THE DESIGN AND OPERATION OF A VARIABLE VALVE TIMING SYSTEM ON A HIGHLY RATED MARINE DIESEL ENGINE

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ABSTRACT

The article describes the design process for a variable valve timing system on an in-line 6 cylinder diesel engine at Pyestock. The mechanism selection, design consideration, rig trials and engine experience are discussed.

Introduction

History

In 1991 work on simulating a Variable Valve Timing (VVT) strategy, together with some basic validation work on a 6 cylinder in-line diesel engine, was undertaken[1,2]. This demonstrated the benefits of reducing the inlet manifold reverse flow from the combustion space, when running at low load. The strategy employed variable valve phasing i.e. the point of Inlet Valve Opening (IVO) was changed whilst maintaining a fixed inlet valve period. No changes were made to the exhaust valve events.

VVT can be used to widen the operating range of a diesel engine, particularly those which are highly rated and often run at low load. Highly rated engines need a wide valve overlap of exhaust and inlet valve opening, to allow sufficient cylinder cooling at high loads. At low loads, this can lead to an adverse pressure gradient between the exhaust and inlet manifolds. This results in reverse gas flow, leading to inlet port fouling and increased maintenance. VVT has the potential to allow the valve overlap to be matched to the applied load.

Research objectives

The objective of the work was to design a VVT system based on the earlier simulation work and to demonstrate its operation on a working engine.

Project funding and progression

The project was undertaken through funding from the Directorate of Operational Requirments (Sea) by means of Applied Research Package 15e, Surface Platform Characteristics. The work was let as an university agreement to Brunel University, where the design and rig work was undertaken. Engine trials were undertaken in the Diesel Test Facility at the Test &Evaluation Establishment Pyestock.

Mechanism selection

Mechanism strategy

The initial work undertaken, indicated that only inlet valve events needed to be controlled to obtain the required performance improvements. A literature survey indicated that both mechanical and hydraulic systems had the potential to offer the required valve event control. The review of mechanical VVT systems indicated that valve phasing control would be easier to implement than a system that provided lift control.

Hydraulic options

Considerable effort was put into devising and modelling hydraulic VVT systems, since it was thought that they would have advantages of compactness and ease of control. However designs using engine lubrication oil were ultimately rejected because of:

- A lack of system stiffness due to oil compressibility.
- The difficulty of ensuring the absence of air from the hydraulic system.
- The need to ensure that the valve closing velocity would not lead to impact damage.

However, it was felt the advantages of a hydraulic system could be realized if they were considered at the design stage of a new engine.

Mechanical options

Two mechanical variable phasing systems were devised, both adopting the same principal of rotating the axis of the roller follower about the camshaft axis, so making it possible to advance/retard the IVO. Since the valve train motion was changed, the mechanism kinematics had to be analysed in order to verify that the valve motion was acceptable, and that the accelerations were within acceptable limits.

In order to keep costs and complications within manageable limits, it was considered important to minimize the modifications to the engine, and in particular to avoid any machining on components that could not be readily removed from the engine.

