NUMERICAL INVESTIGATION OF THE PIPELINES MODELING IN A SMALL-SCALE CONCENTRATED SOLAR COMBINED HEATING AND POWER PLANT

Roberto Tascioni[a,b,]*, Alessia Arteconi[b,c], Luca Del Zotto[b,d], Emanuele Habib[a], Enrico Bocci[d], Ramin Moradi[a], Khamid Mahkamov[e], Carolina Costa[e], Luisa F. Cabeza[f], Alvaro de Gracia[f,g], Piero Pili[h], André C. Mintsa[i], Matteo Pirro[j], Toni Gimbernat[k], Teresa Botargues[l], Elvedin Halimick[m], Luca Cioccolanti[b]

[a]DIAEE, Sapienza Università di Roma, via Eudossiana 18, Rome 00184, Italy
[b]Università Telematica eCampus, Via Isimbardi 10, Novedrate (CO) 22060, Italy
[c]Dipartimento di Ingegneria Industriale e Scienze Matematiche, Università Politecnica delle Marche, Italy
[d]Università degli Studi Guglielmo Marconi, Italy
[e]Department Mechanical Engineering and Construction, Northumbria University, UK,
[f]GREiA Research Group, Universitat de Lleida, Spain
[g]CIRIAF - Interuniversity Research Centre, University of Perugia, Italy
[h]Elianto S.R.L., Italy,
[i]Enogia S.A.S, France
[j]Società per il TRAsferimento TEcnologico e Guida all'Innovation Engineering, S.TRA.TE.G.I.E. srl, Italy
[k]SINAGRO ENGINYERIA S.L.P, Spain
[l]USER FEEDBACK PROGRAM SL, Spain
[m]AAVID Thermacore Europe, UK

*Correspondence author: Roberto Tascioni Email: roberto.tascioni@uniroma1.it

Keywords: Organic Rankine Cycle System; Numerical Investigation; Renewable Energy; Concentrated Solar Power; Micro Combined Heating and Power; Thermal Losses.

ABSTRACT

In this paper four different detailed models of pipelines have been proposed, they are supposed to be used in a quasi steady-state model of a micro solar Combined Heat and Power plant developed under the EU funded project Innova MicroSolar. The integrated plant consists of a Linear Fresnel Reflectors solar field, 3.8 tons of Latent Heat Thermal Energy Storage system equipped with reversible heat pipes and an Organic Rankine Cycle unit designed for a power production of 2 kW_e/18 kW_{th}. Previous numerical analyses carried out by some of the authors have revealed a high incidence of pipelines on the plant performance due to thermal losses and their dynamic behavior. Hence, in this paper four different levels of models detail of such pipelines have been developed and tested based on the micro CHP operating conditions in order to find out which approach provides the most accurate dynamic behavior in small scale CHP systems simulations.

NOMENCLATURE

A	Internal cross section Area of the tube (m^2)
c	specific heat at constant pressure (J/kg K)
D	Diameter (m)
DNI	Direct Normal Irradiance (W/m^2)
fr_D	Darcy friction coefficient
g	Gravity (m^2/s)
h	External convective coefficient (W/m^2 K)
k	Thermal conductivity (W/m K)
L	Length of the receiver (m)
p	Pressure (Pa)
Pr	Prandtl number
R	Thermal resistance (K m/W)
r	Radius (m)
rr	Angular coefficient of the temperature ramp
Ra	Rayleigh number
Re	Reynolds number
S	Internal "wetted" area of the tube (m^2)
T	Temperature (K)
t	Time (s)

u	Velocity of the Heat Transfer Fluid (m/s)
x	Longitudinal axis
$\dot{Q}$	Heat transfer rate (W/m)
Subscripts	
1	Internal part of the tube
2	External part of the tube
2-amb	Mean properties between 2 and amb
amb	External ambient
conv,int	Internal convective
conv,ext	External convective
cond	conductive
del	delay
HTF	Heat Transfer Fluid
i	i-th cross section along x direction
in	inlet
int	internal
ext	external
ins	Insulating material
loss	Losses
Nu	Nusselt number
oil	Diathermic oil
out	outlet
R	Radius
Steel	Stainless Steel metal
th	Thermal
w	Wall
Greek symbols	
Δx	Longitudinal discretization space (m)
Δr	Radial discretization space (m)
Δt	Time discretization (s)
α	Thermal diffusivity (m^2/s)
β	Thermal expansion coefficient (1/K)
ν	Kinematic viscosity (m^2/s)
η	Efficiency
ρ	Density (kg/m^3)

