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Sperimentazione e Calibrazione di Motori a Combustione Interna

TORQUE MODEL CALIBRATION OF A MOTORCYCLE INTERNAL COMBUSTION ENGINE

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Sessione I

A mia nonna Giovanna

che avvera il sogno di avere

una nipote “ingegnere finito”

INDEX

INDEX OF FIGURES

Definition and abbreviations

AFR – Air-Fuel Ratio
AIS – Air Injection System
AuSy – Automation System
BMEP – Brake Mean Effective Pressure
CA – Crank Angle
CAN – Controller Area Network
CCP – CAN Calibration Protocol
DBW – Drive By Wire
ECU – Engine Control Unit
EGO – Exhaust Gas Oxygen
EOI – End Of Injection
ETB – Electronic Throttle Body
ETK – Emulator Tast Kopf
HEGO – Heated Exhaust Gas Oxygen
IMEP – Indicated Mean Effective Pressure over the entire cycle, 720°CA
IMEP COV – Indicated Mean Effective Pressure Covariance
IMEPH – Indicated Mean Effective Pressure High pressure part 360°CA
KP FREQ – Knock Pressure Frequency
MAF – Mass Air Flow
MAP – Manifold Absolute Pressure
MBT – Maximum Brake Torque
MFB50 – Mass Burned Fraction 50%
OBI – On Board Indicating
PLC – Programmable Logic Controllers
SA – Spark Advance
SA EFF – Spark Advance Efficiency
SA OPT – Spark Advance Optimum
SA REF – Spark Advance Reference
SA REF – Spark Advance Reference
TIA – Temperature Intake Air
TPS – Throttle Position Sensor
TQFR – Torque Friction
UEGO – Universal Exhaust Gas Oxygen

Introduction

Optimizing the performance of internal combustion engines increases complexity of control strategies and makes the calibration process long and expensive.

As in most companies, also the DUCATI Engine Tests Room Dept. has, among its various tasks, to perform all tests needed to build up the base engine calibration, in cooperation with the ECU supplier.

DUCATI also has the task of controlling the final work of the supplier, and the aim of this Thesis is properly creating a simplified Torque-based model on the data collected during the calibration campaign and on the strategies already known.

In the first chapter is given an overview on systems for checking the bench, on systems for the control and calibration of the ECU and on the communication protocols between the cell SWs; the second chapter describes the calibration activities performed and the analysis of the data produced: the data were processed with scripts and functions written in Matlab language (using and sometimes modifying already existing DUCATI calibration tools, others were created *ex novo*) and elaborated with software like Excel. In the third chapter is explained the possibility to reduce tests on the bench extrapolating data of spark advance min and extramin from the spark advance dynamic sweep: it has been done a comparison between modelled and measured values. In the fourth and last chapter is analysed the torque structure, is shown the simplified Torque-based model created and are explained in details the “cases of study” and the various simulations realized.

1. The engine test room architecture and calibration instruments

Each engine before production must be properly developed and deliberated, by means of calibration, power curves and durability tests: if it passes the tests successfully it is approved. To do this, DUCATI has several engine test benches managed by PLC systems (Programmable Logic Controllers), in order to allow the determination of the mechanical characteristics of the engines: power, torque and consumption.

The bench is not only used for engine characterization, but also for the development of it and for the calibration setup.

The engine test room is usually divided into two rooms kept separated for safety reasons: the engine room, where the engine under test is allocated with the relative transducers, auxiliary and necessary systems, and the control room where PCs for the tests management are allocated. The advantage of this type of experimentation is the possibility to carry out measurements in a controlled manner, keeping the ambient conditions unchanged and above all using sensors with a much higher accuracy than those used on board.

Fundamentally there are three systems that are necessary to carry out the calibration campaign of an engine:

- systems for the calibration of the ECU (Engine Control Unit);
- systems for checking the bench;
- systems for the analysis of combustion inside the cylinders.

In particular for DUCATI test rooms the software/hardware used are:

- INCA® and ETAS® modules for ECU interfacing and management of UEGO sensors;
- AVL PUMA® for the bench control;
- AVL INDICOM® for combustion analysis.

1.1 The dynamic brake

The test room in which the engine I worked on in these months was a "dynamic cell", so defined for the presence of the dynamic brake inside it.

The dynamic brake is an electric device that acts on the engine shaft and the fundamental difference respect to passive brakes is that this machine can act both as a brake and as a motor .

Figure 1 - Dynamic Brake

The brake is characterized by the main components:

- Rotor: rigidly couples to the drive shaft through a joint;
- Stator: equipped with a load cell to measure the torque applied on the shaft;
- Load regulator: component that regulates the amount of braking torque exerted by the brake;
- Angular speed sensor: instrument necessary to measure the brake speed first and then power of the engine under test.

The braking action allows to bring and keep the engine to a precise operating point. The action as a motor allows the engine to be dragged to make a run-in cycle or for the direct measurement of the frictions of auxiliaries, or to simulate real road gradient and vehicle inertia in the developing tests.

1.2 External sensors

1.2.1 Optical Encoder

Figure 2 - Optical Encoder

The Encoder is an electromechanical device that converts the angular position of its rotating axis into short electrical pulses that are processed by an analysis circuit in the form of digital numeric signals. Connected to the drive shaft, it provides information on the angular position and on the rotation speed of the engine.

We can distinguish two parts:

- the body, the fixed part, with inside the electric components (sensors, circuits, etc.);
- the rotor, the rotating part, which normally ends with a shaft to be connected to the axis to be read.

The electrical output signals transmit the information related to the position or the displacement of the rotor with respect to the body.

In an optical encoder there is a led source that transmits the light to a disk that has slots: this allow the passage of light and prevents it if there is no opening. The presence or absence of light generates a corresponding electrical signal.

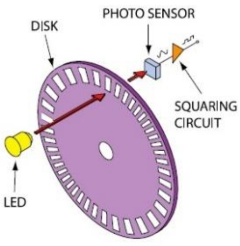


Figure 3 - Functioning of the optical encoder

1.2.2 Thermocouples

Figure 4 - Thermocouple

Thermocouples are thermistors mounted on exhaust manifolds used to measure those high temperatures. They are widely used because they are cheap, easily replaceable and standardized.

The operating principle of a thermocouple is that, in a circuit formed by two conductors of a different nature subjected to a temperature gradient, it establishes a difference of potential. This phenomenon is called *Seebeck effect*. The phenomenon cannot exist in a circuit formed by a single homogeneous conductor and for this reason a thermocouple consists of a pair of electrical conductors of different materials joined together at one point. This junction is conventionally called "hot joint" and is the point at which the temperature to be measured is applied. The other end, constituted by the free ends of the two conductors, is conventionally called "cold joint". When there is a temperature difference between the hot joint area and the cold joint area, an electrical difference of potential can be detected.

There is a great variety of thermocouples, distinguishable according to the two electrical conductors that make up the junction and to the field of application (industrial, scientific, food, medical, etc.). The thermocouples mounted on the engine analyzed are those of type K: (Chromel (Ni-Cr) (+) / Alumel (Ni-Al) (-)). They are thermocouples of general use, economical and available in a large variety of formats. Their measuring range is from -200 ° C to 1260 ° C. The sensitivity is about 41 μV / ° C.

1.2.3 Oil temperature sensor

The oil temperature sensor is a resistance thermometer sensor that uses the variation of the resistivity of some materials when they are subject to temperature changes. There are several types of heat resistance, generally quite resistant to corrosive agents, which can measure temperatures in a good temperature range (even if lower than that of thermocouples) and that above all have excellent linearity.

For metals there is a linear relationship that links resistivity and temperature:

$$\rho(T) = \rho_0 \cdot [1 + \alpha(T - T_0)]$$

where T is the temperature, $\rho(T)$ is the resistivity of the material at the temperature T, ρ_0 is the resistivity of the material at the temperature T_0 and α is a coefficient that depends on the material.

Using the relationship that links resistance and resistivity (through section S and length L of the conductor):

$$R = \frac{\rho L}{S}$$

we get:

$$R(T) = R_0 \cdot [1 + \alpha(T - T_0)]$$

From this last relation, we can get the temperature by a measure of resistance.

The temperature sensor used on the engine analyzed was a Pt100, that is thermoresistance in platinum (Pt), in which the resistance at the temperature of 0°C is respectively 100 Ω.

The working range of this sensor is 200°C.

1.2.4 Oil pressure sensor

All the internal components of the engine are lubricated through a system of pipes and ducts, in which the oil is pumped at high pressure through a pump. Through the ducts and thanks to the centrifugal force, the oil reaches all the parts that need to be lubricated: camshafts, valves, bearings, rods, cylinders, pistons. The pressure sensor allows to define the oil pressure inside the lubrication circuit

The sensor is piezoresistive and works on the physical principle of piezoresistance: the resistive element (a membrane) follows the deformations of the sensor surface to which it is fixed; these deformations (typically elongations and shortenings) cause a variation in the electrical resistivity of the resistor material, and consequently its electrical resistance.

By connecting to this element a measuring system capable of read variations in resistance, it is possible to trace the amount of the deformation, and consequently the amount of the physical quantity that caused them.

The working range of this sensor is about 16 bar.

1.2.5 Combustion chamber pressure sensor

Figure 5 - Example of in chamber pressure sensor

To detect the trend of the cylinder pressure, DUCATI uses sensors facing in the chamber, the installation of which requests drilling the engine head.

Starting from the information of the sensor, it is possible determine the so called "indicated" quantities as they refer to the indicated cycle. The sensor is based on a piezoelectric functioning.

The sensing element facing the chamber, being of piezoelectric material, deforms when a pressure is applied, and charges of opposite sign accumulate on the two faces of the sensor.

The force that determines the deformation is proportional to the pressure acting on the sensor surface.

The sensor generates a voltage equal to the ratio between the amount of charge accumulated and the equivalent capacity of the sensor itself. So that sensor becomes a current generator, and the current is proportional to the pressure variation that acts on it.

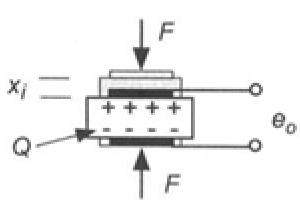


Figure 6 - Piezoelectric functioning

1.2.6 Lambda Probes

The lambda probe is necessary to detect the presence of oxygen in the exhaust gases, to ensure the stoichiometry of the mixture and to perform engine control. It is able to generate an electrical signal according to the concentration of oxygen. The correlation between oxygen concentration and electrical signal depends on the type of probe, therefore three types are distinguished:

- The EGO (Exhaust Gas Oxygen sensor) is a hysterical type sensor: the generated electrical signal (output) is a step one, characterized by a signal transition at $\lambda = 1$, at the corresponding voltage of rich mixtures, approximately 0.9 V, and that relating to lean mixtures, approximately equal to 0.09 V;
- The HEGO (Heated Exhaust Gas Oxygen sensor), has the same operating principle described for the EGO with the only difference of being pre-heated by an internal resistance;
- The UEGO (Universal Exhaust Gas Oxygen sensor) is linear: the electric signal generated has a linear trend as a function of the partial pressure of oxygen in the exhaust gases, for which, based on the voltage value, it is possible to trace the actual value of AFR (Air - Fuel Ratio).

For the engine analyzed we have an HEGO probe per cylinder as ECU sensors and a UEGO probe per cylinder as external sensor.

1.3 ECU sensors

To calibrate the ECU it is necessary to know its sensors and the actuators that allow the control.

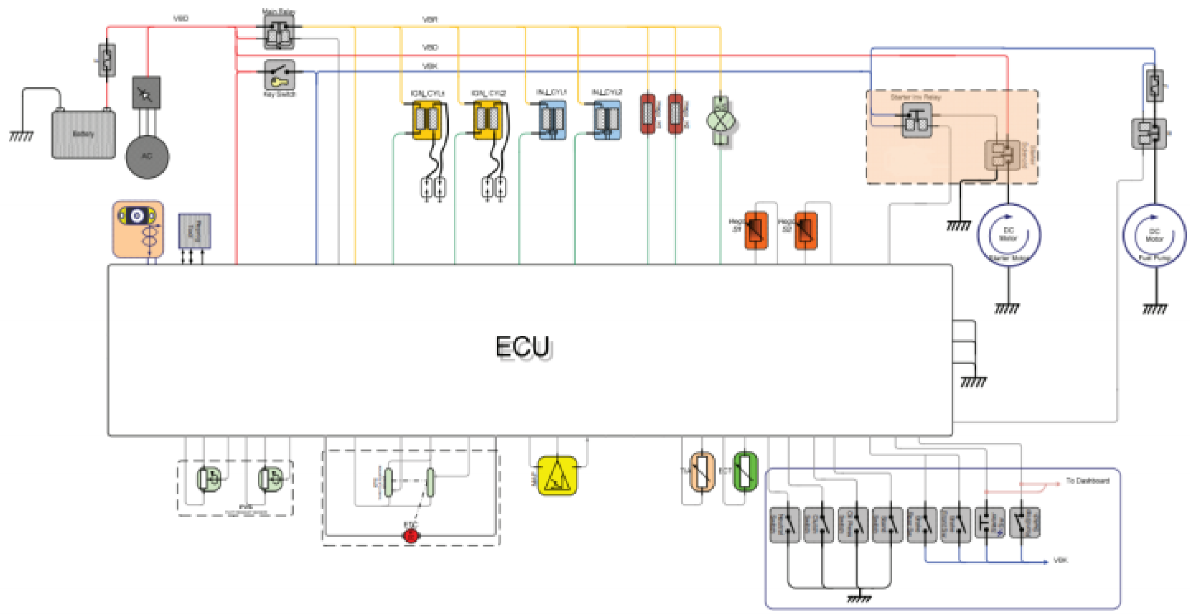


Figure 7 - ECU inputs and outputs in detail

1.3.1 Crank wheel and signals panel

The signals panel of a propulsor describes the behavior of the four-stroke engine, relating the four phases of each cylinder in relation to each other: intake, compression, ignition and exhaust, with angle position reference to the teeth of the phonic wheel put on the drive shaft

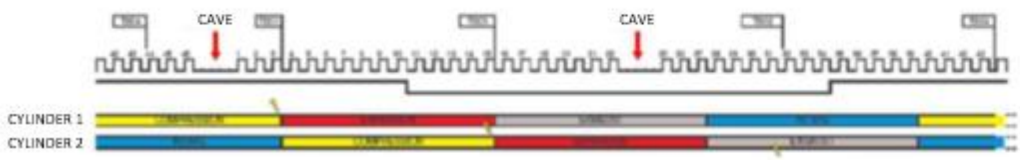


Figure 8 - Example of engine signals panel

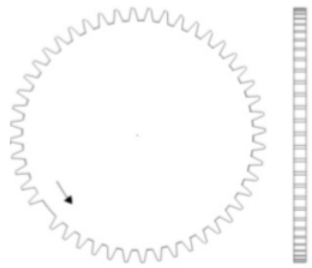


Figure 9 - Phonic wheel

The crank wheel is a gear with 28 teeth and a cave, i.e. two missing teeth, which allows the ECU to identify the rotation speed and the engine position. Electronic ignition/injection and all engine management are scheduled by the signal received from the wheel sensor.

1.3.2 Pick-up

The pick-up is a sensor with variable reluctance, positioned perpendicular to the teeth of the crank wheel and has the task of measuring the rotations of the shaft in order to synchronize the actuations that have to be phased with the angular position of the engine (ignition and injection).

Figure 10 - Pick-up sensor

The sensor consists of a coil wound around a permanent magnet and connected to the angular velocity detection terminal. The passage of the teeth of the phonic wheel causes a variation in the flow of the magnetic field which is transformed by the pick-up into a voltage signal from which it is possible to detect every tooth position and then determine the angular speed of the engine.

To be more precise, the passage of a tooth increases the relative magnetic permeability of the magnetic circuit, with consequent decrease of the magnetic reluctance. Then the magnetic flux increases again.

These sensors provide a very precise, repeatable and fast voltage signal, of the order of microseconds, adapt to be used for this type of application. The characteristics of the sensor, the gap (distance that separates the sensor from the phonic wheel) and the dimensions of the teeth of the crank wheel, determine the amplitude of the signal to be sent to the ECU.

1.3.3 Throttle potentiometer TPS

Figure 11 - Example of ETB (Electronic Throttle Body)

The DUCATI engine analyzed presents the ride-by-wire system (electronic accelerator), in which the throttle control and a linear potentiometer read by the ECU, gives the throttle position feedback to the control system. The opening angle is the result of a calculation chain that evaluates the torque request by the driver and the engine operating point. The features of this system allow engine management through torque requests, improving vehicle driveability, consumption and performance.

1.3.4 MAP sensor

Figure 12 - Example of MAP sensor

The MAP sensor is a transducer that measures the absolute pressure under the throttle, thus allowing the correction of the fueling by atmospheric pressure (and therefore altitude), in addition to identifying which cylinder is in intake at the passage of the phonic wheel cave and the volumetric efficiency of the specific cylinder.

1.3.5 Ignition System

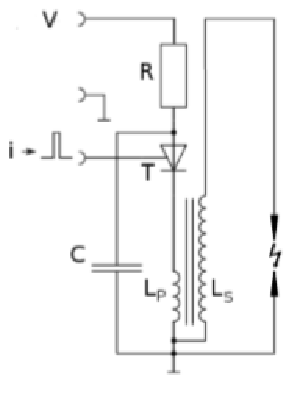


Figure 13 - Electric scheme of ignition

Coils and spark plugs form the ignition system: in particular, the coils are controlled by the ECU with a voltage pulse for a charge time that depends on their characteristics and on the supply voltage (battery charge level). The stored charge allows the spark plug electrodes to ignite the spark in the combustion chamber. In the analyzed engine are used stick-coils, that are modern coils inserted directly into appropriate seats in the engine heads and connected directly to the spark plugs.

The voltage of thousands of volts generated by the coil is transmitted to the spark plug, consequently between the electrodes the difference in voltage increases until it exceeds the insulating capacity of the mixture air/fuel which ionizes. A ionized gas becomes conductor, generating a short but very intense discharge.

The spark causes a local heating of the spark plug with temperatures ranging from 700 to 1000 ° C. The choice of the component is dictated by this feature, since a spark plug with an incorrect heat range[1] can bring pre-ignitions and then detonations.

1.3.6 Injection System

The fuel system consists of three basic elements: fuel pump, pressure regulator and injectors.

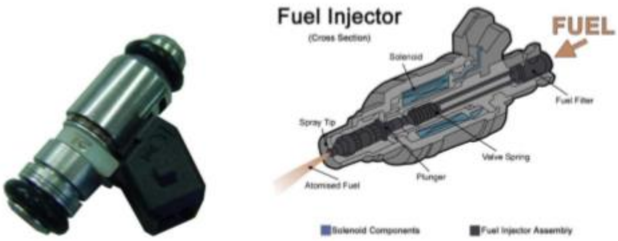


Figure 14 - Injector and its operating scheme

The fuel pump is an electropump located inside the tank that brings the system to a pressure of about 3 bar; it is activated by the ECU and is always powered during engine functioning.

The injector is a solenoid valve whose opening is controlled by an electrical impulse in tension sent by the ECU. The basic characteristics of the injector are:

- flow rate

[1] Depending on its heat range, a spark plug is called "hot" if it has a low ability to disperse heat; instead it is called "cold" if it has a good ability to disperse heat. The right heat dissipation capacity is very important because with a spark plug that is too hot, the resulting overheating would lead to a decline in performance and or self-ignition phenomena that could damage the piston. Vice versa, spark plugs with too low temperatures have more difficult starts with a cold engine and formations of deposits on the electrodes, capable of electrically isolating the two poles preventing the spark.

