

INVESTIGATION OF A DIESEL GAS TURBINE

by

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SUMMARY

The turbocharging of automotive sized two-stroke engines presents a problem to the designer due to the large range of loads and speeds met in service. A mechanically driven blower is usually required to provide a positive pressure drop across the engine at all times, thus providing adequate air for scavenging.

The aim of this investigation is to consider a combination of the two-stroke cycle engine, turbocharger and auxiliary combustion chamber as a prime mover system with a satisfactory torque characteristic and control system. The auxiliary combustion chamber is used for independent control of the turbocharger.

To this end various component combinations are described and assessed. Turbochargers of different builds and frame sizes have been examined to assess the effects of turbocharger matching. Computerised processing for data acquisition and control has been used.

Theoretical work, progressing simultaneously with the experimental programme, has permitted the development of simple computer programs to predict system performance to a good degree of accuracy. The programs are detailed in the appendix.

Experimental results have been presented graphically and compared with those predicted by the computer programs. The work has been extended, by computer techniques, to investigate the effects of charge cooling, heat exchange, and torque 'tailoring' by the control of engine boost levels.

The Diesel Gas Turbine is shown to provide a feasible prime mover system. Significant improvements of power output and torque characteristic have been achieved within constraints of thermal and mechanical loading. The system incurs penalties in specific fuel

consumption at low speeds but these are offset, to some extent, by improvements at high speed. The improved torque characteristics would permit the use of a simpler transmission system, together with the removal of the scavenge blower.

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NOTATION

Abbreviations

ACC	Auxiliary Combustion Chamber
ADC	Analogue to Digital Convertor
DAC	Digital to Analogue Convertor
DAS	Data Acquisition System
DCE	Differential Compound Engine
DDE	Differential Diesel Engine
DFE	Diesel Fuel Equivalent
DGT	Diesel Gas Turbine
DMM	Digital Multi Meter
DVM	Digital Volt Meter
FM	Frequency Modulated
TTL	Transistor - Transistor Logic
VCR	Variable Compression Ratio

Standard Engine

C_{p1}	Mean specific heat over range ΔT
C_v	Calorific value of diesel fuel
D	Scavenge blower speed
H_c	Heat loss to coolant
H_{exh}	Enthalpy of exhaust gas
HP	Horse power of engine

$(HP)_f$	Frictional horse power
$(HP)_i$	Indicated horse power
$(HP)_{sb}$	Scavenge blower horse power
K_1, K_2	Constants
N	Engine speed
n_c	Index of compression
n_e	Index of expansion
P_a	Atmospheric pressure
P_c	Charge inlet pressure
P_{max}	Peak cylinder pressure
R	Gas constant (air)
R_T	Trapped air/fuel ratio
r	Compression ratio
r_c	Boost ratio
r_e	Expansion ratio
sfc	Specific fuel consumption
T_m	Air manifold temperature
$(T_{mean})_c$	Mean temperature of compression
$(T_{mean})_e$	Mean temperature of expansion
V_s	Swept volume
V_T	Volume of trapped air
W	Engine mass flow
W_f	Fuel mass flow
W_T	Mass of trapped air
α	Pressure ratio (dual cycle combustion)
β	Cut-off ratio (dual cycle combustion)
γ	Ratio of specific heats

ΔT	Temperature rise across the engine
η_b	Brake thermal efficiency
η_d	Diagram efficiency
λ	Scavenge ratio
ρ_m	Density of air in manifold
σ	Effective stroke ratio
ϕ	Cylinder filling efficiency

Preliminary Matching

AM	Total turbine mass flow
EM	Engine fuel requirement
ES	Engine speed
e	Charge cooler effectiveness
F	Fuel/air ratio
FM	Mass of auxiliary fuel
h_1	Enthalpy at compressor inlet
h_2	Enthalpy at compressor outlet
h_3	Enthalpy at entry to air manifold
h_5	Enthalpy at turbine inlet
h_6	Enthalpy at turbine exhaust
J	Mechanical equivalent of heat
l	Engine connecting rod length
M	Trapped air mass
n	Number of cylinders
P_a	Atmospheric pressure
P_2	Compressor outlet pressure
P_3	Air manifold pressure

Q	Heat addition
R	Gas constant (air)
R_T	Air/fuel ratio
r_c	Boost ratio
s	Engine stroke
SR	Scavenge ratio
T_g	Exhaust blowdown temperature
T_3	Cylinder inlet temperature
\bar{U}_g	Internal energy of exhaust gases at blowdown
V	Cylinder volume
V_{ec}	Cylinder volume at exhaust port closing
V_{eo}	Cylinder volume at exhaust port opening
V_{ic}	Cylinder volume at inlet port closing
W_c	Compressor work
W_t	Turbine work
X	Fraction of theoretical air
η_c	Compressor efficiency

Diesel Gas Turbine

C_{PA}	Specific heat of inlet air
C_{Pcin}	Specific heat at auxiliary combustion chamber inlet
C_{Peng}	Specific heat of engine exhaust
C_{Ptin}	Specific heat at turbine inlet
C_{Pto}	Specific heat of turbine exhaust
C_{Vp}	Calorific value of propane
ETT	Turbine work factor

e_{ht}	Heat exchanger effectiveness
Fact	Pulse work factor
HPc	Compressor power
H_{cin}	Enthalpy at inlet to auxiliary combustion chamber
H_{eng}	Enthalpy of engine exhaust
H_{in}	Turbine inlet enthalpy
H_{out}	Turbine outlet enthalpy
HP _T	Turbine power
K_3, K_4	Constants
N_T	Turbocharger speed
P	Pulse work contribution
Q_{cc}	Heat addition during auxiliary combustion
r_s	Compressor pressure ratio
T_A	Ambient temperature
T_C	Coolant temperature of charge cooler
T_{ccin}	Temperature at auxiliary combustion chamber inlet
T_{exh}	Engine exhaust temperature
T_M	Air manifold temperature
T_{sup}	Compressor outlet temperature
T_{TIN}	Turbine inlet temperature
T_{TOUT}	Turbine outlet temperature
W_A	Mass of air through compressor
W_{cc}	Mass through auxiliary combustion chamber
W_{eng}	Engine mass flow
W_f	Engine fuel mass flow
W_{facc}	Auxiliary fuel mass flow

W_T	Turbine mass flow
W_{TA}	Trapped air mass flow
ϵ_c	Charge cooler effectiveness
η_c	Compressor efficiency
η_{mech}	Turbocharger mechanical efficiency
η_T	Turbine efficiency

1. Introduction

1.1. Background to Investigation

- 1.1.1. Aim of Investigation.
- 1.1.2. Desirability of torque 'back up'.
- 1.1.3. Comparison of engine types (2 and 4 stroke).
- 1.1.4. Supercharging as a method of improving engine power output.
- 1.1.5. Transient operation of supercharged engines.
- 1.1.6. Turbocharging the two-stroke cycle engine.

1. Introduction

1.1. Background to Investigation

1.1.1. Aim of investigation

The aim concerns the application of a combination of a diesel engine, turbocharger and auxiliary combustion chamber for road vehicle traction. Prime movers for vehicle traction must satisfy the following criteria:-

1. High specific output (low specific weight and volume).
2. Suitable torque/speed characteristics (rising torque with reducing speed).
3. Environmental constraints (low noise and emission levels).
4. Low specific fuel consumption.
5. High reliability (infrequent and cheap maintenance).
6. Ease of control (good 'driveability').

The trend towards higher specific power output has led to an increasing association of the diesel engine with turbomachinery. The individual components:- diesel engine, compressor, turbine and even auxiliary combustion chamber, have been combined in a number of ways from simple turbocharging to complex systems such as differential compounding or the Hyperbar. Simple turbocharging is by far the most common and has been established practice on large and medium sized engines for many years. The advent of smaller, more efficient, turbochargers has extended the application to high speed diesels and nowadays the majority of truck engine manufacturers offer a turbocharged option.

Legal requirements have accelerated the trend towards turbocharging so as to meet the minimum horsepower/ton

legislation with smaller, lighter, units. This can, in turn, allow higher payloads and hence greater profitability. There are further benefits in that turbochargers may help to reduce exhaust noise. For maximum payload increase, the turbocharged two-stroke engine could be favoured for its smaller weight and volume.

The aim of this investigation is to examine the important parameters of a two-stroke diesel gas turbine in order to improve power output and low speed torque 'back up'.

In achieving this aim it is intended to consider the need for a control system such that vehicle operator controls be kept to a minimum (throttle and transmission). It is also intended to remove the scavenge blower to keep mechanical complexity to a minimum. A conventional two-stroke engine cannot operate without a crankshaft driven scavenge blower, which can consume up to 15% of the output power at full load and rated speed. A simply turbocharged two-stroke would still require an auxiliary, mechanically driven, blower for starting and running at conditions of low load or low speed.

Elimination of the scavenge blower would have further advantages as the Roots type blower on this engine has been shown to be responsible for a significant proportion of engine noise that had previously been thought to be a characteristic of a two-stroke (1).

The use of an auxiliary combustion chamber, designed to increase the enthalpy of the gas at the turbine inlet, should assist the turbocharger in providing increased air flow at low engine speed. Thus 'back up' torque should

be improved. Turbocharger assistance could also be provided at part load operation and engine start up could be facilitated by using the turbocharger and auxiliary combustion chamber as a self-sustaining gas turbine, thus eliminating the need for a scavenge blower.

The use of auxiliary combustion for 'torque tailoring', by the control of engine boost, should yield good torque characteristics and permit a reduction in transmission complexity. By using the auxiliary combustion chamber to increase turbocharger speed (hence air flow) on acceleration, prior to increasing engine fuelling, it should be possible to reduce (or eliminate) 'smoke plume' emission and obtain improved acceleration.

1.1.2. Desirability of Torque 'Back-up'.

For traction applications, the drive shaft torque, T , required to move the vehicle is a direct function of the traction resistance, and can be represented by:-

$$\begin{aligned} T &= \text{constant} \times \text{traction resistance} \\ &= C [(a + b)W + cV + dV^2] \end{aligned} \quad (2)$$

where,

T = torque

C = constant (a function of wheel diameter and final drive ratio).

a = a function of the gradient.

b = a function of rolling resistance relative to load and tyre pressure.

c = a function of rolling resistance relative to velocity and tyre pressure.

d = a function of wind resistance relative to
shape and frontal area.

W = vehicle weight.

V = vehicle speed.

The torque requirement for various gradients over the vehicle speed range may be represented by a series of curves as shown in Fig. 1.1a. In addition to matching these requirements the output power of the plant must be matched to suit three conditions:-

- (a) maximum speed requirement at zero gradient;
- (b) maximum gradient starting conditions;
- (c) minimum speed acceptable on intermediate gradients.

Conditions (a) and (c) are the governing conditions, since power transmission arrangements can be made to suit (b).

It is evident that conditions (a) and (c) may be met using a power plant delivering a constant horsepower throughout its speed range. Fig. 1.1c. shows a torque curve for such a power plant, superimposed on the traction torque requirements.

The power output of any plant is a direct function of its ability to convert an appropriate amount of energy into useful work, and the efficiency with which it does this.

$$\text{Power} = (\text{thermal efficiency}) \times (\text{mechanical efficiency}) \\ \times (\text{heat energy consumption}).$$

Systems in which the heat energy supply is not directly related to the output power shaft are capable of producing power at a rate which is independent of the speed of the

output shaft. Such units have the ability to produce constant power output over most of their working speed range.

The air supply (hence heat energy consumption) of the reciprocating internal combustion engine is directly related to the speed of the output shaft.

Consequently power output rises with speed whilst torque remains, more or less, constant. This leads to an instability of operation (Fig. 1.1b.) when using such units for vehicle traction. Stepped transmission ratios are used to overcome this problem. Diesel engines are used in heavy road traction due to their high thermal efficiency (hence low specific fuel consumption). Attempts have been made to improve the efficiency, power output or torque characteristics of these engines.

Chatterton (3) and Wallace (4) have carried out theoretical studies of combinations of compressors, turbines and engines grouped into turbocharged, gas generator and compound arrangements (Fig. 1.2.). In both cases the compounded two-stroke diesel was shown to be the most favourable combination to achieve low specific fuel consumption allied with a torque characteristic approaching the ideal hyperbola. The differential compound engine has been fully examined by Wallace (5, 6, 7, 8, 9). Dawson (10) has also devised an arrangement which approaches the ideal; the differential diesel engine.

Other work concerned with improving the performance of the two-stroke diesel engine and providing turbocharger assistance during periods of low exhaust gas energy has

been conducted by Timoney (2, 11, 12, 13) and Tryhorn (14). This work is also discussed in more detail in sections 2.1.3 and 2.1.4.

Other systems such as the Hyperbar, which utilises a combustion chamber in the exhaust duct of a four-stroke turbocharged diesel (15, 16) and a similar two-stroke based system reported by Blencoe (17, 18) are detailed in sections 2.1.5 and 2.2.1.

1.1.3. Comparison of two-stroke cycle and four-stroke cycle engines.

The relative merits of either two-stroke cycle or four-stroke cycle engines have been the subject of controversy ever since their conception. For outputs above 10,000 hp the supremacy of the two-stroke cycle is undisputed, while, with few exceptions, the field below 100 hp is dominated by the four-stroke cycle engine.

Theoretical work by Chatterton (3), and Smyth and Wallace (6), has demonstrated that a two-stroke can give 80% more power output than a four-stroke of equivalent dimensions running at the same speed. The four-stroke engine can operate at a higher b.m.e.p. but the greater number of working strokes per minute of the two-stroke more than compensates for this. The piston heat flow for a two-stroke is some 70% higher than that for a four-stroke operating at corresponding boost ratios and speeds. Aggregate heat flow to coolant, expressed as a percentage of fuel burned, is lower in an opposed piston, ported two stroke than in valve-in-head two-strokes.

Exhaustive testing at Sulzer Brothers carried out by

Steiger (19), involving both two and four-stroke versions of the same engine has made it possible to draw objective conclusions confirming the merits of each type. The work was carried out with special reference to fuel consumption, smoke emission, thermal and mechanical loading, and acceleration. To keep the two engine builds identical, the two-stroke version was a valve-in-head type. Steiger's conclusions of the relative merits of each type may be summarised as follows:

1) Two-stroke cycle engine

(a) The two stroke cycle engine is capable of higher specific output and is thus most suitable where the highest power concentration, with respect to volume and specific weight, is required.

(b) The two-stroke engine can attain the same power output as the four-stroke at a lower rotational speed.

(c) Because the air throughput of the two-stroke need not be directly related to its speed, as with the four-stroke version, it is capable of developing relatively high b.m.e.ps at low speed, part-load, operation.

2) Four-stroke cycle engine

(a) The better part-load fuel consumption of the four-stroke makes it particularly suitable for application where operation for extended periods below full load may be required.

- (b) The four-stroke engine is required to run at a higher rotational speed to deliver the same power as a two-stroke. This may be more suitable in certain applications.
- (c) Superior acceleration makes the four-stroke well suited to traction applications.
- (d) The four-stroke engine requires a lower specific air throughput and thus has the advantage that a smaller charge cooler may be used.
- (e) The lower thermal loading of the four-stroke at equal specific output could lead to increased reliability.

1.1.4. Supercharging as a method of increasing engine power output.

For an engine of given displacement and speed, the power output can only be improved by increasing the mass of fuel efficiently burned per stroke. An increase in the mass of fuel requires a corresponding increase in the density of the air introduced into the cylinders.

$$\text{Air density } \rho = \frac{P}{RT}$$

thus air density $\rho \propto$ absolute pressure P, and,

$$\rho \propto \frac{1}{(\text{absolute temperature, } T)}$$

However, work by Dicksee (20) has demonstrated that (for a real engine) the weight of air is proportional to

$\frac{1}{T^{\frac{3}{4}}}$ probably due to heating both in the manifold and by residual gases.

Thus air density can be increased by either reducing the temperature or increasing the pressure of the charge air. Dicksee has shown that for both naturally aspirated and supercharged engines,

$$W_s = \frac{W_t \times P_s}{P_t} \times \left(\frac{T_t}{T_s} \right)^{\frac{3}{4}}$$

subscript s refers to the standard engine (or other known condition) and subscript t refers to the test condition.

A reduction in charge temperature of a pressure charged engine can be achieved by coolers of either air/water or air/air type. An increase of the charge pressure can be effected by the use of some form of compressor and several options exist:-

- (1) Mechanically driven compressor of the positive displacement type.

This involves the use of a crankshaft coupled blower. When linked with an engine by a direct geared drive, charge pressure may increase with engine speed. This is due to the fact that the blower delivers a constant volumetric throughput of air whilst engine air acceptance at higher speeds is reduced due to charging and scavenging inefficiencies. This effect is particularly noticeable on a two-stroke where the blower will be matched at low speed to permit starting and satisfactory low speed operation, thus power output increases with engine speed whilst torque remains relatively constant. This situation may be improved with the use of a variable speed drive or some form of differential gearing.

An advantage of a mechanically driven supercharger is the good acceleration performance, but the major drawback to this method of supercharging is the relatively large amount of crankshaft power required to drive the compressor. This may range from 15% of power output at full rated speed and load for modest supercharge (boost = 1.4) to 25% for high levels of supercharging (21). Two-stroke cycle engines require such a blower to provide some degree of supercharge to ensure adequate scavenging of the cylinders.

(2) Exhaust Gas Turbocharging.

In this method of supercharging a compressor is driven directly from a turbine, using the waste heat energy in the engine exhaust. The matching of turbochargers to engines is a complex procedure and has been reported by several workers (22, 23, 24). The increase in charge density can have deleterious effects on smoke emission (25). This is caused by the increase in time required to inject the greater amounts of fuel that may be consumed, thus prolonging the period of combustion. Injection will still be taking place as the gases are expanding and cooling, leading to poor combustion in the latter stages and heavier soot formation. This situation may be alleviated, to some extent, by a change in the injection timing or in the injector plunger/hole sizing (9,25). Too rapid an injection, however, may cause cylinder pressure to rise in an uncontrollable manner.

Turbochargers may be matched at medium engine speed to give improved performance over a broad speed range, or, at low engine speed to give improved torque 'back-up' (24).

Matching at low speed can lead to problems with thermal or mechanical engine constraints at higher speeds. These problems may be alleviated by the use of a 'waste gate' to bypass some of the engine exhaust and hence limit boost levels to give acceptable values of peak cylinder pressure. Alternatively the air/fuel ratio may be maintained at higher levels to reduce thermal loading.

Two-stroke cycle engines require a mechanical blower, in addition to a turbocharger, to permit starting and part-load operation. The turbocharger is a free-running machine and, as such, has the potential to supply a constant air flow matched to optimum turbocharger efficiency. This feature would favour the constant power engine. However, in the classical method of turbocharging, the engine imposes restrictions on the turbocharger due to its air acceptance characteristics.

Turbocharging may be carried out by one of two methods:-

(a) Constant Pressure Turbocharging.

This method of turbocharging is generally accepted to be suitable for boost ratios in excess of 2:1. The method features large volume exhaust ducting to damp out exhaust pulsations. The fitting of large exhaust manifolds to act as expansion chambers can alleviate problems caused by exhaust blowdown pulses from one cylinder interfering with the scavenging process in other cylinders (reverse scavenge). Drawbacks of this system include heat and kinetic energy losses and an increase in

acceleration 'lag' due to the large 'dead' volume of the manifolding.

(b) Pulse Turbocharging.

This method is characterised by short, small diameter, exhaust pipes and multi-entry turbines. The method makes use of the kinetic energy available in the exhaust pulse at blowdown. The method is generally used with boost ratios of 2:1 or less. Usually, two or three cylinders are connected to each of the turbine inlets. The cylinders which are connected together are chosen to fire at regular intervals so as to avoid pulse interference on scavenging. The three cylinder per inlet method is generally considered to be superior as two cylinders per inlet may be prone to 'partial admission' (pressure drops to atmospheric, or below, in exhaust ducts) with a consequent reduction in turbocharger efficiency. Advantages of the pulse system are better acceleration and part-load performance due to the small volume exhaust pipes. A disadvantage may arise on certain configurations due to pulses interfering with the scavenging process.

Much discussion has taken place on the relative merits and drawbacks of each method as outlined above (26, 27) and techniques are now being employed to combine the advantages of both methods (28, 29). Such techniques generally involve the use of a 'pulse convertor'.

The pulse convertor (Fig. 1.3a) is a junction

of two pipes. The junction incorporates a suitable reduction in flow area in the downstream section, converting some of the pressure of the exhaust pulse into kinetic energy. By taking advantage of the high momentum of the gas flow through the inlet nozzles, the effect of a pressure pulse entering the junction on the other pipe is reduced. This reduces the amplitude of pressure variations and eliminates periods of no flow (windage periods). As a result the overall turbine efficiency is increased without unfavourable effects on the exhaust processes of the cylinders.

The pulse convertor has not been found to be load sensitive over normal operating ranges but may lead to poor starting (28). Multi-entry pulse convertors are now in use which are based on a modular system (Fig. 1.3b) and connect all the engine cylinders to a single inlet turbine (27).

Variations of turbocharger practice include two stage turbocharging and register turbocharging (30). Two-stage turbocharging (Fig. 1.4a) consists of two turbochargers in series, one a high pressure, and the other a low pressure, machine. In this way boost levels of up to 8 or 9:1 may be achieved. This usually involves a significant lowering of the engine compression ratio (possibly to as low as 5 or 6:1). Register turbocharging (Fig. 1.4b) involves the use of several turbochargers arranged in parallel with the possibility of connecting one or more of

them to the engine. Thus turbochargers may be 'switched in' at will to change the match throughout the speed range. The system gives improved acceleration and part load performance. It has disadvantages in that control is difficult and switching turbochargers on and off line is prone to mechanical failure in the exhaust ducting. Such sophistication is unlikely to be practical for road vehicle engines.

(3) Pressure Wave Supercharging - BBC Compres.

The BBC Compres system is designed to overcome the drawbacks of mechanical superchargers (high power consumption) and exhaust gas superchargers (acceleration lag and smoke emission). It is a mechanically driven gas-dynamic pressure exchanger or Pressure Wave Supercharger. The device is manufactured by the Brown Boveri Company and its superiority has been widely reported (31, 32, 33).

Principle of operation:-

The device (Fig. 1.5) transfers pressure energy from one gaseous working medium to another by a pressure wave process. The media are working against one another while moving axially in a cylindrical shell, without mechanical separation. The process is governed by connecting the ends of the cell to different gas channels or recesses in a certain cyclic sequence. In the channels the gas pressure remains constant, the pressure wave process taking place only within the cell.

Exhaust gas recirculation (EGR) is possible because air and exhaust gases are not mechanically separated in the cell. Development has reduced the level of EGR to

an acceptable minimum, EGR being almost entirely suppressed at full load. The pressure ratio generally increases with gas temperature and volume flow rate. The entire system reacts almost instantaneously to load changes since a change of conditions in any one of the channels influences the whole and is propagated into the cells at sonic speed (31).

Advantages of the system are good driveability with reduced gear changing and improved acceleration and torque 'back-up', combined with reduced emissions (32). The Comprex gives improved specific fuel consumption at low speeds, but at high speeds it is not as good as in a turbocharged engine (33). The Comprex has a relatively large bulk and weight for small high speed engines and the location is limited by the belt drive. The Comprex also exhibits a characteristic noise in operation.

The Comprex does not give a positive pressure drop across the engine throughout its load range and thus it is not suitable for two-stroke engines without some form of assistance (e.g. a mechanically driven blower (34)).

1.1.5. Transient operation of supercharged engines.

The transient performance of automotive diesel engines is becoming more important. Good driveability and low emissions are becoming increasingly significant for reduced driver fatigue and to meet legislative requirements. Load acceptability is of prime importance in the field of electricity generation to maintain stable voltage outputs.

Accordingly the transient behaviour of supercharged diesel engines has been the subject of much theoretical

(35, 36), and experimental (37, 38, 39) work. Comparisons have also been made between engines fitted with turbochargers and those fitted with pressure wave superchargers (Comprex) (32, 40).

The turbocharged diesel is unable to accept large changes in load due to turbocharger 'lag'. When a sudden load is applied to the engine its speed drops, the governor senses the speed drop and causes more fuel to be injected. If the increase in fuel could be matched by a proportionate increase in the air supplied to the engine, a rapid and efficient increase in power would occur. This is not the case; the turbocharger speed, and hence the air supplied, rise relatively slowly due to turbocharger inertia, resulting in rich conditions and poor combustion (38).

From this it follows that crankshaft coupled superchargers, such as positive displacement blowers or the Comprex, would have superior transient performance and this has been demonstrated (32, 40). However these devices have their relative disadvantages as have previously been discussed. Thus various methods have been tried in an attempt to reduce turbocharger lag.

(a) Register Turbocharging (30).

This involves the use of several turbochargers in parallel which may be 'switched in' at will. The use of several small turbochargers means that turbocharger inertia is kept low and acceleration is improved.

(b) Pelton Wheel/Hydraulic oil assistance (12,13,39).

This involves the use of a small pelton wheel fitted to the turbocharger shaft between the compressor and turbine discs (Fig. 1.6a). A high pressure oil supply is utilised to operate the pelton wheel.

It has been found (39) that the fitting of a pelton wheel has negligible effects on the compressor characteristics and the windage losses are significant. Such a system is more likely to cause engine stall in transient operation. At low speeds and mass flows there is sufficient pelton wheel power to drive the compressor without assistance from the turbine. At high speed ($> 60,000$ rev/min) the pelton wheel does not produce any assistance. The method also suffers from the danger of oil frothing.

Tests on a Ruston & Hornsby 6 YEX II (11.36 litre) engine (39) have shown that transient recovery is better than that of the turbocharged engine. Engine speed recovery is improved from 4 secs to 3 secs and peak smoke occurrence from 3 secs to 2 secs. Oil injection takes place over 1.75 secs (≈ 1.4 pints).

(c) Air Injection (39, 41).

The use of air injection involves a modified compressor housing (Fig.1.6b). The housing is fitted with three air nozzles (41). The normal compression process occurs in the inducer and radial sections of the compressor impeller. At the

impeller tip, air is supplied from an external source. Air from the nozzles mixes with air from the impeller, and the injected air, plus the normal air, passes into the diffuser.

The influence of air injection on the compressor characteristics is to increase the delivery pressure ratio and mass flow rate and reduce the torque required to drive the compressor for the same delivery ratio. The surge point tends to move to the right (increasing mass flow) (41).

Transient engine tests on a Ruston and Hornsby 6 YEX II (11.36 litre, 4-stroke) (39) have shown that peak smoke is reduced from about 3 secs to $\frac{1}{2}$ sec and speed response is improved from 4 sec to 1.5 sec. Air injection takes place over 1 second and uses approximately 0.1 Kg of air.

1.1.6. Turbocharging the two-stroke engine.

With reference to turbocharging, the two-stroke engine has its own peculiar problems. The two-stroke engine requires a positive pressure drop across the cylinders, to provide adequate scavenge air, throughout the full range of speed and load. The use of a positive displacement mechanical blower will satisfy these requirements. However the loss of crankshaft power, required to drive the blower, remains constant at any given engine speed, irrespective of load. This results in extremely poor part-load specific fuel consumption.

Turbocharging improves the specific fuel consumption of the two-stroke and is a common feature of large engines

running over a restricted range of speed and load (e.g. electricity generation prime movers). For such use the turbocharger may be matched at a specific point of load and speed and optimum turbocharger efficiency may be obtained.

Small, high-speed, automotive engines present further problems in that adequate scavenge must be maintained over a wide range of speeds and loads. Many two-strokes scavenge on the 'Kadenacy' principle, i.e. having large area exhaust ports such that the exhaust gas leaves with sufficient momentum to create a depression in the cylinder and thus may tend to draw in a fresh charge without the need for blower assistance. This will permit two-strokes to run naturally aspirated under certain favourable conditions. The Rootes TS3 engine will operate in this manner under suitable conditions (42).

For such operation the exhaust porting is an important factor. High exhaust pulse amplitudes (which are of benefit in 'pulse' turbocharging) conflict with the need for rapid blowdown. The most rapid blowdown is obtained by the highest possible rate of exhaust port opening, thus promoting good scavenge on the Kadenacy principle. This is achieved in practice by increasing the width, as opposed to the length, of the exhaust ports (43).

Opposed piston engines have a high swirl rate, hence incoming air may adhere to the cylinder wall (held by centrifugal effects) leaving the central core unscavenged. This leads to a dual process in which the annular space is perfectly scavenged (i.e. all exhaust gas is pushed out by

a 'piston' of incoming air) and the central core has perfect mixing (i.e. progressive dilution with incoming air, a proportion of the mixture being displaced) (44).

The effective flow area of the exhaust port has an almost linear relationship with port opening. The effective area also increases with engine pressure ratio. Certain exhaust manifolds may increase scavenge problems due to the fact that the major flow resistance may move from the port to the exhaust collection belt or the manifold (45). This may be a significant factor when exhaust piping of a small diameter is used to take maximum advantage of exhaust blowdown pulses in the turbocharger.

Adequate scavenging of a two-stroke engine is promoted by having a large engine pressure drop in order to obtain high values of air velocity and mass flow rate. Not only is this wasteful in terms of compressed air utilisation, but it is in direct conflict with the requirement for a high 'back pressure' to aid high levels of supercharge. The Rootes TS3 engine is less critical in this area as the exhaust ports close prior to the inlet ports, cylinder filling thus being more dependent on absolute charge pressure.

The wide range of speed and load required from an automotive engine results in operating zones in which exhaust energy is very low. On a turbocharged engine this results in insufficient turbine energy to drive the compressor at adequate boost levels. The problem is exacerbated by the lower turbocharger efficiencies available on automotive sized units. This factor is most

significant on a two-stroke engine as scavenge is impaired and engine stall may result. A reduction of scavenge will also result in higher exhaust temperatures due to smaller quantities of air diluting the exhaust gas. Thermal loading may thus reach unacceptable levels. Turbocharged two-stroke automotive engines therefore require some form of 'assistance' during periods of low exhaust gas energy (e.g. starting and part-load operation).

Methods of turbocharger assistance.

Turbocharger assistance may take the form of a positive displacement mechanical blower in series with the turbo-compressor. Such an arrangement may lead to unacceptably high boost levels under certain conditions. This may be overcome by using a system such as that described by Dawson (10), involving a differentially driven mechanical blower to increase boost at low engine speed on a four-stroke engine. Timoney (11, 12, 13) used a variable speed mechanical blower in conjunction with a turbocharger and pelton wheel/hydraulic oil assistance.

Other methods of assistance such as air injection (39, 41) and the use of auxiliary combustion (15) (Hyperbar) have been examined on four-stroke engines. The application of auxiliary combustion on a two-stroke has been briefly mentioned in a later Hyperbar publication (16) and has been the subject of work at Leicester Polytechnic by Blencoe (17, 18) and continued in the present investigation.

Some of these methods will be discussed in more detail in Chapter 2.

2. Survey of Previous Work

2.1. High output engine systems

- 2.1.1. Differential diesel engine (Dawson)
- 2.1.2. Differential compound engine (Wallace)
- 2.1.3. Variable compression ratio engine (Timoney)
- 2.1.4. Blower assisted turbocharging for two-stroke engines (Tryhorn)

2.2. Systems using auxiliary combustion

- 2.2.1. Hyperbar system

2.3. Previous work at Leicester Polytechnic

- 2.3.1. Two-stroke turbocharged engine with auxiliary combustion (Blencoe)

2. Survey of Previous Work

2.1. High output engine systems

The developments to attain high specific power output have led to an increasing association of the diesel engine with turbo-machinery. The two-stroke engine is an inherently good match for a turbocharger and has played an important part in such developments. Some of the more notable systems designed to give high specific output coupled with good torque 'back-up' are discussed, in detail, below.

2.1.1. Differential diesel engine (Dawson) (10)

This work involved a four-stroke cycle diesel engine, supercharged by a differentially driven compressor. The drive was arranged in such a way that the boost (and thus the output torque) increased with decreasing engine speed. The output from the differential diesel engine (DDE) approximated to the constant horsepower ideal. The system arrangement is shown in fig. 2.1.

A prototype DDE was fitted to a truck and the performance was compared with a truck having the same engine in simply turbocharged form. The performance comparison is presented in fig. 2.2. It can be seen that the ideal torque hyperbola is approximately attained. The work has allowed the advantages of the DDE to be assessed as follows:

- 1) The DDE gives better performance in a truck as compared to naturally aspirated and turbocharged engines.
- 2) The DDE was more flexible than conventional engine/gearbox systems and performance was more easily tailored to suit various operating requirements.

- 3) Improved fuel economy was obtained over a major portion of its operating range, particularly at high loads and speeds.
- 4) A lower engine speed was required for a given vehicle speed, leading to longer engine life. Engine life may also have been extended by favourable air/fuel ratios with associated lower component temperatures.
- 5) Operator handling was enhanced with the automatic two pedal control.
- 6) The overall cost of the differential power unit plus torque convertor was substantially less than that of a conventional engine and automatic transmission.

Although the DDE was based on a combination of well developed components, the transmission was mechanically complex. The system also suffered mechanical losses associated with the transmission. Further power losses arose from the mechanical blower and its drive. No use was made of exhaust gas energy and the specific fuel consumption was no better than that of a simply turbocharged engine.

Investigations by Chatterton (3) and Wallace (4,5) have shown that a compound scheme would have favourable operating characteristics. Such a scheme has been combined with elements of the DDE and led to the development of the differential compound engine.

2.1.2. Differential compound engine (Wallace) (5,6,7,8,9)

The differential compound engine(DCE) combined the high specific output of the compound engine with differential gearing to give a package whose output approximated to the

ideal torque hyperbola (fig.2.3). The scheme utilised an opposed piston, two-stroke engine for high thermal efficiency and high unit output. The need for a complex external transmission system was eliminated, this in turn offset the cost of the epicyclic differential gearing.

The original version (fig. 2.4a) used fixed geometry turbines. The power turbine being driven by engine exhaust gas while the auxiliary turbine was driven by surplus air from the compressor and was controlled via a bypass valve. The engine drove the compressor through a fully floating, epicyclic gear train. Both turbines were geared to the output shaft via reduction gearing. An air cooler was used between the compressor and the engine.

The advantages of the system were:-

- 1) High b.m.e.p. ratings resulting in an increase of some 150% over the corresponding naturally aspirated engine.
- 2) Good torque/speed characteristics.
- 3) Effective engine braking could readily be incorporated.
- 4) Response to transients was superior to that of a turbo-charged engine in view of the geared connection between all members of the system.

It was later found (8) that torque and power output were lower than predicted due to parasitic and gearing losses. A later version of the DCE incorporated variable geometry turbines and an auxiliary combustion chamber to augment exhaust gas enthalpy (fig.2.4b).

Thermal loading on the two-stroke engine was quite severe, even with piston cooling and for this reason the DCE concept was applied to four-stroke engines.

As previously mentioned the DCE suffered from a high level of mechanical complexity although the associated high costs may have been offset by the fact that a conventional gearbox was not necessary. There were significant noise problems associated with the mechanically driven, positive displacement, compressor and the epicyclic gearing.

2.1.3. Variable compression ratio engine (Timoney)

(2,11,12,13)

The variable compression ratio (VCR) engine concept was proposed by Timoney as a solution to give a constant horsepower output. The system incorporated a variable speed mechanical blower and a turbocharger with pelton wheel assistance

The engine used for the system was based on a Rootes TS3 horizontally opposed piston, two-stroke. The rocker shafts were fitted with eccentrics which altered the stroke and thus the compression ratio. The engine compression ratio (range 8-18 : 1) varied automatically with load. This was achieved via a spring and damper mechanism (fig.2.5). The mechanism adjusted the compression ratio with respect to peak cylinder pressure (not m.e.p. as might be expected) such that safe mechanical loading was not exceeded (136 bar (2000 psi) peak cylinder pressure). The pistons were made of aluminium with a 6.4 mm ($\frac{1}{4}$ in) alloy steel plate screwed to the top. Oil cooling of the pistons was employed.

The variable speed mechanical blower drive (to increase boost at low speed) was effected using a 'V' belt drive with variable diameter pulleys. This blower was placed in series with a turbocharger (2,11).

The V.C.R. system gave good performance but the variable speed drive to the mechanical blower was complex and bulky and the blower was noisy. Later work (12,13) used two turbochargers in series or a single turbocharger with Pelton Wheel assistance. This consisted of a modified turbocharger having a Pelton wheel fitted to the shaft between the compressor and turbine discs (fig.1.6a). The Pelton wheel was fed with a high pressure oil jet. The system rendered a mechanical blower redundant as starting and part-load operation could be facilitated by Pelton wheel assistance. The system also improved turbocharger 'lag' during transients. The output torque of the V.C.R. system approximated to the ideal and is shown in fig. 2.6.

The engine suffered from lubrication problems on the rocker shafts, due to relatively slow moving oscillations (no rotation) with unidirectional loading. The Pelton wheel assistance system was bulky and complex and may have resulted in a certain amount of oil 'frothing'. Other workers (39) have suggested that there were significant windage losses in certain situations.

2.1.4. Turbocharger with mechanical blower assistance for two-stroke engines. (Tryhorn) (14)

Tryhorn utilised a Roots blower in series with a turbocharger. The blower was used as a second stage compressor. The mechanical blower gave assistance to the turbocharger under acceleration and conditions of low exhaust gas energy (starting and low-load operation).

The Roots blower maintains a constant volumetric throughput of air at the prevailing inlet pressure (turbocharger delivery pressure). The pressure rise across the

blower was found to increase with an increase of inlet pressure. Thus the power absorbed by the blower rose rapidly with the rise of the delivery pressure of the turbocompressor. This led to high power losses and resulted in unacceptable mechanical loads on the engine.

To relieve excess pressure Tryhorn (14) fitted a pipe in parallel with the Roots blower and incorporated a recirculation valve (fig.2.7). The effect of this valve was to maintain a tolerable pressure rise across the blower (.2 - .3 bar (3-4 psi)). The characteristics of the blower being altered from those of a constant displacement type to those of a variable throughput type.

The system yielded a high torque at low speeds and then constant horsepower up to full speed. The disadvantages of the system were that it retained a mechanical blower leading to high noise levels and increased complexity.

2.2. Systems using auxiliary combustion

As previously mentioned turbocharged engines can be required to operate in regimes of low exhaust gas energy. These regimes are especially critical on two-stroke engines and during transients. Auxiliary combustion has been used by some workers to improve turbocharger operating conditions. Auxiliary combustion systems generally incorporate some form of combustion chamber, fitted in the engine exhaust ducting, designed to augment exhaust gas enthalpy prior to the turbine inlet. Some specific systems are discussed in detail below.

2.2.1. Hyperbar system (Andre-Talamon) (15,16)

The hyperbar system is a novel concept for the very high supercharging of four-stroke diesel engines.

It is based on a parallel supercharging method involving the use of single or double stage turbocharging incorporating an auxiliary combustion chamber. Values of boost pressure ratio in excess of 10 : 1 and high overall supercharge efficiencies have been obtained.

The arrangement of the Hyperbar system is shown in figure 2.8. The thermal efficiency of the system has been shown to be good (15) as the proportion of heat lost to coolant decreases with increasing b.m.e.p. and mechanical losses form a smaller proportion of the output. The use of such high levels of supercharge means that the compression ratio of the engine must be substantially reduced (to as low as 6 : 1) to meet mechanical loading constraints.

The above system has certain disadvantages in that it requires major modification of the engine. The use of such low compression ratios may lead to starting difficulties. The system has been most successful on relatively large engines (10 - 60 litres) which utilise relatively large turbochargers having better efficiencies than smaller units. The use of a throttled bypass line, for two-stroke based systems, has been discussed in a later Hyperbar publication (16), although no two-stroke installation is reported.

2.3. Previous work at Leicester Polytechnic

The work at Leicester Polytechnic is based upon earlier work carried out at Bath University by Few (8,9). The programme carried out immediately prior to this investigation was conducted by Blencoe (17,18) and is discussed below.

2.3.1. Two-stroke turbocharged engine with auxiliary combustion (Blencoe) (17,18)

The system adopted by Blencoe involved feeding the air from a turbocompressor to a Rootes TS3, opposed piston, diesel engine (Type 3D215). The engine exhaust configuration, including auxiliary combustion chamber and expansion box ($\frac{1}{2}m^3$), was very complex and is shown in figure 2.9. A Roots type blower was fitted to the engine and this was used to supply air for auxiliary combustion. The engine exhaust bypassed the auxiliary combustion chamber and was mixed with the hot gas from auxiliary combustion prior to the turbine inlet.

The engine was fitted with a DPA type fuel injection pump with a mechanical governor. At full rack setting this pump delivered a fixed amount of fuel at a given engine speed. The engine compression ratio, injection timing and fuel injectors were left as standard.

Blencoe found that it was only necessary to use auxiliary combustion at low speed, obtaining a very modest level of torque 'back-up'. Results from this investigation are shown in figures 2.10 and 2.11. Manufacturer's performance characteristics for this engine in standard trim have been superimposed on Blencoe's results and show a considerable discrepancy, the manufacturer's figures being better than Blencoe's.

The disadvantages of this system are due to the complex and bulky nature of the exhaust and auxiliary combustion ducting and the expansion box. The retention of the noisy mechanical blower introduces further complexity.

The disappointing results, in comparison with the manufacturer's figures, leave the 'base line' of the work open to doubt. This was probably due to the fact that Blencoe's 'base line' was obtained with a worn engine whilst later results were obtained using a reconditioned unit. Also, during the initial stages of the present programme, problems were experienced with the fuel lines and injector pump. These faults required much detailed attention before the standard engine characteristics corresponded with those published by the manufacturer.

Accordingly a more detailed investigation and optimisation of a turbocharged two-stroke engine with auxiliary combustion is justified.

3. Project Approach

3.1. Proposal

3.2. Aim of the programme

3.3. Project objectives

3.3.1. Theoretical

3.3.2. Practical

3.3.3. Computerised processing

3. Project Approach

This section sets out the proposal and states the aim of the research programme and then deals with the approach to the work.

The approach is dealt with in three sections: theoretical, practical and computerised processing.

3.1. Proposal

The use of a two-stroke cycle engine theoretically permits high specific power outputs to be obtained. One problem associated with the two-stroke is the provision of sufficient scavenge air for good combustion. The higher thermal loading of the two-stroke requires that the bulk exhaust temperature must not exceed 600°C (otherwise piston damage may occur (11)) whereas the maximum continuous allowable temperature at the turbine inlet is 700°C (for automotive-sized turbochargers). Operation at part-load conditions will reduce engine exhaust (turbine inlet) enthalpy, resulting in a fall in turbine power and hence a reduction in boost levels.

This reasoning suggests that some form of heat addition (auxiliary combustion) between the engine and turbocharger would be beneficial. Turbocharger output could be adjusted to suit engine running requirements and engine starting could be achieved by using the auxiliary combustion chamber and turbocharger as a self-sustained gas turbine. The inclusion of an auxiliary combustion chamber would also permit some degree of torque 'tailoring' by control of engine boost levels. Transient performance could also be improved by using auxiliary combustion to increase turbocharger speed, hence airflow, just prior to increasing engine fuelling. This would result in reduced smoke emission and turbocharger 'lag'.

Three possible combinations of engine, turbocharger and auxiliary combustion chamber are shown in figure 3.1. Figure 3.1a. shows a series system in which all the engine exhaust gas is passed through the auxiliary combustion chamber. Unless the engine has a 'straight through' passage for air at all positions of crank angle, a mechanical blower would be necessary to allow starting. This configuration has the advantage that exhaust smoke would be subjected to a reburning process and hence emissions would be reduced. Unless very high values of scavenge are used there is a danger of having insufficient oxygen in the engine exhaust to sustain auxiliary combustion.

Figure 3.1b. shows the same system with provision for mixing a bypass air flow with the engine exhaust gas prior to the auxiliary combustion chamber. A mechanical blower is no longer required to start the engine. All the engine exhaust gases are still subjected to a reburning process and extra air is provided for auxiliary combustion. The bypass air supply will not be required when there is no auxiliary combustion.

Figure 3.1c. shows a similar system to 3.1b. Auxiliary combustion utilises clean air only, the engine exhaust gas being mixed with hot gases from auxiliary combustion before the turbine inlet. The system has most of the advantages as that shown in figure 3.1b., but engine exhaust smoke is not subjected to a reburning process.

3.2. Aim of the programme

The aim of the programme is to investigate the performance of combinations of a two-stroke cycle engine with different turbocharger assemblies and an auxiliary combustion chamber, for possible use in a road traction application. The work is intended to

re-evaluate and extend previous work carried out at Leicester Polytechnic by Blencoe (17,18), making use of, and adding to, any existing equipment and computer programs.

3.3. Project objectives

The project was approached with certain objectives in mind, namely:-

To understand the interaction of the various components of the engine-turbocharger-auxiliary combustion chamber system such that a compatible operational system could be achieved. In order to achieve this understanding a versatile theoretical analysis was necessary, to proceed alongside experimental variations, which could be checked at each stage.

It was also necessary to bear in mind the operation of the final system. Thus the engine controls had to be made automatic with feedback of various parameters to permit adjustments to be made. This requirement also affects the choice of instrumentation so as to permit assimilation of the required data for automatic control purposes. The use of computerised data acquisition and control played an important part in achieving these objectives.

The project approach is dealt with in a more detailed manner below, covering theoretical, practical and computerised processing aspects.

3.3.1. Theoretical approach

The theory and equations governing the components of the system are well known and have been detailed by other workers. The characteristics of the Rootes TS3 engine have been well established by Wallace (46). A theoretical analysis, using a quasi-steady flow model, has been published by Wallace (4,46) which was used as the basis for a computer performance prediction program. The thermal and mechanical

limitations of the TS3 engine have been discussed by Timoney (11). The equations governing the behaviour of compressors and turbines have been published by Chatterton (3) and Wallace (4).

Several computer programs for performance prediction have been utilised. A standard engine performance program originating from work done by Wallace (4,46) was inherited from previous workers (Blencoe (17)). The program has been modified only in minor details. The program was tailored to predict the actual performance of the standard engine for later use as part of a much larger program, written to predict the performance of the final engine system.

A computer program developed by Blencoe (17) was used to indicate possible areas of turbocharged engine operation. The program does not predict specific operating points but regions in which favourable conditions exist for matched operation of the turbocharger and engine.

A program was written for the final system, the Diesel Gas Turbine, to predict many aspects of system performance. The program used the previously mentioned Wallace program (4) as a subroutine and incorporated routines to predict turbocharger performance and auxiliary combustion chamber fuelling rates. The program is based on a simple cycle analysis and involves the use of digitised compressor maps as 'look-up' tables to identify the operating point of the specific compressor in use.

3.3.2. Practical approach

The practical approach to the project was to test the standard engine and compare the experimental results with the

published results of other workers and those of the engine manufacturer. If these results proved acceptable, the base line by which later developments would be evaluated would have been established.

The next stage was to investigate the operation of the engine in conjunction with turbochargers, both with and without scavenge blower assistance. A schematic arrangement of this system is given in figure 3.2.

An auxiliary combustion chamber was to be incorporated to augment exhaust gas enthalpy and hence increase turbine work. A schematic arrangement of this system is given in figure 3.3. This system proved unsatisfactory due to flame instability in the auxiliary combustion chamber. The system was then modified to incorporate an air bypass line with a variable throttle. The bypass air was mixed with the engine exhaust gas prior to the auxiliary combustion chamber. This configuration, (figure 3.4.), provided adequate starting and running both with and without scavenge blower assistance. However it proved only a partial solution to flame instability and thus a third and final stage of development was proposed.

The final system utilised only clean, constant pressure air for auxiliary combustion. The engine exhaust gas was mixed with the hot gases from the auxiliary combustion chamber just prior to the turbine inlet. Thus benefit could be obtained from the exhaust pulse energy. The scavenge blower, having been shown to be redundant, was removed and replaced with a blanking plate and a fabricated air manifold. A schematic diagram of this system is shown in figure 3.5.

3.3.3. Computerised processing

The final system, the Diesel Gas Turbine, is a complex system. If it is to have any application in the road traction field, the machine/operator interface should be no more complex than that of any presently available system. Permissible controls, therefore, are an ignition switch for system start-up, an accelerator and transmission control.

To enable the two-stroke Diesel Gas Turbine to fulfil this requirement, some form of complex control system is desirable. Further, the control parameters may differ for different environments; e.g. some applications may require reliability and long life, others may require intermittent high power for emergency use. Thus the control system needs to be flexible.

A computerised control system would have the required flexibility. Changing the control parameters for a particular application would just be a matter of altering the software. Such a computer system would be bulky and costly and, as such, unsuitable for road vehicle application. The advent of cheap microprocessors solves the problems of cost and bulk. Hence the control system envisaged for the research programme was microprocessor based. A satisfactory control system requires feedback of the various operating parameters. For a microprocessor system the feedback signals must be digital. As much instrumentation provides analogue voltage signals (47) some form of data acquisition and conversion is required.

Data acquisition systems are easily available for converting analogue voltages to digital signals which may be readily

interfaced with a computer. A simple control and data acquisition system does not require a high speed of acquisition. However, if the system is to be used for the study of transients, different criteria apply and the speed of data acquisition becomes a significant parameter (48). Certain parameters to be measured (e.g. peak cylinder pressure) dictate that information retrieval should be crankshaft coupled. The study of transients may require that several parameters be logged at each engine cycle. An engine operating at 3000 rev/min with 20 channels of information requiring to be logged each cycle necessitates a data acquisition system having a multiplexing and conversion rate of at least 1 kHz (for a two-stroke cycle engine). Certain other variables may not need such frequent sampling or may require time dependent (real time) rather than crankshaft dependent acquisition. Data acquisition at such a speed implies that a satisfactory control system may be constructed utilising control signals based on an average of, say, 50 cycles. This would damp out control system fluctuations due to cyclic variations of the monitored parameters.

To enable the computerised control to be effected on the engine system it is necessary to employ actuating devices that are amenable to computer controlled (i.e. digital) instructions, if digital to analogue conversion is to be avoided. For this reason, stepper motors were used as they are controllable with standard logic level pulses and do not suffer from instability problems.

A full description of the computer controlled data acquisition and control systems is given in chapter 6.

4. Experimental Apparatus

4.1. Major components

- 4.1.1. Engine
- 4.1.2. Dynamometer
- 4.1.3. Turbochargers
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4.2. Gas flow systems

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4.3. Services

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- 4.3.2. Lubricating oil
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4.4. Instrumentation

- 4.4.1. Instruments - services - mechanical
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- 4.4.4. Instruments - test measurements - electrical/
electronic

4. Experimental Apparatus

The test rig consisted of an engine and hydraulic dynamometer. The engine was first tested in standard form and then modified in several stages as detailed in section 7.

4.1. Major components

The major components of the test rig were as follows:-

4.1.1. Engine

The engine used in the test rig was a Rootes TS3 opposed piston diesel engine, type 3D215. This is a three cylinder unit in which rockers connect the pistons to a single crankshaft. A cross-sectional view of the engine is given in figure 4.1. and a diagrammatic view of the engine linkage layout is shown in figure 4.2.

Detailed engine specifications are given in table 4.1. The engine is a ported type and the desired exhaust piston lead is controlled by the arrangement of the rockers. The port timing of the engine is shown in figure 4.3. Exhaust and inlet port areas, with respect to crank angle, are displayed graphically in figure 4.4. which also indicates their phase relationship.

The engine was modified to accept a pressure transducer in cylinder No.1. by the provision of a special pocket. This was facilitated by boring through the cooling water jacket and into the combustion space of the cylinder. The fitting was based on an arrangement devised by Wright (46) and is illustrated in figure 4.5.

A further engine modification was devised by Blencoe (17), involving the outlet manifold of the scavenge blower. Two stub pipes were welded into position such that the

turbocharger could be connected in series with the scavenge blower. A blanking plate fitted inside the manifold diverted the air to the turbocharger and from there it was delivered to the engine air chest. The modified scavenge blower manifold is illustrated in figure 4.6.

In the later stages of the project (Stage III) the scavenge blower was removed from the engine and replaced with a blanking plate and a fabricated air manifold. The fabricated manifold delivered air directly from the turbocharger to the engine air chest and incorporated a butterfly valve to isolate the engine during system start-up. The manifold is illustrated in figure 4.7.

Engine lubricating oil, which had been specially filtered, was used to lubricate the turbocharger bearing. In addition an auxiliary oil pump was fitted to supply lubricating oil to the turbocharger bearing when the engine was not running. This system is described in detail in section 4.3.3.

The fuel injection system consisted of a CAV DPA type rotary pump with a mechanical governor delivering fuel through one single hole injector per cylinder, the injectors opening at 175 atmospheres. The external fuel supply is described in section 4.3.5.

The engine crankshaft pulley was fitted with a marker disc to permit synchronisation of the measurement of peak cylinder pressure and to facilitate the start of a data acquisition scan. This device is detailed in section 3.4.4.

4.1.2. Dynamometer

A Redman-Heenan-Froude hydraulic dynamometer type DPX4 was used to apply load to the engine output shaft. Flexible couplings were used to minimise vibration and bending loads. The dynamometer was modified to accept a strain gauged cantilever system to give an analogue voltage readout of applied load. Further details are given in section 4.4.4. (see also Appendix 3). The brake was also modified to enable automatic loading by the incorporation of a stepping motor and toothed belt drive. Load could be applied via switches on the stepper motor drive system or under computerised control. The control system is discussed in detail in section 6.3. Figure 4.8. shows the standard and modified dynamometer and figure 4.9. shows the dynamometer characteristics.

The free end of the dynamometer shaft was fitted with toothed discs to permit the measurement of engine speed using optical and magnetic sensors in conjunction with counter/timers and frequency/voltage convertors. Further details are given in section 4.4.4.

4.1.3. Turbochargers

The turbochargers used were Holset radial flow types, 3LD1 and 4LGK, typical sectional views of which are given in figures 4.10 and 4.11 respectively. The turbine wheel and shaft are welded together to form a single unit which is dynamically balanced. The compressor wheel is made as a separate component and is also dynamically balanced. Thus a new compressor wheel may be fitted to a turbine wheel and shaft assembly without the need for subsequent balancing.

The complete rotating assembly runs in a central plain bearing which is fully floating. The stability of the rotor assembly is maintained throughout its speed range by the oil films formed between the bearing, the shaft and the housing. The shaft also includes a thrust ring and grooved sleeve for the provision of a piston ring type seal at the compressor end.

Turbocharger types are denoted by a frame size number and compressor housing/wheel set, e.g. 3LD1, 4LGK. The turbine housing size is denoted by its throat area in square inches and possibly a coding to denote a twin entry type. Thus the full turbocharger designation may be 3LD1 1.57 or 4LGK 2.6 T2 for example.

Several turbocharger assemblies were used during the course of the work. A Holset 3LD1 was used first. This turbocharger was used with turbine housings of 2.5 sq.in. and 1.57 sq.in. throat areas. During the later stages of the project a 1.41 sq.in. throat area housing was also used. Compressor characteristics for the Holset 3LD1 are shown in figure 4.12. A Holset 4LGK compressor with a 2.6 sq.in. throat area turbine housing was also used in the latter stages of the project. The compressor characteristics for this turbocharger are shown in figure 4.13 and the turbine mass flow characteristics for all the turbine housings used during the course of the project are shown in figure 4.14.

All the turbochargers used in the project were fitted with a special magnetised compressor nut which, in combination with a pick-up coil and frequency counter, allowed

the turbocharger speed to be measured. This speed measuring system is described in section 4.4.2.

4.1.4. Auxiliary combustion chamber and housing

The auxiliary combustion chamber, which was used in later tests, was a single chamber unit of a Rolls Royce Tyne gas turbine aircraft engine combustion section. This was housed in a large bore duct, being rigidly attached to a fixing bracket at one end. The other end was supported in a sleeve with a sliding fit to allow for axial thermal expansion.

The chamber consists of a swirl section including a fuel delivery nozzle and a flame tube with primary, secondary and tertiary combustion zones. The chamber was supplied with propane which was fed through a fabricated nozzle and ignited with a spark plug and a capacitive discharge ignition unit. The ignition circuit is described in detail in section 4.3.5. The arrangement of the auxiliary combustion chamber with an enlarged view of the burner nozzle is shown in figure 4.15. and the ignition arrangement is given in figure 4.16.

During the development of the test rig the arrangement of the combustion chamber inlet and outlet transition pieces was changed to suit the modified ducting and different gas flow paths.

4.2. Gas flow systems

In the final arrangement of the test rig there were three gas flow systems. These were the engine air supply, the air supply to the auxiliary combustion chamber and the exhaust system. There was also a slave air supply used for starting the engine but

this will be dealt with separately in section 4.3.5. The engine and auxiliary combustion chamber air supplies were changed during rig development and detailed descriptions of each configuration are given below.

4.2.1. Engine air supply

With the engine in its standard form in the first tests, air was supplied to the engine air chest via a Wade Engineering 'Roots' type scavenge blower. A Preston Meter (9) was fitted to measure air flow. This system is shown in figure 4.17.

With the turbocharger in series with the scavenge blower, the air from the scavenge blower was delivered via one section of the modified blower manifold (figure 4.6) to the turbocharger inlet. The turbocharger air outlet was connected to the engine air chest via the other section of the modified scavenge blower manifold. Both connections to and from the manifold were made with reinforced rubber hose (capable of withstanding the high air temperatures and pressures involved). The arrangement is shown in figure 4.18.

The engine was operated in a freely turbocharged mode (no scavenge blower assistance) by removing the connection from the scavenge blower to the turbocharger such that the turbocompressor took air directly from the laboratory. The air from the scavenge blower was allowed to discharge to atmosphere such that the blower was running unloaded and only absorbing power to overcome frictional forces. The air from the turbocharger was delivered to the engine air chest via the modified scavenge blower manifold.

This arrangement is illustrated in figure 4.19.

In the Stage II development, the air supply was taken through the scavenge blower and turbocharger, as in Stage I, and then split into two parts. One part was passed to the engine air chest via the modified scavenge blower manifold. The other part bypassed the engine and was delivered to the auxiliary combustion chamber via a throttling valve. The valve could be adjusted so as to vary the bypass air flow. This arrangement is illustrated in figure 4.20.

In Stage III system the scavenge blower was removed and a fabricated air manifold (figure 4.7.) substituted. Air was taken directly from the laboratory into the turbo-compressor. From there the air was passed to the engine air chest via the fabricated manifold. The manifold incorporated a butterfly valve to isolate the engine during system start-up. Air was also taken from the compressor via the throttled bypass line to supply the auxiliary combustion chamber. This arrangement is shown in figure 4.21

Tappings for pressure and temperature measurements were incorporated at scavenge blower outlet, turbo-compressor outlet, engine air chest inlet, air bypass and auxiliary combustion chamber inlet.

4.2.2. Air supply to the auxiliary combustion chamber

In the first tests utilising auxiliary combustion (Stage IIa) the auxiliary combustion chamber was placed in the engine exhaust line and thus was not directly supplied with air. The air to sustain auxiliary combustion was present in the exhaust gas due to excess air during firing

and to scavenge air being passed through the engine cylinders during exhaust purging. This arrangement is shown in figure 4.20a.

In later tests (Stage IIb) the air to sustain auxiliary combustion was supplied from the turbocompressor via a throttled bypass line, as previously described, and mixed with engine exhaust gas prior to the auxiliary combustion chamber inlet. This arrangement is shown in figure 4.20b.

The final series of tests (Stage III) utilised only clean air for auxiliary combustion. The air was again supplied from the turbocompressor via a throttled bypass line as previously described. This arrangement is shown in figure 4.21.

4.2.3. Exhaust system

The exhaust system was modified several times during the development of the test rig. During standard engine tests the exhaust was vented directly to atmosphere.

The introduction of a turbocharger into the system necessitated some exhaust modifications to accommodate the turbine. During Stage I development the auxiliary combustion chamber was incorporated in the engine exhaust duct between the engine exhaust manifold and the turbine inlet. The auxiliary combustion chamber was not fuelled and ignited during the early tests. Connections were made using 76 mm (3.0 in) bore piping and transition pieces to accommodate the auxiliary combustion chamber of 127 mm (5.0 in.) bore. A special purpose bellows was used in the exhaust system between the manifold and the

auxiliary combustion chamber to allow for thermal expansion. Tappings for pressure and temperature measurements were incorporated in the exhaust duct just after the engine exhaust manifold and in the transition piece from the auxiliary combustion chamber to the turbine inlet. Three thermocouples were used at this point, placed at 90° to the direction of flow. The thermocouples were placed at different depths in the flow and equispaced around the pipe circumference. The true reading was taken to be an average of the three. Pressure and temperature tappings were also incorporated in the exhaust duct after the turbine. A Bosch Smoke-meter sampling head and tappings for exhaust gas analysis were also incorporated downstream of the turbine. This exhaust arrangement is shown in figure 4.19.

In later tests it proved possible to ignite the auxiliary combustion chamber at all engine speeds in a no-load condition. Any attempt to put load on the engine caused the flame to be extinguished, probably due to increased exhaust pulsation levels resulting in flame instability.

This led to the introduction of another exhaust system in which the air bypass mentioned in section 4.2.2. was connected into the exhaust prior to the auxiliary combustion chamber (figure 4.20). The rest of the exhaust system remained as described above. This configuration had the result of reducing exhaust pulsation effects in the auxiliary combustion chamber. During these tests it proved to be possible to ignite the auxiliary combustion

chamber at all engine speeds and to maintain auxiliary combustion up to approximately $\frac{1}{3}$ of full load conditions, whereupon the flame became unstable and was extinguished. However it proved possible to start the engine system in two modes:

- a) The engine was started in the normal manner and auxiliary combustion was then initiated.
- b) With the scavenge blower disconnected from the air supply circuit, a slave air supply was used to initiate auxiliary combustion. When the gas turbine (formed by the turbocharger and auxiliary combustion chamber) was self-sustained and supplying air to the engine, the slave air supply was disconnected and the engine started in the normal manner. This latter procedure rendered the scavenge blower redundant.

This led to the Stage III development. In this system the engine exhaust was not passed through the auxiliary combustion chamber. Auxiliary combustion utilised only clean, constant pressure air from the air bypass line. The engine exhaust was mixed with hot gases from auxiliary combustion just prior to the turbine inlet. The arrangement of this system is shown in figure 4.21. This system allowed engine exhaust pulses to be utilised in the turbine whilst auxiliary combustion is stable due to the use of clean, constant pressure air. For these tests the scavenge blower was removed and replaced with a blanking plate and fabricated air manifold as described in section 4.2.1. The tests utilised

several different turbocharger assemblies (some of which necessitated the use of a different transition section between the auxiliary combustion chamber and the turbine).

4.3. Services

Services utilised by the test rig were cooling water, brake water, lubricating oil (for both engine and turbocharger), fuel (for both engine and auxiliary combustion chamber) including an auxiliary combustion chamber ignition system, and an external air supply for initiating auxiliary combustion. These supplies will be considered separately as follows:-

4.3.1. Cooling and brake water

The components of the test rig requiring a water supply were the engine, the brake and pressure transducers. The brake was supplied directly from the laboratory mains supply. The waste water from the brake was vented directly to an atmospheric drain. The brake waste pipe incorporated a gate valve to allow a back pressure to be applied. The body of the brake was fitted with a pressure gauge to measure the back pressure. Before the tests commenced a static (i.e. brake shaft not rotating) back pressure of 1.7 bar (25 p.s.i.) was applied. The cooling water for the pressure transducers were also supplied from the laboratory mains and the waste was vented to an atmospheric drain. Water was also used as a heat source for the bottled propane supply to the auxiliary combustion chamber during some tests.

Engine cooling was accomplished by a closed circuit system. The circuit included a header tank fed from the mains via a ball cock valve. Circulation was achieved

by means of the engine driven water pump. Temperature control was effected via a thermostatically operated valve to dump water from the header tank to an atmospheric drain. This caused 'make-up' water to enter the header tank via the ball cock valve. The coolant supply was fitted with a Rotameter type flow indicator and remote reading vapour pressure thermometers were used to monitor coolant temperature. There was also a sensor fitted to activate a warning light and siren when the coolant temperature reached excessive levels (95°C). The water supply arrangements shown in figure 4.22.

4.3.2. Lubricating oil

Lubricating oil was required for the engine and the turbocharger bearing. The engine system was standard utilising an engine driven gear pump, together with an oil cooler and filter.

The turbocharger lubricating oil supply was taken from a tapping at the outlet of the engine oil filter. The oil was taken via a further filter to the turbocharger housing and returned to the engine sump via a gravity drain.

During later developments the turbocharger was run as a self-sustained gas turbine prior to engine start-up. This necessitated an auxiliary oil supply. The auxiliary oil supply was effected by incorporating an external, automotive, vane type, lubricating oil pump driven by an electric motor. Oil was taken from the engine sump to the auxiliary pump. The supply from this pump was tapped into the supply line between the engine oil filter outlet

and an extra oil filter. Non return valves were incorporated to obviate reverse flows under conditions when either the engine lubricating oil system or the auxiliary system were working alone. The auxiliary supply incorporated a pressure relief valve set to operate at 2.7 bar (40 p.s.i.) which returned the oil to the engine sump.

Pressure gauges were incorporated in both the engine supply and the turbocharger supply lines. Both lines incorporated low pressure sensors linked to warning lights and an alarm siren.

The auxiliary supply pump was switched off once the engine was started and engine oil pressure had reached an acceptable level, whereupon the engine system took over the lubrication of the turbocharger bearing. The lubricating oil supply arrangement is shown in figure 4.23.

4.3.3. Engine fuel

Engine fuel oil was gravity fed from a 'constant head' supply tank. The 'constant head' reservoir was maintained by a pumped recirculatory system from the main supply tank. The 'constant head' reservoir supplied fuel to the engine lift pump. From the lift pump the fuel oil was filtered and delivered to the injector pump via a 'Litre Meter' turbine type flowmeter. Careful attention was paid to the spill lines to ensure that only net fuel flow was measured (see Appendix 2). The injector pump was a CAV DPA, rotary type, with a mechanical governor. Fuel was injected into the cylinders via one, single hole, injector per cylinder, opening at 175 bar (2538 p.s.i.).

The fuel oil supply arrangement is shown in figure 4.24.

4.3.4. Auxiliary combustion chamber fuel

The auxiliary combustion chamber utilised propane from a bottled supply. The supply was taken via an adjustable pressure regulator and pressure gauge. The supply also incorporated an on/off solenoid valve to permit computer controlled initiation of the fuel supply.

The propane bottle was immersed in a water bath to act as a heat source and prevent gas bottle icing resulting in a supply pressure drop. The water bath was kept at a constant level by having a mains water supply and an overflow to waste. The gas bottle was suspended, in its water bath, from a gantry using a strain gauged proving ring arrangement permitting computerised gas flow measurements on a gravimetric principle (see Appendix 4). The proving ring arrangement is detailed in section 4.4.2. and the propane supply arrangement is shown in figure 4.25.

4.3.5. Ignition of auxiliary combustion chamber

The ignition of the auxiliary combustion chamber was effected by a Champion N9Y long reach spark plug. The spark energy was supplied by using a proprietary 'Spark-rite' electronic, capacitive discharge, ignition unit in conjunction with an automotive ignition coil. The spark circuit consisted of an RS 555 timer connected in an astable mode. Spark frequency was variable from 1 Hz to 10 Hz. The system operated from a 12v supply (engine batteries). The ignition unit was switched on and off via a relay operated from an open collector TTL integrated circuit. This enabled ignition to be initiated via a manual switch

or automatically under computerised control. The system arrangement is given in figure 4.16 and the timing circuit and relay connections are given in figure 4.26.

4.4. Instrumentation

Instrumentation of the test rig is considered in two major sections. One referring to instrumentation connected with services and the other referring to that connected with test measurements. Each major section is divided into two subdivisions: mechanical instrumentation and electrical/electronic instrumentation. Full details of calibration procedures are given in section 7.5. The instrumentation that was used during the project is also presented in Table 4.2.

4.4.1. Instruments - services - mechanical

a) Cooling water

The temperature of the engine cooling water in the header tank was monitored using a dial indicating, vapour pressure thermometer. Control of this temperature was effected by a thermostatically activated dump valve. An uncalibrated flow meter in the return line to the header tank indicated flow.

The coolant temperature at the engine outlet was measured using a dial indicating, vapour pressure thermometer. An electrical thermostatic warning device was also fitted. The device was connected to a warning lamp and alarm siren to indicate high coolant temperatures (95°C).

A calibrated Bourdon pressure gauge was fitted to the water brake to measure the static water back pressure before the commencement of each test.

b) Lubricating oil

Lubricating oil pressure was monitored at the engine lubricating oil filter outlet and also at the turbocharger bearing inlet using Bourdon gauges. Each oil circuit was fitted with an electrical low pressure switch connected to warning lights and an alarm siren to give warning of low oil pressure (0.7 bar (10 p.s.i.)).

c) Propane pressure

Propane supply pressure to the auxiliary combustion chamber was monitored using a calibrated Bourdon gauge. This gauge was primarily used to give warning of an emptying gas bottle.

4.4.2. Instruments - services - electrical/electronic

a) Diesel fuel oil

Measurement of the flow rate of diesel fuel oil was effected by a turbine type flowmeter. The transducer produces a frequency output, dependent upon flowrate. The relationship is linear with this particular transducer (a Litre Meter LM25GN) for flow rates above 0.2 kg/hr. The frequency output was filtered to remove a d.c. component and then 'cleaned up' using a Schmitt Trigger. The pulses were then TTL compatible and a counter was used. The counter fed a data latch and a digital to analogue convertor. The latch was used such that a constant analogue voltage output was produced. The output voltage was 'updated' at the end of each sampling period (≈ 5 s). A circuit diagram of the frequency to voltage conversion unit is given in figure 4.27 and the timing/reset circuit is shown in figure 4.28.

The output from this unit was suitable for computerised data acquisition. The transducer arrangement is shown in figure 4.24. A more detailed description of this system is given in Appendix 2.

b) Propane flow rate

Various systems were evaluated for propane flow measurement before a gravimetric system was adopted. The propane bottle was suspended from a strain gauged proving ring attached to a gantry. The strain gauges were fitted in accordance with procedures as detailed by Dorsey (49). A Sangamo Weston C56 amplifier provided an analogue output suitable for computerised data acquisition. The output was connected to a 'real time' logging system. The weight of the gas bottle was logged at timed intervals under computer control and propane flow rate calculated in kg/hr.

A general arrangement of the system is given in figure 4.25 and proving ring details in figure 4.29. A more detailed description of this system is given in Appendix 4.

4.4.3. Instruments - Test measurements - mechanical

a) Air and exhaust gas pressure measurements

All gas pressure measurements (both air and exhaust gas) were taken using calibrated Budenburg bourdon type pressure gauges.

b) Air flow measurements

Air flow measurements on all stages of test rig development were conducted using a Preston meter as described by Few (9). A drawing of the Preston meter is given in figure 4.30.

c) Exhaust smoke measurements

Exhaust smoke was measured using a Bosch Smokemeter. This consisted of a sampling head fitted into the exhaust ducting. Smoke samples were obtained on filter papers and these were later analysed, using a reflective, photo-electric, evaluating unit. The positioning of the smoke probe is shown in figure 4.31., the sampling head and evaluating unit in figure 4.32. and a graph indicating acceptable smoke levels in figure 4.33.

d) Engine speed

A measurement of engine speed was available using a tachometer fitted to the dynamometer. This was used for comparison with other measurements only, the preferred measuring system being a digital readout of speed (see section 4.4.4.).

4.4.4. Instruments - test measurements - electrical/
electronic

a) Engine speed

The measurement of engine speed was made in two ways:

(i) A 127 mm (5 in) dia. steel toothed disc was fitted to the free end of the dynamometer.

An 'Orbit' magnetic pick-up was mounted adjacent to the disc and a sine wave output was obtained as the disc rotated. The output was connected to an Orbit counter/timer with a sampling period of 1 sec. A 60 tooth disc was employed, thus giving a reading on the counter/timer directly in rev/min. This arrangement was not suitable for data logging with analogue inputs, thus the following method was also used.

(ii) A further 127 mm (5 in) dia toothed disc was fitted to the free end of the dynamometer shaft. The disc had 120 teeth and the passage of the teeth was indicated by a photodiode/photo-transistor switch. The frequency output from this device was fed to an RS frequency-to-voltage convertor and then to an operational amplifier (741) via a filter, for scaling purposes. The device yielded an analogue output proportional to frequency input. A circuit diagram of this system is given in figure 4.34.

b) Turbocharger speed

An optical method for measuring turbocharger speed, as reported by Goulas and Baker (50), was considered and rejected on grounds of bulk and unreliability. The method adopted involved the use of special compressor nuts, supplied by Holset Engineering, which incorporated a small bar magnet aligned normal to the turbocharger shaft.

A pick-up coil was manufactured, having many thousands of turns of insulated copper wire wound around a soft iron former. The pick-up coil was then impregnated with an epoxy resin compound to make it sufficiently robust to withstand its operating environment. The pick-up head was mounted externally, adjacent to the compressor housing. As the turbocharger shaft rotated a frequency output was obtained which was directly proportional to turbocharger speed. The frequency output was connected to an 'Orbit' counter/timer with a sampling period of 1 sec. Turbocharger speed was indicated in rev/sec.

c) Temperature

All gas flow temperatures were measured using Chromel/Alumel thermocouples welded into stainless steel sheaths. Early tests utilised multi-way switches and a Hewlett Packard Digital Multi-meter (type 3465B) to read individual thermocouple outputs. During later tests the thermocouple outputs were read under computer control via a Solartron A210 DVM with differential input data logger coupled to a PDP 11/05 computer.

d) Brake load

Brake load was measured using a calibrated, strain gauged, cantilever manufactured in accordance with details laid down by Dorsey (49).

The cantilever system fitted to the dynamometer was very rigid and made dynamometer operation much more simple. The cantilever was bolted to the dynamometer base plate and attached to the torque arm via an actuating link incorporating swivel joints to eliminate pre-loaded bending stresses. The arrangement of the cantilever is given in figure 4.35., and the modified dynamometer is shown in figure 4.8.

The strain gauge output was connected to a Sangamo Weston C56 transducer amplifier, which provided an analogue voltage output in the range of 0 → 10v. This output was suitable for computerised data acquisition. The cantilever system is more fully described in Appendix 3.

e) Cylinder pressure

Cylinder pressure was measured using a Southern Instruments T500 inductive FM transducer. The output from the water cooled transducer was displayed on an oscilloscope

via a Southern Instruments FM unit, (type M1860), during early tests to see that the peak cylinder pressure design limits were not exceeded.

During later tests the voltage output from the FM unit was connected to a peak detect circuit feeding a scaling amplifier and a sample and hold circuit. This was controlled via TTL logic and an optical sensor. The sensor was triggered by a 127 mm (5 in.) dia. slotted disc fitted to the engine crankshaft pulley. The hold mode of the circuit was initiated just after peak cylinder pressure had occurred and then held for 4 ms. The circuit was then cleared of the previous reading and returned to the sampling mode. During the hold period the computerised data logger scan was initiated, peak cylinder pressure voltage being the first quantity to be read in the scan. A circuit diagram of the sample and hold/timing unit is given in figures 4.36. and 4.37.

f) Turbine inlet pressure

During certain later tests a piezo-electric pressure transducer was used to investigate engine exhaust pulses at the turbine inlet. A piezo type transducer was used because of its small size and good dynamic response. The transducer (AVL 12QP500C) was water-cooled and was fitted into a further water-cooled jacket situated on the transition section between the auxiliary combustion chamber and the turbine just after the engine exhaust mixing zone. The transducer output was connected to a Southern Instruments charge amplifier (type M05-100) and recorded on a Shandon Southern tungsten light recorder (type F10-650).

g) Exhaust gas analysis

Exhaust gas analysis equipment was available for only limited periods during testing. The analysis was conducted using Beckman equipment. Unburned hydrocarbons and carbon monoxide concentrations were evaluated using a Beckman HC/CO Tester, Model 590, whilst a Beckman Model 955 NO/NO_x Analyser was used for oxides of nitrogen.

5. Theoretical Analysis

5.1. Standard Engine

5.2. Preliminary Matching

5.3. Diesel Gas Turbine

5. Theoretical Analysis

In this section the theoretical analysis of the various parts of the engine system is described. Theoretical analysis of engine systems generally fall into two categories, either a quasi-steady flow treatment or a treatment using unsteady flow processes and step-by-step analysis.

The quasi-steady flow treatment (as used by Wallace (4,46)) is a simple treatment and has the advantage of giving quick results from a computer program which is of low cost in terms of processor time.

The unsteady flow process treatment requires a step-by-step analysis and permits the consideration of exhaust pulsations at the turbine. Such treatments have been used by Benson (51) and Whitehouse et al (52), utilising valves for cylinder/cooling medium heat transfer devised by Feingold (53). The pulse effect is usually accounted for by using the method of 'apparent turbine efficiency' developed by Whitehouse, Stotter and Janota (22). They have shown curves of apparent steady flow efficiency varying with expansion ratio (fig. 5.1.). Such programs are very expensive to run, taking as many as 100 secs. of computer time to simulate 5 secs. of engine time on a main frame computer (35).

Each part of the present work will be considered separately:-

- (i) Standard engine.
- (ii) Preliminary matching.
- (iii) Diesel Gas Turbine.

The equations used will be cross referenced in the relevant computer programs detailed in Appendix 1.

5.1. Standard engine

The treatment used for the standard engine theory is based on the quasi-steady flow analysis developed by Wallace (4,46),

and has been adequately reported. The main points of the analysis are as follows:-

Equations governing cycle performance

(Specific values quoted for variables refer to the particular engine used - a Rootes TS3.)

Volume of trapped air per cycle

$$V_T = \sigma \phi \frac{V_S}{1 - \frac{1}{r}} = 0.78 \times 0.9 \frac{V_S}{1 - \frac{1}{r}} \quad (1)$$

Mass of trapped air in cylinder per minute

$$W_T = \rho_m V_T N = \rho_m 0.78 \times \frac{0.9 V_S N}{1 - \frac{1}{r}} \quad (2)$$

Manifold and trapped charge density

$$\rho_m = \frac{P_c}{R T_m} = \frac{r_c P_a}{R T_m} \quad (3)$$

For constant volume combustion heat addition

$$\frac{1}{1 - \left(\frac{1}{r}\right)} (\gamma - 1) \left[\frac{1}{n_e - 1} \frac{P_{\max}}{r_c P_a} \frac{1}{r} \left[1 - \left(\frac{1}{r}\right)^{(n_e - 1)} \right] + \frac{1}{n_c - 1} \left[1 - r_c^{(n_c - 1)} \right] \right] = \frac{504 C_v}{RR_T T_m} \quad (4)$$

For dual cycle combustion, with cut-off ratio

$$\frac{1}{1 - \frac{\alpha \beta^\gamma - 1 n}{(\alpha - 1) + \gamma \alpha (\beta - 1)} \left(\frac{1}{r}\right)^{\gamma - 1}} \left[\frac{1}{n_e - 1} \frac{P_{\max}}{r_c P_a} \times \left(\frac{\beta}{r}\right) \times \left[1 - \left(\frac{\beta}{r}\right)^{(n_e - 1)} \right] + \frac{P_{\max}}{P_a r_c} \frac{1}{r} (\beta - 1) + \frac{1}{n_c - 1} \left[1 - (r)^{n_e - 1} \right] \right] = \frac{504 C_v}{RR_T T_m} \quad (5)$$

Mean temperatures

$$(T \text{ mean})_e = T_m \frac{P_{\max}}{r_c P_a} \frac{1}{2 r_e} \left[1 + \left(\frac{1}{r_e} \right)^{n_e - 1} \right] \quad (6)$$

where r_e = expansion ratio.

$$(T \text{ mean})_c = \frac{T_m}{2} \left[r^{(n_c - 1)} + 1 \right] \quad (7)$$

Power

$$HP = (HP)_i - (HP)_f - (HP)_{sb} \quad (8)$$

where

$$(HP)_i = \eta_d \left[1 - \frac{1}{r}^{(\gamma - 1)} \right] \frac{W_T}{R_T} C_V \quad (9)$$

for constant volume combustion, or,

$$(HP)_i = \eta_d \left[1 - \frac{\alpha \beta^\delta - 1}{(\alpha - 1) + \delta \alpha (\beta - 1)} \times \left(\frac{1}{r} \right)^{(\gamma - 1)} \right] \frac{W_T}{R_T} C_V \quad (10)$$

for dual cycle combustion.

$$(HP)_f = K_1 N \quad (\text{where } K_1 = 0.008 \text{ for this engine}) \quad (11)$$

$$(HP)_{sb} = K_2 (P_a r_c - P_a) D \quad (12)$$

where $K_2 = 0.00060$ for this engine and

$$D = \text{speed of scavenge blower} = 1.8N$$

Fuel flow

$$W_f = W_T / R_T \quad (13)$$

Engine mass flow

$$W = \lambda W_T \quad (14)$$

Specific fuel consumption

$$sfc = \frac{W_f \times 60}{HP} \quad (15)$$

Temperature rise across the engine

$$\Delta T = \frac{H_{\text{exh}}}{C_{\text{pl}} (\lambda W_T + W_f)} = \frac{C_v \left(1 - \eta_b - \frac{H_c}{C_v W_f} \right)}{C_{\text{pl}} (\lambda R_T + 1)} \quad (16)$$

where C_{pl} is the mean specific heat over the temperature range ΔT , and,

$$\eta_b = \frac{HP}{W_f C_v} \quad (17)$$

The computer program written for this analysis has been developed from much earlier programs originating from Wallace and is detailed in Appendix 1.1. Both the heat loss to cooling water and the C_p and λ curves for varying air/fuel ratios utilise the Wallace curves (4) shown in figures 5.2 and 5.3. The program is based on iterative procedures to realise stable values for various parameters and hence makes much use of sub-routines.

