Heat flux in latent thermal energy storage systems: the influence of fins, thermal conductivity and driving temperature difference

Julian Quenel1 · Burak Atakan1

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Abstract
Phase change materials (PCM) can increase the energy densities in thermal energy storage systems. Heat transfer rates in PCMs are usually limiting, different improvement methods were used previously, such as fins or improved thermal conductivities. Here, the influence of fin geometries, PCM thermal conductivity and discharge temperature of the secondary fluid are investigated by modelling. The analysed outcomes are their influence on stored energy, heat flux and stored exergy. The two-dimensional time dependent energy equation was solved for a rectangular enclosure with a secondary fluid with constant temperature as boundary condition on one side. The modelled PCM data based on a paraffin melting at 44 °C. The different improvement methods increase the heat flux, but the increase is lower than expected according to idealized calculations. The basis for this investigation is the evaluation of a dimensionless heat flux number formed from the heat flux, thermal conductivity, temperature difference from the secondary fluid to the phase change temperature and a characteristic length of the system. The influence of the better thermal conductivity is found to be lower at higher fin volume fractions. The increase of the discharge temperature difference had the strongest impact on the heat flux, but the exergy loss also increased. The exergy loss was also related to the achieved reduction in the discharge time as an indicator for the heat flux. Here, it was seen that the exergy losses outweigh the advantage in discharge time from a higher discharge temperature difference.

1 Introduction
Energy storages offer the possibility to reduce the temporal discrepancy between demand and supply of energy, which results from the natural fluctuation of renewable energy resources such as sun and wind. Thermal energy storages are one possible choice, and they are often cost-effective and easily implemented.

Latent thermal storage systems store large amounts of thermal energy in a small temperature range, due to their high phase change enthalpies, thus, they are discussed as a more effective way to store energy than sensitive storage systems. The discussed field of application is broad, e.g. in the storage of heat from renewable energy sources, especially solar energy [1, 2], in thermal management for buildings [3–5], in industrial applications [6, 7] or for cooling electronics [8]. One problem with latent thermal storage is the often limited heat transfer due to the low thermal conductivity of most PCMs. Therefore, the improvement of heat transfer is the subject of many investigations [9–12].

The three main improvement methods for the heat transfer are a larger surface area for heat transfer, increasing the driving temperature difference or to increase the thermal conductivity of the PCM. Dispersed substances with better thermal conductivity provide better possibilities for heat transfer in the PCM, which is equivalent to a PCM with higher thermal conductivity. Mostly the systems are compared with respect to the discharge time, the liquid fraction, and the temperature distributions [13–16]. But the heat flux is another crucial process variable. This is not addressed sufficiently in the literature, although most technical applications of thermal storages depend on the achievable heat flow rate. Therefore, the present work investigates the influence of these improvement methods on the heat flux.

Various possibilities for attaching fins were investigated like longitudinal [13, 17] or cross fins [14, 18]. Different designs, such as plates or wedge-shaped fins, were also compared [19]. Otherwise, the optimisation of fin systems with respect to the number, spacing and dimensioning of the fins is examined [15, 18]. Another approach is to optimise the natural convection in the PCM, through suitably irregularly distributed fin geometries [20].
An alternative way to improve the heat transfer properties of the PCM is to disperse substances with better thermal conductivity into the PCM. Nanoparticles are usually used for this. The storage capacity is reduced less compared to fins when dispersed substances are used, but the increase in thermal conductivity is lower [21]. Different types of nanoparticles are considered in the literature such as aluminium oxide [21–24], copper and copper oxide [25, 26], titanium dioxide [27], iron oxide [28] and nano graphite [23, 29]. Much of the literature evaluates the properties in relation to the proportion of nanoparticles [21, 22, 25, 26]. Other influences, such as the effect of a magnetic field on the PCM [28] are also considered. Furthermore, different enhancement options are combined, such as nanoparticles together with fins [21], or a porous structure [22, 25, 28].

