Virtual Approach for Control System Design: Integrated Simulation of Battery Cooling and Cabin Comfort Circuits to Develop a BEV Thermal Management Control Logic

G. Boccardo[1], E. Graziano[1], L. De Rosa[1], T. Mrkvica[2], S. Pautasso[3]

1: POWERTECH Engineering S.r.l., Via Carolina Invernizio 6, 10127 Torino, IT

2: SATTELO s.r.o., Stojanova 1334 – Triangl, 686 01, Uherské Hradiště, CZ

3: DIESEL EMISSION CONTROL L.t.d., Office 9, Dunston Business Village, Penkridge, Staffordshire ST18 9AB, U.K

Abstract: The automotive industry is following the worldwide vehicle and powertrain electrification trend, which is expected to increasingly grow in next decades. Electrified vehicles represent the main challenge for OEMs and first-tier suppliers which are requested to operate in a relatively new field, facing additional constraints, issues and requirements.

A key point for hybrid and electric vehicles design is the proper sizing and control strategy development of the cooling and heating system, which is not only required to maintain the battery within the optimal temperature operating range and guarantee appropriate cooling of the other electrical components, but also to ensure thermal comfort for the passengers. In fact, the battery cooling system must be strictly linked to the AC circuit to guarantee cooling capacity whenever the ambient conditions do not allow passive cooling with conventional radiators, and linked to High Voltage Heater loop to provide fast warm up of the battery when operating in cold climate, thus adding several degrees of complexity to the management and calibration of a HEV/BEV cooling system characterized by different interacting sub-systems.

In an attempt to address the difficulties arising in the development of cooling systems for electrified vehicles, this paper proposes an effective approach based on CAE methods to speed up and support the system design and control development.

A fully-physical 1D model of the battery cooling circuit and HVAC loop – representing a BEV system layout currently under study – is built in GT-SUITE and used as a plant for control strategies development. A robust and comprehensive control logic capable of managing the cooling and HVAC systems under different operating conditions is developed in MATLAB-Simulink and virtually validated by means of a coupled simulation with the GT-SUITE model.

Following the normal product development process, the MATLAB-Simulink model could be eventually deployed in a real control unit to assess and furtherly validate the proposed strategy.

Keywords: Thermal Management, BEV, HVAC, Cooling, Battery, MiL, CAE, Control.

1. Introduction

Nowadays the more and more stringent pollutant emissions regulations and challenging fleet CO_2 targets are pushing traditional automakers to strongly introduce electrified powertrains and full electric vehicles in their vehicle line-up. Moreover, new players - often with non-automotive background - are entering in the extremely competitive passenger car market with Battery Electric Vehicles (BEV). Those powertrains bring new design concerns regarding the cooling system and a totally new trade-off between vehicle performance, range and cabin comfort [1][2].

Lithium-ion batteries must be well protected against overheating due to safety and aging concerns by means of an effective cooling system, associated to a power de-rating strategy – which implies lower vehicle performance or higher battery charging time – intended to intervene under the most severe circumstances. However, batteries have an optimal thermal operating range significantly lower than a traditional internal combustion engine which can be close to the external ambient temperature in a hot climate region, thus making ineffective any passive cooling system based on the usual air-coolant radiators that uses external ambient air as heat sink [2]. In order to overcome this limitation, the passive cooling system in BEV is typically coupled with – or even replaced by – the cabin Air Conditioning (AC) system by means of a dedicated battery chiller, capable of providing on-demand active cooling to the battery regardless of the external temperature

Lithium-ion batteries also feature a lower operating temperature boundary, below which vehicle operations are not allowed or strongly limited. Thus, an external heating device is typically installed on the circuit to guarantee an acceptable warm-up time and enable full vehicle performance, especially when the vehicle operates in cold climates.

Merging together all those different constrains results in a very complex battery thermal management system, realized as an integration of a passive and active cooling branch plus a heating branch. Defining the optimal coordination of this system is not a trivial engineering problem, which may require a lot of time consuming and expensive testing to properly size the apparatus and to develop an effective control logic, that guarantees the battery thermal control and addresses the cabin thermal comfort requests.

This work tries to address this problem making extensive use of CAE tools, to develop and tune a battery cooling system control logic from the very early stage of system development, forgoing any testing on prototype systems in this preliminary phase. This work was carried out in close collaboration between Powertech Engineering S.r.l., Diesel Emission Control L.t.d. and Sattelo s.r.o. on a concept study of a thermal management system for premium-class BEV.

With that purpose in mind, a fully physical model of the plant was built in GT-SUITE which has been assessed by several works as a robust and accurate platform for vehicle thermal circuits modelling [3][4][5]. Once the plant model was built, it was coupled with the controller model developed in Matlab-Simulink for an integrated simulation in a Model-In-the-Loop approach (MiL). The controller drives virtual actuators and reads from virtual sensors in the plant as they are supposed to be placed in the real system. In this approach both the controller and the plant are represented by models connected in a closed feedback loop process in which the modelled physical environment affects the controller behaviour and vice versa. Given the possibility of simulating virtually any operating condition, it is an extremely powerful tool for control algorithm verification and, since MiL simulations are performed entirely in a virtual environment, it can get rid of any physical component [6].

In the following sections, the plant model and the control logic will be described and results of the integrated simulation will be presented to assess the potentialities of this methodology for control system development.

