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Effect of extended surfaces on lauric acid melting process in annular cavities

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NUMERICAL INVESTIGATION ON LURIC ACID MELTING PROCESS IN HEAT EXCHANGERS: EFFECT OF THE FIN GEOMETRIC PARAMETERS

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1 **Abstract.** *Thermal energy storage makes it possible to adjust energy availability and*
2 *demand. Systems that use latent heat storage (LHTES) have high energy density and*
3 *low thermal variation during the charge and discharge cycles. The phase change*
4 *materials (PCM) used in these systems generally have low thermal conductivity, which*
5 *makes the energy charge and discharge cycles prolonged. To enhance the heat*
6 *exchange in these systems, several approaches have been presented in the literature.*
7 *Among these different solutions, the use of extended surfaces has shown good results.*
8 *The effect related to the variation of the fins geometric proportions on the PCM melting*
9 *process is still a gap, and the present work aims to analyze the effect of the fins*
10 *positioning and aspect ratio variation, in an annular cavity, filled with lauric acid. In*
11 *total, 46 geometric configurations of the fin were studied, keeping constant its*
12 *transversal area (length x thickness), but varying between 5 aspect ratios (length /*
13 *thickness), for 5 different area ratios (fin area / cavity area). The study was performed*
14 *by numerical simulation with the finite volume method. The numerical model is*
15 *composed of the continuity, momentum conservation, and energy conservation*
16 *equations, plus the enthalpy-porosity phase change model. It was validated with the*
17 *experimental results of the literature. The computational mesh was evaluated using*
18 *the Grid Convergence Index (GCI), resulting in an average index of 0.003%. For the*
19 *analysis of the results, the melting time was considered as a performance indicator.*
20 *The different heat exchange behaviors throughout the phase change process are*
21 *analyzed in terms of liquid fraction vs Fourier and Nusselt number vs. Fourier. The*
22 *melting enhancement of the studied cases is also analyzed. This set of results showed*
23 *that: the total melting time in systems with vertical fin arrangements is approximately*
24 *44% less than in systems with horizontal fin arrangements; the melting rate in systems*
25 *with horizontal fin arrangements is 15% higher than systems with vertical fins, while*

26 *there is solid PCM in the upper region of the cavity; for systems with area ratio*
 27 *$\phi = 0.003$, the increase in the fin aspect ratio entails in a total melting time reduction*
 28 *higher than 11% in vertical fins arrangements, and lower than 2% in horizontal fins*
 29 *arrangements.*

30 **Keywords:** *CFD. PCM. Lauric Acid. Melting. Fins.*

31 **Nomenclature**

| | | | |
|----|-----------|---------------------------|----------------------------|
| 32 | A | area | $[m^2]$ |
| 33 | a | fin width | $[m]$ |
| 34 | b | fin length | $[m]$ |
| 35 | C | mushy zone constant | $[kg\ m^{-3}\ s^{-1}]$ |
| 36 | c_p | specific heat | $[J\ kg^{-1}\ K^{-1}]$ |
| 37 | D | diameter | $[m]$ |
| 38 | dt | time step | $[s]$ |
| 39 | e | error | $[-]$ |
| 40 | Fo | Fourier number | $[-]$ |
| 41 | g | gravity acceleration | $[m\ s^{-2}]$ |
| 42 | GCI | grid convergence index | $[-]$ |
| 43 | h | heat transfer coefficient | $[W\ m^{-2}\ K^{-1}]$ |
| 44 | hr | representative mesh size | $[-]$ |
| 45 | k | thermal conductivity | $[W\ m^{-1}\ K^{-1}]$ |
| 46 | L | latent heat | $[J\ kg^{-1}]$ |
| 47 | L_c | characteristic length | $[m]$ |
| 48 | m | mass | $[kg]$ |
| 49 | N | number of elements | $[-]$ |
| 50 | Nu | Nusselt number | $[-]$ |
| 51 | p | pressure | $[Pa]$ |
| 52 | \dot{Q} | heat transfer rate | $[W]$ |
| 53 | q'' | heat flow | $[W\ m^{-2}]$ |
| 54 | R | radius | $[m]$ |
| 55 | AR | aspect ratio | $[-]$ |
| 56 | \vec{S} | source term | $[Pa\ m^{-1};\ W\ m^{-3}]$ |
| 57 | T | temperature | $[K]$ |

| | | | |
|----|-----------------------------|--------------------------------|---------------------------------------|
| 58 | t | time | [s] |
| 59 | \vec{V} | velocity vector | [m s ⁻¹] |
| 60 | <i>Greek Symbols</i> | | |
| 61 | α | thermal diffusivity | [m ² s ⁻¹] |
| 62 | β | liquid fraction | [-] |
| 63 | γ | volume fraction | [-] |
| 64 | Δ | variation | [-] |
| 65 | ε | numeric constant | [-] |
| 66 | ζ | thermal expansion coefficient | [K ⁻¹] |
| 67 | η_f | fin efficiency | [-] |
| 68 | λ | specific enthalpy | [J kg ⁻¹] |
| 69 | μ | dynamic viscosity | [kg m ⁻¹ s ⁻¹] |
| 70 | ν | kinematic viscosity | [m ² s ⁻¹] |
| 71 | ρ | density | [kg m ⁻³] |
| 72 | ϕ | fraction of fin area in cavity | [-] |
| 73 | <i>Subscripts</i> | | |
| 74 | 0 | without fins | |
| 75 | cr | <i>critical</i> | |
| 76 | ext | external | |
| 77 | f | fin | |
| 78 | i | element | |
| 79 | int | internal | |
| 80 | l | liquid | |
| 81 | m | melting | |
| 82 | min | minimum | |
| 83 | s | solid | |
| 84 | se | section | |
| 85 | w | wall | |

1 Introduction

86 Phase change materials (PCM) systems for heating and cooling solutions are
87 used in a variety of areas such as building construction; vehicle thermal comfort;
88 medical, pharmaceutical and chemical transport solutions; electronic cooling; solar
89 water heating systems, the textile industry [1]. These latent heat storage systems
90 (LHTES) have high energy storage density resulting in a more compact energy storage
91 system [2]. Moreover, they benefit from constant temperature (for pure substances)
92 during energy charge and discharge [3].

93 Thermal comfort solutions normally require a temperature range between 0 and
94 60 °C. This temperature range is compatible with the phase change (solid-liquid) of
95 organic PCM. Which are and widely available in nature, besides being non-toxic, non-
96 corrosive, and chemically stable. However, most PCM has low thermal conductivity.
97 This fact eventually requires techniques for increasing heat transfer to optimize energy
98 loading and unloading rates [2]. Several methods have been proposed in literature
99 such as: bubble agitation [4]; micro-encapsulation [5]; metal-matrix insertion [6]; high-
100 conductivity particle dispersion [7]; nano-particle addition [8]; PCM mixture [9] and
101 metal foam immersion [10]. Among these methods, the application of extended
102 surfaces presents good results for improving the PCM melting rate [1].

103 Annular arrangements are widely used in thermal systems due to less heat loss
104 than other configurations [11]. Thereby, annular section geometries represent more
105 than 70% of LHTES publications. The ratio between the hull and the tube's diameter
106 is linked to the energy storage capacity and the phase change rate of the material. In
107 this configuration, longitudinal, radial, or pin-shaped fins can be used. Among these,
108 the longitudinal fins are the most widespread, probably due to the ease of design and
109 manufacture while providing high heat exchange efficiency [12].

110 Several studies have already been conducted on the effect of the geometry of
111 annular section LHTES. Darzi, Farhadi, and Sedighi [13] and Pahamli et al. [14-15], for
112 example, investigated the effect of the eccentricity of tubes in annular sets. Al-Abidi et
113 al. [16], Mat et al. [17], and Rathod and Banerjee [18] found improvement in the
114 efficiency of LHTES with multiple longitudinal fins. New fin geometries have been
115 proposed by Abdulateef et al. [19] and Sciacovelli, Gagliardi, and Verda [20]. Variations
116 in the geometry of the tube itself were tested by Darzi, Jourabian, and Farhadi [21].
117 Wang et al. [22] and Yuan et al. [23] presented the effect of the fins tilt angle. Mahdi

118 and Nsofor [24-25] studied the use of fins in a system with nanoparticles. The effect of
 119 the length of the fins was addressed by Ji et al. [26]. Optimization studies and
 120 innovative longitudinal fin configurations were presented by Deng et al. [27] and Mahdi
 121 et al [28-29].

122 Finned cylindrical cavities filled with PCM have already proven to be effective
 123 and meet practical needs. Although there are different proposals for improvement, the
 124 fins proportions variation effect, in the PCM melting process, has not yet been explored
 125 in other studies. Thus, this work aims to analyze the effect of the fins proportions and
 126 arrangements on the lauric acid melting process, inside an annular cavity, using bi-
 127 dimensional computational fluid dynamics.

128 2 Problem Presentation

129 The studied heat exchanger has an annular section with internal radius
 130 ($R_{int} = 20$ mm) and external radius ($R_{ext} = 40$ mm). The fins have a length (l_f) and
 131 thickness (e_f). The annular cavity is filled with lauric acid, initially in the solid-state. As
 132 shown in Figs. 1(a-b), were studied two fins arrangements: horizontal and vertical,
 133 respectively.

134 Total melting time (t_m) is the usual performance indicator for latent energy
 135 systems. As shown in the flowchart of Fig. 2, the dimensional definition of the fins intent
 136 to evaluate the design concerning the best efficiency of the system.

