Thermal & Air management for a Bus Engine Compartment

A method for determining boundary conditions for computational fluid dynamics simulation

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Värme & luft hantering för ett motorum i en buss.

En metod för att bestämma randvillkoren för en CFD analys.

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SCANIA

Abstract

In today's procedures of bus development in Scania CV, physical tests are the most common and reliable source of confirmation for new concepts. As of now, the majority of the physical tests are conducted in various places around the globe such as Spain, Brazil and Sweden in order to subject the buses to various climates and environments. Naturally, these tests around the globe demand a lot of resources and are very time consuming leading the bus development department to look for alternative ways of confirmation for these tests. An alternative to testing is in the field of Computational Fluid Dynamics (CFD), by using 3D models to simulate in order to save resources and time. Today, the method of simulating using CFD is becoming more frequent in the development phase. However, it needs improvements.

The purpose of this master's thesis is to develop a method in which the application of CFD can be successfully relied upon, by identifying a set of boundary conditions used as initial data for the CFD simulation. The boundary conditions in this project are the driving conditions and the surface temperatures of the heat sources inside the engine compartment.

Initially a physical test is conducted, in which the surface and surrounding temperatures are measured. Surrounding temperatures will act as comparison between the physical tests and the simulations. Once the boundary conditions have been measured through physical tests, the data acquired is then used for the simulation as initial values. The goal is to achieve a maximum of 10% difference between the results of the physical test and those of the simulations.

From the final results the difference between the physical tests and the simulations is 24% for one of the driving conditions which is considered the best case. However, the results are an improvement compared to the old method used currently by Scania. The method developed in this master's thesis shows an improvement of 21% compared to the old method for the same driving condition which implies a step in the right direction for reliable simulations.

Sammanfattning

Gällande dagens industriella utveckling av bussar på Scania CV, läggs det mycket tyngd och förlitan på fysiska tester. De anses vara väldigt pålitliga när nya koncept ska utvecklas och verifieras. Idag görs majoriteten av de fysiska testerna på olika platser runt om i världen så som Spanien, Brasilien och Sverige. Detta görs för att bussarna ska kunna utsättas för olika klimat och miljöer. Naturligtvis, så krävs det mycket resurser och tid för att kunna utföra dessa tester vilket leder till att avledningen för bussutveckling på Scania bestämt sig för att hitta alternativa metoder för att kunna utvärdera sina koncept. En av dessa metoder är att simulera flödesbilden i ett motorrum med hjälp av Computional Fluid Dynamics. Vilket leder till att företaget sparar resurser och tid. I skrivande stund används CFD för utveckling av motorrumsflödet mer och mer men resultatet är opålitlig på grund av osäker indata.

Syftet med det här examensarbetet är att utveckla en metod där tillämpningarna av CFD kan utnyttjas och är tillförlitlig. Detta görs genom att identifiera randvillkor som används till simuleringarna. Randvillkoren baseras på yttemperaturerna av värmekällorna i motorrummet och körfallet som bussen körs i.

För att bestämma randvillkoren utförs ett antal fysiska tester där yttemperaturer och omgivningstemperaturer mäts med hjälp av termosensorer, där omgviningstemperaturerna kommer att användas för att jämföra de fysiska testeterna med simuleringarna. Mätningar som görs under fysiska tester används sedan för att bygga upp en model och simulera bussen i en CFD programvara. Målet med projektet är att kunna uppnå max 10% skillnad mellan det fysiska testet och simuleringarna, vilket visas vara en tuff utmaning.

Det slutgiltiga resultatet visar en skillnad på 24% mellan det fysiska testet och simuleringen vilket är det bästa resultatet av alla körfall. Dock, så visar resultaten en förbättring jämfört med den gamla metoden som används idag på Scania. Den nya framtagna metoden visar en förbättring med 21% gentemot den gamla metoden för samma körfall, vilket visar att ett steg i rätt riktning har gjorts för att simuleringarna ska anses vara förlitbara.

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Murtaza Lokat & Zain Bashir

Limitations

- Time limitation for this project is 20 weeks
- The project is restricted to the bus model Skruttan provided by Scania
- The driving sessions for the physical test is restricted to Scanias driving course
- Simulation will be performed by another department
- Only steady state simulation is will be considered
- Limitation of resources will be restricted to what Scania can provide
- Some parts of this report have been altered due to it containing confidential information exclusive to Scania CV.

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Nomenclature

Parameter	**Symbol**	**Value**	**Unit**
Time	t		s
Area	A		m^2
Volume	V		m^3
Mass	m		kg
Density	ρ		kg/m^3
Velocity vector	$\boldsymbol{v}$		m/s
Pressure	$\boldsymbol{p}$		Pa
Temperature	T		°C
Momentum	p		$kg \cdot m/s$
Force	F		N
Internal energy	U		J
External work done	W		J
Mass flow	$\dot{m}$		m^3/s
Total energy	E		J
Total enthalpy	H		J
Normal stress	σ_s		Pa
Shear stress	τ		Pa
Thermal conductivity	k		$W/m \cdot K$
Rate of heat transfer	Q		W
Rate of heat flux	$\dot{q}_h$		W
Rate of heat conduction	$\dot{Q}$		J/s
Stefan-Boltzmann constant	σ	$5.67 \cdot 10^8$	W/m^2
Emissivity	ε		
Absorptivity	α		

1 Introduction

1.1 Background

Designing, developing and building a bus is no easy feat. Today's buses are driven in a variety of different environments, with each customer having different requirements and expectations. For Scania to be able to deliver a product that would fulfill these requirements and expectations, effective and accurate methods must be a viable choice. From customer to state and province, the rules and demands for the produced buses differ. Quality, robustness, comfort, air management and noise control all play a part in developing an attractive product.

As time progresses, technology is becoming more and more of an accessible tool for companies and with today's technologies, simulations can be accurate enough to be able to draw conclusions which can further push the project towards the desired goal in the development phase. As of now, Scania is equipped with tools to simulate the airflow through the engine compartment with CFD software's. However, results from these simulations and physical testing show differences which are deemed as unacceptable according to Scania's quality standards. This is due to the boundary conditions not being optimized when running the simulation, thus leading to results being different compared to physical tests. Current boundary conditions used in previous projects are mostly taken from data recovered and measured by the truck department and various physical tests.

1.2 Objectives

The goal of the project is to deliver a method in which one is able to determine a set of reliable boundary conditions for a CFD simulation in an engine compartment for a bus. The boundary conditions will be affected by different load cases of interest to Scania for chassis configurations determined by Scania. Furthermore, the target is to develop a method for simulations that is able to determine boundary conditions which generate results that differs no more than 10% from experimental data. The long term goal for Scania with this projects is to establish a simulation for their concepts that optimizes the use of physical tests, which in turn reduces time and cost for development of projects.

1.3 Project Layout

The project process which in itself is the method can be seen in Figure 1. The figure shows the different stages of the project process.

The first stage is research. Research is a very crucial stage since it lays out the groundwork for what needs to be prepared for and the tools needed to complete the full cycle. Next stage is the physical test where the actual test is carried out. Naturally, after the physical test comes the evaluation of the test results which is used further in the simulations where the physical test is simulated. The last stage is where the comparison between the test and the simulation is to be analyzed. Each of these stages represents steps that must be taken in order to move onto the next stage and in the end complete the cycle.

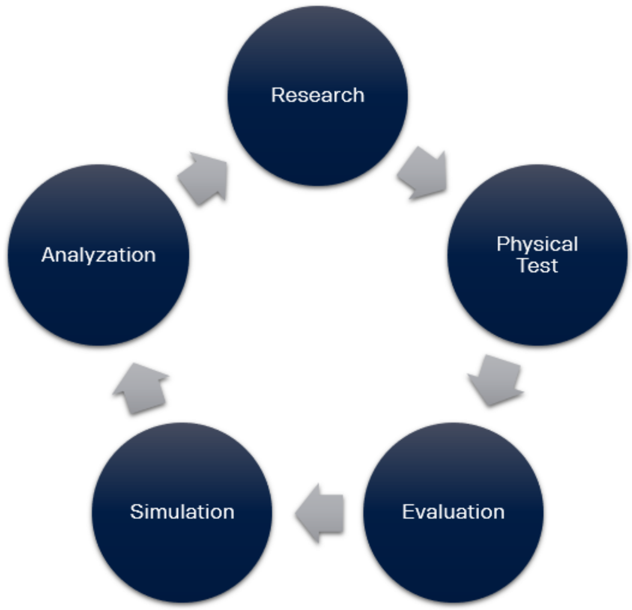

Figure 1: Complete cycle with all the project stages.

2 Literature Study

2.1 Bus

Skruttan is a specific test bus which is part of the Scania Citywide series. The Worldwide Series is a group of busses that are designed for inner cities. The purpose of these busses is to be used in the public transport system. A city bus has a characteristic of having low floor for passengers to effortlessly get in and out. The inner city busses are typically crowded with an intense flow of passengers. Therefore, in order to adjust to the amount of expected passengers these busses have fewer seating areas. This is to be able to take upon as many passengers as possible. It is also known to have many stops along the route with the stops being close to one another.

Figure 2: Illustrative picture of the Citywide series that Skruttan is part of.

Skruttan is designed for high in and outflow traffic of passengers. The bus is equipped with a door configuration of 2-2-2, which specifies three entrances with 2 doors at each entrance. The bus has a total length of twelve meters, with a 4x2 wheel configuration is appropriate. A 4x2 configuration means that the two wheels on each side in the back of the bus, totaling up to four wheels, act as the driving wheels, i.e. they are connected to the shaft in rear. The two wheels on the front, one on each side, act as steering wheels, totaling up to six wheels on the bus.

2.2 Engine

The test vehicle is powered by a seven liter, 6 cylinders diesel engine, equipped with a turbo-compressor, and it is located in the back of the bus.

Figure 3 shows the schematic view over the engine and its relevant components from combustion to exhaust. Is it can be expressed, the components in the figure are directly impacted by the combustion and exhaust gases, and will be a crucial part in determining the heat sources.

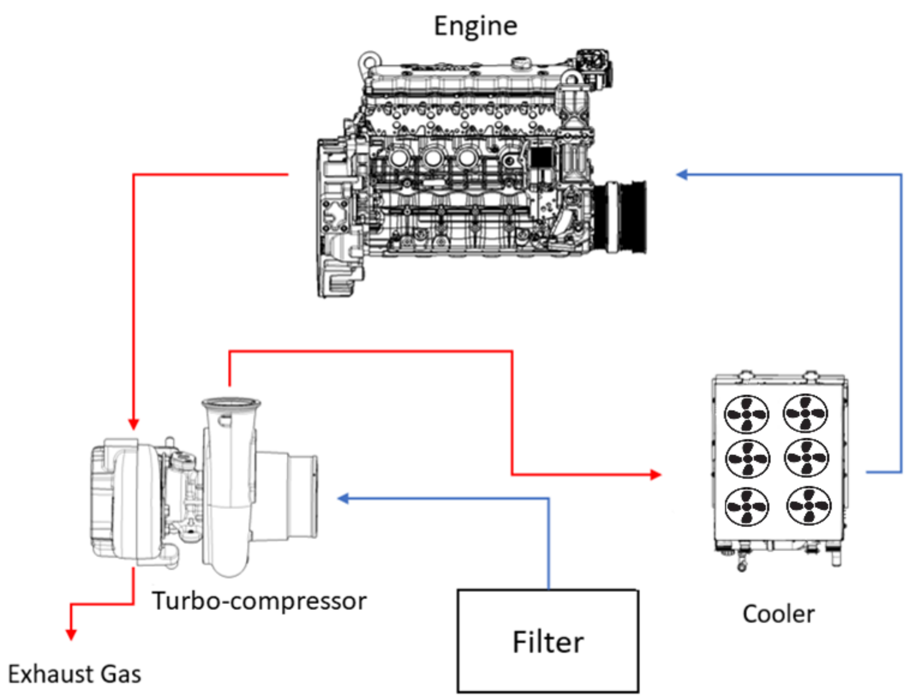

Figure 3: Internal heat exchange of the exhaust gas.

In Figure 3, it can be seen that the air is filtered and then transferred to the turbo-compressor which compresses the gas thus increasing the temperature. The gas then moves through the cooler and into the engine where combustion occurs, the exhaust gases from the combustion is then utilized by the turbo as the turbo uses the gases to drive the turbo-compressor and then expelled out from the exhaust pipe. The reason why this setup is desired due to cooler air entering the engine which enhances the efficiency of the combustion.

2.2.1 Engine Compartment

The engine compartment varies slightly among the different bus models developed by Scania. Skruttan is, as mentioned previously, desires for inner city areas where as much available space as possible is required for the passengers, leading to less space for the engine compartment. This in turn leads the heat distribution inside the compartment to be a lot more compact due to the position of the heat components being closer.

2.2.2 Components of Interest

From previous testings of the temperature distribution in the engine compartment, some components have already been deemed as high heat sources. This part of the literature study will mainly focus on those components and briefly explain their function and contribution to the heat distribution in the engine compartment.

Turbo-Compressor

First of the components of interest is the turbo-compressor, shown in Figure 4. Essentially it is used to increase the efficiency of the combustion engine by forcing in compressed air. The exhaust gas from the engine drives the turbine in the turbo which in turn provides air into the combustion chamber through the compressor. Aside from the efficiency increase, the turbo-compressor also reduces fuel consumption and increases the torque. However, the problem in this case is that the temperature of the exhaust gas leaving the combustion chamber and entering the turbo-compressor is quite high, resulting the turbo-compressor being one of the main heat sources inside the engine compartment, radiating heat to nearby components.

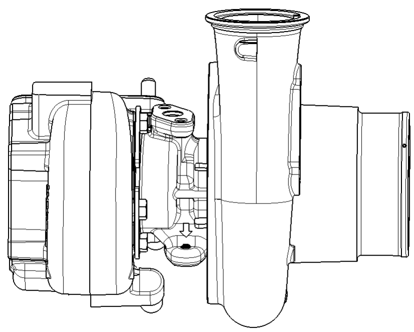

Figure 4: Isometric view of the turbo-compressor. With the left side being the turbo and the right side the turbo compressor.

Alternator

Another component is the alternator, and its purpose is to generate electricity through the use of the engine. It supplies electricity to the battery and other components such as the AC, auxiliary heater, defroster etc. From previous reports in the Scania database, the alternator is deemed as an interesting heat source worthy of a closer examination. The alternator is a component with air circulation and therefore it is a vital component to consider in regards to the heat distribution, as previous tests have shown that the surrounding temperatures reach quite high.

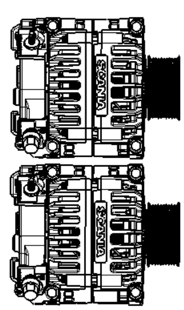

Figure 5: Isometric view of the two alternators.

Silencer

The silencer is also deemed a heat source with high surface temperatures. Although it is secluded from the majority of the components it still has an impact on the overall temperature inside the engine compartment. The main purpose of the silencer is not only to reduce the noise level but it also works as a particle filter. Nowadays the silencer also includes a particle after treatment system that deals with e.g. NO_x emissions thus leading the component to have a high temperature since it is dealing with exhaust gases directly. However, it is necessary for the after treatment system to have high temperature in order to filter out the different kind of chemical emissions in the exhaust gas.

Exhaust Pipe

The last component to handle the exhaust gases directly is the exhaust pipe. With hot gas coming from the combustion chamber reaching high temperatures, the pipe itself is heated up, thus leading the component to act as a heat source. Note that the exhaust pipe and the silencer cannot be illustrated due to confidential reasons.

2.2.3 Measurement Points

With the previous section describing the components of interest, this section will focus on the measurement points and how they are set up. The points are divided in mainly two categories; first, the ones that are directly linked to the boundary conditions for the CFD simulation; and second, the points which are used to compare the results from the simulation to reality. Naturally, the points are correlated with the boundary condition and include the components of interest. The points measuring of surface temperatures are used for the comparison rather than surrounding and fluid temperatures since are more suited for a comparison between reality and simulation. Furthermore, a set of data is measured through the internal software system EMS (Energy Management System) in which crucial information such as loads, speeds, ambient temperature, etc, are measured. The which will be explained in more detail in section 2.7.1.

Table 1: Surface temperature for measurement points where BC stands for boundary condition and CMP for comparison.

Measurement Point	Description	Type
TY_TURBO_COMP	Turbo-compressor	CMP
TY_INL_GEN	Top alternator	BC
TY_LUFT_COMP_SLANG	Air compressor hose	CMP
TY_AVG_SKARV_1	Exhaust pipe joint	BC
TY_AVG_ALU_1	Exhaust pipe aluminum isolation	BC
TY_AVG_ISO	Exhaust pipe isolation	BC
TY_AVG_ALU_2	Exhaust pipe aluminum isolation	BC
TY_AVG_SILENCER	On the exhaust pipe behind the silencer	BC
TY_AVG_SLUT	Exhaust pipe outlet	BC
TY_LUFT_COMP	Air compressor	CMP
TY_TURBO	Turbo	BC
TY_CYL_1	First engine cylinder	BC
TY_CYL_3	Third engine cylinder	BC
TY_CYL_5	Sixth engine cylinder	BC
TY_SILENCER_FRONT	On the bottom of the silencer	BC
TY_SILENCER_OVAN	On top of silencer	BC
TY_SILENCER_VA	On the left side of the silencer	BC
TY_SILENCER_HO	On the right side of the silencer	BC
TY_UPPER_FLOOR	On the floor of the cooling compartment	CMP
TY_CHARGE_HOSE	On the rubber connection to charge air pipe	CMP
TY_WASTEGATE	On the wastegate	CMP
TY_ROR_INL_UT	On the plastic inlet pipe close to turbo compressor	CMP
TY_ROR_INL_2	On the plastic inlet pipe inside the cooling shack	CMP
TY_FILTER_BURK	On the filter	CMP
TY_CAC_ROR	On the charge air cooler pipe	BC
TY_AFI	On the AFI sensor	CMP

Table 1 shows the surface temperature measurement points correlated to the boundary conditions. To clarify, these points are determined through components which are considered a heat source in the engine compartment. However, these points are not the only surface temperature points measured. There are a few surface temperature measurements that do not constitute the boundary conditions, instead they are related to the fragility of the components. These surface temperature measurement points of components which are deemed as fragile articles are under the type "CMP". This includes electric components sensitive to heat which cannot exceed their respective critical limits. Table 2 shows the surrounding temperatures of the measurement points and Table 3 the fluid temperatures, which include the exhaust gases, air and water temperatures. Note that due to the size and configuration of the silencer and exhaust pipe, multiple measurement points have to be place in order to get an accurate surface temperature distribution.

Table 2: Surrounding temperatures of the measurement points for comparison spread out in the engine compartment.

Measurement Point	Description
TL_OMG_UTL_GEN	Temperature around the outlet of top alternator
TL_OMG_STARTM	Temperature around starter
TL_OMG_OVRE_SHACKT	Temperature of the cooling compartment
TL_OMG_INL_GEN_FRAM	Temperature around the inlet of top alternator facing the turbo
TL_OMG_TURBO	Temperature around the turbo
TL_OMG_OVRE_FLAKT	Temperature of the RAD fan in the cooling compartment
TL_OMG_CYL_5	Temperature around the sixth cylinder
TL_OMG_CYL_1	Temperature around the first cylinder
TL_OMG_UNDRE_FLAKT	Temperature of the CAC fan in the cooling compartment
TL_OMG_TAK_TURBO	Temperature around the roof above the turbo
TL_OMG_BAK	Temperature at the ventilation hatch rear side of the engine compartment
TL_OMG_INL_GEN_BAK	Temperature around the inlet of top alternator facing end of the bus

Table 3: Measurement points of the fluid temperature, where TG is for exhaust gases, TL for air and TW for water.

Measurement Point	Description	Type
TG11	Exhaust gas temperature inside turbo	Exhaust gas
TG12	Exhaust gas temperature inside turbo	Exhaust gas
TL23 (KYLARE IN)	Temperature of air in to cooler	Air
TL31	Temperature of air in the inlet pipe	Air
TG21	Temperature of gas after turbine	Exhaust gas
TG50	Temperature of exhaust brake gas	Exhaust gas
TL12	Air temperature before filter	Air
TL15	Air temperature after filter	Air
TL17	Air temperature before compressor	Air

2.3 Metrology

2.3.1 Thermocouple

A thermocouple is an electric sensor that measures temperatures. The device is comprised of two wires, separately surrounded by an outer shelter. One wire is positively charged and the other one is negatively charged. The positive wire measures the surface temperature while the negative wire is used as reference. These wires are then crossed and welded together forming a round tip. When the tip is installed in a hot space, a voltage or a potential difference will occur between the two wires. This voltage is then converted to a temperature using a thermocouple reference table. This measurement technique that is used here is referred to as the Seebeck effect, using this setup gives the thermocouple a high sensitivity and a fast response time.

There are different types of thermocouple sensors. The most common type is Type K, where the K refers to the positive wire containing Nickel-Chromium and the negative one containing Nickel-Aluminium. There exists two different Type K sensors, Thread and Encapsulated. The thread thermocouple sensor, as seen in Figure6 is used to measure surface temperatures while encapsulated thermocouple has a steel plate that covers the measurement sensor and is used to measure fluid temperatures[1].

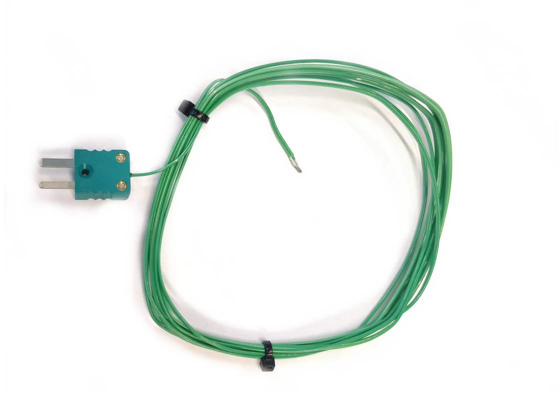

Figure 6: Thread thermocouple sensor.

2.3.2 Measurement Module

The thermocouple is connected to a measurement module which receives the measured signal from the thermocouple. This module is an analog signal receiver for voltage measurements, and it converts this voltage to an analog signal.
There are 16 inputs per module, which allows each module to be able to connect with 16 thermocouple sensors. The voltage from the thermocouple is converted to data that reflects the components temperature inside the measurement module.

Figure 7: Measurement module with SIM-CAN connections for the measurement system [2].

2.3.3 Memorator

Memorator is a two channel controller area network (CAN) bus interference and data logger. This device has the ability to both record data on a memory disc and also be connected directly to a PC. The main features of this device are message filtering, triggers, and error detection. The memorator is also very useful since it converts the data from the measurement modules to the software program in the PC [5].

Figure 8: Memorator used in the test [5].

2.3.4 Complete System

The complete system involves the above mentioned components, but the system has to follow a specific order. All the fluid, surface and surrounding sensors are connected to cables that leads out from the engine compartment to the measurement modules which are located inside the bus. The modules are connected in series which are then connected to the memorator through a VGA type cable, and is powered by an AC/DC source cable. The EMS is also connected to the memorator which enables the memorator to send signals from both the sensors and the EMS being to the computer. IPEmotion is then used to build a complete database with the signals from both the EMS and the module. The software builds a library and sorts out the sensors connected to the module and identifies them, this makes it possible to keep track of the measurements from the sensors and then store them. Finally, the software tool Vision is used to analyze the measurements in real time with the database being exported from IPEmotion. Figure 9 shows the schematic map of the complete system.

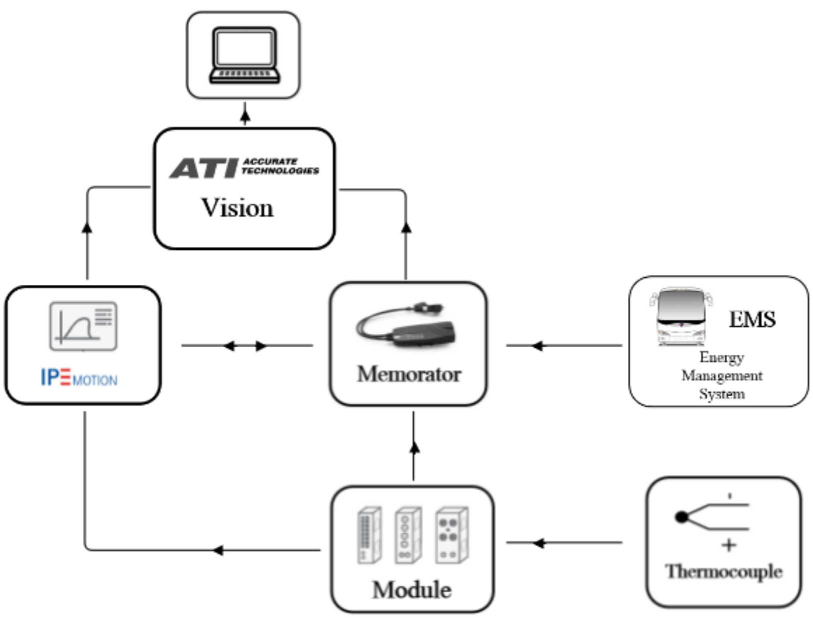

Figure 9: Schematic map of the measurement system.

2.4 Tools

2.4.1 CATIA V5

CATIA is a multiple design software that includes computer aided design (CAD), computer aided manufacturing (CAM), and computer aided engineering (CAE). CATIA is used for product development from conceptualization and design to manufacturing and engineering. This is a standard program used by Scania for development of new concepts [6].

The CFD simulation is performed on CAD model provided to the CFD software STAR-CCM+. The desired measurement points are placed in the CAD model which is exported to the CFD software. These points are converted to triangulated surfaces later on in STAR-CCM+. The desired measurement points must be manually positioned in the existing CAD model of the engine. These points are given as a reference to the CFD department when doing the simulation.

2.4.2 STAR-CCM+

STAR-CCM+ is a multidisciplinary platform with a wide spectrum of application for simulations for different scenarios with real conditions in play. The different platforms that could be of interest for this project or Scania are fluid dynamics, heat transfer, particle flows, multi phase flows and aero acoustics. In this project, the main platforms of interest are fluid dynamics and heat transfer [7].

The software is intended to be used for simulations of the fluid flow together with the heat distribution in the engine room in a replication of the real world testing arrangement. A comparison between simulation and real testing shows the accuracy of the model and gives an answer about the improvements necessary for a reliable model. However, the fault do not necesserly have to be the models alone but could also be in the accuracy of the performance of the physical test.
The simulation is very dependable on the boundary conditions given, and therefore the boundary conditions must be as accurate as possible.

2.4.3 Vision

Vision, a software program developed by Accurate Technology Inc (ATI), is a powerful comprehensive tool to acquire and calibrate data from real time measurements. The software is used to access the data of closed loop systems and components such as the Engine Control Unit (ECU). In this case the post and real-time analysis was done using Vision to build a foundation for the determination of the boundary conditions and verification of the CFD simulation [8].