Lamberg [30] has developed an analytical model to determine the phase boundary and the fin temperatures of a solidification process in a finned two-dimensional PCM storage and compared the results to experimental findings and numerical simulation with the heat capacity method. She found a deviation of 10% in the analytically determined melting fractions compared to the numerically simulated ones and found a better agreement for geometric dimensions in the analytical method where the heat conduction can be approximated one-dimensionally. Bauer [31] also developed an analytical method for determining the solidification time. The analytical results were compared with simulation results and for one case also with experimental work from the literature. Furthermore, he investigated the influence of various dimensionless parameters that represent storage variables such as the thickness, length, number and spacing of the fins. Kamkari and Shokouhmand [16] have experimentally investigated the melting process of a PCM in a rectangular enclosure. This storage was isothermally heated from one side and insulated on the other sides. They investigated different numbers of fins and different wall temperatures. They found that with a higher wall temperature and a lower number of fins, the improvement of heat flow through the fins decreases compared to the case with no fins. In this work, the heat flow rates were addressed, but not the exergy losses.

Gunjo et al. [32] investigated the influence of adding 5% particles of pure copper, copper oxide and aluminium oxide to a paraffin sample. They tested the influence of added particles on the charging and discharging time compared to pure paraffin. For example, copper addition decreased the charging time by a factor of 10 and led to 8 times faster discharging. With their results they calculated thermal conductivities for the mixtures that were about 1 W/(m·K) for paraffin with copper oxide, 2 W/(m·K) for aluminium oxide addition and 20 W/(m·K) for copper addition.

One aspect that is not addressed explicitly is the strong change of charging and discharging heat flux with time and how it is influenced by fins, conductivity, or the driving temperature difference. Further, the relation between the change in heat flux and discharge time and the investigated improvement method remains unclear. Especially larger temperature differences at discharging have the problem that heat flow rates can be increased, but this also leads to larger exergy destruction.

To investigate these issues, a numerical parameter study of the heat transfer from a PCM during the discharge of a rectangular storage is presented, which can be regarded as part of a larger arrangement. The geometry of fins, the thermal conductivity and the discharge temperature were varied. The material data of the used PCM are from RT44 HC, a paraffin with an average melting temperature of 317.15 K from the company Rubitherm [33]. The system is modelled in 2D using the finite volume method to solve the time-dependent heat conduction equation. Due to the small distance between the fins, convection in the PCM is neglected.
here. The focus of the investigations is the impact on the energetic performance with the heat flux and the energy and exergy stored as a function of discharge time.

2 Modelled system

The modelled system is depicted in Fig. 1. The rectangular storage shown in part a of Fig. 1 is the basis for the investigation. The two surfaces on the right and left are plates, functioning as heat exchangers with water as secondary fluid. The PCM and the vertical aluminium fins are placed at regular intervals between the plate heat exchangers. It is assumed that the temperature distribution is periodic in y-direction and symmetric with the fins being the planes of symmetry. Thus, along the line of symmetry in the y-direction, the system can be reduced in size by applying an adiabatic boundary condition. The same condition can be applied at the central symmetry line in x-direction. The system reduced in this way is shown in Fig. 1 in the enlarged view b with the dashed lines as a rectangle. This system is the basis for the simulation as shown in the detailed view in Fig. 2.

The height of the simulated system is $h_{sys} = 5$ mm, while the width is $b_{fin} = 50$ mm. The thickness of the aluminium fin is $l_{fin} = 0.5$ mm. Except for the left side of the system, the boundaries of the system are adiabatic. The PCM in the system is shown in grey. For this, the commercially available PCM RT44 HC from Rubitherm [33] was assumed as base system, where the material properties of the PCM as given by the manufacturer are shown in Table 1. The used properties for aluminium can be found in Table 2 and are based on values in the VDI heat atlas [34] for AlMg2.5. The temperature dependence of the properties of aluminium was neglected due to the relatively small temperature interval considered here.

For all simulations, an initial temperature of $T_0 = 322.15$ K was used, which corresponds to a difference of 5 K to the phase change temperature $T_{melt}$. The internal energy is discharged to a secondary fluid, which is assumed to have a constant temperature of $T_{sf}$ (like seen in Fig. 2) on the whole left side of the system.