2. Requirements

The BEV object of the study sits in the premium vehicles segment, with a battery pack capacity of around 100-120 kWh and with a "low temperature" battery cooling system integrated with the cabin air conditioning loop. A passive-only "high temperature" cooling circuit is used for power electronics, on board battery charger and traction motors, which will not be debated in this paper.

The battery cooling system must provide the best operative condition of the battery pack in the ambient temperature range between -20°C and 45°C and in all the different drive and charging modes. According to [7], the Lithium-Ion battery has an optimal operating window between 15 and 35°C in which the full charging and discharging performance are enabled as schematically shown in Figure 1. This operating area is the target for the thermal management system. Outside of this band and between 10°C and 45°C, the charging performance are de-rated down to no-charging allowed at the opposite ends. In the remaining cold and hot operating regions, only discharging is allowed, which is also progressively limited moving away from the optimal area.

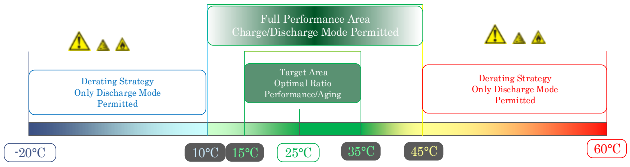

Figure 1 - Battery Pack Operative Temperature

Other than the average battery temperature, a significant variation of cell to cell temperature may cause unbalanced power distribution within the battery pack reducing performance and increasing thermal aging [8]. To guarantee thermal uniformity the cooling system is also asked to limit the temperature difference across the battery pack to 5°C.

3. Cooling system layout and model overview

The BEV thermal management system features two main interacting circuits as shown in Figure 2: a passive circuit flowed by cooling fluid (50% water 50% ethylene glycol) and an active A/C circuit flowed by R134a refrigerant.

3.1. Passive Cooling Circuit

In the passive circuit, the cooling fluid is circulated by means of two parallel electric pumps to maximize the flow rate capacity. It directly exchanges heat with the battery cooling plates and it can be directed to three separate conditioning branches, individually regulated by dedicated fully adjustable Electronic-controlled two-way Valve (EV): the heater or bypass branch, the passive cooling branch and the active cooling branch. The temperature across the battery pack is monitored by two temperature sensors placed at inlet and outlet.

Heater/Bypass Branch

In the heater/bypass branch a high voltage electrical heater is placed. It is used to warm up the battery when operating in cold condition. When the passive cooling strategy is selected, this branch is used as a bypass path (with heater turned off) to modulate the cooling flow to the radiator. Within the model the heater was represented simply via an external heat addition to a pipe with an equivalent wet volume.

Passive Cooling Branch

The passive cooling branch features a fins radiator which exchanges heat with external air. The air flow through the heat exchanger can be controlled by an electrical fan.
In the model, the heat transfer rate is calculated based on the geometrical characterization of the heat exchanger and with the heat transfer coefficients defined by Nusselt correlations directly derived by the experimental data provided by the component supplier [9]. The air side was modelled as a duct in which the air speed at the boundaries is imposed equal to the vehicle speed, mimicking the under-hood air path. Using this approach, the cooling performances are sensitive to the vehicle speed and ambient conditions, making the model suitable to assess the passive cooling performance under real driving conditions. Moreover, using the non-dimensional Nusselt correlations to evaluate the heat transfer coefficients, the model remains predictive if the heat exchanger geometry is scaled, thus becoming a powerful sizing and design tool.
The electrical fan was modelled as a map of volume flow rate and efficiency versus shaft speed and pressure drop. A sketch of the external side of the passive cooling branch is reported in Figure 3.

Active Cooling Branch

In the third branch is placed the battery chiller. It is a heat exchanger between the cooling fluid of the battery circuit and the R134a refrigerant in the A/C loop. The heat exchanger modelling is equal to the passive cooling one, except by the additional complexity introduced by the two-phase nature of the cooling medium.

As far as the battery is concerned, it was modelled with a single lumped thermal mass and a single Thevenin electrical equivalent model, in which the Open Circuit Voltage (OCV) and Internal Resistance (IR) are defined as a function of State of Charge (SOC) and Temperature. The thermal mass exchanges convective heat with a pipe element of the cooling circuit representing the battery cooling plates in terms of volume and surface, and convective and radiative heat with external environment.
Following the same approach, it is possible to improve the battery model discretization down to module or cell level by defining a Thevenin equivalent and a thermal mass per each portion, and representing the detailed flow path of the cooling plates within the battery pack. With this approach, sketched in Figure 4, it would be possible to evaluate cell to cell temperature variation, although some experimental or 3D-CFD validation of the flow distribution may be needed.
Despite a more detailed modelling approach of the battery pack was available, a simplified lumped approach was preferred in this study due to the lack of comprehensive input data at that stage of the development process.
The battery is linked to a dynamic vehicle model which depletes its energy to follow a prescribed vehicle speed trace. The battery power output (discharging) and input (charging) is limited according to the abovementioned temperature limits.

Since the physical system was not yet built at the time of writing, no experimental data were available to validate the model. It could only be validated at the component level by realizing ad-hoc flow bench-like models to compare the predicted components behaviour against the suppliers' specifications. Nevertheless, the uncertainties are related mainly to the piping system, especially for what concerns pressure drop and heat rejection to ambient, which can be considered of secondary importance, given the aim of realizing a realistic model to kick-off the development of the control strategy.

