Heat flow rate based thermal management for electric vehicles using a secondary loop heating and cooling system

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Thermal management energy consumption has crucial impact on the range of an electric vehicle [1]. This requires optimal use of available thermal energy while meeting cabin comfort and component temperature requirements. Secondary loop systems are a promising approach in order to meet these requirements while reducing refrigerant charge and simplifying the vehicle front end. Based on a simplified drivetrain model heat losses are estimated. Using first order thermal mass models and typical component temperature constraints, energy surpluses and energy deficits are calculated. Based on temperature levels and energy balance a central refrigeration unit is then used to provide additional heating power (heat pump) or cooling power (AC mode). An ambient radiator is used as heat sink if required. Simulation results for two different drive cycles show necessary heat flow rates between components and three cooling cycles for typical thermal management use cases. The energy flow rate based thermal management approach remains without further specification of a heating and cooling system (including pumps, valves, heat exchangers and electric heaters). Therefore presented results can be used for system design or lay the foundation for development of control strategies.

NOTATION

EM	Electric machine	CC	Cooling cycle
PE	Power electronics	HC	Heating cycle
BAT	Battery	LC	Low temperature cycle
CHRG	Charging device		
AMBX	Ambient heat exchanger	CAB+	Cabin heating
COP	Coefficient of performance	CAB-	Cabin cooling

INTRODUCTION

Traditionally the main goal of electric vehicle thermal management is to protect powertrain subsystems from overheating. A more general view would be to consider climate control (including safety functions such as defrosting and defogging) as part of the thermal management system. So that the term "thermal management", in this case, refers to the total number of all cooling and heating systems including component cooling, component heating and the HVAC (heating, ventilation, and air conditioning) system. A schematic overview of a typical thermal management system for an electric vehicle is shown in figure 1.

Recent attempts to find energy-efficient thermal management systems for electric and plug-in hybrid electric vehicles have led to secondary loop systems as an alternative approach to meet dynamic heating and cooling demands. Recently, much research has been performed regarding automotive applications of secondary loop systems. Ghodbane et al. [2] present a secondary loop system with the HFC refrigerant R152a. Kowsky et al. [3] demonstrate a hermetic encapsulated central thermal management unit for the use in electric and hybrid vehicles using the battery waste heat as energy source during heat pump mode.

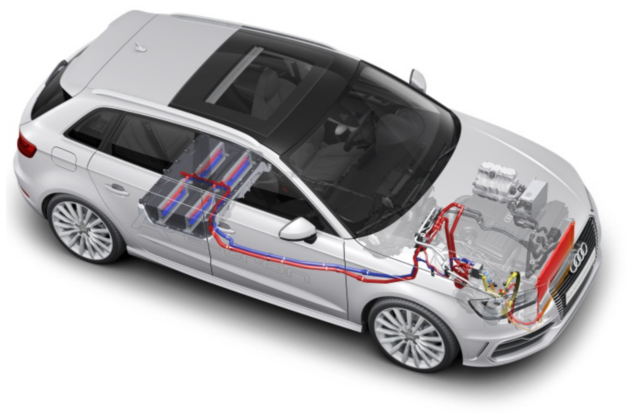


Figure 1 – Typical thermal management system for an electric vehicle

Not always thermal management is considered as an interconnected system in vehicle thermal management, but instead dedicated cooling loops for each subsystem are developed. Much focus has therefore been drawn on isolated systems, instead of a holistic approach, that can potentially benefit from synergetic effects by combining cooling and heating loops [4]. Secondary loop systems hence focus on these benefits.

Figure 2 shows a secondary loop system with a compact refrigeration cycle and a secondary loop flow distribution unit. This work is intended to study thermal energy flow on an abstract level assuming a similar secondary loop system.

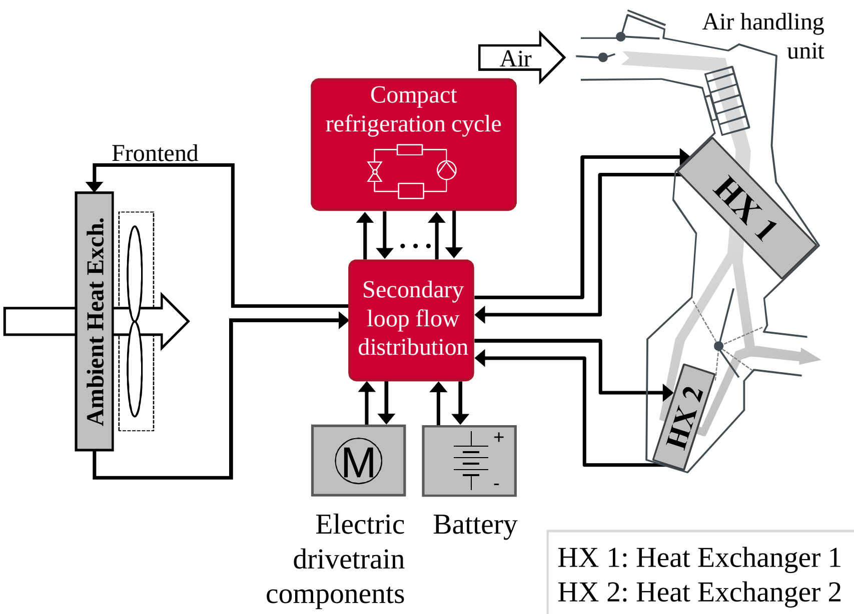


Figure 2 – Thermal management using a compact refrigeration cycle and secondary loop flow distribution unit

1.1 Electric Drivetrain

Thermal losses of electric drivetrain components result in heating of components. This could potentially lead to overheating. To avoid overheating a cooling system ensures that the maximum component temperatures are not exceeded. In case of the battery a given minimum component temperature can also require heating. Figure 3 illustrates a simple scenario for common temperature limits: While power electronics and electric machine usually tolerate a wide temperature range, the battery temperature must be maintained within narrow temperature bounds. This can necessitate to heat the battery if its temperature is too low. In general, typical batteries operate best between 10°C and 30°C [5].

