

ON THE DESIGN AND ADVANTAGES OF A VAPOR CHAMBER FOR BATTERY PACKS AND THE DISCREPANCY IN VERTICAL CHANNEL BOILING CORRELATIONS AT SMALL GAP SPACING

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ABSTRACT

The recent trend of increasing demand for rechargeable energy storage systems leads to the possibility of high power and energy electrical batteries. Especially for mobile applications, minimizing size and weight of the batteries is important, hereby increasing the power and energy density. This leads to higher heat losses, created by the internal resistance of the battery. During discharge, the chemical reactions in most batteries are exothermic which causes an even greater increase in heat generation. To avoid excessive temperatures, which can lead to lower efficiency, reduced lifetime and battery failure, adequate thermal management is needed. This paper introduces battery thermal management for prismatic cells through a vapor chamber. The battery cells are immersed in a non-conducting liquid, contained in a closed vessel. By absorbing heat from the battery, the liquid evaporates at the surface of the battery. The evaporated gas is condensed at the top of the vessel, by releasing heat to the environment. A lumped parameter model is made to simulate the response of a vapor chamber system on several load cycles and to compare with air cooling. Due to the evaporating heat transfer at the battery surface, the local heat transfer coefficient can be increased up to a factor 10 compared to air cooling. Load cycles of batteries often consist of peak currents being drawn for short times. This can cause a temporary increase in battery temperature. To avoid these peak temperatures, thermal buffering has been studied, for example by using the latent heat of melting PCMs. However this also results in an increase of the thermal resistance of the battery system. In a vapor chamber, the latent heat of evaporating refrigerant is used as a thermal buffer, without having a significant increase in thermal resistance. Analysis shows that peak temperature differences between the battery and the environment can be reduced by a factor up to 2 compared to air cooled batteries. The most important factor in modelling and designing the vapor chamber is the heat transfer coefficient of nucleate boiling in vertical rectangular channels and the influence of the distance between the cells on the heat transfer coefficient. Measurements and experimental correlations from literature do not correspond for small channels, where some perceive or predict increased heat transfer while others do not. An experimental setup is designed and being built to experimentally test the heat transfer coefficient for narrow vertical rectangular channels.

NOMENCLATURE

A	[m ²]	Area
Bo	[-]	Bond number
c	[J/kgK]	Specific heat capacity
E	[V]	Open-circuit voltage
g	[m ² /s]	Gravitational acceleration
h	[W/m ² K]	Heat transfer coefficient
H	[J/kg]	Heat of vaporization
I	[A]	Current
k	[W/mK]	Thermal conductivity
L	[m]	Channel height
m	[kg]	Mass
n	[-]	Number of battery cells in pack
Nu	[-]	Nusselt number
Q	[W]	Heat
q	[W/m ²]	Heat flux
R	[Ω]	Internal electrical resistance
Ra	[-]	Rayleigh number
Re	[-]	Reynolds number
T	[K]	Temperature
T	[s]	Time
V	[m ³]	Volume
Special characters		
γ	[-]	Boiling suppression factor
δ	[m]	Channel spacing
ρ	[kg/m ³]	Density
σ	[N/m]	Surface tension
ϕ	[-]	Convective enhancement factor
Subscripts		
a		Ambient
b		Battery
c		Convective
irr		Irreversible
l		Liquid
lat		Latent
nb		Nucleate boiling
r		Refrigerant
rev		Reversible
tot		Total
TP		Two-phase
v		Vapor
vc		Vapor chamber

INTRODUCTION

Electrical energy storage is becoming increasingly important in the global energy context. Next to the buffering of the variable electricity produced by renewable energy sources such as wind

and solar, electrical vehicles are increasingly used as an alternative to traditional combustion engine driven vehicles. For these mobile applications, it is important to have as much energy and power from as little weight and volume as possible, to be able to compete with the traditional vehicles. This demand leads to higher power and energy densities for batteries. The higher power density inevitably leads to higher heat losses in the batteries. Due to these higher heat losses, the temperature of the battery packs increases. Higher temperatures can lead to reduced efficiency, increased ageing and even battery failure through thermal runaway [1]. To maintain the temperature of the batteries within an adequate range, a thermal management system is needed [2]. This paper proposes and investigates the possibility of a vapor chamber as a thermal management system for battery packs.

