Race Car Cooling System Model for Real Time use in a Driving Simulator

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Abstract

Powertrain performance optimization is one of main targets in racecar and road hypercar development. A key activity needed for both endothermal and electric powertrains is the cooling system sizing through simulation to make sure that the temperature limits are not exceeded in the most aggressive conditions minimizing or avoiding power derating. This article describes the implementation of a 1D cooling system simulation model integrated with a vehicle multibody model to be used in real time in the Dallara dynamic driving simulator with human driver. This activity is the result of a collaboration between Dallara which uses the model implemented to develop and optimize the cooling system architecture of its vehicles, and Claytex who develop the libraries used to generate these simulation models. The model has been validated through comparison with real data of an existing vehicle yielding a RMSE of 1.0 °C.

Keywords: cooling system, fluids temperatures, powertrain, derating, real time, simulator.

1 Introduction

Considering the complexity of current vehicles, a holistic approach to analyse the interactions of vehicle dynamics, cooling system dynamics and human drivers in the same simulation can be the key to maximize the overall performance of a high performance car as demonstrated already for many years (Bouvy et al, 2012).

These needs have led to the development of a 1D modular cooling system simulation model which has two main targets:

- To develop for each vehicle project the cooling system solution which has the best trade off among aerodynamics, packaging, weight and motor power (ICE or electric) to reach the max vehicle performance.
- To support the detailed vehicle performance analysis on the Dallara driving simulator (Figure 1) with different boundary conditions such as ambient temperatures, initial oil and coolant temperatures, human drivers.

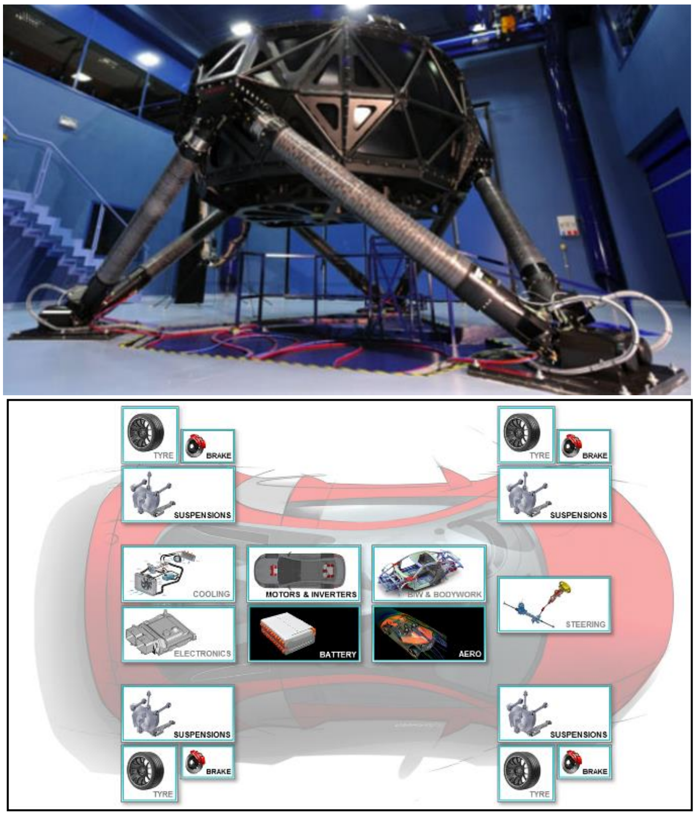

Figure 1. Dallara dynamic driving simulator (top), vehicle model diagram (bottom)

The case study illustrated in this paper refers to an ICE vehicle. Starting from the described architecture it can be modified in a user-friendly way, to develop a cooling system layout for electric and hybrid vehicles.

2 The Library

The Modelica libraries developed at Claytex are designed to be able to include significant levels of model details whilst retaining very robust and efficient simulation performance characteristics. This is done via careful and meticulous development strategies where it's ensured that the models achieve the best possible simulation efficiency from full scale system models right down to component test level scenarios. The Claytex library contains common models that are used in the wide range of libraries developed over the past 25 years.

2.1 Thermofluids

The Claytex.Fluid library (figure 2) is suited to real time performance for HiL (hardware in the loop) and DiL (driver in the loop) applications. It was first applied to driver in the loop models in a demonstrator in 2013 where the whole F1 vehicle according to the new 2014 specs was developed and simulated to test and address a broad range of questions.

The Claytex.Fluid library is based on Modelica.Fluid and the Claytex.Media library is based on Modelica.Media with some streamlining of the models for specific applications and fluid types found in cooling and lubrication. A large part of its development is driven by customer requirements, especially in terms of data input requirements for model parameterisation. This is a fundamental advantage as it is therefore tailored to take in the datasets that are available without requiring extensive calibration of unknown parameters to match test data.

