Lotus Engineering Software –

An Approach to Model-Based Design

R.J. Pearson, M.D. Bassett, N.P. Fleming, and T. Rodemann

Lotus Engineering

Hethel, UK

Abstract

Model-based design processes are key to the reduction of time scales for new vehicle and engine programmes. Lotus Engineering has developed a suite of simulation tools which enable shorter product analysis cycles through the facility to build models rapidly and take the user within close proximity of the final design. The software suite includes Lotus Vehicle Simulation, Lotus Engine Simulation, and Lotus Concept Valve Train. These tools can be used in harness with ADAMS/Engine via direct data links. The present work outlines the application of Lotus Engineering Software in a project environment and identifies their complementary relation to ADAMS packages.

1. Introduction

Computer simulation is now an inextricable component of many automotive engineering projects. At the start of the development of a new engine extensive optimisation is performed using performance simulation and base-engine analysis software which directly drives the prototype design function. The commercially availability of the analysis codes constituting Lotus Engineering Software have arisen from the successful use of these programs on many powertrain and vehicle projects at Lotus over the past 15 years. The philosophy under-pinning Lotus Engineering Software is to offer simulation tools which enable the user to generate models very quickly, using a mixture of embedded design criteria and well-structured interface functionality. Templates are available to guide the user through the model building process. The resulting models can be refined and are then checked at the point of job submission, generating a quality assessment summary for the model.

The three codes currently offered by Lotus Engineering Software are: Lotus Vehicle Simulation, Lotus Engine Simulation, and Lotus Concept Valve Train. Fig. 1 shows the inter-relation of these three codes and how they interact with ADAMS/Engine.

Lotus Vehicle Simulation can be used to specify the torque curve and gearbox specification required to produce a given vehicle performance. Having established a target engine torque curve Lotus Engine Simulation can be used to define the bore / stroke ratio, valve sizes, cam profiles, and intake and exhaust manifold geometry which enable the powertrain unit to meet the performance target. With this first phase of the engine simulation complete the resulting basic engine dimensions, cam period, valve lifts and cylinder gas pressure loading can be fed into the appropriate software for initial component sizing. A starting point for camshaft profile definition is obtained using Lotus Concept Valve Train. This data can be used to set up a valve train sub-system template for use in ADAMS/Engine Valve Train where a full valve train system model can be constructed.

The analysis process is clearly iterative. It may be found that the valve lift targets set by the engine simulation work cannot be met using the cam period specified. A compromised profile will be produced and this will be re-run through Lotus Engine Simulation in order to assess any likely performance detriment. Packaging problems may dictate the re-design of the engine manifolds – this entails further use of Lotus Engine Simulation and may require re-optimisation of cam profiles. Once a physical prototype is built all the analysis codes may be used in subsequent development stages of the project. The actual torque curve achieved by the engine can be fed back in to the vehicle simulation work so that the vehicle performance prediction can be revised. Detailed fuel consumption and emissions maps covering the entire engine load and speed range can be measured from the engine at this stage and used to perform drive-cycle analyses in Lotus Vehicle Simulation.

This paper centres on the role of Lotus Engine Simulation within the framework of a powertrain project. Brief descriptions of the role of Lotus Vehicle Simulation and Lotus Concept Valvetrain in this process will also be presented.

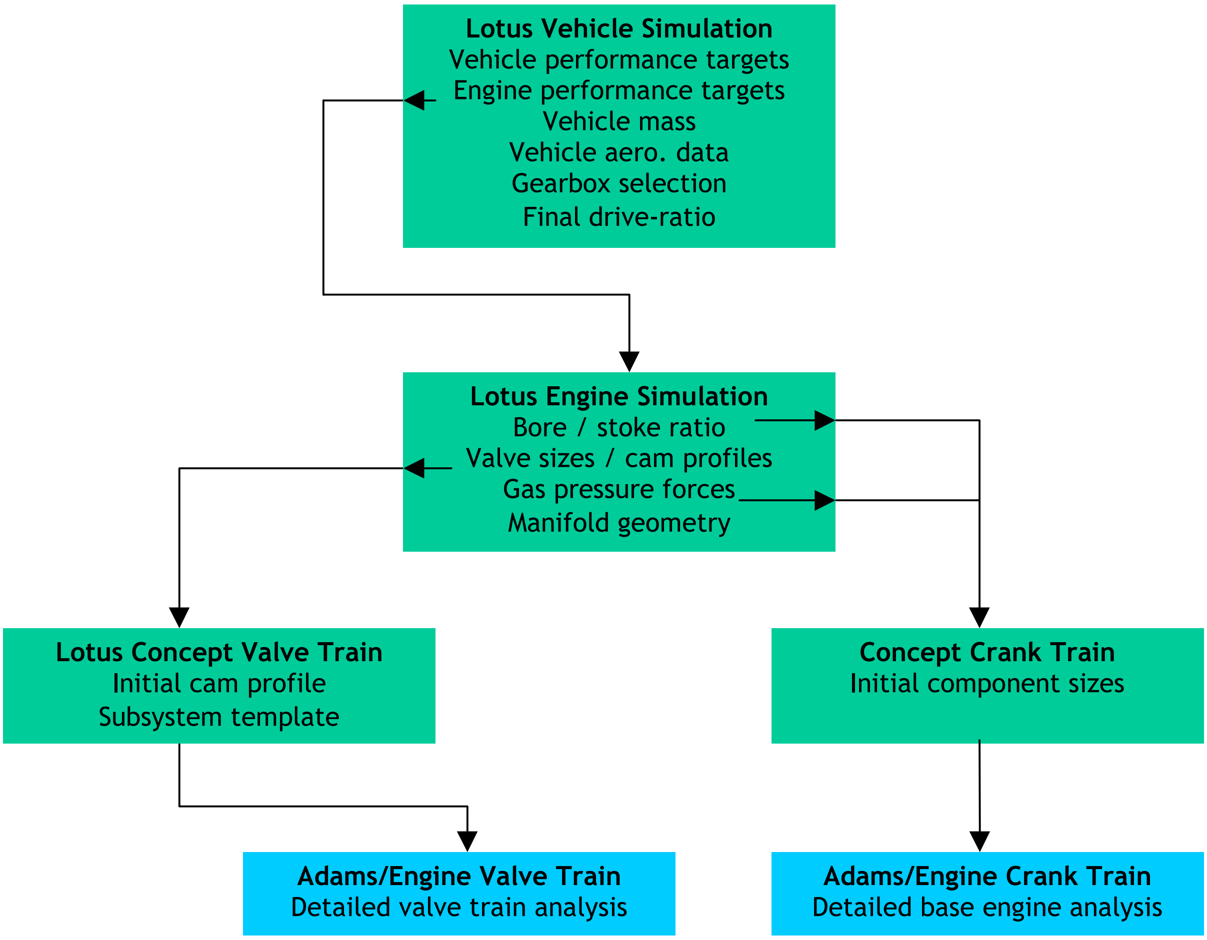

Fig. 1. Inter-relation of Lotus and ADAMS powertrain analysis codes.

2. Vehicle Performance Simulation

At the outset of a vehicle or powertrain engineering project the vehicle performance targets are agreed upon. These targets often specify the so-called 'headline' figures for the car which include the 0-100 km/hr and 'in-gear' acceleration times, and the maximum speed of the vehicle. Lotus Vehicle Simulation enables the investigation of all the key factors affecting the vehicle performance, such as mass, drag coefficient, gear ratios, or engine torque at a given speed. Template-based models of the vehicle architecture can be constructed, as shown in Fig. 2. This process can be assisted by using the Database Wizard, shown in Fig. 3, which makes available a library of component built up by the user.

Note that a full engine performance map is shown in Fig. 2, and this data is necessary as input if vehicle drive-cycle simulations are to be performed. For basic performance prediction tasks, however, the wide-open throttle torque curve will suffice. This reduced data requirement provides the opportunity for torque curve specification required to meet vehicle performance targets as the data can simply be edited in a table of engine speed and torque, or brake-mean effective pressure (BMEP), and the performance of the vehicle re-evaluated in seconds. The data output screen of a vehicle acceleration simulation is shown in Fig. 4. This screen is updated dynamically during the simulation and the engine speed, vehicle speed, and selected gear can be observed during the event. The graph in the bottom-right-hand corner of the screen indicates which region of the torque curve is in use at any particular time.

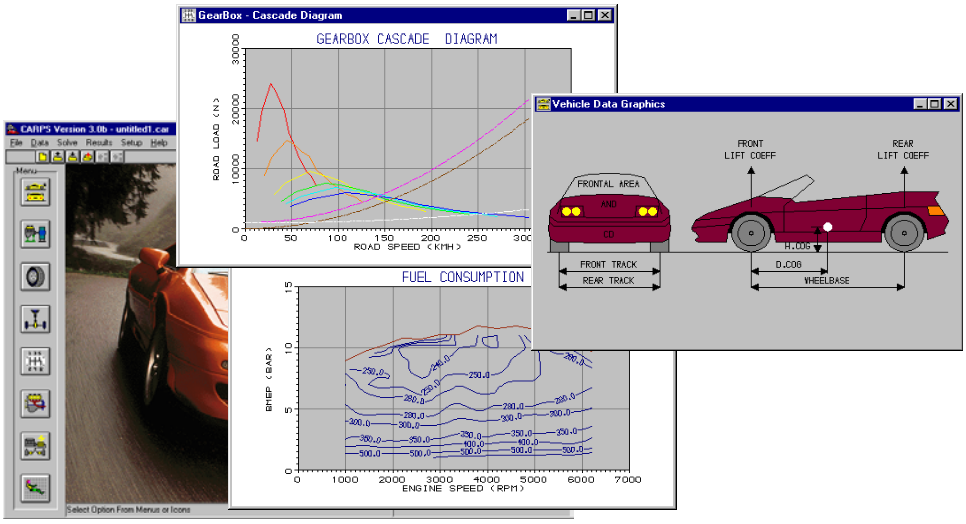

Fig. 2. Input data defining a model in Lotus Vehicle Simulation.

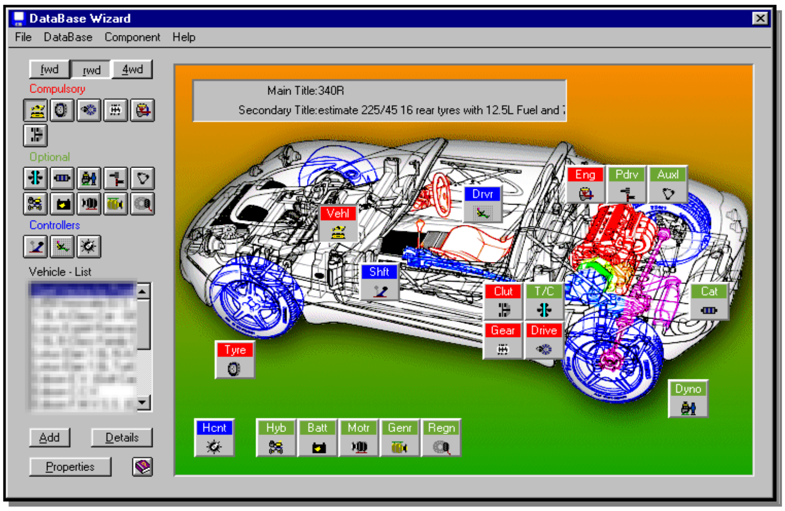

Fig. 3. Database Wizard in Lotus Engine Simulation.

