THE TECHNICAL AND COMMERCIAL FEASIBILITY OF A HIGH SPEED DIESEL ENGINE OF HIGH SPECIFIC OUTPUT

A thesis to be submitted for consideration for the degree of

Doctor of Philosophy

S. J. Charlton, February 1981

Interdisciplinary Higher Degrees Scheme,
University of Aston in Birmingham.

The Technical and Commercial feasibility of a high-speed diesel engine of High Specific output

Stephen John Charlton

A submission for the degree of Doctor of Philosophy 1981

SUMMARY

The objective of this work is to evaluate the new product potential of high specific output, high-speed diesel engines. The research, which was undertaken on the Stafford site of Dorman Diesels Limited, considers technical and commercial aspects of the problem. As such, the project is an example of applied, rather than pure research, seeking to elucidate a multi-dimensional problem of direct interest to the sponsor.

Several methods have been successfully used to increase specific output. The design principle employed for this study was to increase the maximum brake mean effective pressure (bmep) by using a high level of turbocharge. Mechanical loads were contained by lowering the compression ratio. Typically, the most highly rated, commercially available engines operate at bmep's of 14 to 16 bar. This project assesses the feasibility of a current Dorman engine operating at a bmep of 21 bar at fixed speed.

A review of literature was followed by a design study which made use of computer-based engine performance and heat conduction models new to the company. This led to the specification of a research engine which was constructed and operated successfully in the sponsor's development facility.

The technical study was used to form a rudimentary product profile as the basis of a limited assessment of market potential. Consideration was given to the compatibility of the concept with current engine applications, and the suitability of corporate skills and other resources. The broad product characteristics were compared with the needs of identified market segments.

The concept was found to have only limited potential compared with current engine designs, which find almost universal usage.

The project has increased the company's experience and data base, thus allowing more rational decision-making on the future of high output engines in the Dorman product range.

KEYWORDS :

DIESEL TURBOCHARGING HIGH-OUTPUT COMPUTER MARKETING

ACKNOWLEDGEMENTS

This research could not have been completed and presented in this form were it not for the support, patience and goodwill of many, both at Dorman Diesels Limited and the University of Aston in Birmingham.

I am particularly grateful to Mr. Lewis Allsopp, who, as Chief Engineer, initiated the project and invested me with the responsibility for its execution. I would like to acknowledge the guidance and encouragement given by Derek Hickson and the late Tony Proudman.

At Dorman, R.D. Brown, D.L. Jones, S. Oxenham, E.H. Cheers, W.J. Pugh and R.H. Cartwright gave continuous support and many hours of hepful discussion. Thanks are also due to my long-suffering family and to the several typists, in particular, Gwen Saunders.

CONTENTS

LIST OF FIGURES

List of Figures cont'd...

List of Figures Cont'd...

List of Figures contd...

LIST OF TABLES

NOTATION

A	Area
A^*	Effective area for ideal flow
A/F	Air-fuel ratio (mass basis)
C, C_1, C_2	Constants
C_m	Mean piston speed
CR	Compression ratio
C_p, C_v	Specific heat capacities
D	Cylinder diameter ; diameter
F	Equivalence ratio
h	Heat transfer coefficient ; specific enthalpy
k	Thermal conductivity
L	Piston stroke ; length
m	Mass
$\dot{m}$	Mass flow rate
N	Speed of rotation
Nu	Nusselt number
n	Polytropic exponent ; number of cylinders
P	Power
p	Pressure
Q	Heat energy
$\dot{q}$	Heat flux
R	Specific gas constant
Re	Reynolds number
T	Temperature
t	Time

NOTATION CONT'D

U	Internal energy
V	Volume
$\dot{V}$	Volume from rate
v	Specific volume
x	Wall thickness
γ	Ratio of specific heat capacities
ε	Aftercooler effectiveness
η	Efficiency
ρ	Density
φ	Crank angle
ψ	Explosion ratio
ω	Angular velocity
λ_{sc}	Scavenge ratio

NOTATION CONT'D

SUBSCRIPTS

a	Air
b	Measured at brake
bth	Brake thermal
c	Compressor
comp	Compression
cyl	Value in cylinder
e	Exhaust
exp	Expansion
f	Fuel
i	Indicated value
ith	Indicated thermal
m	Mean value
man	Value in manifold
max	Maximum value
mech	Mechanical
min	Minimum value
tot	Total
tr	Trapped by cylinder
vol	Volumetric basis
w	Combustion chamber wall
o	Reference ; ambient value

NOTATION CONT'D

ABBREVIATIONS

tdc	Top dead centre
bdc	Bottom dead centre
atdc	After top dead centre
btdc	Before top dead centre
CA	Crank angle
avo	Inlet valve opens
avc	Inlet valve closes
evo	Exhaust valve opens
evc	Exhaust valve closes
imep	Indicated mean effective pressure
bmep	Brake mean effective pressure
fmep	Friction mean effective pressure
isfc	Indicated specific fuel consumption
bsfc	Brake specific fuel consumption

Energy is eternal delight.

William Blake

CHAPTER 1

INTRODUCTION

1.1. Introduction

This opening chapter discusses the background against which the research was initiated. An historical review of diesel engine progress is given, followed by an account of present day activity. The chapter closes with a description of the sponsoring organisation and the Interdisciplinary Higher Degrees Scheme.

1.2. Historical Perspective

The first engine to operate successfully on the compression-ignition principle did so for the first time on 17 February 1894. It was designed by Dr. Rudolf Diesel and was built and developed under his supervision at the workshops of Maschinenfabrik of Augsburg (1). The first diesel engine achieved fuel injection by forcing fuel and compressed air through a nozzle arrangement. It was not until 1908 that Prosper L'Orange perfected an injection system without air compression whilst working for Benz of Mannheim. Despite the early problems, commercial interest in the new prime mover was immediate, companies such as Sulzer Brothers, Deutz, Krupp and Mirrlees being the first to hold manufacturing licences. Initially, the diesel engine was seen as a cheaper, more efficient and compact replacement for the steam engine. It was soon in demand for electricity generation, marine propulsion and industrial power, and by 1904 was being developed as a potential aeroplane engine !

In 1915, a Swiss engineer, Dr. Alfred Büchi, patented a system which incorporated a diesel engine charged by an axial compressor which was driven by an axial turbine in the exhaust stream of the engine. This stated the principle of exhaust turbocharging as it is widely used today (2). Initial experiments produced a 50 per cent increase in specific power output readily, whilst for short periods, an increase of 100 per cent was attainable. The benefits of exhaust turbocharging, namely reduced capital outlay, size and weight per unit power output encouraged its acceptance by many of the user industries. The first commercial application of the exhaust turbocharged diesel engine was in two passenger ships of the East Prussian service. Büchi further contributed to the development of turbocharging in 1925 when he patented the pressure wave process, today known as pulse turbocharging (3).

Low and medium speed engines require large quantities of charge air and consequently, relatively large turbochargers. The size factor is very relevant to turbocharger design : the smaller the rotor diameter of axial designs, the more significant tip leakage and fluid friction losses become, resulting in poor efficiency (4). For this reason, the most suitable design for smaller, high speed engines is the radial turbine and compressor. It was not until about 1955 that cheap, reliable and efficient cast steel radial flow turbine rotors became available. In the period since then the turbocharger has become widely accepted by users of the high speed diesel engine.

1.3. The Diesel Industry Today

The power range covered by the diesel engine, approximately 2 kW to 36 MW is wider than that of any other prime mover. The design variations are numerous : two or four stroke cycle, direct or indirect injection, low, medium and high speed, light distillate through to heavy fuel oils ; the only unifying element is that combustion occurs spontaneously by injecting fuel into hot, compressed charge air. The most widely used classification is based on the working speed of the engine. Low speed engines generally operate at the propellor speed of large vessels (100-250 rev/min). Above this well-defined group are medium and high speed engines, the division of which is widely accepted as 1000 rev/min.

The prosperity of many parts of the diesel engine industry is largely determined by macro-economic factors such as national growth rate, trade cycles, levels of public expenditure, industrial investment and world oil prices ; therefore it is dynamic and ever-changing. Currently, the world recession in ship-building is having a severe effect upon the makers of low speed marine propulsion engines, forcing them to find other markets, notably stationary power generation plant. In the high speed diesel class, new automobile engine designs are beginning to displace the petrol engine, a causal factor being the good fuel economy of the diesel engine at a time of sharply rising fuel costs.

The world market for diesel-powered generating equipment is worth between £150m and £200m per annum. A high proportion of British generator set production is exported, especially to developing African and Middle Eastern states which do not have a national electrical grid system. This market is particularly sensitive to factors beyond the control of the engine manufacturer. Fluctuation in demand has arisen in the past for a variety of reasons : import tariffs, currency exchange rates and political upheaval being notable examples. The home market is also variable and appears to depend largely on the level of building construction and public spending. There are approximately 250 diesel engine manufacturers throughout the world making some 5000 different designs. World production in 1977 was almost 6 million engines valued at £7500m (5). A breakdown of production for 1977 by application and brake horse power, excluding the Soviet bloc and China is given in Figure 1.1.

1.4. Background to the project

The specific power output of high-speed diesel engines has risen steadily since the introduction of turbocharging. Engines developed for fighting vehicles (6)(7) have shown that very high specific power is achievable by this method. The demands for high power, lightness and compactness are very severe, to maximise the vehicles' mobility in action. Engines developed for the main battle tank typically produce 2.5 to 3.5 times the power available from a naturally aspirated engine of the same swept volume.

The most highly rated commercial engines currently available are moderately turbocharged, giving a little over twice the power of a naturally aspirated equivalent. The benefits of turbocharging are not only a reduction of specific size and weight, but also a reduction of specific cost. The reason for this is straightforward : the cost of the turbocharger and other parts required to uprate a naturally aspirated engine is proportionately less than the resulting increase in power. This cost structure is demonstrated in Figure 1.2.

This project was initiated in 1977 against a background of continually rising commercial engine ratings. The original terms of reference were to design and develop a high output variant of the Dorman "LE" engine to operate up to a brake mean effective pressure (bmep) of $300 lbf/in^2$ (20.7 bar).

The engine had to employ a fixed compression ratio and use single-stage turbocharging, if feasible, in preference to multi-stage schemes. Recognising the inadequacy of this brief, further discussions were held with the sponsor. It was agreed at that stage that the company were interested in the commercial rather than the military potential of highly turbocharged variants of existing ranges. Furthermore, they were not primarily interested in an engine developing a bmep of 21 bar, but more in the knowledge and experience that this would generate. Finally, if they did not market engines of such a high rating, it was hoped that there would be benefits or "spin-off" from the programme for the company's existing range of engines.

1.5. Project Philosophy

It was decided to approach the research from both technical and commercial standpoints. Although a commercial study was not specified in the terms of reference, for many reasons its inclusion was important. The objective was to assess the level of business opportunity that the high output concept might offer the company. This would involve evaluation of production costs and market need, all of which help to create market awareness, which should influence the solution of the technical problems. This emphasises the nature of industrial projects which are rarely amenable to a strictly one-dimensional approach. The final business objective, to generate profits through the satisfaction of human needs, involves the dismantling of traditional disciplinary boundaries. The commercial study was introduced on the premise that, whilst consuming valuable time, it would be more than compensated for because of the greater market orientation of the technical study. The technical work was initiated at the start of the project because of the long lead times anticipated for certain components from outside suppliers. Commercial work was spread over the duration of the project with the occasional extended study. A flow diagram showing the structure of the thesis is given in Figure 1.3.

1.6. The Company

This research project was sponsored by and undertaken on the premises of Dorman Diesels Limited, of Stafford. Dorman is a division of GEC Diesels Limited which is one of several large groupings within GEC Limited.

The company have a long history of designing and manufacturing internal combustion engines. Their earliest designs were spark-ignition engines, the change in emphasis coming in the 1920's when the introduction of tax on petrol created the economic incentive for the development of small, high-speed diesel engines. Dorman have never produced engines in large quantities, their "reputation" being for a quality product adapted to suit the customers' requirements.

Until the last decade, Dorman had sold to a wide range of markets, with marine propulsion, construction, rail traction, earth-moving and industrial applications all taking significant numbers of engines. Throughout the 1970's, Dorman have been very heavily committed to supplying the generating set market which was particularly buoyant around the middle of the decade.

Dorman market engines in the range 35-750 kW, an area which has become highly competitive during recent years.

1.7. The Interdisciplinary Higher Degrees Scheme

The Scheme was established in 1969 to further the ideals of the Swann Report. This said that many graduates were insufficiently attracted to industry, for various reasons. The report called for bold experimentation with the Ph.D. in relation to industry. An objective of the IHD Scheme at the University of Aston in Birmingham is to provide postgraduate training, partly in industry, of a broad nature, as distinct from more specialised education in, for example, Engineering Science, or Business Studies.

The training is organised around a project in industry. The student has a steering committee of University and company supervisors to whom he reports at regular meetings. The University is attended for sessions of coursework, consultations with tutors, and library research. The coursework consists of a series of lectures arranged to occupy three or four days every month. These are generally given by visitors from industry, providing first-hand experience of a range of relevant subjects.

It is the aim of the Scheme that graduates will be well-adapted to industrial management, having an awareness of some of the many areas that impinge upon it, and having had some experience of project management in industry.

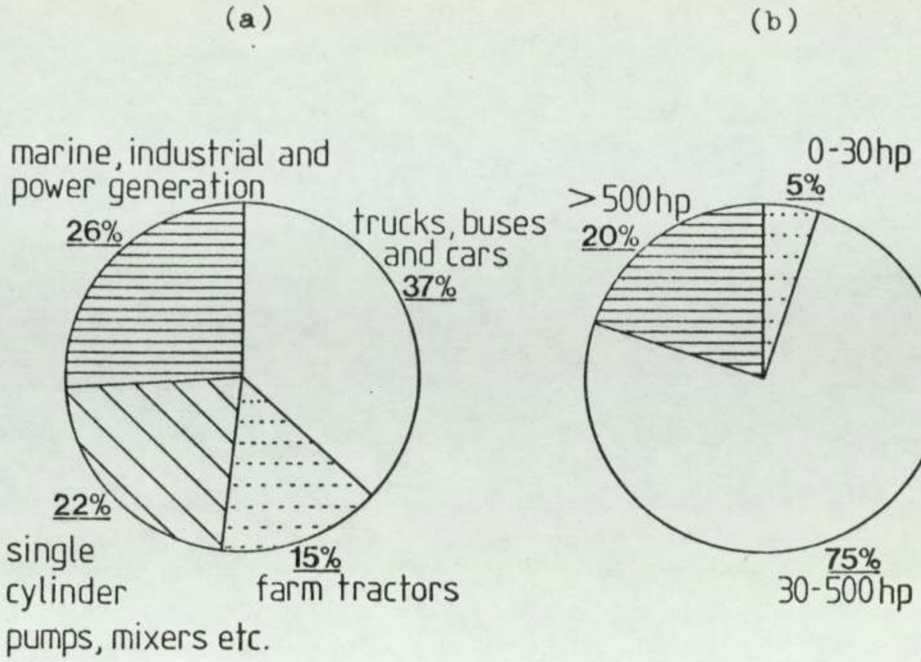


Fig.1·1, Analysis of the world market for diesel engines based on the number of units sold in 1977 (6m). a, by application
b, by horse power, ref(5).

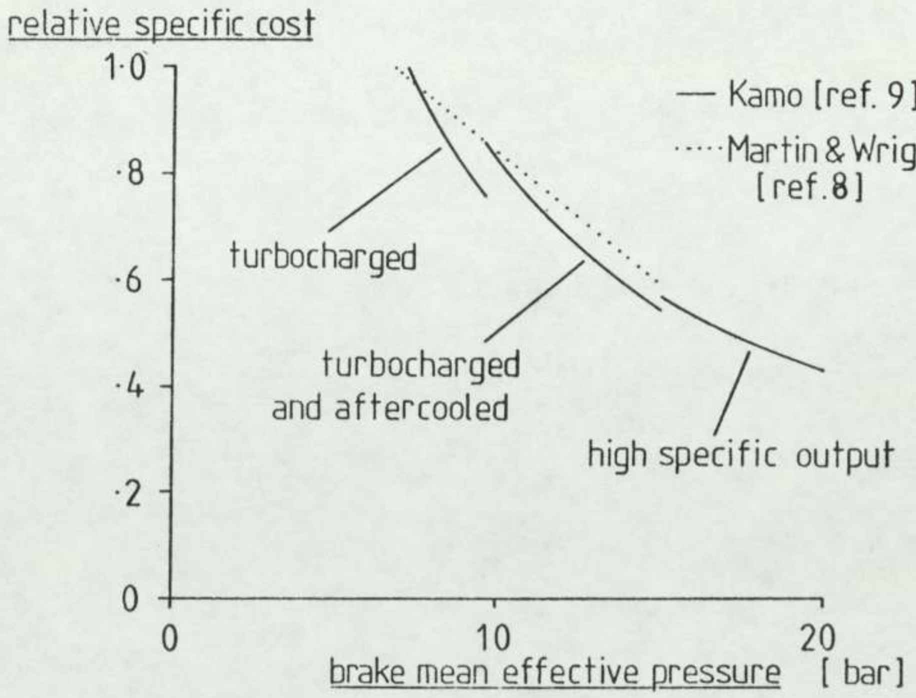


fig.1·2, Relative specific production costs [£/kW], based on nat. aspirated equivalent.

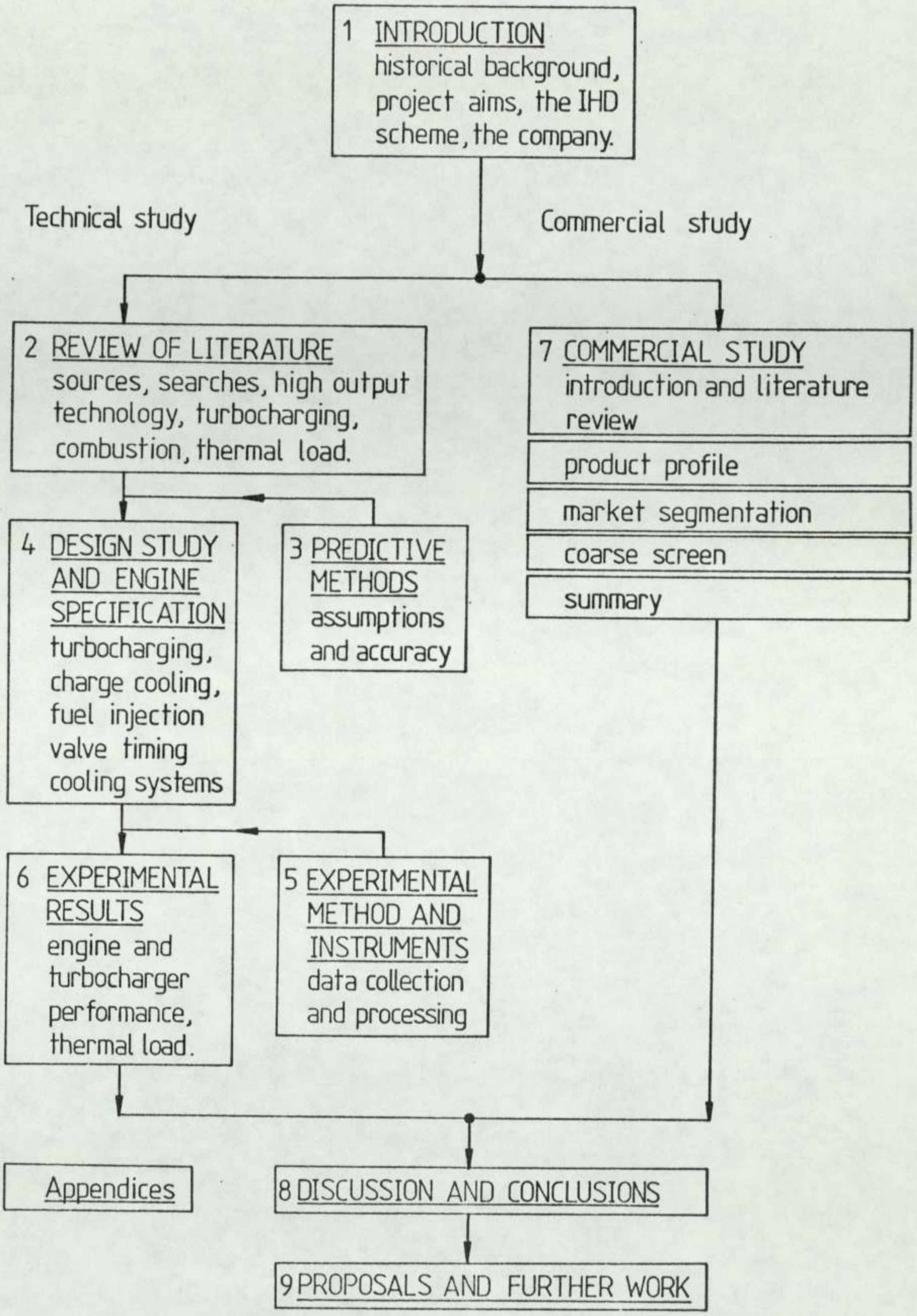


Fig. 1.3, A flow diagram showing the structure of the thesis.

CHAPTER 2

REVIEW OF LITERATURE

"There have been so many publications recently in technical periodicals of all languages, on the construction of the Diesel engine and its various types, that is is hardly possible to give any fresh information on the subject."

Dr. Rudolf Diesel, March 1912.

Chapter 2

REVIEW OF LITERATURE

2.1. Sources and Searches

The diesel engine has been an important prime mover for over seventy years, and during that time has been the subject of a great deal of research and development. Industry, universities and research organisations have all contributed to our understanding of its complex processes. Studies have ranged from the development and analysis of single components to the design and development of whole engines. Thus, the available literature on diesel engine technology is comprehensive.

During the course of the project, literature searches were undertaken on technical and commercial subjects. A general search of high specific output technology and related titles was performed by the libraries of the Institution of Mechanical Engineers, and Ricardo and Company (1927) Limited. Further searches were made of the published papers of the Society of Automotive Engineers, the Diesel Engineers and Users Association, the International Congress on Combustion Engines (CIMAC) and others. The leading journals relating to diesel engines have been retained over many years by the libraries of GEC Diesels Limited. Where relevant, these were consulted. A formal patent search of the years 1968 to 1978, for the U.K. was performed by the GEC Patents Office. This examined the patents filed by :

1. Teledyne Continental Motors ;
2. Rolls Royce Limited ;

3. The Garrett Corporation ;

4. Volvo Limited ;

and 5. Motoren-und Turbinen Union (MTU).

for those concerned with high output technology. This, in fact, was unproductive. Other patents have been obtained by searching the abstracting journals.

There are a large number of potential sources of literature. Those used during the project are listed below.

Sources of Literature

The Institution of Mechanical Engineers :

Library

Conference Papers

Proceedings

Society of Automotive Engineers (USA)

Diesel Engineers and Users Association

American Society of Mechanical Engineers

International Congress on Combustion Engines (CIMAC)

Motor Industry Research Association (MIRA):

Abstracting service

Ricardo and Company (1927) Limited :

Library

Patents Abstracts

British and ISO Standards

Index to British Theses

Diesel and Gas Turbine Worldwide Catalogue

Journals

Diesel and Gas Turbine Progress Worldwide

International Power Generation

Automotive Engineer

The Transport Engineer

Diesel Engineering

Gas and Oil Power

Motor Transport

Marine Engineering/Log

Railway Gazette

Brown Boveri Review

2.2. High Specific Output Engine Design and Development

Specific power output may be defined in several ways : power per unit weight (power/weight ratio), power per unit bulk volume, and power per unit cost are three examples. Brake mean effective pressure (bmep) indicates the degree of utilisation of cylinder capacity and is often used as a surrogate for specific power, since for an engine of given weight, size and cost, at a given speed, specific power will be directly proportional to bmep.

Internal combustion engines transform the chemical energy of the fuel into mechanical work. This transformation is possible only if a suitable quantity of air is made available for combustion. It is, therefore, axiomatic that each increment of power must be accompanied by an increment of air flow unless a lower air-fuel ratio is feasible.

Martin and Wright (8) identified three methods of providing increased power based on greater air flow. These are listed below.

1. Increased engine displacement
2. Increased engine speed
3. Increased charge density.

The first involves an increase of swept volume, either by using a larger cylinder or by having more cylinders per engine. This simply means using a larger engine, and will not, per se, increase specific power since weight, size and cost will also increase.

Increasing engine operating speed as a means of increasing specific output may provide relatively small gains but will be limited ultimately by piston speed constraints. Mansfield (7) discussed the question of engine speed increase and suggested that the disproportionate increase of pumping work, frictional losses and noise were a real disadvantage. A greater problem would be matching the engine speed to market applications. Engines are designed initially to provide a specified output at a given speed or speed range, for example, 1500 and 3000 rev/min for 50Hz power generation. Raising power output by increasing engine speed will usually be commercially non-viable, especially since the use of reduction gears is unacceptable for many applications.

The third means of increasing power output does make possible an increase of specific power judged on weight, size or cost basis. Increasing charge density is readily achieved by the use of relatively cheap, compact and reliable turbochargers. Kamo (9) posed the question "What degree of turbocharging makes economic sense from a standpoint of size, weight, cost, life, performance and sociability ?" He proceeded to define three classes of turbocharged diesel engine based on the bmep.

1. Class I up to a bmep of 140lbf/in^2 (9.7 bar)

2. Class II 140 to 220lbf/in^2

3. Class III above a bmep of 220lbf/in^2 (15.2 bar)

Kamo described many problems to be solved when extending the capability of a Class II engine into Class III. The inevitability of reducing compression ratio to limit gas pressures, and hence mechanical load, would lead to reduced startability and the need for start aids, and the emission of unburned hydrocarbons (white smoke) on light loads. A further disadvantage would be a deterioration of drivability because of turbocharger lag. Kamo was considering the Class III engine for vehicular use only, and reviewed several approaches offering high specific power. These are :

1. Variable compression ratio
2. Fixed low compression ratio
3. Fixed very low compression ratio (hyperbar charged)

and 4. Constant pressure combustion cycle.

He concluded that the use of a fixed "low" compression ratio with the addition of a starting aid represents the most attractive solution. In support of this argument he suggested that existing designs and production tooling could be used as the basis for very wide power ranges, i.e. from Class I through to Class III.

Volvo began a programme of high specific output research in 1970 when they developed an experimental engine known as XD 96. This was based on an existing commercial engine which produced 230 BHP (172 kW) at the same speed, employing a 12 : 1 compression ratio (10). Kamo (9) suggested that high specific output engines could be used to fill gaps in a company's power range, and this is what Volvo did in 1975, based on about five years of research and development. For the British market, Volvo felt that their F88 chassis was underpowered, and to overcome that, they launched the TD100 engine, highly turbocharged and with a compression ratio of 12 : 1 (11). That was the lowest compression ratio used by a commercial engine, and remains so today. For engines of the capacity used in heavy trucks, 13 : 1 is broadly accepted as marginal for reliable starting, therefore the problem facing Volvo was not inconsiderable, especially when we consider the arduous duty and the wide variation of maintenance standards that the truck engine must meet. The problem was solved by using two techniques. Firstly, the engine air intake was fitted with an electrical resistance heater powered by the vehicle's batteries. This was operated manually for one minute prior to cranking in order to heat the mass of charge in the manifold. The second innovation was to incorporate a back pressure regulator immediately downstream of the turbocharger turbine. This restricts the flow of gas from the engine, allowing a proportion of spent gases to remain in the cylinder to heat the next fill. Starting is not the only problem associated with low compression ratio engines : both idling and light loads cause difficulties because of the low turbine energy and consequently low boost temperature giving low compression temperatures and rough combustion. Volvo use the back pressure regulator to ease this problem also, by restricting the exhaust to raise the load on the engine.

Mansfield (7), whilst working for the British Internal Combustion Engine Research Institute, did a great deal to increase understanding of highly turbocharged, high specific output diesel engines. He concluded that turbocharging was the most attractive means available for increasing specific output. An interesting observation made by Mansfield was that in high specific output combustion systems, air movement in the chamber is neither necessary nor desirable. A corollary to that is a reduction in convective heat transfer and hence reduced thermal load and smaller cooling equipment. Unfortunately, no hypothesis for the success of low swirl in this type of engine was proposed. The observation was, however, supported by research at Continental Motors on the AVCR 1100 military engine. Mansfield put forward a cogent argument that thermal loading is not directly related to bmep. He suggested that since the mean heat transfer coefficient and the mean gas-to-metal temperature differential were the determinants of heat flux then, under low compression ratio and normal air-fuel ratio operation, heat transfer would be comparable to a conventional engine. The two central points are that compression ratio controls cylinder pressure and that, in turn with air motion, largely determines the heat transfer coefficient. Therefore, if compression ratio is lowered to maintain normal pressures whilst air motion is reduced or unaltered, the mean heat transfer coefficient must be similar to, or lower than that for the base engine at normal ratings. The second point relates to the mean gas temperature which is largely determined by air-fuel ratio ; since this must be comparable to that of the base engine for performance reasons, then so too must mean gas temperature. The suggestion that thermal load may be reduced by using a high air-fuel ratio has also been made by Kamo (9). The obvious disadvantage of increasing air-fuel ratio is the accompanying increase in charging pressure giving higher mechanical load or the need for a further reduction of compression ratio.

The relationship between compression ratio and thermal efficiency has been widely debated since the use of "low" ratios became attractive. Mansfield made two observations : firstly that frictional losses are substantially independent of bmep, so mechanical efficiency is increased, and secondly, that the possibility of removing heat during the compression process, i.e. after the compressor, reduces the compression work.

Tholen and Killman (12) studied the characteristics of high specific output operation using air-cooled diesel engines. They also noted that mechanical losses did not increase in proportion to bmep at constant speed. They suggested that as the clearance volume is increased to limit peak pressure, the "dead" space becomes less significant. This may not be true, however, in engines that employ a wide valve-overlap period and deep recesses in the piston crown or in the cylinder head.

2.3. Turbocharging Systems

High specific output diesel engines are possible because of the availability of reliable, inexpensive and efficient turbochargers. The choice of turbocharging system is quite wide and will depend ultimately on both technical and commercial criteria. A range of possible systems is given below.

1. Single stage turbocharging with after-cooling
2. Multi-stage turbocharging with inter- and after cooling
3. Variable geometry turbocharging
4. Turbo-compound turbocharging.

Other variables include the design of manifold connecting the cylinder to the turbine. The choice here is between pure pulse, pulse converter or constant pressure systems. A further important consideration is the choice of charge cooling and coolant. The coolant may be air, jacket water, or water from an available cold source (57) (58).

Thus, the number of possible design solutions is large. Fortunately, the findings of several exploratory technical studies have been published in recent years (13)-(21).

Watson et al. (13) concluded that the decision to use single or two-stage (series) turbocharging depended on whether the technical merits of the latter outweighed the cost penalty of two turbochargers and charge coolers. Many of the published papers on turbocharging options have discussed the subject from the standpoint of heavy automotive applications. These require good torque back-up, wide flow range and smoke-free exhaust. The requirements of a constant speed engine are somewhat different. Brock (15), although working with medium speed engines, suggested that even for fixed speed power generating set engines there is a point at which two-stage turbocharging becomes the better solution. He gave several deficiencies of the single-stage system at high pressure ratios, and these are as follows :

1. Narrow surge to choke flow range
2. Deteriorating isentropic efficiency
3. Deterioration of turbine efficiency with the pure pulse system
4. Increasing high frequency noise from the turbocharger.

Watson (13) found that the use of interstage cooling reduces compressor work by lowering the temperature at entry to the second stage. This will provide a greater pressure difference across the engine cylinders, which is an aid to good scavenging and efficient gas exchange. The response of a highly turbocharged engine is generally controlled by the response of the turbocharger. Applications such as power generation load the engine with steps of electrical load. If the magnitude of the load step is high enough the engine may be over-fuelled by the governor and stall before the turbocharger has accelerated to supply the required air to the cylinders. The transient behaviour of two-stage systems is not significantly better than the single-stage equivalent (13) (18). Small gains may be made by careful sizing of the high pressure stage, especially if this can be reduced to a smaller frame size than the equivalent single-stage engine. A further response advantage is the lower speed range of the high pressure stage, since some of the compression will be achieved in the LP stage.

Combustor-assisted, parallel turbocharging (Hyperbar) was developed by a French team in the late 1960's. The system enables operation at very high brake mean effective pressure whilst overcoming some of the disadvantages of more conventional solutions. It is licensed under the name of Hyperbar and, despite its origin as a military engine concept, is now beginning to find application in the commercial field (53). The principle has been described in several published sources (17) (51)-(55), and a schematic drawing is shown in Figure 2.1.(a). The unique features are the engine air by-pass, the combustion chamber and the turbocharger starter motor. These three additions to an otherwise largely conventional engine allow the turbocharger to be operated independently of the diesel engine, i.e. as a gas turbine. The engine employs a fixed, low compression ratio, typically between 5 : 1 and 10 : 1. Starting is facilitated by operating the turbocharger in gas turbine mode to create a supply of hot, compressed charge in the engine manifold, at which time the engine is cranked and started. The by-pass permits optimum operation of the compressor, along the line of highest efficiency by supplementing the speed-dependent engine air flow. This allows the use of high pressure ratio, narrow flow range compressors normally used in small gas turbines. However, under this regime, particularly at low speed and light load, combustion can only be sustained by use of the combustor, thus reducing low speed and part load thermal efficiency as shown in Figure 2.1.(b).

Hyperbar represents a workable, although complicated solution to some of the problems of high bmep operation (up to 30 bar) at the expense of thermal efficiency over much of the load range. The concept is particularly suited to applications requiring low weight or compactness, especially where high grade maintenance is available.

There are three systems in use for transmitting the spent gases from exhaust valve to turbine entry. These are defined by the pressure-time characteristic in the manifold and are as follows :

1. Pure pulse
2. Pulse converted
3. Constant pressure.

The constant pressure system incorporates a high volume exhaust receiver to damp out the pulses issuing from the cylinders, so losing much of the kinetic energy. It is mainly applied to large marine, or fixed speed engines rather than high speed engines because of its large bulk and poor transient response.

Historically, the pure pulse system has been applied to the turbocharged high speed engine. Only relatively recently have the merits of pulse converters been explored. In the pulse system, the blow-down pulse is encouraged to maintain its kinetic energy by providing small diameter exhaust pipes and keeping them as short as possible. Under high bmep operation, the pulses have a large amplitude and are not used with optimum efficiency by the turbine.

A pulse converter is a carefully designed junction, bringing together the gases from two or more cylinder groups. It incorporates a reduction in area, often called a nozzle, in the downstream direction, which converts some of the pressure of the exhaust pulse into kinetic energy. This has the effect of smoothing the pulsations, making possible an increase in turbine efficiency. By taking advantage of the high momentum at exit from a converter the effect of pulse interference between cylinder groups is minimised. Watson and Holness (23) concluded that pulse converters are unlikely to offer any performance advantage over the pulse system except for highly rated engines, or engines with otherwise "difficult" cylinder groupings.

2.4. Combustion and Fuel Injection

The combustion characteristics of an engine are inextricably connected with the fuel injection system and the motion of the charge within the cylinder. The importance of combustion in determining engine performance cannot be overstated ; thermal efficiency, smoke, gaseous emissions, noise, thermal and mechanical load are all largely determined by the combustion process.

The accepted qualitative model of the processes occurring from the start of injection through to the end of combustion is described in reference (24) from which Figure 2.2. is taken. It is a highly generalised representation intended to demonstrate the phases that occur. The relative magnitude of each phase may vary widely, completely altering the shape of the diagram.

Shipinski et al. (25) described the oombustion process similarly, defining four phases which are given below and will be dealt with in turn.

1. Ignition delay
2. Premixed combustion
3. Vaporisation limited combustion
4. Mixing-reaction -rate -limited combustion.

Ignition delay has been the subject of a great deal of research and there are a large number of predictive models available (26) (27). The delay period is partly physical delay whilst the fuel spray disintegrates into small droplets and vaporises, and partly chemical delay, during which chemical reactions proceed so slowly that no effect is discernible. Wolfer (28) and subsequent workers have shown that the most important influences on delay are the prevailing pressure and temperature in the cylinder after the start of injection. The Wolfer expression for ignition delay is given below.

$$ID = \frac{0{\cdot}44e^{\left[\frac{4560}{T}\right]}}{p^{1{\cdot}19}}$$

where ID $[ms]$ T$[K]$ and p $[atm]$

This relationship was determined from work on combustion bombs and has since been shown to give poor correlation with some engines.

Lyn and Valdemanis (56) concluded that pressure, temperature and injection timing were the primary determinants of delay, whilst air motion, fuel injection pressure and nozzle configuration have only a secondary effect.

The second phase is due to premixed combustion. This is characterised by the sharp peak at the start of the heat release diagram. In this stage, the fuel that entered the cylinder at the start of injection, and was subsequently "prepared" by mixing and vaporising, is rapidly consumed by spontaneous combustion initiated at a point within the premixed air and fuel. This phase of combustion is uncontrolled and gives rise to the high rate of cylinder pressure rise responsible for diesel "knock" (30). It is widely held that the proportion of fuel burned in the premixed flame is determined by the length of ignition delay. This explains the smooth combustion of turbocharged engines : the relatively high compression temperatures reduce delay, and hence the magnitude of the premixed peak.

In contrast to the premixed combustion flame, the remaining two phases are characterised by diffusion burning taking place over much of the chamber at the surface of each fuel droplet. Phase three of Shipinski's model suggests that the rate of combustion is determined by the rate at which the liquid centre of the fuel droplets can evaporate at the surface and be consumed in the waiting oxygen. Phase four burning depends on the evaporated fuel finding and mixing with the remaining oxygen. This explains the so-called "tail of combustion".

Khan (31) found that the diffusion flame is responsible for the formation of carbon. This may be released from the engine as soot, which we know as unpleasant exhaust smoke. The formation of carbon is by pyrolysis of unreacted fuel and, at a given time in the combustion process, is being both produced and consumed. The soot leaving the engine will depend on the balance of production and consumption rates, i.e. the net soot release, shown in Figure 2.3. The proportion of fuel burned as a diffusion flame will largely determine the density of exhaust smoke, and this was supported by the work of Khan. As the injection timing is advanced, the delay period lengthens, so increasing the proportion of fuel burned in the premixed phase. Khan suggested that this was the mechanism that causes smoke density to decrease with timing advance.

Mansfield and May (22) found that when operating at low compression ratio, the problem of uncontrolled, premixed burning was exacerbated by the low compression pressure and temperature, and resulting long delay period. They tried three variations of fuel injection : impinging the fuel sprays on the wall of the chamber, introducing approximately 15 per cent of fuel with the charge air and a small pilot injection prior to main injection. They concluded that all three were effective, the first less than the other two. The fuel injection characteristic has a great influence on combustion. However, the complexity of the processes occurring between the start of injection and the end of combustion have prevented the development of a reliable, quantitative model to link the two (59) (61).

Burman and De Luca (60) suggest that the most important injection characteristic is the spray duration. This is especially true at full load, since it directly affects engine power, thermal efficiency and exhaust smoke. They add that for a given chamber and air motion, an acceptable upper and lower duration will exist. Short duration will give good efficiency and a clean exhaust, possibly at the expense of rough running, whilst long duration will give poor efficiency and a smoky exhaust. For the widely used "jerk" system, the controllable parameters that affect duration are many. Table 2.1. is reproduced from reference (60). A second, very important factor is the shape of the injection rate diagram. Although the initial, premixed combustion phase is uncontrolled, the second, diffusion phase is largely influenced by the rate of injection, at least until the point where mixing and oxygen availability take control. Austen and Lyn (29) found that the rate of injection during the early stages, particularly during the delay period, influenced the rate of pressure rise.

Table 2.1 : Effects of fuel injection configuration on duration

	Change	Effect on Duration	Effect on Secondaries
Plunger diameter	increase	decrease	increase
Cam velocity	"	"	"
Nozzle orifice area	"	"	decrease
Tubing bore	decrease	"	"
Tubing length	"	"	"
Trapped fuel volume	"	"	"
Nozzle opening pressure	increase	"	indefinitive
Needle inertia(mass)	decrease	"	decrease
Needle lift	"	"	"
Delivery valve retraction volume	increase	indefinitive	"

2.5. Engine Cooling and Thermal Load

It is a consequence of the second law of thermodynamics that when heat is taken from a hot source to produce work, some of that heat must be rejected to a cold sink. The hot source in an internal combustion engine is the release of chemical energy from the fuel, and the cold sink is ultimately the atmosphere, even if indirectly, via a liquid coolant. There are two important modes of heat rejection : direct to atmosphere with the exhaust gases, and through the engine structure to a coolant. The thermal load of an engine is due to this heat transfer from the hot working gases to the metal chamber walls.

Eichelberg (48) studied the heat transfer in engine cylinders and proposed that forced convection was the primary mode and that radiation was a secondary influence. He suggested that the process was too complex for the formulation of an elegant theoretical model. Instead, he proposed an empirical relationship.

$$h_m = 2{\cdot}1\, c_m^{\frac{1}{3}} (pT)^{\frac{1}{2}}$$

where Cm $\left[\frac{m}{s}\right]$
p $[atm]$
T $[K]$
hm $[kcal/m^2h^oC]$

The use of the term $(pT)^{1/2}$ was an attempt to modify an earlier expression by Nusselt, i.e. $p^{2/3}\ T^{1/3}$ to include the effect of radiation in a simplified manner.

Woschni (49) rejected the pressure and temperature terms of Eichelberg and Nusselt since they were based on free, instead of forced, convection. He worked from the Nusselt number relationship given below, where m = 0.8 for forced turbulent flow.

$$Nu = C\ Re^m$$ where C is a constant.

Woschni also regarded the velocity term $(c_m^{1/3})$ as inadequate and proposed a more comprehensive, two-part term in its place. This was to take account of both organised air motion due to swirl and piston velocity and that due to rapid temperature changes during combustion. The Woschni heat transfer expression is given below.

$$h = 110D^{-0.2} p^{0.8} T^{-0.53}[C_1 c_m + C_2 \frac{V_s T_1}{p_1 V_1}(p - p_o)] \quad ..(2.1)$$

where $D\left[m\right]$ C_1 constant

$P, P_o\ P_1\ \left[Kp/cm^2\right]$ C_2 constant $\left[m/sK\right]$

$T\ \left[K\right]$ $V_s,\ V_1\left[m^3\right]$

$C_m\ \left[m/s\right]$

Of the several predictive models for the heat transfer from engine cylinder gases, those due to Eichelberg and Woschni are most quoted by the literature. The main points of agreement between all who have written on the subject are that the instantaneous heat transfer coefficient varies in some way with gas pressure, temperature and velocity, relative to the wall, and that radiation is present but usually small in effect.

Thermal load is a broad term in need of closer definition. Wu (34) defined it as high absolute temperature and/or steep temperature gradients. High absolute temperature implies two material changes of great importance :

1. Thermal expansion

and 2. Reduction of strength.

A restrained component undergoing expansion will experience thermal stress. If this continues a permanent stress redistribution may occur which, on cooling , will produce a residual tensile stress. This process may lead to thermal fatigue and eventual failure. Howarth (47) outlined the dangers of restrained thermal expansion leading to distortion. This is an acute problem in cylinder heads since they provide a platform for the valves ; any distortion will lead to blow-by and leakage.

A further problem caused by expansion is the reduction or elimination of running clearances which can lead to valve stick or piston seizure. The second high temperature effect, reduced strength, is a clear disadvantage to an engine that generates forces of high magnitude, as the diesel engine does. Table 2.2. presents the widely accepted temperature limitations for engines of conventional design. Howarth states that heat flux is the main cause of thermal load. Given normal coolant temperatures and a favourable coolant side heat transfer coefficient, the gas side metal temperature will be determined by the heat flux and the wall thickness as shown in Figure 2.4.

From this, the importance of taking an overall view of thermal load may be clearly seen. Maximum metal temperature may be reduced by any of the following measures :

1. Reducing gas-to-metal heat transfer coefficient (i.e. gas pressure, temperature or velocity)
2. Reducing gas temperature

3. Reducing wall thickness
4. Increasing material conductivity
5. Increasing coolant-side heat transfer coefficient
6. Reducing the coolant temperature

<u>Table 2.2</u> Widely used thermal limitations of combustion chamber components and materials.

Component	Material	Temperature Limitation	Effect/Location
Cylinder head	Cast iron	400°C	Reduced strength, thermal fatigue e.g. valve bridge
Piston	Aluminium alloy	180°C	Reduced strength, crown support and pin-boss area
Piston	Aluminium Alloy	350°C	Thermal fatigue, leading to crown cracking
Piston and Liner	Aluminium Alloy and Cast iron	240°C	Breakdown of lubrication leading to ring stick or scuffing
Piston	Aluminium Alloy	180°C	Overcooling leading to the formation of acidic compounds and ring groove wear
Exhaust valve	Steel alloys	650-800°C	Distortion, seat wear and thermal fatigue.

The first two measures were alluded to earlier in the chapter. Mansfield (7) and Kamo (9) suggest that high output engines can tolerate less swirl (i.e. reduced velocity at wall) and that gas temperature could be controlled by the use of excess air. The third approach, reducing wall thickness, could only be applied in full knowledge of the mechanical loading, and usually will present little scope. The fourth approach, increasing material conductivity is feasible if a change from cast iron to aluminium is economically acceptable. Howarth states that light alloys are about the most effective materials for cylinder heads because of their high conductivity. The fifth measure, to increase the coolant-to-metal heat transfer coefficient, is essentially a question of providing a high coolant velocity at the metal surface. Howarth quotes typical velocities for water cooling of up to 3 m/s, although this may usually only be achieved through small bore passages. Normally, the velocity will be between 0 and 1 m/s. A graph of waterside metal temperature against water velocity is given in Figure 2.5.

The region of appreciably constant metal temperature is controlled by nucleate boiling, i.e. at low water velocity. Above this, as water velocity is increased, the transition is made from nucleate boiling to forced convective heat transfer. Nucleate boiling is a normal condition at certain spots in highly rated engines, providing a"safety valve", since very high rates of heat transfer are possible under this regime. In areas of high heat flux, it is advisable to provide a machined surface if cast iron is the material, and a high, local water velocity. This can be achieved by providing drilled passages, for example, between the valves. The benefit of machining the as-cast surface from cast iron is the removal of an apparent thermal barrier, although the phenomenon is not fully understood. The final method of reducing metal temperatures, reducing the coolant temperature, offers relatively limited scope since this will require a larger radiator or heat exchanger as the coolant will be rejecting heat at a lower temperature.

The foregoing discusses approaches to cooling when the heat source and sink are on opposite sides of the chamber wall. The piston and valves of a conventional engine pose a greater problem, since they are not amenable to direct cooling (38) - (44). Typically the piston receives heat through the crown and rejects it through the ring pack (38) to the liner, as shown in Figure 2.6.(a) Although there are other heat flow paths, notably through the top land and the upper surface of the rings, the path shown predominates.

Figure 2.6. (b) is a linear representation of the flow, and this demonstrates the accumulation of conductivities and interface heat transfer coefficients which combine to create the problem. It is, therefore, essential to apply additional cooling means at relatively low engine ratings. Munro (36) suggests that approximately 2.2 BHP/in^2 (65 MW/m^2) of piston area is the highest rating that could be tolerated without additional cooling. The accepted means of cooling the pistons of high speed diesel engines is by providing an oil flow to carry away the heat - either to the under-crown area or to a gallery cast into the piston between the crown and the ring pack, as shown in Figure 2.6.(c) and (d). The advantage of the cast-in gallery over under-crown cooling is the proximity of the sink to the source. This reduces the temperature

gradient for a given heat flow. There is, however, a cost penalty of approximately 50 per cent, according to data published by Munro and Griffiths (50). The film heat transfer coefficients of the various oil cooling arrangements are given below in Table 2.3.

Table 2.3. Boundary conditions for oil-cooled pistons

Undercrown	Cast-in gallery	h [kW/m^{2}K]	Ref.
Crankcase atmosphere (dry small end)		.25-.33	(40)
Splash from small end lubrication		2.13-2.78 2.04	(40) (42)
Jet from small end or standing pipe		3.27-9.4 3.68	(40) (42)
	"Cocktail-shaker" (Gallery part full)	2.04-2.86	(40)
	" 100RPM " 300RPM " 2000RPM	.286-.572 .85-1.145 2.86-5.674	(41)
	Solid (gallery full)	1.145	(41)

The part-filled gallery is known as a "cocktailshaker" for obvious reasons. From Table 2.3. it may be seen that its effect is related to engine speed. At all but the lowest speeds, a part-filled gallery is superior to the"solid" full-flow gallery design.

Although other means of piston cooling have been suggested, for example, water cooling and air blast, for small high speed engines lubricating oil is most used because it is relatively easy to apply to existing designs.

2.6. Performance Modelling

Many diesel engine performance models have been proposed, and they vary from highly simplified to complex and extremely descriptive. In the first category are air-standard cycles. These may be used to establish "ideal" performance as a guide to the absolute limitations of practical performance. Used with judgement, the air-standard equations form the basis of a "quick and dirty" model for estimating cycle variables in actual engines.

Digital computers have made the use of very detailed predictive techniques a practical proposition, with their ability to perform a large number of calculations accurately and quickly. The more complex programs model the combustion processes, attempting to predict the rate of combustion and the gaseous species formed by it (68)(69)(70)(73). Maguerdichian and Watson (73) developed a combustion model which divided the fuel jet into a large number of elements, as shown in Figure 2.7. This model considered air motion, spray penetration, evaporation, ignition delay, chemical kinetics and air-fuel mixing. Despite the apparent detail of the program, the results achieved were not of sufficient value to justify the use of such a complex program. Given further development, this type of program may be of value to engine designers, but Watson has reported that even if developed to good accuracy, it would be too large and time-consuming for day-to-day use in engine design and turbocharger matching. The main disadvantages of this type of highly detailed model are the large amount of input data, requiring many skilled man-hours, and the degree of experimental support essential for program development. A compromise between over-simplification and extreme complexity is available in the "filling and emptying" method. This technique considers the engine system as a series of volumes, junctions and shafts, as shown in Figure 2.8. The calculation proceeds from a set of initial conditions in each volume by discrete time steps, numerically solving the system differential equations for internal energy and mass flow. The rate of change of internal energy is given by the following equation :

$$\frac{dU_{cyl}}{d\varphi} = p_{cyl}\frac{dV_{cyl}}{d\varphi} + \sum \frac{dQ_w}{d\varphi} + \frac{dQ_c}{d\varphi} + h_a\frac{dm_a}{d\varphi} + h_e\frac{dm_e}{d\varphi}$$

The main assumptions necessary for simplification are that the fluid properties are constant for each time interval and uniform throughout each volume, thus spatial variations are not considered. The technique is widely used in research institutions and industry (62)(63)(65)(67), probably because it is a sufficiently detailed and representative model for design purposes yet has modest data requirements. Janota stresses that the underlying strength of the method is in predicting the rate of change of engine performance indicators with the design parameter of interest (75). It is not capable of predicting absolute values with acceptable accuracy. Some examples of the studies undertaken by this method are given below.

1. Turbocharger matching
2. Valve timing
3. Injection timing
4. Varying compression ratio
5. Boundary conditions for thermal and mechanical stress programmes
6. Studying the effects of wall temperature.

The filling and emptying method integrates several predictive models, each representing mathematically a function of the engine system. A description of some heat release models taken from the literature, is provided in the following section.

Heat release models

Several "combustion" models have been developed for use in filling and emptying programmes (46)(63)(64)(72). Although essentially these represent combustion, in practice, a model of the heat energy released by combustion is sufficient. This gives rise to the concepts of rate of heat release and heat release diagrams. Three models will be discussed. The first uses the injection rate and ignition delay to compute the rate of heat release, the other two rest on the assumption that the shape of the heat release diagram is determined by the conditions in the engine cylinder when a normally matched fuel system is used.

Possibly the first attempt to model the combustion process was made by Austen and Lyn (46). This considers the injection rate diagram and divides the fuel into a series of masses, each corresponding to a fixed time interval during the injection period. Each fuel mass is "prepared" for combustion according to an empirical formula, and the heat release diagram is then found by equating the fuel prepared during the delay period with the fuel burned in the initial pre-mixed "spike". This may be seen in Figure 2.9.(a). Although the author reported satisfactory correlation from this model it has not been widely used. The most obvious disadvantage is the need to predict the injection rate diagram, which is not easily achieved.

Woschni (64) used a mathematical heat release function proposed by Wiebe, as the basis of a heat release model. The Wiebe function is a universal relationship which can be tailored to a wide range of combustion systems. The Wiebe rate of heat release function is given below.

$$\frac{dQ_c}{d\varphi} = C_2(C_1+1)\,\varphi^{C_1}\exp\left(-C_2\varphi^{(C_1+1)}\right)$$

It is most generalised when two are combined to model both pre-mixed and diffusion flames (76). Woschni, however, found that a single function was adequate to model the heat release diagrams of his medium speed engine, Figure 2.9.(b).

The function may be adapted by changing the constants C_1 and C_2. C_1 is a shape factor, typically between 0.5 and 1.6 ; C_2 is a completeness factor. Varying C_2 within the range 5 to 10 varies completeness from 99.3 to 100 per cent. Throughout his work, Woschni used a value of 6.9.

Starting from an experimental heat release diagram, to which a Wiebe function is fitted, changes in shape and duration are computed from changes in correlation parameters, as shown below.

$$C_1 = C_{1_0}\left[\frac{T_0}{T}\right]\left[\frac{N_0}{N}\right]^{\cdot 3}\left[\frac{ID_0}{ID}\right]^{\cdot 5}\left[\frac{p}{p_0}\right]$$

$$\Delta\varphi = \Delta\varphi_0\left[\frac{A/F_0}{A/F}\right]^{\cdot 6}\left[\frac{N}{N_0}\right]$$

Suffix 0 = reference condition

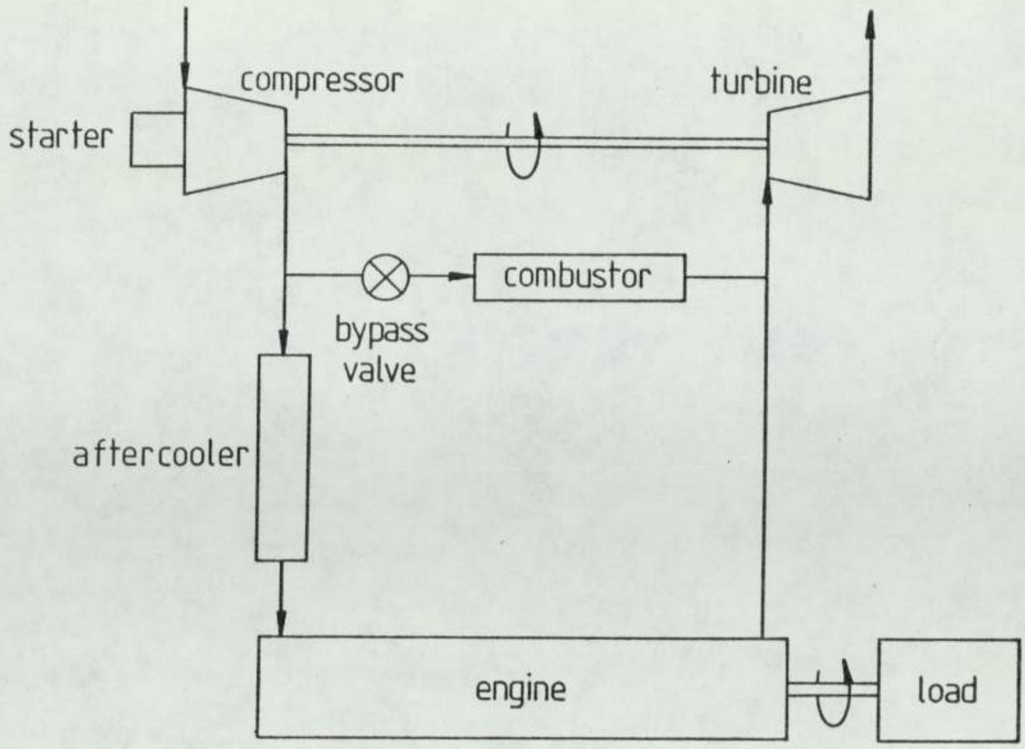
The final heat release model is due to Marzouk and Watson (72) (74)(82). This is more applicable to high speed engines since it includes both premixed and diffusion burning. The diffusion phase is a function capable of very steep rates of rise and decay, typical of diesel combustion. They used high speed engines (Leyland and Perkins) to provide experimental heat release diagrams over a wide load and speed range. The shapes of these diagrams were found to correlate with ignition delay, air-fuel ratio and speed. The rate of heat release function is the sum of the pre-mixed and diffusion equations shown below.

Premixed $$\frac{dQ}{dt} = C_1 C_2 t^{C_1 - 1}(1 - t^{C_1})^{C_2 - 1}\ \beta$$

Diffusion $$\frac{dQ}{dt} = C_3 C_4\ t^{C_4 - 1}\ \exp(-C_3 t^{C_4})(1 - \beta)$$

β is the phase proportionality factor and determines the relative magnitude of pre-mixed and diffusion phases.

For high speed engines, this model would appear to offer more than the Woschni correlation of the Wiebe function. Whereas the previous model did not require injection data because it used a datum heat release diagram, the Marzouk-Watson model assumes a normally matched fuel system. A disadvantage of the latter models is that the combustion duration is not predicted. However, by trial and error methods or using hard data, an acceptable value may be found quite easily. A typical Marzouk-Watson heat release function is given in Figure 2.9.(c).



(a)

(b)

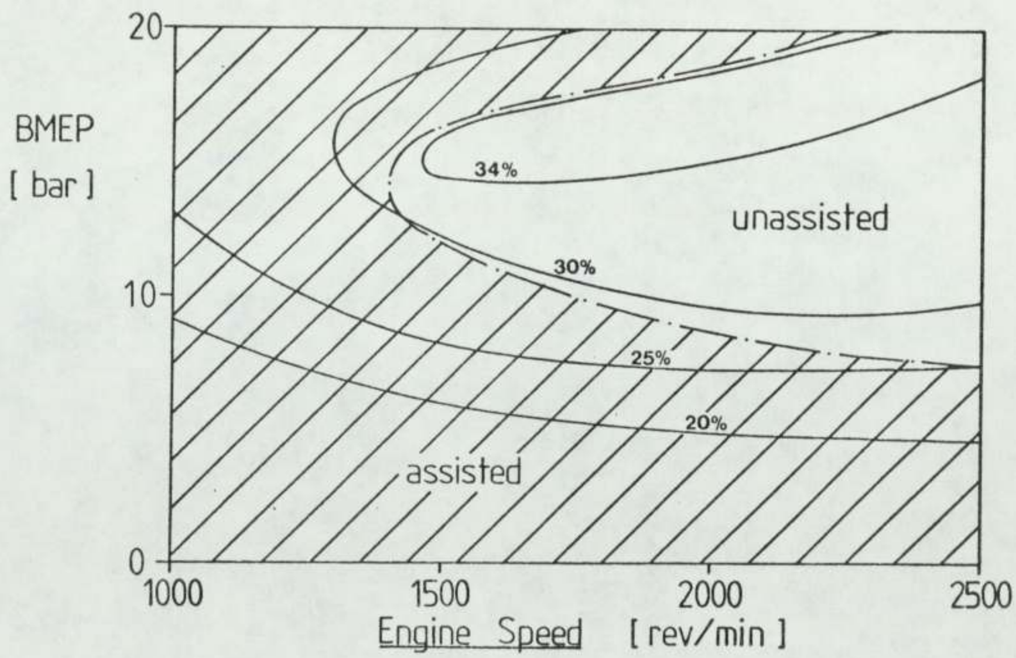
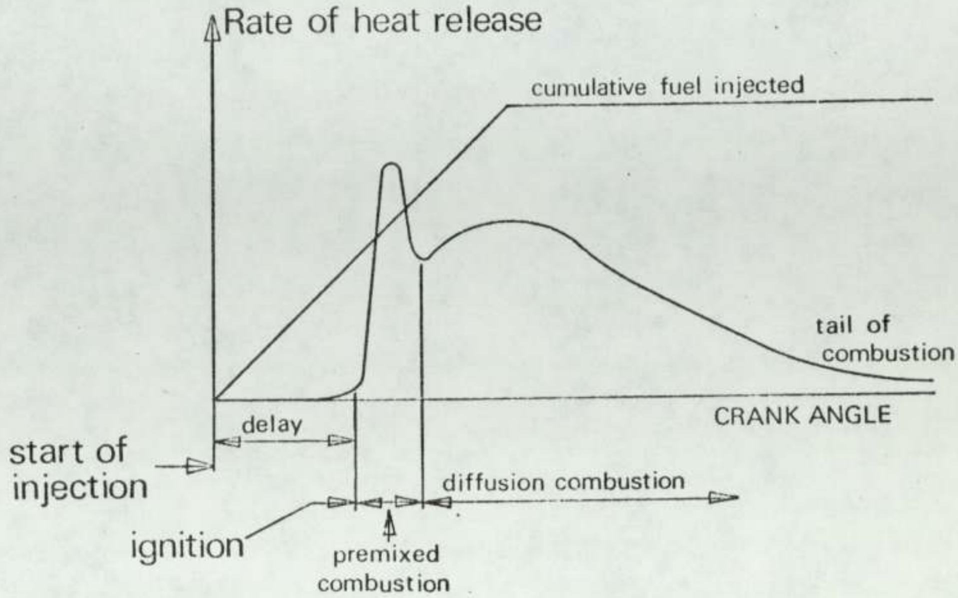


Fig. 2.1, a. Schematic representation of the 'hyperbar' system.
b. Hyperbar performance map, due to Wallace (17), showing region of combustor assistance and lines of iso-efficiency.

Fig.2.2, Qualitative model of diesel engine combustion, taken from ref.(24).



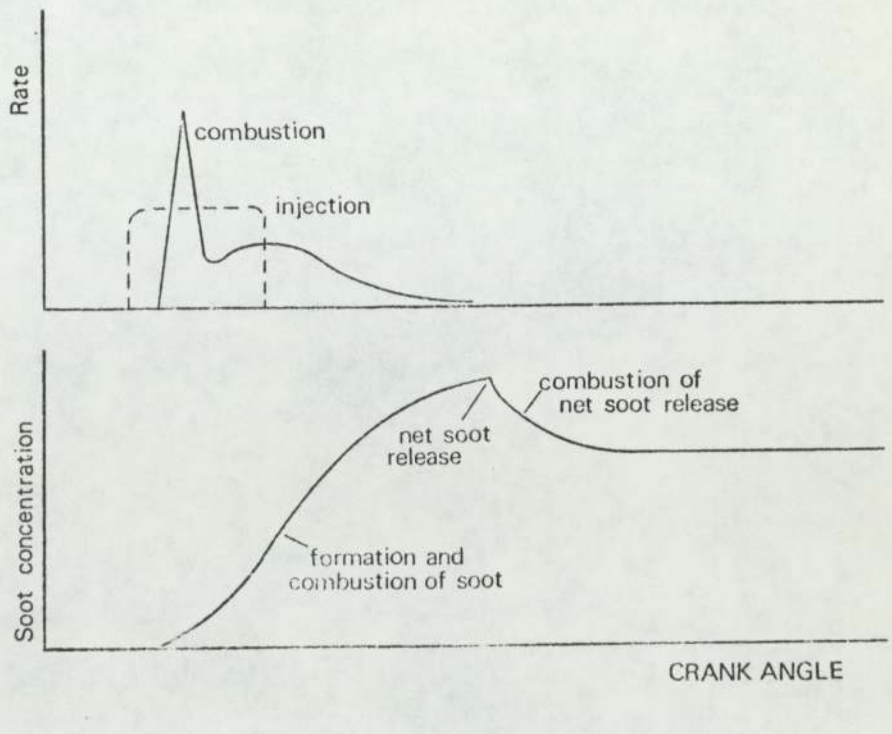


Fig. 2.3, Model of soot formation proposed by Khan (31).

(By permission of the Intitution of Mechanical Engineers)

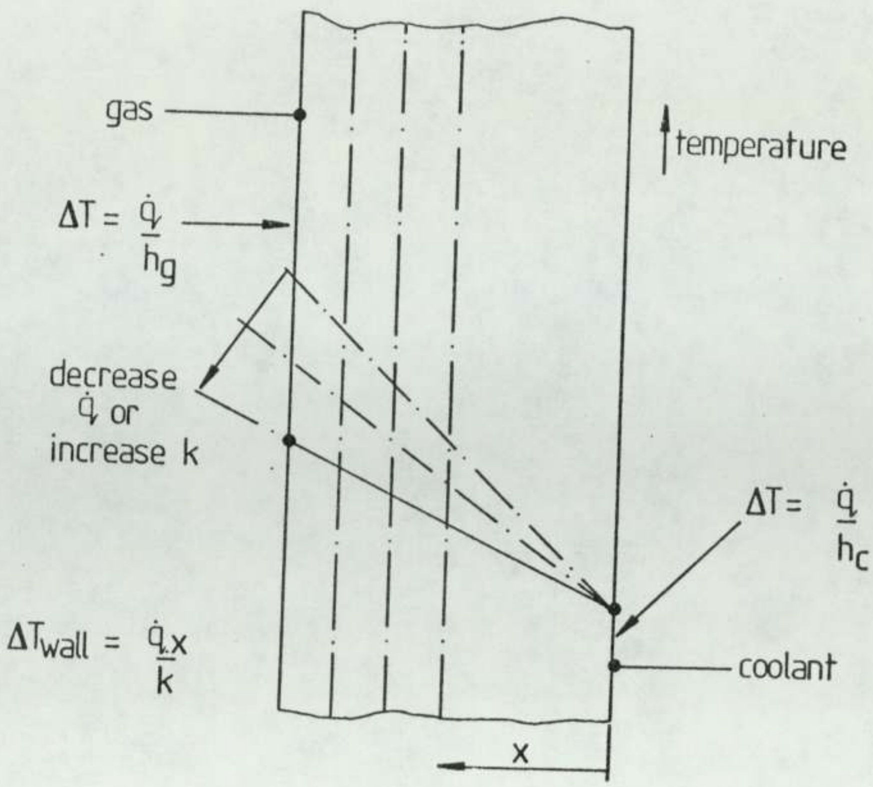


Fig. 2.4, Factors determining the temperature distribution in a plane wall for one dimensional heat flow.

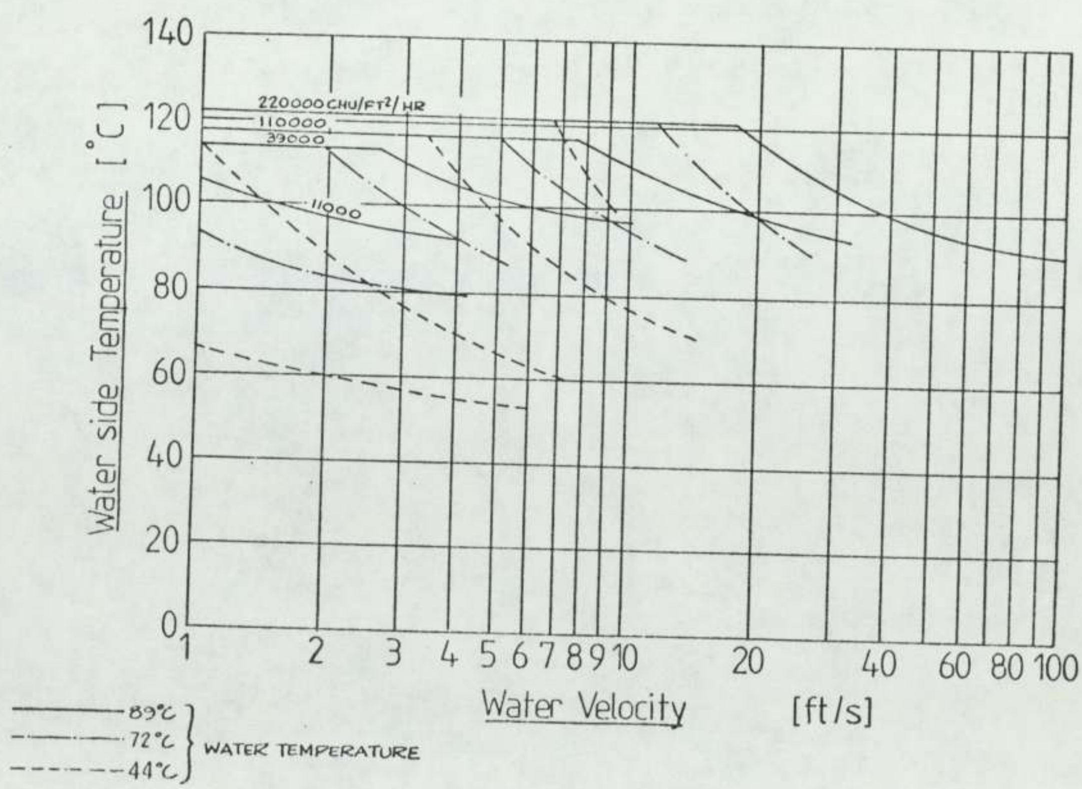
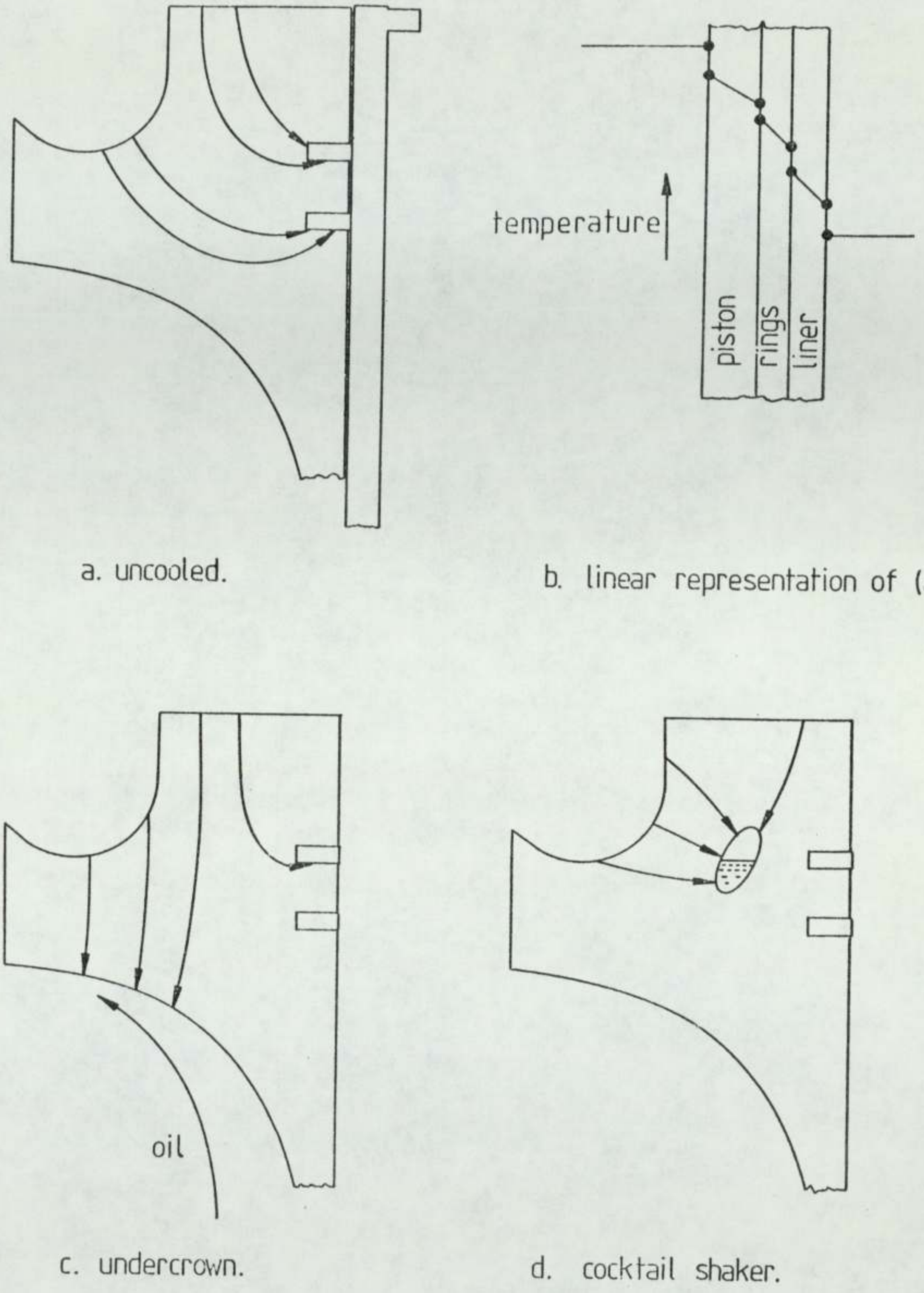


Fig. 2·5, Effect of coolant velocity and temperature on water side wall temperature, at four levels of heat flux. Ref. (47).



a. uncooled.

b. linear representation of (a).

c. undercrown.

d. cocktail shaker.

Fig. 2.6, Heat flow in diesel engine pistons with various cooling provisions.

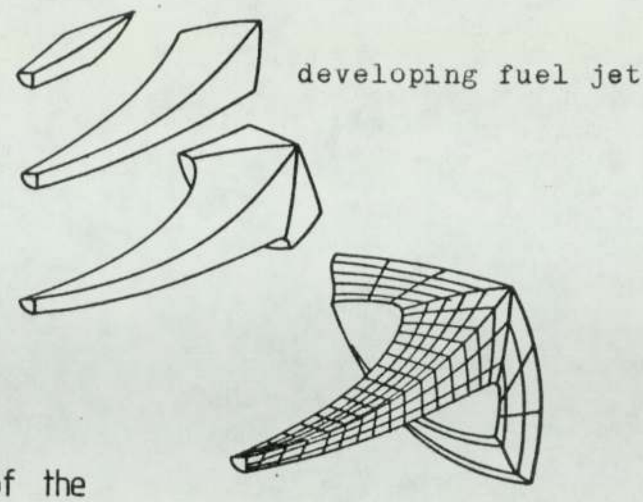


Fig. 2·7, An impression of the combustion model due to Maguerdichian and Watson (73), showing zone motion with deflection and impingement against the bowl.

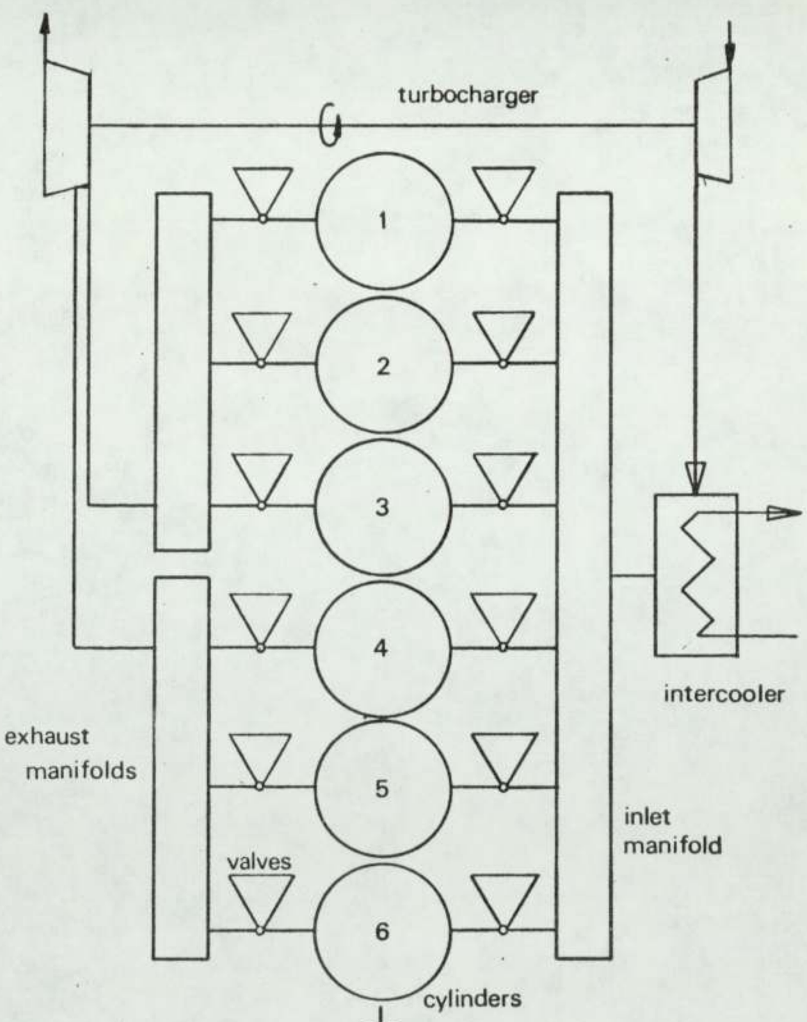


Fig. 2.8, 'Filling and emptying' representation of an in-line engine as a series of volumes, junctions and shafts.

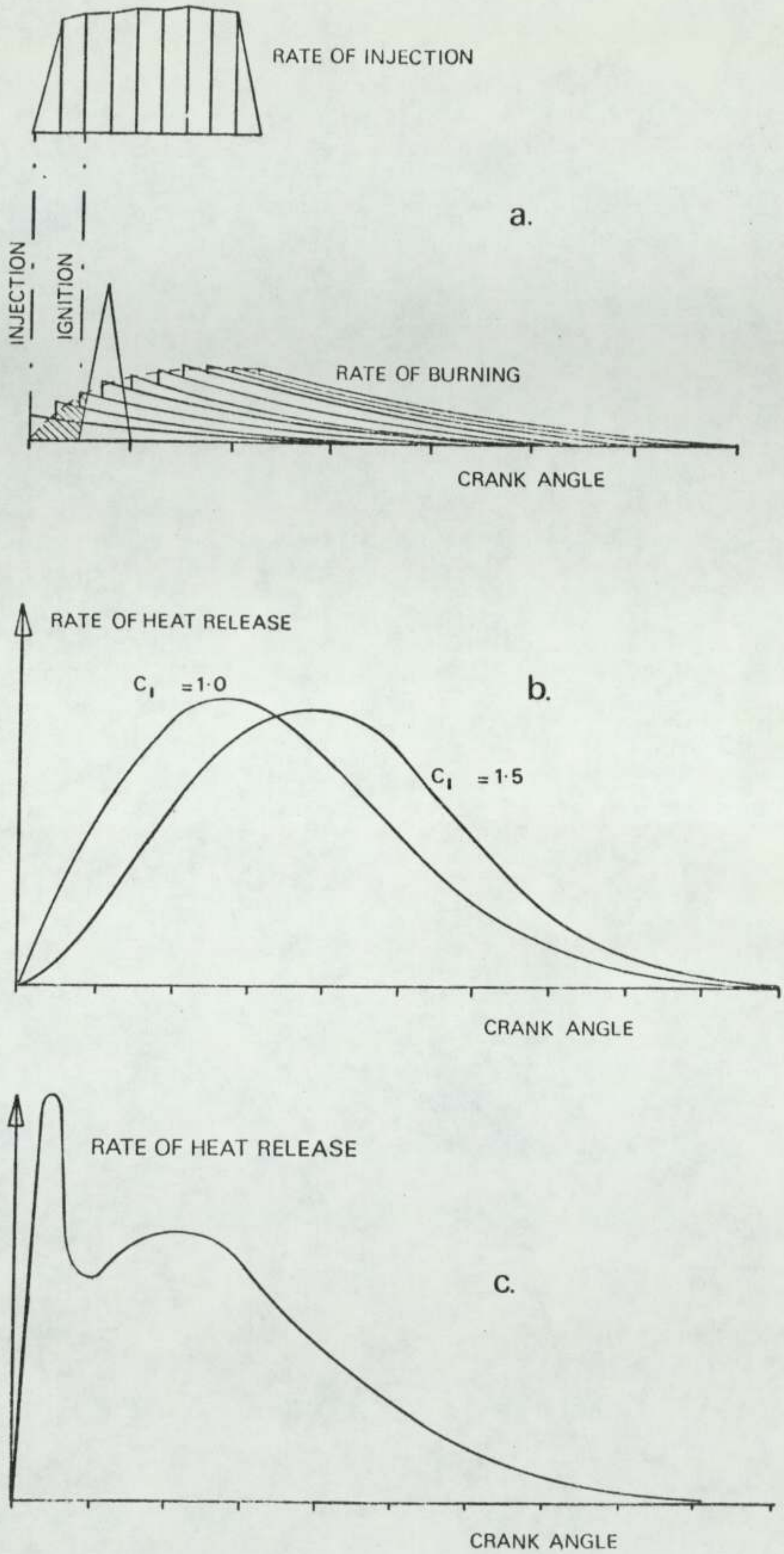


fig.2·9, Heat release models a. Lyn's method (46)
b. Wiebe function c. Marzouk-Watson. (82)

CHAPTER 3

ENGINE PERFORMANCE MODELLING

3.1. Introduction

This chapter discusses the practical issues of performance modelling. In particular, the sensitivity of the 'filling and emptying' method to the necessary assumptions and simplifications, is discussed.

The prediction of the temperature of cylinder components is discussed.

3.2. The "Filling and Emptying" Model

The filling and emptying model is a practical proposition because of the power of digital computers. The routines that form the model are straightforward and uncomplicated, but so numerous that hand calculation is out of the question. In the last twenty years, the great increase in calculating power has been used to solve many engineering problems which previously required expensive experimental study. Of course, computer models do not obviate the need for prototype testing and routine development. Properly used, however, they enable the designer to optimise and make trade-offs, and to perform studies that may be prohibitively expensive to do with hardware - for example, the variation of bore/stroke or valve area ratios.

The contribution that filling and emptying can make to engine design is to reveal the important trends of energy and mass flow as design features are varied over the area of interest. Table 3.1 presents some of the design parameters and performance indicators that may be studied by this method.

Many writers have emphasised that the method cannot accurately predict absolute performance. The principal reason for this is the complexity of the combustion process. Until the effective rate of heat release can be derived from a physical description of fuel injection equipment, cylinder and port geometry, absolute values will have little meaning. At present, highly simplified heat release models are used to bridge this gap in our knowledge and allow the study of performance, relative to some datum.

The filling and emptying method makes many simplifying assumptions, whilst retaining the essential characteristics of the diesel engine. There are two groups of assumptions that should be recognised : those implicit in the method and those made by the modeller in applying the method. These assumptions are listed in Table 3.2. Although the second group are essentially input data, it is unlikely that such information would be available. If the design is completely new, all four inputs would have to be assumed, perhaps by adapting data from similar engine types. If the engine exists in some form, then valve area and swirl data may be available, but it is most unlikely that wall temperature or heat release data would be. The following discusses the sensitivity of the model to the input data most difficult to estimate - valve effective areas, wall temperature, swirl velocity and heat release rate.

Table 3.1. Some of the design parameters and performance indicators that may be considered in a filling and emptying study.

Design Parameters	
Bore/stroke ratio (79)	Compressor efficiency
Conn.rod/throw ratio	Turbine effective area
Compression ratio (62)(79)	Charge cooler effectiveness
Valve areas (79)	Charge coolant temperature
Valve area ratio	Manifold volume
Valve timing (79)(80)	Chamber wall temperature (83)
Engine speed	Swirl velocity
Turbocharger efficiency	Allowable cylinder pressure
Performance Indicators	
Indicated thermal efficiency	Scavenge ratio
Brake thermal efficiency	Charge temperature
Volumetric efficiency	Turbine inlet temperature
Charge pressure	Mean cycle temperature
Peak cylinder pressure	Mean heat transfer coefficient
Turbine expansion ratio	

Effective valve area

The effective valve area for a given lift allows the same mass flow under ideal, frictionless conditions as the actual valve, for the same pressure ratio. Dent and Derham (77) describe a method of obtaining the effective area as a function of lift. The valve flow characteristics are evaluated under steady-flow conditions at a number of discrete lift values. The data so obtained are used to calculate the effective area using the following equation :

$$A^* = \frac{\dot{m}}{\left(\frac{P}{P_o}\right)^{\frac{1}{\gamma}}\sqrt{\left(\frac{2\gamma}{\gamma-1} P_o \rho_o \left(1-\left(\frac{P}{P_o}\right)^{\frac{\gamma-1}{\gamma}}\right)\right)}}$$

Table 3.2. : Filling and emptying programme assumptions

Implicit in the method

1. Gases are ideal
2. Volumes are shapeless and have uniform gas properties which remain constant through each calculation step
3. Heat transfer surface areas of the chamber have uniform, non-varying temperature
4. There is perfect mixing when fresh and spent gases occupy the same volume
5. Pressure wave action is neglected
6. Inertia of gas columns is neglected.

Made in applying the method

1. Mean gas-side temperature of chamber components
2. Swirl rotational velocity for heat transfer calculation
3. Timing, shape and duration of heat release diagram
4. Valve effective areas versus crank angle.

This relationship is based on compressible flow theory. It is allowable to use incompressible flow theory for pressure ratios near to unity. The error introduced for a pressure differential of 10ins (254mm) water is about -2.5 per cent.

Clearly, the flow through the valves of a working engine is not steady. Both valve area and pressure ratio are changing during the induction and exhaust periods. However, this does not lead to significant error since the effective area is substantially independent of pressure ratio as shown in Figure 3.1. taken from Ref. (77). Woods and Khan (81) reported similar results, but with a tendency for effective area to increase slightly with pressure ratio at high lift.

The effective valve area/lift relationships used throughout this work are shown in Figure 3.2.

Wall Temperature

The model differentiates between "constant" and "variable" wall areas. The cylinder head, piston crown and valve heads present a constant surface area to the cylinder gases through the cycle. The cylinder liner area is variable, since varying amounts of it are exposed during the cycle. The total surface area may be represented by any number of smaller areas, but each must have a uniform surface temperature. For example, it is allowable to divide the cylinder cover into two parts, say an area representing the exhaust valve at a temperature of 900K and an area representing the flamedeck at 600K. These may be reduced to an equivalent area (Aeq) at an equivalent temperature (Teq) by using the following equations :

$$A_{eq} = \Sigma(A_i) \qquad T_{eq} = \frac{\Sigma(A_i T_i)}{\Sigma(A_i)}$$

The sensitivity of the model to errors in the assumed mean wall temperature is shown in Figures 3.3 and 3.4. As may be expected, the heat flux is most sensitive. This is because it is calculated from the mean heat transfer coefficient and the temperature drop available between gas and wall. The general engine performance, typified by volumetric and brake thermal efficiencies, exhaust and mean cycle temperature, is largely insensitive.

Swirl

The heat transfer from the working gases was discussed in Section 2.5 where the works of Eichelberg (48) and Woschni (49) were reviewed. They found that gas pressure, temperature and velocity, relative to the wall, largely determine the instantaneous heat transfer coefficient.

Since swirl is the organised rotation of the charge about the cylinder axis, the relative tangential velocity of the gas to the liner will be given by :

$$V_t = \frac{\pi D N_{sw}}{60}$$

The Woschni heat transfer model (Equation 2.1.) may be adapted to include a swirl term in the velocity component. The effect of varying the assumed mean swirl, over a suitably wide range, is shown in Figure 3.5. This shows that the prediction of engine performance generally is unaffected by the swirl assumption. The heat transfer coeffiecient, as expected, increases with swirl speed, causing an increase of heat flux.

It should be possible to estimate the mean swirl to within reasonably close limits using swirl rig data, which means that inaccuracies from this source should be quite small.

Heat Release Function

Despite our almost comprehensive qualitative understanding of the combustion process, the quantitative link with the fuel injection equipment and engine geometry has not been made. The problem is partly overcome by the use of heat release models which assume a relationship between engine parameters, such as air-fuel ratio and speed, and the shape of the heat release diagram. Two such models are described in Section 2.5 ; the Marzouk-Watson model, and the Woschni correlation of the Wiebe function. If we accept the gross simplification of such models in the absence of a more realistic approach, two further assumptions remain : heat release timing and duration.

Lyn (71) studied the influence of heat release shape, timing and duration on the development of pressure in the cylinder. Some of the results of this study are shown in Figure 3.7. Although Lyn varied the three parameters over an unrealistically wide range, the results do demonstrate the effects that the various assumptions may have. The heat release assumption has little effect on the expansion curve followed by the gases after heat release. This confirms the belief that the heat release diagram is relatively unimportant for open-cycle studies such as turbocharger matching and valve optimisation. For closed-cycle studies, the choice of heat release diagram, timing and duration completely determines the final result. This underlines the futility of predicting rate of pressure rise, thermal efficiency or peak cylinder pressure in absolute terms.

Exhaust Manifold

The "filling and emptying" method considers the exhaust manifold as a shapeless volume having uniform thermodynamic properties at any given point in the cycle. The model assumes instantaneous and complete mixing as the cylinders discharge and "fill" the manifold. The actual processes occurring in an engine's exhaust system are quite complex which is why such simplification is necessary. Normally, a pressure wave, or blow-down pulse, leaves the cylinder early in the exhaust period. This travels at sonic velocity towards the turbine entry. At points such as the turbine, or junctions with other branches, reflections can occur which further complicate the situation.

Janota et al. (75) studied the likely errors introduced by this simplification. Figure 3.8 gives some of their results for a wide range of manifold shape and volume. The factor

$$\frac{12 L_e N}{\sqrt{(\bar{\gamma} R T_e)}}$$

indicates manifold length and is actually the elapsed time in crank degrees for a pulse to travel from valve to turbine and back. The filling and emptying method tends to underestimate the exhaust energy under all conditions, but less so for short, low volume manifolds.

They also studied the accuracy of the transient temperature prediction. Generally, the maximum and minimum values were well predicted, the most significant error being a phase-shift. This was thought to have been caused by the instantaneous and complete mixing assumption.

Alternative methods for modelling the events in the exhaust manifold are available, although these are generally rather complex and expensive to process. One such method is the method of charactersitics. This is a mathematical technique for solving the differential equations set up by considering the one-dimensional, compressible, unsteady flow in the manifold.

Calculation Step Size

The filling and emptying method proceeds by a series of finite time steps, solving the system differential equations at discrete points. Between these points, the thermodynamic properties of the working fluid are assumed constant. Therefore, the smaller the time step, the more accurate the numerical solution will be, whilst the computer time will necessarily increase.

Figure 3.6 shows the effect of increasing the step size from the widely used 1 degree of crank rotation up to 6 degrees. The most marked effect is the rapid decrease of cost, a function of computing time only, to less than half the initial cost as the step size is increased from 1 to 3 degrees. The calculated results diverge by up to 5 per cent when the interval is increased to 2 degrees, and do not further diverge until the 6 degree interval.

The average cost at the start of 1980, of running a model of a 6-cylinder engine with turbocharger through three calculation cycles of 1 degree step size was £4. It could be argued that the large number of simplifying assumptions inherent in the method already undermine the predictive accuracy, and that introducing further inaccuracy by increasing step size would be unacceptable. Whilst this is a valid argument, dramatic cost savings may be made with relatively little divergence of predicted result. For large-scale studies requiring many computer runs to reveal broad trends, perhaps cost and predictive accuracy could be traded to good effect.

Model Construction

Filling and emptying models require large computing capacity, and will generally be too large for present day desk-top computers. The program used throughout this work was produced by GEC Diesels Limited and is currently available through Midland Computer Services on the Stafford IBM 370, to which Dorman now have a direct line.

The program is part of a performance prediction suite which also includes heat release analysis of indicator cards, valve area calculations and a Wiebe heat release model.

Constructing a model is relatively straightforward. First, the parts of the engine required to solve the problem are defined. This might be just one cylinder and two manifolds, or the complete engine with turbocharger and charge cooler. This is then represented schematically (see Figure 2.8) as a series of interconnected volumes, junctions and shafts. In this respect, the program is versatile, since any thermo-fluid system describable in this way may be tackled by this method.

The computer handles numerical values only, therefore the various elements are numbered. The input data describes the interconnections and defines which volumes are working (cylinders), which junctions have variable area (valves) and which shafts deliver work from the system (crankshaft). Once the geometric data (bore, stroke, etc.) have been listed, valve area, swirl speed, wall temperature and heat release data are input.