The program assumes heat addition at constant volume until a pressure limit of 120 bar (1800 psi) is reached. The remainder of the heat addition is then assumed to occur at a constant pressure. Thermal efficiency is based on the ideal constant volume or dual combustion cycle (figure 5.4.). The indicated efficiency is assumed to be 70% of the ideal. Frictional horsepower is taken to be proportional to speed.

The program assumes that the effective stroke = 0.78 x total stroke and that the trapped air fills 90% of the trapped volume. The pressure drop across the engine is assumed to be a constant, exhaust pressure being equal to 0.86 x charge pressure. The calorific value of diesel fuel is taken from Goodger (54).

A listing of the computer program is given in Appendix 1.1.

5.2. Preliminary matching

The preliminary matching theory utilises a simple steady flow, first law analysis and predicts the likely matched operating areas of an engine/turbocharger system. The computer program used (Appendix 1.2.) has been developed by Blencoe (17) from earlier work by Picken (55) and the analysis has been published elsewhere (17,18). The analysis does not define specific operating points but rather favourable areas of operation (see figure 5.5.). The analysis also provides for the use of auxiliary combustion to augment exhaust gas enthalpy and this will permit matching at lower values of exhaust blowdown temperature. The effect of this can be seen in figure 5.6.

The main points of the analysis are as follows:-

Fraction of theoretical air

$$X = \frac{1}{14.5F} \quad F = \text{fuel/air ratio} \quad (18)$$

Internal energy of exhaust gas at blowdown (after Feingold (53))

$$\begin{aligned} \bar{U}_g = & - \left(0.1755 + \frac{0.0873}{X^{0.75}} \right) \left(\frac{T_g - 32}{180} \right)^3 \\ & + \left(13.98 + \frac{6.048}{X^{0.8}} \right) \left(\frac{T_g - 32}{180} \right)^2 + \left(881.3 + \frac{83.52}{X^{0.93}} \right) X \\ & \left(\frac{T_g - 32}{180} \right) + 2282 \end{aligned} \quad (19)$$

T_g = blowdown temperature (in Btu/lb mol.)

The cylinder volume at a given crank angle is given by

$$V = \frac{\pi d^2}{4} \left(l + \frac{s}{2} (1 - \cos\theta) - \left[l^2 + \frac{s^2}{4} (\cos\theta - 1) \right]^{\frac{1}{2}} + \frac{s}{r} \right) \quad (20)$$

Specific enthalpy after compression

$$h_2 = \frac{h_2' - h_1}{\eta_c} + h_1 \quad (21)$$

Enthalpy at entry to manifold

$$h_3 = h_2 - e(h_2 - h_1) \quad (22)$$

Air manifold pressure

$$P_3 = P_a r_c \quad (23)$$

Mass of trapped charge

$$M = \frac{P_2 \times V_{ic}}{R \times T_3} \quad (24)$$

V_{ic} = volume at inlet port closure

Engine fuel requirement

$$EM = \frac{\eta \times M \times ES}{R_T} \quad (25)$$

Compressor work

$$W_c = M \times SR \times (h_2 - h_1) \quad (26)$$

for a matched condition this will be equal to the turbine work

$$W_t = W_c \quad (27)$$

Enthalpy at turbine inlet

$$h_5 = \frac{(SR - 1) \times h_3 + \bar{U}_g (1 + F)/m}{(SR + F)} \quad (28)$$

for a two-stroke, or,

$$h_5 = \frac{(SR - 1) \times h_3 + \frac{\bar{U}_g (1 - F)}{m} + \frac{P_3 \times (V_{eo} - V_{ec})}{M \times J}}{(SR + F)} \quad (29)$$

Enthalpy of exhaust gas leaving turbine

$$h_6 = h_5 - \frac{W_c}{M(SR + F)} \quad (30)$$

If the engine exhaust gas cannot supply the enthalpy required to do the necessary turbine work, then a heat addition term is used.

$$Q = \int h \times \dot{m} \times AM \times ES \quad (31)$$

The corresponding fuel requirement for auxiliary combustion is given by

$$FM = \frac{Q}{C_V} \quad (32)$$

The computer program for this analysis is listed in Appendix 1.2. Specific enthalpies are obtained from a linear interpolation of data in 'look up' tables. The data for these tables is taken from Keenan & Kaye (56). The program operates in imperial units with a section for SI unit conversion prior to printout.

5.3. Diesel gas turbine

The complexity of the diesel gas turbine system is minimised by using a simple steady flow analysis. However provision is made for the inclusion of an exhaust pulse energy term. The engine cycle analysis is as detailed in section 5.1. This analysis has proved itself during standard engine tests, the computer program providing accurate predictions and justifying its use as a section of the larger diesel gas turbine program.

Turbocharger outlet temperature

$$T_{sup} = T_A \left[1 + \frac{r_s^{\frac{\gamma-1}{\delta}} - 1}{\eta_c} \right] \quad (33)$$

Compressor work

$$HP_c = W_A C_{PA} T_A \left[\frac{r_s^{\frac{\gamma-1}{\delta}} - 1}{\eta_c} \right] \quad (34)$$

Air manifold temperature

$$T_M = T_{sup} - \epsilon_c (T_{sup} - T_c) \quad (35)$$

The analysis detailed in section 5.1. is then used to evaluate the engine operating parameters. The diesel gas turbine

analysis then continues:-

Turbocharger bearing losses

$$\text{loss} = K_3 \times N_T^2 \quad \text{where } K_3 = \text{a constant dependent} \quad (36)$$

upon turbocharger type.

Necessary turbine power

$$HP_T = HP_c + \text{loss} \quad (37)$$

Mechanical efficiency

$$\eta_{\text{mech}} = HP_c / HP_T \quad (38)$$

Turbine mass flow

$$W_T = W_A + W_f + W_{\text{face}} \quad (39)$$

Auxiliary combustion chamber mass flow

$$W_{\text{cc}} = W_A - (W_{TA} \times SR) + W_{\text{face}} \quad (40)$$

Auxiliary combustion chamber inlet temperature

$$T_{\text{ccin}} = (T_{\text{sup}} - K_4) + (e_{\text{ht}}(T_{\text{Tout}} - (T_{\text{sup}} - K_4))) \quad (41)$$

Enthalpy at turbine outlet

$$H_{\text{out}} = W_T C_{p_{T_{\text{O}}}} T_{\text{Tout}} \quad (42)$$

Enthalpy at turbine inlet

$$H_{\text{in}} = W_T C_{p_{T_{\text{in}}}} T_{\text{Tin}} \quad (43)$$

Enthalpy of engine exhaust

$$H_{\text{eng}} = W_{\text{eng}} C_{p_{\text{eng}}} T_{\text{exh}} \quad (44)$$

Enthalpy at inlet to auxiliary combustion chamber

$$H_{\text{cin}} = (W_A - (W_{TA} SR)) C_{p_{\text{cin}}} T_{\text{ccin}} \quad (45)$$

Heat input of auxiliary combustion chamber

$$Q_{\text{cc}} = (H_{\text{in}} - H_{\text{eng}}) - H_{\text{cin}} \quad (46)$$

Auxiliary combustion chamber fuelling rate

$$W_{\text{face}} = Q_{\text{cc}} / C_{v_p} \quad (47)$$

The new value for W_{facc} is used in an iterative process from equation 39 to 47 until a stable value for W_{facc} is found.

Pulse contribution

$$P = H_o + \text{turbine work} - H_{\text{in}} \quad (48)$$

Turbine efficiency

$$\eta_T = (T_{\text{Tin}} - T_{\text{Tout}'}) / (T_{\text{Tin}} - T_{\text{Tout}}) \quad (49)$$

Pulse work factor

$$\text{Fact} = P / \text{turbine work} \quad (50)$$

Turbine work factor

$$\text{ETT} = (1 / (1 - \text{Fact})) \quad (51)$$

The computer program written for this analysis utilises input data taken from actual compressor maps to determine the compressor operating points. The manufacturers maps for the compressors used are shown in figures 4.12 and 4.13 whilst the digitised versions of these maps, plotted from the input data, are shown in figures 5.7 and 5.8. The computer program then utilises the analysis detailed in section 5.1 to evaluate the engine operating conditions. The program incorporates a section to deal with outputs in the case of poor scavenge by effectively increasing the apparent air/fuel ratio. An operating point may then be determined by matching engine power output and recorded sfc levels. The program contains several iterative loops to obtain stable parameter solutions. Looping has been kept to tolerable levels via the use of looping counters to effect an escape if a divergent solution is obtained. The computer program is listed in Appendix 1.3.

- 6. Computerised Processing
 - 6.1. PDP 11/05 Logging system
 - 6.1.1. Hardware
 - 6.1.2. Software
 - 6.2. Hybrid logger + micro
 - 6.2.1. Hardware
 - 6.2.2. Software
 - 6.3. Control system
 - 6.3.1. Hardware
 - 6.3.2. Software

6. Computerised Processing

Computerised processing was used during the course of the project for two reasons:-

- 1) To ease the task of obtaining large quantities of data from a complex test rig.
- 2) A control system for the Diesel Gas Turbine, which would require feedback of the various influencing parameters, would be necessary before it could be used in a traction system.

Two data acquisition systems were used:-

- a) A real-time acquisition system.
- b) A crankshaft triggered acquisition system.

A real-time logging system was necessary for the recording of propane flow rates to the auxiliary combustion chamber (see Appendix 4). Real-time is also suitable for recording of parameters which are not dependent upon crankshaft position, e.g. mean gas temperatures measured via thermocouples.

A crankshaft triggered acquisition system is desirable for use with rotating machinery in order to effect an efficient control system. It is also necessary for the recording of parameters which are crankshaft dependent, e.g. peak cylinder pressure. The use of crankshaft triggered acquisition also permits control actions to be based on an average cycle value (say an average of 50 cycles), thus eliminating violent control reactions to an unrepresentative data point, whilst still maintaining an adequate speed of response.

An acquisition system to log 20 channels of input at each crankshaft revolution requires a scan rate of at least 1 kHz for a two-stroke engine operating at 3000 rev/min. The available PDP 11/05 computer utilised an acquisition system capable of recording 10 input

channels at a maximum rate of 100 Hz and was thus not suited for crankshaft triggered operation. Accordingly, a data logger was constructed having 16 analogue input channels and a scan rate of ≈ 30 kHz. This logger was linked to a microprocessor and would allow for the development of a flexible control system.

6.1. PDP 11/05 Logging system

The Solartron DVM and logging system utilised by the PDP 11/05 had 10 differential analogue input channels with a scan rate of 10 Hz for a single channel or 100 Hz for all 10 channels.

6.1.1. Hardware

The logging unit utilised a Solartron A210 DVM. The unit had 10 analogue input channels and logging (including choice of voltage range) was program controlled. The differential inputs rendered the unit ideal for recording thermocouple outputs from a number of sheathed thermocouples having their sheaths in electrical contact (a common earth system tends to average the thermocouple outputs). As the computer had a real-time clock facility the logging system was ideal for recording weight loss of a propane bottle over an accurate period of time.

6.1.2. Software

The logging program written for the PDP 11/05 computer allowed a choice of sampling times from 1 to 15 minutes. Values of up to 9 thermocouple readings and propane bottle weight were logged at time zero and approximately one minute intervals thereafter. The internal computer clock timed the intervals to 0.01 sec., such that successive propane bottle weight readings could be adjusted to represent accurate one minute weight losses. The program gives

gas flow rate results based on the average of the one minute values and another based on the weight loss during the total sampling period.

Temperatures are presented as an average of the readings taken at one minute intervals. The output from the program is given on a teletype with an option for punch tape output from a high speed punch if desired. The tape output permits later computer analysis of the data if required.

A listing of the logging program, written in BASAC IV, is given in Appendix 1.4., along with a typical example of data output.

6.2. Hybrid data logger with microprocessor

The data acquisition system linked to the microprocessor had 16 single ended input channels. The system was constructed during the course of the project.

6.2.1. Hardware

The system consisted of a 16 channel multiplexer (Hybrid Systems MUX 204, figure 6.1.) linked to an analogue to digital convertor (Hybrid Systems ADC 550-12E, figure 6.2.). These elements were linked to associated control logic and a microprocessor (Quarndon QMS 8080). The system takes analogue voltages (0-10v) and converts them into 12 bit binary representation for use in the microprocessor. The channel to be read is selected via a 4 bit binary address set up by the control logic. The data acquisition system is shown in figure 6.3. and the system organisation in figure 6.4.

The control logic (figure 6.5.) sets up the multi-

plexer address and increments the address after each channel has been read to give 'scan' capability. The number of channels to be scanned (1-16) is set on thumb-wheel switches on the front of the logger. At the end of each scan the logging system is 'de-activated' until a further signal from a crankshaft trigger initiates a new scan. The crankshaft trigger alone cannot generate a scan unless it occurs in conjunction with a software generated command from the microprocessor, and a valid multiplexer address (VMA).

6.2.2. Software

Although the analogue to digital convertor is a 12 bit device, only the most significant 8 bits are used in the microprocessor. The processor is an 8 bit device and thus the use of 8 bits simplifies the system architecture and is less demanding on storage space.

The software serves a number of functions.

- (i) It controls and monitors the data acquisition system, such that the correct number of analogue channels are sampled.
- (ii) It determines the number of times that the analogue inputs will be scanned (i.e. the number of engine revolutions for which measurements will be taken).
- (iii) It stores the measured values in a table as a series of 8 digit binary numbers.
- (iv) When all the required measurements are completed, it converts the stored values to an equivalent voltage and prints the numbers on a teletype.

A listing of the computer program (in assembly

language for the Intel 8080 processor) is given in Appendix 1.5.

6.3. Control system

The control system is based on stepper motors and a microprocessor.

6.3.1. Hardware

The stepper motors (Evershed & Vignoles FDS4/A51 and FDM4/A52) were driven by a power supply and Digicard 053 drive units (figure 6.6.). Forcing resistors were used to maintain good torque characteristics. The motors were all of the 400 step/rev type such that a positional accuracy of 0.9° could be maintained. Several motors and drive systems were combined to provide a comprehensive control system enabling up to six stepper motors to be driven simultaneously (six channel control). Each channel could be driven manually (using signals internally derived from the drive system) or automatically (using signals from a microprocessor). The circuit for such a system is shown, for a single channel, in figure 6.7.

6.3.2. Software

Only limited software has been written for a Texas 9900 microprocessor to drive stepper motors under speed control and positional control. These programs provide the necessary routines to enable motors to be accelerated to a high running speed and then decelerated to a final position in order to keep control system response time delays to a minimum. The computer programs (written in assembly language for the Texas 9900 processor) are listed in Appendix 1.6.

7. Experimental Procedure
 - 7.1. Standard engine tests
 - 7.2. Stage I development
 - 7.3. Stage II development
 - 7.4. Stage III development
 - 7.5. Calibrations
 - 7.5.1. Pressure gauges
 - 7.5.2. Pressure transducers
 - 7.5.3. Diesel fuel oil flow
 - 7.5.4. Transducer for applied brake load
 - 7.5.5. Thermocouples
 - 7.5.6. Preston meter
 - 7.5.7. Propane flow measurement
 - 7.5.8. Engine speed measurement

7. Experimental Procedure

The experimental programme was divided into four sections:-

- 1) Standard engine tests.
- 2) Stage I development. The incorporation of an exhaust gas turbocharger.
- 3) Stage II development. Turbocharged engine incorporating an auxiliary combustion chamber to augment exhaust gas enthalpy.
- 4) Stage III development (The diesel gas turbine).

Auxiliary combustion utilising only clean by-pass air.

Each stage of development is discussed separately below.

7.1. Standard engine tests

These tests involved running the engine, in its standard form, over the full range of speed and load. From the results of these tests, standard engine performance characteristics were established. These characteristics formed the basis for the comparison of results from subsequent modifications.

The engine was started and run under a light load for 15 minutes to allow it to reach normal operating temperature. The engine fuelling and brake load were then increased until the fuel pump rack was in its maximum position and an engine speed of 1000 rev/min was attained. Results were then taken at this point and brake load was decreased until a new test point was reached. Results were taken at engine speed increments of 200 rev/min up to a maximum of 2400 rev/min. The tests were then repeated and results obtained for $\frac{2}{3}$ and $\frac{1}{3}$ of full load conditions.

The steady state parameters recorded were:-

Engine speed

Air flow

Inlet manifold temperature and pressure
Exhaust manifold temperature and pressure
Dynamometer applied load
Peak cylinder pressure
Diesel fuel oil consumption

The results of these tests are presented graphically in Chapter 8.

7.2. Stage I development

The Stage I system (figure 4.18) incorporated an exhaust gas turbocharger in series with the scavenge blower. A Holset 3LD1 turbocharger was used with turbine housings of 1.57 sq.in. and 2.5 sq.in. throat area.

The test procedure adopted was as that described in section 7.1. Results were taken only at full load conditions. Once results had been obtained with the turbocharger and scavenge blower in series, the connection between the scavenge blower and turbocharger was removed with the engine running at approximately 2000 rev/min. under full load conditions to allow freely turbo-charged (i.e. no scavenge blower assistance) results to be obtained.

The use of the scavenge blower in series with the turbocharger was necessary for engine starting.

The steady state parameters that were recorded were:-

Engine speed
Air flow
Inlet manifold temperature and pressure
Exhaust manifold temperature and pressure
Temperature and pressure between scavenge blower and turbo-compressor

Turbine inlet temperature and pressure
Turbine exhaust temperature and pressure
Turbocharger speed
Applied dynamometer load
Peak cylinder pressure
Exhaust smoke
Diesel fuel oil consumption

The results of these tests are shown graphically in Chapter 8.

7.3. Stage II development

Stage II development (figure 4.20a) included the incorporation of an auxiliary combustion chamber in the exhaust duct between the engine exhaust manifold and the turbocharger turbine. The purpose of the auxiliary combustion chamber was to augment the exhaust gas enthalpy and thus increase turbocharger work.

The engine was started in the normal manner and when it was running satisfactorily, the auxiliary combustion chamber ignition was initiated and its propane fuel supply turned on. Auxiliary combustion could be maintained throughout the full range of engine speed at no-load conditions. Any attempt to increase loading resulted in flame instability and the extinguishing of auxiliary combustion.

A further modification (figure 4.20b) was then introduced. A throttled air bypass link was placed between the turbo-compressor outlet and the auxiliary combustion chamber inlet. Thus a proportion of the air could bypass the engine, being mixed with the engine exhaust gas prior to auxiliary combustion. With this arrangement starting was possible in two modes:-

1) The engine, with the scavenge blower and turbocharger in series, was started in the normal manner. When the engine was running satisfactorily the ignition for the auxiliary combustion chamber was turned on and fuelling was initiated. It was then possible to remove the link (air) between the turbocharger and scavenge blower if desired.

2) With the scavenge blower disconnected from the air supply circuit (i.e. motoring unloaded), an additional air supply was connected to the inlet of the auxiliary combustion chamber. The bypass air throttle was fully opened and the slave air supply turned on. Auxiliary combustion was started, which, together with the turbocharger, acted as a gas turbine. When self-sustained conditions were reached, the slave air supply was turned off and disconnected. The air bypass throttle was then closed until a small, positive, pressure drop was obtained across the engine. The engine was then started with the conventional starter motor.

Both starting systems proved reliable and load could be sustained up to approximately $\frac{1}{3}$ of full load conditions, throughout the full range of engine speed, before the onset of flame instability. The second mode of operation permitted the removal of the scavenge blower as it became redundant.

The parameters monitored were as those in section 7.2. Few formal results are presented in Chapter 8 as the maximum power operating points were unacceptable. The tests were carried out using a Holset 3LD1 turbocharger with a 1.57 sq.in. throat area turbine housing. The experience gained during this phase of testing was used to design a final operating system (the Diesel Gas Turbine).

7.4. Stage III development

The Stage III development (figure 4.21.) included removing the scavenge blower (and substituting a blanking plate and a fabricated air manifold). A controlled throttle air bypass was taken from the turbocompressor outlet to the auxiliary combustion chamber inlet. The engine exhaust duct was connected to the auxiliary combustion circuit just after the auxiliary combustion chamber and prior to the turbine inlet. Hence auxiliary combustion utilised only clean, constant pressure, air. Exhaust pulse energy was attenuated, to some extent, by the length of exhaust ducting ($\approx 1\text{m}$) but the remaining pulse energy was available for utilisation in the turbine.

The tests involved running the engine over as wide an operating range as possible, with a given turbocharger. Tests commenced at the lowest rated engine speed (1000 rev/min) and were increased in 200 rev/min. steps until the engine power output had fallen below that pertaining to the standard engine at that speed. Tests were also carried out at a constant engine speed of 1200 rev/min. with charge inlet pressure being varied. This was achieved by varying the auxiliary fuel input rate.

The test rig starting procedure is detailed in Appendix 9.

Tests were conducted at full load conditions. In this case the apparent air/fuel ratios were in the range from 24:1 \rightarrow 33:1 due to the nature of the injector pump employed (fixed delivery). Part load operating conditions were not investigated as engine fuelling rate was not found to have any significant effect on operating conditions other than auxiliary fuelling rates and specific fuel consumption.

Tests were carried out using a Holset 3LD1 turbocharger with

turbine housings of 1.41, 1.57 and 2.5 sq.in. throat areas. Further tests were carried out using a Holset 4LGK turbocharger with a 2.6 sq.in. throat area turbine housing.

In the tests with the Holset 3LD1 turbocharger using a 1.41 sq.in. throat area turbine housing, the initial start-up procedures could be followed and self-sustained gas turbine conditions reached, but any attempt to start the engine drove the compressor into surge. This showed that the turbocharger was too small.

Steady state parameters monitored during the tests were:-

- Turbo-compressor outlet temperature and pressure
- Engine air inlet temperature and pressure
- Bypass air temperature and pressure
- Engine exhaust temperature and pressure (at exhaust manifold)
- Engine exhaust temperature and pressure (at exhaust/ auxiliary combustion gases mixing zone)
- Auxiliary combustion chamber inlet temperature and pressure
- Turbine inlet temperature and pressure
- Turbine exhaust temperature and pressure
- Air flow rate
- Diesel oil flow rate
- Propane supply pressure
- Propane flow rate
- Dynamometer applied load
- Peak cylinder pressure
- Engine speed

Turbocharger speed

Exhaust emissions

Results from these tests are presented graphically in Chapter 8.

7.5. Calibrations

During the course of test rig development there were several items of both standard, and manufactured, transducers and associated amplifiers that required calibration. The calibration of all flowmeters was carried out in accordance with rules laid out by Hayward (57). The calibration procedure for each item is discussed below.

7.5.1. Pressure gauges

All pressure gauges used were Bourdon types manufactured by Budenburg. The calibration of these gauges was checked using a Barnett Industrial Deadweight Tester (model 3760/74).

7.5.2. Pressure transducers

a) Cylinder pressure transducer

The transducer used to monitor cylinder pressure was a Southern Instruments T500/H524 inductive FM type. The transducer and FM unit (Southern Instruments type M1860) were calibrated using a Barnett Industrial Deadweight Tester (model 3760/74) and a Hewlett Packard HP356A Digital Multi-meter. The digital multi-meter was of the dual ramp integration type, and, as such, its response was not sufficiently fast to read cylinder pressure traces, so test results were taken using an oscilloscope (Tektronix type 564B) or logged via the peak detect/sample and hold module described

in section 4.4.4. The calibration is given in Table 7.1. and shown graphically in figure 7.1.

b) Pulse measurement transducer

The transducer used to investigate exhaust pulses at the turbine inlet was a piezoelectric type pressure transducer (AVL 12QP500C No.1074) and was used in conjunction with a Southern Instruments (MO5 100) charge amplifier and a Shandon Southern Tungsten Light Recorder (type F10-650). Calibration was carried out using both the light recorder and a Hewlett Packard 3465B Digital Multi-meter, with a Barnett Industrial Deadweight Tester (model 3760/74).

7.5.3. Diesel fuel oil flow transducer

This instrument consisted of a Litre Meter LM25GN transducer and a voltage readout unit, designed and constructed at Leicester Polytechnic during the course of the project. The system is described in detail in Appendix 2. The transducer was a rotor type operating on a volumetric principle, delivering a frequency output proportional to volumetric flow rate. The device was found to be very sensitive to delivery pressure thus necessitating a 'constant head' feed which was provided from a reservoir and recirculatory supply system.

The system was calibrated with regard to instructions laid down by Hayward (57), using a weighed amount of diesel oil passed in a given time. The fuel was weighed to 1 oz. (28.5g) on an Avery scale (type 3205ABA) and the period was timed to 1 second using a Smith's stop clock. The output voltage was read on a Hewlett Packard HP3465A

Digital Multi-meter. Fuel was allowed to flow for a variable period of 10 to 30 minutes depending upon flow-rate. A flying start and finish technique was used.

During the calibration period there was a variation in atmospheric pressure of 20 mmHg and in atmospheric temperature of 7°C. These factors were found to have no significant effect within the stated accuracy of $\pm 1.5\%$ of full scale. The calibration figures are given in Tables 7.2. and 7.3. and shown graphically in figure 7.2.

7.5.4. Transducer for applied brake load

Applied brake load was measured using a strain gauged cantilever transducer and a Sangamo Weston C56 transducer amplifier. Voltage output was measured on a Hewlett Packard DMM (type 3465A) or logged on the systems described in Chapter 6

The cantilever system (see Appendix 3) was designed and manufactured during the course of the project and was subjected to rigorous testing and calibration procedures. The cantilever was tested for linearity and calibrated using a Dartec tensile test machine and a Bryans Southern Instruments X-Y Plotter (type 2000). This calibration is shown in figure 7.3. The cantilever was tested using deadweights over a period of 12 hours to investigate strain gauge adhesive creep effects during extended service. This effect was found to be indistinguishable from amplifier 'drift'. The cantilever system was given a final 'in situ' calibration on the brake, using the brake's spring balance to apply the load. This calibration is shown in Table 7.4. and figure 7.4.

7.5.5. Thermocouples

All the thermocouples used were of the standard Chromel/Alumel type. A calibration chart is shown in figure 7.5. The thermocouples used were checked against this chart in a temperature controlled oven, used for metallurgical purposes. The thermocouples used all complied with the chart.

7.5.6. Preston meter

A Preston Meter (9) was used to measure air flow at all stages of rig development. The Preston Meter was constructed using perspex tube and hypodermic tubing. Calibration was carried out on a variable velocity air flow rig, using a water manometer and pilot traverse techniques. The calibration is given in Table 7.5. and figure 7.6.

7.5.7. Propane flow measurement

Several gas and liquid flow measuring devices were investigated and rejected on the grounds of poor accuracy or unsuitability (see Appendix 4). The method used was a gravimetric system using a strain gauged proving ring. The proving ring was calibrated using deadweights and a Sangamo Weston C56 transducer amplifier. The voltage output was read on a Hewlett Packard Digital Multi-meter (type 3465B). The proving ring was subjected to linearity and strain gauge adhesive creep checks demonstrating good linearity and undetectable creep. The calibration of the proving ring is given in Table 7.6. and figure 7.7.

Results during engine testing were obtained by data logging the proving ring output at fixed time intervals,

under computer control, and deriving the flow rate.

The system is fully described in Appendix 4.

7.5.8. Engine speed measurement

A tachogenerator was used to give an analogue voltage output proportional to speed during the early tests but the drive system proved unreliable due to vibration or 'wind-up'. The method finally adopted used a toothed disc/opto switch sensor in conjunction with a frequency to voltage convertor filter and amplifier. The device was designed and constructed during the course of the project. Calibration was carried out using a Farnell Instruments Signal Generator to simulate the input from the optical sensor. The frequency was checked on an Orbit counter/timer. The voltage output was read on a Hewlett Packard 3465A DMM. The filter used in the circuit, to reduce A.C. ripple to acceptable levels, caused the output to be non-linear over an equivalent speed range of 0-3000 rev/min. (0-6 kHz). Linearity was, however, acceptable over an equivalent speed range of 600 to 2400 rev/min. (1.2-4.8 kHz) which covered the range from idling speed to full rated engine speed. The calibration of this device is given in Table 7.7. and presented graphically in figure 7.8.

8. Results

8.1. Experimental

- 8.1.1. Standard engine
- 8.1.2. Stage I and II development
- 8.1.3. Stage III development

8.2. Theoretical

- 8.2.1. Performance predictions of the standard engine
- 8.2.2. Preliminary matching
- 8.2.3. Performance predictions of the diesel gas turbine

8.3. Comparison of results

- 8.3.1. Standard engine
- 8.3.2. Preliminary matching
- 8.3.3. Diesel gas turbine

8.4. Extension of experimental work by theoretical study

8. Results

The results are presented as follows:-

- 1) Presentation of experimental results:
 - (i) Results from the standard engine
 - (ii) Results from the Stage I and II developments
 - (iii) Results from the diesel gas turbine
- 2) Presentation of results obtained from computer prediction programs:
 - (i) Standard engine program
 - (ii) Preliminary matching program
 - (iii) Diesel gas turbine program
- 3) Comparison of the theoretical and experimental results:
 - (i) Standard engine
 - (ii) Preliminary matching
 - (iii) Diesel gas turbine
- 4) Extension of experimental work by theoretical study :
Diesel gas turbine only.

The results obtained from the various stages of the experimental programme, together with those from computer predictions, are presented in graphical form. Boost ratios, scavenge ratios and smoke numbers pertaining to each stage of development are presented in Tables 8.1. (variable speed tests) and 8.2. (constant speed tests).

8.1. Presentation of experimental results

8.1.1. Results from the standard engine

The standard engine was tested over a range of speeds and loads as discussed in Section 7.1. Full load results for power output, torque and specific fuel consumption are given in figures 8.1. and 8.2. These results compare favourably with those published by the engine manufacturers.

Results for power output, torque and specific fuel consumption for conditions of $\frac{1}{3}$ full load, $\frac{2}{3}$ full load and full load are given in figures 8.3., 8.4. and 8.5.

8.1.2. Stage I and II developments

The engine system was modified to include a turbocharger in series with the scavenge blower (figure 4.18) and tests were carried out as described in section 7.2. The tests were undertaken using a Holset 3LD1 turbocharger with two different turbine housings. No freely turbocharged results (without scavenge blower assistance) could be obtained using a Holset 3LD1 turbocharger with a 2.5 sq.in. throat area turbine housing (see figure 4.19). The same turbocharger with a 1.57 sq.in. throat area turbine housing produced freely turbocharged results over a narrow speed range at full load. Results of power output, torque and specific fuel consumption are shown in figures 8.6. and 8.7.

The Stage II development (figure 4.20a.) included the use of an auxiliary combustion chamber. The combustion chamber operated satisfactorily at all engine speeds under no load conditions. Attempts to apply load resulted in flame instability and the extinguishing of auxiliary combustion. Results of charge air density (figure 8.8.) and trapped air flow (figure 8.9.) compare developments I and II with the naturally aspirated engine.

The engine system was modified to mix bypass air with the engine exhaust before entering the auxiliary combustion chamber (figure 4.20b.). It was found that auxiliary combustion could be maintained at all engine speeds up to

approximately $\frac{1}{3}$ of full load conditions. Attempts to increase loading beyond this point resulted in flame instability and the extinguishing of auxiliary combustion. Accordingly the engine system was considered to be unviable and no graphical results are presented.

8.1.3. Stage III development

The engine system was modified such that auxiliary combustion utilised only clean, constant pressure, air (figure 4.21.). The hot gases from auxiliary combustion were mixed with engine exhaust gases just before entering the turbine.

Tests were carried out with four different turbocharger types:-

- 1) A Holset 3LD1 with a 1.41 sq.in. throat area turbine housing.
- 2) A Holset 3LD1 with a 1.57 sq.in. throat area turbine housing.
- 3) A Holset 3LD1 with a 2.5 sq.in. throat area turbine housing.
- 4) A Holset 4LGK with a 2.6 sq.in. throat area turbine housing.

Tests were conducted in two ways:-

- a) With increasing engine speed at full load using each turbocharger. Tests were continued until the system power output fell to that pertaining to the standard engine at that speed. After this point the tests were discontinued as the engine system was not likely to prove viable if the power output was below that of the standard engine.

b) At a constant engine speed (1200 rev/min). The boost level was varied by adjusting the fuelling rate of the auxiliary combustion chamber. This set of tests permitted comparisons between the various turbocharger types employed.

Results of power output, torque, specific fuel consumption, apparent air/fuel ratio, trapped and total air flows, propane consumption, smoke emission, air manifold temperature and b.m.e.p., for the variable engine speed tests are shown in figures 8.10 to 8.15. The operating points of the various turbochargers are shown in figures 8.16 to 8.18. No results have been presented for the Holset 3LD1 with a 1.41 sq.in. throat area turbine housing due to difficulties encountered with compressor surge. These difficulties set the smallest turbocharger limit as the Holset 3LD1 1.57.

Results of power output, torque, specific fuel consumption, apparent air/fuel ratio, air manifold temperature, total and trapped air flows, smoke emission, charge air density, propane consumption and b.m.e.p., for the constant speed tests are shown in figures 8.19 to 8.24. The operating points of the various turbochargers are shown in figures 8.25. to 8.27.

8.2. Theoretical results

Theoretical predictions of engine performance are shown (figures 8.28. to 8.51.). Each section demonstrates the effects of variations in the program input parameters.

8.2.1. Performance predictions of the standard engine

The effects of variations of boost ratio, scavenge ratio, air/fuel ratio, and air manifold temperature on the

program output parameters (power output, specific fuel consumption, exhaust temperature and brake thermal efficiency) at a constant engine speed of 1200 rev/min. are shown in figures 8.28. to 8.33.

The program was also used to predict the effects of variations of scavenge ratio, boost ratio, air/fuel ratio and air manifold temperature on power output and exhaust temperature over the whole range of engine speed. These results are shown in figures 8.34. to 8.37.

8.2.2. Preliminary matching

Results obtained from the preliminary matching program show areas in which turbocharged operation of the engine is feasible and not specific operating points (figures 8.38 to 8.46). The matching aid graphs (figs. 8.43 - 8.46) show curves for exhaust pressure ratio (EPR), inlet manifold pressure (IM), turbine inlet temperature (TI) and engine exhaust temperature (TEX) all with reference to exhaust blowdown temperature. Matched operation is possible in regions where the exhaust pressure ratio (EPR) falls below the inlet manifold pressure (IM). The turbine inlet temperature and engine exhaust temperatures demonstrate whether matched operation is feasible whilst remaining within thermal loading limits and whether auxiliary combustion is desirable.

The preliminary matching program was used to investigate the effects of boost ratio, scavenge ratio and exhaust blowdown temperatures on engine and auxiliary combustion fuel consumption. These results are shown in figures

8.38 to 8.42. Figures 8.43 to 8.46 show matching aid graphs for various turbine and compressor efficiencies, including the use of auxiliary combustion.

8.2.3. Performance prediction of the diesel gas turbine

The effects of changes of the program input parameters, (air/fuel ratio, scavenge ratio, boost ratio and air flow) on the program output parameters (power output, propane consumption, engine exhaust temperature, air manifold temperature and compressor efficiency) were examined at a constant engine speed of 1400 rev/min (figures 8.47 to 8.51).

8.3. Comparison of theoretical and experimental results

The various computer prediction programs were run using experimental data for the input parameters to permit comparisons of the experimental and theoretical results and to validate the programs.

8.3.1. Standard engine

As the standard engine program was to be used as a section of the diesel gas turbine program, it was essential that the predictions should be accurate. The frictional horsepower correction factor was thus adjusted accordingly. Comparisons of engine power output, torque and specific fuel consumption are given in figures 8.52 and 8.53.

8.3.2. Preliminary matching

As previously mentioned, the preliminary matching program is only an aid to indicate feasible areas of matched operation, rather than specific operating points. For this reason, it is not possible to show direct comparisons with experimental results. The importance of

turbine and compressor isentropic efficiencies for matched operation is shown in figures 8.54, 8.55 and 8.56.

8.3.3. Diesel gas turbine

As the diesel gas turbine program was to be used for a theoretical extension of the experimental work, the program had to be well validated. Experimental results were obtained with three different turbocharger configurations at both constant engine speed (1200 rev/min) and variable engine speed.

a) Constant engine speed

The comparisons at constant engine speed are all plotted with reference to boost ratio. Comparisons are shown for the Holset 3LD1 turbocharger with a turbine of 1.57 sq.in. throat area. Comparisons of power output, engine specific fuel consumption, air manifold temperature, air/fuel ratio, propane consumption and total system specific fuel consumption are given in figures 8.57 to 8.59. Reasons for discrepancies in the air/fuel ratio comparison (figure 8.58) are given in Chapter 9.

Similar comparisons for the Holset 3LD1 with a turbine housing of 2.5 sq.in. throat area are given in figures 8.60 to 8.62 and for the Holset 4LGK with a turbine housing of 2.6 sq.in. throat area in figures 8.63 to 8.65.

The program for performance prediction contained a section to predict the contribution of engine exhaust pulse energy to turbine work which is discussed in Chapter 9.

b) Variable engine speed

Comparison for the tests using a Holset 3LD1 turbo-charger with a turbine housing of 1.57 sq.in. throat area are shown of power output, engine specific fuel consumption, air/fuel ratio, air manifold temperature and total system specific fuel consumption. These comparisons are given in figures 8.66 to 8.68.

The comparison of air/fuel ratio (figure 8.67) shows some discrepancies. The reason for these discrepancies is given in Chapter 9. Similar comparisons for the Holset 3LD1 with a turbine housing of 2.5 sq.in. throat area are given in figures 8.69 to 8.71 and for the Holset 4LGK with a turbine housing of 2.6 sq.in. throat area in figures 8.72 to 8.74.

8.4. Use of the computer program to extend the scope of test results

The diesel gas turbine program for predictions displayed good agreement with experimental results. Accordingly it was used to investigate the performance of the diesel gas turbine in regions for which no experimental results were available. These investigations include the use of 20:1 air/fuel ratios, charge cooling and heat exchange. The program input data were chosen using representative values obtained during experimental testing. This is discussed more fully in section 9. Results for the engine using the various turbocharger types are shown in figures 8.75 to 8.80.

An investigation was carried out on the use of the diesel gas turbine for obtaining good torque 'back-up' characteristics. The compressor operating lines are shown in figures 8.81 and 8.82. The results of the investigation are given in figures 8.83 to 8.85.

9. Discussion

9.1. Experimental work

9.1.1. Standard engine tests

9.1.2. Stage I and II developments

9.1.3. Stage III development

9.1.3.1. Variable speed tests

9.1.3.2. Constant speed tests

9.2. Theoretical work (Computer predictions)

9.2.1. Standard engine program

9.2.2. Preliminary matching program

9.2.3. Diesel gas turbine program

9.3. Comparison of experimental and theoretical results

9.3.1. Standard engine

9.3.2. Preliminary matching

9.3.3. Diesel gas turbine

9.4. Extension of the scope of the programme by computer
predictions

9.4.1. Diesel gas turbine

9.5. General comments

9.6. Differences between the diesel gas turbine and the
Hyperbar system

9. Discussion

9.1. Experimental work

9.1.1. Standard engine

The standard engine was tested at full load, $\frac{3}{4}$ full load and $\frac{1}{2}$ full load. The full load performance (figures 8.1, 8.2) compares favourably with the manufacturer's published figures. The part load results (figures 8.3 to 8.5) demonstrate the poor part-load fuel economy. A reduction in load to $\frac{2}{3}$ of full load results in a modest rise in specific fuel consumption of about 3%, whereas a reduction to $\frac{1}{2}$ full load results in an increase of specific fuel consumption from 8% to 45%. Such a fall in fuel economy is to be expected as the mechanical losses form a greater percentage of the engine output (especially the power loss to the scavenge blower) and a proportionally greater amount of heat is lost to the coolant at low b.m.e.p.s. These tests established a base line by which further engine developments could be assessed.

9.1.2. Stage I and II developments

The Stage I development (figure 4.18) incorporated an exhaust gas turbocharger, in series with the mechanical blower. This system configuration demonstrates a useful power increase over most of the speed range (figure 8.6). The maximum increases occur at 1600 rev/min (8% for the 2.5 sq.in. housing and 10% for the 1.57 sq.in. housing.

Freely turbocharged results, using the 1.57 sq.in. housing, were obtained over a limited speed range (1650-2000 rev/min) and demonstrate a maximum power increase of some 4% at 1800 rev/min (figure 8.6). The speed range

could not be extended due to high exhaust temperatures (in excess of 650°C). The improvements discussed above are also reflected in the torque and specific fuel consumption results (figure 8.7). It was not possible to operate the system satisfactorily with larger turbine housings. The Stage II development incorporated auxiliary combustion (figure 4.20a). Auxiliary combustion was maintained throughout the full range of engine speed in a no-load condition. Any attempt to load the engine resulted in flame instability and the termination of auxiliary combustion (viewed through a quartz 'window' in the combustion chamber). Figures 8.8 and 8.9 show improvements of charge air density and trapped air flow throughout the speed range. The most significant improvements occur at an engine speed of 1600 rev/min (20% for the 2.5 sq.in. housing, 28% for the 1.57 sq.in. housing and 42% for the 1.57 sq.in. housing with auxiliary combustion). These improvements in air flow demonstrate a potential for increased power output. The system which would lead to the highest output would include a turbocharger, an auxiliary combustion chamber, and no mechanical scavenge blower.

With the Stage II development, it was found that auxiliary combustion could not be maintained even at light loads. Flame instability was

caused by engine exhaust pulsations reaching the primary (burning) zone in the auxiliary combustion chamber. A modified system was thus proposed (figure 4.20b), in which bypass air, controlled by a throttling valve, was mixed with the engine exhaust gas, upstream of the auxiliary combustion chamber. The benefits of the bypass were threefold:-

- 1) The constant pressure bypass air would reduce the effect of engine exhaust pulsations, thus permitting extended auxiliary combustion.
- 2) Excess air would be available for auxiliary combustion at low engine air/fuel ratios.
- 3) A bypass would permit the use of the turbocharger and auxiliary combustion chamber as a self-sustained gas turbine, allowing an alternative start-up procedure.

Testing of the modified system showed that the range of auxiliary combustion had been extended (up to $\frac{1}{3}$ full load condition) and the use of a modified starting procedure made the scavenge blower unnecessary. The low power outputs achieved, however, made the system unviable, but the experience gained led to the adoption of the final system. This system (figure 4.21) utilised only clean, constant pressure, bypass air for auxiliary combustion.

The scavenge blower was also removed and a blanking plate and fabricated air manifold were fitted in its place.

9.1.3. Stage III development

In this section the major experimental work is considered. Tests were conducted with various turbochargers over a range of engine speeds. They were not carried out

in regions of low power output where diesel gas turbine operation was impossible. Tests were also carried out at a constant engine speed of 1200 rev/min with variable boost (obtained by adjustment of the fuelling rate to the auxiliary combustion chamber).

9.1.3.1. Variable speed tests

Improvements in power output were obtained (figure 8.10). The range and level of improvement is affected by the size of the turbine employed. The larger the turbine throat area, the greater the range and level of improvement. The maximum improvements are generally obtained at 1400 rev/min (8%, 11% and 16%, with increasing turbine size). Similar improvements can be seen in torque and engine specific fuel consumption (figure 8.11). Due to the nature of the injection pump the fuel delivery rate (at full load conditions) is fixed for a given speed. The improvements in power output are thus due to the removal of the scavenge blower and improved combustion due to higher manifold air temperatures. The potential for improvement must be evaluated by examining the trapped air flow and air/fuel ratio (figure 8.12). The various turbochargers deliver similar trapped air flows, hence similar air/fuel ratios prevail. Increases in air/fuel ratio of between 29% and 50% are shown with peak levels of 33:1. Larger turbine housings deliver greater total air flows (figure 8.13).

As this extra air flow is diverted to the auxiliary combustion chamber, it would be expected that larger turbine housings have a greater auxiliary fuelling requirement. This is borne out by the gas (propane) consumption curves and the total system specific fuel consumption (figures 8.13, 8.14). Smoke emission (figure 8.14) increases with increasing engine speed, smaller turbines being more sensitive to speed changes. Since the air/fuel ratios are high, the deterioration in combustion must be caused by inadequate scavenge. Smaller turbines produce greater 'back-up' pressures due to earlier 'choking' which results in the onset of poor scavenge at lower engine speeds. The air/fuel ratios presented in figure 8.12 must, therefore, be considered as 'apparent' values.

Smaller turbochargers give higher air manifold temperatures (figure 8.15) due to the higher boost levels generated. The curtailment of adequate scavenge is reflected in the b.m.e.p. curves (figure 8.15). The compressor operating lines (figures 8.16 to 8.18) demonstrate that the compressors are operating in high efficiency zones. The use of a larger turbine housing on the Holset 3LD1 results in increased air flow at lower efficiency. Table 8.1. shows the boost and scavenge ratios used during the tests. Boost levels may be increased by adjusting the

auxiliary fuelling rate, nevertheless it can be seen that smaller turbine housings tend to produce higher boost levels at a given engine speed.

9.1.3.2. Constant speed tests

Tests were conducted at a constant engine speed of 1200 rev/min with variable boost. These tests permitted more detailed investigations into the effect of turbocharger size.

Higher power outputs (hence lower engine specific fuel consumption) are obtained with larger turbochargers (fig. 8.19). The larger turbochargers operate in more efficient regions and this is reflected in lower air manifold temperatures and increased apparent air/fuel ratios (fig. 8.20). It may be noted that, although two turbochargers display similar air/fuel ratios, the larger turbocharger generates an increased torque output (fig. 8.21). This effect may be due to smaller turbochargers imposing an increased back pressure on the engine, leading to an earlier onset of poor scavenge. Trapped air flow, at a given boost level, is increased by the use of larger turbochargers as is total air flow (fig. 8.22). The specific fuel consumption of the total system (fig. 8.23) decreases with boost suggesting optimum values for lower boost ratios (≈ 1.5), smaller turbochargers yielding a lower specific fuel consumption.

Exhaust smoke emission (fig. 8.23) displays a rising trend with falling boost, suggesting that higher boost levels give better scavenge, or pressure wave reflections which can lead to higher charge densities, for a given turbocharger. Smaller turbochargers display increased smoke

9.2. Theoretical work

9.2.1. Standard engine program

The program (see Appendix 1) was executed by varying propane consumption in the auxiliary combustion

chamber increases with boost level for a given turbocharger (fig. 8.24). Larger turbochargers

lead to increased propane consumption for a given

boost level. The relevant compressor operating lines (figs. 8.25 - 8.27) show that larger turbochargers permit increased air flows but may result in a fall of isentropic efficiency. The effects of varying boost ratio and scavenge ratio are demonstrated in figure 8.29, effects of varying air/fuel ratio in figures 8.30 and 8.31 and the

Generally the two-stroke diesel gas turbine displays some useful power increases over the basic engine. The use of larger turbochargers tends to increase scavenge at a given boost or engine speed, although heavy penalties are paid in terms of propane consumption and hence total specific fuel consumption. The improvements in power output are limited by scavenge problems. Scavenge

air flow may be improved by the use of higher boost levels or increasing turbocharger size. For satisfactory operation the scavenge ratio should increase with engine speed, as cylinder filling efficiency tends to reduce with increasing engine speed. Scavenge air flow could be improved by

detailed attention to the exhaust duct design.

It is well known (58, 59) that exhaust pulsations can give rise to complex positive and negative pressure wave reflections which may lead to scavenging problems.

9.2. Theoretical work

9.2.1. Standard engine program

The program (see Appendix 1) was examined by varying the input parameters at a constant engine speed. This demonstrated the relative sensitivity of the output parameters to errors of input data.

Figure 8.28 demonstrates the effect of varying boost ratio on power output, specific fuel consumption and brake thermal efficiency. The effects of varying boost ratio and scavenge ratio are demonstrated in figure 8.29, effects of varying air/fuel ratio in figures 8.30 and 8.31 and the effects of varying air manifold temperature in figures 8.32 and 8.33. Figure 8.34 demonstrates the effect of varying scavenge ratio on exhaust temperature over the full range of engine speed (presented as a family of scavenge curves). Increases in scavenge ratio have a more pronounced effect at lower scavenge ratios (an increase in scavenge ratio from 1.0 to 1.5 producing a reduction of 39% in exhaust temperature, whilst an increase from 1.5 to 2.0 produces a corresponding reduction of only 26%).

Similar families of curves are shown for variations in boost ratio with reference to power output and exhaust temperature (fig. 8.35). An increase in boost ratio has a more significant effect at lower boost levels. An increase

in boost ratio from 1.0 to 1.5 results in an increase of power output of 42%, and in exhaust temperature of 15%. A corresponding increase from 1.5 to 2.0 produces increases of 25% and 10% respectively. Families of curves for varying air/fuel ratio (fig. 8.36) show that a decrease in air/fuel ratio from 40 to 30 yields a power increase of 52% with an increase of 25% in exhaust temperature. A corresponding decrease from 30 to 20 yields a power increase of 69% with an increase of 37% in exhaust temperature. A similar family of curves is shown for variations in air manifold temperature (fig. 8.37). A decrease of 40°C in air manifold temperature results in an increase in power output of 15% with an increase of 6% in exhaust temperature. Further decreases in air manifold temperature produce proportionate increases in power output and exhaust temperature.

This section demonstrates that power increases may be obtained by increasing boost ratio and/or decreasing air/fuel ratio and air manifold temperature. As the air/fuel ratio for maximum output is usually fixed at $\approx 20:1$, this leaves the options of increasing boost and/or reducing air manifold temperature, with boost being the more significant factor. The thermal loading (as measured by engine exhaust temperature) may be controlled, to some extent, by scavenge ratio, an increase in scavenge ratio causing a decrease in exhaust temperature.

9.2.2. Preliminary matching program

The effects of variations in various input parameters have been examined. The effect of boost ratio on engine

fuelling requirements is shown in figure 8.38, presented as a family of boost curves, for a freely turbocharged system. Figure 8.39 displays corresponding auxiliary combustion chamber fuelling requirements. Boost levels of 1.1 to 2.3 (in steps of 0.3) are shown. A boost level of 1.1 is shown to be very inefficient, whilst an optimum fuel requirement is found at a boost ratio of 1.4. The effect of variations in scavenge ratio (fig. 8.40) demonstrates that scavenge ratios below 1.6 have no auxiliary fuelling requirement. Scavenge ratios in excess of 1.6 have a rapidly increasing fuel requirement, this effect being more pronounced at high engine speeds. The effect of exhaust blowdown temperature (fig. 8.41) demonstrates that auxiliary combustion is unnecessary at blowdown temperatures in excess of 1000°C . Lower levels of blowdown temperature have a rapidly increasing auxiliary fuelling requirement, the effect, again, being more pronounced at high engine speeds. The effect of various boost levels on the total system fuel requirement (fig. 8.42) demonstrates a clear optimum operating region at a boost of 1.4.

Figures 8.43, 8.44 and 8.45 show typical diagrams, produced as matching aids, to indicate feasible areas of turbocharged operation, for different compressor and turbine efficiencies. The curves displayed are of exhaust pressure ratio, turbine inlet temperature and engine exhaust temperature, for a range of scavenge ratios. Matched operation is likely to occur in regions where the exhaust pressure ratio curve lies below the line representing charge inlet pressure (shown dotted). Increase in the total air

For a specific scavenge ratio and blowdown temperature the turbine inlet temperature and engine exhaust temperature may be evaluated. For acceptable operation the engine exhaust temperature must be below that pertaining to the limit of thermal loading. If the turbine inlet temperature is below the engine exhaust temperature, there will be no matched operation at this point as the turbine will accelerate until a higher boost level is reached. Figures 8.43, 8.44 and 8.45 demonstrate that matched operation is more difficult to achieve with lower turbine and compressor efficiencies. Figure 8.46 demonstrates the use of auxiliary combustion to extend the areas of feasible operation for a simple series system.

9.2.3. Diesel gas turbine program

The effects of varying the program input parameters have been examined. The variation of engine air/fuel ratio is seen to affect power output, engine exhaust temperature and auxiliary fuelling requirements (fig. 8.47). As expected, power output and exhaust temperature decrease with increasing air/fuel ratio requiring an increase in auxiliary fuelling rate to maintain the desired boost levels. A variation in scavenge ratio affects the engine exhaust temperature and hence the required auxiliary fuelling rate (fig. 8.48). Increasing boost ratio increases engine power output and exhaust temperature (fig. 8.49) with a corresponding increase in air manifold temperature and decrease in auxiliary fuelling requirement, however, the compressor efficiency may vary, having a well defined optimum 'band' (fig. 8.50). An increase in the total air

flow (presented as Preston meter manometer reading), at a constant boost results in an increase in auxiliary fuelling requirement (fig. 8.51). As the boost level is constant, the trapped air flow will not increase, and the extra air will go to the auxiliary combustion chamber for heating. No variation of compressor efficiency (hence air manifold temperature and power output) was found at the air flow levels examined.

9.3. Comparison of experimental and theoretical results

The comparisons were carried out using experimentally obtained data for the input parameters of the various computer programs. The output parameters may then be directly compared to investigate the validity of the programs.

9.3.1. Standard engine

As the standard engine program was to be used as a subroutine in the later diesel gas turbine program, it was important that it should yield accurate predictions. The computer program was thus adjusted to match experimental results (figs. 8.52, 8.53).

9.3.2. Preliminary matching

As the preliminary matching program demonstrates areas of feasible operation, rather than well defined operating points, it is not possible to make direct comparisons with experimental results. Figure 8.54 demonstrates that turbine and compressor efficiencies, of 75% and 70% respectively, should yield a wide range of matched operation. Turbine and compressor efficiencies of 65% and 55% respectively (fig. 8.55) show only areas of marginally matched operation, matching being possible only at low scavenge ratios and

excessive exhaust temperatures. Figure 8.56 demonstrates that matched operation is not feasible with turbine and compressor efficiencies of 60% and 40% respectively. The curves demonstrate that component efficiencies are critical and matched operation is only a near certainty with an overall turbocharger efficiency of $\approx 50\%$. For certain matching, component efficiencies must remain high over a broad operating band. Such efficiencies are difficult to obtain with small turbochargers suitable for automotive sized engines. Experimental results (fig. 8.6) have demonstrated that freely turbocharged results were obtainable over a very narrow range of engine speed, limited by excessive exhaust temperatures, implying a marginal match situation.

9.3.3. Diesel gas turbine

As this program was developed during the course of the work, the comparisons have been exhaustive, covering three turbocharger sizes at both constant and variable engine speed. During testing it was not possible to measure scavenge ratios. Scavenge ratios used in the computer program were altered to give matched predictions of engine exhaust temperature. During testing it was found that scavenge problems existed at higher engine speeds, resulting in poor combustion and high specific fuel consumption. The computer program catered for this by increasing the effective air/fuel ratio under conditions of low scavenge. Specific fuel consumption was computed from the program input parameter of apparent (i.e. measured) air/fuel ratio.

Constant engine speed

The results for constant engine speed tests are presented for each of the three turbochargers investigated.

(i) Holset 3DL1 1.57

Engine power output and specific fuel consumption are predicted to within 1% and 2% of experimental values respectively (fig. 8.57). Air manifold temperature is predicted to an accuracy of 2% of the experimental value, whilst air/fuel ratio shows maximum discrepancies of 16% at low boost (fig. 8.58). The air/fuel ratio discrepancies occur as the measured value is 'apparent', whilst the predicted values are 'effective'. Inadequate scavenge raises the 'effective' air/fuel ratio of the charge mixture, leaving a residual amount of unburned fuel, giving poor combustion and high smoke emission at lower boost levels (see fig. 8.23). Predictions of gas (propane) consumption of the auxiliary combustion chamber and total system specific fuel consumption are predicted to 4% and 1%, of experimental values, respectively (fig. 8.59).

(ii) Holset 3LD1 2.5

Power output displays a maximum discrepancy of 1.6% at high boost and engine S.F.C. is predicted to better than 1.5% of experimental value (fig. 8.60). Air manifold temperature displays a maximum discrepancy of 5% whilst air/fuel ratios must again be considered as 'effective' and 'apparent' values, with discrepancies of 9% at low boost levels

(fig 8.61). Gas consumption is predicted well until lower boost levels, which show deviations of 15% of experimental value and total system specific fuel consumption is predicted to better than 8% (fig. 8.62).

As the data logging system was not available for this set of tests, gas consumption was measured 'manually' with the electronic equipment with a resultant loss of accuracy.

(iii) Holset 4LGK 2.6 T2

Power output displays a maximum discrepancy of 1.5% of experimental value at high boost levels and there is a similar deviation of 5% in engine specific fuel consumption (fig. 8.63). Air manifold temperature shows a maximum deviation of 3% at low boost levels and air/fuel ratio a maximum deviation of 10% of experimental value at low boost levels (fig. 8.64). This is again due to 'apparent' and 'effective' values. Gas consumption is predicted to better than 9% and total system specific fuel consumption better than 6% of experimental value (fig. 8.65).

In general the performance prediction program performs well, giving predictions mainly within 2% of the experimental values. Gas consumption of the auxiliary combustion chamber is not predicted quite so well. This may be due to inaccuracy in the measured results. Computerised data acquisition of gas flow rates dramatically improves measurement accuracy.

The computer program was also used to calculate the contribution of pulse work to the turbine. This is defined as the heat energy required (evaluated from the compressor work and mechanical efficiency), minus the heat energy supplied (evaluated from turbine inlet and outlet temperatures and mass flow rates). As the pulse energy was calculated as a negative quantity it was termed an 'error' factor. Since the auxiliary fuelling rate (evaluated using measured turbine inlet temperatures) is predicted with reasonable accuracy, some level of doubt is cast on measured turbine outlet conditions. Evaluation of self-sustained gas turbine results (see Appendix 8), carried out on a separate rig developed during the course of the work, has also demonstrated a negative 'error' factor (on a constant pressure process). It thus seems reasonable to suggest that measured turbine conditions were not ideal and contained errors.

Curves have been plotted to show energy supplied to the turbine against the energy demand of the turbine (fig. 9.1). An enthalpy matched curve is shown. This curve applies if all the turbine work is supplied as heat energy (constant pressure turbo-charging). Curves moving to the left of the enthalpy matched condition display increasing 'errors', i.e. more heat is supplied than would appear to be necessary to drive the turbine. The 'error' becomes more pronounced for larger turbine housings.

Figure 9.2 shows the 'error' plotted against turbine expansion ratio. In all cases the 'error' increases with expansion ratio and is again greater for larger turbine housings. Figure 9.3 shows the quotient of error by work demanded, against expansion ratio. The turbine housings in the smaller frame size show a trend of increasing error ratio (P/w) with expansion ratio, the trend being increased with a larger housing. The larger frame size turbine displays a reverse trend, P/w increasing with decreasing expansion ratio. These trends are reflected in figure 9.4, showing a turbine work factor against expansion ratio and work demand. The work factor has been defined as work factor = $1 / [1 - (P/w)]$. For a constant pressure turbo-charging system this would yield a value of 100%. Work factors in excess of 100% signify a positive 'error' (pulse) contribution, values less than 100% signify a negative 'error' contribution.

These curves lead to a postulation that 'error' may be connected, in some way, with mass flow rates. Figure 9.5 shows 'error' plotted against the turbine mass flow parameter. Smaller frame sizes display a similar level of 'error' for different housing sizes (hence different mass flow parameters). Larger frame sizes display increasing error at similar mass flow parameters. Error would thus seem to be dependent upon housing area and expansion ratio and heavily dependent upon frame size.

The measurement of turbine inlet and outlet conditions is notoriously difficult. Problems are exacerbated on a system incorporating auxiliary combustion due to several effects.

- (ii) a) The possible proximity of the flame, to turbine inlet thermocouples, distorting readings.
- b) The relatively small mixing length available to differing gas streams with widely varying temperature profiles (the analysis used in this work has assumed perfect mixing).
- c) The possibility of secondary auxiliary combustion, taking place at the turbine inlet, causing localised hot spots with the possibility of flame passage through the turbine, leading to poor efficiency and unpredictable performance.

Variable engine speed

Tests at variable engine speed were carried out using three different turbochargers.

(i) Holset 3LD1 1.57

Power output is predicted to 1% of experimental values and engine specific fuel consumption shows a maximum deviation of 2.5% (fig. 8.66).

Auxiliary combustion chamber gas (propane) consumption displays a maximum deviation of 4% from experimental values at higher engine speeds

(> 1400 rev/min and the maximum discrepancy between 'apparent' and 'effective' air/fuel ratio is 9% (fig. 8.67)). Total system specific fuel consumption is predicted to within 2% of experimental values,

air manifold temperature is not so well predicted, showing maximum discrepancies of 17% (fig. 8.68).

(iii) Holset 3LD1 Testing with this unit was terminated at an engine speed of 1600 rev/min.

(ii) Holset 3LD1 2.5 speed of 2400 rev/min.

Power output is predicted to 2% accuracy and engine specific fuel consumption shows a maximum deviation from experimental values of 3% at 1400 rev/min (fig. 8.69). Gas (propane) consumption is predicted well at low engine speed (better than 2% up to 1200 rev/min) but shows an increasing deviation with engine speed up to a maximum of 13% of experimental values at 1800 rev/min.

Apparent engine air/fuel ratios show a maximum discrepancy of 3% (fig. 8.70). Total system specific fuel consumption is predicted within 1% up to an engine speed of 1400 rev/min and then displays an increasing deviation up to a maximum of 7% of experimental values at 1800 rev/min.

Air manifold temperature is predicted well at a mid-speed range of 1400 rev/min but displays deviations of - 3% at 1000 rev/min and + 2% at 1800 rev/min (fig. 8.71). Testing with this unit was discontinued at an engine speed of 1800 rev/min.

Discrepancies in total specific fuel consumption are caused by deviations in predicted gas consumption. It has been found that vibrations of the gas measurement system (see Appendix 8) cause a high reading of fuel consumption. Steps were

taken to reduce vibrations by isolating the measurement system.

(iii) Holset 4LGK 2.6 T2

Testing of this unit was continued up to the maximum rated engine speed of 2400 rev/min. Power output is very well predicted to within 1% of experimental values, engine specific fuel consumption has a maximum deviation of 7% at 1400 rev/min (fig. 8.72). Gas consumption is poorly predicted, having an increasing deviation from 6% of experimental values at 1000 rev/min to 75% at 2400 rev/min. Engine 'apparent' air/fuel ratio is predicted within 4% up to 2000 rev/min and then displays a progressive deviation to a maximum of 10% of experimental values at 2400 rev/min (fig. 8.73). The deviations in gas consumption are reflected in the prediction of total system specific fuel consumption, which displays deviations of 5% at 1000 rev/min to 19% of experimental values at 2400 rev/min. Air manifold temperature is predicted within 2% up to 1600 rev/min and then displays an increasing deviation up to a maximum of 6% of experimental values at 2400 rev/min (fig. 8.74).

Generally the prediction program works very well, giving results to within 2% of experimental values for most parameters. Gas (propane) consumption of the auxiliary combustion chamber is predicted more accurately for the constant engine speed tests.

Gas consumption for variable engine speed tests is not predicted so well, especially at high engine speeds. This may be due to increasing vibration levels distorting measured values of gas consumption and giving artificially high values.

9.4. Extension of the scope of the programme by computer predictions

9.4.1. Diesel gas turbine

As the prediction program has been demonstrated to give reasonably accurate results, it was used to extend the scope of the work. One limitation of the project was the fixed delivery fuel injection pump, which did not permit the use of optimum air/fuel ratios. The prediction program has been used to evaluate system performance with air/fuel ratios of 20 : 1, charge cooling and heat exchange. In all cases a scavenge ratio of 1.4 has been assumed.

(i) Holset 3LD1 1.57

Figure 8.75 shows power output and torque. The 'as tested' condition displays increases, of a maximum of 10%, over the standard engine. The use of 20 : 1 air/fuel ratios displays a potential increase of from 42% to 63% over the standard engine. The use of charge cooling permits further increases to be obtained (from 69% at 1000 rev/min to 104% at 1600 rev/min). The investigation was

(ii)

Holset 3LD1 2.3
terminated at 1600 rev/min, due to a lack of experimental input data. Figure 8.76 shows engine exhaust temperature and total system specific fuel consumption. System specific fuel consumption is,

in all cases, higher than that of the standard engine. Reductions in specific fuel consumption can be brought about by decreasing the engine air/fuel ratio to 20 : 1, a further reduction by the use of charge cooling (less air is bypassed) and further still by the use of heat exchange between turbine exhaust and auxiliary combustion chamber air inlet. The most favourable case (air/fuel ratio = 20 : 1, charge cooled, with heat exchange) displays an increase in specific fuel consumption of 13% over the standard engine at 1000 rev/min with a reduction in these increases, to display a 1% improvement at 1600 rev/min, the implication being that speeds in excess of 1600 rev/min require no auxiliary combustion and thus may demonstrate an improvement over the standard engine. The use of a 20 : 1 air/fuel ratio increases the engine exhaust temperature by some 35% (fig. 8.76). This is still within thermal loading limits, however the extension of the operating range above 1600 rev/min would give cause for concern. The use of a charge cooler reduces engine exhaust temperature to some 25% above that of the standard engine. This is well within thermal loading constraints.

(ii) Holset 3LD1 2.5

Figure 8.77 shows power output and torque up to an engine speed of 1800 rev/min. Output in the 'as tested' condition shows an improvement of some 12% over the standard engine. The use of a

20 : 1 air/fuel ratio demonstrates increases of 49% at 1000 rev/min to 70% over the standard engine at 1800 rev/min. Charge cooling further increases output by 80% to 111%. Figure 8.78 shows engine exhaust temperature and total system specific fuel consumption. Decreases in specific fuel consumption (from the 'as tested' condition) are obtained by the use of 20 : 1 air/fuel ratios, further decreases being obtained by charge cooling, and heat exchange. The most favourable system (air/fuel ratio = 20 : 1, charge cooled, with heat exchange), displays fuel consumption increases from 21% at 1000 rev/min to 7% at 1800 rev/min. The trend implies that auxiliary combustion assistance would prove unnecessary at engine speeds in excess of \approx 2000 rev/min. Engine exhaust temperature (fig. 8.78) shows that there is cause for some concern at high engine speeds. The use of a charge cooler reduces exhaust temperature to some 16% above that of the standard engine, which is well within the limits of thermal loading.

(iii) Holset 4LGK 2.6 T2

Figure 8.79 shows power output and torque up to full rated engine speed. The 'as tested' condition displays maximum increases of some 15% over the standard engine. The use of 20 : 1 air/fuel ratios results in increases from 38% at 1000 rev/min to 89% over the standard engine at 2400 rev/min. The use of charge cooling gives increases from 59% to 129% over the standard engine. Figure 8.80 shows the specific

fuel consumption of the total system and engine exhaust temperature. The lowest specific fuel consumption is given by a system using 20 : 1 air/fuel ratio, charge cooling and heat exchange.

Such a system gives fuel consumption increases of 50% at 1000 rev/min to improvements of 9% over the standard engine at 2400 rev/min. A possibility of operating with auxiliary combustion is demonstrated for speeds in excess of 2200 rev/min. A charge cooled system would permit full speed range operation without violating thermal loading constraints.

The extension of the experimental work has shown that the two-stroke diesel gas turbine, working at modest boost levels (up to 2 : 1), can demonstrate power increases of up to 129% over the standard engine (providing that scavenge difficulties can be overcome).

Larger turbochargers demonstrate the greatest levels of improvement, but display larger penalties in specific fuel consumption. The use of larger turbochargers also extends the range of engine speed, over which auxiliary combustion is required. Although the torque is much improved, the torque envelope is essentially 'flat', falling slightly with decreasing engine speed. This is not an ideal situation for traction requirements.

The diesel gas turbine offers scope for torque 'tailoring' by controlling boost levels. This can be effected by adjusting the auxiliary fuelling rate. Thus boost levels may be decreased with increasing

engine speed to provide good 'back-up' torque.

Computer predictions have been obtained for such a system (as follows).

Torque 'tailoring'

Compressor operating lines have been chosen (figs. 8.81, 8.82) to give a boost of 1.5 at an engine speed of 2400 rev/min, rising linearly with decreasing engine speed to 2.2 at 1000 rev/min. Figure 8.83 shows power output and engine exhaust temperature.

Power increases of 73% at 1000 rev/min to 66% over the standard engine at 2400 rev/min are demonstrated.

Charge cooling raises these increases to 113% and 89% respectively. Exhaust temperature reaches a

plateau at 600°C, which is the acceptable thermal loading limit; with charge cooling this is reduced to 550°C. Output torque (fig. 8.84) demonstrates

a rising characteristic with decreasing speed, i.e. very good 'back-up'. The characteristic is

linear, rather than the ideal hyperbola, however it demonstrates a dramatic improvement over the standard

engine. The specific fuel consumption curves (fig. 8.85) demonstrate that auxiliary combustion (hence

bypass air) would be unnecessary at engine speeds in excess of \approx 1800 rev/min. (To maintain falling

boost with increasing speed, a wastegate may be necessary.) Low speed specific fuel consumption is

dramatically increased. The system offering the lowest specific fuel consumption utilises the Holset

3LD1 1.57 turbocharger, charge cooling and heat

ramp function) to exchange. This system demonstrates a maximum increase of specific fuel consumption of 14% over the standard engine, although reductions at high speeds (maximum $\approx 17\%$), could result in overall operating costs of a similar order to those of the standard engine. Specific fuel consumption at full load conditions could be improved by the use of higher boost levels with an associated reduction of engine compression ratio.

9.5.1 General comments

The microprocessor logging system has been shown to operate satisfactorily and, in conjunction with further software, demonstrates good potential for a sophisticated control system.

The stepper motor drive system for control purposes has been shown to operate satisfactorily with the limited microprocessor software produced. Further development of the control software to embrace several stepper motors, and the significant parameters of the diesel gas turbine system is required.

The 'errors' present, due to the difficulties in measuring the turbine operating parameters must be eliminated. This work could be carried out on the existing 'hot' rig. Temperature profiles must be measured in various parts of the ducting to enable the most suitable measurement sites to be evaluated.

The possibility of secondary auxiliary combustion (fire-balls) must also be eliminated. The diesel gas turbine ducting should be improved to give adequate scavenge at all engine speeds.

Once an adequate control system is operational, transient performance should be evaluated. The study of transients would require the use of accurately repeatable input signals (step and

ramp function) to both brake and engine. A microprocessor could perform this function adequately, whilst also facilitating data acquisition.

9.6. Differences between the diesel gas turbine and the hyperbar system

A comparison between the engine systems, described in Chapter 2, and the diesel gas turbine is given in Table 9.1. Table 9.2 gives a more specific comparison with the hyperbar system, which has been described in section 2.2.1. It is compared with the diesel gas turbine as both systems utilise auxiliary combustion. The major differences between the systems are detailed below.

Engine type

Reported hyperbar installations (15) have utilised 6 cylinder, four-stroke, diesel engines of 10.48 and 59.72 litres. The diesel gas turbine uses a three cylinder, two-stroke, engine of 3.52 litres.

Compression ratio

The hyperbar system utilised very low compression ratios, down to 6.8 : 1, whereas the diesel gas turbine utilises the standard engine compression ratio of 16 : 1.

Boost

The hyperbar system employed single or double stage turbo-charging, enabling boost ratios of up to 7 : 1 to be obtained. The diesel gas turbine employs maximum boost ratios of 2.2:1.

Air/fuel ratio

The hyperbar installations used minimum air/fuel ratios of between 38 and 32 : 1, whereas the diesel gas turbine was expected

to use air/fuel ratios of 22 : 1. It was not possible to realise these values, due to the limitations of the fixed delivery, DPA type, injection pump used on the engine. If the pump delivery could be increased, in conjunction with the solution of scavenge problems, there are no problems anticipated in the use of 22 :1 air/fuel ratios.

Auxiliary combustion

In the hyperbar system, auxiliary combustion was undertaken using a mixture of engine exhaust gases and clean bypass air. Accordingly some level of exhaust pulsation will be present at the auxiliary combustion chamber. This may have resulted in flame instability and satisfactory operation may only have been realised by the use of high air/fuel ratios. The instability effect would have been minimised by the use of engines having a relatively large number of cylinders.

The diesel gas turbine uses only clean, constant pressure, air for auxiliary combustion. The engine exhaust is mixed with hot gases from auxiliary combustion just before the turbine inlet. Exhaust pulsations may thus be utilised in the turbine.

The hyperbar system may have benefitted from its auxiliary combustion system, in that engine exhaust smoke was subjected to a reburning process which may have resulted in reduced hydrocarbon and carbon monoxide emissions. The diesel gas turbine does not have this facility, although some beneficial effects may occur from the mixing of engine exhaust gases with the very hot gases from auxiliary combustion.

The hyperbar system had certain zones of operation in which auxiliary combustion was unnecessary. A disadvantage of using a two-stroke is that auxiliary combustion is required over relatively extended zones.

10. Conclusions

10.1. Present investigation

10.1.1. Experimental work.

10.1.2. Theoretical work.

10.1.3. Theoretical prediction of performance.

10.2. Suggestions for further work

10.2.1. Experimental work.

10.2.2. Theoretical work.

10. Conclusions

10.1. Present investigation

The present investigation has involved a substantial experimental programme and the development of a computer program for the prediction of system performance.

10.1.1. Experimental work

The experimental programme proceeded through several stages of development, as detailed in chapter 4, with results being presented in chapter 8. Several conclusions may be drawn:-

- 1) Despite the limitations of the fuel injection pump, modest increases in 'back-up' torque have been demonstrated.
- 2) The mechanically driven scavenge blower has been eliminated from the system. System 'start up' being effected by using the turbocharger and auxiliary combustion chamber as a self-sustained gas turbine, thus providing a pressure drop across the engine, and permitting engine starting in the normal manner.
- 3) Performance may be improved and the operating range extended by the use of larger turbochargers; however these incur penalties in specific fuel consumption.
- 4) Progress has been made with data acquisition and control systems based on microprocessors. A significant amount of work remains to be done before the diesel gas turbine system could be used as a commercially viable traction package.

- 5) The study of transients could be easily facilitated by the use of the microprocessor control and data acquisition systems.

10.1.2. Theoretical work

- 1) The simple 'quasi-steady' engine program has been developed to give performance predictions for the standard engine, which is subsequently used as a section of the larger diesel gas turbine performance prediction program.
- 2) The use of the preliminary matching program described must be undertaken with caution. Optimistic estimates of turbine and compressor efficiencies can produce misleading results, as these factors are critical. Realistic efficiencies pertaining to small size, automotive, turbochargers are more likely, at best, to produce a marginal match for a two-stroke cycle diesel engine.
- 3) The diesel gas turbine prediction program gives very good predictions of system performance. Auxiliary fuel consumption is not predicted so accurately although this is more likely to be due to measurement errors.

10.1.3. Theoretical prediction of performance

- 1) Increases of power and torque output, up to a maximum of some 130% have been shown to be possible, whilst remaining within thermal and mechanical loading constraints (assuming that adequate scavenge can be maintained).

- 2) Torque 'tailoring' is possible, yielding increases of 66% to 73% over the standard engine torque.
The torque characteristic rises with decreasing engine speed, in a linear manner. Whilst this is not the ideal torque hyperbola, it is a significant improvement on the standard characteristic and would permit a reduction in transmission complexity.
- 3) The use of a larger turbocharger results in increased torque output with increased specific fuel consumption.
- 4) The smallest turbocharger examined, a Holset 3LD1 1.57, is capable of giving increased output (66%-73%) with an improved torque characteristic.

Penalties in specific fuel consumption are incurred at lower engine speeds, although these are, to some extent, offset by improvements at high speed and overall operating fuel economy may be similar to that of the standard engine.

The two-stroke diesel gas turbine is a promising traction system yielding improvements of power output and a much improved torque characteristic over that of the standard engine. These improvements can be made whilst remaining within acceptable thermal and mechanical loading constraints.

In practice, other workers (Timoney, Wallace) have found that the two-stroke, in highly rated forms, has demonstrated thermal loading problems resulting in piston damage. The lower thermal loading of the four-stroke may thus make it a better candidate for

the diesel gas turbine system.

The system does not require any major engine alterations (e.g. the standard compression ratio is used). Alterations are confined to exhaust ducting and 'bolt on' components.

10.2. Suggestions for further work

10.2.1. Experimental work

- 1) The problem of inadequate scavenge must be solved. This requires detailed attention to the design of the exhaust ducting. A modified exhaust manifold is required to shorten pipe lengths and reduce the possibility of pulse reflections and interference.
- 2) The DPA type injector pump should be replaced with an 'in line' version. This, together with modified injectors, would provide extra fuelling capacity, enabling performance improvements to be realised.
- 3) Charge cooling may be considered for high outputs. An air/water system would be most suited to the test rig. For very high outputs, consideration may be given to the use of two stage turbocharging with charge cooling, and an engine with reduced compression ratio.
- 4) Tests should be carried out on the separate 'hot' gas turbine rig to assess the most suitable sites for turbine instrumentation in order to eliminate the turbine work 'error' factor. The possibility of secondary auxiliary combustion could be eliminated by monitoring unburned hydrocarbons at various points in the ducting.

- 5) The auxiliary combustion chamber should be converted to use diesel fuel in order to ease fuel consumption measurements and to eliminate the need for two different fuel supplies.
- 6) Transient performance should be investigated with both 'ramp' and 'step' inputs to the engine and brake. This could be facilitated by the use of the microprocessor control system to provide repeatable input signals. The response of the water dynamometer would not be fast enough to cope with 'step' loads and thus may be restricted to investigating 'lugging' modes.
- 7) For true commercial viability more effort must be directed towards compliance with noise and exhaust emission regulations.
- 8) The control system must be developed to produce a reliable system requiring a minimum of operator attention.

10.2.2. Theoretical work

- 1) The computer program for performance prediction should be further developed to account for heat transfer from the ducting and adequate scavenge. Attempts should be made to reduce the amount of empirical data required.
- 2) For transient studies, the program would need to be modified to use a 'step-by-step' analysis.
- 3) The control system software should be developed to enable implementation of the microprocessor based control system.

- 4) The study of transient phenomena would require that the software for the microprocessor logging system be restructured to cope with the change in data acquisition requirements.

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ROOTES TS3

Two stroke, horizontal three cylinder with opposed pistons, direct injection diesel engine.

Type 3D215

Bore	3.375 in (85.7 mm)
Stroke	4.00 in (101.6 mm)
Effective compression ratio	16:1
Capacity	215 cu. in (3520 cc)
BHP	125 (93.2 kw) at 2400 rev/min
No. of cylinders	Three
Firing order	1, 2, 3.
Injection timing (static)	16° B IDC
Max. torque	313 lb ft (425 Nm) at 1200 rev/min
Blower make	Wade 5R034 No. 2679
Type	Roots with two 3 lobe rotors
Injector pump	Lucas/CAV
Type	DPA 3233400Z0 (mechanical governer) CAV BKB 52S656 DN 5206404
Injectors	Single nozzle set to open at 175 atmospheres.

