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OPTIMAL DESIGN OF HIGH-TEMPERATURE THERMAL ENERGY STORE FILLED WITH CERAMIC BALLS

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The momentary amount of the available solar energy and the demand usually are not equal during the usage of solar energy for heating and electric power supply. So it is necessary to store the heat energy. This article shows optimal design of a new construction, sensible heat store filled with solid heat storage material. The planned heat store has cascade system formed a spiral flow-path layout. This is a conceptual model, worked out in case of packed bed with ceramic balls. The aim of the special layout is to realize better overall efficiency than regular sensible heat stores have. The new construction would like to get higher overall efficiency by long flow-way, powerful thermal stratification and spiral flow-path layout which can ensure lower heat loss. The article shows the calculation method of the simulation of the charge and discharge and the calculation method of the overall efficiency using the results of the simulations. The geometric sizes and operating parameters of the thermal energy store with the best overall efficiency were calculated using genetic algorithm (GA). The results of the calculation tasks show that a thermal energy store with long flow-way, with cascade system formed spiral flow-path layout has higher overall efficiency than an one-duct, short flow-way thermal energy store which has equal mass of solid heat storage material as the long flow-way one, mentioned before.

Keywords: solar energy, heat storage, solid charge, sensible heat, optimization

Introduction

The possible thermal energy storing methods are: sensible heat storage, latent heat storage, sorption heat storage and chemical energy storage [1–8].

The simplest way is the storage of sensible heat, by heating a heat storage material without phase changing. The energy density of the sensible heat storage will be high if the specific heat and the density of the heat storage material are great as well [9].

Out of the materials which can be found in the environment in large quantity, the water has the greatest volumetric heat capacity (~4.18 MJ/m^{3}K), but water can be applied at atmospheric pressure up to 100ºC only. The heat transport media of the concentrated solar power systems can be used as heat storage liquids as well. The melt of the solar salt (60% $NaNO_3$ + 40% KNO_3) is used out of these materials in concentrated solar power plants as heat storage material (operating temperature range 260–550ºC, volumetric heat capacity ~2.84 MJ/m^{3}K [11]). It is not flammable, not toxic, and not too expensive.

The volumetric heat capacity of some solid materials (magnesite, corundum) – because of their higher density – come near to the volumetric heat capacity of the water with much higher upper temperature limits (magnesite 3.77 MJ/m^{3}K, corundum 3.3 MJ/m^{3}K, cast iron 4.1 MJ/m^{3}K [10]).

Screened pebble stone, cracked stone (1.5–2.5 MJ/m^{3}K), concrete (0.8–1.8 MJ/m^{3}K), wet soil (3.56 MJ/m^{3}K) [10] are used as sensible heat storage materials, because they are inexpensive.

The sensible heat stores are typical regenerative heat-exchangers. These are instationary thermal state heat-exchangers. The regenerators are long ago applied, great heat capacity heat stores with solid fill and with short charge-discharge cycle time (10–7200 s).

My aims were to study the possible interior structure of the long-term heat stores, the charge-discharge process, to calculate the optimal geometric sizes and operating parameters of those.

Comparison of short (L/D<10) and long (10<L/D) heat stores

The temperature-place function of the heat transport medium is similar to the temperature-place function of the solid heat storage material at a moment (the temperature of the heat transport medium t_f is higher at charge, lower at discharge than the t_s temperature of the solid heat storage material), so it is enough to show the temperature-place functions of the solid heat storage material.

Charge

The hot heat transport medium gives a part of its heat content to the solid heat storage material by flowing through the heat store, which is cold at the beginning of the charge period.

In case of short heat store the outlet temperature of the heat transport medium and the solid heat storage material are increasing soon after the beginning of the charging *(Figure* 1*)*, in case of long heat store they start to increase only at the end of the charge period *(Figure 2)*.

