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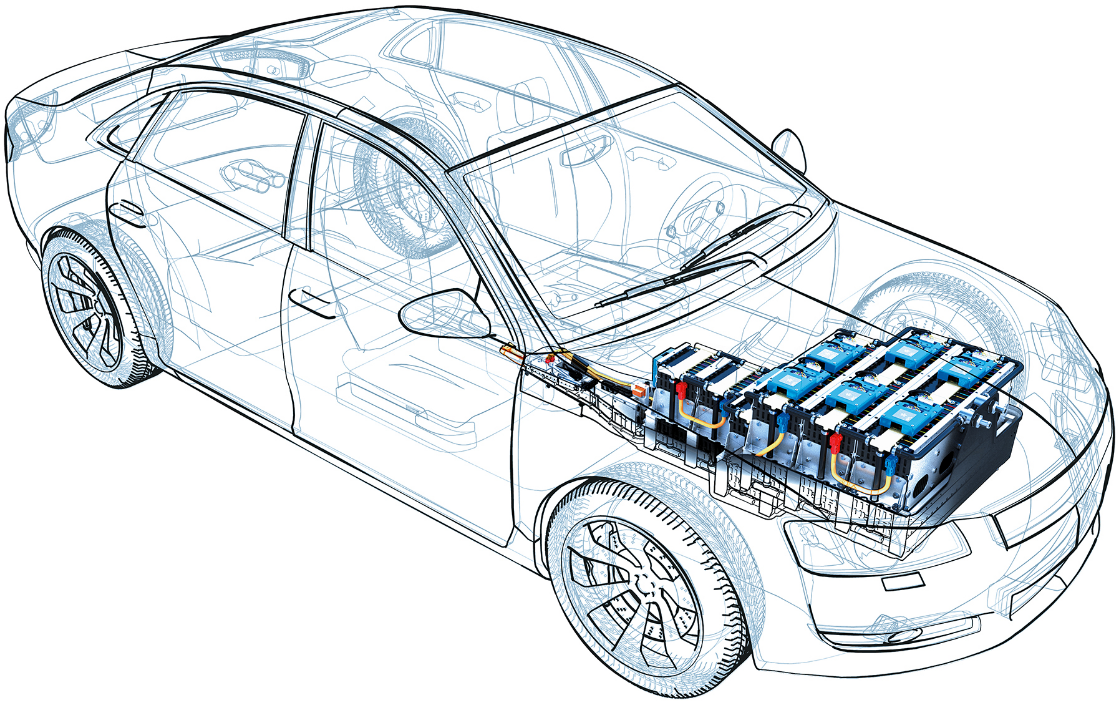

# Dynamic Energy Modeling of Battery Climate System in an Electric Vehicle

Master's thesis in Applied Mechanics

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Varun Venkatesh

Department of Applied Mechanics
CHALMERS UNIVERSITY OF TECHNOLOGY
Gothenburg, Sweden 2017

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cover: High voltage battery of an electric vehicle[23]

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## Abstract

With ever increasing environmental awareness , electric vehicles have gained focus in the current market scenario and are developing fast. Electric vehicles are emission free, have lower energy consumption and maintenance cost. However, there has been constant strive to enhance the range and performance of the electric vehicles as that of conventional combustion vehicles.

The primary challenge associated with an electric vehicle is frequent charging and discharging cycles which causes heat generation within the battery cells. This in turn reduces the performance and lifetime of the battery if not monitored. Thus it is necessary to have a thermal management system to control the thermal behaviour of the battery. However, battery thermal management system also consumes energy from battery affecting the range of the vehicle. Thus it is also necessary to monitor the energy consumption of the thermal management system.

The main aim of this thesis is to model a generic thermal management system for the battery and other auxiliary components of an electric vehicle using a lumped system approach using the tool SIMULINK. The lumped system approach will be benchmarked with the 1D CFD approach by comparing with the model developed using the commercial tool GT-SUITE. Also, the thermal management system model developed in SIMULINK will be integrated with the ÅF's complete vehicle energy model to monitor the energy consumption and range of the vehicle. The model shall be flexible to simulate different types of cooling and heating systems. The model shall be able to accommodate different drive cycles and ambient climatic conditions.

The energy model for thermal management system of battery, propulsion system and passenger compartment has been developed. The model is capable of predicting the thermal behaviour of the system as well as the energy consumption and range of operation. It is flexible for different driving cycles and ambient conditions, which is developed using lumped system analysis. Comparisons with 1D CFD approach was found in good agreement.

Keywords: Electric vehicles, Battery cooling system, Lumped system approach, 1D CFD, SIMULINK, GT-SUITE, Energy model.

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Priyanka Jog & Varun Venkatesh, Gothenburg, June 2017

# List of Symbols and abbreviations

| | |
|---|---|
| $\Delta E_{st}$ | Stored energy of a system [J] |
| $\dot{Q}_{conv}$ | Rate of Convection [W] |
| $h$ | Convective heat transfer coefficient $[W/m^2.K]$ |
| $T_{surf}$ | Surface Temperature [K] |
| $T_{liq}$ | Fluid Temperature [K] |
| $Nu$ | Nusselt number |
| $D$ | Hydraulic diameter [m] |
| $Pr$ | Prandtl number |
| $\mu$ | Dynamic viscosity $[Ns/m^2]$ |
| $C_p$ | Specific heat capacity [J/kg.K] |
| $Re$ | Reynolds number |
| $Q$ | Work done on the system[J] |
| $\rho$ | Density $[kg/m^3]$ |
| $v$ | Flow velocity [m/s] |
| $Q_{rad}$ | Rate of radiation [W] |
| $\epsilon$ | Surface emissivity |
| $\sigma$ | Boltzmann constant $[W/m^2.K^4]$ |
| $T_s$ | Radiation surface temperature [K] |
| $Bi$ | Biot number |
| $L_c$ | Characteristic length [m] |
| $P_{in}$ | Power input to the battery [W] |
| $P_{out}$ | Power output to the battery [W] |
| $W$ | Work done by the system[J] |
| $P_{int}$ | Battery Internal energy [W] |
| $L$ | Battery heat losses [W] |
| $V_{oc}$ | Open circuit voltage of battery [V] |
| $I_{act}$ | Actual current of the battery [A] |
| $V_{act}$ | Actual voltage of the battery [V] |

| | |
|---|---|
| $(UA)_{cb}$ | Overall heat transfer coefficient of coolant inside the battery (W/K) |
| $T_{bat}$ | Temperature of the battery [K] |
| $T_{h1}$ | Temperature at the entry of a component [K] |
| $m_{bat}$ | Mass of the battery [kg] |
| $Cp_{bat}$ | Specific heat capacity of the battery [J/kg K] |
| $\dot{E}_{st}$ | Rate of change of stored energy [W] |
| $Q_{int}$ | internal energy [J] |
| $Q_{in}$ | Energy entering the system [J] |
| $Q_{out}$ | Energy leaving the system [J] |
| $Q_{bat}$ | Energy generated by the battery [J] |
| $T_{h2}$ | Temperature at the exit of a component [K] |
| $\dot{m}_c$ | Coolant mass flow rate [kg/s] |
| $m_c$ | Coolant mass passing through tubes in battery [kg] |
| $h_c$ | Convective heat transfer coefficient of coolant $[W/m^2.K]$ |
| $r_1$ | Coolant tube inner radius [m] |
| $r_2$ | Coolant tube outer radius [m] |
| $\dot{E}_{in}$ | Rate of change of energy entering the system [W] |
| $L_{\text{tube}}$ | Piece Length of coolant tube in battery [m] |
| $n$ | Number of $L_{\text{tube}}$ length tubes |
| $K_{\text{tube}}$ | Thermal conductivity of coolant tube [W/m.K] |
| $K_{\text{bat}}$ | Thermal conductivity of battery [W/m.K] |
| $f$ | Fouling factor |
| $m_{\text{c-rad}}$ | Coolant mass in tubes running through radiator [kg] |
| $(UA)_{\text{c-rad}}$ | Overall heat transfer coefficient of coolant inside the radiator (W/K) |
| $T_{air}$ | Temperature at the ambient air [K] |
| $\dot{m}_{\text{air}}$ | Air mass flow rate [kg/s] |
| $\dot{Q}_{\text{rad}}$ | Cooling capacity of the radiator [W] |
| $\dot{E}_{out}$ | Rate of change of energy leaving the system [W] |
| $\dot{m}_{\text{c-heater}}$ | Coolant mass inside the heater [kg/s] |
| $P_{\text{heater}}$ | Heater rating [W] |
| $\dot{m}_{\text{c-chiller}}$ | Coolant mass inside the chiller [kg/s] |
| $\dot{Q}_{\text{chiller}}$ | Cooling capacity of the chiller [W] |
| $Q_{prop}$ | Energy generated by the propulsion unit [J] |
| $m_{prop}$ | Mass of the propulsion unit [kg] |
| $Cp_{prop}$ | Specific heat capacity of the propulsion unit [J/kg K] |

| | |
|---|---|
| $T_{prop}$ | Temperature of the propulsion unit [K] |
| $T_{req}$ | Required battery temperature [K] |
| $\dot{m}_{\text{p-byp}}$ | Coolant mass flow rate for propulsion bypass [kg/s] |
| $\dot{E}_{gen/diss}$ | Heat losses [W] |
| $\dot{m}_{\text{p-rad}}$ | Coolant mass flow rate for propulsion by radiator [kg/s] |
| $EV$ | Electric Vehicle |
| $HEV$ | Hybrid Electric Vehicle |
| $AC$ | Air conditioned |
| 1D CFD | 1 Dimensional Computational Fluid Dynamics |
| $NiMH$ | Nickel Metal Hydride |
| $SoC$ | State of Charge |
| $ac$ | alternating current |
| $dc$ | direct current |
| $BMU$ | Battery Monitoring Unit |
| $CAE$ | Computer Aided Engineering |
| $BTMS$ | Battery Thermal Management System |
| HVAC | Heat Ventilation & Air Conditioning |
| $COP$ | Co-efficient of Performance |
| $NEDC$ | New European Driving Cycle |
| $EPA75$ | Environmental Protection Agency, a US drive cycle |
| $OEM$ | Original Equipment Manufacturer |

# 1
# Introduction

## 1.1 Background

The global demand for sustainability has made electric vehicles and hybrid electric vehicles alternatives to conventional combustion engine based vehicles. An Electric vehicle is driven by electric motors/generators which are powered by a high voltage battery. The high voltage battery supplies direct current (dc) to the power electronics. The power electronics convert direct current to alternating current (ac) or dc to dc depending on the auxiliary unit.

Batteries are the key to the performance of electric vehicles. There are different kinds of batteries available for automotive vehicles based on their chemical composition, such as lead-acid battery, Nickel-metal hydrite battery and the Lithium-ion battery [1]. Lithium-ion battery has been positioned as one of the best candidate for EV/HEV application [2]. The performance of Li-ion battery strongly relies on the temperature thus making thermal management crucial to safety issues like thermal run-away and over-heating [3]. Lithium ion batteries perform efficiently at temperatures between 25° and 40° and it is desirable to have a temperature difference less that 5° among the modules in a battery pack [4]. Thus thermal management is crucial for a battery to operate at its best efficiency.

Electrical drive motors are the major contributors for the advantages of electric vehicle over conventional vehicle. Electric motors have very high efficiency up to 95% and have excellent torque and output characteristics. Energy losses in electric motors should be given high importance so as to increase efficiency of the electric vehicle. The losses in an electric motor can be significantly higher when it is not operating in a optimum temperature range[14].

In a pure EV, the range of the vehicle is restricted by the energy storage capacity of the battery. The thermal management system consumes power from the battery. Thus it is also crucial to investigate the energy efficiency of the thermal management system for an optimum over-all efficiency of the vehicle.

## 1.2 Objectives

Considering the temperature dependency of the battery and electric motors, the main focus of the thesis is to model a generic thermal management system for the battery and other heat generating components like electric motors, power electronics. It also includes analyzing and evaluating the different configurations of thermal management system for cooling and heating the battery and different control strategies for energy efficiency. The modeling of the system will be done using the lumped system approach in the tool SIMULINK™. This model will be further integrated with the complete vehicle energy model of ÅF, which is a SIMULINK model to analyze the over-all energy efficiency of the vehicle.

The different cooling systems for the battery include passive cooling system and active cooling system. The passive cooling system consists of a radiator as the cooling component in the circuit, whereas an active cooling system consists of an air conditioning circuit. This thesis also investigates the possibility of using the AC system for cooling the battery using a chiller. A final layout of the thermal management system will be chosen so as to provide sufficient cooling and heating for battery, propulsion unit and cabin for different external climatic conditions and drive cycles. The thermal management system will be modeled in a flexible way, where the user will be able to decide the operating state by themselves or the control system decides the operating state based on battery temperature. The model will be analyzed for different drive cycles and initial conditions. The over-all goals of this thesis can be summarized as:

- Modelling different types of cooling and heating systems for the battery, propulsion unit and cabin for an electric vehicle.
- Integrating the thermal management system model with the complete vehicle energy model to investigate over-all efficiency of the vehicle.
- Analyzing and evaluating the model with different configurations of thermal management system and for different external climatic conditions and drive cycles.

## 1.3 Assumptions and Limitations

The main focus of the thesis is to model a generic thermal management system to predict thermal behaviour and energy performance of the battery electric vehicle. The modeling is performed using the lumped system analysis which assumes that the components in the cooling system circuit have uniform material and thermal properties along all the spatial directions. This leads to an assumption that the thermal conductivity of battery and other components are so high that the temperature is equal at all the points and the component can be treated as a lump.

The cooling and heating components in the system are modeled only in terms of energy exchange. That is, the cooling and heating capacity of radiator,chiller and heater are to be given as input to the model as a function of coolant mass flow rate. The chiller and cabin AC were considered to consume power from a single compressor. The refrigerant path and components of the AC loop are not modeled in the thesis.

There will be pressure losses in the coolant circuit and also in the under-hood air path. The pressure loss in the air path is modeled using a simplified thermodynamic approach. But, the pressure loss in the coolant flow is not modeled leading to an assumption of imposed flow through the pump. Thus the energy model of pump is not considering the pressure loss in the coolant circuit.

The thermal model of the battery requires information regarding the architecture and the geometry of the cooling system (tubes) in the battery to calculate the overall heat transfer between battery and the coolant. These parameters were assumed in the current model because of the unavailability of the data.

Modeling the heat generation rate of the battery, electric motor and power electronics were not considered as the scope of this thesis, instead were extracted from ÅF's complete vehicle energy model.

Due to unavailability of the test data, the thermal management system model could not be validated. A simple passive cooling system was modeled using the lumped system analysis in SIMULINK and was compared with the one modeled using 1D CFD approach in GT-SUITE for benchmarking.

# 2

# Theory

This section describes the fundamental concepts about battery, propulsion unit, cooling systems, thermodynamics and heat transfer principles used to model the battery thermal management system.

## 2.1 High Voltage Battery

An electric powertrain consists of mainly four elements: Energy source, power converter, electric motors and mechanical transmission. A high voltage battery pack is the energy source which forms the most important element in an electric vehicle. The battery pack houses several hundred to thousands of cells and are used under dynamic loading [7]. The characteristics of cells play an important role in performance of an electric vehicle. Lithium-ion cells have been popular choice owing to their high power density and charging/discharging efficiency[8]. There are various forms of cell such as prismatic, cylindrical and pouch available for automotive applications. A module consists of several cells connected in series/parallel, packed in a mechanical structure. A battery pack is assembled by connecting multiple modules along with the sensors, controls, battery monitoring unit and thermal management system. Fig 2.1 represents complete Lithium-ion battery pack of Chevrolet Volt.

**Figure 2.1:** Lithium ion battery pack of Chevy Volt[9]

### 2.1.1 Thermal analysis of Li-ion batteries

Temperature range and uniformity are the most important parameters which affect the performance and lifetime of a Lithium-ion battery.

The primary challenge in designing the Lithium-ion battery pack is safety. The charging/discharging process of battery involves various chemical reactions which are highly exothermic in nature. In the hot environment, this might lead to accumulation of heat inside the battery causing significant rise in battery temperature. This will cause hot spots in the battery leading to thermal runaway. Also, Chemical reaction rates vary linearly with respect to temperature. Lower operating temperature leads to decreased rate of reaction causing reduction in the power capacity. Thus it is necessary to maintain the temperature of the battery with in an optimum range.

Additionally, the modules in the battery pack have to operate at a uniform temperature range. Uneven temperature in the pack leads to different charge/discharge behaviour further causing electrically unbalanced modules and reduced performance.

Lithium-ion batteries operate best at temperature range of 25°C to 40°C. At these temperature, they achieve good balance between performance and lifetime. Also, it is desirable to have a temperature distribution of < 5°C between the modules of a battery pack [4].

Fig. 2.2 shows the variation of battery discharge power with temperature. It is evident that the discharge power is very low at sub zero temperatures and it increases with increase in the temperature.

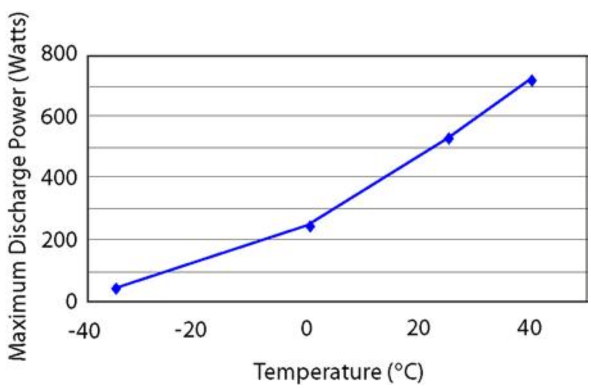



**Figure 2.2:** Maximum discharge power vs temperature for a Panasonic 6.5 Ah NiMH module and 55% SOC [12]

Also, Fig. 2.3 shows the variation of battery life cycle with temperature. The lower the temperature, longer is the life cycle of the battery.

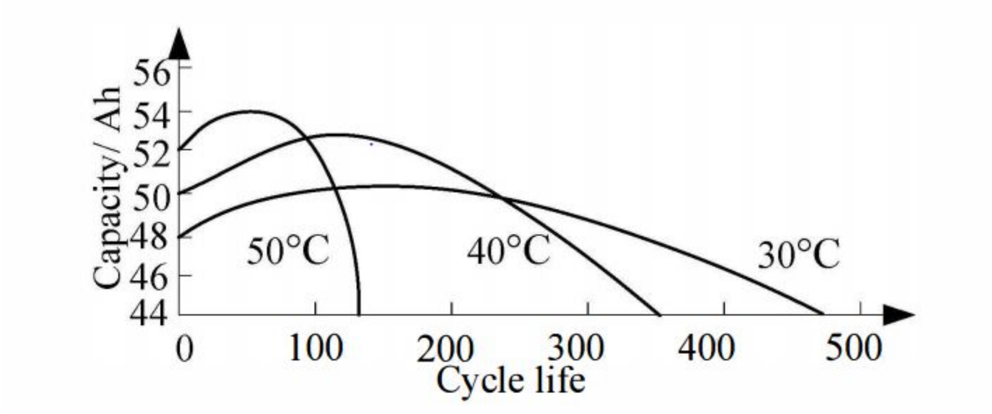



**Figure 2.3:** Cycle life dependence on temperature for lead-acid battery [13]

### 2.1.2 Battery terminology

Some of the important variables which define the condition of the battery are described briefly in this section.

- **State of charge**: It is a parameter which defines the present capacity of the battery as percentage of maximum capacity of the battery.

- **Capacity or nominal capacity**: It is the Amp-hours available when the battery is discharged at particular rate from 100% state of charge to zero.

- **Cycle life**: It represents the number of charging-discharging cycles the battery can experience before it fails to give out certain performance criteria.

- **Open-circuit voltage**: It is the voltage between the terminals of the battery when no load is applied on it. It depends on the battery state of charge.

- **Terminal voltage**: It is the voltage between the terminals of the battery when the load is applied. It depends on state of charge and charge/discharge current.

- **Internal resistance**: It is the resistance within the battery, different for the charging and discharging. Internal resistance depends on state of charge and temperature of battery cell.

## 2.2 Propulsion unit

### 2.2.1 Electric motor/generator

An electric vehicle is mainly driven by electric motors/generators. High voltage battery supplies power to electric motors to drive the vehicle and it also behaves as generator during instances like braking. Energy losses in electric motors are of high importance and temperature is one of the parameter which decides the efficiency of electric motor. Energy losses in an electric motor can be classified as iron losses and winding losses [16]. Winding losses increase with increase in temperature [17] where as Iron losses show a relative decrease with increase in temperature [18]. Thus an electric motor should be maintained at an optimum temperature to operate at highest efficiency.