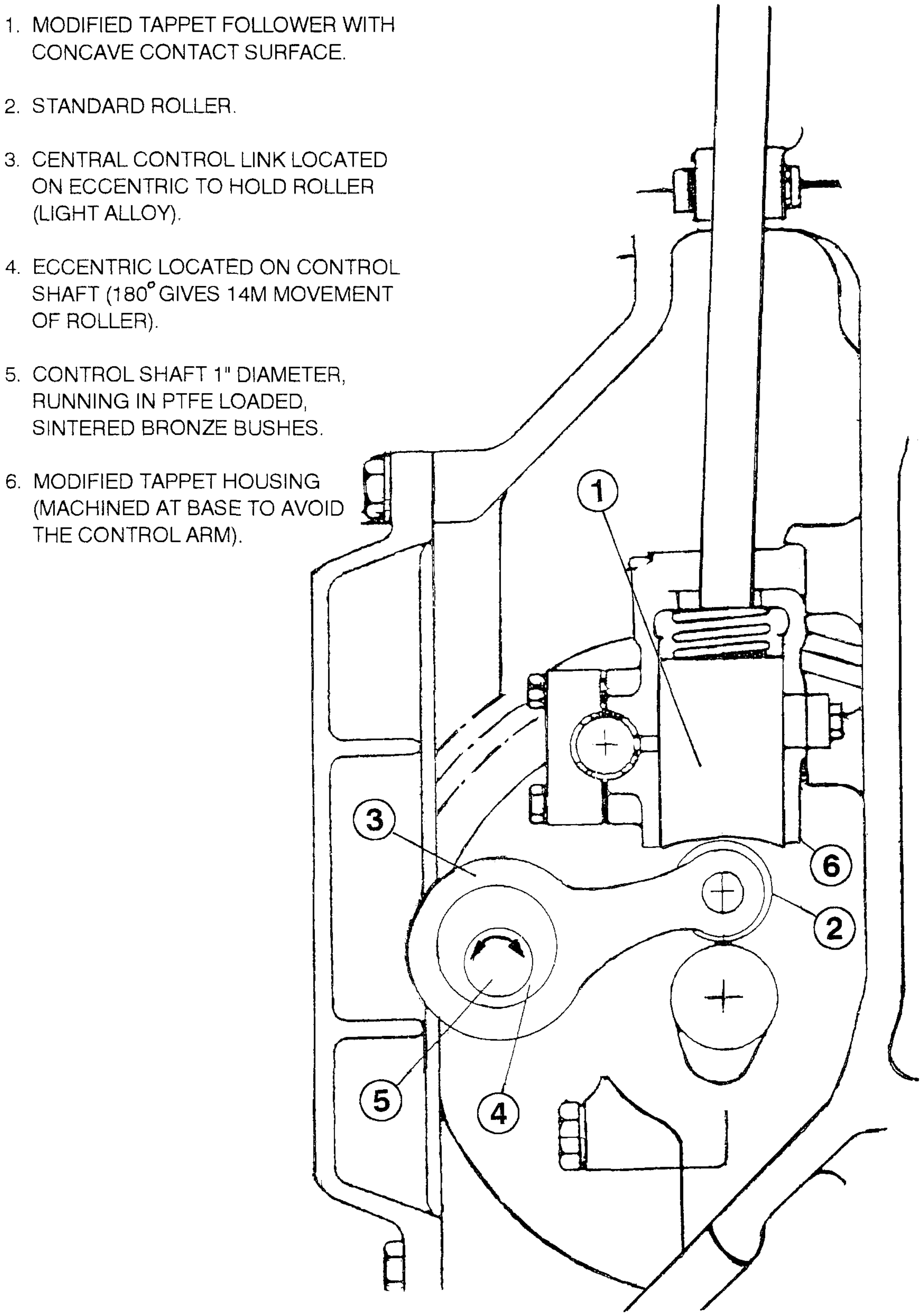

FIG. 1—SECTIONAL DRAWING OF THE VVT MECHANISM USED

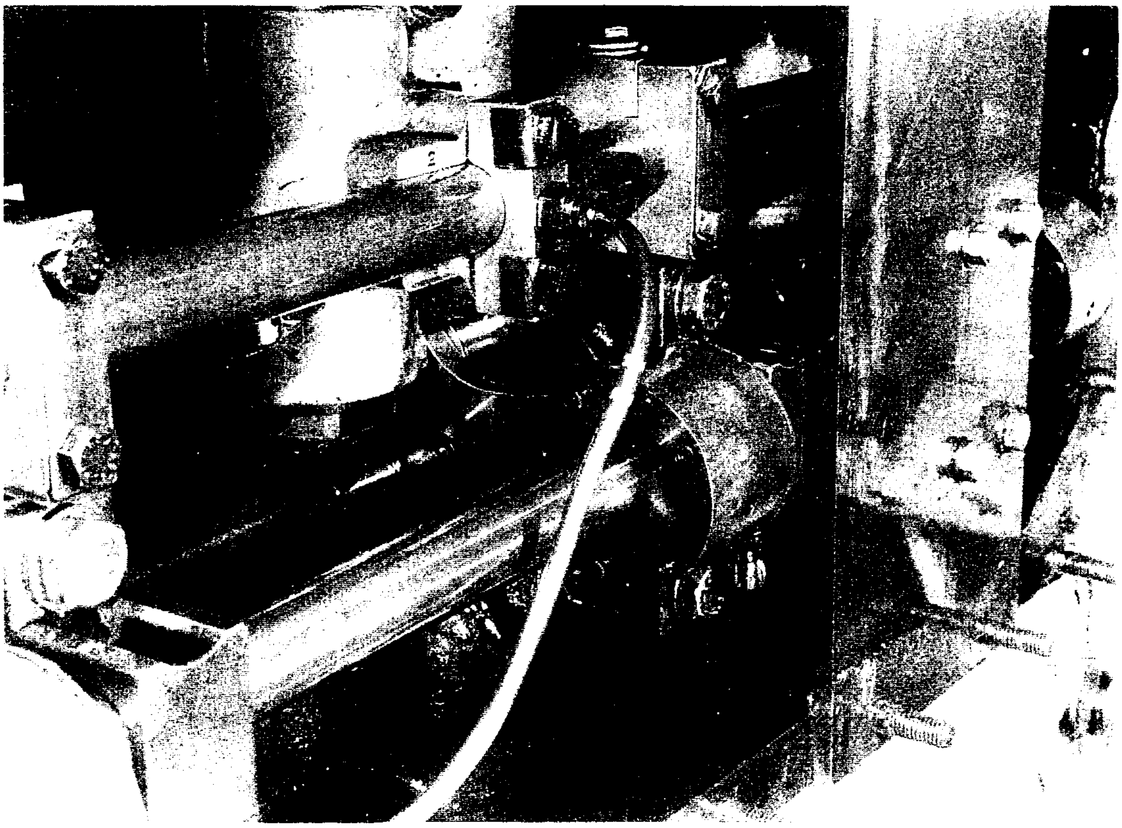

FIG. 2—THE STANDARD TAPPET ARRANGEMENT (EXHAUST) AND MACHINED TAPPET HOUSING FOR THE VVT MECHANISM

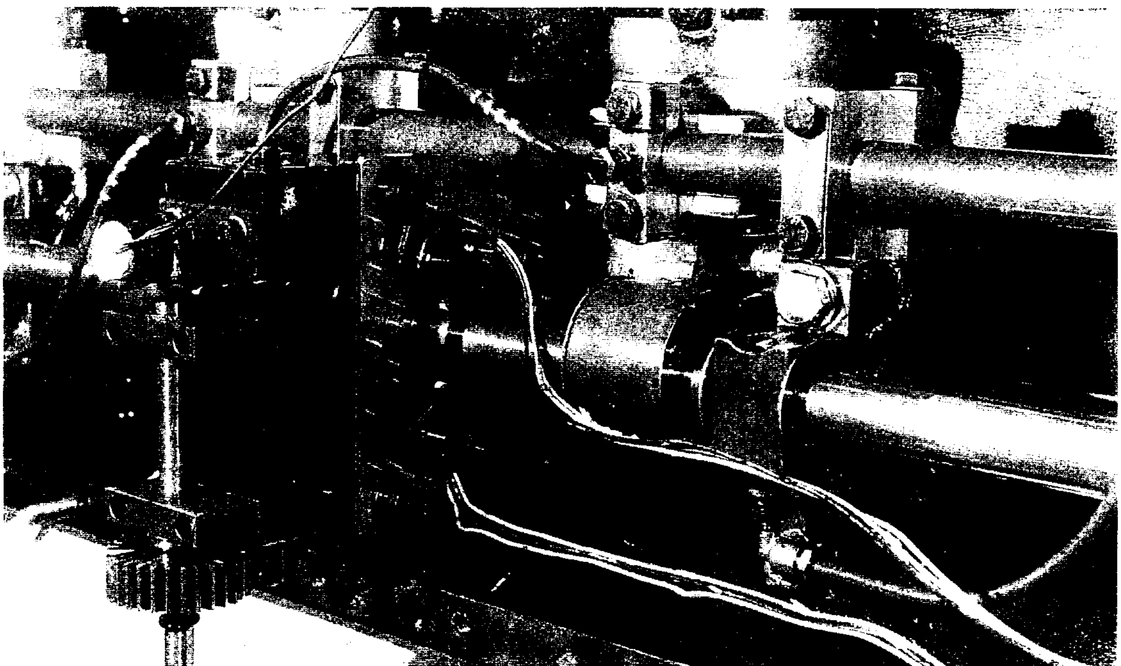

FIG. 3—THE WORM REDUCTION GEARBOX

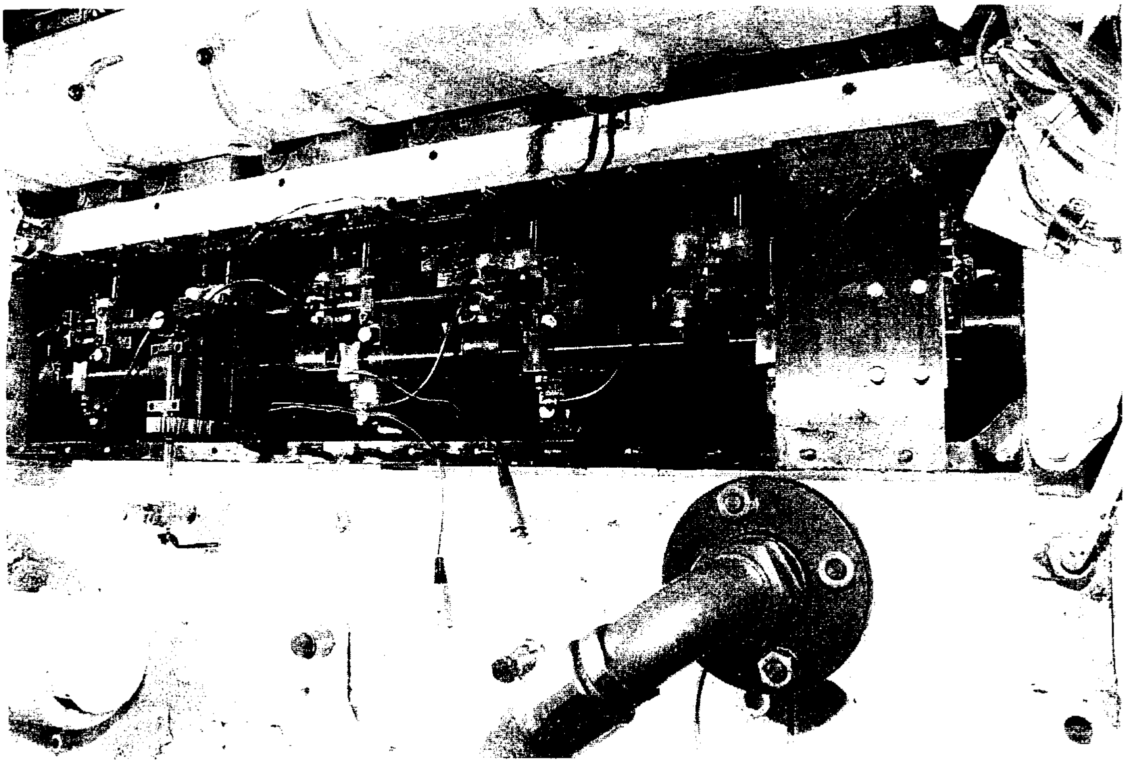

