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Report

Kriti '25

Motor Cooling System Design

Hostel 78

Contents

1 Abstract

This report presents the design and implementation of a thermal management system for a Formula Student Electric Vehicle (FSEV), focusing on efficient heat dissipation from the in-wheel motor (EMRAX 208) and the motor control unit (BAMOCAR-D3). The cooling system integrates a liquid-based loop that uses water as a coolant, coupled with a radiator,fan, and pump to ensure optimal temperature regulation. A lookup table-based control strategy dynamically adjusts the fan speed and coolant flow rate to maintain efficiency and prevent thermal overload. The simulation results validate the system's ability to sustain safe operational temperatures, optimizing performance and energy consumption for high-performance electric mobility applications.

2 Problem Understanding

Efficient thermal management is crucial for the performance and reliability of Formula Student Electric Vehicles (FSEVs), where high-power components like the motor (EMRAX 208) and motor control unit (BAMOCAR-D3) generate significant heat. Without effective cooling, excessive temperatures can reduce efficiency, degrade components, and compromise safety. This project addresses the challenge by developing a liquid-based cooling system capable of dissipating up to 10 kW of heat while maintaining safe operating temperatures. Using water as the primary coolant, the system integrates a radiator, fan, and pump to optimize heat transfer. A lookup table-based control strategy dynamically adjusts coolant flow and fan speed, ensuring efficient temperature regulation while minimizing energy consumption. This approach enhances the system's reliability and performance under dynamic driving conditions.

3 Heat Management Model in Simulink

3.1 Cooling System Structure

- A **closed-loop liquid cooling system** is used to regulate motor and MCU temperatures.
- The system incorporates:
  - **Radiator** for heat dissipation
  - **Fan** to enhance airflow
  - **Pump** for continuous coolant circulation
  - **Coolant pipes** for coolant flow
  - **Control unit** for dynamic adjustments

3.2 Heat Transfer Mechanism

- Heat generated by the motor and MCU is absorbed by the circulating coolant.
- The heated coolant passes through the **radiator**, where it dissipates heat into the surroundings.
- A **fan** increases airflow over the radiator for improved cooling efficiency.
- The **pump** maintains continuous coolant circulation to prevent overheating.

3.3 Control Strategy Implementation

- A **lookup table-based control system** dynamically adjusts:
  - **Fan speed**
  - **Coolant flow rate**
- Adjustments are made based on **real-time temperature readings**.
- This ensures **efficient cooling**, **energy optimization**, and **prevention of thermal overload**.

3.4 Simulation Parameters and Analysis

- The model accounts for **fluid dynamics, heat transfer properties, and component efficiency**.
- Key parameters include:
  - **Coolant flow rate**
  - **Reynolds number**
  - **Pressure losses**
  - **Thermal conductivity**
- The results help assess the effectiveness of the cooling strategy, ensuring optimal temperature regulation.

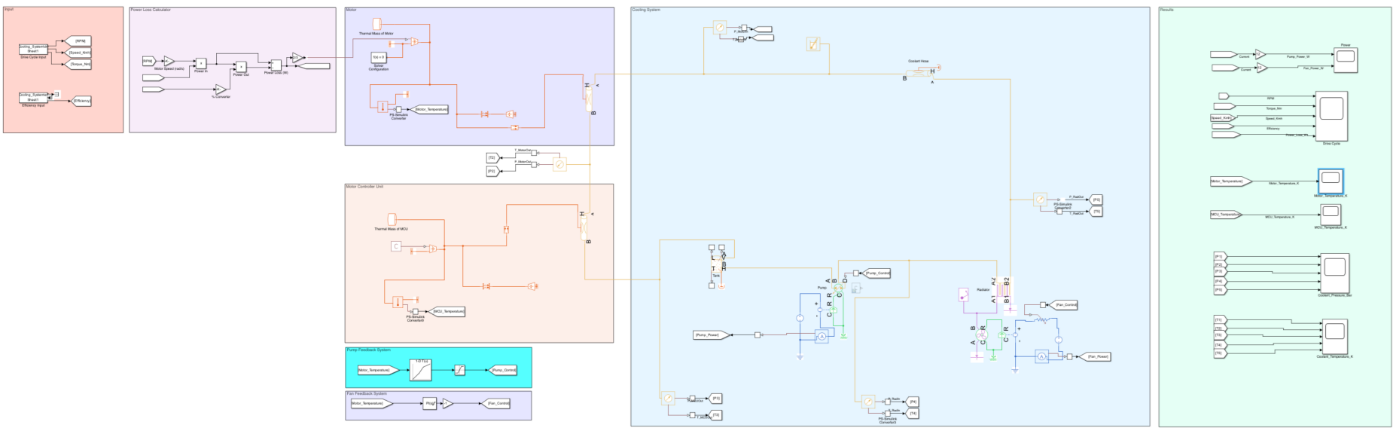

Figure 1: The Simulink Model

4 Design approach

The thermal management system for the Formula Student Electric Vehicle (FSEV) is designed to effectively dissipate the peak heat losses generated by the in-wheel motor (EMRAX 208) and the motor control unit (BAMOCAR-D3), ensuring optimal performance under dynamic driving conditions. This system is specifically designed to handle maximum heat dissipation of 10kW within specified temperature limits. The design approach integrates thermodynamic principles, fluid dynamics, and control strategies to maintain component temperatures within these limits.