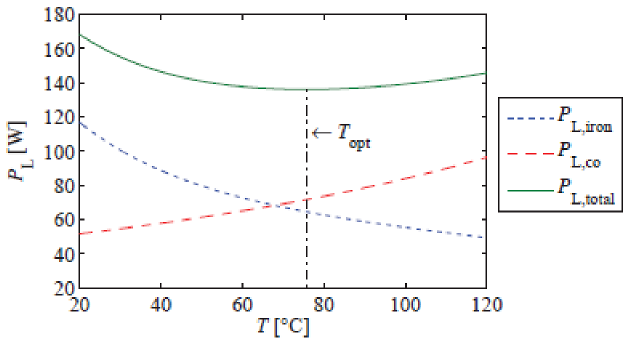



**Figure 2.4:** Temperature dependent losses in an electric motor [16]

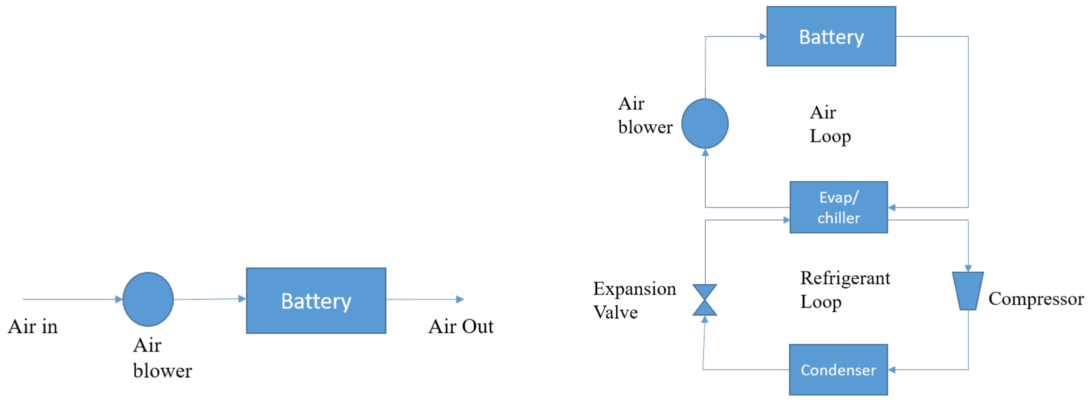
It can be observed from Fig. 2.4 that the optimum temperature where both the iron and winding losses are minimum is in the range of $70 - 80°C$ [16].

### 2.2.2 Power electronics

Power electronics in automotive application employ series of dc-dc and dc-ac converters to drive the electric motor in an electrical vehicle. Today's automotive power electronic converters employ silicon power semiconductors with an upper temperature limit 150°C [19]. In order to establish sufficiently large temperature difference between Si power semiconductor junction and ambience, the temperature of the converter must be lowered to $50 - 70°C$ [20].

## 2.3 Thermal management system

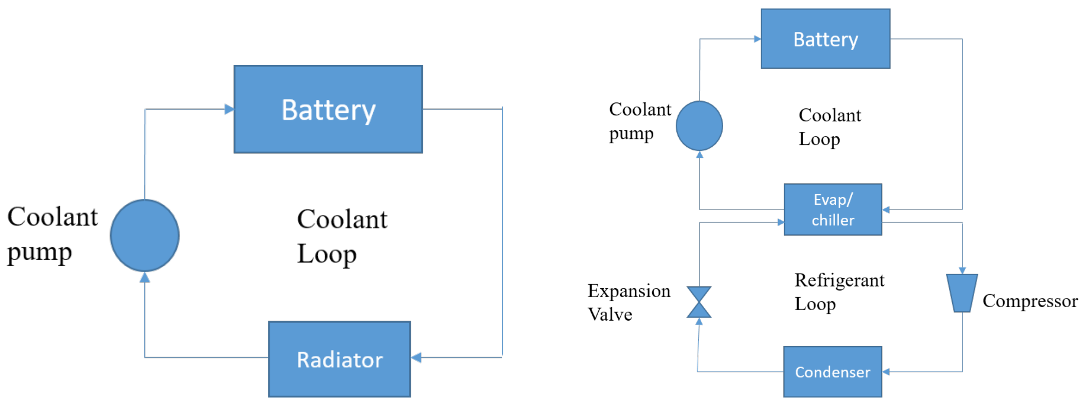
The main goal of the the thermal management system is to maintain the temperature of the battery within certain range. The system may use air for heating, cooling and ventilation or liquid for heating/cooling or the phase changing materials for thermal storage or the combination of methods[12]. The cooling system may be passive cooling system or an active cooling. Passive cooling is the one where ambient air is used for the cooling and active cooling is the one where cooling is provided by additional components and is independent of ambient conditions. Passive cooling system works well in mild climates but an active cooling system is required for extreme climates[15].




**(a)** Passive cooling system **(b)** Active cooling system

**Figure 2.5:** Air cooling systems for battery

Fig. 2.5 represents an example of passive and active air cooling systems for the battery pack. It can be observed that ambient air is directly blowed over the battery pack in a passive cooling system. Active cooling system uses an AC loop for cooling the air making it independent of the ambient conditions.




**(a)** Passive cooling system **(b)** Active cooling system

**Figure 2.6:** Liquid cooling systems for battery

Fig. 2.6 represents an example of passive and active liquid cooling systems for battery. It can be observed that the passive cooling system contains only radiator as the cooling component and it uses ambient air for cooling the coolant. Active cooling system uses an AC loop for cooling the coolant, thus it is independent of ambient conditions.

Air cooling systems are simple and less in cost but less effective than liquid in heat transfer. For same flow rate, liquid has much higher heat transfer coefficient than air. Although liquid cooling is more efficient and it takes less volume, it has some disadvantages. It is more expensive and it involves repair and maintenance costs. Because of the fact that effectiveness of liquid cooling systems outweigh their limitations, liquid cooling systems will appear in future [12].

Hybrid vehicle configurations can be broadly classified as: Series, Parallel, Plug-in, Series-Parallel or split-power hybrid. The degree of hybridization of the car considered in the model is 100%, which means it is a pure electric drive. Electric motor is the only propulsion unit in an electric car. In other configurations, either an engine or a motor or both can be propelling the drive and battery is the energy storage unit. This means there are different requirements of cooling/ heating depending on the type of hybrid vehicle. For an electric vehicle, the main components of a thermal management system can include: radiator, cooling fans for the radiator, electric pump, reserve tank, heater, Chiller, bypass system, flow control valves, hoses and thermostat.

### 2.3.1 Radiator

This is a heat exchanger which is used to cool the coolant that circulates in the battery pack. The cooling medium is air. Air at ambient conditions is allowed to flow over the outer surface of the radiator. The cooling capacity of the radiator can be adjusted by altering the mass flow of the coolant and the air. When the ambient air is very low in temperature, faster cooling takes place and hence involves lesser mass flow rates.

### 2.3.2 Radiator fan

It is a fan which is used to cool the radiator by sucking air towards it. The radiator fan will take care of the air mass flow rate required for effective and efficient cooling. Pressure drops across the grill, radiator and the condenser which account to the major losses across these components are compensated by the fan such that the required mass flow of the air can be maintained.

### 2.3.3 Pump

A pump is used to regulate the coolant mass flow rate throughout the cooling circuit. Centrifugal pump is one of the most common pump that is used in the cooling system applications. The mass flow rate of the coolant can be controlled by varying the speed of the pump. However, there will be pressure losses encountered while maintaining the flow rate of the coolant in the components like hoses, heat exchanging volumes, valves and bends.

### 2.3.4 Reserve tank

When the radiator cap ejects coolant, reserve tank collects the coolant released. This happens when the coolant pressure and temperature are high. Such a situation is seen when the radiator is cooling an engine. In case of an electric vehicle the radiator cools the electrical components only.

### 2.3.5 Heater

This device is used in case of heating requirements. This includes heating of battery up to its working temperature when the ambient air temperature is low. Additionally, the heater may also be used to warm up cabin area. The usage of heater is limited to low ambient temperatures.

### 2.3.6 Chiller

A chiller is a used when the radiator is unable to cool the battery. This is used only when the ambient temperature is more than the required battery temperature. The chiller has an air conditioning unit to provide necessary cooling to the system.

### 2.3.7 Bypass system

A bypass system is used when the coolant must neither be heated nor be cooled, but a uniform flow is to be maintained throughout the circuit. That is to say, the coolant bypasses the heater/ radiator when the temperature of the battery is within the working range. By utilizing the bypass system, some amount of power consumption can be reduced.

### 2.3.8 Flow control valves

The cooling system has different components to cool/heat the battery based on the battery temperature. Hence control valves are required to control the coolant flow direction from one component to another depending on the battery temperature.

### 2.3.9 Hoses

An automotive cooling package can consist of various hoses connecting the components like, radiator hoses, heater hoses etc. It is to be noted that the volume occupied by the coolant in these hoses is not taken into consideration in the model due to lack of packaging and hosing information. However, adding such information can make the model more realistic.

### 2.3.10 Thermostat

Thermostat is used to measure the coolant temperature. Depending on the flow control setting, the thermostat measurement is used to control the flow of coolant either through the heater, radiator, bypass or the chiller unit.

## 2.4 1st Law of thermodynamics

The first law of thermodynamics states that the total energy of a system is conserved. First law of thermodynamics is used to address the ways through which the energy can cross the boundaries of a system. For a closed system, first law of thermodynamics leads to

$$\Delta E_{st} = Q - W \tag{2.1}$$

where, $\Delta E_{st}$ is the energy stored in the system, Q is the work done on the system and W is the work done by the system. Fig. 2.7 shows the schematic representation of the energy flow in a closed system.


**Figure 2.7:** Schematic representation of 1st law for closed system

Considering a control volume at any time instant (t), the rate of thermal energy stored in the control volume is equal to rate of thermal energy entering the control volume minus the rate of thermal energy leaving the control volume plus/minus rate of thermal energy generated/dissipated by the control volume. This statement can be mathematically expressed as

$$\dot{E}_{st} = \dot{E}_{in} - \dot{E}_{out} \pm \dot{E}_{gen/diss} \tag{2.2}$$

The Eqn. 2.2 forms an essential method to solve the heat transfer problems with the proper identification of control volume and control surface. One such application is illustrated in the Fig. 2.8 which shows the energy conservation in a pipe flow with heat addition to it.

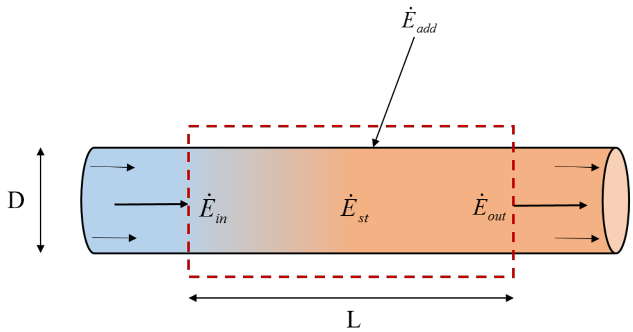



**Figure 2.8:** Energy conservation in a pipe flow with heat addition

## 2.5 Heat transfer

Thermodynamics deals with the end states during the process of energy interaction. The principles of heat transfer provide information regarding the nature of interaction and the rate at which interactions occur. The thermodynamic analysis along with the principles of heat transfer are extensively used to analyse the behaviours of thermal systems. The three modes of heat transfer are conduction, convection and radiation.

### 2.5.1 Conduction

**Conduction** is the transfer of energy from the higher energy particles to the lower energy particles within a substance due to their interactions. The particles may be atoms, molecules and others. Conduction occurs in all the phases of a substance. The rate of heat transfer is governed by Fourier's law. For a 1D substance with temperature distribution of T(x), the rate of heat transfer ($W/m^2$) is given as

$$\dot{Q}_{cond} = -K\frac{dT}{dx} \tag{2.3}$$

Here K is known as thermal conductivity (W/m.K) and it is the property of a material.

### 2.5.2 Convection

**Convection** is a mode of heat transfer where energy is transferred due to the motion of a physical medium over the surface of a solid. Convection comprises of two mechanisms: Energy transfer due to molecular interaction (diffusion) and energy transfer due to bulk motion of the fluid. Convection heat transfer may further be classified into two types based on the type of flow. Convection is forced if the flow is induced from the external sources like fan, pump or winds. Natural convection is the one where the flow is induced due to the buoyancy forces which are due to density change caused by the temperature variations of the fluid. Regardless of the type of convection, the appropriate rate of heat transfer is given as

$$\dot{Q}_{conv} = h.(T_{surf} - T_{liq}) \tag{2.4}$$

where $T_{surf}$ and $T_{liq}$ are the temperatures of the surface and the temperature of the liquid respectively. h is the convective heat transfer coefficient ($W/m^2.K$). It depends on the nature of fluid motion, boundary layer conditions and thermo-fluid properties.

The heat transfer coefficient can be determined by experimental methods or CFD simulations. Many correlations were developed using the dimensionless numbers like Nusselt number, Reynold's number and the Prandtl number based on the type of flow [10].
Nusselt number is the ratio of convective to conductive heat transfer and is expressed as

$$Nu = \frac{h.D}{K} \tag{2.5}$$

where, $h$ is the convective heat transfer of the flow, $D$ is the hydraulic diameter and $K$ is the thermal conductivity of the fluid. $Nu$ can be found using its relationship with Reynolds and Prandtl number.
Prandtl number is the ratio of momentum diffusivity to the thermal diffusivity. It is expressed as

$$Pr = \frac{\mu.C_p}{K} \tag{2.6}$$

where, $\mu$ is the dynamic viscosity ($Ns/m^2$) and $C_p$ is the specific heat capacity ($J/kg.K$).
Reynolds number is the ratio of inertial to viscous force. Reynolds number is expressed as

$$Re = \frac{\rho.v.D}{\mu} \tag{2.7}$$

where, $\rho$ is the density ($kg/m^3$), $v$ is the flow velocity ($m/s$).

### 2.5.3 Radiation

Radiation is the heat energy emitted by any substance at non-zero temperature. The energy transmitted through radiation is by the means of electromagnetic waves. It does not require the presence of material medium to transfer energy through radiation. Although radiation is most significant in solids, it can also be observed in liquids and gases [11]. The rate at which energy is released per unit area is given by Stefan Boltzmann law and is given as

$$Q_{rad} = \epsilon.\sigma.T_s^4 \tag{2.8}$$

Where, $\epsilon$ is the surface emissivity ranged $0 \leq \epsilon \leq 1$, $\sigma$ ($W/m^2.K^4$) is the Stefan-Boltzmann constant and $T_s$ (K) is the absolute temperature of the emitting surface.

## 2.6 Lumped system analysis

Lumped system analysis reduces a thermal system to a single "lump" assuming that the temperature across the lump in all the spatial directions are the same. This analysis helps in simplifying transient thermal analysis. This method reduces partial differential equation to a ordinary differential equation. Assumption of Lumped system method leads to systems conforming to Newton's law of cooling requirement that the heat release rate is directly proportional to temperature difference between the heat source and the fluid.
To determine the validity of the lumped system analysis, dimensionless number called Biot number is used. Biot number is defined as

$$Bi = \frac{h.L_c}{K} \tag{2.9}$$

Where, h is heat transfer coefficient between body and fluid, $L_c$ is the characteristic length and $K$ is the thermal conductivity of the body. Value of Biot number less than 0.1 indicates that thermal conductivity is far higher than convection and temperature gradients inside the body can be ignored. Biot number greater than 0.1 indicates that the assumption of lumped system analysis is not valid.

# 3
# Methods

## 3.1 Layout of thermal management system

The thermal management system should be able to provide sufficient heating and cooling for the battery, propulsion unit and cabin for various climatic conditions and drive cycles. The objective behind the design of the layout was that it has to be generic and flexible. The intention here was not to find the optimum cooling system configuration but rather a simple system that can include all the common components and can be easily adapted to various final configurations.

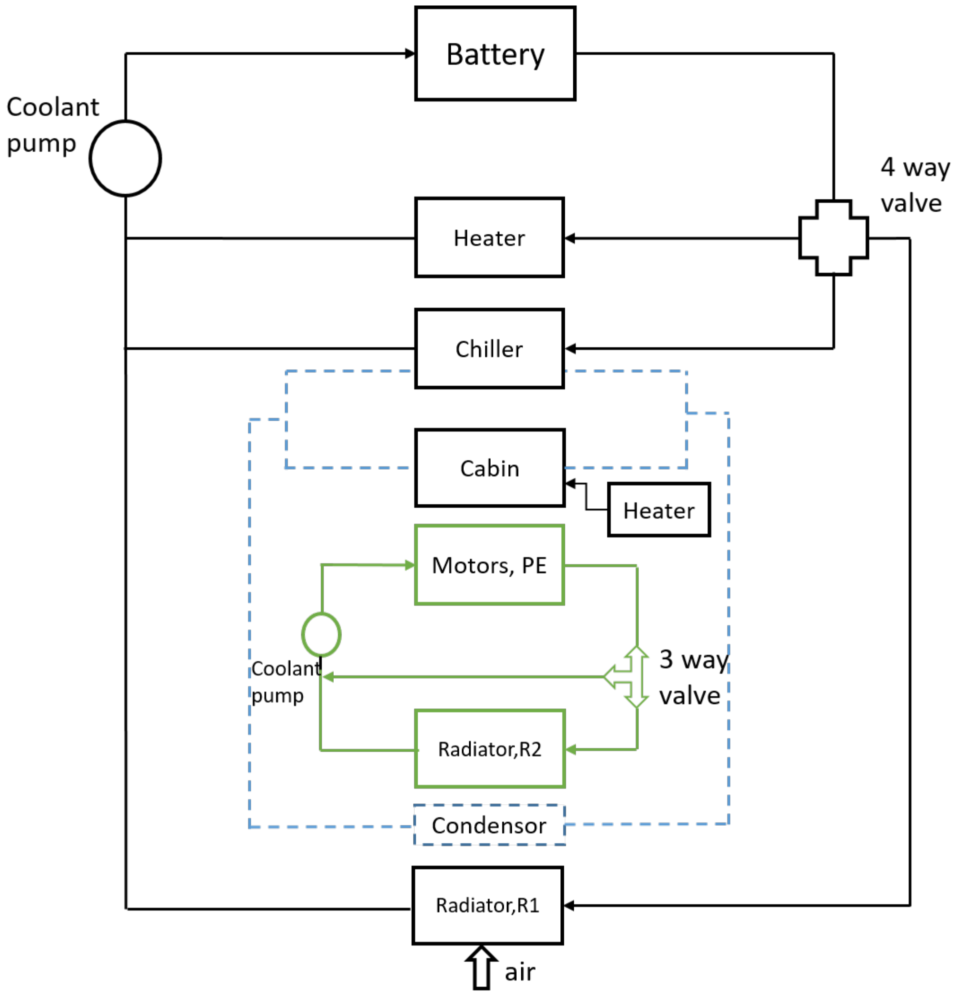



**Figure 3.1:** Layout of the thermal management system

Fig. 3.1 shows the final layout of the thermal management system. Because of the different heating and cooling requirements for battery and propulsion, two different coolant circuits were chosen for the layout of the system. The battery loop (black color in Fig. 3.1) has a radiator which provides passive cooling during mild climate and a chiller providing active cooling during the hot climatic conditions. Additionally, a heater is added in the battery circuit to heat up the battery pack during extreme cold climatic conditions. The propulsion circuit (green color in Fig. 3.1) has a radiator and a bypass line. Because of the high range of optimal operating temperatures, active cooling is not required for the propulsion unit. An additional heater is used to heat the cabin instead of using the same heater used for the battery. One of the main reason for this was to avoid complexity in the circuit. Also, the AC system has an additional evaporator where the cabin air is cooled. The components of the AC loop (dotted blue color in Fig. 3.1) like compressor, expansion valve and condenser have not been modeled in this thermal management system.
It can be observed in the Fig. 3.1 that there is one 4-way valve in the battery circuit and a 3-way valve in the propulsion circuit. The main purpose of the valves are to direct the coolant based on the required operating conditions. The coolant flow is controlled by a control system which decides the path as well as the mass flow rate of the coolant. The control system sends signals to the valves and the pump to choose the coolant path and mass flow rate respectively.

## 3.2 Battery thermal model

The main objective of battery thermal model is to arrive at the battery temperature considering transient heat rate of the battery. In reality, the battery temperature varies from the core to the boundaries in all the three directions. The scope of this thesis does not cater to a three dimensional heat transfer analysis, rather focuses on the lumped system perspective.

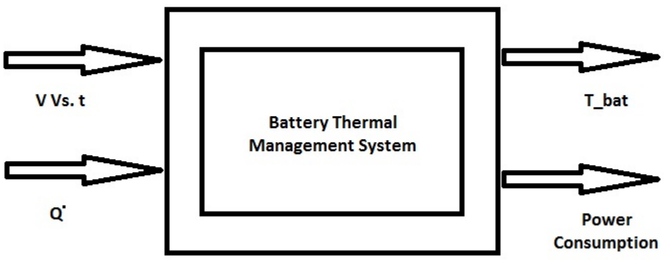



**Figure 3.2:** Schematic representation of the Battery thermal model as a lump system

Considering that the core and the modules in the battery are isothermal due to high thermal conductivity, uniform heat generation is assumed and also that temperature at any point in the battery is same at a given time. When electric current passes through a current carrying conductor, heat is produced. This process is called as Ohmic/Joule heating. During formulating the energy balance equation, Ohmic/Joule heating is considered to be dominant and all the other forms of heating are assumed to be relatively small compared to Joule heating.