BATTERY PACK VAPOR CHAMBER

Battery packs typically consists of dozens of batteries, connected in series and/or parallel to comply to the demands of the application. The most common types of cells are either cylindrical or prismatic. Since prismatic cells can be combined to battery packs in a more compact manner, these types of cells are considered in this paper.

For high current applications, the heat generation of the battery can cause the temperature of the battery to increase to higher temperatures than allowed for safe battery utilization if the thermal management system is not adequately designed. To cool the batteries, forced convective air cooling and water cooling have been applied [2]. For the reduction of peak temperature and the improvement of the battery pack safety, thermal management systems using thermal buffering with phase change materials (PCM's) have been designed [3]. However, this technique increases the total thermal resistance between the battery and the ambient. Another technique that has been proposed recently is two-phase immersion cooling of batteries [4]. A two-phase immersion cooling system is also known as a vapor chamber. The batteries are immersed in a dielectric liquid, which boils due to the heat dissipation of the batteries. The batteries and fluid are contained within an enclosure, and vapor bubbles rise to the top of the enclosure. Here, the vapor is condensed again and droplets fall back to the liquid surface. An example of a battery pack vapor chamber is shown in Figure 1. The cooling of the top of the vapor chamber can be by different means, depending on the application: natural or forced air cooling with or without fins or liquid cooling.

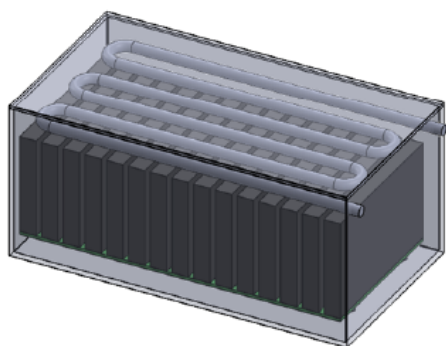


Figure 1 Example of a battery pack vapor chamber

A vapor chamber for battery cooling has several advantages over the more common air or liquid cooling techniques:

- The local heat transfer coefficient at the battery surface is a lot higher for pool boiling in the vapor chamber than for natural or forced convection with water or air.
- By immersing the batteries, the entire outer surface of the battery can be cooled. For forced convective air cooling, only the surfaces where the air flow passes are cooled effectively. For liquid cooling, the battery surface that is cooled is limited to area where the coolant channels are provided. Cooling the entire surface of the battery reduces the risk of excessive temperature on hot spots of the battery, which can reduce the battery lifetime.
- For highly dynamic load current profiles, peak loads can cause peak temperatures which jeopardize the safety of the battery operation by inducing thermal runaway [1]. Due to the evaporation of the dielectric fluid in a vapor chamber system, large amounts of latent heat can be stored when the total amount of vapor in the chamber increases. This effect can be used to buffer the peak load heat dissipation, thereby reducing the peak temperatures of the battery.

Another advantage of the vapor chamber thermal management technique is that it has the potential of taking up less volume than a battery pack with a traditional thermal management system. Instead of using several fans or pumps to transport the liquid across all the batteries, only the top plate of the battery pack vapor chamber has to be cooled, with a single fan or pump (or with natural convection). In addition to this, the batteries in the pack can be positioned closer together, since no air or structure containing the coolant has to be in between. The minimal spacing between the batteries in a vapor chamber is limited by the critical heat flux. This is the heat flux at which a vapor film forms on the battery surface, hereby dramatically increasing the battery temperature. With lower spacing, the critical heat flux reduces and thereby the risk of damaging the battery due to excessive temperatures increases. Adequate design of the battery pack and battery spacing is therefore essential.

BATTERY THERMAL MODELLING

The battery chosen in the modelling is the polymer lithium-ion battery TENERGY, model 30123. It is a pouch-type cell, nearly prismatic in shape. The main battery characteristics are given by Table I. The battery thermal characteristics were investigated in detail by Nieto et al. [5].