The temperature $T_{sf}$, the thermal conductivity of the PCM and the type of included fin were varied throughout the simulations. $T_{sf}$ varies from 315.15 K to 302.15 K, which corresponds to a difference of 2 to 15 K to the phase change temperature $T_{melt}$. To investigate the influence of thermal conductivity of the PCM, the values were varied from 0.3 to 1 and 1.5 W/(m·K). The used fin geometries are shown in Fig. 3.

Aluminium is shown in black and the PCM in grey. The fin volume fraction varies for the different systems and increases from top to the bottom in Fig. 3. The system with crossed fins has the highest aluminium content; by using vertical fins with $b_{fin, cross} = 10$ mm that are placed on the horizontal fin. These fins are regularly distributed in the system, resulting in 5 voids with a width of $b_{PCM, cross} = 9.1$ mm filled with PCM. The aluminium on the right side of the system has only half the thickness of $b_{fin, cross}$ due to the symmetry of the entire system.

The system with a continuous fin, a temperature in the secondary fluid $T_{sf} = 312.5$ K and a thermal conductivity of 0.3 W/(m·K) is taken as the reference for the present investigations.

3 Model

The two-dimensional time dependent energy balance Eq. (1) is solved with a finite volume solver, neglecting convection. The rectangular mesh consists of 450 cells in the

### Table 1 Thermophysical properties of RT44 HC [33]

| Property | Value |
|----------|-------|
| $c_p$ (liquid) | 2 000 J/(K·kg) |
| $c_p$ (solid) | 2 000 J/(K·kg) |
| $\rho$ (liquid) | 800 kg/m$^3$ |
| $\rho$ (solid) | 800 kg/m$^3$ |
| $\lambda$ (liquid) | 0.3 W/(m·K) |
| $\lambda$ (solid) | 0.3 W/(m·K) |
| $T_{melt}$ | 317.15 K |
| $h_{melt}$ | 250 000 J/kg |

### Table 2 Thermophysical properties of Aluminium [34]

| Property | Value |
|----------|-------|
| $c_p$ | 870 J/(K·kg) |
| $\rho$ | 2680 kg/m$^3$ |
| $\lambda$ | 140 W/(m·K) |
y-direction and 250 in the x-direction. Regarding to the dimensions of the system this leads to cell edge length of the cells of $d_{cell,y} = 0.0111$ mm in the y-direction and $d_{cell,x} = 0.2$ mm in the x-direction. The heat conduction Eq. (1) is solved using the python package FiPy [35] and the time dependent temperature distribution is calculated.

$$\rho \cdot c(T) \cdot \frac{\partial T}{\partial t} = \nabla \left[ \lambda(T) \cdot \left( i \cdot \frac{\partial T}{\partial x} + j \cdot \frac{\partial T}{\partial y} \right) \right]$$  \hspace{1cm} (1)

All further quantities of interest, such as the heat flux and the energy can be derived from the cell temperatures and their gradients. For simulating the phase change, the heat capacity method is used. With this method the phase change occurs over a temperature interval and thus a heat capacity can be defined along the phase change. With the normal distribution as a basis, Eq. (2) was used for the heat capacity $c$.

$$c(T) = \frac{1}{\sqrt{2 \cdot \pi \cdot \sigma^2}} \cdot \exp\left\{ \frac{-(T - T_{melt})^2}{2 \cdot \sigma^2} \right\} \cdot h_{PCM, melt} + c_{p, liquid}$$  \hspace{1cm} (2)

The temperature interval for the phase change and thus the value of the heat capacity is determined via the variance $\sigma^2$. This value of $\sigma$ was set to 0.1 K in the present work and leads to a good reproduction of the analytical solidification solution of the one dimensional system without fins, the Stefan-problem [36].