The sum of rate of internal energy ($P_{int}$) and the losses dissipated in the form of heat (L) can be accounted using the above input and output power equations as shown below:

$$P_{\text{in}} - P_{\text{out}} = P_{\text{int}} + L \tag{3.1}$$

The maximum power output of the battery is:

$$P_{\text{in}} = V_{\text{oc}} I_{\text{act}} \tag{3.2}$$

Here, $V_{oc}$ is open circuit voltage and $I_{act}$ is the actual current. The actual output power generated by the battery is:

$$P_{\text{out}} = V_{\text{act}} I_{\text{act}} \tag{3.3}$$

where, $V_{act}$ is the actual terminal voltage. Using the Newton's law of cooling, Losses disspated are given as

$$L = (UA)_{cb}(T_{\text{bat}} - T_{\text{h1}}) \tag{3.4}$$

Where, $T_{h1}$ is the temperature of the coolant and

$$P_{\text{int}} = m_{\text{bat}} Cp_{bat} \frac{dT_{\text{bat}}}{dt} \tag{3.5}$$

Applying the first law of thermodynamics, the final energy balance equation for the battery is:

$$P_{\text{in}} - P_{\text{out}} = m_{\text{bat}} Cp_{bat} \frac{dT_{\text{bat}}}{dt} + (UA)_{cb}(T_{\text{bat}} - T_{\text{h1}}) \tag{3.6}$$

Assuming that the battery pack and the piping system are well-insulated, radiation mode of heat transfer has been dismissed and combination of conductive and convective heat transfer is considered. Therefore, the product $UA_{cb}$ is a result of computing the conductive and convective heat transfer.

Knowing the battery properties such as mass ($m_{\text{bat}}$), specific heat capacity ($Cp_{bat}$), overall heat transfer co-efficient between the battery and the coolant ($UA_{cb}$), the temperature of the battery can be calculated.

Solving for time derivative of $T_{\text{bat}}$, re-arranging the Eqn. 3.6:

$$\frac{dT_{\text{bat}}}{dt} = \frac{(P_{\text{in}} - P_{\text{out}}) - (UA)_{cb}(T_{\text{bat}} - T_{\text{h1}})}{m_{\text{bat}} Cp_{bat}} \tag{3.7}$$

Integrating Eqn. 3.7,

$$\int \frac{dT_{\text{bat}}}{dt} dt = \int \Big[ \frac{(P_{\text{in}} - P_{\text{out}}) - (UA)_{cb}(T_{\text{bat}} - T_{\text{h1}})}{m_{\text{bat}} Cp_{bat}} \Big] dt \tag{3.8}$$

$$T_{\text{bat}} = \frac{1}{m_{\text{bat}} Cp_{bat}} \int \Big[ (P_{\text{in}} - P_{\text{out}}) - (UA)_{cb}(T_{\text{bat}} - T_{\text{h1}}) \Big] dt \tag{3.9}$$

The difference of battery input power and the output power i.e. $(P_{\text{in}} - P_{\text{out}})$, is the heat generation rate of the battery and derived from the Battery Monitoring Unit (BMU) in ÅF's complete vehicle energy model. The heat transfer capacity of the surface i.e. $UA_{cb}$ is calculated with respect to the coolant path through the battery module, depending on the type of the cooling system used.

The are many different types of cooling system arrangements inside the battery. In this model, it is assumed that the coolant flows in the coolant tubes which are embedded in the battery back between the cells and the modules. The temperature of the coolant flowing out of the battery is highly influenced by the heat generation rate of the battery. The coolant exit temperature is calculated by formulating an energy equation between the battery and the coolant flowing in it.

By applying the energy balance between the battery pack and the coolant flow, we get

$$\dot{Q}_{int} = \dot{Q}_{in} - \dot{Q}_{out} + \dot{Q}_{bat} \tag{3.10}$$

$$m_c C_{pc} \frac{dT_{h2}}{dt} = \dot{m}_c C_{pc}(T_{h1} - T_{h2}) + (UA)_{cb}(T_{bat} - T_{h1}) \tag{3.11}$$

Integrating and rearranging Eqn. 3.11,

$$T_{h2} = \frac{1}{m_c.C_{pc}} \int \Big[\dot{m}_c C_{pc}(T_{h1} - T_{h2}) + (UA)_{cb}(T_{bat} - T_{h1})\Big].dt \tag{3.12}$$

Here, $m_c$ refers to the mass of coolant in the battery cooling tubes, $C_{pc}$ refers to the specific heat capacity of the coolant and $\dot{m}_c$ refers to the mass flow rate of the coolant. Eqn. 3.9 and 3.12 are the two equations that were used to calculate the battery temperature and the temperature of the coolant at exit of the battery.

### 3.2.1 Calculation of $UA$

In order to develop a thermal model for a particular battery, it is essential to know the battery characteristics like the type, composition, mass, specific heat and also the arrangement of cooling tubes through the modules of the battery. Acquiring data regarding the battery module architecture is tedious as most of such data is proprietary belonging to the OEM. Hence, some assumptions were made where necessary, like geometry of cooling tubes in the battery etc.

Knowing the coolant tube geometry, the conductive and convective heat transfer co-coefficients of the coolant and the dimensionless parameters of the coolant flow like Nusselt number (Nu), Reynolds number (Re) and Prandtl number (Pr), $UA_{cb}$ of the battery can be determined.

For the flow of coolant through the battery, $(UA)_{\text{cb}}$ can be calculated as follows[5]:

$$\frac{1}{(UA)_{cb}} = \frac{1}{h_c(2\pi r_1 L_{\text{tube}} n)} + \frac{\log(r_2/r_1)}{2\pi K_{\text{tube}} L_{\text{tube}} n} + \frac{1}{K_{\text{bat}}(2\pi r_2 L_{\text{tube}})} \tag{3.13}$$

The term $h_c$ is the heat transfer coefficient between the coolant and the coolant tube, $r_1$ and $r_2$ are the inner and outer radiator of the cooling tube. The product of $L$ and $n$ gives the total length of the cooling tube running through the battery module.

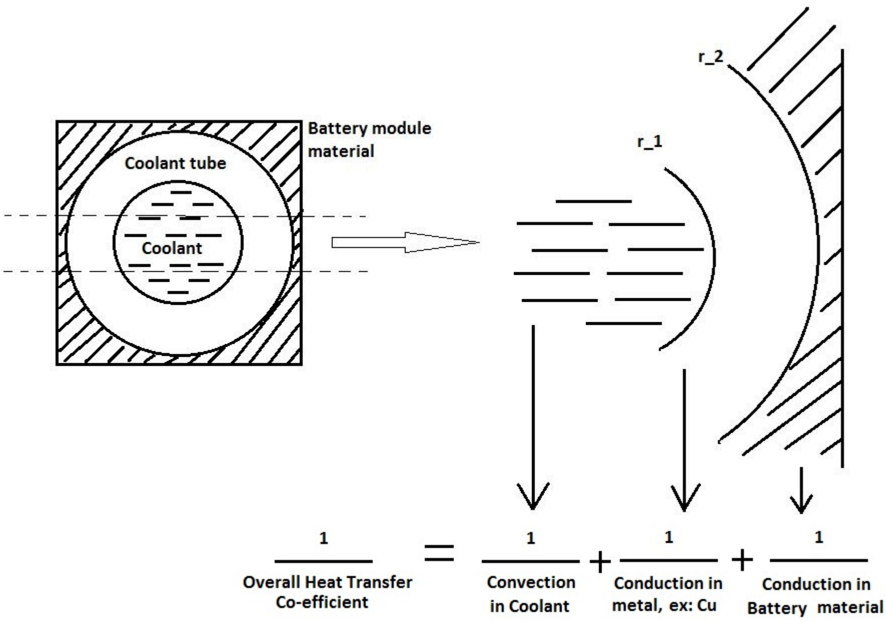



**Figure 3.3:** Schematic representation of thermal resistances involving coolant, cooling tube and the material enclosing the tube

The thermal resistance offered by the cooling tube can be calculated using the second term in the Eqn. 3.13. The third part of the equation indicates contact resistance between the outer surface of the cooling tube and the battery module material. In this case, it is considered that cooling tube is embedded into the module. The contact resistance is neglected, assuming that the contact pressure between the tube and the module material is very high such that $K_{\text{bat}}$ becomes infinity which makes the third term zero[6]. All the terms in thermal resistance equation are constants, except $h_c$. It is determined by using the dimensionless numbers, Nusselt, Reynold's and Prandtl number. The correlation between these numbers is shown below.

For Laminar flow [11]:

$$Nu = 3.66 \tag{3.14}$$

For turbulent flow [21]:

$$Nu = \frac{(f/8)(Re - 1000)Pr}{1 + 1.27(f/8)^{0.5}(Pr^{2/3} - 1)} \tag{3.15}$$

## 3.3 Passive cooling system for the battery

The passive cooling system for the battery includes radiator as the main component for cooling.

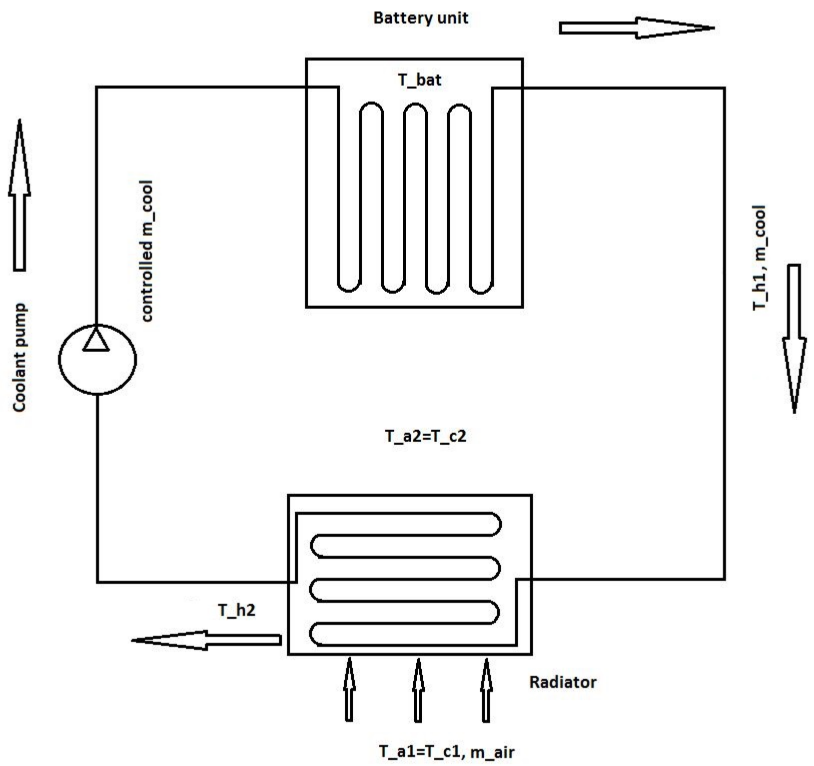



**Figure 3.4:** Schematic layout of passive cooling system

While developing a battery thermal model in SIMULINK, the initial value of the battery temperature is set in order to get the cyclic calculation going. An initial value of the battery temperature is chosen assuming that the battery has not been operated for quite some time so that its temperature can be equated to the ambient temperature. From equation 3.12, the coolant temperature at the exit of the battery module is calculated.

The hot coolant from the battery flows to radiator where it is cooled using the ambient air as the cooling medium. There are two methods of modeling the heat exchange between the coolant and air flowing over the radiator.

**Method 1:** From the previous section, it has been established that the initial conditions for the battery and the cooling fluid temperature are set in the SIMULINK model. An ideal cooling system should be capable of picking up all the heat dissipated by the battery. Referring to the Fig. 3.5, it is clear that at the start of the simulation, $T_{\mathrm{h1}}$, which is the temperature of the coolant passing out of the battery is equal to the battery temperature. This value is fed to the radiator model as the inlet coolant temperature and the temperature of coolant at the exit of the radiator is solved.

The heat picked up by the coolant inside the battery can be expressed as:

$$q = (UA)_{\text{cb}}(T_{\text{bat}} - T_{\text{h2}}) \tag{3.16}$$

For simplification, it can be considered that the radiator is effective enough to cool all the heat picked by the coolant. Applying this idea, the heat transferred by the radiator is equated to $q$ from the Eqn. 3.16.

$$q = (UA)_{\text{crad}}\Delta T_{\text{am}} \tag{3.17}$$

Hence,

$$(UA)_{\text{cb}}(T_{\text{bat}} - T_{\text{h2}}) = (UA)_{\text{crad}}\Delta T_{\text{am}} \tag{3.18}$$

Where, $(UA)_{\text{crad}}$, $(UA)_{\text{cb}}$ are the overall heat transfer co-efficients of the coolant over the area of radiator and battery respectively.

It is to be noted that Arithmetic mean temperature difference (AMTD), $\Delta T_{\text{am}}$, is used instead of Logarithmic mean temperature difference (LMTD), $\Delta T_{\text{lm}}$, because of the type of heat exchanger considered for this model. That is to say, LMTD holds good for parallel and counter flow heat exchangers whereas the radiator is a cross flow heat exchanger. Hence, the use of AMTD.

$$\Delta T_{\text{am}} = \frac{(T_{\text{h1}} + T_{\text{h2}})}{2} - \frac{(T_{\text{c1}} + T_{\text{c2}})}{2} \tag{3.19}$$

Where, $T_{\text{h1}}$, $T_{\text{h2}}$, $T_{\text{c1}}$, $T_{\text{c2}}$ have been indicated in the Fig. 3.4. Assuming that $T_{\text{c1}}$ and $T_{\text{c2}}$ are equal considering the air circulating around the radiator disperses into the atmosphere with a minor increase in the temperature. Therefore, the AMTD equation changes as

$$\Delta T_{\text{am}} = \frac{(T_{\text{h1}} + T_{\text{h2}})}{2} - T_{\text{c1}} \tag{3.20}$$

Re-arranging and solving for $T_{\text{h2}}$,

$$T_{\text{h2}} = 2(\Delta T_{\text{am}} + T_{\text{c1}}) - T_{\text{h1}} \tag{3.21}$$

This $T_{\text{h2}}$ is the new coolant temperature that the cooling system should adjust so as to provide effective cooling of the battery. In case of the radiator, the atmospheric air temperature cannot be controlled. Hence, controlling the mass flow rate of the coolant can deliver the required cooling. This term is adjustable using the heat transfer co-efficient $U$.

**Method 2:** The problem with using a direct heat exchanger equation of energy balance is that it is difficult to allot initial conditions to the variables like coolant temperatures at the entry and exit of the radiator. Additionally, no direct measurement from a sensor or a thermocouple is available to feed the thermal model, which calls for supplying initial values to the variables $T_{\text{bat}}$, $T_{\text{h1}}$ and $T_{\text{h2}}$ (i.e. battery and coolant temperatures at radiator entry and exit respectively).
In such case, it is suitable to use the concept of energy balance between the coolant and the air flowing over the radiator. When the coolant passes through the battery

unit, heat is being added to it and heat is rejected from it when it passes through the radiator. This is important to note as this decides the sign convention of the heat transfer rate throughout the coolant's path. The temperature of the coolant at the exit of the radiator is calculated by solving the equation below:

$$T_{\text{h2}} = \frac{1}{m_{\text{c-rad}} C_{\text{pc}}} \int (\dot{m}_c C_{\text{pc}} (T_{\text{h1}} - T_{\text{h2}}) - (UA)_{\text{c-rad}} (T_{\text{h1}} - T_{\text{air}}) \tag{3.22}$$

where, $T_{\text{h1}}$ and $T_{\text{h2}}$ are the hot coolant entry and exit temperatures across the radiator respectively. The term $m_{\text{c-rad1}}$ is the thermal mass of the coolant inside the radiator.
The $(UA)_{\text{c-rad}}$ is the overall heat transfer co-efficient of the radiator. The overall heat transfer of the radiator is a function of coolant mass flow rate and the air mass flow rate.

$$(UA)_{c-rad} = f(\dot{m}_c, \dot{m}_{air}) \tag{3.23}$$

$UA_{c-rad}$ is calculated directly from the radiator data provided by ÅF, which can be seen in the Appendix.

$$(UA)_{\text{c-rad}} = \frac{\dot{Q}_{\text{rad}}}{ETD} \tag{3.24}$$

The heat transfer for a given value of coolant and air flow rate is given in the radiator data, which is denoted by $\dot{Q}_{\text{rad}}$. ETD is the entering temperature difference i.e. algebraic difference between the coolant entry and exit temperatures across the radiator.
The mathematical model developed in SIMULINK can be illustrated using the Fig. 3.5.

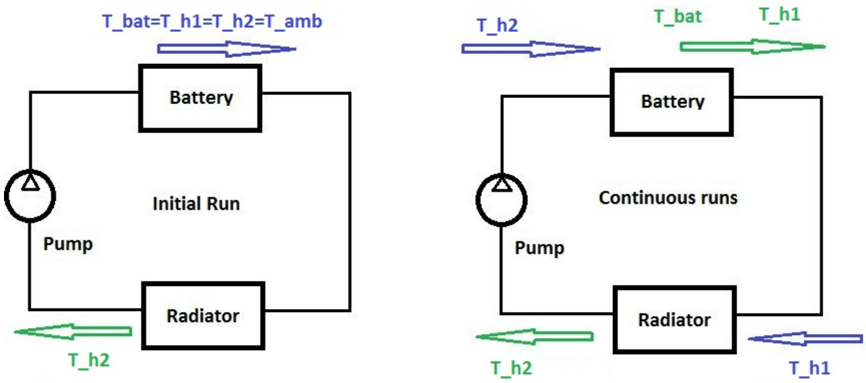



**Figure 3.5:** Schematic representation of Sequence of input acquired for the initial and continuous runs from the second datapoint.

The blue arrows indicate the input and the green arrows represent output from the battery and the radiator.

## 3.4 Inclusion of heater in the battery circuit

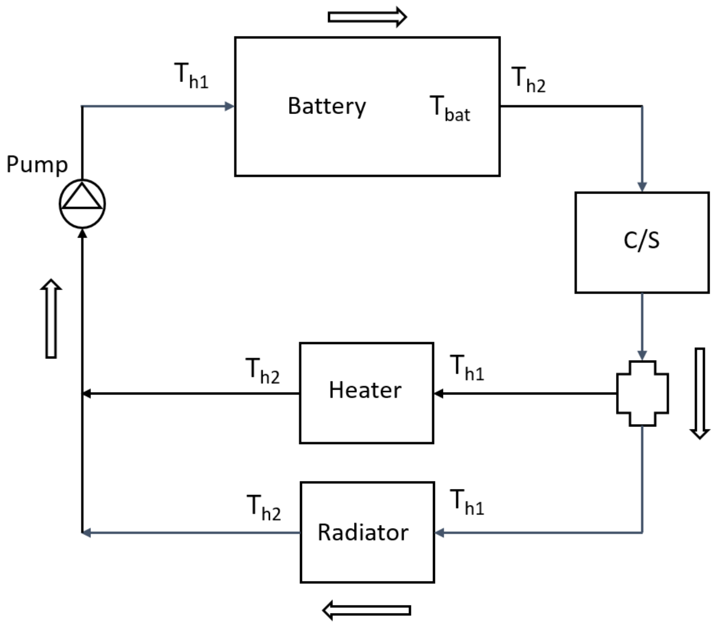



**Figure 3.6:** Passive cooling system with an heater and a bypass loop

When different ambient conditions ranging from cold to hot temperatures are considered, the battery needs both heating and cooling. Hence, a heater is included in the battery circuit in order to maintain the battery temperature between certain range. The temperature of the coolant at the exit of the heater is calculated by applying the first law of thermodynamics, as done previously.
The change in internal energy of the coolant as it flows through the heater is calculated:

$$m_{\text{c-heater}} C_{\text{pc}} \frac{dT_{\text{h2}}}{dt} = P_{\text{heater}} + \dot{m}_c C_{\text{pc}} (T_{\text{h1}} - T_{\text{h2}}) \tag{3.25}$$

Simplifying,

$$T_{\text{h2}} = \frac{1}{m_{\text{c-heater}} C_{\text{pc}}} \int \Big[ P_{\text{heater}} + \dot{m}_c C_{\text{pc}} (T_{\text{h1}} - T_{\text{h2}}) \Big] dt \tag{3.26}$$

$m_{\text{c-heater}}$ is the mass of the coolant inside the heater. The heat addition rate of the heater, $P_{\text{heater}}$ is calculated using the power rating of the heater. The flow control system works in such a way that it directs the coolant flow through heater, bypass (heater off) or the radiator depending on the battery temperature. The control system directs the coolant to flow through the heater circuit with heater on or heater off depending on the battery temperature. If the battery needs cooling, then the coolant path will be changed to the radiator. The conditions for the direction of the coolant flow are specified in the control system.

