VARIABLE VALVE TIMING FOR HIGHLY RATED MARINE DIESEL ENGINES

BY

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ABSTRACT

The article describes the use of an internal combustion engine simulation program to investigate the reverse flow of exhaust gases on a highly rated turbocharged engine at part load. The effect of valve timing changes to eliminate this flow are investigated. The article then goes on to describe the validation of the program using different valve overlap camshafts and fuel on a Paxman 6RP200 and finishes by discussing inlet valve opening strategies.

Introduction

The Problem

Valve timing is often optimized for engine operation at the rated speed and load, an operating point that is rarely encountered. In naval applications of the Valenta Engine (propulsion and generation) a range of fixed valve timings is used, with up to 140° valve overlap. The aim of this work is to investigate the valve timings as a means of optimizing an engine, for both performance at full power and good combustion at low load.

In highly rated diesel engines there is substantial valve overlap around top dead centre. This provides excellent scavenging and cooling of the components at the rated speed and load, since the turbocharger boost pressure is greater than the exhaust back pressure. However, at light loads the boost pressure is less than the back pressure, and significantly less valve overlap would reduce the back flow into the inlet port, hence reducing fouling and improving efficiency.

Research Objectives

The main objectives of the research were:

(*a*) to establish the scope offered by Variable Valve Timing (VVT) in improving the performance and reducing the inlet port fouling of diesel engines at part load;

(*b*) to develop a model of the Paxman Valenta diesel engine, thereby minimizing the need for engine testing of any proposals;

(*c*) to examine the many VVT systems and assess whether they can meet the required variations in timing. This performance appraisal is also to include preliminary assessments of reliability, maintainability and cost.

The following subsidiary aims were also identified:

(*a*) to determine the potential for reducing ignition delay;

(*b*) to assess improvements that might be made to cold starting;

(*c*) to examine correlations for ignition delay, and investigate cycle-by-cycle variations in the ignition delay;

(*d*) to consider the VVT control strategy required to optimize performance during transients.

Project Funding and Progression

The work was funded by a joint SERC/MOD contract lasting two years and beginning in January 1989. A computer simulation of the Paxman 6RP200 was first established in steady conditions with two different valve timings. This was then validated by running the engine at the conditions simulated and comparing the results. Finally some transient simulation and validation work was conducted.

The Paxman Valenta Engine and its Simulation

The Paxman Valenta 6RP200 Engine

Readers will be familiar with the RP200 Valenta range of engines produced by Paxman Diesels Ltd. The engine used for this work was a straight 6, similar to that fitted to the SRMH. The turbocharger fitted was a Napier NA150. The valve timing details are shown in TABLE I.

TABLE I—*Valve timings used in the experimental program*

build	*valve overlap* degrees	*evo bbdc* degrees	*evc atdc* degrees	*ivo btdc* degrees	*ivc abdc* degrees	*effective compression ratio*
Standard	120	65	60	60	45	11·4
Low Overlap	46	82	27	19	33	12·1
Low Overlap phased	46	62	47	—1	53	10·6

abdc: after bottom dead centre
atdc: after top dead centre
bbdc: before bottom dead centre
btdc: before top dead centre
evc: exhaust valve closes
evo: exhaust valve opens
ivc: inlet valve closes
ivo: inlet valve opens

Simulation of the Paxman Valenta 6RP200 Engine

The Simulation Program for Internal Combustion Engines (SPICE) is based on the well-established 'filling and emptying approach' to satisfy the principles of mass and energy conservation[1]. A series of volumes represents the engine to be modelled, and they are interconnected by a system of shafts and/or flow junctions (FIG. 1). SPICE has a modular approach to building the model, so that it is well suited to simulating a turbocharged, intercooled diesel engine. SPICE allows the user to select from a range of sub-models that predict processes within the engine. For example, the in-cylinder heat transfer can be modelled by the Eichelberg[2], Woschni[3], Hohenburg[4] or Annand[5] correlations. The combustion has been modelled here using the Watson correlation[6] with prescribed values of ignition delay from the experimental data.

In SPICE, the turbocharger is modelled from compressor and turbocharger maps that are constructed from tabulated data. The compressor data were from steady flow tests, whilst the turbine data were one-dimensional steady flow predictions using the Ainley and Mathieson method, as modified by Dunham and Came[7]. The intercooler was modelled from manufacturer's data. However, in the light of experimental data from the engine testing, the turbocharger maps were redrawn. The most significant changes were to the efficiency contours on the compressor map in the low boost region. The differences between the rig testing and on-engine performance of the compressor are attributable to several factors:

(*a*) When installed on an engine the compressor performance has to include filter and ducting losses.

(*b*) The compressor is subject to quasi-steady flow; the turbocharger speed is varying continuously.

