Race Car Braking System Thermal Model for Real Time Use in a Driving Simulator

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Abstract

Vehicle performance optimisation is the primary target in the development of race cars and road hypercars. Considering the complexity of modern vehicles, a holistic approach to analyse the interaction of vehicle dynamics, powertrain cooling systems, brake cooling, and human drivers in the same simulation can be crucial to maximise the overall performance (*Bouvy et al., 2012*). This paper is a follow up of a previous article written by Dallara and Claytex *("Race Car Cooling System Model for Real Time use in a Driving Simulator", Stellato et al., 2023*), which describes a one-dimensional (1D) cooling system model integrated with a vehicle multibody model, used in the Dallara dynamic driving simulator (Figure 1). This collaboration has led to a further step, where Dallara has developed a model to optimise the brake cooling of its vehicles, with Claytex's VeSyMA suite used for the auxiliary vehicle systems and code compilation. The model was validated through the comparison with real-vehicle data, demonstrating good accuracy in i.) sizing a race car braking system; and ii.) providing a refined assessment of the global vehicle performance on the driving simulator.

Keywords: brakes cooling, cooling systems, real-time, driving simulator

1 Introduction

Optimising vehicle performance is a primary objective in the development of high-performance vehicles. A holistic simulation approach that integrates key component models, such as powertrain and brake cooling systems, with vehicle dynamics and driver behaviour, is essential for maximising overall performance. With this regard, in a previous paper *(Stellato et al., 2023),* a car cooling system model was integrated inside the vehicle multi-body model. This integration allowed the definition of the best cooling system architecture for every project, targeting the trade-off to maximise the vehicle performance, especially in a professional driving simulator with a human driver. As a further step, the previous model was extended with the braking system thermal model, to predict the braking system's main components' temperatures (disc and caliper) and neighbouring components (disc bells and rims). As for the cooling system, this integration allows the evaluation of several layouts for air intakes, brake ducts, disc, and caliper size in different boundary conditions, finding the best configuration for aerodynamic, handling, and cooling targets, based on the vehicle mission and the driving style.

In the first design phase, the simulation model may run offline, using vehicle speed and brake pressure profiles coming from:

- Track data acquisition, if the activity involves an existing vehicle update, and thus, there is already data available.
- Quasi-static lap time simulations, in the case of designing a new vehicle.

Then, in a more advanced phase, it can be integrated directly into the Dallara dynamic driving simulator to obtain more accurate and precise feedback from the driver.

In the remainder, this article is organised as follows: Section II presents the Claytex VeSyMA suite utilised in the project; Section III delves into the proposed models; Section IV discusses the main results; and Section V summarises the main conclusions.

Figure 1: Dallara dynamic driving simulator

2 Claytex VeSyMA suite

The VeSyMA (**Ve**hicle **Sy**stems **M**odelling and **A**nalysis) simulation suite was used to form the backbone of the Dallara vehicle model. Comprising various subject-specific Modelica libraries built on top of a common base, the VeSyMA suite was originally developed by Claytex using Dymola and Modelica. More than 25 years of experience in the motorsport world supporting customers from Formula 1, IndyCar and NASCAR with Dymola and Modelica simulations was drawn upon in the creation of VeSyMA (*Hammond-Scott and Dempsey, 2018*).

Conceived and built in a hierarchical structure, all subject-specific libraries (such as Suspensions for Vehicle Dynamics analysis, Motorsport for motorsport-specific requirements, PTDynamics for the simulation of drive trains and gearboxes, etc.) inherit from the base VeSyMA library. This fully leverages the concepts of model reuse and inheritance present in the object-oriented Modelica language, helping users to efficiently manage their own simulation libraries.

The cornerstone of model reuse in the VeSyMA suite is the utilisation of the open-source Vehicle Interfaces library. All vehicle models found within VeSyMA are built from templates located in the VeSyMA library, which are built up of component templates from Vehicle Interfaces. Integrating third-party libraries built on the same open-source Vehicle Interfaces library into VeSyMA base vehicle models is thus a simplified task. Such a feature is of use in a study such as this, as the implementation of customer-developed libraries into VeSyMA vehicle models is straightforward.

Beyond templates, the VeSyMA suite contains packages of commonly found vehicle components, with simplified versions found in the VeSyMA library, and detailed versions found within the subject-specific libraries. Scalability of the detail in a model is therefore possible by selecting which component is used, which is of benefit as it enables vehicle models to be configured to focus on the detail where required. This can be done both for intellectual clarity and efficient use of computational resources. All component and vehicle models found in the VeSyMA suite have been built from the outset with performance in mind, with optimisations made in key areas such as state selection.