- number of holes
- angle of the spray cone

1.4 Bench management PCs and communication protocols between SWs

1.4.1 AVL PUMA®

PUMA is a commercial HW-SW architecture of AVL that allows total control of the bench in both manual and automatic mode. It is one of the most used applications in the industrial field, in modern engine testing rooms. Inside the tool all the information for the interaction and control of the brake, the cell equipment and the engine actuators are provided. In fact, the PUMA, by controlling the resistant torque exerted by the brake and acting on the handle actuator, regulates the engine rotation speed and the torque resultant.

Through the PLC (system that allows the management of the room and the alarm limits), the system regulates the temperatures of the coolant water, oil and air drawn in by the engine.

Figure 15 - Example of PUMA operator interface panel

The operator can select which parameters visualize, by means of digital indicators or with a virtual analogical instrumentation, arranging the various indicators as he wants. Furthermore, the graphic nature is an easy control of the test execution point.

All the acquired quantities can be saved for data monitoring and analysis.

1.4.2 OBI® and AVL IndiCom®

Figure 16 – OBI hardware

OBI (On Board Indicating), is a tool that allows to make an indicating analysis in real time. With “indicating analysis” we mean the determination of the parameters characterizing the combustion inherent to a given cycle. All the analysis is based on the pressure signal in the combustion chamber, which is detected on the basis of the engine phase, amplified and analyzed by the OBI hardware. From these two information it is possible to calculate quantities related to the indicated cycle: IMEP, IMEPH, MFB50 and KP_FREQ. Since the OBI is connected directly to the PC in real time, the hardware inputs (amplified cylinder pressure, phase sensor and eventually inputs for konocking analysis) can be seen on the screen using dedicated software.

The software used in the test room where I worked is IndiCom of AVL. The software allows the display of the pressure value in the cylinders and the analysis of the combustion itself.

1.4.3 ECU Management

The management of the engine actuators, on the bench as well as in the vehicle, is demanded to the ECU, a real control system equipped with a processor with its inputs, outputs and control parameters. The inputs correspond to signals acquired by the ECU from the numerous sensors present on the engine (crank wheel signal, intake pressure, etc.), while the outputs represent the values of the control levers available on the engine (ignition timing, injection timing, valve opening, etc.). The control parameters are the variables that allow the calibrator to decide which reaction (output) the ECU must have under certain operating conditions (input). These variables can be modified during the calibration phase (when development ECU is available) and, once optimized, are deliberated to be implemented in the standard ECUs and installed on production vehicles.

1.4.4 Development ECU

Typically, an additional electronic component is inserted in a development ECU, which takes the name of ETK (Emulator TastKopf), but in the case of the engine analyzed, we communicated and written to the ECU's memory by CCP (CAN Calibration Protocol). The CCP is, as the name indicates, a protocol for calibration and data acquisition from electronic control units.

The protocol is defined by ASAM (formerly known as ASAP (Arbeitskreis zur Standardisierung von Applikationssystemen). This is an international organization consisting of a number of significant vehicle manufacturers i.e. Audi, BMW, VW etc. Until now different technical solutions have been used for developing, calibration, production and servicing of ECU hardware and software. The aim of ASAM is to create a common tool for all levels of development of the computer hardware and software.

CCP supports the following functions:

- Reads and writes to ECU memory.
- Simultaneous calibration and data acquisition.
- Flash programming.
- Protection of resources.

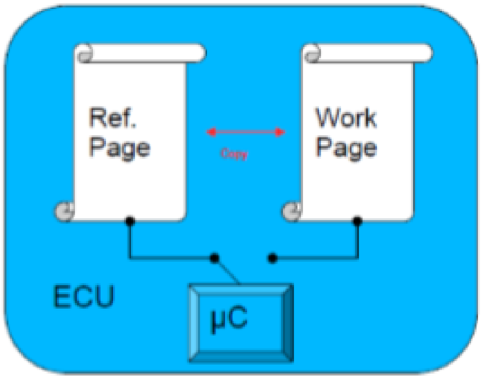


Figure 17 - Scheme for the management of ECU data sets

In a development ECU, two sets of data are stored:

1. Working Page (WP): is the data set in which are made the changes of the operating parameters during the calibration procedure.
2. Reference Page (RP): is a write-protected data set.

1.4.5 ETAS INCA®

INCA is the commercial software of ETAS, which allows the calibrator to read the values of the inputs and outputs, defined Measurements, and act on the control parameters, defined Calibrations, in order to identify the values that allow the best functioning of the engine. Through it, is possible to manage both RPs and WPs locally (on the ECU management PC) and it is possible to download inside ECU new software that needs to be tested/validated (ECU flashing operation). Without going too much into the operation of the SW, an example of Experiment is shown in the figure below, that is the interface of INCA from which it is possible to check Measurements and Calibrations. From the same figure we can see how the Calibrations can be of different shape and size; in fact, there are boolean (1), scalar (2), vector (3) and matrix (4) calibrations, while the Measurements (5) are generally scalars.

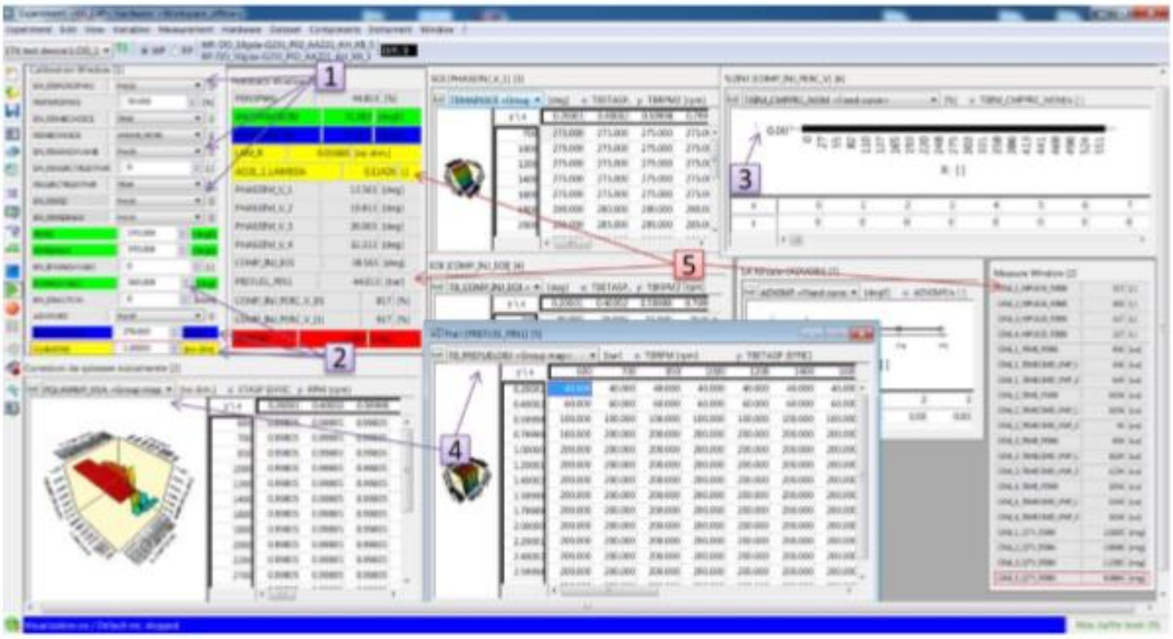


Figure 18 - Interface of the Experiment: we recognize the Calibrations from the Measurements because the first have a white background and the second are gray

1.4.6 AVL CAMEO®

CAMEO is an automatic and iterative calibration software. It is used to minimize bench measurement efforts, with the variation of different inputs at the same time and has rapid intervention on the exceeding limit thresholds imposed by the calibrator.

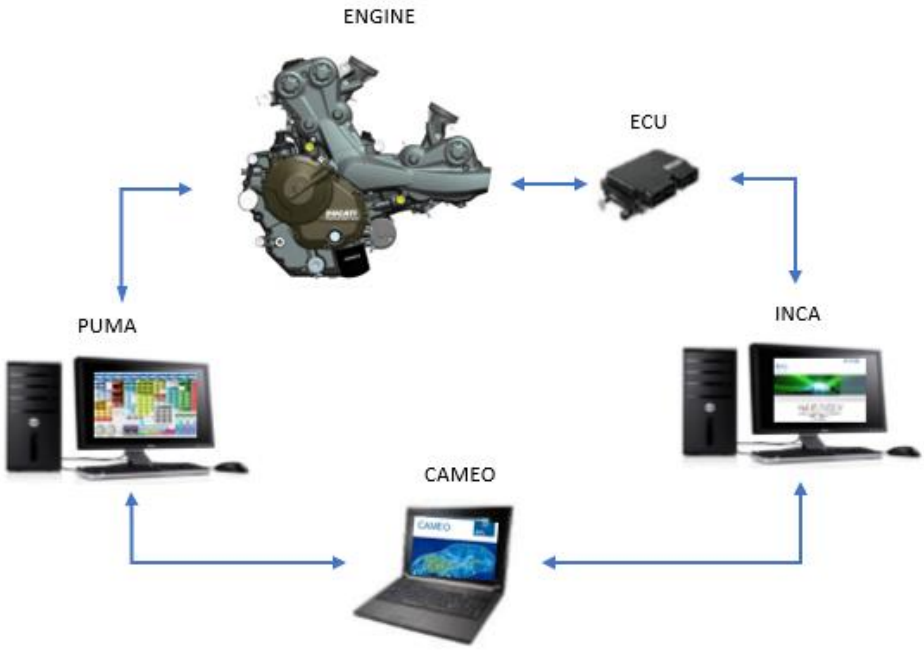


Figure 19 - CAMEO integration with engine and other tools

This software supports the entire calibration process, from the generation of DoE (Design of Experiment), to the production of the results and the creation of maps.

1.4.7 DoE

One of the most common approaches to the study of a complex system composed of many variables is to change one factor at time while keeping all the other fixed ones. The results often provide a limited amount of information with excessive working time.

With an appropriate application of the DoE we can drastically reduce the costs of the tests, obtaining a lot of useful information from the results.

The plans of experiment can be very different and dependent on the complexity of the model. The three main ones are:

1. Grid, in which tests are carried out with every possible combination of the parameters to be optimized (the number of tests can be very high);

2. Cross, in which a base point is set and one input is changed at time, but it is not suitable for complex physical systems;
3. Space-filling, is the method with which the entire space under test is uniformly covered, with a limited number of tests.

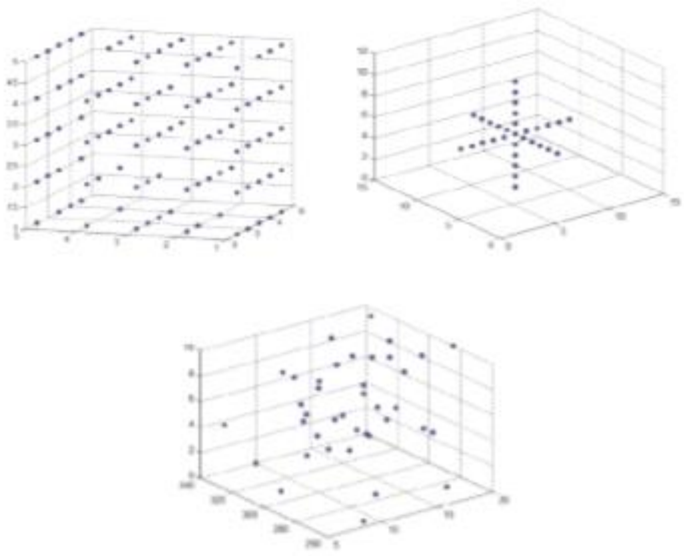


Figure 20 - Types of Plan of Experiment: Grid, Cross and Space-Filling

Statistical methods cover an important role in the planning, execution and analysis of the results of a complex system.

It is important to note that not all factors have the same weight on performance of the model: some may have high influences, other medium influences, or no influence. So an experiment needs to be carefully planned to understand what factors and how much weight have they.

For each DoE a series of specific parameters are defined:

- Factors: parameters on which we intend to act and object of study (controllable);

- Range: minimum and maximum variation value within which the factors will be changed. The range can be complete for a given factor or limited to a specific range of values if are already aware of its partial effect on the measurements;
- Measurements: measurements made on the system during the variation of the factors and object of optimization (not controllable).
- Experiment matrix: has in column the factors taken into account, and in each row the different values to analyze (the single line will be defined as point DoE).
- Test conditions: they represent the conditions in which the tests will be performed (speed, load, engine water temperature, acquisition time, acquisition frequency, etc.).

The execution of the DoE involves the simultaneous modification of the factors, in order to observe the corresponding changes.

The main advantages are:

- greater reliability of results;
- reduction of working time;
- reduction of costs;
- reduction of development times;
- stronger link between factors and measures.

1.4.8 ASAM ASAP3

In this paragraph I want to give the basic information on the integration of the engine with electronic systems in a test bench system (AuSy - Automation System) through an electronic calibration system (MC System - Measurement Calibration). The current integration takes place via a data connection of two types:

1. RS232 serial;

2. Ethernet with TCP/IP protocol.

The figure below shows the basic hardware structure of this integration. It shows that the MC system is used as a repeater between the AuSy and the control unit parameter management system. The communication is essentially based on a flow of requests by the AuSy of the parameter name and the size and its dimension in physical units (for example, air flow and m^3/h). All names or labels used in the AuSy and MC system are defined in the data description file .a21 which conforms to the ASAP2 ECU standard. When the INCA ASAP3 interface is started, we start writing commands and detailed responses to all parameters and can monitor the communication between the bench and INCA.

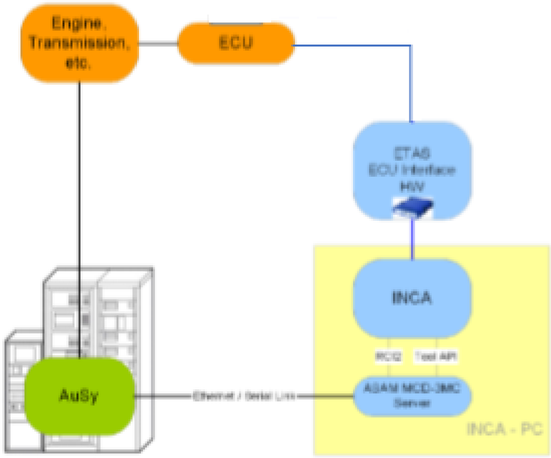


Figure 21 - Layout of ASAM ASAP3 standard with INCA

1.4.9 CAN Database

The Controller Area Network, also known as CAN-bus, is a serial standard bus introduced in the 1980s by Robert Bosch GmbH to connect several electronic control units. The CAN has been expressly designed to work without problems in areas strongly disturbed by the presence of electromagnetic waves. Noise immunity can be further increased by using twisted pair cables. The bit rate can reach 1 Mbit/s for networks less than 40 m long. Lower speeds allow to reach greater distances (e.g. 125 Kbit/s for 500 m).

To translate the data frame, a CAN device must be provided with a database that describes the signals contained in the message. The CAN Database File (CANdb file) is a text file containing this information. It allows to find data in a frame and convert it to engineering units. The data field can range from 0 to 8 bytes and can contain multiple CAN signals. The characteristics of the frame and of the signal when creating a .dbc file must be inserted correctly analyzing both the type of signal and the management of the signal itself from the source taken into consideration. For example, if the source is the PUMA, the names of the signals inside the file will represent the PUMA Input and Output variables.

2. Calibration activity and Data analysis

The data acquisition phase in the test room generates a lot of information on the functioning of the engine; therefore these tests serve not only for calibration but also for identifying any hardware or software anomalies.

The procedure for preparing and carrying out a test is defined in the Calibration Standard, an instructions document referred to a particular control unit.

Almost every map that needs to be calibrated requires a series of personalized instructions. The instructions sent to the test room technicians is common and consists of the following sections:

- Hardware set up: possible modifications to engine components, special sensors to be used, additional measuring instruments;
- Disabling: control functions that must be deactivated in order not to influence the measurement;
- Definition of points: engine points (rpm, load) in which it is necessary to carry out the tests[2];
- Procedure: sequence of operations to be followed for the test.

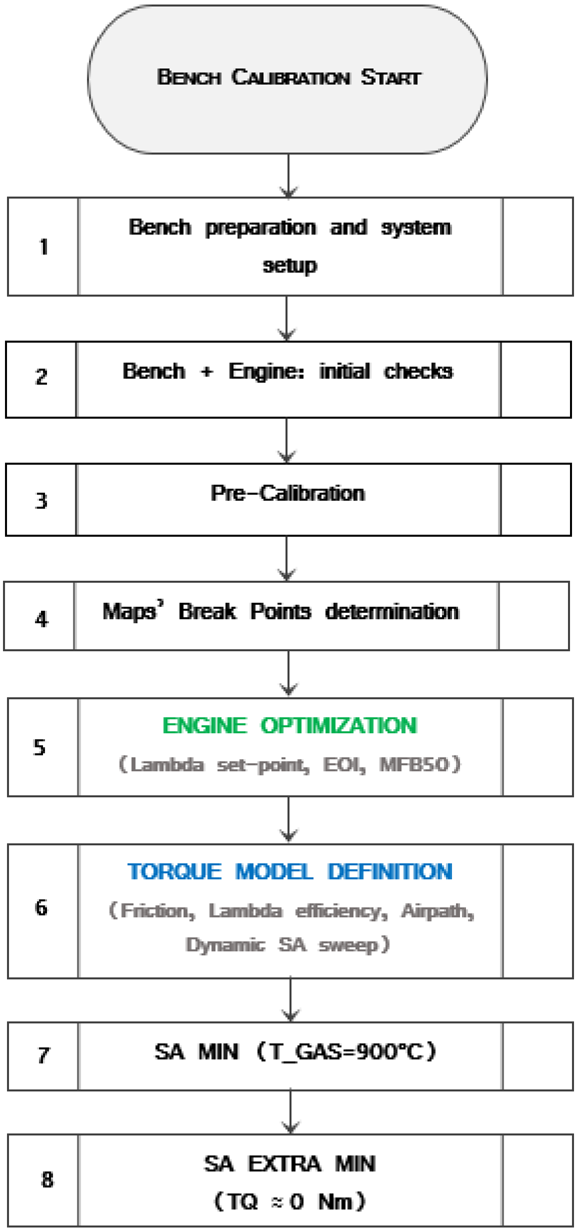
Once the tests on the bench have been performed, the data produced are analyzed. In the case of this Thesis activity, as mentioned before, MATLAB scripts were used.

From the files containing the data recorded by the cell operator, Excel files are created in which each row represents a different engine point, while each column contains the information of a

[2] These points are called K-points (barycentric points for engine optimization) depends on the stability of the output in that particular area of the map: the more the value tends to vary irregularly, the density of measurement points will be high.

certain signal. These documents can be easily read from MATLAB scripts thanks to functions for importing spreadsheets (for example `xlsread`).

2.1 Calibration Campaign



2.2 Bench preparation and system setup

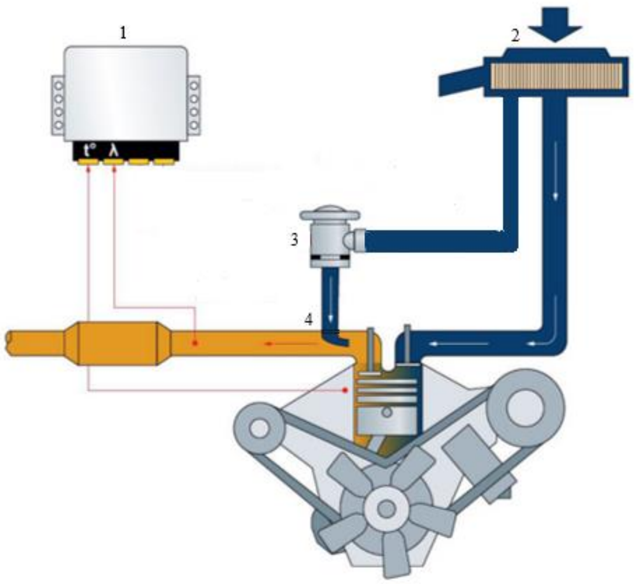
Before starting the calibration campaign, a check-up of the pressure and temperature sensors is performed.