2.4.4 Matlab

Matlab (Matrix Laboratory) is a software tool with its own language used for simulation and numerical computing. Matlab allows matrix manipulation, plotting of functions and data, simulation etc. It is a great engineering tool and has a plethora of uses, in this case Matlab is used for plotting data and analysis of the data [9].

2.5 Computional Fluid Dynamics

2.5.1 History

Computational fluid dynamics began its application in the beginning of the 1970's. It was first used to simulate inviscid transonic flow based on the solution of the nonlinear potential equation. A decade later the two-dimensional and three-dimensional applications of computational science became feasible through the application of Euler equations. With a combination of the Navier-stokes equations, the method became applicable for viscous flow analysis which paved the way for complex simulations in aeronautics and turbomachinery. With time, numerical methods and chemistry started to play a bigger role in the advancement of the method. Where the CFD now also considers the molecular structure of the fluid and its heat characteristics in the analysis. This made it very useful when modelling gas turbines and engines [10].

The long development journey has now lead to CFD being a major part of industrial development design process of all types of vehicles, aircrafts and turbomachinery. The reason why it is so desirable is because one can simulate the design with virtual models before actually having to physically model it. This saves both time and resources which is one of the reasons for this thesis.

Even though CFD is at a very advance stage it still requires further research and development. Example of that are the physical phenomenon such as heat transfer and modeling of turbulence flow which are both very complex [13].

2.5.2 Governing Equations

Fluid dynamics describes the motion of a large amount of particles which together form a fluid, through the application of fundamentals concepts and equations. When these particles build a continuum with high density, there exists a mean velocity and a mean kinetic energy which can define relevant physical quantities such as velocity, pressure, temperature etc.

The derivation of the governing equations of fluid dynamics are based upon the behaviour of the fluid and is determined by the conservation laws. These law states that the mass of a fluid is conserved, the change in momentum equals to the sum of the forces on a fluid particle, and the rate of energy is equal to the sum of the rate of the heat added and the rate of work done on the fluid particle.

The quantities of the fluid particles are seen as functions of space and time. The quantities will be written as $\rho(x, y, z, t)$ for density, $p(x, y, z, t)$ for pressure, $T(x, y, z, t)$ for temperature and $\vec{v}(x, y, z, t)$ for velocity. Where $\vec{v}(x) = u$, $\vec{v}(y) = v$ and $\vec{v}(z) = w$.

Mass conservation for a three dimensional control volume (Ω); the rate of increase of mass in the fluid element equals the net rate of flow of mass into the fluid element:

$$\frac{\partial}{\partial t} \int_{\Omega} \rho \, d\Omega + \oint_{\partial\Omega} \rho(\vec{v} \cdot \vec{n}) \, dS = 0, \tag{1}$$

where equation 1 represents the unsteady mass continuity equation at a point in a compressible fluid. The first term on the right is the change of density over time, and the rest of the terms can be written in a compact vector $\frac{\partial \rho u}{\partial x} + \frac{\partial \rho v}{\partial y} + \frac{\partial \rho w}{\partial z} = div(\rho u)$, which represents the net flow of mass out of the control volume across its boundaries [11].

Conservation of Momentum is described by Newton's second law stating that "the rate of change of momentum of a fluid particle equals to the sum of the forces on the particle". Applying the law of conservation will yield that, the rate of increase of momentum of a fluid particle is equal to the sum of forces on the fluid particle:

$$\frac{\partial}{\partial t} \int_{\Omega} \rho \vec{v} \, d\Omega + \oint_{\partial\Omega} \rho \vec{v}(\vec{v} \cdot \vec{n}) \, dS = 0, \tag{2}$$

where the first term is the variation of momentum in time within the control volume Ω. The second term is the contribution of the flex tensor to the conservation of momentum. The fluid will be exposed to both external and surface forces which are included in equation (2), [11].

Conservation of Energy is derived from the first law of thermodynamics. The equation of energy conservation states that the rate of increase of energy of fluid particle is equal to the net rate of heat added to the fluid particle and net rate of work done on the fluid particle [11].

$$\frac{\partial}{\partial t}\int_{\Omega}\rho E\, d\Omega + \oint_{\partial\Omega}\rho H(\vec{v}\cdot\vec{n})\, dS = 0, \tag{3}$$

where E in the first term of equation (3) is the total energy per unit mass and is obtained by adding its internal energy per unit mass. In equation (5), the first term to define the total energy for a control volume depending on time, and the second term defines the total enthalpy equation, where H is the relation between the total enthalpy, total energy and pressure with $H = E + \frac{p}{\rho}$ [13].

2.5.3 Navier-Stokes Equations

As previously stated, fluid dynamics describes the motion of large number of particles which can be seen as a fluid or a gas. It is with Navier-Stokes equations that these motions and behaviours are defined. Navier-Stokes equations use the laws of conservation of mass (1), momentum (2) and energy (3) and gather these equations in one system;

$$\frac{\partial}{\partial t}\int_{\Omega}\vec{W}\, d\Omega + \oint_{\partial\Omega}(\vec{F}_c - \vec{F}_v)\, dS = \int_{\Omega}\vec{Q}\, d\Omega, \tag{4}$$

where the vector $\vec{W}$ (5) is referred to as the conservative variables, which consist by three dimensions with five components. These components are the first terms from equations (1), (2) and (3) which describe the change in mass continuity, momentum and total energy inside the control volume.

$$\vec{W} = \begin{bmatrix} \rho \\ \rho u \\ \rho v \\ \rho w \\ \rho E \end{bmatrix}, \tag{5}$$

There is also two new flux vectors introduced, $\vec{F}_c$ and $\vec{F}_v$. The first vector is related to a convective transport of quantities in the fluid and is usually referred to as the vector of convective fluxes. This component relates to the second terms from equation (1), (2) and (3). However, for the momentum and energy equation a pressure term will also be included.

$$\vec{F}_c = \begin{bmatrix} \rho V \\ \rho u V + n_x p \\ \rho v V + n_y p \\ \rho w V + n_z p \\ \rho H V \end{bmatrix}, \tag{6}$$

where V is the contravariant velocity which is the velocity normal to the surface element dS.

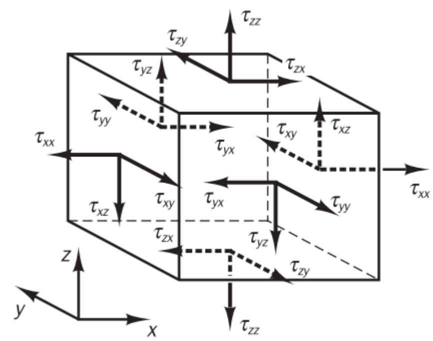

Figure 10: Normal and shear stresses on a fluid element [13].

The second flux vector $\vec{F_v}$ is called the vector of viscous fluxes. This vector contains the viscous stresses and the heat diffusion as seen in Figure 10.

$$\vec{F_v} = \begin{bmatrix} 0 \\ n_x\tau_{xx} + n_y\tau_{xy} + n_z\tau_{xz} \\ n_x\tau_{yx} + n_y\tau_{yy} + n_z\tau_{yz} \\ n_x\tau_{zx} + n_y\tau_{zy} + n_z\tau_{zz} \\ n_x\Theta_x + n_y\Theta_y + n_z\Theta_z \end{bmatrix}, \tag{7}$$

where equations (8) represents terms that describe both the viscous stresses and the heat conduction in the fluid.

$$\begin{aligned} \Theta_x &= u_{xx} + vu_{xy} + wu_{xz} + k\frac{\partial T}{\partial x}, \\ \Theta_y &= u_{yx} + vu_{yy} + wu_{yz} + k\frac{\partial T}{\partial y}, \\ \Theta_z &= u_{zx} + vu_{zy} + wu_{zz} + k\frac{\partial T}{\partial x}, \end{aligned} \tag{8}$$

Last but not least, the last term in equation (4) is a source term, which sums all volume sources due to body forces and heating.

$$\vec{Q} = \begin{bmatrix} 0 \\ \rho f_{e,x} \\ \rho f_{e,y} \\ \rho f_{e,z} \\ \rho\vec{f_e}\cdot\vec{v} + \dot{q}_h \end{bmatrix}. \tag{9}$$

The three dimensional Navier Stokes equations are a system of five known variables, $\rho, \rho u, \rho v, \rho w$ and ρE, and two unknowns which are p and T. These are solved by the thermodynamic relations under the assumption of thermodynamic equilibrium that can be described by the Equations of state.

2.5.4 Turbulence Modeling

Turbulence modeling is a crucial part to get an accurate CFD simulation, due to the fact that the flow is in frequently inherently turbulent in reality, compared to theory where one is able to separate turbulent from laminar flows. RANS (Reynolds-Averaged Navier-Stokes) is a method to model turbulence with the help of the Navier-Stokes equations. The concept is to take a time average of the flow equations, i.e averaging over small-scale fluctuations and model nonlinear influence from small-scale fluctuations on the governing Navier-Stokes equation, which in turn can alter large-scale fluid motion. However, it is worth noting that RANS requires an input model before computing. The reason for this is that the method creates terms which are different for each scenario. Generally, there is no turbulence model that works universally [11].

2.6 Heat Transfer

It is well known from the first law of thermodynamics that energy cannot be created nor destroyed in an isolated system, it can only change its form. In this section the change of energy will be in the form of heat, which is the form of energy that can be transferred from one system to another as a result of temperature difference. The physical phenomenon that explains this energy transfer is heat transfer, which will always occur in the direction of the hotter system to the colder one. Furthermore, the three aspects of heat transfer will be discussed: conduction, convection and radiation.

2.6.1 Conduction

Conduction is heat exchange between particles. The energy is transferred from the particle with higher energy to the lower one until they reach equilibrium. The heat is exchanges through physical contact. In solid form heat is transferred with a combination of vibrations of the molecules and energy transported by free electrons. Liquid and gases conduct heat due to collision and diffusion of the molecules.

Conduction is described by **Fourier's law**:

$$\dot{Q}_{cond} = -kA\frac{dT}{dx}, \tag{10}$$

which is also known as the law of heat conduction. Equation (10) describes the rate of heat conduction $\dot{Q}_{cond}$ being proportional to the area times the temperature difference divided by the thickness of the materials where k is the thermal conductivity of the material, and dT/dx is the temperature gradient and A being the area [12].

2.6.2 Convection

Convection is heat transfer between a solid surface and a liquid or a gas. Convection is a combination of fluid motion and conduction. As the motion of the fluid increases, the greater the convection of heat transfer between the surface and the fluid. When the fluid is not in motion, the heat transfer between the solid and the fluid is purely by conduction. However, with fluid motion the heat transfer increases between the solid surface and the fluid due to a bigger difference in temperature.

There are two ways heat transfers with convection. The first is called forced convection, which happens when the fluid is forced over the surface. The second is called natural convection, that occurs if the fluid motion is caused by buoyancy forces that are induced by the density difference, which happens due to change in temperature.

The rate of heat convection is expressed by **Newton's law of cooling** as

$$\dot{Q}_{cond} = hA_s(T_s - T_\infty), \tag{11}$$

where h is the convection heat transfer coefficient, A_s is the surface area through which the convection heat is transferred, T_s is the surface temperature and T_∞ is the fluid temperature. In spite of the complexity of convection, the rate of convection heat transfer is observed to be proportional to the temperature difference.

2.6.3 Radiation

Radiation is heat transfer between two bodies at different temperatures with a distance between them, where heat is transferred through electromagnetic waves between the bodies. Heat transfer by radiation does not require the existence of an intervening medium, as compared to conduction and convection. Radiation is the fastest heat transferring aspect. All bodies with an absolute temperature above zero radiate thermal energy. The maximum rate of heat that can be emitted from a body is from a black-body which is described by the **Boltzmann law**

$$\dot{Q}_{emit} = \sigma A_s T_s^4, \tag{12}$$

where σ is the Boltzmann constant. The black-body is the idealized surface from where the maximum rate of radiation occurs, all real surfaces radiates less heat than the black-body which is why it is used as a reference in the following equation

$$\dot{Q}_{emit} = \varepsilon \sigma A_s T_s^4, \tag{13}$$

where ε is the emissivity of the surface. The range of emissivity is $0 \leq \varepsilon \leq 1$, depending on the surface. Emissivity describes how close the characteristics of the surface is to a black-body, where $\varepsilon = 1$ is defined as a black-body.

2.6.4 In Between Medium

A very important property when talking about radiation is absorptivity α. It is a measure of how much of the radiation energy is absorbed by a surface or a fluid. Like emissivity, the range of the absorptivity is $0 \leq \alpha \leq 1$, which is a very important concept when talking about the medium in between radiation surfaces.

Clean air in theory does not have any absorptivity since it is mostly composed by nitrogen and oxygen which are transparent to radiation. In an engine room, clean air is contaminated with particles such as carbon dioxide, water vapor and dust particles. These particles will have an absorptivity and therefore absorb heat transferred through radiation. The non-radiating components in the engine room will then experience radiation from the heat sources together with the surrounding air. For simplicity, the total heat transfer to or from a surface is expressed as

$$\dot{Q}_{total} = h_{combined} A_s (T_s - T_\infty), \tag{14}$$

where the variable $h_{combined}$ is a combined heat transfer coefficient that includes the effects from both convection and radiation [12].

2.7 Physical Test and Simulation

2.7.1 Driving Conditions

In order to replicate real life scenarios, a set of driving conditions were pre-determined in which the vehicle operates and experiences all the applied loads that accompany each driving condition. The course which can be seen in Figure 11 shows the test ground where all the driving conditions will be satisfied. The first driving condition called Hot Shut Down (HSD), where the bus is exposed to heavy use prior to shutting the engine. Temperatures inside the engine compartment are then measured, this is to examine the fluctuating temperatures and evaluate the different cooling times. Furthermore, a suburban driving condition is needed to replicate commercial use. The bus is driven around a test course with stops to represent real life urban scenarios in which the distance from each stop is enough to consider a realistic use in urban areas. The Suburban course has a variety of terrain configurations which the bus undergoes, uphill and downhill drive, sharp turns, high velocity etc. These different parts of the course require different properties from the engine components, therefore giving a wide range of data in different scenarios. It is worth noting that the tests are done after the bus has been driven until temperatures of the engine components reach a stable value.

Another driving condition is when the vehicle stands idle i.e. the engine is running but the throttle is left untouched. The main purpose here is to examine the behaviour of the components when they are not subjected to any loads. Since buses remain idle a lot in normal circumstances, it is worth knowing how the temperature of the components behave in terms of how long it takes for the temperature to decrease, and what components start to cool down or heat up. The opposite of letting the bus be in an idle state is to drive it uphill. By driving it up a slope, one is able to subject the bus to heavy loads and examine the components when they are exerted to heavy use.

The last driving condition is a constant speed of 50 km/h, in order to eliminate the impact of accelerating and decelerating effects. Instead, the focus here is to examine the applied loads subjected to the bus in a constant speed and determine the effects on each component.

2.7.2 Test Course

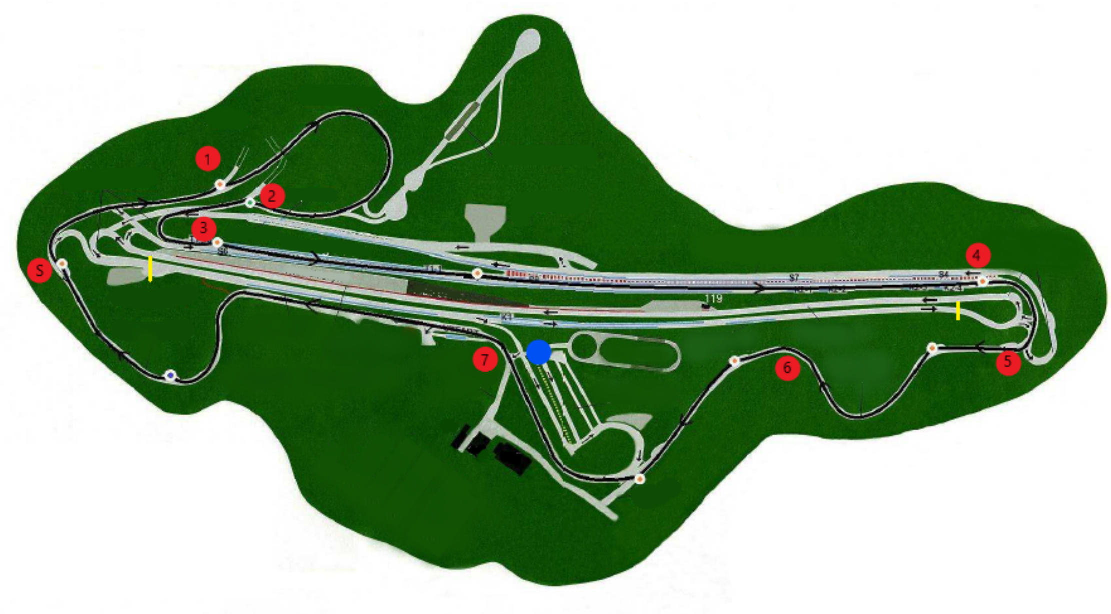

Figure 11: The test ground with all the driving courses.

The driving course has a wide range of terrains and it is designed to test vehicles for realistic environments throughout the lifespan of the vehicles. Figure 11 shows the driving course and all the important aspects, where "S" in the red dot indicates the start and end of a lap and the numbers indicate each stop. The stops are mostly relevant to the suburban run and are there to replicate real life stops in commercial traffic. Each stop is timed to 20 seconds.

The blue dot indicates the slope which is intended for the uphill driving condition, the slope is not long in terms of distance traveled meaning, that the temperatures of the components do not have enough time to reach their true value. Therefore, it is necessary to drive the bus up and down the slope as many times as possible until the temperatures within the engine compartment reach a stable value for data collection.

In the center of the course, marked between two yellow lines, there is a high speed road where the speed limit is at 110 km/h. This part of the course can be used to exert the bus to heavier loads which in turn would warm up components to a stable value faster, but its main purpose is for the driving condition of Constant 50. The bus can undisturbedly drive in a straight path without worrying about turns and obstacles on the road that results in speed variations.

Note that the test course is not needed for the HSD and idle driving conditions since both scenarios require the bus to stand still for a long period of time. In order to avoid disrupting other test vehicles, the previous mentioned driving conditions should be tested outside the test course or on the side of the road.

2.7.3 Previous Simulations

The previous results results obtained through simulations have been deemed to be unacceptable. Scania believes the main reason for these poor results has to do with the boundary conditions and how they are set up before the simulations. Boundary conditions used for the simulations for the busses were taken from a wide range of sources, as the main source of these boundary conditions are from their trucks.

In 2017, a physical test was conducted in Spain on the bus Spartacus [3]. The bus is quite similar to Skruttan as both buses have a diesel engine configuration with 6 cylinders, are intended for urban usage and has same external geometries. The physical test was done in a city environment with stops along the way, which is similar to the City Suburban driving condition. Furthermore, a simulation was done using the previous boundary conditions. Figure 12 shows the results and the differences between the physical test and simulation for Spartacus. However, the measurement set up was different for Spartacus compered to Skruttan which means that comparison can only be made between the points that match. This comparison will be presented later in this report.

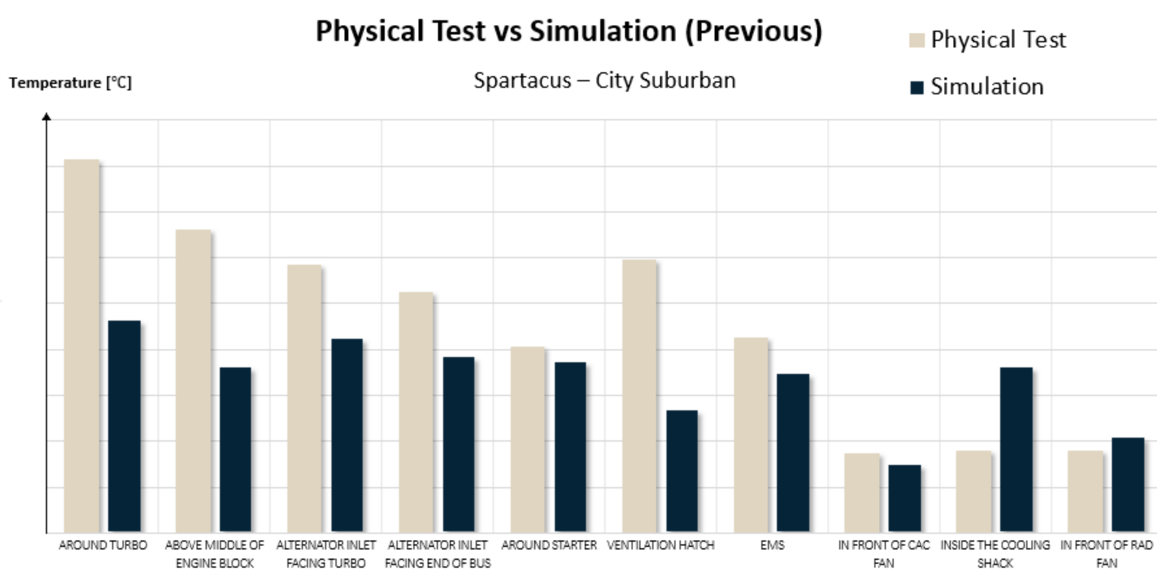

Figure 12: Comparison of physical test and simulation for the bus Spartacus in a City Suburban driving condition.

Figure 13 shows the delta between the physical test and simulation results for Spartacus in the City Suburban driving condition, as well as the median. The biggest difference in percentage is seen in the comparison point "Ventilation Hatch" and the biggest difference in terms of temperature is seen in the "Around turbo" point with the median being 37% and the total average 46% for all the points.

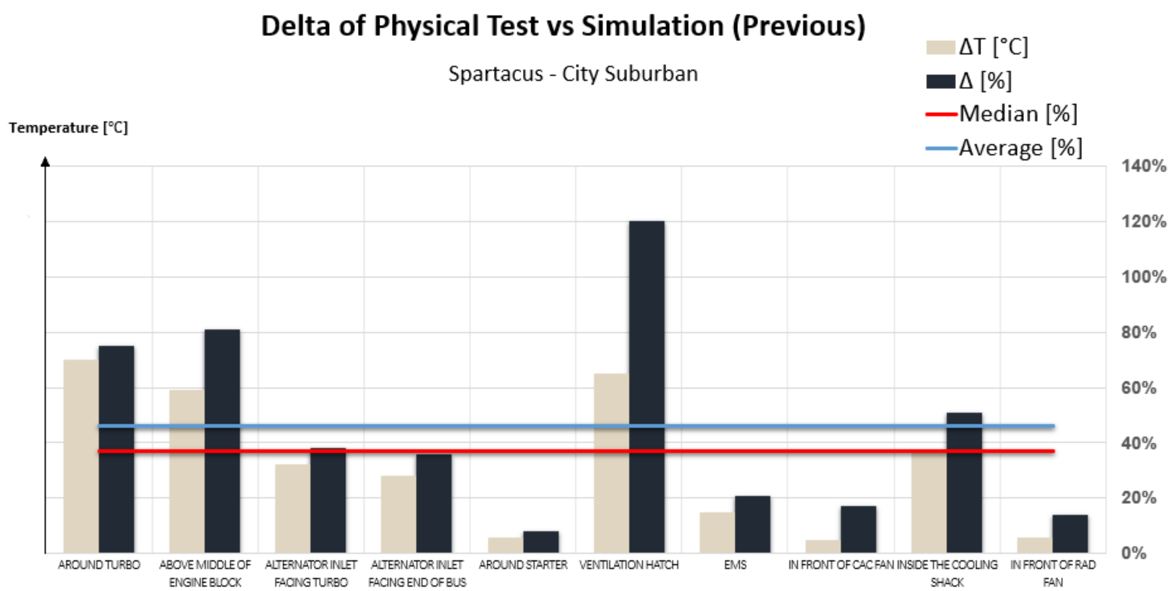

Figure 13: Delta and the median between the physical test and simulation for each comparison point in the City Suburban driving condition for Spartacus.

3 Methodology

3.1 Measurement Procedure

As mentioned in section 2.2.3, the measurement points were divided into two categories. Boundary conditions and comparison points, both containing surface, surrounding and fluid temperatures. This section will describe the steps and procedures taken during the preparation phase.

The surface temperatures were taken by mounting the tip of the sensor against the surface of interest with aluminum tape holding it in place. In an ideal case the tips are supposed to be welded on to the components, but due to time restrictions, this was not possible. In order to get a reliable measurement, the tip had to be taped down securely, so that the sensors were measuring the surface temperature and not the surrounding temperature, but to ensure that the tip is not disconnected from the surface due to hard and unexpected vibrations during the test drive. Another crucial step is to make sure that the tip is not under the tape, which would end up causing the sensor to measure the heat trapped under the tape. Figure 14 illustrates an example of how the sensors were secured during the preparation phase of the project. It was later discovered that some of the components reached quite high temperatures, melting the adhesive on the tape and loosening the grip of the tape. Therefore, a quick check up of the articles deemed as risky measurements was always done before each course run, which would be avoidable if the tips were welded in instead. The component that showed to be the most troublesome regarding the loosening of the tape was the turbo.

Figure 14: The installation of a surface temperature.

The surrounding temperatures were measured in a similar fashion, but instead of touching the surface of any component the importance here lies in the sensor not moving around in the engine compartment, in order to avoid fluctuations in the measurements. In order to have a stable measurement of the surrounding temperature close to a component, the sensors were taped and also attached to nearby cables or hoses with the help of zip ties to ensure no major movements throughout the course run compared to vibrations that gives small changes in the sensor measurements. It is also important to understand how the airflow inside the compartment behaves and to avoid moving parts. Figure 15, shows an example of how a surrounding temperature measurement was done.

Figure 15: The installation of a surrounding temperature

Fluid temperatures were measured directly inside the pipes between components, with a hole drilled in the pipe and the sensor pushed through the hole and held at place with a bolt and a nut to ensure no leakage of the fluid. Fluid temperature measurements are the most time and resource consuming of the three types of measurements. As it turns out the bus already had the important fluid temperature points placed inside the pipes from a previous project, thus, reducing the time spent on fluid measurements in the preparation phase.

All of the sensors were then wired to inside the bus and connected to measurement modules and the measurement system is assembled as shown in Figure 9.

3.2 Physical Testing

As mentioned in section 2.7.1 and 2.7.2, the physical testing was carried out on the driving course. Each driving condition is tested and the temperatures of the components measured with the software program Vision throughout the runs. The first driving condition tested was the suburban case, during which the bus was driven around the course for three laps stopping at each stop for 20 seconds. During the first test, some problems arose regarding temperature measurement. One of the surface temperature sensors was found to not having been attached properly to the surface of the exhaust pipe behind the silencer, leading to temperatures well below the expected value. Furthermore, issues arose with the software leading to a few measurements not being able to be recorded during the driving session. The temperature for the exhaust gas at the exit of the exhaust pipe was not recorded due to some issue with the module not recognizing the connected sensor ultimately leading to a lost measurement point. Furthermore, the measurement points which measured the surrounding temperature on the left side of the silencer and the surrounding temperature around the EMS were only included in the suburban driving condition due to them being neglected in the other driving conditions this is simply due to the hardware not recognizing these two measurement points when the test was conducted during the other driving conditions.

