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# DEVELOPMENT OF CONTINUOUSLY VARIABLE INTAKE MANIFOLD FOR FORMULA STUDENT RACING ENGINE

**ABSTRACT:** After several years of research and development of Formula Student's air mass flow restricted racing engine at the Internal Combustion Engines Department of the Faculty of Mechanical Engineering, University of Belgrade, the design process of a new intake manifold for the 2014 competition season was set off. Through several seasons, the intake manifolds of the YAMAHA YZF-R6 high performance engine evolved from a dual volume, into a single volume concept and finally to the continuously variable intake manifold (CVIM) design. Comparative analysis of data obtained during in-laboratory engine testing and data logged from ECU during the races gave some guidelines in the design of CVIM. The main goal of this research is increasing the number of engine operating points with resonant supercharging. The Ricardo WAVE engine mathematical model is improved and particular attention is dedicated to the approximation of the adopted CVIM concept using Helmholtz Resonance Theory. This paper describes the correlation between optimal intake runner length and manifold volume over engine speed at wide open throttle as well as their influence on volumetric efficiency and engine effective parameters.

**KEYWORDS:** engine testing, variable intake manifold, optimization, resonant supercharging, Formula Student.

## INTRODUCTION

Formula Student (FS) is a global student engineering competition, whose rules are prescribed by the Society of Automotive Engineers (Formula SAE) [1] and individual amendments to the rules and by the other competition organizers (Formula ATA [2], FSA [3], FSG [4], FSH [5], etc.). Students at different study levels, B.Sc., M.Sc. and Ph.D., at the Internal Combustion Engines Department (ICED), Faculty of Mechanical Engineering, University of Belgrade are involved in this contest taking a part in the development, production and testing of powertrain and electronic systems of the racing vehicle. The main goal of this paper is presenting results of the research dedicated to continuously variable geometry intake manifold (CVIM), engine test results and providing guidelines for the design process.

The applied technical solutions that will be used in the 2014 competition season are obtained taking into account the experience gained in previous years of development and research aiming at performance increases in terms of engine effective torque over certain RPM. Using Ricardo WAVE simulation environment, engine mathematical

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model is constantly improved through an iterative process of model validation and calibration coupled with in-house developed MATLAB post-processing and decision-making codes.

The design process of complex engine's systems such as CVIM is divided into several stages. It is very important to use high-quality data obtained during in-laboratory engine testing based on the predefined engine test plan in order to achieve the best possible engine mathematical model matching. Also, the availability of the test equipment, "know how" resources, production technology and affordable budget are key features. Considerable attention is paid to logging setup and to the analysis of the data logged during the races. In this manner, important conclusions about the entire powertrain system are drawn. Additionally, this paper will briefly present theoretical bases of resonant supercharging, CVIM concept review in the way of actuation and part of the simulation results which are used for control look-up tables.

# ENGINE TESTING

For the purpose of the Engine Control Unit (ECU) [6] optimal control maps determination, the testing of naturally aspirated air mass flow restricted Yamaha YZF-R6 engine was performed during the competition seasons of 2012 and 2013. The idea of CVIM implementation to the Yamaha's power unit is intended for the 2014 season and all calculations, assumptions and results obtained during testing were used in this sense. The basic characteristics of the test object are given in the Table 1.

Engine intake must be equipped with a 20 mm diameter restrictor section which is in accordance with the FS competition rules. The engine was coupled to a Schenck W130 Eddy current dynamometer with a corresponding control module that enables required stable operating regime.

***Table 1*** Engine and ECU specifications

| | | |
|---|---|---|
| Engine | YAMAHA YZF-R6, four stroke, spark ignition, in-line four cylinder, naturally aspirated, water cooled | |
| Ignition order | 1-2-4-3 | |
| Bore | 67 mm | |
| Stroke | 42.5 mm | |
| Compression ratio | 13.1 [-] | |
| Valves per cylinder | 4 | |
| Restrictor diameter | 20 mm | |
| Competition Season | 2013 | 2014 |
| Runner length (RL) | Optional<br>180 mm, 210 mm and 240 mm | Continuously Variable<br>164 mm to 260 mm |
| Intake plenum volume (V) | Fixed<br>3.4 lit | Continuously Variable<br>4.3 lit to 3.7 lit |
| Engine Control Unit (ECU) | DTA S60 PRO | DTA S80 PRO |
| ECU interface software | DTASwin 63.10 | DTASwin 67.00 |
| Injection | Sequential | |
| Ignition | Sequential | |
| Lambda sensor | Wide band | |
| Communication | Serial, CAN | |

Because of the growing idea to develop a CVIM system, during the design process of the intake manifold for the 2013 competition season and during in-laboratory engine testing, it was very important to collect as much data as possible with different configurations of the intake manifold. Regarding this, the optimization of the advance timing and the injection duration was conducted with three different runner lengths: 180 mm, 210 mm and 240 mm (not including the length of intake line within the cylinder head) and constant plenum volume of 3 liters. The engine operating regimes on which the experiments were conducted are very similar to the ones explained in the previous studies of the author [7, 8]. Because of the inability to avoid detonation at high load and low RPM regimes, and nearly zero value of effective torque at high RPM and no load regimes, some values of control maps are determined based on previous experience.

## Data acquisition

All sensor readings were set in time domain to simultaneous 1k sample per second. Readings from stock engine sensors, which were used as ECU inputs, and some output (control) parameters were available on the high speed ECU's CAN data stream (total of 24 channels within 6 messages). Established CAN communication enabled data-flow of all available ECU's data to the acquisition system based on the National Instruments (NI) PXI hardware with maximum frequency of 50 Hz. Serial communication between the ECU and PC allows real-time adjustment of engine control parameters (advance timing, injection duration and others) and monitoring of sensor readings (engine speed, throttle position, lambda sensor reading, intake air, coolant and oil temperature, intake air and oil pressure, manifold absolute pressure, battery voltage and others). Besides listed engine parameters, by using an open ECU interface, it was possible to adjust PID coefficients of all closed loop corrections and additional look-up tables (engine start fueling, idling, temperature compensations, etc.). The engine test stand was equipped with some inevitable sensors, which provided monitoring of the parameters, like effective brake torque, transmission output shaft angular speed, exhaust gas temperature, intake air temperature, coolant pressure and temperatures, oil pressure and temperatures and A/F ratio. The acquisition of the analog signals was improved by utilizing in-house developed analog input multi-channel amplifier modules [9]. Along with the data acquisition, the NI PXI multifunctional acquisition devices were used for controlling and supervising of the engine test bench subsystems, like intake air conditioning system, engine throttle positioning, oil cooling and fuel supply system, evacuation of exhaust gases and additional engine cooling. Generally, the data acquisition was conducted via several data channels (Serial, CAN, AIO, DIO).

## Data sets and post-processing

Time and funds-limited projects, such as this one, require slightly different approach to engine test bench run procedure and data acquisition. Control maps were optimized at quasi-stationary operating regimes using "n-const" and "M-const" dynamometer modes [10]. Engine testing was conducted with great respect of the procedures defined by the standard ISO 1585 [11] and ISO 15550 [12] with one difference, which is related to the stabilization of the engine operating regime. Because of relatively large number of test regimes (about 150 regimes for each set of runners) time dedicated for exhaust temperature stabilization was minimized. Instead of waiting for the particular operating regime to become satisfactorily stable, it was decided to perform continuous data logging and after that carrying out more advanced post processing and analysis techniques. In this way, it was possible to preserve engine from excessive wear and obtain sufficiently accurate measuring.