FIG. 4—GENERAL VIEW OF THE INSTALLED VVT MECHANISM

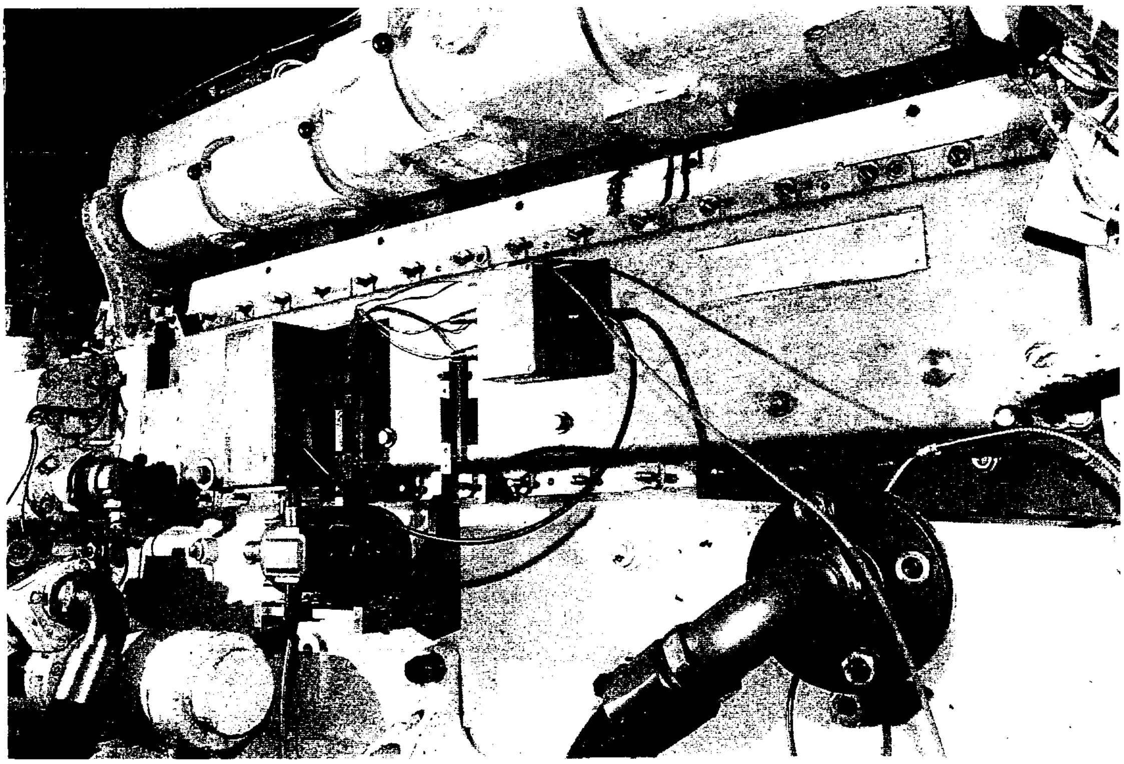

FIG. 5—THE SIDE OF THE ENGINE WITH THE MODIFIED COVER IN PLACE TOGETHER WITH THE REDUCTION GEARBOX AND ELECTRIC MOTOR

Mechanism design

Mechanism philosophy

The mechanism selected[3], (FIGS 1-5), was particularly compact. By allowing the roller support arm to rotate about the eccentric, the number of elements in the mechanism were minimized. It was to fit within the existing valve gear cover. A few modifications were required to the camshaft cover, to accommodate the greater depth required for the eccentric shaft and to support the actuating motor.