4.1 Thermal Requirements

1. **Peak Heat Losses:**
   (a) Motor: 6 kW
   (b) Motor Control Unit (MCU): 4 kW
2. **Temperature Constraints:**
   (a) Maximum allowable motor temperature: 318.15 K (45°C)
   (b) Maximum allowable MCU temperature: 343.15 K (70°C)
   (c) Coolant temperature must remain below 363.15 K (90°C) to prevent vaporization.

These parameters establish the thermal load that the cooling system must handle while ensuring operational safety and efficiency.

4.2 Component Selection

1. **Coolant:** Deionized water
   Water is selected as the primary coolant due to its favorable thermal properties:
   (a) **High Thermal Conductivity:** 0.6 W/m·K, facilitating efficient heat transfer.
   (b) **Specific Heat Capacity:** 4.18 J/Kg. K, allowing significant energy absorption with minimal temperature rise.
   (c) **Cost-Effectiveness:** Readily available and economical for prototype and large-scale applications.
   (d) **Material Compatibility:** Non-corrosive to most components of the cooling system, reducing maintenance concerns.
2. **Pipe System**
   (a) **Inner Diameter:** 12 mm
   (b) **Total Length:** 3 m

   These dimensions optimize the coolant flow rate while minimizing pressure drops and ensuring a sufficient surface area for heat transfer. The material selection for pipes will ensure durability and thermal insulation to minimize heat losses to the environment.
3. **Radiator:**
   The radiator is critical for dissipating the absorbed heat from the coolant to the ambient environment. The selection criteria include:
   (a) **Heat Dissipation Capacity:** Sufficient to handle a combined peak heat load of 10 kW (motor + MCU).
   (b) **Material:** Aluminum, due to its high thermal conductivity and lightweight nature.
   (c) **Design:** Multi-pass radiator with finned surfaces to maximize surface area and enhance convective heat transfer.
   (d) **Placement:** Positioned to receive maximum airflow for improved cooling efficiency.

General Specifications

- **Heat Load (Q):** 10 kW (10,000 W)
- **Ambient Temperature ($T_{ambient}$):** 298.15 K (25°C)
- **Dimensions:** 35 cm x 30 cm x 3 cm (L x H x D)
- **Fan Power:** 0.11 horsepower
- **Material:** Aluminum (for both tubes and fins)
- **Objective:** Balance in cooling & budget

Physical Properties

- **Air Density (ρ):** 1.225 kg/m^3 (at sea level)
- **Specific Heat of Air (c_p):** 1005 J/kg-K
- **Thermal Conductivity of Aluminium (k_{Cu}):** 237 W/m-K

Gas 1 (Air) Parameters

Flow Characteristics

The air serves as the primary cooling medium, and its performance is crucial for the effectiveness of the heat exchanger.

Minimum Free-Flow Area

The free-flow area is determined by the fin spacing and the arrangement of the tubes in the heat exchanger. For optimal airflow:

$$A_{free-flow} = A_{total} - (N_t \cdot d_{fin} \cdot L)$$

Where:

- N_t = Number of tubes = **45**
- d_{fin} = Fin spacing = **15 mm = 0.0015 m**
- L = Length of the heat exchanger = **0.35 m**

Calculating:

$$A_{free-flow} = 0.0735 - (45 \cdot 0.0015 \cdot 0.35) = 0.0735 - 0.023625 = 0.049875\,\mathrm{m}^2 \approx 0.05\,\mathrm{m}^2$$

Hydraulic Diameter and Pressure Loss

The hydraulic diameter for a rectangular fin-and-tube heat exchanger can be approximated:

$$D_h = \frac{4 \cdot A_{free-flow}}{P_{wetted}}$$

Where P_{wetted} is the wetted perimeter.

