The variable valve timing mechanism for the Rover K16 engine
Part 2: application to the engine and the performance obtained

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Abstract: Part 1 of this paper described the principles and basic design of the Rover variable valve timing system (VVC). This second part describes in detail the mechanical design which was developed to suit the particular requirements of the planned 1.8 litre version of the Rover K16 engine. The testing programme undertaken to establish durability and prove the product is covered and the performance obtained from the VVC engine is compared with the corresponding base engine which uses standard valve gear.

Keywords: variable valve timing, VVC

NOTATION

AFR	air–fuel ratio
ATDC	after top dead centre (deg)
BDC	bottom dead centre
b.m.e.p.	brake mean effective pressure (bar)
ECU	electronic control unit
IVC	inlet valve closing
IVO	inlet valve opening
WOT	wide open throttle

1 INTRODUCTION

Part 1 of this paper [1] described the principles of the operation and introduced the chosen arrangement of the mechanisms for the variable valve timing system used on the VVC version of the Rover 1.8 litre K16 engine.

The particular inlet valve timing required at a given engine condition is obtained by varying the angular velocity ratio of the cams relative to the engine crankshaft as they rotate. The amplitude and phase of this cyclic variation is adjusted to change the timing. Each cylinder requires an individual mechanism to synchronize with the engine firing order and as explained in reference [1] the compact design of the K series engine led to a decision to situate one double variable mechanism on each end and use the exhaust camshaft as a layshaft to take the drive to the rear mechanism. Even then the space available is restricted both radially and axially, which made the design and development of the system very challenging.

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2 MECHANICAL DESIGN

2.1 VVC components

Figure 1 is a plan view of the camshaft drive and the VVC arrangement on the engine, which shows how the mechanisms have been fitted into the space. Figure 2 is an exploded isometric view of the rear double eccentric unit and Fig. 3 is a photograph of the engine cylinder head sectioned for exhibition. These three figures can be studied in conjunction with the following notes, which list the main mechanical design features:

1. The double eccentric units have been made as compact as possible in the axial direction by arranging for components to overlap each other where possible. This can be seen in Fig. 2. The various cut-outs in the components, which allow for the relative motion, have been designed to clear within the chosen range of valve period adjustment.
2. The front and rear assemblies are similar but not identical. The fact that the two assemblies are opposite-hand coupled with the four-cylinder engine firing order, namely 1–3–4–2, results in differences

