging. First, clamshells are used as primary packaging and placed into trays. Different
 36 types, dimensions, and materials of clamshell are used (Schudel et al., 2022). The most
 37 commonly used packages are vented clamshells made of polyethylene terephthalate (PET)
 38 (Shrivastava et al., 2023).

39 Several studies were carried out to study the effect of different designs of PET clamshells and
 40 trays (Anderson et al., 2004; Ferrua and Singh, 2011; Nalbandi et al., 2016). The characteristics
 41 of the primary (clamshells) and secondary (trays) packaging have an impact on the cooling rate
 42 and uniformity and should be designed together to improve the performance of the cooling
 43 (Anderson et al., 2004). Ferrua and Singh (2011) proposed an alternative package design to
 44 increase the homogeneity of strawberries temperature inside the clamshell and between the
 45 clamshells at different positions. Most of these studies focused on the products temperature
 46 evolution in the packages only during the precooling process. Schudel et al. (2022) compared

47 the performance of clamshell designs on strawberries quality during a whole cold chain from
48 farm to retailer.

49 For fruits like strawberries, maintaining an appropriate low temperature is not a sufficient
50 condition; controlling CO₂ and O₂ levels and humidity inside the clamshell can increase the
51 shelf life throughout the post-harvest chain by reducing the heat of respiration. In this context,
52 modified atmosphere packaging (MAP) is considered a promising technology.

53 The majority of the studies considered one single airtight clamshell and focused on the impact
54 of CO₂ and O₂ concentration, storage temperature and humidity, headspace and fruit number
55 on the strawberries mass loss and shelf life (Bovi et al., 2018; Joshi et al., 2019; Matar et al.,
56 2018; Sousa-Gallagher and Mahajan, 2013). These studies took into account the effect of
57 product respiration rate and package permeability to CO₂, O₂ and H₂O on metabolic activity
58 and deterioration of the food product.

59 However, in the case of MAP, heat transfer is limited by the airtight packaging, therefore, the
60 product temperature evolution is highly dependent on the convective heat transfer coefficient
61 (CHTC) on the packaging walls. In a pallet, the CHTC is heterogeneous and is mainly
62 influenced by the airflow. Therefore, secondary packaging (tray) characteristics and clamshell
63 arrangement within the trays become critical factors to the CHTC as they directly affect the
64 local airflow characteristics around packages which in turn affect the atmosphere conditions
65 within the airtight clamshells (AC) via transfer mechanisms through AC walls (Nasser Eddine
66 et al., 2022).

67 CHTC is one of the performance parameters used to understand the effect of the packaging
68 design or staking patterns on cooling performance and heterogeneity of individual packages
69 (Berry et al., 2016; Berry et al., 2017; Defraeye et al., 2014; Defraeye et al., 2013; Wu et al.,
70 2018). In these studies, the CHTC was predicted using a CFD model and applying a steady state
71 method. Other studies used experimental methods to analyze the CHTC on the surface of

72 different products using transient (Dincer, 1994) or steady state (Merai et al., 2019; Willix et
73 al., 2006) methods.

74 However, to our knowledge, the CHTC on the wall of airtight clamshells placed in a pallet
75 were characterized neither experimentally nor numerically, although their impact on the
76 product temperature and quality evolution.

77 The aim of this study was to measure the CHTC on the different sides of an airtight clamshell
78 (AC) inside a tray of a ventilated pallet. Measurements were carried out on different AC
79 positions inside the trays representing a pallet level. Four different tray characteristics were
80 studied. This makes it possible to evaluate the cooling performance of different tray designs
81 in terms of CHTC levels and heterogeneity.

82 The analysis of experimental results also allowed to derive a relationship between the local heat
83 transfer coefficient and the airflow in the main orifice supplying the tray.

84 Another overall objective of this study is to assess the ability of CFD approach to predict local
85 airflow and heat transfer mechanisms around ACs in a pallet. After adequate validation, the
86 CFD model could be used as a design tool to optimize clamshells and trays in order to improve
87 the cooling processes of products, reducing the need for expensive and time-consuming
88 experiments.

89 This study, focused on CHTC, is part of a research project whose objective is to predict the
90 evolution of the product (strawberries) in terms of temperature, water content and quality along
91 the cold chain using MAP. The next step is the experimental and numerical characterization of
92 the heat and mass transfer at the scale of a clamshell. For this step, the knowledge of the CHTC
93 on the different wall sides is essential, especially, for weight loss and condensation issues. For
94 example, if the product is warmer than the blown air, the product is expected to dehydrate and
95 the produced water vapor to condense on the internal wall of highest CHTC. Further CO₂ and

96 O₂ transfers and product metabolism will be integrated in the model to predict quality evolution
97 and remaining shell life.

98 **2. Materials and methods**

99 **2.1. Experimental setup**

100 For the purpose of this study, empty airtight clamshells (AC) were used. Due to the symmetric
101 arrangement of the trays on a pallet layer, the experimental setup represented half of a layer,
102 i.e. 2 trays were considered (Figure 1). Each tray contained 16 ACs.

103 The experimental setup was built inside a room where the temperature can be controlled and
104 set at 15°C. The setup consists of a rectangular duct (Tunnel) with, successively, an inlet
105 section, a middle section containing the packaging, an outlet section (Figure 1a). Two
106 transparent poly-(methyl-methacrylate) (PMMA) trays with 583 mm x 370 mm x 100 mm as
107 internal dimensions and 593 mm x 380 mm x 105 mm as external dimensions (Figure 1-b) were
108 placed in series inside the tunnel (middle section). Figure 1c represents the external dimensions
109 of the AC used. The AC's were arranged in a pattern to allow a regular 4 mm air gap between
110 the AC edges in the transversal direction and a 2 mm air gap along the airflow.

111 Two fans (RS PRO 24 V DC) fixed at the outlet section create a differential pressure between
112 the inlet and the outlet to let the air flow through the packages. The power supplied to the fans
113 was regulated to ensure the desired air speed at the tunnel entrance. The system was sealed to
114 ensure no infiltrations. A honeycomb was placed at the inlet of the tunnel to further stabilize
115 the airflow.

116 **2.2. Tray designs**

117 Performances of an actual open-top tray design called TD1 is compared to three alternative
118 designs (TD2, TD3, and TD4). These alternatives are proposed to evaluate the effect of
119 additional orifices and air gap above the ACs on the heterogeneity of heat transfer coefficients
120 around packages. The actual design (TD1) used in the market has a lateral trapezoidal orifice

121 with a 5 mm air gap above the ACs. TD2 is similar in size to TD1, the only difference is that
122 circular vent holes with a diameter of 30 mm have been added to the longitudinal face of the
123 tray (Figure 2). In the case of TD3 and TD4, the height has been increased to 128 mm to
124 provide a larger air gap (28 mm).