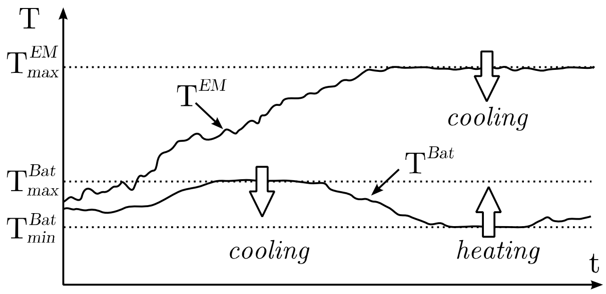


Figure 3 – Component cooling and heating

1.2 Air conditioning components

Usually the air handling unit is composed of a heat exchanger dedicated for cooling and a heat exchanger dedicated for heating, a blower and various air flaps. This allows heating and cooling of the passenger compartment as well as dehumidification of inflow air in a process called reheat. During reheat air is first cooled close to 0°C in order to expel saturated water and afterwards reheated to comfort temperature. Contrary to conventional A/C systems the described system uses a secondary loop heat exchanger as cooling device. Therefore, no refrigerant is entering the interior of the vehicle. This is done by providing cold and hot coolant for cabin heating and cooling.

1.3 Primary loop cycle

A refrigeration cycle is able to move heat from a lower temperature level to a higher temperature level using mechanical work. It can therefore be used as a refrigeration device for cooling or a heat pump device for heating depending on the perspective. In automotive application the refrigeration cycle is mainly used for air conditioning. However, for electric vehicles in many cases waste heat is only available at lower temperature level because the overall drivetrain efficiency is much higher compared to conventional vehicles. Therefore, both heating and cooling are required applications for a refrigeration cycle.

The vapor compression refrigeration cycle contains an electrical driven compressor, an (indirect) condenser, an expansion valve, an (indirect) evaporator and a refrigerant accumulator. Using a simple and compact refrigeration cycle allows reduction of refrigerant [6] and hermetic encapsulation. “Indirect” refers to the fact that the compact refrigeration cycle only provides hot and cold coolant as secondary fluid. This means that all connection logic has to be implemented by a coolant distribution system. In order to identify the necessary topological configurations, this work focuses on simulation of an electric vehicle thermal management system based on thermal energy flow rate between subsystems using simplified models.

METHODOLOGY

All simulations are carried out using a simplified powertrain model, similar to [7], to calculate powertrain losses and an energy flow rate based thermal management model based on first order thermal masses and time-dependent solution of the energy balance equations in MATLAB/Simulink. HVAC requirements are calculated by assuming heating and cooling loads of a standard size passenger car with comparable comfort requirements as a conventional vehicle heated by heat losses from a combustion engine. Simulation of (temperature dependent) drivetrain losses is decoupled from the simulation of the thermal management system. Some models might be unsuited for general statements on energy consumption due to oversimplification. The main focus here is to describe required heat flow on a fundamental level. This justifies the use of such simplified models.

The simulation is performed for two different scenarios with different thermal management requirements: A winter scenario at average temperature of -3°C and a summer scenario at average temperature of 31°C. In both cases the battery is charged for 45 min. After 60 min a Motorway-150 CADC (Common Artemis Driving Cycle) is repeated three times resulting in a drive cycle of 53 min at 100 km/h on average.

SIMULATION MODEL

A compact electric vehicle with a range of 150 km and an average energy consumption of 15 kWh/100 km is assumed. Basic vehicle parameters are summarized in Table 1.

		Vehicle	
Parameter	*Unit*	*Description*	*Value*
m_v	[kg]	vehicle mass	1750
f	[-]	friction coefficient	0.01
r_{tire}	[m]	tire radius	0.3125
c_w	[-]	air drag coefficient	0.27
A	[m^2]	frontal area	2.23
n_{gear}	[-]	transmission rate	5.66
		Environment	
ρ	[kg/m^3]	air density	1.0133
v_w	[m/s]	wind speed	0
g	[m/s^2]	standard gravity	9.81

Table 1 - Assumed vehicle und environment parameters

The simulation model is broken down into three parts: Simulation of drivetrain losses, simulation of HVAC requirements and simulation of the thermal management system.

3.1 Drivetrain model and thermal losses

Figure 4 shows the powertrain components of an electric vehicle, including one or more electric machines (EM), power electronics (PE) and a high-voltage battery. All of these components have in common that their efficiency characteristics and therefore thermal losses vary depending on component temperatures and power requirements.

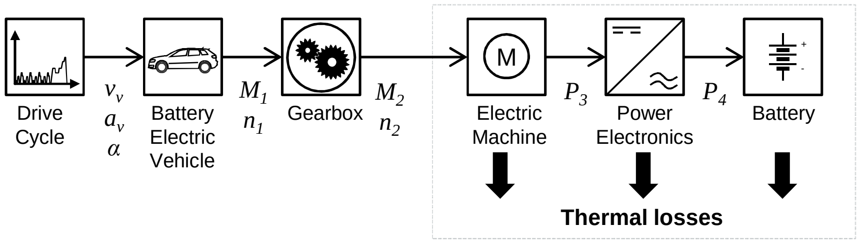


Figure 4 – Overview drivetrain model

The power requirements for each component are calculated based on the power requirements of the previous component. Feedback between components, that ensures that power requirements by one component can be met by the next component, is not considered. Plausibility checks show that for the considered drive cycles each component is able to deliver the required power. Vehicle speed v_v, acceleration a_v and slope α are given by the drive cycle. The overall drive resistance force is calculated by the sum of drive resistance forces [8]:

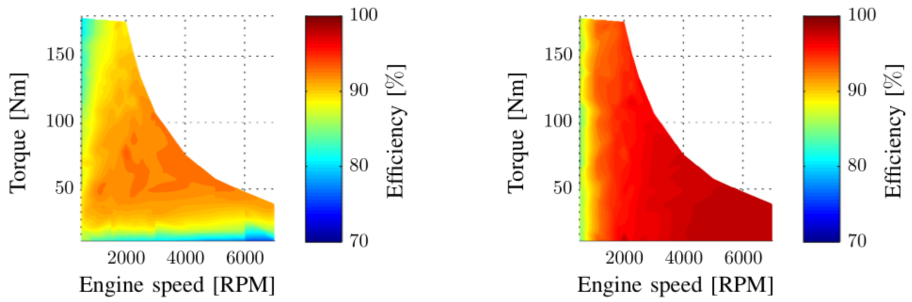
$$F_{drivetrain} = \underbrace{f \cdot m_V \cdot g \cdot \cos(\alpha)}_{F_{roll}} + \underbrace{m_V \cdot g \cdot \sin(\alpha)}_{F_{elev}} + \underbrace{c_w A \tfrac{\rho}{2} (v_V + v_W)^2}_{F_{air}} + \underbrace{m_V a_V}_{F_{acc}}$$

where F_{roll} is rolling resistance, F_{elev} is resistance caused by road slope, F_{air} is air resistance and F_{acc} is resistance due to acceleration or deacceleration. Mechanical power requirements are given by $P = F_{drivetrain} \cdot v_v$. For tire radius r_{tire} torque $M = F_{drivetrain} \cdot r_{tire}$ and rotational speed $n = 2\pi \cdot r_{tire}$ are then calculated. A fixed transmission n_{gear} of 5.66 is assumed resulting in a higher rotational speed and reduced torque. Based on efficiency maps $\varepsilon = f(n, M)$ for a synchronous electric machine and power electronics of a 100 kW drivetrain the required power drawn from the battery is estimated (Figure 5 a/b). Values for vehicle parameters m_v, f, c_w, A and environment assumptions g, ρ, v_w are given in table 1.

Based on the power each component provides $P_{provided}$ and the efficiency ε, (thermal) power loss P_{loss} (Q_{loss}) and required power $P_{required}$ are calculated.

$$P_{loss} = (1 - \varepsilon)/\varepsilon \cdot P_{provided} = Q_{loss}$$

$$P_{required} = 1/\varepsilon \cdot P_{provided}$$



(a) Electric Machine (b) Power Electronics

Figure 5 – Efficiency maps and thermal power losses within the powertrain [9]

A 26.5 kWh lithium-ion battery pack with 3 parallel stacks of 84 series-connected cells is assumed (table 2). For the battery an equivalent circuit-based battery model is used

$$V_o = V_{oc} - I_d R_{int} = f(SOC) - I_d R_{int}(T_{Bat}, SOC)$$

where output voltage V_o depends on open circuit voltage V_{oc} and an equivalent internal resistance R_{int}.

Battery			
Parameter	*Unit*	*Description*	*Value*
Type		Lithium-ion 3p84s	
C	[kWh]	Capacity	26.5
SOC_0	[-]	Initial state of charge	0.65

Table 2 - Assumed battery parameters

While internal resistance depends on average cell temperature T_{Bat} and state of charge SOC, cell temperature in turn, varies with battery internal state and external stress [10]. Any battery operation generates heat due to internal resistance when the cell delivers power to electric loads or is charged. These losses can be calculated from

$$P_{loss} = {I_d}^2 R_{int}.$$

Moreover, additional auxiliary loads of 450W are assumed for e. g. ECUs, lighting, entertainment. The charging device efficiency ε_{chrg} is assumed to be constant at 0.92.

3.2 Thermal Component Model

A simplified lumped first order model is used. The component temperature $T_m(t)$ is given by the differential equation

$$mc_{p,m}\, dT_m(t)/dt = Q = Q_{dis} + Q_{add/rem} + Q_{loss}$$

where $mc_{p,m}$ is the thermal mass of the component, $Q_{add/rem}$ is added or removed heat by the thermal management system and Q_{loss} is thermal loss of the component caused by non ideal operation. Dissipation losses to the environment Q_{dis} are estimated with overall heat transfer coefficient *k* and area *A* from

$$Q_{dis} = kA \cdot (T_\infty - T_m).$$

The temperature of the thermal mass T_m has to stay within given boundaries:

$$\mathrm{T}_{min} \leq \mathrm{T}_m \leq \mathrm{T}_{max}$$

Temperature boundaries, thermal masses mc_p and heat transfer coefficients kA are given in table 3.

Component	T_{min} [°C]	T_{max} [°C]	mc_p [J/K]	kA [W/K]
Electric machine	-30	65	60000	1
Power electronics	-30	85	2000	3
Battery	-5	35	130000	3
Charging device	-25	70	2000	4

Table 3 – Allowed temperature range, thermal mass and heat transfer coefficient

3.3 Cabin heating and cooling demand

Cabin heating and cooling demands are calculated based on a first order cabin model. Only loss to the environment Q_{loss} and heating or cooling effort Q_{hvac} are considered.

$$Q_{loss}(t) + z \cdot Q_{hvac}(t) = mc_{p,cab} \frac{dT_{cab}(t)}{dt}$$

$$Q_{loss}(t) = kA_{cab}(T_{cab}(t) - T_{amb})$$

Equivalent thermal capacity of the cabin $mc_{p,cab} = 123\, kJ/K$ and heat transfer coefficient between cabin and environment $kA_{cab} = 90\, W/K$ are estimated based on simulation data from a higher order simulation model. Mass flow in and out of the cabin is captured implicitly. During “ignition on” (z = 1) the heating and cooling power Q_{hvac} that will drive T_{cab} towards the comfort temperature $T_{comfort}$ is modeled by a PI controller

$$Q_{hvac}(t) = K_p(e(t) + \frac{1}{T_i}\int_0^t e(\tau) \cdot d\tau)$$

with error $e(t)$ defined as

$$e(t) = T_{cab}(t) - T_{comfort}(T_{amb}, \Phi_{amb}).$$

Controller parameters are chosen to meet typical dynamic heating and cooling requirements. Different gains K_p for heating and cooling ($K_{p,heating} = 286$ and $K_{p,cooling} = 212$) are chosen to consider different dynamic behaviors during cooling and heating. The integral time is set to $T_i = 1400\, s$. Based on ambient temperature, humidity and solar radiation the comfort target temperature $T_{comfort}$ and the coolant side temperature level T_{cab} at which heating or cooling is provided are calculated using a model of an air conditioning control unit.