The throw of the eccentric was chosen so that any IVO timing between 60° and 20° Before Top Dead Centre (BTDC), could be selected whilst the engine was running. The obvious disadvantage of the mechanism was that there was no longer pure rolling between the cam and the roller follower. This was because at the tappet/roller and roller/cam interfaces, where there were combinations of rolling and sliding. An issue that required some investigation.

Tappet and roller selection

The tappets to be used in the engine were rough machined from EN 43, stress relieved, finish machined and then Sulfinuz treated. This comprised of immersion of the components for one and a half hours in a bath of cyanide and sulphide salts at 570 °C. The Sulfinuz treatment resulted in deep surface layers of iron carbide and iron nitride, which increased the hardness and wear resistance. There was also a surface layer of iron sulphide, which provided a low friction surface to assist running-in. After the Sulfinuz treatment, the tappets were induction hardened and oil quenched.

The Sulfinuz treatment resulted in a deterioration of the surface finish from that obtained through standard means. This required the tappet to be polished prior to operation, since otherwise local scuffing was found to occur. The roller was made from a softer material with the intention that it, rather than the tappet, should wear. This being uniform due to its rotating motion. It was also found to be necessary to barrel the rollers by 25 μm, since there was no positive location for the tappet block on the engine housing.

The final tests with the barrelled roller (EN 8 with 50 HRC) and the Sulfinuzed and induction hardened tappet (EN 43 with 58 HRC) indicated that: a highly polished finish (0.14 μm) would deteriorate to a finish of 0.23-0.33 μm, whilst a surface that was initially rougher than this would improve during the running-in period.

Tribological implications

The oil film thickness was computed for both the standard valve gear and the VVT system. In the case of the standard valve gear, with the engine running at 1200 rev/min, the roller was free to rotate at a speed of 840 rev/min on the base circle, rising to 1366 rev/min on the nose circle. For the proposed mechanism where the roller was in contact with the concave tappet, the maximum velocity was reduced only slightly from 1366 rev/min, but when the roller was on the base circle the angular velocity fell to 229 rev/min. Fortunately the contact forces on the base circle were very low and so this was not a problem. For the roller, there was a negligible difference in the minimum oil film thickness in comparison with the standard roller arrangement.

Mechanism actuation

The control shaft was rotated by a DC electrically driven worm reduction gearbox, mounted on the control shaft. A counter-shaft arrangement was used so that the drive shaft axis was clear of the engine side.

Mechanism control

Ultimately it was envisaged that the mechanism would be controlled by an on engine system. For the duration of this project, the mechanism was manually positioned from the control room, using electrical means. This allowed any valve timing to be selected with the engine stopped or running. The armature current to the DC motor, driving the worm reduction gearbox, was monitored and from calibration tests this could be equated to the torque demand.

Experimental rig results

Test rig description

The test rig was designed to simulate the engine block and to accommodate the inlet valve train of a single cylinder of an in-line six cylinder engine consisting of:

- The tappet.
- Pushrod.
- Rocker arm.
- Bridge piece.
- Two valves and springs.

In an attempt to compensate for the lower running temperature of the rig, compared to the engine, a lubricant of lower viscosity (10W/30) was used. However the use of a lower viscosity lubrication oil does not necessarily ensure that the viscosity is correctly matched or modelled. Since the viscosity measurements are made under low shear, the dependence of viscosity on pressure will vary, and the relevant temperatures would be those of the oil film (which could be subject to transient local heating by viscous dissipation).

The test rig was comprehensively instrumented for:

- Valve lift.
- Valve acceleration.
- Camshaft position and speed.
- Torque reaction on the eccentric control shaft.
- Roller speed measurement.

VVT mechanism rig results

Baseline measurements of the standard valve-train were taken, in order to verify the instrumentation and data analysis. These tests showed the instrumentation to be working well, but with significant differences between the measured and predicted valve motion at high speeds.

The VVT mechanism was tested at camshaft speeds of 150 rev/min and 750 rev/min. The results at 150 rev/min showed good agreement with earlier simulation predictions,[1] while at 750 rev/min there was some evidence of loss of contact between the valve stems and the cam. There was good evidence to indicate that the mechanism was giving appropriate phase changes to the valve events. Valuable tribological experience as discussed above was also gained from the rig work.

Engine changes

Engine modifications

For the engine trials some engine modifications were necessary, the most notable of which was the fitting of a new camshaft, with the inlet cam lobe timing adjusted to allow full movement for the VVT mechanism. The camshaft was manufactured to a new specification, that included phosphating after the heat treatment and final machining. The inlet valve push rods required shortening to allow the finger follower to be inserted, Figure 2. Minor modifications had to be made to the oil lubrication system. The turbocharger and fuel injection equipments were left as standard.