## 3.5 Active cooling system for the battery

Passive cooling system with heater performs basic function of heating and cooling the battery during cold and mild ambient temperature. But when the ambient temperature is extremely hot, radiator will fail in providing sufficient cooling because of which a chiller is added to the battery thermal management system.

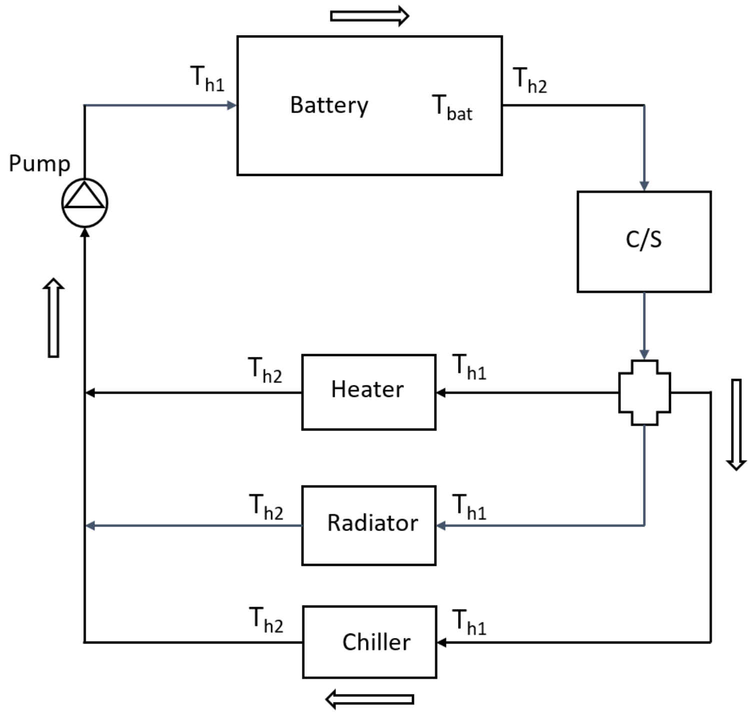



**Figure 3.7:** Active and passive cooling for the battery

When the control system directs the coolant to flow through the chiller, the temperature at the entry of the chiller will be $T_{h1}$. Chiller removes certain amount of heat from the coolant thus reducing the coolant temperature. The temperature of the coolant at the exit of the chiller is calculated by applying the principle of energy balance.

$$m_{\text{c-chiller}} C_{\text{pc}} \frac{dT_{\text{h2}}}{dt} = \dot{m}_c C_{\text{pc}}(T_{\text{h1}} - T_{\text{h2}}) - \dot{Q}_{\text{chiller}} \tag{3.27}$$

Simplifying,

$$T_{\text{h2}} = \frac{1}{m_{\text{c-chiller}} C_{\text{pc}}} \int \left[ (\dot{m}_c C_{\text{pc}}(T_{\text{h1}} - T_{\text{h2}}) - \dot{Q}_{\text{chiller}} \right] dt \tag{3.28}$$

Here, $m_{c-chiller}$ is the mass of the coolant in the chiller, $\dot{m}_c$ is the coolant mass flow rate and $\dot{Q}_{chiller}$ is the heat removal rate of the chiller. The heat removal rate of chiller is a function of the coolant mass flow rate. The heat transfer data of the chiller must be given as input to the model. The chiller heat transfer data for testing the model was provided by ÅF and it can be seen in appendix.

## 3.6 Benchmarking

To validate the approach used for mathematical modeling of the thermal systems, a benchmarking study of a simple passive cooling system was done. The system included the battery, radiator and a coolant pump. The model developed in the SIMULINK was compared with a similar model developed in GT Suite for battery and coolant temperature. GT-SUITE is one of the widely used CAE tool in the automotive industry for a wide range of applications. It uses 1D CFD approach to solve the governing equations of fluid dynamics. Since the current study is limited to heat transfer and pressure drop, a simplified model was developed in GT-SUITE. The procedure used to model the cooling system in GT-SUITE can described in a brief way.
**Battery** is the heat generating source in this system. To model the thermal behaviour of the battery, a template called 'EngineBlock' was used in GT-SUITE. This template replicated a similar kind of cooling architecture used in the battery. In this template, heat (Q) is rejected from the engine to a lumped thermal mass as shown in the Fig.

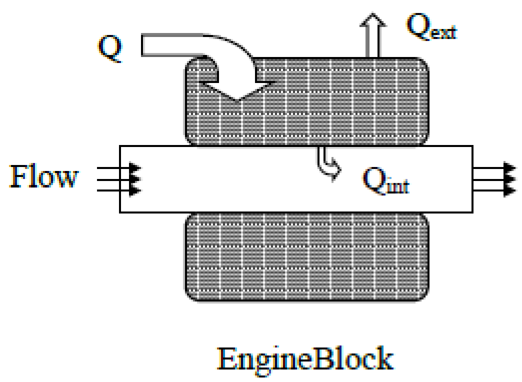



**Figure 3.8:** Template EngineBlock in GT-SUITE [22]

There is heat transfer from the thermal lump to the coolant ($Q_{\text{int}}$) and also to the external ambient ($Q_{\text{ext}}$). The heat transfer between coolant and the engine is calculated based on RLT dependence [22]. This engine block template was used and all the battery properties were provided as input to replicate a similar battery geometry which was considered in SIMULINK. A uniform heat generation rate of $4000W$ was applied to the battery thermal model.

**Radiator**: Heat exchange between two fluids can be modeled using the templates 'HxMaster' and 'HxSlave' in GT-SUITE. Heat exchanger modeling can be done either using lumped system modeling or by discretizing the heat exchanger block. In this case, the heat exchanger was discretized to 3 subvolumes to get better accuracy. GT-SUITE can model parallel flow, counter flow or cross flow heat exchanger. Since the radiator is a cross flow heat exchanger, a single pass unmixed cross flow heat exchanger was chosen in the template. To calculate the heat transfer coefficients between air and the coolant, default heat exchanger file *.hx* available in tutorial was given as input to the 'HeatExchangerSpecs' object.

**Pump**: To model the pump in a simple way, template called 'imposedFlow' was used in GT-SUITE. A particular mass flow rate can be given as input to the model. This model is not capable of using the pump performance curve to calculate the pressure loss since it is a constant mass flow rate flow.

**Pipe**: To complete the circuit of the cooling system, all three components should be connected using the pipes. To model the pipe in GT-SUITE, 'pipeRound' template was used. The template requires main material, geometrical and thermal parameters as input. The material properties includes the type of material and surface finish. The geometrical parameters include inner and outer dia and length of the pipe. Thermal properties like mode of heat transfer can be specified in the template. The initial boundary conditions used to run the model are summarized below.

| | |
|---|---|
| Heat generation rate | 4000 w |
| Initial battery temperature | 293 K |
| Ambient temperature | 293 K |
| Mass flow rate | 1.5 Kg/s |
| Ambient pressure | 101325 pa |

**Table 3.1:** Boundary conditions for GT-SUITE

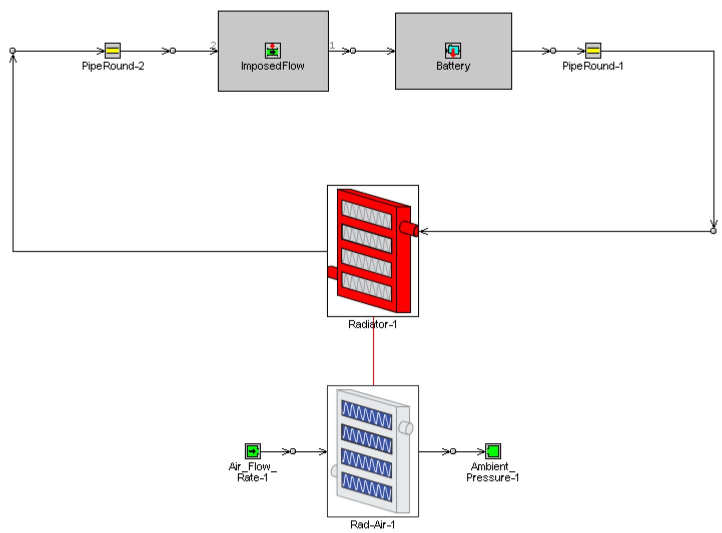



**Figure 3.9:** Schematic arrangement of a simple passive cooling model in GT Suite

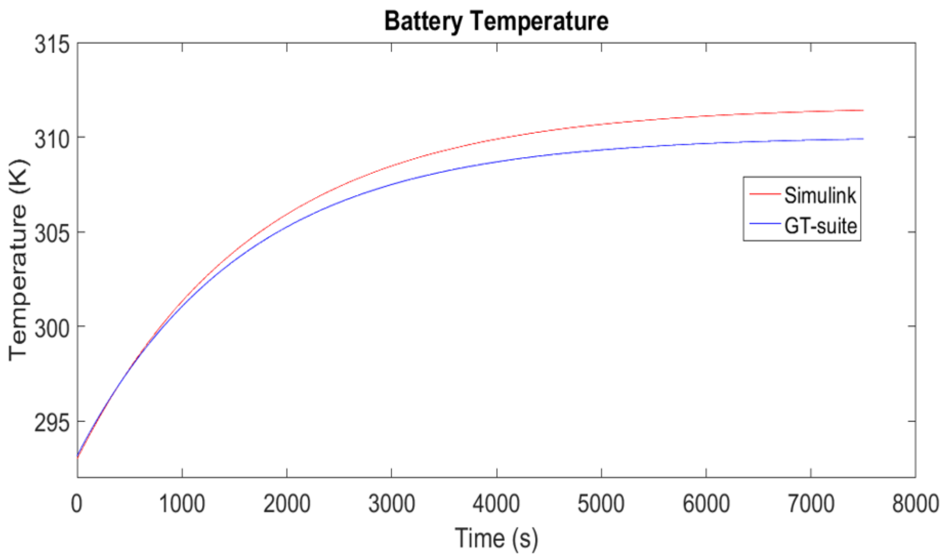



**Figure 3.10:** Comparison of battery temperature between GT-SUITE and SIMULINK

Fig. 3.9 represents the layout of the model built in GT-SUITE. Also, Fig. 3.10 shows the comparison of battery temperature between the model in GT-SUITE and SIMULINK. Since constant heat generation has been applied to battery, it is obvious that the temperature of the battery comes to a steady state after some time. It can be seen from the plot that the battery temperature is approaching a steady state after 6000s. There is a temperature difference of around $2K$ between the temperature given by SIMULINK and GT-SUITE. This is because of the difference in the way the heat transfer coefficient is calculated in the both the codes. Though there is a slight difference, observing the trend of temperature pattern of both the models, it was concluded that the lumped system approach used for modeling the thermal management system agrees well with the 1D CFD approach in GT-SUITE.

## 3.7 Propulsion unit thermal model

To predict the thermal behaviour of the electric motors and power electronics, a thermal model for propulsion unit was modeled. The principle of thermal modeling was similar to that used for the battery. Since, modeling the propulsion is not the major focus of this thesis, a simplified approach was used.

The original ÅF's model is shown in the figure below. This simulink model has been used for analysis of performance and drive quality. Additionally it provides energy / fuel consumption, $CO_2$ emissions and driving range. It consists of different models of propulsion (e.g.: engine, electric motors), battery, transmission, vehicle dynamics (tires, suspension) and the auxiliary and monitoring units. The thermal sources that are under the scope of this thesis, have been marked as 'T'. It can be observed from the Fig. 3.11 that there are four electric motor/generators and a auxiliary electronic unit. The propulsion unit was assumed to be a thermal lump which generates same amount of heat as generated by four motors and power electronics. The unit is assumed to have a cooling architecture with coolant flowing through it.

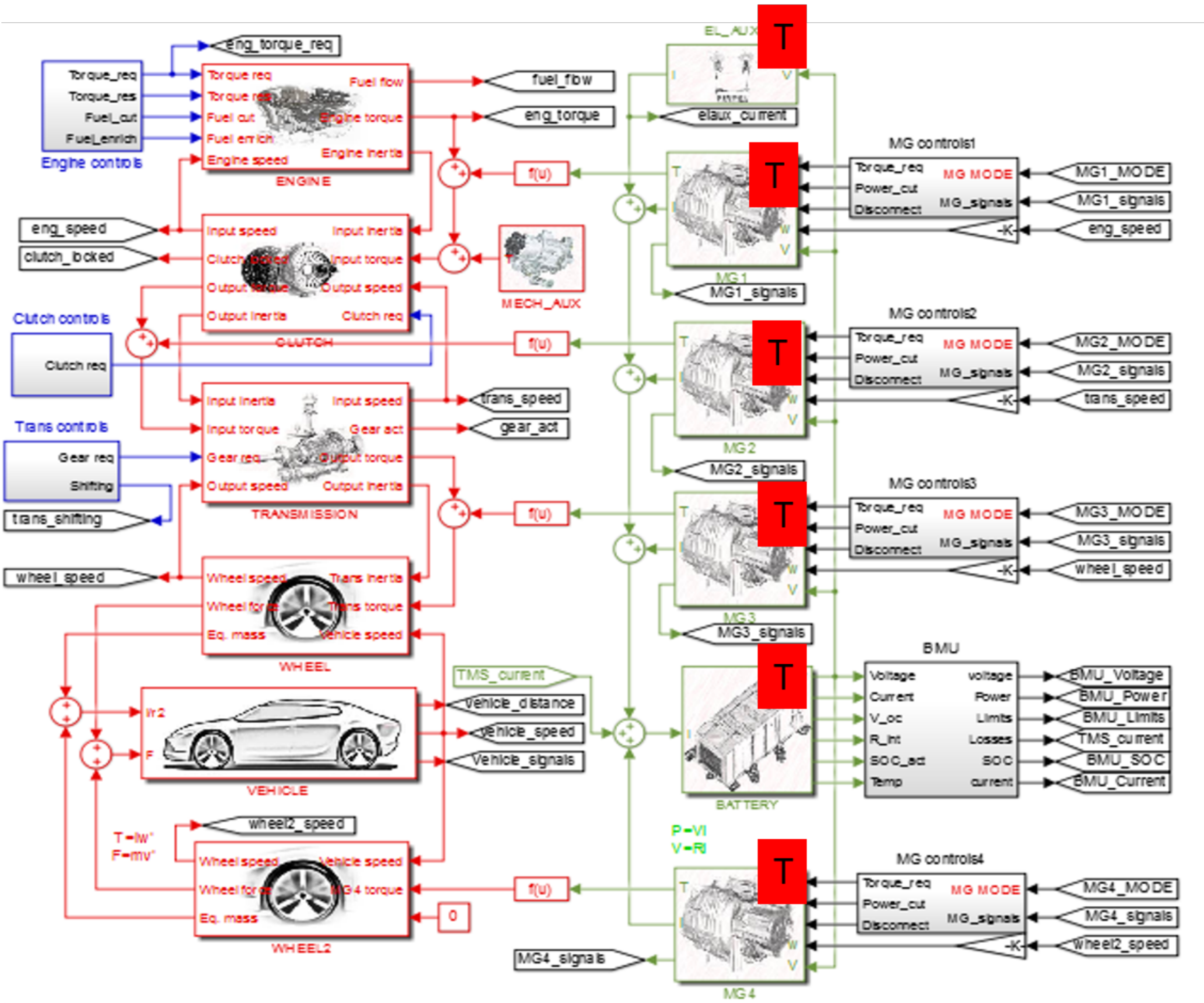



**Figure 3.11:** Thermal sources in the energy model

Using the principle of 1st law of thermodynamics, the energy balance equation for the propulsion unit can be written as

$$\dot{Q}_{int} = \dot{Q}_{gen} - \dot{Q}_{diss} \tag{3.29}$$

$$m_{prop}.Cp_{prop}\frac{dT_{prop}}{dt} = \dot{Q}_{gen} - UA_{prop}(T_{prop} - T_{h1}) \tag{3.30}$$

Integrating and rearranging the Eqn. 3.30,

$$T_{prop} = \frac{1}{m_{prop}Cp_{prop}}.\int \Big[\dot{Q}_{gen} - UA_{prop}(T_{prop} - T_{h1})\Big]dt \tag{3.31}$$

Here, $T_{prop}$ and $T_{h1}$ represents temperature of the propulsion unit and coolant respectively. $m_{prop}$ and $cp_{prop}$ represents the mass and specific heat capacity of the electric motors and power electronics. A mass weighted average value of the specific heat capacity can be considered. $\dot{Q}_{gen}$ represents the sum of heat generated by all four electric motors and power electronics and is given as

$$\dot{Q}_{gen} = \dot{Q}_{mot1} + \dot{Q}_{mot2} + \dot{Q}_{mot3} + \dot{Q}_{mot4} + \dot{Q}_{PE} \tag{3.32}$$

By applying the energy balance between the propulsion unit and the coolant flow, we get

$$\dot{Q}_{int} = \dot{Q}_{in} - \dot{Q}_{out} + \dot{Q}_{prop} \tag{3.33}$$

$$m_c C_{pc} \frac{dT_{h2}}{dt} = \dot{m}_c C_{pc}(T_{h1} - T_{h2}) + UA_{prop}(T_{prop} - T_{h1}) \tag{3.34}$$

Integrating and rearranging Eqn. 3.34,

$$T_{h2} = \frac{1}{m_c C_{pc}} \int \Big[\dot{m}_c C_{pc}(T_{h1} - T_{h2}) + UA_{prop}(T_{prop} - T_{h1})\Big].dt \tag{3.35}$$

Here, $m_c$ and $C_{pc}$ are the mass of the coolant in the propulsion unit and specific heat capacity of the coolant respectively. $\dot{m}_c$ refers to the mass flow rate of the coolant. $UA_{prop}$ represents the over all heat transfer coefficient of the propulsion unit and is calculated by correlation using the battery coefficient. Eqn. 3.31 and Eqn. 3.35 are collectively solved to arrive at propulsion unit temperature and coolant outflow temperature respectively as function of time.

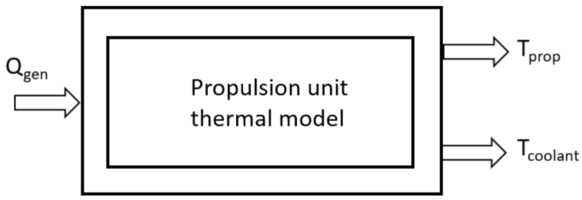



**Figure 3.12:** Outline of the propulsion thermal model

## 3.8 Passive cooling system for propulsion unit

Owing to the high heat generation rate of propulsion unit, only passive cooling system was chosen for the propulsion cooling circuit. Fig. 3.13 shows the thermal management system model for the propulsion unit.

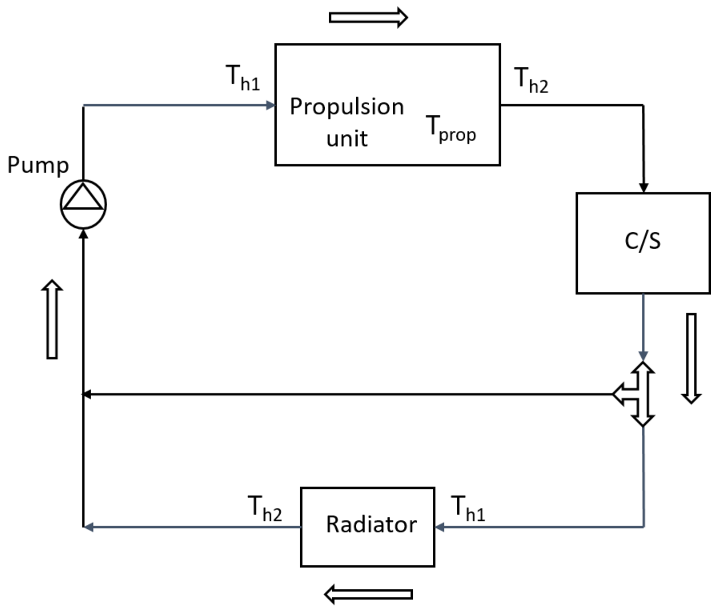



**Figure 3.13:** Passive cooling for propulsion unit

When the coolant flows through the radiator, it enters the radiator with a temperature $T_{h1}$. The radiator takes some amount of heat from the coolant decreasing the coolant temperature. The coolant temperature at the exit of the radiator is calculated by applying the energy balance.

$$m_{\text{c-rad2}} C_{\text{pc}} \frac{dT_{\text{h2}}}{dt} = \dot{m}_c C_{\text{pc}} (T_{\text{h1}} - T_{\text{h2}}) - (UA)_{crad2} (T_{h1} - T_{air}) \tag{3.36}$$

Simplifying,

$$T_{\text{h2}} = \frac{1}{m_{\text{c-rad2}} C_{\text{pc}}} \int \Big[ \dot{m}_c C_{\text{pc}} (T_{\text{h1}} - T_{\text{h2}}) - (UA)_{\text{crad2}} (T_{\text{h1}} - T_{\text{air}}) \Big] dt \tag{3.37}$$

Here, $m_{c-rad2}$ refers to the mass of coolant in the propulsion radiator, $UA_{crad2}$ refers to the over all heat transfer coefficient of the radiator. It is calculated in the similar way as it is calculated for the battery radiator. $\dot{m}_c$ refers to the coolant mass flow rate in the propulsion circuit.
When the temperature of propulsion unit is not above the range of specified critical temperature, the control system directs the coolant to flow in the bypass.