Because Claytex.Fluid is compatible with Modelica.Fluid, this broadens the range of components the user has access to for building the fluids system models. Below is a snapshot of the top level packages within Claytex, Claytex.Fluid and Claytex.Media.

The modelica libraries used to represent the vehicle model are also developed by Claytex. The VeSyMA – Motorsports library was used (Claytex). This library provides all the required suspension and chassis related components to allow the user to build a full multi-body suspension vehicle model based on the customisable VeSyMA model vehicle architecture templates (Claytex; Hammond-Scot et al, 2018). In a similar way to the fluids and media libraries which have been described, all models have been designed with mathematical and numerical efficiency in mind whilst still capturing all the required details. The VeSyMA - Motorsports library is used within motorsports series such as Formula 1, NASCAR and IndyCar including driver in the loop applications and lends itself very well to the study presented in this paper (Dempsey, 2016).

One of the key components in the modelling were the heat exchangers. The Heat Exchanger models take in tabular data from the user table of Gc values (Heat transfer area * heat transfer coefficient) in W/K as a function of the actual mass flow rates (kg/s) on the primary side and secondary side of the heat exchanger. This value is then multiplied by dT, the temperature difference between the primary and secondary side inlets to yield the heat flow between the primary and secondary fluid.

$$HD_{rad} = A \cdot h \cdot dT$$

HD_{rad} = radiator heat dissipation [W]
A = heat transfer area [m^2]
h = heat transfer coefficient [$W/(m^2.K)$]
dT = temperature difference of inlet primary and inlet secondary [K]
A·h = f(m_flow_primary, m_flow_secondary)

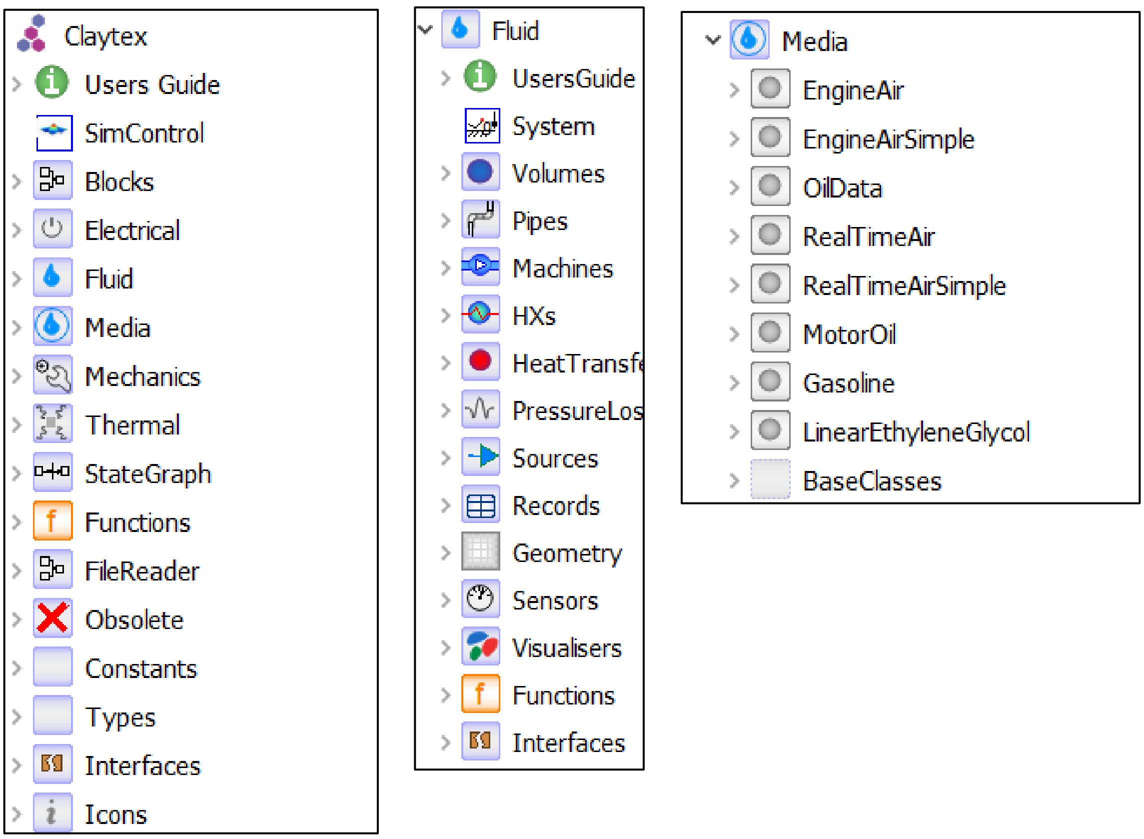

Figure 2. Claytex (left), Claytex.Fluid (middle) and Claytex.Media (right) library structure.