1. INTRODUCTION

In order to facilitate the transition towards a cleaner energy context, the European Union has updated its energy policy by fixing for the 2030: (i) a binding renewable energy target of at least 32% and (ii) an energy efficiency target of at least 32.5%, with a possible upward revision in 2023 [1].

Among the different technologies to efficiently convert and supply energy into buildings, Combined Heat and Power (CHP) plants in combination with district heating (DH) networks have already proved their appreciable benefits [2][3]. Despite this, rooms of improvement exist to increase their potential in reducing primary energy consumptions and curbing CO2 emissions. Indeed, their proper design and operation are of paramount importance to limit the significant thermal losses of the pipelines. So far, many researchers have addressed such issues by modeling the thermal dynamic behavior of the pipelines networks. For instance, B. van der Heijde [4] presented the mathematical derivation and software implementation in Modelica of a thermo-hydraulic plug-flow model of thermal networks. They highlighted that the advantages of the plug-flow model in comparison to the use of multiple control volumes is the grid size and time step independence with flow velocity and no numerical diffusion. Another example of improvement concerning the plug-flow model is explained by Denari et. al [5]. In particular, their approach provides the same accuracy of the finest-discretization finite volume method (FVM), while being 10^3 times faster and without introducing the smoothing effect of sharp temperature variations.

Instead, Del Hoyo Arce et. al [6], developed a distribution pipe model, simpler than detailed models and less computationally costly. They obtained an error less than 5 %, and 0.02 % in the mass flow rate and temperature respectively, while the transient analysis showed differences in the pure delay in temperatures within the network. Chertkov and Novitsky [7] considered the dynamic/transient advection diffusion-losses equations, by keeping the velocity flow steady and adjusting the temperature at the heat producing source. This formulation bypasses the computationally expensive Partial Differential Equations (PDEs) solution. An alternative to the FVM is the Finite difference method. In [8], the authors applied the third order Euler discretization method with the Total Variation Diminishing (TVD) to assess the thermal behavior of the pipe in a DH network. In particular, they proved an effective elimination of the numerical dissipation and dispersion even in rather coarse grids. Indeed, the TVD prevents oscillations of solution near the sharp front of temperature variation where other numerical schemes, such as Lax Friedrich, Lax Wendroff, Crank Nicolson, fail [9]. One of the most used numerical scheme for solving the advection-diffusion equation is the Upwind Method thanks to its simple applicability and good accuracy also during fast rising or falling edge in temperature. Anyhow, the major problem of this solving method consists of an additional numerical diffusion due to the Courant-number and grid refinement. Wanga et al. [10] compared the implicit Upwind scheme with the characteristic line method, finding that the first one is unconditionally stable and more informative in simulating the temperature distribution along the pipeline. While an extensive research activity is carried out with respect to pipelines in DH network, to the best of the authors' knowledge this aspect is still not investigated in case of small-scale CHP plants.

Among the different renewable energy resources to be used in CHP systems, solar energy is considered the one with the greatest potential thanks to its worldwide diffusion. However, it varies significantly with time, it is intermittent and has low density which requires the support of thermal storage technologies to extend its exploitation. In a previous work [11] some of the authors of the present paper developed a mathematical model of a micro-solar Organic Rankine Cycle plant for residential applications as designed within the EU funded project Innova Microsolar [12] finding a significant amount of thermal losses in the pipelines connecting the different subsystems. In this case every single pipe has been modeled as single node for the sake of simplicity. However, this hypothesis might introduce a significant approximation of the actual thermal behavior of the whole system. Indeed, shutdowns and restarts, as well as the flow rate variations, affect the ability of the solar field to produce more thermal power, or the ability of the Thermal Energy Storage to rise in temperature because of the thermal gradient along the pipe itself [13]. The thermal inertia of the pipelines, the multiple supply temperature levels of the transfer medium and the variability of local renewable energy sources entail significant fluctuations of their operation [13].

