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Introduction of the PCM Flux concept for latent heat storage

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Abstract

The focus in the development of storage systems using phase change materials is on the implementation of heat transfer concepts compensating the limited heat conductivity of the storage materials. This paper introduces the PCM Flux concept as a new alternative for latent heat energy storage. Here, the storage material is separated mechanically by an intermediate fluid layer from the heat transfer surfaces. This approach avoids the formation of a growing layer of solidified storage material covering the heat transfer structure, which limits the heat flux. This paper outlines the PCM Flux concept and gives results of its theoretical analysis.

Keywords: Thermal Energy Storage; Phase Change Material (PCM); Stationary/Moving PCM Storage; Separation of Power and Capacity

1. Introduction

In times of a rising share of fluctuating renewables in the worldwide energy mix, there is a growing demand for both cost and energy efficient energy storage units allowing harmonization of electricity production and consumption. Using phase change materials (PCM) in isothermal processes to store heat promises a high overall efficiency due to minimization of exergy losses resulting from temperature differences. This high efficiency, however, is associated with the need to compensate the poor thermal conductivity in cost-effective PCMs. Today, the required amount of heat transfer structure (HTS) to achieve a sufficient effective thermal conductivity is coupled to the storage capacity and upscaling a storage unit is not expected to reduce costs significantly. While discharging, the maximum heat flux

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decreases with time due to a growing and isolating layer of solidified PCM on the heat exchanger modules. As a result of this, the heat exchanger has to be designed for a small heat flux at the end of the discharging process leading to a high necessary amount of HTS in order to reach the claimed nominal power output. One possibility to overcome this conceptual problem is the development of new concepts with transportable PCM as the storage material. In such systems, the isolating layer of PCM is moved away from areas with the highest heat transfer. The heat flux can thereby be controlled accurately by the PCM`s velocity. In these concepts, the design of the heat exchanger can be based on higher heat fluxes leading to a minimization of expensive HTS.

This paper gives an overview and a categorization of known latent heat storage concepts and their specific ways to deal with the poor heat transfer within the storage material. Moreover, the PCM Flux system is outlined and the theoretical background for this concept is given.

2. Categorization of latent heat storage systems

The design of latent heat storage has to account for the poor thermal conductivity of the storage material. KNO_3, $NaNO_3$ and their combinations are typical PCMs for high temperature storage applications with an average thermal conductivity of 0.5 – 0.59 W/mK [1,2]. Especially while discharging the thermal energy storage, the thermal conductivity is critical. In this operational mode, the frozen PCM suppresses convection effects close to the heat transfer structures and heat conduction is the dominating heat transfer mechanism. There are several ways to enhance heat transfer in PCM storage units to compensate this effect. References [2,3] give an overview of ways to enhance heat transfer in PCM storage systems. There is a new class of PCM storage systems with moving pure storage material. This leads to an expanded categorization, see Fig. 1.

There are two main groups within this categorization. Concepts with locally fixed storage material are defined as *stationary PCM* systems, whereas systems in which the PCM does not stay in one place are called *moving PCM* storage concepts. All of the latent heat storage concepts shown in Fig. 1 are briefly described in the next chapter.

2.1. Stationary PCM Concepts

Stationary PCM concepts can be subdivided into *Surface Concepts* and *Heat Conductivity Concepts*. According to Fourier`s Law of Conduction, there are two possibilities to enhance heat transfer in conduction processes. Assuming a constant temperature difference, either the active surface participating in heat transfer or the effective thermal conductivity have to be enlarged to improve heat conduction.

One way to enlarge the surface as part of the surface concepts is to embed *Fins* connected to the heat transfer modules into the PCM. This system was developed and tested by the German Aerospace Center (DLR) [4,5]. The feasibility of this concept was demonstrated by operating the system for approx. 3000 h.