## 3.9 Control strategy

The control strategy for the thermal management of the battery pack and the other heat generating components in an electric vehicle can be employed based on the variable being controlled. The main reason to develop a thermal system is to regulate the battery temperature irrespective of the ambient conditions and also to recover heat if it can be utilized in another component, for example: the cabin interior. The battery works well when it operates within a certain temperature range. The degree of temperature fluctuation of the battery for any given cycle can be controlled depending on the application.

The indicators that are important to design the control system are:

- Battery temperature
- Rate of Heat transfer (mass flow rate)

In order to get a smooth trend of the battery temperature, it is important to regulate the cooling capacity of the radiator/ chiller. This can be done by regulating the mass flow rate of the coolant. Considering the heat transfer equation as shown below

$$Q = \dot{m}_{\text{cool}} C p \delta T \tag{3.38}$$

$\delta T$ is the temperature difference between the battery and the coolant temperatures. It can be seen that heat transfer is mainly dependent on the mass flow rate of coolant.

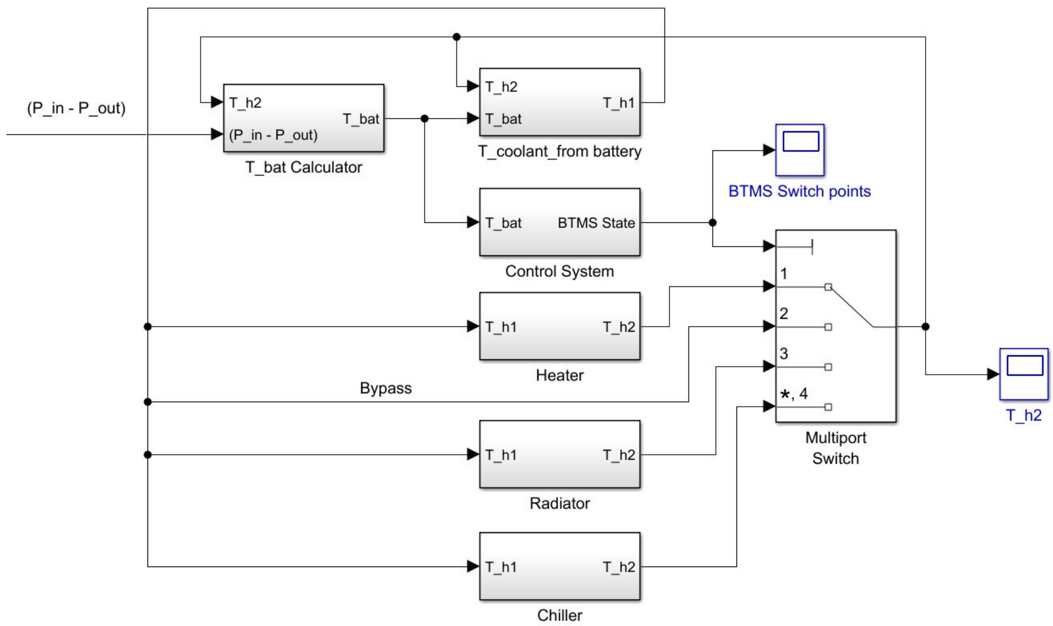



**Figure 3.14:** Overview of Battery Cooling Model developed in Simulink

| Mode | TMS state | Battery temperature (C) | Coolant mass flow rate (kg/s) |
|---|---|---|---|
| Heater ON | 1 | $<T_{req}$-15 | $\dot{m}_{c-heater}$ |
| Bypass | 2 | $T_{req}$-15 to $T_{req}$+2 | $\dot{m}_{c-bypass}$ |
| Radiator ON | 3 | $T_{req}$+2 to $T_{req}$+6 | $\dot{m}_{c-radiator}$ |
| Chiller ON | 4 | $>T_{req}$+6 | $\dot{m}_{c-chiller}$ |

**Table 3.2:** Control strategy for the battery thermal management

There are four components in the SIMULINK model which heat/cool the battery: Heater, bypass, radiator and a chiller. Initially, the control system chooses the coolant path depending on the battery temperature. If the ambient temperature is lower than the lowest working temperature of the battery, then the heater has to be switched on for a duration that is enough to raise the battery temperature

| Mode | TMS state | Propulsion system temperature (C) | Coolant mass flow rate (kg/s) |
|---|---|---|---|
| Bypass | 1 | $<T_{prop.req}$ | $\dot{m}_{c-bypass}$ |
| Radiator ON | 2 | $>T_{prop.req}$ | $\dot{m}_{c-radiator}$ |

**Table 3.3:** Control strategy for the propulsion thermal management

to the recommended value. When the battery operates within its working temperature range, there is no need for either heating or cooling. Hence, the coolant is recirculated into the battery through the bypass system (heater off). As the battery temperature increases past its highest working temperature value, the radiator is switched on. In case of either higher ambient temperatures or when the radiator cannot provide enough cooling, the chiller unit is switched on, which is connected to an air-conditioner loop.

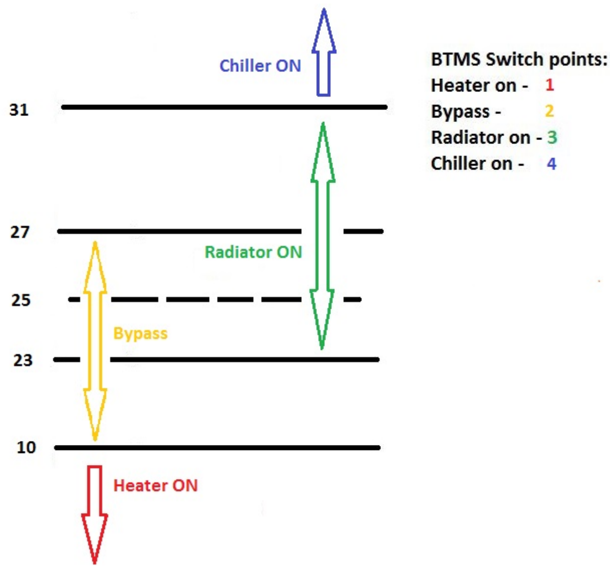



**Figure 3.15:** Designation of BTMS circuits

The aim is to operate at a thermal state which will give better vehicle range, better energy efficiency and at the same time maintain the battery working temperature. To elaborate this concept, consider a scenario where the ambient temperature is 18°C and the lower limit of battery working temperature is 20°C. At the start of the cycle the battery temperature will be 18°C.
There can be two ways of increasing the battery temperature. One method involves using a heater and the other method is allowing the battery to raise its temperature by 2°C on its own. The former method consumes some power, whereas the latter consumes lesser power, however the battery might take extra time to increase by 2°C. The heating capacity required in this example is less. If the ambient temperature is 10°C, the required heating capacity is higher, hence the usage of heater is more desirable when the temperature difference between required battery temperature and the ambient temperature is higher.

Consider a test case: When the ambient temperature is 18°C, the control system allows the coolant to pass through bypass till the battery temperature rises to 27°C. At 27°C, coolant is made to flow through the radiator up to 31°C and then the chiller takes over. Suppose, the ambient temperature is 35°C, the coolant is directly cooled by the chiller unit right from the start of the driving cycle. Since the ambient temperature is high, the difference between coolant and ambient air temperature reduces, hence, the radiator becomes ineffective and the coolant is once again made to flow through the bypass till the battery temperature rises to 27°C.

The generic power consumption of the components used in the thermal circuit assuming the ambient temperature to be 10°C are as shown (exclusively for battery):

$P_{\text{pump}} < P_{\text{radiator-fan}} < P_{\text{heater}} < P_{\text{ac loop}}$

This trend varies when more heat sources like cabin's HVAC, power electronics and electric motor are added.

## 3.10 Air path model

With all the components, connecting pipes, valves, joints etc., arranged in the cooling circuit, in a practical system, there are always many major and minor losses encountered in all the physical components. These losses can influence the reliability of the model. Hence, it is important to identify the different losses and also include them so as to make the model more accurate. The losses observed in the cooling system included in this thesis are mentioned below:

**Radiator**:

To determine the losses occurring in a radiator, pressure drop and air flow across the radiator have to be analyzed. This can be termed as the study of underhood thermal management. In a practical cooling package, there will be more components between the grill and the fan. For example, there could be a air-condition condenser behind the radiator, or there could be two radiators and a charge air cooler one after the other. The method of calculation is similar irrespective of the number of components between the grill and the fan.

For a simplified approach, it is a good idea to only consider a system consisting of grill, the radiator itself and the cooling fan in the same order. The progressive pressure drops after each of these volumes can be determined using a series of calculation. The objective of performing this calculation is to arrive at a required air mass flow rate that can make the radiator deliver the required performance. The air path model is described below.

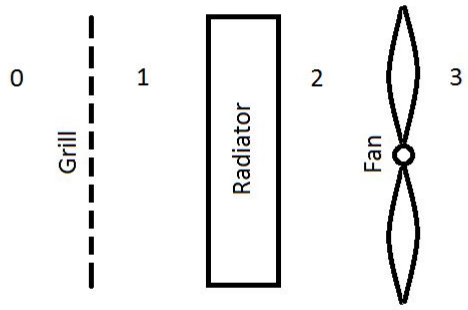



**Figure 3.16:** Simplified Cooling Package

The sections 0, 1, 2, and 3 are considered to determine various air properties like pressure, temperature and density. This will further aid in calculation of the required air mass flow rate in order to get the desired heat transfer from the radiator.
At section 0, ambient properties including static and dynamic air pressure are considered. Pressure drops are simultaneously calculated at section 1 i.e. after the grill, at 2, after the radiator. The pressure rise generated by the fan is calculated at section 3. The air properties like static pressure, temperature are assumed accordingly. The cross-sectional dimension of the cooling package i.e., radiator is assumed and can be altered in the input interface. The fan diameter and its speed has to be prescribed in order to have the required air mass flow rate.
The Ideal gas law is used to calculate the air density. It states that

$$\rho_0 = \frac{P_{\text{static}}}{R.(T_0 + 273)} \tag{3.39}$$

R is the Universal gas constant of air, which is equal to 287 $Jkg^{-1}K^{-1}$. The total pressure at 0, is the stagnation pressure when the vehicle is moving at velocity, $v$ is calculated as

$$P_0 = P_{\text{tot}} = P_{\text{static}} + P_{\text{dynamic}} P_{\text{dynamic}} = \frac{1}{2}\rho.v^2 \tag{3.40}$$

It is assumed that there is no density change after the grill. Therefore,

$$\rho_1 = \rho_0 \tag{3.41}$$

To calculate the pressure drop after the air passes through the grill, the change in air velocity is to be determined. Also, it is assumed that this drop can be about half of the dynamic pressure at section 0.

$$v_g = \frac{\dot{m}_{\text{air}}}{\rho_0 A_g} \tag{3.42}$$

Likewise, the grill loss factor is shown below:

$$\Delta P_1 = \frac{1}{2} P_{\text{dynamic}} \tag{3.43}$$

The pressure drop across the radiator is given by the equation that is suitable for both laminar and turbulent flow conditions.

$$\Delta P_2 = \frac{\dot{m}_{\mathrm{air}}^2 K_2}{\rho_1} + \frac{\dot{m}_{\mathrm{air}} K_1}{\rho_1} + K_0 \tag{3.44}$$

The constants $K_2, K_1, K_0$ are specific to the heat exchanger i.e., radiator installed in the vehicle.
The air mass flow rate is the required variable that is to be calculated in order to ensure the required cooling output from the radiator. Initially, it is assumed to facilitate running of the simulation. At the end of the calculation in SIMULINK, the mass flow rate when a particular condition of air pressure is achieved, is set as the air mass flow rate in the cooling package. This condition is described below.
The air density changes after it has passed through the radiator and the fan, due to the pressure and temperature changes. The pressure balancing equation for the cooling package can be expressed as:

$$P_3 = P_0 - \Delta P_1 - \Delta P_2 + \Delta P_3 \tag{3.45}$$

The fan is said to have worked effectively if this final pressure at the back of the fan is equal to the initial static pressure of air at section 0. Hence ideally,

$$P_3 - P_0 = 0 \tag{3.46}$$

However, there can be other pressure losses that have not been accounted for in the model, hence it is safe to say that this difference should be more than a certain pressure value.

## 3.11 Energy modeling

To account the energy consumption of the thermal management system, it is important to model the energy consumption of all the components in the thermal management system. The energy consuming components in the thermal management circuit are

- Battery and propulsion circuit pumps
- Battery heater
- Chiller
- Cabin heater and AC
- Radiator fan

**Pump:** It is assumed that an electric pump is used for pumping the coolant in the circuit. The power output of a pump is given as [24]

$$P_{out} = \rho g H Q \tag{3.47}$$

Here, $\rho$ is the density of the coolant ($kg/m^3$), g is acceleration due to gravity ($m^2/s$), H is the head generated by the pump ($m$) and Q is the volumetric mass flow rate of the coolant ($m^3/s$).

$$P_{out} = Q\Delta P \tag{3.48}$$

Where, $\Delta P$ is the pressure rise of the pump and is expressed as $\Delta P = \rho g H$.
The final energy consumption of the pump is expressed as

$$P_{in} = \frac{P_{out}}{\eta_{pump}} \tag{3.49}$$

$\eta_{pump}$ represents the efficiency of the pump. The volumetric flow rate of the coolant is a known parameter in Eqn. 3.48. The pressure rise of the pump depends on the type of pump used and it is characteristic variable of specific type of the pump. A performance curve of the pump will have the variation of pressure rise of pump as a function of volumetric flow rate. The performance curve of a particular pump was fed as input to the model. The performance curve of the pump used for testing the model can be seen in appendix.

**Heater:** A simplified approach is used to model the energy consumption of the heater. It is assumed that an electric resistance heater is used for heating the battery and the cabin. In case of an electric resistance heater, all the power input is used as the useful work done i.e. the efficiency of heater is 100 %. It was assumed that the heater has a constant power rating and the power rating of the heater was given as input parameter to the model.

**Chiller:** The energy efficiency of the chiller is usually expressed using the coefficient of performance (COP). It is expressed as

$$COP = \frac{Q_{chiller}}{P_{in}} \tag{3.50}$$

Where, $Q_{chiller}$ is the heat transfer rate of the chiller and $P_{in}$ the power input to the chiller unit.

$$P_{in} = \frac{Q_{chiller}}{COP} \tag{3.51}$$

The heat transfer rate chiller ($Q_{chiller}$) and the COP were given as input to the energy model of the chiller.

**Cabin heater and AC:** The cabin heater and AC are among the major energy consuming components. As it was discussed that the AC loop modeling is not considered in this thesis, the cabin heater and AC are modeled only in terms of energy consumption. The heating or cooling power required to maintain the cabin climatic conditions depend on ambient temperature as well as the vehicle speed. The data regarding cabin energy consumption was provided by ÅF in terms of energy consumption as a function of ambient temperature and the vehicle speed. This data was provided as input to the model for the purpose of testing.

**Fan:** The energy consumption of a fan can be calculated using the same principle as that of the pump. The power output of a fan is expressed as

$$P_{out} = Q\Delta P \tag{3.52}$$

Where, $Q$ is the volumetric flow rate of air and $\Delta P$ is the pressure rise of the fan. The total energy consumption of fan is calculated as

$$P_{in} = \frac{P_{out}}{\eta_{fan}} \tag{3.53}$$

$\eta_{fan}$ represents the efficiency of the fan. The pressure rise of fan is function of volumetric flow rate of air and it is a variable that depends on specific type of fan. The performance curve of fan was used to determine the pressure rise as function of flow rate of air. The performance curve that was used for testing the model can be seen in appendix.

# 4

# Results

This chapter describes the results obtained from testing the thermal management system model for two drive cycles and various ambient conditions. The results of the model can be broadly classified to two types, Thermal perspective and energy perspective. Thermal results include temperature, mass flow rate and heat transfer rate. Energy results include power consumption and vehicle range.

The results of NEDC cycle for mild, hot and cold climatic conditions are presented at the beginning of the chapter. A comparison study of the two drive cycles in terms of energy consumption and drive cycle are presented in the later part of the chapter. It is to be noted that the battery properties like weight, make etc., and coolant and coolant tube properties are kept the same for all the simulations. The initial conditions for running the integrated thermal management model is shown below:

| | |
|---|---|
| Max battery capacity | 100 Ah |
| Initial SOC | 90 % |
| Final SOC | 1 % |

**Table 4.1:** Battery initial conditions

## 4.1 NEDC driving cycle

Fig. 4.1, 4.2 and 4.3 show the plot of vehicle speed for NEDC cycle, heat generation rate of battery and propulsion unit.

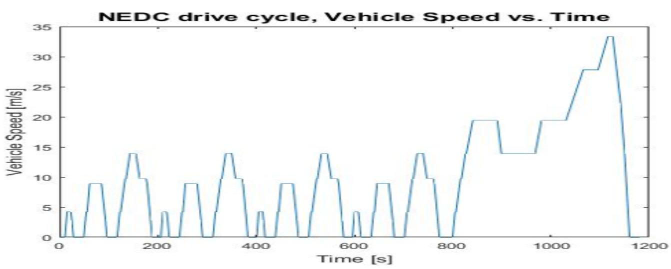



**Figure 4.1:** Vehicle speed

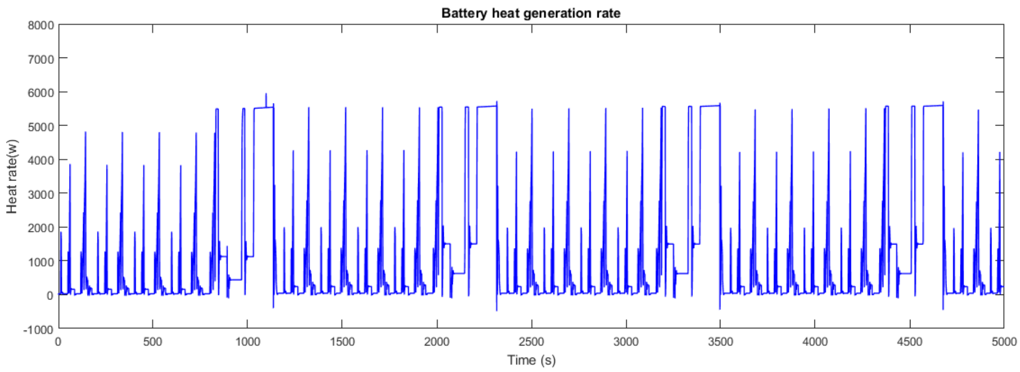



**Figure 4.2:** Heat generation rate of the battery

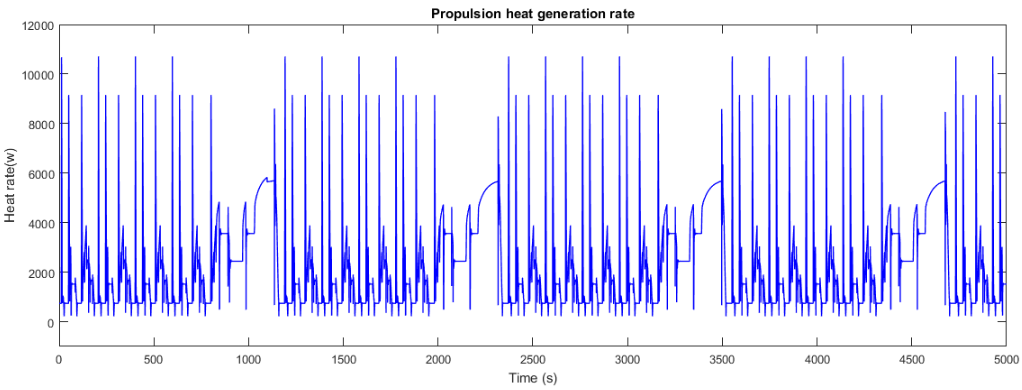



**Figure 4.3:** Heat generation of the propulsion unit

### 4.1.1 Mild climate

For the mild ambient climate case, a temperature of $25°C$ was chosen as the initial temperature for the battery, propulsion unit and the ambient climate.