When modelling the turbocharger, one of three levels of sophistication is possible. The most basic simply modifies the inlet temperature and pressure to simulate the presence of a compressor and cooler. The exhaust is then restricted by a fixed area junction to represent the turbine. The next level of sophistication allows the mean flow rate to be specified. Continuity is achieved by allowing the compressor to find the required pressure ratio, consistent with the volumetric efficiency of the engine. The temperature rise through the compressor is calculated using the pressure ratio and a specified efficiency. Charge cooling may be included and is defined by the effectiveness, coolant temperature and pressure loss. The turbine is represented by a swallowing curve, defined by the mass flow at choke. Turbocharger power balance is achieved by the external control of the modeller, normally by adjusting the turbine area.

The third level of sophistication involves describing the full characteristics of both compressor and turbine. The program allows the turbocharger to "float" and find power balance, influenced only by the engine, as in reality. The performance maps of the compressor and turbine are input in tabular form in which the program then interpolates, when necessary.

The level of turbocharger model that may be used will normally be determined by the availability of performance data. Most engine makers buy turbochargers from outside suppliers. Therefore, access to the required data could be limited, especially for turbine maps which are jealously guarded by the makers. This is generally not for commercial secrecy, but to prevent misinterpretation, since steady-flow maps should be treated cautiously when translating to pulsating engine flow conditions. This emphasises the difficulties of turbine modelling and casts doubt on the value of full-map modelling if the data are compiled under steady-flow conditions.

In the early stages of this project the basic model was used and later, when swallowing curves were released by the manufacturer, the second level model was tried. Experience now indicates that the benefits of the second model over the first (specifying mass flow and charge cooling) are a significant advantage, whilst the use of swallowing data is of little help in predicting turbine energy under pulsating flow. The most satisfactory approach is to adjust the turbine characteristic externally to give the required energy for power balance.

Having constructed the model this far, there only remains to specify initial conditions for shafts and volumes. The precise details of the foregoing are given in Reference (84).

3.3. Thermal Predictive Models

The temperature fields within combustion chamber components are an important consideration in diesel engine design. High temperatures, or gradients, reduce component life, or may lead to catastrophic failure caused by thermal fatigue, loss of strength, distortion or loss of lubrication.

In an earlier section, the feasibility of detailed performance models was accredited to the coming of powerful digital computers. This is also true of heat conduction models, which require many straightforward calculations, far beyond the scope of hand calculation. Generally, the methods of temperature prediction divide the component into a number of cells or elements. The modeller is then required to specify the heat transfer conditions at the boundaries of the region. The program's solution algorithm then interprets these boundary conditions and derives a temperature field that satisfies the principle of energy conservation.

Provided the component is adequately represented, by careful subdivision, the overall accuracy will depend on the quality of the boundary conditions.

Boundary Conditions

The heat source is the combustion of fuel in the cylinder gases which gives rise to rapidly varying pressure and temperature. Thus, the heat transfer boundary conditions on the gas-face of the cylinder components are not steady-state. However, little error is incurred by assuming a quasi steady-state condition, using the arithmetic time-mean temperature and heat transfer coefficient. The gas-side boundary conditions used for the piston and liner study reported in 4.3 and Appendix 2, were derived from the filling and emptying model of the engine, using the Woschni (49) heat transfer model.

The boundary conditions between the piston, the rings and the liner were taken from the work of Woolley (44), and are shown in Figure 3.9. The undercrown heat transfer coefficient, with and without oil jet cooling, was taken from the same source, and is given in Table 2.3. The research engine that Woolley used to study the heat transfer was an earlier marque of the Dorman "L" series, a 3-cylinder "LB" engine.

The piston and liner were analysed separately. Allowance was made for the heat transfer between the two by weighting the coefficients by the residence time that the nodal areas of each are in contact. The water-side heat transfer coefficient was calculated from the Nusselt number relation for turbulent flow, given below.

$$h_c = 0{\cdot}023\,\frac{k^{0.6}}{d^{0.2}}(\rho v)^{0.8}\left(\frac{C_p}{\mu}\right)^{0.4}$$

where the constant (.023) is dimensionless.

The water velocity was assumed to be only 60 mm/s around the liner. However, the gas-side conditions largely determine the heat transfer, so that the calculated result is relatively insensitive to this assumption. Figure 3.10 shows that for a 300 per cent increase of the water-side coefficient, the calculated metal temperature decreased by only 42^{o}C. This calculation was performed on the I.H.D. Department's Hewlett Packard System 45 micro-computer, using a program written by Mr. D.C. Hickson.

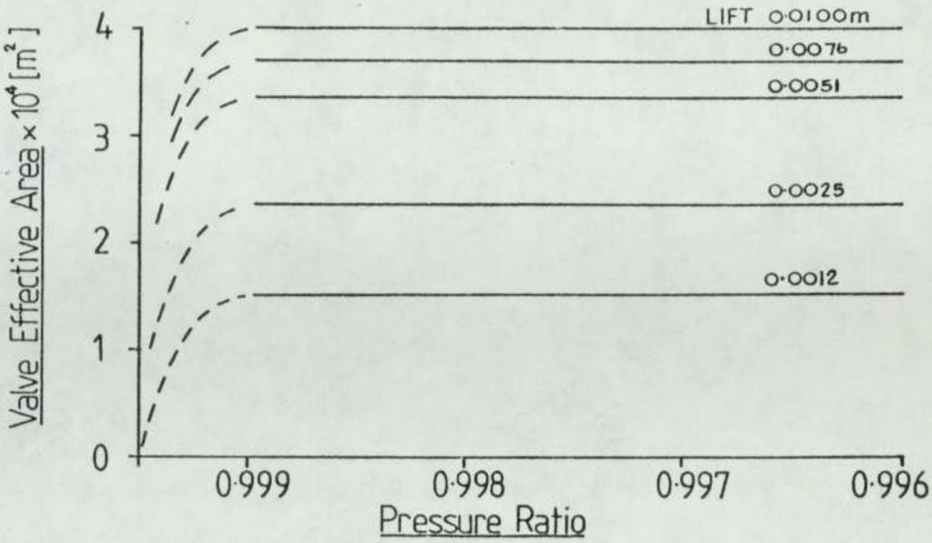


Fig.3.1, Valve effective area versus pressure ratio across the valve. From Dent and Derham, ref.(77).
(By permission of the Institution of Mechanical Engineers)

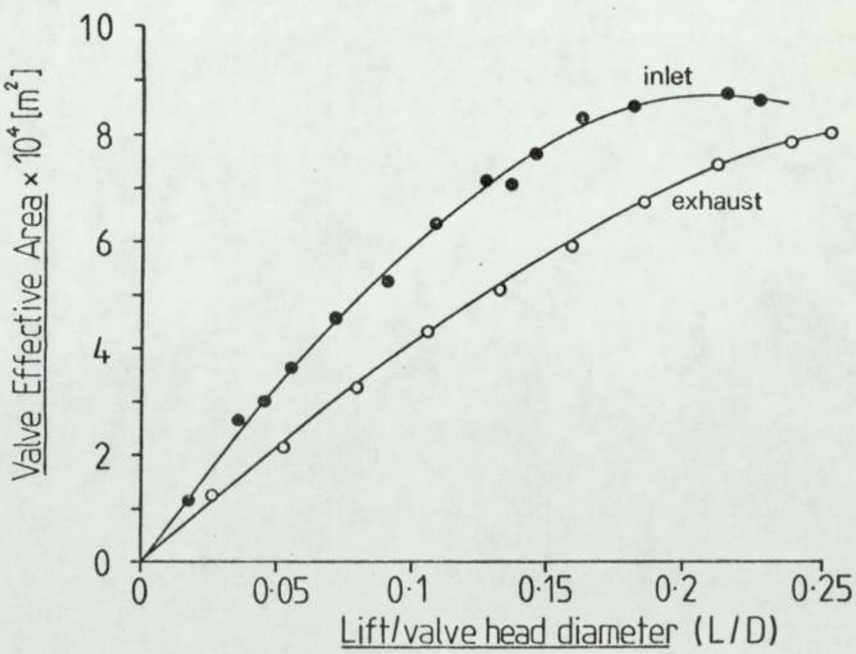


Fig. 3.2, Valve effective area against non-dimensional lift for the Dorman 'LE' engine. Derived from steady-flow experiments.

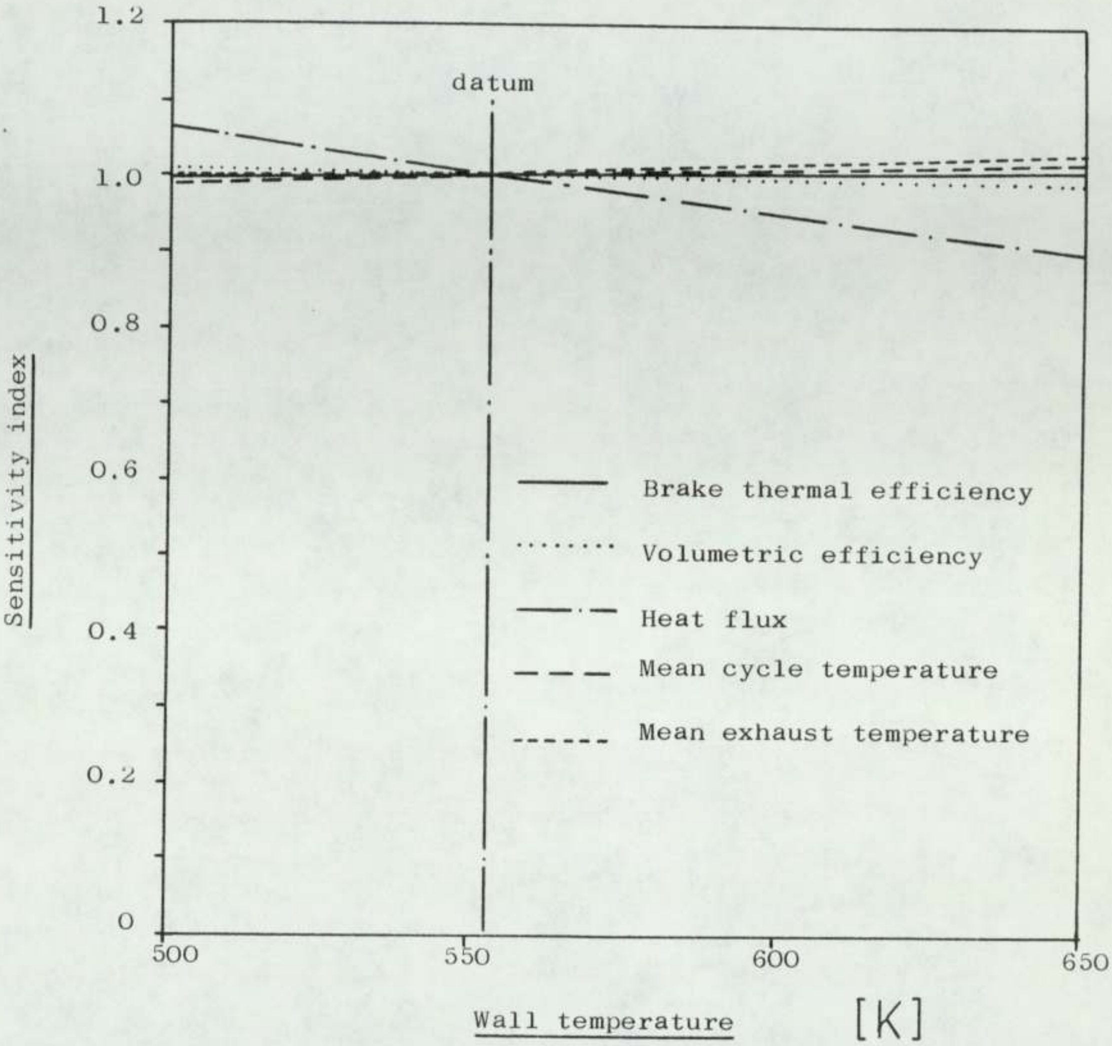


Figure 3.3. : Sensitivity of the "filling and emptying" model to the assumed wall temperature.

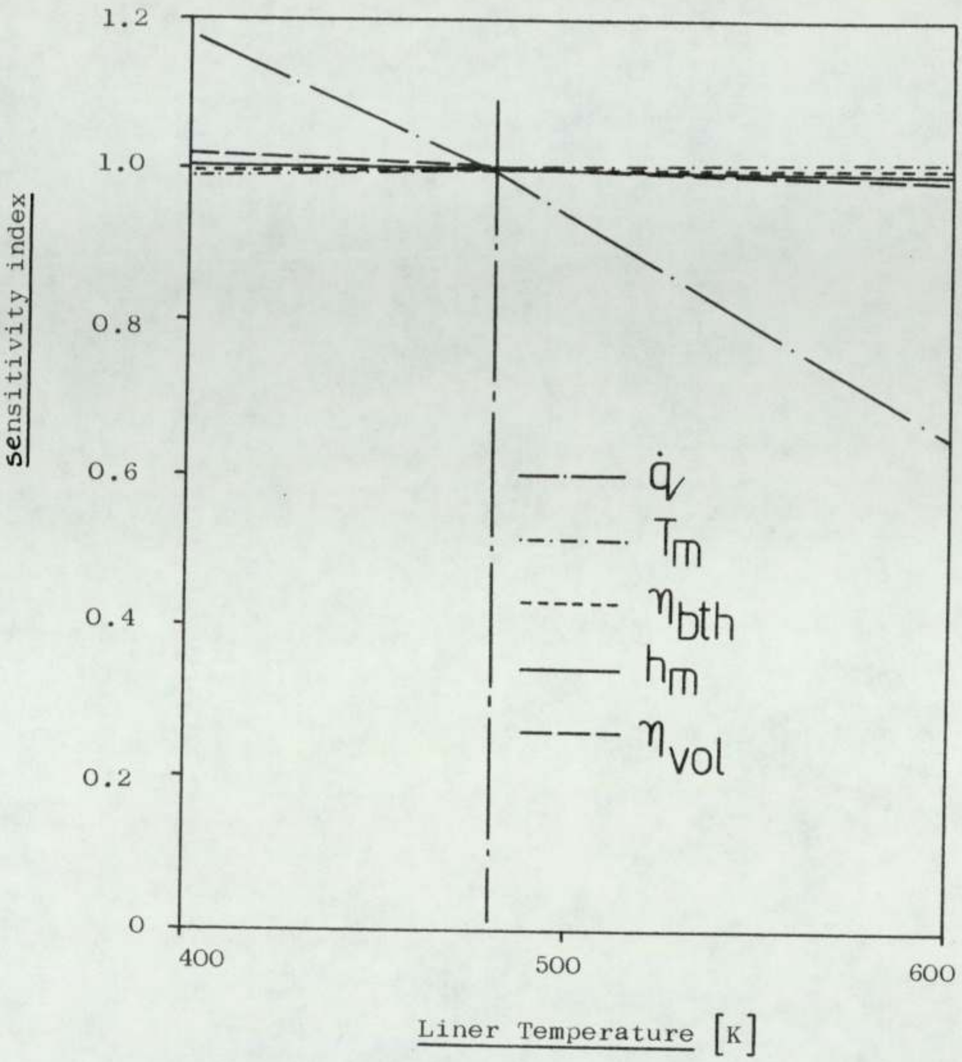


Figure 3.4. : Sensitivity of the "filling and emptying" model to assumed liner temperature

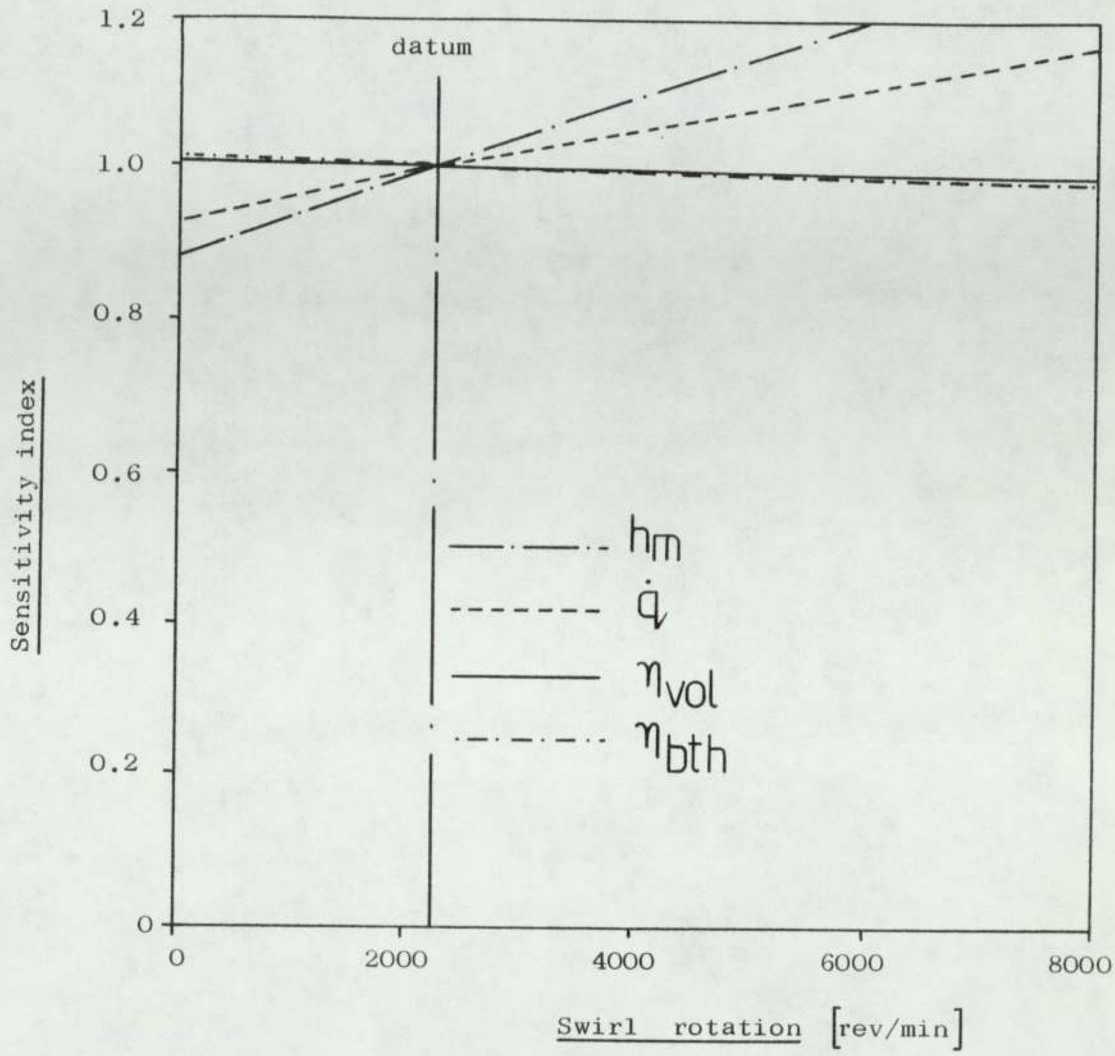
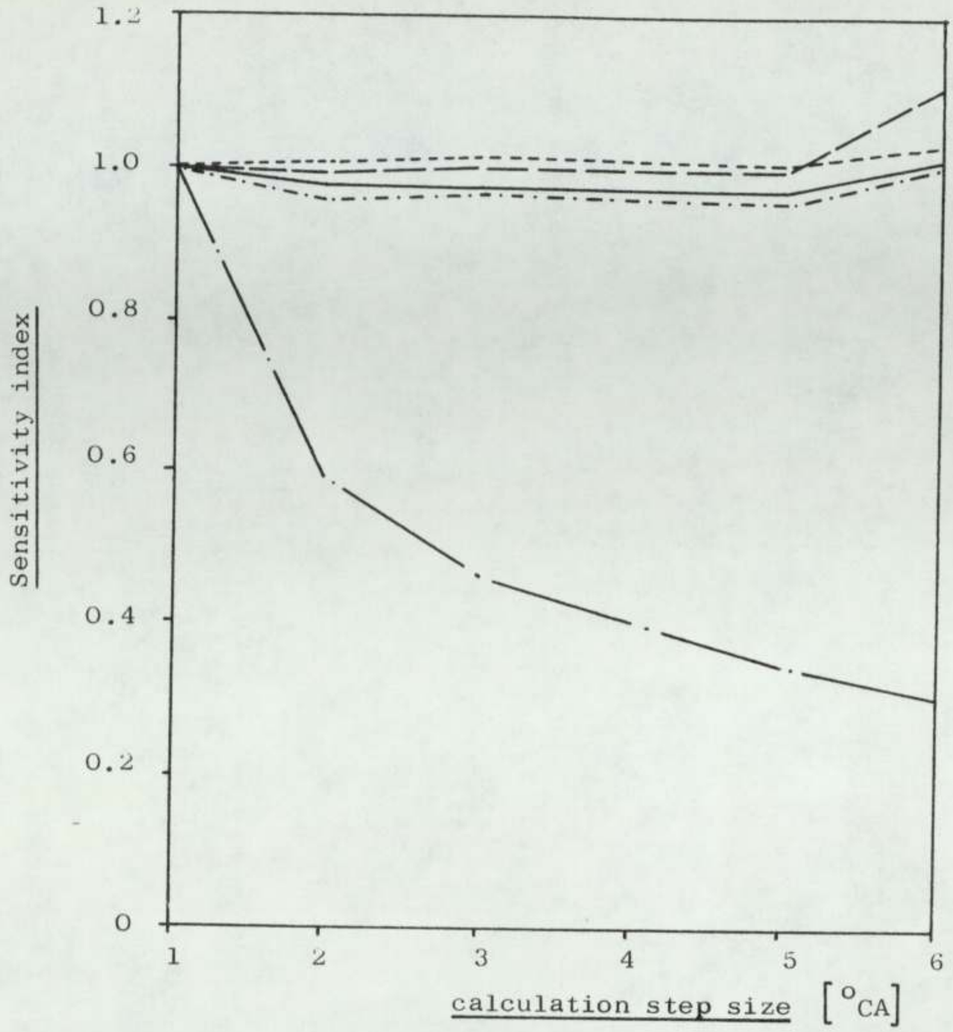


Figure 3.5. : Effect of the swirl assumption on the results predicted by the "filling and emptying" model.



------ Mean heat transfer coefficient

——— Maximum cycle pressure

——— Brake mean effective pressure

·—·—·— Mean exhaust temperature

·———·· Computing cost

Figure 3.6. : Sensitivity of the "filling and emptying" model solution to the calculation step size.

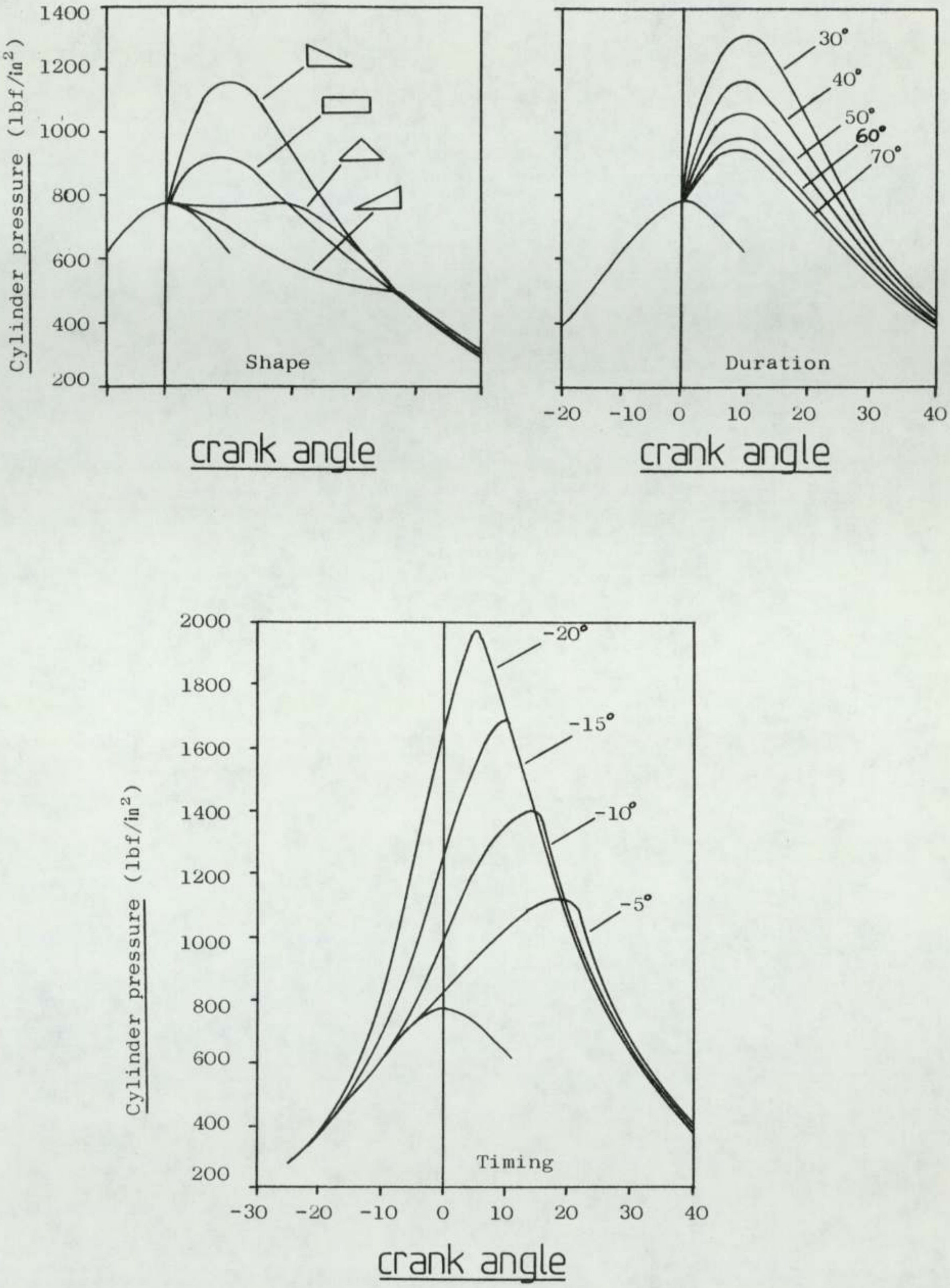


Figure 3.7. : Variation of cylinder pressure development with assumed heat release diagram shape, duration and timing. Due to Lyn (71).

(By permission of the Institution of Mechanical Engineers)

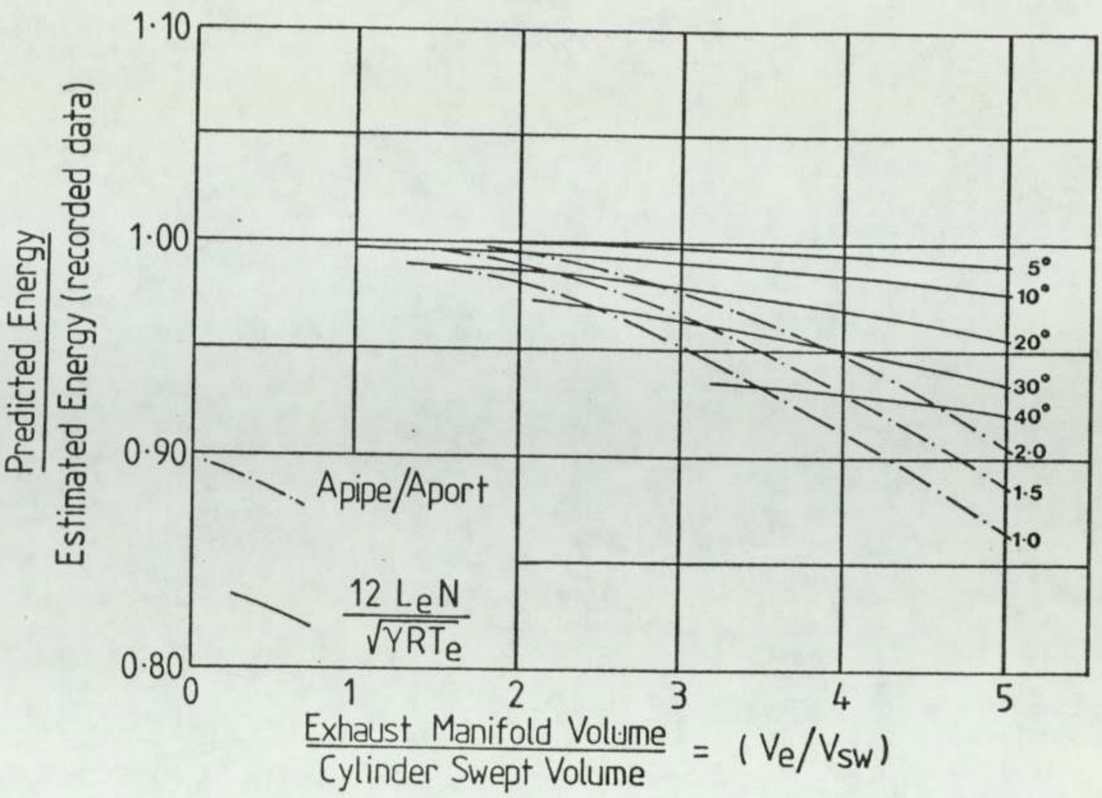
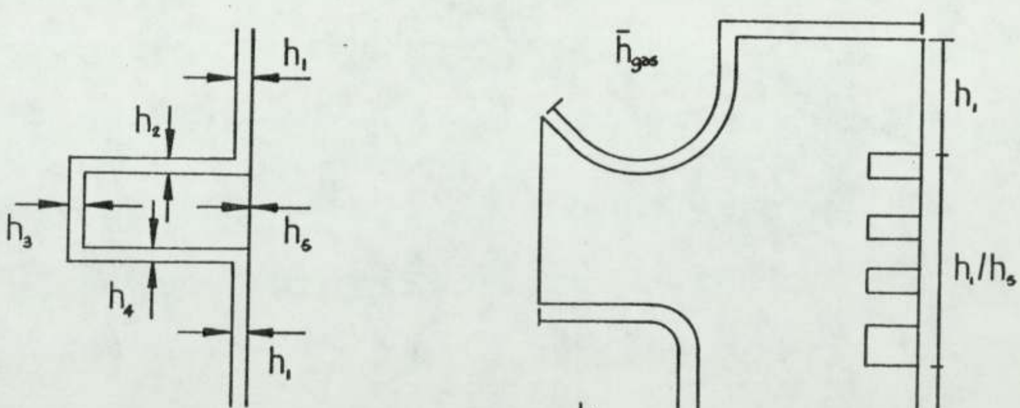


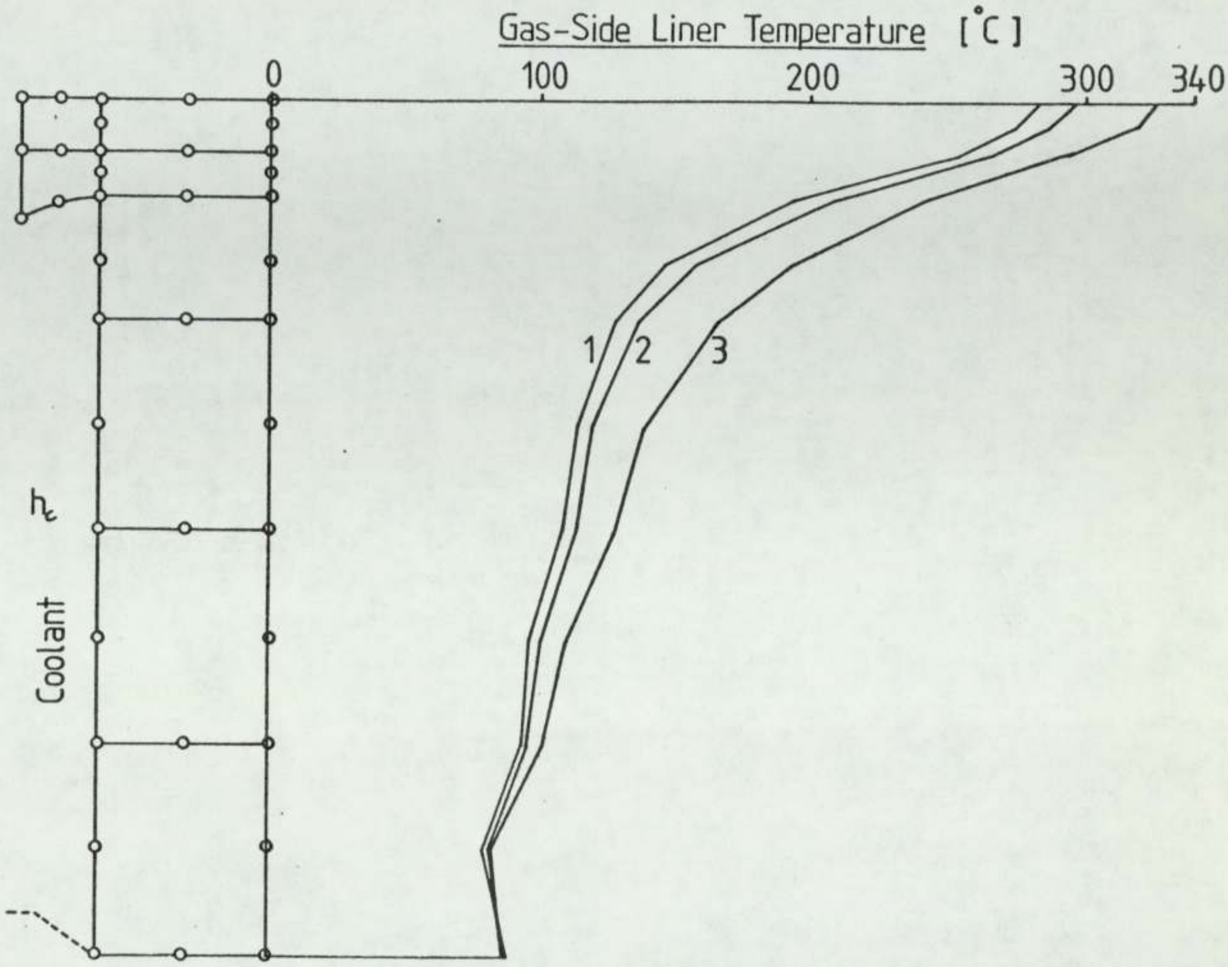
Fig. 3.8, Accuracy of the exhaust energy prediction for a single cylinder engine. From Janota et al, ref. (75).

(By permission of the Institution of Mechanical Engineers)



h_1	0·695 $kWm^{-2}K^{-1}$
h_2	0·695 "
h_3	0·695 "
h_4	11·69 "
h_5	69·50 "

Fig. 3·9, Film heat transfer coefficients used in the prediction of piston temperature. Due to Woolley (44).



1	h_c	=	8·52	$kWm^{-2}K^{-1}$	(1500	$CHU\,ft^{-2}h^{-1}K^{-1}$)
2	h_c	=	5·68	"	(1000	")
3	h_c	=	2·84	"	(500	")

Fig. 3.10, Prediction of cylinder liner temperature, showing the relative insensitivity to the water-side heat transfer coefficient. Finite element program due to Hickson.

CHAPTER 4

Design Study and Research Engine Specification

4.1. Introduction

The project brief given by the sponsor required the evaluation of high output technology through an experimental programme based on a current production engine. Further discussion produced the following terms of reference.

1. The research vehicle will be the Dorman in-line, six-cylinder "LE" engine.
2. The target performance will be a bmep of 300lbf/in^2 (20.7 bar) at a fixed speed of 1500 rev/min.
3. The compression ratio will be fixed.
4. Single stage turbocharging is to be used, if feasible.
5. The concept should be studied primarily with the needs of commercial, rather than military, users in mind.

This broad definition was used to guide the design study and to specify the research engine's detail design. Each item will be discussed in turn :

1. Base engine : this would appear clear and unambiguous but the question arises - how much may be changed ? The answer used in design decision-making was - as little as possible. An intrinsic value of a high output variant would be that it make fullest use of existing production facilities and components.

2. Target performance : the question of rating (continuous or intermittent duty) cannot be satisfactorily answered at this stage. The design objective was to operate the engine satisfactorily at a steady-state bmep of 21 bar.
3. Compression ratio : this rules out the use of variable compression ratio schemes.
4. Turbocharging : The simplicity, cost, size and weight advantages of single stage turbocharging are preferred over multi-stage alternatives.
5. End use : this is the least tangible guideline, but was taken to mean that the traditional virtues of the diesel engine should be preserved so far as is practicable.
   a) Design solutions should be simple, rather than highly "sophisticated" and avoid the use of exotic materials.
   b) Thermal efficiency, reliability, durability and sociability should be carefully considered in design decisions, where this is possible.

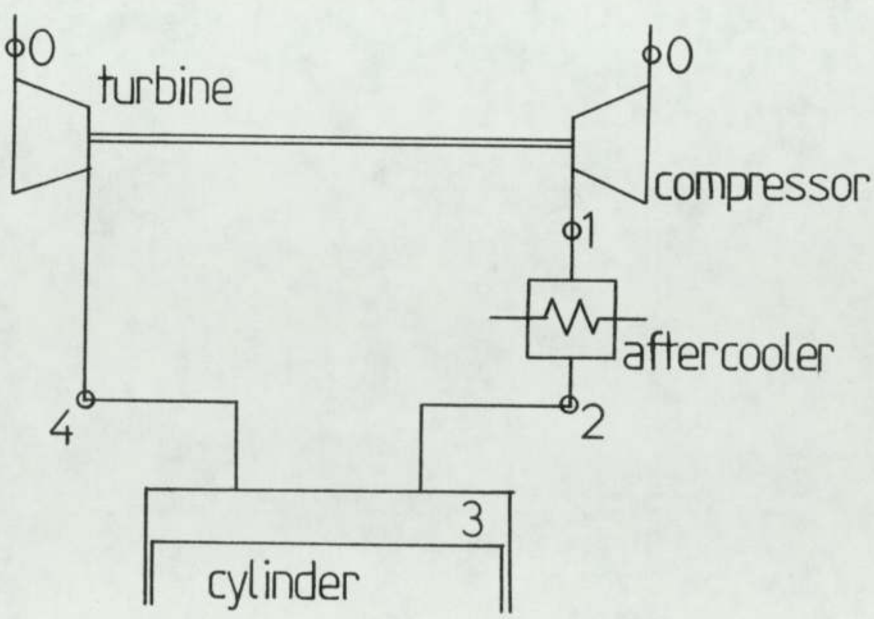
Many engine design problems may be studied by hand calculation and this was the approach, when suitable. Some problems are mathematically too complicated or simply too long-winded, necessitating the use of computer programs. In particular, gas exchange, turbocharging, heat release and heat transfer calculations were handled in this way.

The design study attempts to reveal the underlying trends of thermal efficiency, mechanical and thermal load and other performance indicators, with variations in design. In this process, the engine is reduced to manageable sub-systems for analysis. By contrast, the research engine specification is the synthesis of these sub-systems, it must achieve the target performance quickly because of the time constraints, and have the design flexibility to allow for contingencies and a degree of experimentation.

In some areas, notably turbocharging and fuel injection, the equipment is manufactured by other companies. In such cases, the expertise of the maker's representative was recognised and components were specified in the light of their recommendations. Specific areas of expertise within the company were used where appropriate, for example, in translating the required valve motions into a working cam shaft and preparing the engine layout and installation.

4.2. Preliminary considerations

The following analysis is intended to enable study of the key engine parameters and their relevance to increasing bmep. It is highly simplified and contains assumptions which may be questionable if accuracy is sought. These are pointed out where appropriate.



1. Air demand :

$$\dot{m}_f = isfc.\,P_i \qquad 4.1.$$

also

$$P_i = imep.D^2.L.N.n\frac{\pi}{4} \qquad 4.2.$$

and by definition

$$A/F = \frac{\dot{m}_{a,tr}}{\dot{m}_f} \qquad 4.3.$$

combining 4.1, 4.2, 4.3 :

$$\dot{m}_{a,tr} = A/F_{tr}\, isfc\, imep\, D^2 L N n.\frac{\pi}{4} \qquad 4.4.$$

2. Air supply :

$$\dot{m}_{a,tr} = \eta_{vol,tr}.\rho_2.D^2 L\frac{N}{2}n\frac{\pi}{4} \qquad 4.5.$$

$$\rho_2 = \left(\frac{P}{RT}\right)_2 \qquad 4.6.$$

neglecting pressure loss through the aftercooler

$$T_1 = T_o\left[\left\{\left(\frac{P}{P_o}\right)^{\frac{\gamma-1}{\gamma}} - 1\right\}/\eta_c + 1\right]$$

assuming the aftercooler coolant is at the ambient air temperature

$$T_2 = T_o\left[\left\{\left(\frac{P}{P_o}\right)^{\frac{\gamma-1}{\gamma}} - 1\right\}(1-\varepsilon)/\eta_c + 1\right] \qquad 4.7.$$

substituting 4.7 into 4.6 and 4.6 into 4.5 and letting Po = 1 bar

$$\dot{m}_{a,tr} = \frac{\eta_{vol,tr}\eta_c P_2 \pi D^2 L N n}{T_o\left[(P_2^{\frac{\gamma-1}{\gamma}} - 1)(1-\varepsilon)+\eta_c\right]} \qquad 4.8.$$

equating air supply and demand - 4.4 and 4.8 and rearranging for imep

$$imep = \frac{\eta_{vol,tr}\eta_c P_2}{A/F_{tr}\, isfc\, T_o\left[(P_2^{\frac{\gamma-1}{\gamma}} - 1)(1-\varepsilon)+\eta_c\right]2} \qquad 4.9.$$

3. Mechanical load :

This may be simplified without undue loss, to consideration of the maximum cylinder pressure.

explosion ratio $$\psi = \frac{P_{max}}{P_{comp}} \qquad 4.10.$$

This is the ratio of peak cycle pressure to hypothetical compression pressure at tdc. To simplify the analysis, let compression commence at bdc, and:

$$P_{bdc} = P_2 \qquad 4.11.$$

combining 4.10 and 4.11 and introducing the polytropic relation for compression (n = 1.35) :

$$P_{max} = P_2\, \psi\, CR^{1\cdot 35} \qquad 4.12.$$

Rearranging 4.12 for P_2 and substituting into 4.9 :

$$imep = \frac{\eta_{vol,tr}\eta_c P_{max}}{isfc\, T_o\left[\left\{\left(\frac{P_{max}}{\psi\, CR^{1\cdot 35}}\right)^{\frac{\gamma-1}{\gamma}} - 1\right\}(1-\varepsilon)+\eta_c\right]A/F_{tr}\,\psi\, CR^{1\cdot 35}.2} \qquad 4.13$$

:

This expression relates the major design parameters to the indicated specific output. The incalculable nature of combustion is contained in the only performance parameter in the equation, isfc.

The isfc is primarily a function of the combustion system but secondary influences exist, such as air-fuel ratio, compression ratio and explosion ratio. The latter may be taken as a measure of injection timing for a given combustion system. If isfc could be taken at some reference condition, as a constant, it could be included in equation 4.13 with correction terms as follows :

$$isfc = isfc_o \cdot f\left(\frac{CR}{CR_o}\right) \cdot f\left(\frac{A/F}{A/F_o}\right) \cdot f\left(\frac{\psi}{\psi_o}\right)$$

A typical correction equation for air-fuel ratio is the Wanschiedt equation given in reference (3). Corrections for compression ratio and explosion ratio could acceptably be deduced from the ideal dual cycle. To maintain simplicity, isfc was assumed constant over the range of variables.

The reference values assumed for the parameters of equation 4.13 are given in the following table.

Parameter	Reference value
p_{max}	120 bar
CR	12 : 1
T_o	303 K
A/F_{tr}	24 : 1
$\eta_{vol,tr}$	0.84
η_{comp}	0.72
ψ	1.4
isfc	.198kg/kWh

Equation 4.13 was used to derive Figures 4.1 and 4.2 (a) and (b). Figure 4.1 shows the influence of aftercooling on the imep attainable over a range of compression ratio for a fixed maximum cylinder pressure. The "degree of aftercooling" is the temperature drop achieved by the cooler as a fraction of the temperature rise produced by the compressor. This allows the study of cooling effect as determined by aftercooler effectiveness and coolant temperature.

Figure 4.1 demonstrates the importance of efficient charge cooling since, for a given load, maximum cylinder pressure and air excess, mechanical load is reduced, allowing a higher compression ratio to be used. The less efficient the charge cooling, the higher the compressor pressure ratio must be to maintain manifold density. This greatly influences the feasibility of single-stage turbocharging at this level of output.

Commercially available, mass produced turbocharger compressors have deteriorating performance above a pressure ratio of 3 : 1. The maximum pressure ratio attainable by such machines is about 3.5 : 1 and is accompanied by a narrow surge-to-choke flow range and low efficiency. From Figure 4.1 the trade-off between pressure ratio and the degree of aftercooling may be seen. At an imep of 22 bar, the required pressure ratio increases from 3.0 to 3.4 : 1 as the degree of aftercooling is reduced from 1.0 to 0.8.

Efficient charge cooling allows operation at a reduced compressor pressure ratio, thus a higher compression ratio may be used for a given bmep and mechanical load. By reducing gas temperatures throughout the cycle thermal load is also reduced. The conclusion is that an aftercooling system of the highest performance, consistent with cost considerations, should be specified. The need for a high level of performance and single-stage turbocharging prevents the use of the engine's jacket cooling water as the charge coolant. This may be seen in Figure 4.1, Line 'A'.

The imep attainable for a given maximum cylinder pressure and compression ratio would depend on the explosion ratio and the air-fuel ratio. This is shown in Figures 4.2 (a) and (b) which are derived from equation 4.13. The lower the explosion ratio or air-fuel ratio, the higher the imep may be. However, since this would generally reduce thermal efficiency, it would essentially represent a trade-off between specific output and fuel consumption.

4.3. Computer Modelling

The computer models were constructed by the methods outlined in Chapter 3. Three programs were used : the GEC "filling and emptying" program ; the MEL thermal analogue network solution (86) on the GEC IBM 370 computer, and a closed cycle engine model described in Appendix 5, developed on the Dorman HP9825A micro-computer. The performance model was used because it provides more information than the simpler air-standard cycle calculations. In particular, heat transfer and mass flow may be considered. Heat transfer data from the "filling and emptying" program was used to predict piston and liner temperatures and to assess the need for piston cooling.

Compression Ratio Study

When the bmep of an engine is increased, excluding the use of combustion retard or reduced air-fuel ratio, the maximum cylinder pressure also increases. This is, in part, due to the increase in charge pressure needed to supply the increment of air flow, and in part due to the increased pressure rise from combustion.

Figure 4.3 presents the results of the study of compression ratio. Maximum cylinder pressure and air excess were maintained constant while bmep, charge density and compression ratio were varied over a wide range. Heat flux was calculated using the assumption that the waterside heat transfer conditions remained constant. Although this is not strictly true, because of boiling and non-boiling regimes, it introduces only a small error since gas-side conditions dominate. The relationship used for the calculation of mean heat flux is given below.

$$\text{Mean heat flux} = \frac{T_g - T_c}{\frac{1}{h_g} + \frac{x}{k} + \frac{1}{h_c}}$$

where T_g = mean gas temperature

T_c = cooling water bulk temperature

h_g = mean gas-metal heat transfer coefficient

h_c = mean waterside heat transfer coefficient

k, x = hypothetical mean wall conductivity and thickness taking account of head, liner and piston.

Alcock (45) states that the limiting thermal load of an engine is determined by the heat flux at local "hot spots". This analysis, however, relies on the mean heat flux to indicate changes in thermal load from one operating point to another.

In Figure 4.3, the mean gas temperature in the cylinder is substantially constant over the range considered. This reinforces the view that charge temperature and air-fuel ratio are the determining factors at fixed speed. The calculated mean heat transfer coefficient rises with output, but is less than doubled whilst bmep increases by a factor of four. This indicates that when compression ratio is lowered to allow an increase of power, thermal load also increases, but less than in direct proportion. Thus, engines of this sort will require less cooling capacity per unit of output.

Mean turbine inlet temperature is an important consideration in turbocharged diesel engine design. The following equation gives the work available to the turbine, which is proportional to inlet temperature.

$$W_t = m C_p T \left[\left(\frac{p}{p_o} \right)^{\frac{\gamma - 1}{\gamma}} - 1 \right]$$

For a given work requirement, inlet temperature could be traded for pressure ratio, but this will adversely affect pumping work and scavenging. Therefore, within the limits of material strength, the exhaust temperature should be as high as possible. At present, turbine inlet temperature is constrained to about 950K for alloy steel rotors under continuous duty. From Figure 4.3, the predicted trend is for the mean exhaust temperature to slowly increase as load is increased and compression ratio reduced. This occurs because the effective expansion from combustion is reduced by the increase of expansion through the turbine. The absolute value of the mean exhaust temperature cannot be reliably predicted because of its dependence on combustion duration and timing. The rate of change of mean exhaust temperature assumes constant combustion characteristics, which is questionable in view of the relationship between compression ratio and ignition delay.

Elementary thermodynamics shows that indicated thermal efficiency decreases as compression ratio is decreased, at least for ideal air-standard cycles. The same is largely true for actual engines, and this explains the inferior efficiency of spark ignition engines. Therefore, reducing the compression ratio of a diesel engine may be seen as a retrograde step as far as efficiency is concerned. However, when compression ratio is reduced to accommodate a higher power output, the losses are partly offset by an increase of mechanical efficiency. Figure 4.3 shows predicted and indicated brake thermal efficiencies. The brake thermal efficiency curve is flat over most of the range but reduces increasingly rapidly below 11 : 1. Over the area of interest, 11 : 1 to 15 : 1, the variation is about 1 per cent of thermal efficiency.

Air-fuel ratio study

The chemically correct, or stoichiometric air-fuel ratio for light distillate diesel fuel is about 14.5 : 1. In the diesel engine, the fuel and air have to mix rapidly for efficient and clean combustion, unlike the spark ignition engine which has a mixture prepared throughout induction and compression. Perfect mixing can only be approached in the diesel engine because of "dead" pockets of charge around valves, etc., and especially in high speed engines, for lack of time. This mixing problem is overcome by providing an excess of air. Most engines operate with a minimum air-fuel ratio of 24 or 25, some can tolerate 20 to 24, but only a very limited number less than 20 : 1.

Figure 4.4 is the predicted performance over a range of air-fuel ratios, whilst bmep and compression ratio were held constant. As air-fuel ratio is increased, higher cylinder pressures cause an increase of mean heat transfer coefficient. This adverse trend is offset by the decreasing mean gas temperature, and the net result is constant heat flux over the range of air-fuel ratio considered.

This result is supported by the findings of Alcock (45) who found that for the three engines studied, heat flux varied as the 0.75 power of gross fuel consumption, regardless of air-fuel ratio, speed and charge pressure, the effect of compression ratio was not studied.

Kamo (9) has reported that thermal load can be reduced by trading air-fuel ratio for compression ratio at fixed peak cylinder pressure. From Figure 4.4, this would appear to be feasible since, if maximum gas pressure were constrained, the mean heat transfer coefficient would have increased less, if not declined, giving a net decrease of heat flux.

In conclusion, air-fuel ratio alone offers no reduction of thermal load, whilst increasing mechanical load. Air-fuel ratio traded for compression ratio does allow a reduction of thermal load for constant mechanical load. Kamo's suggested increase of air-fuel ratio from 25 to 30-32 would require a reduction of approximately 2 units of compression ratio. For a given fuel input, the increase in boost pressure required is more than directly proportional to the change in air-fuel ratio because of charge heating effects. The use of 30 : 1 air-fuel ratio would probably require a two-stage turbocharging system.

Valve timing study

The exhaust valve opening event (EVO) is an important consideration in engine design because of its influence on thermal efficiency. The problem may be reduced to balancing the loss of expansion work due to early opening, with the increase of pumping work resulting from late opening. Janota (74) had argued the desirability of timing the EVO so that the blow-down pulse peak occurs at, or near to the bdc after the power stroke.

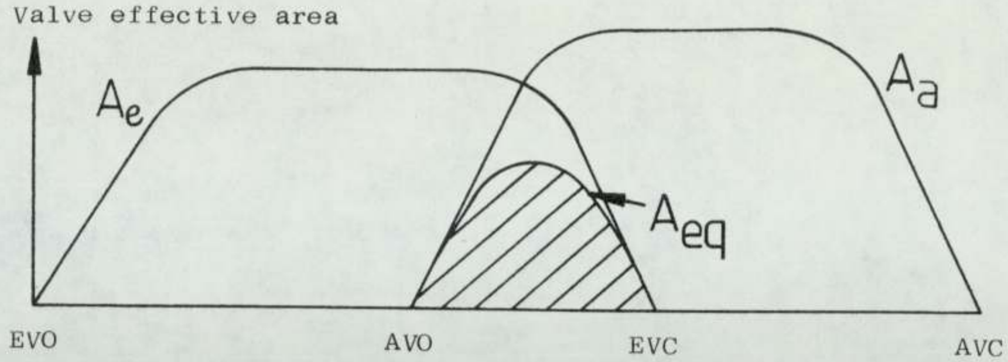
Figures 4.5 and 4.6 summarise the results of a "filling and emptying" study of EVO. Figure 4.5 shows the predicted gain in thermal efficiency by advancing the EVO from the "LE" standard timing of 44^{o} before bdc. The optimum occurs at 80^{o} bbdc which is some 30 to 40^{o} earlier than conventional engine practice. This is due to the low expansion ratio and higher charge mass, whilst exhaust valve size remains unchanged. Low expansion ratio implies a high pressure towards EVO and to vent this optimally, the valve must open earlier.

This surprisingly early EVO was also found to be optimum for the AVCR 1360-2 (6). This engine, which used a variable compression ratio (9 : 1 to 16 : 1), was found by experiment to operate most efficiently at an EVO of 75^{o} bbdc. The authors suggested that earlier completion of blow-down and reduced pumping work gave improved efficiency over the original timing of 40^{o} bbdc.

Figure 4.6 contrasts the transient pressure in the cylinder and exhaust manifold predicted by the model for EVO's of 44 and 76^{o} bbdc.

The gas exchange processes of a four-stroke engine occupy a complete engine revolution. An efficiently turbocharged engine will develop a pressure differential between inlet and exhaust manifolds which the designer can use to good advantage. If it can be arranged for inlet and exhaust valves to be open simultaneously, thus linking these manifolds, a flow of fresh charge will be created, scavenging the clearance volume of spent products from the previous cycle.

This action inceases the fresh charge trapped by the cylinder, thus making better use of the inlet manifold conditions. The through-flow of charge may have a cooling effect, especially for the exhaust valve (6) (67) (80), but this is questionable since overlap denies the valve its normal heat sink by keeping it off its seat. The scavenging effect depends largely on two factors : the equivalent flow area and the nominal pressure differential across the cylinder. The equivalent area is a hypothetical flow passage which connects inlet and exhaust manifolds throughout the cycle, and allows the same flow rate as the valves during the overlap period. This concept is useful for comparing the scavenging potential of systems with different overlap periods and valve velocities. It is described fully by Zinner (3), and calculations for this exercise are given in Appendix 1 . Essentially, the scavenge flow area will depend on the period of overlap and the velocity of the valves during overlap, as shown below.



The nominal pressure differential from inlet to exhaust manifold is determined by the efficiency of the turbocharger and by the temperature rise across the engine. Valve overlap will, however, necessitate a reduction of pressure differential for power balance, since it reduces the exhaust temperature.

Figure 4.7 gives a prediction of scavenging for a range of valve overlap periods. The significant trend is the rapid increase of scavenge flow at overlap periods in excess of 70degCA. This is, in part, due to the near square law relationship between equivalent area and overlap period. The program assumes perfect mixing of products and fresh charge during scavenging. Trapped volumetric efficiency rises significantly between 70 and 100° of valve overlap, as the scavenging of products is increasingly effective, but beyond this there is little further increase, indicating almost complete scavenge. Ryti (80) predicted a very similar volumetric efficiency trend with complete scavenging at only 80° of overlap. The difference is probably due to valve velocity considerations.

Figure 4.8 shows the predicted transient cylinder conditions during scavenging. The rapid drop in temperature of the cylinder contents is caused by the inrush of cool charge which displaces and mixes with the products of the previous cycle.

When studying valve overlap or exhaust valve performance in general, it is important to recognise the limitations of the "filling and emptying" model. In particular, the exhaust manifold is treated as a uniform volume. In reality, the exhaust manifold is a pipe with a length many times greater than its diameter along which primary and reflected pressure waves travel with finite velocity.

<u>Heat release timing</u>

The objective of this study was to examine the relationship between the timing of the effective heat release and thermal load, mechanical load and efficiency. The development engineer frequently makes trade-offs in this area when optimising engine performance, particularly between peak cylinder pressure, efficiency, turbine inlet temperature and exhaust smoke. The program is of only limited usefulness for this problem because exhaust smoke cannot be predicted, but it does make heat transfer, and hence thermal load parameters available. Although effective heat release rates are known to vary with timing via ignition delay, the assumption of a fixed heat release diagram is thought to introduce only a small error over the range considered.

Figure 4.9 presents the results of the study. The explosion ratio increases with timing advance. The compression pressure is constant, but the maximum cylinder pressure increases almost linearly at 3.7 bar/deg. Brake thermal efficiency reaches a maximum at the most advanced timing, although the curve is relatively flat over the range considered. This confirms the conventional wisdom, which is to trade efficiency and mechanical load by timing retard. Dorman turbocharged engines typically operate at the right-hand side of Figure 4.9 to maximise the power output from a given structure. The heat flux, which indicates thermal load, decreases slightly as timing is retarded, thus mechanical and thermal considerations are compatible in this respect. The exhaust temperature, and probably the exhaust valve temperature, increase with timing retard, giving an increase of turbine energy.

Although mechanical and thermal load are reduced by timing retard with some loss of thermal efficiency, the turbine inlet temperature and smoke trends will normally provide a limiting acceptable retard, therefore the optimum will be a compromise.

Component temperature prediction

Both thermal analogue network and finite element methods have been used to calculate component temperatures during the project. The construction and accuracy of such models are discussed in Chapter 3.

Throughout the discussion of computer modelling, it has been stressed that the predicted trends, and not the absolute values should be considered because of the inherent difficulties of combustion simulation. Whilst this is true, the uncertainty of predicted absolute values could be accounted for by the use of probability concepts. This would involve defining an upper and lower performance between which the actual engine has an acceptably high probability of falling. This would be especially useful for temperature and stress calculations, since we would normally wish to proceed with an engine design defining in absolute terms the cooling and structural detail. This approach was not adopted here, but experience now indicates that the use of a performance band is fundamentally more sound than a single point solution.

Figure 4.10 gives the piston temperature field predicted by a thermal analogue network, with and without undercrown cooling. The gas-side heat transfer conditions were predicted by the filling and emptying model, which used the Woschni relationship (49). The undercrown, outer diameter and ring heat transfer conditions are due to Woolley (44) who used a Dorman "LB" engine for his research. The effectiveness of undercrown cooling is greatest at the centre of the piston beneath the bowl, reducing temperatures there by about 60K. In the more critical areas around the bowl upper edge and above the top ring, the cooling is less effective, amounting to a reduction of approximately 25K.

It has been reported that undercrown cooling increases the thermal stress in the unsupported crown area. This region may be approximated to a restrained beam with lateral heat flow. In that case, the maximum thermal stress is longitudinal and directly proportional to the lateral temperature gradient. On that basis, the radial thermal stress in the piston will be about 50 per cent greater when oil-cooled, due to the greater axial temperature gradient.

Considering the predicted absolute temperatures, the uncooled piston is marginal in all respects. The top ring area would probably suffer poor lubrication, the load-bearing pin-boss area exceeds the recommended 180°C limitation, and the bowl edge is close to the allowable maximum of 350°C. Although predictions of this sort are not reliable enough to allow detailed conclusions to be drawn, the value of undercrown oil-cooling in a marginal situation may be seen.

Figure 4.11 gives the result of a cylinder liner analysis at the target output of 21 bar bmep at 1500 rev/min. The interesting feature is the sharp rise in temperature towards the top. This is caused by high heat flux from the hottest part of the cycle, and relatively poor local cooling. This emphasises the need to allow the coolant as near to the liner top as is practicable. This temperature profile will cause little trouble to the lubricant film, especially at the top ring position where the predicted surface temperature is 175°C.

Piston and liner temperatures were studied under the same operating conditions, and these are given in the following table.

bmep	21 bar
speed	1500 rev/min
air-fuel ratio	28 : 1
compression ratio	11.7 : 1
max. pressure	138 bar
<u>Gas</u>	
hm	670 W/m^2K
Tm	707°C
<u>Oil splash</u>	
hm	409 W/m^2K
Tm	80°C
<u>Oil jet</u>	
hm	3680 W/m^2K
Tm	80°C
<u>Water</u>	
hm	3400 W/m^2K
Tm	60°C

4.4. Research Engine Specification

Valve timing

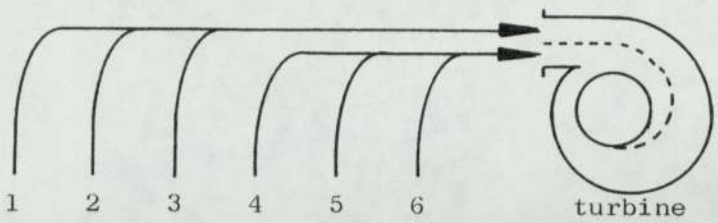
The valve events chosen were based largely on the results of the "filling and emptying" study. The valve events of the cam shaft specified for the research engine are given below in Table 4.1.

Table 4.1.

	Valve timing	[deg atdc]
Event	Standard "LE"	High output
EVO	134	105
AVO	346	296
(Overlap)	(22.5)	(127)
EVC	369	425
AVC	581	584

The exhaust valve opening event selected was some 30 degrees earlier than the standard timing for the engine. It is near to the optimum predicted, and should allow the blow-down pulse to leave the cylinder efficiently. The valve overlap period chosen should be adequate to thoroughly scavenge the clearance volume of combustion products. It is thought that 11 to 14 per cent of the total air-flow will pass through the cylinder during valve overlap.

The conventional exhaust arrangement for an in-line, six-cylinder engine is two 3-cylinder groups. These empty into separate manifolds and then into a double-entry turbine, as shown below.



This is intended to prevent interference between consecutively firing cylinders. The early EVO and wide overlap selected for this engine makes interference possible, even with this exhaust arrangement, as shown in Figure 4.12. Figure 4.13 gives the predicted transient pressure conditions during valve overlap, in the cylinder and manifolds. The pressure rise in the exhaust manifold causes little loss of scavenging or induction since the exhaust valve is almost closed. This is, however, thought to be the longest exhaust period that could be employed without serious interference.

Engine and turbocharger

Having defined the valve timing, the interaction between compressor, charge cooler, engine and turbine, may be examined in greater detail. The conclusion of the design study was that the aftercooler performance should be maximised within size and cost constraints. This would minimise the required charge pressure, increasing the feasibility of a single-stage turbocharging system, and reduce both thermal and mechanical load. The need for high performance rules out the use of the engine's jacket cooling water as a charge coolant, as confirmed by figure 4.1. Although many applications employ air-to-air charge cooling, the medium used experimentally matters little. For convenience, a water-to-air design was favoured. The unit specified has a high effectiveness (approximately 0.9) and the capacity to handle the air flow with minimal pressure loss. The aftercooler was made by, and selected in consultation with DAW Limited.

The air-fuel ratio of turbocharged engines decreases as load is increased, thus reaching a minimum at full-load. Dorman engines tend to operate with a full-load, air-fuel ratio in the range 22 to 25 : 1, at which point smoke and turbine inlet temperature become limiting factors. The "LE" combustion system is of the classical toroidal open chamber type. This relies on the orderly rotation of the charge, generated during induction, to effect mixing of the fuel and air throughout combustion. A disadvantage of high valve overlap is the need to recess the piston crown to accommodate the valve heads at top dead centre. These recesses are thought to break up swirl, particularly when the piston is near the top of its stroke, and to create pockets of charge inaccessible to the fuel sprays. These effects are likely to lower the smoke threshold so that a higher air-fuel ratio may be needed for acceptable smoke emission. Decisions relating to combustion and smoke emission are rather arbitrary at the design stage, but such decisions have to be made if only to be modified during development. A trapped air-fuel ratio of 28 : 1 was chosen, and turbocharger match and engine loading calculations were based thereupon.

The characteristics of fuel injection equipment cannot be translated into an effective heat release rate diagram because of the complexity of the processes involved. To overcome the problem and allow the compression ratio to be estimated, the concept of explosion ratio was used. This is simply the ratio of peak to compression pressure, and may be used to link peak pressure to charge pressure via compression ratio. Dorman engines typically operate with explosion ratios in the range 1.35 to 1.50.

Modern turbocharged diesel engines can be designed with high mechanical strength, hence high allowable cylinder pressure, without undue economic penalty. Advances in bearing design and materials, cylinder head sealing and analytical stress methods mean that cylinder pressures of 2000 lbf/in^2 (138 bar) are readily achievable. Although the maximum cylinder pressure of the "LE" engine is limited, for commercial application, to 1750 lbf/in^2 (121 bar), the design limit for the high output exercise was raised to 2000 lbf/in^2 without modification.

The compressor operating point is determined from the air flow requirement, degree of aftercooling, volumetric efficiency, speed and swept volume. Since a wide valve overlap period is being used, the total air flow will be the sum of the trapped and through-flows. This makes air flow more difficult to predict. Partly because of the dependence on the available pressure differential across the cylinder, which is related to turbocharger efficiency, but also because of the difficulty of predicting the extent of cylinder scavenging. Figure 4.14 shows the predicted full-load operating point for an overall turbocharger efficiency of 53.8 per cent, superimposed on the selected compressor map. This compressor match is not ideal, but is an acceptable compromise between surge margin and isentropic efficiency at high pressure ratio. If the turbocharger efficiency is less than expected, the surge margin of only 6 per cent will be further reduced. The 72 per cent efficiency island extends to a pressure ratio of 3.4, beyond which there is a rapid deterioration. This confirms the need for a high degree of charge cooling at this rating - unless a two-stage system is acceptable.

Having defined the compressor operating point, and hence, power requirement, it is possible to consider the turbine specification. The turbine effective area controls the upstream pressure head and thus, the gas velocity at entry to the rotor, thereby affording control over the energy available to the turbocharger. The engine-turbocharger match is optimised experimentally, normally by using a number of turbine housings covering a range of effective area. The turbocharger was supplied by Garrett Airesearch Limited, and under their advice, two turbine housings were specified. The final turbocharger specification was :

Frame size	TV 71
Compressor	A-8
Turbine housings	1.41 and 1.08 A/R "E" trim

Table 4.2 outlines the salient design features and predicted performance, resulting from valve timing, turbocharger and charge cooler specification. The final design was modelled to provide a basis for comparison between the "filling and emptying" method, as applied to this engine, and experimental data.

Fuel injection equipment

Dorman's main supplier of fuel injection equipment is, and has been for many years, CAV Limited, part of the Lucas Group. CAV have recently introduced a new fuel injection pump, the Maximec, capable of high rates of injection, with adequate capacity for this application. Consultation with their engineers revealed that the pump was

Table 4.2. : Salient design features and predicted performance of the research engine

Parameter	Value
bmep	21.1 bar
speed	1500 rev/min
brake power	303 kW (406 bhp)
compression ratio	11.7 : 1
maximum cylinder pressure	135 bar
explosion ratio	1.56
air-fuel ratio	27.6 : 1 (trapped)
valve overlap	127 degrees
volumetric efficiency	99% (trapped)
scavenge ratio	1.144
air mass flow rate	0.562 kg/s
compressor pressure ratio	3.14
aftercooler effectiveness	0.90
turbine expansion ratio	2.80
mean cylinder temperature	969 K
mean turbine inlet temperature	792 K
indicated thermal efficiency	41.9%
mechanical efficiency	91.3%
overall turbocharger efficiency	53.8%

well-suited to the task and that they were amenable to producing a one-off for research purposes. In view of the performance capability and anticipated long production life ahead, this pump was selected for the research engine.

The quality and duration of the fuel spray which emerges from an injector nozzle under given cylinder conditions, cannot be accurately predicted from a physical description of the fuel injection system. In any case, the performance of a system could not be guaranteed on this alone, since it depends on the marriage of fuel injection equipment, combustion chamber geometry and air motion. Apart from the consideration of flow capacity, selecting fuel injection components is largely the development engineer's "art". For this reason, the advice of those experienced in such matters was sought, particularly the engineers of CAV Limited who are clearly specialists in this field.

The essential features of the "jerk" pump system are the plunger, barrel, delivery valve, fuel pipe and injector nozzle. At the design stage, the equipment is "sized" and a range of nozzle configurations, thought most likely to succeed, is specified. During development, plunger diameter, nozzles, delivery valves, etc. are varied intuitively, and by experiment the system is optimised.

Figure 4.15 gives the lift-velocity characteristic of the fuel pump cam specified for the research engine. The inlet port is closed by the plunger after 4mm of lift, at which point a pressure wave is set up which travels to the injector and delivers the fuel. Statically, the plunger will deliver the required 320mm^3 of fuel at a mean rate of $41mm^3$/degree of cam shaft rotation. This translates to a spray duration of almost 16 degrees of crankshaft rotation. However, the normal assumption that liquids are incompressible, is not valid over the pressure range (0-1100 bar) of fuel injection, thus, large deviation from this nominal rate can occur. Table 4.3 gives the specification of the fuel injection equipment selected for the research engine.

Table 4.3 : Specifications of the fuel injection equipment

Fuel pump	CAV Maximec
Cam form	CDS 162- see Figure 4.15
Plunger lift	12mm
Plunger diameter	12mm
Inlet stroke	4mm
Delivery valve	$125mm^3$ unloading volume
Fuel pipe	8mm O/D x 2.5mm I/D
Pipe lengths	1370mm (all equal)
Fuel injectors	CAV "low spring" type
Nozzle range	4x0.39, 0.40 and 0.42mm diameter
Spray angle	140 degrees
Opening pressure	260atm (263.4 bar)

Cooling systems specification

The objective of an engine cooling system is to maintain the components surrounding the hot working gases at a temperature consistent with strength, distortion and lubrication considerations. The thermal analysis presented in 4.3 showed that the uncooled "LE" piston was likely to be in distress in several respects if operated at the target full-load. The benefits of undercrown oil-jet cooling were estimated and shown to offer some improvement in all areas. Wellworthy Limited, the piston manufacturers, have studied the need for direct piston cooling (35)(36) over a wide range of cylinder size and bmep. They propose a hierarchy of cooling measures. In order of increasing bmep, these are : uncooled, undercrown oil-jet and soluble core "cocktail shaker" designs.

Based on the Wellworthy data, the "LE" piston should be undercrown cooled above a bmep of 12 bar, and require a cooling gallery above 21 bar. At the target output the pistons will require cooling, but only actual test data will determine by which method. Having discussed this question with Wellworthy engineers, it was decided to proceed with the undercrown design initially.

The most important design parameters in oil-jet cooling are the flow rate and velocity of the jet. The flow rate should be high enough to prevent overheating of the oil whilst in contact with the piston. The velocity should be greater than the maximum piston velocity to ensure jet impingement throughout the cycle.

$$\Delta T = \frac{Q}{\dot{V} C_p \rho} = \frac{0{\cdot}35.5{\cdot}9}{V.2.820}$$

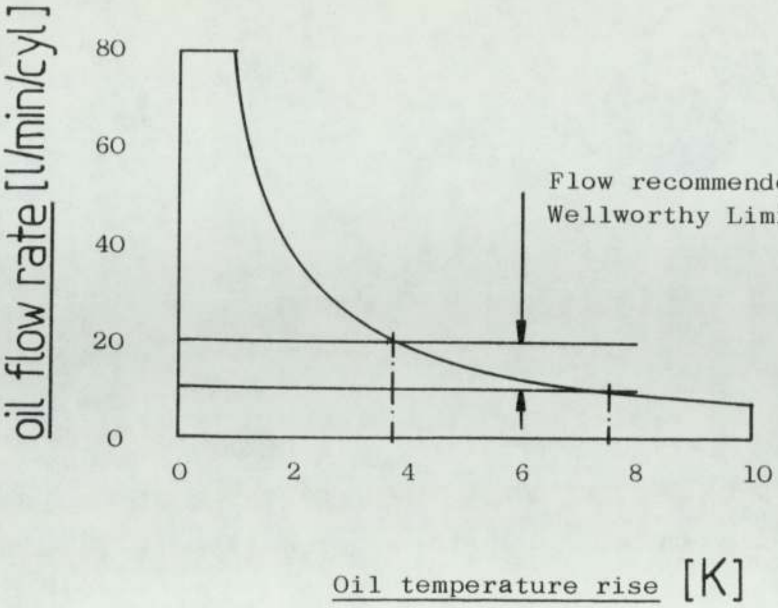
where predicted piston heat flow Q = 5.9kW

predicted fraction of heat to oil = 0.35

specific heat of oil C_p = 2 kJ/kgK

density of oil ρ = 820 kg/m³

which has been used to produce the figure below.



Temperature rise of piston cooling oil plotted against

oil flow rate

The jet diameter was determined by satisfying the requirements for velocity ($>$12.2 m/s) and flow rate (10 to 20 l/min/cyl.)

$$d_{jet} = \sqrt{\left(\frac{V}{\dot{V}}\frac{4}{\pi}\right)}$$

A parallel jet of 5/32" (3.97mm) inside diameter was selected. A gear type oil pump with a nominal swept volume of 91 l/min (15.2/piston) at 1500 rev/min was chosen. During development, the oil delivery could be measured and corrected, if necessary, by changing the oil relief pressure or pump speed. A standard Dorman oil cooler was incorporated, to allow heat exchange from oil to jacket water. The system is shown in Figure 4.16.

The "LE" is a water-cooled design. The heat loss through the chamber and port walls, and the heat collected by the oil is transferred to the engine's closed water system. This includes a heat exchanger which enables the heat to be rejected, either direct to the atmosphere or via a liquid coolant. Experimentally, the choice between water-to-air (radiator) and water-to-water heat exchangers matters very little. For convenience, and to remove doubts about cooling fan power absorption, a water-to-water system was specified.

The most important design considerations are :

1. To ensure adequate flow to prevent bulk boiling.
2. To provide high velocity at local "hot spots" to prevent film boiling and scaling.
3. To prevent steam pockets.

The predicted engine energy balance is given in Table 4.4 and in Figure 4.17. Predicting the energy input and distribution depends largely on the heat release rate assumed, and hence is not accurately predicted ahead of test-bed development.

The cooling equipment for a commercial engine should be well-matched to minimise costs, but since this is less important for an experimental engine, some over-capacity can be tolerated to account for uncertainty. Accordingly, a shell-type heat exchanger, normally used on the Dorman "Q" series, was chosen. This could dissipate up to 250kW.

Table 4.4. : Predicted energy balance of research engine

Work output	303kW	0.375
Aftercooling	81kW	0.100
Friction	29kW	0.036
Heat loss to walls & ports	103kW	0.127
Exhaust	252kW	0.312
Radiation etc.	40kW	0.050
Total	808kW	1.000 unit

Howarth (47) suggests that the water flow through a closed jacket cooling system should be as high as 20gal/bhp.h (122 l/kWh). The requirement to prevent bulk boiling is much less than this, but to produce a lively, high velocity flow through the jacket and cylinder head, such quantities are generally required. The charge cooled variants of the Dorman 6LE and 6Q engines circulate 6.9 and 11.3 gal/bhp.h respectively at their continuous full-loads, at 1500 rev/min. The flow of the "LE" engine was thought to be inadequate for the high rating of the research engine, and measures were taken to increase it. A larger water pump (Dorman 6Q) was specified, the drillings between the crankcase and the cylinder head were increased and generous pipe sizes were chosen.

Starting

The range of compression ratio (10 to 13) to be used will make starting the engine difficult unless aids are provided. In any case, it will be of value to the overall project objective to evaluate startability and the degree of assistance that will be required to give commercially acceptable starting. The aids on the market for diesel engine starting include : ethyl ether fumigation, manifold air and block heaters, and fuel pump controls to optimise injection timing. Provision was made to insert a 3kW mains block heater into an external water pipe. The manifolds were fitted with proprietary ether spray nozzles and fuel burners.

Restricting the passage of exhaust gases, once combustion is initiated, is thought to assist acceleration and reduce white smoke emission by holding back hot gases to heat the next fill. This is readily provided since each test-bed is fitted with a moveable flap in the exhaust pipe. The effectiveness of retarded fuel injection timing for starting can be evaluated by manually resetting for start-up. It is intended to evaluate such aids during development, albeit at normal shop ambient conditions.

Experimental provisions

The foregoing describes a "median" build of the research engine thought most likely to achieve the target output satisfactorily. It is necessary to provide flexibility, both for contingencies and to allow a degree of experimentation at the target load.

The compression ratio may be changed by altering the clearance volume, most of which is contained in the toroidal combustion chamber in the piston crown. With this in mind, pistons giving compression ratios of 10, 11, 11.4, 11.7, and 13 were made available for the experimental programme, each, of course, giving a different cumbustion chamber size and aspect ratio which may give insight into the effect of these factors on performance. Two turbocharger builds are possible by changing the turbine housing to give "high" or "low" boost pressure.

The fuel injection equipment is more flexible. There are three nozzle configurations which are easily changed. Delivery valve retraction volume is equally easy to change over a wide range. Less easy, but feasible, is a change of pump element. The available range is 11, 12 and 13mm diameter.

An outline drawing of the engine, showing the layout of the various systems, is given in Figure 4.18.

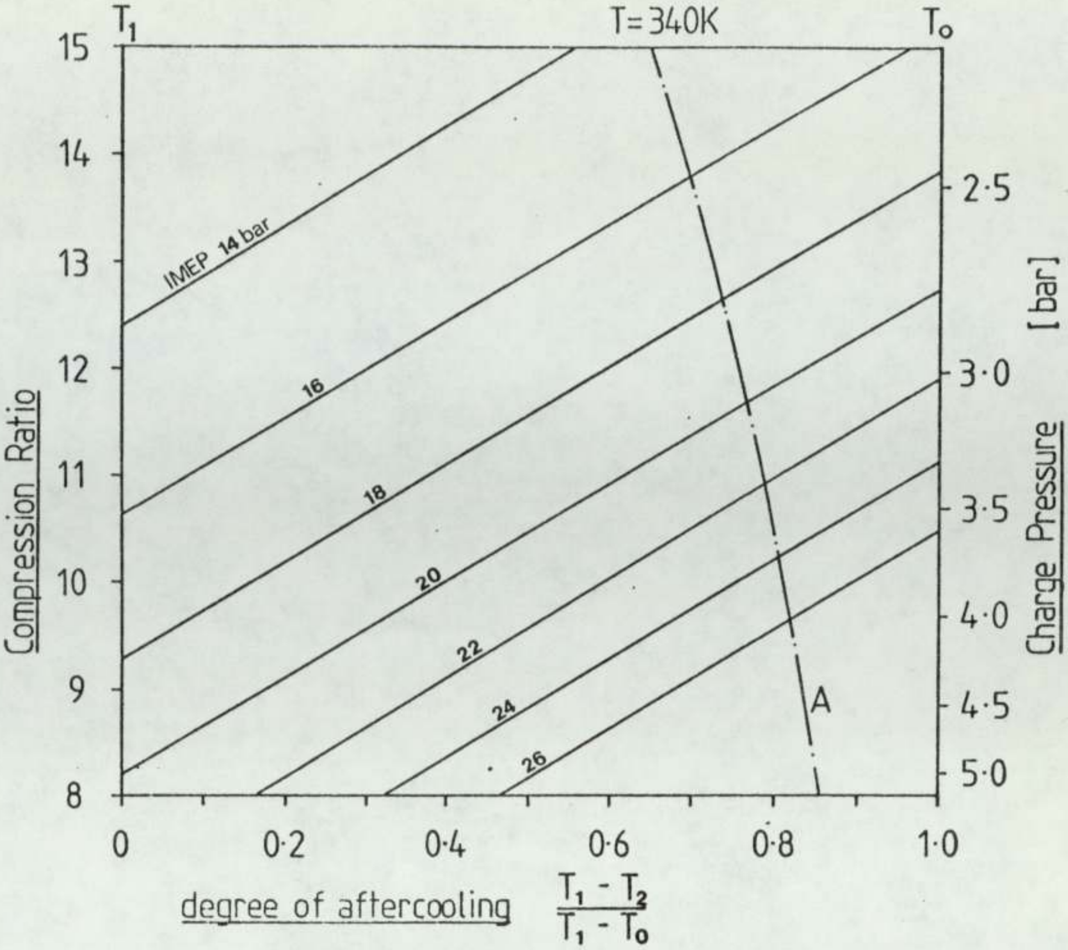


Fig.4.1, Calculated relationship between degree of aftercooling and compression ratio; constant air excess and peak pressure. 'A' is the limiting performance using jacket coolant at 340K.

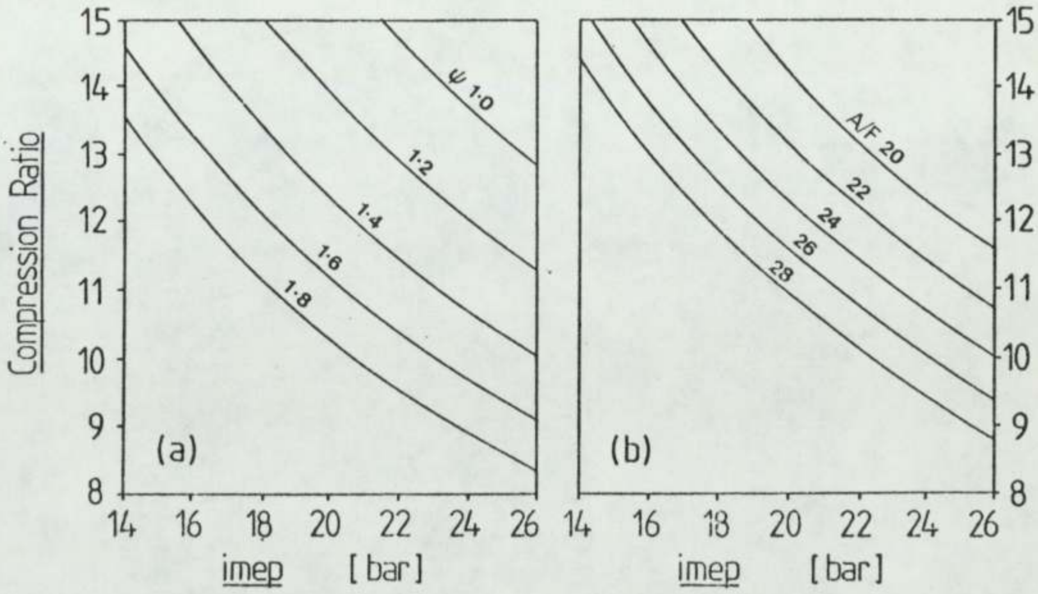


Fig.4.2, The imep attainable from a given compression ratio and maximum cylinder pressure depends on, (a) explosion ratio, and (b) air-fuel ratio.

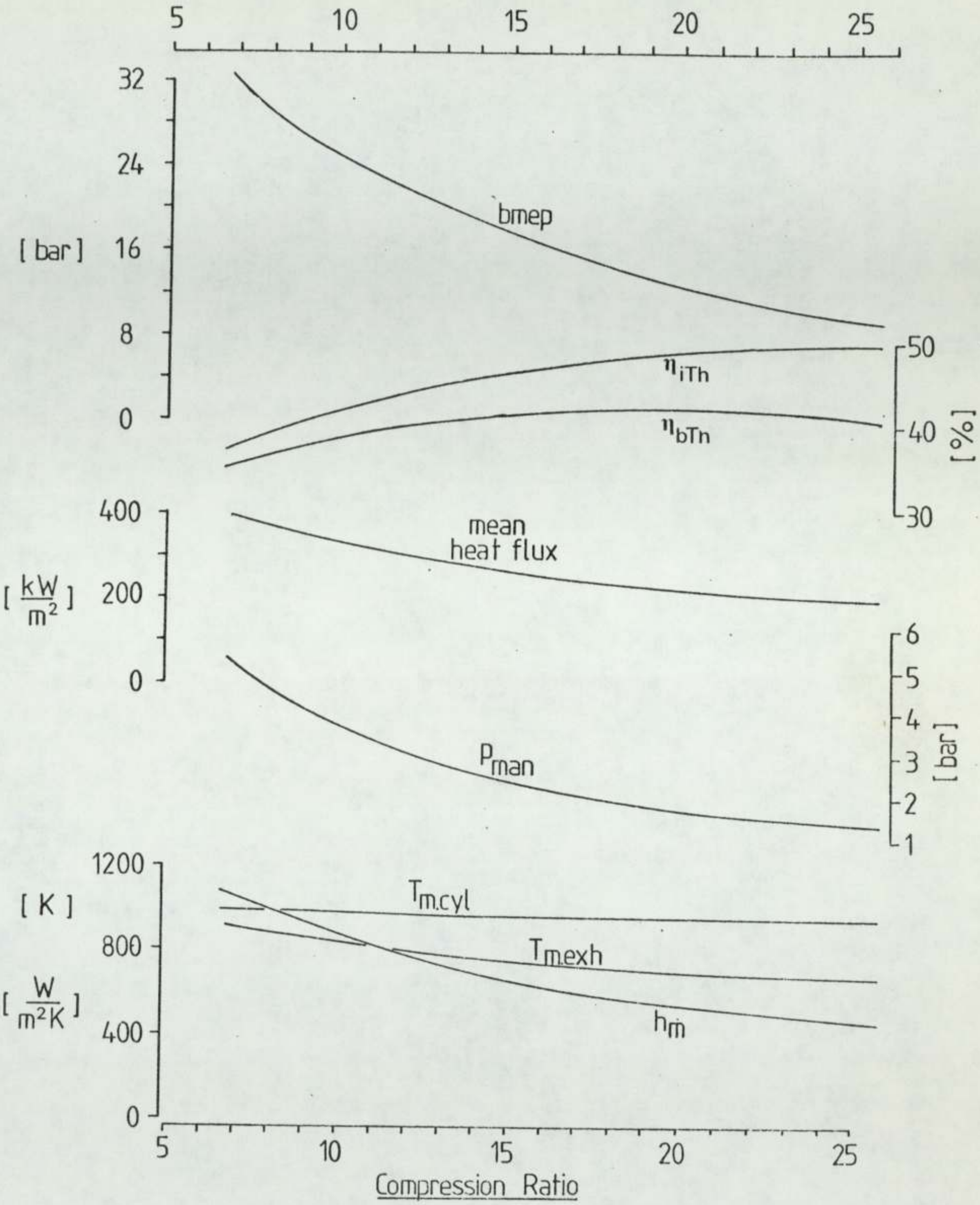


Fig.4.3, Predicted performance trends - compression ratio and bmep varied with peak pressure and air-fuel ratio constant.

P_{max} = 138 bar   A/F = 28:1

heat release function $= 1 - \exp(-6.9[\frac{709-\theta}{75}]^{2.2})$

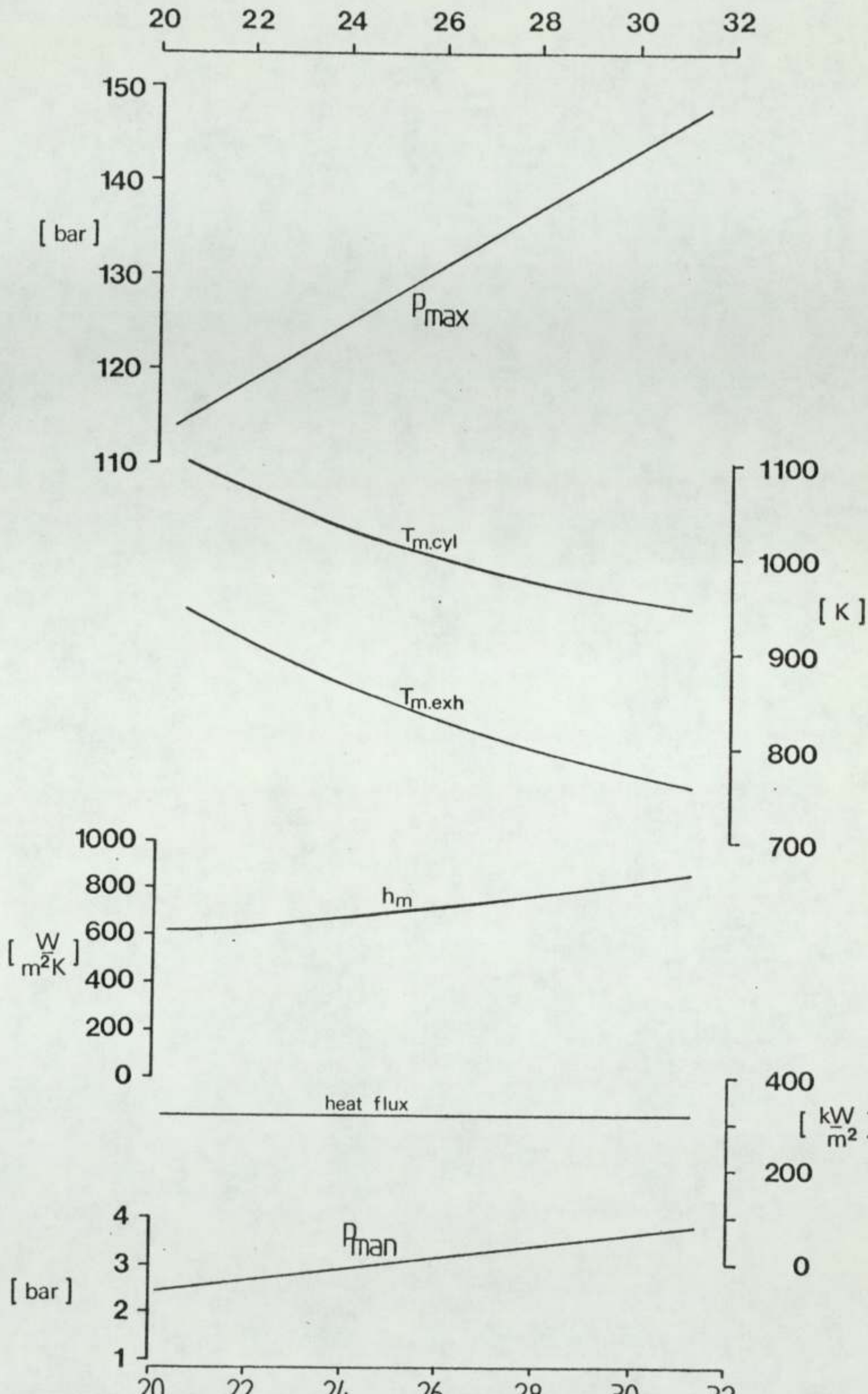


Fig. 4.4, Predicted effect of air-fuel ratio on performance, with constant fuel input and compression ratio.