In this work, a detailed model of the pipelines connecting the different subsystems has been developed. In particular, a first order explicit Upwind method has been adopted for the fluid, whilst a first order in time and a second order in space Euler method has been used for the insulating walls. The choice fell on these numerical schemes because of their simple applicability and good effectiveness. In dynamic simulations the computational time is a tighten constrain, consequently simplified models have been also considered. The following models have been developed: (i) 2 D, (ii) 1 D radial (iii) 1 D longitudinal model and finally (iv) a lumped model. Thus, the accuracy of the different models has been tested for linear inlet temperature ramps with different slopes and at different fluid velocity. Comparing these results with the fluid regime obtained by the dynamic simulation of the micro solar CHP (as temperature derivative and flow rate), it is possible to compare the models in terms of accuracy and computational time.

2. ANALYSIS AND MODELLING

2.1 Two dimensional tube model

Predicting the thermal transient of the oil inside the pipeline means solving the transport equation of energy throughout the pipe itself. As shown in equation (1), both internal energy and heat losses depend on the axial position in the pipe and the time t:

$$\frac{\partial(\rho c_p T A)}{\partial t}+\frac{\partial(\rho u(c_p T+p/\rho)\, A)}{\partial x} = uA\frac{\partial p}{\partial x}+\frac{\rho u^2|u|}{2} fr_D S + \frac{\partial}{\partial x}\left(k_{HTF}A\frac{\partial T}{\partial x}\right)-\dot{Q}_{loss} \tag{1}$$

The source term $\dot{Q}_{loss}$ included in the advection equation 1 can be positive in case of heat losses from the internal fluid to the ambient and vice-versa. Since the pipelines under investigation are referred to a micro-scale CHP plant a detailed solution as for DH networks is not required while robustness and velocity of the code are preferred. As a consequence, all the terms on the second member of eq. 1 can be deleted and the source term excluded, with good approximation. Indeed, Van der Heijde et al. [4] have shown that the diffusivity term can be neglected, while the pressure difference and wall friction are not relevant in comparison to the total energy in common operating conditions. In light of these assumptions, the resulting advection equation can be re-written as:

$$\frac{\partial(\rho c_p T A)}{\partial t}+\frac{\partial(\rho u c_p T A)}{\partial x} = -\dot{Q}_{loss} \tag{2}$$

According to the Fourier's law, in an isotropic medium the 1 D heat transfer equation can be written in cylindrical coordinates as

$$\frac{1}{\alpha}\frac{\partial T}{\partial t}=\frac{1}{r}\frac{\partial T}{\partial r}+\frac{\partial^2 T}{\partial r^2} \tag{3}$$

In order to ensure the correct heat transfer across the interface between the HTF-insulation and insulation-ambient a Von Neumann boundary condition needs

to be included. Hence, the equations are discretized using the Finite Difference Method (FDM) by dividing the pipeline into o circumferential sections along the radial direction and m segments in the axial direction. The time and the spatial derivatives of the oil temperature, instead, are replaced by difference quotients.
Hence, in this work the one order explicit Upwind scheme was applied for solving the advection equation (2) as shown below:

$$\frac{T_{i,j}^{k+1}-T_{i,j}^{k}}{\Delta t}+u_i\frac{T_{i,j}^{k}-T_{i-1,j}^{k}}{\Delta x}=-\frac{T_{i,j}^{k}-T_{iw}^{k}}{\mathrm{R}^{k}{}_{\mathrm{conv_i},int}\rho^k c_{p_i}^k A} \quad (4)$$

where k = 1,2,…,N denotes the number of time step, i = 1,2,…,M denotes the number of longitudinal segments, while j = 1 since it is evaluating the fluid and it is the first nodal point along the radial discretization.
The thermophysical properties of the fluid are evaluated as a function of its temperature in the previous time step.
On the other hand, the one order explicit Euler scheme is used for the time derivative whilst the second order scheme is adopted for the spatial derivative of eq. (3). Therefore:

$$\frac{1}{\alpha^k}\frac{T_{i,j}^{k+1}-T_{i,j}^{k}}{\Delta t}=\frac{1}{r_i}\frac{T_{i,j+1}^{k}-T_{i,j-1}^{k}}{2\Delta r}+\frac{T_{i,j+1}^{k}-2T_{i,j}^{k}+T_{i,j-1}^{k}}{\Delta r^2} \quad (5)$$