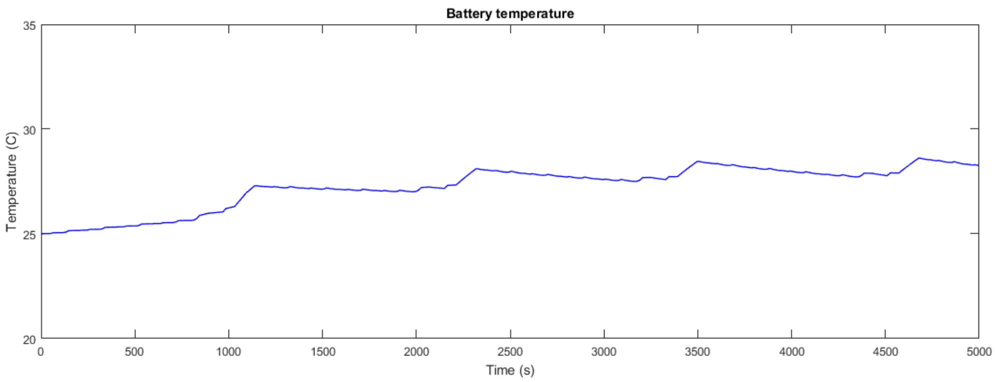



**Figure 4.4:** Battery temperature

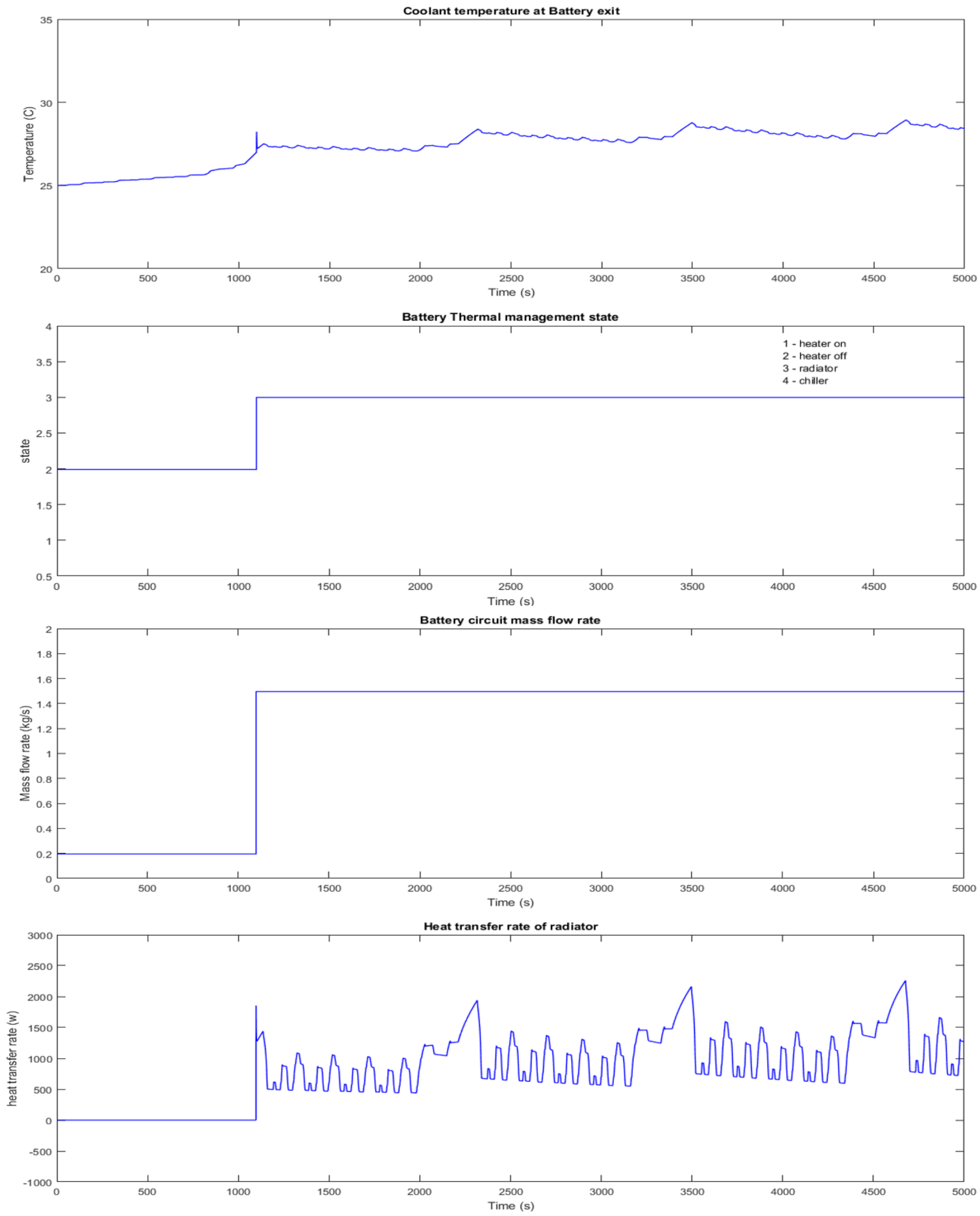



**Figure 4.5:** plots of thermal results

Fig. 4.4 and Fig. 4.5 show the plot of battery temperature and coolant temperature at the exit of battery. Fig. 4.5 represents the state of the thermal management system. It can be observed that the state is 1 till 1200s and then state goes to 2. It means that the coolant is flowing through the bypass line where there is no heat addition or removal and then it's path changes to radiator. It can be seen in Fig. 4.5 that there is no heat transfer happening until 1200s and then the heat transfer rate increases as the coolant starts flowing through radiator. Also, it can be seen that there is change in the mass flow rate of the coolant when the state of the thermal management changes from bypass to radiator. This mass flow rate is controlled through the control system.

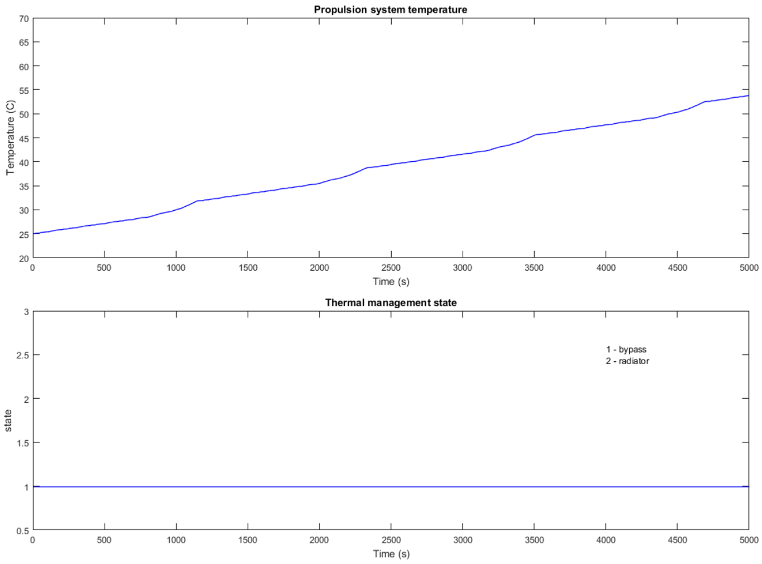



**Figure 4.6:** propulsion unit temperature and state

Fig. 4.6 shows the plot of propulsion unit temperature and thermal management state. It can be observed that the temperature of the propulsion unit is increasing, this is because of the self heat up of the motors and power electronics. As the upper limit temperature of $65°C$ was given to the control system to switch the loop to the radiator, the coolant is flowing only in the bypass loop.

Fig.4.7 shows the total power consumption of the cabin HVAC. It includes power consumption of both cabin heater and the cabin AC system. In the mild climate, both are used minimalistically. Fig. 4.8 shows the plot of total power consumption of thermal management system. It includes power consumption of battery and propulsion circuit pumps, radiator fan and cabin HVAC.

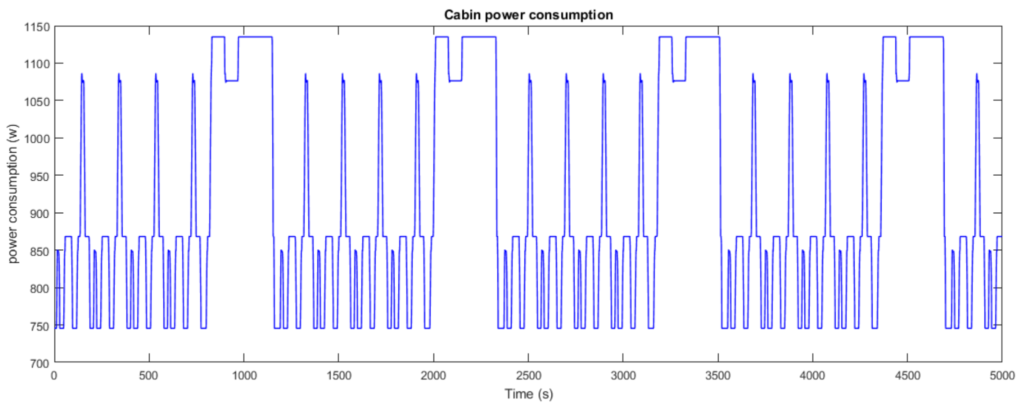



**Figure 4.7:** Total power consumption of cabin

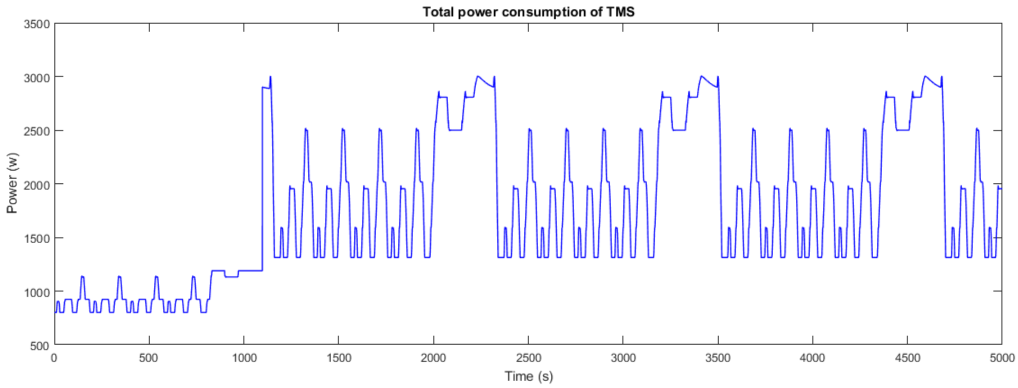



**Figure 4.8:** Total power consumption of thermal management system

### 4.1.2 Hot climate

For the hot ambient climate case, a temperature of $40°C$ was chosen as the initial temperature for the battery, propulsion unit and the ambient climate.

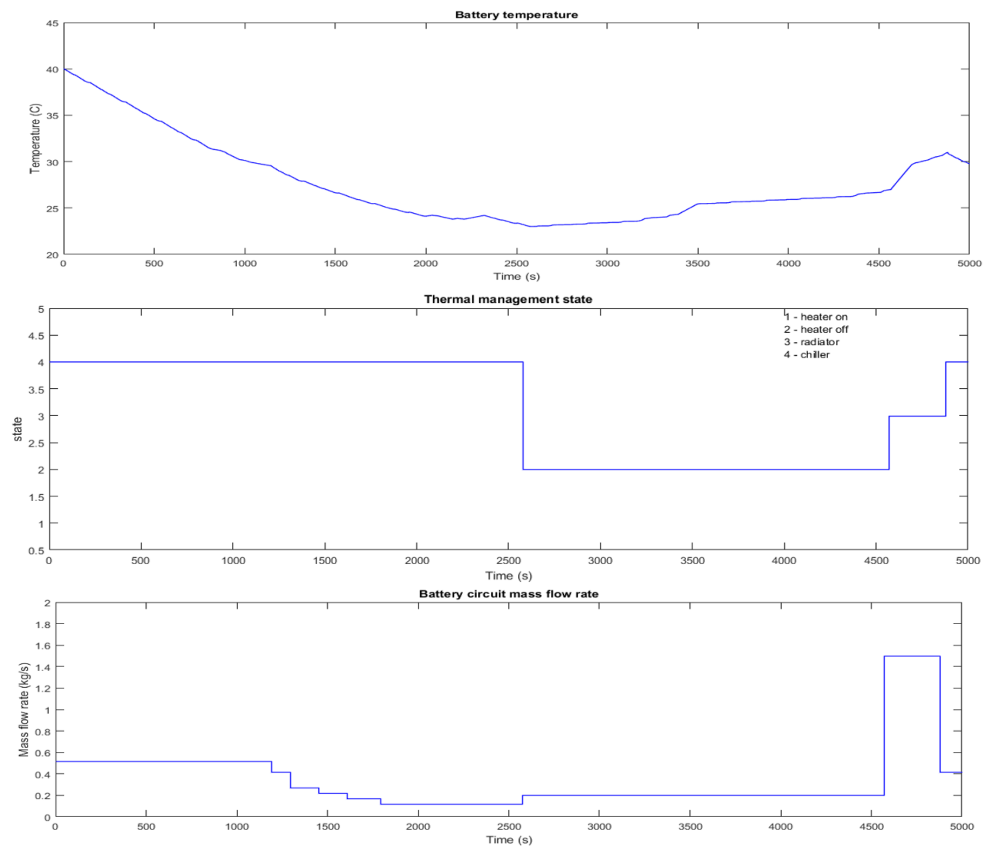



**Figure 4.9:** Thermal results for battery thermal management system

Fig. 4.9 shows the plot of battery temperature, state of thermal management system and mass flow rate of coolant in the battery circuit. Because of the high ambient temperature, radiator will not be able to cool the battery. Thus the thermal management state is operating at chiller mode. When the coolant is flowing through the chiller, its mass flow rate depends on the battery temperature as modeled in the control system. Thus it can be observed in the Fig. 4.9 that as the battery temperature is decreasing, its mass flow rate also decreases.

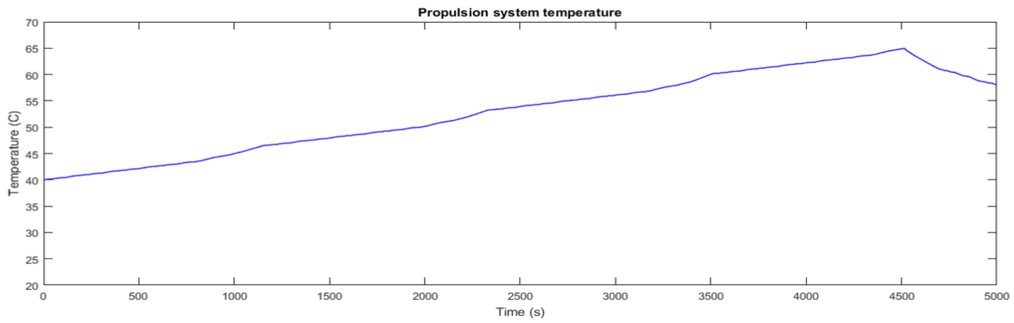



**Figure 4.10:** Temperature of propulsion unit

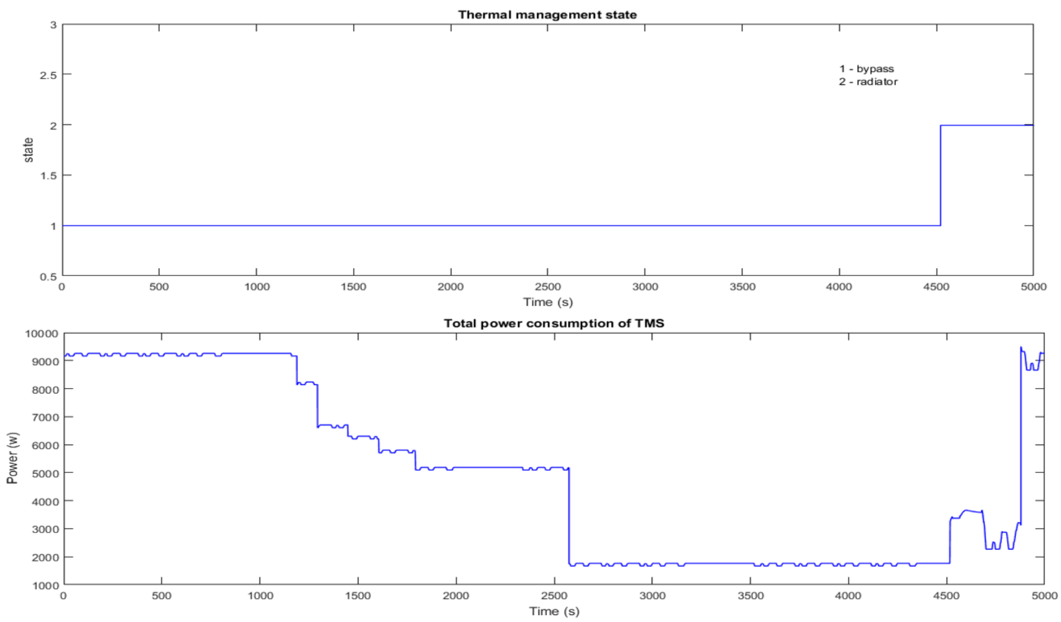



**Figure 4.11:** Plots of propulsion unit

Fig. 4.10 shows the plot of temperature variation of propulsion unit and Fig. 4.11 represents thermal management state of propulsion unit and total power consumption of thermal management system. It can be observed that temperature of the propulsion unit starts to decrease after 4000s. This is because the coolant path has changed from bypass to radiator. Also, it can be observed that the overall power consumption follows the trend of battery thermal management state. It shows that chiller is the highest power consuming component among all other components.

### 4.1.3 Cold climate

For the case of cold ambient, a temperature of $-10°C$ was chosen as the initial temperature for the battery, propulsion unit and the ambient climate.

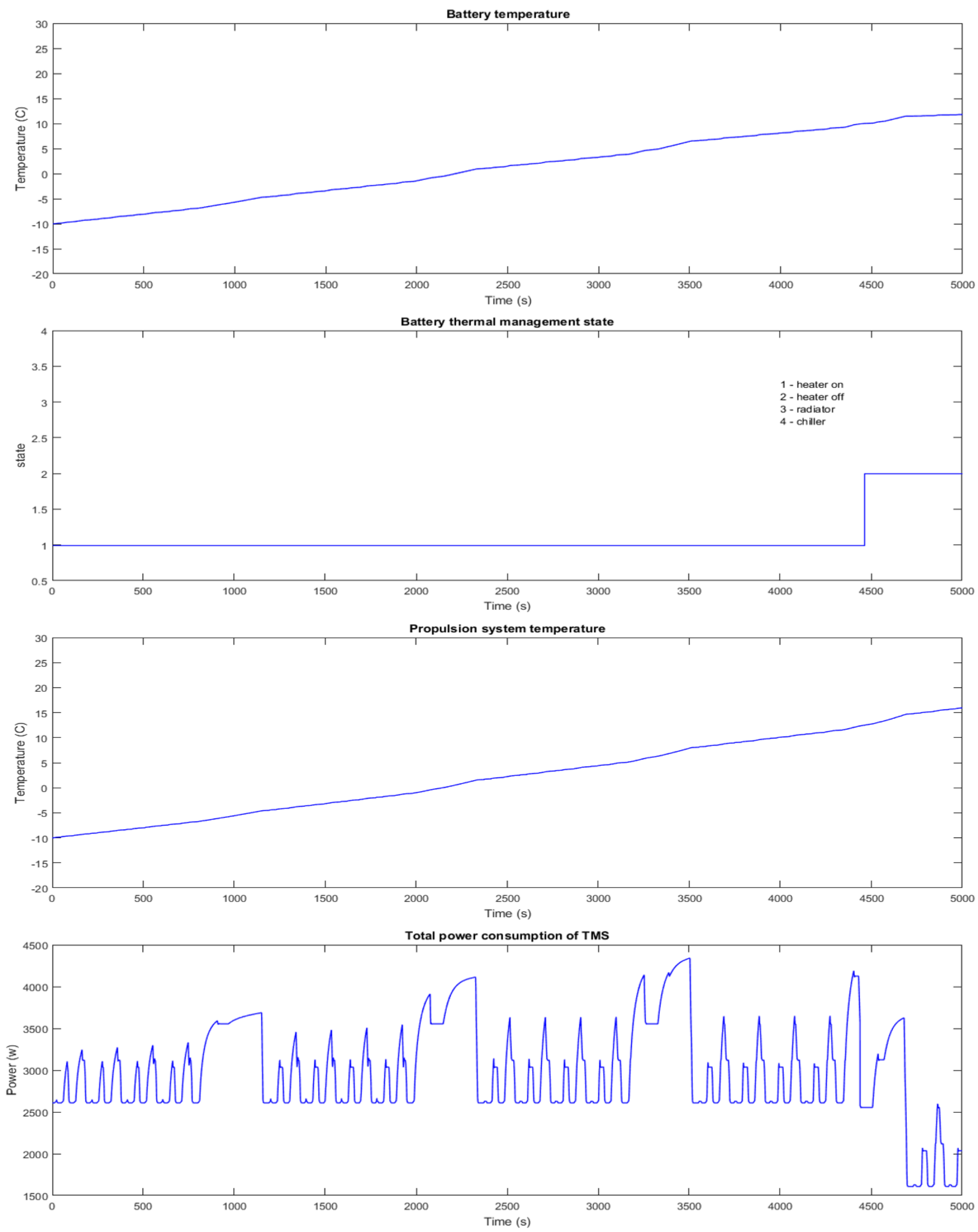



**Figure 4.12:** Plots for cold climate

Fig. 4.12 shows the results of temperatures and power consumption for cold climate case. It can be observed that the battery temperature is increasing because the coolant is flowing through the heater. The heater is on till 4500s and then the bypass mode in on. This is because of the fact that the battery heat generation is sufficient for the further heat up of the battery. The propulsion unit temperature is increasing in almost the same rate as that of the battery though there is no heater included in the propulsion circuit. This is because the heat generation rate of the propulsion is much higher in magnitude than the battery pack. This was one of the primary reason for not including the heater in propulsion thermal management circuit. The only power consumption units during the cold climate are pumps, battery heater and the cabin heater. Thus the total power consumption is less in magnitude when compared to hot climate case.

