A COMPARISON BETWEEN EXPERIMENTAL AND ANALYTICAL TRANSIENT TEST RESULTS FOR A TURBOCHARGED DIESEL ENGINE

D.E. WINTERBONE, BSc, PhD, CEng, MIMechE

R.S. BENSON, DSc(Eng), PhD, CEng,FIMechE, FIMarE, MemSAE

G.D. CLOSS, MSc, PhD

A.G. MORTIMER, BEng

SYNOPSIS A computer controlled diesel engine test-bed is described. The system is capable of excursions into regions of very low air-fuel ratio operation without damage to the engine; this enables the engine "steady-state" characteristics to be measured for off-design operating conditions. A computer-based data acquisition system was developed to acquire test-bed results under engine transient operating conditions. An analogue computer model of the engine was constructed using the steady-state characteristics of the engine as obtained from on- and off-design. This model operated in real-time and was suitable for both studies of engine control systems and the testing of control hardware. Results obtained from the test-bed and the model are compared. It is shown that good agreement can be achieved by minor modifications to the model based on steady state results.

1. INTRODUCTION

The turbocharging of diesel engines is now very common and in the future it is probable that the majority of engines will be turbocharged. Turbocharging improves the steady state output of the engines by increasing its specific power output; it also lowers the specific fuel consumption over a wide load range. However, such benefits do not accrue to the transient performance and as the engine rating is increased by turbocharging its ability to accept sudden increases in load deteriorates. Engines rated at about 14 bar b.m.e.p. are very likely to stall when subjected to a 100% step load application.

Such transient load applications occur in:

(a) Marine power units, where rapid load variations take place when changing from full ahead to full astern using a reversing gear box.

(b) Electrical generating equipment, when under certain conditions, large loads are applied almost instantaneously.

The turbocharged diesel engine is unable to accept a large step change of load due to turbocharger lag. When a sudden load is applied to the engine its speed drops; the governor senses this drop in speed and causes more fuel to be injected to compensate the error. If the increase in fuel could be matched by a proportionate increase in the air supplied to the engine a rapid and efficient increase in power would occur. This is not the case, the turbocharger speed, and hence the air supplied, rise relatively slowly due to turbocharger inertia, resulting in very rich conditions and poor combustion. The power output of the engine will only increase when the air supplied by the turbocharger satisfies the demand of the fuel. If the power output of the engine does not equal the load the engine stalls; if the output exceeds the load the engine will accelerate to its steady state speed droop. During such transients it is probable that the engine will enter regions of severe overfuelling and its output under these conditions will be of paramount importance.

The high thermal loads encountered preclude measurement of such outputs by normal steady state testing. To obtain these values a test rig was constructed which allowed a turbocharged diesel engine to operate under transient load conditions. The rig comprises a six-cylinder engine coupled to a fast response hydraulic dynamometer. Both the dynamometer and the engine fuel system are controlled by a computer which also acts as a high speed data acquisition system. The speed of the control and acquisition system also enabled excursions to be made into regions of very low air-fuel running without damage to the engine. In this way both transient and off-design results were obtained for the engine.

Analogue and digital computer models have been constructed to represent the engine; these were based on data obtained from the abovementioned tests. The former model only will be discussed in this paper. The models are based on the technique described by Ledger et al (ref.1,2,3)* , which in turn was largely based on work by Whitehouse et.al (ref. 4). The digital model was a direct development of one written by Furukawa (ref.5) whilst the analogue one was based on the technique described in (ref.6). The model is based on a quasi-steady calculation technique which assumes that the airflow can be evaluated by steady-state matching methods and that differential equations are required only for the mechanical components of the engine. Such programs are very dependent on empirical data and these were obtained from the test bed. The experimental results were processed into the form required by the programs and are presented thus in this paper.

2. GENERAL DESCRIPTION OF APPARATUS

Basic requirements

The test bed had to be capable of performing the following tasks:

(1) Measurement of the engine response to transient step changes in load

* references given in Appendix 1.

(2) Measurement of the off-design characteristics of the engine by short-term excursions into regions of low air-fuel ratio.

These requirements were met by coupling the engine to a fast-response water brake and logging the data by digital computer. Fig.1. is a schematic diagram of the overall system, from the engine test bed through to the print-out of results. The system may be conveniently split into three sections; the test-bed, comprising the mechanical components; the data acquisition system, including analogue-digital conversion equipment and the computer; instrumentation. These are described below.

To achieve the low air-fuel ratios required in (ii) above it was necessary to remove the turbocharger and run the engine on shop-air or naturally aspirated.

Engine test-bed

The engine used was a Ruston and Hornsby 6YEX Mk.II turbocharged diesel engine. This is a six cylinder water-cooled four stroke engine with open type combustion chambers and direct injection. The salient parameters are given in Appendix 2.

A Heenan and Froude F027 fast response hydraulic dynamometer was also used. This is a variable fill machine for which the load absorbed is independent of the speed and controlled only by the setting of a butterfly valve in the water output system. The engine and dynamometer were coupled by a British Hovercraft Ltd. strain-gauge torque meter. The turbocharger fitted was a Holset model 4LEV-305 consisting of a single stage centrifugal compressor and double entry single stage radial turbine. The characteristics of the turbine and compressor are shown in Figs.2 and 4 respectively. These were measured on a "cold" rig at U.M.I.S.T. The standard engine is fitted with a CAV 'NN" series monoblock fuel pump. A maximum-speed governing system is built into the pump. For these tests it was desirable to be able to vary the governor characteristics; to this end the following modifications were made. A printed circuit motor, Printed Motors Ltd., Model G19M4KT, of 1kW continuous output was mounted on the engine and the armature was connected to the exposed end of the fuel pump rack; a linear position transducer was also fitted to the rack. A simple control system was developed using these components and is shown in Fig.3. The speed-droop characteristic was matched to that of the original mechanical governor and generated electrically. Because of the dynamics of the system components the overall response was effectively controlled by the filter.

A Celesco Model 103 smokemeter was fitted into the exhaust pipe downstream of the turbine. This is a photo-electric device which measures the opacity of the gas passing through the exhaust pipe. Its response time was much faster than the gross rate of change of smoke density.

To enable the engine to run at low air-fuel ratios it was necessary to remove the turbocharger and run the engine either naturally aspirated or using a pressurized air supply. Removal of the turbocharger necessitated the inclusion of a variable area orifice in the exhaust pipe to simulate the turbine.

Both the oil and water temperatures were controlled by motorized flow splitter valves which altered the quantity of liquid passing through heat exchangers.

Data acquisition and control

Data acquisition and control of the test-bed were both performed by a digital computer. This was necessitated by the requirement for synchronizing these activities. The computer used was a Digital Equipment Corporation (DEC) PDP-15 machine which was connected to the test-bed via Microconsultants analogue-digital converters (ADC) and digital-analogue converters (DAC). The overall arrangement is shown diagrammatically in Fig.1.

Data acquisition

The project specification required acquisition of gross engine performance during a load transient and also acquisition of detailed pressure-crankangle data under off-design conditions. These were two quite different requirements and necessitated different data acquisition philosophies; both used the Microconsultants interface which was located in the engine test cell- this was advantageous during initial equipment tests.

An existing computer program was modified to enable pressure-crankangle data to be obtained from the Ruston 6YEX. This program has been described in (7).

A new program was written to acquire the gross engine parameters. This took the form of a 'look-up' table which could be accessed every crankangle degree. The table was essentially an array of channel numbers versus crankangle; entry to the table switched the acquisition system to the channel whose number had been read, the system then acquired and stored the value of the parameter on that channel. The number of channels being accessed during this project was 15, the acquisition sequence was entered 6 times every engine revolution.

Control

Both acquisition programs required synchronization with the engine. The pressure-crankangle program was synchronized manually.

The transient test program was more complicated. It was necessary for the computer to control the engine operating conditions, viz engine speed and brake load. This was achieved by generating the speed and load demand in the transient acquisition program and feeding these signals as reference points to local analogue control loops on the engine via DACs in the interface. When the engine has been switched to computer control it was possible to change the operating conditions from the computer by typing new conditions on the console teletype.

A typical test sequence consisted of switching the engine to computer control, setting the speed and load (usually nominal no-load), running the engine for a specified number of cycles at this low load, stepping the dynamometer load demand to full load, continuing the test for 15-20 seconds and then slowly removing the engine load. Throughout the test data was acquired by the program. Immediately after the test response graphs could be plotted on a visual display unit (VDU) coupled to the computer.

The computer and interface

Both of these components are described in Appendix 3. The maximum rate of data acquisition of the

overall system was limited to 24K words/second by the rate of data transfer from computer core to disk.

Instrumentation

The instrumentation was based on techniques which had proved successful in previous work. Much of this has been described by Benson (ref.7) and is listed in appendix 4.

3. EXPERIMENTAL PROCEDURE

The different natures of the two prime objectives necessitated very different test procedures. These are described below.