3 The Model

Figure 2 shows the cooling and braking system models connected to the vehicle model and the human driver. The interface with the multi-body vehicle model allows further dynamic analysis to improve vehicle performance.

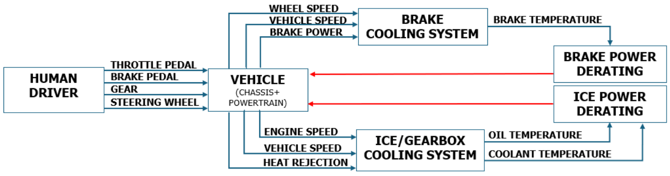

Figure 2: Cooling and braking system interactions with vehicle model and human driver

Regarding the cooling system model, the coolant and oil temperatures are defined as model outputs. Therefore, they affect the internal combustion engine (ICE) power curve according to the derating strategies agreed with the engine supplier. Instead, for the braking system model, the outputs are the disc and caliper temperatures, which will affect the friction coefficient inside the vehicle model. A temperature increase will decrease the coefficient, forcing the driver to increase the braking distance.

3.1 Architecture and Physics

Figure 3 shows the components of the braking system model and describes how the heat is propagated among them.

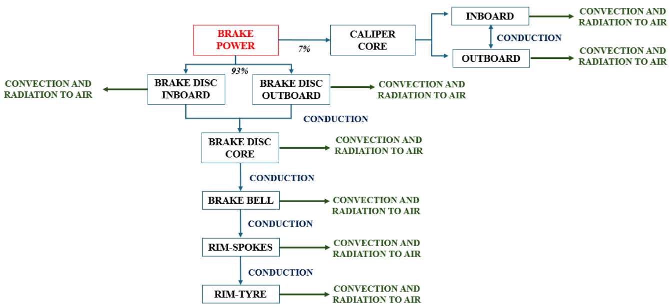

Figure 3: Heat fluxes in the simulation model

The first block is the "Brake Power", where the brake power, and thus the overall system heat rejection, is calculated by the following equations at each time step:

$$P_B = \mu \, F_n \, R \, \omega \tag{1}$$

$$R = \frac{R_e + R_i}{2} \tag{2}$$

$$F_n = p \, A \tag{3}$$

P_B = brake power [W]
μ = friction coefficient [-]
F_n = disc normal force [N]
R = disc mean radius [m]
ω = wheel angular speed [rad/s]
R_e =disc external radius [m]
R_i =disc internal radius [m]
p = brake pressure [Pa]
A = brake pistons area [m^2]

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A 7% fraction of the heat rejection produced is assigned to the caliper core. This fraction is exchanged to the inboard and outboard areas through conduction and finally dissipated by convection into the environment. This percentage was estimated after several validation processes on different types of Dallara vehicles (race cars, hypercars, road cars). To align the track data with the simulation model, the values fell in the range of 6.5% to 7.5%. This matches well with both the standard estimation of 5-10% and the values reported in recent studies (*Vdovin and Sebben, 2018*). As a further step in the model, the implementation of a disc rotational model in contact with the pads could be considered, where this percentage will be given as output of the boundary conditions.

The remaining 93% of heat rejection is assigned to the inboard and outboard brake disc area, equally distributed. The heat is then conducted to the disc core, and thereafter, it passes through the brake bell, rim-spokes and rim-tyre. All these components dissipate the heat by convection and radiation into the environment.

To evaluate the temperature of each component, the first principle of thermodynamics is applied. First, the heat transferred to the component is calculated. This is the instantaneous difference between the heat rejection (HR), produced by the braking system or exchanged from the neighbouring component by conduction, and the heat dissipated by convection to air (HD):

$$HF = HR - HD \tag{4}$$

HF = overall heat flux [W]
HR = heat rejection [W]
HD = heat dissipation [W]

At this point, the component temperature can be calculated at each time step by:

$$T_{t1} = T_{t0} + \frac{HF\,(t_1 - t_0)}{C} \tag{5}$$

T_{t1} = temperature at time t_1 [K]
T_{t0} = temperature at time t_0 [K]
C = thermal capacity [J/K]
t = time [s]