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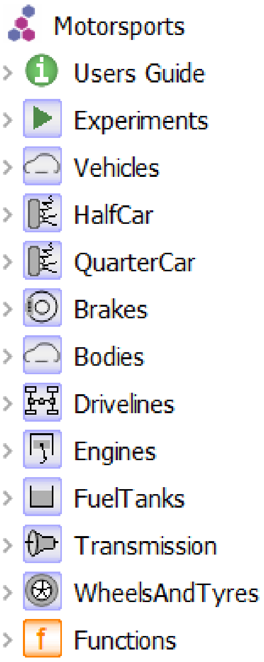


Figure 3. VeSyMA – Motorsports library top level package structure

3 The Model

Figure 4 shows the cooling system model connected to the vehicle model and human driver. The ICE power curve of the vehicle case study in this paper is affected by coolant and oil temperatures which are output of the cooling system model according to the graph reported in figure 5. The heat rejections and airflow across the radiators are function of vehicle speed, throttle and engine speed which are output of vehicle model.

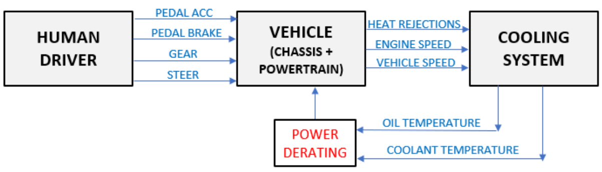


Figure 4. cooling system interactions with vehicle model and human driver

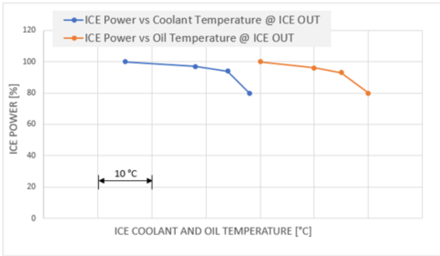


Figure 5. ICE power derating

Coolant and oil flow rates are consequences of pumps head curves, instantaneous engine speed and pressure drops for all components (ICE, pipes, heat exchangers). The fluids (coolant and oil) heat transfer and hence temperatures are calculated by the following equations at each time step.

$$HF_n = HR_{ICE_n} - HD_{rad_n}$$

$$T_{coolant_n} = T_{coolant_{n-1}} + \frac{HF_n \cdot (t_n - t_{n-1})}{TC}$$

HF = Heat Flow [W]
HR_{ICE} = ICE Heat Rejection [W]
TC = Thermal capacity [J/K]
t = time [s]

HF is the instantaneous difference between the heat rejection produced by the engine and the heat dissipation of the radiators. The thermal capacity [J/K] takes into account the total coolant volume, components (ICE, radiators, pipes) materials and weight.

3.1 The modular architecture

The model is composed of multiple air/coolant radiators, air/oil coolers and coolant/oil heat exchangers in a modular way allowing to study different cooling system configurations where coolant and oil are cooled together in a single loop or separately in more loops. In each loop air/fluid radiators and coolant/oil heat exchanger can be arranged in series or in parallel connection.

In figure 6 is reported the cooling scheme of the Dallara ICE vehicle case study of this paper, which is composed by two air/coolant radiators and one coolant/oil heat exchanger.

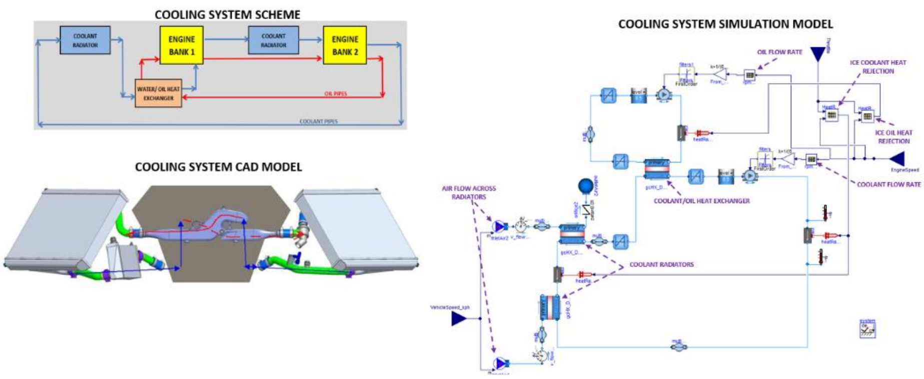


Figure 6. Cooling system of a Dallara vehicle

3.2 Parameters

The model parameters are obtained from CFD simulations and experimental tests:

- The air flow ratio across the radiators is the output of dedicated CFD simulations which take into account all the vehicle geometry or wind tunnel test (Figure 7).
- The ICE heat rejection is provided by the engine manufacturer as function of throttle and engine speed (Figure 8).
- Air/coolant radiator and coolant/oil heat exchanger efficiency has been measured by experimental tests on the Dallara cooling test rig (Figure 9).

Many coolant radiator specifications have been analysed. The chosen radiator specification is the result of the best trade off among air pressure drops, coolant pressure drops and heat dissipation which ensure the optimal cooling operating point for the case study.

The thermal behaviour of an ICE is defined by its capacities, heat transfer and thermal conductivities as well as its surrounding conditions (Morawietz at Al, 2005). The ICE heat capacity is modeled with a lumped thermal element storing heat. This parameter together with the amounts of coolant and oil volumes plays an important role as it affects the thermal inertia and therefore the time before reaching the maximum temperature values (Stellato et al, 2017).

The more coolant in the system and the higher heat capacity thus allowing higher vehicle accelerations for longer periods of time, before the available ICE power decreases for the derating.

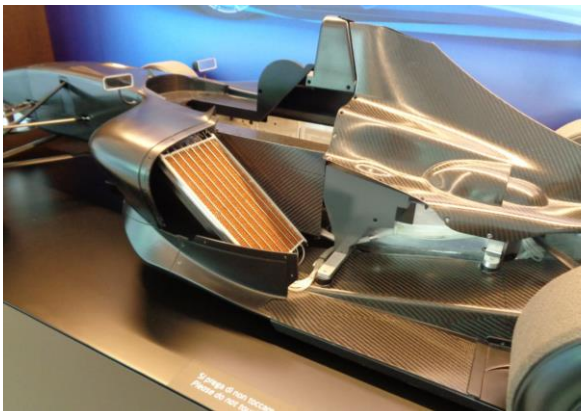


Figure 7. Typical wind tunnel vehicle test physical model

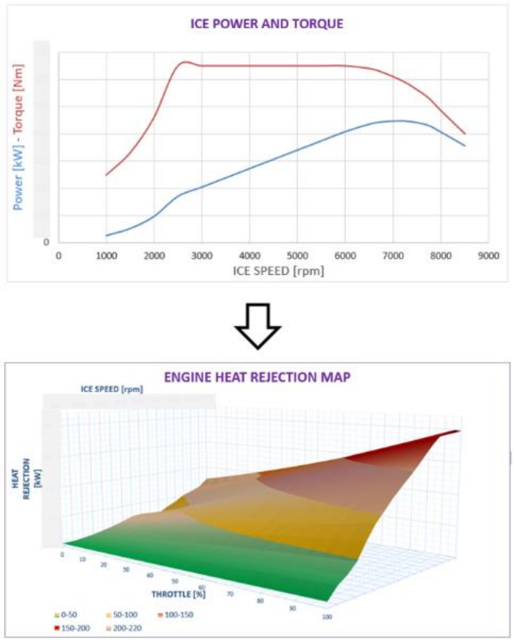
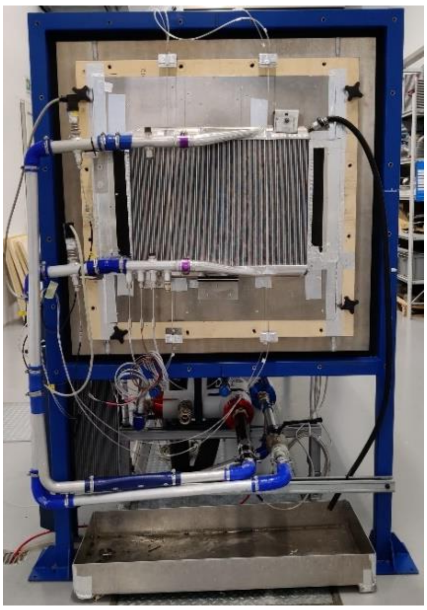


Figure 8. ICE power and heat rejection maps



INPUT - SINGLE PASS - H2O RADIATOR - ROLLED TUBES

EXTERNAL FPI	20
THICKNESS [mm]	55
MAIN CHANNEL LENGTH [mm]	359
STACK HEIGHT [mm]	515

Coolant Temperature @ Radiator inlet = 90 °C
Air Temperature @ Radiator inlet = 35 °C