125 The different tray designs were evaluated and compared under the same airflow rate. In the case
126 where each clamshell would contain 250 g of strawberries, the chosen reference airflow rate
127 per unit of fruit mass would be $1.1 \text{ L}\cdot\text{s}^{-1}\cdot\text{kg}^{-1}$. The air velocity at the inlet of the tunnel leading
128 to the ventilated trays was measured using an LDV device (Dantec FlowExplorer-2D), in order
129 to calculate the inlet airflow and to ensure that the same airflow rate is flowing through the
130 different package designs.

131 **2.3. Convective heat transfer coefficient measurements**

132 The method used to measure the local CHTC on the surfaces of the AC is similar to that applied
133 in previous studies (Merai et al., 2019; Willix et al., 2006). A circular (30 mm diameter) copper-
134 surface heat flux sensor (Captec-France) with a heating resistance bonded to the surface of the
135 flux-meter was used (Figure 3a). The calibrated accuracy of the flux-meter is assured by the
136 manufacturer to 3% of the read value. A specific value obtained for each heat flux sensor makes
137 it possible to convert the output signal from μV into $\text{W}\cdot\text{m}^{-2}$. Five sides of one instrumented AC
138 were equipped with heat flux sensors and thermocouples. The CHTC for each side is calculated
139 using the measured heat flux and the recorded temperature difference between the surface of
140 the flux-meter and the surrounding air according to Newton's law:

$$q/A = h(T_s - T_{\text{air}}) \quad (1)$$

141 Temperatures were measured using T-type thermocouples with a precision of $\pm 0.1 \text{ K}$ after
142 individual calibration. The air temperature was measured in the tunnel before the trays (inlet
143 section). A thermocouple was bonded between the AC surface and the heating resistance as
144 shown in Figure 3b. The measured temperature: T_{measured} is slightly higher than at the surface of

145 the flux-meter. Some experiments were conducted to determine this small difference (by adding
 146 a thermocouple at the surface of the flux-meter). This allowed to establish an empirical
 147 correlation (eq. 2) which takes into account the heat transfer conductance through the flux-
 148 meter and the heating flux. This correlation was used to calculate the temperature of the surface
 149 of the flux-meter T_s (surface exposed to airflow).

$$T_s = T_{\text{measured}} - \frac{q}{h'A} \text{ with } h' = 240 \text{ W.m}^{-2}.\text{K}^{-1} \quad (2)$$

150 Considering the symmetry of the packaging system, the measurements were conducted on half
 151 of the tray (highlighted in yellow in the Figure 3c). The instrumented AC was displaced at
 152 different positions into the tray in order to characterize the corresponding CHTC. Figure 3c
 153 shows the numbering of the AC positions inside the two trays and the names of the different
 154 sides.

155 Each experiment consisted in heating continuously the resistance until a steady state was
 156 reached. The time to reach a constant temperature depended on the side and the position of the
 157 AC. It varied between 30 minutes and 420 minutes. Each run measured the CHTC on fives
 158 sides of one AC position. Temperature and flux-meter output signal acquisitions were
 159 performed every 10 s using a data-logger (Keysight DAQ970A) and acquisition software
 160 (Keysight BenchVue).

161 **Evaluation of CHTC heterogeneities**

162 Several average CHTC levels and heterogeneity indicators were defined to compare the
 163 performance of different tray designs. These functions were calculated from the measurements
 164 made on each surface of the AC and at the 16 different positions (index i):

Average CHTC for vertical sides of AC at position i

$$\text{CHTC}_{\text{avg,vert,i}} = \frac{\text{CHTC}_{\text{right,i}} + \text{CHTC}_{\text{back,i}} + \text{CHTC}_{\text{left,i}} + \text{CHTC}_{\text{front,i}}}{4} \quad (3)$$

Average CHTC of the five sides of AC at position i

$$\text{CHTC}_{\text{avg},i} = \frac{\text{CHTC}_{\text{right},i} + \text{CHTC}_{\text{back},i} + \text{CHTC}_{\text{left},i} + \text{CHTC}_{\text{front},i} + \text{CHTC}_{\text{top},i}}{5} \quad (4)$$

Overall CHCT average of all AC in the trays

$$\text{CHTC}_{\text{avg}} = \frac{\sum \text{CHTC}_{\text{avg},i}}{16} \quad (5)$$

Deviation of the $\text{CHTC}_{\text{avg},i}$ from the overall average (Position Heterogeneity Index)

$$\text{PHI}_i = \frac{\text{CHTC}_{\text{avg},i} - \text{CHTC}_{\text{avg}}}{\text{CHTC}_{\text{avg}}} \quad (6)$$

Maximal difference of the CHCT between the different sides of AC at position i (Side Heterogeneity Index)

$$\text{SHI}_i = \frac{\text{CHTC}_{\text{max},i} - \text{CHTC}_{\text{min},i}}{\text{CHTC}_{\text{avg},i}} \quad (7)$$

Standard deviation

$$\text{SD} = \sqrt{\frac{\sum (\text{CHTC}_{\text{avg},i} - \text{CHTC}_{\text{avg}})^2}{16}} \quad (8)$$

165 2.4. Relationship between local heat transfer coefficient and airflow in the orifice

166 A series of CHCT measurements for two AC positions inside tray design (TD1) were carried
 167 out with different airflow rates and different heating powers. In order to develop a
 168 dimensionless correlation, a Reynolds number based on the hydraulic diameter of the
 169 trapezoidal orifice and the average velocity in this orifice was defined. The Nusselt and
 170 Rayleigh numbers were based on the diameter of the flux-meter. The Reynolds and Rayleigh
 171 number varied respectively in range [0, 3700] and [4000, 14000]. The two AC positions 9 and
 172 12 were chosen because they have the highest and lowest average CHCT respectively.

$$\text{Nu} = \frac{h \times D_f}{\lambda_a} \quad (9)$$

$$\text{Re} = \frac{u \times D_o}{\nu} \quad (10)$$

$$\text{Ra} = \frac{g\beta(T_s - T_a)D_f^3}{\nu\alpha} \quad (11)$$

173 Due to low velocity at some positions, free convection cannot be ignored. Therefore, a mixed
174 convection correlation was proposed, in the form:

$$\text{Nu} = \varphi \text{Re}^\beta + \delta \text{Ra}^\gamma \quad (12)$$

175 with φ , β , δ , γ coefficients to be determined.

176 **2.5. Numerical simulations**

177 A three-dimensional model was developed to predict the CHTC on the surfaces of the AC
178 placed inside the two trays using the CFD code ANSYS 21.R1. Only half of the trays was
179 modeled based on the assumption that the packaging system is symmetric. The experimental
180 geometry was reproduced using SpaceClaim 2021.R1. The computational domain is composed
181 of the free airflow fluid zone and the solid tray zones. The AC walls are modeled as surfaces
182 with no thickness. Conditions of no-slip are assumed at the surfaces of the trays and AC walls.
183 A circular surface representing the surface of the flux-meter was created on each side of the
184 AC. The model was applied for two configurations TD3 and TD4.

185 A hybrid mesh (tetrahedral and hexahedral cells) was created for the numerical study with
186 7.3×10^6 cells for TD3 and 7.4×10^6 cells for TD4.