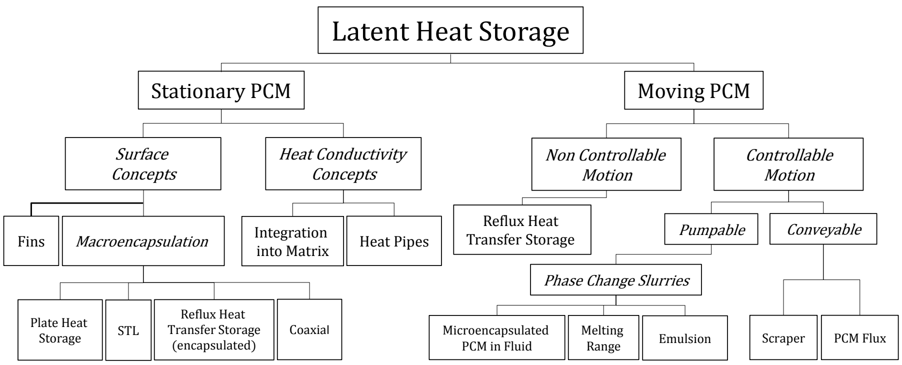

Fig. 1. Categorization of latent heat storage systems

This is the only large scale high temperature PCM storage system demonstrated so far. Because of that, it is defined as the state-of-the-art PCM storage system to be referred to.

There are many other concepts using the extension of heat transfer area that can be summarized under the topic of *Macroencapsulation*. In these systems, the storage material is enclosed in containers, boxes or spheres of the size of millimeters to some centimeters aiming to increase the contact area of the heat transfer fluid (HTF). These single elements packed upon one another show a bigger effective porosity of the storage inventory than the PCM directly put into the storage tank itself. There are different ways to arrange the encapsulated PCM modules. Either they are randomly put into a tank creating a packed bed of bigger granular size than with PCM only or they are strictly located in a calculated order to improve the flow of HTF through the individual elements. One of the packed bed concept is called *STL* (*stockage latent;* French for latent heat storage) and is outlined in [6]. The packed bed is thereby directly perfused by the HTF and the surface of each PCM module is in contact with the HTF. Another concept is the encapsulated *Reflux Heat Transfer Storage*. The storage material is arranged the same way as it is in the STL system. The difference is the way of heat transfer. The PCM capsules are not in direct contact with the HTF. An additional fluid within the storage tank changes phase from liquid to vapor or vice versa while transferring the heat [7].

There are several designs of thermal energy storages that take the strict arrangement of encapsulated PCM as the basis. The *Plate Heat Storage* concept works like a regular heat exchanger, whereby the secondary fluid between the corresponding plates is replaced by enclosed PCM, for details see reference [8]. The arrangement of PCM along the outer face of a pipe carrying HTF inside bounded by another pipe of bigger diameter is called a *Coaxial* concept, details can be found in references [8,9].

To enhance heat transfer of the PCM, the thermal conductivity can be manipulated by different methods to achieve a better overall effective thermal conductivity. One possibility to do so, is to integrate the PCM into a matrix of highly conductive materials, e.g. into graphite. Details can be found in reference [10]. Another way to improve heat conduction is to bridge longer distances between heat source/ heat sink and the area of phase changing storage material by heat pipes, see references [11,12].

All of these stationary PCM concepts have one disadvantage in common: Especially while discharging the storage unit, the thermal output power cannot be kept constant at constant pressure. The growing frozen layer of PCM represents an increasing thermal resistance that results in a significant drop of power. This is a systematic problem which can be solved by continuous removal of the frozen layer from the heat transfer area. This is the motivation of moving PCM concepts.

2.2. Moving PCM Concepts

As can be seen in Fig. 1, the different moving PCM concepts are divided into systems with *Non Controllable Motion* and systems with *Controllable Motion*. The *Reflux Heat Transfer Storage (non encapsulated)* has a similar operating strategy to the encapsulated version (see 2.1). In this concept, the movement of the storage material is based on gravity and convection effects and therefore cannot be controlled directly. For details see references [7,13,14].

In the field of controllable motion concepts, one can differentiate the concepts by the means of moving the storage material. In some concepts, the storage material stays pumpable all the time and can be directly stored in tanks. *Phase Change Slurries* are a combination of latent and sensible heat storage. In the case of the *Microencapsulated PCM in Fluid* concept, the sensible heat of a fluid is used to transport heat. This fluid is usually combined with PCM microencapsulated in spheres. The latent heat of the encapsulated PCM thereby is used, resulting in a higher effective heat capacity of the slurry [15–17]. Using *Emulsions* as both HTF and storage material is another possibility for direct moving PCM storage. The operating principle is similar to the microencapsulated PCM in Fluid concept. But in emulsions, the small particles of PCM material are not physically separated from the surrounding fluid. Due to the physical properties of the involved materials, the two phases are stable and do not get mixed, for details see references [15,18]. Another way to realize pumpable moving PCM storages is the use of a non-eutectic mixture of a PCM combination. Mainly the sensible heat of the medium above the eutectic melting temperature is used. While cooling the completely melted PCM to a temperature still above the melting temperature of the whole mixture, one fraction with a different composition freezes according to the phase diagram of the mixture. Only the latent heat of this fraction is used. Overall, the whole fluid stays liquid and therefore pumpable. This kind of a moving PCM concept uses the effects of a *Melting Range*. There is little referenced information available on that topic but some slides can be found in [19,20] and a recent article about the theory and possibilities of melting range thermal energy storage systems in [21].