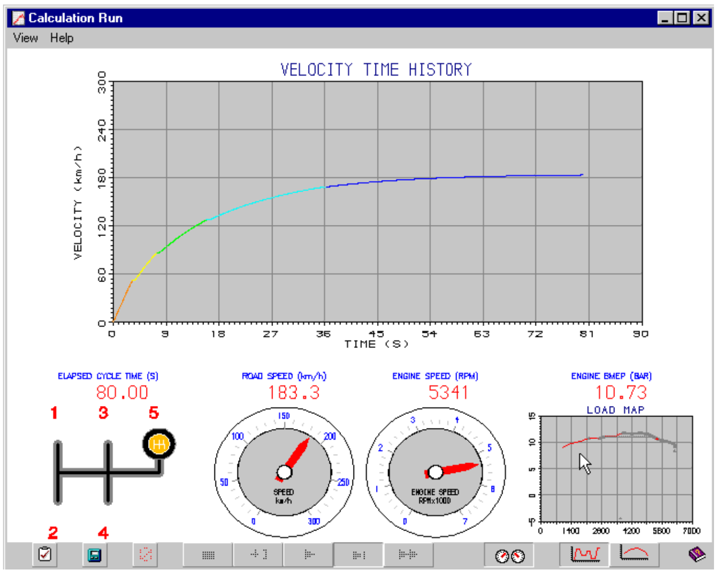

Fig. 4. Output data screen showing vehicle acceleration simulation.

The short run times of the code (complete drive-cycles run in a matter of secconds) make extensive parametric analysis a routine task. Fig. 5 shows the results, plotted directly in the simulation code, of a parametric study of vehicle 0-60 miles/hr. acceleration time as a function of vehicle drag coefficient and final drive ratio.

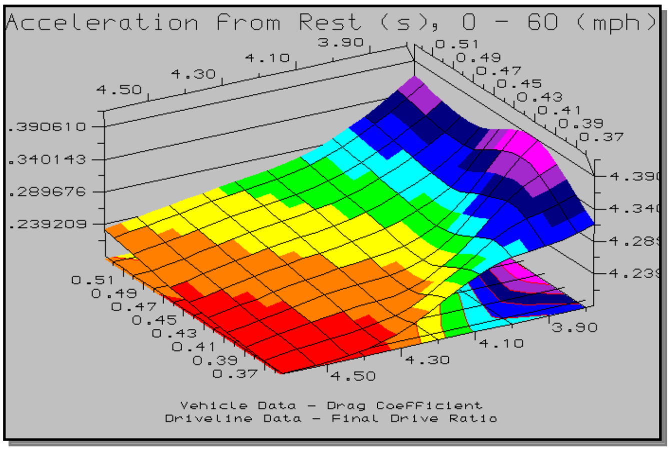

Fig. 5. Output from parametric study using Lotus Vehicle Simulation.

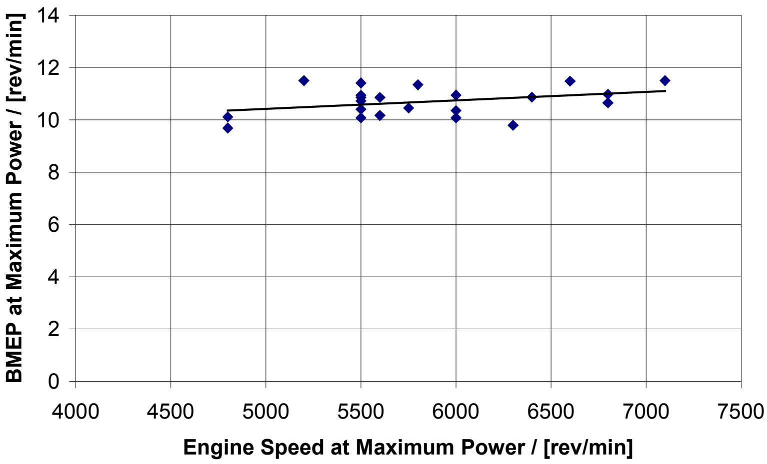

Fig. 6a. Variation of BMEP at maximum power with engine speed for naturally aspirated V6 gasoline engines.

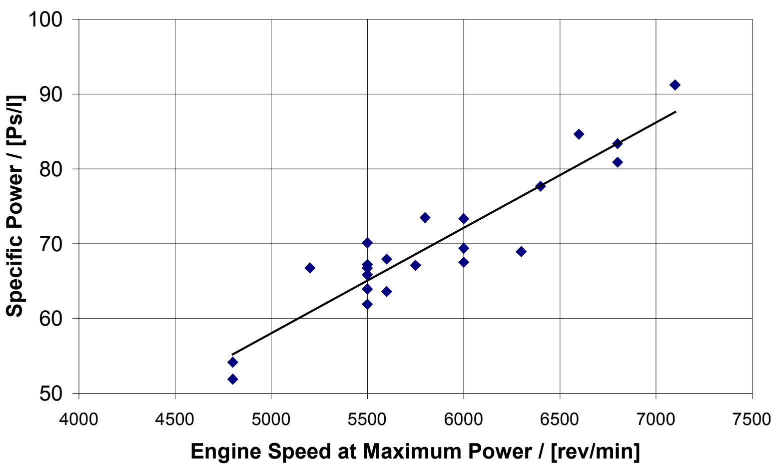

Fig. 6b. Variation of Specific power with engine speed for naturally aspirated V6 gasoline engines.

3. Engine Performance Simulation

3.1. Basic Parameters Defining Engine Performance

The basic architecture of an automotive engine is defined by its swept volume, the number of cylinders and configuration the engine is to have, and its rated power. Engine power [kW] is given by

$$\dot{W} = \frac{\overline{p}\, V_{\text{swept}}\, N}{1200}, \qquad (1)$$

where $\overline{p}$ is the brake mean effective pressure (BMEP) [bar], V_{swept} is the engine swept volume [litres] and N is the engine speed [rev/min]. The BMEP of an engine operating at a fixed air-fuel ratio (AFR) and given fuel specific heating value (Q_v), is proportional to the volumetric, or breathing, efficiency, the thermal efficiency, and the combustion efficiency, i.e.

$$\overline{p} \propto \frac{Q_v}{\text{AFR}} \eta_v \eta_{\text{th}} \eta_{\text{comb}} \quad . \qquad (2)$$

For naturally aspirated automotive spark-ignition engines the volumetric, thermal, and combustion efficiencies usually vary by relatively small amounts (assuming contemporary engines are being assessed) so that the BMEP levels at peak torque and peak power of current powertrains usually fall within a range of about 2 bar. It follows from equation (1) that the primary factor in determining the power output from an engine of a given swept volume is its maximum rated speed. Fig. 6a shows the BMEP at maximum power of a number of current naturally aspirated V6 gasoline engines. It can be seen that there is only a slight increase in BMEP with engine speed, whereas Fig. 6b shows that the specific power of the engines analysed increases significantly with engine speed.

Together with the number of cylinders and swept volume, the maximum rated speed defines the performance characteristic of the engine. These factors may be pre-ordained by the requirement to use an existing powertrain unit as a development platform or may be open to specification at the start of a project. In either case 'market placement' issues dominate the decision criteria.

3.2. Lotus Engine Simulation - The Model Building Process

Equations (1) and (2) show the relationship between the key factors controlling engine performance. Although it was stated in Section 3.1 that power is substantially determined by the maximum rated speed of the engine, the challenge is to maintain a high volumetric efficiency at this speed. The flows in the manifolds of reciprocating internal combustion engines are highly unsteady due to the propagation of pressure waves initiated during the cylinder charging and discharging processes. By modifying the pressure ratio upstream of the intake valves and downstream of the exhaust valves these pressure waves can have a significant effect on the filling and emptying of the cylinders, and hence the charge mass of air and fuel trapped in the cylinder. Careful design of the manifold systems and cam profiles enables the charge mass to be augmented across the engine speed range.

Lotus Engine Simulation provides a Concept Building Tool for naturally aspirated engines which enables a user to build a complete engine simulation model by specifying the three key parameters: number of cylinders, swept volume, and engine speed at maximum power. Having entered this data (in the three boxes highlighted at the top of the screen shot in Fig. 7) the engine configuration is selected from a number of templates and a simulation model can then be loaded into the interface of Lotus Engine Simulation. Screen shots illustrating the construction of a V6 engine model are shown in Fig. 7. Options for an in-line engine with open intake trumpets and a 'V' configuration with an intake plenum are shown as examples of the available templates – the V option was selected in the case shown. Similar options are available for the exhaust system. The entire model building process is completed in a matter of seconds.

In the main screen of the Concept Building Tool (top left-hand part of Fig. 7) all dimensions salient to the performance of the engine are defined, such as the bore / stroke ratio, the inlet runner pipe length, the inlet valve size, etc. The Concept Building Tool uses empirical formulae for making first estimates of component sizes: the Helmholtz resonator model is used for intake manifold tuning [1], and the Mach Index of Livengood et. al. [2] is used for analysing valve sizes. First approximations to suitable cam profiles are also selected. Users can over-ride these relationships in order to impose their own equations. Lotus design criteria are used in order to provide starting values for parameters such as the engine bore and stroke, and to impose data checking when values are over-ridden in order to adhere to a pre-defined project 'hard-point'. Parameters can be fixed by applying 'locks' to the data which are activated by clicking on the 'pad-lock' icons to the left of the text boxes in Fig. 7. The data checking is affected by highlighting in red any values which fall outside Lotus recommended practice.