(*c*) With low boost pressures small changes in temperature (possibly due to experimental errors) lead to large changes in the isentropic efficiency.

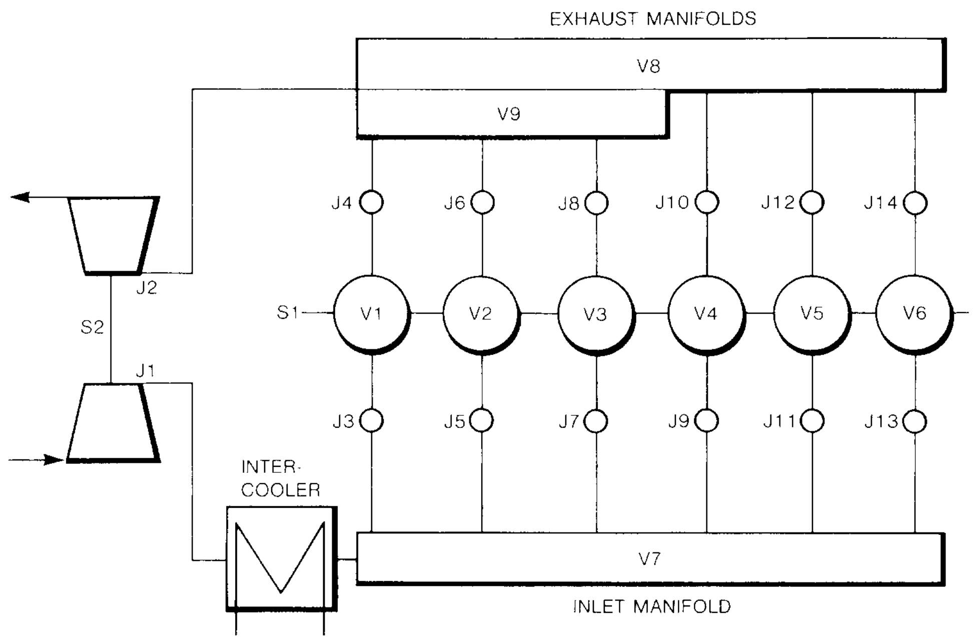

FIG. 1—REPRESENTATION OF THE PAXMAN VALENTA ENGINE BY A SERIES OF SHAFTS (S), JUNCTIONS (J) AND VOLUMES (V)

Model Validation

The model developed with SPICE was validated by tuning the model to match the experimental with the simulated results for the standard camshaft across the load range. The fuelling levels used as inputs to SPICE were the experimentally recorded fuelling rates for 100%, 75%, 50% and 25% load. These fuelling levels were then kept constant during the valve timing investigations, and this led to slight departures from the nominal loads (invariably less than 1%). Without changing the model, it was then used to predict the performance of the other two experimental camshaft timings. FIG. 2 shows a comparison of the simulated and measured engine performance at a speed of 1500 rev/min across the load range. It shows that the valve timing has a negligible effect on the maximum cylinder pressure, and the largest effect is on the reduced air flow for the shorter period inlet valve event. The reduction in air flow leads directly to an increase in the exhaust temperature, thereby tending to maintain the turbocharger power at its original level. FIG. 2 demonstrates excellent agreement between the experimental and simulation results.

