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IMPROVING THE POWER CHARACTERISTICS OF AN INTERNAL COMBUSTION ENGINE WITH THE HELP OF A 0D/1D ENGINE MODEL

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Abstract: During the initial design phase of an ic engine it is not important to exactly identify all the thermodynamic, fluid mechanical and chemical reactions within the engine in question. What is of interest is the whole system that needs to be optimized to reach the preset targets with sufficient precision. In this work the validation of an engine model is presented and the results of the simulation and measurements are compared and examined. The outcome of the comparison is the clear identification of the cause of the dip in the torque characteristics of the modelled test engine while also presenting a possible solution for this special problem.
Keywords: engine, simulation, power, torque, computer model

1. INTRODUCTION

During the initial design phase of an IC engine it is not important to exactly identify all the thermodynamic, fluid mechanical and chemical reactions within the engine in question. It is more important to optimise the whole system to reach specific targets with sufficient precision. This defines the path of further development of subsystems or parts. Therefore the aforementioned reactions in each component are modelled with a certain level of approximation [4]. In the simulation, components belong to two greater groups: connection elements and devices. Connection elements connect all the devices, and represent the pipes and any other pathways in that air, air and fuel mixture or exhaust gas can move along.

During modelling the direction of flow is necessary information. In order to get results within a reasonable timeframe with the usually available computational background only the equations describing the flow in the direction of the axis of the connection elements are solved. Therefore the equations describing the transport of particles perpendicular to the longitudinal axis of the connection elements (y and z directions) are missing from the solutions. Hence they are named 1D element.

On the other hand devices, such as valves, plenums, branches, etc. may only have information on eg. their volume, branching angles etc.. In these entities most properties are treated as component averaged scalar values and no spatial information is available. Therefore these are the 0D components of the model.

In special cases the 0D/1D simulation software can be coupled parallel to 3D CFD software to model certain components (air filter box, catalyser, etc.) [5]. This way the input values of the CFD simulation are produced by the 0D/1D engine model while the output values from the CFD model are fed back to the engine simulation software. The accuracy of the engine simulation can be increased greatly but there is a price to pay: the time needed to complete the modelling along with the costs of software and hardware is also increased.

2. INTRODUCING THE ENGINE

A motorcycle engine had been chosen for the tests. It was a Suzuki SV 650 manufactured in 2003. The technical specifications of the engine can be found in Table 1 [7]. It is important to note that this particular engine has only one spark plug per cylinder that facilitates the wider use of the simulation results to other engines.

Table 1. Technical specifications of 2003 Suzuki SV650

Configuration:	90 deg., 2 cyl V, 4 stroke wet sump
Valve actuation:	DOHC, with inverted bucket follower
Number of intake/exhaust valves per cylinder:	2/2
Compression ratio:	11.5:1

3. SPECIAL REQUIREMENTS DURING THE TESTS

To facilitate straightforward simulations a simple-to-model engine was needed so the OEM exhaust silencer was removed. This with its chamber system would have complicated the simulation process, therefore it was replaced with straight through pipes. This way the wave propagation along the exhaust pipe was not affected by any way and they could be identified easier. In order to fit the engine model more accurately to the measured values two sets of measurements were made: the first with a 550 *mm* long exhaust collector pipe, the second with a 785 *mm* pipe. The inner diameter was 56 *mm* that connected to the original collector that had 50 *mm* inner diameter. This diameter difference proved to be useful as it generated small but significant pressure waves that lead to the final design.

4. RECORDING THE DATA NECESSARY TO VALIDATE THE MODEL

For registrating the engine parameters that can be used to validate the model a rolling road dynamometer was used. The actual measurements were conducted on a Superflow CycleDyn Pro (SF-250)- WynDyn 3.2 dynamometer. It has a special feature namely that the tested vehicle's air intake openings are supplied by air at a speed that would exactly correspond to the road speed of the vehicle under the test. This reproduces the actual road conditions and increases the accuracy of the test. During the tests it was considered to be fundamental to record parameters that could be compared to parameters produced by the simulation program. These were: power, torque, exhaust gas temperature in the header pipes, air-fuel ratio (lambda), wheel slip, parasitic losses in the driveline of the vehicle.

Table 2. comparison of published and measured engine parameters

Parameter	Published manufacturer data		Measured data	
Power	54,7 kW	8800 rpm	54,05 kW	9250 rpm
Torque	64 Nm	7000 rpm	58,5 Nm	8750 rpm

To get accurate information on the state of the engine to be tested the first recordings were taken with the OEM exhaust system in place. The collected data was corresponding well to the parameters published in the vehicle's service manual [7]. The comparison of these parameters can be found in Table 2.