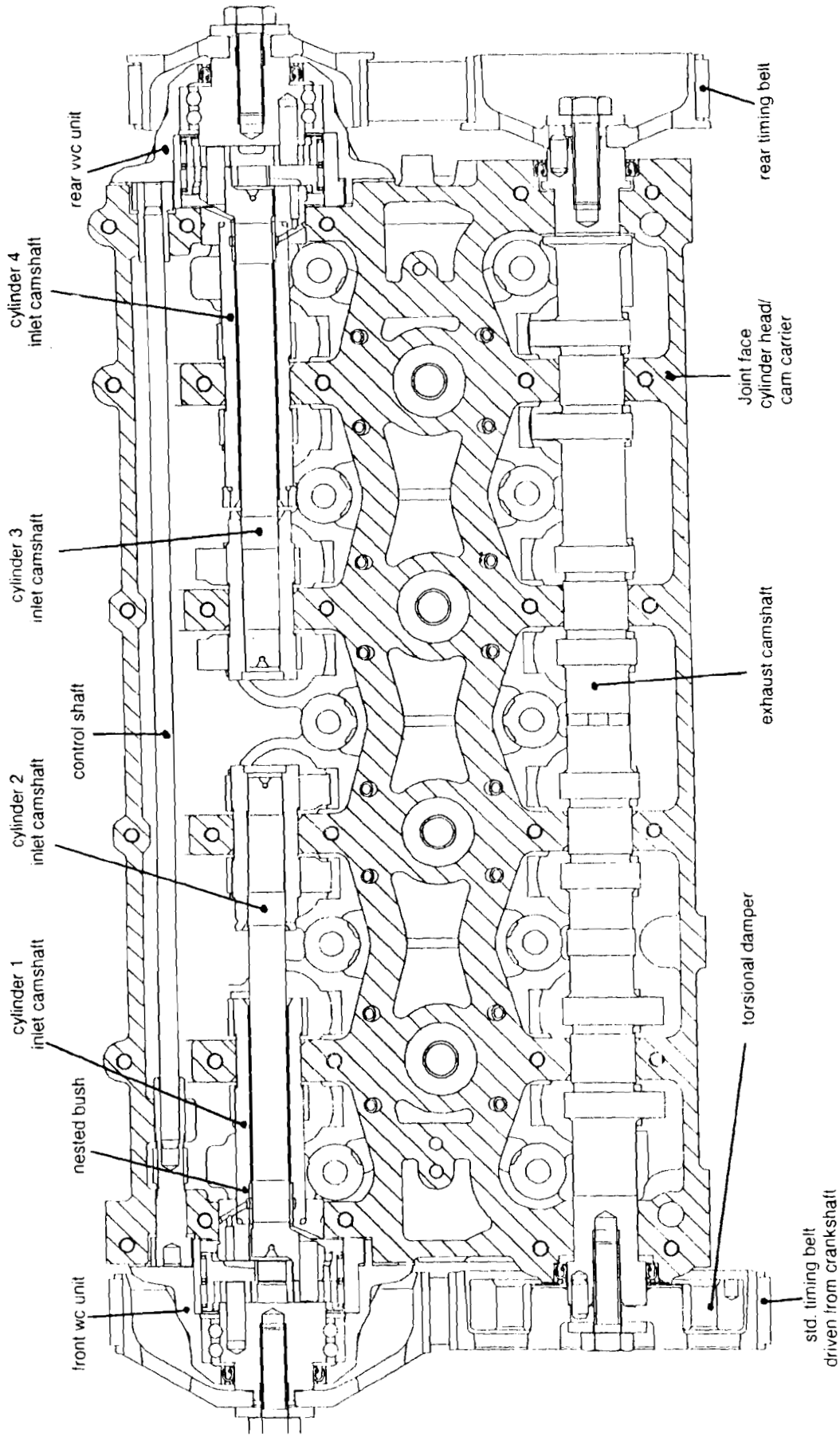

Fig. 1 Mechanical arrangement of the VVC cylinder head: plan view with the cam carrier removed

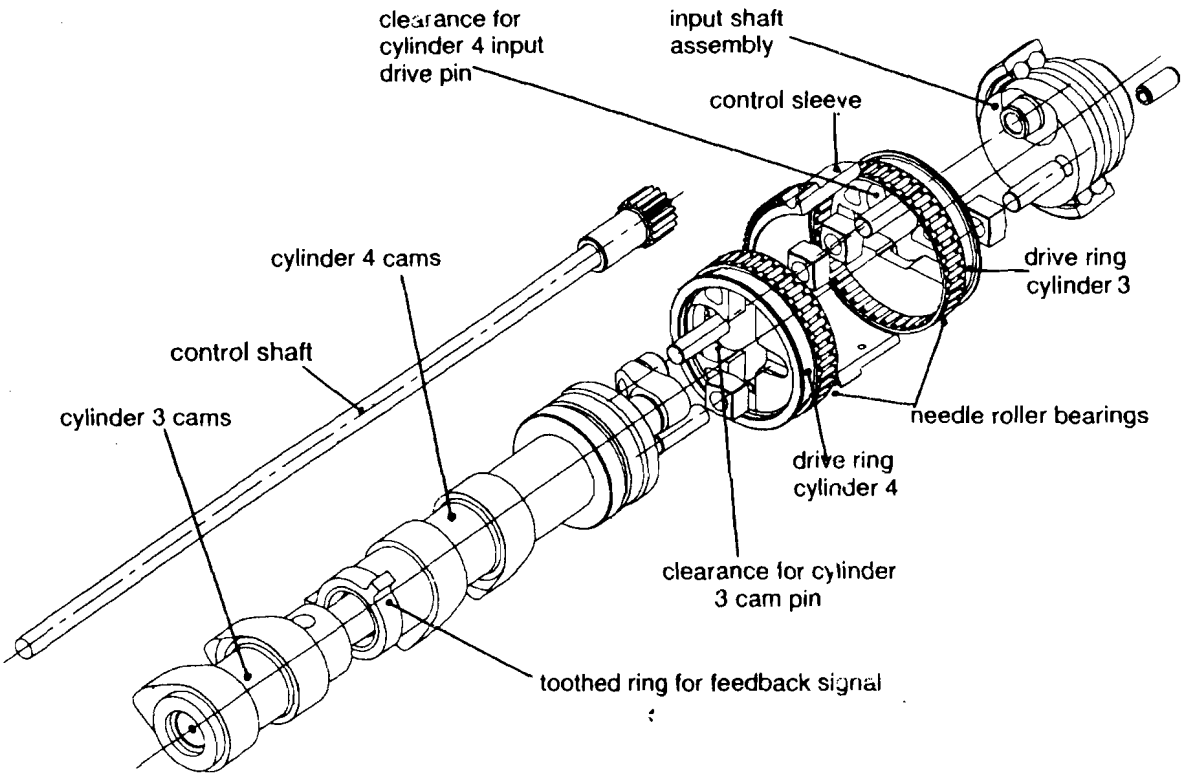

Fig. 2 VVC mechanism: rear double unit

between the front and rear housings, input drive shafts, control sleeves and camshafts. However, the drive rings, bearings and drive blocks are common.