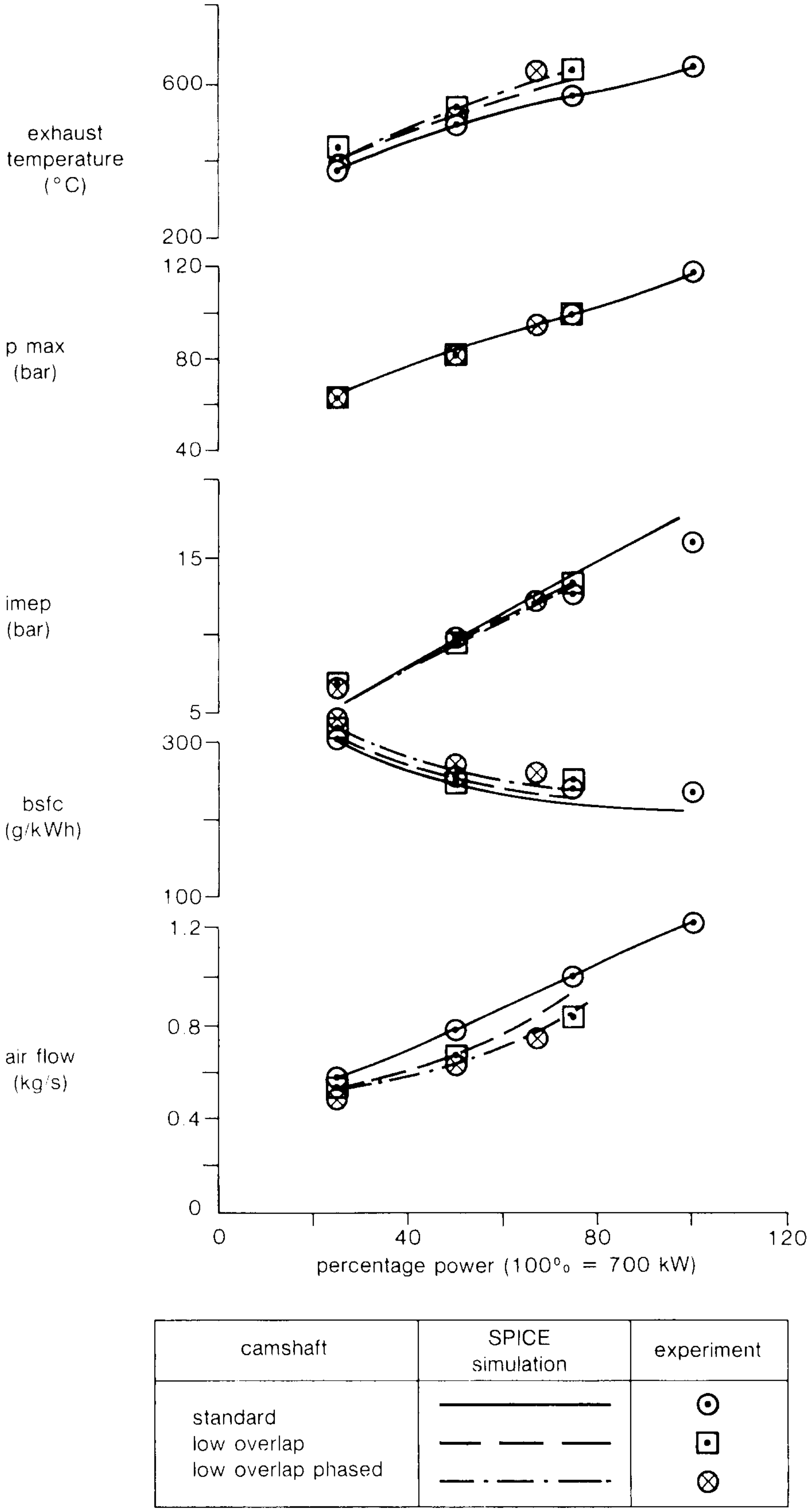

FIG. 2—COMPARISON BETWEEN THE SIMULATED AND EXPERIMENTAL PERFORMANCE OF THE PAXMAN VALENTA 6RP200 ACROSS THE LOAD RANGE AT 1500 REV/MIN