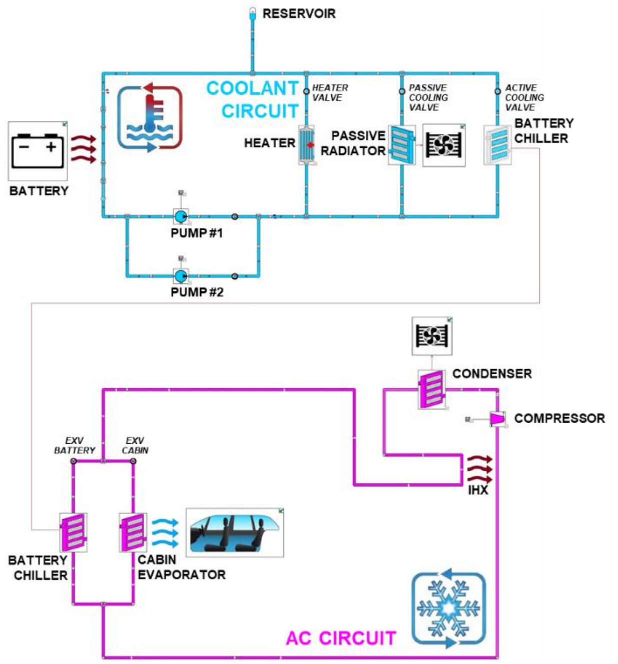

Figure 2 - Cooling System Layout

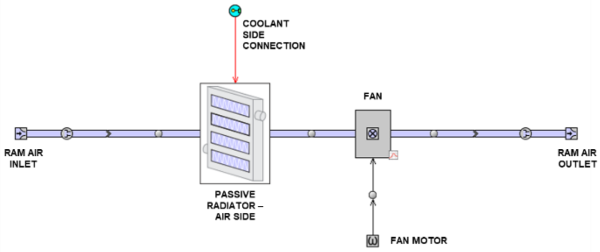

Figure 3 - Passive Cooling Branch / Air Side

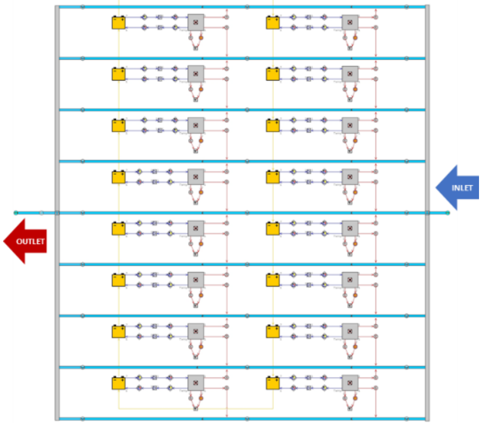

Figure 4 - Cell level battery discretization

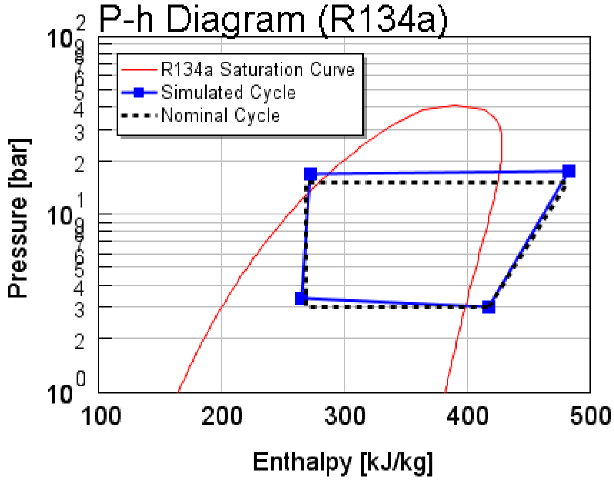

Figure 5 - R134a Cycle

3.2. Active Cooling Circuit

The active cooling circuit serves as cooling source for the cabin and for the battery when the conventional air-cooling is not enough or not applicable. The superheated R134a refrigerant fluid is compressed through a High Voltage (HV) electric compressor modelled with a map of speed, mass flow, pressure ratio and efficiency. The compressed and superheated gas passes through the condenser, exchanging heat with external air and being cooled down to the liquid phase (the condenser external air flow can be increased by a dedicated fan). The modelling of heat exchanger and external duct is equivalent to the passive cooling radiator.
Downstream of the condenser is placed an internal heat exchanger (IHX) to increase the enthalpy difference across the evaporator and reducing refrigerant quality to reduce the pressure drop, resulting in increased system efficiency and cooling performance [10]. Since no information was available on this component, this was simply represented with a lumped thermal mass connecting the two parts of the refrigerant circuit.
Downstream the IHX, the flow path splits in two parallel branches: one directed to the battery chiller and another to the cabin evaporator, which exchanges heat with the passengers compartment. The flow in the two branches is controlled via two fully adjustable Electrical Expansion Valves (EXV) which were modelled as variable diameter orifices.
The two separate branches are connected back together downstream the heat exchangers and the flow, after passing through the low temperature side of the IHX, enters the compressor to complete the thermodynamic cycle.

A target thermodynamic cycle in the design point was provided, which was reproduced to validate the model obtaining a good correlation as shown in Figure 5.

To monitor the thermodynamic cycle, the control strategy can rely on a pressure and temperature sensor placed at compressor inlet (downstream IHX), and a pressure sensor placed downstream the condenser.

