

Bosch Motorsport has finally brought its much anticipated engine simulation software to market. Its author talks us through what the new package is designed to achieve

By **Chris van Ruten**

Bosch has developed an engine simulation package to extend its LapSim vehicle simulation product. The goal is to supply an easy-to-use engine simulation package capable of generating efficiency, as well as torque and power curves. The model should be able to simulate any four-stroke spark ignition (SI) race engine currently on the market, either with or without air restrictor(s). From the modelled engine's air consumption, the program should also be capable of generating a fuel map dependent on rpm and throttle position.

The 3D fuel map, combined with a chassis simulation, creates the opportunity to compare lap time with fuel used per lap of the vehicle/engine combination. Fuel economy is an issue, especially in endurance racing, because it determines the frequency of pit stops, and refinement of gear ratios/shift points strategy for fuel saving is possible. Feasibility studies also become more accurate and it is possible that an engine/vehicle combination, which may not necessarily have the fastest lap time, can still be a race winner due to better fuel economy.

In line with the general approach of LapSim, the engine software avoids a very complex model with vast numbers of variables in order to define every engine detail, as it would automatically become very expensive and hard to use, therefore strongly limiting the potential customer base. To summarise, the software is aiming for 95 per cent accuracy but with only five per cent of the effort of complex engine software packages. All the figures in this article are made by 'print screen' shots from the Bosch software. Nothing is specially designed for this article.

The model has been tested with many existing race engines. This includes two, four and five-valve heads, low (by regulation) and high

compression ratios and low and high rpm engines with large and small individual cylinder capacities. While testing the model, the internal constants have remained equal for all the engines trialled. With a relatively small number of input parameters, one is able to describe all kinds of racing engines, but still able to generate surprisingly accurate performance curves.

Valve resistance

The output of an engine is largely influenced by the amount of air that is processed by the cylinders. This is actually the amount of air that the cylinder holds once the valves have closed. In order to determine the amount of air captured, it

“CAPABLE OF GENERATING EFFICIENCY, TORQUE AND POWER CURVES”

is important to simulate the dynamics of the intake and exhaust tract. These dynamics create the changing pressure differences between the intake, cylinder and exhaust that cause flow. However, the flow is primarily limited by the resistance of the valves.

To keep the model simple, the valve resistance is assumed mainly dependent on the effective area of the poppet valve opening, multiplied by a resistance factor. Much attention has been given to this effective area. It depends strongly on the motion, angle and the position of the valves in the cylinder head. Figure 1 shows the resulting airflow of the intake valves, dependent on valve lift.

The resistance factor is adjustable in the software to facilitate accurately matching cylinder heads that have been flowbench tested. However, to validate the model for the different engine configurations, it has been kept constant and the accuracy of the model for the wide variety of configurations is still surprising.

Camshaft profile

With the resistance of the valves determined, the next step is to relate valve position to the crankshaft angle. This is done with an adjustable camshaft diagram, in which duration and lobe centre position, as well as opening and closing acceleration, can be specified.

Additionally, valve clearance is set with both length and height of the starting ramps specified, leading to an accurate and effective motion description of the valves in regard to crankshaft position. Naturally, intake and exhaust timing are specified independently. A further special option caters for engines in which maximum valve lift is limited by regulations. In this case, users specify how long the cam dwells at its maximum lift.

Intake and exhaust dynamics

With the valve motion now specified, as well as the resistance of the valves determined as a function of the valve lift, the next step is to model intake and exhaust dynamics. The computational goal of the model is to generate a cylinder pressure diagram over the full 720 degrees of crankshaft rotation. In order to do so, the amount and composition of air admitted to each cylinder needs to be defined. Of course, this varies with engine rpm.

The cylinder bore, stroke and connecting rod length determines the exact position of the piston, and thereby the cylinder volume, at any

crank angle. With rpm, these inputs also determine the piston speed, which has great influence on the dynamics.

Both intake and exhaust are represented by spring/mass systems, with properties defined by the lengths and diameters of the intake runners and primary exhaust branches respectively. While this is a simplification, Bosch has taken into account the fact that both intake manifold and exhaust configuration are more complex, and is of the view that the gain in user friendliness and calculation speed outweigh the loss in accuracy.