137 The transversal areas of the fins (A_f) and cavity (A_{se}) are obtained by Eq. (1)
 138 and (2), respectively, considering the heat exchanger with the annular section shown
 139 in Fig. 1. The cavity area is constant in all cases. The area of the fins varies according
 140 to the different proportions of occupation (ϕ) obtained by Eq. (3). The degrees of
 141 freedom and parameters of the system are defined by the aspect ratio (AR), according
 142 to Eq. (4), where $\Delta R = R_{ext} - R_{int}$:

$$143 \quad A_f = H_f e_f \quad (1)$$

$$144 \quad A_{se} = \pi(R_{ext}^2 - R_{int}^2) \quad (2)$$

$$146 \quad \phi = \frac{A_f}{A_{se}} \quad (3)$$

147

148

$$AR = \frac{l_f}{\Delta R}. \quad (4)$$

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Initially, 5 different values of ϕ were defined: 0.003, 0.005, 0.01, 0.02 and 0.03. Also, 5 values of AR were defined: 1/8, 1/4, 1/2, 3/4, and 7/8. Since A_{se} is constant, 5 different values of A_f were obtained from Eq. (3). For each value of ϕ , 5 vertical fins and 5 horizontal fins were studied, and the values of l_f were obtained from eq. (4). Thus, e_f was obtained as $e_f = A_f / L_f$. All the dimensions analyzed are shown in Tab. 1. However, e_f value for $AR = 1/8$ and $\phi = 0.02$ and 0.03 is not physically viable, as it exceeds the value of ΔR .

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The lauric acid is a saturated fatty acid used on an industrial scale. It is characterized by biodegradability, chemical stability, non-toxicity, high availability, and low cost. Besides, it presents a small volumetric variation during phase change [30-31]. Table 2 present the thermophysical properties of lauric acid, with the values of specific heat (c_p), latent heat (L), melting temperature (T_m), thermal conductivity (k), thermal expansion coefficient (ζ), density (ρ), and dynamic viscosity (μ).

163

2.1 Mathematical Model

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The mathematical model is composed of the equations of continuity (5), momentum (6), and energy conservation (7) plus the *enthalpy-porosity* phase change model (8-12) of Voller e Prakash [32].

168

$$\frac{\partial \rho}{\partial t} + \nabla(\rho \vec{V}) = 0 \quad (5)$$

169

170

$$\frac{\partial \rho \vec{V}}{\partial t} + \nabla(\rho \vec{V} \vec{V}) = -\nabla p + \nabla(\mu \nabla \vec{V}) + \rho \vec{g} + \vec{S} \quad (6)$$

171

172

$$\frac{\partial(\rho \lambda)}{\partial t} + \nabla(\rho \vec{V} \lambda) = \nabla(k \nabla T), \quad (7)$$

173

174 Where t is the time, p is the pressure, \vec{g} is the acceleration of gravity, λ is the total
 175 enthalpy, \vec{V} is the velocity vector, ρ is the density, and \vec{S} is the source term given by
 176 Eq. (8)

$$177 \quad \vec{S} = \frac{(1-\gamma)^2}{(\gamma^3 + \varepsilon)} C \vec{V}. \quad (8)$$

178 In Eq. (8), $\varepsilon = 0.001$ is a constant, to avoid division by zero, C is the constant of the
 179 porous zone, related to its morphology [32], γ is the volume fraction of solid and liquid,
 180 obtained through Eq. (9),

$$181 \quad \gamma = \begin{cases} 0 & \text{se } T < T_s \\ 1 & \text{se } T > T_l \\ \frac{T-T_s}{T_l-T_s} & \text{se } T_s < T < T_l \end{cases} \quad (9)$$

where T_s is the temperature of the solid and T_l the temperature of the liquid.

The total enthalpy (λ) is obtained by adding the sensitive enthalpy (λ_{sen}) and the variation of the enthalpy in the phase change (λ_L). The sensitive enthalpy (λ_{sen}) is given by Eq. (10):

$$182 \quad \lambda_{sen} = \lambda_{ref} + \int_{T_{ref}}^T C_p dT \quad (10)$$

183 Where c_p is the specific heat at constant pressure, and λ_{ref} is the enthalpy at the
 184 reference temperature ($T_{ref} = 293.15$ K). The variation in enthalpy in the phase change
 185 (λ_L) is a function of temperature, obtained by Eq. (11) or by Eq. (12)

$$186 \quad \lambda_L = \gamma L \quad (11)$$

$$187 \quad \lambda_L = \begin{cases} 0 & \text{if } T < T_s \\ L & \text{if } T > T_l \\ \gamma L & \text{if } T_s < T < T_l \end{cases} \quad (12)$$

188 2.2 Initial and Boundary Conditions

189 The initial conditions consisted of $V(x, y, 0) = 0$ and $T(x, y, 0) = 20^\circ\text{C}$. The
190 boundary conditions used were:

$$191 \quad T|_{r=R_{in}} = T_w \quad V|_{r=R_{in}} = 0 \quad (13)$$

$$192 \quad q''|_{r=R_{ex}} = 0 \quad V|_{r=R_{ex}} = 0 \quad (14)$$

193 where $T_w = 80^\circ\text{C}$ and q''_w are the temperature and the heat flow in the wall,
194 respectively. The following boundary conditions were considered in the fin walls:

$$195 \quad T = T_w = 80^\circ\text{C} \quad \text{and} \quad V = 0 \quad (15)$$

196 2.3 Numerical Method and Validation

197 The simulations were performed in ANSYS Fluent software 18.2. The pressure
198 was defined as *PRESTO*, the *SIMPLE* method was used for the pressure-velocity
199 coupling and the gradient of spatial discretization as *Least Squares Cell-Based*. The
200 relaxation factors used for pressure, density, field forces, momentum, liquid fraction,
201 and energy were, respectively: 0.3; 1.0; 0.5; 0.5; 0.6 and 1.0. The energy and
202 momentum were established as *Second-Order Upwind*. With a limit of 1000
203 interactions per time step, the time step was 0.01 s. The absolute convergence criterion
204 established was 10^{-6} for mass and velocity and 10^{-8} for the energy equation.

205 The continuous growth of publications related to CFD and the significant
206 advances in computational techniques and technologies have also improved methods
207 for validating results, ensuring the credibility of works involving computational fluid
208 dynamics. Among the mesh validation methods, GCI (Grid Convergence Index) is
209 credible and recommended, having been proven through hundreds of cases [33].
210 Three computational meshes were created (*M1*, *M2*, *M3*), with 28457, 14520, and
211 10118 volumes respectively, applying this methodology. GCI was calculated for each
212 mesh and evaluated. For *M1*, GCI had an average value of 0.003%, affirming the

213 quality of this mesh. Mesh *M1* is composed of quadrilateral elements and is shown in
214 Fig. 3. Also seen in Fig. 3 is the great mesh refinement in active surfaces: Detail (A)
215 shows the refining gradient on the fin walls while Detail (B) shows the refinement of
216 the elements close to the inner cylinder wall. All meshes used in the study of fin aspect
217 ratio followed the same criteria of distribution and refinement of the elements.

218 For numerical validation, experimental data of Al-Abidi et al. [34] and
219 Yuan et al. [23] were used. Both cases are for an annular heat exchanger with internal
220 fins, filled with PCM. Quantitative analysis of the Al-Abidi et al. [34] data was performed
221 using the values of local temperature (T) as a time (t) function. Fig. 4 shows an
222 approximate temperature behavior between the present work and the experimental
223 data. The average percentage difference between both is 4.9%. This is a relatively
224 small difference because it is a local temperature value. Quantitative analysis of the
225 Yuan et al. [23] data was performed using the values of the liquid fraction (β) as a
226 function of the Fourier number ($Fo = \alpha t / L_c^2$), where $\alpha = 7.53 \cdot 10^{-8} \text{ m}^2/\text{s}$ is the thermal
227 diffusivity, and $L_c = 0.04 \text{ m}$ is the characteristic length established by the author. A
228 comparison of the results obtained by this work and those presented by the reference
229 are shown in Fig. 5. It presents an average percentage difference of approximately
230 1.5%, which proves a good agreement between results.

231 Qualitative analysis comparing the results from this work and Yuan et al. [23] is
232 shown in Figs. 6(a-b), respectively. The figures present the β fields at different melting
233 stages, where the blue and red represent the solid and the liquid phase of the PCM,
234 respectively. Regardless of running time, the similarity between results was significant,
235 showing the satisfactory agreement of the numerical model.

236 As previously presented, the mesh analysis revealed a very low GCI, giving
237 credibility to the results. A triple validation with experimental data from two different
238 authors was presented. In view of the great results obtained, the model was considered
239 validated and adequate for the case study.

240 **3 Results and Discussions**

241 In the initial moments of the melting process, heat transfer is predominantly
242 through conduction due to the solid layer's direct contact with the heated wall. As the
243 liquid layer forms between the heated wall and the solid, the participation of natural
244 convection in the heat transfer process increases. This process is shown in Figs. 7(a-

245 b) for $AR = 7/8$ and $\phi = 0.003$ in systems with vertical and horizontal fins, respectively.
 246 The right side of the section shows streamlines, and the left side shows temperature
 247 fields. The streamlines are superimposed on the liquid fraction plane, where the black
 248 color indicates the solid phase and white the liquid phase. The Details shown in Figs.
 249 7(c-h) present the velocity vectors.