3.3. Cabin Circuit

Since no information on the cabin was available, a representative cabin model already available was assumed, with the layout reported in Figure 6. A blower sucks air from the cabin and from external ambient with a fixed split of around 50%. Downstream the blower (modelled as a fan) is placed the cabin evaporator and a PTC (Positive Temperature Coefficient) heater which blows out the conditioned air in the cabin. The cabin was modelled with a lumped 0D approach, as a volume characterized by a single averaged temperature, connected with different thermal masses representing seats, dashboard, frame, etc. The air in the cabin exchanges heat with internal components, absorbs solar radiation from the sun and exchanges convective heat with external environment.

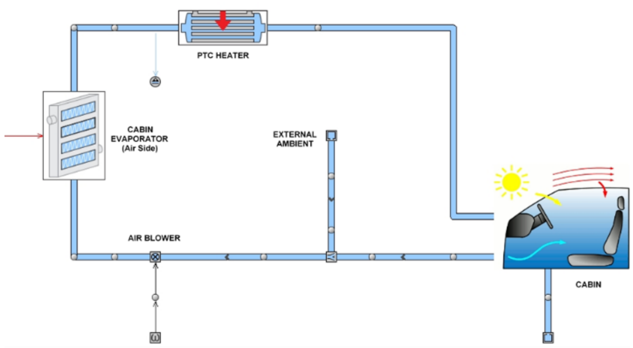

Figure 6 – Cabin Circuit

4. Control System Overview

The control logic was developed in Matlab-Simulink and coupled with the GT-SUITE plant model as shown in Figure 7 and described in detail in the following paragraphs.

The control logic reads from the sensor placed in the model and actuates valves, pumps, compressor and fans with the primary goal of keeping the battery in the optimal temperature range and to guarantee an inlet-outlet temperature differential lower than 5°C, without compromising whenever possible the passengers comfort in the cabin compartment.

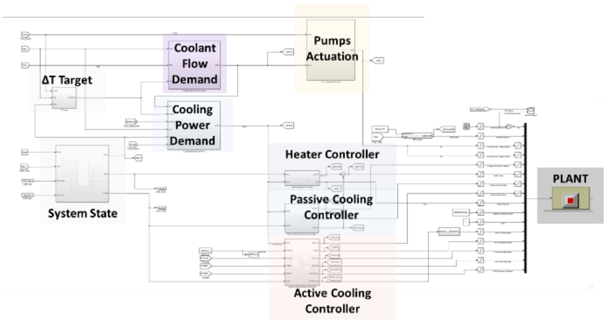

Figure 7 - Controller overview

The thermal management strategy determines the control actions by means of three main sub-controllers: the coolant flow demand, the cooling power demand and the system state.

<u>Coolant Flow Demand Strategy</u>

The coolant flow demand determines the desired coolant flow using a closed loop PI controller targeting the desired temperature difference across the battery pack (5°C in most of the operating conditions and reduced in some cases to increase the coolant flow and the heat exchange). The coolant flow output of the controller is used to lookup the pumps speed needed to achieve the desired flow, based on the actual cooling or heating strategy adopted.

<u>Cooling Power Demand Strategy</u>

The cooling or heating power demand strategy determines the cooling or heating power needed to achieve the target battery outlet temperature. It works with a Feedback (FB) plus Feed Forward (FF) loop: the feedforward loop imposes the estimated battery heat rejection from the Battery Management System (BMS) in the plant model, while the Feedback part closes the loop by means of a PI controller targeting the desired battery outlet temperature.

<u>System State Determination</u>

By means of a States Machine the controller first defines whether the battery has to be cooled or heated, checking the battery outlet temperature. To avoid instabilities a hysteresis logic is applied between 20°C and 30°C. To limit the waste of energy, since heating and cooling power (in case of active cooling) is drawn from the battery, the target battery outlet temperature is different in the case of cooling or heating: in heating case the controller targets a battery outlet temperature of 20°C, while in cooling the target is set to 30°C. This results in a dead band within the hysteresis values in which the controller doesn’t command any cooling or heating action.
In case cooling power is requested, the states machine decides either to use the passive cooling or the active cooling system based on the difference between ambient temperature and actual battery temperature. A safety strategy is implemented to automatically switch to active cooling if the passive cooling is found to be ineffective in reducing the battery temperature to the target value.

Depending on the mode selected by the state machine different controllers are activated:

<u>Heater Controller</u>

In case heating mode is selected, the heating power demand is directly imposed to the heater as electrical load, with the corresponding energy drawn from the battery.

<u>Passive Cooling Controller</u>

The passive cooling controller works as an engine-like cooling circuit, splitting the flow between the passive cooling branch, in which the coolant exchanges heat with external air, and the heater branch (with the cooler off) which is used as a radiator bypass. The flow is regulated by the two-way valves downstream heater and radiator. To increase the cooling power, according to the cooling power demand strategy, the system progressively opens the passive cooling branch to drive more coolant through the radiator while closing the heater one. Once the heater branch is fully closed and the passive cooling one is fully open, the system activates and increases the radiator fan speed until the desired heat transfer rate is achieved.

<u>Active Cooling Controller</u>

The Active Cooling Controller works computing a thermal budget taking into account the maximum available cooling power and the cooling power demand from the cabin and from the battery. The system initially tries to satisfy both the clients (battery and cabin) up to the saturation of the maximum cooling capacity of the plant. Once the maximum cooling power is reached, it starts de-rating the cooling to the cabin privileging the battery for components safety reasons. Once the thermal budget is computed, the compressor speed is looked up based on the total cooling request.