The behaviour of the intake and exhaust dynamics of the model can be analysed in figure 3, where the green line represents the relative position of the piston. This can be compared with the relative flow of the intake, represented by the two blue lines. The darker blue line monitors the flow of air in the middle of the intake runner, while the lighter blue line shows the flow of air over the valves. So, where the light blue line ends (signifying that the valves are closed), indicates how much air has been trapped. The difference between the two blue lines demonstrates how much the intake air is compressed at the end of the intake runner, due to the dynamics. The red line is cylinder pressure. The plots in figure 3 show the changing behaviour of the model due to increasing rpm. It can clearly be seen that due to

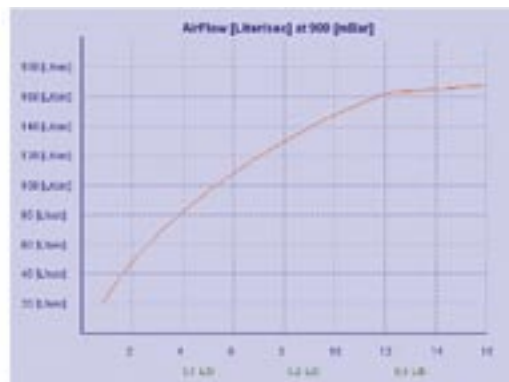
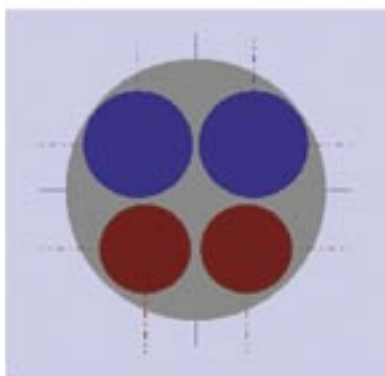


Figure 1: overhead view of cylinder head with valve arrangement and the subsequent airflow dependent on valve lift. Airflow is mainly determined by the minimum opening area and resistance coefficient

favourable dynamics in this case, air delivery increases with speed over an extended range. This continues until the point where the resistance of the valves and the time available from cam duration combine to leave the cylinder 'short of breath'. Furthermore, the figures show that with increasing rpm, the dynamics become more volatile, evident from the larger cylinder pressure changes. A different combination of intake dimensions and cam timing would change the characteristics.

The result of the intake and exhaust dynamics is a fill rate of the cylinder in combination with a fresh air ratio. Some of the cylinder contents will

be residual exhaust, determined by interaction between the intake and exhaust dynamics and valve restrictions. This changes significantly with part-throttle operation and the operation of an air restrictor.

The results for the rpm sweep can be seen in figure 4. In the figure, it can be seen that dynamic effects dominate from 4500rpm onwards, whereas the resistance of the valves start to limit the amount of air entering the cylinder when the engine revs above 8000. The grey/black line shows the amount of fresh air that goes directly into the exhaust, and the sensitivity to intake and exhaust dynamics of the overlap timing of the

Figure 2: camshaft profile. Duration and lobe centre, as well as opening form and closing area are given. Independent opening and closing accelerations may specify non-symmetrical cam shape