Assuming a perimeter based on the tube arrangement yields a typical value of about 40 m:

$$D_h = \frac{4 \cdot 0.05}{40} \approx 0.5\,\text{cm}$$

Reynolds Number for Gas 1 The flow type (laminar or turbulent) can be assessed using the Reynolds number (Re):

$$Re = \frac{\rho \cdot v \cdot D_h}{\mu}$$

Assuming a dynamic viscosity μ of air at 291.15 K: - $\mu = 1.81 \times 10^{-5}\,\text{Pa} \cdot \text{s}$

Calculating using the previously found air velocity $v = 12.28\,\text{m/s}$:

$$Re = \frac{1.225 \cdot 12.28 \cdot 0.005}{1.81 \times 10^{-5}} \approx 4155\,(\text{Turbulent Flow})$$

Pressure Loss Calculation Utilizing the Darcy-Weisbach equation:

$$\Delta P = f \cdot \frac{L}{D_h} \cdot \frac{\rho v^2}{2}$$

Assuming $f = 0.03$ (friction factor for turbulent flow):

$$\Delta P = 0.03 \cdot \frac{0.9}{0.0266} \cdot \frac{1.225 \cdot (12.28)^2}{2} \approx 500.8\,\text{Pa}$$

Thermal Liquid 2 (Coolant) Parameters

The chosen coolant will significantly influence the thermal performance of the heat exchanger.

Coolant Properties

We consider water as the coolant due to its advantageous thermal properties.

- **Specific Heat Capacity ($c_{p,water}$):** 4182 J/kg-K
- **Density (ρ_{water}):** 1000 kg/m^3 (at room temperature)
- **Thermal Conductivity (k):** 0.606 W/m-K

Flow Characteristics

Minimum Free-Flow Area and Hydraulic Diameter

For the coolant flowing through the tubes:

$$A_{tube} = \frac{\pi \cdot D_i^2}{4}$$

Where $D_i = 0.006\,\text{m}$:

$$A_{tube} = \frac{\pi \cdot (0.006)^2}{4} \approx 2.827 \times 10^{-5}\,\text{m}^2$$

The hydraulic diameter for flow inside a tube:

$$D_h = D_i = 0.006\,\text{m}$$

Reynolds Number for Liquid 2 Similarly, we calculate the Reynolds number for the coolant:

$$Re = \frac{\rho_{water} \cdot v_{water} \cdot D_h}{\mu_{water}}$$

Assuming a typical velocity for coolant flow $v_{water} = 0.61\,\text{m/s}$ and dynamic viscosity $\mu_{water} \approx 0.001\,\text{Pa} \cdot \text{s}$:

$$Re = \frac{1000 \cdot 0.61 \cdot 0.006}{0.001} = 3,660\,\text{(Turbulent Flow)}$$

Pressure Loss Calculation Using the same Darcy-Weisbach equation: Assuming $f = 0.035$ for turbulent flow in smooth tubes:

$$\Delta P = f \cdot \frac{L}{D_h} \cdot \frac{\rho_{water} v_{water}^2}{2}$$

$$\Delta P = 0.035 \cdot \frac{1.0}{0.006} \cdot \frac{1000 \cdot (0.61)^2}{2} = 1.09\,\text{KPa}$$

Coolant Flow Rate Calculating the required coolant mass flow rate ($\dot{m}_{water}$):

$$Q = \dot{m}_{water} \cdot c_{p,water} \cdot \Delta T_{coolant}$$

Assuming a temperature rise $\Delta T_{coolant} = 17.3\,\text{K}$:

$$10,000 = \dot{m}_{water} \cdot 4182 \cdot 17.3 \implies \dot{m}_{water} = \frac{10,000}{17.3 * 4182} \approx 0.13777\,\text{kg/s}$$

Final Results Summary

- **Air Flow Rate:** 0.312 m³/s
- **Air Velocity:** 6.24 m/s
- **Mass Flow Rate (Air):** 0.382 kg/s
- **Temperature Rise (Air):** 26.1 K
- **Total Heat Transfer Area:** 7.0 m²

- **Overall Heat Transfer Coefficient:** 112.5 W/m²K
- **Pressure Drop (Air Side):** 41.0 Pa
- **Required Coolant Flow Rate (Water):** 0.13777 kg/s
- **Pressure Drop (Coolant Side):** 1.09 kPa

4. **Fan**
The fan enhances airflow across the radiator to improve heat dissipation, especially under low-speed or stationary conditions.

(a) **Fan Specifications**

- **Type:** Axial Fan (commonly used in motor cooling due to high flow rates and compact design).
- **Operating Voltage:** 12V (aligned with system power constraints).
- **Shaft Power:** 0.11 HP (selected based on required air movement).
- **Fan Diameter:** 350 mm (ensures proper coverage of the heat exchanger surface).

(b) **Nominal Flow Rate Calculation** To ensure effective cooling for a 35 cm × 30 cm heat exchanger:

i. **Circular Inlet/Outlet Area:**

$$A = \frac{\pi D^2}{4}, \quad D = 0.35 \text{ m} \tag{1}$$

$$A = \frac{3.1416 \times (0.35)^2}{4} = 0.0962 \text{ m}^2 \tag{2}$$

ii. **Face Velocity Estimation:** Assuming a typical cooling system axial fan face velocity of 8 m/s:

$$Q = \text{Face Velocity} \times A = 8 \times 0.0962 = 0.7696 \text{ m}^3/s \tag{3}$$

iii. **Volumetric Flow Rate:**

$$0.7696 \times 1000 = 769.6 \text{ L/s} = 1631 \text{ CFM} \tag{4}$$

Thus, the nominal volumetric flow rate is 0.77 m³/s (770 L/s or 1631 CFM).