3. The valve operating loads are taken directly on the standard bearings situated between the two cams associated with each cylinder. This is exactly the same as for the base engine. Because the inlet camshafts for each cylinder are separate components the axial thrust loads have to be taken individually on the upper bearing caps situated in the cam carrier, which is bolted on to the cylinder head. This differs from the base engine where the camshaft thrust is taken on dedicated faces situated on each side of the rear bearing. The hydraulic tappets from the base engine were specified in the initial design, but subsequently modified versions

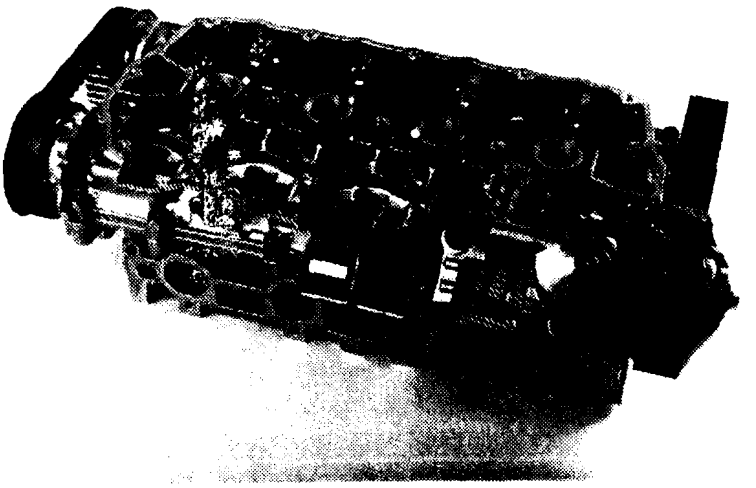

Fig. 3 Cylinder head (sectioned example used for exhibition)

from the same supplier, which were 17 per cent lighter, were substituted to reduce maximum cam drive torque.

4. The exhaust camshaft is made in chilled cast iron exactly as those in the standard K series engines. However, in order to permit individual material choices to suit the differing functions of each feature, the individual inlet camshafts use composite construction. The iron cam lobes are sintered on to tubular steel shafts. This ensures that relatively thin-walled shafts with adequate bore diameter for the nested shaft arrangement can be used, but having sufficient tensile strength and ductility for the interference fits. However, the vital cam–tappet interface is similar to that of the proven base engine, namely iron cams against hard steel tappets. The end flanges of the outer cylinder camshafts are made in a sintered copper iron alloy, furnace brazed to the tubular shafts. Unlike cast iron, this alloy has adequate tensile strength to properly support the interference needed to secure the drive pins.
5. An extra bearing, in the form of a thin-walled bronze bush to support the extended inner camshaft, is nested between the two camshafts at each end of the engine. These bushes are only subject to the relative velocity between the two camshafts. This is, of course, zero when the mechanisms are concentric and cyclically varying backwards or forwards at other settings.
6. The end housings carry the eccentric bores for the control sleeves, i.e. eccentricity OA in Fig. 3 of Part 1 of this paper [1]. All the bores in the cylinder head are concentric with the normal camshaft bearings.
7. In order to reduce friction, the eccentric drive rings run on needle roller bearings mounted in the control sleeves. The drive rings are investment castings with the accurate slot for the drive blocks finished by broaching and the outer diameter hardened and ground to form the inner track for the roller bearings.
8. The input drive shafts are mounted on two row angular contact ball bearings. These shafts are supplied as integral subassemblies from the bearing manufacturer with the inner tracks ground directly on to the shaft surfaces. These bearings have to withstand a combination of drive belt loads and the drive pin loads from the camshafts. These loads combine to produce offset alternating bending so the bearing specification calls for relatively high load capacity in the limited space and also close control of clearances.
9. The front timing belt could not be moved from its standard position. The diameter of the camshaft pulley and the number of teeth also had to remain the same as for the standard engine. The mechanism was therefore designed so that the housing would fit inside the deeply dished pulley shown in Fig. 1. The rear pulleys were made smaller with a less deep dish since the axial position of the rear belt could be chosen to suit.
10. The standard K16 camshaft oil seals with their associated design features, which have proved satisfactory in service, are used on the input drive shafts.
11. The small drive blocks are sintered iron with the holes for the drive pins accurately reamed and the side surface ground to match the drive ring slots. The blocks are shaped so that they cannot be fitted wrongly in the slots.
12. The base K16 engine has its ignition distributor mounted on the end of the inlet camshaft. While it would be possible to mount this outboard of the rear toothed belt, due to vehicle packaging constraints it was decided to introduce direct ignition on the VVC engine and eliminate the distributor drive. This uses double output coils, one for cylinders 1 and 4, the other for cylinders 2 and 3. The engine management system times the switching of the low tension current from the crankshaft encoder. The two spark plugs on each coil fire together, which implies a wasted spark during the exhaust stroke. The spark plugs specified have platinum tips on both electrodes to permit a life of 66 000 miles between changes.
13. The two control sleeves are synchronized by gears mounted on a shaft which runs the length of the cylinder head. The valve timing is adjusted by rotating this shaft. The associated control system is described in detail in the next section.

