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A numerical model for pressure drop and flow distribution in a solar collector with U-connected absorber pipes

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Abstract

This study presents a numerical model calculating the pressure drop and flow distribution in a solar collector with U-type harp configuration in isothermal conditions. The flow maldistribution in the absorber pipes, caused by the different hydraulic resistances, was considered to evaluate the pressure drop across the collector. The model was developed in Matlab and is based on correlations found in literature for both friction losses and local losses, and was compared in terms of overall pressure drop against experimental measurements carried out on an Arcon Sunmark HT 35/10 solar collector at different flow rates and temperatures for water and water/propylene glycol mixture. For collector pressure drops higher than 1.4 kPa, the relative difference between the model and measurements was within 5% for water and 7% for water/propylene glycol mixture. For lower pressure drops the relative difference increased, but remained within the accuracy of the differential pressure sensor. The flow distribution was mainly affected by the flow regime in the manifolds. Turbulent regime throughout the manifolds entailed a more uniform distribution across the absorber pipes compared to laminar regime. The comparison between calculated flow distributions and results from previous literature showed a good agreement.

Keywords: solar collector, U-configuration, pressure drop, flow distribution.

1. Introduction

The pressure drop over a solar collector should be known and taken into account, when connecting collectors in a solar collector field. In fact, the pressure drop impacts the flow distribution throughout the field, affecting the heat transfer rate in the collectors and hence their efficiency, and determines the requirements of the pumps that need to be installed to supply the field in the most efficient way.

The standard norm ISO 9806 does not strictly prescribe a specific fluid when performing the pressure drop test of a solar collector. It states that “the fluid used in the collector for the test shall be water or a mixture water/glycol (60/40), or a mixture recommended by the manufacturer. The temperature of the fluid shall be (20 ± 2) °C” (ISO 9806). Therefore, if different fluids are used to test different collectors, the test results cannot be directly compared (Kovacs et al., 2012). Additionally, the conditions during the test may differ significantly from the actual operating conditions of the collector, both in terms of fluid type and operating temperature. For this reason, it could be useful to have a model able to derive the pressure drop of a collector for different temperatures and fluids, starting from a single test carried out in a specific operating condition.

When considering large flat plate collectors, the most common design is a number of parallel pipes connecting two manifolds. This configuration allows a high ratio between pipe length and absorber area, without too large pressure drops. On the other hand, the parallel pipe design has the disadvantage of non-uniform flow distribution in the different parallel pipes. The flow distribution in flat plate collectors with parallel pipes has been the topic of many investigations, as it may strongly affect the collector efficiency. In fact, several studies show that the efficiency decreases for higher flow maldistribution, as a consequence of non-uniform outlet temperatures for the different collector pipes. Chiou (1982) developed a method to determine how much the collector efficiency is deteriorated by flow maldistribution, but he did not calculate the flow distributions himself. He defines a *flow nonuniformity parameter* as the root mean square deviation of the absorber pipe flow rates and concludes that the deterioration of the collector efficiency is proportional to approximately the square of such parameter. Wang and Wu (1990) propose a discrete numerical model to predict the flow and temperature distribution in collector arrays with vertical pipes, both in U-type and Z-

47 type configuration, taking into account buoyancy force. The U-type array presents a higher maldistribution
48 and hence its efficiency is more penalized compared to the Z-type array. The flow rate in the absorber pipes
49 for the U-type array decreases monotonically with the distance from the manifold inlet. The same trend is
50 found by Jones and Lior (1994), who considered a single collector with vertical pipes, instead of an entire
51 array, and neglected buoyancy effect. The flow is assumed isothermal, as buoyancy and temperature
52 dependence of the thermophysical properties are expected to play a negligible role, as argued in the authors'
53 previous work (Jones and Lior, 1987). Weitbrecht et al. (2002) carried out both an experimental and
54 analytical study on the flow distribution in a Z-type collector in isothermal conditions. They investigated the
55 influence that the inset of a pipe into the manifold has on the pressure drop across the tee junction. However,
56 only laminar flow and one specific tee geometry were considered. Still, the effort of investigating pressure
57 drops in tees with inset is noteworthy. In fact, the pipe inset is often neglected or simplified in other studies,
58 but can play a role in the flow distribution by increasing significantly the pressure drop compared to tees
59 with sharp edges (Ohnewein et al., 2015).

60 Fan et al. (2007) studied the flow and temperature distribution in a large solar collector with 16 U-connected
61 horizontal tubes with quadrilateral cross section. A numerical model, based on CFD calculations and taking
62 into account buoyancy, and experimental measurements are compared. The results show that the flow
63 distribution is dominated by friction (and hence buoyancy can be neglected), if the velocity in the collector
64 pipes is high compared to the temperature rise across the collector. In large solar collector fields in Denmark
65 each row usually consists of 10 to 25 collectors (Windeleff and Nielsen, 2014), so the temperature rise in
66 each collector is relatively small, while the flow rate is relatively high. Consequently, it can be considered
67 that buoyancy plays a minor role in the flow distribution in this kind of installations. As the focus of the
68 present study was on large solar collectors for collector field applications, the assumption of no buoyancy
69 was made and the flow was assumed isothermal.