Moving PCM concepts that make use of pure PCM as the storage material and of a complete phase change can be found in Fig. 1 under the topic *Conveyable*. To remove the frozen PCM from the heat exchanger continuously, *Scrapers* can be used. These scrapers can have different shapes, but basically these instruments scratch the sticking and solid PCM off the heat exchanging surface. The thereby resulting PCM fragments are collected, moved to another place and stored externally. While charging the storage, these solid fragments are transported back to the heat exchanger successively and are molten again inside. One example for such a concept is the *Screw Heat Exchanger* [22,23].

PCM Flux, the concept that is presented in this paper, fits into the same category as the latter concept. It is described in detail in the following chapter.

3. PCM Flux as a moving PCM storage

PCM Flux as a moving PCM concept combines several advantages. First of all, an accurate control of the thermal in- and output power over the whole operating time is possible. Additionally, the heat flux does not decrease with time. PCM Flux represents a latent heat storage system with strict separation of power and capacity. The following chapters describe the system, give the theoretical background and show simulation results.

3.1. Description of PCM Flux

PCM Flux consists of macro encapsulated PCM blocks that are arranged as parallel layers, for details see Fig. 2 (a). The PCM blocks are separated by a fluid that ensures thermal conduction while the blocks are decoupled from the rest of the system to stay movable [24]. The layers are separated by heat exchanger pipes, fins and insulation. The PCM blocks are moved slowly in one direction for charging the storage and in the other for discharging. They thereby pass the fixed heat exchanger pipes. Once the material arrives at the next pipe, the storage is fully charged or respectively discharged. Due to the enormous amounts of storage material necessary for full scale thermal energy storage, moving the PCM provokes significant parasitics. The PCM Flux concept reduces this problem by letting the encapsulated PCM blocks float in the thermal conduction fluid. The velocity of the PCM blocks controls the actual thermal power of the storage system. If this velocity is chosen within a specific interval, the phase change interface of the PCM is locally fixed and a quasi-stationary state is established. This state leads to a constant thermal power in- and output of the system. Depending on the location of the interface, variable partial load states can be achieved. Fig. 2 (b) shows exemplarily the influence of moving the PCM blocks on the resulting heat flux. If the PCM blocks are moved, the heat flux stagnates on a plateau, whereby in case of non-moving PCM blocks (“PCM Flux stationary”) the heat flux curve shows the for stationary latent heat storage systems typical decrease over time.

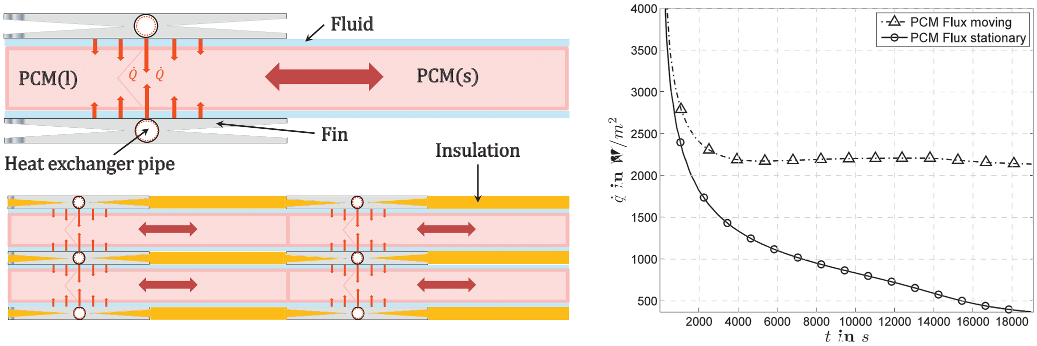

Fig. 2. (a) PCM Flux module and the structure of the stack; (b) Heat flux curve of moving and stationary PCM Flux over time (for configuration data see Table 1)