(c) **Static Pressure Gain Selection** Axial fans generally operate at low static pressures. Based on similar applications:

- **Nominal Static Pressure Gain:** 90 Pa (sufficient for a heat exchanger of this size).
- **Maximum Static Pressure Gain:** 135 Pa (for cases of increased resistance, such as clogged fins).

(d) **Maximum Flow Rate Calculation** Considering a 10% margin for optimal conditions:

$$Q_{\max} = 0.773 \times 1.1 = 0.8503 \text{ m}^3/s \tag{5}$$

$$0.8503 \times 1000 = 850.3 \text{ L/s} = 1,802 \text{ CFM} \tag{6}$$

Thus, the maximum volumetric flow rate is 0.85 m³/s (850 L/s or 1.802 CFM).

(e) **Rotational Speed Considerations** Automobile fans in this power range typically operate between 3.000 RPM and 4000 RPM.

- **Reference Shaft Speed:** 4000 RPM (balancing performance and noise).
- **Minimum Shaft Speed:** 500 RPM (to prevent stall conditions).

5. **Pump**
The pump ensures continuous coolant circulation throughout the cooling loop:

   (a) **Type:** Variable-Displacement Pump (TL)
   (b) **Operating Voltage:** 12V
   (c) A displacement threshold of 1.5 cm^3/rev is chosen in line with the motor datasheet's minimum flow rate requirement, ensuring that the pump adapts dynamically to cooling demand.
   (d) A nominal shaft angular velocity of 4000 rpm is selected to balance flow rate and energy efficiency while maintaining effective cooling.
   (e) The nominal pressure gain of 2 bar ensures the pump can overcome resistance in the cooling circuit, delivering adequate pressure for smooth fluid circulation.
   (f) A nominal dynamic viscosity of 1 cP is chosen based on typical cooling fluids, ensuring low resistance and minimal pumping losses.
   (g) A volumetric efficiency of 0.92 minimizes internal leakage, ensuring that most of the displaced fluid contributes to effective cooling.
   (h) A mechanical efficiency of 0.88 reduces power losses while maintaining reliable pump performance.
   (i) A no-load torque of 0.1 N·m ensures that the pump starts efficiently without requiring excessive input torque.
   (j) A cross-sectional area of 0.000111 m^2 at ports A and B minimizes pressure drops while maintaining efficient fluid flow.

6. **Coolant Reservoir (Tank)**
The reservoir acts as a buffer to accommodate thermal expansion of the coolant and maintain system pressure:

   (a) **Design:** Cylindrical tank with a constant cross-sectional area.
   (b) **Dimensions:**
      i. Cross-sectional area: 0.02 m^2
      ii. Height: 120 mm
   (c) **Pressure:** Assumed to operate at atmospheric pressure to simplify the model and avoid the need for pressurization mechanisms.

5 Assumptions

To ensure a well-defined thermal model for the motor cooling system, the following assumptions have been made based on system constraints, component characteristics, and standard thermodynamic principles.

5.1 Motor Control Unit (MCU)

1. The maximum power loss in the MCU is assumed to be 4 kW, as no explicit heat loss equation is provided in the datasheet.
2. The specific heat capacity of the MCU is assumed to be 900 J/Kg. K, based on general values for electronic components and heat sink materials.
3. The cooling path length for the MCU is assumed to be 0.5 m, which represents the effective path through which heat is dissipated.

5.2 Motor (EMRAX 208)

1. All inefficiencies in the motor are assumed to be converted into heat loss, meaning no additional energy dissipation modes (such as sound or vibration losses) are considered.
2. The specific heat capacity of the motor is assumed to be 780 J/Kg. K, approximating common values for aluminum and copper components used in electric motors.
3. The cooling path length for the motor is assumed to be 0.5 m, representing the effective heat conduction path before reaching the coolant hose.

5.3 Pipe System

1. Total length of the coolant pipes is assumed to be 3 m, ensuring sufficient distance for effective heat exchange.
2. The inner diameter of the pipes is 12 mm, optimizing flow rates while minimizing pressure drops and material usage.

5.4 Ambient Conditions

1. The ambient temperature is assumed to be 298.15 K (25°C) to standardize thermal analysis under typical environmental conditions.
2. The system operates at atmospheric pressure (101.325 kPa), meaning no pressurization mechanisms are incorporated in the cooling loop.

5.5 Power Supply

A 12V DC power source is used to drive both the cooling fan and pump, ensuring compatibility with standard low-voltage automotive electrical systems.

5.6 Heat Generation Assumptions

1. The maximum heat dissipation is assumed to be 5.22 kW for the motor and 4 kW for the MCU, representing peak load conditions.
2. These values serve as worst-case scenarios for thermal performance evaluation.