70 The aim of the study was to develop a numerical model for calculating the pressure drop and flow
71 distribution in a U-type harp collector under isothermal conditions. The model, whose source code is
72 publicly available online (Bava, 2015) and can be easily customized by other users, takes only a couple of
73 seconds to perform a simulation. The model is based on a very common design of solar collector, so it can be
74 used to evaluate geometry improvements to achieve more uniform flow distribution. Both laminar and
75 turbulent regimes were considered, and an effort was made to take into account the effect of the pipe inset
76 into the manifolds. The reliability of the model was verified by comparing its results against experimental
77 measurements and previous literature findings.

78 **2. Material and method**

79 *2.1. Numerical model*

80 *2.1.1 Pressure drop correlations*

81 In a solar collector, as well as in any other hydraulic circuit, the total pressure drop is given by the sum of
82 two types of pressure losses: friction (or major) losses and local (or minor) losses. Friction losses occur in
83 pipe flow because of viscous effects generated by the pipe surface. Local losses are due to variations of
84 momentum. Valves, bends, tees and abrupt changes in pipe cross section are examples of components
85 causing local pressure losses.

86 The friction loss along a straight pipe of constant cross section can be calculated by the Darcy-Weisbach
87 equation:

$$88 \quad \Delta p = \lambda \frac{l}{D_h} \frac{\rho w^2}{2} \quad (\text{eq.1})$$

89 where Δp is the pressure drop [Pa],

90 λ is the Darcy friction factor [-],

91 l is the pipe length [m],

92 D_h is the pipe hydraulic diameter, which equals the inner diameter for a full flow circular pipe [m],

93 ρ is the fluid density [kg m^{-3}],

94 w is the mean fluid velocity [m s^{-1}].

95 The friction factor depends on the type of flow regime and, if this is turbulent, on the roughness of the pipe
96 as well. The parameter indicating whether a flow is laminar or turbulent is the Reynolds number, defined as

$$97 \quad Re = \frac{w\rho D_h}{\mu} \quad (\text{eq.2})$$

98 where Re is the Reynolds number [-],

99 μ is the fluid dynamic viscosity [Pa s].

100 Density and viscosity are properties characteristic of each fluid and are dependent on the temperature. Water
101 and propylene glycol/water mixtures are the most common fluids used in solar thermal applications.

102 Thermophysical properties of water are well known and are easily found in literature. For the density, the
103 correlation (eq.3) proposed by Furbo (2015) was used, while the dynamic viscosity was evaluated through
104 equation (eq.4) proposed by Kestin (1978):

$$105 \quad \rho = 1000.6 - 0.0128 T^{1.76} \quad (\text{eq.3})$$

$$106 \quad \log\{\mu(T)/\mu(T=20\text{ }^\circ\text{C})\} = \{1.2378 - 1.303 \cdot 10^{-3} \cdot (20-T) + 3.06 \cdot 10^{-6} \cdot (20-T)^2 + \\ 107 \quad + 2.55 \cdot 10^{-8} \cdot (20-T)^3\} \cdot (20-T)/(T+96) \quad (\text{eq.4})$$

108 where T is the fluid temperature [$^\circ\text{C}$].

109 Given the large variability of the properties of propylene glycol/water mixtures found in literature and in
110 product datasheets, these were experimentally determined with an Anton Paar DMA 4100 densimeter and an
111 Anton Paar AMV 200 viscometer. Three samples with glycol concentration of 40%, 45% and 50% were
112 tested at temperatures between 20 $^\circ\text{C}$ to 80 $^\circ\text{C}$ with an intermediate step of 10 $^\circ\text{C}$. The experimental data
113 points were then interpolated with the polynomial expressions (eq.5) and (eq.6):

$$114 \quad \rho = 1013 - 0.2682 T + 0.7225 x - 1.94 \cdot 10^{-3} T^2 - 4.964 \cdot 10^{-3} x T \quad (\text{eq.5})$$

$$115 \quad \mu = (-2.881 - 6.721 \cdot 10^{-3} T + 0.2839 x + 1.959 \cdot 10^{-3} T^2 + \\ 116 \quad - 7.036 \cdot 10^{-3} x T - 1.883 \cdot 10^{-5} T^3 + 4.862 \cdot 10^{-5} x T^2) \cdot 10^{-3} \quad (\text{eq.6})$$

117 where x is the mass concentration of propylene glycol in the mixture [%].

118 Laminar flow regime is characterized by low values of Reynolds number. In literature it is often stated that,
119 for fully developed flow in a circular pipe, laminar flow occurs for $Re < 2300$, and turbulent flow for $Re > 4000$
120 (Holman, 2002). The flow regime between laminar and turbulent is referred to as transitional regime. In
121 reality, the exact value at which change in flow regime occurs is extremely difficult to determine and
122 depends on whether small disturbances are present. In the developed model, the flow was assumed laminar
123 for $Re < 2300$, and turbulent for $Re > 3100$, as a result of a series of tests carried out to evaluate transition in
124 flow regime in the HT collector pipes. More detailed information about how these tests were performed and
125 analyzed is reported in Appendix.

126 In case of laminar flow, the Darcy friction factor was calculated through the Hagen-Poiseuille law:

$$127 \quad \lambda = 64/Re \quad (\text{eq.7})$$

128 For turbulent flow the Blasius correlation for smooth pipes (eq.8) was used, as the collector manifolds and
129 pipes were made of commercial copper tubes, characterized by very small absolute roughness (Binder,
130 1973).

$$131 \quad \lambda = 0.3164/Re^{0.25} \quad (\text{eq.8})$$

132 Beside the Blasius correlation, other friction factor correlations were implemented in the model, such as
133 those proposed by Colebrook (1939) and Haaland (1983) for turbulent flow in pipes of known roughness,
134 and by Joseph and Yang (2010) for any flow regime in smooth pipes.