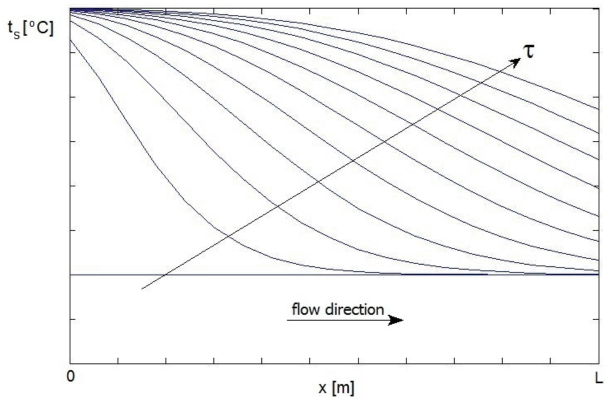

Figure 1: The temperature-place functions of the solid heat storage material during charge, in case of short heat store

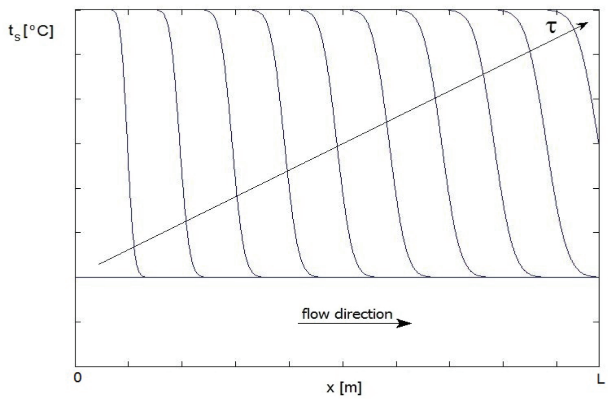

Figure 2: The temperature-place functions of the solid heat storage material during charge, in case of long heat store

Discharge

In the discharge period the cold heat transport medium flows through the hot heat store in opposite flow direction of the charge.

In case of short heat store the outlet temperatures of the heat transport medium and the solid heat storage material are decreasing soon after the beginning of the discharging *(Figure 3)*, in case of long heat store they start to decrease only at the end of the discharge period *(Figure 4)*.

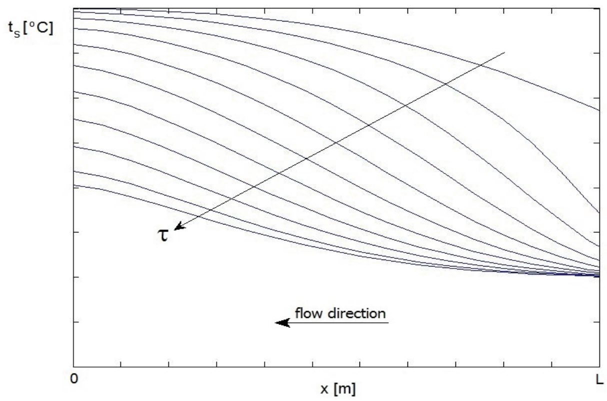

Figure 3: The temperature-place functions of the solid heat storage material during discharge, in case of short heat store

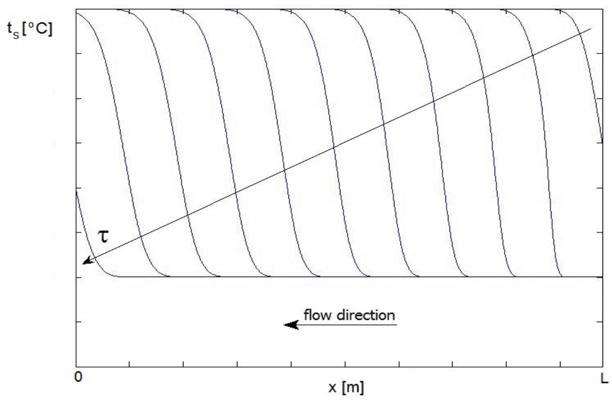

Figure 4: The temperature-place functions of the solid heat storage material during discharge, in case of long heat store

Basic idea of the cascade system heat store

During the charging and discharging of the long heat store the thermocline zone is located only in a part of the length of the heat store. It is plausible solution to divide to sections the heat store and allow knocking-off the sections from the flow-path of the heat transport medium. Let's call these sections as ducts. In case of the cascade system heat store the heat transport medium must flow through only the ducts where the thermocline zone is going along. The transport power demand of the heat transport medium can be reduced by this solution.

The heat-loss of the heat store into the environment will be small if the heat store has small specific surface. From the prismatic bodies the cylinder with H/D=1 ratio has the smallest specific surface, followed by the regular n-sided prism with H/S=1 ratio (assuming that the heat-loss flux is approximately equal in all sides of the body).

The higher heat-loss of the long heat store (because of its greater specific surface) can be reduced by using cascade system of the ducts formed a spiral flow-path layout *(see later on Figure 5)*.

Out of the regular n-sided prisms the six-sided can be built from cylinders with the best space utilization.