2.2 VVC control system

A new version of the Rover engine management system positions the control sleeves in the VVC units to set the required valve period at each engine speed and load. These settings are optimized during engine performance and emission development and are stored in the ECU as a map of values at chosen sites defined by speed and inlet manifold absolute pressure.

During steady state running the required accuracy of the valve period setting is within two degrees of target. It is not, however, necessary to remain exactly in step as the map sites are passed during fast transients, provided there is no significant loss of engine torque and that the fuel delivery properly matches the actual amount of air flowing at any instant. It was therefore decided that a very fast response servo to move the control sleeves would not be justified and that by careful development of a suitable algorithm for the ECU a relatively simple control system would suffice.

The valve period setting is measured in the ECU by continuously calculating the ratio of the time between the start of valve opening and the finish of valve closing.

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for one representative cylinder relative to the time for one camshaft revolution. This ratio is directly proportional to the valve period. The toothed ring mounted on the camshaft of cylinder 4, which generates suitable timing signals in a variable reluctance sensor connected to the ECU, is shown in Fig. 2. It is important to note that the measurement of period can only be updated every camshaft revolution and a continuous instantaneous reading is not available. The first prototypes used an electric stepping motor to rotate the control shaft via a worm gear, but the available speed of response was disappointing. A decision was therefore taken to change to an electrohydraulic actuator capable of applying more power. Figure 4 shows a diagram of this actuator. Engine lubricating oil at the prevailing oil gallery pressure is switched by a four-way twin solenoid spool valve to move a hydraulic piston, which in turn rotates the control shaft via a rack and pinion. When both solenoids are switched off the piston is hydraulically locked at a particular setting. The control valve solenoids are simple on/off devices. The basic principle of the algorithm is that while the setting remains beyond a predetermined outer band the appropriate solenoid can remain switched on to move as quickly as possible towards the target. Once the setting is between the outer band and the inner dead band, adjustment is only made using minimum pulse times on the appropriate solenoid and then waiting for the effect to be fully recorded in the updated feedback readings. The values used for the minimum pulse times are effected mainly by oil temperature but also by whether the period is increasing or decreasing. The other main feature of the algorithm is the compensation applied to the fuel delivery to allow for airflow differences caused by differences between the actual valve period prevailing at any instant and that implicit in the ECU fuelling maps. These differences can be due to the relatively slow adjustment during transients as discussed above or due to mechanism malfunction. The compensation uses a table of calibrated coefficients, which relate the airflow at any period setting to that at a nominal median period for the range of engine speeds.

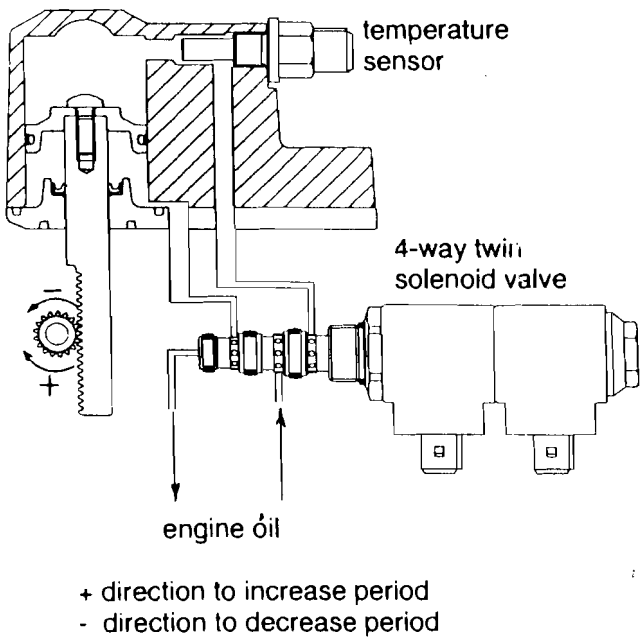

Fig. 4 VVC control actuator

3 PRODUCT DEVELOPMENT

The overall test programme for the 1.8 litre K16 VVC engine followed standard Rover practice [2, 3]. However, in this case there was special emphasis on early rig test and fatigue testing to ensure a high confidence in overall product integrity. The testing process is discussed in the next four sections.

3.1 Fatigue testing

Individual components, identified from failure mode and effect analyses as being fatigue critical, were developed separately using a conventional electrohydraulic fatigue testing machine. These critical components were the input drive shaft assembly, the outer cylinder camshafts and the inner cylinder camshafts. An alternating load was applied at a frequency in the 10–15 Hz range. This meant that cycles could not be accumulated as quickly as on an engine running at high speed, but these tests have the advantage of allowing an overload to be applied to establish a definite endurance limit. Correlation with the results from the cylinder head motoring rig, described in the next section, established an appropriate load that the components had to withstand, for ten million (10^7) cycles, in order to meet the durability standard.

