

**OPTIMAL CONTROL OF ENGINE TEST-BEDS
BY MICROCOMPUTER NETWORKS**

By

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ERRATA

Page	Line top bottom	Original	Correction
10	17	introducing	introducing
31	10	imlementation	implementation
31	13	patters	patterns
35	7	near	new
36	6	correction	corruption
38	11	provide reliable	provide reliable data transfers in the case of a link failure but rather to ensure reliable
47	2	characterisitcs	characteristics
50	5	selective	selected
50	11	maxium	maximum
53	Equations (1), (3)	d	∂
54	Equations (8), (11)	d	∂
55	1	dependant	dependent
59	3	(35)	(25)
61	14	current	instantaneous
76	3	(56)	(55)
90	Equation (82)	$1 + x_2 p_1(t) - \beta x_2 p_2(t)$	$1 + x_2 p_1(t) - \beta x_2 p_2(t) + u(t) p_2(t)$
90	Equation (87)	$\frac{M_1}{\beta}$	$\frac{M_1}{\beta}$
101	11	by	be
101	9	every	every S
110	10	inertia	stator inertia
130	Fig. 7.13	1500 RPM	2000 RPM
137	Fig. 7.22	10 mkp	70 mkp
124 and 125		time-optimal own excitation	-----
		analog PID, CEB 105	-----
		software PID, BEC 12/45	— . — . — . —
168	1	DAC	ADC

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Abstract

The complexity of modern transient engine test procedures and the required operational flexibility of engine test-beds are investigated. A cost efficient solution for test-bed control is presented by way of microcomputer networks. Distributed processing potential allows for at the same time sophistication in the data-logging procedures and software control of the test-bed components.

A model for the transient response of eddy-current dynamometers has been presented and an optimal controller implemented for it.

Time optimal analysis has also been applied to software stepping motor control.

A new protocol has been developed to provide communication within the microcomputer network. The Data-logging subsystem has also been optimized for speed in data transfers.

Tests carried out on two different engines and test-beds show good repeatability and flexibility in the testing procedures.

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CHAPTER 1

Introduction

1.1 Introduction to Automated Test Systems

The need for more effective test methods has long been accepted among researchers and has spread to many other fields of manufacture and service. The requirements of the modern world for more efficient machines, improvements in performance levels and the simplification of operating procedures have resulted in the design and development of more complex equipment.

It is this complexity that increases the importance of testing throughout the many engineering disciplines and at all levels of R and D production and operation of equipment. Testing technology in its widest sense, of not only measuring and presenting test data but also of comparing, interpreting and applying such data, presents a continuing challenge to the ingenuity of all concerned.

Any attempt to categorize test facilities or techniques by individual test objectives (e.g. production testing, development testing, maintenance testing) is bound to be complicated by the duplication of test processes in many of these roles. A more effective method of classification is to consider the nature and origin of test equipment, its application to the item that is to be tested and its mode of operation. The development of test facilities must depend on the characteristics of the item to be tested. Such items vary in size, operational characteristics and complexity (e.g. a two stage turbocharged diesel engine and a naturally aspirated petrol engine). A term, now generally accepted, describing all such items is "unit(s) under test" (UUT).

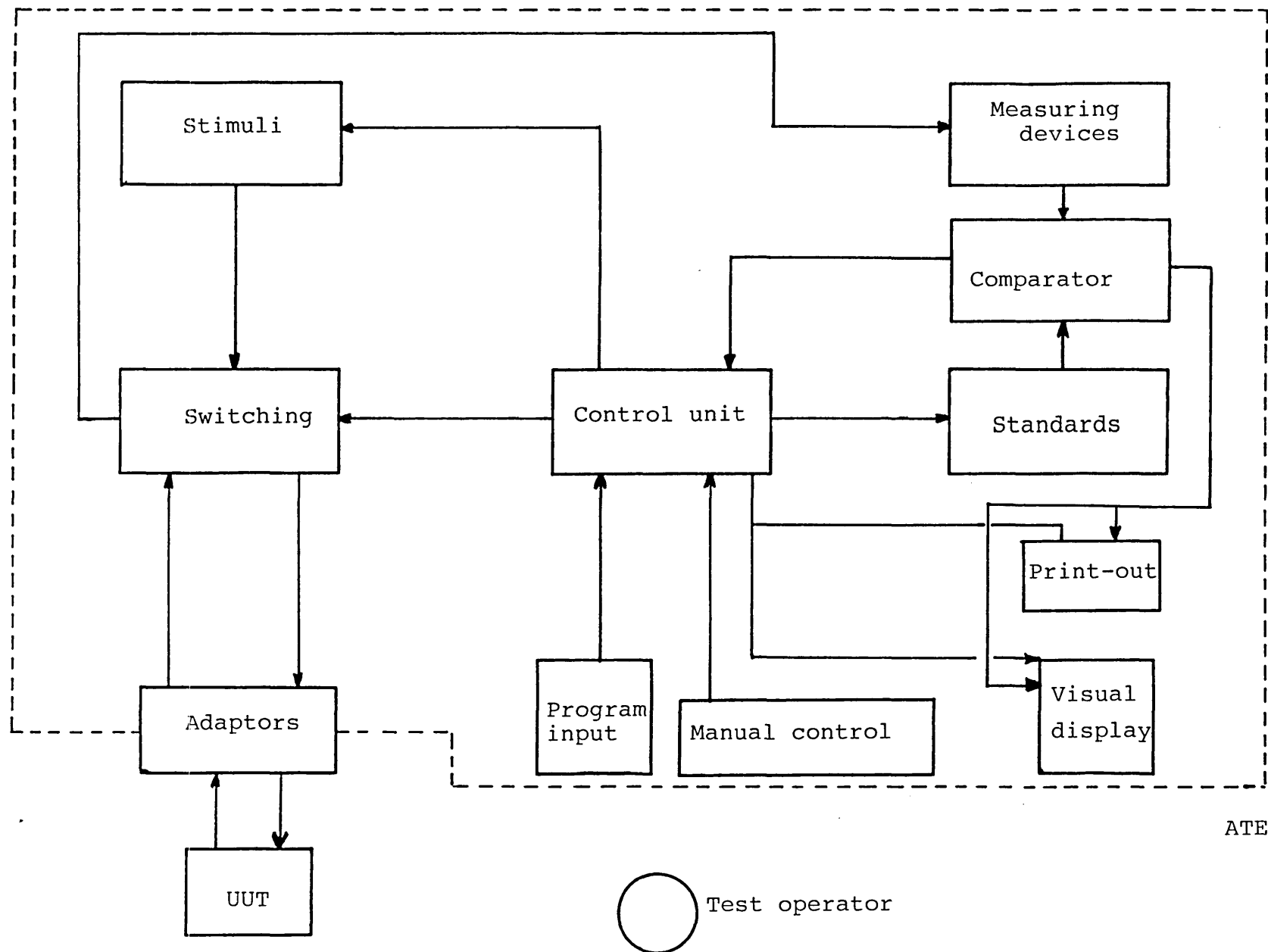


Fig 1.1 Schematic diagram of typical automatic test system

The design phase of ATE (automated test equipment) involves primarily the definition of all candidate UUT, and modes of operation. This process will help design the least general purpose system, and hence the most economical, that will conform with the set specification.

1.2 The Reasons for Automatic Testing

The concept of applying automatic control to variable programmed test procedures dates from soon after the end of the Second World War. Wartime developments in, for example, aerial navigation, radar, computer controlled gun laying, automatic bombsights and stabilised control systems had created many new test problems [A-6].

Initially the pressures for introducing ATE were primarily concerned with the speed of testing in conjunction with an exploding number of test-points [K-5]. The second thrust towards ATE came from requirements on accuracy and repeatability. The accuracy required of a test depends on the type of test and the particular test function. Tests carried out on individual components or data-collection for research purposes are intended to maintain close tolerances and therefore test accuracy requirements are high. On the other hand the ultimate accuracy of any test system is limited by the quality of its measurement devices and by any uncompensated losses in the complete system (e.g. engine torque measurement subject to vibration). In this respect automatic systems are as dependent on the choice of measurement devices and the application of appropriate engineering standards as manual methods of testing. Manual methods are, however, prone to considerable human errors which automatic control should eliminate. Many test procedures are tedious, and this leads to a fall-off in the accuracy of human performance. Errors in dial settings, meter readings and interpretability of displays are only

some of the serious operator failings which frequently occur and make recorded results suspect.

It must be remembered, however, that measurement problems which arise in manual testing can be magnified in a complex automatic system. Working tolerances for measuring devices and sensors should be carefully selected to ensure repeatable measurements and data integrity.

Cycle testing has evolved as an important means of assessing new equipment characteristics and achieve a reliable foundation for comparison purposes. In order to achieve these objectives a high degree of repeatability is required and in most cases more than one controlled inputs to the UUT are involved. When the cycle involves fast transient phenomena it is impossible for the human operator to respond quickly enough not to mention the degree of his unrepeatability. The only solution to repeatability is automatic control of the actuating inputs through the ATE.

Overshadowing all of these reasons for introducing ATE has been, and still is, the vital need for economy in the use of technical manpower. There is an overall shortage of qualified personnel and of suitable candidates with the aptitude for training in the advanced technologies. Automatic testing cannot overcome this fundamental problem but it can alleviate its effects in particular areas. Automatic data collection, control of the test, and processing of the data can extend the range of personnel capable of supervising a test.

This can be beneficial in production and quality control environments or even universities where untrained students could use the ATE for their project UUT without getting involved with the ATE knowledge or development.

1.3 Trends in Engine Research

The seventies have been marked by two, often contradictory, tendencies in engine research objectives. The first one was the soaring price of oil and the requirement for more fuel-efficient internal combustion engines. The second one emerged from an increasing awareness, worldwide, about pollution and the subsequent legislation to limit it.

Historically the second objective was the one to be looked upon first, as mandatory regulations in the United States required vehicle manufacturers to meet limits on the hydrocarbon (HC), oxides of nitrogen (NO_x), and carbon monoxide (CO) emissions from motor vehicles. The 1970 US Clean Air Act required the automobile industry to produce vehicles in 1975 which emitted essentially 90% less pollutants than those of pre-1970 years. The 1973 oil crisis occurred in the middle of this development and less stringent standards were implemented, as a compromise to fuel consumption. Far reaching consequences were already on the way:

- 1) The timescale given to car companies to try and meet the legislation was very tight, which meant that very little attention was paid to the effect that the emission control technology being applied, was having on fuel consumption and driveability.
- 2) The diesel engine, was seriously considered as a commercially viable alternative to the automotive petrol engine.

It should be noted here that whilst the USA was undoubtedly the leader in the field of vehicle exhaust pollution control, other nations of the world soon began to be concerned because of urban smog problems and noticeable deterioration in urban air quality, so that today European nations, Canada, Japan and Australia have

introduced similar legislation [L-7].

While pollution legislation was becoming more and more stringent, in 1981 new legislation was enacted for fuel consumption reduction, which required a 'fleet' average of 25 MPG (miles per U.S. gallon) by 1985 for all cars sold in the U.S. by a specific manufacturer. Even without the introduction of the legislation, market research had shown that inefficient cars were becoming increasingly unpopular.

So far only some historical background on the legislation has been given without any reference to the actual solutions and requirements in engine research.

The projected limits of the 1970 US Clean Air Act were so low that automobile manufacturers had to resort to the chemical treatment in the exhaust systems using noble metal catalysts. As a consequence of this catalytic control route, unleaded petrol had to be made available as lead poisons the catalysts. The use of unleaded fuel resulted to lower compression ratio engines necessitating the use of 90-91 RON (Research Octane Number) petrol. This had a significant fuel penalty due to the poor engine efficiency at low compression ratios and high exhaust back pressure. If in turn it is decided to keep a minimum compression ratio and refine the petrol to 95-96 RON without the use of lead then a big refinery investment programme and increased petrol costs would be necessary.

Some research is still being conducted to develop lead tolerant catalysts [O-1]. The increase in RON will enable engine manufacturers to increase their compression ratio and improve fuel efficiency. But in view of the growing concern for health hazards from lead, it is doubtful whether leaded fuel will be tolerated much longer. The bulk of internal combustion engine research is conducted towards an improvement of the engine as a system, by improving

combustion characteristics, friction losses, specific power, engine control and gaining the benefits of increased efficiency.

It is to this particular direction that research is being carried out at the author's research establishment. The present work is confined to the requirements such research is imposing, and presents solutions within that context, which will be highlighted in the next chapters.

1.4 Automated Test-Beds in Engine Research and Development

In a research establishment investigations are carried out on a wide variety of engines and configurations. In addition the nature and purpose of the investigations can differ enormously, although a particular type of test may predominate for a time.

Jones [J-3] and Marzouk [M-5] have stressed the requirements on petrol and diesel engine research respectively and pointed out that high speed data logging and computer control of transient testing were prerequisites for extending knowledge into engine behaviour when simulating normal driving conditions. Research contractors shared the same view [N-1] and developed automated test-beds with good data logging capability and computer control of the entire test bed [R-7][P-2].

The need for high speed data logging comes mainly as a requirement for accurate determination of heat release rate which involves logging of the cylinder pressure diagram. Brown [B-12] states the requirements for accurate determination of heat release rate which involves logging of the cylinder pressure diagram. Brown [B-12] states the requirements for accuracy in these measurements with respect to distortion of imep. Pischinger [P-5] relates the accuracy of pressure diagrams and sampling rates with the distortion effected on heat release diagrams. He implemented a computer

controlled fast data logging facility dedicated to data acquisition, and demonstrated sampling effects on the rate of heat release calculation. Lancaster [L-5] underlines the importance of fast data logging of pressure and refers to the accuracy provided by piezoelectric pressure transducers and the combination of digital data acquisition to improve the utility of that data.

The widespread use of turbochargers in both diesel and more recently petrol engines resulted in a different transient behaviour from naturally aspirated engines, and therefore extensive investigations are required in order to establish the interaction between emission control, fuel consumption, and driveability. As Nightingale [N-1] points out, the key to the evaluation of transient conditions is repeatability and accuracy of the testing procedures. His suggestion for achieving these goals has been the computer control of the transient testing operation. Other researchers like Cichy [C-8], Akiyama [A-2], Diffenbach [D-4] shared the same view regarding the need to automate testing procedures by introducing the computer as a controller. The common approach has been, and still is, to use hardware local controllers for the torque or speed setting and the computer to feed these controllers with the desired output level of some parameter without intervening into the actual control procedure. This of course frees the computer from control considerations and confines it to overall test bed management. It will be shown in a later chapter that such an approach in testing is inflexible. It was therefore decided to use software control where possible at Imperial College, as a means of enhancing the flexibility and control procedures of engine test-beds. It is within this context that the present work has been conducted.

1.5 Software Control and Distributed Processing

As mentioned earlier the prevalent method of control of test-bed parameters (e.g. speed, torque, injection or ignition advance) has been by local hardware controllers receiving the desired level signals and effecting necessary corrections to the actuators. If the control task is shifted to the computer then this can involve excessive requirements in terms of computing power due to the real-time constraints of the controlled processes. The characteristics to be optimised within an R and D test bed are data logging speed, control quality and flexibility. All these requirements are conflicting with the limited computing power of modern minicomputers in that:

- a) Increased data logging speed increases computing power requirements.
- b) Good control quality requires small sampling periods therefore more control loops per unit of time.
- c) Flexibility in the software requires a certain degree of sophistication in the programming techniques and this is of course at the expense of the complexity of the executed program.

Until 1976 computing power was still expensive. The use of medium-sized computers for engine research was prohibitive, and therefore no one contemplated introducing software control, where a hardware three term controller was adequate. The advent of cheap computing power with the introduction of microprocessors and microcomputers imposed a reassessment of the situation. By building either a multiprocessor system or a local network one could have virtually unlimited computational power to perform even the most complicated control tasks. It was therefore decided to investigate

the software control of test bed equipment via optimal controller algorithms for the torque and governor setting. These optimal controllers would have the advantage of theoretically zero overshoot, minimum time to reach target and software controlled variable gain, required for the efficient and flexible operation of the test bed.

Communication between the microcomputers had to be resolved. Existing microcomputers at Imperial College were already configured to perform dedicated tasks like data-logging or control and a network software had to be designed along those requirements leaving room for future expansion. A review of network-like solutions to engine test bed automation is given in Chapter 2.

1.6 Eddy Current Dynamometers and Stepping Motors

The control task on an engine test-bed consists of presenting the engine with certain inputs, notably torque and governor settings (rack/throttle position and injection/ignition advance) and control engine speed. The actuators and devices historically used for that purpose are discussed below.

The first dynamometers to be used in engine testing were water brakes. Electrical braking methods were introduced next, first by means of Direct current motors/generators and later on through eddy-current brakes. Lately hydraulic braking systems have been introduced, using a hydraulic motor as the braking component. It is difficult to declare one method better than the others. There is always a field with its own operating characteristics that justifies one or the other type of dynamometer as the most suitable.

Concentrating on engine research requirements, with characteristics such as:

- a) good response in order to conduct transient testing

- b) smooth torque generation
- c) motoring capability for certain types of cycle testing
- d) simplicity of operation and maintenance
- e) low inertia
- f) low specific cost (£/KW)
- g) no regeneration requirement

It becomes extremely difficult to select any one system as most suitable.

Water dynamometers are cheap, rugged and reliable, offer smooth torque, but do not have motoring capability and are difficult to control during fast transients.

The DC machines have a good transient response, smooth torque, motoring capability but relatively high acquisition costs (sometimes justified when energy recovery can be included in the amortisation estimate [N-8]), limitation in speed, and high rotor inertia.

Eddy-current dynamometers have reasonable transient response, smooth torque, low inertia, relatively low cost, they are compact and reliable but do not have motoring capability (some manufacturers will fit a separate motoring facility on the same unit).

Hydraulic braking appears to have the best response among various braking methods (very few results have been published so far) [B-13]. It has low inertia, can be configured with motoring capability and regeneration equipment, but the cost is high, it requires a motoring facility even to achieve zero torque, offers a slightly fluctuating torque of a frequency $p \times$ (shaft speed) where p is the number of pistons in the hydraulic motor, and has speed limitation, therefore an intermediate gearbox is required in some cases. Hydraulic dynamometers are relatively new and possible shortcomings may appear in practice.

Water and eddy-current dynamometers remain most popular for engine testing. Eddy-current units are used extensively for transient work, even though they do not offer motoring capability [N-1].

The potential of considerable optimisation in the control of eddy current brakes, in addition to an already big investment in such brakes at Imperial College and elsewhere, led the author to study optimal control of eddy-current systems, and usage in test-bed automation.

Governor settings have accurate positional requirements. Early automated test-beds [B-6] used an electric d.c. motor with positional feedback to move the throttle. The use of an hydraulic pneumatic servo motor was also usual [N-1]. The advent of cheap and reliable permanent magnet stepping motors led to their adoption in test-beds at Imperial College.

A system with stepping motors offers at least the following advantages [K-10]:

- a) A step motor is inherently a discrete motion device. It is more compatible with modern digital control and is more easily adaptable for interfacing with other digital components.
- b) The positional error in a step motor is non-cummulative
- c) It is possible to achieve accurate position and speed control in an open-loop system, thus avoiding ordinary instability problems of positional servos. Feedback transducers can often be eliminated.
- d) Design procedure is simpler for a stepping motor control system.

There are also some disadvantages in their use:

- a) The choice of a stepping motor will always involve a compromise between speed and torque therefore high speed combined with high torque applications are ruled out.
- b) Being an electrical device it is susceptible to heat and moisture.
- c) In open-loop configurations the loss of a step will never be detected by the controller.

Stepping motors at Imperial College were controlled by a downcounter interface card. The computer loaded the downcounter with the required number of steps, and the hardware controlled the acceleration and deceleration rates. This mode of operation did not allow for computer control of the acceleration/deceleration characteristics and in addition once the interface was loaded with a specified number of steps, could not be cleared without a loss of position. This mode was geared to a specific type of test, step input on the governor setting [S-12]. Modern transient testing requires flexibility in actuator control. Firstly, stepping motors should be able to receive a new command even when in motion and optimize their positioning without loss of steps. Secondly, it should be easy to change acceleration/deceleration characteristics in order to emulate variable gain actuators. These requirements were to be incorporated in the test-bed designed by the author using microcomputer power to implement time-optimal software control.

1.7 Outline of the Present Work

The concurrent growth, during the past few years, of engine testing requirements and the computer industry, have led to a

reassessment of test-bed management and control techniques. The synthetic task of reviewing various technological achievements and incorporating them into engine testing has been undertaken by the author.

The behaviour of eddy-current brakes, used at Imperial College for engine test-beds, has been theoretically investigated and the time-optimal control derived and implemented.

The time-optimal control of stepping motors has also been studied and implemented for throttle actuation.

Software control requirements for the implementation of the optimal algorithms were resolved by resorting to distributed processing. A three microcomputer local network has been implemented and the communication software developed for it, catering for an open-ended configuration.

The overall test-bed management, control and data-logging task subdivision has been performed by appropriate programming techniques.

Finally tests have been carried out on two different engines and test-beds to demonstrate the flexibility and transportability of the controllers.

CHAPTER 2

The Microcomputer Network

2.1 The Introduction of Networks in Engine Testing

Microcomputer networks and multiprocessor systems were introduced in engine test-beds in order to meet the requirements of increasingly complex data logging and control procedures, while retaining the microcomputer as a base unit, for its economic advantages. This approach, if implemented in a modular way, is able to cope with future increases in data logging capacity or increased sophistication in the test-bed controllers or even the introduction of new concepts in engine research (on-line, recursive system identification).

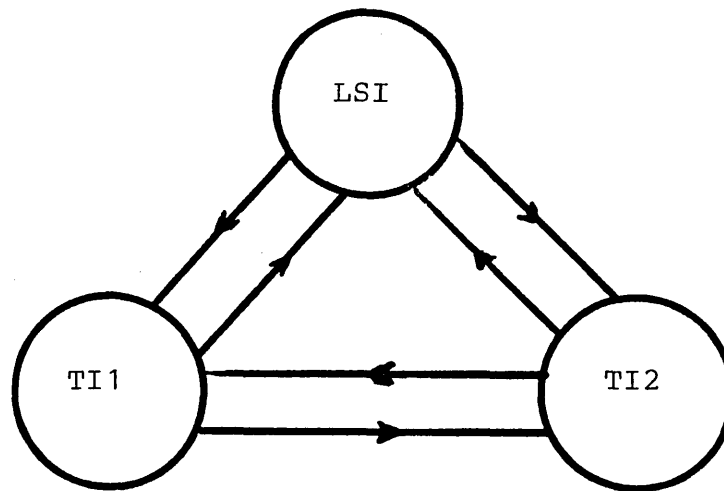
Distributed processing was introduced in engine testing after the 1977 computer industry 'boom' and provided a very economic and flexible alternative to concentrated mainframe processing power. Recent literature on the subject provides examples of such an approach. The internal combustion engines group at Osaka-City University [A-2] used a multi-microprocessor system to improve their data logging facility and control engine speed during transients. Their configuration consisted of a host CPU (Central processing unit) that controlled the system bus and three slave CPUs performing control tasks. All CPUs were connected on a common bus and shared the same main memory. Such a system requires identical processors (same make and type) to ensure compatibility on memory access and bus handshaking. Beckman Instruments [G-12] used the distributed system approach to build a vehicle emissions measurement system. The computational complexity required for their instruments, necessitated a solution which caters for economical computing power through an increased number of processors. Cichy [C-8] describes a test-bed

control system using two communicating microcomputers. Pajdla [P-2] describes what the author considers as being the prevailing view for a distributed microprocessor system for engine testing. The proposed architecture is fully hierarchical and every lower level implementation is completely transparent to the higher level. The higher level controller is responsible for overall test scheduling and control, and communicates with a set of intelligent devices which in turn can be implemented around a microcomputer. Every intelligent device performs a specific task and by the nature of hierarchical structures the protocol of communication with the higher level is rigidly fixed.

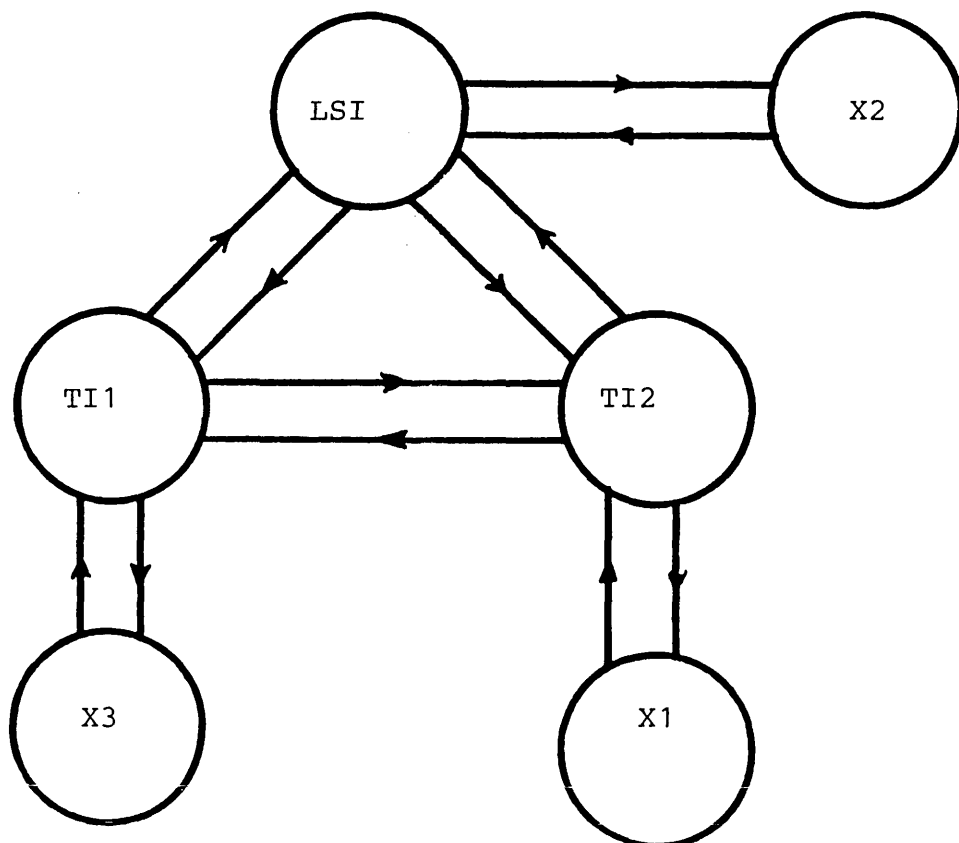
It is generally agreed [P-2, S-8, W-4] that process control and data acquisition applications require a hierarchical structure so that modularity can be enhanced. It should, however, be pointed out that an hierarchy in the interconnection structure, should not isolate every lower level branch from non adjacent upper levels. This consideration is very important when such a system (like the Imperial College test-bed controller) is to be used as a nucleus where other applications will be built around it and not as a system with strictly and rigidly defined objectives. It had to be anticipated that eventually more microcomputers would join the test-bed system and to allow for their introduction with minimal disturbance to the existing configuration and with maximum use of already existing resources in terms of interfaces and measuring devices.

2.2 Design Considerations and Objectives

The use of two different microcomputers (PDP 11/23, TMS 9900) during early development work at Imperial College for data logging and control respectively, did not allow for multiprocessor



existing configuration



possible future configuration

LSI : LSI 11/23 microcomputer - data logging and supervision
 TI1 : TMS 9900 microcomputer - stepping motor controller
 TI2 : TMS 9900 microcomputer - torque controller
 X1 : wastegate and Turbocharger inlet controller
 X2 : exhaust gas analyser
 X3 : multivariable engine controller

Fig. 2.1 Network configurations

configuration where absolute bus structure compatibility is essential. A multi-processor configuration would not be desirable anyhow, in that the different subsystems cannot be isolated during development and new achievements in microcomputer technology cannot be integrated within the system, unless a complete redesign takes place, which greatly reduces the maintainability of the system. Network structures were therefore investigated within the engine test-bed controller context.

An important consideration in the design of the network should be the ability to adapt to changing (expanding) topologies. To be more specific, it is desirable to allow for extra microcomputers to join the existing 3-node configuration, without fully interconnecting them to the existing microcomputers.

The nodes should not necessarily be operational at the same time, in case they correspond to physically different test-beds. It should be possible to transmit information in a way completely transparent to the processes running at any node without limiting the communication capacity to adjacent microcomputers. This requirement will greatly reduce extra instrumentation and interfacing in the sense that the newly inserted node can be presented with information available at another node, without equipment duplication. If, for example, node X3 (Fig.2.1) requires the torque available at node TI2 it should acquire it through the system communication rather than through an additional interface. Transparency of the communication process to all intermediate nodes is a prerequisite. It is essential that the communication between any two nodes does not interfere with the task executing on any other intermediate node, otherwise every new insertion in the network or even a change in its communication requirements would assume a modification of every single node software. It is also important to allow for a mechanism of fast

notification of a node in case of extraordinary conditions. That can be an abort, synchronisation or some kind of handshaking message. This type of message will be referred to as supervisory. Supervisory messages should have precedence over data transfers and be able to interrupt long messages.

A communication system should be fault tolerable. Bearing in mind that the use of electrically noisy spark ignition engines is an eventuality, the communication software should include some self-checking and recovery mechanisms from corruption. These mechanisms should proceed in a way completely transparent to the end-transmitting nodes. Sustained corruption should be notified to the rest of the network and result in an abort condition. It is desirable for reasonable message lengths to be transmitted as a single logical entity.

The test-bed network nodes are not dedicated communication processors and therefore the communication overhead should be kept to a minimum. This implies a relatively simple protocol to ensure fast execution. Simplicity in the communication design is more important than the ability to recover through all possible failures. It is the extremely complex nature of network software that made its use prohibitive in engine test-beds. The author attempted to rationalise the requirements of the particular environment and to design a system tailored to these needs.

There are various constraints dictated by the structure of the communicating microcomputers, the most important being the 16-bit word architecture. This will influence some elements in the software and hardware implementation.

2.3 The Network Communication Structure

It is desirable to maintain 'equivalence' among different nodes. This does not interfere with a functional hierarchical structure but provides flexibility and transparency in the exchange of information. A packet switched network protocol has been chosen to cope with the objectives set in section 2.2.

A packet switched network splits the messages into fixed length packets hence does not require buffers for storage of transit messages. In contrast with message switched networks, it transmits the data in packets which can be treated as independent entities as far as their handling is concerned. The basic Transport facility will be a Virtual Circuit. [S-7, D-2]. This ensures that packets will arrive in sequence. The service is similar to the permanent Virtual Circuits recommended by CCITT. The virtual circuit number will be the association of sender and destination identification. A global namespace will be implemented, being the most appropriate for the expected size of the network [D-2].

Another important consideration is the routing technique, which resolves a packet immediate destination. In case the destination node is not adjacent to the sender then the next hop should be decided. Fixed directory routing will be used as the simplest solution with minimum computational overheads. Bearing in mind that engine test-bed networks are not primarily communication networks but multiplexed controllers and communication processors, adaptive routing is not considered. A new node introduction only requires an additional entry in the directory. This amounts to minimal alteration in the communication software.

2.4 The Data Exchange Protocol

Once the general guidelines, at a network level, have been fixed in section 2.3, the detailed structure of packets and associated protocols will be investigated within implementational restrictions.

The first decision to be made involved the choice between serial and parallel transmission. Commercially available UARTs (Universal Asynchronous Receiver Transmitter) could handle 8-bit messages, at a time. That would slow down considerably the data interchange rate and in addition the link would be very noise prone. In order to increase the interchange rate it was decided to implement a 16-bit parallel bidirectional data bus and an interrupt request line so that the whole process could be interrupt driven. This choice imposed immediately a limit of 16 bits per packet. One byte (8 bits) of information was desirable to be transferred, per packet. The remaining 8 bits had to contain all the necessary information for packet routing and identification. These restrictions were observed when a new full-duplex hop-to-hop protocol with end-to-end services was designed. The author termed this new protocol 'PCNP' (Parallel Communication Network Protocol). The PCNP specification is presented below.

2.4.1 Data Transfer Control

The PCNP frame format is shown in fig. 2.2. Any node starting a message transfer initially sends a request which is a supervisory packet with the data flag bit equal to 0 and the message count bits declare the message length in bytes. The next node replies with an acknowledgement packet and the message transfer begins in a handshaking mode until the originally declared message bytes are transmitted. There is an upper limit of 127 bytes per message, deriving mainly from the bit field restriction.

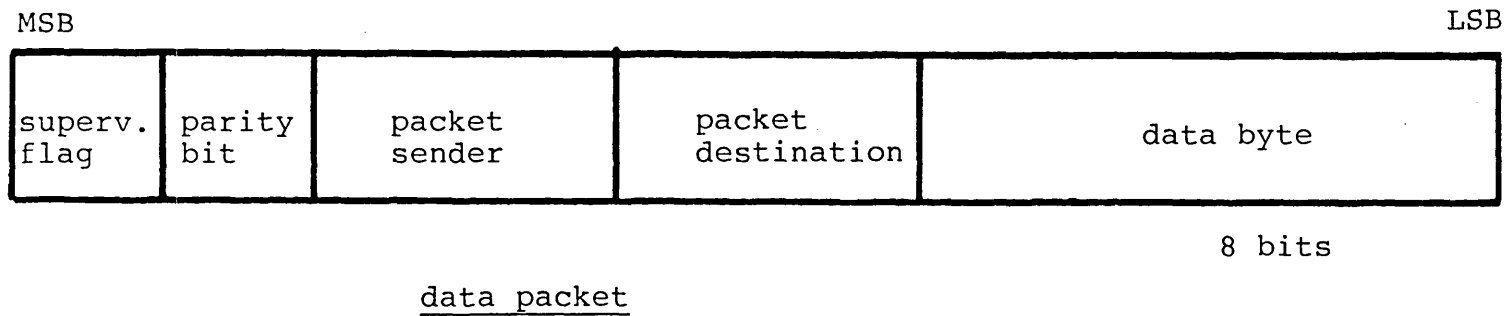
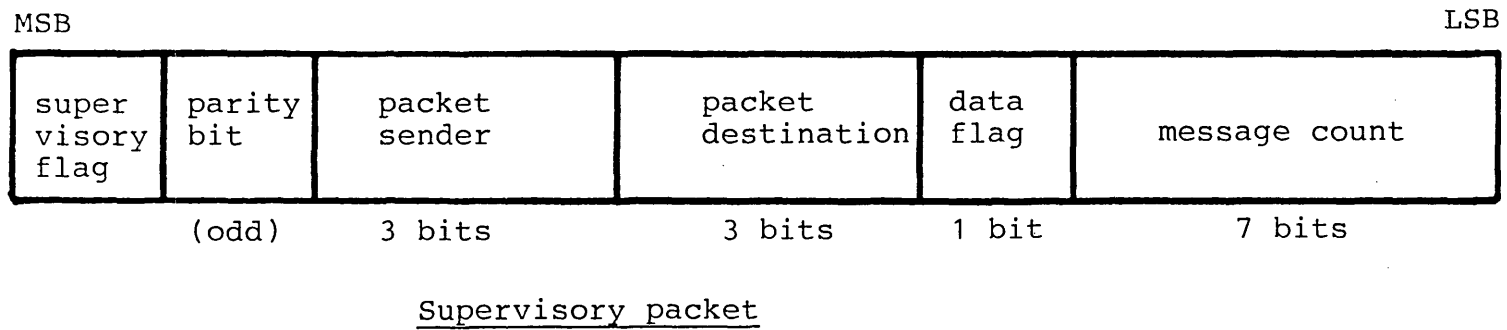


Fig. 2.2 PCNP frame format

A supervisory packet with the data flag bit high is considered as a pure supervisory packet and can be used to transfer emergency information among processes in the network. Two bit patterns are reserved for system's use $7F_{16} = \text{NAK}$, $7E_{16} = \text{ACK}$). Any other 7 bit pattern can be used by the application processes.

Transmission is full duplex, with messages exchanged concurrently from two neighbouring nodes. A Stop and Wait ARQ (Automatic Repeat reQuest) is used to acknowledge (ACK) or not acknowledge (NAK) transmitted packets. In order to avoid heavy packet traffic resulting from data and ACK packets, a node will not issue an ACK if the output queues to that node are non-empty. The receiving node will consider as an acknowledgement of its previous data packet any correct packet except from a NAK. The nodes involved in an error recovery are prohibited from transmitting data until the interrupt driven retransmission results to reliable data exchange and clears the corruption flag. A typical data exchange example is shown in Fig. 2.3. By the nature of the routing technique and the virtual circuit architecture packets are ensured to arrive in sequence, therefore data packets do not require a sequence number.

2.4.2 Error Checking and Recovery

As the actual link between adjacent nodes is a parallel bus and every transfer will generate an interrupt irrespective of data correctness, timeout support is not essential and due to lack of supporting hardware (programmable real-time clock) it has not been implemented. The error checking and recovery procedures are designed on the assumption that a packet, correct or not, will always arrive at the next node.

Parallel communication does not require CRC (Cyclic Redundancy Check) bits. A parity bit has been implemented (Fig.2.2) to ensure

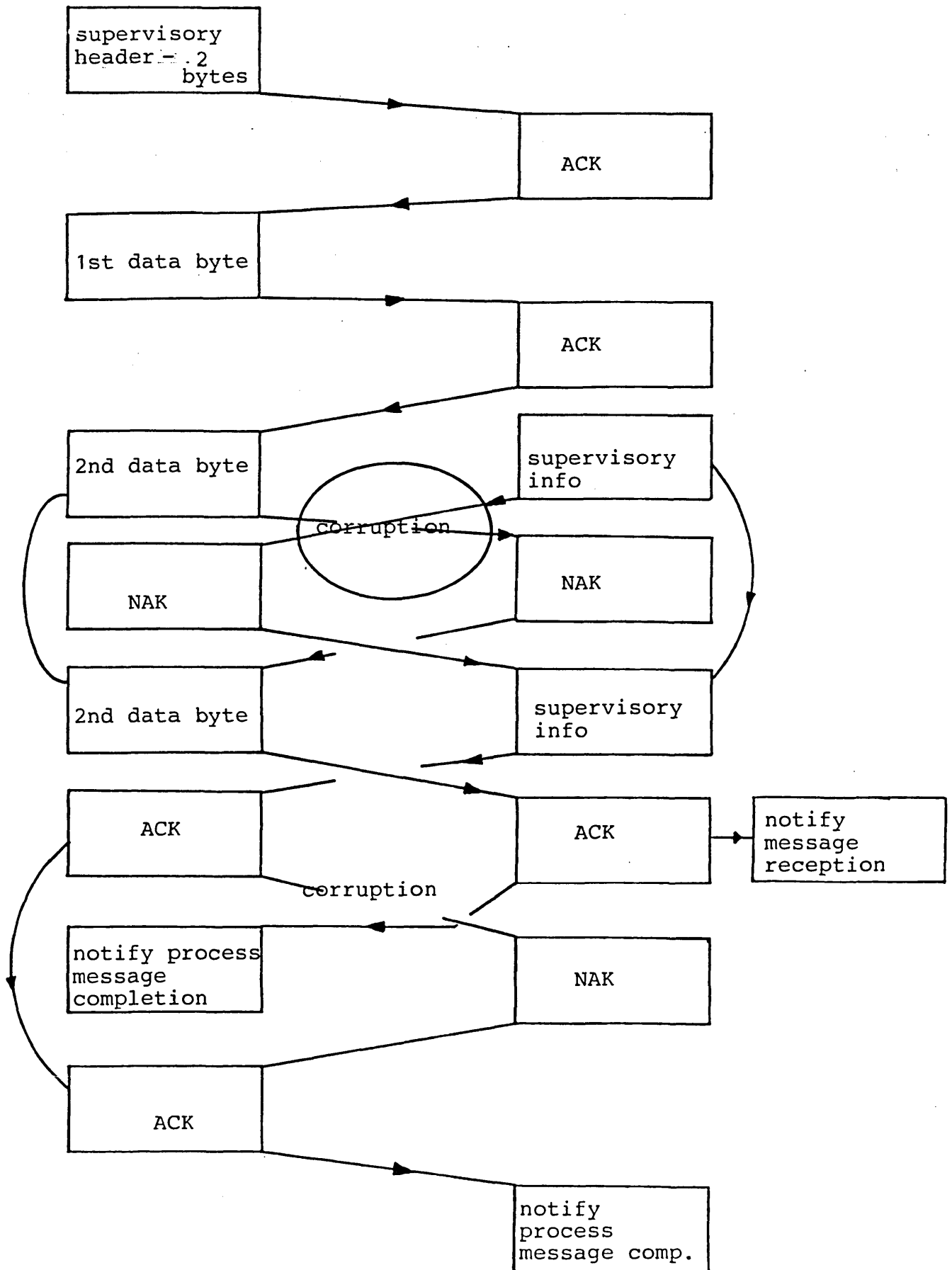
NODE ANODE B

Fig. 2.3 Data exchange in PCNP.

odd parity. It will be used as safety for high noise contention. When an error is detected a NAK message is sent to the transmitting node. Then the transmitting node will repeat the last packet sent, up to a predetermined maximum number of times. If re-transmission exceeds this number (selected as 3 for PCNP) of times consecutively then a user specified abort procedure is initiated, signalling the high noise contention of the environment.

2.4.3 Summary of System Characteristics

The message structure is completely transparent to the protocol. Any 8-bit pattern can be transferred per packet, the parity bit being outside this field. The count header allows for any block of data (up to 127 bytes) to be transmitted. In case of supervisory data then the 7-bit patterns of 7F16 and 7E16 are reserved for system use. The address bits can support only up to eight nodes in the network. Timeouts are not handled due to the lack of supporting hardware and increased complexity. Finally, line utilization can be improved by the full duplex mode, and by including acknowledgements in opposite direction messages (cf. DDCMP).

2.5 Software Implementation in a Layered Structure

The ISO Open Systems Interconnection recommendation defines the hierarchical (layered) structure of a communication network. A layer, according to the recommendation, consists of a set of service units (entities) which provides services to the next higher layer, and it does so by making use of the services of the layer immediately below. The service units constituting a layer are part of the data stations forming the network and so can be considered as a distributed sub-system, co-operating to provide a service. The level of a layer is an ordinal number associated with the layer. The

numbering of layers starts at the transmission line, therefore a 'lower' layer is closer to the transmission line, and a 'higher' closer to the application.

The various standards organisations have defined the services provided by each layer, though these definitions are by no means static. A number of layers could be amalgamated into a single one to improve efficiency, or the services provided by some layers may not be required in smaller networks. PCNP provides services up to Transport Control layer. The transport service isolates the higher levels (application processes or users) from the actual transmission facilities used in the network and provides the higher levels with the means of exchanging information. PCNP provides a telegram service i.e. exchange of independent messages. The Transport control layer is responsible for addressing. Each user of the transport service has a unique transport address (port). The transport service is responsible for the physical transparency of the transport address, by isolating the user process from the physical network structure. Another function is the connection control. It establishes and terminates suitable connections to provide telegram service. Priority control is also resolved at this level. The multi-level traffic handling with different priorities is implemented within the transport service. The widely accepted layers below the Transport Control are the Network, Datalink and Physical control.

The Network and Datalink layers have been functionally simplified in PCNP. The Network Control layer lost its major importance by introducing fixed directory routing. The Datalink Control layer is responsible for the exchange of link frames between two adjacent stations. The offered service is to convert any error prone physical circuit into a relatively error free logical link. The PCNP data link specification ensures transmission reliability by specifying the

error control and recovery procedures.

Overall, the software should provide Transport Station facilities. A Transport station consists of the Transport, Network, datalink and physical layers, within a single data station. The Transport stations communicate according to the end-to-end protocol. This relieves user processes requiring the transport service from problems of sharing transmission resources or how they are implemented.

2.6 Functional Description of Layers

To understand the interaction between the different routines implementing a layer, it is necessary to describe the underlying queue structure. Every logical connection requires a set of data which is gathered into a NCB (Node Control Block). The NCB contains all necessary information about the link and is shared by all layers in a protected access manner. The NCB are initially kept in a linked free list. The Transport and Network control layers will extract NCBs from the free list according to the requirements, and append them on to the main system queues. The system keeps four distinct queues per node, to reduce search time. There are two input queues, one for supervisory and another for data packets, hence supervisory communication is enhanced and transit delays minimised. The other two queues are used for output and their structure and functionality is similar to the input queues. It should be mentioned that the queues and their associated handlers implemented a FIFO (First-in-First-out) environment.

2.6.1 The Transport Control

This layer presents the user interface and is built around the two network handling primitives REMSND (Remote Send) and REMRCV

(Remote Receive). The user process must notify the Transport station about its intention to use the station facilities. In addition it should provide all necessary information about the message in order to enable the Transport station to perform its internal checks and effect its resource allocation (NCB assignment). More specifically the user process is expected to supply the following information:

- a) Buffer address where the message is to be stored (reception) or found (transmission)
- b) The address of a flag which will be set to .TRUE. upon completion of the transfer.
- c) The node involved in the data exchange (sender node, in case of reception, destination node in case of transmission)
- d) The length of the message (in bytes) (a zero length message is considered to be supervisory).

The Transport Control layer will return a signal to the user process to notify that resource allocation was successful and eventually the transfer will be completed. In the case of reception the user is not expected to be aware exactly of the message length. The Transport station will return the length of the message but will abort if the received message exceeds the declared buffer length.

NCBs are initialised by the Transport Control layer. All control fields will be filled by the user and system information at the moment of the call. After this layer is executed, lower layers have all necessary information about routing and the message itself so that data exchange can proceed. A Directory lookup is effected within this layer for the sake of effectiveness.

2.6.2 The Network Control

This layer initiates transfers prepared by the Transport Control. Two routines NETSND (network send) and NETRCV (network receive) implement the functions of this layer. Here the NCB queues are scanned and differentiation between data, supervisory and transit packets takes place. The sender and destination bits are also adjusted within this layer to resolve the queuing of each individual packet and its association with an existing or near NCB. Data exchange control is completely handled within this layer, once Transport Control performed the initial allocation. The user process is ultimately notified from this layer about completion of the transfer using information in the NCB. The assembly of an outward packet takes place here, as well.