Measurement of engine performance under over-fuelling conditions

To measure the output of the engine under conditions of extreme over-fuelling it was necessary to operate the engine with insufficient air for complete combustion. Such conditions can be achieved with a turbocharged diesel engine under transient operation. If the turbocharger is removed and the engine connected to a separate compressed air supply any combination of air and fuel may be achieved. However, over-fuelling leads to severe thermal loading with ultimate failure of engine due to piston cracking or seizure; these failures can be avoided if the duration of over-fuelling is short enough.

The automation of the test-bed made it ideal for the performance of low air-fuel ratio tests. These were performed as follows. The dynamometer was set in a control mode enabling it to maintain constant engine speed and the engine output was controlled by altering the rack position, x_d, in Fig.3. This meant that the only effect of increasing the fuel input was to increase the load; the dynamometer maintained the speed constant. Hence it was possible to go practically instataneously from a low load to full load at a fixed engine speed; this is essentially what happens during the initial stages of a transient before either engine or turbocharger speed have a chance to respond to a load change. At this stage of the project gross engine parameters were recorded by Solartron data-loggers with an acquisition rate of 1 data channel/second, for this reason over-fuelling conditions were maintained for approximately 10 seconds. Pressure-crankangle diagrams were recorded during this period using the high-speed data acquisition system; 8 consecutive cycles were acquired to reduce effects of cyclic irregularity.

The following gross engine parameters were recorded by low-speed data-loggers:

Inlet Manifold Pressure
Inlet Manifold Temperature
Exhaust manifold temperature (one in each manifold).
Engine torque (by torquemeter).
Fuel pump rack position.

Engine oil and water temperatures were also monitored on data-loggers to avoid possibility of engine damage.

Measurement of transient response

To limit the total number of tests it was decided to investigate the "constant speed" load application performance of the engine. The engine was run under computer control for all of these tests. Ideally the tests should have been between no-load (0 bar bmep) and a high load but, due to the characteristics of the dynamometer, it was necessary to operate from a minimum load of 1 bar bmep. This was effectively a compromise between minimum load and maximum response rate. The tests were "timed" by counting the number of engine cycles; typical timing was:

50 cycles at 1 bar bmep (initial conditions)
Step load change to 10 bar bmep.
300 cycles at 10 bar bmep (final conditions)
Gentle load reduction to 1 bar bmep.
At 1800 RPM this would take 23 seconds.

During this period the following readings were recorded at 6 times per engine revolution by the computer acquisition system:

Engine speed
Turbocharger Speed
Turbine Inlet Temperatures (2 low speed thermocouples, 1 fast thermocouple).
Applied Load (dynamometer load cell).
Shaft Torque (torquemeter).
Rack Position.
Venturi Nozzle Throat Depression.
Inlet Manifold Pressure.
Inlet Manifold Temperature.
Exhaust Smoke.

Oil and water temperatures were maintained constant by a closed loop control system.

Twenty transient load application tests were performed over a speed range from 1000-1800 RPM. These tests include some no-load to intermediate load step changes. Detailed results are given in Section 4.2 for full-load applications at 1000 RPM and 1800 RPM, the extremes of the speed range. The engine only stalled for a full-load application at 1000 RPM; this case is considered below.

4. ANALOGUE COMPUTER MODEL

The analogue computer model was based on the method described by Ledger et al (1,2,3). This is essentially a quasi-steady model of the system which extrapolates steady-state engine and turbocharger data into regions of transient operation. Ledger discusses the simulation of a highly-rated turbocharged Ruston and Hornsby 6APC diesel engine. In common with most highly turbocharged engines this machine had a large valve overlap to give effective scavenge; this created an airflow modelling problem which was overcome by a two-orifice scavenge model suggested by Whitehouse et.al. (ref.4). The agreement between analytical predictions and empirical results provides confidence in this modelling technique.

The analogue simulation developed for this project differs from that of Ledger in the following respects:

(a) A simplified scavenge model was used because of the small valve overlap. The scavenge mass flow was estimated as 5% of the "pumped" mass flow.

(b) The Holset inward-flow radial turbine was modelled.

(c) Turbocharger pulse factor was redefined (see section 5.1.4).

(d) A new model of engine temperature rise was formulated to allow for exhaust manifold heating. (See section 5.1.3).

(e) A refined air mass flow balancing technique was introduced based on (6). This enabled a real-time analogue simulation to be achieved.

(f) Curved compressor characteristic speed lines were simulated using variable diode function generators.

The model was implemented on two Redifon-Astrodata Ci-175 analogue computers which were "slaved" together. This made available 160 high gain amplifiers, 24 multipliers/dividers and a total of 120 potentiometers. The model also utilized variable diode function generators to represent non-linear characteristics. The analogue computers were interfaced to a Digital Equipment Corporation (DEC) PDP-10 digital computer which was used for automatic setting of potentiometers, analogue computer component checks and automated data-logging of results.

5. EXPERIMENTAL RESULTS

The engine was initially run in turbocharged form to obtain the system steady state characteristics. These are shown on the compressor map, Fig.4.

Steady state tests including over-fuelling

The tests performed are summarized in Appen.5. These include both on- and off-design conditions and were aimed at evaluating the steady-state engine performance over a very wide range of air-fuel ratios. The effect of speed was investigated over the range 1000-1800 RPM at effective boosts of 1.0, 1.4, and 1.8, achieved by means of a compressed air supply. The results were processed into the form required by the simulation model (section 4) and are presented thus in this paper.

Willan's lines

Fig.5 shows Willan's lines drawn for the engine under steady state naturally aspirated conditions. The fmep of the engine at various speeds was evaluated by these means and compared with the values obtained from pressure-crankangle diagrams. The agreement between the two methods was reasonable.

Relative thermal efficiency

Fig.6 shows the variation of imep with fuel injected per cycle. The imep was calculated as the sum of bmep and fmep, the latter being evaluated from Fig.5. The diagram shows that imep is practically independent of engine speed and, above an air-fuel ratio of 30:1, is independent of boost and directly proportional to the mass of fuel injected. As air-fuel ratio falls below 30:1 the imep ceases to be directly proportional to fuel injected and less imep is obtained than is potentially available. The ratio of actual imep to potential imep has been termed 'relative thermal efficiency' by Ledger and Walmsley (ref.1). Fig.7. shows a graph of this parameter plotted against air-fuel ratio for this engine; it was considered to be unity at an air-fuel ratio of 30:1.

Engine temperature rise

A crucial factor in assessing the transient response of a turbocharged engine is the rate of acceleration of the turbocharger. This is directly proportional to the available torque on the turbocharger shaft, viz, the difference between turbine and compressor torques, which is in turn very dependent on the engine temperature rise. Ledger and Walmsley (ref.1) showed that engine temperature rise, ΔT, may be expressed as:

$$\Delta T = \frac{K}{1 + (A/F)_{trapped}}$$

where K is essentially a measure of the proportion of fuel energy passed to the exhaust pipe; K was evaluated from steady state results. During these tests it was observed that although a step change of fuel brought about a rapid change in engine temperature rise this did not correspond to that measured on steady state tests. A typical variation of non-dimensional exhaust temperature with time is shown in Fig.8, approximately 65% of the total change occurs in the first 10 seconds, the remaining 35% of the change taking approximately 10 minutes. Detailed examination of the results showed that the thermcouple time constant was less than 10 seconds.

Hence the long term transient must be due to other causes: two are suggested here:

(1) Heat transfer to the manifold
(2) Radiation and conduction losses from the thermocouple.

Calculation showed that, considering very unfavourable operating conditions and estimating heat transfer parameters pessimistically, the radiation error for the thermocouple was about 35°C. Conduction losses were negligible for this particular system.

Although radiation losses undoubtedly affected the value of temperature indicated by the thermocouple it is postulated that the major effect was due to heat transfer to the manifold. This means that the equation for ΔT derived by Ledger is a simplification of that existing during a transient and would overestimate the response of the turbocharger. These tests suggested that ΔT should be modelled as follows:

$$\Delta T = \frac{K + K_D (1-e^{-at})}{1 + A/F}$$

where K is a factor evaluated from the temperature rise 10 seconds after a load change (imposed by thermocouple response) and shown as a function of air-fuel ratio in Fig.9, and K_D and a are constants evaluated from the long term thermocouple transient.

The initial and secondary (steady-state) temperature rise is shown as a function of air-fuel ratio on Fig.10. Steady-state results could only be obtained for tests where the air-fuel ratio was greater than 17; at lower values damage to the engine became likely.