The geometry and operating of the heat store filled with ball particles

The ducts of the heat store filled with ball particles are cylinders made of metal sheet with outer thermal insulation *(Fig. 5)*. The metal shell holds the bed of the particles, and the heat transport medium. The thermal insulation supports the thermal stratification in radial direction. To choose cylindrical shape for the ducts is necessary – because of the pressure of the packed bed of bulk particles.

The outer geometry of the heat store is regular hexagonal prism with $H/S_t \approx 1$ ratio and cascade system of the ducts formed a spiral flow-path layout.

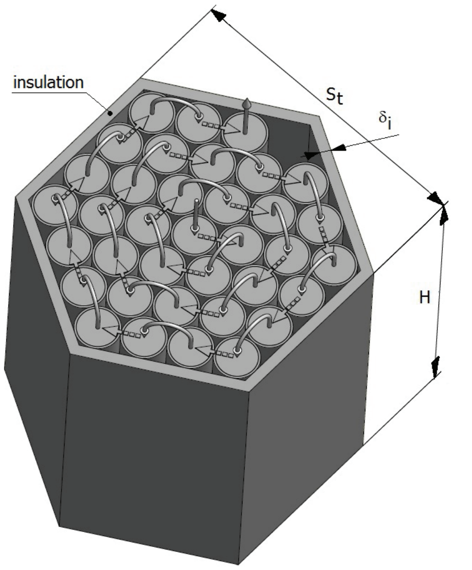

Figure 5: The construction layout of the heat store filled with ball particles and arrangement in the cascade system with spiral connection
S_t – side distance of the heat store, H – bed height of a duct, δ_i – thickness of the outer thermal insulation

The number of ducts N_j is an odd one in case of full filling, but leaving out the last duct we get an even number, so it is possible to lock out or join in the heat transport medium flow into any duct-pairs (duct-pair means a pair of ducts, one downwards and another upwards). This way the place of the last duct could be used e.g. for service purpose.

The hot heat transport medium is put in at the top of the middle duct at the beginning of charge and it flows downwards, it turns in the return band flows into the next duct (the lower connecting of the ducts is signed with dashed arrow) and flows trough that upwards. The heat transport medium coming out from the second duct can be led to the next pair of ducts. The heat transport medium must flow through only the ducts where the thermocline zone is going along. In the discharge period the cold heat transport medium flows through the hot heat store opposite to the flow-direction of the charge (from outside to middle).

The main sizes of a duct of the heat store can be seen in *Figure 6*.

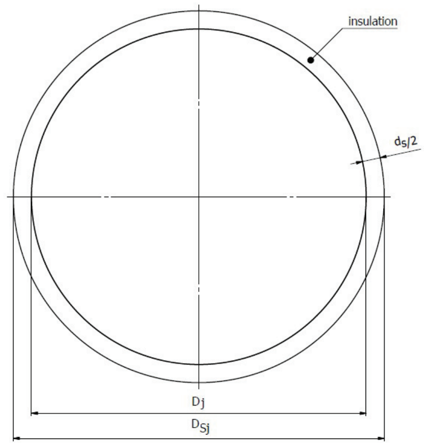

Figure 6: Top view of a duct
D_{Sj} – outside diameter of the insulated duct, D_j – inside diameter of the duct, d_s – whole thickness of the thermal insulation between two ducts

Basic differential equations of the heat transport in the heat store filled with ball particles

K. A. R. Ismail, and R. Stuginsky Jr. [126] have reported an excellent comparative analysis of several models for description of the heat transfer in the heat transport medium and the solid heat storage material.

For both media I have chosen a bit simpler model than the general, one-dimensional model. The heat-loss boundary condition is not included in the differential equations, because it takes effect only at the outer ducts, the effect of it will be calculated separately from the calculation of the temperature-place functions.

The differential equation for the description of the heat transfer in the heat transport medium is

$$\varepsilon\ \rho_f\ c_f\left(\frac{\partial t_f}{\partial \tau} + w_f\ \frac{\partial t_f}{\partial x}\right) = \alpha_f\ a_p(t_s - t_f)\,, \qquad (1)$$

where:

- t_f – temperature of the flowing heat transport medium,
- t_s – temperature of the solid heat storage material,
- ρ_f – density of the heat transport medium,
- c_f – specific heat of the heat transport medium,
- w_f –average velocity of the heat transport medium in the flow-channels,
- a_p – superficial particle area per unit bed volume,
- α_f – heat transfer coefficient between the flowing heat transport medium and the solid heat storage material,
- ε –void fraction of the bed.

