MATHEMATICAL MODELING OF HEAT EXCHANGE PROCESSES IN THE UNDERGROUND THERMAL ACCUMULATOR

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Abstract

The article presents the thermal technical calculation of the underground heat accumulator, which is used to prevent a sharp increase in air temperature in hothouses on sunny days and accumulate solar energy. The coefficient of heat transfer by air-pipe and pipe-soil boundary layers between 20-35 ^{0}C of the air-driven to the heat accumulator with the help of a ventilator was calculated in the analytical method. The mathematical model of the change in the time and length of the temperature during the movement of hot air through the pipe is developed (based on boundary conditions). By applying a submerged heat accumulator of hot air, it is possible to accumulate heat 3 - 4,5 MJ, depending on the speed of movement of air on sunny days.

Keywords

heat accumulator, coil, heat exchange, heat transfer coefficient, heat exchange efficiency coefficient.

At present, the heat load of 1 m² of a solar greenhouse is 0.18–0.2 kW, and during one heating season, it amounts to 70–78 kg of standard fuel. The total heat load of the solar greenhouse is estimated based on the heat loss through its enclosing structures, that is,

$$Q_t = k_{g.w} \cdot F_g \cdot K_b (t_i - t_e) \cdot K_{inf} \qquad (1)$$

where, $k_{g.w}$ – is the heat transfer coefficient of the solar greenhouse wall, which for a double-layer polyethylene film is $k_{g.w}$=5.8W/(m^2·°C) ; F_g – is the working area of the solar greenhouse, m²; K_b – is the barrier coefficient, which is evaluated as the ratio of the total external wall area of the solar greenhouse to its working area; t_i, t_e – are the temperatures of the internal and external environments of the solar greenhouse, respectively, °C; K_{inf} – is the infiltration coefficient [1].

One of the main elements of solar installations is the thermal accumulator.Analysis shows that various designs of solar thermal accumulators have been developed, in which materials such as water, soil, gravel, metal fragments, and others have been used as heat storage media [2–5]. However, the conducted studies revealed that there are few or no scientific sources available on the thermal-physical properties and mineralogical composition of inexpensive and locally available accumulator materials.

During sunny days, the excessive rise in the internal air temperature of solar greenhouses negatively affects plant growth and development. Therefore, it is an urgent task to develop a system for accumulating this excess heat and supplying it to the solar greenhouse during nighttime hours.

Under the influence of solar energy, the volumetric air heating collector–solar greenhouse underground thermal accumulator is made of a composite material that provides maximum heat transfer around the pipe. The thermal accumulator pipe is embedded in a substrate filled symmetrically at a depth of 50 cm below the soil surface, and scientific research is aimed at studying its thermophysical and energy characteristics (Fig. 1) [6].

The main characteristic of the heat exchange process around the underground thermal accumulator pipe is that the heat propagation in the soil layer close to the pipe has a higher gradient compared to the layers farther from the pipe. As the distance from the pipe increases, the intensity of heat propagation gradually decreases. In the air–pipe boundary of the underground thermal accumulator located inside the solar greenhouse, heat exchange occurs through forced convection. When the air velocity in the pipe of a given diameter is v_a >1m/s the air flow becomes fully turbulent.
The heat transfer in the pipe is determined by the following empirical relationship [7,8].

$$\mathrm{Nu} = 0{,}018 \cdot Re^{0{,}8} \approx 0{,}02 \cdot Re^{0{,}8} \qquad (2)$$

Research has shown that equation (2) leads to an underestimation of the heat transfer coefficient at the air–pipe boundary by 20–25%. The obtained values are consistent with the calculation results derived from the following expression taken from the literature [9]:

$$\mathrm{Nu} = 0{,}0126 \cdot Re^{0{,}875} \cdot Pr^{0{,}36} \qquad (3)$$

Here, $\mathrm{Nu} = a_{pipe} d_{pipe} / \lambda_a$, $\mathrm{Re} = v_a d_{pipe1} / V_a$, Pr = 0,71. By simplifying expression (3), it can be written in the following form:

$$a_{pipe} = 0{,}01138 \cdot \frac{\lambda_a}{V_a^{0,875}} \cdot \frac{v_a^{0,875}}{d_{pipe1}^{0,125}} \qquad (4)$$

The values of λ_a and v_a are taken from the table of physical parameters of air corresponding to the arithmetic mean temperature ($t_{average} = (t_{hot\ air} + t_p)/2$) between the hot air flowing inside the pipe and the inner surface temperature of the pipe. During the charging period, the amount of accumulated heat is determined using the following expressions:

– from the temperature drop along the pipe length [8]:

$$Q = G_a \cdot c_a\,(t_{i.a1} - t_{i.a2}) \cdot T, \quad G_a = v_a \cdot p_a \cdot \frac{\pi \cdot d_{pipe1}^2}{4} \cdot N, \qquad (5)$$