187 The inlet boundary condition was defined as velocity inlet and set to the value measured by the
188 LDV. The temperature at the inlet was set to 15°C, as the experimentally measured temperature.
189 The outlet boundary condition was set to the atmospheric pressure.

190 With regard to the thermal conditions of the circular surfaces added to the sides of the AC, two
191 approaches were performed:

192 1- Considering that the type of convection is mixed inside the tray, therefore, buoyancy
193 effect was taking into account in the momentum equation in order to model heat transfer
194 by natural convection. In this case, to calculate the CHTC of each side for a particular
195 AC position, the heat fluxes of each side that were found experimentally at steady state
196 were imposed to its corresponding circular surfaces.

197 2- Assuming that the forced convection is predominant in all zones inside the trays.
198 Accordingly, buoyancy effect was not considered. In this case, a fixed temperature
199 (20°C) was imposed to the circular surface of each side for a particular position.

200 In total 16 steady-state simulations were run to obtain the CHTCs of the different sides for
201 the different positions.

202 Regarding the turbulence model, Realizable k-ε model with enhanced wall treatment was
203 applied in this study. The SIMPLE algorithm with second-order upwind technique is
204 applied to solve the pressure-velocity-temperature coupled equations.

205 Simulations were run using a computer with 2.4 GHz Intel® Xeon® Silver 4210R CPU and
206 256 GB of RAM.

207 The validity of the model under the two approaches was evaluated by comparing the CHTC
208 values obtained from models to the CHCT measurements performed experimentally using
209 the root-mean-square error (RMSE); for side k:

$$RMSE = \sqrt{\frac{1}{16} \sum_{i=1}^{16} (CHTC_{k,i,num} - CHTC_{k,i,exp})^2} \quad (13)$$

210 3. Results and discussion

211 3.1. CHTC levels and heterogeneities

212 Figure 5 represents the heat transfer heterogeneities inside the different tray designs, by
213 displaying the different indicators defined above in paragraph 2.4. These results were obtained
214 for the reference airflow rate: 8.8 L.s⁻¹ (1.1 L.s⁻¹.kg⁻¹ on the basis of 250 g/AC). For tray design
215 1 (TD1) this corresponds to an average air velocity in the trapezoidal orifice of 0.9 m.s⁻¹ and a
216 Reynolds number of 3700. The temperature difference between the flux-meter and the air was
217 kept between 2 and 4 K (by adjusting the heating power) in order to minimize free convection.
218 For example, concerning the bottom right AC (n°2) for the tray design 1 (TD1)

219 $CHTC_{avg,i}=18$, $CHTC_{top,i}=11$, $CHTC_{avg,vert,i}=20$ means that the side-averaged CHTC
220 (average over the 5 sides) for this AC is $18 \text{ W.m}^{-2}.\text{K}^{-1}$, the top wall CHTC is only $11 \text{ W.m}^{-2}.\text{K}^{-1}$
221 and on the vertical sides walls it is averagely $20 \text{ W.m}^{-2}.\text{K}^{-1}$.
222 $SHI_i=0.79$ (Side Heterogeneity Index) means that the difference between highest and lowest
223 CHTC on the 5 sides of this AC is 79% of the side-averaged value.
224 $PHI_i=-0.15$ (Position Heterogeneity Index) means that this AC has a side-averaged CHTC
225 15% lower than the overall CHTC (for all the sides of all the ACs).
226 Regarding the actual tray design TD1, the most important heterogeneity is found in the width
227 positions (transversal direction) for the two trays. The ACs placed in the middle (near the
228 symmetry plane) have $CHTC_{avg,i}$ from 23 to $35 \text{ W.m}^{-2}.\text{K}^{-1}$ and those at the border only from
229 14 to $18 \text{ W.m}^{-2}.\text{K}^{-1}$. The highest $CHTC_{avg,i}$ is 2.5 times higher than the lowest. The variations
230 of the CHCT are mainly related to the characteristics of the airflow. Therefore, the
231 heterogeneity can be explained by the fact that the ACs positioned at the border are less
232 ventilated since most of the airflow passes in the middle of the tray downstream of the vent
233 hole (Figure 1b). Second and third column of Figure 5 reflects the effect of the airflow
234 behavior on the heat transfer of different sides. Comparing the CHTC of the top wall between
235 the ACs placed in the middle and those placed at the border, a maximal difference of $31 \text{ W.m}^{-2}.\text{K}^{-1}$
236 can be observed between the AC 9 and AC 10. This difference can be explained by the
237 fact that AC 9 is located in the core air jet flowing through the vent hole of the tray while AC
238 10 is located in a supposed weakly ventilated recirculation area induced by the same jet. In
239 this configuration, there is a primary airflow represented by an air-jet issuing the vent hole
240 that develops in the air headspace above the ACs. The secondary airflow passing between the
241 ACs also results in a significant heterogeneity between the $CHTC_{avg,vert,i}$ of the ACs
242 positioned in the middle and those placed at the border where a maximum difference of 17
243 $\text{W.m}^{-2}.\text{K}^{-1}$ can be observed between the AC 1 and AC 6.

244 A second heterogeneity can be noticed between the different sides of each AC especially for
245 positions 7 and 10 as the relative difference between the maximum and minimum CHTC on
246 the same AC reaches about 100% (1.00 for position 7 and 1.02 for position 10). The thermal
247 behavior of the AC 7 can be explained by the fact that it is placed near the outlet of the first
248 tray, the flow is accelerating towards the vent hole. Therefore, the right, left, back, and top
249 side are subjected to higher velocities compared to the front side where it is facing a relatively
250 dead zone, which has resulted in a higher CHTC on different sides except the front side.
251 A small heterogeneity is found along the flow direction: The CHTC is higher near the inlet
252 and outlet sections of the trays where the airflow accelerates when passing through the
253 opening.

254 **3.2. Influence of tray design on CHTC level and homogeneity**

255 Adding vent holes in TD2 in comparison to TD1 has increased the overall averaged CHTC
256 from 21 to 24 $\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$ (Table 1). This increase concerns both top and vertical sides.
257 Moreover, on the top, the CHTC values become less heterogeneous between middle and
258 border lines ACs. Therefore, this leads to a better global homogeneity: the maximum PHI
259 decreases from 0.64 for TD1 to 0.40 for TD2. Locally, the major improvements were seen on
260 the ACs positioned near the added vent holes (AC n°1-2-7-8-9-10-15-16) as it can be noticed
261 in Figure 5. However, it should be noted that the heterogeneity between the different sides of
262 each AC increased. The relative difference between the maximum and minimum CHTC (SHI)
263 for AC 10 increased from 1.02 to 1.5 (Figure 5). This can be explained by the fact that the
264 CHTC of the front side significantly increased due to its position in front of the added vent
265 hole (outlet of tray 1 and inlet of tray 2) contrary to the CHTC of back side. Figure 6 presents
266 boxplots showing the dispersion of the CHTC values on each side under different packaging
267 designs. By comparison with TD1, the dispersion observed for TD2 is smaller at top but
268 larger at the front which confirms previous remarks.