The energy per cell is calculated with Eq. (3) using the properties of the cell material. The sum of the energies of all cells is the current given total internal energy in the system at a certain time.

$$E_{cell} \left\{ \begin{array}{ll}
d_{cell,x} \cdot d_{cell,y} \cdot \rho \cdot \int_{T_{sf}}^{T_{cell}} c_{PCM} \, dT \\
d_{cell,x} \cdot d_{cell,y} \cdot \rho \cdot c \cdot (T_{cell} - T_{sf})
\end{array} \right\} \hspace{1cm} \text{energy PCM} \hspace{1cm} \text{energy aluminium}$$  \hspace{1cm} (3)

At the start of the simulation, all cells have the initial temperature 322.15 K. For the left side of the system, a constant temperature $T_{sf}$ is used as a boundary condition. Starting from this initial condition, the simulation is carried out until each cell has reached the condition $T_{cell} \leq T_{sf} + 0.5$ K. In this way, the time required for the simulation is kept within reasonable limits.

4 Discussion and results

The reference system with a continuous fin, thermal conductivity of 0.3 W/(m·K) and $T_{sf} = 312.15$ K is analysed, the other systems are then compared to this reference. Figure 4 shows the heat flux and the internal energy of the system as a function of the discharging time.

Figure 4 shows the heat flux and the energy over time in minutes. The time to discharge all cells to 0.5 K above the secondary fluid temperature is 75 min. The heat flux drops steeply from more than 70 kWm$^{-2}$ to 5 kWm$^{-2}$ within the first 3 min. The heat fluxes, as typical for two-dimensional calculations, are for a length of one meter in the orthogonal z-direction. After the first 3 min, the drop in heat flux flattens out and there is a continuous reduction over time. Overall, the heat flux is low with values below 5 kWm$^{-2}$ after 3 min. The energy curve is slightly concave, but unlike the heat flux, it does not show an initial steep drop in stored energy.

The reason for the strong decrease of the heat flux at the beginning of the process can be found in the rapid formation of a solidified layer. The formation of this layer can be traced in Fig. 5.

There, the temperature distributions in the system are shown for 0.2 min, 3.25 min and approximately 50 min
graphically. After 0.2 min a solidified layer is seen in dark blue at the left and lower left system side. The distance to the heat transferring left side is short for those cells so that the heat flux is high, as seen in Fig. 4 and the PCM solidifies quickly there. The layer starts forming from the left side and at the bottom left corner and lowers the heat flux due to the lower temperature difference to the secondary fluid. It can also be seen that the temperature in the area near the fin drops more quickly along the length of the storage, due to the significantly higher thermal conductivity of the metal which leads to higher temperature gradients in y-direction. It is noticeable that the temperature decreases faster along the fin than in the PCM in x-direction. However, the fin doesn’t reach $T_{sf}$ quickly over the complete length as it could be expected with its higher thermal conductivity. Instead, a curved shape of the phase transition area is observed over the complete discharging time with significant lower fin temperatures for only some mm in x-direction. Furthermore, the amount of the sensible thermal energy in the system after reaching 5 kWm$^{-2}$ is small, because nearly each cell is at least at a temperature of 317 K, which corresponds to the phase transition temperature. This part of the initial energy of the system can be discharged way quicker than the phase change energy. So, the faster retrievable part of the sensible heat and the formation of the solidified layer as an insulator are responsible for steep drop of heat flux at the beginning of the discharge.

To investigate the influence of different fin geometries on the discharge process, the storage with a continuous fin is compared with systems that have no fin, half a fin or additionally vertical fins, as shown in Fig. 3. The heat fluxes over time for the different fin types at reference conditions (0.3 W/(m·K) and $T_{sf} = 312.15$ K) are shown in Fig. 6.

In Fig. 6, the solid line shows the heat flux evolution for the reference case with a continuous fin, the dotted line for a system without fin, the dashed line for a system with a half fin and the dashed-dotted line for the crossed fin system. The initial drop of the heat flux described before also occurs with all fin geometries.

However, a difference in the heat flux evolution is recognized after the steep drop. With increasing fin volume,
the heat flux after the initial drop increases. As an example, in the storage without fin after 3.5 min a heat flux of approximately 778 W/m² is found, whereas for the cross-fin system values of 3.7 to 3 kW/m² between 5 and 7.5 min is reached. This can be related to the insulating effect of the already solidified PCM. A system with fins provides a larger area over which energy can be transferred vertically from the PCM, which leads to an increase in heat flux even if an insulating layer is formed. The difference in the progress of the phase change can be seen in Fig. 7. There, the temperatures in a system without a fin and one with a continuous fin are plotted at reference conditions after 50 min of discharging.