- Intake pressure sensor (MAP sensor);
- Average Pressure (hole in airbox) and temperature before the throttle: pressure and temperature sensors in the air box (between air filter and throttle);
- Average pressure and temperature exhaust gas cylinder 1 & 2;

MAP sensors were installed with pressure sample pipes equivalent to vehicle in terms of length and diameter, respecting the expected mounting layout approved by the engine control system supplier.

Checks are made on the optical encoder, ignition coils signals and injection signals; the whole communication system between the bench management PCs; it is verified that the components in the engine are in frozen configuration: in particular air-box, exhaust system, camshafts and pistons; finally during bench preparation, also wiring harness is prepared and checked later on.

2.3 AIS Sensivity



1 ECU - 2 Air Filter - 3 Combined valve - 4 Lamellar valve

Figure 22 - AIS System

In some areas of the engine cycle, in particular those at low rpm and low loads, to improve the stability of combustion it is used to enrich the mixture ($\lambda < 1$). Enriching the mixture means producing more carbon monoxide and unburned hydrocarbons at the exhaust. To reduce the emission of pollutants, fresh air ("secondary air") is blown into the exhaust gas manifold for post-oxidation the HC.

Furthermore, the heat generated by the post-combustion of the HC heats the catalyst in the cold start phase and allows to reduce the waiting time for the regulation of the λ.

The Secondary Air System consists of a line that comes out from the airbox and is connected to a combined valve (in the figure the line leading to the other cylinder is not indicated). This valve, when opened, introduces fresh air into the exhaust manifold, blocking the exhaust fumes rising towards the airbox with lamellar valve. The lamellar valve opens when a depression wave is created in the exhaust manifold, allowing the passage of fresh air.

For the AIS sensivity test, engine runs in idle condition with clutch disengaged and deactivated secondary air. Deactivated all λ correction in order to not have any correction factor active and avoid mixture enrichment. Then activated secondary air and did not adjust λ; kept secondary air active, adjusted air path in order to have measured λ=λ target; deactivated one more time secondary air and didn't adjust λ (kept λ=λ target, then measured differences to λ=1: mixture revealed to be rich, λ<1).

10% of measured λ variation has been considered the best compromise for secondary air.

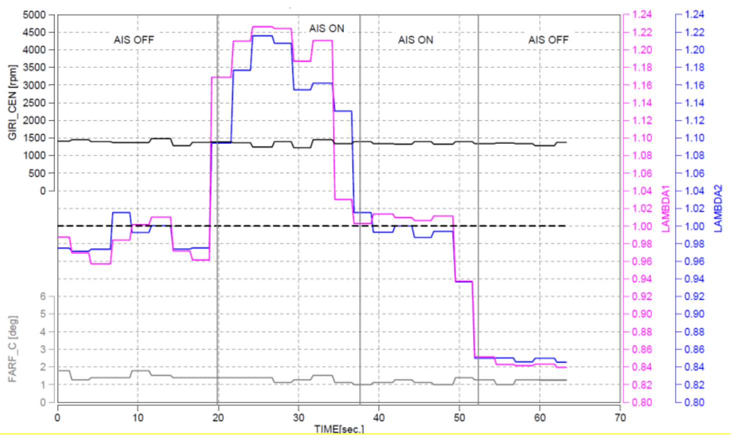


Figure 23 - AIS Sensivity Test

2.4 Pre-Calibration of Airpath and Spark Advance

Before starting the calibration campaign of the engine under investigation, it was needed to perform a pre-calibration of both air path (MAF) and spark advance (SA), considering the system as α - *n* one.

In this case, the calibration isn't the optimal one, but such a pre-calibration will allow to perform the entire campaign with a safer starting calibration. Moreover, it will also allow to run the engine in safer condition, by having already an overview of engine behavior over the entire engine working area.

In order to complete this step, it was necessary to start from a carry-over calibration, coming from a previous similar application. A carry-over is important in order to have at least a basic definition of lambda and EOI (End Of Injection) setpoint, that could be helpful to manage properly both air path and SA.

2.5 K-Points selection and stoichiometric area identification (λ = 1)

The most representative engine working points are called K-Points. The K-Points were chosen in order to represent in the best way each engine working area, depending on the main requests the engine has to fulfill on the selected areas like emission, performance and drivability.

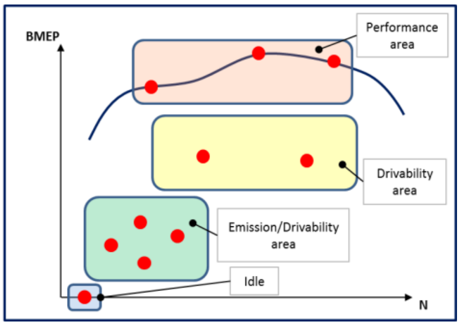


Figure 24 - Qualitative example of K-Points selection

K-Points are application dependent and their number may vary.

All K-Points within the emission area were provided by means of a tailored homologation cycle simulation of the application under investigation.

K-Points were provided in terms of engine speed and brake mean effective pressure (NxBMEP). On these points a tailored setpoint optimization was performed, in accordance with the main requests depending on the involved area.

2.6 RPM breakpoints determination

An upward ramp of 90 sec was performed, keeping the engine on the chosen working point for 10 sec, then performed a downward ramp of 90 sec. The working points went from N_{MIN} to N_{MAX} for different values of throttle.

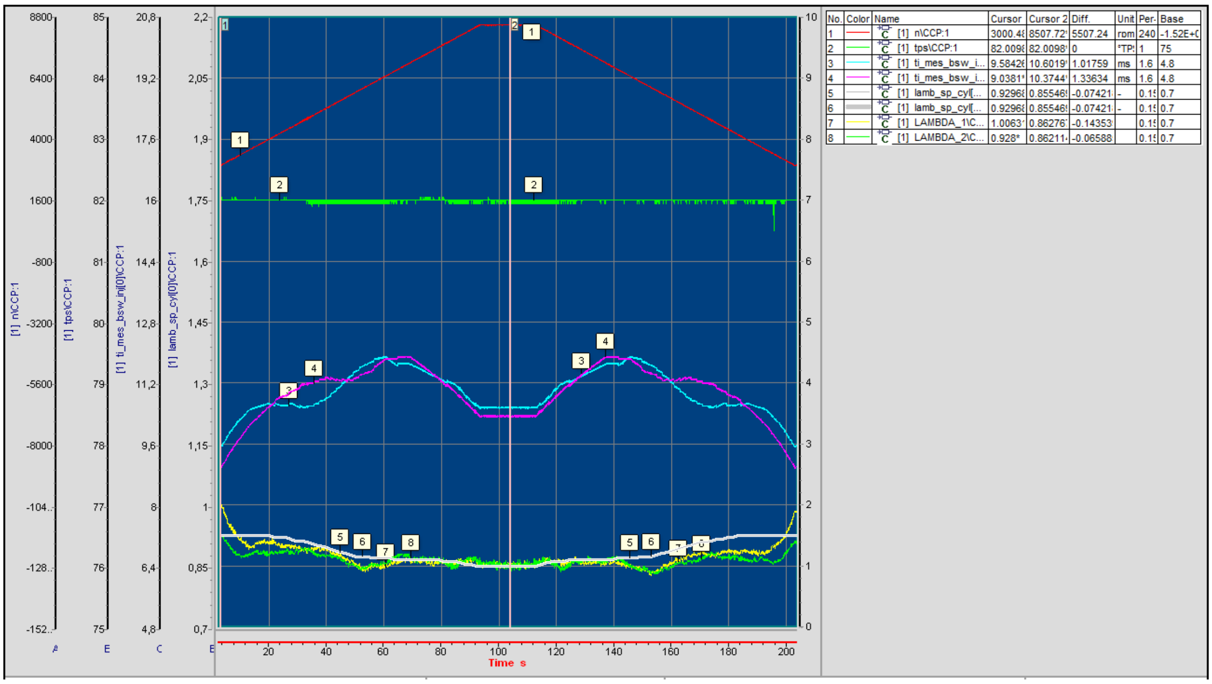


Figure 25 - Example of engine speed ramp at 82° TPS, from 3000 rpm to 8500 rpm

The purpose of the test was to evaluate real MAF behavior and select RPM breakpoints accordingly, in order to not loose accuracy due to possible non-linearity.

2.7 LOAD breakpoints determination

Like for RPM BREAKPOINTS DETERMINATION, upwards and downwards TPS ramps have been performed. These ramps were performed on 4 different engine speed:

- Minimum reachable engine speed
- Max torque point
- Max power point
- High engine speed

Per each engine speed, the TPS ramp was made from minimum possible angular position in steady state conditions to full load (WOT).

The ramp was continuous and slow (50/60 sec).

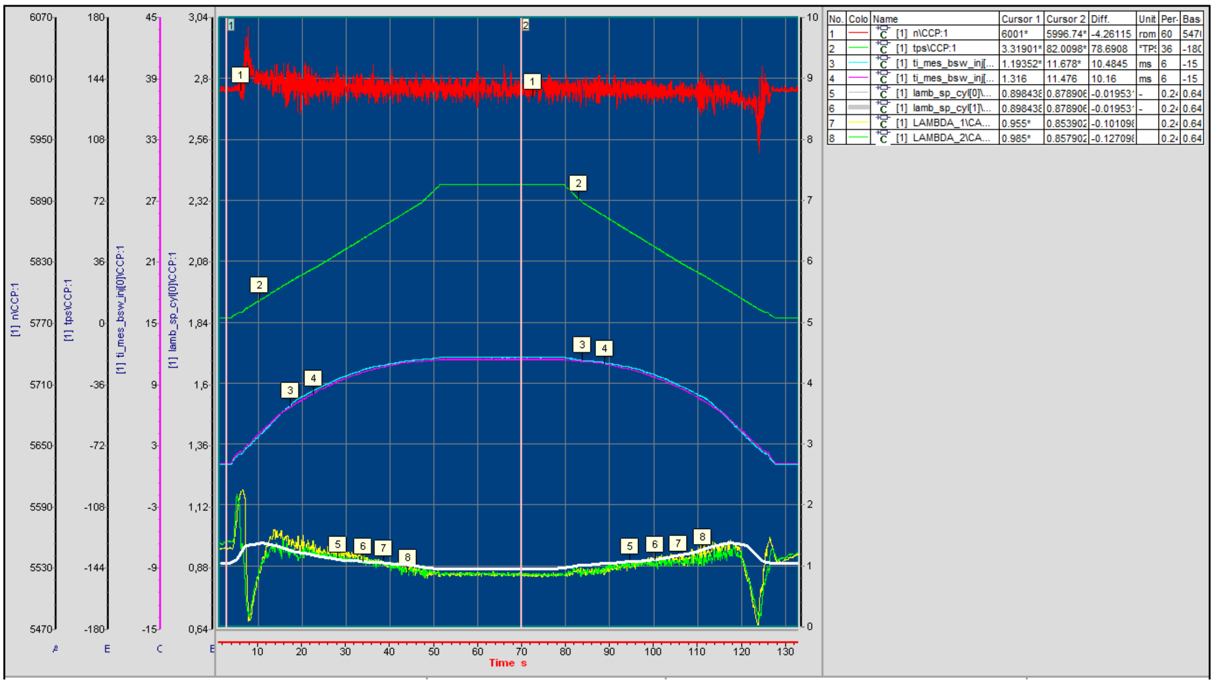


Figure 26 - Example of TPS ramp @ 6000 rpm

This test has been performed using last updated calibration in terms of Air Path and Spark Advance.

Like the test described before, also this test had the purpose of evaluate real MAF behavior and select TPS/MAF breakpoints accordingly, in order to not loose accuracy due to possible non-linearity.

2.8 LAMBDA setpoint determination

A lambda sweep between 0.7 and 1.1 was done on 4 engine working points:

- Max Power point
- Max Torque point (or all points where a torque peak was present)
- Minimum engine speed (i.e. 3000rpm) at which WOT was possible
- Maximum engine speed (close to physical speed limiter) and minimum throttle position (Blow-By ≈ 50 l/min)

The SA has been adjusted for each lambda value in order to have maximum Torque and Power (at the best/safe possible MFB50, 8-12° CA).

Lambda set point determination																						
pedal/N	1100	1500	2000	2500	3000	3500	3700	3950	4500	4750	5000	5250	5500	6000	6200	6500	7000	7250	7500	7800	8000	8500
2%																						
4%																						
6%																						
8%																						
10%								LAMBDA = 1														
13%																						
15%																						
25%																						
35%																						
40%																						
50%																						
65%																						
80%																						
100%																						

Figure 27 - Example of engine working point selection for lambda set point determination and respective stoichiometric area extension

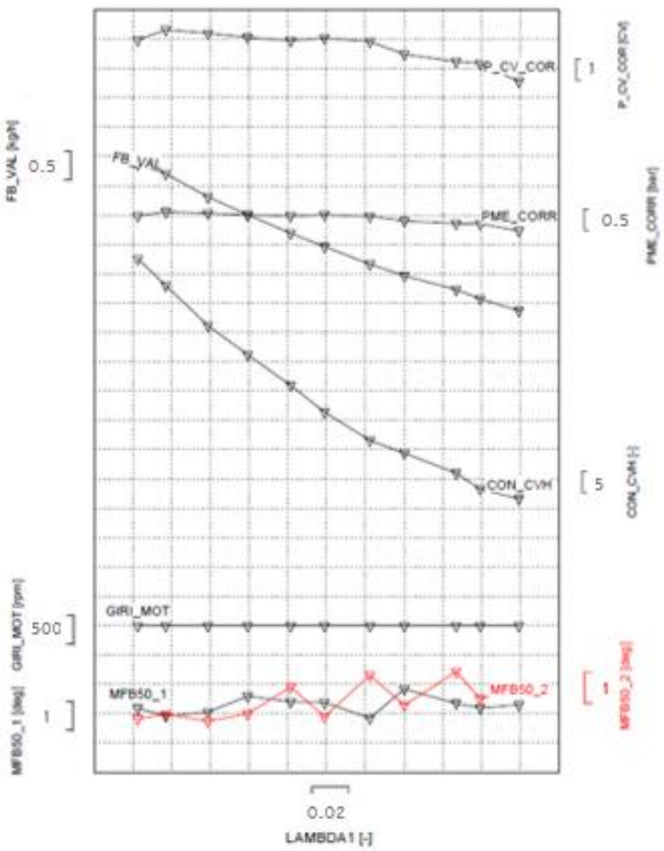


Figure 28 - Example of Lambda sweep (with quite constant MFB50)

For slightly rich mixture (λ = 0.8 ÷ 0.9) the optimum SA angle is lower than that calibrated at λ = 1.1, because he combustion speed is higher and, to maintain the MFB50 angle centered around values of 8-12°CA after the TDC, the combustion must start a few moments later. Continuing to enrich the mixture, the combustion speed is slowed down again due to poor mixture quality.

The SA must be instead increased for lean mixtures, whose combustion speed is lower.

At high load a rich mixture is needed to prevent knock effect and contain exhaust gas temperature, in order to increase the performances.

2.9 EOI Calibration

After completed lambda set point determination, it has been calibrated the injection phase, performing EOI sweep (EOI: End Of Injection) from TDC overlap cycle *i* to TDC overlap cycle *i*+1, with step of 30°CA, in all K-Points chosen as above.

On all K-Points within emission cycle area, the EOI sweep has been performed with both AIS ON and OFF.

EOI Calibration																						
pedal/N	1100	1350	2000	2700	3000	3500	3700	3950	4500	4750	5000	5250	5500	6000	6200	6500	7000	7250	7500	7800	8000	8500
2%																						
4%																						
6%		AIS ON							AIS ON													
8%																						
10%				AIS ON/OFF																		
13%																						
15%									AIS ON/OFF													AIS OFF
25%																AIS OFF						
35%																						
40%																						
50%																						
65%																						
80%																						
100%					AIS OFF				AIS OFF							AIS OFF			AIS OFF			

Figure 29 - Example of selected operating points for EOI optimization

The engine was run in λ closed loop (where λ_target=1) and setted SA corresponding to MFB50=10°/12°CA on the first EOI, investigating the whole range keeping MFB50 constant.

EOI optimization is a compromise among Torque/Emission/IMEP CoV/Fuel Consumption.

Obviously more attention has been given to the low rpm – low load points where the injection window is very small and it is important to know how to control it (at full load the injection window is almost as big as all the indicated cycle).

As results of the test it was obtained that for EMISSION area an optimal EOI value should lie during intake valve closure phase (about 130° ÷ 180°CA), while for PERFORMANCE area a better comprise should be given injecting with intake valve opened (close to 400° ÷ 500°CA). So generally, the EOI window is +130°CA/+500°CA, depending on the area under investigation.

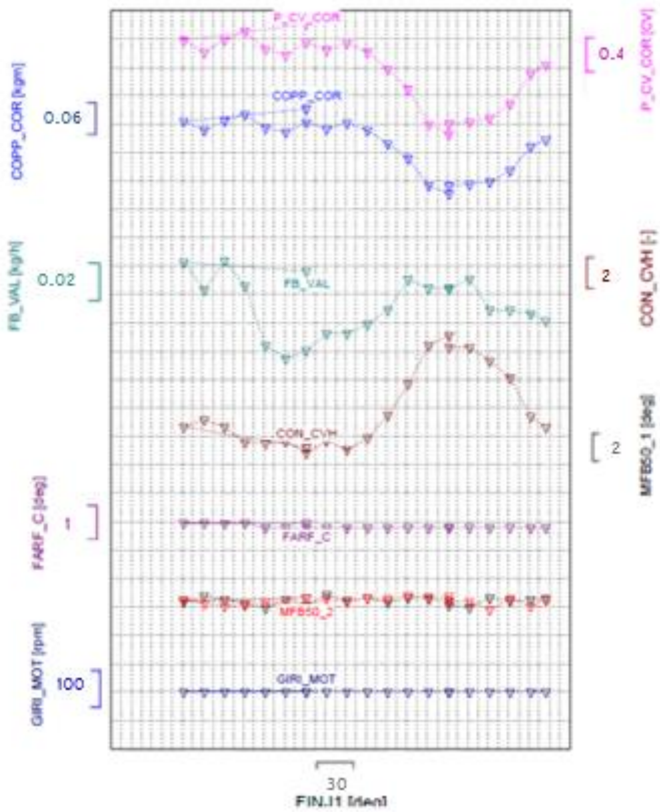


Figure 30 - Example of EOI sweep

2.10 Combustion Title Umbrella Curve (λ Efficiency)

Parameter λ is of fundamental importance for operation and for engine control. It is defined as the ratio between the mass of air and the fuel mass in the combustion chamber $\lambda=(A/F)$: the stoichiometric value is identified by the chemical reaction of combustion and depends on the type of fuel. For "rich" mixtures it is $\lambda < 1$, for "lean" mixtures it is $\lambda > 1$.

Lambda sweeps with SA = cost are executed to measure torque behavior depending on lambda variation. They were executed on 6 points:

- Idle running (dyno clutched)
- Max Power point
- Max Torque point
- High RPM - Low Load

- Low RPM - High Load
- Low RPM - Low load

	1100	1350	2000	2700	3000	3500	3700	3950	4500	4750	5000	5250	5500	6000	6200	6500	7000	7250	7500	8000	8700	9200
2%																						
4%		Idle running																				
6%					Min TPS																	
8%																						
10%																						Min TPS
13%																						
15%																						
25%																						
35%																						
40%																						
50%																						
65%																						
80%																						
100%					λ=0.7 to 1.1													λ=0.7 to 1.1				λ=0.7 to 1.1

Figure 31 - Example of operating point selection for lambda efficiency curve test

λ has been changed from 0.7 to 1.1, with step of 0.02 and kept on the point for 10 sec.