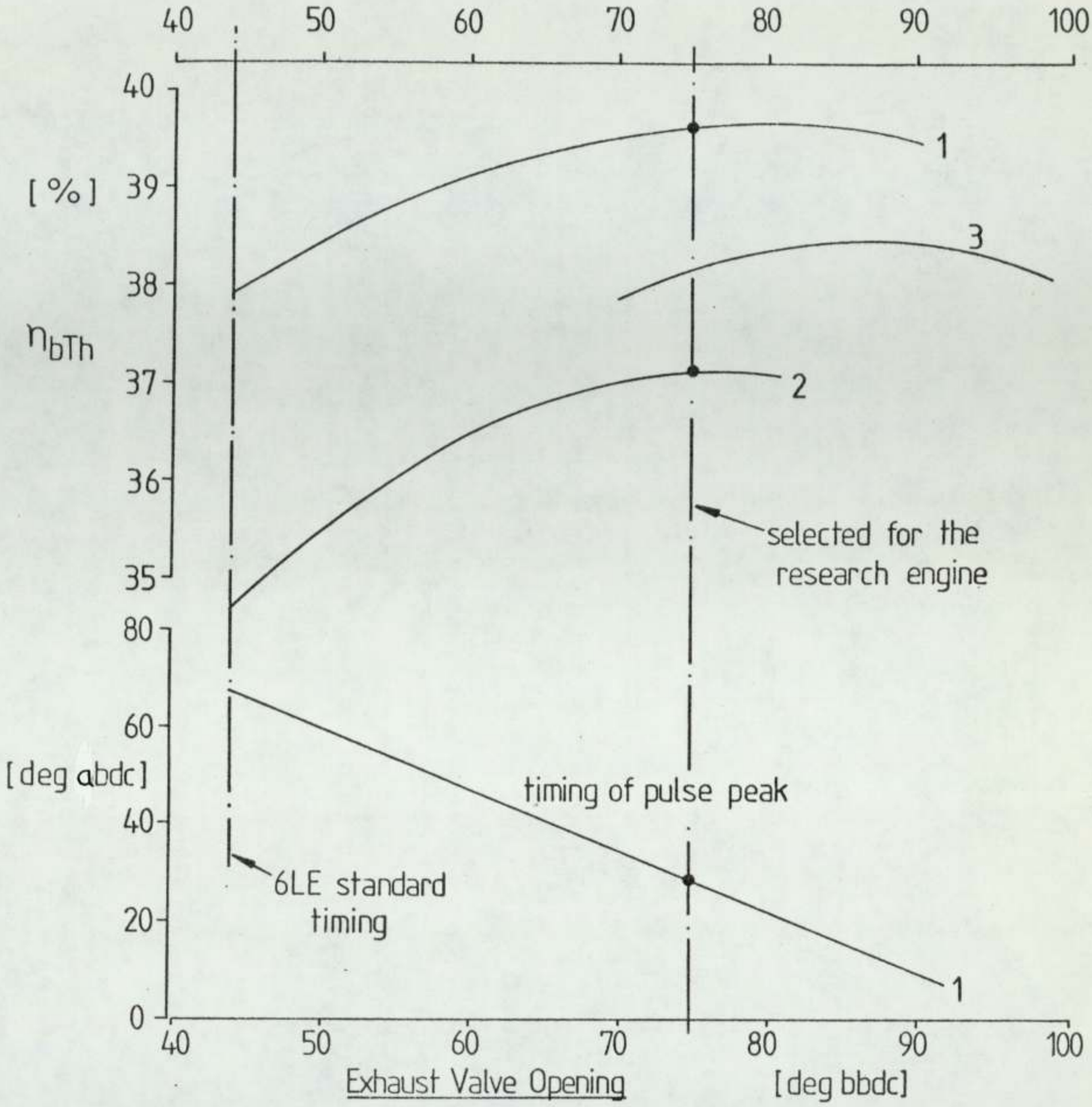
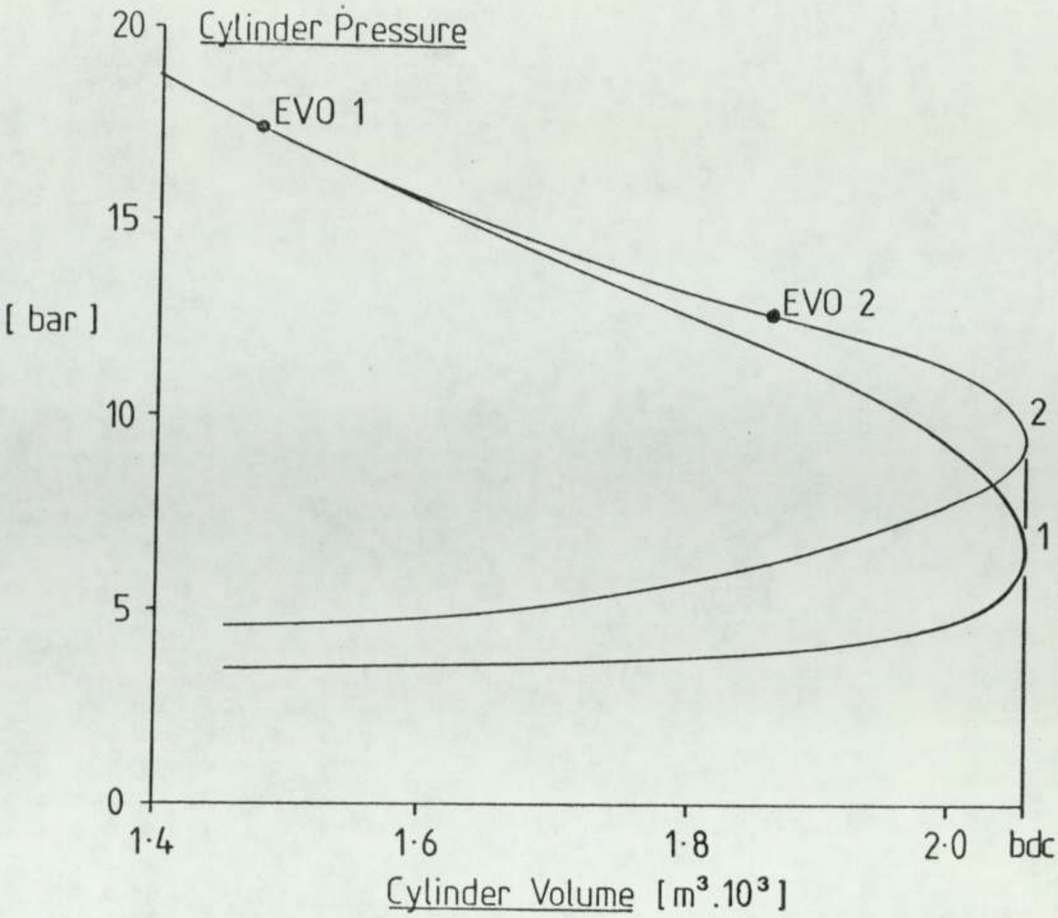


Fig. 4.5, Predicted effect of exhaust valve opening event on brake thermal efficiency and exhaust pulse timing.

1. compression ratio = 11·7, bmep = 21 bar.
2. " " = 11·7, bmep = 10 bar.
3. " " = 9·3, bmep = 26 bar.



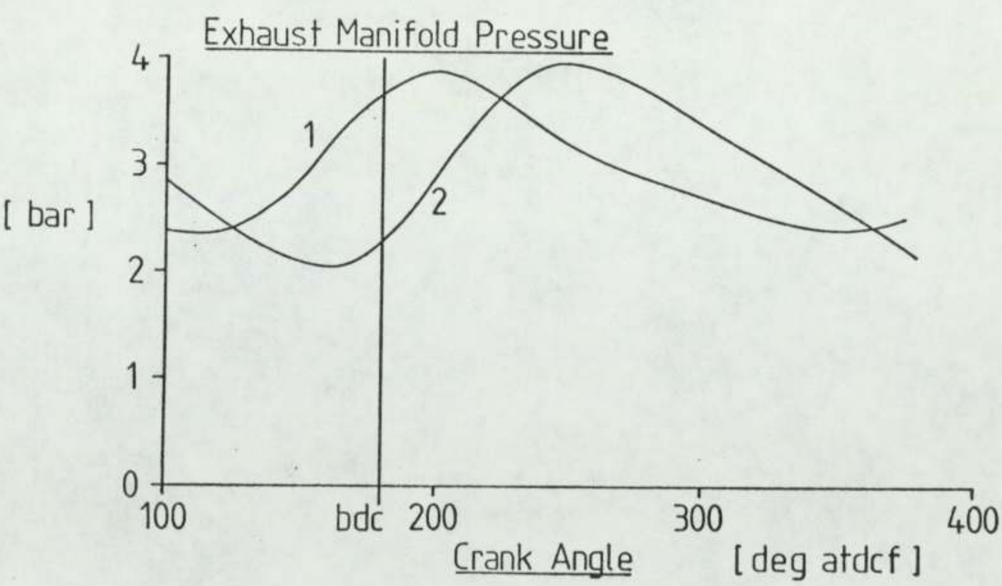


Fig.4.6, Predicted effect of exhaust valve opening event (EVO) on transient pressure at the end of the power stroke and timing of the blowdown pulse.

compression ratio = 11·7

1. EVO = 76 [deg bbdc]
2. EVO = 44 " "

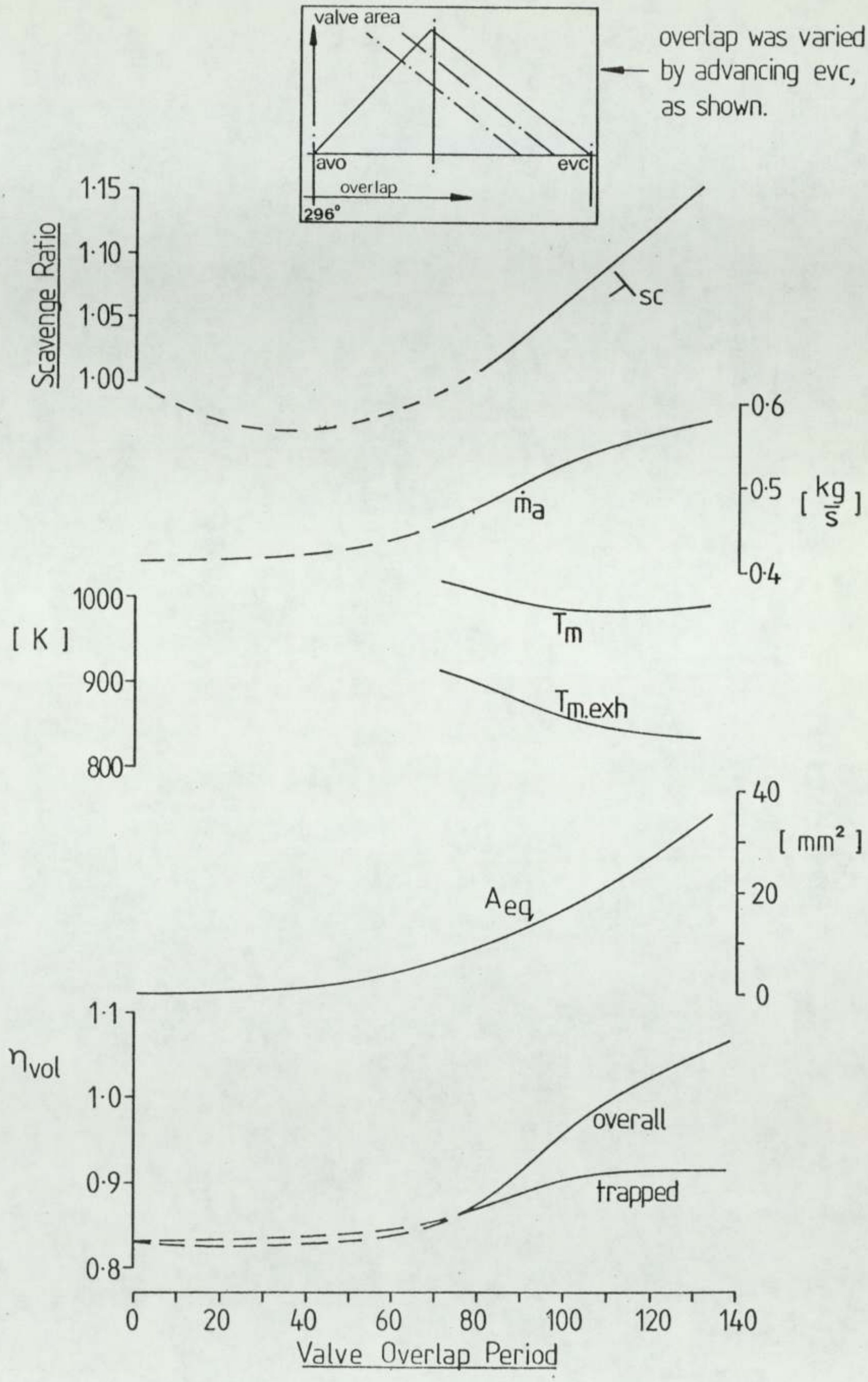


Fig.4.7, 'Filling and emptying' calculation of breathing and scavenging, with variation of valve overlap period.

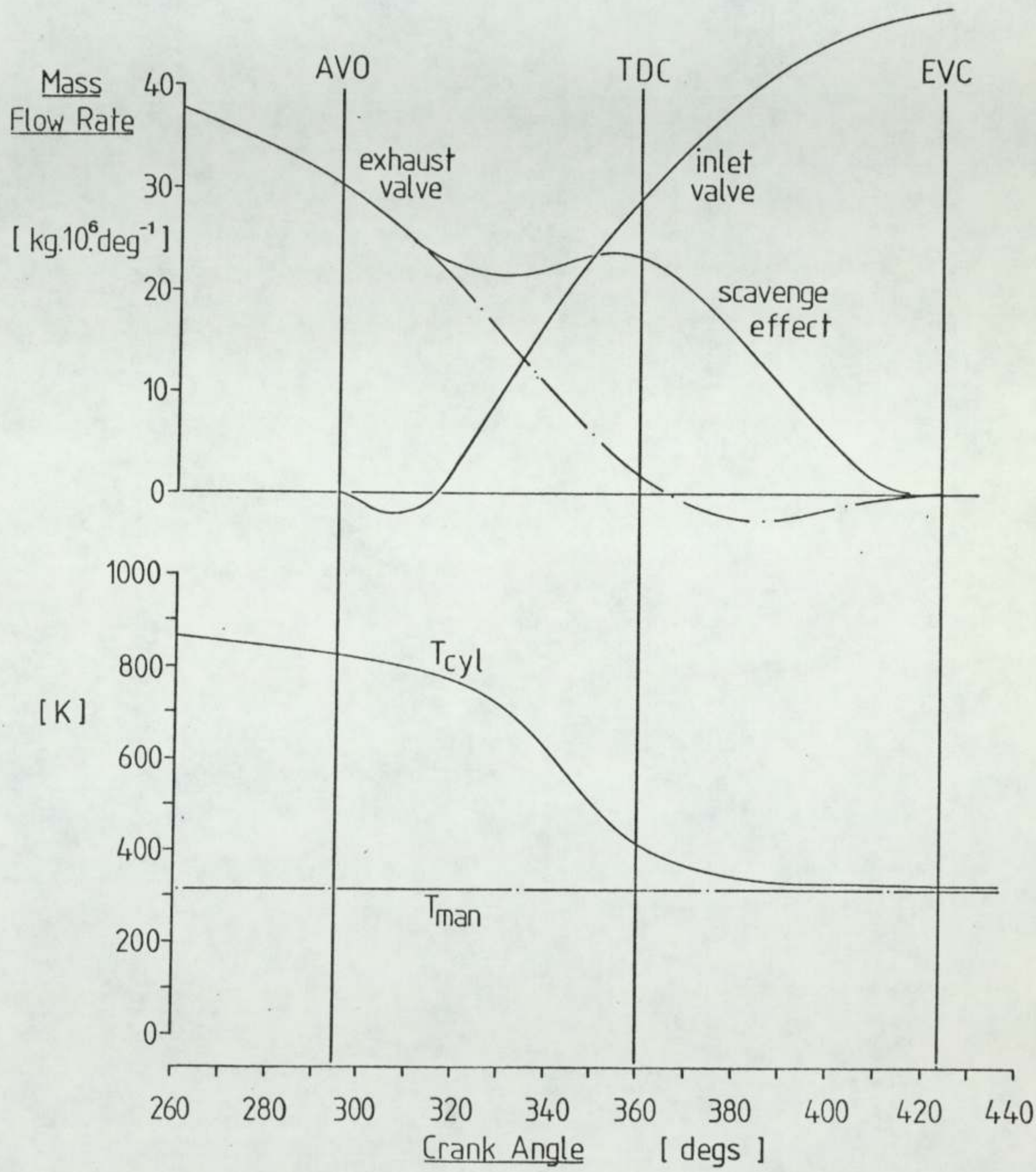


Fig.4.8, Transient mass flow and cylinder temperature during the valve overlap period, predicted by the 'filling and emptying' model.

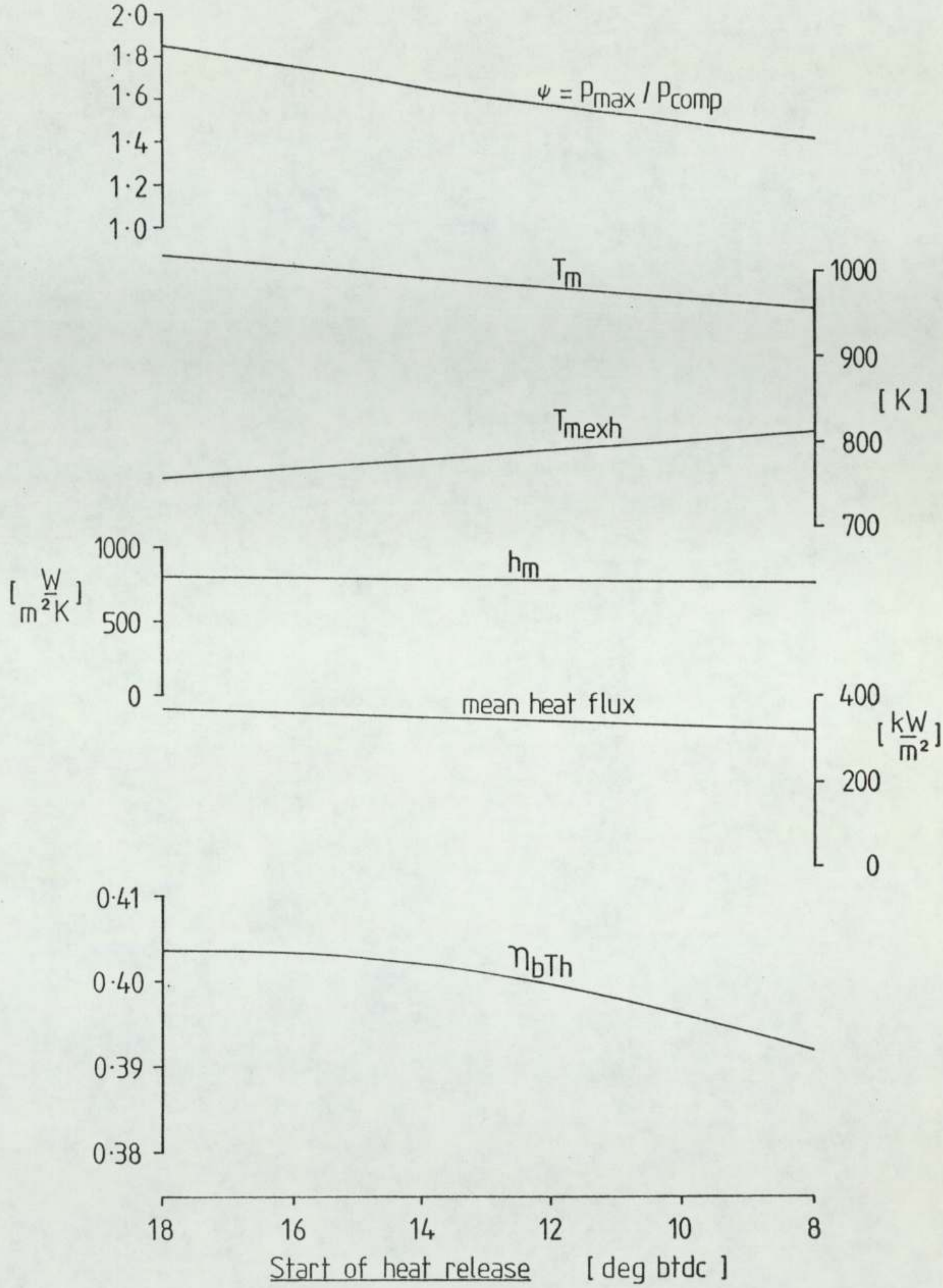


Fig.4.9 , Predicted effect of start of heat release on engine performance.

Wiebe heat release function $= 1 - \exp(-6\cdot9\,[\frac{\varphi_1 - \varphi}{75}]^{2\cdot2})$

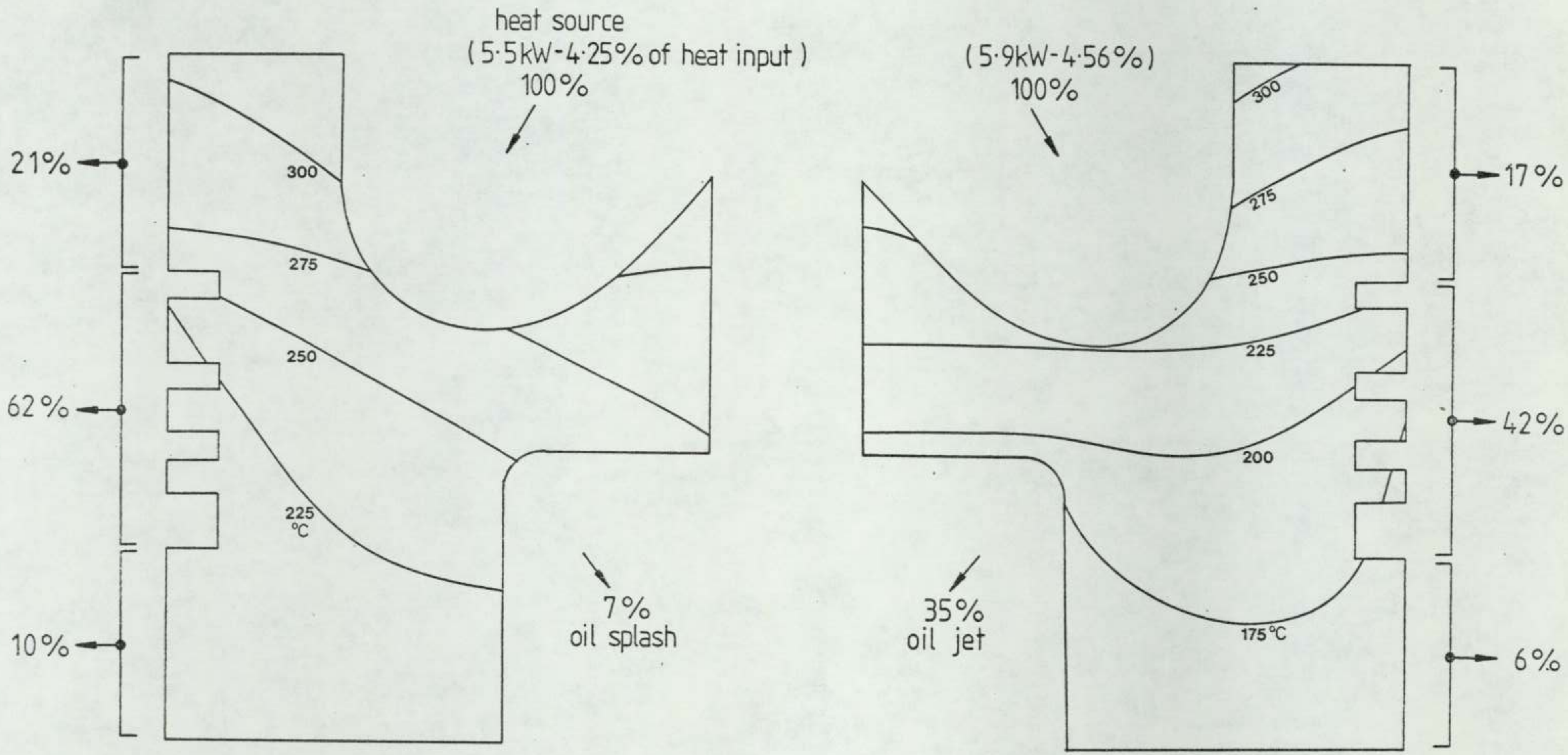


Fig.4.10, Temperature fields, predicted by thermal analogue network method, for uncooled and undercrown oil jet cooled piston at a bmep of 21 bar.

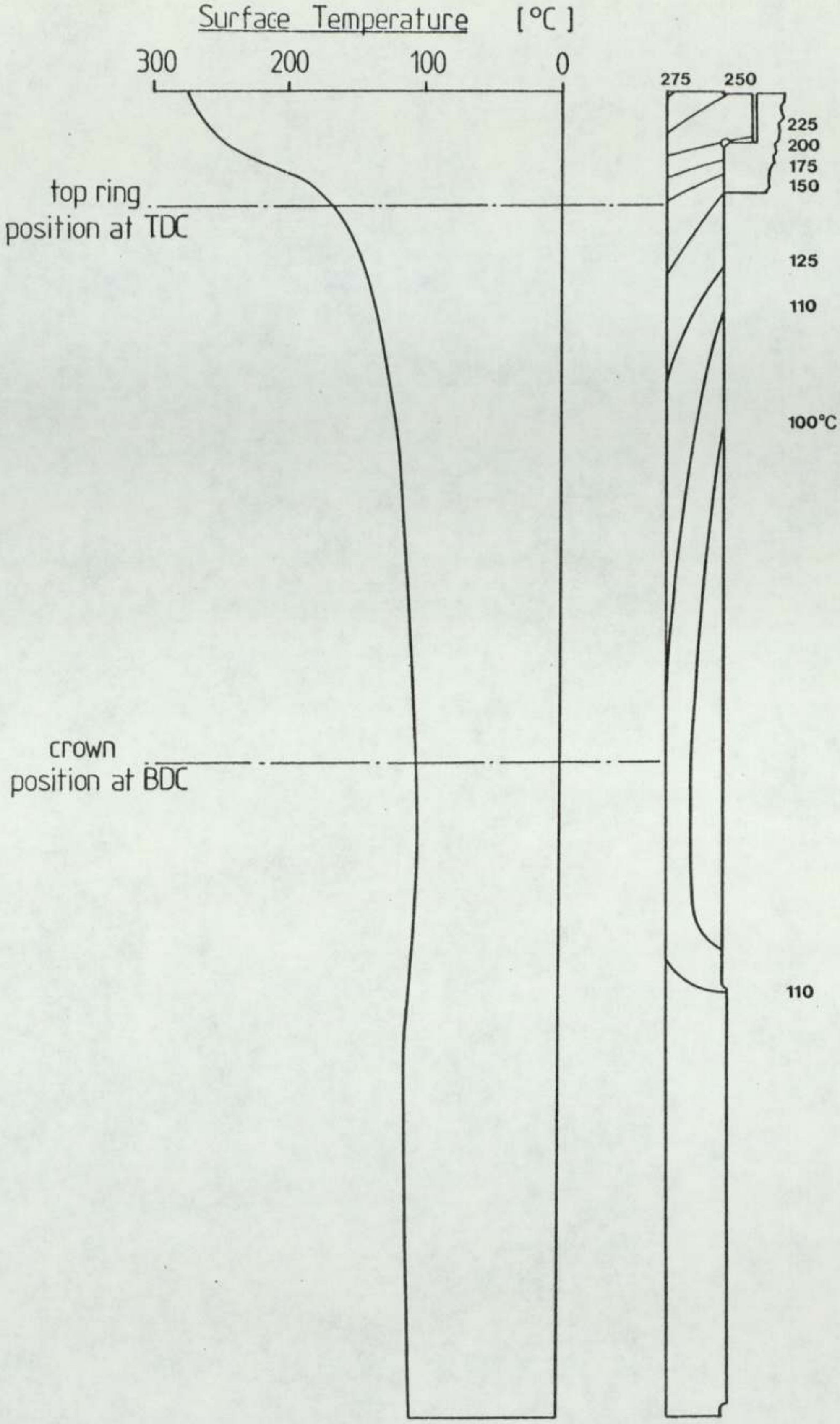


Fig. 4.11 , Cylinder liner temperature predicted by the thermal analogue network method, at a bmep of 21 bar.

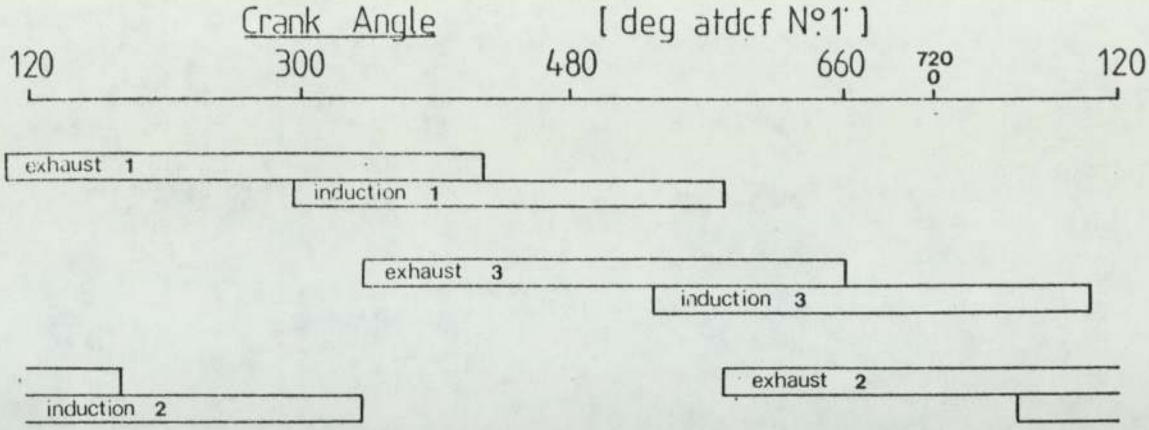


Fig. 4.12 , Valve events of cylinders emptying into the front manifold.

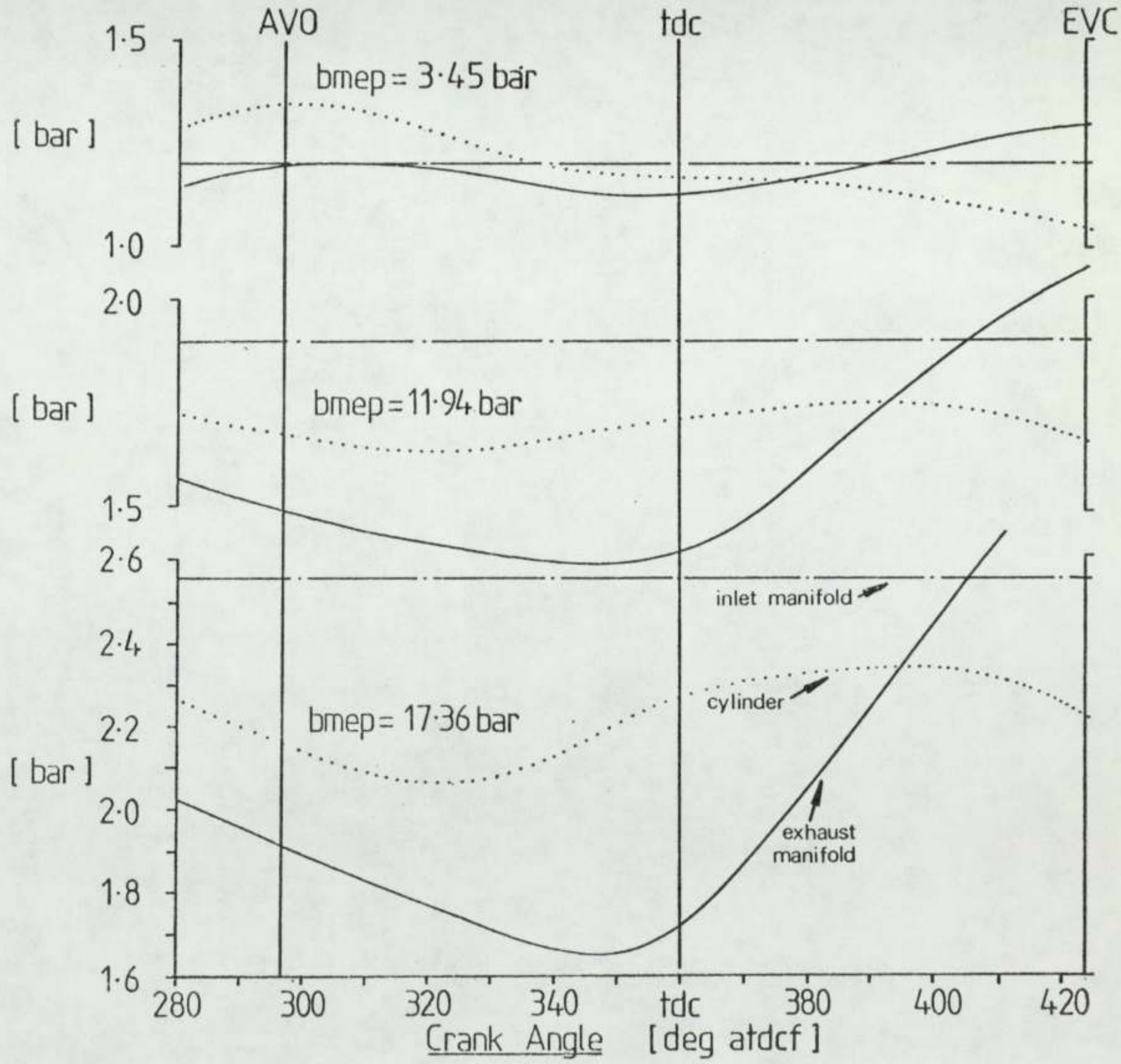


Fig. 4.13 , Computed transient pressure conditions during valve overlap, in the cylinder and manifolds.

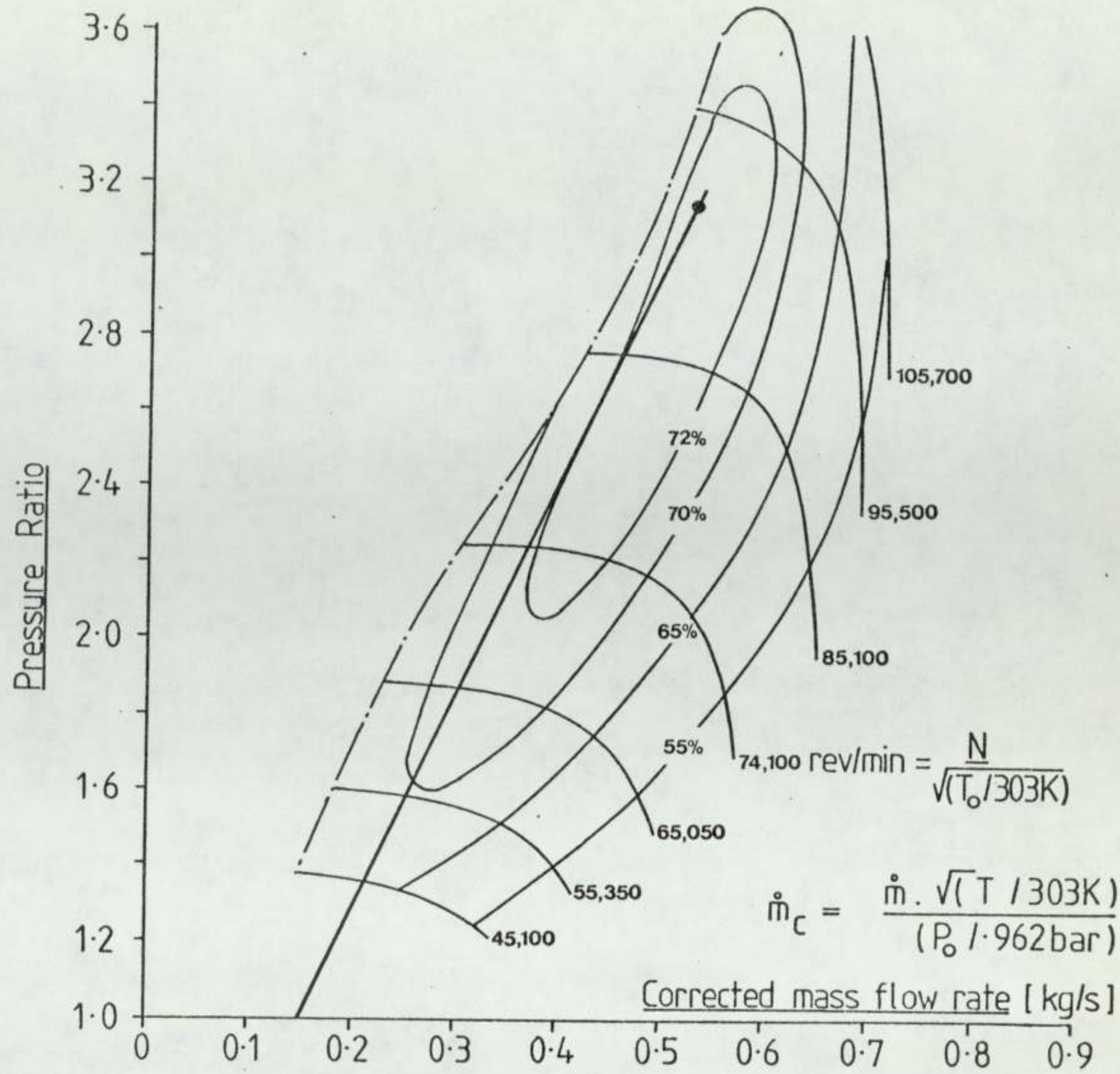


Fig. 4.14 , Performance map of Garrett Airesearch compressor specified for the research engine, showing predicted operating line.

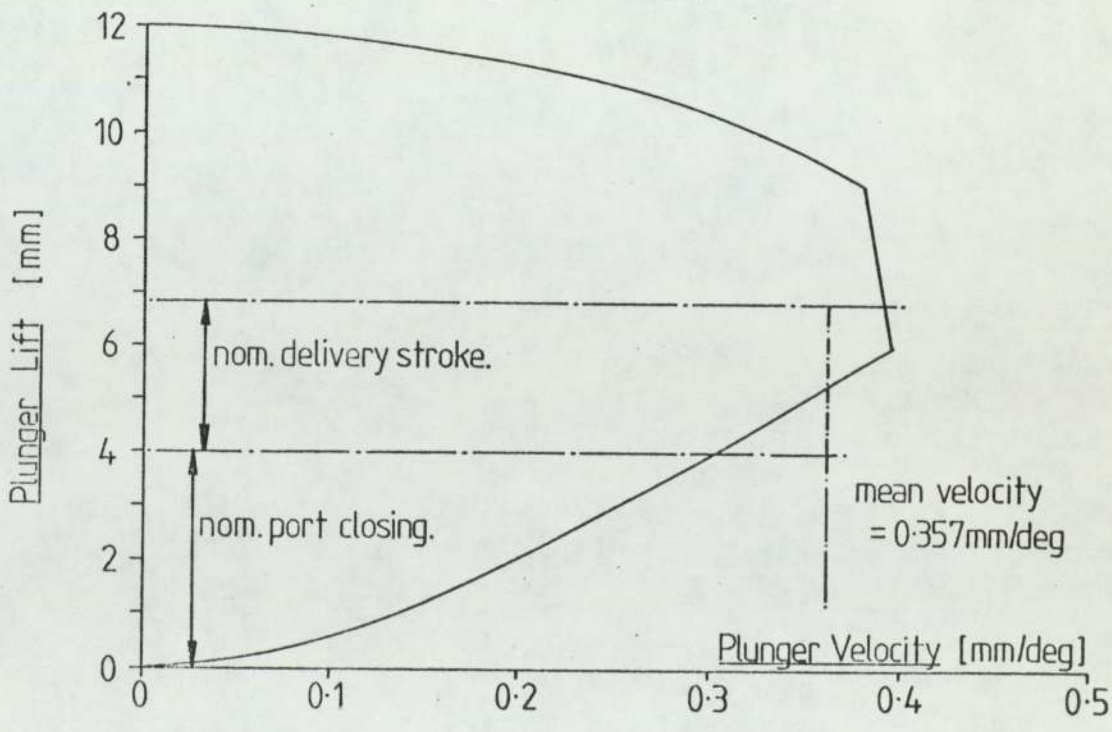


Fig. 4.15 , Characteristic of the specified fuel pump cam form.

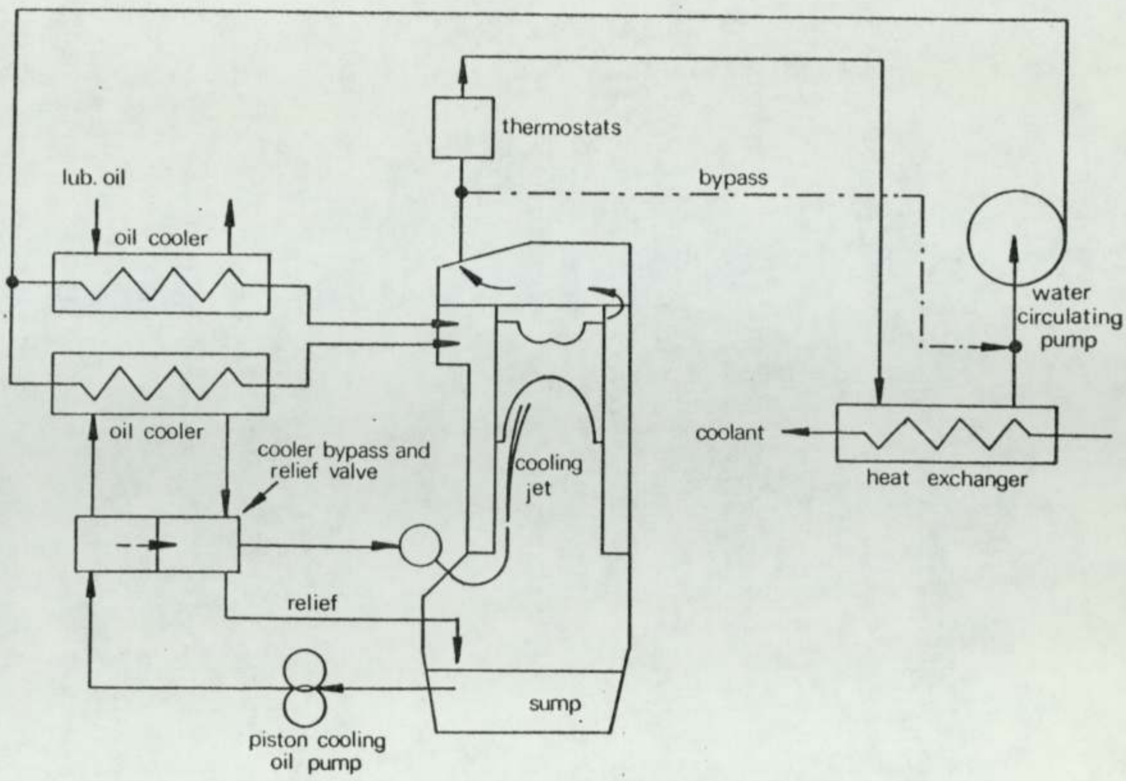


Fig.4.16 , Schematic representation jacket and piston cooling systems.

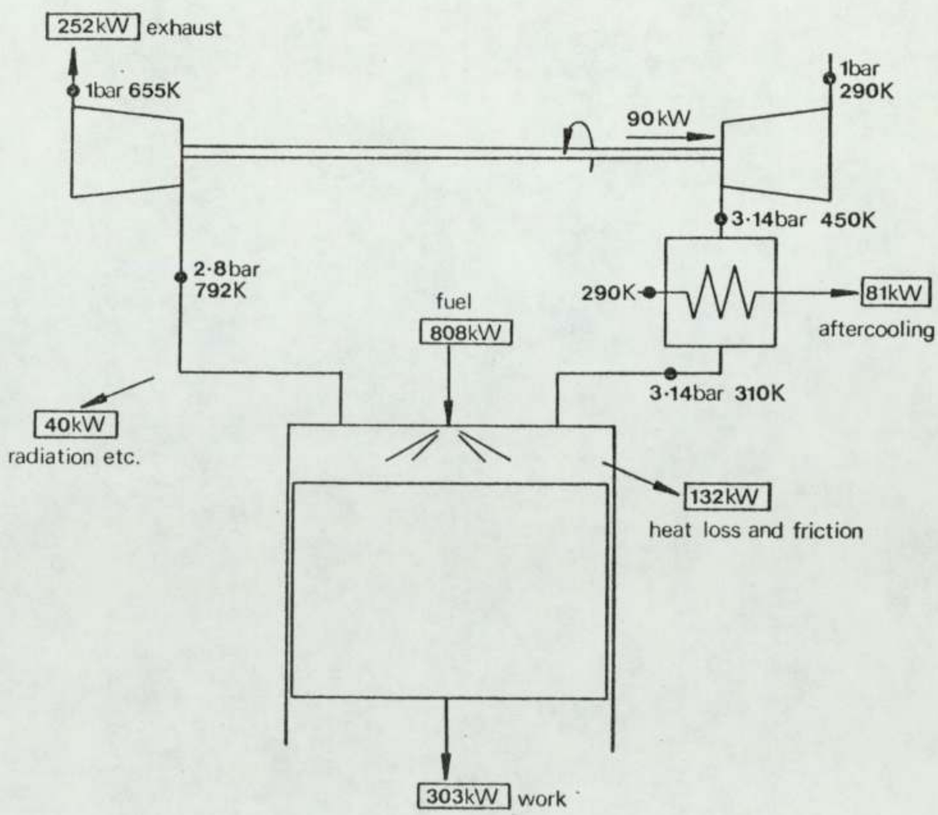


Fig.4.17 , Energy flow [kW] , mean pressures [bar] and temperatures [K] predicted for the research engine at a bmep of 21bar.

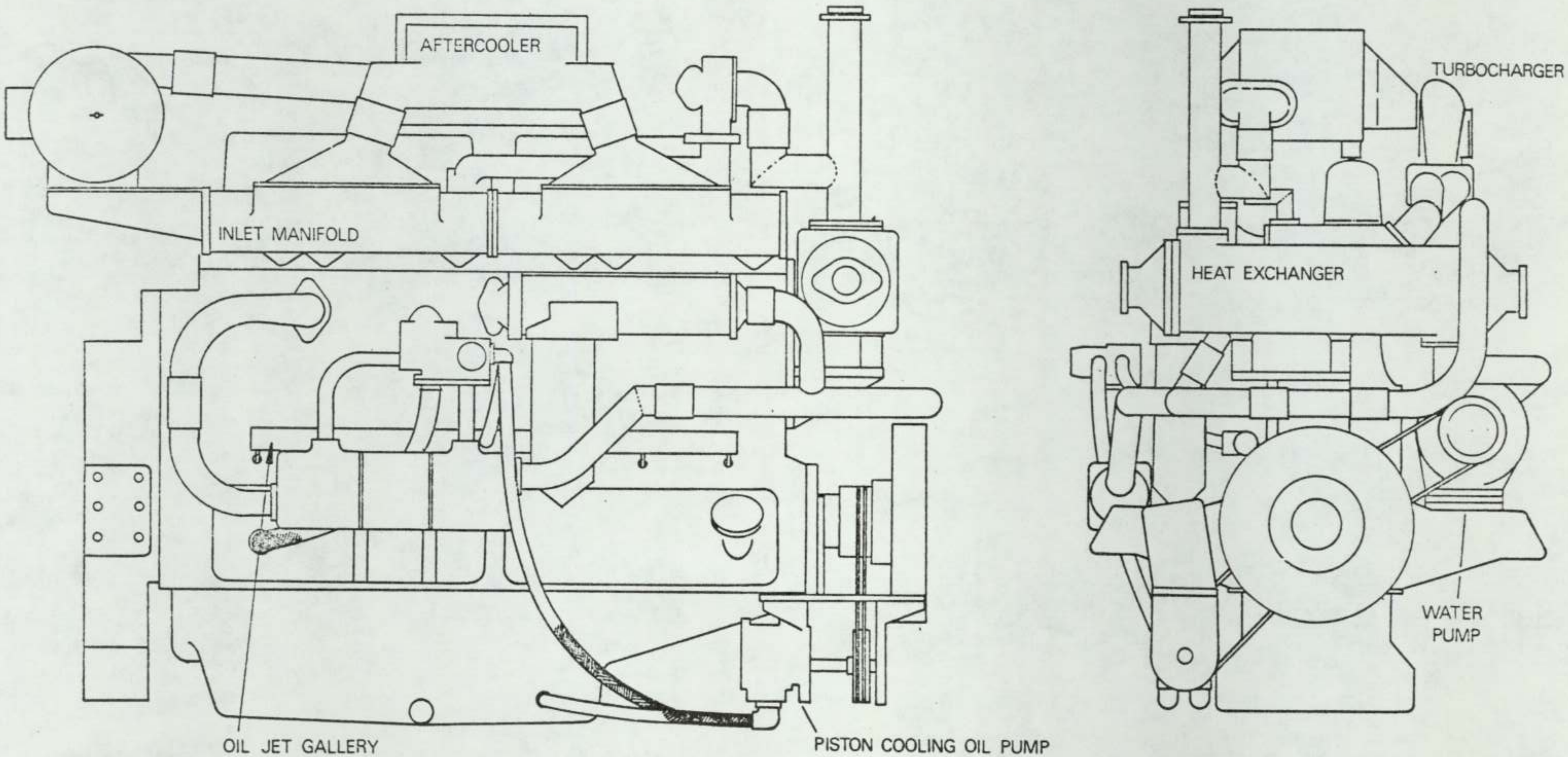


Fig.4.18, Outline drawing of the research engine, showing the layout of cooling and induction systems.

CHAPTER 5

Experimental Methodology and Instrumentation

5.1. Introduction

Very few technological products can be designed and put into quantity production without experimentation where failures will not damage the company's business potential - in the R & D department or in field trials. This widespread practice reflects the limitations of the designer ; he cannot foresee every operational detail of the equipment he designs, no matter how well tried or understood the individual elements are.

Diesel engines are no exception to this general rule. On the contrary, the presence of design elements which are not fully understood make development mandatory. For example, combustion, fuel injection and turbocharger match are optimised during development when the functioning of the whole engine is scrutinised. This process allows for feedback from the development engineer to the designer.

This procedure is relevant when the engine concept and design point are within the company's experience. This project extends the company's area of experience by extrapolating the performance level of an existing engine concept. Therefore, although the practical work will involve development techniques, particularly in uprating the engine initially, the exploratory work at the target load will be of a research nature.

5.2. Objectives

The objective of the project as a whole is to assess the feasibility both technically and commercially, of high specific output variants of conventional designs. The company requires first-hand experience of the concept to establish the characteristics, the problems, and the opportunities it presents.

To assume that the findings of this study could be conclusive is optimistic in view of the limited time scale. Engine performance is not fully defined by the level of specific output, thermal efficiency or other immediately measurable parameters. It is the sum of many factors, some, such as service life and reliability requiring thousands of hours of operating experience to establish. Others, such as startability and load acceptance would require research programmes of their own. Given unlimited resources, all of these questions could be answered within the time available by allowing such programmes to proceed simultaneously. Since this was not permissible, the limited scope was taken into account when setting objectives.

The prime objective of the experimental programme was to demonstrate the feasibility of extending the performance of a conventional Dorman engine up to a bmep of 21 bar at fixed speed. Once this level had been achieved, the objective was to collect as much relevant data as the time scale would permit, particularly related to durability, for example, piston and exhaust valve thermal load. In addition, the design would be varied where feasible, to evaluate the effects on performance. This would include compression ratio, charge pressure and fuel injection characteristics.

5.3. Planning

Planning is an important precursor to all industrial activity and is essential when dealing with valuable resources. The planner should seek to utilise these effectively to meet the objectives in question. The resources available to the development programme were time, skilled manpower and facilities.

Planning was a continuous process throughout the development programme, requiring frequent updating as events unfolded. The criterion used to form plans was to minimise "dead" time by scheduling components and instruments ahead of need, performing experiments in an order that minimised rebuilding and streamlining data collection within the scope available. Thus, short-term planning involved maximising the data collected per unit time.

Longer term plans had to ensure that objectives were achieved within the overall time scale. R & D, by definition, involves uncertainty and plans cannot, therefore, be too rigid. For example, the unexpected failure of a part within the engine may consume over one man-week, or more if replacement parts are not readily obtainable. It would be difficult to estimate the duration of the first phase, to achieve target output, since the problems cannot entirely be foreseen. Almost inevitably, forecasts of time requirement are optimistic. Thus, if the time available is fixed, priorities have to be considered and some items cut from the programme. Conversely, should a time forecast be pessimistic, either the work ends ahead of time or other relevant areas are introduced. The first priority of this programme was to operate the engine and collect data at the design output. Secondary considerations were studies of starting and load acceptance. Thus, the time available could be usefully exploited over a wide range of progress rates.

5.4. Experimental Methods and Instrumentation

The normal approach to engine development within the company is to proceed towards a specified performance by a series of objective experiments, more commonly called "trial and error". In this process, the intuition and experience of the development engineer have a leading role, since reliable analytical methods are non-existent in areas such as combustion and fuel injection optimisation. This calls for a high degree of empathy and observation, followed by the formation of hypotheses, and experiment, all features of "scientific method".

Engine experiments are usually performed at constant speed with the load increasing in steps until some prescribed limiting factor is reached. Data are collected manually at each load point by the technician responsible for the engine. The data are then studied and further experiments are proposed until the objective is achieved. This requires close consultation with the engineers of suppliers, particularly fuel injection and turbocharger manufacturers.

To effectively evaluate the performance of a particular configuration it is necessary to measure, or derive, the most informative parameters. Some, for example smoke density are the result of a single measurement, whilst others, such as volumetric efficiency, are derived from several measurements. Figure 5.1 lists the most definitive performance indicators and the physical variables needed to calculate them. These parameters form a "basic" data set which allows a rapid evaluation of performance for development decision-making. At the start of the programme, a computer program (described in Appendix 4) was written, to reduce the data quickly and accurately. This also acts as a check for gross instrument or reading errors. The instrumentation was specified to provide the basic performance feedback of Figure 5.1 for each test. More specialised instruments, for example transient pressure and injector needle displacement transducers, were made available when required.

5.5. Basic Instrumentation

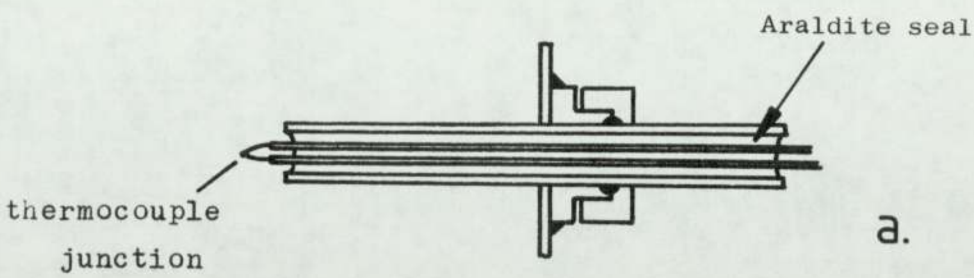
Pressure

Pressure measurements were made by both mercury and water manometers. Low pressures eg, air-flow measurement, air cleaner depression and silencer back pressure were measured using water, whilst for higher pressures in the manifolds, mercury was used. Because of the unsteady flow nature of reciprocating engines, few parameters remain constant. The temperature and pressure in the exhaust manifold fluctuate over a wide range as the cylinders empty. Since the liquid column cannot respond to the frequency of these fluctuations, the manometer will record a steady value near to the time-mean value of the pressure signal. To minimise errors due to friction loss, the pipes connecting manifold and manometer were kept as short as possible.

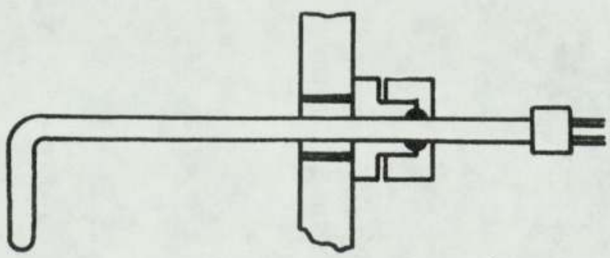
Pulse turbocharging aims to conserve the kinetic energy of the blowdown pulses, and this velocity should be accounted for when calculating available turbine energy. This is most readily achieved using a total pressure probe which faces upstream and brings the flow, locally, to rest. For scavenging calculations, the static pressure is of greater value, therefore, both total and static pressure probes were provided in each manifold branch.

Temperature

Fluid temperatures throughout the system were measured by Chromel-Alumel thermocouple junctions, which is standard practice in the company. For moderate temperatures (oil, water and charge air) the junctions were made by hand using lengths of proprietary thermocouple wire, as shown in (a) below. In the exhaust manifold, stainless steel sheathed thermocouples were used to resist the unfavourable conditions. The design of these is shown in (b) below.



a.



b.

Two effects leading to temperature measurement error are conduction of heat away from the sensor through the probe structure, and temperature rise in the boundary layer as the stream velocity approaches zero. The conduction effect should be negligible because of the length of the probe and the poor means of heat transfer away from the area. The velocities encountered in typical exhaust manifold flow could add of the order of 5 to 10K to the static temperature, which would introduce a small, but correctable error. These effects offset each other, partly, and likely errors should be a small overestimation of static temperature.

The thermocouples were read by a digital reader with electronic cold junction and compensation for the inherently non-linear sensitivity.

Engine Torque

The engine load was applied and measured by a hydraulic dynamometer. The energy was carried away as heat by water which circulates. The torque created by hydraulic resistance, bearing and gland friction is balanced by a torque reaction which is measured by a weighing apparatus at the end of a lever arm. The principle is inherently accurate provided static balance is achieved.

Engine Speed

Engine speed was monitored by a Racal 9520 digital counter. This receives signals from a magnetic pick-up in proximity to a gear wheel with 60 teeth, in the shaft system. The counting period is one second, which gives a direct reading in revolutions per minute, updated each second.

Fuel Flow

Fuel flow was measured on a volumetric basis, using a standard glass measure of one pint, installed in the fuel line. The fuel supply was allowed to run from the glass measure by operating taps in the system. The upper and lower levels were sensed photoelectrically to operate an automatic digital timing device. Thus, the measurment represents the mean rate over the period required to consume one pint.

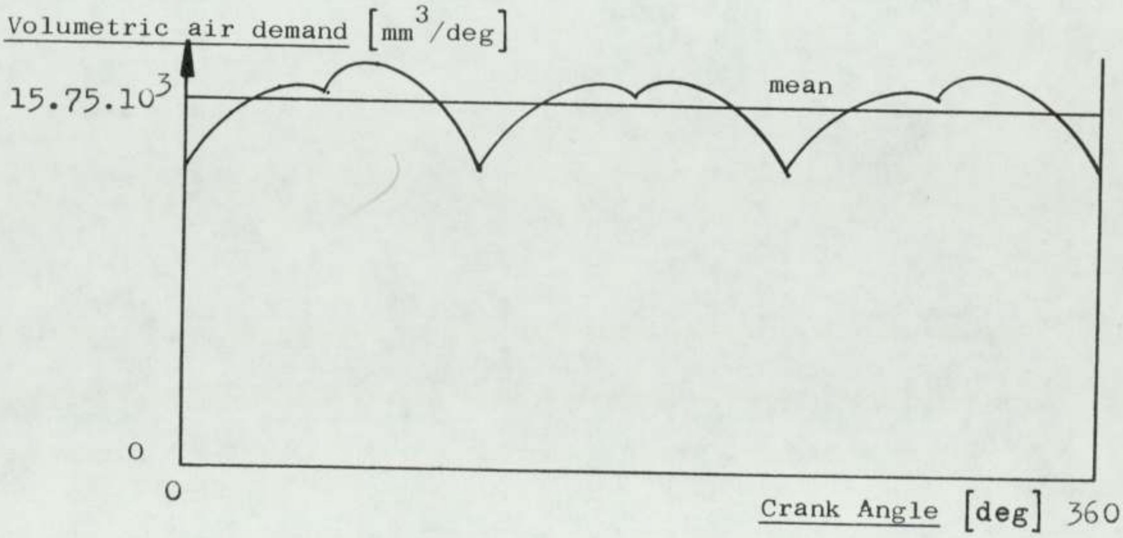
Exhaust Smoke

Exhaust smoke is the presence of particles of soot formed in the fuel-rich areas of the combustion chamber by pyrolysis. The concentration of soot in the exhaust stream was evaluated by a manual Bosch smoke meter. This incorporates a piston which is made to draw a volume (330 ml) of exhaust gas through a disc of white filter paper. The soot particles present in the sample lodge in the filter and darken it, depending on the quantity present. The darkness of the filter paper is then measured using a combined light source and sensor. The scale reads between zero for complete reflection (clear exhaust) and ten for complete absorption. This method does not give a direct measure of soot concentration, which would be more difficult to make, but does give a consistent measure, albeit on an arbitrary scale.

Air Flow Measurement

Accurate air flow measurement is essential for the analysis of all combustion engines, and particularly important when turbocharger performance is to be considered. Measuring the air flow through reciprocating engines is often problematic because of the unsteady nature of the flow. This is a particular problem for naturally aspirated engines with few cylinders. To overcome this it is usual to provide a large chamber in the induction system to attenuate the pulsations. The flow may then be measured using orifice plates or other flow restriction devices. An alternative approach is the Alcock viscous-flow air meter described by Ricardo and Hempson (87).

Air flow was measured by a standard flow nozzle and manometer supplied by Holset Engineering Limited, turbocharger manufacturers. The factors that make a flow restriction device acceptably accurate in this situation are the number of cylinders and the presence of the compressor. Six cylinders with equal firing intervals have a reasonably smooth air demand, as shown below. A parallel duct was provided for one metre downstream of the nozzle to reduce unwanted flow effects.



Two nozzle sizes were used to give the required sensitivity over the wide flow range.

Liquid Flow Measurement

The measurement of oil and jacket water flow was required for control of the piston cooling system and energy balance calculations. All oil and water flow rates were measured using turbine-type flow meters. These devices have a turbine fitted into a short duct which is installed in the fluid circuit. The turbine rotates at a rate dependent on the fluid flow rate. At the opposite end of the turbine shaft is a magnetic sensor which generates one electrical pulse per revolution of the turbine. The reading device converts the frequency of the signal into a DC level, which is displayed by a conventional meter calibrated in the units of volume flow rate.

5.6. Specialised Instrumentation

The foregoing described the basic instrumentation set needed for a rapid assessment of overall performance during development work. For a more detailed appraisal of the individual systems, more specialised and generally, more sophisticated and expensive instruments are required.

Dynamic Pressure Measurement

A transient pressure trace from the cylinder or exhaust manifold provides far more information than a single peak or "time-mean" measurement. A pressure-time diagram gives a quantitative and qualitative description of the processes occurring.

The hostile environment within the diesel engine has made transient pressure measurement a difficult problem. The qualities of piezo-electric transducers have, in recent years, led to their widespread use for internal combustion engines. The main advantages are high frequency response (6kHz) and linearity over a wide operating range. Among the disadvantages are temperature drift, causing datum instability, and a low electrical output.

High pressure cylinder transients were recorded by a Kistler 6121 miniature transducer, charge amplifier, digital storage 'scope and pen recorder. The Kistler 6121 is a quartz transducer designed to withstand soaked temperatures up to 350^{o}C and intermittent flash temperatures up to 2500^{o}C. Its robustness makes it ideal for engine work and permits installation directly into the cylinder without direct cooling. The installation in the research engine is shown in Figure 5.2.

Low pressure transients, or "light spring" diagrams, were obtained from a Kistler 601a quartz transducer mounted in a water-cooled "clipper" adaptor, type 644. The clipper adaptor incorporates a valve and reference pressure source which are used to calibrate the absolute level of the trace.

Fuel line pressures were studied using an AVL piezo-electric transducer. This was inserted into a high pressure fuel line adjacent to the injector inlet connection. Calibration was achieved statically by use of a dead-weight tester.

Injector Needle Lift Measurment

The motion of the injector needle from its seat indicates high fuel pressure and delivery from the nozzle. Transient needle displacement is essential for calculating the instantaneous fuel delivery rate. More commonly, the timing and duration of the injection events are needed for evaluation of injection and ignition delay, mean rate of injection and the presence of secondary injections. These data do not require calibration of the displacement signal.

The method most widely used for needle lift studies is based on the electrical inductance of a coil, which may be modified by the movement of a metal slug within it. This small effect is made to modify a 2MHz carrier wave. A section through the injector used for this purpose is shown in Figure 5.3.

Component Temperature Measurement

The accurate measurement of cylinder component temperatures is an essential part of engine research and development. A simple and rapid method of collecting such data is by inserting specially prepared plugs of metal alloy into the areas of interest. Proprietary plugs, known as "Templugs" and developed by Shell Research, were used. These are made in grub screw form and fitted in flat-bottomed holes made in the component. The lower face is later used for hardness measurement. The technique relies on the permanent hardness changes that some alloys undergo when heated (88). To achieve the desired effect, the engine should be run steadily at the load of interest for a minimum of only one hour. Thus, the greater part of the exercise usually involves stripping and rebuilding the engine. The method gives point measurements, which is a disadvantage since many plugs would be needed to build up a coherent picture of the temperature field. Shell specify an accuracy of $\pm 5K$.

More comprehensive information can be obtained if whole components are made of such alloys. Piston aluminium alloys exhibit this property, but need in excess of 25 hours of steady running to produce a significant hardness change. This is a widely used method, but was rejected in favour of "Templugs" because of time constraints. However, the principle was used for inlet and exhaust valve thermal surveys. Thermometric valves of hardened EN52 steel alloy were run at steady load for 4 to 6 hours. This allows the material to temper and the resulting hardness distribution may then be related, using material calibration curves, to the temperature distribution during operation.

Other methods of metal temperature measurement allow data to be collected whilst the engine is running. These usually involve thermocouple junctions or resistance thermometers in intimate contact with the area of interest. When measurements inside the component are needed, small drillings are made to allow insertion of the sensor. Heat flux may be estimated using a traversing thermocouple (24)(45). This may be moved along a drilling in the chamber wall, thus establishing the temperature gradient in the direction of the hole. However, the presence of the hole introduces a small error. Another method of local heat flux measurement involves inserting a specially prepared plug of material into the wall. This is fitted with thermocouple junctions near to the inner and outer faces, a known distance apart.

If the material conductivity is known, the heat flux is calculable from the measured temperature difference (89). Moving parts may be studied in this way using link mechanisms (24) or contacts which touch at one extreme of the motion (44).

Having surveyed the flamedeck temperature distribution using "Templugs", the hottest area was found to be around the exhaust valve seat. To extend study of this area, eight thermocouples were fitted to the rear cylinder head, as shown in Figure 5.4. A difficulty of point temperature measurement, using a sensor of finite size, is the averaging effect over the area of contact, especially where steep gradients are present. This effect was minimised by using thermocouples of 1mm diameter and inserting them approximately normal to the direction of the heat flow. The measuring point was the end of the thermocouple, which was of the grounded type. Conduction effects were minimised by providing a slight clearance between the hole and the thermocouple sheath. Adequate contact pressure between the thermocouple and the flat-bottomed hole was achieved by the screw fitting, as shown in Figure 5.4.

5.7. Discussion of Experimental Accuracy

The level of accuracy required of an experimental measurement will depend on the use to which the result is to be put. If it is desired to measure a fundamental quantity such as the speed of sound or the acceleration due to gravity, strictly for the advancement of knowledge, then extra care should be taken to minimise errors and to correct for those that cannot be eliminated.

In applied research, the highest accuracy is sought because the results are often used as the basis of empirical models, to be used in design calculations. Whilst the experimental accuracy of this project was considered important, it was also recognised that economic constraints would force some compromise. For example, data logging, or transducers for pressure and flow measurement were not feasible. The physical variables listed in Figure 5.1 were recorded manually, using basic instruments such as manometers, thermocouples and flow nozzles.

The reading accuracy of these methods generally depends on the percentage of the full-scale deflection, since a "fixed" reading error is most probable. For example, a monometer could probably be read to within ±3 mm regardless of the scale deflection. Similarly, temperature could be read to within ±0.5 K.

The engine was allowed 20 minutes to stabilise prior to the first set of readings at a given load. This was usually followed by a second set of data after 30 minutes, after which the load would be reset.

The reading accuracy of a single measurement would typically be within ± 2.5 per cent, at worst ± 5 per cent, whilst the accuracy of the derived performance indicators would depend on the accuracy of the variables from which it is calculated.

performance indicators / physical variables	Engine Speed	Brake Load	Air Flow Rate	Fuel Flow Rate	Smoke Density	Maximum Cylinder Pressure	Compressor Entry Temperature	Compressor Exit Temperature	Aftercooler Exit Temperature	Aftercooler Coolant Temp.	Turbine Entry Temperature	Turbine Exit Temperature	Compressor Entry Pressure	Compressor Exit Pressure	Aftercooler Exit Pressure	Turbine Entry Pressure	Turbine Exit Pressure	Jacket Coolant Flow	Coolant Temperature Rise
Brake Mean Effective Pressure		O																	
Brake Thermal Efficiency	O	O		O															
Volumetric Efficiency	O		O						O						O				
Smoke Density					O														
Maximum Cylinder Pressure						O													
Compressor Pressure Ratio													O	O					
Compressor Corrected Flow			O				O						O						
Compressor Efficiency							O	O					O	O					
Compressor Power			O				O	O											
Aftercooler Effectivemess								O	O	O									
Turbine Pressure Ratio																O	O		
Turbine Corrected Flow			O	O							O					O			
Available Turbine Energy			O	O							O					O	O		
Apparent Turbine Efficiency			O	O				O			O					O	O		
Air-Fuel Ratio			O	O															
Heat Rejected to Exhaust			O	O			O		O			O							
Heat Rejected to Aftercooler			O					O											
Heat Rejected to Coolant																		O	O

Fig.5.1, Physical variables and performance indicators that form the basic data set.

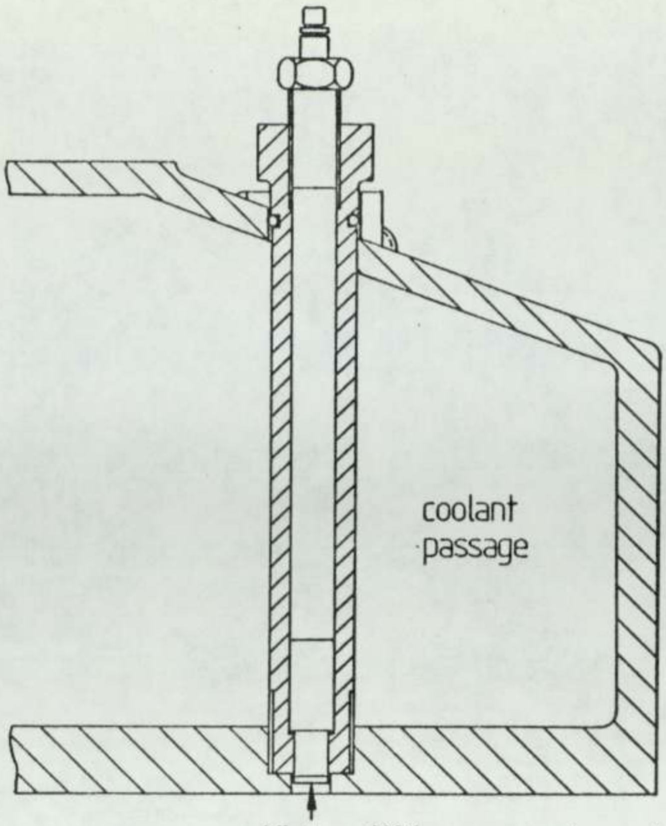


Fig. 5.2, Part section of a cylinder head showing the installation of the pressure transducer directly into the cylinder.

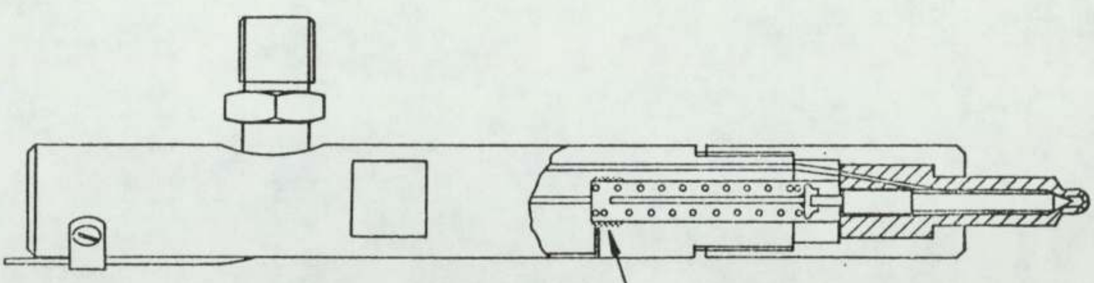


Fig. 5.3, Section through a low spring injector instrumented for needle lift measurement, showing the location of the induction coil.

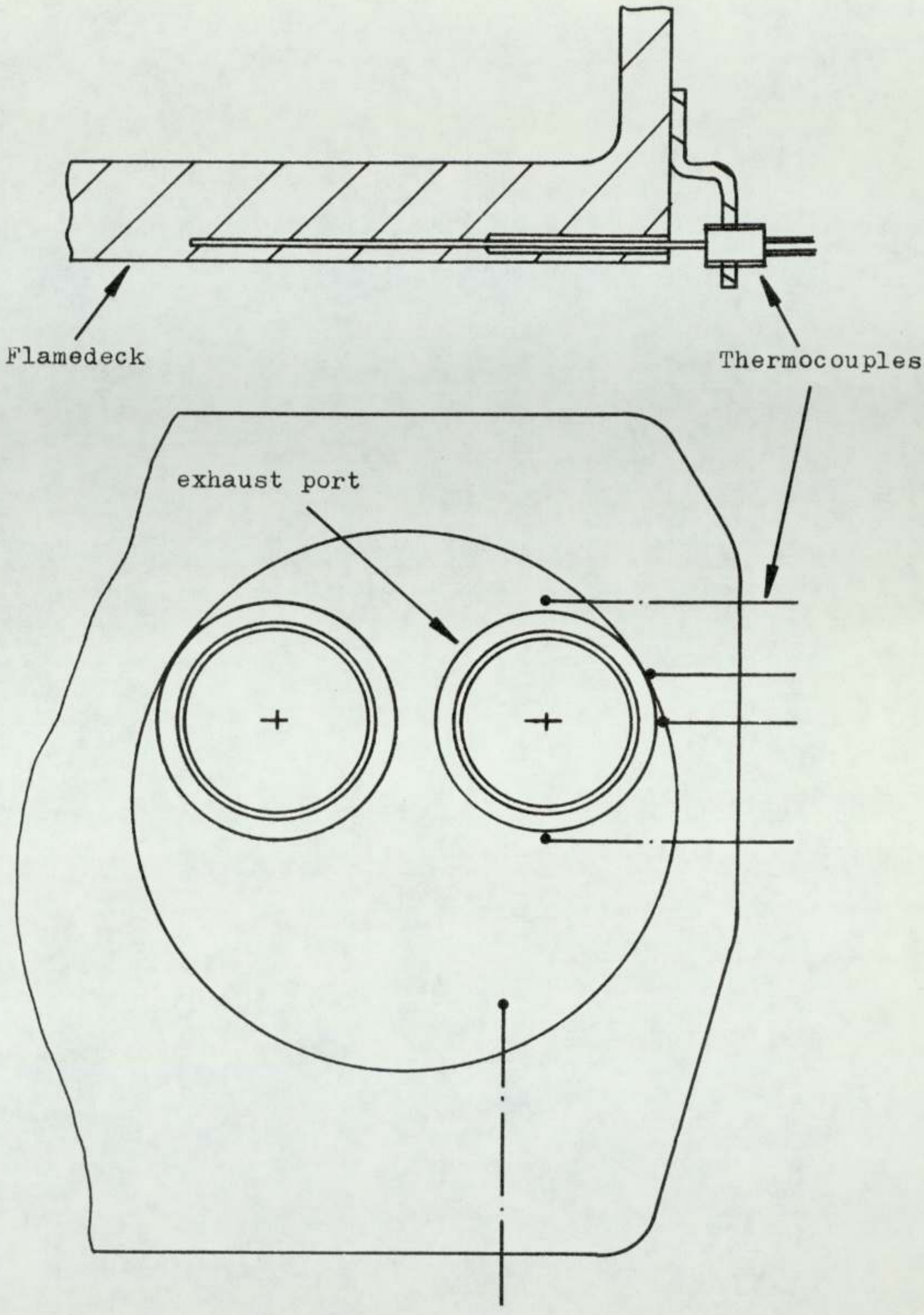


Fig.5.4, Installation and location of the thermocouples in the cylinder head flamedeck.

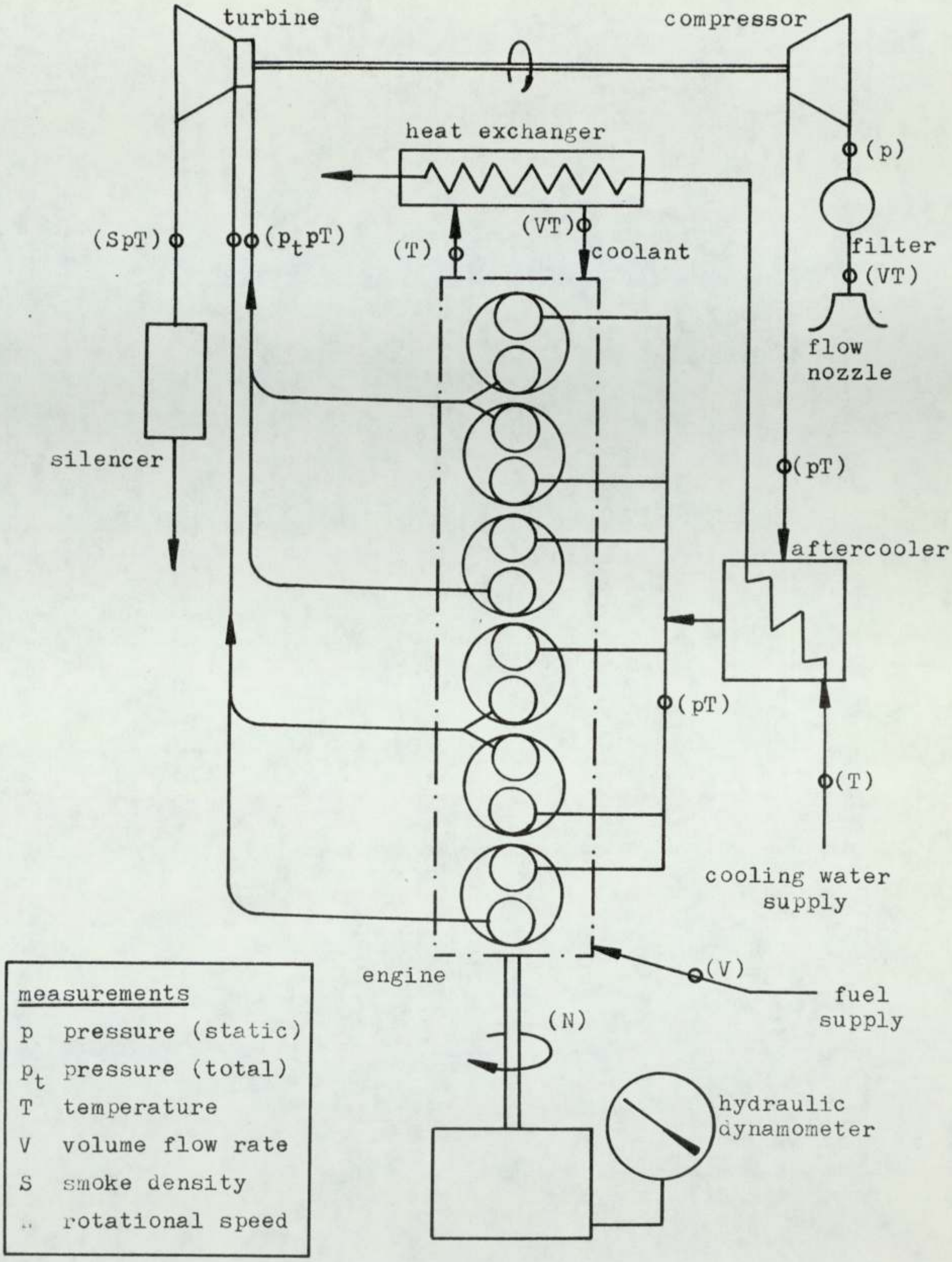


Fig.5.5, Schematic representation of the test bed installation.

CHAPTER 6

Experimental Results

6.1. Introduction

This chapter presents and interprets the results of the experimental programme. It is intended to complement Chapter 5, which discusses the practical issues of experimentation and data gathering. The objectives of the experimental programme were to increase the company's knowledge and experience of the high output concept, applied to one of their current products, as an aid to sound decision-making. Specific power output of this order is arousing the active interest of several high-speed engine manufacturers ; it is, therefore, desirable, for competitive reasons, to be aware of the problems to be solved and the opportunities it presents.

The experimental programme can contribute by defining the performance characteristics, the areas of distress and the areas where "special" components or exotic materials may be needed in an engine built for such power output. Of course, the "LE" engine is a particular engine of a particular size and configuration, and it would not be prudent to make generalisations about other Dorman engines of different design. For example, turbocharging and scavenging three cylinder groups is not the same as two cylinder groups. Larger cylinders require less swirl and have a higher chamber volume/surface area ratio. Larger components require more stringent cooling measures, etc.

These are good reasons for feeding back data from the programme to allow development of the predictive methods used at the design stage. This will surely allow greater design effectiveness in a future where "not getting it right first time" is becoming increasingly expensive.

The experimentation proceeded essentially by building the engine in a given configuration and putting it through a series of steady-state loads at constant speed. The variable design elements were compression ratio , fuel injection equipment and turbocharger build. In all, 17 builds, excluding static timing and speed variations were used. Table 6.1. gives the design specification of each build, numbering 1 to 17. This forms the basis of a shorthand notation to the data presented in this chapter. Rather then give all the details of the build with each graph, the build number and static timing will be given : for example [build 15, spill 22^{o}]. The engine speed throughout was 1500 rev/min unless stated otherwise.

6.2. Engine Development

At the start of the project, the turbocharged and after-cooled "LE" engine was marketed up to a maximum continuous bmep of 13 bar at 1500 rev/min. The design study described in Chapter 4 indicated the need for many modifications to enable the target output of 21 bar at 1500 rev/min to be achieved. The research engine specified for the experimental programme featured a new turbocharger and

build number	compression ratio	turbocharger		fuel pump (element)	injector nozzle	camshaft valve overlap
		compressor	turbine			
1	11·9	Tv71 A-8	1·41E	MAJORMEC 12mm	4x0·35mm	22·5°
2			1·08E			
3						127·0°
4				MAXIMEC 12mm		
5		Tv71 0-1				
6					4×0·39	
7					4×0·40	
8					4×0·42	
9	10·7				4×0·39	
10					4×0·40	
11					4×0·42	
12		Tv61 W-4	1·15G		4×0·39	
13			1·00G			
14	11·6					
15			1·15G			
16				MAXIMEC 13mm		
17					4×0·42	

Table 6.1, Configurations of the research engine during the experimental programme.

after-cooler, fuel system, cam shaft, compression ratio (hence combustion chamber) and jacket cooling components. The number of changes ensured that routine development would be needed to smooth out unforeseen problems. The progress rate may be seen in Figure 6.1.

The compressor match originally specified gave a full load surge margin of about 6 per cent, based on mass flow. Figure 6.2 (a) compares the actual and predicted matches. The "LE" low overlap cam shaft was used early in the programme whilst awaiting delivery of the high overlap cam shaft. With low overlap the compressor quickly ran into surge. When the overlap was increased to 127 degrees of crank rotation, scavenging and through-flow increased the volumetric efficiency with load, thus reducing the gradient of the operating line. Qualitatively, the effect shown in Figure 6.2 (a) was accurately predicted by the design study. Quantitatively, the prediction was in error, overestimating the volumetric efficiency of the engine by about 5 per cent. The reasons for this error are discussed in Section 6.7. This reduced the expected surge margin to almost zero, and necessitated rematching. The original compressor was at the lower end of the TV71 frame size range and the only alternative offering greater surge margin is shown in Figure 6.2 (b). This gave only 6 to 8 per cent surge margin but was fitted because it was immediately available and would allow development to proceed. Having hard data available, the turbocharger makers proposed a change of frame size - downwards from TV71 to the TV61. Figure 6.2 (c) shows the TV61 compressor match, the surge margin being increased to almost 20 per cent.

To assist turbocharger matching, two turbine housings having different effective areas were available for each turbocharger, the TV71 and later TV61. The flow characteristics of these turbine configurations, under pulsating flow from the engine, are given in Figure 6.3. Curves 1 and 2 are TV71 turbines, curves 3 and 4 being TV61 turbines.

Turbine performance throughout this work was calculated on a quasi-steady state basis, using values of pressure and temperature in the exhaust pipe measured by lagging instruments. This introduces and underestimation of turbine energy and hence an overestimation of turbine efficiency (3). Therefore, turbine energy and efficiency are not actual values but "apparent" values.

The available turbine energy may be controlled by the effective area of the nozzles in the turbine housing, thus varying the pressure head. In practice, this is achieved by producing a series of turbine housings for each rotor covering a range of effective flow area. Curves 1 to 4 in Figure 6.3 have decreasing area, hence 4 is seen to choke at a lower mass flow than 1, and will provide greater available energy. Figure 6.4 shows the effect of turbine area on available power and the resulting compressor pressure ratio. Insofar as engine and turbocharger efficiency remain constant, the air-fuel ratio is determined by the turbine characteristic, at any given load. As the load is increased at fixed engine speed, the increment of fuel increases the turbine energy causing the turbocharger to accelerate and provide an increment of air. However, the fuel flow increases at a higher rate than the air flow and causes the air-fuel ratio to decrease with load. Turbine 1 gave an inadequate air supply and was rejected early in the programme. Turbine 2 was better, but was regarded as marginal. When the TV61 turbocharger was ordered, the performance of turbines 1 and 2 were used as a guide to the required turbine area, which led to the specification of turbines 3 and 4.

The development phase occupied eight months during which the maximum bmep that could be sustained satisfactorily was increased to 21 bar. This does not imply that the engine was highly developed, rather that the individual systems were functional and that the engine as a whole was stable at the target specific output.

During this phase, the engine was fitted with the components arriving behind schedule, notably the fuel injection equipment, and the TV61 turbocharger. Fuel system performance was not evaluated until later in the programme. This was because of the late arrival of the CAV low spring injectors, and difficulties with the needle lift instrumentation. When measured, the mean rate of injection was less than anticipated, and modifications were made. The fuel injection injection characteristics are discussed in Section 6.4.

6.3. Turbocharging and Induction

The turbocharging and induction system is taken to include the turbocharger, aftercooler, manifolds, ports and valves. The gas exchange process, the degree of scavenging and the air supply characteristics are all determined by the design of these components.

During the design study, the performance of the turbocharging and induction system was modelled by the "filling and emptying" method. Figure 6.5 compares the actual and predicted performance after rematching the turbocharger.

The greatest source of error was the overestimation of the trapped volumetric efficiency. This caused a significant miscalculation of air consumption which led to the turbocharger rematching outlined in 6.2. The design trapped air-fuel ratio was 28 which, with the supposed trapped volumetric efficiency of almost 100 per cent on a swept volume basis, was achievable with single-stage turbocharging and a high degree of after-cooling. In practice, the trapped volumetric efficiency was about 87 per cent. This prevented the use of a high air-fuel ratio, unless a two-stage turbocharging system was adopted. A trapped air-fuel ratio of 28 would require a compressor pressure ratio of around 3.8 and a reduction of compression ratio from 11.7 to 10.1. Neither of these measures were regarded as acceptable and the alternative, to reduce air-fuel ratio was pursued.

The characteristics of four configurations capable of operation at high output are given in Figures 6.6 and 6.7. At no-load, the overall air-fuel ratio is as high as 80. This falls rapidly to 40 at a bmep of 5 bar and thereafter decreases gradually to 28 at 21 bar. It has been estimated, using the "filling and emptying" method, that the trapped air-fuel ratio at full load is approximately 24.5 giving a scavenge ratio of 13 to 15 per cent, as predicted during the design study. A benefit of wide valve overlap is the increase of trapped volumetric efficiency, approximately 4 per cent in this case due to scavenging. This means that the required trapped air mass can be delivered by a lower charge pressure than with low valve overlap.

If the low overlap cam shaft were used at full load, the charge pressure would have to be increased by about 0.15 bar to compensate, thus causing an increase of peak cylinder pressure of about 5 bar or necessitating a reduction of compression ratio of 0.5.

At the start of the experimental programme, the opportunity was taken to run the engine with the "LE" standard "low" overlap cam shaft. This has a nominal overlap period of 22.5^{o} which, in practice, isolates the exhaust and inlet manifolds preventing scavenging flow. At that stage of development, the maximum bmep obtainable was limited to 14 bar. However, the results offer some insight and are presented in Figure 6.8.

The volumetric efficiency of the low overlap build was less than expected for a mid-range speed. It was approximately constant over the load range at 84 per cent, based on swept volume and the density at port entry. The high valve overlap cam shaft modified the volumetric efficiency of the engine, making it partly dependent upon the pressure differential across the cylinders. Figure 6.8 shows the volumetric efficiency rising from less than 80 per cent, at a bmep of 4 bar to 98 per cent at 14 bar. At the lower loads when the pressure differential is low or negative, significant flow reversal occurs through either or both valves, which reduces the volumetric efficiency to less than the low overlap case. This effect is shown in Figure 6.9, which describes the gas exchange processes at no-load. At higher loads, a positive pressure differential is developed and through-flow is increased. This may be seen from Figure 6.11 which shows the gas exchange processes at a bmep of 21 bar.

Zinner (3) presents a simplified model of scavenge flow in four-stroke engines. An equivalent area is derived from the valve effective areas during overlap which hypothetically connects the inlet and exhaust manifolds throughout the cycle. The flow through this area is then calculated in the normal way for a simple orifice, using the mean pressure differential between the two manifolds. This method is outlined in Appendix 1.

The cam shaft designed for the research engine had a nominal valve overlap period of 127 degrees of crank rotation. The crank angle - valve area integral during valve overlap, which depends on valve velocity as well as overlap duration, was 23,000mm^2 degrees. This translates to an equivalent area per cylinder of 31.8mm^2 or 191mm^2 for the whole engine, connecting the inlet to the exhaust manifold throughout the cycle. Figure 6.12 compares experimental data with the results calculated by this method, using both compressible and incompressible flow theory. The volumetric efficiency characteristic calculated in this way correlates well with the experimental data, tending to overestimate the through-flow, whether positive or negative. Discrepancies of this order should be expected because of the fluctuating nature of the pressure in the exhaust port, and the assumptions made by modelling the valves and clearance volume as a simple orifice. The method does, however, provide a benchmark against which the performance of a scavenging system may be judged. It should be possible to exceed the predicted performance, since during overlap the pressure differential is usually greater than the mean value, because the exhaust pressure has decayed to its lowest level immediately before the next cylinder discharges, especially for three-cylinder groups. That the degree of scavenging and through-flow depends on the time-area integral for a given pressure differential, may be seen from Figure 6.10. At the lower speeds, more time is available for scavenging, and hence the overall volumatric efficiency is much greater than at the higher speeds.

Unfortunately, it is not possible to calculate or measure, with any certainty, the contribution that scavenging makes to the trapped volumetric efficiency. Models have been proposed which assume perfect mixing with, or perfect displacement of the end gases. In reality, both effects will probably occur. The performance of a scavenging system will depend on factors other than the pressure differential and equivalent area. Equally important are the clearance volume, bowl location and geometry, induced gas motion and the shrouding effect of the valve recesses in the piston.

The research engine operated on the pulse turbocharging system throughout the experimental programme. This is the method adopted for all of the company's turbocharged engines. The pulse system allows the pressure wave set up when the exhaust valve opens to travel at high velocity to the turbine. Thus the turbine expansion ratio varies throughout the cycle as successive pulses are consumed. It is a well-known disadvantage of pulse turbocharging at high specific output, that pulses of high magnitude are handled inefficiently by the conventional radial in-flow turbine. This is mainly because of the high frictional losses and off-design flow angles. Figure 6.11 shows the exhaust pressure trace recorded at the exit from cylinders 4 and 5 under the full load bmep of 21 bar.

The maximum expansion ratio is 4 : 1 and the minimum 2.3 : 1. Although the relevant turbine efficiency-gas velocity curves are not available, it is probable that the efficiency at the highest pressure ratio is less than optimum. Figure 6.13 tends to confirm this, showing the apparent turbine efficiency deteriorating after a bmep of 14 bar, from 73 to 70 per cent. The apparent efficiency is calculated from the mean pressure and temperature at turbine entry and includes the frictional losses of the rotor assembly.