The possible reason for the differences is the usually generous description of the vehicles' parameters by the manufacturers, which is a well known marketing strategy. Aside from that the test engine was assessed to be in reasonable condition to continue the validation work.

After this stage the OEM exhaust silencer was removed and replaced with the straight through pipes. As can be seen in Figure1 none of the replacement pipes improve on the peak power figures but considerably affected the midrange torque, unfortunately not for the better. This decrease in torque is worst in the range between 4200-6000 *rpm* and is highlighted most with the 785 *mm* exhaust pipe.

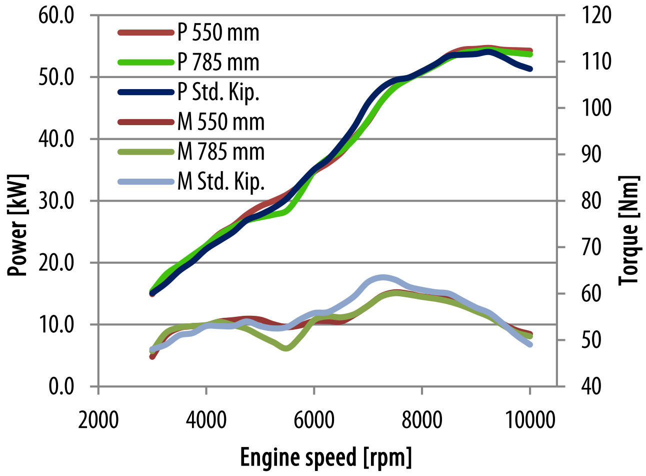

Figure 1. Power and torque characteristics of the engine with the OEM and the two manufactured exhaust systems

5. STRUCTURE OF THE SIMULATION MODEL

The simulation model was built with components described in the "INTRODUCTION" section of this present paper. It looks like a network of different elements (Figure 2). Data describing the individual parts can be input through dedicated pop-up panels. This does not mean any problem with pieces that can be measured easily in reality. These are intake and exhaust pipes, connecting rod centre to centre distance, compression ratio etc.

To define the Coefficient of Discharge of the intake and exhaust valves needed special equipment. The highest pressure ratio of this apparatus was 1,1 but the actual pressure ratios occurring across valves of internal combustion engines are much higher than this causing flows reaching sonic conditions locally. Since the value of Cd is affected by the velocity of flow the recorded discharge coefficient values were extrapolated up to a pressure ratio of 2 using a built-in target function of the simulation software [6]. The returned result was a surface of Cd values as a function of relative valve lift and pressure ratio (Figure 3).

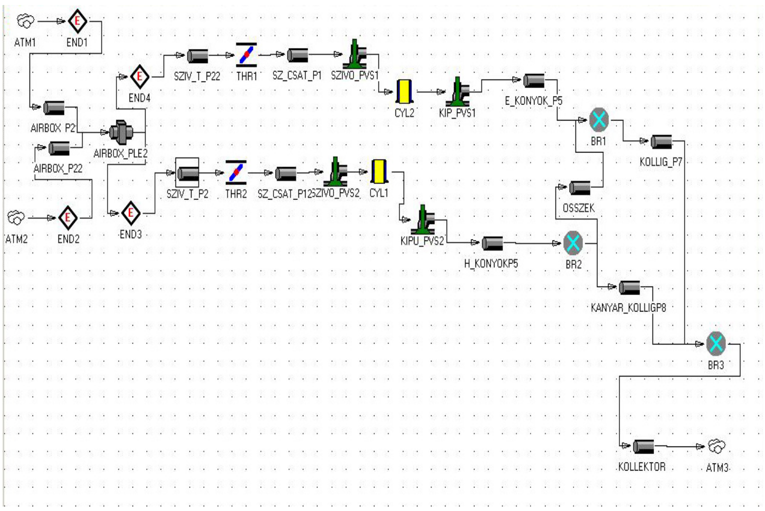

Figure 2. Outlay of engine simulation model

All valve systems have two Cd surfaces: one for normal direction of flow and one for reverse flow. Normal means inflow for intake valves and outflow for exhaust valves, while reverse means outflow for intake valves and inflow for exhaust valves considering the cylinder as datum. These later Cd maps in the reverse direction define the amount of fresh charge escaping from the cylinders during blow back through the intake valve while at the exhaust valves the reverse Cd maps describe the amount of exhaust gas recirculated internally in the engine.

