Journal of Physics: Conference Series

PAPER • **OPEN ACCESS**

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To cite this article: Kirill Bolotin *et al* 2023 *J. Phys.: Conf. Ser.* **2423** 012013

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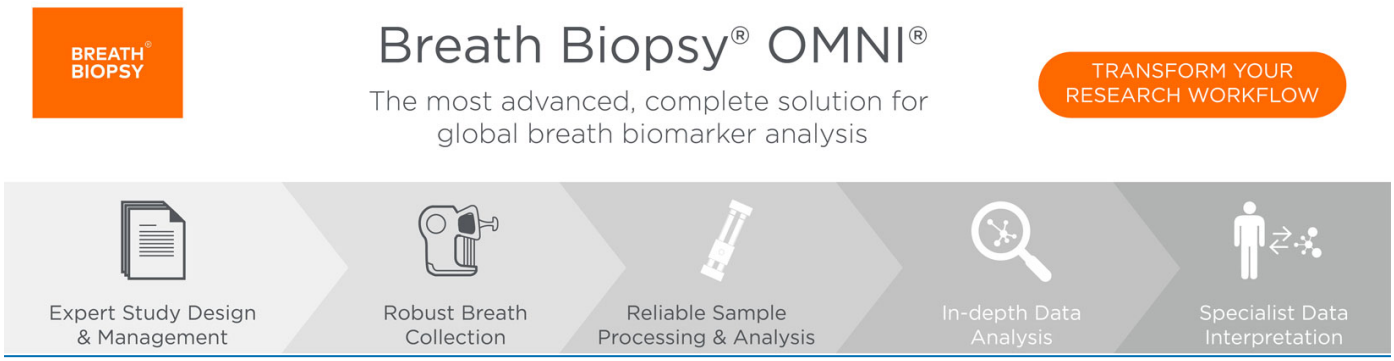

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Model of radiant capillary heating and cooling system

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Abstract. Paper is devoted to the development and verification of a tool for modeling and designing radiant-panel heating and cooling systems. Source of radiation is capillary mats, which consist of tubes through which water flows with a temperature of 18°C to 32°C, depending on the operating mode. The tool is an analytical model that takes into account the radiation from capillary mats located on the ceiling and walls, the mutual radiation of all surfaces in the room, including doors and windows, simplified convection, as well as third-party sources of thermal energy, such as the sun, electronics and people. The model was compared with a similar numerical model created in Comsol Multiphysics, and also verified by experimental data.

1. Introduction

Optimization of cooling and heating systems for residential and non-residential premises helps to save energy while maintaining the comfort, in this regard, traditional schemes are being replaced by new solutions [1, 2, 3]

This work is focused on the radiant-panel cooling and heating system based on capillary mats. Principle of its operation lies in the flow of the heat carrying agent through capillary tubes, assembled in mats, which can be placed on any room internal surface, in most cases - on the ceiling. Its main advantages include a relatively low operating temperature (28-30°C) in heating mode, as well as the ability to operate in cooling mode at temperatures of 16-18°C. Such temperature ranges can significantly increase the energy efficiency of systems especially in combination with renewable energy sources like heat pumps, e.g. ground, water or air heat pumps with a liquid agent in an internal loop [4, 5].

Main disadvantage is that system requires more detailed design in order to avoid errors that can lead to additional costs to correct the situation and increase the cost of its operation in the future. This is due to the fact that radiant systems are sensitive to their position relative to other surfaces in the room, as well as to the size of the radiant. In this regard, the standard approach with the search for a zero balance between the thermal energy of the system and losses through structures to the external atmosphere will not give a sufficiently accurate result.

Idea of creating such models is not new and there are a number of publications that touch on this topic in one form or another [6, 7, 8]. However, our goal was to develope, verify and describe a tool that will allow to perform all the calculations necessary for the design of radiant-panel capillary heating and cooling system that third-party users can use.

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2. Analytic model

Developed tool had to meet several requirements: the ability to use without additional knowledge, sufficient accuracy, low computation time, functionality and accessibility.

Since the main users will be design-engineers, who may not have specific knowledge in the field of mathematics and physics, the tool should be a ready-made package into which parameters are entered to get the result. At the same time, the accuracy should be sufficient for the designed system to fulfill the task of heating or cooling the room, but not excessive, since the package will not be used to solve scientific problems. It follows from this that the speed of the solution and the computational resources expended should be minimal.