Turbocharger pulse effect

The turbine characteristics shown in Fig.2. were obtained on a constant pressure test rig. When mounted on the engine the turbocharger is subjected to rapid fluctuations in pressure. Owing to the inertia of the rotor assembly the blade speed remains practically constant; hence the efficiency changes due to the changing ratio of gas to blade velocity. Whitehouse (ref.4) introduced an 'apparent' constant pressure system to estimate the turbine efficiency under pulsating conditions. This allowed the ratio of actual turbine power to isentropic turbine power, termed the pulse factor, to be evaluated. Ledger and Walmsley (ref.1) applied this technique to an axial turbine and obtained the relationship shown in Fig.11a. The

pulse factor for this particular system is shown in Fig.11b; it is a function of both pressure ratio and engine speed. Similar effects have been noticed previously (ref,8,9) with radial turbines operating under pulsed conditions.

To simplify the inclusion of the pulse effect in the simulation programs a redefinition of the problem was attempted. Instead of evaluating the pulse factor as actual power divided by isentropic power it was decided to evaluate an "effective turbine expansion ratio" defined as $K(p_{ex}/p_o)$ where K varies as shown in Fig.12. This reformulation enables the pulsation effect to be related simply to $\dot{m}\sqrt{T_{ex}}/p_o$ N_{eng}, the independent variable is effectively a "cyclic mass flow parameter". The ratio may vary with number of cylinders and exhaust configuration and further investigation is necessary.

Transient results

Having processed and included the steady-state results in the quasi-steady computer model it is possible to compare its transient predictions with engine transient results. All of the initial comparisons showed that the model was very optimistic compared to the experimental results. The effect of increasing the "firing delay" was tested on a digital computer program; no significant improvement was noted at reasonable delays. Finally it was considered that the only explanation of the discrepancy could be that certain steady state results differ from those encountered during a transient due to 'off-design' heat transfer and friction effects. These losses were introduced into the model by reducing the instantaneous value of imep and engine temperature rise by 5% during the transient. The results discussed below include this modification of these two parameters, otherwise all values are obtaine direct from steady-state tests.

Two cases are presented here. Both are from nominal no-load (1 bar bmep) to 10 bar bmep. at speeds of 1800 RPM and 1000 RPM. For the former case the engine accepts the load change and runs satisfactory. In the latter, the load is too great for the engine at the low speed and it stalls. The computer predicts both of these responses. The agreement between measured and computer results is typical of that achieved throughout the whole spectrum of tests.

Considering Fig.13 the following features should be noted. The exhaust gas temperature calculated does not include any allowance for the thermocouple time constant; hence this is not a true comparison. The dynamometer response has been simplified in the computer model; this could have an effect on the initial response of the engine. The measured turbocharger results all exhibit a sharp rate of change of response at approximately 1½ seconds; it has not been possible to produce this effect on the models and no wholly satisfactory explanation has been suggested.

Fig.14 shows the results for a load application of 10 bar bmep with the engine running at 1000 RPM. The computer model quite accurately predicts that the engine will stall. The results should only be compared up to about 3 seconds because thereafter the dynamometer fails to maintain the nominal 10 bar load. Over this period the agreement is good.

6. FUTURE WORK

This test-bed will be used for investigations into methods of improving the transient performance of diesel engines, both from their ability to accept rapid load changes and a diminution of smoke during such changes. The first method to be tried will be air injection into the turbocharger compressor, a technique already investigated analytically with another engine (ref.5). Secondly acceleration of the turbocharger using an oil-driven Pelton wheel will be investigated. Such novel devices will require a different approach to engine control system design and these new controllers will be assessed on this test-bed.

7. CONCLUSIONS

Fully automated engine test-beds for transient investigations enable detailed engine data to be recorded at sufficiently high acquisition rates to describe the engine response. Such beds are also useful for obtaining data from the engine operating in regions of extremely low air-fuel ratio.

It is possible to develop useful mathematical models of turbocharged diesel engines based on a sufficiently wide range of steady-state data. In these quasi-steady models it is necessary to inc include an allowance for the pulsations in the exhaust manifold; this is more complex for radial flow turbines than axial flow ones.

It is necessary to "trim" the steady-state data obtained from the test-bed to get good agreement for engine transient response. A diminution of 5% steady-state values was applied to two parameters during transient calculations.

It is unlikely that quasi-steady models of turbocharged diesel engines will ever achieve more than a qualitative predicting accuracy at the design stage of an engine system. Future work may improve this situation by cross-fertilizing the quasi-steady model from a more comprehensive one (ref.10).

8. ACKNOWLEDGEMENTS

The authors would like to thank all members of the Mechanical Engineering Department or Control Systems Centre who helped in any way with this project.

Particular thanks go to Professor H.H. Rosenbrock, Control Systems Centre, for permission to use the analogue computer; Messrs. Shamsi, Mandl and Kenyon for allowing their results to be used; and Messrs. Quare and Wrigley for writing the computer programs.

9. APPENDIX 1

References

1. LEDGER, J.D. and WALMSLEY, S. Computer Simulation of a Turbocharged Diesel Engine Operating under Transient Load Conditions. S.A.E. Paper No. 710177. January 1971.

2. BENSON, R.S., LEDGER, J.D., WHITEHOUSE, N.D. and WALMSLEY, S. Comparison of Experimental and Simulated Transient Responses of a Turbocharged Diesel Engine. S.A.E. Paper No.730666, June 1973.

3. BENSON, R.S., LEDGER, J.D. and WHITEHOUSE, N.D. Dynamic Modelling of a Turbocharged Diesel Engine. I.Mech.E. Conf. on Engine Performance Modelling, London, May 1972.

4. WHITEHOUSE, N.D., STOTTER, A. and JANOTA, M. S. Estimating the Effects of Attitude, Ambient Temperature and Turbocharger Match on Engine Performance. Proc. Instn. Mech. Engrs. 1963-64, 178 (1), p.483.

5. LEDGER, J.D., BENSON, R.S. and FURUKAWA, H. Improvement in Transient Performance of a Turbocharged Diesel Engine by Air Injection into the Compressor. S.A.E. Paper No.730665, June 1973.

6. BASTO, G. A Hybrid Simulation of an Automotive-Type Gas Turbine. M.Sc. Dissertation, University of Manchester, January 1973.

7. BENSON, R.S. and PICK, R. Recent Advances in Internal Combustion Engine Instrumentation with particular reference to High-Speed Data Acquisition and Automated Test Bed. S.A.E. Paper No.740695, 1974.

8. BENSON, R.S. and SCRIMSHAW, K.H. An Experimental Investigation of Non-Steady Flow in a Radial Gas Turbine. Proc. Instn. Mech. Engrs. 1965-66, 180 (3J), pp.74-85.

9. MIYASHITA, T., TOMITA, T. and ISHIHARA, D. Performance of Inward Radial Flow Turbine under Unsteady Flow Conditions - Influence of Pulse Frequency. IHI Engineering Review, Vol.7, No.1. January 1974.

10. BENSON, R.S. A Comprehensive Digital Computer Program to Simulate a Compression Ignition Engine including Intake and Exhaust Systems. S.A.E. Paper No.710173, January 1971.

10. APPENDIX 2

Engine and turbocharger data

Engine make and model:	Ruston and Hornsby, 6YEX, Mk.II.
Type:	4 stroke, compression-ignition, turbocharged.
Number of cylinders:	Six
Bore:	127 mm.
Stroke:	149 mm.
Compression ratio:	14.4:1
Speed range:	1000-1800 rpm
Maximum bmep:	10.3 bar.
Combustion chamber:	Open-type, direct injection.
Valve timing:	IVO 10° btdc. IVC 35° abdc. EVO 50° btdc. EVC 15° atdc.
Moment of inertia:	4.97726×10^{-1} kg.m.sec^2.
Turbocharger make and model:	Holset 4LEV-305.
Turbine:	Double entry, single stage radial.
Compressor:	Single stage, radial.
Moment of inertia:	4.0×10^{-5} kg.m.sec^2.

11. APPENDIX 3

Specification of computer and interface used for data acquisition - Digital Equipment Corporation (DEC) PDP15 Digital Computer

Word length	18 bit	
Core size	32K words	
Execution times:	18 bit ADD	1.6 μs
	18 bit MULTIPLY	7.0 μs
	18 bit DIVIDE	7.25 μs

Peripheral devices:

- 2 Disk: 256K words
- 4 Dectape handlers
- 1 Tektronix Visual Display Unit (VDU)
- 1 Console Teletype
- 1 Paper tape reader
- 1 Card reader
- 1 Output lineprinter.

Micro consultants interface

- 1 Multiplexed analogue-digital converter (ADC)
  - 32 ADC channels
  - 15 bit accuracy
  - 40 kHz conversion rate
- 2 Digital-Analogue converter channels (DAC)
  - 8 bit accuracy.

12. APPENDIX 4

Engine test-bed instrumentation

Crankangle Moore-Reed optical encoder coupled to crankshaft.

Cylinder pressures Two Vibrometer 8QP.500.CA. Quartz Piezoelectric transducers.

Fuel injection pressure Intersonde VT series strain gauge transducer fitted into injector body and exposed to bottom gallery.