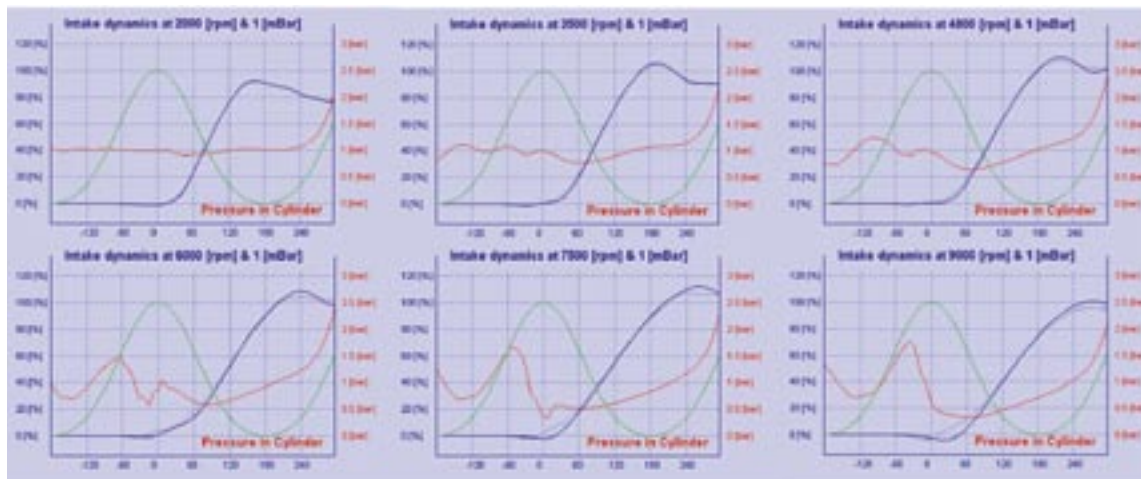
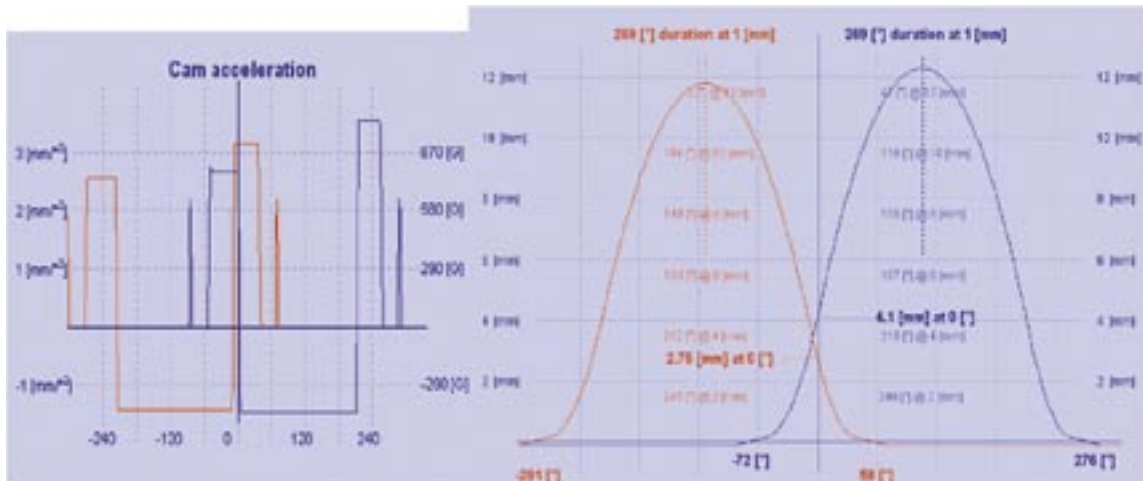


Figure 3: visualisation of intake dynamics. Parameters are plotted against crank angle (0 = TDC) and figures from top to bottom show the intake dynamics at 2000, 3500, 4800, 6000, 7500 and 9000rpm

inlet and outlet cams is clearly evident in the trace.

The air box pressure is shown in green on figure 4 to summarise the effect of an air restrictor. The example shown is without an air restrictor.

Two-zone burn model

With the amount of fresh air and the residual exhaust gasses in the combustion chamber determined, the compression and expansion strokes can be calculated. As the end of the compression stroke approaches, combustion is initiated (combustion is neither an instant nor a constant process).

The easiest solution is to use a constant burn rate or a constant burn profile, but it was considered too big a simplification if all the different types of race engines are to be simulated properly. Furthermore, model goals include simulation of part-load conditions. The errors made by a combustion profile proved to be too

“THE ENGINE SOFTWARE AVOIDS A VERY COMPLEX MODEL”

large with these constraints. So, a more elaborate burn model was developed.

The burn model consists of two zones: a fresh mixture zone and the burned zone. Each zone has its own temperatures as a degree of freedom. The burn speed of the fresh mixture zone depends mainly on the instantaneous temperature of the burned gas, as well as the temperature and pressure of the unburned gas. The model takes the amount of turbulence in the combustion chamber into account, by looking at the air speed over the valves.

With the two-zone method, the form and size of the chamber geometry can have a significant influence on burn progression. A smaller bore, for instance, will lead to shorter burn duration. A bad chamber shape will lead to longer burn duration and thereby an efficiency loss. The user determines the shape of the combustion chamber not just by bore, stroke and compression ratio, but with chamber height and radius, too.