Other important parameter of valve systems is the opening and closing point of the valves relative to the momentary piston position and the actual time areas of the opening processes. Input of these parameters was only possible after the accurate measurement of the camshafts. The details of the measurement can be found in the literature [3].

The cylinders are the central parts of the model. Beside of the obvious measures and relative ignition sequence the frictional mean effective pressure (FMEP) needed to be given, too. The internal friction characteristics of the engine could not be recorded directly only the values obtained during the dyno test were available. These were masked heavily by the driveline losses of the vehicle therefore these were used only as base for the actual data input during the simulation runs. There is a debate in the literature on the exact shape of the frictional losses diagram: Blair [1] suggests linear increase of losses with increasing engine speed while Yagi et al [8,10] conducted researches based on motorcycle engines and suggests progressive increase with speed. Since the test engine is a motorcycle engine the actual values used in the simulation were chosen based on the data presented in the works of Yagi et al, resulting in progressively increasing FMEP values of 0,41 *bar*@1000 *rpm* and 1,8 *bar*@10000 *rpm*.

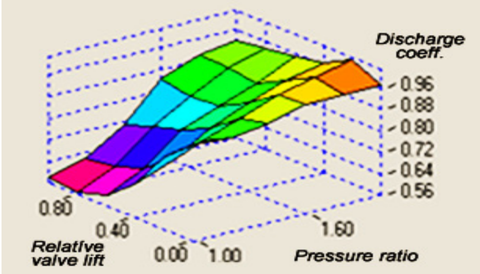

Figure 3. Extrapolated map of Discharge Coefficient as a function of relative valve lift and pressure ratio across the valve

6. RESULTS OF MODELLING

After setting up the model the fourth run produced acceptable results that quite well matched the measured power and torque characteristics of the engine. Seven more runs needed to refine the model with the 785 *mm* collector pipe to reach a maximum variation of ± 2,9 kW between measured and simulated power values. This equalled an average error of 5%, which value is considered to be sufficient precision [2] (Figure 4). The torque characteristics followed a similar nature and the average error stayed within 5% in this case as well (Figure 5).

As a check - the same way as was done in reality – the exhaust collector length was decreased to 550 mm and the model was run again. In this case the error value between simulation and measurement was 24% at 3000 rpm but at the most critical 5000 rpm the error decreased to 1% and never exceeded 7% in average which was true for the power and also for the torque values.

In the torque curves it can be clearly seen that general trends, troughs, etc. occur at the same engine speed in the model as in the measurements. Based on this general behaviour of the simulated engine it can be concluded that the model reflects the real engine parameters with appropriate precision.

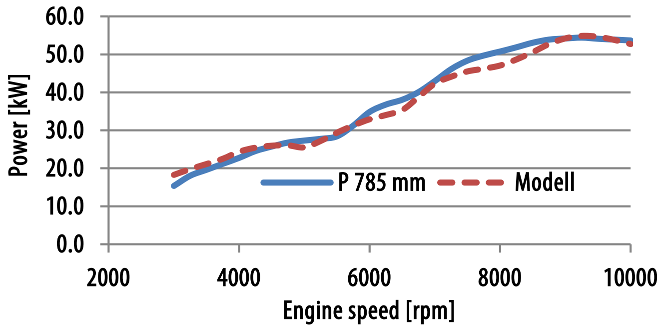

Figure 4. Comparison of measured and simulated power curves obtained using the 785 mm exhaust pipe

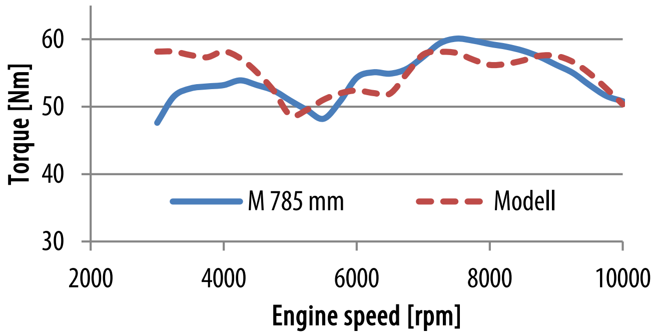

Figure 5. Comparison of measured and simulated torque curves obtained using the 785 mm exhaust pipe

7. EXAMINATION OF THE MODEL

During further simulation the object was to find the reason for the great decrease in the torque between 5-6000 *rpm*. Using the possibilities of the simulation software the Delivery Ratio (DR) curves were investigated cylinder by cylinder basis. This showed that the rearward facing cylinder, which is number 1 considering the ignition sequence, has its DR far from ideal (Figure 6).