The early concept designs of the mechanism had inadequate torque capacity and failed either by bending fatigue of the cantilevered drive pins or fatigue of the structures supporting those pins in the input drive shafts and the camshafts. The fatigue test results along with finite element stressing were used to develop the designs so that the relative proportions of the drive pins and their surrounding structures were carefully optimized within the available space. This, together with proper material selection, appropriate specification of heat treatment and careful attention to manufacturing processes, resulted in the very compact mechanism necessary to fit into the available space on the engine, but having sufficient torque capacity to withstand all engine durability cycles.

3.2 Rig testing

Having established confidence through careful analysis and fatigue testing of individual components, durability could be proven by targeted testing on a cylinder head motoring rig. This consisted of an isolated cylinder head assembly mounted on an aluminium base plate through which it was serviced with oil and water heated to engine operating temperature. The camshafts were driven using the standard timing belt by an electric motor at speeds up to the maximum engine speed of approximately 7500 r/min. An adapted engine management system was utilized to adjust the cam period specified at each speed during the running of a range of duty cycles. The dynamic loads generated on this rig were very similar to those experienced on a firing engine, but component failure would not lead to engine termination and so replacements could be fitted quickly in order to continue testing. Specific test regimes were developed to assess a range of engine speeds and cam period settings beyond the harshest customer usage in order to guarantee component life.

In order to assess the long-term durability of the control system a separate rig was used that allowed cycling tests on the hydraulic control unit. This rig was serviced with heated engine oil and appropriate signals to the spool valve solenoids. The unit was subjected to repeated actuations from one extreme to the other, i.e. from minimum period to maximum period settings. The electrohydraulic spool valve, the piston rack pinion mechanism and the oil seals were shown to have adequate durability using this rig.

3.3 Test bed running

In parallel with the test programme, a test bed durability programme was undertaken on the engine. In the early stages the objective was to establish ultimate durability of the VVC mechanism, but later the objective became reliability confirmation through the use of more but shorter tests. In this way, accelerated testing separate from ongoing vehicle work could be conducted. This technique is discussed in greater depth in reference [2].

3.4 Vehicle testing

Early product development engines were installed in a range of simulator and engineering vehicles. The first objective was to subject the VVC engines to real user conditions, including hot and cold climates, high-speed autobahn driving, as well as extended mileage of normal road driving. The second objective was to allow development of whole vehicle packages. Applications of the new power unit included both the mid-engined sports car and front wheel drive vehicles. Therefore the various engine and vehicle component teams needed to optimize installation and equipment to take into account the associated differences in intake and exhaust systems, underbonnet heat management, etc. Later specification engines were then fitted to the intended vehicles and subjected to the appropriate development and sign-off tests.

3.5 Engine refinement

The VVC engine is part of the Rover K series family and thus shares the refinement benefits and characteristics derived from the structural features, mechanical details and combustion system developed for earlier versions. The key area for refinement work on the VVC project was therefore the novel valve train. The important items were cam profile design, material selection, limits, fits and clearances. These are discussed in the rest of this section.

Cam profile design is the usual compromise between a desire to increase valve lift for maximum performance and restricting valve acceleration to limit drive belt loads, cam contact stress and noise. The addition of VVC means that the full range of valve period adjustment has to be considered in the compromise. The peak loads in the front drive belt are a problem at short period settings. This is due to three additive effects. Near minimum period setting, which is about 220° crank angle, the interval between IVO on any particular cylinder and IVC for the previous cylinder in the firing order, is roughly 20° cam angle compared with about 32° for the base engine with a fixed 244° cam period. This phase difference prevents the normal cancellation of the positive torque from the opening valve by the negative torque from the closing valve, which increases the load on the front drive belt. The production belt was of course rated for the standard engine. This effect occurs in any variable system which permits valve periods below about 230°, which is applied to a four-cylinder engine. There are, however, two more effects with VVC. For low period settings the drive torque is higher at any given engine speed due to the relative speeding of the cam as it passes the tappet. Also, the use of exhaust camshaft as a layshaft to drive the rear VVC unit via another belt provides a further degree of freedom and produces an unwelcome extra torsional natural frequency, which occurs in the particular speed range where the shorter period settings are required to maximize engine output torque. Detailed analysis of design iterations produced a special asymmetric cam profile which carefully limited valve accelerations. With the help of the reduced weight tappets, minimum inertia pulleys and addition of a torsional damper to the front exhaust cam pulley, shown in Fig. 1, an inlet valve lift of 9.5 mm was achieved compared with 8.8 mm for the standard engine.

It is absolutely essential to closely control the clearance in the two-row angular contact ball bearing supporting the input drive shaft. If the clearance increases,

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the torque reversals on the drive pins resulting from the cam loads generate unacceptable noise. The housings for the double VVC units are therefore made in cast iron to closely maintain the desired clamping load on the bearing outer races over a temperature range from below −20 to over 130 °C. The cast iron also damps some of the noise transmitted along the camshafts.