The oil supply to the VVT mechanism was via copper pipes, fed from the saddles on the main oil gallery tube in the camshaft chest.

The oil supply to the roller/tappet interface came via drilled holes in the underside of the tappet. The oil supply to the tappet was in the usual way from the tappet block. The timing slots and internal passages were modified to increase the oil flow to the tappet, so as to ensure an adequate oil supply to both the valve train and the roller/tappet interface.

Engine test results

Performance work

The engine tests were performed at a speed of 1200 rev/min, and power was varied from a minimum of 30kW up to a maximum of 500kW with the full range of inlet valve event timing. A maximum power of 650kW was also noted for the mechanism at the fully advanced position (corresponding to the baseline IVO of 60°BTDC). As expected the maximum power of 650 kW could not be achieved with the retarded inlet valve events, since the reduced airflow would lead to compressor surge in the turbocharger and/or overheating of the exhaust system components.

As predicted, most of the key engine performance parameters were not much affected by the phasing of the inlet valve events. For example, the fall in effective compression ratio was compensated for by an increase in the compressor boost pressure. Thus the maximum cylinder pressure and its location, and the maximum rate of pressure rise and its location, were not affected by the phasing of the inlet valve events. It might be thought that the lower effective compression ratio, when the valve events are retarded, might lead to a longer ignition delay but this was not found to be the case.

The tests were performed at constant Brake Mean Effective Pressure (BMEP) with measurements of the Indicated Mean Effective Pressure (IMEP), and Pumping Mean Effective Pressure (PMEP) made. Retarding the IVO showed a slightly adverse effect on the PMEP at the lowest load (30 kW). However, for the higher loads (200, 400 and 500 kW) there was evidence that retarding the IVO has a beneficial effect.

There were negligible changes in the brake specific fuel consumption values.

The averaged exhaust temperatures were seen to increase as IVO was delayed, especially at the higher power values. This being due to the reduced breathing ability of the cylinder.

(FIG.6) shows the effect on manifold pressures of delaying the inlet valve events. The pressure difference between the exhaust and the inlet manifolds have been plotted (relative to IVO), for the two extremes of inlet valve phasing. Also shown is the relative value, from which it can be deduced that the delayed IVO reduces the opportunity for reverse flow into the inlet manifold. Similar results are plotted in (FIG. 7), except that an absolute basis has been used for the manifold pressures and the crankangle. Figure 7 shows that delaying IVO leads to a:

- Slight reduction in the exhaust back pressure.
- Slight increase in the compressor boost pressure.
- Shorter time after IVO before the adverse pressure gradient between the exhaust and inlet manifolds is eliminated.

All three of these effects will contribute to a reduction of the reverse flow into the inlet manifold, that can lead to inlet port fouling. The differences in the exhaust pressure records of the two manifolds are due to cylinders 1, 2 and 3 exhausting into manifold A, which is of smaller volume and closer to the turbocharger than