Exhaust temperature and knock were kept under control (if knock limit was reached in lean condition with a given SA, we reduced SA until killing knock effect and perform the whole sweep with that SA value). These data were acquired in order to build lambda efficiency curves.

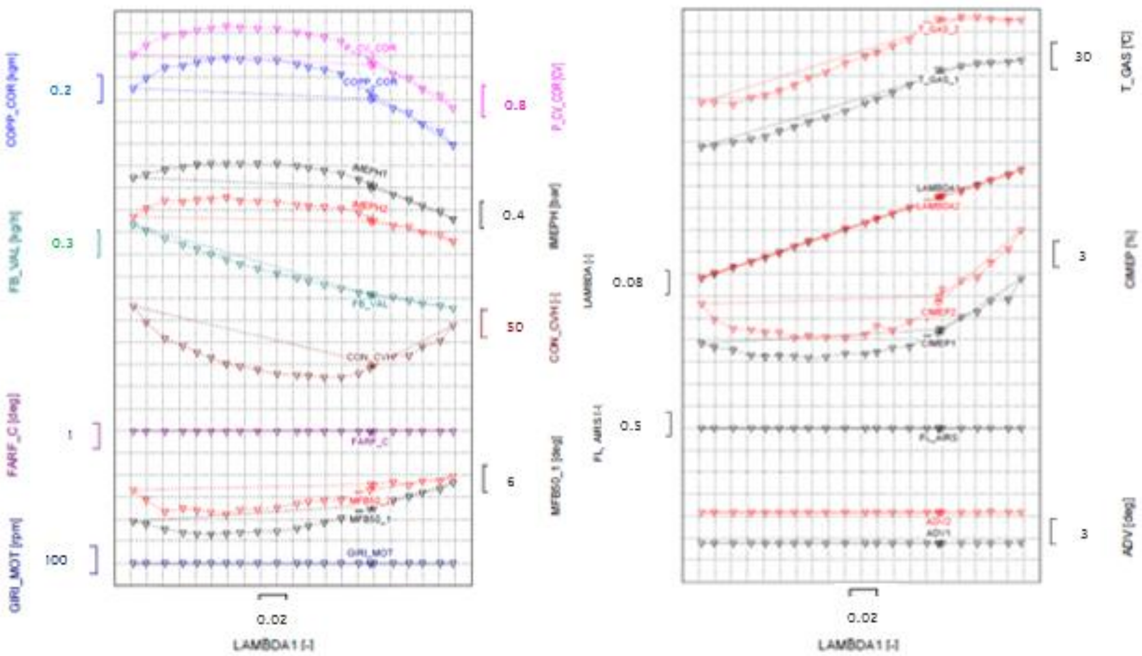


Figure 32 - Example of λ sweep performed (with SA constant)

The variation of lambda has an effect on the amount of torque transferred and on the ignition advance angle. The highest torque is normally obtained for lightly rich mixtures ($\lambda = 0.9$) while with lean mixtures the performances decrease even if, thanks to the possibility of completely burn the injected fuel, benefits are obtained in terms of consumption. The torque is maximum for rich mixture values because more oxygen molecules are present thanks to the dissociation of the combustion products and it is therefore possible to burn more fuel than the quantity established from the chemical reaction of combustion.

The effect on the spark advance is seen because according to the air/fuel ratio the combustion rate changes (in particular with λ close to 0.9 the higher speeds occur). The optimum ignition angle must then be moved to maintain the MFB50 (angle where half of the fuel injected was burned) in a window between 8°CA and 12°CA after the top dead center; in this way the torque efficiency will be maximum.

2.11 SA Sweep for Emission Optimization on K-Points (NxBMEP)

SA sweep for emission optimization purpose has been performed on all K-Points defined as in paragraph 2.5.

As already mentioned, K-Points are established in terms of NxBMEP coordinates and they are the most representative points of the homologation cycle performed for emission measurements, such as for mid load area and full power.

So SA sweep has been performed on all K-Points related to the following areas:

- Emission
- Driveability
- Idle

In this case SA sweep has been done keeping BMEP (Torque) constant and the throttle has been changed in order to compensate SA effect.

If all K-Points chosen for SA sweep belong to the emission area (λ target = 1), it was worth to make it mainly with AIS ON**.**

After completed spark advance sweep for emission optimization, SA_OPT (or MFB50_OPT) has been determined in the low load area by means of tailored analysis based on a trade-off among Emission / IMEP CoV / Fuel consumption**.** On the other hand, moving towards the high load area, the SA_OPT (MFB50_OPT) has been chosen mainly basing the selection criteria on Performances.

The results obtained and the considerations made for the low load area are summarized in the next page:

EMISSION
- SA ↑ (MFB50 ↓) ⇒ NOx ↑ , CO ↑ , THC ↑ — WORST COMBUSTION, BUT BETTER FUEL CONSUMPTION
- COMBUSTION IMPROVES IF SA ↓ (MFB50 ↑)

DRIVEABILITY
- DRIVEABILITY IS HIGHLIGHTED BY IMEP COV:
- IF IMEP COV ↑ ⇒ DRIVEABILITY ↓

IDLE
- THE IDLE CONTROL IS HIGHLIGHTED BY RPM COV:
- IF RPM COV ↑ THE IDLE IS BAD CONTROLLED

Since MFB50 is an operating engine parameter, it shall be easier to establish the MFB50 trend on the entire engine working area.

Therefore, the creation of such a map made possible an easier calibration of SA_OPT during the test **“AIR PATH & SA_BAS CALIBRATION AIS OFF/ON”**, because of the presence of an MFB50 target to be reached in each engine running point.

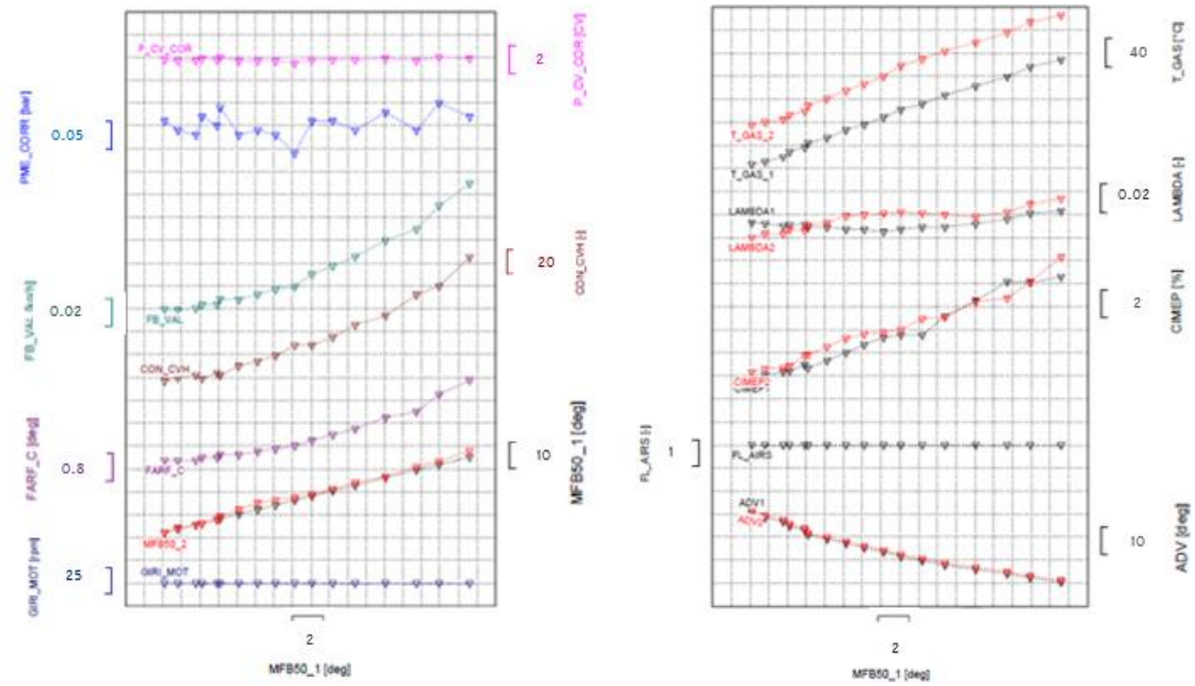


Figure 33 - Engine operating parameters during SA sweep

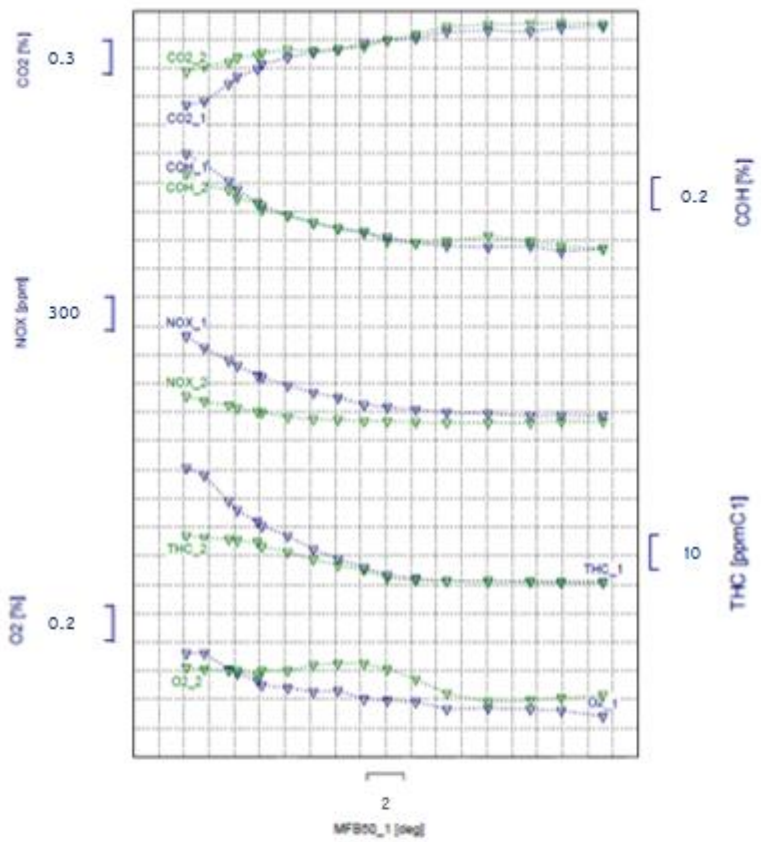


Figure 34 - Engine-out emissions during SA sweep

2.12 Airpath & Basic Spark Advance Calibration (AIS OFF/ON)

With the air model it was calculated the intake air mass value according to the rotation speed, throttle position and the intake manifold pressure.

In this part of the campaign, the calibration methodology is explained with both AIS OFF and AIS ON.

Therefore, MAF and SA adjustments have been done on all the RPM-TPS breakpoints.

2.12.1 Air Path + SA Calibration AIS OFF

- MAF ADJUSTMENT

In order to adjust MAF, it has been used the lambda defined in paragraph 2.4 and adjusted the MAF_OPT maps on the tested point to reach the lambda target.

Based on those data, both air path tables (Speed-Density/Speed-Throttle) have been built for each cylinder.

- SA ADJUSTMENT

The ignition was adjusted in the N-TPS tables, and reconverted in the N-MAF tables after post-treatment. If full load ignition retardation was needed in order to avoid knock (in the area where the MAF is flat), it has been put in the N-TPS table.

The nominal crank angle for the optimal MFB50 (corresponding to the MBT condition or emission / fuel consumption / IMEP CoV optimum) has been defined in paragraph 2.2.5, and it is so defined per each running point, by interpolating the map of Optimal MFB50 in N-TPS. Therefore, SA has been adjusted in order to guarantee such value.

The test has been performed with:

- AIS OFF over the whole engine working area
- AIS ON only where lambda set-point is 1

2.12.2 Air Path + SA Calibration AIS ON

The same test above has been repeated with AIS ON**,** only where λ target = 1

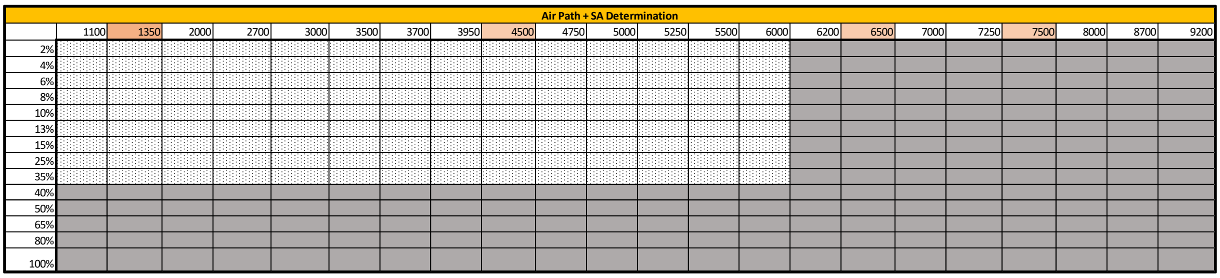


Figure 35 - Example of N/TPS area with λ target = 1

To create the Airpath map, it has been used a script that reads the values of some signals recorded at the bench from an Excel sheet, and in a few calculations generates the required calibration.

The steps are as follows:

1. First of all, it is indicated the document from which read the test data;
2. The breakpoints of the maps to be generated have been chosen *a priori*;
3. Defined some Boundary Conditions;
4. At this point the script is able to generate the map;
5. As a last step, is possible to save the calibration in an Excel worksheet.

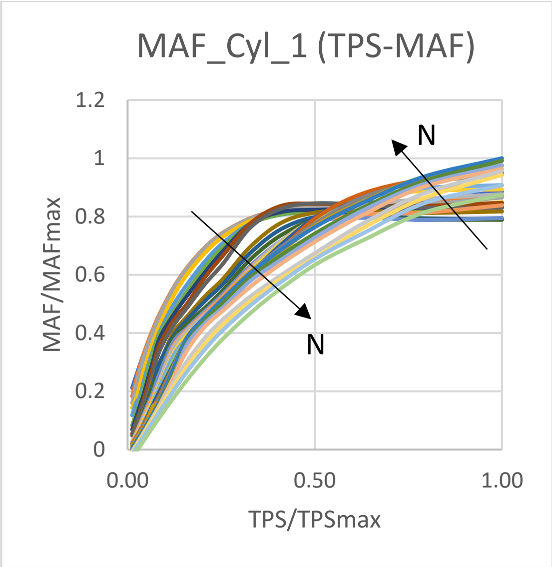


Figure 36 - Final airpath: MAF according to TPS

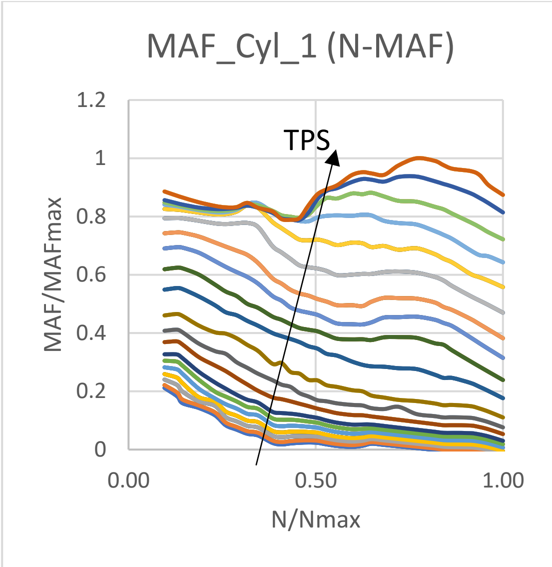


Figure 37 - Final airpath: MAF according to N

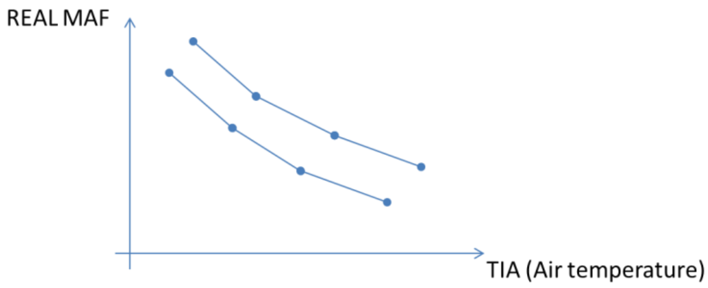
The graph on the right shows the effect of the throttle on the volumetric efficiency (cyl 1 is taken in consideration but the observations are also valid for the cyl 2): where the curves present a knee are the typical air flows at low rpm where sonic block occurs, i.e. even continuing to increase the opening the MAF does not increase. What makes the flow chock is not the throttle but the critical area defined by the valve.

The graph on the left shows the effect of the rpm: the inflection point indicates the resonance zones characteristic of the acoustic phenomena that occur in the intake manifold.

2.13 Temperature Model

This model was needed to evaluate and adjust air model (MAF) in hot and cold condition.

A correlation between real MAF and air temperature has been evidenced in different operating points, in terms of NxTPS and also N-MAP_RATIO (MAP_RATIO = ratio between MAP and Environmental Pressure).



The test has been performed in different operating points like those reported below:

- 1350rpm @Idle position
- 1350rpm @2%
- 2500rpm @4%
- 2500rpm @10%
- 2500rpm @25%
- 4000rpm @7%
- 7500rpm @20%
- Max rpm @30%
- 4000rpm @82%
- Max rpm @82%

Starting with TIA = 25°C (as reference condition) (TIA : Intake Air Temperature), we made a record for each point reported above in terms of N-TPS and for each of them recorded the respective value of MAP_RATIO for each cylinder. In this way, a corresponding list of N-MAP_RATIO points has been created.

Once the test at TIA=25°C has been completed, we performed the same test also at TIA=Minimum°C and TIA=Maximum°C on each N-TPS point. If the new MAP_RATIO value registered with new TIA level differed from that measured at 25°C in the same operating point (N-TPS), then, after completed the measurement on the given N-TPS point, the TPS has been

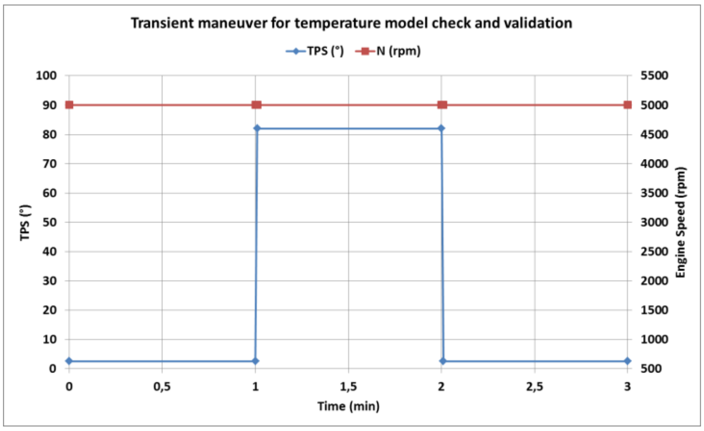


Figure 38 - Final maneuver to perform at the end of the test

adjusted in order to obtain the corresponding N- MAP_RATIO point and made other measurements.

After completed all measurements for each TIA level, we performed the following 2 specific tests, with continuous recorder:

1. at middle rpm (4000-5000rpm) run the engine with very low throttle opening for 1 minute, then opened quickly the throttle to wide open and waited 1 minute
2. at middle rpm (4000-5000rpm) run the engine with wide opened throttle for 10s, then close quickly the throttle and waited 1 minute.