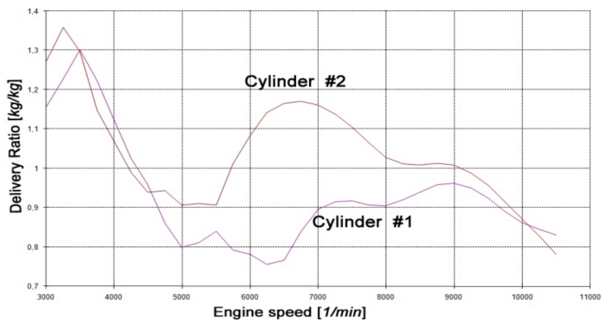

Figure 6. Delivery ratio curves for both cylinders

To find the cause of the deficit in Delivery Ratio the animation utility of the simulation software was used (Figure 7). This function made possible to visually check the pressure waves in motion within the gases of the engine while also showing the valves at their appropriate position relative to the piston.

With this method it was discovered that the shortage of DR of cylinder #1 is caused by a pressure wave emanating from itself. This wave propagates not only towards the open end of the system but enter s all other openings within the exhaust system. In this case it intrudes into the pipe system of cylinder #2 as well through the cross connecting "balance" pipe. Since this is a V engine the exhaust valves of cylinder #2 are closed when the pressure wave arrives there and the wave is reflected back with the same attitude eg. as a compression wave, the same way as it arrived from. So it travels again towards the open end of the system but also enters any pipes, now to the header of cylinder #1. It travels up to the valve but arrives there during valve overlap.

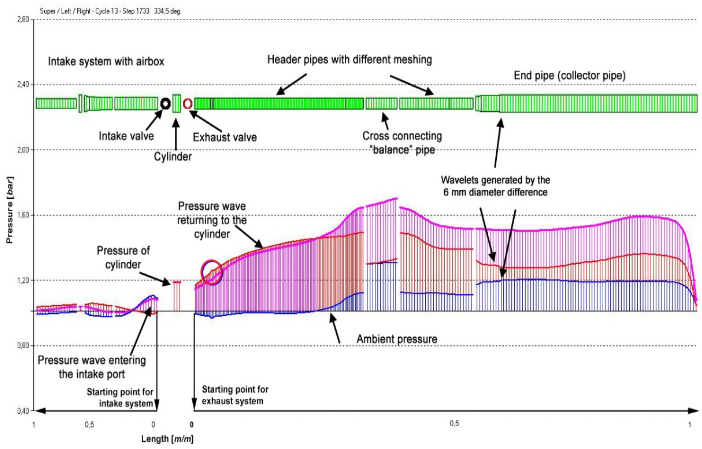

Figure 7. Animation showing cylinder #1 at when the exhaust pressure wave enters the intake port causing back flow and diluting fresh charge with exhaust gas. The red circle shows wavelets produced by a welding protruding into the exhaust flow. Purple shows the superponated pressure, blue the returning (leftward) waves, red is the escaping (rightward) pressure waves

The result of this is that the arriving compression wave partially blocks the exhaust process and the outflow from cylinder #1 happens at a much lower speed than from cylinder #2. To worsen things the arriving wave pushes the fresh charge back to the intake port diluting it with exhaust gas. Therefore the combustion process is lower efficiency in this cylinder.

8. ACHIEVED IMPROVEMENTS

To solve the problem caused by the badly timed reflected wave blocking the balance pipe totally seemed to be a good idea. It really would flatten the torque curve at 5000 *rpm* for a price of loosing around 5 *kW* from the peak power. As an alternative solution the length of the intake tracts were changed but this would only replace the depression with the same amount of lost kW's from peak power value.

To get the most horizontal torque characteristics a collector pipe shape was found in the simulation environment that contains a 80 *mm* long conical section where in reality the 6 *mm* step in diameters was between the OEM collector and the manufactured end pipe. Continued search for the flattest torque curve lead to a solution where the pressure waves originated from cylinder #2 are used to help the gas exchange process in cylinder #1. This was achieved by an 860 *mm* collector pipe. As a result the dip in the torque characteristics could be decreased and the cylinders now can work with the least differing amount of air.