The condenser fan speed is controlled via a FB+FF loop closed on the pressure measured downstream of the condenser.
The control of the two EXVs placed on the battery chiller and cabin evaporator parallel branches works on two levels: it firstly determines an equivalent total throat area across the two valves using a PI controller targeting the superheat value at the compressor inlet, then, the equivalent area is split proportionally to the cooling request on the two valves. The superheat value at compressor inlet is determined by computing the difference between actual compressor intel temperature and the R134a saturation temperature at the actual compressor inlet pressure.

Cabin Controller

The cabin thermal control strategy was kept very simple since it was not the main focus of the work.
In the first simulated cases, in which the cabin temperature was not considered, the blower air mass flow was simply regulated to extract a certain cooling power out of the evaporator.
Later on, for comprehensive vehicle performance and thermal comfort simulations, the cabin thermal power request was regulated by a PI controller to meet a target cabin temperature. In this case the blower air mass flow was regulated to blow in the cabin a conditioned air temperature 10°C higher than the cabin target in case of heating, and 10°C lower in case of cooling.

5. Model Setup

Once the physical model had been completed and validated, it was coupled with the controller developed in Simulink. The connection between the two simulation platforms is made by means of a special component named “Simulink Harness”, which makes the GT-SUITE model as a black-box function in Simulink. Within this component the Input and Output (I/O) signals which are exchanged between the controller and the plant model are defined:

- GT-SUITE plant model outputs (corresponding to Simulink model inputs) represent the measures from the different sensors placed along the model.
- Simulink model outputs (i.e., GT-SUITE model inputs) represent the signals governing the actuators placed in the plant model

Since GT-SUITE and Simulink can run with different timesteps, here is also defined the communication interval. In this case the communication interval was set equal to the Simulink timestep to 0.5 seconds. The implicit solver of the GT-Suite model was instead set to run with a timestep of 0.2 seconds, while the spatial discretization of the circuits was set to 25mm, resulting in around 700 flow volumes. With these settings, the simulation model model was able to run 2-5 times faster than real-time on an Intel i7 3.60 GHz CPU.

6. Results Assessment

To evaluate the results of the developed control strategy, some steady state “worst case scenarios” were run. The condition with the highest thermal load for the cooling system corresponds to the case of battery recharging in a fast charging station at 100 kW, with external hot ambient air temperature (45 °C) and the passengers staying in the car, thus asking for cabin cooling. To replicate those conditions in a test bench-like approach the battery heat rejection and demanded cabin cooling power were fixed to the value reported in Table 1. The battery heat rejection was anyhow limited according to the battery temperature limitations (Figure 1): 4.5kW of battery heat rejection was assumed to correspond to the maximum charging power, but this value was imposed only in the temperature window between 15 and 35°C, de-rated linearly between 15°C and 10°C, and 35°C and 45°C and imposed to zero (no charging allowed) outside these boundaries. On the other hand, the active cooling thermal budget strategy was kept on, therefore the cabin cooling power request is de-rated according to the total available cooling power, giving priority to battery cooling.
Moreover, the system temperature at the beginning of the simulation was imposed equal to ambient temperature to initialize the cool-down transient.

Table 1: Steady-State Cases

Ambient Temperature: 45°C			
Design Case ID	**Battery Heat Rejection**	**Cabin Cooling Demand**	**Total Cooling Demand**
##	[kW]	[kW]	[kW]
R_45-1	0.5	4.0	4.5
R_45-2	1.0	5.0	6.0
R_45-3	2.0	3.0	5.0
R_45-4	2.0	6.0	8.0
R_45-5	2.5	4.0	6.5
R_45-6	3.0	3.0	6.0
R_45-7	3.5	3.5	7.0
R_45-8	4.0	4.0	8.0

The envelop of the results in terms of battery outlet temperature, temperature difference across the battery pack and heat exchanged by the battery chiller and cabin evaporator is reported in Figure 8.

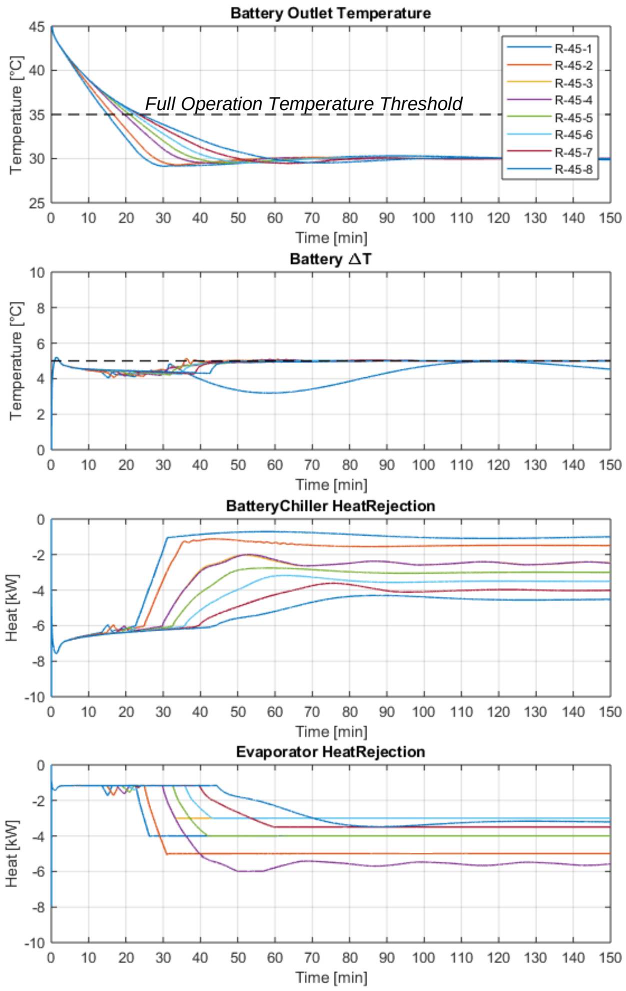

Figure 8 - Steady State Cases Envelop Results

As shown, the controller can effectively regulate the coolant mass flow rate to keep the battery temperature differential below 5°C in all the cases. In case R-45-1 the temperature difference is significantly lower than for the other cases, because the battery heat rejection is very low (0.5 kW), thus, even though the pumps are running at the minimum speed, the mass flow is higher than what is needed to keep 5°C.