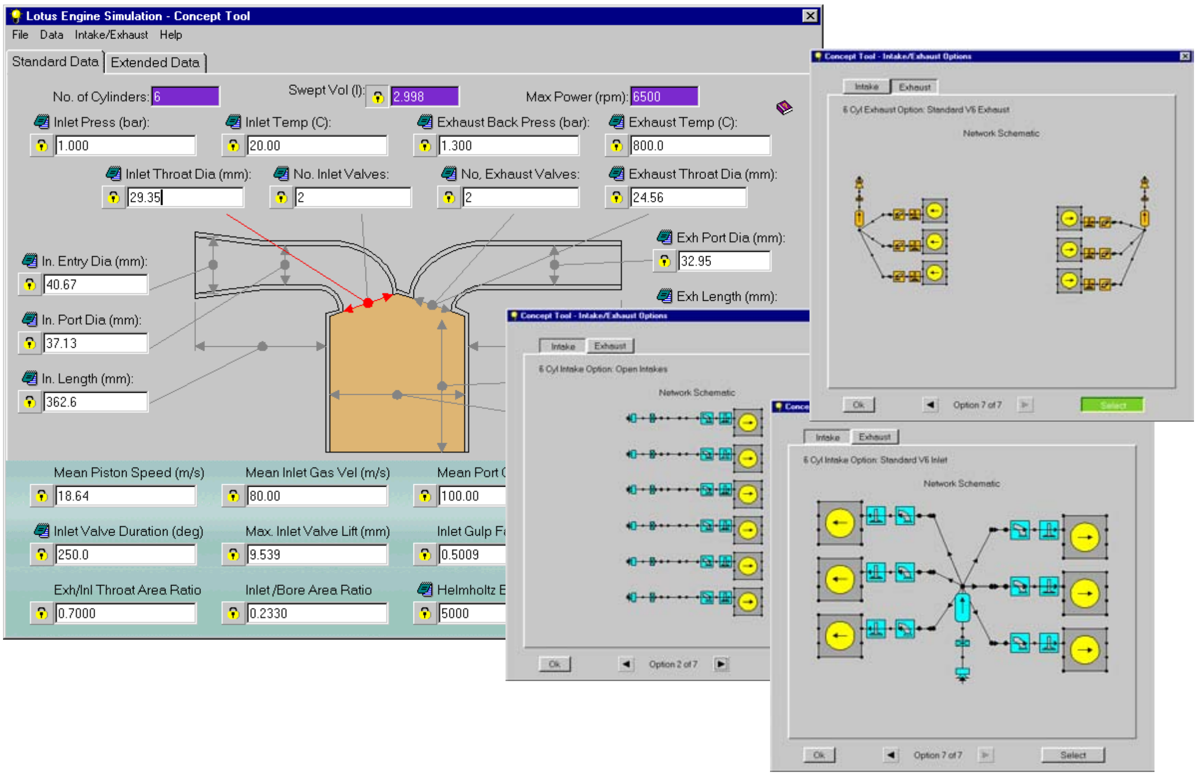

Fig. 7. Concept Building Tool in Lotus Engine Simulation

Once the basic configuration of the engine has been defined, the model can be loaded directly into Lotus Engine Simulation. During this process two engine test points are defined which the simulation is to run at the wide-open-throttle (WOT) load condition: the first point is an estimate of the peak torque speed of the engine and the second point is defined by the maximum power speed specified as input. Alternatively, a full 'power curve' can be set up using the Test Conditions Data Wizard which sets up the operating speed range to be simulated, specifying the air-fuel ratio, and ambient conditions.

Models can also be put together using the drag-and-drop model-builder environment in which any pipe network system can be constructed from individual components available from a 'Tool Kit'. This environment is often used to refine models produced by the Concept Building Tool by a adding elements representing catalysts or silencers.

3.3. Lotus Engine Simulation – The Solution Process

For engine simulation codes to capture the pressure wave phenomena described in Section 3.2 they must be capable of resolving both the spatial and temporal variation in gas properties in the manifold systems. The hierarchy of models available range from three-dimensional models with some mode of characterising the effects of turbulence, to filling and emptying models that represent the engine manifolds as lumped volumes. An effective compromise, in terms of simulation speed and accuracy, is to assume the flow is compressible and quasi-one-dimensional. This type of model considers the effects on the flow of the variation of cross-sectional area (F) along the axis of the pipes, as shown in Fig. 8, and includes the effects of pipe wall friction and heat transfer as source terms. The governing equations then become

continuity

$$\frac{\partial(\rho F)}{\partial t} + \frac{\partial(\rho u F)}{\partial x} = 0\,; \qquad (3)$$

momentum

$$\frac{\partial(\rho u F)}{\partial t} + \frac{\partial\left(\rho u^2 + p\right)F}{\partial x} - p\frac{\mathrm{d}F}{\mathrm{d}x} + \frac{1}{2}\rho u^2 f \pi D = 0\,; \qquad (4)$$

energy

$$\frac{\partial(\rho e_0 F)}{\partial t}+\frac{\partial(\rho u h_0 F)}{\partial x}-q\rho F=0\,. \tag{5}$$

These relationships constitute a set of non-linear hyperbolic partial differential equations. They can be written in vector form as

$$\frac{\partial \mathbf{W}}{\partial t}+\frac{\partial \mathbf{F(W)}}{\partial x}+\mathbf{C}=0, \tag{6}$$

where

$$\mathbf{W}=\begin{bmatrix}\rho F\\ \rho u F\\ \rho e_0 F\end{bmatrix},\qquad \mathbf{F(W)}=\begin{bmatrix}\rho u F\\ (\rho u^2+p)F\\ \rho u h_0 F\end{bmatrix},\qquad \mathbf{C}=\begin{bmatrix}0\\ -p\dfrac{\mathrm{d}F}{\mathrm{d}x}\\ 0\end{bmatrix}+\begin{bmatrix}0\\ \rho G F\\ -\rho q F\end{bmatrix}. \tag{7}$$

In these equations the pipe wall friction factor is included in the G term as

$$G=\frac{1}{2}u|u|f\frac{4}{D}, \tag{8}$$

and the term $u|u|$ is used to ensure that the pipe wall friction always opposes the fluid motion.

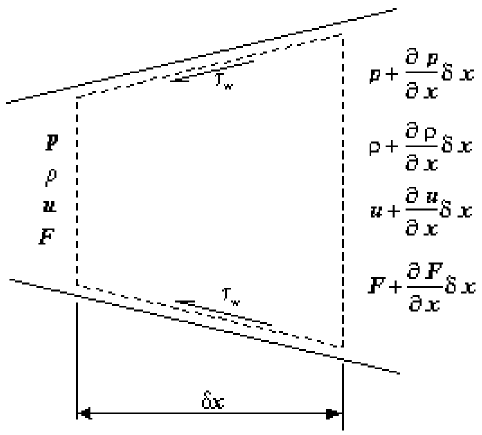

Fig. 8. Control volume for flow in a duct.

The numerical method used in the *Lotus Engine Simulatio*n program is based on the two-step Lax-Wendroff scheme, used in conjunction with a symmetric non-linear flux limiter, giving second-order spatial and temporal accuracy. This scheme is a member of the class of shock-capturing finite difference schemes which are capable of handling shock waves and super-sonic flows that can occur in the manifolds of high-performance engines [3]. The flux limiter, which is based on the total variation diminishing (TVD) criterion (TVD) (see later), helps to prevent the occurrence of spurious oscillations in the solution when shock waves and contact discontinuities are encountered.

The two-step Lax-Wendroff method is a space-centred scheme based on the computational stencil shown below in Fig. 9.

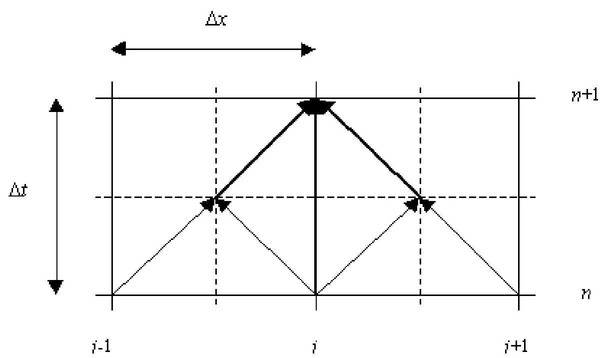

Fig. 9. Computational stencil for two-step Lax-Wendroff scheme.

The first step of the scheme uses a space-centred differences about the points $[(i+1/2)\Delta x,n\Delta t]$ and $[(i-1/2)\Delta x,n\Delta t]$ whilst the second step is a calculation which uses a time difference centred about the point $(i\Delta x,\ (n+1/2)\Delta t)$. Thus the scheme can be expressed in the form

$$\mathbf{W}_{i+1/2}^{n+1/2} = \frac{1}{2}\left(\mathbf{W}_{i+1}^{n} + \mathbf{W}_{i}^{n}\right) - \frac{\Delta t}{2\Delta x}\left(\mathbf{F}_{i+1}^{n} - \mathbf{F}_{i}^{n}\right) - \frac{\Delta t}{4}\left(\mathbf{C}_{i+1}^{n} + \mathbf{C}_{i}^{n}\right); \qquad (9)$$

$$\mathbf{W}_{i+1/2}^{n+1/2} = \frac{1}{2}\left(\mathbf{W}_{i+1}^{n} + \mathbf{W}_{i}^{n}\right) - \frac{\Delta t}{2\Delta x}\left(\mathbf{F}_{i+1}^{n} - \mathbf{F}_{i}^{n}\right) - \frac{\Delta t}{4}\left(\mathbf{C}_{i+1}^{n} + \mathbf{C}_{i}^{n}\right) \qquad (10)$$

and

$$\mathbf{W}_{i}^{n+1} = \mathbf{W}_{i}^{n} - \frac{\Delta t}{\Delta x}\left(\mathbf{F}_{i+1/2}^{n+1/2} - \mathbf{F}_{i-1/2}^{n+1/2}\right) - \frac{\Delta t}{2}\left(\mathbf{C}_{i+1/2}^{n+1/2} + \mathbf{C}_{i-1/2}^{n+1/2}\right). \qquad (11)$$

The Godunov Theorem [3] states that all second-order schemes having *constant coefficients* will generate spurious oscillations at discontinuities such as shock waves and contact surfaces. This obstacle to the development of numerical methods for hyperbolic equations can be circumvented by the construction of *non-linear* difference schemes in which the coefficients of the scheme are functions of the solution itself. One approach to constructing non-linear difference schemes is based on the total variation diminishing (TVD) criterion which is a measure of the variation of the solution at any given time step, given by

$$\mathrm{TV}\left(\mathbf{W}^{n}\right) = \sum_{i}\left|\mathbf{W}_{i+1}^{n} - \mathbf{W}_{i}^{n}\right|. \qquad (12)$$

In order to prevent the occurrence of spurious oscillations the total variation of the solution must satisfy the condition

$$\mathrm{TV}\left(\mathbf{W}^{n+1}\right) \le \mathrm{TV}\left(\mathbf{W}^{n}\right). \qquad (13)$$

This criterion can be utilised in a numerical scheme in the form of a ‘smoothness monitor’ which tests the sign of consecutive gradients of the solution between pipe meshes.