To arrive to this design about 130 simulation runs were conducted. During this work it became clear that the engine could not respond to any changes to produce not only better torque and power distribution with fixed peak values but to improve on the peak figures, too. It was observed that modifications aimed at the physical dimensions of its gas exchange system were not enough to produce the required result. After almost all solutions tried and excluded, the reasons were pinpointed to be the inadequate time area values for the valve events. At this point three possibilities were evaluated:

» changing the intake valve time area,
» changing the exhaust vale time area,
» changing the time area of both valves.

On the grounds of getting the most out of the engine, camshafts should have been changed. This would have been essential in achieving the desired valve lift and duration characteristics. As described in [3] the intake valve springs are at their limit at full valve lift therefore increasing the intake valve lift any further would require very expensive modifications. For that reason changing the timing and lift values on the intake side or both sides together were not tested.

In contrary to the intake side the exhaust valve spring has some margin at full lift for some improvements. Therefore exhaust valve open duration was lengthened by 22 Crank Degrees (CA°) and the lift were increased by 1,3 *mm*. This modification immediately returned an overall improvement of power and torque in every aspect.

Checking on the p-V diagram (Figure 8. a-b.) the reason of the improvement could be easily identified. The area in the diagram corresponding to the work consumed by the exhaust stroke has been decreased in both cylinders. This change also improved the engine volumetric efficiency and proved to be one of those rare occasions when improvement in performance figures does directly not translate to increased fuel consumption. In fact rather the opposite is true: To get the same road performance less fuel is needed thanks to the improved efficiency of the engine. In cylinder #1 the effects of pressure wave tuning achieved with the 860 mm tapered exhaust pipe could also been identified. Because of the positive simulation results a decision was made to implement these changes in the material world.

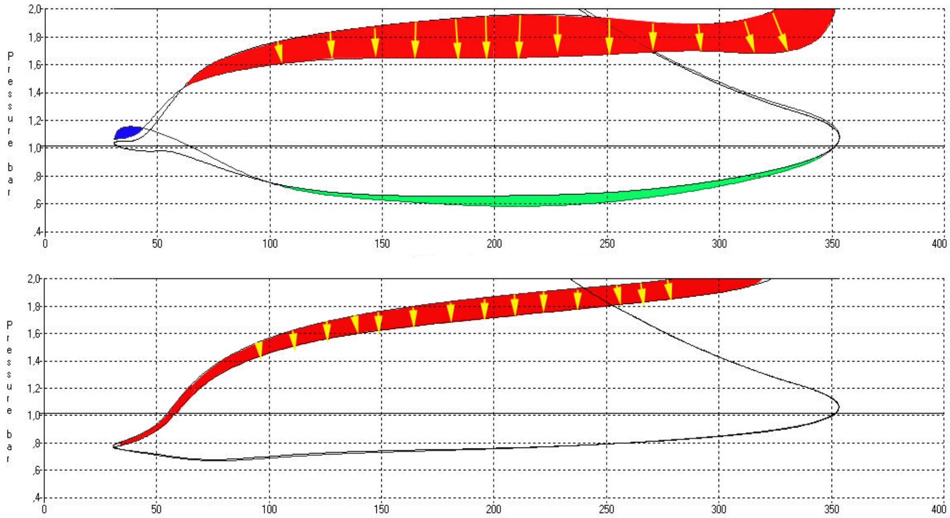

Figure 8. a-b. P-V diagrams of the gas exchange process showing the decrease in pumping losses in red. The green area represents a small improvement in the strength of the intake stroke realized by the smaller amount of exhaust gas back flow.
a: Cylinder #1 b: Cylinder #2

Given that the engine should be used within a vehicle an absorption silencer was added to the model to check its effect on engine parameters. The dimensions of the silencer can be found in Table 3. It was simulated as a series of interconnected chambers, which is shown in Figure 9.

Table 3. Main dimensions of the absorption silencer

Inner diameter of perforated pipe	60 mm
Diameter of perforation holes	5 mm
Inner diameter of silencer shell	130 mm
Length of silencer	450 mm

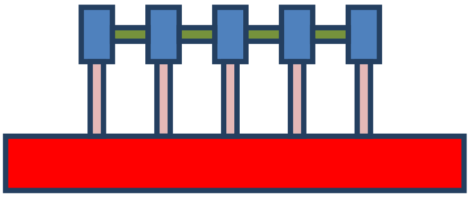

Figure 9. Representation of the silencer adopted in the simulation environment

The simulation produced better results than expected as the silencer's body, represented by a system of chambers, reflected waves that really helped to fill-in the depression in the torque curve. Peak simulated values of torque and power were significantly increased as well. (Figure 11-12).