Needle lift Inductive coil operated by injector valve spindle extension.

Inlet manifold pressure Bell and Howell 366 series. strain gauge transducer with temperature compensation.

Air mass flow rate 2½" diameter Venturi nozzle. Throat depression measured by SE Labs. SE1150 variable reluctance differential transducer.

Temperatures Pyrotenax Nickel Chrome/Nickel Aluminium thermocouples. Thermocouple with outer sheath removed used for fast response exhaust temperature measurement.

Engine torque Two independent means of measuring engine torque were used.

(a) British Hovercraft strain gauge transducer TT2.4.EA. coupled between engine and dynamometer.
(b) BLH C3P1 strain gauge load cell fitted to dynamometer.

Engine speed Two independent means of measuring engine speed were also available.

(a) Magnetic pick-up and 60 tooth wheel on dynamometer.
(b) Processing encoder crankangle signal.

Turbocharger speed Induction coil sensing movement of magnetised nut fitted to turbocharger shaft.

Smoke measurement Celesco model 103 'in-line' opacity meter fitted in exhaust pipe. Response time approximately 20 ms.

Filtering All gross signals containing cyclic variations were filtered to give a representative mean value. The cut-off frequency was chosen to exclude cyclic effects without adversely reducing the instrument response to transient phenomena.

13. APPENDIX 5

Table of transient tests performed

No.	Engine Set Speed (rpm)	Maximum App. Load (bars)	Dynamometer Time Constant (ms)
1	1,008	8.00	500.0
2	1,000	8.67	300.0
3	1,000	9.44	500.0
4	1,004	10.00	300.0
5	1,200	8.00	300.0
6	1,200	10.00	350.0
7	1,200	10.00	400.0

cont'd...

Table of transient tests performed (continued)

No.	Engine Set Speed (rpm)	Maximum App. Load (bars)	Dynamometer Time Constant (ms)
8	1,395	8.00	350.0
9	1,400	10.00	300.0
10	1,600	5.00	500.0
11	1,600	7.00	350.0
12	1,600	8.07	300.0
13	1,600	9.51	350.0
14	1,600	10.06	300.0
15	1,600	11.00	300.0
16	1,800	5.00	500.0
17	1,800	7.00	300.0
18	1,800	8.00	300.0
19	1,800	10.00	300.0

• Nominal No. Load (bars) 1.0.

This paper is presented for written discussion. The MS was received on 24th February 1975 and accepted for publication on 18th December 1975. 23

Communications are invited for publication in the Proceedings. Contributors should read the instructions an page ii of cover.