The design of the fins made of aluminum attached to the heat exchanger pipes has to be optimized. The less material used, the lower the costs. The shape of these fins in Fig. 2 (a) is just an example and will be an important subject of future research. Choosing a fluid that can permanently withstand the high temperature is a challenging task. Depending on the storage material, e.g. in the case of $NaNO_3$, temperatures of 315°C are reached while charging the storage. Most thermal oils need a pressurized ambience to prevent evaporation at these temperatures. This would make the system both more complex and expensive. Moreover, the thermal conductivity of most thermal oils is even worse than that of PCMs. As a result of these reasons, another PCM with a lower melting temperature than the actual storage material is chosen as temperature-resistant fluid.

The heat has to pass different areas to get from the heat exchanger pipes into the storage material and vice versa. In the case of charging the storage, the heat is conducted from the heat exchanger pipe through the aluminum fins, the fluid and the container wall until it reaches the PCM, see Fig. 2 (a). All these thermal resistances have to be minimized in order to reach a maximum heat flux. The next chapter shows how the different influences correlate with each other.

3.2. Simulation results and theory

For numerical simulation of the PCM Flux concept, a finite-difference-method tool based on the Enthalpy-Method was set up. To accelerate the computational speed, an implicit approach was chosen to enable bigger time steps. The program regards different materials, such as different PCMs and different sorts of aluminum and steel. The implementation of the moving parts of the PCM Flux concept was given a high priority. It was realized by the implementation of flexible boundary conditions that change with time, depending on the velocity v_{PCM} of the PCM blocks. The program is a full-value 2-D thermal conduction tool specialized in PCMs. However, heating and cooling of sensible materials and PCMs can be described as well. The program is very flexible, shapes of any parts, materials and dimensions can be changed or adjusted very quickly. In this way, variation of parameters can easily be performed. The tool is based on the enthalpy form of the two dimensional heat conduction equation:

$$\frac{\partial H}{\partial t} = \lambda \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right) \tag{1}$$

Equation (1) contains the volumetric enthalpy H, the time t, the thermal conductivity λ, the temperature T and the directions x and y. The outer borders of the model are implemented with adiabatic boundary conditions. The temperature at the inner heat exchanger pipe surface is imposed by a boundary condition of first order, due to the constant temperature of the phase changing water/steam inside the pipes. Table 1 states all important data for further considerations in this paper. These data reflect an example configuration of the PCM Flux system; the optimized configuration is not disclosed yet, due to a pending patent.

Table 1. Data of an example PCM Flux configuration referred to in this paper

Configuration data	Value	Unit	Configuration data	Value	Unit
$A_{PCM}, A_P, A_C, A_F, A_{Fin}$	0.04, 0.064, 0.002, 0.18, 0.0023	m^2	$\Delta T, \Delta T_F, \Delta T_{eff,fin}, \Delta T_{eff}$	10, 0.77, 8.59, 7.82	K
$B_F, S_{Fin}, d_P, P_{Fin}$	0.001, 0.18, 0.0204, 1	m	λ_F, λ_{Fin}	0.5, 210	W/mK
ρ_{PCM}, ρ_C	2017.5, 7872	kg/m^3	$v_{PCM,max}$	$1{,}395 \cdot 10^{-5}$	m/s
c_{PCM}, c_C	1421, 481	J/kgK	$\dot{q}_{max}$	2167	W/m^2
L	108000	J/kg	K_{Flux}	1,403	–

Simulation results show a linear correlation between the velocity of the PCM blocks v_{PCM} and the resulting heat flux $\dot{q}$ at the inner surface of the heat exchanger pipes. This correlation is shown via a general characteristic curve $\dot{q}$ over v_{PCM} that can be seen in Fig. 3 (a). The simulated characteristic curve based on the data of Table 1 can be seen in Fig. 3 (b) “Simulation”. These figures clearly show the mainly linear correlation of $\dot{q}$ and v_{PCM} from a minimum state on until $\dot{q}_{max}$ as the nominal power of the system. After exceeding this point by a further increase of v_{PCM}, the characteristic curve leaves its linear shape significantly. Knowing the slope of the linear part of the characteristic curve shown in Fig. 3 (a) and the location of $\dot{q}_{max}$, the PCM Flux system can be designed and calculated. The nonlinear beginning of the characteristic curve for small v_{PCM}, see Fig. 3 (a)+(b), is marginal and can therefore be neglected.