At this stage the model changes were implemented to the actual engine. Figure 10 shows the actual silencer that was created based on the model information. To check the quality of modelling the results were validated using the dynamometer mentioned above. The outcome of the test showed that the behaviour of the engine model matches very closely the real mechanism.

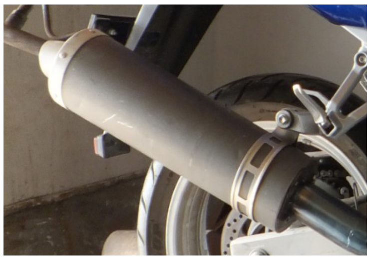

Figure 10. Silencer manufactured based on the results of simulation

With the final set of implemented design changes the average model error is 3% between power simulation and measurement, while it is 4% in the torque characteristics (Figure 11-12.C, D curves). The reason for the differences seen in the lower half of the rpm range is probably caused by the overly rich air-fuel mixture commanded by the OEM Engine Control Unit.

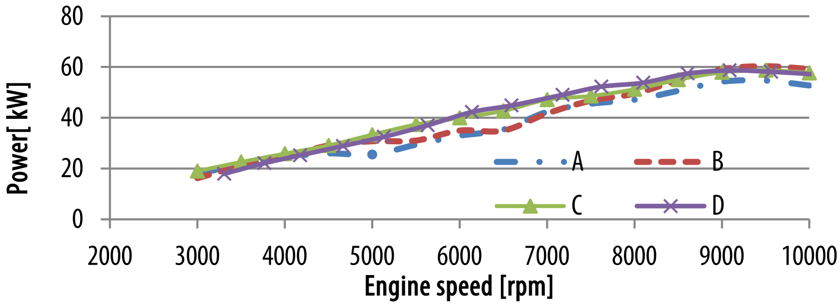

Figure 11. Comparison of power characteristics of simulations and measurement
A: Simulation I. (785 mm straight pipe), B: Simulation II. (860 mm straight pipe + exh. cam changed)
C: Simulation III. (860 mm pipe with silencer + exh. cam changed), D: Measurement (Silencer with exh. cam)

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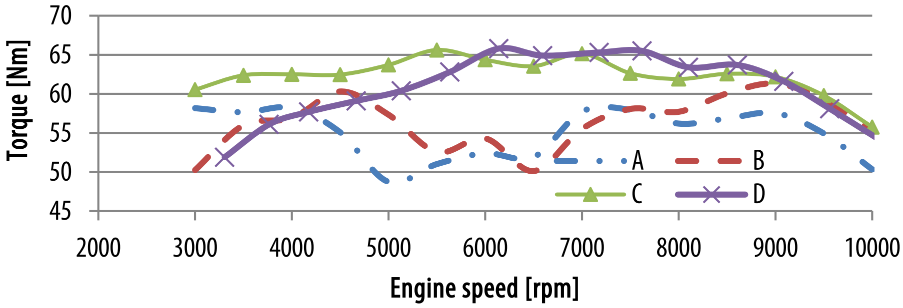

Figure 12. Comparison of torque characteristics of simulations and measurement
A: Simulation I. (785 mm straight pipe), B: Simulation II. (860 mm straight pipe + exh. cam changed)
C: Simulation III. (860 mm pipe with silencer + exh. cam changed), D: Measurement (Silencer with exh. cam)

9. CONCLUSIONS

Further improvements could be achieved by changing the programming of the Engine Control Unit or through modifications to the intake and exhaust ports to improve their flow characteristics. Though these changes would increase cost and fuel consumption. The changes described in this paper work on decreasing the engine's pumping losses hence not requiring additional fuel to return the expected improvements in engine parameters while also decreasing emissions.

As has been shown it is not necessary to know the exact 3D fluid dynamics in an engine's subsystems to produce a model with sufficient accuracy. The method presented demonstrates the possibilities that help the designer to track down the gas dynamic phenomenon in a running engine. With this knowledge a properly designed gas exchange system has been suggested that takes into account the peculiar gas dynamic nature of the multiple cross connected exhaust system, and the asymmetric wave motion in the presented engine, having V cylinder configuration.

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