2.6.3 The Data Link Control

The purpose of the Data link layer and its two associated routines DLRCV (Data link receiver) and DLXMIT (Data link Transmitter) is to ensure transparency in the recovery procedures of information corruption. This layer is responsible for checking and setting of the parity for the entire packet. The Data link layer issues a NAK in case of a reception error, and increments the corruption counter for the particular link in order to decide whether to pursue abort procedures. It keeps a copy of the latest transmitted message until acknowledged. Higher layers are only called when an excessively noisy environment fails to sustain reliable communication. The positive acknowledgement (ACK) is issued at this level to inform the sender node about error free reception, unless the output queues to the node are non-empty.

The reason for adoption of a Stop and Wait ARQ technique was implied by the lack of a buffering front-end processor which could

probably handle the whole network traffic. This layer will keep information, about the communication condition with each node, independently, so that individual evaluation about the transmission line can be effected.

2.6.4 The Physical Layer

The physical layer is implemented through the hardware interface that will bring the computer signals to the transmission line. It is very much implementation dependent and differs on each type of computer in the network. The requirements set by this particular PCNP implentation is the existence of a 16-bit bidirectional data bus and and IRQ (interrupt request) line to ensure the nature of PCNP handshaking at the Data link layer and cope with the lack of timeout support.

2.7 The PCNP Philosophy

Since the emergence of computer networks, the design of protocols had as its main objective the achievement of data transfer reliability, network throughput, low system delays and other user oriented measures. [M-3]. The underlying hypothesis in such designs was that a network is a loosely-coupled system [S-8] facing all perils deriving from its distributed nature. The risks in data transmission within such systems included the complete loss of a packet, the duplication of a packet, complete disruption of a link and correction during transmission. In other words a network protocol was designed to cope with anything that could disrupt the communication. It is obvious that such an approach requires extremely complex recovery procedures.

At the other end of the taxonomy appear multiprocessor systems, including several processors in close physical proximity and

communicating with each other with shared memory or by high-speed common bus. This is also known as a tightly coupled system. A multiprocessor system is characterised by the fact that all processors can directly communicate with each other (either by common bus or by a fully interconnected topology) so that no store and forward operation is required for data transmission to an arbitrary node.

The Imperial College engine testing arrangement was somewhat different. There were microcomputer systems, rather than just processors, in close proximity, communicating with each other by high speed point to point rather than broadcast buses. Muhleman [M-19] considered an operating system Kernel without process management which he extended for the multiprocessor case. He defined two non-local synchronisation primitives (REMINC, REMDEC) which implied the existence of a broadcast bus. He is therefore attempting to transpose the problem into a hardware handshaking design. Muhleman's paper was quoted as, to the author's knowledge, it is the only simple communication scheme suggested for a network. A further refinement by Muhleman [M-20] suggests a model of communication without topology and interconnection restrictions but with the increased overhead of requiring a complete operating system Kernel with sophisticated process management.

Processing requirements of a network operating system imply the need for a dedicated communication processor to handle the network traffic. [P-8]. At the same time it is absurd to consider a front-end communication microcomputer for engine test-bed requirements. The solution lies in the rationalisation of the needs of the specific application, the assessment of the risks involved in information exchange, and while the integrity and flexibility of the data interchange remain intact, exception handling and recovery are

downgraded to allow for a feasible implementation. PCNP was developed around these guidelines. The addressing generality at the Transport control layer was maintained. Multiprocess addressing was omitted as the microcomputers are unlikely to run on a multi process mode. An important omission was the end-to-end protocol. An internal check is performed instead, to confirm that the running process expects information from the particular node. This implies that processors running on different nodes are aware of each others structure and if crucial information has been delayed they can initiate abort procedures. It should be stressed that a network in engine test-beds should not provide reliable communication assuming the links are operational (though not noise free). This assumption led to the end-to-end protocol omission. Once a packet initiates a data exchange it is bound either to arrive at its destination or cause an abort. The reasons for this are:

- a) If a node transmits a packet then the next node should respond with either an ACK or NAK, assuming it is operational.
- b) All internal queues are implemented in a FIFO environment, therefore no message will be forever pending.
- c) Excessive transmission line noise will generate repeated transmissions that will force a system abort. Even if supervisory abort messages cannot be transmitted because of noise, the communication failure will generate an implied abort.

The omission of end-to-end protocol will not decrease data integrity but will cause an abort when a more sophisticated handshaking might be able to recover from the fault.

Some existing protocols that have been reviewed include the CCITT X.25, IBM SNA (System Network Architecture) and DECNET (Digital Equipment Corporation Network Architecture). Their common characteristic is the use of either a virtual link or a single master node. Each specification hides the reasons for which the protocol was primarily designed. The only protocol with some degree of applicability in a microcomputer network is DECNET DDCMP (Digital Data Communications Protocol) that manages the data link layer. One shortcoming is that the DDCMP header format does not have both source and destination address, assuming point to point connections or multipoint with a single master node. PCNP includes both source and destination in the packet and a two-level traffic priority for supervisory communication in line with the ANSI recommendation in ADCCP (ANSI Advanced Data Communication Control Procedures) for control applications, to reduce supervisory information transit time.

As a conclusion to the PCNP description it should be noted that there is yet no panacea in network protocols. Most specifications have been derived from a specific need, or to accommodate a new piece of intelligent hardware [S-8].

PCNP was specifically designed for local loosely coupled controllers, to serve engine test-bed communication.

CHAPTER 3

Test-bed Control Strategies

3.1 Task Division within the Network

Engine testing can be seen as the achievement of three distinct objectives:

- a) the management of all data logging devices and handling of I/O for this data
- b) the application of a certain torque to the engine by means of a dynamometer
- c) the control of the inputs to an engine (e.g. fuelling, injection advance) to achieve some objective (e.g. speed, emissions, consumption).

Each of these tasks has been confined to a microcomputer in the network. The actuators and devices used for the data logging system together with their requirements are extensively covered elsewhere [R-5, M-5, L-5, P-5, B-12]. The present work investigates the performance of load and speed actuators. Engine testing necessities are evaluated in order to examine the actuator control feasibility.

3.2 The Torque Control

The control of load application to the engine is particularly important when carrying out research on the transient behaviour of the engine. The requirements are for a rapid (step) load application with the minimum possible overshoot.

Overshoot is undesirable because the momentary increase in absorbed power will introduce a hysteresis in the measurements and may mask or distort the transient performance of the engine itself.

The mathematical formulation of the control problem can be stated as follows:

Given the control system defined by:

$$\frac{dx}{dt} = f(x, u) \quad (1)$$

where x is an n -vector for the n th order lumped parameter eddy-current dynamometer model (assumed to exist) and f a vector function (also assumed to exist) of x and the scalar control input u (the input current to the dynamometer windings), find the optimal control function $u^0 = u(t)$ which minimizes the performance index

$$J[u] = \int_0^{t_1} f_0(x, u) dt \quad (2)$$

where f_0 is the cost function (in this context cost means the prescribed penalty)[A-9, T-3] subject to the constraints:

(a) $0 \leq u \leq I$ (input current limit)

(b) $x \in X$ where X is the set of admissible states

to move the control object (dynamometer) from the initial state $x(0)$ to the target state $x(t_1)$.

The only term that is not clearly defined in this problem formulation is the cost function f_0 . It is reasonable to assume that f_0 should not have a cost for the input, reflecting the fact that in practice there is no penalty for an increase in the electric field intensity, and the resulting increase in electric power consumption. But when the input cost disappears from the cost function then one is left with a Bang-Bang controller. Furthermore the requirement for a minimal (zero) overshoot during step changes could well be met if the cost function is unity transforming the problem into a time-optimal

one, and the controlled object (f) is of a certain special form [A-9]. With such a controller, it will be proved that the time to move the control object from an initial state, to a final state is minimum and the theoretical overshoot on the movement from state $x(T_1, 0, 0, \dots)$ to the state $x(T_2, 0, 0, \dots)$, where T_1, T_2 are initial and target torque, is zero.

Up to now eddy current torque controllers have been three term controllers [V-1, P-7] and some heuristics have been tried [N-1, W-10] to improve step response. A three term controller can be optimal under the following assumptions:

- (a) the cost function is of the form

$$J = \frac{1}{2} \sum_{k=0}^N \left[x_k^T Q_1 x_k + u_k^T Q_2 u_k \right] \quad (3)$$

where Q_1, Q_2 are symmetric non-negative, constant, $n \times n$ matrices and in addition Q_2 is positive (the cost on the input cannot be zero). This is known as a quadratic cost function.

- (b) there are no constraints on the input
(c) the controlled object is of the form

$$\frac{dx}{dt} = A x + K u \quad (4)$$

where A is a constant $n \times n$ matrix

- (d) the order of the matrix A is 2
(e) the parameters of the control matrix are the optimal ones

Assumption (a) is unrealistic and unnecessary as explained before. Assumption (b) cannot be met as in practice there are physical limits on the electric and electronic actuators. Assumption

(c) is correct as the model derived for the dynamometer is of that form. Assumption (d) is also correct as a viable approximation to the real model based on the observability of the states.

Assumption (e) although can be met by the model derived by the author, has never been achieved and the tuning of the three term controllers was based on the Ziegler-Nichols recommendations which imply ITAE optimality for a first order system with a delay [P-7, W-10].

It should also be stated that assumption (c) is correct when the concerned state is the flux and its derivatives and not the torque. This is the first attempt, to the best of the author's knowledge, to use the flux as a state variable for the control of eddy-current dynamometers.

Another field that has been investigated is the benefit of preview action. The preview control problem is one in which a controller can use future as well as present and past information to determine the control.

Engine test cycles are in general predefined therefore preview methods are applicable. The derived optimal controller can be easily adapted to include preview action and the benefits are demonstrated by experiment and simulation.

3.3 The Engine Control

A little over 100 years ago, James Clerk Maxwell, read his classic paper 'On Governors' before the Royal Society [M-10]. He gave mathematical analyses of the dynamic behaviour of three designs of governors then in use. He was concerned with stability. Most of the early research on governors has been devoted to the problem of keeping a steady rotation of the controlled object when an external disturbance (load) acts to shift it from the set point. Now that

automotive engines are required to achieve more complex objectives like consumption and emission minimisation this crude control of the early stages is no longer sufficient. Welbourn [W-5] first predicted the beneficial use of a derivative term in the existing P (proportional) and PI (proportional-integral) compression-ignition engine governors as well as load feedback. Webb and Janota [W-2] further correlated the necessary loop gain as a function of the disturbing load on governors with load feedback. Other heuristic approaches led to fuelling limiters of one sort or another. Flower [F-5] used earlier proposed discrete models for turbocharged diesel engines to design an optimum controller that minimises a quadratic performance index accounting for the error in the speed, the cost of control, and the air-fuel ratio. This approach will give rise to an optimal linear regulator but the reconstruction of the state variables from engine and turbocharger speed does not seem feasible.

A discrete three term controller was used for the engine, with the setpoint band modification proposed by Cox [C-10]. A torque feedback feature has been included based on torque information communicated between the torque and engine controllers and a pre-compiled table of the required change in rack position. The resulting disturbance rejection improved dramatically as is demonstrated in Chapter 7.

The author has not attempted any mathematical modelling of the engine, as this is a subject on its own. The three term controller used together with the modifications of the setpoint band and load sensing will cover most requirements and the software simulation of existing governors. The important component of the test-bed is the actuator, moving the throttle, the fuel rack or the governor setting. As previously mentioned a stepping motor was used for that purpose. The time-optimal software controller has been designed for that

actuator. The time optimal nature of the controller ensures zero positional overshoot, and the shortest possible time to reach the new state. The software nature allows for extreme flexibility as far as the performance is concerned, allowing different governor gains to be investigated by the simple alteration of some software constants. Hazell and Flower [H-5] as well as Bowns [B-10] investigated the compression-ignition engine as a sampled-data system and pointed out the importance of how the fuel rack movement can influence performance. Such investigations would find the gain simulations offered by the software stepping motor controller useful.

3.4 Decoupling of Speed and Torque Control

Strictly speaking the control task on an engine test bed is a multivariable one. The torque application influences the speed, the change in speed influences the torque, the change in speed will influence rack position which itself is going to create a secondary change in speed. In order to command the outputs T_e (torque) and V_e (engine speed) and eventually E_e (emission levels) the torque controller should be aware of the speed (V_e) and the speed request (V_d) so as to compensate the action of the feed forward path G_1 (Fig. 3.2). That in itself would imply the incorporation of the engine and speed controller models in the torque controller. This requirement, though necessary for an overall optimisation, would render the structure of the torque controller extremely complex and still require a comprehensive engine model which, unless small perturbations are examined, does not exist in a universal form realisable within real-time constraints. The same rationale applies to the engine controller. If of course someone would opt for a perfect driver model then this approach is the only one to solve what has been described by Webb and Janota [W-2] as the most undesirable

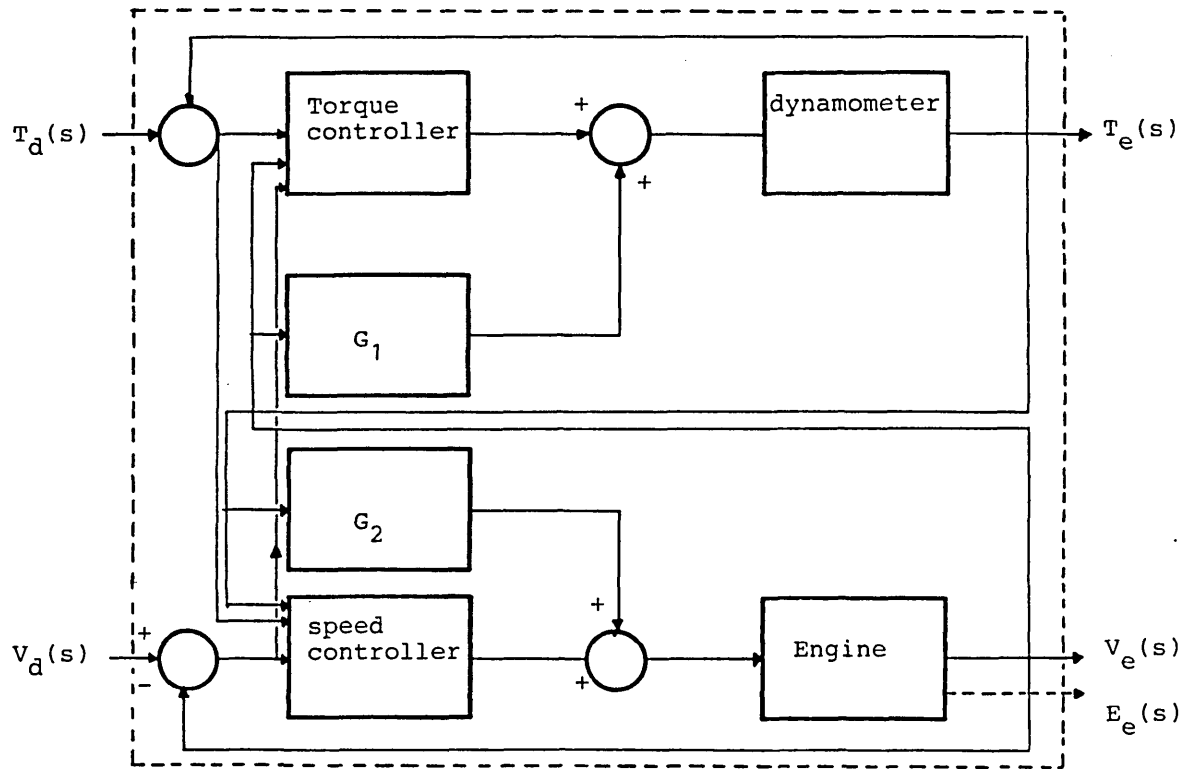


Fig. 3.1 The engine test-bed coupling

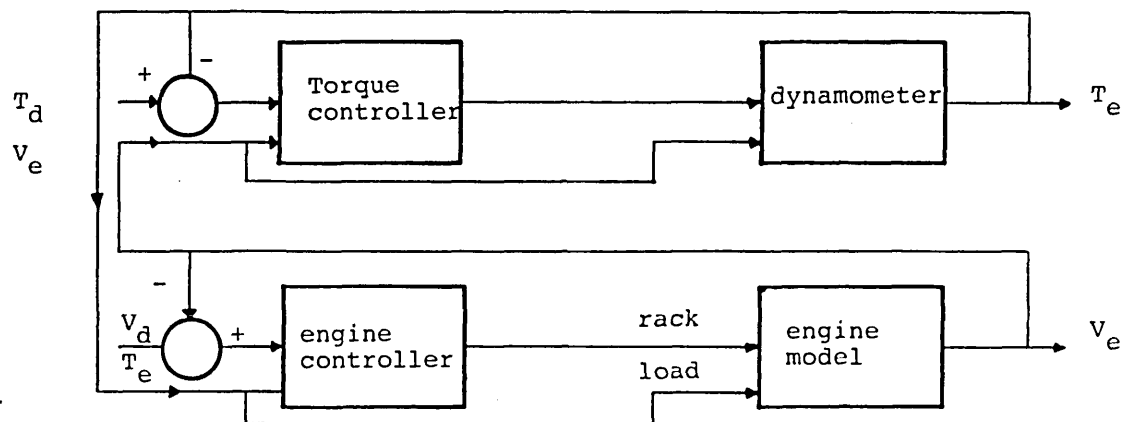


Fig. 3.2 The engine test-bed decoupled

feature of the *modus operandi* of the conventional governor compared with the speed control of a road vehicle with conventional transmission:

"Let us suppose that the vehicle is being driven fast on a motorway; the throttle will be well open and the engine will be overcoming a considerable load. To simulate the sudden removal of load from the engine, the driver simply has to depress the clutch pedal. An experienced driver would automatically release the accelerator pedal at the same time as he depresses the clutch pedal. This is what the conventional governor cannot do because, for reasons of stability, the gain of the system cannot be made large enough to prevent a time lag between the equivalent of clutch and accelerator movements."

Webb presents load sensing as a solution to this problem. Nevertheless the driver and the governor (even with load sensing) are not equivalent systems dimensionally, in the sense that the driver has additional information about the movement of T_d and V_d which any conventional governor has not, relying purely on the feedback.

An overall optimisation strategy does not appear feasible at present and therefore the interaction, as far as optimisation is concerned, between the Engine and speed controllers will be limited to the feedback values of speed and torque (Fig. 3.2)

All existing eddy-current dynamometer controllers do not have speed input to the torque controller, in order to avoid multivariable control techniques. [W-10]. Changes in speed are treated as exogenous disturbances which are compensated by integral action. Such an approach is bound to have considerably low attenuation on disturbance rejection characteristics. There is an alternative approach to the problem. What is eventually controlled is the

torque. What is responsible for the generation of the torque is flux. The flux itself is a function of the mmf on the windings and the reactive field which is a function of the speed.

If the speed and torque are mapped into flux then the control problem becomes equivalent to a single input-output, without losing the speed information. In addition the dynamometer model for the flux will be shown to be approximately linear whereas no torque-speed related model has been presented so far.

The engine controller has the same theoretical problems. As mentioned earlier there is no comprehensive real-time engine model, therefore simple models representing the engine as a single time constant process with a delay have been assumed giving rise to three term controllers tuned by the Ziegler-Nichols recommendations. The torque feedback has been used in an open-loop configuration assuming corrective action by the speed controller for the final speed adjustment.

The following chapters will examine the decoupled system. The torque controller will be derived after modelling the eddy-current dynamometer. The stepping motor controller will be investigated as the most important on-engine test bed actuator. The integration of the stepping-motor into the overall engine control will be discussed in Chapter 6.

CHAPTER 4

Eddy Current Dynamometer

Modelling and Control

4.1 Introduction

Eddy current dynamometers have been used almost exclusively for steady state work. It was only recently that requirements for transient testing emerged. The lack of a cheap alternative together with the high acquisition cost of dynamometers in general, led to the adaptation of eddy-current dynamometers for transient testing.

In some cases the advent of new power electronics gave birth to new excitation units [V-1, N-1] to accelerate the torque increase rate.

Until now engine research establishments and industry have considered eddy current dynamometers as a black box [W-10, M-7, B-8]. All industrial controllers [B-8] or purpose built DDC algorithms [W-10] were, to the author's best knowledge, three term controllers, analog or digital but with essentially the same response characteristics. The reason for the adoption of such controllers has been the lack of any comprehensive model for the behaviour of the dynamometer. The optimal control of an object requires a transfer function or state equation, describing that object. Once such a description exists then significant control improvement can be achieved. In addition, the limitations of eddy current dynamometers can be highlighted regarding response.

Eddy current dynamometers constitute an important module of numerous test beds. The emerging requirement for transient tests prompted the author to analyse their theoretical performance. This analysis is presented in this chapter.

4.2 Steady State Characteristics

The steady state theory of eddy current dynamometers has already been explored both from an analytical point of view [D-1] and as a requirement for the design of such dynamometers by various companies [C-5]. Selective parts of their work will be presented in order to relate torque with magnetic flux.

Assuming a steady state condition where the toothed rotor in Fig 4.1 rotates at constant speed, while a constant magnetic field has been applied in a suitable manner, then the temporal variation of the flux density has the form shown in Fig. 4.1 and alternates between a minimum and a maximum corresponding to a slot or tooth being located immediately underneath the point in question.

The flux through the small area S is:

$$\Phi_0 = \int_S \vec{B} \cdot d\vec{s}$$

The eddy currents introduced in the loss drum are proportional to the rate of change of flux, therefore the power absorbed is proportional to

$$\text{Power absorbed} \propto \frac{d\Phi_0}{dt}$$

This is the principle of operation of eddy current machines. Considering that $dB/dz = 0$ (where z = axial length) the losses due to rotor movement are the only ones to determine the torque. All other losses in the main mass of the magnetic circuit (far from the loss drum) are heating the body without any torque effect.

Davies [D-1] has derived general equations for the flux dependance on dynamometer parameters and operating conditions. The toothed rotor movement is seen to create a reactive field which itself is a function of dynamometer characteristics and operating

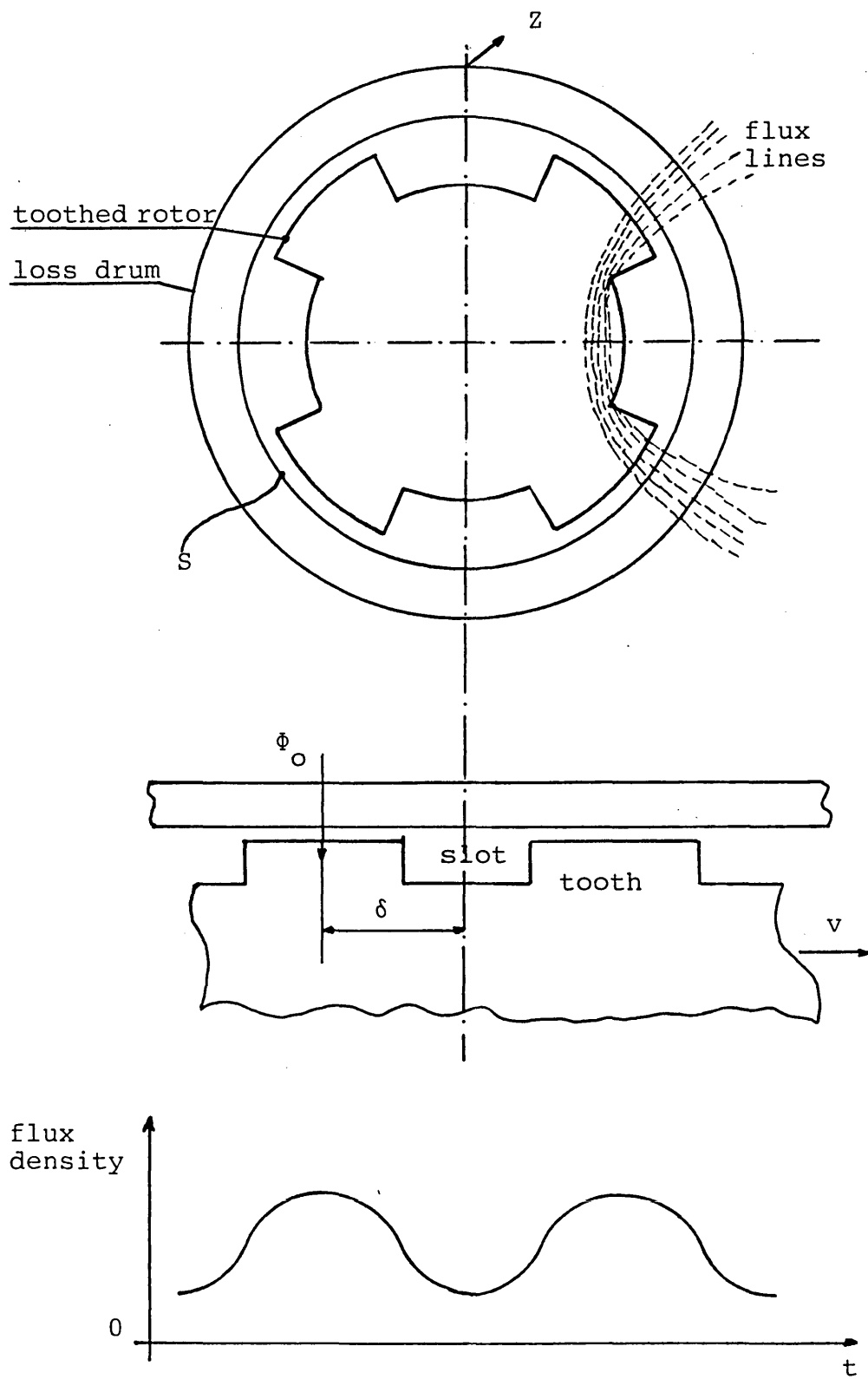


Fig. 4.1 Typical cross-section of eddy current dynamometer and flux density variation.

conditions. When the dynamometer is excited, the imposed mmf should overcome the reactive mmf and create at the same time the required field intensity to generate the required flux. Davies suggests that this armature reaction is some exponential function of the flux and rotor velocity. In this study the reactive flux will be neglected on the basis of low speed, low flux application and a mmf sizeably larger during the transient operation [D-1].

The instantaneous torque developed by the dynamometer depends on the value of the developed flux. The flux rise and fall will be investigated in the following paragraphs.

4.3 The Transient Flux Response of Solid Iron

Eddy current dynamometers are unlaminated machines. Their behaviour is not easily predictable. [P-6, K-4, S-11, C-7]. The general analysis of flux behaviour in solid iron is very difficult as can be readily seen by the complexity of the derived results [K-4]. These complex mathematical expressions are of little use to control modelling when feedback regulators are considered and considerable approximations have to be effected, if used, which in return need to be reconsidered after comparison of experiment and theory [C-7]. The approach adopted by the author was to study the transient behaviour of a semi-infinite body and derive some behavioural characteristics which in turn will help the construction of a model with close characteristics to the real dynamometer. What is required is the flux behaviour and more particularly the behaviour of the first and second derivatives. The analytic form will be derived without assumptions regarding the flux density penetration[C-7].

Theoretical Models and Assumptions

The magnetising windings placed close to the solid iron core can be represented by the theoretical model shown in Fig. 4.2 already used by other researchers [K-4, C-7]. Here the solid iron core is considered to be an infinite half space with specific resistance ρ and permeability μ . The magnetising winding is represented by a system of straight conductors parallel to the boundary air/solid-iron. No transverse air gap has been considered as the same analysis can be applied by considering a modified permeability for the same system according to the airgap dimensions. Eddy currents belong to the region of quasi-stationary effects (displacement current is neglected compared to conductive current). It is therefore possible to use the modified Maxwell equations for quasi-stationary state:

$$\text{curl } \vec{E} = - \mu \frac{\partial \vec{H}}{\partial t} \quad (1)$$

$$\text{curl } \vec{H} = \vec{J} \quad (2)$$

Equation (1) can be written as:

$$\text{curl } \vec{E} = - \frac{d\vec{B}}{dt} \quad (3)$$

assuming an unsaturated region.

It is also necessary to state the generalised ohm's law:

$$\vec{E} = \rho \cdot \vec{J} \quad (4)$$

substituting in (2) the flux density:

$$\text{curl } \vec{B} = \mu \cdot \vec{J} \quad (5)$$

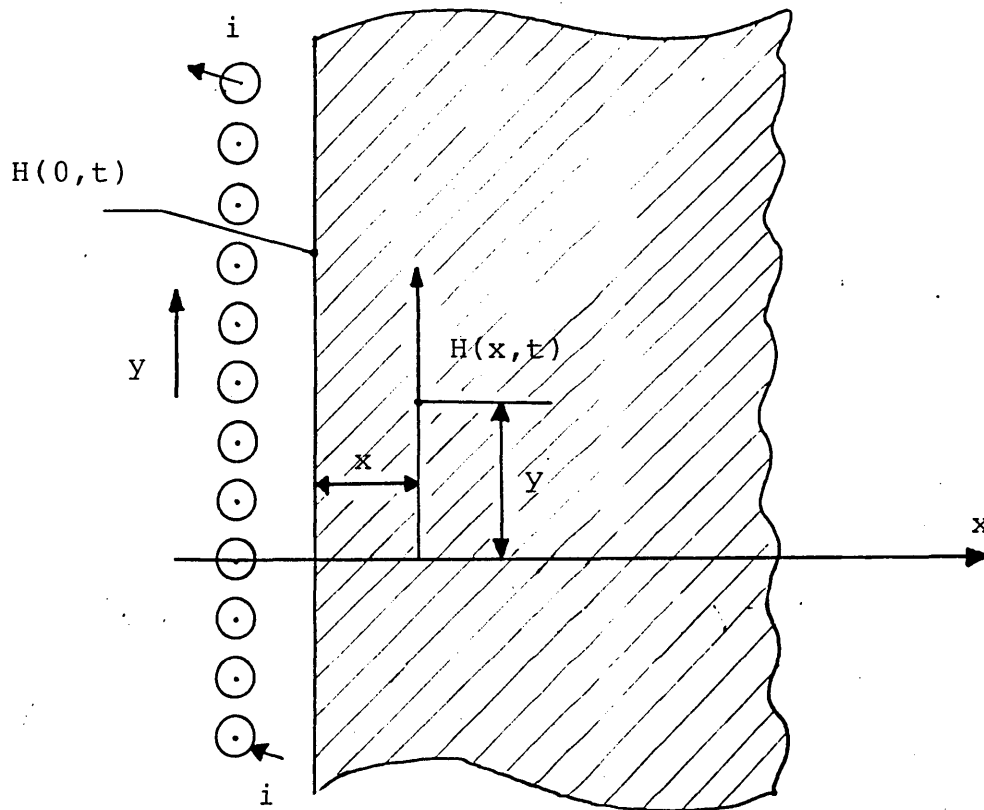


Fig. 4.2 Semi infinite plate model

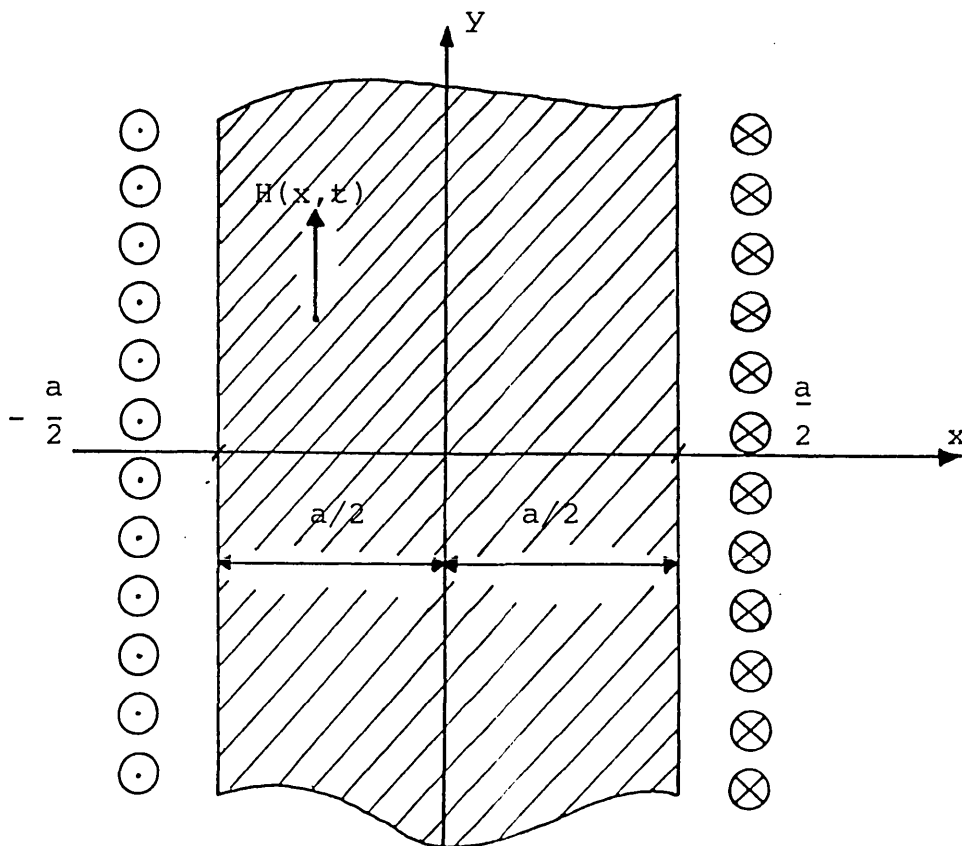


Fig. 4.3 The finite width plate model

substituting J from (4):

$$\text{curl } \vec{B} = \frac{\mu}{\rho} \vec{E} \quad (6)$$

By applying a curl to both members of (6)

$$\text{curl curl } \vec{B} = \frac{\mu}{\rho} \text{curl } \vec{E} \quad (7)$$

Substituting curl E from (3)

$$\text{curl curl } \vec{B} = - \frac{\mu}{\rho} \frac{\partial \vec{B}}{\partial t} \quad (8)$$

By the nature of the magnetic field (Gaussian):

$$\text{div } B = 0 \quad (9)$$

Applying (9) on (8):

$$- \nabla^2 B = - \frac{\mu}{\rho} \frac{\partial B}{\partial t} \quad (10)$$

which is the general flux density partial differential equation for quasi-stationary time varying fields.

The application of (10) to the theoretical model of Fig. 4.2 yields:

$$\frac{\partial^2 B}{\partial x^2} = \frac{\mu}{\rho} \frac{\partial B}{\partial t} \quad (11)$$

as $\frac{\partial B}{\partial y}$, $\frac{\partial B}{\partial z}$ are zero because of symmetry in these directions.

Considering the conductors very near to the core it is possible to ignore for the moment any leakage flux. The boundary condition to solve equation (11) is in general time dependant.

The current source required to achieve this input will not be examined at present.

Equation (11) will be solved with the help of Laplace transforms. The flux density is bounded in value so:

$$|B(x,t)| < M e^{\gamma t} \text{ for } x, t \in D$$

where M, γ are real constants and D represents the domain of B .

$$L(11) - s \cdot \beta - B(x, 0) = \frac{\rho}{\mu} \frac{d^2 \beta}{dx^2} \quad (12)$$

$$\Rightarrow \frac{d^2 \beta}{dx^2} - \frac{\mu}{\rho} s \beta = 0 \quad (13)$$

Equation (13) is considered as an ordinary differential equation and has the following general solution:

$$\beta(x, s) = C_1 e^{\sqrt{\frac{\mu}{\rho} s} x} + C_2 e^{-\sqrt{\frac{\mu}{\rho} s} x} \quad (14)$$

(14) should be bounded for any $x \in D$

In order to have:

$$\lim_{x \rightarrow \infty} \beta(x, s) < M \quad M \in \mathbb{R}$$

the coefficient C_1 must be zero ($C_1 = 0$). Thus

$$\beta(x, s) = C_2 e^{-\sqrt{\frac{\mu}{\rho} s} x} \quad (15)$$

Assuming that the input function (boundary condition) in (12) has a laplace transform

$$L [B (0,t)] = \beta (0,s) \quad (16)$$

then at $x = 0$ and combining (15) and (16)

$$\beta (0,s) = C_2$$

The general s -plane solution for the flux density can be rewritten as:

$$\beta (x,s) = \beta (0,s) e^{-\sqrt{\frac{\mu}{\rho}} s x} \quad (17)$$

The total flux through the semi infinite plate for unity width is:

$$\Phi(s) = \int_0^{\infty} \beta(x, s) dx \quad (18)$$

If the boundary condition is not a function of x (which it is not, being uniform) (18) can be rewritten as:

$$\Phi(s) = \beta (0,s) \int_0^{\infty} e^{-\sqrt{\frac{\mu}{\rho}} s x} dx \quad (19)$$

The evaluation of the integral leads to the following equation:

$$\Phi(s) = \beta (0,s) \frac{1}{\sqrt{\frac{\mu}{\rho}} s} \quad (20)$$

The only assumptions made to deduce this formula were that $B(0,s)$ is not a function of x depending only on time (s) and that its Laplace transform exists. Therefore (20) can be used as a general transfer function for the flux. By considering a step and a ramp inputs the following time-domain solutions are obtained:

$$\text{step input } \phi_s(t) = B_0 \sqrt{\frac{\rho}{\mu}} \frac{\sqrt{t}}{\Gamma(3/2)} \quad t \geq 0 \quad (21)$$

$$\text{ramp input } \phi_r(t) = B_0 \sqrt{\frac{\rho}{\mu}} \frac{t \sqrt{t}}{\Gamma(5/2)} \quad t \geq 0 \quad (22)$$

Chura [C-7] derived the same flux equations by effecting approximations and assuming a maximum depth penetration.

It is interesting to study the time derivatives of the flux as in an electric circuit the back-emf from the coil will be proportional to the rate of change of flux.

$$\text{step input } \frac{d\phi_s}{dt} = B_0 \sqrt{\frac{\rho}{\mu}} \frac{1}{2\Gamma(3/2)} \frac{1}{\sqrt{t}} \quad (23)$$

$$\text{ramp input } \frac{d\phi_r}{dt} = B_0 \sqrt{\frac{\rho}{\mu}} \frac{3}{2\Gamma(5/2)} \sqrt{t} \quad (24)$$

$$\lim_{t \rightarrow 0+} \frac{d\phi_s}{dt} = +\infty \quad (25)$$

$$\lim_{t \rightarrow 0+} \frac{d\phi_r}{dt} = 0 \quad (26)$$

The limits of the derivatives at $t = 0$ highlight some interesting results. From (25) it can be seen that the back-emf generated by a step increase in current is infinitely large and is practically unachievable. Equation (26) shows that for a ramp current input the effective inductance of the system will appear infinitely small.

Practically the current will shoot up to its V/R value until back emf is generated by the rise in flux.

4.3 The Case of a Finite Width Iron Plate

In the previous section the flux behaviour has been investigated for the semi-infinite plate. This was used to derive exact formulas and deduction of back-emf behaviour. It also set a framework within which further approximations of the real model can be achieved. Nevertheless it is not desirable to start the approximation from the infinite plate model as it does not reflect elements in the real model, such as steady state behaviour. It is much more realistic to start from a finite width plate and use some of the information on the infinite plate in order to simplify the results.

The finite width plate model is shown in Fig. 4.3. The same assumptions will be made as for the semi-infinite plate. The difference will be in the boundary conditions. Again $\frac{\partial B}{\partial y}$, $\frac{\partial B}{\partial z}$ are zero and (10) is written as:

$$\frac{\partial^2 B}{\partial x^2} = \frac{\mu}{\rho} \frac{\partial B}{\partial t} \quad (27)$$

The new boundary conditions are:

$$B \left[-\frac{a}{2}, t \right] = B \left[\frac{a}{2}, t \right] = G(t) \quad (28)$$

where $G(t)$ is a function solely of time. Applying again the Laplace transform on (35):

$$s.\beta - B(x, 0) = K \frac{d^2 \beta}{dx^2} \quad (29)$$

$$K = \frac{\rho}{\mu} \quad (30)$$

The general solution of (29) assuming $B(x,0) = 0$ (zero initial condition is:

$$\beta(x,s) = C_1 e^{\sqrt{s/k} x} + C_2 e^{-\sqrt{s/k} x} \quad (31)$$

By symmetry $\beta(x,s) = \beta(-x,s)$ therefore

$$C_1 = C_2$$

(31) can be rewritten for $x = a/2$:

$$\beta \left[\frac{a}{2}, s \right] = C \left[e^{\sqrt{s/k} \frac{a}{2}} + e^{-\sqrt{s/k} \frac{a}{2}} \right] = g(s) \quad (32)$$

where $g(s)$ is the Laplace transform of $G(t)$, the input function.

Combining (32) and (31)

$$\beta(x,s) = \frac{g(s)}{e^{\sqrt{s/k} a/2} + e^{-\sqrt{s/k} a/2}} \cdot \left[e^{\sqrt{s/k} x} + e^{-\sqrt{s/k} x} \right] \quad (33)$$

(33) can be rewritten in terms of hyperbolic functions.

$$\beta(x,s) = g(s) \frac{\cosh(\sqrt{s/k} x)}{\cosh(\sqrt{s/k} a/2)} \quad (34)$$

The unit width flux can now be evaluated in the s -plane:

$$\Phi(s) = \int_{-a/2}^{a/2} g(s) \frac{\cosh \sqrt{s/k} x}{\cosh \sqrt{s/k} a/2} dx \quad (35)$$

$$\Phi(s) = \frac{g(s)}{\sqrt{s/k} \cosh(\sqrt{s/k} a/2)} \cdot \int_{-\sqrt{s/k} a/2}^{\sqrt{s/k} a/2} \cosh u \cdot du \quad (36)$$

$$\Phi(s) = 2 \sqrt{k} g(s) \cdot \frac{\tanh(\sqrt{s/k} a/2)}{\sqrt{s}} \quad (37)$$

(37) is the generalised transfer function for the unit width flux on the finite width plate. The time domain solution for $g(s) = 1$ (impulse) is:

$$\phi_s(t) = \bar{L}^{-1}\{\phi(s)\} \wedge g(s) = 1 \wedge \frac{a}{2\sqrt{k}} = m \quad (37a)$$

$$\phi_s(t) = 2\sqrt{k} \frac{4\sqrt{k}}{a} \sum_{n=1}^{\infty} e^{-(2n-1)^2\pi^2 t/4m^2} \quad (38)$$

$$\phi_u(t) = \int_0^t \phi_s \cdot dt \quad (39)$$

By combining (38), using the linearity property of the integral and the uniform convergence of the series [K-7]:

$$\phi_u(t) = 8a \cdot \sum_{n=1}^{\infty} \frac{1}{(2n-1)^2\pi^2} \left[1 - e^{-(2n-1)^2\pi^2 t/4m^2} \right] \quad (40)$$

$$\phi_u(t) = 8a \left[\sum_{n=1}^{\infty} \frac{1}{\pi^2(2n-1)^2} - \sum_{n=1}^{\infty} \frac{e^{-(2n-1)^2\pi^2 t/4m^2}}{\pi^2(2n-1)^2} \right] \quad (41)$$

This series always converges. The first summation depends only on n , whereas the second one depends on n , primarily, then on t and m . The error, as a function of the quantity $\pi^2 t/4m^2$ and the n th order approximation of ϕ_u will be investigated.

The flux response can be considered as an infinite time constant system where the time constants are given by:

$$\tau = \frac{4m^2}{(2n-1)^2\pi^2} \quad n = 1, 2, \dots \quad (42)$$

It is therefore expected that the finite order approximation will be inadequate for very small time t .

The relative error for the approximation is:

$$\epsilon_r = \frac{\phi_u - \phi_{app}}{\phi_u}$$

$$\epsilon_r = \frac{\sum_{n=1}^{\infty} \frac{1 - e^{-(2n-1)^2 \pi^2 t / 4m^2}}{\pi^2 (2n-1)^2}}{\sum_{n=1}^{\infty} \frac{1 - e^{-(2n-1)^2 \pi^2 t / 4m^2}}{\pi^2 (2n-1)^2}} \quad (43)$$

By applying l'Hopital's rule to the quotient

$$\lim_{t \rightarrow 0} \epsilon_r = 1 \quad (44)$$

The fact that the relative error is 1 for very small t reflects the fact that the approximation does not incorporate infinitely small time constants. At the same time the absolute error is infinitely small and in that respect this mismatching for very small t does not become important. Figs. 4.4 and 4.5 illustrate the error as a function of time and the order of approximation. The steady state error is also an important consideration. If the approximating series is normalised for a zero steady state error then this new error is shown in Fig. 4.6. Although the relative error is more important for the theoretical behaviour of the approximation, the absolute error is more important from an engineering point of view. Measurements of engine-dynamometer torque (which is directly related to the flux) are known to be rather fluctuating (See Chapter 6). Transducer manufacturers give a 1% error relative to full scale measurement [V-1]. Therefore approximations of the same order are acceptable.

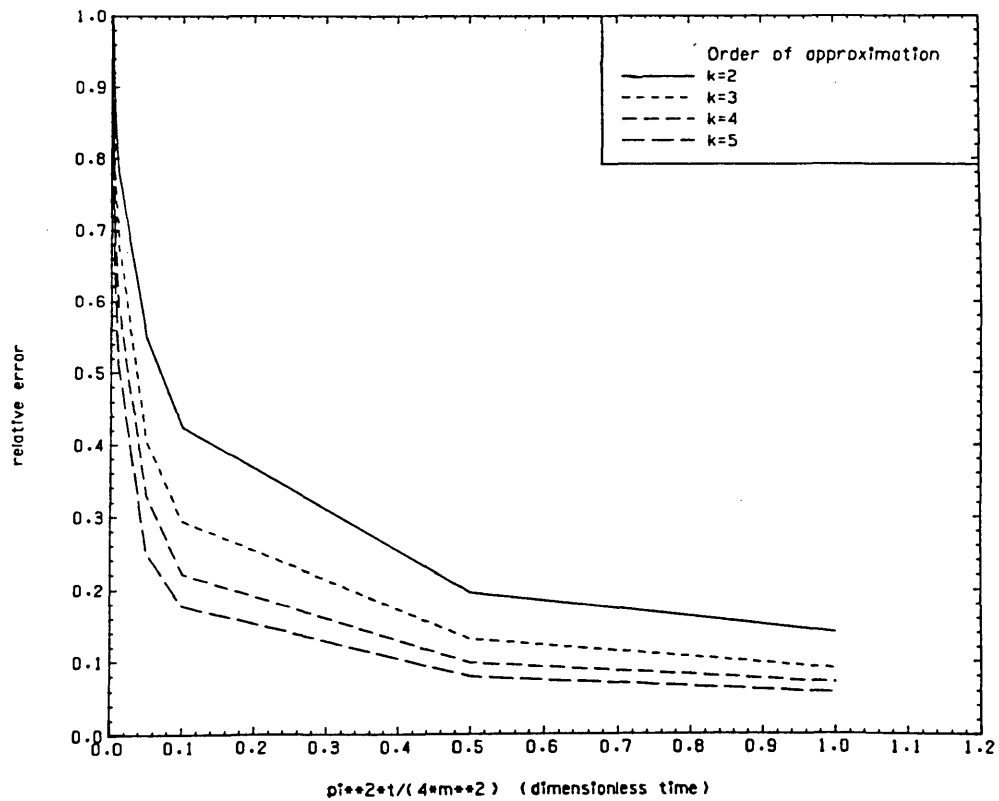


Fig. 4.4 Relative error as a function of the order of approximation and time for a step input.