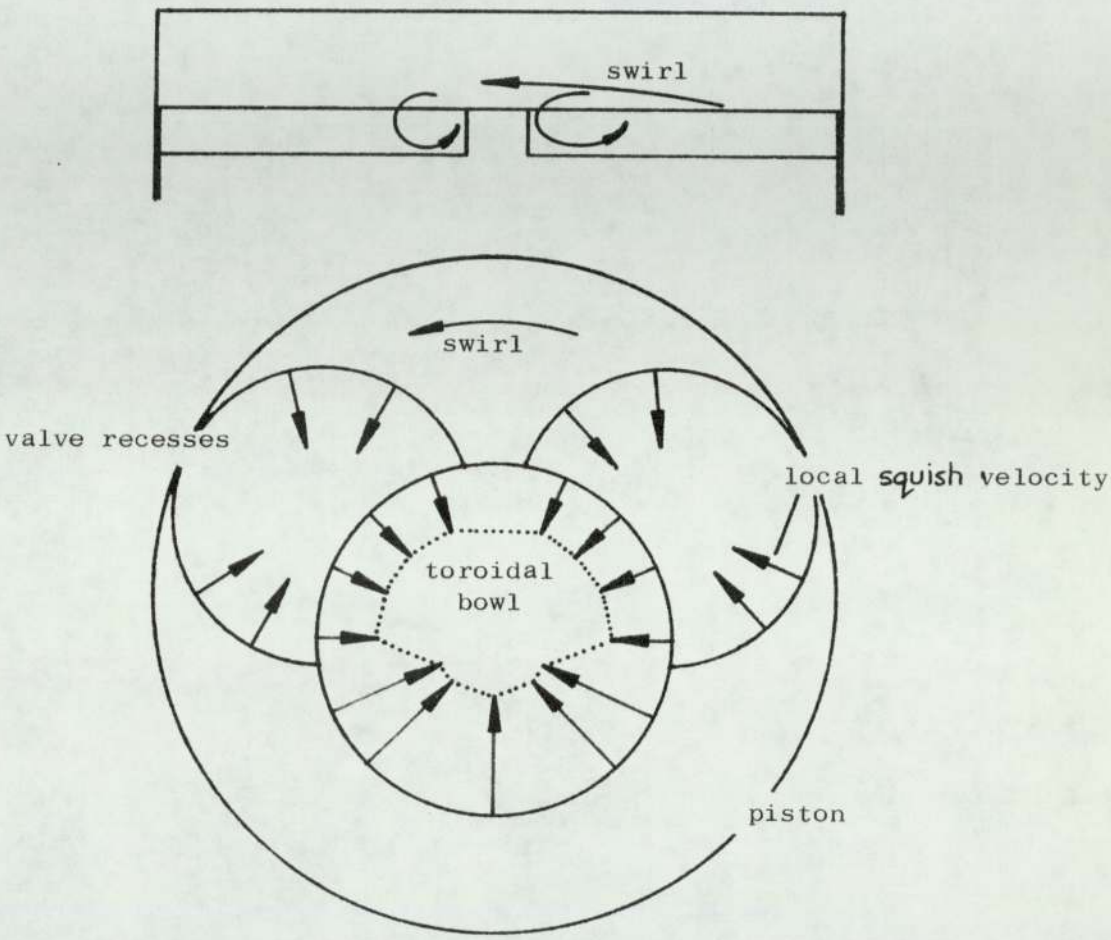
The loss of turbine efficiency, combined with the slowly deteriorating compressor efficiency, adversely affects engine performance as might be expected. In Figure 6.6 the data relating to the 11.6 compression ratio build shows a rapid increase of fuel consumption and exhaust temperature, and a corresponding reduction of air-fuel ratio. This is symptomatic of deteriorating turbocharger efficiency.

6.4. Fuel Injection and Combustion

The fuel injection and combustion results will be presented together, since the former has such a great influence over the latter. The instrumentation used to form an impression of the processes occurring was typical of that used by many engine development laboratories. The measured parameters were cylinder pressure development over the compression and expansion strokes, injector needle lift timing, fuel line pressure, exhaust smoke, gaseous emissions and basic data such as air and fuel flow rates, engine speed and load. Such data does not allow study of the detail of combustion, rather, they provide an overview, treating the complex phenomena in a "black box" fashion.

The use of significant valve overlap makes necessary the provision of recesses in the piston crown to accommodate the valve heads during scavenging. These recesses may be detrimental to combustion in two ways. Firstly, they will disturb the pattern of air motion, thus affecting the mixing of fuel and air. Secondly, the recesses hold air which will be largely inaccessible for combustion until the piston has descended appreciably. Figure 6.14 gives the details of the combustion chambers used during the experimental programme. For design 2, the recesses account for more than 9 per cent of the total clearance volume. If this were unavailable for combustion, the full-load trapped air-fuel ratio would effectively be reduced from about 24 : 1 to less than 22 : 1. An insight of the effects of valve recesses on charge motion and mixing may be gained from the results obtained from the turbocharged "LE" engine. Two high overlap cam shafts were made and the "spare" was evaluated in the current turbocharged development engine. The performance with and without valve recesses, both with a compression ratio of 14.5 is presented in Figure 6.15. The most marked change is the increase of exhaust smoke, despite the increase of total and, probably, trapped air flow. The smoke was present over a wide load range as a high "background" level. The most likely reason for this deterioration is that the recesses disturbed the mixing process, since, at the lower load, the air-fuel ratio in the bowl alone should have been sufficient for clean combustion. Conversely, if fuel spray entered the recesses it could have caused smoke through a locally high air-fuel ratio.

The presence of valve recesses is likely to affect both swirl and squish. Swirl is an organised charge rotation and squish the radial motion from above the piston periphery into the bowl, towards top dead centre. These effects are shown schematically below.



The high output research engine exhibited a smoke characteristic similar to that of the turbocharged engine, both in shape and magnitude, as may be seen from Figure 6.6 and Figure 6.7.

The fuel injection equipment described in Chapter 4 could not be fully evaluated until late in the programme because of difficulties with the instrumentation. Therefore, when the injection characteristics were obtained, little time remained for modification. The performance of the specified system is given in Figure 6.17 which shows that the duration of injection at full load approaches 30 degrees of crankshaft rotation as compared with the nominal period of almost 15^{o} based on the velocity of the plunger and incompressible flow. The difference between the nominal and actual rate is caused by the expansion of the containment, particularly the fuel pipe, and the compressibility of the fuel. Howarth (47) indicates the difficulties of attaining a high rate of injection and likens it to "playing billiards with an eel for a cue !" It is difficult to judge whether this rate of injection is optimum or near optimum until other rates of injection have been tried. The symptoms of a low rate of injection are poor fuel consumption and high exhaust temperatures, whilst an excessively high rate will be indicated by high rates of pressure rise and rough running, and possibly high peak cylinder pressure. Judged subjectively, the engine ran smoothly and with an acceptable noise level, except shortly after a cold start, and when accelerating from a cold start. Under these conditions, the "diesel knock" caused by the high rate of pressure rise was akin to a rapid succession of hammer blows inside the engine. This condition was less severe when the engine was started "warm", and ceased under all conditions when 15 to 30 per cent load was applied. This indicates that low compression temperature was the primary cause, as may be expected.

Figure 6.18 gives overall noise measurements in the bmep range 4 to 18 bar, with a compression ratio of 11.6. These are compared with the noise levels calculated by the model proposed by Hawksley and Anderton (30) which is given overleaf.

$$\text{Overall noise (dBA)} = 30\log_{10}N + 50\log_{10}D + (2.\text{IRPR}-13)\log_{10}\left(\frac{5455}{N}\right) - 103$$

where N is the engine speed (rev/min)

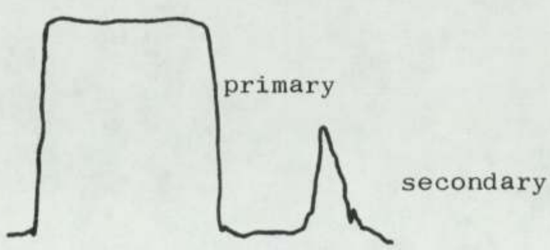
D is the engine bore (mm)

and RPR is the rate of pressure rise (bar/degCA)

The measurements were taken at six points round the engine at a distance of lm, in an open shop, which would be classed as a semi-reverberant field. Thus the results cannot be assumed to be definitive.

The smoothness of the engine under load suggests that the rate of injection achieved by the originally specified equipment was not too high. Conversely, the fuel consumption and exhaust temperatures do not immediately suggest that the rate of injection was unduly low. The fuel consumption recorded throughout the programme was close to 225g/kWh (.37 lb/BHP hr) at full load, including the "low" compression ratio (10.7) build. Exhaust temperatures generally remained between 625 and 700^{o}C at full load, but this was assisted by significant scavenge flow.

Towards the end of the programme, the CAV "Maximec" fuel pump was rebuilt with elements of 13mm diameter, replacing the original 12mm elements. Nominally, this would have increased the rate of injection by 17 per cent. The rates of injection with 12 and 13mm elements are contrasted in Figure 6.16. This shows that the rate was increased as expected, but above 80 per cent load, the larger plungers were inferior because of secondary injections as shown below.



Needle lift curve at a bmep of 20 bar, showing secondary injection, with 13mm fuel pump elements

This was probably caused by high residual pressure In the fuel pipe after injection, allowing a reflected wave to return at a high pressure level, and open the nozzle. This could have been improved by using a larger retraction volume to unload the piping at the end of injection.

The fuel line peak pressure represents an ultimate limit to the rate of injection, since it determines the stress levels throughout the system. Figure 6.19 gives the fuel line pressure recorded at the injector end of the fuel pipe. At a bmep of 20 bar, the peak fuel pressure approaches 700 bar, which is well within the capacity of the Maximec fuel pump, nominally 1000 bar. By making a number of simplifying assumptions, it is possible to estimate the potential rate of injection if the full capability of the pump is exploited.

Considering the nozzle as a simple orifice with a discharge coefficient of 0.7 :

$$\dot{V} = \frac{0.7\ A_N}{6N} \sqrt{\left[\frac{2\Delta p\ 10^5}{\rho_f}\right]} \cdot 10^3 \qquad \text{mm}^3/\text{deg} \qquad (6.1)$$

where N = engine speed (rev/min)

A_N = nozzle area (mm^3)

ρ_f = fuel density (kg/m^3)

$\dot{V}$ = volume flow rate (mm^3/deg)

Δp = pressure drop across the nozzle (bar)

If we assume that the mean injection pressure is between 0.5 and 0.7 of the peak injection pressure, we have :

$$500 \leqslant \bar{P}_f \leqslant 700 \text{ bar} \quad \text{when } \hat{P}_f = 1000 \text{ bar}$$

if we further assume that the mean cylinder pressure during injection is between 80 and 100 bar, then

$$400 \leqslant \overline{\Delta P}_f \leqslant 620 \text{ bar}$$

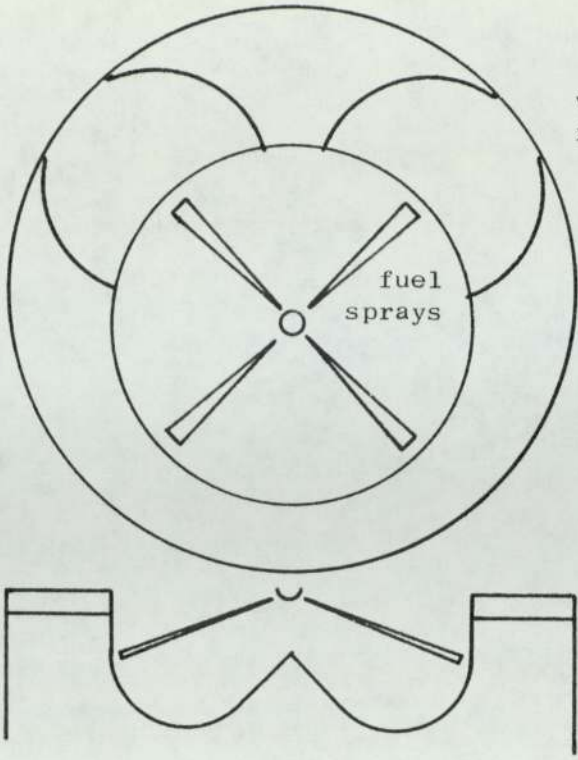
Using equation 6.1

$$12.63 \leqslant \dot{V} \leqslant 15.39 \quad mm^3/deg$$

This gives the possible period of injection in the range 19.5 to 24 degCA, which compares with the 28 to 30 degCA achieved with a maximum fuel pressure of 700 bar.

Thus, there is scope to increase the rate of injection, the benefits of which would be an improvement of thermal efficiency and exhaust smoke, but at the expense of rate of pressure rise, and hence noise.

A well-optimised combustion system is the result of painstaking attention to detail. Among such details is the design of the injector nozzle. The design variables are : number of holes per injector ; hole length and diameter ; sac volume ; cone angle of the spray pattern ; and the direction of sprays if the chamber is non-axisymmetric. Thus, finding the optimum combination for any given design requirement would involve a development exercise on a scale well beyond the scope of this project. As described in Chapter 4, three nozzle designs were specified for the programme, the only difference between them being the hole size. The spray angle of the four hole pattern was 140°, compared with the "LE" nozzle which is 150°. This was intended to reduce the likelihood of spraying directly into the valve recesses. The spray arrangement relative to the combustion chamber is given below. Figure 6.20 shows the effect of nozzle size on the fuel consumption, smoke and maximum cylinder pressure. The smoke level is poor with all three configurations, probably because of the disturbing influence of the valve recesses as discussed earlier.



However, the largest nozzle (4 x 0.42mm) gives an inferior smoke characteristic with little between the smaller nozzles (4 x 0.39mm and 4 x 0.40mm).

At part load, the smallest nozzle gives the better fuel consumption but is tending to be less efficient at full load. The maximum cylinder pressure is unchanged by the nozzle design, within experimental accuracy over the load range considered. Although the delay period, over the range of nozzles, was not measured, it is unlikely that it was greatly affected since that would have caused the peak cylinder pressures to vary similarly. It is common experience that the spray mean droplet size increases with hole size ; it is also likely that the penetration of the spray will increase, since the spray momentum increases as the square of nozzle diameter, whilst the circumferential area subjected to friction increases linearly (60). The inferior smoke characteristic of the larger nozzle could, therefore, have been caused by the low surface area of the fuel giving poor air utilisation, or perhaps the increased penetration allowing more fuel to enter the valve recesses.

Figure 6.21 gives the fuel consumption, exhaust smoke and maximum cylinder pressure for a range of injection timings plotted against bmep. This shows that at low compression ratio (11.6) and up to a bmep of 21 bar, the performance trends remain the same as a conventional design. Smoke is reduced by timing advance, as is fuel consumption, but at the expense of maximum cylinder pressure. Figure 6.22 gives some of the corresponding transient pressure diagrams to which Figure 6.21 refers. The injection timing is seen to control the pressure rise due to combustion, thus varying the explosion ratio. This effect exercises some control over the gas temperatures achieved during the combustion and expansion phases.

Figure 6.23 gives the results of a limited exhaust emissions survey of the engine, with a compression ratio of 11.6. Nitric oxide (NO) emission is shown to be sensitive to injection timing, improving significantly with timing retard. This is a trend observed in many engines, and results from the gas temperature relationship with timing discussed earlier, NO being formed more readily at high temperatures. Benson and Whitehouse (24) give a table of exhaust emission levels "representative" of diesel combustion, which is reproduced below.

Oxides of Nitrogen	1000 to 4000 ppm
Carbon Monoxide	0.1 per cent by volume
Hydrocarbons	300 ppm

When compared with Figure 6.23 it may be seen that the high output engine offers a low NO characteristic, a "normal" CO characteristic but a very poor Hydrocarbon (HC) characteristic below 30 per cent load. The favourable NO emission is probably due to a combination of low compression ratio and a high degree of charge cooling, which combine to lower the peak cycle temperature.

The CO emission is controlled by oxygen availability, and since diesel engines operate with an excess, CO is quickly transformed into Carbon Dioxide (CO_2). By contrast, spark ignition engines have a low, or negative oxygen excess and emit up to 10 per cent, by volume, of CO. Figure 6.23 shows the CO level rising at either end of the load range. At the upper end, the increase is due to a reduction of air-fuel ratio and hence oxygen availability. The relatively high concentration towards no-load cannot be explained similarly since the air-fuel ratio exceeds 50 : 1.

The HC characteristic demonstrates unequivocally the marginal combustion conditions of low compression ratio engines under light load. Below a bmep of 6 bar, the HC emission increases rapidly because of the low compression temperature and incomplete combustion. This coincides with the emission of white smoke as droplets of fuel condense in the cold air at exhaust outlet. This effect is particularly severe on light load immediately after starting from cold, as shown in Figure 6.33. This shows the HC level decreasing rapidly for the first ten minutes as the engine "warms up". However, after twenty minutes it is still decreasing as the structure becomes "soaked" and nears thermal equilibrium.

Under steady-state operation at a bmep of 1 bar, the application of back pressure, using an adjustable flap valve in the exhaust duct decreased HC from 1950 to 1620 ppmC, whilst the pressure at the exhaust valve was increased from 1.18 to 1.25 bar. This reduces air flow and raises the gas temperature in the cylinder by retaining products from the previous cycle. However, it is to some extent self-defeating since the fuelling must be increased to maintain engine speed against greater pumping losses.

6.5. Energy Flow and Thermal Load

Of the total energy supplied to a conventional diesel engine, only about 40 per cent is transformed into useful work ; the remainder is largely rejected as heat. The fraction rejected to the coolant is of considerable interest since this determines the size and cost of the cooling provisions.

Many researchers have found that the heat rejected to the coolant per unit output decreases as the engine specific output is increased. Thus, a naturally aspirated engine will require a larger heat exchanger than a highly rated turbocharged engine developing the same power. This effect may be seen in Figure 6.24 which shows the energy distribution of the research engine, with compression ratios of 11.6 and 10.7. Whilst the bmep is increased from 4 bar to 21 bar, the proportion of heat dissipated by the coolant decreases from 30 to 15 per cent. Comparing the energy distribution at the present-day bmep norm of 14 bar with that at 21 bar, the coolant handles 18 per cent and 15 per cent of the total respectively. The after-cooler dissipates 7 per cent at 14 bar and 10 per cent at 21 bar. Effectively, the total cooling requirement has remained the same at 25 per cent, but the relative proportions taken by the main cooling system and the after cooler have changed.

These data suggest that the energy distribution is not highly sensitive to compression ratio. However, in this case it should be noted that a measured reduction of compression ratio of 0.9 failed to lower the maximum cylinder pressure. The compression pressure responded according to the polytropic relationship for compression and was reduced by some 10 bar. The combustion process proved highly sensitive to compression ratio over this relatively small range. The ignition delay period increased from 7.8 degrees at 11.6 to 9.3 degrees at 10.7, both at a bmep of 21 bar. As a direct result, the rate of pressure rise at 10.7 was 8.3 bar/deg, whereas at 11.6 it was 6.8 bar/deg. This gave the same peak cycle pressure in each case although they were arrived at by different means. Therefore, since heat transfer from the trapped gases depends on pressure, temperature and air motion, none of which changed significantly, it is not remarkable that the energy distribution remained unchanged. The cylinder pressure diagrams for 10.7 and 11.6 compression ratios are compared in Figure 6.25.

In Chapter 2, thermal load was described as the absolute temperature or temperature gradients present within a component, since these primarily determine the distress due to thermal effects. High temperature generally leads to loss of strength and expansion whilst gradients produce internal thermal stresses. Chapter 5 described the methods of metal temperature measurement used in the experimental programme. The piston, cylinder head and liner top were studied using Shell "Templugs" ; the valves were analysed using the hardness relaxation method and the flamedeck was fitted with fixed thermocouple junctions, mainly around the exhaust valve seat. Figure 6.26 gives a set of results from the thermocouples located in the flamedeck. These recorded the temperatures at 3.8mm from the gas face, therefore they do not represent the highest temperature in the flamedeck at each location. However, they do provide an insight into the variation of metal temperature with load and location. Not unexpectedly, the temperature rises with load, although the curve is thought to be modified by the coolant flow rate, particularly at full load where a fault caused a reduction of flow, and the temperature is seen to rise quite sharply. This effect would be expected, since coolant flow rate influences velocity, and that, in turn, controls the waterside heat transfer coefficient. The coolant enters the cylinder head through vertical drillings from the crankcase ; these connect with horizontal drillings which act as jets, projecting the coolant between the ports at high velocity. Probably as a result of this directed jet of coolant, the temperature between the valves was not the highest recorded. The hottest point of those recorded by either thermocouple or templug was, significantly, adjacent to the exhaust valve insert on the side opposite the coolant jet.

Templugs record the temperature at the end of the flat-bottomed hole into which they are screwed. Since these are 4mm long and mounted flush with the surface, they recorded the temperature at almost the same depth as the thermocouples. By combining the data from these sources, a more comprehensive picture of the flamedeck temperature distribution may be built up. Temperature measurements at bmep's of 18, 20 and 21 bar were normalised by division by the temperature in ^{o}C at a point common to both Templug and thermocouple sets. The ratios so obtained were then combined to produce Figure 6.27.

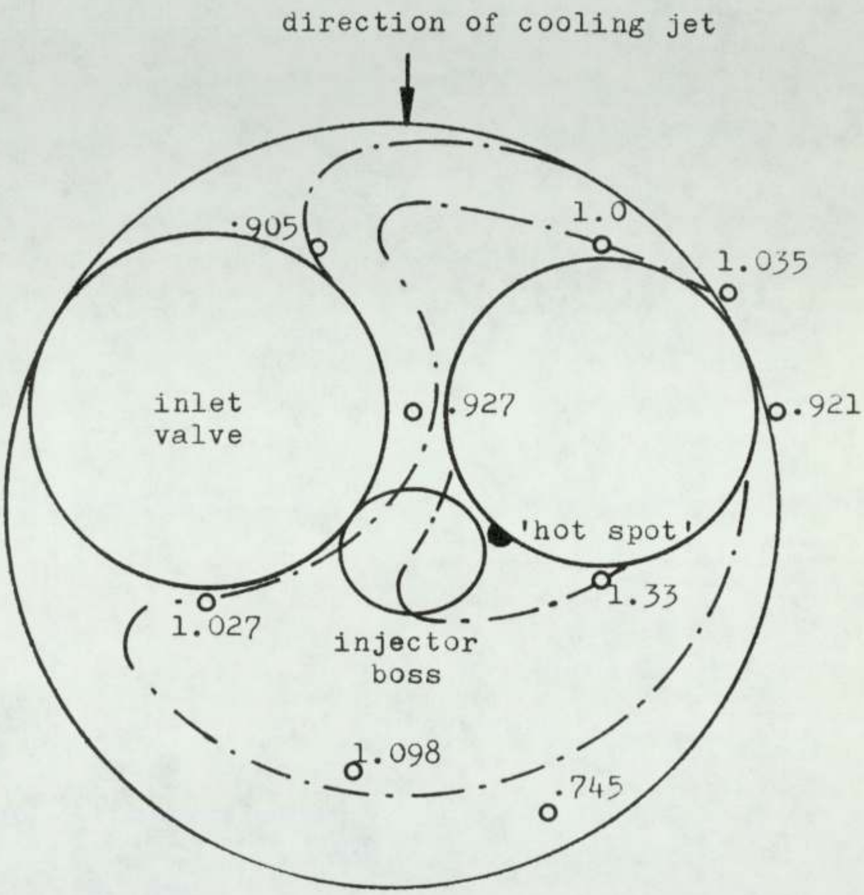


Figure 6.27 : Estimated normalised isothermal map of the cylinder head

The isothermals of Figure 6.27 are based on supposition and represent a possible map based on the point measurements. The most obvious conclusion is that the side opposite the coolant jet is appreciably the hotter. Furthermore, the exhaust port appears to act as a heat source whilst the inlet acts as a heat sink. This effect does, unfortunately, lead to sharp temperature gradients across the valve bridge. It is likely that the "hot spot" is between the exhaust port and the injector boss, an area of high metal thickness.

Figures 6.28 to 6.30 give the results of the thermal surveys at bmep's of 18, 20 and 21 bar respectively. Figures 6.29 and 6.30 are the average of the results from cylinders 2 and 5, whereas Figure 6.28 is from cylinder 5 only. These all show the temperature at the critical area of the head to be above $300^{o}C$ and, at 21 bar, as high as $360^{o}C$. Remembering that this is not the highest temperature at that location, and that it may well not be the hottest location, the cylinder head in its present form would appear to be in some distress in this area. The cylinder head is cast from Grade 17 cast iron, which exhibits a severe loss of strength above $400^{o}C$.

Piston temperatures were assessed using a number of templugs inserted along three radial sections, as shown in Figures 6.28 to 6.30. Three thermal surveys were carried out and the salient details of both design and performance are given in Table 6.2. The first exercise was at a bmep of 18 bar, since that was the maximum attainable load at that time. The second exercise was at a bmep of 20 bar - restricted below full load because of difficulties with the joint between manifold and turbine. The third exercise was at the target bmep of 21 bar.

The temperatures recorded at all three loads are acceptable in several respects. Firstly, the accepted maximum allowable temperature for the material (Aluminium 52-Alloy) is $350^{o}C$. Only one templug

Table 6.2. : Details of design and performance of the thermal survey builds

Engine speed rev/min	1500	1500	1500
bmep bar	17.95	19.73	21.01
Compression ratio	11.90	10.70	10.70
Charge pressure bar	2.88	3.13	3.14
Maximum pressure	128	120	123
Explosion ratio	1.57	1.563	1.597
Overall air/fuel ratio	27.92	27.89	26.80
Volumetric efficiency (percentage)	101	99.8	103
bsfc g/kWh	226.1	224.5	228
Piston cooling jet temp.k	356	356	354
Cooling jet flow (l/s/piston)	0.189	0.163	0.189
Jet velocity/max piston velocity	1.275	1.10	1.275
Fuel pump/plunger diameter mm	Maximec/12	Maximec/12	Maximec/12
Injector nozzle	4x0.39mm	4x0.39mm	4x0.39mm
Turbocharger	TV71	TV61	TV61
Compressor	0 - 1	W - 4	W - 4
Turbine	1.08E	1.00G	1.15G

recorded in excess of 300°C, that was 335°C at the bowl edge at a bmep of 18 bar. At full load, the highest recorded temperature was 278°C at the bowl edge of piston 2. Secondly, the lubrication conditions rely on a temperature at the top ring of between 180°C and 240°C. All 13 measurements above the top ring were between 202°C and 232°C. At the surface above the top ring the temperature would be slightly less than this. Finally, the strength of the material deteriorates at temperatures above 180°C and accordingly, the gudgeon pin boss should be kept below this limit. Although templugs were not used in this area, the second land temperature was consistently between 190°C and 200°C, therefore the pin boss would have been safely below 180°C. Figure 6.31 shows a piston section with isothermals "faired in", as was done for the cylinder head.

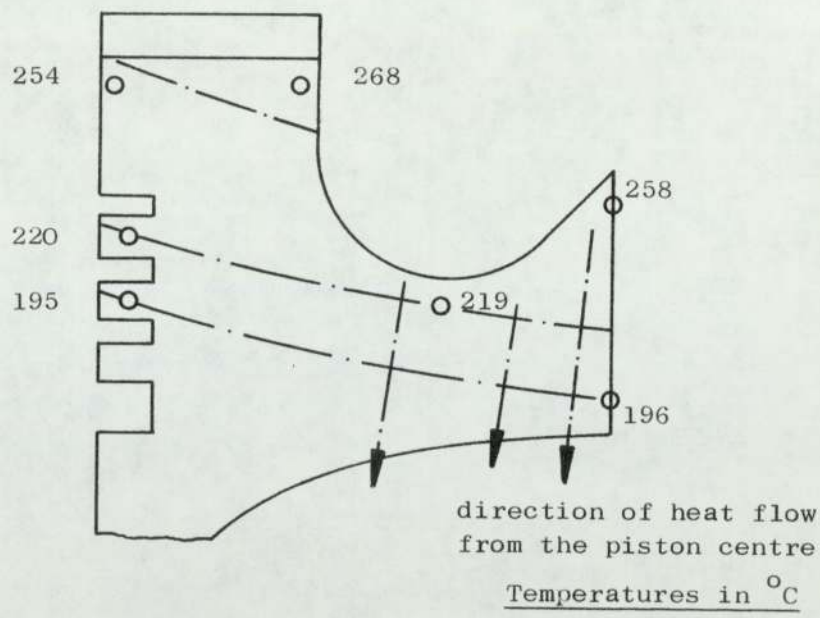


Figure 6.31. : Section through the piston, showing measured temperatures and the estimated isothermal map at a bmep of 21 bar.

This shows that the heat flow from the centre of the piston was to the undercrown oil jet.

The liner bore, piston exterior design and ring pack used throughout this work were exactly as used in the "LE" turbocharged and after-cooled engine in current production. The parts were of production quality and were not selectively fitted. At each piston change, the used sets were examined by the maker's technical representatives. One set, after 50 hours of operation at 75 per cent load and above, were showing signs of scarring and tightness around the top land. Otherwise, the pistons were sound. The measured oil consumption - with jet cooling - was near to 0.5 per cent of fuel consumption ; normal for high speed engines.

The results of the thermal surveys of the valves are shown in Figures 6.28 to 6.30. The temperatures were measured using thermometric valves of hardened steel alloy (EN52 and EN18), as described in Section 5.4. Figure 6.28 gives results from cylinder 5, as does Figure 6.29. Figure 6.30 gives the result from cylinder 2, since it was the hotter, in that case.

The most obvious conclusion from these results is that none of the valves rotated freely during the tests. This is undesirable for the valve head, since the thermal loading is effectively increased on one side. Valve manufacturers, TRW Ltd., who supply Dorman valves, and who analysed these experimental valves, stated that the absolute temperatures of the valves were "fairly low for a diesel engine." Whilst this is encouraging, the other determinant of thermal load, temperature gradient, gives cause for concern. A measure of temperature gradient is the differential between the seat face and the highest recorded temperature. At all three loads, this exceeded 200^{o}C. This probably contributed to some distortion of the valve head, which took the form of "cupping". This measured between .025 and .076mm (.001 - .003").

Perhaps the most serious problem is valve stem lubrication. The problem here is the large pressure differential that exists along the valve guide, caused by the relatively high inlet and exhaust manifold pressure necessary for high output. Lubricating oil is discouraged from entering the guide, and conditions are made worse on the exhaust side by the influx of carbon at the lower end. This effect was seen on the thermometric valves as slight scoring of the stem, and carbon packing. This is not a new problem, but it is one that will have to be solved if satisfactory component life is to be achieved.

6.6. Starting with Low Compression Ratio

It is inevitable when highly turbocharging a conventional engine that the compression ratio must be lowered to contain the mechanical loading. During this programme, three 'low' ratios were used : 10.7, 11.6 and 11.9. These are below the marginal compression ratio for unaided starting of this size of cylinder, which is about 13.5 : 1.

A formal study of startability and an evaluation of the many aids and techniques available could not be undertaken because of time constraints. However, the opportunity was taken to collect a limited amount of data during routine starts for the experimental programme. Although these data are not structured for "scientific" analysis, they nonetheless provide an insight into the problem. Several aids were used at different times, essentially to give a rapid and clean start for experimental work.

Tables 6.3 and 6.4 present the results obtained with compression ratios of 10.7 and 11.6 respectively. A conventional high-speed, direct injection engine will start reliably, cleanly, and achieve operating speed in 3 to 5 seconds after the commencement of cranking. The results obtained from this programme, with low compression ratios, demonstrate the need for assistance even at ambient temperatures of 20^{o}C. The most rapid start achieved by the 10.7 build was 7 seconds from rest to 1500 rev/min, and that was with block heating to 46^{o}C. Increasing the compression ratio to 11.6 did not reduce the minimum start time, but gave a more consistent start within the range 8 to 22 seconds, using a high level of block heating. The best results were achieved by using Ethyl Ether fumigation. This improved the time to the range 3 to 10 seconds.

The use of CAV thermostart, a fuel-burning manifold heater, proved ineffective as installed. The performance of this device is known to be sensitive to positioning in the manifold : firstly, to provide even distribution to all cylinders, and secondly, to be in active air but not be extinguished. It is possible that if positioned differently, it could function satisfactorily. The positioning of the burners in the manifold is shown in Figure 6.32(b).

Restricting the exhaust pipe to create back pressure was tried, but gave inconclusive results. The hope was that air flow would be reduced and hot products, once formed, would remain to heat the next charge.

Table 6.3. : Data from routine starts during development programme

Compression ratio : 10.7 : 1

Manifold temperature (^{o}C)	Jacket temperature (^{o}C)	Fuelling	Special aids	Static injection (^{o}btdc)	Time to 1500 rev/min (S)
21	24	Excess	a	22	no start
21	40	"	b	22	66
21	28	"	abc	22	18.5
21	42	"	b	22	12.0
21	46	"	b	22	7.0
21	32	"	bc	22	71.0
22	31	"	bc	22	71.0
19	26	"	abc	22	no start
17	54	"	b	14	13.6
17	52	"	b	14	11.8
18	52	"	b	14	10.0
20	44	"	b	24	59.0

a CAV "Thermostart" fuel-burning manifold heater

b Jacket heater

c Back pressure restricted

Table 6.4. : Data from routine starts during experimental programme

Compression ratio : 11.6 : 1

Manifold temperature (^{o}C)	Flamedeck temperature (^{o}C)	Fuelling	Special aids	Static injection (^{o}btdc)	First fire /1500 rev/ min (S)
25	35	Excess	b	22	20/-
19	55	"	b	22	1/8
20	41	"	b	18	1/22
16	38	"	b	26	1/15
10	32	"	b	26	2/14
15	53	"	b	20	2/14
19	43	"	b	22	1/14
26	21	"	d	20	.25/3
		13mm plungers			
24	12	Excess	d	20	2/7
31	21	"	d	20	1.7/5
28	16	"	d	20	2/10
10	37	"	bd	20	2/7
8	16	"	d	20	.5/6

b Jacket heater

d Start Pilot Ether system

Ethyl Ether has been used for many years as an aid to the cold starting of conventional engines. As a result, there are several proprietary systems on the market. Two such systems were tried. These were the series 450 made by Start Pilot Ltd., and the measured shot system made by KBI Diesel Start of the USA.

These two systems represent contrasting methods of delivering ether to the manifold. The Start Pilot system is well-engineered, robust and the more expensive by a factor of 3. It features a battery-driven, positive displacement compressor which forces air through a mixing valve above an ether reservoir to form an ether/air mixture. This enters the manifold through relatively large bore (.5mm) nozzles. Control is effected by the size of the mixing valve orifice and nozzle diameter. The KBI system is lightly constructed to facilitate cheap, high volume production for the North American truck market. It dispenses with the electrical compressor, instead using a consumable, pressurised ether reservoir. This is tapped by a battery-powered solenoid valve which allows a chamber to fill with ether. The solenoid then closes automatically. The chamber is connected to small bore (.05mm) nozzles in the manifold through which ether flows under a steadily decaying pressure head. Thus, the ether flow rate varies over time. The delivered quantity or "measured shot" may be controlled by the volume of the chamber.

In practice, the Start Pilot system functioned well, whereas the KBI system tended to malfunction and failed to start the engine when working. With the Start Pilot system, it was possible to run the engine at up to 590 rev/min without fuel injection. Pressure diagrams from cylinder 2 under this mode of operation are shown in Figure 6.34. The ether is seen to ignite at a pressure of 10 to 12 bar, which corresponds to a gas temperature of only $200^{o}C$ to $240^{o}C$.

Ether has a mixed reputation amongst engine users. Applied carelessly to the air intake by proprietary aerosol spray, it may cause drastic damage to pistons, or even connecting rods. Used in a controlled manner, it provides a rapid start in very adverse conditions. The maximum cylinder pressure, without fuelling, may be seen from Figure 6.34 to be approximately 33 per cent of the allowable maximum (120 bar). With fuel, this was increased to 40 to 55 per cent. The location of the spray nozzles in the manifold wall is given in Figure 6.32 (a).

6.7. Discussion of the application of predictive models

The use of computer based predictive techniques prior to the experimental programme undoubtedly consumed valuable time. It may also be said to have saved considerable time and money by allowing efficient use of the experimental facilities. It is less expensive to design an engine on a drawing board, so far as possible, than to proceed by trial and error on a development test bed. The design study, and in particular, the "filling and emptying" and thermal analogue models, served both to elucidate and solve design problems.

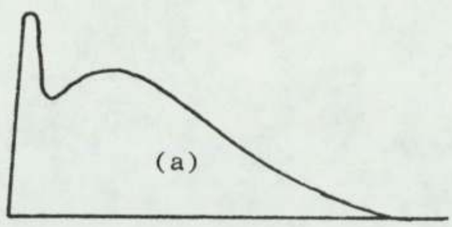
The extent to which these models accurately predicted the performance was rather limited, in fact signficant differences are immediately obvious in the area of volumetric efficiency. Despite these discrepancies, the engine specification resulting from the design study did achieve the required specific output, except for a re-specified compressor wheel and housing. The successful application of such methods can only result from continued use, continued feedback, and learning by experience. The remainder of this chapter will be devoted to discussion of the predicted and experimental performance, and some notes on the progress made since the availability of more comprehensive experimental data.

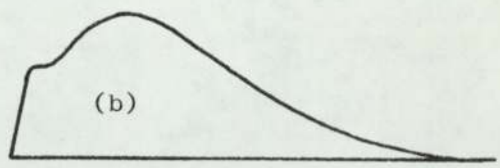
Of the assumptions made when modelling the high output engine during the design study, the heat release diagram was probably the most uncertain. It was based, optimistically, on a Wiebe function found to fit existing data from the "LE" turbocharged and after-cooled engine. This data consisted of maximum cylinder pressure, thermal efficiency and exhaust temperature ; pressure diagrams were not available. Having obtained a reasonable fit, the shape was maintained for the high output study, whilst the duration was shortened to account, in a crude way, for the higher air-fuel ratio. This approach is rather arbitrary, but for some aspects of the study, the heat release function is relatively unimportant. Conversely, for the calculation of thermal loading, the heat release assumption is most important since it controls the development of pressure and hence the mean heat transfer coefficient, and mean gas temperature. A comparison of the Wiebe function, used throughout the design study, and a heat release curve derived from experimental results at a bmep of 21 bar, is given in Figure 6.35(a). The corresponding pressure diagrams are given in Figure 6.35(b). The compression ratios are not the same in each case (11.6 and 11.7) because of manufacturing tolerances.

These show that the heat release diagram used for the study was realistic, since it produced a pressure curve of the approximate shape and magnitude achieved experimentally. This was, however, fortuitous, and not the result of a reliable or repeatable procedure.

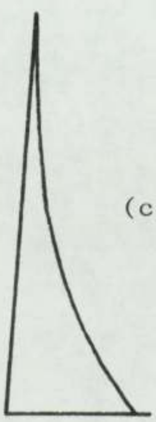
The method of Marzouk and Watson (72) offers a more rational approach to the problem. This model recognises that engines of a given type with normally matched fuel injection equipment, have characteristic heat release patterns which correlate, in broad terms, with certain engine parameters. The Marzouk-Watson model uses ignition delay period, air-fuel ratio and engine speed to derive a two-phase, heat release rate diagram. This provides a diagram shape characteristic of a high-speed, direct injection engine under the prevailing conditions. A range of such diagrams is given below.

Rate of heat release





crank angle



	air-fuel	ignition delay	engine speed
a.	24:1	5^{o}CA	1500 rev/min
b.	27:1	3^{o}CA	1500 rev/min
c.	33:1	14^{o}CA	1500 rev/min

Heat release diagrams derived from the Marzouk-Watson model (ref. 72).

Figure 6.36(a) and (b) compare experimental and calculated cylinder pressure curves. Although the model was established using a conventional "high" compression ratio truck engine, the results at a compression ratio of 10.7 compare well with the experimental data. Ideally, both the ignition delay relationship, the shape and proportionality constants in the model should be evaluated for current Dorman engines over a suitably wide range of speed, load and compression ratio . However, in the short term, the model would appear to give important advantages over the Wiebe function. These are : a two-phase characteristic typical of high-speed engine combustion, relative simplicity, and a realistic response to the main causal factors.

The main discrepancy between predicted and experimental data was an overestimation of volumetric efficiency. The effective areas of inlet and exhaust valves were derived from steady-flow rig experiments on the valve and port combination. This method is widely recommended by the literature and is thought to be a sound assumption. The filling and emptying model used for the design study simplified the induction processes by neglecting the kinetic energy of the fluid flow, and inertial or "ram" effects at inlet valve closure. The ram effect would tend to increase the volumetric efficiency above that predicted, which was not the case. A further simplification, again supported by common practice, was to neglect charge heating effects in the induction passages. This was probably the main cause of the error. The main supporting evidence to this effect is the work by Jones (90) on the development of induction systems for high-speed diesel engines. Figure 6.37(a) is reproduced from this source, which shows that considerable charge heating was present in that engine, resulting in a reduction in volumetric efficiency from 96 to 82 per cent at 1500 rev/min. Although this was a naturally aspirated engine with an unfavourable temperature difference between the charge and the port wall and valve head, it does serve to demonstrate the potential heat pick-up. The extent of heat transfer in the ducting will depend largely on the available temperature difference between gas and metal. Dicksee (91) compares the volumetric efficiency of an engine at full load and motored, as shown in Figure 6.37(b). This shows the significant effect of heat pick-up on volumetric efficiency, in this case the total difference is due to contact with cylinder walls as well as the induction system.

To reduce errors in this area of the model, a volume representing the six ports was inserted between the manifold and the valve junctions. Controlled amounts of heat could be released into this to simulate port heating. For the high output research engine, the heat released into the 'ports', to give correlation with experimental data, was equivalent to 1.85 per cent of the total fuel energy supply to the engine. This value will vary from case to case, and predicting the heating effect at the design stage will require careful consideration.

Figure 6.38 compares the gas exchange diagram predicted at the design stage, with a gas exchange diagram recorded experimentally. Although there are clear differences throughout, the overall similarity provides confirmation of the model, at least for design purposes. The main departures are caused by the simplifying assumption neglecting wave action, especially in the exhaust manifold. In particular, the exhaust pulses decay more rapidly than the model predicts, because in reality, the pulse travels along the pipe as a wave, rather than filling the entire volume, having equal effect in all parts. Since the pulse decays less rapidly, the cylinder pressure has to follow, and both are overestimated towards the end of blowdown. A further difference between the predicted and experimental diagrams is that the model fails to account for pressure waves reflected from other junctions along the pipe, and the turbine. Reflected waves of small magnitude may be seen to increase both exhaust manifold and cylinder pressure between pulses, whilst the model predicts a steady decay. The rise in cylinder pressure after top dead centre is caused by the pulse from the next cylinder exhausting into the pipe. Here, the model predicts a pressure rise of much smaller magnitude. Though the model would appear to be representative enough for design purposes, the possibility of adverse effects due to wave action, especially at low speed or in long pipes, should be considered.

During the design study, the piston temperature field was predicted under the target load conditions, with and without undercrown oil cooling. A computer based thermal analogue network or "lumped parameter" model was used, and the boundary conditions on the gas face were taken from the filling and emptying performance prediction. Since the model was two-dimensional and axisymmetric, neglecting bowl off-set and valve recesses, the predicted isothermal map may be shown by a single section. Figure 6.39 contrasts this prediction with "templug" point temperature measurements at a bmep of 21 bar. The point temperatures are the mean values of all measurements at each location, for pistons 2 and 5. The prediction is generally higher than the experimental results by as much as $30^{\circ}C$ at the bowl edge and down to the top ring. The two distributions are approximately consistent with each other, which tends to confirm the isothermal map and heat flow paths which may be inferred from it. Thus the heat entering the piston centre is extracted by the undercrown cooling oil. From this limited exercise it would appear that the method could be a highly effective design tool.

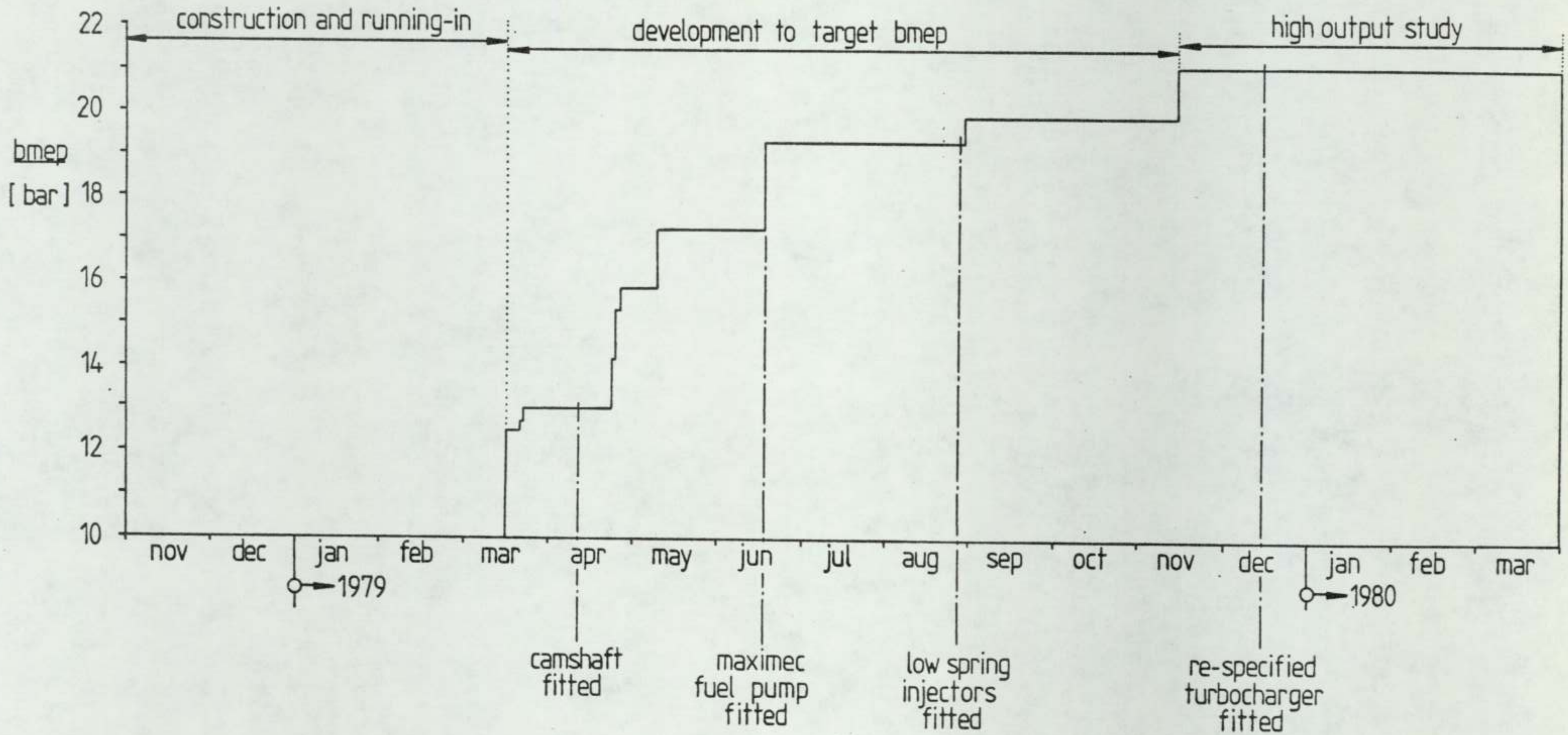


Fig. 6.1, The progress rate was controlled by the availability of components.

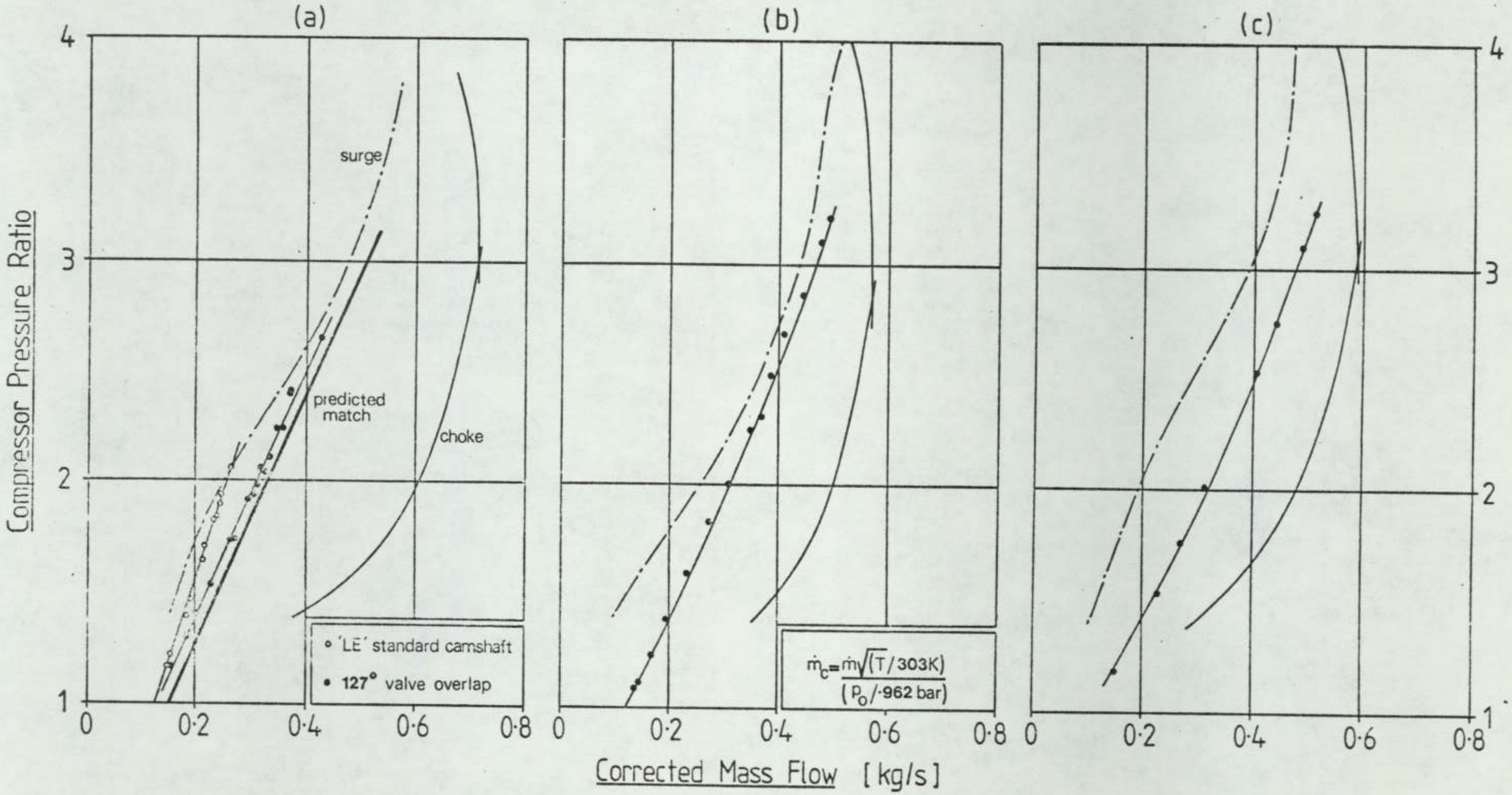


Fig.6.2, Development of the compressor-engine match, (a) originally specified.(b) intermediate measure. (c) re-specified match. All at 1500rev/min.

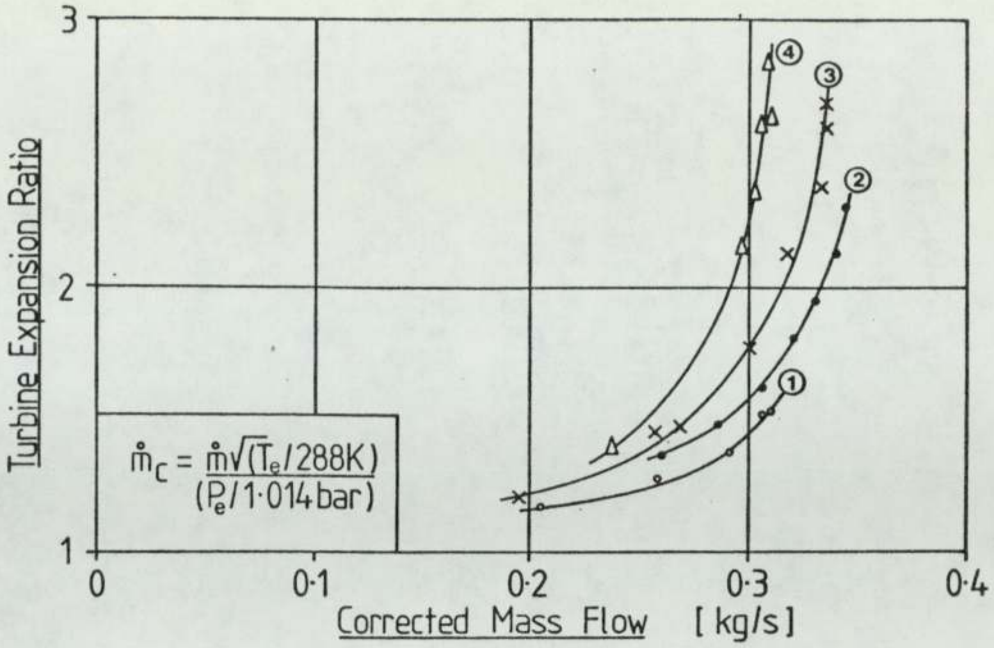


Fig.6.3, Characteristics of the turbine configurations used during development, calculated on quasi-steady basis. Engine speed 1500 rev/min.

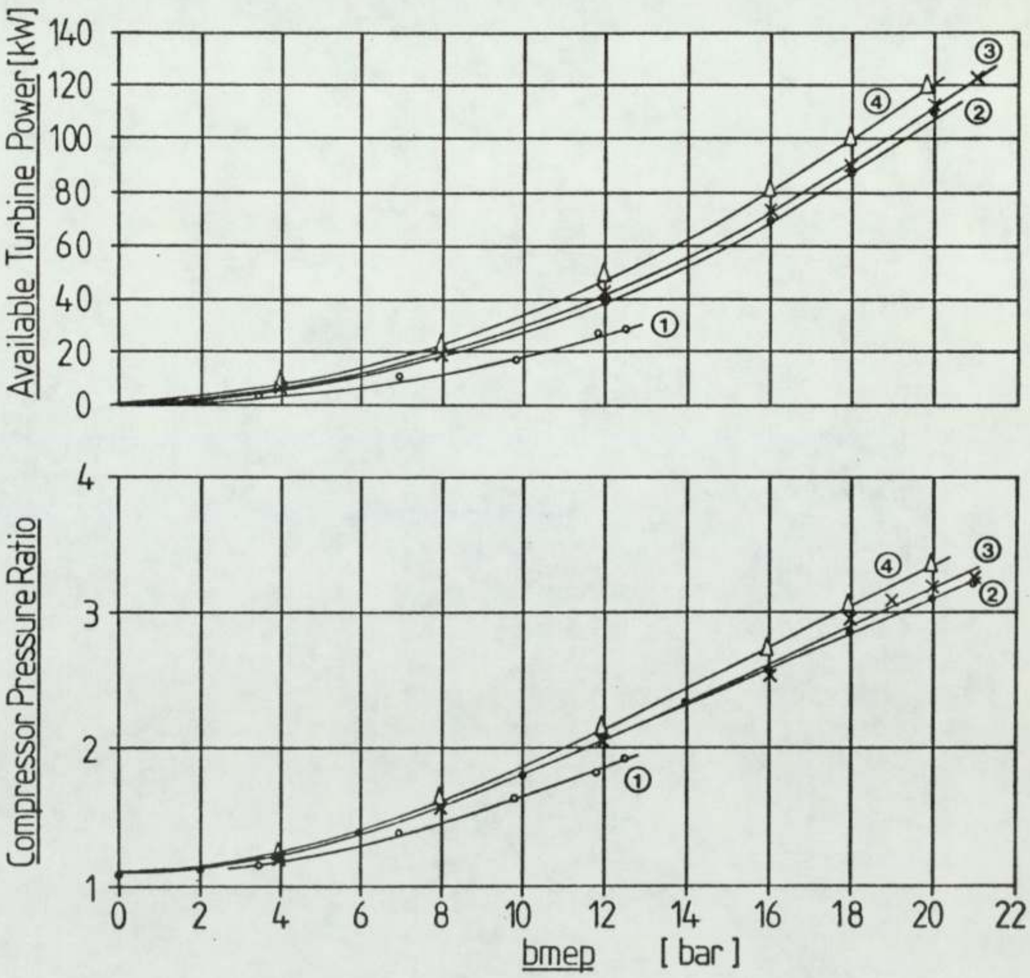
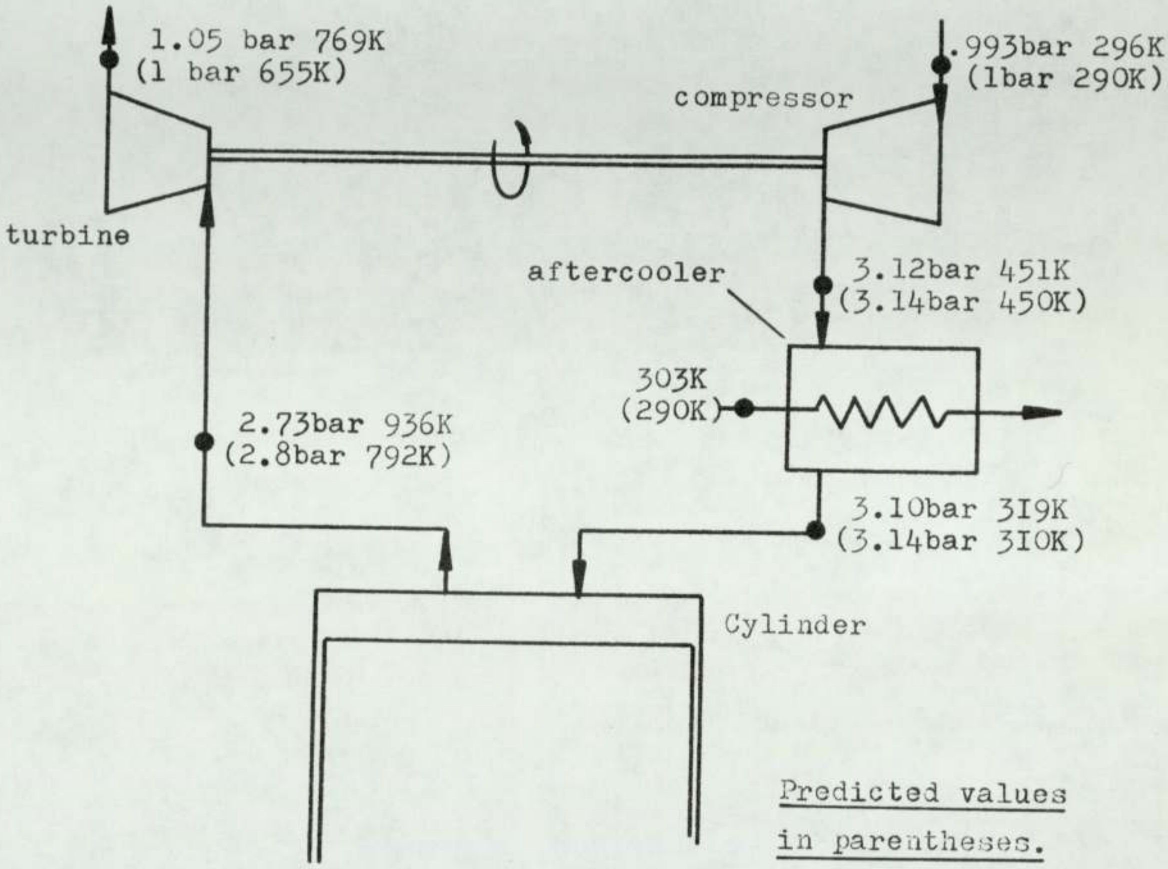


Fig.6.4, Available turbine power and the resulting compressor pressure ratio, for the four configurations of fig.6.3



Air-Fuel Ratio	24 (28)
Compression Ratio	11.6 (11.7)
bmep	21.01 (21.38) bar
Aftercooler Effectiveness	.892 (.900)
Air Flow	.494 (.562) kg/s
Trapped Vol.Efficiency (estimated)	.87 (.99)

Fig.6.5 Comparison of experimental and predicted performance of turbocharging and induction system.

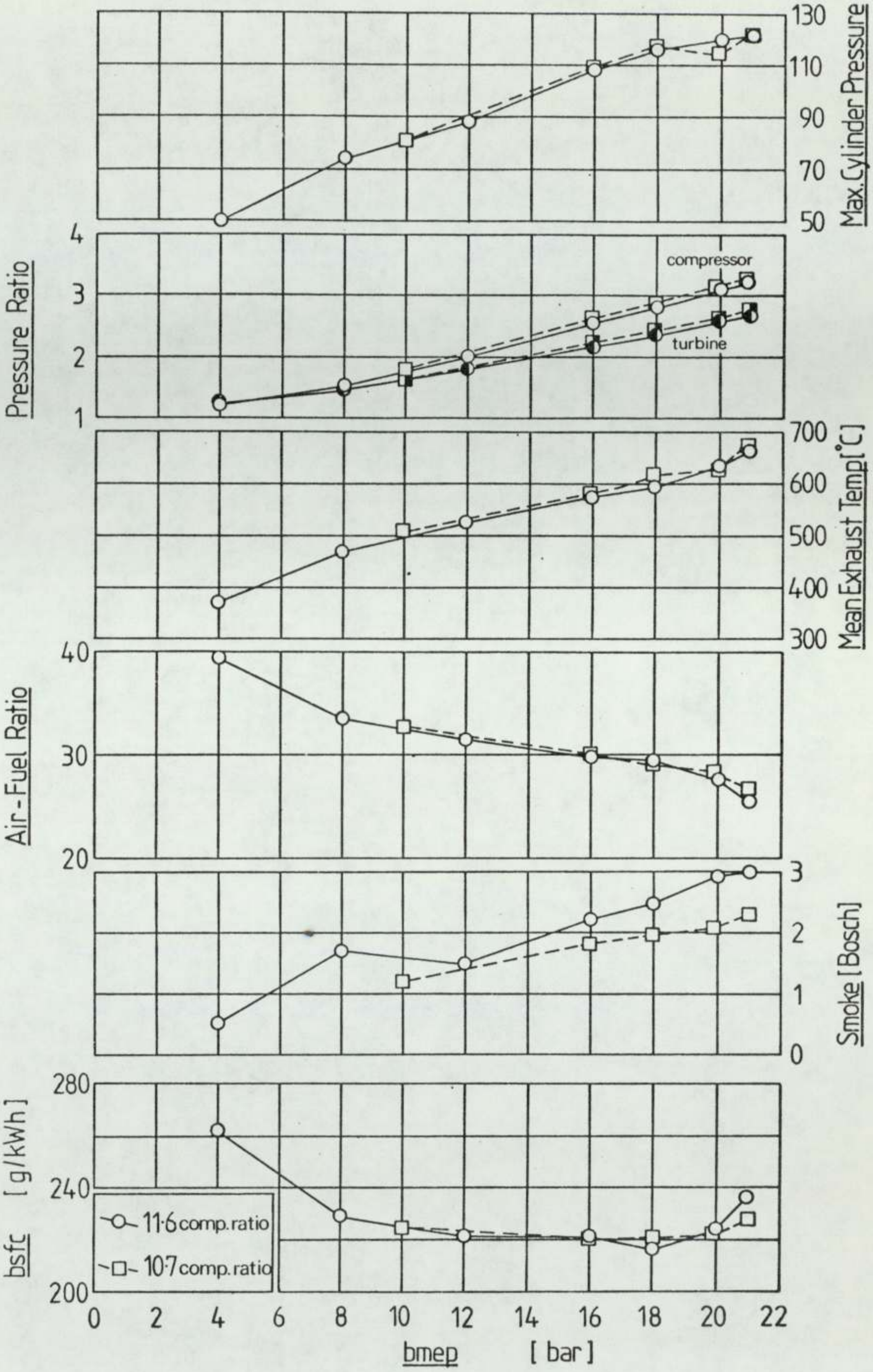


Fig. 6.6, Performance with compression ratios of 10·7 and 11·6.
[builds 12 and 15, spill 22°]

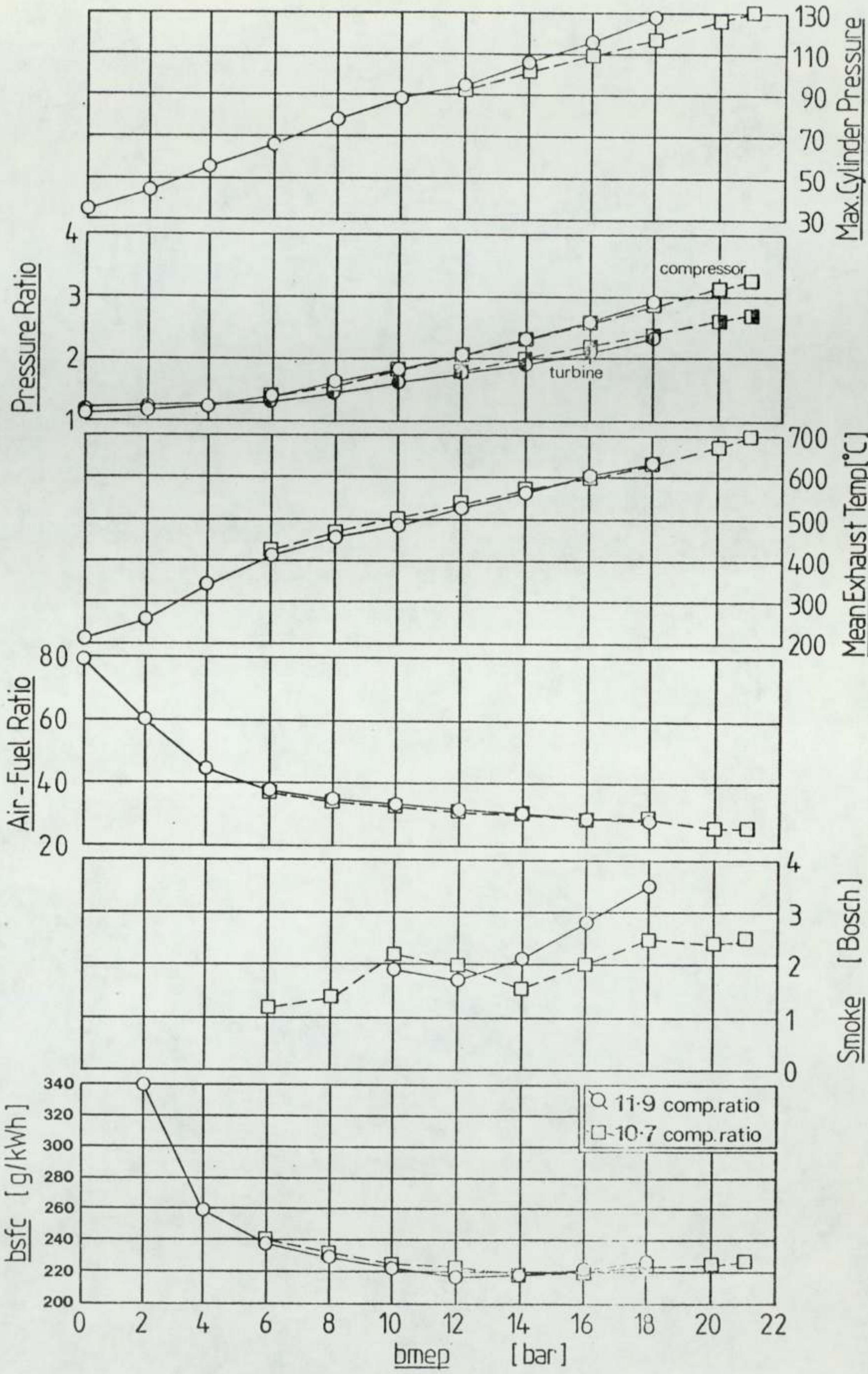


Fig.6.7, Performance with compression ratios of 10·7 and 11·9.

[builds 6 and 9, spill 22°]

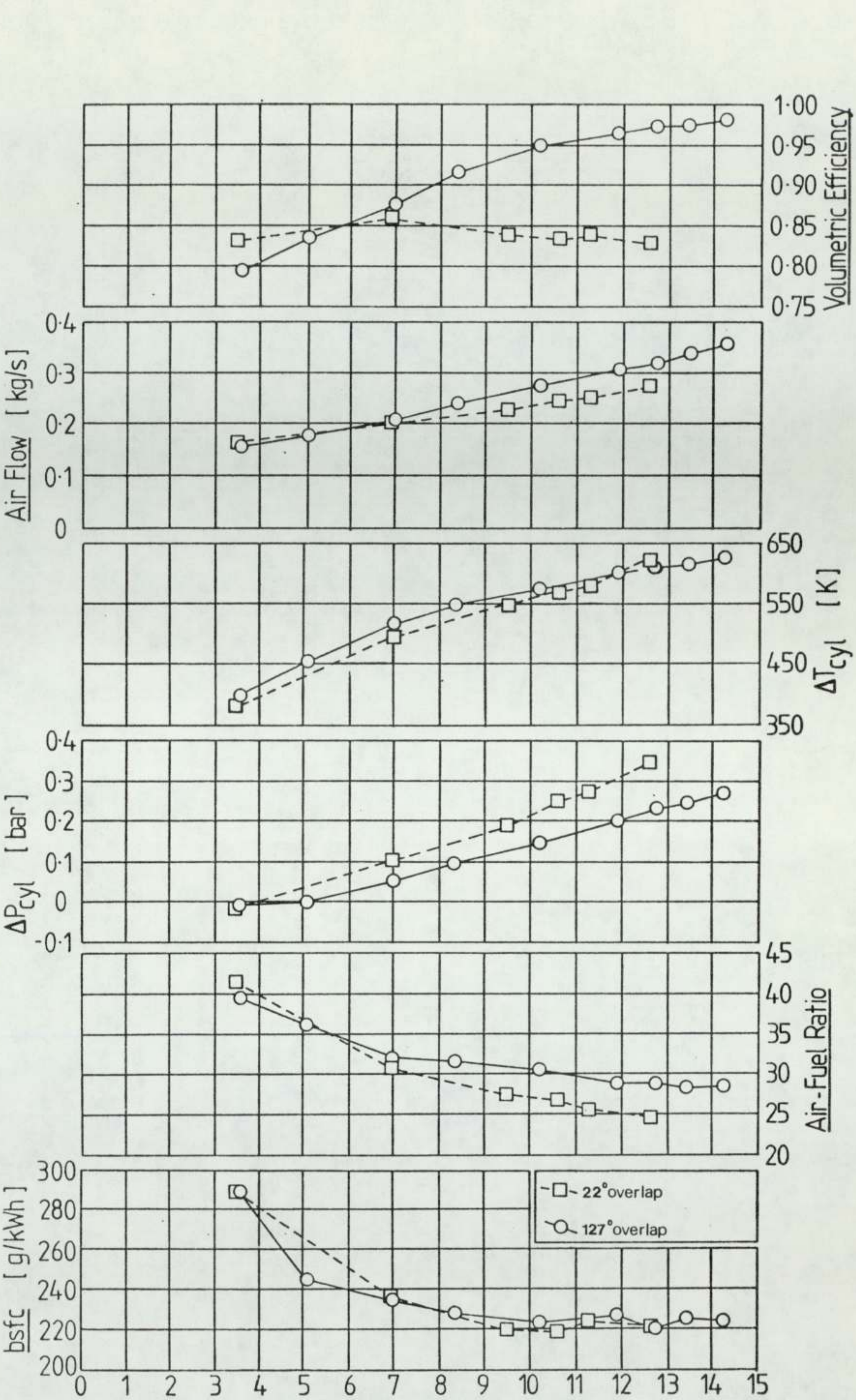


Fig. 6. 8, Effect of camshaft design, particularly valve overlap period, on research engine performance. [builds 2 and 3, spill 28°]

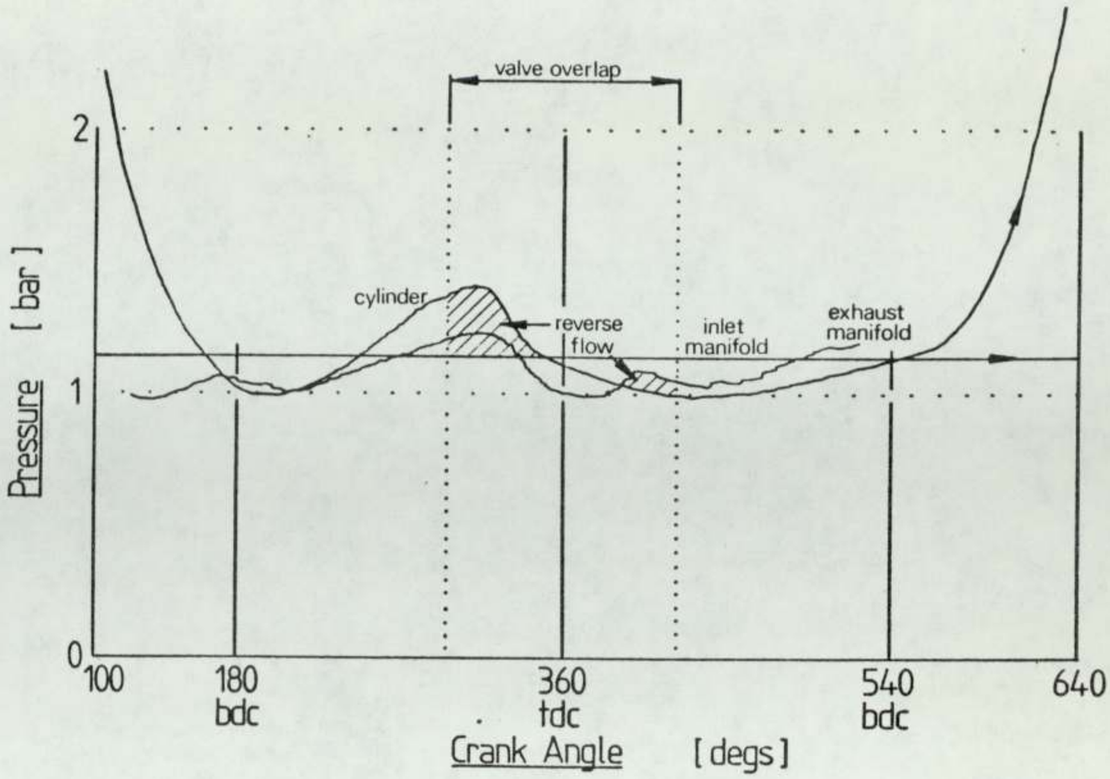


Fig. 6.9, Gas exchange processes at no-load, 1500 rev/min, showing the adverse pressure gradients which cause flow reversal. [build 16]

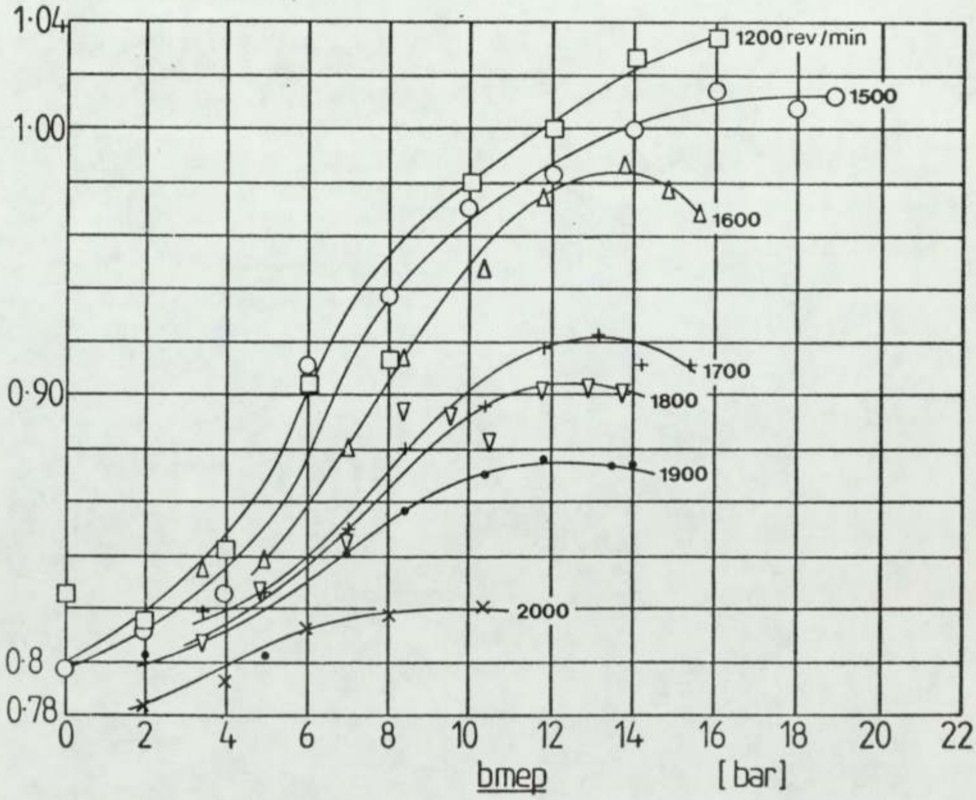


Fig. 6.10, Volumetric efficiency varies with load and engine speed when high valve overlap is used. [builds 3 and 5]

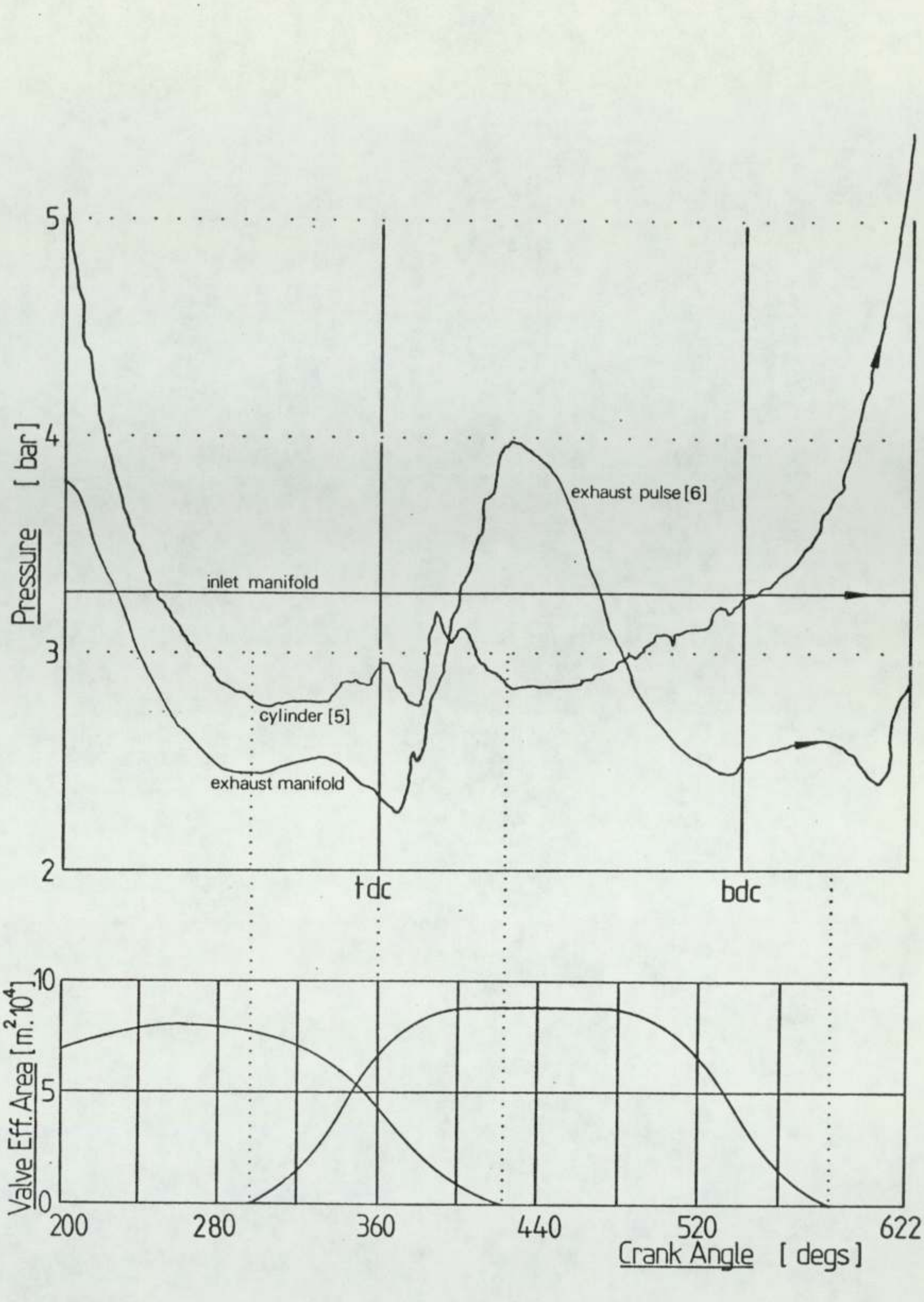
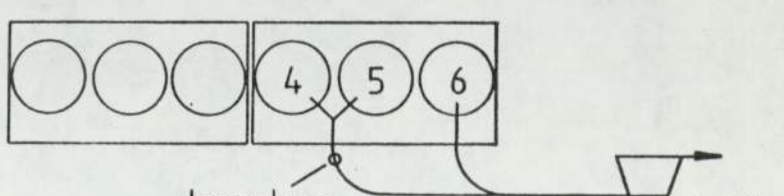


Fig. 6.11, Gas exchange processes at a bmep of 21 bar showing some interference, between cylinders 5 and 6, towards the end of valve overlap. [build 12]



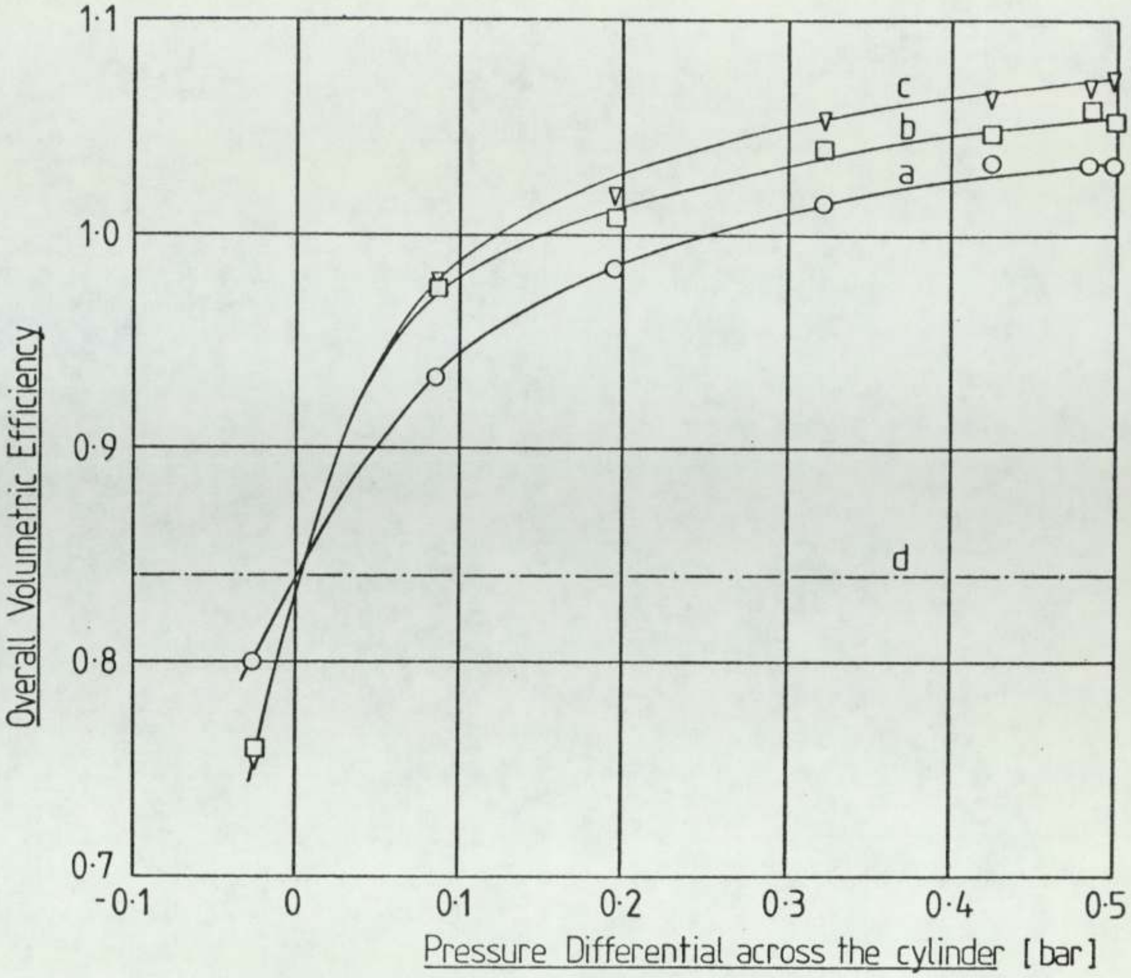


Fig.6.12, Volumetric efficiency plotted against mean static pressure drop across the cylinder for the 127° valve overlap camshaft. a, experimental data. b and c, calculated by Zinners method(3) using compressible and incompressible flow theory respectively. d, volumetric efficiency with 22·5° valve overlap. [build 17]

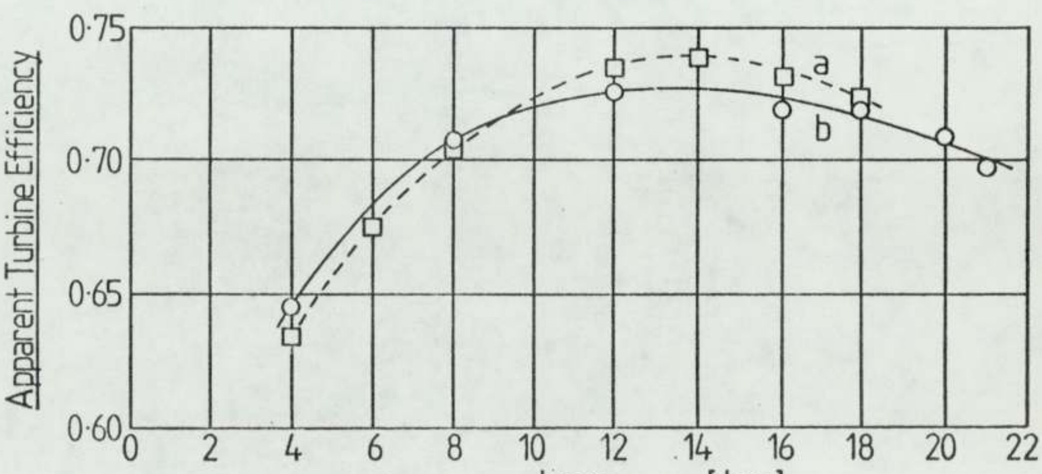
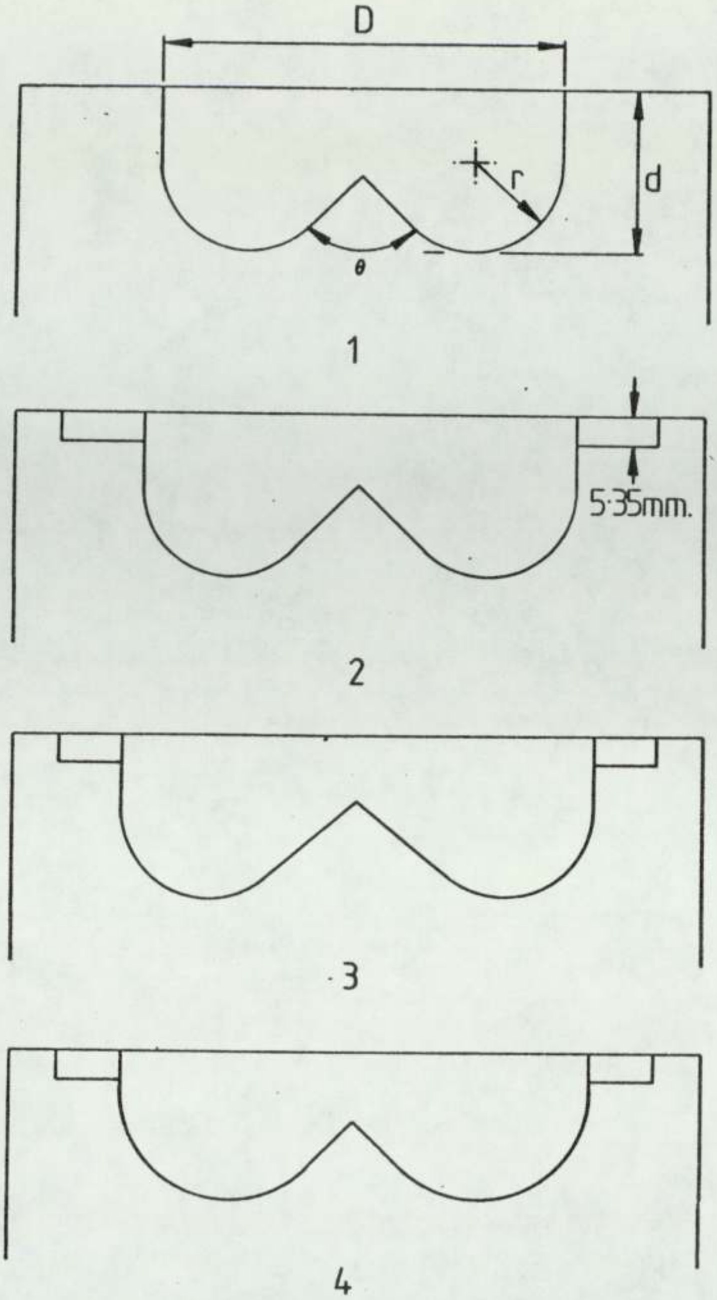


Fig.6.13, Apparent turbine efficiency – load characteristics. a, TV61 'high' boost. b, TV61 'low' boost. [builds 14 and 15]



piston	comp. ratio	D	d	r	θ	V_c^*	V_r^*	V_r %
1	14·5	73·7mm	30·5mm	15·9mm	90°	140	0	0
2	11·9	79·3	"	"	"	173	16·1	9·31
3	10·7	86·9	"	"	"	195	13·5	6·94
4	11·6	86·2	27·3	20·3	100	178	13·7	7·68

* V_c = total clearance volume. [$m^3 10^6$] V_r = volume of recesses in the crown.

Fig. 6.14, Details of the toroidal combustion chambers used during the experimental programme. Design 1 is the current 'LE' chamber.

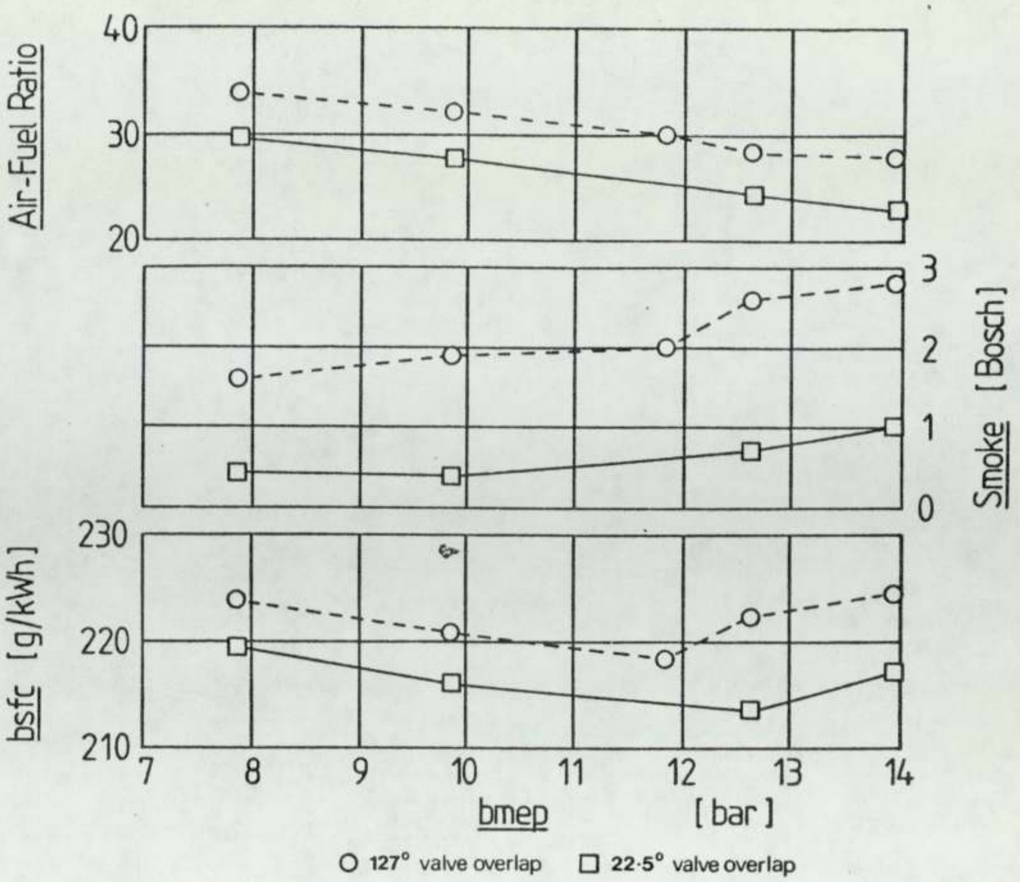


Fig. 6.15, Effect of valve overlap and valve recesses in the piston crown on the performance of the Dorman 'LE' turbocharged engine.