250 The left side of Figs. 7(a-b) shows contours of the temperature of the phase-
 251 change process. In the vertical fin geometry of Fig. 7(a), a temperature gradient can
 252 be seen in the solid phase (blue to yellow/green). This gradient occurs because the
 253 PCM is sub-cooled at the initial condition. In the liquid region (red), horizontal thermal
 254 stratification is observed. Temperature gradients are also seen close to the solid-liquid
 255 interface and the internal wall of the exchanger, depicting a tendency of upward drag
 256 along the internal and descending walls and near the solid-liquid interface. This
 257 behavior is confirmed in the vectors shown in Figs. 7(c-d), where ascending and
 258 descending convective currents can be observed. Upward convective currents are a
 259 direct result of the buoyancy from the reduction of the PCM density as the internal
 260 surface of the exchanger is heated concerning the PCM. Note that a downward
 261 convective current must also be present to preserve the conservation of the mass, as
 262 verified.

263 In this work, the number of Rayleigh was defined as $Ra =$
 264 $[g \zeta L_c^3 (T_w - T_m)] / (\nu \alpha)$, where ν is the kinematic viscosity. Since the temperature
 265 and the characteristic length ($L_c = R_{ext}$) are the same for all cases studied, $Ra = 34364$.
 266 This value is greater than the critical Rayleigh ($Ra_{cr} = 12250$) presented by Yigit et al.
 267 [35], to an equivalent annular geometry, which justifies the appearance of Rayleigh-
 268 Bénard cells. In the horizontal fin geometry of Fig. 7(b) temperature contours also show
 269 a thermal gradient in the solid phase as per the initial sub-cooled state. However, the
 270 liquid phase has a diffuse temperature gradient. Such behavior is following the
 271 expected Rayleigh-Bénard convective behavior of a heated surface. This fact becomes
 272 evident from characteristic recirculations of the Rayleigh-Bénard behavior shown by
 273 the streamlines over the horizontal heated surfaces, as well as vectors of Figs. 7(e, g).
 274 In this convective process, the heated liquid rises in multiple regions perpendicular to
 275 the heat source, coming into contact and fusing the solid interface as it exchanges heat
 276 and cools. Due to the conservation of mass, cooled liquid moves downwards towards
 277 the heated plane, establishing a recirculation zone and an efficient thermal exchange
 278 cycle. Fig. 7(f) shows an upward flow near the heated surface of the tube and

279 consequently, downward flow due to the presence of Rayleigh-Bénard cells at the top
 280 of the cavity. The shear region between ascending and descending flows can also be
 281 seen in Fig. 7(h). This resulting intense thermal exchange system persists if there is a
 282 solid material in the region above the heated surfaces.

283 The evolution of the melting process for a sample of the studied cases is shown
 284 in Figs. 8(a-e), where the liquid-solid interfaces can be observed at 1 min, 10 min, 20
 285 min, 30 min, 40 min, and 60 min. It is possible to observe the progress of cases with
 286 $AR = 1/4$, $1/2$ and $7/8$ for all ϕ , both in terms of melting and the geometric aspect
 287 evolution with the variation of AR and ϕ . The first three columns cover cases with
 288 vertical fins, while the last three refer to cases with horizontal fins. Observing the lines
 289 for horizontal fins at $t = 1$ min, 10 min, and 20 min, the liquid region advances
 290 significantly in comparison to the respective cases with vertical fins. This trend is in line
 291 with the natural convection process shown in Fig. 7.

292 The liquid fraction lines of Figs. 8(a-e) at $t = 30$ min, 40 min, and 60 min show a
 293 deceleration in the melting speed of the cases with horizontal fins compared to vertical
 294 fins as the solid phase moves away from heated surfaces. It is noted that the total
 295 melting process concluded before 40 min for the vertical fin, while for the horizontal fin,
 296 the melting process extends beyond the 60 min when analyzing specifically the cases
 297 with $AR = 7/8$ of Fig. 8(e).

298 **Figures 9(a-e)** shows the variation of β vs. Fourier number ($Fo = \alpha t/L_c^2$), for
 299 horizontal and vertical fins with $AR = 7/8$, $3/4$, $1/2$, $1/4$, and $1/8$ for $\phi = 0.003$, 0.005 ,
 300 0.01 , 0.02 , and 0.03 . In this work, the characteristic length was defined as $L_c = R_{ext}$.
 301 **Figures 9(a-e)** shows that for all cases up to approximately $Fo = 0.006$, the values of
 302 β are practically superimposed before drifting apart as Fo increases. Between
 303 $Fo = 0.006$ and $Fo = 0.05$, melting is favored by horizontal fins, with the faster melting
 304 process occurring in the fin with $AR = 7/8$ for all the values of ϕ . From $Fo = 0.05$ and
 305 higher, there is a transition in the slope of the horizontal fins, with the reduction of the
 306 melting rate in these configurations. Afterward, vertical fins complete the melting
 307 process ahead of horizontal fins starting from the highest $AR = 7/8$ and proceeding in
 308 an orderly way to the lowest $AR = 1/8$. Only after the total melting of all cases with
 309 vertically oriented fins, the cases with horizontally oriented fins begin to reach $\beta = 1$,
 310 also in an orderly manner, from the cases with the highest AR , down to the lowest.
 311 **This behavior of β is similar for all ϕ values, with only a subtle increase in the melting**
 312 **rate as ϕ increases.**

313 Figures 10(a-c) present the liquid fraction fields (right side) and temperature
 314 fields (left side) at $t = 600$ s and 2270 s, for $\phi = 0.03$ and $AR = 7/8, 1/2$ and $1/4$,
 315 respectively. The initial instant (600 s) enables us to view the heat exchange
 316 mechanisms present at the beginning of the melting process, while the final instant
 317 (2270 s) refers to the total melting time of the fastest case (vertical fins with $AR = 7/8$).
 318 This observation allows a comparison of the other cases melting state at the same
 319 instant as when the fastest case had already concluded. The β value is displayed at
 320 the center of each geometry. Observing the temperature fields at $t = 600$ s, it is visible
 321 the greater dynamism of the cases with horizontal fins as the weaving outlines are
 322 consistent with recirculations of the Rayleigh-Bénard convective behavior. In contrast,
 323 in the cases with vertically oriented fins, it is possible to observe relatively stratified
 324 temperature fields and upward flow behavior only in the region close to the fins and
 325 the heated wall. Consequently, the β value is higher for cases with horizontal fins.

326 Still, concerning Figs. 10(a-c) at $t = 2270$ s, it can be seen in all cases that the
 327 temperature fields show a well-defined thermal step between solid and liquid phase
 328 despite presenting a small thermal stratification close to the solid-liquid interface.
 329 However, when comparing the β values between vertical and horizontal fins, it is
 330 observed that the vertical fins have a higher melting rate than the horizontal fins
 331 because the vertical fins are closer to the solid mass, compared to the horizontal fins.

332 **Figures 11(a-e)** shows the behavior of the Nusselt number ($\overline{Nu} = \bar{h} \cdot L_c / k_l$) as
 333 a function of Fo , for horizontal and vertical fins with $AR = 7/8, 3/4, 1/2, 1/4$, and $1/8$ for
 334 $\phi = 0.003, 0.005, 0.01, 0.02$, and 0.03 . The average heat transfer coefficient
 335 [$\bar{h} = q'' / (T_w - T_m)$] is calculated from is the total heat flow (q'') while the temperatures
 336 are taken as $T_w = 80^\circ\text{C}$ at the active wall and $T_m = 44.2^\circ\text{C}$ as the melting temperature
 337 of the PCM. **Figures 11(a-e)** can be divided into three behavior patterns indicated by I,
 338 II, and III. As observed by Ji et al. [26], these patterns are related to the heat exchange
 339 mechanisms during the phase change of the PCM. **In addition to the analysis of Nu**
 340 **values, Figs. 12(a-c) shows the fields of temperature, liquid fraction, and streamlines**
 341 **of an example case. The joint analysis of Figs. 11(a-e) with Figs. 12(a-c) allows better**
 342 **compression of the three-behavior described below.** Region I start with \overline{Nu} higher than
 343 12 for all cases. This fact is due to the process beginning when there is direct contact
 344 between the solid PCM and the heat source. As the PCM melts, the liquid interface is
 345 formed between the solid and the heat source. The increasing size of the liquid layer

346 increases the thermal resistance by conduction between the heated wall and the
 347 melting material with a gradual decrease to $Fo = 0.003$, as can be observed in the
 348 Fig. 12(a). Region II is characterized by the beginning of fluctuations in the behavior of
 349 \overline{Nu} . These fluctuations are due to a further increase in the liquid layer's size and the
 350 beginning of an intense convective process, which can be observed in the streamlines
 351 of the Fig. 12(b). The greater dynamism of the convective process on the horizontally
 352 oriented fins, its due the Rayleigh-Bénard convective pattern. In line with this
 353 phenomenon, the highest values of \overline{Nu} that occur in this region are relative to the
 354 horizontal fins (dashed lines). In region III, from $Fo = 0.05$, \overline{Nu} decreases rapidly for
 355 cases with a horizontal fin. As shown in Fig. 12(c), this behavior makes the heat
 356 transfer process restricted to a practically conductive in the molten and stationary
 357 liquid. As the melting process approaches the end, cases with vertically oriented fins
 358 presents a gradual drop in the value of \overline{Nu} , until zero. This behavior of Nu is similar for
 359 all ϕ values, with only a subtle increase in region II, according to the increase in ϕ .