ENGINE SPECIFICATIONS

<u>PARAMETER</u>	<u>INSTRUMENT TYPE</u>	<u>OPERATING PRINCIPLE</u>
Brake Load	Heenan Froude dynamometer DPx4 with strain gauged cantilever, Sangamo C56 amplifier.	Change in fluid momentum.
Engine Speed 1)	Orbit controls counter-timer.	Induced voltage from magnetic pick-up.
2)	Frequency to voltage converter; DVM.	Optical switch & electronic signal conditioning.
3)	Tachometer.	Magnetically coupled dial indicator.
Turbocharger Speed	Orbit controls counter/timer and magnetic nut.	Induced voltage from magnetic search coil.
Air Flow	Preston meter and manometer.	Averaging pitot static tubes.
Cylinder Pressure	Southern Instruments T500 Pressure Transducer Readout Unit M1681, Tektronix 5648 Oscilloscope or Peak Detect/sample & hold system.	Inductive FM system.
Exhaust Pulses	AVL quartz pressure transducer 12QP 500C, SI charge amp M05100 tungsten light recorder.	Piezo electric.
Gas Pressures	Bourdon gauges.	
Gas Temperatures	Thermocouples	Thermoelectric effect on dissimilar metals.
Water Temperatures	Dial indicating thermometers.	Vapour pressure.
Exhaust Gas Analysis	Bosch meter.	Reflected light from filter paper (opacity).
	Beckman model 590 HC, CO.	Absorption of infra red radiation.
	Beckman NO/NO _x analyser.	Chemiluminescence technique. (The emission of light resulting from a chemical reaction).
Fuel Flow	Litre meter and readout unit.	Volumetric rotor type (frequency output). Frequency count, digital to analogue conversion.
Gas Flow (propane)	Strain gauged proving ring. Sangamo Weston amplifier. DVM/logger.	Resistance change in full bridge. Dual ramp integration.

TEST RIG INSTRUMENTATION

Pressure (Bar)	Output Voltage (Volts)				Average
	Up	Down	Up	Down	
0	0.00	0.00	0.00	0.01	0.00
10	0.14	0.14	0.13	0.14	0.14
20	0.27	0.27	0.27	0.28	0.27
30	0.40	0.41	0.40	0.41	0.41
40	0.54	0.55	0.54	0.55	0.55
50	0.68	0.69	0.69	0.69	0.69
60	0.82	0.83	0.83	0.83	0.83
70	0.97	0.97	0.97	0.98	0.97
80	1.11	1.12	1.11	1.11	1.11
90	1.25	1.26	1.26	1.26	1.26
100	1.39	1.40	1.40	1.41	1.40
110	1.54	1.54	1.54	1.54	1.54
120	1.68	1.68	1.68	1.68	1.68

Calibration of Southern Instruments T500/H524

Pressure Transducer using:-

Barnet Industrial Deadweight Tester No 3760/74

Southern Instruments F.M. Unit M1860 (No18240)

Zero Setting 0.00v

Reference 1.50v

CALIBRATION OF SI T500 PRESSURE TRANSDUCER

FUEL FLOW RATE SYSTEM CALIBRATION

Voltage Output Volts	Weight of Can lb/oz	Weight of Can+Fuel lb/oz	Time min/sec	Weight of Fuel lb	Time min	Fuel Per Min lb	Fuel Per Hr lb	Fuel Per Hr kg
0.79	10/4	12/8	25/0	2.250	25.00	0.090	5.40	2.45
1.10	10/5	12/0	15/33	1.688	15.55	0.109	6.51	2.95
1.41	10/4	12/4	15/0	2.000	15.00	0.133	8.00	3.63
1.88	12/4	15/10	21/2	3.375	21.03	0.161	9.63	4.37
2.19	14/0	18/3	22/15	4.188	22.25	0.188	11.29	5.12
2.51	10/5	14/10	21/0	4.313	21.00	0.205	12.32	5.59
2.98	17/13	20/14	13/30	3.063	13.50	0.227	13.61	6.17
3.45	15/10	19/4	14/0	3.625	14.00	0.259	15.54	7.05
3.91	14/8	17/13	11/30	3.313	11.50	0.288	17.29	7.84
4.22	10/5	14/11	13/46	4.375	13.77	0.318	19.06	8.65
4.70	15/15	21/2	15/0	5.188	15.00	0.346	20.75	9.41
5.00	10/5	15/6	14/0	5.063	14.00	0.362	21.70	9.84
5.15	10/6	14/8	11/0	4.125	11.00	0.375	22.50	10.21
5.48	10/5	15/15	14/29	5.625	14.48	0.389	23.31	10.57
6.09	20/2	25/11	13/0	5.563	13.00	0.428	25.68	11.65
6.55	12/0	16/7	9/43	4.438	9.72	0.457	27.40	12.43

TABLE 7.2

FUEL FLOW RATE SYSTEM CALIBRATION

Voltage Output Volts	Weight of Can lb/oz	Weight of Can+Fuel lb/oz	Time min/sec	Weight of Fuel lb	Time min	Fuel Per Min lb	Fuel Per Hr lb	Fuel Per Hr kg
7.17	22/2	27/5	10/30	5.188	10.50	0.494	29.66	13.45
7.50	17/9	20/11	6/0	3.125	6.00	0.521	31.25	14.17
7.98	10/6	17/9	13/0	7.188	13.00	0.553	33.18	15.05
8.27	16/7	22/2	10/0	5.688	10.00	0.569	34.13	15.48
8.59	17/3	23/2	10/14	5.938	10.23	0.580	34.82	15.79
8.91	16/14	22/5	9/0	5.438	9.00	0.604	36.25	16.44
9.52	14/10	20/8	9/0	5.875	9.00	0.653	39.17	17.77
9.84	10/8	17/3	10/0	6.688	10.00	0.669	40.13	18.20
10.47	20/8	24/0	5/0	3.500	5.00	0.700	42.00	19.05
10.96	10/5	16/14	9/0	6.563	9.00	0.729	43.75	19.84
11.43	10/4	16/11	8/30	6.438	8.50	0.757	45.44	20.61
11.87	14/7	18/6	5/0	3.938	5.00	0.788	47.26	21.44
12.34	10/5	14/7	5/0	4.125	5.00	0.825	49.50	22.45

Calibration of Fuel Flow Rate System:- Ambient 18°C 762.55 mm Hg

Avery Type 3205 ABA Scale

Smiths Stop Clock

TABLE 7.4

LOAD LB	OUTPUT VOLTAGE V		
	UP	DOWN	AVERAGE
0	0	0	0
3.5	0.09	0.10	0.095
7.0	0.19	0.20	0.195
10.5	0.29	0.29	0.29
13.5	0.38	0.38	0.38
15.5	0.44	0.45	0.445
18.5	0.52	0.52	0.52
20.5	0.57	0.58	0.575
23.5	0.66	0.67	0.665
28.5	0.81	0.81	0.81
33.5	0.95	0.96	0.955
38.5	1.10	1.11	1.105
43.5	1.25	1.25	1.25
48.5	1.40	1.40	1.40
53.5	1.55	1.55	1.55
58.5	1.70	1.70	1.70
63.5	1.83	1.84	1.835
68.5	1.98	1.98	1.98
73.5	2.12	2.12	2.12
78.5	2.26	2.27	2.265
83.5	2.41	2.41	2.41
88.5	2.55	2.55	2.55
93.5	2.69	2.70	2.695
98.5	2.83	2.84	2.835
103.5	2.98	2.98	2.98

Sangamo Weston channel 1.

Gauge factor 4.46 Range 0.25.

Calibration: Load (N) = 154.48 x V

'INSITU' CALIBRATION OF STRAIN GAUGED CANTILEVER

TABLE 7.5

Manometer Reading (x) ΔP mmH ₂ O	Air velocity (y) m/s
0	0
2	5.5
3	6.8
4	9.6
7	11.9
10	14.9
15	17.9
21	20.3
26	23.1
32	25.2
35	26.7
39	27.6
44	28.7
49	29.5
51	30.2
56	30.7

Atmospheric conditions 759.6 mm Hg 18°C.

From polynomial curve fitting routine:

$$\text{Velocity (y)} = 0.1298450 + 3.0057677x - 0.2483596x^2 \\ + 0.0123399x^3 - 0.0003226x^4 + 0.0000042x^5$$

For dimensions of preston meter used:

$$\text{Air flow rate} = 30.378 \times y \text{ Kg/hr.}$$

PRESTON METER CALIBRATION

Load (Bar)	Output Voltage (Volts)				Average
	Up	Down	Up	Down	
0	+0.001	-0.006	-0.006	-0.004	-0.004
5	0.498	0.499	0.493	0.498	0.497
10	0.997	0.999	0.993	1.001	0.997
15	1.485	1.500	1.493	1.510	1.497
20	1.986	1.999	1.993	2.010	1.997
25	2.490	2.509	2.486	2.510	2.499
30	2.990	3.007	2.991	3.010	2.999
35	3.490	3.505	3.494	3.513	3.501
40	3.995	4.010	3.994	4.013	4.001
45	4.504	4.507	4.494	4.505	4.503
50	5.001	5.001	4.997	4.997	4.999

Calibration using deadweights Sangamo Weston C56 Channel 4

Cal. Setting 5.000 V Range 0.25 Gauge Factor 2.46

Hewlett Packard HP 3465B Digital Multi Meter

Calibration:- Weight = 10 x Voltage Output

CALIBRATION OF PROVING RING FOR GAS FLOW MEASUREMENT

TABLE 7.7

Frequency Hz	Engine speed equivalent Rev/min	Output voltage V
200	100	0.195
400	200	0.535
600	300	0.895
800	400	1.253
1000	500	1.609
1600	800	2.672
2000	1000	3.378
2400	1200	4.076
2800	1400	4.776
3200	1600	5.468
3600	1800	6.151
4000	2000	6.826
4400	2200	7.493
4800	2400	8.142
5200	2600	8.772
5600	2800	9.380
6000	3000	9.998

Calibration of F-V converter:

Farnell Instruments signal generator

Orbit Controls counter timer

Hewlett Packard HP 3465B digital multi meter.

TABLE 8,1

SPEED Rev/min	STANDARD	STAGE 1 (T/C only)	STAGE 1 (TC + SB 1.57)	STAGE 1 (TC + SB 2.5)	STAGE 2 (TC + SB + ACC 1.57)	STAGE 3 1.57	STAGE 3 2.5	STAGE 2.6
<u>Boost Levels</u>								
1000	1.10	-	1.22	1.18	1.4	1.72	1.80	1.54
1200	1.10	-	1.40	1.28	1.51	2.06	1.85	1.76
1400	1.10	1.50*	1.55	1.36	1.59	2.01	1.91	1.76
1600	1.10	1.55†	1.68	1.45	1.74	2.08	2.00	1.91
1800	1.18	1.53	1.80	1.58	1.88	-	2.02	1.94
2000	1.23	1.58	1.96	1.72	-	-	-	1.89
2200	1.30	-	1.15	1.84	-	-	-	1.93
2400	1.41	-	1.30	1.98	-	-	-	1.92
<u>Scavenge Ratios</u>								
							Program estimation	
1000	1.53	-	1.84	1.92	1.41	1.20	1.35	1.65
1200	1.57	-	1.78	1.79	1.64	1.30	1.10	1.45
1400	1.61	-	1.79	1.75	1.61	1.10	1.30	1.40
1600	1.65	-	1.93	1.98	1.51	1.20	1.10	1.40
1800	1.63	-	1.82	1.93	1.20	-	0.90	1.00
2000	1.52	-	1.58	1.72	-	-	-	1.00
2200	1.56	-	1.10	1.60	-	-	-	1.20
2400	1.48	-	0.94	1.51	-	-	-	1.00
<u>Smoke Emission (Bosch)</u>								
					**			
1000	-	-	6.0	5.8	2.6	2.2	2.0	2.1
1200	-	-	4.5	4.9	1.4	2.2	2.1	1.5
1400	-	5.5*	3.5	4.2	1.4	3.7	3.2	2.4
1600	-	5.4†	3.5	4.0	1.2	5.2	4.5	3.8
1800	-	5.7	4.0	4.9	1.9	-	5.8	4.5
2000	-	6.2	4.7	5.5	-	-	-	5.4
2200	-	-	5.0	5.8	-	-	-	6.2
2400	-	-	5.3	6.0	-	-	-	7.3

* 1650 rev/min

† 1700 rev/min

** Load 1+5 kw

EXPERIMENTAL RESULTS (VARIABLE ENGINE SPEED)

TABLE 8.2

<u>Boost</u> <u>Ratio</u>	<u>Scavenge</u> <u>Ratio *</u>	<u>Smoke</u> <u>Emission (Bosch)</u>	<u>Power</u> <u>kw</u>
------------------------------	-----------------------------------	-----------------------------------------	---------------------------

Holset 3LD1 1.57

2.06	1.30	2.2	59.82
1.95	1.30	2.7	59.04
1.86	1.20	2.9	58.66
1.75	1.20	3.2	58.27
1.70	1.10	3.5	57.23
1.62	1.10	4.4	56.07
1.55	1.10	5.0	54.77
1.46	1.10	6.1	53.35
1.39	-	7.0	51.02

Holset 3LD1 2.5

1.85	1.20	2.1	57.62
1.71	1.20	2.7	56.32
1.62	1.20	2.9	55.68
1.53	1.20	3.3	54.90
1.47	1.20	4.4	53.09
1.40	1.20	5.0	51.27
1.33	-	6.1	48.56
1.28	-	7.0	45.32

Holset 4LGK 2.6 T2

1.76	1.30	1.4	57.49
1.70	1.30	1.5	57.1
1.63	1.25	1.6	56.71
1.57	1.20	1.9	55.94
1.51	1.20	2.4	54.90
1.46	1.20	2.7	53.22
1.41	1.10	3.4	51.40
1.36	1.10	4.3	49.20
1.31	1.10	5.1	46.22

* Computer program estimation.

EXPERIMENTAL RESULTS CONSTANT SPEED (1200 Rev/min)

TABLE 9,1

	DDE Dawson	DCE Wallace	VCR Timoney	Recircu- lation Tryhorn	Hyperbar	Blencoe	DGT (Theoretical)
Two or four stroke	4	2	2	2	4	2	2
Power output of standard engine	+++	++++	++++	++	+++	---	+
Torque character- istic of ideal hyperbola	+++	+++	+++	++	----	---	++
Complexity of exhaust ducting	++++	----	++	++	++	----	--
Complexity of transmission	+++	+++	++	++	++	----	++
Mechanical complexity	----	----	--	++	++	++	+++
Control complexity	++++	---	--	++++	--	---	----
Modification to engine structure	--	++	----	++++	----	++++	++++
Use of noise generating components (gears & blowers)	----	----	--	--	++	--	++++
Auxiliary combustion	NO	YES	NO	NO	YES	YES	YES
Air coolers	NO	YES	YES	NO	YES	NO	YES

++++ Very good

---- Very bad

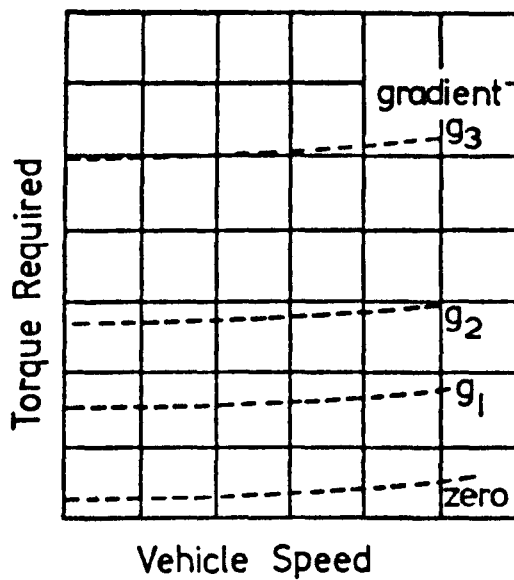
COMPARISON OF ENGINE SYSTEMS

	DGT	HYPERBAR
Boost ratio	2.3:1 (max)	8:1 *
Compression ratio	16:1 +	5:1
Engine	2 stroke	4 stroke
Minimum air/fuel ratio	22:1 (proposed)	30-35:1
Auxiliary combustion	Clean air from bypass	Engine exhaust + clean air from bypass

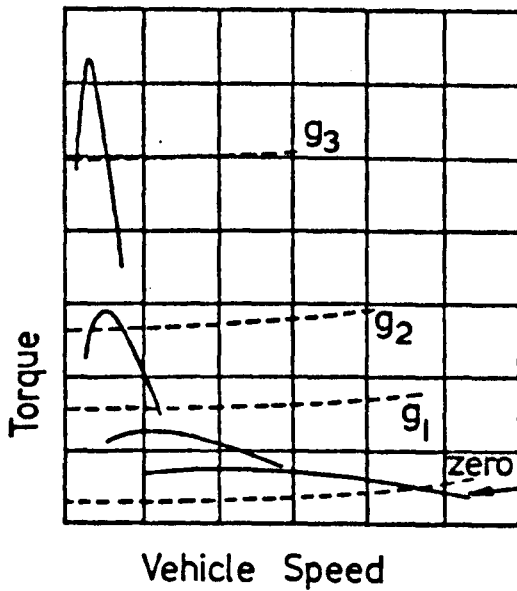
* The Hyperbar system utilises two stage turbocharging.

+ Standard engine compression ratio.

[Timoney (2)]

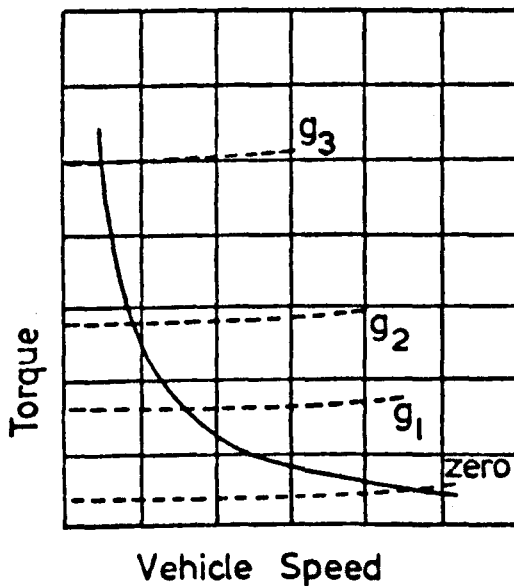


(a) VEHICLE TORQUE REQUIREMENT ON VARYING GRADIENTS



(b) CONVENTIONAL ENGINE TORQUE MATCH

(Torque curves shown for transmission with 4 speed gear box)

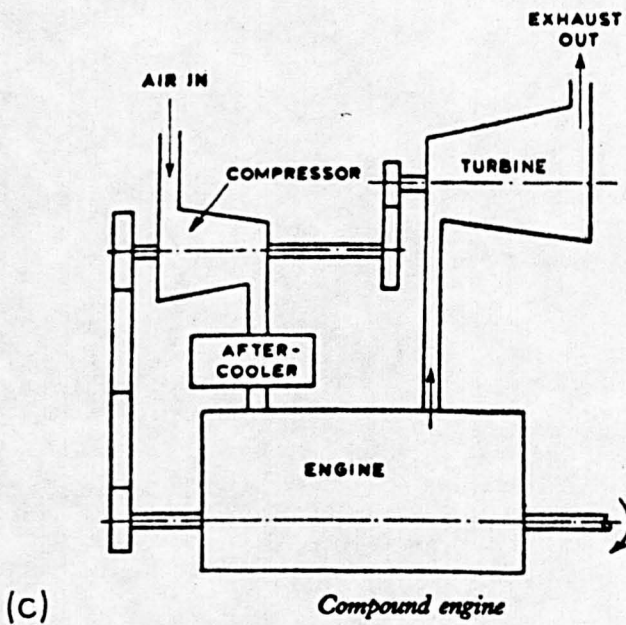
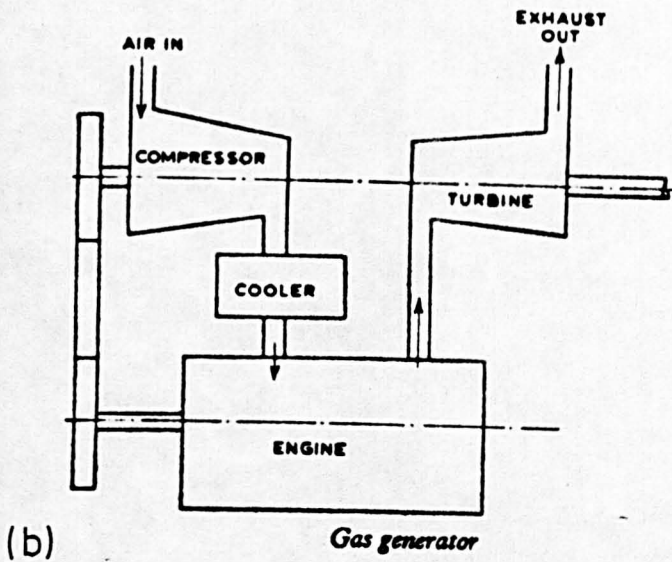
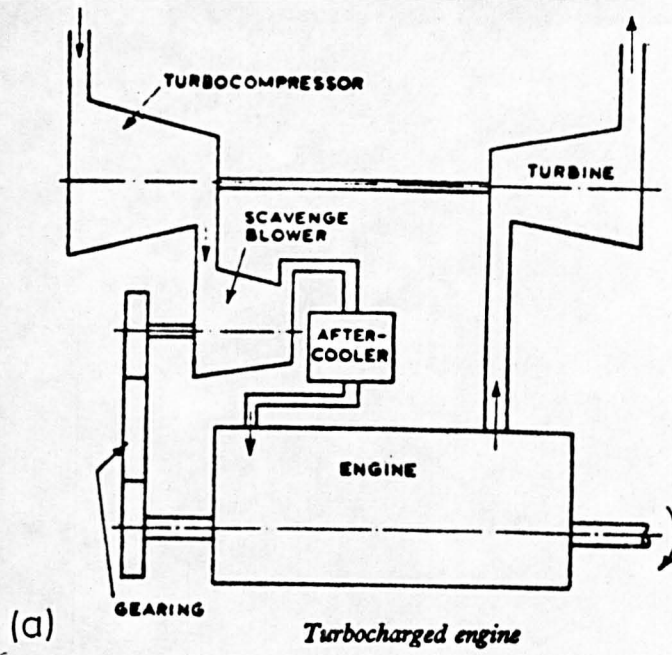


(c) TORQUE MATCH OF CONSTANT POWER OUTPUT ENGINE

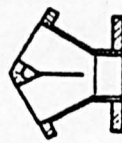
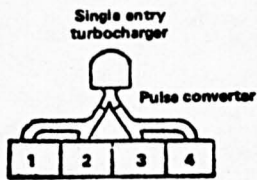
(Torque curve shown for constant horse power)

FIGURE 1.2

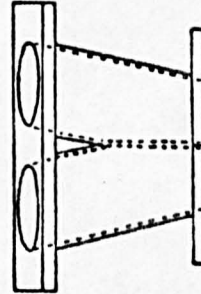
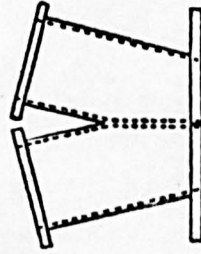
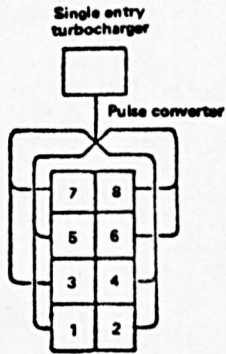
[Wallace (4)]



SUPERCHARGED ENGINE SYSTEMS

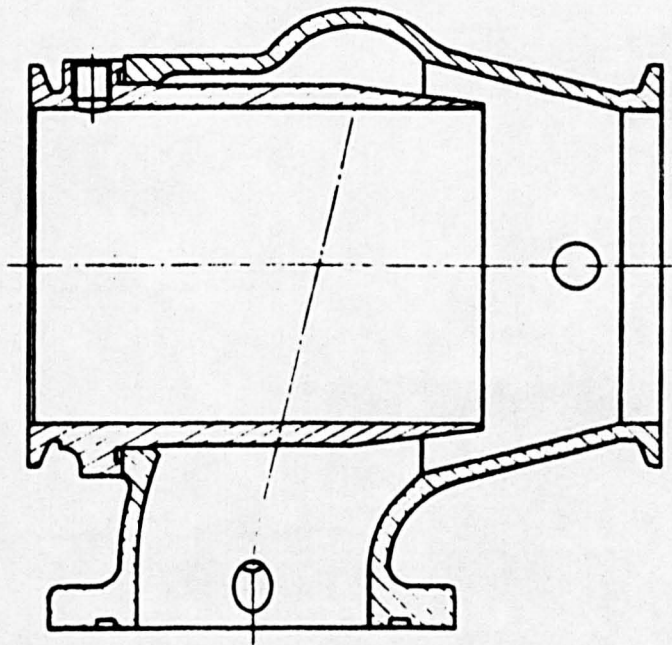


Twin entry bifurcated pulse converter



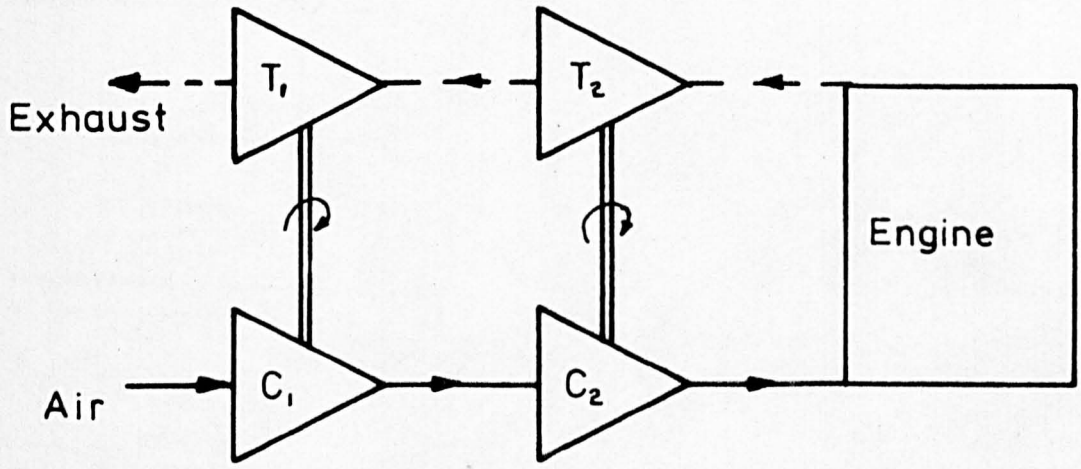
Four entry pulse converter

(a) Typical pulse converters and engine arrangements
[Watson & Holness (29)]

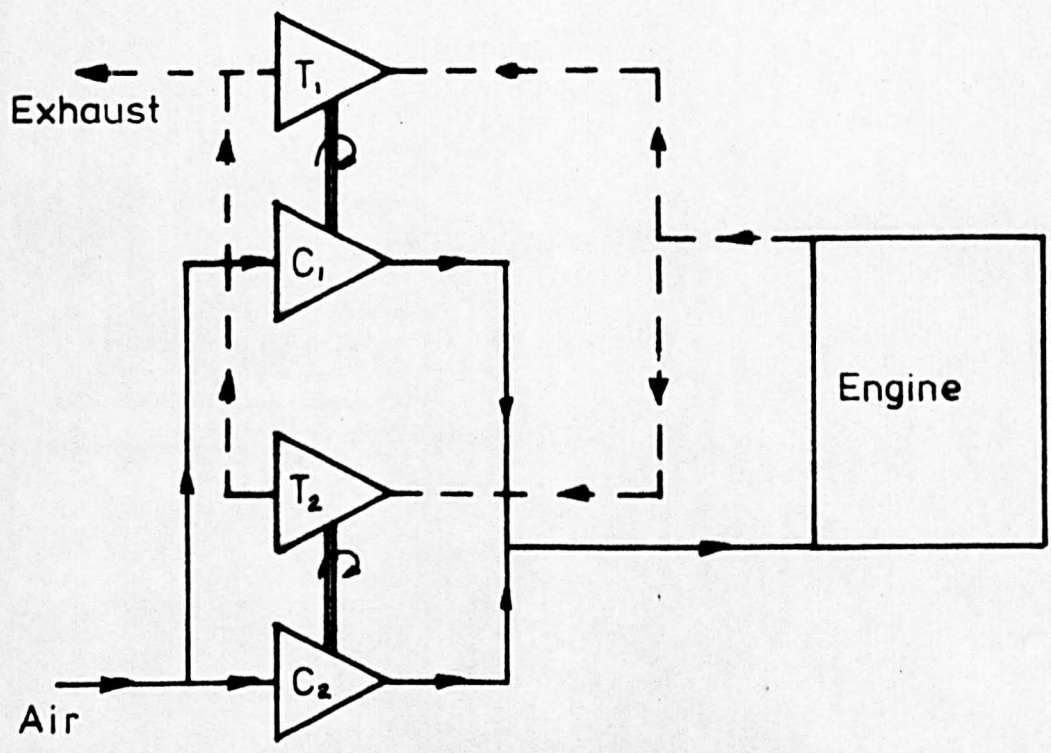


(b) Modular pulse converter
[Curtis & Magnet (27)]

PULSE CONVERTERS



(a) Two Stage (Series) Turbocharging

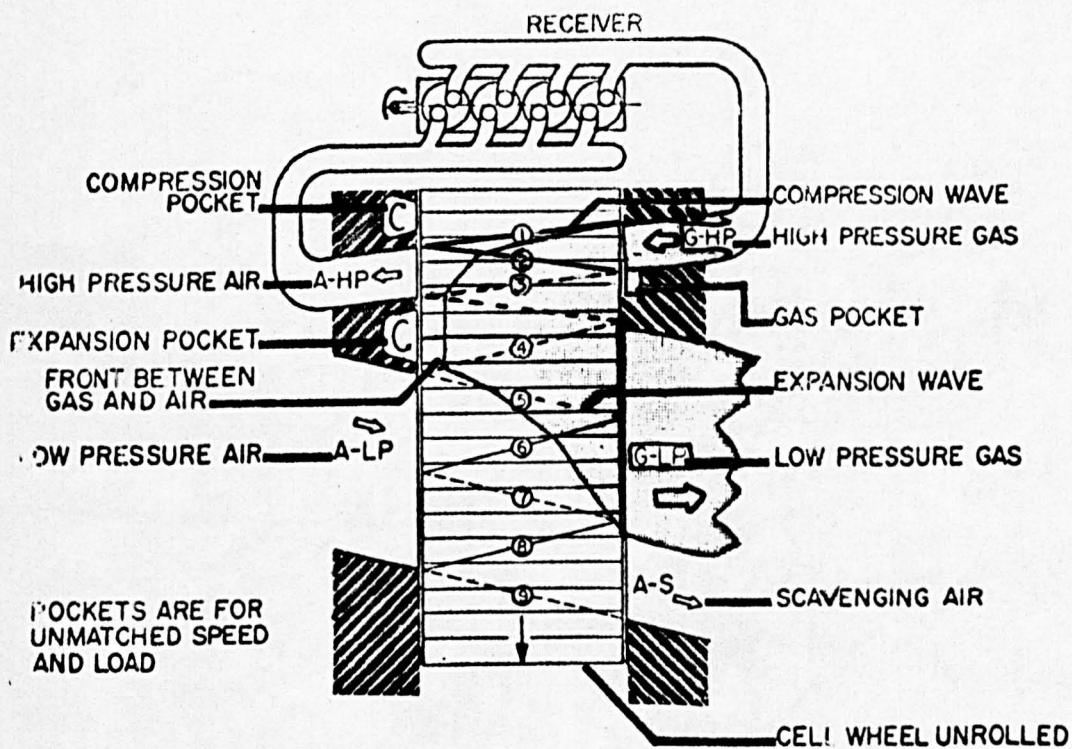
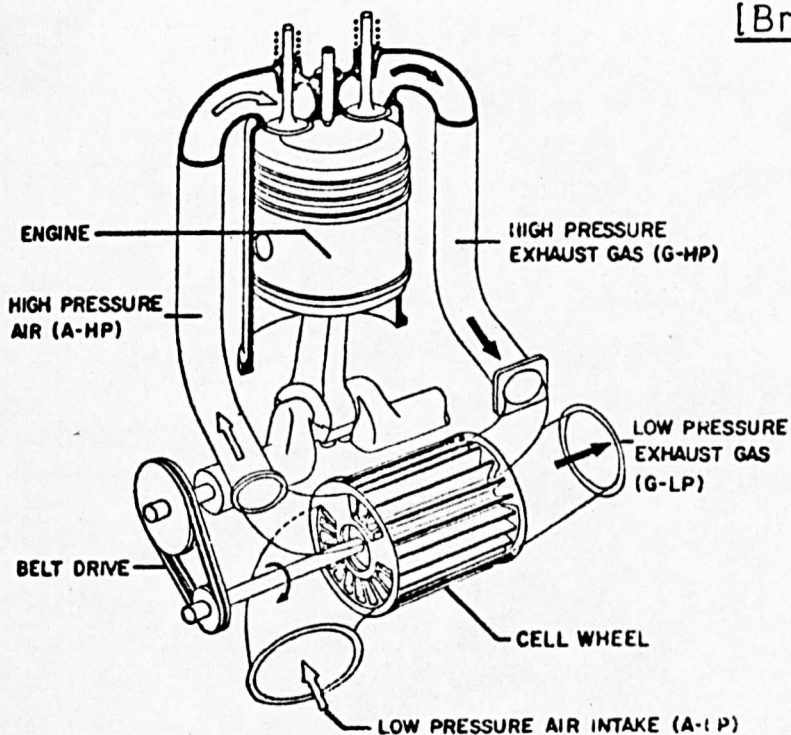


(b) Register (Parallel) Turbocharging

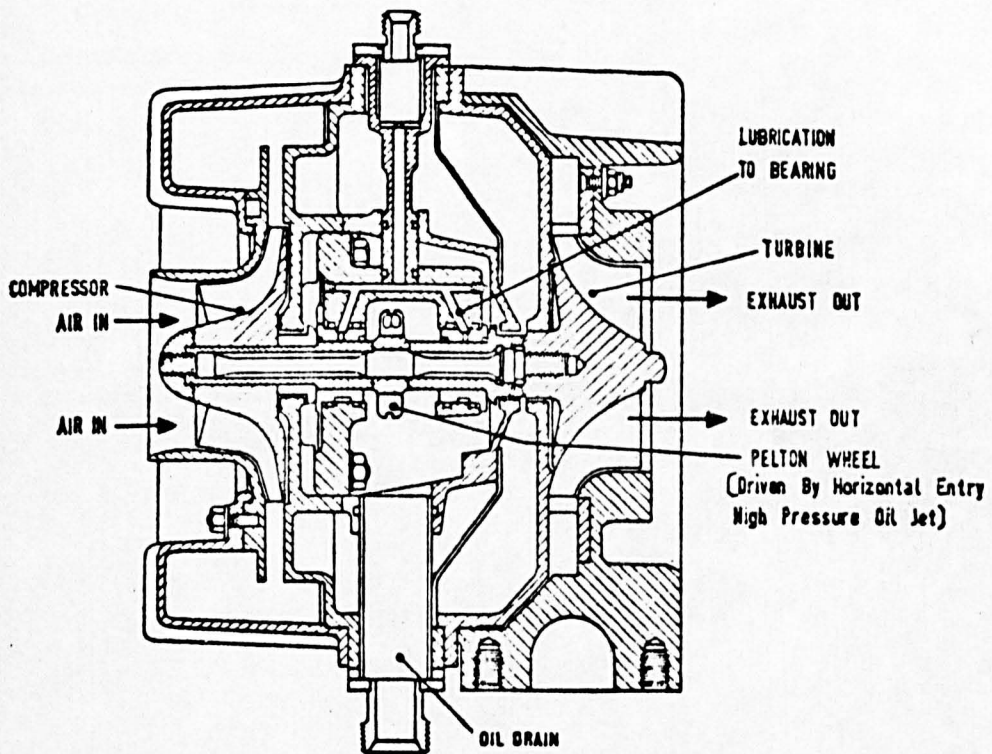
SERIES & PARALLEL TURBOCHARGING

FIGURE 1,5

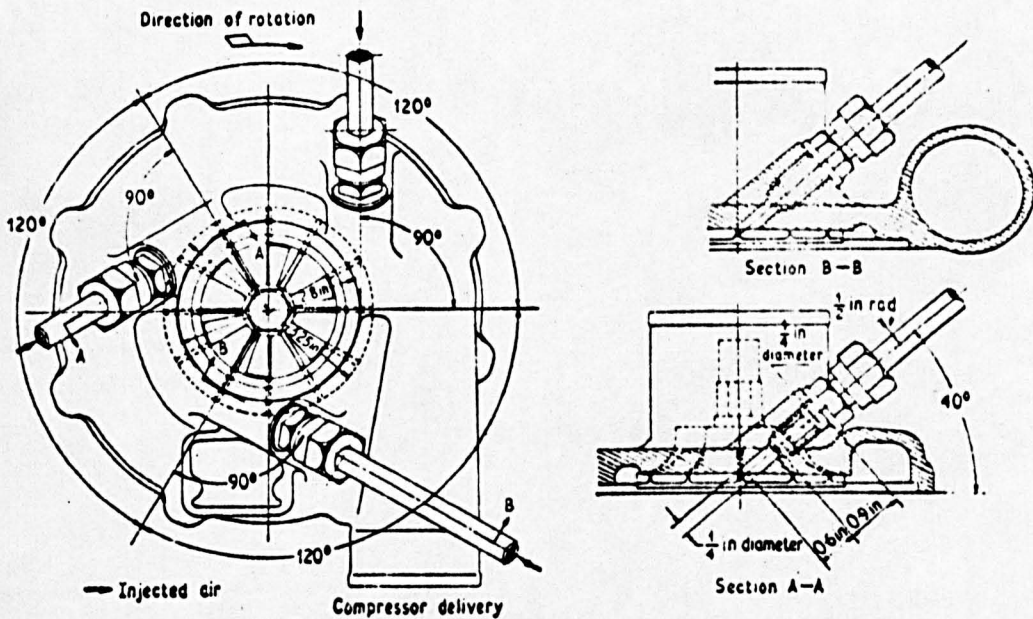
[Brown Boveri (32)]



BBC COMPLEX



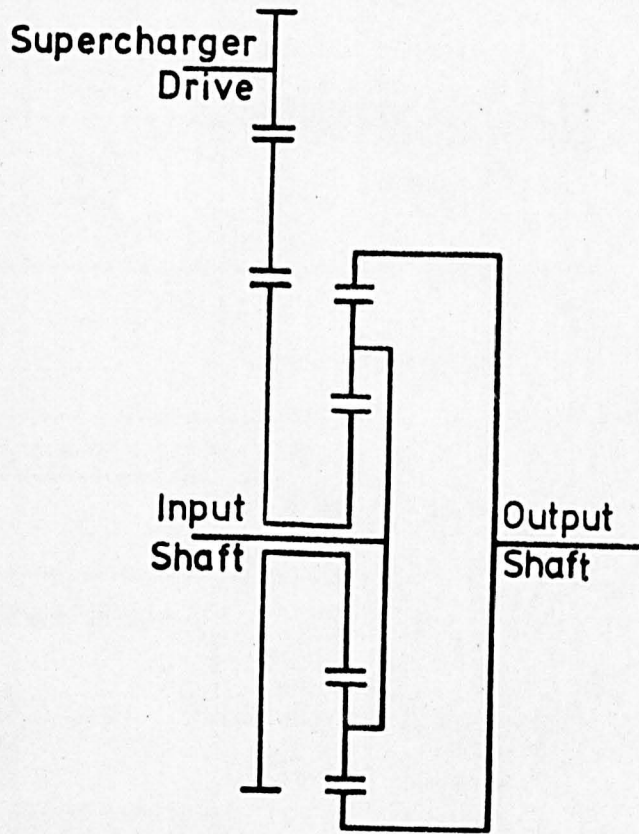
(a) *Arrangement of Pelton wheel drive on turbo-charger shaft*
 [Timoney (2)]



(b) *Compressor with injection nozzles*
 [Ledger, Benson & Furukawa (41)]

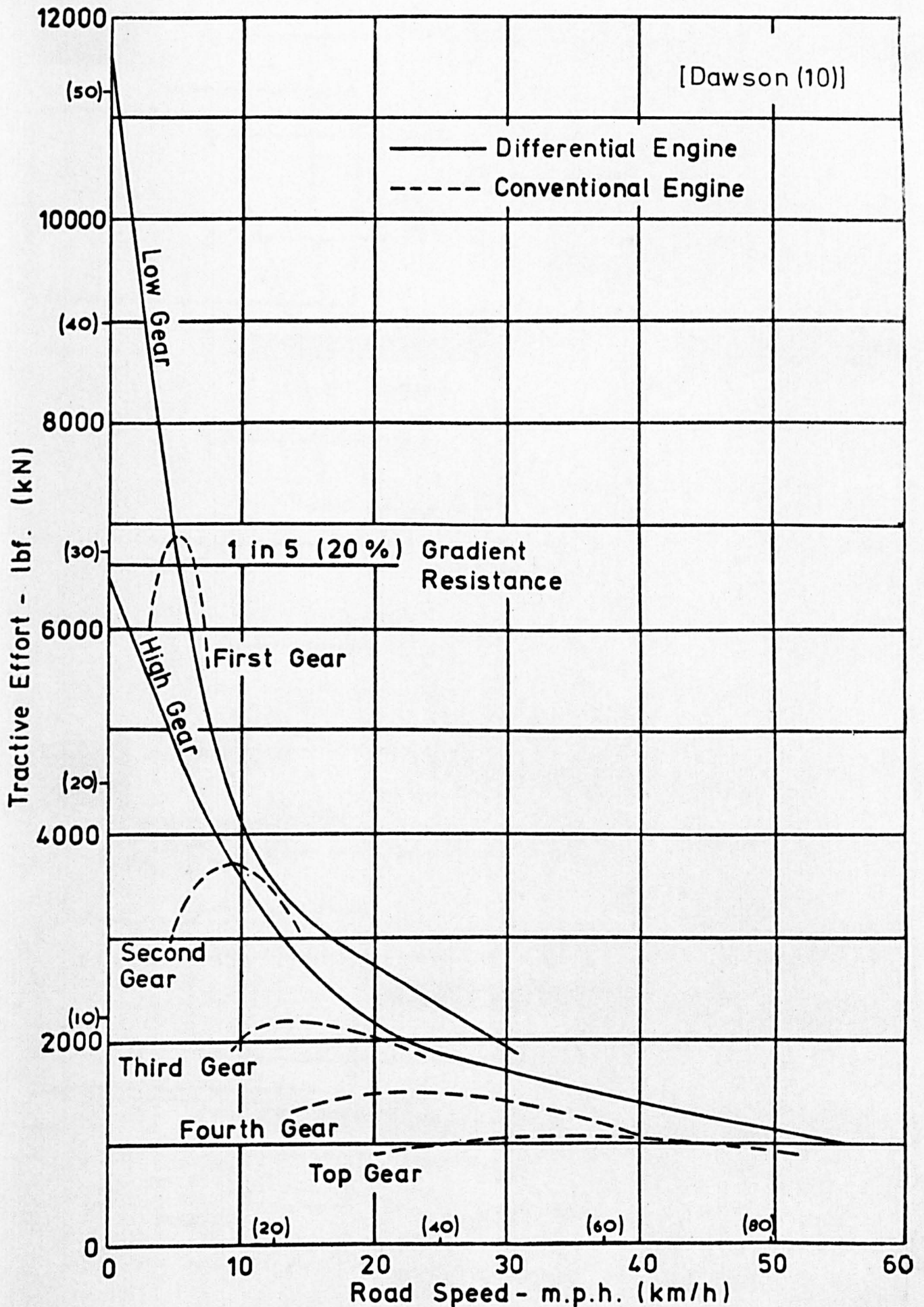
TURBOCHARGER ASSISTANCE

FIGURE 2.1



DIFFERENTIAL DIESEL DRIVE ARRANGEMENT (DAWSON) (10)

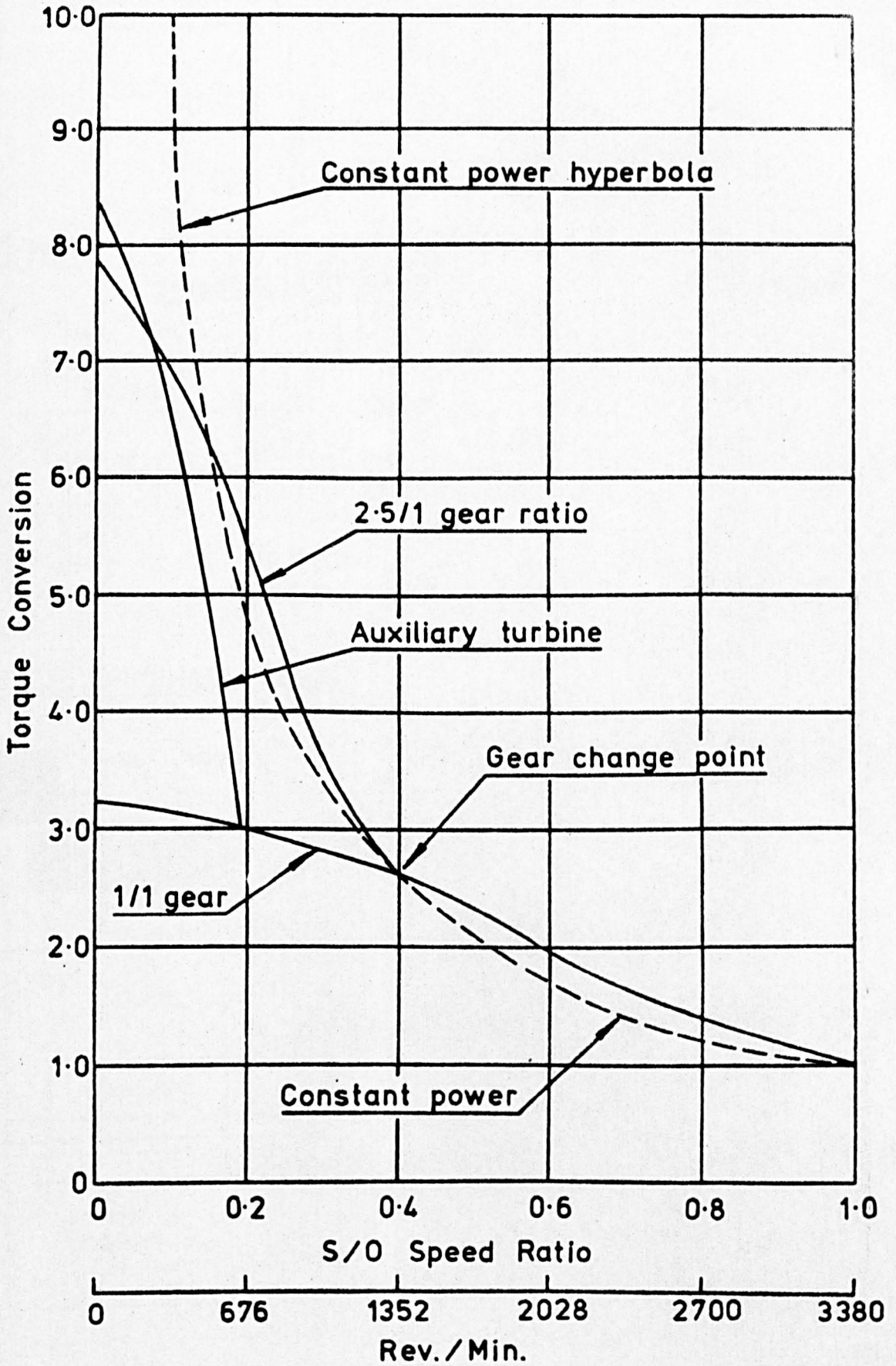
FIGURE 2.2



D.D.E. PERFORMANCE CHARACTERISTICS

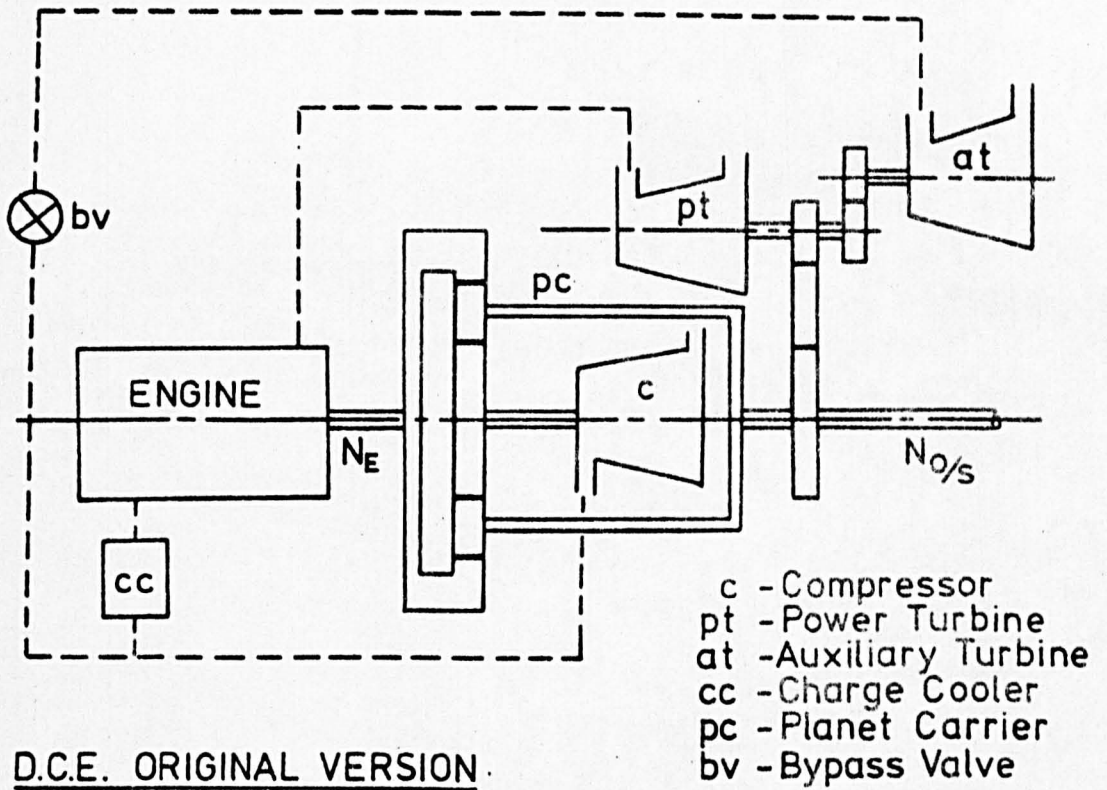
FIGURE 2,3

(Wallace)

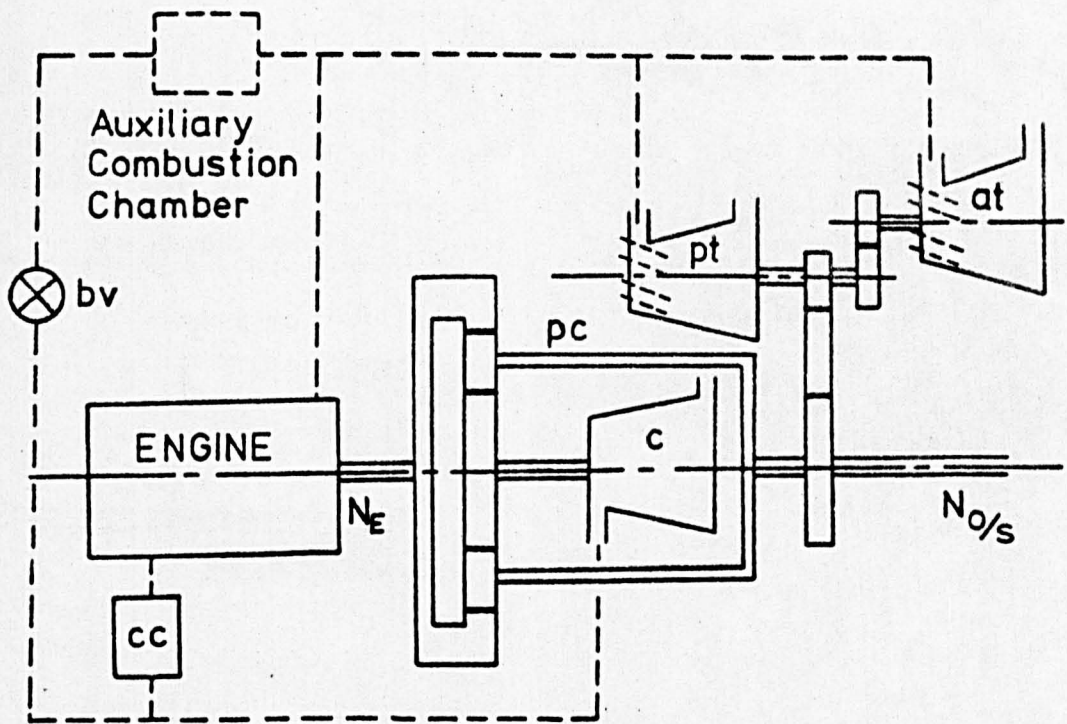


D.C.E. TORQUE CHARACTERISTICS

(Wallace)

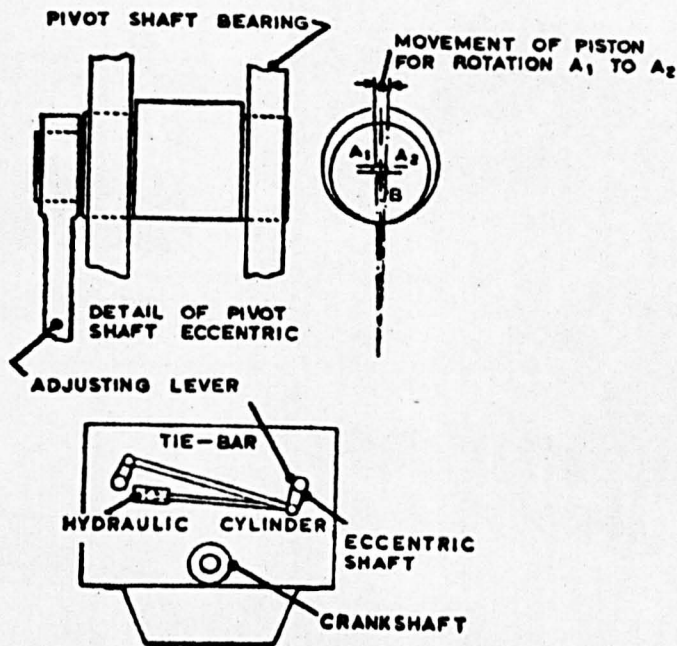
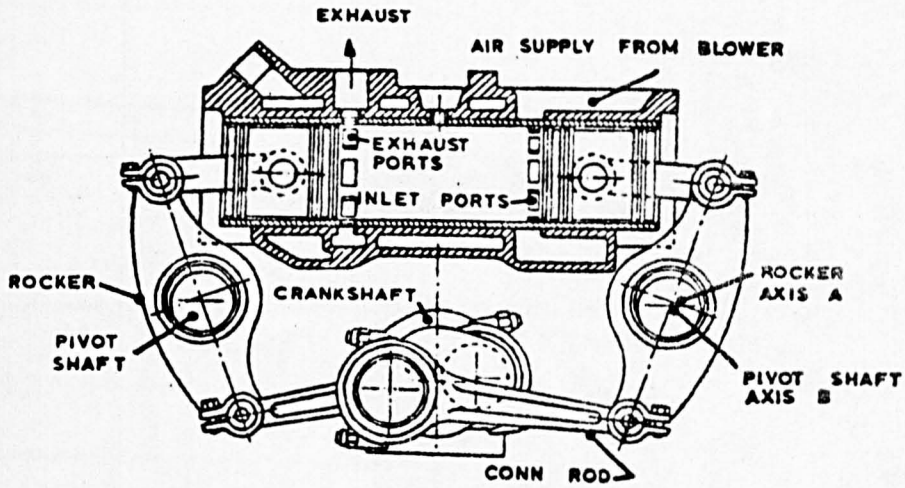


(a) D.C.E. ORIGINAL VERSION.



(b) D.C.E. WITH VARIABLE GEOMETRY TURBINES

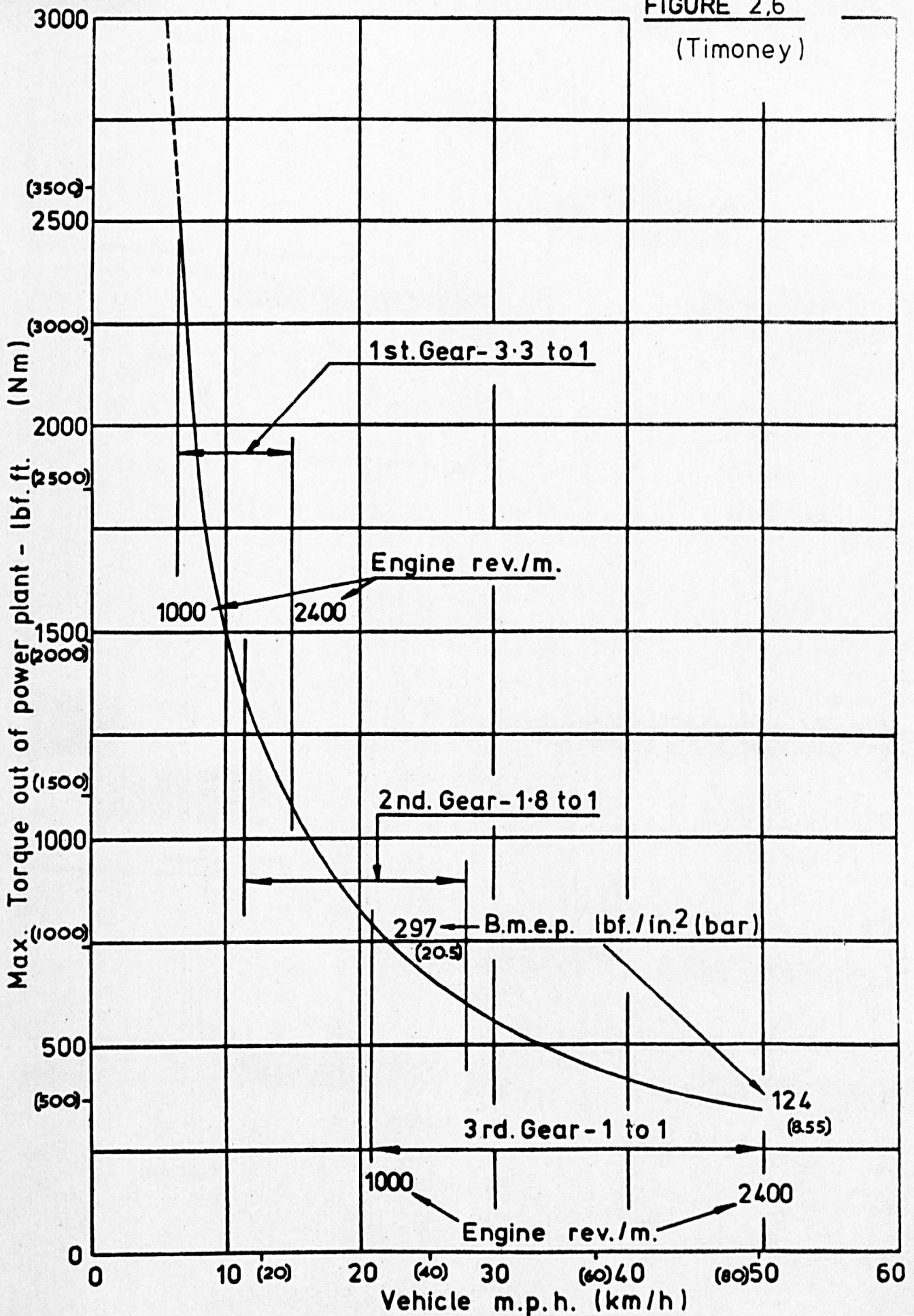
FIGURE 2.5
(Timoney)



VARIABLE COMPRESSION RATIO ENGINE

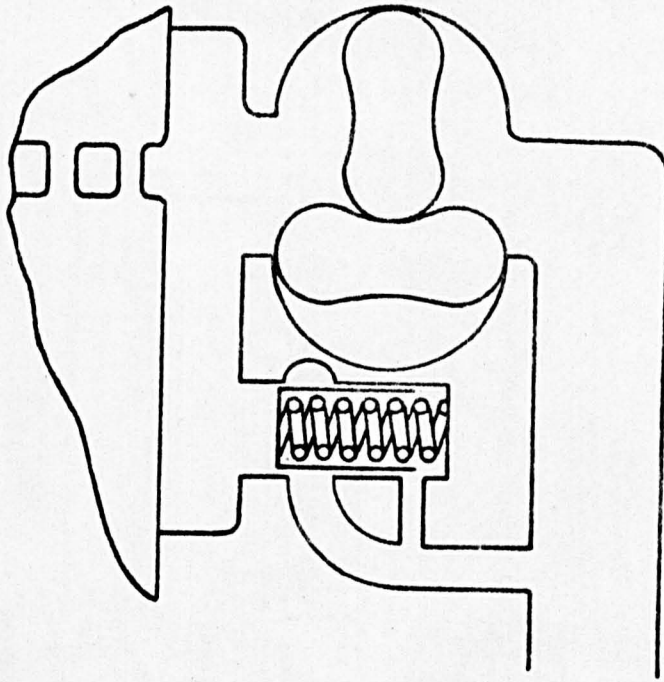
FIGURE 2.6

(Timoney)



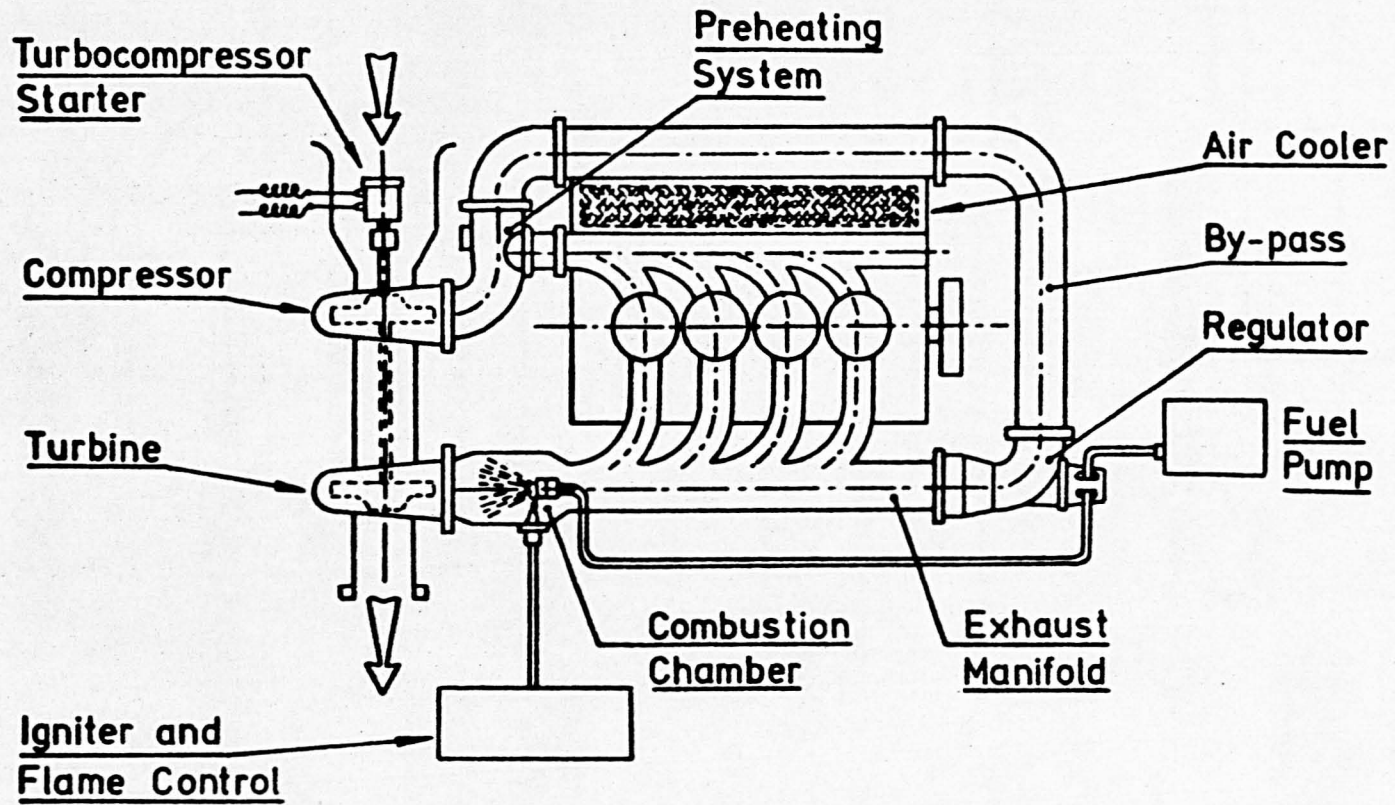
VARIABLE COMPRESSION RATIO ENGINE TORQUE CHARACTERISTICS (With Pelton Wheel assisted Turbocharger drive)

FIGURE 2.7



ROOTS BLOWER WITH RECIRCULATION VALVE (TRYHORN) (14)

HYPERBAR SYSTEM



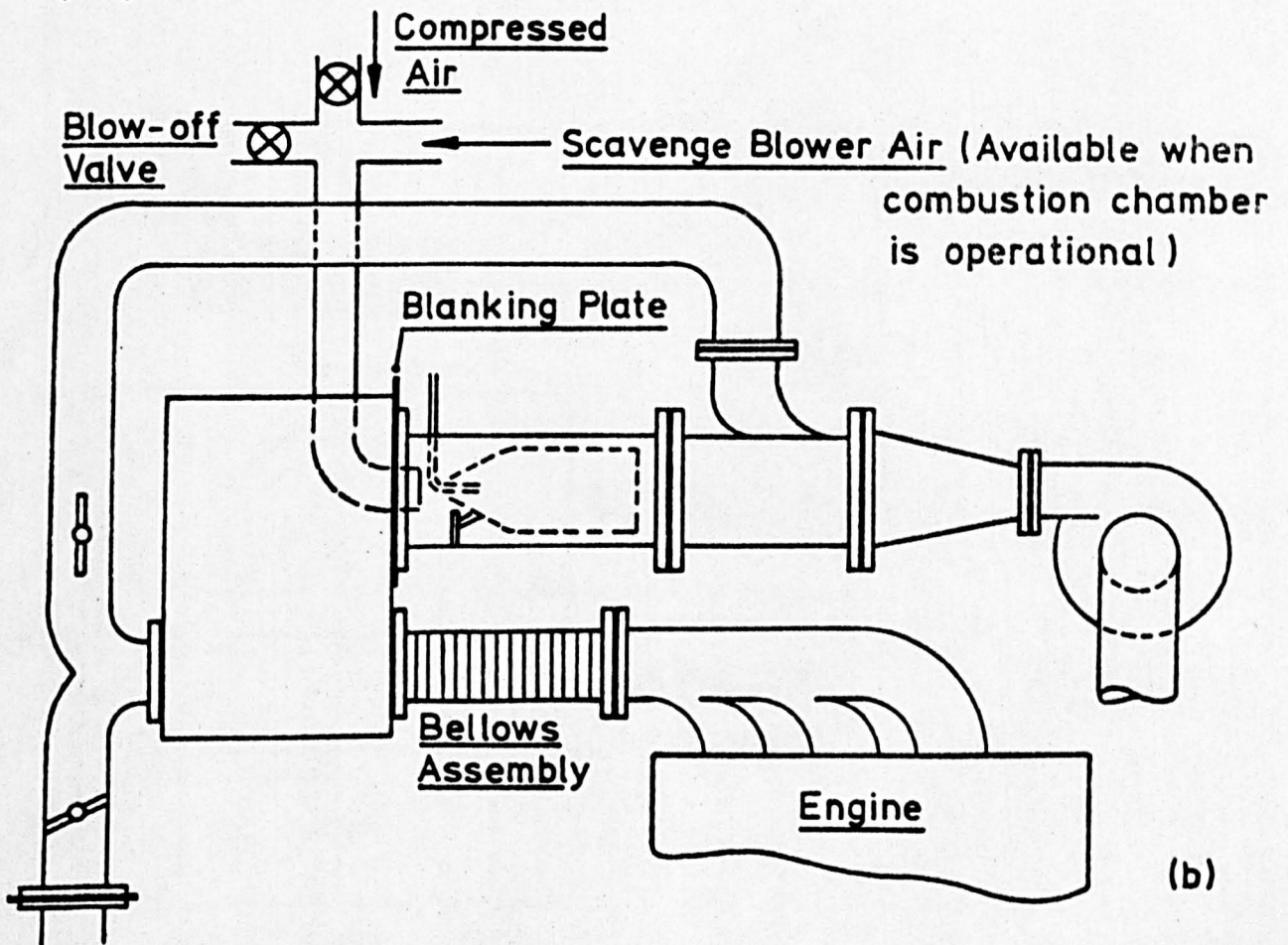
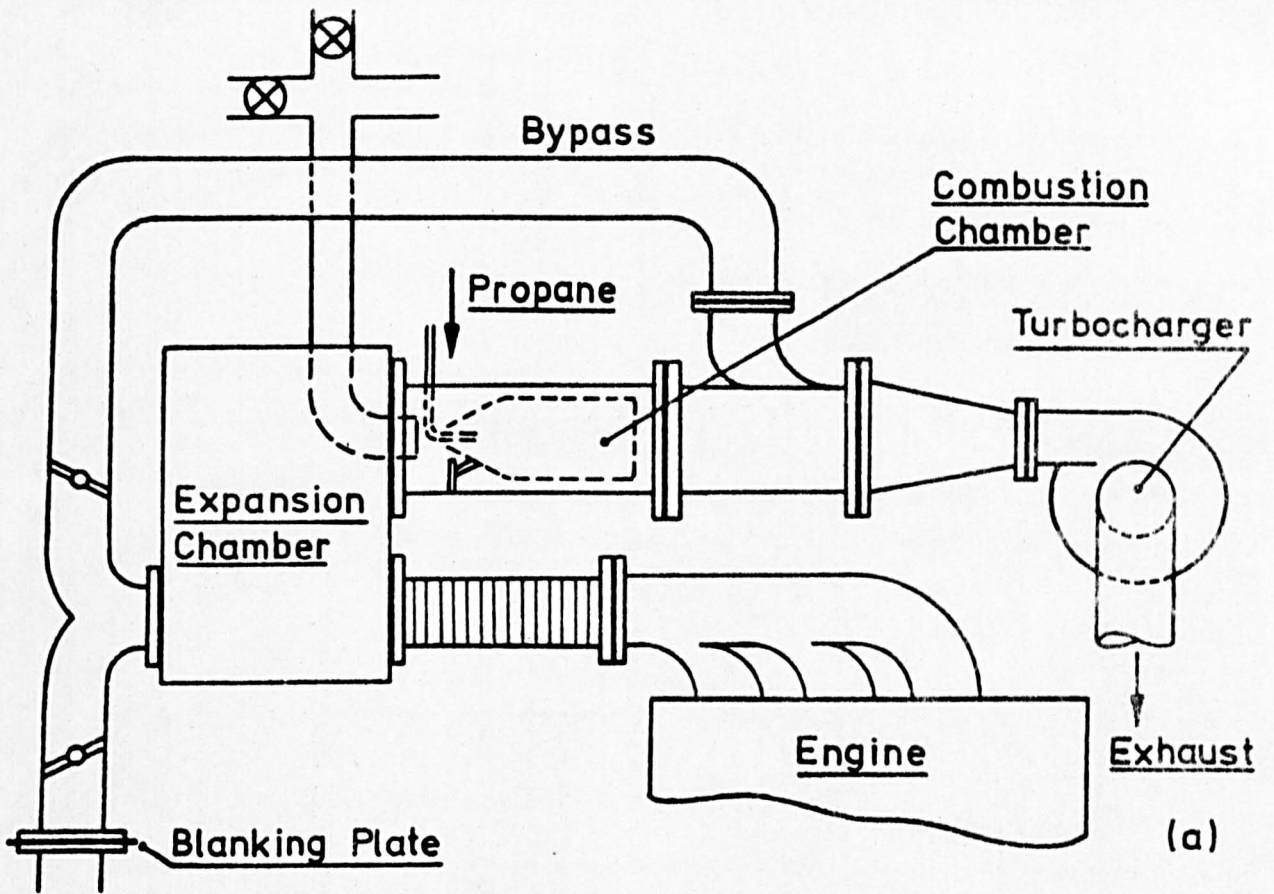
HYPERBAR SYSTEM

[Melchior & Andre-Talamon (15)]

FIGURE 2.8

FIGURE 2.9

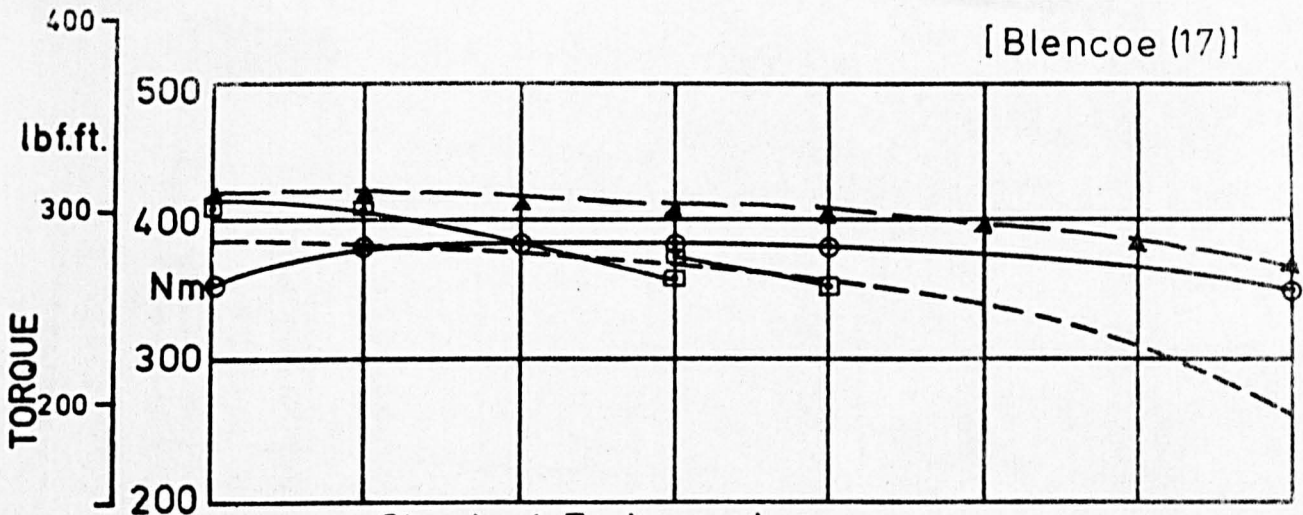
[Blencoe (17)]



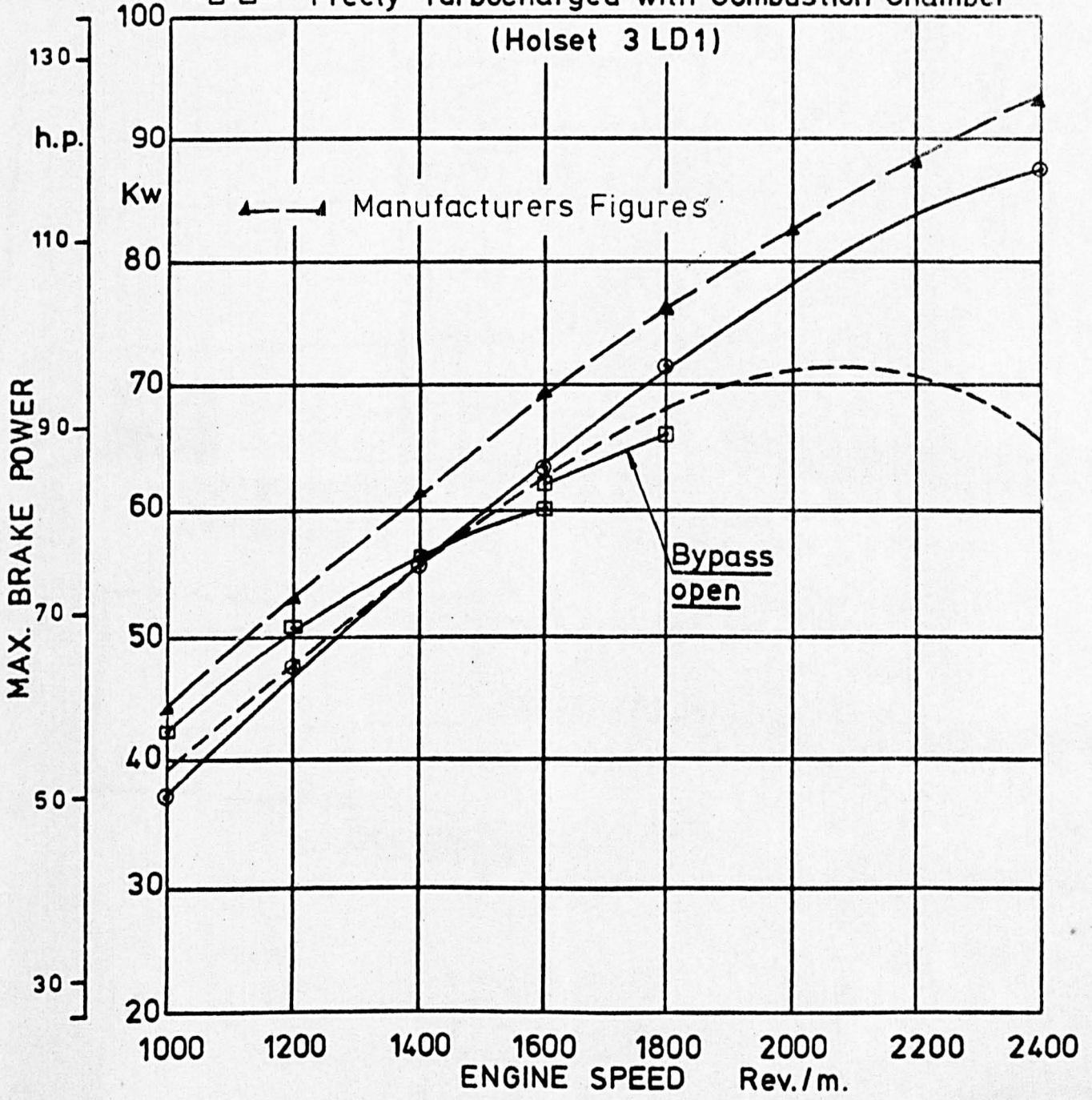
ARRANGEMENT OF EXHAUST GAS FLOW DUCTING

FIGURE 2.10

[Blencoe (17)]