$$T_{comfort} = f_1(T_{amb}, \Phi_{amb})$$

$$T_{cab} = f_2(T_{amb}, \varphi_{amb}, \Phi_{amb})$$

3.5 Thermal management strategy

The thermal management concept is based on thermal energy flow. At each time step the optimal energy flow is calculated assuming steady state. Heat is transferred between components and coolant cycles. Each coolant cycle is balanced to net power of 0. Coolant cycles are divided into three sub cycles: A **heating cycle (HC)**, corresponding to the hottest system temperature – usually the cabin heater, a **cooling cycle (CC)** which is connected to the ambient heat exchanger and a **low temperature cycle (LC)** corresponding to the lowest system temperature. All powertrain components (on board charging device, battery, power electronics, electric machine) can be included in the **cooling cycle** for “passive cooling” or in the **low temperature cycle** for “active cooling”. Electric machine and power electronics can also be included in the **heating cycle** as heat source for cabin heating depending on the current temperature level and heating/cooling requirements. It is assumed that the thermal management will only actively interfere when maximum or minimum allowed temperature are reached.

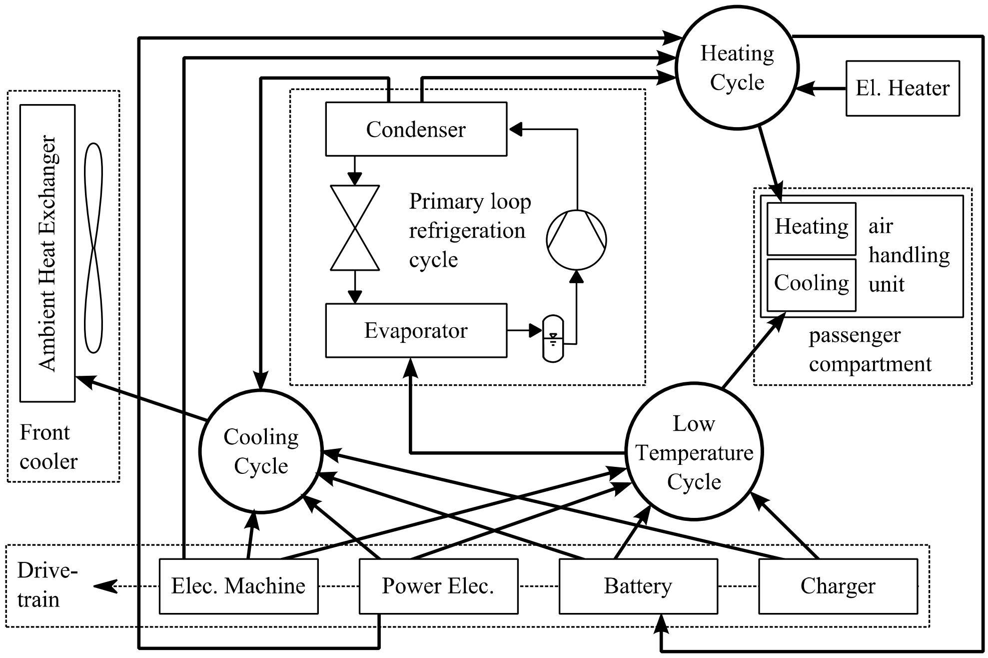


Figure 6 - Possible energy flows between components and coolant cycles

During passive cooling (not to be confused with "passive" air cooling) coolant circulates between components that must be cooled and an outside air heat exchanger. During active cooling the refrigeration cycle is used as heat sink. Active cooling allows cooling of components below ambient temperature T_{amb}. Active cooling is also required for air conditioning by providing coolant below ambient temperature for the cabin cooler. In heat pump mode the low temperature cycle serves as heat source for heat at low temperature, which is then transferred to a higher temperature level.

The thermal management system remains without further specification of a technical realization which would consist of pumps, valves, heat exchangers, heaters, expansion devices and a compressor. A purely energy flow based approach allows to derive basic understanding of the thermal demands of an electric vehicle being true for a wide range of systems in contrast to an approach solely suited for a specific system. In order to capture essential effects of thermal management, the following assumptions are made:

- Each component can only be associated with one or no coolant cycle.
- The cooling cycle is always associated with the ambient heat exchanger.
- The condenser of the compact refrigeration cycle (HOT) is either connected to the heating cycle or to the cooling cycle.
- The evaporator (COLD) is always connected to the low temperature cycle.
- Cabin heater (CAB+) and cabin cooler (CAB-) are always connected to heating and low temperature cycle respectively.
- When sufficient heat is available by thermal losses of components inside the powertrain, heat from the hottest component is used for cabin heating.
- A refrigeration cycle (A/C) is used for cabin cooling or (active compressor) battery cooling. Abundant waste heat is dissipated to the environment by the front cooler.
- When component temperature is high enough, waste heat is directly used for cabin heating if necessary.
- No heating/cooling is done until min./max. allowed temperature is reached.

Table 4 summarizes the criteria.

	BAT Battery	**EM** El. Machine	**PE** Power. Elec.	**CHRG** Charger	**CAB+** Cabin heat.	**CAB-** Cabin cool.	**HOT** ← COP →	**COLD**	**AMBX** Cooler
Heating Cycle (HC)	Active battery heating $T_{Bat} < T_{min}$	Heating with EM $T_{HC} < T_{EM}$ *deficiency HC*	Heating with PE $T_{HC} < T_{PE}$ *deficiency HC*	Heating with CHRG $T_{HC} < T_{CHG}$ *deficiency HC*	Heating (reheat) $Q_{heat} > 0$		heat source (heat pump)		
Cooling Cycle (CC)	Passive Cooling $T_{Bat} > T_{max}$ $T_{Bat} > T_{amb}$	Passive cooling $T_{EM} > T_{max}$	Passive cooling $T_{PE} > T_{max}$	Passive cooling $T_{CHG} > T_{max}$			heat source (A/C)		heat sink
Low Temp. Cycle (LC)	Active cooling $T_{Bat} > T_{max}$ $T_{Bat} < T_{amb}$	Heat pump *deficiency LC*	Heat pump *deficiency LC*	Heat pump *deficiency LC*		Cooling (reheat) $Q_{cool} > 0$		heat sink	
None	Circulation $T_{min} < T_{Bat} < T_{max}$	Circulation $T_{EM} < T_{max}$	Circulation $T_{PE} < T_{max}$	Circulation $T_{CHG} < T_{max}$	Cooling only $Q_{heat} = 0$	Heating only $Q_{cool} = 0$			