Two burn sequences for different ignition points are shown in figure 5. The top sequence is the normal predicted ignition point, whereas the bottom sequence shows the results of delayed ignition. The rate of burn slows significantly with retarded timing, so complete combustion is not achieved until the piston is well down the expansion stroke.

Since the burn speed is a degree of freedom based on parameters in each zone, the model can calculate the influence of advancing or retarding the ignition, also displayed in figure 6. The left

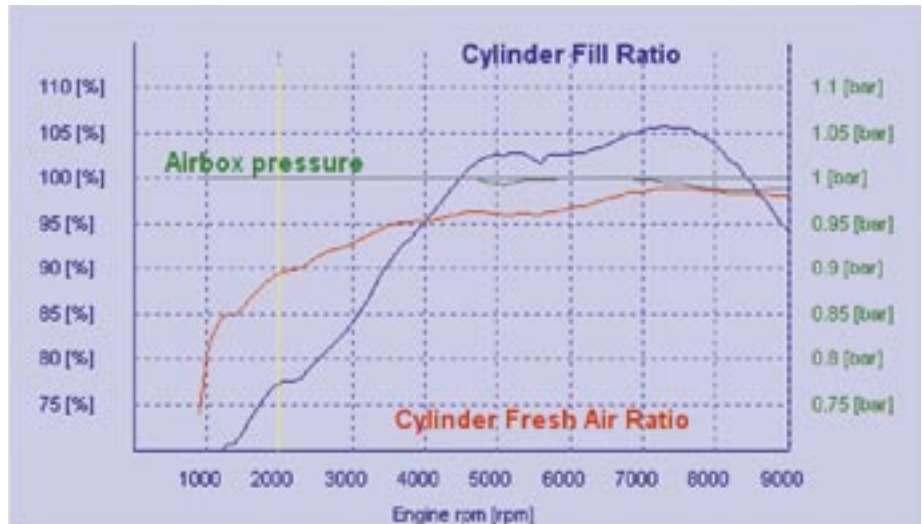


Figure 4: result of the intake and exhaust dynamics, shown over the complete rpm range. Fill ratio of the cylinder (blue line), percentage of fresh air (red line), and amount of fresh air that goes directly into the exhaust (grey line). This occurs between 4800 and 5500rpm, as well as from 7000rpm onwards

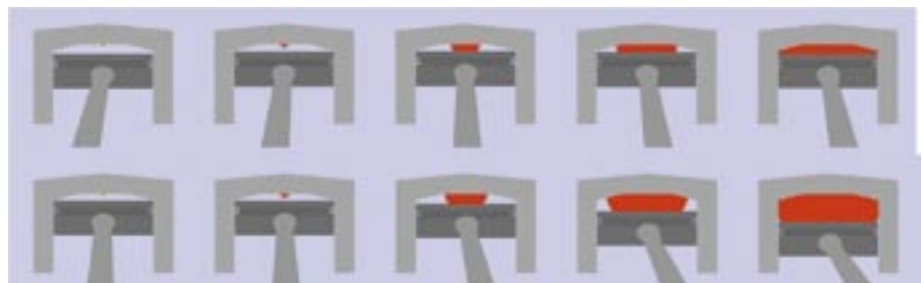


Figure 5: the burn sequence visualised. A two-zone burn model takes the chamber form into account. In both cases the left most picture is the moment of ignition, following (to the right) the progress of the flame front in the chamber. The top sequence is at 21 degrees ignition, the bottom sequence at nine degrees ignition (on a high compression 14.8:1 engine)