– from the temperature drop at the air–pipe boundary:

$$Q = a_{pipe} \cdot \pi \cdot d_{pipe1} \cdot l \cdot N\,(t_{i.a} - t_{pipe1}) \cdot T \qquad (6)$$

– from the temperature drop at the pipe–soil boundary:

$$Q = a_{soil} \cdot \pi \cdot d_{pipe2} \cdot l \cdot N\,(t_{pipe2} - t_{soil}) \cdot T \qquad (7)$$

– from the temperature variation in the soil layer:

$$Q = T_{soil} \cdot c_{soil} \cdot \Delta t_{soil}, \quad T_{soil} = \pi \cdot d_{pipe2} \cdot \delta_{soil} \cdot l \cdot N \qquad (8)$$

– from the temperature drop between the air inside the thermal accumulator pipe and the surrounding soil layer:

$$Q = \frac{1}{\frac{1}{a_{pipe}} + \frac{d_{pipe2} - d_{pipe1}}{2 \cdot \lambda_{pipe}} + \frac{1}{a_{soil}}} \cdot \pi \cdot d_{pipe1} \cdot l \cdot N\,(t_{i.a} - t_{soil}) \cdot \tau \qquad (9)$$

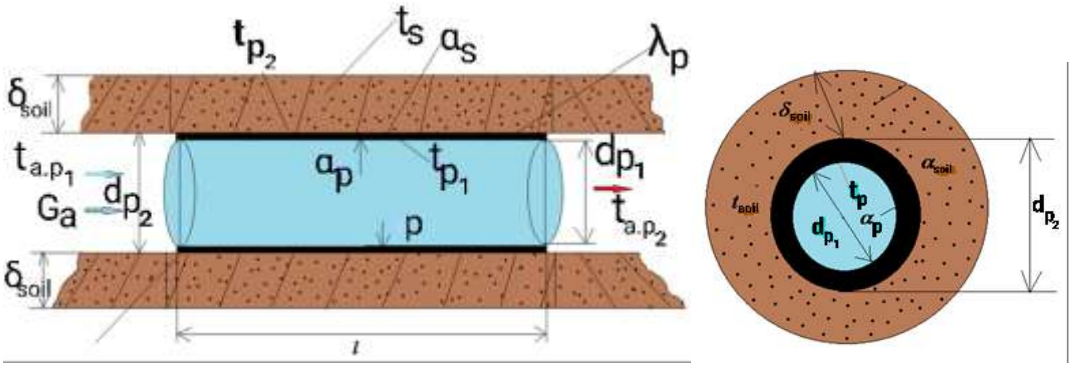

Figure 1. Schematic diagram for calculating heat transfer through the heat exchange pipe in the underground thermal accumulator.

For the air–pipe–soil system (Fig. 1), we formulate the heat exchange equation. The following assumptions are made:

– the thermal–physical properties of the soil and air are constant;

– the thermal resistance of the pipe wall is negligibly small, therefore, the **in**ner and outer pipe wall temperatures are assumed to be equal;

– the temperature gradient along the height and length of the thermal accumulator is insignificant. Hence, the pipe temperature is considered uniform throughout its volume, i.e., the temperature is evenly distributed**.**

The heat balance equation at the air–pipe boundary is expressed as follows [8]:

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$$v_a \cdot c_a \cdot p_a \cdot d_{pipe1} \cdot \frac{dt_{i.a\,(X,\ \tau\,)}}{dx} = 2\, a_{pipe}(t_{i.a}(\tau\,) - \ t_{pipe}(\tau\,)) \qquad (10)$$

$dt_{i.a\,(0,\ \ \tau\,)} = t_g(\tau\,)$ at the boundary condition, equation (10) takes the following form:

$$t_{pipe}(\tau\,) - t_{i.a}\,(\mathrm{x},\ \tau\,) = [t_{pipe}(\tau\,) - t_g(\tau\,)] \cdot exp(-\frac{2\alpha_{pipe}}{\vartheta_a \cdot c_a \cdot \rho_a \cdot d_{pipe1}} x) \qquad (11)$$

The air temperature inside the pipe varies linearly along its length

$$t_{i.a\,(x,\tau\,)} = \ t_g(\tau\,) + \mathrm{xn}, \qquad (12)$$

Here, n<0 corresponds to the charging process, and n>0 corresponds to the discharging process.