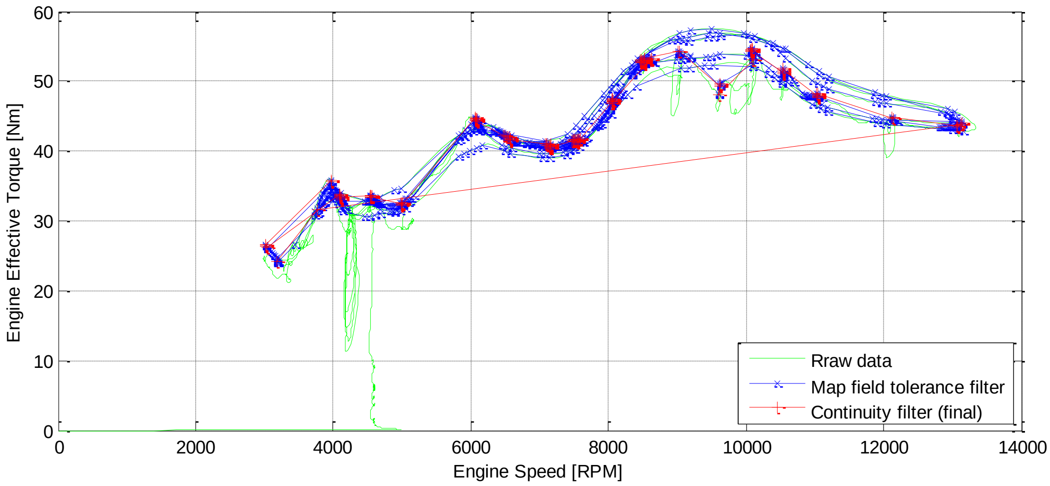


***Figure 1*** Engine Effective Torque [Nm] over Engine Speed [RPM] at TPS=100%, Runner length 210 mm. Data post-processing and applying filters. Test results of 2013 season.

Data sets were acquired by the following test procedure:

- A certain set of intake runners was mounted on the engine;
- All systems of the test bench were checked and the engine was warmed up;
- Dynamometer was set to the required operating mode and a limit for brake torque or RPM was set;
- Engine load was increased slightly to the desired value of throttle opening (throttle and RPM stops are defined by engine control maps). This is done for the whole range of TPS, from 0% to 100%;
- By adjusting dynamometer control parameters, desired engine RPM was reached. Engine speed was varied from 3000 RPM up to 13500 RPM;
- For each pair of TPS and RPM values, ignition advance timing was varied in certain steps to achieve maximum brake torque;
- Injection duration was adjusted in the way of obtaining predefined air to fuel equivalence ratio (AFR).

The same procedure was repeated for all sets of runners. Acquired data sets were recorded independently for each TPS value and for a whole range of RPM. Because of continuous sampling, a large amount of data was obtained and some additional kind of post-processing needed to be applied. Besides moving-average filter, additional ones were generated. Forthcoming analysis was applied only to the data that had been filtered, which refers to the prescribed tolerance of engine speed, throttle position and optimal advance timing. Additionally, one of the conditions was that the filtered data needed to be logged continuously for a certain period of time. In this way, engine testing time was significantly reduced. The engine brake torque data for the engine configuration with 210 mm runner length is used to describe the mentioned methodology and results are shown in Figure 1. All relevant engine data sets were treated in this manner.

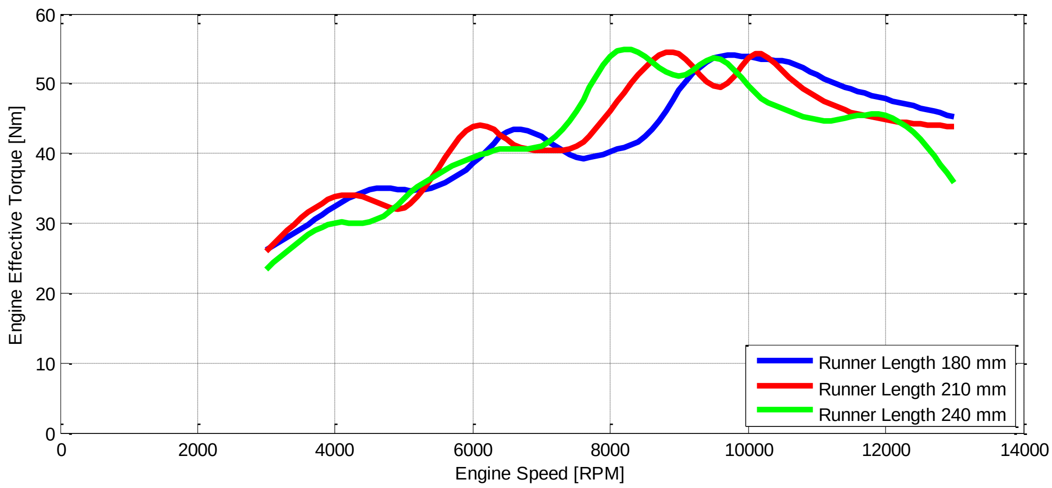


***Figure 2*** Engine Effective Torque [Nm] over Engine Speed [RPM] at TPS=100% for different Runner Lengths, Final data. Test results of 2013 season.

Taking into account the complexity of a racing vehicle, as a whole, and numerous influential factors that affect overall performance, many hours of testing of the entire system have been conducted. With different engine torque curves it is expected the car behavior on the track will be different. Considering different final drive ratios, the tractive effort characteristic was examined in parallel with data analysis obtained during the race. More details about race data log will be shown below. The engine intake manifold in configuration with a 210 mm runner length turned out to be the best option, which was used during the competition.