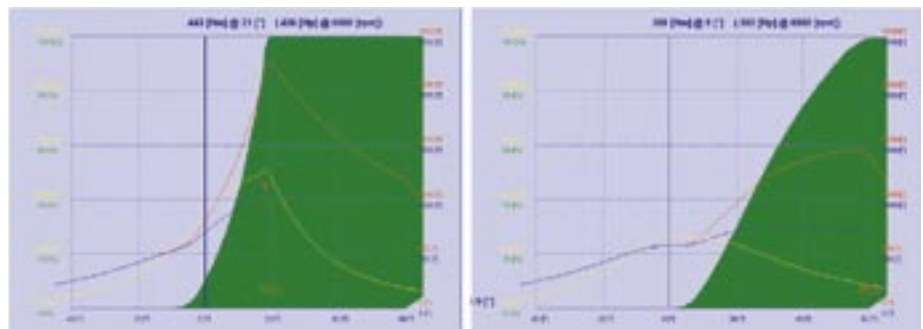


Figure 6: overview of the burn sequence. The red line is temperature burned gas, the yellow line cylinder pressure and the blue line temperature unburned gas. The red star is an indication of where the unburned mixture exceeds the knock temperature, while the green area is the percentage of mixture which is burned. The right figure shows the influence of retarding the ignition on the burn process

figure shows the burn sequence at normal ignition timing (21 degrees) versus inefficient combustion when the ignition is retarded (nine degrees).

Torque generation falls from 443Nm to 308Nm, a reduction of about 30 per cent. The figure shows not only the overall duration extension, but also the altered form of the burn sequence.

A feature of the model is an ignition reduction curve, displayed in figure 7 for full load at 6900rpm. The model makes an ignition sweep and shows the resulting torque for each ignition point. In the figure, the red zone indicates when the maximum cylinder pressure exceeds the limit

specified by the user, 80bar in this case.

With the combustion model's multiple degrees of freedom, knocking can be simulated, wherein the burn speed is very high and subsequently the pressure rise becomes too great. The model explores this danger zone without destroying the engine, to see if further advance results in theoretical torque/efficiency gains.

Resulting cylinder pressure diagram

With combustion modelled, all the parts can be combined, generating a pressure curve over the full 720 degrees of crankshaft rotation. □

Figure 7: reduction curve engine torque versus ignition point. The red area indicates where the cylinder pressure exceeds the specified maximum pressure (80bar in this case), while the second plot corresponds with the right plot in figure 6

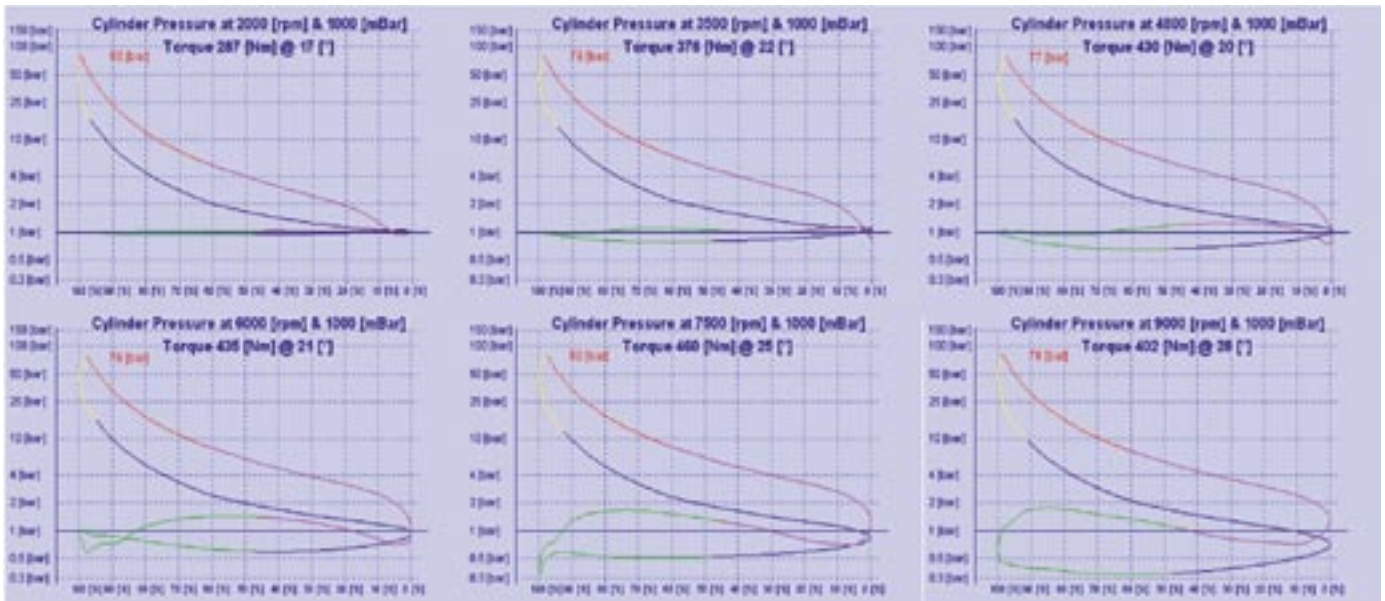
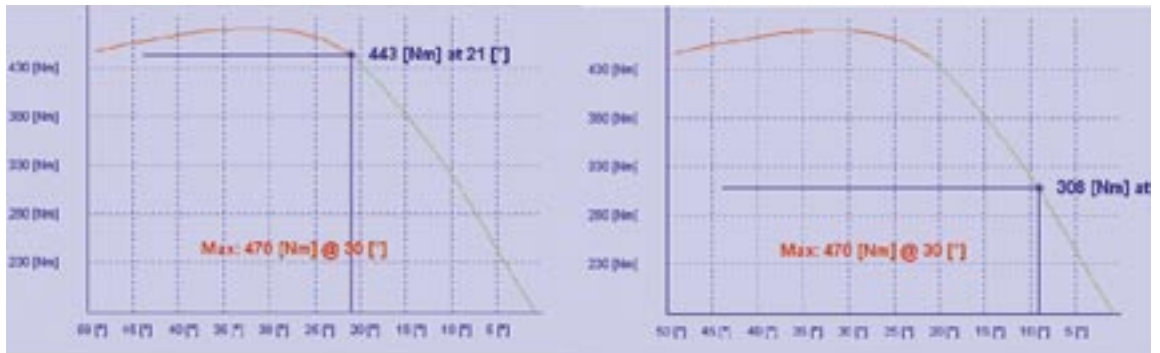