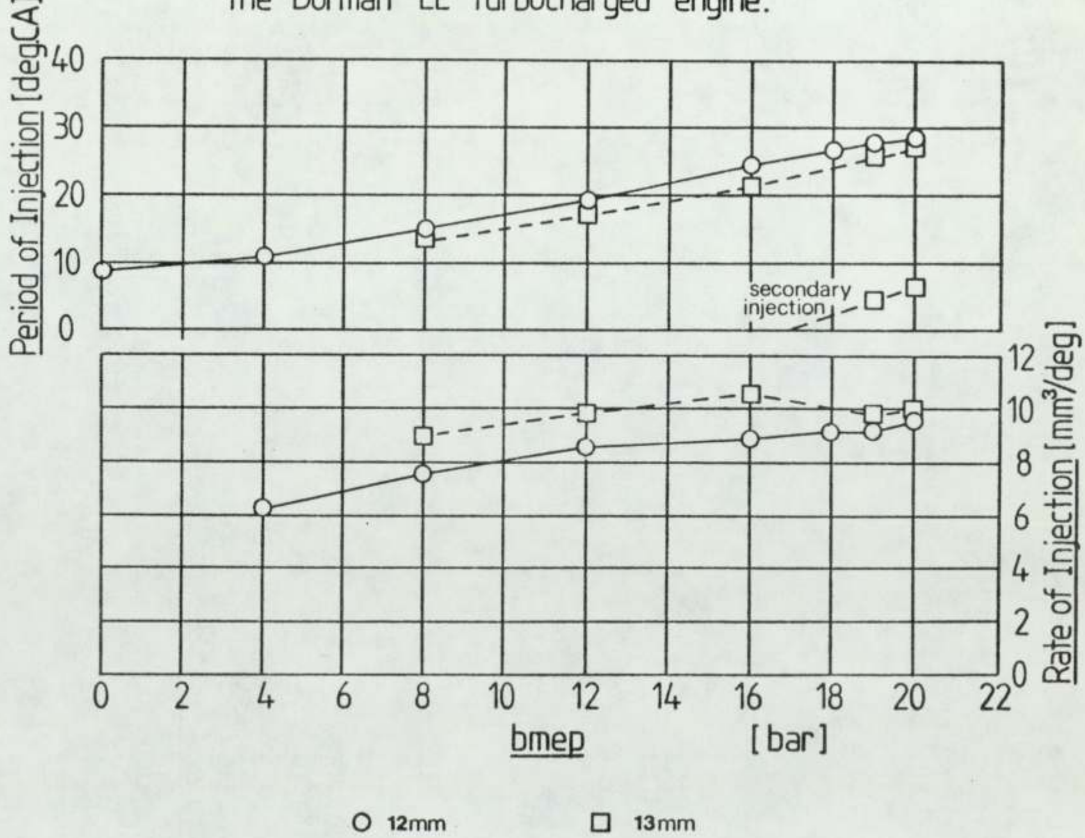


Fig. 6.16, Comparison of the measured injection characteristics of the CAV'Maximec' fuel pump with 12mm and 13mm diameter plungers. [builds 15 and 16]

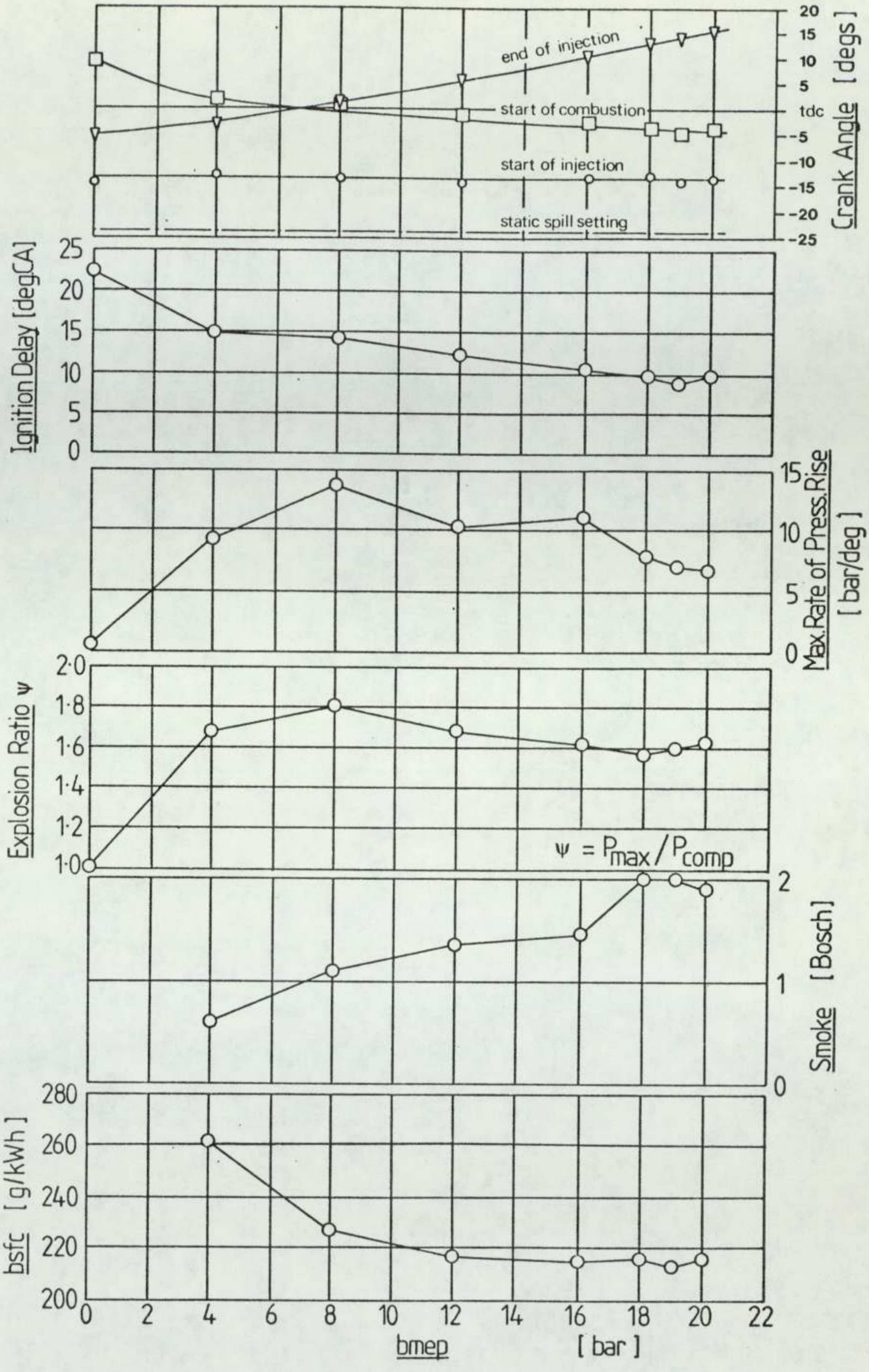
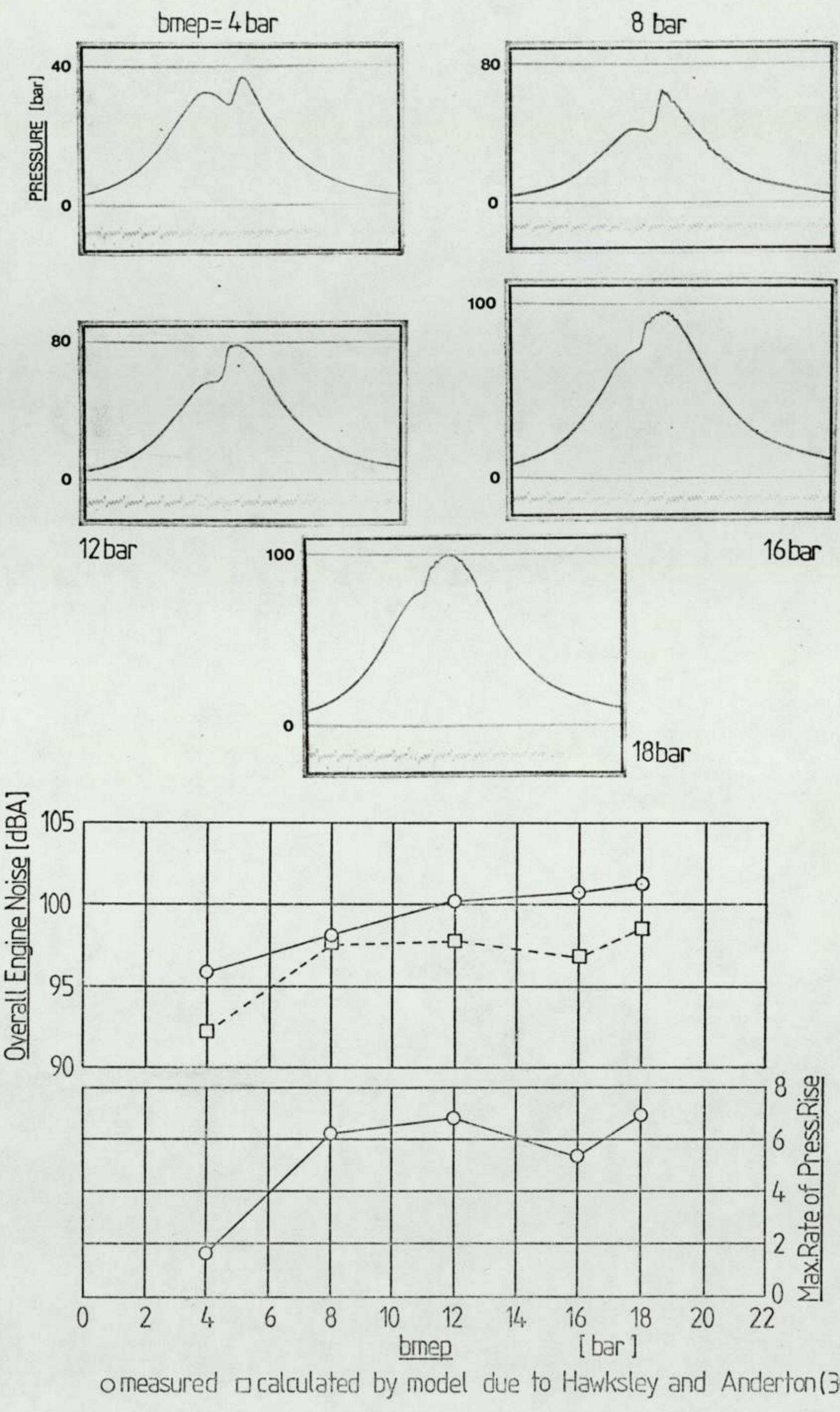


Fig.6.17, Performance of the fuel injection equipment originally specified. [build 15, spill 24°]



o measured □ calculated by model due to Hawksley and Anderton (30)

Fig.6.18 Noise levels with a compression ratio of 11·6. [build 15, spill 20°]

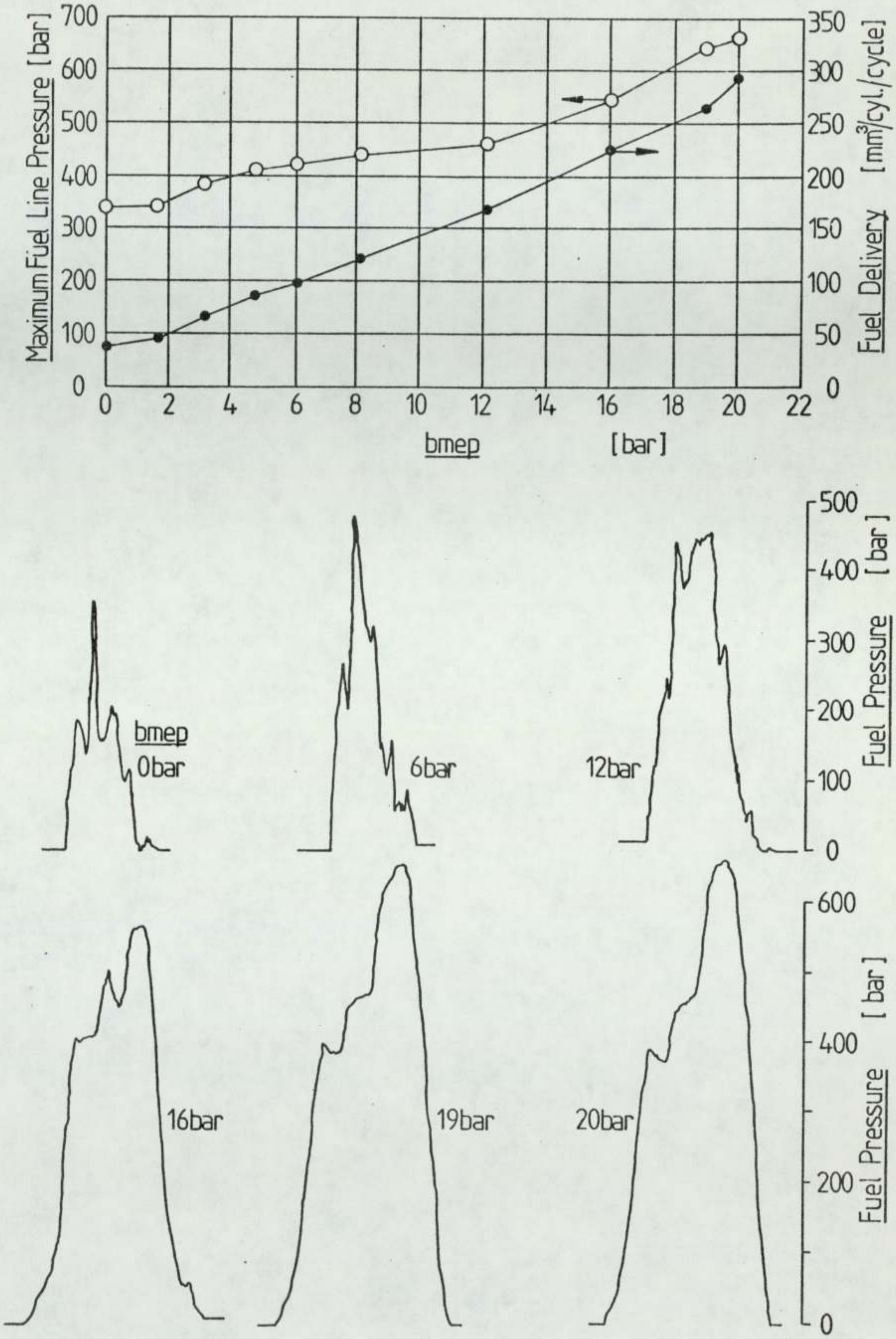
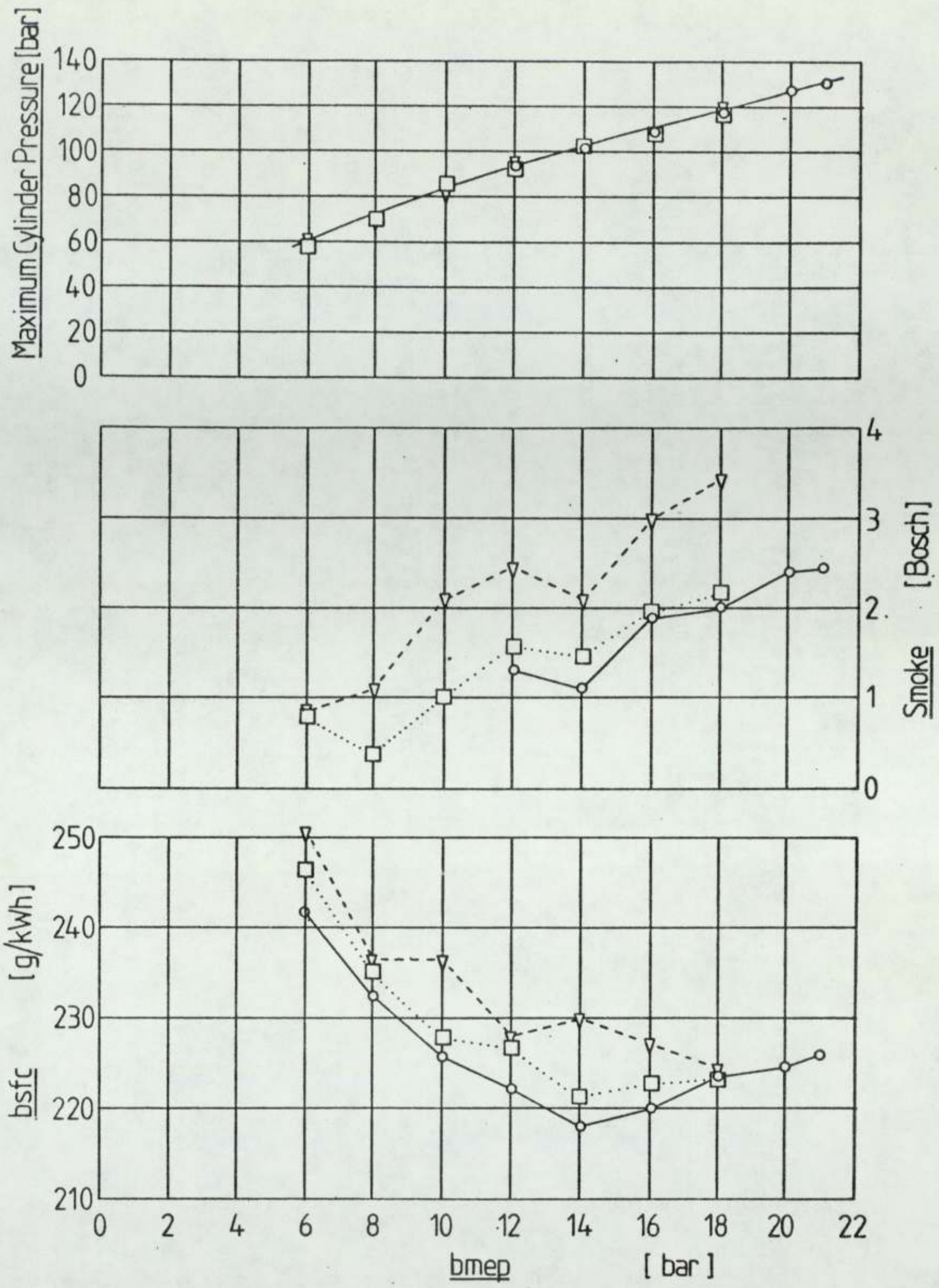


Fig. 6.19, Fuel line pressure waves recorded at the injector end. 12mm diameter plunger, 4 × 0·40mm diameter nozzle. [build 7, spill 24°]



Injector nozzle configuration:

○ 4 holes × 0·39mm diameter

□ 4 " 0·40mm "

▽ 4 " 0·42mm "

Fig. 6.20, Effect of injector nozzle size on performance. [builds 9, 10 and 11, spill 22°]

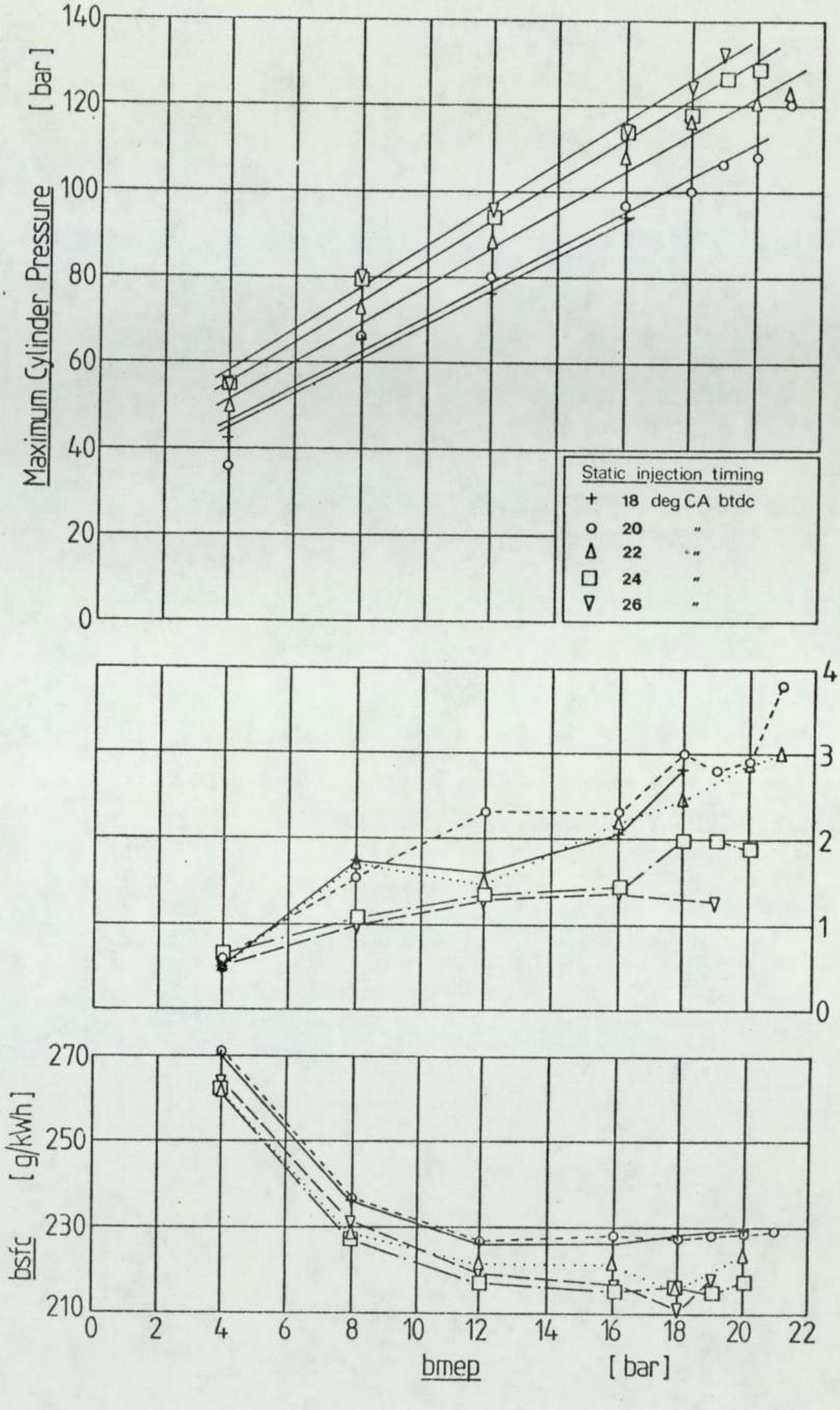


Fig. 6.21, Effect of static injection timing on performance. [build 15]

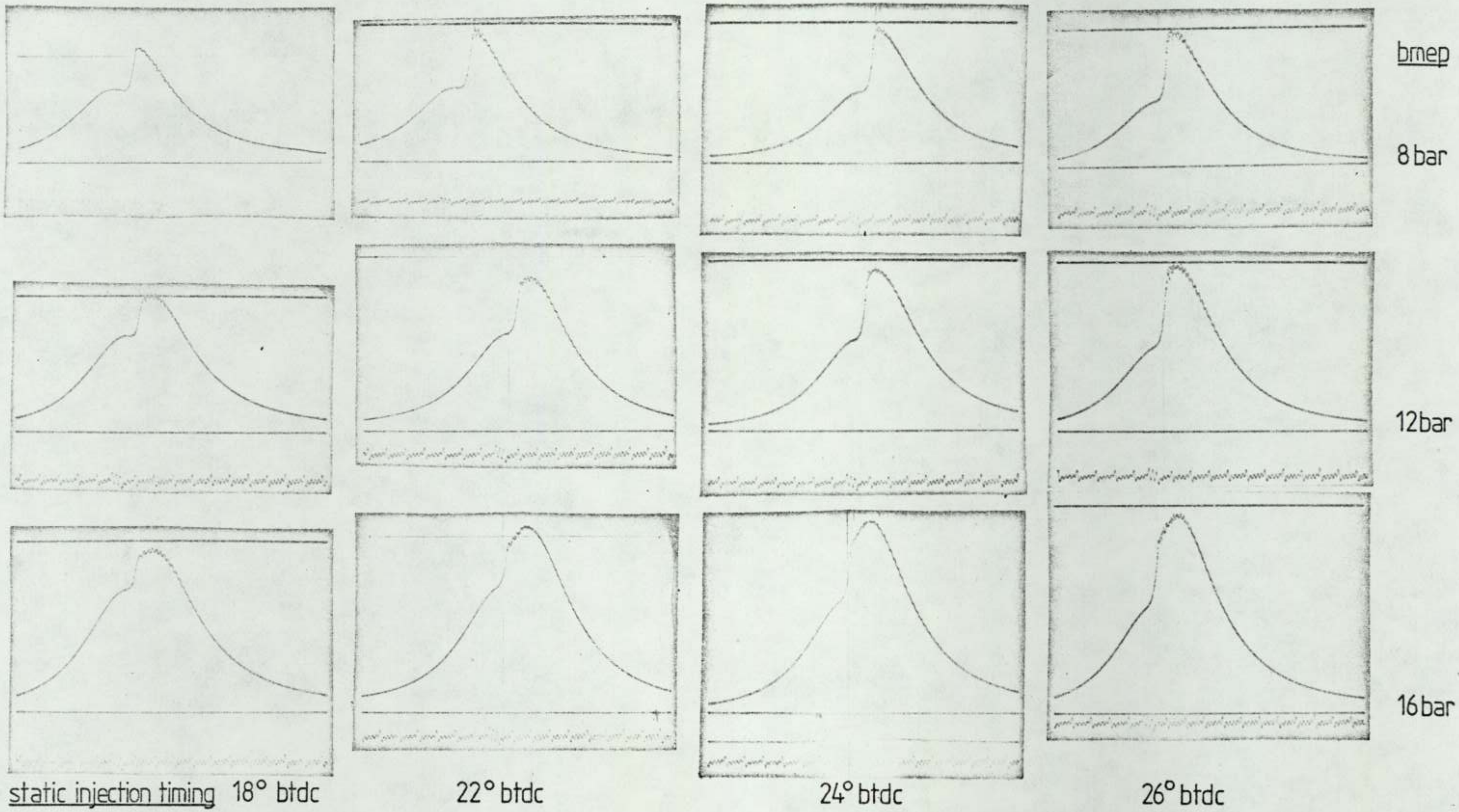


Fig.6.22, Effect of static injection timing on pressure development during combustion. 0 40 80 120bar [build 15]

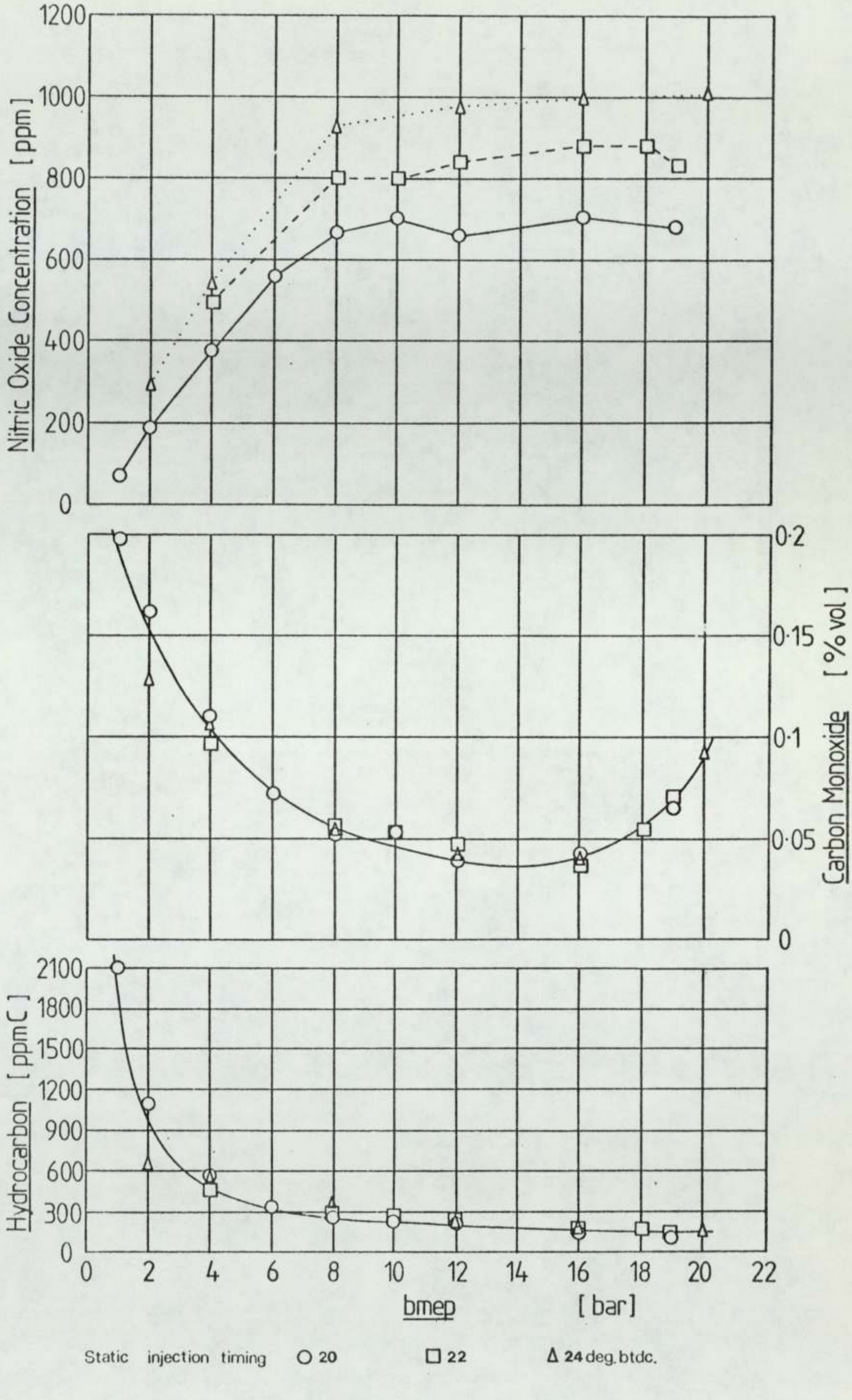


Fig. 6. 23, Effect of static injection timing on gaseous exhaust emmissions. [build 16]

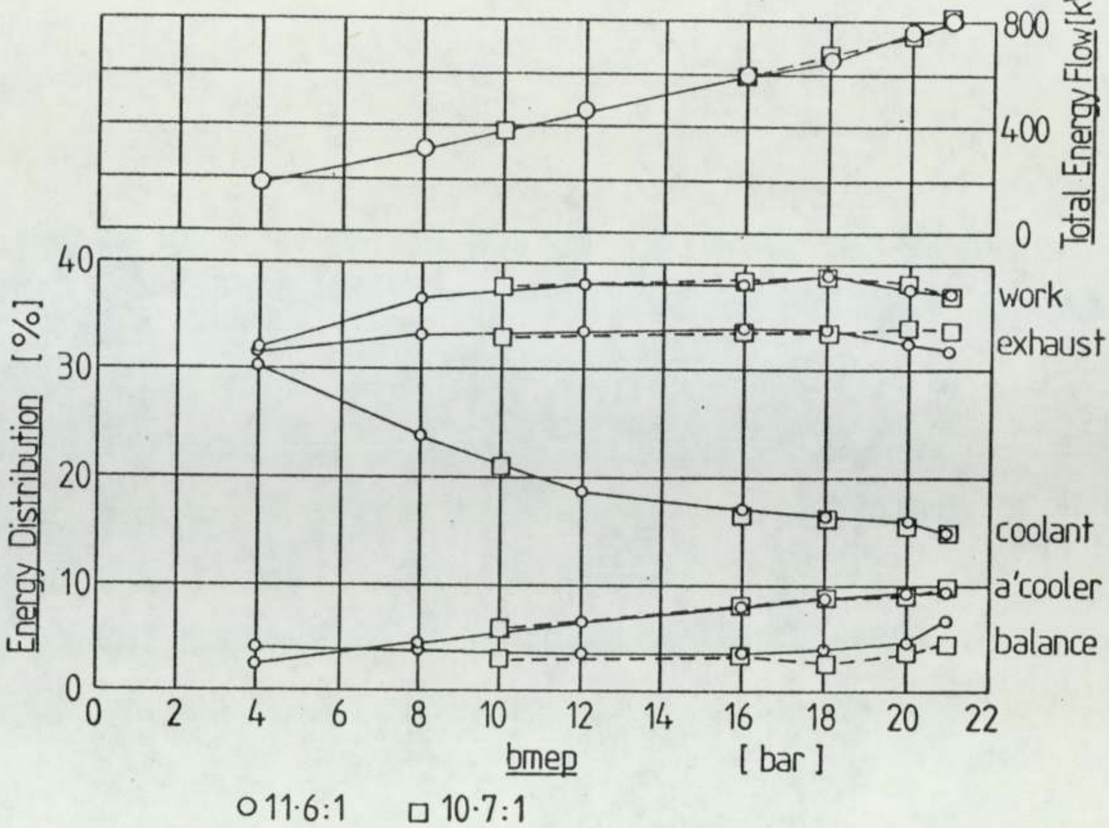


Fig.6.24 Comparison of energy flow with 11·6 and 10·7 compression ratio. [builds 12 and 15, spill 22°]

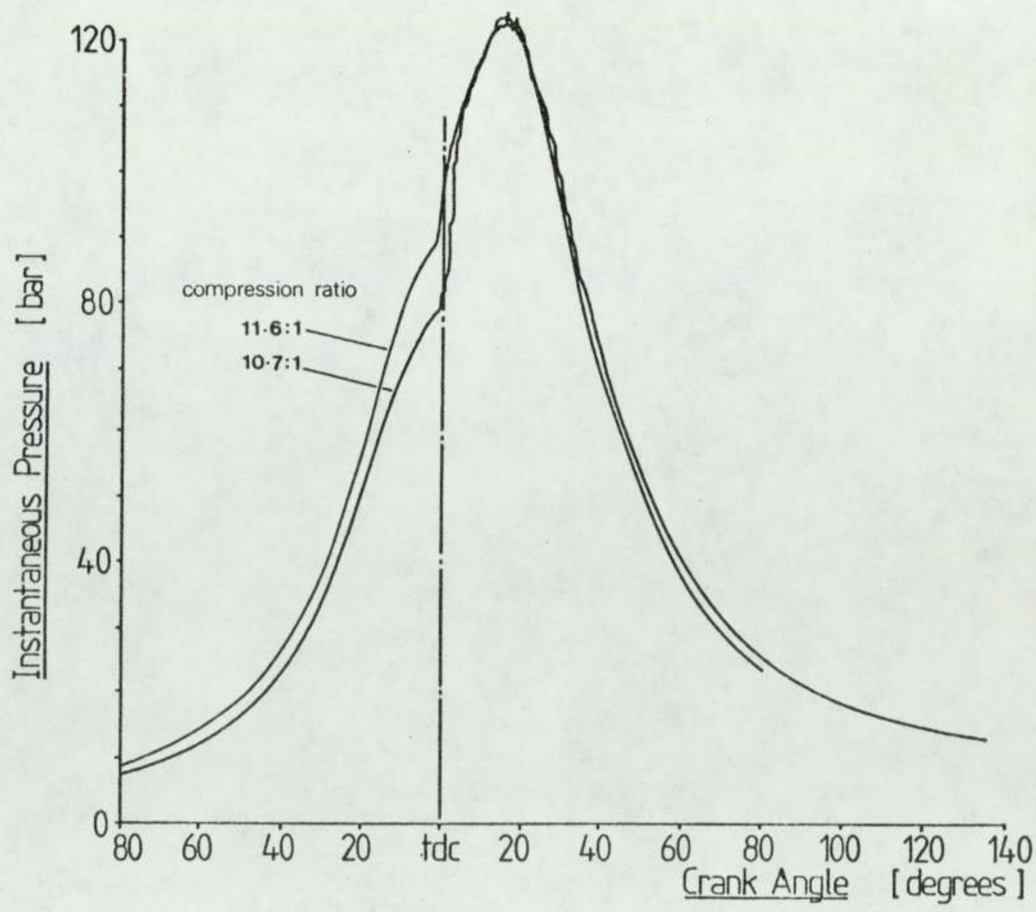


Fig.6.25, Pressure development at a bmep of 21 bar. [builds 12 and 15, spill 22°]

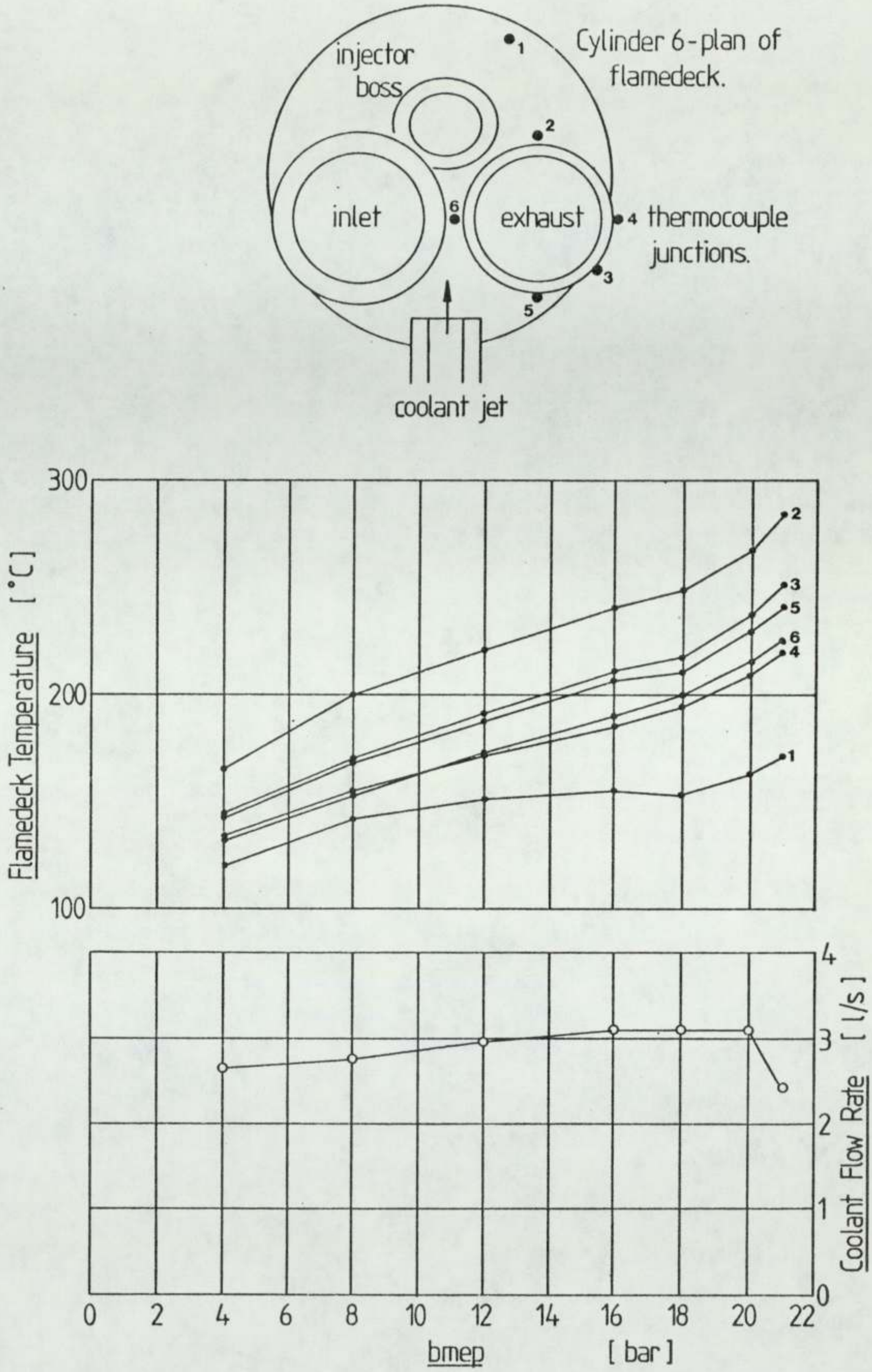
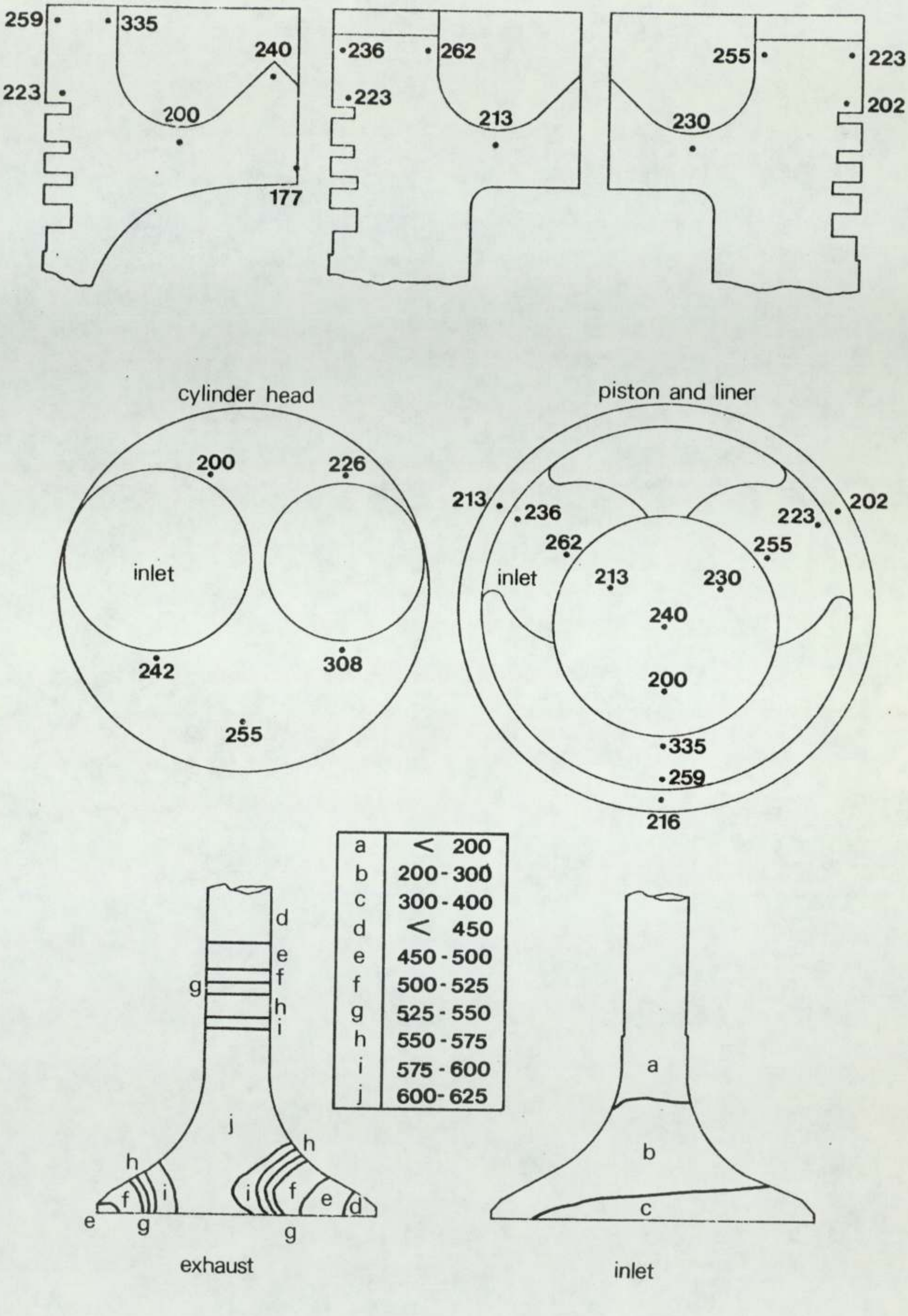


Fig. 6.26, Cylinder head temperature, 3·8mm from the gas face, and coolant flow rate versus load. [build 15, spill 22°]



a	< 200
b	200 - 300
c	300 - 400
d	< 450
e	450 - 500
f	500 - 525
g	525 - 550
h	550 - 575
i	575 - 600
j	600 - 625

Fig. 6.28 Combustion chamber component temperatures(°C) measured at a bmep of 18 bar. [build 6, spill 22°]

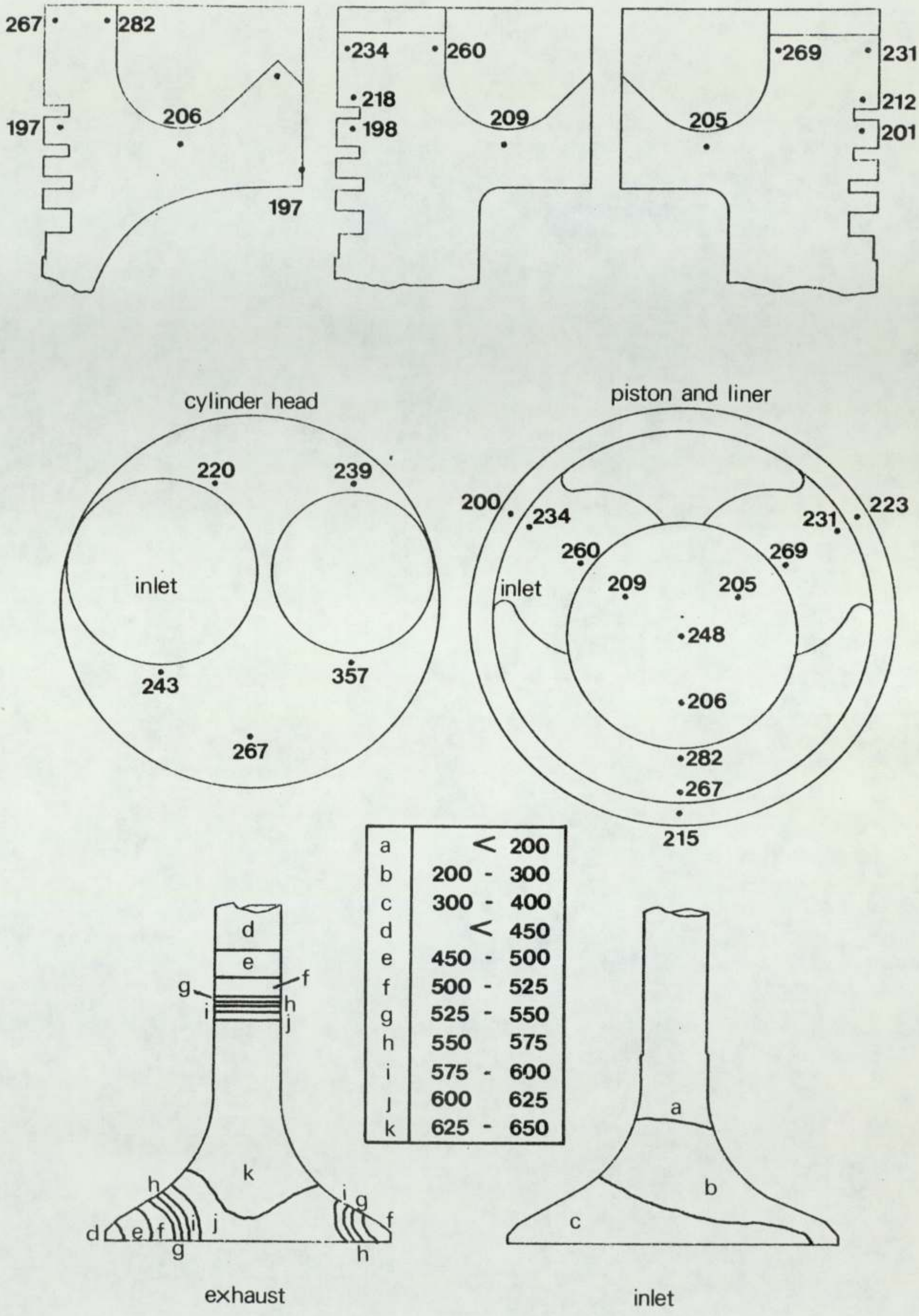


Fig.6.29 Combustion chamber component temperatures(°C) measured at a bmep of 20 bar. [build 12, spill 22°]

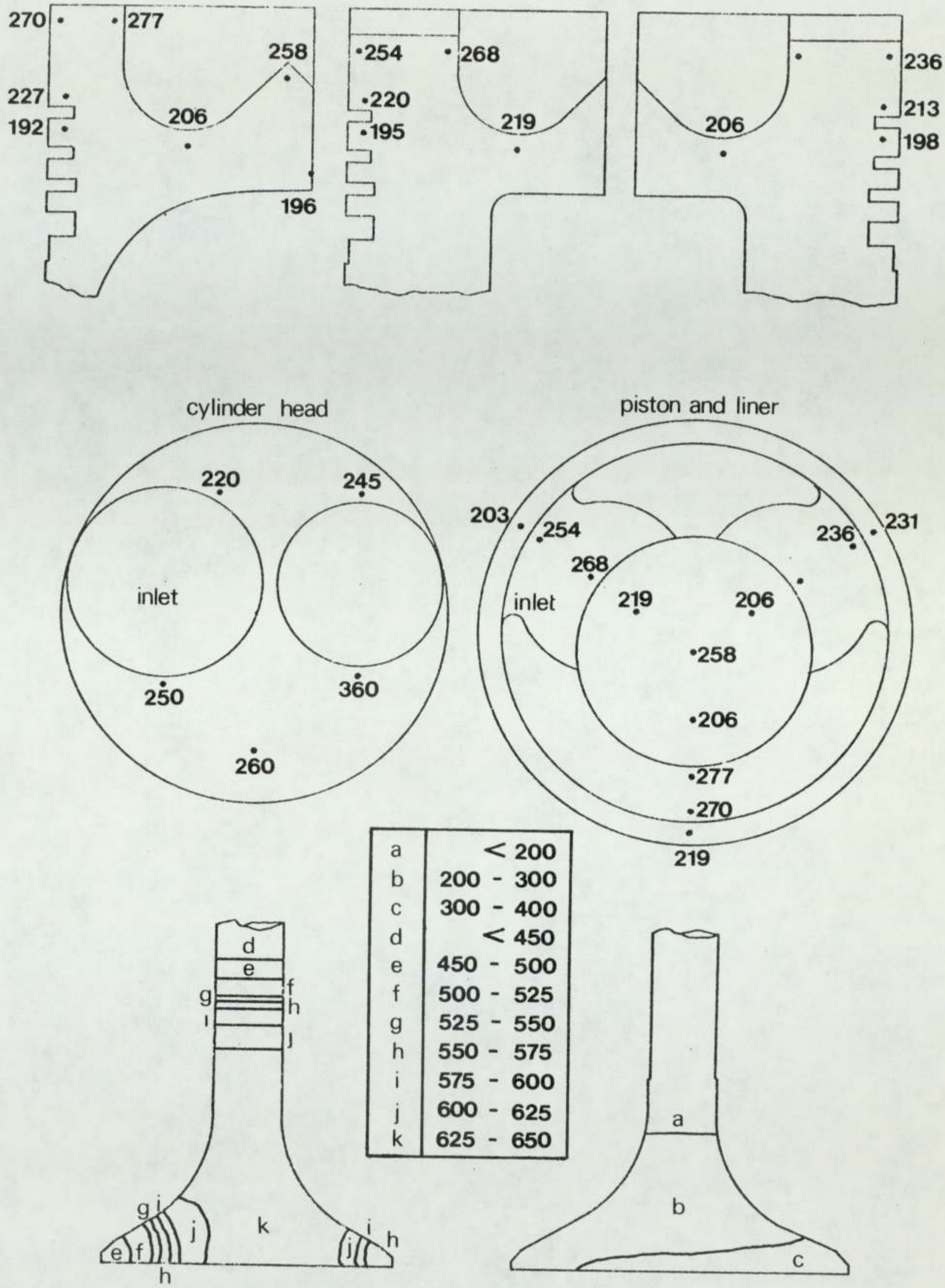


Fig.6.30 Combustion chamber component temperatures(°C) at a bmep of 21 bar. [build 13, spill 22°]

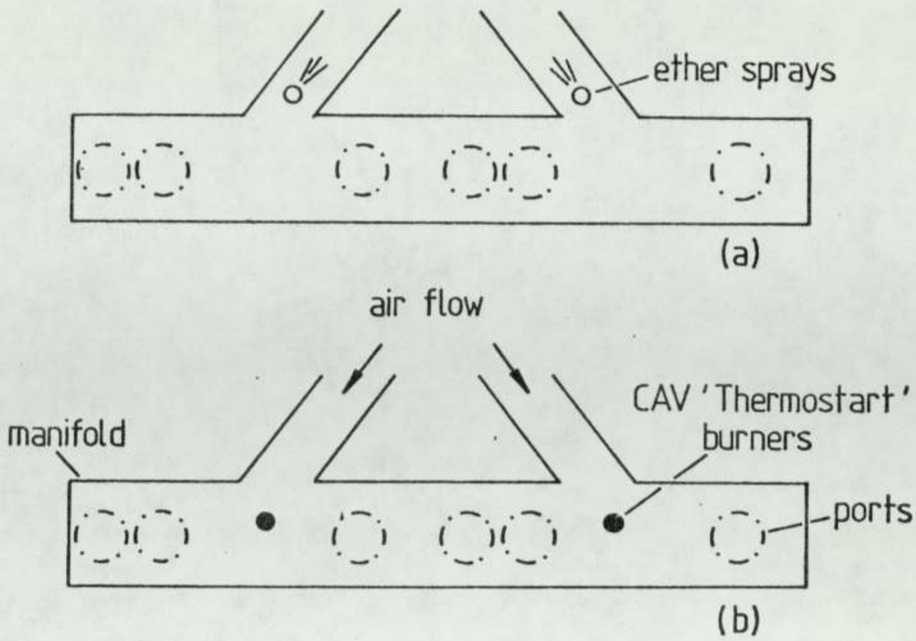


Fig. 6.32 Schematic drawing of the inlet manifold showing the position of (a) ether sprays and (b) fuel burners.

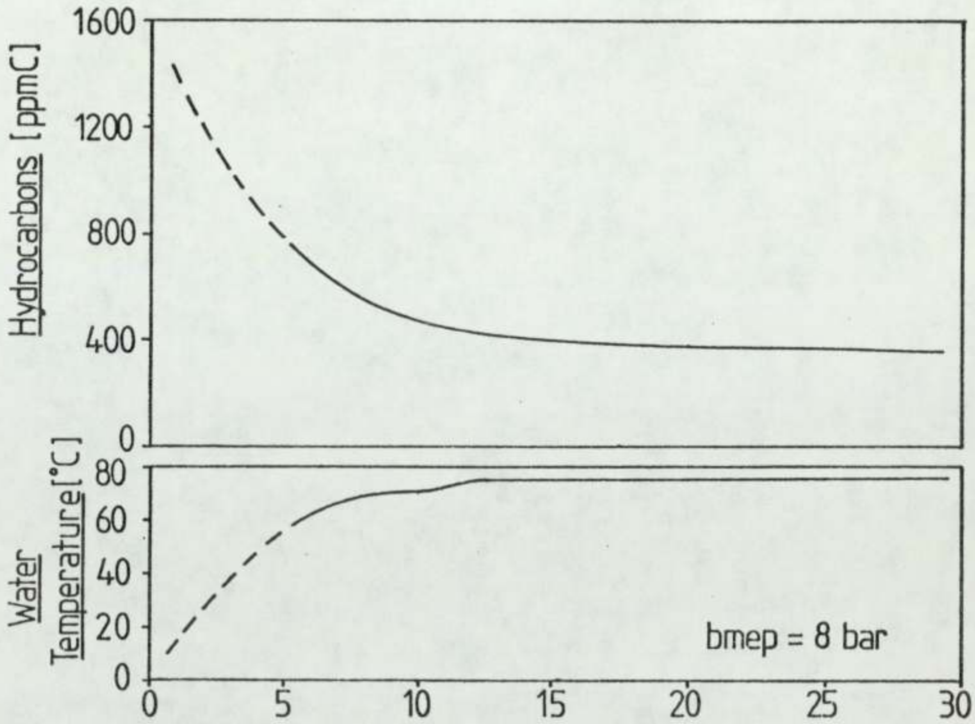


Fig. 6.33, Hydrocarbon and white smoke emission varied with engine temperature. [build 16, spill 20°]

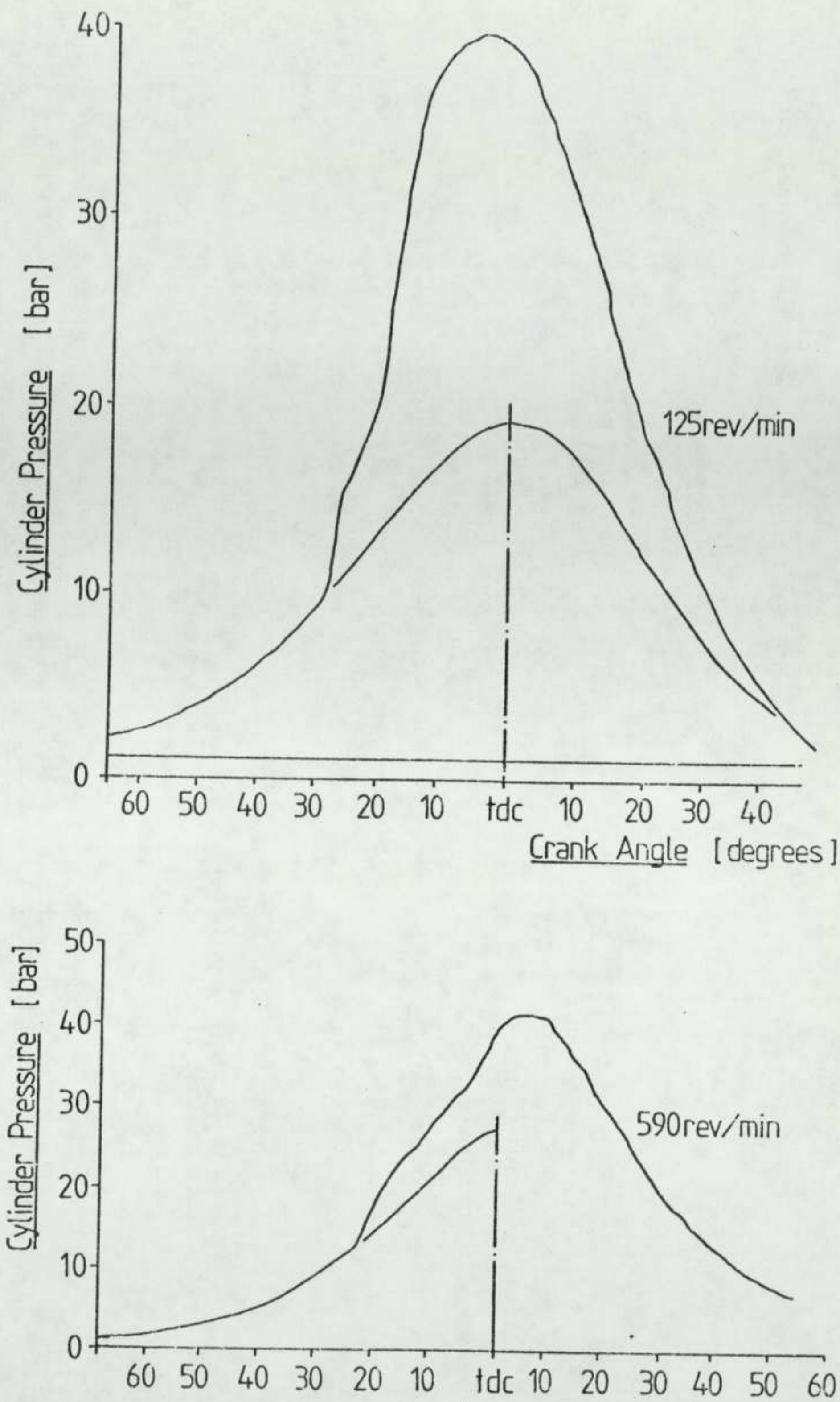
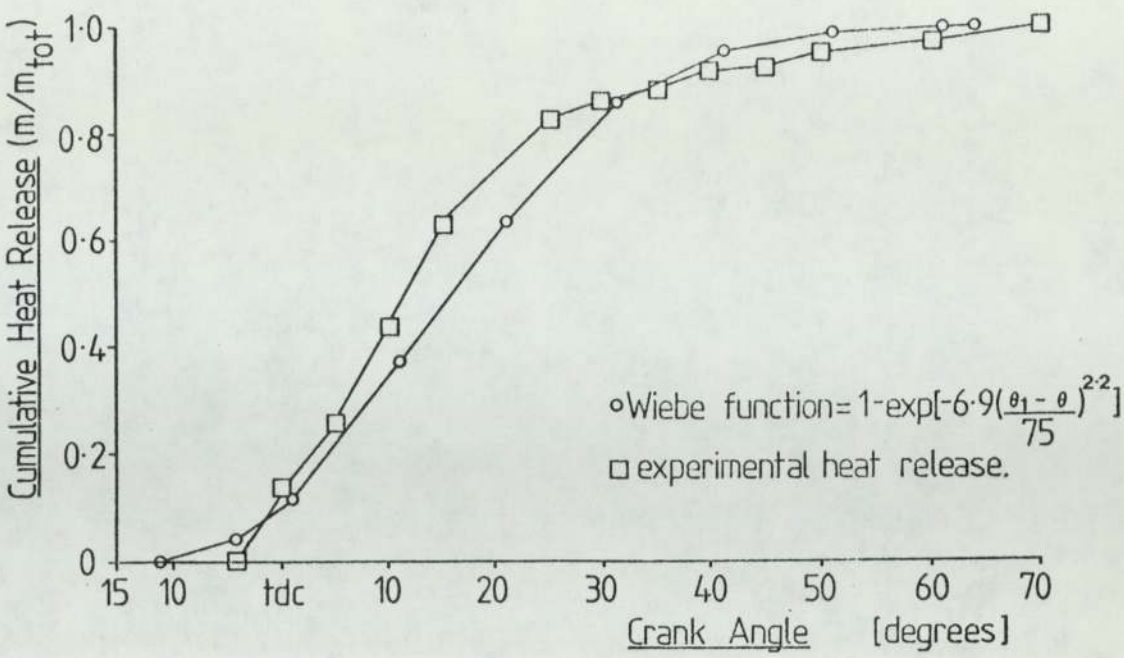


Fig.6.34 Cylinder pressure diagrams with ethyl ether manifold fumigation, no fuel injection, showing the early point of ignition. Compression ratio = 11·6:1.



(a)

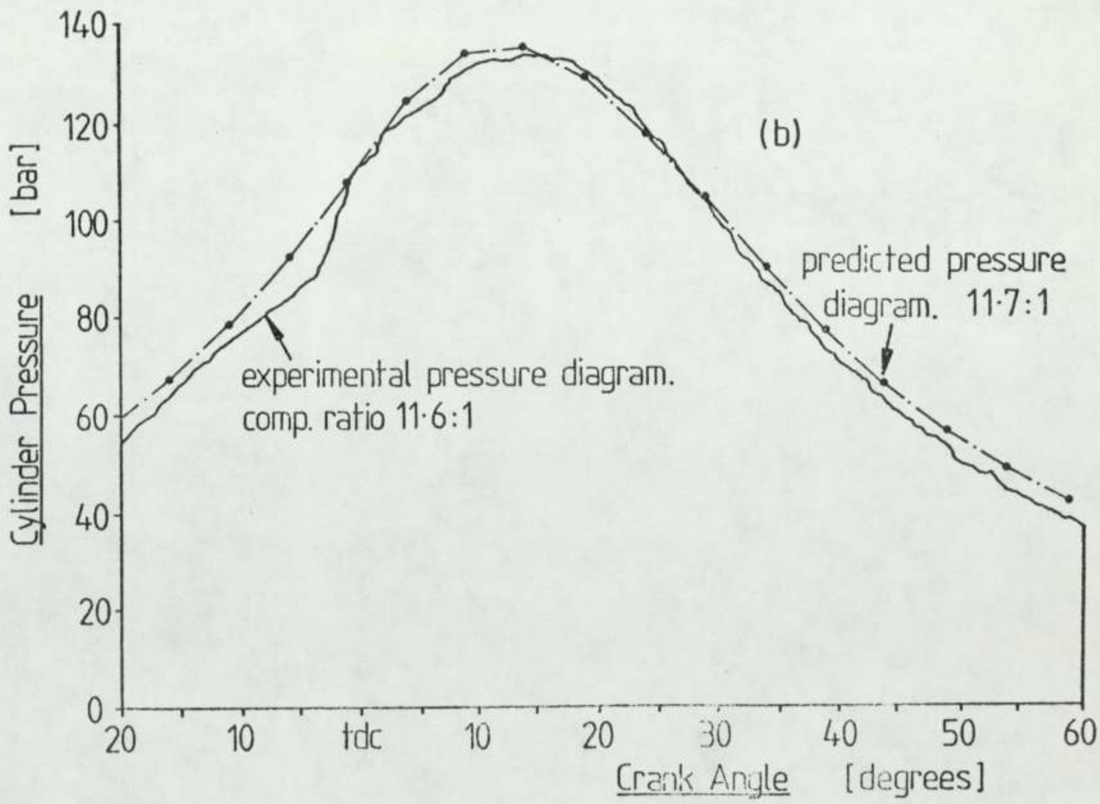
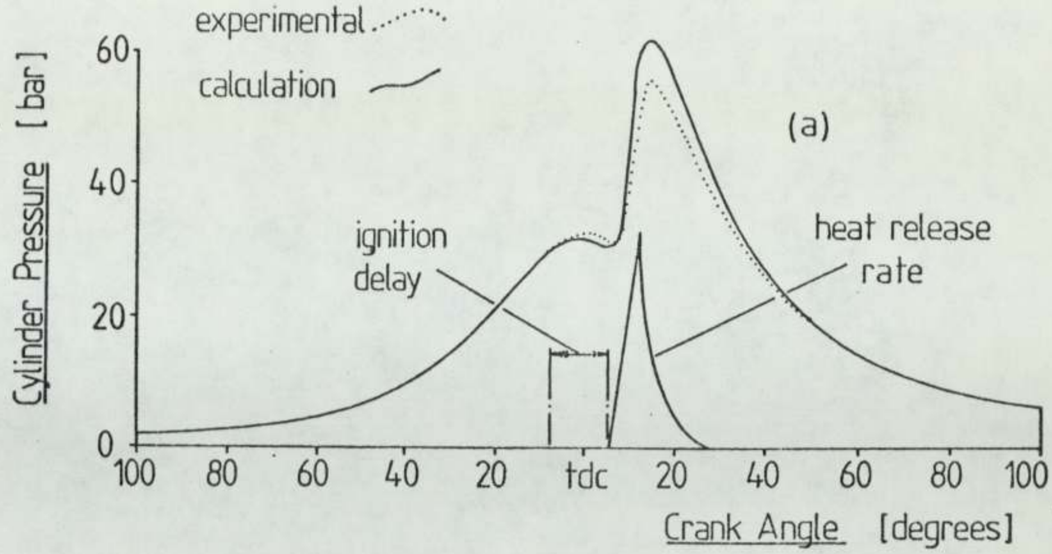


Fig.6.35, Comparison of predicted and experimental data at a bmep of 21 bar. [build 17, spill 24°]



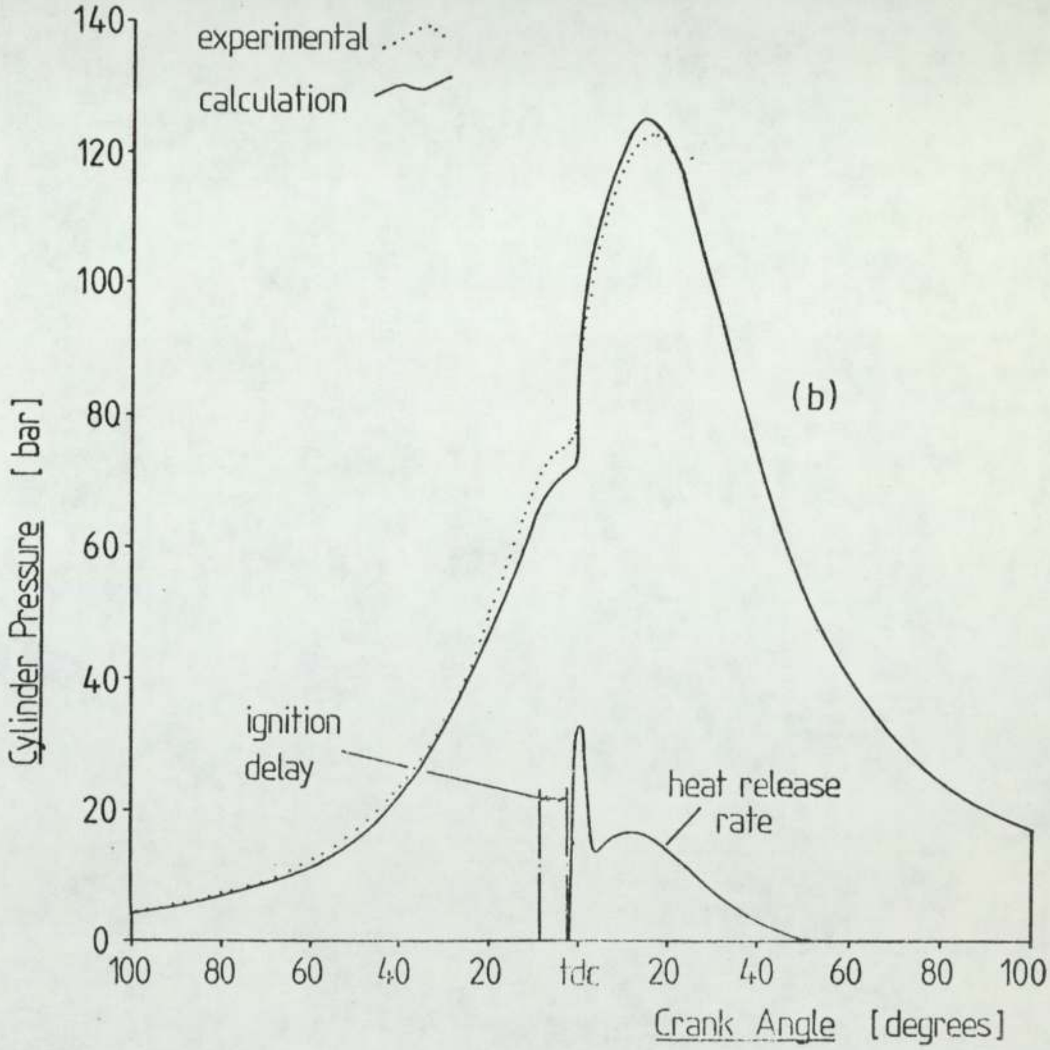
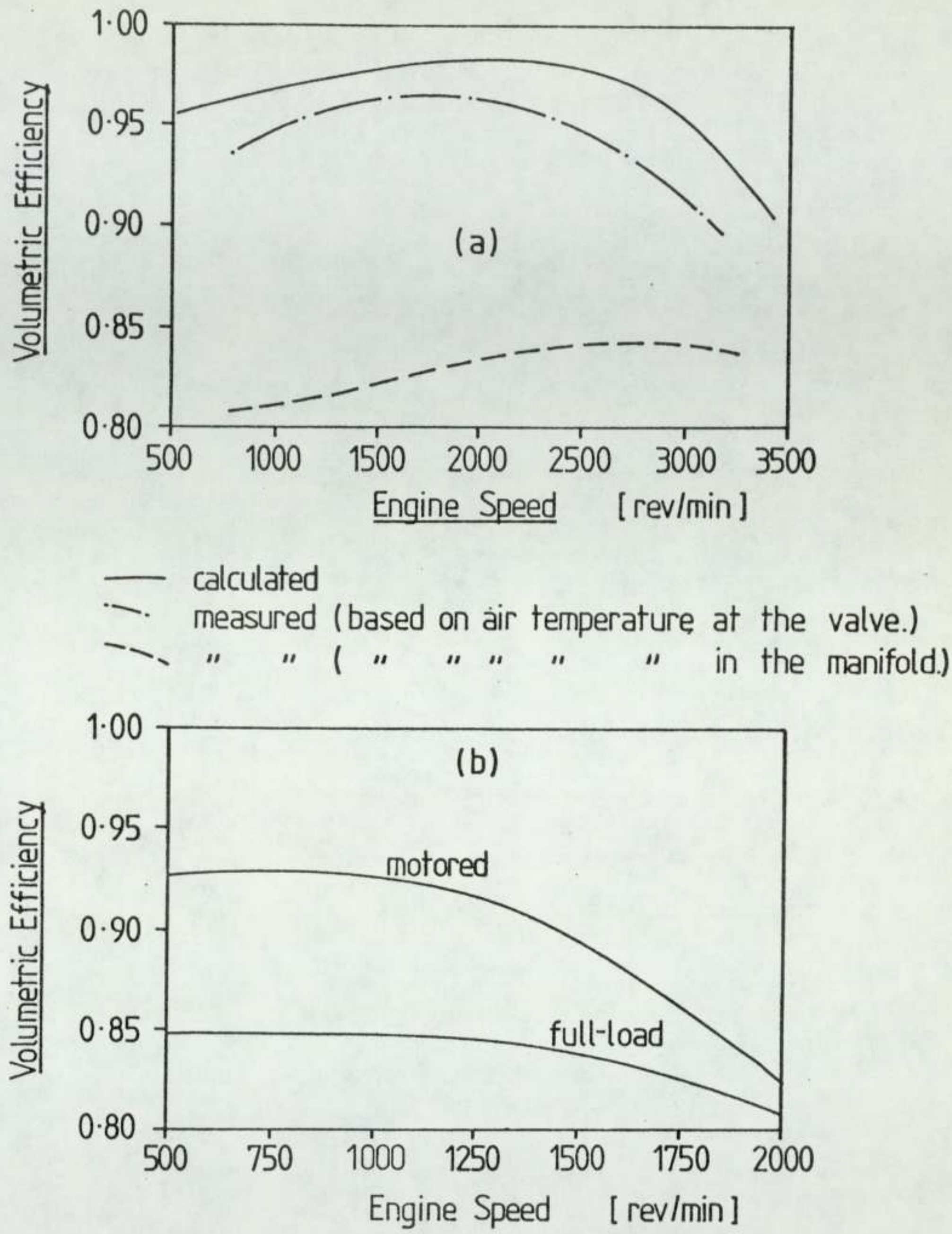


Fig.6.36, Cylinder pressure diagrams calculated retrospectively using the Marzouk-Watson heat release model. Compression ratio=10·7, bmep a. 6 bar b. 21 bar



((a) by permission of the Institution of Mechanical Engineers)

Fig. 6.37, Graphs showing a significant reduction of volumetric efficiency, caused by heat pick-up. (a) Jones ref (90). (b) Dicksee ref (91).

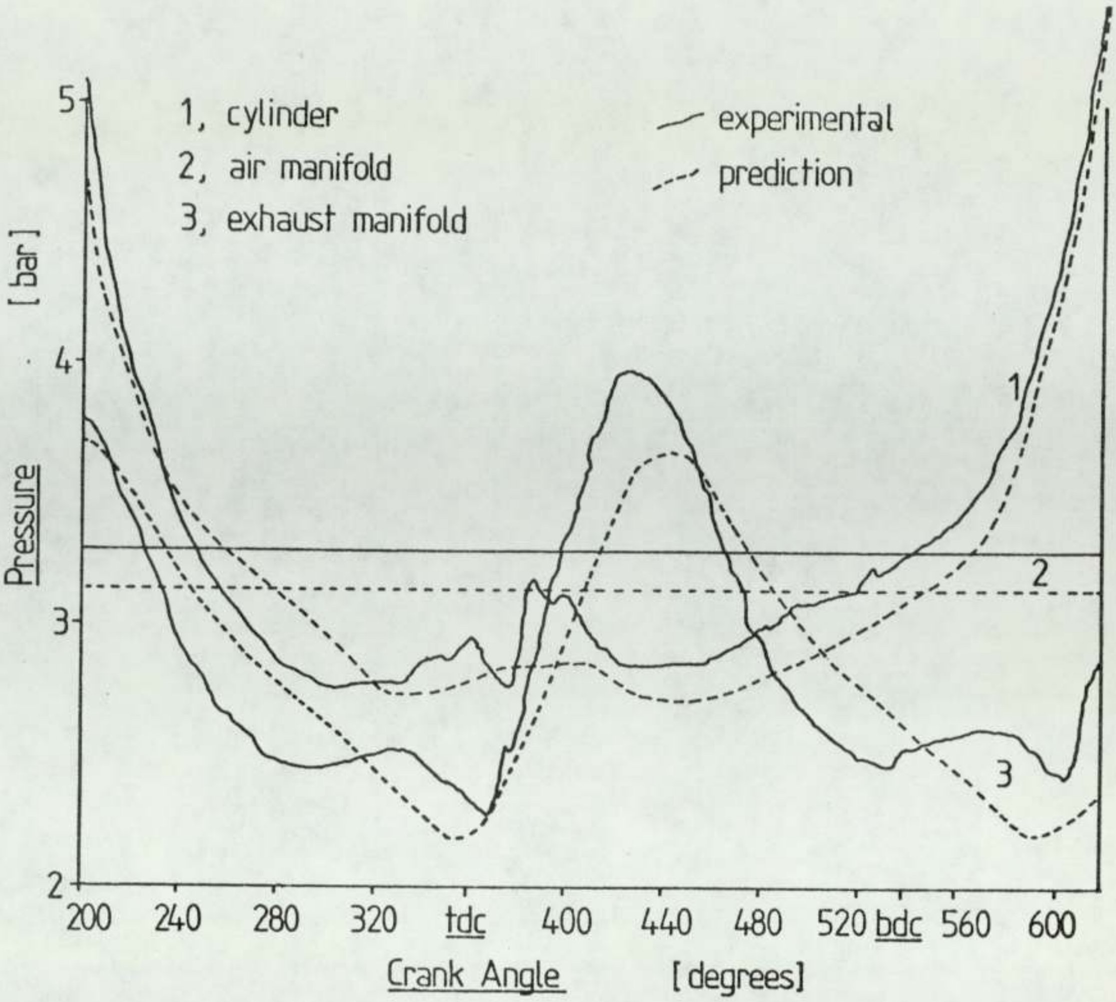


Fig.6.38, Comparison of predicted and experimental gas exchange at a bmep of 21bar.

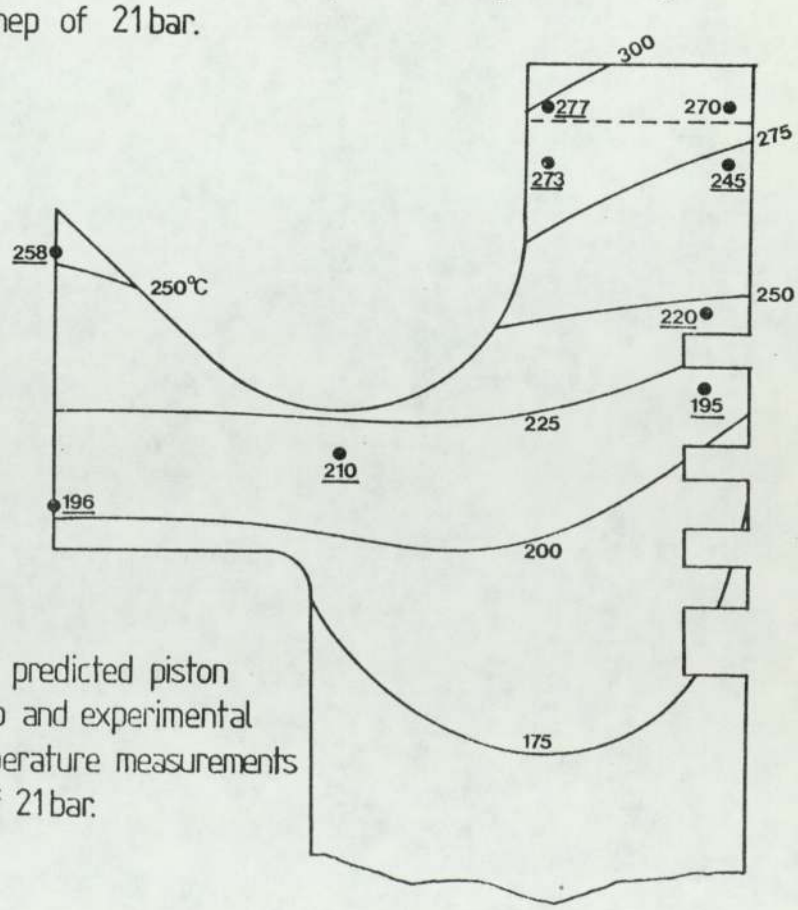


Fig.6.39, Comparison of predicted piston isothermal map and experimental 'templug' temperature measurements at a bmep of 21bar.

CHAPTER 7

COMMERCIAL STUDY

7.1. Introduction

The original project brief was to develop an existing Dorman engine to operate at a bmep of 21 bar at 1500 rev/min. Although this was later qualified by the sponsor to more closely define the scope and area of interest, the initial project definition was purely technical. During the first six months, the project was broadened to include commercial feasibility. Thus the final project objective was to study the technical and commercial feasibility of the high output concept. The major questions to be answered were :

1. Could the company manufacture an engine to operate at high specific output, based on existing designs ?
2. Would market acceptance be found ?
3. Would economic criteria be satisfied ?

These are, of course, very big questions in their own right and could not all be considered fully in the time available. The technical contribution has been presented in Chapters 2 to 6. This chapter is a contribution mainly to the question of market acceptance.

Unlike the technical development of the diesel engine which has been well-documented over the last seventy years, published accounts of their commercialisation are almost non-existent. It is suspected that this is partly because marketing and a "scientific" approach to business problems are relatively recent developments, and partly because of a perceived need for secrecy in all matters commercial. Although there is a lack of literature dealing specifically with the diesel industry, there are a number of published works which suggest general approaches to innovation and industrial marketing.

It is widely agreed that most companies need to innovate to remain competitive. This is a natural consequence of the basic human needs for progress and an improving quality of life. A company that relies on existing products will eventually be eclipsed by a more progressive competitor with products that more closely fulfill the market need. This gives rise to the concept of product life cycle, or more seriously, technology life cycle. An example of the latter was the demise of the electro-mechanical calculator in the wake of the electronics revolution.

The Advisory Council for Applied Research and Development (92) define industrial innovation as :

1. improvement of existing products or processes
2. introduction of novel production methods based on new technology
3. introduction of novel products or processes.

The first is the more usual ; the other two are exceptional since they usually require a technical breakthrough. Twiss (96) makes the distinction between invention and innovation. Invention is to conceive an idea, whereas innovation is the process of commercialisation by which an invention is translated into the economy.

It is now possible to identify this project as a feasibility study for an innovative new product. Innovative because it is a significant development of an existing product. New products can only follow from ideas or concepts, in this case the result of a technological possibility rather than the converse : an identified market need. The notion of high specific power - leading to lighter, more compact and cheaper engines - has developed in the corporate, technical mind of the diesel industry since the turbocharger was invented. Figure 7.1. shows schematically the phases in the development of the high-speed diesel engine. This shows the relationship of the area of interest to the historical trends of military and commercially available high-speed engines. During the early history of the diesel engine, prior to turbocharging, the specific power output was constrained by the limitation of breathing direct from the atmosphere. This caused the levelling-off towards 1955. Immediately after this, performance increased but was constrained by the low efficiency of the early turbochargers and the conservatism of the market. Later , the increase of specific output was given further impetus by the introduction of charge cooling. Today the constraints are startability and transient response. The most highly rated engines have the lowest compression ratio, consistent with rapid and clean starting, and many require block heaters or manifold heaters under slightly adverse conditions. Few of the most highly rated engines can accept their nominal full-load as a step function. This is usually applied in two or three steps. Thus the charge-cooled engine of today may be approaching a limiting specific output, as the naturally aspirated engine did before 1955.

Introducing new products is one of the most difficult tasks of management. This stems from the inherent uncertainties, for example, the future ; demand ; competition, and new technology. Kotler (97) estimates that, of the projects that reach development, as few as one in seven becomes a successful addition to the product line. Cooper (100) studied 114 industrial new products which had failed to meet profit criteria. Not surprisingly, the main general reason was that expected sales did not materialise. Underlying this general reason were more specific causes :

a) firmly entrenched competition - hence difficulties breaking into the market

b) potential users overestimated.

He concluded that too many organisations embarking on new product ventures lacked understanding of the market place, the customer or the competition.

Twiss (96) lists the factors which have been linked most often with successful innovation :

1. Market orientation.
2. Relevance to organisations corporate objectives.
3. Effective project selection and evaluation.
4. Effective project management and control.
5. A source of creative ideas.
6. An organisation receptive to innovation.
7. Commitment by one or a few individuals.

Market orientation is extended by Twiss to mean a company-wide attitude, especially with a free exchange of ideas between R & D and marketing. Blois and Cowell (94) also stress the mutual dependence of the R & D and marketing functions for successful innovation. They criticise the view that ideas for new products should only result from study of the market and defend R & D as a source of ideas. These usually evolve from technical developments and require a marketing effort to evaluate the new product potential.

Thus there are two clear possibilities : a market need in search of a technical solution and a technical possibility in search of a market need. This project is more like a technical possibility requiring a market. Rothwell (93) estimates that of all successful innovations, about 75 per cent arise in response to a market need (market-pull). The remainder follow a new technological potential (technology-push). He suggests that successful "technology-push" innovators determine that a need exists before they proceed with the project, and take great care to establish precise user needs, and to interpret them in the design of the new product.

There are few absolutes in the innovation process. There are no laws of nature to invoke ; no formulae to apply. It is more of an art than a science. Unlike pure or applied scientific research where solutions are readily framed in numerical terms and repeatable over time, innovation calls for subjective judgements - usually on incomplete data about a dynamic system at some point in an uncertain future.

Industrial marketing is a relatively new aid to management decision-making. It is increasingly necessary to use "scientific"methods to elucidate business problems because of the growing complexity and and competitiveness of industrial markets. Wilson (99) suggests that marketing should not replace entrepreneural flair, rather, it should complement it at a time when the breadth of knowledge and capacity necessary for sound decision-making are unlikely to be found in one individual.

Marketing research provides the substance on which marketing totally depends. Kotler (97) defines it as :

> "...systematic problem analysis, model building and fact-finding for the purposes of improved decision-making and control in the marketing of goods and services."

Of the literature available on the subject, the vast majority relates to consumer rather than industrial marketing. Although there are similarities between the two, the differences are significant enough to prevent the direct transfer of methodology.

A large part of the resources allocated to marketing research are used to establish market size, segmentation and trends. This is by no means the total activity, but it is important since these data will form a framework for the project and assist in setting realistic sales forecasts, R & D and marketing budgets. Other activities include identifying market needs, isolating the factors that bear on the buying decision, so that company resources may be directed most advantageously.

Marketing research makes use of primary and secondary sources. Primary data results from original research which makes new facts available. Secondary data results from a search of available publications, either from inside or outside the organisation. These may include government or other official statistical and bibliographical data, reports, surveys and internal records. There are many sources of potentially useful data. Wilson (99) gives an exhaustive list. Most projects require primary and secondary data, a fact common to most forms of research.

Primary data is made available by sampling. This involves defining the population of interest, e.g. users of electric motors or a sub-set of these, selecting a representative sample and devising a means of obtaining the required information. The most widely used method is the postal questionnaire. Other methods include the structured and semi-structured interview, telephone questionnaires and interviews and observation techniques. Chisnall (98) and Wilson (99) discuss these techniques in considerable detail. A summary of the recognised advantages and disadvantages of the postal questionnaire is given below.

Advantages

1. Cost effectiveness.
2. Respondents have time to consider replies.
3. Anonymity may be offered.
4. Elimination of interview bias.
5. Ease of analysis.

Disadvantages

1. Limited number of questions.
2. Difficulty in reaching target respondent.
3. Possibility of ambiguity or lack of understanding.
4. No chance to redirect or follow up questions.
5. Low response rate.

Reference (95) is an example of a new product evaluation. The main objective was to predict the likelihood of electric vehicles penetrating the commercial vehicle market. The problem was analysed in two parts. Firstly, having quantified the salient characteristics of the "state-of-the-art" electric vehicle, its technical compatibility with existing truck applications was studied. This revealed that 11 per cent of applications were within the performance envelope of the electric vehicle, as defined. The second part of the problem related to buyer attitude towards the new concept. Although the electric vehicle may meet usage requirements, a buyer may not wish to change for other reasons which bear on the buying decision.

The researchers used a technique known as conjoint measurement to predict whether the perceived attributes (low operating costs, low noise and air pollution) would outweigh the limitations (limited speed and range and high first cost). As a result of the analysis, each parameter could be assigned a utility, which is measure of its importance to the buyer. The results are given below.

Parameter	Electric vehicle	Existing vehicle
Speed and range	-1.426	+1.426
Operating costs	+0.928	-0.928
First cost	-0.901	+0.901
Pollution	+0.544	-0.544
Type of propulsion	+0.019	-0.019
net utility	-0.836	+0.836

This showed that when forced to make trade-offs between electric and conventional vehicles, the speed and range parameter had the most utility, whereas the type of propulsion system, per se, was relatively unimportant. The researchers concluded that despite fulfilling the technical requirements of 11 per cent of applications, the electric vehicle would not significantly penetrate the market because buyers were unwilling to trade speed and range for reduced operating costs and pollution.

This study is interesting because it is an example of the technological possibility looking for a market need. It is worthwhile noting the phases of the project :

1. "State-of-the-art" electric vehicle defined.
2. Technical compatibility with current applications assessed.
3. Buyer attitudes to new product studied.
4. Probability of market demand evaluated.

This class of problem arises with each new technological advance. For example, colour/monochrome television and jet/internal combustion engines for aircraft propulsion. Typically, the new product has recognisable advantages, but may have disadvantages, compared with the established product. The problem is deciding if the utility of the new exceeds that of the established, and will therefore present a new product opportunity.

7.2. High Output Engine Profile

Essential to any discussion of new product viability is the need to define as clearly as possible, the leading technical and economic parameters of the new product. The parameters thought to be most important to this study are listed below.

1. Production cost
2. Operating cost
3. Transient response

4. Noise

5. Exhaust emissions

6. Startability

7. Specific weight and overall size

8. Reliability.

These are not all readily quantified for a current production engine ; it is still more difficult for a prospective design. To some extent, the technical study provides answers to these questions. However, comprehensive study of operating costs, reliability, transient response and starting were beyond the scope of the project. Where available, estimates will be based on other data sources. Each parameter will be discussed in turn and compared with the "typical" conventional turbocharged and charge cooled engine. Where appropriate, comparison will be made with a Dorman "LE" turbocharged and charge cooled engine.

Production costs

This estimate of production costs makes use of the cost data available within the company, and cost estimates and quotations from suppliers, where applicable. The technical study indicates that a high proportion of "LE" engine components would be suitable for a high output variant. Therefore, the most convenient costing procedure is to modify the cost of the current charge cooled engine by adding the differential cost of new and enhanced systems. The possible modifications and additions required to convert from charge cooled to high specific output are given in Table 7.1. The costs of these components as a percentage of the charge cooled "LE" total cost in

December 1978 are given in Table 7.2. At that time, the total production cost was almost £4,000. The differential costs of conversion are given in the right-hand column. Since two-stage turbocharging is not of immediate interest, the costs incurred have been omitted. The effects of low cost items such as filters, have been neglected. In areas where hard data is not available, for example, oil-cooling jets and the extra machining to the crankcase, the costs have been estimated. The production volume on which these costs are based is approximately 200 engines per year, this being the production rate of charge cooled engines at that time. From Table 7.2 it is possible to build up the cost of a high output variant for marine or fixed speed industrial applications, i.e. raw water and air-cooled respectively. By assuming the use of the more expensive solutions in each case, e.g. steel cam shaft in place of cast iron, the expected maximum production cost may be derived. Similarly, by adopting the least expensive solution, a minimum production cost can be obtained. By dividing the new relative total cost by the relative power, the specific cost index is arrived at. This provides a comparison between the specific cost (£/kW) of a high output engine with a normally rated engine.