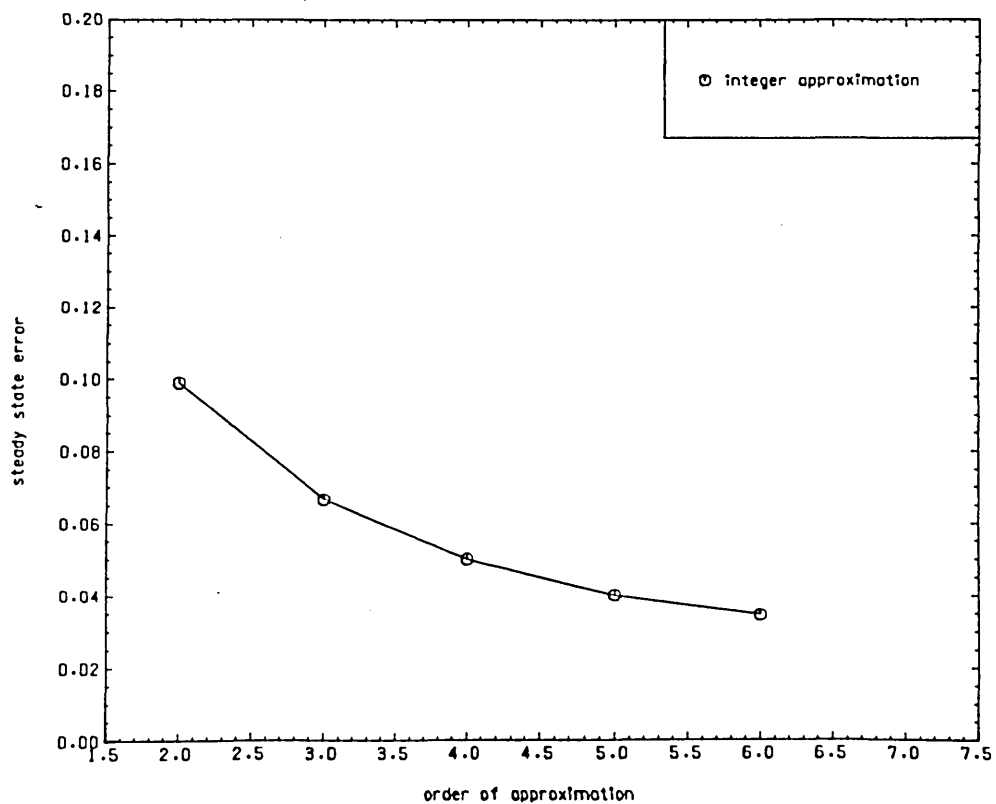


Fig. 4.5 Steady state error as a function of the order of approximation for a step input.

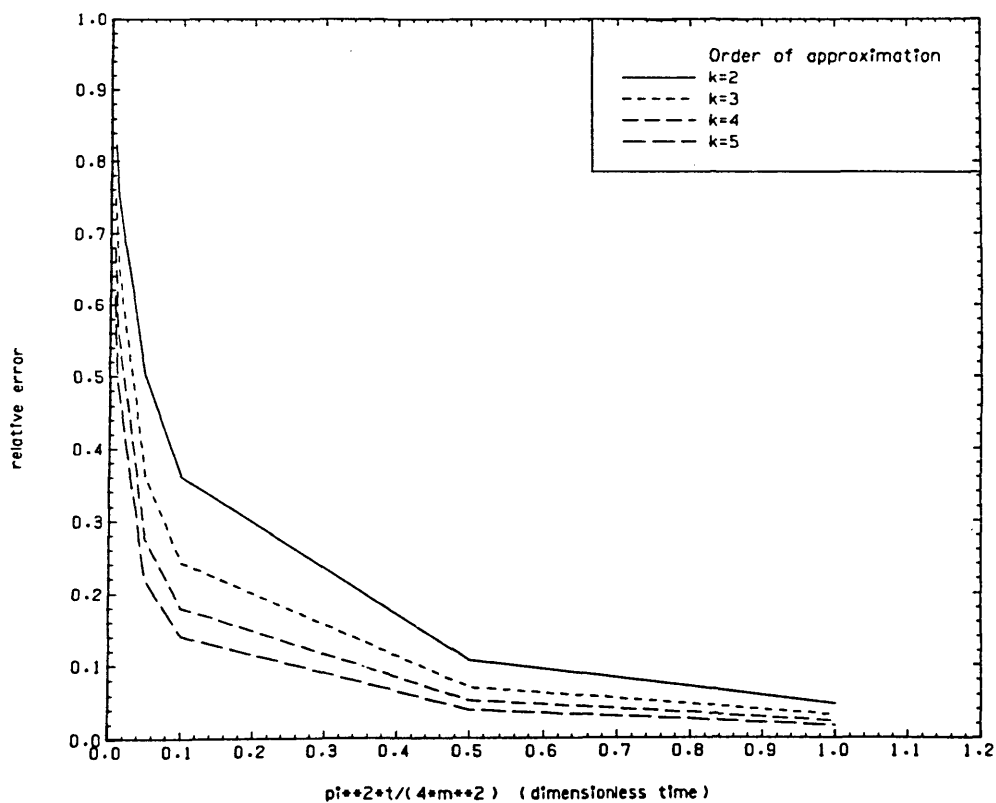


Fig. 4.6 Relative error for the normalized approximation as a function of the order of approximation and time for a step input.

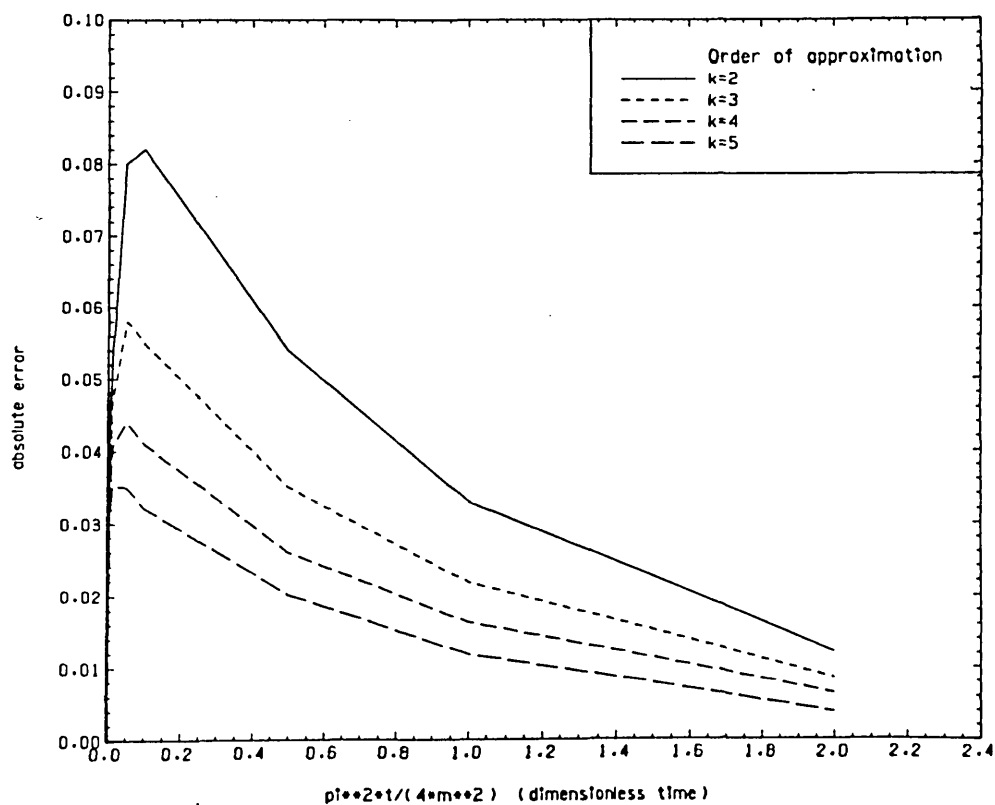


Fig. 4.7 Absolute error for the normalized approximation, for a step input.

The behaviour of the approximation has also been investigated with a ramp input:

$$\phi_r(t) = 8a \left[\sum_{n=1}^{\infty} \frac{t}{\pi^2(2n-1)^2} - \sum_{n=1}^{\infty} \frac{4m^2(1-e^{-(2n-1)^2\pi^2 t/4m^2})}{\pi^4(2n-1)^4} \right]$$

The relative error limit at $t \rightarrow 0$ becomes also 1. What is interesting is to deduce the relative steady state error for the ramp input

$$\epsilon_{rss} = \frac{\sum_{n=k+1}^{\infty} \frac{1}{(2n-1)^2}}{\sum_{n=1}^{\infty} \frac{1}{(2n-1)^2}}$$

which is the same as the step input steady state error. The normalized approximation will therefore have a zero ramp input steady state error. The derived results will help in the choice of the order of the approximating function.

4.5 The Approximating Function

The k th order approximation has been investigated with regard to the two most common inputs (step and ramp). The k th order approximation is the solution of a linear differential equation of order k . It is possible to revert to the transfer function of the approximating model by applying Laplace transforms on the k th order series for the step input:

$$\begin{aligned} \phi_{app}(t) &= \sum_{n=1}^k \frac{1}{\pi^2(2n-1)^2} - \sum_{n=1}^k \frac{e^{-(2n-1)^2\pi^2 t/4m^2}}{\pi^2(2n-1)^2} \\ L(\phi_{app}) &= \frac{1}{s} \sum_{n=1}^k \frac{1}{\pi^2(2n-1)^2} - \sum_{n=1}^k \frac{1/\pi^2(2n-1)^2}{s + (2n-1)^2\pi^2/4m^2} \end{aligned} \quad (46)$$

Bearing in mind that (46) is the transfer function multiplied by the step input $1/s$:

$$\Phi(s) = g(s) \left\{ \sum_{n=1}^k \frac{1}{\pi^2(2n-1)^2} - \sum_{n=1}^k \frac{s \cdot 1/\pi^2(2n-1)^2}{s + \pi^2(2n-1)^2/4m^2} \right\} \quad (47)$$

and by writing (47) in quotient form.

$$\Phi(s) = g(s) \left\{ \sum_{n=1}^k \frac{1}{\pi^2(2n-1)^2} \right\} \frac{\prod_{n=1}^k (s + (2n-1)^2\pi^2/4m^2) - \sum_{n=1}^k \left\{ \frac{s}{\pi^2(2n-1)^2} \prod_{j=1, j \neq n}^k (s + (2j-1)^2\pi^2/4m^2) \right\}}{\prod_{n=1}^k (s + (2n-1)^2\pi^2/4m^2)} \quad (48)$$

where $g(s)$ is the input function (in the s -plane). This k th order linear differential equation needs k initial conditions. For an optimal controller it is required to observe k inputs or be able to reconstruct them. In that respect the degree of the approximation will be limited by the number of states that can be introduced.

4.6 Linearity of the model

It will be shown that for (37) the linearity property holds. If g_1 , g_2 are two input functions then if their Laplace transform exists:

From the linearity of the Laplace operator:

$$g_1(t) = L^{-1}\{G_1(s)\} \quad (37) \quad \phi_1(t) = L^{-1}\left\{2\sqrt{k} \frac{\tanh \sqrt{s/k} \cdot a/2}{\sqrt{s}} G_1(s)\right\}$$

$$g_2(t) = L^{-1}\{G_2(s)\} \quad (37) \quad \phi_2(t) = L^{-1}\left\{2\sqrt{k} \frac{\tanh \sqrt{s/k} \cdot a/2}{\sqrt{s}} G_2(s)\right\}$$

$$g_1 + g_2 = L^{-1} \{ G_1 + G_2 \} - \phi_{12}(t) = L^{-1} \left\{ 2\sqrt{k} \frac{\tanh \sqrt{s/k} a/2}{\sqrt{s}} (G_1 + G_2) \right\}$$

$$\phi_{12}(t) = L^{-1} \left\{ 2\sqrt{k} \frac{\tanh \sqrt{s/k} a/2}{\sqrt{s}} G_1 \right\} + L^{-1} \left\{ 2\sqrt{k} \frac{\tanh \sqrt{s/k} a/2}{\sqrt{s}} G_2 \right\}$$

$$\phi_{12}(t) = \phi_1(t) + \phi_2(t)$$

which proves the linearity condition.

Linearity holds mainly because a linear relation has been assumed between B and H. This assumption is true when operation is far from the saturated region. Bearing in mind that dynamometers are air-gap machines saturation is further prevented.

Another assumption relates to the distributed nature of the system in question. What is of interest is the flux, but this flux is the integral of the flux density vector which, in general, is not uniform. When equation (31) has been solved with the assumption of a zero initial condition (or constant) the effect of the current shape of the flux density distribution on future behaviour has been hidden by assuming a uniform distribution. This assumption cannot be overcome unless the system is deliberately treated as a distributed one. The complexity introduced is not justified for the practical realisation of a real-time controller.

4.7 Evaluation of Model Behaviour and Degree of Approximation

Many eddy current dynamometers do not have a layout allowing a precise definition of the magnetic paths. However, there is always the dominant path which presents the dominant time constant of the system. To discover this path is not thought to be particularly difficult and for the case of the dynamometers at Imperial College such a path is reasonably clear (Fig. 4.8). The dominant dimension appears to be the width between the magnetising coils. Considering

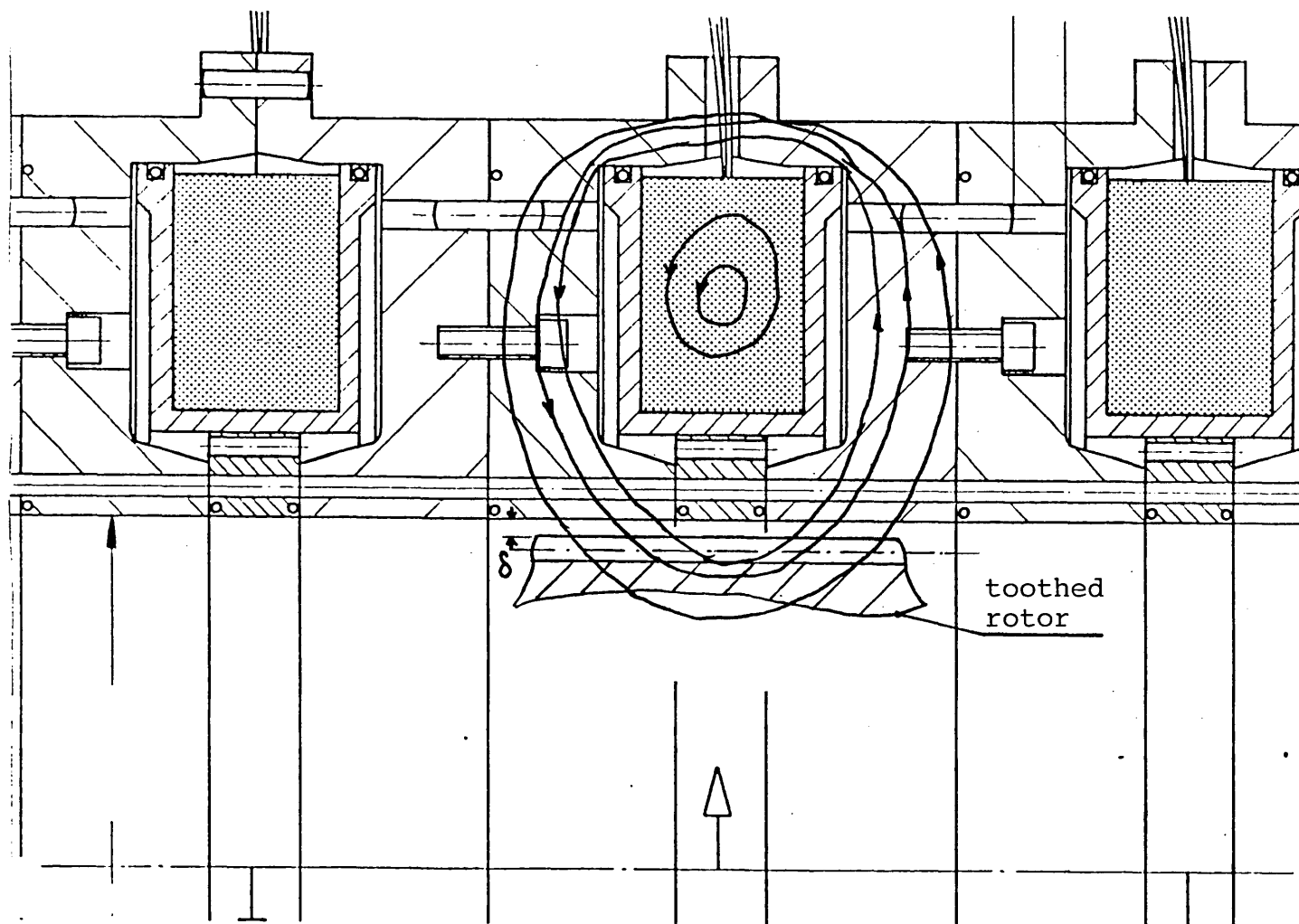


Fig. 4.8 Eddy current dynamometer cross sections and flux line indication

the idealised magnetic circuit of Fig. 4.9 to incorporate the effect of the air gap in the circuit, it is possible to deduce the corrected relative permeability [F-2].

$$\mu_{re} = \frac{\mu_r}{1 + \frac{\mu_r \cdot \delta}{l}}$$

It is now possible to estimate the basic model parameter m , given by equation (37a). Other required data for the estimation are:

- (1) the equivalent plate width (a)
- (2) the half length of the magnetic circuit (l)
- (3) the rotor-stator air gap (δ)
- (4) the relative permeability of the magnetic material (μ_r)
- (5) the resistivity of the magnetic material (ρ)

From (37a) and (49)

$$m = \frac{a}{2} \sqrt{\frac{\mu_r \cdot \mu_0}{\rho(1 + \frac{\mu_r \cdot \delta}{l})}} \quad (50)$$

The results for two types of dynamometers are shown in Table 4.1.

Assuming that the flux not contributing to torque generation is small in a dynamometer then the instantaneous torque is directly related to the instantaneous flux. It is therefore possible to monitor a step torque response and deduce the flux response based on steady state characteristics, between torque and flux (again here there is an implicit assumption about the effects of the non-uniformity of the flux density). Such tests have been carried out and the flux response is plotted together with the different order approximations. (Fig. 4.10 and Fig. 4.11). The results are in good agreement with predictions though in the low end the accuracy of the measurements is questionable. The different order approximations coincide after about 1.5 seconds. The current step input was

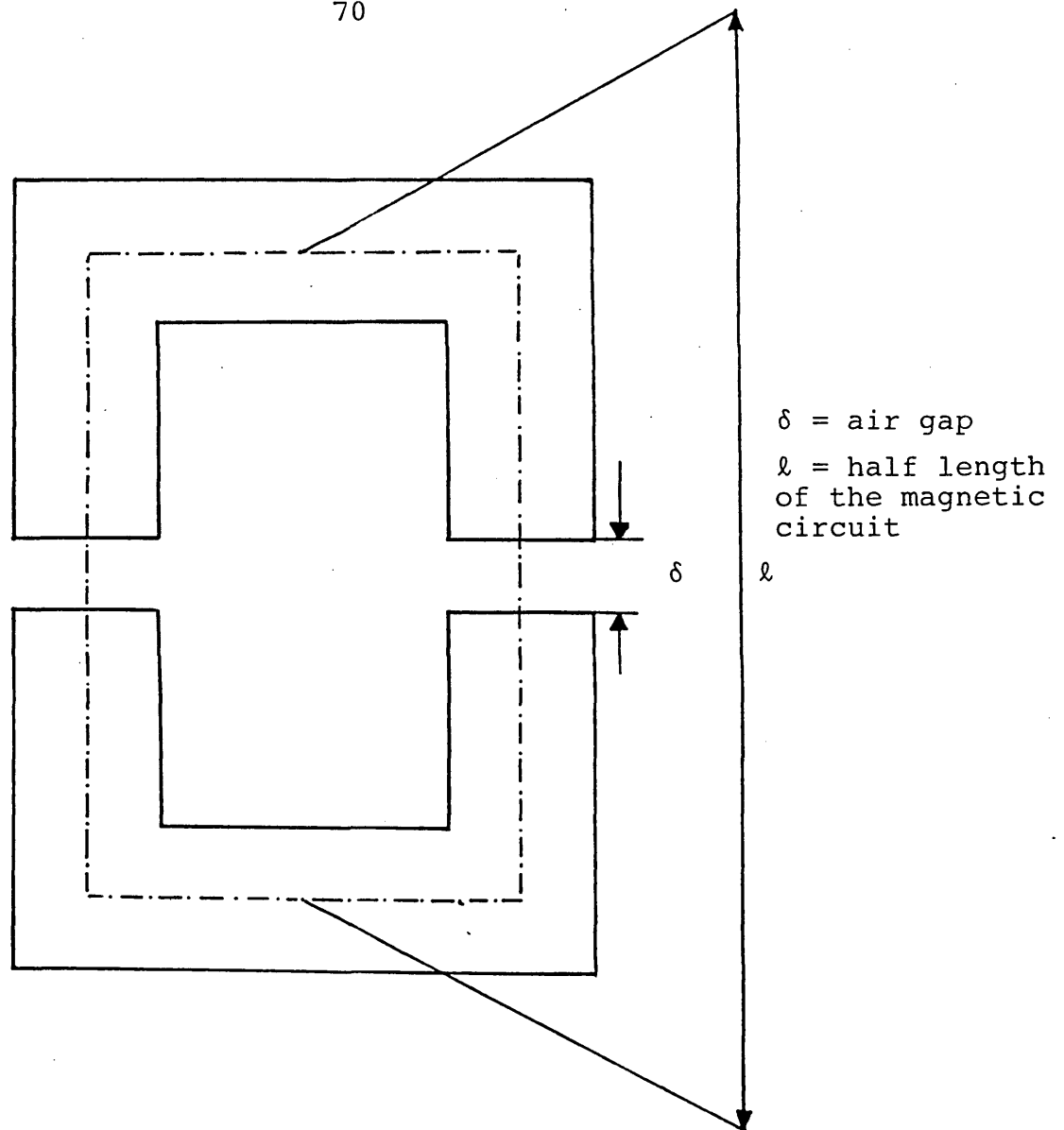


Fig. 4.9 The idealized magnetic circuit

	a	μ_0	δ	ℓ	μ_r	ρ	m
WB15	.07	$4\pi 10^{-7}$	10^{-3}	.25	1500	$5 \cdot 10^{-8}$	2.57
WB25	0.09	$4\pi 10^{-7}$	$1.5 \cdot 10^{-3}$	.4	1500	$5 \cdot 10^{-8}$	3.39

Table 4.1 m-parameter derivation for mild steel bodied dynamometers.

realised by a high voltage current controller and the estimated build-up time was of the order of 20ms. It is therefore considered as an adequate step simulation when objects of much larger time constant are investigated. As far as model accuracy is concerned even the second order approximation ($k=2$) predicts real behaviour closely and in that respect is adequate as an approximating function.

Investigating the problem from a control point of view the order of approximation will lead to particular types of controllers. The first order approximation is nothing more than a first order lag. The optimal controller for this configuration will be bang-bang and the steady state accuracy will depend on the limit cycle which is a function of the sampling time [T-3]. This sampling time cannot decrease below 20ms because the torque measurement is integrated within that period to reduce signal scatter (due to vibration etc.) and the microprocessor used as a controller cannot operate much faster bearing in mind its other tasks. The first order model implies a system without numerator dynamics and this implication removes much of the usefulness of the derived model.

The second order model is much more representative of the real behaviour incorporating the effects of lag, double time constant and derivative 'kick'. The time-optimal controller for this model leads to a dual-mode control which for large settings will operate first in its bang-bang mode and then when the states are sufficiently small (within the target set) will enter a pseudo-singular mode [A-3, A-9, S-1] and remain in this mode when reaching the target.

To implement the optimal control law for the preceding problem, requires a computer to store the characteristics of the switching hypersurface, to decide when to switch from $u = \max$ to $u = \min$ and vice versa. On reaching the target the control is a state feedback one and ensures improved steady state smoothness relative to the

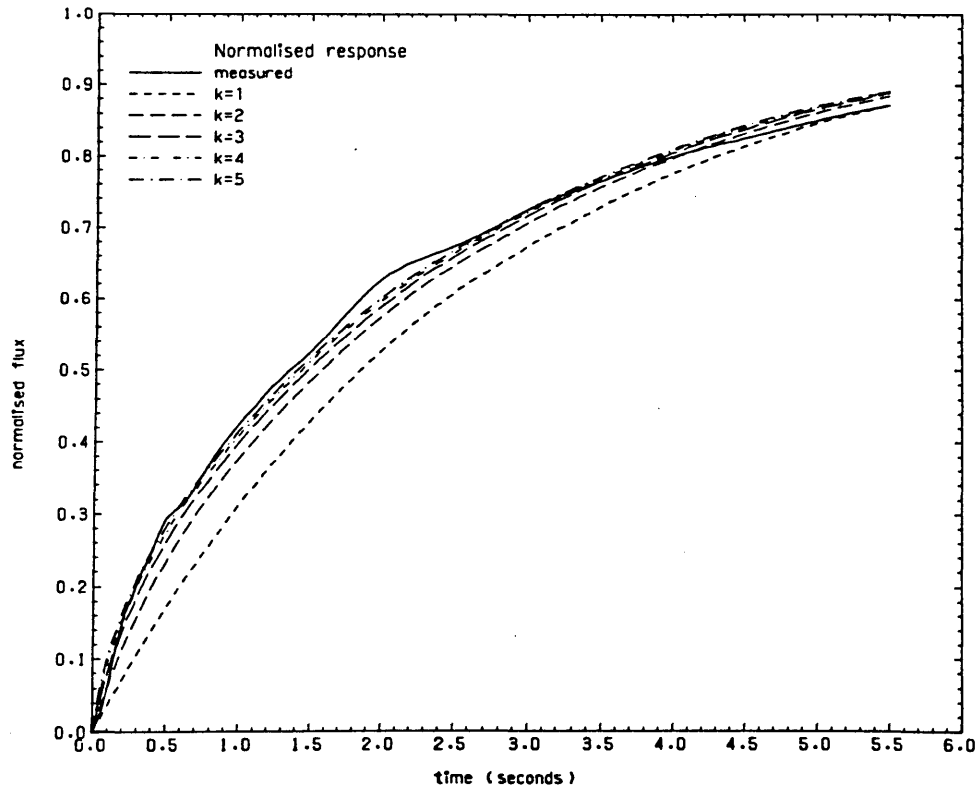


Fig. 4.11 Step flux response on the 3WB15 dynamometer.

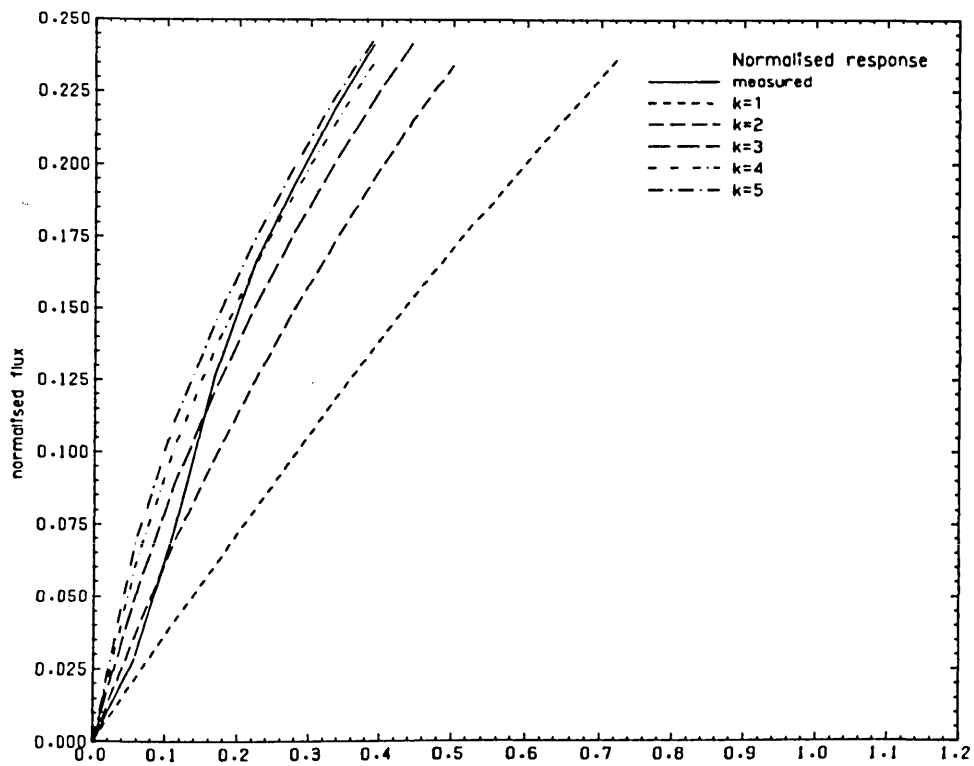


Fig. 4.10 Step flux response on the 3WB15 dynamometer.

ON-OFF control achieved by the first order controller.

The complexity of the second order controller is much greater than the first order one. Increasing the order to three would make the controller even more complex and an order of four or more would lead to transcendental equations for the switching hypersurface [A-9]. The fundamental benefit of incorporating a state feedback control near the target is achieved with the second order model and it is the one used herein.

4.8 Time-optimal control of the second order model

The second order model can be written as a second order differential equation of the flux:

$$\frac{d^2\phi}{dt^2} - (\lambda_1 + \lambda_2) \frac{d\phi}{dt} + \lambda_1 \lambda_2 \phi = Ka \frac{du}{dt} + Ku \quad (51)$$

where ϕ is the flux

$$a = \frac{5\pi^2}{4m^2} > 0 \quad (\text{numerator zero})$$

$$\lambda_1 = -\frac{\pi^2}{4m^2} < 0 \quad (\text{first pole})$$

$$\lambda_2 = -\frac{9\pi^2}{4m^2} < 0 \quad (\text{second pole})$$

$$K = \lambda_1 \lambda_2 \quad (\text{gain})$$

u = control input

and m given by (50).

To simplify the solution procedure the flux will be substituted by the error to reach the target so that the new target can be the origin.

$$\ddot{y} - (\lambda_1 + \lambda_2) \dot{y} + \lambda_1 \lambda_2 y = K a \dot{u}^* + K u^* \quad (52)$$

$y = \phi_T - \phi$ where ϕ_T is the target flux.

$u^* = -u + m$ where $m = u_T$, u_T is target steady state control.

The control problem can be stated now formally:

Given the system with output y and control u^* related by the 2nd order time invariant linear differential equation (52) find the control $u^*(t)$ such that if $G \subset \mathbb{R}^2$ is a target set for the final state

$$G = \{(y, \dot{y}) : y \in Y_T, \dot{y} \in \dot{Y}_T\}, \quad Y_T, \dot{Y}_T \text{ are target intervals}$$

and if the system is at an arbitrary point in the state space $(y_a, \dot{y}_a)$ the cost functional

$$J = \int_0^T dt \quad (T \text{ is free and } (y_T, \dot{y}_T) \in G)$$

is minimised subject to the constraint

$$-m_1 \leq u^* \leq m_2 \quad m_1, m_2 > 0$$

Equation (52) will be further transformed to reach the canonical form.

The assumed target set is the origin $(0, 0)$.

The new variables z_1, z_2 are defined as follows:

$$z_1 = y \quad (53)$$

$$z_2 = \dot{y} - K a u^* \quad (54)$$

The initial target set (origin) is mapped by (53), (54) into

$$G_Z = \{(Z_1, Z_2) : Z_1 = 0, Z_2 \in [-K\alpha_2, K\alpha_1]\} \quad (55)$$

The differential equation for z_1, z_2 by substitution of (53), (54) in (52) is in matrix form:

$$\begin{bmatrix} \dot{Z}_1 \\ \dot{Z}_2 \end{bmatrix} = \begin{bmatrix} 0 & 1 \\ -\lambda_1 \lambda_2 & \lambda_1 + \lambda_2 \end{bmatrix} \begin{bmatrix} Z_1 \\ Z_2 \end{bmatrix} + \begin{bmatrix} K\alpha \\ K\alpha (\lambda_1 + \lambda_2) + K \end{bmatrix} u^* \quad (57)$$

(57) will be further transformed into the canonical form. If

$$A = \begin{bmatrix} 0 & 1 \\ -\lambda_1 \lambda_2 & \lambda_1 + \lambda_2 \end{bmatrix}$$

then the eigenvalues of A are λ_1, λ_2 .

If Λ denotes the diagonal matrix

$$\Lambda = \begin{bmatrix} \lambda_1 & 0 \\ 0 & \lambda_2 \end{bmatrix}$$

then there exists a non singular matrix P ,

(known also as Vandermonde matrix) with inverse

$$P^{-1} = \frac{1}{\lambda_2 - \lambda_1} \begin{bmatrix} \lambda_2 & -1 \\ -\lambda_1 & 1 \end{bmatrix}$$

such that $\Lambda = P^{-1} A P$

By using the similarity transformation

$$X = P^{-1} Z$$

then the components x_1, x_2 can be expressed as

$$x_1 = \frac{1}{\lambda_2 - \lambda_1} (\lambda_2 Z_1 - Z_2) \quad (58)$$

$$x_2 = \frac{1}{\lambda_2 - \lambda_1} (-\lambda_1 Z_1 + Z_2) \quad (59)$$

and (57) is mapped as

$$\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \end{bmatrix} = \begin{bmatrix} \lambda_1 & 0 \\ 0 & \lambda_2 \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} + \frac{K}{\lambda_2 - \lambda_1} \begin{bmatrix} -1 & -a\lambda_1 \\ 1 & +a\lambda_2 \end{bmatrix} u^* \quad (60)$$

and the target set (56) is mapped into G_X :

$$G_X = \left\{ (x_1, x_2) : x_1 \in \left[-\frac{Kam_2}{\lambda_1 - \lambda_2}, \frac{Kam_1}{\lambda_1 - \lambda_2} \right], \right. \\ \left. x_2 \in \left[-\frac{Kam_1}{\lambda_1 - \lambda_2}, \frac{Kam_2}{\lambda_1 - \lambda_2} \right] \right\} \quad (61)$$

The system described by (60) is controllable and normal [A-9] therefore the requirements of Theorem 6-2 in [A-9] are satisfied and the time-optimal control to reach the target is Bang-Bang:

$$u^*(t) = -\operatorname{sgn} \{B^T [x^*(t), t] \cdot p^*(t)\}$$

where u^* is the time optimal control, x^* , p^* are the corresponding state trajectory and costate, and B the gain matrix.

When the target is reached then by substituting (53), (54) in (58) the control is defined by:

$$u^* = \frac{x_1 (\lambda_2 - \lambda_1)}{Ka} \quad (62)$$

To determine the optimal control it is necessary to find the switching curve i.e. the set of states (x_1, x_2) such that the target set G_X is reached with a single control ($u^* = \text{maximum}$ or $u^* = \text{minimum}$). This requires the solution of the differential equations defined by (60).

$$\dot{x}_1 = \lambda_1 x_1 + \frac{K(1 + a\lambda_1)}{(\lambda_1 - \lambda_2)} u^* \quad (63)$$

$$\dot{x}_2 = \lambda_2 x_2 - \frac{K(1 + a\lambda_2)}{(\lambda_1 - \lambda_2)} u^* \quad (64)$$

The solution for (63), (64) is

$$x_1(t) = x_1(0) e^{\lambda_1 t} + \int_0^t e^{\lambda_1(t-\tau)} \frac{K(1+a\lambda_1)}{\lambda_1 - \lambda_2} u^* d\tau \quad (65)$$

$$x_2(t) = x_2(0) e^{\lambda_2 t} + \int_0^t e^{\lambda_2(t-\tau)} \frac{K(1+a\lambda_2)}{\lambda_1 - \lambda_2} u^* d\tau \quad (66)$$

and if $x_1(0) = \xi_1$, $x_2(0) = \xi_2$ are the initial states

$$x_1 = \left\{ \xi_1 + \frac{K(1 + a\lambda_1)}{\lambda_1 (\lambda_1 - \lambda_2)} u^* \right\} e^{\lambda_1 t} - \frac{K(1 + a\lambda_1)}{\lambda_1 (\lambda_1 - \lambda_2)} u^* \quad (67)$$

$$x_2 = \left\{ \xi_2 - \frac{K(1 + a\lambda_2)}{\lambda_2 (\lambda_1 - \lambda_2)} u^* \right\} e^{\lambda_2 t} - \frac{K(1 + a\lambda_2)}{\lambda_2 (\lambda_1 - \lambda_2)} u^* \quad (68)$$

By eliminating the time t from (67), (68)

$$\left[\frac{x_1 + \frac{K(1 + a\lambda_1)u^*}{\lambda_1(\lambda_1 - \lambda_2)}}{\xi_1 + \frac{K(1 + a\lambda_1)u^*}{\lambda_1(\lambda_1 - \lambda_2)}} \right]^{\lambda_2/\lambda_1} = \frac{x_2 - \frac{K(1+a\lambda_2)u^*}{\lambda_2(\lambda_1-\lambda_2)}}{\xi_2 - \frac{K(1 + a\lambda_2)u^*}{\lambda_2(\lambda_1 - \lambda_2)}} \quad (69)$$

The switching curves are now defined as

$$\gamma^- = \left\{ (\xi_1, \xi_2) : \xi_1 > \frac{Kam_1}{\lambda_1 - \lambda_2} \text{ and } \right.$$

$$\left. \xi_2 = \frac{Km_2}{\lambda_2(\lambda_1 - \lambda_2)} + \frac{Kam_2}{\lambda_1 - \lambda_2} + \left[\frac{\xi_1 + \frac{Km_2}{\lambda_1(\lambda_1 - \lambda_2)} + \frac{Kam_2}{\lambda_1 - \lambda_2}}{\frac{Km_2}{\lambda_1(\lambda_1 - \lambda_2)}} \right]^{\lambda_2/\lambda_1} \left(-\frac{Km_2}{\lambda_2(\lambda_1 - \lambda_2)} \right) \right\}$$

$$\gamma^+ = \left\{ (\xi_1, \xi_2) : \xi_1 < \frac{-Kam_2}{\lambda_1 - \lambda_2} \text{ and } \right.$$

$$\left. \xi_2 = \frac{-Km_1}{\lambda_2(\lambda_1 - \lambda_2)} - \frac{Kam_1}{\lambda_1 - \lambda_2} + \left[\frac{\xi_1 - \frac{Km_1}{\lambda_1(\lambda_1 - \lambda_2)} - \frac{Kam_1}{\lambda_1 - \lambda_2}}{-\frac{Km_1}{\lambda_1(\lambda_1 - \lambda_2)}} \right]^{\lambda_2/\lambda_1} \frac{Km_1}{\lambda_2(\lambda_1 - \lambda_2)} \right\}$$

The switching curves together with the target set G_X are shown in Fig. 4.12.

The preceding analysis leads to the following control law:

$$\begin{aligned} u^* &= m_2 \text{ if } (x_1, x_2) \in \gamma^- \cup R^- \\ u^* &= m_1 \text{ if } (x_1, x_2) \in \gamma^+ \cup R^+ \\ u^* &= x_1(\lambda_2 - \lambda_1)/Ka \text{ if } (x_1, x_2) \in G_X \end{aligned}$$

where x_1, x_2 are the states of the system described by (60); γ^- and γ^+ are the minimum and maximum switching curves respectively and R^-, R^+ are the sets defined by:

$$\begin{aligned} R^- &= \{(x_1, x_2) : \text{if } (x_1, x_2') \in \gamma \text{ then } x_2 < x_2' \\ &\quad \text{and } (x_1, x_2) \in G_X\} \\ R^+ &= \{(x_1, x_2) : \text{if } (x_1, x_2') \in \gamma \text{ then } x_2 > x_2' \\ &\quad \text{and } (x_1, x_2) \in G_X\} \\ \gamma &= \gamma^- \cup \gamma^+ \end{aligned}$$

4.9 The Equivalent Tracking Problem

The solution of the complete tracking problem cannot be easily reduced to a feedback controller form. Though it is possible to adapt the tracking problem to the optimal controller of section 4.8 for a class of certain trajectories.

Rewriting (51)

$$\ddot{y} - (\lambda_1 + \lambda_2) \dot{y} + \lambda_1 \lambda_2 y = Ka \dot{u} + Ku \quad (70)$$

where y is the output state and u the control let τ be the class of trajectories satisfying the following linear differential equation:

$$\dot{\tau} - (\lambda_1 + \lambda_2) \tau + \lambda_1 \lambda_2 \tau = \tau_0 \quad \tau_0 \in R \quad (71)$$

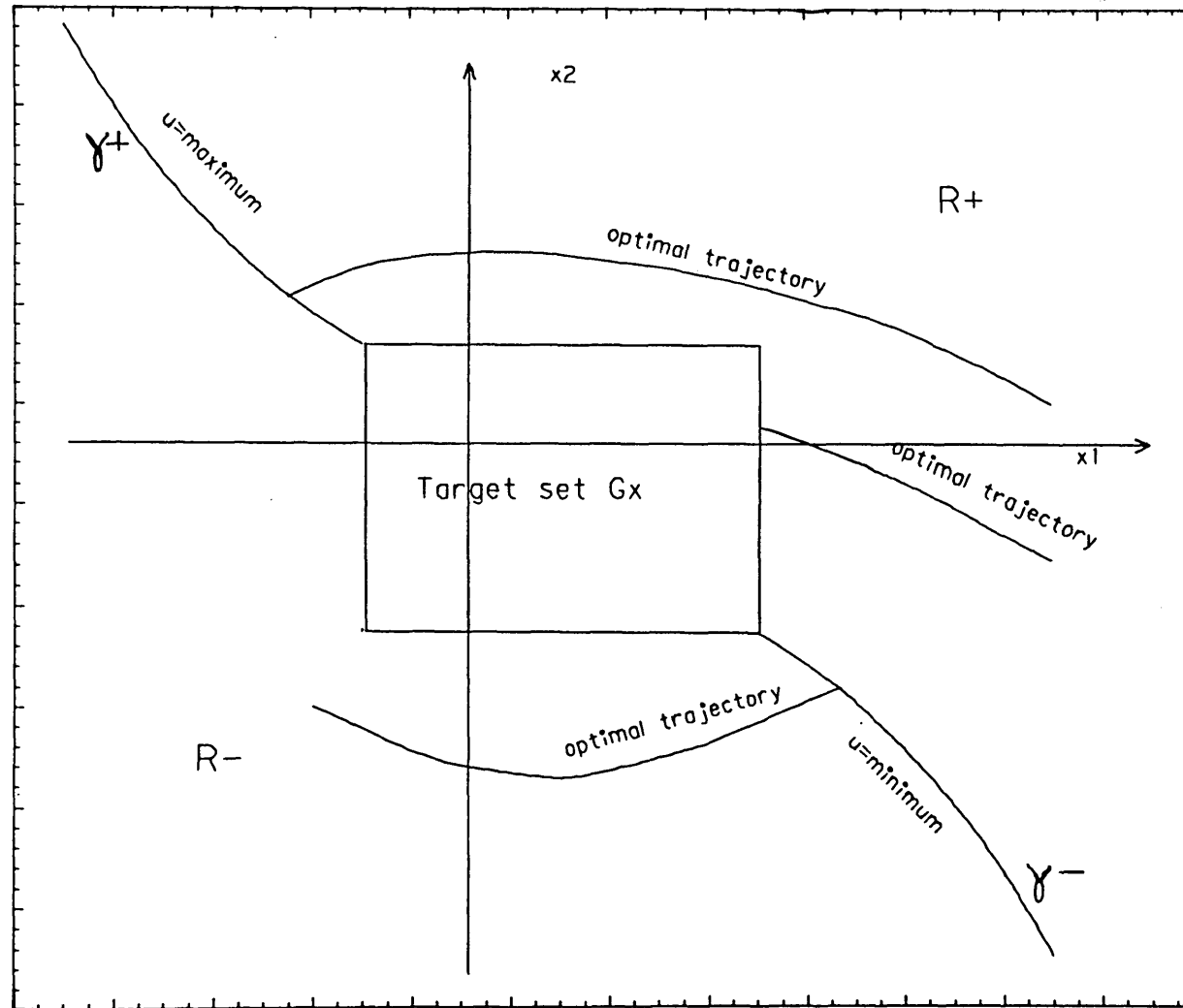


Fig. 4.12 State space indicating the optimal trajectories and switching curves.

By transforming

$$e = \tau - y$$

(71) and (72) can be written as:

$$\ddot{e} - (\lambda_1 + \lambda_2)\dot{e} + \lambda_1\lambda_2 e = -K\dot{u} - Ku + \tau_0 \quad (72)$$

If the transformation

$$u = -u^* + m \quad (73)$$

is effected in (72) and m is chosen as:

$$m = \frac{\tau_0}{K} \quad (74)$$

then (72) is equivalent to

$$\ddot{e} - (\lambda_1 + \lambda_2)\dot{e} + \lambda_1\lambda_2 e = K\dot{u}^* + Ku^* \quad (75)$$

The meaning of (75) is that the error of the trajectory from the output obeys equation (75) for which the optimal control has been derived and therefore a controller implementing the control law of section 4.8 will track the class of trajectories described by (71) in minimum time.