CASE A				
COLD SIDE		HOT SIDE		HD (kW/m2/°C ETD)
Mean Core Air Face Velocity (m/s)	Air Side Pressure Drop Across Core (Pa)	Water Flow Rate (l/min)	Water side Pressure Drop (mbar)	
2.5	119	20	2.4	2.34
5.0	348	20	2.4	3.70
7.6	650	20	2.4	4.56
10.0	1054	20	2.4	5.16
2.5	119	60	17.3	2.40
5.0	348	60	17.3	4.13
7.6	650	60	17.3	5.43
10.0	1054	60	17.3	6.49
2.5	119	100	44.6	2.47
5.0	348	100	44.6	4.47
7.6	650	100	44.6	6.08
10.0	1054	100	44.6	7.50
2.5	119	140	84.3	2.53
5.0	348	140	84.3	4.71
7.6	650	140	84.3	6.53
10.0	1054	140	84.3	8.17

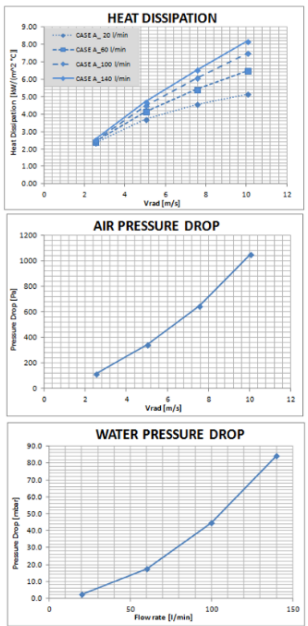


Figure 9. Dallara cooling test bench setup (left) and typical results (right)

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3.3 Realtime Features

The fluids models within the cooling system were set to run with fixed step Euler solver at a rate of 1kHz. With the exception of some initialization spikes, the turnaround time was approximately 0.1ms throughout the desktop test runs (figure 10).

The desktop runs were performed on a 2.8GHz Core I7 processor with SSD giving the confidence of being able to run on the hardware used in the PCs in the DiL setup at Dallara simulator which has superior characteristics.

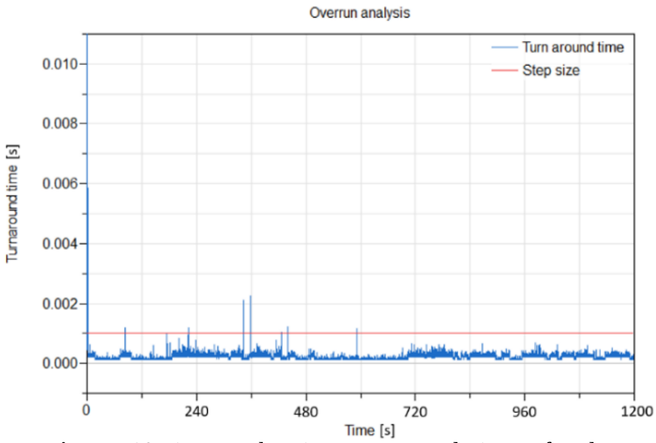

Figure 10. image showing turnaround time of a lap simulation of Monza

The cooling models were compiled with a mixed explicit/implicit Euler method with advanced inline integration settings using the Claytex library Real-time configuration functions to further enhance real time perfomance and model robustness.

Such real time configuration functions also make use of flags described in Section 11.3.3 of the Dymola Full User Manual.

It was not necessary to resort to multicore simulation for the entire fluids system. In fact there would be enough processing power capacity with the aforementioned processor spec to also include a full multibody suspension model of the vehicle and still run in real time at 1kHz and have a physical connector based coupling between the cooling system and the vehicle and powertrain systems.

Figure 11 shows the comparison between the coolant temperature profile simulated vs real logged data. The maximum CoolTempDiff, defined as CoolantTempSimulated - CoolantTempMeasured, is 2.7 °C, the average is 0.2 °C, RMSE is 1.0 °C.