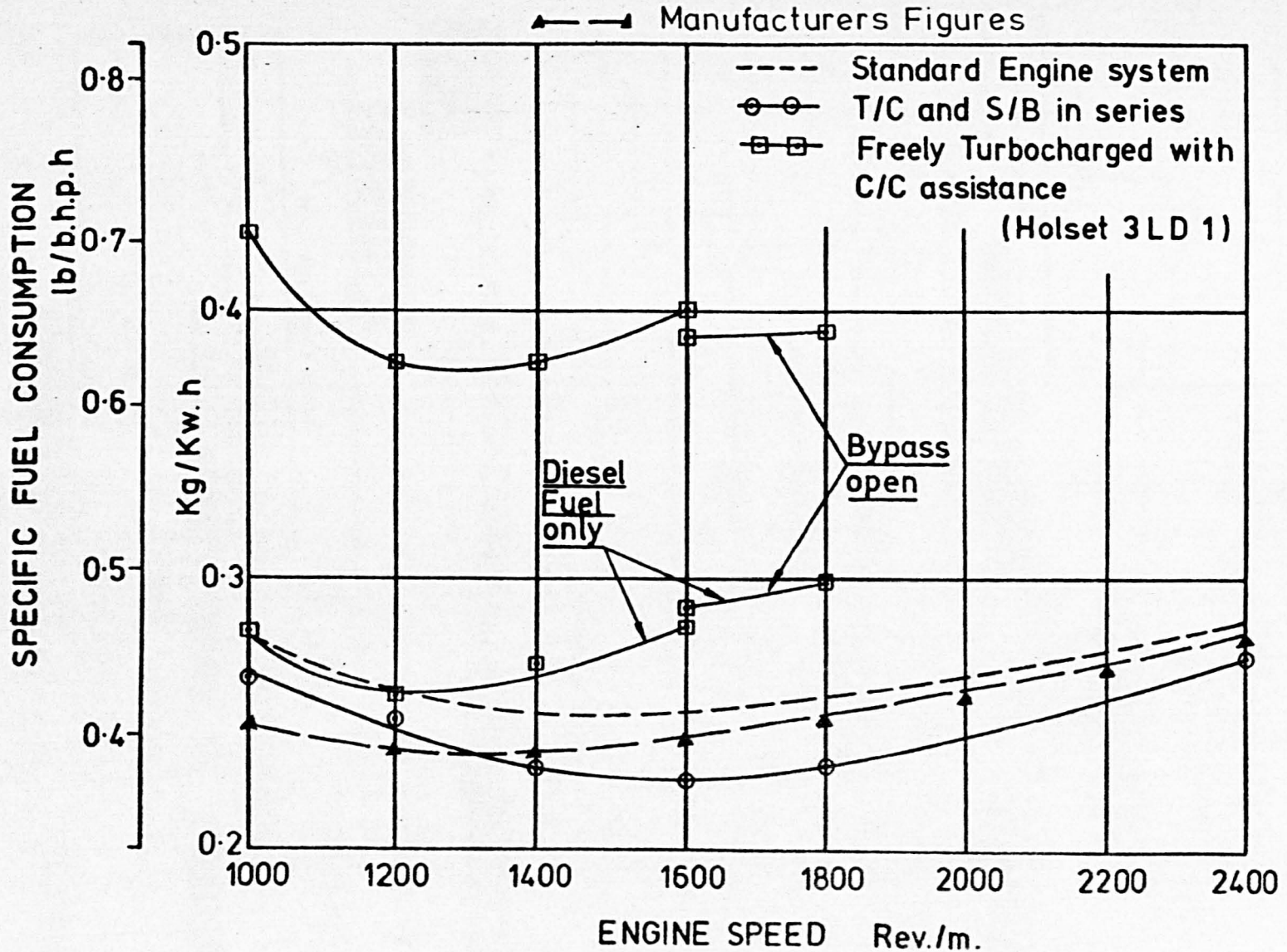


- Standard Engine system
- Turbocharger and Scavenge Blower in series
- Freely Turbocharged with Combustion Chamber



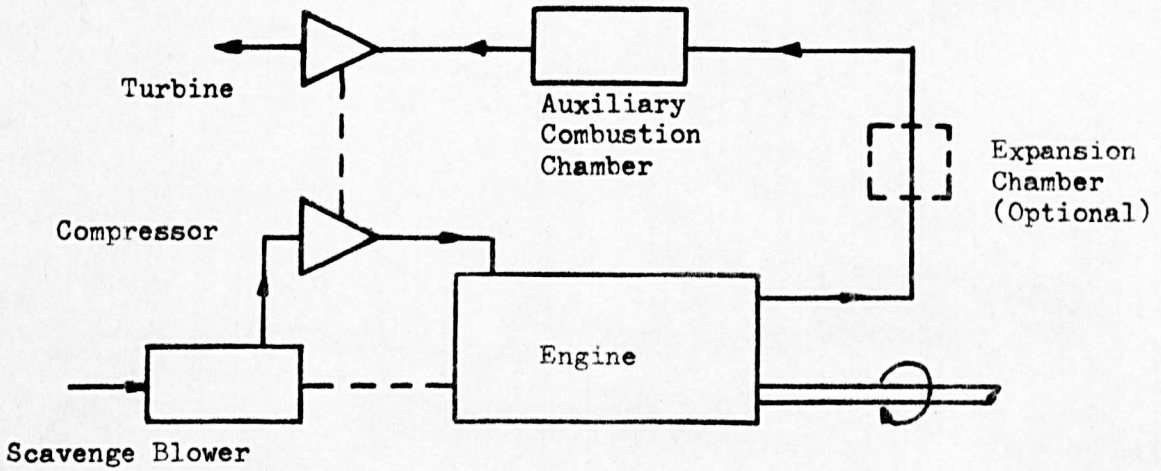
ENGINE PERFORMANCE

SPECIFIC FUEL CONSUMPTION

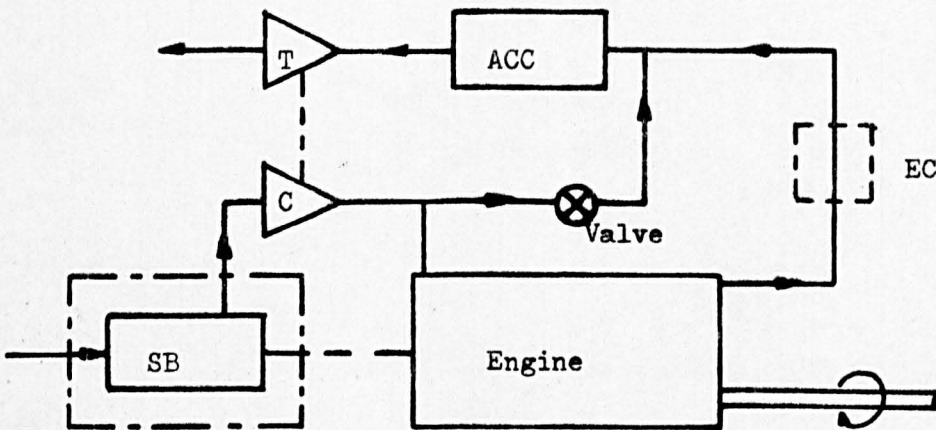


[Blencoe (17)]

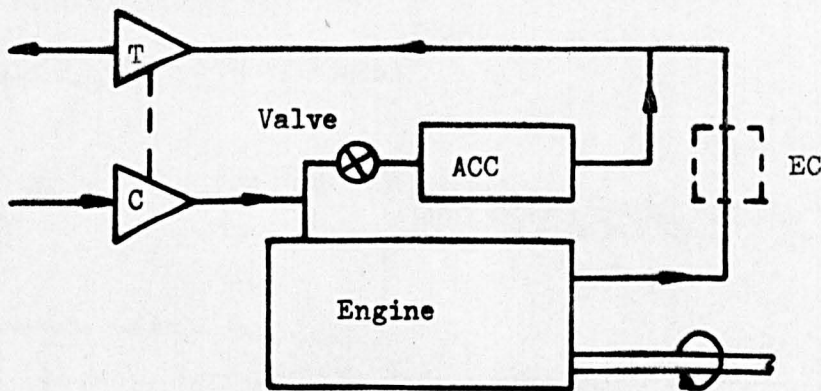
FIGURE 2.11



a) Simple Series System

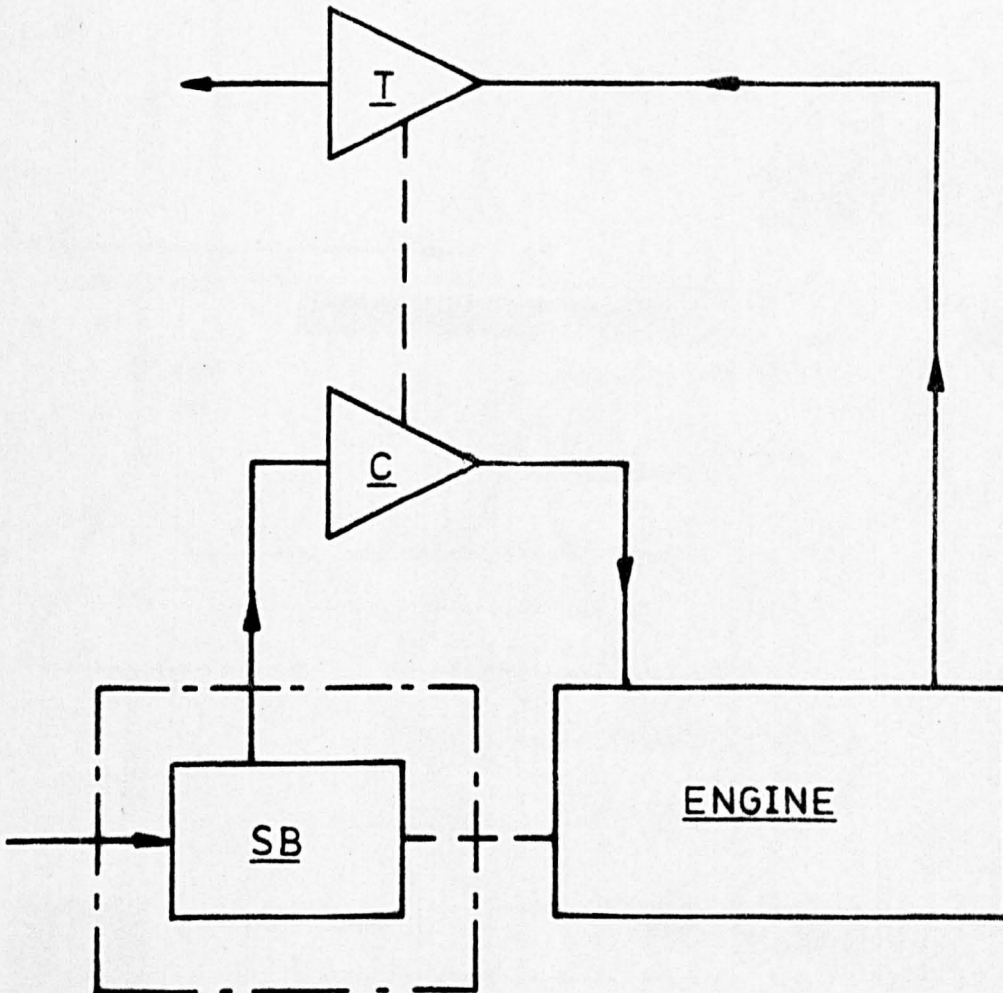


b) System with Throttled Air Bypass



c) System using only clean air for Auxiliary Combustion

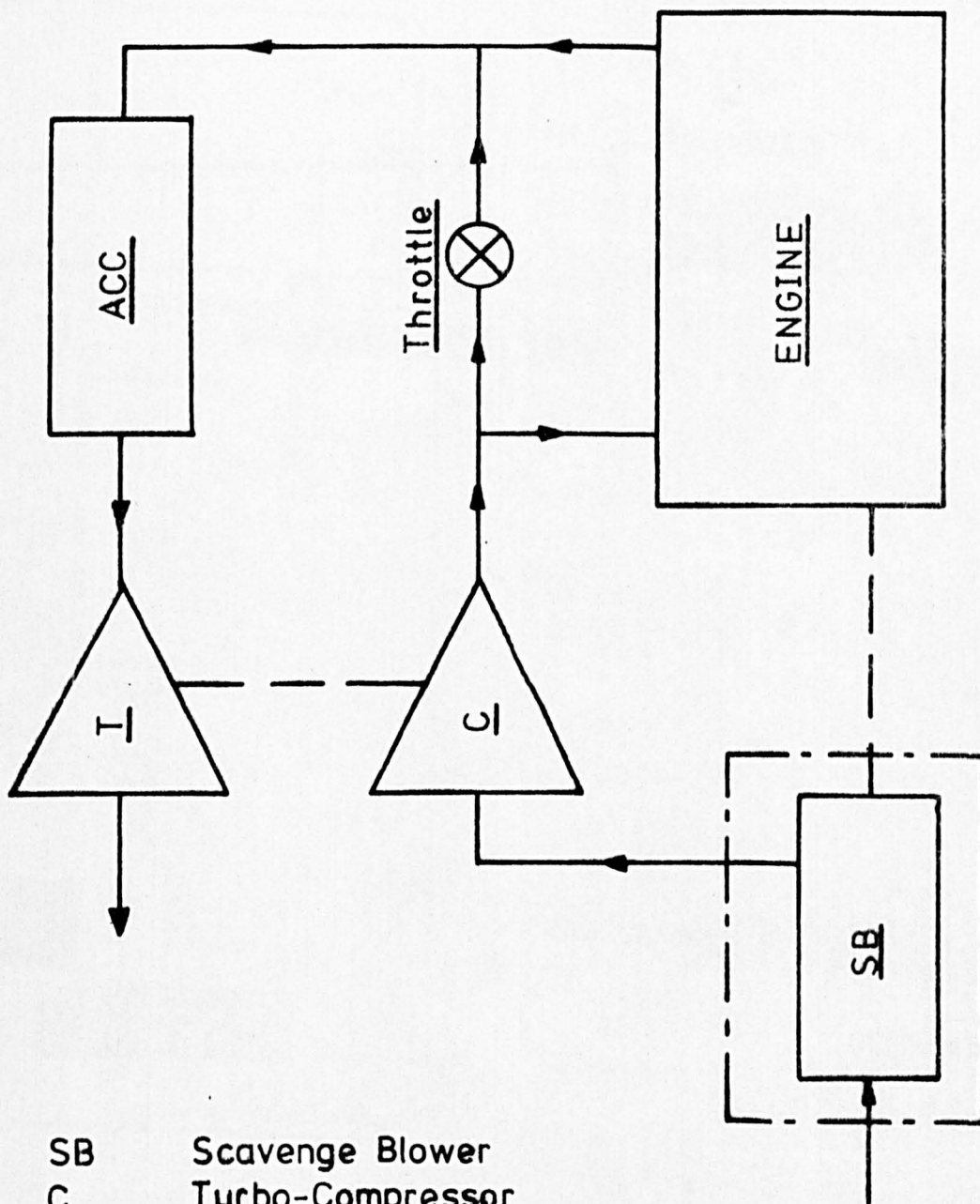
POSSIBLE CONFIGURATIONS OF TWO-STROKE ENGINE, TURBOCHARGER AND AUXILIARY COMBUSTION CHAMBER SYSTEMS



- SB Scavenge Blower
- C Turbo-Compressor
- T Turbine
- - - Mechanical Link
- ▶— Thermodynamic Link

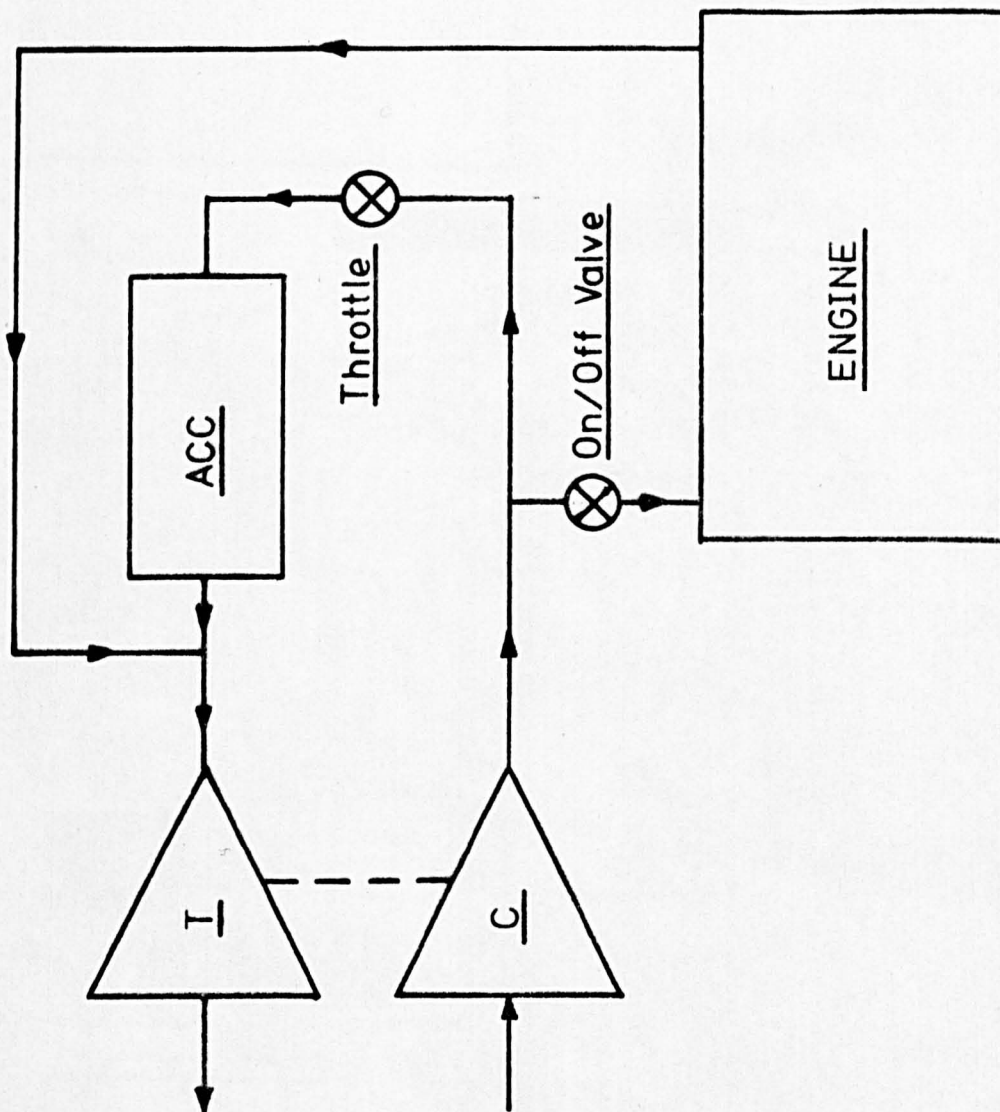
ENGINE SYSTEM: STAGE I DEVELOPMENT

FIGURE 3.4



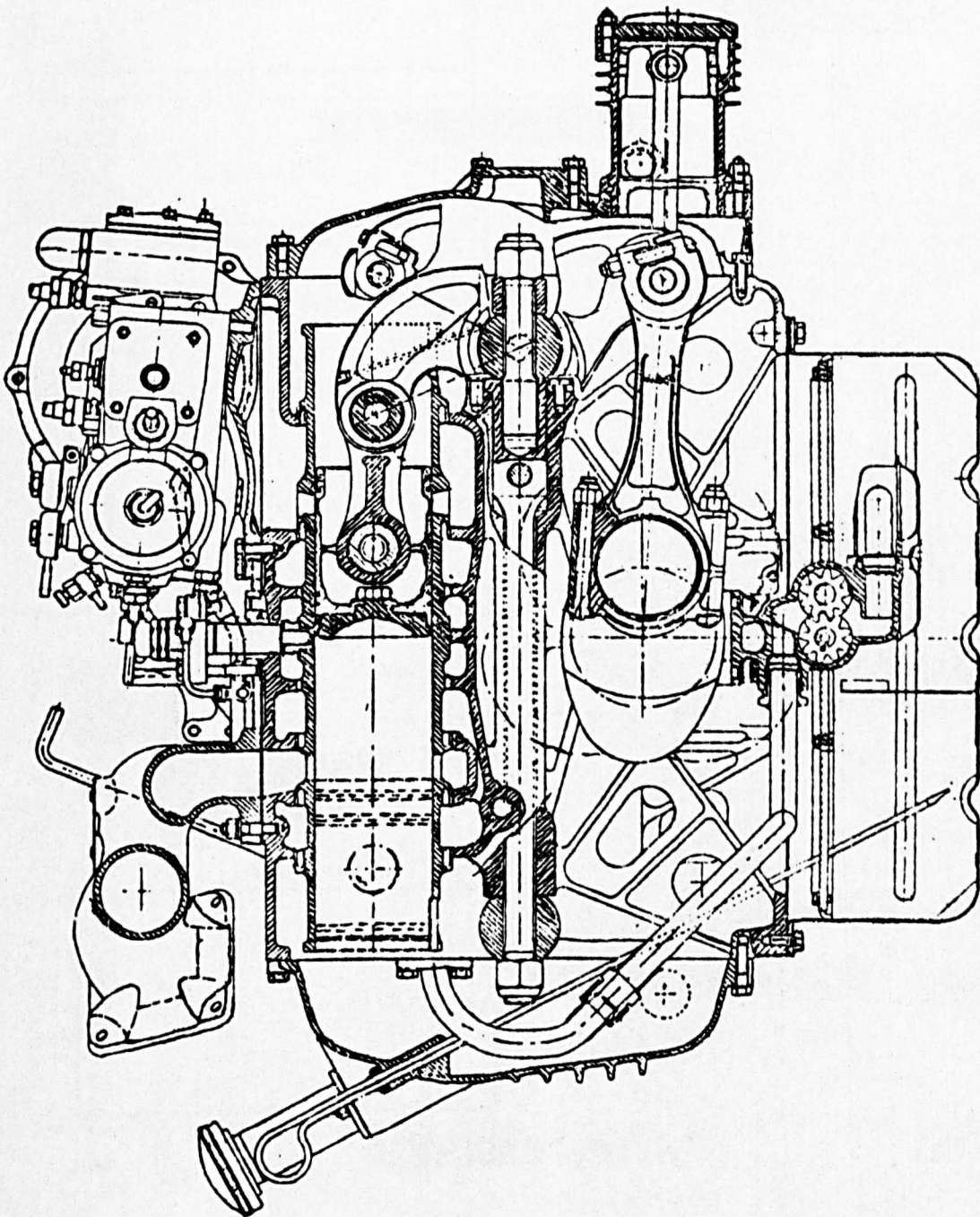
- SB Scavenge Blower
- C Turbo-Compressor
- ACC Auxiliary Combustion Chamber
- T Turbine
- — Mechanical Link
- Thermodynamic Link

ENGINE SYSTEM: STAGE II DEVELOPMENT B



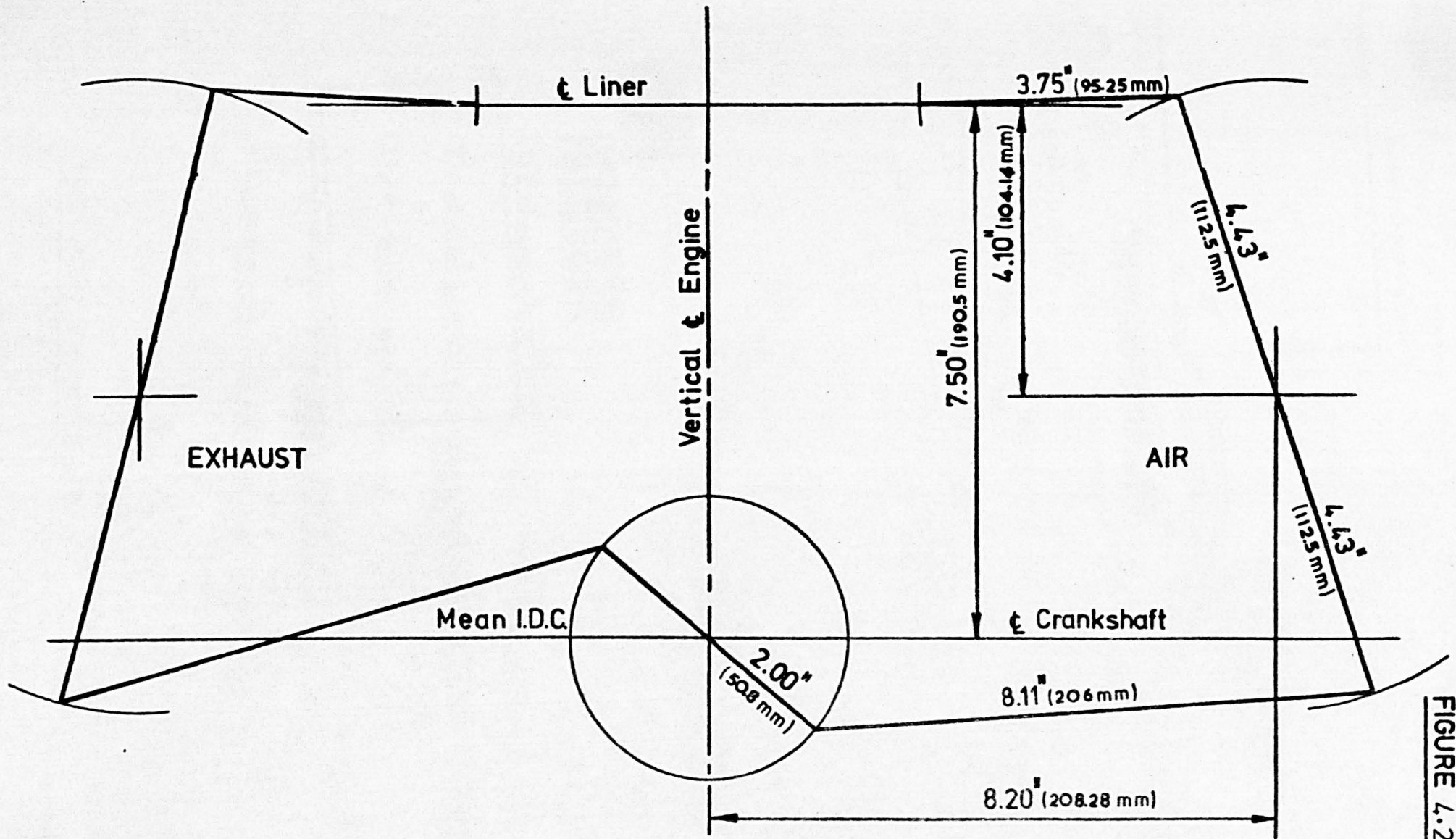
- C Turbo Compressor
- ACC Auxiliary Combustion Chamber
- T Turbine
- Mechanical Link
- Thermodynamic Link

FIGURE 4.1



ROOTES TS 3 ENGINE

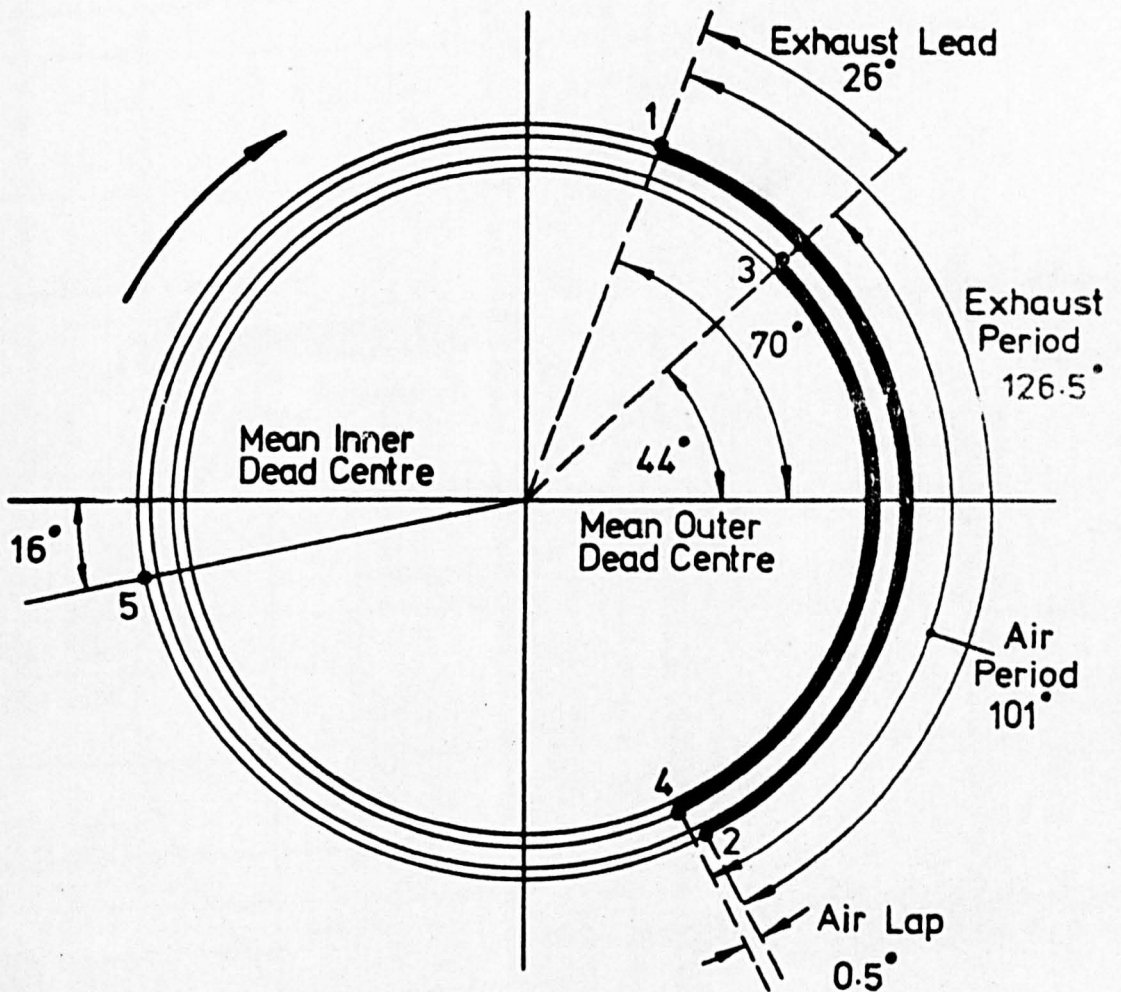
SECTIONED VIEW OF ROOTES TS3 ENGINE



TS 3 ENGINE LINKAGE LAYOUT

FIGURE 4.2

FIGURE 4.3



- (1) Exhaust ports open 110° A.M.I.D.C.
- (2) Exhaust ports close 236.5° A.M.I.D.C.
- (3) Air ports open 136° A.M.I.D.C.
- (4) Air ports close 237° A.M.I.D.C.
- (5) Fuel injection commences 16° B.M.I.D.C.

TS 3 PORT TIMING DIAGRAM

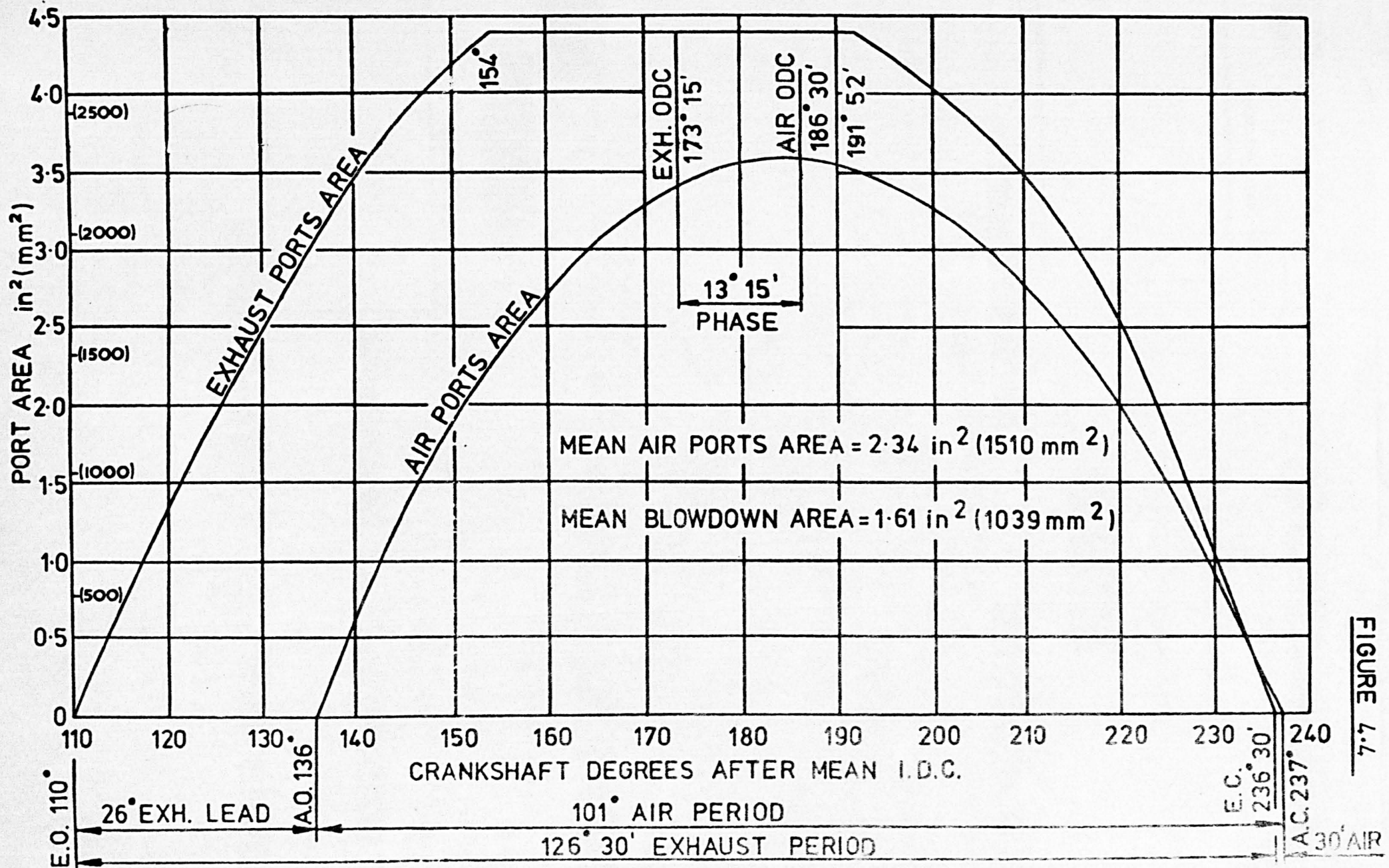
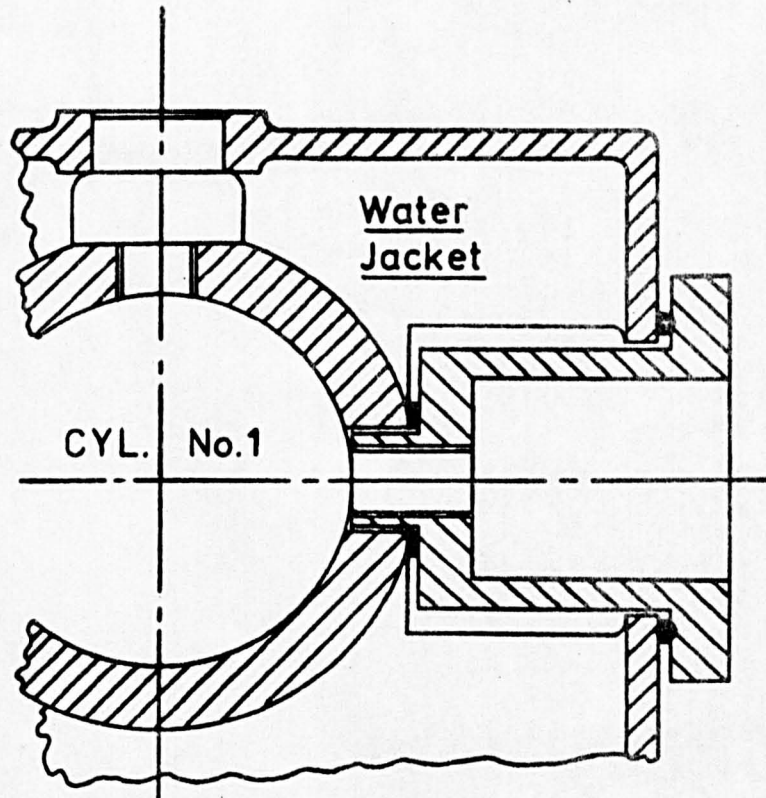


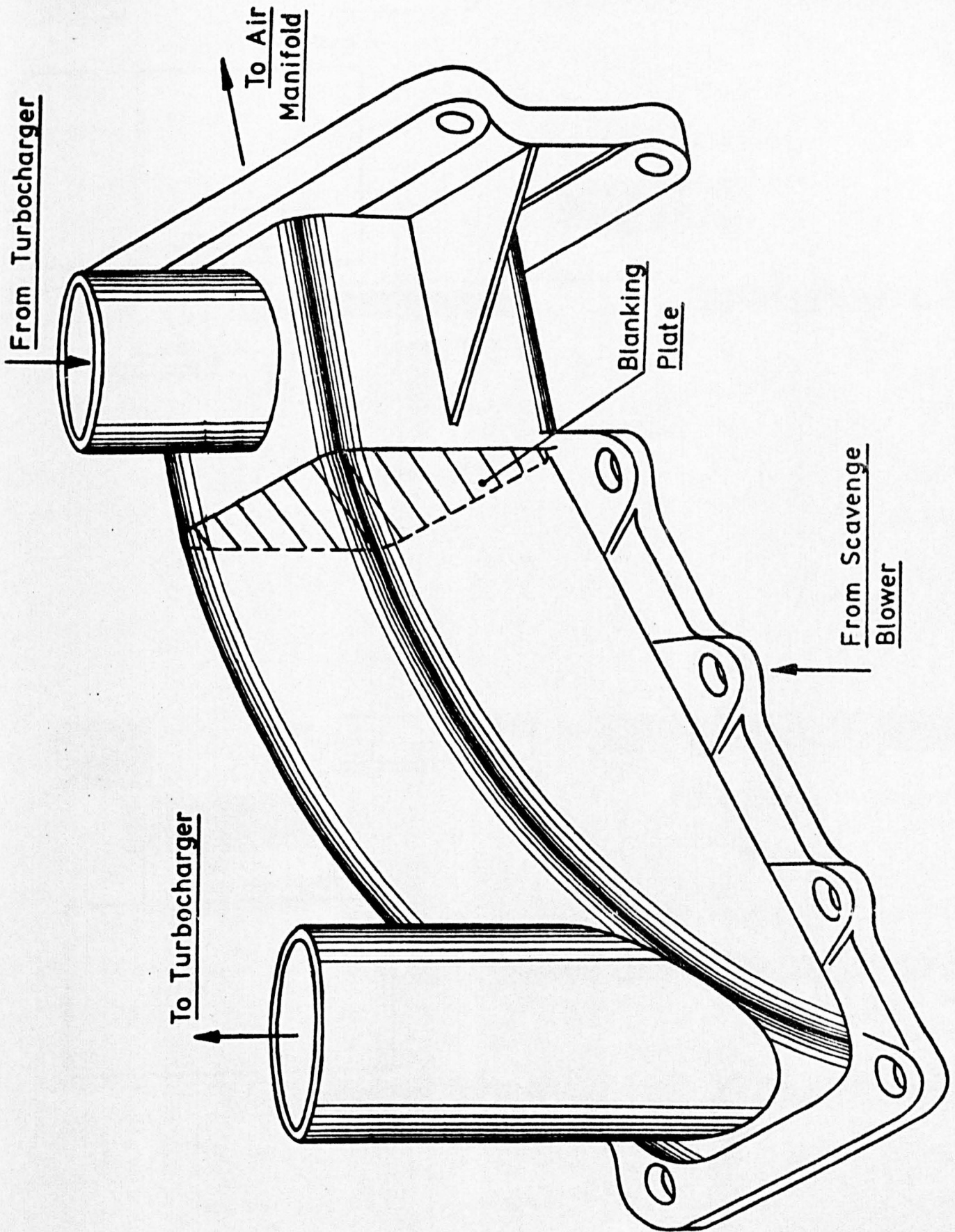
FIGURE 7.7

FIGURE 4.5

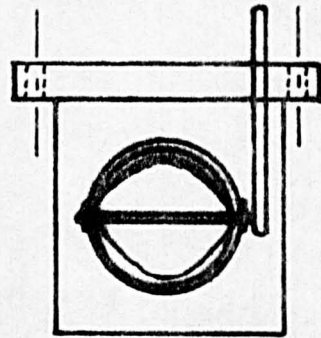
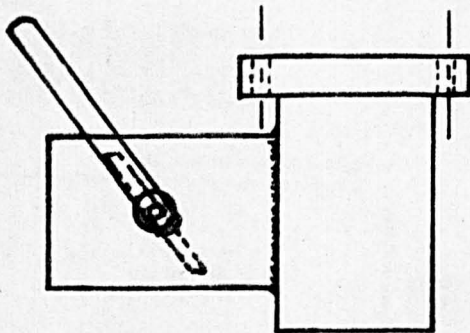
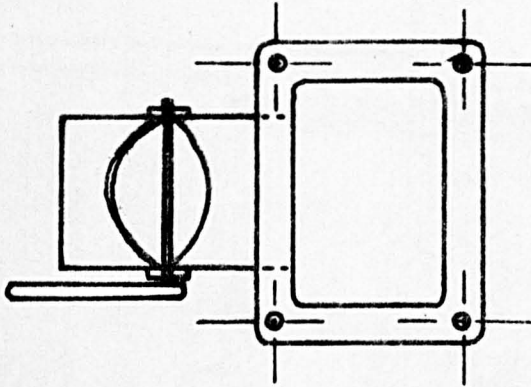


PRESSURE TRANSDUCER FITTING
(Cylinder Pressure)

FIGURE 4.6

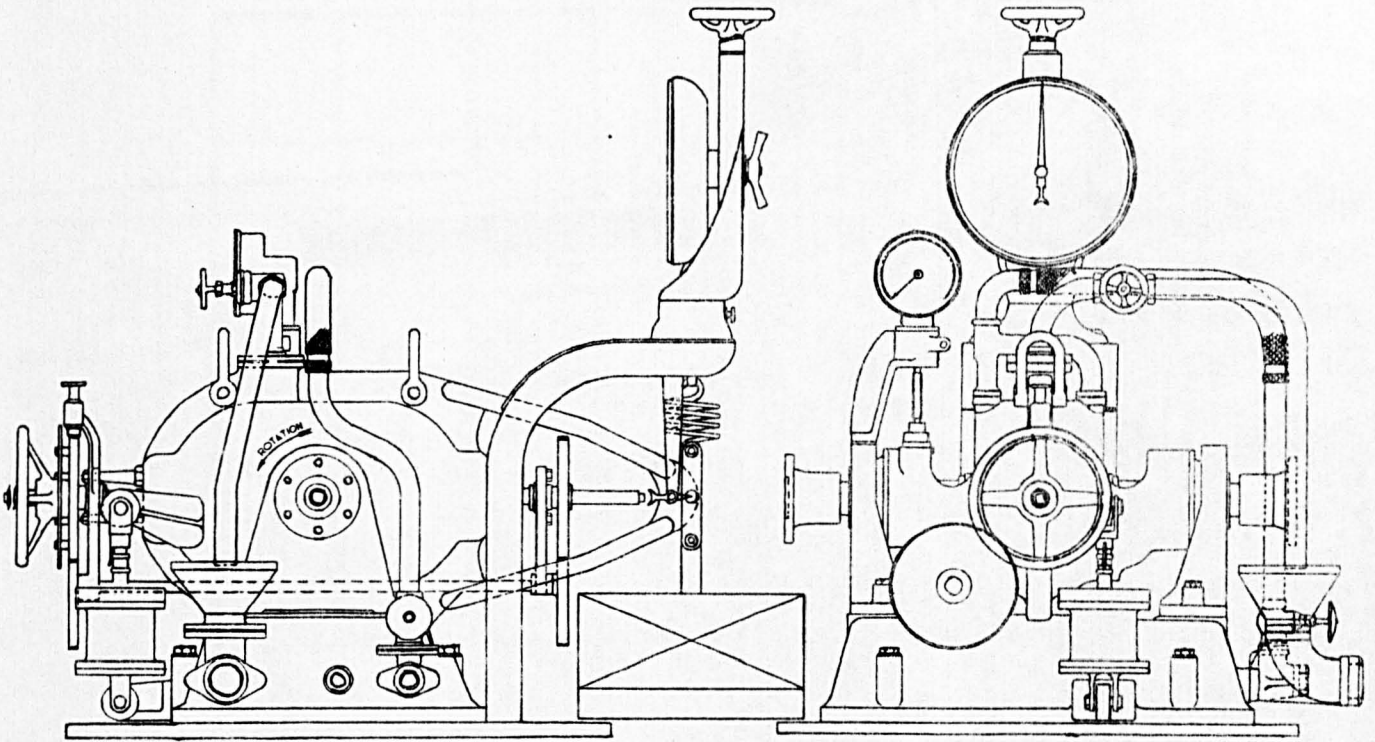


MODIFICATION TO SCAVENGE BLOWER
OUTLET MANIFOLD

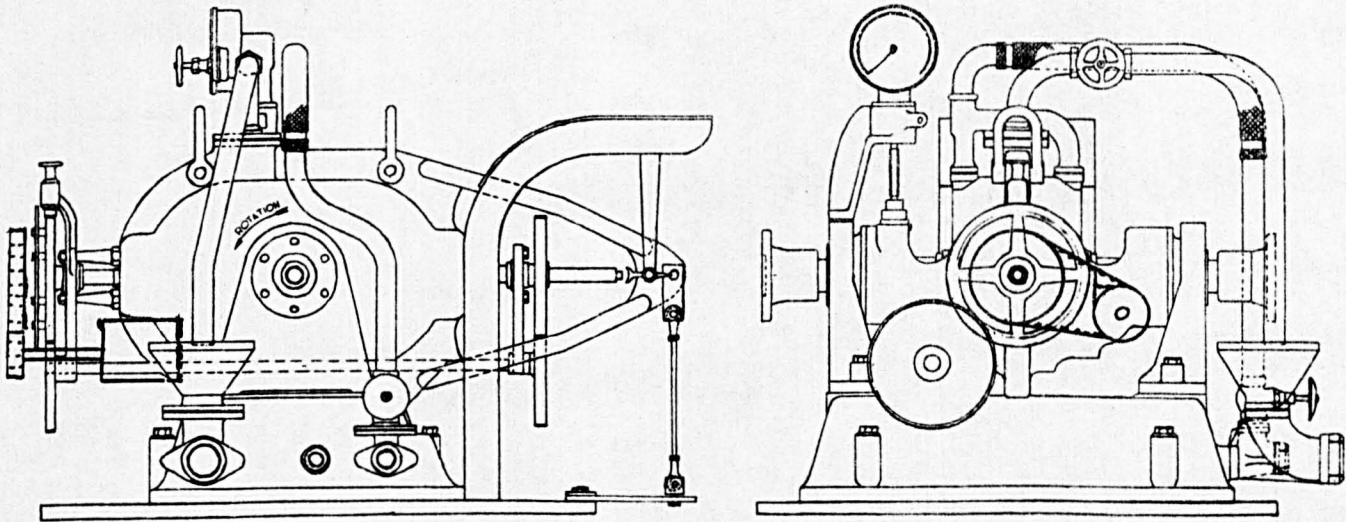


FABRICATED AIR MANIFOLD

FIGURE 4.8



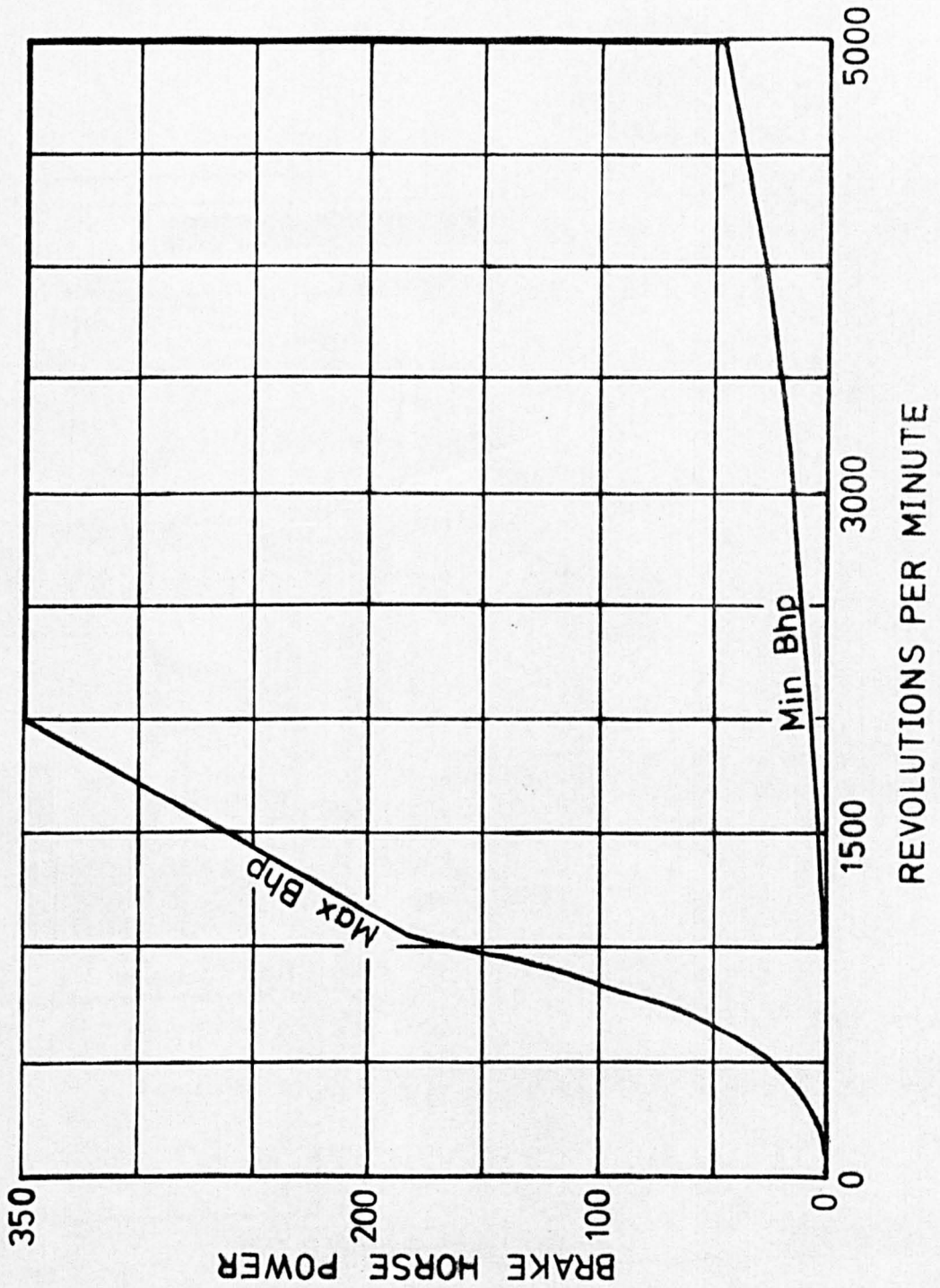
STANDARD DYNAMOMETER



ADAPTED DYNAMOMETER

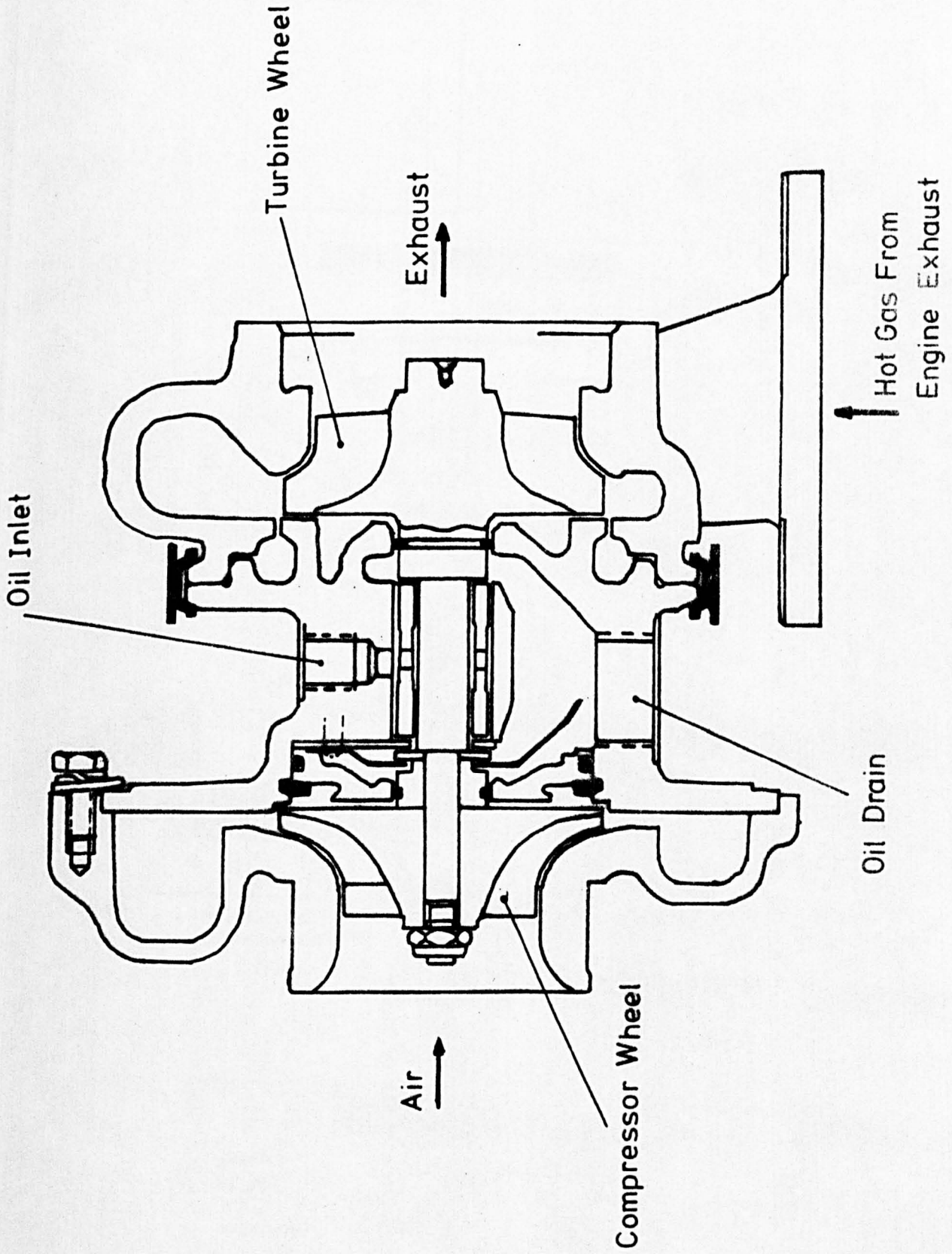
STANDARD AND ADAPTED
DPX 4 HYDRAULIC DYNAMOMETER

FIGURE 4.9



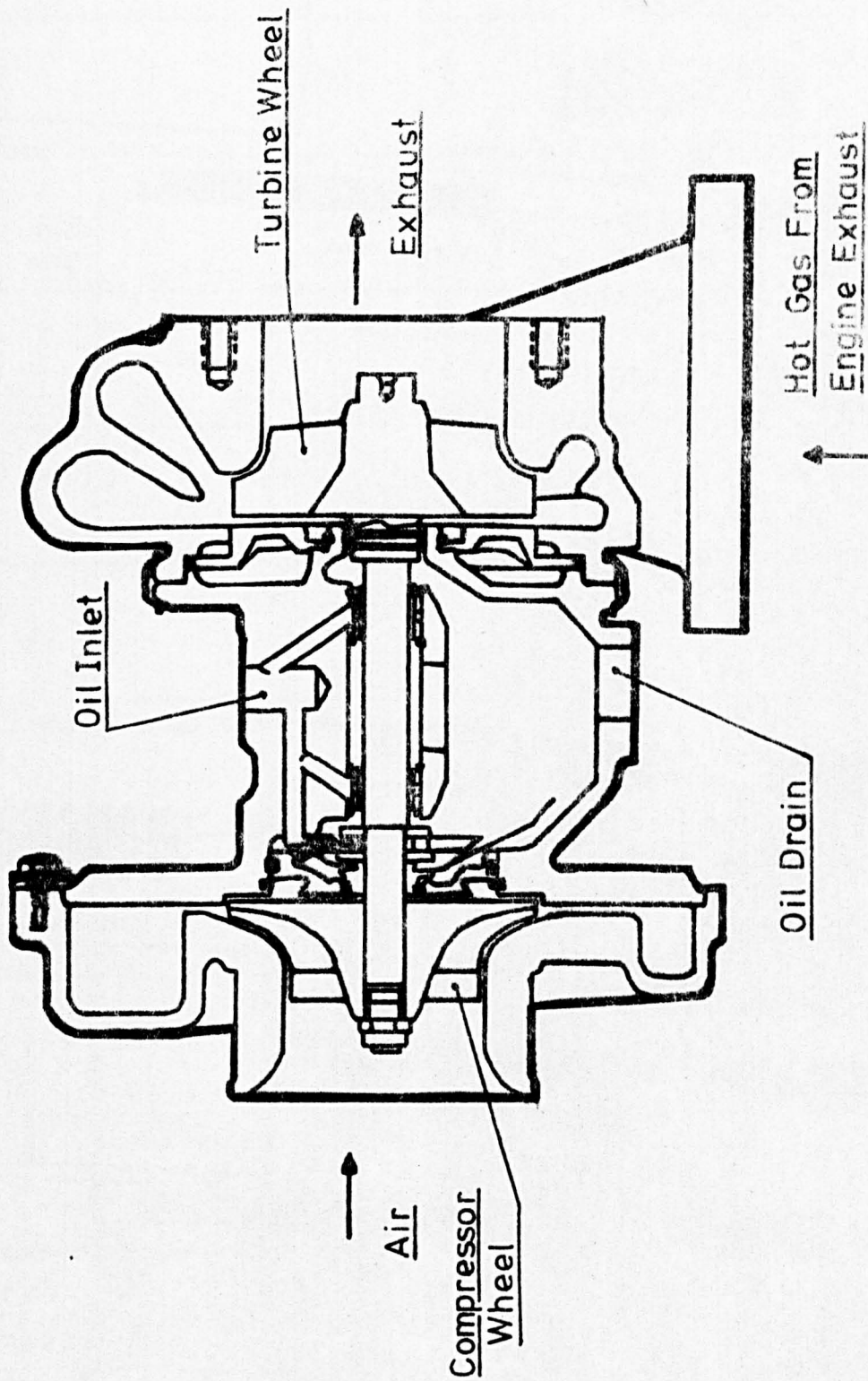
CHARACTERISTICS OF FROUDE DPX 4
HYDRAULIC DYNAMOMETER (OPEN FLOW)

FIGURE 4,10



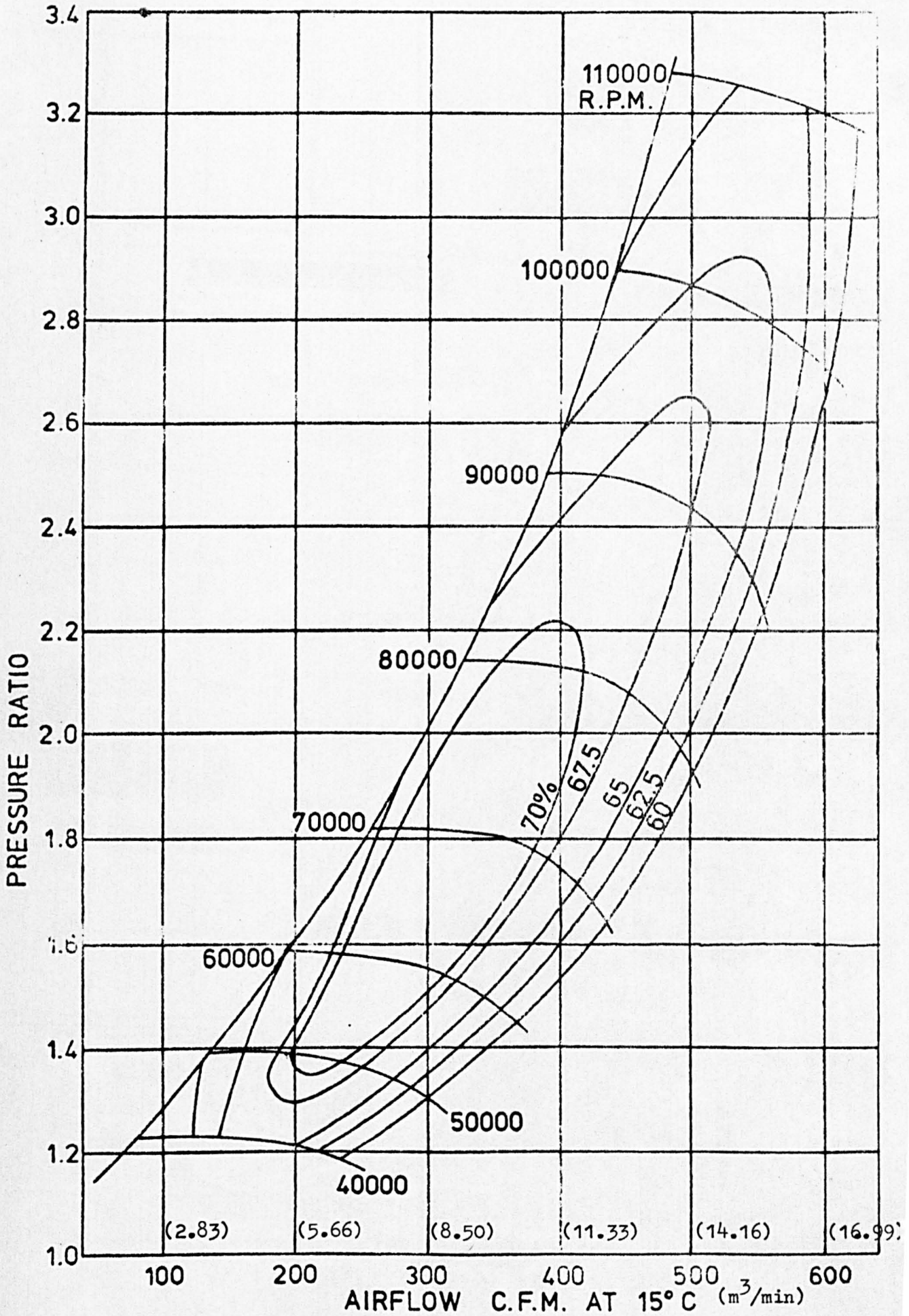
SECTIONAL VIEW HOLSET 3LD1

FIGURE 4.11



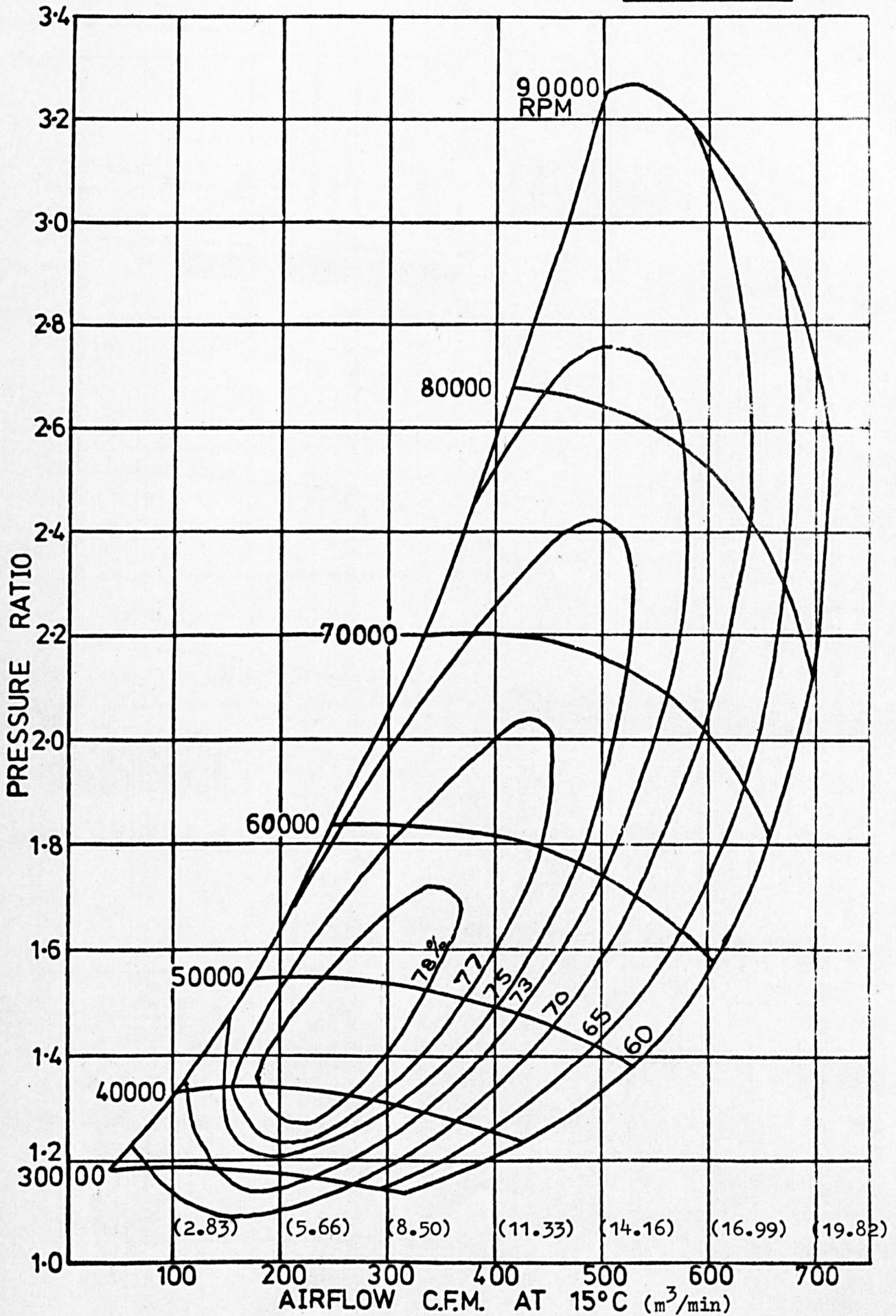
SECTIONAL VIEW HOLSET 4LGK

FIGURE 4,12



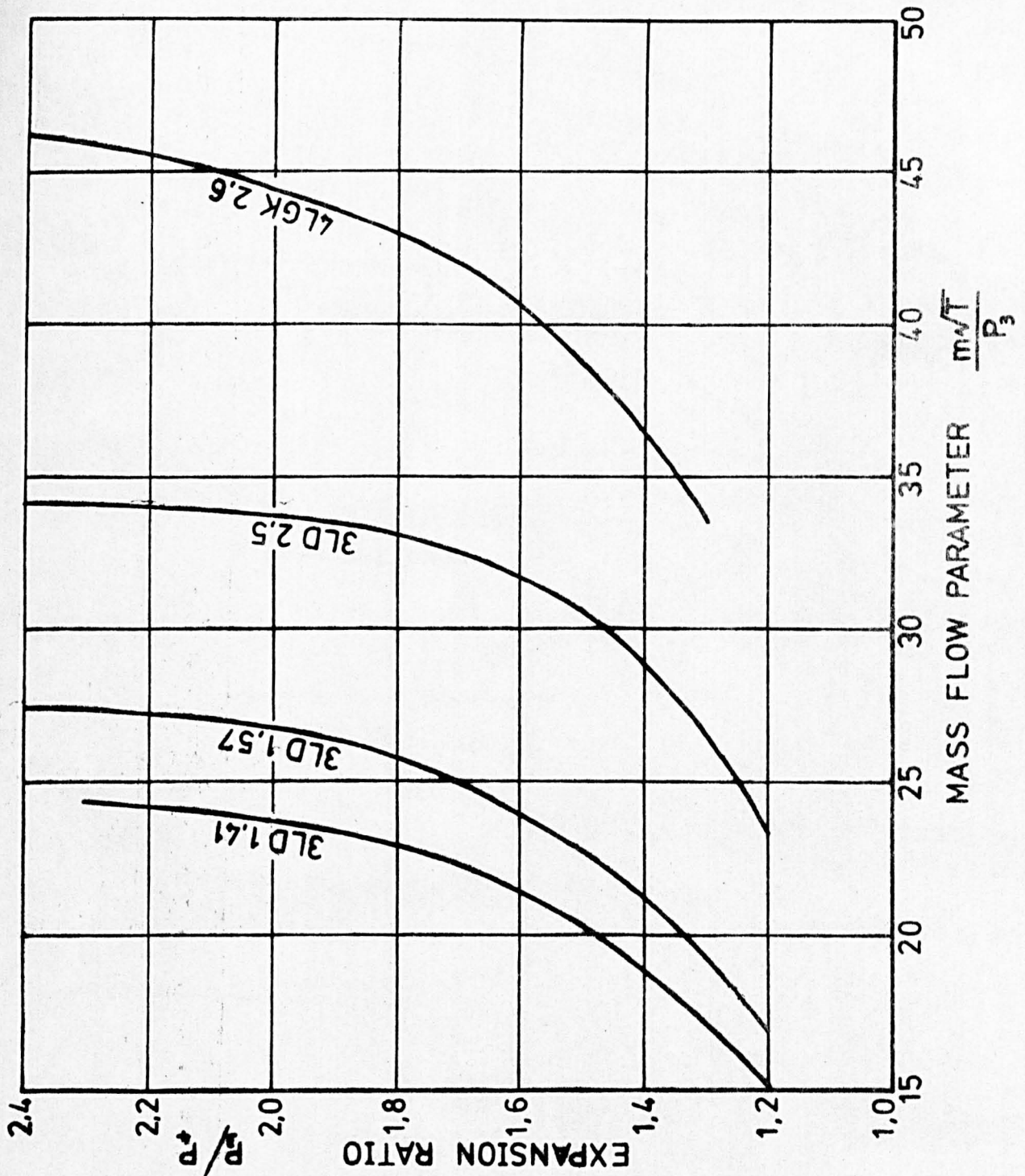
HOLSET 3LD-1 COMPRESSOR PERFORMANCE

FIGURE 4,13



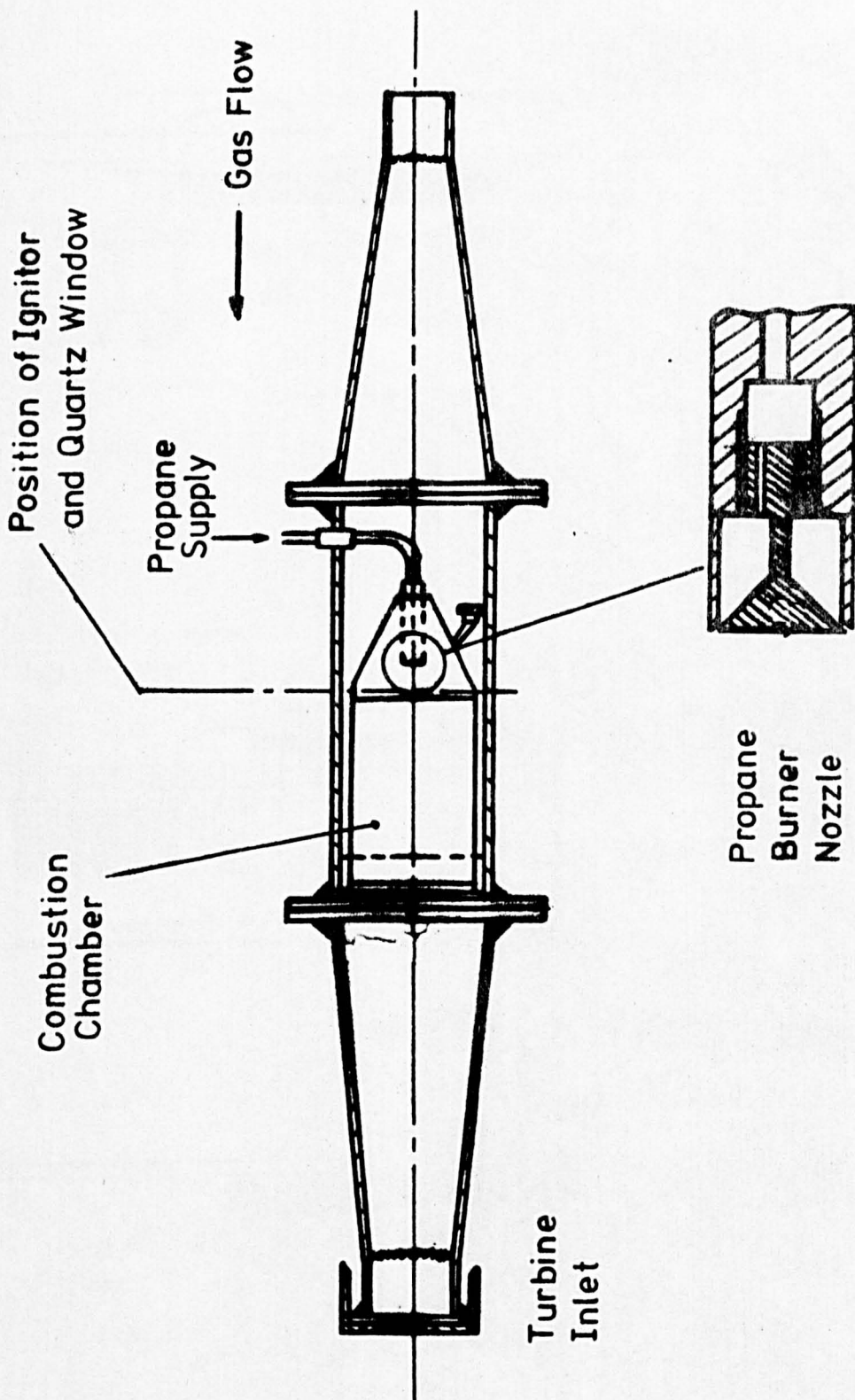
HOLSET 4L6K COMPRESSOR PERFORMANCE

FIGURE 4.14



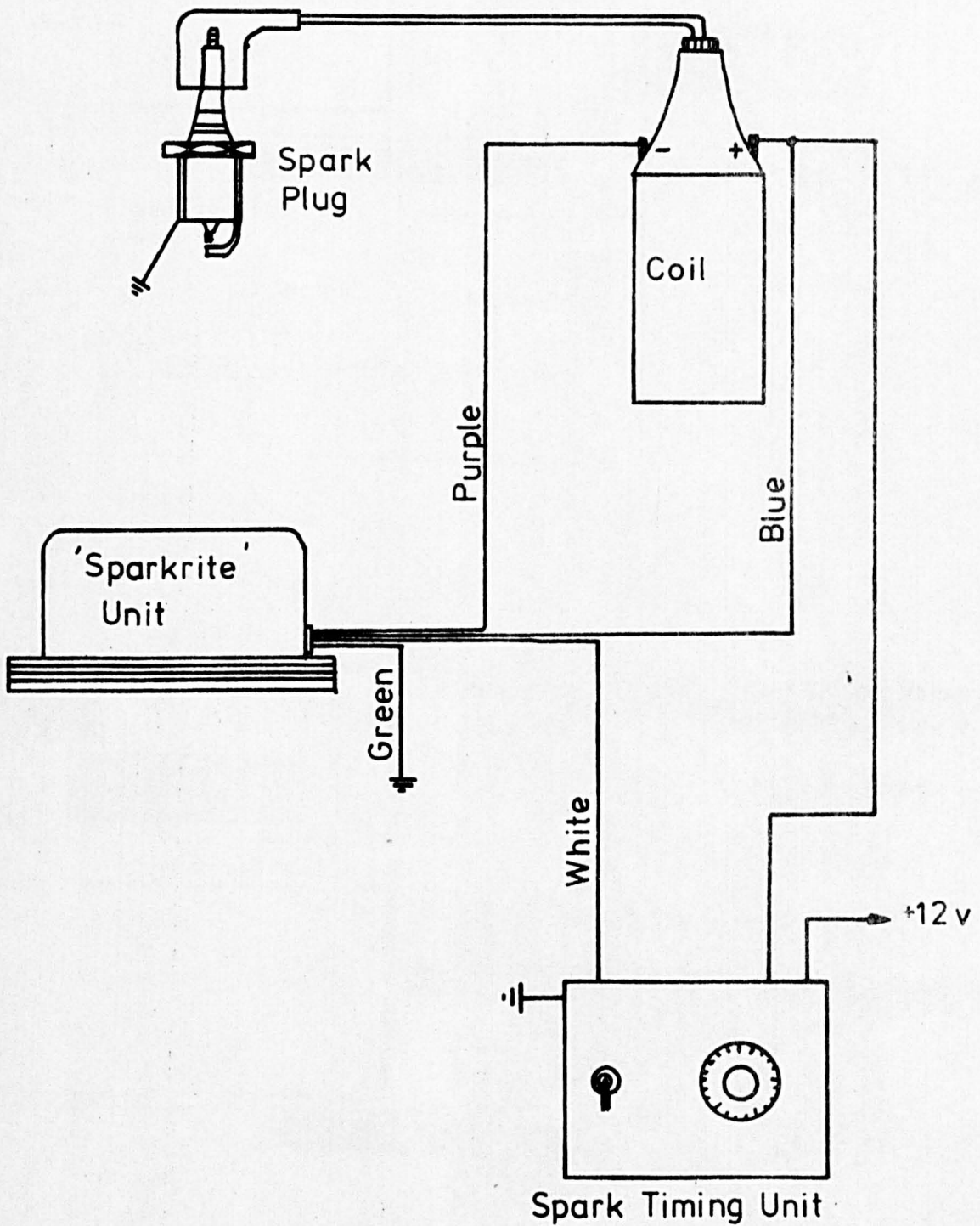
T - Turb Inlet Temp (K) P_3 - Turb Inlet Press (psi abs)
 P_2 - Turb Outlet Press (psi abs) m - Mass Flow (lbs/min)

TURBINE SWALLOWING CAPACITY PERFORMANCE



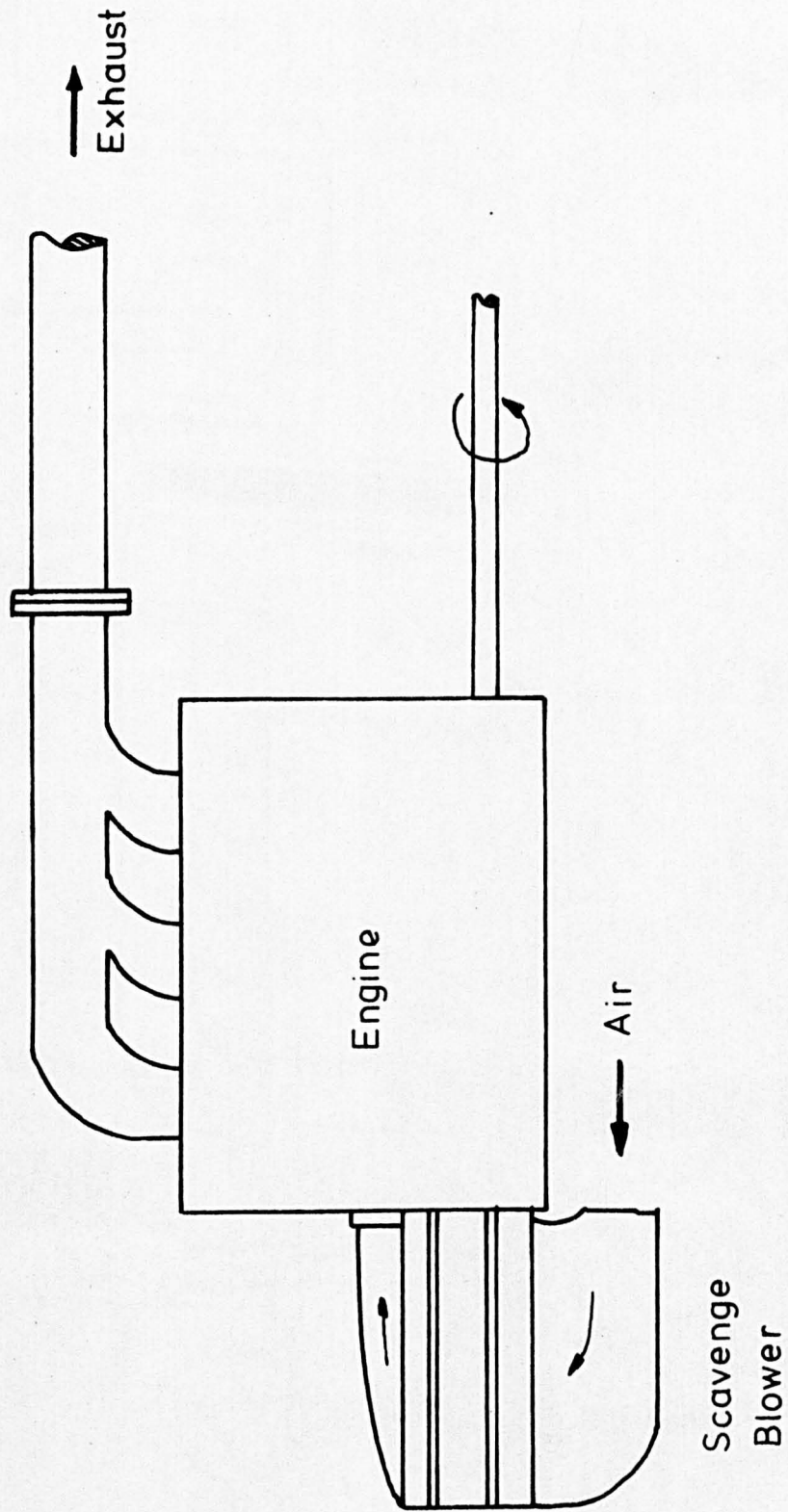
AUXILIARY COMBUSTION CHAMBER ARRANGEMENT