269 Comparing the tray designs in terms of air headspace above the ACs, the results show that
270 TD3 when compared to TD1 has lower CHTC values with similar difference between the
271 ACs placed in the middle and at the border (higher in the middle). This could be explained by
272 the effect of the higher headspace of TD3 compared to TD1 implying lower velocity
273 magnitude above the ACs (for the same flow rate) and therefore lower CHTC values for top
274 values on the whole tray. Due to this airflow behavior, the heterogeneity between the different
275 sides is high in the middle, SHI reaches 1.39 for AC 1. However, the position heterogeneity is
276 lower: the maximum PHI equals 0.37 for TD3 when it was 0.64 for TD1. Figure 6 shows that
277 for most of the sides, the CHTC is less dispersed.

278 For TD4, the additional vent holes also improved the CHTC compared to TD3. The $CHTC_{avg,i}$
279 varied from 6 to 12 $W.m^{-2}.K^{-1}$ and from 8 to 22 $W.m^{-2}.K^{-1}$ for TD3 and TD4 respectively. As
280 shown on figure 6, the CHTC of the front, left and right sides increased.

281 Such a study allows to evaluate the heat transfer characteristics of different packaging
282 designs. It helps to identify the most sensitive areas (AC positions) inside a tray, which is
283 essential to assess them in terms of cooling rate and fruits quality. Through this study, we
284 observed that decreasing the air headspace height above the ACs allows higher CHTC levels
285 for a given airflow rate. Globally, since the section is reduced, averagely the velocity is
286 increased allowing better ventilation. The impact of additional vent holes is relevant for
287 increasing the average heat transfer but the effect in our case remains limited. Locally,
288 additional vent holes mainly concerns the ACs placed in front of the orifices. It is therefore
289 worthwhile to study the effect of the position of the orifices on the performance of packaging.
290 Moreover, a high heat transfer coefficient can lead to fast cooling, however, this is not always
291 beneficial for the quality of the fruit. Defraeye et al. (2013) stated that the optimal packaging
292 design in terms of cooling is not necessarily the best in terms of fruit quality. In addition, a

293 higher heterogeneity between the sides (higher $SHI_{i,max}$) is observed with the additional vent
294 holes.

295 **3.3. Nu-Re-Ra correlation**

296 In practice, along the cold chain, the air flowrate is often much less than $1.1 \text{ L}\cdot\text{s}^{-1}\cdot\text{kg}^{-1}$ (our
297 reference flowrate) and free convection can become significant or even dominant compared to
298 forced convection. This is why the effect of air flowrate (from 0 to $1.1 \text{ L}\cdot\text{s}^{-1}\cdot\text{kg}^{-1}$) was studied
299 in some positions. The experimental data obtained for ACs 9 and 12 (highest and lowest
300 $CHTC_{avg,i}$) in Tray Design 1 (TD1) were used for the development of a correlation between Nu,
301 Re and Ra as given in eq. (12).

302 The power value related to Reynold number (β) was estimated by regression using only data
303 for which forced convection is predominant (high flow rate, low heating power). The obtained
304 value: $\beta = 0.60$ is close to the one found by Saha et al. (2006) ($\beta = 0.65$) and Pham et al. (2019)
305 ($\beta = 0.63$). The power value related to Rayleigh number (γ) was assumed to be $1/4$ as in similar
306 problems (Özışık, 1985).

307 Table 2 presents the coefficients δ and φ (for different positions). The value of δ does not
308 depend on position because free convection is induced by the heated flux-meter, not by the local
309 velocity due to the forced airflow. The values are close whatever the sides.

310 The values of φ characterize the intensity of heat transfer related to local velocity (depending
311 on AC position and side). The values vary a lot from 0.008 to 0.364, this clearly indicates the
312 importance of local airflow characteristics. The highest value is for the front side of AC 9 and
313 can be explained by the fact that the front side of this AC is facing the trapezoidal orifice where
314 a high velocity is expected. On the other hand, the back face of AC 12 is certainly in a relatively
315 dead zone. There is no simple way to express how φ depends on the orientation and positions
316 of the AC, this reflects a complex flow pattern as could be observed by simulation.

317 Figure 7 represents the Nusselt number calculated using the correlation and the experimental
318 results for different Re. A good agreement is observed. The values observed for $Re=0$
319 correspond to pure free convection (no airflow). In the dead zones like the top side of AC 12,
320 the CHTC increased slightly (less than 50 %) with flowrate. In contrast, in the zones where the
321 forced convection is predominant like front, right and left sides of the AC 9, Nu rose largely
322 (for example from around 10 to 50 when the Re increased from 0 to 3700 for the left side of
323 AC 9).

324 **3.4. Numerical model**

325 Figure 8 and Figure 9 compare the experimental CHTCs and the predicted values from the
326 numerical model (including natural convection) for TD3 and TD4.

327 Globally, there is a good agreement: most of the CHTCs are predicted at +/- 20%. For TD3, the
328 model overestimated the CHTC of the top sides and underestimated the CHCT of the vertical
329 sides, which may be due to the overestimation of the airflow through the headspace by the
330 model.

331 Table 3 compares RMSE between experimental and numerical results with and without taking
332 into account natural convection for TD3. Excepted for the top side, RMSE is lower for mixed
333 convection. This highlights that free convection cannot be ignored especially on the side walls.
334 Different factors influencing simulation and experiment can be responsible for the
335 discrepancies observed in some points. For example, the actual geometry is not perfectly
336 symmetric, the air gaps are not exactly the same, the surface of the ACs is not totally flat and
337 smooth, and the cables can slightly modify the airflow around the ACs. Also the turbulence
338 model and the boundary treatment are perhaps not completely appropriate notably in the zones
339 of very low velocity (locally laminar flow).

340 **4. Conclusion**

341 Experiments were conducted to measure the local convective heat transfer coefficient (CHTC)
342 of an airtight clamshell walls for different tray designs representing a half pallet level.

343 Two major levels of heterogeneity were identified: according to the AC position in the tray
344 design and the considered side (top, front, back, left, and right). These heterogeneities are
345 induced by the airflow behavior in the headspace and its diffusion between the ACs. Two
346 main zones inside the tray are identified: the middle of the tray where the main opening is
347 located and where high CHTCs are observed and the poorly ventilated zone at the border of
348 the tray.

349 This study allowed the assessment of the effect of two factors: air headspace height and
350 additional vent hole, on the cooling performance in terms of CHTC levels and heterogeneity.

351 It was concluded that both of these factors should be simultaneously considered in a manner
352 that allows for an equal airflow distribution between the headspace and the circular orifices.

353 The dimensionless correlation established between Nu, Re and Ra shows that free convection
354 cannot be ignored especially on the sides placed in the poorly ventilated zone. The
355 coefficients of this correlation depend on the side orientation and the position of the AC.