The driving course consisted of 8 stops for the suburban case, in order to analyse and understand the data the stops played a crucial part since they acted as reference points between the stops. The data achieved from the measurements was divided up between each stop, the focus was to examine how the components behave during three cases depending on from which position the bus starts to drive. The cases were divided up to acceleration, brake and plateau. The acceleration case covered all the acceleration phases on the course, the brake covered all the brake phases, and the plateau case covered the phases between the acceleration and brake phase. As mentioned previously, the cases were applied to the bus depending on its initial position when at the stop i.e. uphill, downhill and flat road. Each of the cases was applied to the initial positions totaling up to 9 different cases and scenarios for the suburban data set.

As for the testing of the HSD and idle state, no major issues were present except that the previously mentioned measurement points were missing. The driving conditions were tested in the parking space outside the course with both conditions being measured in real time for 10 minutes each. Furthermore, when the HSD driving condition was being measured, the engine was mistakenly turned on due to the button for the ignition reacting to touches. However, the data showed no changes to the temperatures other than that the CAC fan was running which can be due to a prior fan setting. The fan did not have any impact on the temperatures due to there being no fluid inside the pipes to cool. Note that the HSD condition was measured after the measurement for the idle condition was conducted and not directly after a warm-up run.

The uphill testing was done by driving up and down the hill with the intent of exerting the components to higher loads. The repetitive sequence was due to the fact that the hill was not long enough for the temperatures to reach maximum, so the focus was instead to drive the bus until the temperatures were stable enough to get some valuable data. In an ideal scenario, the bus would have been loaded with extra weight and the hill would be infinitely long so that the applied load would reach around 100 percent. Due to lack of resources and time the bus could not be loaded with extra weight.

Lastly, the constant 50 km/h driving condition faced some initial problems due to the shape and limits of the course. With sharp corners, different speed limits and traffic, reaching a constant speed of 50 km/h proved to be quite difficult without putting the bus and the passengers at risk. In order to get reliable data for the 50 km/h driving condition, the high speed road on the course was utilized. However, caution was necessary due to the speed limit being 110 km/h which would lead to risk of disrupting other vehicles physical tests due to the bus building up unnecessary traffic. To surpass the issue, the bus was driven on the high speed road during lunch time to reduce the risk of encountering traffic and priority was given to other vehicles which intended to reach higher speeds during the run.

3.3 Simulation

As required by the simulations department, which holds the responsibility of performing the CFD simulations for Scania, a set of required data was needed. First, the boundary conditions in order to perform a steady state simulation, and a second set of coordinates in a 3D model corresponding to real life locations of the measurement points. The boundary conditions were given in the form of temperatures from the measurements, with a mean value taken from the different driving conditions and the corresponding bus speed, engine speed, fan speed, etc. Using MATLAB, the data was imported, sorted and evaluated. A grid was made with all the driving conditions and the corresponding mean values of the measurements, this includes the suburban driving condition with the different cases and scenarios. Data is shown graphically in Appendix A and Appendix B for the matrices.

Once the temperatures are gathered for each component, a set of coordinates were gathered from CATIA. Using a 3D model of the engine compartment, a group of spheres are placed in the location of the sensors from the physical test. Note that the only relevant sensors here are the ones that measure the surrounding temperatures which acts as comparison points. With the spheres places and their respective coordinates saved, a list was then sent to the simulation group, along with a 3D model which the simulations are based on.

After the measurement point coordinates were given to the simulation group the next step of the simulation preparations was to decide a driving course. Due to the sheer complexity of the driving course used in the physical test, a much simpler course was decided for the simulation which was a straight road. This is due to the time and resources that has to be utilized to build a complex driving course with various terrain alternatives.

As for the simulation itself, as mentioned in Section 2.6 there are three major aspects of heat transfer. Conduction, convection and radiation. The simulation done on all the driving conditions were done only considering convection, meaning that two important physical aspects of heat transfer were neglected. This is an important factor to consider when looking at the result as heat transfer is the major physical phenomenon taking place inside the engine compartment. In particular, when the temperatures for the components acting as boundary conditions were applied there was no temperature gradient for the components due to conductivity being neglected. Instead, the temperature was set as a mean value which was taken from the physical test measurements. Furthermore, due to the fact that radiation and conduction were not considered, the surface temperatures intended for comparisons were neglected since the results from the simulation would have been inconclusive.

4 Results

4.1 Results from Physical Testing

Table 4 shows the results from the physical test containing the defining data for the driving conditions. As mentioned previously , the City Suburban case was divided in different cases and scenarios, however for the simulations the mean value was taken for only one case. This is mostly due to the fact that the majority of the components did not differ in temperature depending on the case and scenario, and the ones that did could be neglected due to not showing much difference.

Table 4: Driving conditions for the physical tests.

Case	**City Suburban**	**Idle**	**Hot Shut Down**	**Uphill**	**Constant 50**
Engine Speed [rpm]	912	600	0	1149	1064
Vehicle Speed [km/h]	34	0	0	18	52
CAC Fan Speed [rpm]	2264	1500	2000	1931	2130
RAD Fan Speed [rpm]	1904	150	150	2065	882
Engine Load [%]	40	16	0	51	29
Massflow Exhaust Pipe [kg/s]	5	2	0	7	5
Ambient Temperature [°C]	9	14	14	8	8

Table 5 is to represent the components and their surface temperatures. They will act as boundary conditions along with the driving condition. The temperatures are hidden in Table 5 since the data belongs to Scania and is confidential. Two new additions to the boundary conditions has been added compared to the speculated ones from section 2.2.3 and some taken out. From the results of the measurements, some components did not live up to the expected temperatures and were not considered as crucial for the temperature distribution in the engine compartment, such as the charge air cooler pipe. However, other components as the wastegate and the air compressor showed quite high surface temperatures. For that reason they were considered as important components for the boundary conditions. Furthermore, the fluid temperatures taken will be addressed in Appendix B. This is due to the simulation not providing any values for the fluid temperatures, therefore are not as relevant for the comparison. Note that the measurement point which is referred as Engine Block is a mean value surface temperature of all the measured cylinders, due to the surface temperatures for the cylinders are very similar.

Table 5: Measured surface temperatures in [°C] for the boundary conditions from the physical tests.

Case	City Suburban	Idle	Hot Shut Down	Uphill	Constant 50
Turbo					
Alternator					
Exhaust Pipe Alu 1					
Exhaust Pipe Alu 2					
Exhaust Pipe Skarv 1					
Exhaust Pipe Behind Silencer					
Exhaust Pipe End					
Engine Block					
Silencer Front					
Silencer Top					
Silencer Left					
Silencer Right					
Wastegate					
Air Compressor					

Table 6 is to shows the measurement results for the surrounding temperatures which act as comparison points. The data belongs to Scania and is confidential, therefore no data is shown in Table 6. The simulation provides the surrounding temperatures at the same position reflecting reality regarding the sensors, and the comparison will be discussed in Section 4.2. Furthermore, the surrounding temperatures were also divided up to cases and scenarios, a long with the surface temperatures but given in terms of mean values for one lap of the test track for the same reason mentioned above. In fact, the difference between the scenarios was minimal regarding the surrounding temperature.

Table 6: Measured surrounding temperatures in [°C] for the comparison points from the physical tests

Case	City Suburban	Idle	Hot Shut Down	Uphill	Constant 50
Alternator Inlet Facing End of Bus					
Alternator Inlet Facing Turbo					
Alternator Outlet					
Cylinder 1					
Cylinder 6					
Above Middle of Engine Block					
Above Turbo					
Around Turbo					
In front of RAD Fan					
In front of CAC Fan					
Inside the Cooling Shack					
Around Starter					
End of the Bus					
EMS					
Left Side of Silencer					

4.2 Results from Simulation

Table 7: The comparison temperature points from the simulation in [°C] for all the driving conditions.

Case:	City Suburban	Idle	Hot Shut Down	Uphill	Constant 50
Alternator Inlet Facing End of Bus					
Alternator Inlet Facing Turbo					
Alternator Outlet					
Cylinder 1					
Cylinder 6					
Above Middle of Engine Block					
Around Turbo					
In front of RAD Fan					
In front of CAC Fan					
Inside the Cooling Shack					
Around Starter					
End of the Bus					
EMS					
Left Side of Silencer					

5 Comparison of results

In this section of the report, the results from the physical test and the simulation are be compared, as well as the previous simulation results and the current simulation results. Before the comparison can be made, some general information about the temperature distribution in the simulation is provided.

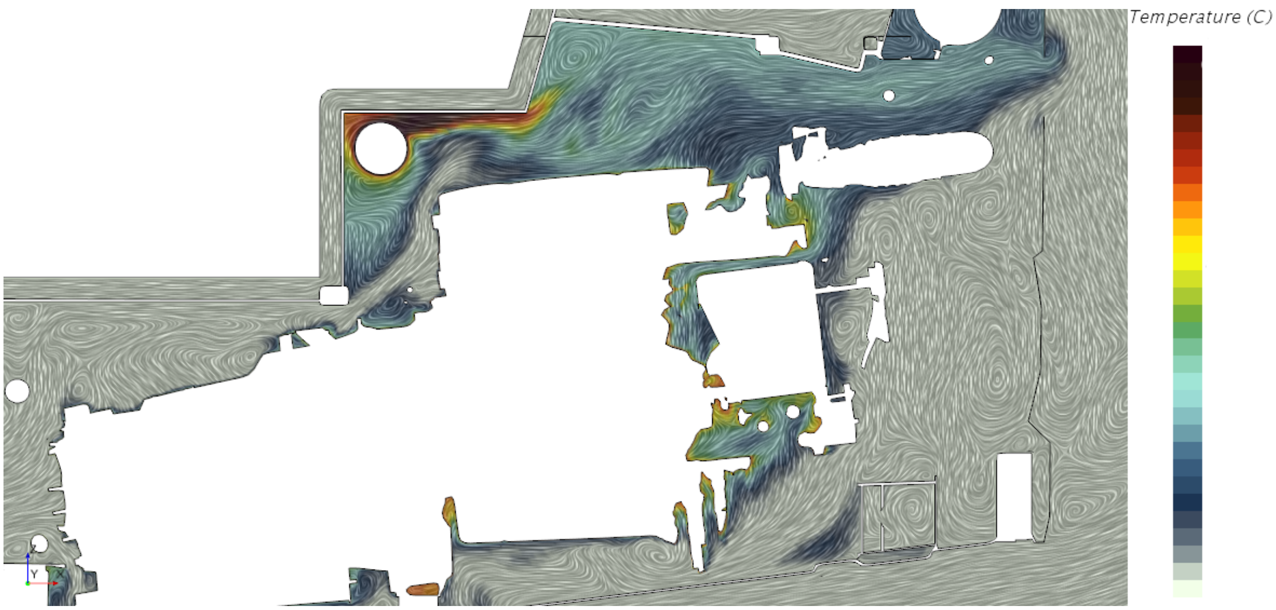

Figure 16: Temperature distribution across the engine compartment.

Figure 16 shows the temperature distribution obtained during the simulations. The temperature distribution across the engine compartment is quite low while the majority of the heat is condensed around the engine components such as the turbo and engine block. Note that the temperature gradient is quite sharp, thus leading the temperature variation to be large within a short distance interval. As top left corner of the engine compartment, a heat pocket is seen with a quite extreme temperature gradient, and a distance of just a few centimeters can be the difference between being above 100 °C and below 50 °C.

5.1 City Suburban

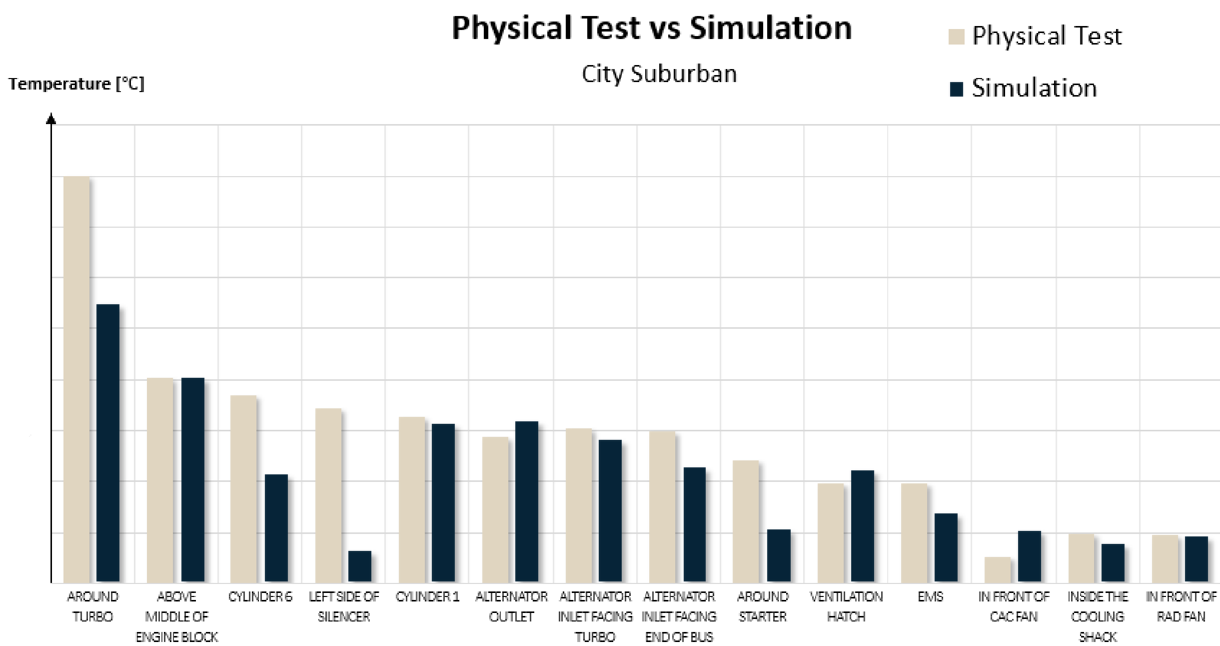

Figure 17: Comparison of the results for the physical test and simulation for the City Suburban driving condition.

Figure 17 represents the comparison between the physical test and the simulation of the City Suburban driving condition. As be seen in the bar chart, there are four comparison points with large deviations, which are the Turbo, Cylinder 6, Starter and Silencer, the rest of the comparison points have minor differences between the physical test and the simulation. The total average difference between the physical test and the simulation for the City Suburban condition is 25%, which is well over the initial goal. It is worth mentioning that the "Left side of silencer" point has been excluded, the reason for this is that after completing the simulation, it was discovered that the model used for the silencer in the simulation was not the same as in the physical model, which explains the big contrast in temperatures. Figure 18 shows the delta between the physical test and the simulation for each comparison point in terms of degree Celsius and percentage. The biggest difference in percentage is seen on the point "In front of CAC fan". However, the comparison point with the biggest difference in terms of temperature is the "Around turbo" with the exception of "Left side of silencer" which has been removed from the average and median due to it being considered a defect measurement point. Furthermore, it can be seen that the median for the City Suburban condition is 17%.

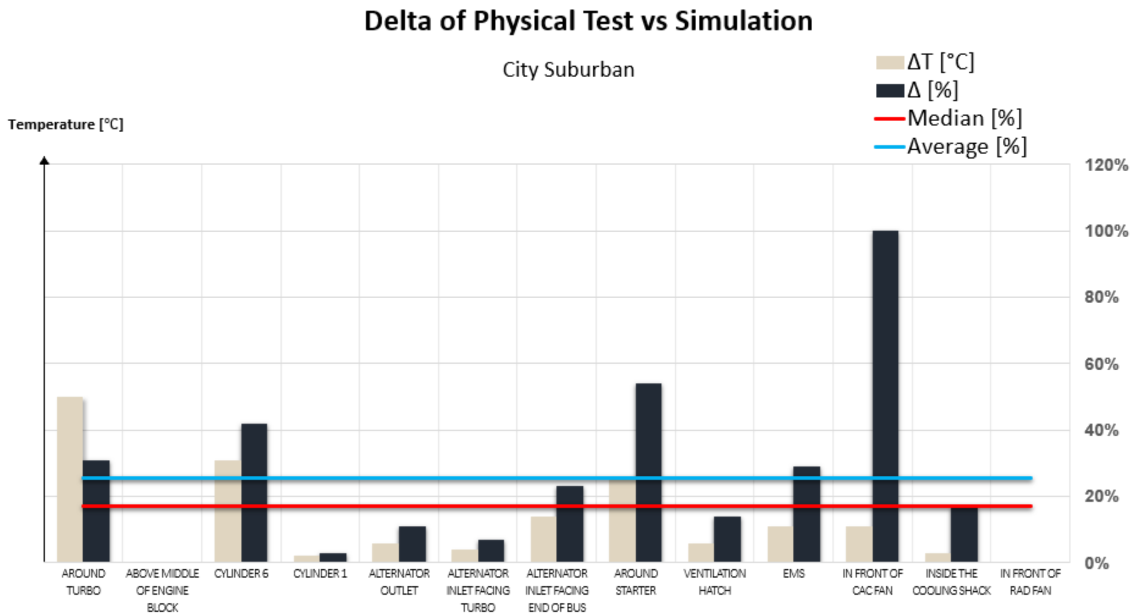

Figure 18: Delta and the median between the physical test and simulation for each comparison point in the City Suburban driving condition.

Figure 19 presents the measurement point of the turbo in the simulation, and the surrounding temperature distribution. A close inspection of the measurement point shows, that the temperature gradient is not evenly distributed inside the sphere. Furthermore, the temperature variations inside the sphere is quite extreme in some areas of the sphere and the way STAR-CCM+ calculates the temperature is by averaging every point inside the specified volume. Since the simulation is done in steady state, the stream lines are stationary, meaning that there are no fluctuation, confirming that the positioning of the virtual measurement point is crucial in getting an accurate result that is in line with the physical test. Determine the exact location of the placed measurement sensors from the physical test and to find those locations in the 3D model is an extremely hard task to complete. So naturally, the location of all the comparison points differ by some random margin between the physical test and the simulation.

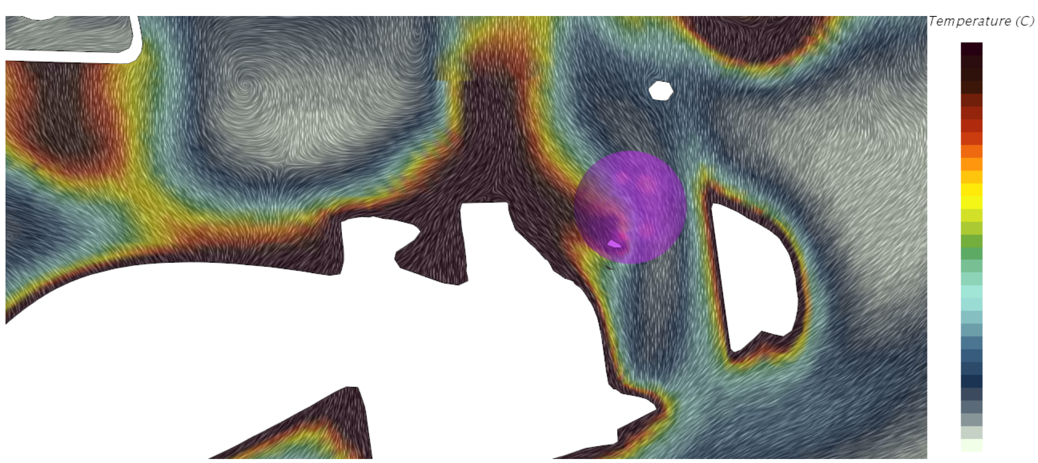

Figure 19: Temperature distribution around the turbo with its corresponding measurement point.

As for the Starter, the temperature distribution around the measurement point, as it can be seen in Figure 20, is quite different from the physical test which is reflected in the results. Note that the position of the measurement points in the simulation and the physical test are dissimilar for the Starter. However, even if the positions do not match, the reason why the temperatures diverge is not clear. This is due to the fact that the positions for both cases are still inside the same area regarding the temperature distribution where the temperature is the same in the simulation.

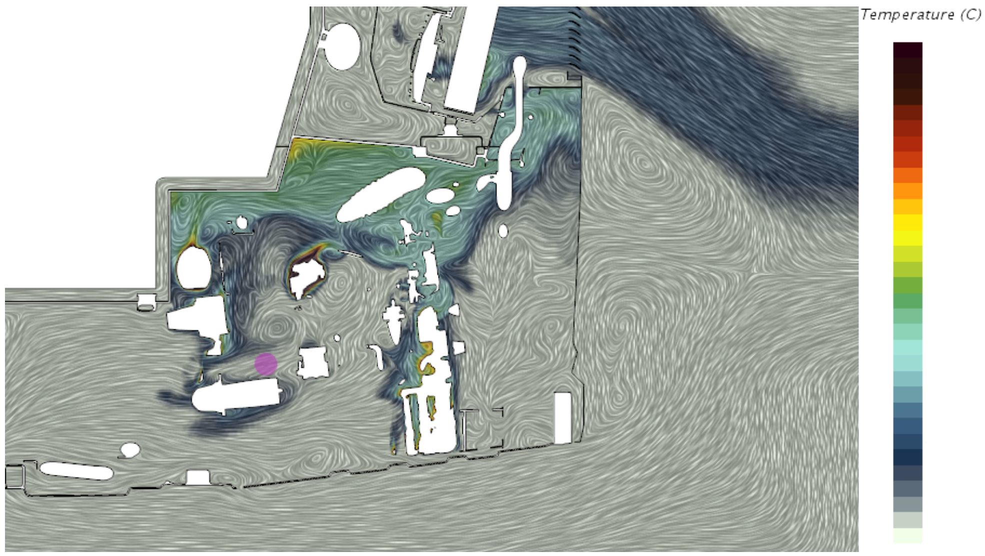

Figure 20: Temperature distribution around the starter with its corresponding measurement point (the purple circle).

5.2 Idle

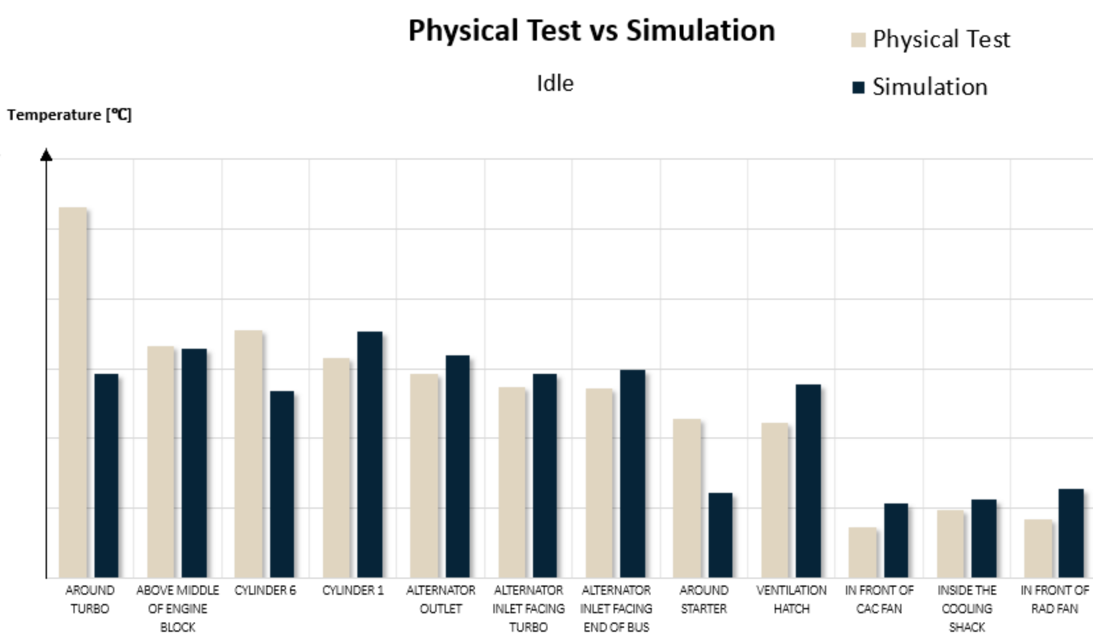

Figure 21: Comparison of the results for the physical test and simulation for the Idle driving condition.

Figure 21 shows the comparison between the physical tests and the simulation for the Idle driving condition. Here, the same pattern for the same comparison points can be seen in terms of the deviation when comparing reality to simulation. As mentioned previously, the position of the virtual measurement point is important in a steady state simulation in order to get an accurate result. This argumentation is further strengthened by the fact that the same comparison points show deviations for the City Suburban condition as for the Idle condition.

For the Idle case there is no forced ventilation throughout the engine compartment, as the bus is standing still, and this is due to the fact that Skruttan does not have a forced air circulation except from the alternators. Note that the speed of the bus is 3.6 m/s in the simulation, this is due to the way STAR-CCM+ simulates, thus required the bus to have a low speed which initiates some small forces, stream lines of air etc. This applies to both the Idle and Hot Shut Down driving conditions. Despite the dissimilarities, the total average difference between the physical test and the simulation for this condition is 25%, which is considered a good results compered to the other driving conditions.

5.3 Hot Shut Down

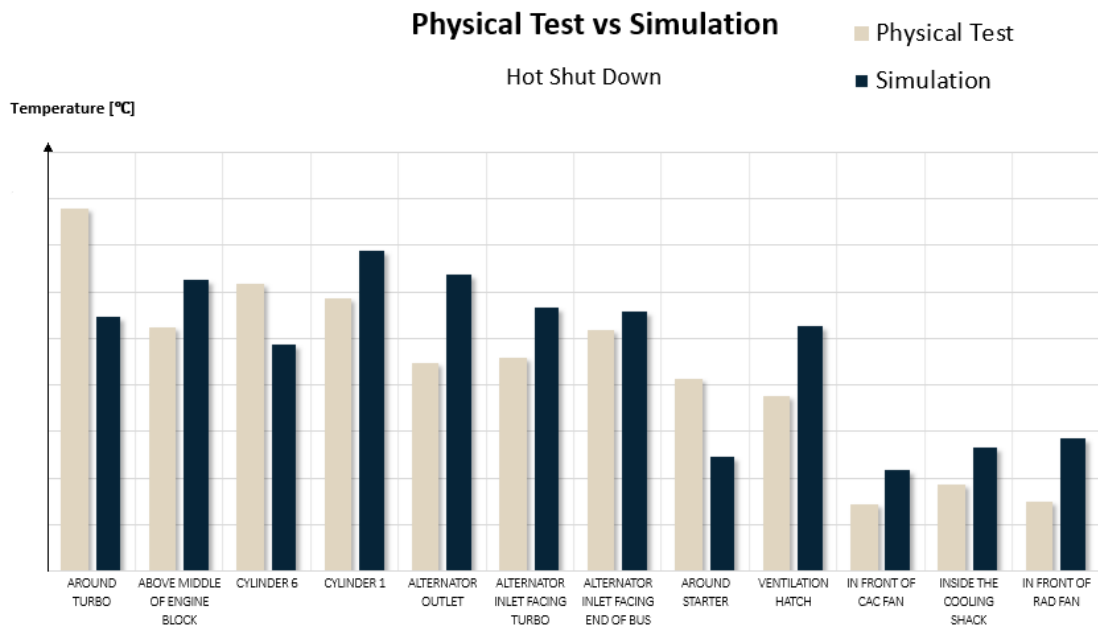

Figure 22: Comparison of the results for the physical test and simulation for the Hot Shut Down driving condition.

