Design and application of sensitive thermal energy storage from concrete.

D Rusovs¹, S Jaundālders¹ and P Stanka¹
¹Riga Technical University, Department of Heat and Power Engineering Systems, Kalku Str. 1, Riga, LV-1658, Latvia
E-mail: dmitrijs.rusovs@rtu.lv

Abstract. The main objectives of the authors’ current paper include thermal sizing of a regenerative concrete heat exchanger for coupling of the biomass plants and the heat engine like ORC (Organic Rankine Cycles). The main challenge for regenerators is the temperature difference between charging and discharging process. Heat energy charging can be done by hot flue gas or high temperature fluid circulation through solid modules of storage. Optionally heat supply can be obtained by using of electrical energy. The concrete modules contain embedded piping for charging and discharging and definition of the geometric parameters like piping size and distance represented in current work. The application of unsteady heat transfer equations for heat storage description was checked by comparison with experimental and theoretical results. The paper offers equations for evaluation of storage efficiency for different heat transfer coefficient, dimensions of storage and time duration of warming and cooling process.

1. Introduction
The need for energy storage is created by the intensive use of renewable energy resources [1,2]. The waver energy generation requires storage in order to provide stable supply in grid and network. Because the energy consumption of today is also fluctuating, energy is no longer used when it is produced. Nowadays production gear and load gear are no longer the same. This means that there is a surplus at sometimes and a lack at other times. It is the job of storage to restore the balance between these two. The term “load gear” refers to the time course of the power consumption by the end users. Another reasons for energy storages is energy price high volatility [3,4] and reduction of feed-in-tariff for “green” energy. Furthermore, more heat accumulators are used to reduce the emission of carbon dioxide in the heat sector [5,6]. The wide adopted water sensitive energy storage for district heating systems are designed for operation at water saturation temperature at ambient pressure. Due to that water heat accumulator had season bounding and had restriction in thermal energy conversation to electrical power. Rapid development of the heat-and-power coupling systems request application of material with higher operational temperature. In particular, for application of concentrated solar power were created high temperature concrete for sensible heat storage system [7]. However, the cracking due to hoop stress of concrete need to be limited at high temperature application. At temperature 340°C concrete Heatcrete® (data adopted from [8]) ensure heat capacity 0.75 kWh/(m³·K) and thermal conductivity 2.2 W/(m·K). After high temperature testing the storage elements from Heatcrete® were cut and investigation show absence of visible thermal crack [8].
2. Heat-to-power ORC cycle with sensitive heat energy storage.

The focus of this paper is heat to energy conversation by application of solid storage, mainly from concrete with embedded pipes system. The Organic Rankin Cycle (ORC) become common solution for electrical power generation from renewable energy [9]. For high-temperature range typical efficiency of ORC is 15-20%, but when operating temperature drop to 130-160 °C efficiency become only 10-13%. The Fig.1 represent combination of energy supply and storage together with ORC turbo generator 5.

![Diagram of ORC with heat energy storage](image.png)

**Figure 1.** Organic Renkin cycle with heat energy storage.

The energy from heat source 1 (solar, biomass combustion, geothermal etc.) is supplied by heat transfer fluid (HTF) to storage system 3. The design of the geometry parameters like pipe diameter and distance between them in storage is crucial important for efficient output of charging-discharging of energy storage. During charging cycle pump 2 provide circulation of HTF through source 1 and storage 3. When storage reach temperature close to HTF inlet temperature it is possible to start discharge process, then heat energy will be desorbed from accumulator and will provide evaporation of working fluid in boiler 4 to supply steam for ORC operation. After expansion in the turbine organic steam will condensate in heat exchanger 6 and pump 7 will ensure circulation through boiler 4. To make figure simpler we do not show energy recuperation in ORC cycle, as we not going to cover in this paper efficiency of ORC performance. There are three possible operation mode in Fig.1 first is when heat energy come directly from heat generator 1, second is when heat come from accumulator 3 and third one is combination of accumulator and limited heat power from generator 1. The combination of accumulator and heat generator is possible only when heat power not limited by time of day like for solar energy, but it is necessary to follow given heat consumption as it could be with biomass boiler and heating system. In this case become possible keep permanent ORC performance with limited fuel consumption by desorbing energy from heat accumulator. Due to that paper deal with two issue: how geometry and flow parameters of HTF influence accumulator output and how accumulator absorption and desorption of power will depend from duration of charging-discharging process.
3. Sensitive energy storage description as regenerative heat exchanger.