2.14 Friction Determination (HOT/COLD)

2.14.1 Hot Friction

Frictions represent the amount of energy transferred to the piston that is not available to the shaft for transmission to the wheels.

Hot frictions are generated by two different contributions:

1) Pumping losses: these are visible in the graph p - V in the area between the intake line (pressure lower than the atmospheric one) and the exhaust line. This type of frictions is bigger when the engine operates at low loads, i.e. when the throttle is almost closed, and offers a strong resistance to the air flow;
2) Mechanical losses: it is the amount of energy spent to win the dissipations that arise from the relative movements of the components. They included the friction between the piston rings and the cylinder walls, the friction in the bearings of the rotating shafts, the

friction of the mechanisms of the intake and exhaust valves, the frictions of gears and belts and auxiliaries such as oil and water pumps.

From the data acquired on the bench, it is not possible to distinguish between frictions coming from pumping losses and mechanical losses, but from the trend of the curves we see that at high rpm and high MAF relative speed between moving parts increases, so mechanical losses increase; at low rpm, an increase in the MAF implies a wider throttle opening and a reduction in pumping losses.

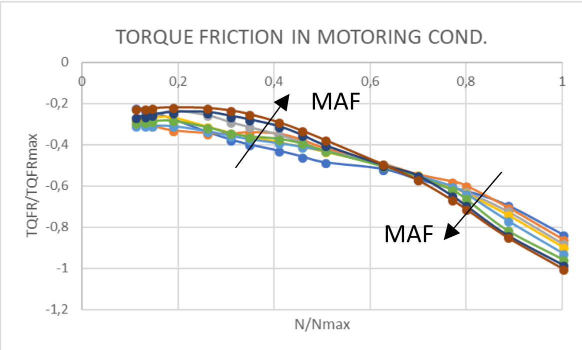


Figure 39 - Example of torque losses measurement according to N

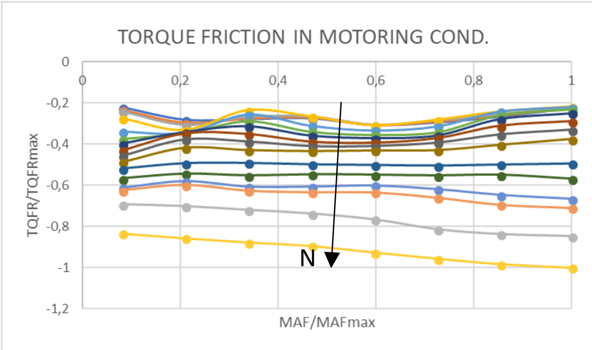


Figure 40 - Example of torque losses measurement according to MAF

To acquire bench data and calculate engine friction, it is possible to proceed in various ways. In our case the engine was brought to the desired point (N,TPS), performing ramps in the hot engine condition (TCO≈85°C) (TCO: engine coolant temperature) from 1350 rpm to max engine speed in 60 sec and turning off the cylinders to enter in the cut-off condition (injection time set to 0), imposing on the brake (which is able to carry the engine) to continue to rotate the shaft at the same rotation speed of the instants before the cut-off. In this way, by reading the torque value necessary to the brake to guarantee the constant motion of the engine, it gets exactly the job to win the frictions and allow the entry of air into the cylinder. Then we kept the engine at max rpm for 10 sec with injection active, then went down from max rpm to 1350 rpm in 60 sec again without injection.

Once the air-path has been fulfilled, TPS has been converted in MAF and the table of the Torque Friction been calibrated.

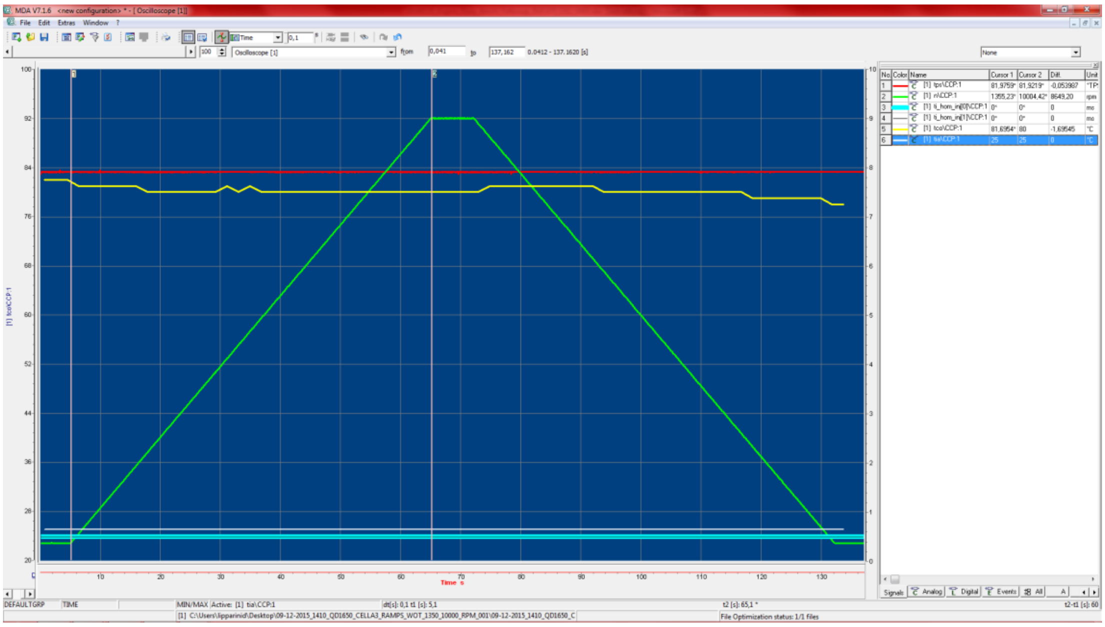


Figure 41 - Example of test execution for the hot friction determination

The engine points analyzed were:

- Min TPS (tuned TPS in order to keep the engine in low idle)
- 4 %
- 10%
- 30%
- 50%
- WOT

The study of hot frictions is necessary in order to identify the correct amount of torque that the engine must get to satisfy the driver's request.

2.14.2 Cold Friction

Cold friction test has a double goal:

1. Measure torque losses in cold condition
2. Measure the enrichment needed during warm-up phase

The test has been performed as follows.

After selected a given engine working point (N,TPS), a warmup ramp was performed, from TCO=25°C up to fully warmed up engine (TCO≈90°C) with an injection cut-off of 10 sec every 5°/10° of TCO, in three or four different engine operating points.

	TPS / N	IDLE	3000		5000		7000
TCO cold => fuel cut off for some seconds during WUP every 5°/10° of TCO	**Min TPS**	X					
	10°						X
	20°		X		X		
	WOT						

Figure 42 - Example of chosen operating points for warm-up evaluation

The figure below shows how the test has been performed.

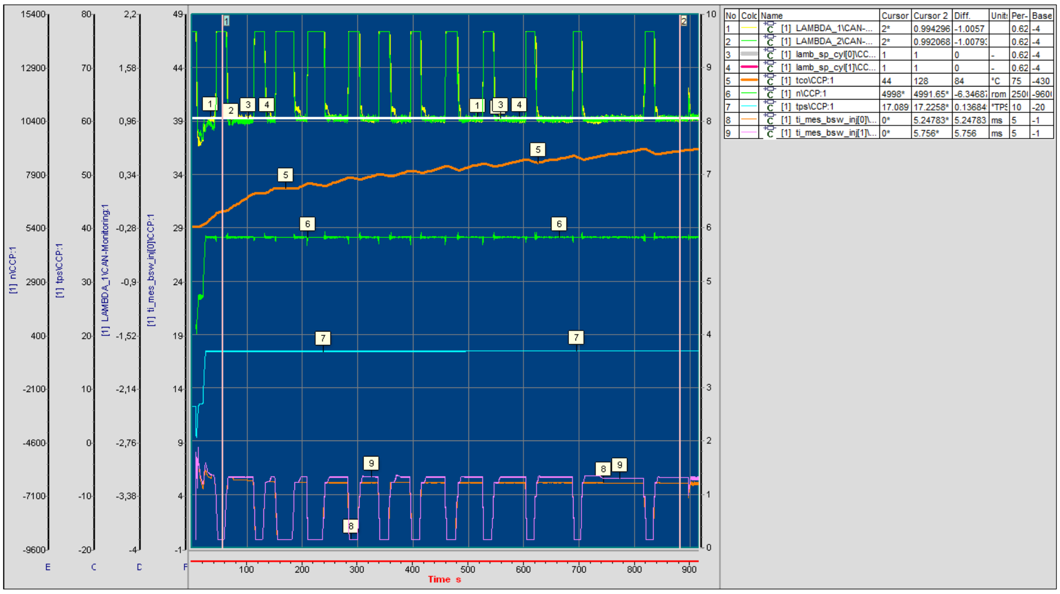


Figure 43 - Cold friction test @ 5000 rpm x 17° TPS

2.15 SA Dynamic Sweep

Among the last tests followed at the bench there is the SA dynamic sweep, an activity certainly longer than the others, useful to determine the effect that different angles of ignition have on combustion and therefore on the torque.

This test was needed for the definition of the following parameters:

- **SA EFFICIENCY**: map representing the torque variation (with the same intake air and injected fuel) based on the variation of SA. The SA variation as a function of efficiency is necessary in torque control because the system calculates SA degradation to be applied to satisfy the driver's request, ensuring the other torque requests.
- **SA REF**: corresponding to the spark advance that provides the maximum engine torque, also called Maximum Brake Torque (MBT), for each engine operating point. Even if in some points SA REF cannot be physically reached because of knocking, it is computed by looking at the torque shape as function of the SA SWEEP performed in that specific point.
- **TQI REF**: maximum torque values (obtained by implementing the reference spark advance) for each engine point. From this, by multiplying the value of a given engine point with the efficiency obtained by implementing the SA_BAS, the basic torque is obtained.
- **SA EXTRA MIN (only at low load)**: minimum spark advance to have a break torque equal to 0 ± 5 Nm. It can be physically reached only in those engine working points where maximum exhaust temperature (880 °C ÷ 900 °C) is not exceeded. By means of this test, SA EXTRA MIN cannot be reached everywhere because of the presence of the catalyst, but only at low speed - low load working points, where zero torque is obtained

without exceeding the maximum temperature indicated. On other points, it can be reached only if an exhaust system is used, as indicated in chapter 2.17.

Dynamic sweeps of spark advance have been performed, starting from the maximum allowed spark advance on cylinders, limited by knock (KP_FREQ ≤ 1÷2 %), until the minimum allowed one, limited by maximum exhaust temperature (T_GAS≤ 880÷900 °C) or zero torque (BRAKE TORQUE = 0 ± 5 Nm).

The test has been done keeping measured λ=λ target, adjusting λ by means of MAF maps on the tested point.

Torque, exhaust temperature, knock limit and measured λ has been acquired by means of continuous recording.

SA SWEEP data acquisition requires a particularly high computational effort. The tool used to perform this test was AVL CAMEO (referred paragraph 1.4.6 for further information on this software).

As said, one of the objectives of the test is to identify the efficiency curve that will be inserted in the control unit and will be used for all engine points. For this reason it is essential that this calibration correctly approximates as many of the curves obtained to the various point (N/TPS) as possible and it is for this reason that the tool performs more calculation iterations.

Once the SA value, at which the maximum torque corresponds, is determined for each engine point, the torque trends are considered in relation to ΔSA applied, with respect to that at maximum torque.

Therefore, the same torque efficiency trend is considered, but normalizing the torque with respect to its maximum value in each point in relation to ΔSA as mentioned above.

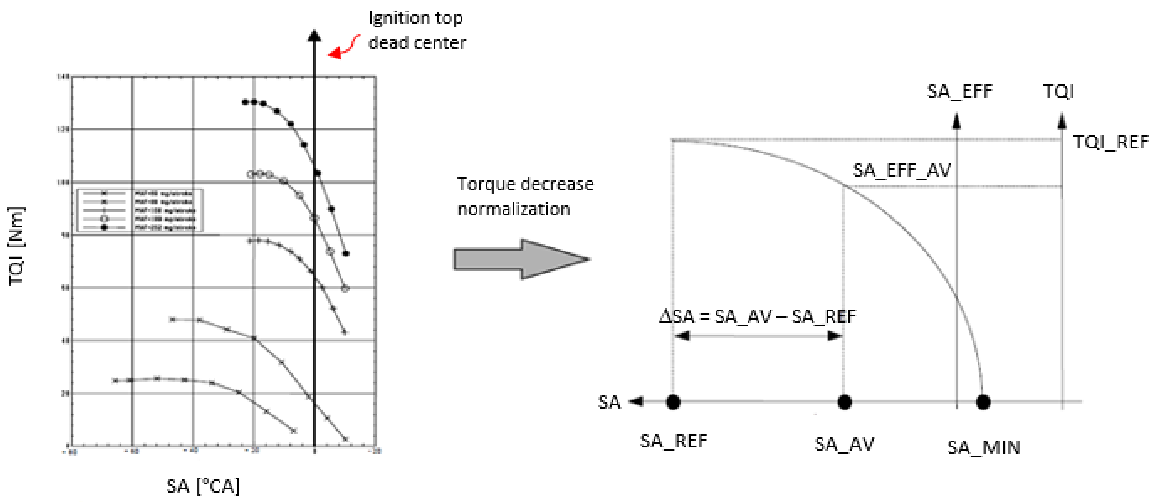


Figure 44 - Homogeneous Torque Equation/Single ignition efficiency

$$SA\ EFF\ AV = \frac{TQI\ (SA)}{TQI\ REF\ (SA\ REF)}$$

The equation above represents the combustion efficiency with respect to the application of the SA optimum (SA_REF). This information is properly the curve that is typically stored in the ECU to know the effects of the spark advance implemented (compared to the optimum) on the torque produced.

2.16 SA MIN Determination

This test identifies minimum spark advance that would make the engine running with constant exhaust temperature close to 880 °C ÷ 900 °C and can be used for longer time without incurring in catalyst damaging. This value of spark advance is called SA MIN. These tables limit the minimum spark advance that the algorithm of the torque structure could ask during normal working condition, in order to guarantee exhaust components protection.

The test has been performed making a sweep of spark advance of 5°CA/10°CA, moving together the spark advance of the two cylinders.

When the exhaust temperature of one cylinder reached 880°C÷900°C, the test was stopped retarding SA on that cylinder, but kept going on the other cylinder until the same exhaust temperature is reached. In this way, SA MIN is well defined for both cylinders with the final SA step. See the example shown below.

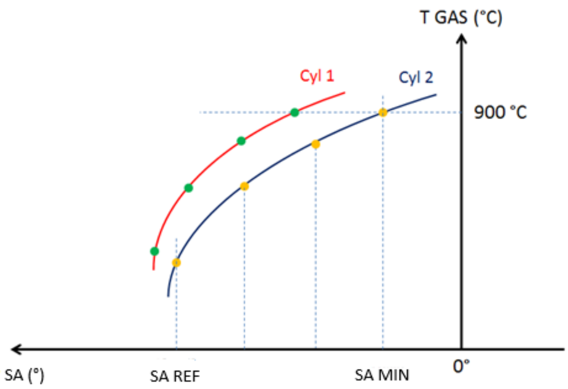


Figure 45 - Example of SA MIN Determination

2.17 SA EXTRAMIN Determination

SA EXTRA MIN test has been performed as final one, because of the high probability of damaging the engine.

SA EXTRA MIN represents the minimum spark advance that provide a null brake torque (Engine Torque = 0). It can be obtained retarding a lot the spark advance in a given N/TPS operating point. Such a spark advance degradation will strongly increase exhaust temperature close to 1200°C or more.

SA EXTRA MIN is applied during normal driving condition when a sudden request of zero torque is needed, and it can be applied only for very short time interval.

For example, zero torque could be requested by “Quick Shift” strategies, which has the aim to allow gear change without using the clutch.

Such a test can only be performed with a dedicated exhaust system, without catalyst, in order to avoid catalyst melting. During the test, per each running point, the measurement could last more than 3/5s, which is much longer than the time interval during which SA EXTRA MIN is being applied on vehicle. For this reason, it is better to make the test without catalysts.

The test was performed as quickly as possible in order to not stress the cylinder head close to the exhaust valve side, too.

3. SA MIN/EXTRAMIN: comparison between SA Sweep extrapolated data and measured data

After completing the calibration campaign and going to the data analysis at the desk, we realized the possibility of avoiding the tests of SA MIN and EXTRAMIN determination and extrapolate these values from the SA continuous sweep, obtaining the trend of IMEPH = f (SA) and TGAS = f (SA).

3.1 Data Modeling

What we did was to develop a MATLAB code that could read from a .xlsx file some parameters produced by the dynamic spark advance sweep:

```
Input_Names = {'n' 'tps' 'SA_cyl_[1]' 'SA_cyl_[2]'};

Output_Names = {'IMEP_H_1' 'IMEP_H_2' 'T_GAS_1' 'T_GAS_2'};
```

The code requires input definition of N and TPS breakpoints (specifying a tolerance on these, that in our case has been fixed to `toll_bkp = 0.01`) and obviously a temperature target, being SA MIN one of the result to be achieved, mainly dependent on a maximum discharge temperature. A `TGAS_tgt = 890 [°C]` has been chosen[3].

We used the function `[P, S] = polyfit (X, Y, n)` which determines the polynomial of order `n` that better fit the couple of points `X,Y`.

```
[P_IMEPH,S_IMEPH] = polyfit(SA,IMEPH,3);

[P_TGAS,S_TGAS] = polyfit(SA,TGAS,2);
```

---

[3] Actually, considering that at the bench, in the high-speed and high-load area, we've maintained lower temperatures (obtaining higher SA), in the model we set that up to 50 TPS the `TGAS_tgt` was equal to `890 ° C`; from 50 TPS onwards `TGAS_tgt = 880 ° C`

```
if  TPS_bkp > 50

    TGAS_tgt = 880; %[°C]
else
    TGAS_tgt = 890; %[°C]
end
```

With the function `roots (P)` were found the polynomials roots; the command `polyval` determines the result deriving from the application of the polynomial.

```
% Calculation of polynomial roots

            roots_dIMEPH = roots([P_IMEPH(1)*3 P_IMEPH(2)*2 P_IMEPH(3)]);
            roots_TGAS = roots([P_TGAS(1:2) P_TGAS(2+1)-TGAS_tgt]);

% Calculation SA MIN @ T_GAS = 890 °C

            pos_SA_MIN=find(roots_TGAS<90 & roots_TGAS>-30);
            SA_MIN=min(roots_TGAS(pos_SA_MIN));
            IMEPH_MIN = polyval(P_IMEPH,SA_MIN);

% Calculation SA_EXTRAMIN

            TQI = (-1)*interp2(TQFR_X_axis,TQFR_Y_axis,TQFR_map,N,maf)/2;
            IMEPH_EXTRAMIN = (TQI*4*pi*10/Vd);
            SA_EXTRAMIN = fzero(@(x) polyval([P_IMEPH(1:3) P_IMEPH(3+1)+
                                                   - IMEPH_EXTRAMIN],x),0);
```

The values of SA MIN and SA EXTRAMIN thus obtained were compared with the bench values. The comparison was made simply by making the difference:

SA MEASURED - SA MODELED

If the difference is negative it means that the model is more conservative, i.e. the value of modeled SA is higher than the one measured; if the difference is positive it means a lower SA has been modeled.