As seen in Figure 22 the deviations for the Hot Shut Down driving condition differ from the previous mentioned conditions. This is due to the major differences between the physical test and simulation. During the physical test, all components inside the engine room were turned off as the engine was shut down. However, in the simulation the two alternators were not shut down resulting in a forced circulation of air around the alternators. The reason for this is that the simulation setup from the beginning is made this way and there was not an alternative where they could be totally shutdown. So it had become clear at this point that the simulation had limits in some aspects regarding the idle and shut down cases. This is reflected upon the results for the alternator outlet and ventilation hatch where the temperatures for the physical test and simulation differs. This is reflected in the total average difference between the physical test and simulation for the HSD driving condition which is 36%.

5.4 Uphill

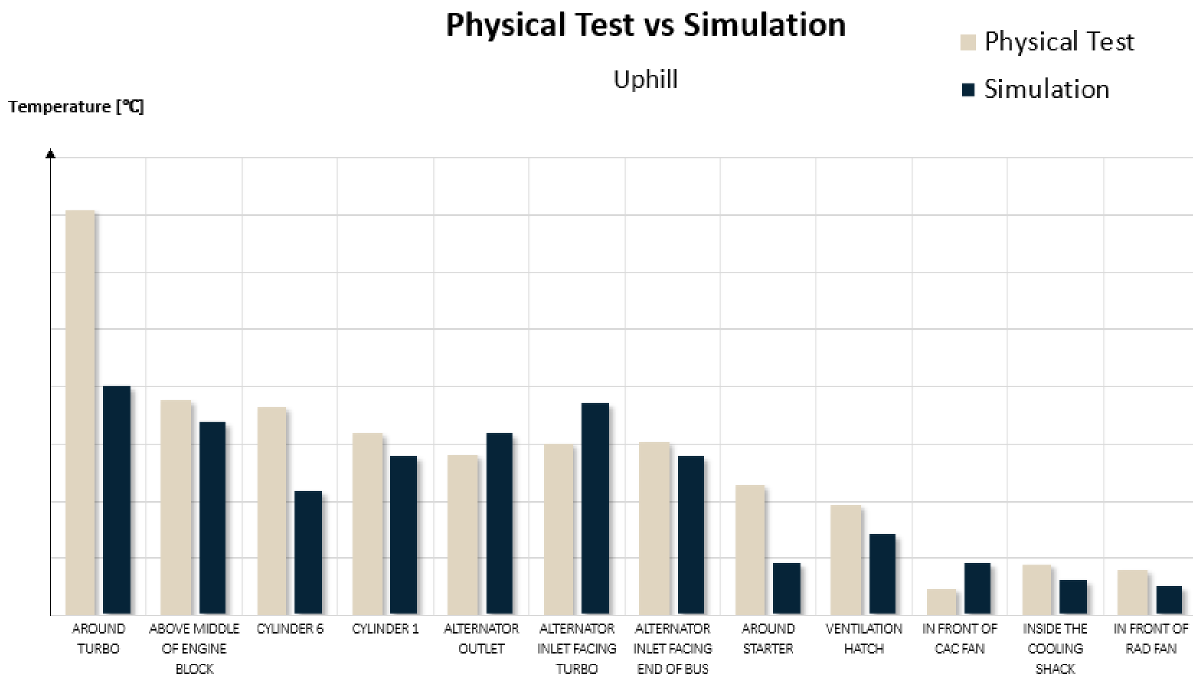

Figure 23: Comparison of the results for the physical test and simulation for the Uphill driving condition.

For the Uphill driving condition, the same deviations occur as for the City Suburban condition. As seen in Figure 23, the Turbo, Starter, and Cylinder 6 show the same type of behaviour. Again, this is due to the virtual measurement point location and the direction of the stream lines. The total average difference between the physical test and simulation for the Uphill driving condition is 42%, which is the highest for all the driving conditions. This uphill driving case is the most extreme case since it requires the most amount of work out of the engine, which will automatically lead to higher local temperatures from the components. This will result to giving higher surrounding air temperature through radiation in the engine compartment. The first major difference is that the simulation and physical test has been executed differently. Second difference has to be that, since this is the most extreme case, small dissimilarities or faults from simulation and test will result in an higher delta between the two. This delta difference is relative to the driving condition, where the delta increases as the as the conditions becomes more complex.

5.5 Constant 50

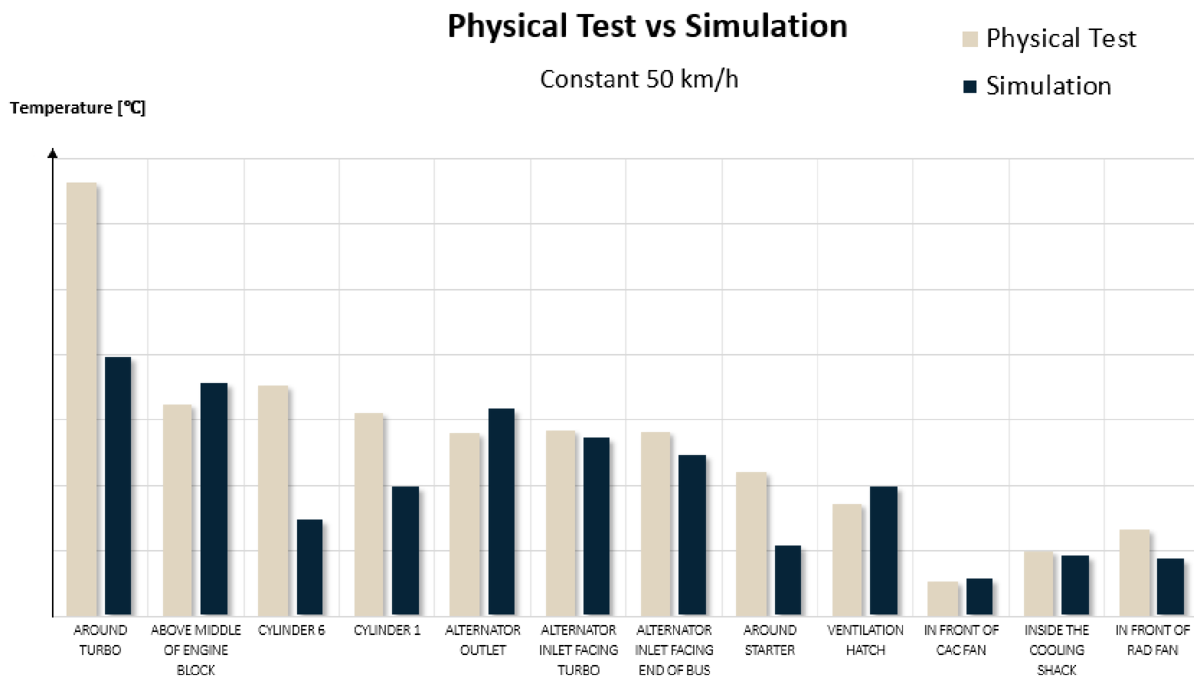

Figure 24: Comparison of the results for the physical test and simulation for the Constant 50 driving condition.

The comparison points for the driving condition Constant 50 shows the same type of behaviour as Uphill and City Suburban. As seen in Figure 24, the same points differs for the physical test and simulation with the exception of Cylinder 1. The total average difference between the physical test and simulation for the Constant 50 driving condition is 24%, which is the best result in terms of the difference for all the driving conditions. Note that during the simulation for the Constant 50 driving condition, the temperature experienced a lot of oscillations leading to rougher estimates compared to the other driving conditions. However, the simulation for the Constant 50 condition reflects the physical test better in terms of the driving course which is essentially the same for both. This shows that even though the the test and the simulation match, it still gives large differences on some of the points, such as the Turbo and the Cylinder 6, where the simulation really underestimates the temperatures. This is a universal problem for all the cases which indicates that the main fault lies in the simulation rather then the test, and more exactly around the Turbo and Cylinders.

5.6 Previous Simulation

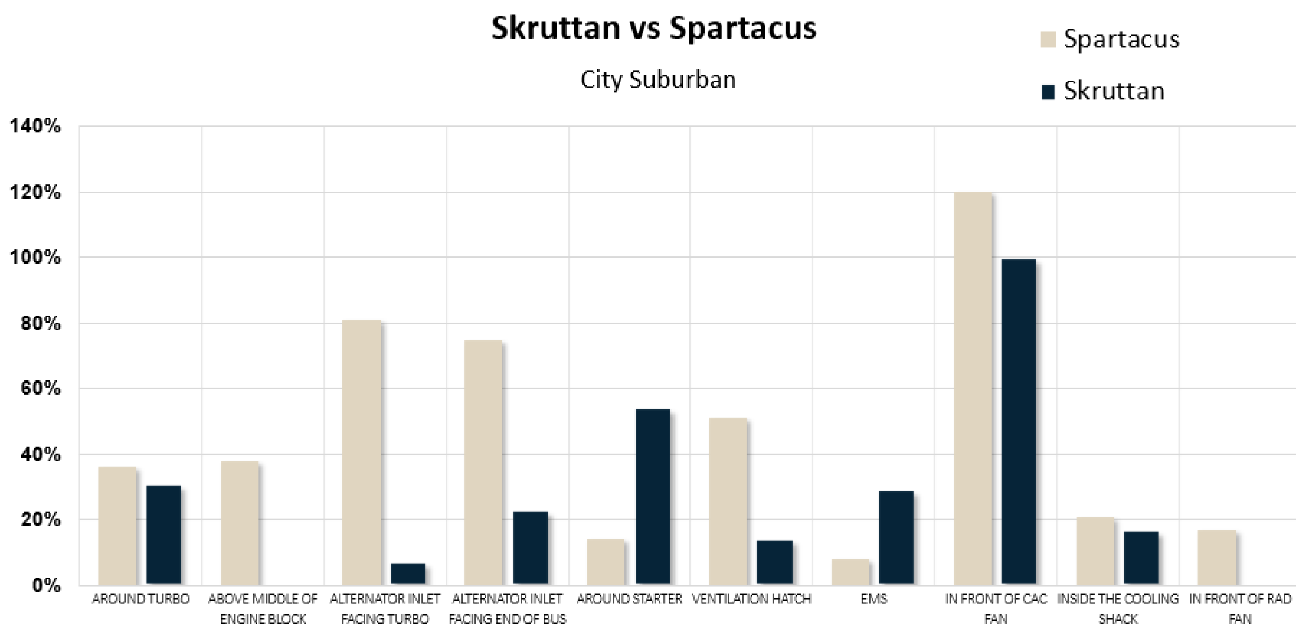

Figure 25: Comparison of the difference in % between physical test and simulation for Skruttan and Spartacus.

Figure 25 shows the differences between the comparison of the physical test and simulation for the bus Skruttan and Spartacus. As can be seen, the difference for Skruttan deviates less for the majority of the measurement points compared to Spartacus. In total the average difference for Skruttan is 25% compared to the average difference for Spartacus which is 46%. This means that an improvement of 54% has been made with the new method. While looking at the median which is 17% for Skruttan and 37% for Spartacus leading to a difference of 46%. Note that the comparison is only made between the measurement points that match between the both buses, hence why some measurement points are disregarded in the figure compared to the previous figures which contains all the measurement points that are relevant to Skruttan.

6 Conclusion

In conclusion, when looking back at the goals for this thesis, the question remains if a method has been found that will result in, less than 10% difference between the physical test and the simulation? From Section 5.6, the overall difference between reality and simulation is not within the ideal goal of less than 10%. However, comparing Spartacus using the old method, it is confirmed that the newly method is a big improvement and definitely a way in the right direction. The new method has shown an improvement of 54% in average. It is advised to also consider the median when comparing Skruttan to Spartacus where the difference is 46%, which provides confidence in the result in the new method.

When compering all the driving conditions with one other, it is seen that the best result between the physical test and the simulation is given by the Constant 50 driving condition with a total average difference of 24%. The second best is the Idle together with City Suburban condition which gives a difference of 25%. Lastly, the driving conditions Hot Shut Down and Uphill gives 36% and 42% difference respectively.

It is worth remembering that throughout the project, compromises were made for both the simulations and physical tests, which are reflected on the results since the final result could not live up to the ideal goal. However, it is worth mentioning that even though these compromises were made the result itself is an improvement leading to the fact that the new method is an improvement and a success.

7 Discussion

The project has been very educative and a great learning experience, as we came into the project with an academic mindset that was very different to the mindset of a large company such as Scania. The major differences were that, in the university you are not as dependant on other people as you are in a company. This was a culture shock for us, as we had to depend on other departments and colleagues in order to advance in the project. In retrospect there are a few improvements that we would like to discuss and how we could do things differently throughout the master's thesis work.

Firstly, the practical aspect of the project. Initially, we had a limited and general knowledge about the bus, its functionality and its major components. As we advanced in the project towards the physical test, we missed quite a lot of valuable information regarding the internal systems of the bus. This is mainly due to the fact that the systems are developed by different departments leading to the information of the systems not being as available to us as we wanted. This proven to be a minor complication during the physical test, and the data evaluation, where we had some components behaving in a manner which we would like to have known before conducting the physical test. If we had known about how these systems operated beforehand, we would be able to do the tests in a more resourceful way. One of those systems was the cooling system which turned out to be a very crucial part for the simulation but was completely overlooked in the physical test. Luckily, the CFD department was able to gather sufficient data to reflect our driving conditions from other sources. However, this is naturally something that must be taken into consideration for the next iteration of the project cycle as described in Section 1.3.

Another improvement that we would like to have implemented was the weight distribution inside the bus. By placing large water tanks in a distribution that reflects a crowded bus we would be able to increase the applied load in order to increase the overall temperature in the engine compartment. Which could also be done by testing in an environment with higher ambient temperatures. Furthermore, we would like to have tested the bus inside the climate chambers in Scania's facility, where we would have been able to apply a desired constant load which would essentially replace the driving conditions Uphill and Constant 50. By replacing these driving conditions with the climate chamber we would be able to match the driving courses between the simulation and physical tests since both cases utilizes a straight infinite road. Compared to our current situation, where the simulation had a driving course with a straight road while the physical test had roads with various terrain. Ideally, the driving courses for the physical test and simulation should be identical or closely representative.

Naturally, the execution of the physical test itself could have been improved. This would facilitate the similarities between reality and simulation. By acquiring better data by utilizing the driving sessions more, we would have been able to have a set of data that would have been more representative of the driving conditions. With more tests conducted the mean average of the data would have been more accurate in the sense that the noise and unexpected deviations would be reduced.

Strictly speaking of the simulation, there are a few improvements to implement there as well. Such as also performing a transient simulation since all of the driving conditions may not be able to to be represented by a steady state simulation. The transient simulation would provide the aspect of time which accurately represents reality and its impact would give a better understanding of the temperature distribution. Furthermore, we would like to advice to include radiation and conduction in the simulation. Since as of now, there is no clear answer as to how much the results are effected by neglecting two major aspects of heat transfer. We suspect that including these two physical aspects to the simulation would lead a better and more accurate representation of reality and in the comparison between the physical test and simulation.

We would think that implementing the above mentioned improvements would lead to achieving more accurate results when comparing simulations with the physical test. Initially, some of these improvements could have been implemented by us during the project timeline where we would be able to re-access the changes on the second iteration. This was of course intended, sadly the test vehicle Skruttan was given a driving ban due to some unforeseen issues with the wheel and gear box leading us to dismiss three weeks of the scheduled testing period. Hence, the reason for why all the measurement points did not match between the driving conditions and new measurement points were not able to be added to further strengthen our method. Finally, we would like to clarify that the simulation must be developed further while the physical tests must be improved.

8 Future Work

Moving forward for future projects a few things must change in order to achieve a better and accurate result. Note that the method itself is a step in the right direction and the project layout as described in Section 1.3 should be the foundation for any future work regarding this topic. By following the project layout and implementing the changes mentioned in Section 7, the method can be improved by completing iterations of the project cycle. Since as of now, the simulation has to develop on a few aspects and by acquiring new data from the physical tests one is able to continuously provide the necessary information in order to build a model that reflects reality more accurately. However, the improvements for the physical test must also be implemented as they will act as a reference for the simulation and cannot be neglected if progress of the method is desired. This can be done by utilizing the climate chambers in Scania. Note that the climate chambers must be developed further in order to reflect reality better in terms of the streamline behaviours. By combining the climate chambers and the CFD simulations, small steps can be taken in order to complete the method and minimize the difference in results between the physical tests and simulation. A good way to start is by running the simulation again with measurement points that reflects the position of the physical test more accurately and by focusing on making sure that the steady state simulation results are completely reliable before moving onto the transient simulations. The recommendation here is to get the average difference between all the comparison points closer to the median.

Furthermore, Scania should focus on expanding the method so that an universal method can be developed and used for any desired vehicles. Naturally, the boundary conditions may differ depending on the bus configuration but could always be adjusted accordingly by following the method.

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A Appendix: Measurement Points

Surface Temperatures

Turbo

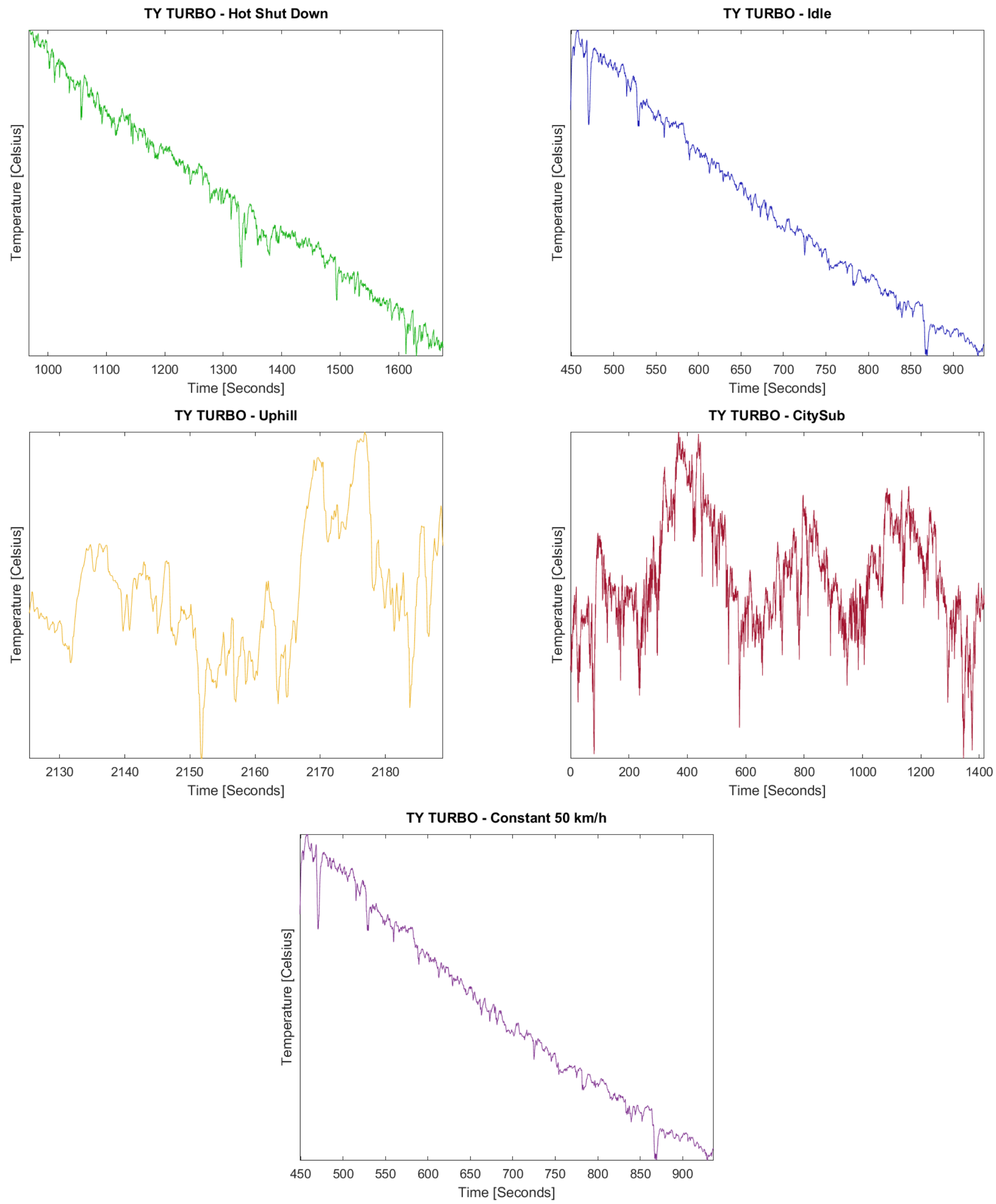

Figure 26: Surface temperatures for turbo in various driving conditions.

Alternator

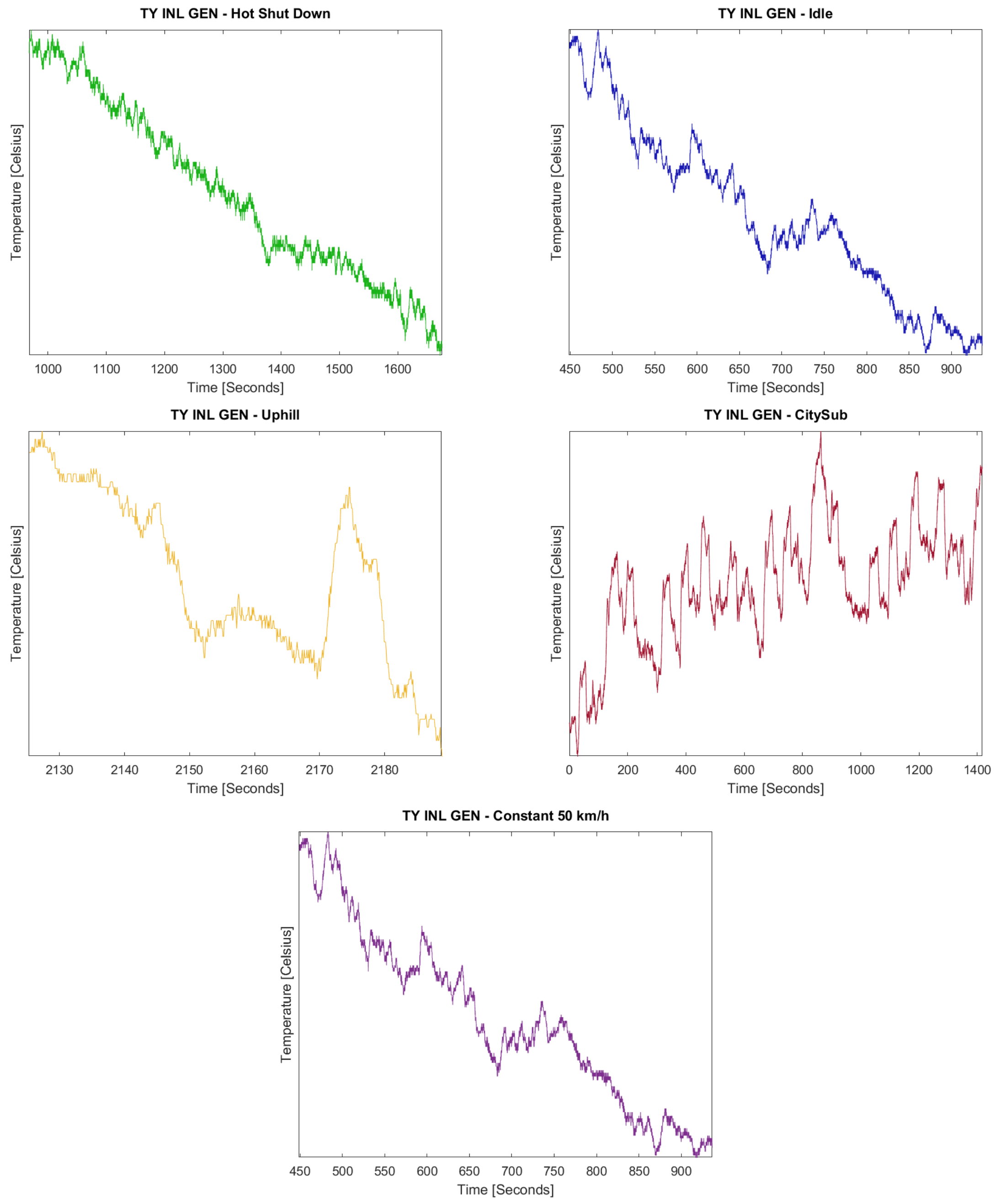

Figure 27: Surface temperatures for alternator in various driving conditions.

Exhaust Pipe

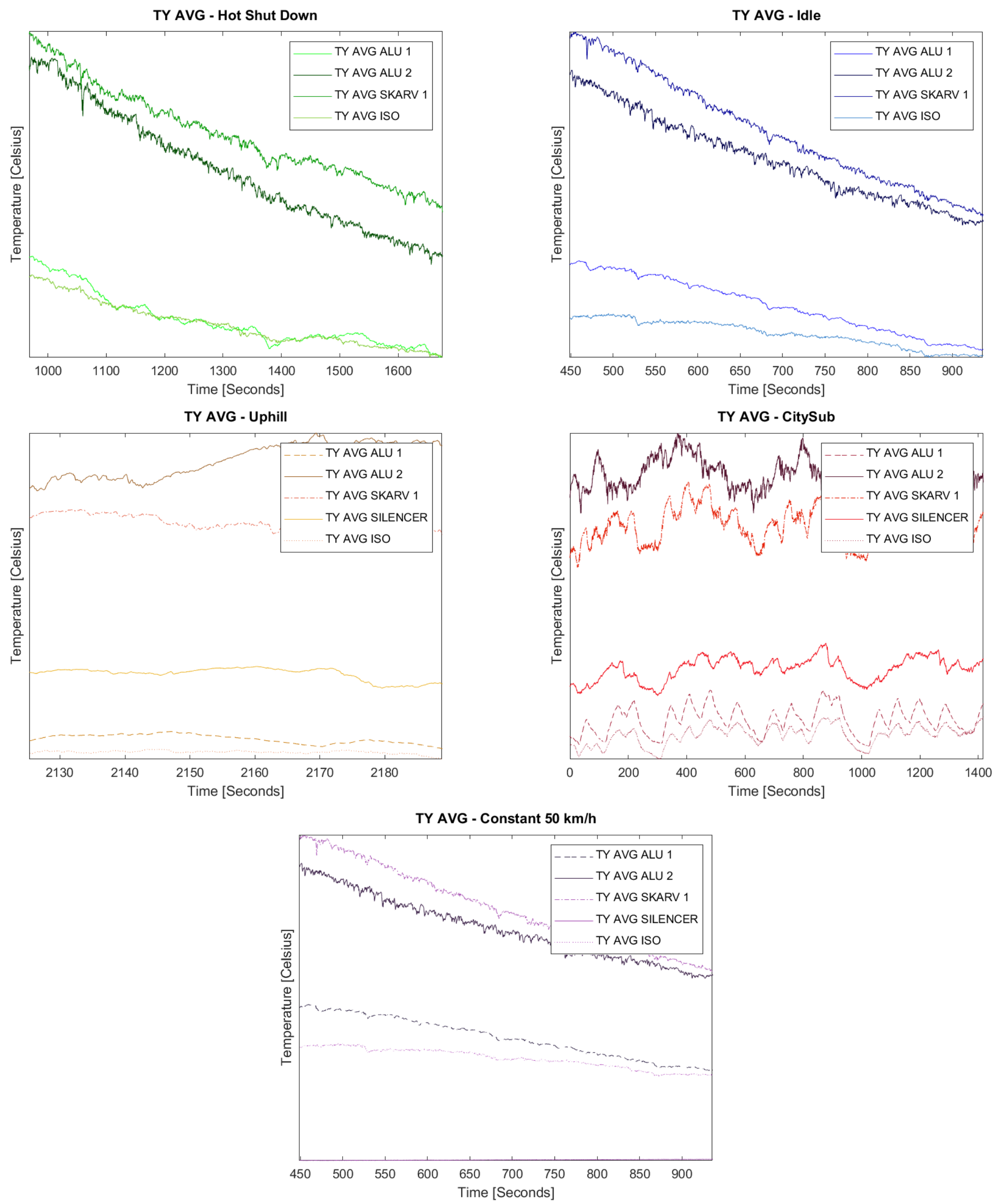

Figure 28: Surface temperatures for exhaust pipe in various driving conditions.