### 4.1.4 Battery resistance

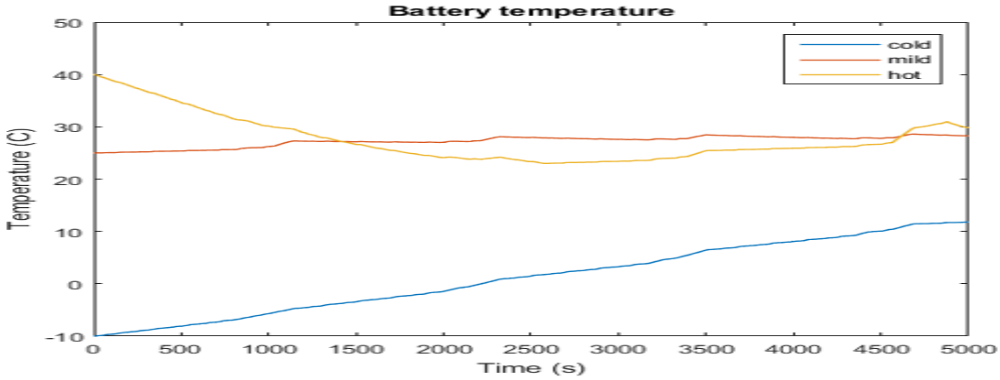



**Figure 4.13:** Battery temperature for all climatic conditions

Fig. 4.13 shows the plot of battery temperature variation for three climatic cases for NEDC driving cycle. Also Fig.4.14 shows the plot of battery resistance for the similar cases.

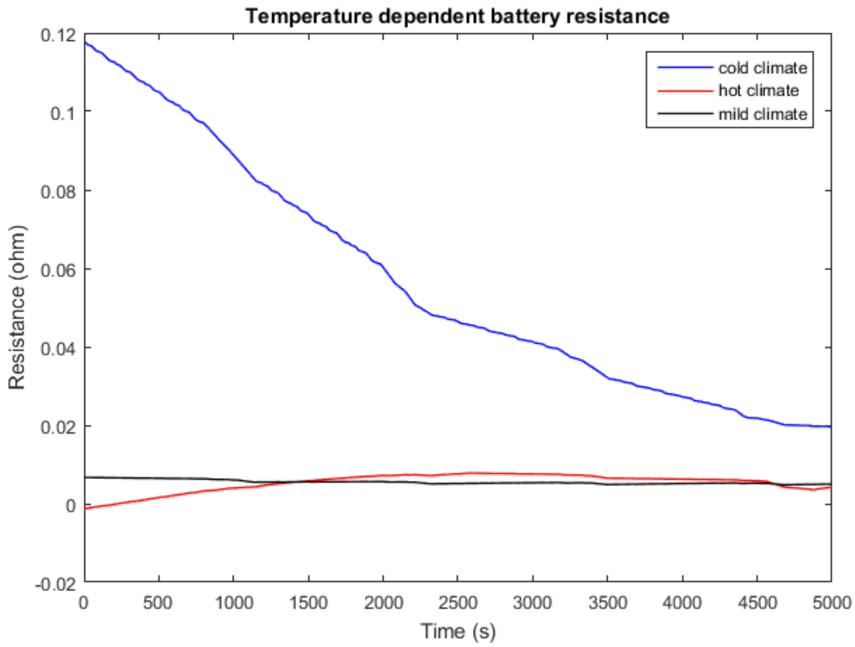



**Figure 4.14:** Temperature dependent battery resistance

It is explained in the theory section that the battery capacity increases with increase in temperature. To investigate the effect of temperature on the battery capacity, resistance as a function of temperature is plotted. Fig. 4.14 shows the plot of battery resistance as a function of battery temperature for all three cases of climatic conditions. It can be clearly observed that the resistance is maximum at the beginning of the cold climate case and decreases gradually. This is because of the increase in temperature of the battery due to heater. Also it can be observed that the resistance is minimum at the beginning of hot climate case due to the high battery temperature.

## 4.2 Energy consumption

The thermal management system model was tested for two modified versions of driving cycles NEDC and EPA75 for ambient temperature varying from $-10°C$ to $40°C$ for energy consumption and vehicle range. The initial temperature of battery and propulsion unit is considered to be same as that of ambient temperature.

The energy consumed by the thermal management system can be expressed in terms of kWh in order to determine the total energy utilized to maintain a working thermal environment for the total discharge of the battery. As explained earlier, the initial SoC of the battery is 90% and all the results are recorded till the point where the final SoC drops to 1%. The drive cycles that have been used as input are chosen such that two different kinds of demands can be run. The portion of the NEDC cycle used is smoother and involves lesser number of changes in acceleration/ deceleration. The EPA75 drive cycle used is more demanding in terms of sudden changes in velocity. The maximum velocity in either of the driving cycles is more or less similar. The reason for choosing these particular portions of the standard drive cycles is to observe noticeable changes in energy consumption and also the range for one parameter, i.e. number of times the velocity alters and not the magnitude of the velocity.

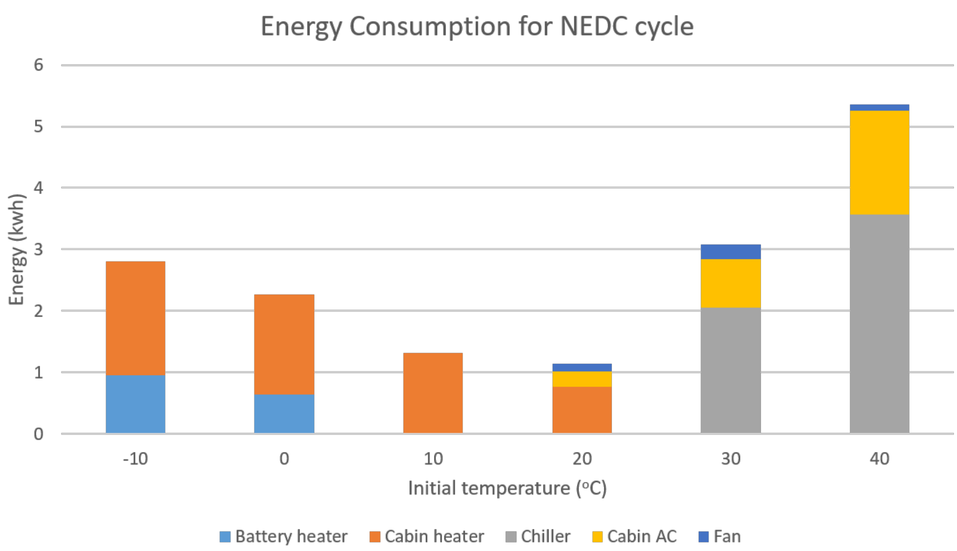



**Figure 4.15:** Component energy consumption for NEDC

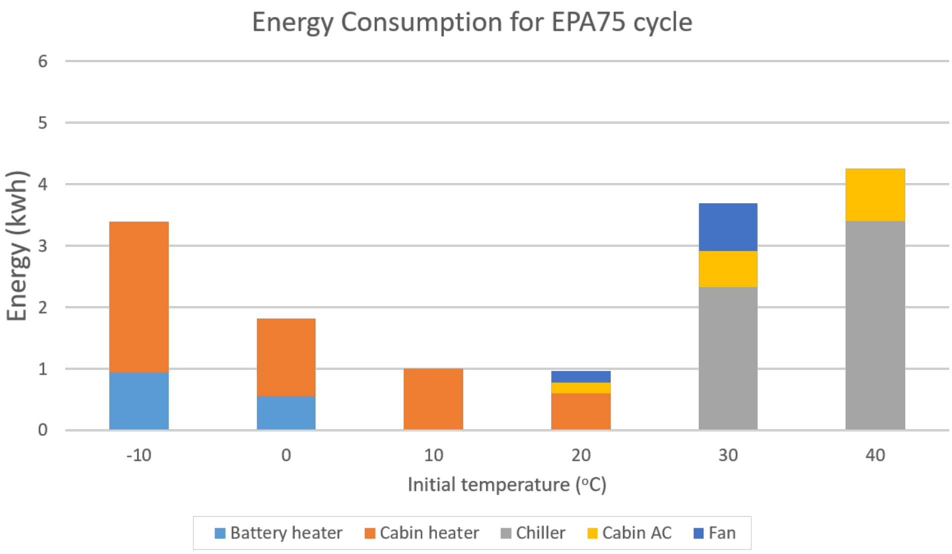



**Figure 4.16:** Component energy consumption for EPA75 driving cycle

Fig. 4.15 and Fig. 4.16 shows the energy consumption of all the heating and cooling components in the thermal management system for NEDC and EPA75 driving cycle respectively. The figure helps in comparing the power consumption of components for different ambient conditions. It is quite obvious that the heating components which are cabin heater and battery heater consume energy during the cold climatic conditions and cooling components, chiller, cabin AC and radiator fan consume energy during hot climatic conditions. The interesting thing to be observed here is the total amount of energy consumption for different ambient conditions. From the figure it is clear that chiller is the most highest energy consuming unit. This gives the user an idea of what component to focus on when designing a control strategy. Temperature range of around $10°C$ to $20°C$ is where the thermal management system consumes least energy.

| Initial temperature (°C) | Thermal management system power consumption (kWh) | |
|---|---|---|
| | NEDC | US |
| -10 | 2.85 | 3.04 |
| 0 | 2.31 | 1.84 |
| 10 | 1.45 | 1.1 |
| 20 | 1.22 | 1.03 |
| 30 | 3.15 | 3.03 |
| 40 | 4.61 | 4.31 |

**Table 4.2:** Thermal management system energy consumption

The average velocity for NEDC cycle is 6.77m/s from start to finish and for the EPA75 cycle it is 12.96 m/s. As a result, the model runs on EPA75 cycle for shorter time compared to NEDC cycle. This also supports the result shown in the Tab. 4.2 that the total energy consumption of the 'thermal management system' of the battery and propulsion is lesser for EPA75 cycle compared to NEDC. Also, the work done by the vehicle is more during the EPA75 cycle when compared to NEDC for the same amount of time, because of which more range is achieved for the same battery capacity and discharge. This can be seen in the Tab. 4.3.

| Initial Temperature | NEDC | | EPA75 | |
|---|---|---|---|---|
| (°C) | Range (km) | Time (mins) | Range (km) | Time (mins) |
| -10 | 25.7 | 56.6 | 28 | 45.16 |
| 0 | 29.14 | 57 | 30.21 | 42.05 |
| 10 | 34.3 | 66 | 35 | 46.91 |
| 20 | 35.25 | 67.1 | 35.61 | 47.36 |
| 30 | 24.96 | 51.4 | 24.08 | 37.05 |
| 40 | 16.55 | 35.28 | 16.58 | 30.5 |

**Table 4.3:** Vehicle range comparison

# 5
# Conclusion

A generic model of thermal management system for battery, propulsion unit and the cabin has been modeled using the lumped system approach in the tool SIMULINK. The thermal management system was integrated with the ÅF's complete vehicle energy model to calculate the energy consumption and the range of the vehicle.

The mathematical modeling has been developed based on the principle of lumped system approach and this method was compared with 1D CFD approach using the tool GT-SUITE. A simple passive cooling system was built using both the methods and the results were compared. Though there is minute deviation in the magnitude, the thermal behaviour matches for both the models.

The thermal management system for the battery has passive cooling, heater and active cooling included in it. A control system has been modeled to control the mode of operation of the cooling system. The control parameters have been modeled as variables that can be entered by the user so as to make the model more flexible.

The losses in the air path across the radiator and the cooling package have been modeled using the simple thermodynamic approach. The model gives the mass flow rate of the air and the pressure rise required by the fan to maintain that particular mass flow rate as output.

Finally, the model was tested with the two driving cycles NEDC and EPA75 for various range of ambient temperatures. Both the thermal behaviour of the system as well as the energy consumption and vehicle range have been presented as results.

# 6

# Future Scope

The main goal of the thesis itself limits the scope as to model a generic and flexible climate system for the battery and other heat generating sources in an electric vehicle. Thus a very simplified approach is used for mathematical modeling. Though the modeling approach was compared with 1D CFD approach, the accuracy and the reliability of the model can be better understood by comparing the results with the test data.

**Accuracy:**

- The accuracy of the model can be further increased by having a coupling with the 3D CFD models to predict the heat transfer of the battery, radiator, chiller and the propulsion unit.
- The battery pack is considered to have a isothermal temperature distribution across all the spatial directions. In practicality, the battery pack will have an uneven temperature distribution. The model can be made more accurate by modeling the temperature distribution across the battery pack.
- Also, the mass flow rate of the air across the radiator and the cooling package is calculated using basic thermodynamic approach. The accuracy can be increased by linking a 3D CFD model of the underhood air flow.
- The pressure losses in the coolant path has not been modeled in the thermal management system. The accuracy in the pump energy consumption can be increased by modeling the pressure loss in the coolant path
- Other electric losses like the losses involved across the battery terminals can also be accounted. Some heat is dissipated through these terminals apart from the other parts of the battery pack, which are assumed to be well insulated.

**Reliability:**

- The propulsion unit is considered as a lump which generates equivalent heat as generated by the motors and power electronics. The model can be made reliable and accurate by modeling the electric motors and the power electronics separately as the individual components.
- The model predicts the affect of temperature on the battery capacity, but does not give any idea about the cycle life of the battery. It would be better to monitor both the battery capacity and cycle life as a function of the battery temperature.
- The model right now has a one way coupling between the ambient conditions and the thermal management system. It can be made more reliable by including a two way coupling between the ambient and the thermal system.

**Results:**

- The main goal of the thesis was to model the climate system which must be flexible and not for a particular type of system. Thus the results presented in the report are just qualitative analysis of a particular type of system. The model can be used for the quantitative analysis if all the particulars and data of the system are predefined and available.
- The results presented for the current thermal management system are for one particular control strategy. However, an optimization study can be made by optimizing the control strategy for the best efficiency.

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# A

# Input manual

## A.1 Introduction

The Matlab / Simulink model for the battery thermal management system consists of several inputs that are written in the input script and the output is shown in the GUI. Most of the inputs written have a fixed value, for example: mass of the battery, `TMS_mbatt`. Physical properties of the components and the coolants have been fixed considering literature that have been referenced in the bibliography.

## A.2 Unit of measurement

All the units are according to the International Sysytem of Units (SI) throughout the script and the model. The radiator, chiller, HVAC data etc, have been converted to SI units wherever necessary.

## A.3 Description of Input script

`INPUT_26`.m consists of all the controlling parameters related to the energy model. The script line numbers from : 422 to 510 contain the variables used for the thermal management system. The variable names start with the term that suggests the functional aspect of the overall model, i.e., TMS, which stands for Thermal Management System. In this case it includes the thermal management of the battery and the propulsion system including the motors and the power electronics.

```
422     %---------------- Thermal management system-----------
423 -   TMS_enable = 1;                  % 0-disable, 1-enable
424
```

**Figure A.1:** TMS: Enable / Disable

The battery thermal model specifies variables like the battery mass, its specific heat capacity etc., as shown below. The parameters required to calculate the coolant volume enclosed inside the coolant tubes running through the battery have been listed in the Fig. A.2.

```
%----Battery thermal model-----
TMS_Tamb = 20;                   % Ambient temperature (K)
TMS_Tbattini = 20;               % Battery initial temperature (K)
TMS_mbatt = 500;                 % Mass of battery (kg)
TMS_cpbatt = 1040;               % Specific heat capacity of battery (J/kg K)900
TMS_r1tube = 0.007;              % inner radius of the tube (m)0.007
TMS_r2tube = 0.009;              % outer radius of the tube (m)0.009
TMS_ntube = 4;                   % number of tubes 5
TMS_Ltube = 0.6;                 % Length of the tubes (m)2
TMS_ktube = 401;                 % Thermal conductivity of tube material (W/m K)200
```

**Figure A.2:** Battery Thermal Model

The coolant properties are assumed as shown in the figure below.

```
%---Battery cooling system-----
TMS_kcoolbatt = 0.402;          % Thermal conductivity of coolant(W/mK) 0.25
TMS_mucoolbatt = 4.87e-3;       % Dynamic viscosity of coolant(Ns/m^2)2.2e-3
TMS_cpcoolbatt = 3260;          % specific heat capacity of coolant(J/kg K)3410
TMS_rhocoolbatt = 1082;         %Density of the coolant (kg/m^3)1130
TMS_Tcoolinibatt = 20;          % Initial temperature of coolant (K)
```

**Figure A.3:** Battery Cooling System

Pump data received from ÅF is loaded. It has pressure rise vs volumetric flow rate of a specific pump.

```
%---RESS_Pump--------
TMS_RESSpumpdp = load('data\tms\pump.dat');  % pressure rise of pump (psi)
```

**Figure A.4:** Pump

Radiator data is loaded and the coolant volume inside the radiator is assumed. This value can be altered by the customer if geometric data about the radiator is available. The radiator data gives the cooling rate as a function of coolant mass flow rate and air velocity.

```
%---LT_Radiator----
TMS_VolumeLTrad = 0.005;        % Volume of fluid in radiator (m^3)0.005
TMS_UALTrad = load('data\tms\radiator.dat');
```

**Figure A.5:** LT Radiator

Similar calculation of coolant volume inside the heater is used considering convection of heat into the coolant passing through heated tubes. The power wattage of heater is used.

```
%---RESS_Heater------
TMS_pRESSheater = 1000;          % Power input of heater (w)1000
TMS_nuRESSheater = 0.9;          % effeciency of heater (%)0.9
TMS_volRESSheater = 0.005;       % volume of coolant in heater (m^3)0.005
```

**Figure A.6:** Heater

Chiller data is loaded based on the plots received from ÅF. The data has the chiller cooling rate as a function of coolant flow rate.

```
%---Chiller------
TMS_volchiller = 0.005;        % Volume of coolant in chiller (m^3)0.005
TMS_Qchiller = load('data\tms\chiller.dat');
TMS_copchiller = 1;
```

**Figure A.7:** Chiller

Coolant properties and volume inside the tubes running through the propulsion system are assumed.

```
%---Propulsion thermal model--------
TMS_Tpropini = 20;              % Initial temp of propulsion system (K)
TMS_mcpprop = 400000;           % Thermal Mass of propulsion system (J/K)

%---Propulsion cooling system-------
TMS_cpcoolprop = 1000;          % Heat capacity of propusion coolant loop (J/kg K)
TMS_volcoolprop = 0.01;         % Volume of coolant in propulsion system (m^3)
TMS_UAprop = 2000;              % Heat transfer coefficient of propulsion system (w/K)
TMS_rhocoolprop = 1082;         % Density of the coolant in propulsion system (kg/m^3)
TMS_Tcooliniprop = 20;          % Initial temperature of coolant in propulsion system (K)
```

**Figure A.8:** Propulsion Thermal Model

Same radiator data has been used for both LT and HT radiator.

```
%---HT_radiator------
TMS_volHTrad = 0.005;           % Volume of coolant in HT radiator (m^3)
TMS_UAHTrad = load('data\tms\radiator.dat');
```

**Figure A.9:** HT Radiator

The pressure loss in the air path is modeled using the air path model called as fan model.

```
%---Fan-----
TMS_airpath_Enable = 1;           % 0 = Disbale air path model, 1 = Enable air path model
```

**Figure A.10:** Fan

This model loads the fan data with respect to its pressure rise with air flow rate. The pressure drop in the cooling package is calculated using this input.

```
% To be specified if air path model is disabled
TMS_CONSTairvel = 5;            % Velocity of air through packge (m/s)
TMS_CONSTvdot = 2.5;            % Volumetric flow rate of air through package (m^3/s)
% To be specified if air path model is enabled
TMS_fanOriginalSpeed = 40;      % Speed of the refernce fan (rps) (available data)
TMS_fanOriginalDia = 0.78;      % Dia of the refernce fan (m)
TMS_fancurve = load('data\tms\Fancurve_D780.dat'); % Fan curve (flow rate vs pressure rise)
TMS_fanspeed = 40;              % Fan speed (rps)
TMS_fanDia = 0.5;               % Fan diameter (m)
[V_3,dp_3] = Fancurve(TMS_fanDia, TMS_fanOriginalDia, TMS_fanspeed, TMS_fanOriginalSpeed, TMS_fancurve);
```

**Figure A.11:** Air path

```
%---Propulsion_Pump----
TMS_PROPpumpdp = load('data\tms\pump.dat');  % pressure rise of pump (psi)
```

**Figure A.12:** Pump for propulsion unit

For the control system the controlling parameters are battery temperature and the coolant mass flow rates applicable for its passage through the different components of the thermal management system.

```
%---Control system----------
TMS_cs_Enable = 1;               % 0 = Disble control system, 1 = Enable control system
% To be specified if control system is disabled
TMS_state_batt = 3;             %Battery circuit coolant flow direction
                                % 1 = heater, 2 = bypass, 3 = radiator, 4 = chiller
TMS_state_prop = 1;             %Propulsion circuit coolant flow direction
                                % 1 = bypass, 2 = radiator
% To be specified if control system is enabled
TMS_Treqbatt = 25;              %Required temperature of battery (K)
TMS_Tcritprop = 65;             %Critical temperature for propulsion system (K)
TMS_mdotRESSrad = 1.5;          %mass flow rate when coolant in RESS radiator (kg/s)
TMS_mdotRESSbyp = 0.2;          %mass flow rate when coolant in RESS bypass (kg/s)
TMS_mdotPROPrad = 1.5;          %mass flow rate when coolant in propulsion radiator(Kg/s)
TMS_mdotPROPbyp = 0.2;          %mass flow rate when coolant in propulsion bypass (Kg/s)
```

**Figure A.13:** Control system

Cabin HVAC data is loaded from the plots received from ÅF for the purpose of calculating the cooling rate of the cabin AC and heater.

```
%---Cabin HVAC----
TMS_compressor = load('data\tms\compressor.dat');
TMS_HVACheater = load('data\tms\heater_hvac.dat');
```

**Figure A.14:** Cabin HVAC

# B

# Appendix

## B.1 Radiator data

The figure below shows the radiator data used for testing the model. The table data refers to the heat transfer rate of the radiator in kW as function of coolant flow rate and velocity of the air.

| flow rate of coolant ($m^3/s$) \ Velocity of air (m/s) | **1.886** | **3.69** | **5.5** | **7.3** | **9.1** | **10.9** |
|---|---|---|---|---|---|---|
| **0.0004** | 10.48 | 13.16 | 14.4 | 15.11 | 15.57 | 15.89 |
| **0.0008** | 13.89 | 19.15 | 21.96 | 23.7 | 24.89 | 25.76 |
| **0.0011** | 15.8 | 23.06 | 27.31 | 30.1 | 32.06 | 33.52 |
| **0.0015** | 17.05 | 25.87 | 31.39 | 35.16 | 37.89 | 39.97 |
| **0.00197** | 17.94 | 28.01 | 34.63 | 39.31 | 42.78 | 45.45 |
| **0.0023** | 18.61 | 29.7 | 37.29 | 42.79 | 46.95 | 50.22 |
| **0.0027** | 19.13 | 31.09 | 39.52 | 45.77 | 50.58 | 54.41 |
| **0.0031** | 19.56 | 32.24 | 41.42 | 48.35 | 53.77 | 58.13 |
| **0.0035** | 19.91 | 33.22 | 43.07 | 50.63 | 56.61 | 61.47 |

**Figure B.1:** Heat transfer data of the radiator

## B.2 Chiller data

The plot below was provided for the chiller heat transfer rate as the function of coolant volumetric flow rate.