This class of trajectories is restricted but it is possible to approximate any trajectory over a small interval by a least squares fit of a curve described by (71). If this procedure is effected in advance (off-line) and the T_0 constants stored by the controller then piecewise tracking can be achieved. A computer simulation is shown in fig 4.13 where the desired trajectory is shown together with the

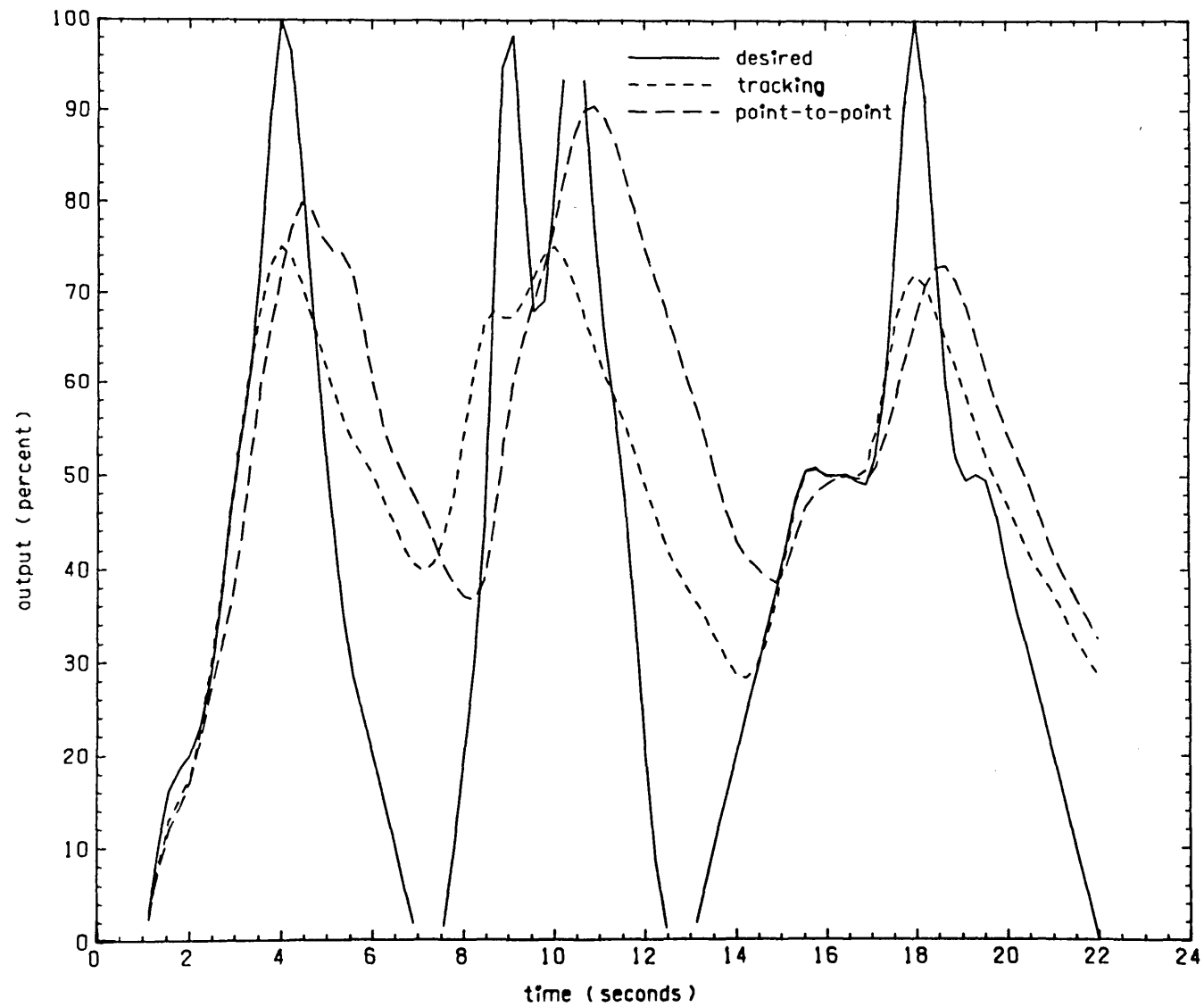


Fig. 4.13 Computer simulation of the tracking and point-to-point optimal controllers for a second order process.

ones achieved by the tracking and non tracking controllers. Experiments confirmed the superiority of the tracking controller and are shown in a subsequent chapter.

CHAPTER 5

The Software Control of Stepping Motors

5.1 Advantages of Software Control

Stepping motors can be easily controlled by digital electronics as the motor power driver responds to voltage pulses to increment position. Direction is also a digital signal indicating clockwise or counterclockwise rotation.

Stepping motors require controlled acceleration and deceleration for best use of their capability as their torque-speed characteristics together with the inertia of the load do not allow for high start/stop rates and in any case maximum speed is well above the start/stop rate. A technique called motor-clock ramping is used to minimise movement duration. In ramping, the frequency of the control pulses is varied to accelerate or decelerate the motor from zero to maximum speed.

A conventional hardware accelerator is shown in Fig. 5.1. The controlling device loads the step counter with the number of steps required to move the load over a prescribed distance. The counter decrements for each clock pulse applied to the motor. The motor stops when the counter decrements to zero. During the move the acceleration/deceleration controller monitors the number of steps remaining in the counter and determines how long acceleration should continue and when deceleration should begin. The acceleration/deceleration controller provides the input for the ramping oscillator which generates pulses controlling motor stepping rate. Although an exponential velocity ramp is considered ideal for many applications (demonstrated in the following sections) the ramping circuit generates simpler quasilinear curves so that hardware

complexity is minimized [L-1]. The logic gate controls the ramping oscillator by passing clock pulses to the motor and decrementing the counter. When the counter reaches zero, the control logic blocks further pulses to the motor. One of the drawbacks of the hardwired ramping system is that the acceleration/deceleration controller and ramping oscillator are difficult to synchronise. Consequently the motor may not stop at the required step rate and may overshoot. To avoid such a possibility, conventional acceleration/deceleration controllers produce a fine period of very slow motor speed just prior to the last step, that is within the motor's start/stop rate (Fig. 5.2). The pedestal technique requires motor operation at the slowest rate, which is less than ideal for time-critical applications.

A microprocessor ramping program can replace all custom-designed circuitry except the phase sequencer and power driver. Software generated pulses for the phase sequencer can be supplied directly from microprocessor output ports. A microprocessor register can replace the step counter and a memory table can provide the optimal acceleration/deceleration values to replace the acceleration/deceleration controller and ramping oscillator. The software control allows for the optimal velocity profile to be applied instead of the compromised one dictated by hardwired control, and ageing of components does not modify proper operation.

5.2 Open Versus Closed Loop

The controllers in Fig. 5.1 are both open loop. In conventional control theory and practice, open loop is associated with inaccurate operation, inability to cope with disturbances, component ageing and exogenous factors in general. Such shortcomings would render open loop control unacceptable for the positional requirements of engine test-beds.

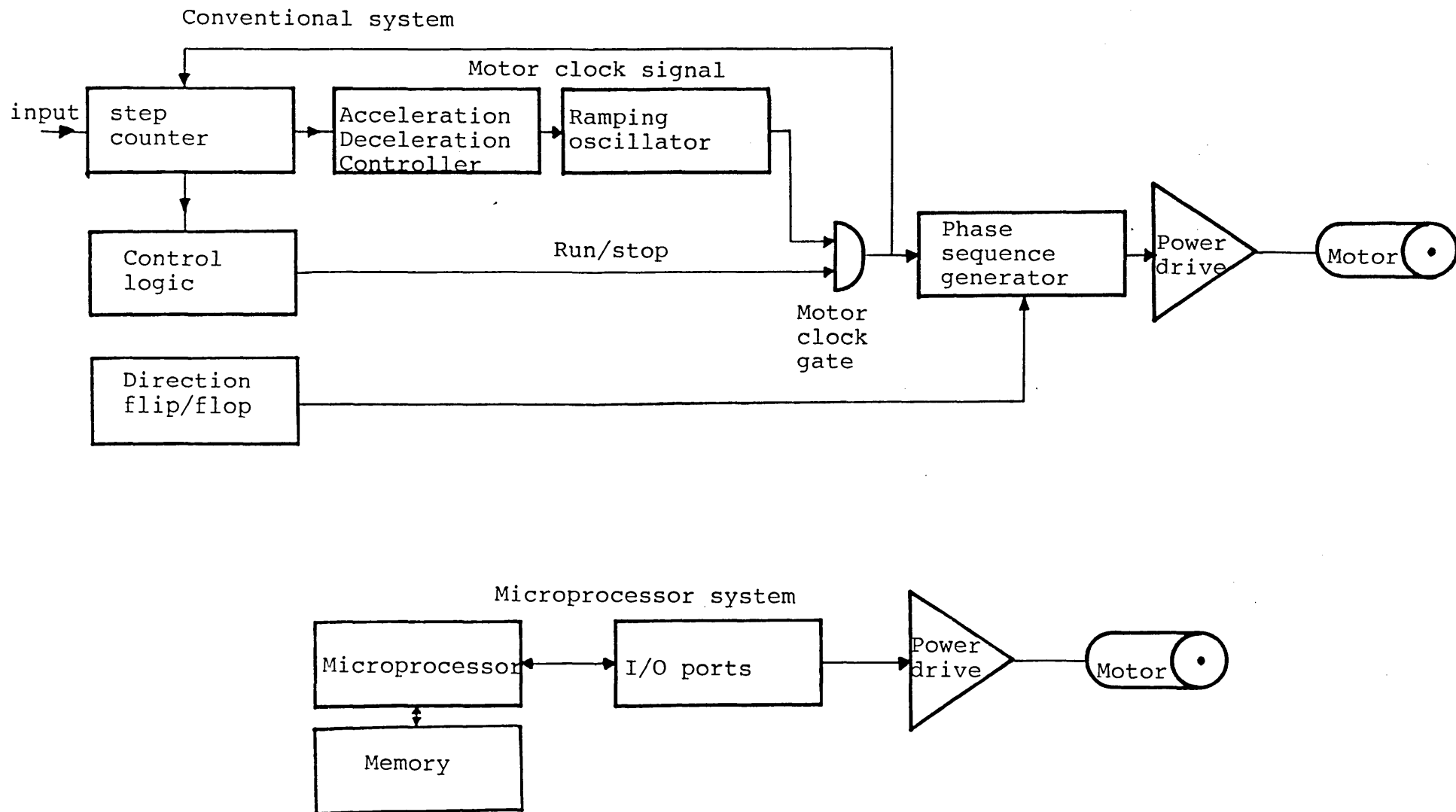


Fig. 5.1 Conventional and Microprocessor accelerator.

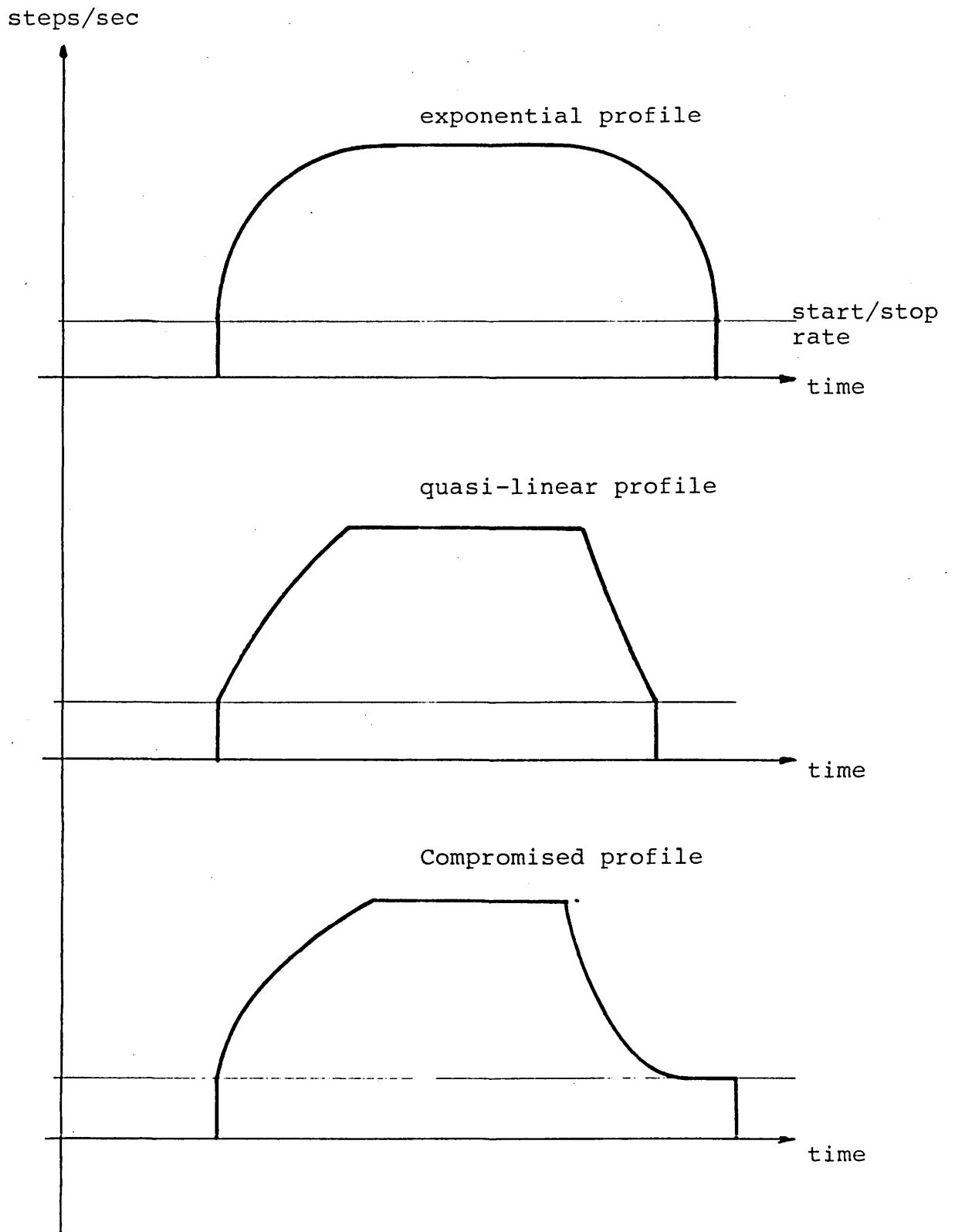


Fig. 5.2 Acceleration/deceleration profiles.

Stepping motors have some operational characteristics that under certain circumstances eliminate open loop control disadvantages. External disturbances in the form of torque disturbances, can be contained if they do not exceed the holding torque threshold. Component ageing is of no importance because of the digital nature of the response. Such ageing might alter the holding torque threshold but what is then altered is the disturbance rejection characteristic and not positional accuracy.

Open loop control can therefore be acceptable if a high disturbance rejection is kept.

This is achieved when:

- (1) the motor does not exceed a certain speed that will render disturbance rejection (reserve torque) low.
- (2) a stepping motor with high holding torque is selected with, consequently, large time constants.

A closed loop system could accelerate the stepping motor to the limit of its capability as the positional error would not be given by step counting (Fig 5.1) but through the position feedback measurement. As Fredriksen notes [F-10] the open loop control gives a smoother maximum speed operation than the closed loop when the target speed is kept constant. However, the closed loop gives more secure and faster positional control.

The choice between open and closed loop is a cost versus function decision. If the motor is removed from the open loop system, the controls still execute and the computer will never know the difference. If the motor is stalled in the closed loop system during an operation and held for a period of time, then released, the motor will continue and complete the objective without error. Hence, the cost of failure in the process will, in most cases, dictate whether to use open or closed loop systems. In engine testing there are always

mechanical limiters and the cost of failure will merely amount to an abortive test. Open loop control can readily be used and the appropriate thresholds will be determined in each case by trial and error on the particular test-bed, to ensure reliable operation.

5.3 The Time-Optimal Control of Stepping Motors

The full benefit of software control of stepping motors cannot be achieved unless the optimal acceleration/deceleration rates are determined.

However the true optimal rates can only be found if there is complete knowledge about the overall motor-load system with its damping, inertia and load torque. When the load displacement is linear, inertia reduced to the axis of rotation will be variable. Under these circumstances the determination of optimal trajectories becomes difficult, to determine analytically. Therefore the simplified control problem will be investigated, and the shape of the optimal rates will be determined with constant inertia. Once the shape is known it is possible to determine the near-optimal trajectories of the real system experimentally.

Let I be the total system Inertia reduced to the stepping motor axis of rotation. Let T_a be the maximum torque available for acceleration, and $\omega_{\max}$ the maximum speed the motor can reach without loss of steps. Let ζ be the system damping coefficient.

$$\dot{\omega}I + \omega\zeta = T_a \quad (76)$$

By substituting

$$\dot{S} = \omega.R \quad (77)$$

Where S is the distance and R the reference radius, (76) becomes:

$$\frac{I}{R} \ddot{S} + \frac{\zeta}{R} \dot{S} = T_a \quad (78)$$

By setting:

$$x_1 = S$$

$$x_2 = \dot{S}$$

(78) is transformed into the system

$$\dot{x}_1 = x_2$$

$$\dot{x}_2 + \zeta/I x_2 = u(t) \quad (79)$$

The control problem can now be stated as:

Given the system described by (79), with output x_1 and control u , find the control u^* such that if G is the final state ($G = (0,0)$) and if the system is at an arbitrary point (x_1, x_2) in the state space, the cost functional

$$J = \int_0^T dt \quad (T \text{ is free})$$

is minimised subject to the constraints

$$|u(t)| \leq T_a \cdot R/I \quad (80)$$

$$|x_2| \leq R \cdot \omega_{\max} \quad (81)$$

The system Hamiltonian is

$$H = 1 + x_2 p_1(t) - \beta x_2 p_2(t) \quad (82)$$

where $\beta = \zeta/I$, p_1, p_2 the two costate variables. From (82) it is obvious that the H-minimal control is given by:

$$u(t) = \text{sgn} \{p_2(t)\} \cdot \Delta = \pm \Delta = \pm \frac{T a \cdot R}{I} \quad (83)$$

The costate variables satisfy the equations:

$$\dot{p}_1 = - \frac{\partial H}{\partial x_1} = 0 \quad (84)$$

$$\dot{p}_2 = - \frac{\partial H}{\partial x_2} = - p_1 + \beta p_2 \quad (85)$$

If $p_1(0) = \Pi_1$ then from (84) $p_1(t) = \Pi_1$ and (85) can be written as:

$$\dot{p}_2 = - \Pi_1 + \beta p_2 \quad (86)$$

If $p_2(0) = \Pi_2$ then from (86)

$$p_2(t) = \frac{\Pi_1}{\beta} (1 - e^{\beta t}) + \Pi_2 e^{\beta t}$$

$$p_2(t) = \frac{\Pi_1}{\beta} + \left[\Pi_2 - \frac{\Pi_1}{\beta} \right] e^{\beta t} \quad (87)$$

According to Π_1 , Π_2 values, p_2 might be always positive or always negative or have at most one root (change of sign).

The minimisation of the Hamiltonian is only a necessary condition as it does not take into account the state of constraint (81). Omission of this constraint, at present, leads to the following locus of trajectories:

$$\xi_1 = \frac{\Delta}{\beta^2} \ln \frac{\Delta/\beta}{\Delta/\beta - \xi_2} - \frac{\xi_2}{\beta} \quad (88)$$

Where ξ_1, ξ_2 are the states driven to or from the origin by the control Δ . It can be seen that

$$\begin{aligned} \beta > 0 \quad \Delta > 0 \quad \lim_{\xi_2 \rightarrow -(\Delta/\beta)-} \xi_1 &= +\infty \end{aligned}$$

reflecting the fact that it takes an infinite time to achieve speed x_2 which corresponds to torque $T_a = \omega \cdot \zeta$ (friction torque).

During deceleration the sign of β will change as the friction torque contributes to the deceleration torque, therefore any state can be forced in finite time to the origin.

In Fig. 5.3 the state space representation of the switching curves is shown. These switching curves are obtained with $\beta < 0$ reflecting the deceleration required to stop the object. The optimal path of the stepping motor at rest ($x_2 = 0$ and $x_1 = A$) to move to the origin, is ABCDO (Fig 5.3).

If the constraint (81) is greater than the maximum x_2 along ABCDO then ABCDO is indeed an optimal solution subject to the constraints (80), (81). If (81) leads to an $x_{2\text{lim}}$ lower than the maximum x_2 the unconstrained path BCD does not meet (81). It will be shown that path BDO is the time optimal one subject to (81), (82).

Assume that the optimal path is BXO, and BXO is continuous in (x_1, x_2) . Let L be the point of intersection of BXO with A'D where A'DO is the optimal path. Because of the principle of optimality LXO $\equiv$ LDO. The problem therefore is reduced to the proof that path BD is time optimal. This is true because from (79)

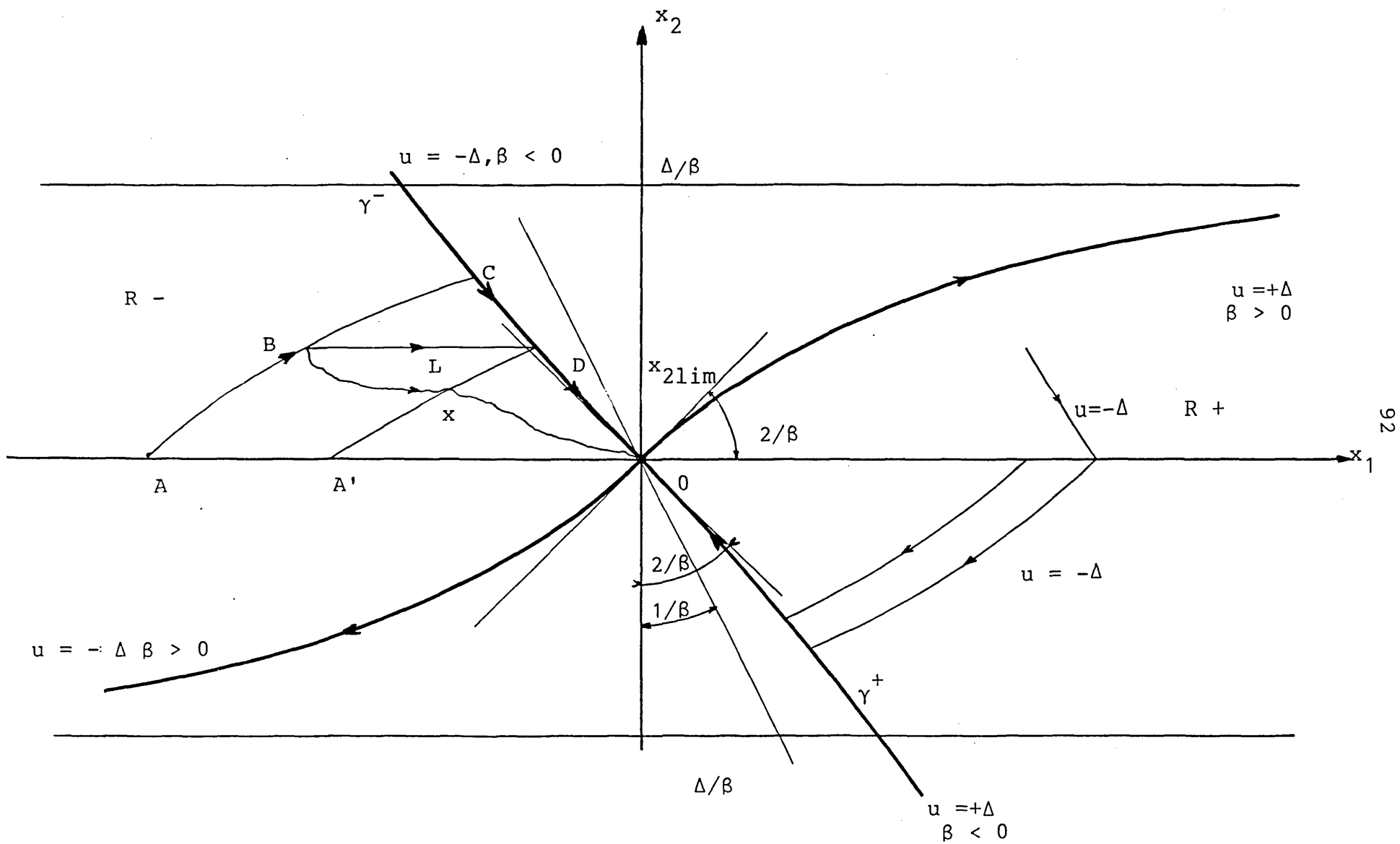


Fig. 5.3 State space representation.

$$t = t_0 + \int_B^D \frac{dx_1}{x_2} \quad (89)$$

and for $x_2 \geq 0$ the value that minimises the integral is $\max(x_2)$ which for BD is equal to X_{2lim} and therefore the segment BD is time optimal. It is not useful to derive the actual control necessary to implement the optimal controller as the software controller will use the rate of acceleration. The solution of (79) gives:

$$x_2(t) = x_2(0)e^{-\zeta t/I} + \frac{Ta.R}{\zeta} (1 - e^{-\zeta t/I}) \quad (90)$$

$$x_2(t) = x_2(0)e^{\zeta t/I} - \frac{Ta.R}{\zeta} (1 - e^{\zeta t/I}) \quad (91)$$

For the acceleration (90) $x_2(0) = x_{20}$

$$x_2(t) = \left[x_{20} - \frac{Ta.R}{\zeta} \right] e^{-\zeta t/I} + \frac{Ta.R}{\zeta} \quad (92)$$

and for the deceleration

$$x_2(t) = \left[x_{20} + \frac{Ta.R}{\zeta} \right] e^{\zeta t/I} - \frac{Ta.R}{\zeta} \quad (93)$$

The software open loop controller should therefore follow the optimum rate algorithm:

If $(x_{10}, x_{20}) \in R^+ \cup \gamma^-$ and $x_{20} > 0$ and $x_{10} \neq 0$

$$\text{then } x_2(t) = \left[x_{20} - \frac{Ta.R}{\zeta} \right] e^{\zeta t/I} + \frac{Ta.R}{\zeta} \text{ (deceleration)} \\ (u = -\Delta)$$

If $(x_{10}, x_{20}) \in R^+$ and $x_{20} \leq 0$ and $x_{10} \neq 0$

$$\text{then } x_2(t) = \left[x_{20} + \frac{Ta.R}{\zeta} \right] e^{-\zeta t/I} - \frac{Ta.R}{\zeta} \text{ (acceleration)} \\ (u = -\Delta)$$

If $(x_{10}, x_{20}) \in R^- \cup \gamma^+$ and $x_{20} < 0$ and $x_{10} \neq 0$

$$\text{then } x_2(t) = \left[x_{20} + \frac{Ta.R}{\zeta} \right] e^{\zeta t/I} - \frac{Ta.R}{\zeta} \text{ (acceleration)} \\ (u = + \Delta)$$

If $(x_1, x_2) \in R^-$ and $x_2 \geq 0$ and $x_1 \neq 0$

$$\text{then } x_2(t) = \left[x_{20} - \frac{Ta.R}{\zeta} \right] e^{-\zeta t/I} + \frac{Ta.R}{\zeta} \text{ (acceleration)} \\ (u = + \Delta)$$

if $(x_1, x_2) = (0, 0)$ then $x_2 = 0$

if $|x_2| > R.\omega_{\max}$ then $x_2 = \text{sgn}(x_2).R.\omega_{\max}$.

It is obvious that all the optimal speed profiles are exponential and the deceleration profile is in general different and steeper.

5.4 Stepwise Control Approximation

The preceding analysis is for the continuous control mode. Stepping motors are discrete motion devices in the sense that once a command has been given to the power driver to advance one step, a finite period of time should elapse before another command is given [K-11]. The software implementation of a stepping motor controller should compute the optimum rate at a given moment in time, and issue a command pulse to the power driver after a delay equivalent to the optimum computed rate. Another possibility is to stepwise approximate the optimal path and with the help of an interface issue a number of steps at a fixed step frequency. (Fig. 5.4). This will alleviate the processor from step by step computations freeing it for other duties.

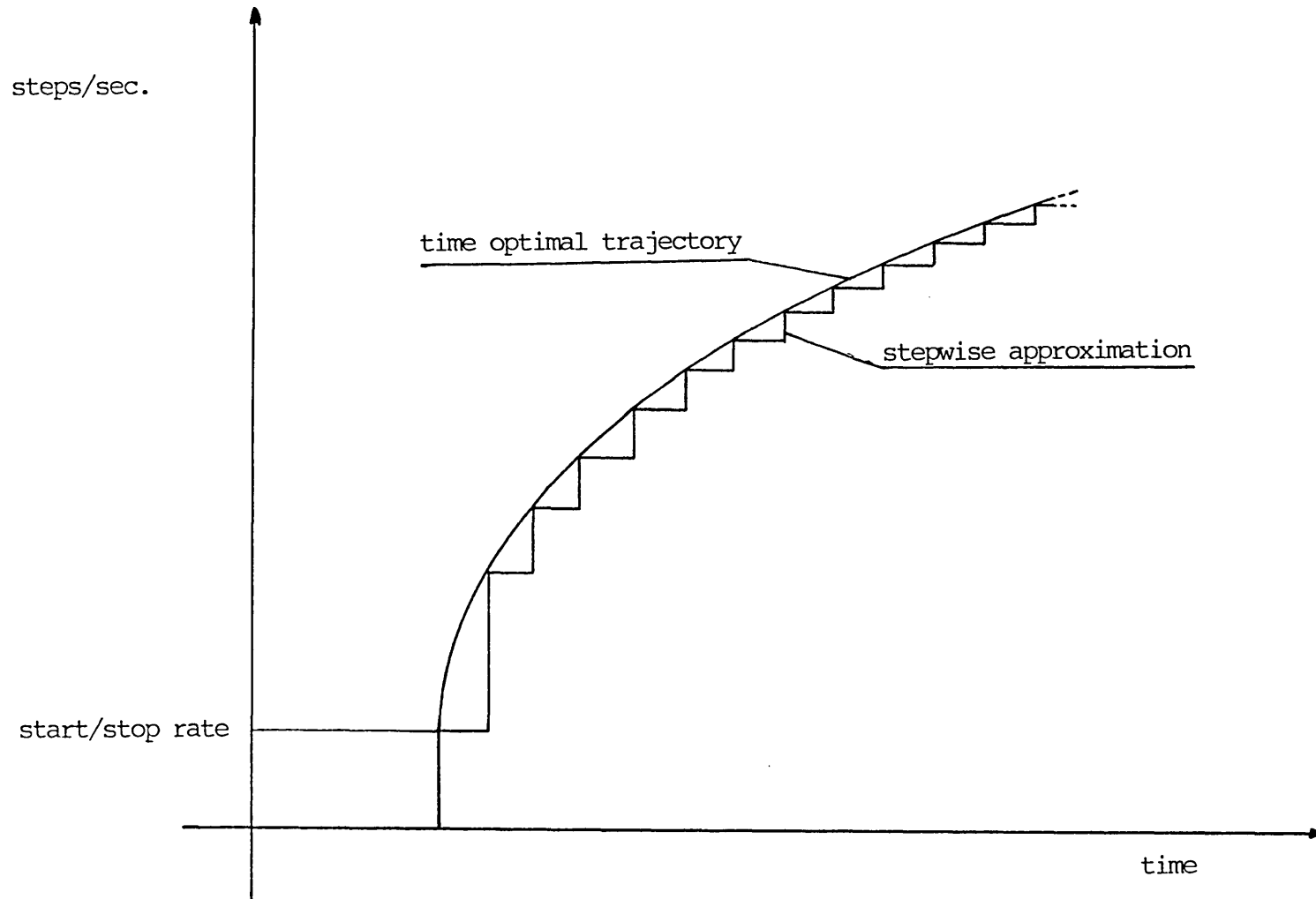


Fig. 5.4 Optimal rate and stepwise approximation.

CHAPTER 6

The Implementation of the Automated Test-bed

6.1 Overall Description of Test Bed Components and their Interaction

During Testing

The aim of the present author has been the design of a test-bed controller, flexible and adaptable to different physical test-beds and engine/dynamometer combinations. In Fig. 6.1 the overall test-bed arrangement is shown. The part of the test bed enclosed with dashed lines is the one subject to change between test beds.

It is obvious that the requirement for adaptability implies a certain standardisation in components and actuators. The engine sensors should be within the data-logging interface specifications (this can almost always be achieved through amplification and calibration). The shaft encoder providing the speed and synchronisation signals should be standardised [R-5]. The on-engine actuators like the stepping motor should be compatible with the power-drivers used. The dynamometer should be an eddy-current one if the Node 2 software is to be used intact. In case another Node is to be added in the system then the 16-bit bidirectional bus should be used for the physical connection, plus the PCNP (Chapter 2) protocol for the communication handler.

The rest of the system is transparent to the U.U.T. (Chapter 1). Mass storage devices are connected only to Node 0. Program loading to Nodes 1 and 2 is effected through the RS232 serial interface which is the mode of communication used by the monitor on these Nodes. This operation is effected off-line, prior to test initialisation, by an automatic monitor handler that simulates a file oriented operating system (using Node 0 mass storage devices) to a user accessing the

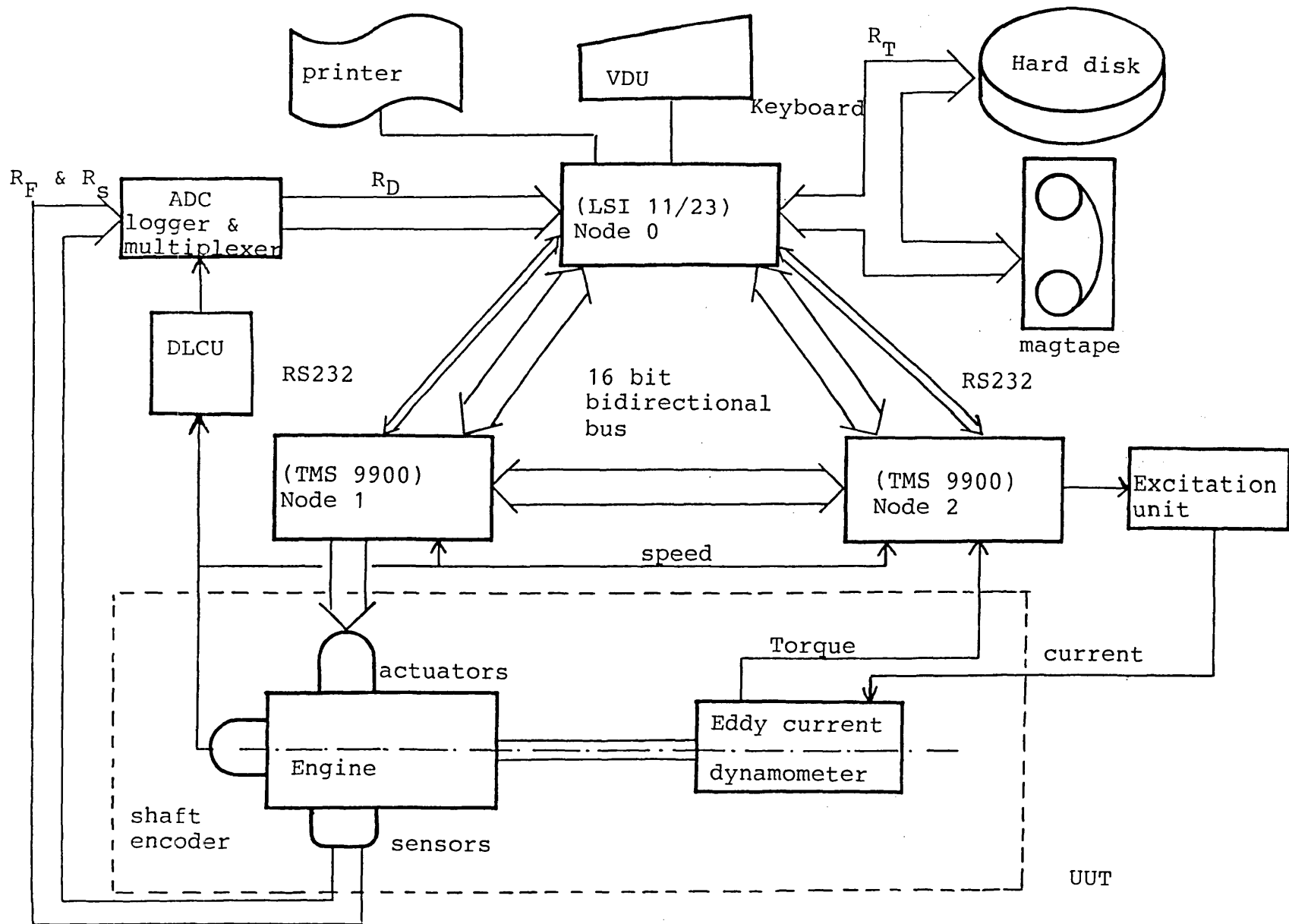


Fig. 6.1 Overall schematic of test-bed arrangement

Node 1 or 2 monitor through the Node 0 keyboard or application program.

Node 0 will ensure that Nodes 1 and 2 are loaded with the run-time programs, then initiate execution all over the network.

During a test the following data transfers and information exchange take place:

- (a) Node 0 - 1: Node 0 will send every half second the new engine controller directive (setpoint) in the form of a data packet and a supervisory synchronisation packet after ensuring that the previous directive has been received. Node 1 will send the current engine speed at a system generation selected rate (currently 40 ms). Either node will check for an end-of-test or abort signal from the other and if positive, the abort procedure will be initiated.
- (b) Node 0 - 2: Node 0 will send every half second the new torque controller directive together with a supervisory synchronisation packet after ensuring that the previous directive and synchronisation packet has been received. Node 2 will send the current torque measurement at a system generation selected rate (currently 20 ms). Either node will check for an end-of-test or abort signal from the other and if positive, the abort procedure will be initiated.
- (c) Node 1 - 2: Node 2 will send a torque measurement to Node 1 at a system generation selected rate (currently 20 ms). Either Node will check for an abort signal from the other and if positive, the abort procedure will be initiated.
- (d) Data logger - Node 0: Node 0 fills up its data buffers with data from the logger acquired either on-line for slowly varying inputs or off-line, then transferred using the DMA. (Direct Memory Access).

(e) Node 0 - Hard disk: Data buffers in Node 0 have to be saved in a mass storage device, if sizeable amounts of information are to be recorded. This is effected through double buffering and asynchronous transfers to the hard disk. At the same time test information (test schedule specification) is transferred asynchronously into the Node 0 test specification buffers from the disk file, using the same double buffering technique.

Node 1 will perform all on-engine actuator control like stepping motor handling. Node 2 will output through a DAC (Digital to Analog Converter) a signal to the dynamometer excitation unit to request a current on the dynamometer windings.

The detailed design philosophy, operation and measurement technique at each Node will be dealt with in the following sections.

Five different testing modes have been implemented:

- (1) Steady state: The engine will reach a steady state speed, the dynamometer will apply a steady state torque and when the system stabilises data-logging begins according to the specified test procedure. On completion of the test the load is removed and the engine driven to idle. Post-processing produces averaged results for the specified channels and standard deviation as an aid for acceptance or rejection of the test by the operator. The formatted results are stored in the magnetic tape for further processing and plotting at another site.
- (2) Transient tracking mode: The engine and torque controller go through a user predefined sequence of settings while data logging is performed as per the test specification. This mode uses preview methods to optimise the torque schedule, and involves significant off-line processing. On completion the load is removed and the engine driven to idle. Post-processing produces batch by batch results available for inspection on the VDU in the

form of histograms, for every sampled channel. The formatted results are handled as for the steady state mode.

- (3) Non-tracking transient mode: Identical to the transient tracking mode but the torque schedule is not optimised with preview methods.
- (4) Power law mode: Similar to the transient modes except that the torque controller is fed initially with the coefficients of a third order speed polynomial and the instantaneous torque request computed as a speed function.
- (5) Speed control mode: Similar to the transient modes except that the engine controller receives directives for the absolute positioning of the actuator (stepping motor) and the torque controller adjusts the torque to achieve a speed request.

The different test mode requirements are handled by the test generator program which performs logical checks on the user requirements and specification so that parameter contradiction cannot be created.

6.2 The Data Logging Node (Node 0)

The general requirements of data-logging have been described by Roberts [R-5]. The data logging and control unit (DLCU) that handles data logging transparently from Node 0 has been described thoroughly in [R-5] and [M-5]. The novel requirements in the new test bed are the following:

- (a) The transient testing specification has to be stored in a mass storage device (e.g. hard disk) so that virtually unlimited (timewise) test sequences can be executed (discussed in 6.1).
- (b) Test data should be stored in a mass storage device at the fastest possible rate.

Both requirements imply that the data logging program should handle I/O (input/output) in the most efficient way to avoid an I/O induced 'bottleneck'.

The data logging is characterised by the following rates of transfer.

- 1) Peak sampling rate which is achieved through the off-line ADC logger [R-5]. Let this rate be R_F (words/cycle). The total data transfer rate is equal to $R_F \cdot \eta_F$ where η_F is the number of Fast adc channels sampled.
- 2) Slow sampling rate which is achieved through the on-line ADC logger. Let this rate be R_S (words/sec). The total data transfer rate is equal to $R_S \cdot \eta_S$ where η_S is the number of slow adc channels sampled.
- 3) DMA rate which is the rate that data is transferred to Node 0 memory from the off-line ADC memory. Let this rate be R_D .
- 4) Disk transfer rate which is the rate achieved during transfers to hard disk. Let this rate be R_T .

If the engine runs at f cycles/sec and sampling is effected every cycles (skipcycle number) then the following inequality can be written:

$$R_T \geq \frac{f}{S} R_F \cdot \eta_F + R_S \cdot \eta_S \quad (94)$$

$$\text{if } \frac{f}{S} \cdot R_F \cdot \eta_F < R_D \quad (95)$$

Inequality (95) does not present an important constraint as by the nature of DLCU synchronisation the theoretical skipcycle limit is 2, and practical DMA rates outperform ADC conversion rates. Inequality (94) shows that for given f , η_F and for the equal sign holding, the

only way to increase the peak sampling rate is by either increasing the skipcycle number or by increasing the disk transfer rate R_T . The increase in skipcycle number is a solution of last resort as in transient testing it is important to keep the closest possible track of some quickly varying parameters. The term $R_g \cdot \eta_g$ is in general small and does not affect throughput greatly. What is therefore left to optimise is the disk transfer rate.

The existing hard disk is a moving head one, and the specification give transfer rates up to 64K words/sec. The peak transfer rate is achieved when the positional delay (seek time) is zero i.e. the disk arm does not have to move, and the transfer rate is based on the average rotational delay (latency). Experiments confirmed that when transfer was effected simultaneously on two files (normally on different sectors) a maximum transfer rate of 4K words/sec could be achieved whereas 64K words/sec were achieved on single file transfers. This result dictated that one output file should be used and the data organisation should be such that one file operation becomes feasible.

The data logged is only 10 bits long whereas the memory word is 16 bits long. In that respect the top 6 bits are cleared for the data and 16 bit headers are used to differentiate between fast and slow ADC batches. The actual batch length after the header, is fixed at test generation time and is used by the output file analyser program to determine the positional significance.

The data-logging system is essentially a sophisticated I/O handler and for this reason the actual data transfer design philosophy has been described.

Other tasks of Node 0 include the communication handling as described in 6.1. All events are handled through interrupt routines and the critical regions have been carefully selected to avoid

interrupt latency and ensure correct resource sharing and protection. Software techniques have been used to simulate software skipcycle numbers, variable within a test sequence enabling flexibility in the isolation of a particular part within a test. Skipcycle numbers for fast and slow sampling have been completely disassociated and specified independently. Time based sampling has been provided for slow ADC measurements as an alternative or in conjunction with skipcycle based sampling.

The Node 0 data logging capability has been tested by an engine speed simulator. The system performance has been as follows:

speed: 3000 RPM (1500 cycles/min on 4X)

Fast adc: 1440 words/cycle . 5 channels

Slow adc: 50 words/sec . (32 + 6) channels

Minimum skipcycle number: 4

Average disk transfer rate: 48.5 K words/sec.

Roberts [R-5] achieved through the same hardware a skipcycle number of 6 only for steady state tests.

The PDP-15 based system [S-12] achieved a minimum of 7 skipcycles at speeds up to 2000 RPM. The improvement has been of the order of 50% by the reorganisation of the data logging procedure and I/O.

6.3 Node 1 and the Engine Control

The functions of this node are the control of the on-engine actuator (stepping motor), the control of the engine itself and the communication handling of the processes.

Stepping motor controller

The time-optimal acceleration profile has been derived in chapter 5. An exact implementation of the controller would require the computation of the state trajectory at every moment in time. This is

beyond the microprocessor capabilities when used in a multitask configuration. In addition most requests to the controller will start from rest therefore the stepping motor will always require an acceleration from rest trajectory or a deceleration one (on the switching curves). It is therefore possible to precompute these trajectories and store them in the computer memory so that an incremental indexed access leads to the generation of the optimal stepping rates. The profile shape is computed off-line by the

following formula:

$$V = V_1 + \frac{(V_F - V_1)(1 - e^{-T/S_K})}{1 - e^{-T_F/S_K}} \quad (96)$$

Where V_1 is the start/stop rate (steps/sec)

V_F is the final step rate (steps/sec)

T is time in seconds (sec)

T_F is the duration of acceleration (sec)

S_K is the shape factor.

Equation (96) is the normalised version of (92) when $x_{20} = 0$ (at rest). The shape factor is the ζ/I ratio indicating that a high inertia object will have a very pronounced exponential velocity profile, whereas a low inertia one will tend to be linear when all other parameters are kept constant and the duration of acceleration normalised. The controller configuration does not use a different deceleration profile for the sake of simplicity. During test generation the user is prompted with the questions necessary for the profile computation which is then automatically loaded in Node 1 memory.