The thermal conduction is evaluated by the following equation:

$$\dot{Q}_{cond} = K\,\frac{A}{d}\,\Delta T \tag{6}$$

$\dot{Q}_{cond}$ = thermal conduction [W]
K = conductivity coefficient [W/mK]
A = component reference area [m^2]
d = component reference length [m]
ΔT = delta temperature [K]

While the thermal convection is given by:

$$\dot{Q}_{conv} = HTC\,(T - T_{air}) \tag{7}$$

$\dot{Q}_{conv}$ = thermal convection [W]
HTC = heat transfer coefficient [W/K]
T = component temperature [K]
T_{air} = air temperature [K]

3.2 Parameters

The model input parameters are obtained from computational fluid dynamics simulations (CFD), experimental tests, and CAD data (computer-aided design). In detail:

- The heat transfer coefficient (HTC) for each component (disc, caliper, brake bell, rim-spokes and rim-tyre) is the output of CFD simulations that consider both convection and radiation. The HTC is a function of vehicle speed and component heat rejection: for each component, 9 different HTCs are evaluated at 3 different speeds (100, 180, and 300 km/h) and 3 heat rejection rates (minimum, average, and maximum). As an example, Table 1 illustrates an HTC map for one component. The instantaneous value in the 1D simulation is then extrapolated or interpolated from these tables. Figure 4 shows the outputs of the CFD simulations, where the variability of HTCs can be appreciated: for the inboard disc surface, close to the air duct, the values are practically doubled with respect to those of the outboard.

HTC [W/K]		Heat Rejection [kW]		
		2.4	7.2	12
Vehicle speed [km/h]	100	7.52	6.96	7.2
	180	11.6	10.56	10.16
	300	16.72	15.52	14.56

Table 1: HTC map for 3 different heat rejections and vehicle speeds

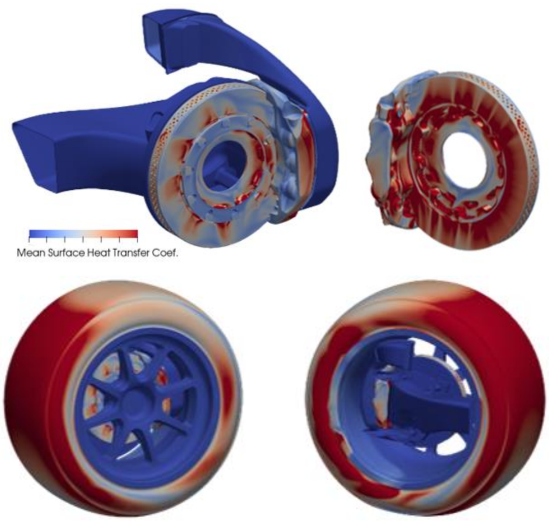
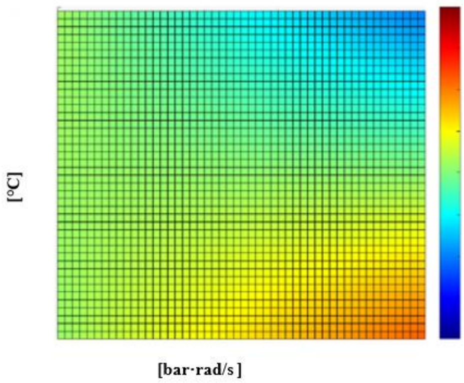


Figure 4: HTCs values from a CFD simulation

- The brake friction coefficient is provided by the manufacturer and is described in a table as a function of surface temperature and brake pressure. Figure 5 reports an example: it has to be reported that the unit of measurement of the x-axis is [bar·rad/s], as the map takes into account also the disc angular speed.



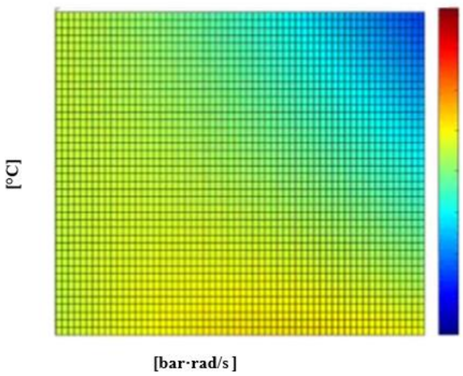


Figure 5: Brake friction coefficient maps for front brakes (top) and rear ones (bottom)

- The component's physical properties, such as density, emissivity, conductivity, and specific heat, are obtained from the supplier's datasheets. The evaluation of the reference area and length for the calculation of the heat exchanged by thermal conduction is done by direct analysis of CAD data. The main parameters for each component are summarised in Table 2. An interesting note concerns the rim. To have a better discretisation, it has been divided into two parts: rim-spokes and rim-tyre. The first part considers the spokes and the area close to the brake bell. The other one is the part close to the tyre (Figure 6). Each part has its own HTCs.