Charge Hose

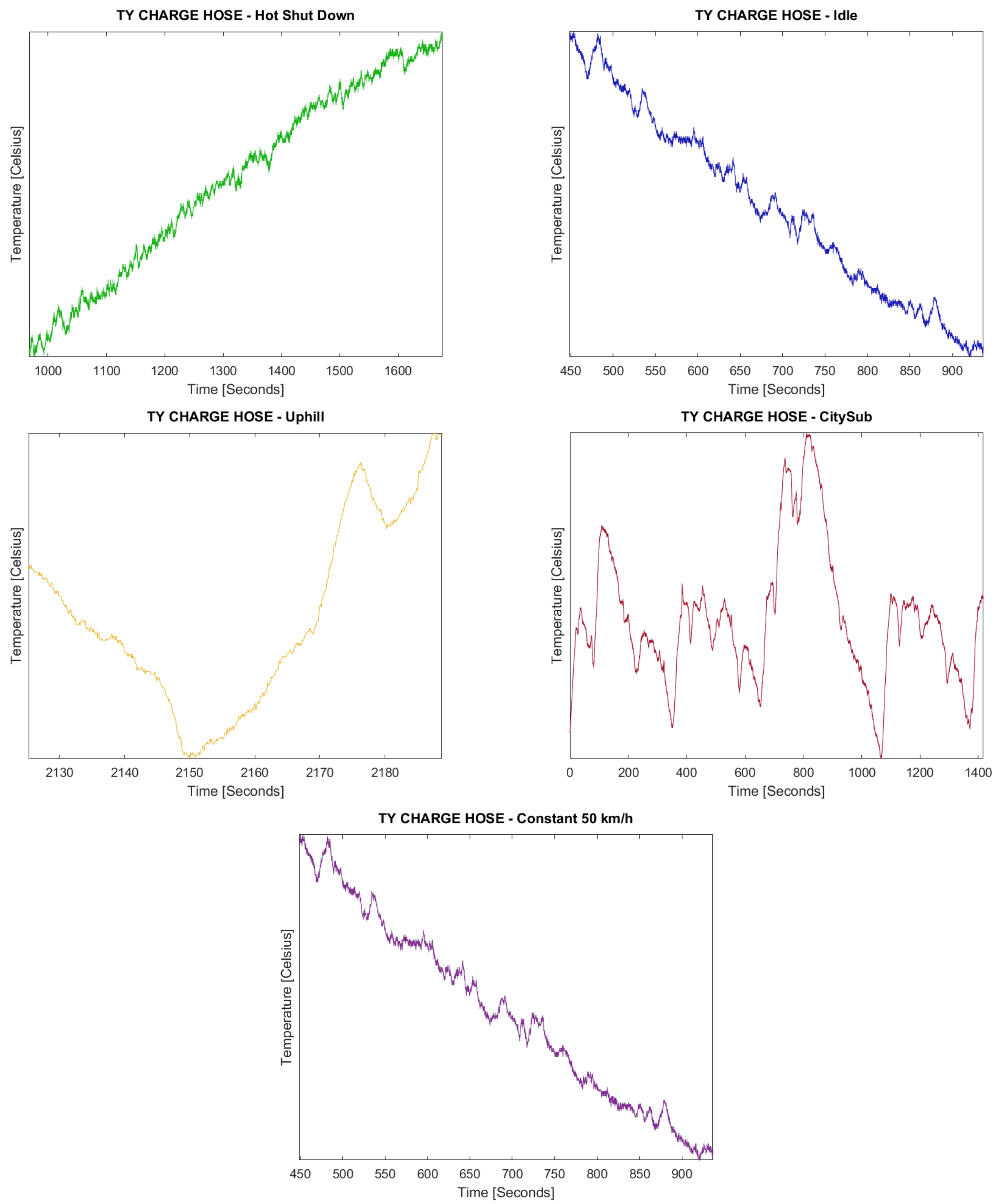

Figure 29: Surface temperatures for charge-hose in various driving conditions.

Engine Block

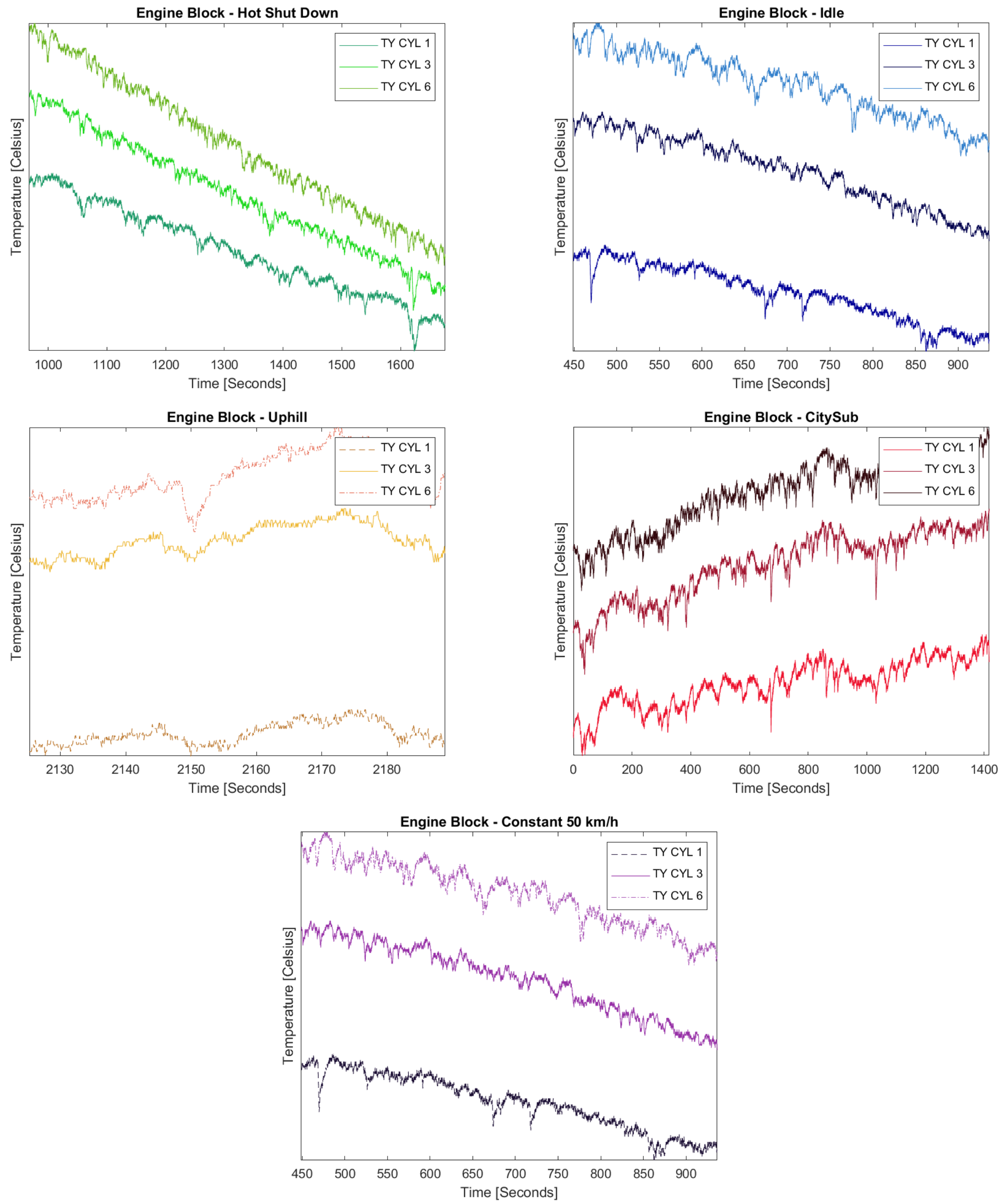

Figure 30: Surface temperatures for engine-block in various driving conditions.

Filter

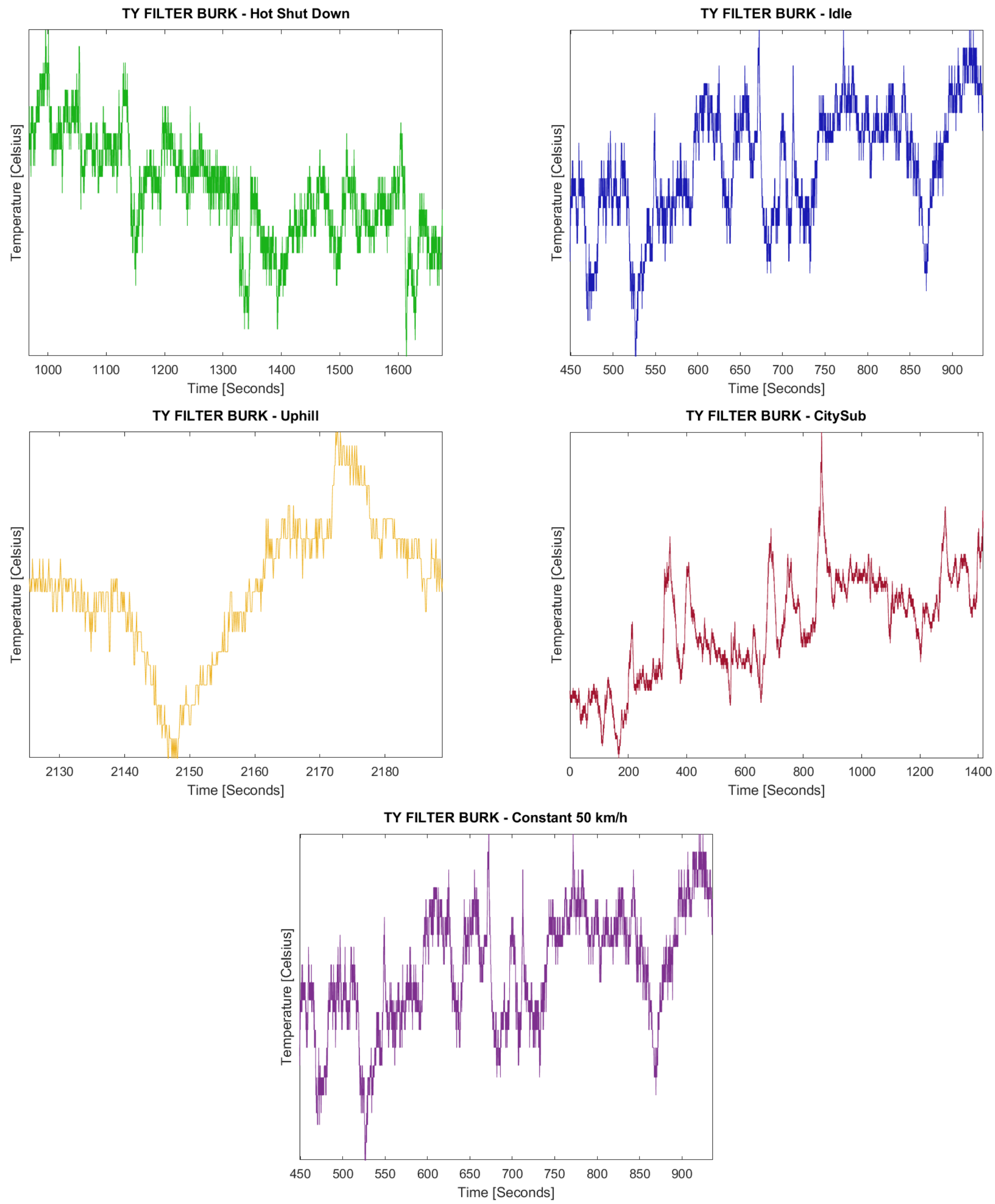

Figure 31: Surface temperatures for filter in various driving conditions.

Air Compressor

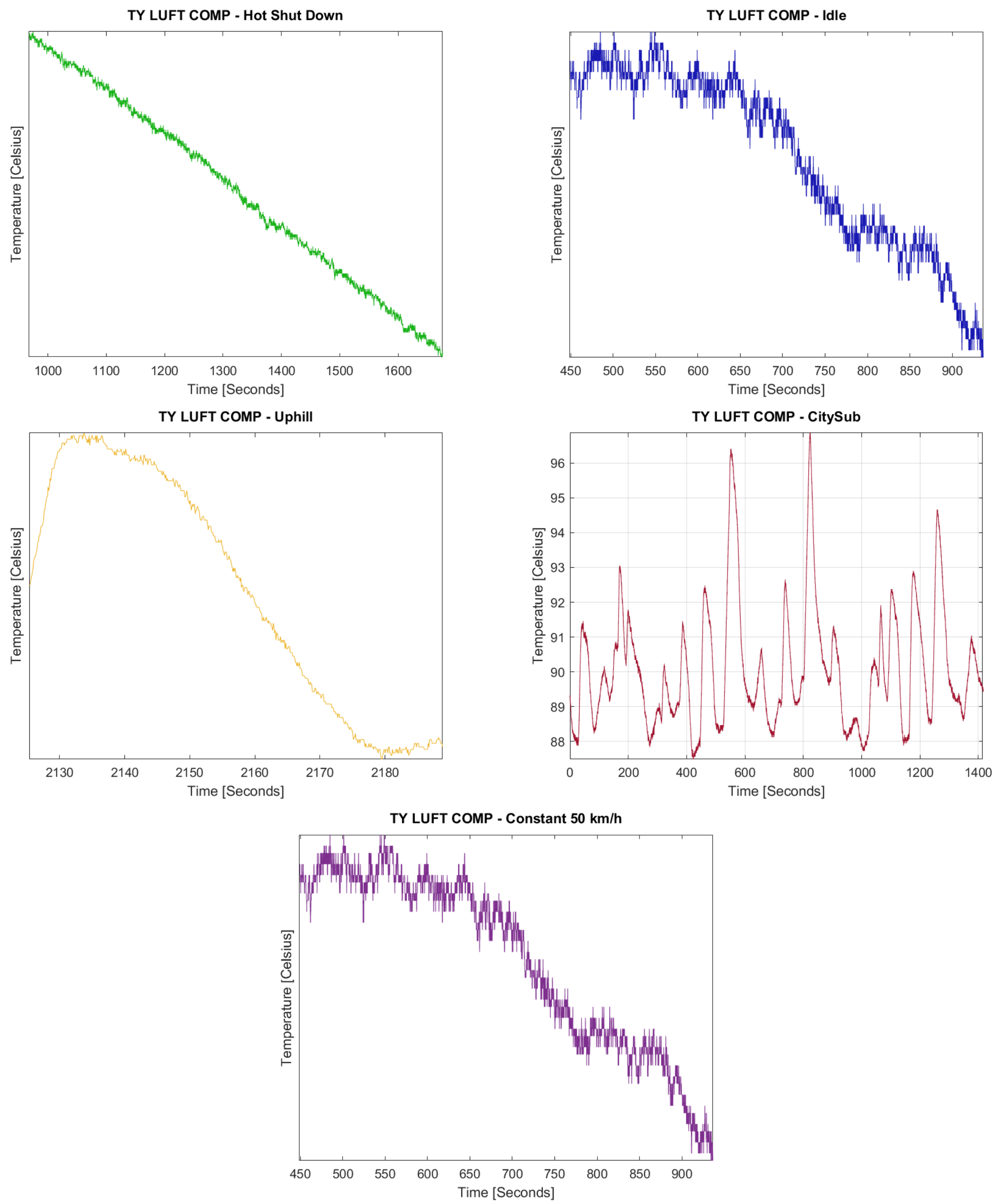

Figure 32: Surface temperatures for air-compressor in various driving conditions.

Air Compressor Hose

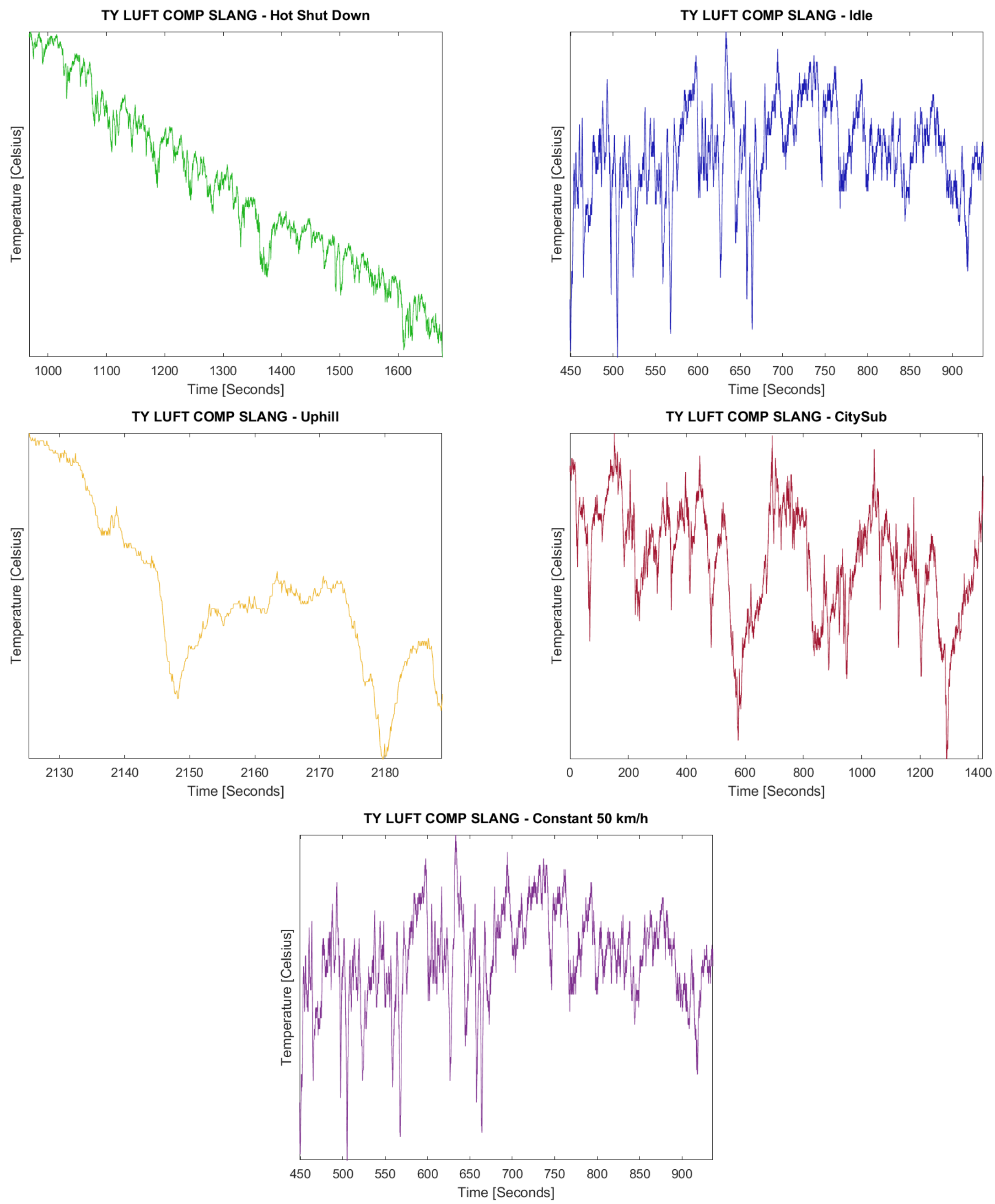

Figure 33: Surface temperatures for air-compressor-hose in various driving conditions.

Rubber inlet-pipe Compressor

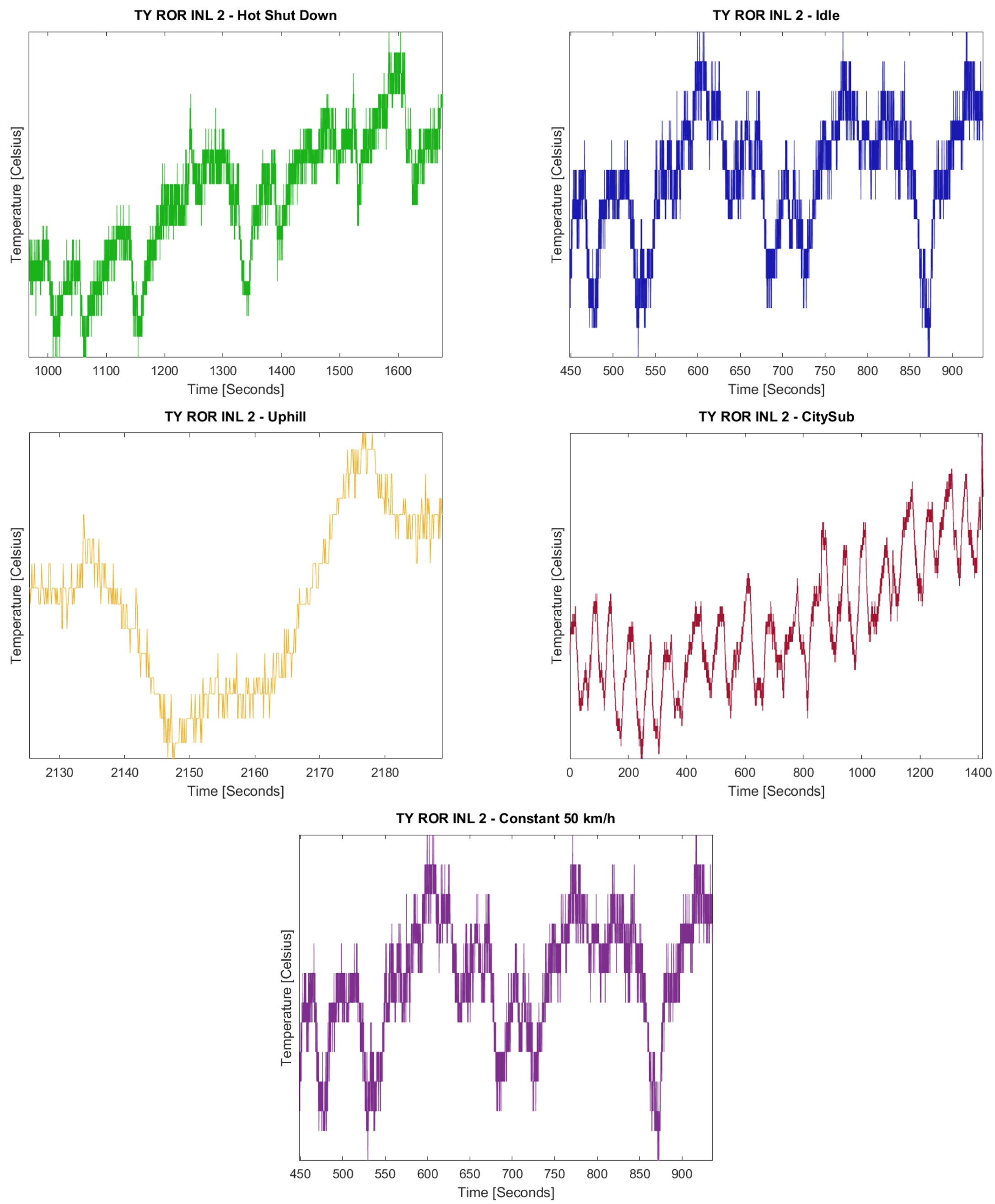

Figure 34: Surface temperatures for inlet of the rubber-pipe going into the turbo-compressor, in various driving conditions.

Rubber outlet-pipe Compressor

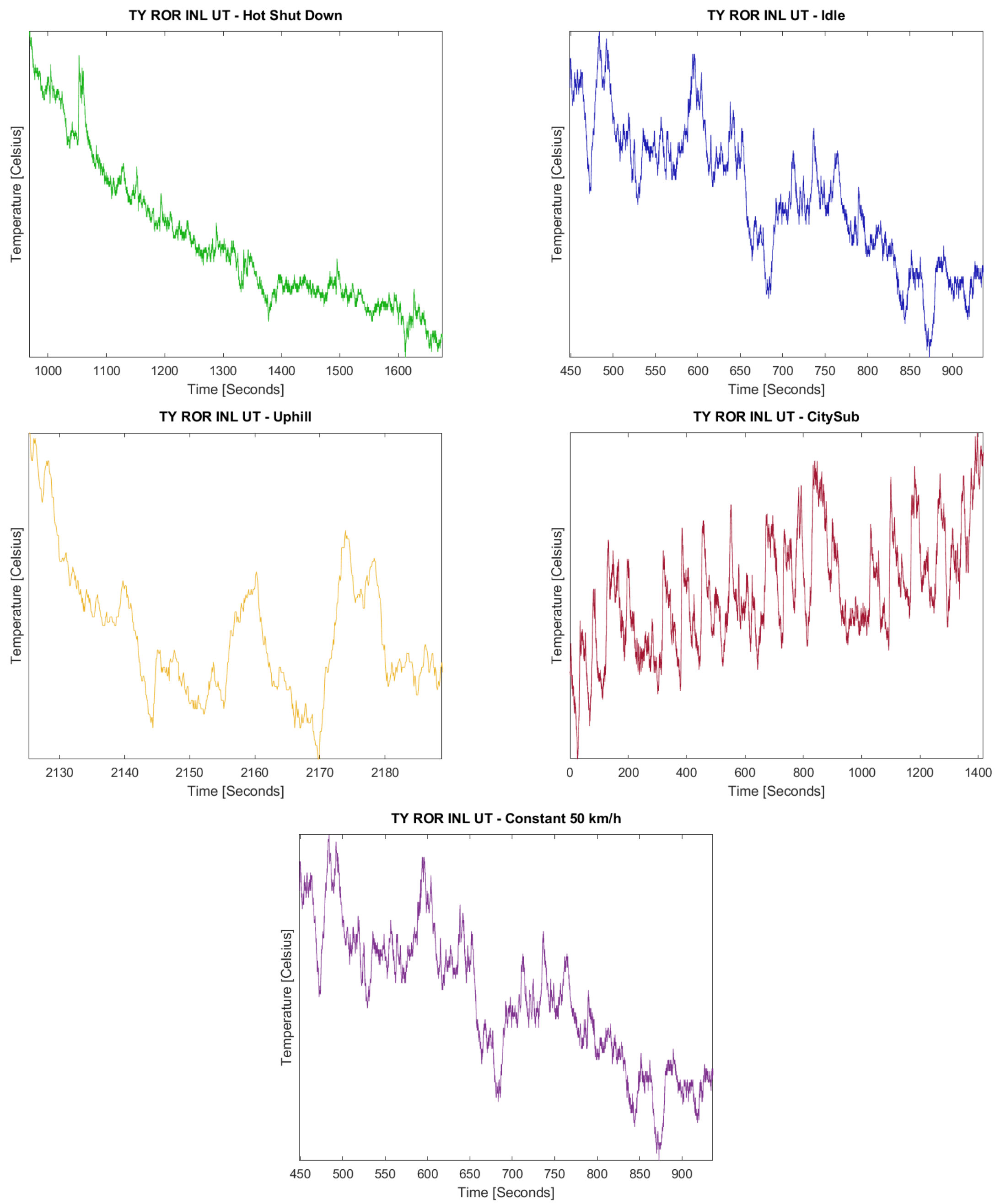

Figure 35: Surface temperatures for outlet of the rubber-pipe going into the turbo-compressor, in various driving conditions.