Table I Battery characteristics (TENERGY 30123)

Property	Value	Unit
Height	0,0595	m
Width	0,1570	m
Thickness	0,0098	m
Weight	0,210	kg
Nominal electric capacity	10	Ah
Max peak discharge current	100	A
Max continuous discharge current	50	A
Max continuous charge current	10	A
Heat capacity [5]	1015	J/kgK

For the battery thermal modelling, the two basic heat generation effects are taken into account: the irreversible heat and the reversible heat (equation (1)). The irreversible heat is modelled as Joule losses by an internal electrical resistance (equation (2)). The reversible heat is generated due to the entropy change related to the chemical reactions in the battery during charging and discharging (equation (3)). The irreversible heat is always positive, which means that heat is generated by the battery. The reversible heat can be positive or negative, which means heat can be generated or taken up by the battery.

$$Q_b = Q_{irr} + Q_{rev} \quad (1)$$

$$Q_{irr} = RI^2 \quad (2)$$

$$Q_{rev} = -IT \frac{\partial E}{\partial T} \quad (3)$$

In these equations, Q_b is the heat generated by the battery, Q_{irr} is the irreversible heat, Q_{rev} is the reversible heat, R is the internal electrical resistance, I is the current (chosen positive during charging), T is the temperature in Kelvin, and E is the open-circuit voltage. The internal electrical resistance is gathered from [5], dependent on the battery temperature. The derivative of the open-circuit voltage with respect to temperature is also called the entropic coefficient. This coefficient is also gathered from [5] and is dependent on the State of Charge (SoC) of the battery.

To analyze the thermal behavior of the battery, a load cycle is needed for the current in equations (2) and (3). The cycle is a full charge and discharge with the maximal continuous charge and discharge current, based on the battery datasheet. This cycle is shown in Figure 2. The State of Charge (SoC) shown in the figure is a measure for the charge of the battery (100% is fully charged, 0% is fully discharged).

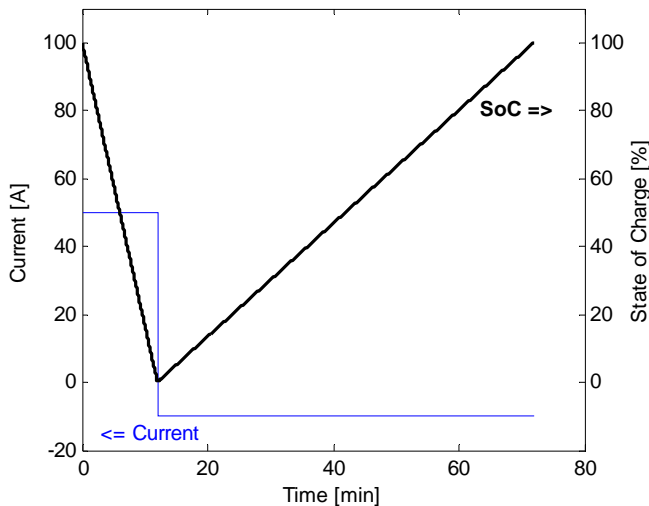


Figure 2 Charge-discharge load cycle

The entire battery is assumed to be at a constant temperature in this model. This is not completely consistent with reports from literature [6], however this paper does not intend to analyze in detail the temperature distribution of the battery. The aim to have a global analysis of the effect of the different thermal management systems on the overall battery temperature. The

heat generated by the battery from equation (1) is either stored as sensible heat in the battery or removed to the vapor chamber, as indicated by equation (4).

$$Q_b = m_b c_b \frac{dT_b}{dt} + A_b q_{vc} \quad (4)$$

In this equation, m_b is the mass of the battery, c_b is the specific heat capacity of the battery, t is time, A_b is the total surface area of the battery and q_{vc} is the heat flux from the battery. The total battery area is simplified to two times the product of the height and the width of the battery, since the battery thickness is small compared to the other dimensions.