Based on all of the above, decided to abandon the use of finite element packages, since they are not only redundant for the task, but also require significant material investments for their purchase. The free versions, on the other hand, lack a graphical interface, which reduces the availability of the package and requires a large investment to create it.

Our solution was based on an analytical model created in the free mathematical editor SMath Studio.

2.1. Model Description

Model allows to study the radiant heat transfer between the radiant surfaces (CM, CM1-CM4), floor (F), four walls (C1-C4), wall openings (W1-W4), ceiling area (CCM) not occupied by the mat, and heat transfer to external environment. In addition, the model contains a simplified account of convective heat exchange with air inside the room and additional sources of thermal energy, such as the sun, electrical appliances and people in the room. Scheme of interactions shown in Figure 1.

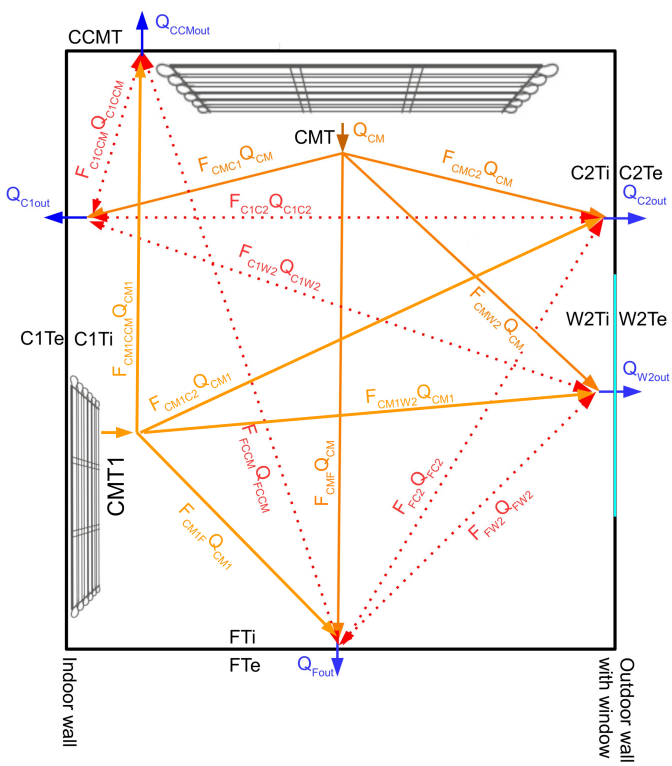

Figure 1. Scheme of mutual exchange of heat energy inside the room and with the external environment.

Although a two-dimensional scheme is presented, the model has a three-dimensional statement. For the two remaining walls, all the statements presented below will be valid.

It is also worth clarifying what is meant by the term "radiant surface". In most cases, the capillary mat is closed with a finish, due to which its temperature and the temperature of the surface in contact with the room are somewhat different. In this work, we are talking about the latter.

Main sources of thermal energy in the model are radiant surfaces, which can be located on the ceiling and walls. Their temperature (RST, RST1, etc.) is known and is a constant, the second constant value is outside temperature on the room's enclosing structures (C1Te, FTe, etc.). Unknown value is the temperature on the enclosing structures inner surface (C1Ti, FTi, etc). To relate these quantities, the standard expressions describing power of radiative and heat transfer exchanges were used, which are given below. For radiative:

$$Q_{RS_nC_m} = F \times \sigma \times \frac{(RSTn^4 - CmTi^4)}{\frac{1}{\varepsilon_{CM_n}} + \frac{1}{\varepsilon_{C_m}} - 1}, \tag{1}$$

where $Q_{RS_nC_m}$ (W) - power of radiative heat exchange between n radiative surface and surface of m wall, F (m^2) - mutual exposure coefficient, σ ($Wm^{-2}K^{-4}$) - Stefan–Boltzmann constant, RSTn and CmTi (K) - temperatures of both surfaces, ε_{CM_n} and ε_{C_m} (1) - emissivity of them.