FIGURE 4.16



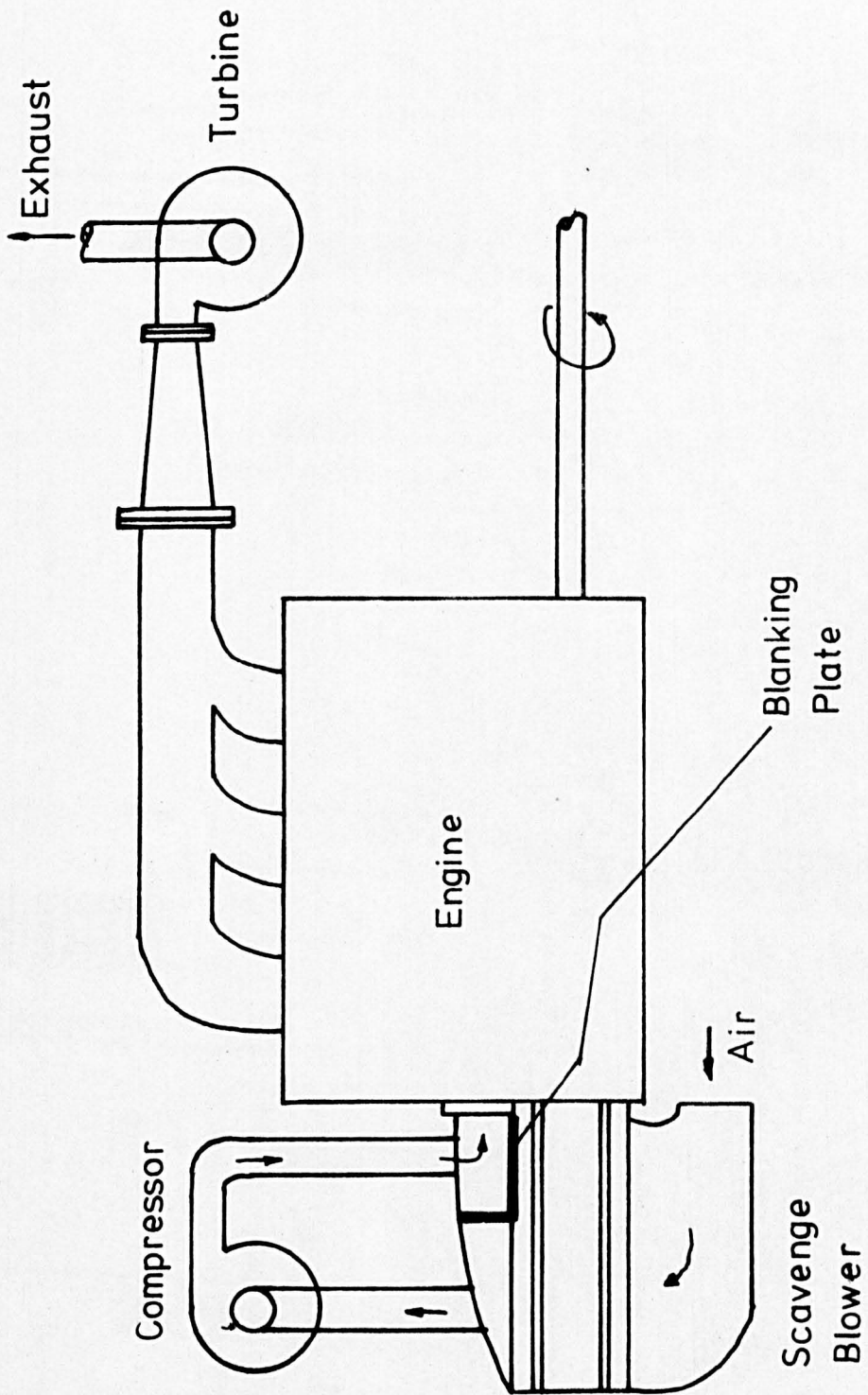
AUXILIARY COMBUSTION CHAMBER
IGNITION SYSTEM ARRANGEMENT

FIGURE 4.17

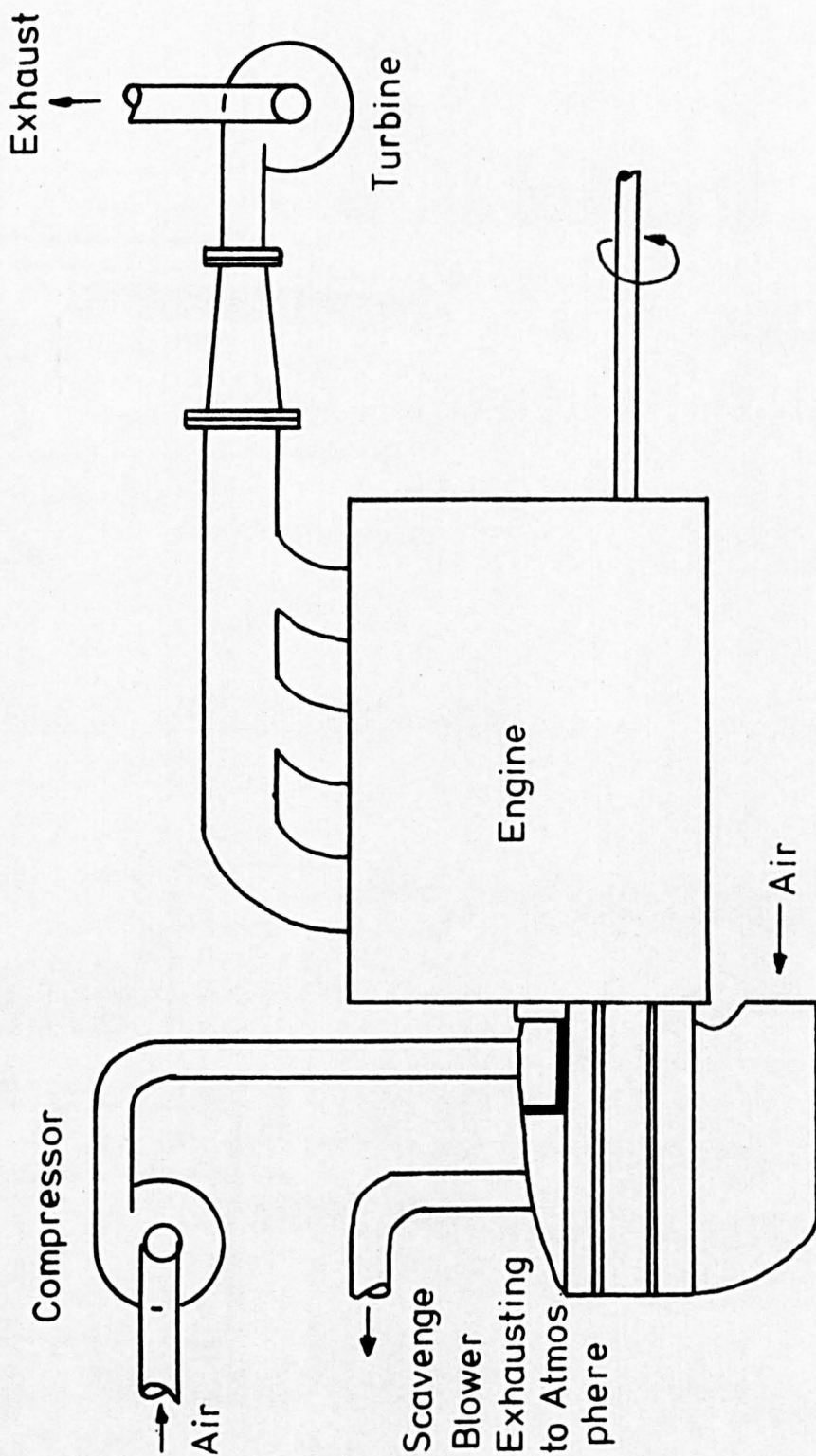


STANDARD ENGINE ARRANGEMENT

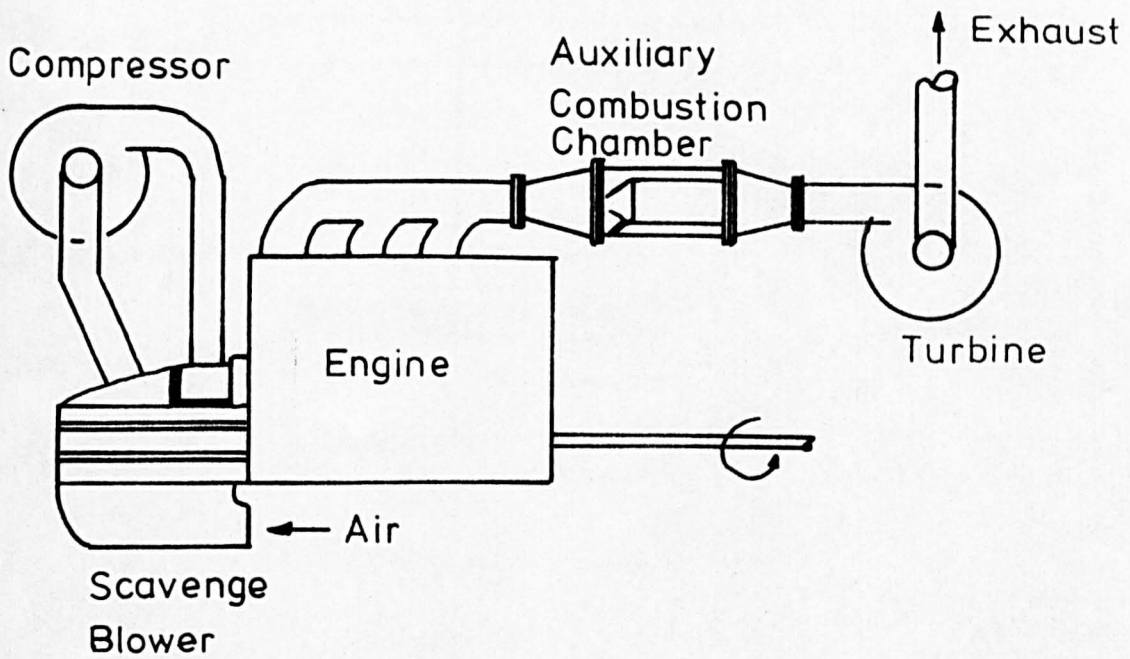
FIGURE 4.18



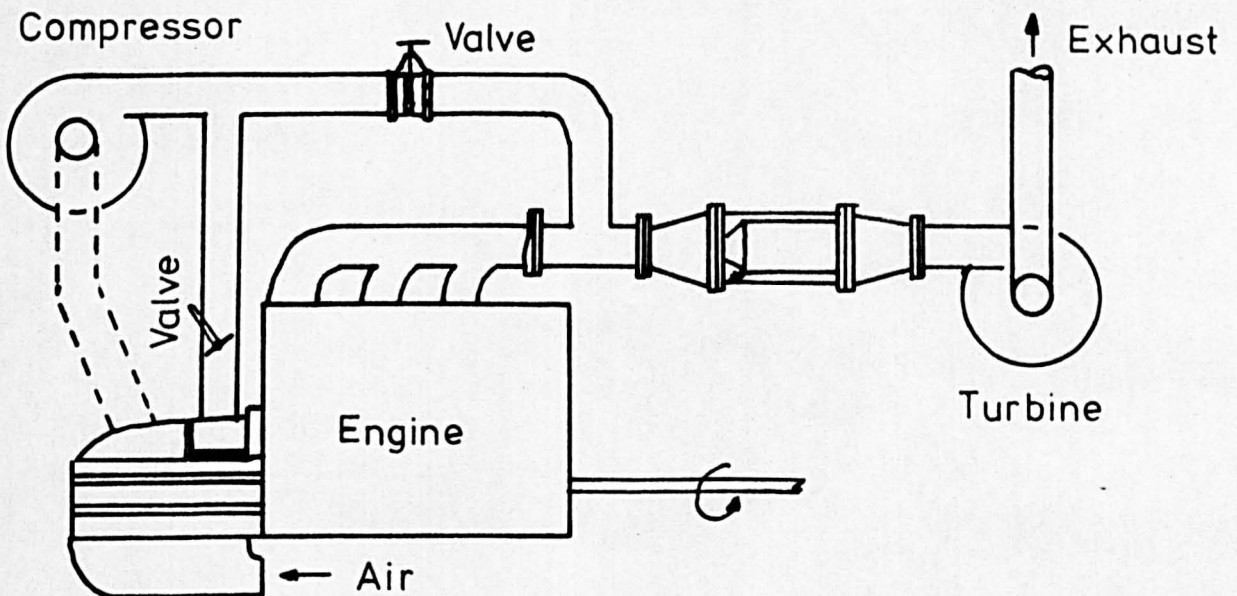
DETAIL OF ENGINE WITH TURBOCHARGER
AND SCAVENGE BLOWER IN SERIES



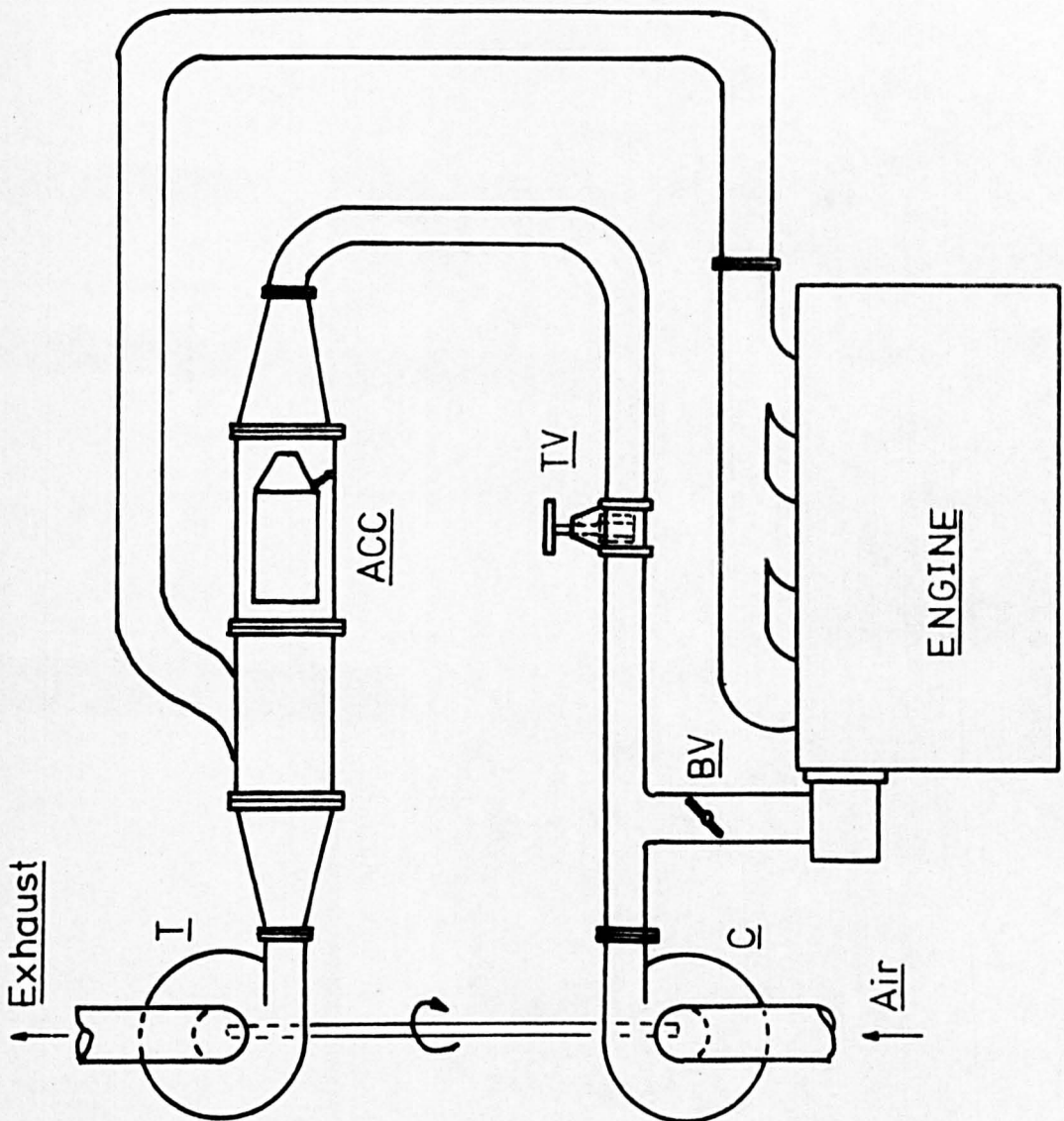
SYSTEM ARRANGEMENT FOR FREELY TURBOCHARGED TESTS



(a) STAGE II SYSTEM (no bypass)



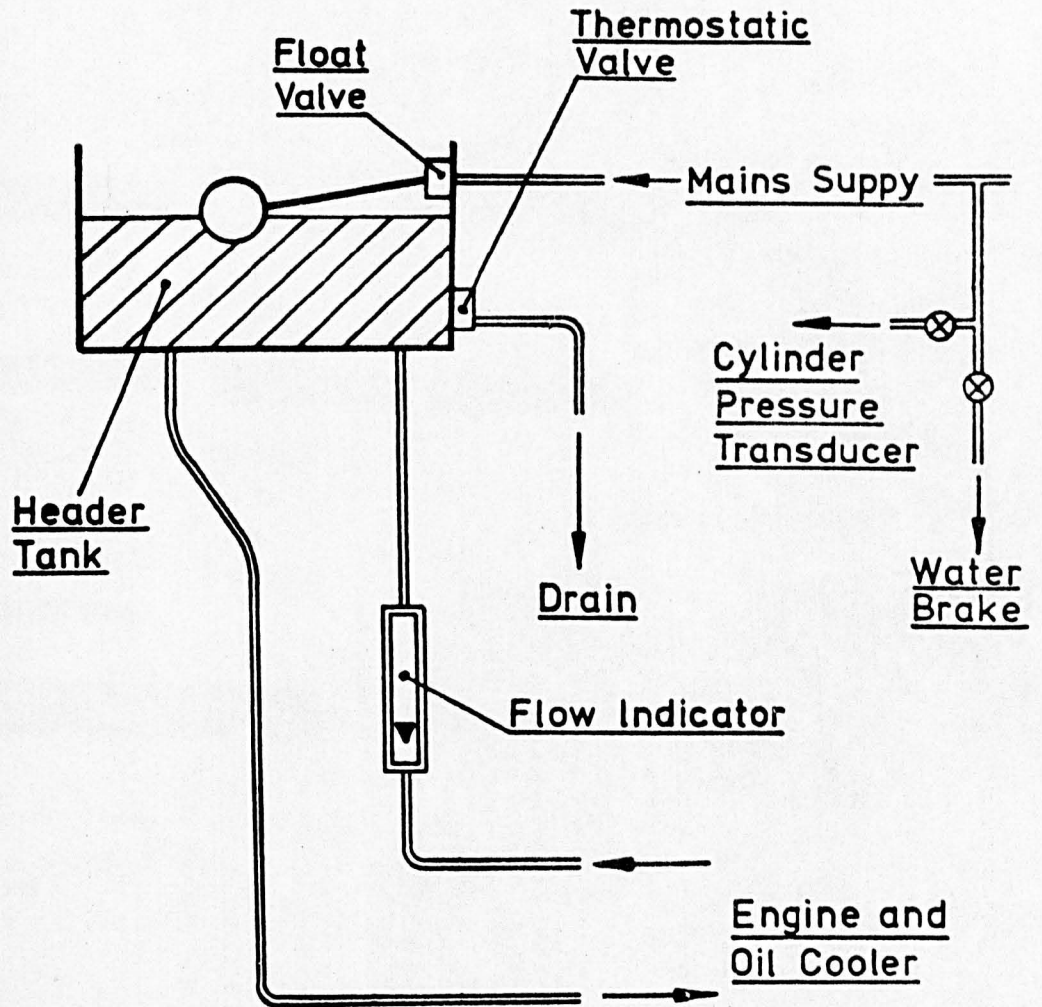
(b) STAGE II SYSTEM (with throttled bypass)



- C Compressor
- T Turbine
- ACC Auxiliary Combustion Chamber
- BV Butterfly Valve
- TV Throttling Valve

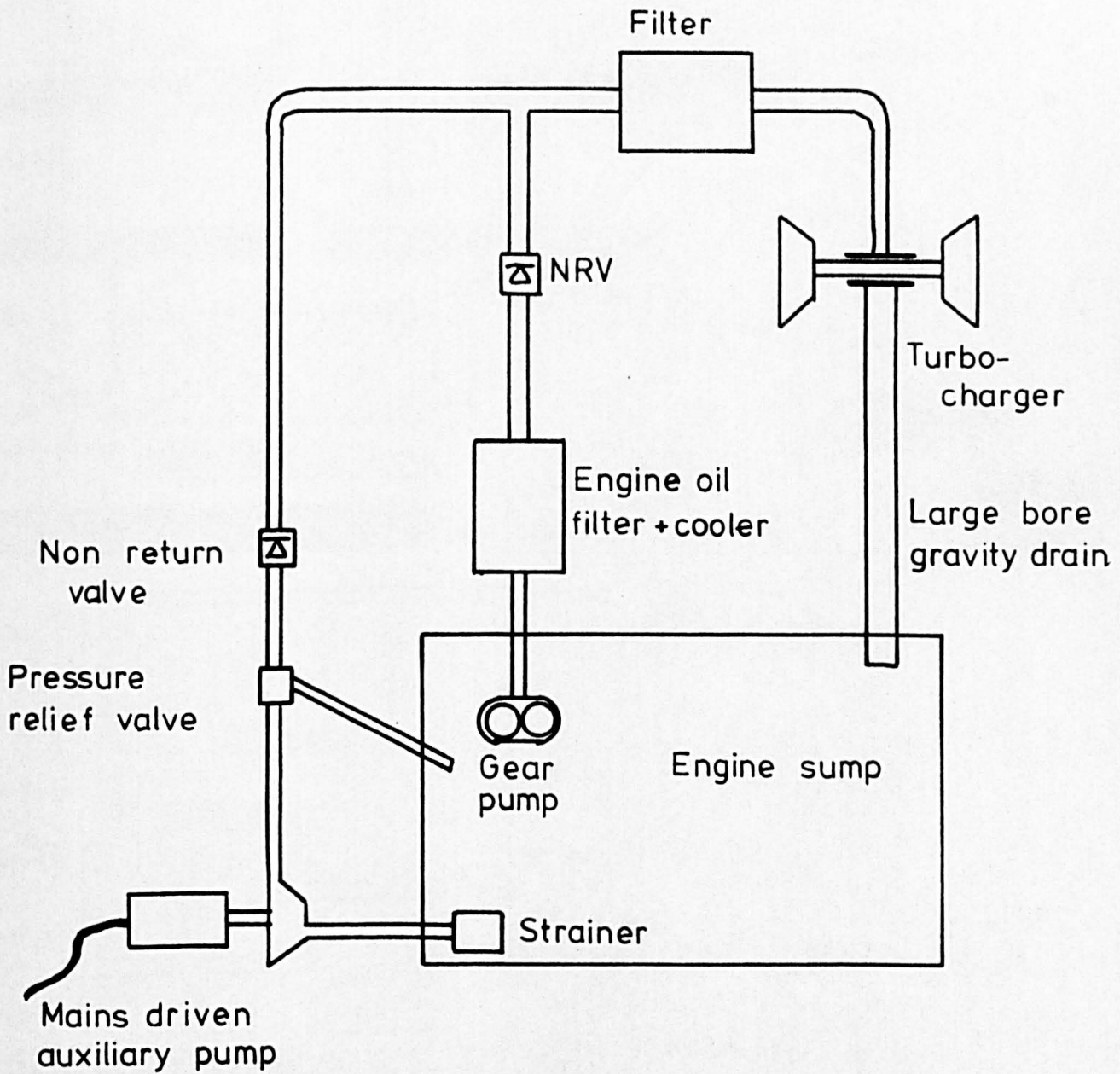
DETAIL OF DGT SYSTEM STAGE III

FIGURE 4.22

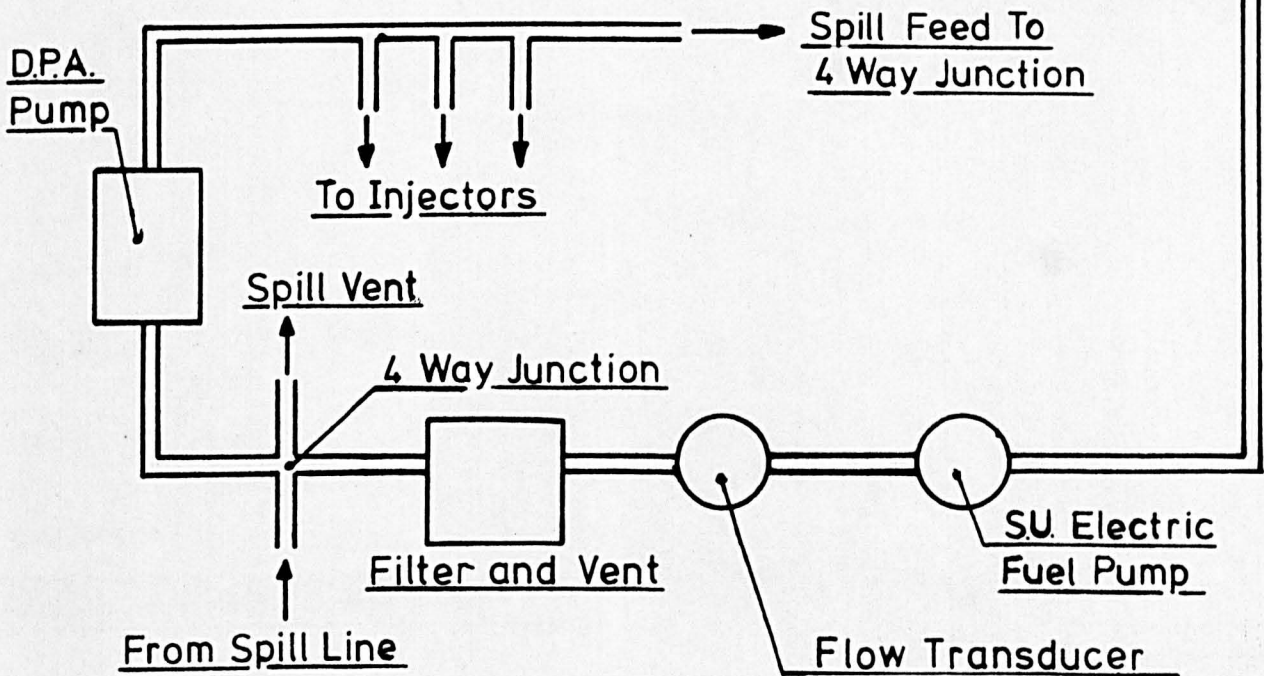
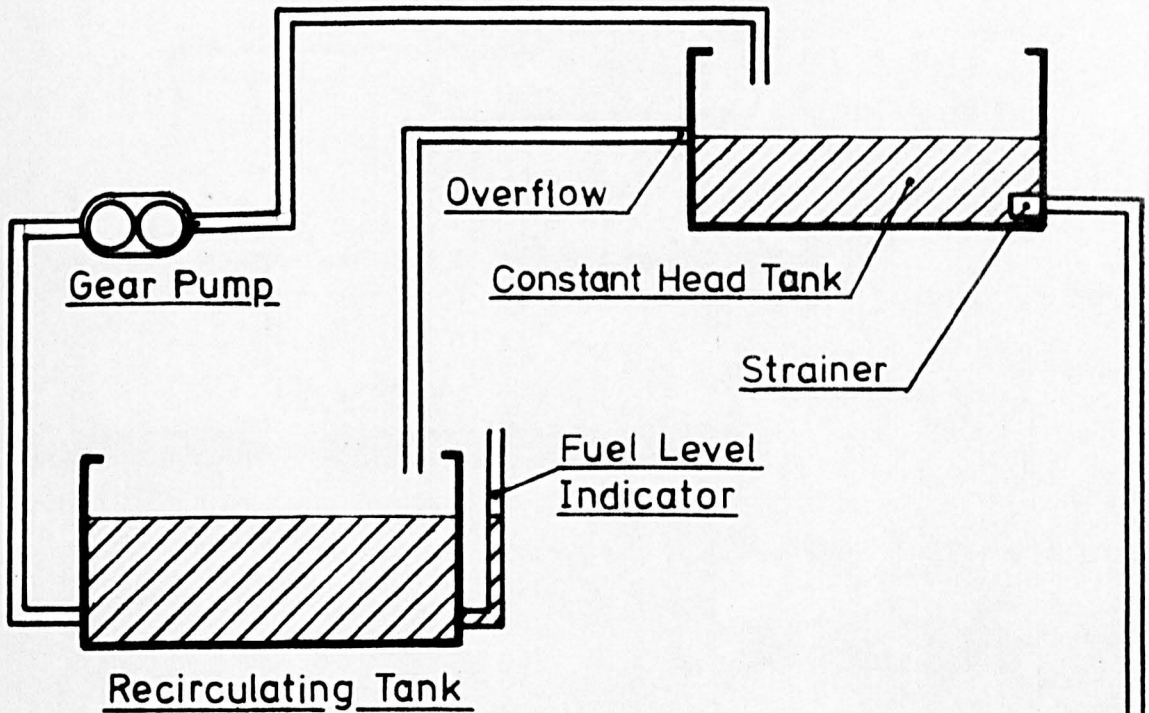


COOLING WATER SYSTEM

FIGURE 4,23

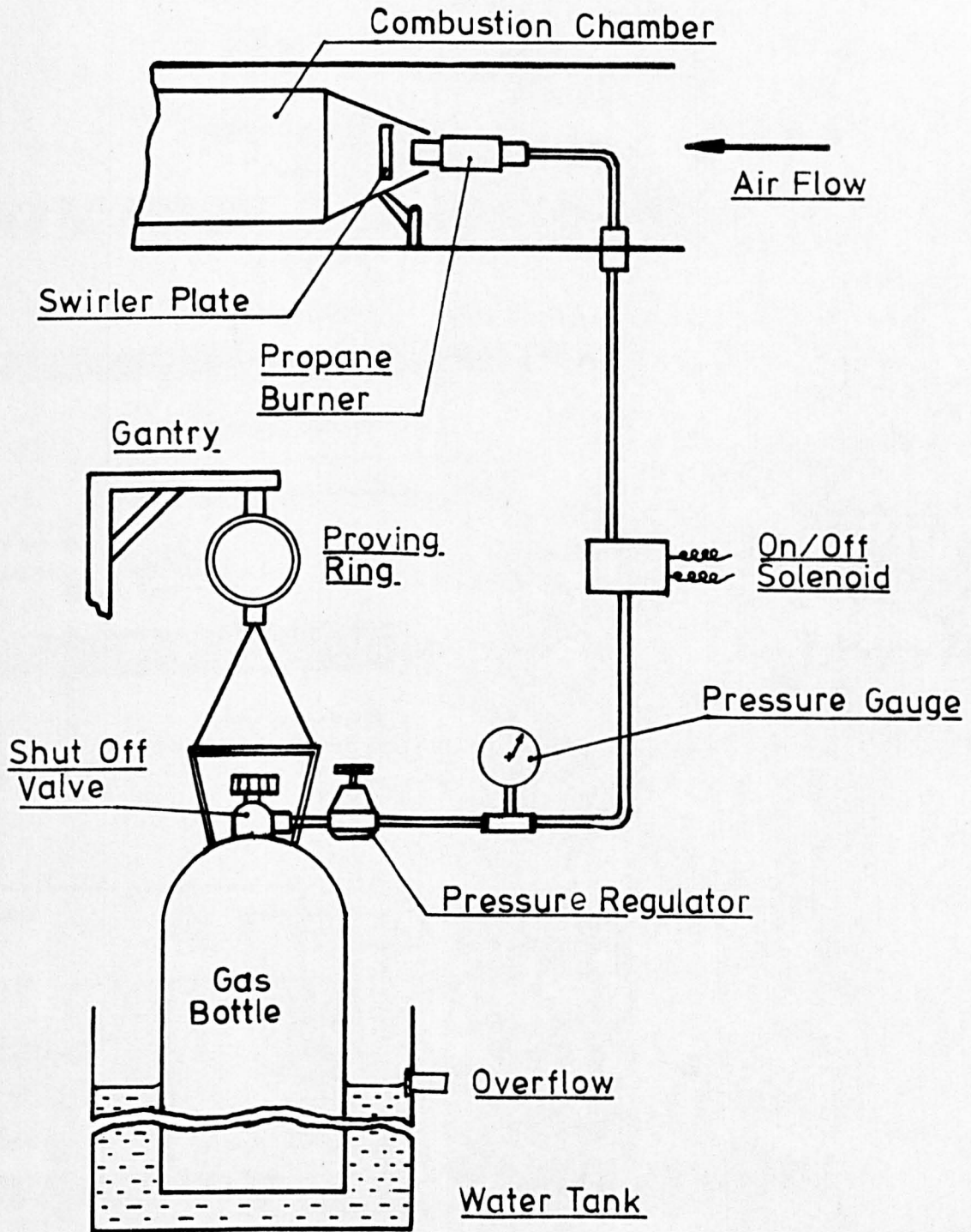


TEST RIG LUBRICATING OIL SUPPLY ARRANGEMENT



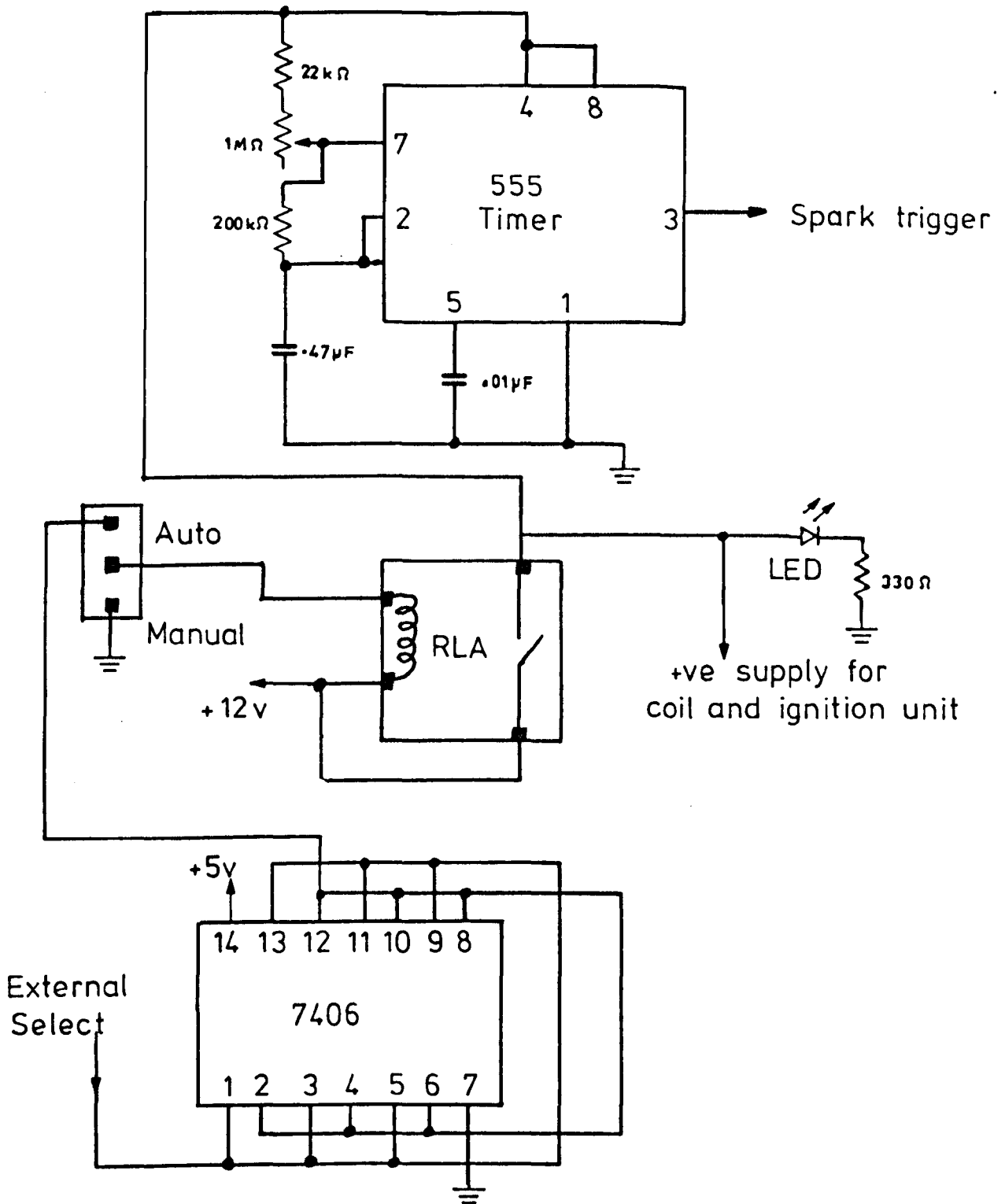
ENGINE FUEL OIL SYSTEM

FIGURE 4,25



COMBUSTION CHAMBER FUEL SYSTEM

FIGURE 4.26



TIMING AND RELAY CIRCUIT FOR AUXILIARY
COMBUSTION CHAMBER IGNITION

FUEL FLOW READOUT UNIT
CIRCUIT DIAGRAM

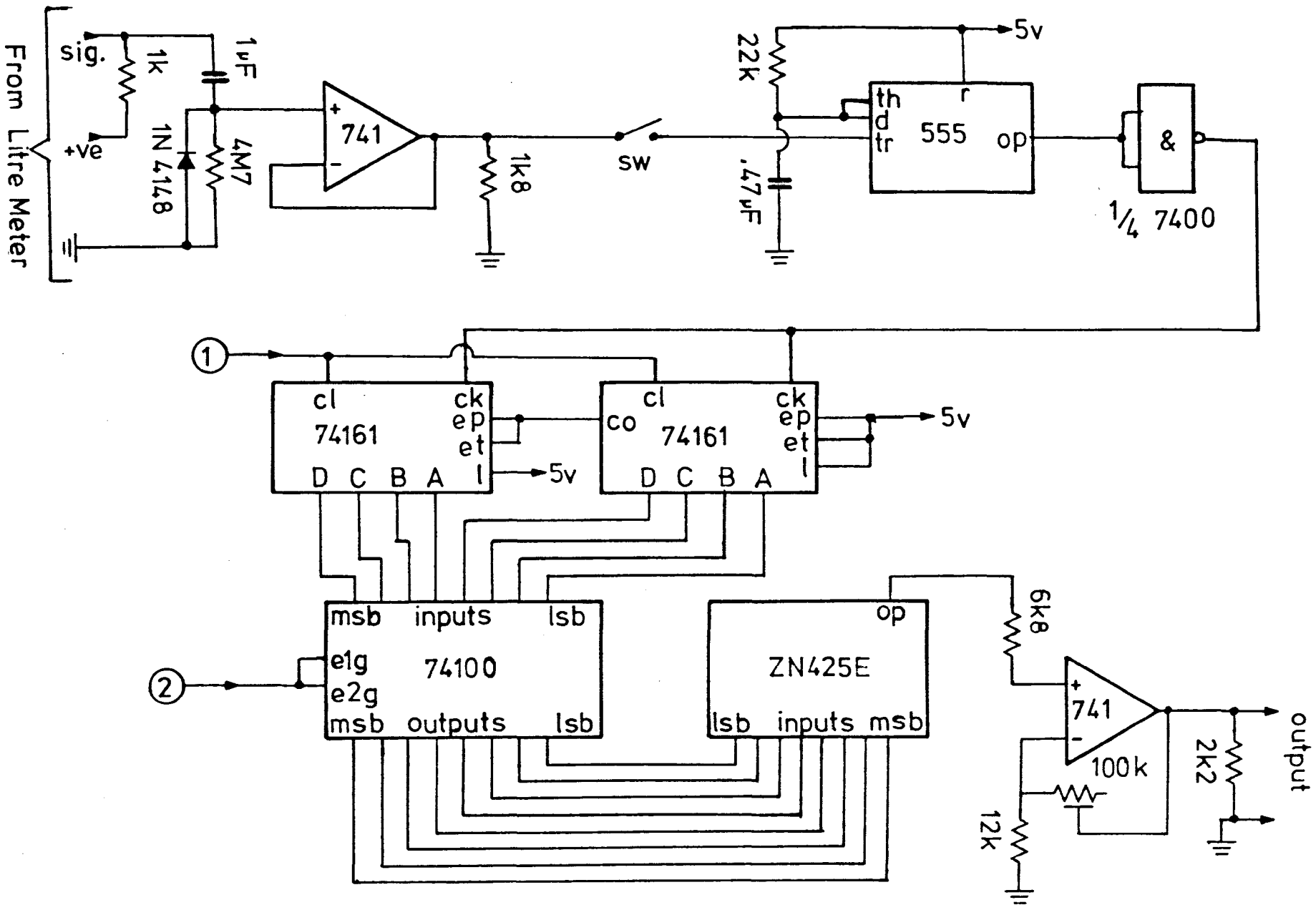
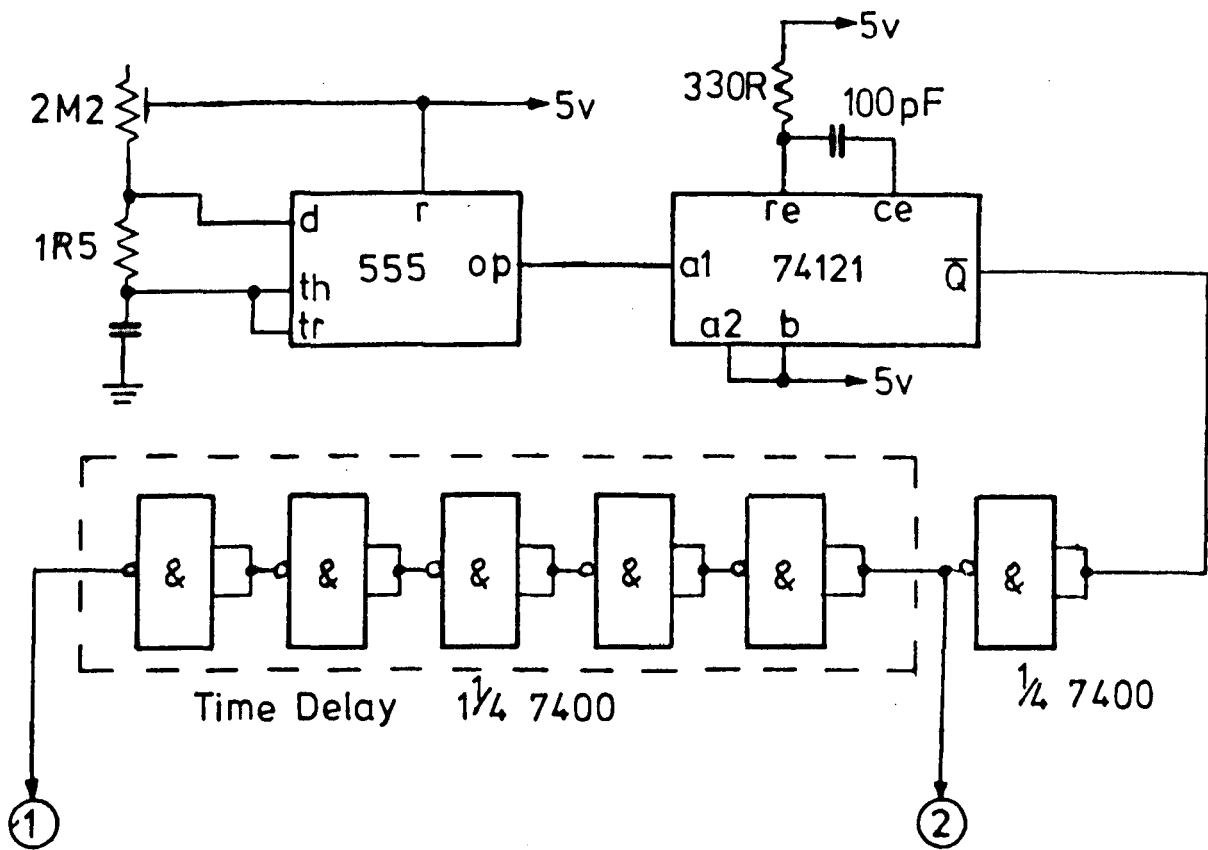


FIGURE 4.27

FIGURE 4,28

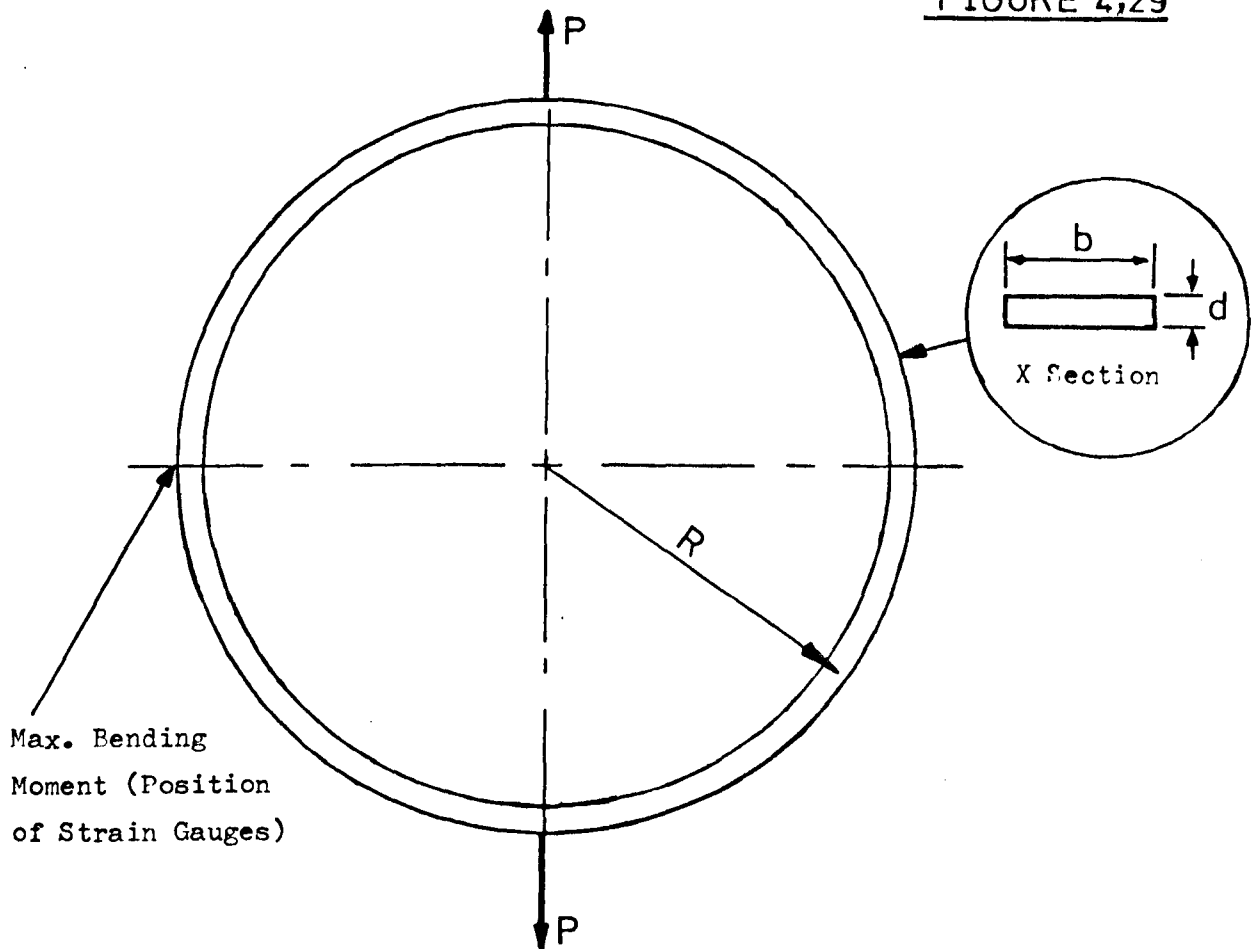


KEY

- | | | | |
|---------|-----------------------|---------|----------------------|
| sw | Switch | th | Threshold |
| d | Discharge | tr | Trigger |
| r | Reset | op | Output |
| cl | Clear | ck | Clock Input |
| ep | Enable P Input | et | Enable T Input |
| l | Load Input | msb | Most Significant Bit |
| lsb | Least Significant Bit | e1g,e2g | Enable Data Transfer |
| re | External Resistor | ce | External Capacitor |
| Q̄ | Monostable Output | a1,a2,b | Monostable Inputs |
| A,B,C,D | Counter Outputs | | |

FUEL FLOW SYSTEM
SAMPLE TIMING & RESET CIRCUIT

FIGURE 4,29



Max. Bending
Moment (Position
of Strain Gauges)

$$\text{Maximum Bending Moment} = \frac{PR}{\pi}$$

$$\therefore \sigma = \frac{PR}{\pi} \cdot \frac{d}{2} \cdot \frac{12}{bd^3} = \frac{6PR}{\pi bd^3}$$

$$\epsilon = \frac{\sigma}{E} \quad \therefore \epsilon = \frac{6PR}{\pi bd^3 E}$$

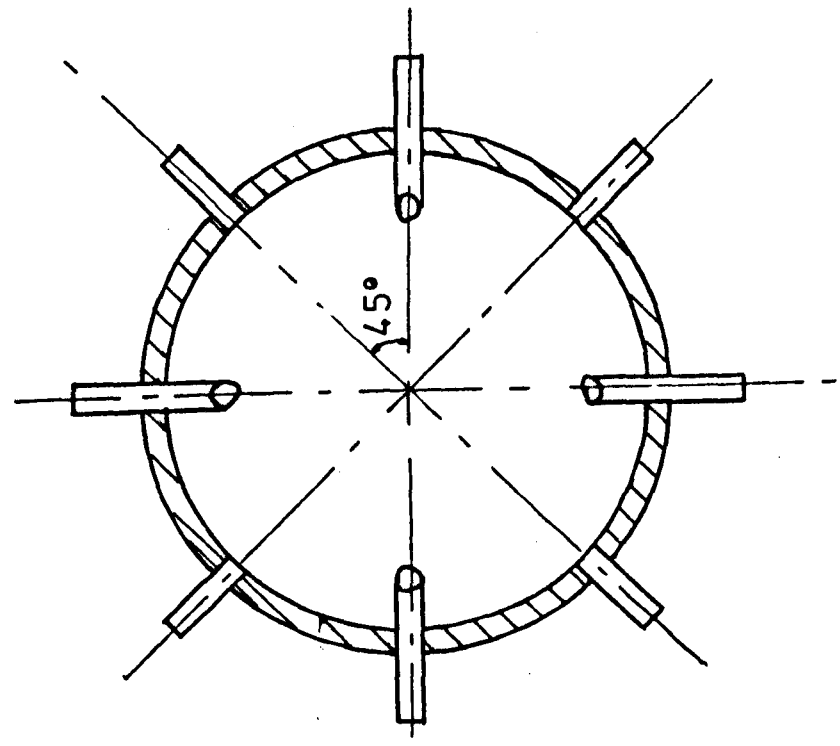
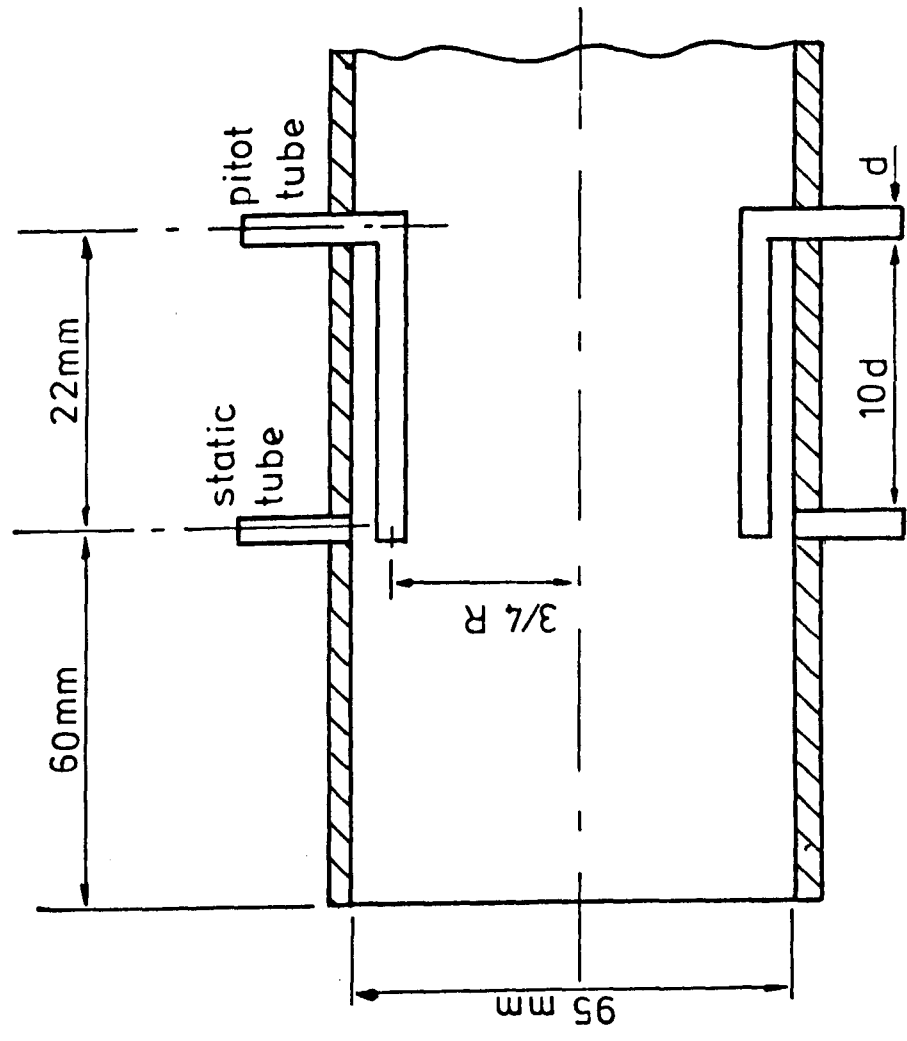
With :- $P = 50 \text{ kg (490.5 N)}$ $R = 50 \text{ mm}$ $b = 25 \text{ mm}$
 $d = 3.5 \text{ mm}$ $E = 207 \text{ GN/m}^2$

$$= \frac{6 \times 490.5 \times 50 \times 10^{-3}}{\pi \times 25 \times 10^{-3} \times 3.5^2 \times 10^{-6} \times 207 \times 10^9}$$

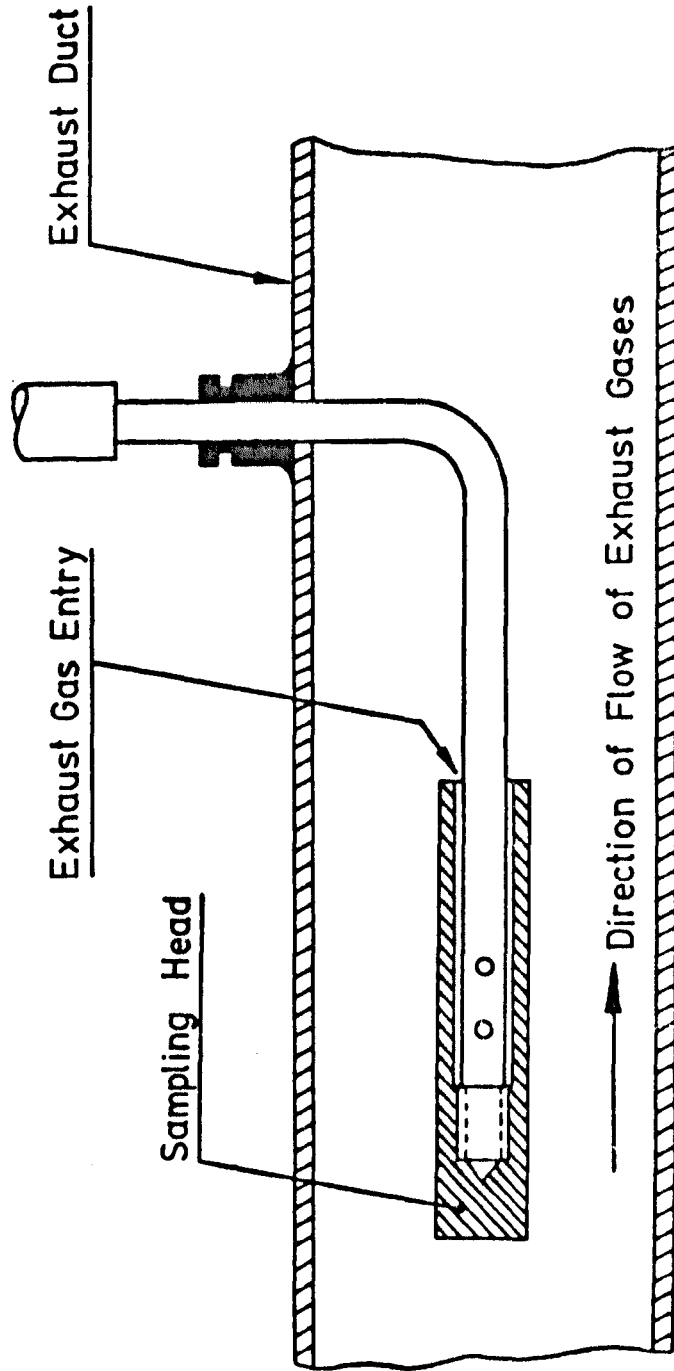
$$\epsilon = 739 \mu\epsilon$$

DETAILS OF PROVING RING

FIGURE 4.30



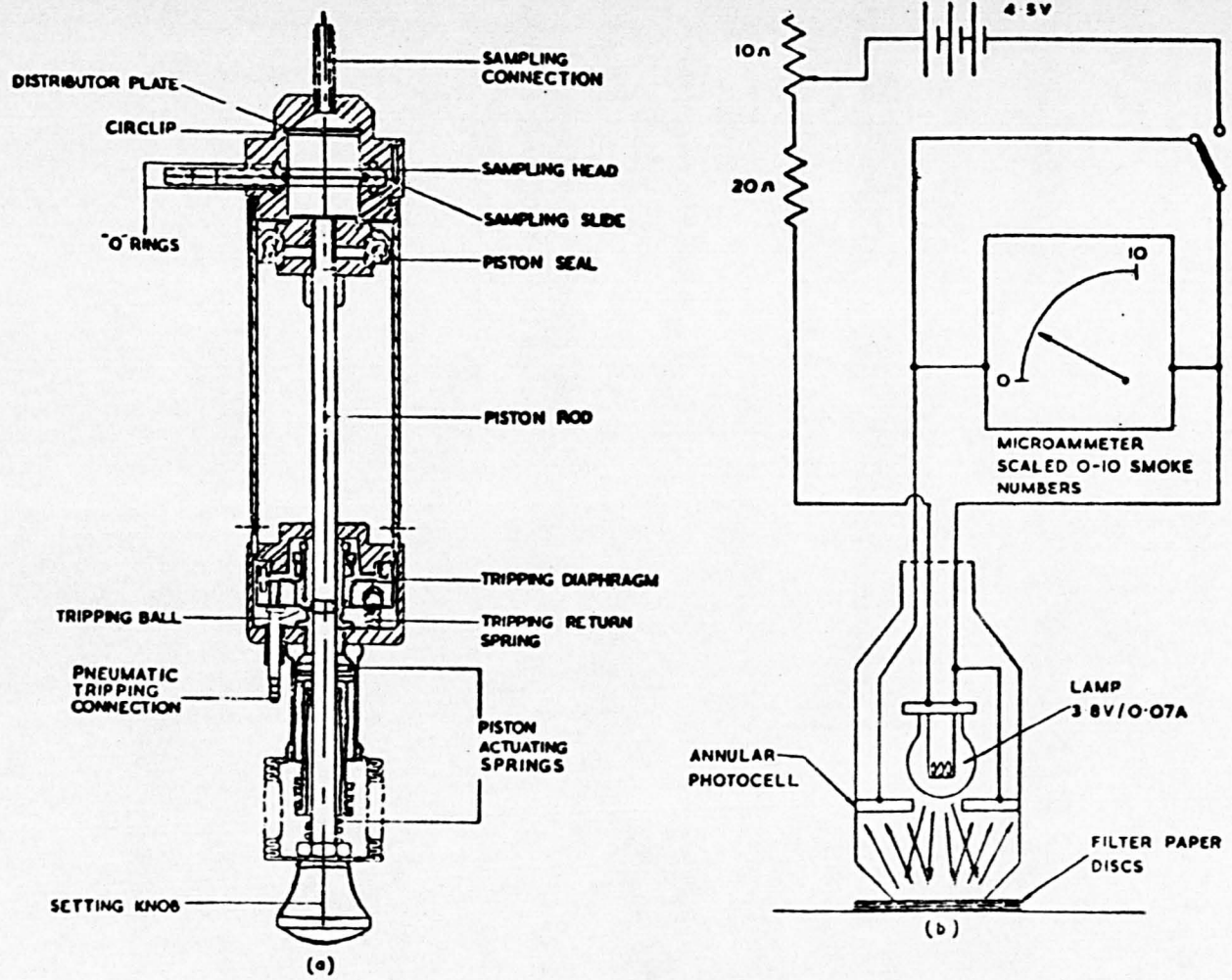
PRESTON FLOWMETER



BOSCH SMOKEMETER

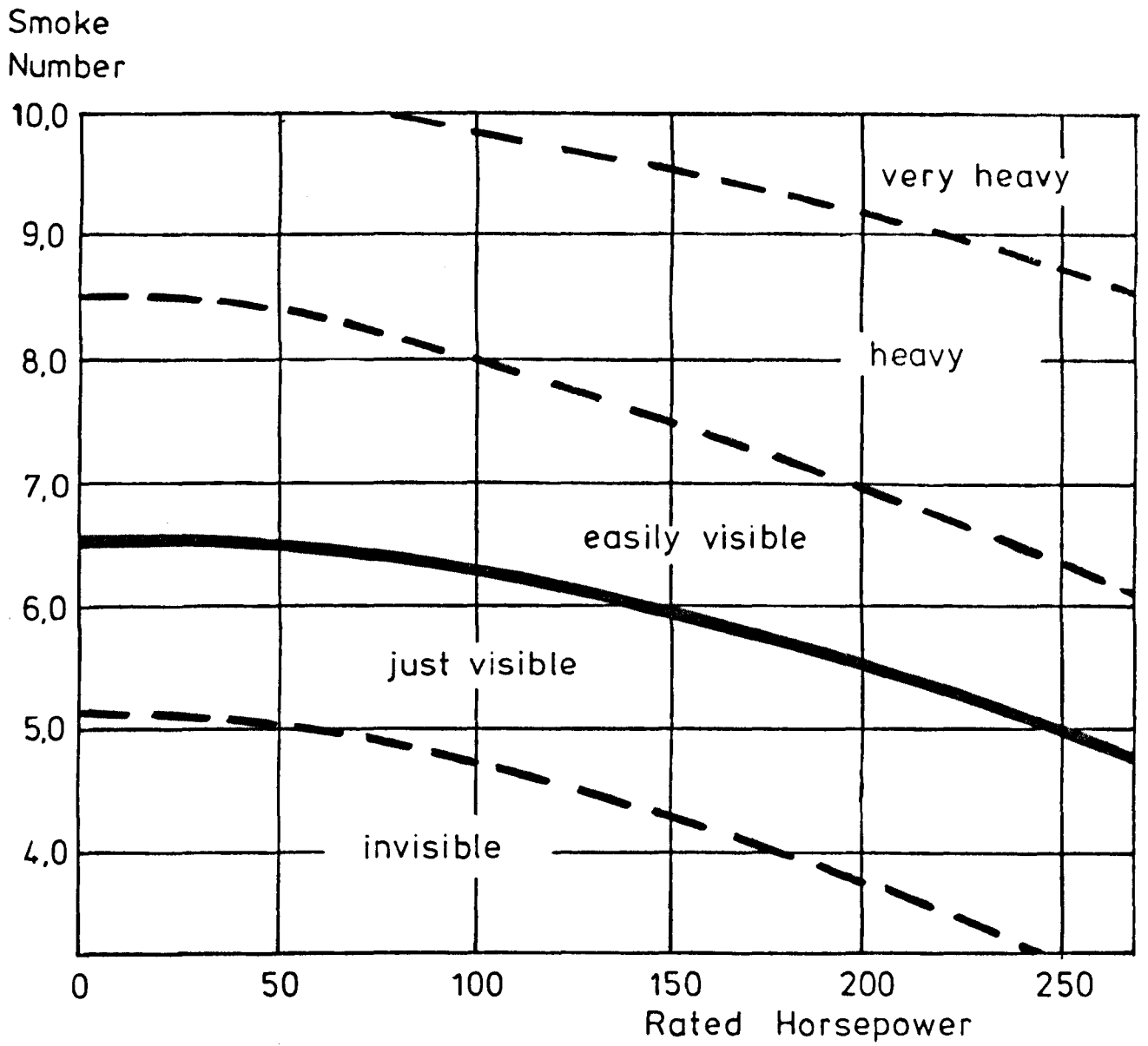
(Fitting the Sampling Head)

BOSCH FILTER-TYPE SMOKEMETER



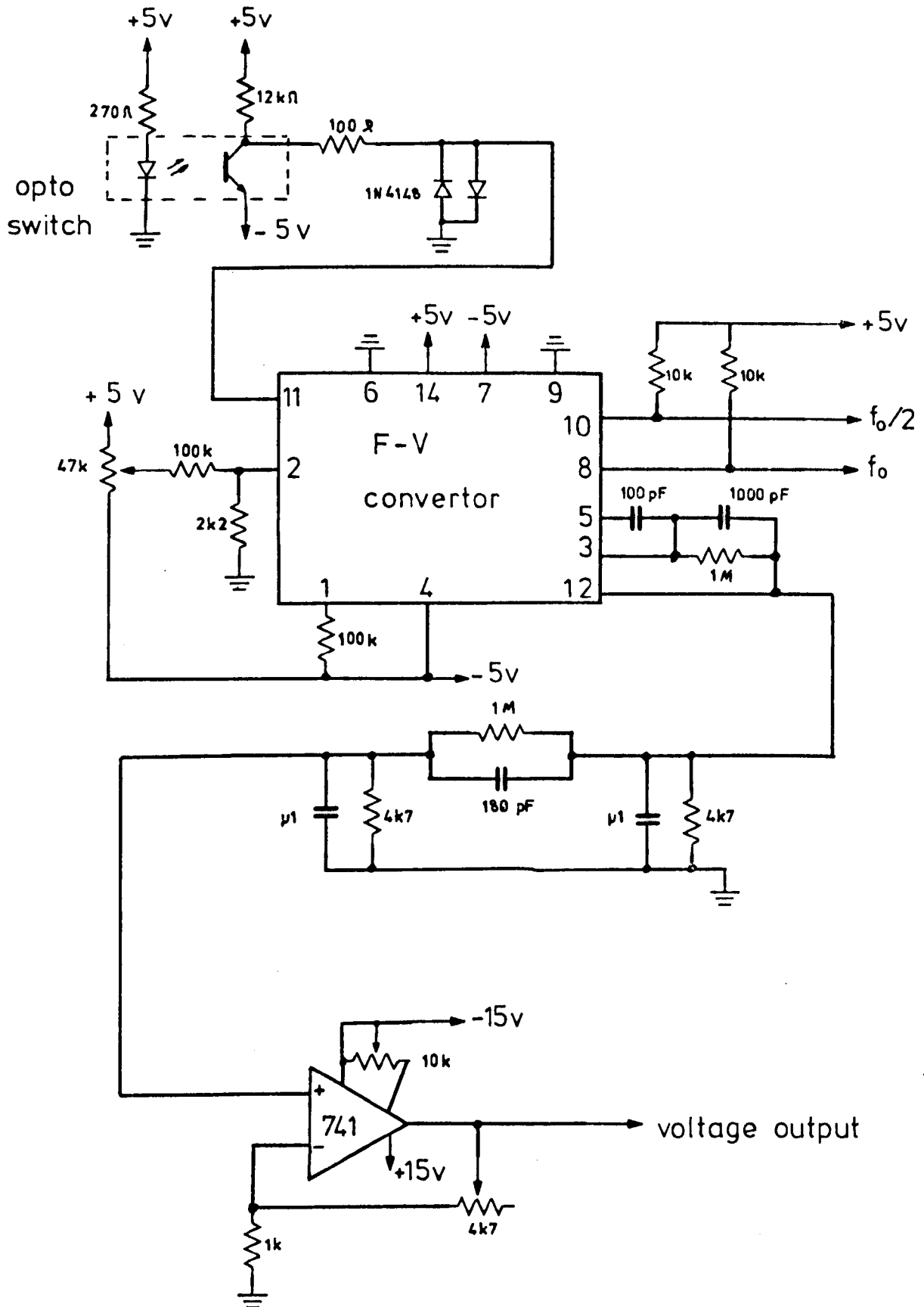
(a) Sampling unit
(b) Evaluating unit

FIGURE 4,33



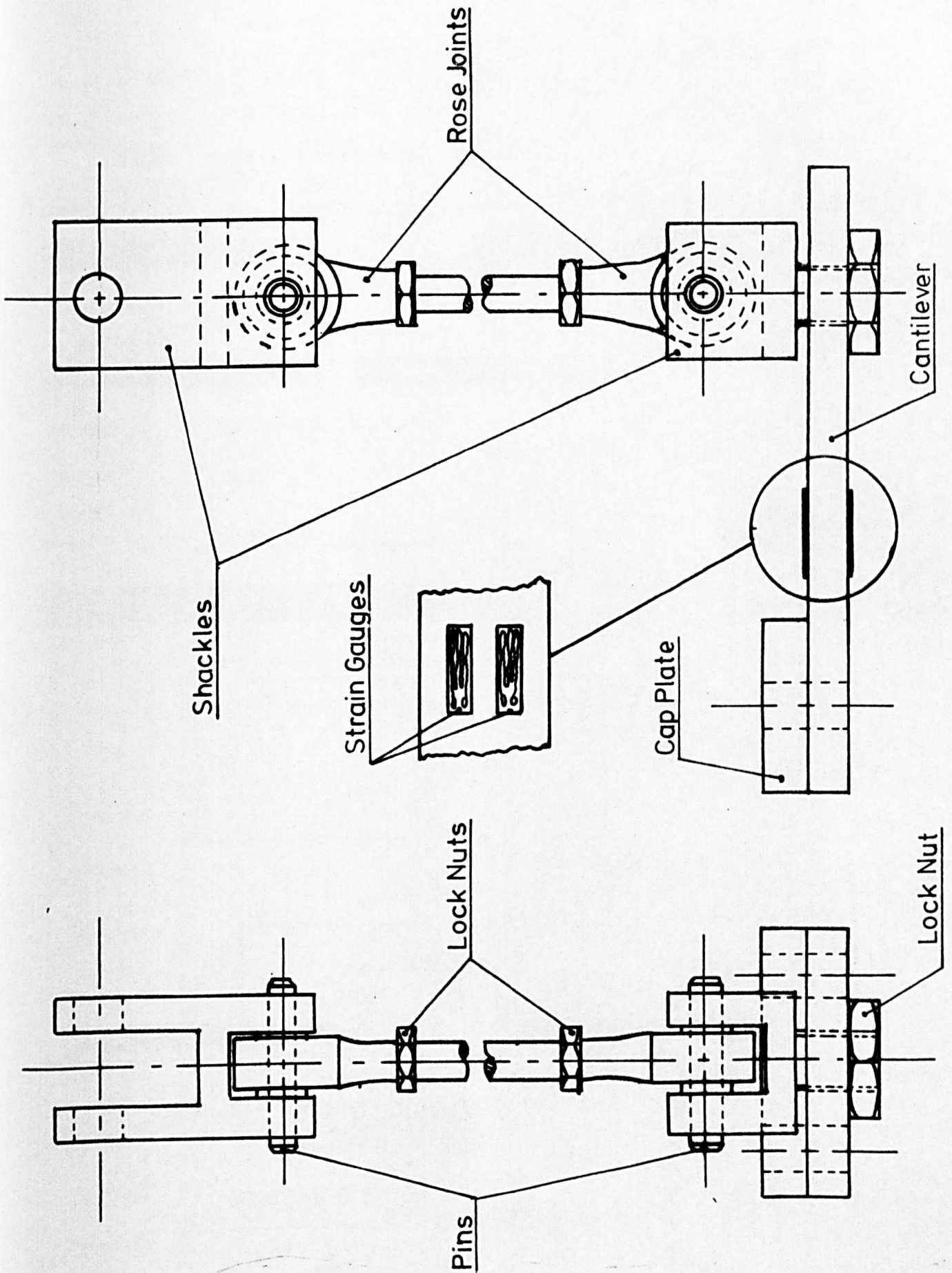
BOSCH SMOKE NUMBER GRAPH

FIGURE 4.34



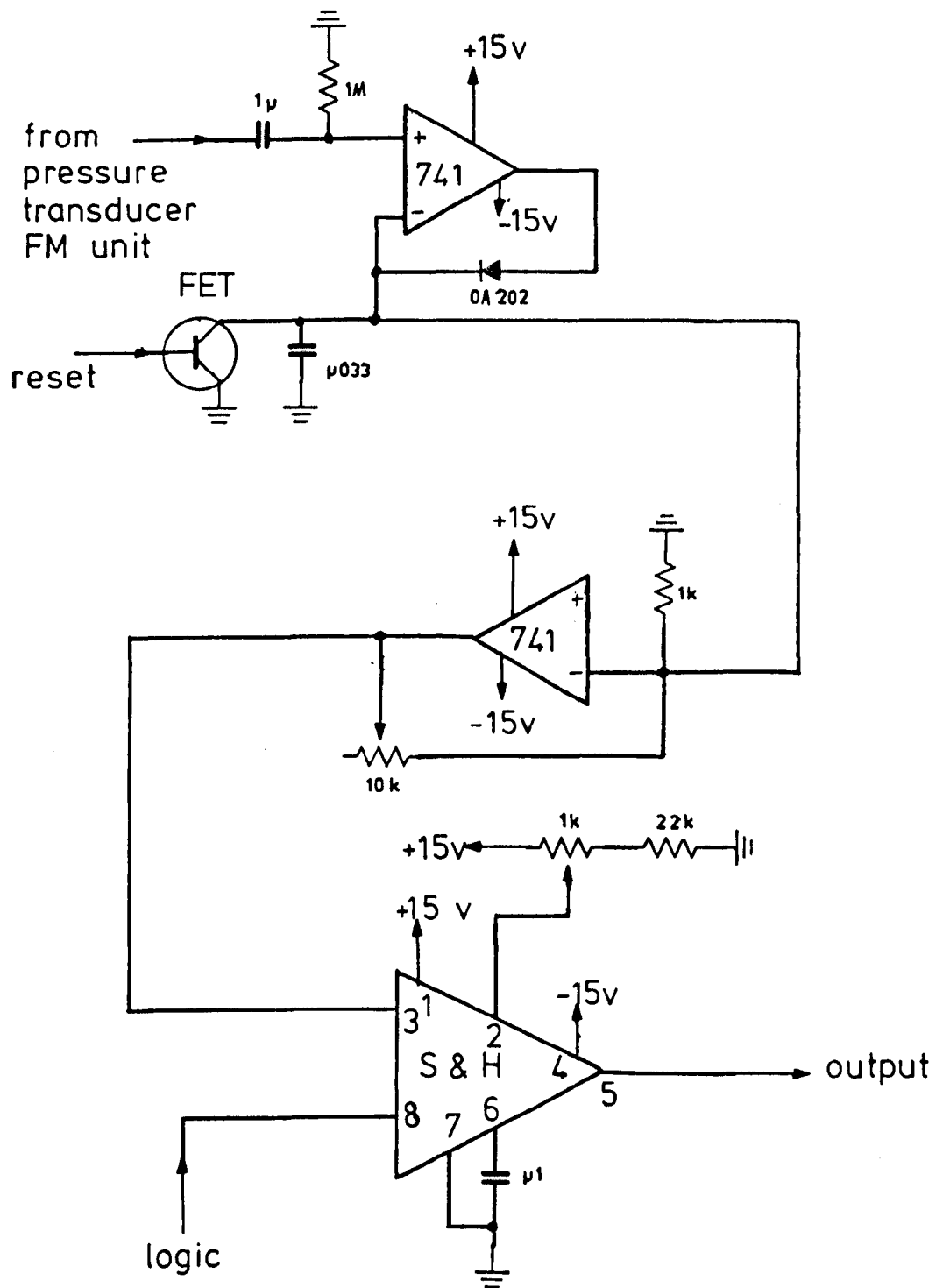
CIRCUIT FOR ANALOGUE VOLTAGE READOUT
OF ENGINE SPEED

FIGURE 4.35



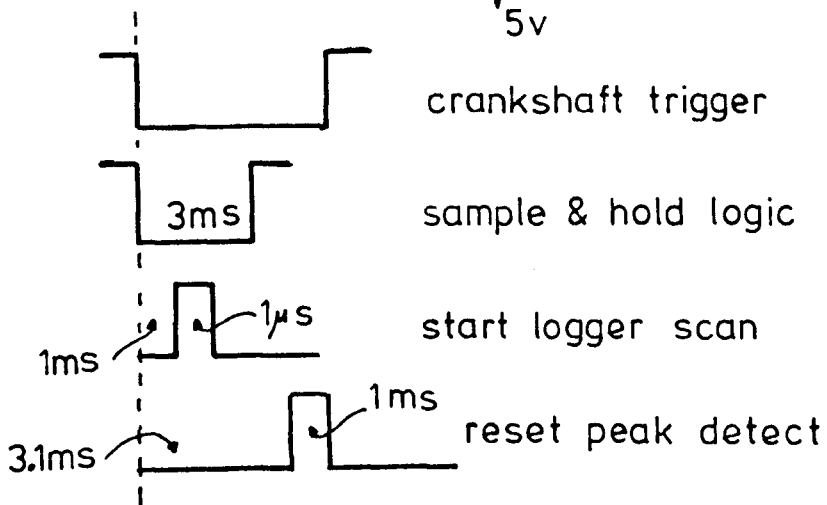
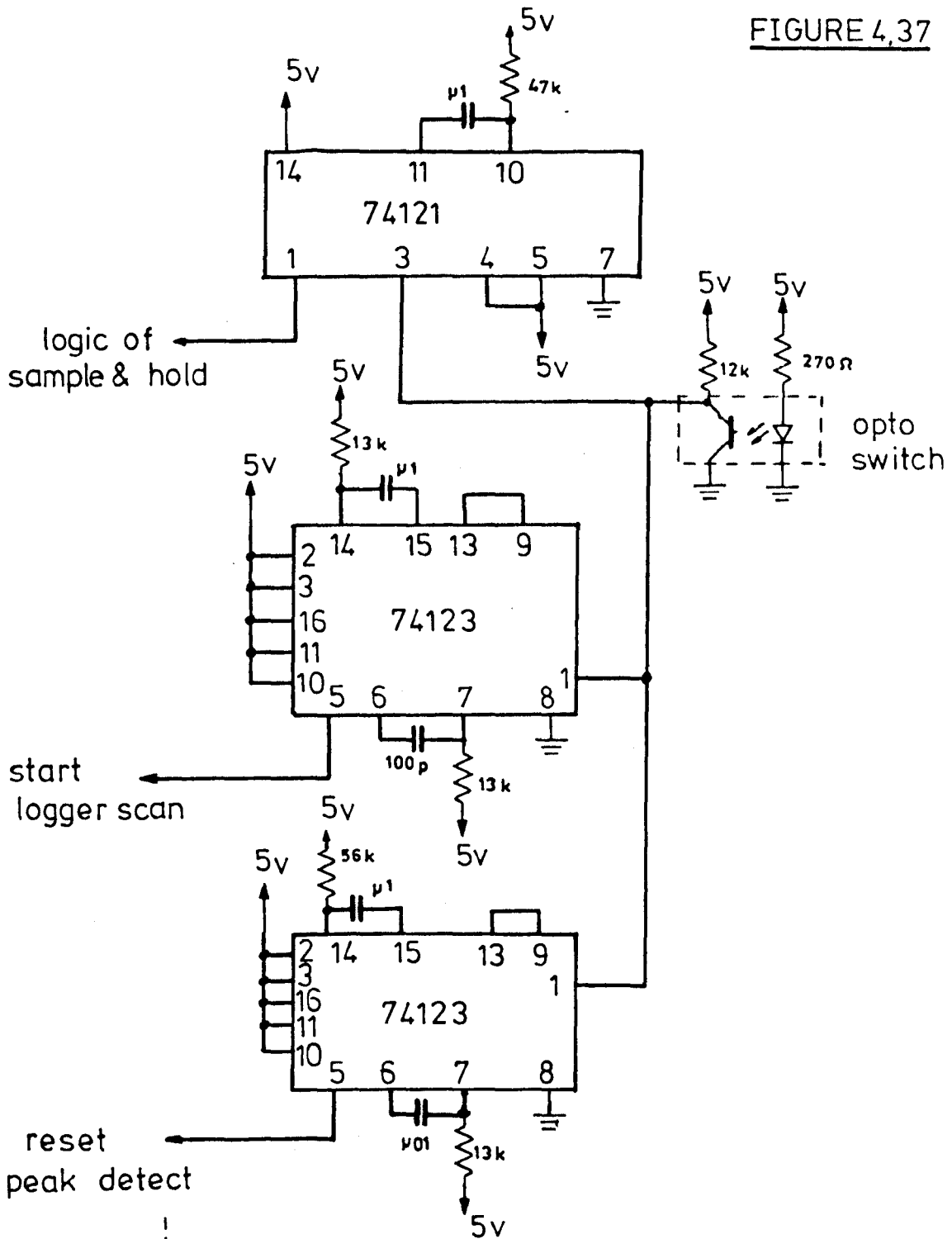
STRAIN GAUGED CANTILEVER SYSTEM