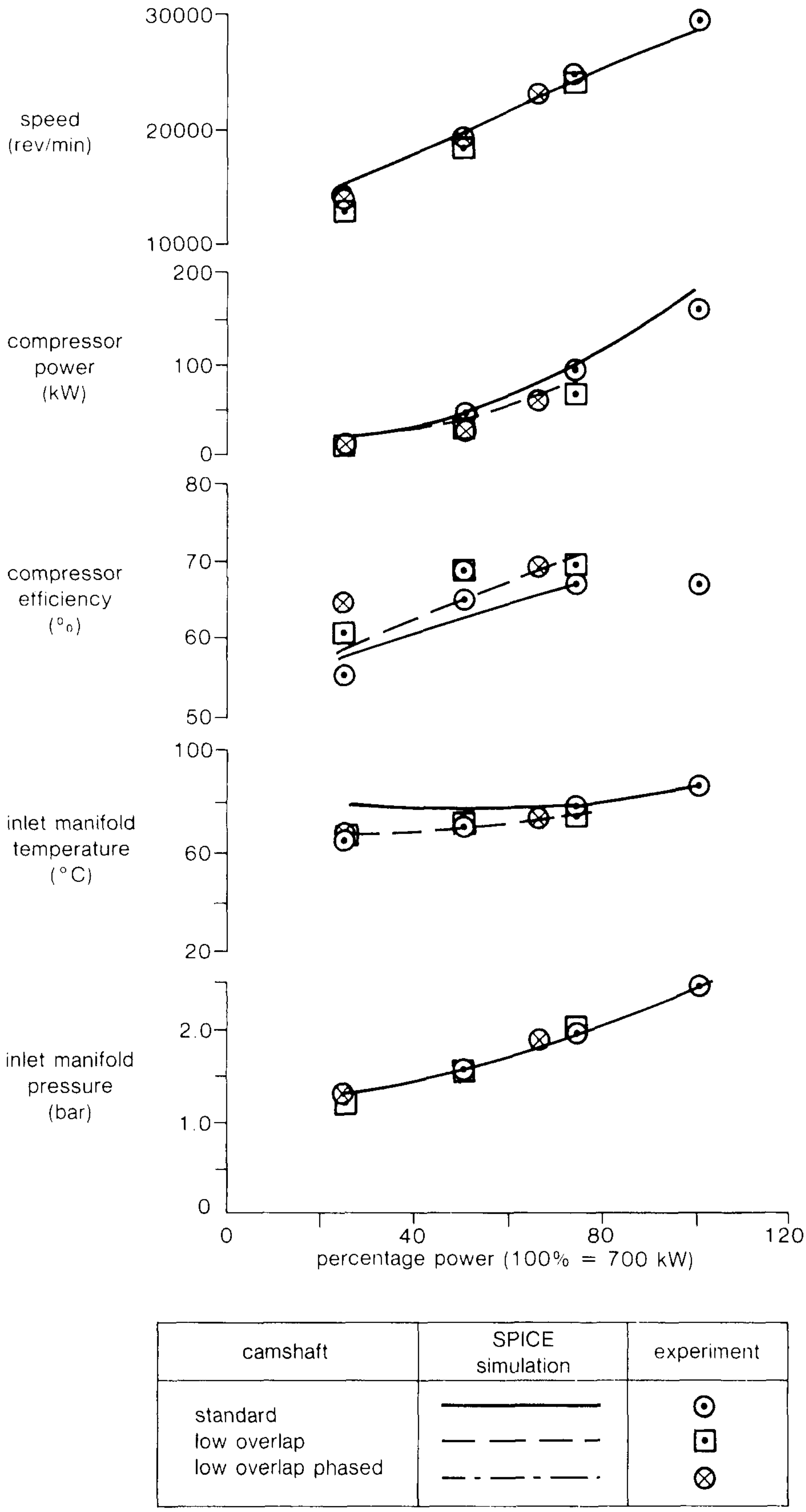

FIG. 3—COMPARISON BETWEEN THE SIMULATED AND EXPERIMENTAL PERFORMANCE OF THE TURBOCHARGER FITTED TO THE PAXMAN VALENTA 6RP200 ACROSS THE LOAD RANGE AT 1500 REV/MIN

FIG. 3 shows a comparison of the measured and predicted turbocharger performance for the same conditions as FIG. 2. It shows that the turbocharger speed and inlet manifold pressure were not affected by the change in valve timing. There was a slight fall in the turbocharger power, and in the inlet manifold temperature for the lower flow rates associated with the reduced inlet valve opening. Again, these results show excellent agreement between the experimental and simulation results. There appears to be poor agreement between the measured and simulated compressor isentropic efficiency, especially at the low fuelling levels. However, under these conditions the pressure ratio across the compressor is very small, and any errors in measuring the compressor delivery pressure or temperature have a profound effect on the value of the isentropic efficiency.

FIGS. 2 and 3 show that the model of the Paxman Valenta engine with SPICE should be capable of giving good predictions of engine behaviour with different valve timings.

Heat release was analysed with data obtained from a Kistler 6121 piezoelectric pressure transducer, injector needle lift and TDC pulses. These were recorded on both an AVL indiscop 647 and PC fitted with a Computerscope data acquisition card. Readings were taken at 0·5° crankangle.

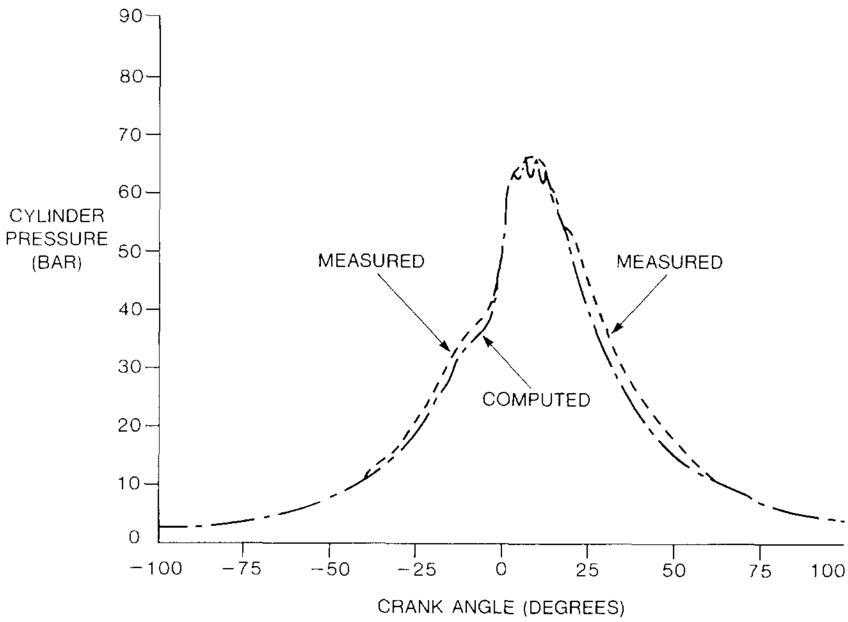

FIG. 4 COMPARISON BETWEEN MEASURED AND SIMULATED CYLINDER PRESSURE AT 25% LOAD, 1500 REV/MIN

A comparison between the measured and simulated cylinder pressure is shown in FIG. 4 for the 25% load fuelling level. This level has been selected since the ignition delay is greatest at light load, and this leads to a significant rapid combustion phase. Good agreement is shown between the measured and predicted cylinder pressure measurements, at an operating point that can be difficult to model. The discrepancies could be due to inadequate modelling of the in-cylinder heat transfer, or errors in the experimental pressure measurement.