Silencer

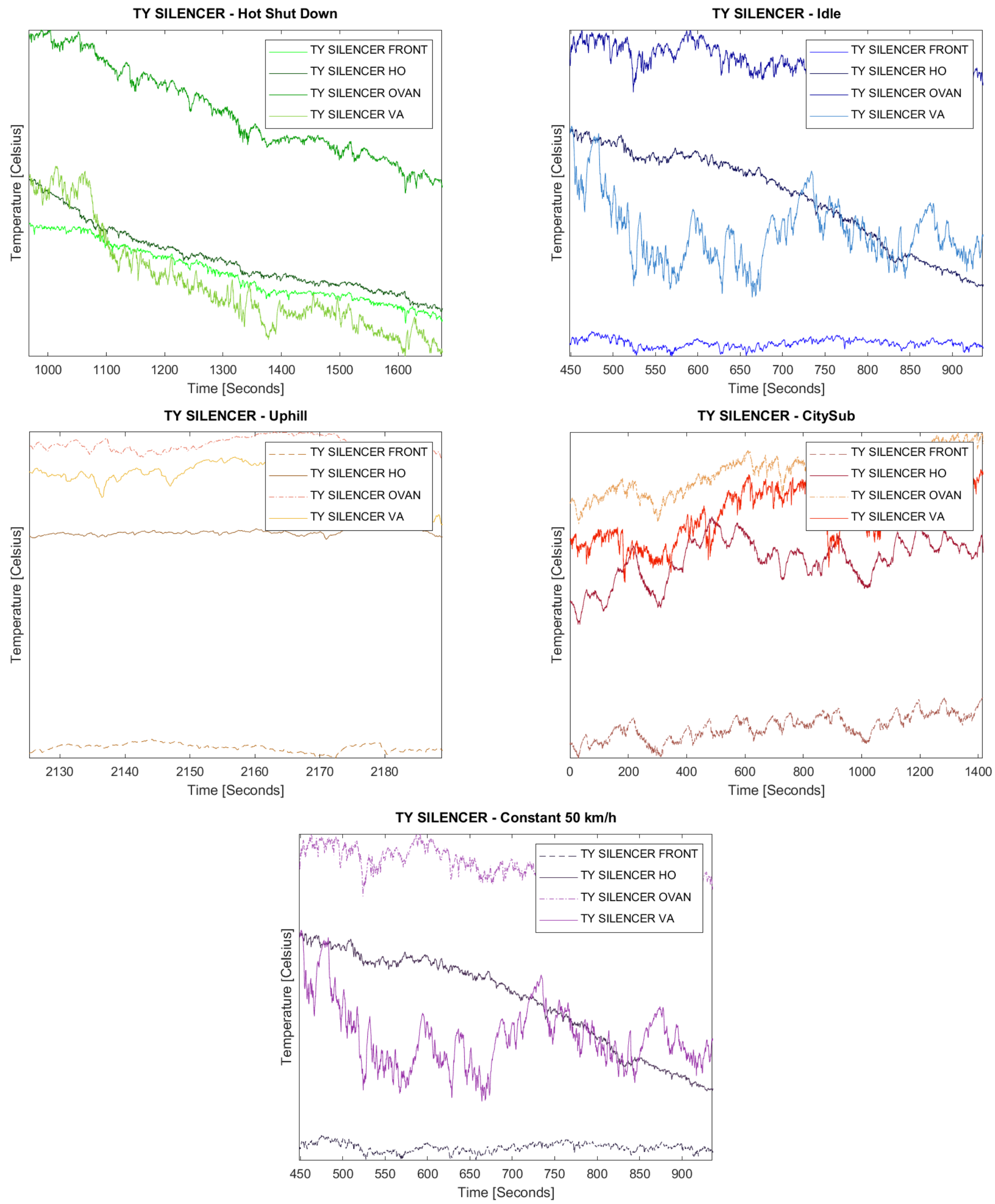

Figure 36: Surface temperatures for Silencer in various driving conditions.

Turbo Compressor

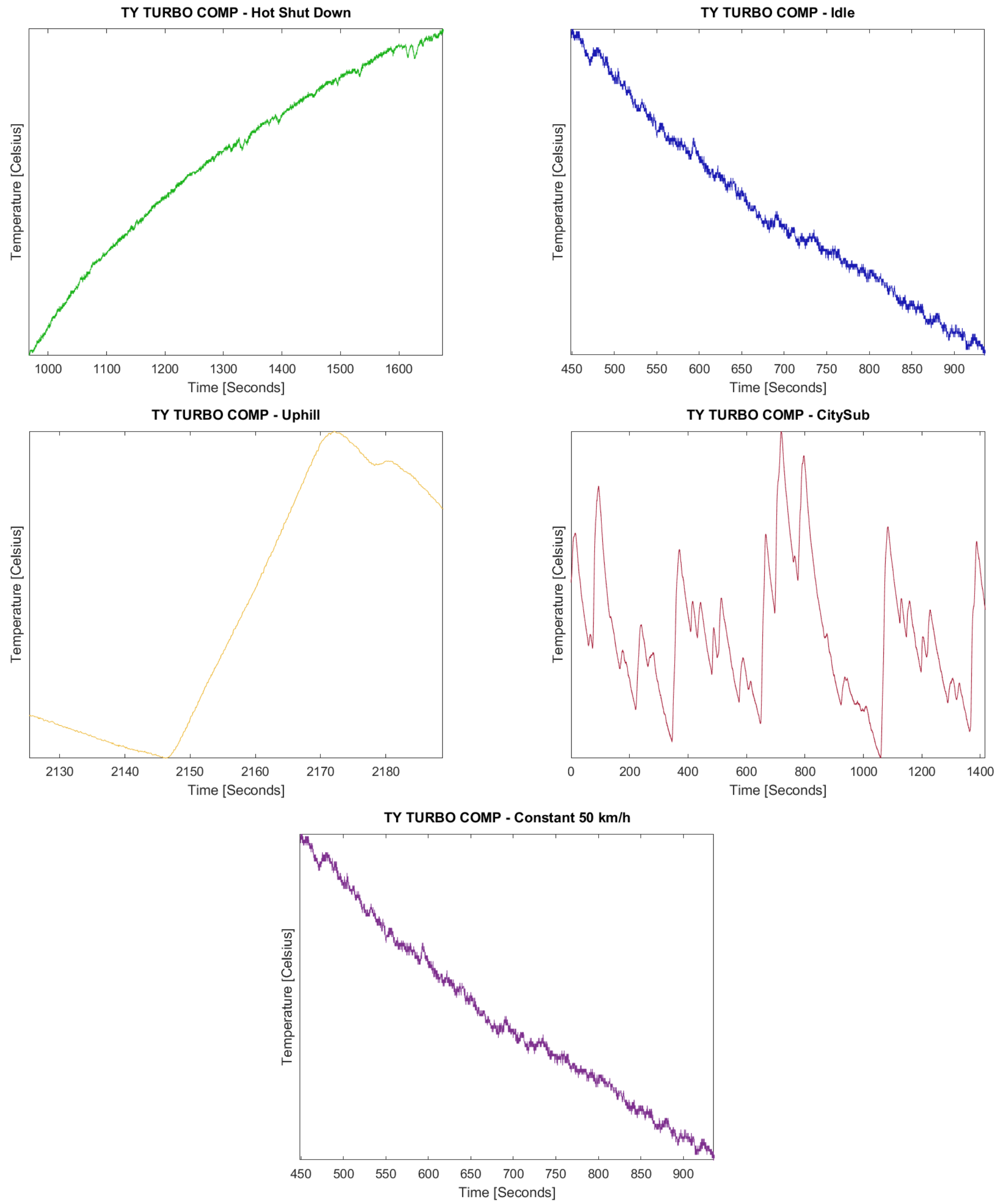

Figure 37: Surface temperatures for turbo-compressor in various driving conditions.

Wastegate

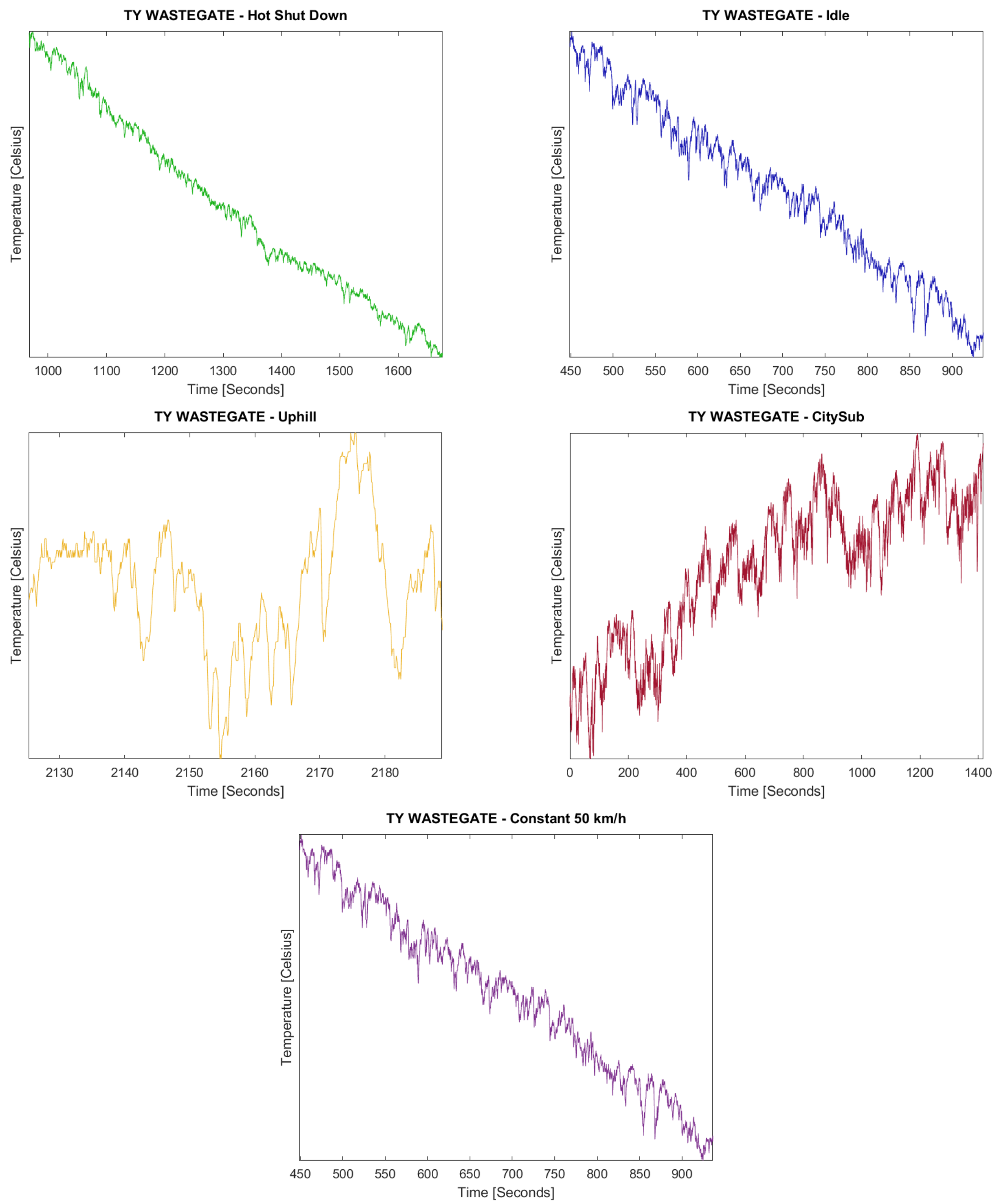

Figure 38: Surface temperatures for wastegate in various driving conditions.

Surrounding Temperatures

Ventilation mid rear

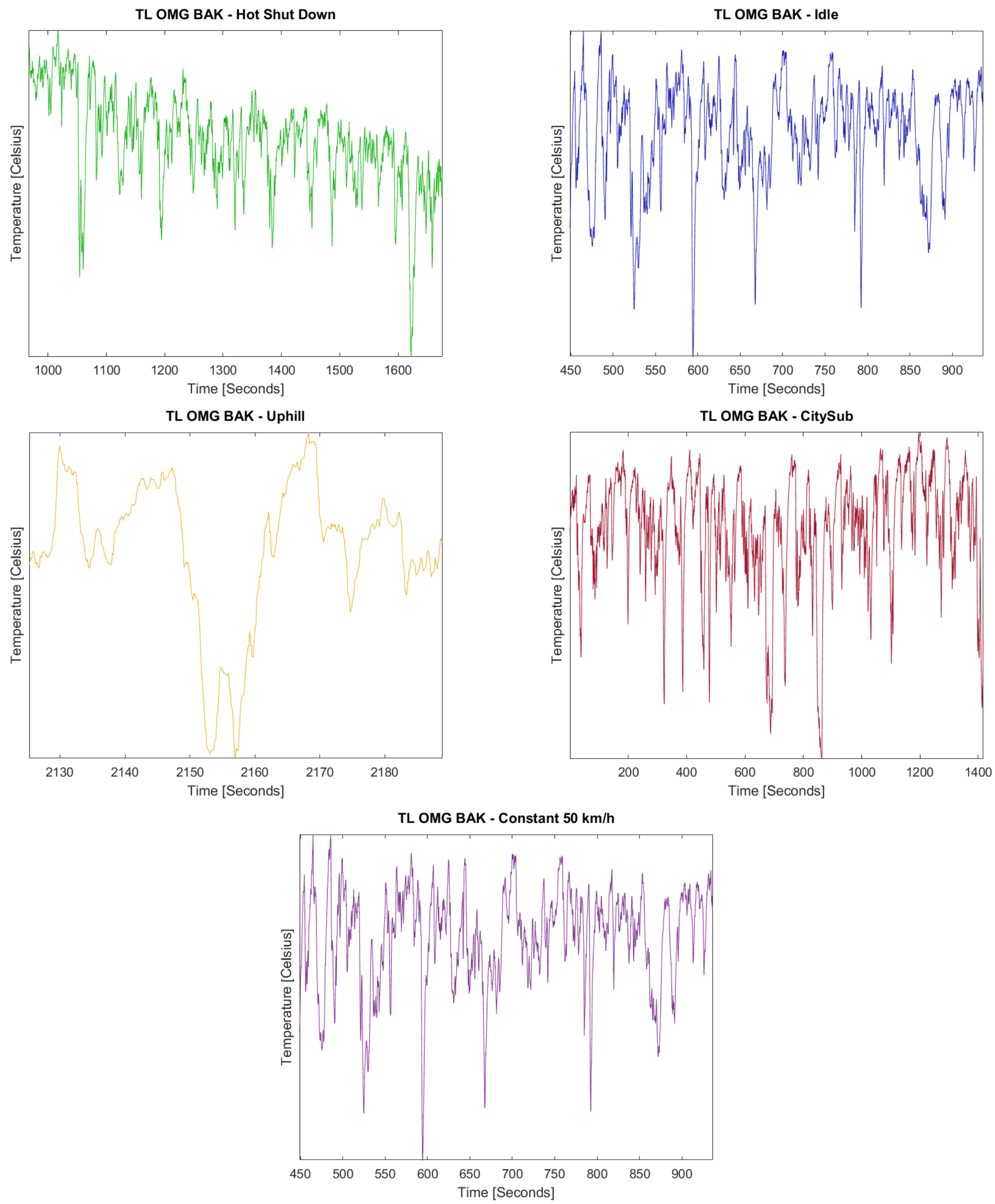

Figure 39: Surrounding temperatures for back-ventilation-opening in various driving conditions.

Cylinders

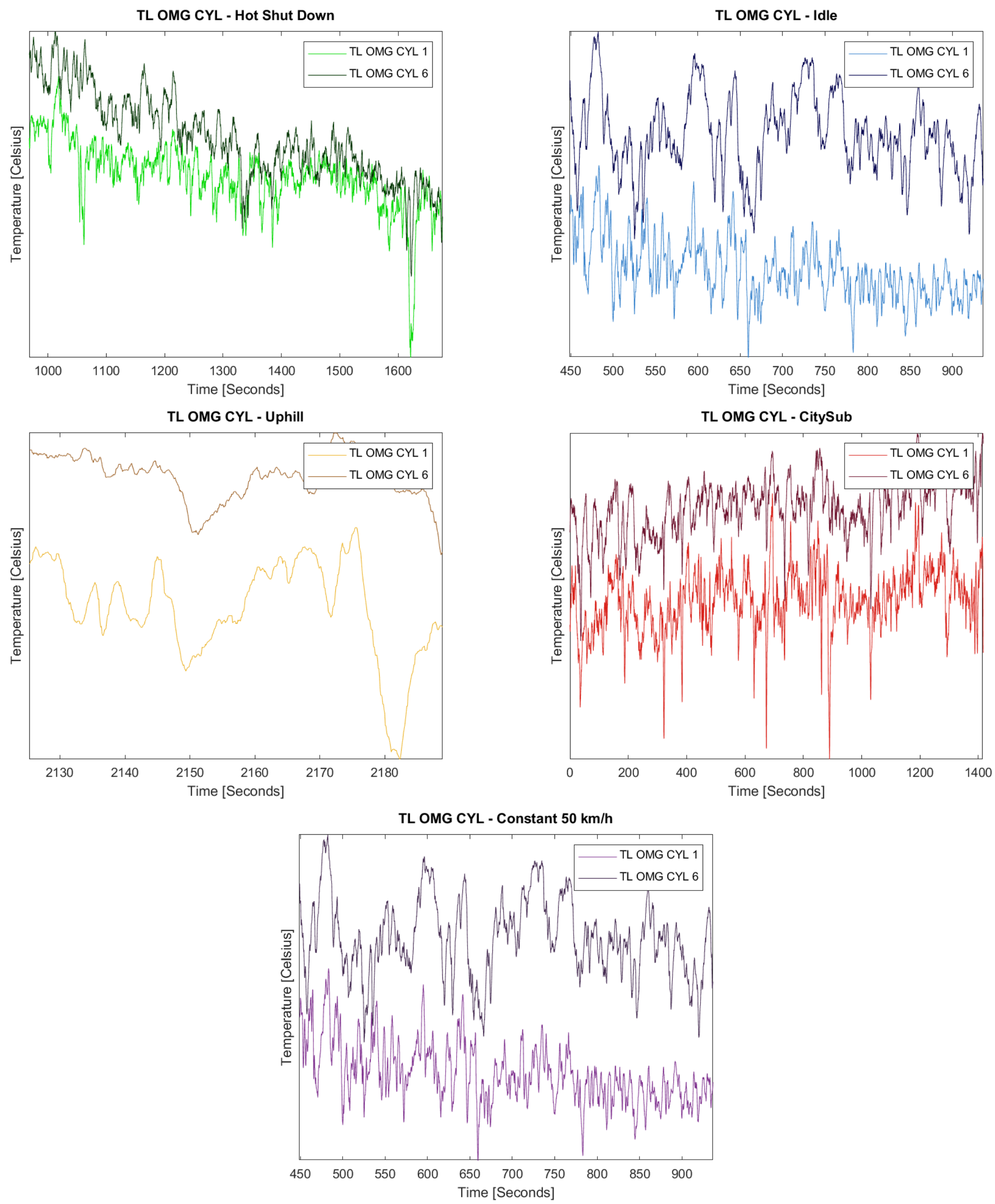

Figure 40: Surrounding temperatures for cylinders in various driving conditions.

Alternator Inlet

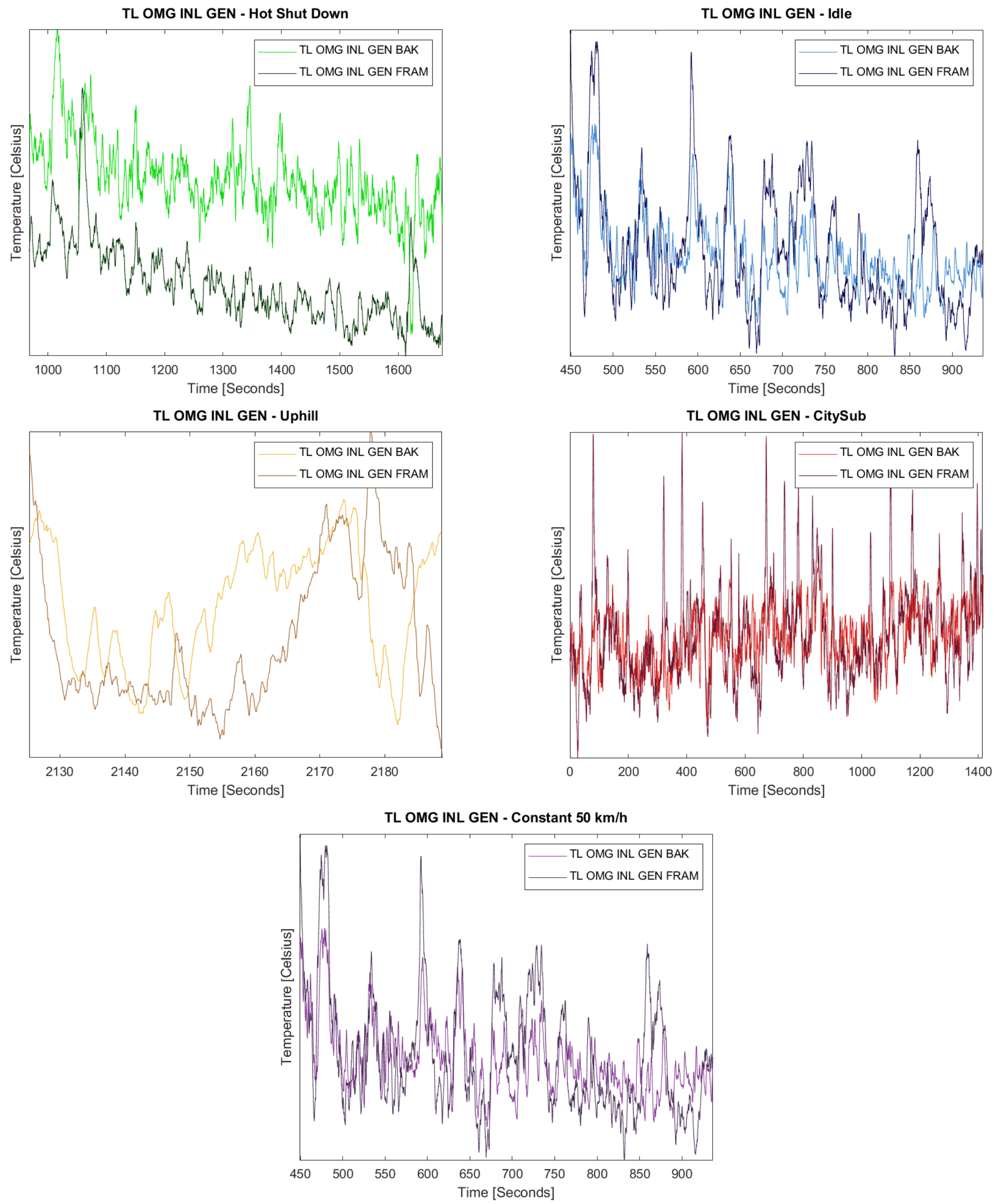

Figure 41: Surrounding temperatures for alternator-inlet in various driving conditions.

Fan

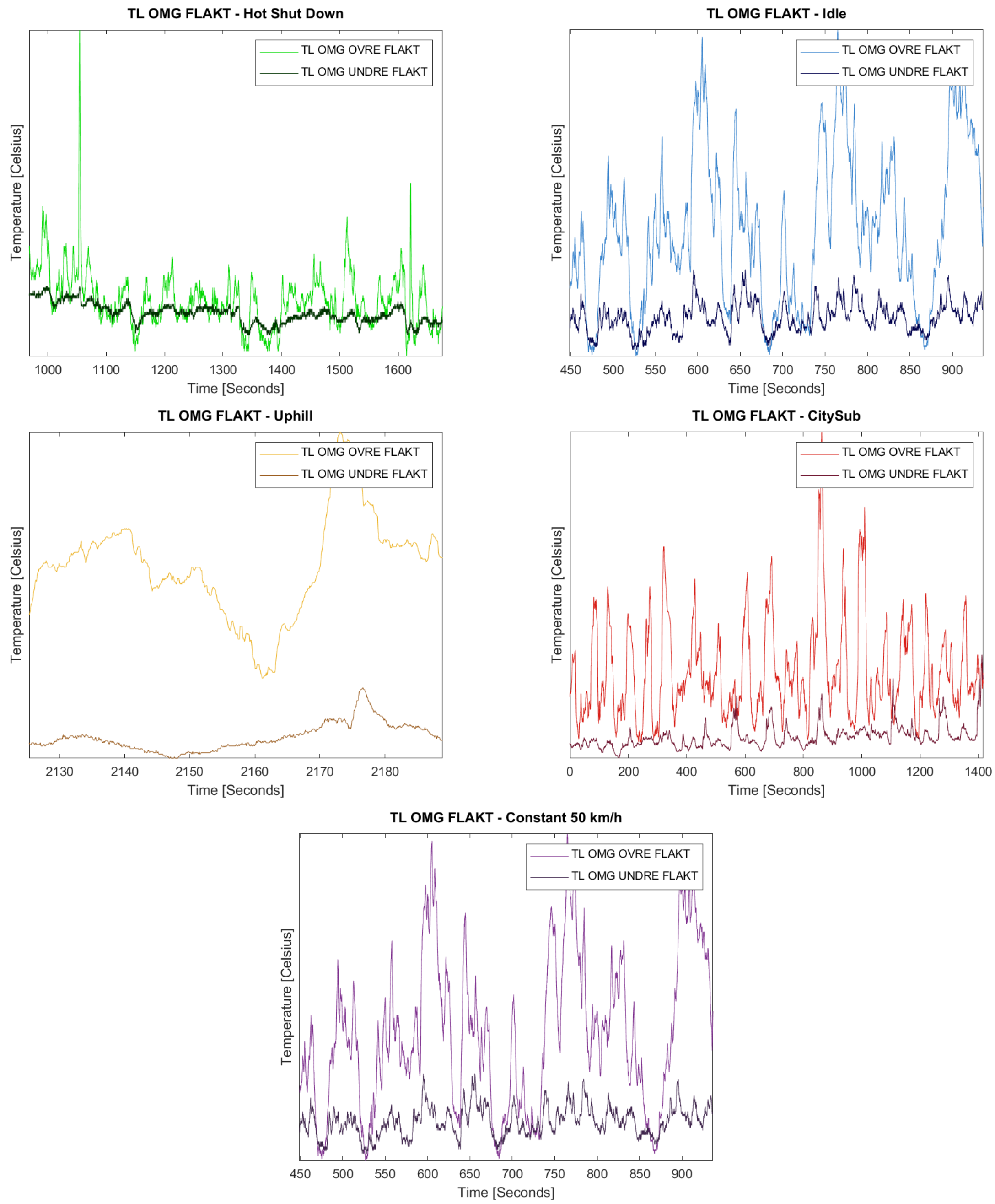

Figure 42: Surrounding temperatures for fan in various driving conditions.