FIGURE 4.36



PEAK DETECT WITH SAMPLE & HOLD CIRCUIT

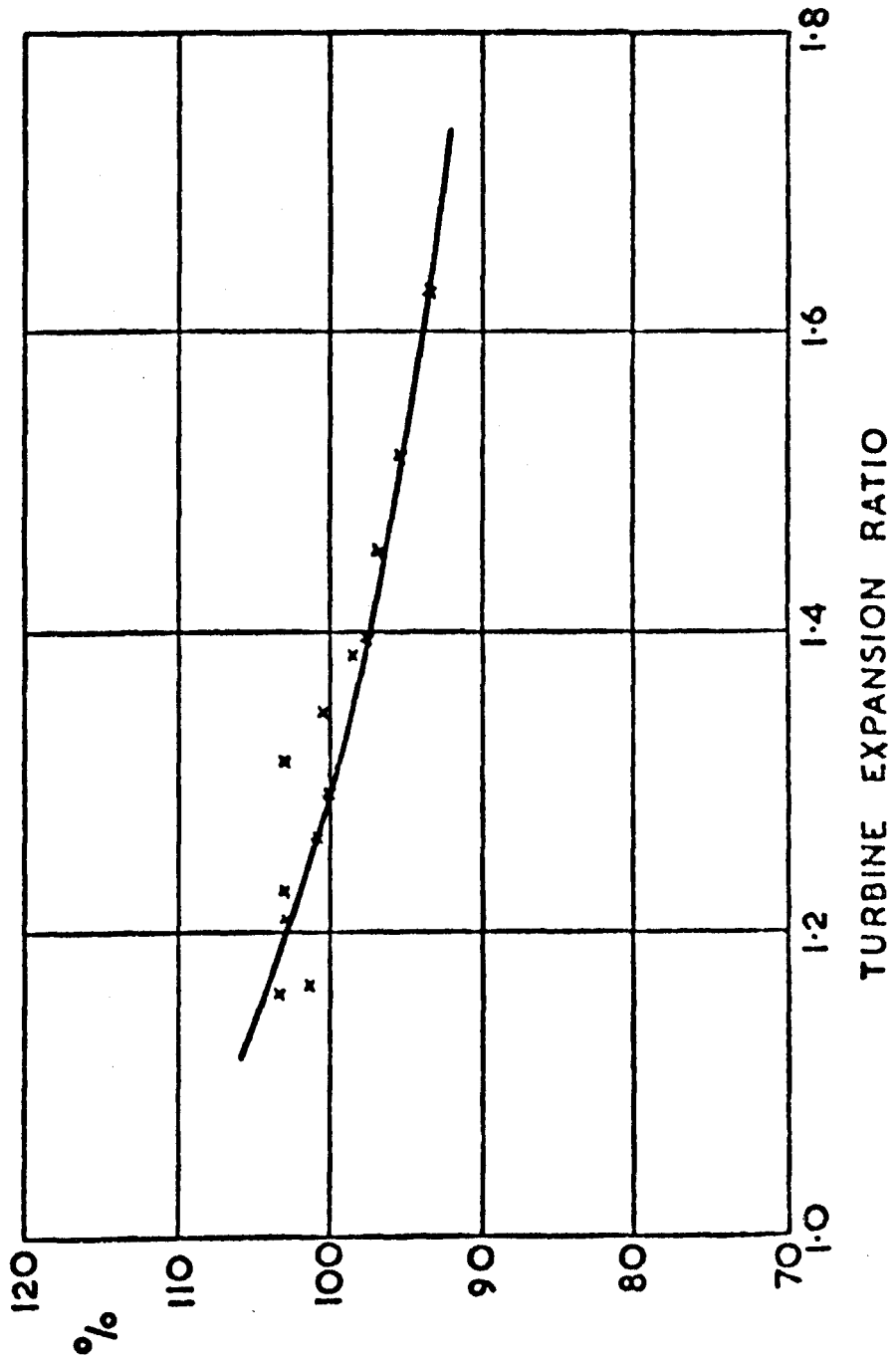
FIGURE 4.37



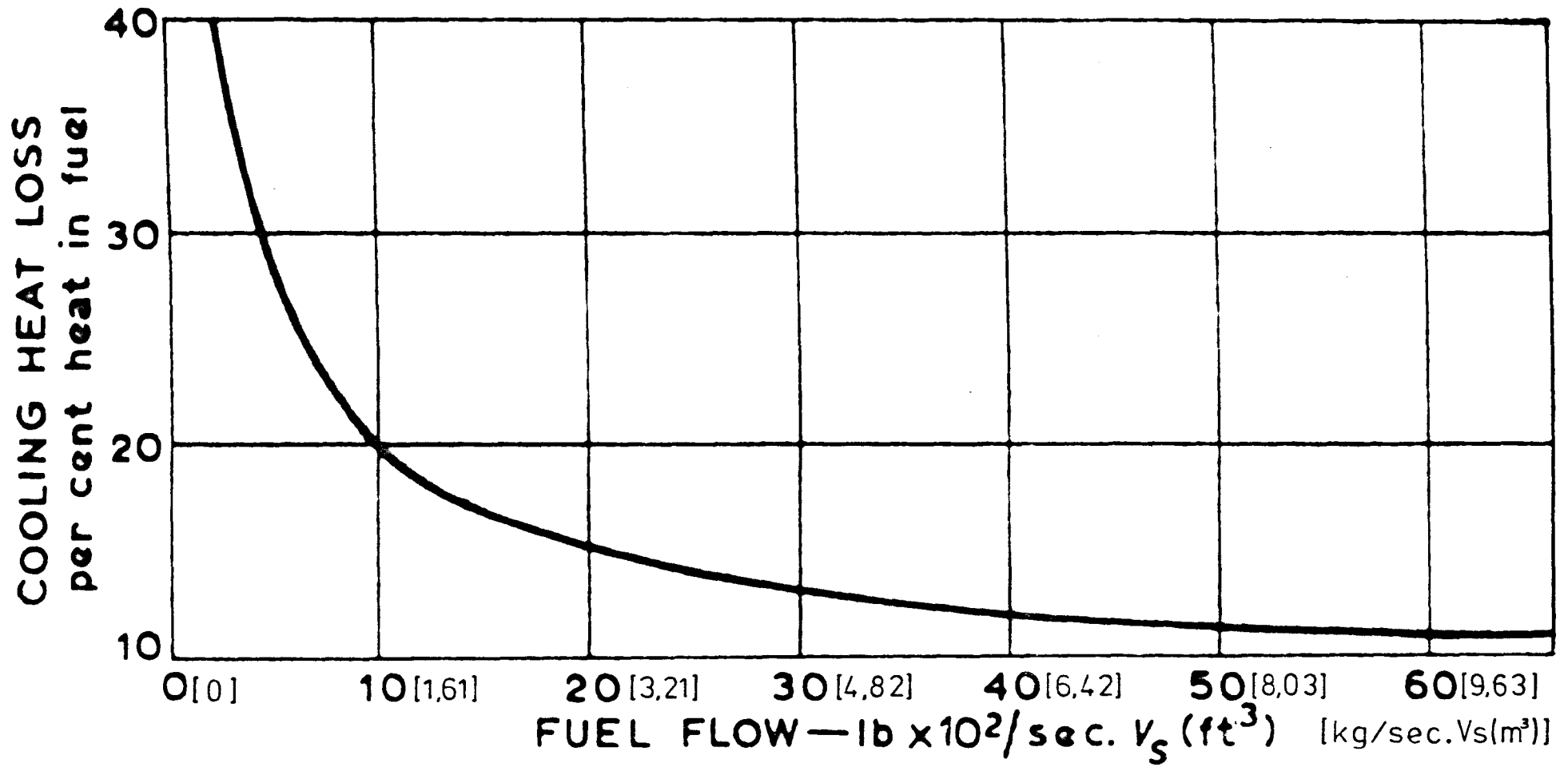
SAMPLE & HOLD/PEAK DETECT TIMING CIRCUIT

FIGURE 5.1

RELATIVE TURBINE APPARENT EFFICIENCY



TURBINE APPARENT EFFICIENCY



Cooling water heat loss

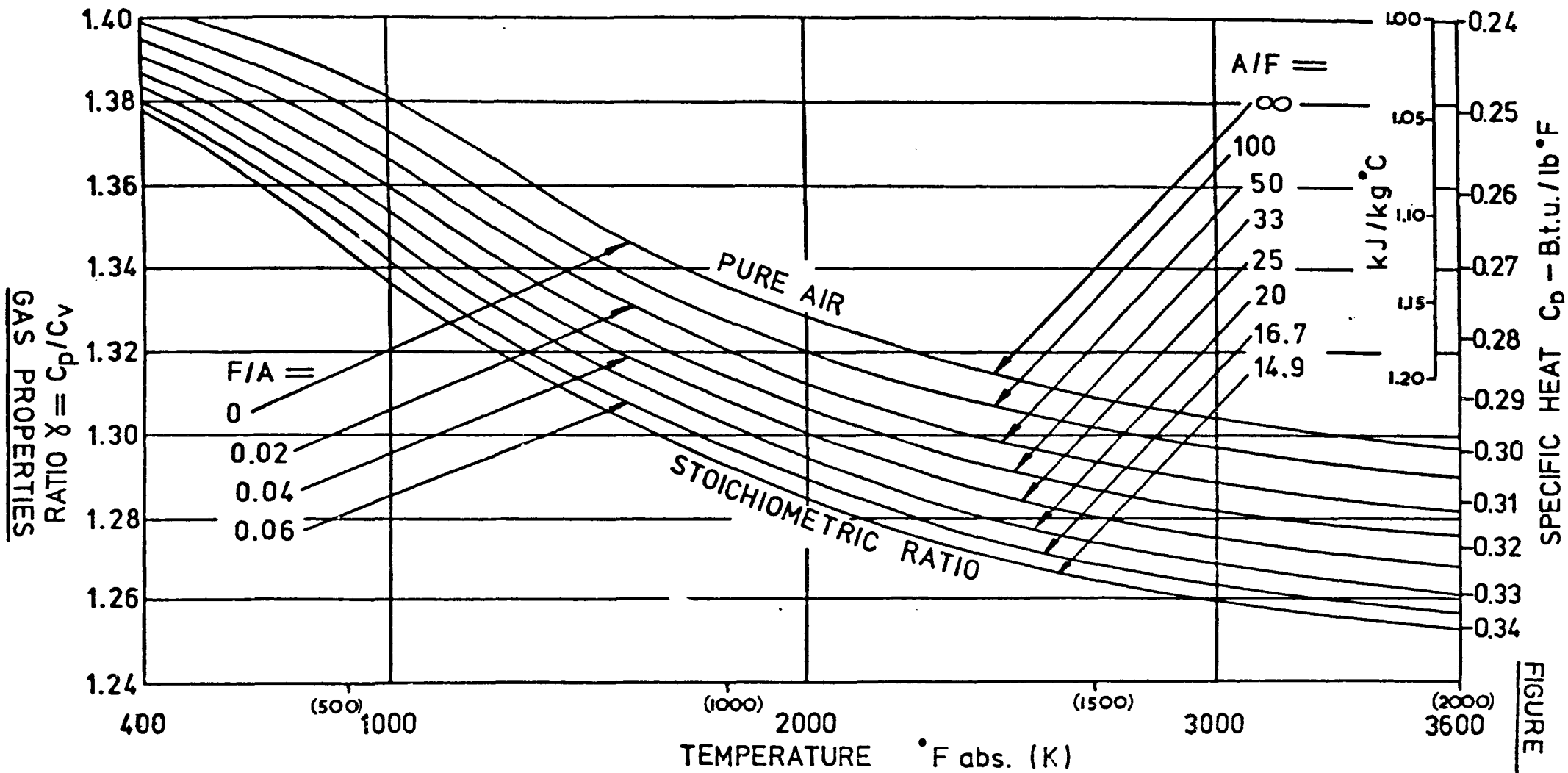
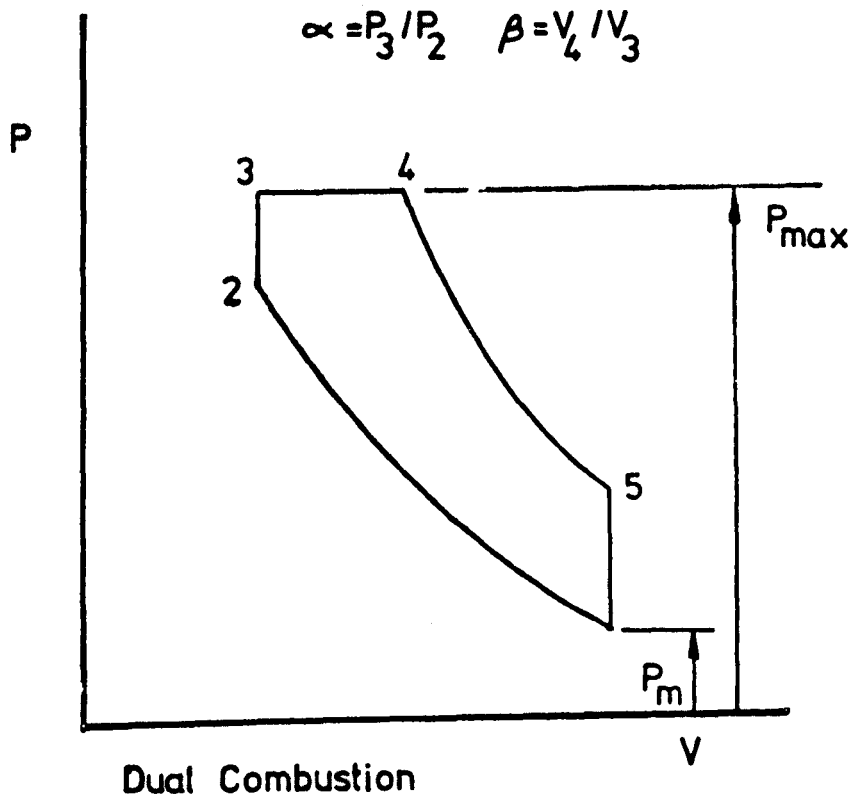
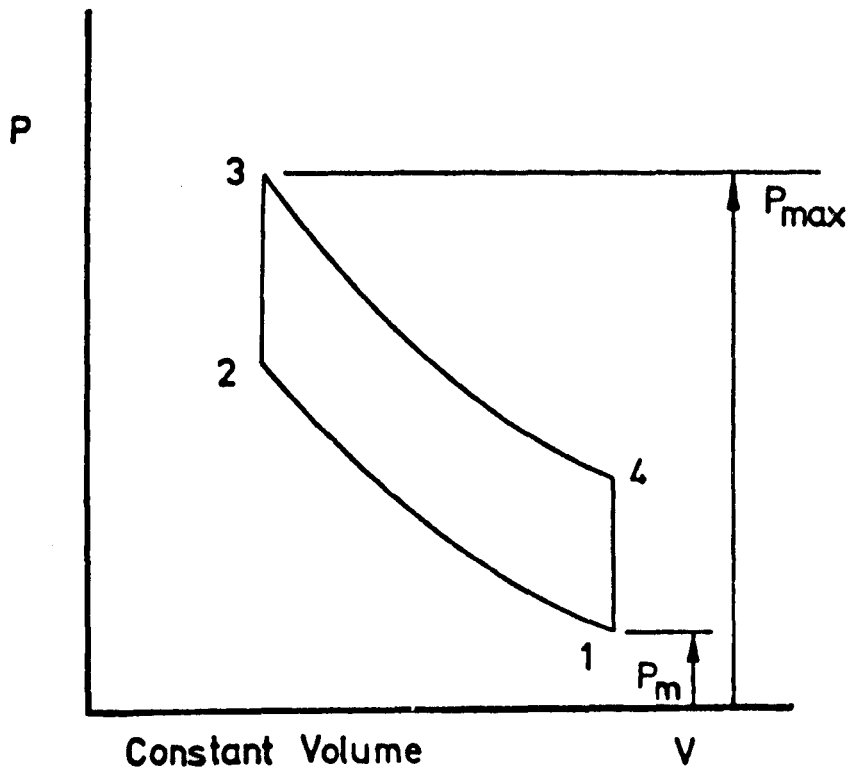


FIGURE 5.3

FIGURE 5.4



GRAPH TO ASSIST TURBOCHARGER MATCHING

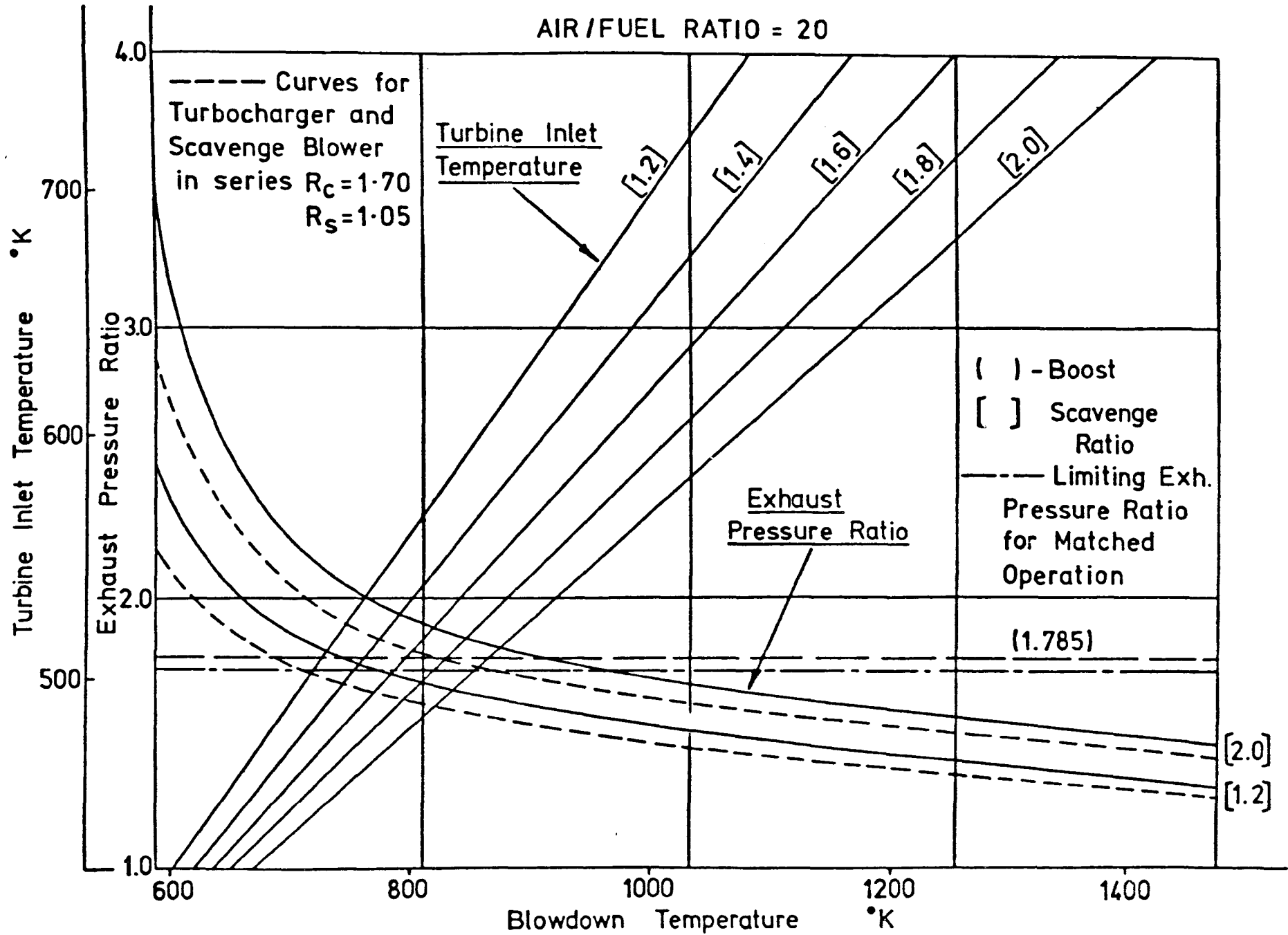
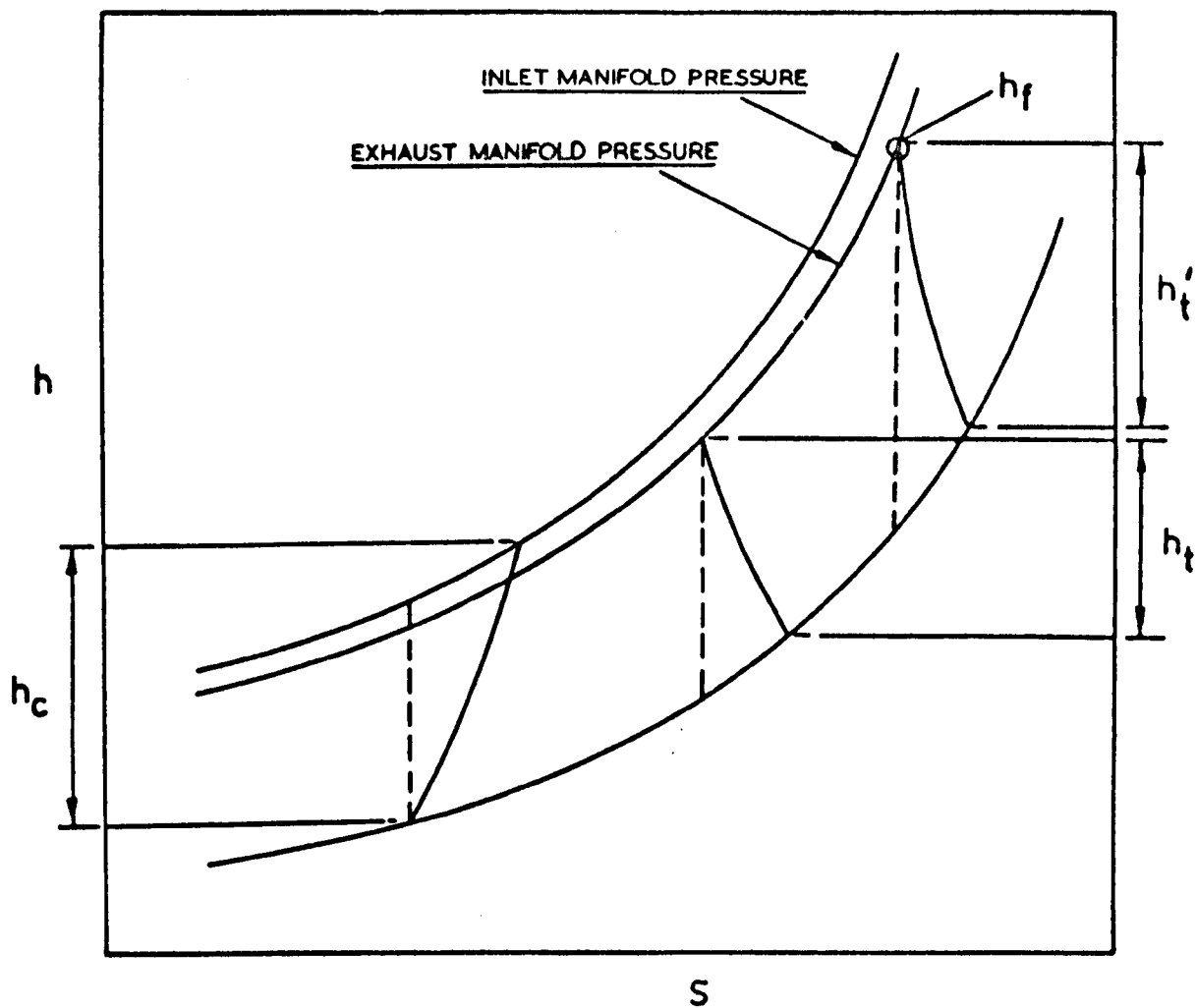


FIGURE 5.5



ENTHALPY / ENTROPY DIAGRAM
(for turbocharger)

h_c — Compressor Work

h_t — Turbine Work

h'_t — Turbine Work (with Combustion Chamber)

FIGURE 5.7

COMPUTED COMPRESSOR MAP
FOR HØLSET 3LD1

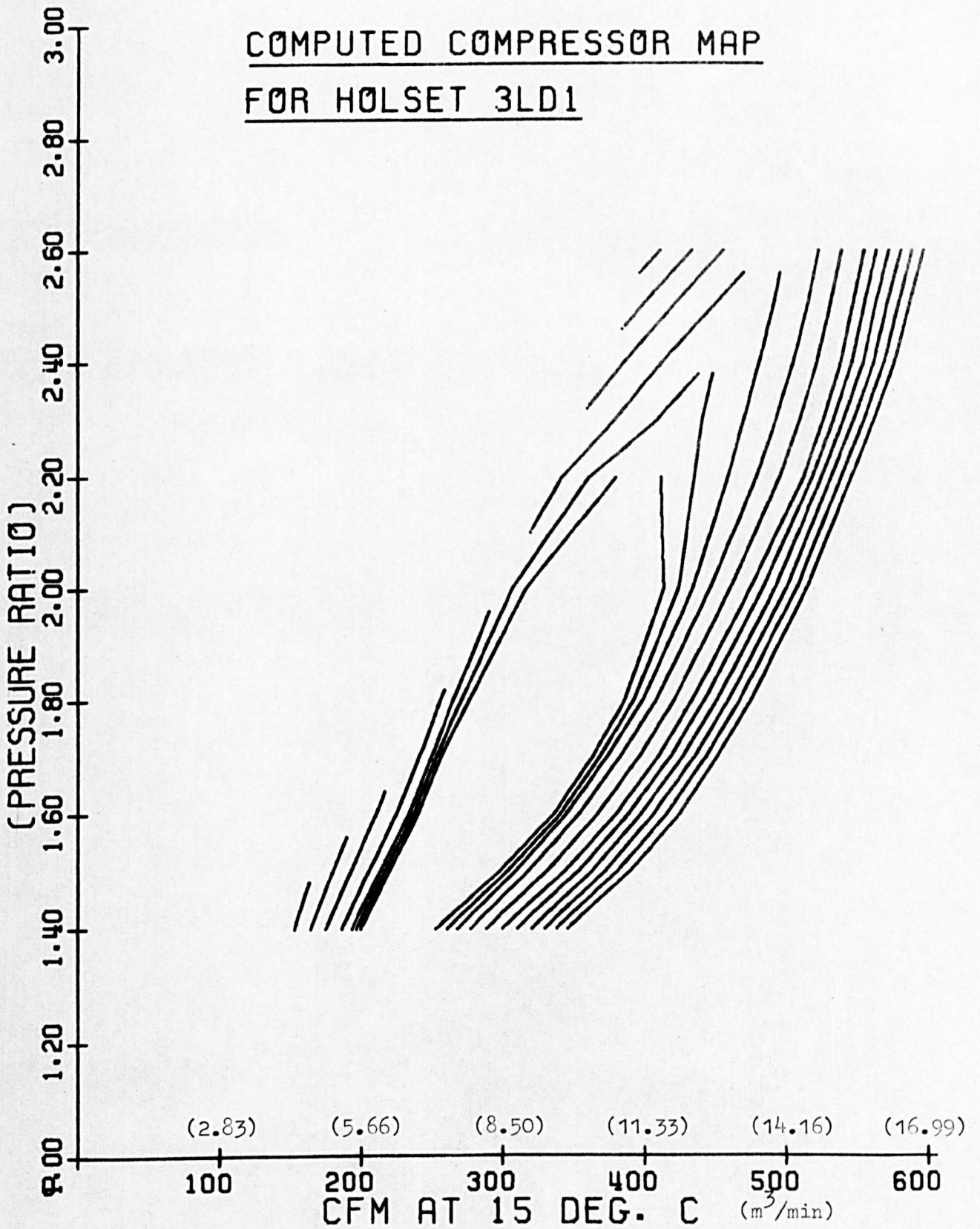
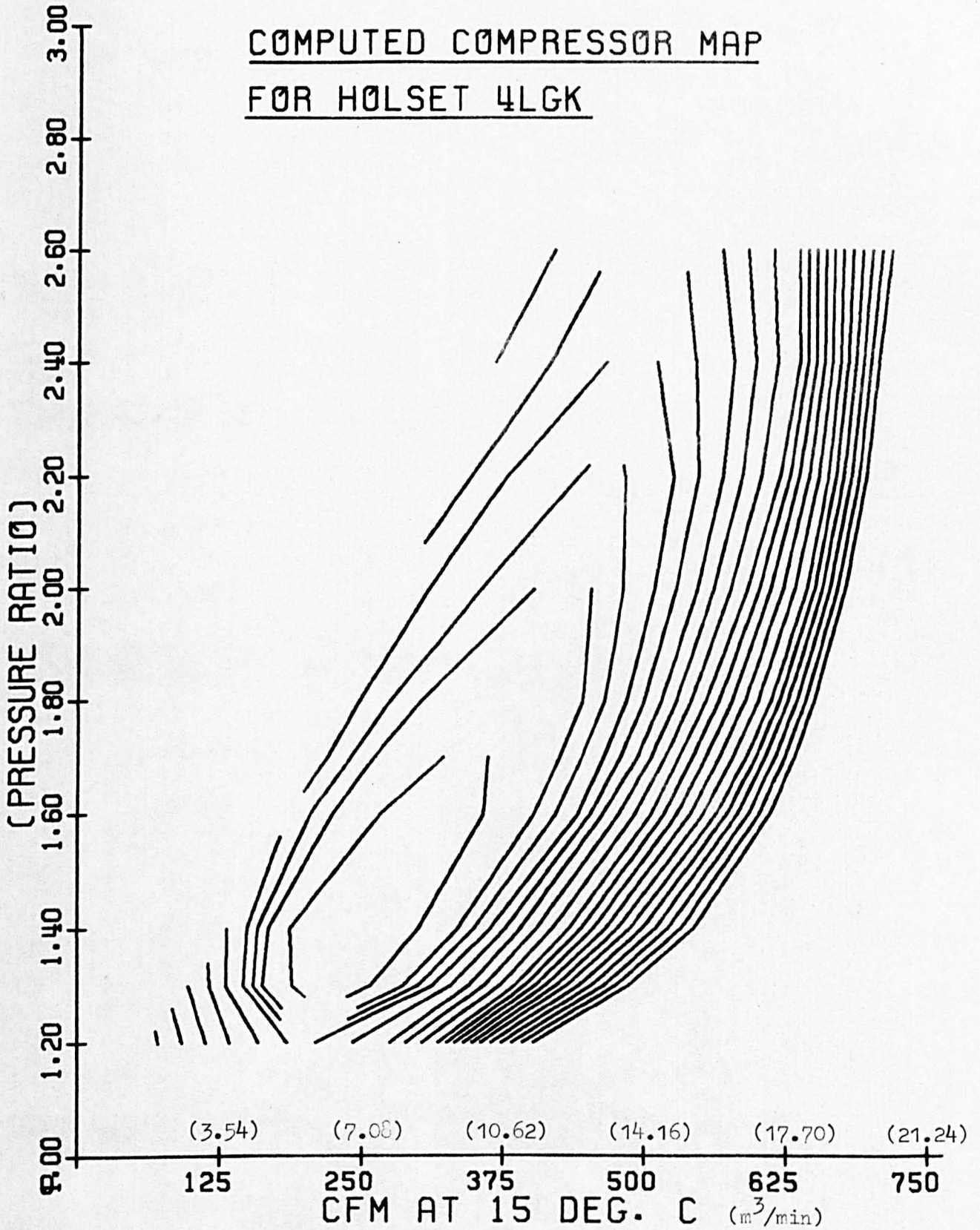


FIGURE 58

COMPUTED COMPRESSOR MAP
FØR HØLSET 4LGK



8 + 16 Channel Analog Multiplexer

CHARACTERISTICS

($V_s = \pm 15V$, $T_A = +25^\circ C$ unless otherwise noted)

MODEL	MUX 202-M/MUX 203
CHANNELS	8
ANALOG STATIC	
Ron ¹ Max. (+25° C)	1.5K Ω Max. ^A
MUX202-M (-55° C to +125° C)	1.8K Ω
Roff	.3pF 10 ¹¹ Ω
On Input Leakage (+25° C)	$\pm 1nA$
Input Range ²	$\pm 15V$
Off Input Leakage	$\pm .03nA$
Off Output Leakage (+25° C)	$\pm 1.0nA$
(-55° C to +125° C)	$\pm 250nA$ Max.
Output Capacitance	25pF
Input Capacitance	5pF
Address Input Capacitance	5pF

ACCURACY	
Gain	1.0
Gain Accuracy ³	.01% FS
Cross Talk	.010% @ 10kHz Max.
Common Mode ⁴	92DB

DYNAMICS	
Settling to .01%	4 μ sec Max.
Throughput Rate	250kHz
Address Time	.5 μ sec

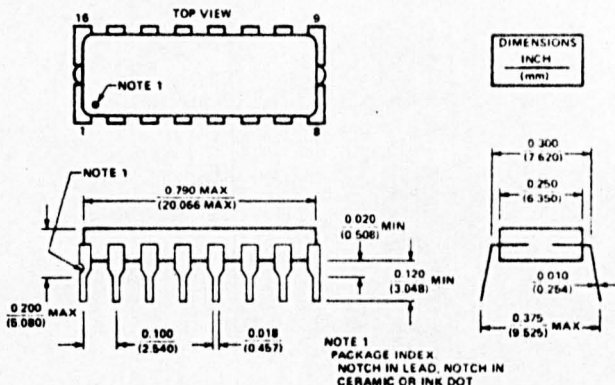
DIGITAL INPUT	
Break before make delay	80ns
Inputs ⁵	TTL ⁶ /CMOS ⁷ Compatible
Code	3 Bit + enable

POWER SUPPLY	
Standby Supply Current ^{8B}	+15V @ 2.0 mA -15V @ .5mA

ENVIRONMENTAL	
Operating Temperature Range	-55° C to +125° C
Storage Temperature Range	-65° C to +150° C

MECHANICAL	
Case dimensions	0.79" x 0.30" x 0.2"
Case Style	16 Pin DIP

MECHANICAL



NOTES:

- $V_{out} = \pm 10V$, $I_{out} = -100\mu A$
- Max safe overvoltage: $\pm V$ supply $\pm 20V$
- 20 meg load, $\pm 10V$ swing
- Applicable for usage of two devices in differential configuration.
- Max rating $V^+ = +4V$, $V^- = -4V$
- For TTL use 1K pull up to +5V
- "1" $> 4V$, "0" $< 0.8V$ @ 1 μA Max
- Max voltage between supply pins: 40V

PIN	FUNCTION	PIN	FUNCTION
1	A0	9	INB
2	ENABLE	10	IN 7
3	VDC	11	IN 6
4	IN 1	12	IN 5
5	IN 2	13	-VEE
6	IN 3	14	GND
7	IN 4	15	A1
8	OUT	16	A1

ORDERING INFORMATION

MODEL NUMBER	MIL-STD-883A PROCESSING
MUX 202-M/C	Class C
MUX 202-M/B	Class B
MUX 203	Commercial (0 to +70° C)

MODEL	MUX204
CHANNELS	16
INPUT/OUTPUT	
Voltage Range	$\pm 15V$
ON Resistance ¹	1.5k (1.8k max.)
OFF Resistance	0.3pF 10 ¹¹ Ω
ON Input Leakage	100pA
OFF Input Leakage	$\pm 0.3nA$
OFF Output Leakage	4nA
Output Capacitance	50pF
Input Capacitance	5pF

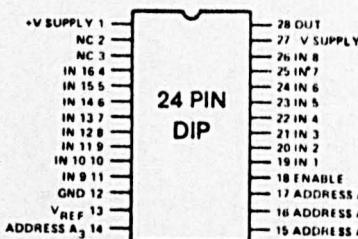
THROUGHPUT	
Settling Time to 0.01%	4 μ s max.
Throughput Rate	250 kHz
Addressing Time	500ns

ACCURACY	
Gain	1.0
Gain Accuracy ²	0.01% F.S.
Crosstalk	0.01% @ 10 kHz

CODING	
Inputs	MOS ³ /CMOS ⁴ /DTL/TTL ⁵ Compatible
Code	4 bit binary + enable

ENVIRONMENTAL	
Temperature Range	0 to 70° C
Storage	-65° C to +150° C
Power Supply	+15V @ 5mA; -15V @ 2mA

- $V_{OUT} = \pm 10V$, $I_{OUT} = -100\mu A$
- 20 meg load, $\pm 10V$ swing
- $V_{REF} = +10V$, "1" $> 6V$, "0" $< 0.8V$ @ 5 μA
- Logic "1" $> 4V$, Logic "0" $< 0.8V$ @ 5 μA
- For TTL/DTL use 1k Ω pull up to +5V



A ₃	A ₂	A ₁	IN	CH/OUT
H	L	L	H	1
L	L	L	H	2
L	L	L	H	3
L	L	L	H	4
L	L	L	H	5
L	L	L	H	6
L	L	L	H	7
L	L	L	H	8
H	L	L	H	9
H	L	L	H	10
H	L	L	H	11
H	L	L	H	12
H	L	L	H	13
H	L	L	H	14
H	H	L	H	15
H	H	L	H	16

ORDERING INFORMATION

MODEL	NUMBER OF CHANNELS
MUX204	16 SINGLE-ENDED

FIGURE 6.2

SPECIFICATIONS

MODEL	ADC 550		
MEASUREMENT TYPE	General Purpose		
DIGITAL			
Conversion Type	Successive Approximation		
Output Codes	Two's Complement, Offset Binary, Binary*		
Resolution	12 Bits — All Units; Parallel & Serial Out		
Conversion Time	30µs (12 Bits)		
Output Type	TTL Compatible; drives up to 4 TTL loads		
Strobe Input (Convert)	Positive Pulse — 100 nS wide min, Standard TTL Leading edge ("0" to "1") resets converter Trailing edge ("1" to "0") initiates conversion		
Busy Bit Output (Status)	"1" during conversion; "0" after conversion		
ANALOG			
Standard Input	±5V, 0 to +10V, Input Z > 6K		
Option — G	±10V; Input Z > 12K		
Internal Reference	Yes		
ACCURACY & STABILITY			
Relative Accuracy ADC 550-12	±0.0125%		
ADC 550-10	±0.05%		
Scale Factor & Offset	0.1% F.S. nominal; adjustable to zero error		
STABILITY OPTIONS			
	— E	— S	— LD
Accuracy vs Temp**	50	30	15 PPM/°C
Linearity vs Temp	15	15	7 PPM/°C
Precision Network Type	Matched Metal Film		
RELIABILITY			
Construction	Standard Encapsulated, Factory Repairable		
Active Component Types	All Hermetically Sealed — No Plastics		
Factory Burn-In	72 Hour min.		
Warranty Period	3 Years		
ENVIRONMENTAL			
Operating Temperature	0 to 70°C		
— MIL Versions	-55°C to +125°C		
Power Supply	+15V @ 30mA, -15V @ 30mA, +5V @ 200mA		
Power Supply Rejection	±0.002% /% for -15V, ±0.002% /% for +15V		

* Selection by pin interconnection.
 ** Accuracy vs. temperature includes all effects of offset, gain and linearity.

ORDERING INFORMATION

MODEL	TEMPCO	PRICE (1-9)
ADC550-10-E	50PPM/°C	\$ 85.00
ADC550-10-S	30PPM/°C	95.00
ADC550-10-LD	15PPM/°C	150.00

For -55°C to +125°C operation add suffix -MIL. \$45.00 ext

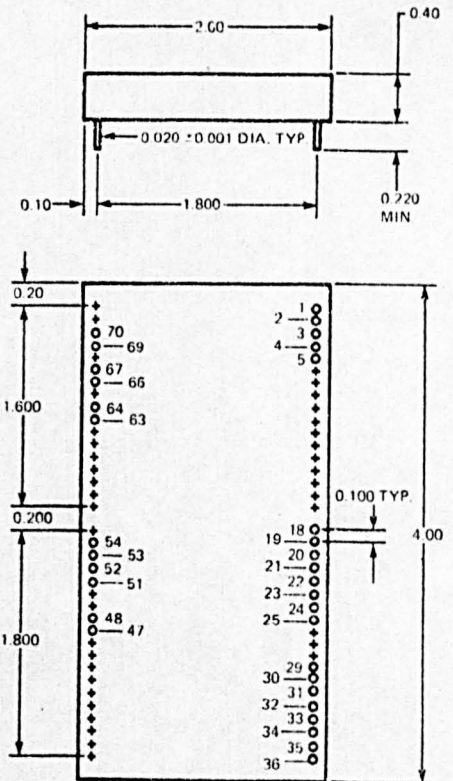
MODEL	TEMPCO	PRICE (1-9)
ADC550-12-E	50PPM/°C	\$115.00
ADC550-12-S	30PPM/°C	125.00
ADC550-12-LD	15PPM/°C	175.00

For -55°C to +125°C operation add suffix -MIL. \$90.00 ext

Options:
 ±10V Input--Add suffix -G. No extra charge.

-MIL VERSIONS
 -MIL versions are more mechanically rugged but electrically identical to the commercial ones. Continuous operation above 80°C is not recommended in that internal precision resistor networks could be permanently affected. Accuracy vs. Temperature for -MIL units over the range of -20°C to +70°C is the same as for commercial ones. Outside this range the temperature coefficients may significantly increase.

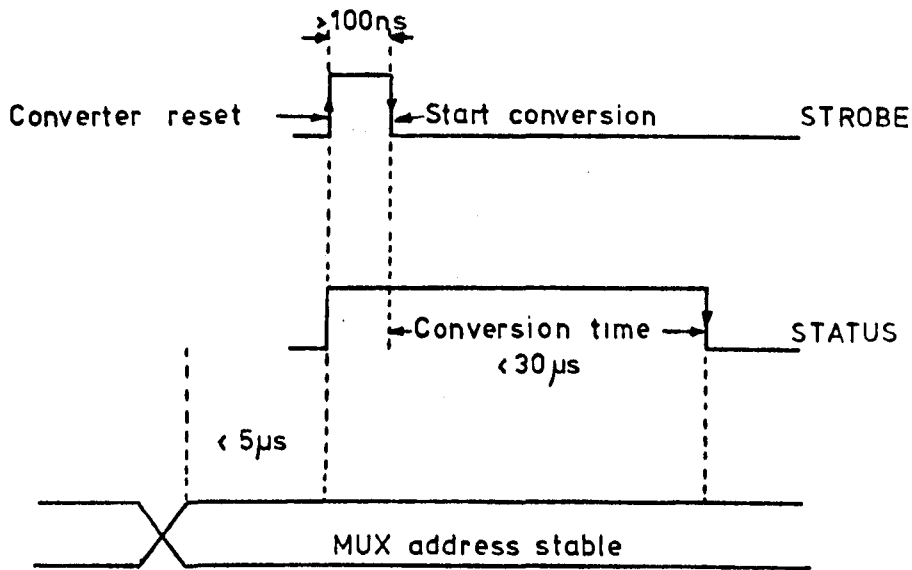
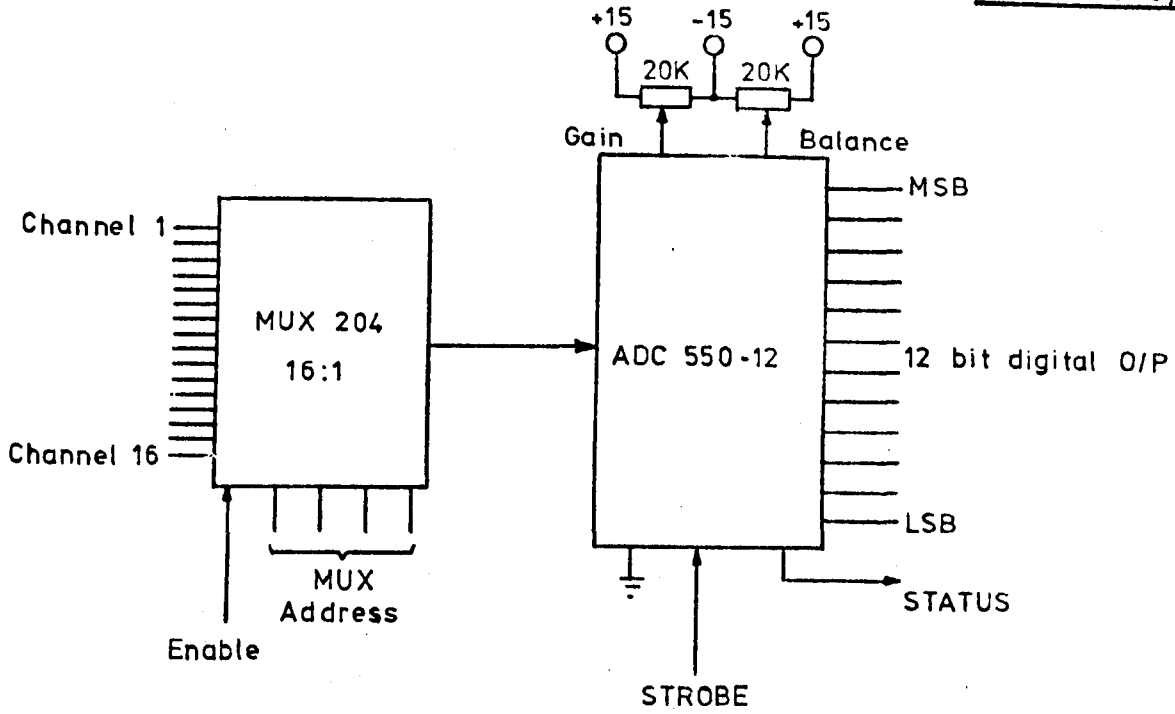
MECHANICAL



NOTES:
 1. UNLESS OTHERWISE SPECIFIED DIMENSION TOLERANCES ARE, DECIMALLY, .XX ± 0.02, .XXX ± 0.005
 2. (0) DENOTES A PIN, (1) DENOTES NO PIN.

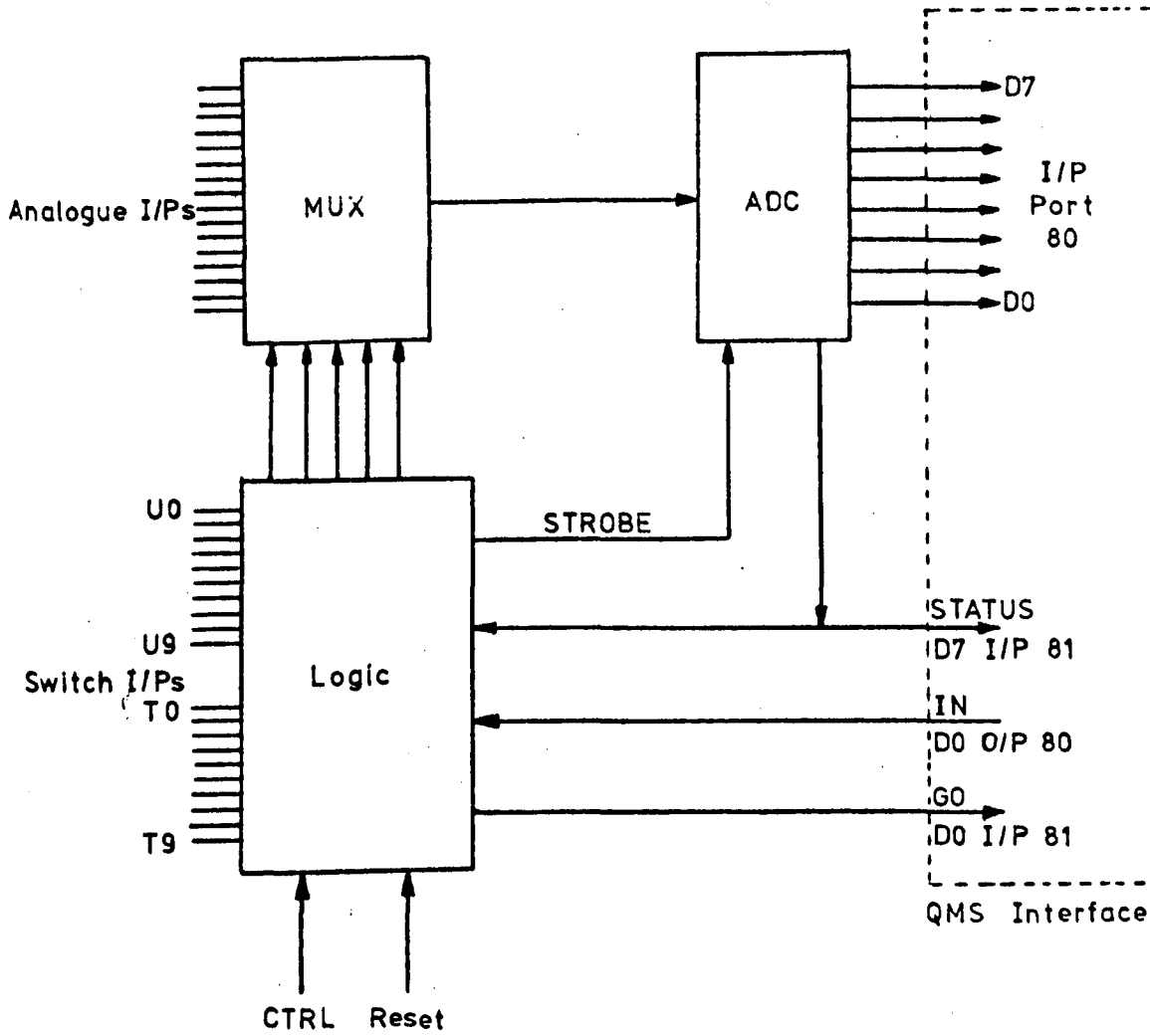
ANALOGUE TO DIGITAL CONVERTOR

FIGURE 6.3

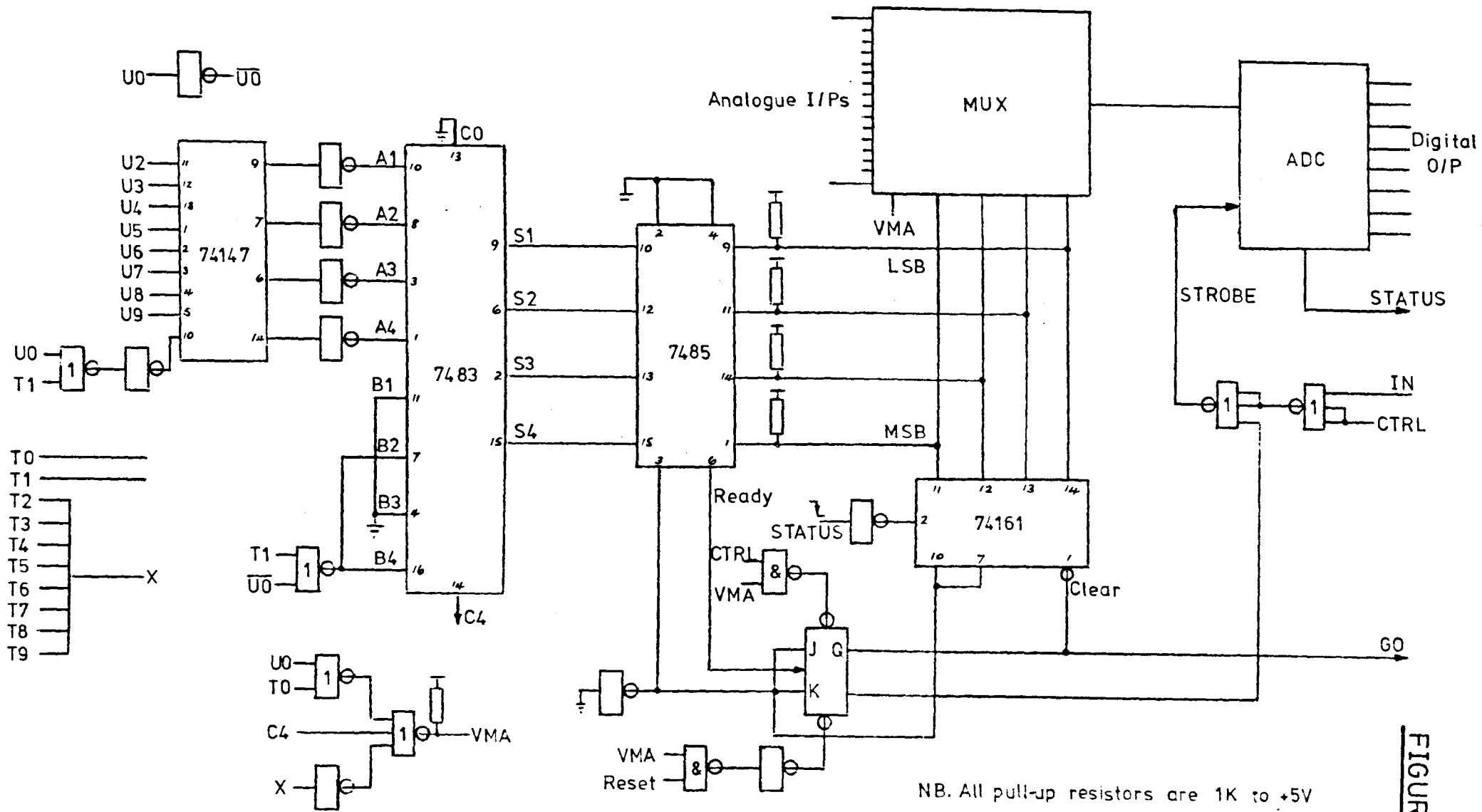


HYBRID DATA ACQUISITION SYSTEM

FIGURE 6.4



DATA ACQUISITION SYSTEM ORGANISATION



CONTROL LOGIC

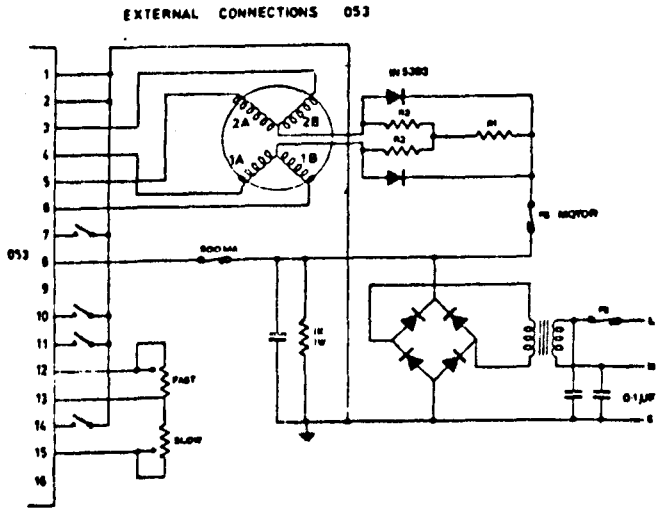
FIGURE 6.5

FIGURE 6,6

General Details

Ambient temperature range
Board Edge Connector (standard)
Mating Connector
Alternative board connector

0 - 45°C
16 way Varicon
Varicon 5007-016-166-001
Screw type - 16 way Klippon



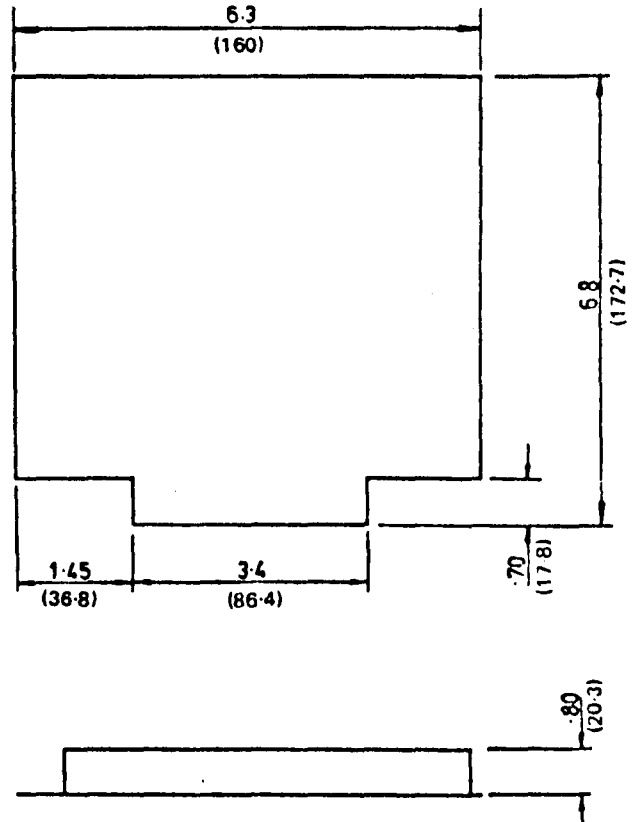
$R1 = R/2 \quad R2 = R$

Where $R = \frac{24 - V_m}{2 I_{ph}}$

& V_m = Rated motor coil voltage
 I_{ph} = Rated current/phase when two phases are energised.

Wattage $R1$ & $R2 = (24 - V_m) I_{ph}$ watts.

Note It is recommended that resistors of approximately twice the calculated wattage are normally used to avoid very high surface temperatures.



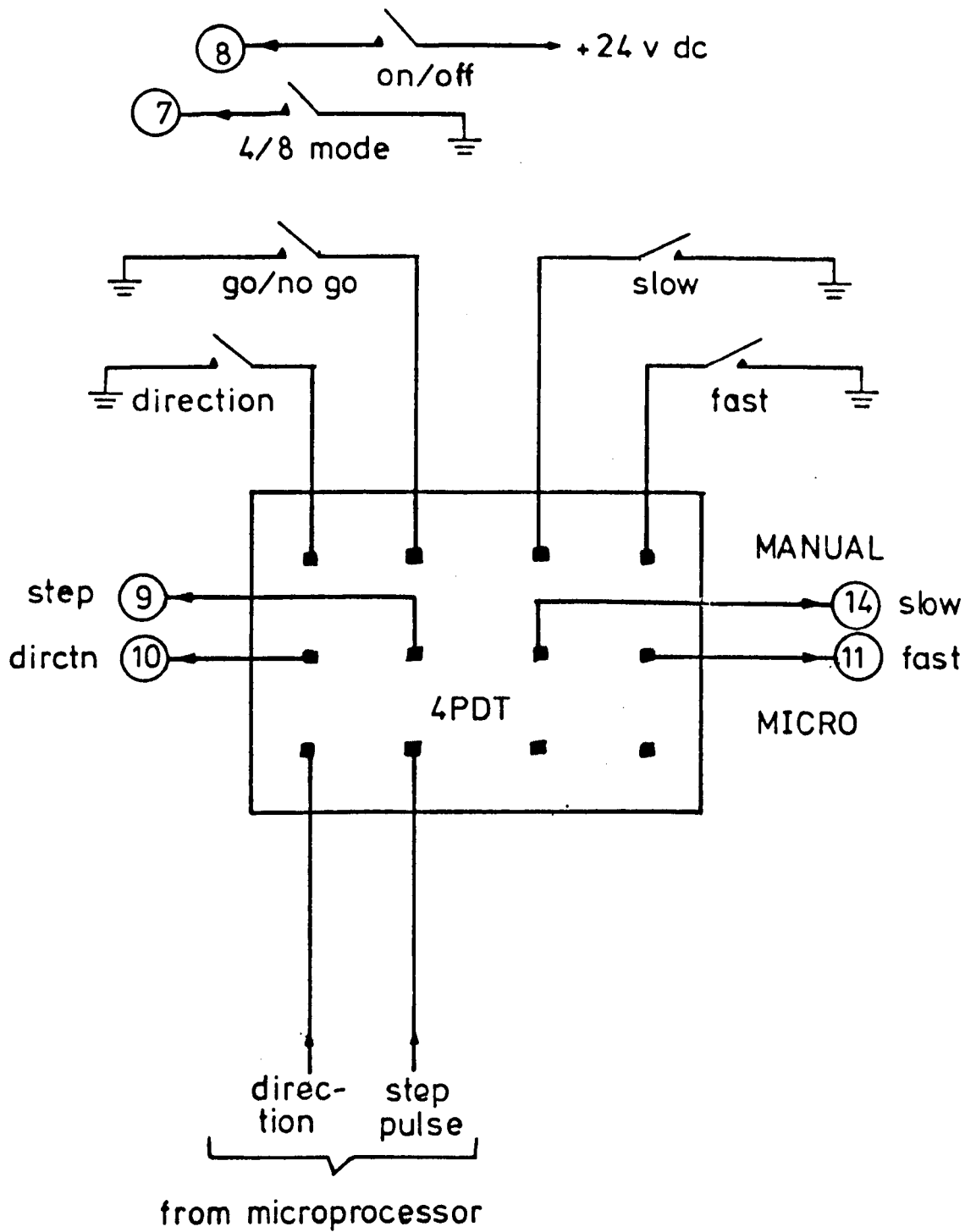
WEIGHT 12 OZ.
DIMS. IN INS. (mm)

Edge Connector Designation

PIN	CONNECTOR
1	0V
2	0V
3	MOTOR COIL 2B
4	MOTOR COIL 1A
5	MOTOR COIL 2A
6	MOTOR COIL 1B
7	4/8 MODE CONTROL
8	+24V dc
9	STEP PULSE
10	DIRECTION
11	FAST CONTROL
12	EXT. FAST ADJUST
13	EXT. SPEED ADJ. COMMON
14	SLOW CONTROL
15	EXT. SLOW ADJUST
16	+5V dc (monitor)

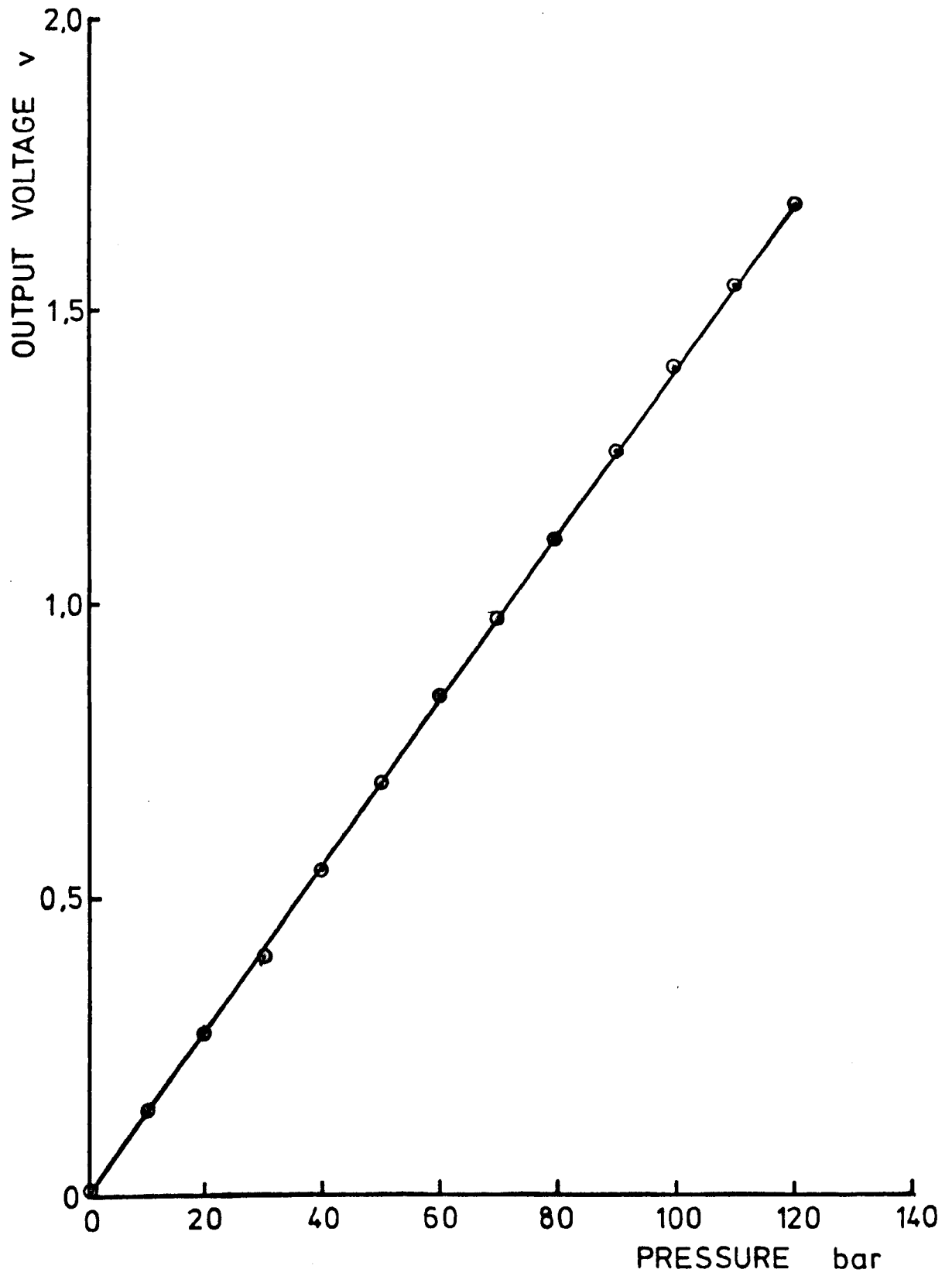
STEPPER MOTOR DRIVE

FIGURE 6.7



STEPPER MOTOR CONTROL CIRCUIT

FIGURE 7.1



S.I. T500/H524 PRESSURE TRANSDUCER CALIBRATION

FIGURE 7.2

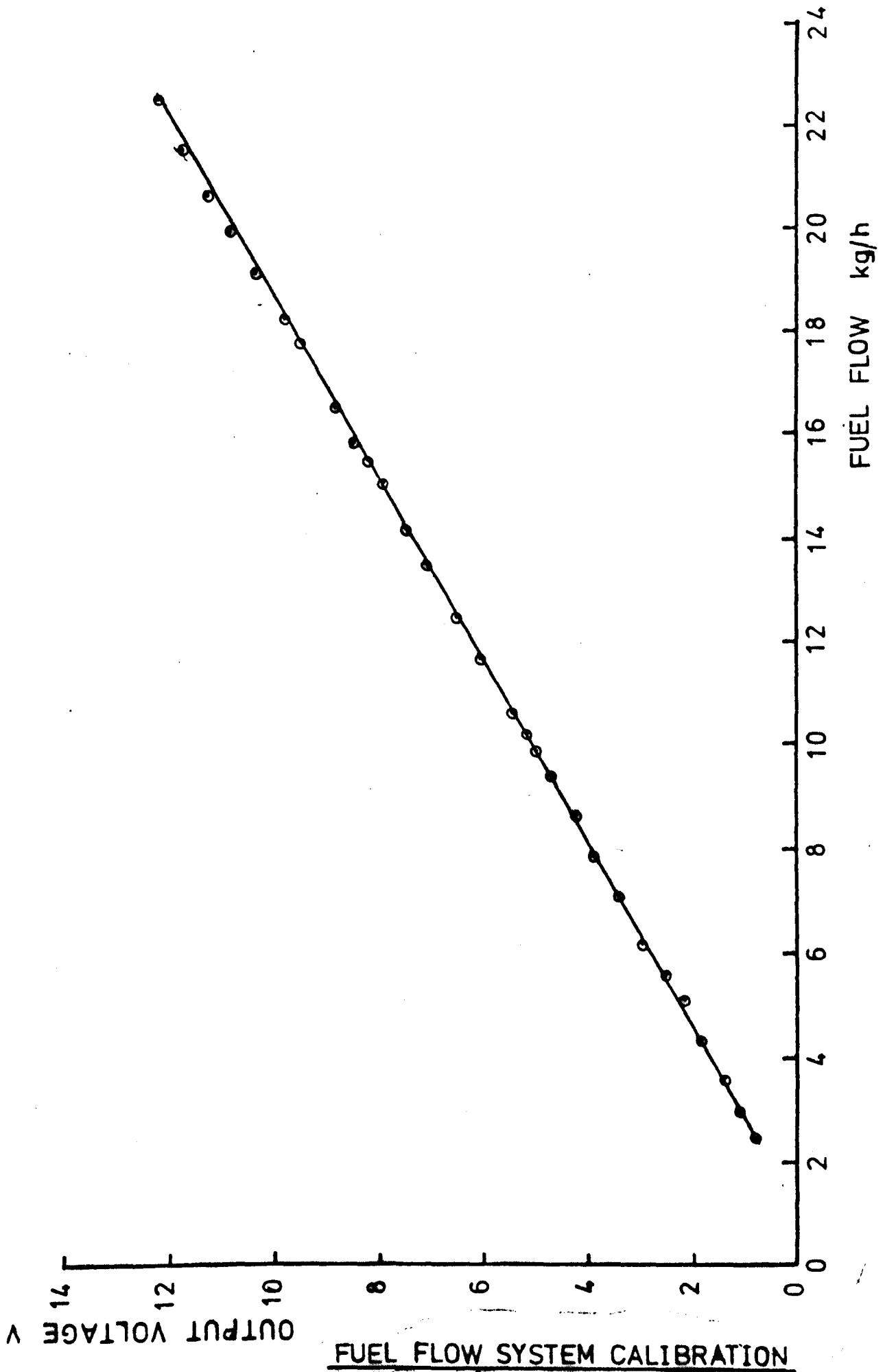
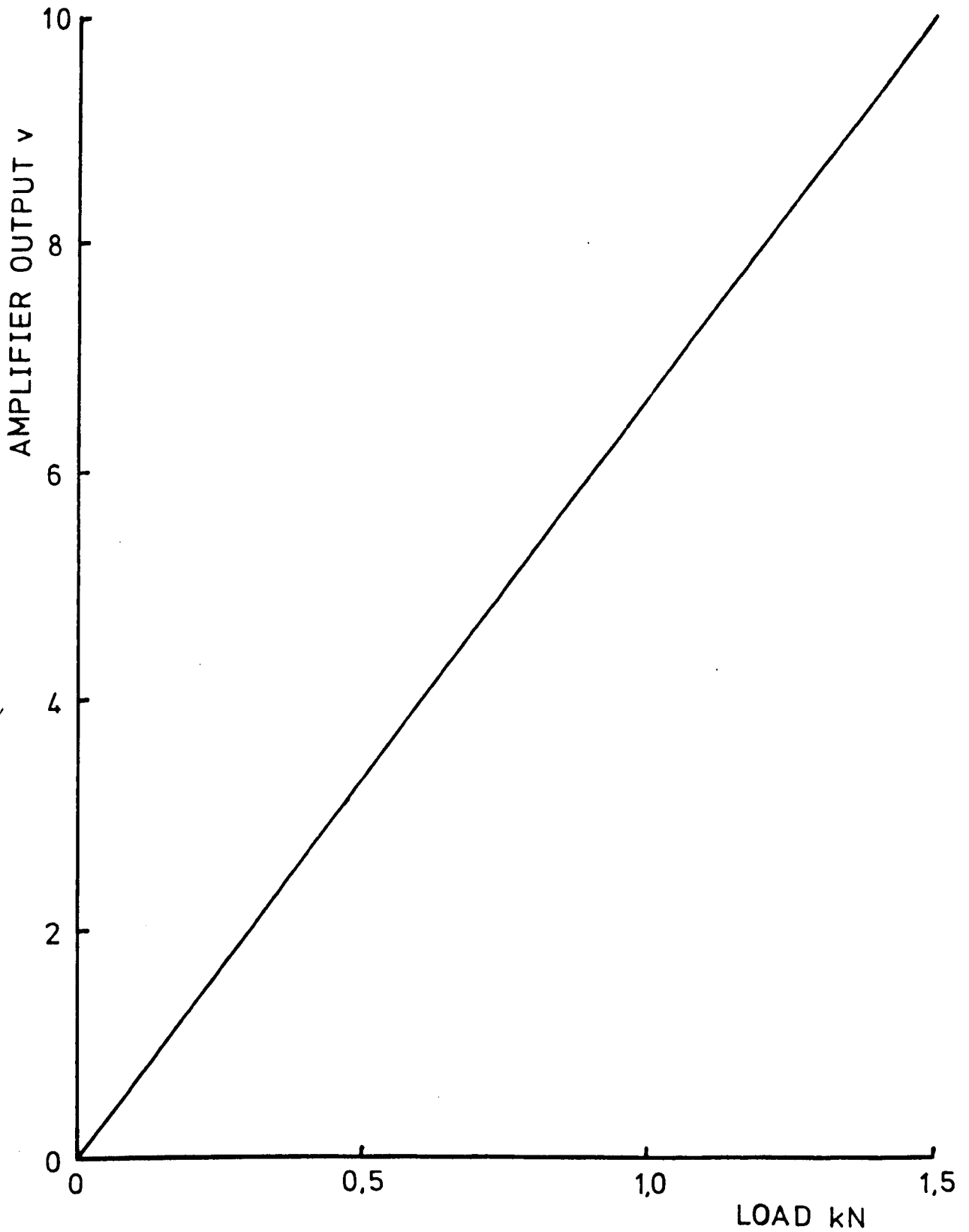
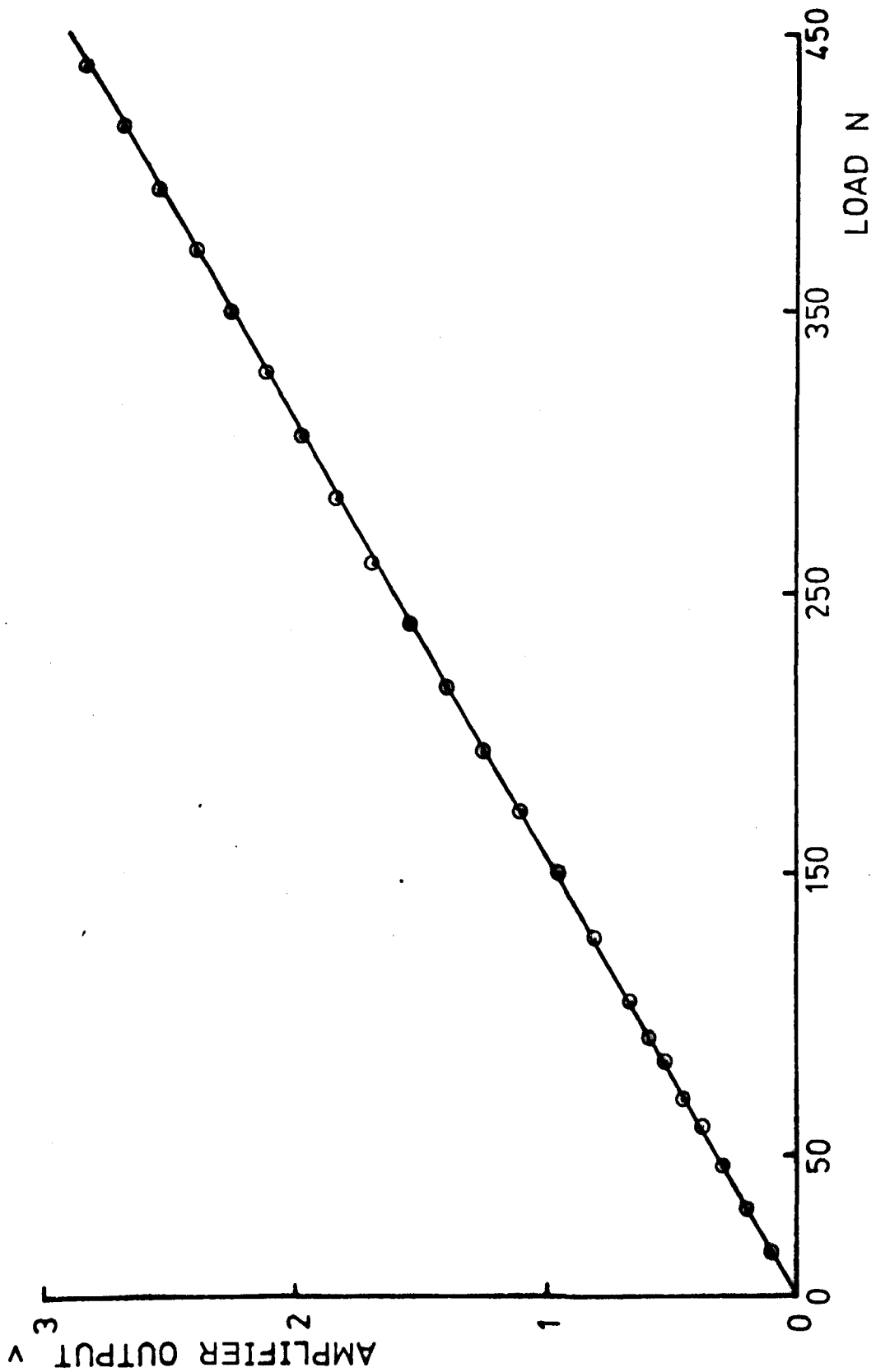


FIGURE 7.3



STRAIN GAUGED CANTILEVER CALIBRATION
[Tensile Test Machine and X-Y Plotter]

FIGURE 7,4



'IN SITU' STRAIN GAUGED CANTILEVER CALIBRATION
[Loaded by Spring Balance]

0
1
2
3
4
5
6
7
8
9
10
11
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THERMOCOUPLE CALIBRATION