SPICE does not include any wave action effects and it is important to establish that this is acceptable, since the gas exchange processes will depend on the instantaneous values of the manifold and cylinder pressures. Watson and Janota[8] suggest that wave action effects can be neglected if the pressure wave takes less than 15° to 20° crank angle to travel the length of the manifold and

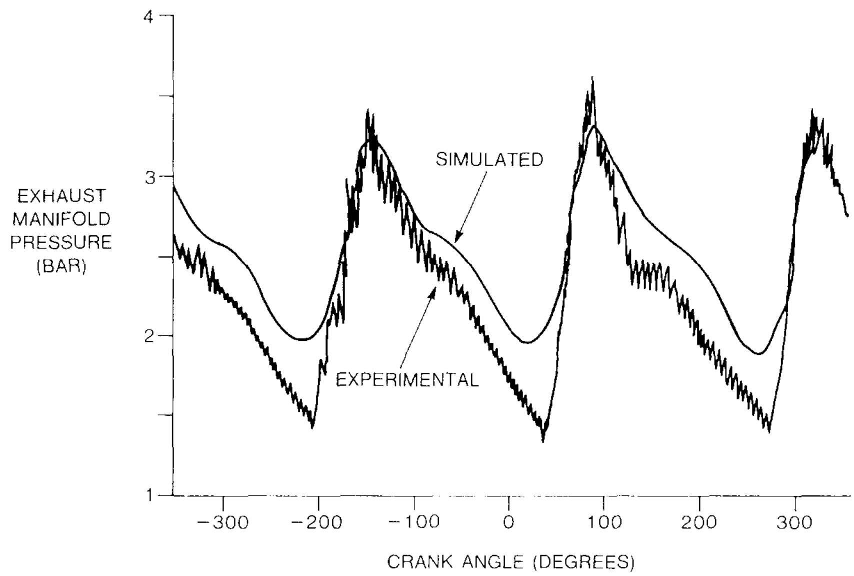

FIG. 5—COMPARISON BETWEEN MEASURED AND SIMULATED EXHAUST MANIFOLD PRESSURE AT FULL LOAD (700 KW), 1500 REV/MIN

back. In the case of this Paxman Valenta engine the exhaust manifold is compact. Even so a typical wave travel time is about 40° crank angle, so it might be thought that wave action effects could be significant. As the exhaust manifold is subject to the greater fluctuations in pressure, a comparison between the measured and simulated exhaust manifold pressure is given in FIG. 5. Full load has been chosen since it gives the greatest fluctuations in pressure; it can be seen that SPICE has given a reliable prediction, from which it was concluded that the absence of wave action modelling was not significant.