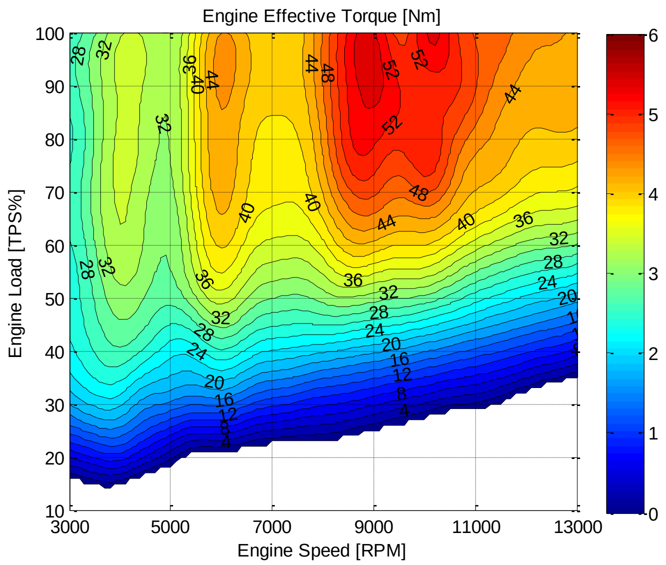


***Figure 3*** Engine Effective Torque [Nm] as a function of Engine Speed [RPM] and Load [TPS%], Runner Length 210 mm.

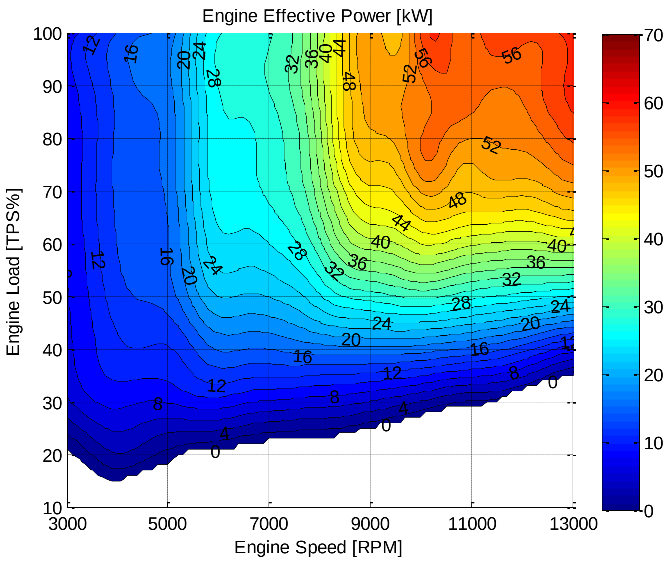


***Figure 4*** Engine Effective Power [kW] as a function of Engine Speed [RPM] and Load [TPS%], Runner Length 210 mm.

Detailed analysis of the obtained data was carried out using self-developed MATLAB codes dedicated to NI LabVIEW *.tdms files processing. The engine universal performance characteristics maps were effectively drawn by implementing the Design and Analysis of Computer Experiments (DACE) [13], a MATLAB approximation toolbox. Some of the experimental results, such as engine torque and power, exhaust gas temperature, air mass flow, manifold absolute pressure, brake specific fuel consumption, ignition timing and injection duration are shown in Figures 3 to 10.

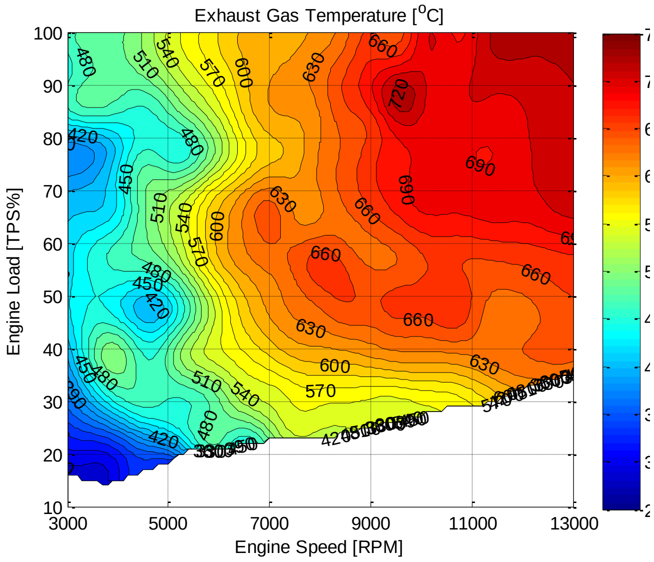


***Figure 5*** Exhaust Gas Temperature [°C] as a function of Engine Speed [RPM] and Load [TPS%], Runner Length 210 mm.

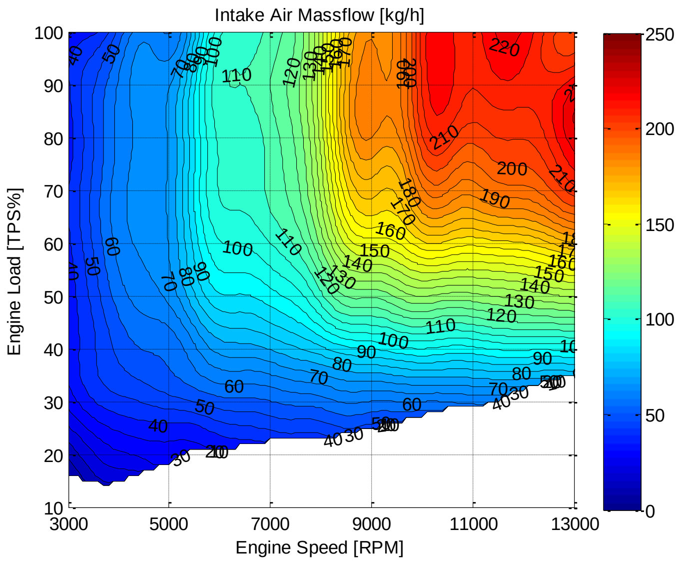


***Figure 6*** Intake Air Mass Flow [kg/h] as a function of Engine Speed [RPM] and Load [TPS%], Runner Length 210 mm.

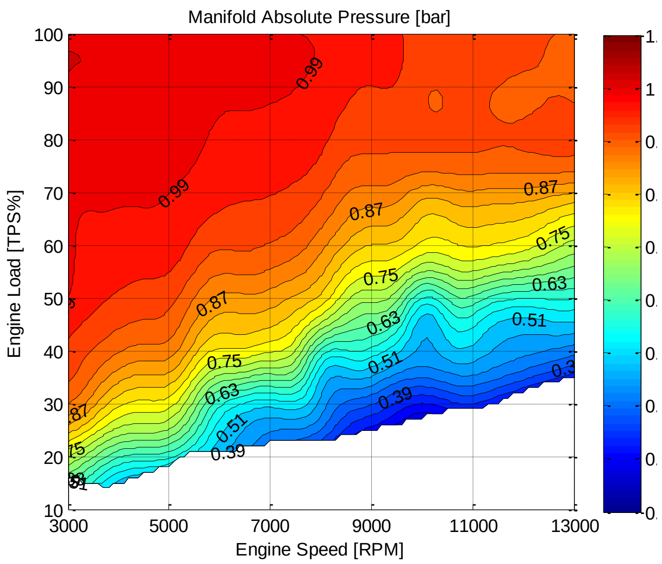


***Figure 7*** Manifold Absolute Pressure [bar] as a function of Engine Speed [RPM] and Load [TPS%], Runner Length 210 mm.

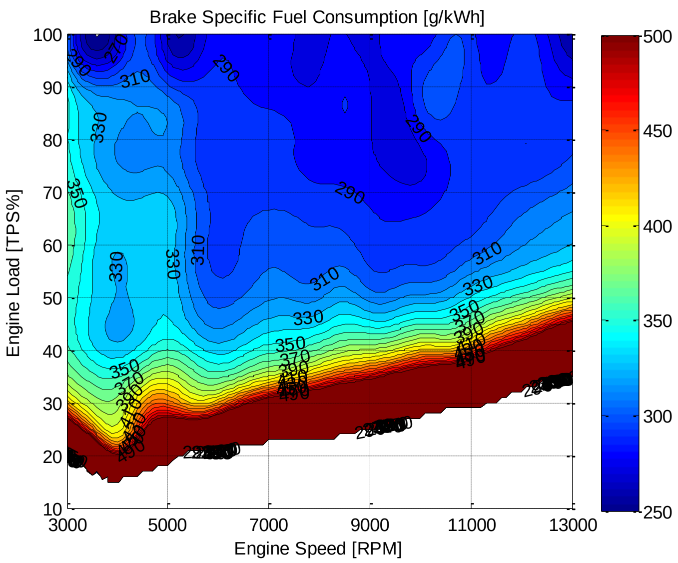


***Figure 8*** Brake Specific Fuel Consumption [g/kWh] as a function of Engine Speed [RPM] and Load [TPS%], Runner Length 210 mm.