135 Following the example of Jones and Lior (1994), the friction factor in the transition region ($2300 < Re < 3100$)
136 was calculated by linear interpolation between the value obtained from (eq.7) for $Re = 2300$ and that obtained

137 from (eq.8) for $Re=3100$.

138 Regarding local losses, in the solar collector under investigation the only discontinuities were represented by
139 the tees connecting the absorber pipes to the manifolds. The correlations used to model the pressure losses in
140 tees were mainly obtained by Idelchik (1994). For the sake of simplicity, these correlations are not reported
141 here, but can be found as part of the code of the developed model (Bava, 2015). Though, Idelchik's
142 correlations refer to tees with sharp edges and without any inset, while the collector under investigation had
143 2-3 mm inset of the absorber pipes into the manifold. As even short insets have been proven to be able to
144 affect the pressure drop across tees (Ohnewein et al., 2015), Idelchik's correlations were corrected based on
145 the results from Ohnewein et al. (2015). In Ohnewein et al. (2015) Figures 33-38 present the measured
146 pressure drop coefficients in different types of tee junctions for different inset lengths and different flow rate
147 ratios between side passage and combined passage of the tee junction. In the same diagrams also Idelchik's
148 pressure drop coefficients for turbulent conditions are reported, even when the Reynolds number in the
149 combined passage of the tee junction is very low (for example $Re=500$ in Figure 37). For this reason, in order
150 to make a fair comparison, the coefficients proposed by Ohnewein et al. were compared to those from
151 Idelchik, using turbulent or laminar correlations accordingly. Among the different pressure drop coefficients
152 proposed by Ohnewein et al., those with an inset length/manifold diameter ratio closer to that of the
153 investigated collector (see Section 2.2) were chosen. Based on the comparison, the following conclusions
154 were drawn:

- 155 • good agreement was found between Ohnewein et al.'s and Idelchik's pressure drop coefficients in
156 case of straight passage in a diverging tee junction both in laminar and turbulent conditions;
- 157 • good agreement was found in case of side passage in a diverging tee junction in laminar conditions,
158 assuming that at $Re=3500$ in the combined passage of the tee junction the flow is still laminar;
- 159 • in case of side passage of a diverging tee junction in turbulent conditions, Ohnewein et al.
160 measured pressure drop coefficients about 0.75 times those predicted by Idelchik;
- 161 • in case of straight passage of a combining tee junction in turbulent conditions, Ohnewein et al.
162 measured pressure drop coefficients about 2.2 times higher than those predicted by Idelchik.

163 Hence, Idelchik's correlations were corrected accordingly and implemented in the model. Given the collector
164 geometry, the above mentioned pressure drop coefficients for isolated tee junctions could be used to evaluate
165 the pressure losses along the manifolds, which, according to Miller (2009), is allowed when the distance
166 between two consecutive tees is three times longer than the manifold diameter.

167 2.1.2 Matlab implementation

168 Using the pressure drop correlations described in the previous section, a numerical model was developed in
169 Matlab to compute the flow distribution and the resulting pressure drop across a collector. The input data to
170 the model are the dimensions of the collector hydraulics and operating conditions of the heat transfer fluid
171 (glycol content, flow rate and fluid temperature). The fluid temperature, which is assumed to be constant
172 throughout the collector, is used to determine the density and viscosity of the fluid. Additionally, the flow is
173 considered to be fully developed.

174 Given an initial flow rate as input, the model assumes that the flow is uniformly distributed in all absorber
175 pipes. A uniform flow distribution would cause the pressure drop to increase from one strip to the next (see
176 strip numbering in Figure 1), as the fluid path becomes longer due to additional manifold segments. In
177 reality, the pressure drop must be the same, irrespective of the path the fluid follows, resulting in an
178 adjustment of the flow rate in each absorber pipe. In order to calculate the true flow distribution, the
179 numerical model solves iteratively the set of equations (eq.9), which impose both the conservation of mass
180 across the collector (first line in (eq.9)) and the uniformity of pressure drop along the different hydraulic
181 paths (from second line downward in (eq.9)):

$$182 \begin{pmatrix} 1 & 1 & 1 & 1 & 1 & 1 \\ k_{1,j} \cdot \dot{m}_{1,j} & -k_{2,j} \cdot \dot{m}_{2,j} & 0 & 0 & 0 & 0 \\ 0 & k_{2,j} \cdot \dot{m}_{2,j} & -k_{3,j} \cdot \dot{m}_{3,j} & 0 & 0 & 0 \\ \vdots & \vdots & \vdots & \vdots & \vdots & \vdots \\ 0 & 0 & 0 & 0 & k_{N-1,j} \cdot \dot{m}_{N-1,j} & -k_{N,j} \cdot \dot{m}_{N,j} \end{pmatrix} \begin{pmatrix} \dot{m}_{1,j+1} \\ \dot{m}_{2,j+1} \\ \dot{m}_{3,j+1} \\ \vdots \\ \dot{m}_{N,j+1} \end{pmatrix} = \begin{pmatrix} \dot{m}_{tot} \\ 0 \\ 0 \\ \vdots \\ 0 \end{pmatrix} \quad (\text{eq.9})$$

183 where \dot{m} is the mass flow rate [kg s^{-1}],
184 k is a hydraulic resistance coefficient [$\text{kg}^{-1} \text{m}^{-1}$], defined so that the product ($k_i \cdot \dot{m}_i^2$) corresponds
185 to the pressure drop (in Pascal) across the i -th hydraulic path. The factor k_i is representative of the
186 pressure drop along the i -th absorber pipe, which is proportional to the square of the flow rate \dot{m}_i , as
187 well as the pressure drop across the manifold segments and tees included in the i -th hydraulic path,
188 properly normalized to the flow rate \dot{m}_i ,
189 1, 2, ..., N subscripts denote the hydraulic path and N is the total number of hydraulic paths,
190 tot subscript refers to the total flow rate supplied to the collector,
191 j subscript denotes the iteration number.