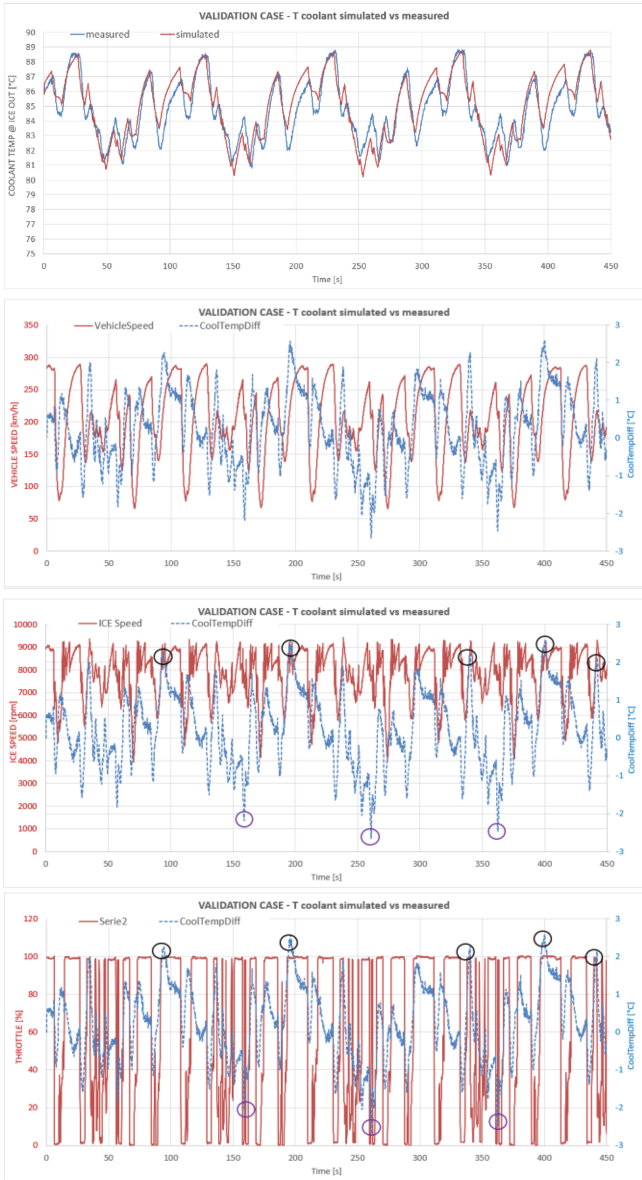

Figure 11. Simulated coolant temperature vs real logged data, vehicle speed, throttle and ICE speed for the validation case

4 Virtual Validation

The validation was a very important phase of this activity, it consisted of the following steps:

- Implementation of a cooling system architecture relative to an existing vehicle.
- Offline simulation using as inputs the measured vehicle speed, throttle, engine speed and ambient temperature recorded on the track.
- Comparison between the simulated coolant temperature profile and measured coolant temperature profile recorded on the track.

The results accuracy is considered acceptable to size a racecar cooling systems and also for a refined assessment of the global vehicle performance on the driving simulator.

In an attempt to further explain the reasons for the discrepancy with the real data, a possible explanation lies in the fact that the ICE heat rejection computed in the simulation transients is the result of an ICE measurement at test bench in fully stabilized conditions. Consequently, the model doesn't consider any delay between the throttle pedal and the heat rejection produced by the engine.

This hypothesis appears confirmed by acknowledging that the points with higher CoolTempDiff (+2.2/+2.7 °C → simulated temperature being higher than real) occur when ICE is in high speed (>8500) and max throttle, as shown in the black circles at time 95s, 190s, 340s, 400s, 440s.

This hypothesis appears also validated in the opposite direction (low ICE speed and throttle close to zero), when the throttle pedal releases and the ICE heat rejection dramatically reduces. In this second situation CoolTempDiff reaches the minimum values (-2.2/-2.7 °C → measured being higher than simulated) exactly in those working points characterized by low ICE speed (<7000) and zero or low throttle (<10%), as shown in the purple circles at time 160s, 260s, 360s.

In order to introduce the effect described above, as a further step of refinement, a first order filter calibrated according to transient test data, could be applied to the Heat Rejection maps.

5 Results

The simulation model, as developed and validated, is currently used at Dallara to size the cooling systems and to analyse the vehicle performance on the driving simulator for different boundary conditions, human drivers and tracks.

The case study under discussion concerns the global vehicle performance of a Dallara car simulated in the Driving Simulator facility, driven by a professional driver in Monza track (Figure 12). Three different boundary conditions, hereinafter referred to as “OUTINGS” (Figure 13 and 14), are considered and below described:

OUTING 1

Air ambient temperature 25 °C, coolant and oil initial temperature 89 °C (no precooling), 10 laps.

OUTING 2

Air ambient temperature 35 °C, coolant and oil initial temperature 80 °C (precooling), 10 laps.

OUTING 3

Air ambient temperature 35 °C, coolant and oil initial temperature 89 °C (no precooling), 10 laps.

In all the three outings the vehicle is able to run without derating in the first lap.

OUTING 1 shows that when the ambient temperature is 25 °C the vehicle runs all the 10 laps at max ICE power without derating.