VAPOR CHAMBER MODELLING

The vapor chamber is modelled as a constant mass and volume of refrigerant. Heat is transferred between the batteries and the refrigerant and between the refrigerant and an external cooling circuit by condensation of the refrigerant at the top of the vapor chamber. Latent heat is stored in the vapor chamber by the increase of vapor inside the chamber, sensible heat can be stored by a temperature increase in the vapor chamber. The heat balance of the vapor chamber is given by equation (5).

$$nA_b q_{vc} = h_a A_a (T_r - T_a) + (m_v c_v + m_l c_l) \frac{dT_r}{dt} + H_{lat} \frac{dm_v}{dt} \quad (5)$$

In this equation, n is the number of batteries, h_a is the overall heat transfer coefficient to the ambient, A_a is the heat transfer area between refrigerant and ambient, T_r is the refrigerant temperature, T_a is the ambient temperature, m_v is the mass of gaseous refrigerant, c_v is the specific heat capacity of the vapor, m_l is the mass of liquid refrigerant, c_l is the specific heat capacity of the liquid and H_{lat} is the latent heat of vaporization of the refrigerant.

The heat transfer from the refrigerant to the cooling circuit is through condensation of the refrigerant, conduction through the coolant tubing and convection to the liquid coolant. Convection to the coolant is assumed to be the largest thermal resistance and is estimated using a convection coefficient of $1000 \text{ W/m}^2\text{K}$. The heat transfer area is calculated by multiplying the width of the battery with the length of the battery pack. The length is equal to the sum of the battery thickness and spacing, multiplied by the number of batteries in the pack.

Since the mass and the volume of the refrigerant in the chamber is constant, the mass of vapor is directly related to the temperature of the refrigerant (assuming that the temperature of the liquid and gaseous refrigerant are uniform and equal), and is given by equation (6).

$$m_v = \frac{\rho_l V_{tot} - m_{tot}}{\frac{\rho_l}{\rho_v} - 1} \quad (6)$$

In this equation, ρ_l is the liquid density, V_{tot} and m_{tot} are respectively the total volume and the total mass of the refrigerant in the vapor chamber, and ρ_v is the vapor density. The densities are only dependent on the refrigerant temperature when both liquid and vapor phase are present in the chamber.

The heat transfer mechanism between the battery and the refrigerant is dependent on the temperature difference between the battery and the refrigerant. If the battery temperature is lower

than the refrigerant temperature, heat is transferred through natural convection, modelled by equation (7) [7], with L the battery height, k_l the liquid thermal conductivity and Ra the Rayleigh number. If the battery temperature is higher than the refrigerant temperature, heat is transferred through vertical channel pool boiling (equation (8)), with h_{TP} the boiling heat transfer coefficient. At low heat fluxes, boiling will not occur but heat will be transferred through natural convection. Since the focus in this paper is on high heat fluxes, only two-phase heat transfer is implemented in this model. Several methods to model heat transfer through vertical channel pool boiling are described in the next section.

$$\text{if } T_b < T_r: \quad \frac{q_{vc}}{T_b - T_r} \frac{L}{k_l} = 0.59Ra^{\frac{1}{4}} \quad (7)$$

$$\text{else:} \quad q_{vc} = h_{TP}(T_b - T_r) \quad (8)$$

VERTICAL CHANNEL POOL BOILING

When the batteries dissipate enough heat to the saturated liquid in the vapor chamber, the liquid at the surface will start to evaporate. Due to the density difference between the vapor and liquid refrigerant, the bubbles will rise to the surface. The heat flux that is transferred from the battery surface to the refrigerant by this process is related to the difference between the surface temperature and the refrigerant temperature, also called the excess temperature. When the batteries are located close together, bubbles from the two surfaces will interact and the heat transfer will be affected. In literature, three authors proposed different correlations for vertical channel boiling with narrow gap spacing: Bar-Cohen and Schweitzer [8], Fujita et al. [9] and Chan et al. [10].

Bar-Cohen and Schweitzer [8] proposed a correlation for the heat transfer coefficient based on the superposition of two contributions: a convective contribution h_c and a boiling contribution h_{nb} , as in equation (9). The convective contribution is based on the Dittus-Boelter correlation [11], while the boiling contribution is based on the Rohsenow correlation [12]. An enhancement factor ϕ for the convective contribution is implemented to take into account the effect of the velocity of the vapor core. A correlation is used, based on the mass flux of refrigerant through the vertical channel. This mass flux is determined from a pressure balance over the channel. The boiling suppression factor γ is chosen as 1.