For the sake of clarity, it is worth to highlight that in all the plots in which a thermal power is reported, negative values represent removed heat from water or refrigerant, while positive values represent a heat addition to the fluid.

Looking at the Battery Chiller and Evaporator Heat Rejection, it results clear how the system immediately reacts to the high initial battery temperature by pushing the battery chiller to the maximum power, in order to cool the battery down in the optimal operation area, where charging is allowed, while limiting the cabin cooling. As soon as the battery temperature falls within the optimal operating range (<35°C), the battery cooling request starts to be modulated while the allowed cabin cooling is increased accordingly. Even though the active cooling system was designed to withstand a total cooling of 8 kW, in some cases (R-45-4 and R-45-8) the cabin cooling power remains de-rated also when the battery temperature is at target. This is due to the fact that the battery cooling system, other than dispose the battery heat rejection, must also counterbalance the external heat from the external (hot) environment into the system.

To analyse deeper how the controller behaves, a detailed breakdown of the case R-45-5 is reported in Figure 9. In this case the battery heat rejection is imposed to 2.5 kW, the cabin requests 4kW and all the system is soaked at 45°C at the beginning of the simulation. At key-on, immediately, the system recognizes the battery is too hot to be charged and asks the maximum cooling power out of the battery chiller (7kW). The cabin cooling request is de-rated down to 1kW to saturate the total cooling capacity of the system. At the same time the cooling flow demand strategy actuates the coolant pumps to the maximum speed, to maximize the heat exchange. On the refrigerant side, the superheat controller regulates the equivalent total EXV diameter to achieve 25°C of superheat at the compressor inlet, which is achieved in around 5 minutes.

As soon as the battery outlet temperature approaches the target, the battery cooling power demand starts to be reduced and the cabin cooling is increased up to the requested value of 4kW. To achieve the requested cooling power split between battery chiller and cabin evaporator, the EXVs are actuated proportionally. In the cooling power request plot of Figure 9, together with the requested value in solid line, the actual heat rejection (predicted by the GT-SUITE model) for the battery chiller and cabin evaporator are reported in dashed lines. While for the cabin evaporator the actual value overlaps exactly the demand, since the blower mass flow is adjusted to extract exactly the requested cooling power, for the battery chiller some deviation is observed. This deviation has to do with the actual effectiveness of the heat exchanger which is not considered when computing the power split between battery and evaporator.

To evaluate the heating and passive cooling strategy behaviour, an additional case was run in which the system is soaked at the ambient temperature of -10°C and a battery heat rejection of 4kW was imposed (de-rated according to the temperature limits). The results of this case are reported in Figure 10.

After key-on, the State machine recognizes that the battery needs to be warmed and switches to heating mode (System State = 1, Cooling State = 0). The heater branch valve is fully opened (Heater EV=1) and the maximum heating request of 7kW is addressed to the electrical heater. As soon as the battery outlet temperature approaches the lower battery temperature threshold of 20°C, the heating power request is reduced down to 0 until the target is reached. After this point the system enters in the dead band: the battery target temperature is still 20°C and the cooling power request computes a cooling need, but the system remains in heating mode, thus the cooling request could not be accomplished and the battery temperature keeps increasing since the internal heat rejection is not dissipated. This was done to avoid any waste of energy due to heating or cooling whenever the battery is within its optimal operating temperature range.

As soon as the battery outlet temperature exceeds 30°C the system switches in cooling mode (System State = 2) and, since the external ambient air temperature is low, the system decides to use the passive cooling (Cooling State = 1). After this point the cooling power request is reset and the target temperature is set to 30°C. The heater EV valve starts closing while the passive cooling valve is opened. Once the passive cooling valve is fully open the radiator fan speed is increased until the battery temperature reaches the target.