Figure 8: resulting cylinder pressure over the complete four-stroke cycle. Maroon is exhaust valve open, green is both exhaust and intake valve open, blue is intake and compression until ignition, yellow is combustion and, finally, red is power stroke with valves closed. The figures show the indicator curves at 2000, 3500, 4800, 6000, 7500 and 9000rpm. The increasing influence of the pumping losses become very visible in this figure. Please note that the y-axis is a log scale

Integration of the pressure curve over these 720 crank degrees gives indicated torque, automatically integrating the pumping losses. Only the mechanical losses, identified by the user as an engine braking torque, need to be subtracted from the indicated torque in order to calculate the flywheel output torque.

Figure 8 shows an overview of the pressure diagrams for several rpm levels, to analyse the functioning of the model. It can be seen that the pumping losses at low rpm are very small, whereas they become significant at higher rpm levels. However, the log scale accentuates this to purposefully show the pumping losses.

The pressure trace is colour coded to amplify the engine's operational performance. Yellow represents the burn phase, green is the overlap between intake and exhaust valves and maroon displays when only the exhaust is open. Additionally, the maximum cylinder pressure is noted in red. In this simulation, the maximum value was 80bar. The model has selected the ignition timing so as not to exceed this pressure. In the different figures, it is seen that the ignition point changes with rpm.