5.7 Thermal Mass Representation

Both the motor and MCU are modeled as thermal masses with controlled heat flow rate sources, meaning their temperatures are determined by the balance of heat input (losses) and heat extraction (cooling efficiency).

5.8 Fluid Flow Assumptions

1. Turbulent flow is expected in the cooling circuit, as the estimated Reynolds number exceeds the laminar limit of 2300. This is due to relatively high flow velocity and narrow tubing diameters commonly used in compact thermal systems.
2. Flow behavior is modeled using standard Navier-Stokes equations with turbulence effects implicitly captured, though detailed turbulence modeling is not explicitly implemented to keep the Simulink model computationally efficient.

5.9 Air Properties for Heat Exchange

Standard properties of air at 25°C (298.15 K) are assumed for convective heat transfer calculations in the radiator-fan interaction. These include:

1. **Density (rho):** 1.184 kg/m^3
2. **Thermal Conductivity (k):** 0.0257 W/m·K
3. **Specific Heat Capacity (Cp):** 1005 J/Kg. K
4. **Viscosity (mu):** 1.846 × 10e-5 Pa·s

6 Control Strategy

6.1 Overview of the Control System

The cooling system for the motor is designed to regulate fan speed and volumetric flow rate based on motor temperature. This ensures:

1. **Optimized cooling efficiency** – The system provides cooling only when needed.
2. **Energy savings** – Reducing power consumption when cooling demand is lower.
3. **Extended component lifespan** – Minimizes unnecessary wear and tear.
4. **Improved reliability** – Ensures stable cooling performance across different conditions.

Instead of using fixed-speed fans or constant volumetric flow rates, we implemented a lookup table-based control system for controlling the volumetric flow rate of the coolant based on predefined values for temperature, and a PI controller for controlling the fan speed (via resistance).

6.2 How the Control System Works

1. **Fan Speed Control:**
   (a) A PI controller is installed to control the rheostat of the fan circuit.
   (b) Adjusted based on motor temperature using a 12V supply with a variable resistor.
   (c) Instead of controlling current directly, resistance is varied, which indirectly adjusts the fan speed.
2. **Volumetric Flow Rate Control:**
   (a) A lookup table is installed to control the variable displacement pump's flow rate.
   (b) Adjusted based on motor temperature to ensure proper cooling.
   (c) The volumetric rate changes between 6 to 11 l/min based on predefined breakpoints.

6.3 Lookup Table Implementation

Motor Temperature (K) $\rightarrow$ Volumetric Flow Rate (cm^3/rev)

Lookup Table Values:

1. Breakpoints (Motor Temperature in K): [310.15 313.15 315.15 318.15]
2. Volumetric Flow Rate (cm^3/rev): [1.5 2 2.5 2.75]

6.4 PID Controller Implementation

Motor Temperature (K) $\rightarrow$ Fan Circuit Resistance (Ω)

PID Control Parameters:

1. **Output Saturation Limits:** [25Ω (minimum) – 1000Ω (maximum)]
2. **PID Gains:**
   - Proportional Gain (K_p): 17
   - Integral Gain (K_i): 3
   - Derivative Gain (K_d): 0
3. **Controller Sample Time:** 0.1 s
4. **Control Mode:** Continuous PID (Simulink)

6.5 Rationale for Hybrid Control Approach: Lookup Table for Pump, PID for Fan

A **lookup table** was chosen for controlling the pump's volumetric flow rate due to its *simplicity, stability, and computational efficiency.* Unlike PID controllers, which require continuous tuning of proportional, integral, and derivative gains, a lookup table operates on predefined mappings, ensuring predictable and consistent behavior without the risk of instability or overshoot. It retrieves stored values instantly, avoiding the need for real-time calculations. This is particularly advantageous in embedded systems or simulation environments where computational load must be minimized. Additionally, lookup tables prevent the hunting behavior often observed with poorly tuned PID controllers, maintaining smooth transitions and energy-efficient performance.

However, **PID control** was selectively implemented for the fan because *air-side cooling requires faster dynamic response* compared to fluid-side systems. The fan directly affects surface cooling and must react quickly to sudden temperature spikes, especially during rapid acceleration or transient load conditions. In contrast, the pump's influence on motor temperature is governed by the thermal inertia of the coolant loop, which responds more slowly. This makes PID control unnecessary for the pump, but highly effective for the fan, where continuous and fine-grained adjustment improves thermal stability and responsiveness.

7 Subsystem Description

7.1 Input

The system requires two key inputs:

1. **Drive Cycle:** We have tested the model with 2 different drive cycles. The first drive cycle was as provided by the organizers (80s). The second version is an extrapolated version of the same drive cycle, for 1000s.