The two-step Lax-Wendroff scheme can be modified to fulfil the TVD criterion by appending the term

$$\left[\overline{\mathbf{G}}_{i+1/2}^{+}\left(r_{i}^{+}\right) + \overline{\mathbf{G}}_{i+1/2}^{-}\left(r_{i+1}^{-}\right)\right]\Delta\mathbf{W}_{i+1/2}^{n} - \left[\overline{\mathbf{G}}_{i-1/2}^{+}\left(r_{i-1}^{+}\right) + \overline{\mathbf{G}}_{i-1/2}^{-}\left(r_{i}^{-}\right)\right]\Delta\mathbf{W}_{i-1/2}^{n} \qquad (14)$$

after the second-step (equation (11)) where

$$\overline{\mathbf{G}}^{\pm}(r_{i}^{\pm}) = \frac{1}{2}C(v)\left[1 - \phi(r_{i}^{\pm})\right]; \qquad (15)$$

and

$$\left.\begin{aligned} r_{i-1}^{+} &= \frac{\left[\Delta\mathbf{W}_{i-3/2}^{n}, \Delta\mathbf{W}_{i-1/2}^{n}\right]}{\left[\Delta\mathbf{W}_{i-1/2}^{n}, \Delta\mathbf{W}_{i-1/2}^{n}\right]}, r_{i}^{-} = \frac{\left[\Delta\mathbf{W}_{i-1/2}^{n}, \Delta\mathbf{W}_{i+1/2}^{n}\right]}{\left[\Delta\mathbf{W}_{i-1/2}^{n}, \Delta\mathbf{W}_{i-1/2}^{n}\right]} \\ r_{i}^{+} &= \frac{\left[\Delta\mathbf{W}_{i-1/2}^{n}, \Delta\mathbf{W}_{i+1/2}^{n}\right]}{\left[\Delta\mathbf{W}_{i+1/2}^{n}, \Delta\mathbf{W}_{i+1/2}^{n}\right]}, r_{i+1}^{-} \frac{\left[\Delta\mathbf{W}_{i+1/2}^{n}, \Delta\mathbf{W}_{i+3/2}^{n}\right]}{\left[\Delta\mathbf{W}_{i+1/2}^{n}, \Delta\mathbf{W}_{i+1/2}^{n}\right]} \end{aligned}\right\} . \qquad (16)$$

This approach to producing a symmetric TVD scheme was proposed by Davis [4,5]

The local Courant number is defined as

$$\nu = \max_{k}\left|\lambda_{k}\right|\frac{\Delta t}{\Delta x} \qquad (17)$$

where $C(\nu)$ is given by

$$C(v) = \begin{cases} v(1-v), & v \le 0.5 \\ 0.25, & v > 0.5 \end{cases}. \quad (18)$$

The flux limiter can be defined as

$$\phi(r) = \begin{cases} \min(2r, 1), & r > 0 \\ 0, & r \le 0. \end{cases} \quad (19)$$

This limiter constrains the Courant number of the scheme to 0.7.

The interface between the intra-pipe gas dynamic calculations and the boundary conditions is dealt with by using the Mesh Method of Characteristics [3,4].

Once the trapped mass of air, fuel, and exhaust residual gas in the engine cylinder has been calculated the next task is to simulate the energy release rate during the combustion process. Lotus Engine Simulation uses a 'heat release' model for this purpose, in which the mass fraction of burnt fuel at any instant is evaluated using a Wiebe function, or the burnt mass fraction / engine crank angle diagram can be entered directly. For concept studies default trend lines for the combustion phasing and duration are utilized so that the simulation is not overly constrained.

Heat transfer to the three major surfaces comprising the combustion chamber (cylinder head, piston, and liner) is evaluated from the difference between the instantaneous gas temperature value and the metal surface temperature. The latter value is inferred from a simple thermal network calculation. The convective heat transfer coefficients used in these calculations are derived from the well known semi-empirical relationships of Annand, Woschni, or Eichelberg [5].

Simulation of the energy release mechanism from the fuel and air mixture and the heat transfer processes enables the cylinder pressure during the combustion event to be predicted and thus a complete cycle simulation is performed. Thus gas pressure loads can be predicted and used for crank-train analysis using ADAMS/Engine. Lotus Engine Simulation can write gas pressure files in Teimorbit format which can be read directly into ADAMS/Engine.

3.4. Intake System Design and Cam Profile Specification

A six-cylinder engine of 2.998 litre swept volume, with a projected maximum power speed of 6500 rev/min was specified in the Concept Building Tool. Choosing the V-configuration options, as in Fig. 7, resulted in the model shown in Fig. 5, which is a screen shot from the Lotus Engine Simulation model-builder. This model connects all cylinders to a common intake plenum and has two separate simplified exhaust systems with plenum elements representing 'under-floor' catalysts.

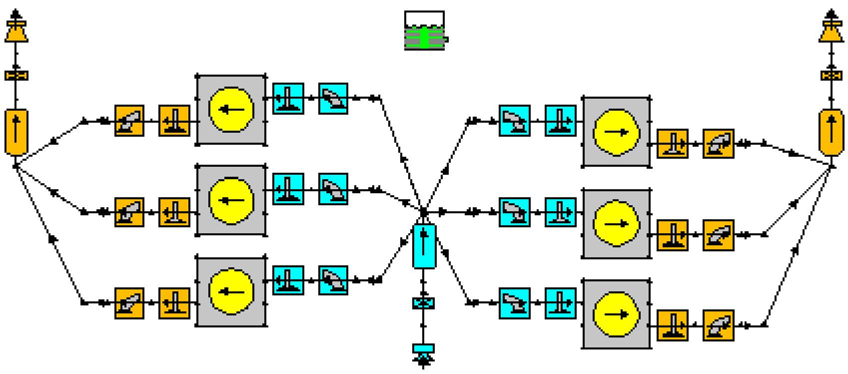

Fig. 10. Model of V6 engine generated by Concept Building Tool.

This section will centre of a limited 'optimisation' of the engine intake manifold and cam profile. An aspect of the intake system to which the engine performance is highly sensitive is the intake manifold runner length and the analysis will consider this parameter in tandem with the timing of the intake closing event, to which the engine performance is also sensitive. The model produced by the Concept Building Tool has inlet manifold runner lengths of 260 mm (in addition to the port length of 98 mm) and an intake valve closing angle of 58 degrees after bottom-dead-centre (ABDC). The volumetric efficiency and torque predicted by this model are shown in Fig. 11. It can be seen that the peak volumetric efficiency value is about 106 percent, giving a peak torque level of around 280 Nm at 5000 rev/min., this corresponds to a peak BMEP level of 11.7 bar. The peak power of the engine of 169.5 kW (227 bhp) is actually produced at 6000 rev/min.

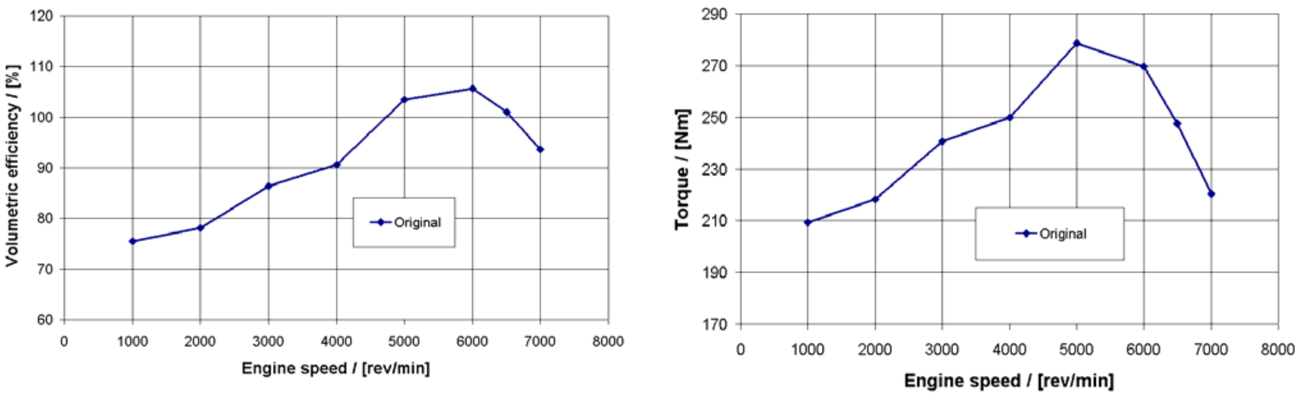

Fig. 11. Variation of volumetric efficiency and torque with engine speed for original model.

A parametric study of the inlet system was then conducted by varying the intake manifold runner length between 200 and 400 mm, in steps of 50 mm, and the intake valve closing timing between 30 and 70 degrees after bottom-dead-centre, in steps of 10 degrees. Thus a two-dimensional matrix of simulation points was established and a batch run which submitted all these variants automatically was then run from within Lotus Engine Simulation. Note that simple scaling rules will be applied to the intake cam profile in order to stretch and compress it as required in the parametric study. The detailed specification of the cam profile is carried out later on in the design programme.

Intake manifold runner length and the timing of the intake valve closing point have a strong effect on the engine volumetric efficiency, and hence the torque. Large variations in trapped mass, and residual gas concentration cause significant variations in the phasing and duration of the combustion event within a cylinder. Therefore it is prudent to study the response of the engine in terms of its volumetric efficiency during this type of analysis. Fig. 12 shows the spectrum of the volumetric efficiency curves from the 2-d parametric analysis. The curve depicting the results from the initial simulation run is highlighted in pink.

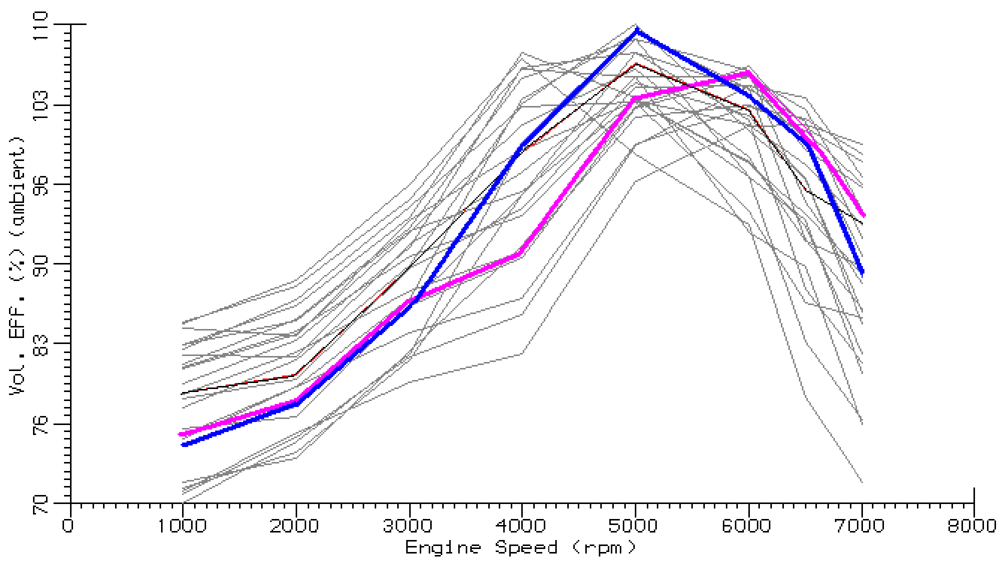

Fig. 12. Variation of volumetric efficiency with engine speed for 2-d parametric analysis.

An alternative way of presenting this data is to plot volumetric efficiency contours as a function of intake runner length and intake valve closing event, as shown in Fig. 13a, b, and c.

