ACTA TEHNICA CORVINIENSIS – Bulletin of Engineering
Tome VII [2014] Fascicule 1 [January – March]
ISSN: 2067 – 3809

[1.] Tibor BORBÉLY

OPTIMAL DESIGN OF SEASONAL PIPE-CHANNELLED THERMAL ENERGY STORE WITH GAS HEAT TRANSPORT MEDIUM

[1.] University of Pannonia, Institute of Mechanical Engineering, Veszprém, HUNGARY

Abstract: *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 pipe-channelled construction. 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 was 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 much 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 100 ^{o}C 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 260-550 ^{o}C, 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 temperature of the solid heat storage material t_s), 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).

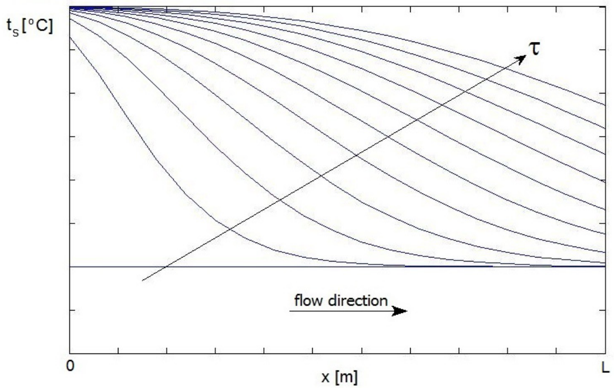

Figure 1. The temperature-place functions of the solid heat storage material during charge, in case of short heat store

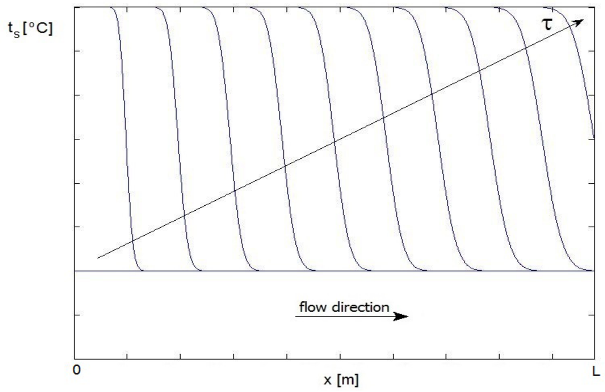

Figure 2. The temperature-place functions of the solid heat storage material during charge, in case of long heat store

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).

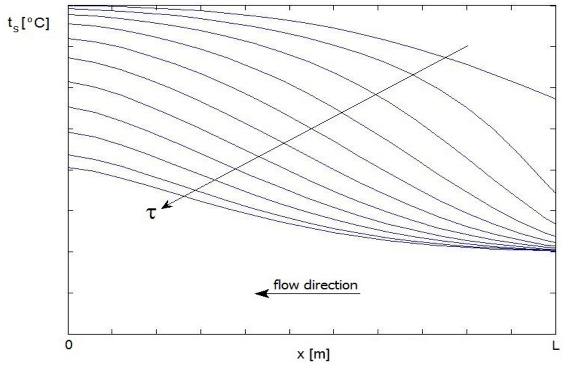

Figure 3. The temperature-place functions of the solid heat storage material during discharge, in case of short heat store

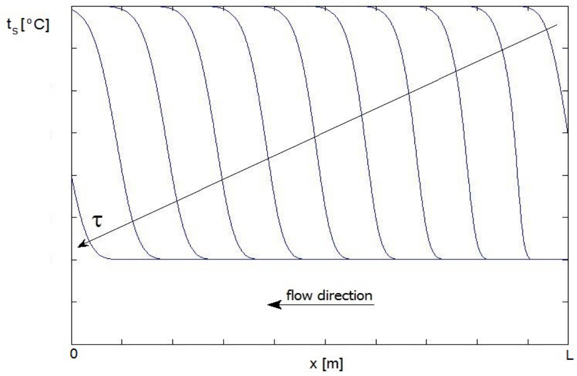

Figure 4. The temperature-place functions of the solid heat storage material during discharge, in case of long heat store

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 three-sided, four-sided and six-sided are suitable to build from them regular n-sided prisms without gaps.