Mutual exposure coefficient performs the same role as the view factor, making it possible to estimate the degree of radiation of one surface on another, depending on their geometric dimensions and positions relative to each other. Two analytical expressions were used to calculate them [9]. First for two perpendicular surfaces:

$$F = \sum_{i=1}^{2}\sum_{j=1}^{2}\sum_{k=1}^{2}\sum_{l=1}^{2}(\frac{(-1)^{i+j+k+l}}{Pi} \times (\frac{1}{8} \times ((x_k - a_l)^2 - (y_i)^2 - (b_j)^2) \times ln((x_k - a_l)^2 \tag{2}$$

$$+(y_i)^2 + (b_j)^2) + (\frac{1}{2} \times (x_k - a_l) \times \sqrt{(y_i)^2 + (b_j)^2} \times arctg(\frac{x_k - a_l}{\sqrt{(y_i)^2 + (b_j)^2}}))),$$

where x_k, y_i, a_l, b_j - coordinates defining surfaces.

Second for two parallel surfaces placed at a distance h from each other::

$$F = \sum_{i=1}^{2}\sum_{j=1}^{2}\sum_{k=1}^{2}\sum_{l=1}^{2}(\frac{(-1)^{i+j+k+l}}{2 \times Pi} \times ((b_k - y_l) \times \sqrt{(x_j - a_i)^2 + h^2} \times arctg(\frac{b_k - y_l}{\sqrt{(x_j - a_i)^2 + h^2}}) \tag{3}$$

$$+(x_j - a_i) \times \sqrt{(b_k - y_l)^2 + h^2} \times arctg(\frac{x_j - a_i}{\sqrt{(b_k - y_l)^2 + h^2}}) - \frac{h^2}{2} \times ln((b_k - y_l)^2 + (x_j - a_i)^2 + h^2))),$$

where x_j, y_l, a_i, b_k - coordinates defining surfaces, h - distance between surfaces.

Thus, coefficient calculations were performed that were used in the model to describe the interaction of all surfaces inside the room. In addition, an expression was used to describe losses to ambient, that is, connecting internal and external temperatures:

$$Q_{OUT} = U \times (CmTe - CmTi) \times SCm, \tag{4}$$

where U ($Wm^{-2}K^{-1}$) - reciprocal of thermal resistance, describing the effectiveness of the thermal insulation of the room constructions, CmTi and CmTe - internal and external temperatures of construction surfaces, S - construction area interacting with the ambient.

A similar expression is used for convective heat transfer between room air and its surfaces. Free or forced convection of the air itself is not calculated, since this would take too much time, computational resources and would not allow using the stationary problem formulation. To find unknown temperatures, for each surface and the room as a whole, heat balance equations are compiled, which are then combined into a system, the solution of which allows you to find the desired values.

In general, the heat balance expression for the entire room looks like:

$$Q_{CM} + Q_{CONV} + Q_S - Q_{OUT} = 0. \tag{5}$$

In it, Q_{OUT} - thermal energy losses through walls, floors, ceilings and openings, Q_{CONV} and Q_S - optional components of convective heat transfer and third-party energy sources, and Q_{CM} - describing the interaction of capillary mats with all internal surfaces.

Based on the formulation, the following assumptions were made when developing the model:

- All room constructions is represented by a thin layer;
- There is no horizontal and vertical edge heat exchange between the openings and the walls in which they are located;
- There are no losses through the areas occupied by radiating surfaces;
- All used parameters is averaged.

First, second and fourth assumptions are related to the simplification of the design of room to speed up the calculations. Third assumption was made based on the fact that a separate model will be created to calculate the temperature of the water inside the capillary mat, based on the power flows directed into and out of the room.

2.2. Realization

Once the basic principles underlying the model were formulated, it was necessary to translate them into an application that would be used by the design engineer. For the convenience of understanding the progress of its work, Fig. 2 shows its block diagram.