Table 4 – Criteria for assigning a component to a certain coolant cycle

Required heating and cooling demand for each component is calculated at every time step from

$$Q_{rem} = mc_{p,m}(T_m - T_{max}) \text{ when } T_m > T_{max}$$

$$Q_{add} = mc_{p,m}(T_m - T_{min}) \text{ when } T_m < T_{min}$$

Dynamic behavior is derived from the steady state thermal management model by only transferring 10% of the required heating or cooling demand. This can results in violation of the temperature constraints (figure 7), but gives results closer to a real system. The primary loop cycle is used for balancing missing heating and cooling demand. The mutual relationship between heating power Q_{hot}, cooling power Q_{cold} and compressor work P_{comp} for the refrigeration cycle at steady state is given by the steady state coefficient of perfomance (COP):

$$Q_{hot} = COP \cdot P_{comp} \text{ and } Q_{cold} = (1 + COP) \cdot P_{comp}$$

$$Q_{hot} = COP/(1 + COP) \cdot Q_{cold}.$$

Therefore, a given cooling Q_{cold} or heating load Q_{hot} (depending on demand) will result in a certain heating or cooling power that must be considered. Primarily the COP depends on the temperature levels between which the refrigeration cycle is operating. For this work a simplified linear relationship is used

$$COP = 3.509 - 0.0494 \cdot T_c + 0.0461 \cdot T_e$$

where T_c is the inlet temperature of the condenser and T_e is the inlet temperature of the evaporator. Coefficients are derived from steady state measurements performed with a comparable refrigeration cycle.

RESULTS

Simulation results for the hot summer scenario are shown in figure 7. The first graph shows the vehicle velocity with time frame for charging and driving. While driving the "ignition on" signal indicates the start and end of the drive cycle. The next three graphs show the ambient conditions with 31°C temperature, 36% relative humidity and 329 W/m^2 sun radiation.

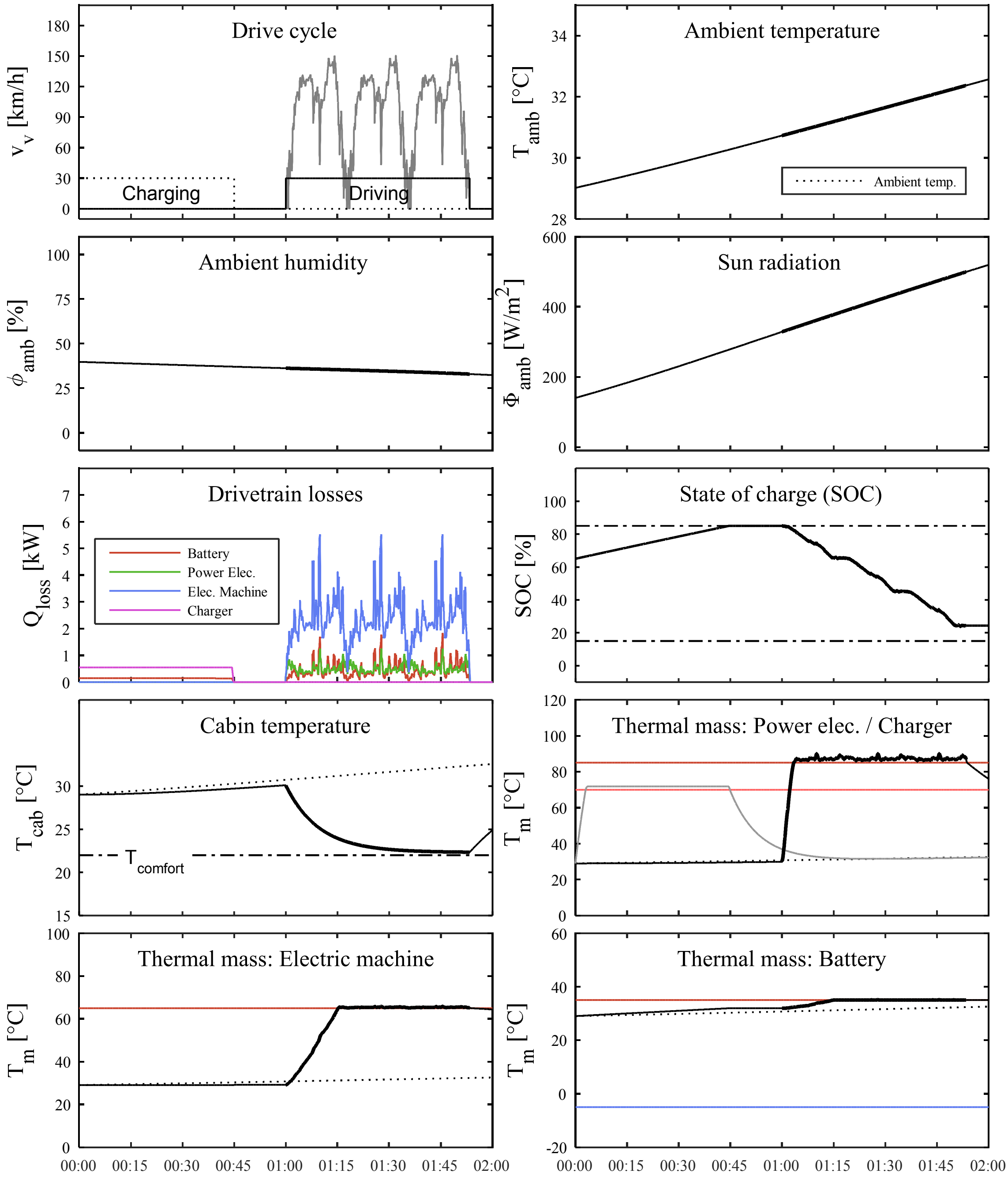


Figure 7 – Simulation results for the summer scenario

Thermal power train losses for battery, power electronics, electric machine and charging device are shown in the next graphs. Next the state of charge (SOC) is shown. After charging the maximum allowed amount of energy is stored in the battery at a SOC of

85%. During the drive cycle the battery is discharged to approximately 24%. Subsequently cabin temperature (together with ambient temperature and comfort temperature) is shown. The last three graphs comprise the temperature of the drivetrain components. Maximum/minimum allowed temperatures are indicated as well. Starting at approximately 25°C all components reach their maximum allowed temperature with power electronics first, then electric machine, then battery. The charging device only heats up during the 45 min charging period. During that period the charging device is cooled most of the time.