192 The iterative procedure is continued until the difference in the absorber pipe flow rates between two
193 consecutive iterations is lower than 0.1%. With this convergence constraint a typical calculation is performed
194 in a couple of seconds for a computer with quad-core CPU, 2.4 GHz CPU frequency and 8 GB memory.

195 The entire Matlab code used to carry out the present study is publicly available online (Bava, 2015).

196 2.2. Experimental setup for pressure drop measurement

197 The developed model was based on the design of U-type large scale collectors. These collectors have a
198 number of horizontal absorber pipes welded to absorber strips and connecting two vertical manifolds located
199 along the sides of the collector with their outlet at the top corners (Figure 1).

200 The selection of this design was motivated by the fact that this is the most frequently adopted when
201 manufacturing large collectors for solar heating field applications. In fact, these collectors are easy to
202 assemble and quick to connect to one another when installed side by side in a field.

203 *Figure 1: Sketch of a U-type absorber with horizontal absorber pipes (adapted from Fan et al., 2007).*

204 To validate the model in terms of overall collector pressure drop, the results were compared to measurements
205 carried out on a large scale collector in different operating conditions. The tested collector was an Arcon
206 Sunmark HT-SA 35/10 module, having an aperture area of 12.60 m^2 . The collector piping was made of
207 copper and had circular cross sections. The 18 absorber pipes were 5.80 m long, had an inner diameter of 9.1
208 mm and an intermediate spacing of 122 mm. The manifolds had an inner diameter of 32.9 mm.

209 A differential pressure sensor TA-SCOPE from IMI Hydronics was used to measure the pressure drop across
210 the collector. The instrument nominal accuracy is given by the higher value between 0.1 kPa and 1% of the
211 measured value.

212 The volume flow rate supplied to the collector was measured by a Kamstrup MP115 electromagnetic flow
213 meter. Its accuracy is stated to be within $\pm 0.50\%$. Additionally, the instrument was calibrated with water and
214 propylene glycol/water mixture at different fluid temperatures and flow rates, and its nameplate calibration
215 factor was confirmed.

216 Transparent plastic tubes were connected to the inlet and outlet of the collector at one end and to the pressure
217 sensor at the other end, as shown in Figure 2. The transparent pipes made it easier to verify that no air was
218 present in the circuit, which otherwise would alter both the flow rate and differential pressure measurements.
219 As the plastic pipes could not be connected exactly at the inlet and outlet of the solar collector (Figure 3), the
220 pressure losses occurring in between needed to be evaluated, in order not to assign them to the collector.

221 *Figure 2: Back of the HT-SA 35/10 solar collector with plastic pipes arrangement for pressure drop measurement.*

222 *Figure 3: Detail of collector outlet with connection for the differential pressure sensor.*

223 Fluid temperature was measured both at the inlet and outlet of the collector by type T thermocouples
224 (copper/constantan junction).

225 Pressure drop measurements were carried out for flow rates between 0.08 and 0.72 liters s^{-1} (0.3 - 2.6 $\text{m}^3 \text{h}^{-1}$)
226 and at temperatures of approximately 25 °C, 55 °C and 70 °C, with water and a 50% propylene glycol/water
227 mixture as heat transfer fluid. The tests were performed in cloudy sky conditions, so that the temperature rise
228 across the collector was negligible and the assumption of isothermal flow introduced in the model was

229 fulfilled.

230 3. Results

231 3.1. Collector pressure drop

232 The comparison between the measured and calculated pressure drops as function of the flow rate is shown in
233 Figure 4 and Figure 5. Given the configuration of the hydraulic circuit, it was possible to test volume flow
234 rates up to $2.6 \text{ m}^3 \text{ h}^{-1}$, corresponding to pressure drops of approximately 9-10 kPa. The lower boundary of the
235 tested flow rate was determined by the error of the differential pressure sensor, which became increasingly
236 predominant when measuring pressure drops lower than 1 kPa. The error bars for the measured pressure
237 drops represent the maximum value between the accuracy of the differential pressure sensor (i.e. the higher
238 value between 0.1 kPa and 1% of the measured value) and the standard deviation of the measured values. For
239 sake of simplicity of the diagrams, the error bars relative to the measured flow rates are not shown, as they
240 are much smaller compared to the uncertainty of the differential pressure measurements. In fact, for all tested
241 flow rates, the highest of the ratios between standard deviation and mean value was 0.43%, and most of them
242 were lower than 0.20%. Consequently, they were always lower than the nameplate accuracy of the flow
243 meter ($\pm 0.50\%$) and more than twice as small as the accuracy of the differential pressure sensor.

244 *Figure 4: Comparison between measured and calculated pressure drops for water.*

245 *Figure 5: Comparison between measured and calculated pressure drops for 50% propylene glycol/water mixture.*

246 The pressure drop values given by the model matched the measured values with a reasonable accuracy. For
247 values higher than 1 kPa, the highest relative difference between model and measurements was equal to 5%
248 for water. Regarding the glycol/water mixture, the highest relative difference was 10% for a measured
249 pressure drop of 1.37 kPa, while in the other cases it was within 7%.