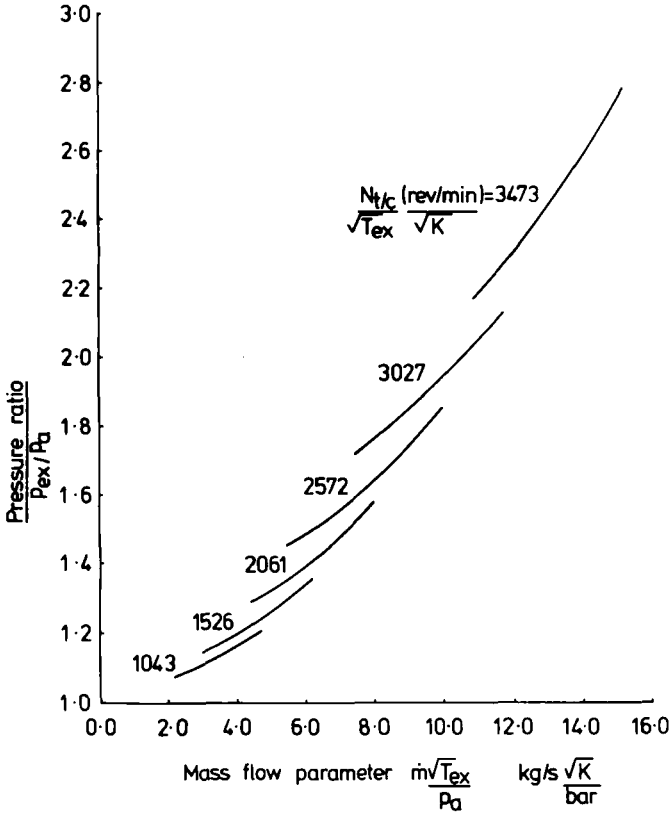

Fig. 2: Turbine charateristics

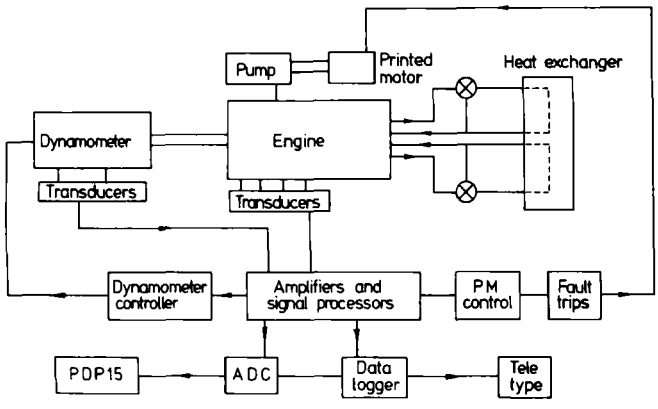

Fig. 1: Schematic of complete system

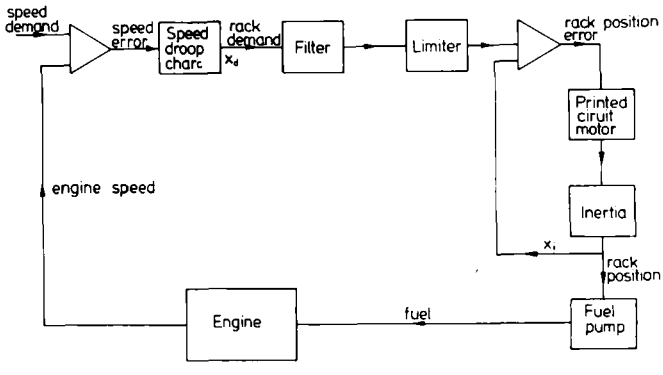

Fig. 3: Rack control system

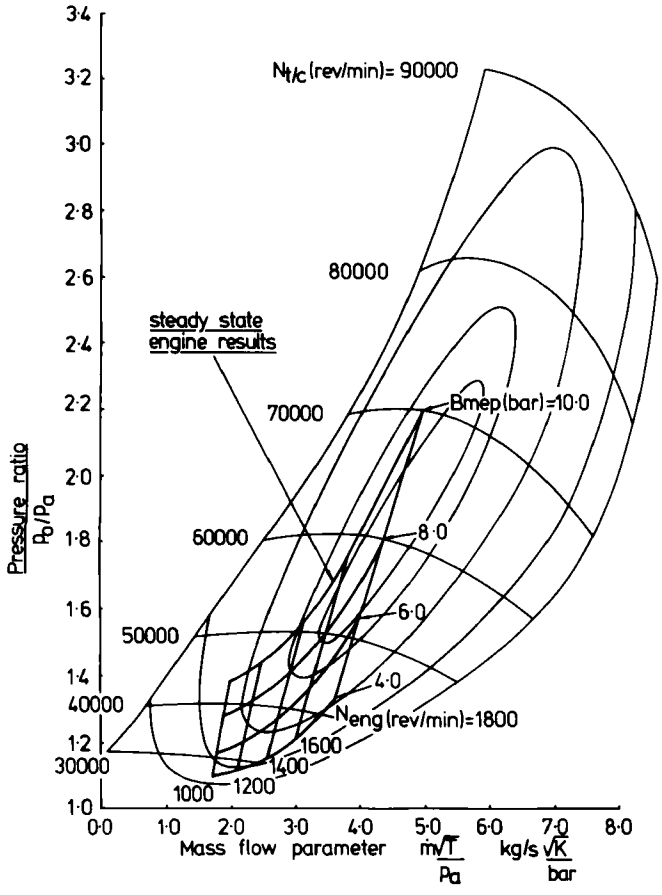

Fig. 4: Compressor characteristics

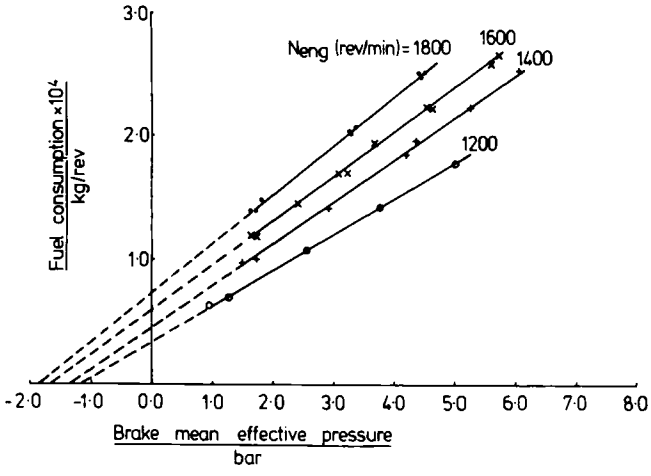

Fig. 5: Willan's lines

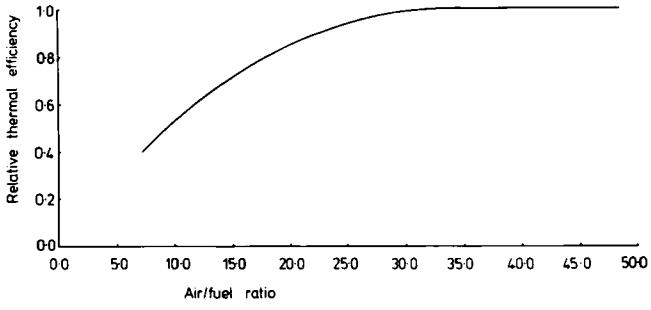

Fig. 7: Relative thermal efficiency

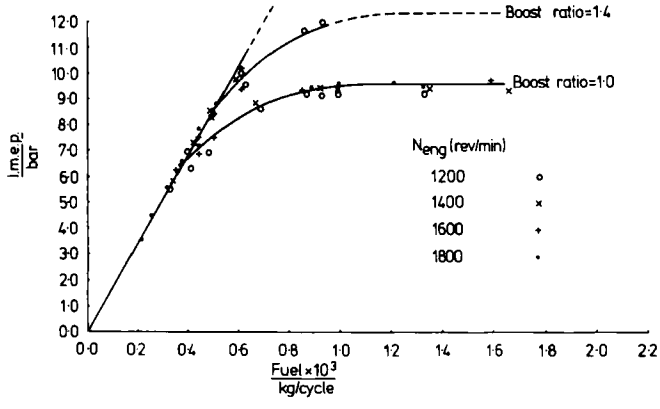

Fig. 6: Engine imep

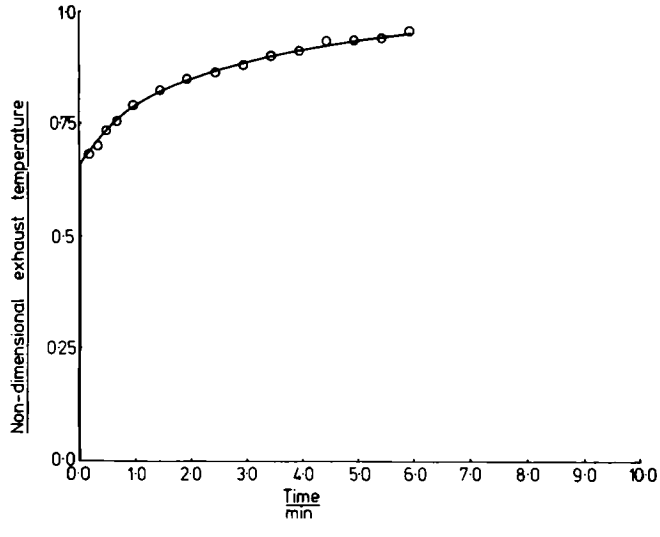

Fig. 8: Exhaust temperature rise

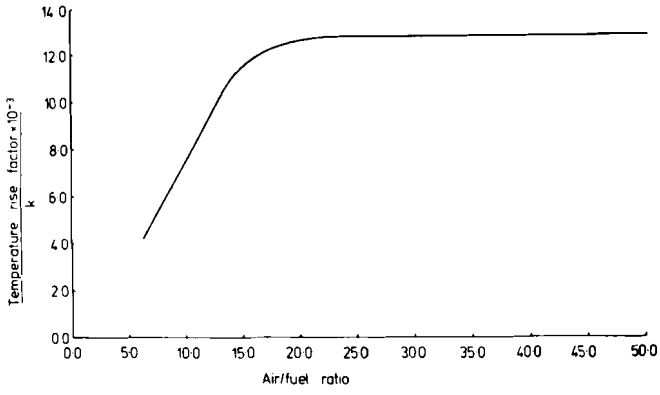

Fig. 9: Engine temperature rise factor

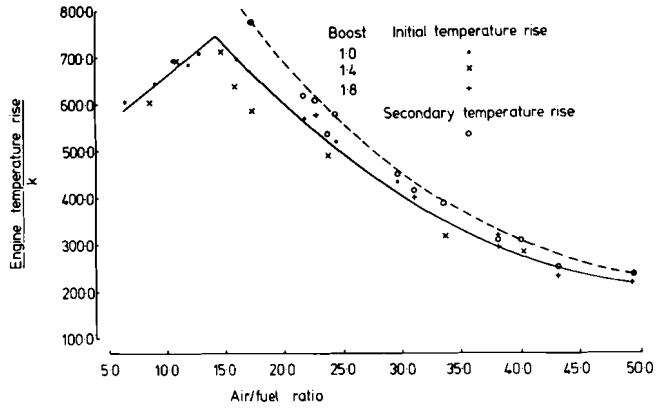

Fig. 10: Engine temperature rise

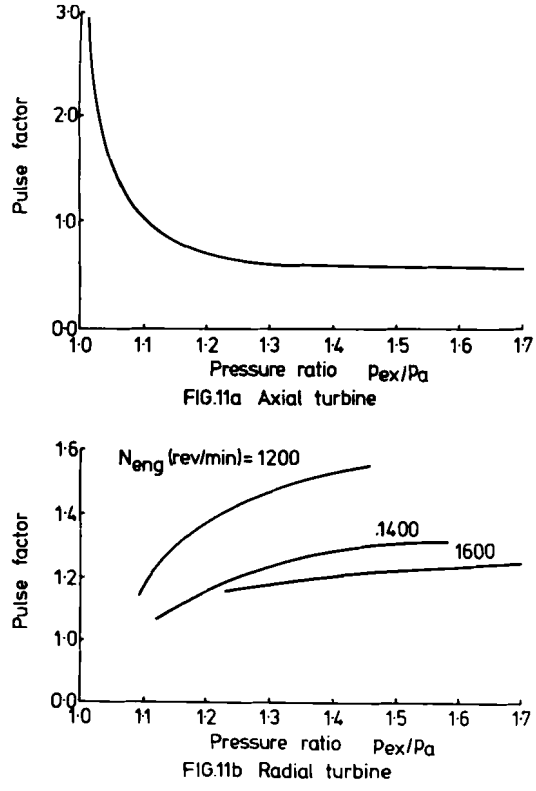

Fig. 11: Pulse factor for axial and radial turbines

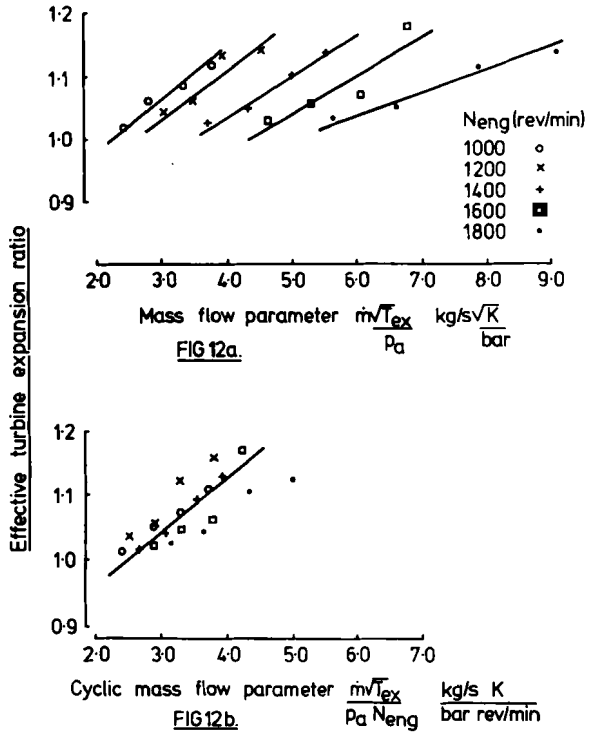

Fig. 12: Variation of turbine expansion ratio with engine operating coditions

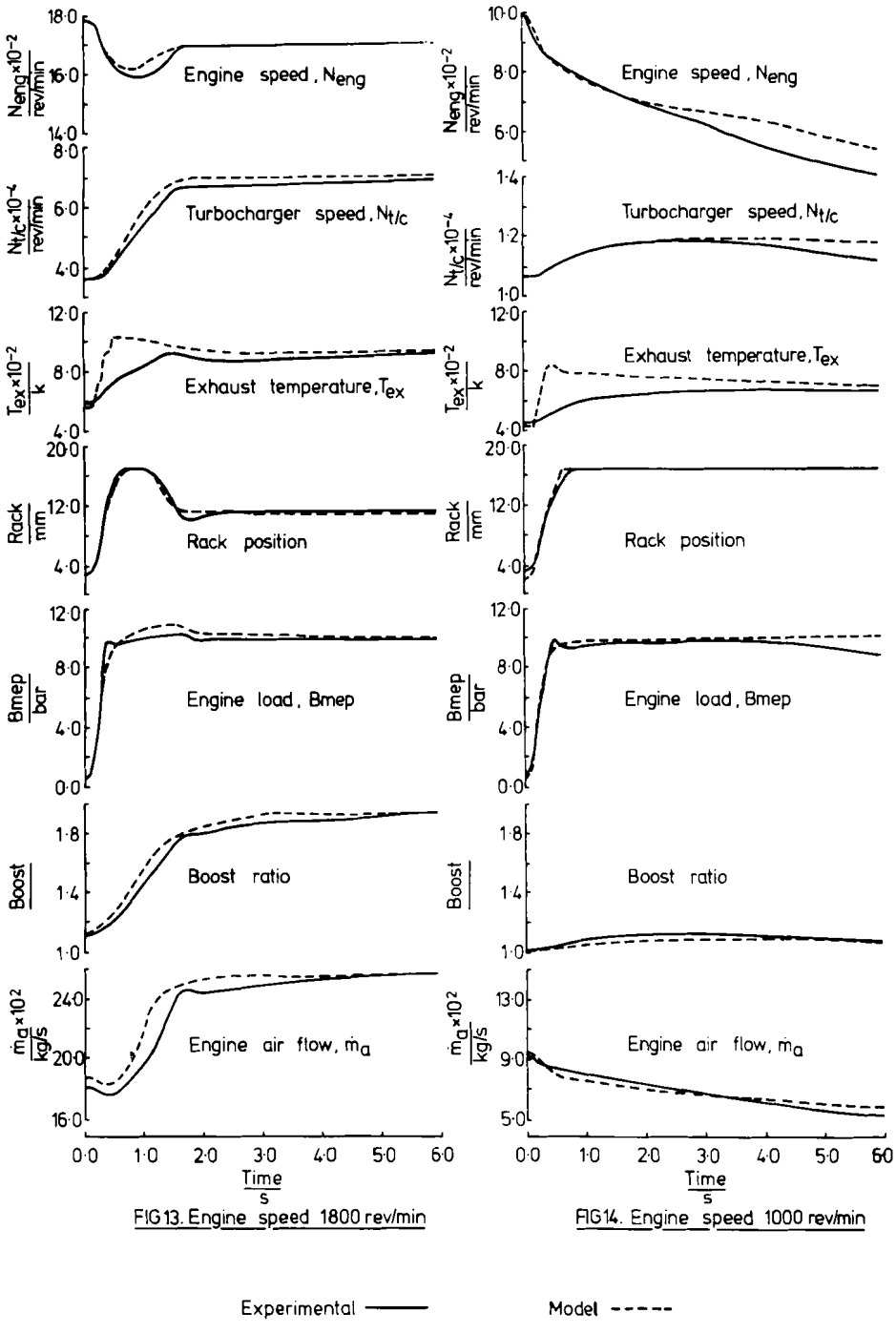

FIG 13. Engine speed 1800 rev/min

FIG 14. Engine speed 1000 rev/min

Figs. 13 & 14: Engine transient response for load application 1-10 bar bmep

Discussion

F. S. Bhinder Fellow

Interest in transient response of engines has grown rapidly during the last decade. However, because of the high cost of the installations only a few laboratories are equipped to do this work, consequently test results are scarce. This paper, therefore, provides useful and, perhaps, much needed data. However, I have a number of points on which further comments by the authors would be much appreciated.

On the question of Willan's lines, the forces on various parts of an engine would change significantly due to turbocharging. Therefore the friction bmep in the turbocharged mode would be different from that in the naturally aspirated mode. Why did the authors obtain these measurements in the naturally aspirated mode? Secondly it is not clear whether the cylinder head used for this test was for the naturally aspirated or the turbocharged version. The matching of the fuel injection equipment and combustion characteristics would, most certainly, be different for the two cases.

The authors read in Fig. 6, 'things' which are not shown, but with which they are of course familiar. For instance it is very difficult to deduce air/fuel ratio from the fuel consumed per cycle. The mass flow of air per cycle cannot be found from the compressor mass flow parameter, Fig. 4, because P_1 and T_1 are not given.

Engine temperature rise depends primarily on the air/fuel ratio and the type of combustion chamber employed, but the valve timing and overlap, and the pressure difference between the inlet and the exhaust manifolds are also known to affect ΔT. Therefore it is difficult to give an universal formula relating ΔT with air/fuel ratio. It would be helpful if the authors will give some idea of the expected ranges of their constants K, K_D and a used in the equation for ΔT.

The pulse factor, Fig. 11, for the axial turbine increases sharply below an expansion ratio of 1·15:1, whereas for the radial turbine it decreases with decreasing expansion ratio for all engine speeds. Some explanation of this phenomenon would be appreciated.

According to the text Fig. 12 is supposed to give the variation of K with the mass flow parameter, but from the caption it is not clear whether the y-axis shows K (P_{ex}/P_o) or simply (P_{ex}/P_o). Also there is no explanation for the difference between (a) and (b) of Fig. 12.

I would like to know how the continuous variation of turbocharger speed shown in Figs. 13 and 14 was recorded. Usually this speed is measured by means of a magnetic pick up which counts the number of pulses per revolution provided by either the blades or flats machined on the special nose nut fitted on the compressor end. Therefore the speed is measured over discrete time intervals during which the gate in the frequency meter remains open. The length of the interval is limited by the capabilities of the frequency counter. Hence the recording of continuously varying speed by this method is not technically possible.