The superficial ball particle area per unit bed volume a_p is

$$a_p = \frac{6}{d_b}(1-\varepsilon)\,, \qquad (2)$$

where: d_b – diameter of the ball particle.

The differential equation of the heat transfer in the heat transport medium was discretized by applying explicit forward difference scheme in time and upwind difference scheme in space [13, 14].

The differential equation for the description of the heat transfer in the solid heat storage material is

$$(1-\varepsilon)\ \rho_s\ c_s \frac{\partial t_s}{\partial \tau} = \lambda_{seffx} \frac{\partial^2 t_s}{\partial x^2} + \alpha_f\ a_p (t_f - t_s)\,, \quad (3)$$

where:

ρ_s – density of the solid heat storage material,

c_s – specific heat of the solid heat storage material,

λ_{seffx} – effective axial thermal conductivity of the solid heat storage material.

The effective axial thermal conductivity of the solid heat storage material λ_{seffx} is

$$\lambda_{seffx} \frac{1}{\dfrac{\varepsilon}{\lambda_f} + \dfrac{(1-\varepsilon)}{\lambda_s}}\,, \quad (4)$$

where:

λ_f – thermal conductivity of the heat transport medium,

λ_s – thermal conductivity of the solid heat storage material.

The differential equation of the heat transfer in the solid heat storage material was discretized by applying explicit forward difference scheme in time and centred difference scheme in space (FTCS) [13, 14].

The heat transfer coefficient between the flowing heat transport medium and the solid heat storage material was calculated according to [15].

The design variable

It is practical to choose the diameter of the ball particle d_b for optimization design variable: $x_1=d_b$. The flow velocity of the heat transport medium w_f has the greatest influence on the charging of the heat store, but it depends on the void fraction ε only (at given inside diameter of the duct D_j). In case of the bed filled with uniform ball particles the void fraction ε is independent from the diameter of the particle d_b.

Definition of the restrictions

Geometric restrictions

The lower limit of the diameter of the ball particle d_b is restricted by the manufacturing technology and the pressure drop, and the upper limit is also restricted by the minimal way-length of the heat transfer in the heat storage material.

Limits for x_1 design variable are

$$0.01\ \text{m} \le x_1 \le 0.1\ \text{m}\,. \quad (5)$$

For the minimal specific surface the best value of the geometric ratio could be

$$\frac{H}{S_t} \approx 1\,. \quad (6)$$

Pressure drop restriction

The pressure drop of the heat transport medium flowing through the packed bed in case of incompressible medium, according to [16] is

$$\Delta p' = L \left[150 \frac{(1-\varepsilon)^2}{\varepsilon^2} \frac{\rho_f\ \nu_f}{d_b^{\,2}} w_f + 1{,}75 \frac{1-\varepsilon}{\varepsilon} \frac{\rho_f}{d_b} w_f^{\,2} \right], \quad (7)$$

where:

ν_f – kinematic viscosity of the heat transport medium.

The pressure drop of the heat transport medium must be restricted in order not to be necessary to install pressure resistant shell, not to get high transport work demand, can be negligible the compressibility of the gas and may be sufficient to use ventilator instead of blower or compressor in case of gas heat transport medium.

The upper pressure drop limit for L=2H flow-way length according to the previous requirements is

$$\Delta p'_{2H} \le 10000\ \text{Pa} = 0.1\ \text{bar}\,. \quad (8)$$

Composition of the objective function

The objective function is the overall efficiency of the heat storage, which is suitable to compare the variants of the heat stores. The optimal sizes and operating parameters could be got from the maximum-point of the objective function.

In the calculation of the overall efficiency I relate the part of the extractable heat quantity which can be used for heating and electric power production to the sensible heat storage capacity of the heat store as it is here:

$$\eta_o = \frac{Q_{hid} - Q_l - Q_{tr}}{Q_{cap}}\,, \quad (9)$$

where:

Q_{hid} – extractable heat quantity from the heat store during a charge-discharge cycle without heat-loss,

Q_l – heat-loss to the environment through the boundary surfaces during a charge-discharge cycle,

Q_{tr} – heat-equivalent of the transport work demand during a charge-discharge cycle,

Q_{cap} – sensible heat storage capacity of the heat store between the inlet and outlet temperature of the heat transport medium at the charge.

The optimal value of the design variable can be searched by optimization using the calculation of the temperature-place functions during the whole length of the charge-discharge cycle.