250 For values lower than 1 kPa, the relative differences were slightly higher. The highest deviation (15%) was
251 found for the lowest pressure drop (0.2 kPa). However, this deviation was within the accuracy range of the
252 differential pressure sensor (0.1 kPa). Apart from this, no particular trend of discrepancy could be identified
253 with respect to flow rate, temperature or fluid type, as both positive and negative deviations are found (see
254 Figure 4 and Figure 5). Figure 4 shows that for water at high flow rates ($V \geq 1.5 \text{ m}^3 \text{ h}^{-1}$ at $25 \text{ }^\circ\text{C}$ and $V \geq 2.0 \text{ m}^3$
255 h^{-1} for at $55 \text{ }^\circ\text{C}$ and $70 \text{ }^\circ\text{C}$) the model underestimated the actual pressure drop by 3%-5%. An
256 underestimation of the friction factor in the corresponding range of Reynolds number ($Re_{\text{pipe}} > 5200$) was at
257 first assumed as possible cause of these deviations. However, other data points in the same range of
258 Reynolds number did not show similar deviations. Additionally, in order to increase the friction factor for
259 this range of Reynolds number in such a way that these deviations were minimized, the roughness of copper
260 pipes should have been assumed 10 times higher than that usually reported in literature (Binder, 1973). Such
261 a correction did not seem acceptable, also based on the results of some surface roughness tests carried out on
262 similar collector pipes. Hence, the assumption of smooth pipes was maintained.

263 Beside the accuracy of the instruments, other sources of error might be the assumptions made in the model,
264 such as linear interpolation between laminar and turbulent conditions in case of transitional regime and fully
265 developed flow throughout the pipes.

266 In both diagrams, the typical quadratic relation between pressure drop and flow rate can be observed, as the
267 different groups of points are approximately aligned along parabolic trajectories. Additionally, the influence
268 of the fluid temperature on the pressure drop across the collector can be noted by comparing the different
269 series of points within the same diagram. As expected, a fluid flow at lower temperature caused a higher
270 pressure drop for the same flow rate.

271 3.2. Flow distribution

272 The flow distribution was expressed in terms of dimensionless flow rate in the absorber pipes, defined by
273 (eq.10) as:

$$274 \quad V'_i = \frac{V_i}{(\sum_{i=1}^N V_i)/N} \quad (\text{eq.10})$$

275 where V'_i is the dimensionless flow rate flowing in the i -th absorber pipe [-],

276 V_i is the volume flow rate in the i -th absorber pipe [$\text{m}^3 \text{h}^{-1}$],

277 N is the number of absorber pipes [-].

278 3.2.1. Comparison with previous literature

279 As no validation in terms of flow distribution could be performed in the test rig, the model results were
280 compared with data found in literature. Results from the simulation model of Jones and Lior (1994) were
281 chosen for comparison, because of the similarity in collector design as well as the identical assumption of
282 isothermal flow. In Jones and Lior's paper, model results obtained for a U-type harp collector having 8 or 16
283 absorber pipes are presented. The absorber pipes are 1.83 m long and can have a diameter of 0.25, 0.50 and
284 0.75 times the manifold diameter, which is 2.54 cm. The manifolds length is 0.915 m. As heat transfer fluid
285 water at 60 °C is assumed and the flow rate is varied in such a way that the Reynolds number at the manifold
286 inlet is 3210, 9640 and 16100 in different cases. Darcy friction factors are evaluated using Colebrook
287 equation (Colebrook, 1939) for $Re > 3000$ and Hagen-Poiseuille law (eq.7) for $Re < 2100$. Linear interpolation
288 between the two is used for the transition regime. The roughness of the pipe material is not specified in the
289 paper, but it is said to be intermediate between that for clean copper and steel. A value of 0.016 mm was
290 therefore selected in the model proposed in the present paper. Regarding tee junctions, the schemes reported
291 by Jones and Lior imply that sharp edge tees were considered. The pressure change across the tee junctions is
292 evaluated based on momentum conservation and constant pressure regain coefficients ($\gamma = 0$ for converging
293 tee junction and $\gamma = 0.9$ for diverging tee), with no distinction between laminar and turbulent regime.
294 Additionally, the energy loss coefficient at the entrance and exit of the absorber pipes is assumed to be
295 constant and equal to 1.2. The last two assumptions may not be very accurate, as pressure drops in tees are
296 shown to be very sensitive to the tee geometry and flow characteristics (Idelchik, 1994; Ohnewein et al.,
297 2015). However, to make the comparison between the two models fair, the same assumption was made in the
298 proposed model and the pressure drop correlations for tees in turbulent regime were used also in the laminar
299 region.

300 The comparison between the flow distributions found by Jones and Lior and those given by the proposed
301 model are shown in Figure 6 for two different cases. The first refers to a collector with 8 pipes and a ratio
302 between pipe and manifold diameters of 0.50. The second case refers to a collector with 16 pipes and a
303 diameters ratio of 0.25. This resembles more closely the geometry of a large scale collector on which the
304 present model was based. In both cases the Reynolds number at the manifold inlet was 9640.