THE GEOMETRY AND OPERATING OF THE HEAT STORE MADE FROM PIPE-CHANNELLED BRICKS

The pipe-channelled ducts of the heat store are regular hexagonal prisms with metal shell and outer thermal insulation. The metal shell holds the heat transport medium in. The thermal insulation supports the thermal stratification in radial direction.

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

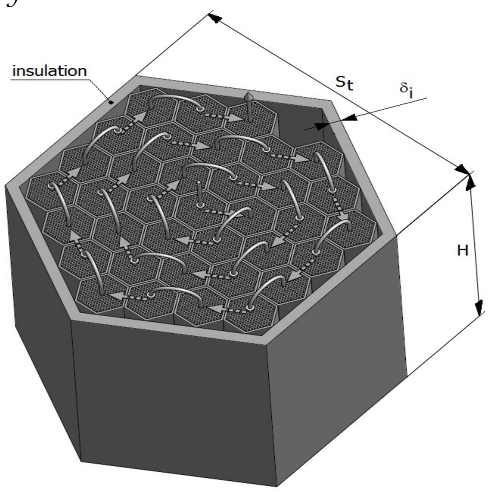

Figure 5. The constructional layout of the pipe-channelled heat store and arrangement in the cascade system with spiral connection

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.

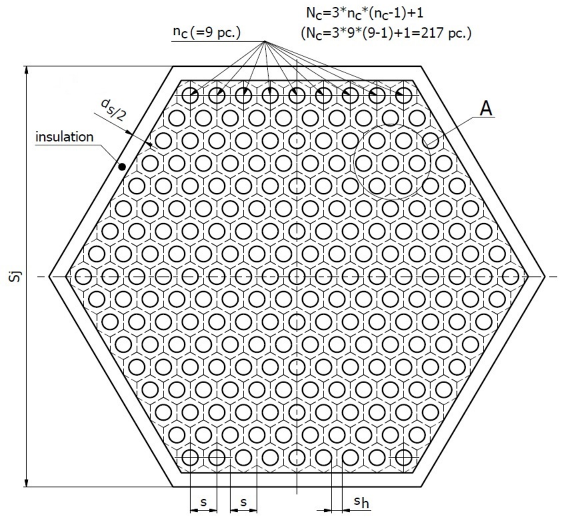

Figure 6. Top view of a pipe-channelled duct with the main sizes

S_j – side-distance of the insulated duct, d_s – whole thickness of the thermal insulation between two ducts, n_c – number of channels along a side-length of a duct, N_c – total number of channels of a duct, s – distance between the center of two pipe-channels, s_h – minimal material-thickness between two pipe-channels.

The centres of the pipe-channels form an equal-sided triange shape. The heat storage material which belongs to a pipe-channel can be replaced by a pipe of equal solid volume (drawn with dashed line in Figure 7).

The pipe-channels of the bricks of a duct are arranged so that they make the pipe-channels connected along the whole duct. The whole cross-

section of a duct can not be made from a single brick – because of the cross-sectional size of the duct.

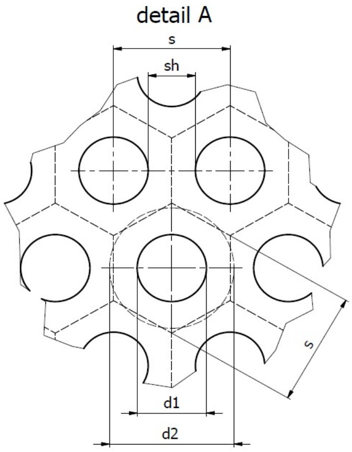

Figure 7. The part of the heat storage medium which belongs to a pipe-channel and approximation principle of it
d_1 – inside diameter of a pipe-channel, d_2 – equivalent outside diameter of the substitutional heat storage pipe.