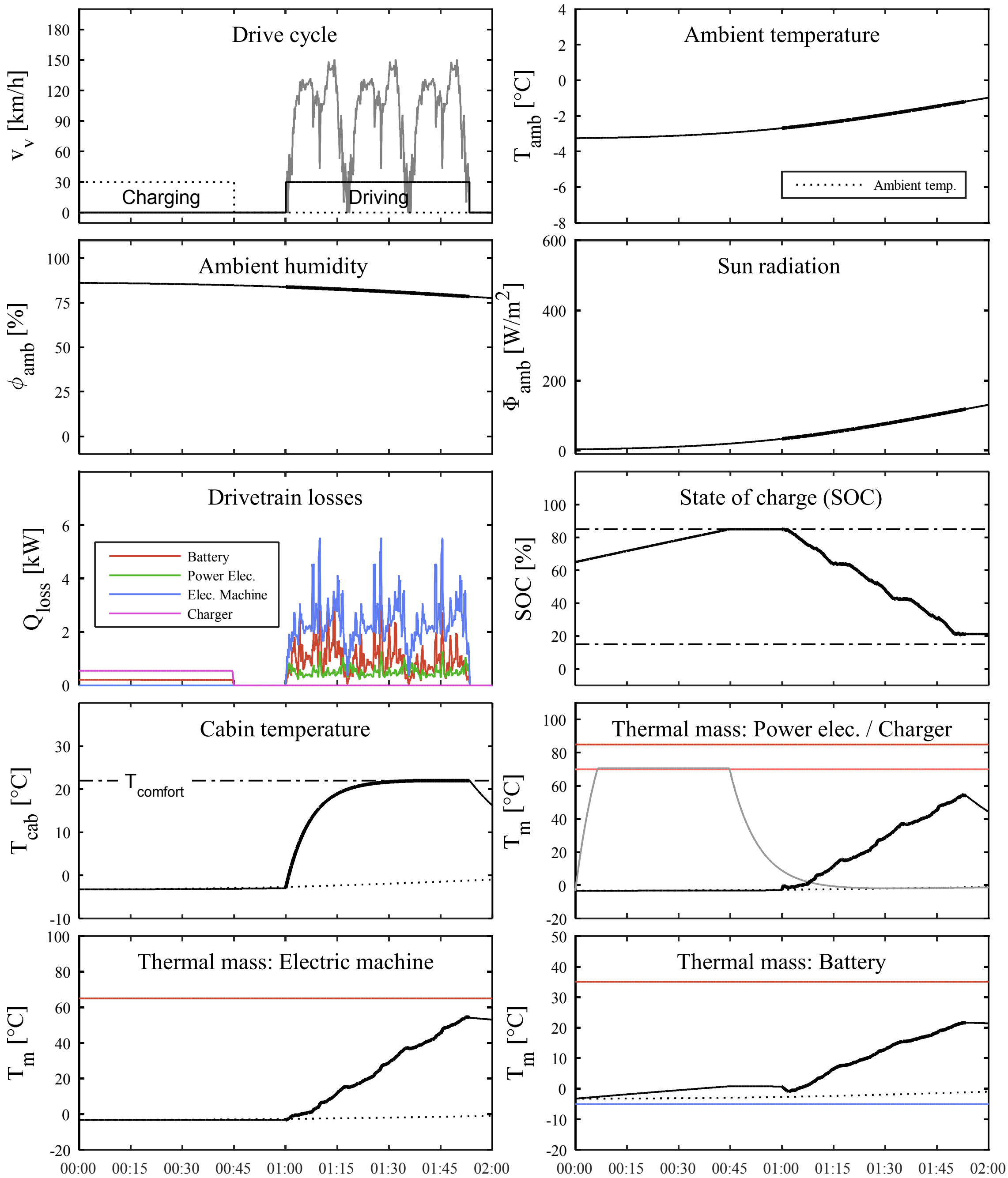


Figure 8 – Simulation results for the winter scenario

When the maximum allowed temperature is reached, the thermal management system draws heat from each component so that component temperature stays at maximum temperature. Simulation results for the cold winter scenario are presented in the same order (Figure 8). It can be seen that component heat up is delayed. This is mainly due to cabin heating requirements that result in a utilization of waste heat either as low temperature heat source for the heat pump or as heat source that can be used directly at high temperature level.