3.2 Results and Considerations

The results obtained are shown below:

SA_MIN_1_Measured - SA_MIN_1_Modeled																					
	3300	**3600**	**4000**	**4250**	**4500**	**4750**	**5000**	**5250**	**5500**	**5800**	**6200**	**6500**	**6750**	**7100**	**7500**	**8000**	**8500**	**8750**	**9000**	**9500**	**9750**	**10000**
10							3,75	3,84	7,45	7,51	10,26	3,40	6,35	8,25								
13			6,61	4,59	4,76	5,85	3,39	4,09	2,71	2,87	4,54	8,49	7,34	6,55	4,31	7,52	2,53					
17		5,14	5,53	4,72	4,16	3,81	3,25	4,05	3,18	1,43	2,39	2,28	3,92	2,95	4,60	5,92	1,45					
21	6,00	6,35	6,32	4,34	5,03	3,37	3,25	2,75	4,36	3,32	2,68	4,36	5,49	3,04	2,54	3,51	2,60					
27	5,35	7,91	8,33	5,58	4,58	4,07	3,43	4,95	4,34	4,14	4,24	4,48	4,50	3,92	3,56	3,95	5,37	3,99	3,83			
34	5,69	8,41		6,06	3,39	4,48	2,84	3,80	4,06	4,51	3,13	2,06	3,51	2,93	3,20	3,70	4,37	3,00	3,00	2,20	1,01	0,43
42	6,65	7,46	10,44	5,36	3,95			3,02	4,26	4,88	4,09	1,70	1,89	3,20	3,54	1,36	3,17	2,16	2,92	2,53	1,63	0,11
52	7,40	7,70	8,81	5,44	3,10	2,51	4,50	2,68	3,44	4,36	4,78	3,53	3,02	2,81	2,64	4,44	4,31	2,67	2,61	4,92	1,68	0,52
65	7,39	7,14	7,42	4,59	2,76	2,05	3,31	3,00	2,54		4,18	4,61	3,54		0,98	4,25	5,80	3,13		4,10	2,13	2,84
82	9,21	8,03	9,10	5,41	3,21	1,92	3,85	2,83	2,99	2,90	4,16	4,19	3,85	2,38	1,42	1,98	3,91	2,73	2,80	3,76	1,81	0,56

Figure 46 - Results of comparison between SA_MIN meas and mdl on cyl 1

SA_MIN_2_Measured - SA_MIN_2_Modeled																						
	3300	3600	4000	4250	4500	4750	5000	5250	5500	5800	6200	6500	6750	7100	7500	8000	8500	8750	9000	9500	9750	10000
10							4,38	5,66	5,34	5,49	9,45	6,71	6,71	7,93								
13			5,97	4,28	4,99	5,44	3,58	5,32	4,15	6,95	6,79	6,42	5,59	4,97	2,87	3,51	8,42					
17		5,96	6,40	5,15	5,43	4,74	3,90	4,77	3,67	4,20	5,26	5,29	6,64	6,27	3,35	2,95	5,53					
21	5,43	6,63	7,74	5,16	5,21	4,97	3,25	3,60	2,92	3,91	5,45	5,55	7,11	6,88	4,57	5,26	4,43					
27	6,98	7,34	8,96	5,68	4,74	5,97	3,72	2,65	2,66	3,79	4,75	5,50	5,20	6,57	4,50	1,89	3,11	3,83	2,79			
34	6,21	6,63		5,31	6,19	3,78	2,84	2,53	2,43	2,94	4,40	4,95	4,32	3,96	3,47	2,01	3,86	2,84	2,79	3,32	2,80	1,87
42	7,00	5,57	9,84	7,08	6,75			3,29	3,43	5,15	5,71	4,10	4,54	3,98	2,37	2,28	2,35	2,09	1,44	4,93	1,81	1,94
52	7,29	7,82	9,03	6,37	6,99	3,90	4,41	3,96	4,60	5,71	6,95	3,88	4,40	3,87	2,98	3,47	3,20	2,28	3,06	3,32	4,35	3,14
65	7,76	6,53	7,16	6,12	5,19	4,22	2,99	4,10	3,73		6,35	3,34	5,22		3,59	5,62	5,84	3,15		3,33	4,04	4,55
82	7,80	6,18	6,57	5,64	6,08	4,10	3,43	2,42	3,76	3,73	4,64	2,14	3,42	3,36	3,17	3,89	3,45	2,56	2,00	2,63	3,28	1,90

Figure 47 - Results of comparison between SA_MIN meas and mdl on cyl 2

SA_EXTRAMIN_1_Measured - SA_EXTRAMIN_1_Modeled																						
	3300	3600	4000	4250	4500	4750	5000	5250	5500	5800	6200	6500	6750	7100	7500	8000	8500	8750	9000	9500	9750	10000
10							0,80	3,56	5,66					-0,62								
13			2,75	2,37	1,05	1,34	0,39	3,87	4,33	4,27	4,68	7,23	4,95	4,45	0,06	-1,39	-4,96	-5,63				
17		-0,78	4,19	1,40	-0,34	2,03	2,29	3,95	6,36	4,79	4,88	5,92	4,53	1,61	0,99	0,36	-3,16	-3,20				
21	-0,13	0,23	5,18	1,14	0,74	1,34	-1,40	3,21	3,68	4,90	4,69	4,55	3,88	3,64	2,20	1,25	0,07	0,43				
27	-1,24	0,52	2,48	0,93	0,55	-0,10	-0,48	-0,32	1,02	3,64	2,79	2,72	2,72	4,78	1,77	1,26	1,12	1,92	-1,20			
34		-2,23	2,63	-1,46	2,20	-0,48	0,14	-1,15	0,59	0,68	1,81	1,37	1,95	3,90	0,94	1,14	2,68	0,50	0,12	0,09	-0,13	-1,05
42	-2,52	-3,12	2,67	-1,55	-2,49			-1,79	0,63	1,44	1,14	2,35	0,79	1,33	3,12	0,43	1,04	1,71	0,41	1,47	-0,87	-1,24
52	-2,26	-3,40	-1,21	0,17	-0,34	-0,39	-2,23	-2,38	-2,24	-1,52	1,50	2,42	-0,22	3,19	0,26	3,18	0,41	1,85	1,62	2,12	0,56	-1,83
65	-1,63	-3,14	-1,32	-2,28	0,48	-1,62	-0,48	-2,85	-5,93		2,58	2,65	1,00		-0,48	1,04	2,84	0,95		2,26	-0,11	-0,48
82		-3,06	-1,87	-1,79	-1,68	-2,53	-0,91	-3,38	-2,85	-1,92	0,83	0,92	2,40	1,20	1,38	1,22	3,34	1,03	3,04		-2,95	-0,39

Figure 48 - Results of comparison between SA_EXTRAMIN meas and mdl on cyl 1

SA_EXTRAMIN_2_Measured - SA_EXTRAMIN_2_Modeled																						
	3300	3600	4000	4250	4500	4750	5000	5250	5500	5800	6200	6500	6750	7100	7500	8000	8500	8750	9000	9500	9750	10000
10							-2,18	0,76	-0,33	-0,27	2,67	6,63	6,40	8,49								
13			2,08	1,40	0,76	-0,18	-0,57	0,93	-1,25	1,11	-0,92	1,79	2,46	5,98	8,46	11,14	12,63					
17		0,83	3,11	0,29	0,57	0,41	-2,33	-1,00	-0,55	-0,48	0,20	4,79	5,84	7,99	5,75	9,63	13,26	12,98				
21	0,37	1,43	4,44	-0,53	0,19	0,90	-2,93	0,25	-1,19	-0,63	1,69	2,92	4,87	6,08	3,47	6,42	6,31	7,21				
27	0,04	2,44	3,23	0,73	0,89	-0,07	-1,25	-1,34	-0,41	1,99	2,78	3,00	2,76	5,44	5,11	4,69	5,19	6,09	3,86			
34	0,03	1,45	3,47	1,61	1,75	-0,10	0,26	-1,08	0,04	0,89	2,79	2,42	1,84	4,91	1,96	3,45	4,50	4,63	4,26	4,92	4,24	2,43
42	1,08	-0,08	5,34	1,52	1,24			-1,10	2,35	2,70	3,16	4,05	3,04	2,17	3,71	2,16	1,99	3,36	3,68	3,95	2,97	1,89
52	1,37	0,32	3,59	2,41	1,04	0,59	1,98	0,48	2,12	1,75	3,45	3,56	1,78	2,78	1,51	2,31	1,10	2,82	2,76	3,89	3,58	1,64
65	1,18	0,36	3,31	-0,46	1,97	0,35	0,76	-1,05	-2,22		1,79	3,33	2,23		0,80	2,50	3,46	2,48		2,60	1,93	3,42
82		0,15	0,45	0,71	0,06	0,15	0,19	-0,26	0,55	-0,74	1,75	1,52	3,67	1,69	2,40	2,94	3,46	1,25	1,90		2,98	1,85

Figure 49 - Results of comparison between SA_EXTRAMIN meas and mdl on cyl 2

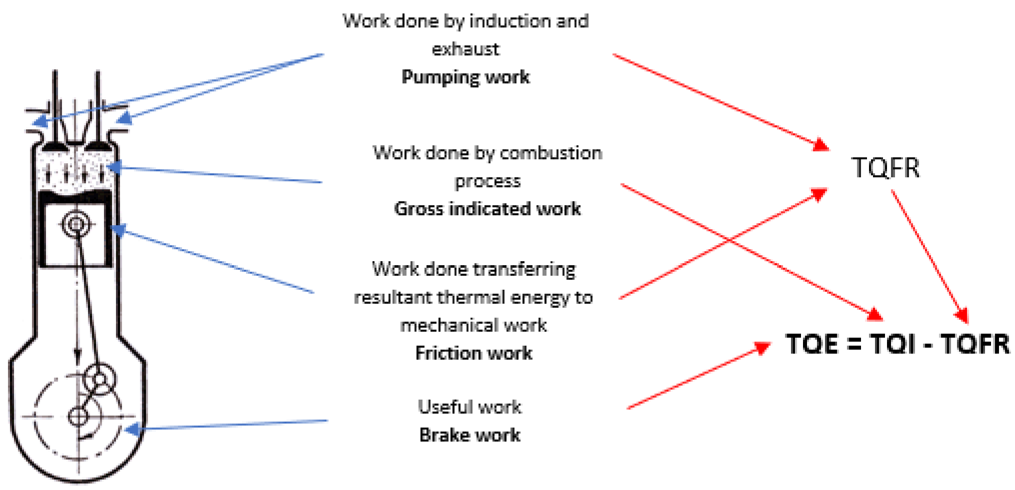
As can notice, the values resulting from the operation of difference are low, which means that the model is acceptable and that extrapolation works.

As described in paragraphs 2.16 and 2.17, the execution of the tests for the determination of SA_MIN and SA_EXTRAMIN subjects the engine to high stress. The result we have arrived puts in the hypothesis of avoiding these two tests at the bench, saving hours of work and avoiding the problems that usually occur in the execution of these activities and reinvest rather on experimenting new functionalities.

4. Torque-Based Structure

4.1 Thermodynamic Behaviour and Torque Calculation

The purpose of an internal combustion engine is to develop a certain power, determined by the torque available at the clutch and by the rotation speed. The clutch torque is that developed by the engine during the combustion phase, except for frictions due to pumping losses, mechanical losses and auxiliaries. Then the torque available to the wheels is clutch torque multiplied by the transmission ratio.



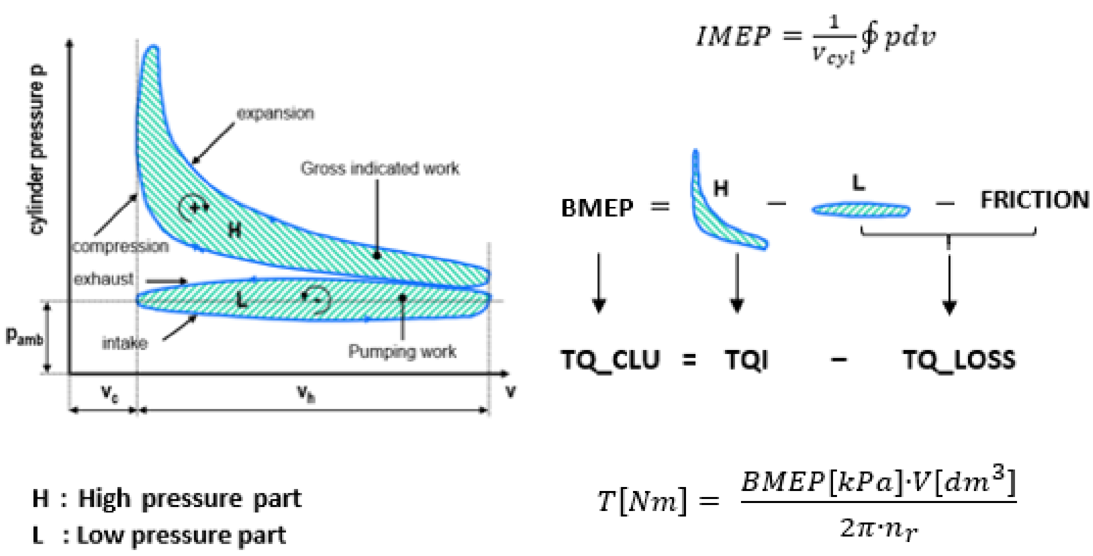
In the next page is shown an indicated diagram of which is considered unnecessary give a detailed description: just the IMEP is expressed, which is the work of the thermodynamic indicated cycle for unit of displacement.

Notice that:

V_{cyl} $[dm^3]$ is the displacement of the engine;

p $[kPa]$ is the pressure in the combustion chamber, point by point, during the working cycle;

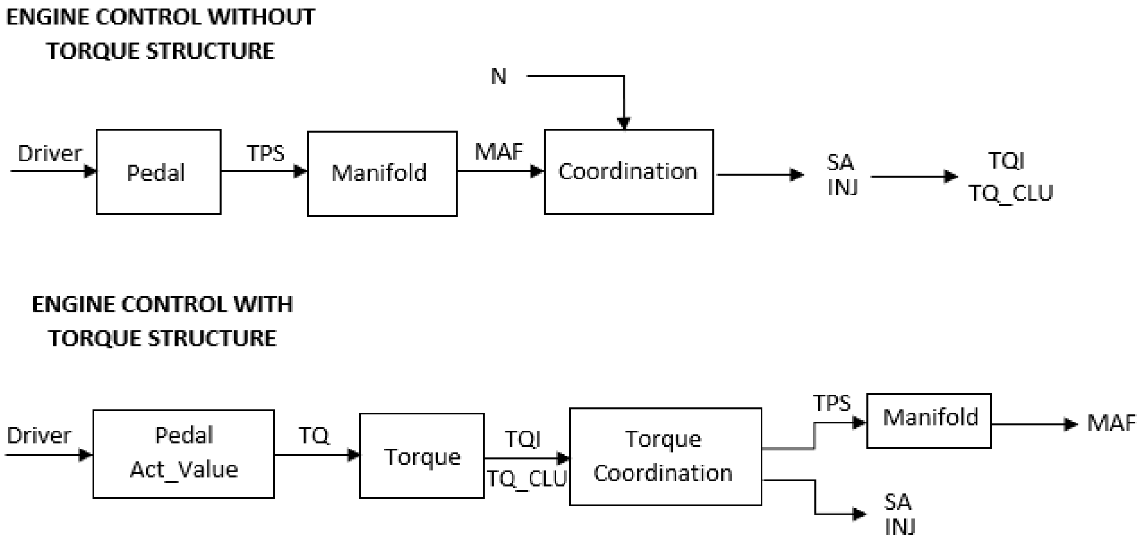
$dv\ [dm^3]$ is the infinitesimal volume swept by the cylinder between the BDC and TDC.



4.2 Torque-based System

The original control strategy was based on the direct relationship between the throttle body and the driver action on the handle. The other interventions (warmup, traction control, speed limiter, etc.) contributed to the definitive calculation of the actuations in the form of throttle opening offset; the ignition angle was chosen with the aim of guarantee the maximum efficiencies during the combustion phase. This did not ensure a precise and immediate control of torque.

With the need to control the engine in a precise manner, it was essential to implement the Torque-based system and introduce the electronically controlled throttle (DBW, drive-by-wire). In this way it was possible to separate the driver's torque request from the actuation itself. The driver request, while remaining the main target of the control strategy, is only one of the numerous torque request that reach the engine at each cycle; other functions contribute to the best choice of actuations cycle by cycle.

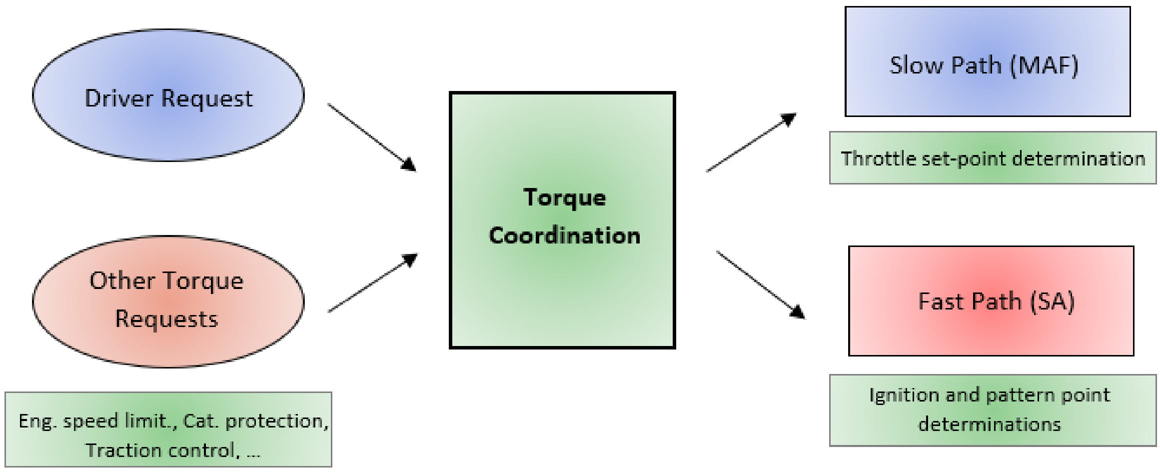


So the control system is a “Torque Coordinator” and its control levers are ignition and throttle position.

4.3 Torque Coordination

An instantaneous adjustment of the torque (so-called "fast path”) is represented by the variation of SA (the torque regulation through variation of the intake air mass is considered "slow path” as the dynamics of filling and emptying the intake manifolds are not instantaneous.). The value of the SA can be changed from one combustion cycle to the other and heavily affects the torque. In the ECU there is the SA_EFF_Curve, which allows the system to establish the ignition angle necessary to guarantee a certain efficiency. This term refers to the ratio between the maximum producible torque, obtained by applying the optimum spark advance, and the produced torque, by implementing an advance different from the optimal one.

It is clear that the management of the actuations necessary for the correct functioning of the engine is assigned to the ECU. The tasks of the ECU, or better, of the control system, are to guarantee that the driver request is transformed into commands to the engine actuators, to monitor the operation of the various sensors and to intervene in the event of preserving the driver safety or engine integrity. It regulates all the engine functions in order to control emissions and consumption, too.



So at the base of the functioning of the ECU there are the principles of automation, automatic controls and computer programming.

4.4 General example of vehicle torque chain

The supplier of control units and control software for the DUCATI engine analyzed is one of the leading companies in the electronics industry.

The control unit uses data from the sensors to monitor the state of the engine in all its parts with very short time intervals (in millisecond) to allow real-time operation of the control.

The first step towards defining the requests that the ECU needs to transfer to the engine is to establish what the driver is asking. The software receives the information from the potentiometers placed on the throttle handle and converts it into a torque request through a calibration that depends on the engine revolutions, the pedal map.