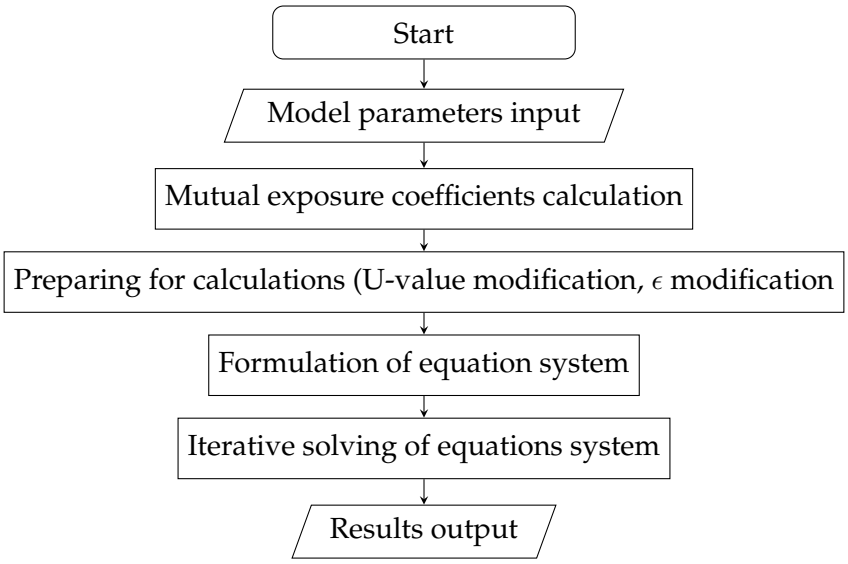

Figure 2. Block diagram of the analytical application work and calculations.

For a correct calculation, the user must enter the main geometric and physical parameters of the room under study, the temperature of the environment and radiating surfaces, as well as indicate the amount of thermal energy from external sources, if any, and whether the calculation should be carried out taking into account convective heat transfer. Geometric parameters include the dimensions of the room, the dimensions of the radiating surfaces, openings and their location. Physical parameters are heat transmittance (U-value), emissivity (ϵ) and heat transfer coefficient (α-value).

Largest amount of calculations in the model is associated with mutual exposure coefficients, since it is necessary to take into account the interaction of all surfaces. In order to reduce costs, the properties of view factors were used that the value for an area is equal to the sum of the values for the subareas of which it consists, for example, a wall in which there is one opening and one radiating surface was divided into 3 sub-areas: two occupied by other objects and one free part of the wall.

After completing all the preparatory stages, a system of equations is compiled, which is solved by an interactive method. The temperature of the internal surface of the floor is used as the value to be sorted out, and the evaluation criterion is the heat balance equation of the room, which must be equal to zero.

As a result, temperature values on all internal surfaces are displayed, as well as power and flows from all sources.

3. Numerical model

Although we have abandoned the use of numerical models based on the finite element method, it is convenient to use such a model for verification. Room was taken as the basis of model geometry in Comsol Multiphysics, which was used to obtain experimental data for verification. The result is shown in Fig. 3.

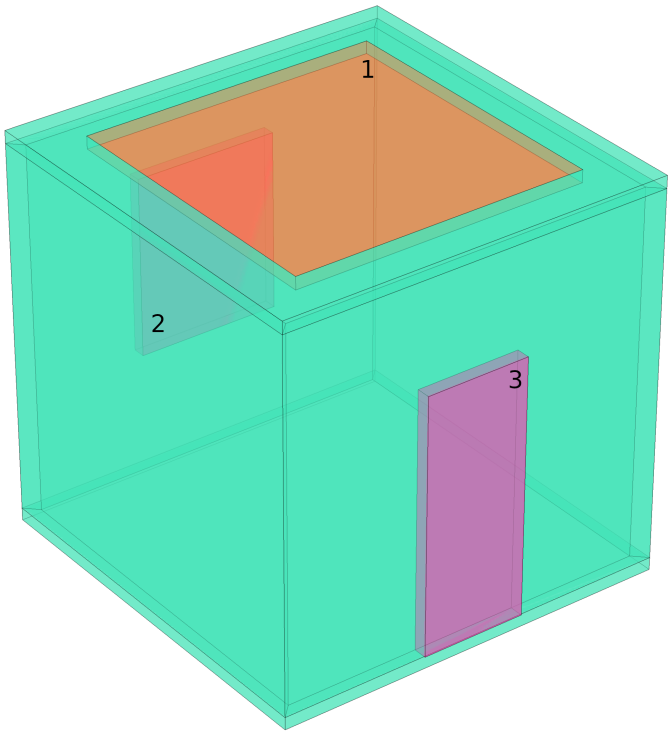

Figure 3. 3D geometry used in the model. 1 - radiating surface, 2 - window, 3 - door.