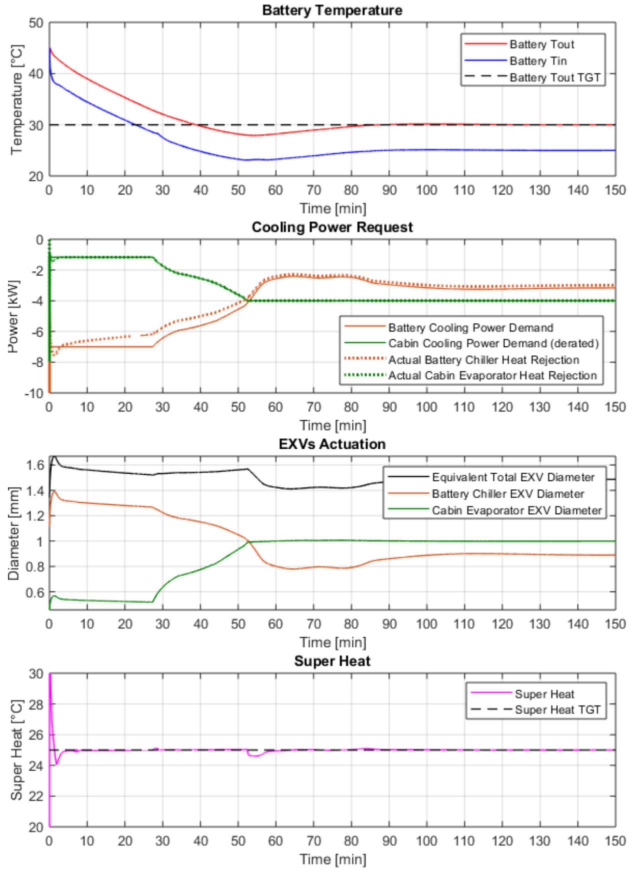

Figure 9 - R-45-5 detailed results

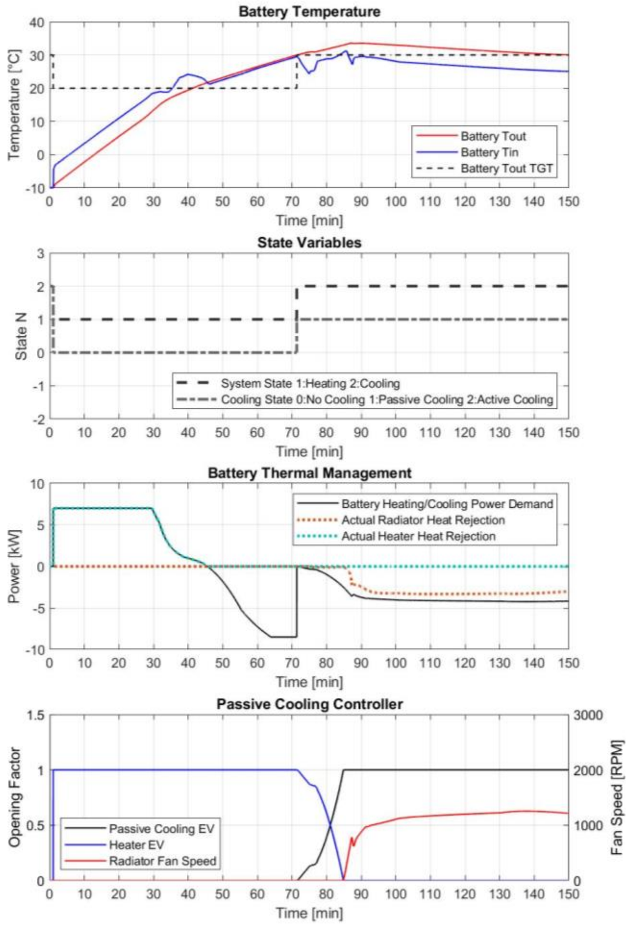

Figure 10 - Case A-10 Battery Heating and Passive Cooling

To evaluate the system behaviour in a more dynamic situation, the model was set to run a driving cycle. All the electrical loads (fan, compressor, pumps, heaters, etc) were connected to the battery, and the cabin cooling control strategy, which was previously configured to impose the heat rejection, was modified to control the cabin temperature.

Figure 11 reports the results of this case, in which the vehicle is soaked at 40°C and runs a series of RTS 95 cycles starting from fully charged battery. Five passengers are supposed to be in the car, thus a 500 W heat source is placed in the cabin [11].

As the simulation starts, both for cabin and battery a high cooling power is requested. However, as shown before, the battery cooling has higher priority and the cabin cooling is limited to around 1 kW, which is pretty ineffective in cooling the cabin, as shown by the cabin temperature profile. As soon as the battery temperature reaches the optimal thermal window (at around 10 minutes), some cooling power can be freed from the battery and transferred to the cabin, that can now be effectively cooled, reaching the target temperature at around 13 minutes.

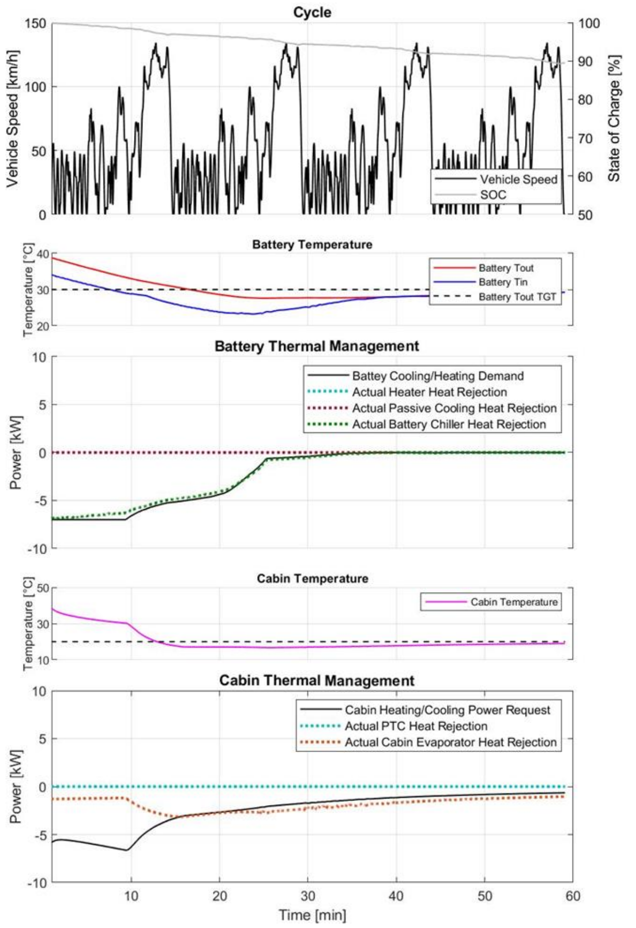

Figure 11 - RTS 95 results

7. Conclusion and Next Steps

BEV thermal management is a not trivial engineering problem which requires the coordination of different circuits and the management of the trade-off between vehicle performance, component protection, energy saving and cabin comfort.