356 CFD model was applied to predict the CHTC on the different sides of the ACs inside the two
357 tray designs TD3 and TD4. The numerical results of the approach using natural convection
358 showed a better agreement with the experimental results with an overall RMSE equal to 2.2
359 $\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$.

360 Further studies will concern experimental characterization of heat and mass transfer inside the
361 clamshells (strawberry temperature and weight loss measurements). The modelling at the
362 scale of a clamshell will then use the present results as boundary conditions. The difference in
363 CHTC depending on the sides could notably determine the walls where condensation issues
364 are probable.

365 **5. Acknowledgements**

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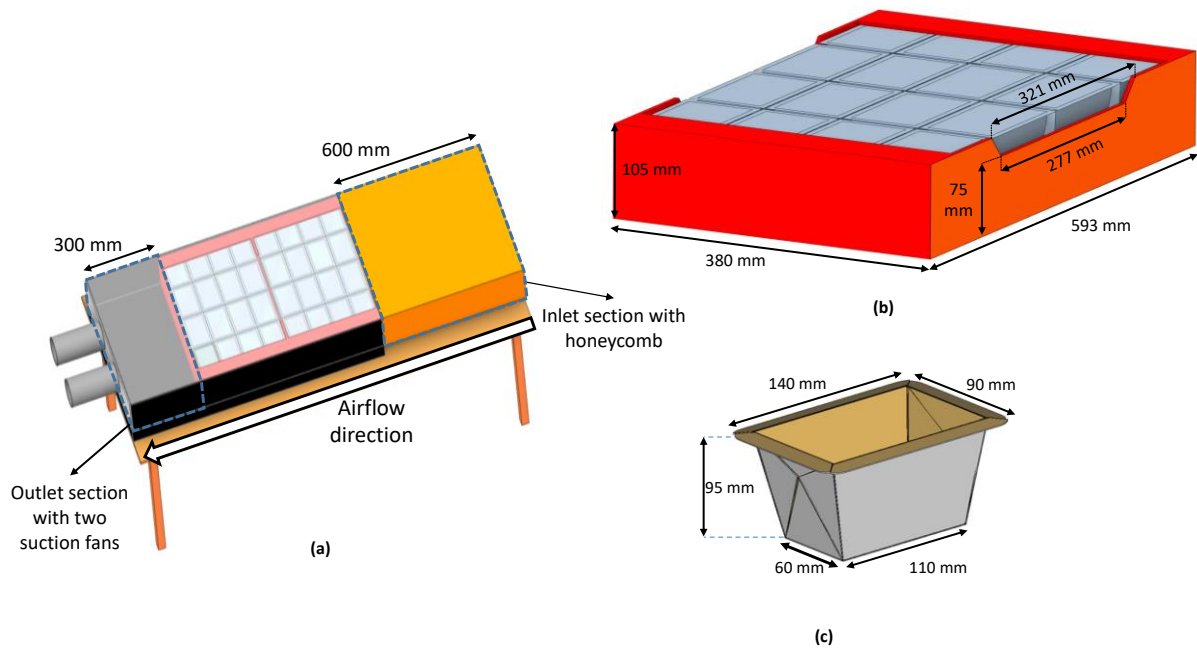
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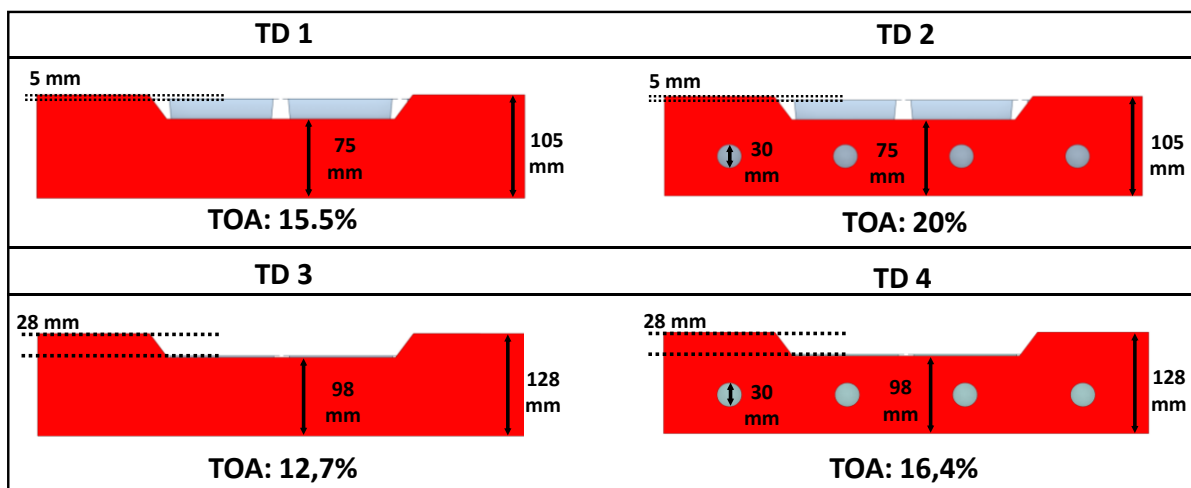


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Figure 1: (a) Experimental setup (top view), (b) dimensions of the existing tray, (c) dimensions of the AC