360 One way to qualify the different systems analyzed is by their effectiveness (η_f)
 361 in relation to the system without fins. In this work, this effectiveness is defined as: $\eta_f =$
 362 $(\beta/\beta_0) - 1$, where β_0 is the liquid fraction of a case without fins at the same instant in
 363 time. Figures 13(a-e) shows the variation of η_f as a function of Fo for vertical and
 364 horizontal fins with $AR = 1/8, 1/4, 1/2, 3/4,$ and $7/8$ for $\phi = 0.003, 0.005, 0.01, 0.02,$ and
 365 0.03 . It is initially observed that all cases present their highest η_f at the beginning of the
 366 melting process. Ideally, η_f would remain high throughout the melting process.
 367 However, the cases show two patterns of decrease, first more accentuated and then
 368 gradually. The effectiveness of horizontal and vertical fins, represented by dotted and
 369 continuous lines, respectively, is quantitatively similar for each value of AR . The value
 370 of η_f is initially higher for horizontal fins, but for $Fo > 0.06$, vertical fins become more
 371 efficient. This inversion occurs for all AR presented. Horizontal fins have η_f tending to
 372 0 with increasing Fo . This tendency indicates the fins' inefficiency in the melting
 373 process since the null value of η_f is equivalent to a system without fins. However,
 374 vertical fins keep the η_f value practically stable between $Fo = 0.06$ and the ending of
 375 the melting process. For these geometries, higher AR values reveal greater
 376 effectiveness and lower Fo value after the melting process. The η_f behavior is
 377 equivalent to all ϕ values, with a directly proportional increase between the η_f and ϕ
 378 values.

379 Figures 14(a-b) show the time to reach complete melting (t_m) as a function of
 380 AR , for horizontal and vertical fins, respectively, with $\phi = 0.003, 0.005, 0.01, 0.02$ and
 381 0.03 . For better visualization of the results, the t_m range is different between the cases.
 382 Figures 14(a-b) shows that for both horizontal and vertical fins, the decrease in t_m is
 383 practically linear with AR .

384 For cases with horizontal fins, Fig. 14(a) shows a decrease in t_m with an increase
 385 in AR below 1000 s for all values of ϕ . This small period indicates the low
 386 representativeness of the aspect ratio in the total melting time when the fins are
 387 arranged horizontally. This fact is related to what is observed in Figs. 13(a-e), where it
 388 can be observed that η_f is practically null, for all AR values at the end of the melting
 389 process of the cases with horizontal fins. Even if cases with horizontal fins promote a
 390 higher melting rate at the beginning of the process, the conclusion of the melting
 391 process stretches further due to the gap between the solid PCM and the heat source.
 392 Increases in AR in horizontal fins do not reduce this gap.

393 The total melting time of cases with vertical fins of Fig. 14(b) varies between
 394 2000 s and 8000 s. The higher amplitude in this ordinate axis is due to the more
 395 significant influence of the AR variation on the total melting time. As seen in Figs. 13(a-
 396 e), the increase in AR results in greater effectiveness at the end of the melting process
 397 for vertical fins. Since vertical fins maintain proximity between the heat source and the
 398 solid PCM until the end of the melting process, increases in AR also increase the
 399 proximity of the heated wall to the solid PCM.

400 As seen in Figs. 14(a-b), the shortest total melting time ($t_{m,min}$) occurs for
 401 $AR = 0.88$ (7/8). In these figures, although the tendency of $t_{m,min}$ to decrease with
 402 increasing ϕ can be observed, a more accurate assessment is shown in Figs. 15(a-b),
 403 which show the variation of $t_{m,min}$ as a function of ϕ , for horizontal and vertical fins,
 404 respectively, at the shortest melting time of $AR = 0.88$. Figures 15(a-b) demonstrate
 405 that $t_{m,min}$ is inversely proportional to ϕ regardless of the fin layout. This can be due to
 406 the volume reduction of PCM in the cavity since the increase in ϕ implies an increase
 407 in the volume occupied by the fin. It is also observed that the range of variation of $t_{m,min}$
 408 is higher for vertical fins (Fig.15(b)) than for horizontal fins (Fig. 15(a)). The difference
 409 between the time ranges is the result of prolonged load cycles in systems with
 410 horizontal fins and shorter cycles in systems with vertical fins, as seen in Figs. 14(a-
 411 b).

412 5 Conclusions

413 This work sought to analyze the behavior of the melting process of lauric acid
414 PCM in a cylindrical cavity concerning the dimensional variation fins. The numerical
415 model was validated with numerical and experimental results provided by literature.
416 The computational mesh analysis was performed using the GCI method, which
417 resulted in an average value of 0.003%. Under these conditions, the main results were:

- 418 • Horizontal fins had a melting rate up to 15% higher than vertical fins if there was
419 solid PCM in the region above the layer of PCM already fused next to the horizontal
420 fin. This occurred for all values of ϕ and was due to localized Rayleigh-Bénard
421 convective effect;
- 422 • For all values of ϕ , the total time for complete melting was, on average, 44% lower
423 for vertical fins than for horizontal fins. In a vertical fin configuration, buoyancy
424 effects were less pronounced in the active wall (circular internal) and the vertical
425 fin. Consequently, this region was the last to be melted, and higher temperature
426 differences develop between PCM and active walls, promoting higher rates of heat
427 transfer throughout the melting process;
- 428 • Increases of the aspect ratio, or thinness, of the fin, reduced the time of the total
429 melting process by the maximum values of 11.3% for vertical fins and 1.2% for
430 horizontal fins, with $\phi = 0.003$;
- 431 • Increases in ϕ resulted in an approximately linear decrease in the total melting time
432 of the PCM in the heat exchanger. This was due to the larger area of the fin's
433 occupancy in the cavity.

434

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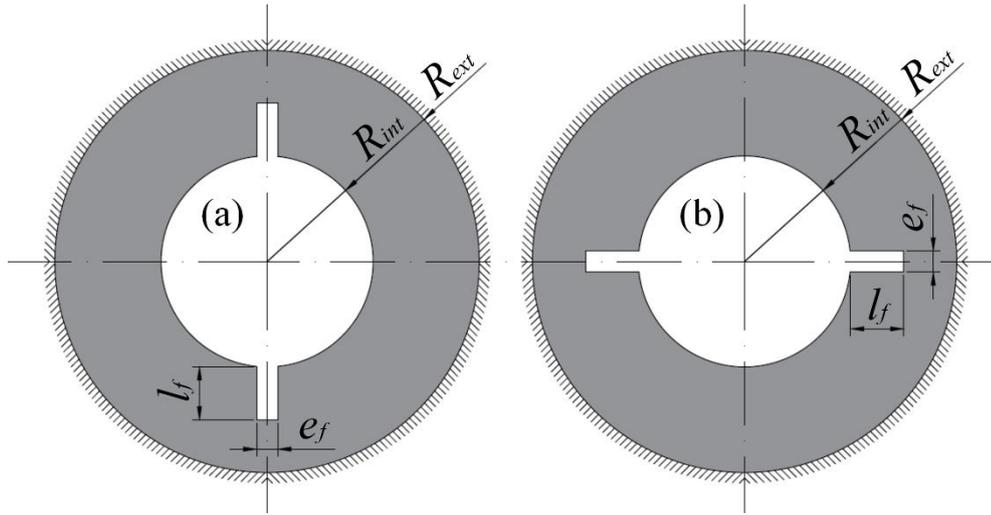
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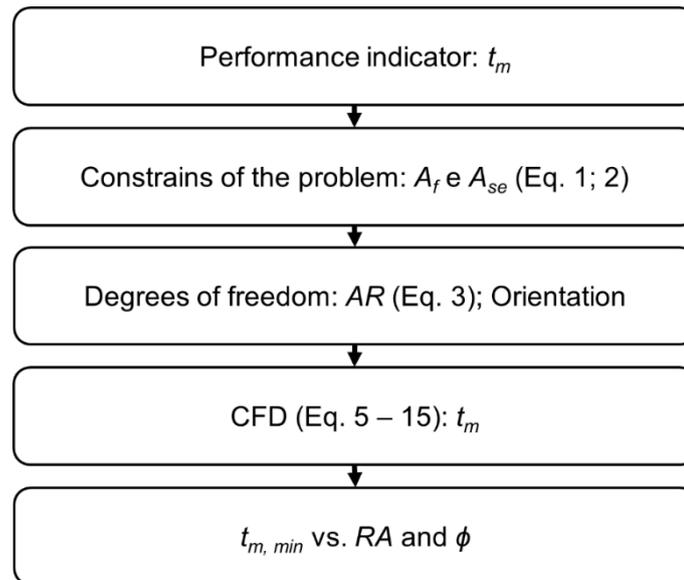
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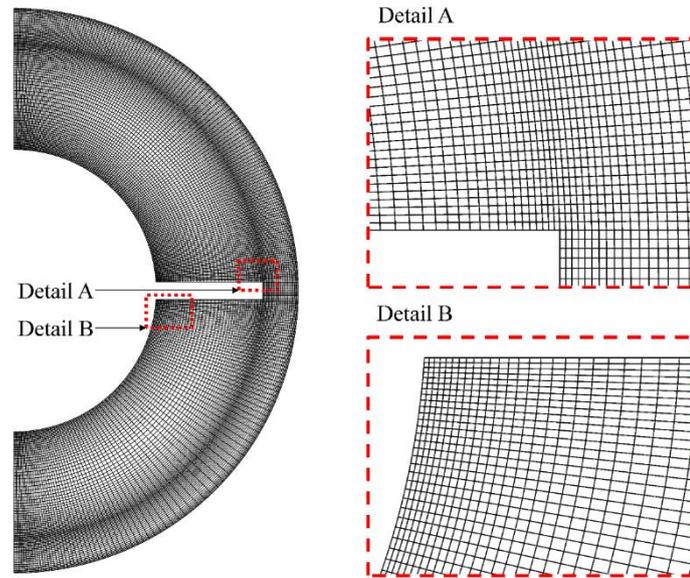
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Figure 1 - Annular tube with: (a) vertical and (b) horizontal fins.