BASIC DIFFERENTIAL EQUATIONS OF THE HEAT TRANSPORT IN THE PIPE-CHANNELLED HEAT STORE

Ismail, K. A. R. and Stuginsky Jr., R. [12] 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 and then I have converted it to be suitable for the case of the pipe-channelled 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

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

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

The void fraction ε in case of pipe-channelled heat store (see Figure 7) is

$$\varepsilon=\frac{\frac{d_1^2\,\pi}{4}}{\frac{\sqrt{3}}{2}s^2}=\frac{\frac{d_1^2\,\pi}{4}}{\frac{d_2^2\,\pi}{4}}=\left(\frac{d_1}{d_2}\right)^2. \quad (2)$$

The superficial heat transfer surface area per unit bed volume a_p in case of pipe-channelled heat store is (see Figure 7)

$$a_p=\frac{d_1\,\pi}{\frac{\sqrt{3}}{2}s^2}=\frac{d_1\,\pi}{\frac{d_2^2\,\pi}{4}}=\frac{4d_1}{d_2^2}. \quad (3)$$

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

$$(1-\varepsilon)\ \rho_s\ c_s\frac{\partial t_s}{\partial \tau}=\lambda_{seffx}\frac{\partial^2 t_s}{\partial x^2}+\alpha_f\ a_p(t_f-t_s), \quad (4)$$

where ρ_s is density of the solid heat storage material, c_s is specific heat of the solid heat storage material, λ_{seffx} is effective axial thermal conductivity of the solid heat storage material.

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

$$\lambda_{seffx}=(1-\varepsilon)\ \lambda_s, \quad (5)$$

where λ_s is 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 [16].

THE DESIGN VARIABLES

The velocity of the flowing heat transport medium w_f has the most influence on the charge and discharge. Out of sizes in Figure 7. d_1 and s or s_h are the most important geometrical sizes which can be chosen in order to be optimization variables together with the velocity of the heat transport medium w_f. In the w_f-d_1-s or the w_f-d_1-s_h groups there are only two independent design variables.

The economical flow velocity in pipelines (nonlinearly) depends on the inside diameter of the pipe [15], but there is such a k exhibitor, with that w_{fmax}/d_1^k and w_{fmin}/d_1^k are nearly constant against to d_1. This value for the exhibitor is k=0.6. The w_{fmax}/d_1^k and w_{fmin}/d_1^k can be used as restriction limits. The design variables are the following

$$x_1 = \frac{w_f}{d_1^{0.6}}, \; x_2 = s_h. \qquad (6)$$

DEFINITION OF THE RESTRICTIONS

Geometric restrictions

In case of a pipeline the upper limit of the variable x_1 would be restricted by the operation cost, the lower limit would be restricted by the investment cost. In case of the pipe-channelled heat store there is not such lower limit, so I have decreased the lower limit.

Limits for design variable x_1 are given by

$$1\frac{m^{0.4}}{s} \le x_1 \le 110\frac{m^{0.4}}{s}. \qquad (7)$$

Lower limit of the size s_h is restricted by the manufacturing technology, and the upper limit is also restricted by the minimal way-length of the heat transfer in the heat storage material.

Limits for design variable x_2 are are given by

$$0.01 \text{ m} \le x_2 \le 0.1 \text{ m}. \qquad (8)$$

The lower limit of the diameter of the pipe-channel d_1 is restricted by the manufacturing technology and the mountability

$$0.01 \; m \le d_1. \qquad (9)$$

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

$$\frac{H}{S_t} \approx 1. \qquad (10)$$

Integral restriction

The number of channels along a side-length of a duct n_c must be integer.

Pressure drop restriction

The pressure drop of the heat transport medium flowing through the pipe-channels in case of incompressible medium is

$$\Delta p' = \lambda_{fr} \frac{L}{d_1}\frac{\rho_f}{2} w_f^{\,2}, \qquad (11)$$

where λ_{fr} is friction factor.

In case of compressible medium and if the pressure drop is lower than 10 percent of the entire pressure this equation could be used [17].