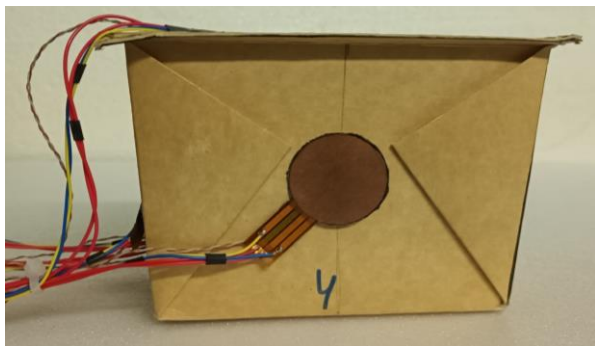


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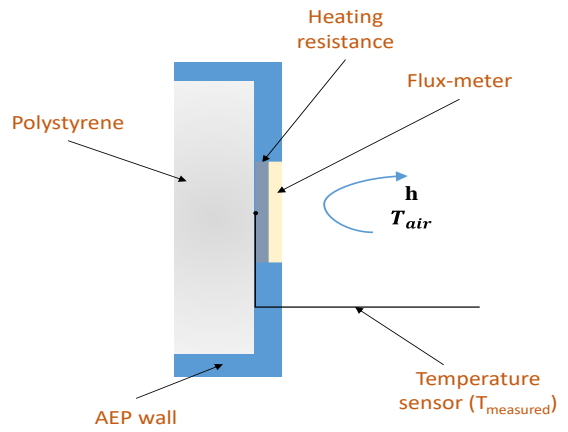
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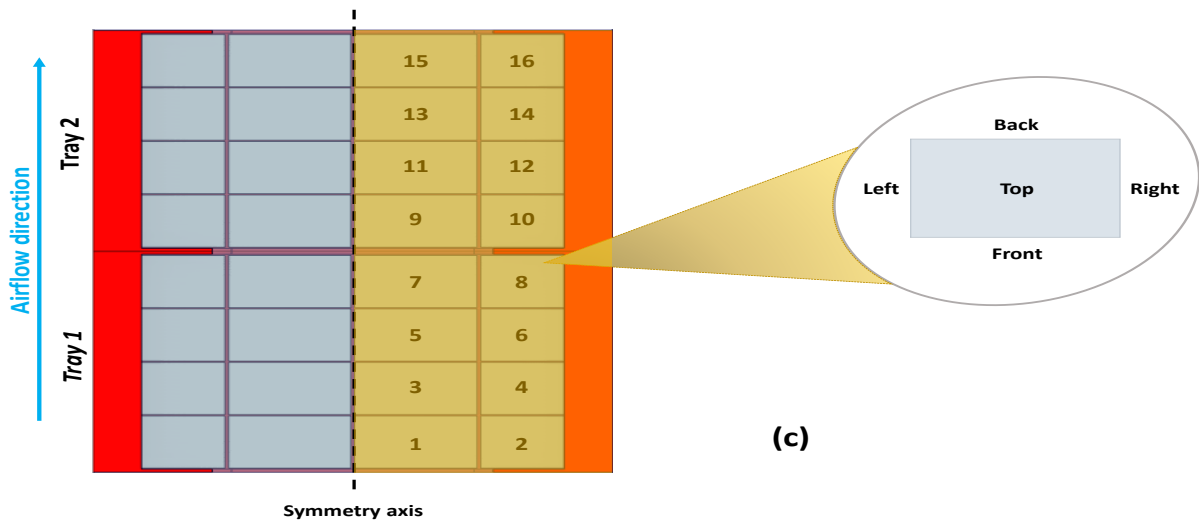
Figure 2: Dimensions and the total opening area (TOA) of the different tray designs



(a)



(b)



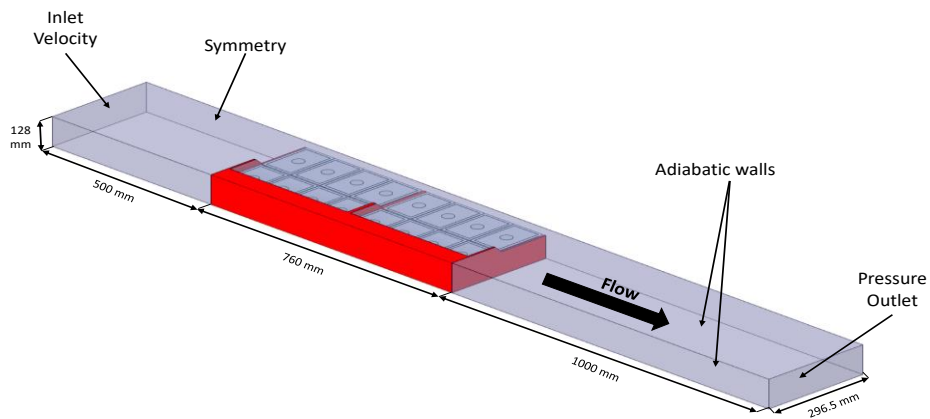
(c)

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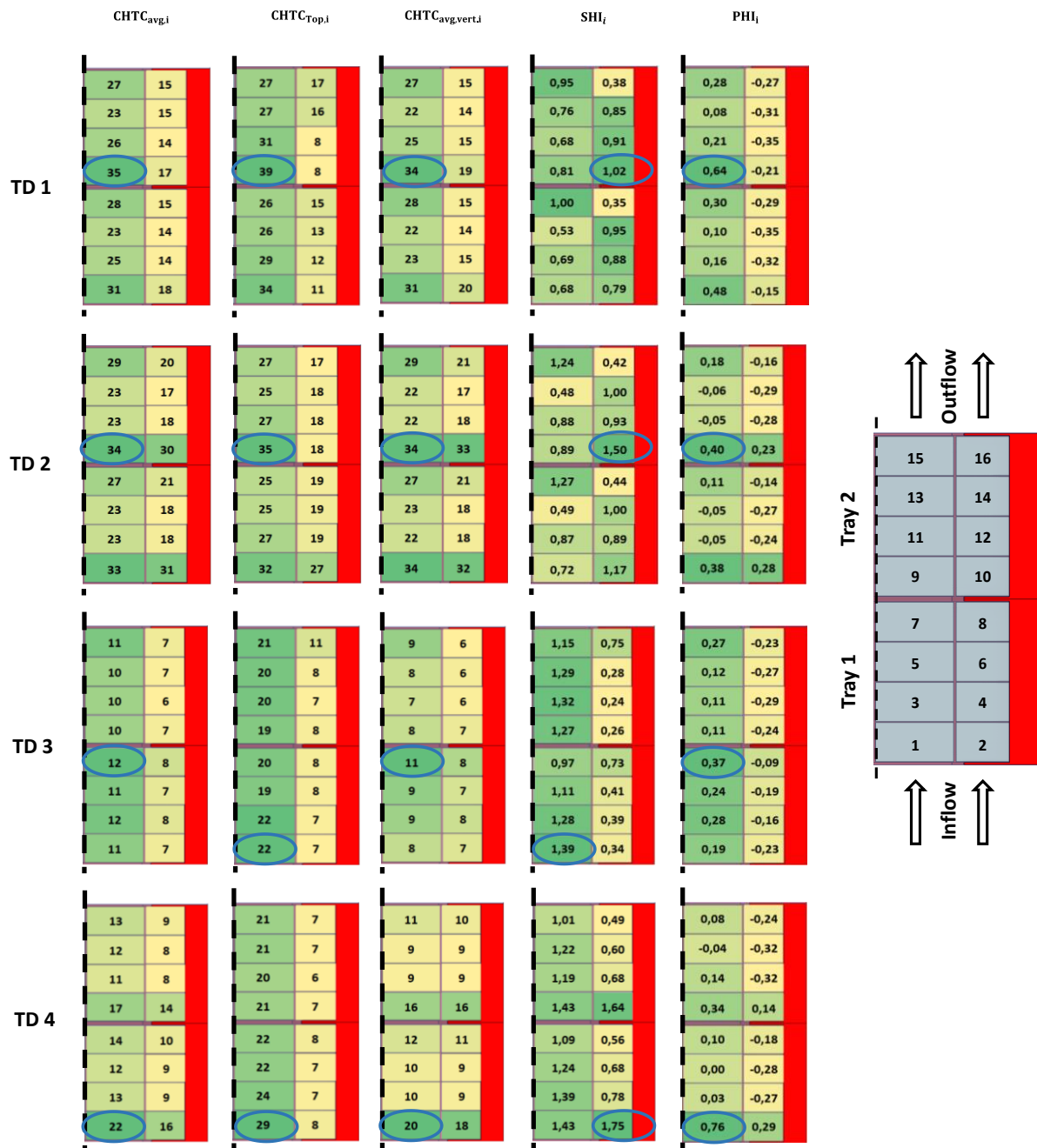
Figure 3: (a) instrumented AC with flux-meters and thermocouples, (b) schematic of a section cutting the flux-meter, (c) AC numbering and side names