Where the upper case and lower case have the same meaning mentioned before, while j = 1,2…,O represents the circumferential discretization of the pipeline. Also in this case the thermal diffusivity depends on the thermophysical properties of the previous time step: $\alpha^k = {k_{i,ins}}^K/({\rho_{ins}}^K c_{p,ins_i}^k)$.
With respect to the internal interface between the heat transfer fluid and the insulation, the boundary condition can be written as:

$$\frac{T_{i,j=1}^{k}-T_{i,j=2}^{k}}{R_{i(conv,int)}^{k}}=\frac{T_{i,j=2}^{k}-T_{i,j=3}^{k}}{R_{i(cond)}^{k}} \quad (6)$$

The same procedure is performed for the external interface where the radiative heat transfer can be neglected:

$$\frac{T_{i,j=O-1}^{k}-T_{i,j=O}^{k}}{R_{i(cond)}^{k}}=\frac{T_{i,j=O}^{k}-T_{amb}^{k}}{R_{i(conv,ext)}^{k}} \quad (7)$$

and the external convective resistance evaluated in case of wind and no wind conditions:

$$R_{(conv,ext)}=\frac{1}{h_2\pi D_2} \quad (8)$$

in case of no wind, the Churchill and Chu's correlation is used to estimate the Nusselt number [14]:

$$\overline{Nu_2}=\left\{0.6+\frac{0.387(Ra_2)^{1/6}}{\left[1+\left(\frac{0.559}{\overline{Pr}_{2-amb}}\right)^{9/16}\right]^{8/27}}\right\}^2 \quad (9)$$

Where:

$$Ra_2=\frac{g\beta\left(T_{i,j=O}^{k}-T_{amb}^{k}\right)D_2^3}{\alpha_2\nu_2} \quad (10)$$

$$\beta=1/\overline{T_{2-amb}} \quad (11)$$

$$\overline{Pr_2}=\frac{\nu_{2-amb}}{\alpha_{2-amb}} \quad (12)$$

with $10^5 < Ra_2 < 10^{12}$.
In case of forced convection, instead, the Nusselt number is estimated according to the Zhukauskas' correlation:

$$\overline{Nu}_2=CRe_2^m\,\mathrm{Pr}_{amb}^n\left(\frac{Pr_{amb}}{Pr_2}\right)^{1/4} \quad (13)$$

with

Re_2	C	M
1 ~ 40	0.75	0.4
40 ~ 10^3	0.51	0.5
10^3~ $2\cdot10^5$	0.26	0.6
$2\cdot10^5$~ 10^6	0.076	0.7

And $\begin{cases}\mathrm{n} = 0.37 \text{ for Pr } \le 10\\ \mathrm{n} = 0.36 \text{ for Pr} > 10\end{cases}$

which is valid for $0.7 < Pr_{ext} < 500, and\ 1 < Re_{D_2} < 10^6$. As regards the internal convective resistance:

$$R_{(conv,int)} = \frac{1}{h_1 \pi D_1} \quad (14)$$

where the convective heat transfer coefficient can be written as:

$$h_1 = \frac{k_{HTF}}{D_1} Nu_1 \quad (15)$$

In case the HTF regime has a Re > 2300, the Nusselt number can be expressed according to the Gnielinski's correlation:

$$Nu_{D_2} = \frac{\frac{f_2}{8}(Re_2 - 1000)\mathrm{Pr}_1}{1 + 12.7\sqrt{\frac{f_2}{8}}\left(Pr_1^{2/3} - 1\right)} \left(\frac{\mathrm{Pr}_1}{\mathrm{Pr}_2}\right)^{0.11} \quad (16)$$

where: $f_2 = (1.82 \log_{10}(Re_2) - 1.64)^{-2}$ (17)

valid for turbulent conditions when $0.5 < Pr_1 < 2000$ and $2300 < Re_2 < 5 \cdot 10^6$. On the contrary, when the HTF flow is laminar this value is fixed to 4.36.