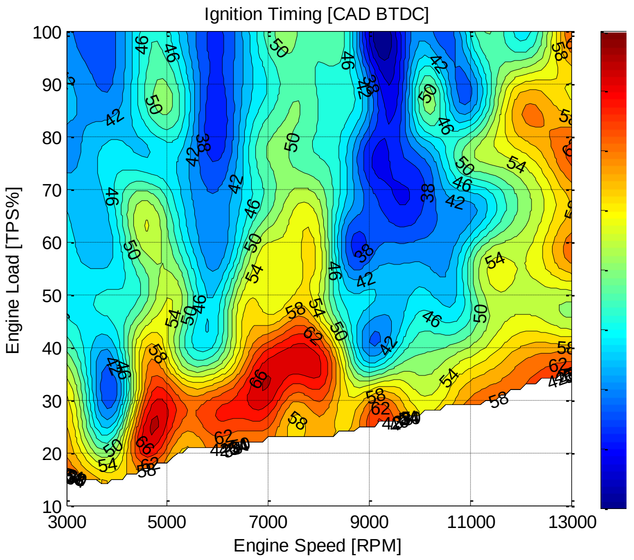


***Figure 9*** Ignition Timing [CAD BTDC] as a function of Engine Speed [RPM] and Load [TPS%], Runner Length 210 mm.

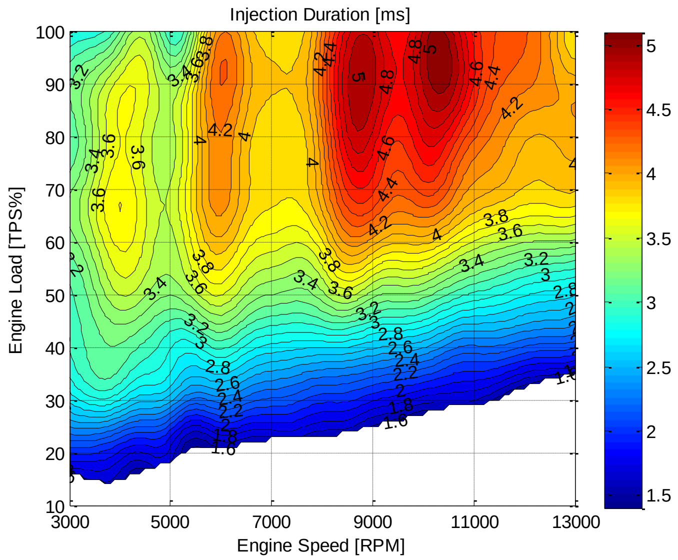


***Figure 10*** Injection Duration [ms] as a function of Engine Speed [RPM] and Load [TPS%], Runner Length 210 mm.

## In-vehicle data logging and analysis

Significant attention was dedicated to in-vehicle data log and analysis of the gathered data. In the first place, the most frequent engine operating regimes needed to be defined because of getting the starting point for the development of the engine mathematical model. Data logging was performed using ECU's built-in log option combined with E-Race analysis software [14]. Simultaneous sampling of 34 different engine and vehicle parameters with a sampling frequency of 10Hz was used. The most important data were the ones related to over-time changes of engine speed, throttle opening, lambda reading, water and oil temperatures, driven and undriven wheel speed, etc. Some of the log results acquired at the competition during the race are shown in Figure 11.

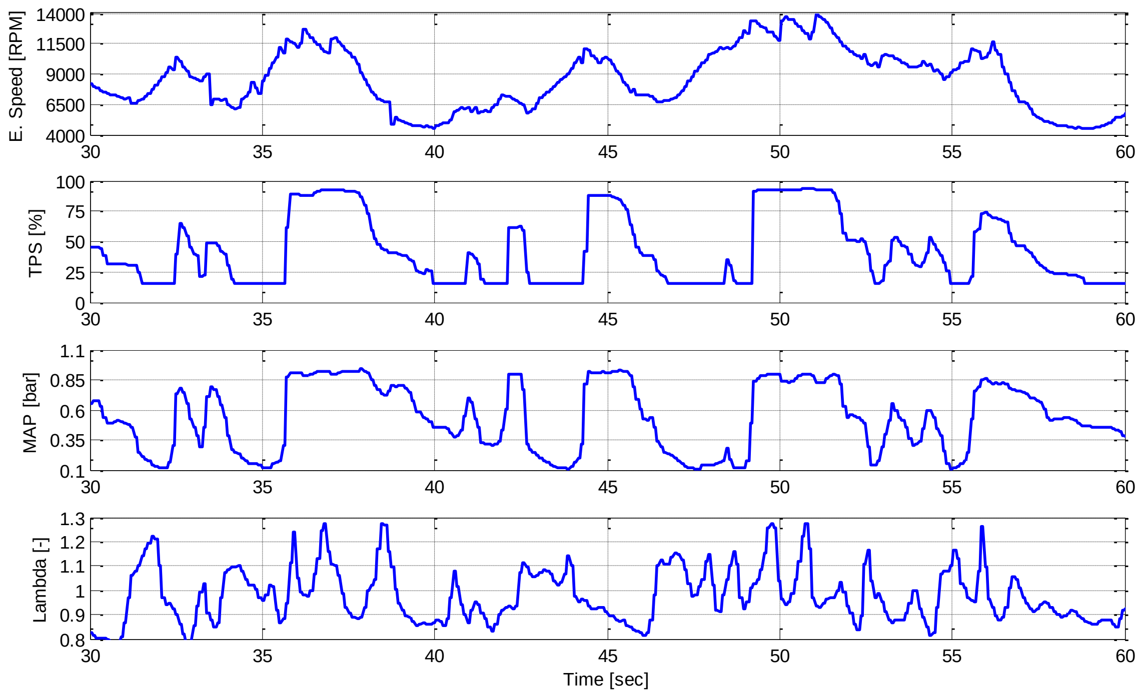


***Figure 11*** Race track data log. Engine Speed [RPM], Throttle Opening [%], Intake Manifold Pressure [bar] and Lambda reading [-] changing over time during the race.

Statistical analysis of the data mentioned provides important guidelines for the forthcoming design and development process. Also, the results of the analysis of the dynamic behavior of the entire system are of great importance. The rate of change of certain parameters, such as engine speed and load, has influenced the CVIM building concept, the way of actuation and further control algorithms. The statistical distribution of engine speed, throttle position and lambda readings during the one complete race, which is 22 km long, is shown in Figure 12.

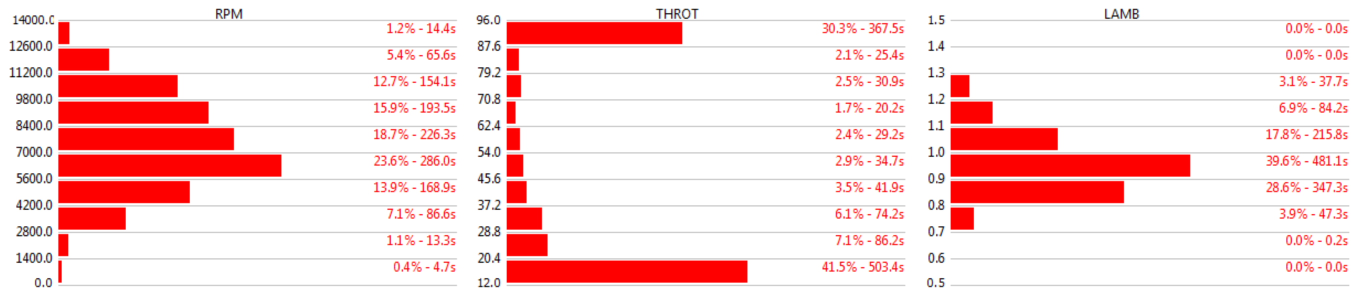


***Figure 12*** Statistical distribution of relevant engine parameters logged during the race.

## Theoretical approach to engine resonant charging

An approach towards defining optimal intake plenum volume and intake runners length so that the effect of resonant charging could be utilized lies on the basis of Helmholtz resonant theory.