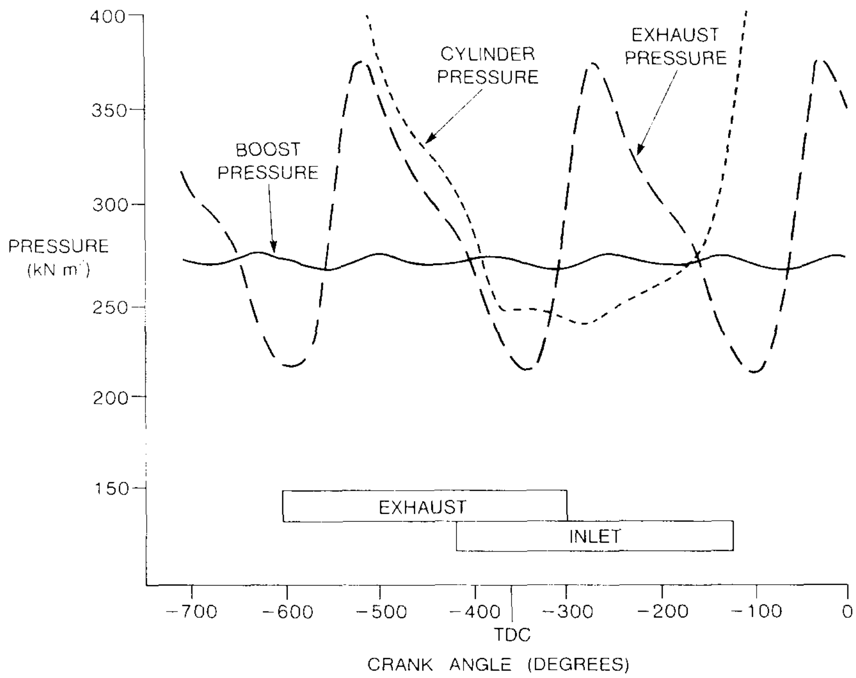

FIG. 6—CYLINDER, INLET MANIFOLD AND EXHAUST MANIFOLD PRESSURES DURING THE GAS EXCHANGE PROCESS AT FULL LOAD (700 KW) AND 1500 REV/MIN FOR THE STANDARD 120° OVERLAP CAMSHAFT

Simulation Results for Valve Timing Control

The Standard Valve Timing Situation

FIG. 6 shows the cylinder, exhaust and inlet pressures, along with the valve events for the standard camshaft (ivo 60° btdc) at full load. It can be seen that initially the cylinder pressure is greater than the inlet manifold pressure. However, as the inlet valve opens slowly there is only a small amount of reverse flow for about 25° crank angle (some 0·4% of the ultimate trapped mass). FIG. 7 is equivalent to FIG. 6, except that the fuelling has been reduced to the level for 25% load. The compressor boost pressure level has fallen significantly (from 2·7 to 1·3 bar absolute), and, although the cylinder pressure is lower, it does not equal the inlet manifold pressure until about tdc. An immediate consequence is that there is reverse flow of exhaust gases into the inlet manifold for about 60° crank angle, and this amounts to about 3·9% of the ultimate trapped mass. It is this flow of exhaust gases into the inlet system that can cause the port fouling problems.

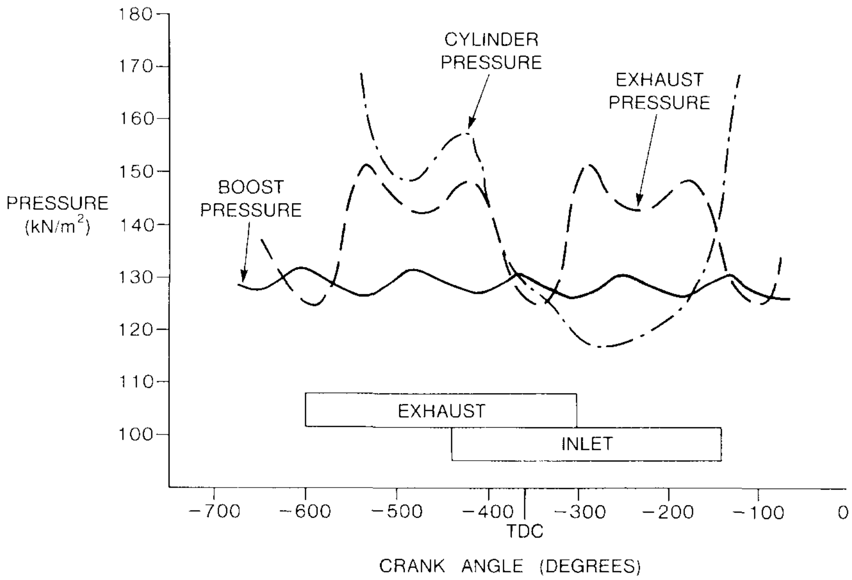

FIG. 7—CYLINDER, INLET MANIFOLD AND EXHAUST MANIFOLD PRESSURE DURING THE GAS EXCHANGE PROCESS AT 25% LOAD AND 1500 REV/MIN FOR THE STANDARD CAMSHAFT

Identification of Valve Opening Strategy

A preliminary modelling study that included the effects of valve overlap control considered variation of both the angle of inlet valve opening and exhaust valve closure. This study suggested that there was no merit in varying the exhaust valve closure time. Indeed, earlier closing of the exhaust valve led to compression of the exhaust residuals at the end of the exhaust stroke, and this would tend to increase the likelihood of exhaust gas flowing into the inlet manifold. Thus the results considered here are for variations to the inlet valve timing alone. This is a useful simplification if a variable valve timing system is to be implemented: there are two types of variation that can be considered. Firstly, the inlet valve closure time can be fixed, such that the valve period is varied. Secondly, the inlet valve period can be fixed, and as the phasing of the valve events is changed, then the inlet valve opening and closure are affected equally.

Discussion of the Inlet Valve Opening Strategies

The effect of delaying the inlet valve opening by 40° crank angle is illustrated by FIG. 8, which shows the flow through the inlet valve for three cases, all with the 25% fuelling level and with constant exhaust valve timing (see TABLE I standard):

(*a*) the standard camshaft with a valve overlap of 120°;
(*b*) inlet valve opening occurring 40° late at 20° btdc;
(*c*) inlet valve opening and closing both occurring 40° late (ivo at 20° btdc and ivc at 85° abdc).