Despite the external convective heat transfer coefficient is not negligible, the predominant term of the total thermal resistance is the insulating material which can be calculated as follows:

$$R_{i,cond} = \frac{\ln\left(\frac{r_{i+1}}{r_i}\right)}{2\pi k_{ins}(T_{r_i})} \quad (18)$$

where the thermal conductivity of the mineral wool $k_{ins}(T)$ is strongly dependent from its temperature as reported in table:

Table 1

Temperature (ºC)	Thermal conductivity (W/m·K)
50	0.042
100	0.049
150	0.059
200	0.071
250	0.086
300	0.105

2.2 One dimensional longitudinal model

With respect to the 1 D longitudinal model, equation 3 previously reported includes also the total thermal resistance towards the ambient while the thermal capacity of the insulation can be neglected. Therefore, the energy balance can be written as:

$$\frac{T_{i,j}^{k+1} - T_{i,j}^{k}}{\Delta t} + u_i \frac{T_{i,j}^{k} - T_{i-1,j}^{k}}{\Delta x} = -\frac{T_{i,j}^{k} - T_{amb}}{(\mathrm{R}_{\mathrm{conv_i},ext} + R_{i,cond})\rho^k c_{p_i}^k A} \quad (19)$$

where k = 1,2,…,N is the number of time step, i = 1,2,…,M the number of longitudinal segments while j = 1 ∀ k.

In this case the internal convective resistance has been neglected while the conductive resistance evaluated at the mean temperature between the HTF and the ambient.

2.3 One dimensional radial model

In the 1 D radial model, a linear temperature distribution along the tube length has been assumed. So, the energy balance equation in a single pipeline segment can be written as:

$$\dot{Q}_{in} - \dot{Q}_{out} - \dot{Q}_{loss} = \rho A L c_{p_i}^{K} \frac{\partial \bar{T}}{\partial t} \quad (20)$$

where the first two terms depend on the mean temperature of the pipeline segment. Hence:

$$\dot{Q}_{in} - \dot{Q}_{out} = 2\dot{m} c_p (T_{in}^k - \overline{T^k}) \quad (21)$$

By combining eqs. (20) and (21) and using eq (5) for the internal domain of the insulating material, the governing equation of the 1 D radial model is:

$$2\dot{m} c_p\left(T_{in}^k - \overline{T^k}\right) - \left(\frac{\overline{T^k}}{R_{(conv,int)}} - \frac{1}{R_{(conv,int)}} \cdot \frac{(R_{cond}\overline{T^k} R_{(conv,int)} T_{j=2}^k}{R_{(conv,int)} + R_{cond}}\right) = \rho A L c_{p_i}^{K} \frac{\partial \overline{T^k}}{\partial t} \quad (22)$$

2.4 Lumped model

In the lumped model a linear temperature distribution of the HTF is assumed as in the case of the radial model since there is not any discretization along the radial direction and all the thermal resistances are included into a single equation:

$$2\dot{m}c_p\left(T_{in}^k - \overline{T^k}\right) - \left(2\pi L \frac{\overline{T^k} - T_{amb}}{\frac{1}{k_{ins}} ln\left(\frac{r_2}{r_1}\right) + \frac{1}{h_2 r_2}}\right) = \rho A L c_p \frac{\partial \overline{T^k}}{\partial t} \quad (23)$$