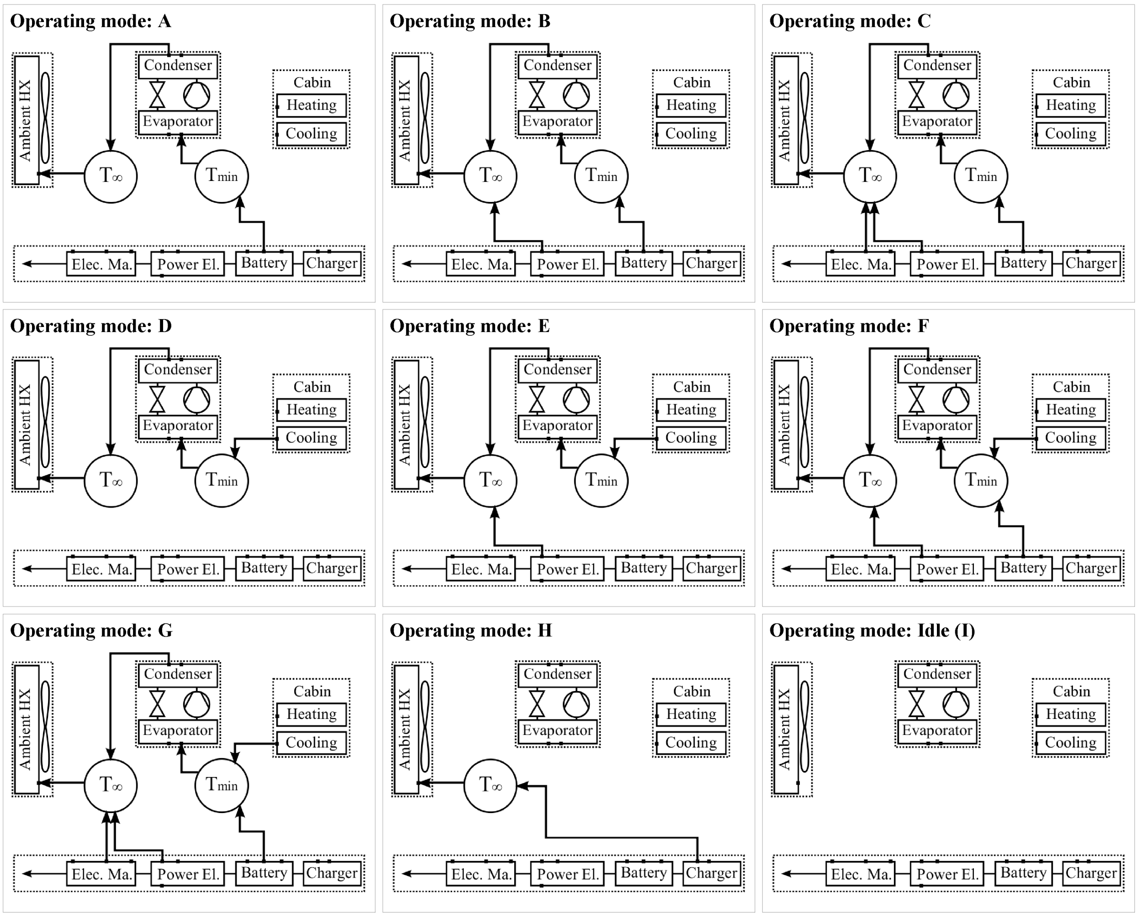


Figure 9 – Overview of operating modes for the summer scenario

Different cooling requirements and component conditions result in different thermal energy flow requirements during the drive cycle. These operating modes are shown in figure 9. Figure 10 shows the active operating mode including cooling of the battery charger between 0:00 and 0:45. The drive cycle itself is highlighted as “ignition on”.

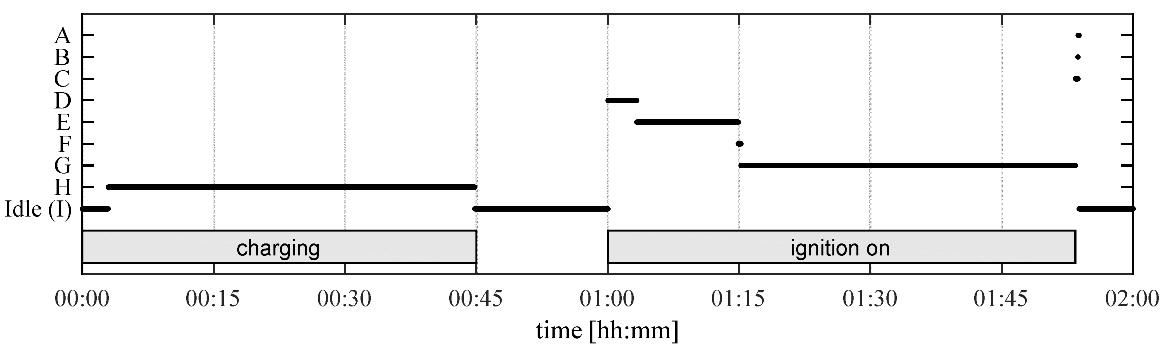


Figure 10 - Operating modes over time summer scenario

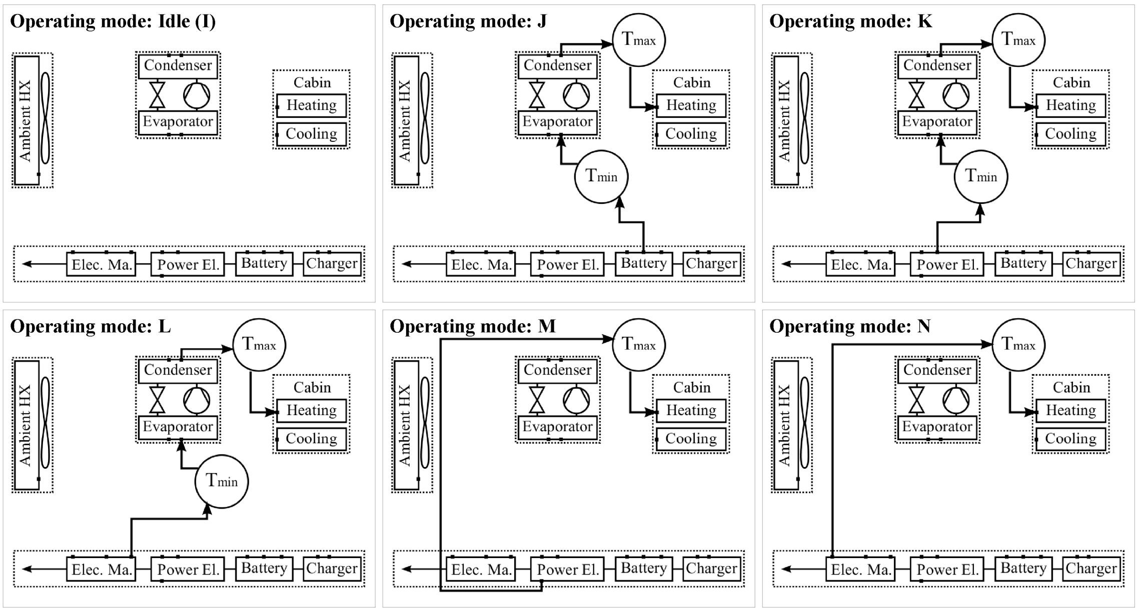


Figure 11 – Overview of operation modes for the winter scenario

Accordingly figure 11 and 12 show required thermal energy flows and operating modes for the winter scenario. Cooling of the battery charging device is included as well (operating mode H). Operating mode I for both scenarios represents an idle case where no thermal management is required.