Room is a cube with a side of 3 m, the U-value for the floor, walls and ceiling is U=0.16, for the window and door U=0.8, the emission for all surfaces is ϵ=0.9. The window size is W=1.16 by H=1.43 m, the door size is W=0.84 by H=2.07, both openings are in the center of their walls. The size of the radiating surface is W=2.38 by H=2.38 m, it is also located in the center of the ceiling.

Model used two physics "Surface to surface radiation" to describe radiation and "Heat transfer in solids" to calculate temperatures on all surfaces. The mesh of the model consisted of 48 thousand elements and had 102 thousand degrees of freedom.

The following assumptions were made:

- There is no interaction between openings and radiated surfaces in the model with the walls on which they are located;
- Mutual exposure coefficients were calculated using the "Hemicube" method with radiation resolution equal 256;
- Model does not take into account convective heat transfer and external energy sources.

4. Verification

Experimental room, the main geometric dimensions and physical parameters of which are presented in the previous section, is equipped with measuring instruments that allow obtaining the following data: water temperature in the capillary mat and heat flux from it, temperature inside the room, heat flux through the window from solar radiation. An outside weather station allows to get the temperature outside. An extended description of the measuring system is given in [10].

In paper used average data for the period from 07.01.22 to 07.05.2022 for the same time interval from 13:00 to 15:00, since during this period they were the most stable. System operation mode - cooling. Data is shown in Figures 4 and 5.

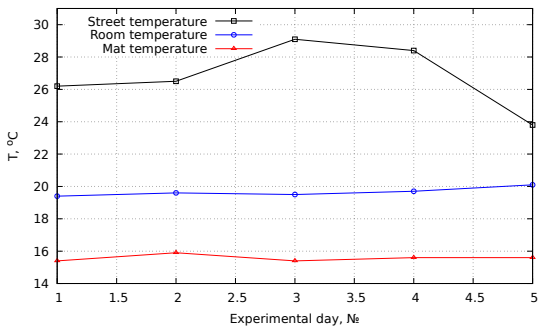

Figure 4. Experimental data of temperatures by days.

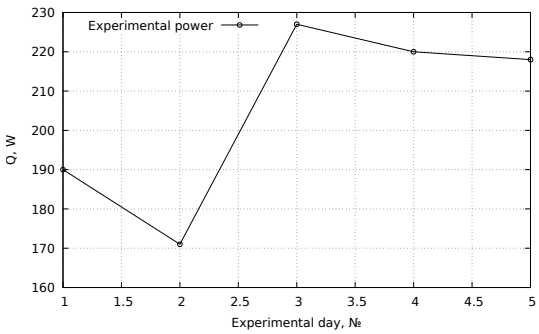

Figure 5. Experimental data of power by days.

Due to the limitations imposed by the application of the numerical model based on Comsol, only the radiative exchange of thermal energy was taken into account at the first stage of verification. Results shown in Figures 6 and 7 were obtained.

As can be seen, the temperature values are quite close, the deviation does not exceed 10%, which allows us to speak about the high accuracy of both models. However, the power results differ significantly from the experimental data. It was suggested that this is due to the lack of consideration of additional heat sources, namely the sun and the electronic components of the capillary installation itself. For each measurement period, its own value of additional power

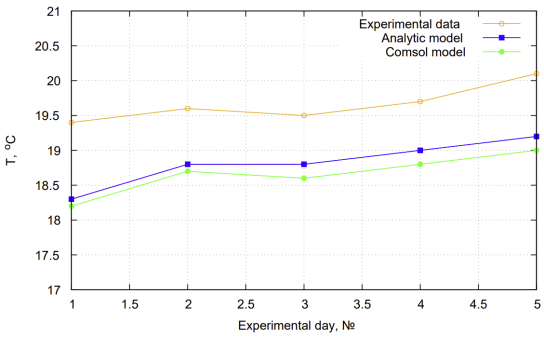

Figure 6. Comparison of calculated and experimental room temperatures values.