   Drive Cycle Original Link

   Drive Cycle Extrapolated Link

2. **Motor Efficiency:** Obtained using the Efficiency_Finder.m These are inputted into the system using 'From spreadsheet' block.
   To determine the efficiency, we use a custom MATLAB program, Efficiency_Finder.m, which compares the instantaneous torque and motor speed with the efficiency chart from the motor datasheet. The program generates efficiency values for each instant throughout the drive cycle, which are then stored in a spreadsheet and used as an input.

   The efficiency chart data can be obtained in the following

   Efficiency Chart Data Link

   The efficiency output data for the drive cycles can be obtained on the following:

   Link for Original Drive Cycle Efficiency

   Link for Extrapolated Drive Cycle Efficiency

7.2 Power Loss

This is essential for evaluating the motor cooling system by quantifying the power losses that contribute to heat generation.

7.3 Motor

The motor is modeled as a thermal mass with a controlled heat flow rate source, which takes input from the power loss calculated in the Power Loss Calculation block. The thermal pipe block is used to represent the motor's cooling jacket, simulating the heat dissipation through the cooling system.

7.4 Motor Controller Unit (MCU)

The motor controller is also modeled as a thermal mass but with a constant heat flow rate source of 4 kW. The thermal pipe block represents the cooling plate of the motor controller unit, facilitating heat transfer to the cooling system.

7.5 Cooling System

The cooling system consists of the following components:

1. **Tank:** Stores the coolant and maintains the required coolant volume in the system.
2. **Radiator:** Facilitates heat dissipation from the coolant to the surroundings.
3. **Fan:** Increases airflow through the radiator to enhance cooling.
4. **Pump:** Circulates the coolant through the system to maintain efficient heat transfer.
5. **Coolant Hose:** Connects various components and ensures proper coolant flow.

Both the pump and fan operate on a 12V power supply, and their power consumption is calculated based on operating conditions.

7.6 Pump and Fan Feedback System

A control system has been implemented using a lookup table, which adjusts the pump and fan operation based on system conditions to optimize cooling performance.

7.7 Output

This subsystem collects and displays the key results of the simulation, including:

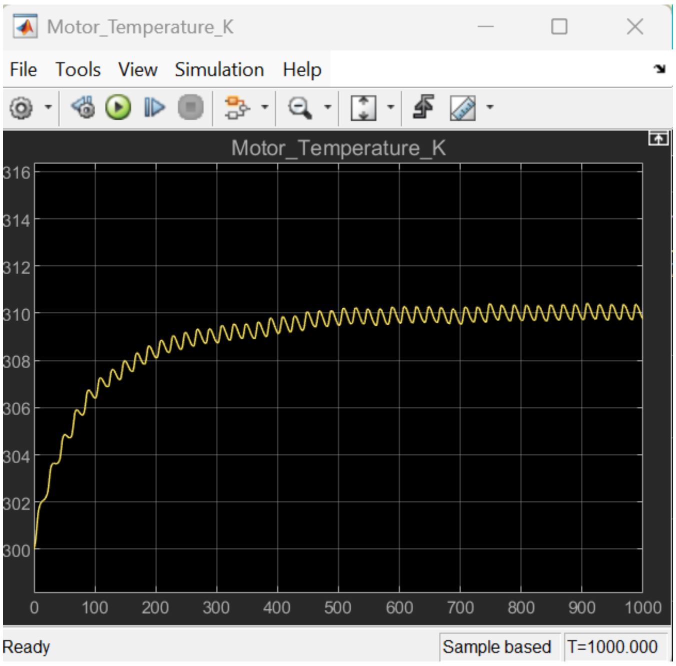

Figure 2: **Motor Temperature**

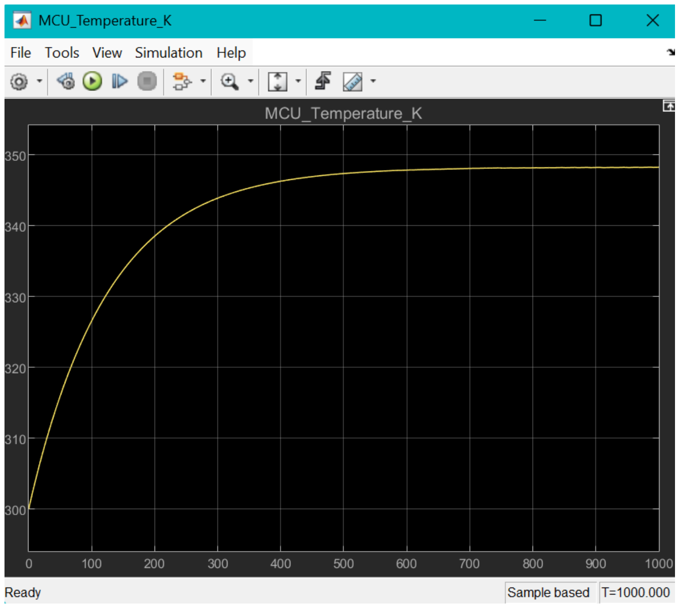

Figure 3: **MCU Temperature**

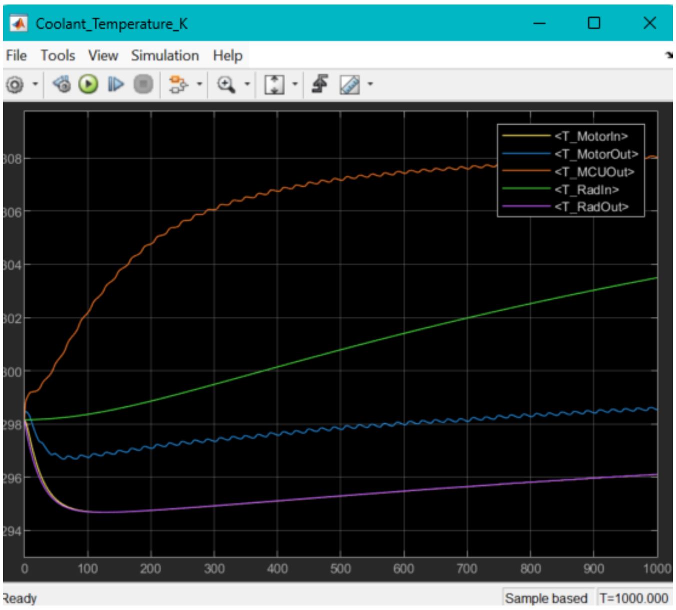

Figure 4: **Coolant Temperature**

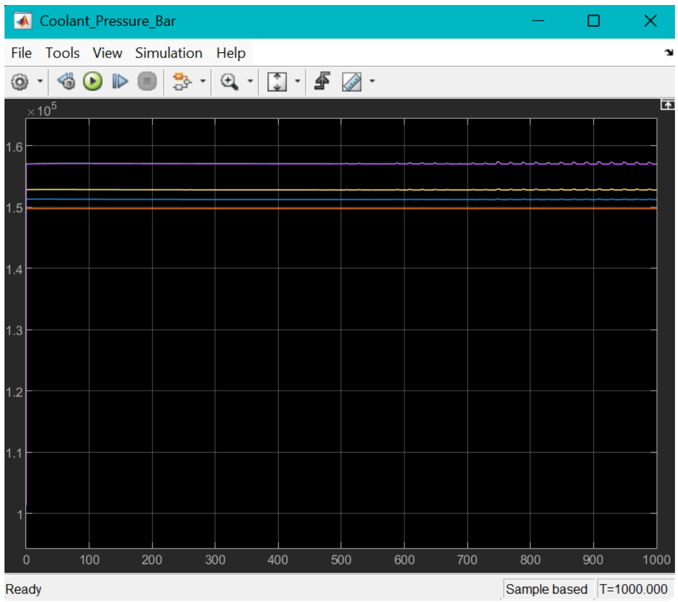

Figure 5: **Coolant Pressure**

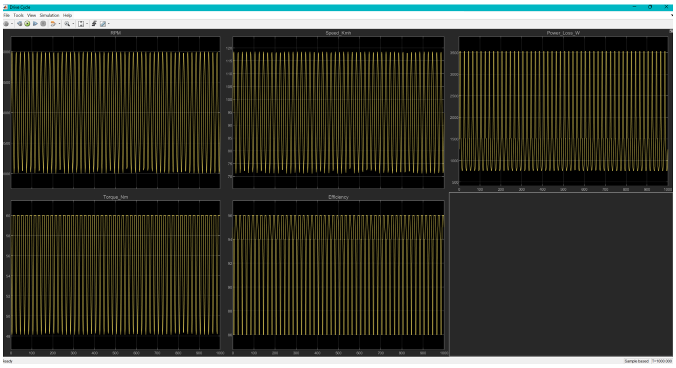

Figure 6: **Drive Cycle**

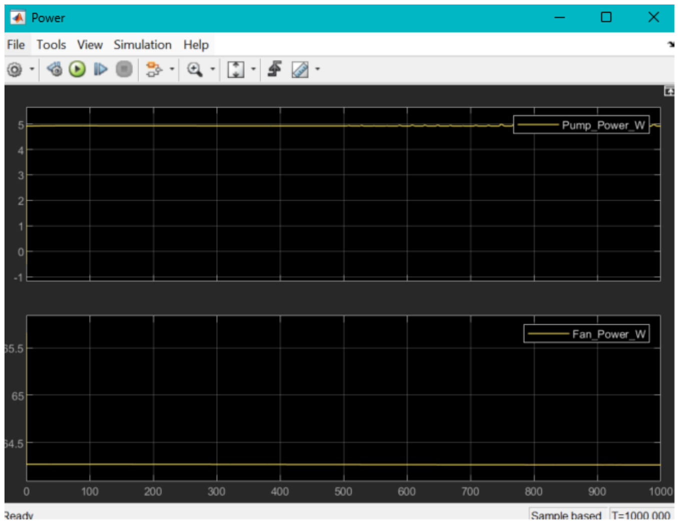

Figure 7: **Power**

These outputs help evaluate the performance of the motor cooling system under different operating conditions.