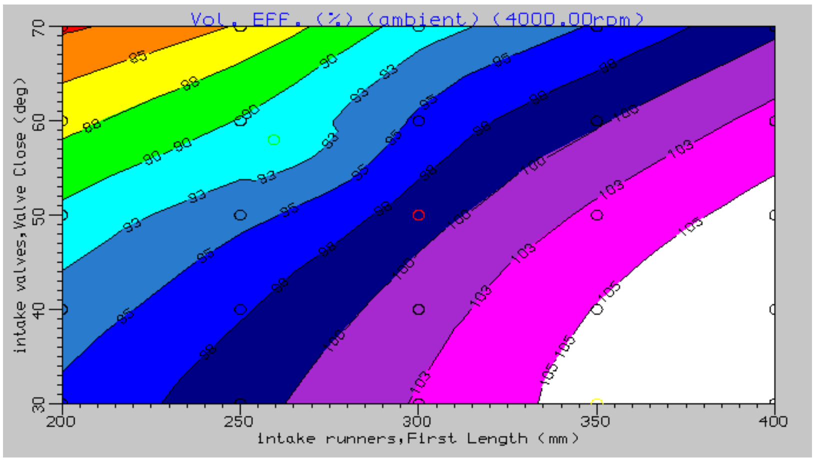

Fig. 13a. Volumetric efficiency contours at 4000 rev/min.

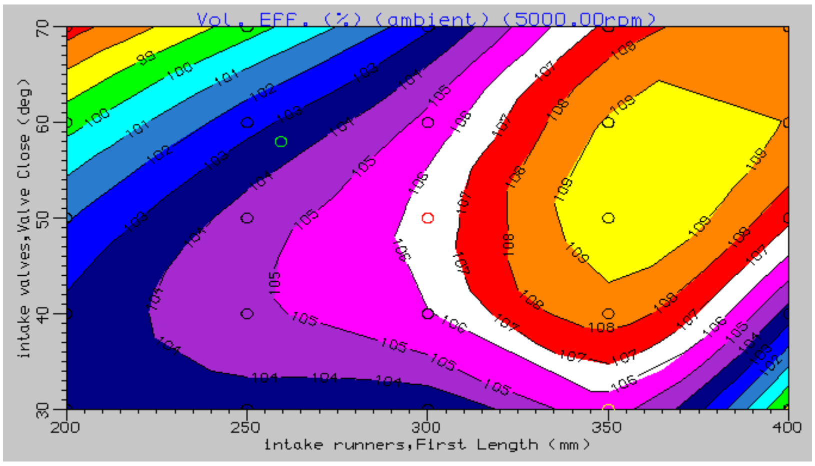

Fig. 13b Volumetric efficiency contours at 5000 rev/min.

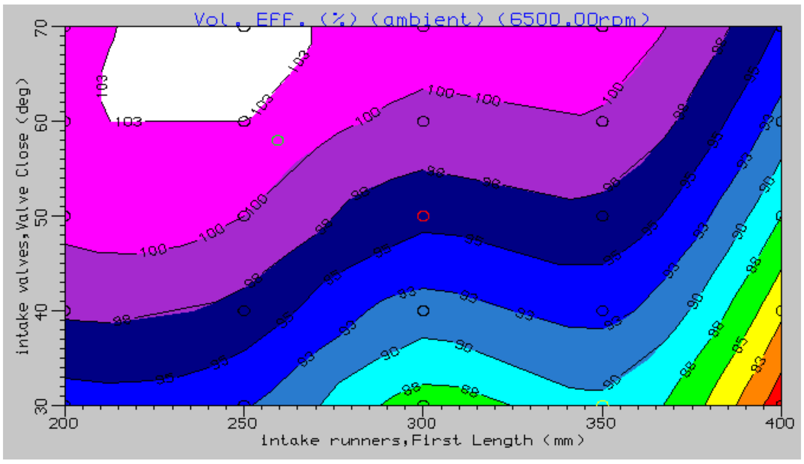

Fig. 13c Volumetric efficiency contours at 6500 rev/min.

Fig. 13a shows the volumetric efficiency contours at 4000 rev/min, plotted within the Parametric Tool in Lotus Engine Simulation. It is clear that, for this engine speed, the highest volumetric efficiency is obtained using an intake runner length of 400 mm and an intake valve closing timing of 30 degrees ABDC. Fig. 13b reveals that, at 5000 rev/min., the optimum combination of intake runner length and intake valve closing timing has changed to 350 mm and 50 degrees ABDC. Finally, at 6500 rev/min.,

Fig. 13c shows that the an intake runner length of 250 mm and an inlet valve closing angle of 60 or 70 degrees ABDC gives the highest volumetric efficiency. These latter values are approximately those chosen by the Concept Building Tool when it was set to size components to produce maximum engine power at 6500 rev/min.. Considering Figs. 13a to c simultaneously leads to the choice of an inlet runner length of 350 mm with an intake valve closing angle of 60 degrees ABDC as a good compromise which gives high volumetric efficiency at all the speeds considered. This combination of values produces the volumetric efficiency / speed curve highlighted in blue in Fig. 12. It can be seen that the volumetric efficiency has been increased substantially at 4000 and 5000 rev/min. without the values at 6000, 6500, and 7000 rev/min suffering to any significant degree.

The above procedure would constitute only a small part of a full engine optimisation process. The effects of runner cross-sectional area variation, intake valve lift and the intake valve opening timing have not been considered. The latter parameter can only investigated in conjunction with studying the exhaust cam profile. The timing of the exhaust valve opening event determines the compromise between expansion work and pumping work done by the engine. The intake valve opening and exhaust valve closing events determine the duration of the valve overlap period and this, in turn, needs to be set whilst optimising the exhaust system ‘header’ lengths which determine the phasing of the major pressure wave events used for tuning exhaust systems. An illustration has been provided, however, of how the process of specifying an intake cam profile is begun.

If the optimisation of the intake valve opening event and lift is ignored it can be assumed that the basic definition of the intake cam profile has been specified. These values can now be used within Lotus Concept Valve Train in order to begin the detailed design of the cam profile. This will be briefly described in Section 4.

Once the profile has been fixed the phasing of the intake cam shaft profile with respect to the position of its maximum opening point (MOP) must be optimised. Fig. 12 shows that the combination of intake runner length and intake valve closing timing chosen in the previous analysis in order to give high performance at relatively high engine speeds still gives a rather low level of volumetric efficiency at low speeds. At 2000 rev/min. the volumetric efficiency level is almost identical to that given by the original specification, at about 77%. This gives a torque level of only 218 Nm, which corresponds to a BMEP value of 9.1 bar. When a continuously variable cam phasing mechanism is specified the low speed torque of the engine can be enhanced by optimising the phasing of the cam profiles. This process can be undertaken using a 1-d parametric analysis.

Fig. 14 shows the results of changing the maximum opening point (MOP) of the cam profile from 90 degrees after top-dead-centre (atdc) to 125 degrees atdc. The arrows on the diagram indicate the direction of increasing MOP (later phasing). Early phasing of the cam profile boosts the engine volumetric efficiency in the lower and central regions of the speed range whilst late phasing of the cam results in greater volumetric efficiency levels at high engine speeds. This set of curves can be used to specify the position of the continuously variable cam-phasing device which gives the highest volumetric efficiency at each engine speed.

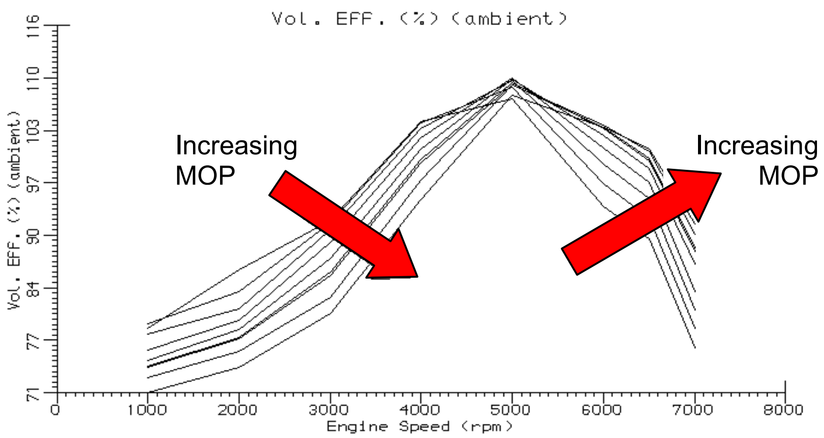

Fig. 14. Variation of volumetric efficiency with engine speed for various MOP values.

Fig. 15 indicates how the model shown in Fig. 10 can be modified to run through the engine speed range using the cam phasing values (MOPs) inferred from the results presented in Fig. 10. A sensor (the green icon) is attached to the engine cylinder to sense the crank speed. This device feeds its input signal to an ‘actuator’ (the yellow circular icon) which uses a 1-d look-up table to set the phasing of the cam. Only one actuator is required as the valve elements have been assigned to a common group. Engine speed values are contained in the ‘X’ column in the table shown in Fig. 15, and the corresponding MOP values are specified in the ‘Y’ column. The model will interpolate if speeds are run which do not coincide with those given in the table.

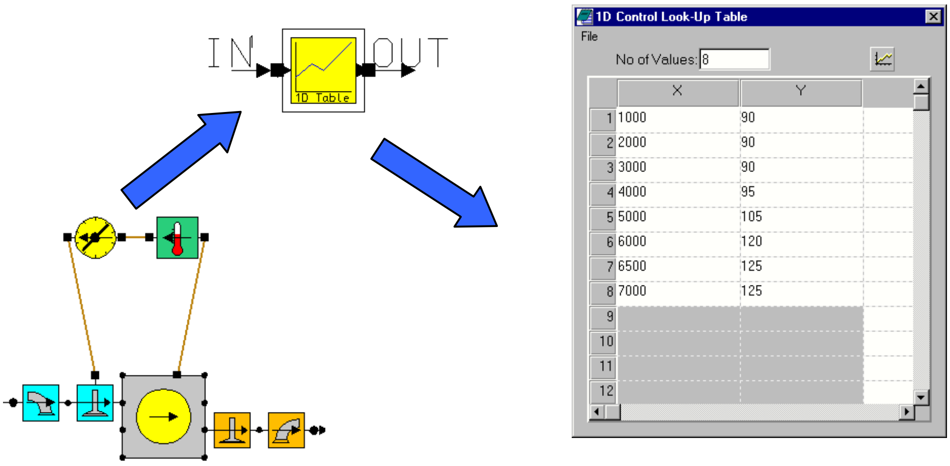

Fig. 15. Use of sensors and actuators in Lotus Engine Simulation to simulate an engine with a continuously variable cam phasing system.