The heat quantity Q_{hid}

The heat quantity Q_{hid} is the difference between the heat-content of the heat store after charge and after discharge without heat-loss.

It is necessary to know the temperature-place functions of the heat store at the end of the charge and at the end of the discharge.

The heat quantity Q_{hid} is

$$Q_{hid} = \int_0^{N_j H} c_s \, \rho_s \, (1-\varepsilon) \, \frac{D_j^2 \pi}{4} \, \left(t_s(x,\tau_c) - t_s(x,\tau_c+\tau_d)\right) dx \, , \quad (10)$$

where:

τ_c – term of charge,
τ_d – term of discharge.

The final temperature-place function of the charge of the heat store is the initial condition of the discharge. In the discharge period the heat transport medium flows through the heat store in opposite flow direction of the charge.

The heat quantity Q_l

The heat quantity Q_l is the heat-loss into the environment through the boundary surfaces during a charge-discharge cycle is

$$Q_l = Q_{lr} + Q_{ls} + Q_{lb} \, , \quad (11)$$

where:

Q_{lr} – heat-loss to the environment through the roof surface,
Q_{ls} – heat-loss to the environment through the side surfaces,
Q_{lb} – heat-loss to the environment through the bottom and the ambient ground.

The calculation of the heat-loss has taken into account the temperature of the heat store changing in place and time.

The heat quantity Q_{tr}

The heat quantity Q_{tr} is the heat-equivalent of the transport work demand during a charge-discharge cycle. The heat quantity Q_{tr} is approximated

$$Q_{tr} = \frac{(N_{j2Hc} \, \tau_c + N_{j2Hd} \, \tau_d) \dot{m}_f \, \rho_f \, \Delta p'_{2H}}{\eta_{oh}} \, , \quad (12)$$

where:

N_{j2Hc} – average number of duct-pairs which are simultaneously used during the charge,
N_{j2Hd} – average number of duct-pairs which are simultaneously used during the discharge,
$\dot{m}_f$ – mass flow rate of the heat transport medium,
$\Delta p'_{2H}$ – pressure drop of the heat transport medium on L=2H flow-way length,
η_{oh} – overall efficiency of the electric power production in a heat power station.

The heat quantity Q_{cap}

The heat quantity Q_{cap} is the sensible heat storage capacity of the heat store between the inlet and outlet temperature of the heat transport medium at the charge:

$$Q_{cap} = \dot{Q}_f \, \tau_c = \dot{m}_f \, c_f (t_{f,ci} - t_{f,co}) \, \tau_c = m_s \, c_s (t_{s,ce} - t_{s,cs}) \, , \quad (13)$$

where:

$\dot{Q}_f$ – heat current during the charge,
$t_{f,ci}$ – inlet temperature of the heat transport medium at charge,
$t_{f,co}$ – outlet temperature of the heat transport medium at charge,
m_s – mass of the heat storage material,
$t_{s,cs}$ – (homogeneous) temperature of the solid heat storage material at the start of the charging,
$t_{s,ce}$ – (homogeneous) temperature of the solid heat storage material at the end of the charging.

Basic data of the optimization task

I have made the calculations during the optimization with the following main data:

τ_c=63 day=1512 h=5 443 200 s, $\dot{Q}_f$ =2 MW, $t_{f,ci}$=400°C, $t_{s,cs}$=100 °C, d_s=0.2 m.
τ_d=58 day=1392 h=5 011 200 s.
The solid heat storage material is magnesite, its physical properties are at $t_{s,mid}$ [10]:
$t_{s,mid}=(t_{s,cs}+t_{s,ce})/2$=(100°C+400°C)/2=250°C
λ_s=23.26 W/mK, ρ_s=3500 kg/m^3, c_s=1077.5 J/kgK.
The required mass of the heat storage material for an ideal heat store: m_s=33 678 t, ε=0.3.
The heat transport medium is nearly ambient pressure air, its physical properties are at $t_{f,mid}$ [10]:
$t_{f,mid}=(t_{f,ci}+t_{f,co})/2$=(400°C+100°C)/2=250°C
λ_f=0.0425 W/mK, ρ_f=0.6715 kg/m^3, c_f=1038.5 J/kgK, $\nu_f=4.1525\cdot10^{-5}$ m^2/s.

The final temperature-place function of the charge of the heat store is the initial condition of the discharge.

The mass flow rate of the heat transport medium is constant during the charge-discharge process.