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Figure 4 : CFD model boundary conditions



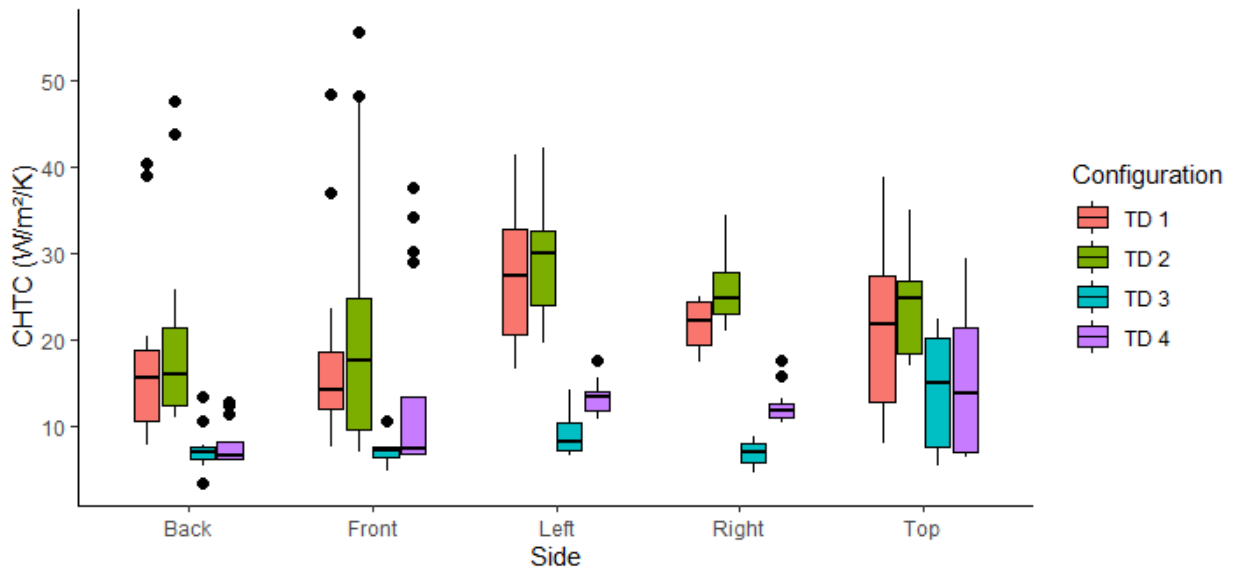
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457 **Figure 5:** Cartographies illustrating the heterogeneities of CHTC for TD1, TD2, TD3, and
 458 TD4 at 1.1 L.s⁻¹.kg⁻¹. The maximum values are circled.

459 **Table 1:** Overall average CHTCs of the ACs and their standard deviation for four tray
 460 designs

	TD1	TD2	TD3	TD4
CHTC _{avg} (W.m ⁻² .K ⁻¹)	21	24	9	12
SD (W.m ⁻² .K ⁻¹)	9	10	4	7

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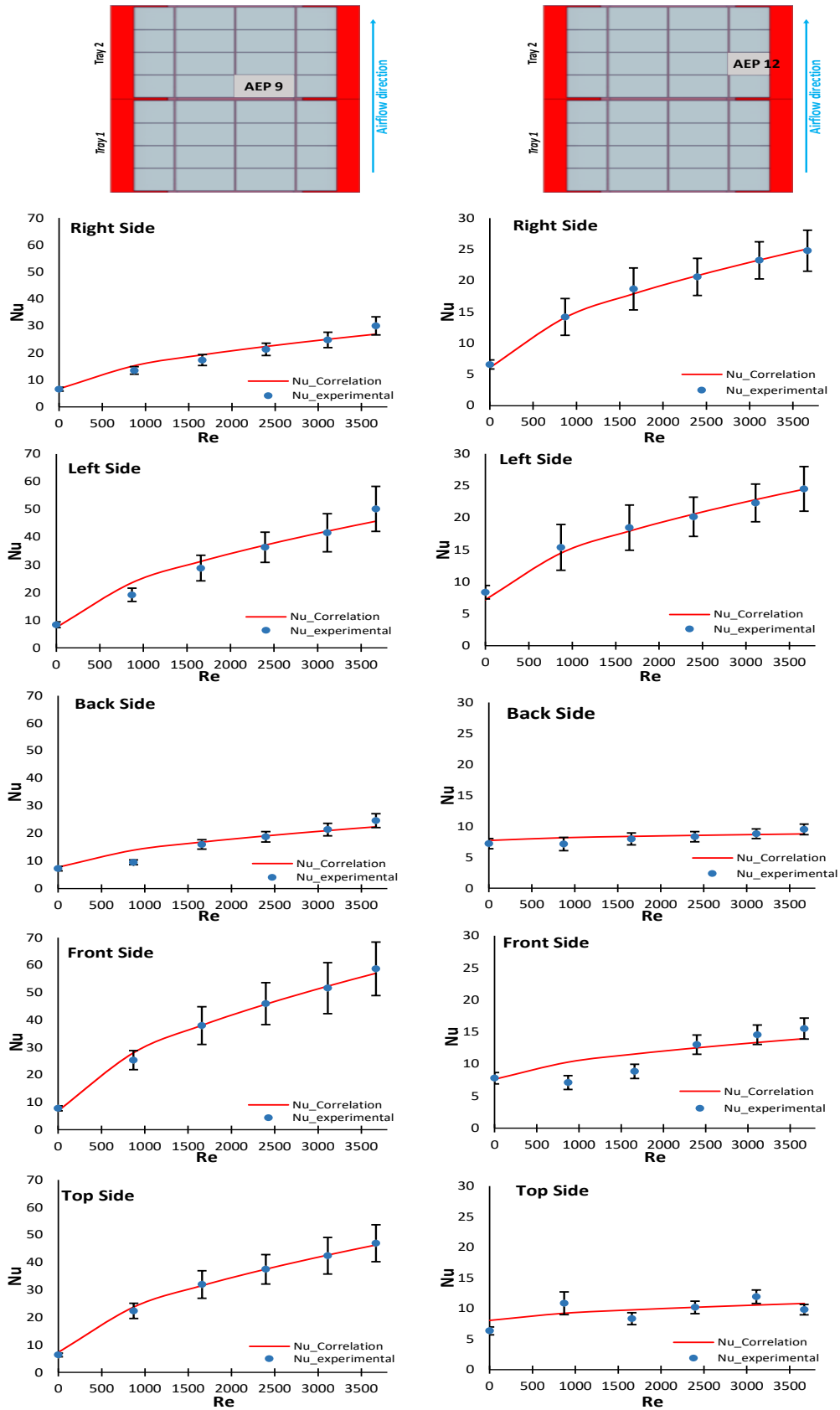
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463 **Figure 6:** Boxplots showing the dispersion of the CHTC values on each side under different
 464 packaging designs. Outliers are represented by black points