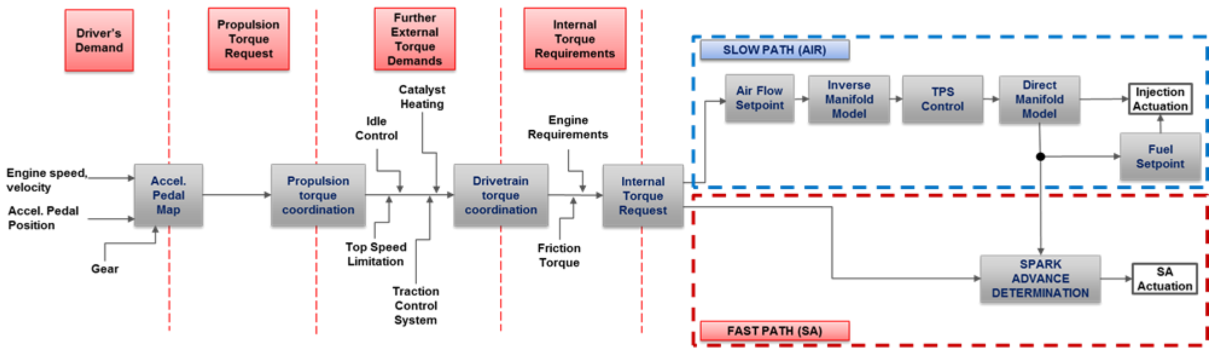


Figure 50 - Example of vehicle torque chain

To the driver request are submitted other external torque requests to be coordinated (idle control, traction control, cat-heating, etc.) from which the torque at the clutch is obtained. Frictions are added to this value and the internal torque is obtained. The internal torque is managed by two control levers: the relative load of air that demand a fuel setpoint (slow path), and a SA value is calculated starting from the value of reference spark advance degraded to obtain the desired torque and limited by the one coming from the base maps (fast path).

4.5 Inverse torque model for simulation

The model realized in this Thesis was developed by doing reverse engineering. In fact, as reported in the diagram below, we are gone backwards the model described before.

We started from air-path, SA maps and the λ set-point, whit given rpm and load values. Once the efficiency data of SA and λ have been obtained, these have been multiplied to the reference internal torque value to obtain a gross internal torque. Once this value has been reduced by friction (in the diagram shown as added but notice that they are negative values), the actual external torque (effective torque at clutch) is obtained.

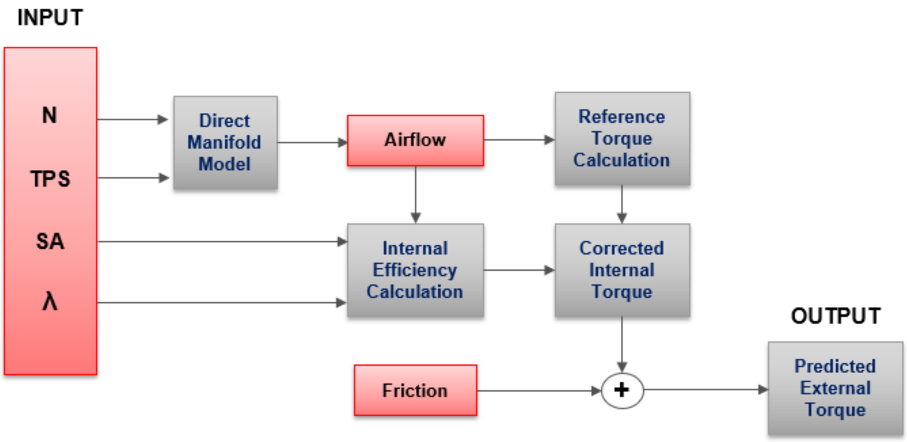


Figure 51 - Inverse simplified torque model

4.6 Model Simulation and "Cases of Study"

The simulation of the model was carried out by writing and executing MATLAB codes: it has been an extremely important tool for the development of the project. In fact, it was possible to evaluate the model's response to the inputs given, such as the data collected during the bench activity and final calibration maps.

To make the model "universal", that is to make it valid for other engines different from the one analyzed, some of the parts of the script are represented by modifiable parameters.

By developing such a model it could be possible to control the work of the supplier who, based on data provided by DUCATI during the calibration campaign, realizes the final calibration to be enabled on vehicle.

Obviously, a good simulation requires the model validation. We must be sure that the virtual outputs are the same as the physical we will have. To do this, the coefficient of determination R^2 was used[4].

4.6.1 Case 1: SA_BAS with SA_EFF_Curve and TQFR MOTORING

The first simulation was made by launching the torque model with the values of SA_BAS which is the best spark advance in terms of engine functionality (consumption, emissions, etc.) not to be confused with SA_REF (MBT) and SA_OPT which is, in a certain manner, the compromise between functionality and maximum torque.

What is obtained are macroscopic errors on friction with unacceptable values of R^2:

[4] The coefficient of determination R^2 is a proportion between the variability of the measured data and the correctness of the statistical model used:

$R^2 = \frac{ESS}{TSS} = 1 - \frac{RSS}{TSS}$ where:

$ESS = \sum_{i=1}^{n}(\hat{y}_i - \bar{y})^2$ is the deviance explained by the model (*Explained Sum of Squares*)

$TSS = \sum_{i=1}^{n}(y_i - \bar{y})^2$ is the total deviance (*Total Sum of Squares*)

$RSS = \sum_{i=1}^{n}(y_i - \hat{y}_i)^2$ is the residual deviance (*Residual Sum of Squares*)

- y_i are the data observed
- $\bar{y}$ are their average
- $\hat{y}_i$ are the data estimated from the model obtained from regression

R^2 varies between 0 and 1: when 0 the model does not explain the data at all; when it is 1 the model explains the data perfectly.

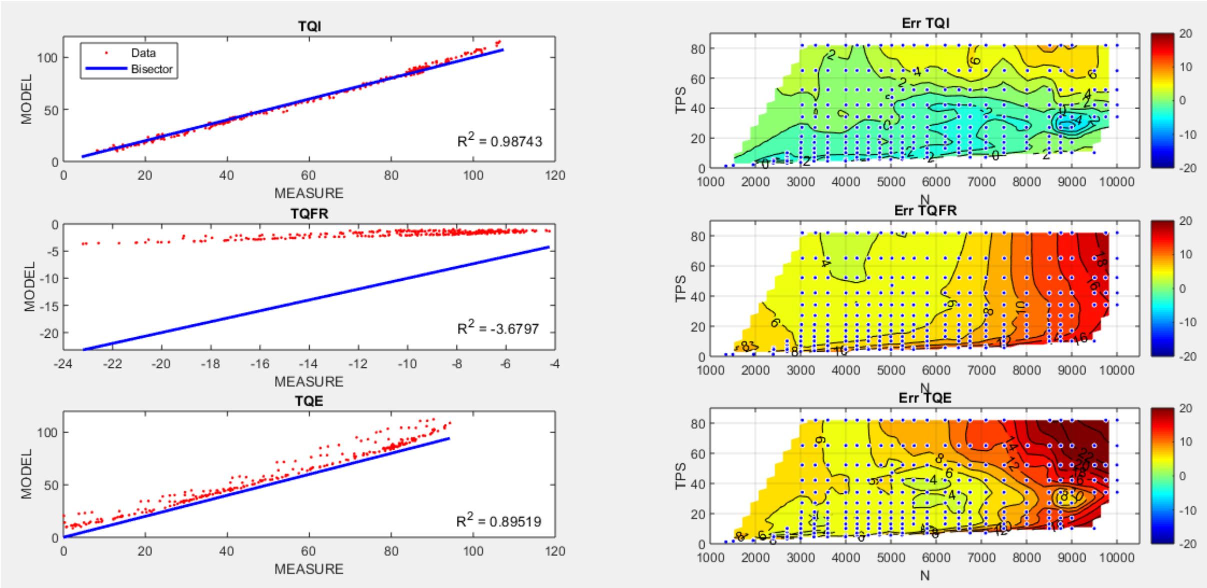


Figure 52 - Case 1: SA_BAS w/ SA_EFF_Curve and TQFR MOTORING

4.6.2 Case 2: SA_BAS with SA_EFF_Curve and TQFR FIRING

In the second simulation, having ascertained that the problem was on friction, the model is relaunched using the friction map in FIRING. The results obtained are much better:

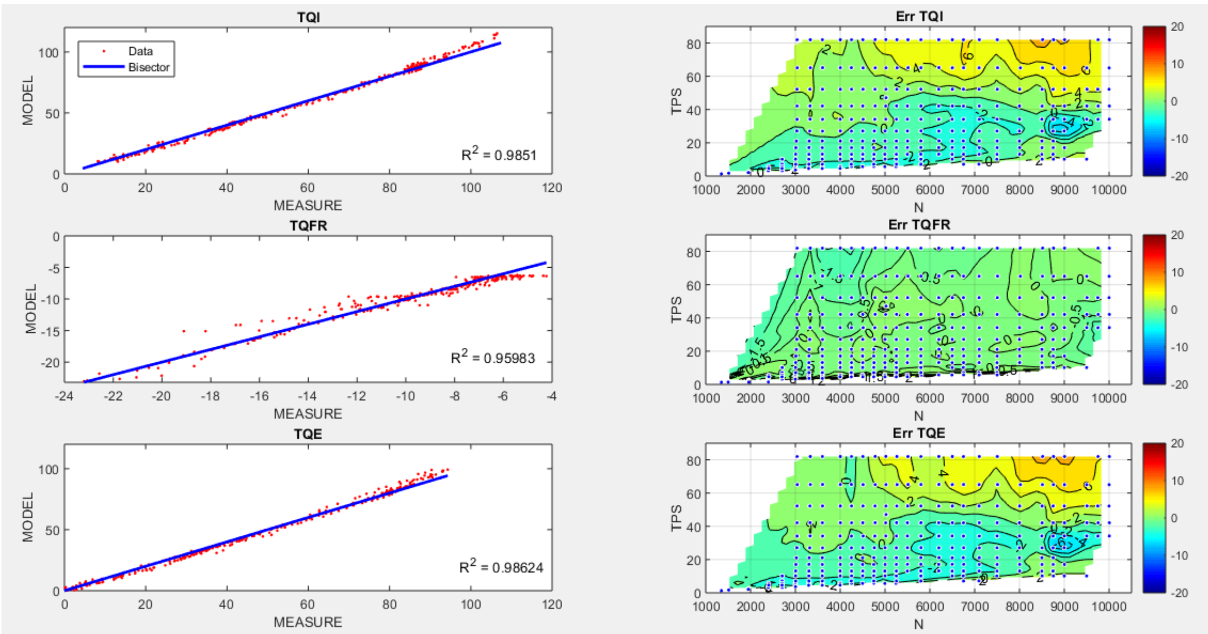


Figure 53 - Case 2: SA_BAS w/ SA_EFF_Curve and TQFR FIRING

What happened? Friction represents the amount of energy transferred to the piston that is not available on the shaft for transmission to the wheels. They are divided into pumping losses and mechanical losses.

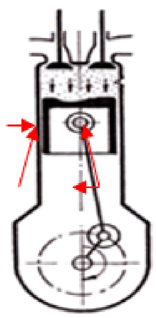
FRICTION

PUMPING LOSSES: They represent the net work of the piston on the gas present in the cylinder during the intake and discharge phases.

MECHANICAL LOSSES: Amount of energy spent to overcome the dissipations that arise from the relative movements of the components.

Since on the bench the measure of friction is done by bringing the engine into cut-off condition, i.e. removing for a certain time the injection to the cylinders, there is no combustion (MOTORING condition) and then sensitivity on the mechanical friction is lost.

In FIRING condition (with combustion) the peak pressures in the chamber are greater and therefore the friction between piston segments and cylinder is much higher:



Friction MOTORING << Friction FIRING

4.6.3 Case 3: SA_SWEEP with SA_EFF_Curve and TQFR FIRING

For this simulation, the model was launched with the values deriving from the test of the dynamic SA sweep.

NOTE: The graphic did not show the contour plot used to display the model errors on the quoted plane as done for the SA_BAS data: it would not have sense as the dots are no longer precisely engine operating points but are continuous sweep.

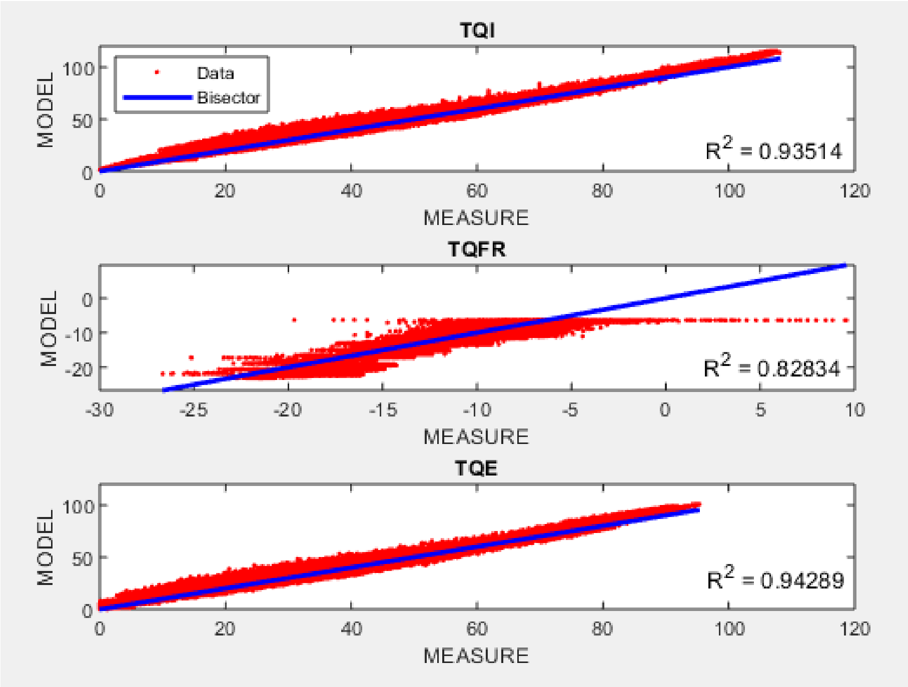


Figure 54 - Case 3: SA_SWEEP w/ SA_EFF_Curve and TQFR FIRING

We found, particularly looking at the results on the external torque TQE, a more or less considerable dispersion of the data produced by the model respect the linear regression.

4.6.4 Case 4: SA_SWEEP with SA_EFF_Map and TQFR FIRING

To avoid the problem, we took into consideration the SA efficiency map instead of the curve with a discreet improvement of the data produced by the model: always looking at the results of the TQE, the coefficient R^2 has in fact passed from 0.94 to 0.96:

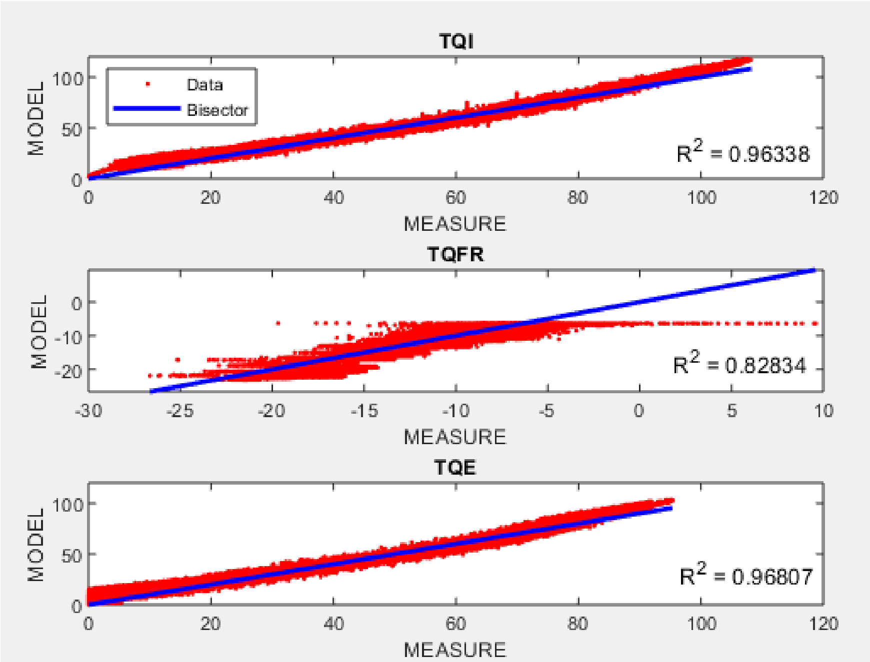


Figure 55 - Case 4: SA_SWEEP w/ SA_EFF_Map and TQFR FIRING

Some engine operating points are analyzed in detail. In particular, for the comparison between the results produced using the SA_EFF_Curve and those produced using the SA_EFF_Map. The points under investigation are summarized in the following table:

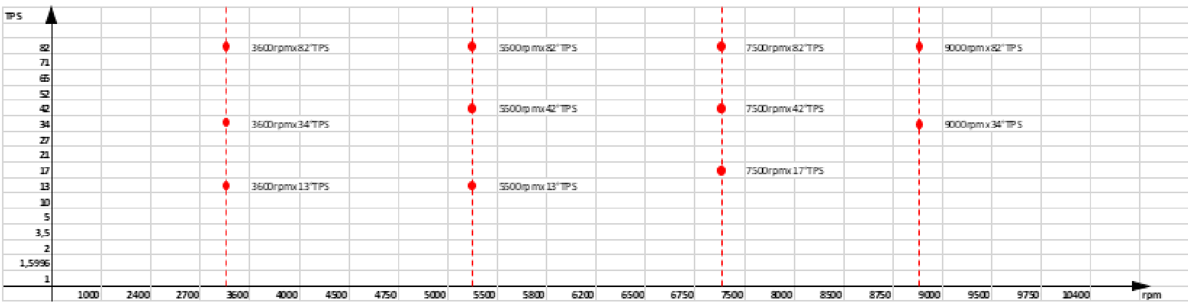


Figure 56 - Case 4: table of engine points analyzed in detail

Comparing the TQE Error produced by implementing the curve or the map of SA_EFF, it is clear that the map has a better potential than the curve in representing the engine behaviour.

Below are the graphs showing the results obtained and, after these, again the summary table of the analysis considerations reached.