	Relative cost	Relative power	Cost Index
Marine build	1.2 - 1.29	1.35 - 1.54	.78 - .96
Industrial build	1.21 - 1.31	1.35 - 1.54	.79 - .97

The range of power increase results from the flexibility that exists in rating the engine. Assuming that the target power of 293 kW at 1500 rev/min could be rated for continuous, one-hour or stand-by duty, the percentage increase could vary from 35 to 54 per cent. Thus,

Table 7.1. : Possible modifications and additions to a charge cooled engine for high output operation

New Systems

Starting and light load aids

Piston cooling

Second turbocharger and charge cooler

Pulse converter junctions

Enhanced Systems

Engine structure	-	bearing shells, exhaust valves, cam shaft, casting modification
Air supply	-	air cleaner, ducting, charge cooler, turbocharger
Fuel supply	-	injection pump, H.P. pipes, filters, injectors
Heat dissipation	-	radiator, heat exchanger, water pipes, thermostats and housing, water pump
Torsional vibration control	-	viscous damper

Table 7.2.

datum cost £4,000	TCC cost%	HSO cost%	Diff. cost%
Engine structure			
Bearing shells	0.31	0.46	0.15
Cam shaft (steel)	-	2.50	1.55
Cam shaft (C.I.)	0.95	0.95	O
Exhaust valves (standard)			
Exhaust valves (Bimetallic)			
Air supply			
Turbocharger	2.45	2.45	O
Charge cooler (water/air)	3.87	5.75	1.88
Charge cooler (air/air) and radiator	13.60	18.75	5.15
Fuel supply			
Fuel pump	8.35	10.63	2.28
Fuel injectors	0.85	1.13	0.28
Heat dissipation			
Water pump	0.8	1.20	0.40
Water/water heat exchanger	2.9	4.98	2.08
Torsional vibration			
Viscous damper	0.76	4.23	3.47
Pistons and jet cooling			
Piston (standard)	2.0	2.0	O
Piston (with gallery)	-	2.8	0.80
Oil pump	1.53	2.6	1.07
Oil cooler	1.63	3.6	1.97
Jets and provisions	-	1.2	1.2
Low compression aids			
Starting	-	2-4	2-4
Light load running	-	2-4	2-4
Other costs			
Large water and air pipes, casting modifications, etc.	-	1 - 3	1-3

if the engine were rated most conservatively and constructed of the more expensive components, the production cost per kW would be only 3 or 4 per cent less than a conventional turbocharged and charge cooled engine (TCC).

Conversely, rating the engine to operate continuously at 293 kW and specifying standard components would produce a production cost saving of over 20 per cent/brake kW. It is not possible to be more specific because of the large number uf uncertainties involved.

Operating costs

Operating costs result from operating the engine for a given number of hours or producing a given useful output (kWh or miles). This would include the cost of fuel and oil, spare parts and maintenance. The information to calculate this is not available.

It is probable that the spare parts and maintenance requirement would be greater than for a conventional design at today's commercial ratings. However, this is speculation since this factor can only be quantified after exhaustive development testing and/or field trials.

The indications are that both fuel and oil consumption will increase slightly because of the reduced compression ratio and the need to use oil to cool the pistons.

It is highly improbable that a high output engine of this concept would offer any benefit over conventional designs so far as operating costs are concerned. It would be judicious to rate this factor as having a neutral or negative effect, in the absence of hard data.

Transient response

It is unquestionable that, unaided, the transient response of a high specific output engine is inferior to a conventional or normally rated design. Generally, transient response deteriorates with the degree of turbocharging, or the full load bmep of an engine. This primarily results from the need to accelerate the turbocharger through a speed range before the air demand is satisfied. If the load change is great enough and the turbocharger is unaided, the engine may be over-fuelled by the governor and eventually stall.

Highly turbocharged engines are unable to accept their rated full load as a step function. Figure 7.2. is reproduced from a draft ISO standard on engine performance.

The curve was established from a number of "representative" engines, and clearly rests on the observation that a four-stroke diesel engine will accept an instantaneous load change equivalent to a bmep of only 8 bar. This is arguably a little conservative, but the precise magnitude will depend on the allowable engine speed drop immediately after load application, which, in this case, was not stated.

From Figure 7.2 it is possible to compare the transient response of a normally rated and high specific output engine at bmeps of 15 bar and 21 bar. This shows that the lower rated engine would accept about 53 per cent of its full load, whereas the highly rated engine would accept only 38 per cent.

For application duties other then fixed speed, for example, variable speed truck application, the question of transient response or drivability, is less easily quantified, but nontheless an important consideration.

Noise emission

Although some noise measurements were made during the experimental programme, these are neither comprehensive nor accurate enough for firm conclusions to be drawn on the noise emission characteristics of this type of engine. The accuracy is questionable because the measurements were made in an open shop with other engines.

The findings of Hawksley and Anderton (30) explain the principal causes of combustion generated noise, and these are summarised by a simple model. This model relates engine speed (rev/min), cylinder diameter (mm) and maximum rate of pressure rise (bar/deg.CA) to overall engine noise (dBA).

$$\text{Overall engine noise} = 30 \log_{10} N + 50 \log_{10} D + (2.1\ \text{RPR} - 13) \log_{10} \left(\frac{5455}{N}\right) - 103$$

Thus, in comparing a conventional charge cooled engine with a low compression ratio, high output engine, the important factors from a noise standpoint are cylinder bore and maximum rate of pressure rise. For a given bore/stroke ratio, the bmep required to produce a given power is inversely proportional to the cube of the cylinder diameter. Therefore, the "LE" engine (127 mm bore) should be compared with an engine of 142 mm bore, to properly contrast engines operating at bmeps of 15 and 21 bar. By putting these values of cylinder diameter into the relevant terms in the equation, the effect on overall noise level may be predicted.

$$50 \log_{10}(142) - 50 \log_{10}(127) = 2.424 \text{ dBA}$$

Of greater importance to this discussion is the effect of rate of pressure rise. It is widely accepted that low compression temperature, associated with low compression ratio and charge cooling, cause high rates of pressure rise. This is due to the increase of ignition delay period and hence the high proportion of fuel consumed in the rapid, pre-mixed combustion phase. The maximum rate of pressure rise recorded during the experimental programme, with a compression ratio of 11.6, covered a wide range, depending on injection timing and induction temperature. The maximum tended to occur at about 40 per cent of full load, and at worst, was 14 bar/deg. This compares with values of up to 10 bar/deg., and often lower for a conventional, normally rated engine. At a fixed speed of 1500 rev/min the predicted contribution to overall noise level from this source is given by :

$$2.1 \times 4 \log_{10} \left(\frac{5455}{N}\right) = 4.7 \text{ dBA}$$

This is a prediction based on a highly simplistic model. However, it does confirm the conventional wisdom that maximum rate of pressure rise is a major contributor to overall engine noise. This is, of course, the source of diesel "knock". Therefore, when comparing the noise emission of a normally rated engine with a "low" compression ratio, high output engine, the latter should be considered inferior.

Visible exhaust emission

Diesel engines tend to produce soot which leaves the engine as a visible emission or "smoke". This is the result of incomplete combustion, usually caused by poor mixing of air and fuel or an inadequate air supply.

Although the experimental high output engine produced smoke which would be at best, marginal and at worst, totally unacceptable to engine users, this was not intrinsic to the high output concept. A poor smoke characteristic may be intrinsic to the use of deep valve recesses in the piston crown which, in turn, may be essential to the use of wide valve overlap, but this is a secondary consideration, although important.

There is no reason why the visible exhaust emission from a high output engine should be, per se, any better or worse than a conventional, normally rated engine.

Gaseous emissions

The gaseous emissions of greatest concern to public health are oxides of nitrogen (NO and NO_2), hydrocarbons and carbon monoxide. Both No and No_2 are toxic in low concentrations ; hydrocarbons are odorous and thought to be carcinogenic and carbon monoxide is toxic in high concentrations (102).

The diesel engine produces little carbon monoxide because of the relatively high availability of oxygen. This encourages the formation of carbon dioxide which is relatively harmless. A high output engine would require the same level of excess air as a normally rated engine, thus the emission of carbon monoxide would not be expected to deteriorate.

The production of oxides of nitrogen depends largely on the maximum gas temperature reached during the cycle. The maximum gas temperature will depend on several factors, e.g. degree of charge cooling, compression ratio, injection timing, engine speed and air fuel ratio. The emission survey carried out on the high output research engine over a range of injection timing compared favourably with data from other Dorman engines (103). The characteristic oxides of nitrogen emission* of normally rated Dorman engines falls in the range 7 to 12 g/kWh. This compares with a measured characteristic emission of about 9 g/kWh for the research engine. The characteristic emission of non-charge cooled, turbocharged Dorman engines is generally in the range 10 to 20 g/kWh. A "low" compression ratio, high output engine would probably have an oxides of nitrogen emission no worse than the normally rated, charge cooled equivalent.

The presence of hydrocarbons in the exhaust is usually a result of incomplete combustion. If sufficient hydrocarbon is present, it may condense in the cold air at exhaust outlet and be visible as a fog of fuel droplets, known as white smoke. Hydrocarbon emission is closely associated with low compression temperature, often occurring in diesel engines immediately after starting from cold. A turbocharged engine operating at a low load, or idling, will breathe effectively as a naturally aspirated engine does, direct from the

* Characteristic emission defined by DEMA (3 mode test)

atmosphere. This is because the available turbine energy is insufficient to operate the turbocharger effectively. If a "low" compression ratio is used (typically less than 13.0 : 1), incomplete combustion and white smoke will result. White smoke was produced over the range of compression ratios used in the experimental programme (10.7, 11.6 and 11.9) below a bmep of 6 bar. The hydrocarbon emission correlated with the white smoke and helps to quantify the problem - at a bmep of 6 bar it was measured at 300 ppmC and at 1 bar had risen to 2100ppmC. This level of unburnt fuel in the exhaust would be unacceptable to engine users and manufacturers alike. This situation could be improved by the use of charge heating, exhaust recirculation or exhaust restriction, as practised by Volvo (10)(11). The adoption of a suitable method is essential to the feasibility of high output engines of this concept. If a satisfactory solution is found, the low load hydrocarbon or white smoke emission need be no worse than that of a normally rated engine.

Startability

A compression ratio of 13.0 : 1 is generally regarded as the minimum for acceptable rapid and clean starting of high speed engines. To reduce startability to a single number is, of course, a gross simplification. Many factors combine to determine startability, for example, cylinder volume/surface area ratio, cranking speed, and fuel injection characteristics.

To produce a bmep of 21 bar within the capacity of a conventional high speed engine structure (limited by bearing areas and casting design) would require a compression ratio of less than 12.5 : 1. This under the most favourable design and ambient conditions, would be difficult, if not impossible, to start "unaided". However, the term "unaided" is a relative one since even engines of the highest compression ratio do not start themselves. If we define a starting system that comprises components whose main or only function is to start the engine, this would include lead/acid batteries, starter motor, excess fuelling devices, injection retard elements, block heaters, etc. The contribution of these items to production costs is not inconsiderable.

It may only be a semantic point, but the requirement of a high output engine is not for a starting aid, but for an extension to the capability of the existing start system. This extended capability could take one of several forms, e.g. charge heating or ethyl ether fumigation, or be a combination of measures. Given a suitable research and development effort, this problem should be solvable.

If we assume the problem can be solved, and compare the startability of high output and conventional engines, then the former should be judged inferior, because the presence of additional equipment would undermine system reliability and hence reduce the probability of a successful start.

Specific weight and specific overall size

Specific weight and specific overall size may be defined as the weight and box volume of an engine per unit of power output. They are determined by the bmep and the attention to detail of the design as a whole. To a great extent, the achievement of low weight and size depend on the importance attached to them at the design stage. Often these are subordinated to cost criteria, hence heavier and bulkier solutions are selected if they are less costly than lighter, more compact solutions.

Undoubtedly, high bmep can contribute to low specific weight and overall size ; the need for these attributes by the market is another question.

Reliability

Reliability may be defined as the useful output of an engine divided by the number of service failures during the achievement of that output, which caused a loss of availability. It may also be described as the tendency to break down, or the probability of successfully completing a particular mission or cycle of operations.

Reliability has been adjudged by many engine users to be one of the most important attributes that the diesel engine has over other prime movers. Although this is mainly due to the intrinsic reliability of the diesel engine, it is related to the "quality" of design solutions which also, in turn, affect the production cost. Reliability is also a function of the cumulative experience of the manufacturer with the product. Figure 7.3 is reproduced from reference (101) and indicates the contribution of specific engine systems to unreliability.

High bmep operation may have adverse effects on several of the systems shown in Figure 7.3. Of particular interest are those common to other engines in the range, such as engine structure, liners, crankshaft and connecting rods, or the turbocharger which is expected to operate towards the upper limit of its design envelope. For the high output concept to function commercially, production costs must be carefully controlled. This implies the greatest number of common parts with other engines in the range and the use of readily available, unsophisticated components from other manufacturers.

Because of the need to extend the engine's capability whilst minimising costs, reliability may deteriorate compared with established standards. If a satisfactory balance of cost and reliability cannot be found, this alone could invalidate the concept.

Comparison of high output with normally rated engines

The technical and economic parameters thought to be most relevant to commercialisation of the high output concept have been discussed. To elucidate the comparison between the high output concept and the normally rated engine, each parameter was quantified on an arbitrary scale expressing the perceived degree of advantage or disadvantage. This provides a profile of the high output concept, relative to the established product, the normally rated, charge cooled engine. This is shown overleaf in Figure 7.4.

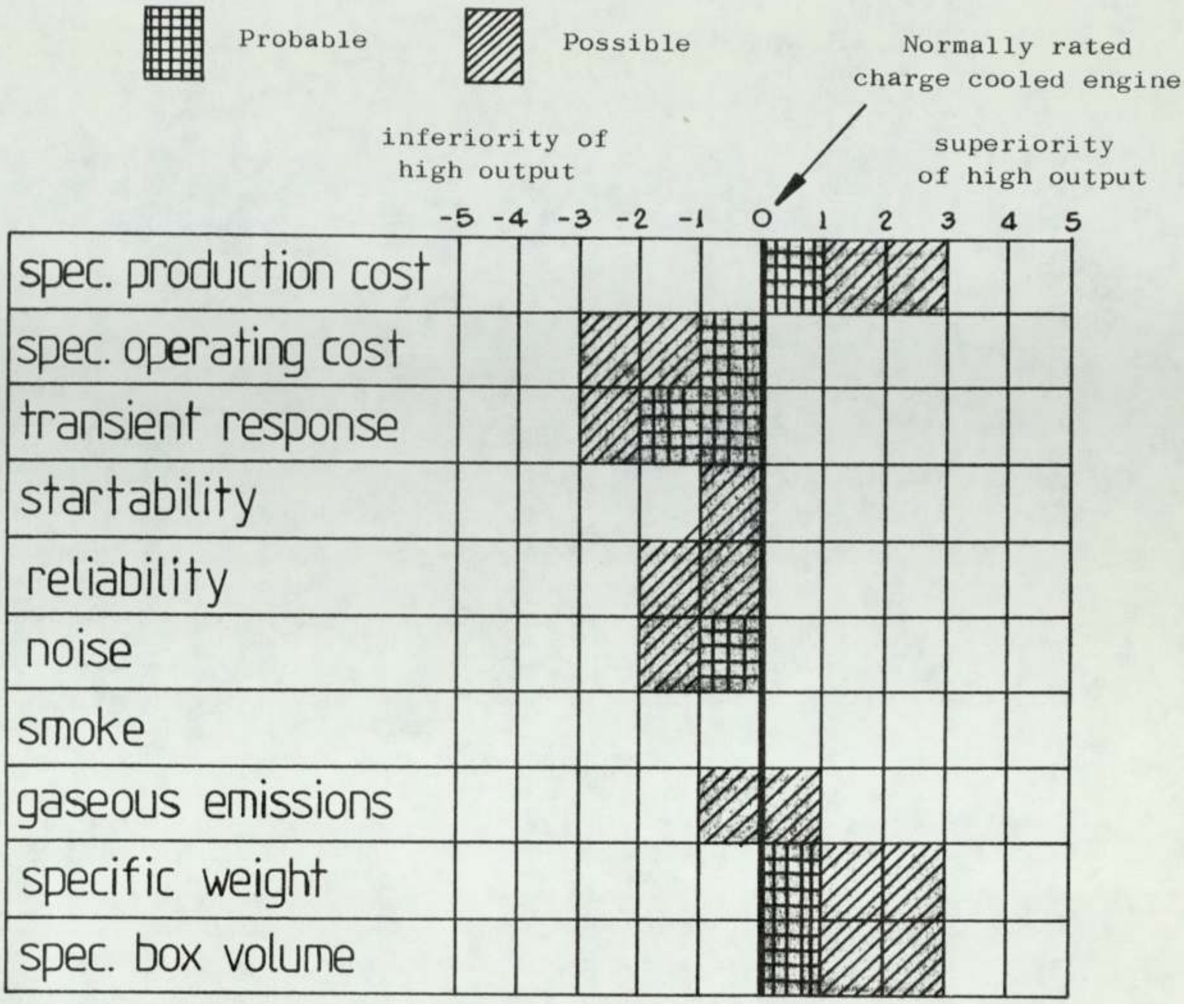


Figure 7.4 : Comparison of the high output concept with a normally rated charge cooled engine.

7.3. Market Segmentation

The diesel engine has been so successful over the past seventy years that it is now used in almost every application requiring mechanical power. Diesel engines are made for a very wide power range (2 kW to 36 MW) and vary considerably in design. As a further complication, they are sold internationally to a wide range of customers.

Market segmentation is a marketing tool intended to reduce large and complex markets into near-homogenous segments for the purpose of marketing research and policy-making. Industrial markets are often segmented in one, or a combination of the following ways :

a) customer size

b) customer location

c) industrial classification

d) usage rate.

This exercise requires that existing engine markets are screened to identify those to whom no additional benefit or utility is offered by high output technology. Thus, at this stage of the evaluation, the most appropriate basis for segmentation is end use.

7.4. New Product Screening

New product screening essentially compares a new product idea with the criteria thought most necessary for successful innovation. Most of the published literature discusses the choice between competing product ideas. This project is the converse : screening a number of market segments to evaluate the potential of a new product idea.

Usually, more than one screen is used. First, a coarse screen to eliminate the obvious "non-starters". This is followed by screens of increasing refinement to test the idea against technical and economic criteria. For this project, three screening criteria have been identified : technical compatibility ; utility, perceived by the buyer, compared with alternative products ; and compatibility with company resources.

Screening Criteria

1. Technological compatibility of the high output concept with existing applications.
2. Utility of high output concept (does it offer significant advantages over alternative products ?).
3. Compatibility with company resources.

Technical compatibility means : can the new product perform the duty less well, equally well or better than existing alternatives ? Product utility is less tangible. It may comprise one, or many factors, will normally change over time and vary from customer to customer. Utility is the perceived overall satisfaction that buying the product confers on the buyer. It is thought that a buyer seeks to maximise utility when making a purchase. Thus, not surprisingly, when comparing competing alternatives, the one seen to offer the greatest satisfaction of the need felt, is purchased.

Company resources embody the assets of human and corporate expertise and skills, the buildings, production facilities, distribution networks and the availability of working capital.

Automotive Applications

Automotive applications in the 200 to 750 kW range include on-highway trucks, off-highway vehicles and some military vehicles. On-highway truck engines, with few exceptions, are manufactured by volume production methods. They are often designed for transfer line machining and assembly by unskilled labour. Truck engines are generally robust to cope with the arduous conditions of continuously varying load and speed. They require good fuel economy, clean combustion, reliability and maintainability.

In the UK, diesel engines were first used in trucks in the 1920's when they gained an economic advantage over spark-ignited engines, due to the taxation of petrol. Despite the introduction of turbocharging around 1955, the vast majority of truck engines remained naturally aspirated until the last decade. Today, the advantages of turbocharging are more widely accepted by truck users. Increasing numbers are turbocharged and some are charge cooled. Kamo (9) has studied the feasibility of high bmep truck engines for Cummins Engine Company. He concluded that engines operating at bmeps of 15 bar and above were feasible, and offered certain advantages for automotive applications. The advantages that Kamo delineated are given below.

1. A high output variant would extend the power range covered by an engine series. This would make the low volume top-of-the-range engine more economical to produce.
2. Reduced specific costs particularly in the bmep range, 17 to 21 bar.
3. Reduced specific size and weight.

The problem areas, however, were not inconsiderable, and included : turbocharger matching for torque back-up ; transient response (drivability); starting ; and white smoke.

Assuming the high output, on-highway truck engine is technically feasible, the questions that remain are : does it offer a significant advantage to the user, and would it be compatible with company resources ? The first question would require a marketing research effort to estimate whether the concept offered greater utility to the buyer than existing engines. The question of compatibility with company resources is the crucial one, in this case. Dorman have no recent experience of the truck engine market. They have no established base engine suitable for, or proven in this application. Therefore, the high output concept as an extension of an existing engine series would not be possible so far as the automotive market is concerned. The question of corporate experience is vitally important for this market since it is especially demanding and unforgiving. The demands are for economic, reliable technology with excellent service and spares support. If a product fails to meet these demands, the after-effect could be long-term since, in this industry, reputations count.

On the basis of company resources alone, the probability of Dorman successfully entering this market with a high bmep engine is low. In general terms, the new product/new market combination is the most difficult with which to successfully innovate.

Off-highway trucks, to some extent, require the same engine characteristics as their on-highway counterparts. They must be durable, tolerant to adverse and widely varying conditions, operate over a wide speed range and have good torque back-up.

However, this is not a high volume market and covers a wider power range than the on-highway truck. The range is approximately 200 to 750 kW which coincides with the range of current Dorman engines.

Aveling Barford of Grantham who make vehicles in the 200 to 450 kW range, estimate that about 25 per cent of off-highway truck engines are turbocharged, and a smaller proportion are charge cooled. Engine weight and size are apparently not critical considerations because of the high weight of the vehicle as a whole, and lack of size restrictions. Most off-highway trucks travel as "wide loads" on British roads, the width being a function of the load-carrying part of the vehicle, not the engine.

Often, this type of vehicle will be engaged on major construction projects having exacting time targets. As a result, the utility of an engine may depend less on cost considerations and more on reliability and performance. With this requirement for an engine that will keep working under the most adverse conditions is the need to support the product with spare parts and service. Caterpillar Tractor Company claim that their world-wide network of almost 1000 sales, service and parts facilities is the foundation of their business. For construction equipment, product support represents significant utility to the buyer.

Dorman have sold engines for this application in the past and therefore have some limited experience of the problem faced. On the debit side, it is a comparatively small market, dominated by a large and sophisticated organisation, Caterpillar Tractor. The fact that less than 25 per cent of current usage is charge cooled does not suggest rapid uptake of the high output concept. In conclusion, the opportunity could, at best, be described as limited, and the probability of successful innovation relatively low.

Military non-combat vehicles tend to be based on conventional commercial technology. By contrast, combat vehicles have a history of using "state-of-the-art" technology. The most highly rated diesel engines to enter service have powered main battle tanks. This is because of the severe restrictions on size and weight and the need for a high power/weight ratio. Typically, the priorities in battle tank design are for fire-power, mobility and protection. Service life, capital cost and operating cost are all subordinated to achieving the highest power from the least engine space and weight. Perhaps the only proviso is for high reliability over the stated operating life, albeit a short one.

Ministry of Defence contractors must demonstrate high standards of manufacture and quality control, whilst engines are subjected to rigorous type-testing. It is possible that the average engine manufacturer would have to up-grade procedures to meet these requirements.

If we include in the definition of company resource the existing product range, then only two engines suggest themselves as the basis for a highly rated combat vehicle engine. These are the 90 degree vee-form "J" series engines of 8 and 12 cylinders. They were designed with a wide speed range (up to 2200 rev/min) and high allowable cylinder pressure (138 bar). The power output at 2200 rev/min for six levels of bmep are given below.

bmep	7 bar	15 bar	18 bar	21 bar	24 bar	30 bar
8J	170 kW	365 kW	438 kW	511 kW	584 kW	730 kW
12J	255 kW	547 kW	657 kW	766 kW	876 kW	1095 kW

The power requirement for battle tanks varies with vehicle weight. The usual requirement is for 20 to 30 BHP/ton (14.7 to 22 kW/tonne). Consequently, engine power is normally in the range 560 to 1120 kW. On this basis, the "8J" may be too small for this application.

Probably the main problem an engine manufacturer would face in trying to enter this market would be finding a vehicle maker to team up with. This may be the Ministry of Defence or a private concern such as Vickers. However, the existing links between vehicle and engine maker are strong because of the capital investment and design effort of integrating the two.

The high output concept is compatible with the demands of fighting vehicles, and Dorman may have an engine suitable for such a project. However, success in this field could only result from a high degree of collaboration from the outset, with a combat vehicle manufacturer.

Construction and Earthmoving

The construction and earthmoving market includes equipment such as excavators, cranes, bulldozers, scrapers, graders and loaders. The engine requirements are similar to those for the off-highway truck. The engine must be reliable and tolerant to poor operating conditions. The power range is wide - the smallest machines using less than 75 kW and the largest up to 1000 kW.

In the UK, the leading excavator and crane manufacturers are Priestman Brothers of Hull and Ransome and Rapier of Ipswich. Neither company has yet used a charge cooled engine in their equipment, although they do use turbocharged engines. Weight is not important, size and shape are only important insofar as the engine must be compatible with existing units. The required engine characteristics are ruggedness and durability under negligible maintenance schedules.

The world leader in earth moving equipment is the Caterpillar Tractor Company, who manufacture both engine and vehicle and sell engines to other vehicle makers.

The probability of stimulating demand for a high bmep engine in this market is very low. Technically, the concept would probably not cope well with the rapid load application as the vehicle meets the load to be moved, unless turbo assisted ; furthermore, if the Industry does not make significant use of charge cooled engines, there would be little scope for penetration by a high output variant.

Industrial

The "industrial" segment includes many varied applications not readily classified under any other heading. Fluid pumping (for process plant, fire-fighting or surface water), compressors, stone-crushers, paper and saw mills are examples. Because the technical requirements are so varied, it is impossible to generalise on the extent of compatibility with high output technology.

Perhaps the most interesting one would be fluid pumping. Technically, this would be broadly compatible because of the propellor law load/speed relationship. Under normal circumstances, the engine air flow requirement may be met by a single stage turbocharging system.

This market segment alone may not be large enough to justify the manufacture of a special product, but it may provide a contribution to the total demand.

Rail Traction

GEC Diesels Limited has a significant presence in the rail traction field. Each British Rail high-speed train has two 12-cylinder Paxman Valenta engines developing 2250 BHP (1678 kW) at 1500 rev/min. Ruston Diesels Limited of Newton-Le-Willows, formerly English Electric, have a history of supplying engines for main-line and shunting locomotives for export and the home market. GEC's interest in rail traction does not rest with engines alone. GEC Traction Limited, manufacture diesel and overhead and third-rail electric locomotives. In the past, Dorman have supplied many engines for the low power end of the market, mainly for small industrial and mining locomotives of powers up to 300 kW. In the 1960's, the company supplied prototype engines for British Rail's Southern Region multiple units. Each engine was rated at about 450 kW. Despite extensive trials, the project was abandoned. Today, the only on-going involvement by Dorman is the supply of 12-cylinder vee engines of 39 litres swept volume to GEC Traction for use in shunting locomotives.

To generalise, the virtues of an engine for this application should be :

1. Low operating costs
2. Reliability
3. Durability
4. Maintainability
5. Low first cost.

First cost is perhaps less important than the other factors, particularly for the larger, main-line locomotives. This is largely because the engine accounts for only about 25 per cent of the total locomotive cost, and usually, the operating costs will exceed the first cost within one year. Furthermore, the design life is often as much as 15 years or 50,000 hours.

Engine first cost, size and weight are important considerations for railcars and multiple units. This is emphasised by the work of British Rail engineers at Derby, who are experimenting with the Leyland "National" bus body in an attempt to reduce production costs and keep the overall weight down. The most interesting development in this field is the prototype "hyperbar" powered, French SNCF railcar (53). This uses two Poyaud 520 8-cylinder engines rated at 590 kW (150 kW naturally aspirated). The railcar works the shuttle service between Paris and Le Touquet airport. The first progress report by the railway operators compared the Poyaud engine favourably with the engine it replaced, although this was a 1955 vintage MGO unit. The significant results were that the engine was well-suited to rapid and frequent changes in load and speed, and that over the first 1400 hours reliability was no worse than the original engine. Fuel consumption was 10 per cent higher than the MGO engine, but the operators claim this was mainly due to governing and gearbox losses.

Transient response is already a problem for the designers of the most highly rated rail traction engines. The demand for acceleration is greatest when leaving a station and over-fuelling due to turbo-charge lag will not only limit acceleration but produce unpleasant exhaust smoke in a public place. This, of course, is not a problem for the turbo-assisted hyperbor concept, which is why such an engine may succeed where an unassisted engine may not. It is quite common for a fuel limiting device to be fitted between driver and engine. This reduces transient exhaust smoke but lengthens acceleration and hence, total journey time. Larger engines have used a mechanical drive between engine and turbocharger (107). Under low load or low speed conditions, the engine drives the turbocharger to provide a given level of charge pressure. A further problem would be controll-ing white smoke emission during the long idling periods to which the rail traction engine is often submitted.

The technical compatibility of the high output concept to rail traction applications is not high, especially when compared with conventional, normally rated engines. Transient response, unless turbo assisted, would deteriorate and operating costs would probably increase. Reliability and maintainability are largely determined by detail design. Although achieving the high standards of reliability demanded by this application may undermine the production cost advantage of high output over conventional engines.

Successful innovation in this market with a high output engine would depend on demonstrating technical compatibility and showing a clear technical or economic advantage over other prime movers.

Marine applications

Marine applications are extremely diverse, covering a wide range of power for propulsion and auxiliary use, burning a wide range of fuels and competed for by most forms of prime mover. Diesel, steam turbine and gas turbine all find usage in different applications. There are, broadly, three classes of application - military or naval, merchant and work-boats, pleasure craft. Military vessels at the low power end include patrol boats, landing craft and in-shore mine-sweepers. Working vessels are typified by pilot and police launches, crew and supply boats for off-shore oil installations, tenders, harbour tugs and in-shore fishing craft.

Technically, the high output concept would be reasonably well-suited to propulsion applications. The propellor-law load/speed characteristic would generally allow the use of a single-stage turbocharger system. The only possible exception here are vessels which change their load/speed characteristic adversely due to towing or even fouling. The availability of a low temperature liquid coolant would also be an advantage. Transient response and startability may be a disadvantage to craft requiring rapid start and acceleration to a high cruising speed, such as launches and patrol craft. A high output engine designed for marine propulsion, with careful attention to detail would be technically compatible with a high proportion of applications.

The attributes of such an engine would be low first cost, overall size and weight. However, these would have to be off-set against a likely increase in operating costs - the magnitude of which is unknown at this time.

An important consideration for both auxiliary and main propulsion engines is the need to meet the detailed standards and rules of the naval engine buyers and marine insurance underwriters. The basis of their standards is that total loss of engine power should be a highly improbable event. The reasoning is that a large investment such as a ship (and its cargo) should not be exposed to the danger of loss of power or propulsion through an unreliable or overrated engine - perhaps costing a fraction of the total. Lloyds of London have devised rules that diesel engines must meet to qualify for insurance by them. Although these cover many aspects of engine design and installation, including fire risk, the most relevant to discussion of high output technology is the crankshaft rating rule. This considers the crankshaft material properties and detail design and determines the allowable loading, defined by maximum cylinder pressure and bmep. This has been used to produce Figure 7.5, which is a graph of bmep against maximum cylinder pressure and gives the maximum rating lines for current standard Dorman crankshafts. Also shown is the approximate operating area of current Dorman charge cooled engines and the high output area of interest.

Today's engine ratings comfortably satisfy the Lloyds formulae. The high output concept exceeds the limit for the in-line six "LE" design, is marginal for the three vee-twelve designs and acceptable for the vee-eight designs. The variation of allowable rating is mainly a function of the crankshaft geometry, particularly the journal and pin overlap.

Figure 7.5 applies to both main propulsion and auxiliary engines. Engines for naval applications have to undergo rigorous type-testing to verify the rating given by the manufacturers and assess reliability, performance and often, maintainability.

Traditionally, marine engines have been modestly rated and relatively low-speed. There are, however, indications that the benefits of the high-speed, turbocharged engine are now more widely recognised (108). Smaller engines require less engine room space and hence, allow more pay-load. Even so, the most highly rated, non-military, marine high-speed engines develop little more than a bmep of 15 bar. In the naval sphere, highly rated engines are required for fast patrol or attack craft (54). However, these generally have an installed power beyond the Dorman range, even in a multi-engine configuration. This application is also often fulfilled to greater effect by the gas turbine.

The company have supplied marine engines for many years and hence have applications engineering and sales experience. The high output concept is broadly compatible with marine propulsion, although meeting the requirements of the insurance underwriters should be carefully considered. The demand for such a product may depend on the willingness of operators to trade increased operating costs for reduced capital outlay, size and weight.

Electrical power generation

A wide range of prime movers are used for the generation of electrical power. These include steam turbines, gas turbines, diesel, petrol, gas and dual-fuel internal combustion engines. On a national scale, the steam turbine is almost unchallenged. Power stations not linked to a national grid, perhaps supplying a single community or industrial site are powered by gas turbines or large diesel engine sets. Below 1MW the high speed diesel engine is most widely used, the only exceptions being a minority of gas and small petrol engines.

The duties for diesel generators in private industry below 1MW include base load, peak lopping and mains failure. The application may be stationary, for example in hospitals, office blocks or manufacturing plant, or mobile, powering floodlights, welding sets, or starting aircraft. In the last decade the industry has flourished, especially in the lucrative developing markets such as Nigeria, Iraq and Iran. These countries do not have electrical distribution networks but require electrical power locally for their developing industries and community services. Towards the end of the 1970's, upheaval in Iran and heavy import tariffs by Nigeria have drastically reduced demand by these markets. However, the underlying demand for diesel generators is said to be strong (5) despite this setback.

Developed countries have demand for generating sets but generally have a strong indigenous industry. In the UK, the loss of mains electricity on occasion, through industrial action, has increased the demand for mains-failure equipment. There is also a steady demand for generating sets for hospitals, computer centres, telecommunications and public buildings, where loss of power would present a hazard eg.those with lifts.

The high-speed diesel generator industry in the UK comprises 35 to 50 small to medium companies, the largest, employing about 200 to 400 people. Typically, these companies buy engines and alternators on the open market, and mount them together on a base-frame with the necessary controls and ancillary equipment. Most offer a standard range of products featuring engines from two or three manufacturers. Some of the larger companies also produce gas turbine and medium-speed diesel engine sets. The more sophisticated companies offer complete installation, commissioning and after-sales service.

Naturally aspirated, turbocharged and charge cooled engines find application in this market segment. No commercial market has exploited turbocharging more than power generation. The most highly rated, charge cooled engines available are used, particularly in mains failure sets. For this reason, the power generation segment was chosen for closer study. This involved visiting and talking to set builders and engine makers, and the use of the postal questionnaire as shown in Figures 7.6(a) and (b)

The objectives of this study were :

1. To determine the factors that bear upon the buying decision and their relative importance
2. To assess the compatibility of the high output concept with power generation applications

3. To further understanding of the power generation market.

The questionnaire was an integral part of this study, aimed at elucidating some of the above questions. The demand for industrial products derives ultimately from a social need, although this may be quite remote from the product being manufactured. For example, the needs for mobility and health care result in road-building and hospital projects, which in turn require earthmoving and emergency power equipment - which require diesel engines - which require raw materials, etc. In the "ideal" market study, these patterns of demand would be of great interest. However, this is rarely a practical proposition. This study attempts to make use of the market knowledge acquired by the set makers and rests largely on the assumption that they more closely understand the satisfactions of the end users. The end users were not studied because they are diverse and difficult to locate, particularly the buying-decision makers.

The companies invited to take part in the study are listed in Table 7.3. This shows those that responded, those choosing not to, despite reminders, and those refusing on commercial grounds. Of the 28 companies approached, 13 returned the completed questionnaire, and all respondents claimed to be involved with selecting engines for their companies' product range. A description of the questionnaire design is given in Appendix 8.

The results of the postal questionnaire are given in Figure 7.7 (a), (b) and Figure 7.8 (a) and (b). The data has been analysed in two groups. The first group includes all 13 respondent companies, the second includes only the 7 largest companies. The total annual production of the 13 companies was approximately 8000 units. The 7 largest produced a total of about 4100 units. This subdivision is thought to be informative because it largely excludes the manufacturers of small generators using such engines as Petter, Lister and Perkins. The 7 companies, whilst not exclusively using engines of the Dorman power range, are heavily committed to the larger, high-speed engines typified by Cummins, Volvo, Rolls Royce, Deutz and MAN. The 7 companies included in this sub-group were :

1. Auto Diesels Braby Ltd.
2. Dawson Keith Ltd.
3. Elequip Ltd.
4. Petbow Ltd.
5. Grahame Puttick Ltd.
6. Thomas W. Ward Ltd.
7. Welding Industries Ltd.

Considering first the general data, Figure 7.7 (a), relating to the 7 largest companies shows that a very high proportion ($\simeq$ 85 per cent) of the electrical generators produced by this group operate at 1500 rev/min. Mains failure duty is dominant, accounting for over half of the total production. The data emphasises the requirement for rapid uptake of load in the event of mains-failure. Although no-break applications are rare, about 50 per cent require the restoration of electrical service within 10 seconds, and over 80 per cent within 20 seconds. The technology used by this market segment is perhaps the most advanced. Turbocharged and charge cooled variants account for approximately 40 per cent of applications each, whilst naturally aspirated engines fulfill the remaining 20 per cent.

The relative importance of the technical and economic parameters listed in the last question, as perceived by the respondents, is given in Table 7.8 (b). The most important design-influenced parameter for mains-failure sets are :

1. First cost
2. Reliability
3. Mains down-time
4. Noise levels

These are not entirely compatible one with another, and are not all improved by the adoption of the high output concept. Of the four, only first cost per kW is likely to offer any advantage over current products. Mains down-time will deteriorate until both start aids and turbo-assistance are used, thus undermining the cost advantage. In the context of mains-failure, successfully starting and accepting the load would have great influence on overall reliability. Thus, the need for start aids, however well-engineered, would probably reduce the reliability of this phase of the duty cycle. Noise levels, too, would normally be higher than the existing products, both through combustion generated noise at part loads and mechanical noise, not least from the turbocharger.

The most important non-technical factors are :

1. Spares and service support
2. Reputation of engine maker
3. Delivery period.

Baseload and peak-lopping applications would require different engine attributes to mains-failure applications. The emphasis would probably be on operating costs, service life and reliability, with less exacting rapid start requirements.

Technically, the high output concept is compatible with power generation applications. The fixed speed characteristic would allow the use of a single-stage, turbocharging system up to a bmep of 21 bar and possibly higher. The load characteristic, which can impose step changes in torque demand for motor starting, etc. would be a problem because of turbocharger lag. A two-stage, turbocharging system or turbine-assist device such as an oil-driven pelton wheel, air injection or exhaust combustor, may improve load acceptance but only by increasing first cost. The need for rapid starting for mains-failure sets, with a reduced compression ratio, would require additional measures or a combination of measures. For example, block heating or manifold heating.

A high proportion of the engines currently used by this market are highly rated charge cooled engines with "marginal" compression ratios of 13.3 to 14 : 1. Many of these require start aids under normal engine room conditions. The generating set manufacturers consulted during the study suggest that if a "low" compression ratio engine were fitted with a reliable and effective start system, then it would be acceptable to the market. A concern expressed by some in the industry is that "sophisticated" engines are not well adapted to the low level of maintenance, service and operative skills in some developing countries. However, the difference between a high output and normally rated engine is largely a question of degree rather than design principle.

The company has supplied many engines for this market in the last decade and in 1979, recommenced generating set manufacture and marketing in their own right.

7.5. Summary

The foregoing discussion of the commercial feasibility of the high output concept was essentially a "coarse screen" intended to remove those market segments of low potential. The discussion is summarised below in Figure 7.9, which uses a subjective scale to rate the three screening criteria for each segment. Those lacking adequate information for even a subjective assessment have been left blank.

Market segment	Technical compatibility	Market need	Company resources
Construction and earthmoving	average	poor	average
Automotive			
On-highway	average	-	very poor
Off-highway	average	poor	average
fighting vehicles	good	-	average
Rail traction	average	poor	very good
Industrial	average	-	good
Marine			
Propulsion	good	average	very good
Auxiliary power	good	average	very good
Auxiliary general	average	-	good
Generating			
Baseload & peak-lopping	good	-	very good
Mains-failure	good	good	very good

Figure 7.9. : Summary of coarse screen of diesel engine market segments.

This coarse screen is rather crude, much more detailed and quantifiable data could be obtained, such as :

1. Effect on existing product range
2. Market size
3. Expected sales/cost/profit relationships
4. Required investment.

However, at this stage, a more detailed approach would be inappropriate and unprofitable. This evaluation is based on discussions with users, OEM's and manufacturers, sales literature and articles, and visits to exhibitions.

The outcome of the commercial study is not encouraging and the scope for introducing such a new product is, at present, limited. The areas which are most interesting are :

1. Fighting vehicles
2. Marine auxiliary power and propulsion
3. Electrical power generation.

Table 7.3. : Companies participating in the questionnaire

1. Hampson Automation Ltd.		N	
2. Graham Puttick Ltd.	Y		
3. Elequip Ltd.	Y		
4. Thomas Ward Ltd.	Y		
5. Jonlaw Engineering Ltd.	Y		
6. Wysepower Ltd.		N	
7. Auto Diesels Braby Ltd.	Y		
8. Newton Derby Ltd.	Y		
9. Power Units Hindle Ltd.	Y		
10. Workman Reed Ltd.	Y		
11. Countryman Power Plant Ltd.		N	
12. British Int. Industries Ltd.		N	
13. Fitzcroft Ltd.		N	
14. Sentinel Power Systems Ltd	Y		
15. Shannon Power Services Ltd.		N	
16. Mitchell Diesel Ltd.			R
17. G & M Power Plant Ltd.		N	
18. Dale Electric Ltd.			R
19. Welding Industries Ltd.	Y		
20. Atlanta Engineering Ltd.	Y		
21. Lewis Electric Generators Ltd.		N	
22. RTD Swan Ltd.		N	
23. IPS Ltd.		N	
24. Transunits Ltd.		N	
25. Petbow Ltd.	Y		
26. Raynar Ltd.			R
27. Dawson-Keith Ltd.	Y		
28. Hawker-Siddeley Power Plant Ltd.			R

Y - Response

N - No response

R - Refused on commercial grounds

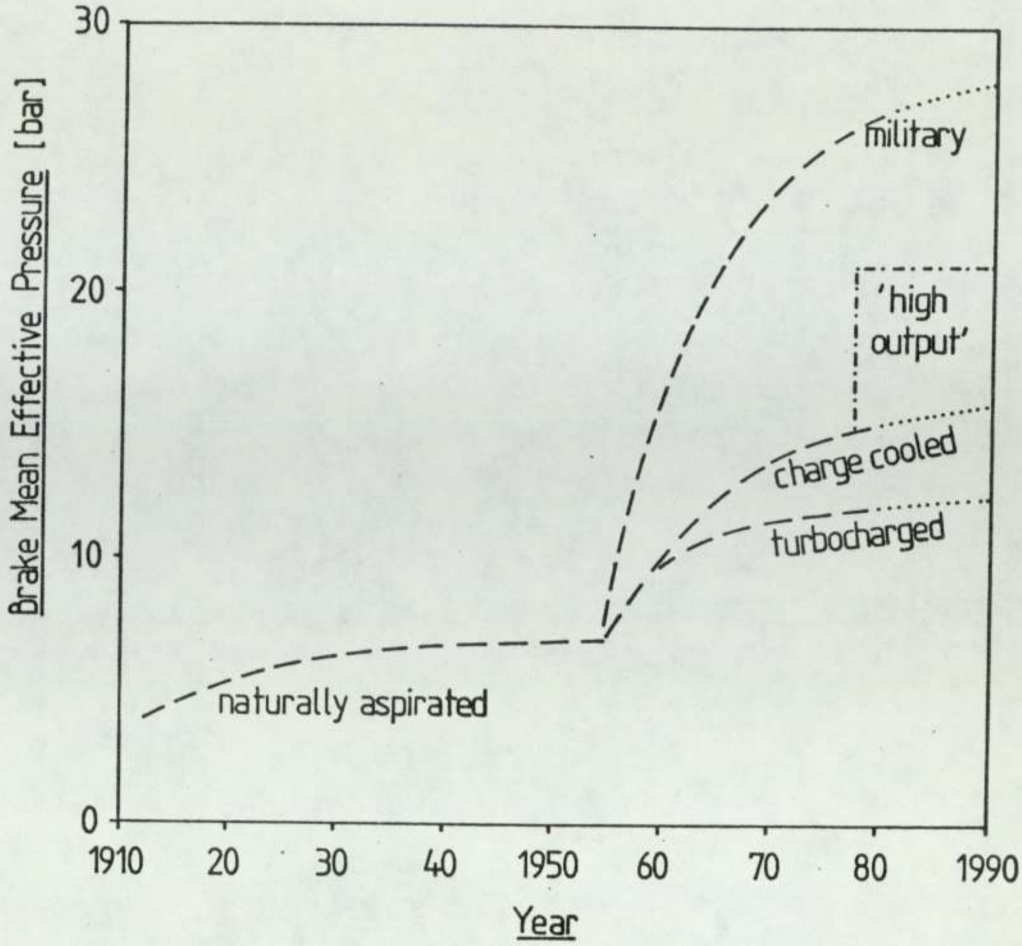


Fig.7.1 Progress of the high-speed diesel engine, highly simplified, showing the relationship of the high output concept to current trends.

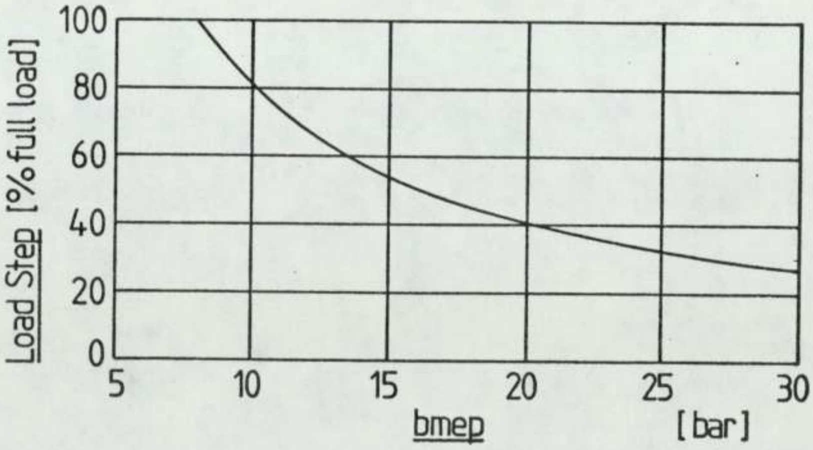


Fig.7.2 'Maximum possible' sudden load increase, from zero, for turbocharged, 4-stroke engines, as a function of bmep. [from draft ISO/DIS/3046/ IV]

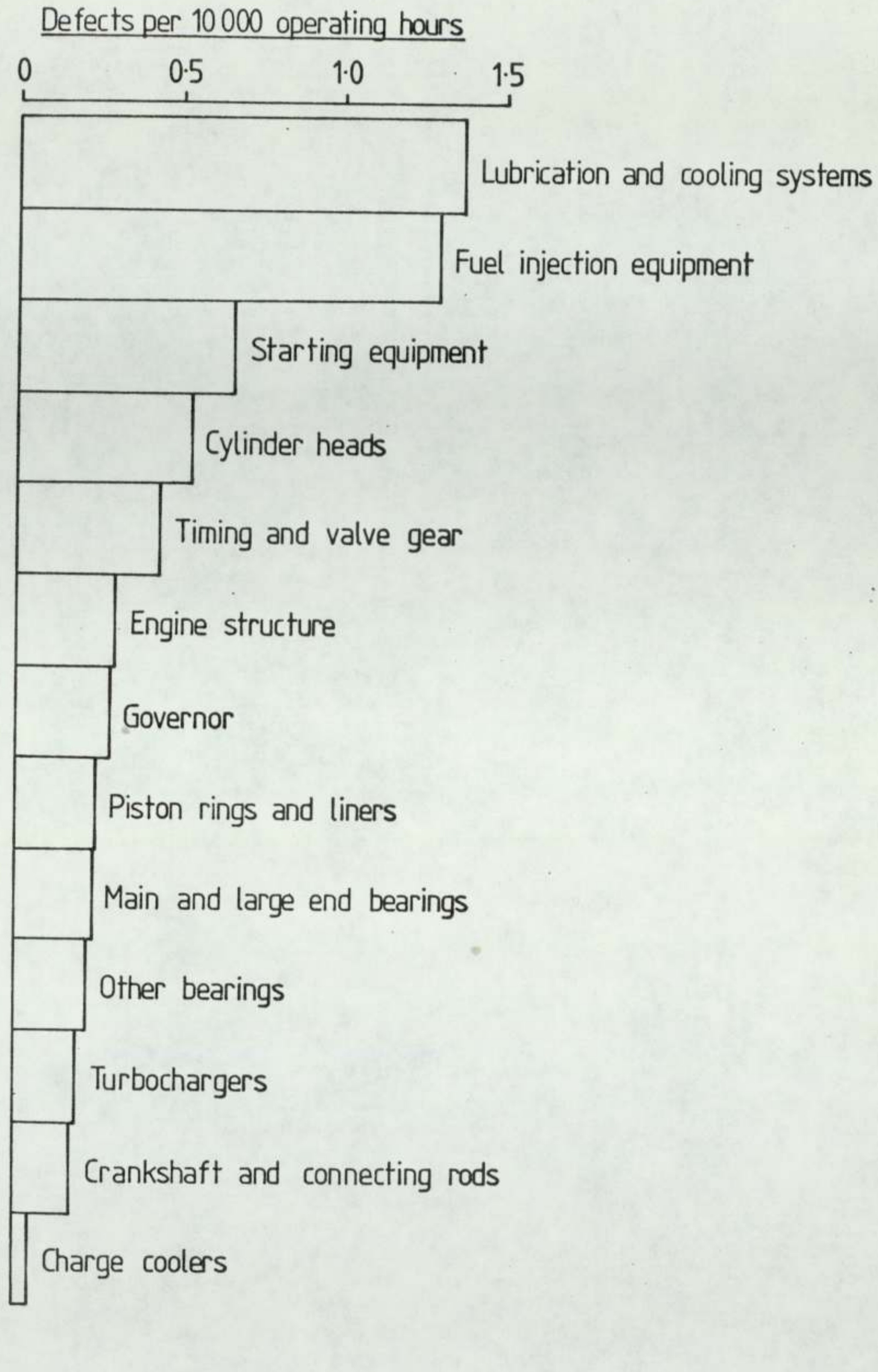


Fig.7.3 Contribution of the various components and systems to the unreliability of a conventional diesel engine, from ref(101).

engine —	6LE	8J	8Q	12J	12Q	12S
bore [mm]	127	130	158·75	130	158·75	158·75
stroke [mm]	149·2	129	165·10	129	165·10	165·10
configuration	str.6	90° vee 8	90° vee 8	90° vee 12	90° vee 12	90° vee 12

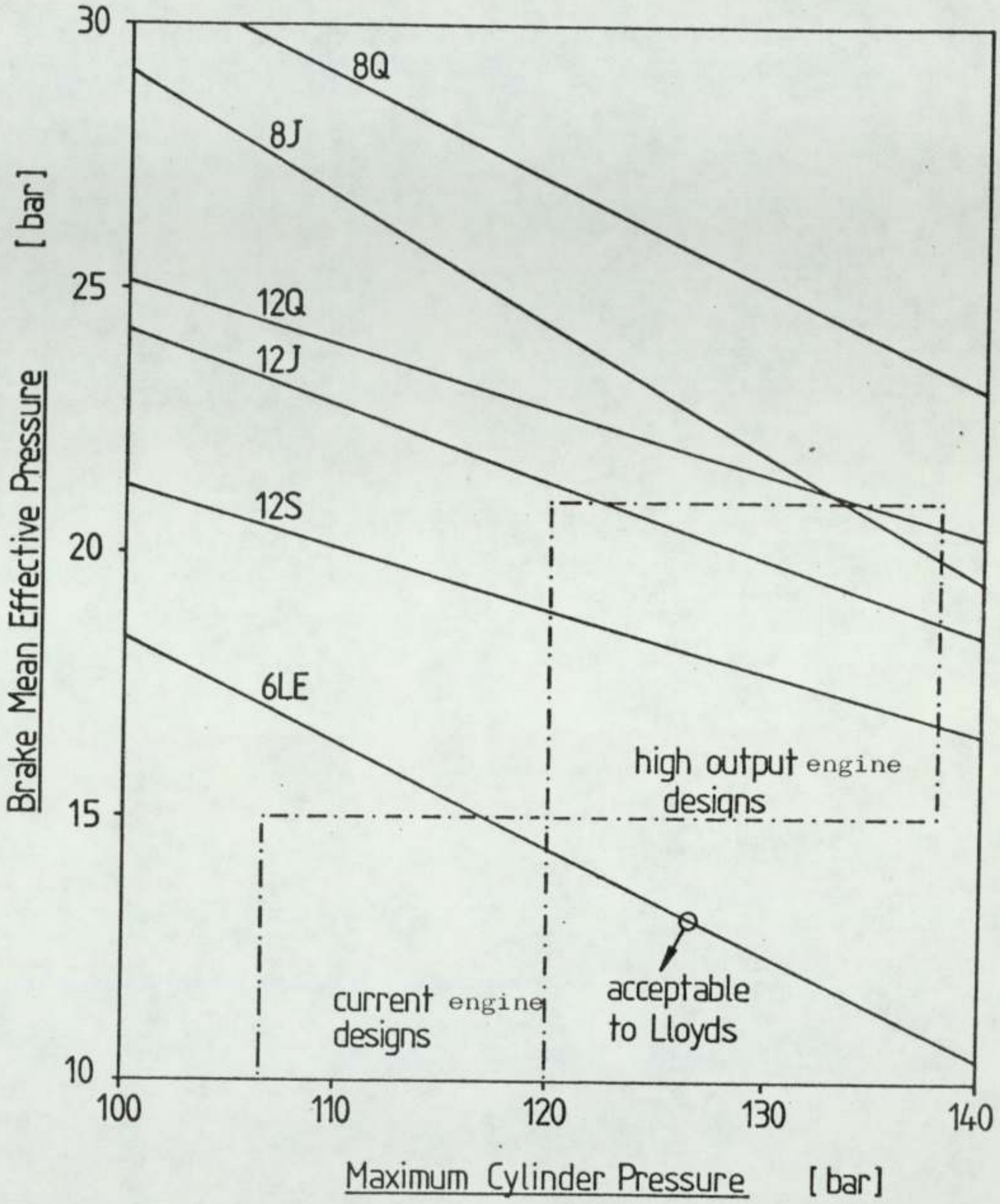


Fig.7.5 Maximum rating of current Dorman crankshafts, to meet the 'rules' of Lloyds of London.

THE UNIVERSITY
OF ASTON
IN BIRMINGHAM

Approximately how does your Company's production sub-divide by the five classifications given below?

1. Set Application	MAINS FAILURE	☐ %
	BASELOAD	☐ %
	OTHERS?	☐ %
2. Diesel Technology	NATURALLY ASPIRATED	☐ %
	TURBOCHARGED	☐ %
	TURBOCHARGED & INTERCOOLED	☐ %
3. Set Application	1500 RPM (50 Hz)	☐ %
	1800 RPM (60 Hz)	☐ %
	D.C. MACHINES	☐ %
	OTHERS?	☐ %
4. Markets	HOME	☐ %
	OVERSEAS	☐ %
5. Mains Failure Sets	NO BREAK	☐ %
	LOAD-ON IN UNDER 10 Sec.	☐ %
	LOAD-ON IN UNDER 20 Sec.	☐ %
	LOAD-ON IN UNDER 60 Sec.	☐ %

Fig.7·6(a) Marketing questionnaire (sheet 1)

Rate the following engine parameters according to their relevance to selection for your Company's STANDBY Generating Sets.

	Very Important				Unimportant
Noise level	5	4	3	2	1
First Cost (£/kVA)	5	4	3	2	1
Weight	5	4	3	2	1
Reputation of engine maker	5	4	3	2	1
Operating cost (£/kWh)	5	4	3	2	1
Exhaust emissions	5	4	3	2	1
Overall engine size	5	4	3	2	1
Service life	5	4	3	2	1
Delivery period	5	4	3	2	1
Load-on time from cold	5	4	3	2	1
Spares service back-up	5	4	3	2	1
Fuel consumption (Kg/kWh)	5	4	3	2	1
Engine reliability	5	4	3	2	1

Do you have any comments to add? ....

....

....

....

Name of Company ....
Range covered ....
No. of units produced annually ....
Name of respondent ....
Position held ....
Do you contribute towards engine selection? Yes/No

Fig.7·6(b), Marketing questionnaire (sheet 2)

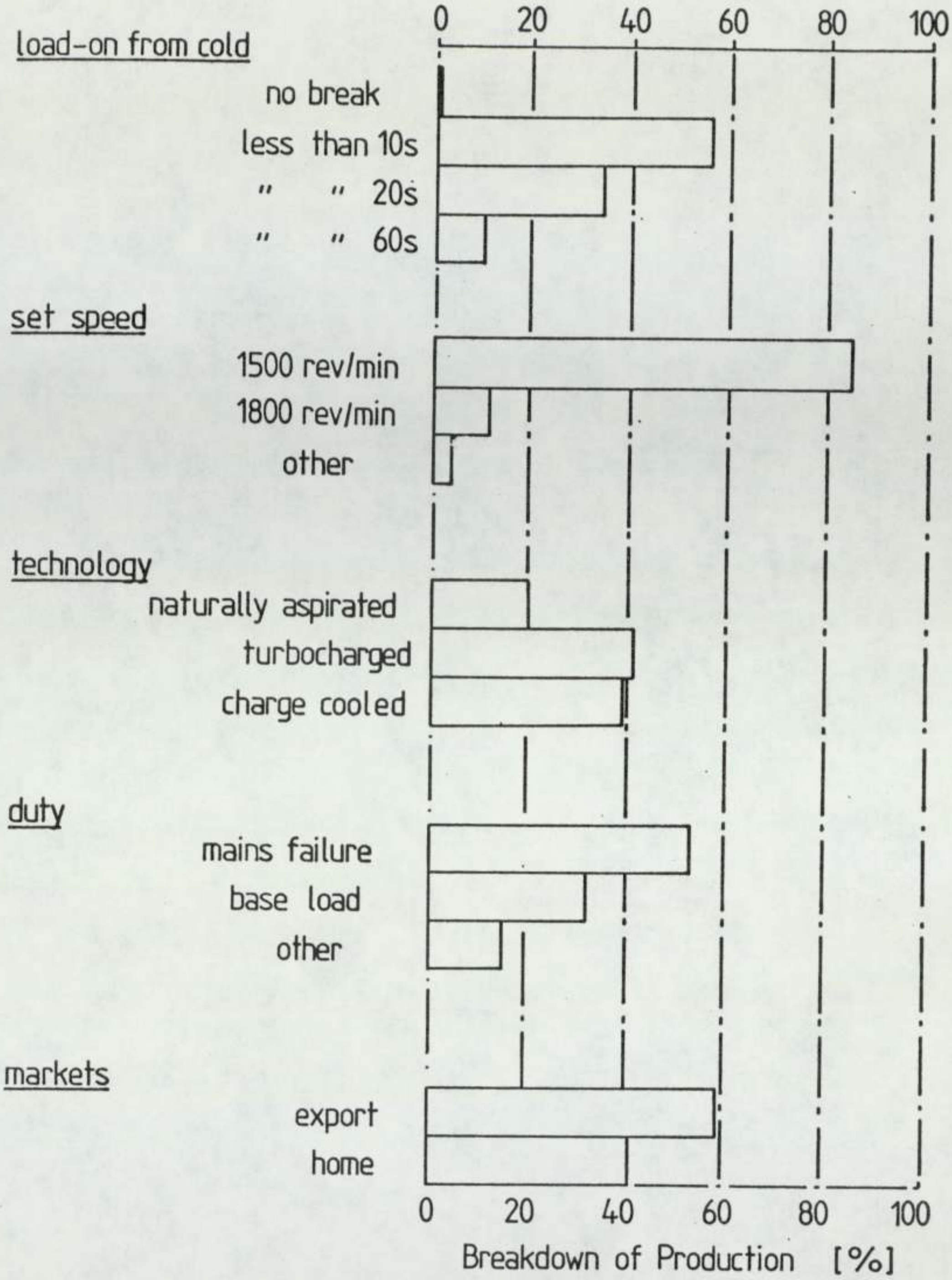


Fig.7.7 (a), Subdivision of production of the seven largest companies in the questionaire sample.

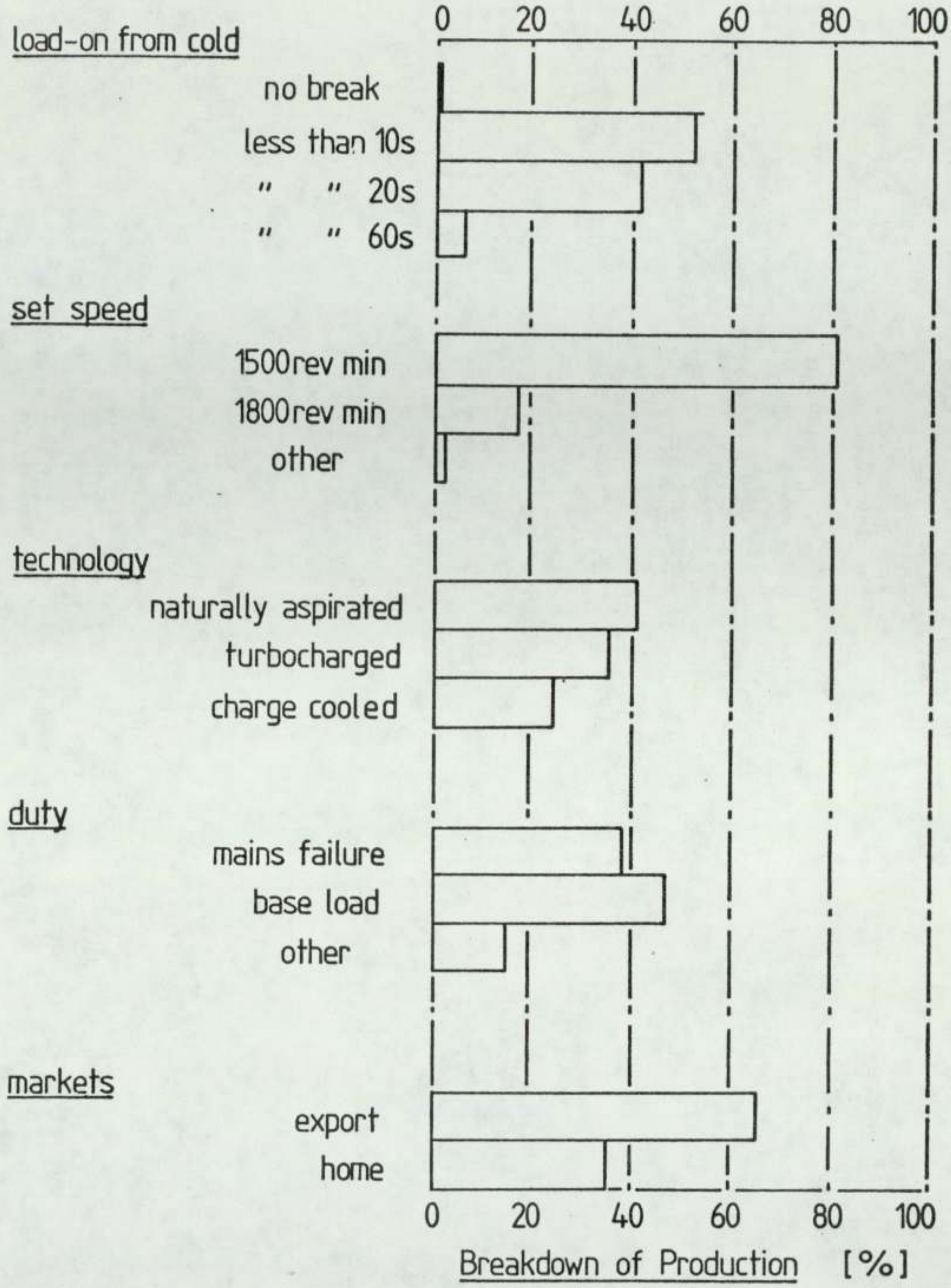


Fig.7.7 (b), Subdivion of production of all questionnaire respondents.

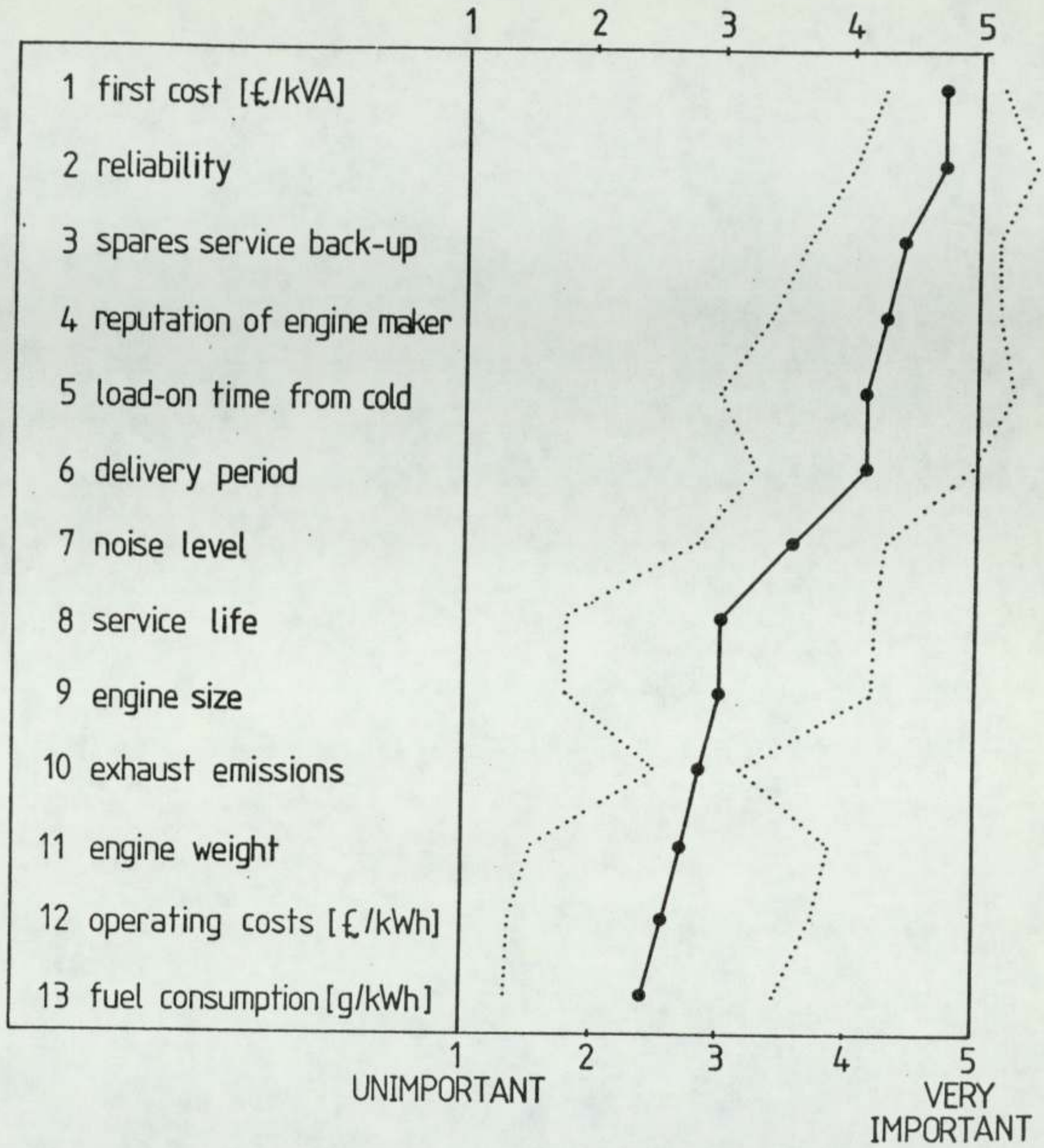


Fig.7.8 (a), Rank and relative importance of 13 factors, to the selection of an engine for mains-failure duty, showing the standard deviation. [7 largest companies]

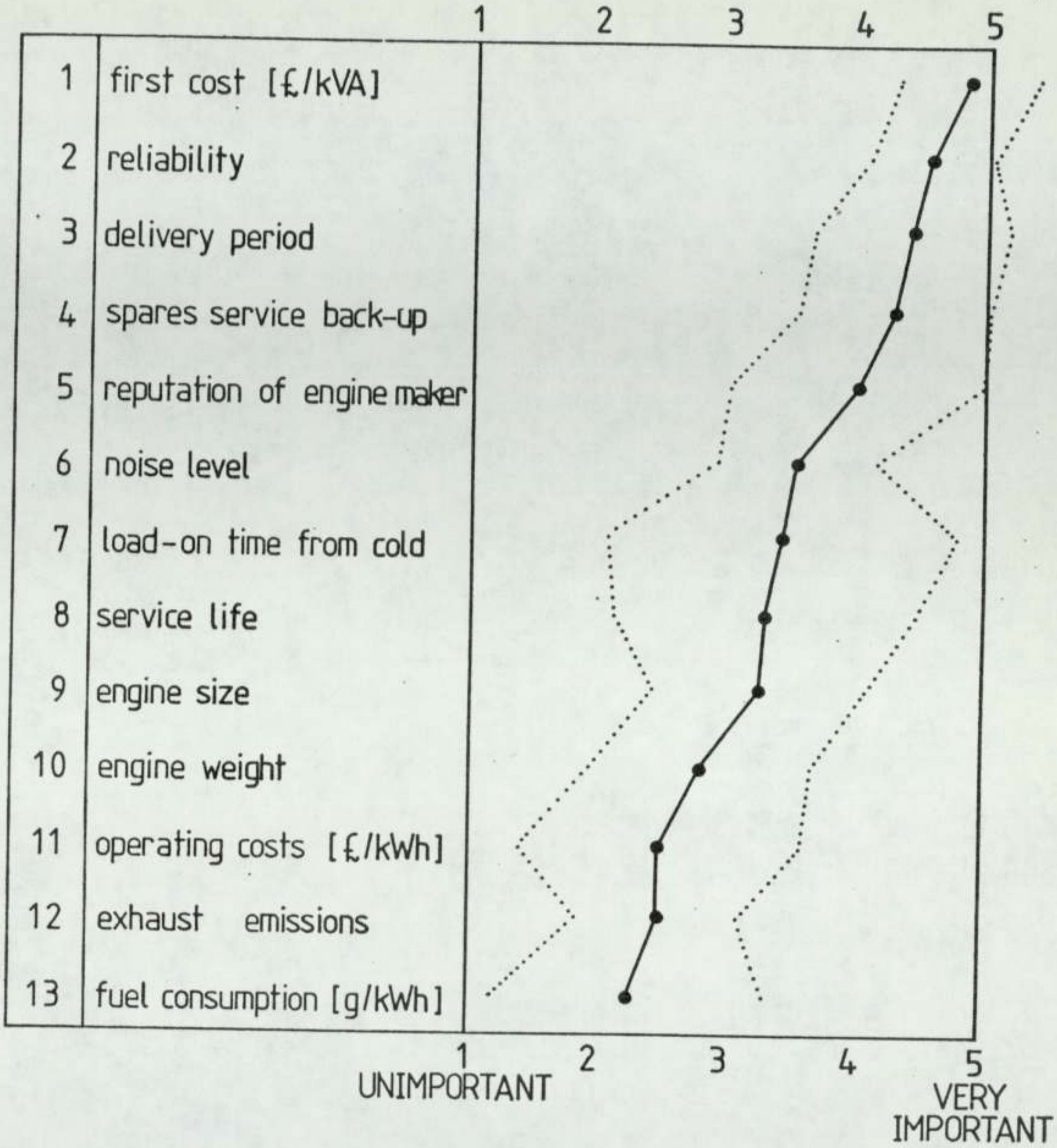


Fig.7.8 (b), Result of all 13 replies, showing standard deviation.

CHAPTER 8

DISCUSSION AND CONCLUSIONS

8.1. Contribution to Knowledge

It is difficult to measure the effectiveness or "productivity" of research, since its output is knowledge, unlike most other functions of business whose output is more readily measured in monetary terms. It is useful, however, to ask - what have we learned ? This may be split broadly into two areas : firstly, the knowledge that directly serves the objectives of the research, and secondly, knowledge gained incidentally in performing the research.

Perhaps the only knowledge of interest in pure research is that which was previously unknown to mankind, no matter how seemingly un-useful that knowledge may be. In applied research, the area in which the knowledge is sought, and the purpose to which it will be put, are more closely defined. Research in an industrial organisation is rarely aimed at creating new knowledge, more usually it seeks to apply existing external knowledge to the activities of the organisation in an original way. That is the context of this project.

The high output concept is not new to the field of knowledge, nor is it totally new to Dorman Diesels Limited. However, applied to the "LE" engine with specific performance and design criteria, and a view to commercialisation, it is both new and meaningful.

The main objective was to increase the company's knowledge and experience of the high output concept as an aid to sound decision-making. Undoubtedly, knowledge has been increased, but to what extent ? The concept is not now fully understood, therefore decisions could not be based on perfect information. Neither technical nor commercial study is complete. Realistically, the project has given the company a technical and commercial introduction to the concept. Technically, it has demonstrated that a high level of bmep is achievable in short-term laboratory operation, using not especially "sophisticated" design solutions and readily available components. Commercially, the project helps to elucidate a complex problem and accomplishes the first stages of a formalised new product appraisal procedure.

This work increases the information base, allowing management to more rationally decide whether to continue the project to find out more, or to terminate it.

In performing the research, techniques new to the company were used to study the underlying trends of engine design and performance, and to assist the specification of a working prototype. This project is an example of the way that computer modelling may be used to increase design effectiveness and minimise costly test bed development.

8.2. Technical Discussion

At the start of the project, few of the technical questions relating to high output operation could be answered with any certainty. For example, is low compression ratio synonymous with low thermal efficiency ? Does high bmep imply high component temperatures ? And can high output only be achieved with "sophisticated" design solutions ? At the end of the project, some of these questions have been answered, some have not. In particular, questions of durability over 5000 or 10000 hours, transient response and startability remain unanswered.

The brake thermal efficiency of the research engine was consistently in the range 37 to 38.5 per cent at full load, including the "low" compression ratio (10.7) build. Allowing for the addition of a cooling fan, the specific fuel consumption would be between 229 and 237 g/kWh (.376 - .39 lb/bhph). This compares well with the quoted specific fuel consumption of the current charge-cooled "LE" production engine which is 221.4 g/kWh (.364 lb/bhph), at a bmep of 13.7 bar, at the same speed. Considering that the research engine was constructed of production-quality basic components (pistons, cylinder liners, etc.) not selectively fitted, and did not have an optimised fuel system, this result is encouraging.

High thermal efficiency is fundamental to good engine design for several reasons. Firstly, the diesel engine's success is due mainly to its economic use of fuel. For the majority of applications, fuel economy is a considerable selling point. Secondly, the less fuel that is injected to produce a given output, the lower the thermal load, the smaller and less expensive the cooling equipment need be, and the lower the air demand for a given air-fuel ratio.

The use of a wide valve overlap period (127^{o}CA) gave the advantages that were forecast during the design study. However, certain disadvantages arose, such as an increase of exhaust smoke which greatly influenced its feasibility. A list of the advantages and disadvantages of high valve overlap is given below.

Advantages of high valve overlap

1. Scavenging of the clearance volume.
2. Increase of trapped volumetric efficiency.
3. Reduction of turbine inlet temperature.
4. Probable cooling of exhaust valve and valve bridge.

Disadvantages of high valve overlap

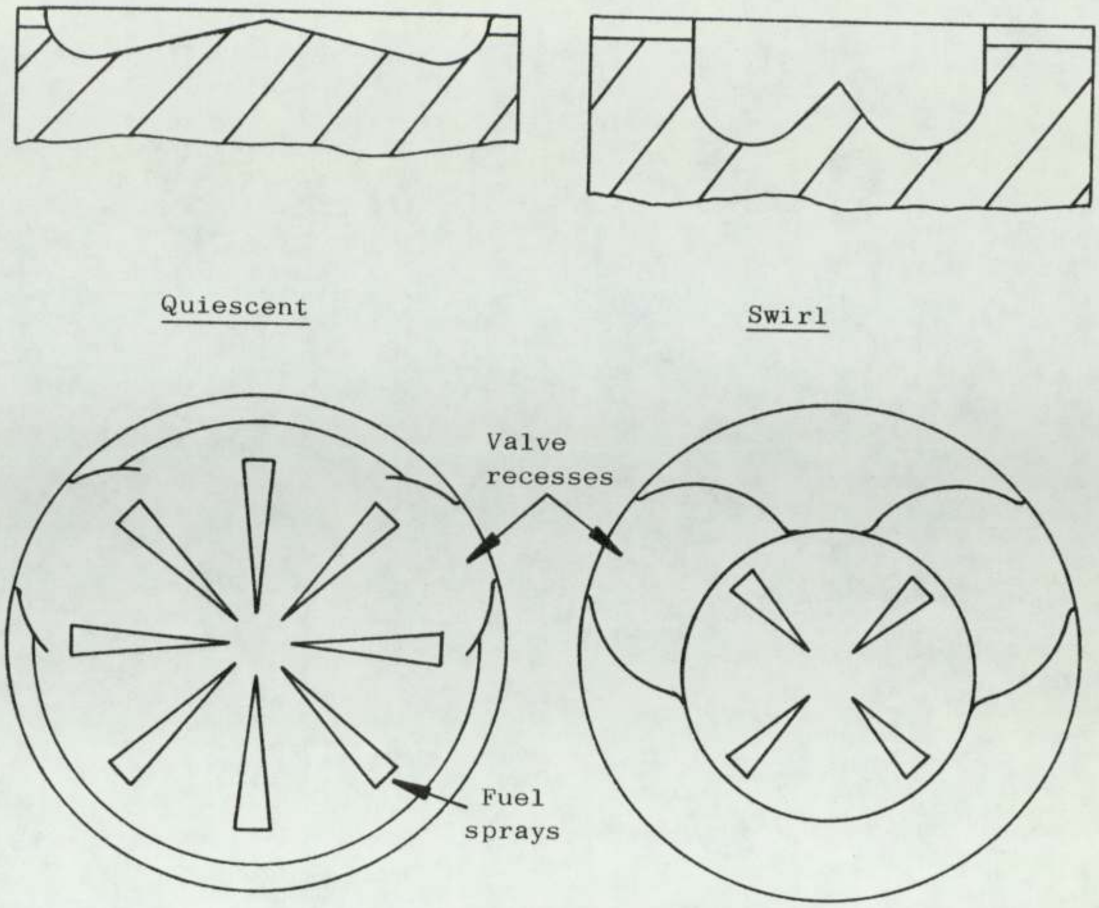
1. Need for larger components in the induction system to accommodate the increase of air flow.
2. Larger compressor may mean higher rotor inertia, especially if frame size is increased.
3. Larger, more expensive charge cooler.
4. Possibility of flow reversals, particularly under light load operation.
5. Need for valve recesses, which is thought to cause exhaust smoke.
6. Reduction of the pressure drop across the cylinder (necessary for turbocharger power balance).

Valve overlap exploits the favourable pressure gradient that exists across the cylinders of an efficiently turbocharged engine. The argument for or against its use is not straightforward, at least, not for high speed engines. The problem centres on the combustion system.

The typical high-speed, direct injection, diesel engine combustion system uses a deep bowl, usually toroidal, in the piston crown. This is necessary to promote swirl and squish during fuel injection and combustion to effect air-fuel mixing. Because the bowl is deep, it has a diameter often only 50 or 60 per cent of the cylinder diameter. Thus, only a small fraction of the plan area of the valves is directly above the chamber. If recesses are needed they must necessarily contain a signficant percentage of the total clearance volume. This is thought to lead to the inaccessibility of part of the charge, leading to the emission of smoke.

Larger engines in the medium-speed class usually have quiescent combustion systems. The charge is not encouraged to move rapidly and air-fuel mixing, which is no less important, is achieved by dispersing the fuel throughout the chamber, using a large number of fuel sprays. The combustion bowl is normally much shallower and covers most of the piston crown. Thus, valve recesses are less of a handicap to this form of chamber, since they enclose only a small fraction of the available air, and the low level of charge motion is not adversely affected by them. A diagrammatic comparison of the two systems is shown overleaf.

It would, of course, be possible to achieve high output without



significant valve overlap. However, there is little doubt that the research engine as built, would not have achieved the target output without valve overlap. Output would have been limited to a bmep of about 18 bar, primarily because of high turbine inlet temperature. This could be offset by increasing the rate of injection and/or the air-fuel ratio. The first is achievable (Section 6.4) and probably desirable with or without valve overlap. The second would lead to higher cylinder pressures or a reduction of compression ratio. The absence of clearance volume scavenging would reduce the trapped volumetric efficiency and so necessitate higher manifold pressure for the same air flow, again increasing mechanical load. High valve overlap is not essential for high output operation, but when the traditional diesel engine virtues of economy, durability and reliability are sought, its advantages should not be rejected lightly.

Two alternatives are possible if wide valve overlap is to be used in high output variants of Dorman engines :

1. Retain the deep bowl, high swirl combustion system and attempt to optimise the details of chamber shape, air motion and injection to reduce exhaust smoke
2. Convert to a form of shallow bowl, quiescent combustion system.

Neither of these options is certain to provide a solution. The first would require an extensive development exercise involving "trial and error" methods, which would almost certainly lead to some improvement. However, Howarth (47), p.57, suggests that a satisfactory solution does not exist. The second approach is more radical, but if it could be adopted, other benefits would accrue. For example, the heat loss from the cylinder would be signficantly reduced, easing thermal load, and the removal of the means of producing swirl (valve mask or contorted port) would give an increase of volumetric efficiency.

The level of air motion required by an engine is broadly related to the time available for air-fuel mixing and combustion. The small, high-speed engines (3000 rev/min) generally require high levels of charge motion, and often employ swirl pre-combustion chambers to achieve it. Truck engines in the speed range 1500 to 2200 rev/min. need less charge motion, often employing swirl of only 1.5 to 2.0 times engine speed. Engines of 150 to 200mm bore need little or no air movement.

For the larger Dorman engines of 125 to 160mm bore, it would be contrary to conventional wisdom to attempt to use a quiescent chamber. However, Mansfield (7) found that at high bmep, air motion is neither necessary nor desirable.

Equally important to the success of the quiescent system is the fuel injection equipment. Unlike the classical deep bowl system used in all current Dorman engines, which typically uses 4-hole injector nozzles, the quiescent system requires between 5 and 9 holes. To maintain comparable spray quality, the hole diameter must be reduced as the number of holes increases. If the 4 x 0.4mm diameter nozzle used in the experimental programme, were to be replaced by an 8-hole nozzle of the same flow area, the hole diameter would be only 0.28mm. This would cause manufacturing difficulties, but nozzles with holes of only 0.25mm diameter are in production for the smaller high-speed diesels.

Although there was insufficient time available for a thorough investigation of the effects of thermal and mechanical load on durability, some observations will be discussed.

High bmep operation implies high pressures in the inlet and exhaust manifolds. This pressure level is one of the chief differences between high output and normally rated engines. It could also be a source of considerable difficulties. The least serious encounttered during the experimental programme, is sealing the joints in both inlet and exhaust manifolds. The standard "LE" seals proved inadequate for pressures of 2.75 bar and above. Particularly vulnerable was the joint with the turbine which has a double entry, although in turn, each of the exhaust manifold joints failed. This is probably only a question of detailed flange and joint design. On the inlet side, the conventional hose and jubilee clip connection was equally inadequate at high output. A more appropriate solution is the "dog-bone", which uses "o" rings to seal against a turned inside diameter on each component.

A more serious problem resulting from the high pressure in the manifolds was discussed in Section 6.5. - valve stem lubrication. Oil will not penetrate the lower reaches of the valve guide against the boost and exhaust pressures required for high bmep. In the research engine, the problem was exacerbated by the lack of valve rotation, which also causes an uneven distribution of thermal load on the valve head. British Patent 1 412 075, filed by the French Defence Ministry, in connection with the development of "hyperbar", describes a means of overcoming the adverse pressure gradient that exists along the valve stem. It consists of a recess in the inner diameter of the valve guide, a small distance from the lower end. This is connected via drillings to an external groove that runs from the recess up to the rocker box. This relieves the pressure and allows the upper portion of the stem to be normally lubricated. The patent also suggests the possibility of a pressurised oil feed to the recess.

The thermal load of an engine component is a function of its absolute temperature and/or temperature gradients. Although the absolute temperature of a component may not exceed prescribed limits, the gradients present may be steep enough to cause thermal fatigue failure in the longer term. The components of the research engine were generally at acceptable absolute temperatures, with the exception of a localised area of the cylinder head (Section 6.5). The thermal gradients in the piston crown area, which was cooled by an oil jet, and the exhaust valve head near to the seat face, were higher than in a normally rated engine. This could be problematic if the engine were operated for a long period or subjected to load cycles. The seriousness of this effect could only be assessed accurately by controlled endurance testing. If problems did arise, improvement could be effected by placing the heat sink nearer the source : for example by using "cocktail shaker" piston cooling instead of undercrown cooling, and direct valve cooling (perhaps using sodium) to supplement the indirect cooling through the seat and stem. Such measures would, of course, impact upon the product cost.

8.3. High Output Design and Market Needs

The structure and running gear of the conventional high-speed diesel engine are remarkably versatile. Typically, the same basic components are used to produce the naturally aspirated, turbocharged and charge cooled variants. The high output concept aims to extend this power range still further. The "LE" naturally aspirated engine develops 97 kW at 1500 rev/min, whereas the research engine produced almost 300 kW at the same speed, with the same basic structure and running gear. Twenty-five years ago when the naturally aspirated engine was unchallenged, the designer had little scope to vary the characteristics of the product. By contrast, turbocharging gives the designer greater freedom to balance engine characteristics in a way that more closely meets the needs of specific market segments. Consider the versatility of the Rolls Royce CV8 which is sold in different configurations for on-highway trucks, electrical power generation and fighting vehicle applications. Figure 8.1 presents some of the design options that are possible. On the basis of these

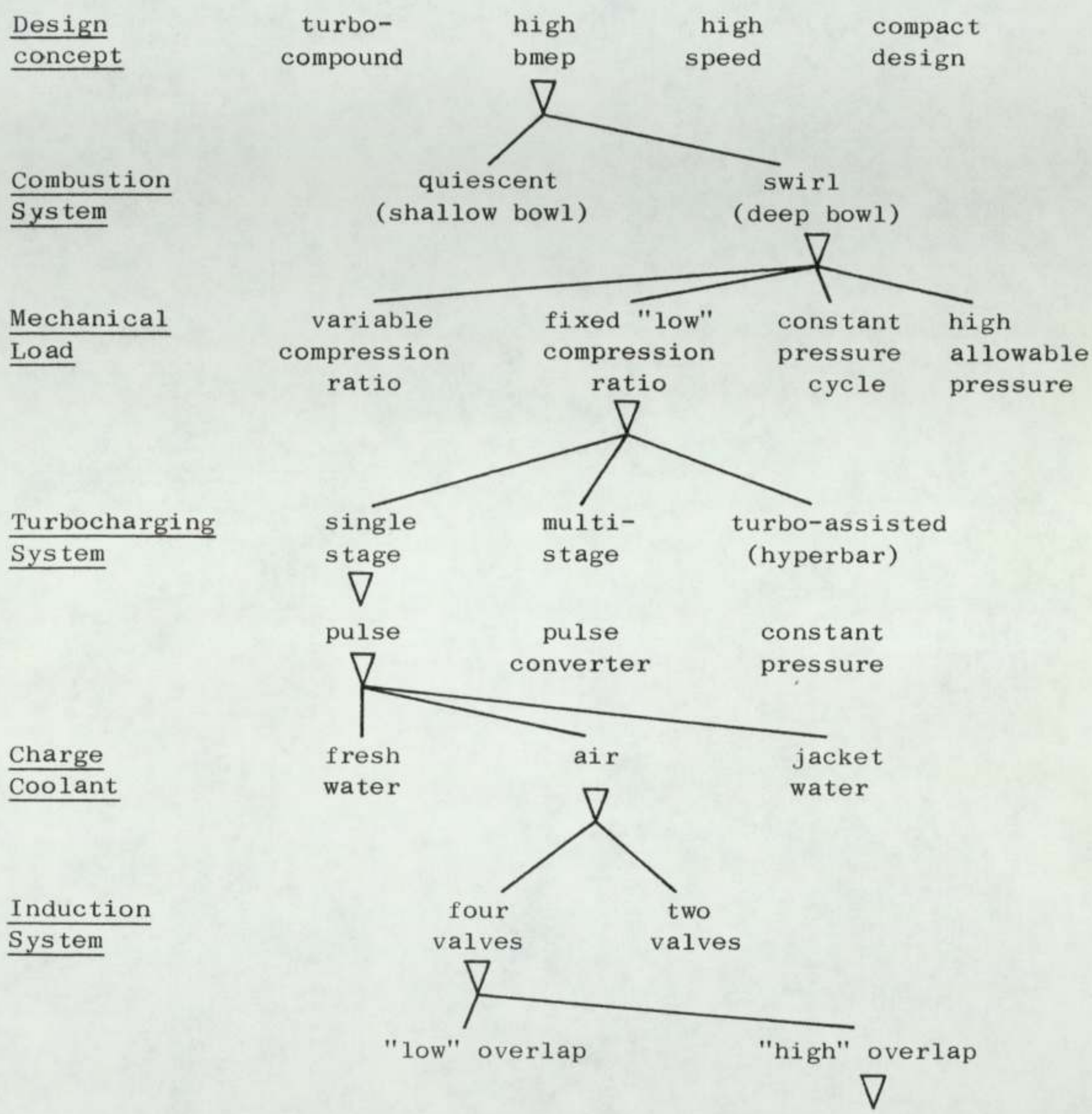
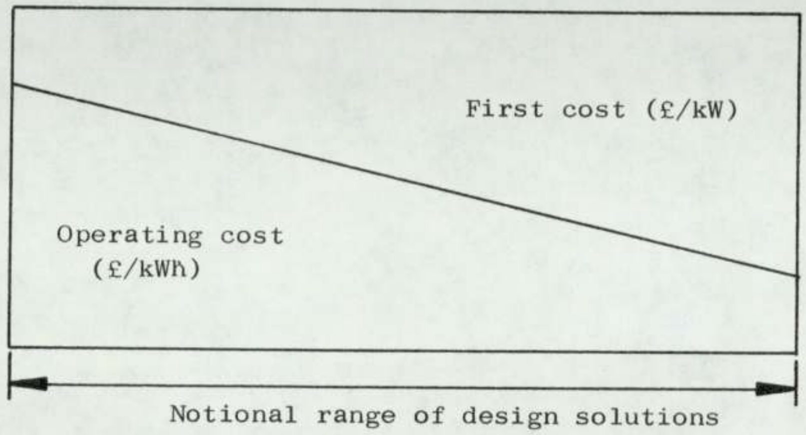


Figure 8.1. : The range of design solutions for high specific output is very wide.

decisions alone, there are potentially 864 configurations capable of achieving high bmep.