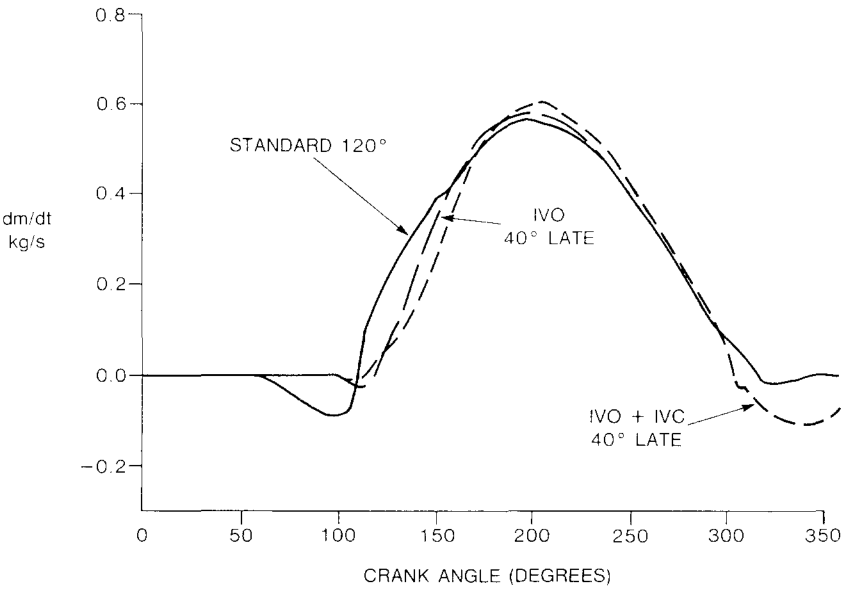

FIG. 8—AIR FLOW RATE THROUGH THE INLET VALVE, WITH A FUELLING LEVEL FOR 25% LOAD AND A SPEED OF 1500 REV/MIN, FOR:
- STANDARD CAMSHAFT TIMING
- INLET VALVE OPENING DELAYED BY 40° CRANK ANGLE
- INLET VALVES OPENING AND CLOSING BOTH DELAYED BY 40° CRANK ANGLE

When the valve events are retarded (so that the inlet valve does not open until close to tdc) then the delayed closure of the inlet valve leads to a significant backflow into the inlet manifold before inlet valve closure. Since this backflow is mostly air, it should not lead to an inlet port fouling problem. The backflow also reduces the air flow and the trapped mass in the cylinder. As the fuelling rate in the simulation is fixed the reduced air flow leads to a higher exhaust temperature (though still significantly lower than the full load exhaust temperatures), but the rise in exhaust temperature does not fully compensate for the reduced exhaust flow, and the turbocharger power falls. However, as with the fixed inlet valve closure case, there is little effect on either the mean compressor boost pressure or the turbine back pressure.

If FIG. 9 is compared with FIG. 10, then it can be seen that when the inlet valve opening is delayed by phasing the valve events (FIG. 10), then the pressure difference across the inlet valve is greater than when the inlet valve opening is varied without changing the inlet valve closure (FIG. 9). With a fuelling level for 25% load, the reverse flow before the inlet valve closure is 17·84% of the ultimate trapped mass. However, it would seem that this reverse flow from one cylinder coincides with the inlet valve opening of the next cylinder, so the reverse flow helps to maintain the inlet manifold pressure at an important time, thereby reducing the reverse flow of exhaust residuals from the cylinder into the inlet manifold at inlet valve opening. This results in the reverse flow at the 25% load level being reduced from 3·9 to 0·12% of the final trapped mass.

When the inlet valve opening is delayed by 40° crank angle (from 60° btdc to 20° btdc) there is a negligible effect on the specific fuel consumption. However, when this delay is obtained by a phasing variation, then a 1·5% increase is predicted in the fuel consumption. This requires to be validated by an engine test, as it may be an artefact of the simulation.