The average air temperature along the length of the pipe:

$$t_{pipe}(\tau) = t_g(\tau) + \frac{nl}{2} \qquad (13)$$

At x=l, the average temperature value along the pipe length is:

$$t_{pipe}(\tau\,) = t_{i.a}\ (\tau\,) - [t_{pipe}(\tau\,) - t_g(\tau\,)] \cdot \frac{v_a \cdot c_a \cdot p_a \cdot d_{pipe1} \cdot [\exp\left(-\frac{2\, a_{pipe}}{v_a \cdot c_a \cdot p_a \cdot d_{pipe1}} l\right) - 1]}{2 \cdot a_{pipe} \cdot \mathrm{l}}$$

(14)

Using expression (13), we obtain the following:

$$t_{pipe}\,(\mathrm{x},\ \tau\,) = t_g(\tau\,) + \mathrm{c}, \qquad (15)$$

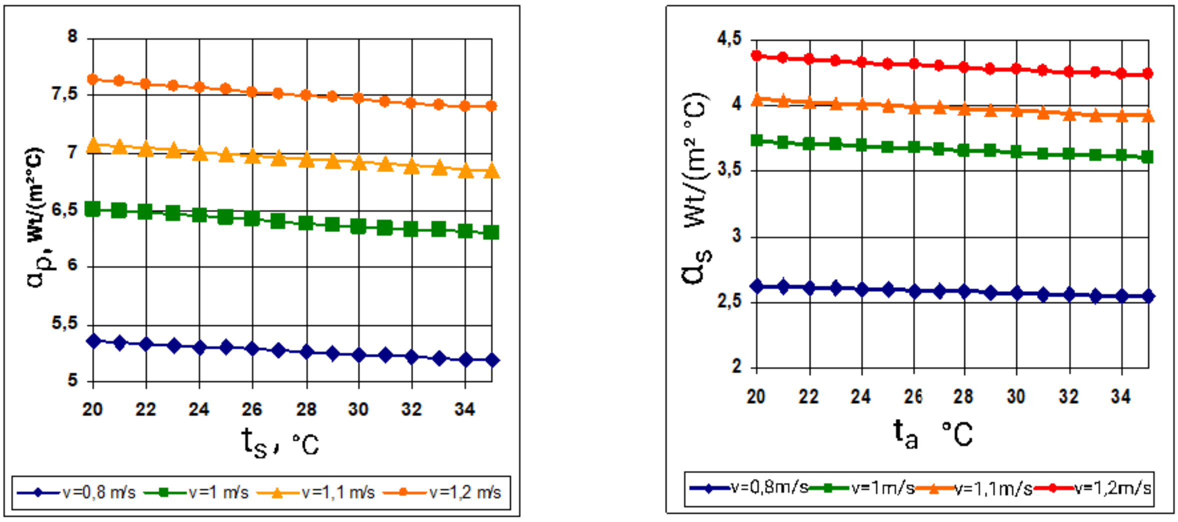
Here,

$$C = \frac{n \cdot l^2 \cdot \alpha_{pipe}}{\vartheta_a \cdot c_a \cdot \rho_a \cdot d_{pipe1} \cdot \left[exp(-\frac{2\alpha_{pipe}}{\vartheta_a \cdot c_a \cdot \rho_a \cdot d_{pipe1}} l) + \frac{2\alpha_{pipe} l}{\vartheta_a \cdot c_a \cdot \rho_a \cdot d_{pipe1}} - 1 \right]}$$

When the air temperature inside the solar greenhouse exceeds **25 °C**, the fan of the underground thermal accumulator is activated, forcing the warm air from the upper part of the greenhouse into the accumulator pipes. As the air temperature changes along the pipe length and over time, the temperature of the pipe surface also varies. This variation is calculated using expression (14)**.** The air blown into the underground thermal accumulator cools down as it passes through the accumulator and then rises back into the greenhouse, thereby creating a moderate microclimate inside. The heat accumulation process begins around 10:30 a.m. on sunny days and continues until approximately 4:00 p.m. At night, when the air temperature inside the greenhouse drops below 15 °C, the internal air is again directed into the underground thermal accumulator by the fan. As the air moves through the pipe, it is heated by the stored heat in the soil, then returns to the greenhouse, maintaining a stable and favorable temperature regime.