It would also be helpful to know how and where the air mass flow rate was measured. Would it be safe to assume that the air mass flow rate measured by means of a nozzle or an orifice upstream of the compressor and the actual mass flow rate into the cylinder would be the same during such short time interval as to give continuous traces similar to those shown in Figs. 13 and 14.

The following data have been extracted from Figs. 4, 13 and 14:

Engine speed (rev/min)	bmep (bar)	turbocharger speed (rev/min)	
		from Fig. 4	from Figs. 13 and 14
1 000	1	*	10 300
	10	43 000	12 000
1 800	1	22 000**	36 000
	10	70 000	70 000

*The point lies outside the graph
**extrapolated

I am a little concerned about the differences in the final speeds attained by the turbocharger at the end of the transients and its speeds at the corresponding steady state conditions. For example at low engine speed and full load condition the final speed of the turbocharger at the end of the transient is only 12 000 rev/min but for the same condition the steady state data, Fig. 4, shows the speed to be approximately 43 000 rev/min. Similarly at low load and 1800 rev/min engine speed the difference is nearly 14 000 rev/min. These differences are too large to be ignored and deserve some explanation. In my view the measurement of turbocharger speed under transient conditions is perhaps unreliable.

It is interesting to note that at both conditions the turbocharger attains the final speed in almost exactly the same time ($\approx$ 1·5 s). The values for the resulting acceleration are 118·68 rad/s^2 and 2373·65 rad/s^2. With these accelerations, it does not seem just to blame the turbocharger for a slow response of the aggregate to step changes in the applied load.

N. Watson Member, **M. Marzouk** London

At Imperial College we have also been engaged in a programme of work on the transient response of turbocharged diesel engines. In order to obtain greater insight into the complex processes involved, we used a computer controlled test-bed in conjunction with a more advanced data-logging system than that described in the paper. This enabled us, for example, to record cylinder pressure diagrams during a transient run. The system is described in (11). We have also

developed a transient response model (12) based on quite different ideas.

In concept, the model described in the paper is quite simple, but it may have important limitations. Consider first the generality of the model. It contains a large number of empirical functions and constants, many obtained from tests at off-design conditions with simulated turbocharging. To avoid repeating these tests on various engines, these values must truly be constant or the usefulness of the model is reduced. Comparison of Fig. 6 with Fig. 7 of reference (3) shows that different types of engines may have imep curves with wholly different characteristics. In addition, the scavenge flow parameter and relative thermal efficiencies seem markedly different.

We have attempted to overcome this difficulty by developing a model that follows the cyclic nature of the process occurring in the engine (12). Essentially this is an extension of the well known 'filling and emptying' concept to the transient situation. The advantage of this approach is that since it models actual processes more realistically, the requirement for numerous empirical constants obtained from off-design tests is eliminated. This, in turn, widens its applicability to different types of engines. We would emphasize that where semi-empirical formulae have been used in the model (for example, heat transfer from the cylinder) values of 'constants' have been taken from the literature without modification. Fig. 15 compares predicted response to a change in speed demand and load with that measured.