The theory is described in a machine system called Helmholtz resonator, which consists of one spherical cavity with a single tube of defined length and diameter. The main idea of a Helmholtz resonator is that with the right frequency of disturbance on the open end of the tube, the complete system gets intoresonance. The equation (1) that defines the frequency needed for getting the system with a known cavity volume, diameter and tube length in resonance is:

$$f_H = \frac{c}{2\pi}\sqrt{\frac{A}{VL}} \tag{1}$$

Where:

- $f_H$ is the resonant frequency [Hz];
- $c$ is the speed of sound [m/s];
- $A$ is the cross-sectional area of the tube [m$^2$];
- $V$ is the static volume of the cavity [m$^3$], and
- $L$ is the length of the tube [m].

The speed of sound is calculated from the ideal gas equation as:

$$c = \sqrt{\kappa R T} \tag{2}$$

Where:

- $\kappa$ is the adiabatic index (equals 1.4 for air [15]);
- $R$ is the ideal gas constant (equals 287.1 J/kgK [15]), and
- $T$ is the temperature in [K].

From given equations, it can be seen that there is no significant influence of the intake manifold pressure change on the resonant charging via this approach. Resonant frequency $f_H$, is the frequency of intake valves opening. As known, the frequency of camshaft rotation of a four-stroke engine is equal to half of the crankshaft rotation frequency. Intake air temperature influence was considered taking into account readings from sensors that were installed on the laboratory installation during engine testing. The readings were in the range of 30°C to 50°C. It was agreed that it would be similar to on-track conditions, as races are held in the summer time. Temperature variations affect the calculated speed of sound by no more than 3%, so the complete influence of the temperature change is declared as non-important for the following analysis, noting that future control algorithms should have an integrated temperature compensation. Intake air temperature was only considered for determining injection time compensation, without which the engine would have bad management.

The cross-sectional area of the tube was kept the same as in the previous year (diameter of 40mm), as there was an intention to avoid the possible effect of tube diameter change on the engine effective parameters. The main goal was to evaluate the influence of intake plenum volume and intake runners length variations on the engine effective performance.

The approach to defining needed intake plenum volume (V) and runner length (RL) lied in the development of a MATLAB code for the assumption of resonant fields in a V versus RL map for given engine speed, which was the baseline of the engine work cycle simulation. From data acquired earlier, it was possible to identify the engine working regimes on which the resonant charging phenomenon was utilized. As the tests were performed with three different sets of runner lengths, the resonant charging phenomenon was identified at 9400, 8700 and 8100 RPM, respectively. The next step was to analyze these regimes using the Helmholtz resonant theory equation (1).

The differences that were seen were discussed as the difference between the physical design of the Helmholtz resonator and the intake manifold for a four-cylinder engine.

The firstly mentioned is one-end open system, with spherical volume and one tube, and the other is an open thermodynamic system with one input tube (the intake restrictor section) and four output tubes (leading to the engine cylinders).

Due to the high intake air velocity (the nominal air mass flow was defined as 240 kg/h, which corresponds to 13500 RPM and WOT, after which the restrictor chokes), the correlation was defined so that there must be reduction in the in-cavity air volume that is acting as one defined in eq.(1). The trend of this volume reduction is defined for the regimes mentioned, and a resemblance was noted with air-mass flow through the intake manifold. A new variable, $\Delta V_{dyn}$ was integrated into an algorithm, so the work could carry on defining resonant fields across the engine work field.

The input volume of the intake plenum was set in the range from 0.1 $dm^3$ to 6 $dm^3$, with steps of 0.1 $dm^3$, from which the needed intake runners length for resonant charging have been identified, as seen in Figure 13:

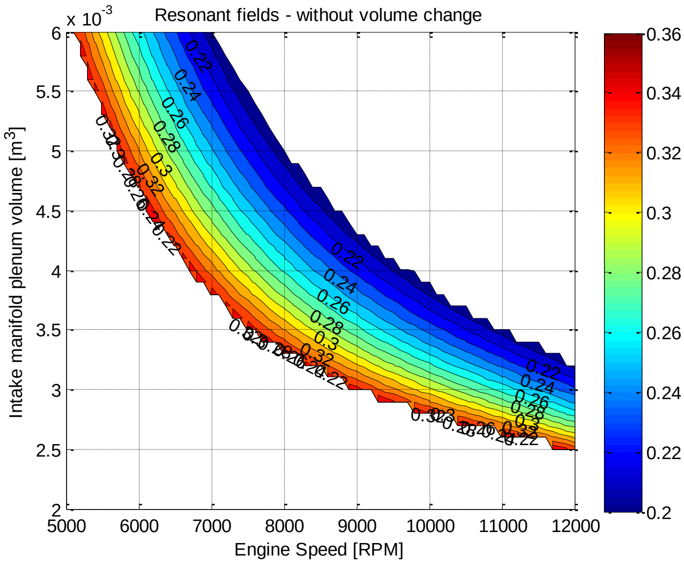


***Figure 13*** Calculated Runner Length [m] as a function of Engine Speed [RPM] and Intake manifold volume [$m^3$] without volume change that is present in the physical model of CVIM.

The next objective was to define the approach behind the active change of intake runners length, for race track use. The principle is set to be telescopic tubes which would change their length inside the intake plenum volume, as any other principle would produce a need for grave reconstruction of the existing intake manifold concept, with which the benchmark would be lost.

The main problem with the telescopic tubes approach is that when the intake runners extend, the intake plenum volume decreases. As shown in equation (1), the cavity volume and the tube length have the same influence on the resonance frequency.

The effect of the intake manifold geometry change with the change of intake runners length is that the resonant fields, shown in Figure 14, are narrower than those which are calculated with constant volume, as shown in the Figure 13. The physical meaning of the noted problem is that it is needed to elongate intake runners more than it would be needed if the volume was kept constant.

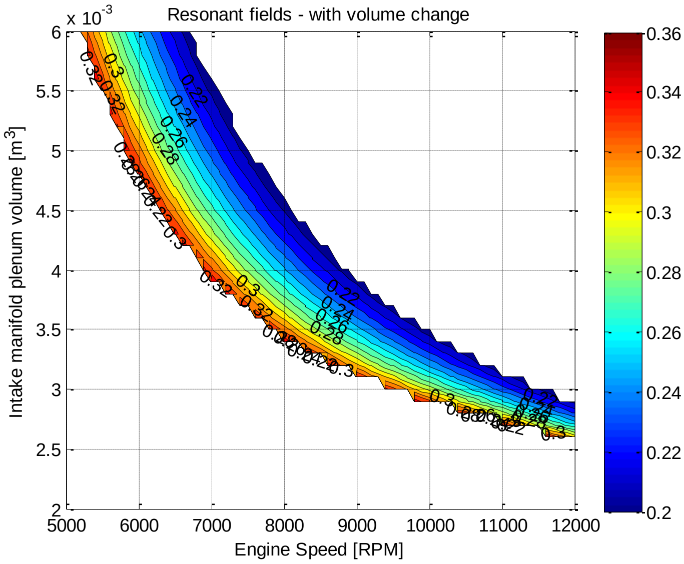


***Figure 14*** Calculated Runner Length [m] as a function of Engine Speed [RPM] and Intake manifold volume [$m^3$] with volume change that is present in the physical model of CVIM

The baseline plenum volume for the designed CVIM was set to the level of half of the maximal runners length elongation (stroke), and it was set at 4 $dm^3$. It was set to this value with the idea of having resonant charging in the widest area shown in the Figure 14, or in the as widest as possible range of engine speed. Runners length was adopted using earlier discussion to vary from 160 to 260 mm, but during the design process the variable length was reduced to 96mm in total, so the final runners length was in the range of 164mm to 260mm.