Analyzing the simulation results of different PCM Flux configurations and examining the dimensions and influences of the different factors shown in Table 1, the dimensionless number K_{Flux} was developed and is introduced together with its calculation rule via equation (2).

$$K_{Flux} = \frac{v_{PCM} \cdot L \cdot \rho_{PCM}}{\dot{q}} = \frac{\overbrace{A_{PCM} \cdot v_{PCM} \cdot L \cdot \rho_{PCM}}^{\dot{Q}_L}}{\dot{q} \cdot A_P} \cdot \left(\frac{A_P}{A_{PCM}}\right) = \frac{A_P \cdot L \cdot \rho_{PCM}}{A_{PCM} \cdot \rho_{PCM}\,(L + c_{PCM} \cdot \Delta T) + A_C \cdot \rho_C \cdot c_C \cdot \Delta T} \tag{2}$$

Equation (2) contains the velocity of the PCM block v_{PCM}, the latent heat and density of the storage material L and ρ_{PCM}, the heat flux related to the inner pipe surface of the heat exchanger $\dot{q}$, the inner pipe surface A_P, the cross sectional area of the PCM block and the container walls in direction of movement A_{PCM} and A_C, the specific heat capacity of the PCM and the container material c_{PCM} and c_C, the density of the container material ρ_C and the imposed temperature difference ΔT at the inner surface of the heat exchanger pipes related to the melting temperature of the storage material. For details and visualization see Fig. 4. Rearranging the first part of equation (2) leads to the slope shown in Fig. 3 (a). After expanding this fraction by A_P and A_{PCM}, $\dot{Q}_L$ and $(\dot{q} \cdot A_P)$ can be identified. $\dot{Q}_L$ thereby describes the heat flow necessary for the phase change of the storage material only. This heat flow is divided by $(\dot{q} \cdot A_P)$, describing the whole heat flow passing through the heat exchanger pipe including the heat flow for phase change $\dot{Q}_L$ and additionally the heat flows for the sensible heating or cooling of the storage and container material $\dot{Q}_{s,PCM}$ and $\dot{Q}_{s,C}$. According to first law of thermodynamics, these heat flows can be calculated via equation (3).

$$\dot{q} \cdot A_P = \dot{Q}_L + \dot{Q}_{s,PCM} + \dot{Q}_{s,C} = \dot{Q}_L + A_{PCM} \cdot v_{PCM} \cdot c_{PCM} \cdot \rho_{PCM} \cdot \Delta T + A_C \cdot v_{PCM} \cdot c_C \cdot \rho_C \cdot \Delta T \tag{3}$$

Inserting equation (3) into the second part of equation (2) and cancelling out v_{PCM} leads to the definition shown at the end of equation (2). K_{Flux} now can be calculated with just the properties of the involved materials and the geometry data of the PCM Flux modules. Knowing K_{Flux}, the slope of the linear part of the characteristic curve is known and therewith the corresponding pairs of $\dot{q}$ and v_{PCM} can be found immediately in the area of operation, see Fig. 3 (a)+(b).

When the slope of the characteristic curve is known, the maximum point $\dot{q}_{max}$ where the even line of the characteristic curve drops is calculated. Being able to determine the corresponding pair $\dot{q}_{max}$ and $v_{PCM,max}$ is essential. In case $v_{PCM,max}$ is exceeded, the PCM block is moved too fast and the storage material cannot change phase completely while passing the heat exchanger fins. In literature [25], an analytical approach to calculate the phase change time t_{pc} of a PCM block can be found as the *Neumann Solution*. This is a 1-D solution and thus has to be adapted to the PCM Flux specific circumstances. Assuming a stationary PCM block that does not move, one can calculate t_{pc} using the Neumann Solution. t_{pc} is the minimum time the PCM block has to be imposed with the fully ΔT allowing the PCM block to change phase completely. Taking the length of the heat exchanger fins S_{Fin} into account, the maximum velocity $v_{PCM,max}$ can be calculated via length per time. This correlation is only valid, if the fins show a perfect thermal conductivity and if no temperature gradient occurs along these fins. Only in this case, the PCM block would be imposed by the complete ΔT over the whole length S_{Fin}, while moving the PCM towards the heat exchanger pipes along the fins. This assumption of course does not match with reality. A solution for this problem is given later. So far, the fluid layer is not considered in the model. This layer is responsible for a significant temperature drop ΔT_F, due to its small thermal conductivity and must not be neglected. Unfortunately, ΔT_F itself is a function of $\dot{q}_{max}$ according to Fourier`s Law of Conduction. That is why an iterative solution for determining $\dot{q}_{max}$ is required.