Figure 6: Rim-tyre part

CALIPER		BRAKE BELL	
Core Mass	[kg]	Mass	[kg]
Inboard/Outboard Mass	[kg]	Heat Coefficient	[J/kgK]
Heat Coefficient	[J/kgK]	Initial Temperature	[K]
Initial Temperature	[K]	Material	
Material		Conductivity	[W/(mK)]
Conductivity	[W/(mK)]	Density	[kg/m^3]
Density	[kg/m^3]	Emissivity	[-]
Emissivity	[-]	Reference Area	[m^2]
Reference Area	[m^2]	Reference Length	[m]
Reference Length	[m]	**RIM-SPOKES**	
Piston Area	[m^2]	Mass	[kg]
DISC		Heat Coefficient	[J/kgK]
Core Mass	[kg]	Initial Temperature	[K]
Inboard/Outboard Mass	[kg]	Material	
Disc Heat Coefficient	[J/kgK]	Conductivity	[W/(mK)]
Surface Initial Temperature	[K]	Density	[kg/m^3]
Core Initial Temperature	[K]	Emissivity	[-]
Material		Reference Area	[m^2]
Density	[kg/m^3]	Reference Length	[m]
Emissivity	[-]	**RIM-TYRE**	
Conductivity	[W/(mK)]	Mass	[kg]
Reference Area	[m^2]	Heat Coefficient	[J/kgK]
Reference Length	[m]	Initial Temperature	[K]
External Radius	[m]	Material	
Internal Radius	[m]	Conductivity	[W/(mK)]
Thickness	[m]	Density	[kg/m^3]
		Emissivity	[-]
		Reference Area	[m^2]
		Reference Length	[m]

Table 2: Components geometrical and physical properties

Overall, if the model is executed offline, e.g., outside the Dallara simulator, it requires input data for the brake pressure at each caliper, the vehicle speed, and the angular velocity profiles of each wheel during a representative track lap. Otherwise, if deployed online in the Dallara dynamic driving simulator, the human driver and the vehicle model will directly provide these inputs.

3.3 Real-time Features

In terms of real-time setup, the proposed model follows the same methodology as the previous paper on this subject (*Stellato et al., 2023*). Dymola's fixed-step Euler solver was employed, running at a sampling time of 1 ms. Preliminary testing of the model was undertaken on a 2.8 GHz I7 processor SSD-equipped desktop workstation; successful real-time simulation on this machine ensured the model would be suitable when deployed on the more powerful hardware associated with Dallara's DiL simulator.

The VeSyMA – Driver-in-the-Loop library was utilised for the compilation of the vehicle and cooling models into a real-time app capable of running on the Dallara simulator. A 2^{nd} order Runge-Kutta explicit solver method was used for the real-time application, with advanced inline settings (*Dymola manual*) contributing to a robust and high-performance real-time model (*Hammond-Scott and Dempsey, 2018*).

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Regarding the brake thermal model in real-time use, some first-order filters (with a 5 ms time constant) were added when solving non-linear equations.

4 Virtual Validation

4.1 Offline Simulation

For any simulation activity, the validation phase represents the crucial moment to assess the correctness of the model. For this work, it consisted of the following steps:

- Implementation of the braking system architecture of an existing vehicle.
- Offline simulation using measured vehicle speed, brake pressure and ambient temperature as input data.
- Comparison between the simulated temperature profiles and the measured ones on the track.

Figure 7 shows the comparison between the simulated disc temperature profile and the real logged data of a hypercar with more than 1000 horsepower at Vallelunga track. The average difference between the measured temperature and the simulated one is ~12°C, with a root mean square error (RMSE) of ~15 °C, and a mean relative error of ~2%. This level of accuracy, given the high temperature range profile (from 400 to 780 °C), is considered acceptable for sizing the braking system and analysing its performance. Similarly, Figure 8 shows the comparison for the caliper temperature profile. In this case, the average temperature difference between the measured and the simulated one is ~3.3°C, the RMSE is ~3.9 °C and the mean relative error is ~2.7%.

Overall, the accuracy of the model is considered acceptable for sizing a race car braking system and doing a refined assessment of the global vehicle performance on the driving simulator.