## Engine mathematical model

Due to the impossibility of putting the engine on a test bench, it was necessary to perform an advanced engine mathematical model simulation. The mathematical model was set in Ricardo WAVE simulation software. The complete analysis was done for Wide Open Throttle (WOT). Test-bench results were used to calibrate the model of the engine from the previous season. That was a fundamental step for further design and optimization of the CVIM for season 2014. The model calibration was performed for various parameters, especially the engine effective torque, advance timing, air mass flow, A/F ratio, etc.

The first step in the model setting up was mapping of the whole geometry of the intake and exhaust system, such as runner lengths and diameters, volume of the plenum, length and diameters of the restrictor section, the position of the injectors, types of materials used, wall thicknesses, heat transfer coefficients etc. Internal engine geometry such as bore, stroke, connecting rod length, wrist-pin offset and the compression ratio were taken from the manufacturer's specifications [16]. Initial conditions such as exhaust gas, intake air and wall temperatures should be entered as some reasonable values. For the purpose of this project, the values were taken as recommended from the WAVE Knowledge Center [17]. Special attention was given to the plenum geometry, which was meshed from CAD model using the WaveMesher software.

Because the engine was not indicated during testing, particular attention was given to the Friction Mean Effective Pressure (FMEP) assessment which was calculated with an equation which approximates total friction correlation [18]. The equation is defined by two constants and mean piston speed, i.e. it correlates FMEP and the engine speed linearly. This was proved to be just enough accurate assumption for the purposes of development of the model.

The equation states:

$$FMEP = a + b \cdot Sp \tag{3}$$

Where $a$ and $b$ are friction correlation constants dependent on the type of the engine and $Sp$ is mean piston speed.
The engine heat-release model, i.e. its parameters such as BDUR, CA50, and WEXP were determined using a Single Wiebe model. It was the most appropriate approach to the problem, considering possible options. The equation (4), which represents the Wiebe function [19], is given by:

$$W = 1 - e^{-AWI \cdot \left(\frac{\Delta\Theta}{BDUR}\right)^{WEXP+1}} \tag{4}$$

Where:

- $W$ is cumulative mass fraction burned [-];
- $\Delta\Theta$ is crank degrees past start of combustion [CAD];
- $BDUR$ is user-entered 10-90 percent burn duration [CAD];
- $WEXP$ is user-entered Wiebe exponent [-];
- $AWI$ is internally calculated parameter to allow BDUR to cover the 10-90 percent range [-];
- $CA50$ is user-entered 50 percent burn location [CAD ATDC].

Values for BDUR and WEXP were adopted from Blair's findings [20]. The WAVE inputs for Wiebe function operate exclusively – either Start of Combustion (SOC), either CA50 [17]. The CA50 values were not considered because of existing data for the Start of Combustion (SOC) in WAVE terms or ignition timing in general. In addition, if a user enters both values at the same time, WAVE prefers SOC values. The CA50 was only considered as a control parameter. Blair's recommendation for BDUR parameter was overruled, since they are too low and complete combustion process would have ended even before TDC leading to irrationally high cycle peak-pressure values. So the next step was to find the appropriate BDUR values across the range of RPM at WOT.

Once all relevant data that was available had been set, whether recorded or recommended, the simulation of the model was initialized. Simulations were performed for the regimes from 4000 to 13500 RPM, which was the engine RPM range used in the prior season. The BDUR parameter was varied from 40 to 95 CAD. As previously said, the calibration of the model was conducted, so the appropriate value of BDUR for recorded torque was aimed through the simulation. The results were exported and then the post-processing of those data was carried out in the MATLAB programming environment.

One of the outputs of the engine model calibration were overlapped torque curves at WOT, which are shown in the **Figure 15**. For the engine speed regimes of interest, a relatively good fit was achieved and this was a starting point for further development of the CVIM.

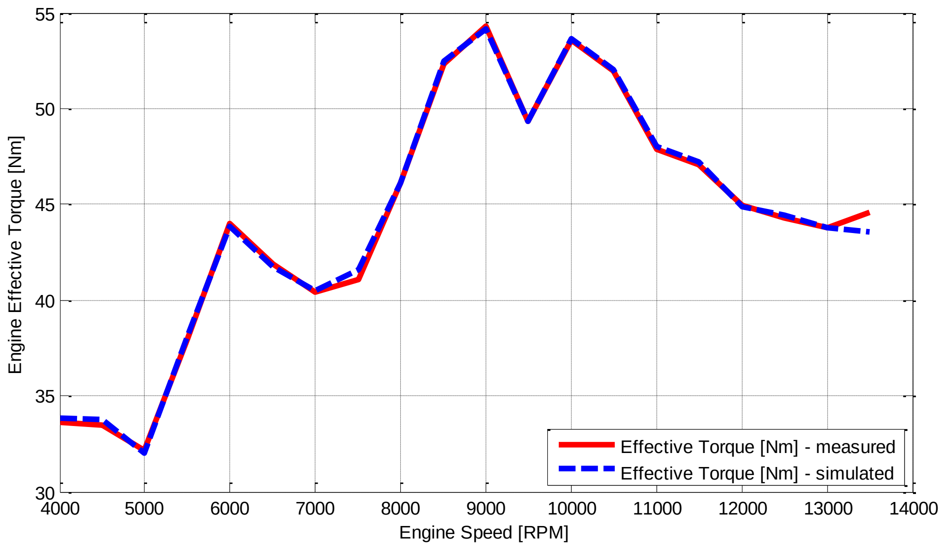


***Figure 15*** Engine Effective Torque [Nm] over Engine Speed [RPM]. Matching of the simulation results and measured values.

After the engine model calibration, the next step was the redesign of the intake and exhaust manifold geometry. The manifolds needed to be optimized in order to improve engine torque characteristics over the wide RPM range and especially between 6000 and 8500 RPM. In this manner, tractive characteristics of the vehicle would be greatly improved.

# Post processing of data obtained from WAVE

The outputs from the Ricardo WAVE analysis were effective engine torque characteristics as a function of engine speed, load and runners length. For every pair of engine speed and throttle position, 20 different runners length values were simulated in steps of 5 mm, with their repercussion on the plenum volume. One of those sets of effective torque data (given for WOT over the engine speed range from 4000 to 13500 RPM) is shown in the Figure 16.