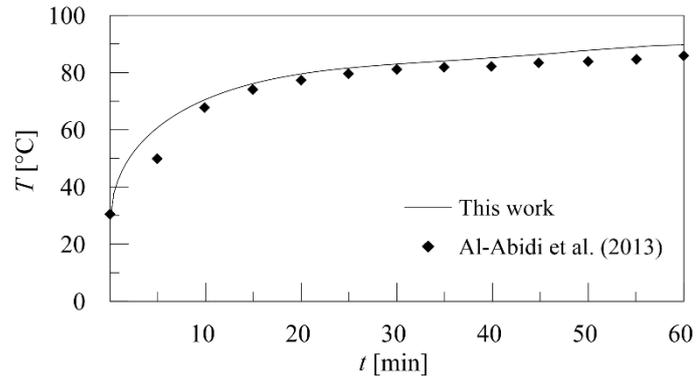


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Figure 2 – Flowchart of fin studies conducted.



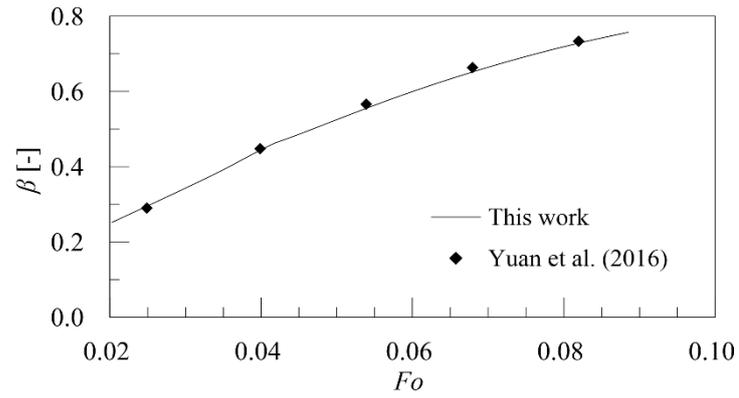
564 Figure 3 – Computational mesh with details at the fin end (A) and the fin base and
565 inner cylinder wall (B).



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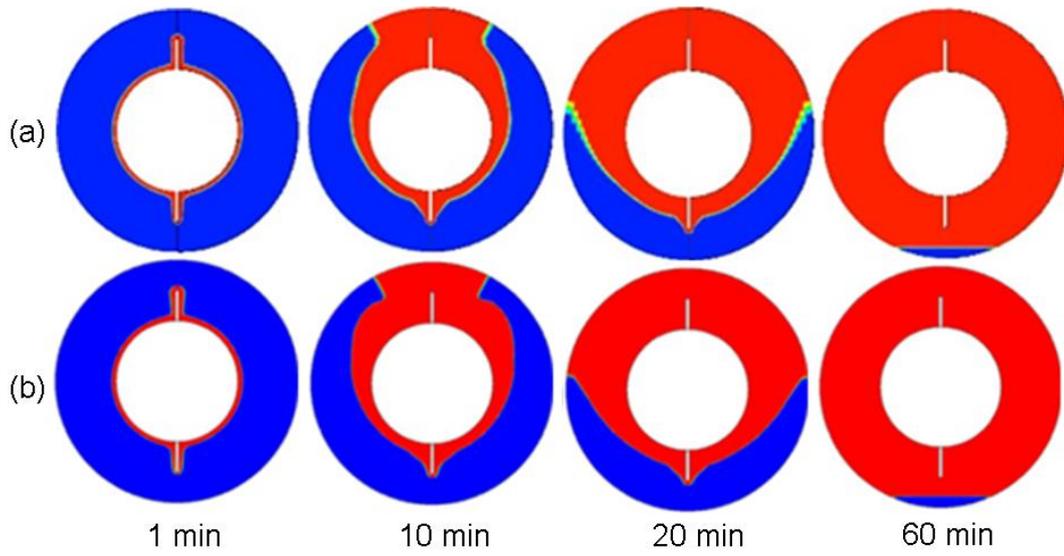
Figure 4 – Temperature (T) vs. (t): current numerical simulation and experimental data from Al-Abidi et al. [23].

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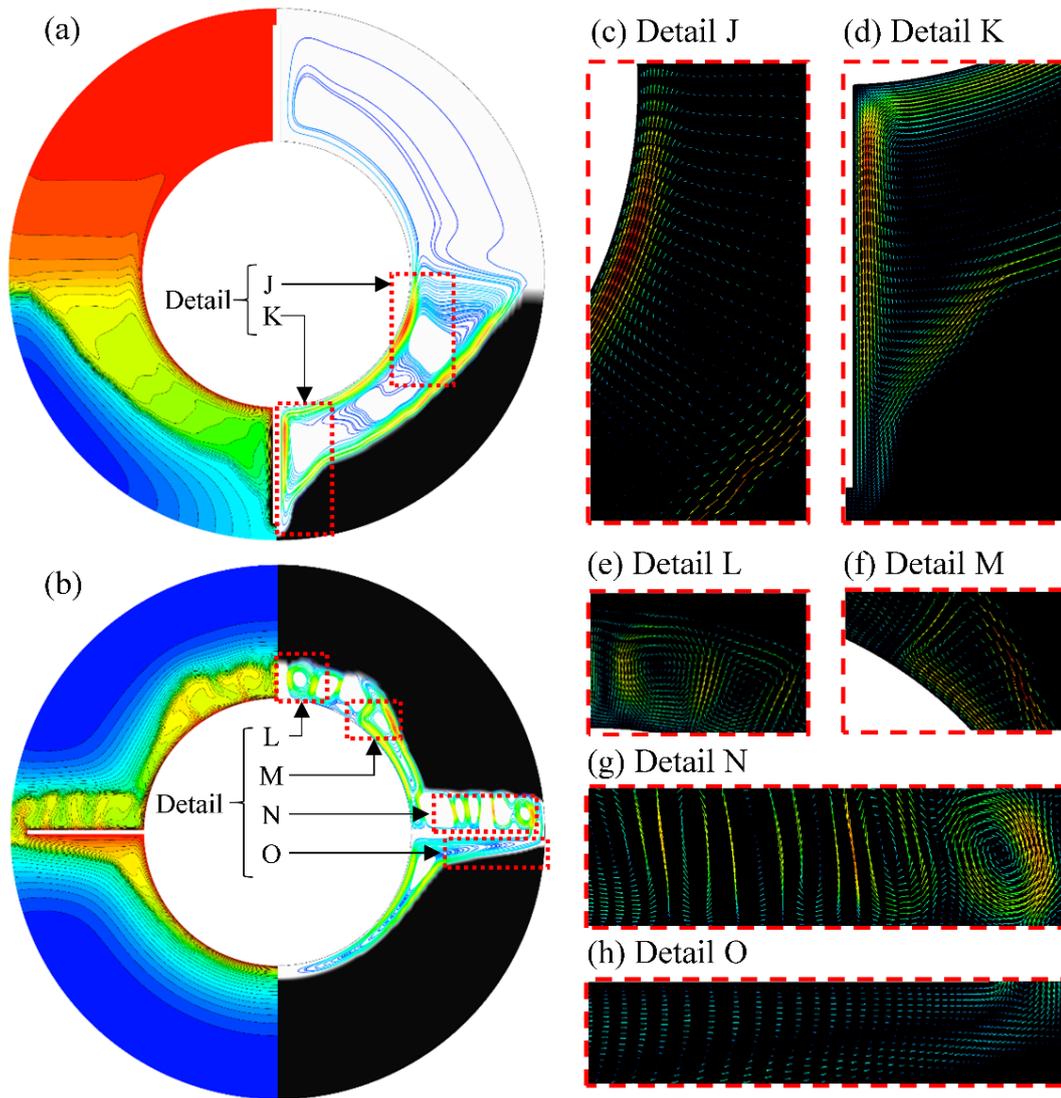
568 Figure 5 – Liquid fraction (β) vs. Fo : current numerical simulation and experimental
569 data from Yuan et al. [23].

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572 Figure 6 $-\beta$ fields at $t = 1; 10; 20$ and 60 min: (a) current numerical simulation and (b)
573 experimental data from Yuan et al. [23].