Energy storage in thermal accumulator can be described as regenerative heat exchanger performance [10]. Dimensionless number used for heat transfer can be calculated as

\[ NTU_0 = \frac{1}{C_{\text{min}}} \left[ \frac{1}{(hA)_c} + \frac{1}{(hA)_d} \right] \]  

(1)

where \( C_{\text{min}} = m \dot{C} \) is minimum heat capacity ratio of heat carrier flow during charging (heating) and discharging (cooling) process, \((hA)_c\) and \((hA)_d\) are products of heat transfer coefficient to heat transfer surface area during charge-discharge process.

Fixed-matrix regenerator wall heat capacity rate calculated as

\[ C_r = \frac{M_W C_W}{P_t} \]  

(2)

where \( M_W \) is mass of solid regenerator and \( C_W \) is heat capacity, \( P_t \) is charging and discharging cycle time. Dimensionless \( C_r^* \) value for solid storage equal to

\[ C_r^* = \frac{C_r}{C_{\text{min}}} \]  

(3)

If both flows in charge and discharge process had equal capacity ratio than efficiency of counter flow regenerator can be determined as

\[ \varepsilon = \frac{NTU_0}{NTU_0 + 1} \]  

(4)

The influence of \( C_r^* \) value of efficiency given by equation (3) for case when \( \varepsilon \) is less than 90% represented by empirical expression adopted from work [10]

\[ \varepsilon_r = \varepsilon \left( 1 - \frac{1}{9(C_r^*)^{1.93}} \right) \]  

(5)

If substitute data from work [1], where charging and discharging cycle time was around \( P_t = 24 \) hours, but mass of concrete wall was \( M_w = 324 \) kg, heat capacity of concrete \( C_W = 0.940 \) kJ/(kg·K), heat transfer surface was \( A = 1.37 \) m\(^2\) (we take for consideration only one arrow of eight concrete module in series [1]) and mass rate of air flow for speed 4.4 m/s equal to 0.126 kg/s, than \( C_{\text{min}} = 125 \) W/K and heat transfer coefficient was \( h = 16.4 \) W/(m\(^2\)·K). Performance of this concrete accumulator was described by numerical model proved later by experimental measurement.

Calculation by equation (5) will give negative value for efficiency of storage performance if substitute values from considered work [1]. This result show that regenerator calculation approach not valid for considered case. The regenerator description not include transient heat transfer in solid wall by conductivity, what can be valid for small thickness of wall, but in case of massive wall and long charging period is not acceptable.

Since heating and cooling period in solid storage can be as long as 12 to 24 hours, unsteady heating and cooling of storage wall should be involved also in estimation temperature ramp rate in accumulator.
4. Unsteady heat transfer application for description of energy storage design.

Transient heat conduction involve time as variable [11]. The purpose of calculation is to determine temperature distribution as function of position and time within solid body. Assume the thermal conductivity k of solid body and average convection heat transfer coefficient h as constant value. If represent concrete module as slab, then characteristic length will be obtained by dividing the volume of module V by A- heat transfer surface area.

\[ L = \frac{V}{A} \]  

From data considered [1] we take value for L=0.1 m as average between full volume of module and reduced by volume of air duct inside module.

Reduce to 1D unsteady heat conduction in a flat plate wall of thickness L from initial temperature \( T_i \) and with one surface insulated and the other subjected to convective heat transfer into contacting surrounding with temperature \( T_0 \). Introduction of the dimensionless parameters will simplify the solution of temperature distribution. Biot number represent the ratio of conductive heat flux resistance to resistance of convective heat flux. When Biot number is less than 0.1 the transient problem can be considered as “lumped thermal capacity” method or solid body had single average temperature.