Engine controller

Engine control is performed through a discrete PID controller with the setpoint band modification [C-10] and torque feedback incorporated in the offset of a positional form algorithm [W-10]. The positional form has been adopted so that the engine map can be directly related to actuator position when the offset incorporates other forms of feedback or preview, non linear over the engine range. The engine map is a three-dimensional one between speed-torque-position designed to cope with torque feedback and speed preview but actual tests have been carried out with torque feedback only. The map is stored as a data table of a particular structure and 3-D interpolation by the method of smoothest gradient is performed on-line. Saturation of the controller is avoided by resetting the controller status when the actuator hits the limit switches. Speed measurement is performed digitally by counting the clock pulses of a fixed frequency clock between each revolution. The novelty of this system lies in the derivation of the revolution pulse by a pulse divider of the crank-angle pulses and not by the revolution pulse supplied by the optical shaft encoder. This method is more fault tolerant as the loss of a pulse will produce a $1/720$ error whereas before the error would be $1/2$. [W-10, W-14].

Tuning of the PID has been performed through the Ziegler-Nichols recommendations [R-6]. The engine and the stepping motor have to be considered as a single object for tuning purposes therefore tuning parameters are only valid for the particular stepping motor controller characteristics selected during the step-input application.

Communication controller

Communication is based around the PCNP primitives. Data transfers are directed to other nodes and directives forwarded to the engine or stepping motor controllers. Node 1 management is effected according to Node 0-2 information and abort procedures initiated.

Node 1 is multiplexed between the following tasks:

- (a) Stepping motor control with the second highest priority interrupt service routine so that no loss of synchronisation occurs.
- (b) Engine control with the third highest priority interrupt, entered through a fixed frequency clock currently at 50 ms.
- (c) Communication handling with the lowest priority interrupts, to service the data transfers to the other nodes.
- (d) Exception interrupts with the highest priority interrupt level, to service engine malfunction alarms and actuator limit switches.

Because of the priority structure it is obvious that the system assumes a steady state capacity to handle the tasks. This capacity would be impaired if the stepping motor controller operated at extremely high stepping motor rates. To prevent this from happening a software limiter does not allow the profile generation program to exceed an interrupt frequency of 1 kHz for the stepping motor controller. This frequency is fixed at system generation.

6.4 The Dynamometer Control Node (Node 2)

The functions of this node include the dynamometer control and the communication handling. The discretisation of the optimal algorithm will be briefly examined:

The discrete controller stability

The system described by (60) has the following discrete time form
(T is the sampling interval)

$$\begin{bmatrix} x_{1,K+1} \\ x_{2,K+1} \end{bmatrix} = \begin{bmatrix} e^{\lambda_1 T} & 0 \\ 0 & e^{\lambda_2 T} \end{bmatrix} \begin{bmatrix} x_{1,K} \\ x_{2,K} \end{bmatrix} + \begin{bmatrix} \frac{e^{\lambda_1 T} - 1}{\lambda_1} & 0 \\ 0 & \frac{e^{\lambda_2 T} - 1}{\lambda_2} \end{bmatrix} \begin{bmatrix} -1 - a\lambda_1 \\ 1 + a\lambda_2 \end{bmatrix} \frac{x_{1,K}}{a}$$

The control signal has been substituted by the state feedback form.

By effecting the matrix computations:

$$x_{1,K+1} = e^{\lambda_1 T} x_{1,K} + \left[\frac{e^{\lambda_1 T} - 1}{a\lambda_1} \right] \left[-1 - a\lambda_1 \right] x_{1,K} \quad (97)$$

$$x_{2,K+1} = e^{\lambda_2 T} x_{2,K} + \left[\frac{e^{\lambda_2 T} - 1}{a\lambda_2} \right] \left[1 + a\lambda_2 \right] x_{1,K} \quad (98)$$

(97) can be rewritten as

$$\frac{x_{1,K+1}}{x_{1,K}} = 1 - \frac{e^{\lambda_1 T} - 1}{a\lambda_1} = \beta \quad (99)$$

In Chapter 4 it was shown that $\lambda_1 < 0$ and $a > 0$.

In order to have:

$$-1 < 1 - \frac{e^{\lambda_1 T} - 1}{a\lambda_1} < 1 \quad (100)$$

$$e^{\lambda_1 T} \geq 1 + 2a\lambda_1 \quad (101)$$

(101) must hold. If (101) holds then $|\beta| < 1$ and

$$\lim_{K \rightarrow \infty} x_{1,K} = 0 \quad (102)$$

therefore the x_1 state reaches the origin when the state feedback control is applied.

Similarly for the x_2 state:

$$\frac{x_{2,K+2}-d \cdot x_{2,K+1}}{x_{2,K+1}-d \cdot x_{2,K}} = \frac{x_{1,K+1}}{x_{1,K}} \quad (103)$$

$$0 < d < 1$$

From (103) the $x_{2,K}$ sequence is bounded and monotonic therefore converges. (Bolzano-Weierstrass Theorem). In addition because of (98) and (102) the limit is zero:

$$\lim_{K \rightarrow \infty} x_{2,K} = 0 \quad (104)$$

therefore x_2 reaches the origin with the state feedback control. The necessary and sufficient condition for stability is (101). The choice of the sampling time has also to take into account bang-bang mode operation. If the sampling time is too large then the controller will never enter the state feedback mode of control. Such an effect has not been experienced. The choice of the optimal constants reduces (101) into:

$$e^{\lambda_1 T} \geq 1 - 10\lambda_1^2 \quad (105)$$

which is always true for $|\lambda_1| > 0.32$.

The torque controller

The control algorithm derived in section 4.8 has been implemented together with the tracking mode modifications suggested in 4.9. The basic sampling time is 20 ms and the setpoint lookahead time is one half second. The tracking controller performs a 4-point (2 seconds) preview action. A speed compensation loop has been written so that the torque controller under the tracking mode presents zero steady state error. This is necessary as the tracking mode uses precompiled

flux rather than torque schedules and the flux computation assumes a speed schedule. This loop can be omitted if a speed controller with integral action is used and speed tracking is adequate.

The speed control mode has been implemented by superimposing a PID on the optimal controller. The sampling rate for the PID controller can be set at any multiple of 20 ms. Such an option is selected at system generation and the current rate is 40 ms. Tuning of the PID can be effected by the Ziegler-Nichols recommendations when a step load is applied to an engine without speed control.

Flux mapping

The optimal controller requires a flux input to operate. Mapping is performed by steady state tests relating speed, torque and steady state excitation considered proportional to the flux. Initially the flux was expressed as a second order speed-torque function and the constant coefficients derived by a least-squares fitting technique over the experimental data. This was later abandoned as it could not provide accurate predictions over the whole range. A 3-D interpolation program was used instead, similar to the speed controller one (see section 6.3), using the measured values directly. The interpolation method assumes linearity within samples and consequently it becomes inaccurate towards low torque conditions [D-1]. A tradeoff is therefore necessary between available memory for table storage and desired accuracy.

Torque measurement

Torque measurement has been reported to have considerable noise [W-10, C-3]. The severity of the noise level is a function of the design of the test bed and the anti-vibration measures taken. The mechanical noise has two origins. One is test-bed vibration and the

second is the cyclic irregularity of the torque produced by the engine. Cyclic irregularity can only be reduced by increasing the inertia of the system, which is undesirable. Filtering is not possible as in most cases the fluctuation is of the order of the torque sampling time.

The best solution to avoid vibration noise is the provision of good anti-vibration measures on the test-bed. Nevertheless if such measures are impractical or impossible, an alternative filtering technique should be used. The First low-pass filter for the torque is the dynamometer itself. Its inertia I (which is high) in conjunction with the load cell arm length d and the load cell spring constant K determine the cutoff frequency.

$$f = \frac{1}{2\pi} \sqrt{\frac{K \cdot d}{I}} \quad (106)$$

The value of f is practically large and preliminary computations for the 3WB15 dynamometer give a value around 200 Hz.

The second stage of filtering has usually been an electronic low-pass filter [V-1, W-10]. This method rejects all frequency components above the cutoff frequency but also attenuates components constituting the required signal itself.

Vibration noise is not Gaussian as the primary exciting input is not Gaussian. It can be considered as the sinusoidal input response of a second order inertial system represented by the dynamometer body and attachments. Filtering of this signal would require a high rejection of the input frequency which is variable according to test bed resonance characteristics. Such filtering could be effected by software but previous experience [W-10] showed that it is processor-time consuming and would require an almost dedicated processor.

A hardware integrator was finally implemented providing the time average of the measured torque and the integration frequency was set to 20 ms. This has a strong 50Hz rejection which also helps eliminate the load cell power supply electrical noise. The delay introduced in the control loop did not practically affect system stability. The integration signal is converted to a 10-bit number available to the processor.

Initial experiments were effected on the PERKINS test bed described in the Appendix. The lack of anti-vibration mounting created extreme vibration which is displayed by Fig. 6.2. The frequency was of the same order as the integration time therefore filtering was not effective. From Fig. 6.2 it is possible to determine multiple resonances not only related to engine speed but to load levels as well. The same experiment on the LEYLAND test bed (see appendix B) which has much better anti-vibration construction showed an almost zero standard deviation throughout the speed range.

Dynamometer excitation

The control of the dynamometer is effected by varying the excitation current. The processor controls this current by outputting a word to the DAC presenting the input to the excitation unit. This excitation current is achieved in a closed loop mode although the processor does not have a direct current feedback. This is unnecessary as the excitation unit achieves the required current level before the next sampling period. (Appendix C).

Communication handling

Communication is identical to Node 1. The PCNP implementation uses the bidirectional 16-bit data buses provided for this purpose. Message communication involves torque readings to Nodes 0-1 and

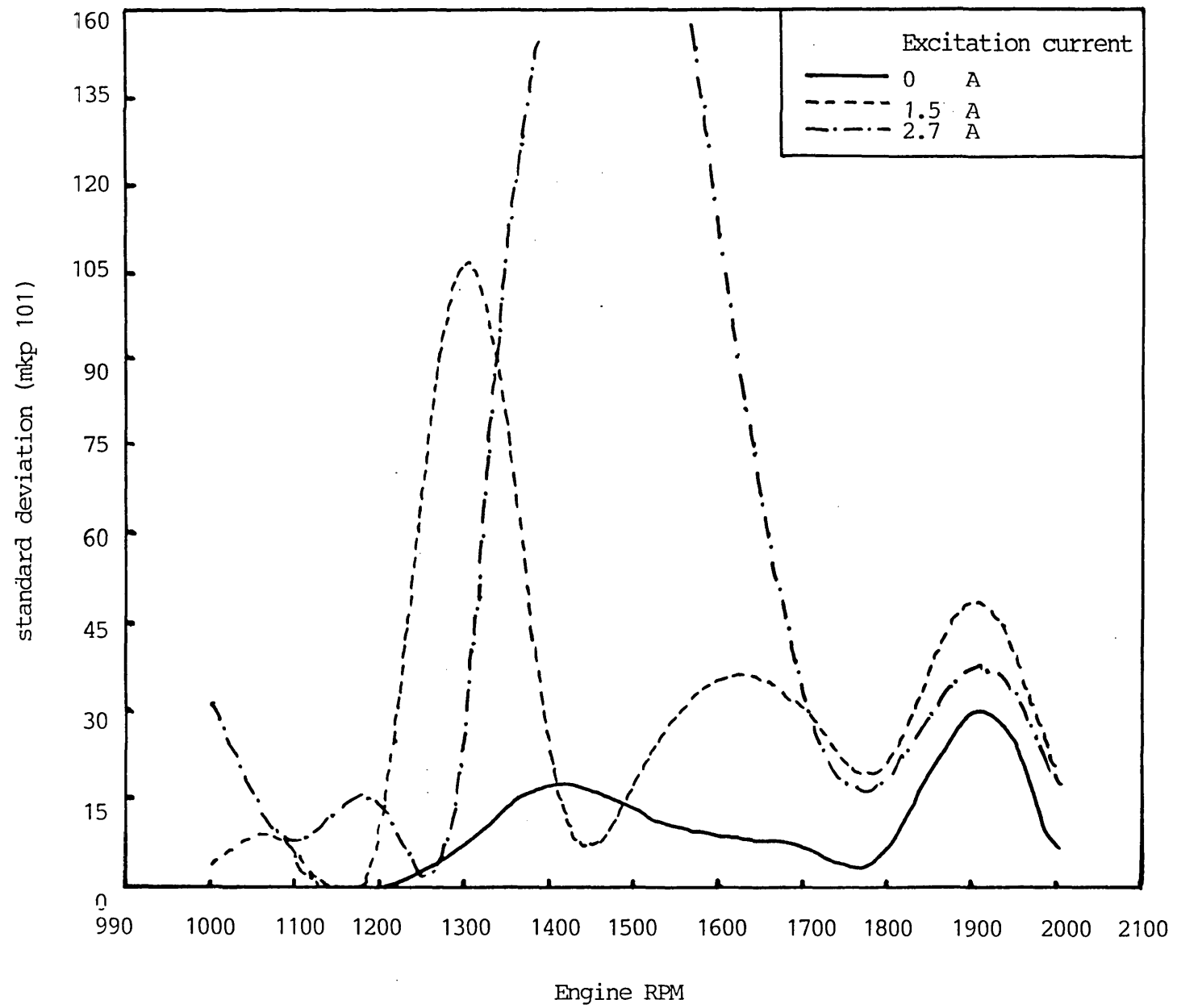


Fig. 6.2 Standard deviation of torque measurements on the PERKINS test-bed.

torque setpoint from Node 0.

6.5 Test Generation Procedure

In Chapter 1 the benefits of automatic testing were put forward. One of the reasons for introducing automatic test equipment is the extension of the range of users of such test equipment. It should be possible to execute a test without detailed knowledge of the system though this should not impair flexibility and versatility of the testing procedures.

These objectives were met by the Automatic test generation and post-processing software package. They constitute chained [R-5] programs invoked according to test-specification requirements.

Three basic modules have been provided

1) Test data file generation

With this module the user specifies or inspects a test sequence for any of the five available modes of testing. The data file creation is performed by an interactive dialogue where the user is prompted for setpoint values specified every half second. Inspection of old test data files is available and printout held as an option.

2) Test specification file generation

The user is prompted with questions regarding a newly specified test. It is possible to create a new test, modify an existing test, or execute an existing test. Test summary with optional printout is also available to provide a hard copy of test specification. Data required for test specification is divided into five categories:

- a) General where test name, mode and associated test data file are specified.

- b) Fast ADC specification where the mode of operation of the off-line logger is specified [R-5].
- c) Slow ADC specification where the on-line logged channels are specified together with the sampling mode combination (time based or cycle based or both).
- d) Engine controller specification where Engine PID constants, interpolation file for mapping, idle RPM and stepping motor characteristics are specified.
- e) Torque controller specification where dynamometer constants, maximum excitation, flux interpolation file, torque calibration constant, offset and speed PID constants (when in speed control mode) are specified.

The test specification module can invoke the test initialisation module that handles automatically the bootstrapping of the network by loading Nodes 1 and 2 through the RS232 line. This module chains automatically to the data logging and control program. All off-line processing is performed in Node 0 as the only one with mass storage peripherals.

3) Post-processing software

On the successful completion of a test this module is chained automatically and processes the recorded data using the calibration constants, offsets and positional significance specified during the test specification file generation. Steady-state tests are averaged per cycle and the standard deviation computed.

Transient tests are processed cycle by cycle for fast ADC recordings and time based for slow ADC readings. Inspection can be random for any cycle and channel, and can be displayed in histogram form or as actual values of the recorded variable. This module

provides for the automatic transfer of the processed results on to a file structured magnetic tape as a means of long-term data storage and transfer to the central computer plotting facilities.

The modules described above cover the requirements set for the Automatic Test Equipment. The user is completely isolated from the details of the testing and processing procedure and the information he is required to input, reflects purely the description of his desired test and do not involve any knowledge whatsoever of the on-line or off-line system operation. All tests carried out on the two test-beds described in Appendix B were generated and processed using these facilities. The data was transferred to magnetic tape, then to the ICCC (Imperial College Computer Centre) in order to use the comprehensive plotting facilities and obtain the graphs of Chapter 7.

6.6 Overview of programming languages used

High level programming languages have a definite advantage over assembly level languages as far as development is concerned. Powerful instructions are included in their repertoire and run-time checks inserted by the compiler, minimize suprious corruption of data and code. This is unfortunately at the cost of run-time efficiency of execution. As described in 6.2, 6.3 and 6.4 there are many time critical processes where interrupt latency (the maximum time that an interrupt is disabled) should be kept to a minimum therefore execution speed is the determining factor. The choice of the programming language was therefore dictated by the real-time requirements of the control systems.

As a general rule all the software written for control purposes and data-logging was in assembly language. The LSI 11/23 software was written in MACRO-11 and assembly was done through the operating

system (RT-11) utilities. The TI 9900 (Nodes 1 and 2) did not have an assembler on the RT-11 therefore a cross-assembler was developed by the Author written in FORTRAN IV from that developed by WIJEYAKUMAR [W-10].

The test generation procedure which involved considerable screen formatting and character handling was written in FORTRAN IV. There were no time critical considerations at that point and ease of programming prevailed. The same considerations led to the use of FORTRAN IV in the post-processing and graphics package. FORTRAN is not the ideal language for character and file handling but most engineers are acquainted with it and program updating could therefore be made easier.

CHAPTER 7

Evaluation of Results from Engine Testing

7.1 General Test Description

A series of tests was conducted on two distinct test-beds described in Appendix B. Most of the tests were repeated on the two test-beds so that comparative results could be obtained. The aim of the tests was twofold:

- a) To investigate controller behaviour
- b) To demonstrate test bed flexibility regarding the nature of supported tests and automatic testing capability.

Each of the two test beds had a distinct problem. The PERKINS engine test-bed did not have any anti-vibration protection and as a result the torque signal scatter was excessive except in the vicinity of 1200 RPM (Fig. 6.2). This is the reason that step loading tests for this test bed are presented only for the 1200 RPM region. The LEYLAND engine test-bed was used with an on-engine governor with proportional and derivative action and with a boost pressure rack limiter. Tuning of the engine controller was impossible as effectively two governors (software and hardware) were competing for control. This was particularly felt during simultaneous torque and speed requests.

The drawbacks of each test-bed led to a good speed - poor torque control for the PERKINS and a poor speed - good torque control of the LEYLAND. These drawbacks have to be kept in mind when examining the test results.

0 = used X = not used

Fig. No.	P I D	Torque feedback	Torque Control Tracking
7.1	0 X 0	X	X
7.2	0 0 0	X	X
7.3	0 0 0	X	X
7.7/8/9/10	X X X	X	X
7.11/12/13	0 0 0	X	X
7.14/15	X X X	X	X
7.16 ~ 24	0 0 0	X	X
7.25/26	0 0 0	0	X
7.27/28	0 0 0	X	X
7.29 ~ 33	0 0 0	X	X
7.34/35	0 0 0	X	X
7.36 ~ 39	0 0 0	0	0
7.40/45	0 0 0	0	X
7.50/53	0 0 0	X	X

Table 7.1 Table of Control Parameters in tests of Chapter 7

7.2 Step Loading Tests

Step loading tests provide a good indication of controller behaviour as they can be directly compared with the theoretical response of the model to a step input. These tests have been conducted at different loads and speeds to investigate the real system behaviour.

Fig. 7.1 and 7.2 display a step loading test at 22 and 10 mkp respectively. In both cases the starting speed was 1200 RPM which presents the lowest torque reading scatter on the PERKINS test-bed. The speed controller in 7.1 has only P, D terms therefore once the load is applied and a speed droop established, there is no further fluctuation. In 7.2 the speed controller also has an integral term (PID) therefore the load application acts as a disturbance to the speed setting and the speed fluctuates for about 2.4 seconds. This fluctuation in turn induces a small torque fluctuation until the system stabilises at 10 mkp - 1200 RPM. Overshoot in both cases is below 4%.

The dynamometer manufacturer uses a synchronous motor instead of a diesel engine for dynamometer control tests, to achieve a variable torque at fixed speed and avoid the disturbance to the torque controller induced by the speed fluctuation or droop.

Fig. 7.3. and 7.4 display step loading-unloading of the dynamometer with the speed controller incorporating integral term (PID) hence speed is always kept around 1200 RPM. No overshoot or undershoot were experienced during these tests. The torque decay, especially at the lower range, is beyond control, and reflects the flux decay characteristics. The torque curve shown in Fig. 7.4 from 15s onwards displays the limitation of eddy current dynamometers which can be overcome only through motoring capability.

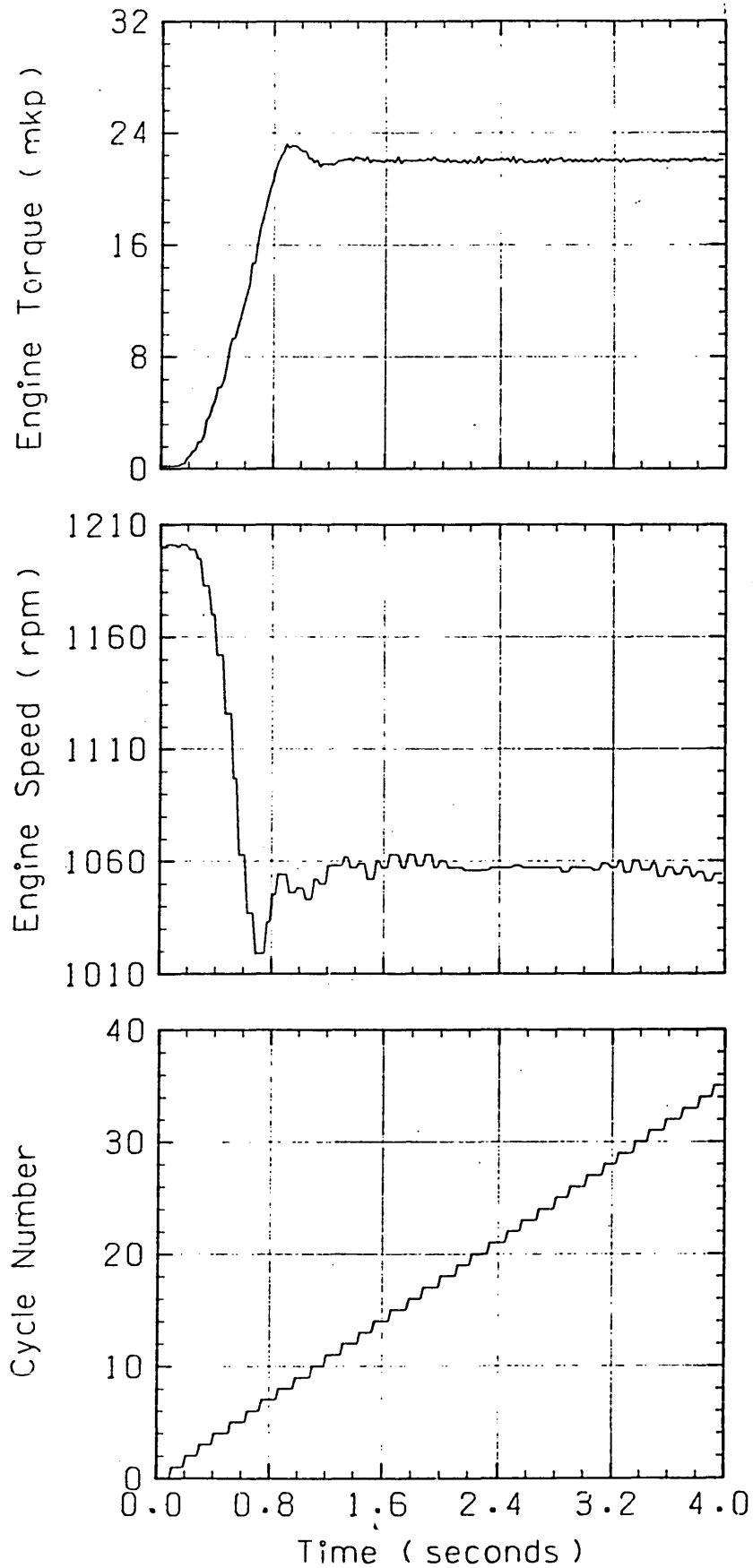


Fig. 7.1 PERKINS test-bed step loading (22 mkp).

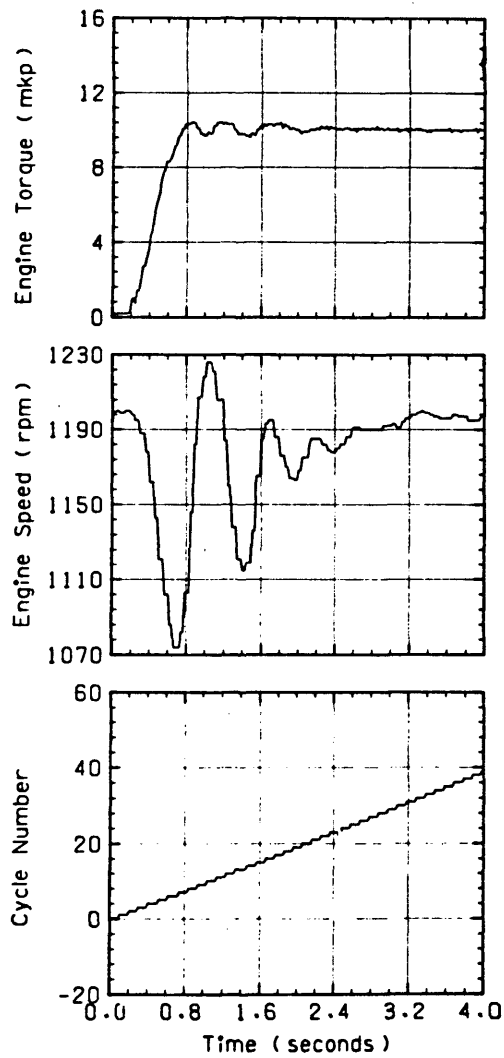


Fig. 7.2 PERKINS test-bed step loading (10 mkp)

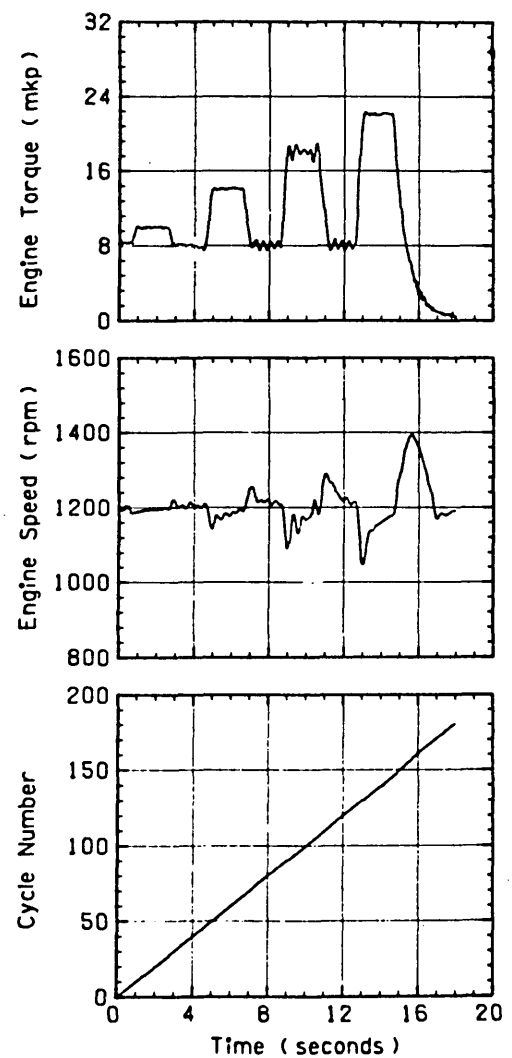


Fig. 7.3 PERKINS test-bed step loading-unloading.

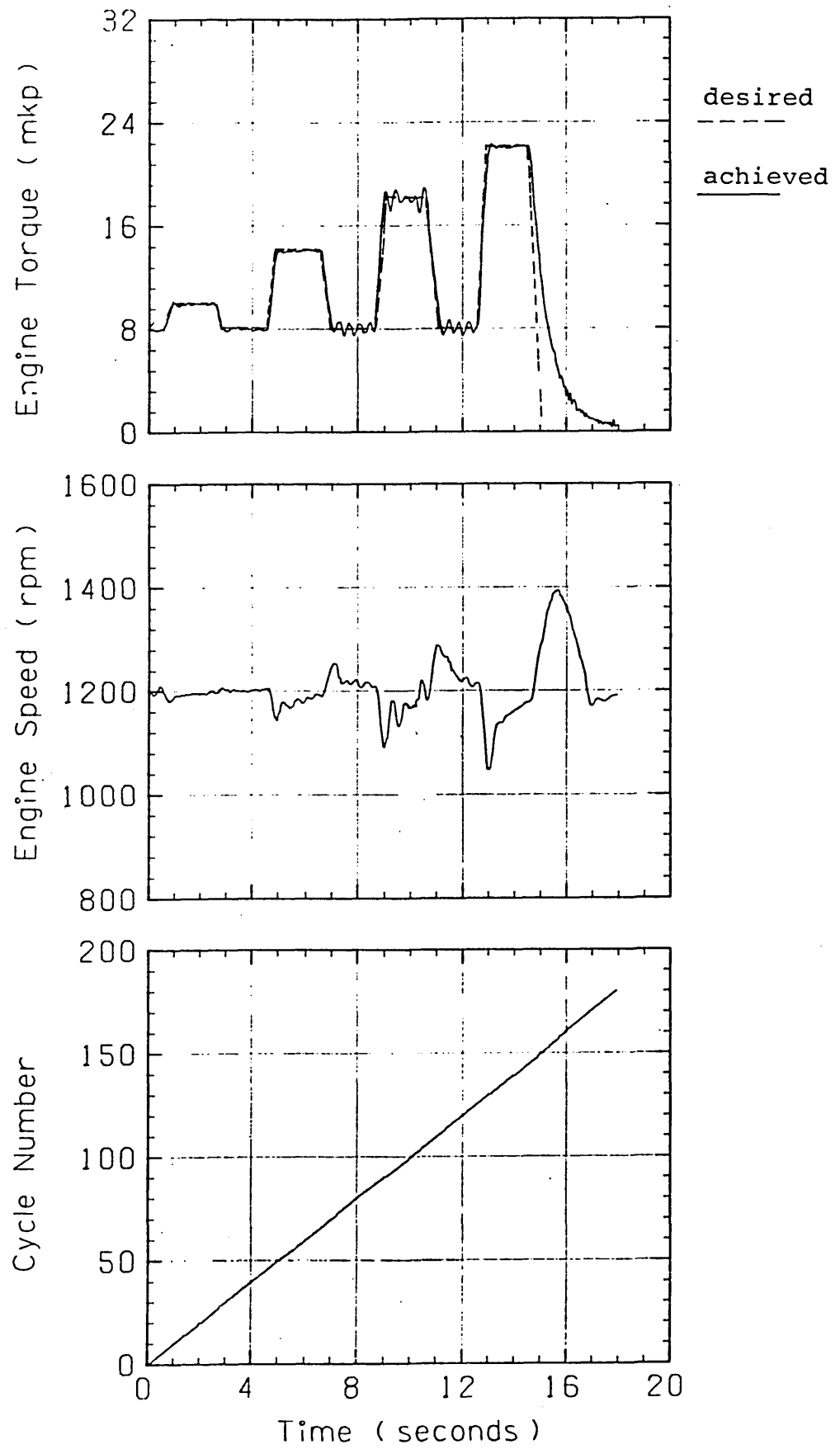


Fig. 7.4 PERKINS test-bed step loading-unloading

Fig 7.5 and 7.6 illustrate the behaviour of different dynamometer controllers. All three curves have been normalised and shifted in the t axis so that they reach 10% of target torque at the same time. They correspond to the following step load conditions:

	Target torque (mkp)	RPM
Time optimal controller (test of Fig. 7.1)	22	1200
Software PID, BEC 12/45 excitation unit.	15	1200
Analog PID CEB 105 excitation	5	2000

The Analog PID has a better rise time than the time-optimal controller but this is partly due to the fact that the Analog PID curve was obtained at 2000 RPM whereas the time-optimal test was carried out at 1200 RPM. The Analog PID data was taken from the manufacturer's tests in which a synchronous load was used, therefore its loading conditions are idealised. In spite of this fact the superiority of the time optimal response is obvious. The poor software PID response achieved by Wijeyakumar [W-10] is partly due to the nature of the excitation unit used at the time.

The step loading tests on the LEYLAND test-bed were more comprehensive regarding the speed and load range as the torque measurement was noise free. Figures 7.7 through 7.15 display different combinations of step loads and speeds. The effect of the on-engine governor cannot be eliminated on the LEYLAND test bed and is superimposed to the speed controller. This means that the engine speed cannot be stabilised within the torque application period and therefore instead of having a step load request only, a combined step load - speed change request is in effect. This results in the small oscillations of the torque around the requested step load, while the

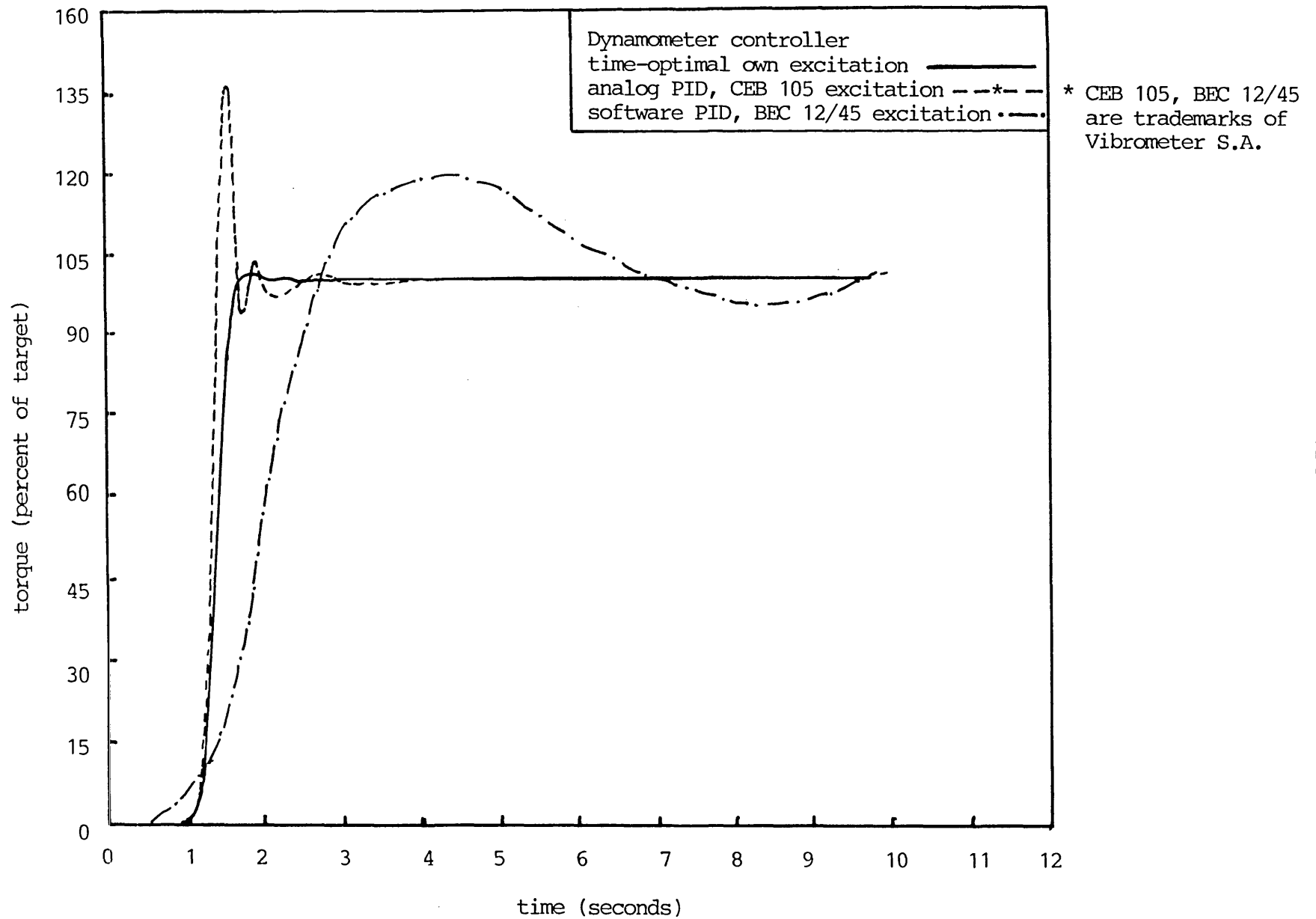
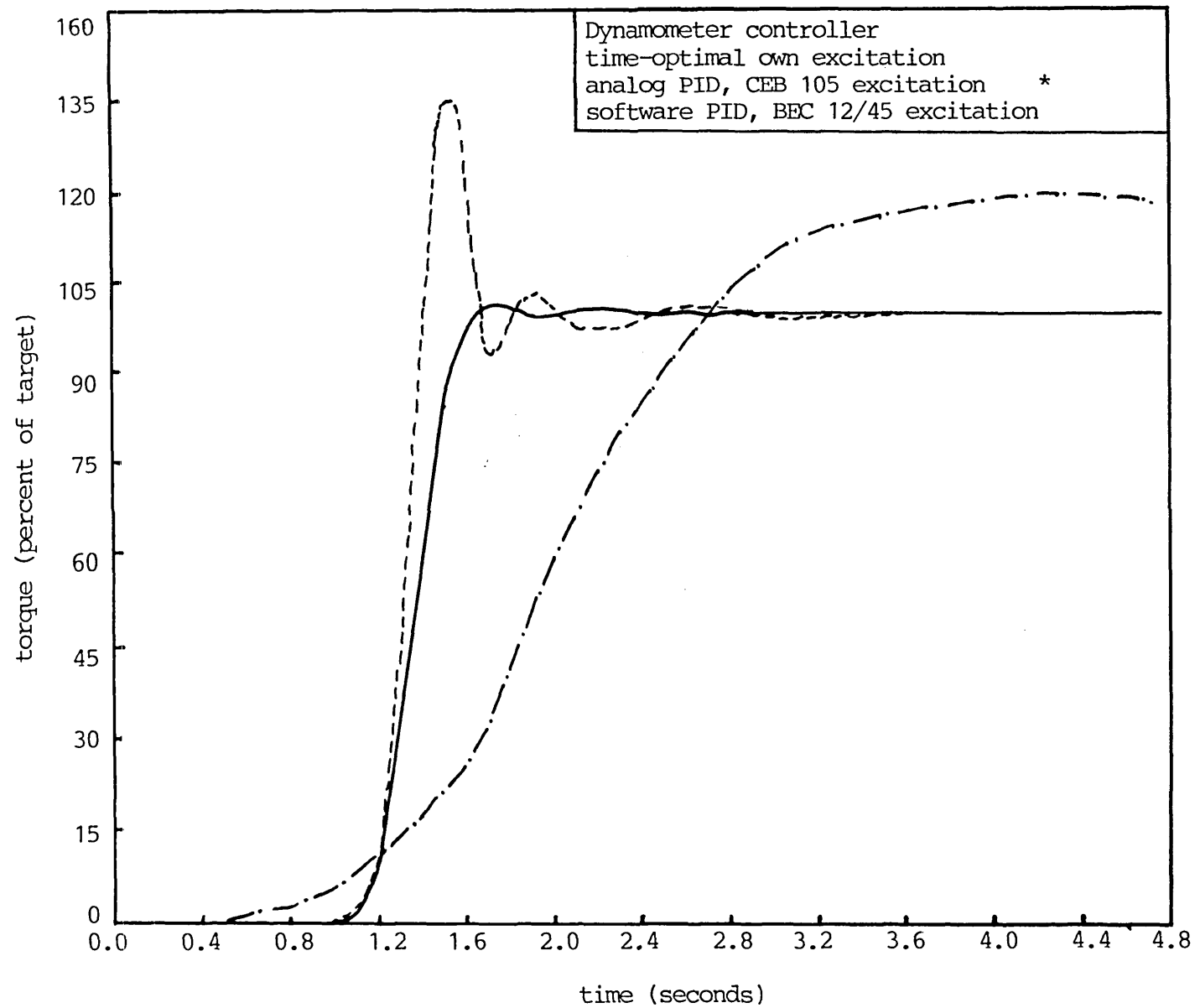


Fig. 7.5 Comparative step loading.



* CEB 105, BEC 12/45
are trademarks of
Vibrometer S.A.

Fig. 7.6 Comparative step-loading

speed is varying.

Tests 7.7, 7.8, 7.9, 7.10, 7.14 and 7.15 were carried out with an inactive speed controller (fixed governor setting) therefore what is shown as engine response is the on-engine governor response. Tests 7.11, 7.12, 7.13 use a PID speed control therefore the engine as a whole is kept at a constant speed without droop by means of the integral term action.

Fig. 7.16 through 7.18 demonstrate step loading and unloading at different speeds with an active PID speed control (notice that speed at the beginning and end of test are the same although the loads are different which indicates the integral action). The no overshoot or undershoot behaviour displayed in Fig. 7.4 for the PERKINS test bed is reconfirmed.

Fig. 7.19 demonstrates the capabilities and limitations, regarding transient load application, of the torque controller and eddy current dynamometer combination. This test is combined with transient speed requests in order to demonstrate the torque controller capability both under steady state and varying speed conditions. The test sequence comprises an initial loading-unloading at steady speed then loading-unloading simultaneously with acceleration-deceleration, loading-unloading under deceleration-acceleration, then step load requests to the limit of the system response capabilities.

The step-loading tests described are believed to cover most of the possible combinations of speed-torque application. More indicative results for the torque controller could have been obtained, had it been possible for the speed of the load to be kept constant by means of a synchronous motor.

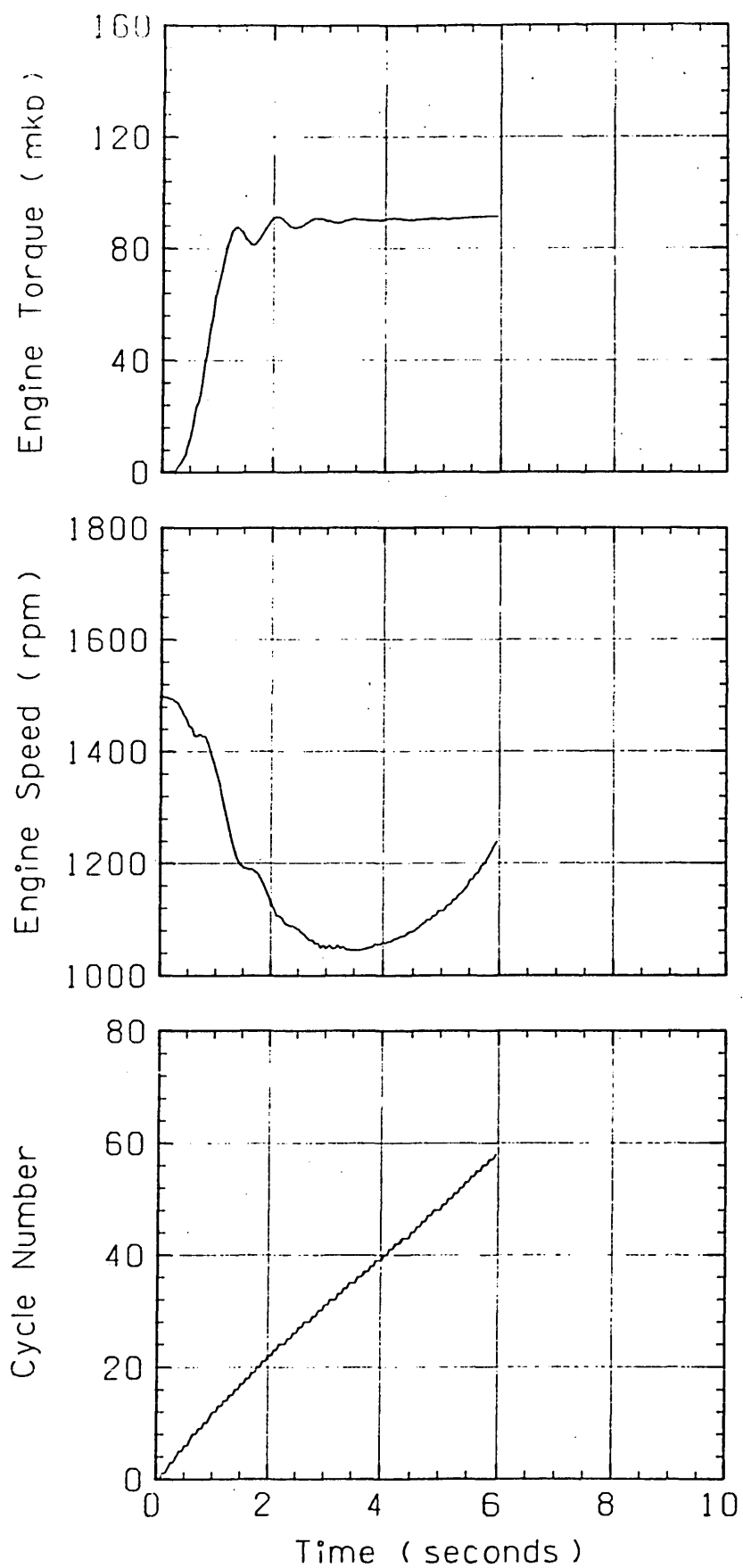


Fig. 7.7 LEYLAND test-bed step loading (90 mkp - 1500 RPM).