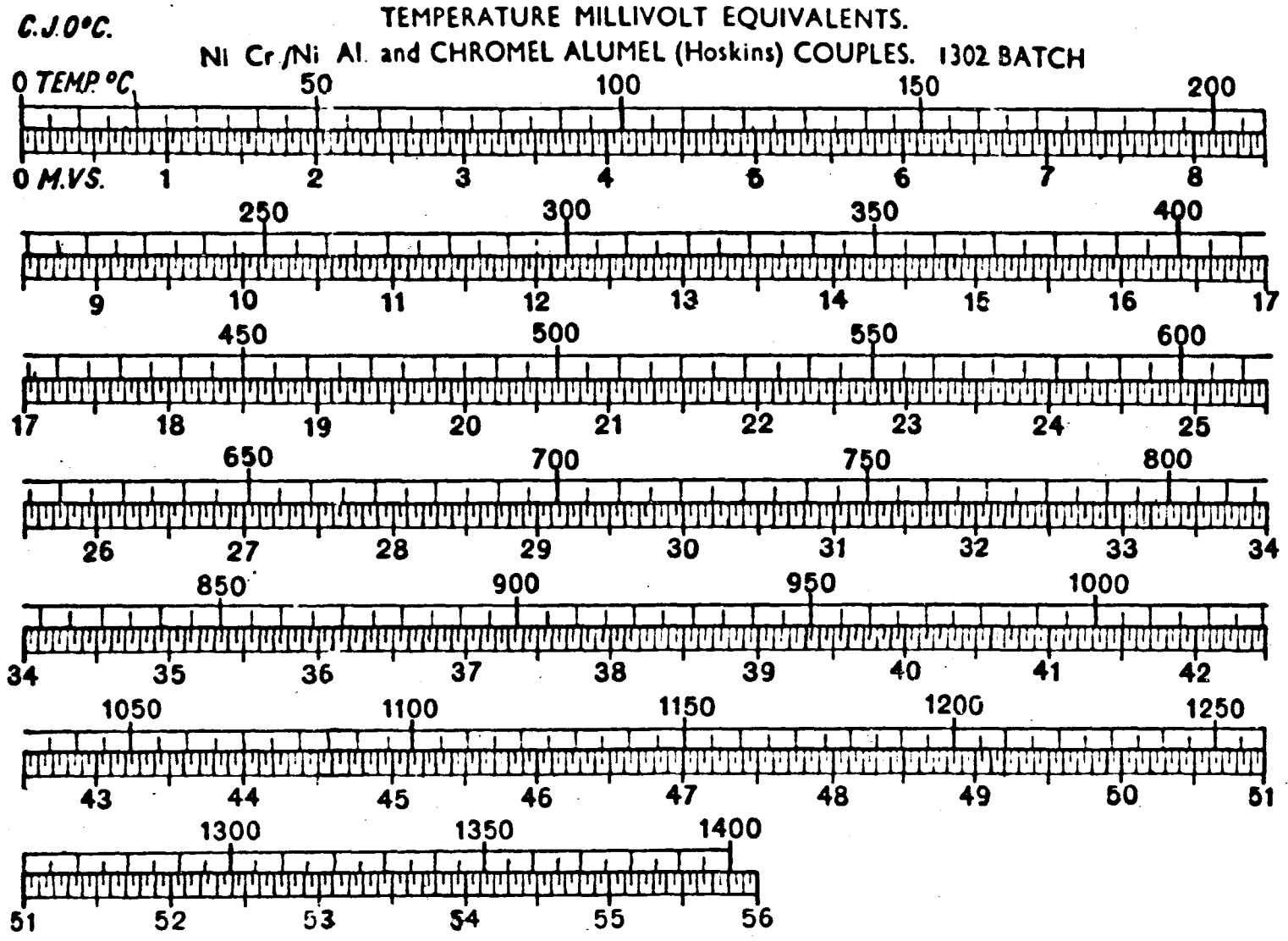
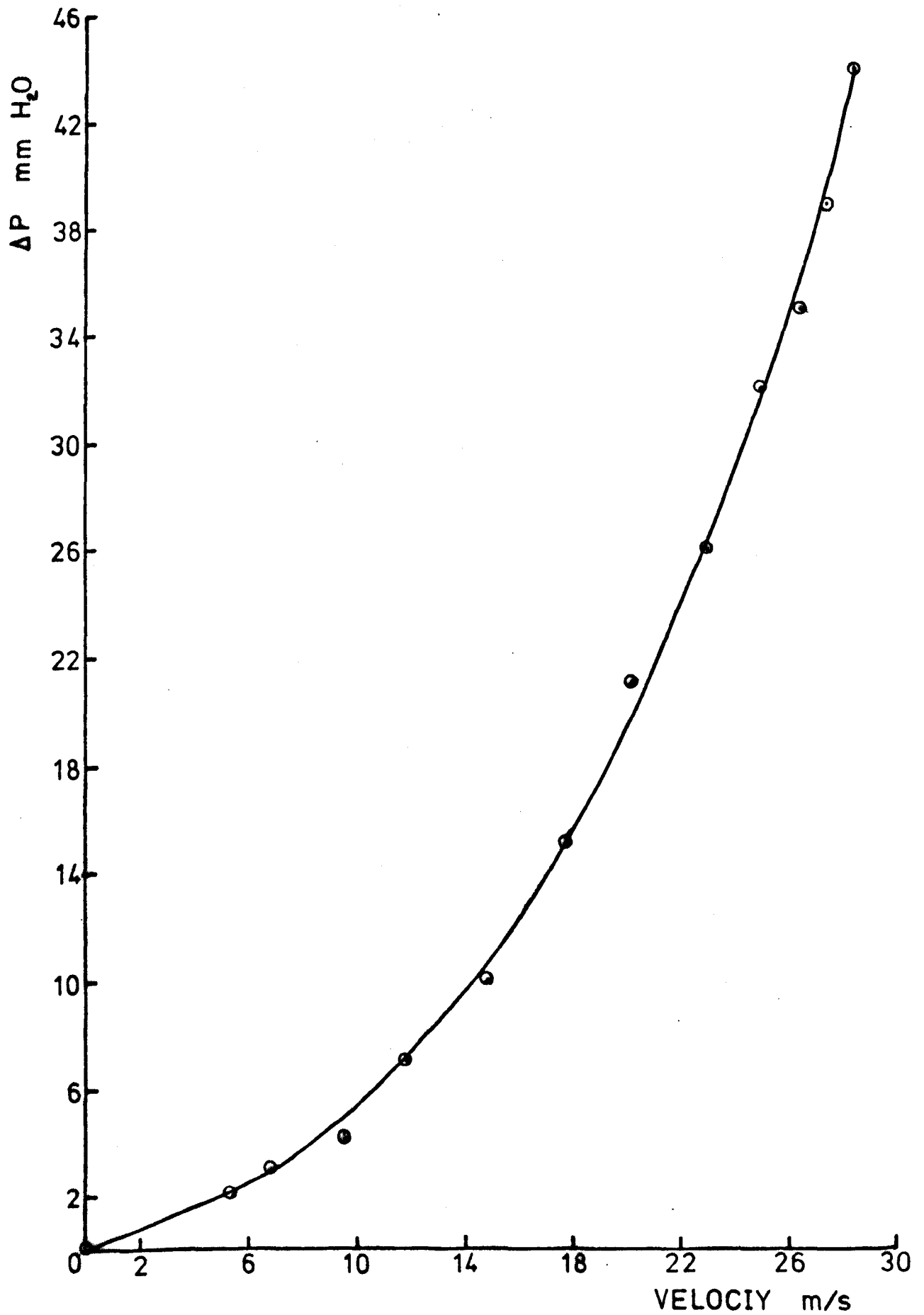


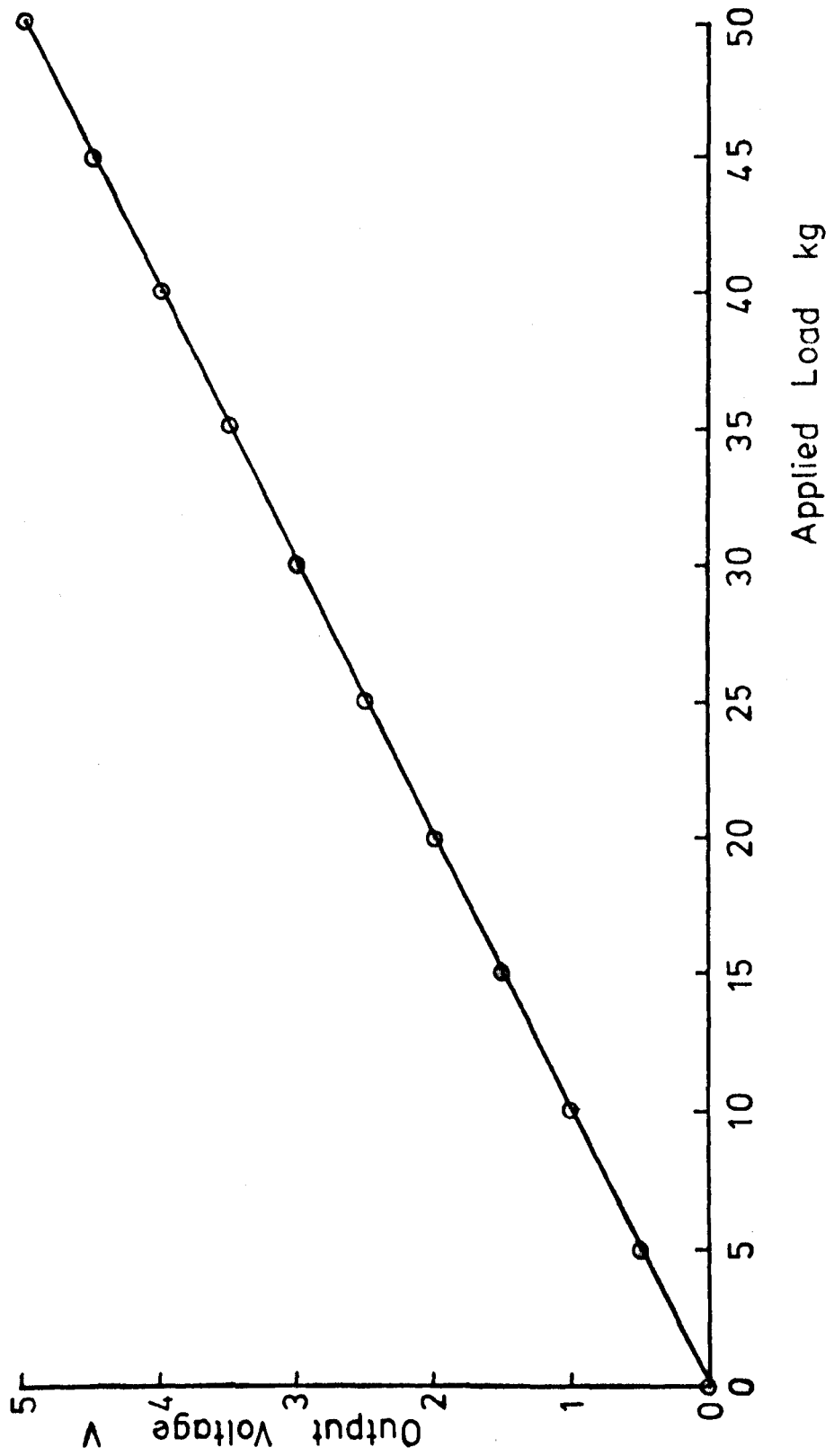
FIGURE 75

FIGURE 7.6



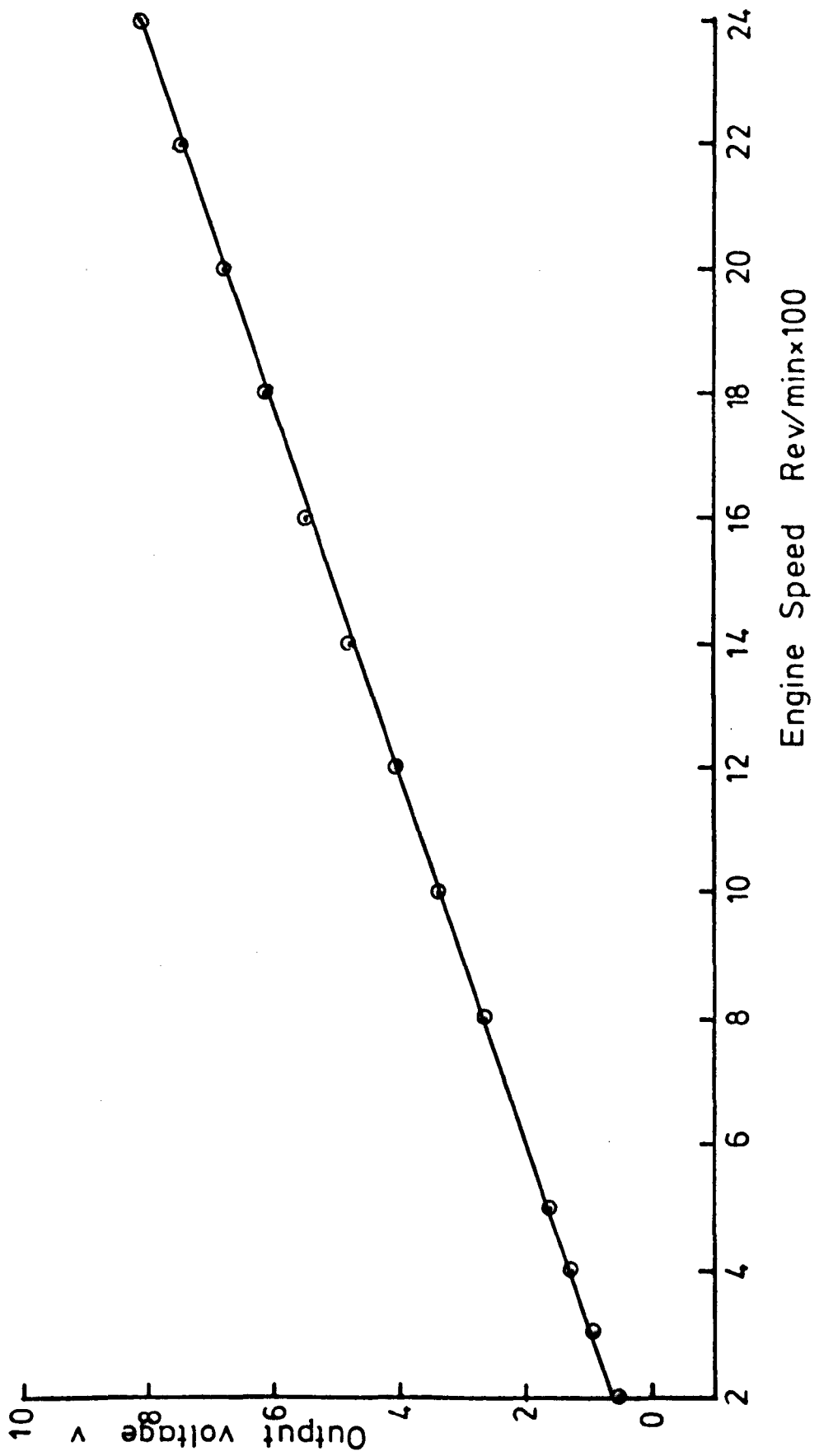
PRESTON FLOW METER CALIBRATION

FIGURE 7.7



CALIBRATION OF PROVING RING

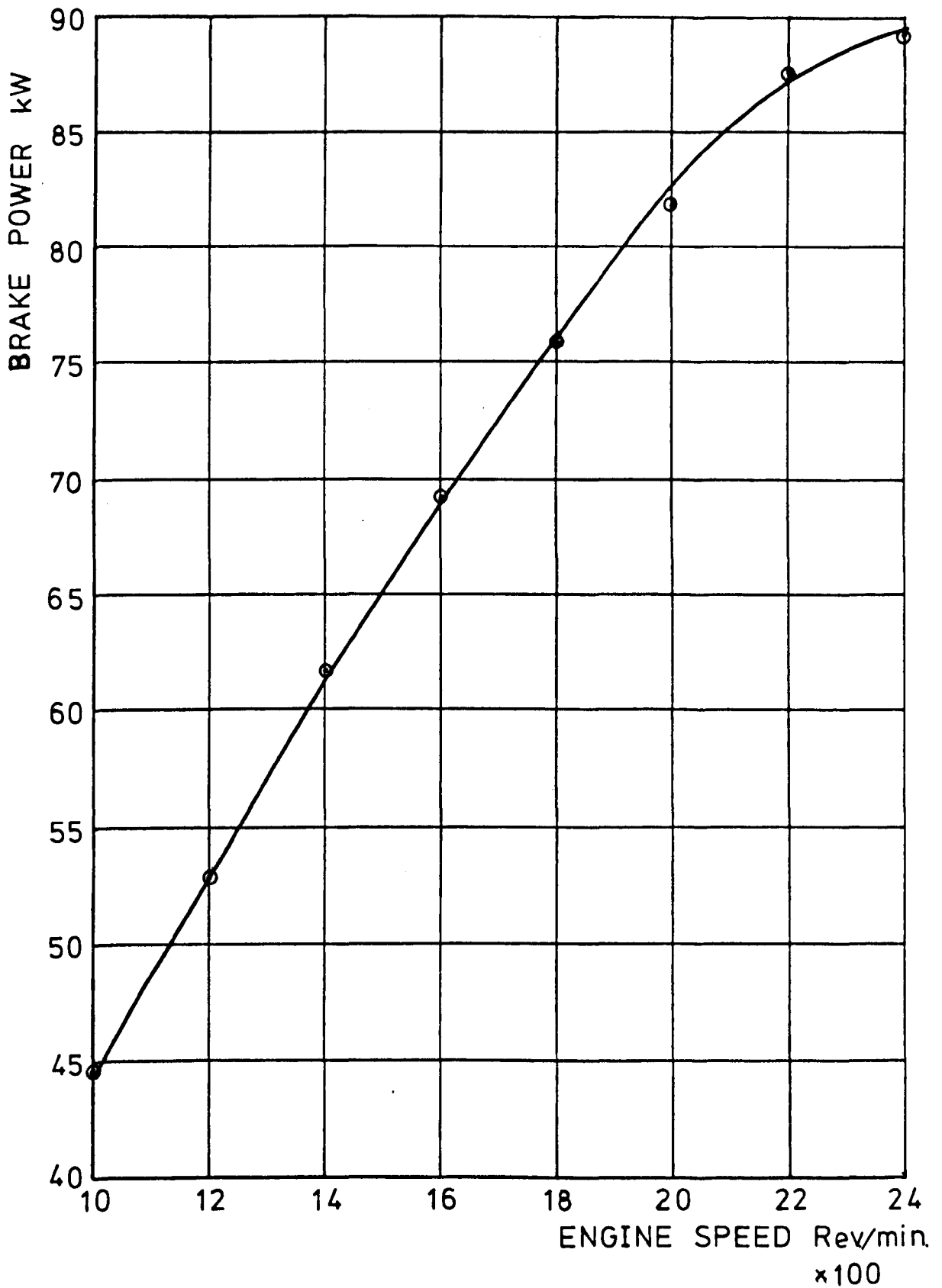
FIGURE 7,8



ELECTRONIC ENGINE SPEED

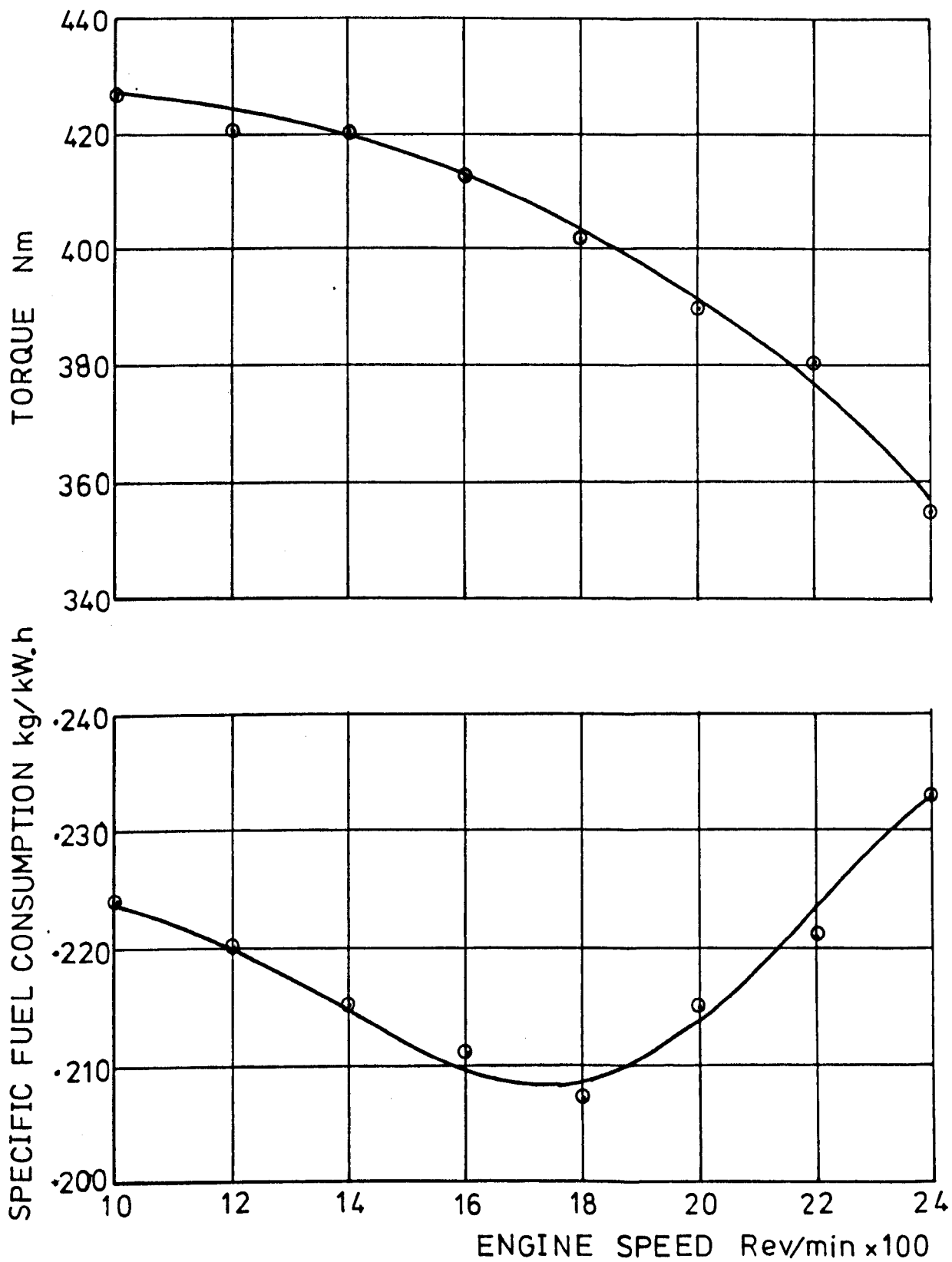
MEASUREMENT CALIBRATION

FIGURE 8.1



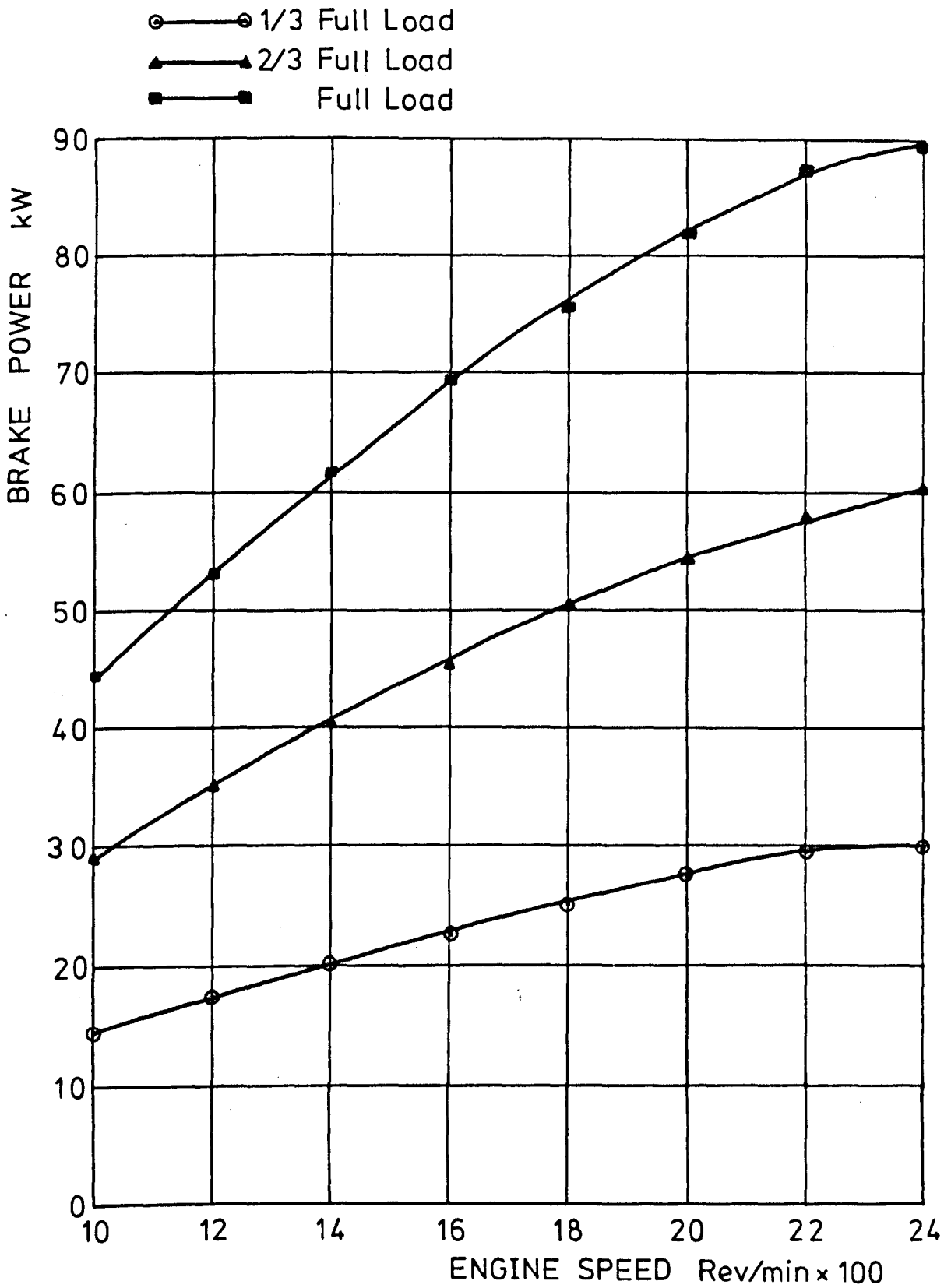
BASIC ENGINE: FULL LOAD PERFORMANCE

FIGURE 8.2



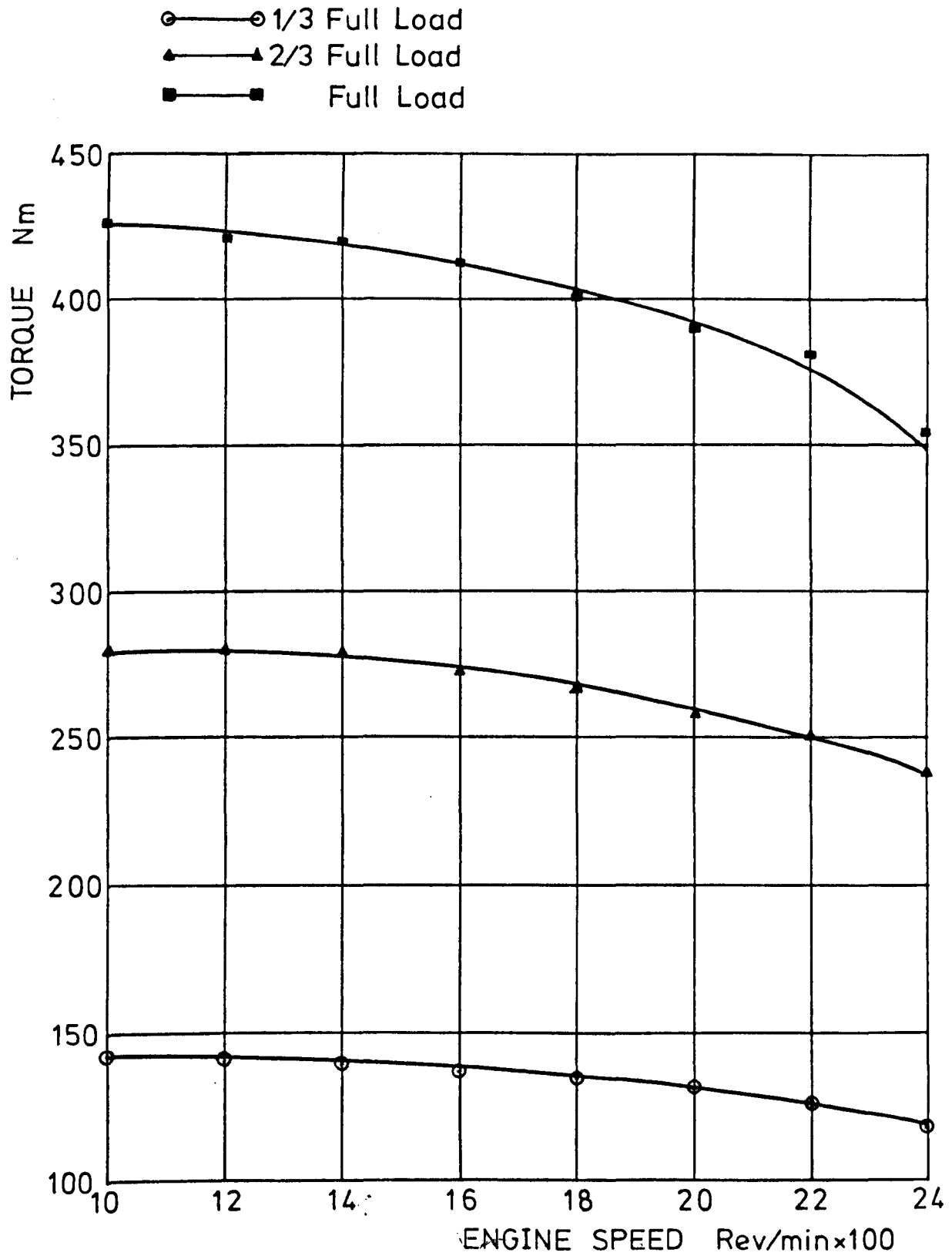
BASIC ENGINE: FULL LOAD PERFORMANCE

FIGURE 8.3



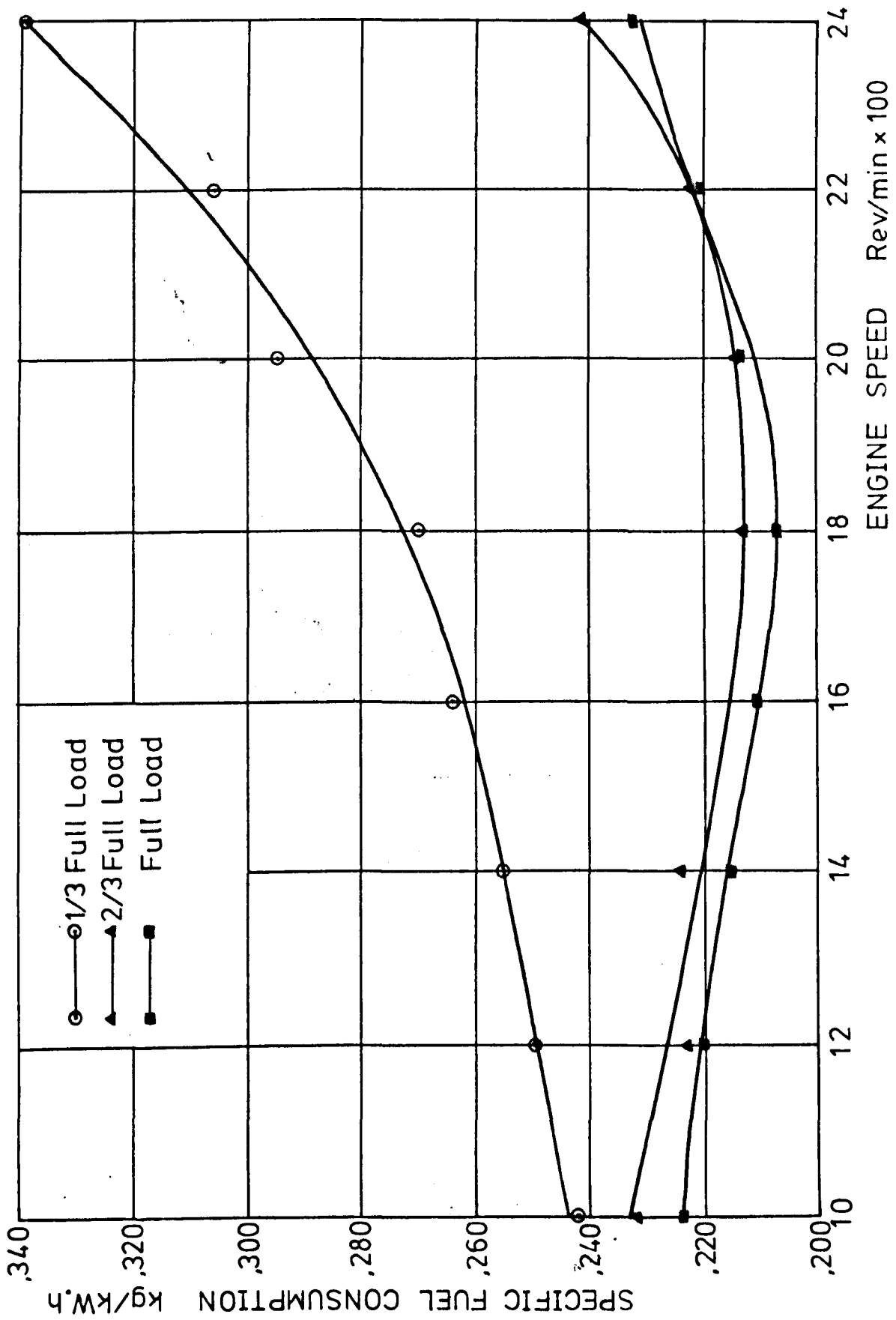
BASIC ENGINE PERFORMANCE

FIGURE 8.4



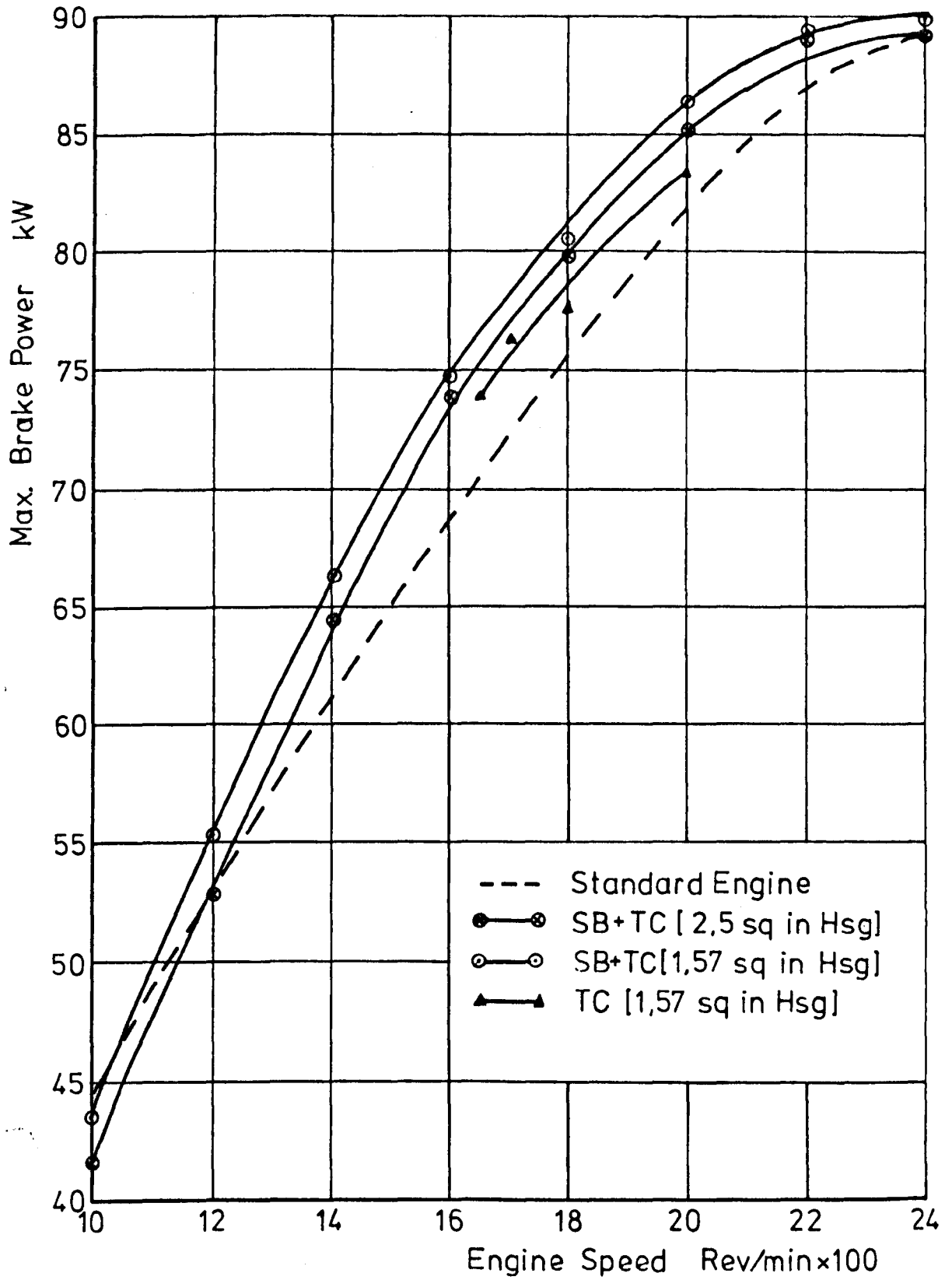
BASIC ENGINE PERFORMANCE

FIGURE 8.5



BASIC ENGINE PERFORMANCE

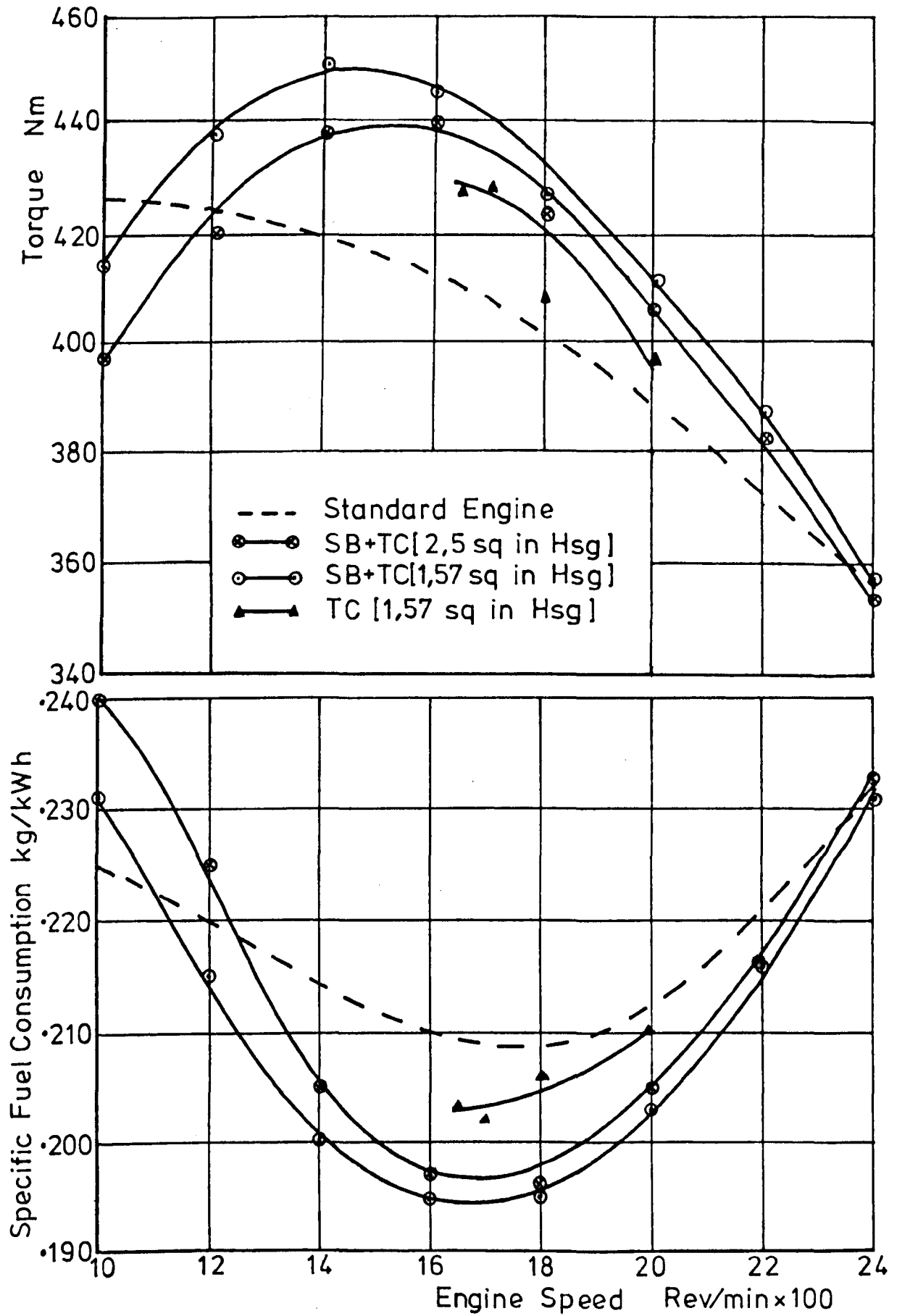
FIGURE 8.6



ENGINE SYSTEM PERFORMANCE

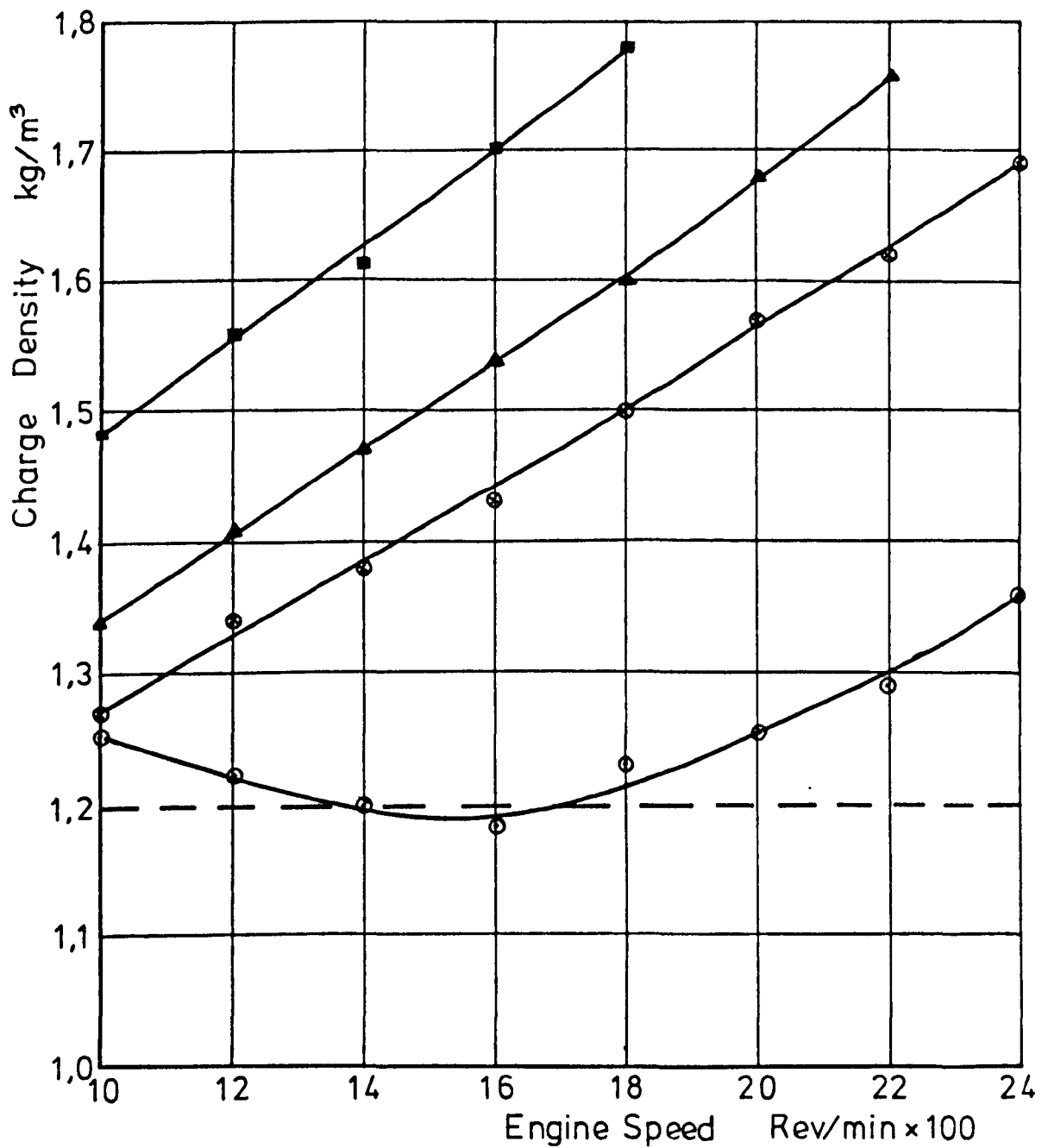
STAGE I DEVELOPMENT

FIGURE 8.7



ENGINE SYSTEM PERFORMANCE STAGE I DEVELOPMENT

FIGURE 8.8

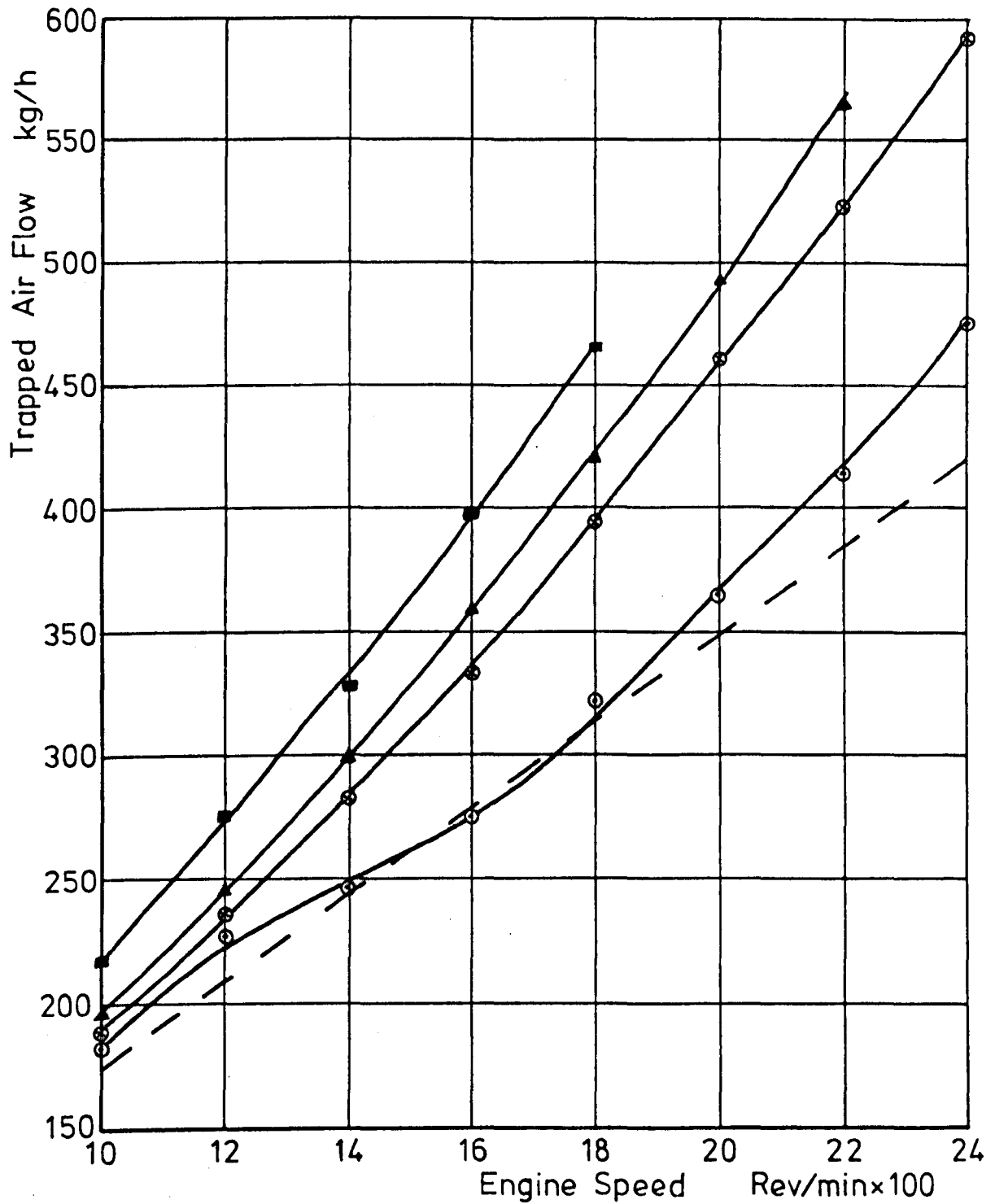


- Standard Engine
- SB+TC [2,5 sq in Hsg]
- ▲—▲ SB+TC [1,57 sq in Hsg]
- SB+TC+ACC [1,57 sq in Hsg]
- - - Naturally Aspirated Conditions

CHARGE AIR DENSITY

DEVELOPMENTS I AND II

FIGURE 8.9

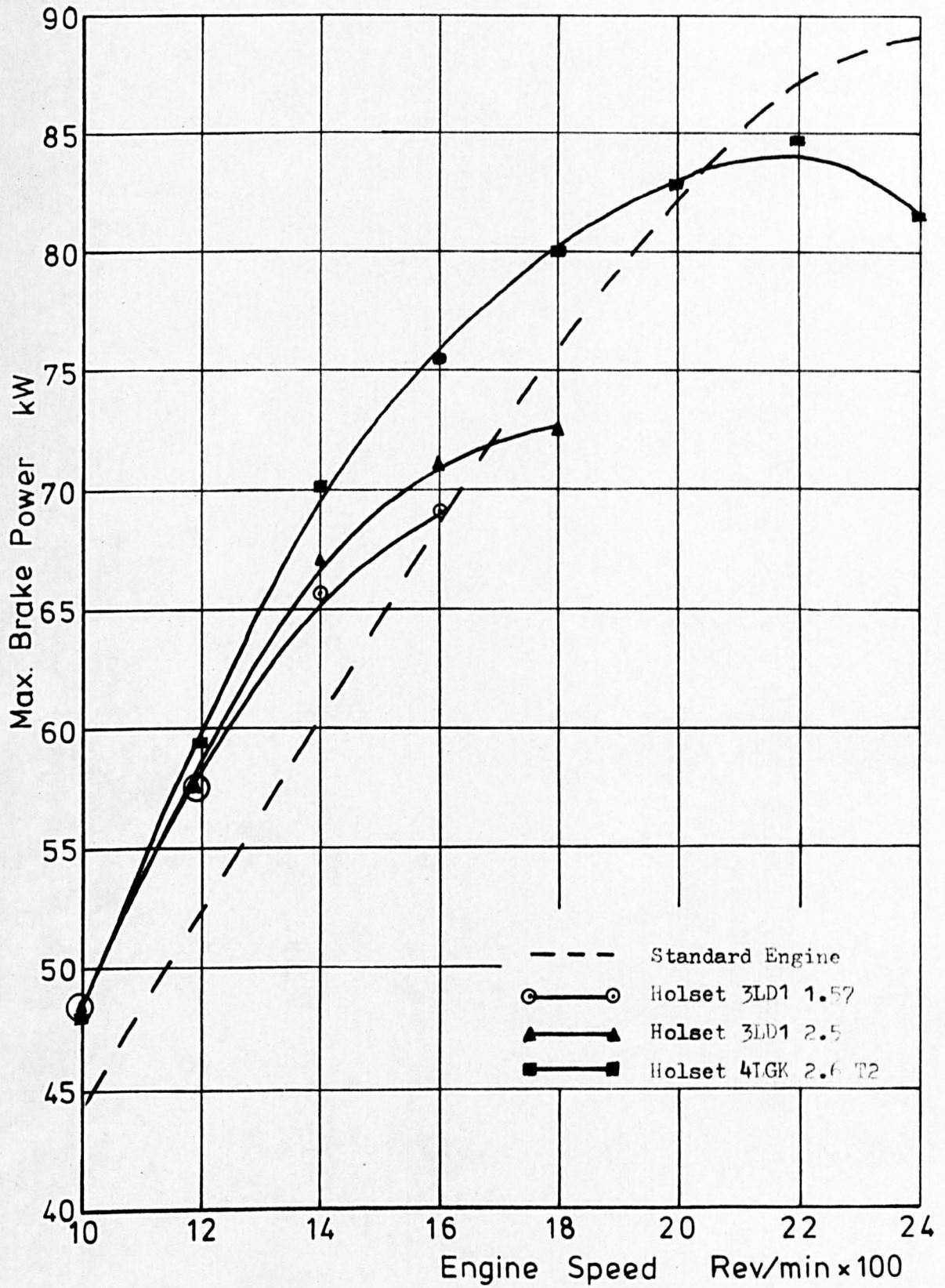


- Standard Engine
- SB+TC [2,5 sq in Hsg]
- ▲—▲ SB+TC [1,57 sq in Hsg]
- SB+TC+ACC [1,57 sq in Hsg]
- - - Naturally Aspirated Conditions

TRAPPED AIR FLOW

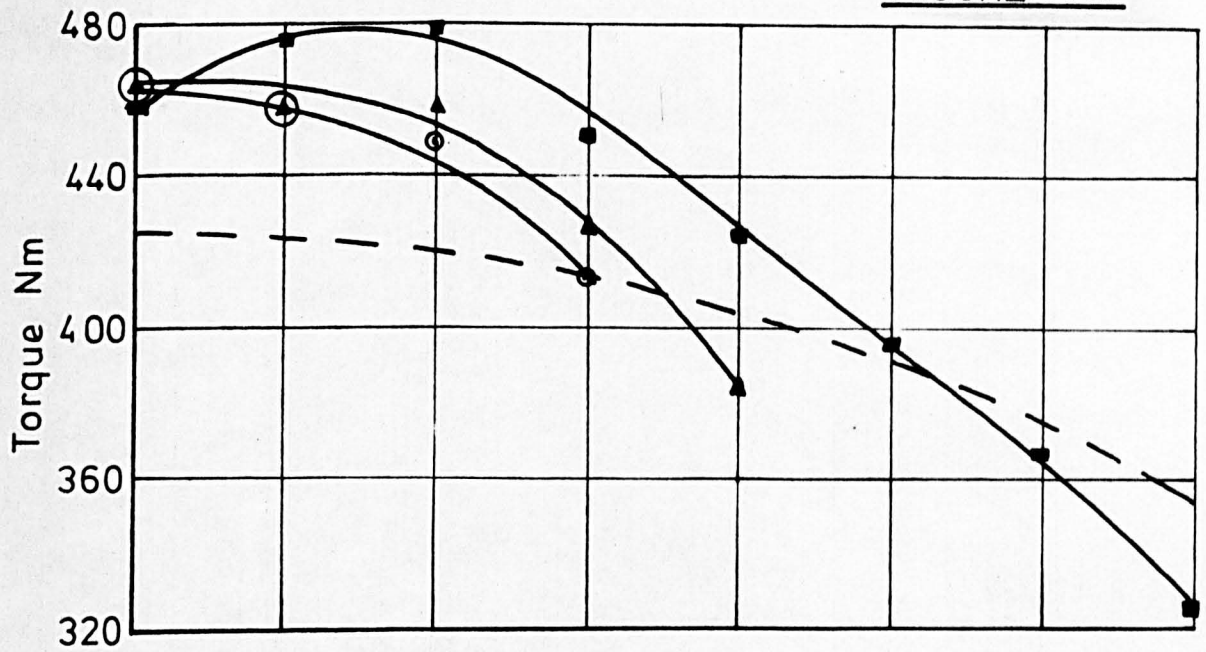
DEVELOPMENTS I AND II

FIGURE 8.10

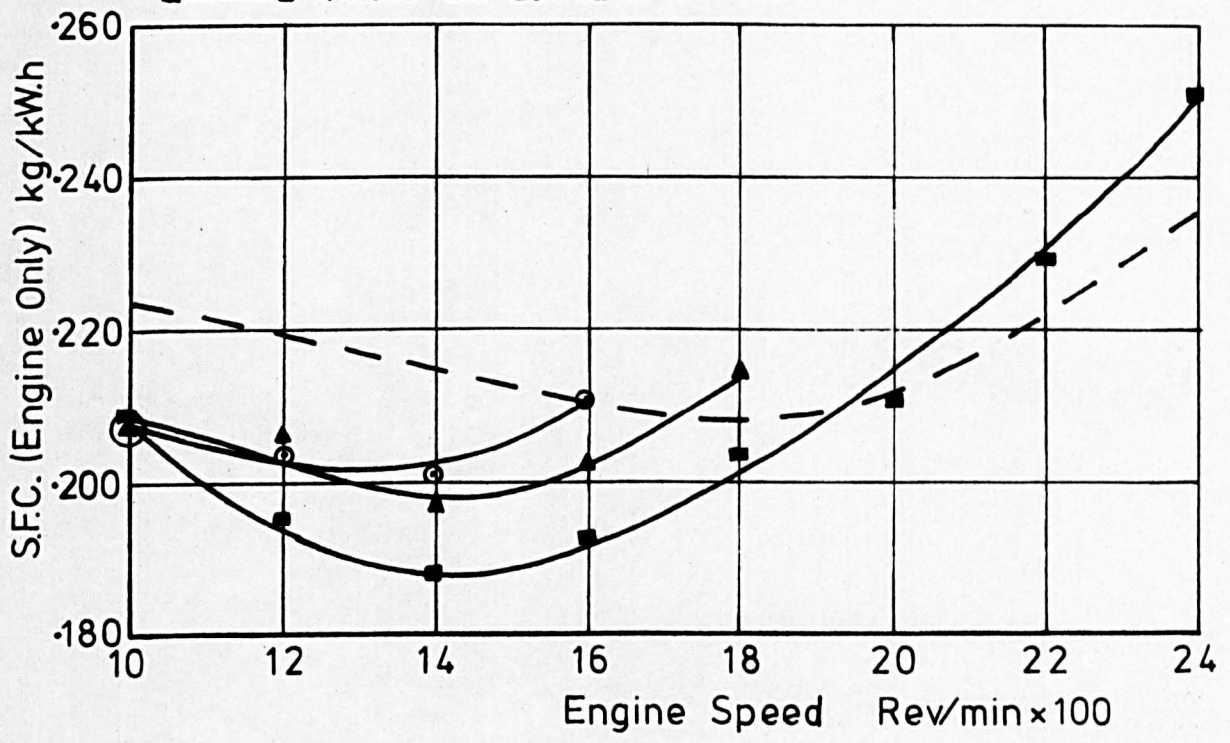


STAGE III [DGT] PERFORMANCE

FIGURE 8.11

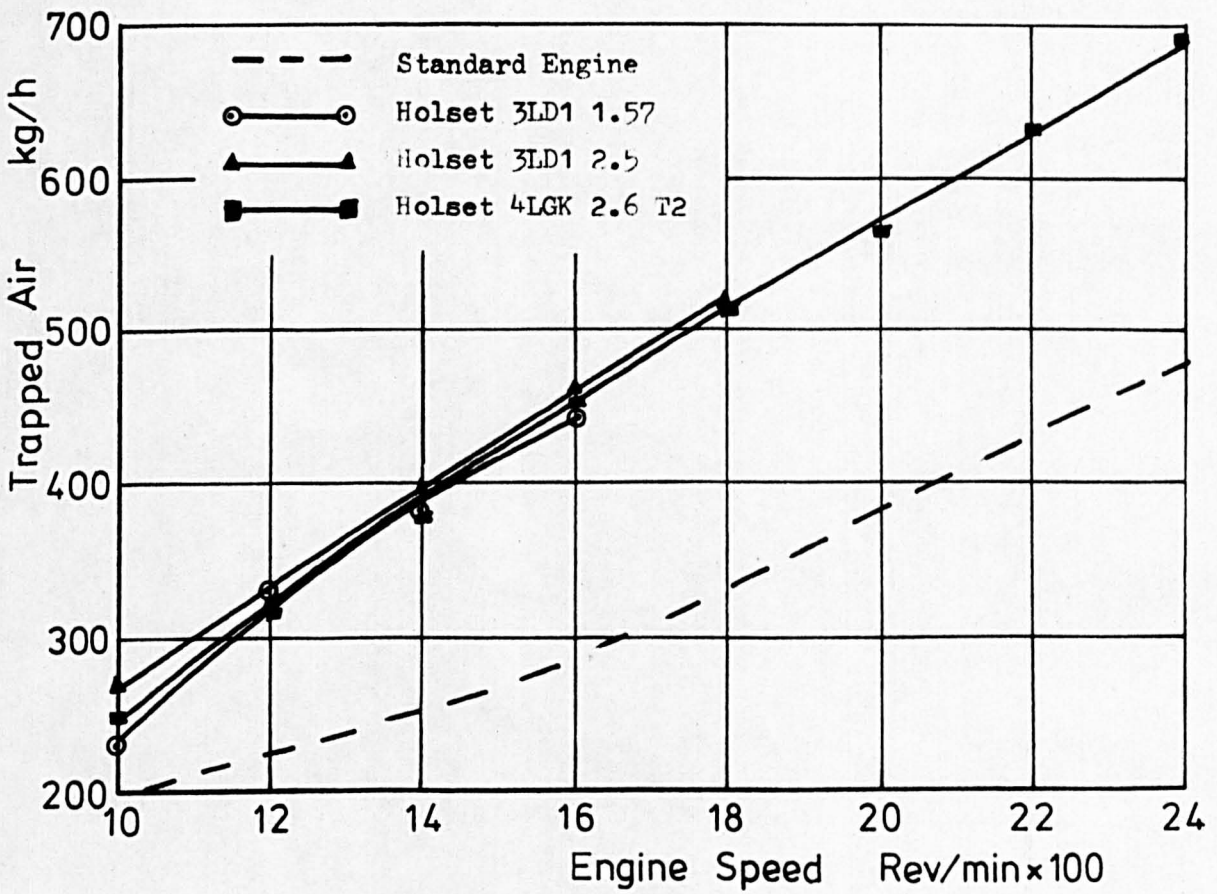
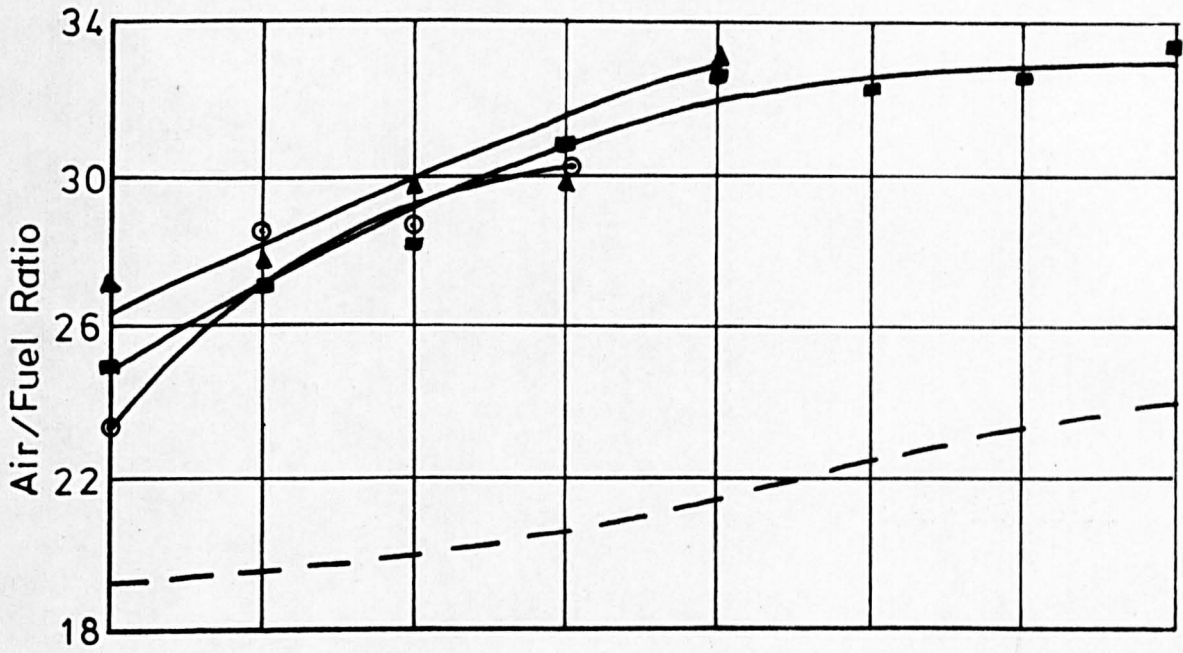


- - - Standard Engine
- Holset 3LD1 1.57
- ▲—▲ Holset 3LD1 2.5
- Holset 4LGK 2.6 T2



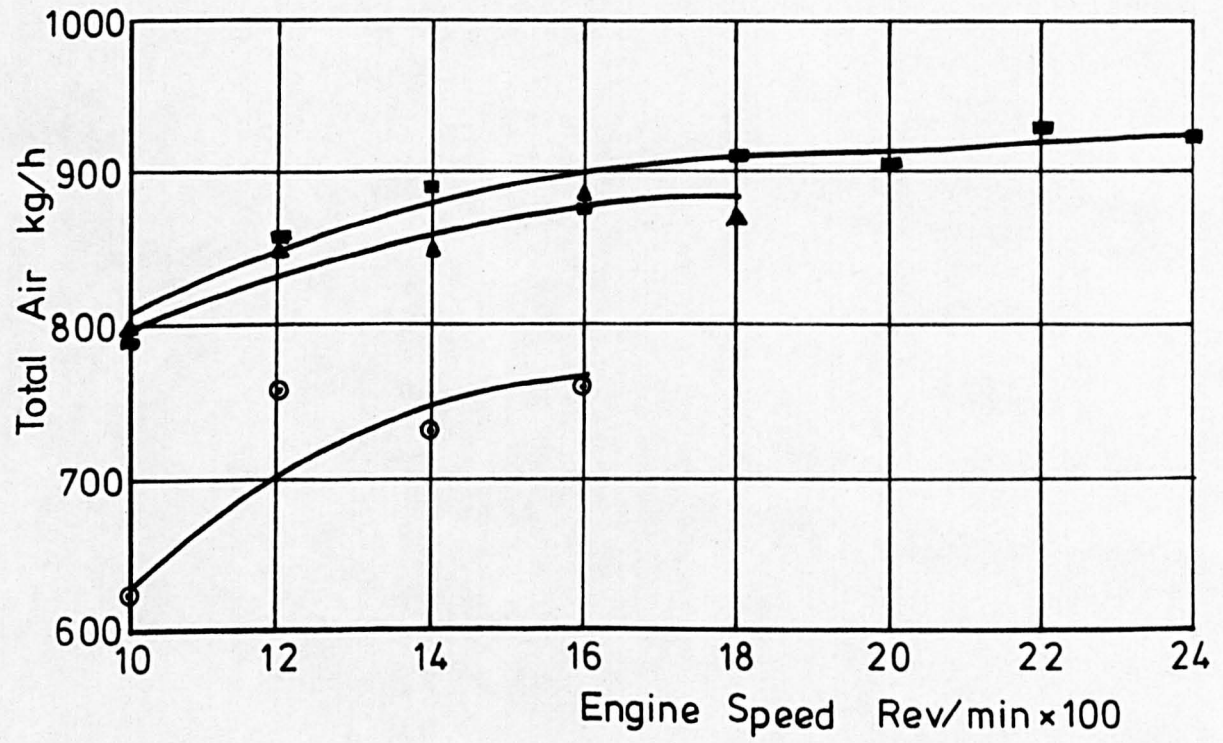
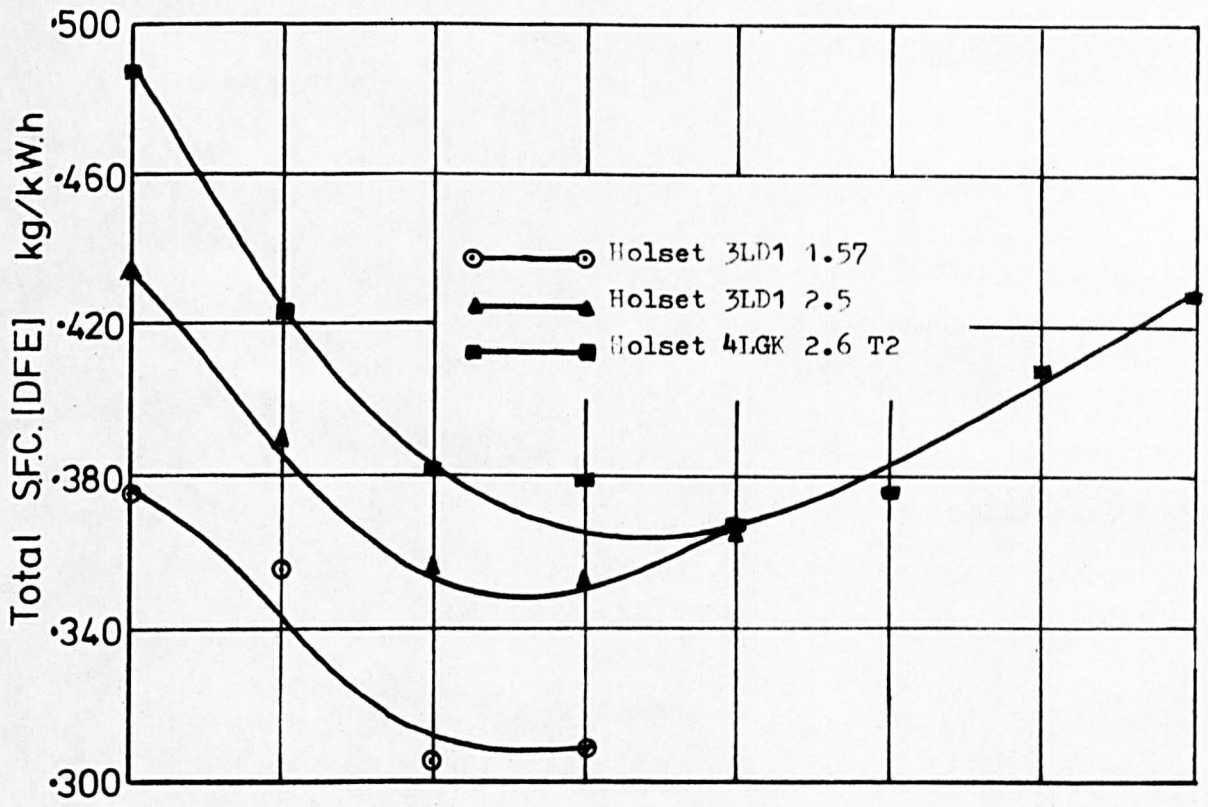
STAGE III [DGT] PERFORMANCE

FIGURE 8,12



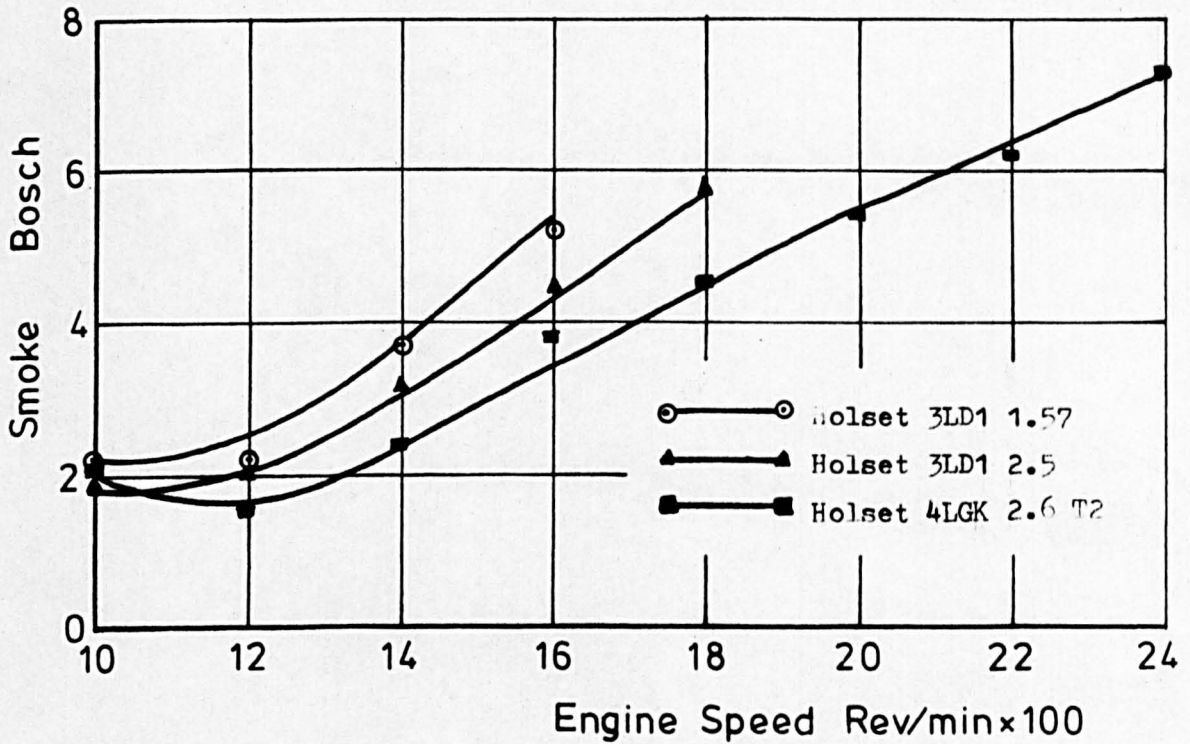
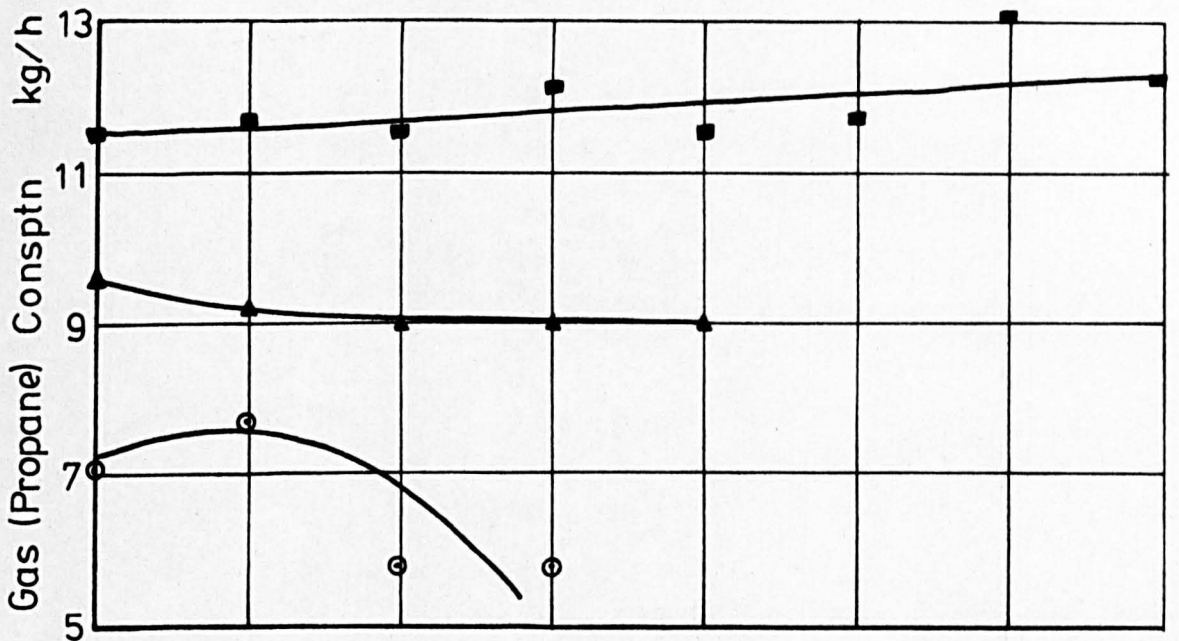
STAGE III [DGT] PERFORMANCE

FIGURE 8,13



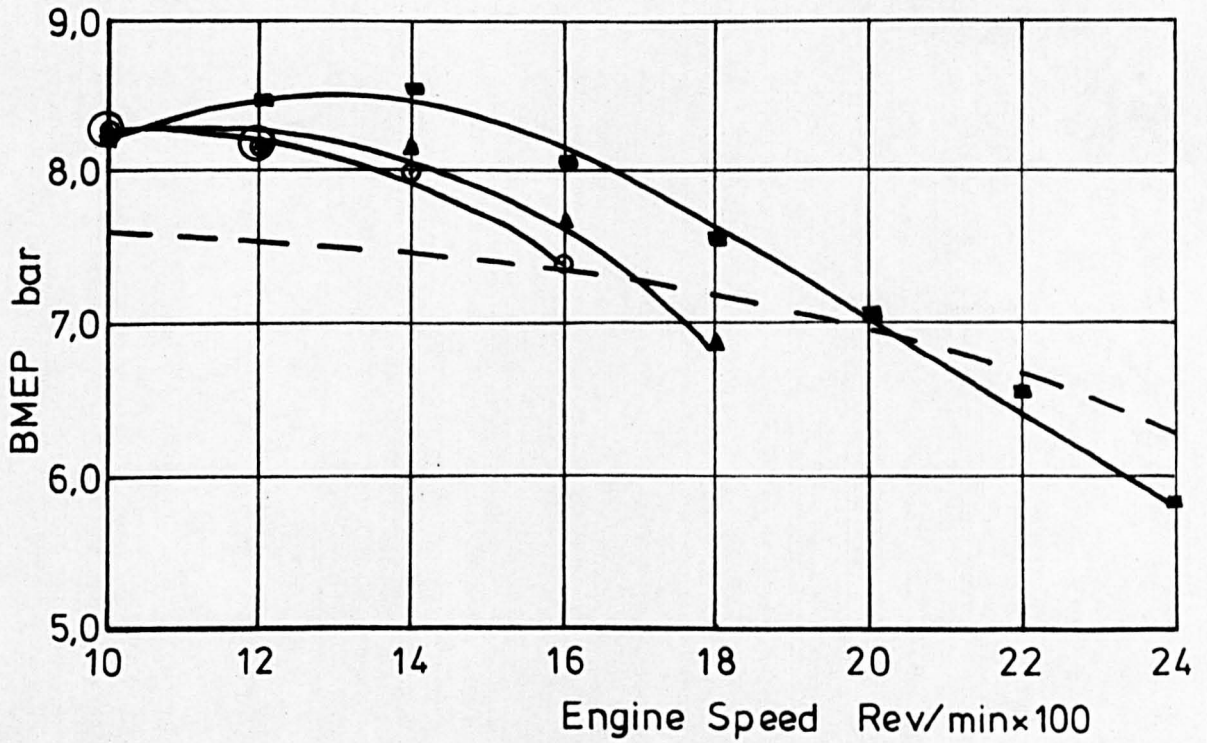
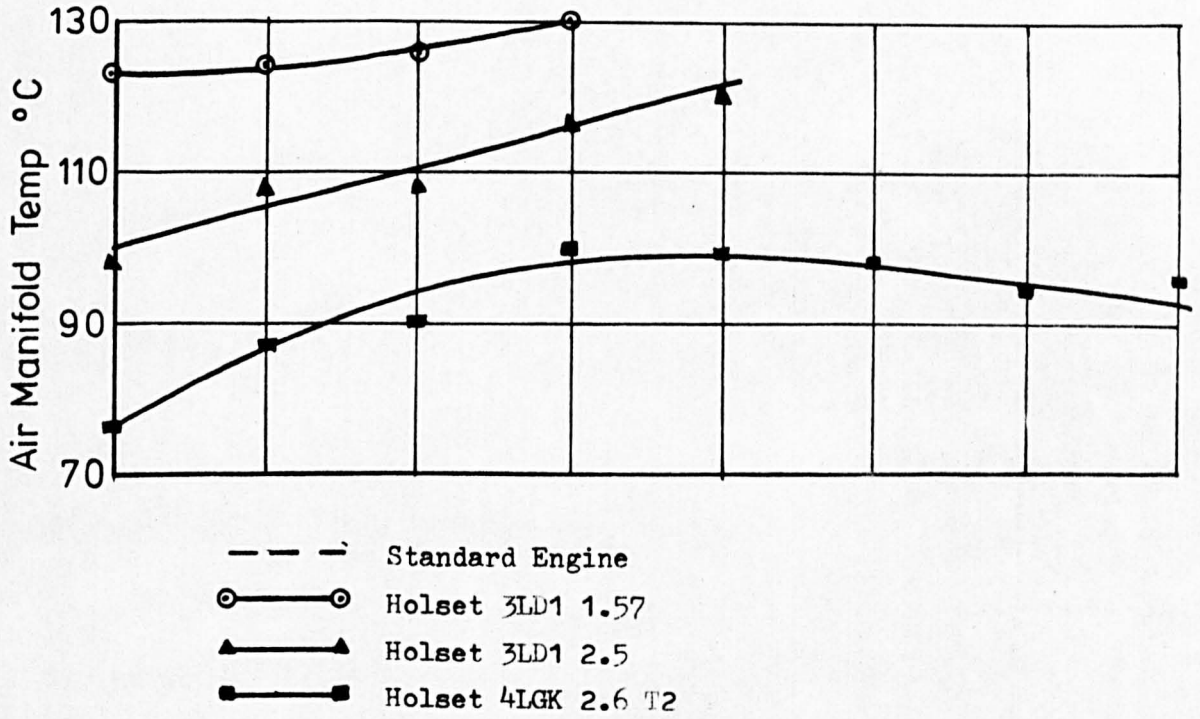
STAGE III [DGT] PERFORMANCE