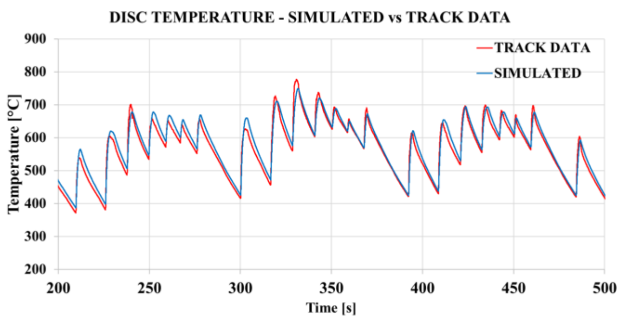

Figure 7: Simulated disc temperature vs real logged data

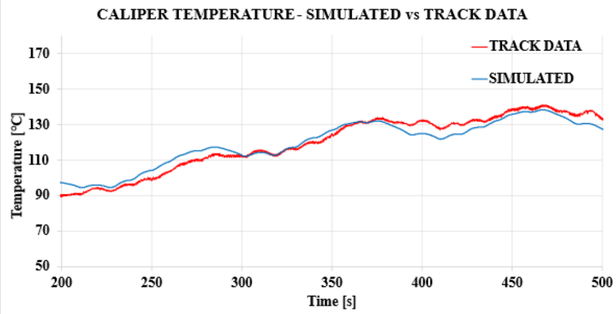

Figure 8: Simulated caliper temperature vs real logged data

4.2 Real-time Simulation in the Dallara Dynamic Driving Simulator

The same model was also deployed directly on the Dallara Driving Simulator, where the human driver performed a series of laps at maximum vehicle performance on the Zandvoort track. From this experimental campaign, the main target is the evaluation of:

- Disc temperature before the braking point. This is a value assessed by the supplier to preserve the disc and avoid excessive wear of the component.
- Friction coefficient reduction (due to the increased disc temperature), and thus the brake power.

The following figures show the results of a simulator session, where the driver performed a run (a total of 8 laps) at maximum performance to test the braking consistency and effectiveness. In Figure 9, the front left (FL) and front right (FR) brake disc outboard temperatures are plotted: after 400 seconds, the peak temperature stabilises between 900 and 1000 °C.

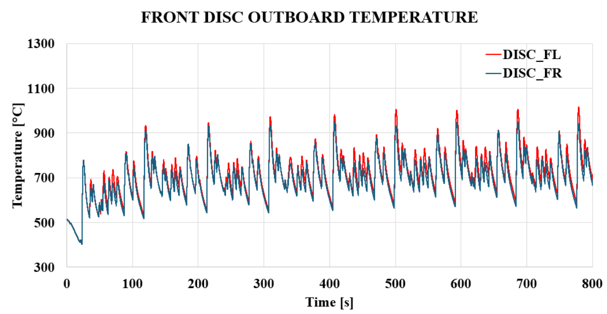

Figure 9: Front discs outboard temperature in a driving simulator session

It can be noted that the left disc is warmer than the right one in the heaviest braking zone. This is attributable to the different brake pressure values for these turns (Figure 10), caused by the ABS (anti-lock braking system). However, from the driver's point of view, this temperature imbalance did not affect the performance.

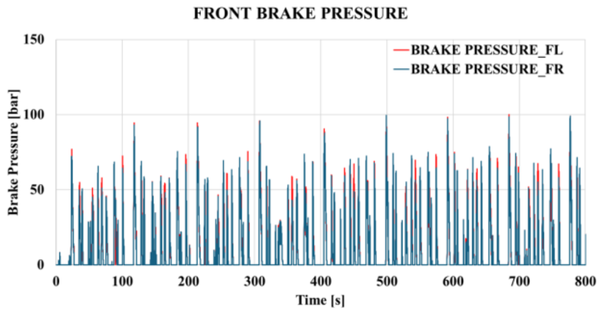

Figure 10: Front brakes pressure in a driving simulator session

The most interesting part turns out to be the friction coefficient trend. As mentioned in the previous paragraph, inside the model, it is discretised as a look-up table, function of brake pressure and disc temperature. Figure 11 shows the actual friction coefficient during the whole run. As expected, increased temperature leads to overall friction degradation, therefore, a reduced braking performance. In the track's hardest braking zone, the disc temperature reaches 1000°C, and the coefficient is reduced by more than 20% from its optimum value.