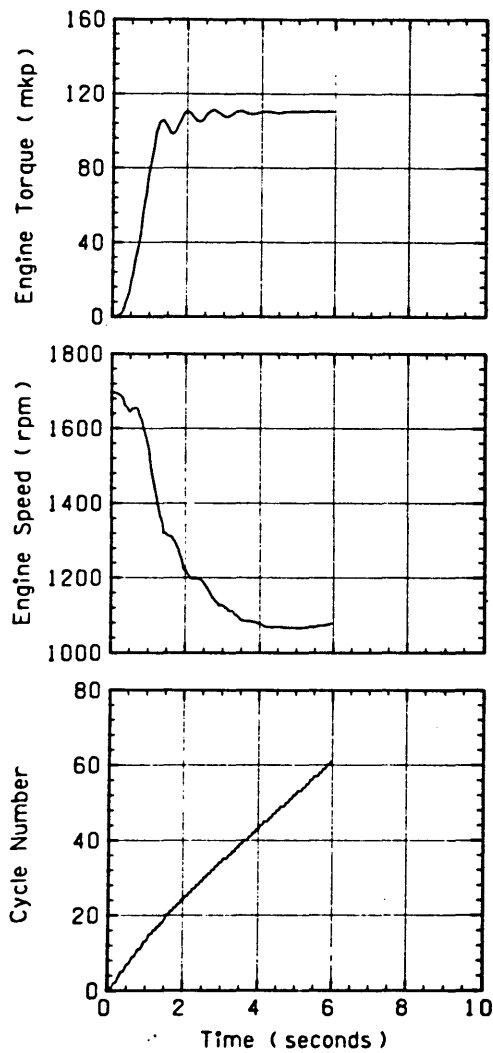


Fig. 7.8 LEYLAND test-bed step loading (110 mkp - 1700 RPM)

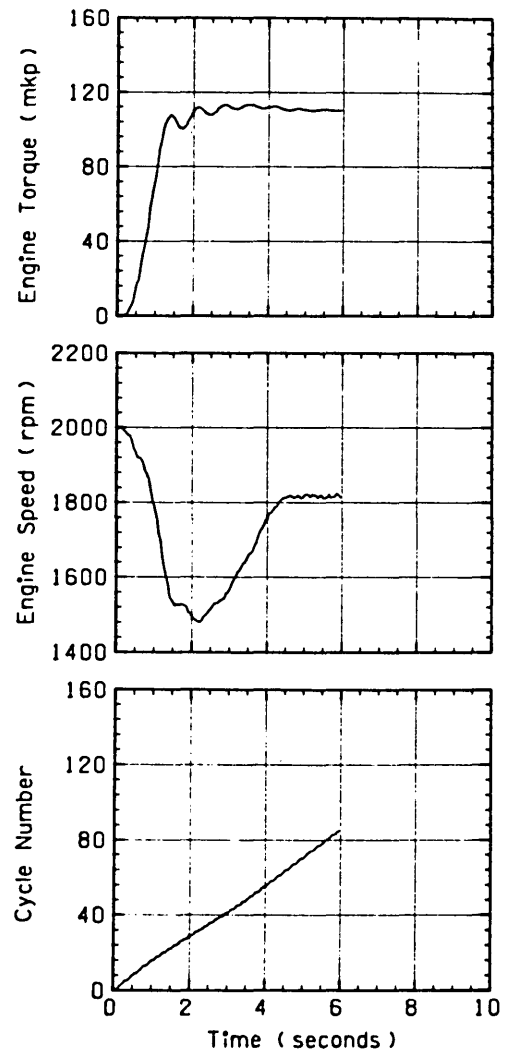


Fig. 7.9 LEYLAND test-bed step loading (110 mkp - 2000 RPM)

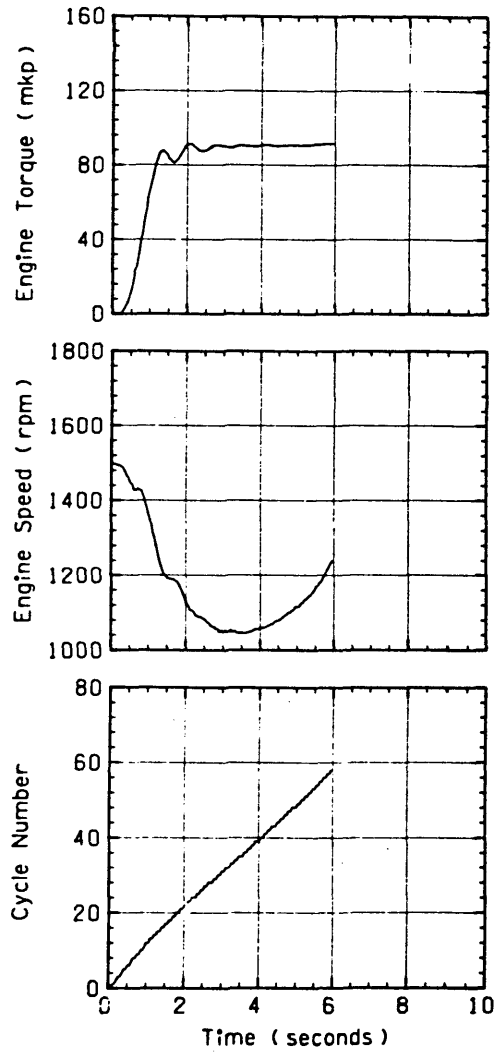


Fig. 7.10 LEYLAND test-bed
step loading (90 mkp
- 1500 RPM)

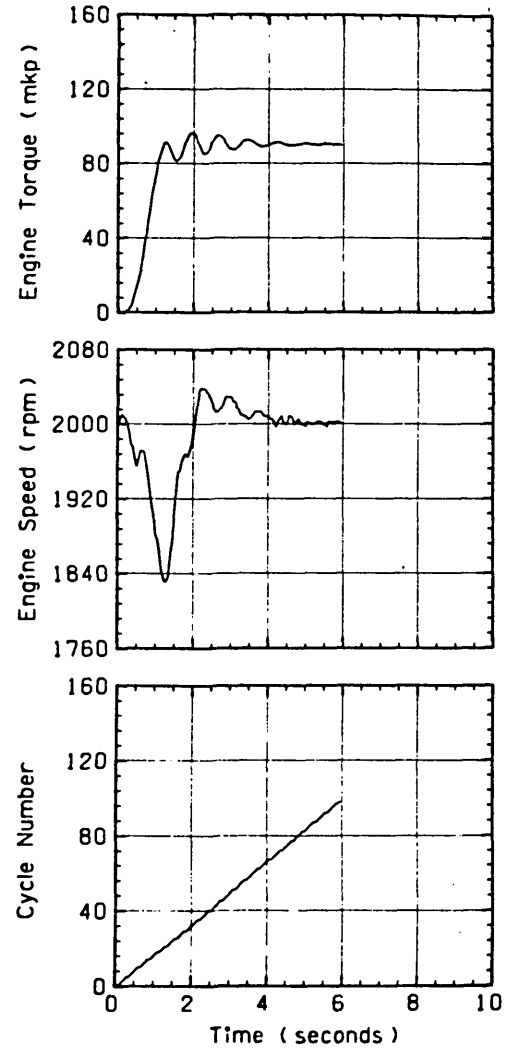


Fig. 7.11 LEYLAND test-bed
step loading (90 mkp
- 2000 RPM)

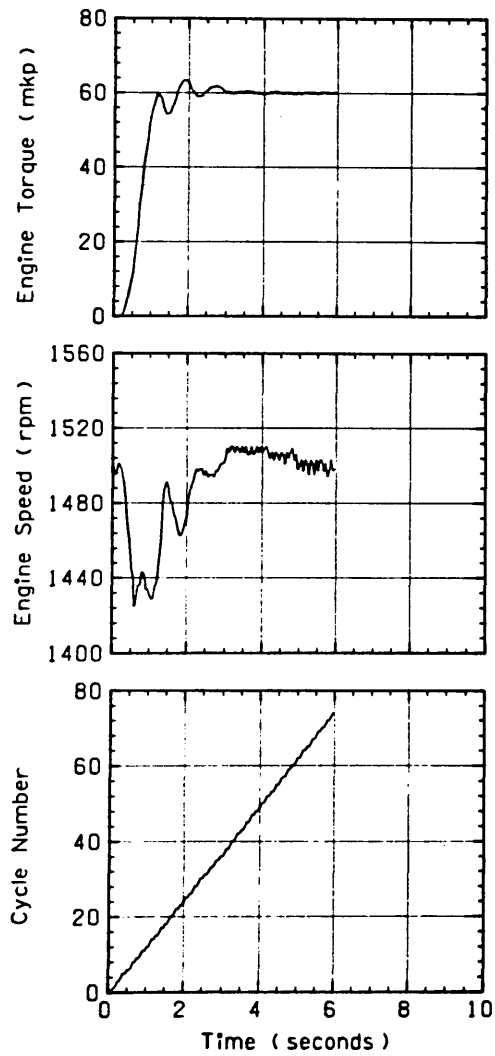


Fig. 7.12 LEYLAND test-bed
step loading (60 mkp
- 1500 RPM).

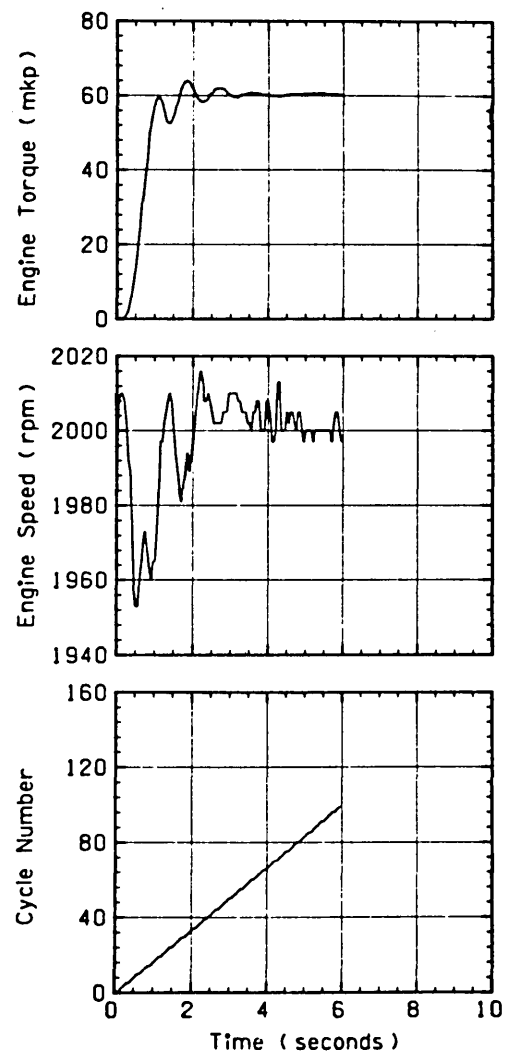


Fig. 7.13 LEYLAND test-bed
step loading (60 mkp
- 1500 RPM).

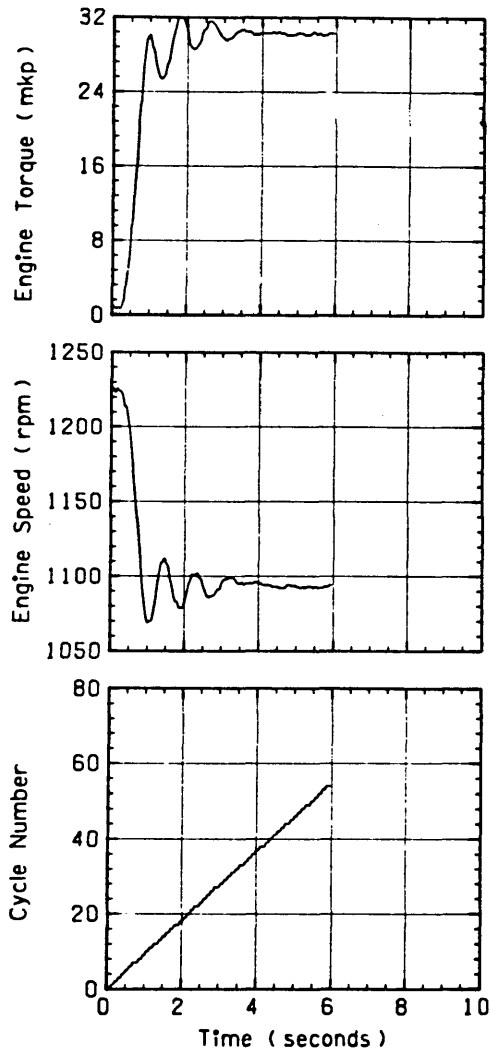


Fig. 7.14 LEYLAND test-bed
step loading (30 mkp
- 1200 RPM).

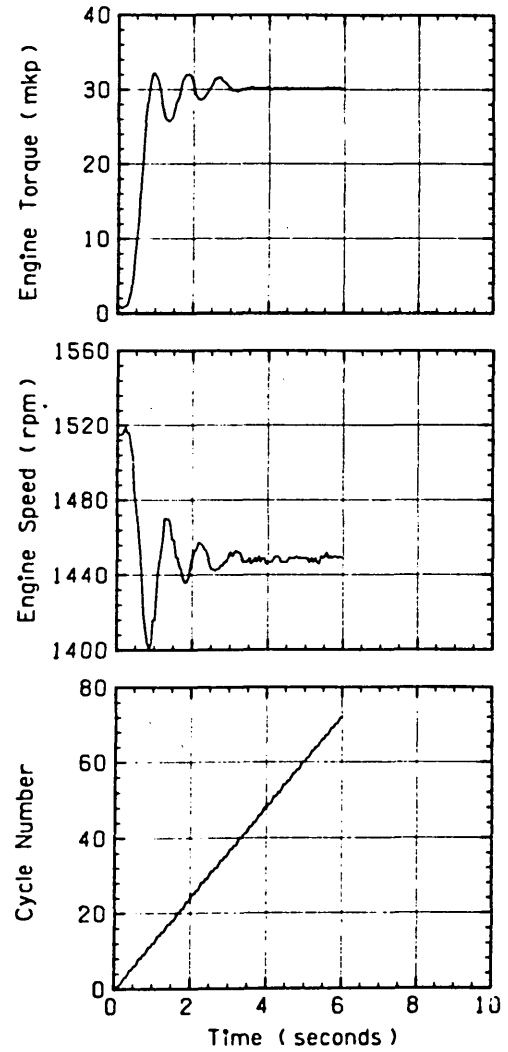


Fig. 7.15 LEYLAND test-bed
step loading (30 mkp
- 1500 RPM).

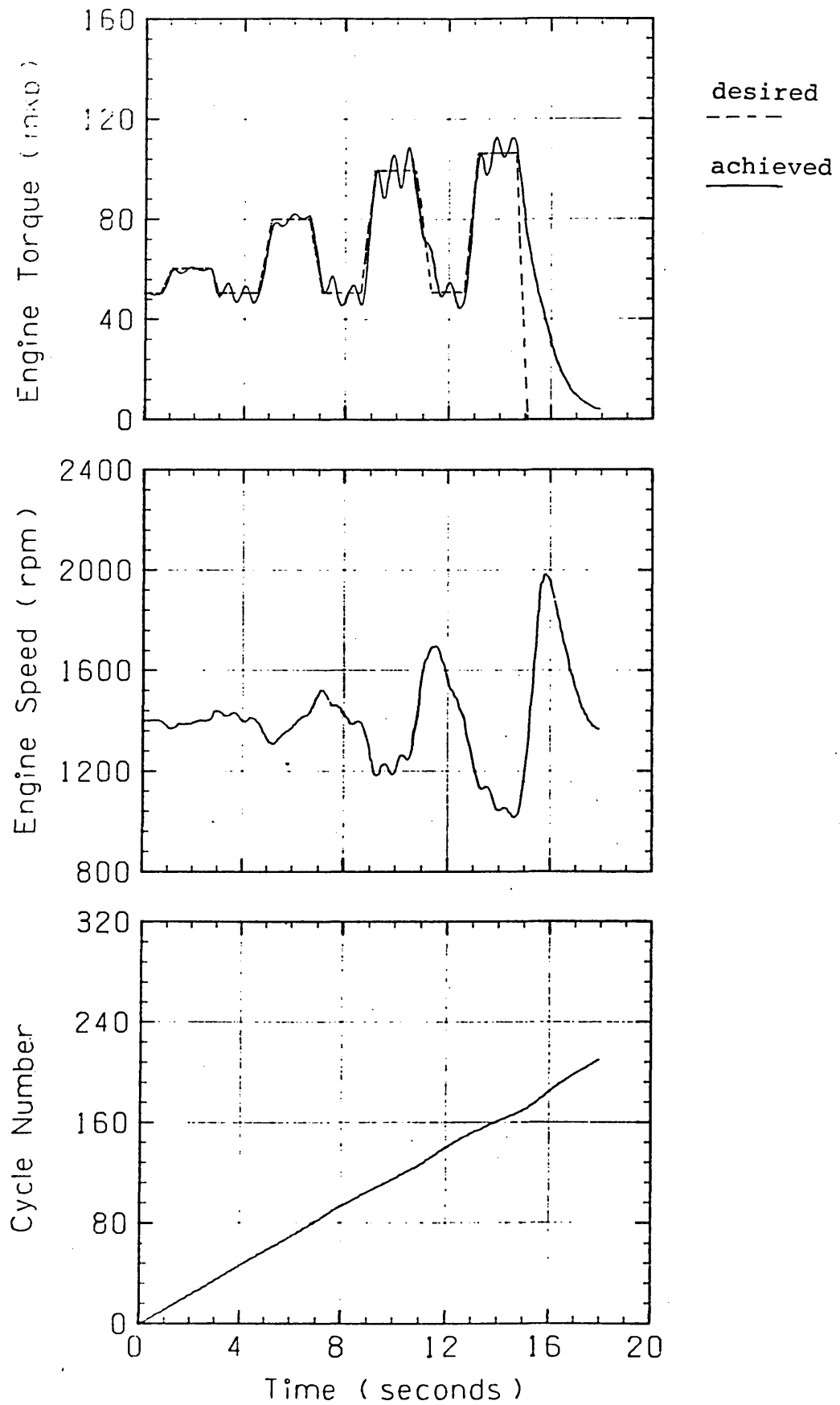


Fig. 7.16 LEYLAND test-bed step loading-unloading (1400 RPM).

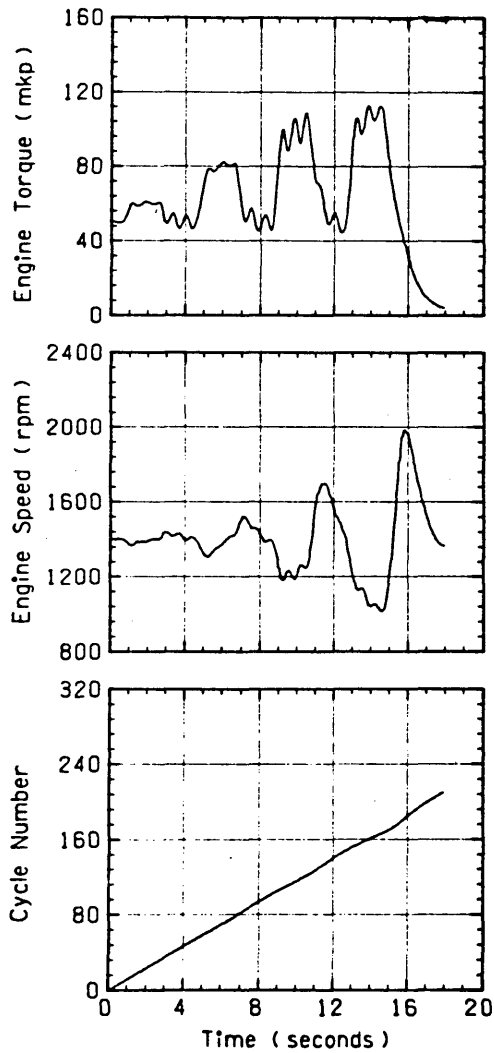


Fig. 7.17 LEYLAND test-bed
step loading-unloading
(1400 RPM).

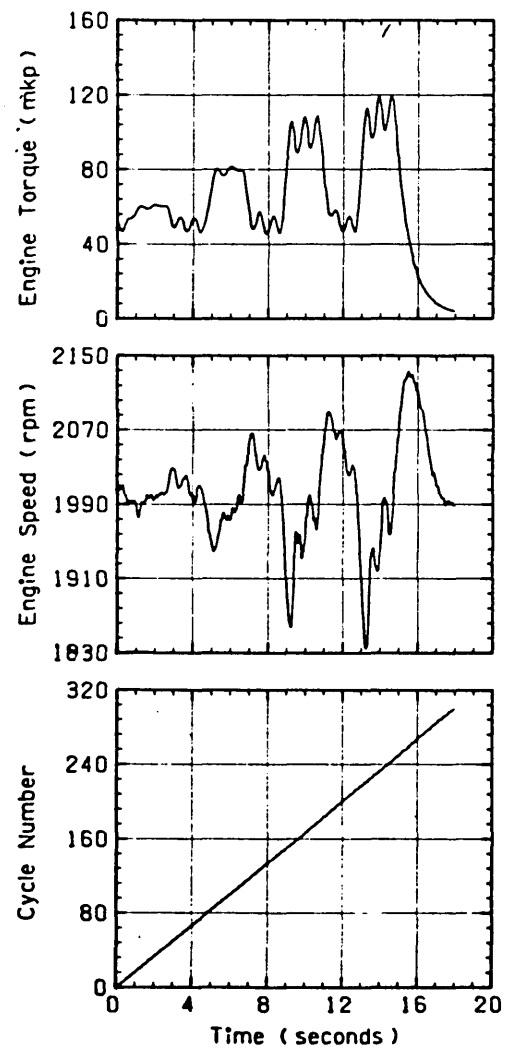


Fig. 7.18 LEYLAND test-bed
step loading-unloading
(2000 RPM).

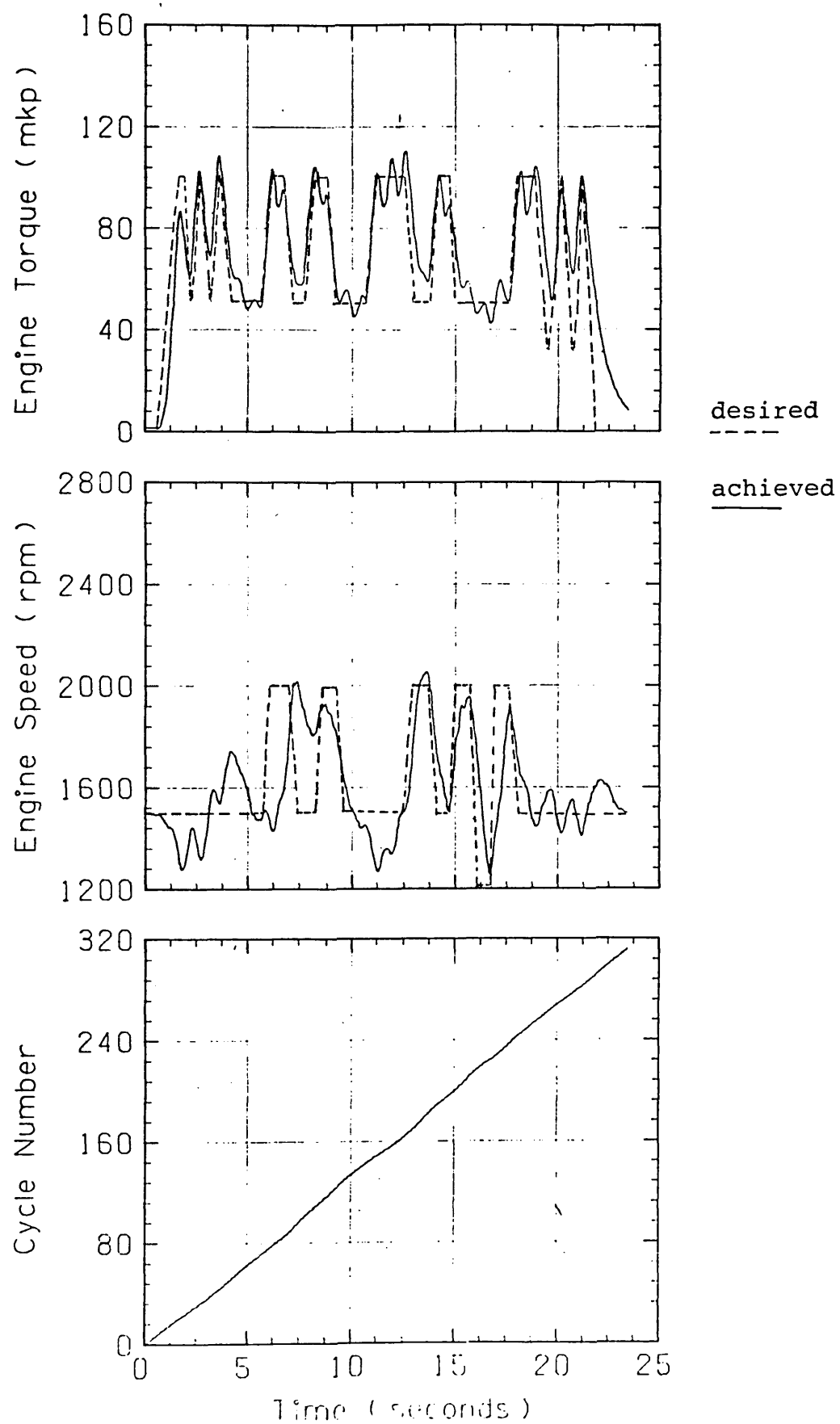


Fig. 7.19 LEYLAND test-bed rapid transient testing.

7.3 Disturbance Rejection Tests

An important controller characteristic is the ability to maintain a setpoint when external disturbances force a move from equilibrium.

Disturbance rejection tests have been carried out for the dynamometer controller by a constant torque request and rapid speed movements induced by the speed controller. Such tests are shown in Fig. 7.20 through 7.24. The maximum overshoot or undershoot for the PERKINS test bed was of the order of 10% of full scale load. This can be seen in Fig. 7.20 and 7.21. The overshoot/undershoot for the LEYLAND test-bed was of the order of 7% for an acceleration rate of about 1400 RPM/S. This is shown in Fig 7.22, 7.23 and 7.24. Shahin [S-12] reported an overshoot of 16% for a speed acceleration rate of about 600 RPM/s for exactly the same physical test bed (LEYLAND) using a hardware PID dynamometer controller.

The software speed controller has been provided with a torque feedback facility. This was expected to improve the engine speed stability during sudden load changes. Tests were carried out on the PERKINS test bed to confirm this behaviour. In Fig. 7.25 and 7.26 the engine controller is requested to keep a constant 1200 RPM setting while a 22 mkp load is suddenly applied then removed. The response without feedback shows a peak-to-peak speed fluctuation of 420 RPM whereas the equivalent figure with torque feedback is 220 RPM which amounts to almost halving the fluctuation.

Load disturbance rejection tests were not carried out on the LEYLAND test bed as the on-engine governor would have masked the software controller behaviour.

7.4 Power Law Tests

These tests were generated by a selection of constants for a third order speed polynomial.

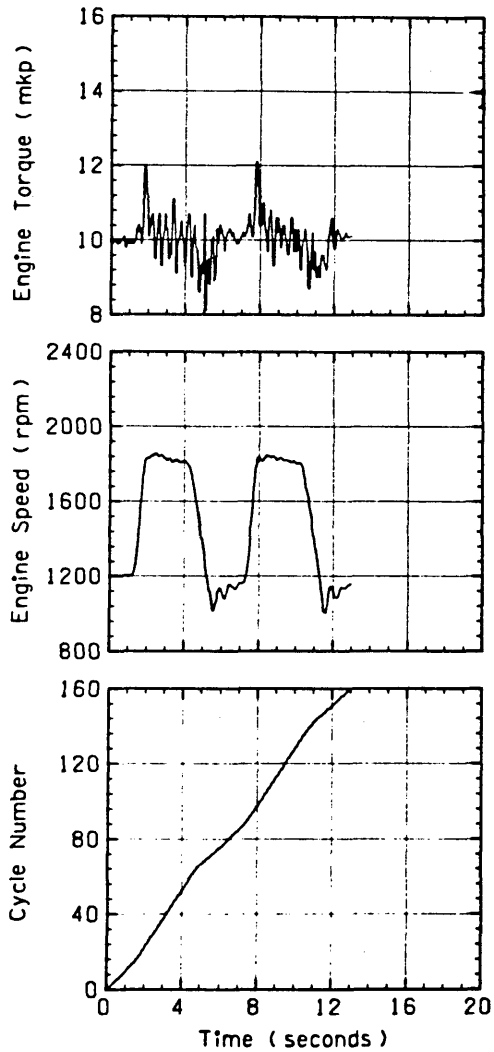


Fig. 7.20 PERKINS test-bed speed disturbance rejection (10 mkp).

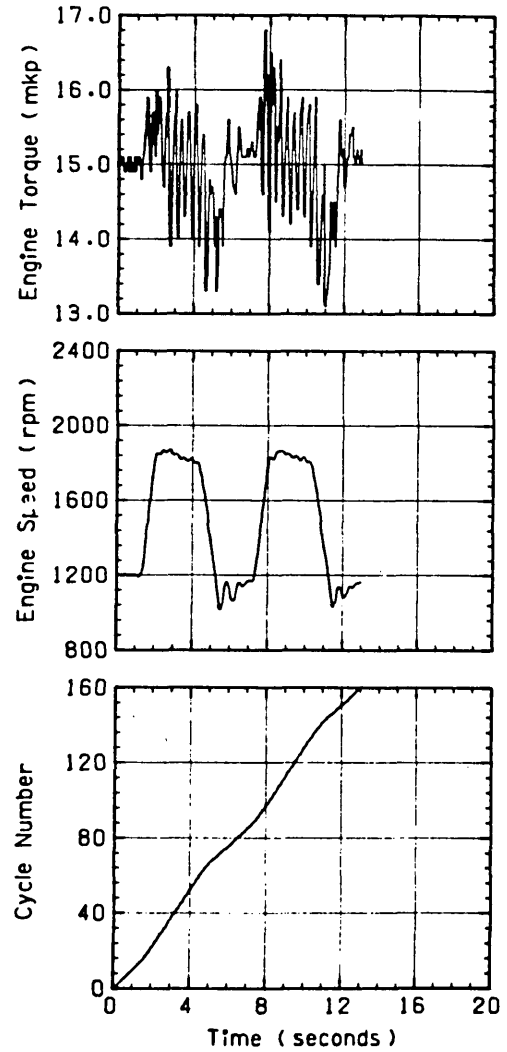


Fig. 7.21 PERKINS test-bed speed disturbance rejection (15 mkp).

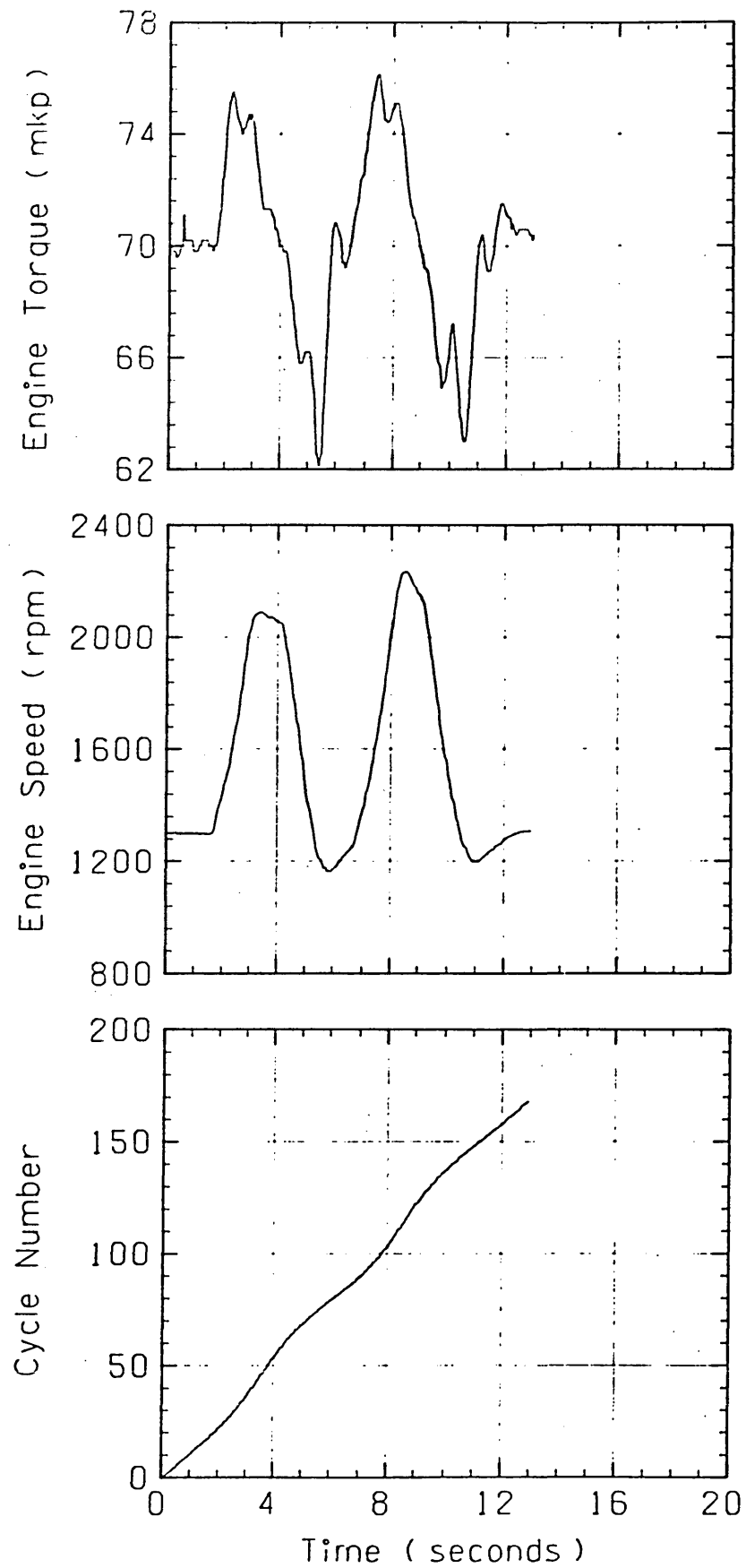


Fig. 7.22 LEYLAND test-bed speed disturbance rejection (10 mkp).

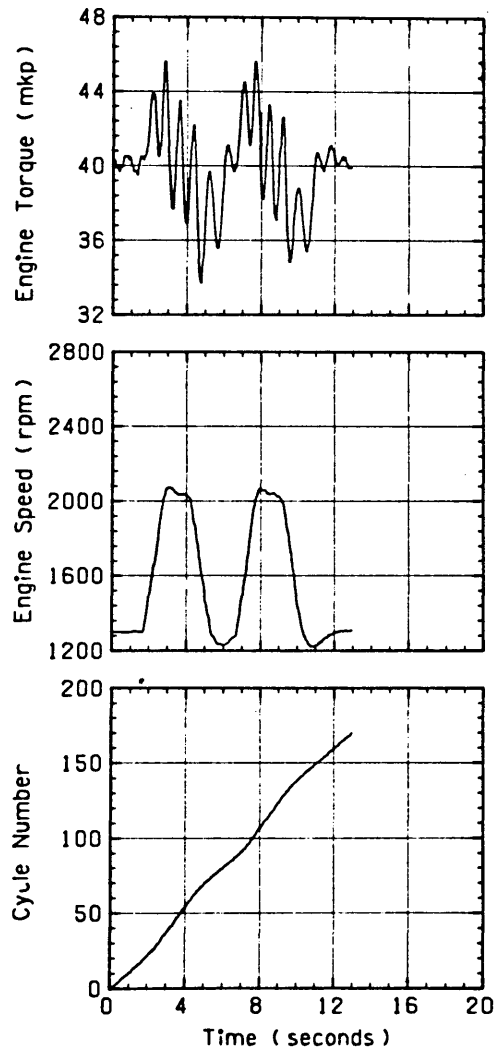


Fig. 7.23 LEYLAND test-bed speed disturbance rejection (40 mkp).

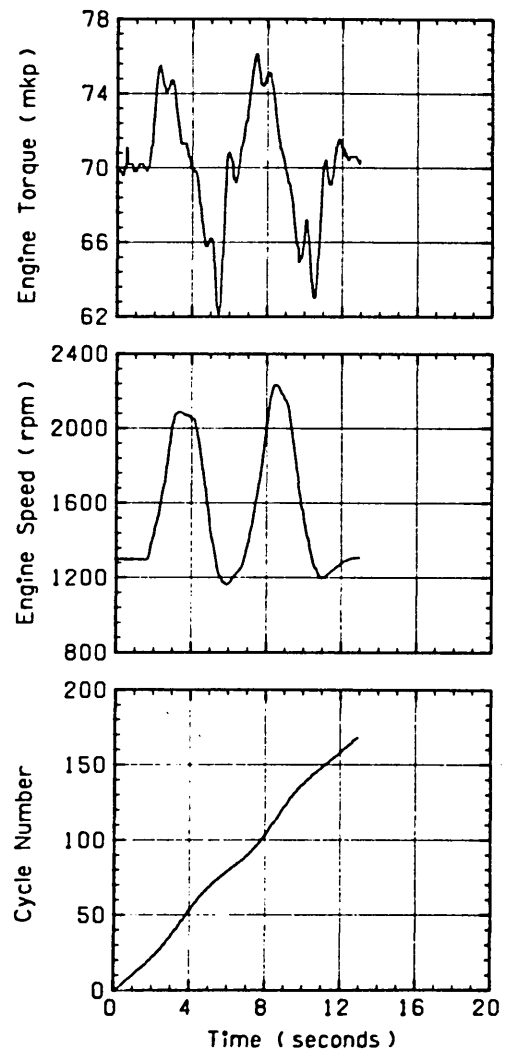


Fig. 7.24 LEYLAND test-bed speed disturbance rejection (70 mkp).

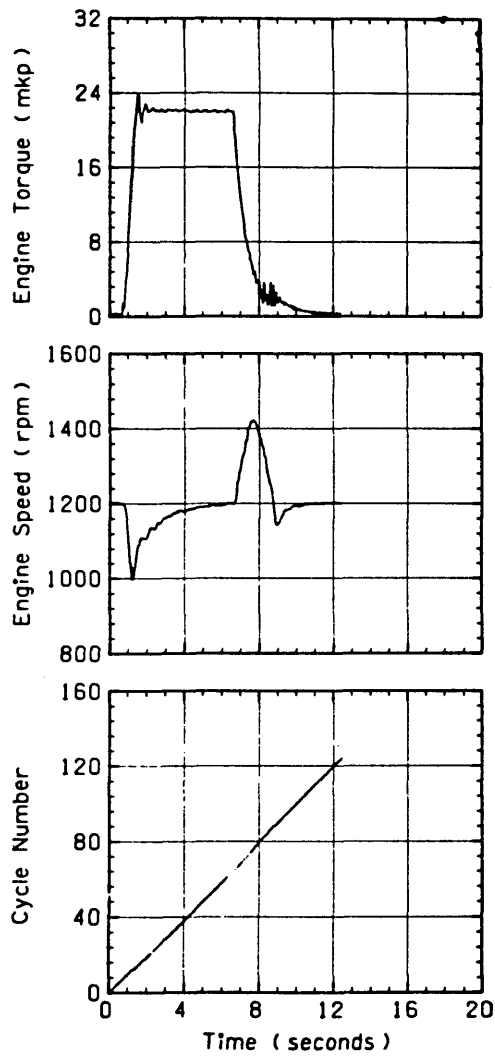


Fig. 7.25 PERKINS test-bed load disturbance rejection without torque feedback.

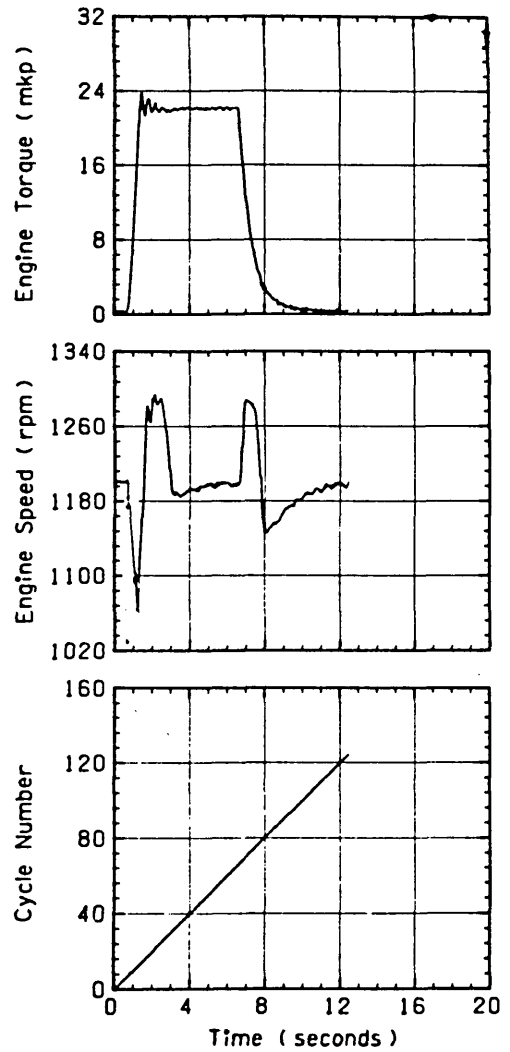


Fig. 7.26 PERKINS test-bed load disturbance rejection with torque feedback.

The torque is given as:

$$T(\text{mkp}) = C_0 + C_1 \cdot \frac{\text{RPM}}{1000} + C_2 \cdot \left[\frac{\text{RPM}}{1000} \right]^2 + C_3 \cdot \left[\frac{\text{RPM}}{1000} \right]^3 \quad (107)$$

where C_0 , C_1 , C_2 , C_3 are test selectable constants. For test 7.27 these constants were:

$$C_0 = -5.70 \quad C_1 = 2.0 \quad C_2 = 1.50 \quad C_3 = 2.21$$

A speed schedule was selected as seen in Fig. 7.27 and the torque controller computed the instantaneous torque according to (107). The torque input to the controller is shown as the desired torque versus the output which is plotted as achieved (measured) torque. The speed controller is in a full PID mode. The desired and achieved torque have a lag which is due to the torque controller lag. There is no lookahead for this test as the torque controller responds to the instantaneous torque demand derived from (107) through the instantaneous speed measurement.

For test 7.28 on the LEYLAND test-bed the constants selected were:

$$C_0 = 30.0 \quad C_1 = 0 \quad C_2 = 20.0 \quad C_3 = 0$$

These constants correspond to a second order torque characteristic. By choosing a linear speed increase and decrease as an input, a parabola would be expected for the torque. Test results (Fig. 7.28) confirm our expectation. The lag between input and measured torque is smaller than in 7.27 mainly because the rate of speed increase is smaller.

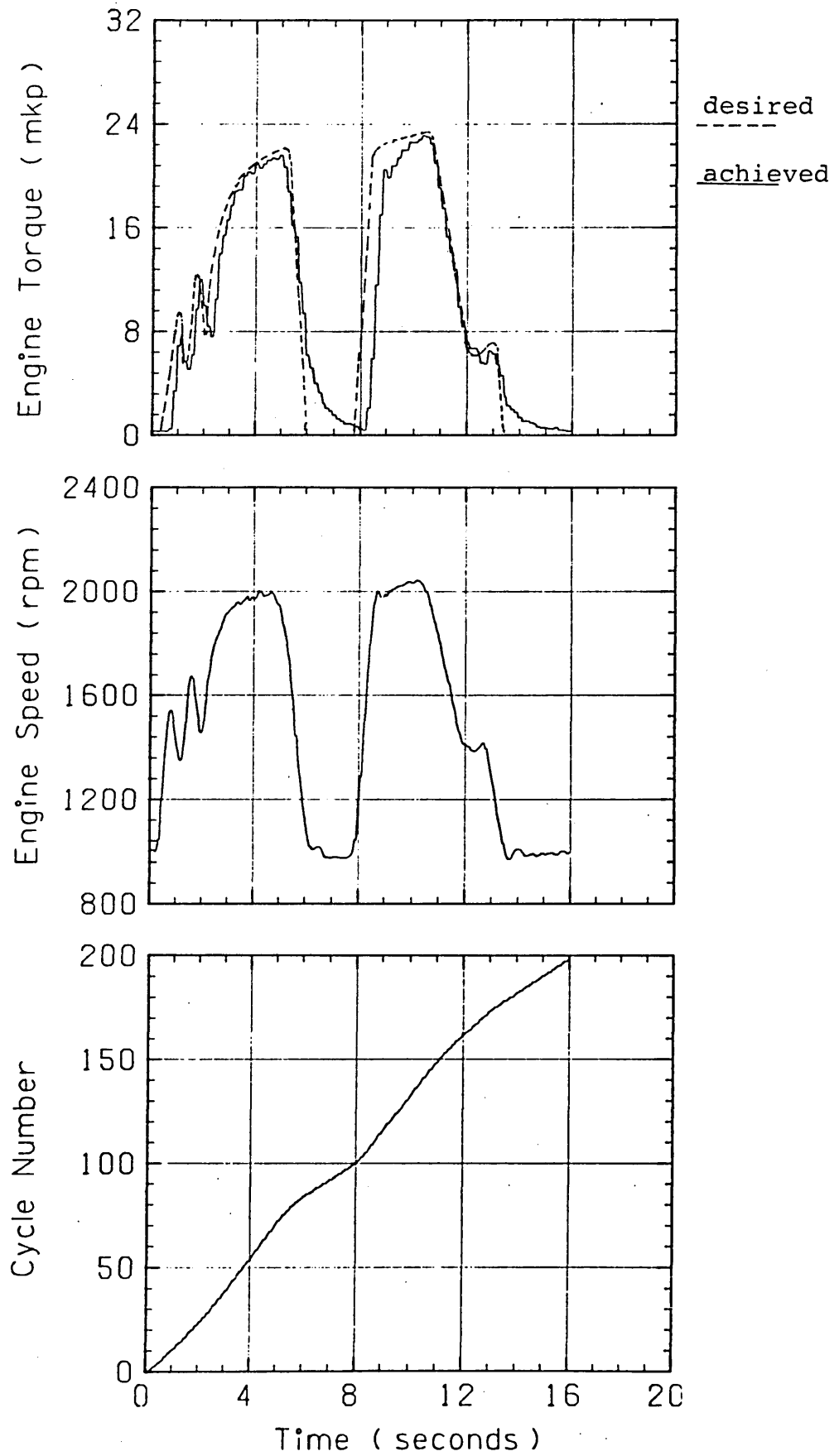


Fig. 7.27 PERKINS test-bed Power law test.