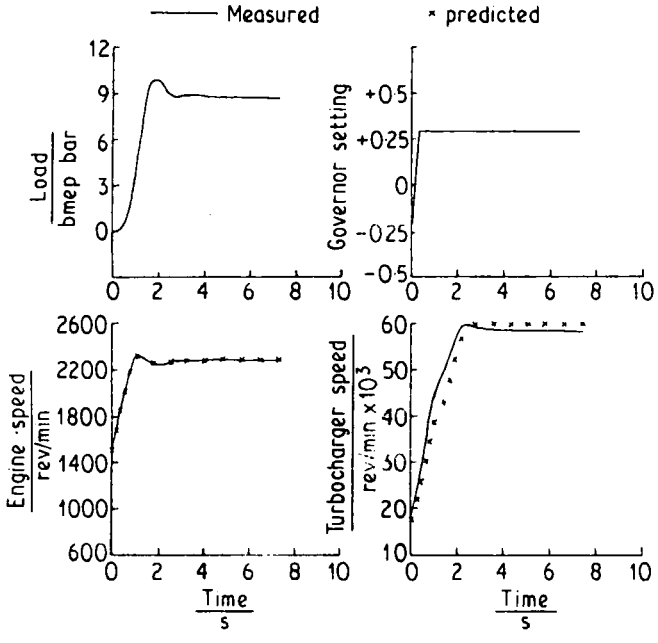

Fig. 15. Comparison of predicted and measured engine response to a speed demand and load increase.

Derivation of the correct turbine inlet temperature seems to be a problem inherent in the simpler model used by the authors. Are we correct in believing that transient tests must be conducted to derive K? Although not required for our own model, we have monitored turbine inlet temperature, with rather different results to those in the paper. Our data, Fig. 16, which was obtained on a similar

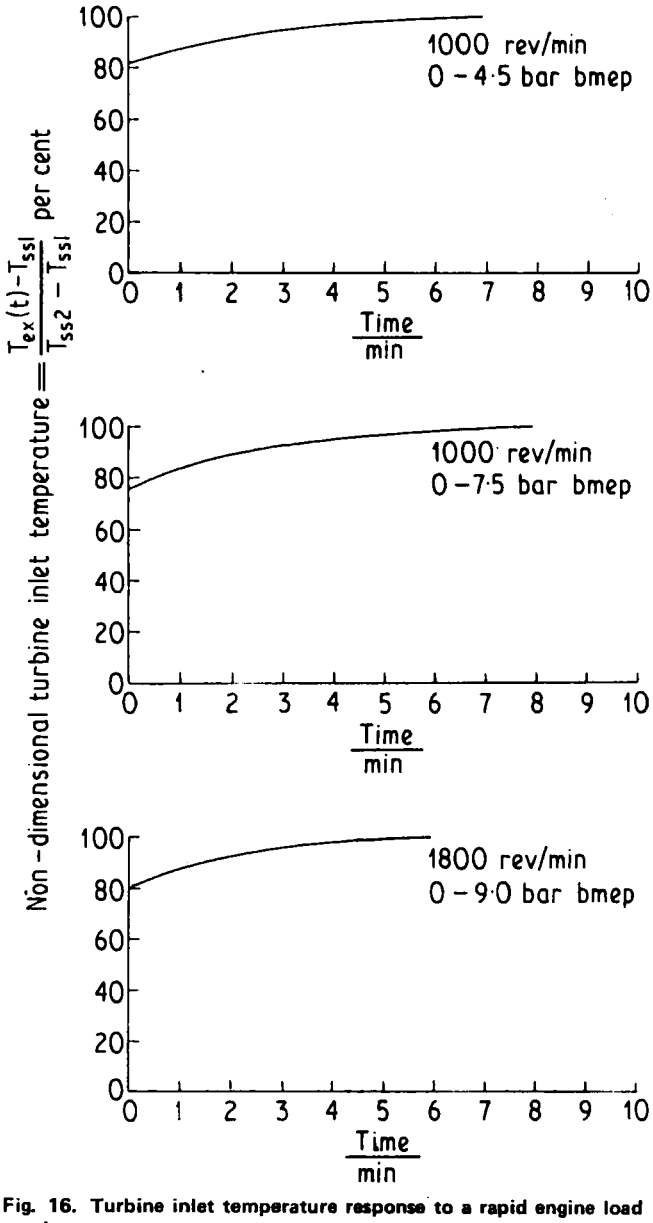

Fig. 16. Turbine inlet temperature response to a rapid engine load change.

size of engine, suggests that about 80 per cent of the temperature rise takes place in the first few seconds, but that the exact figure varies with speed and load.

We would have expected the lag in turbine inlet temperature rise to have been caused partly by a reduction in heat transfer from the cylinder as the walls heat up. We have looked at the effect of this by artificially altering the cylinder-wall thermal capacitance, such that the wall-temperature response time is of the same order of magnitude as engine speed. Fig. 17 shows the predicted effect of this change on turbocharger speed and hence boost pressure. Engine speed response is unaffected since the engine is speed governed.

We believe that the authors may have been fortunate in not being able to operate their test-bed from a load of less than 1 bar bmep. At no-load, the pressure ratio of the compressor may be less than unity, hence the usual definition of efficiency becomes meaningless. We have had to

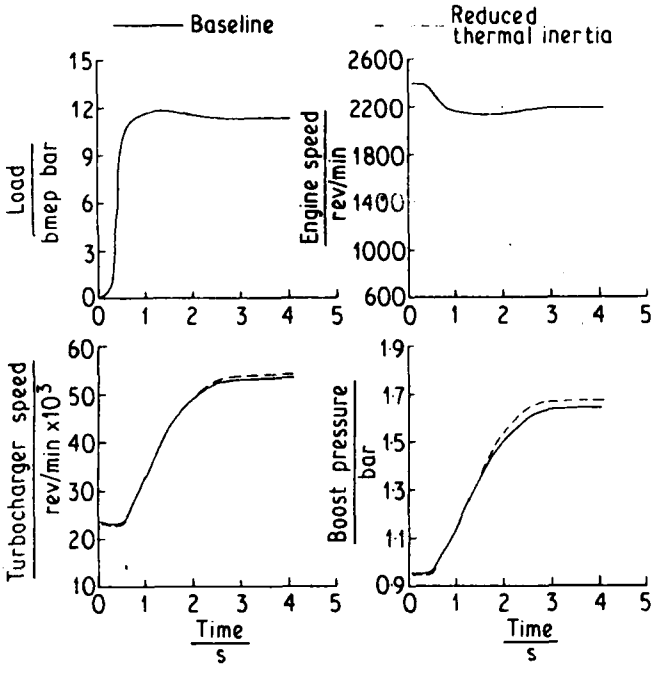

Fig. 17. Effect of cylinder wall thermal inertia on engine response.

re-define compressor work under these circumstances and wonder if the authors have had similar difficulties.

REFERENCES

(11) MARZOUK, M. and WATSON, N. 'Some problems in diesel engine research with special reference to computer control and data acquisition'. *Proc. Instn. Mech. Engrs*, **190**, 137–151.

(12) WATSON, N. and MARZOUK, M. 'A non-linear digital simulation of turbocharged diesel engines under transient conditions'. SAE Paper No. 770123, 1977.

A. E. W. Austen Fellow

I assume that the authors' only purpose may have been to put on record their *technique* for the study of transient performance. If so, and for this reason they regard concern with their results to be outside their 'terms of reference' I hope they will accept my apologies for having found the results of even greater potential interest than the technique.

The authors would appear to have chosen rather unrealistic conditions for their transient tests if they are concerned with marine and generator applications. With a generator it is reasonable to suppose that full torque can be applied suddenly from near-zero torque at maximum speed. But surely the generator would represent a significant addition of rotational inertia, which was apparently omitted in their tests. In the marine case they cite, moving quickly from full ahead to full astern, would present very different conditions to the engine from steady running at low torque to suddenly-applied full torque used by the authors.

In either marine or generator use it is not easy to see how the switch they use from low to full torque at a little over half speed could be effected; it is still more difficult to see why anyone should want to do it.

In most turbocharger applications, the critical factor is the amount of air the turbocharger provides; there is no difficulty about supplying plenty of fuel. It is therefore to be regretted that the authors present their results (for example Figs 5, 6 and 7) only in terms of how much of the fuel is used; air available and fraction burnt would be more informative. And it would have been useful if the authors had included in Fig. 4 the 1 bar bmep from which they started their transient tests. It seems probable that a useful approximate answer to the question asked of whether or not the engine will stall could be obtained from air available at initial turbocharger speed, maximum air that can be burnt, hence imep and bmep. If the bmep is equal to or exceeds that corresponding to the applied torque the engine may be expected to continue to run.

The authors must have the information necessary to try this simple hypothesis. They might care to supply it in replying to the discussion and to try out this suggestion. They could no doubt suggest some other simple but less crude approximation to estimate the speed at which the engine will just tolerate their load switch.