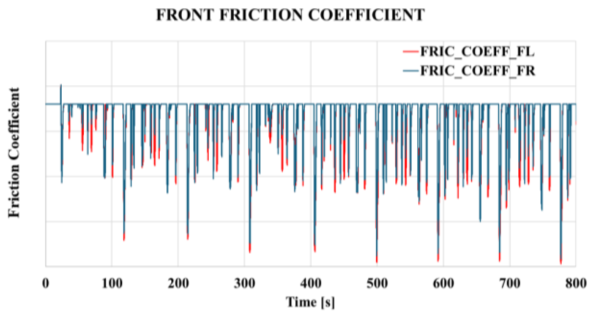

Figure 11: Front brakes friction coefficient in a driving simulator session

The same analysis can also be done for the rear brakes, where the disc outboard peak temperature is approximately 350 °C lower than the front ones (Figure 12). Also, in this case, the left side is slightly warmer than the right one, but with a smaller gap than before.

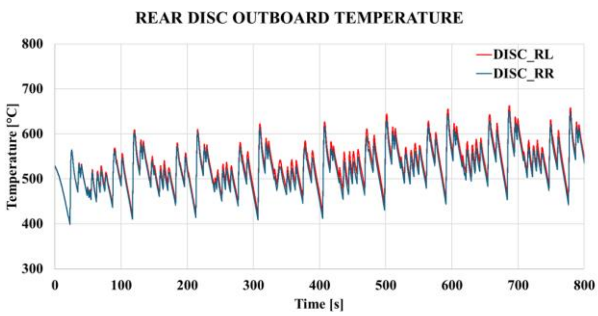

Figure 12: Rear discs outboard temperature in a driving simulator session

However, the mean lower temperature results in a different friction coefficient trend. If for the front brakes, its value varies over the lap course, then for the rear ones it is practically constant, with only two turns leading to a reduction of about 10 % (Figure 13).

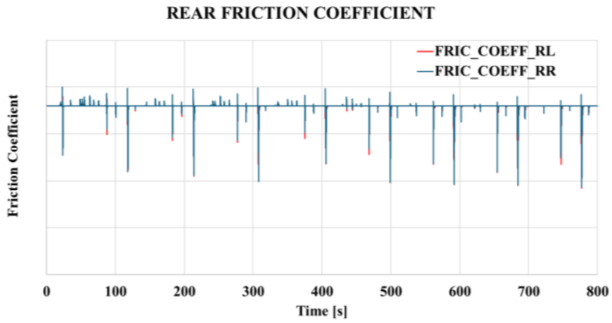

Figure 13: Rear brakes friction coefficient in a driving simulator session

The driving simulator and this simulation model can also be useful in preparing for race weekend by evaluating the ideal air duct size, if necessary. Several blanking percentages for both front and rear brakes can be tested, accounting for different ambient temperatures and race conditions (qualifying, traffic, free air, etc.). So, based on the situations they face, track engineers have a precise assessment of the necessary blanking, which ensures the best trade-off between performance and reliability.

5 Conclusions and Further Developments

The proposed simulation model, as developed and validated, is currently used at Dallara to size the braking system and analyse the vehicle performance on the driving simulator for different tracks, driving styles and boundary conditions (brake size, materials, aero configurations, ambient temperature). As all these parameters affect the overall results, the simulation model is useful to evaluate each effect to develop the best braking system architecture for every project, targeting the maximisation of vehicle performance (*Stellato et al., 2023*). The duty cycle analysed for the case study is a track lap at maximum performance, but additional critical driving cycles can be studied on the simulator, such as the brake warm-up during the out lap. Certainly, a key aspect turns out to be the friction coefficient sensitivity. The temperature calculation allows the thermal effect to be considered, defining the ideal design for the cooling ducts in each track. However, the variable friction coefficient makes it possible to evaluate the impact on lap time and vehicle performance. For example, in some tracks, the air duct size could be decreased to reduce the aerodynamic drag. Consequently, the temperature will increase, and the friction coefficient will decrease, resulting in worse

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braking performance. However, thanks to the improved aerodynamics, the lap time will improve. This scenario represents the typical trade-off that is analysed by the simulation model.

Further refinement and improvements are planned for the simulation model. Currently, the thermal discretisation stops at the rim. The next step will be to predict the thermal conduction from the rim to the tyre and integrate the tyre thermal model. This latter feature will further expand the holistic approach and enable a greater optimisation of vehicle performance by considering the temperature effect on the tyre's lateral and longitudinal grip and its correlation to the brake thermal performance.

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