$$h_{TP} = \phi h_c + \gamma h_{nb} \quad (9)$$

Fujita et al. [9] constructed a correlation based on the work of Ishibashi and Nishikawa [13]. A modified Reynolds number Re is used, taking into account the heat flux. The Nusselt number Nu is correlated to the Reynolds number by equation (10). The gap spacing is used as the characteristic length.

$$Nu = 16[Re^{1/2}]^{2/3} \quad (10)$$

Chan et al. [10] used the Cooper correlation [14] and added a correction (equation (12)). This correction uses the Bond number Bo , defined by equation (11), which represents the ratio of the bubble departure diameter to the gap spacing. In these equations, σ_l is the liquid surface tension, ρ_l and ρ_v are the liquid and vapor density and g the gravitational acceleration.

$$Bo = \frac{\delta^2}{\left[\frac{\sigma_l}{(\rho_l - \rho_v)g} \right]} \quad (11)$$

$$h_{TP} = h_{nb}(1 + 0.3Bo^{-0.6}) \quad (12)$$

The general experimental observations for vertical channel pool boiling in narrow gap spacing are that at low heat fluxes, heat transfer is enhanced, while at high heat fluxes, heat transfer is deteriorated [15,16]. All the correlations described previously present enhanced heat transfer for narrow gap spacing, independent of heat flux. The effect of deterioration at higher heat fluxes is thus not covered by the correlations and may lead to modelling and design errors. In future work, the authors of this paper intend to investigate this effect with new experimental work.

REFRIGERANT SELECTION

To simulate the performance of the vapor chamber, Novec649 (dodecafluoro-2-methylpentan-3-one) was chosen as refrigerant. The selection was made based on the following criteria:

- It has no ozone depletion potential (ODP=0), has low global warming potential (GWP=1) and is not flammable.
- The boiling point is low (49°C at atmospheric pressure), which means the pressure in the vapor chamber does not have to be too low.
- It is a replacement for FC-72, which has been used frequently for electronics cooling.

ANALYSIS OF BATTERY PACK VAPOR CHAMBER

The simulation results presented are for a battery pack with twenty cells and an ambient temperature of 20°C. As a reference to compare the battery back vapor chamber thermal management system, a battery pack with forced convective heat transfer is simulated. The spacing between the battery cells is chosen to be 5mm and the air velocity in between the cells is 5m/s. The convective heat transfer coefficient is calculated from the Gnielinski correlation [17]. The simulated battery temperature during one load cycle is shown in Figure 3 for an ambient temperature of 20°C. The maximal battery temperature is reached at the end of the discharge phase and is equal to 50 °C.

For the simulation of the battery pack vapor chamber, the battery spacing is an important parameter. Using the correlation for the critical heat flux by Bonjour and Lallemand [18] and the maximal heat flux from equation (1), the minimal battery spacing is calculated for which the critical heat flux is higher than the maximal battery heat flux. From the calculations a battery spacing of 0.1mm is attained. In the following simulations, a battery spacing of 0.5mm is used. This allows for a safety margin on the critical heat flux and is more feasible to be implemented in practice. The simulation of the load cycle is performed for the three different correlations and is shown in Figure 4. The results of the model using the three different correlations match within less than 1°C. The maximal battery temperature is reduced to 36°C, which is 14°C lower than with forced air cooling, or a reduction of half the temperature

difference between the battery and the ambient. Next to the lower battery temperature, the spacing in between the batteries can be reduced by a factor 10, thereby reducing the width of the battery pack by 30%.