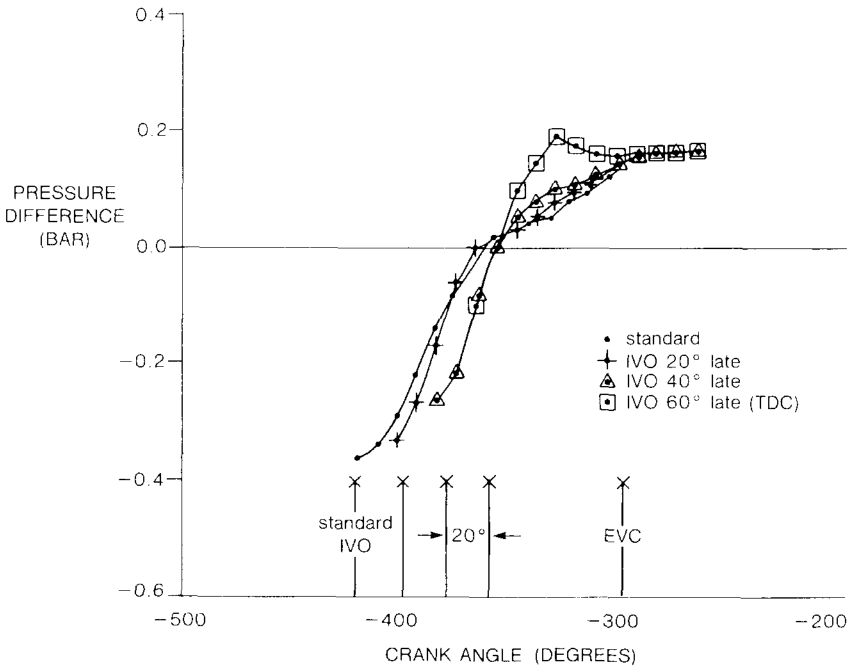

FIG. 9—THE EFFECT ON THE INSTANTANEOUS PRESSURE DIFFERENCE ACROSS THE INLET VALVE OF DELAYING THE INLET VALVE OPENING, WITH 25% LOAD AT 1500 REV/MIN

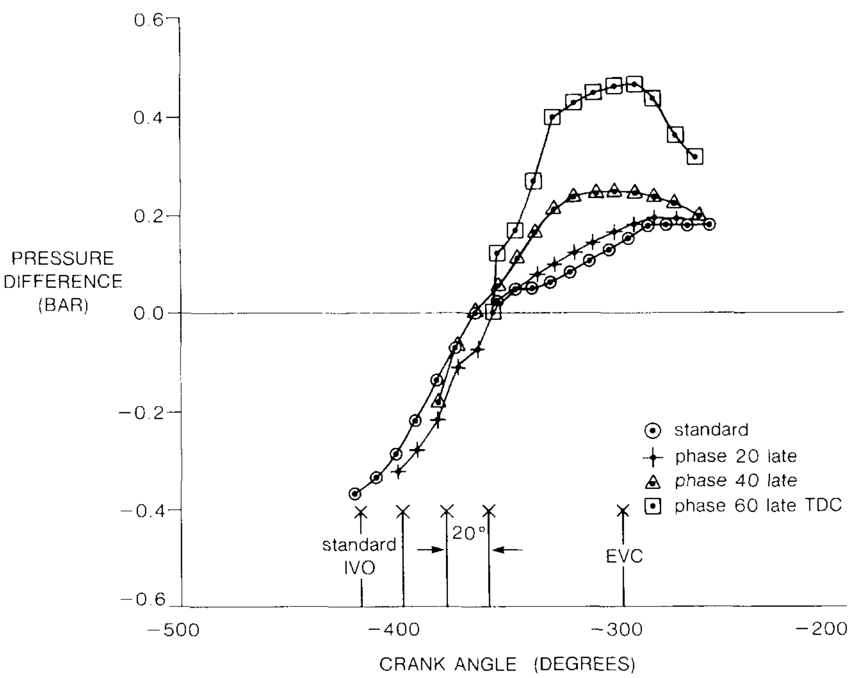

FIG. 10—THE EFFECT ON THE INSTANTANEOUS PRESSURE DIFFERENCE ACROSS THE INLET VALVE OF PHASING THE INLET VALVE OPENING, WITH 25% LOAD AT 1500 REV/MIN

Phasing the inlet valve events by 40° crank angle delays the inlet valve closure from 45 to 85° abdc, and the effective compression ratio is reduced from 11·4 to 7·8. This leads to questions of how the efficiency be affected by a lower compression, and how the ignition delay be affected. This is discussed further by Charlton *et al.*[9] and Leonard[10].

Transients

The study of transient performance (for a step increase in load from 0 to 75%) indicated that the valve overlap should be increased as soon as the increased load was demanded. This was contrary to expectations, and may be a result specific to the load/engine/turbocharger combination. This is discussed further by Charlton *et al.*[11]

The Future

The work undertaken on the MOD/SERC contract is now being continued by a DRA funded University agreement. This work is to identify a suitable VVT mechanism, carry out rig trials and finally build a system for trials on the 6RP200 engine at Pyestock. It is scheduled for completion in 1994.

Conclusions

The conclusions of the modelling and validation work so far are that a variable valve timing technique can make a useful contribution to the reduction of inlet port blowback and hence an improvement in operating efficiency of a high speed diesel engine. The simulation used represented the engine well and is now able to provide a sound base from which to develop this work. When differences did exist between the simulated and measured performance they were as likely to be measurement as simulation errors.

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