By varying an engine's design within the scope of Figure 8.1, it should be possible to vary the characteristics over a range, which at one extreme emphasises low operating cost, and at the other extreme, low first cost. This is shown by the following diagram. It is also likely that "low" first cost and "low" operating cost are only attainable at the expense of each other.



The needs of diesel engine users vary considerably. The most important attribute of an engine for a mains failure set has been shown to be low first cost. Rail traction and on-highway truck applications require a greater balance of these costs and would probably appear at the right hand side of the diagram above. Military engines are not subject to the same commercial forces ; the primary requirement is for high power from an engine of low box volume and low weight. Perhaps the only factor of universal importance is reliability.

Considering the flexibility of the modern diesel engine, and the varying needs of the diesel engine market, is it sound business practice to compromise an engine design to suit all markets ? The alternative is to produce perhaps three or four different specifications. Few manufacturers are prepared to do this (apart from superficial applications components) because of the proliferation of parts, less efficient production, increased stock and control problems. The result is that the engine is a basic power unit that fails to discriminate between the fundamental needs of the different user groups.

If the high output concept is to become a successful innovation, the product would have to be more specialised, aimed at identified sectors of the market. Compromising the design to appeal to everyone would mean that it offered significant advantage to no-one.

8.4. Conclusions

1. It is feasible to build a high output variant of the Dorman "LE" engine, to operate at fixed speed up to a bmep of 21 bar. This has been demonstrated, using readily available, inexpensive components, in short-run tests under controlled conditions in the development department of the sponsor.

2. The high output concept is unlikely to penetrate all segments of the high speed diesel engine market. To achieve this, the new product would have to offer clear advantages over current products, and the only areas where such an advantage is feasible are : specific cost (£/kW), specific weight and box volume. The vast majority of high-speed engines are purchased for their low

operating costs (£/kW) and high reliability. Since a high output engine, working on this design principle, would almost certainly have higher operating costs than its normally rated equivalent, penetration will be limited to those markets that rate first cost, size or weight above operating costs. Three markets have been shown to be worthy of further consideration :

a) Electrical power generation

b) Fighting vehicles

c) Marine propulsion.

3. The thermal efficiency of the research engine at a bmep of 21 bar was less than that of the current "LE" production engine at its full load of 13.7 bar, both at 1500 rev/min. If allowance is made for the addition of a cooling fan to the research engine, the brake thermal efficiency would be about 36 per cent, which compares with 38 per cent for the production engine. Although this order of deterioration would be expected, since the compression ratio was reduced, the difference between the two engines would probably be less if the high output engine were as highly developed.

4. The use of a wide valve overlap (127^{o}CA) successfully accomplished a high degree of through-flow and, probably, clearance volume scavenging. The apparent volumetric efficiency of the engine was between 98 and 102 per cent at full load. This compares with only 82 to 85 per cent with the standard 22.5^{o}CA overlap cam shaft. The valve recesses in the piston crown necessary for wide overlap are thought to have caused a marked increase in exhaust smoke.

5. Component temperatures do not rise in direct proportion to bmep.

Piston and valve temperature measurements were comfortably within accepted limits. The only notable "hot spot" was in the cylinder head flamedeck in the area between the exhaust valve seat and the injector boss. This was thought to be related to the detail design of the cylinder head, rather than intrinsic to high bmep operation. Although the absolute temperature levels were generally acceptable, the thermal gradients in the piston crown and near the exhaust valve seat could cause distress if the engine were operated over a longer period, or subjected to load cycles. Undercrown oil jet cooling was successfully applied to this engine. From point temperature measurements, it is estimated that the heat entering the piston centre was transferred to the oil, thus lowering the temperature throughout. The oil flow was equivalent to 12.8 l/kWh (2.1 galls/bhph) at a velocity of 1.17 x maximum piston velocity.

6. The "LE" engine, with compression ratio in the range 10.7 to 11.9, is extremely difficult to start without additional "aids". At best, the engine must be cranked for long periods ($>$ 3 minutes) which tends to create large pockets of liquid fuel from the cylinder into the exhaust ducting. Thus, if a start is achieved, a cloud of thick, grey smoke is issued for up to 20 seconds. Of the aids used for routine starting during the experimental programme, controlled ethyl ether fumigation was the most effective - giving a clean start and acceleration up to 1500 rev/min in 3 to 10 seconds. Block heating was a useful secondary aid, but at these low compression ratios, could not induce clean, rapid starting unless set at 45 to 55^{o}C.

7. The high manifold pressures (2.5 to 3.4 bar) are the cause of poor valve stem lubrication. This appears to be serious enough to lead to premature failure. The valves are not made to rotate, and this is generally unhelpful. Inlet and exhaust manifold sealing was a minor problem also caused by the high manifold pressures. This could be eliminated by minor detail design changes.

8. Computer modelling techniques can make a real contribution to the design and development of high-speed diesel engines. Considering that this exercise is the first by the company to use predictive models, the correlation between experiment and prediction is encouraging.
   The use of the "filling and emptying" model undoubtedly reduced the research engine's development phase, thus conserving valuable resources. The specification resulting from the design study achieved the target output with only the change of turbocharger compressor specification.
   These programs are not "too sophisticated" for the daily design problems of an engine manufacturer, and do not require advanced experimental facilities for program development and verification.

CHAPTER 9

Proposals and Recommendations for Further Work

9.1. Decision Options

The objective of this project was to increase the company's knowledge of the high output concept, as expressed in the brief, to assist management decision-making. The decision options that arise from this research are :

1. Terminate all interest in high output technology.
2. Continue the research in some form to further increase the data base, or
3. Commence product development as a precursor to field trials and eventual market launch.

The most obvious conclusion is that the state of knowledge is inadequate for option 3 to be instituted. To be in a position to develop the final product, further market, financial and technical data would have to be obtained. For example, a reasonably tight product outline specification and a sound estimate of the sales volume/cost/profit relationship.

At this stage, the only realistic decision is between project termination and continued research. Ending the study would not necessarily be the "easy" or "safe" option, since this could deny the chance of early involvement if a latent need for this type of product later became apparent. Missing such opportunities may be considered to "cost" the company the lost revenue ; this is known as opportunity cost.

The second option is to find out more by continuing the research in some form, to allow the decision to be made in the future. The decision to continue should be carefully considered against the background of the R & D programme as a whole. A project that is wrongly selected for research effort deprives other projects of valuable resources. The area that requires greater definition, to allow rational decision-making is the market, and the demand that such a product may stimulate.

9.2. Proposals

The main proposal is that the project should be continued, with the proviso that the allocation of resources can be justified against competing alternatives. Although the market appears to be limited to only a small fraction of the total high-speed diesel market, this could nevertheless represent a significant opportunity to a small/medium sized company such as Dorman. The company's resources are well-suited to the concept, having existing engines that could be extended to high output operation and the necessary tooling and manufacturing skills.

Immediate follow-up should be in the marketing area, to establish the potential of those segments highlighted in Chapter 7. This should provide the level of intelligence needed to make the decision whether or not to proceed into product development, or terminate the project.

If the project is not rejected after closer study of the market, then performance targets (first cost, operating cost, durability, etc.) could be established to form the basis of the product development programme.

Further involvement with high output technology would have the secondary benefit of improving market information and adding to the company's technical experience. This could offer "spin-off" for the current product range.

A further proposal, not directly related to high output technology, is that the company should actively follow-up this introduction to computer-based engine design methods. Such techniques are not an academic curiosity, but a valuable, cost-effective contribution to engine design.

9.3. Recommendations for Further Work

The preceding section dealt, in broad terms, with the proposals for the future of the project. This section discusses some of the areas that may be important if the project proceeds.

The most important areas for further work are :

a) Durability

b) Transient response and

c) Starting.

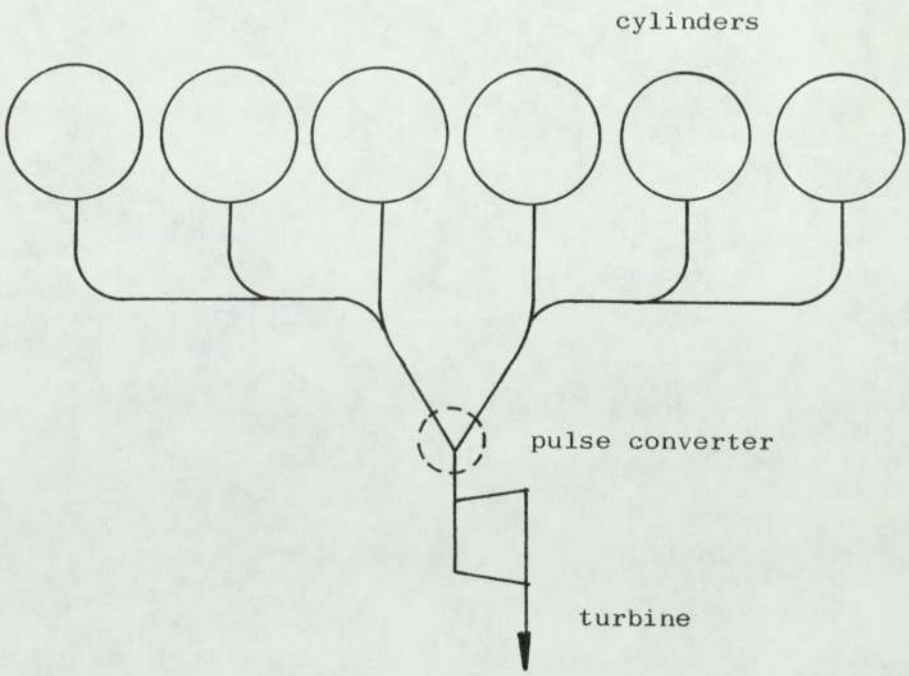
It may be necessary to quantify (a) and (b) before a detailed evaluation of market potential can be made. The first could be achieved by controlled endurance testing, putting the engine through a realistic cycle of operation. Performance could be monitored continuously and components checked for wear, thermal cracking, scuffing, etc., at regular intervals.

Transient response could be assessed using an electrical dynamometer, or generator, capable of applying sudden increases in load. The transient response could probably be improved by attention to details such as manifold shape and volume, and turbocharger rotor inertia, but unless assisted, such highly turbocharged engines would ultimately be limited to bmep steps of 8 to 12 bar. Therefore, further work should evaluate turbocharger assist devices such as combustors, oil-driven pelton wheels, high pressure air reservoirs and air jets (82)(109)(110).

Future development of the larger Dorman engines (127 to 159mm bore) may be limited or even handicapped by the use of the deep bowl swirl combustion system. This effectively prevents the use of wide valve overlap, encourages heat loss and thermal load, and the restriction necessary to induce swirl incurs pumping work and loss of volumetric efficiency. The alternative, the quiescent system, is difficult to apply to this size of engine. However, a relatively inexpensive experimental programme could quickly establish whether it was feasible. The chances of making the system work increase with cylinder size. It is possible that only the larger Dorman engines, which are restricted to speeds below 1500 rev/min could successfully adopt it. However, the benefits are not inconsiderable, especially when high output is sought.

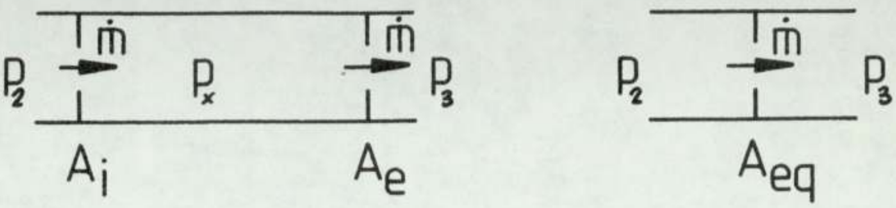
The use of a four-valve cylinder head arrangement on high output variants is worthy of full consideration. This would probably improve gas exchange and volumetric efficiency and reduce exhaust valve thermal load, since they would be smaller in diameter. This would allow the fuel injector to be located in the centre of the cylinder, which is compatible with the adoption of a shallow bowl, quiescent combustion system.

Under high bmep operation, the exhaust pulses issuing from the cylinders become too large for efficient utilisation by the turbine. This leads to a deterioration of the engine performance. A pulse converter is a carefully designed junction that gradually restricts the exhaust flow, smoothing the pressure fluctuation. A suggested arrangement for the "LE" engine is given below.



Zinner's Model for the Determination of Scavenge Flow

For the analysis of scavenging, the valve overlap period need only be considered. The first stage is to replace the inlet and exhaust valves by an equivalent area.



Simplifying assumptions:

1. The fluid is incompressible.
2. The kinetic energy generated in the first restriction is completely lost.
3. The coefficients of flow restriction are equal.

For continuity:

$$A_i\sqrt{(p_2 - p_x)} = A_e\sqrt{(p_x - p_3)} = A_{eq}\sqrt{(p_2 - p_3)} = C$$

$$\therefore \quad p_2 - p_x = \frac{C^2}{A_i^2} \qquad p_x - p_3 = \frac{C^2}{A_e^2} \qquad p_2 - p_3 = \frac{C^2}{A_{eq}^2}$$

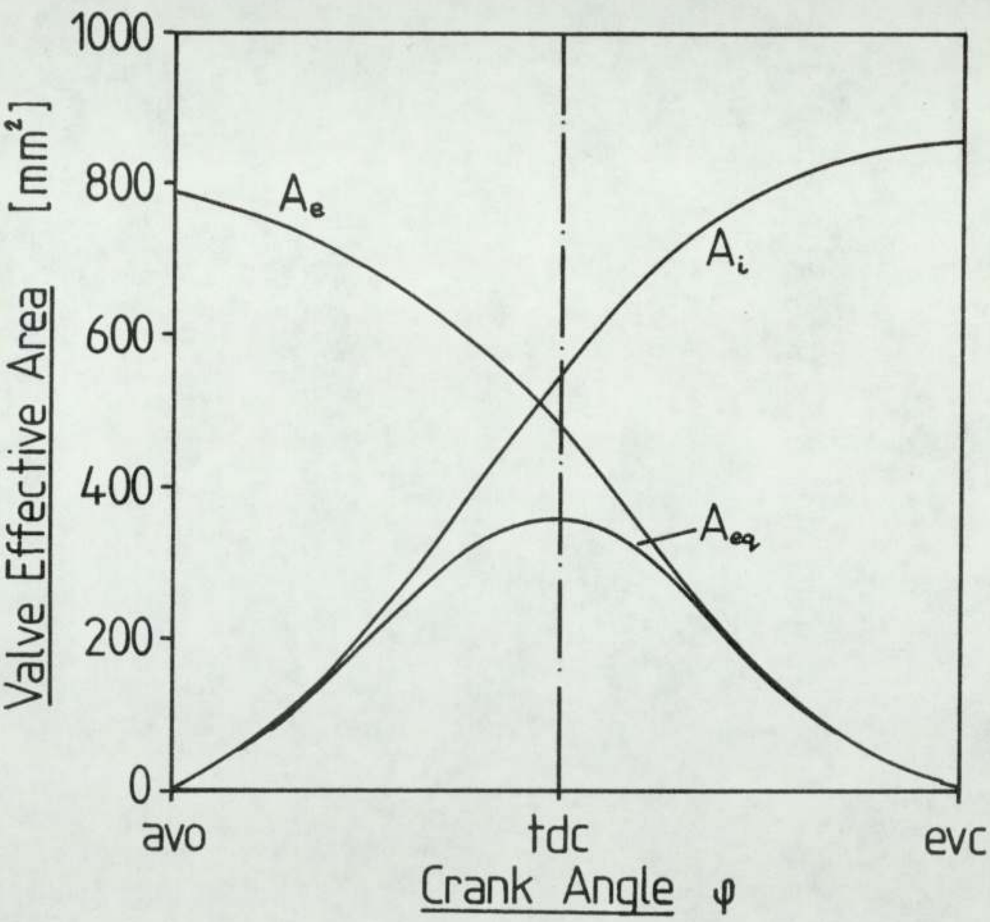
from which:

$$A_{eq} = \sqrt{\frac{A_i^2 A_e^2}{A_i^2 + A_e^2}}$$

As A_i and A_e are functions of crank **angle**, A_{eq} is an instantaneous value. The mean equivalent area, open throughout the cycle, is $\bar{A}_{eq}$, given by:

$$\bar{A}_{eq} = \int \frac{A_{eq}\,d\varphi}{720\,\text{deg}}$$

The following example is based on the 'LE' valve effective areas and gives the mean effective area used in section 6.3.

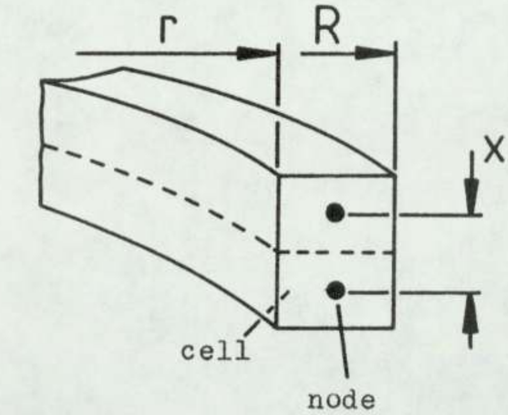


$$\int A_{eq}\,d\varphi = 23\ 000\ \text{mm}^2\text{deg}$$

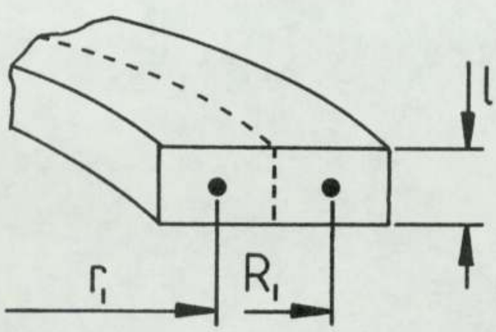
$$\bar{A}_{eq} = 31.8\ \text{mm}^2/\text{cyl} \equiv 191\ \text{mm}^2/\text{engine}$$

Detail of the Piston and Liner Thermal Analogue Models

The analogue network model represents the component to be studied as a series of interconnected cells, or 'lumped' masses. The network is chosen so that cell mass and conductivity are readily calculable. These are usually square, rectangular or triangular. Both the piston and the liner were assumed to be axisymmetrical, to simplify the problem. The inter-cell conductances for a rectangular axisymmetric network are given below.



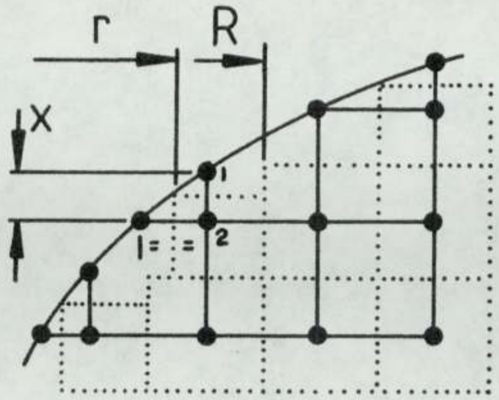
$$G = \pi (R^2 - r^2) \frac{k}{x}$$



$$G = \frac{2\pi k l}{\log_e (R_1 / r_1)}$$

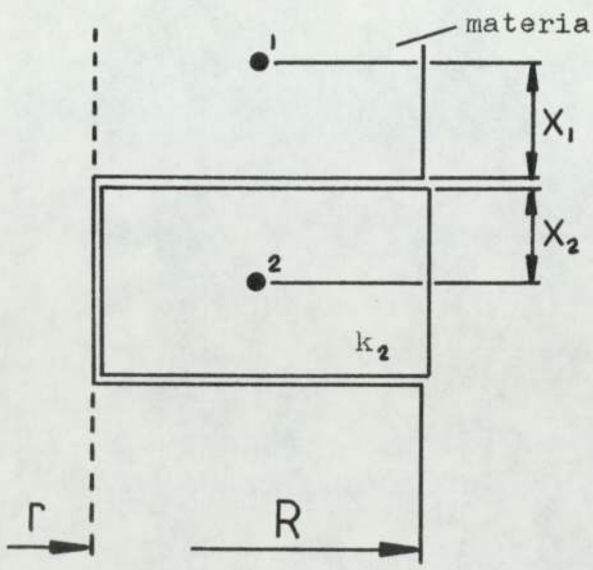
where k is the material thermal conductivity

At curved or otherwise irregular boundaries conductances are not so readily calculated. The following method was suggested by Chapman (104).



$$G_{12} = \frac{k\pi(R^2 - r^2)}{X}$$

For transient problems it is necessary to calculate the cell thermal capacity as well as the inter-cell conductance. Since this is not required for steady-state solutions, it will not be discussed further. In areas where there is a change of material or an interface, between nodes the conductance is calculated as follows.

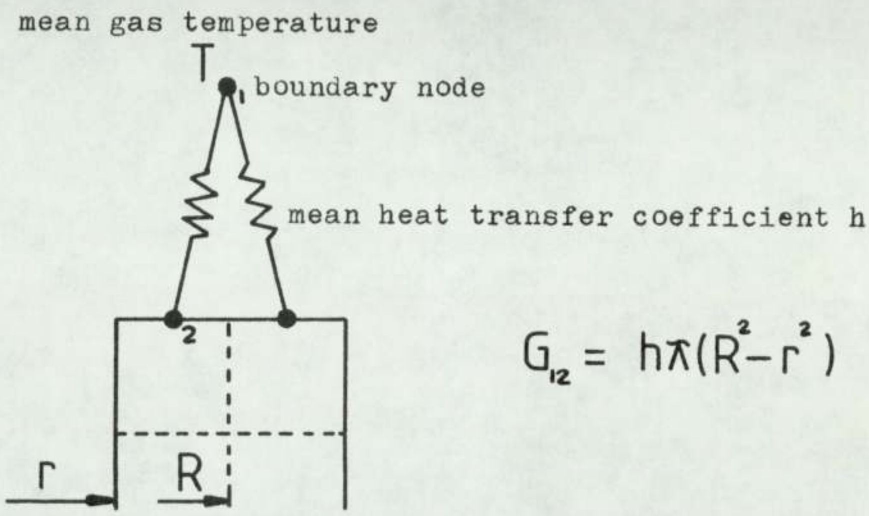


film coefficient h

$$\frac{1}{G_{12}} = \frac{1}{G_1} + \frac{1}{G_f} + \frac{1}{G_2}$$

$$G_{12} = \frac{\pi(R^2 - r^2)}{\frac{X_1}{k_1} + \frac{1}{h} + \frac{X_2}{k_2}}$$

At the boundary the heat transfer is represented by a conductance, calculated in the normal way, and the temperature of the surrounding medium. In combustion engine work it is usual to consider the cylinder gas temperature and heat transfer coefficient as being constant at their mean values. The cycle frequency and component thermal inertia mean that only the first 3 to 5 mm below the surface has transient heat flow, during the cycle.



$$G_{12} = h\pi(R^2 - r^2)$$

Two piston networks were constructed during the project. The first was used to make the prediction of piston heat flow and temperature reported in Chapter 4. Later, this was found to be unnecessarily approximate around the irregular toroidal bowl edge. Model 2 was constructed to eliminate this error, but was not used due to time constraint. The second model is operational and stored on tape with the Dorman Data Processing Dept. Figures A2.I,A2.2 and A2.3 show the networks of the piston and the liner.

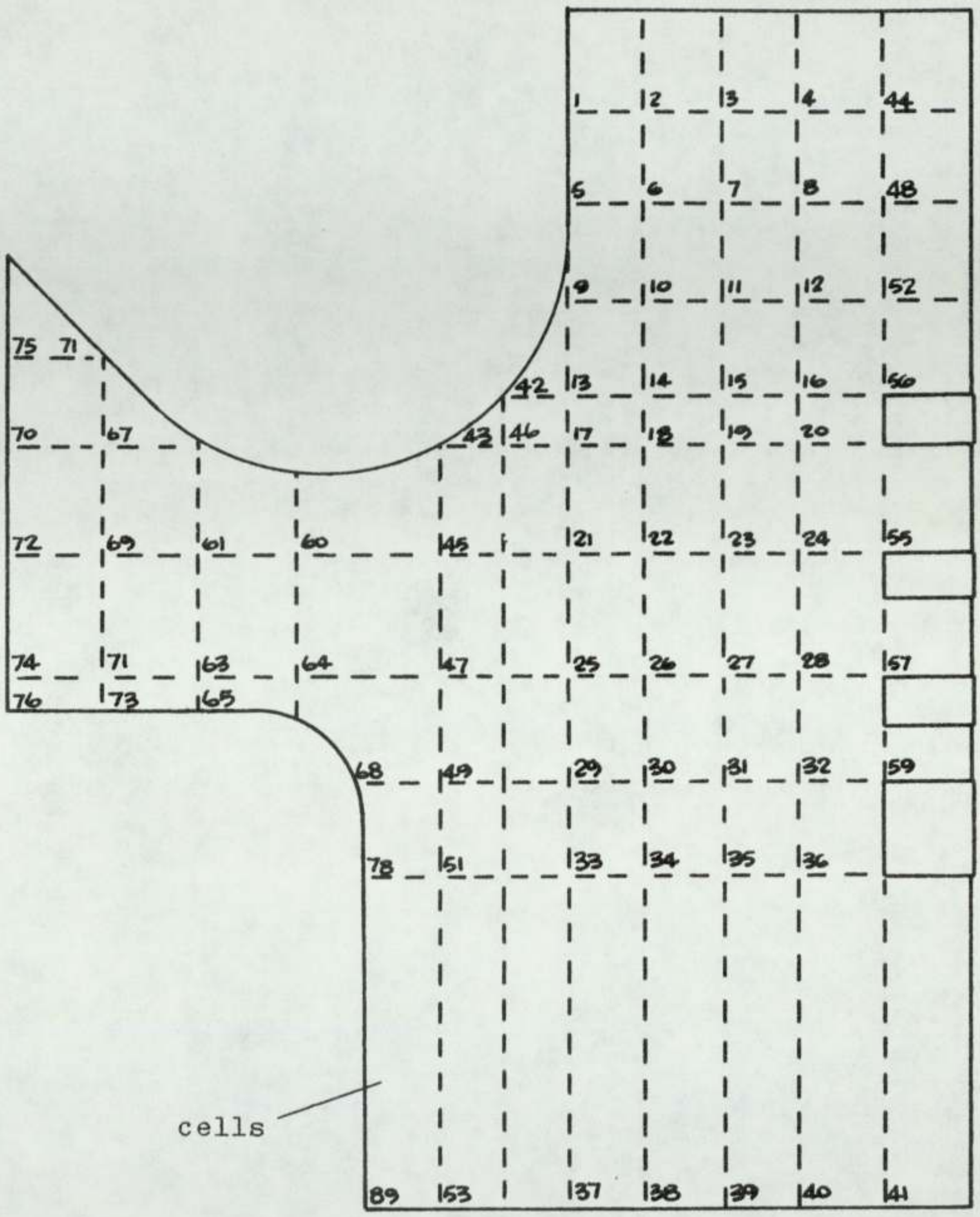


Fig.A2.1, First piston thermal analogue network. This was used to make the heat flow and temperature prediction presented in chapter 4.

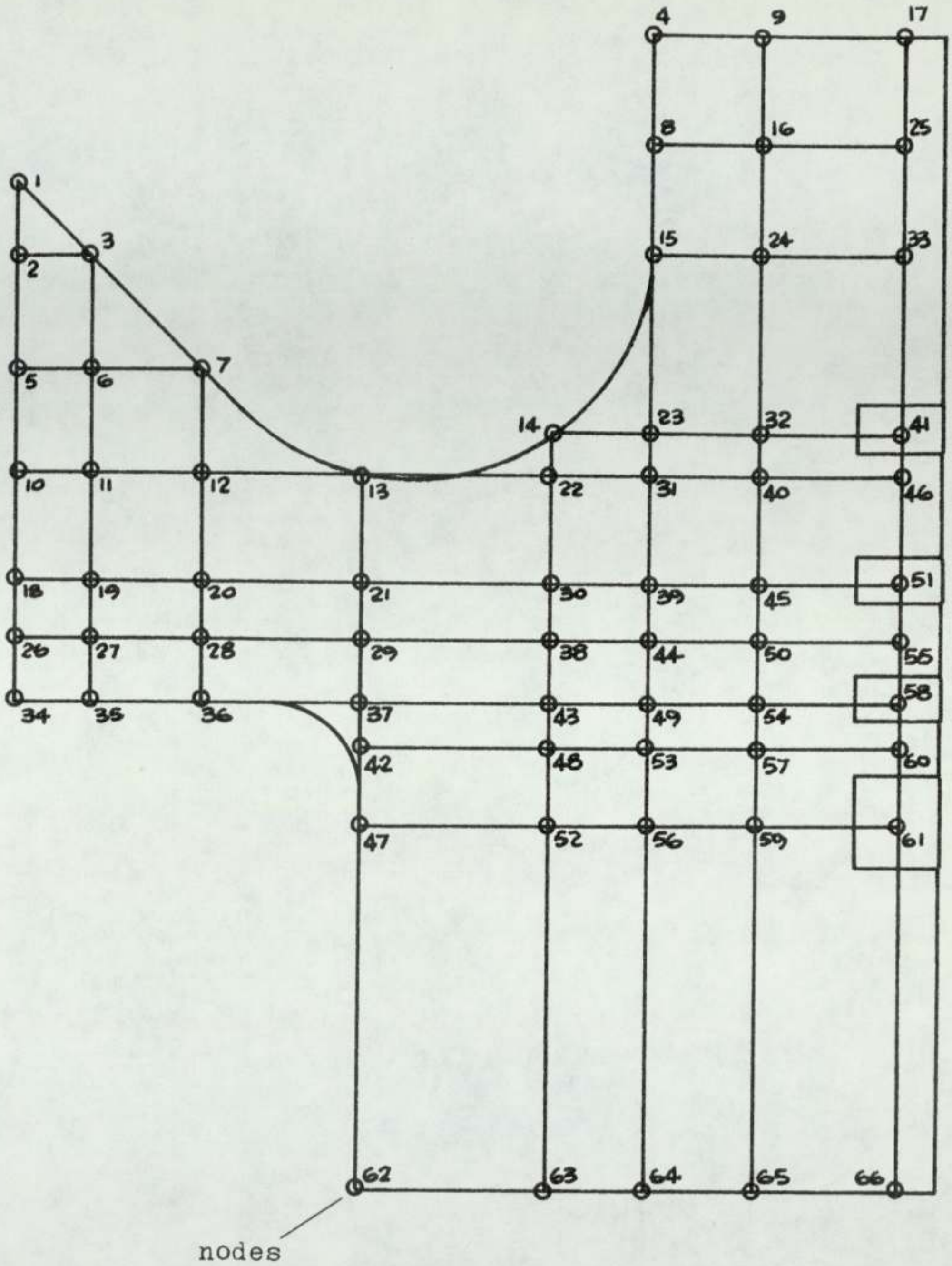


Fig.A2.2, Second piston thermal analogue network.

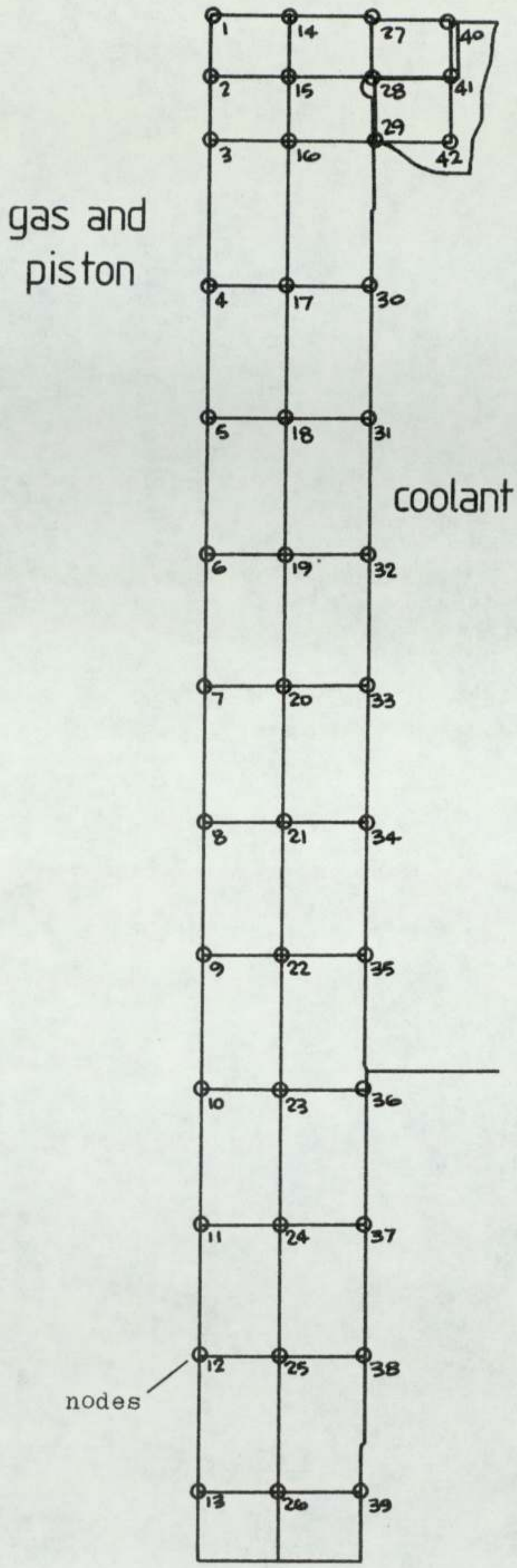


Fig.A2.3, Thermal analogue network of the 'LE' cylinder liner, used to make the temperature prediction.

APPENDIX 3

Calculation of C_p and γ for Exhaust Gas Mixtures

For the calculation of apparent exhaust energy and the heat rejected in the exhaust gases, it is necessary to derive C_p and γ. C_p is the specific heat of the mixture at constant pressure. γ is the ratio of specific heats:

$$\gamma = \frac{C_p}{C_v}$$

$$R = C_p - C_v$$

Where C_v is the specific heat at constant volume.

R is the specific gas constant.

The factors that have an important influence over the gas 'constants' are the mixture composition and the temperature. Mixture composition is largely determined by the air-fuel ratio, if dissociation is neglected:
Where x is the air-fuel ratio.

$$\text{exhaust mass} = x + 1$$
$$\text{charge air mass} = x$$
$$\text{fuel mass} = 1$$

assume composition of air, on mass basis is:

$N_2 = 76.7\ \%$ $\qquad$ $O_2 = 23.3\ \%$

assume composition of light diesel fuel is:

$$C = 87\ \% \qquad H_2 = 13\ \%$$

Then:

$$\frac{76 \cdot 7}{100} x N_2 + \frac{23 \cdot 3}{100} x O_2 + \frac{87}{100} C + \frac{13}{100} H_2 \longrightarrow \frac{23 \cdot 3 x}{100} O_2$$

$$- \frac{13.8}{100} O_2 + \frac{76 \cdot 7}{100} x N_2 + \frac{87.44}{100.12} CO_2 + \frac{13.18}{100.2} H_2O - \frac{87.8}{100.3} O_2$$

Relative proportions in the exhaust:

$$O_2 \sim \frac{0 \cdot 233 x - 3 \cdot 36}{x + 1} \qquad N_2 \sim \frac{0 \cdot 767 x}{x + 1}$$

$$CO_2 \sim \frac{3 \cdot 19}{x + 1} \qquad H_2O \sim \frac{1 \cdot 17}{x + 1}$$

From Rogers and Mayhew, ref(4) page 272:

$$C_{p,mix} = \sum \frac{m_i}{m} C_{p_i}$$

$$R_{mix} = \sum \frac{m_i}{m} R = \sum \frac{m_i}{m} \frac{R_o}{M_i}$$

Thus, the gas constants C_p and R may be derived from the constants of the constituents, weighted by their mass proportion. The ratio of specific heats γ may be derived as follows:

$$\gamma_{mix} = \frac{C_{p,mix}}{C_{p,mix} - R_{mix}}$$

The values of C_p for the exhaust constituents considered, assuming H_2O in the vapour phase, are given in ref(I06). R_o is the universal gas constant (8.3I43 kJ/kmolK) and M is the relative molecular mass. A computer program was written to perform this calculation over a range of temperature. This later became a sub-routine of the data reduction program, appendix 4. The program was written in a form of the 'BASIC' language, for the company's Hewlett Packard HP9825A micro computer. The program is listed in figure A3.I and examples of the calculated results are given in figure A3.2.

```
0: "FILE...23  TRACK 0":
1: "GAS CONSTANTS FOR EXHAUST MIXTURES → ASSUMING NO DISSOCIATION":
2: dim N[8],A$[5];1→A
3: "START":
4: wrt 6," "
5: wrt 6," "
6: wrt 6," "
7: ent "AIR-FUEL RATIO?",U
8: wrt 6,"AIR/FUEL RATIO =',U
9: wrt 6,"=============="
10: wrt 6," "
11: wrt 6," "
12: wrt 6,"            TEMP [degC]            Cp [kJ/kgK]            GAMMA"
13: for T=573 to 1023 by 25
14: "Cp":
15:
16: 1.0724-272.4tn↑(-6)T+607.1tn↑(-9)T↑2-240tn↑(-12)T↑3→N[1]
17: .81592+337.3tn↑(-6)T-11.01tn↑(-9)T↑2-50tn↑(-12)T↑3→N[2]
18: 1.8111-55.18tn↑(-6)T+831.6tn↑(-9)T↑2-300tn↑(-12)T↑3→N[3]
19: .474+541.5tn↑(-6)T-68.64tn↑(-9)T↑2+290tn↑(-12)T↑3→N[4]
20:
21:
22:
23:
24: N[1]/(N[1]-.297)→N[5]
25: N[2]/(N[2]-.26)→N[6]
26: N[3]/(N[3]-.231)→N[7]
27: N[4]/(N[4]-.189)→N[8]
28: .767N[5]U/(U+1)+(.233U-3.36)N[6]/(U+1)+1.17N[7]/(U+1)+3.19N[8]/(U+1)→G
29: .767N[1]U/(U+1)+(.233U-3.36)N[2]/(U+1)+1.17N[3]/(U+1)+3.19N[4]/(U+1)→N[2]
30: fxd 4
31: wrt 6,T-273,N[2],G
32: next T
33: ent "ANOTHER RUN? [Y/N]",A$;if cap(A$[1,1])="Y";gto "START"
34: if cap(A$[1,1])#"N";jmp -1
35: end
```

Fig. A3.1, Listing of gas constants program.

TEMP [degC]	Cp [kJ/kgK]	GAMMA
300.0000	1.0788	1.3554
325.0000	1.0865	1.3521
350.0000	1.0944	1.3487
375.0000	1.1026	1.3454
400.0000	1.1110	1.3420
425.0000	1.1196	1.3387
450.0000	1.1284	1.3354
475.0000	1.1374	1.3321
500.0000	1.1466	1.3289
525.0000	1.1559	1.3257
550.0000	1.1654	1.3226
575.0000	1.1750	1.3195
600.0000	1.1847	1.3165
625.0000	1.1946	1.3135
650.0000	1.2045	1.3107
675.0000	1.2145	1.3079
700.0000	1.2246	1.3052
725.0000	1.2347	1.3025
750.0000	1.2448	1.3000

A/F = 20

TEMP [degC]	Cp [kJ/kgK]	GAMMA
300.0000	1.0739	1.3595
325.0000	1.0811	1.3563
350.0000	1.0885	1.3531
375.0000	1.0962	1.3499
400.0000	1.1041	1.3467
425.0000	1.1121	1.3435
450.0000	1.1203	1.3404
475.0000	1.1287	1.3372
500.0000	1.1373	1.3341
525.0000	1.1459	1.3310
550.0000	1.1547	1.3280
575.0000	1.1636	1.3251
600.0000	1.1726	1.3222
625.0000	1.1816	1.3193
650.0000	1.1907	1.3166
675.0000	1.1999	1.3139
700.0000	1.2091	1.3113
725.0000	1.2182	1.3087
750.0000	1.2274	1.3063

A/F = 25

Fig. A3.2, Calculated results with air-fuel ratios of 20 and 25.

APPENDIX 4

Experimental Data Reduction Program

This program converts the raw experimental data into a set of performance indicators. Generally these are fundamental quantities, such as brake thermal efficiency, volumetric efficiency, compressor and apparent turbine efficiencies, explosion and air-fuel ratios. These are calculated very rapidly and provide a valuable 'overview' of engine performance. This gives greater insight and better decision-making during test-bed development, when time is at a premium.

The program was written to meet the needs of the high output research engine and it's instrumentation. The program is not highly polished, and is presented as used during the project. If the program were generalised to cope with a range of engines, instrumentation sets and units of measurement, it would prove an invaluable development aid.

Figure A4.I is the program flow diagram. The gas constants were calculated using the method given in appendix 3, which takes account of the air-fuel ratio and gas temperature. Figure A4.2(a) to (d) give the program listing and figure A4.3 is an example of the results print-out. The program was written in a form of the 'BASIC' language, to be compatible with the company's Hewlett-Packard 9825A micro computer.

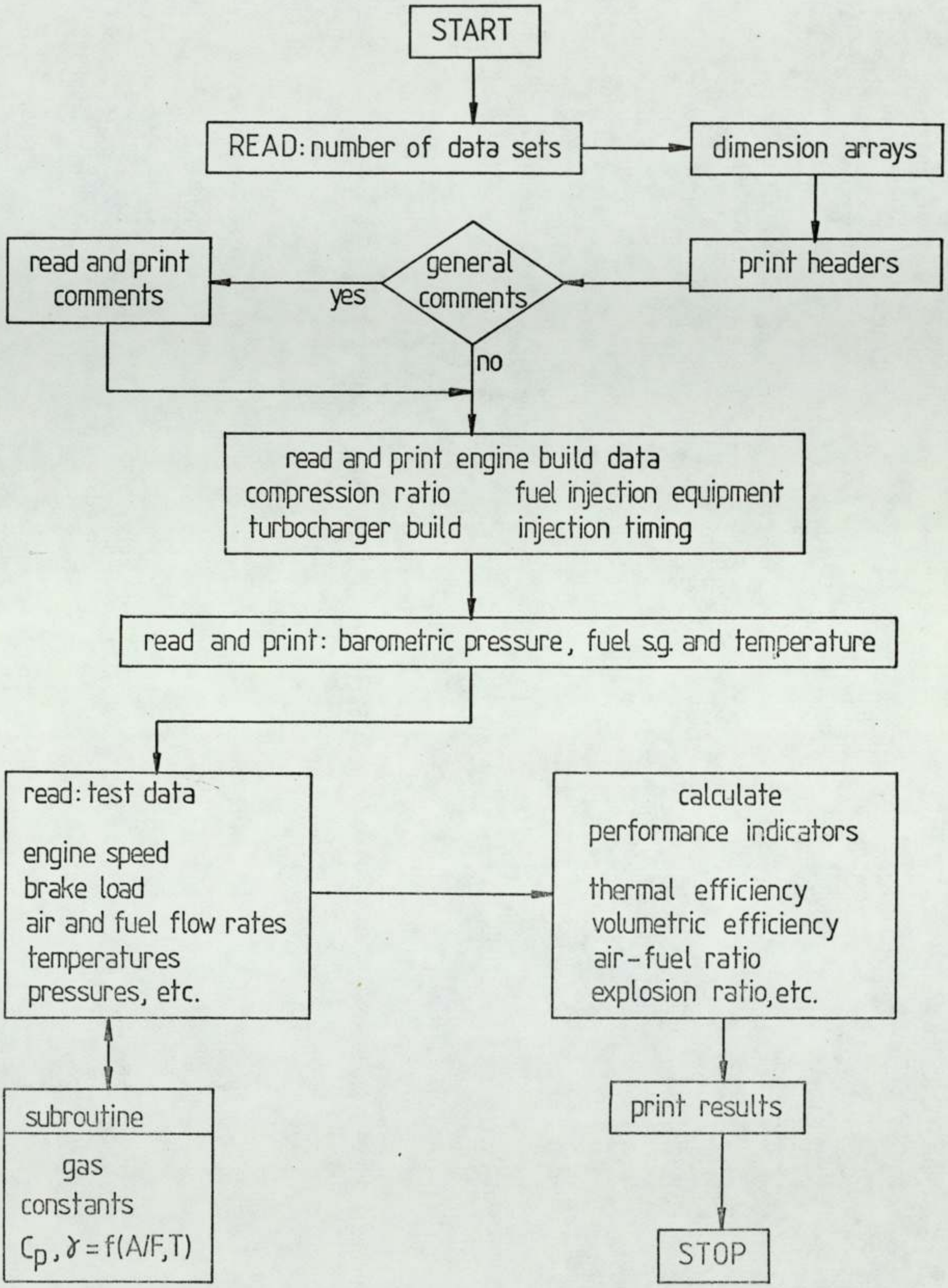


Fig. A4.1, Flow diagram of experimental data reduction program.

```
0: "FILE...2":
1: ent "NUMBER OF DATA SETS?",B
2: wrt 6," "
3: wrt 6," "
4: dim A[20,B],B[30,B]
5: dim A$[85],N[8],B$[20]
6: fxd 0;ent "LOG SHEET NUMBER",F
7: ent "     DATE       ",B$
8: "6LE HIGH OUTPUT EXPERIMENTAL ENGINE        LOG SHEET No."→A$
9: str(F)→A$[len(A$)+1];"      "→A$[len(A$)+1];B$→A$[len(A$)+1]
10: wrt 6,A$
11: "================================"→A$;wrt 6,A$
12: fxd 3
13: wrt 6," "
14: ent "No. OF LINES OF REMARKS?",F
15: for A=1 to F
16: ent "REMARKS",A$;wrt 6,A$
17: next A
18: wrt 6," "
19: fxd 1;ent "COMPRESSION RATIO",Y;wrt 6,"COMPRESSSION RATIO",Y
20: fxd 3;ent "TURBOCHARGER BUILD",A$;wrt 6,"TURBOCHARGER BUILD          ",A$
21: wrt 6,"FUEL PUMP   MAXIMEC,12mm Lift,13mm Dia Plungers,4mm inlet stroke."
22: fxd 3;ent "INJECTOR NOZZLES",A$;wrt 6,"INJECTOR NOZZLES                ",A$
23: fxd 0;ent "SPILL TIMING btdc",F;wrt 6,"SPILL TIMING btdc",F
24: wrt 6," "
25: fxd 1
26: ent "BAROMETER (mm.Hg)",C;wrt 6,"BAROMETER              ",C,"(mm)";C/25.4→C
27: fxd 3;ent "FUEL SPECIFIC GRAVITY",D;wrt 6,"FUEL S.G.                 ",D
28: fxd 1
29: ent "FUEL TEMPERATURE (degC)",E;wrt 6,"FUEL TEMP.             ",E,"(degC)"
30: wrt 6," "
31:
32:
33:
34:
35:
36: for A=1 to B
37: ent "ENGINE SPEED (RPM)",A[1,A]
38: next A
39: for A=1 to B
40: ent "BRAKE LOAD (lbf)",A[2,A];if A[2,A]=0;.1→A[2,A]
41: next A
42: for A=1 to B
43: ent "FUEL (SEC/PINT/2)",A[3,A]
44: next A
45: for A=1 to B
46: ent "SMOKE LEVEL (bosch units)?",A[4,A]
47: next A
48: for A=1 to B
49: ent "MAXIMUM PRESSURE (lbf/in2)",A[5,A]
50: next A
51: for A=1 to B
52: ent "AIR CONSUMPTION (cfm)",A[6,A]
53: next A
54: for A=1 to B
55: ent "AIR CLEANER DEPRESSION (cm.H2O)",A[7,A];A[7,A]/2.54→A[7,A]
56: if A[7,A]>20;dsp "EXCESSIVE AIR CLEANER DEPRESSION";beep;wait 1500
57: C-A[7,A]/13.6→A[7,A]
58: next A
59: for A=1 to B
60: ent "SILENCER BACK PRESSURE (cm.H2O)",A[8,A];A[8,A]/2.54→A[8,A]
61: if A[8,A]>20;dsp "EXCESSIVE BACK PRESSURE";beep;wait 1500
62: C+A[8,A]/13.6→A[8,A]
63: next A
64: for A=1 to B
65: "MANOMETER CORR.FACTOR=1.0139":
66: ent "BOOST INTO COOLER (cm.Hg)",A[9,A];1.0139*A[9,A]/2.54→A[9,A]
67: A[9,A]+C→A[9,A]
68: next A
69: for A=1 to B
70: ent "BOOST INTO ENGINE (cm.Hg)",A[10,A];1.0139*A[10,A]/2.54→A[10,A]
71: A[10,A]+C→A[10,A]
72: next A
73: for A=1 to B
74: ent "STATIC P AT TURBINE (cm.Hg)",A[20,A];1.0139*A[20,A]/2.54→A[20,A]
75: A[20,A]+C→A[20,A]
76: next A
77: for A=1 to B
78: ent "TOTAL P AT TURBINE (cm.Hg)",A[11,A];1.0139*A[11,A]/2.54→A[11,A]
79: A[11,A]+C→A[11,A]
80: next A
```

Fig.A4.2(a) Part of data reduction program.

```
80: next A
81: for A=1 to B
82: ent "TEMPERATURE INTO COMPRESSOR (degC)",A[12,A]
83: next A
84: for A=1 to B
85: ent "TEMPERATURE INTO COOLER (degC)",A[13,A]
86: next A.
87: for A=1 to B
88: ent "TEMPERATURE INTO ENGINE (degC)",A[14,A]
89: next A
90: for A=1 to B
91: ent "RAW WATER TEMPERATURE",A[16,A]
92: next A
93: for A=1 to B
94: ent "TEMPERATURE INTO TURBINE (degC)",A[15,A]
95: next A
96: for A=1 to B
97: ent "TEMPERATURE AFTER TURBINE (degC)",A[17,A]
98: A[14,A]+273→A[14,A];A[12,A]+273→A[12,A];A[13,A]+273→A[13,A]
99: A[15,A]+273→A[15,A];A[16,A]+273→A[16,A];A[17,A]+273→A[17,A]
100: next A
101: for A=1 to B
102: ent "COOLANT FLOW (GPM)                    ",A[18,A]
103: next A
104: for A=1 to B
105: ent "TEMP.RISE ACROSS ENG. (degC)",A[19,A]
106: next A
107:
108:
109:
110:
111:
112: for A=1 to 25
113: for T=1 to B
114: 0→B[A,T]
115: next T
116: next A
117:
118:
119:
120:
121:
122:
123: for A=1 to B
124: "B1=BMEP":1144.29A[2,A]/(400*14.5)→B[1,A]
125: "B5=POWER":A[2,A]A[1,A].7457/400→B[5,A]
126: "B23=BSFC":(D-(E-15).00038)2255.05*1000/2.205B[5,A]A[3,A]→B[23,A]
127: "B2=Pmax":A[5,A]/14.5→B[2,A]
128: "B6=bsac":20052A[6,A]C/A[12,A]B[5,A]→B[6,A]
129: "B3=AIR/FUEL":B[6,A]/B[23,A]→B[3,A]
130: "B4=EXP.RATIO":A[11,A]/A[8,A]→B[4,A]
131: A[15,B]→T
132: gsb 203
133: "B7=CORR.TURB.FLOW":
134: (B[23,A]+B[6,A])B[5,A]r(A[15,A]/288.3)29.92/A[11,A]60000→B[7,A]
135: 2.205*B[7,A]→B[7,A]
136: "B8=TURB.POWER":
137: (B[23,A]+B[6,A])B[5,A]N[2]A[15,A](1-B[4,A]↑((1-G)/G))→B[8,A]
138: B[8,A]/3600000→B[8,A]
139: A[9,A]/A[7,A]→T
140: A[12,A](T↑.2846-1)/(A[13,A]-A[12,A])→R
141: A[6,A]22.047*1.008(T↑.2846-1)/60R2.205→Q
142: "B9=TURB.EFF.":Q/B[8,A]→B[9,A]
143: "B10=VOL.EFF.":A[6,A]A[14,A]C4*2*12↑3→B[10,A]
144: B[10,A]/(A[10,A]*25*π*5.875*6*A[1,A]A[12,A])→B[10,A]
145: "B11=B.TH.EFF.":3600*1000/(42800*B[23,A])→B[11,A]
146: "B12=BOOST RATIO":A[9,A]/A[7,A]→B[12,A]
147: "B13=COMP.EFF.":A[12,A](T↑.2846-1)/(A[13,A]-A[12,A])→B[13,A]
148: "B14=A/C EFF.":(A[13,A]-A[14,A])/(A[13,A]-A[16,A])→B[14,A]
149: "B15=COMP.POWER":A[6,A]22.047*1.008(T↑.2846-1)/60R2.205→B[15,A]
150: "B22=COMP.CORR.MASS":
151: A[6,A]626.13r(A[12,A]/303)/A[7,A]A[12,A]→B[22,A]
152: "B16=COMB.RATIO":
153: A[5,A]/14.5/(A[10,A].0254*.136*9.81*Y↑1.35)→B[16,A]
154: "B17=DENS.RATIO COMP.":A[9,A]A[12,A]/A[13,A]C→B[17,A]
155: "B18=DENS.RATIO A/C":A[10,A]A[13,A]/A[9,A]A[14,A]→B[18,A]
156: "B19=MM3/SHOT":B[23,A]B[5,A]5.556/A[1,A]D→B[19,A]
157: "B20=TURB.ISEN.EFF.":
158: (A[15,A]-A[17,A])/A[15,A](1-B[4,A]↑((1-G)/G))→B[20,A]
159: "B21=P.DROP ACROSS CYL":.03388(A[10,A]-A[20,A])→B[21,A]
160: "B24=HEAT TO COOLANT":A[18,A].3077A[19,A]→B[24,A]
```

Fig.A4.2(b) Part of data reduction program.

```
160: "B24=HEAT TO COOLANT":A[18,A].3077A[19,A]→B[24,A]
161: "DENS.H20=969@85C":
162: (A[13,A]+A[14,A])/2→T
163: gsb 274
164: "B25=HEAT TO A'COOLER":B[6,A]B[5,A]N[2](A[13,A]-A[14,A])/3600000→B[25,A]
165: (A[17,A]+A[12,A])/2→T
166: gsb 203
167: "B26=HEAT TO STACK":
168: (B[23,A]+B[6,A])B[5,A]N[2](A[17,A]-A[12,A])/3600000→B[26,A]
169:
170:
171:
172:
173: next A
174:
175:
176:
177: for F=1 to 78;"-"→A$[F,F];next F;wrt 6,A$
178: "|                                             |                   TURBINE     "→A$
179: "              |"→A$[len(A$)+1];wrt 6,A$;" "→A$[1,80]
180: for F=1 to 78;"-"→A$[F,F];next F;wrt 6,A$
181: "SPEED BMEP  POWER  BSFC   Pmax  BSAC    AIR/ c.flow EXP"→A$
182: "  Pt   Tit  EFFtm  Pit"→A$[len(A$)+1]
183: wrt 6,A$
184: "(RPM) (bar) (bkW) (g/kWh) (bar) (g/kWh) FUEL(lb/mn)RATIO (kW) (K)"→A$
185: "          (bar)"→A$[len(A$)+1]
186: wrt 6,A$
187: fmt 1,f4.0,f7.2,f5.0,f8.1,f5.0,f7.0,f8.2,f5.1,f6.2,f6.1,f4.0,f7.3,f6.3
188:
189:
190: for A=1 to B
191: B[7,A]→J;B[8,A]→P;B[9,A]→O
192: A[11,A].03389→Z;B[4,A]→S;B[5,A]→W;B[6,A]→T
193: wrt 6.1,A[1,A],B[1,A],W,B[23,A],B[2,A],T,B[3,A],J,S,P,A[15,A],O,Z
194: next A
195: gto 210
196:
197:
198:
199:
200:
201:
202:
203: "GAMMA":
204: B[3,A]→U
205: 1.0724-272.4tn↑(-6)T+607.1tn↑(-9)T↑2-240tn↑(-12)T↑3→N[1]
206: .81592+337.3tn↑(-6)T-11.01tn↑(-9)T↑2-50tn↑(-12)T↑3→N[2]
207: 1.8111-55.18tn↑(-6)T+831.6tn↑(-9)T↑2-300tn↑(-12)T↑3→N[3]
208: .474+541.5tn↑(-6)T-68.64tn↑(-9)T↑2+290tn↑(-12)T↑3→N[4]
209:
210: N[1]/(N[1]-.297)→N[5]
211: N[2]/(N[2]-.26)→N[6]
212: N[3]/(N[3]-.231)→N[7]
213: N[4]/(N[4]-.189)→N[8]
214: .767N[5]U/(U+1)+(.233U-3.36)N[6]/(U+1)+1.17N[7]/(U+1)+3.19N[8]/(U+1)→G
215: .767N[1]U/(U+1)+(.233U-3.36)N[2]/(U+1)+1.17N[3]/(U+1)+3.19N[4]/(U+1)→N[2]
216: ret
217:
218: wrt 6," "
219: wrt 6," "
220: wrt 6," "
221: for F=1 to 78;"-"→A$[F,F];next F;wrt 6,A$
222: "|                                        |  '                COMPRESSOR"→A$
223: "                  |"→A$[len(A$)+1];wrt 6,A$;" "→A$[1,80]
224: for F=1 to 78;"-"→A$[F,F];next F;wrt 6,A$
225: " VOL.EFF  A/C       Trw   B.Th    COMB  BOOST c.flow  Pc   EFFc   Tic"→A$
226: "     Pic"→A$[len(A$)+1]
227: wrt 6,A$
228: "              EFF.     (K)     EFF.   RATIO RATIO (lb/mn)(kW)            (K)"→A$
229: "    (bar)"→A$[len(A$)+1]
230: wrt 6,A$
231: fmt 2,f7.3,f8.3,f8.1,f7.3,f7.2,f6.2,f7.1,f6.1,f7.3,f6.1,f7.3
*7852
```

Fig.A4·2(c), Part of the program for the reduction of experimental data.

```
240: fmt 2,f7.3,f8.3,f8.1,f7.3,f7.2,f6.2,f7.1,f6.1,f7.3,f6.1,f7.3
241:
242:
243:
244: for A=1 to B
245: B[10,A]→N;B[14,A]→L;B[11,A]→M;B[16,A]→I;B[12,A]→T
246: B[22,A]→K;B[15,A]→Q;B[13,A]→R
247: wrt 6.2,N,L,A[16,A],M,I,T,K,Q,R,A[12,A],A[7,A].03388
248: next A
249: wrt 6," "
250: wrt 6," "
251: wrt 6," "
252: wrt 6," "
253: for F=1 to 78;"-"→A$[F,F];next F;wrt 6,A$
254: wrt 6,"        DENSITY RATIO        FUEL   Teng   dPe  SMOKE  TURB ISEN"
255: wrt 6,"  COMP    A/C    OVERALL mm3/shot (K)  (bar) (bsch)  EFF."
256: for F=1 to 78;"-"→A$[F,F];next F;wrt 6,A$
257: fmt 3,f6.3,f7.3,f9.3,f8.1,f7.1,f6.3,f5.1,f9.3
258:
259:
260:
261: for A=1 to B
262: B[17,A]→H;B[18,A]→V;B[19,A]→X;B[20,A]→T
263: wrt 6.3,H,V,A[10,A]A[12,A]/CA[14,A],X,A[14,A],B[21,A],A[4,A],T
264: next A
265: wrt 6," "
266: wrt 6," "
267: for F=1 to 78;"-"→A$[F,F];next F;wrt 6,A$
268: "|              ENERGY BALANCE (OR LACK OF)              |                "→A$
269: "           |"→A$[len(A$)+1];wrt 6,A$;" "→A$[1,80]
270: for F=1 to 78;"-"→A$[F,F];next F;wrt 6,A$
271: "TOTAL   WORK   STACK   COOLANT  A'COOLER  BALANCE        "→A$
272: wrt 6,A$
273: " kW       %       %        %        %        %           "→A$
274: "                              "→A$[len(A$)+1]
275: wrt 6,A$
276: fmt 1,f4.0,f8.3,f8.3,f9.3,f9.3,f10.3
277: for A=1 to B
278: B[5,A]/B[11,A]→T;1-(B[5,A]+B[24,A]+B[25,A]+B[26,A])/T→J
279: wrt 6.1,T,B[11,A],B[26,A]/T,B[24,A]/T,B[25,A]/T,J
280: next A
281: wrt 6," "
282: jmp 6
283: "CP air":if T>450;1.021→N[2]
284: if T>400;1.017→N[2]
285: if T>350;1.0094→N[2]
286: if T>300;1.0056→N[2]
287: ret
288: end
```

Fig.A4.2(d) Part of data reduction program.

6LE HIGH OUTPUT EXPERIMENTAL ENGINE LOG SHEET No. 80 13.2.80

COMPRESSSION RATIO	11.6
TURBOCHARGER BUILD	Tv61 W-4/1.15G
FUEL PUMP	MAXIMEC, 12mm Lift, 12mm Dia Plungers, 4mm inlet stroke.
INJECTOR NOZZLES	L5386
SPILL TIMING btdc	18
BAROMETER	757.2(mm)
FUEL S.G.	0.830
FUEL TEMP.	27.0(degC)

							TURBINE					
SPEED (RPM)	BMEP (bar)	POWER (bkW)	BSFC (g/kWh)	Pmax (bar)	BSAC (g/kWh)	AIR/ FUEL	c.flow (lb/mn)	EXP RATIO	Pt (kW)	Tit (K)	EFFtm	Pit (bar)
1500	4.04	57	271.7	42	10535	38.77	28.0	1.24	6.9	669	0.660	1.257
1500	4.04	57	268.2	42	10535	39.27	27.9	1.24	6.9	666	0.663	1.257
1500	7.99	113	235.9	65	7792	33.03	36.3	1.49	21.3	766	0.723	1.520
1500	7.99	113	236.6	65	7818	33.04	36.4	1.50	21.5	768	0.717	1.523
1500	12.03	171	226.0	76	7116	31.49	41.5	1.85	46.7	818	0.730	1.895
1500	12.03	171	226.0	76	7141	31.60	41.5	1.85	46.8	816	0.724	1.896
1500	15.98	227	225.9	94	6769	29.97	43.8	2.26	80.6	864	0.711	2.333
1500	15.98	227	225.9	94	6719	29.75	43.8	2.24	79.7	867	0.713	2.320

					COMPRESSOR					
VOL.EFF	A/C EFF.	Trw (K)	B.Th EFF.	COMB RATIO	BOOST RATIO	c.flow (lb/mn)	Pc (kW)	EFFc	Tic (K)	Pic (bar)
0.835	0.957	296.0	0.310	1.27	1.21	20.9	4.6	0.598	292.0	1.006
0.838	0.913	296.0	0.314	1.27	1.21	20.9	4.6	0.598	292.0	1.006
0.956	0.946	300.0	0.357	1.51	1.58	30.8	15.4	0.660	294.0	1.002
0.956	0.927	300.0	0.355	1.50	1.59	30.8	15.4	0.665	293.0	1.002
1.011	0.915	299.0	0.372	1.34	2.10	42.6	34.1	0.687	293.0	0.995
1.010	0.904	297.0	0.372	1.34	2.10	42.6	33.9	0.695	292.0	0.995
1.031	0.899	296.0	0.372	1.33	2.65	54.2	57.3	0.701	292.0	0.986
1.020	0.900	296.0	0.372	1.33	2.66	53.9	56.9	0.706	293.0	0.986

DENSITY RATIO							
COMP	A/C	OVERALL	FUEL mm3/shot	Teng (K)	dPe (bar)	SMOKE (bsch)	TURB ISEN EFF.
1.102	1.067	1.175	69.5	297.0	-0.026	0.5	1.465
1.102	1.063	1.171	68.6	298.0	-0.026	0.5	1.413
1.295	1.167	1.511	119.2	303.0	0.095	2.0	1.093
1.301	1.162	1.511	119.6	304.0	0.101	1.5	1.113
1.540	1.272	1.959	172.0	307.0	0.250	1.5	0.931
1.547	1.268	1.961	172.0	306.0	0.250	1.7	0.924
1.776	1.361	2.418	228.3	309.0	0.360	2.1	0.844
1.784	1.364	2.434	228.3	309.0	0.369	2.0	0.847

ENERGY BALANCE (OR LACK OF)					
TOTAL kW	WORK %	STACK %	COOLANT %	A'COOLER %	BALANCE %
185	0.310	0.327	0.299	0.020	0.044
183	0.314	0.330	0.303	0.019	0.034
318	0.357	0.339	0.227	0.041	0.037
319	0.355	0.340	0.232	0.040	0.033
458	0.372	0.345	0.193	0.064	0.027
458	0.372	0.346	0.193	0.063	0.026
608	0.372	0.347	0.166	0.082	0.033
608	0.372	0.346	0.166	0.082	0.034

Fig. A4.3, Typical print-out from the data reduction program.

APPENDIX 5

Closed Cycle Performance Model

This program was written for the company's Hewlett Packard 9825a micro-computer. It was produced initially to establish the effects of the heat release assumption on the development of pressure and temperature during the closed cycle. This could be achieved quickly, and then used as reliable input to the GEC 'filling and emptying' model.

The program essentially calculates the change of internal energy of the cylinder gases at one degree intervals between inlet valve closes and exhaust valve opens. Models are used to predict the energy flow due to heat transfer through the chamber wall, and the rate of heat addition due to combustion. The Woschni (49) heat transfer relation, as described in section 2.6, and the Marzouk-Watson (82) heat release model, described in section 2.7, are used. The predictor-corrector method is used to provide an accurate and rapid numerical solution.

The initial conditions, at inlet valve closure, may be input directly or derived from the required bmep and the design concept, based on assumptions of volumetric efficiency, heat pick-up and pressure loss between the manifold and the cylinder.

An overall flow diagram is given in figure A5.1 and the program listing is given in figures A5.2(a) to (g). Examples of print-out and graph plots are given in figures A5.3(a) to(c).

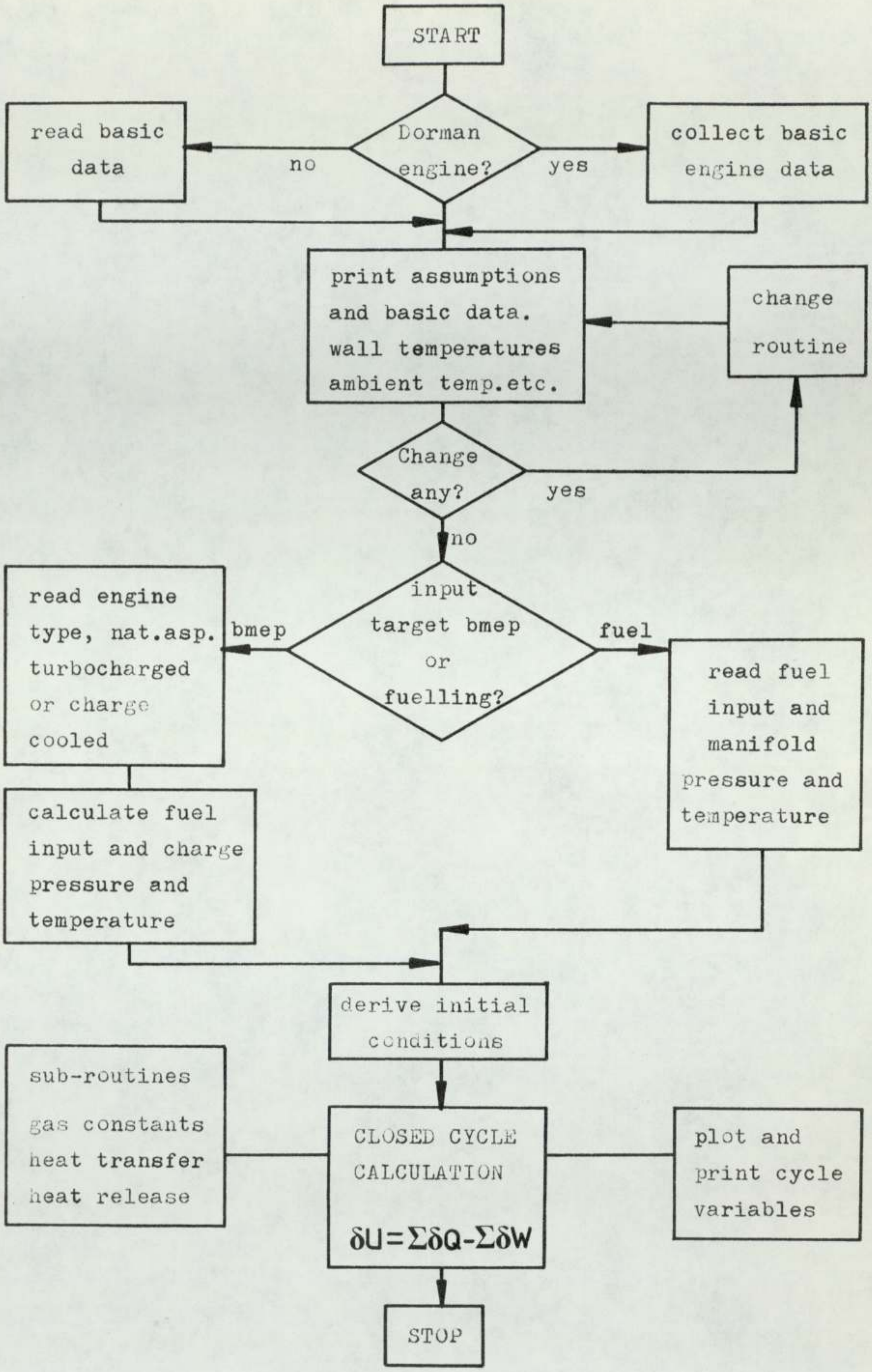


Fig.A5.I, Flow diagram of the closed-cycle performance model.

```
0: "FILE 4 CLOSED CYCLE PREDICTION ":
1: wrt 6," "
2:
3:
4:
5: "DIM":
6: dim V[3],P[12],T[12],Z$[80],E[9],M[10],C[3],O[12],R[2],H[300]
7: dim I[2],V$[20],W$[20],X$[20],Y$[20],U$[82],J[181]
8: dim B[2],A[2],D[5],W[2],N[3]
9: dim A$[6],B$[5],C$[40]
10:
11: wrt 6," "
12: wrt 6,"                              CLOSED CYCLE PERFORMANCE MODEL"
13: wrt 6,"                              =============================="
14: wrt 6," "
15: wrt 6," "
16: ent "DORMAN ENGINE---(1)  OTHER---(2)",G
17: if G=1;ent "(DA)(F)(LD)(LE)(J)(6Q)(QV)(S)?",X$;jmp 5
18: if G=2;ent "BORE(ins)",D,"STROKE(ins)",L,"CONN-ROD LENGTH(ins)",J
19: if G=2;-120→O[8];120→O[9]
20: if G=2;2.3πDD/4→A[1];550→W[1];400→W[2];gto 37
21: jmp -5
22: if cap(X$)="DA";gsb "DA"
23: if cap(X$)="F";gsb "F"
24: if cap(X$)="L";gsb "LE"
25: if cap(X$)="LE";gsb "LE"
26: if cap(X$)="LD";gsb "LD"
27: if cap(X$)="J";gsb "J"
28: if cap(X$)="6Q";gsb "6Q"
29: if cap(X$)="QV";gsb "QV"
30: if cap(X$)="S";gsb "S"
31: wrt 6,"DORMAN   '",cap(X$),"'   DIESEL ENGINE."
32: wrt 6,"--------------------------------"
33: wrt 6," "
34: wrt 6," "
35:
36:
37: ent "ENTER HEADING",U$
38: wrt 6,U$
39: wrt 6," "
40: wrt 6," "
41: ent "NUMBER OF CYLINDERS?",N[1]
42: ent "START OF INJECTION? deg.rel tdc",F
43: .85→X
44: 293→I
45: 50→K
46: 1.5→r6
47: 550→W[1]
48: 400→W[2]
49:
50: wrt 6,"ASSUMPTIONS AND BASIC DATA"
51: wrt 6,"--------------------------"
52: wrt 6," "
53: fmt 6,c30,f8.2,c16
54:
55: wrt 6.6," 1  INLET VALVE CLOSES          ",O[8]," degs rel.tdc   "
56: wrt 6.6," 2  EXHAUST VALVE OPENS         ",O[9]," degs rel.tdc   "
57: wrt 6.6," 3  FIXED WALL AREA             ",A[1]," [in2]          "
58: wrt 6.6," 4  FIXED WALL TEMPERATURE      ",W[1]," [K]            "
59: wrt 6.6," 5  CYLINDER LINER TEMPERATURE",W[2]," [K]            "
60: wrt 6.6," 6  SWIRL RATIO [Nsw/Neng]      ",r6
61: wrt 6.6," 7  CHARGE HEATING,MAN-to-BDC ",K," [K]            "
62: wrt 6.6," 8  CYLINDER BORE               ",D," [ins]          "
63: wrt 6.6," 9  STROKE                      ",L," [ins]          "
64: wrt 6.6,"10  CONN.ROD CENTRES            ",J," [ins]          "
65: wrt 6.6,"11  VOLUMETRIC EFFICIENCY       ",100X," [%]            "
66: wrt 6.6,"12  AMBIENT TEMPERATURE         ",I," [K]            "
67: wrt 6.6,"13  START OF INJECTION          ",F," degs rel.tdc   "
68: wrt 6.6,"14  NUMBER OF CYLINDERS         ",N[1],"               "
69: wrt 6," "
70: wrt 6," "
71: ent "CHANGE ANY ASSUMPTIONS? YES/NO",X$
72: if cap(X$)="NO";gto "SPEED"
73: if cap(X$)#"YES";jmp -2
74: ent "HOW MANY CHANGES?",W
75: for U=1 to W;ent "ENTER ITEM No. AND 'CONTINUE'",G
76: jmp G+2
77: next U
78: gto 50
79: ent "AVC  NB:[-ve btdc]",O[8];gto 77
*32614
```

Fig. A5·2 (a), Closed cycle program listing.

```
80: ent  "EVO [+ve atdc]",O[9];gto 77
81: ent  "FIXED WALL AREA [in2]",A[1];gto 77
82: ent  "FIXED WALL TEMP [K]",W[1];gto 77
83: ent  "CYLINDER LINER TEMP[K]",W[2];gto 77
84: ent  "SWIRL RATIO",r6;gto 77
85: ent  "CHARGE HEATING [K]",K;gto 77
86: ent  "CYLINDER BORE [ins]",D;gto 77
87: ent  "STROKE [ins]",L;gto 77
88: ent  "CONN.ROD CENTRES [ins]",J;gto 77
89: ent  "VOL.EFFICIENCY [%]",X;X/100→X;gto 77
90: ent  "AMBIENT TEMPERATURE [K]",I;gto 77
91: ent  "START OF INJECTION      ",F;gto 77
92: ent  "NUMBER OF CYLINDERS?    ",N[1];gto 77
93:
94:
95: "SPEED":L/2→R
96: ent  "SPEED (RPM)",N,"COMP.RATIO",C;.172→C[3]
97: "1=WOSCHNI  2=EICHELBERG":1→r21
98: "Vol.bdc":πDDL(1+1/(C-1))/4→V
99: "Vol.Swept":πDDL/4→V[1]
100:
101:
102: "INITIAL CONDITIONS":
103: ent  "SPECIFY-1.REQD.BMEP or 2.FUELING",W;if W=1;gto "BMEP"
104: if W#2;jmp -1
105: "FUELING":4→W
106: ent  "FUEL ENERGY per SHOT (CHU)",r25
107: ent  "MANIFOLD PRESSURE (psi)",P[1];P[1]→P;P/14.5→U
108: ent  "MANIFOLD TEMPERATURE (K)",T[1];T[1]→T
109: "AIR FLOW lb/min":.000434XPV[1]NN[1]/T→M[3]
110: "AIR-FUEL":20444M[3]/r25NN[1]→A
111: if A<18;dsp "OVERFUELLED AIR-FUEL=",A;wait 3000;gto "FUELING"
112: gto "I"
113:
114: "BMEP":
115: ent  "TARGET BMEP (psi)",O;O+(C-4+.007N+1.5(LN/6000)↑2)→B
116: "Fuel [CHU]":108.13tn↑(-6)BLDD→r25
117: ent  "1-N/A  2-TURBO   3-CHARGE-COOLED",W
118: if W=1;14.5→P
119: if W=1;.000434PXV[1]NN[1]/I→M[3];1→U
120: if W=1;20444M[3]/r25NN[1]→A
121: if W=1;if A<18;dsp "OVERFUELLED: A/F =",A;wait 3000;gto "BMEP"
122: if W=2 or W=3;ent "REQD.AIR/FUEL RATIO",A
123: if W=3;ent "CHARGE COOLANT TEMP.(K)",H,"EFFECTIVENESS [%]",E;E/100→E
124: if W=2;for U=1 to 6 by .01
125: if W=2;if 14.5UX/((U↑.2857-1)1.43+1)I>=Ar25/8.8788V[1];gto "I"
126: if W=2;next U
127:
128: if·W=3;for U=1 to 6 by .01
129: if W=3;if 14.5UX/((((U↑.286-1)1.4+1)I-H)(1-E)+H)>Ar25/8.879V[1];gto "I"
130: if W=3;next U
131: if U>=6;dsp "REQD.BOOST > 6:1";wait 4000;gto "BMEP"
132: "I":
133: if W=1;I+K→T→T[1]
134: if W=2;((U↑.2857-1)1.43+1)I+K→T→T[1];14.5U→P
135: if W=3;(((U↑.2857-1)1.43+1)I-H)(1-E)+H+K→T→T[1];14.5U→P
136: if W=2 or 3;.000434XPV[1]NN[1]/(T-K)→M[3]
137: if W=4;T+K→T→T[1];jmp 2
138: P-1→P;.0008688PV/T→M→M[1]
139:
140:
141:
142: "DURATION":
143: if W=4;ent "APPROX.COMBUSTION DURATION degCA",R[2];jmp 8
144: if W=1;15+60*B/140→R[2]
145: if W=2;15+60*B/220→R[2]
146: if W=3;15+60*B/270→R[2]
147: dsp "COMBUSTION DURATION =",R[2];wait 3000
148: ent  "CHANGE?",X$;if cap(X$)="NO";jmp 3
149: if cap(X$)#"YES";jmp -2
150: ent  "COMBUSTION DURATION?",R[2]
151: int(R[2]2.2)→R[2]
152: if W#1;if W#2;if W#3;if W#4;gto 117
153:
154: "SWIRL FACTOR=1.5":r6N/60→r6
155: P→J[180+O[8]]
156:
157:
158:
159:
*1798
```

Fig. A5·2 (b) Closed cycle program listing (cont'd)

```
160:
161: wrt 6,"DESIGN VARIABLES AND DERIVED DATA"
162: wrt 6,"---------------------------------"
163: fmt 6,c30,f8.2,c16
164: wrt 6.6," 1 CRANKSHAFT SPEED             ",N," [rev/min]      "
165: wrt 6.6," 2 HEAT RELEASE DURATION        ",int(R[2]/2.2)," [degs]         "
166: wrt 6.6," 3 COMPRESSION RATIO            ",C,"               "
167: wrt 6.6," 4 AIR-FUEL RATIO               ",A,"               "
168: wrt 6.6," 5 FUEL INPUT                   ",r25," [CHU]          "
169: wrt 6.6," 6 SWIRL ROTATION               ",60r6," [rev/min]      "
170: wrt 6.6," 7 TARGET BMEP                  ",O," [psi]          "
171: wrt 6.6," 8 AIR-FLOW                     ",M[3]," [lb/min]       "
172: wrt 6.6," 9 MANIFOLD PRESSURE            ",U14.5," [psi]          "
173: wrt 6.6,"10 MANIFOLD TEMPERATURE         ",T-K," [K]            "
174: M[3]I1.005(U↑.2857-1)/(2.205*60*.7457)→O
175: if W=3 or 3;wrt 6.6,"11 COMPRESSOR POWER            ",O," [BHP]          "
176: if W=3;wrt 6.6,"12 CHARGE COOLANT TEMPERATURE ",H," [K]            "
177: if W=3;wrt 6.6,"13 CHARGE COOLER EFFECTIVENESS",E," [K]            "
178: wrt 6," "
179: wrt 6," "
180: if W=1;"NAT. ASPIRATED"→U$
181: if W=2;"TURBOCHARGED  "→U$
182: if W=3;"CHARGE COOLED "→U$
183: if W=1 or 2 or 3;wrt 6,"DESIGN CONCEPT:";U$
184: if W=1 or 2 or 3;wrt 6,"-------------- "
185: wrt 6," "
186: wrt 6," "
187:
188: gsb "BDC-AVC"
189:
190:
191:
192:
193:
194: gto "END"
195:
196: "DA":4.1339→D;4.7255→L;8.7756→J;-115→O[8];115→O[9];2.3πDD/4→A[1];ret
197: "F":4.3307→D;4.7244→L;9.2756→J;-115→O[8];115→O[9];2.3πDD/4→A[1];ret
198: "LD":5→D;5.118→L;10.51→J;-139→O[8];134→O[9];2.3πDD/4→A[1];ret
199: "LE":5→D;5.875→L;10.51→J;-139→O[8];134→O[9];2.3πDD/4→A[1];ret
200: "J":5.118→D;4.9213→L;10.4724→J;-140→O[8];140→O[9];2.3πDD/4→A[1];ret
201: "6Q":6.25→D;6.5→L;11.7441→J;-138→O[8];129→O[9];2.3πDD/4→A[1];ret
202: "QV":6.25→D;6.5→L;12.626→J;-138→O[8];129→O[9];2.3πDD/4→A[1];ret
203: "S":6.25→D;7.5→L;13.626→J;-138→O[8];129→O[9];2.3πDD/4→A[1];ret
204:
205:
206:
207: "BDC-AVC":
208: for G=-180 to O[8] by 1
209: πDD(R+J-Rcos(G)-√(JJ-RRsin(G)↑2)+L/(C-1))/4→W
210: P(V/W)↑1.35→J[181+G]
211: if G=O[8];P(V/W)↑1.35→P→P[1];T(V/W)↑.35→T→T[1]
212: next G
213: ret
214:
215:
216:
217:
218: "END":ent "TABLE OF CYCLE VARIABLES? YES/NO",X$
219: if cap(X$)="NO";sfg 5;gto 225
220: if cap(X$)#"YES";jmp -2
221: wrt 6,"TABLE OF CYCLE VARIABLES"
222: wrt 6,"------------------------"
223: wrt 6,"CRANK ANGLE VOLUME PRESSURE TEMPERATURE"
224: wrt 6,"  [degs]     [in3]   [psi]      [K]    "
225: 1→Y;trk 0;ldf 5
*5436
```

Fig. A5·2 (c), Closed cycle program listing (cont'd)

```
0: "FILE 5 CLOSED CYCLE PREDICTION [cont'd]":
1: if Y=0;trk 0;ldf 4
2: dim F[-180:180]
3: dim K[-180:180]
4: 0→Q→I→K
5: O[8]→O[1];1→O[2];gsb "VOLUME"
6: gto 17
7:
8: "IGN.DELAY":if r25=0;ret
9: P[1]/14.5+S→S;T[1]+O→O
10: S/(O[1]-F+1)→r30;O/(O[1]-F+1)→X
11: "ID-degs":.006N3.52exp(2100/X)/r30↑1.022→I
12: if int(I+.5)<=O[1]-F+1;int(I+.5)→I;F+I→H[1];gsb "M-W"
13: ret
14:
15:
16:
17: "DRAW":
18: 1→Y
19:
20:
21:
22:
23:
24: "RUN":
25: if Y=2;gsb "VOLUME"
26: if O[1]>O[9];gto "END"
27: if H[1]#0 and O[1]<=F+I+r23;gto "BURN"
28:
29:
30:
31:
32:
33: "WORK":
34: -P[1]V[1]/16800→E[1]
35:
36:
37:
38:
39:
40: "HR":0→E[3];if r21=1;gto "WOSCHNI"
41: gto "EICH"
42:
43:
44:
45: "BURN":0→E[2]
46: if r25=0;gto "WORK"
47: "r23 is duration to 99.5%":
48: if O[1]<H[1];gto "WORK"
49: r25H[(O[1]-F-I)2+2]→E[2];2r23+2→H
50: if Y=1;E[2]/10222.22→M[2];M[1]+M[2]→M[1]
51: gto "WORK"
52:
53:
54:
55: "EICH":.48(.00084667LN)↑(1/3)(T[1]P[1]/14.7)↑(1/2)→K
56: gto 65
57:
58:
59:
60: "WOSCHNI":110(.0254D)↑(-.2)(P[1].0703)↑.8T[1]↑(-.53)→r8
61: if r25=0 or O[1]<H[1];0→r10;gto "K"
62: .00324πTDDL(P[1]-P(V[2]/V)↑1.35)/4PV→r10
63: "K":(2.28NL.00084667+.0245773Dr6+r10)↑.8→r9;.2048124r8r9→K
64: 4(V[2]-πDDL/4(C-1))/D→A[2]
65: for G=1 to 2;-KA[G](T[1]-W[G])/144N21600+E[3]→E[3];next G
66:
67:
68:
69: "GAMMA":(M[1]-M)/M→Z
70: if T[1]>=600;.0881T[1]-.00002165T[1]↑2+Z(.3417T[1]-.00005583T[1]↑2)→C[3]
71: if T[1]>=600;.1378+Z.1383+C[3]/1000→C[3];gto 76
72: if T[1]>=480;.1507+.000053T[1]+Z(.583+.000268T[1])→C[3];gto 76
73: .1617+.00003T[1]+Z(.0667+.0004817T[1])→C[3]
74:
75:
76: "NUMERICAL SOLUTION":
77:
78: E[1]+E[2]+E[3]→E[4];E[4]/M[1]C[3]→T[2];1+Y→Y;if Y>2;gto "P-C"
79: "P=mRT/V":108M[1]53.3(T[1]+T[2])/5V[2]→P[3]
*17801
```

Fig. A5·2 (d), Closed cycle program listing (cont'd)

```
80: T[1]+T[2]→T[3]
81:
82: T[1]→T[10];P[1]→P[10];O[1]→O[10];T[2]→T[11];M[1]→M[10]
83: T[3]→T[1];P[3]→P[1];O[1]+1→O[1]
84: gto "RUN"
85:
86: "P-C":
87: (T[2]+T[11])/2→T[2]
88: "P=mRT/V":108M[1]53.3(T[10]+T[2])/5V[2]→P[1]
89: T[10]+T[2]→T[1]
90: dsp O[1],P[1],T[1],Y
91: if abs(T[2]/T[11])>1.1 or abs(T[2]/T[11])<.9;T[2]→T[11];gto "RUN"
92: "WALL HEAT LOSS":
93: E[3]+Q→Q
94: if O[1]<=0;P[1]→J[181+O[1]]
95: if O[1]>0;(P[1]-J[181-O[1]])V[1]+I[2]→I[2]
96: "PRESSURE LOG":
97: P[1]→F[O[1]]
98: T[1]→K[O[1]]
99: if O[1]>=F and D[2]=0;gsb "IGN.DELAY"
100: "Pmax":if P[1]>P[4];P[1]→P[4]
101: if O[1]/5#int(O[1]/5);jmp 3
102:
103:
104:
105:
106:
107:
108: if flg5;gto "DRAW"
109: if O[1]/5#int(O[1]/5);gto "DRAW"
110: fmt 8,f7.0,f11.1,f8.1,f10.1
111: wrt 6.8,O[1],V[2],P[1],T[1]
112: gto "DRAW"
113:
114:
115:
116:
117:
118:
119:
120: "END":gsb "EVO-BDC"
121: wrt 6," "
122: wrt 6," "
123: wrt 6,"CYCLE RESULTS"
124: wrt 6,"-------------"
125: C-4+.007N+1.5(LN/6000)↑2→W
126: fmt 6,c30,f8.2,c16
127: wrt 6.6,"BMEP                          ",I[2]-W," [psi]          "
128: wrt 6.6,"IMEP                          ",I[2]," [psi]          "
129: wrt 6.6,"FMEP                          ",W," [psi]          "
130: wrt 6.6,"MECHANICAL EFFICIENCY         ",(I[2]-W)/I[2]," [%]          "
131: wrt 6.6,"HEAT LOSS                     ",Q100/r25," [%]          "
132: wrt 6.6,"MAX.CYCLE PRESSURE            ",P[4]," [psi]          "
133: P[4]/14.5UC↑1.35→P[4]
134: wrt 6.6,"EXPLOSION RATIO  [Pmax/Pcomp] ",P[4]
135: (I[2]-W)πLDDNN[1]/3.186tn↑6→P[4]
136: wrt 6.6,"BRAKE POWER                   ",P[4]," [BHP]          "
137: r25NN[1]30/10222P[4]→P[4]
138: fmt 7,c30,f8.4,c16
139: wrt 6.7,"BSFC                          ",P[4]," [lb/BHPh]      "
140: wrt 6.6,"IGNITION DELAY                ",I," [degs]         "
141: wrt 6,"";wrt 6,"";wrt 6,""
142:
143:
144:
145:
146:
147: ent "HEAT RELEASE TABLE? YES/NO",X$
148: if cap(X$)="YES";gsb "TABLE"
149: if cap(X$)#"NO";jmp -2
150:
151: " "→X$;ent "PRESSURE/CRANK ANGLE PLOT?",X$
152: if cap(X$[1,1])="Y";gto "PLOT"
153: if cap(X$[1,1])#"N";jmp -2
154: stp
155:
156: "TABLE":
157: fmt 4,10x,f16.6,10x,f16.6,10x,f16.6
158: 0→I→N
159: fxd 5;0→W
*31894
```

Fig.A5·2 (e), Closed cycle program listing (cont'd)

```
for H=3 to 2r23+1 by 2
N+1→N
H[H+1]+W→W
if N/2#int(N/2);wrt 6.4,H[H],H[H+1],W
next H
ret



"VOLUME":0→W
for G=O[1]-.5 to O[1]+.5 by .5;W+1→W
πDD(R+J-Rcos(G)-√(JJ-RRsin(G)↑2)+L/(C-1))/4→V[W]
next G
V[3]-V[1]→V[1]
ret



"M-W":
2+1.25tn↑(-8)(I/.006)↑2.4→D[1]
5000→D[2]
"EQUIVALENCE RATIO":14.5/A→A
A↑(-.644)*14.2→D[3]
.791D[3]↑.248→D[4]
"PREMIXED BURNING FRACTION":1-.926A↑.37/(I/.006N)↑.26→D[5]
if D[5]<0;0→D[5]
0→K;cfg 1
for G=0 to 1 by 1/R[2]
G→T[8]
if flg1;0→B[2];jmp 3
"PREMIXED":D[5]D[1]D[2]T[8]↑(D[1]-1)(1-T[8]↑D[1])↑(D[2]-1)→B[2]
if B[2]<tn↑(-10) and T[8]>0;sfg 1
"DIFFUSION":(1-D[5])D[3]D[4]T[8]↑(D[4]-1)exp(-D[3]T[8]↑D[4])+B[2]→B[1]
B[1]/R[2]→H[2G*R[2]+2];F+I+GR[2]→H[2GR[2]+1]
B[1]/R[2]+K→K
if K>.995;gto "END1"
next G
"END1":R[2]G→r23
ret






"EVO-BDC":
V[2]→V
for G=O[9]+1 to 180;0→H
for W=G-.5 to G+.5 by .5;H+1→H
πDD(R+J-Rcos(W)-√(JJ-RRsin(W)↑2)+L/(C-1))/4→V[H]
next W
(P[1](V/V[2])↑1.35-J[181-G])(V[3]-V[1])+I[2]→I[2]
next G
4I[2]/πDDL→I[2]
ret


"POS":"     "→B$
str(B)→A$;len(A$)→r33
A$[1,r33]→B$[6-r33,5]
for E=1 to 5
if B$[E,E]="0";"O"→B$[E,E]
next E
ret


"PLOT":fxd 0;cfg 6
ent "NEW PLOTTING SHEET?",X$
if cap(X$)="NO";sfg 6;ent "LINE STYLE? 0-6",X;line X;gto "CURV"
if cap(X$)#"YES";gto "PLOT"
max(F[*])→r31;200int(r31/200)+200→r32
wrt 705,"IN"
scl 1.5O[8],1.1O[9],-.25r32,1.25r32

plt O[9],r32;plt O[9],0;plt O[8],0;plt O[8],r32,-1
plt 0,r32;plt 0,0,-1;plt O[9],r32;plt O[8],r32,-1

for B=r32 to 0 by -200
gsb "POS"
plt 1.2O[8],B,0;lbl B$;plt 1.025O[8],B;plt O[8],B,-1
*6194
```

Fig. A5·2 (f), Closed cycle program listing (cont'd)

```
next B
20int(O[8]/20)→r34;20int(O[9]/20)→r35
for B=r34+20 to r35 by 20
gsb "POS"
plt B,0;plt B,-.02r31,-1;plt B-.05(O[9]-O[8]),-.06r31,0;lbl B$
next B
csiz 1.5,2,1,90
plt 1.22O[8],.2r32,0;lbl "CYLINDER PRESSURE - lb/sq.in"
csiz 1.5,2,1,0
plt O[8]-.03(O[9]-O[8]),-.12r31,0;lbl "AVC"
plt O[9]-.03(O[9]-O[8]),-.12r31,0;lbl "EVO"
csiz 2,2,1,0
plt O[8]+.4(O[9]-O[8]),-.2r31,0;lbl "CRANK ANGLE"
"CURV":pen
for B=O[8]+1 to O[9]
if B<-50 and frc(B/2)#0;gto "NB"
if B>50 and frc(B/2)#0;gto "NB"
plt B,F[B]
"NB":next B
pen
ent "HEAT RELEASE PLOT?",X$;if cap(X$)="NO";jmp 9
if cap(X$)#"YES";jmp -1
0→N[3]
for G=2 to R[2] by 2
if H[G]>N[3];H[G]→N[3]
next G
for G=2 to R[2] by 2;H[G]400/N[3]→H[G];next G
for G=2 to R[2] by 2
if H[G-1]=0;pen;jmp 2
plt H[G-1],H[G];if H[G-1]=0;pen
next G
pen
ent "TEMPERATURE PLOT?",X$;if cap(X$)="NO";gto "HEADING"
if cap(X$)#"YES";jmp -1
if flg6;gto "TCURV"
max(K[*])→r36;200int(r36/200)+200→r37
for G=O[8] to O[9];K[G]r32/r37→K[G];next G
for B=0 to r37 by 200
plt O[9],Br32/r37;plt 1.025O[9],Br32/r37,-1
next B
for B=r37 to 0 by -200
csiz 1.5,2,1,0
gsb "POS"
plt 1O[9],Br32/r37,0;lbl B$;next B
csiz 1.5,2,1,90
plt 1.2O[9],.2r32,0;lbl "CYLINDER TEMPERATURE [K]"
dsp "CHANGE PEN?";stp
"TCURV":
for B=O[8]+1 to O[9]
if B<-50 and frc(B/2)#0;gto "TNB"
if B>50 and frc(B/2)#0;gto "TNB"
plt B,K[B]
"TNB":next B
pen
pen
"HEADING":
csiz 2.8,2,1,0;" "→C$
ent "Enter your heading",C$
plt O[8],1.1r32,0;lbl C$
plt 1.1O[9],1.25r32,0;pen
end
*11106
```

Fig. A5·2 (g) Closed cycle program listing (cont'd)

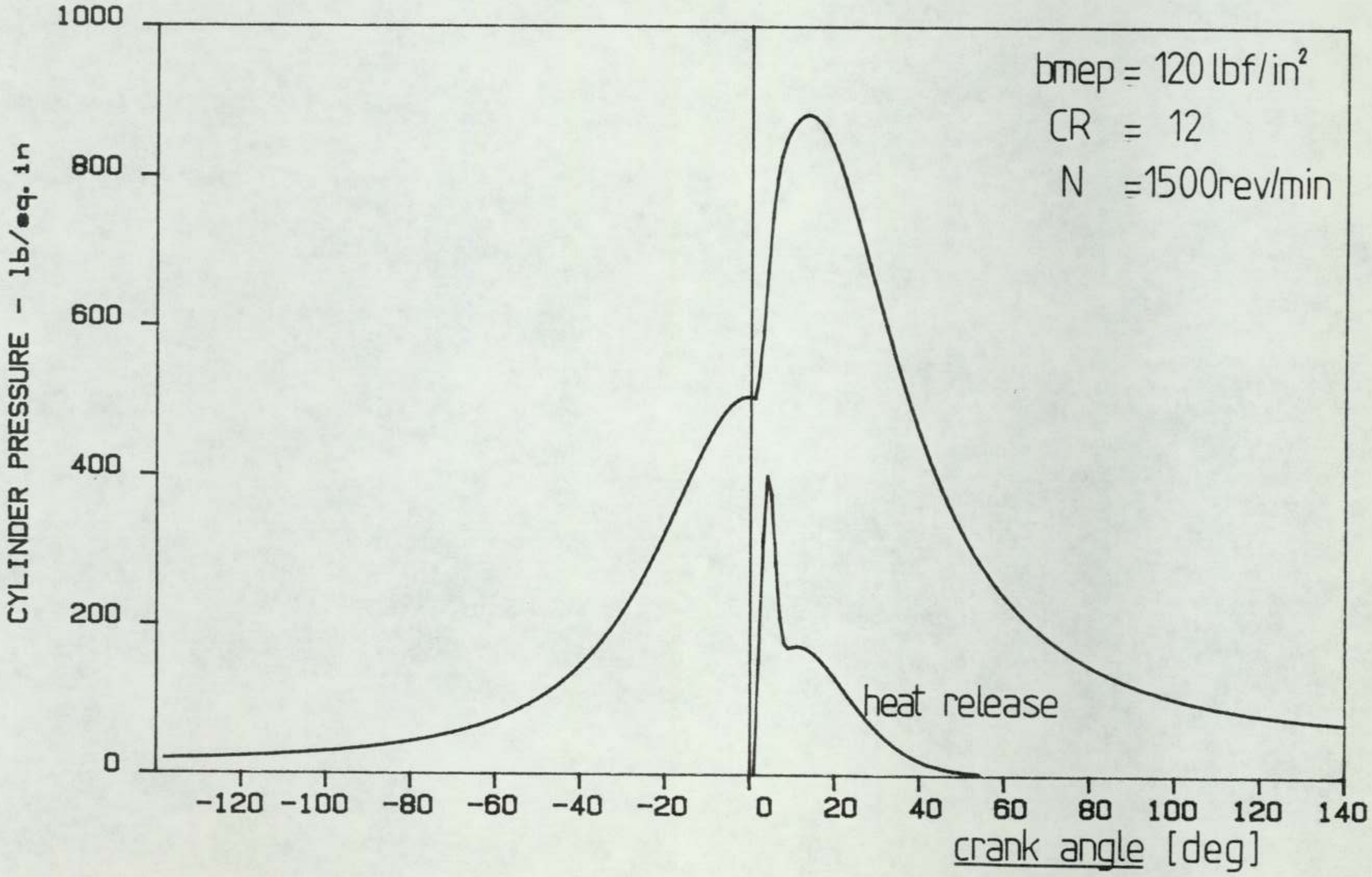


Fig. A5·3 (a), Example of the output of the closed-cycle program.