Upper Compartment

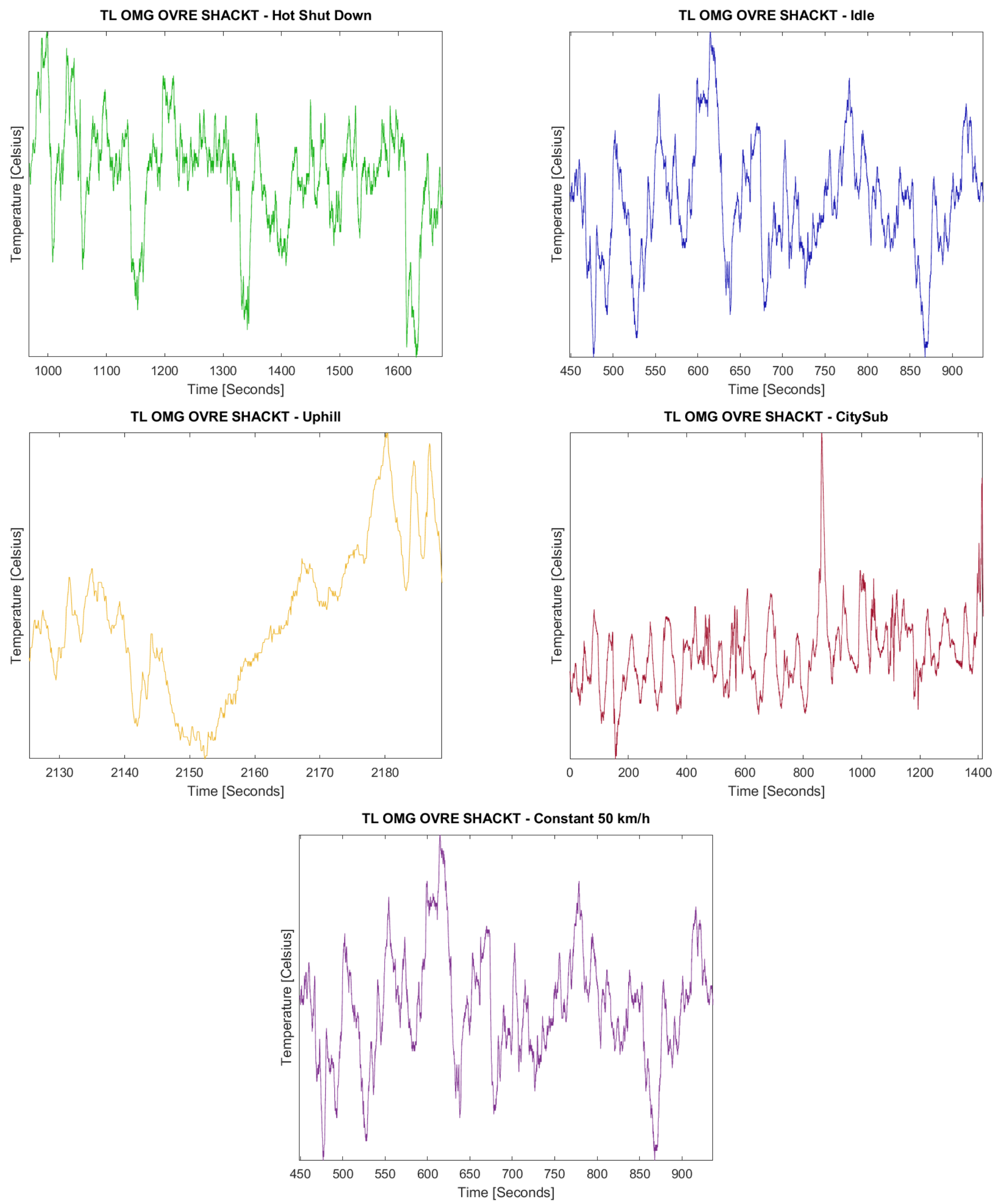

Figure 43: Surrounding temperatures for upper-compartment in various driving conditions.

Starter

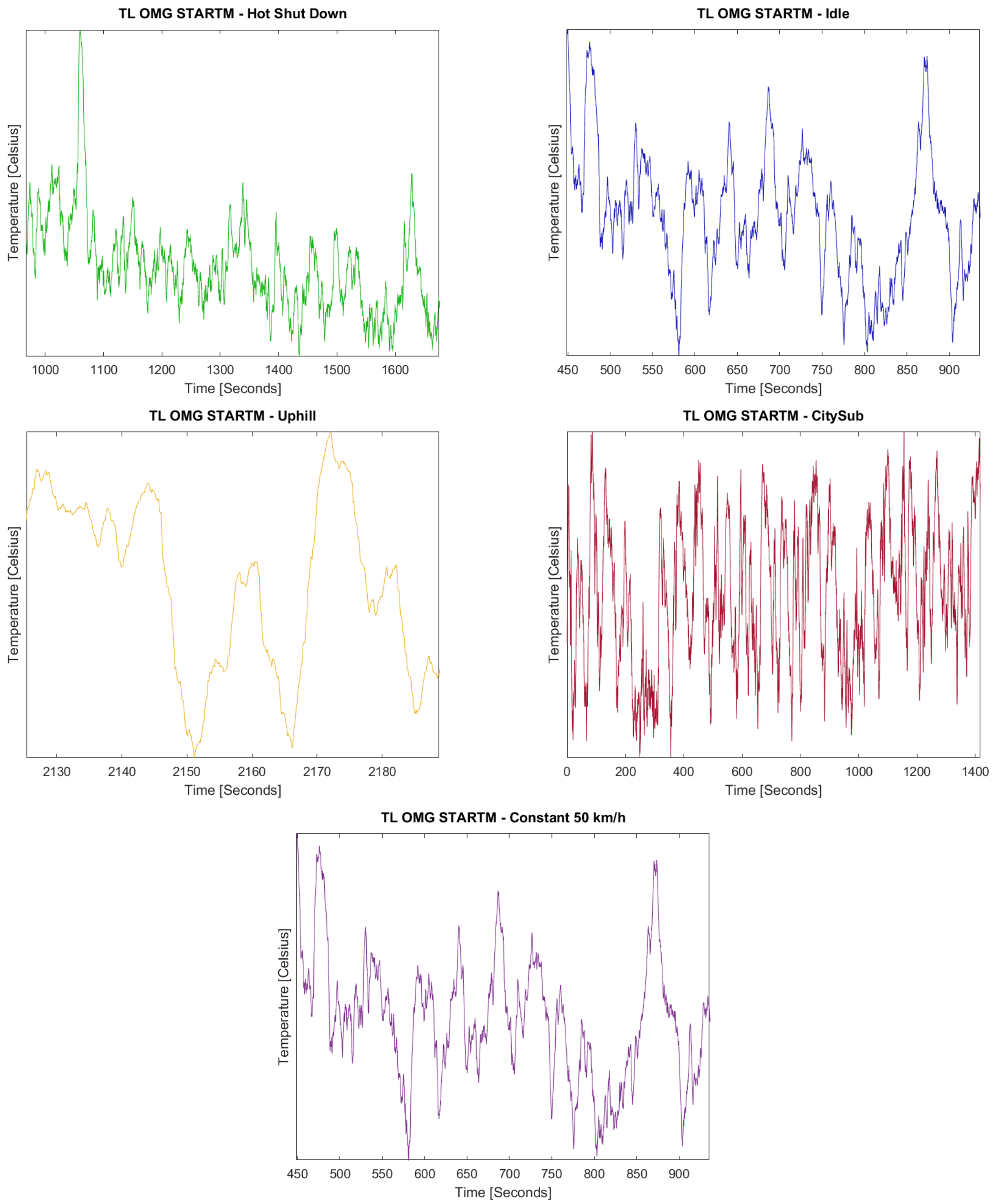

Figure 44: Surrounding temperatures for starter in various driving conditions.

Roof at Engine Block

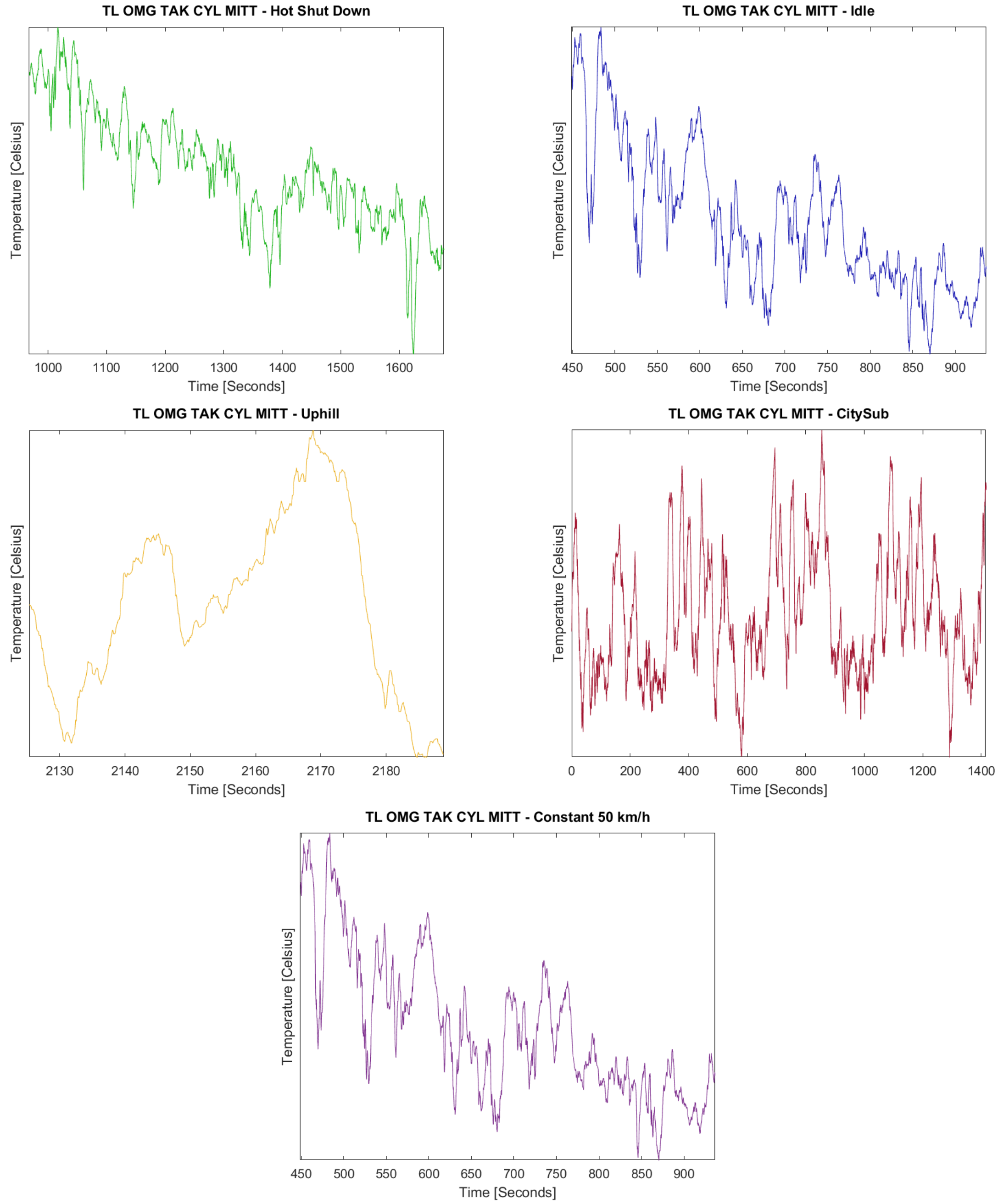

Figure 45: Surrounding temperatures for roof-engine-block in various driving conditions.

Roof at Turbo

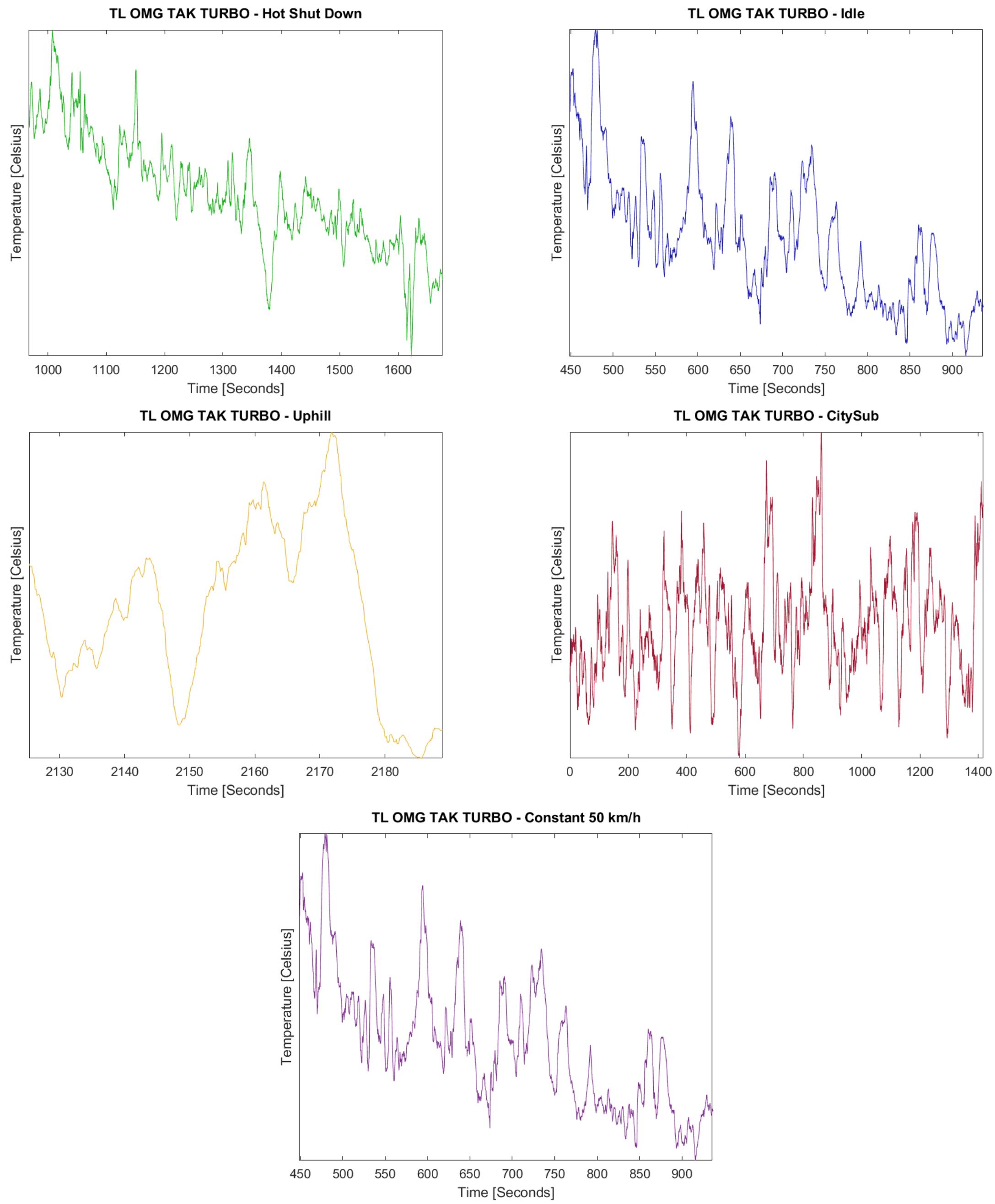

Figure 46: Surrounding temperatures for roof at turbo in various driving conditions.

Turbo

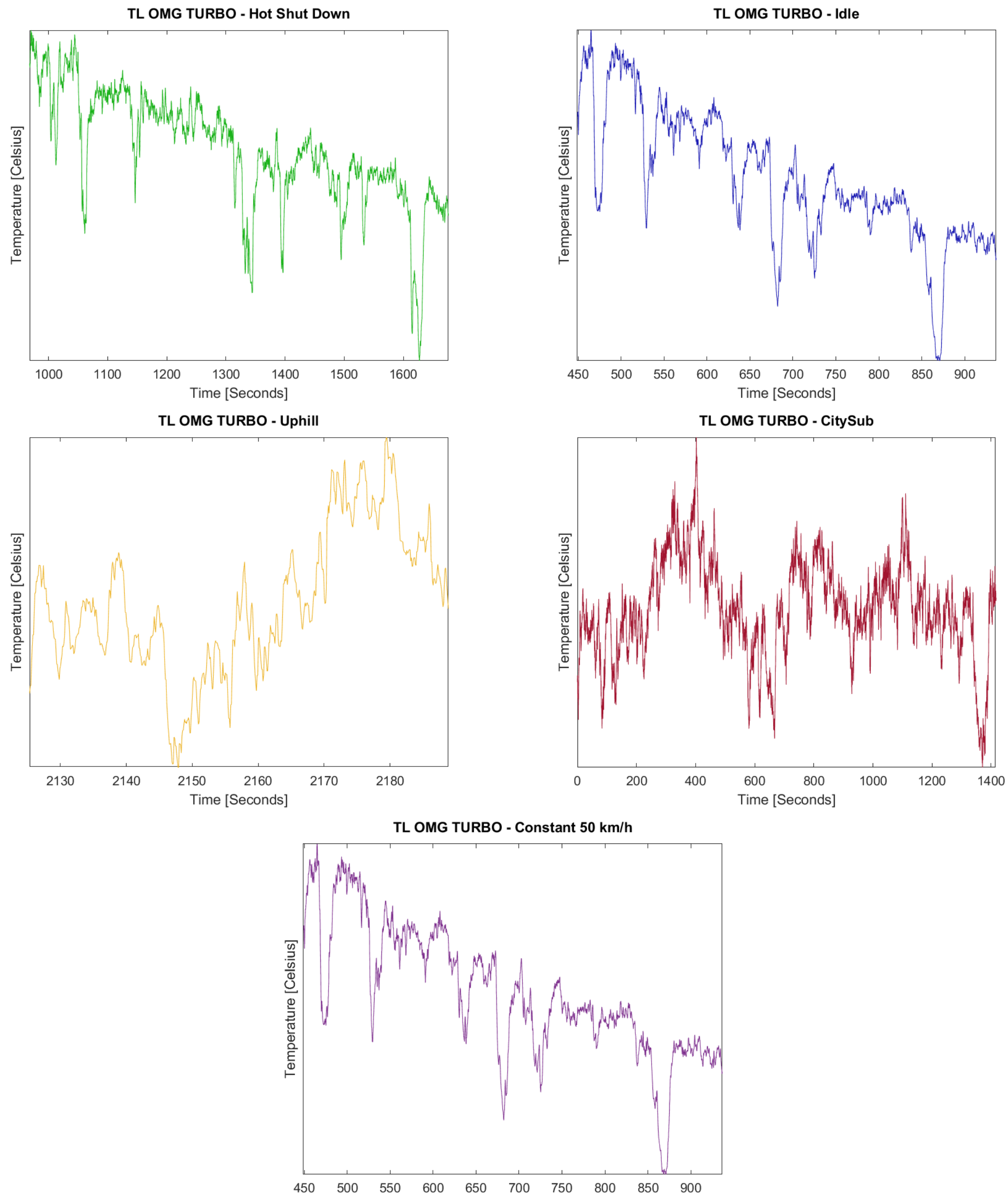

Figure 47: Surrounding temperatures for turbo in various driving conditions.

Alternator Outlet

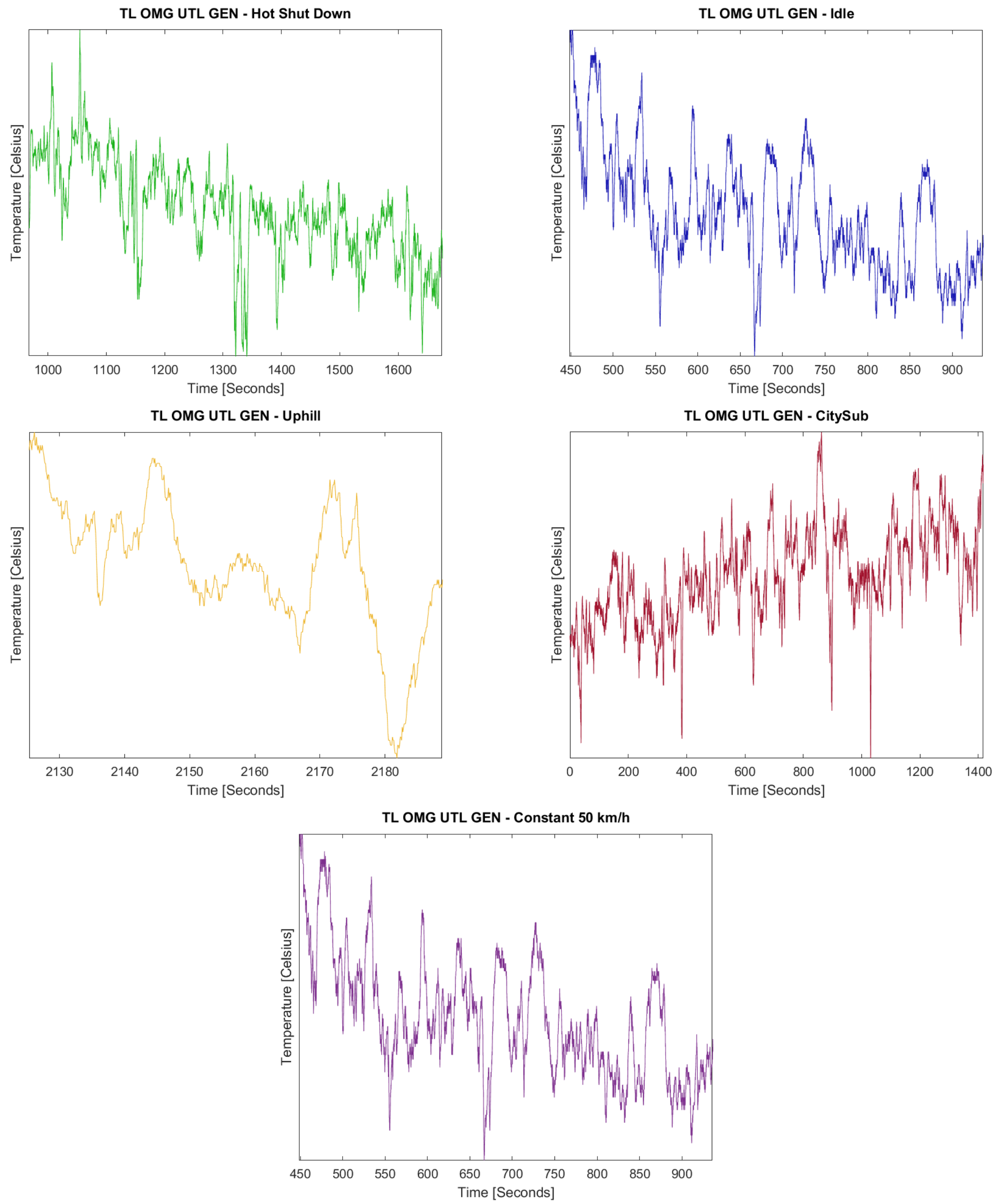

Figure 48: Surrounding temperatures for alternator-outlet in various driving conditions.

EMS

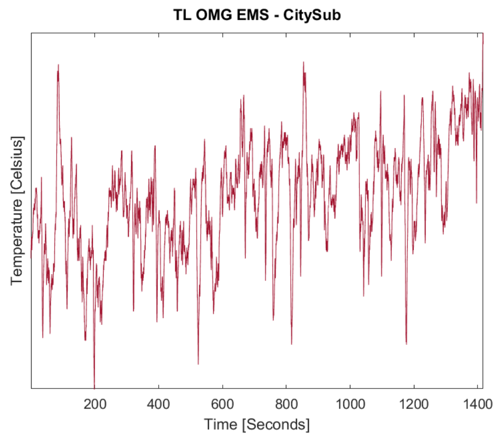

Figure 49: Surrounding temperatures for EMS in various driving conditions.

Silencer

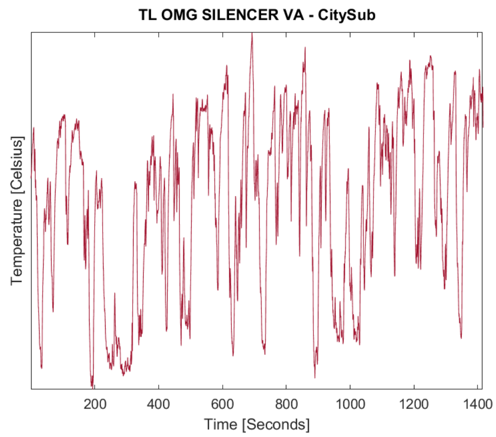

Figure 50: Surrounding temperatures for Silencer in various driving conditions.