In Fig. 16 the benefits of the overall optimisation can be seen. The blue curves show the predicted volumetric efficiency and torque of the original model generated by the Concept Building Tool in Lotus Engine Simulation. The red curves show the volumetric efficiency and torque values which result from running the model in which the optimised intake runner length and cam profile duration established in the 2-d parametric study have been used. In addition, the optimum cam phasing inferred from the 1-d parametric study is set at each speed point in the simulation. It is clear that even these limited studies have resulted in large improvements in the volumetric efficiency and torque curves. The peak torque level has increased from 280 Nm to 296 Nm at 5000 rev/min.. The value of torque at 4000 rev/min. in the optimised model also exceeds the peak value of the original model.

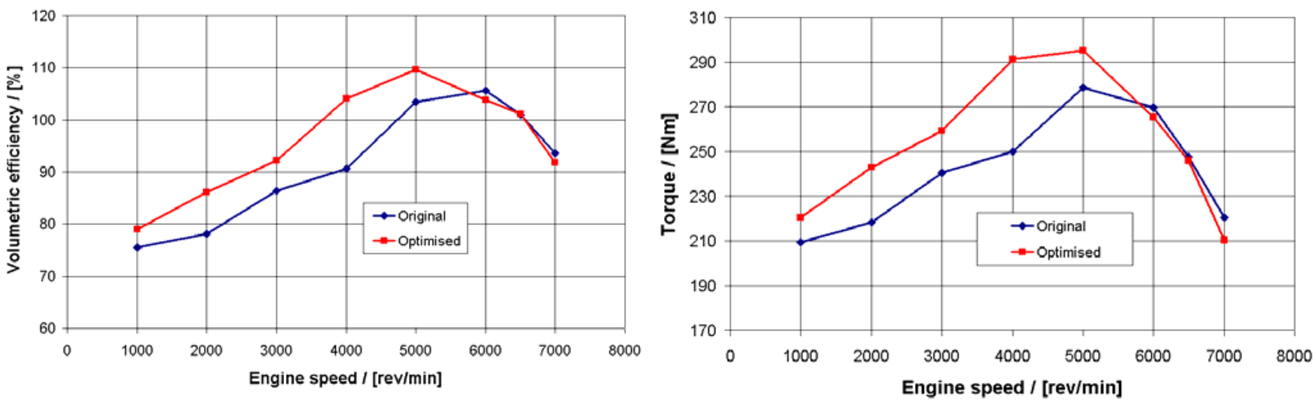

Fig. 16. Improvement of engine performance due to intake system optimisation process.

As part of the engine performance prediction Lotus Engine Simulation calculates the spatial and temporal variations in gas properties throughout the engine system included in the model. Figure 17 shows a contour plot of the pressure level in the model at an instant in the engine operating cycle at 5000 rev/min. The extent of the spatial variation of pressure throughout the engine system is clear from this figure.

The central cylinder of the right-hand bank in Fig. 17 is just beginning its intake stroke. The intake valve is open (it has turned white in the display) and the downward movement of the piston has created a depression in the cylinder (low pressure is indicted by the blue colour, high pressure by red). The suction in the cylinder has, in turn, has created a rarefaction wave in the intake runner pipe, which propagates into the intake system toward the plenum where it will be reflected as a compression wave which travels back towards the cylinder. When the system is well tuned this compression wave will arrive at the intake valve between bottom-dead-centre of the intake stroke and the point of inlet valve closing, as shown in Fig. 18. Thus the pressure ratio across the intake valve is increased in this crucial period of the induction process, and the trapped mass of air and fuel in the cylinder is augmented. This type of plot can be produced directly using the post-processing facilities within Lotus Engine Simulation and are of use in understanding the detailed gas dynamic mechanisms which effect the engine performance.

Cylinder pressure / crankangle data at any engine operating condition can be accessed via this mechanism and used to define the gas pressure loads as part of a mechanical system simulation of the engine in ADAMS/Engine Crank Train, as indicated in Fig. 1. The data can be produced in Teimorbit format which can be read directly by ADAMS/Engine. An example of a similar process using Lotus Concept Valve Train will be described in Section 4.

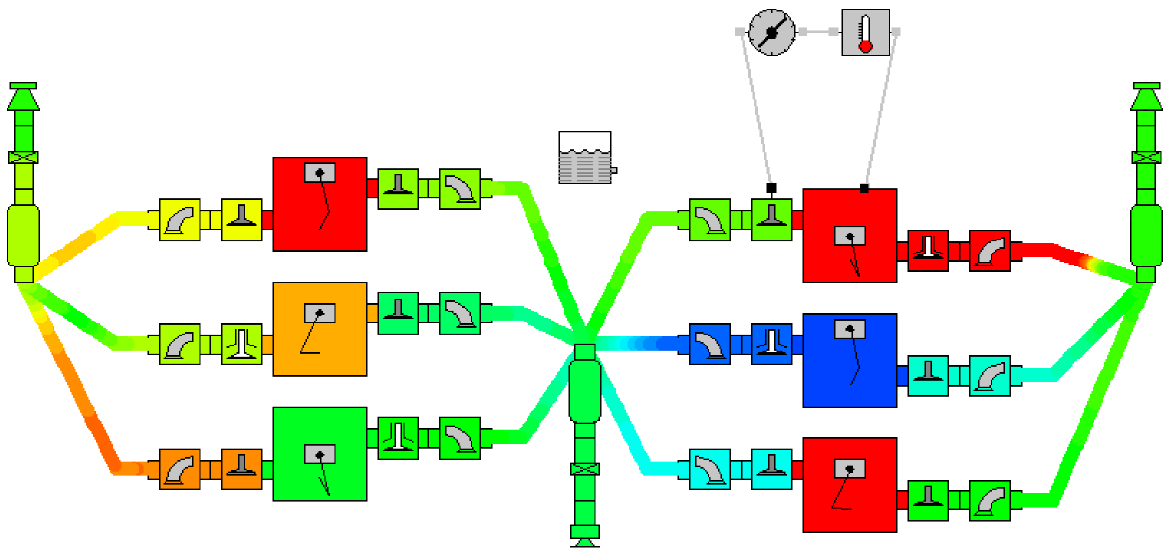

Fig. 17. Variation of pressure throughout the engine manifold systems at 5000 rev/min.

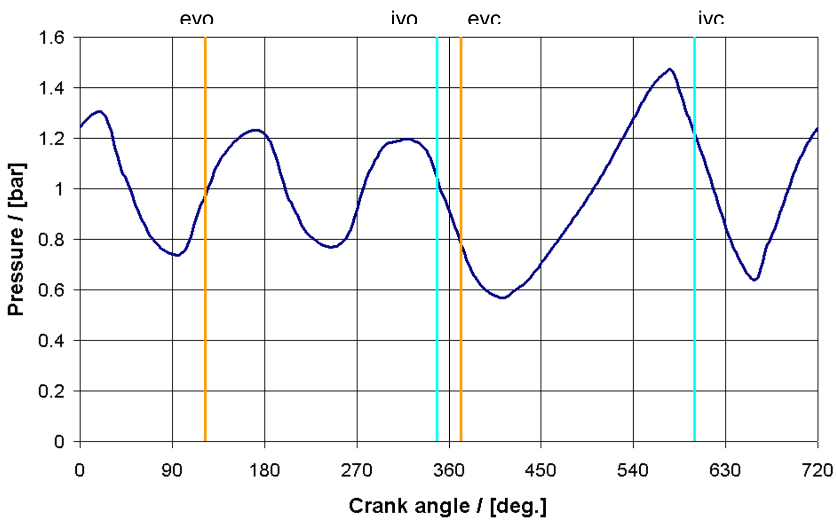

Fig. 18. Predicted variation of pressure with crankangle at the intake valves.

4. Cam Profile Design

In the previous section it was shown how it is possible to use Lotus Engine Simulation in order define the intake valve closing point, and hence the valve opening duration, required in order to meet a given engine performance target. A more extensive optimisation procedure would have encompassed varying the intake valve opening timing, the valve lift, and the interaction with the exhaust valve closing point, which determines the valve overlap period.

The Parametric Tool within Lotus Engine Simulation applies simple scaling rules to a basic valve lift profile but does not test the mechanical integrity of the valve train system on each iteration required to produce each profile. Having arrived at an optimum specification the design of the valve train system can be initiated. The first stage in this process is to specify the type of valve actuation mechanism to be used. Lotus Concept Valve Train incorporates four basic templates for direct-acting, finger-follower, centre-rocker, or push-rod systems. The selection of the actuation mechanism is made from the interface panel shown in Fig. 19. The polynomial function for the cam profile is also determined at this point.

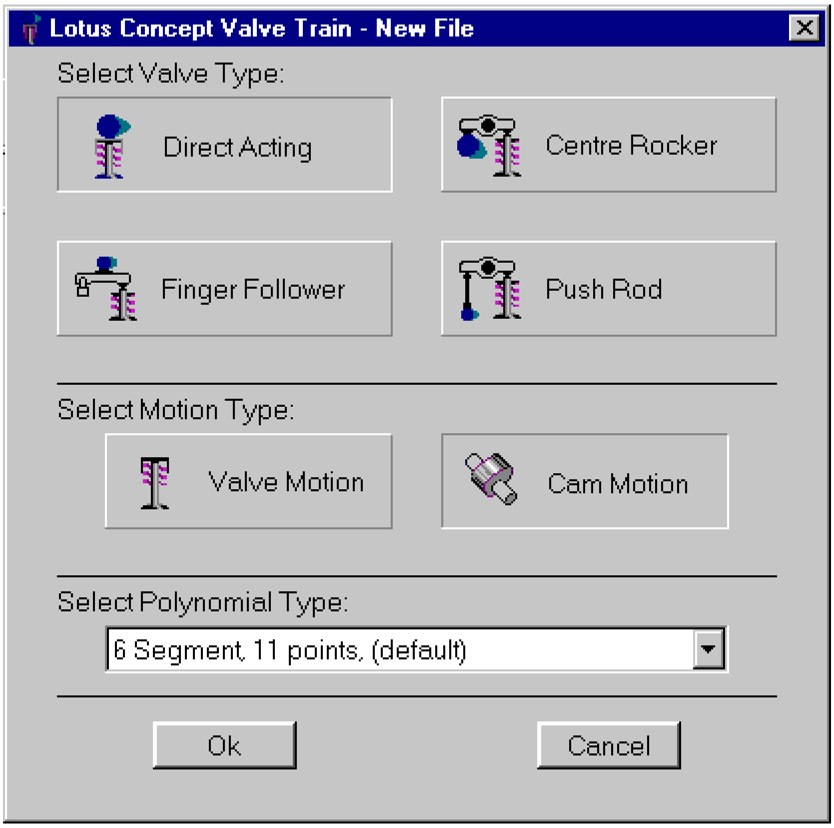

Fig. 19. Panel for selecting valve actuation mechanism in Lotus Concept Valve Train

Once the valve actuation type has been chosen the mechanism dimensions must be specified. Fig. 20 shows the mechanism for a push-rod system, drawn to scale, as displayed in the interface of Lotus Concept Valve Train. For the engine performance described in Section 3 a direct-acting or finger-follower system would probably be used. A push-rod system is described here as it comprises the most components, and is therefore the most visually interesting, of the mechanism types. The user can edit the component geometry, such as the valve angle, the push-rod length, or the cam base-circle radius, directly (as shown in Fig. 20) or via text boxes on the data property sheet. The co-ordinates of, for example, the camshaft centre and the follower pivot point are also specified here.