In case we are consider Biot [1] number is equal to

\[ Bi = \frac{hL}{k} \]  

where \( k = 1.7 \) W/(m·K) is thermal conductivity for concrete in module, \( h = 16.4 \) W/(m²·K) air flow convective coefficient according [1]. Since Bi =0.96 lumped system analysis will be not valid.

Dimensionless temperature distribution \( \theta(x,t) \) in wall of module can be reduced to consideration of flat plane wall with thickness L.

\[ \theta(x,t) = \frac{T(x,t)-T_i}{T_0-T_i} = A_1 e^{(-\mu_1^2 Fo)} \cos(\mu_1 \frac{x}{L}) \]  

where \( x=0 \) represent surface of flat plane in contact with flow, while coordinate \( x=L \) for isolated surface of wall. This equation is valid if dimensionless time or Fourier number is more than 0.2

\[ Fo = \frac{a t}{L^2} \]  

where a is thermal diffusivity for concrete equal to ratio of thermal conductivity k to product of specific heat (940 J/(kg·K)) and density of concrete (2820 kg/m³)[1]. As result a = 6.410⁻⁷m²/s. The constant in equation (8) \( A_1 \) and \( \mu_1 \) are function of the Biot number only and their value are tabulated.

The paper [1] present theoretical model for calculation of mean temperature of accumulator for different heat convection coefficient values. For the purpose to calculate mean temperature \( \bar{T} \) (t) of flat plain wall temperature are used equation

\[ \bar{\theta}(t) = \frac{\bar{T}(t)-T_i}{T_0-T_i} = \frac{2\sin^2(\mu_1)}{\mu_1^2+\mu_1^4 \sin(\mu_1) \cos(\mu_1)} e^{(-\mu_1^2 Fo)} \]  

During heating cycle (charge process) mean temperature of accumulator can be calculated as

\[ \bar{T}_{AC}(t) = T_{OC} - \bar{\theta}(t) \left( T_{OC} - T_{AO} \right) \]
where $\overline{T}_{AC}(t)$ is average temperature of plain wall during charging (heating) of accumulator, $T Oc = 60^\circ C$ is temperature of air flowing in duct and $T AO = 20^\circ C$ is initial accumulator temperature.

Similar way is possible to calculate temperature during cooling (discharge of accumulator)

$$\overline{T}_{Ad}(t) = T Od + \theta(t) (T_{AC} - T Od)$$

where $\overline{T}_{Ad}(t)$ is average temperature of plain wall during discharging (cooling) of accumulator, $T Od = 20^\circ C$ is inlet temperature of flow warming from accumulator (during discharge process) and $T_{AC}$ is temperature of accumulator after end of charging cycle.

Fig. 3 represent mean temperature distribution in concrete wall calculated for condition of [1]. It is obvious compatibility or theoretical – experimental results predicted in [1] fig.2 and calculated for unsteady heat transfer fig.3. The numerical and qualitative coincidence ensure validity of our developed approach by transient conductivity mathematical model. Also our suggestion about accumulator representation by flat wall with given characteristic length can be considered as proven.

Heat power absorbed by accumulator during charging cycle possible to estimate as function of heating time during charging process

$$N_C(t) = h A \left( T_{OC} - T_A(x = 0, t) \right) = h A \left( T_{OC} - T_{AO} \right) A_1 e^{-\mu_1^2 F_0 t}$$

where $h$ is convection coefficient, $A$ is heat transfer surface and $T_A(x=0, t)$ temperature of accumulator surface calculated by (8).
Similarly, possible to calculate heat power absorbed by carrier (gas or liquid) flow from accumulator for discharge process

\[ N_d(t) = h A(T_A(x=0,t) - T_{OD}) = h A(T_{AC} - T_{OD}) A_1 e^{-\mu_1^2 F_0} \] (14)

where \( T_{OD} \) is temperature of carrier which absorb heat from accumulator during discharging time \( t \), and \( T_{AC} \) is mean temperature of accumulator before discharging.