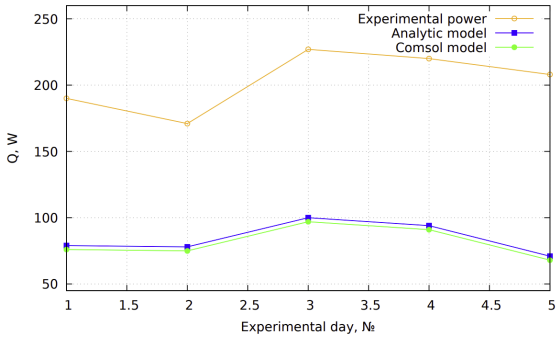

Figure 7. Comparison of calculated and experimental radiant surface power values.

was chosen based on the measured data on solarization and power of electric components of installation

In addition, it was decided to add convective heat transfer to the analytical model, since in the case of cooling in the room, natural convection occurs to move hot air from the floor to cold air on the ceiling.

Results obtained after the refinement of the model are shown in Figures 8 and 9. As can be seen, in both cases, the accuracy of calculations for the analytical model has increased and the deviation is less than 4%.

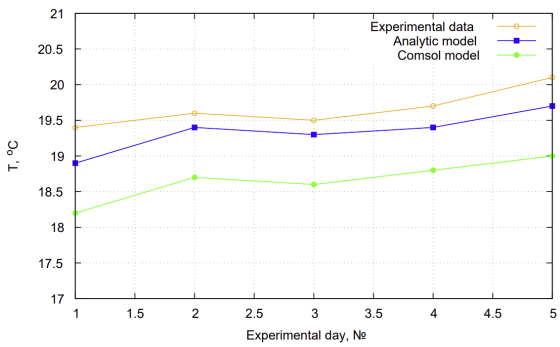

Figure 8. Comparison of calculated and experimental room temperatures values.

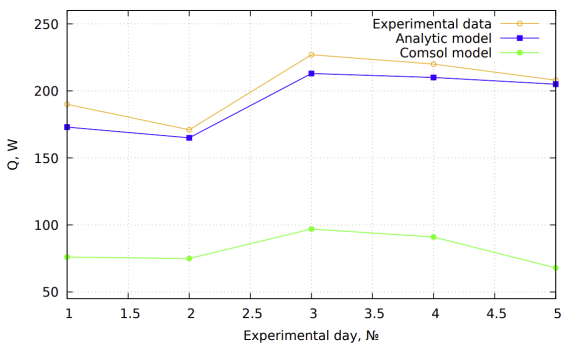

Figure 9. Comparison of calculated and experimental radiant surface power values.

Another important parameter is the calculation time and the spent computing power. Both models were run on the same computer station. For the first set of calculations, the total time for the analytical model was 12 min 38 sec, for the numerical model 34 min 12 sec. At the same time, it should be taken into account that our application does not implement an interface that would allow calculation for a set of values and part of the time was spent saving the results and introducing new initial values. At the same time, Comsol has such a function and all the time was spent on making calculations.

Second series of calculations on the analytical model (convection + additional heat sources) took 15 min 05 sec. In the numerical model, they were not fulfilled due to a significant increase in computational costs, which also affected the calculation time, which, approximately, should have increased by 13-15 times.

5. Conclusion

In this paper, we considered the creation of an application for performing engineering calculations for the design of capillary heating and cooling systems for residential and non-residential premises.

An analytical model was developed, which is based on the heat balance equation that takes into account the radiative exchange of thermal energy, as well as allowing you to optionally include the accounting for convective heat transfer and external sources of thermal energy.

A similar numerical model and experimental data were used for verification.

As a result, the developed model has an accuracy sufficient for engineering calculations (the deviation in temperature and power of the radiating surface does not exceed 4%), and allows calculations to be carried out without significant expenditure of computing resources in a minimum time, compared to the numerical model. At the same time, all results are averaged values, which saves time and computing power, but complicates the process of their analysis.

In the future, it is planned to add the ability to calculate the temperature distribution on surfaces depending on the distance from the radiating surface. Calculation of the water temperature inside the capillary tubes, since there can be different finishes between them and the radiating surface, and it is the water temperature that is important for design. And develop an interface that will allow to perform calculations for a set of initial values.

Acknowledgement

This work has been financially supported by the ERDF project "Development and approbation of complex solutions for optimal inclusion of capillary heat exchangers in nearly zero energy building systems and reduction of primary energy consumption for heating and cooling" No. 1.1.1.1/19/A/102

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