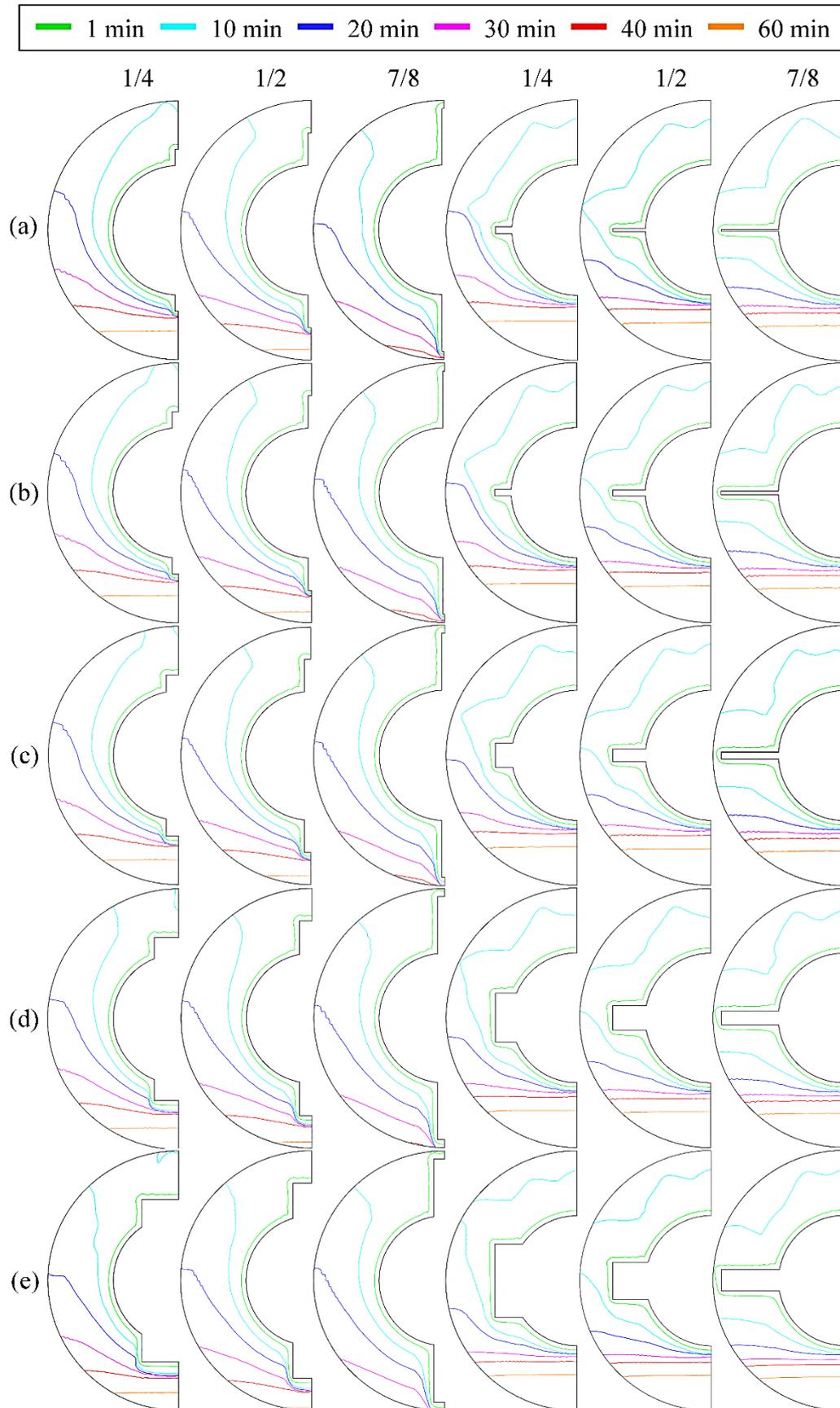
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Figure 7 – T fields and streamlines, with details of the velocity vectors for $AR = 7/8$ and $\phi = 0.003$: (a) vertical fins, (b) horizontal fins and (c, d, e, f, g, h) related to Details J, K, L, M, N and O respectively.

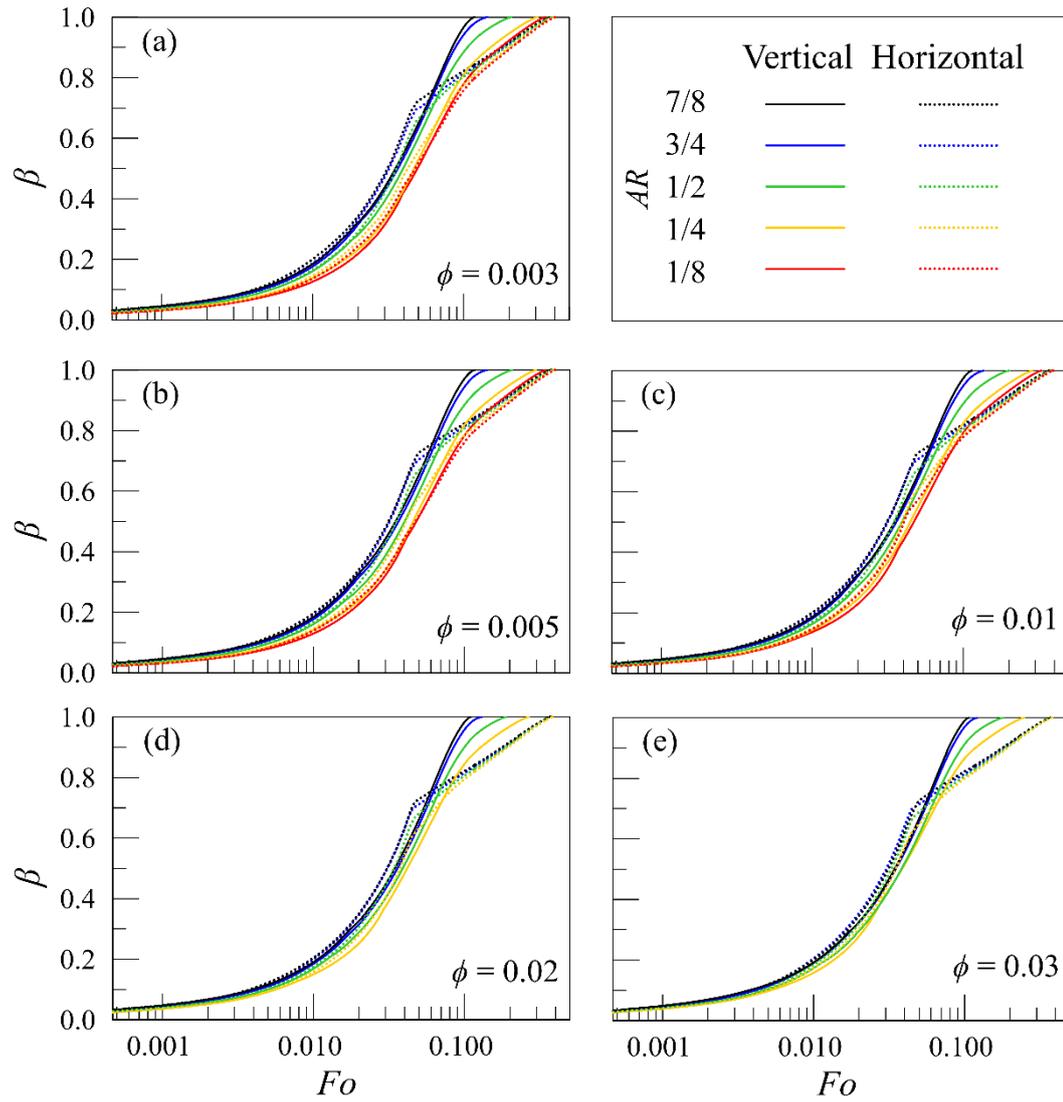


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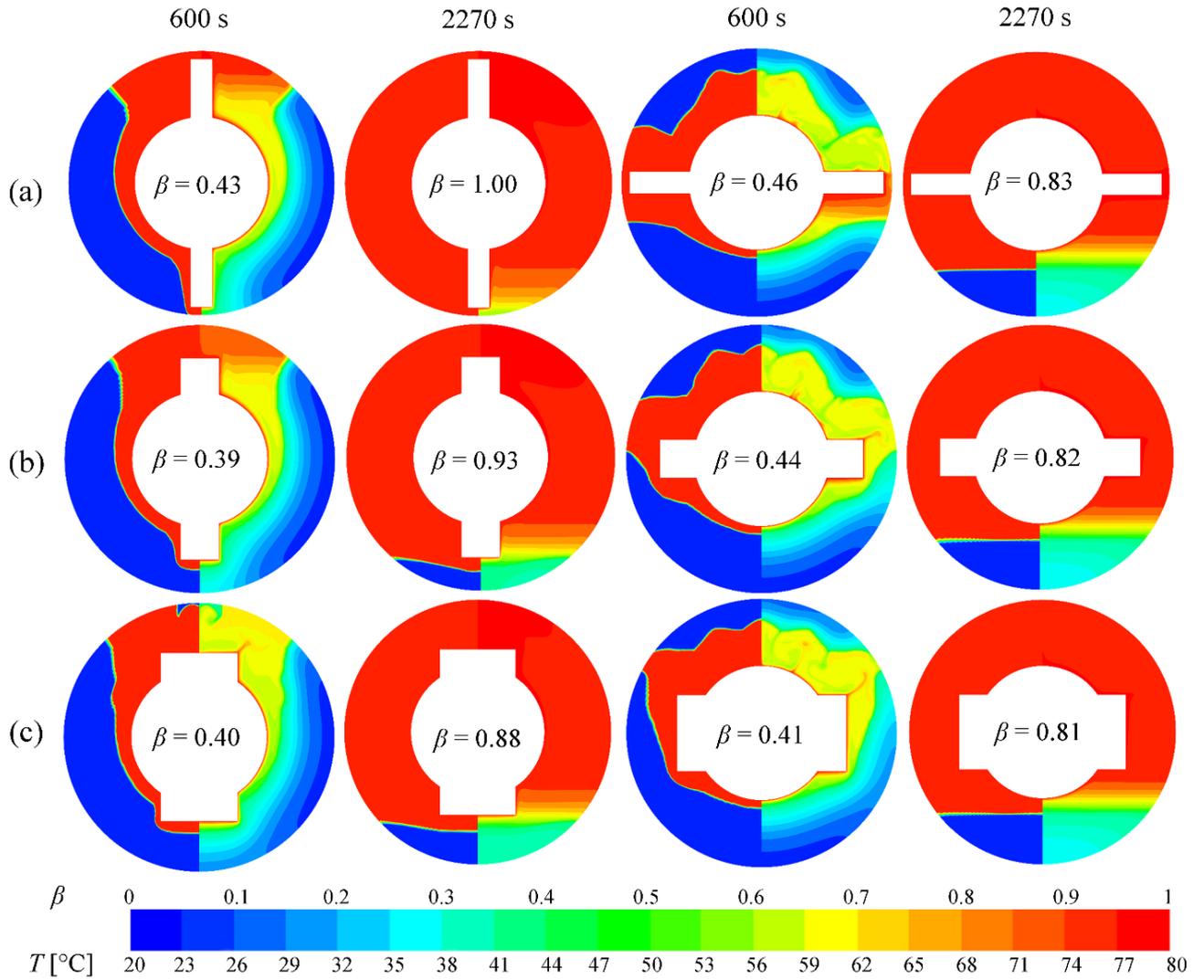
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Figure 8 – Solid-liquid interface for different moments of the melting process for $AR = 1/4, 1/2$ and $7/8$ and $\phi =$ (a) 0.003, (b) 0.005, (c) 0.01, (d) 0.02 and (e) 0.03.