It is clearly seen that in the system without fin there is only a temperature gradient in x-direction and the phase change does not progress as fast as in the case with a continuous fin. With an increasing proportion of fins, the advantage in discharge time is smaller. This can be better observed in Fig. 8, where the energy for the 4 different systems is plotted against time.

As seen in Fig. 8, a storage system with a half fin requires only 29.5% of the discharge time compared to a system without a fin, whereas a system with cross fins requires 75.7% of the discharge time of a system with a continuous fin. Thus, the effectiveness of the additional fin material decreases. The possible amount of stored energy and the volume fractions of PCM and aluminium are shown in Table 3. The stored energy is calculated with Eq. (3); the volume fractions depend on the geometry of the fins and decreases with the PCM volume fraction. Because of the low impact of the sensible energy of aluminium on the stored energy, the energy drops nearly as much as the volume fraction of the PCM in the system. A comparison of the stored energy for cross-fin systems to the storages without fins demonstrates the coincidence, at 81.9% PCM volume fraction the energy storage capacity drops to 83.9%.

The decrease in stored energy is lower than the decrease of the discharge time with increasing volume fraction of aluminium. The discharge time is 24.3% lower for crossed fins compared to continuous fins, but the stored energy is only 7.9% lower. In the considered cases the advantage of a larger fin volume is given. But it can be assumed that
the effectiveness of additional fin volume will continue to decrease. So, there will be a point, where the reduction of discharge time with increasing fin volume will be outweighed by the continuous loss of stored energy due to the reduction in PCM volume fraction.

Furthermore, the influence of the thermal conductivity of the PCM on the heat flux is investigated. For this purpose, Fig. 9 compares a system with a continuous fin and a system without a fin, whereby the heat flux over time is examined for thermal conductivities of 0.3 W/(m·K), 1 W/(m·K) and 1.5 W/(m·K).

The upper group of curves in Fig. 9 belongs to the system with a continuous fin, while the lower group belongs to the system without a fin. In the system without a fin, a proportional increase of the heat flux with increasing thermal conductivity is observed, while for a system with a continuous fin, the heat flux is increased 10 to 20% until a discharge time of 15 min. From this point on, the heat fluxes are the same throughout the systems until 48 min. After this, the time for full discharge is lower for higher thermal conductivity. This discharge behaviour suggests that the increase in thermal conductivity becomes more ineffective with increasing fin volume fraction and that the impact of the fin on discharge outweighs the effect of increasing thermal conductivity.

For a better comparison of the influence of the heat exchange enhancement methods on the discharge process, a dimensionless heat flux number $M$ is calculated from the heat flux $\dot{q}$, the thermal conductivity $\lambda$, the temperature difference $\Delta T$ and a characteristic length $L$, as shown in Eq. (4).

$$M = \frac{\dot{q}}{\lambda \cdot \Delta T/L} \quad (4)$$

Here the length in the y-direction of the system was used as the characteristic length since the phase change is last completed in this direction.

In order to quantify the influence of the increase in thermal conductivity more precisely for the two systems discussed above, Fig. 10 shows the dimensionless heat flux number $M$ as a function of the Fourier number for the same thermal conductivities as in Fig. 9. The characteristic length needed to calculate the Fourier number is also the length in y-direction.