Furthermore, it can be seen that due to the

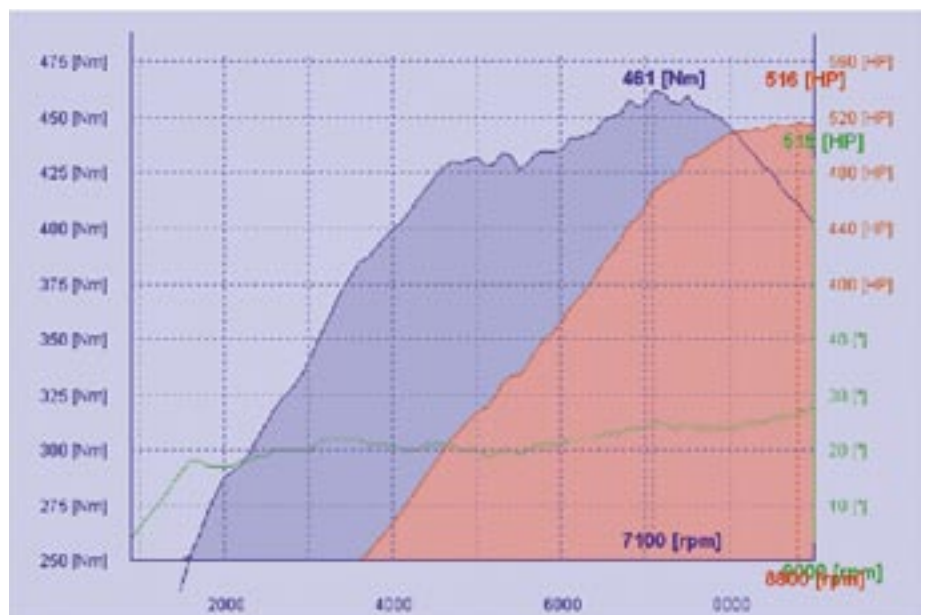


Figure 10: open throttle engine torque, power and ignition, calculated by integrating cylinder pressure over the 720 degree rotation of the crankshaft

early opening of the exhaust to suit high-speed running, not all of the energy in the expansion stroke is used at low rpm. This is evident from the sudden reduction of pressure to atmospheric

before the piston has reached BDC.

At 7500rpm it can be seen that at TDC with both valves opened, the inertia of the exhaust gasses creates low pressure in the cylinder, thereby

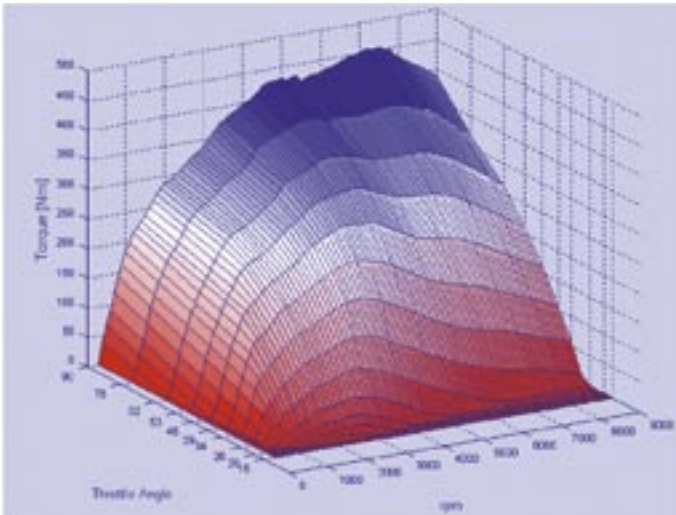


Figure 11: engine torque as a function of engine rpm and throttle angle

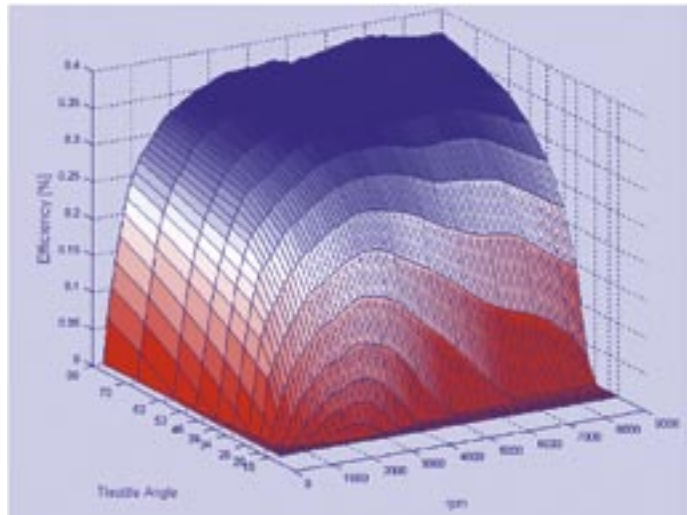


Figure 12: engine efficiency curve as a function of throttle and rpm

“A SCIENTIFIC MEANS OF PREDICTING RACE PERFORMANCE”

helping the acceleration of the intake air. It is at this rpm that the maximum torque occurs.