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

$$\Delta p'_{2H} \le 10000 \;\; Pa = 0.1 \; bar. \qquad (12)$$

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:

$$\eta_o = \frac{Q_{hid} - Q_l - Q_{tr}}{Q_{cap}}, \qquad (13)$$

where Q_{hid} is extractable heat quantity from the heat store during a charge-discharge cycle without heat-loss, Q_l is heat-loss to the environment through the boundary surfaces during a charge-discharge cycle, Q_{tr} is heat-equivalent of the transport work demand during a charge-discharge cycle, Q_{cap} is 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

$$Q_{hid} = \int_0^{N_j H} c_s \, \rho_s \, N_c \left(\frac{\sqrt{3}}{2} s^2 - \frac{d_1^2 \pi}{4} \right) \left(t_s(x,\tau_c) - t_s(x,\tau_c + \tau_d) \right) dx, \quad (14)$$

where τ_c is term of charge, τ_d is 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

$$Q_l = Q_{lr} + Q_{ls} + Q_{lb}, \quad (15)$$

where Q_{lr} is heat-loss to the environment through the roof surface, Q_{ls} is heat-loss to the environment through the side surfaces, Q_{lb} is 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

$$Q_{tr} = \frac{(N_{j2Hc}\,\tau_c + N_{j2Hd}\,\tau_d)\,\dot{m}_f \;\; \rho_f \;\; \Delta p'_{2H}}{\eta_{oh}}, \quad (16)$$

where N_{j2Hc} is average number of duct-pairs which are simultaneously used during the charge, N_{j2Hd} is average number of duct-pairs which are simultaneously used during the discharge, $\dot{m}_f$ is mass flow rate of the heat transport medium, $\Delta p'_{2H}$ is pressure drop of the heat transport medium on L=2H flow-way length, η_{oh} is 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 (Eq.17).

$$Q_{cap} = \dot{Q}_f \;\; \tau_c = \dot{m}_f \;\; c_f(t_{f,ci} - t_{f,co}) \;\; \tau_c = m_s \;\; c_s(t_{s,ce} - t_{s,cs}), \quad (17)$$

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

Basic data of the optimization task

We 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.

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, ν_f=4.1525 $\cdot 10^{-5}$ 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 PIPE-CHANNELLED THERMAL ENERGY STORE WITH THE BEST OVERALL EFFICIENCY

The optimization process has been executed with number of ducts N_j=1, 6, 18, 36, 60, 90, 126, 168. The results are summarized in Table 1. and Figure 8.

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

N_j [-]	1	6	18	36
x_1 [$m^{0.4}/s$]	1	5	8	10
x_2 [m]	0.01	0.01	0.01	0.01
H [m]	25.7	29.6	29.8	31.1
L [m]	25.7	177.7	536.1	1121.3
S_j [m]	24.5	9.74	6.12	4.62
S_t [m]	26.5	30.1	30.3	31.3
N_c [-]	1230721	155269	37969	13669
d_1 [mm]	11.9	14.2	20.4	27.7
d_2 [mm]	23.0	25.4	31.9	39.6
s [mm]	21.9	24.2	30.4	37.7
s_h [mm]	10.0	10.0	10.0	10.0
w_f [m/s]	0.07	0.39	0.77	1.16
Q_{cap} [PJ]	10.89	10.89	10.89	10.89
Q_{hid} [PJ]	7.09	9.28	9.89	10.07
Q_l [PJ]	0.94	1.12	1.08	1.11
Q_{tr} [PJ]	0.004	0.09	0.13	0.16
η_o [-]	0.5644	0.7417	0.7970	0.8083

N_j [-]	60	90	126	168
x_1 [$m^{0.4}/s$]	11	15	9	7
x_2 [m]	0.01	0.01	0.01	0.01
H [m]	32.4	33.2	36.4	41.6
L [m]	1945.8	2990.7	4591.8	6988.4
S_j [m]	3.82	3.18	3.09	2.90
S_t [m]	32.9	33.2	37.7	40.6
N_c [-]	5941	3571	1141	397
d_1 [mm]	36.8	39.7	74.9	123.8
d_2 [mm]	49.1	52.2	89.2	140.5
s [mm]	46.8	49.7	84.9	133.8
s_h [mm]	10.0	10.0	10.0	10.0
w_f [m/s]	1.52	2.16	1.90	2.00
Q_{cap} [PJ]	10.89	10.89	10.89	10.89
Q_{hid} [PJ]	10.12	10.15	10.19	10.25
Q_l [PJ]	1.17	1.18	1.40	1.57
Q_{tr} [PJ]	0.19	0.32	0.29	0.23
η_o [-]	0.8049	0.7951	0.7808	0.7764