The curves for the system without a fin are all shown with continuous lines, as these almost overlap indicating that the increase in heat flux is proportional to the increase of the values from the boundary conditions in the denominator. The systems with continuous fins show different results. For this condition the dimensionless heat flux number decreases with increasing thermal conductivity and comes closer to the curve for the system without a fin. This means in contrast to the results of the system without a fin that the increase of heat flux does not increase proportional to the thermal conductivity and increasing the thermal conductivity in systems

Table 3 Volume fraction of PCM and aluminium, and total amount of stored energy at $T_{cell} = T_{0} = 322.15$ K for considered fin geometries

| Volume fraction | Volume fraction aluminium | Stored energy at $T_{0}$ [kJ] |
|-----------------|--------------------------|-------------------------------|
| Without fin     | 1                        | 0                             | 54.00                        |
| Half fin        | 0.95                     | 0.05                          | 51.59                        |
| Continuous fin  | 0.9                      | 0.1                           | 49.18                        |
| Cross fins      | 0.819                    | 0.181                         | 45.28                        |
with fins loses effectiveness and is less needed, as summarized in Table 4, where the discharge time of the system is shown depending on the fin geometry and the thermal conductivity.

The discharge time decreases with higher thermal conductivity and fin volume fractions. However, the ratios between the discharge times at 0.3 W/(m·K) and 1 W/(m·K), shown in the third row, shows that the discharge time decreases less with increasing fin volume fraction. For a system without fin, the higher thermal conductivity leads to a ratio of 0.30, which means that the discharge is 70% faster, while with cross fins as the system with the highest fin volume, the discharge is only 18% faster with a ratio of 0.82 between the two thermal conductivities. Accordingly, the influence of the thermal conductivity decreases significantly with increasing fin volume. Furthermore, Fig. 10 shows that the Fourier number increases for the continuous fin with increasing thermal conductivity. This shows that for a system with one specific fin case the decrease in discharge time is not proportional to the increase in thermal conductivity.

The influence of \( T_{sf} \) was also investigated, because the heat fluxes can be increased with larger driving temperature differences. The temperature differences to the phase change temperature of 5 K, 10 K and 15 K were investigated. In Fig. 11 the time dependence of the heat flux is presented for a system with a continuous fin and a thermal conductivity of 0.3 W/(m·K).

An increase in the heat flux and a reduction in the discharge time of approximately 46% can be observed comparing the temperature differences (\( \Delta T \)) of 5 K with the one of 10 K. Thereby, the heat flux at \( \Delta T = 10 \) K falls below the level of \( \Delta T = 5 \) K at about 27 min. The heat flux at \( \Delta T = 15 \) K is for approximately 14 min higher than the heat flux at 10 K. Overall the change in heat flux is higher when changing from 5 to 10 K than from 10 to 15 K, as also seen for the discharge times. There the discharge time is lower for higher temperature differences, which leads to the intersection of the curves.

Figure 12 shows the dimensionless heat flux \( M \) plotted as the function of the Fourier number for the three temperature differences for a continuous fin and a thermal conductivity of 0.3 W/(m·K).

The level of the dimensionless heat flux decreases with an increasing temperature difference of \( T_{sf} \) and the phase change temperature of the PCM. As also seen in Fig. 11, there is a larger drop of the heat flux number from \( \Delta T = 5 \) to \( \Delta T = 10 \) K than from 10 to 15 K. Similar to the variation of thermal conductivity, the heat flux is not increased proportionally to the temperature difference to the secondary fluid.

Furthermore, the comparison of the influences of higher thermal conductivity and lower discharge temperatures is of interest. A comparison is seen in Fig. 13 for the possible combinations of 0.3 W/(m·K) and 1 W/(m·K) at 5 K and 10 K for a system with a continuous fin.

In comparison the system with \( \Delta T = 5 \) K and 0.3 W/(m·K) has the highest dimensionless heat flux for \( Fo > 30 \), but the

| Table 4 | Discharge times in min for two thermal conductivities and different fin geometries for \( T_{sf} = 312.15 \) K |
|---------|-------------------|-------------------|-------------------|-------------------|
| \( \lambda \) [W/(m·K)] | Without fin | Half fin | Continuous fin | Cross fins |
| 0.3    | 2895           | 865           | 75              | 57              |
| 1.0    | 855            | 270           | 56              | 47              |
| Ratio  | 0.30           | 0.31          | 0.75            | 0.82            |

Fig. 11 Heat flux as a function of time for temperature differences of 5 K, 10 K and 15 K in a system with continuous fin and 0.3 W/(m·K)
second lowest Fourier number at full discharge. The system with a conductivity of 0.3 W/(m·K) and a difference of 10 K has the second highest dimensionless heat flux level as well as the lowest Fourier number at full discharge. The combination of high driving temperature differences and high conductivities leads to quite low dimensionless heat fluxes and is worse than just applying only one method to increase the heat flux.