305 *Figure 6: Comparison in flow distribution between Jones and Lior's model and the proposed model.*

306 In Figure 6 it can be seen that the two couples of profiles presented a very similar trend, with the presented
307 model predicting a more uniform flow distribution compared to Jones and Lior's. The difference was
308 relatively small, with maximum deviations of 5% and 1% in case of 8 and 16 pipes respectively, and it was
309 most likely due to the different correlations used to estimate the energy losses in the tee junctions.

310 Another difference between the two models was the trend of the curves in the bottom pipes. Due to the
311 assumption made by Jones and Lior about the constant pressure regain coefficients, the profiles returned by
312 their model were strictly decreasing. On the other hand, the pressure drop coefficients from Idelchick (1994)
313 used in the proposed model allowed a slight increase in flow in the last pipes, as shown in the case with $N=8$
314 in Figure 6. This happened because the pressure drop across the side passage of diverging tee junctions
315 decreased with the reduction in the manifold flow rate more significantly than the increase in hydraulic
316 resistance given by the other hydraulic components (tee junctions and manifold segments).

317 A good agreement between the two models was found for most of the other combinations of Reynolds
318 number and diameters ratio proposed by Jones and Lior in their paper, with maximum deviations of 5%.
319 Exceptions were the results obtained for a diameters ratio of 0.75. In this case, Jones and Lior's model
320 predicted a much stronger non-uniformity, with the top pipe having a flow rate up to four times higher than
321 in the bottom pipe. In the same conditions, the proposed model predicted a ratio between of the two flow
322 rates of approximately two. The difference between the models was caused by the different correlations used
323 to evaluate the pressure drops in the tee junctions. In case of a large diameters ratio, the pressure drop across
324 the absorber pipes is not predominant any more, and the pressure drops across the tee junctions play a more
325 important role in determining the flow distribution.

326 3.2.2 Effect of flow regime on flow distribution

327 Using the developed model, it was possible to calculate the flow distribution inside the HT 35/10 collector at
328 different flow rates and fluid types. Two cases were considered, one with water at 70 °C (Figure 7) and the
329 other with water at 20 °C (Figure 8). Due to the different fluid viscosity at different temperatures, these two
330 cases allowed to study the flow distribution at similar flow rates, but in different flow regimes.

331 The flow distributions shown in Figure 7 and Figure 8 decrease from the top to the bottom pipe, but with
332 different profiles. In case of water at 70 °C (Figure 7), the distribution was more uniform, especially at flow
333 rates higher or equal to 1.5 m³ h⁻¹, for which the parameter V' ranged from 0.92 to 1.09. At these flow rates
334 the flow across the absorber pipes and in the largest part of the manifold was turbulent. At a flow rate of 0.7
335 m³ h⁻¹, the flow regime in the manifold was turbulent in the first 15 pipes and consequently the flow
336 distribution was still roughly uniform. Conversely, in the last three manifold segments the flow regime
337 switched to laminar, causing a stronger decrease of the flow rates in the last absorber pipes and a wider
338 spread of the V' values.

339 *Figure 7: Flow distribution in the absorber pipes at different flow rates for water at 70 °C.*

340 *Figure 8: Flow distribution in the absorber pipes at different flow rates for water at 20 °C.*

341 In case of water at 20 °C (Figure 8), the laminar regime in the pipes and most of the manifolds caused a
342 much less uniform flow distribution. This is particularly evident at the lower flow rates of 0.5 and 1 m³ h⁻¹,
343 where V' varied more strongly, in a range between 0.66 and 1.25. For higher flow rates the portion of
344 manifolds experiencing turbulent conditions increased and the flow distribution became more uniform,
345 similar to the profiles shown in Figure 7.

346 Through the developed model, it was also possible to study the contribution to the total pressure drop of the
347 single hydraulic components, such as absorber pipes, manifolds and tee junctions. Although the pressure
348 drop across the absorber pipes played the most significant role in both cases, its importance for the different
349 paths was quite different, depending on the flow regime. In case of turbulent regime in all absorber pipes, the
350 pressure drop along these represented 86%-90% of the total collector pressure drop for the top pipe, and
351 66%-71% for the bottom pipe. The remaining part was caused by forward and return manifolds as well as tee
352 junctions. In case of laminar regime, the pressure drop in the absorber pipe was 87%-92% of the total for the
353 top pipe, and 44%-50% for the bottom pipe.

354 4. Discussion

355 4.1 Pressure drop

356 As expected, when testing the two heat transfer fluids at the same temperature and flow rate, the pressure
357 drop for the glycol/water mixture was larger than for water, because of the much higher viscosity. For the
358 same reason, the pressure drop for both fluids was higher at lower temperatures. Nevertheless, this effect was
359 much more relevant for the 50% glycol/water mixture than for water. This was due to two main factors.
360 Firstly, the kinematic viscosity of water decreases by a factor of 2.7 when the temperatures increases from 20
361 °C to 80 °C, while that of a 50% glycol/water mixture decreases by a factor of 5 for the same temperature
362 variation. Secondly and more importantly, the higher viscosity of the glycol/water mixture at about 20 °C
363 caused the flow regime in the absorber pipes to be laminar, even at the highest tested flow rate of 2.5 m³ h⁻¹.
364 This entailed much higher friction factors compared to those obtained at higher temperatures.

365 Despite its secondary importance, the higher density also contributed in increasing the pressure drop at lower
366 temperatures. The effect was more important for the glycol/water mixture, as its density varies by about 4%
367 in the range 20 °C-80 °C, compared to 2% for water.