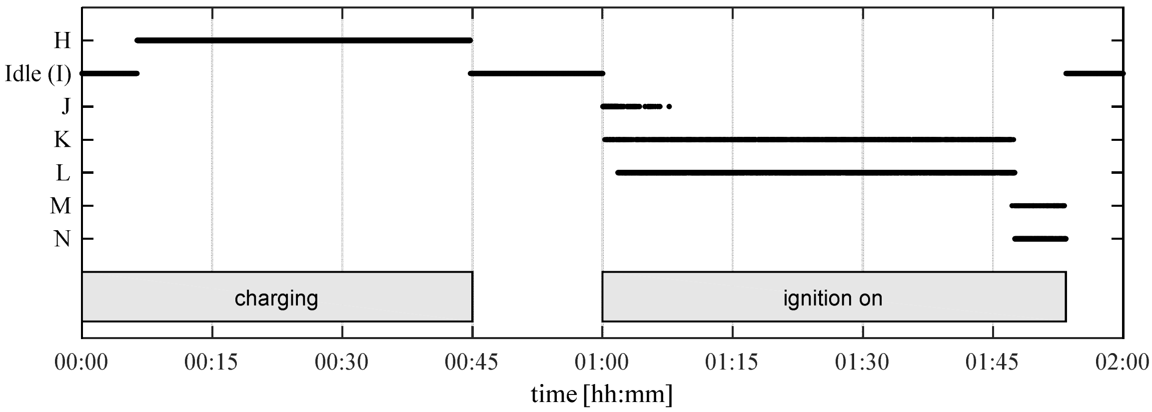


Figure 12 - Operating modes over time winter scenario

The percentage share of each operating mode is given in figure 13 and 14 for both scenarios. As well the total sum of active time for each operating mode is indicated. Idle mode (I) is excluded from the calculation of percentage share.

During the summer drive cycle operating mode A, B, C and F are relatively rare with accumulated active time below 1 min each. Operating mode G is mainly observed. It represents cooling at very high load. Beside active battery and cabin cooling, cooling of power electronics and electric machine is required. During the winter drive cycle different usage of components can be observed. For example power electronics and electric machine are used as low temperature heat source first (K, L) and later as heat source at high temperature (M, N). For both scenarios charger cooling is the most common operating mode due to a long charging period.

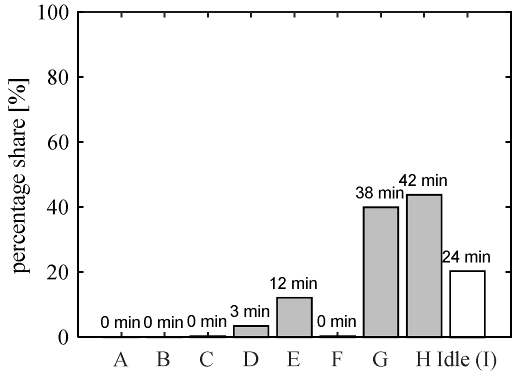


Figure 13 - Histogram summer scenario

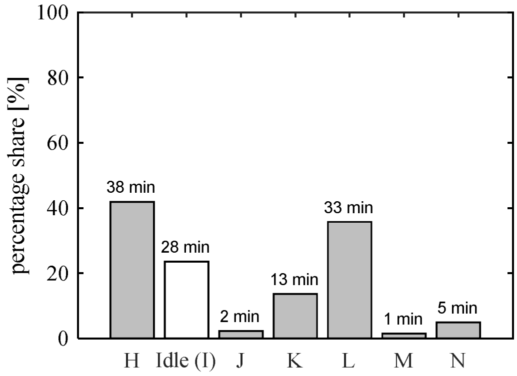


Figure 14 - Histogram winter scenario

DISCUSSION

The results indicate that many topological configurations can be useful and are practical use cases for vehicle thermal management. However, reduction of complexity should be an important goal. Therefore, not every operating mode must or should be implemented. The results clearly show that some operating modes are of higher importance compared to others. Especially operating modes with low frequency of occurrence (e.g. A-C, F, J and M) must be checked for necessity. Also not every energy flow rate configuration must result in an individual piping. For example in most cases (e.g. heat pump mode) power electronics and electric machine can be encapsulated in one reconcilable unit.

CONCLUSION

Based on simplified models drivetrain heat losses and thermal requirements for cabin comfort are estimated. Using first order thermal mass models and typical component temperature constraints, energy surpluses and energy deficits are calculated. A heat flow rate based thermal management strategy then transfers heat between components and three coolant cycles. Simulation results for a realistic winter and summer scenario are presented. Up to 14 useful operating modes are identified and importance is prioritized by calculating the total active time of each operating mode. The results remain without further specification of a heating and cooling system design (including pumps, valves, heat exchangers and electric heaters). Obviously, the provided results depend on the assumptions that are made. Thermal capacity of components, different driving behavior and drive cycles will result in new operating modes.

The main goal of a heating and cooling system should be low cost and low complexity as more topological states result in higher switching time, more complex control schemes and finally more development and testing time. Therefore, more advanced studies should focus on overall system performance when selected operating modes are prohibited in order to reduce system complexity. Furthermore focus of further consideration will be a holistic evaluation including statistical profound driving behavior and weather conditions. Especially the number of operating modes will increase when reheating and more driving scenarios are included. However this will reflect the overall distribution of operating modes more accurately and result in more representative result for overall system energy consumption.

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