OUTING 3 is the most critical case due to the higher ambient temperature and the initial fluids (coolant, oil) temperature.

OUTING 2 shows that a precooling phase, which implies lower initial coolant and oil temperatures, allows the vehicle to run more time with higher power at high ambient temperature until it reaches the same stabilized conditions of Outing 3.

The power derating in the stabilized laps of Outing 3 (ambient temperature 35 °C), due the high coolant and oil temperatures, affects the lap time by 2.3 seconds with respect to a not de-rated lap.

All the results reported are marginally affected by the human driver who is the same in all 3 outings, but he doesn’t drive always exactly in the same way. This can be noted in the lap 2 and 3 of outing 1, where despite the power derating doesn’t occour the laptime is higher than lap 1 and lap 10.

The precooling considered on the initial fluids temperature (coolant and oil) is only 9 °C with respect to the case without precooling, so the consequent effect on vehicle performance is minimal, it can be noted analyzing the power profile in the lap 2.

As a next step, two configurations (with and without precooling) with a larger temperature difference will be tested also taking into account the higher friction losses.

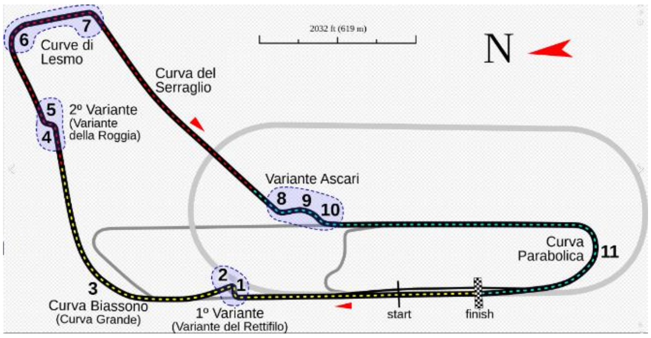

Figure 12. Monza track layout and table results

	INPUT		OUTPUT															
OUTING	T AMB [°C]	PRE COOLING [°C]	MAX POWER DERATING [%]				AVE ICE POWER [%]				TOP SPEED [km/h]				LAPTIME [s]			
			LAP 1	LAP 2	LAP 3	LAP 10	LAP 1	LAP 2	LAP 3	LAP 10	LAP 1	LAP 2	LAP 3	LAP 10	LAP 1	LAP 2	LAP 3	LAP 10
1	25	0	0	0	0	0	66	65	65	66	REF	REF	REF	REF	REF	REF+0.3	REF+0.2	REF
2	35	-9	0	-8	-13	-17	66	64	61	58	REF	REF	REF-6	REF-8	REF	REF+0.6	REF+1.4	REF+2.3
3	35	0	0	-9	-13	-17	66	63	61	58	REF	REF	REF-6	REF-8	REF	REF+0.8	REF+1.4	REF+2.3