368 It can also be noted that the pressure drop for glycol/water mixture at 70 °C was very similar to that of pure
369 water at 20 °C. This was due to the fact that the two fluids have very similar kinematic viscosity and density
370 in these conditions ($\nu \approx 1 \cdot 10^{-6}$ m² s⁻¹, $\rho \approx 1000$ kg m⁻³). This means that the pressure drop curves for
371 glycol/water mixtures at relatively high temperature may be indirectly evaluated using water at sufficiently
372 low temperature.

373 4.2 Flow distribution

374 The results given by the model showed a decrease in flow rate from the top to the bottom pipes, because only
375 friction was considered as driving force, so that the longer the hydraulic path, the higher the resistance. This
376 is in agreement with the findings of Wang and Wu (1990), Jones and Lior (1994) and Fan et al. (2007).

377 A large difference appeared to exist when comparing flow distributions characterized by a different flow
378 regime. Turbulent flow in the manifolds entailed relatively small pressure losses both in manifold segments
379 and tees, compared to those occurring across the absorber pipes. So, these additional pressure drops could be
380 compensated by a slight unbalance in the collector flow distribution. This is shown in Figure 7 by the flow
381 distributions referring to the flow rates of 2.5 and 3.5 m³ h⁻¹. The extent of the unbalance in flow distribution,
382 which was needed to compensate the pressure drop caused by manifolds segments and tees, was also affected
383 by the flow regime in the absorber pipes. Due to the geometry of the HT 35/10 collector, the flow regime in
384 all absorber pipes was turbulent, when it was turbulent in the entire manifold. Given the weak dependence of
385 the friction factor in the turbulent region on the Reynolds number (see (eq.8)), the friction pressure drop in
386 the absorber pipes can be considered proportional to the square of the fluid velocity, so that small variations
387 in flow rate cause larger variations in pressure drop. For this reason, a slight unbalance in flow distribution is
388 sufficient to give the right pressure drop along each hydraulic path.

389 Figure 7 also shows that when turbulent regime was established throughout the collector, then the flow
390 distribution was not significantly affected by the total flow rate, which is in agreement with previous findings
391 (Jones and Lior, 1994).

392 At lower flow rates ($V \leq 1.5$ m³ h⁻¹ in Figure 7), only part of the manifolds was in turbulent conditions, so that
393 laminar equations applied to the last manifold segments and tees. As both pressure drop coefficients for tees
394 and friction factor coefficients are much higher in laminar regime than in turbulent regime, the flow rate in
395 the last absorber pipes needed to diminish abruptly in order to cause the same pressure drop as the previous
396 hydraulic paths, causing a higher flow non-uniformity.

397 On the other hand, when using water at 20 °C (Figure 8), the higher viscosity resulted into laminar regime in
398 the absorber pipes and in larger part of the manifolds. At lower flow rates ($V \leq 1$ m³ h⁻¹), the flow regime in all
399 absorber pipes was laminar, while it was initially turbulent in the manifold and then, as more fluid was
400 diverted to the absorber pipes, laminar. The precise tee junction after which the change in flow regime
401 occurred can be identified looking at the change in slope in the curves in Figure 8: a milder slope
402 corresponds to turbulent flow in the manifold, while a steeper slope to laminar flow. The flow distribution at
403 these flow rates was much less uniform than in the turbulent case (Figure 7). The main reason for this
404 behavior was that the local loss coefficients for tees in laminar regime are more sensitive to flow conditions
405 than in turbulent regime, so they varied more significantly from one pipe to the next. As the hydraulic
406 resistance between two consecutive pipes differed more significantly, this needed to be compensated by a
407 larger difference in flow rates. A secondary reason is the linear dependence of the friction pressure drop on
408 the fluid velocity in laminar conditions (see (eq.1) and (eq.7)), which, compared to the quadratic dependence
409 in turbulent conditions, required larger variations in flow rate to compensate the varying pressure drop across
410 the absorber pipes.

411 The distribution became more uniform at higher flow rates, as a longer part of the manifolds experienced
412 turbulent conditions and the regime inside the absorber pipes became transitional ($V=1.5$ m³ h⁻¹) and then
413 turbulent ($V=2$ m³ h⁻¹).

414 The fact that the modeled flow distributions become more uniform at higher flow rates seems to be in
415 disagreement with previous literature. For example, Jones and Lior (1994) show that the flow distribution in
416 a harp collector (both in Z- and U-configuration) is approximately independent of the Reynolds number at
417 the manifold inlet, and hence of the total flow rate. Only for the highest ratio between absorber pipe and
418 manifold diameters ($d/D=0.75$), the flow distribution at the lowest flow rate ($Re_{inlet}=3210$) was slightly more
419 uniform than at higher flow rates ($Re_{inlet}=9640$ and $Re_{inlet}=16100$). The reason for the different trend found
420 through the proposed model was mainly the use of different correlations for tee junctions, depending on the
421 flow regime. As an example, Figure 9 shows the modeled flow distributions in a HT-SA 35/10 collector at
422 different manifold inlet Reynolds numbers, using turbulent correlations for tee junctions regardless of the
423 actual flow regime. In this case the trend of the flow distributions was in agreement with Jones and Lior
424 (1994). However, given the good agreement between the measurements from Ohnewein et al. at low

425 Reynolds number and Idelchik's laminar correlations, it was decided to keep the latter in the model.

426 *Figure 9: Flow distribution in the absorber pipes of a HT-SA 35/10 collector for different manifold inlet Reynolds numbers.*