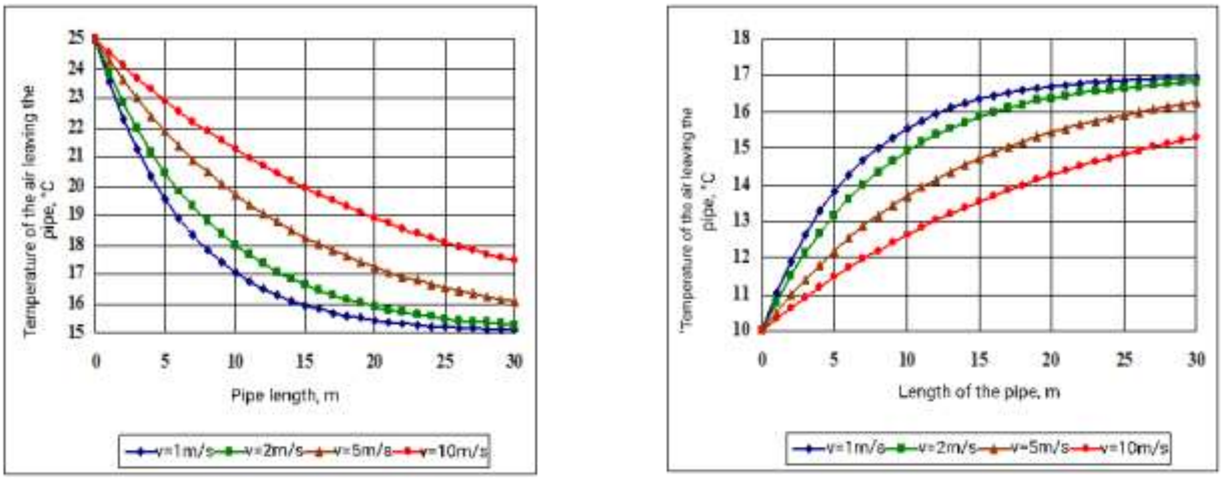
Using expression (14), the heat transfer coefficients at the air–pipe boundary are determined based on the measured air temperatures inside the pipe (ranging from 20–35 °C) and the pipe's inner surface temperature. The physical parameters of air corresponding to their arithmetic mean temperature are taken into account, and the results are presented in Figure 2(a). By equating expressions (6) and (7), and using the values obtained from expression (4), the heat transfer coefficient from the outer surface of the pipe to the surrounding soil layer (a_{soil}) is determined from the measured soil temperatures at the pipe–soil interface, and its dependence on various temperatures is illustrated in Figure 2(b).

In the underground thermal accumulator, the air-flow pipe has a diameter d_{pipe} = 100 mm and a length l_{pipe} = 150 mm, and consists of five sections. The charging mode lasts 6 hours, i.e., from 10:00 to 16:00. The variation of the temperature of the air exiting the underground thermal accumulator with pipe length at different flow velocities is shown in Figure 3, and the amount of heat accumulated is presented in Table 1.



a) **b)**

Figure 2. Dependence of the heat transfer coefficients on air temperature for the air moving in the underground thermal accumulator pipe: (a) heat transfer coefficient from the air to the pipe wall; (b) heat transfer coefficient from the pipe wall to the surrounding soil layer.



a) **b)**

Figure 3. Variation of the air temperature along the pipe length at different air velocities in the underground thermal accumulator: (a) charging mode, (b) discharging mode.

Table 1

Results of the calculation of the accumulated heat

№	Air umetric rate, s	Air city,	Air inlet perature, °C	Air outlet perature,	Accumulat eat, W	Daily umulated , MJ
1	0,00785	1	25	15,0	510	11
2	0,0157	2	25	15,1	1010	21,8
3	0,03925	5	25	16,1	2269,7	49
4	0,0785	10	25	17,5	3825,3	82,6

By using an underground thermal accumulator in solar greenhouses, it is possible to accumulate excess heat on sunny days, regulate the air temperature inside the greenhouse simultaneously, and efficiently utilize the stored heat during nighttime conditions, thereby partially saving fuel and energy resources. As shown in Figure 2, the heat transfer coefficient depends on the air velocity inside the pipe and increases with increasing air velocity. As a result, during the charging mode (τ = 5.5 hours), when the air velocity is v_a= 1m/s, $Q_{\rm ac}$ = 11MJ, v_a= 5m/s, $Q_{\rm ac}$ = 11MJ, v_a= 5m/s up to , $Q_{\rm ac}$ = 82,6 MJ of heat can be accumulated. The temperature drop of the air moving inside the pipe also depends on its velocity. When the air velocity reaches 10 m/s, it is found that installing a fan with a power of 0.5 kW is the most suitable and efficient option.

The heat load of a solar greenhouse depends on the outside air temperature and solar radiation, and during daytime the greenhouse heat demand can be met by solar energy. When the average outside air temperature is $t_{outside}$ = 0∘C, the nighttime heat load of the solar greenhouse is calculated as follows:

$$Q_{heat} = k_{inf}k_{wall}KF_{heat}(t_{heat} - t_{outside}) = 1{,}11 \cdot 1{,}25 \cdot 5{,}8 \cdot 100(18 - 0) = 14{,}485\ kW\ .$$

The amount of heat accumulated during the day in the underground thermal accumulator installed inside the greenhouse depends on the air velocity in the pipe; the calculation results are presented in Table 1. At an air velocity of 1.0 m/s, the accumulator can cover 3.5% of the greenhouse heat load; at 2.0 m/s — 7.0%; at 5.0 m/s — 15.7%; and at 10.0 m/s — 26.4%.

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