In Fig. 10 it was observed that the decrease of discharge time is not proportional to the increase in thermal conductivity. In addition to that, Fig. 13 shows that the increase in

\[ Q_{\text{trans}} = \frac{T_{\text{ambient}}}{T_{\text{sf}}} \cdot Q_{\text{trans}} \]  

\( Q_{\text{trans}} \) is the heat transferred over the whole discharge time. \( T_{\text{ambient}} \) was taken as 298.15 K. \( T_{\text{sf}} \) is constant due to the boundary condition. Thus, the factor before \( Q_{\text{trans}} \) is constant for a specific temperature difference. In the following it is called \( f \). \( E_{\text{trans}} \) can be compared with the exergy \( E_{\text{total}} \) in the storage at the initial temperature. \( E_{\text{total}} \) is the same for each case because the start temperature was not varied. The exergetic efficiency is defined as \( \eta_{\text{Ex}} = E_{\text{trans}} / E_{\text{total}} \). The values for the different considered \( \Delta T \) can be found in Table 5.

The transferred heat slightly increases from 52.60 kJ at 2 K to 57.29 kJ at 15 K due to the higher temperature difference and thus a higher sensible heat fraction. In contrast, the transferred exergy decreases from 2.84 kJ to 0.76 kJ. The exergetic efficiency drops from 69.27% to 18.45%, as the final temperature approaches \( T_{\text{ambient}} \). Thus, the exergetic efficiency is already relatively low at a difference of 2 K and drops strongly with increasing \( \Delta T \). In contrast to this decreasing \( \eta_{\text{Ex}} \), is the decreasing discharge time and with it the increasing heat fluxes. A calculated exergetic efficiency of 0.6927 at 2 K or 0.5872 at 5 K are relatively high for a thermal energy storage like this, but it should be considered that these efficiencies are achieved with the condition of an isothermal secondary fluid. Overall, there is a trade-off

| \( \Delta T [\text{K}] \) | \( f [-] \) | \( Q_{\text{trans}} [\text{kJm}^{-2}] \) | \( E_{\text{trans}} [\text{kJm}^{-2}] \) | \( E_{\text{total}} [\text{kJm}^{-2}] \) | \( \eta_{\text{Ex}} [-] \) |
|----------------|-------------|-----------------|-----------------|-----------------|-----------|
| 2              | 0.0539      | 52.60           | 2.84            | 4.1             | 0.6927    |
| 5              | 0.0446      | 53.68           | 2.41            | 4.1             | 0.5872    |
| 10             | 0.0293      | 55.40           | 1.62            | 4.1             | 0.3959    |
| 15             | 0.0132      | 57.29           | 0.76            | 4.1             | 0.1845    |
between the efficiency and the discharge time because a high exergy efficiency with very long discharge times is just as unsuitable for applications as a fast discharging, with high exergy losses.

For a comparison, $E_{\text{trans}}$ and the discharge time $t$ are listed in Table 6 for temperature differences of 2 to 15 K for a system with a continuous fin and 0.3 W/(m·K). Furthermore, the transferred exergy and the discharge time are calculated relative to the values for $\Delta T = 2$ K. The relative exergy and relative discharge time are shown in Fig. 14.

In Fig. 14 it is seen that the relative exergy decreases nearly linearly with increasing $\Delta T$, while the relative discharge time decreases fast at first, but with higher $\Delta T$ the slope decrease. The comparison of these two curves shows the trade-off in exergy efficiency and lower discharge times with higher temperature differences. First the relative discharge time decreases faster than the exergy loss until a $\Delta T$ of around 6 K, which means that it is beneficial to use higher $\Delta T$ until this driving temperature difference, if discharge time is the crucial parameter. But with higher temperature differences than 6 K, the transferred exergy is 20% lower than for 2 K and decreases for higher $\Delta T$, which can outweigh the reduction in discharge time for some applications. Considering this, it can be concluded that increasing the temperature difference is only reasonable up to a certain point and the trade-off between exergy loss and lower discharge times should be weighed carefully for every system and application.