427 As mentioned, isothermal flow was assumed in the model. In real operating conditions, the temperature of
428 the heat transfer fluid would increase along the absorber pipes. If the temperature rise is relatively small, the
429 fluid properties do not change significantly and using the mean fluid temperature is an acceptable
430 simplification. Nevertheless, in case of uneven flow distribution and uniform solar irradiance on the
431 collector, a pipe with a lower flow would experience a larger temperature rise and a higher mean fluid
432 temperature. The fluid viscosity would be lower, and thereby the friction factor and hydraulic resistance
433 would be smaller. Consequently, the flow distribution in case of non-isothermal conditions would partly
434 balance itself, resulting in a more uniform distribution compared to isothermal conditions. The effect is
435 expected to be more relevant in case of laminar flow, because of the stronger flow maldistribution (Figure 8)
436 and the direct proportionality between friction factor and fluid viscosity (see (eq.2) and (eq.7)).

437 Flow maldistribution in a harp collector causes a decrease in the collector efficiency. In fact, absorber pipes
438 having lower flow rates and hence higher temperatures suffer higher heat losses. The warmer fluid at the exit
439 of these pipes is tempered down by the colder fluid coming from the absorber pipes supplied with higher
440 flow rates. The mixing of flows at different temperatures causes a lower temperature rise across the collector
441 compared to the case with uniform flow distribution and identical temperature rise across each absorber pipe.
442 Consequently, the overall collector efficiency decreases. The negative effect of the flow maldistribution on
443 the collector efficiency is increased primarily by the degree of maldistribution and secondly by the collector
444 heat loss coefficient. The quantitative relationship between flow maldistribution and collector efficiency
445 degradation is out of the scope of this investigation. However, an estimation of this effect can be found in
446 Chiou (1982). In fact, defined a *flow nonuniformity parameter* as the root mean square deviation of the
447 absorber pipe flow rates, the deterioration of the collector efficiency is expected to be proportional to
448 approximately the square of such parameter.

449 **5. Conclusions and future perspective**

450 A model for estimating the pressure drop and flow distribution across a solar collector with U-type
451 configuration in isothermal conditions was developed. The pressure drops calculated by the model were
452 compared to measurements carried out on a solar collector. The flow distributions were compared to
453 previous literature findings.

454 For pressure drops higher than 1 kPa, all relative differences between model and measurements were within
455 $\pm 7\%$, apart from one point. For lower pressure drops the relative difference increased, but always within the
456 accuracy of the differential pressure sensor.

457 Flow rate and viscosity were the main factors influencing the pressure drop, so different fluids having similar
458 values for these two parameters gave almost identical pressure drops.

459 The flow distribution was mainly affected by the flow regime in the manifolds. Turbulent regime throughout
460 the manifolds entailed a more uniform flow distribution compared to laminar flow. This was mainly due to
461 the strong dependence of the local losses for the tee junctions in laminar conditions. The comparison
462 between calculated flow distributions and results from literature gave a good agreement in most of the cases.

463 The presented model is planned to be used for the development of a numerical model for evaluating the flow
464 distribution in large solar collector fields for district heating applications. In fact, flow rate and temperature
465 levels in such fields vary depending of several boundary conditions, such as solar irradiance, fluid type,
466 forward and return temperatures of the district heating network. Moreover, due to the temperature rise across
467 a collector row, each collector experiences a different mean fluid temperature. In order to evaluate the flow
468 distribution across the collector field, a flexible and accurate model is needed to calculate the pressure drop
469 across each collector in different operating conditions.

470 Additionally, regarding the flow distribution, the model can be used to optimize the design of collector.
471 Aspects which may be investigated are the ratio between manifold diameter and absorber pipe diameter,
472 length of the absorber pipes and effect of pipe inset into the manifold. The results could be used by collector

473 manufacturers to design their products in such a way that a more uniform flow distribution is achieved under
474 typical operation, and thereby improve the collector efficiency.

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521 **Appendix**

522 To evaluate the Reynolds number at which transition in flow regime occurred, the following procedure was
523 followed. The pressure drop across the collector was measured while varying gradually the fluid inlet
524 temperature and keeping a constant flow rate. The test was performed in cloudy sky conditions, so that the
525 inlet and outlet temperature were approximately the same. Due to non-uniform flow distribution, the
526 different absorber pipes had different flow rates. Using the presented numerical model, it was possible to
527 identify the pipe having a value of flow rate closer to the flow rate which would occur in case of perfectly
528 uniform flow distribution. For the selected pipe, the numerical model was also used to isolate the pressure
529 drop caused by the absorber pipe from that caused by the manifolds and tee junctions. The Darcy friction
530 factor was finally calculated through (eq.1), using the pipe pressure drop and a fluid velocity corresponding
531 to a perfectly uniform flow distribution.

532 The comparison between the so calculated Darcy friction factor and the theoretical one is shown in Figure
533 A1. In the diagram the curved referred to as “theoretical” friction factor is obtained from (eq.7) for $Re < 2300$,
534 from (eq.8) for $Re > 3100$ and from linear interpolation between the two for the intermediate Reynolds
535 numbers. The data points refer to the results of a test performed with a 42% glycol/water mixture, a total
536 flow rate of 1.98 m³/h and a mean fluid temperature increasing from 30 °C to 70 °C during 2.5 hours.

537 *Figure A1: Comparison between theoretical and calculated Darcy friction factor. The theoretical friction factor curve is obtained*
538 *from (eq.7) for $Re < 2300$, (eq.8) for $Re > 3100$ and linear interpolation between the two in between these values.*