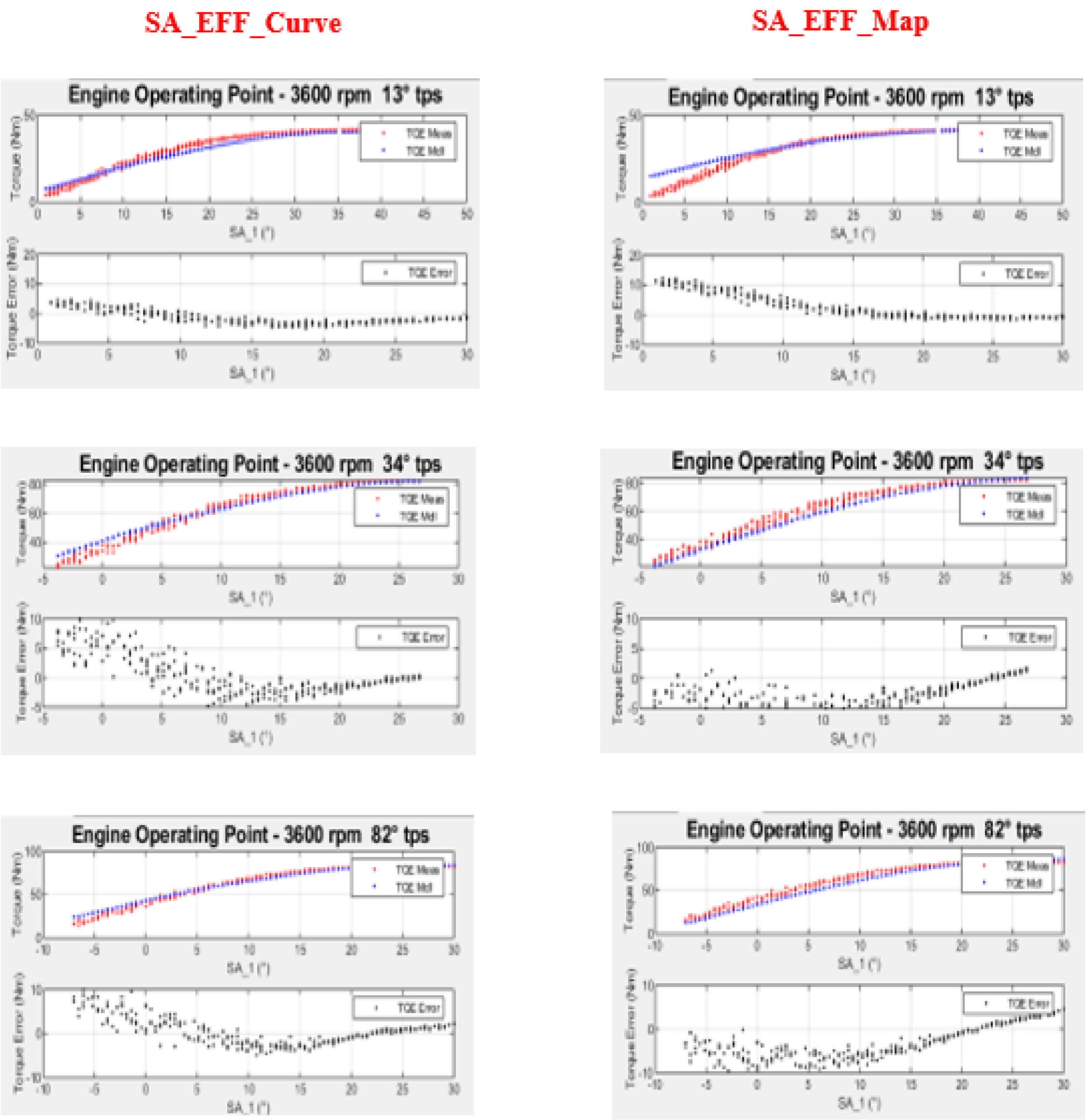


Figure 57 - Curve and Map Efficiency @ 3600 rpm x 13/34/82 °TPS

SA_EFF_Curve **SA_EFF_Map**

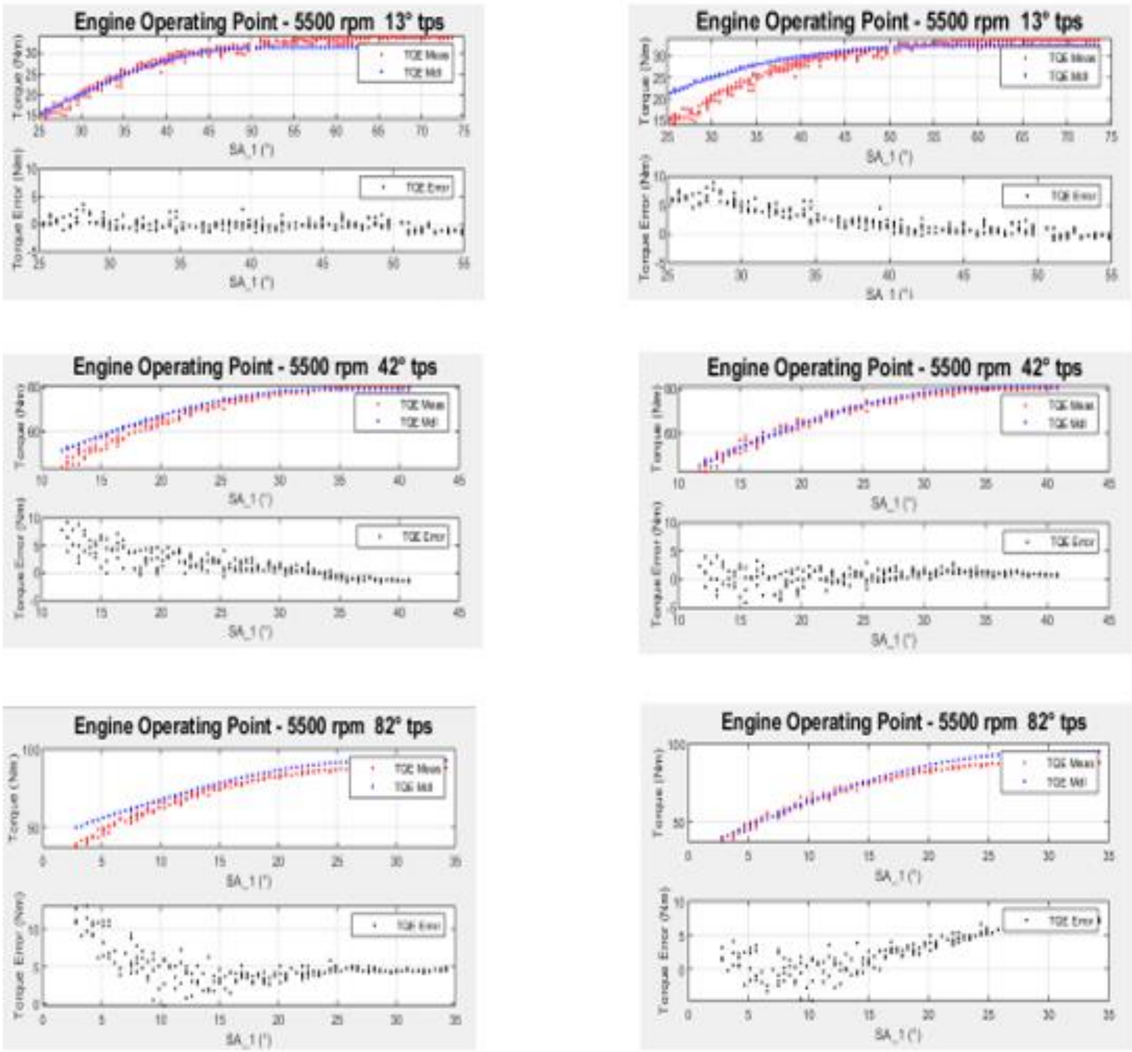


Figure 58 - Curve and Map Efficiency @ 5500 rpm x 13/42/82 °TPS

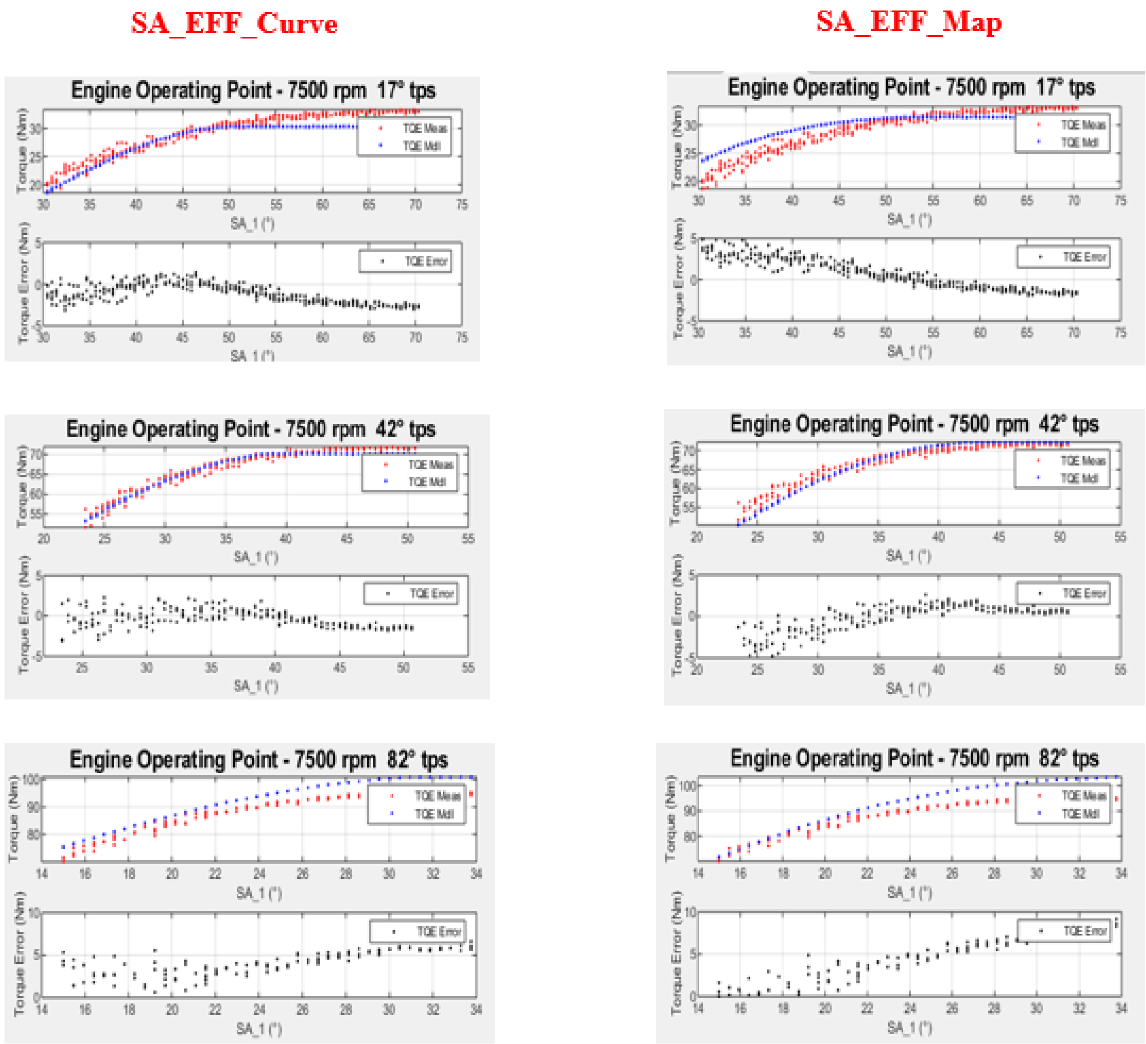


Figure 59 - Curve and Map Efficiency @ 7500 rpm x 17/42/82 °TPS

SA_EFF_Curve **SA_EFF_Map**

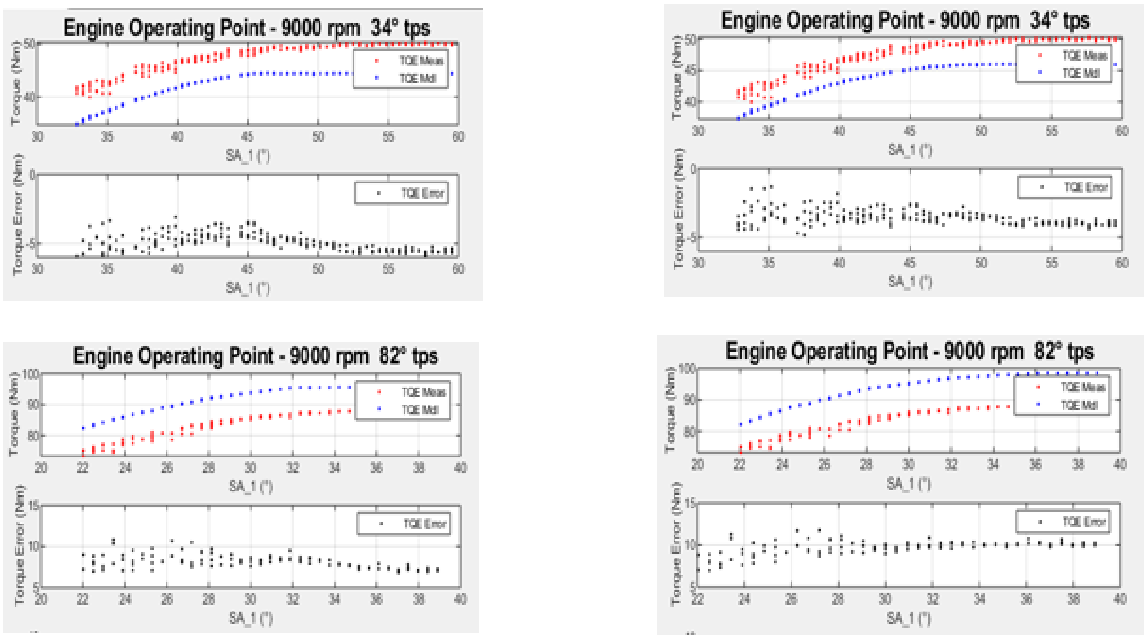


Figure 60 - Curve and Map Efficiency @ 9000 rpm x 34/82 °TPS

4.6.5 Case 5: SA_BAS with SA_EFF_Map and TQFR FIRING

The last case analyzed was the one that takes the data coming from the SA_BAS file but we tried in this case to implement the SA_EFF map instead of the curve. The simulation was made to see if there could be any improvement on resultant TQE.

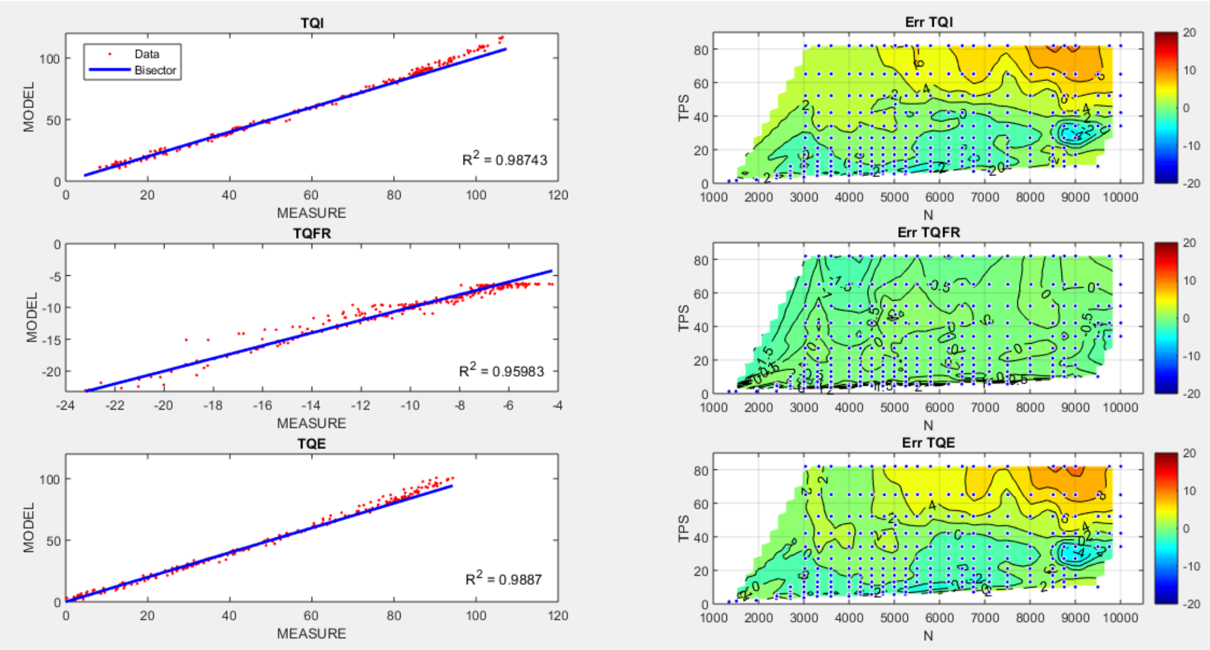


Figure 61 - Case 5: SA_BAS w/ SA_EFF_Map and TQFR FIRING

Unlike what happened for SA_SWEEP, in this case no significant improvements are noted. The explanation for this is in the fact that, during the continuous sweep test, a whole range of spark advances is covered, working with very degraded SA and almost zero torque; the area of SA_BAS is instead limited to a high-torque zone in which using the curve or the map of SA_EFF makes no great difference.

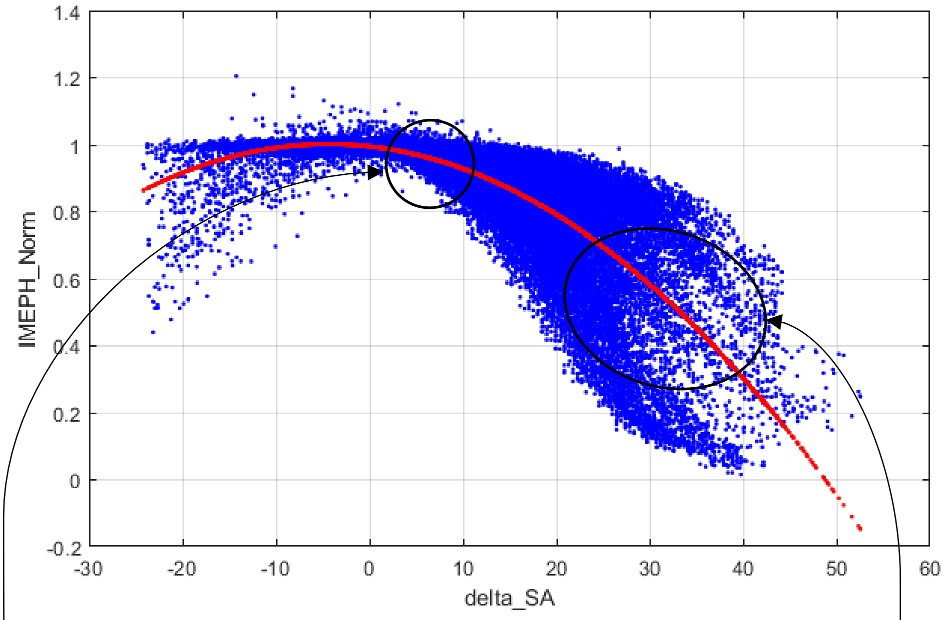


Figure 62 - Example of SA_EFF_Curve and Map (ΔSA/Torque)

For the simulations in which SA_BAS was used, both the map and the SA_EFF curve are acceptable

For the simulations with SA_SWEEP they were reached SA also in this area where the SA_EFF curve is not very accurate → Major error on modeled torque

5. Conclusions

The work carried out in these months of internship in DUCATI gave me the opportunity to deepen my knowledge on the calibration of internal combustion engines, on the sensors of a tests room, on the PCs that interface and control the cell, but above all they taught me how to deal with the technical problems that arise when an engine is on the bench.

The part of post-processing data through specific tools allowed me to learn how to produce reports adequately, to understand how torque structure works, and to improve my judgment and analysis data skills.

At the end of the collaboration the established objectives have been reached.

The project puts under the hypothesis of avoiding some tests on the bench, tests that currently subject the engine to strong thermal stress with all the problems that result and that require many hours of activity. The possibility of saving on working hours and costs allows solving data acquisition "vacuums" and problems that sometimes prevented completion of the tests themselves.

The Torque-based model realized was undoubtedly the core of the work of these months.

Based on a MATLAB code that reads .xls files in input and gives the external torque in output, it's easy to use. The greatest result is the model ability to determine the behavior and the response of the engine control unit and the added value is to have editable parameters that make the model universal and applicable to different cases of study. The calculation of the coefficient of determination R^2 allows an immediate evaluation of the error produced by the model and the graphs help to make the model "user friendly". This offers the possibility of a quick check and a validation of the work that comes back from the supplier of the control unit that will be implemented on the vehicle.

There could be margins for improvement on the accuracy of the results provided by the model, and these are to be found in the setting up of some maps on which the model develops (SA_REF, SA_EFF and TQFR determination), but overall the result of an R^2 = 0.98 affirmed that it is a robust model.

Definitively this Thesis project is proposed to support the calibration and analysis of data produced during the bench activity.

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