465 **Table 2:** Coefficients φ and δ in Nu-Re-Ra correlation for different sides of two positions 9
 466 and 12

	Position	Right Side	Back Side	Left Side	Front Side	Top Side
δ	9,12	0,685	0,761	0,824	0,790	0,800
φ	9	0,147	0,106	0,279	0,364	0,284
	12	0,138	0,008	0,125	0,046	0,020

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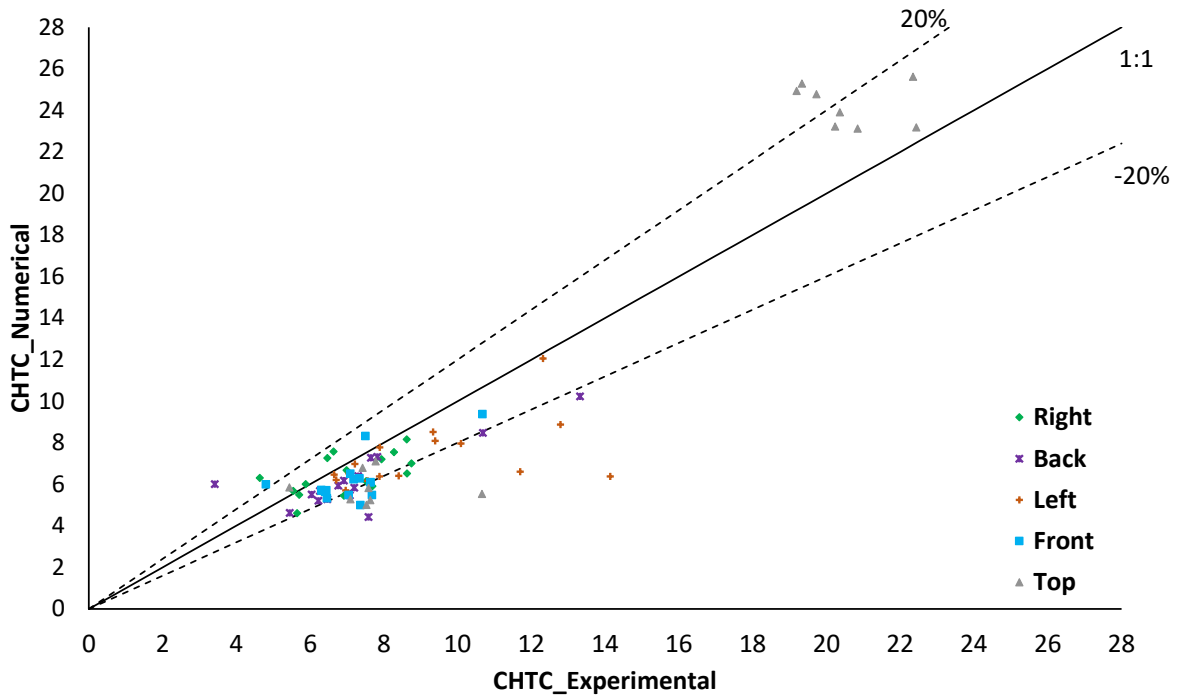
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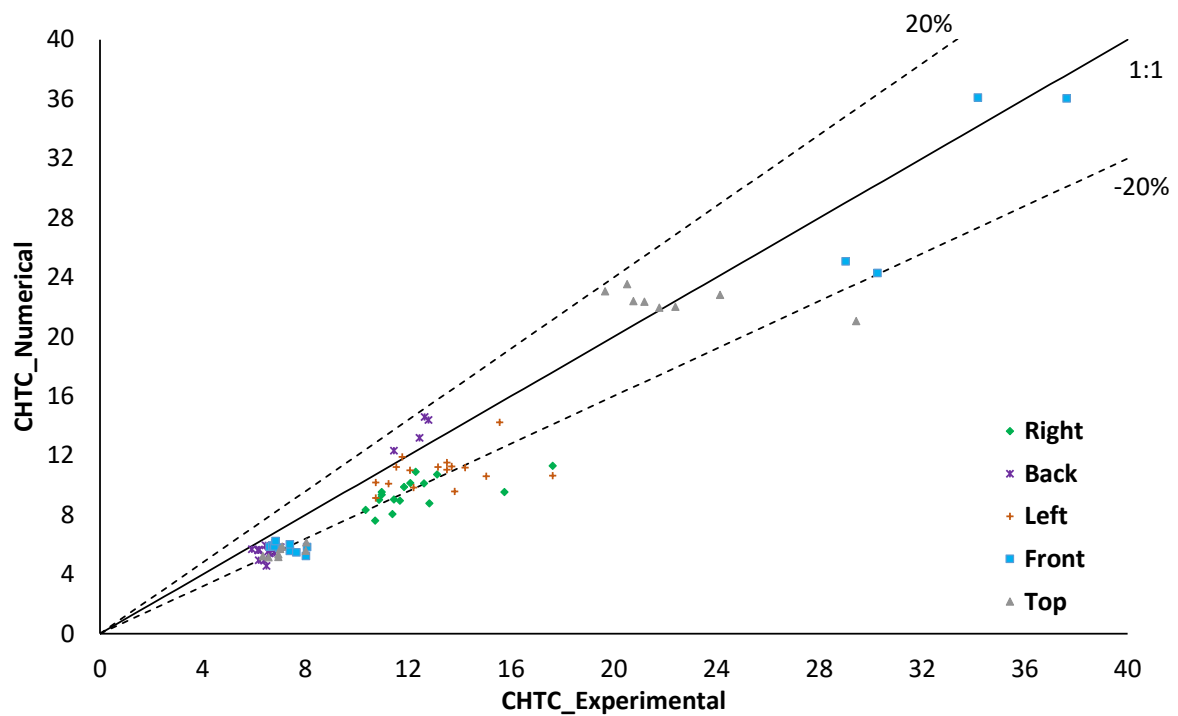
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Figure 7: Nu for each sides of AC 9 and 12 as function of Re (at the orifice) for the actual tray design (TD1). The bars represent the uncertainty of measurements.



472

473 **Figure 8:** Comparison of the CHCT obtained from the experimental data and from the
 474 numerical model (the effect of natural convection is included) for all sides of ACs placed
 475 inside TD 3



476

477 **Figure 9:** Comparison of the CHCT obtained from the experimental data and from the
 478 numerical model (the effect of natural convection is included) for all sides of ACs placed
 479 inside TD 4

480 **Table 3:** Results for RMSE ($\text{W}\cdot\text{m}^{-2}\text{K}^{-1}$) for tray design TD 3

	Right	Back	Left	Front	Top	Overall
Experiments vs model with Forced convection	1,9	2,8	3,6	2,9	2,6	2,8
Experiments vs model with Mixed convection	1,1	1,5	2,8	1,3	3,3	2,2

481