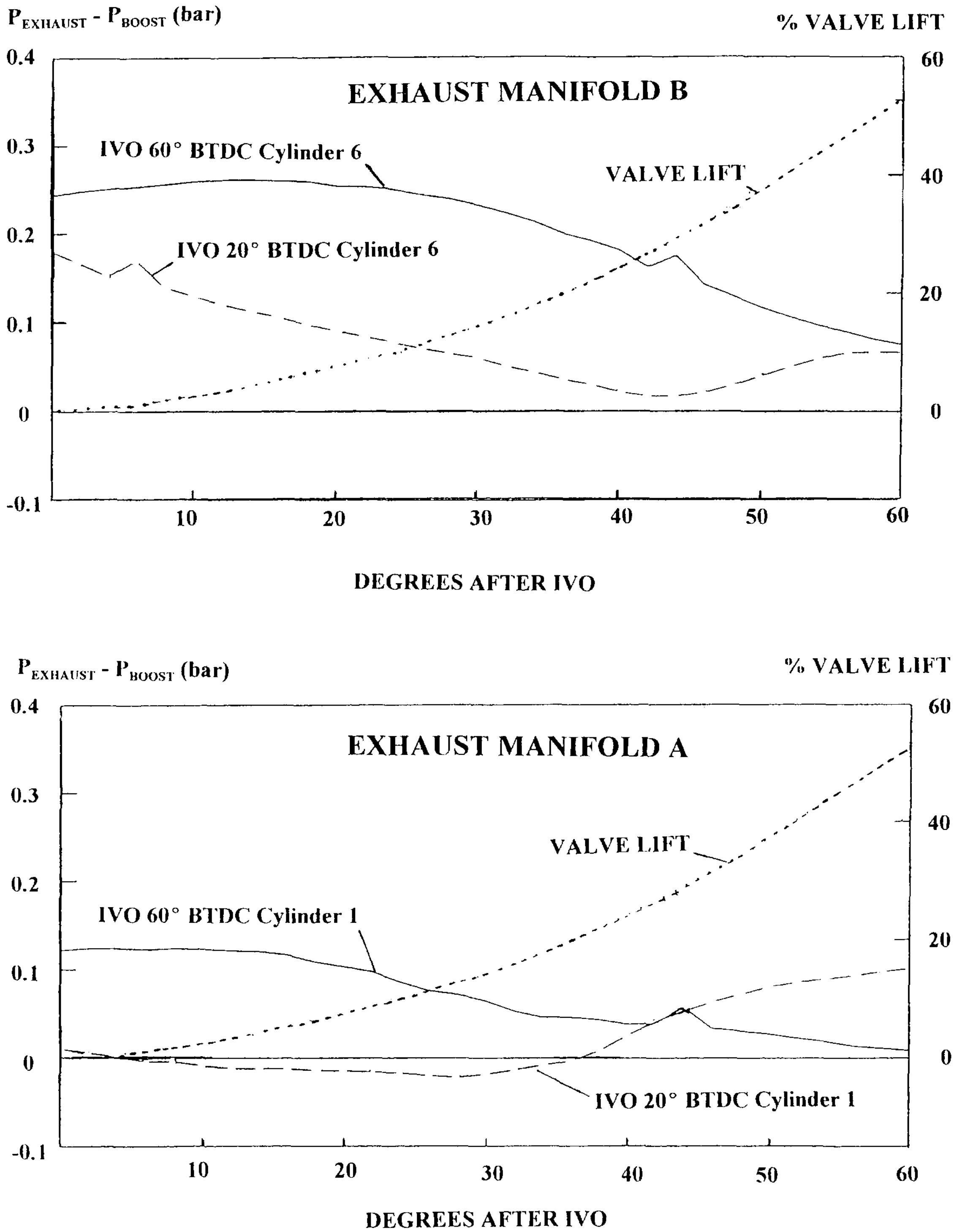

FIG. 6—THE INSTANTANEOUS PRESSURE DIFFERENCE BETWEEN THE BOOST PRESSURE FROM THE COMPRESSOR AND THE BACK PRESSURE FROM THE TURBINE, PLOTTED RELATIVE TO IVO
IVOs of 20° and 60° BTDC, at a power of 30 KW and speed of 1200 rpm for:
(*a*) MANIFOLD A—CYLINDERS 1,2 AND 3.
(*b*) MANIFOLD B—CYLINDERS 4,5 AND 6.
VALVE LIFT IS ALSO SHOWN

manifold B (for cylinders 4, 5 and 6). This creates a larger pressure rise following blowdown in manifold A and a quicker return of any reflected pressure waves.

From Figures 6 and 7, it is clear that delaying the valve events reduces the adverse pressure gradients at valve opening. In other words the difference between the exhaust back pressure and the compressor boost pressure has been