2.5 Inlet conditions

In the above equation the inlet temperature T_{in} is given at time t = 0:

$$T_{in}^k = T_{in}^{k=1} + rr \cdot t \quad (24)$$

where rr is the angular coefficient of the temperature ramp.
Both in the 1 D longitudinal model and in the 0 D model the heat capacity of the insulating material is neglected. Eq. (23) is solved analytically obtaining the mean temperature of the HTF in the single cell. Then based on the linear temperature distribution assumed, the outlet temperature is calculated. In the 0 D model, since the term of the equation used for the transport properties has been neglected, a delayed output is introduced. This delay is calculated as:

$$\Delta t_{del} = L/u \quad (25)$$

3. RESULTS AND DISCUSSION

The yearly dynamic simulation of the Innova MicroSolar plant has been run aimed at showing the HTF regimes (temperature derivative and flow velocity) inside the pipelines during the operative conditions. Comparing the dynamic response of the plant with the output of the four proposed models, the model accuracy can be assessed in that specific working condition.
The simulation has been carried out for Lleida, Spain, (where the plant is currently being built), and the pipeline model used is the lumped model already available in the Simulink library called Simscape, in this case a single thermal node represents a tube. All the tubes have an internal diameter of 6 cm, 3 cm of insulating material. To avoid redundancy on the results, the fluid regime has been recorded at the outlet of the solar field (12 m long) and the Organic Rankine cycle (5 m long). For both the tubes the derivative of the outlet temperature with respect to time has been calculated as well as the oil flow rate. The simulation time step has been fixed at 10 seconds to better appreciate the variation of these parameters with time. The results are shown in Figure 1 and Figure 2.

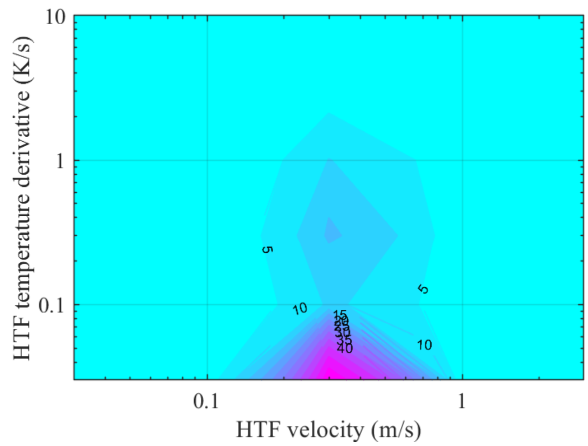

Figure 1
HTF regime in percent of total time, ORC outlet

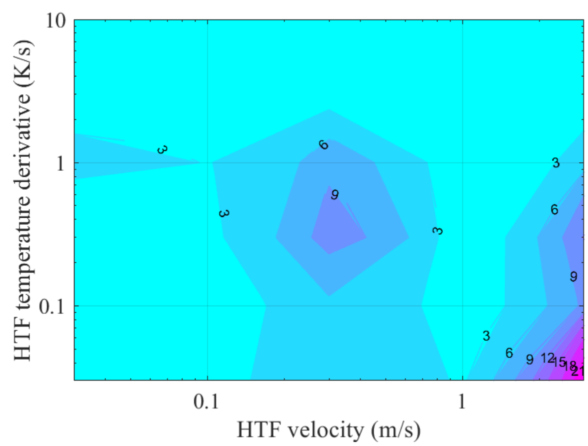

Figure 2
HTF regime in percent of total time, LFR outlet

The contour plots depicted in the previous figures address the operating points in a circumscribed region with rr < 1 K/s and u > 0.1 m/s most of the time.
On the basis of the four proposed models, a test with an increasing inlet temperature is carried on: the fluid inside pipeline at an initial temperature 300 K experiences a temperature rise up to 600 K. It has been chosen a tube length of 10 m, with the same internal diameter and insulating material of the Innova plant. A sensitivity analysis is accomplished by means of the variation of the temperature ramp slope and of the HTF velocity, an example is depicted in Figure 3.