The temperature drop over the fluid layer is calculated using K_{Flux} and Fourier`s Law of Conduction.

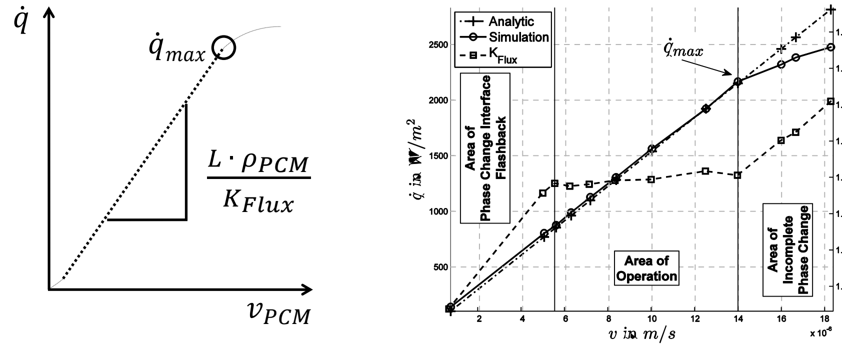

Fig. 3. (a) General characteristic curve of the PCM Flux system showing $\dot{q}$ over v_{PCM} with its slope and rated power $\dot{q}_{max}$; (b) Simulation results of the characteristic curve, the analytic approximation of the characteristic curve, $\dot{q}_{max}$ and K_{Flux}

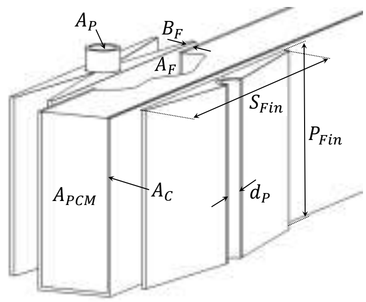

Fig. 4. Visualization of the main construction parameters of one PCM Flux module

Once the first guess for $v_{PCM,max}$ is available according to the Neumann Solution (S_{Fin}/t_{pc}), the corresponding heat flux $\dot{q}_{max}$ under the use of K_{Flux} is determined, see equation (2). This heat flux is used to calculate ΔT_F via Fourier`s Law of Conduction, see equation (4). A_F represents the contact area of the heat exchanger fins and the fluid. A_P corresponds to two fluid layers beside the pipe. Calculating ΔT_F over one fluid layer, the half of A_P is used, see Fig. 4.

$$\Delta T_F = \frac{B_F}{\lambda_F \cdot A_F} \cdot \frac{(\dot{q}_{max} \cdot A_P)}{2} \tag{4}$$

Expanding the model by the already mentioned temperature drop along the heat exchanger fins, one can use a modified version of existing fin efficiency equations. The original imposed ΔT at the inner pipe surface can be assumed to be applied at fin areas close to the pipe only. Along the fins with growing distance to the heat exchanger pipe, the effective ΔT decreases and t_{pc} determined by the stationary Neumann Solution respecting ΔT gives a too optimistic guess. In [26], a correlation for calculating the temperature curve along different shaped fins is given. The assumptions that lead to these equations can only be partly applied for the heat exchanger fins in the PCM Flux concept. The heat is not transferred by convection but by conduction through the fluid and only one side of the fin takes part in heat transfer. Modifying the general fin equation for constant base temperature and adiabatic tip gives the following equations for estimating the temperature curve from the pipe to the tip of the fins.

$$m = \sqrt{\frac{\left(\frac{\lambda_F}{B_F}\right) \cdot P_{Fin} \cdot \Delta T_F}{\lambda_{Fin} \cdot A_{Fin} \cdot \Delta T}} \tag{5}$$

$$\frac{\Delta T_{Fin}}{\Delta T} = \frac{\cosh(m \cdot (a - x))}{\cosh(m \cdot a)} \qquad with\colon \quad a = \frac{S_{Fin} - d_P}{2} \tag{6}$$