581

582 Figure 9 – Liquid fraction (β) vs. Fo , for vertical and horizontal fins, with $AR = 7/8$,583 $3/4$, $1/2$, $1/4$, and $1/8$ for ϕ : (a) 0.003, (b) 0.005, (c) 0.01, (d) 0.02, and (e) 0.03.

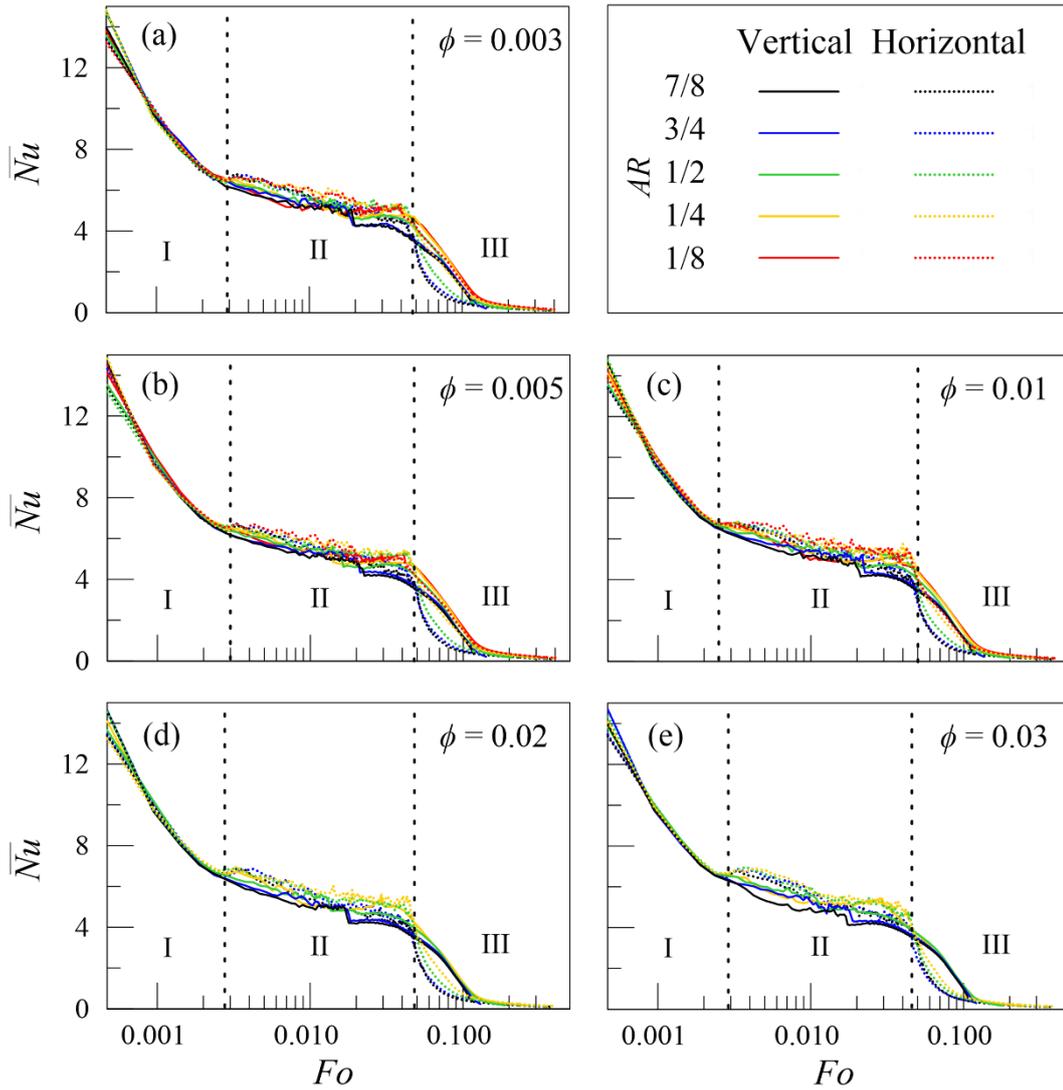


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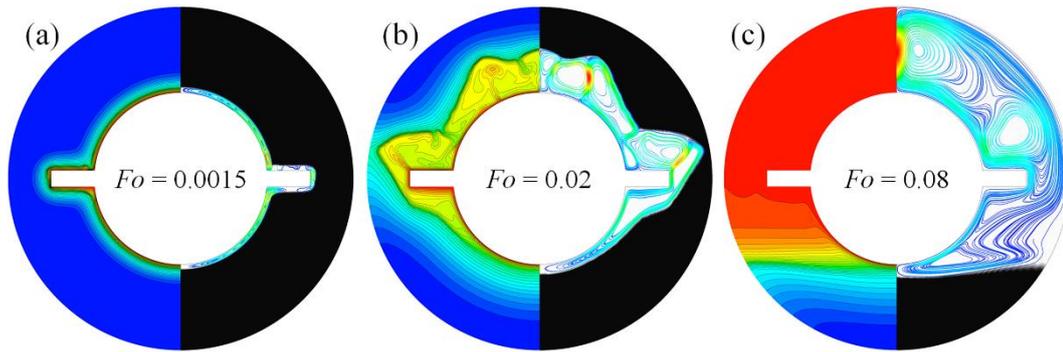
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Figure 10 – Fields of β (left) and T (right), at $t = 600$ and $2,270$ s, for vertical and horizontal fins, with $\phi = 0.03$ and AR : (a) $7/8$, (b) $1/2$, and (c) $1/4$.



587

588 Figure 4 – \overline{Nu} vs. Fo for vertical and horizontal fins, with $AR = 7/8, 3/4, 1/2, 1/4,$ and
 589 $1/8$ for ϕ : (a) 0.003, (b) 0.005, (c) 0.01, (d) 0.02, and (e) 0.03.