Stringent manufacturing tolerances have been applied to all rotating bearing and sliding components, since any extra clearance over that required for smooth operation leads to increased flexibility which would change the valve motion compared with that intended and also generate noise. Detailed analysis showed that the clearance between the drive blocks and their parent slots in the drive rings would need to be controlled through a graded assembly process. This process, which is similar to that normally used for the pistons in their bores, allowed achievable manufacturing tolerances, but at the same time enabled stringent in-house refinement targets to be met. The necessity for close tolerances and surface finishes also applies to the meshing gears on the control sleeves, control shaft pinions and the hydraulic control unit rack. These teeth are not in the direct valve actuation path, but the load transmitted is not inconsiderable and correct mesh is critical for satisfactory and refined operation.

4 ENGINE PERFORMANCE

In this section the performance of the K 1.8 litre VVC engine is compared with that of the base K 1.8 litre and reasons for the differences analysed. Both engines are as specified in the MGF sports car, including the full intake and exhaust systems.

As far as possible the two engines are identical. However, in order to make best use of the variable valve timing characteristics three other changes directly affecting performance behaviour were made to the VVC engines in addition to fitting the mechanisms described in this paper:

1. In order to improve breathing capability, the diameter of the inlet valves was increased from 27.3 to 31.3 mm and the diameter of the exhaust valves was increased from 24.0 to 27.3 mm.
2. In order to increase flow capability the cross-sectional area of the inlet manifold tracts was increased by 12 per cent.
3. The exhaust cam period was increased from 240 to 252° and the exhaust valve lift from 8.8 to 9.2 mm. The exhaust cam is set to close at 20° ATDC compared with 12° for the base engine. See Fig. 6 of reference [1] for details of the valve timings used.

4.1 Behaviour at low-speed/low-load conditions

The two test conditions used to represent this area of engine operation are 2000 r min 2 bar b.m.e.p. and 2400 r min 3.5 bar b.m.e.p. At these conditions the VVC is set to inlet valve timings quite close to those of the fixed camshaft, which is used in the base engine. The later exhaust valve closing referred to in the last para-