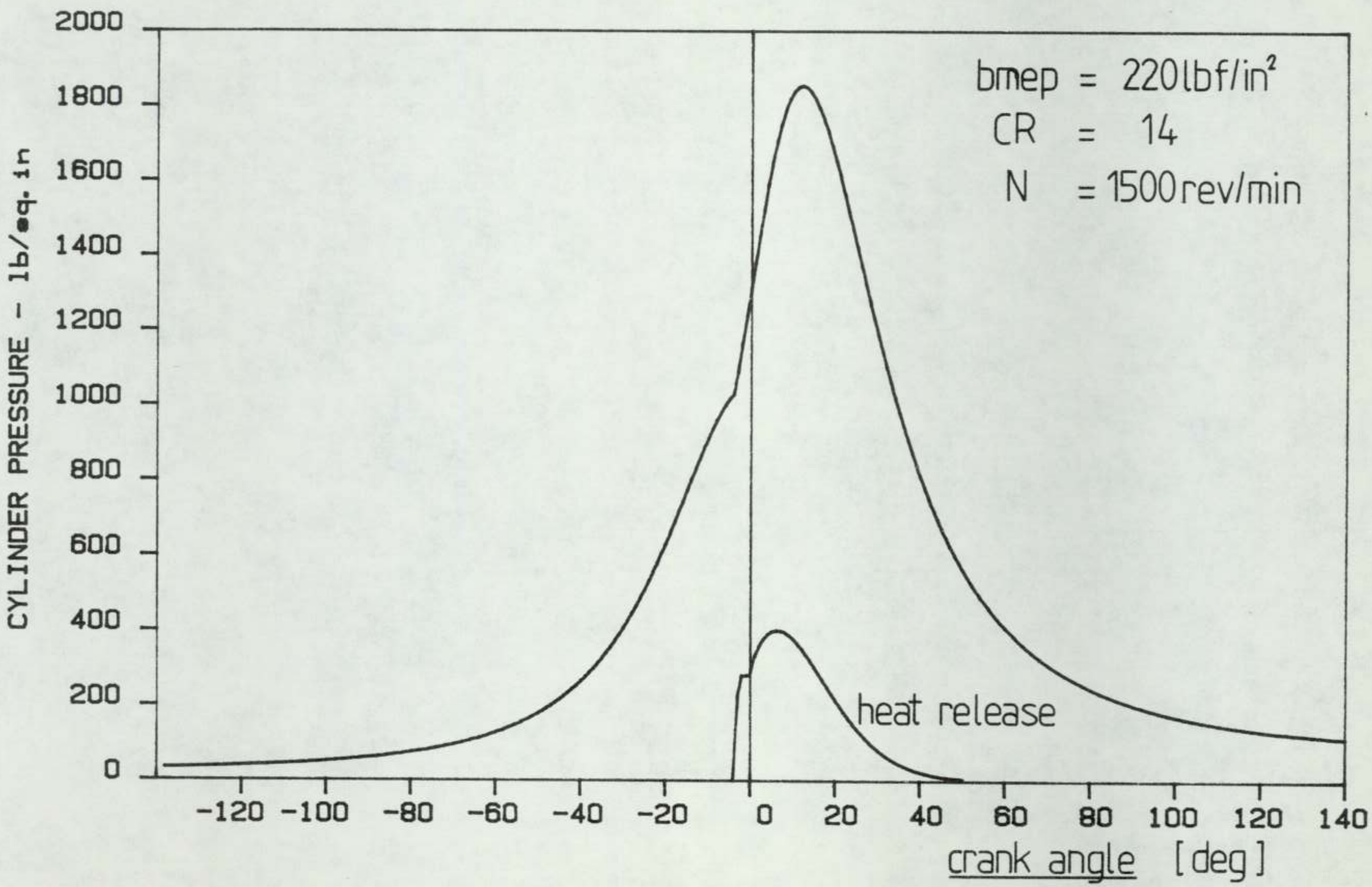


Fig. A5·3 (b), Example of the output the closed-cycle program.

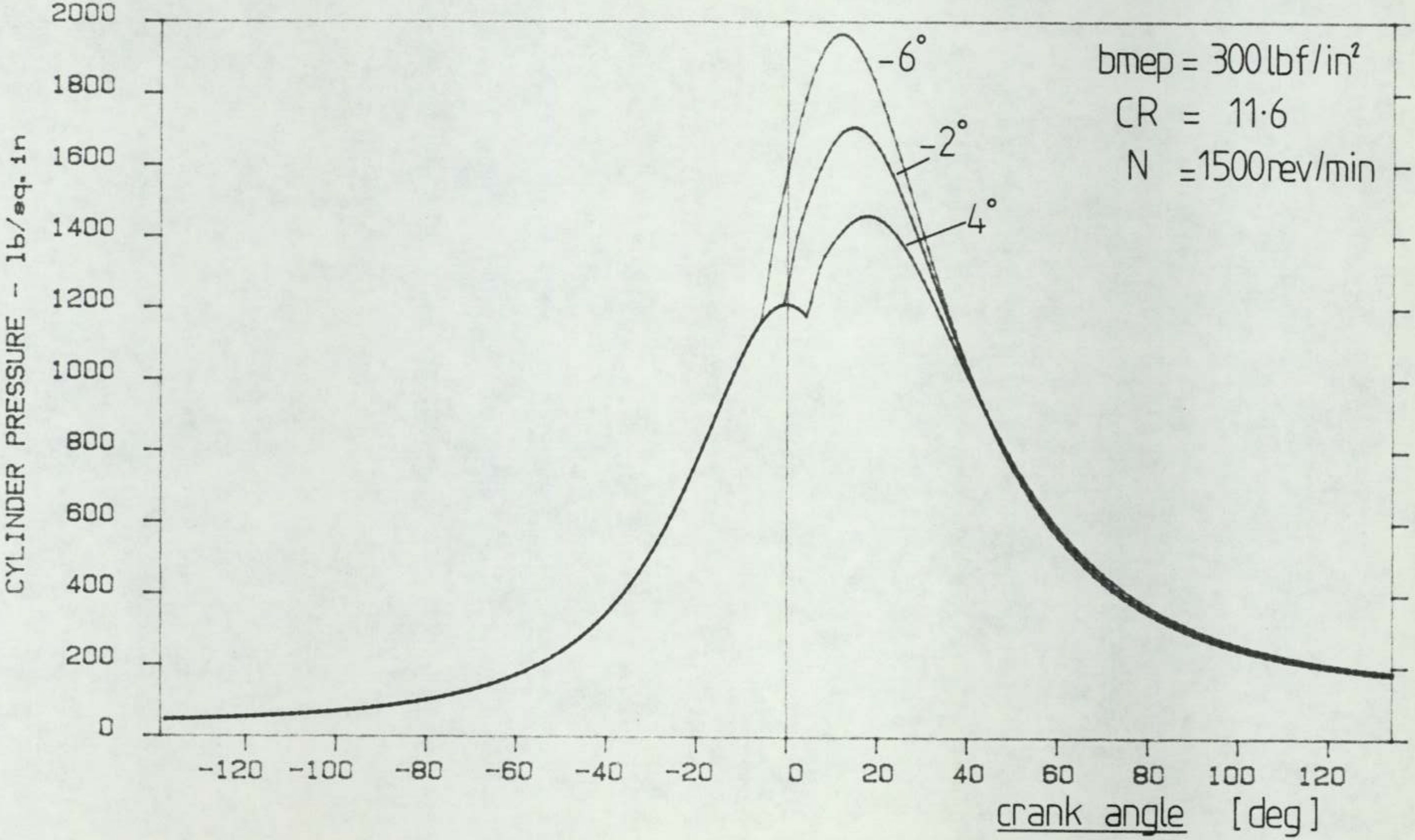


Fig. 5·3 (c), Study of heat release timing using the closed-cycle program.

APPENDIX 6

Experimental Results

The experimental results that contribute to the understanding of the high output concept and its operation are presented graphically in Chapter 6. In this appendix a small number of sets of processed data are given. However, the other sets of reduced data and the raw data, which did not warrant inclusion in the thesis, may be viewed in the company's Development Department.

6LE HIGH OUTPUT EXPERIMENTAL ENGINE LOG SHEET No. 72 18.12.79

Light spring diagram recorded.

COMPRESSSION RATIO 10.7
TURBOCHARGER BUILD Tv61 W-4/1.15G
FUEL PUMP MAXIMEC,12mm Lift,12mm Dia Plungers,4mm inlet stroke.
INJECTOR NOZZLES L5386
SPILL TIMING btdc 22

BAROMETER 749.5(mm)
FUEL S.G. 0.832
FUEL TEMP. 22.0(degC)

								TURBINE				
SPEED (RPM)	BMEP (bar)	POWER (bkW)	BSFC (g/kWh)	Pmax (bar)	BSAC (g/kWh)	AIR/ FUEL	c.flow (lb/mn)	EXP RATIO	Pt (kW)	Tit (K)	EFFtm	Pit (bar)
1500	15.98	227	220.9	0	6623	29.98	43.9	2.24	77.7	858	0.718	2.272
1500	17.95	254	220.0	0	6455	29.34	44.3	2.46	97.4	888	0.708	2.508
1500	21.01	298	224.2	0	6085	27.14	44.6	2.76	127.7	944	0.705	2.838
1500	21.01	298	229.7	0	6146	26.76	44.7	2.80	131.7	953	0.706	2.880
1500	21.01	298	231.5	0	6137	26.51	44.6	2.80	131.4	952	0.703	2.880
1500	21.01	298	227.8	0	6120	26.86	44.5	2.78	129.2	944	0.701	2.864

						COMPRESSOR				
VOL.EFF	A/C EFF.	Trw (K)	B.Th EFF.	COMB RATIO	BOOST RATIO	c.flow (lb/mn)	Pc (kW)	EFFc	Tic (K)	Pic (bar)
1.028	0.913	297.0	0.381	0.00	2.61	54.1	55.8	0.699	292.0	0.975
1.028	0.908	299.0	0.382	0.00	2.90	59.6	69.0	0.698	292.0	0.970
1.021	0.895	297.0	0.375	0.00	3.28	66.4	90.0	0.676	294.0	0.963
1.025	0.888	295.0	0.366	0.00	3.32	67.2	93.0	0.668	294.0	0.962
1.021	0.888	293.0	0.363	0.00	3.30	67.1	92.4	0.669	294.0	0.962
1.021	0.893	292.0	0.369	0.00	3.27	66.9	90.5	0.674	294.0	0.962

DENSITY RATIO COMP	A/C	OVERALL	FUEL mm3/shot	Teng (K)	dPe (bar)	SMOKE (bsch)	TURB ISEN EFF.
1.755	1.366	2.397	222.8	308.0	0.257	1.8	0.915
1.867	1.406	2.625	249.2	312.0	0.296	2.0	0.881
1.984	1.480	2.936	297.3	315.0	0.307	2.3	0.850
1.984	1.489	2.953	304.5	315.0	0.284	2.4	0.843
1.979	1.495	2.960	307.0	313.0	0.270	2.2	0.854
1.975	1.495	2.953	302.1	311.0	0.260	2.4	0.855

ENERGY BALANCE					
TOTAL kW	WORK %	STACK %	COOLANT %	A'COOLER %	BALANCE %
595	0.381	0.337	0.159	0.081	0.042
666	0.382	0.343	0.158	0.088	0.028
794	0.375	0.343	0.145	0.098	0.039
813	0.366	0.343	0.142	0.099	0.049
820	0.363	0.338	0.157	0.099	0.043
807	0.369	0.338	0.145	0.100	0.048

Fig.A6·1 (a) Experimental results

6LE HIGH OUTPUT EXPERIMENTAL ENGINE   LOG SHEET No. 75   23.1.80

COMPRESSSION RATIO	11.6
TURBOCHARGER BUILD	Tv61 W-4/1.00G
FUEL PUMP	MAXIMEC, 12mm Lift, 12mm Dia Plungers, 4mm inlet stroke.
INJECTOR NOZZLES	L5386
SPILL TIMING btdc	22
BAROMETER	737.7(mm)
FUEL S.G.	0.834
FUEL TEMP.	18.0(degC)

									TURBINE			
SPEED (RPM)	BMEP (bar)	POWER (bkW)	BSFC (g/kWh)	Pmax (bar)	BSAC (g/kWh)	AIR/ FUEL	c.flow (lb/mn)	EXP RATIO	Pt (kW)	Tit (K)	EFFtm	Pit (bar)
1500	5.92	84	250.1	40	8583	34.32	31.2	1.42	14.3	725	0.652	1.404
1500	17.76	252	223.4	114	6651	29.77	40.9	2.65	102.0	853	0.738	2.709
1500	19.73	280	224.0	120	6305	28.15	40.5	2.88	121.2	895	0.730	2.959
1500	19.73	280	224.8	120	6262	27.86	40.7	2.87	121.4	905	0.732	2.946

						COMPRESSOR				
VOL.EFF	A/C EFF.	Trw (K)	B.Th EFF.	COMB RATIO	BOOST RATIO	c.flow (lb/mn)	Pc (kW)	EFFc	Tic (K)	Pic (bar)
0.898	0.925	301.0	0.336	1.07	1.41	26.5	9.3	0.680	296.0	0.978
1.004	0.892	295.0	0.377	1.43	3.07	62.2	75.3	0.692	287.0	0.954
0.988	0.897	298.0	0.376	1.39	3.36	66.0	88.5	0.683	289.0	0.950
1.001	0.896	303.0	0.374	1.39	3.34	65.8	88.9	0.677	291.0	0.951

DENSITY RATIO COMP	A/C	OVERALL	FUEL mm3/shot	Teng (K)	dPe (bar)	SMOKE (bsch)	TURB ISEN EFF.
1.219	1.113	1.356	93.2	304.0	-0.014	0.0	1.109
1.930	1.418	2.737	249.7	311.0	0.219	0.0	0.814
2.023	1.457	2.948	278.2	315.0	0.230	0.0	0.789
2.011	1.447	2.909	279.2	320.0	0.230	2.3	0.798

ENERGY BALANCE (OR LACK OF)

TOTAL kW	WORK %	STACK %	COOLANT %	A'COOLER %	BALANCE %
249	0.336	0.313	0.222	0.030	0.099
668	0.377	0.317	0.167	0.092	0.047
745	0.376	0.321	0.150	0.098	0.056
747	0.374	0.321	0.149	0.096	0.059

Fig. A6.1 (b) Experimental results

6LE HIGH OUTPUT EXPERIMENTAL ENGINE LOG SHEET No. 79 13.2.80

COMPRESSSION RATIO 11.6
TURBOCHARGER BUILD TV61 W-4/1.00G
FUEL PUMP MAXIMEC,12mm Lift,12mm Dia Plungers,4mm inlet stroke.
INJECTOR NOZZLES L5386
SPILL TIMING btdc 22

BAROMETER 757.2(mm)
FUEL S.G. 0.830
FUEL TEMP. 19.0(degC)

									TURBINE			
SPEED (RPM)	BMEP (bar)	POWER (bkW)	BSFC (g/kWh)	Pmax (bar)	BSAC (g/kWh)	AIR/FUEL	c.flow (lb/mn)	EXP RATIO	Pt (kW)	Tit (K)	EFFtm	Pit (bar)
1500	7.99	113	227.4	76	7571	33.30	35.5	1.46	18.7	736	0.707	1.482
1500	7.99	113	227.4	76	7689	33.81	35.8	1.47	19.4	738	0.705	1.495
1500	17.95	254	214.8	116	6359	29.60	44.1	2.36	88.9	862	0.721	2.442
1500	17.95	254	219.1	116	6355	29.01	44.1	2.37	90.2	869	0.720	2.456
1500	20.03	284	226.1	120	6189	27.37	44.4	2.59	111.7	906	0.712	2.707
1500	20.03	284	222.8	121	6189	27.78	44.5	2.59	111.9	908	0.710	2.707
1500	21.01	298	222.8	122	6008	26.97	44.4	2.68	121.2	935	0.701	2.807

							COMPRESSOR			
VOL.EFF	A/C EFF.	Trw (K)	B.Th EFF.	COMB RATIO	BOOST RATIO	c.flow (lb/mn)	Pc (kW)	EFFc	Tic (K)	Pic (bar)
0.924	0.930	288.0	0.370	1.82	1.52	29.7	13.3	0.670	290.0	1.003
0.937	0.914	287.0	0.370	1.82	1.53	30.1	13.7	0.668	289.0	1.003
1.024	0.880	289.0	0.392	1.56	2.79	57.1	64.1	0.697	290.0	0.984
1,026	0.883	290.0	0.384	1.56	2.80	57.3	65.0	0.694	292.0	0.984
1.024	0.877	292.0	0.372	1.46	3.09	62.7	79.5	0.690	293.0	0.978
1.024	0.882	293.0	0.378	1.46	3.09	62.7	79.5	0.690	293.0	0.978
1.030	0.871	296.0	0.378	1.44	3.22	64.4	85.0	0.693	297.0	0.976

DENSITY RATIO							
COMP	A/C	OVERALL	FUEL mm3/shot	Teng (K)	dPe (bar)	SMOKE (bsch)	TURB ISEN EFF.
1.271	1.179	1.498	114.9	292.0	0.078	1.3	1.052
1.275	1.173	1.496	114.9	292.0	0.078	1.5	1.073
1.826	1.397	2.552	243.9	306.0	0.384	2.5	0.837
1.827	1.403	2.564	248.8	307.0	0.375	2.4	0.868
1.933	1.447	2.798	286.4	312.0	0.414	3.1	0.906
1.933	1.449	2.800	282.2	312.0	0.421	2.7	0.914
1.981	1.450	2.872	296.1	318.0	0.426	3.0	0.962

ENERGY BALANCE (OR LACK OF)					
TOTAL kW	WORK %	STACK %	COOLANT %	A'COOLER %	BALANCE %
306	0.370	0.317	0.217	0.041	0.055
306	0.370	0.321	0.217	0.042	0.050
650	0.392	0.329	0.175	0.087	0.017
663	0.384	0.322	0.152	0.087	0.055
763	0.372	0.312	0.149	0.091	0.076
752	0.378	0.316	0.168	0.093	0.046
789	0.378	0.310	0.150	0.094	0.069

Fig. A6·1 (c) Experimental results

6LE HIGH OUTPUT EXPERIMENTAL ENGINE   LOG SHEET No. 82   15.2.80

COMPRESSSION RATIO 11.6
TURBOCHARGER BUILD TV W-4/1.00G
FUEL PUMP MAXIMEC,12mm Lift,12mm Dia Plungers,4mm inlet stroke.
INJECTOR NOZZLES L5386
SPILL TIMING btdc 26

BAROMETER 755.2(mm)
FUEL S.G. 0.832
FUEL TEMP. 25.0(degC)

								TURBINE				
SPEED (RPM)	BMEP (bar)	POWER (bkW)	BSFC (g/kWh)	Pmax (bar)	BSAC (g/kWh)	AIR/FUEL	c.flow (lb/mn)	EXP RATIO	Pt (kW)	Tit (K)	EFFtm	Pit (bar)
1500	4.04	57	263.4	55	9595	36.43	25.7	1.21	5.4	642	0.488	1.224
1500	4.04	57	263.4	55	9807	37.24	26.1	1.22	5.9	652	0.482	1.238
1500	7.99	113	230.8	80	7172	31.07	34.3	1.45	17.6	749	0.623	1.468
1500	7.99	113	230.8	80	7060	30.59	33.9	1.44	17.3	751	0.624	1.465
1500	12.03	171	219.7	96	6533	29.74	39.8	1.76	39.0	804	0.669	1.800
1500	12.03	171	219.7	96	6476	29.48	39.8	1.76	38.6	809	0.669	1.792
1500	15.98	227	216.2	114	6130	28.36	42.2	2.11	65.8	845	0.678	2.171
1500	15.98	227	216.2	114	6083	28.14	42.0	2.11	65.4	847	0.676	2.171

						COMPRESSOR				
VOL.EFF	A/C EFF.	Trw (K)	B.Th EFF.	COMB RATIO	BOOST RATIO	c.flow (lb/mn)	Pc (kW)	EFFc	Tic (K)	Pic (bar)
0.802	0.750	298.0	0.319	1.73	1.16	19.3	2.6	0.764	297.0	1.004
0.811	0.778	298.0	0.319	1.71	1.18	19.8	2.9	0.793	298.0	1.004
0.916	0.896	298.0	0.364	1.94	1.52	28.7	11.0	0.784	298.0	1.000
0.910	0.896	299.0	0.364	1.95	1.52	28.3	10.8	0.786	299.0	1.000
0.983	0.904	298.0	0.383	1.80	1.97	39.5	26.1	0.766	298.0	0.994
0.980	0.894	297.0	0.383	1.81	1.97	39.3	25.8	0.769	299.0	0.994
1.005	0.878	297.0	0.389	1.72	2.47	49.6	44.6	0.766	298.0	0.987
0.994	0.880	296.0	0.389	1.72	2.46	49.4	44.3	0.766	299.0	0.987

DENSITY RATIO COMP	A/C	OVERALL	FUEL mm3/shot	Teng (K)	dPe (bar)	SMOKE (bsch)	TURB ISEN EFF.
1.096	1.037	1.137	67.2	302.0	-0.043	0.5	1.702
1.108	1.042	1.154	67.2	302.0	-0.041	0.6	1.555
1.298	1.136	1.475	116.4	303.0	0.070	1.1	1.144
1.299	1.129	1.467	116.4	304.0	0.070	1.1	1.132
1.523	1.238	1.886	166.8	306.0	0.200	1.2	0.977
1.525	1.235	1.882	166.8	306.0	0.189	1.3	0.998
1.749	1.314	2.299	218.0	311.0	0.306	1.4	0.858
1.744	1.326	2.314	218.0	310.0	0.314	1.5	0.856

	ENERGY BALANCE (OR LACK OF)				
TOTAL kW	WORK %	STACK %	COOLANT %	A'COOLER %	BALANCE %
179	0.319	0.267	0.240	0.010	0.164
179	0.319	0.282	0.240	0.012	0.146
311	0.364	0.295	0.198	0.031	0.111
311	0.364	0.293	0.238	0.031	0.075
446	0.383	0.304	0.198	0.052	0.062
446	0.383	0.304	0.174	0.053	0.087
582	0.389	0.308	0.195	0.067	0.041
592	0.389	0.306	0.195	0.068	0.041

Fig. A6·1 (d) Experimental results

6LE HIGH OUTPUT EXPERIMENTAL ENGINE   LOG SHEET No. 84   20.2.80

COMPRESSSION RATIO   11.6
TURBOCHARGER BUILD   Tv61 W-4/1.00G
FUEL PUMP   MAXIMEC,12mm Lift,12mm Dia Plungers,4mm inlet stroke.
INJECTOR NOZZLES   L5386
SPILL TIMING btdc   24

BAROMETER   758.1(mm)
FUEL S.G.   0.832
FUEL TEMP.   23.5(degC)

									TURBINE			
SPEED (RPM)	BMEP (bar)	POWER (bkW)	BSFC (g/kWh)	Pmax (bar)	BSAC (g/kWh)	AIR/ FUEL	c.flow (lb/mn)	EXP RATIO	Pt (kW)	Tit (K)	EFFtm	Pit (bar)
1500	4.04	57	258.0	55	9568	37.08	25.3	1.22	5.7	647	0.485	1.242
1500	4.04	57	262.2	55	9568	36.50	25.4	1.22	5.7	649	0.483	1.242
1500	7.99	113	226.8	78	7135	31.46	34.0	1.43	16.8	736	0.661	1.458
1500	7.99	113	226.1	78	7208	31.88	34.1	1.44	17.2	733	0.652	1.469
1500	12.03	171	211.4	94	6612	31.27	39.6	1.77	38.7	787	0.688	1.810
1500	12.03	171	216.0	94	6612	30.60	39.7	1.76	38.6	787	0.689	1.807
1500	15.98	227	213.8	114	6195	28.97	42.6	2.10	65.7	840	0.695	2.170
1500	15.98	227	213.8	114	6198	28.99	42.5	2.11	66.1	842	0.691	2.175
1500	17.95	254	214.9	118	6027	28.04	43.2	2.29	82.2	869	0.691	2.381

							COMPRESSOR			
VOL.EFF	A/C EFF.	Trw (K)	B.Th EFF.	COMB RATIO	BOOST RATIO	c.flow (lb/mn)	Pc (kW)	EFFc	Tic (K)	Pic (bar)
0.778	0.826	294.0	0.326	1.70	1.18	19.2	2.8	0.804	299.0	1.008
0.781	0.783	294.0	0.321	1.70	1.18	19.2	2.8	0.803	299.0	1.008
0.897	0.900	286.0	0.371	1.93	1.48	28.2	11.1	0.720	297.0	1.004
0.893	0.912	286.0	0.372	1.91	1.50	28.4	11.2	0.730	294.0	1.004
0.962	0.913	286.0	0.398	1.77	1.96	39.5	26.6	0.737	294.0	0.998
0.965	0.902	286.0	0.389	1.77	1.96	39.5	26.6	0.740	294.0	0.998
0.995	0.888	287.0	0.393	1.74	2.43	49.6	45.7	0.732	296.0	0.991
0.992	0.893	289.0	0.393	1.73	2.44	49.6	45.7	0.735	295.0	0.990
0.992	0.891	291.0	0.391	1.62	2.71	54.6	56.8	0.739	297.0	0.986

DENSITY RATIO COMP	A/C	OVERALL	FUEL mm3/shot	Teng (K)	dPe (bar)	SMOKE (bsch)	TURB ISEN EFF.
1.110	1.057	1.172	65.9	298.0	-0.043	0.6	1.644
1.110	1.053	1.169	66.9	299.0	-0.043	0.7	1.545
1.265	1.176	1.488	114.3	292.0	0.041	1.1	1.152
1.274	1.172	1.494	114.0	291.0	0.054	1.1	1.135
1.502	1.277	1.917	160.6	294.0	0.179	1.3	0.961
1.506	1.269	1.911	164.1	295.0	0.181	1.4	0.945
1.708	1.360	2.322	215.6	301.0	0.276	1.4	0.886
1.716	1.353	2.322	215.6	302.0	0.289	1.5	0.881
1.831	1.395	2.554	243.4	306.0	0.349	2.0	0.844

ENERGY BALANCE (OR LACK OF)					
TOTAL kW	WORK %	STACK %	COOLANT %	A'COOLER %	BALANCE %
176	0.326	0.273	0.252	0.017	0.132
179	0.321	0.274	0.248	0.015	0.142
305	0.371	0.291	0.149	0.040	0.149
304	0.372	0.295	0.187	0.039	0.107
429	0.398	0.312	0.191	0.062	0.038
438	0.389	0.307	0.160	0.060	0.084
576	0.393	0.310	0.168	0.076	0.053
576	0.393	0.312	0.164	0.074	0.057
650	0.391	0.313	0.136	0.081	0.078

Fig. A6.1 (e), Experimental results

6LE HIGH OUTPUT EXPERIMENTAL ENGINE   LOG SHEET No. 87   25.2.80

COMPRESSSION RATIO 11.6
TURBOCHARGER BUILD Tv61 W-4/1.00G
FUEL PUMP MAXIMEC,12mm Lift,12mm Dia Plungers,4mm inlet stroke.
INJECTOR NOZZLES L5386
SPILL TIMING btdc 20

BAROMETER 763.0(mm)
FUEL S.G. 0.836
FUEL TEMP. 19.0(degC)

									TURBINE			
SPEED (RPM)	BMEP (bar)	POWER (bkW)	BSFC (g/kWh)	Pmax (bar)	BSAC (g/kWh)	AIR/FUEL	c.flow (lb/mn)	EXP RATIO	Pt (kW)	Tit (K)	EFFtm	Pit (bar)
1500	4.04	57	271.2	36	9780	36.07	26.0	1.24	6.7	681	0.562	1.271
1500	7.99	113	237.0	66	7591	32.03	35.3	1.49	20.9	773	0.654	1.532
1500	7.99	113	235.5	66	7482	31.77	34.9	1.49	20.4	772	0.672	1.525
1500	12.03	171	226.4	80	6910	30.52	40.7	1.84	45.6	832	0.685	1.895
1500	12.03	171	226.4	80	6863	30.31	40.5	1.83	45.4	834	0.684	1.893
1500	15.98	227	228.3	98	6573	28.78	43.2	2.23	79.6	887	0.678	2.330
1500	15.98	227	228.3	98	6551	28.69	43.4	2.22	78.7	887	0.679	2.313
1500	17.95	254	228.1	100	6376	27.95	43.7	2.43	98.1	916	0.673	2.555
1500	17.95	254	228.1	100	6376	27.95	43.6	2.44	98.4	916	0.671	2.562
1500	18.94	268	227.1	106	6322	27.84	44.2	2.52	107.9	928	0.676	2.658
1500	20.03	284	228.7	108	6147	26.88	44.0	2.62	116.2	939	0.670	2.767
1500	20.03	284	229.5	108	6168	26.87	43.9	2.63	117.1	940	0.671	2.781

						COMPRESSOR				
VOL.EFF	A/C EFF.	Trw (K)	B.Th EFF.	COMB RATIO	BOOST RATIO	c.flow (lb/mn)	Pc (kW)	EFFc	Tic (K)	Pic (bar)
0.771	0.848	294.0	0.310	1.08	1.21	19.5	3.8	0.700	303.0	1.014
0.924	0.903	297.0	0.355	1.52	1.58	29.9	13.7	0.738	302.0	1.010
0.921	0.898	299.0	0.357	1.53	1.58	29.4	13.7	0.714	300.0	1.010
0.995	0.903	302.0	0.372	1.41	2.08	41.2	31.2	0.732	300.0	1.003
0.994	0.912	304.0	0.372	1.41	2.09	40.9	31.0	0.737	300.0	1.003
1.013	0.902	307.0	0.368	1.36	2.67	52.4	54.0	0.742	299.0	0.994
1.025	0.907	311.0	0.368	1.37	2.66	52.3	53.4	0.745	300.0	0.994
1.016	0.908	312.0	0.369	1.26	2.97	57.2	66.1	0.739	297.0	0.989
1.019	0.908	313.0	0.369	1.26	2.96	57.2	66.1	0.737	297.0	0.989
1.026	0.905	313.0	0.370	1.28	3.10	59.8	72.9	0.730	296.0	0.988
1.008	0.885	302.0	0.368	1.27	3.19	62.0	77.8	0.731	299.0	0.985
1.009	0.875	300.0	0.366	1.26	3.21	62.3	78.6	0.731	299.0	0.983

DENSITY RATIO COMP	A/C	OVERALL	FUEL mm3/shot	Teng (K)	dPe (bar)	SMOKE (bsch)	TURB ISEN EFF.
1.116	1.091	1.218	68.9	299.0	-0.026	0.6	1.723
1.320	1.177	1.553	118.9	303.0	0.367	1.5	1.213
1.310	1.164	1.525	118.2	305.0	0.092	1.6	1.170
1.557	1.262	1.964	171.1	311.0	0.247	2.4	1.003
1.565	1.248	1.953	171.1	312.0	0.243	2.2	1.002
1.818	1.337	2.429	229.2	319.0	0.406	2.0	0.915
1.814	1.323	2.400	229.2	322.0	0.410	2.4	0.909
1.935	1.355	2.622	257.2	324.0	0.467	2.6	0.889
1.932	1.353	2.614	257.2	325.0	0.464	2.8	0.893
1.978	1.368	2.706	270.1	326.0	0.487	2.7	0.873
2.010	1.422	2.859	287.5	320.0	0.473	3.0	0.865
2.015	1.423	2.866	288.6	320.0	0.464	2.5	0.855

ENERGY BALANCE (OR LACK OF)					
TOTAL kW	WORK %	STACK %	COOLANT %	A'COOLER %	BALANCE %
185	0.310	0.286	0.240	0.024	0.141
319	0.355	0.311	0.220	0.042	0.072
317	0.357	0.313	0.221	0.040	0.069
459	0.372	0.324	0.153	0.060	0.091
459	0.372	0.324	0.188	0.059	0.058
615	0.368	0.328	0.140	0.074	0.090
615	0.368	0.327	0.168	0.072	0.064
690	0.369	0.331	0.146	0.078	0.076
690	0.369	0.330	0.143	0.077	0.081
725	0.370	0.336	0.153	0.081	0.060
772	0.368	0.327	0.187	0.088	0.031
775	0.366	0.328	0.155	0.088	0.062

Fig. A6.1(f) Experimental results

APPENDIX 7

Input and Output of the GEC computer Programs used during the Project

Figure A7.1(a) to (f) gives a summary of a typical print-out of the GEC 'filling and emptying' program. Figure A7.2(a) to (d) presents a summary of the print-out from the GEC 'Meltan' thermal analogue program, for the liner network given in figure A2.3. These programs are available through the Dorman data processing department which has a direct line to the IBM-370 of Midland Computer Services Ltd. of Stafford.

```
RUN-190-SIMP.TURBO-55LB/MIN-11.7:1-LOG.SHT.26-17.36BAR.
    1.00000    5.00000    4.00000    0.0        0.0        9.00000    15.00000    2.00000    4.00000    5.00000
    1.00000    1.00000    7.00000    8.00000    9.00000    1.00000    2.00000    13.00000   14.00000   15.00000
    1.00000    0.0


SHAFT DATA
    1.00000
  CRANK
    1.00000    0.0        0.0

    2.00000
  TURBO
    0.0        0.10000    1.00000


VOLUME DATA
    1.00000
CYLINDER
    1.00000    1.00000    0.0        1.00000    1.00000

    2.00000
CYLINDER
    1.00000    1.00000    480.00000  1.00000    1.00000

    3.00000
CYLINDER
    1.00000    1.00000    240.00000  1.00000    1.00000

    4.00000
CYLINDER
    1.00000    1.00000    600.00000  1.00000    1.00000

    5.00000
CYLINDER
    1.00000    1.00000    120.00000  1.00000    1.00000

    6.00000
CYLINDER
    1.00000    1.00000    360.00000  1.00000    1.00000

    7.00000
AIRCHEST
    0.0        0.46600    0.0        0.0

    8.00000
 EXMAN-1
    0.0        0.09760    1.00000    6.00000    352.00000  0.0

    9.00000
```

Fig. A7.1 (a) Typical input data print-out from the GEC 'filling and emptying' program.

```
EXMAN-2
    0.0        0.06700     1.00000     6.00000   225.00000     0.0


JUNCTION DATA
    1.00000
INVAL-1
    2.00000    7.00000     1.00000     1.00000     1.00000

    2.00000
EXVAL-1
    2.00000    1.00000     8.00000     2.00000     1.00000

    3.00000
INVAL-2
    2.00000    7.00000     2.00000     1.00000     1.00000

    4.00000
EXVAL-2
    2.00000    2.00000     8.00000     2.00000     1.00000

    5.00000
INVAL-3
    2.00000    7.00000     3.00000     1.00000     1.00000

    6.00000
EXVAL-3
    2.00000    3.00000     8.00000     2.00000     1.00000

    7.00000
INVAL-4
    2.00000    7.00000     4.00000     1.00000     1.00000

    8.00000
EXVAL-4
    2.00000    4.00000     9.00000     2.00000     1.00000

    9.00000
INVAL-5
    2.00000    7.00000     5.00000     1.00000     1.00000

   10.00000
EXVAL-5
    2.00000    5.00000     9.00000     2.00000     1.00000

   11.00000
INVAL-6
    2.00000    7.00000     6.00000     1.00000     1.00000
```

Fig. A7.1 (b) Typical input data print-out from the GEC 'filling and emptying' program.

```
    12.00000
 EXVAL-6
     2.00000     6.00000     9.00000     2.00000     1.00000

    13.00000
  TURB-1
     6.00000     8.00000     0.0         0.47000     0.65300     1.00000     2.00000

    14.00000
  COMP
     5.00000     0.0         7.00000     1.13160     0.90400     0.25000   298.00000     0.74000     2.00000

    15.00000
  TURB-2
     6.00000     9.00000     0.0         0.47000     0.65300     1.00000     2.00000

ENGINE DIMENSION DATA
     1.00000
     4.00000     1.00000     1.00000     5.00000     5.87500    10.51000     0.00624     1.00000     1.00000    48.70000
   550.00000     1.00000   480.00000     0.80000    25.00000

HEAT RELEASE DATA
     1.00000
     4.50000 10222.00000   -10.00000     5.00000     0.0         0.10000    10.00000     0.40000    19.00000     0.20000
    27.00000     0.15000   100.00000     0.15000

VARIABLE AREA ORIFICE DATA
     1.00000
   296.00000   584.00000     8.00000    19.00000     0.0         0.03810     0.09160     0.19300     0.51200     0.44300
     0.61000     0.76300     0.92400     1.04500     1.14200     1.20900     1.26500     1.29700     1.32800     1.34300
     1.35300     1.35900     1.35900     0.0

     2.00000
   104.00000   424.00000    10.00000    17.00000     0.0         0.03500     0.11500     0.22100     0.35900     0.51900
     0.67200     0.81300     0.93300     1.01900     1.08400     1.14300     1.18300     1.21500     1.23700     1.24700
     0.02510     0.0

INITIAL VALUES
  1500.00000 89255.00000    14.02000   301.00000    14.70000   290.00000    42.49998   365.00000   200.00000  1330.00000
    30.00000   610.00000   690.00000   822.00000    29.00000   327.00000    56.00000   935.00000    38.49998   311.00000
    26.00000   820.00000    39.00000   990.00000
```

Fig. A7.1 (c) Typical input data print-out from the GEC 'filling and emptying' program.

```
   CYLINDER              EXMAN-1             INVAL-1                     TURB-1                    TURB-2
           AIRCHEST              EXMAN-2                  EXVAL-1                        COMP                      CRANK
 -131.     44.1    38.3    25.5    39.9   0.0        0.0        0.2225E-03  0.5081E-03  0.3221E-03  1500.
          369.2   310.6   805.8   988.5                          0.4327E 70  0.4327E 70  0.4327E 70  -0.03
                                                               -0.1085E 58 -0.1085E 58 -0.1085E 58
          235.
 -126.     46.0    38.1    25.2    40.2   0.0        0.0        0.4419E-03  0.1016E-02  0.6501E-03  1500.
          374.0   310.2   790.9   982.1                          0.4327E 70  0.4327E 70  0.4327E 70  -0.04
                                                               -0.1085E 58 -0.1085E 58 -0.1085E 58
          477.
 -121.     48.3    38.0    25.2    39.8   0.0        0.0        0.6610E-03  0.1524E-02  0.9792E-03  1500.
          379.4   309.8   779.1   972.6                          0.4327E 70  0.4327E 70  0.4327E 70  -0.04
                                                               -0.1085E 58 -0.1085E 58 -0.1085E 58
          725.
 -116.     50.9    37.8    25.5    39.1   0.0        0.0        0.8831E-03  0.2032E-02  0.1306E-02  1500.
          385.4   309.4   772.3   961.3                          0.4327E 70  0.4327E 70  0.4327E 70  -0.01
                                                               -0.1085E 58 -0.1085E 58 -0.1085E 58
          981.
 -111.     54.0    37.7    26.0    38.3   0.0        0.0        0.1111E-02  0.2541E-02  0.1628E-02  1500.
          392.2   309.1   771.6   949.3                          0.4327E 70  0.4327E 70  0.4327E 70   0.06
                                                               -0.1085E 58 -0.1085E 58 -0.1085E 58
         1247.
 -106.     57.6    37.6    26.8    37.3   0.0        0.0        0.1346E-02  0.3049E-02  0.1945E-02  1500.
          399.7   308.9   776.6   937.2                          0.4327E 70  0.4327E 70  0.4327E 70   0.17
                                                               -0.1085E 58 -0.1085E 58 -0.1085E 58
         1523.
 -101.     61.9    37.5    27.9    35.6   0.0        0.0        0.1591E-02  0.3557E-02  0.2255E-02  1500.
          408.2   308.6   787.2   921.8                          0.4327E 70  0.4327E 70  0.4327E 70   0.32
                                                               -0.1085E 58 -0.1085E 58 -0.1085E 58
         1812.
  -96.     67.0    37.4    29.1    32.7   0.0        0.0        0.1845E-02  0.4065E-02  0.2549E-02  1500.
          417.6   308.4   802.6   900.5                          0.4327E 70  0.4327E 70  0.4327E 70   0.49
                                                               -0.1085E 58 -0.1085E 58 -0.1085E 58
         2115.
  -91.     73.1    37.3    30.4    30.9   0.0        0.0        0.2111E-02  0.4573E-02  0.2822E-02  1500.
          428.1   308.3   821.5   885.4                          0.4327E 70  0.4327E 70  0.4327E 70   0.67
                                                               -0.1085E 58 -0.1085E 58 -0.1085E 58
         2435.
  -86.     80.3    37.3    31.9    31.4   0.0        0.0        0.2387E-02  0.5081E-02  0.3088E-02  1500.
          439.9   308.2   842.8   886.5                          0.4327E 70  0.4327E 70  0.4327E 70   0.83
                                                               -0.1085E 58 -0.1085E 58 -0.1085E 58
         2773.
  -81.     89.1    37.2    33.4    32.2   0.0        0.0        0.2673E-02  0.5589E-02  0.3362E-02  1500.
          453.1   308.1   864.4   887.7                          0.4327E 70  0.4327E 70  0.4327E 70   0.98
                                                               -0.1085E 58 -0.1085E 58 -0.1085E 58
         3134.
  -76.     99.6    37.2    34.8    32.3   0.0        0.0        0.2969E-02  0.6097E-02  0.3641E-02  1500.
          467.7   308.1   885.1   883.9                          0.4327E 70  0.4327E 70  0.4327E 70   1.11
                                                               -0.1085E 58 -0.1085E 58 -0.1085E 58
         3521.
  -71.    112.6    37.2    36.2    32.1   0.0        0.0        0.3273E-02  0.6605E-02  0.3920E-02  1500.
          484.1   308.1   902.8   877.0                          0.4327E 70  0.4327E 70  0.4327E 70   1.22
                                                               -0.1085E 58 -0.1085E 58 -0.1085E 58
         3938.
  -66.    128.5    37.2    37.4    31.6   0.0        0.0        0.3585E-02  0.7114E-02  0.4197E-02  1500.
          502.4   308.2   916.9   868.5                          0.4327E 70  0.4327E 70  0.4327E 70   1.32
                                                               -0.1085E 58 -0.1085E 58 -0.1085E 58
         4391.
  -61.    148.3    37.3    38.6    30.9   0.0        0.0        0.3905E-02  0.7622E-02  0.4470E-02  1500.
          522.8   308.3   927.6   859.2                          0.4327E 70  0.4327E 70  0.4327E 70   1.40
                                                               -0.1085E 58 -0.1085E 58 -0.1085E 58
         4885.
  -56.    173.0    37.3    39.6    30.2   0.0        0.0        0.4233E-02  0.8130E-02  0.4738E-02  1500.
          545.5   308.4   935.2   849.5                          0.4327E 70  0.4327E 70  0.4327E 70   1.46
                                                               -0.1085E 58 -0.1085E 58 -0.1085E 58
         5430.
  -51.    204.1    37.3    40.4    29.4   0.0        0.0        0.4568E-02  0.8638E-02  0.5000E-02  1500.
          571.0   308.4   939.5   839.4                          0.4327E 70  0.4327E 70  0.4327E 70   1.51
                                                               -0.1085E 58 -0.1085E 58 -0.1085E 58
         6035.
  -46.    243.6    37.4    40.9    28.6   0.0        0.0        0.4907E-02  0.9146E-02  0.5256E-02  1500.
          599.1   308.6   940.7   828.9                          0.4327E 70  0.4327E 70  0.4327E 70   1.54
                                                               -0.1085E 58 -0.1085E 58
         6711.
  -41.    293.9    37.4    41.1    27.8   0.0        0.0        0.5250E-02  0.9654E-0
          630.2   308.7   939.1   817.6                          0.4327E 70  0.4327E 7
```

Fig.A7.1 (d) Typical results print-out from the GEC 'filling and emptying' program.

TABLE 1 - FIXED VOLUMES

IDENT	STT PRES	STT TEMP	FIN PRES	FIN TEMP	HT LOST	FL INJCD	FL MASS	MN PRES	MN TEMP
AIRCHEST	34.6	308.2	34.4	309.2	0.0	0.0	0.0	34.3	308.2
EXMAN-1	24.2	759.0	23.9	766.4	0.18	0.0	0.0	31.7	869.9
EXMAN-2	39.0	950.4	38.6	960.9	0.12	0.0	0.0	31.8	876.1

TABLE 2 - WORKING VOLUMES

IDENT	STT PRES	STT TEMP	FIN PRES	FIN TEMP	HT LOST	FL INJCD	FL MASS	MAX PRES	AT ANG	HM	TM
CYLINDER	39.8	360.4	39.5	361.6	0.62	4.5000	0.00044	1803.7	10.0	119.7	1045.7
CYLINDER	195.7	1338.7	194.9	1352.2	0.62	4.5000	0.00044	1796.8	10.0	119.2	1047.1
CYLINDER	29.6	492.6	29.3	493.5	0.62	4.5000	0.00044	1799.7	10.0	119.3	1047.0
CYLINDER	745.0	809.0	737.5	811.7	0.62	4.5000	0.00044	1802.8	10.0	119.1	1049.9
CYLINDER	30.3	330.1	30.1	331.4	0.62	4.5000	0.00044	1798.4	10.0	118.9	1051.0
CYLINDER	43.5	898.0	43.1	908.0	0.62	4.5000	0.00044	1794.8	10.0	118.7	1051.8

TABLE 3 - JUNCTIONS

IDENT	POS MASS FL	NEG MASS FL	NET MASS FL	MN PRS RAT	MN FL PARAM	MN EFFIC	EXIT TEMP
INVAL-1	0.124548E-01	0.110467E-03	0.123444E-01				
EXVAL-1	0.129689E-01	0.588296E-04	0.129101E-01				
INVAL-2	0.124867E-01	0.111004E-03	0.123757E-01				
EXVAL-2	0.130375E-01	0.589430E-04	0.129786E-01				
INVAL-3	0.125314E-01	0.111733E-03	0.124197E-01				
EXVAL-3	0.129269E-01	0.590436E-04	0.128678E-01				
INVAL-4	0.123942E-01	0.112185E-03	0.122820E-01				
EXVAL-4	0.129651E-01	0.977273E-04	0.128674E-01				
INVAL-5	0.123981E-01	0.113523E-03	0.122846E-01				
EXVAL-5	0.129023E-01	0.976760E-04	0.128046E-01				
INVAL-6	0.124334E-01	0.112773E-03	0.123206E-01				
EXVAL-6	0.129553E-01	0.977387E-04	0.128576E-01				
TURB-1	0.388590E-01	0.0	0.388590E-01	2.1544	0.451999E 00	0.0	718.43
COMP	0.731651E-01	0.0	0.731651E-01	2.4631	0.113168E 01	0.7400	420.47
TURB-2	0.386099E-01	0.0	0.386099E-01	2.1605	0.448581E 00	0.0	720.30

TABLE 4 - WORK

SECTION 1 - VOLUMES

VOL ID	IMEP	BMEP	ISFC	BSFC	TR A/F	DA A/F	BSAC	SH ID
CYLINDER	273.39	244.80	0.3317	0.3704	24.64	28.04	10.39	CRANK
CYLINDER	273.21	244.64	0.3319	0.3706	24.43	28.11	10.42	CRANK
CYLINDER	273.47	244.87	0.3316	0.3703	24.52	28.21	10.45	CRANK
CYLINDER	275.14	246.37	0.3296	0.3680	24.51	27.90	10.27	CRANK
CYLINDER	273.60	245.17	0.3312	0.3699	24.37	27.91	10.32	CRANK
CYLINDER	273.43	244.84	0.3316	0.3703	24.26	27.99	10.36	CRANK

SECTION 2 - JUNCTIONS

JUN ID	ENERG	SH ID
TURB-1	1.680	TURBO
COMP	2.096	TURBO
TURB-2	1.714	TURBO

TABLE 5 - SHAFTS

IDENT	MN SPD	MAX SPD	MIN SPD	FIN SPD	NET ENERG	MCH EFF
CRANK	1499.92	1500.00	1500.00	1500.00	10.093	0.895
TURBO	90740.63	91049.50	90468.13	91049.50	1.298	1.000

TABLE 6 - MEAN VALUES

SET	HEAT LOST	MAX PRESS	HM	TM	IMEP	BMEP	ISFC	BSFC	TR A/F	DA A/F	BSAC
1	0.62	1799.4	119.2	1048.8	273.74	245.12	0.3312	0.3699	24.46	28.03	10.37

TABLE 7 - LINER HTC

ANGLE	HTC	TEMP
5.	113.16	1043.00
10.	105.3[illegible]	1034.75

Fig A7.1 (e) Typical results print-out from the GEC 'filling and emptying' program.

15.	97.24	1012.56
20.	89.67	983.19
25.	82.82	949.63
30.	76.75	915.42
35.	71.51	885.88
40.	66.97	862.05
45.	62.99	842.89
50.	59.45	827.46
55.	56.25	814.98
60.	53.32	804.86
65.	50.62	796.61
70.	48.10	789.88
75.	45.72	784.38
80.	43.46	779.87
85.	41.29	776.14
90.	39.20	773.01
95.	37.18	770.48
100.	35.24	768.62
105.	33.37	767.65
110.	31.09	759.13
115.	28.86	750.78
120.	26.68	742.46
125.	24.54	734.09
130.	22.43	725.54
135.	20.36	716.67
140.	18.56	702.68
145.	16.84	686.06
150.	15.14	667.28
155.	13.45	645.37
160.	11.78	618.81
165.	10.11	585.02
170.	8.46	539.33
175.	6.80	472.54
180.	5.14	363.49

TABLE 8 - HEAT TRANSFER TO AREAS OF REF VOL

1	0.44
SL	0.18

Fig A7.1 (f) Typical results print-out from the GEC 'filling and emptying' program.

```
BCD 3THERMAL LPCS
BCD 6S LINER,SCALED FROM 6L
END
BCD 3NODE DATA
GEN 1,5,1,230.,1.,1.,1.,-1.
GEN 6,6,1,180.,1.,1.,1.,-1.
GEN 14,5,1,220.,1.,1.,1.,-1.
GEN 19,6,1,180.,1.,1.,1.,-1.
GEN 27,5,1,210.,1.,1.,1.,-1.
GEN 32,6,1,170.,1.,1.,1.,-1.
    40,200.,-1.
    41,200.,-1.
    42,200.,-1.
    -50,83.,0.
    -51,285.,0.
    -52,256.,0.
    -53,208.,0.
    -54,179.,0.
    -55,177.,0.
    -56,179.,0.
    -57,180.,0.
    -58,181.,0.
    -59,179.,0.
    -60,160.,0.
    -61,144.,0.
    -64,797.,0.
    -65,702.,0.
    -66,627.,0.
    -67,577.,0.
    -68,562.,0.
    -69,527.,0.
    -70,482.,0.
END

RELATIVE NODE NUMBERS                    ACTUAL NODE NUMBERS

   1  THRU   10          1     2     3     4     5     6     7     8     9    10
  11  THRU   20         11    14    15    16    17    18    19    20    21    22
  21  THRU   30         23    24    27    28    29    30    31    32    33    34
  31  THRU   40         35    36    37    40    41    42    50    51    52    53
  41  THRU   50         54    55    56    57    58    59    60    61    64    65
  51  THRU   55         66    67    68    69    70
NODE ANALYSIS...  DIFFUSION =     0,  ARITHMETIC =    36,  BOUNDARY =    19,  TOTAL =    55

BCD 3CONDUCTOR DATA
    1,1,2,3.355
    2,2,3,3.394
    3,3,4,1.312
GEN 4,7,1,4,1,5,1,1.4763,-1.,1.,-1.
    13,14,15,7.008
    14,15,16,7.088
    15,16,17,2.74
GEN 16,7,1,17,1,18,1,2.741,-1.,1.,-1.
    25,27,28,7.4046
    26,28,29,6.64
    27,29,30,1.429
GEN 28,7,1,30,1,31,1,1.607,-1.,1.,-1.
```

Fig. A7.2 (a) Typical input data print-out from the GEC 'MELTAN' program.

```
      388,41,42,16.2
      38,1,14,29.29
      39,2,15,58.24
      40,3,16,103.8
      41,4,17,141.45
GEN   42,7,1,5,1,18,1,133.13,-1.,1.,-1.
      51,14,27,30.995
      52,15,28,61.64
      53,16,29,109.9
      54,17,30,149.7
GEN   55,7,1,18,1,31,1,140.9,-1.,1.,-1.
      64,27,40,32.7
      65,28,41,61.02
REM WATERSIDE HTC=750 CHU/FT2HRK
      78,42,50,31.55
      70,29,50,27.15
      71,30,50,51.33
GEN   72,5,1,31,1,50,0,48.3,1.,1.,1.
REM GAS-TO-METAL HTC=117 TO 20 CHU/FT2HRK
      80,51,1,.0422
      81,52,2,.216
      82,53,3,4.835
      83,54,4,21.82
      84,55,5,14.18
      85,56,6,12.15
      86,57,7,12.26
      87,58,8,15.2
      88,59,9,20.06
      89,60,10,7.03
      90,61,11,.351
REM LINER/PISTON
      93,64,1,1.254
      94,65,2,1.943
      95,66,3,2.650
      96,67,4,2.839
      97,68,5,2.19
      98,69,6,1.67
      99,70,7,1.15
      100,29,42,32.66
END

RELATIVE CONDUCTOR NUMBERS                              ACTUAL CONDUCTOR NUMBERS

   1  THRU   10           1      2      3      4      5      6      7      8      9     10
  11  THRU   20          13     14     15     16     17     18     19     20     21     22
  21  THRU   30          25     26     27     28     29     30     31     32     33     34
  31  THRU   40          37    388     38     39     40     41     42     43     44     45
  41  THRU   50          46     47     48     51     52     53     54     55     56     57
  51  THRU   60          58     59     60     61     64     65     78     70     71     72
  61  THRU   70          73     74     75     76     80     81     82     83     84     85
  71  THRU   80          86     87     88     89     90     93     94     95     96     97
  81  THRU   83          98     99    100
CONDUCTOR ANALYSIS...  LINEAR =    83,  RADIATION =     0,  TOTAL =    83,  CONNECTIONS =    83

BCD 3CONSTANTS DATA
      ARLXCA,.01
      NLOOP,500
END
CONSTANTS ANALYSIS...  USER =     0,  ADDED =     0        0        0,  TOTAL =

BCD 3ARRAY DATA
```

Fig. A7.2 (b) Typical input data print-out from the GEC 'MELTAN' program.

```
      END
      ARRAY ANALYSIS...  NUMBER OF ARRAYS =     0  TOTAL LENGTH =     0

      BCD 3EXECUTION
          CORE3K
          ARYMPY(56,G1,1.25,G1)
          ARYMPY(27,G78,1.5625,G78)
          CENTFL
          CSGDMP
          PUNCHT
          QPRINT
      END
      BCD 3VARIABLES 1
      END
      BCD 3VARIABLES 2
      END
      BCD 3OUTPUT CALLS
          TSEG
      END
 END OF PREPROCESSING
```

Fig. A7.2 (c) Typical input data print-out from the GEC 'MELTAN' program.

```
                      GEC MECHANICAL ENGINEERING LABORATORY -  THERMAL ANALOGUE NETWORK
                      PROGRAM MELTAN-S                   VERSION JL00A                 OCT79                    PAGE   1
   S LINER,SCALED FROM 6L

 *   *   *   *
 TIME  0.0           DTIMEU  0.0           CSGMIN(    0)   0.0           DTMPCC(    0)   0.0           ARLXCC(    0)   0.0

  T     1= 2.300E+02   T     2= 2.300E+02   T     3= 2.300E+02   T     4= 2.300E+02   T     5= 2.300E+02   T     6= 1.800E+02
  T     7= 1.800E+02   T     8= 1.800E+02   T     9= 1.800E+02   T    10= 1.800E+02   T    11= 1.800E+02   T    14= 2.200E+02
  T    15= 2.200E+02   T    16= 2.200E+02   T    17= 2.200E+02   T    18= 2.200E+02   T    19= 1.800E+02   T    20= 1.800E+02
  T    21= 1.800E+02   T    22= 1.800E+02   T    23= 1.800E+02   T    24= 1.800E+02   T    27= 2.100E+02   T    28= 2.100E+02
  T    29= 2.100E+02   T    30= 2.100E+02   T    31= 2.100E+02   T    32= 1.700E+02   T    33= 1.700E+02   T    34= 1.700E+02
  T    35= 1.700E+02   T    36= 1.700E+02   T    37= 1.700E+02   T    40= 2.000E+02   T    41= 2.000E+02   T    42= 2.000E+02
  T    50: 8.300E+01   T    51: 2.850E+02   T    52: 2.560E+02   T    53: 2.080E+02   T    54: 1.790E+02   T    55: 1.770E+02
  T    56: 1.790E+02   T    57: 1.800E+02   T    58: 1.810E+02   T    59: 1.790E+02   T    60: 1.600E+02   T    61: 1.440E+02
  T    64: 7.970E+02   T    65: 7.020E+02   T    66: 6.270E+02   T    67: 5.770E+02   T    68: 5.620E+02   T    69: 5.270E+02
  T    70: 4.820E+02

 *   *   *   *
 TIME  0.0           DTIMEU  0.0           CSGMIN(    0)   0.0           DTMPCC(    0)   0.0           ARLXCC(   11)   8.80432E-03

  T     1= 2.728E+02   T     2= 2.327E+02   T     3= 1.766E+02   T     4= 1.512E+02   T     5= 1.399E+02   T     6= 1.315E+02
  T     7= 1.253E+02   T     8= 1.202E+02   T     9= 1.266E+02   T    10= 1.432E+02   T    11= 1.429E+02   T    14= 2.493E+02
  T    15= 2.140E+02   T    16= 1.589E+02   T    17= 1.351E+02   T    18= 1.262E+02   T    19= 1.198E+02   T    20= 1.151E+02
  T    21= 1.114E+02   T    22= 1.166E+02   T    23= 1.423E+02   T    24= 1.429E+02   T    27= 2.350E+02   T    28= 1.987E+02
  T    29= 1.392E+02   T    30= 1.195E+02   T    31= 1.133E+02   T    32= 1.088E+02   T    33= 1.055E+02   T    34= 1.030E+02
  T    35= 1.068E+02   T    36= 1.419E+02   T    37= 1.429E+02   T    40= 2.298E+02   T    41= 1.852E+02   T    42= 1.240E+02
  T    50: 8.300E+01   T    51: 2.850E+02   T    52: 2.560E+02   T    53: 2.080E+02   T    54: 1.790E+02   T    55: 1.770E+02
  T    56: 1.790E+02   T    57: 1.800E+02   T    58: 1.810E+02   T    59: 1.790E+02   T    60: 1.600E+02   T    61: 1.440E+02
  T    64: 7.970E+02   T    65: 7.020E+02   T    66: 6.270E+02   T    67: 5.770E+02   T    68: 5.620E+02   T    69: 5.270E+02
  T    70: 4.820E+02

 LOOPCT =      80 ENGBAL =  0.42175E+01

 *   *   *   *
 TIME  0.0         - DTIMEU  0.0           CSGMIN(    0)   0.0           DTMPCC(    0)   0.0           ARLXCC(   11)   8.80432E-03

  T     1= 2.728E+02   T     2= 2.327E+02   T     3= 1.766E+02   T     4= 1.512E+02   T     5= 1.399E+02   T     6= 1.315E+02
  T     7= 1.253E+02   T     8= 1.202E+02   T     9= 1.266E+02   T    10= 1.432E+02   T    11= 1.429E+02   T    14= 2.493E+02
  T    15= 2.140E+02   T    16= 1.589E+02   T    17= 1.351E+02   T    18= 1.262E+02   T    19= 1.198E+02   T    20= 1.151E+02
  T    21= 1.114E+02   T    22= 1.166E+02   T    23= 1.423E+02   T    24= 1.429E+02   T    27= 2.350E+02   T    28= 1.987E+02
  T    29= 1.392E+02   T    30= 1.195E+02   T    31= 1.133E+02   T    32= 1.088E+02   T    33= 1.055E+02   T    34= 1.030E+02
  T    35= 1.068E+02   T    36= 1.419E+02   T    37= 1.429E+02   T    40= 2.298E+02   T    41= 1.852E+02   T    42= 1.240E+02
  T    50: 8.300E+01   T    51: 2.850E+02   T    52: 2.560E+02   T    53: 2.080E+02   T    54: 1.790E+02   T    55: 1.770E+02
  T    56: 1.790E+02   T    57: 1.800E+02   T    58: 1.810E+02   T    59: 1.790E+02   T    60: 1.600E+02   T    61: 1.440E+02
  T    64: 7.970E+02   T    65: 7.020E+02   T    66: 6.270E+02   T    67: 5.770E+02   T    68: 5.620E+02   T    69: 5.270E+02
  T    70: 4.820E+02
 ARITHMETIC NODE PSEUDO-COMPUTE SEQUENCE FOLLOWS
 NODE     1 TO ARTH NODE     2 THRU LIN COND     1,COND VALUE IS 4.19375E+00
```

Fig. A7.2 (d) Typical results print-out from the GEC 'MELTAN' program.

APPENDIX 8

Detail of the Marketing Questionnaire Construction

A major disadvantage of the postal questionnaire is that the response rate is normally very low, (less than 35%). Assuming the population to be sampled is large, the number of questionnaires may be chosen to allow for this low rate of response, and give a satisfactory result. For this study the population was limited to British electrical generating set manufacturers, who number between 35 and 50. Therefore a high response rate was essential to achieve a meaningful result. The questionnaire was designed with that aim.

1. Questions were limited to two sides of A4 paper. Wilson suggests that each question reduces response by about 5%, ref(99).
2. Questions did not require detailed answers; reference to records was unnecessary.
3. Each respondent was 'warmed up' with a telephone call and asked to participate.
4. The questionnaire was type-set and kept clear and simple, to give a high level of presentation.
5. Personal details were asked at the end, since this often puts respondents on their guard.
6. A copy of the final marketing report was offered.

7. Confidentiality was guaranteed.

The questionnaire was followed-up with a telephone reminder if a reply was not received in 2 to 3 weeks. This usually resulted in people claiming not to have recieved one, in such cases a second copy was sent.

A second disadvantage of the postal questionnaire is reaching the desired respondents in the organisation. Addressing a letter to the sales or technical department may only result in a rapid reply from a junior clerk. To a large extent this was overcome by the telephone 'warm-up', which usually allowed the Technical or Marketing Manager to be identified by name.

The last question, rating the parameters for selection of a mains-failure engine, uses a Likert, or semantic differential scale (98). This allows a large amount of data to be collected with minimal effort on the part of the respondent.

Although the response rate was 46 %, this represented only 13 replies. Thus, a statistical analysis of the results would have been inappropriate.

APPENDIX 9

Dorman 6LE engine basic data

Number of cylinders	six
Firing order	1 5 3 6 2 4
Bore	127 mm
Stroke	149.225 mm
Operating cycle	4 stroke
Maximum rated speed	2100 rev/min
Number of valves	two per cylinder
Valvegear	push-rod operated by a 'low' camshaft.
Valve timing (cold)	avo 347 deg atdc avc 581 " evo 134 " evc 368 "
Cooling system	closed 'jacket' system with circulating pump.

Component	Leading details	Material
Crankcase		Cast iron BS1452-grade12
Crankshaft	Main journal dia. 95.2mm Crankpin dia. 82.5mm	Alloy steel BS970-790M40S (EN19S)
Cylinder liner	.05mm chromium plated bore	Centrifugally cast iron to BS1452-grade14
Cylinder head	one head per 3 cylinders	Cast iron to BS1452-grade17
Piston	3 compression and 2 oil control rings.	Aluminium alloy (52-alloy)
Exhaust valve	seat angle 45 head dia.47.8mm lift 12.1mm	BS970-349S52 (21-4N)
Inlet valve	seat angle 45 head dia.55.7mm lift 12.1mm	BS970-817M40 (EN24)
Connecting rod	Centres 267 mm	BS970-605M36T (EN16T)

APPENDIX 10

Estimation of engine pumping and frictional losses

Figure A10.1 shows a Willans line plotted from experimental results from the research engine at 1500rev/min and a compression ratio of 10.7. This indicates a motoring loss mep of 1.6bar ($23.2\,lbf/in^2$).

This may be compared with the simple model proposed by Millington and Hartles(105), shown below.

$$F = A + 7.0 \frac{N}{1000} + 1.5 \left(\frac{v}{1000}\right)^2$$

where:

F motoring loss $[lbf/in^2]$

A compression ratio - 4

N engine speed $[rev/min]$

v mean piston speed $[ft/min]$

$$F = 10.7-4 + 7.0 \frac{1500}{1000} + 1.5 \left(\frac{1469}{1000}\right)^2$$

$$F = \underline{20.44\ lbf/in^2}$$

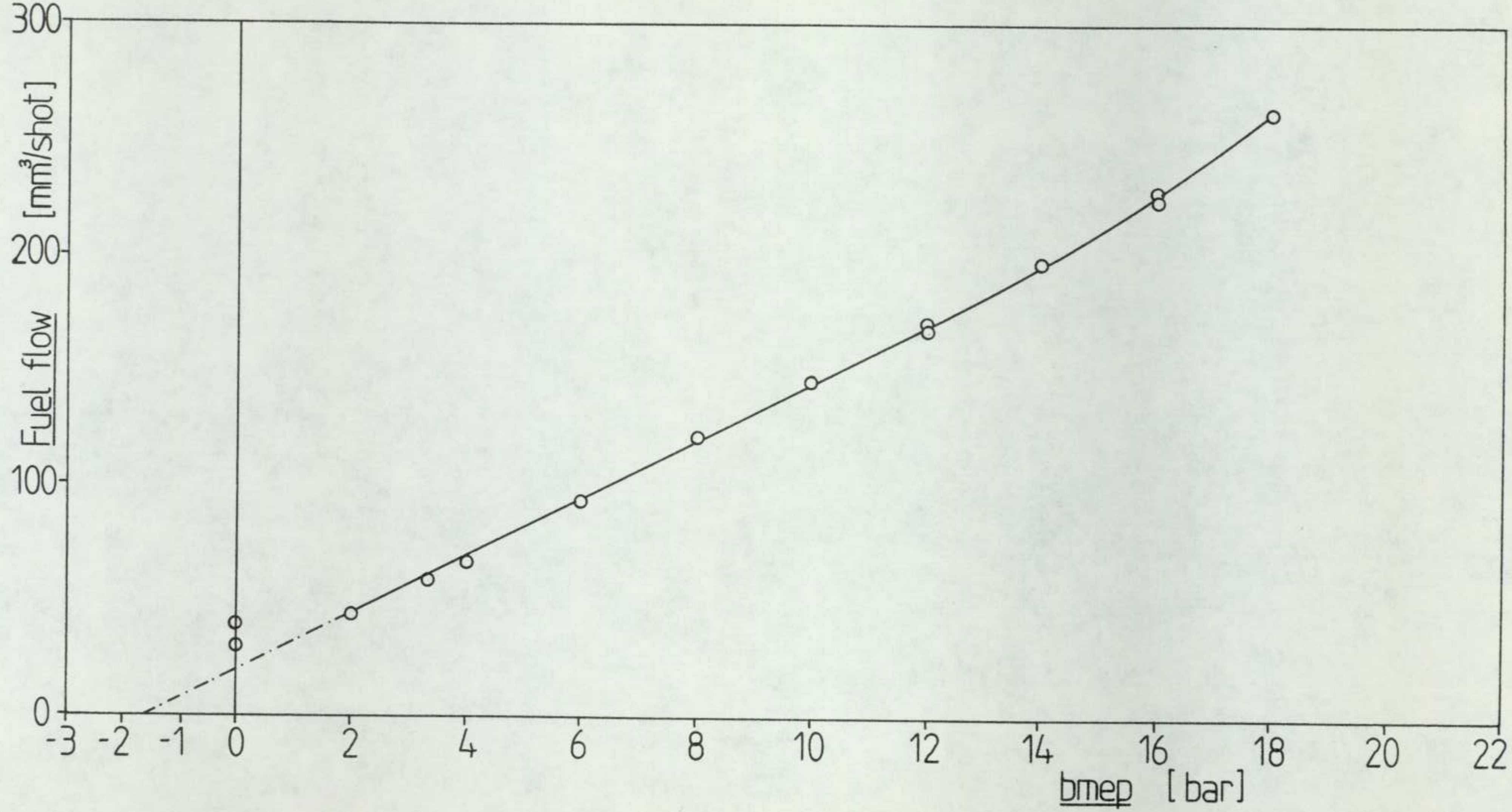


Fig.A10·1, Willans line estimation of pumping and frictional losses. [build 8, spill 22°]

Appendix 11

Fuel Specification

The fuel used throughout this work conformed to the specification of Class A1 light distillate diesel fuel, given in BS 2869:1970. The following table is reproduced from that source.

Viscosity, kinematic at 37.8°C (100°F), centistokes	min.	1.6
	max.	6.0
Cetane Number	min.	50
Carbon residue, Conradson, % by mass	max.	-
Distillation, recovery at 357°C or 675°F, % by volume	min.	90
Flash point, closed, Pensky-Martens,	min.	55°C or 130°F
Water content, % by volume	max.	0.05
Sediment, % by mass	max.	0.01
Ash, % by mass	max.	0.01
Sulphur content, % by mass	max.	0.5
Copper corrosion test,		1
Cloud point		
	summer	0°C (32°F) March/November inclusive
	winter	-7°C (20°F) Dec./February inclusive
Pour point	max.	-

APPENDIX 12

Lubricating Oil Specification

The lubricating oil used throughout this work complied with the American standard for diesel engine crankcase oil - MIL-L-2104C The results of a quality control test by the manufacturer are given below.

SAE Number	30
Specific Gravity, 60/60°F	0.896
Flash Point, °F(°C)	425(218)
Pour Point, °F	-5(-21)
Viscosity	
cP @ 0°F(-18°C)	-
SSU @100°F	575
SSU @210°F	68
cSt @ 38°C	124
cSt @ 99°C	12.4
VI_E (ASTM D 2270)	99
Colour, ASTM	8
Sulphated Ash, %wt	0.98

REFERENCES

1. NITSKE, W.R., and WILSON, C.N., "Rudolf Diesel", University of Oklahoma Press, 1965.

2. JANOTA, M.S., Opening Address "Turbochargers and Turbocharging", I.Mech.E. Conference publication, 1978 - 2.

3. ZINNER, K., "Supercharging of Internal Combustion Engines", Springer-Verlag, 1978.

4. ROGERS, G.F.C, and MAYHEW, Y.R., "Engineering Thermodynamics, Work and Heat Transfer", Longman, 1972.

5. OWEN, G., "Diesel Engines", Financial Times Survey, 8.1.79.

6. GRUNDY, J.R. at al., "AVCR 1360-2 High Specific Output - Variable Compression Ratio Diesel Engine", Report No. 760051, Society of Automotive Engineers (S.A.E.), 1976.

7. MANSFIELD, W.P.,"Transport Engines of Exceptionally High Specific Output", Proc. I.Mech.E., 1968-69, Vol. 183, Pt. 3B.

8. MARTIN, R. and WRIGHT, G., "High Output Diesel Engine Design Philosophy", Report No. 770755, S.A.E., 1977.

9. KAMO, R.,"Higher BMEP Prospects for Vehicular Diesels", I.Mech.E. 1978, C62/78.

10. MOON, J., "Volvo Engine Design Projects include Low Compression Diesels", Diesel and Gas Turbine Progress, Sept.-Oct. 1970.

11. WALLAGE, P., "The Perils of being a Pioneer", Motor Transport, July 7, 1978.

12. THOLEN, P., and KILLMAN, I., "Investigations on highly turbocharged air cooled diesel engines", CIMAC Conference, Tokyo, 1977, Paper All.

13. WATSON, N. et al., "An evaluation of two-stage turbocharging for efficient, high-output diesel engines", A.S.M.E. Conference Paper, November 1978.

14. WATSON, N. et al., "Turbocharger System Option for Vehicle Engines", I.Mech.E.,1978, C61/78.

15. BROCK, E.K., "Turbocharging of four-stroke diesel engines related to power generation", Diesel Engineers & Users Association, publication 379, May 1977.

16. WALLACE, F.J. et al., "Matching of high output turbocharged engines for maximum torque back-up and emission reduction based on the use of variable geometry compressors and turbines", Report No. 740738, S.A.E., 1974.

17. WALLACE, F.J. et al., "A critical comparison of two stage turbocharged, Hyperbar, and differential compound engines", Report No. 770756, S.A.E., 1977.

18. WALLACE, F.J.,et al., "Variable Geometry vs. Two stage Turbocharging of high output diesel engines", I.Mech.E., 1978, C63/78.

19. NANCARROW, J.H., "Influence of turbocharger characteristics on the supply of air for high speed diesel engines", Report No. 660133, S.A.E., 1966.

20. LANG, W., "What can the turbocharger do for the engine ?", Report No. 660473, S.A.E., 1966.

21. "5 to 1 ratio turbocharger", GAS AND OIL POWER, Spring 1974.

22. MANSFIELD, W.P. and MAY, W.S., "Diesel combustion at high MEP with low compression ratio", Report No. 660343, S.A.E., 1966.

23. WATSON, N. and HOLNESS, B.S., "Engine and turbocharger interaction with multi-entry pulse converters", I.Mech.E., 1978, C65/78.

24. BENSON, R.S. and WHITEHOUSE, N.D., "Internal Combustion Engines", Pergamon Press, 1979 (Text).

25. SHIPINSKI, J. et al., "A spray droplet model for diesel combustion", I.Mech.E., Vol. 184, Pt. 3J, 1969/70.

26. HENEIN, N.A. and BOLT, J.A., "Ignition Delay in Diesel Engines", Report No. 670007, S.A.E., 1967.

27. HENEIN, N.A. and BOLT, J.A.,"Correlation of air charge temperature and ignition delay for several fuels in a diesel engine", Report No. 690252, S.A.E., 1969.

28. WOLFER, H.H., "Ignition lag in the diesel engine"; VDI - Forschungsheft No. 392, 1938, Trans. by R.A.E., Farnborough Library, No. 358, August 1959, U.D.C. No. 621 - 436.047.

29. AUSTEN, A.E.W. and LYN, W-T., "Relation between fuel injection and heat release in a direct injection engine, and the nature of the combustion process", Proc. I.Mech.E., No.1, 1960-61.

30. HAWKESLEY, G.J. and ANDERTON, D., "Studies into combustion and noise in turbocharged engines", I.Mech.E., 1978, C72/78.

31. KHAN, I.M., "Formation and Combustion of Carbon in a Diesel Engine", Proc. I.Mech.E., Vol. 184, Pt. 3J, 1969-70.

32. BLACKWOOD, A.K. and PERSON, F.W., "Designing Liquid Cooling Systems for Engines", Publication SP152, S.A.E., 1958.

33. HARTLEY, J., "Engine Design Series - Cooling Systems", Automotive Engineer, April/May 1978.

34. WU, H.L., "Prediction of wall temperatures inside diesel engines by digital computer", The Engineer, November 25, 1966.

35. "Cylinder Components", DIESEL ENGINEERING, Autumn, 1978.

36. MUNRO, R., "Open gallery oil-cooled piston design", Wellworthy Topics, (date unknown).

37. GAZZARD, S.T., "Piston Crown Cracking - Finding a New Solution", Diesel and Gas Turbine Progress, (date unknown).

38. PACHERNEGG, S.J., "Heat Flow in Engine Pistons", Report No. 670928, S.A.E., 1967.

39. LAW, D.A. and DAY, R.A., "Oil-cooled Aluminium Alloy Diesel Engine Pistons - A New Approach", Report No. 690749, S.A.E., 1969.

40. STOTTER, A., "Heat Transfer in Piston Cooling", Report No. 660757, S.A.E., 1966.

41. FRENCH, C.C.J., "Piston Cooling", Report No. 720024, S.A.E., 1972.

42. WOOLLEY, K.S., "Boundary Heat Transfer in Piston Cavities", GEC - M.E.L. Report No. W/M (3B), p.1624, 1968.

43. WOOLLEY, K.S., "The calculation of piston and cylinder liner temperatures", GEC - M.E.L. Report No. W/M (3B), p.1630, 1968.

44. WOOLLEY, K.S., "Piston Temperatures - Theoretical and Experimental Investigations", GEC - M.E.L. Report No. W/M (3B), p.1295, 1968.

45. ALCOCK, J.F. et al., "Distribution of heat flow in high duty internal combustion engines", CIMAC Conference, 1957.

46. AUSTEN, A.E.W., and LYN, W.T., "The application of heat release analysis to engine combustion study", CIMAC Conference, 1962.

47. HOWARTH, M.H., "Design of High Speed Diesel Engines", Constable and Co. Ltd., 1966 (Text).

48. EICHELBERG, G., "Some new investigations on old combustion engine problems", Engineering, Oct-Dec 1939, (five parts).

49. WOSCHNI, G., "A universally applicable equation for the instantaneous heat transfer coefficient in the internal combustion engine", Report No. 670931, S.A.E., 1967.

50. MUNRO, R., and GRIFFITHS, W.J., "The application of predictive techniques in the design and development of medium speed diesel engine pistons", 13th. CIMAC Conference, Vienna, May 1979.

51. MELCHIOR, J. and ANDRE-TALAMON, T., "The Hyperbar System of High Supercharge", Report No. 740723, S.A.E., 1974.

52. "Hyperbar Turbocharged Engines", GAS AND OIL POWER, Spring 1975.

53. "Hyperbar Supercharger Assessed in an SNCF Railcar", RAILWAY GAZETTE INTERNATIONAL, February 1979.

54. "Forty Cylinder Hyperbar-Charged Engine", DIESEL ENGINEERING, Spring 1979.

55. Hyperbar Technical sales literature, Societe Suralmo, Neuilly sur-Seine, France, 1979.

56. LYN, W.T. and VALDMANIS, E., "The Effects of Physical Factors on Ignition Delay", Proc. I.Mech.E., Automotive Division, 1966.

57. "Intercooling for small turbocharged diesel engines", GAS AND OIL POWER, Summer 1972.

58. HARDENBURG, H. and FRAENKLE, G., "The effect of charge air cooling on exhaust emissions and power output of turbocharged diesel engines", I.Mech.E., 1978, C71/78.

59. KNIGHT, B.E., "Fuel Injection Systems Calculations", Proc. I.Mech.E., Automobile Division, 1960-61.

60. BURMAN, P.G. and DE LUCA, F., "Fuel injection and controls for internal combustion engines", The Technical Press, 1962, (Text).

61. WHITEHOUSE, N.D. and WAY, R.J.B., "Rate of heat release in diesel engines and its correlation with fuel injection data", Proc. I.Mech.E., 1969-70, Vol. 184, pt. 3J.

62. GLANZMANN, W., "Calculation of diesel cycles with reduced compression ratio", Brown Boveri Review, 4-77, 1977.

63. METZ, J.R. et al., "Contribution to the calculation of the diesel cycle with the aid of digital computers", Proc. I.Mech.E., 1967-68, Vol. 182, pt. 3L.

64. WOSCHNI, G. and ANISITS, F., "Experimental investigation and mathematical presentation of rate of heat release in diesel engines dependent upon engine operating conditions", Report No. 740086, S.A.E., 1974.

65. WOSCHNI, G., "Computer programs to determine the relationship between pressure flow, heat release and thermal load in diesel engines", Report No. 650450, S.A.E., 1965.

66. HALLAM, A.J., and COTTAM, S., "Computer program to predict the gas exchange process of a diesel engine", Computer Aided Design, Vol. 7, No. 2, April 1975.

67. WOSCHNI, G., "Electronic calculation of the time curve of pressure temperature and mass flow rate in the cylinder of a diesel engine", Proc. I.Mech.E., Vol. 182, Pt. 3L, 1967-68.

68. WHITEHOUSE, N.D. and ABUGHRES, S.M., "Calculation of fuel-air mixing in a diesel engine with swirl for the purpose of heat release prediction", I.Mech.E., 1975, C97/75.

69. SHAHED, S.M., CHIU, W.S. and LYN, W.T., "A Mathematical Model of Diesel Combustion", I.Mech.E., 1975, C94/75.

70. HODGETTS, D. and SHROFF, H.D., "More on the formation of nitric oxide in a diesel engine", I.Mech.E., 1975, C95/75.

71. LYN, W.T., "Calculations of the effect of rate of heat release on the shape of cylinder pressure diagram, and cycle efficiency", Proc. I.Mech.E., Automobile Division, 1960-61.

72. MARZOUK, M. and WATSON, N., "A non-linear digital simulation of turbocharged diesel engines under transient conditions", Report No. 770123, S.A.E., 1977.

73. WATSON, N. and MAGUERDICHIAN, M.,"Prediction of mixture formation and heat release in diesel engines", Report No. 780225, S.A.E. 1978.

74. Course Notes - "Turbocharging the internal combustion engine", Sept. 1979, Imperial College, London.

75. JANOTA, M.S., HALLAM, A.J., BROCK, E.K. and DEXTER, S.G.,"The prediction of diesel engine performance and combustion chamber component temperatures using digital computers", Proc. I.Mech.E., Vol. 182, Pt. 2L, 1967-68.

76. MURAYAMA, et al., "The relation between nitric oxide formation and combustion process in diesel engines", 12th CIMAC Conference, Tokyo, Paper B12, 1977.

77. DENT, J.C. and DERHAM, J.A., "Air motion in a four-stroke, direct injection diesel engine", Proc. I.Mech.E., Vol. 188, 21/74, 1974.

78. RICARDO and Co.(1927) Ltd., "Analysis of steady state flow rig results",Report no.DP20534,12 November 1975.

79. BORMAN, G.L., and WEBER, H.G., "Parametric studies using a mathematically simulated diesel engine cycle", Report No. 670480, S.A.E., 1967.

80. RYTI, M., "Computing the gas exchange process of pressure charged internal combustion engines", Proc. I.Mech.E., Vol. 182, Pt. 3L, 1967-68.

81. WOODS, W.A. and KHAN, S.R., "An experimental study of flow through poppet valves", Proc. I.Mech.E., Vol. 180, Pt. 3N, 1965.

82. MARZOUK, M., "Simulation of turbocharged diesel engines under transient conditions", Ph.D. thesis, Imperial College, University of London, 1976.

83. KAMO, R. and BRYZIK, W., "Adiabatic Turbocompound Engine Performance Prediction", Report No. 780068, S.A.E., 1978.

84. HALLAM, A.J., "GEC Performance Prediction Suite - Users' Manual", GEC Diesels, Report No.80-RN-1, January 1980.

85. HIGGINS, P.A., "Heat release in a four-stroke diesel engine under pressure charged conditions", M.Sc. thesis, University of Manchester Institute of Science and Technology, 1968.

86. "MELTAN - a general computer program for non-linear field problems", GEC Journal of Science and Technology, Vol.44, No. 2., 1976-77.

87. RICARDO, H.R. and HEMPSON, J.G.G., "The High-Speed Internal Combustion Engine", Blackie, 1968, (Text).

88. BELCHER, P.R., "Templugs", The Engineer, February 1966.

89. HOHENBURG, G.F., "Advanced approaches for heat transfer calculations", Report No. 790825, S.A.E. Publication SP-79, 1979.

90. JONES, P.E., "Induction system development for high performance, direct injection diesel engines", Proc. I.Mech.E., Vol. 180, Pt. 3N,

91. DICKSEE, C.B., "High-Speed Compression Ignition Engine", Blackie, 1944, (Text).

92. "Industrial Innovation", Advisory Council for Applied Research and Development,HMSO, December 1978.

93. ROTHWELL, R., "The characteristics of successful innovators and technically progressive firms", R & D Management, 7, 3, 1977.

94. BLOIS, K.J. and COWELL, D.W., "Marketing research for new product ideas arising from R & D departments", R & D Management, 9, 2, 1979.

95. HARGREAVES, G.,et al., "New product evaluation : electric vehicles for commercial applications", Journal of Marketing, January 1976.

96. TWISS, B.C., "Managing Technological Innovation", Longman, 1974, (Text).

97. KOTLER, P., "Marketing Management - Analysis and Measurement", McGraw-Hill, 1973, (Text).

99. WILSON, A., "The Assessment of Industrial Markets", Hutchinson Marketing Library, 1968.

100. COOPER, R.G., "Why New Industrial Products Fail", Marketing Management and Administrative Action, Britt and Boyd (editors), 4th. Edition, McGraw-Hill, 1978.

101. "The reliability of diesel engines and its impact on cost", I.Mech.E., Conference Publication, 7-8 November 1972.

102. DEXTER, S.G., "Diesel Gen-set Exhaust Emissions", International Power Generation, April/May 1978.

103. CHARLTON, S.J., "Survey of emissions of engines currently under development", Dorman Report No. 16700/2.0, September 1975.

104. CHAPMAN, A.J., "Heat Transfer", 2nd. Edition, McMillan, 1960.

105. MILLINGTON, B.W. and HARTLES, E.R., "Frictional losses in diesel engines", Report No. 680590, S.A.E., 1968.

106. MAYHEW, Y.R. and ROGERS, G.F.C., "Thermodynamic and transport properties of fluids", 3rd. Edition, Basil Blackwell, 1980.

107. EPHRAIM, M., "A personal philosophy on diesel engine design", Railway Gazette International, February 1979.

108. "HIGH-SPEED DIESELS : New units pack more power into less space", Marine Engineering /Log, April 1978.

109. WINTERBONE, D.E. et al., "Transient Response of Turbocharged Diesel Engines", Report No. 770122, S.A.E., 1977.

110. GARRETT CORPORATION - Hydraulically driven slave compressor for transient response. UK Patent 1 509 744