### 5 Conclusion

Methods to improve heat transfer in latent thermal storage systems were investigated by modelling. The considered cases were the introduction of fins into the system, increasing the thermal conductivity of the PCM and a greater temperature difference between the secondary fluid and the phase change temperature of the material. A section of a larger rectangular storage was regarded, in which convection within the material was neglected and where the temperature of the secondary fluid was considered constant.

The study investigated the influence of the fin geometry, the thermal conductivity and the driving temperature difference on the heat flux and discharge time. For this purpose, the absolute data as well as a dimensionless heat flux was considered. Furthermore, the proportionality of the heat flux and discharge time to the growing heat transfer improvement and the trade-off between exergy loss and higher driving temperature differences were investigated.

It was shown that the heat fluxes drop steeply in the first few minutes due to the solidified PCM layer as an insulator. After that, discharging takes place with moderate heat fluxes. With increasing fin volume, the general level of heat flux increased after the initial drop, but the advantage decreased with increasing fin volume. For the thermal conductivity a value of 0.3 W/(m·K), typical for a paraffin as PCM, was compared with better values of 1 and 1.5 W/(m·K). It was observed that the influence decreases with larger fin volumes and the discharge time does not decrease proportional with higher thermal conductivity. This was reflected in the decreasing dimensionless heat flux number, less decreasing discharge times and higher Fourier numbers at full discharge. The increase in the temperature difference brought the greatest advantage for the heat flux of the considered improvement methods, but there was also a decreasing dimensionless heat flux number and thus a decreasing benefit at higher driving temperatures.

| $\Delta T$ [K] | $E_{\text{trans}}$ [kJm$^{-2}$] | $t$ [min] | $E_{\text{trans}}/E_{\text{trans},2K}$ [-] | $t/t_{2K}$ [-] |
|---------------|-------------------------------|----------|---------------------------------|-----------------|
| 2             | 2.84                          | 179      | 1                               | 1               |
| 3             | 2.70                          | 122      | 0.95                             | 0.68            |
| 4             | 2.55                          | 93       | 0.904                            | 0.52            |
| 5             | 2.41                          | 75       | 0.845                            | 0.42            |
| 6             | 2.26                          | 63       | 0.80                             | 0.35            |
| 7             | 2.10                          | 55       | 0.74                             | 0.31            |
| 8             | 1.95                          | 49       | 0.69                             | 0.27            |
| 9             | 1.78                          | 44       | 0.63                             | 0.24            |
| 10            | 1.62                          | 40       | 0.57                             | 0.22            |
| 11            | 1.45                          | 36       | 0.51                             | 0.20            |
| 12            | 1.29                          | 34       | 0.45                             | 0.19            |
| 13            | 1.11                          | 31       | 0.39                             | 0.18            |
| 14            | 0.94                          | 30       | 0.33                             | 0.17            |
| 15            | 0.76                          | 28       | 0.27                             | 0.16            |

**Fig. 14** Relative exergy and relative discharge time from Table 6 as a function of $\Delta T$ for a system with continuous fin and 0.3 W/(m·K)
temperature differences. The combination of lower discharge temperatures and increasing thermal conductivity turned out to be the worst variant in the comparison. This was shown by lower dimensionless heat flux numbers for this case than for using only one enhancement method. Within the scope of the investigations for the discharge temperature, the exergetic side was also evaluated. It was shown that higher driving temperature differences were advantageous at first, but the benefit of lower discharge time decreased until an $\Delta T$ of around 6 K after which the decreasing slope of the relative discharge time was lower than for the exergy loss. Thus, the trade-off between the extractable exergy and the reduction of discharge time for increasing $\Delta T$ is important and should be considered in addition to the relation between the change in heat flux and discharge time for different latent thermal energy storage improvement methods.

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**Declarations**

**Conflict of interest** On behalf of all authors, the corresponding author states that there is no conflict of interest.

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