Figure 13. Main results at Monza

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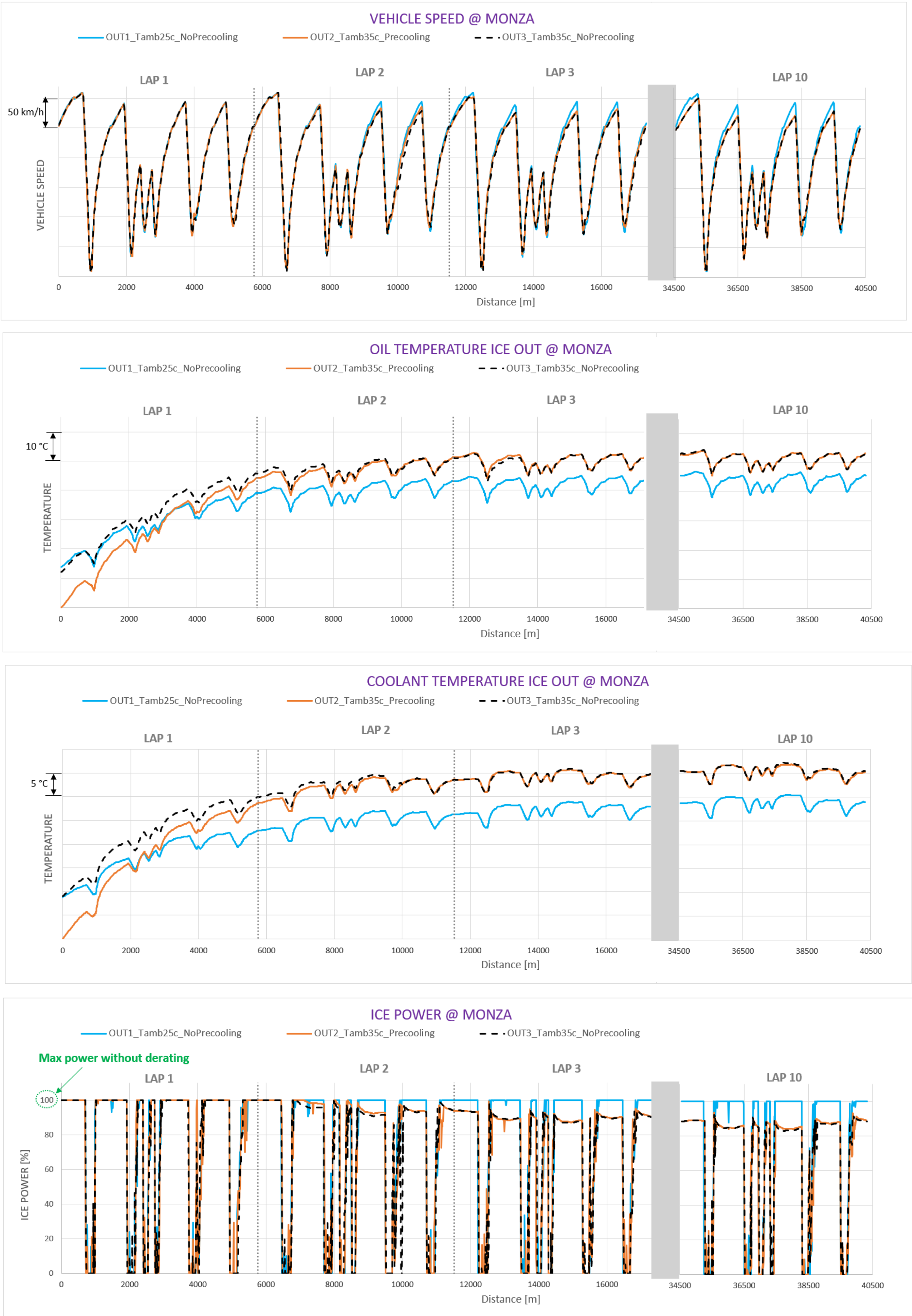

Figure 14. Speed, temperature and power profiles on the Dallara driving simulator

The simulation model, as it is described on this paper, allows to put into the equations different sizes of radiators, considering not only the thermal effect but also their different weight and vehicle aero efficiency.

For example: a configuration with bigger radiator core areas (+15%) is able to improve the laptime by 0.5sec at an ambient temperature of 35 °C (predominant thermal effect because it massively reduces de-rating), but resulted in a 0.3sec slower lap time at an ambient temperature of 25 °C (predominant weight increases and aero efficiency reduction effects). While a configuration with smaller radiator core areas (-15%) resulted slower laps at both ambient temperatures of 25 °C (+0.4sec) and 35 °C (+0.6 sec).

6 Conclusions

The fluids temperature management in high performance vehicles has a crucial impact not only on the reliability but also on the performance and drivability. The obvious answer to that would land to bigger heat exchangers and/or higher air flows, needed to achieve lower fluids (coolant, oil) temperatures. But this implies more weight, drag and a negative impact on packaging constraints. The aforementioned input conditions return a complex and often over constrained or multivariable problem and trade off. Moreover, depending on the mission of the vehicle and on the peculiar driving style it could be more interesting to emphasize the “on power” behavior (typically more favorite for the “track day” drivers) or the “handling” behavior (more important for “professional” drivers). The first case requiring bigger radiators targeting no derating, the second case requiring an overall weight reduction and aero efficiency optimization.

The implemented and here described simulation model is useful to evaluate all these effects together to develop the cooling system architecture for every project targeting the best trade off to maximize the vehicle performance also accounting the driving style and drivability on a professional driving simulator with a real driver.

Powertrain cooling performance is affected by many parameters, this simulation model is useful to analyse the vehicle performance in each condition of ambient temperature, precooling and different human drivers.

The duty cycle analyzed for the case study is a track lap at maximum vehicle performance, but additional critical driving cycles can be studied on the simulator with the model developed, for example, safety car conditions, pit lane or race traffic and grid formation.

The described case study focuses on the coolant and oil cooling system of a high performance naturally aspirated ICE vehicle, clearly similar or greater problems and trade off occur also for charge air temperature in the case of a turbo engine, or in the optimization of an electric powertrain performance (Stellato et al, 2017).

For these reasons, the approach described in this paper, can be considered a good method in the optimization of different vehicle propulsions (ICE, HEV, BEV, FCV, …).

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