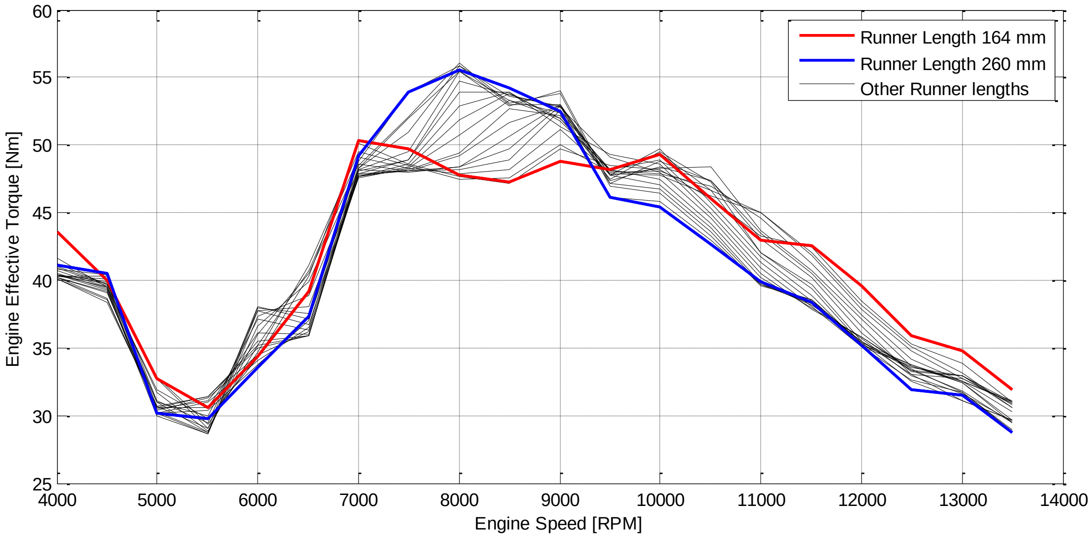


***Figure 16*** Engine effective torque [Nm] as a function of Engine speed [RPM] and for a range of runner length values simulated at WOT.

The approach to determining a control algorithm for runners length variations over engine speed was a multi-step process. The first step was to identify the runners length for the maximum effective torque at the engine speed range previously defined. The next step was to develop a filter so that the change in runners length values over consecutive engine speeds should be observed with an as lower as possible gradient, either positive or negative. The main condition in the filtering process was that the alternative runners length position had an as little as possible influence on the maximum effective torque for the given engine speed.

The first part of the filter consisted of an effective torque map over engine speed and runners length, from which areas with less than 90% of maximum torque at given engine speed were removed . With this approach it was possible to identify the combinations of engine speed and runners length in which the system should not operate. Afterwards, the map was divided into three sections in order to improve the filtration process.

The section of the map in the engine speed range from 9000 to 13500 RPM was identified as one for which the first control algorithm showed that the runners length should be in the upper position, with the tendency of shortening. There were no abrupt changes in control value gradient or its direction, so that part of the filter was set to be the same as the original.

Some minor deviations in the needed runners length position were noticed in the maximal effective torque region, i.e. between 7500 and 9000 RPM. Taking this into account, the analysis showed that there would be no significant difference (<1%), had the runners length been kept at a constant level.

Runner lengths for middle and high regimes are mainly linear. However, the most challenging section for the optimization of required variable runners positioning was in the engine speed range from 4000 to 7500 RPM. The same principle of making the control pattern as linear as possible, with minor influence on the engine effective torque, was utilized. Because this part of map was problematic, it was divided into two sections that were simultaneously resolved.

Values for runners length positions are chosen in such a manner so that the effective torque difference is kept under 5% of the potentially maximum value wherever possible and also the consecutive values do not differ more than 20mm. After a detailed analysis, the complete runners length value over engine speed look-up table was defined and finally implemented into the control algorithm of the designed continually variable geometry intake manifold.

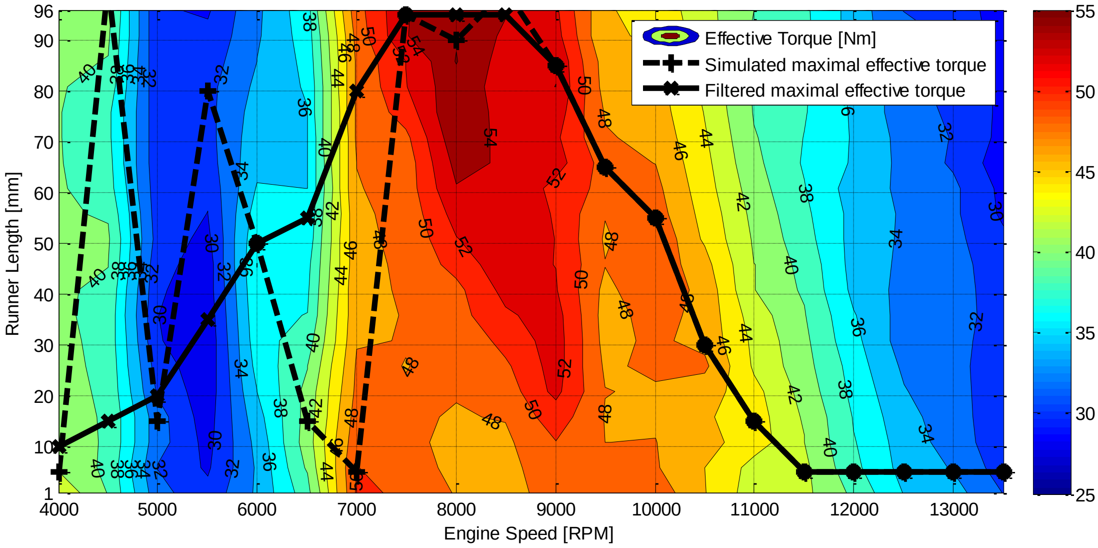


***Figure 17*** Simulated Engine Effective Torque [Nm] and required Runner Length [mm] values over Engine Speed [RPM] at WOT.

Quantitative differences between effective torque and effective power over engine speed from last season's engine testing, as well as the simulated maximal effective torque and power with the implemented CVIM system with raw and optimized RL positioning are shown in the Figure 18:

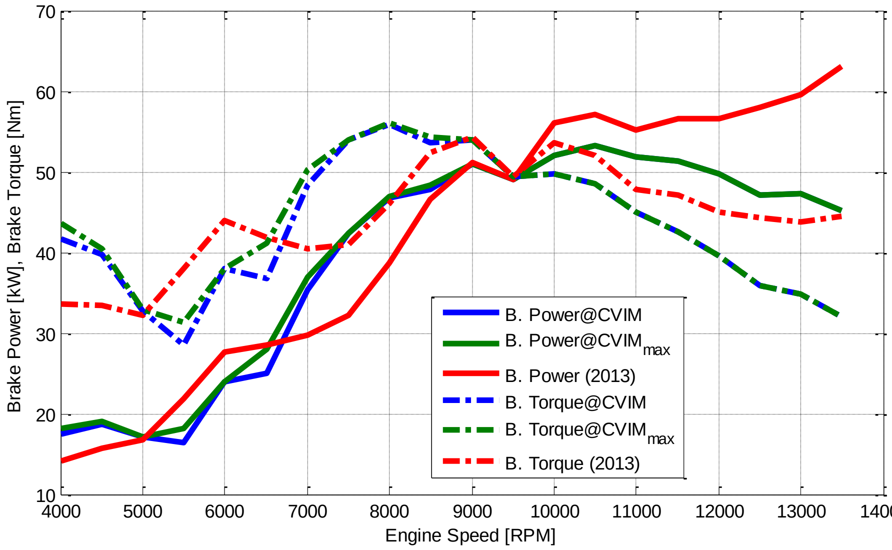


***Figure 18*** Engine Effective Torque [Nm] and Power [kW] curves over Engine Speed comparison of test and simulation results at WOT

## Variable intake manifold concept

While designing such a system, considerations regarding the actuation, control, production technologies, reliability and cost are very important. Taking into account the numerous assumptions about weight and speed of movable parts, the forces in the mechanism and the actuation system needed to be defined. For this kind of analysis, the use of LinMot - Designer software [21] was very useful. Generally, actuation by a linear magnetic actuator, DC motor and stepper motor were all considered. The final solution is based on the use of two Futaba S3306MG high-power servo motors and a set of liners and livers. The intake manifold, runners and the restrictor section were made of high resistant ABS plastic using 3D printing technology.