The cam profile itself must now be determined. For each valve actuation type a segmented polynomial curve is pre-defined for the cam profile in such a way that the user can modify its major features and impose additional constraints if required. The default option is to use a 6 segment profile with eleven point definition but this can be increased to 10 segments with 15 point definition. Each segment of the profile is constructed from a polynomial function, the order of which is determined by the number of boundary conditions imposed. Fig. 21 illustrates how the profiles are defined within Lotus Concept Valve Train. The profile can be made either symmetric or asymmetric. The maximum lift and event length values have been established using Lotus Engine Simulation. Specifying the ramp heights, on the opening and closing flanks, the ramp velocities, and the ramp-period, fully defines the profile.

The four graphs in Fig. 21 show the complete displacement curve for both the cam surface and the valve lift of this push-rod system. The first three derivatives, namely the velocity, acceleration, and jerk, of these displacement curves are also shown. Updating any of the values in the text boxes immediately updates the graphs. The graphs can also be edited directly from their graphics boxes.

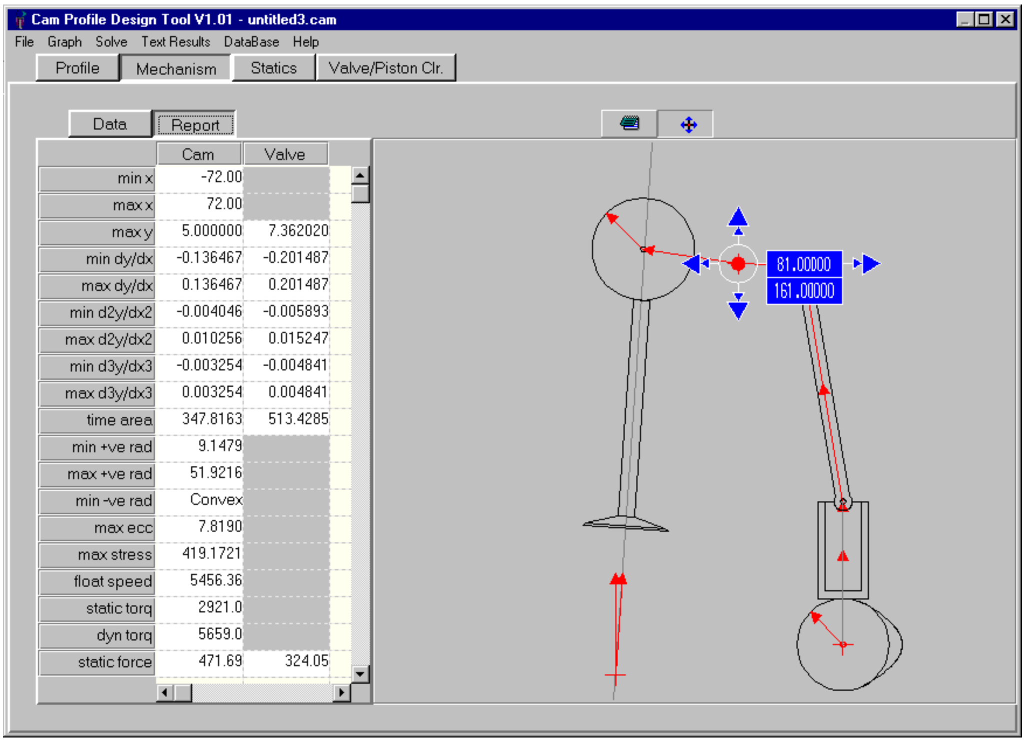

Fig. 20. Mechanism definition in Lotus Concept Valve Train

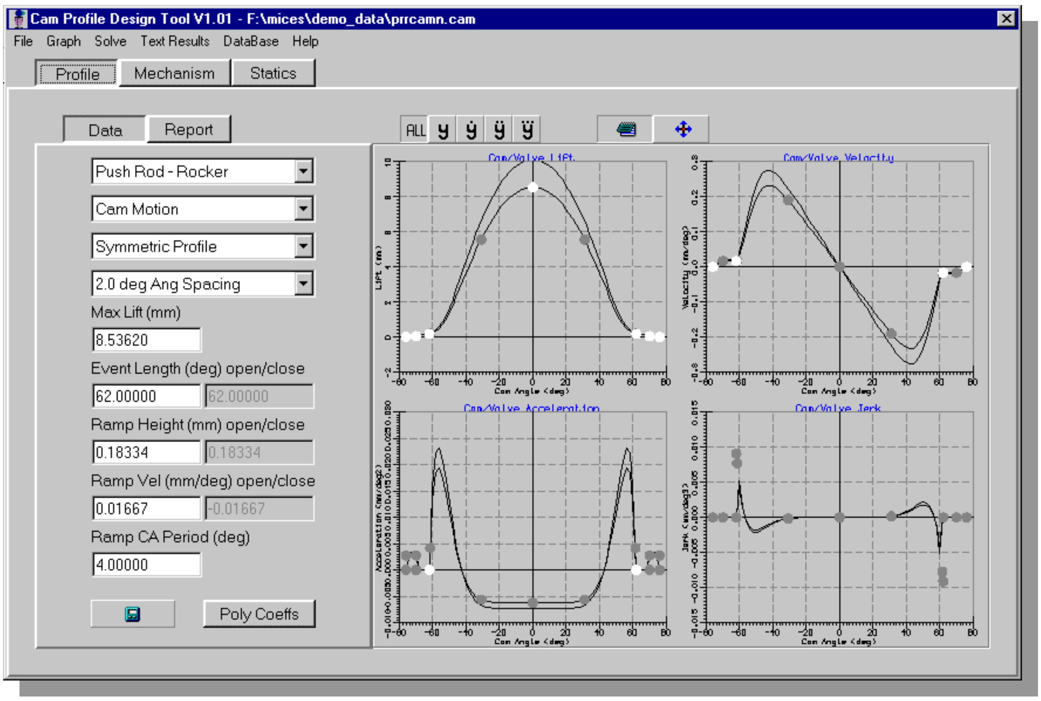

Fig. 21. Cam profile definition in Lotus Concept Valve Train

A kinematic analysis of the motion of the valve train mechanism is performed by Lotus Concept Valve Train – this enables the rapid assessment of systems via interactive design where the user can see the results of geometry or profile changes immediately. In order to make initial predictions of camshaft drive torques, eccentricity, contact stress values, oil film thickness data such as spring rates and pre-loads, the system effective mass, material properties and the coefficient of sliding friction is required.

Fig. 22 shows how the results of this kinematic analysis are displayed within the interface. A System Report is also produced, an example of which is shown on the left-hand side of Fig. 20. If the design of any aspect of the valve train system is outside the range of accepted practice the relevant text box is highlighted in red in the System Report.

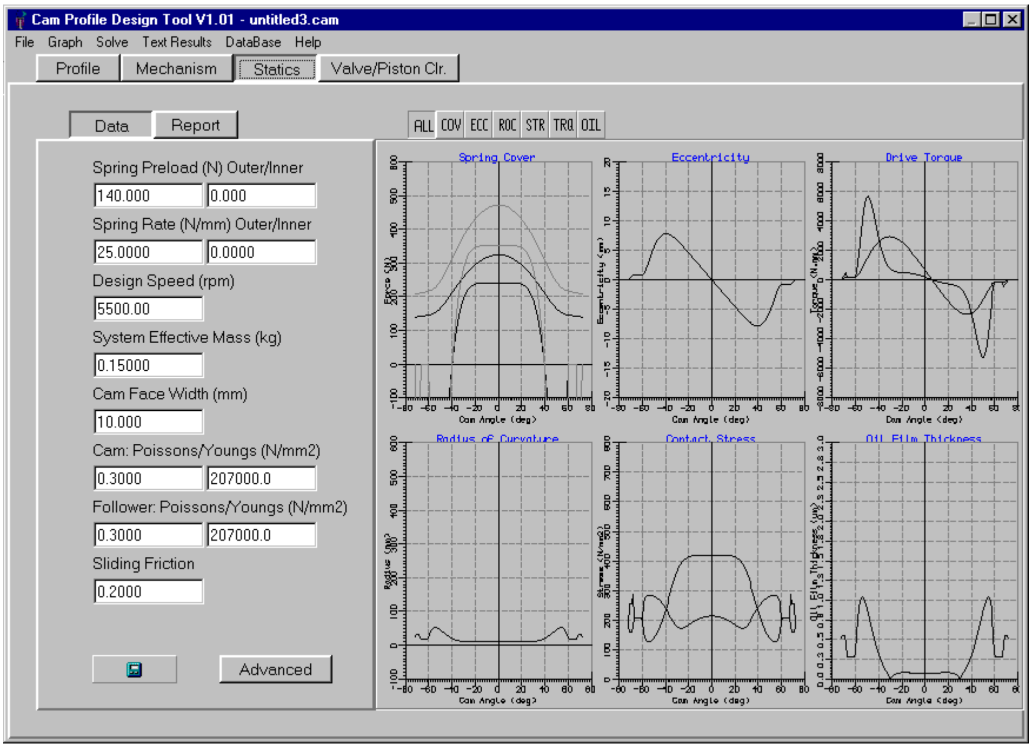

Fig. 22. Static analysis results in Lotus Concept Valve Train

The static analysis described above includes spring characteristics in order to calculate the spring loads on the mechanism. These can be entered as known values, estimates, or can be calculated using the Spring Design Tool in Lotus Concept Valve Train. This facility allows the user to design and analyse a conventional automotive valve spring which can be either constant or progressive rate, and either a single spring or a concentric pair.

The key dimensions of the spring can be edited either directly, or via text boxes, in a manner similar to the valve actuation mechanism. The task of spring design can be approached from a number of different directions, using controlling criteria which may be the wire diameter, the spring fitted length, the number of active coils, the spring rate, or the allowable stress range. Fig. 23 shows the main interface screen from the Spring Design Tool.

Whilst the design of the valve train mechanism is underway a continual check should be made on the valve / piston clearance. This can be done easily using Lotus Concept Valve Train which super-imposes the valve head motion on the locus of the piston top. The minimum clearance is calculated for the defined valve timing and at various timings up to 50 degrees either side of that defined.

By this stage the cam profile has been defined and the initial design of the mechanism has been performed using a simplified kinematic / static analysis using Lotus Concept Valve Train. A detailed dynamic analysis of the mechanism can now be performed in ADAMS/Engine. Lotus Concept Valve Train is supplied with modified ADAMS/Engine templates and can write the appropriate Profile Files and Sub-System Files to enable an ADAMS model to be constructed very quickly. Fig. 24 shows the interface panel from Lotus Concept Valve Train for exporting a solid tappet Sub-System File to ADAMS/Engine.