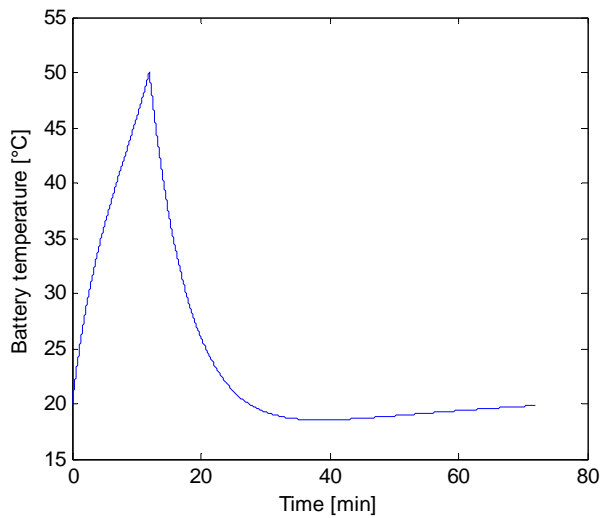


Figure 3 Battery temperature with forced air cooling

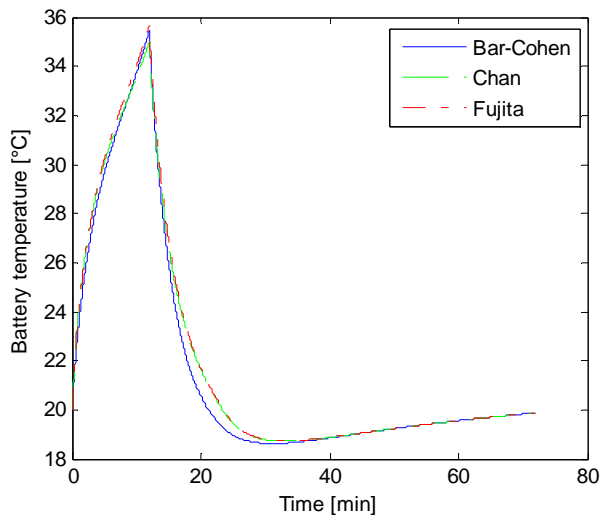


Figure 4 Battery temperature with vapor chamber

Figure 5 shows the battery and refrigerant temperature for the simulation using the correlation by Chan. The graph shows that the temperature difference between the battery and the refrigerant is always smaller than 6°C. The maximal heat transfer coefficient between the battery and the refrigerant is 260 W/m²K, which is about 10 times higher than the heat transfer coefficient with forced air convection. However, the difference in temperature between the battery and the ambient is higher than 6°C, due to the temperature increase of the refrigerant. This temperature increase is due to the evaporation of the refrigerant, causing the pressure in the vapor chamber to rise and thereby increasing the saturation temperature. The effect can be reduced by designing the vapor chamber with a larger mass of refrigerant, making the temperature increase slower. This design is a tradeoff

between a lower battery temperature and a smaller battery pack, which can be optimized based on the application.

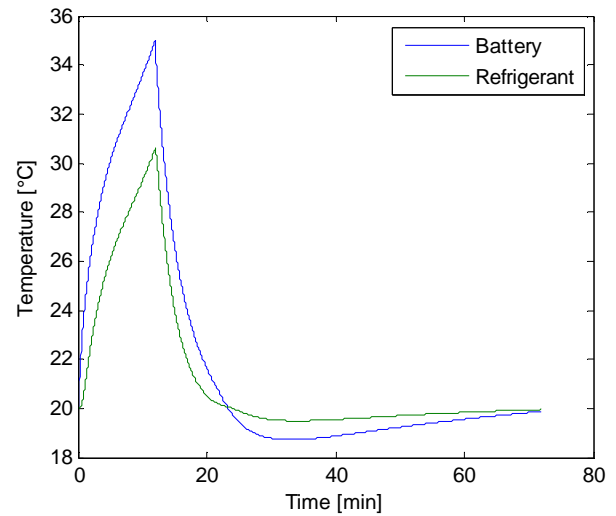


Figure 5 Battery and refrigerant temperature in the vapor chamber

CONCLUSION

A new battery thermal management system using a vapor chamber has been proposed in this paper. The system has been modelled and simulated for a simple charge-discharge cycle. For the boiling heat transfer model in the narrow vertical channel in between the battery cells, three different correlations from literature have been used. The results of the simulations using the different correlations match well with each other. When comparing the vapor chamber battery pack with a forced air cooled battery pack, the maximal battery temperature is reduced by 14°C, or a reduction of the temperature difference between the battery and the ambient with a factor 2. The local heat transfer coefficient of the boiling phenomenon can be up to 10 times higher.

Although the different correlations match very well, experiments have shown different heat transfer behavior at high heat fluxes, which is not adequately predicted by the correlations. To investigate the heat transfer behavior in narrow vertical channels and to validate the battery pack vapor chamber model, a setup is currently under construction.

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