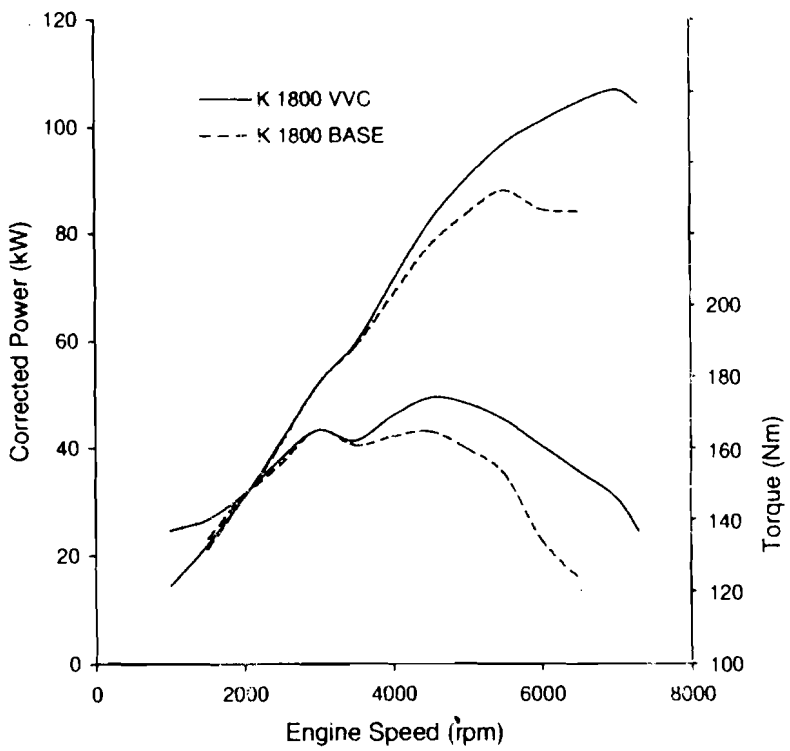

Fig. 5 Power and torque – comparison of VVC and base engines

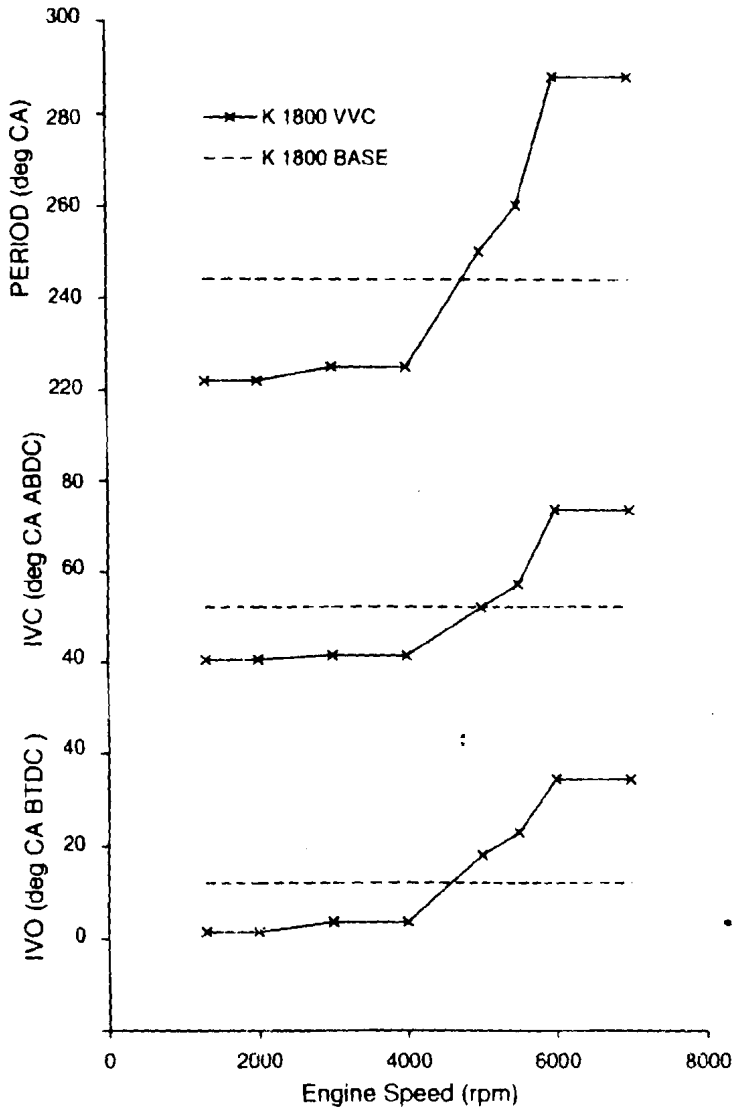

Fig. 6 VVC settings at WOT

graph means that the VVC engine has increased overlap at these settings.

For both conditions the specific fuel consumption is closely similar for both engines. This result is expected since both operate at stoichiometric AFR to produce the required exhaust gas composition for the three-way catalysts fitted to the vehicles, and combustion chamber heat losses and engine mechanical friction losses should be similar. The out-of-engine NO_x emissions of the VVC engine are about 30 per cent less than the base engine due to the higher in-cylinder exhaust residuals resulting from the increased overlap. The hydrocarbon emissions at each condition for the two engines are similar.

At idle the minimum period setting of the VVC is used, which brings the overlap back to that of the standard engine despite the later exhaust valve closing. As a result idle stability is similar for both engines.

4.2 Wide open throttle (WOT) performance

Figure 5 shows the power and torque plotted against speed for the two engines. It is apparent that a substantial increase in power has been obtained from the VVC engine without loss of torque at lower speeds. The VVC settings used to achieve these benefits are shown in Fig. 6. Figure 7 shows the results of analysing the two torque curves for brake specific air consumption (b.s.a.c.) and indicated specific air consumption (i.s.a.c.) using the method described in detail in reference [4]. Salient features of the volumetric efficiency and i.s.a.c. curves are discussed in the next two paragraphs.

The VVC engine shows superior volumetric efficiency at high speed, e.g. 7 per cent at 6000 r/min. This would be expected since the VVC system permits valve timings at high speed normally associated with competition engines. On the one hand, the large overlap due to the later closing of the exhaust valve and the early opening of the