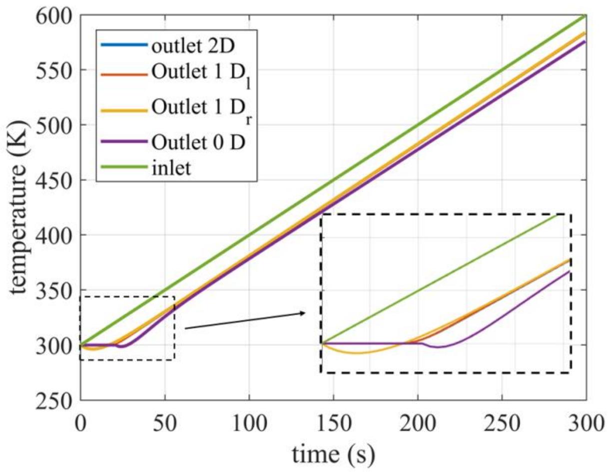

Figure 3
Response time test of the four models

It is worth to notice that the linear approximation of the temperature profile along the pipe length leads to an unphysical decrement of the outlet temperature during the initial transitory for the 0 D and the radial model. As expected, this is more relevant with low flow rated and strong slope of the inlet temperature, when the temperature gradient along the axial direction is higher. On the basis of the results attained by the 2 D model, the deviations of the others models can be evaluated. The good accuracy of the 1 D model is shown in Figure 4 . The effect of the radial discretization becomes negligible above 0.1 m/s for any rr.

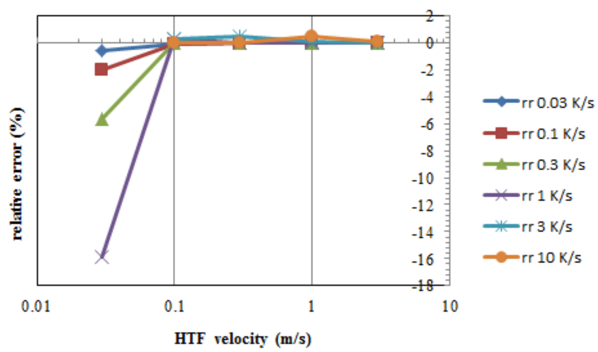

Figure 4
Relative error in the 1 D longitudinal model

It cannot be said for the radial one where rr > 3 causes significant deviations on the output temperature even with a flow rate of 0.3 m/s.

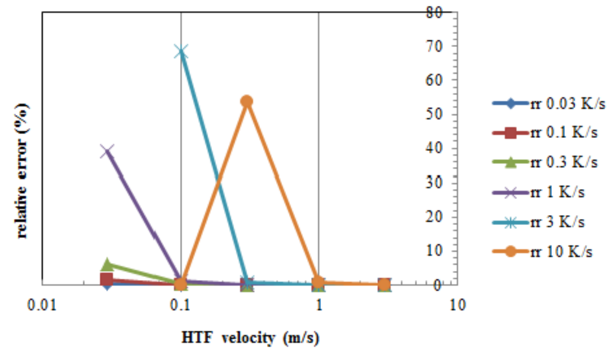

Figure 5
Relative error in the 1 D radial model

The lumped model exhibits good accuracy when the axial gradient is low, that is when the temperature ramp is not steep and/or the fluid has a high velocity, as shown in Figure 6.

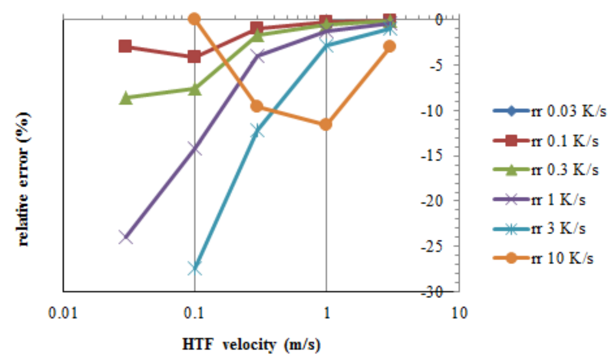

Figure 6
Relative error in the 0 D model

Summarizing, into the operative range of the Innova plant with rr < 1 K/s and u > 0.1 m/s, the only model whose output does not agree with the most detailed one is the lumped version, while the 1 D radial model can lead to not physical values during changing temperature derivative. Furthermore, the latter one has no advantage in terms of CPU time respect to the 1 D longitudinal model, as shown in Figure 7.