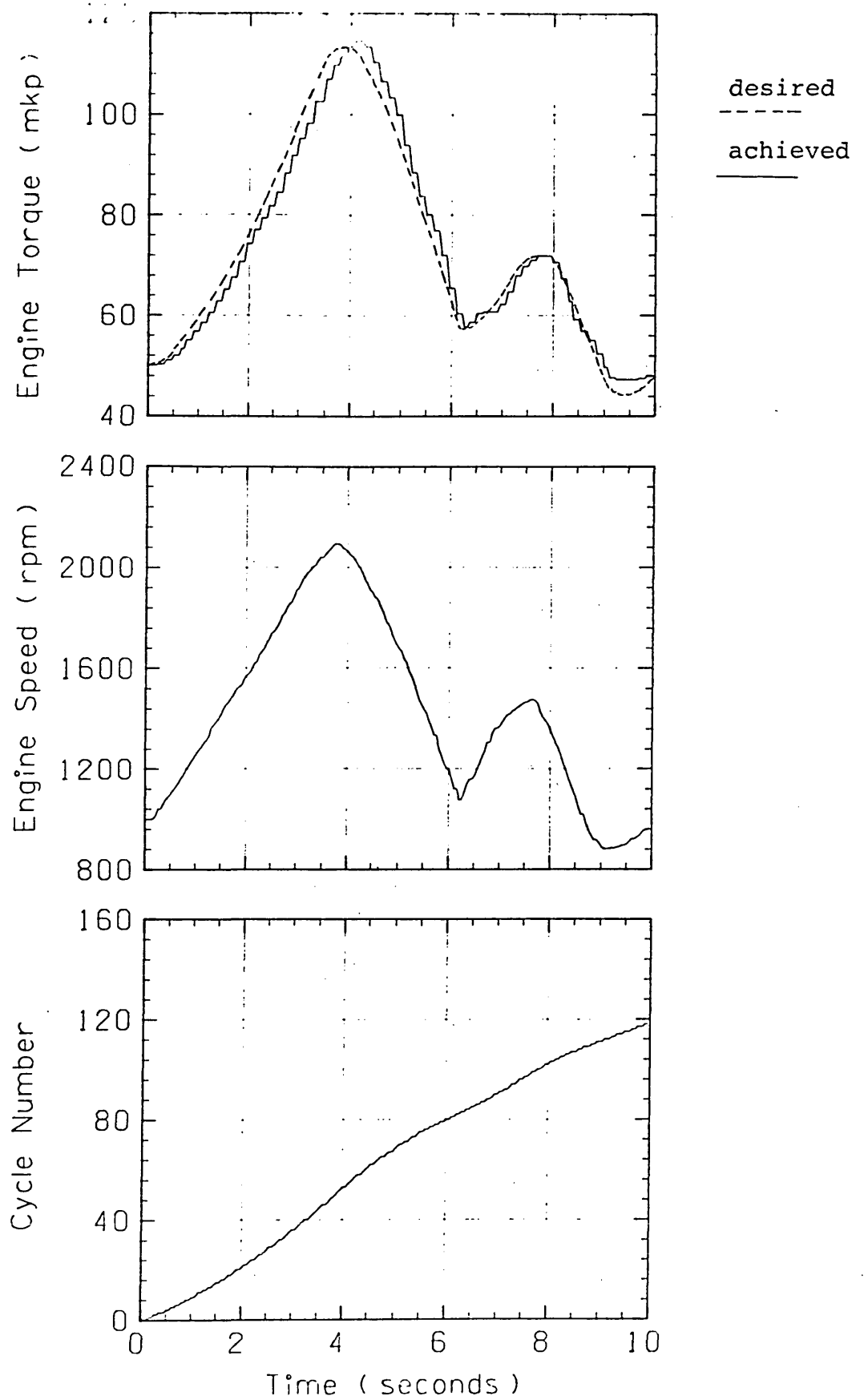


Fig. 7.28

LEYLAND test-bed Power law test.

7.5 Speed Control Mode Tests

The speed control mode tests were divided in two categories:

- a) follow a certain speed schedule when the throttle remains constant.
- b) follow a certain speed schedule when the throttle moves independently.

The speed control mode is based on a speed PID controller superimposed on the load time-optimal one. Tuning of the speed controller is effected by the step loading speed response dead-time and rate of increase. Tuning was almost imposssible for the LEYLAND test-bed for the reasons already stated in 7.1 and 7.2.

Fig. 7.29 shows a category (a) test. The speed tends to reach the 1200 level around 2 seconds with a torque of about 18 mkp. The 1000 RPM level is not reached as the required torque for such a speed is in excess of 24 mkp which is the PERKINS test-bed recommended load limit.

Speed control tests depend very much on the tuning of the speed PID controller. Fig. 7.31 through 7.33 show the same category (b) test with different PID coefficient combination. The effect of the integral and derivative coefficients to the response is obvious.

Although tuning was not possible for the LEYLAND test-bed a category (a) test has been performed and the results shown on fig. 7.34. The performance, although unsatisfactory, confirms the speed control underlying priciple, for this test-bed.

7.6 Tracking controller tests.

All the tests described up to now were conducted without the tracking version of the controller. The difference between the

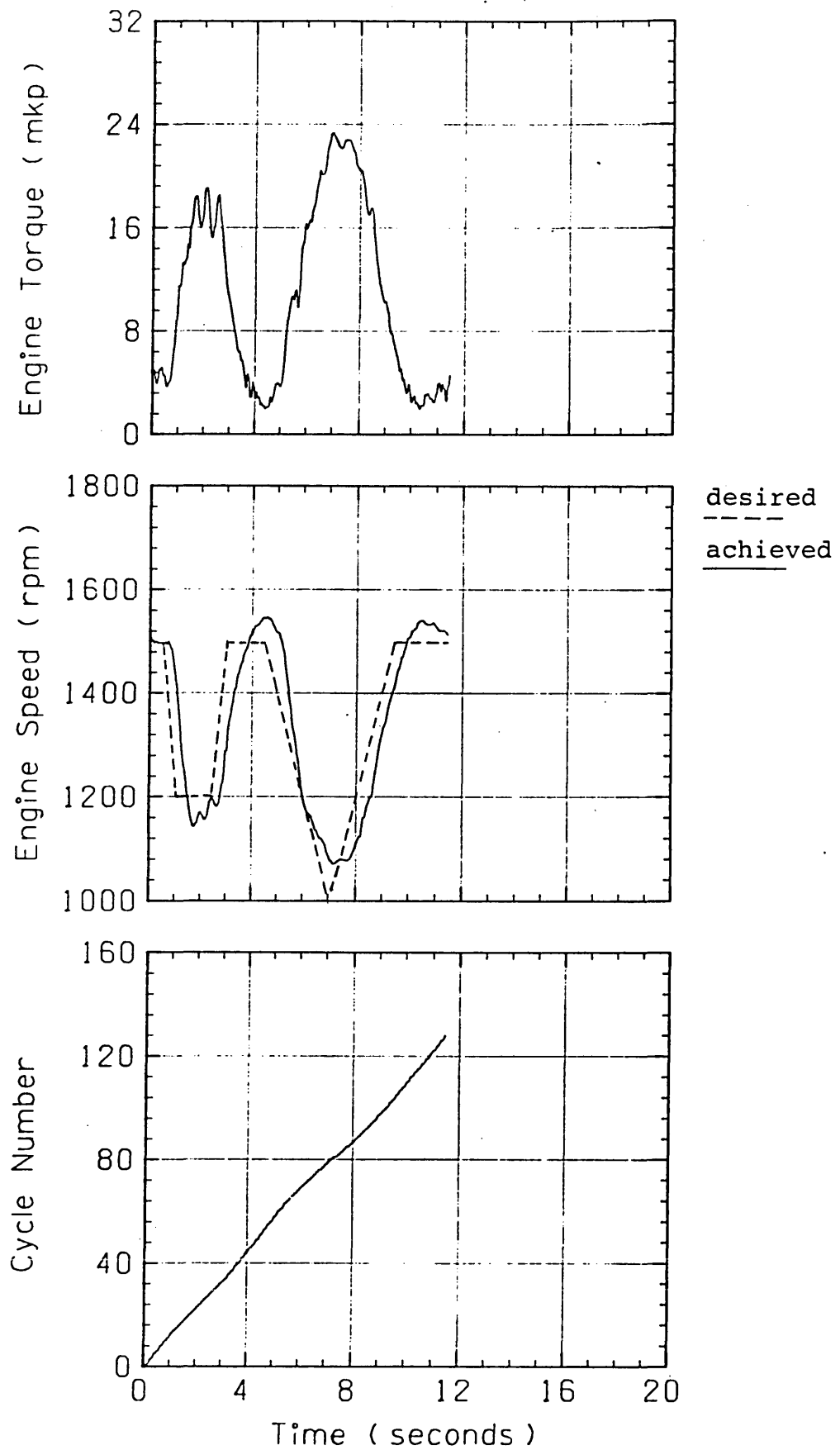


Fig. 7.29

PERKINS test-bed speed control with constant rack position.

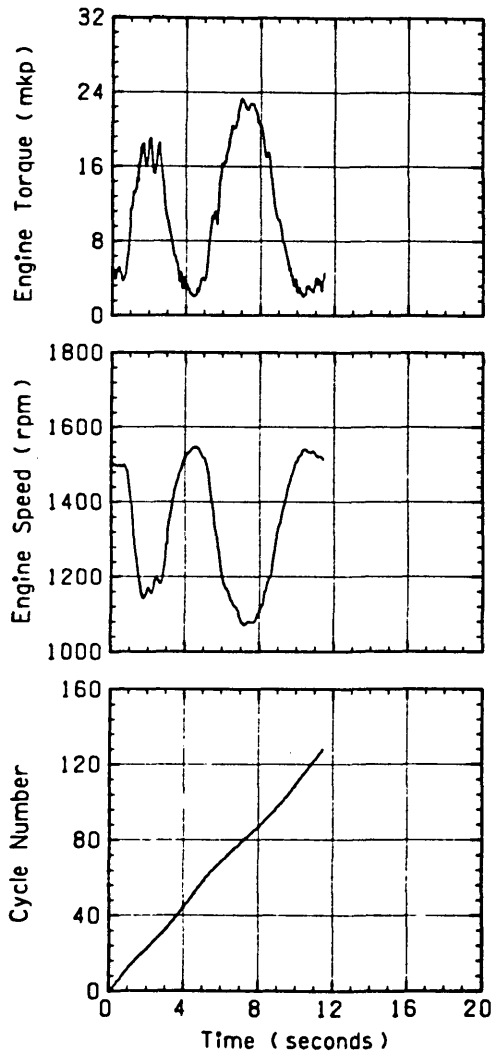
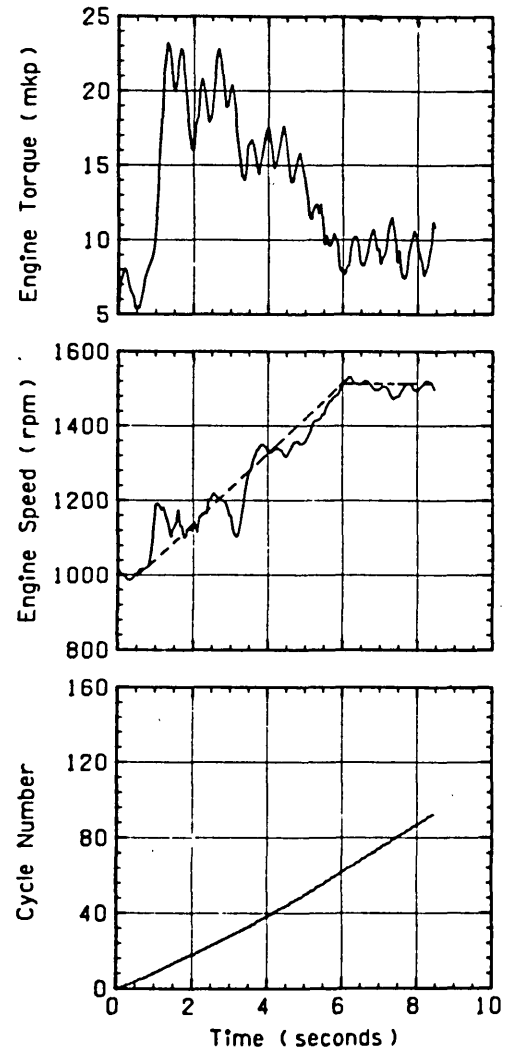


Fig. 7.30 PERKINS test-bed speed control with constant rack position.



---- desired

— achieved

Fig. 7.31 PERKINS test-bed speed control with a step in rack position
(PID = $.72/.062/2.65$).

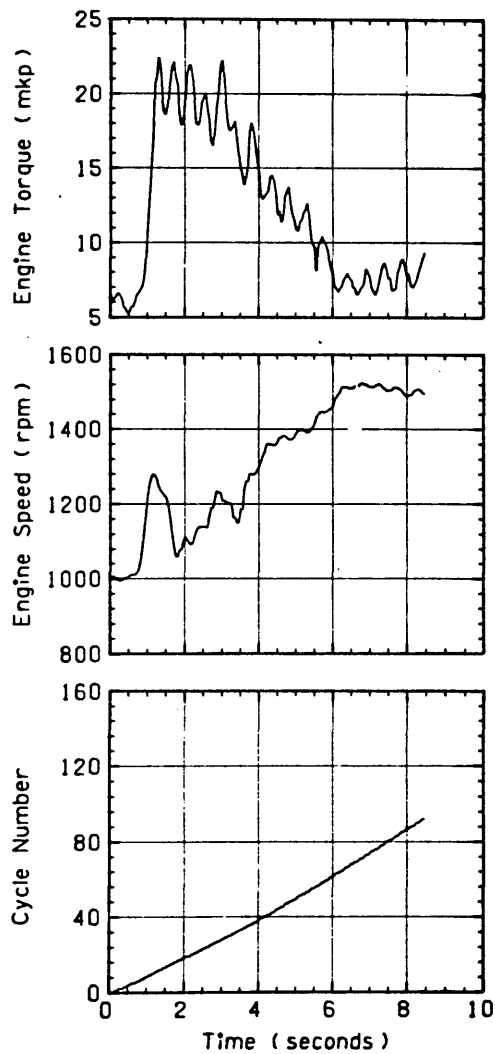


Fig. 7.32 PERKINS test-bed
speed control with
step in rack position
(PID = .065/.042/2.65).

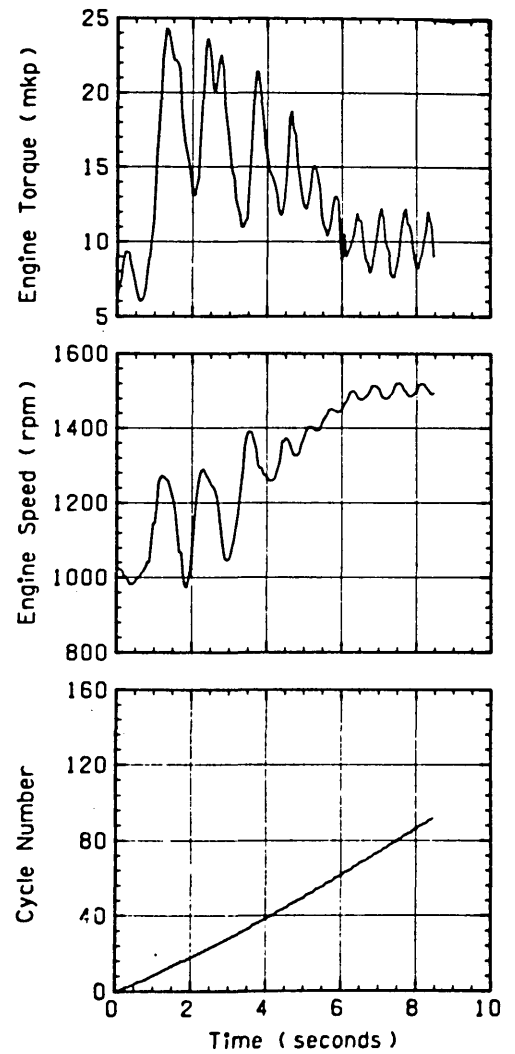
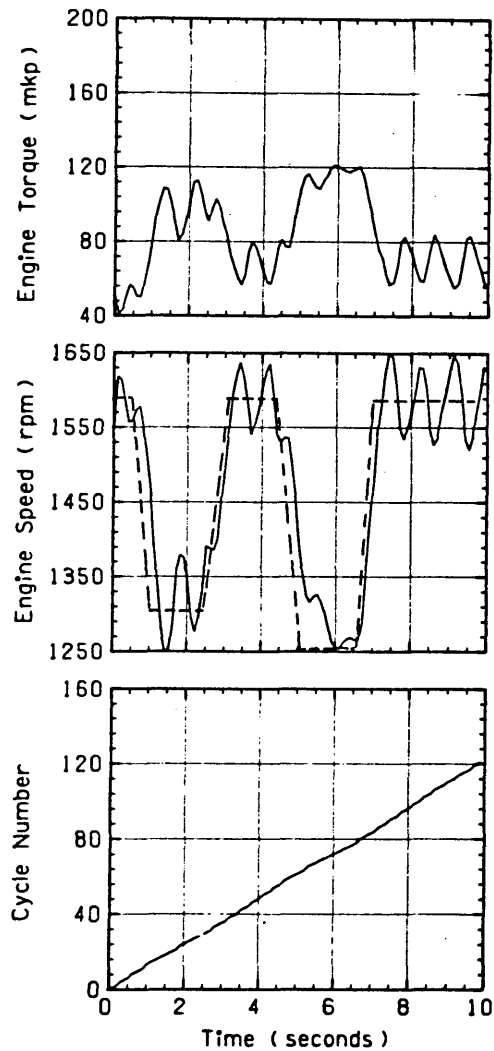


Fig. 7.33 PERKINS test-bed
speed control with
step in rack position
(PID = .83/.086/2.4).



---- desired
 ——— achieved

Fig. 7.34 LEYLAND test-bed speed control with constant throttle position.

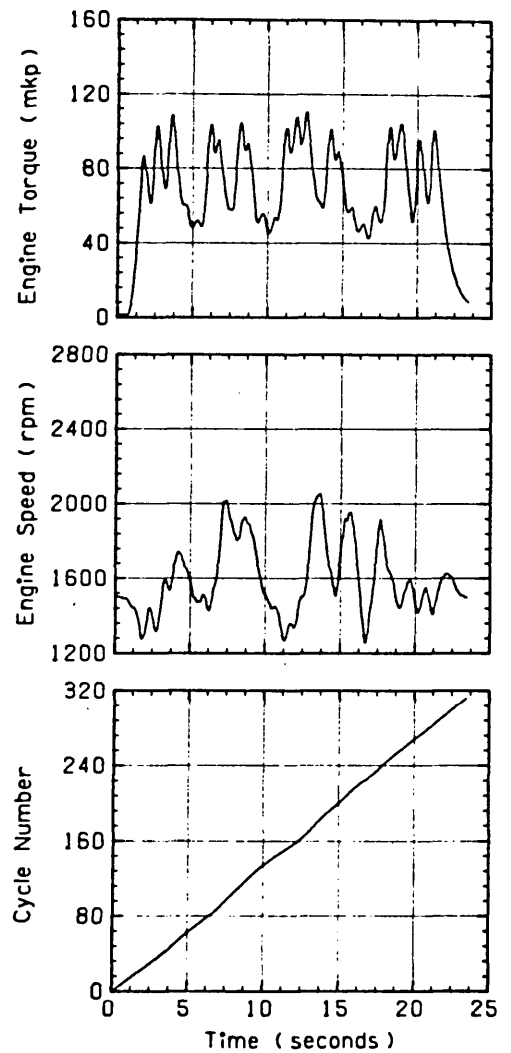


Fig. 7.35 LEYLAND test-bed rapid transient testing.

tracking and non-tracking versions of the torque controller is that without tracking a half second preview is performed whereas with tracking a two second preview action is used with an update every half second. This was expected to give smoother torque transients. The disadvantage of the tracking controller is an initial assumption about the speed transient behaviour which, although compensated, if large, will distort tracking action.

Fig. 7.36 and 7.37 show the same transient schedule under constant engine speed request, with the two controllers. The requested schedule is a linear increase and decrease of torque. The response during the tracking transients is smoother especially when torque is reduced. The same pattern of behaviour is repeated in Fig. 7.38 and 7.39 on the LEYLAND test-bed, confirming the original expectations.

7.7 Federal Cycle Tests

Automatic test equipment has the advantage of good repeatability. This has been demonstrated by multiple runs of the same transient test and analysis of the results. The selected test schedule was the first part (1-90 seconds) of the proposed USA Federal Smoke and Emissions test to be enforced in 1983 (EPA Engine-dynamometer for heavy-duty Diesel Engines). The motoring parts of the cycle could not be reproduced with a power absorbing only dynamometer. A no-load condition has been assumed where motoring was indicated. Eight consecutive tests were conducted on each test bed and the results analysed without any selection process, in order to evaluate the overall statistical behaviour of the test-beds.

Fig. 7.40 shows the torque schedule on the PERKINS test bed. The previously achieved [W-10] torque on exactly the same engine-dynamometer combination is also plotted and used as a

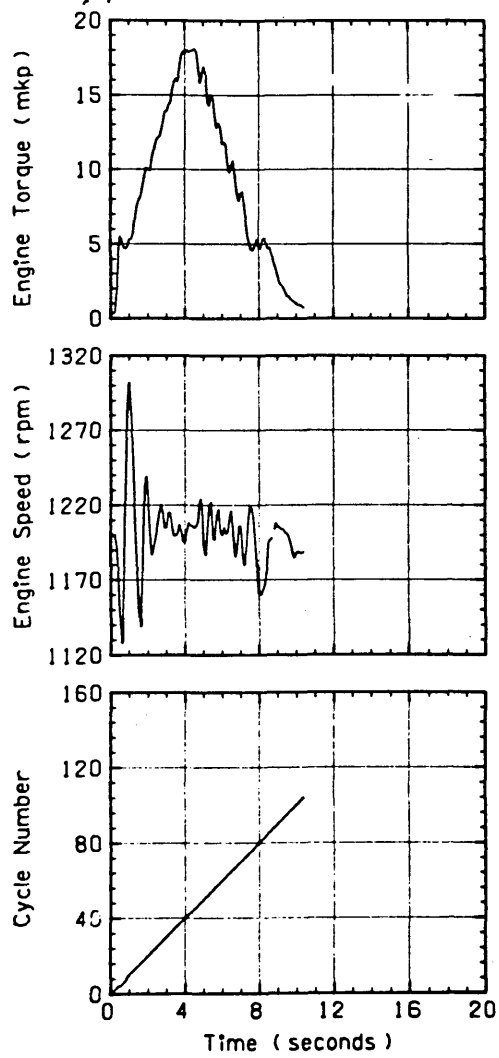


Fig. 7.36 PERKINS test-bed non-tracking transient.

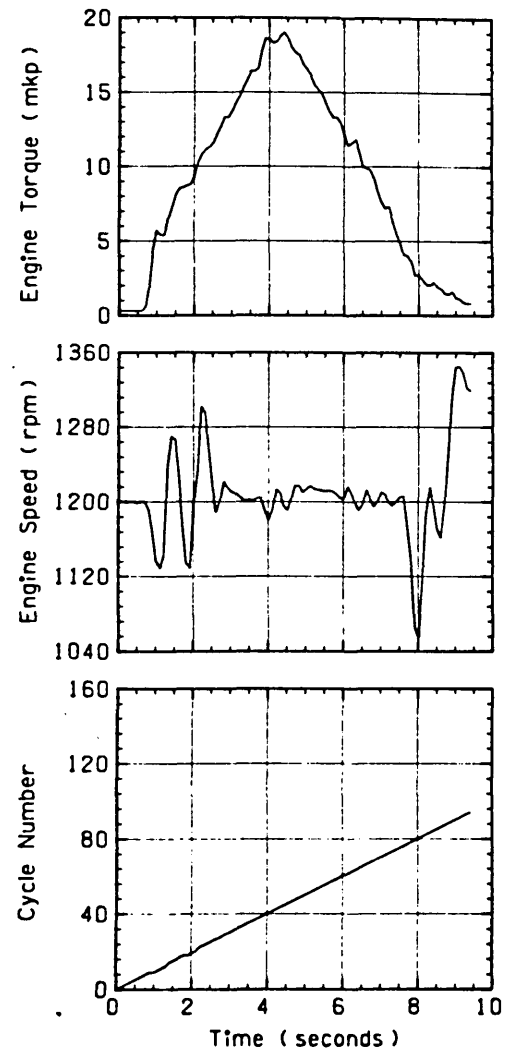


Fig. 7.37 PERKINS test-bed tracking transient.

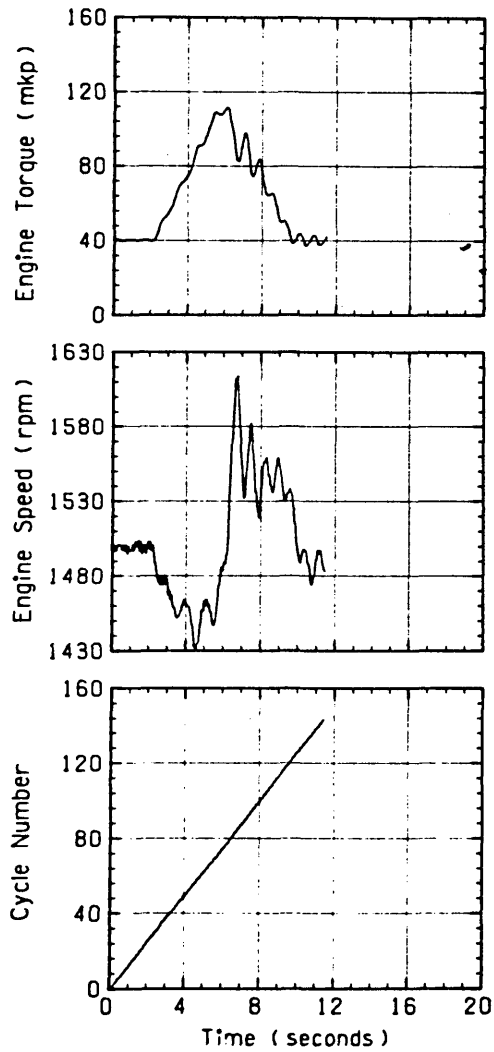


Fig. 7.38 LEYLAND test-bed non-tracking transient.

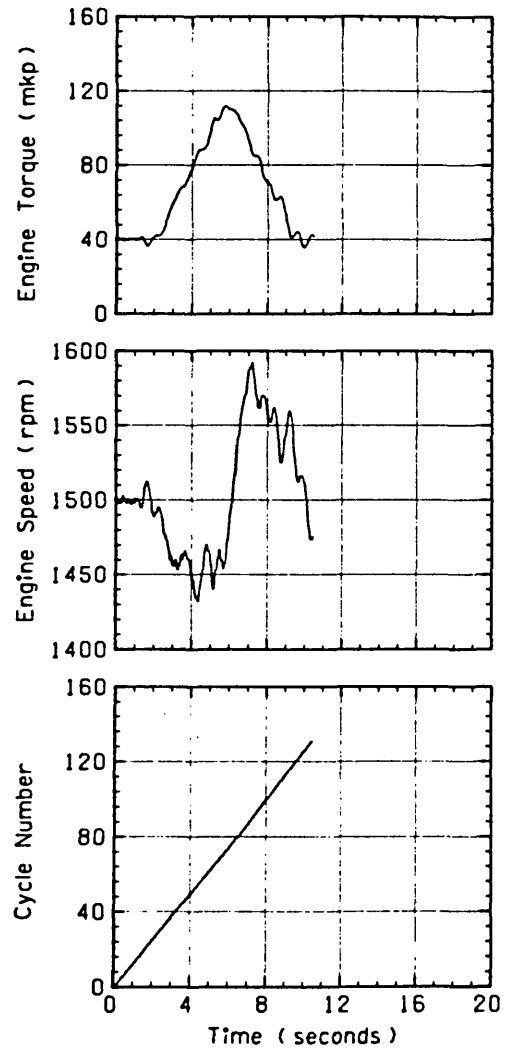


Fig. 7.39 LEYLAND test-bed tracking transient.

reference for comparisons of controller behaviour.

Fig. 7.41 and 7.42 show that, in general, the currently achieved maximum and minimum relative errors are less than the ones achieved previously. The peaks on the currently achieved relative errors correspond to minimal absolute errors and therefore are of no importance.

The absolute error plot of Fig. 7.43 confirms that in general the maximum currently achieved absolute error is better than the mean error achieved previously. Fig. 6.2 displayed the torque reading scatter for the PERKINS test-bed. Fig. 7.44 displays the standard deviation of the measurements (based on eight samples, one from each test) together with the error over the whole test interval. The scale of the standard deviation in fig. 7.44 is in percentage terms, the ratio of the standard deviation torque derived by using the eight tests at the same point in time over the maximum scale torque which for the PERKINS test bed was 22 mkp. It is obvious that the peaks of torque measurement standard deviation coincide with the error peaks. This indicates a measurement problem rather than an inability to follow the test sequence. This argument has been further validated by tests on the LEYLAND test-bed.

Fig. 7.45 displays the speed control performance. The previously achieved speed is not shown because of different assumptions regarding speed during motoring. Fig. 7.47 shows the comparative errors outside of the motoring zones. The currently achieved error is in general superior to the previously achieved one though the comparison cannot be perfectly valid because previously an idle speed was assumed while motoring whereas the schedule speed was assumed for the present tests. The peaks (Fig. 7.49) of the speed measurement standard deviation coincide with standard deviation peaks in fig. 7.44. This indicates the interaction between torque application and

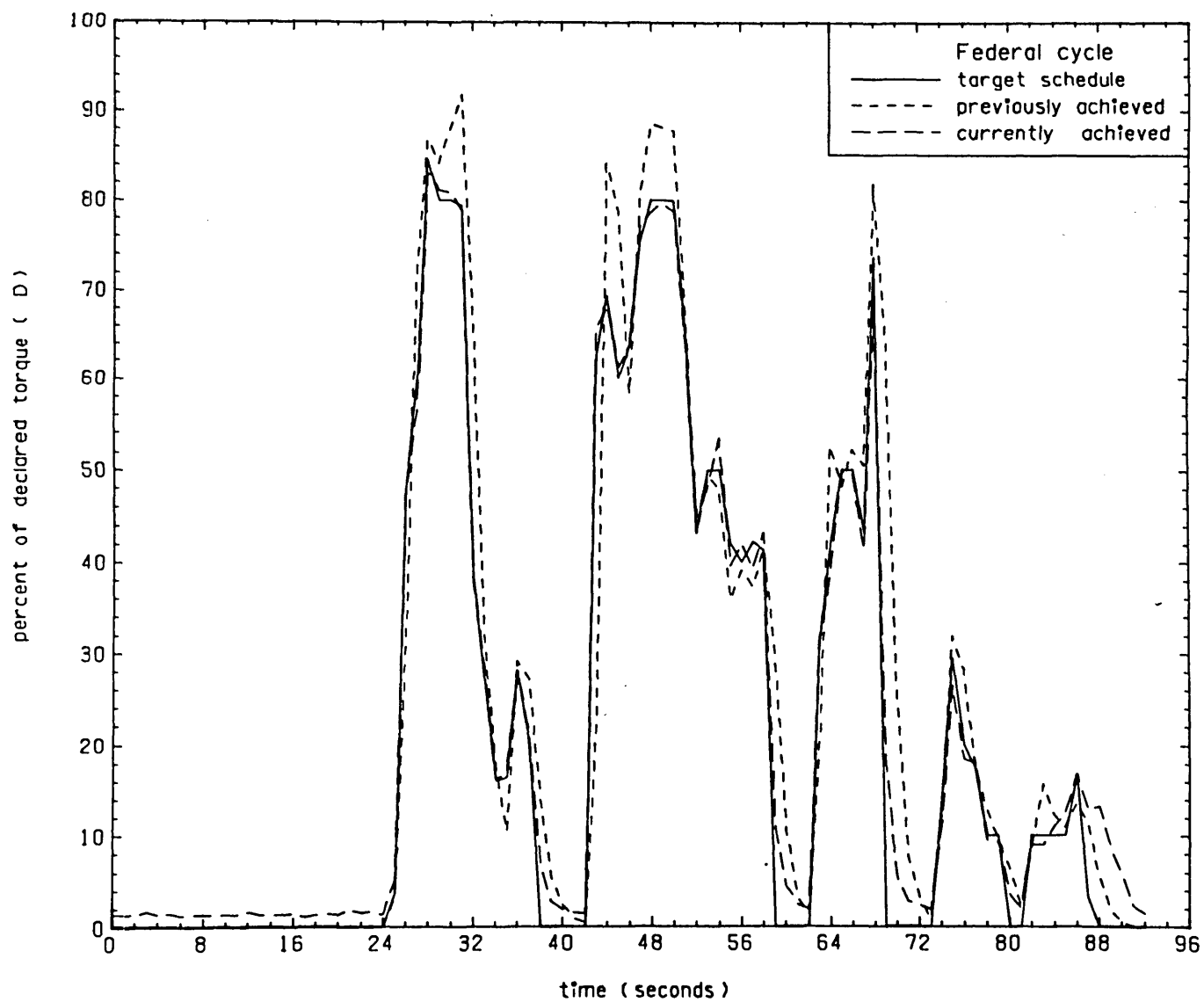


Fig. 7.40 PERKINS test-bed - Federal cycle torque schedule.

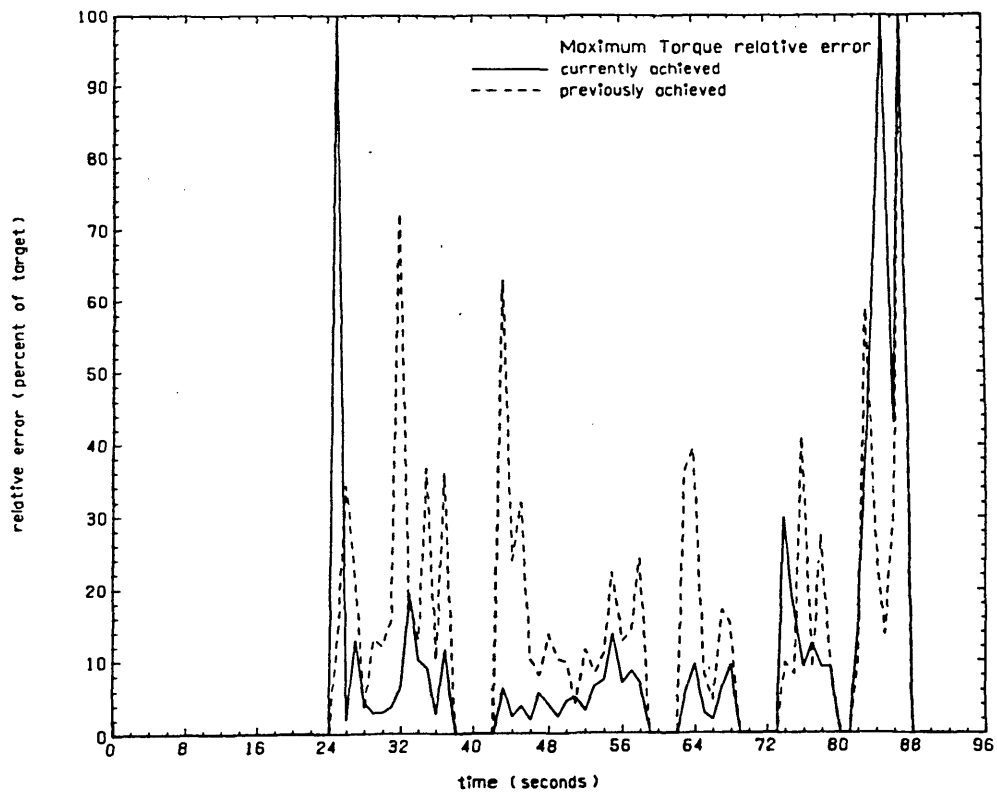


Fig. 7.41 PERKINS test-bed - Federal Cycle analysis

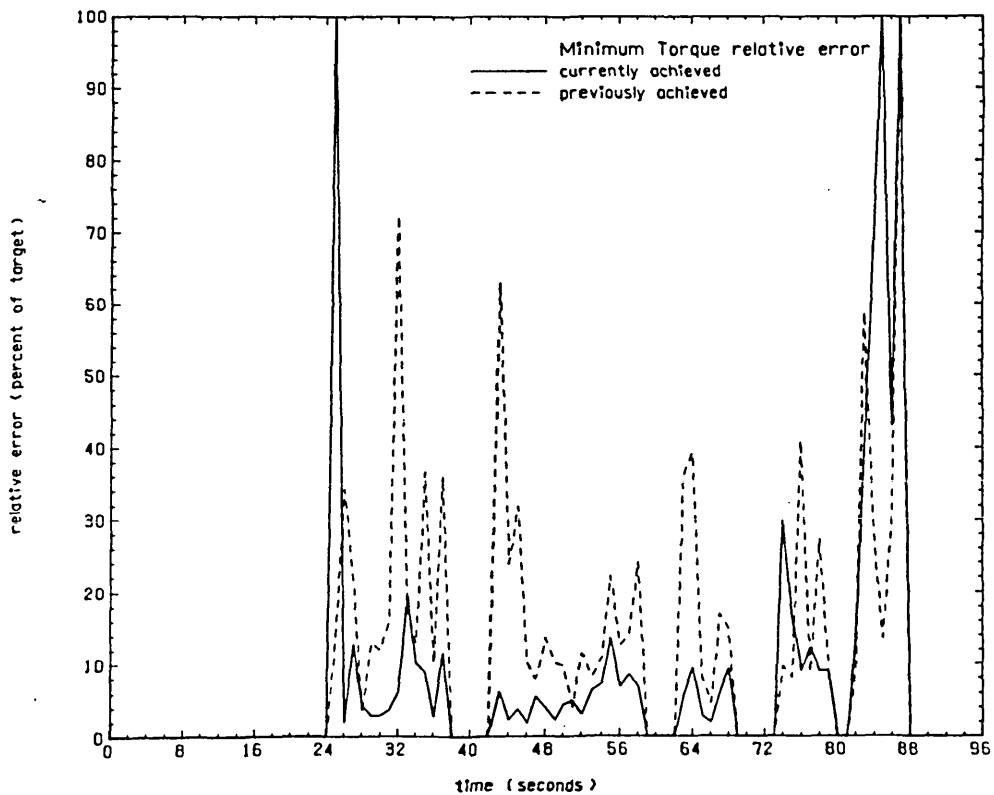


Fig. 7.42 PERKINS test-bed - Federal Cycle analysis.

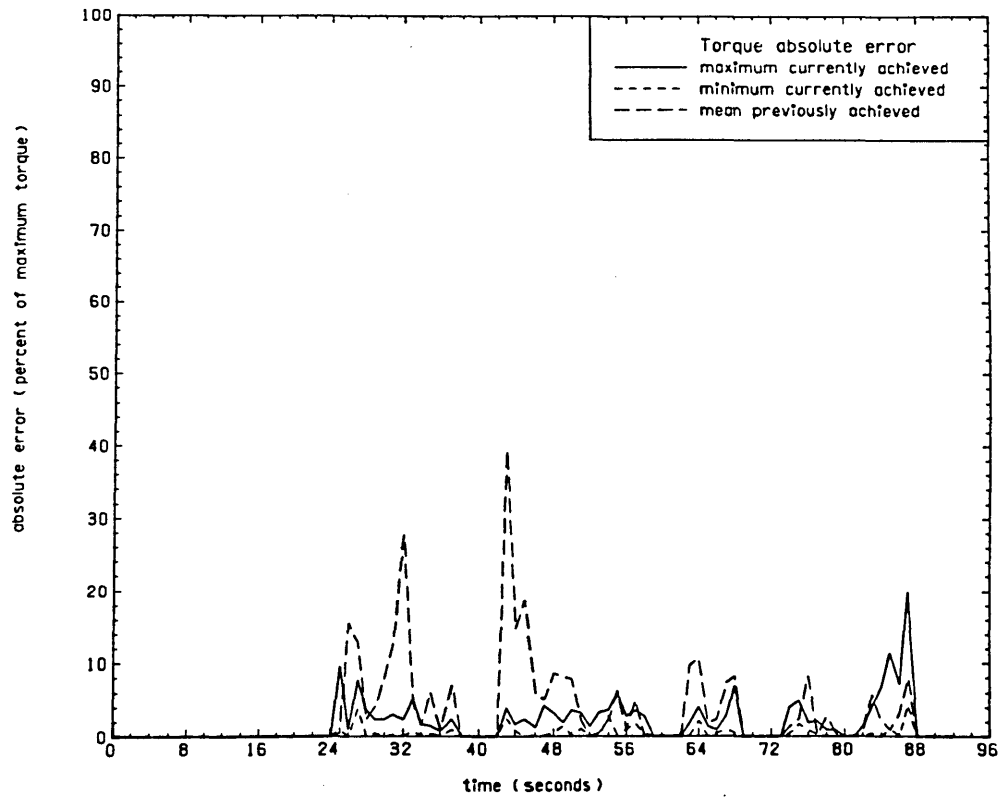


Fig. 7.43 PERKINS test-bed - Federal Cycle analysis

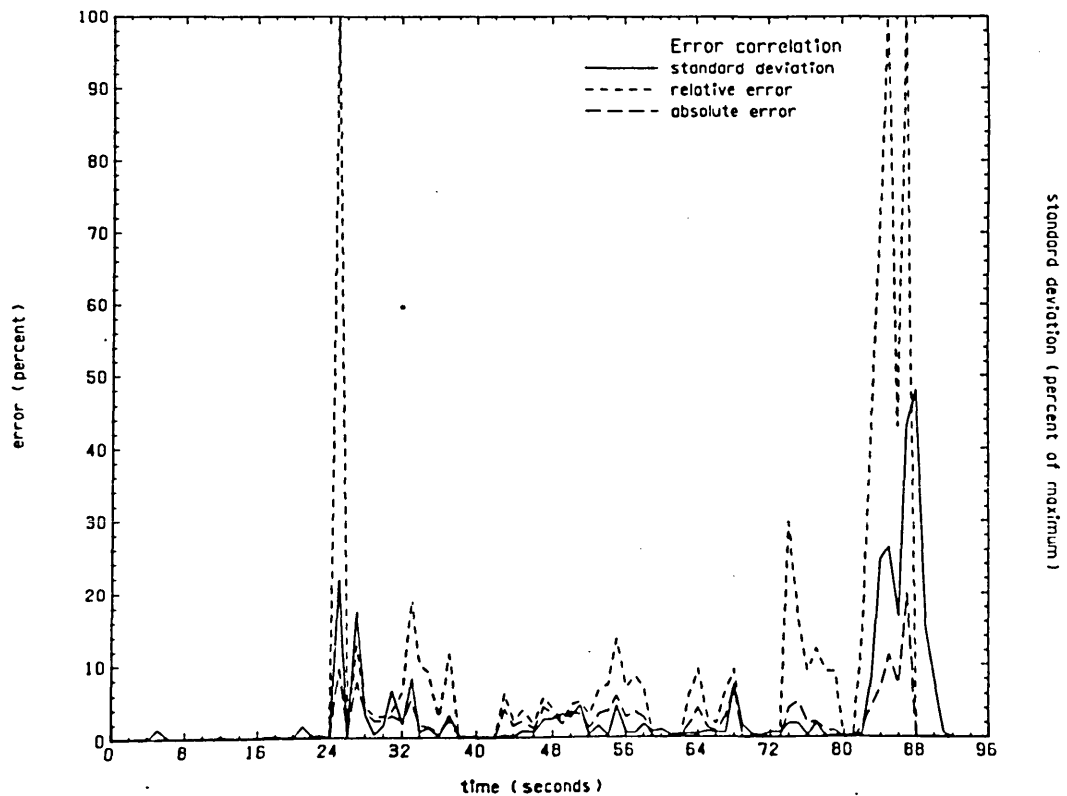


Fig. 7.44 PERKINS test-bed - Federal Cycle analysis.