D. E. Winterbone Member, R. S. Benson Fellow, G. D. Closs, A. G. Mortimer (Authors)

We should like to thank the contributors for their searching and illuminating questions. These have highlighted some deficiencies in the original presentation, for which we must apologize, and also given us some guidance for future work.

We thank Dr. Bhinder for his comments and also for the attention which he paid to the detail of the paper. He queried the use of Willan's Lines for the measurement of friction mean effective pressure, especially under turbocharged conditions. This is a valid criticism and highlights a very real difficulty in the measurement of engine performance. The real nub of the question is what effect does load have on fmep. We are unable to answer this categorically at present but would refer Dr. Bhinder to the paper by Chen and Flyn (**13**) which relates fmep to peak cylinder pressure and speed. This suggests that over the operating range of this engine the effect of load is much smaller than that of speed. Our future work does include a project to investigate the variation of combustion efficiency and fmep under transient conditions so we may be able to answer this question then.

Referring to Fig. 6 we would like to make it clear that these results were obtained using shop-air to boost the engine, thus removing turbocharger effects. Fuel/cycle is then a reasonable parameter because the air mass flow is directly a function of boost. The reference conditions for Fig. 4 are 1·013 bar and 293 K.

As Dr. Bhinder concedes, engine temperature rise is primarily a function of air-fuel ratio, particularly at medium and low-load. As the load increases the other salient parameters become valve overlap and combustion chamber design; it is these secondary parameters which make it impossible to obtain a universal relationship between temperature rise and air-fuel ratio. The function given in the paper is specific to the Ruston 6YEX and could not be used for any other engine without some modification. The value of K is given in Fig. 9, and for complete combustion, is approximately 12 000. This was obtained from steady state tests. K_D can be abstracted from Fig. 10 and is itself a function of air-fuel ratio. It is the shortfall between the initial transient temperature and the corresponding steady state value; this has to be measured during short-term over-fuelling tests. K_D varies from about 25°C at A/F = 50 to 100°C at A/F = 20. The value of a is given in Fig. 8 and is approximately 140 s; this is the time constant of the exhaust system.

It is interesting to compare our experiences in this region with those of Imperial College. In Fig. 16 Drs. Watson and Marzouk present diagrams which are very similar to our Fig. 8. These indicate that a large proportion of the temperature change during a transient occurs almost instantaneously, but then there is a long tail of increasing temperature over about five minutes before steady state is reached – this vindicates the practice of waiting long periods for conditions to steady. The size of the instantaneous rise will be a function of the thermal capacity of the relevant components. We suspect that the Imperial College engine is of a lighter construction than the Ruston 6YEX. We have tried to simulate such an effect by considering the capacitance of the manifold but could not reproduce such a large effect.

The values of pulse factor shown in Fig. 11 were obtained empirically. It was found by Whitehouse (4) that for single stage axial turbines the effect of pressure pulses on turbine power at low exhaust pressures was very much more marked than at higher pressures. This is really an indication of the useful ranges of pulse versus constant pressure turbocharging. A similar approach applied to this engine and turbocharger produced the results shown in Fig. 11b. The radial turbine did not seem to benefit from pressure pulses at low exhaust back pressures; there is insufficient data to suggest why this might be, but it could be due to the large volumes in the critical regions of the turbine or the reaction nature of the machine.

Fig. 12 shows K alone as a function of mass flow parameter. Figs. 12a and b are the same data plotted in terms of mass flow parameter and cyclic mass flow parameter respectively. The use of the latter function reduces the complexity of the graph.

Turbocharger speed was measured using a 'diode pump' circuit which essentially converts the digital pulses into an analogue voltage signal. It will measure speed continuously and is quite capable of tracking turbocharger acceleration. Frequency to voltage converters are now readily obtainable commercially. The measurement of air mass flow rate was by a nozzle and it is unlikely that this device would accurately measure air mass flow under pulsating or acceleration conditions. These measurements should be treated more qualitatively than others.

Dr. Bhinder has pinpointed an error in Fig. 14. The turbocharger scale is incorrect and should go from 15 000 rev/min at the bottom end to 35 000 rev/min at the top end. This gives a turbocharger speed of 20 000 rev/min at 1000 rev/min, 1 bar. The poor agreement between the final value of turbocharger speed at 10 bar bmep in Fig. 14 and that obtained from the steady state characteristics is that the engine has stalled in the first instance and does not achieve steady state. Referring now to the 1800 rev/min results, there is poor agreement between the extrapolated value of steady state turbocharger speed at 1 bar bmep and the initial value in Fig. 13. Repeatability of low load readings has always been a problem area especially when plotting on a compressor map; if mass flow parameter and pressure ratio are plotted an error always exists in the implied turbocharger speed. However, the high load values tend to be in good agreement as demonstrated by Dr. Bhinder.

Unfortunately Fig. 4 was obtained very early in the project when commissioning the rig and the 1 bar results were not recorded. These might have made certain regions of the paper more easy to follow.

The final point raised by Dr. Bhinder is that the turbocharger response is not particularly slow, especially when compared to other dynamics of the system, e.g. engine speed. This is true, but in reality turbocharger lag must be measured relative to the rate of load application; it is here that the turbocharger deficiency manifests itself.

As Mr. Austen says, the results are interesting but the bias of the paper was towards the experimental and data processing aspects of the work. Similar results have been discussed more fully in (1, 2, 3). We consider that a step test is a good criterion for transient response because of its repeatability, although it may not be an exact simulation of the conditions existing in a commercial application. We feel that our tests are a realistic representation of the generator case, even though the inertia of the system is rather low. This engine is quite lowly rated (10 bar bmep) and the neglect of the inertia of the generator gives a similar effect to a higher rating. In the marine situation the ahead–astern manoeuvre places very severe loading on the engine as demonstrated by Gillespie (14); the step test is not an ideal method for assessing the ability of an engine to survive in such an application but it does provide quite a large amount of useful data.

The different engine speeds were chosen to show how dependent the load acceptance is on speed. This is to be expected since the stored energy is a function of speed squared.

Fig. 7 effectively shows the fraction of fuel burnt as the relative thermal efficiency. A value of 1·0 indicates that all of the fuel is burned, this occurring at above 30:1 air-fuel ratio. Below that mixture strength incomplete combustion occurs. This might, in part, answer Mr. Austen's queries on available air and fraction burned.

Mr. Austen's hypothesis is quite a valid means of assessing whether an engine has any chance of accepting a particular load. The shortcoming of the technique is that it does not give any information on transient droop and recovery time – both important parameters in governing.

Drs. Watson and Marzouk of Imperial College have followed a similar line of research to ourselves. We are well aware of their test-bed (11) and have recently obtained a copy of their latest paper on modelling. We accept that quasi-steady models are highly empirical and that it is not possible to obtain truly universal constants for such models. However, the data for a quasi-steady model could be calculated on a more comprehensive model for example the UMIST engine simulation programme (10). Quasi-steady models do have some applications where computer time is limited. Such models can achieve real-time running on a reasonably modest digital computer (DEC PDP-15) and can be incorporated into larger system simulations e.g. truck simulations (15).

We agree with the comments concerning the greater power of wholly dynamic models based on the filling and emptying technique and have developed just such a model at UMIST (16). It is gratifying to see that we recognized the general need at the same time. The results shown by the contributors in Fig. 15 show a change in governor speed demand; this is an easier case to simulate than the typical regulator case because the fuel pump spends most of its time on the end stop.

Surely Drs. Watson and Marzouk mean that there is a reduction in *energy* transfer from the cylinder as the walls warm up, not *heat* transfer.

We were relatively 'lucky' that we could not achieve true no-load running. This was due to the incipient drag on the dynamometer and was a compromise between no-load and speed of response. We did not encounter problems with compressor efficiency but would be interested to hear how such problems might be overcome. Imperial College were also fortunate with their dynamometer, which seems unable to apply full load in less than 2 s. It would be interesting to assess the Imperial College simulation under the rapid load changes affected here (viz ¼ s to full-load).

Work done since the publication of this paper has enabled the simultaneous collection of both gross and in-cylinder transient parameters. This feature will enable us to investigate the variation of in-cyclinder conditions during transients and forms the basis of a current two year project.

REFERENCES

(13) CHEN, S. K. and FLYN, T. Development of a compression ignition research engine. SAE Paper No. 650733. 1965.

(14) GILLESPIE, D. A. and HUGHES, V. R. 'Geared propulsion systems using engines of 745–3730 kW'. Instn. Mar. Engrs. 1973.

(15) BENSON, R. S., MORTIMER, A. G., STOTTER, A. and WINTERBONE, D. E. 'Transient response of turbocharged diesel engines: Pt. 2. application to a heavy truck; a study of dynamic performance of the engine'. SAE Paper No. 770122.

(16) WINTERBONE, D. E., THIRUAROORAN, C. and WELLSTEAD, P. E. 'A wholly dynamic model of a turbocharged diesel engine for transfer function evaluation'. SAE Paper No. 770124.