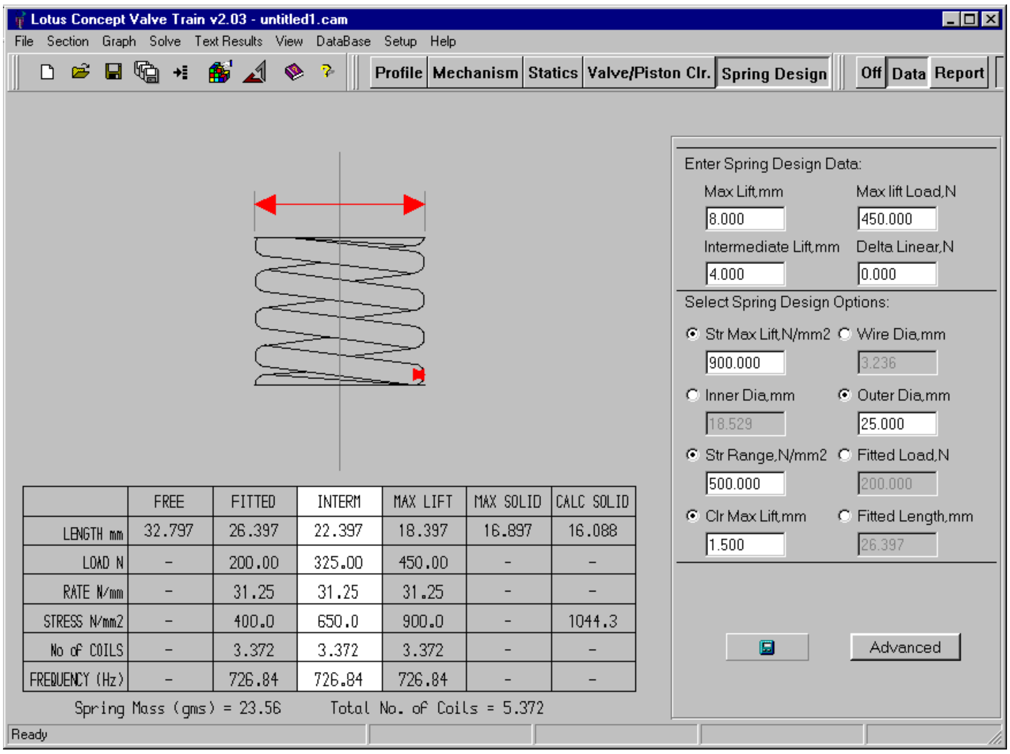

Fig. 23. Spring Design Tool in Lotus Concept Valve Train

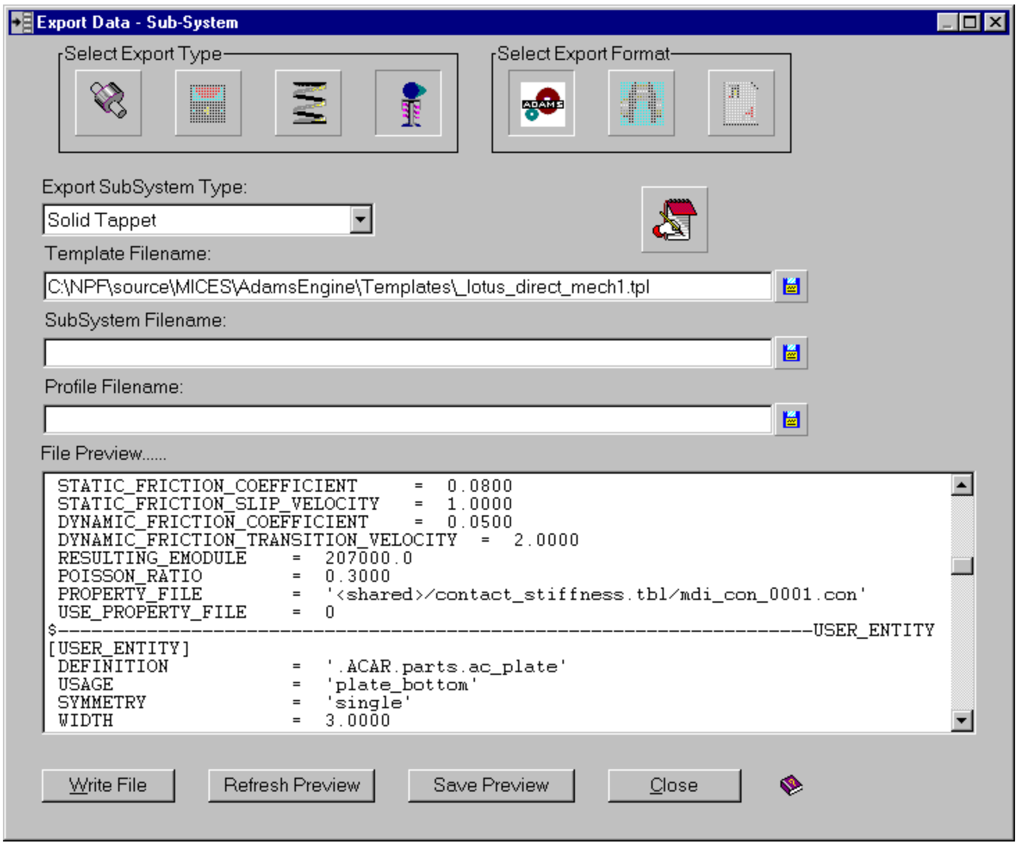

Fig. 24. Sub-System export to ADAMS/Engine from Lotus Concept Valve Train

Fig. 25 illustrates the creation of the full valve train mechanism, including the spring, in ADAMS/Engine directly from Lotus Concept Valve Train. Once the complete unit of the valve actuation mechanism has been built in ADAMS/Engine an entire valve train model can be constructed so that a full dynamic analysis of the system can be performed. This can ultimately be extended to a full engine mechanical model with gas pressure loading applied via output from Lotus Engine Simulation.

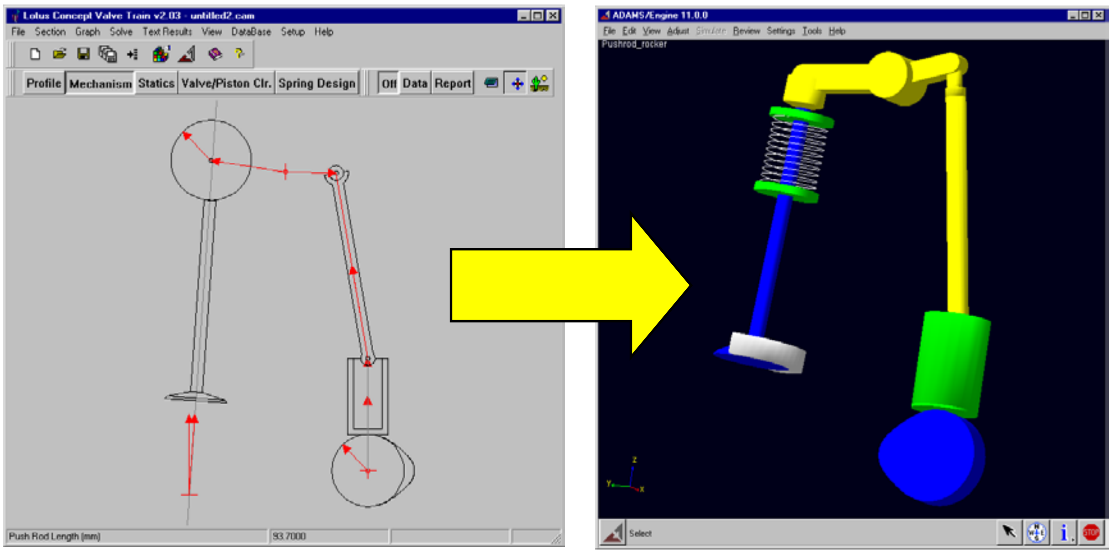

Fig. 25. Creation of ADAMS/Engine model from Lotus Concept Valve Train.

5. Conclusions

The paper has shown how Lotus Engineering Software can be used as an integral part of the powertrain design and development process. Lotus Vehicle Simulation can be used to set engine performance targets required to achieve a stipulated vehicle performance envelope. Lotus Engine Simulation can then be used in order to specify the key engine and manifold dimensions required to realise the desired performance level using parametric analysis techniques. Once the valve lift and opening duration have been defined Lotus Concept Valve Train can be used to design a cam profile in minimal time. This cam profile cam will be sufficiently viable to use in physical prototype engines whilst a detailed dynamic analysis of the full valve train is performed using an ADAMS/Engine model which has been constructed from Sub-System Files written from Lotus Concept Valve Train.

6. References

1. Winterbone, D.E., and Pearson, R.J., Design techniques for engine manifolds: Wave action methods for IC engines. Professional Engineering Publications Ltd, London, 1999. ISBN 1 86058 179 X.
2. Livengood, J.C., Rogowski, A.R., and Taylor, C.F., The volumetric efficiency of four-stroke engines. SAE Quat. Trans., 60, 617-632, 1952.
3. Winterbone, D.E., and Pearson, R.J., Theory of engine manifold design: Wave action methods for IC engines. Professional Engineering Publications Ltd, London, 2000. ISBN 1 86058 209 5.
4. Benson, R.S., The thermodynamics and gas dynamics of internal combustion engines (Volume 1), Clarendon Press, 1982. (ISBN 0-19-856210-1)
5. Benson, R.S., and Whitehouse, N.D., Internal combustion engines. Pergammon Press, Oxford, 1979. ISBN 0 08 022717 1.
6. Davis, S.F. TVD finite difference schemes and artificial viscosity. NASA CR 172373, 1984.
7. Davis, S.F. A simplified TVD finite difference scheme via artificial viscosity. SIAM J. Sci. Stat. Comput., 8, 1, 1-18, 1987.

7. Notation

Symbol	Meaning	Units
a	speed of sound	m/s
AFR	air-fuel ratio	-
BMEP	brake-mean effective pressure	bar
$\mathbf{C}$	vector of source terms	-
e_o	specific stagnation intrinsic internal energy	J/kg
f	pipe wall friction factor	-
F	pipe cross-sectional area	m^2
$\mathbf{F}$	flux vector	-
h_o	specific stagnation enthalpy	J/kg
N	engine speed	rev/min
$\overline{p}$	brake mean effective pressure	bar
q	heat transfer rate per unit mass	J/kg
Q_v	specific heating value	MJ/kg
t	time	s
TV	total variation	-
u	velocity	m/s
V_{swept}	swept volume	litres
$\dot{W}$	power	kW
$\mathbf{W}$	vector of conserved variables	-
x	distance	m
η_{comb}	combustion efficiency	-
η_{th}	thermal efficiency	-
η_v	volumetric efficiency	-
λ	Eigen value (‘wave’ speed) $= u \pm a, u$	m/s
ν	Courant number	-
ρ	density	kg/m^3
ϕ	flux limiter	-