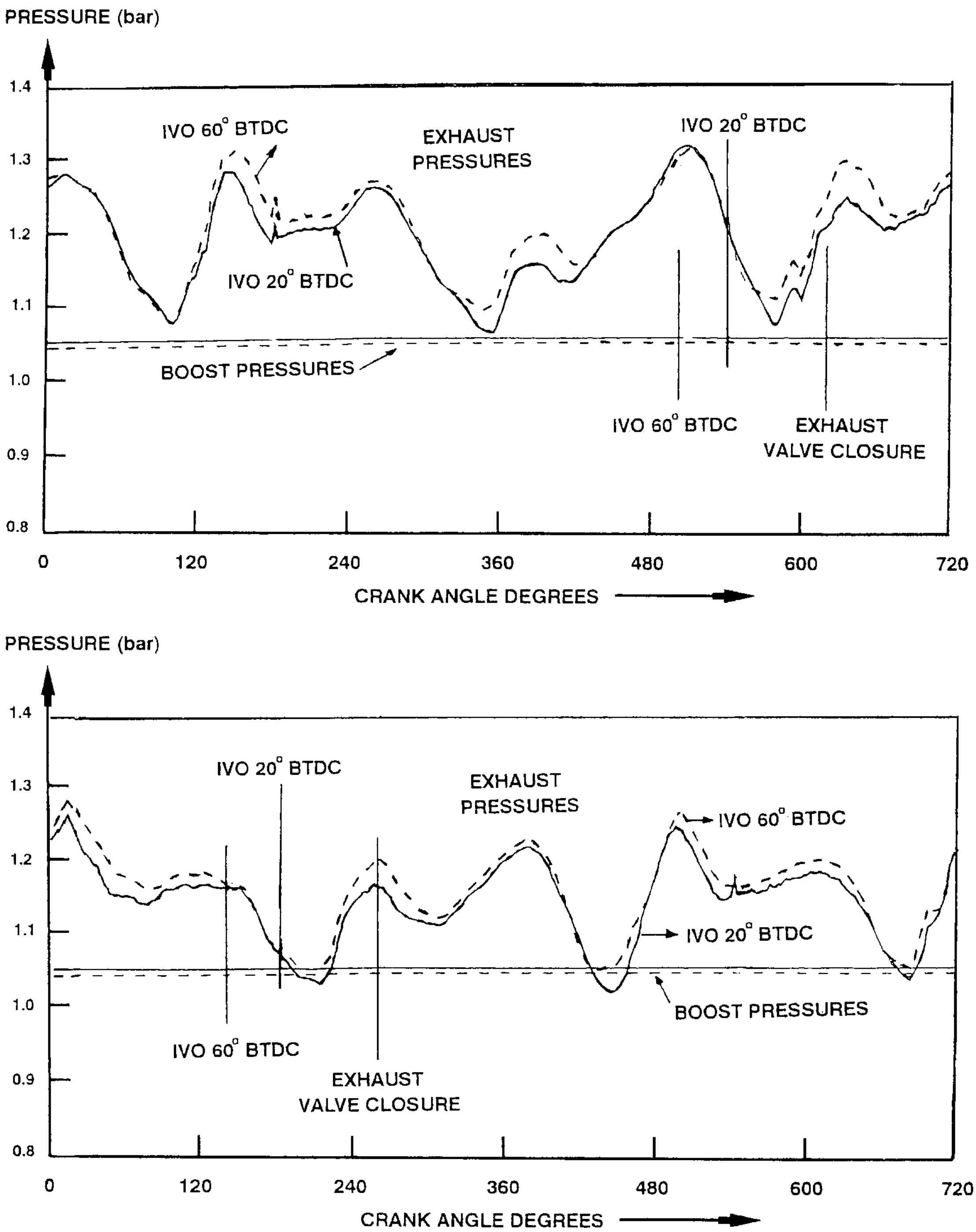

FIG. 7—THE INSTANTANEOUS PRESSURES IN THE INLET AND EXHAUST MANIFOLDS, PLOTTED AGAINST ABSOLUTE CRANKANGLE FOR IVOS OF 20° AND 60° BTDC, AT A POWER OF 30 KW AND SPEED OF 1200 RPM FOR:
(*a*) MANIFOLD A—CYLINDERS 1, 2 AND 3.
(*b*) MANIFOLD B—CYLINDERS 4, 5 AND 6.

reduced. This should help to alleviate the problem of reverse flow into the inlet manifold.

Endurance tests

Endurance tests were performed to assess the durability of the mechanism. The load and the inlet valve phasing were both changed during the tests. After approximately seven hours it became clear that some of the roll pins, which held the eccentrics onto the control rod, had failed due to a combination of fatigue and shear forces. All the roll pins were replaced and the tests continued. Further consideration would have to be given to the manner in which the eccentrics are held onto the control shaft in any future work.

After 27 hours, the surface of the roller followers was found to be excessively pitted due to metal-to-metal contact between either the camshaft or the tappet follower. It is thought that this is attributable to the initial condition of the camshaft prior to installation. Surface roughness figures for before and after engine running are shown in Table 1.

Table 1—*Component surface roughnesses before and after running.*

	Inlet cam lobes	Exhaust cam lobes	Rollers	Tappets
Before	2.4μm	2.4μm	0.4μm	0.4μm
After	0.86μm	0.5μm	1.9μm	1.5μm

The deterioration in surface finish of the tappets and rollers suggests that there was excessive metal-to-metal contact.

Since the preliminary tests with the baseline camshaft indicated an essentially satisfactory performance, it could be concluded that the surface finish on the new camshaft was probably responsible for the wear encountered in the endurance tests. The measurements indicate that a better surface finish on the camshaft would be needed, or alternatively the camshaft has to be run-in with a free roller follower, before the VVT mechanism is installed.

Comparison between measured and modelled engine performance

Comparison of modelled engine performance and the measured results showed good correlations for the inlet and exhaust manifold pressures. Examination of the compressor data also confirmed that delayed inlet valve timings must not be used at high loads, if compressor surge is to be avoided. The model predictions showing the reductions in reverse flow through the inlet valve at low loads, when the IVO angle is delayed, was also proven.

Conclusions

A mechanical VVT mechanism has been built and demonstrated on a 6 cylinder in-line diesel engine. The mechanism was firstly run on a single cylinder test rig prior to full installation on the engine. Some thirty hours of operation were achieved during which it was shown that the valve timing could be varied infinitely with the engine on load. This allowed the benefits of reduced reverse flow from the cylinder into the inlet manifold to be proven, so giving the engine a wider operating range with improved performance at low load. Some mechanical difficulty was experienced with the mechanism, and this needs further development attention. A mechanism control strategy also needs to be investigated.

References

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