Maximum amount of energy, which can be absorbed by accumulator equal to

\[ Q_{MAX} = M_W C_W (T_{OC} - T_{AO}) \] (15)

Actual amount of energy charged in accumulator when mean temperature of accumulator will reach value \( T_{AC} \)

\[ Q_C = M_W C_W (T_{AC} - T_{AO}) \] (16)
Then efficiency of charging during time $t_C$

$$\eta_C = \frac{(T_{AC} - T_{AO})}{(T_{OC} - T_{AO})} = 1 - \theta(t_C) \quad (17)$$

where function $\theta(t_C)$ according equation (10) will describe mean temperature of accumulator as time function.

During discharge process heat amount absorbed by carrier flow from accumulator

$$Q_D = M_W C_W (T_{AC} - T_{AD}) \quad (18)$$

where $T_{AD}$ is temperature at the end of discharge process during time $t_D$, then efficiency of discharge become equal to

$$\eta_D = \frac{(T_{AC} - T_{AD})}{(T_{OC} - T_{OD})} = 1 - \theta(t_d) \quad (19)$$

The total efficiency of charge- discharge cycle will be function of charge time $t_C$ and discharge time $t_d$ will be represented as product of equations (17) and (19)

$$\eta_{CD} = (1 - \theta(t_C))(1 - \theta(t_d)) \quad (20)$$

In case of steady flow through accumulator during charge-recharge process duration of this cycles will determine efficiency together with geometrical size of storage.

For application described in Fig.1 liquid flow of heat transfer oil will be used as heat carrier therefore heat transfer coefficient calculated by well-known equation for Nusselt number in turbulent flow

$$Nu = \frac{h_0 L}{K_0} = 0.023Re^{0.8}Pr^{0.4} \quad (21)$$

where $h_0$ is convective heat transfer coefficient, $L=0.1$ m is characteristic length, $K_0 =0.1$ W/(m K) thermal conductivity for oil in pipe with diameter $d=0.05$ m, $Re=dW/\nu$ represent Reynolds number and $Pr = 8.6$ is Prandtl number. Assume oil flow velocity in pipe as $W=2$ m/s. If take data for oil [12] where kinematic viscosity $\nu = 4 \cdot 10^{-7}$ m$^2$/s, then value of $h_0 = 2201$ W/(m$^2$·K).

The Biot number value according (7) become $200 > 100$ and for mean temperature of accumulator will be used following equation

$$\bar{\theta}(t) = \sum_{n=1}^{\infty} \frac{8}{\pi^2(2n-1)^2} \exp \left(-\frac{(2n-1)^2}{2}\pi^2 Fo\right) \quad (22)$$

If value of Fourier number (9) will be more than 0.3 is possible to reduce expression (22) to only first term, than efficiency of charge- discharge cycle for accumulator can be expressed as

$$\eta_{CD} = (1 - 0.81 \exp (-2.47Fo_c))(1 - 0.81 \exp (-2.47Fo_d)) \quad (23)$$

where $Fo_c$ and $Fo_d$ will represent time of charge and discharge process for accumulator.

5. Summary

The concrete energy storage is an approach to provide heat-to-energy flexible operation for renewable energy sources like solar, biomass, geothermal heat generation by coupling with thermal engine via
accumulator. The paper show limitation of regenerator theory application in case of massive regenerator with long accumulation period.

According to the suggested unsteady heat transfer model is possible to describe solid accumulator performance as function of dimensionless numbers Fourier and Biot. This numbers contain information about properties of the storage – heat conductivity of solid material and flowing media, viscosity and density, geometry of system piping size and distance between them in concrete, flow velocity and duration of process. The design of storage size become easy to couple with efficiency of charge-discharge process by application of given equations and tabulated values of parameters.

Definitely, is important to conduct more comparison with experimental condition for different media than air, but it is significant to understand that changing condition of flow - velocity, temperature will create more challenge for simple mathematical model of storage. In case of steady flow and fixed inlet temperature of heat carrier our results show that heat power of absorption-desorption of accumulator will quickly decrease. That is fundamental drawback for solid sensitive storage. Therefore, evaluation of transient heat transfer processes is crucial important for total efficiency of energy accumulation.

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