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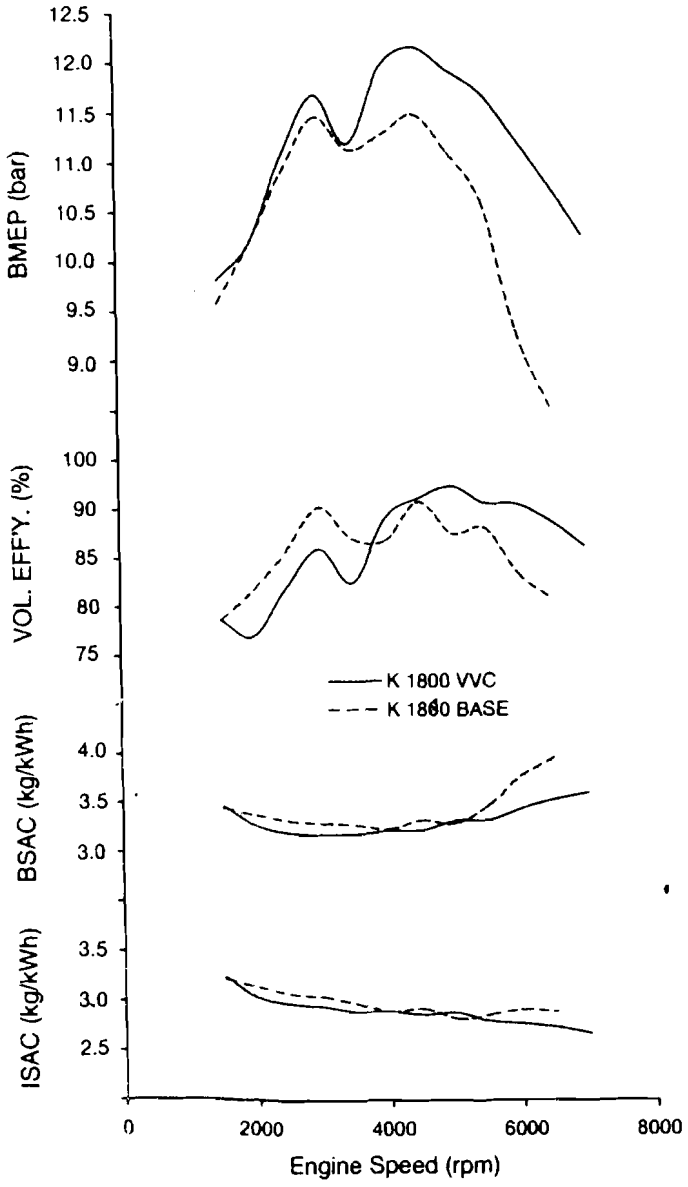

Fig. 7 Volumetric efficiency and air consumption analysis

inlet expedites cylinder scavenging and, on the other, the late closing of the inlet valve makes full use of the manifold ramming capability. However, at lower speeds, e.g. 2500 r/min, the volumetric of the VVC engine is 4 per cent less than that of the base engine. Here the short period setting of the VVC is used to reduce backflow from the cylinder after BDC, but this does not fully compensate for the reduced ramming velocity in the increased area inlet tracts.

The i.s.a.c. curves in Fig. 7 show that the VVC engine makes better use of the air induced at both high and low speeds. At high speeds this supplements the increased volumetric efficiency which increases b.m.e.p. further and at low speeds compensates for the reduced volumetric efficiency so that the b.m.e.p. of both engines is almost identical. The reduced i.s.a.c. of the VVC engine is the result of better combustion. This in turn allows the VVC engine to run at an AFR between 0.5 and 1.0 leaner at WOT than the base engine without exceeding the exhaust temperature limit for the catalyst at high speed or premature knock at low speed. Figure 8 shows that the WOT fuel consumption of the VVC engine benefits from both the improved i.s.a.c. and the leaner AFR and is lower at all speeds compared with the base engine.

CONCLUSIONS

This paper has firstly presented the practical mechanical design that was evolved to adapt the chosen system to

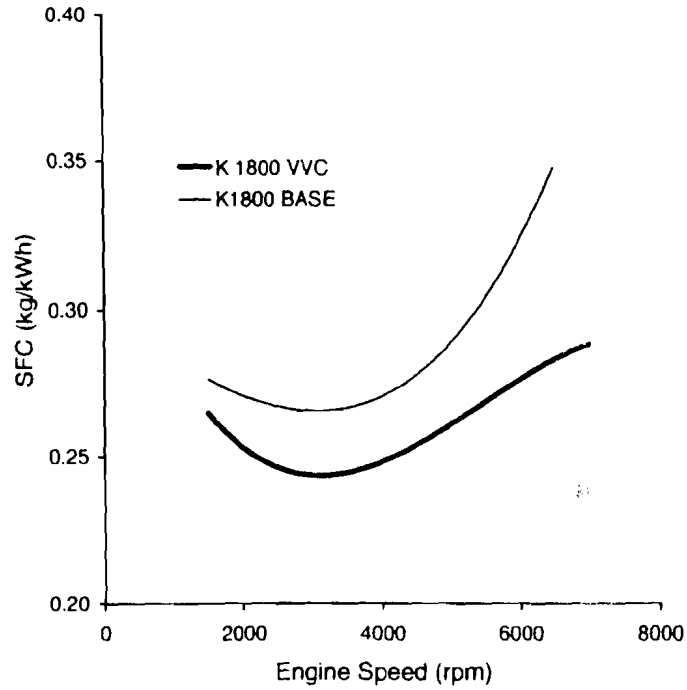

Fig. 8 WOT specific fuel consumption comparison of VVC and base engines

the established Rover K16 engine. This was followed by an outline of the mechanical test work which was undertaken to prove the integrity of the design. Emphasis was placed on the particular areas that differed from the standard engine, which was already proven. The last section reported the performance obtained from the VVC engine and compared it with that of the base engine which has standard valve gear.

The paper, together with the associated Part 1, has shown the wide scope of work that had to be undertaken to develop a novel variable valve timing concept into a completed vehicle application ready to be offered for sale to customers.

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