Integration of the cylinder pressure over 720 degrees for each rpm step leads to an engine torque diagram for the complete rpm range. Power is calculated from torque and rpm. In figure 10, ignition angle is also plotted (in green). The example engine has a relatively high CR of 14.8. Therefore, ignition advance is relatively low. Since the model is also functional at partial throttle, it is capable of generating a complete 3D curve of torque, as shown in figure 11. This predicted torque curve can be used in the lap simulation model to calculate a lap time when this engine is mounted in a specified car.

Figure 12 shows the engine efficiency. This may be used to determine an initial fuel map, and thereby also predict the fuel used over a lap. The last figure shows the ignition angle of the engine for the complete rpm/throttle range.

Conclusion

At Bosch Motorsport we feel we have created a simulation program that will automate initial tuning maps as well as enabling a scientific means of predicting race performance.

The accuracy of the model has been developed and tested against known engines giving very good results. Not only were horsepower figures checked, but also the behaviour at much lower revs, even checking idle rpm. Furthermore, partial load results of the model: ignition timing; max cylinder pressures; ignition reduction curves; manifold pressure etc. were compared with real engines, giving very good correlation.

However, it should be remembered that any simulation program is only a model of reality. A

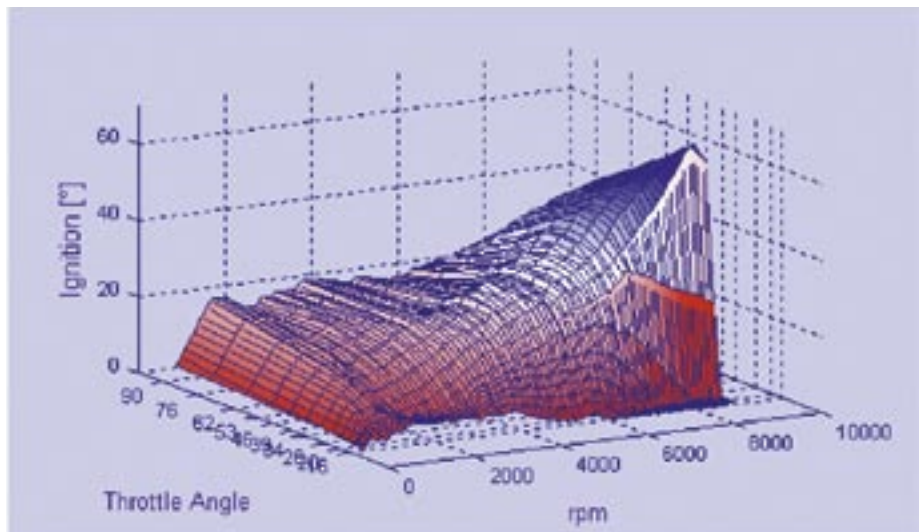


Figure 13: ignition angle as function of throttle and rpm


simplification by definition. Complex design variables are omitted for the sake of user ease and computational efficiency. The initial target was 95 per cent accuracy, but the model often exceeded this figure considerably, whilst still computing very rapidly. A full throttle curve up to 10,000rpm will take about 10 seconds on an average computer and a 3D map up to 10,000rpm will take little over two minutes to calculate.

By simulating the engine, both fuel consumption and lap times can be estimated (time spent in the pits refuelling can be added to that required to complete the race distance). A feasible assessment of which direction an initial design or development program might go is therefore possible.

Furthermore, the software easily enables development analysis. Different camshaft timing, with or without changes in inlet/exhaust layout, might lead to faster lap times, or better efficiency on certain tracks. As is widely known, ultimate horsepower does not always generate the fastest lap times. Achieving this is a complex puzzle, because each engine characteristic should be

paired with appropriate gearbox ratios.

Fortunately, testing time is almost unlimited on the computer, at greatly reduced cost, and might just offer that leading edge to win the race.

With the current state of the model, the next step is not too far away. Having the main fuel and ignition engine maps, one could imagine Bosch software generating a base calibration set for the ECU. It would then give the user a clear head start and could prove to be the ultimate in handling ease for the requirements in motorsport. 

You can try out the software yourself by visiting the Bosch Motorsport website where the engine module is also part of the free download of LapSim, the chassis simulation package. Just pay a visit to www.bosch-motorsport.com and click on Catalogues and software in the Publications and downloads section of the blue panel on the left of the homepage. The fully functioning version sells for 5300 Euros.