In this paper it was shown how, making an extensive use of CAE tools, it is possible to develop an effective thermal management control strategy without the needs of any prototype build. The control strategy handles the battery cooling and the cabin conditioning in any load profile and realistic external boundary conditions, by regulating a passive cooling and heating circuit plus an active cooling circuit that serves both the cabin and the battery. The controller is able to cool down the battery from 45°C to 35°C (where its performances are not limited) in between 10 and 25 minutes (depending on the load) at 45°C external temperature. In a real-driving condition, the cabin can be effectively cooled 10 minutes after key on if the vehicle operates at 40°C external temperature. To shorten this conditioning time, a higher sized active cooling circuit would be needed.

To summarize, this approach allows the developer to keep tracking of performance, safety and comfort at the same time, making it a powerful engineering tool. Given the possibility to monitor all those different aspects, it gives the best results if the control developing process proceeds along with the system design, in order to dynamically resize the system to accommodate the needs that comes out during the virtual testing.

The developed thermal control strategy will be deployed in a real Electronic Control Unit (ECU) and used to drive a real system prototype which was being built at the time of writing. As next step the model will be refined and validated together with the control strategy, thanks to the data coming from the test bench.

Acknowledgement

The authors gratefully acknowledge the precious contribution to this work by Roberto Cardano from Sattelo.

References

[1] S. Chowdhury *et al.*, “Total Thermal Management of Battery Electric Vehicles (BEVs),” in *SAE Technical Paper Series*, 2018, vol. 1, no. May, pp. 1–7, DOI: 10.4271/2018-37-0026.

[2] A. A. Pesaran, “Battery Thermal Management in EVs and HEVs : Issues and Solutions,” *Advanced Automotive Battery Conference, Las Vegas, Nevada*, 2011.

[3] C. N. Grimaldi, C. Poggiani, A. Cimarello, M. De Cesare, and G. Osbat, “An Integrated Simulation Methodology of Thermal Management Systems for the CO 2 Reduction after Engine Cold Start,” *SAE Technical Paper Series*, vol. 1, 2015, DOI: 10.4271/2015-01-0343.

[4] F. Millo, S. Caputo, C. Cubito, A. Calamiello, D. Mercuri, and M. Rimondi, “Numerical Simulation of the Warm-Up of a Passenger Car Diesel Engine Equipped with an Advanced Cooling System,” *SAE Technical Paper Series*, vol. 1, 2016, DOI: 10.4271/2016-01-0555.

[5] G. Seider, J. Mehring, and C. Weber, *A high-resolution warm-up simulation model for a gasoline engine with advanced thermal control*. Woodhead Publishing Limited, 2011, DOI: 10.1533/9781780630489.3.189.

[6] R. Matinnejad, S. Nejati, L. Briand, T. Bruckmann, and C. Poull, “Automated model-in-the-loop testing of continuous controllers using search,” *Lecture Notes in Computer Science*, vol. 8084 LNCS, pp. 141–157, 2013, DOI: 10.1007/978-3-642-39742-4_12.

[7] A. Pesaran, “Battery Pack Thermal Design” in *Advanced Automotive Battery Conference, Detroit, MiI*, 2016.

[8] L. H. Saw, A. A. O. Tay, and L. W. Zhang, “Thermal management of lithium-ion battery pack with liquid cooling,” *Annual IEEE Semiconductor Thermal Measurement and Management Symposium*, vol. 2015-April, no. March, pp. 298–302, 2015, DOI: 10.1109/SEMI-THERM.2015.7100176.

[9] G. Technologies, *GT-Suite Cooling System and Thermal Management Application Manual and Tutorial*. 2018.

[10] S. Kurata, T. Suzuki, and K. Ogura, “Double-pipe Internal Heat Exchanger for Efficiency Improvement in Front Automotive Air Conditioning System,” *SAE Technical Paper Series*, vol. 1, no. 724, 2010, DOI: 10.4271/2007-01-1523.

[11] C. Binggeli, *Building Systems for Interior Designers*. 2011, ISBN 0471417335.

Glossary

0D: Zero-Dimensional

AC: Air Conditioning

BEV: Battery Electric Vehicle

BMS: Battery Management System

CAE: Computer Aided Engineering

CFD: Computational Fluid Dynamics

ECU: Electronic Control Unit

EV: Electronic-controlled two-way Valve

EXV: Electrical Expansion Valve

FB: Feed Back

FF: Feed Forward

HEV: Hybrid Electric Vehicle

HV: High Voltage

HVAC: Heating Ventilation and Air Conditioning

I/O: Input and Output

IHX: Internal Heat Exchanger

IR: Internal Resistance

MiL: Model in the Loop

OCV: Open Circuit Voltage

OEM: Original Equipment Manufacturer

PTC: Positive Temperature Coefficient

SOC: State of Charge