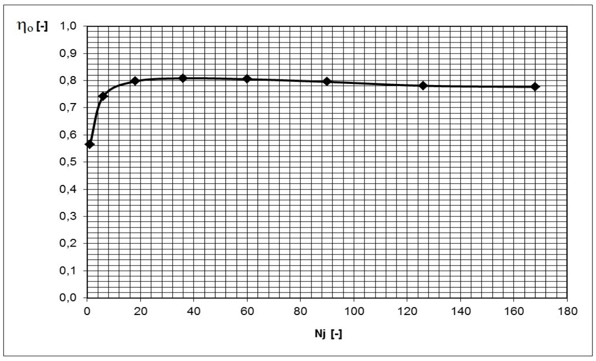

Figure 8. The optimal overall efficiency against the number of ducts

MAIN RESULTS

The best overall efficiency can be reached with 36 ducts. This is much better than the overall efficiency of one-duct type and six-duct type, it is slightly better than the overall efficiency of 18-, 60-, 90-, 126-, 168-duct types. The reason of the decrease of the overall efficiency in case of larger number of ducts is the increasing transport work demand of the heat transport medium and the increasing heat-loss.

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

The location of the optimal combination of the design variables can be seen in Figure 9: a version N_j=1; b version N_j=6, N_j=18, N_j=36, N_j=60; c version N_j=90, N_j=126, N_j=168. The grey area is the available range of the design variables.

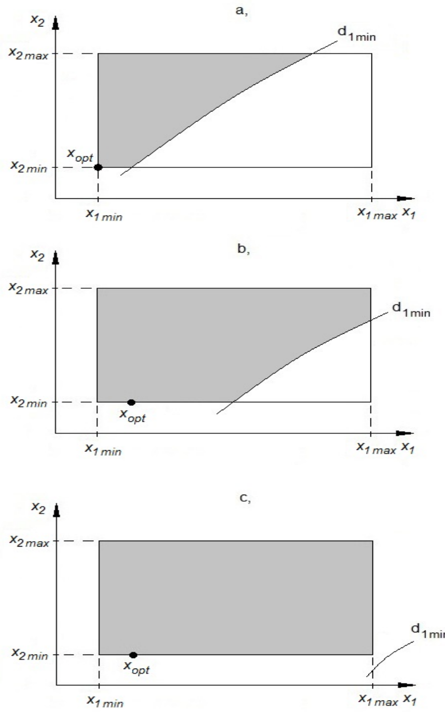

Figure 9. The location of the optimal combination of the design variables

In case of technical tasks the location of the optimal combination of the design variables is often at the border of the available territory. The location of the opimum may move by changing the basic data. In this case it is easy to get general design principles which are suitable to design nearly optimal construction without any optimization process.

The results show that the small minimal material-thickness between two pipe-channels s_h is advantageous.

Greater flow velocity of the gas heat transport medium w_f coluld make shorter thermocline zone, but the optimal flow velocity is lower than 20 percent of the economic flow velocity in pipelines w_{fmin} – because of the greater transport work demand of the greater flow velocity w_f.

The diamaters of the pipe-channel d_1 of the optimal solutions are greater than the lower limit restricted by the manufacturing technology and the mountability.

CONCLUSIONS

I have worked out the constructional and the mathematical model of the long flow-way, pipe-channelled sensible heat store with cascade system. The temperature-place functions and the overall efficiency can be calculated 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 much higher overall efficiency can be reached in case of 36-duct type than in case of one-duct type pipe-channelled heat store.

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

ACKNOWLEDGEMENTS

This work was supported by the research projects TÁMOP-4.2.2/A-11/1KONV-2012-0071 and TÁMOP-4.1.1.C-12/1/KONV-2012-0017.

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