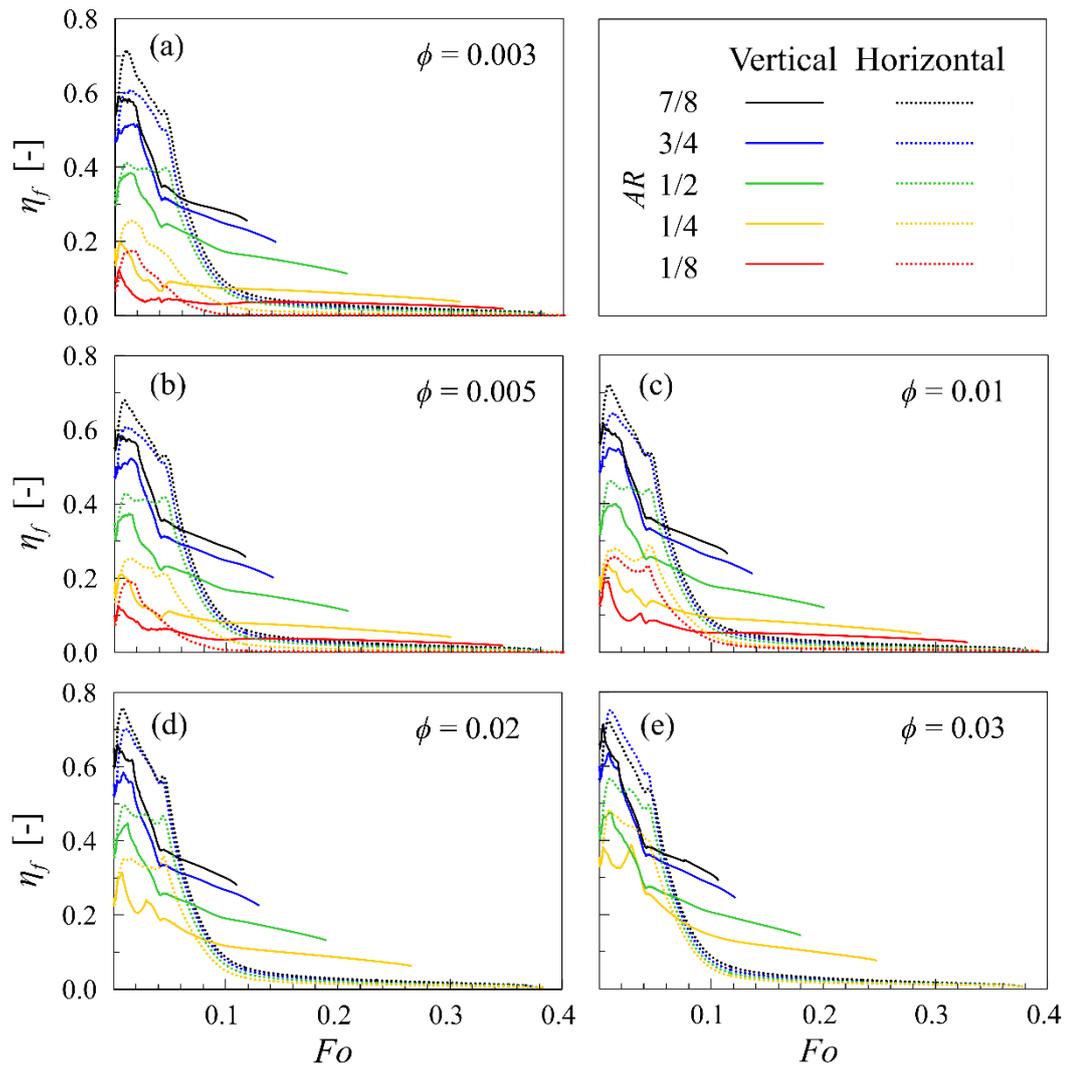
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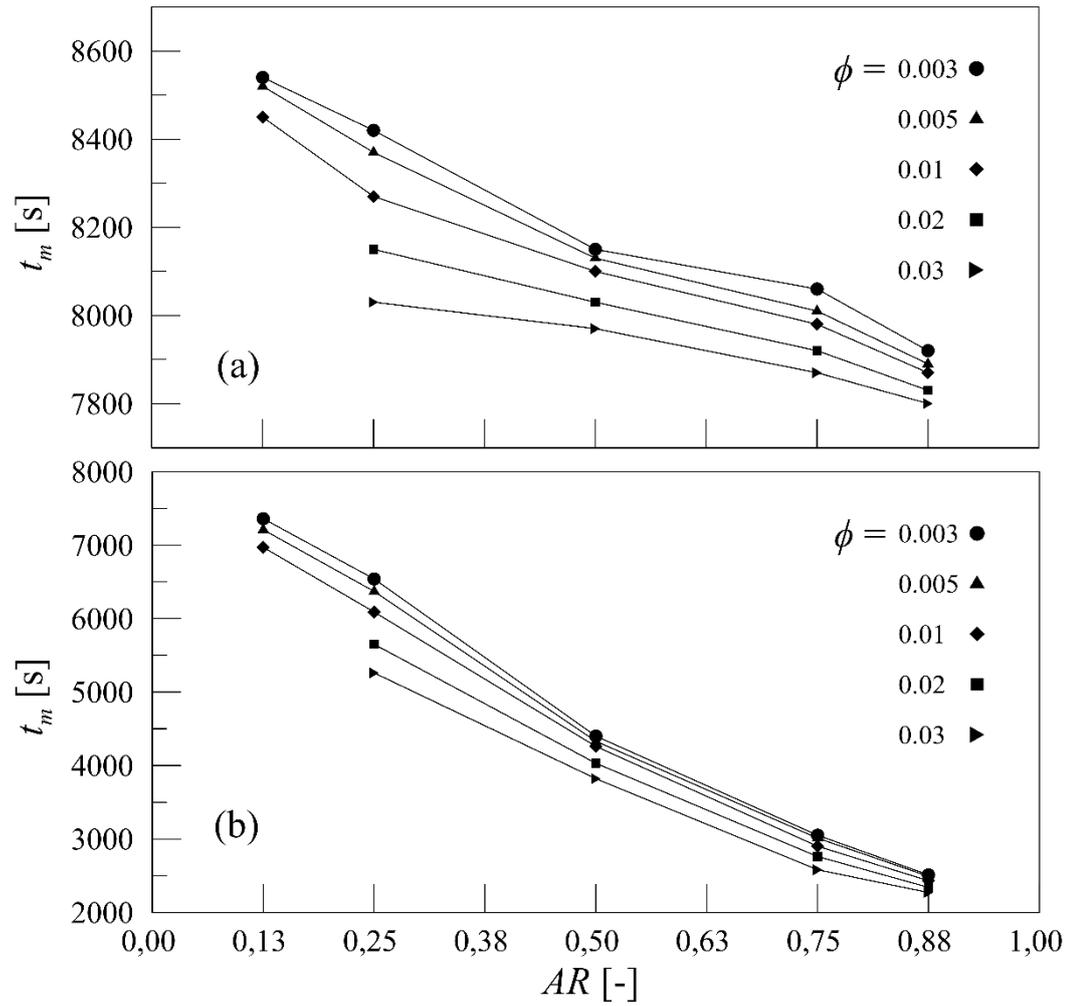
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Figure 12 - Fields of T (left) and streamlines (right), for horizontal fins, with $\phi = 0.01$, and $AR = 1/2$, at Fo : (a) 0.0015, (b) 0.02, and (c) 0.08.

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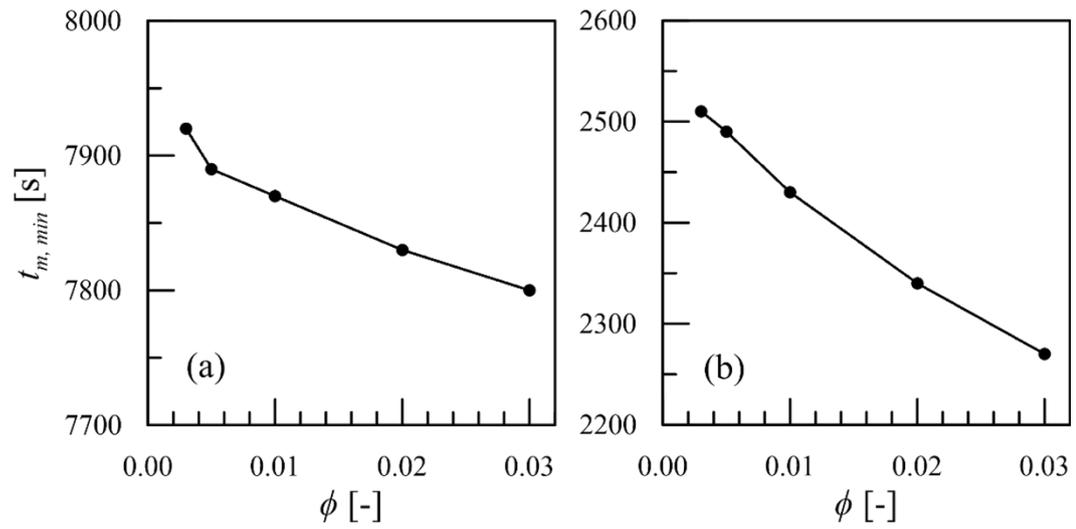


594 Figure 5 – Melting rate effectiveness (η_f) vs. Fo , for vertical and horizontal fins, with
 595 $AR = 7/8, 3/4, 1/2, 1/4,$ and $1/8$ for ϕ : (a) 0.003, (b) 0.005, (c) 0.01, (d) 0.02, and
 596 (e) 0.03.



597
598

Figure 6 – Total melting time (t_m) vs. AR for $\phi = 0.003, 0.005, 0.01, 0.02,$ and 0.03 :
(a) horizontal fins and (b) vertical fins.



599 Figure 7 – Minimum total melting time ($t_{m, min}$) vs. ϕ for: (a) horizontal fins and
600 (b) vertical fins, both for $AR = 0.88$ (7/8).

601
602

Table 1 – Fin dimensions.

| | | AR | | | | | |
|--------|-------|-------|-------|-------|-------|-------|-------|
| | | 1/8 | 1/4 | 1/2 | 3/4 | 7/8 | |
| ϕ | 0.003 | e_f | 4.00 | 2.00 | 1.00 | 0.67 | 0.57 |
| | | l_f | 2.50 | 5.00 | 10.00 | 15.00 | 17.50 |
| | 0.005 | e_f | 7.54 | 3.77 | 1.88 | 1.26 | 1.08 |
| | | l_f | 2.50 | 5.00 | 10.00 | 15.00 | 17.50 |
| | 0.01 | e_f | 15.08 | 7.54 | 3.77 | 2.51 | 2.15 |
| | | l_f | 2.50 | 5.00 | 10.00 | 15.00 | 17.50 |
| | 0.02 | e_f | 30.16 | 15.08 | 7.54 | 5.03 | 4.31 |
| | | l_f | 2.50 | 5.00 | 10.00 | 15.00 | 17.50 |
| | 0.03 | e_f | 45.24 | 22.62 | 11.31 | 7.54 | 6.46 |
| | | l_f | 2.50 | 5.00 | 10.00 | 15.00 | 17.50 |

603

Table 2 – Thermophysical properties of lauric acid (Yuan et al. [23])

| T_m | c_p | k | Z | L | ρ | μ | | |
|-------|---------------------------------------|--------------------------------------|--------------------|-----------------------|----------------------|---------------------------------------|---------|---------|
| [°C] | [J kg ⁻¹ K ⁻¹] | [W m ⁻¹ K ⁻¹] | [K ⁻¹] | [J kg ⁻¹] | [kg/m ³] | [kg m ⁻¹ s ⁻¹] | | |
| | | | | | 60 °C | 60 °C | 70 °C | 80 °C |
| 44.2 | 2300 | 0.147 | 0.000615 | 173800 | 863 | 0.00534 | 0.00427 | 0.00347 |