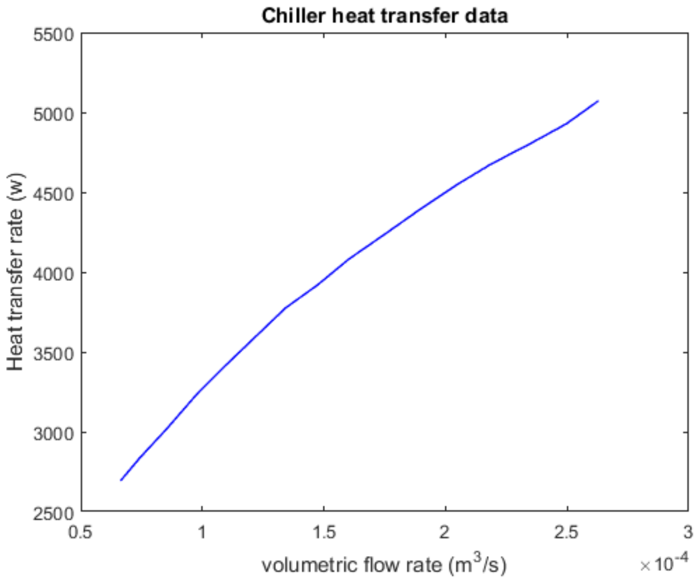


**Figure B.2:** Heat transfer data of the chiller

## B.3 Pump performance curve

The plot below was provided as the pump performance curve. The plot gives pressure rise of the pump as a function of volumetric flow rate of coolant.

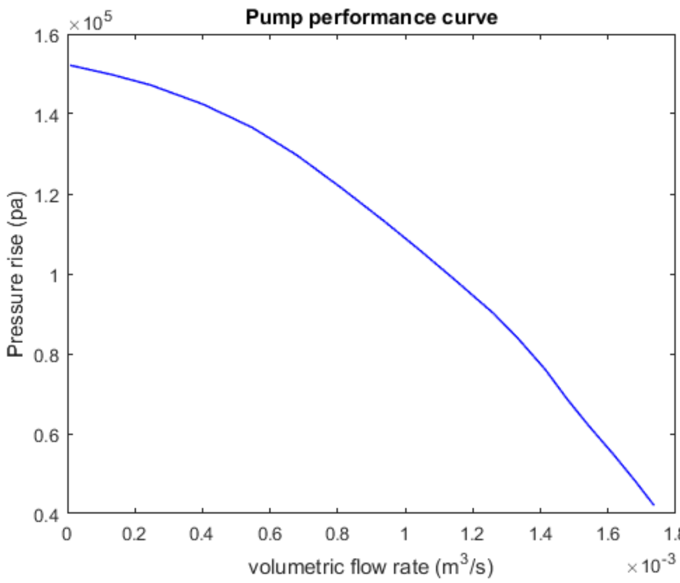


**Figure B.3:** Heat transfer data of the chiller

## B.4 Fan performance curve

The plot below was provided as the fan performance curve. The plot gives pressure rise of the fan as a function of volumetric flow rate of air.

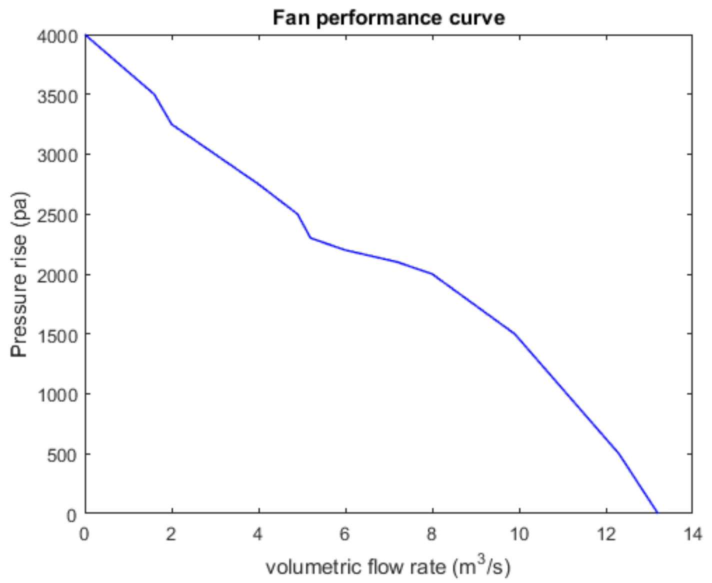


**Figure B.4:** Heat transfer data of the chiller

## B.5 EPA75 driving cycle

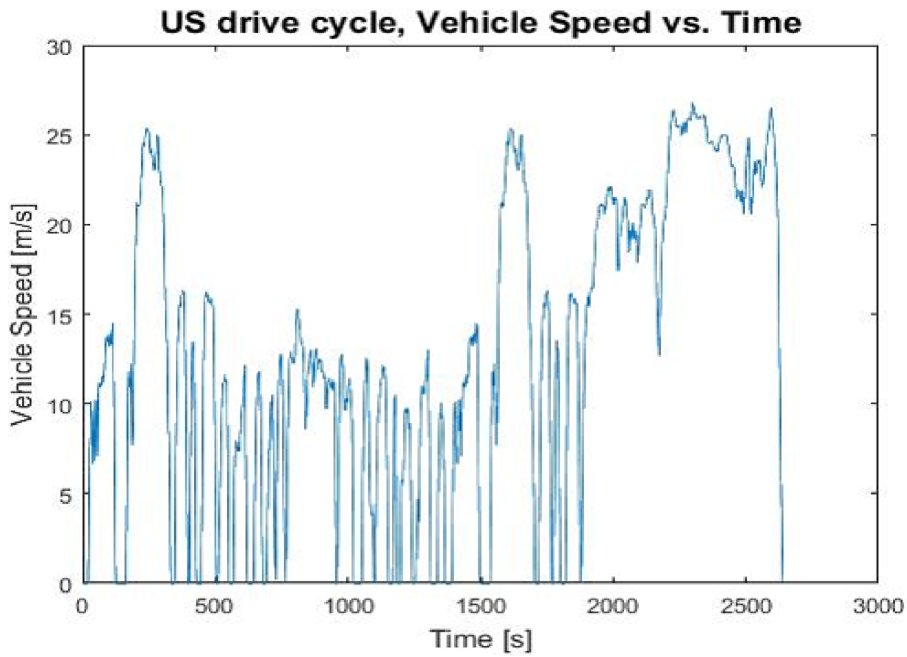


**Figure B.5:** cycle speed for EPA75 drive cycle

### B.5.1 Cold climate

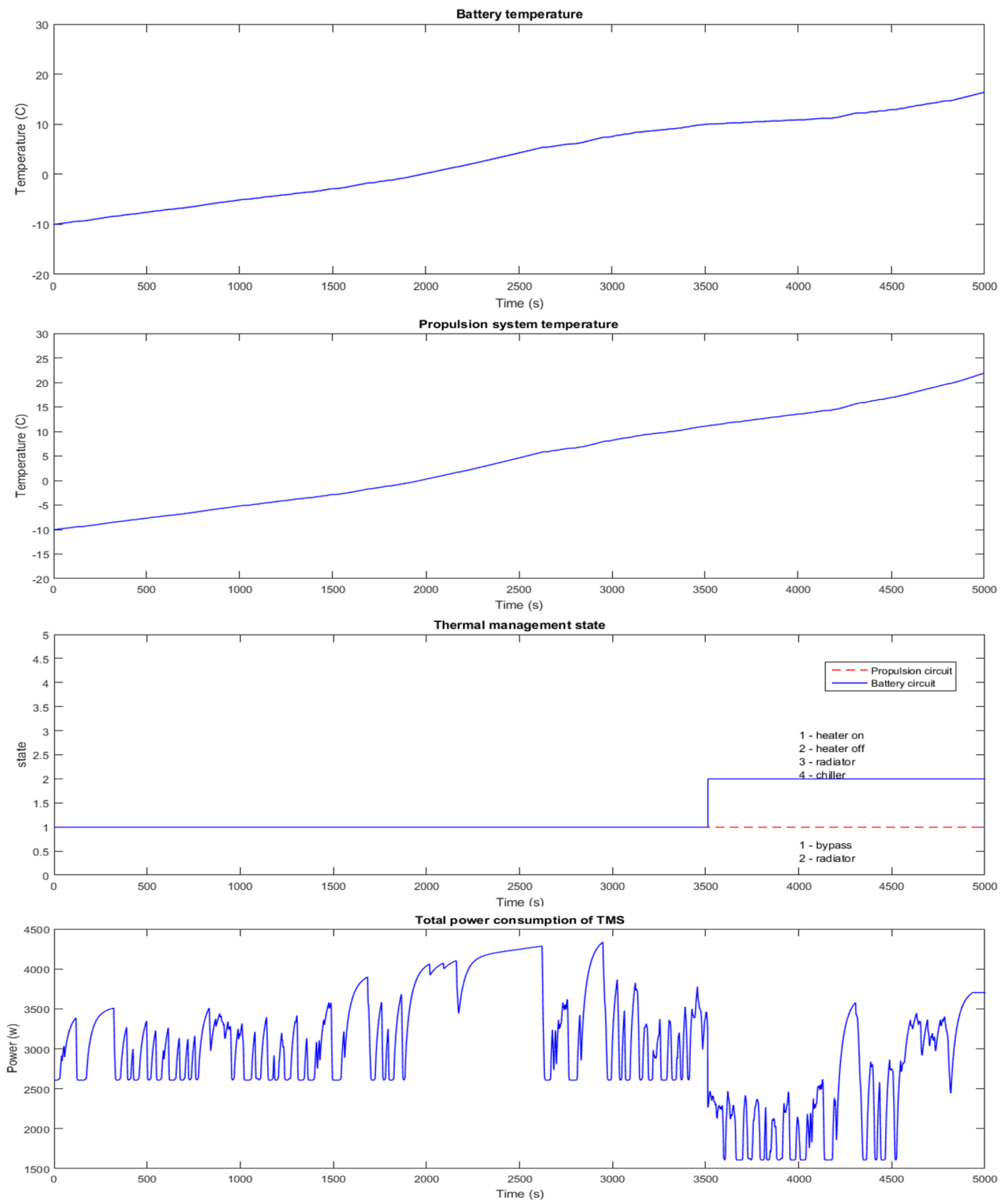


**Figure B.6:** Plots for cold climate case - EPA75 drive cycle

### B.5.2 Mild climate

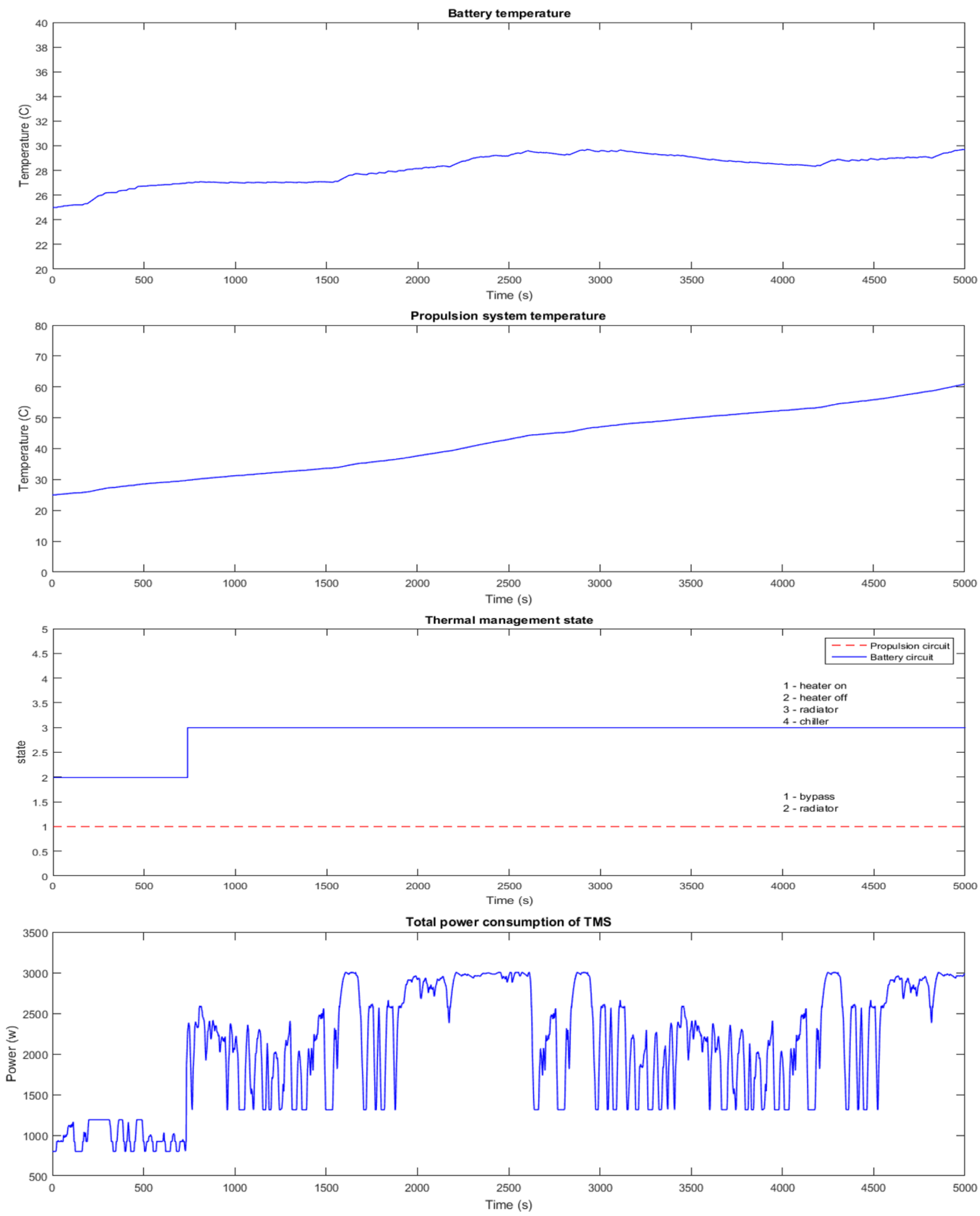


**Figure B.7:** Plots for mild climate case - EPA75 drive cycle

### B.5.3 Hot climate

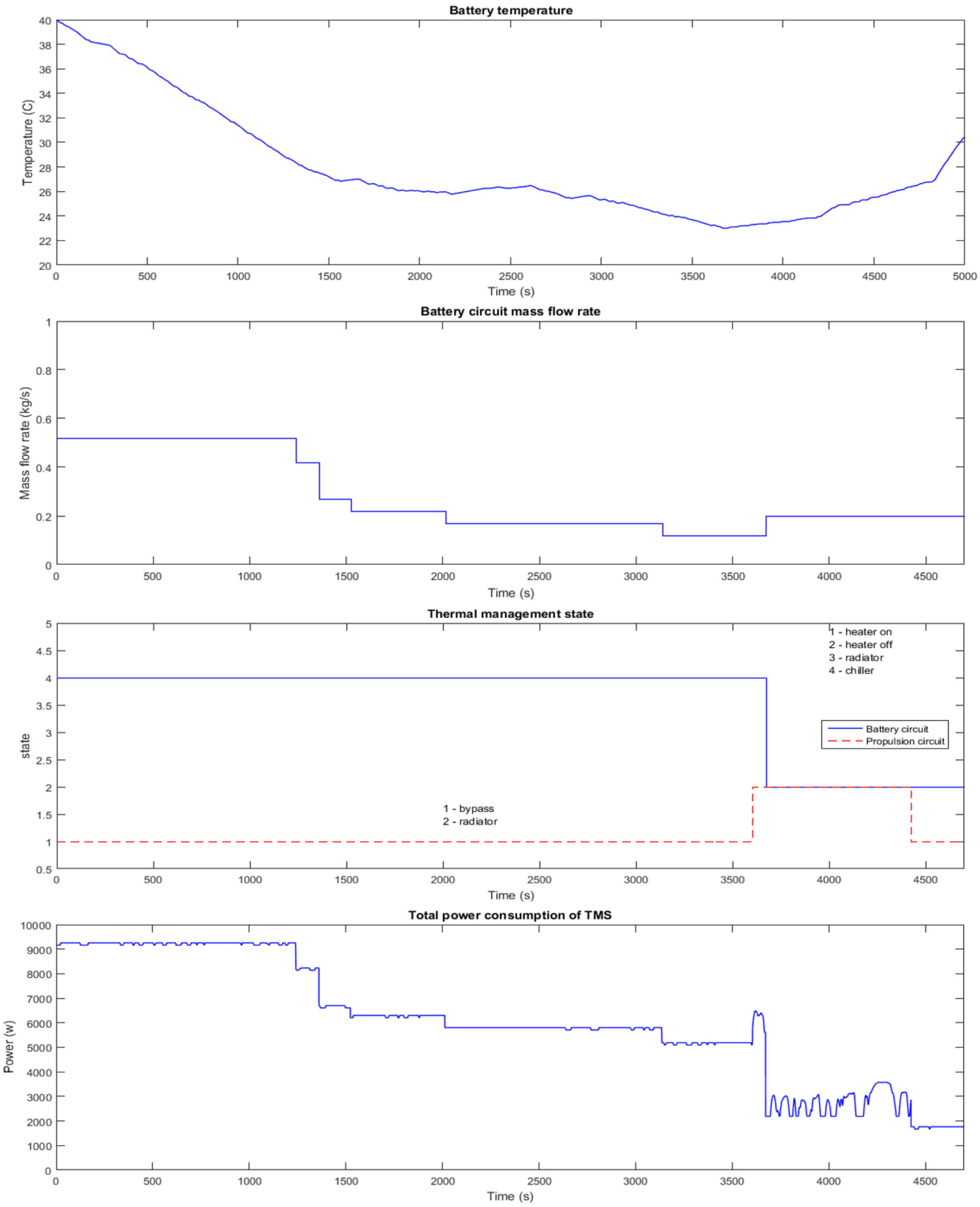


**Figure B.8:** Plots for hot climate case - EPA75 drive cycle