The inlet temperature of the heat transport medium during discharge is: $t_{f,di}$=100°C.

Data of the outer thermal insulation: λ_i=0.0468 W/mK, δ_i=1 m.
η_{oh}=0.3.

I have applied the genetic optimization algorithm of the Matlab software in order to find the optimal geometric sizes and operating parameters of the thermal energy store with the best overall efficiency.

Optimal sizes and operating parameters of the thermal energy store with the best overall efficiency

The optimization process has been executed with number of ducts N_j=1, 6, 18. The results are summarized in *Table 1* and *Figure 7*.

Table 1: Optimal sizes and overall efficiencies with several number of ducts

N_j [-]	1	6	18
x_l [m]	0.045	0.1	0.1
H [m]	27.5	29.6	28.8
L [m]	27.5	177.6	518.4
D_{Sj} [m]	25.5	10.1	6.0
S_t [m]	27.5	29.6	28.8
d_b [mm]	45	100	100
w_f [m/s]	0.064	0.412	1.203
Q_{cap} [PJ]	10.89	10.89	10.89
Q_{hid} [PJ]	8.45	9.38	9.98
Q_l [PJ]	0.92	1.11	1.01
Q_{tr} [PJ]	0.01	0.25	2.87
η_o [-]	0.6908	0.7363	0.5601

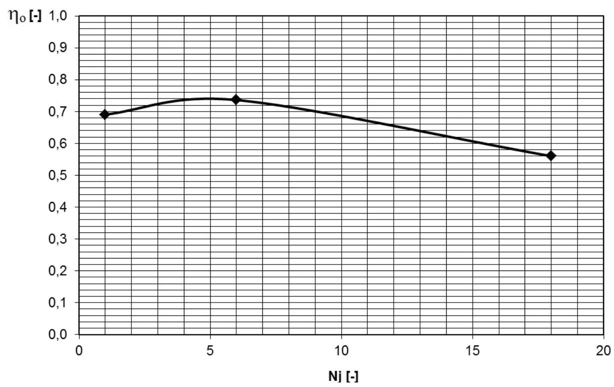

Figure 7: The optimal overall efficiency against the number of ducts

Main results

The best overall efficiency can be reached with six ducts. It is better than the overall efficiency of one-duct type and eighteen-duct type. The expected significant increasing of the overall efficiency was ruined by the increasing transport work demand of the heat transport medium and the increasing heat-loss.

The advantage of the six-duct type against the one-duct type is the smaller inside diameter of a duct D_j. It is easier to distribute the stream of the heat transport medium along a smaller flowing cross-section than along a larger one.

The overall efficiency decreases with increasing of the number of ducts, the main reason of it is the powerful increasing of the transport work demand.

In case of liquid heat transport medium the transport work demand would be much smaller, so better overall efficiency could be reached with greater number of ducts. The upper limit of the overall efficiency would be restricted by the heat-loss.

The void fraction of a packed bed, filled with uniform balls, in case of exact space-filling ε=0.26 is, in practice $\varepsilon\approx$0.3 is! The void fraction of the volume of the heat store would have to fill with heat transport liquid.

This large liquid volume brings up the idea of the establishment of a heat store with liquid heat storage medium, built from cylindrical tanks, in cascade system formed a spiral flow-path layout. The thermal stratification of the heat store with liquid heat storage material would be sharper, the overall efficiency would be higher than with solid heat storage material.

Conclusions

I have worked out the constructional and the mathematical model of the long flow-way, sensible heat store with cascade system. The temperature-place functions and the overall efficiency can be calculated by using the mathematical model.

I have used genetic optimization algorithm in order to find the optimal variant of the thermal energy store with the best overall efficiency.

The chargeable and the dischargeable heat quantity of the multi-duct, long flow-way heat store is more than of the one-duct, short flow-way thermal energy store with equal mass of solid heat storage material. The temperature level of the outgoing heat transport medium is more advantageous in case of the multi-duct heat store than in case of the one-duct type during the whole charge-discharge cycle.

The heat-loss can be reduced by using heat store with small specific surface and by allowing charge from middle to outside and discharge from outside to middle.

The transport power demand of the heat transport medium can be reduced by making the flow of the transport medium only through the ducts where the thermocline zone is going along.

According to the results of the optimization higher overall efficiency can be reached in case of six-duct type heat store than in case of one-duct type, filled with ceramic balls.

The overall efficiency decreases with increasing of the number of ducts – because of the strong increasing of the transport work demand.

Acknowledgements

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