Equations (5) – (6) contain the thermal conductivities of the fluid λ_F and of the fins λ_{Fin}, the thickness of the fluid layer B_F and the resulting temperature drop ΔT_F conducting heat through it, the perimeter of the heat exchanging part of the fins P_{Fin} and the imposed temperature difference ΔT. S_{Fin} represents the fins` length, A_{Fin} their mean cross sectional area and d_p the diameter of the pipes, see Fig. 4. Knowing the temperature curve along the fins ΔT_{Fin} by equation (6), $\Delta T_{eff,Fin}$ as an integral averaged effective temperature difference from the pipe until the fin tip (integration boundaries: 0 to a) is calculated. In the opposite to the original ΔT, $\Delta T_{eff,Fin}$ respects the temperature drop along the fins. Calculating t_{pc} via the Neumann Solution does not consider the whole and original temperature difference ΔT, but $\Delta T_{eff,Fin}$. Using $\Delta T_{eff,Fin}$ instead of ΔT for calculating t_{pc} results in a modified and smaller maximum velocity $v_{PCM,max}$.

Finally, the averaged real temperature difference imposed to the PCM block over the whole length of the fins considering temperature drops over the fluid layer and along the fins, ΔT_{eff} can be computed following equation (7).

$$\Delta T_{eff} = \Delta T_{eff,fin} - \Delta T_F \tag{7}$$

ΔT_{eff} is used to calculate again t_{pc} via the Neumann Solution. S_{Fin} is divided by the resulting t_{pc}, leading, after some iteration steps, to an accurate prediction of $v_{PCM,max}$. The iteration process follows the path:

- Calculate $v_{PCM,max}$ via Neumann Solution (S_{Fin}/t_{pc}); first iteration step: $\Delta T_{eff} = \Delta T$
- Determine $\dot{q}_{max}$ using K_{Flux} via equation (2)
- Compute ΔT_F via Fourier`s Law of Conduction with equation (4)
- Calculate $\Delta T_{eff,Fin}$ as the - from pipe to fin tip - integral averaged value of equations (5)-(6)
- Determine the overall effective temperature gradient ΔT_{eff} via equation (7)
- Start again at the first step, now using the new ΔT_{eff} leading to a new t_{pc}

All occurring parameters, see Table 1, can be examined analytically. The analytical results show a very good agreement with the simulation, see Fig. 3 (b) “Analytic” and “Simulation”. Each state in the area of operation, see Fig. 3 (b), can be predicted with a deviation of less than 2.2%. If v_{PCM} is too big or too small, the area of operation is left and the analytic solution presented in this chapter is not valid anymore. The analytic solution is only applicable if the phase change interface is locally fixed in the region of the heat exchanger fins. Incomplete phase change or migration of the phase change interface contrary to the velocity vector of the PCM block (flashback) due to very small values of v_{PCM}, are not regarded by the analytical solution. At the optimized configuration, a flashback only occurs at very small partial load states and can be neglected. Operating the system in the incomplete phase change area must be avoided to secure the utilization of the whole storage capacity. In these two mentioned areas, the value of K_{Flux} varies, see Fig. 3 (b). This fact explains the nonlinear characteristic curve in these areas. Whereas in the area of operation, K_{Flux} takes on a nearly constant value. This proves a linear behavior of the characteristic curve and shows the validity of the theory presented in this chapter for the area of operation.

4. Conclusion and Outlook

Current PCM storage systems mainly have a conceptual disadvantage while discharging. The thermal power decreases over time due to a growing and isolating layer of crystallized PCM on the heat exchanger. Systems with locally fixed storage material, all of which show this mentioned behavior, are defined as *stationary PCM* concepts. *Moving PCM* concepts by contrast, make use of locally unfixed PCM offering the possibility to remove the isolating layer of frozen PCM. Thus a constant thermal power over time is possible.

PCM Flux, as a new representative of moving PCM concepts, is described. The theoretical background including all necessary equations are shown. The dimensionless number K_{Flux} as a correlation of the PCM block`s velocity and the resulting thermal power is introduced. An example of a non-optimized configuration stated in this article shows a nearly constant power over time at a level of 2167 W/m$^2$, that can be easily and fully controlled by the storage material`s velocity. PCM Flux is considered to be a very promising concept, eliminating the problem of decreasing thermal power with time. Together with its full control and the high maximum possible heat fluxes, PCM Flux has the potential to decrease the storage unit costs for direct steam generating solar power plants significantly and to increase the storage`s performance compared to the state-of-the-art system. Further research will deal with an experimental setup to compare the presented theory with practice.

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