The implemented CVIM control look-up tables are relied on optimized runners length as a function of engine speed and throttle position, but for this study only WOT regimes are considered. After the production and assembly process, a calibration of the physical model needed to be conducted. Control PWM signals were adjusted with an offset because of the need to bring the system in perfect balance.

Before the implementation of the CVIM into the vehicle, a real-time functionality test was carried out. The simulation of control signals was based on the predefined correlation of required runner length over engine speed which was varied over time on the basis of the data logs which were mentioned before.

Rendered CAD models of the intake manifold that was constructed are shown in the Figure 19. On the left-hand side, the intake manifold with an adjustable runner length that was used during engine testing and competition season of 2013 is shown. The middle one and the picture on the right-hand side show the constructed CVIM in the two end positions of the runners with servos position, injector mounts, restrictor section position and throttle body position.

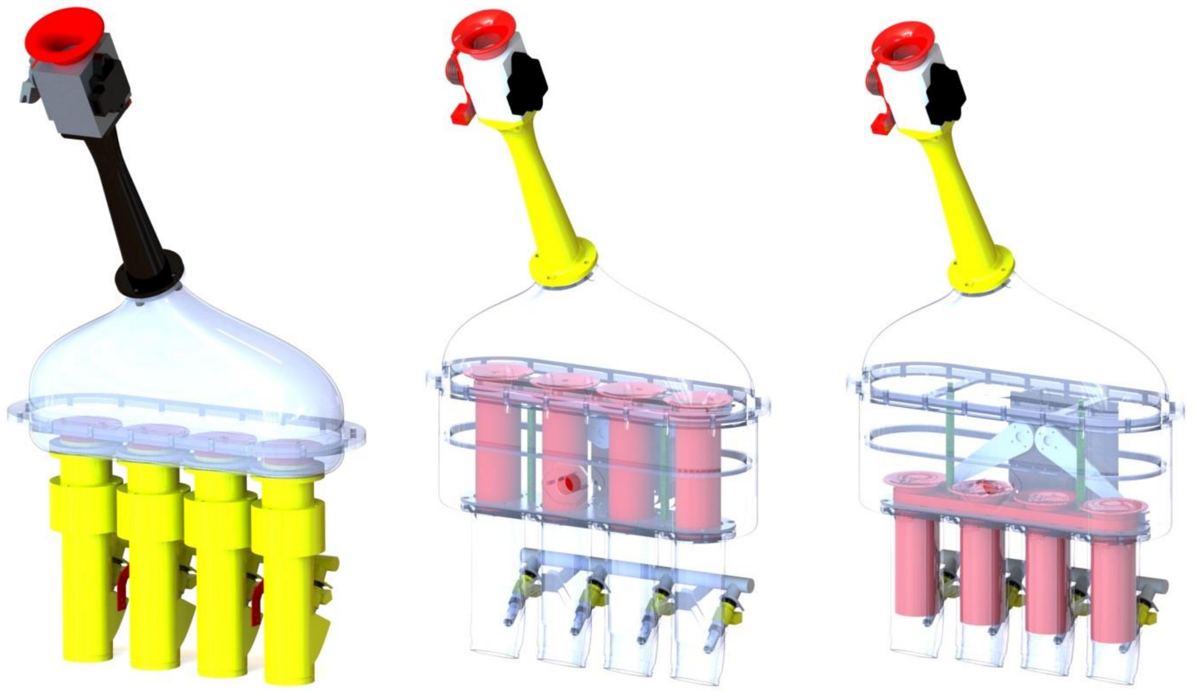

***Figure 19*** Rendered CAD models of the Intake manifold from the 2013 competition season (left), CVIM with extended runners (middle) - 2014, CVIM with contracted runners (right) - 2014.

# CONCLUSIONS

Engine testing is a very expensive and time consuming process, especially if a large number of operating regimes need to be considered. Different engine speeds and loads, combined with the fuelling, ignition timing and intake runner length adjustments could give a large number of optimal combinations in terms of maximum effective performance.

This is supported by the fact that high-revs engines, such as Yamaha's R6, are very delicate and there is a great probability of an incident occurring if something goes wrong during the testing. On the other hand, data obtained during engine testing are very important for the calibration process of the engine mathematical model.

Several years of development of engine systems intended for racing applications, using this approach, resulted in a high-fidelity engine mathematical model and many useful conclusions.

This paper briefly presents some of the analysis results obtained during the CVIM development process. Generally, simulations of the engine work cycle were performed also for partial load regimes. From the aspect of the engine management system, only the use of optimized look-up tables matters. Regarding this, the correlation between injection timing and effective engine torque was formed based on data obtained during engine testing. Having that in mind, the new injection control map was adjusted for the new powertrain system.

Plans for the future include performing crank-angle resolved cylinder pressure measurement at a certain number of engine operation regimes. Also, there is a need to perform intake and exhaust ducts indicating in order to achieve better mathematical to physical model matching.

# REFERENCES


[1] Group of authors, 2014 Formula SAE® Rules, SAE International, 2014.
[2] Group of authors, Formula SAE Italy & Formula Electric Italy 2014, Italian Automotive Technical Association, 2014.
[3] Group of authors, Formula Student Austria, FSA Rules 2014, 2014 .
[4] Group of authors, Formula Student Germany, Formula Student Combustion Rules 2014, 2014.
[5] Group of authors, Formula Student Hungary, FSH 2014 Rules, 2014.
[6] Group of authors, S Series User Manual, DTA Competition Engine Management Systems, 2012.
[7] S. J. Popović, N. L. Miljić, Četvorokanalni univerzalni modul za kondicioniranje signala MSGA-41, 2007.
[8] P. Mrdja, N. Miljić, M. Kitanović, S. Popović, M. Tomić, Model based approach in Yamaha R6 Formula Student engine control parameters optimisation, Interantional Congress Motor Vehicles & Motors 2012, MVM 2012.
[9] P. Mrdja, V. Petrović, N. Miljić, S. Popović, M. Kitanović, Combustion parameters calibration and intake manifold redesign for formula student Yamaha YF-R6 engine, International Conference on Accomplishments in Electrical and mechanical Engineering and Information Technology, DEMI 2013.
[10] Group of authors, Bedienungsanleitung Carl Schenck W130, 1984.
[11] Group of authors, ISO 1585:1992, Road vehicles - Engine test code - Net power, 2010.
[12] Group of authors, ISO 15550:2002 Internal combustion engines - Determination and method for the measurement of engine power - General requirements, 2010.
[13] S. N. Lophaven, H. B. Nielsen, J. Sondergaard, DACE - A Matlab Kriging Toolbox, 2002.
[14] Group of authors, E-Race v3 User Manual, XAP Electronics, 2008.
[15] B. Vasiljević, M. Banjac, Mapa za termodinamiku - Mašinski fakultet Beograd, 2008.
[16] Group of authors, Yamaha YZFR6V(C) Service manual, 2008.
[17] Group of authors, WAVE Knowledge Center v2013.1 Ricardo Software. 2013.
[18] M. Tomic, Fundamentals of the Internal Combustion Engines, Handouts as addition to the lectures, 2011.
[19] D. Cordon, C. Dean, J. Steciak and S. Beyerlein, One-dimensional engine modeling and validation using Ricardo WAVE, Final Report, University of Idaho, 2007.
[20] Blair, Gordon P., Design and Simulation of Four-Stroke Engines. Society of Automotive Engineers, 1999.
[21] Group of authors, LinMot Designer tutorial, NTI AG, 2013.