8 Cost Analysis

8.1 Capital Expenditure

1. Coolant Tank: Considering the total coolant volume in the system, a minimum 2L coolant tank is required.

- It amounts to a cost of **INR 600**

2. Pump: A coolant pump with a flow rate of 11 L/min, powered by 12V, is required.

- It amounts to a cost of **INR 3300**

3. Axial Fan: A radiator fan with the required airflow capacity 1800 CFM, powered by 12V power supply is required.

- It amounts to a cost of **INR 9,700**

4. Control System Components

- **Microcontroller:** Used to process temperature data and control the pump and fan based on the input from the temperature sensor. It amounts to a cost of **INR 450**
- **PWM Controller:** Controls the speed of the radiator fan and coolant pump by adjusting the voltage supplied to them. It amounts to a cost of **INR 150**
- **Relay or Motor Driver:** Switches the high-current pumps and fans on and off or provides motor control. It amounts to a cost of **INR 160**

  Hence, total cost for control system is **INR 760**.

5. Coolant Hose: A minimum of 3 meters of coolant hose is required.

- It amounts to a cost of **INR 1,650**

6. Radiator:

- **Tubes**
  - Tube length per segment = 0.35 m
  - Number of tubes = 45
  - Hence total tube length = $0.35 \times 45 = 15.75$ m
  - Tube inner diameter = 0.006 m
  - Tube thickness = 0.0015 m, Hence volume of tube required would be

$$V = \text{length} \times \pi \times \text{inner diameter} \times \text{thickness} \tag{7}$$

$$V = 15.75 \times \pi \times 0.006 \times 0.0015 \tag{8}$$

$$V = 445cm^3 \tag{9}$$

  - Material used for tubes: **Aluminium**

- Density of aluminium = $2.7 g/cm^3$ Hence, Mass of aluminium required:

$$M = 445 \times 2.7 = \mathbf{1.20\ kg} \tag{10}$$

- As per the London Metal Exchange, the cost of Aluminium metal is INR 217.5 per kg. Hence, Total cost =

$$1.20 \times 217.5 = \mathbf{INR\ 216.0} \tag{11}$$

- **Fins**
  - Number of fins = 45
  - Total fin surface area facing the air = $0.956\, m^2$
  - Total fin surface area facing the coolant (inside the tubes) = $0.390\, m^2$
  - Material used for fins: **Aluminium** (Light and cost-effective)
  - Assuming fin thickness = 1mm, Volume = Surface Area × thickness

$$V = (0.956 + 0.390) \times 0.001 = 1346\, cm^3 \tag{12}$$

  - Density of aluminium = $2.7\, g/cm^3$. Hence, Mass =

$$1346 \times 2.7 = \mathbf{3.63 \quad kg} \tag{13}$$

  - As per the London Metal Exchange, the cost of Aluminium metal is INR 217.5 per kg. Hence, Total cost =

$$3.63 \times 217.5 = \mathbf{INR\ 790.7} \tag{14}$$

Hence, total cost for radiator = Cost for Tubing + Cost of Fins

$$\mathbf{INR\ 261 + INR\ 790.7 = INR\ 1{,}051.7} \tag{15}$$

Given the above considerations, the cost breakdown and total cost comes to a sum of

$$\mathbf{INR\ 1{,}051.2} \tag{16}$$

Component	**Cost**
Coolant Tank	600
Pump	3,300
Axial Fan	9,700
Control system	760
Coolant Hose	1,650
Radiator	1,051.2
Total	**17,061.2**

Table 1: Cost breakdown of components

9 Conclusion

This report presents the design and simulation of an efficient motor cooling system for a Formula Student Electric Vehicle (FSEV) utilizing an EMRAX 208 in-wheel motor and BAMOCAR-D3 motor controller. The system is engineered to manage peak heat dissipation of 10 kW while maintaining component temperatures within safe operational limits.

The design approach incorporates a single-loop liquid cooling system with water as the coolant, optimized for efficient thermal regulation. The integration of a radiator, fan, pump, and coolant reservoir ensures effective heat dissipation while minimizing energy consumption. A lookup table-based control strategy is implemented in MATLAB/Simulink, dynamically adjusting fan speed and coolant flow based on motor temperature to enhance efficiency and prevent thermal overload.

Through detailed thermal modeling, fluid dynamics analysis, and control strategy implementation, the proposed cooling system effectively maintains motor and controller temperatures within safe limits under dynamic driving conditions. The results validate the system's capability to ensure reliable performance, energy efficiency, and component longevity, making it a viable solution for high-performance electric mobility applications.