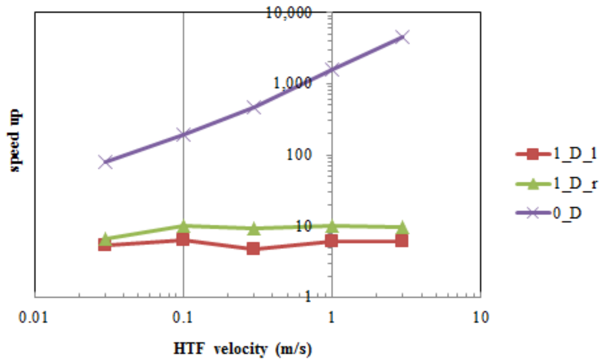

Figure 7
Speed up Vs 2 D model

4. CONCLUSIONS

Dynamic thermal distribution systems, as district heating and cooling, cogeneration and trigeneration plants, need an appropriate modeling of the pipeline networks. In this study four different mathematical models of the tubes have been analysed on the basis of the fluid regime (as temperature derivative at the inlet and flow velocity) obtained for the yearly dynamic simulation of the Innova Microsolar plant. All the outlet temperatures have been compared with the 2 D model within the fluid regime got before, the results revels that the best compromise between high accuracy and low CPU time is achieved using the 1 D longitudinal model.

ACKNOWLEDGMENTS

This study is a part of the Innova MicroSolar Project, funded in the framework of the European Union's Horizon 2020 Research and Innovation Programme (grant agreement No 723596)

REFERENCES

[1] European Union commission, "Clean Energy for All Europeans," last modified 2019, accessed July 8, 2019, https://ec.europa.eu/energy/en/topics/energy-strategy-and-energy-union/clean-energy-all-europeans.

[2] Werner, S., "E C O H E A T C O O L."

[3] Persson, U., and Werner, S., 2012, "District Heating in Sequential Energy Supply," Appl. Energy.

[4] van der Heijde, B., Fuchs, M., Ribas Tugores, C., Schweiger, G., Sartor, K., Basciotti, D., Müller, D., Nytsch-Geusen, C., Wetter, M., and Helsen, L., 2017, "Dynamic Equation-Based Thermo-Hydraulic Pipe Model for District Heating and Cooling Systems," Energy Convers. Manag.

[5] Dénarié, A., Aprile, M., and Motta, M., 2019, "Heat Transmission over Long Pipes: New Model for Fast and Accurate District Heating Simulations," Energy.

[6] del Hoyo Arce, I., Herrero López, S., López Perez, S., Rämä, M., Klobut, K., and Febres, J. A., 2018, "Models for Fast Modelling of District Heating and Cooling Networks," Renew. Sustain. Energy Rev.

[7] Chertkov, M., and Novitsky, N. N., 2018, "Thermal Transients in District Heating Systems," Energy.

[8] Wang, H., and Meng, H., 2018, "Improved Thermal Transient Modeling with New 3-Order Numerical Solution for a District Heating Network with Consideration of the Pipe Wall's Thermal Inertia," Energy.

[9] Hudson, J., Sweby, P. K., and Baines, M. J., "Numerical Techniques for Conservation Laws with Source Terms."

[10] Wang, Y., You, S., Zhang, H., Zheng, X., Zheng, W., Miao, Q., and Lu, G., 2017, "Thermal Transient Prediction of District Heating Pipeline: Optimal Selection of the Time and Spatial Steps for Fast and Accurate Calculation," Appl. Energy.

[11] Cioccolanti, L., Tascioni, R., and Arteconi, A., 2018, "Mathematical Modelling of Operation Modes and Performance Evaluation of an Innovative Small-Scale Concentrated Solar Organic Rankine Cycle Plant," Appl. Energy.

[12] "Innovative Micro Solar Heat and Power System for Domestic and Small Business Residential Buildings," accessed July 8, 2019, http://innova-microsolar.eu/.

[13] Wei, L., Lei, Q., Zhao, J., Dong, H., and Yang, L., 2018, "Numerical Simulation for the Heat Transfer Behavior of Oil Pipeline during the Shutdown and Restart Process," Case Stud. Therm. Eng.

[14] Frank P. Incropera, D. P. D., 1990, *Fundamentals of Heat and Mass Transfer*, la University of Michigan.