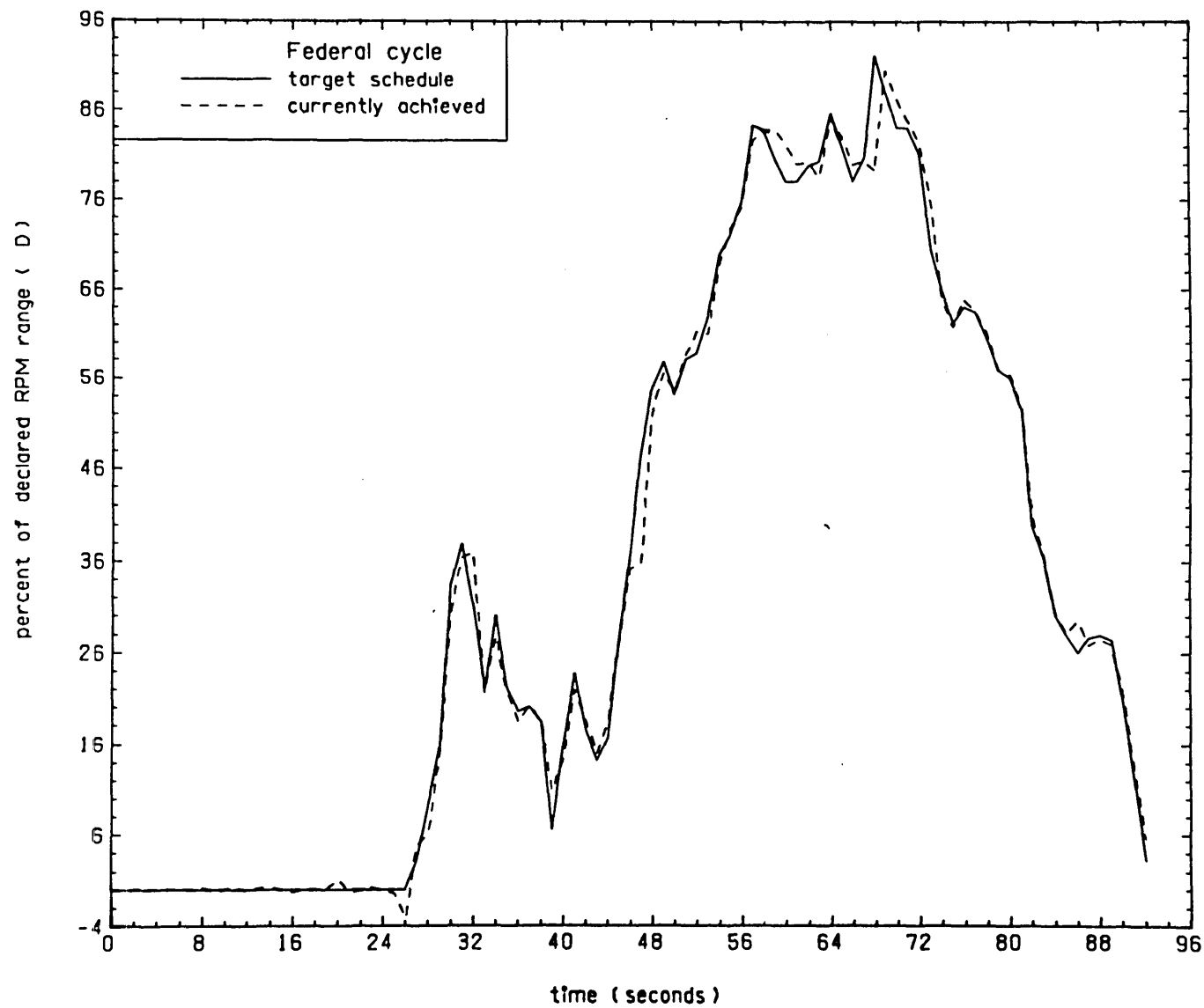


Fig. 7.45 PERKINS test-bed - Federal Cycle speed schedule.

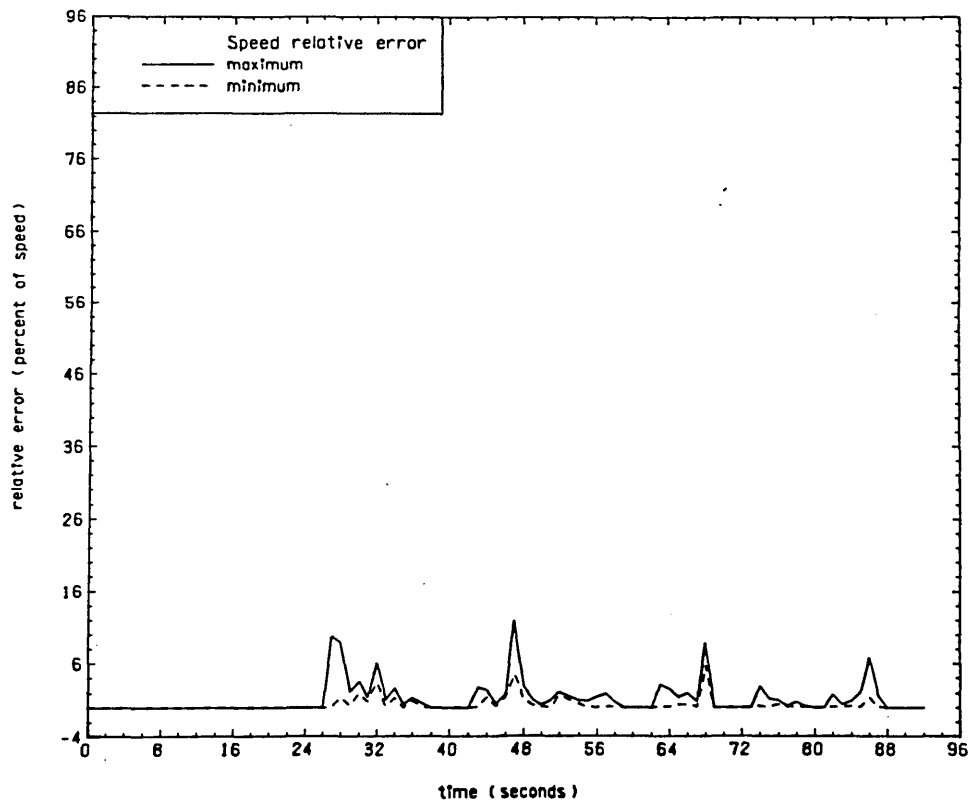


Fig. 7.46 PERKINS test-bed - Federal Cycle analysis.

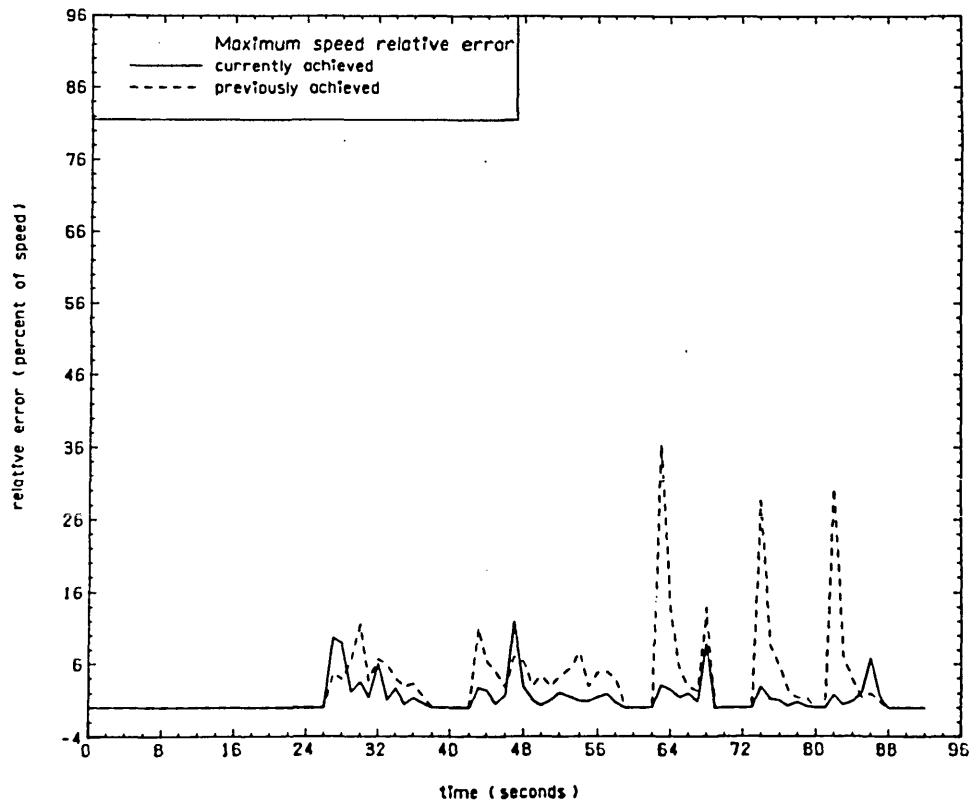


Fig. 7.47 PERKINS test-bed - Federal Cycle analysis.

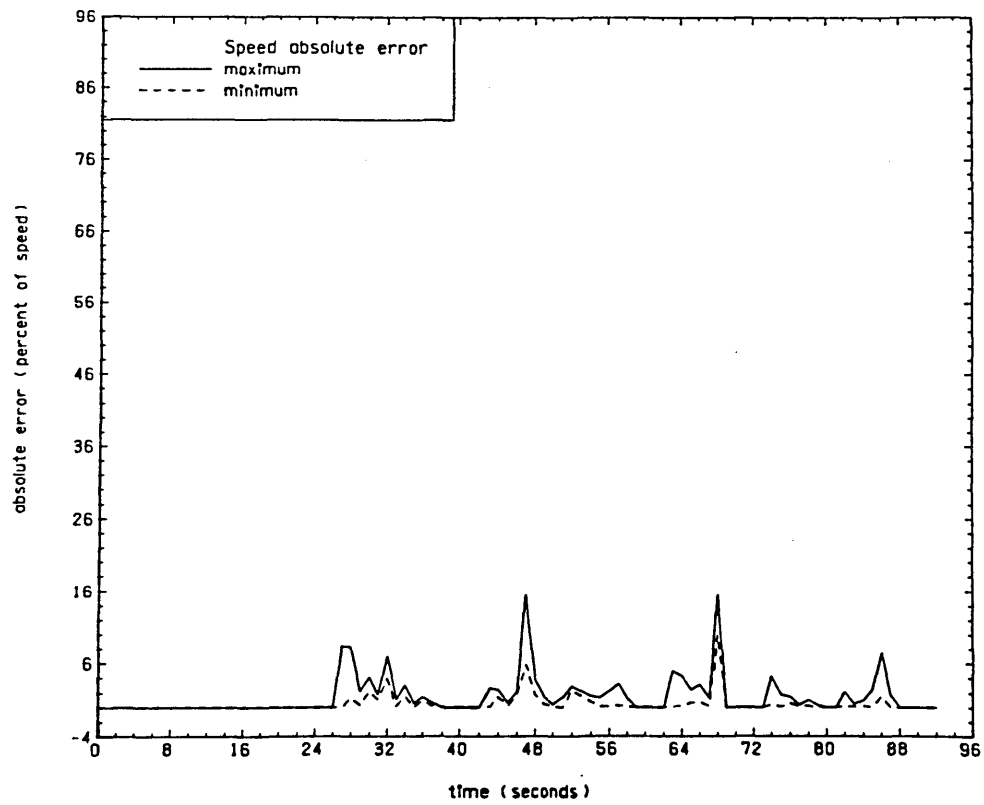


Fig. 7.48 PERKINS test-bed - Federal Cycle analysis.

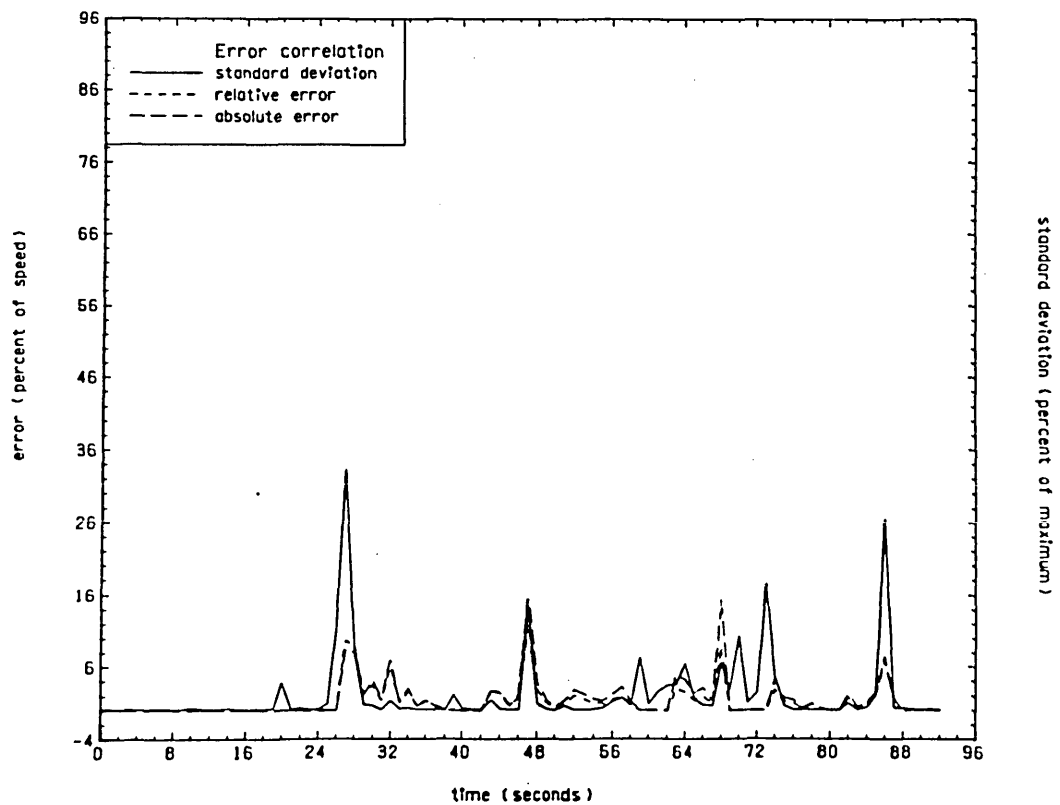


Fig. 7.49 PERKINS test-bed - Federal Cycle analysis.

speed droop as such scatter cannot be attributed to speed measurement precision.

The same tests with the same assumptions were conducted on the LEYLAND test bed. Fig. 7.50 displays the torque schedule measurements which, apart from the unloading parts where response is dictated by the system decay characteristics, follows the target schedule closely. The relative error (fig 7.51) presents some peaks around 25, 76 and 87 seconds. The absolute error which is more important for engineering considerations does not exceed 10% anywhere in the cycle (fig. 7.52).

The repeatability of the tests is illustrated by the standard deviation (fig. 7.56). The good anti-vibration design of the LEYLAND test-bed gave a maximum standard deviation of 4.29% throughout the test. The relative error peaks are not significant as the underlying absolute error is small (fig. 7.52 and 7.56). The error peak at 87 seconds reflects the system decay characteristics and not the controller behaviour.

The speed control problems of the LEYLAND test-bed were also experienced during the Federal Cycle tests. Fig. 7.53 shows the speed over the cycle. The main error peak occurs during the first loading-unloading acceleration-deceleration sequence. This engine is fitted with an exhaust gas turbocharger and a device to limit fuelling as a function of boost pressure. At no load the engine boost pressure is low therefore when the engine is loaded the boost pressure fuel controller does not allow full fuelling immediately. Combining that with an acceleration request causes the engine speed transient behaviour to be governed by the fuel controller. The control input (governor setting) is therefore saturated when attempting to increase speed. This causes the overshoot around $t=32$ seconds. Once boost pressure has built up the fuel controller action

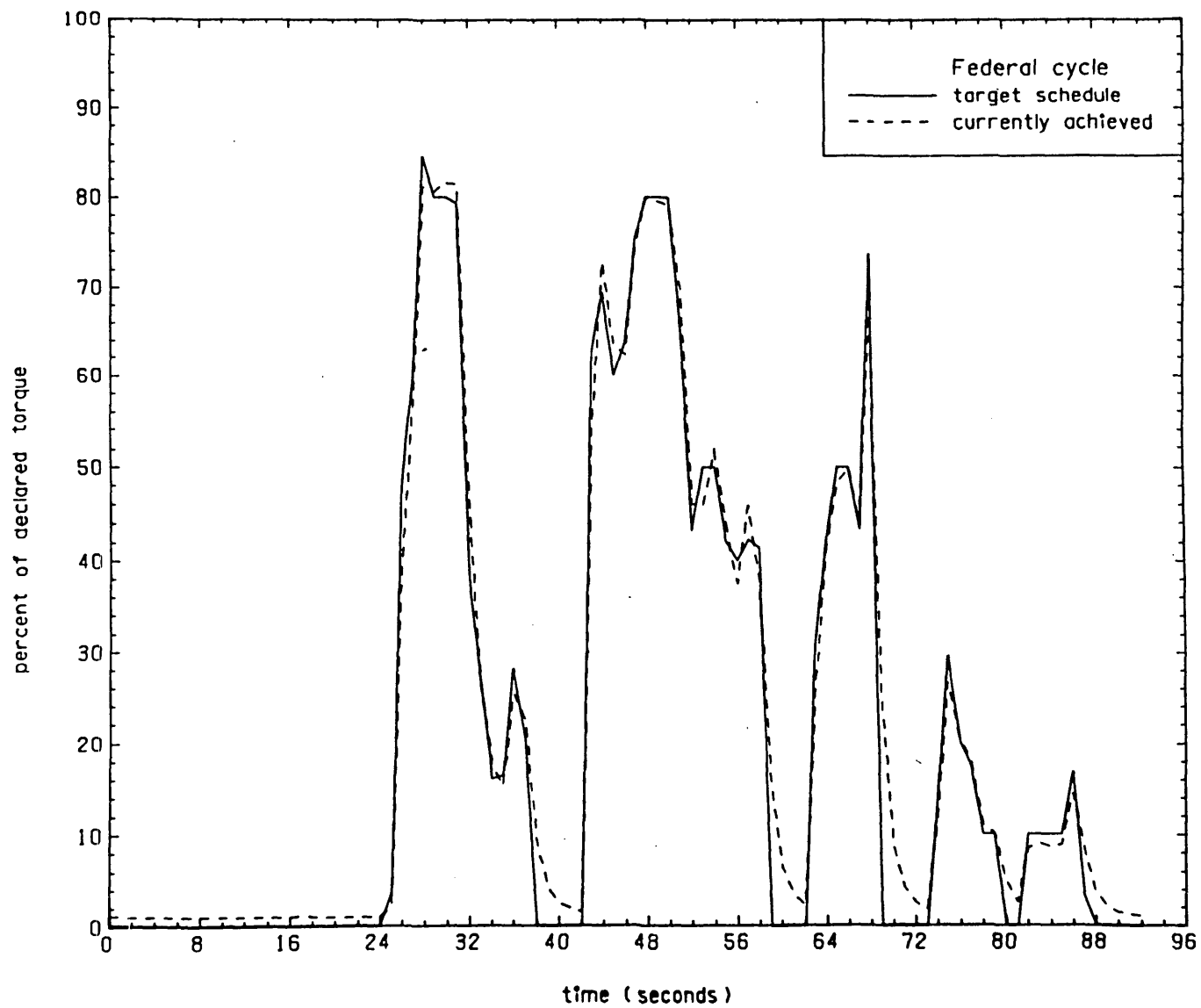


Fig. 7.50 LEYLAND test-bed - Federal Cycle torque schedule.

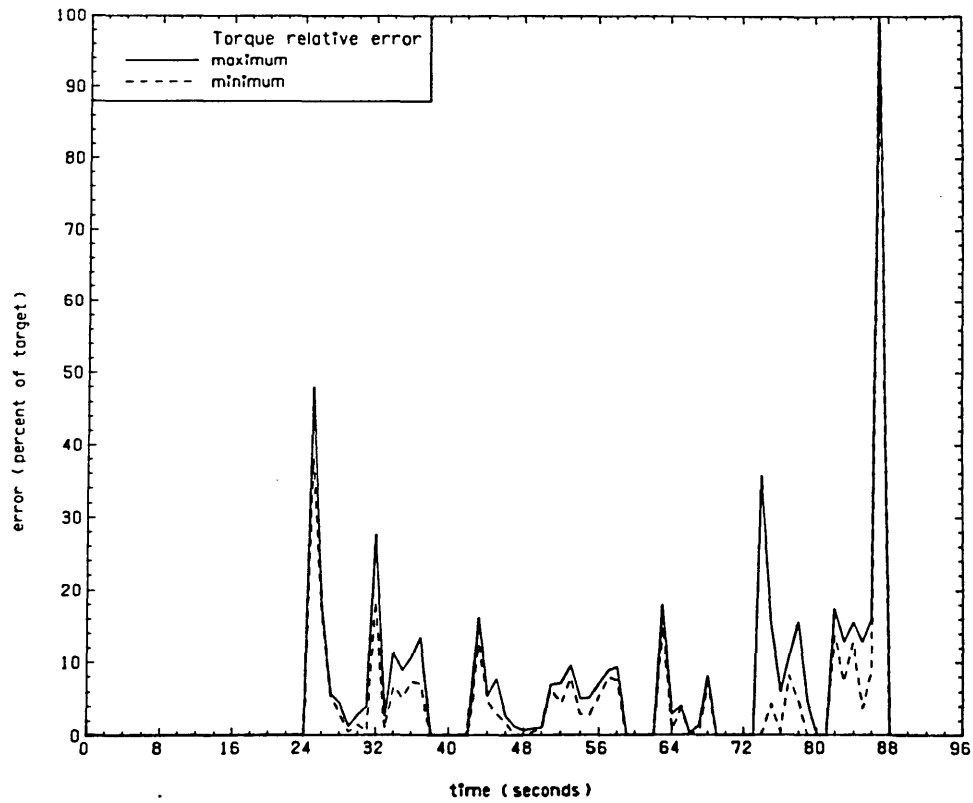


Fig. 7.51 LEYLAND test-bed - Federal Cycle analysis

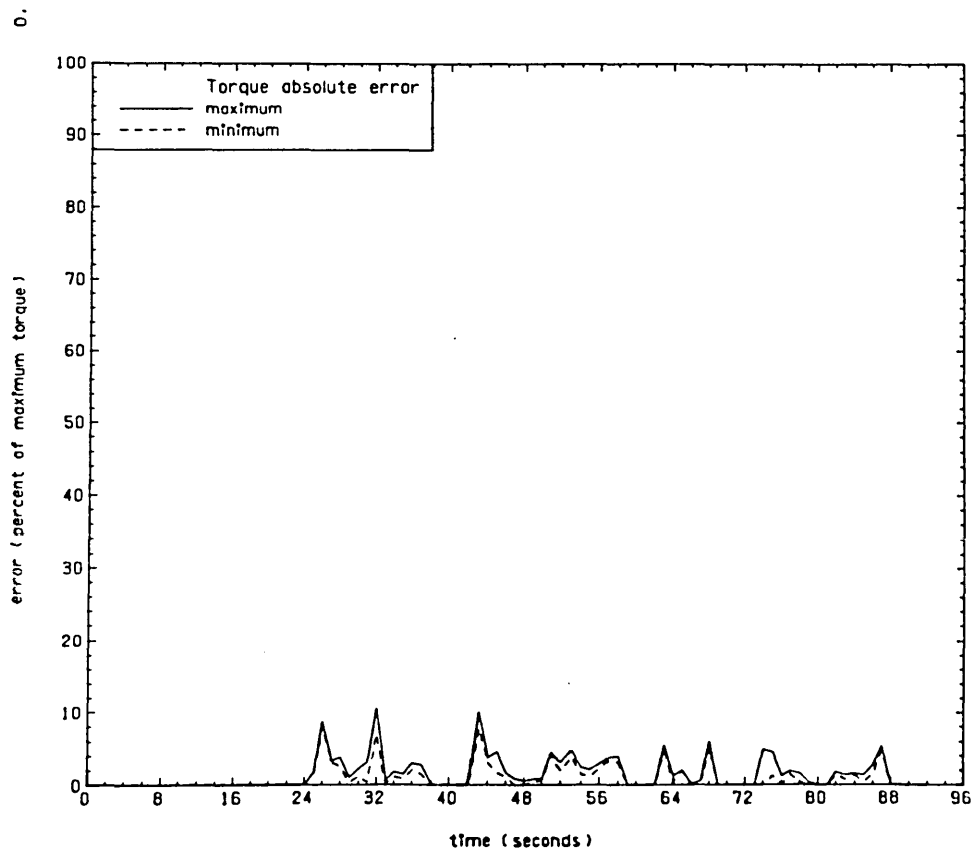


Fig. 7.52 LEYLAND test-bed - Federal Cycle analysis.

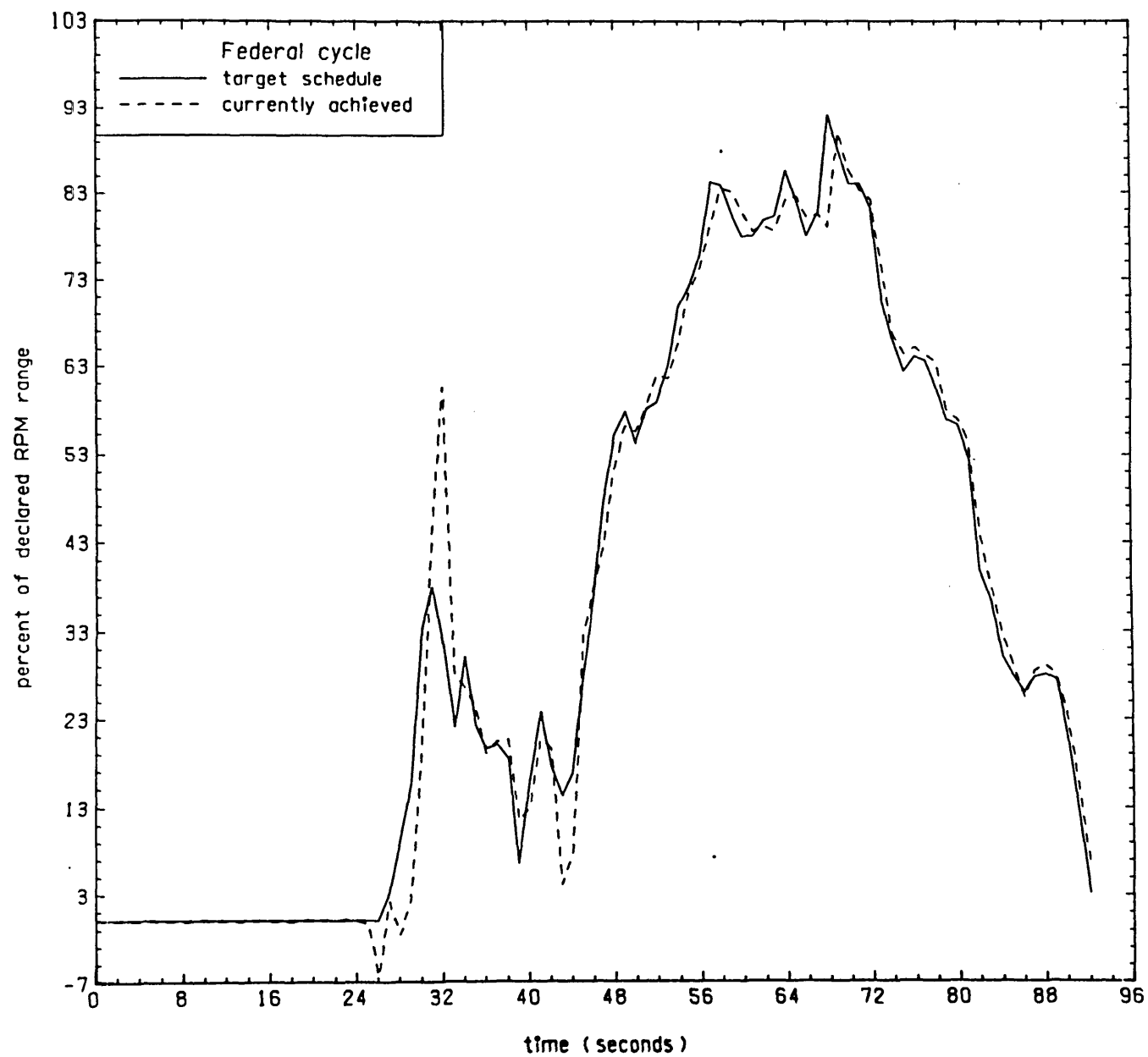


Fig. 7.53 LEYLAND test-bed - Federal Cycle speed schedule.

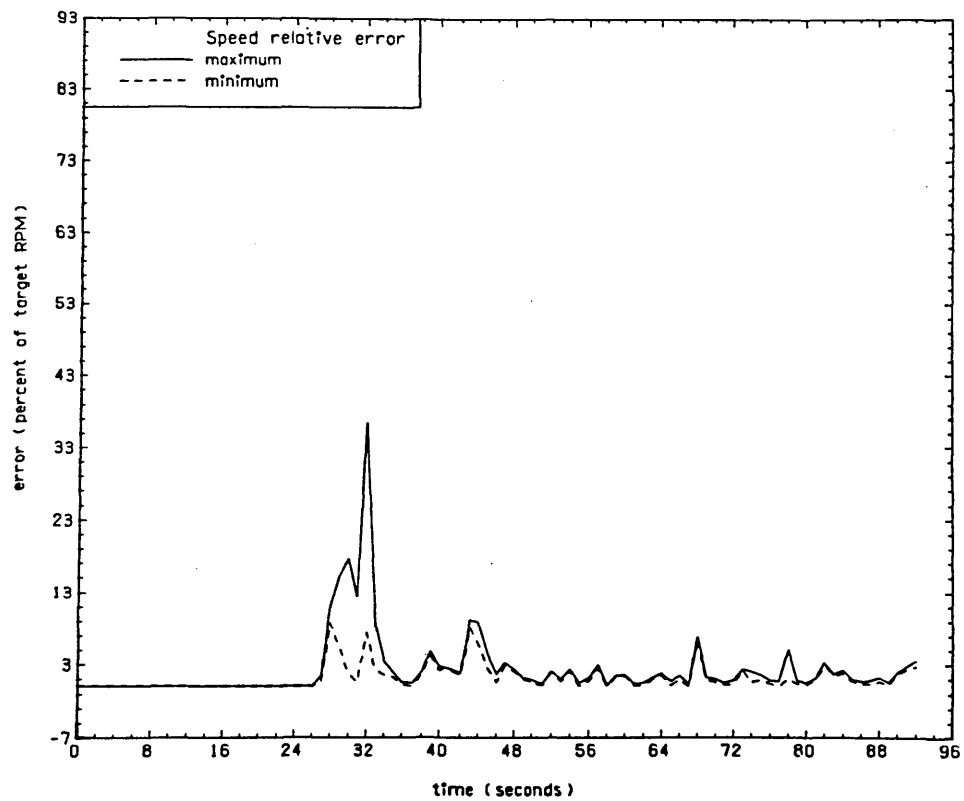


Fig. 7.54 LEYLAND test-bed - Federal Cycle analysis.

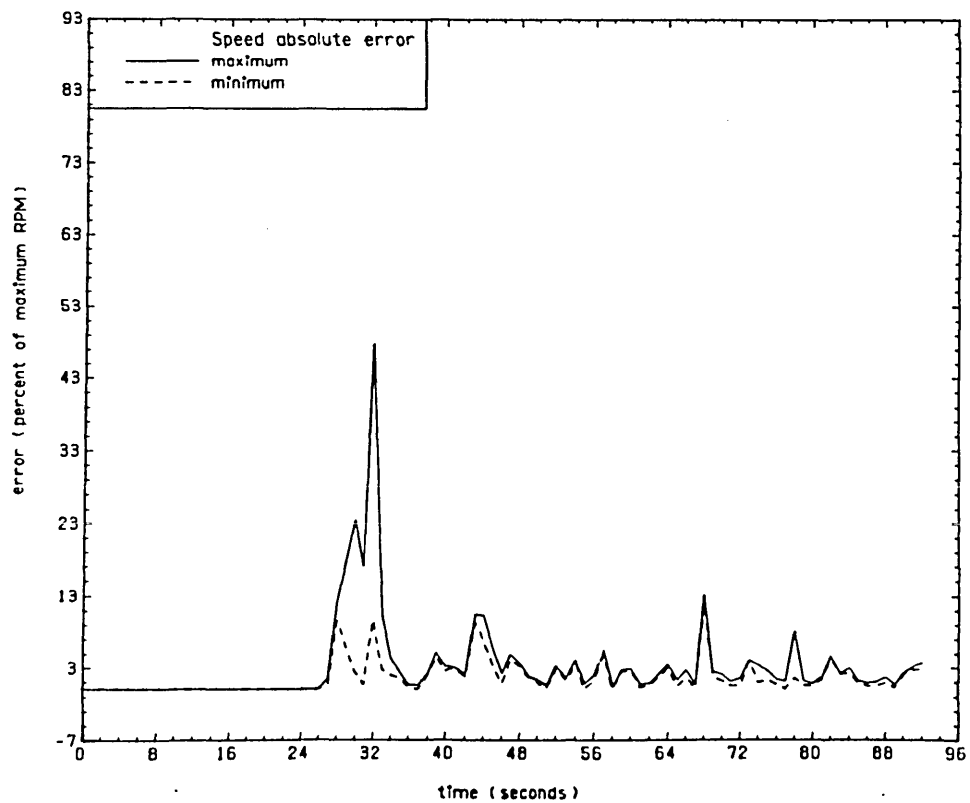


Fig. 7.55 LEYLAND test-bed - Federal Cycle analysis.

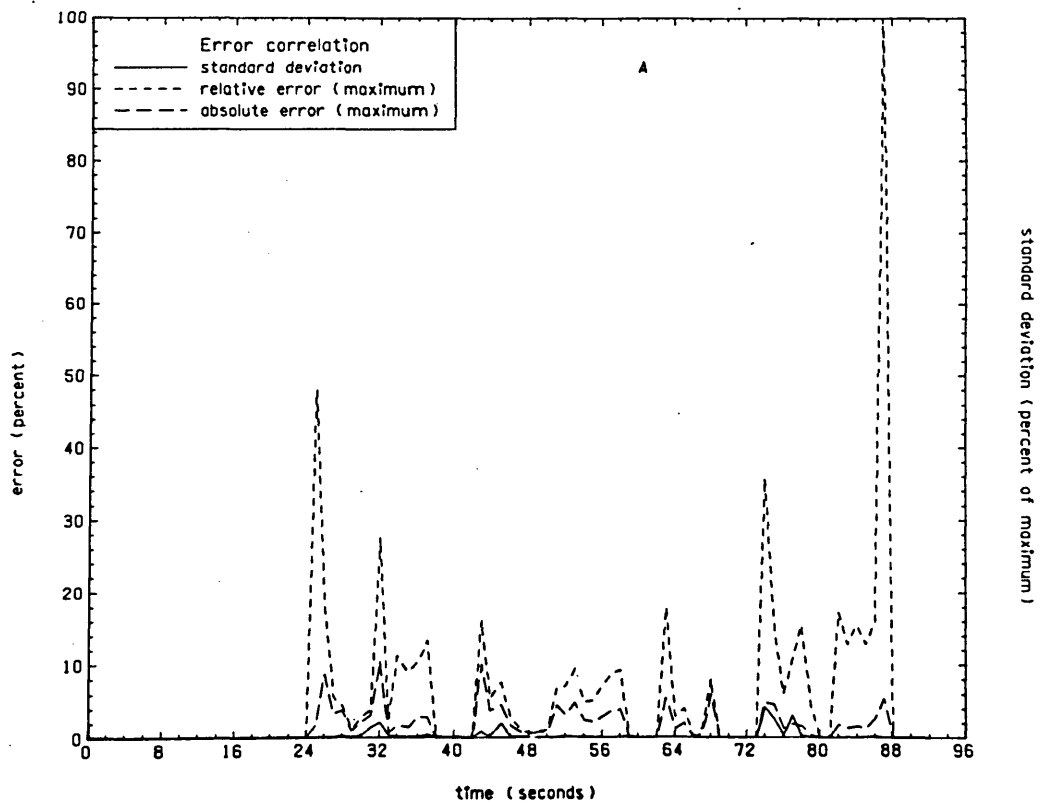


Fig. 7.56 LEYLAND test-bed - Federal Cycle analysis torque measurement and control.

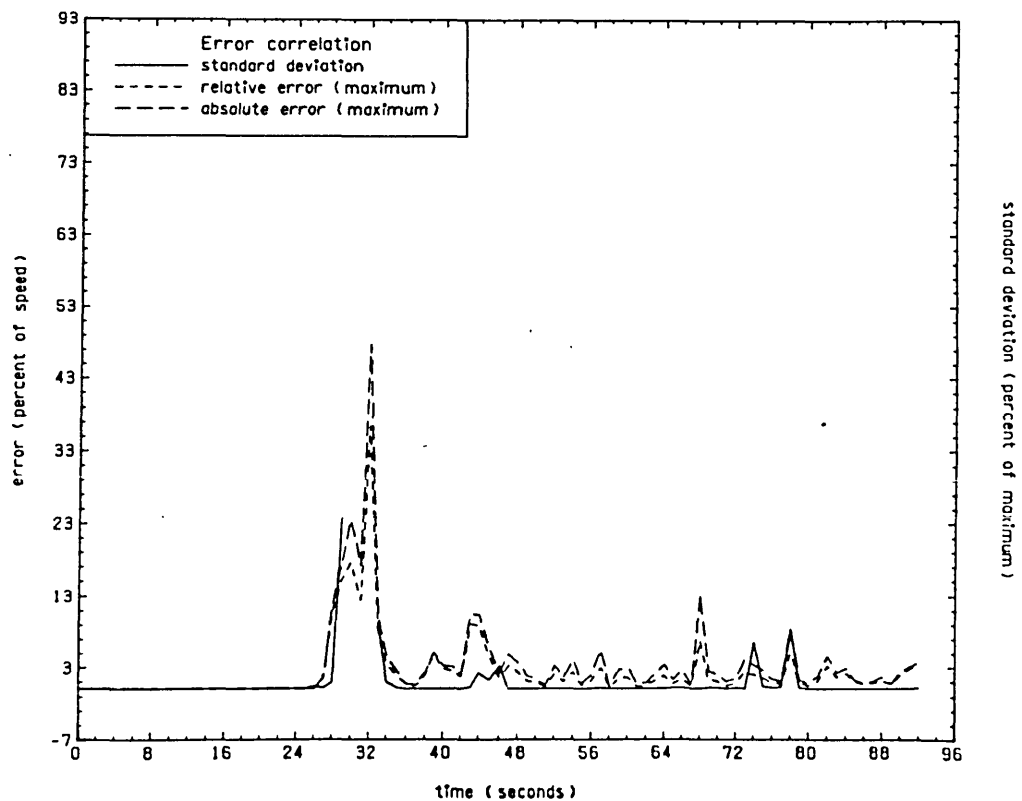


Fig. 7.57 LEYLAND test-bed - Federal Cycle analysis speed measurement and control.

is removed and the schedule speed is followed correctly. This reasoning is further supported by the diagrams of fig. 7.54, 7.55 and 7.57.

The Federal Cycle tests on both test-beds demonstrated the capabilities and limitations of the test bed controller. Also demonstrated were the importance of good test bed design (low vibration of torque measurement system) and the influence of the characteristics of the engine itself.

CHAPTER 8**Conclusions****8.1 Objectives**

The objectives of this work were:

- a) To review internal combustion engine testing and control methods and formulate the requirements for such testing especially with regard to the study of transient behaviour.
- b) To investigate the possibility of creating a local microcomputer network as a test-bed controller and attempt to link the different existing microcomputers at the Internal Combustion Engines Laboratory of Imperial College in a way that communication is made flexible and reliable.
- c) To use the resulting distributed computing power to improve the control methods on the test-bed both for the speed and the torque in terms of response, accuracy and flexibility and enhance data-acquisition in terms of throughput reliability and data integrity.
- d) To automate the engine testing procedure so that it becomes possible to perform repeatable tests, make the network structure and the control system both test-bed independent and transparent to the user and transfer into software all those functions requiring frequent modification to enhance testing flexibility.
- e) To perform a comprehensive series of tests to demonstrate the achievements in control improvement, testing flexibility and confirm the transportability of the system.

8.2 Achievements and limitations

The initial objective of achieving faster response from the eddy current dynamometers has been met by the new excitation unit design and optimal control techniques. Rise time improved to less than 0.5s from about 1.5s (fig. 7.6) and maximum overshoot to less than 5% from about 20% when compared to the previous excitation system with a software PID control.

The software optimal control of the stepping motor resulted in faster response, by means of accelerating the motor to a sustainable pulse rate far greater than the start-stop rate which was used by WIJEYAKUMAR [W-10] without loss of synchronisation.

Precision of control was improved both in the torque and the speed domain. The optimal torque controller improved speed disturbance rejection by incorporating the speed signal in the flux computation. Preview methods led to the design of a tracking controller that smoothed the torque schedule by looking ahead for 2s and optimising the control.

Speed control was improved by incorporating a torque feedback and torque disturbance rejection improved by a factor of two.

Repeatability of the system was confirmed by running eight consecutive tests of the first part of the Proposed USA Federal Smoke and emissions test (1983) for heavy duty truck engines and analysing the standard deviation of the measurements.

Transportability was confirmed by conducting tests on two distinct test beds. No software modifications were required and only parameter adjustment was necessary.

Flexibility was greatly enhanced by the following means:

- a) Software oriented controllers are readily adjusted as far as gain, damping, time constants and control limits are concerned.
- b) Reliability of software controllers is much greater as the

digital nature of the computer results in an either 'work or fail' condition without the component ageing phase, typical of analog controllers.

- c) The network structure made the system modular so that if need be, one node can be substituted without affecting the rest of the system.
- d) The interactive nature of the test setting up procedure made the system more user friendly and at the same time ensured that the operator did not omit any parameter specification. Different internal checks were performed by the system during test generation to validate data and parameters entered by the user.

The conducted tests and system analysis pointed out that the new test bed system still has some limitations:

- a) Torque application is still limited by the rate of increase of flux in solid iron and although faster than before, cannot be considered instantaneous.
- b) Torque removal on the eddy current dynamometer is limited by the flux decay time constant and cannot be compensated. In addition there is not motoring capability and cycle tests like the 1983 Federal Cycle which include motoring parts cannot be fully reproduced.
- c) The use of stepping motors as actuators for the throttle implies a trade-off between holding torque and response.
- d) The frequency of control loops is directly related to control stability. When very responsive engines or dynamometers are used the control loop frequency should be increased if control is not to be degraded and there is an upper limit to that increase, dictated by processor execution speed. Sampling frequency is

still limited by the hardware conversion time of the DAC and by DMA and disk transfer rates. Schedule specification is currently limited to 0.5s per setpoint although this can be easily modified.

8.3 Recommendations

8.3.1 Torque Control

Although a comprehensive set of tests has been conducted, it is desirable to test the time-optimal controller on a constant speed load i.e. a synchronous motor, to evaluate its performance more accurately. It is also necessary to test eddy-current dynamometers of other makers to investigate the influence of geometrical and constructional characteristics on performance.

8.3.2 Speed Control

Experience with the LEYLAND test-bed showed that speed control is impaired when a software and a hardware governor compete for the control of the engine. It should be possible to isolate the effect of each control loop when control performance is investigated. The LEYLAND engine was used with a far from optimized boost pressure - fuel limiting aneroid controller due to time constraints. This led to a serious response lag that distorted controller performance. It is recommended that the speed control tests on the LEYLAND be repeated after engine optimisation.

8.3.3 Data logging system

The data logging and control unit is not set currently by the computer. It requires manual intervention. A new unit is now

designed allowing for full computer control. Necessary provisions in the test-generation program have been made regarding the data structures to contain the change. The data logging program should be modified according to the new DLCU characteristics.

Appendix A Node Hardware Specification

Data Logging Node (Node 0)

Processor: LSI 11/23 (16 bit) with 128 K bytes of dynamic RAM
(Digital Equipment) and floating point hardware.

Peripherals: RK05 hard disk 5 Mbytes (Digital Equipment)
 ELITE 3052 visual display unit
 DECwriter LA36 132 column dot matrix serial printer
 Racal T7000 magnetic tape (7-track)

Operating Single user RT11-V04 without extended memory support.
system: Fortran V04 compiler with extended memory support
 (stand-alone).

Engine control and dynamometer control nodes (Nodes 1 and 2)

Processor: TMS 9900 (16 bit) with 8 and 16 K bytes of static RAM
 respectively.

Peripherals: RS232 interface to DLV11-J serial line interface of
 Node 0

Operating TIBUG monitor, and terminal simulation program
system: extending TIBUG with file structured requests.
 Cross-compiler running under RT11-V04 on Node 0.

APPENDIX B Test-bed specification.

Test bed:	PERKINS	LEYLAND
Engine:	Perkins 4.236	Modified Leyland 500 series
Bore:	98.43 mm	118 mm
Stroke:	127 mm	125 mm
Cubic capacity:	3.86 l	7.743 l
Compression ratio:	16.1:1	12.8:1
Firing order:	1,3,4,2	1,5,3,6,2,4
Power rating:	80 bhp at 2800 RPM	> 300 bhp
maximum torque:	26.68 mkp at 1400 RPM	150 mkp
Fuel pump:	CAV distributor type with max/min governor	SIGMA with P, D Compensation and rack limiter
Eddy-current dynamometer:	VIBROMETER 3WB15	VIBROMETER 4WB25
Dynamometer torque:	41.5 mkp at 2300 RPM	203.9 mkp at 1800 RPM
Excitation unit:	See Appendix C	
Stepping motor:	SIGMA 3437 D200-F075 dual angle type (1.8°/0.9°) reduction 56.25:1, bipolar excitation	
Optimal incremental shaft encoder:	Ferranti 28 H2/M4 3 tracks Track 1: revolution pulse Track 2 & 3: 720 pulses/rev. per track displaced by 90°	

10

Rise time: 20 ms for the 3WB15

VIBRO-METER dynamometer

Load impedance: 4Ω (max)

Rise time will be affected
if greater

Control voltage: 0 - -10V for

0 - 12A

Nomenclature

A	$n \times n$ constant matrix
a	constant denoting width, numerator zero
B	flux density
C	constant
d	constant
E	field intensity
e	error
f	function
G	function, set
g	Laplace transform of G
H	flux intensity, Hamiltonian
I	inertia
J	current density, cost function
K	gain
k	constant
l	length
L	Laplace operator
L^{-1}	inverse Laplace operator
m	dynamometer configuration parameter
P_1, P_2	costate variables
Q	quadratic form matrix
R^+, R^-	set of positive, negative real numbers
R	radius
S	surface, distance
s	complex variable in Laplace transforms
T	torque
t	time
u	control input
V, v	speed

x, y, z orthogonal system co-ordinates

Greek symbols and operators

β	Laplace transform of B, constant
Γ	Gamma function
γ^+, γ^-	switching curves
Δ	switching value
δ	air gap
ϵ	relative error, "belongs" symbol
ζ	damping (friction) coefficient
η	efficiency
λ_1, λ_2	poles in control system
μ	permeability
μ_0, μ_T, μ_{Te}	permeabilities of space, relative, corrected relative
ξ_1, ξ_2	initial states in state the space
π	3.14
Π	multiplication series symbol
Π_1, Π_2	constants
ρ	resistivity
Σ	summation series symbol
τ	time constant
Φ	flux
ϕ	Laplace transform of flux
ω	angular velocity
$\subset$	subset symbol
∇^2	operator $\equiv \left[\frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial y^2} + \frac{\partial^2}{\partial z^2} \right]$

Subscripts

s	step
r	ramp, relative.

δ	impulse
u	step
T	target
max	maximum
min	minimum
rss	relative steady state

Superscripts

T	transpose of matrix
$*, 0$	optimal
$-$	vector sign

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