Fluid Temperatures

TG11

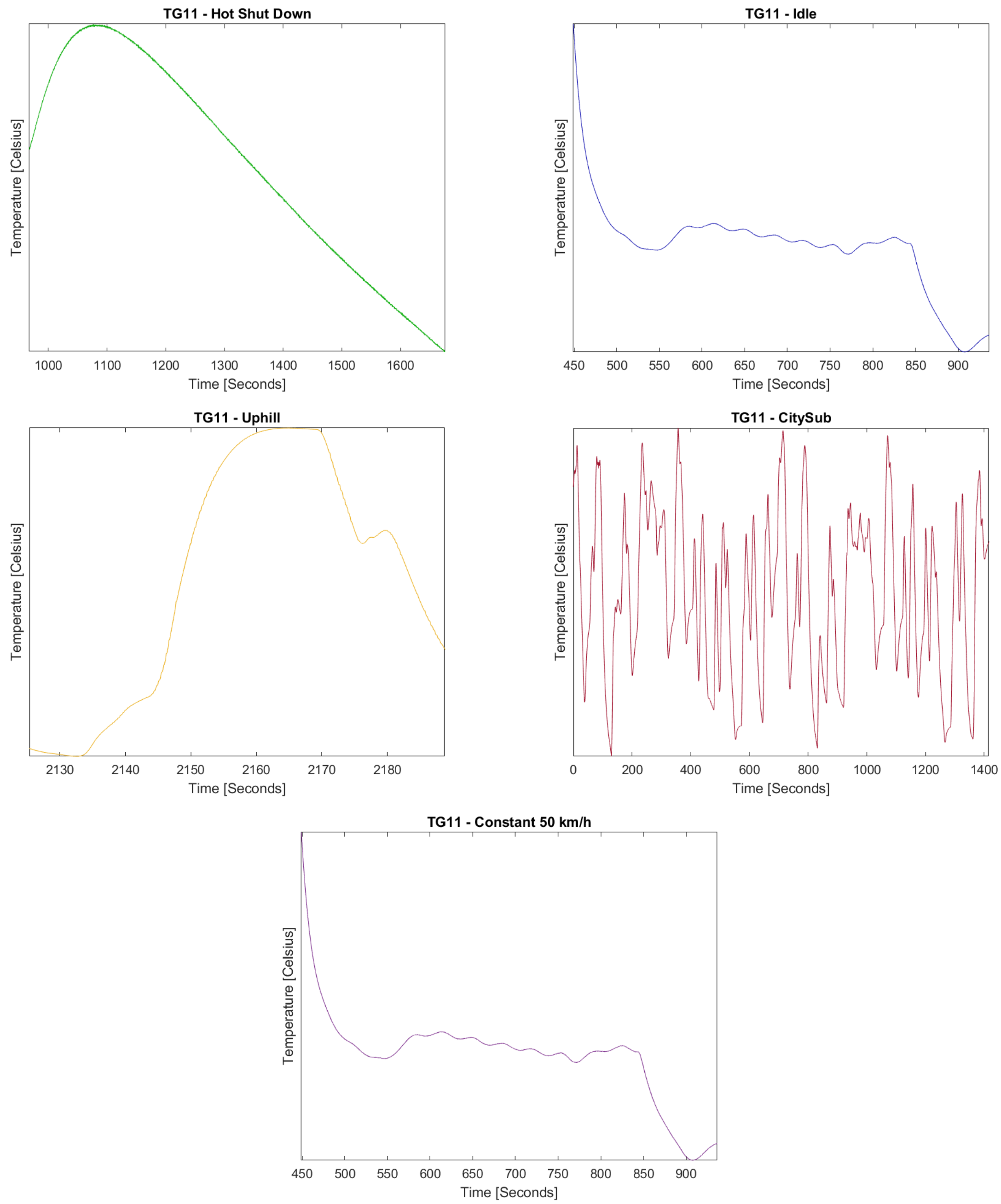

Figure 51: Fluid temperatures for TG11 in various driving conditions.

TG12

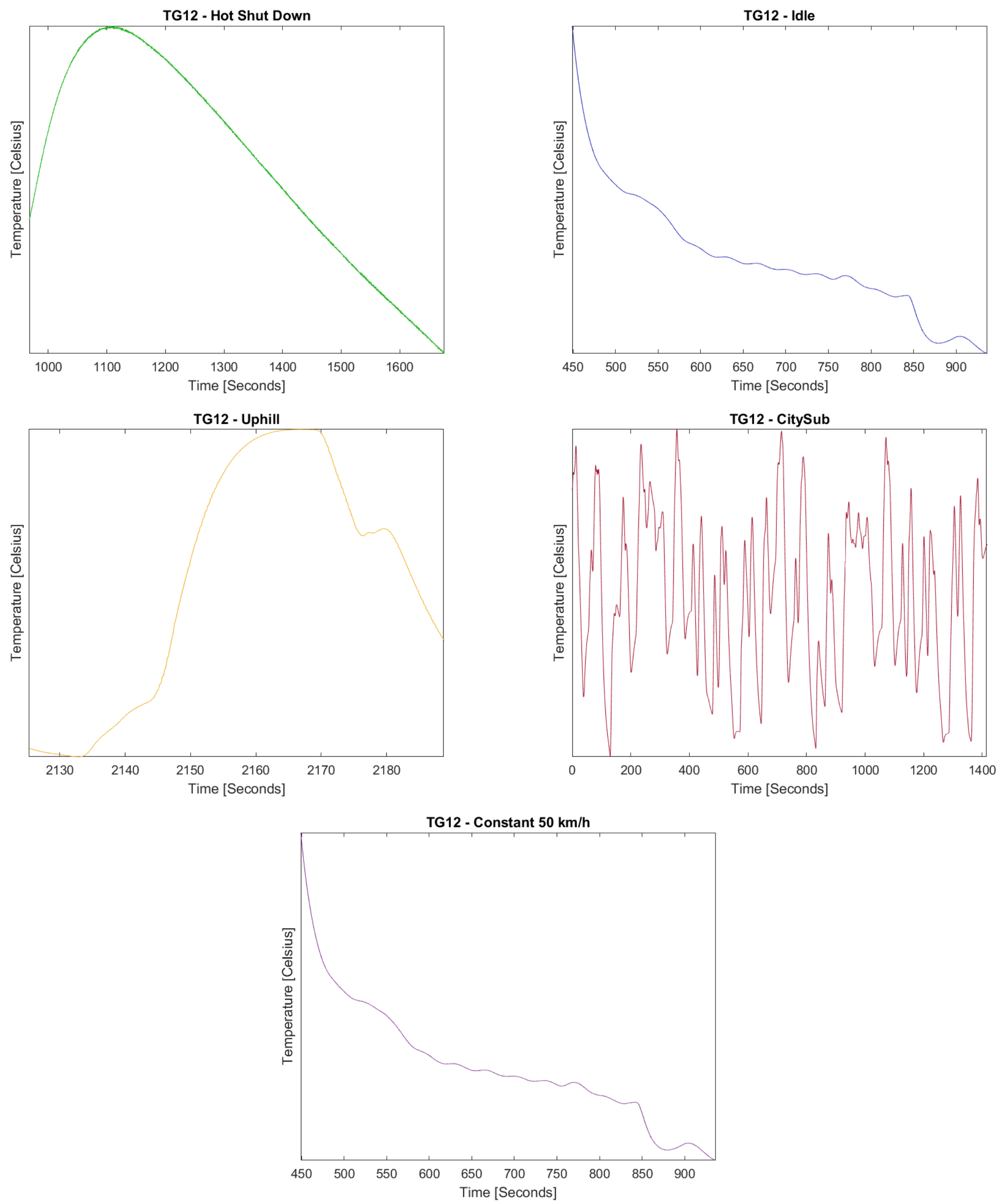

Figure 52: Fluid temperatures for TG12 in various driving conditions.

TG50

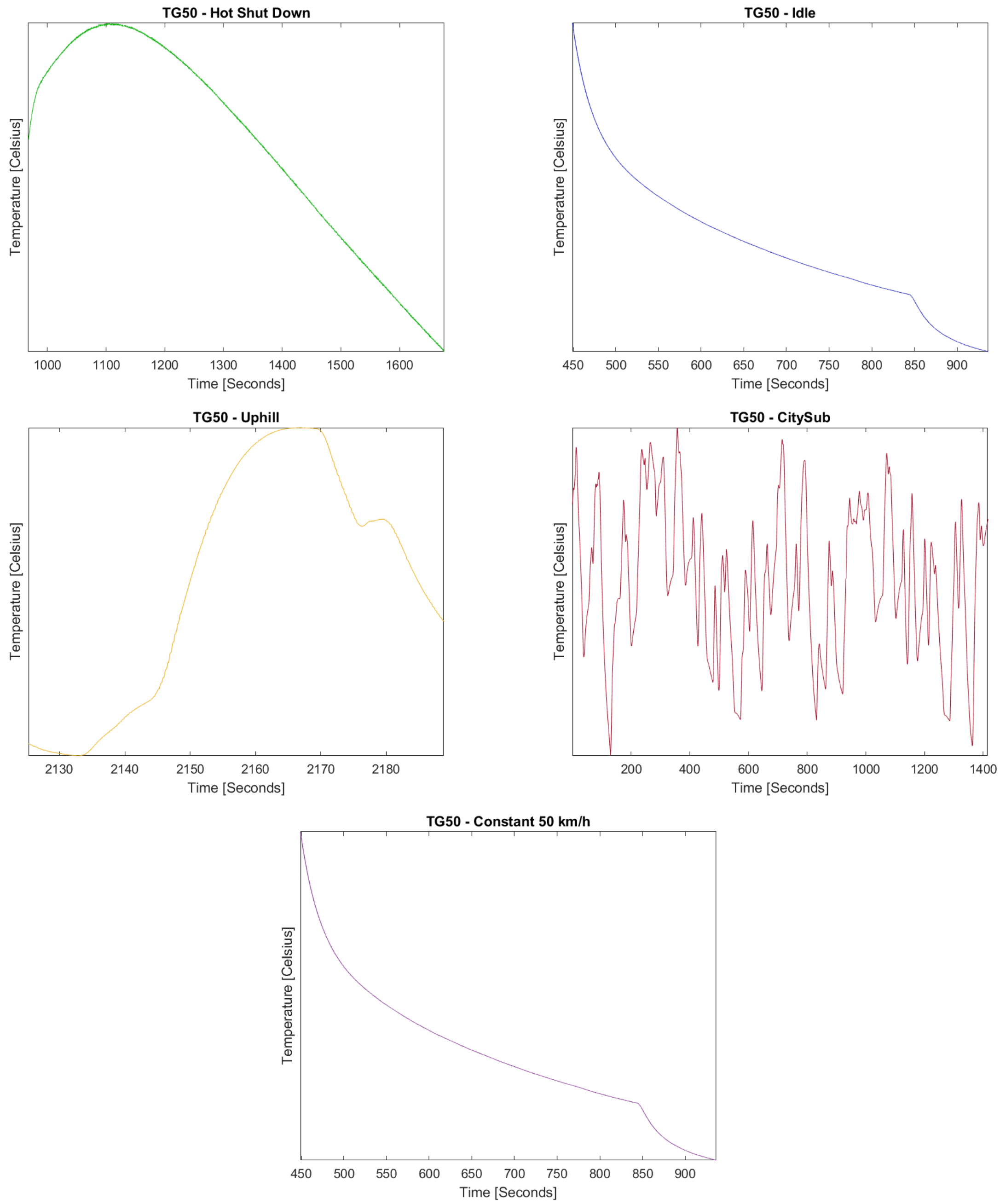

Figure 53: Fluid temperatures for TG50 in various driving conditions.

TL12

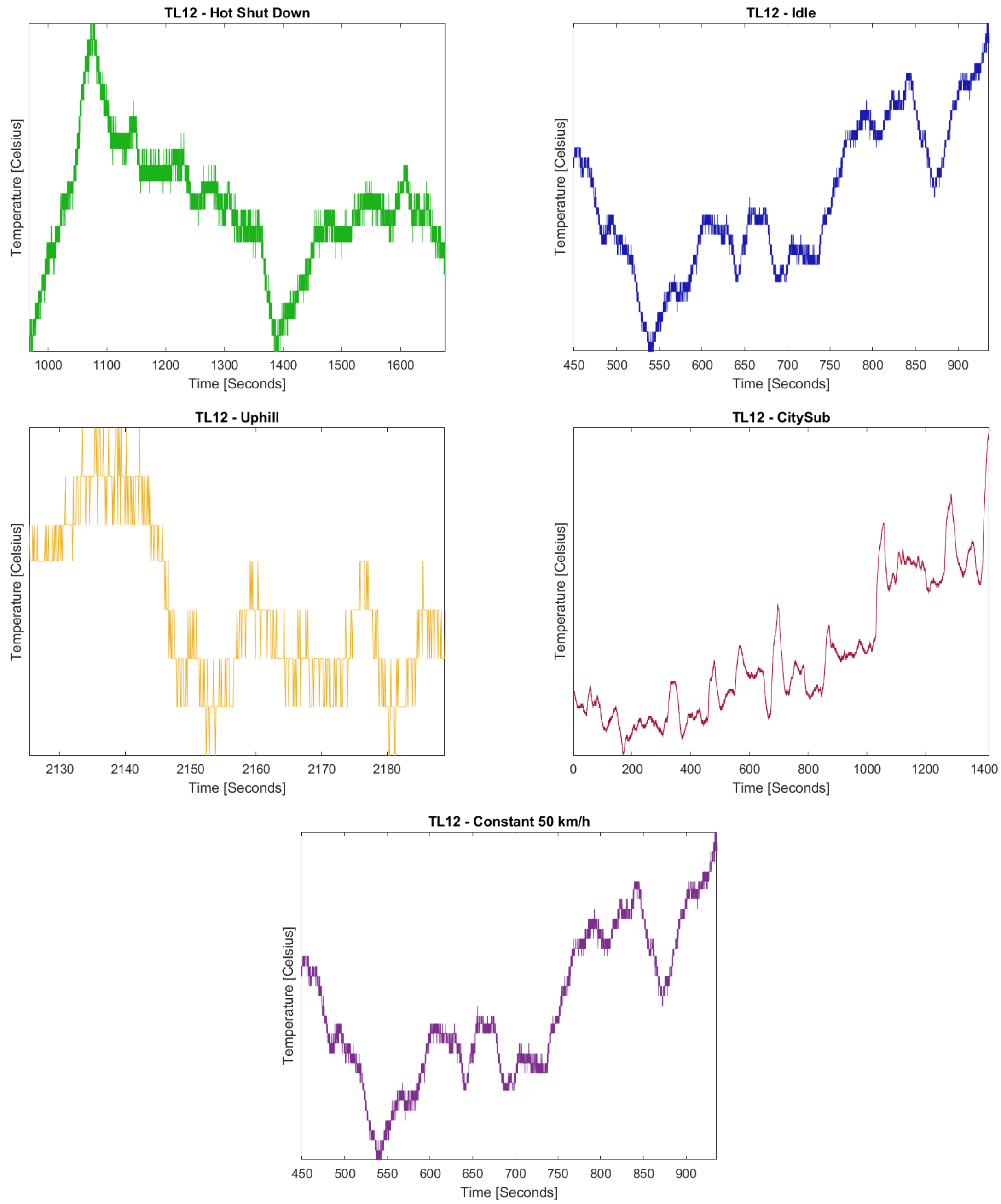

Figure 54: Fluid temperatures for TL12 in various driving conditions.

TL23

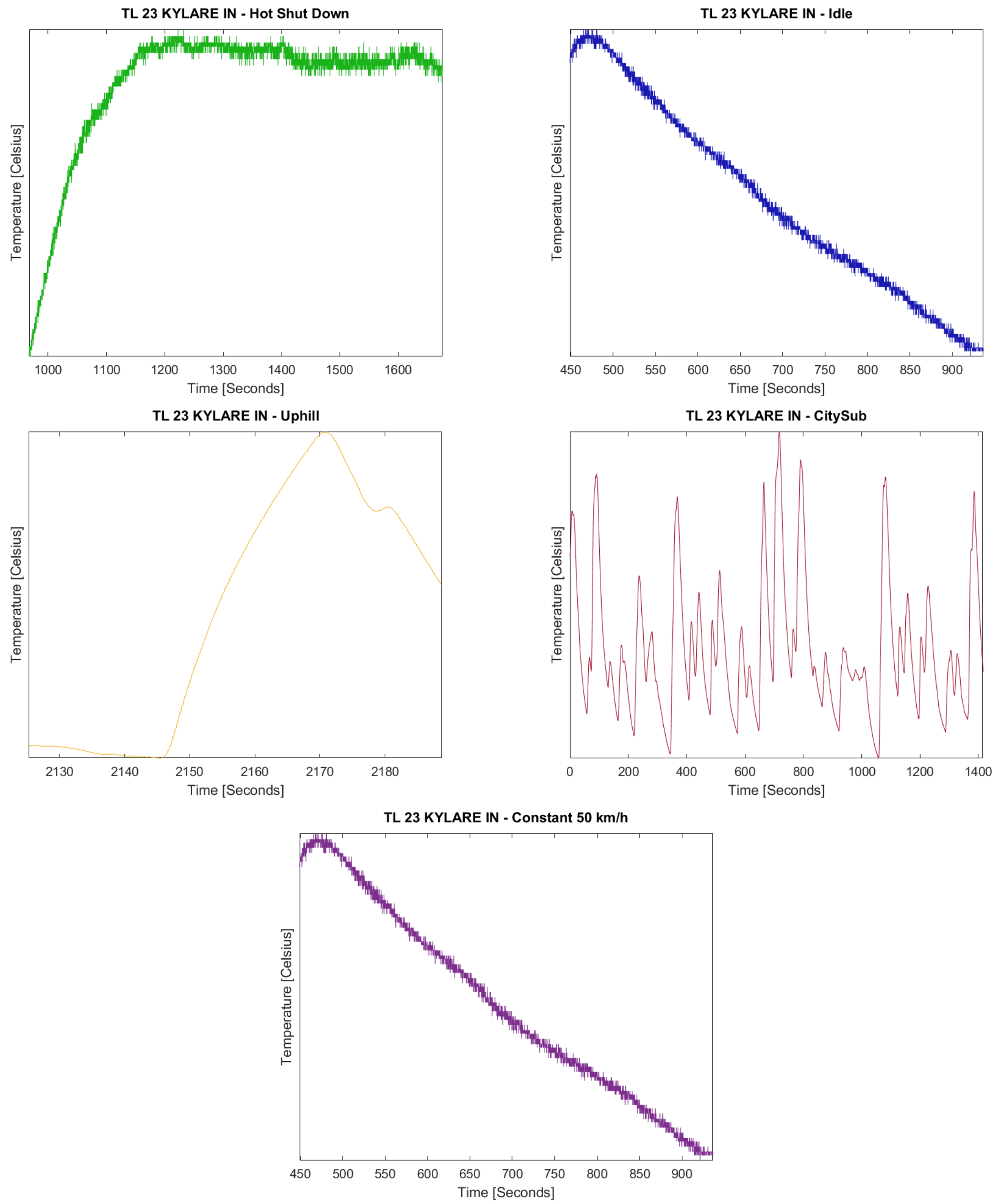

Figure 55: Fluid temperatures for TL23 in various driving conditions.

TL27

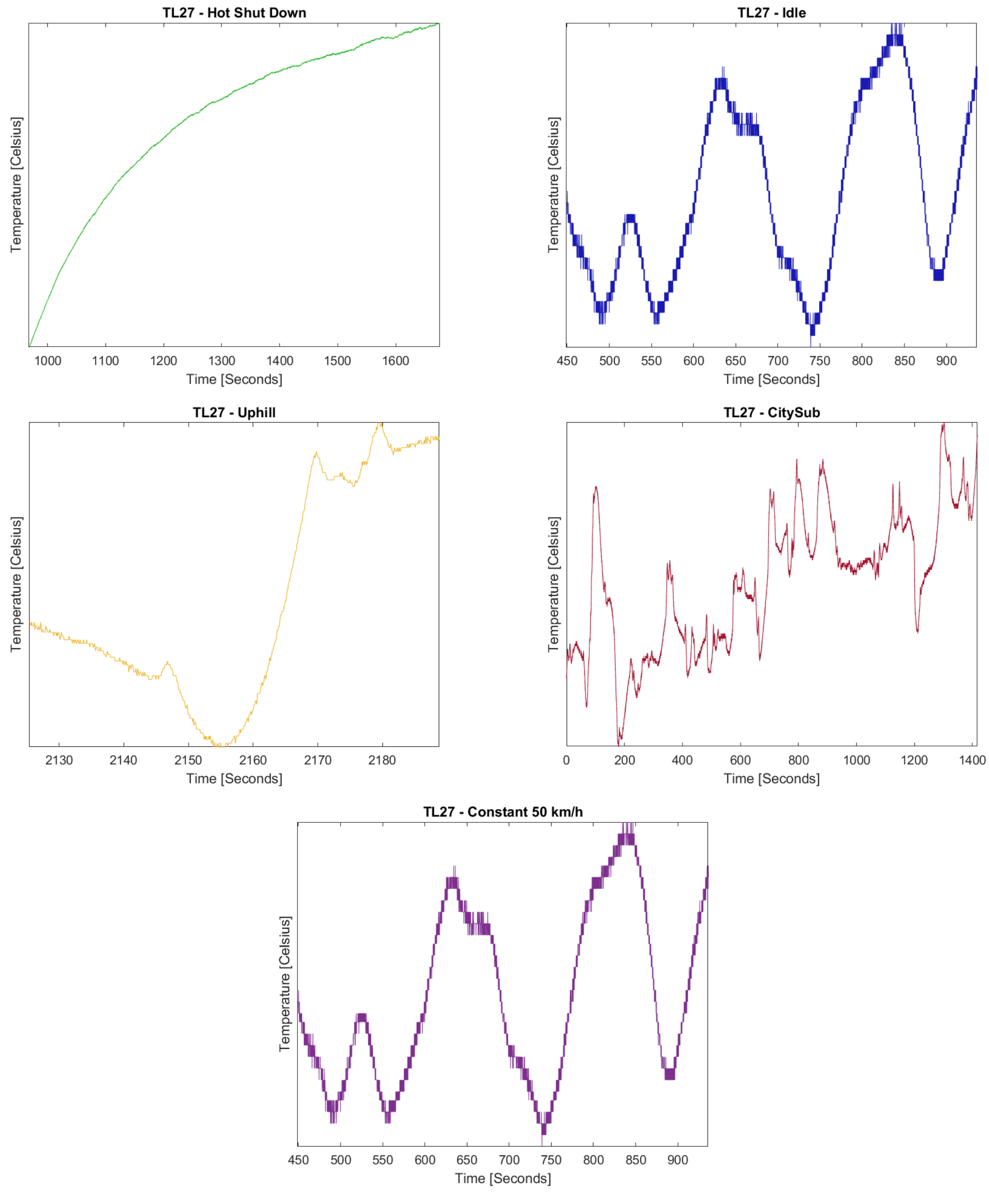

Figure 56: Fluid temperatures for TL27 in various driving conditions.

B Appendix: Complete table for all driving conditions

Table 8: Complete table of all the values from the CitySub Case with different scenarios

Case:	CitySub:	Start Uphill		CitySub:	Start Downhill		CitySub:	Start Flat	
Scenario:	Acceleration	Braking	Plateau	Acceleration	Braking	Plateau	Acceleration	Braking	Plateau
Driving Conditions:									
Engine Speed [Rpm]	1049.79	704.76	912.46	1108.25	796.19	1103.62	1091.85	710.83	922.24
Velocity [Km/h]	30.54	14.32	34.06	31.89	18.18	39.93	26.30	13.86	33.24
CAC Fan Speed [Rpm]	2197	1839	2264	2227	1921	2326	2128	1857	2336
RAD Fan Speed [Rpm]	1353	1403	1904	673	1045	1375	1120	1264	978
Applied Load [%]	69.51	20.91	39.52	45.91	23.15	39.02	49.25	15.48	31.84
Exhaust Gas Massflow [Kg/s]	7.09	2.63	5.21	5.88	3.03	5.11	5.54	2.51	3.93
Ambient Temperature [°C]	9.22	9.22	9.22	9.22	9.22	9.22	9.22	9.22	9.22
Boundary Condition [°C]									
TY_TURBO									
TY_INL_GEN									
TY_AVG_ALU_1									
TY_AVG_ALU_2									
TY_AVG_ISO									
TY_AVG-SKARV-1									
TY_AVG_SILENCER									
Engine Block (TY_CYL_1, TY_CYL_3, TY_CYL_6)									
TY_SILENCER_FRONT									
TY_SILENCER_ABOVE									
TY_SILENCER_VA									
TY_SILENCER-HO									
TY_WASTEGATE									
TY_LUFT_COMP									
Surrounding Comparison Points[°C]									
TL_OMG_INL_GEN_BAK									
TL_OMG_INL_GEN_FRAM									
TL_OMG_UTL_GEN									
TL_OMG_CYL_1									
TL_OMG_CYL_6									
TL_OMG_TAK_CYL_MITT									
TL_OMG_TAK_TURBO									
TL_OMG_TURBO									
TL_OMG_OVRE_FLAKT									
TL_OMG_UNDER_FLAKT									
TL_OMG_OVRE_SHACKT									
TL_OMG_STARTM									
TL_OMG_BAK									
TL_OMG_SILENCER_VA									
TL_OMG_EMS									
Surface Comparison Points[°C]									
TY_LUFT_COMP_SLANG									
TY_CHARGE_HOSE									
TY_FILTER_BURK									
TY_ROR_INL_2									
TY_ROR-INL-UT									
TY_OVRE_SHACKT									
Fluid Temperature Points [°C]									
TG11									
TG12									
TG50									
TL12									
TL23									
TL27									

Table 9: Complete table of all the values from the Idle, HSD, Uphill and Constant 50 driving conditions

Case:	Idle	Hot Shut Down	Uphill	Constant 50
Metadata:				
Engine Speed [Rpm]	599.80	0.00	1148.68	1064.28
Velocity [Km/h]	0.00	0.00	18.11	51.76
CAC Fan Speed[Rpm]	1500	2000	1931	2130
RAD Fan Speed[Rpm]	150	150	2065	882
Applied Load [%]	15.92	0.00	50.86	29.01
Exhaust Gas Massflow [Kg/s]	1.91	0.00	7.08	4.63
Ambient Temperature [°C]	14.02	13.81	7.68	8.05
Boundary Condition [°C]				
TY_TURBO				
TY_INL_GEN				
TY_AVG_ALU_1				
TY_AVG_ALU_2				
TY_AVG_ISO				
TY_AVG_SKARV_1				
TY_AVG_SILENCER				
TY_AVG_SLUT				
Engine Block (TY_CYL_1, TY_CYL_3, TY_CYL_3)				
TY_SILENCER_FRONT				
TY_SILENCER_ABOVE				
TY_SILENCER_VA				
TY_SILENCER_HO				
TY_WASTEGATE				
TY_LUFT_COMP				
Surrounding Comparison Points[°C]				
TL_OMG_INL_GEN_BAK				
TL_OMG_INL_GEN_FRAM				
TL_OMG_UTL_GEN				
TL_OMG_CYL_1				
TL_OMG_CYL_6				
TL_OMG_TAK_CYL_MITT				
TL_OMG_TAK_TURBO				
TL_OMG_TURBO				
TL_OMG_OVRE_FLAKT				
TL_OMG_UNDER_FLAKT				
TL_OMG_OVRE_SHACKT				
TL_OMG_STARTM				
TL_OMG_BAK				
Surface Comparison Points[°C]				
TY_LUFT_COMP_SLANG				
TY_CHARGE_HOSE				
TY_FILTER_BURK				
TY_ROR_INL_2				
TY_ROR_INL_UT				
TY_OVRE_SHACKT				
Fluid Temperature Points [°C]				
TG11				
TG12				
TG50				
TL12				
TL23				
TL27				