FIGURE 8,14



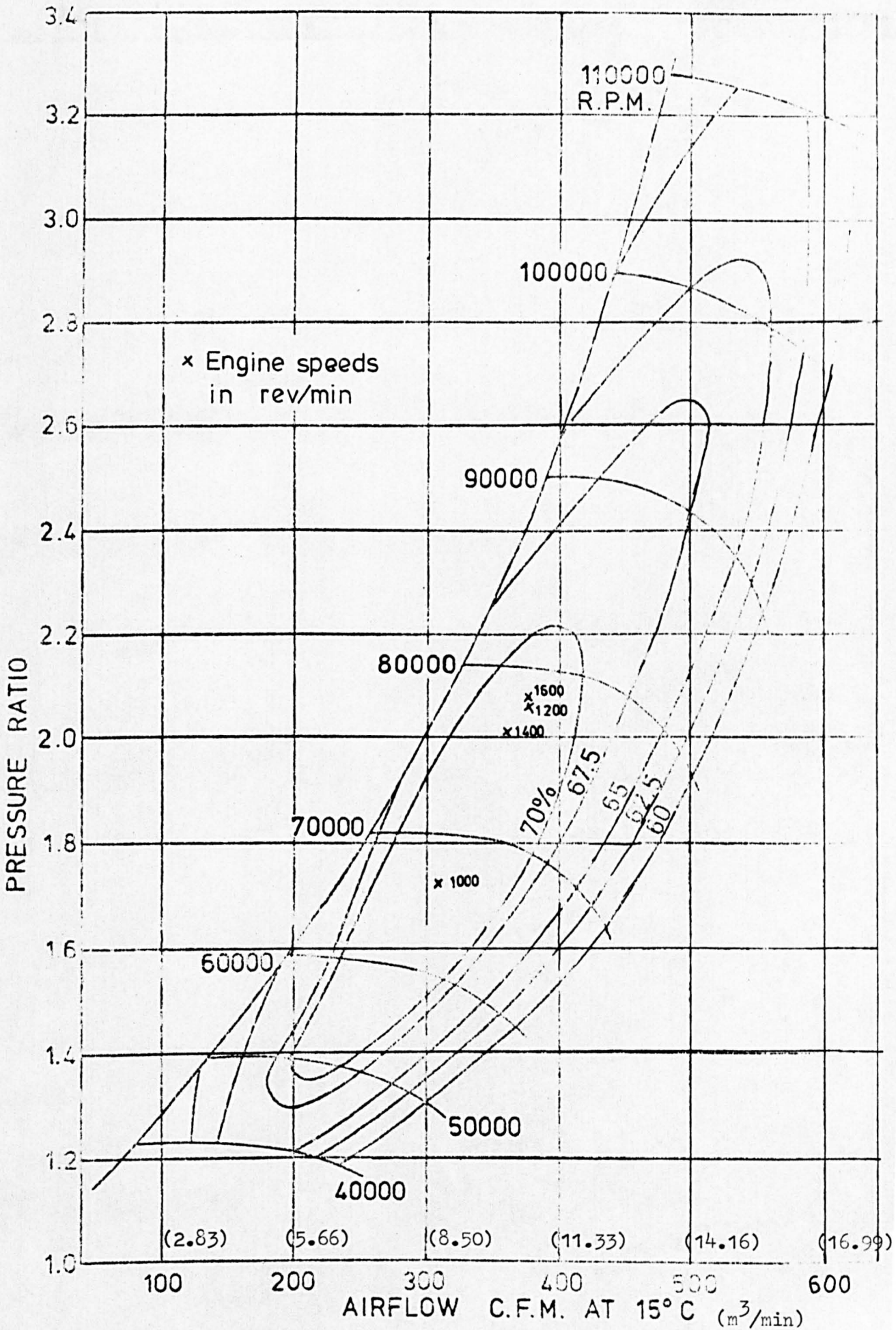
STAGE III [DGT] PERFORMANCE

FIGURE 8.15



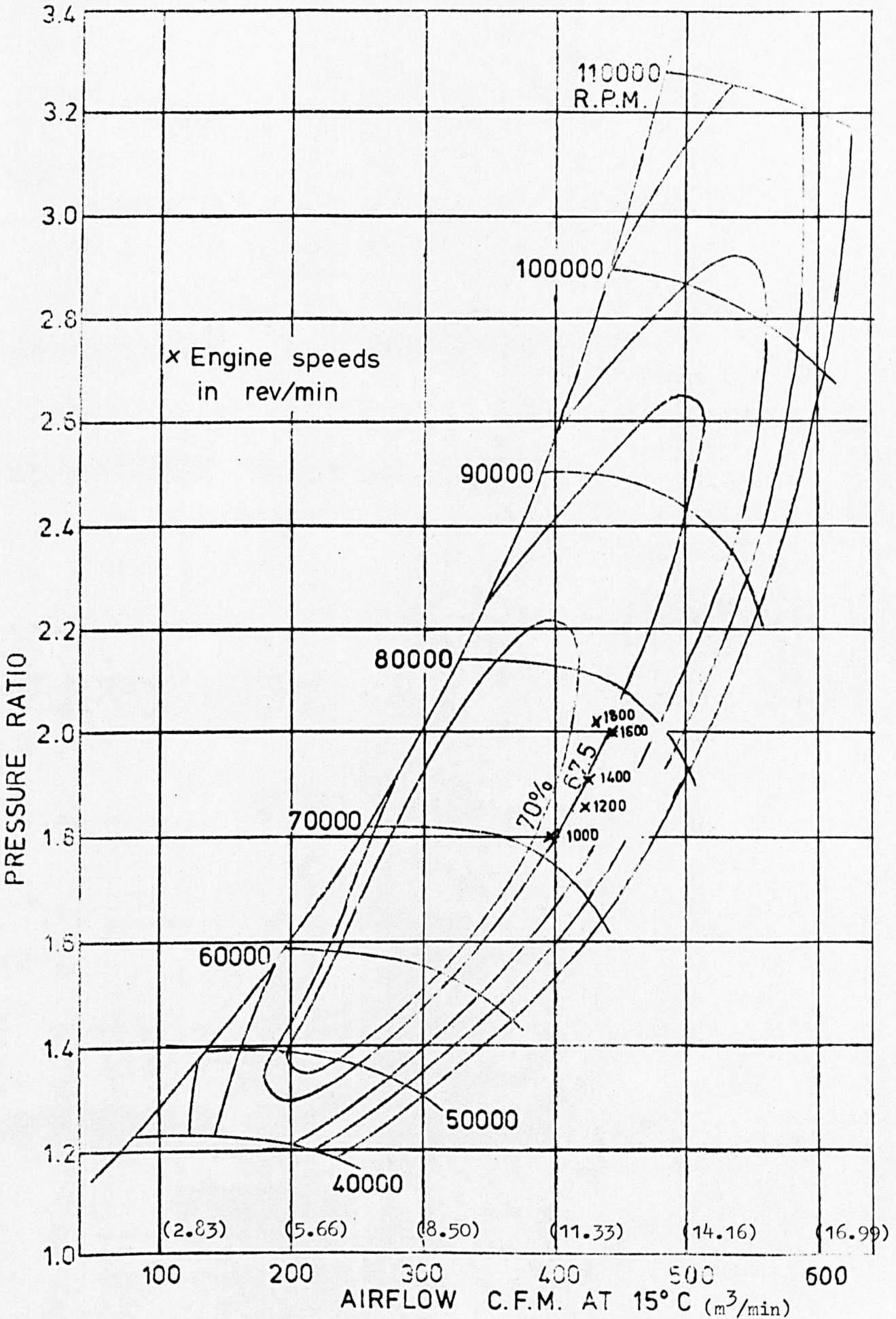
STAGE III [DGT] PERFORMANCE

FIGURE 8.16



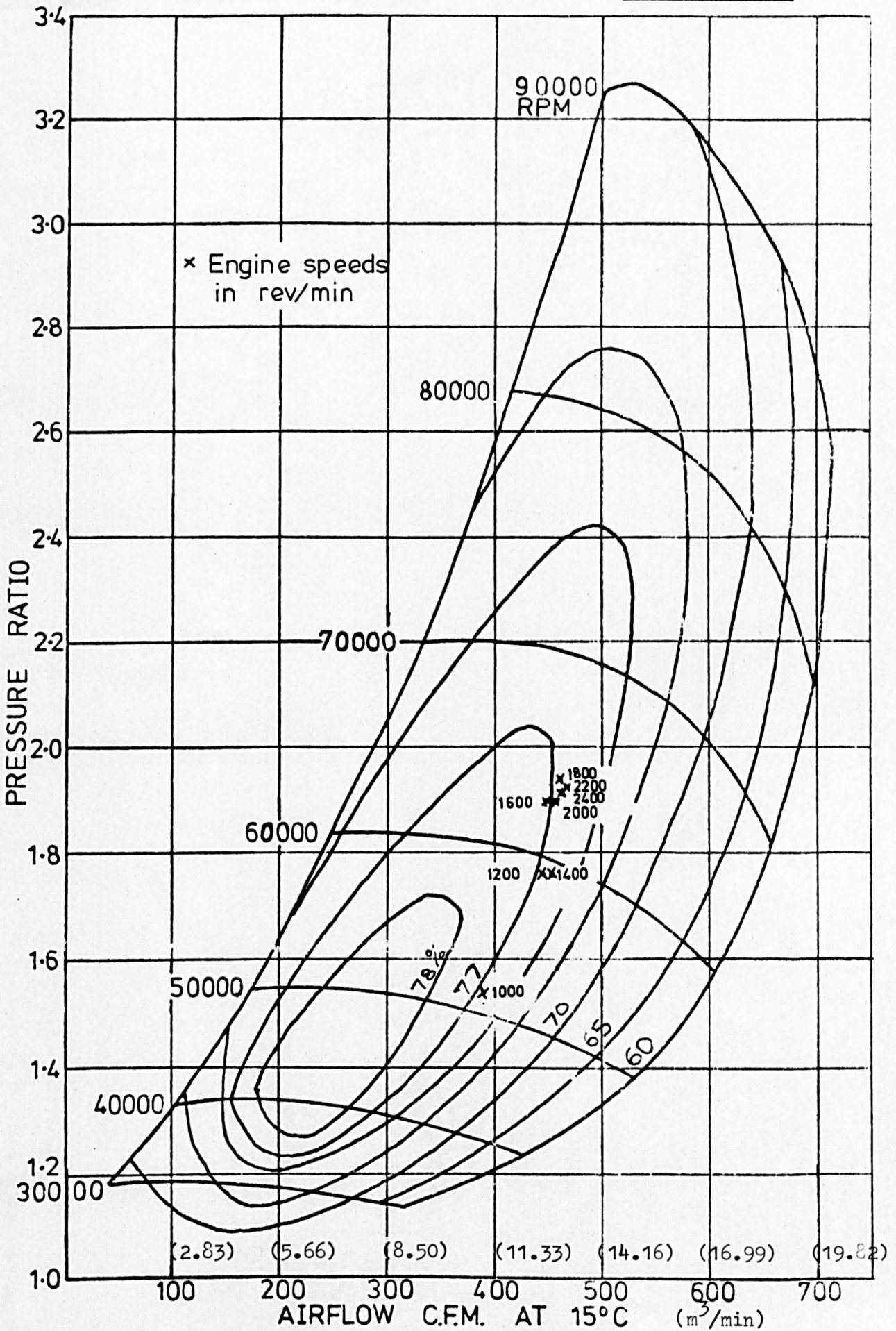
STAGE III VARIABLE SPEED HOLSET 3LD1 1.57

FIGURE 8,17



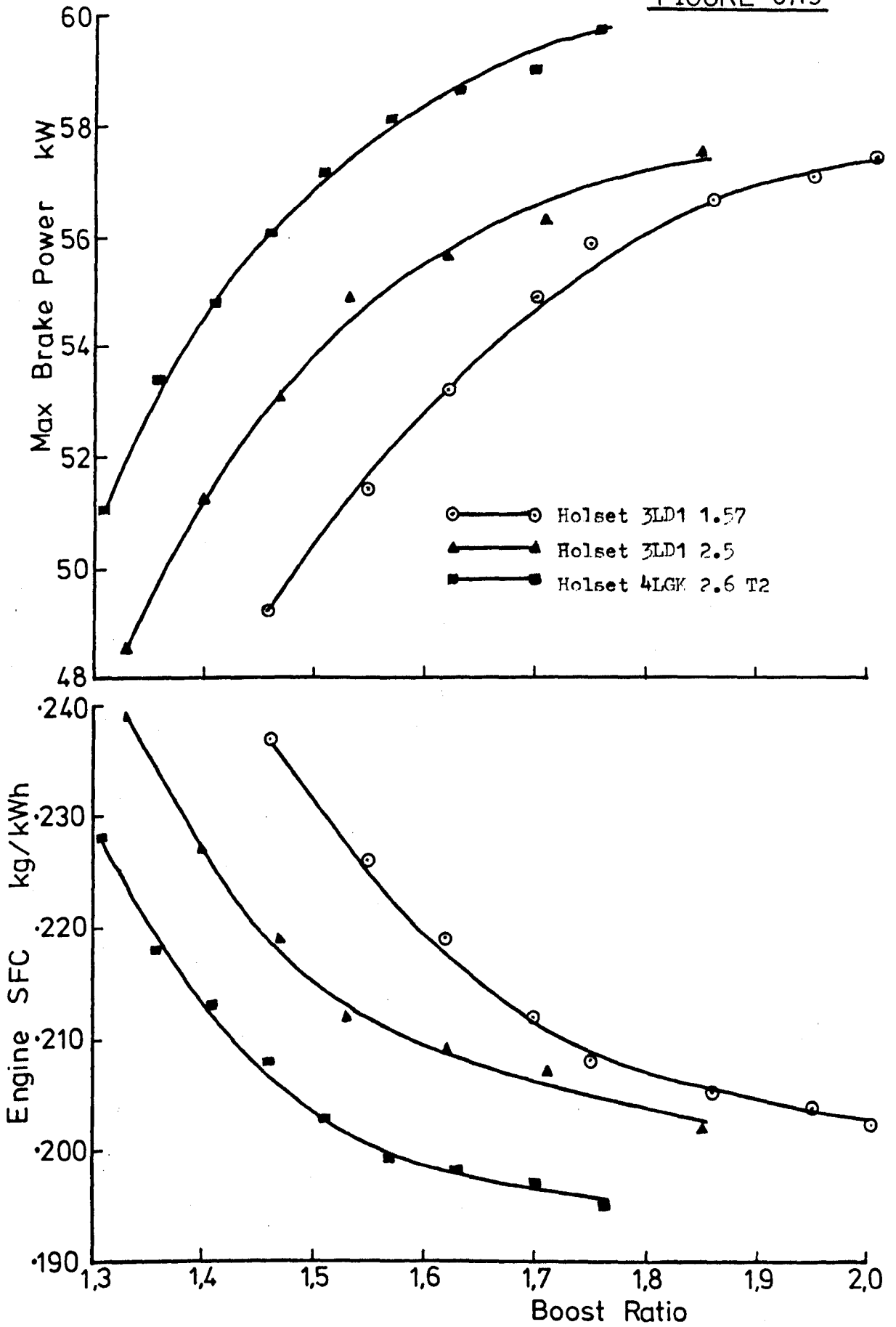
STAGE III VARIABLE SPEED HOLSET 3LD1 2.5

FIGURE 8,18



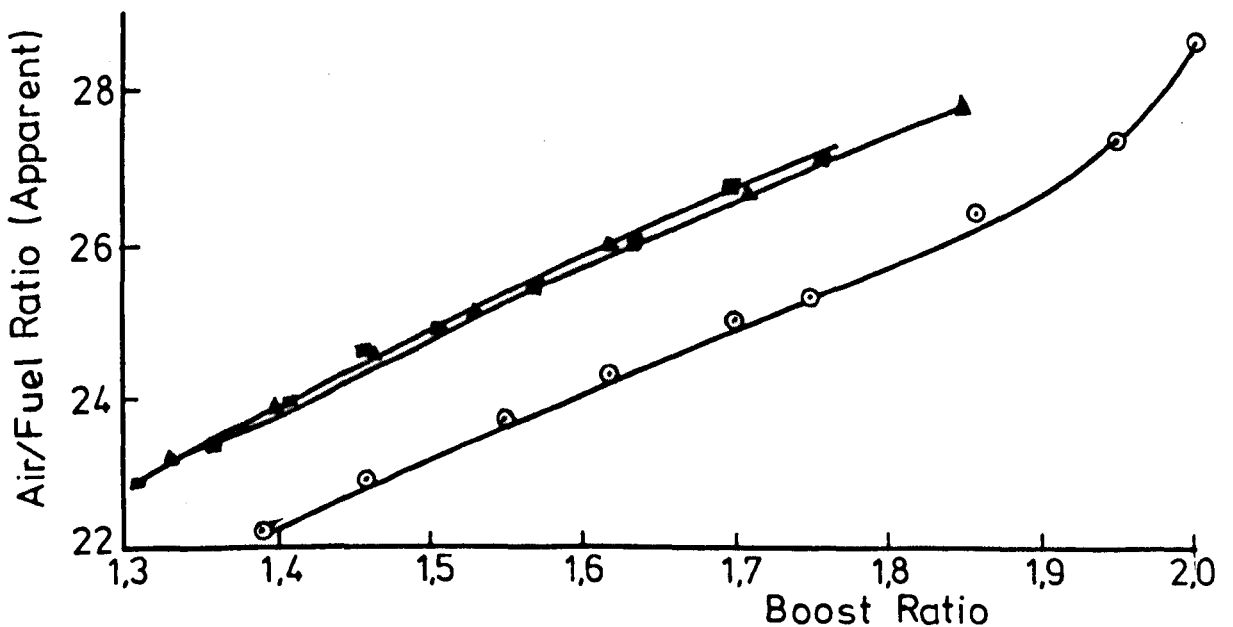
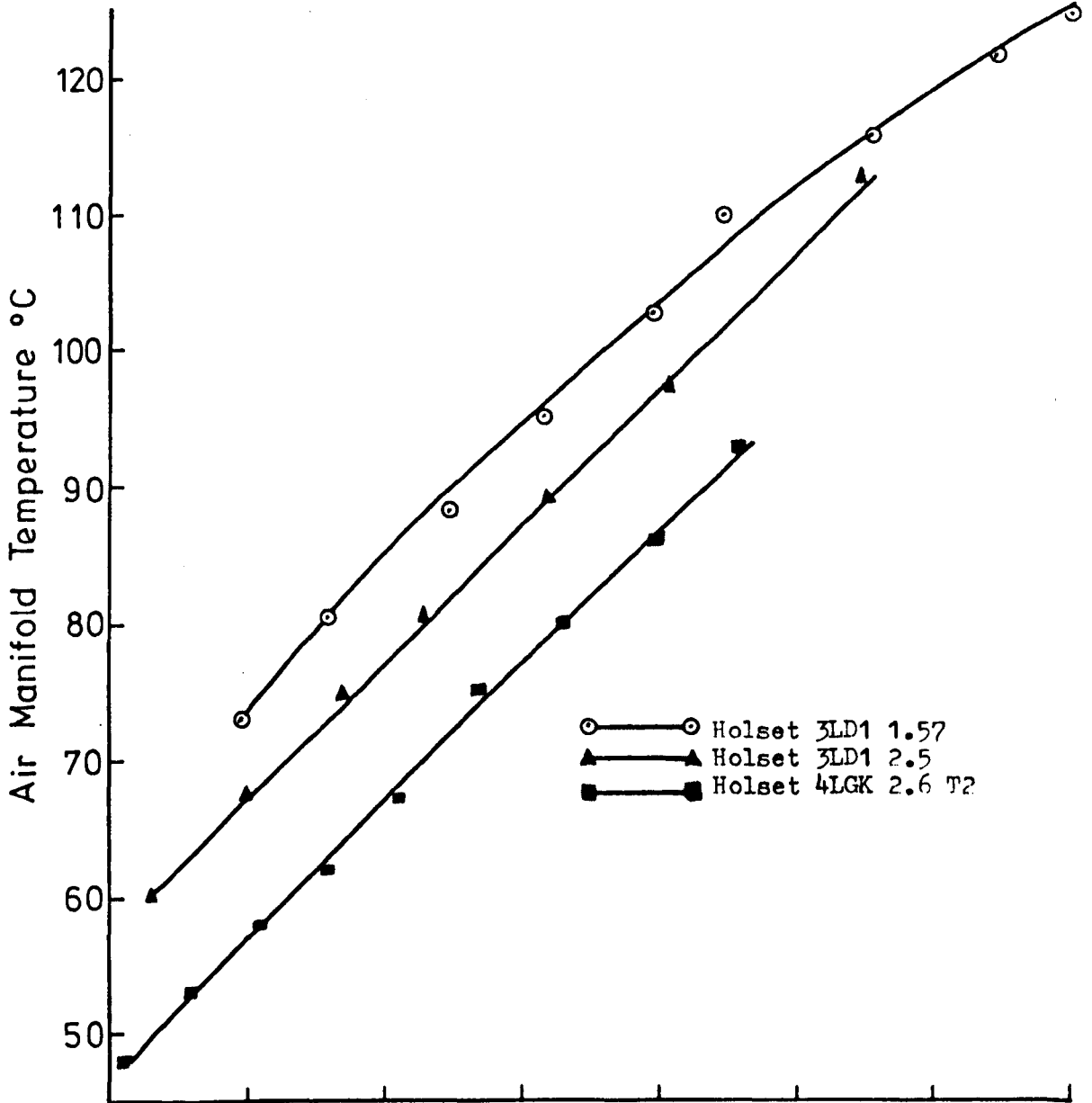
STAGE III VARIABLE SPEED HOLSET 4LGK 2.6 T2

FIGURE 8.19



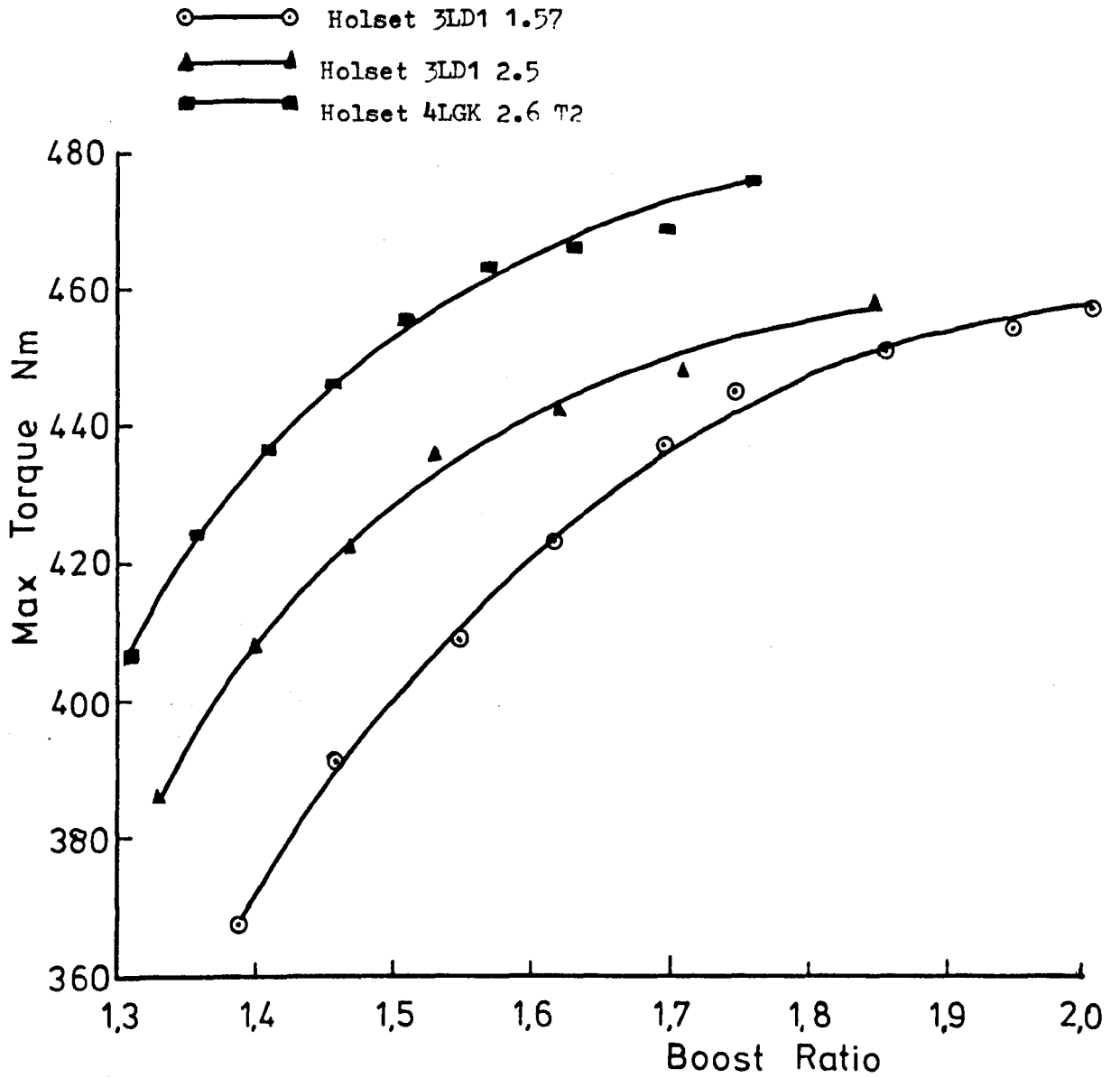
STAGE III [DGT] PERFORMANCE
(Engine Speed = 1200 rev/min)

FIGURE 8.20



STAGE III [DGT] PERFORMANCE
(Engine Speed = 1200 rev/min)

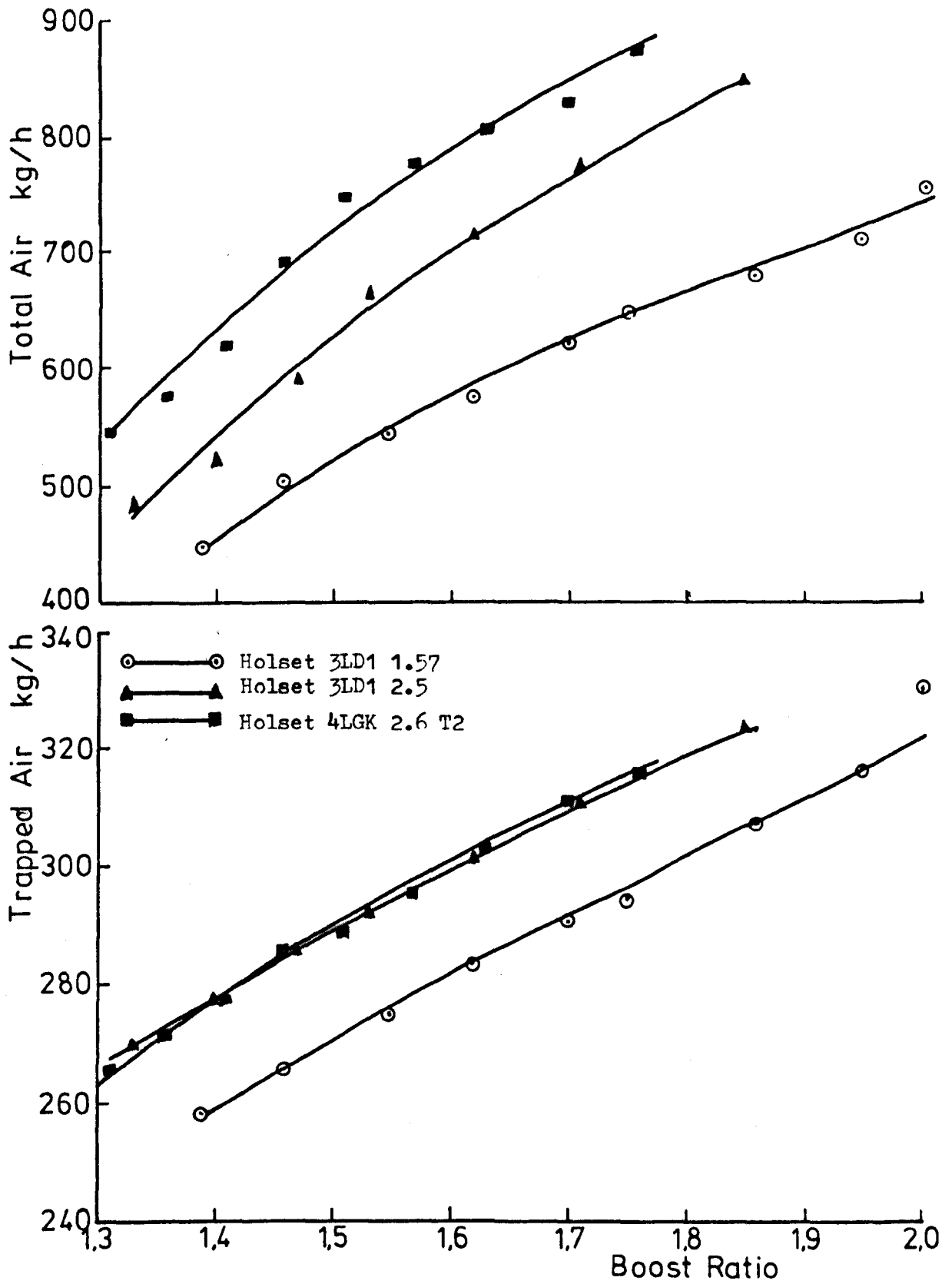
FIGURE 8.21



STAGE III [DGT] PERFORMANCE

(Engine Speed=1200 rev/min)

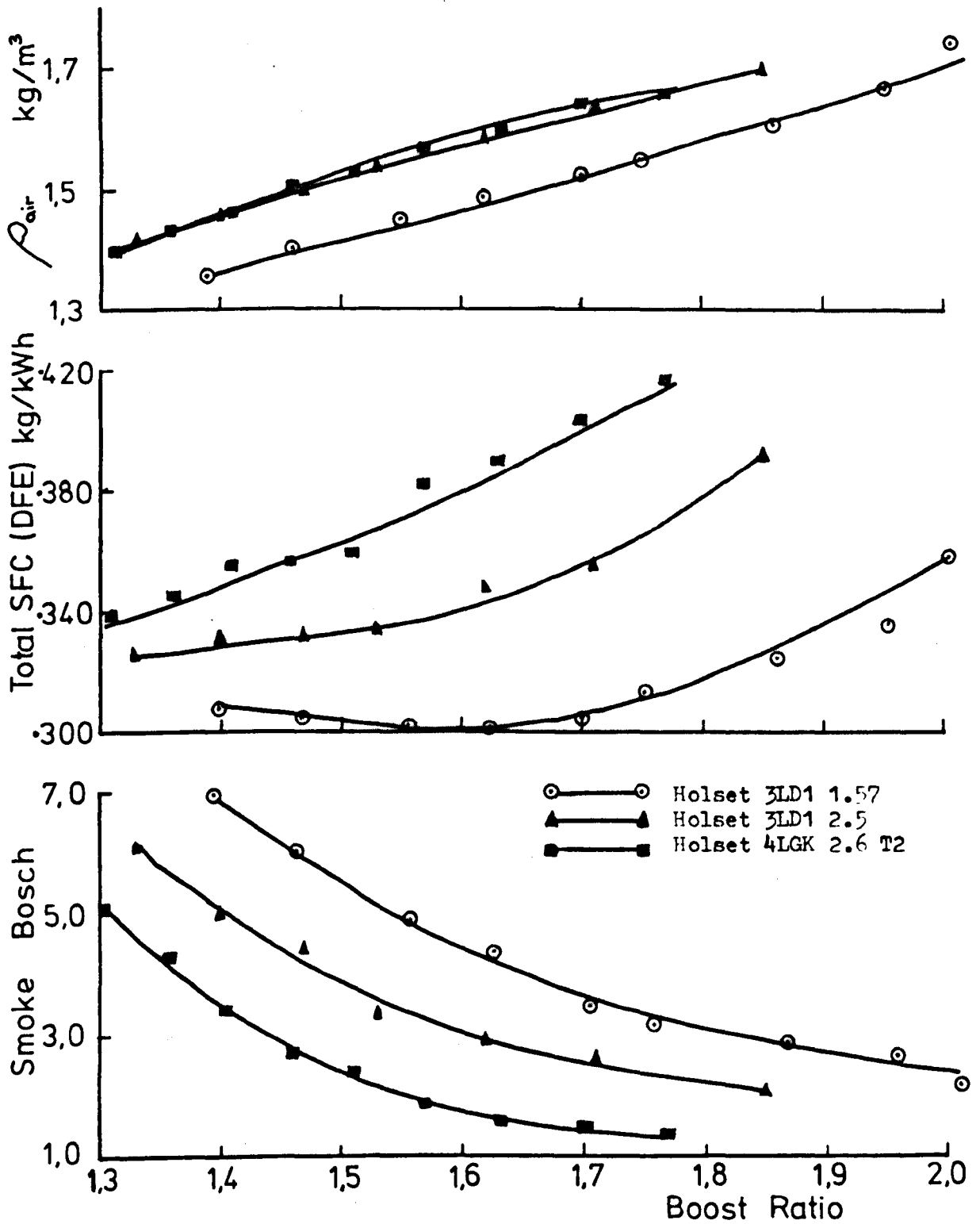
FIGURE 8.22



STAGE III [DGT] PERFORMANCE

(Engine Speed = 1200 rev/min)

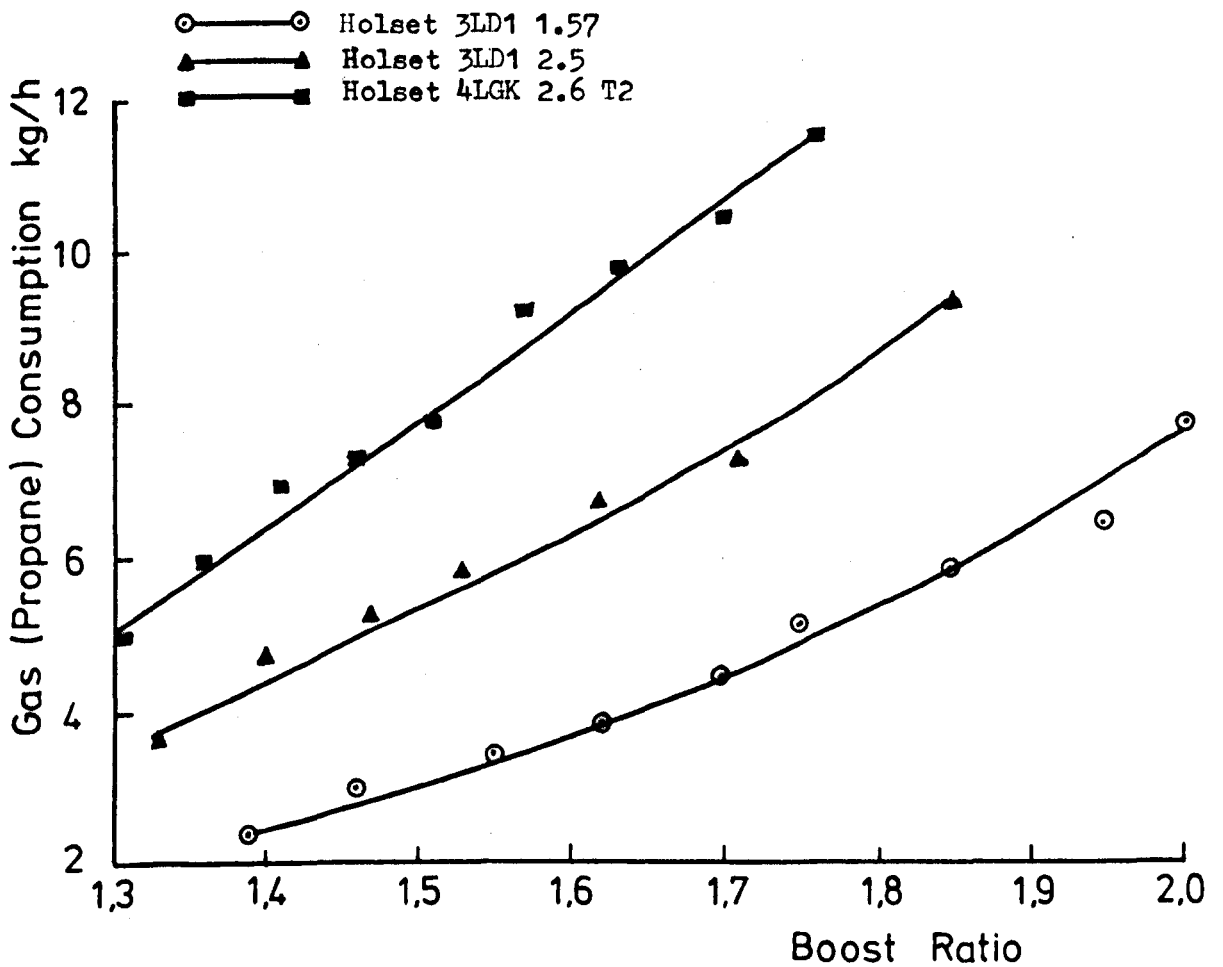
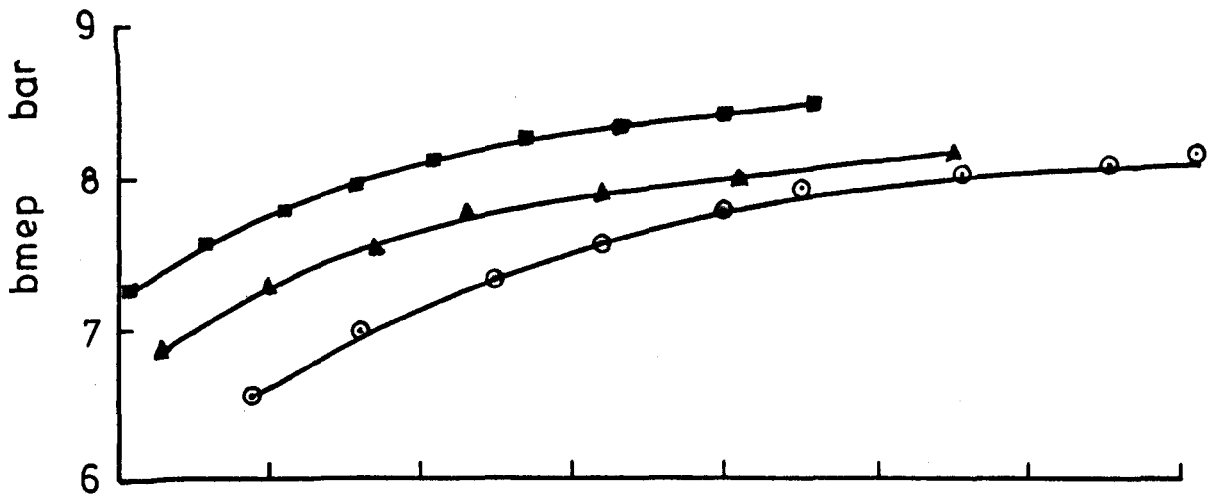
FIGURE 8.23



STAGE III [DGT] PERFORMANCE

(Engine Speed = 1200 rev/min)

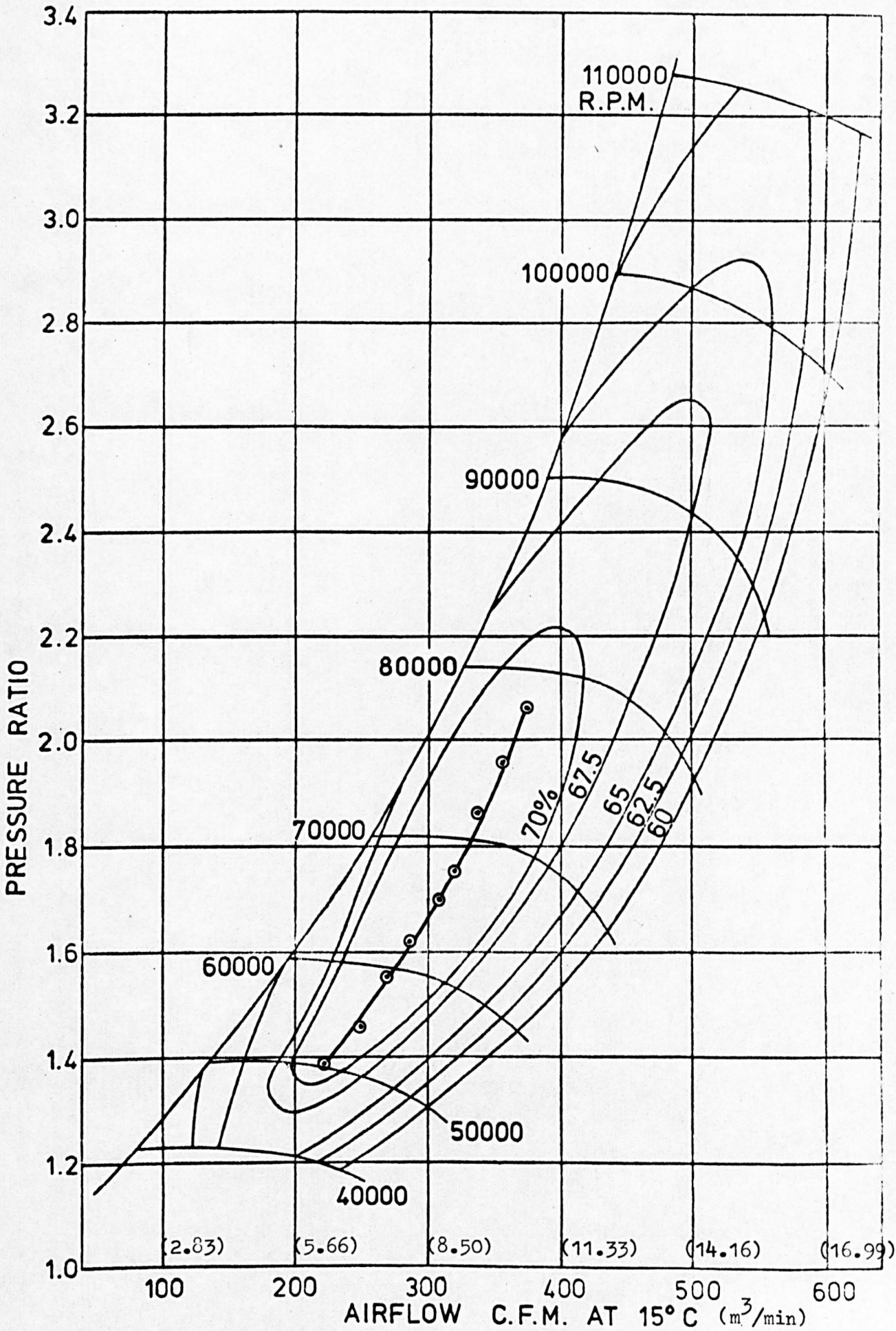
FIGURE 8.24



STAGE III [DGT] PERFORMANCE

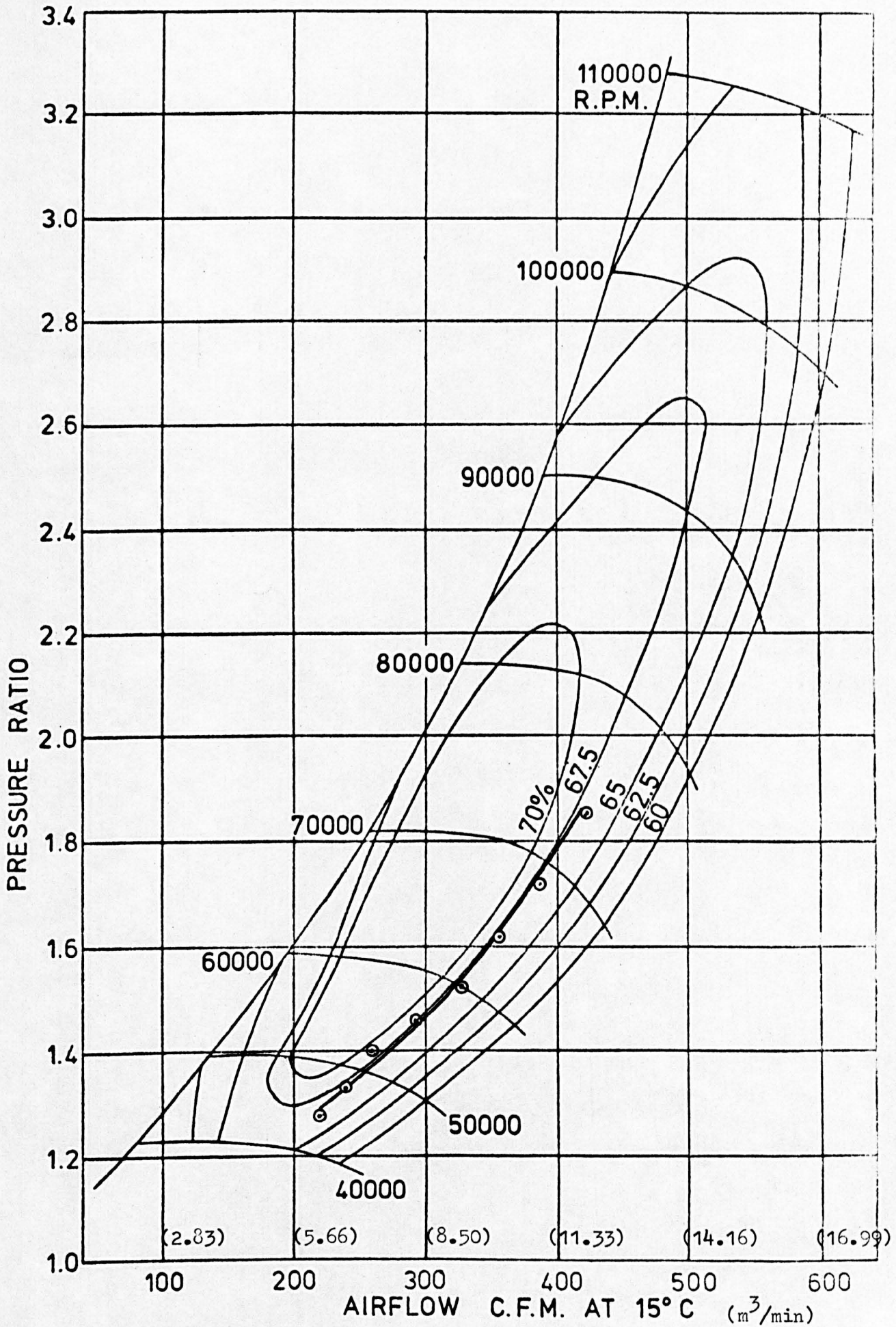
(Engine Speed = 1200 rev/min)

FIGURE 8,25



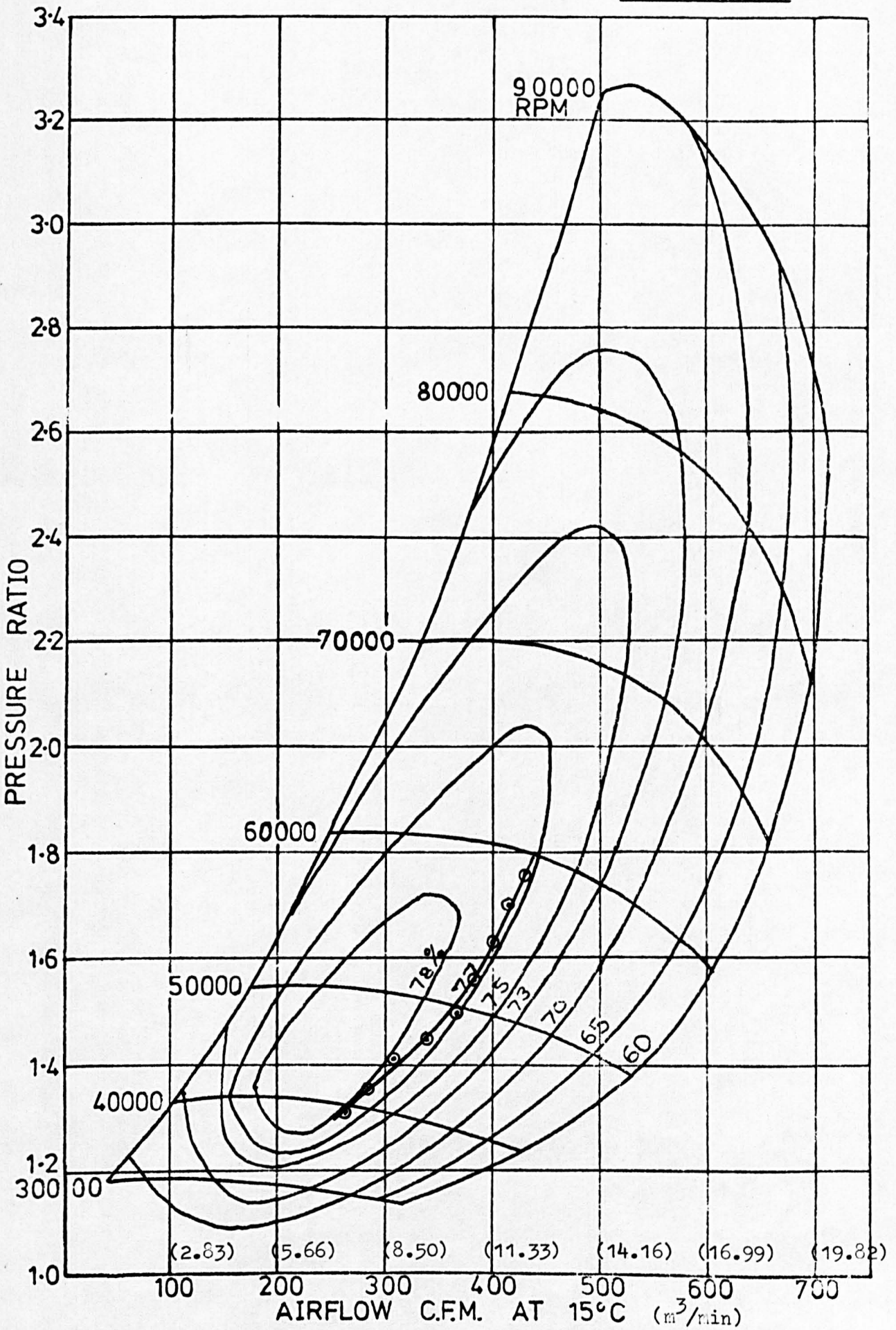
STAGE III (1200 Rev/min) HOLSET 3LD1 1,57

FIGURE 8.26



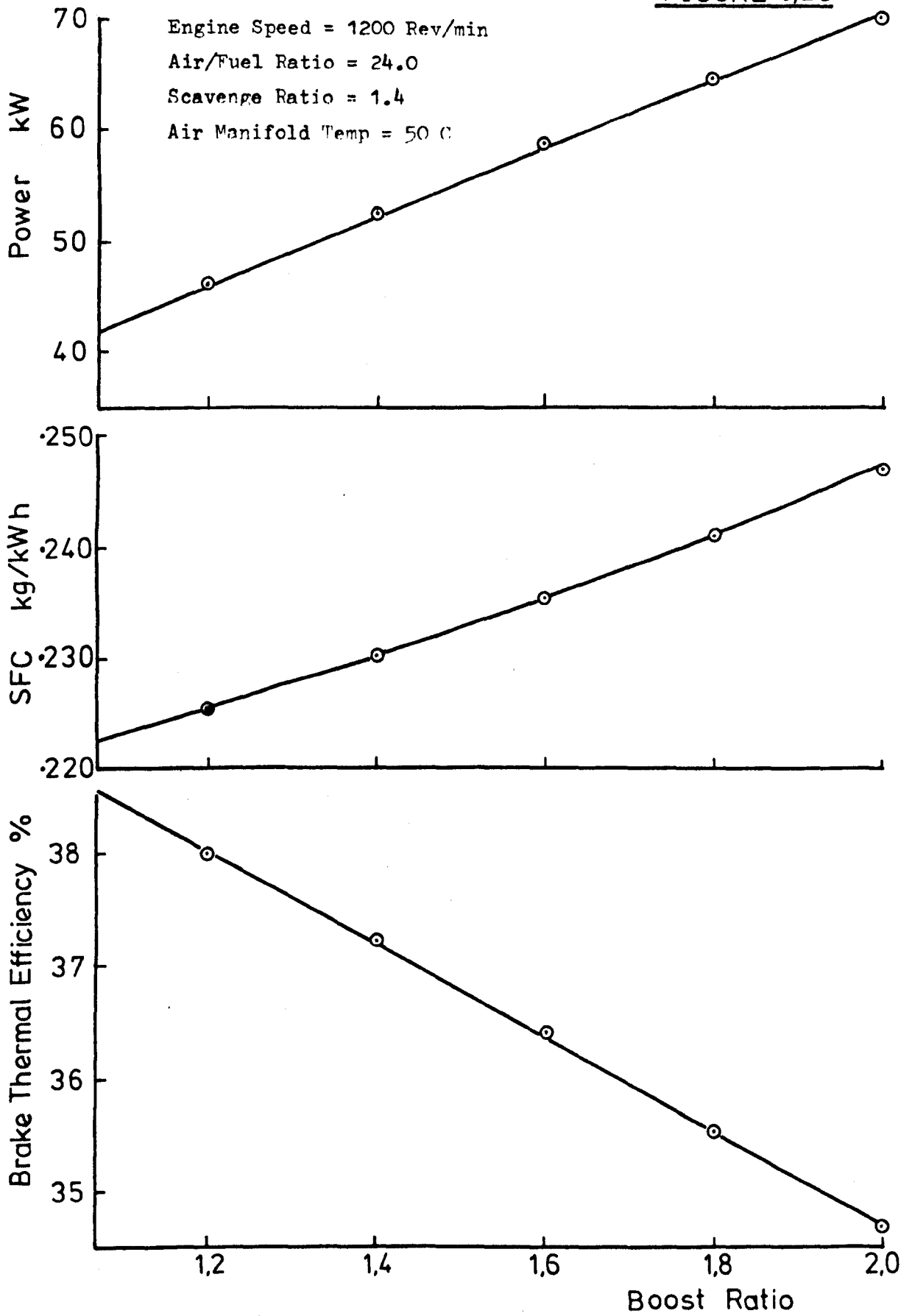
STAGE III (1200 Rev/min) HOLSET 3LD1 2.5

FIGURE 8,27



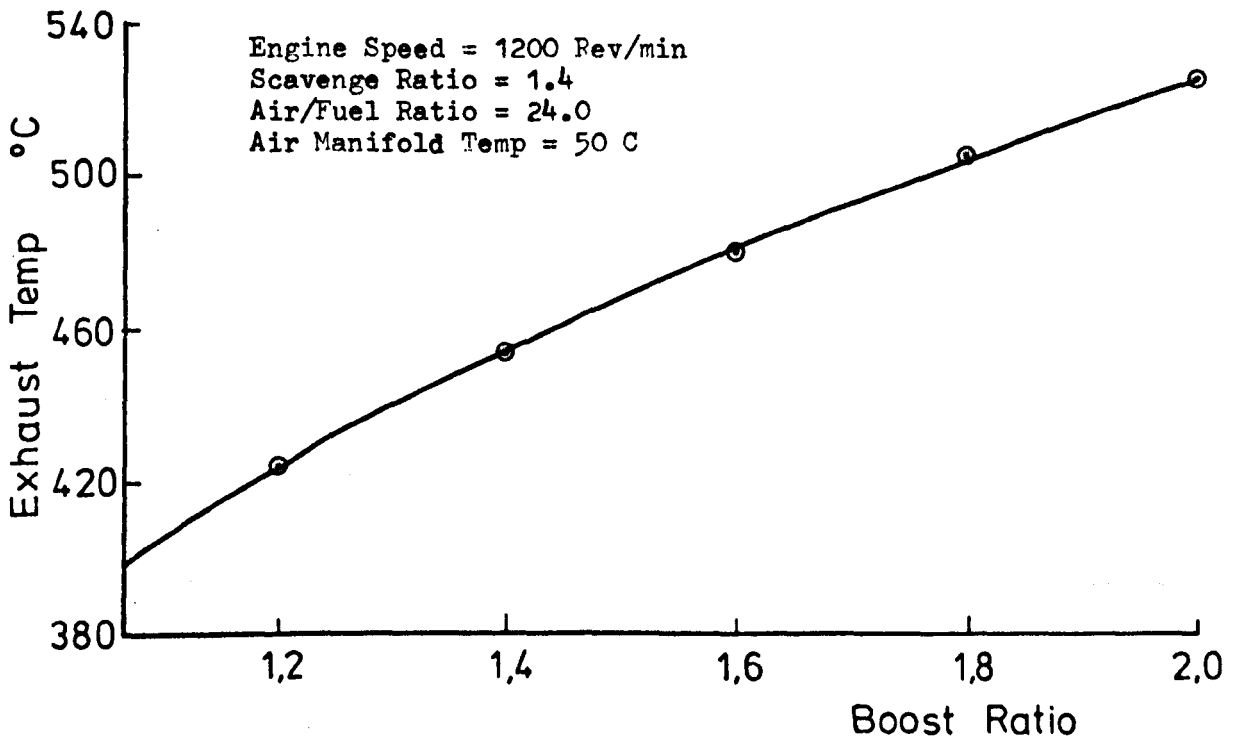
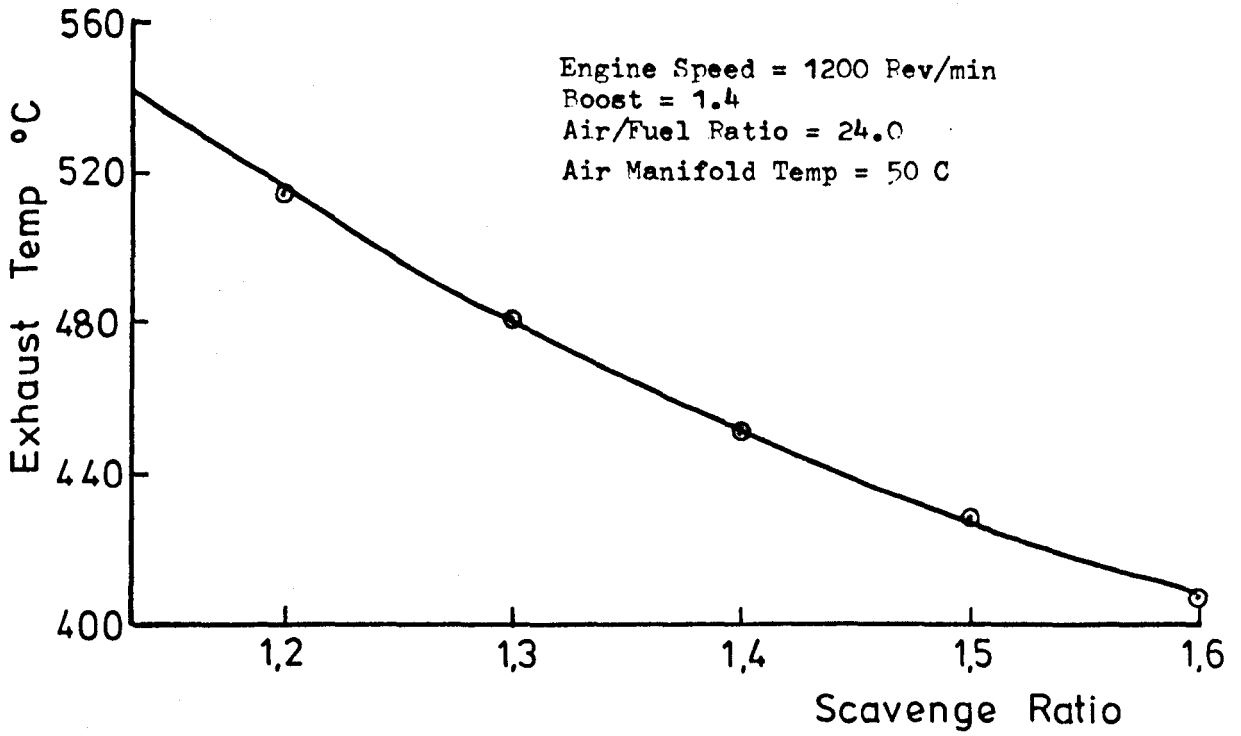
STAGE III (1200 Rev/min) HOLSET 4LGK 2.6 T2

FIGURE 8,28



THEORETICAL PREDICTIONS
(Standard Engine Program)

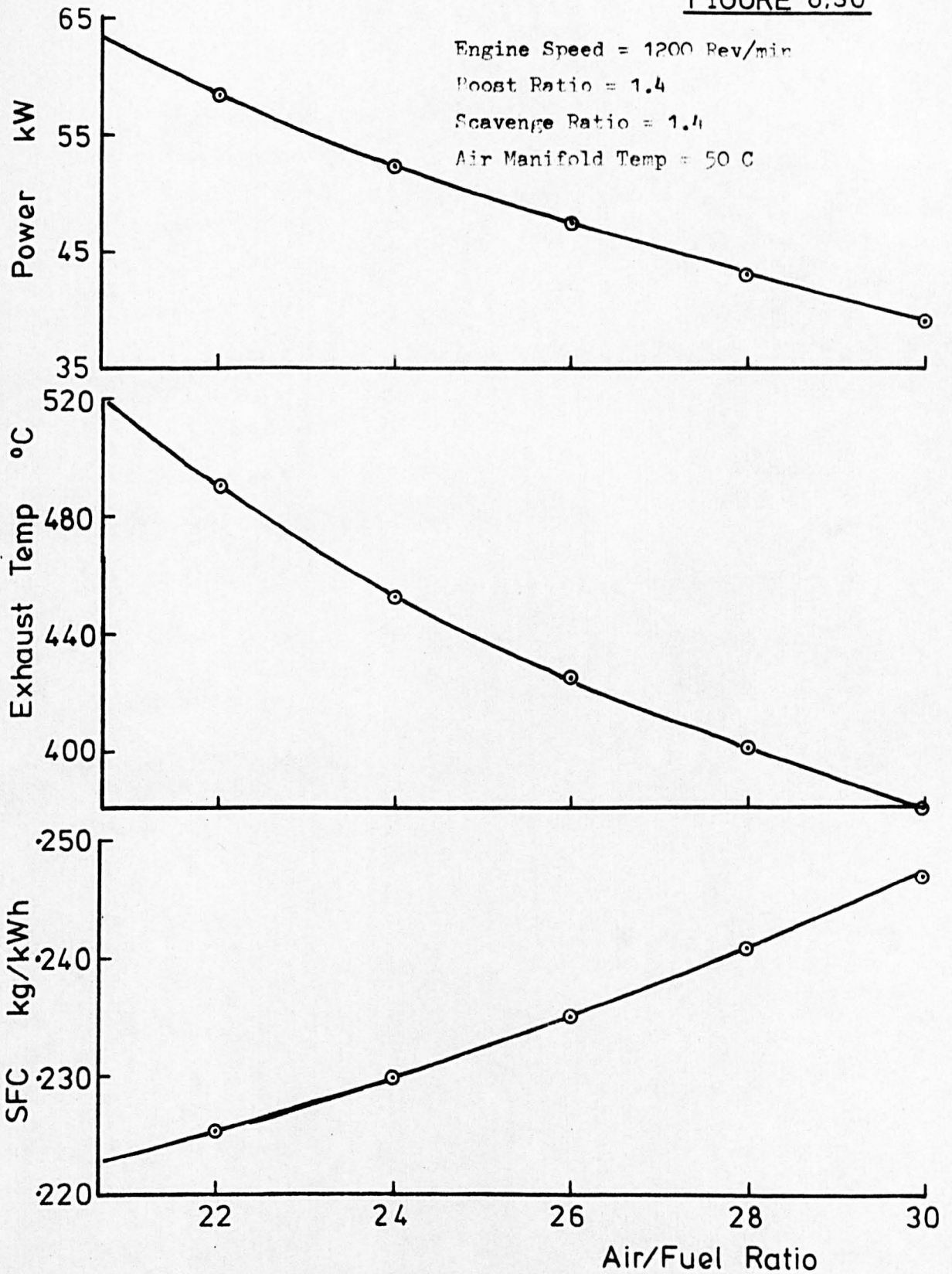
FIGURE 8.29



THEORETICAL PREDICTIONS

(Standard Engine Program)

FIGURE 8.30



THEORETICAL PREDICTIONS

(Standard Engine Program)

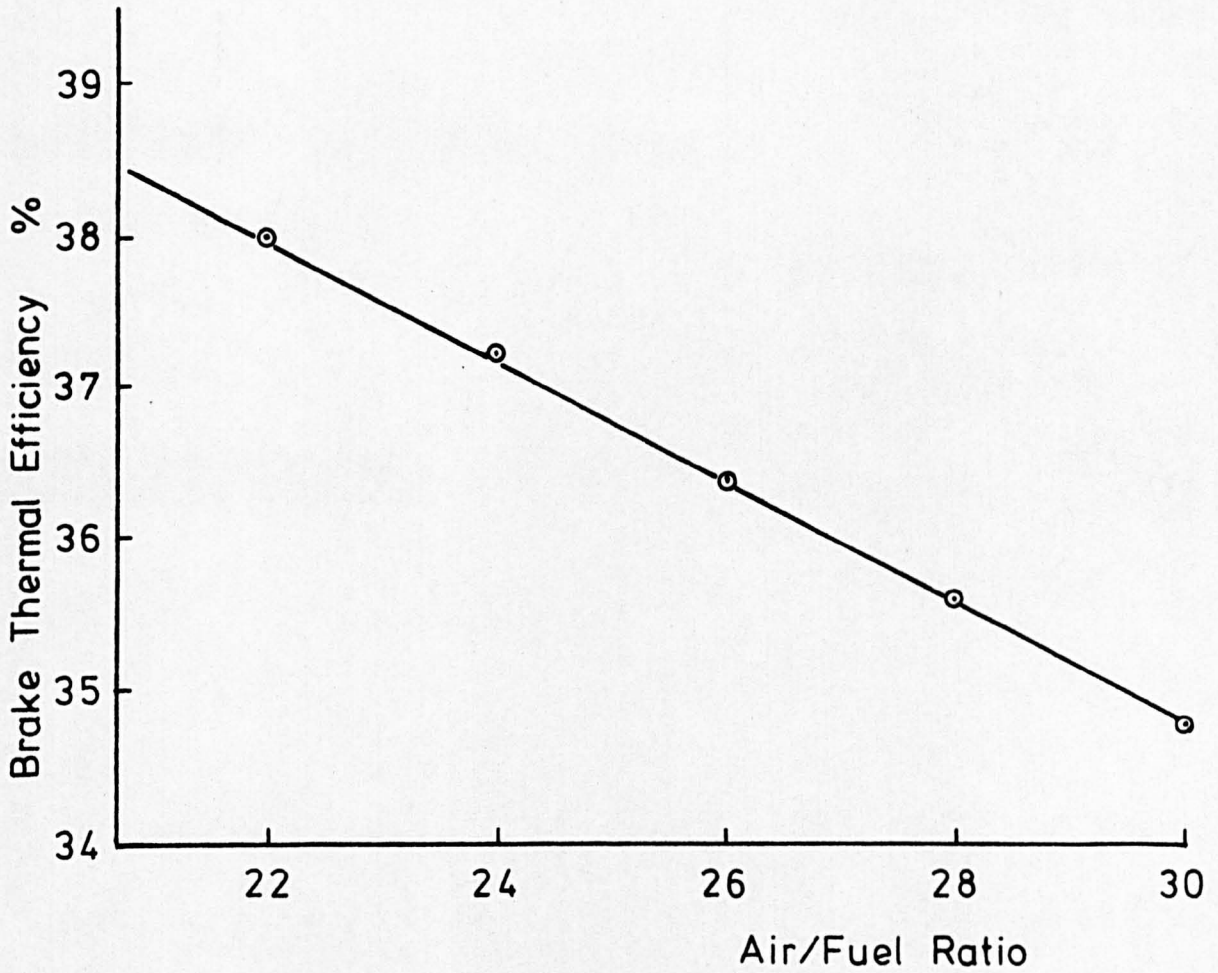
FIGURE 8.31

Engine Speed = 1200 Rev/min

Boost Ratio = 1.4

Scavenge Ratio = 1.4

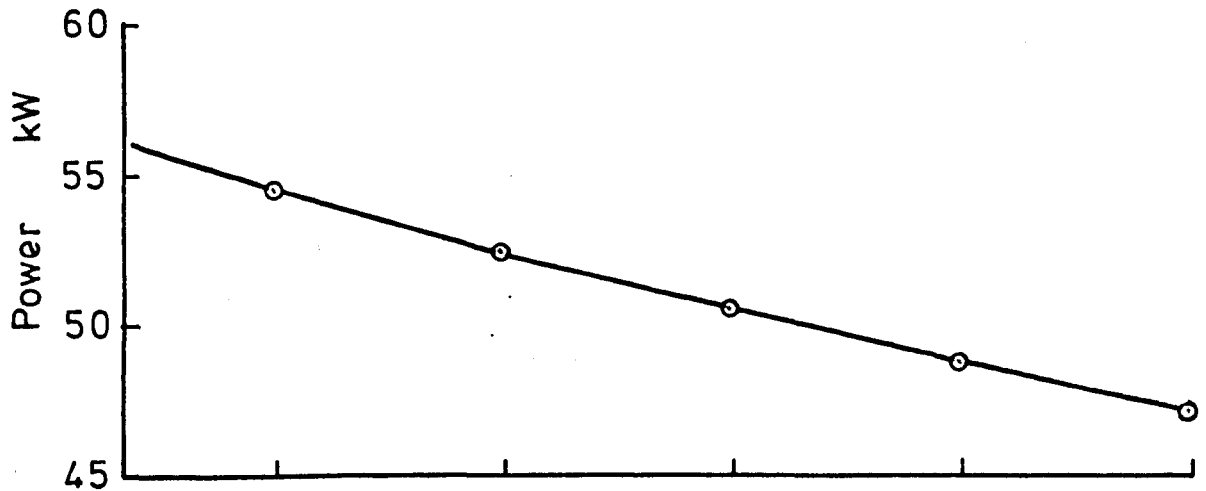
Air Manifold Temperature 50 C



THEORETICAL PREDICTIONS

(Standard Engine Program)

FIGURE 8.32

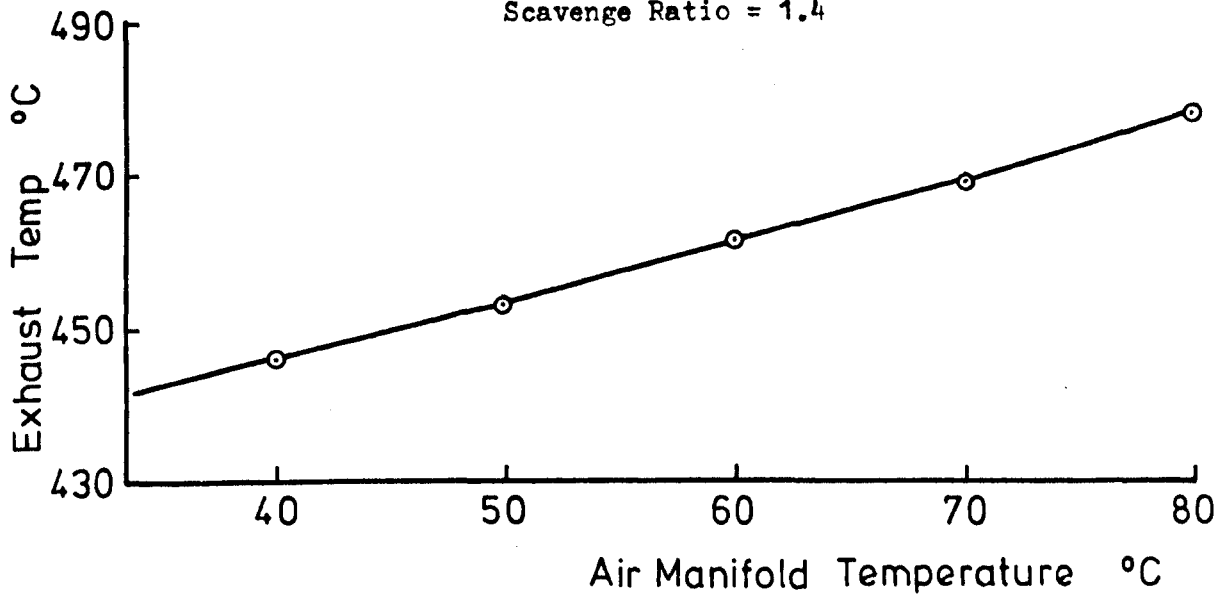


Engine Speed = 1200 Rev/min

Air/Fuel Ratio = 24.0

Boost Ratio = 1.4

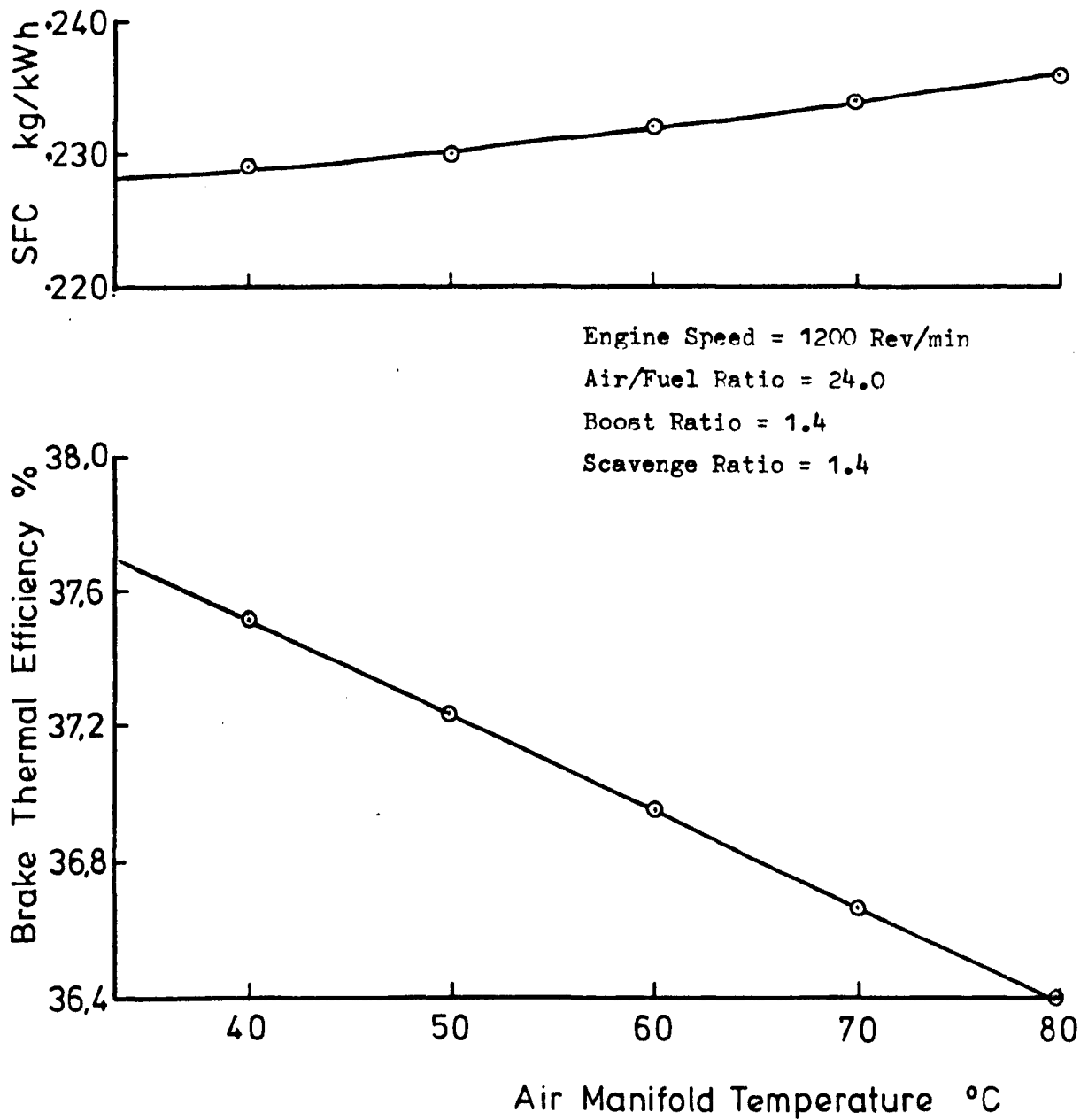
Scavenge Ratio = 1.4



THEORETICAL PREDICTIONS

(Standard Engine Program)

FIGURE 8.33



THEORETICAL PREDICTIONS

(Standard Engine Program)

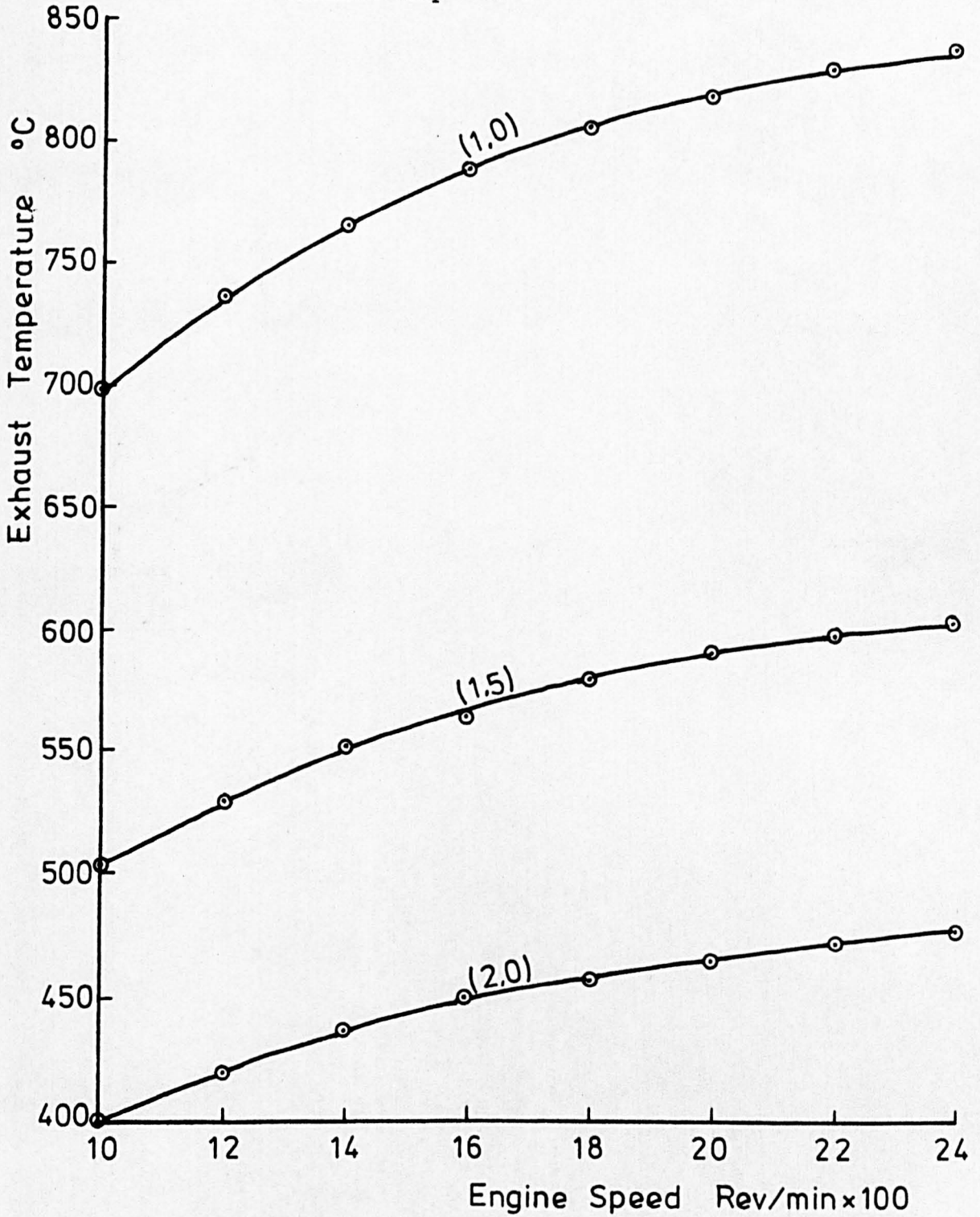
FIGURE 8,34

Boost Ratio = 1.5

Air/Fuel Ratio = 20.0

Air Manifold Temp = 60 C

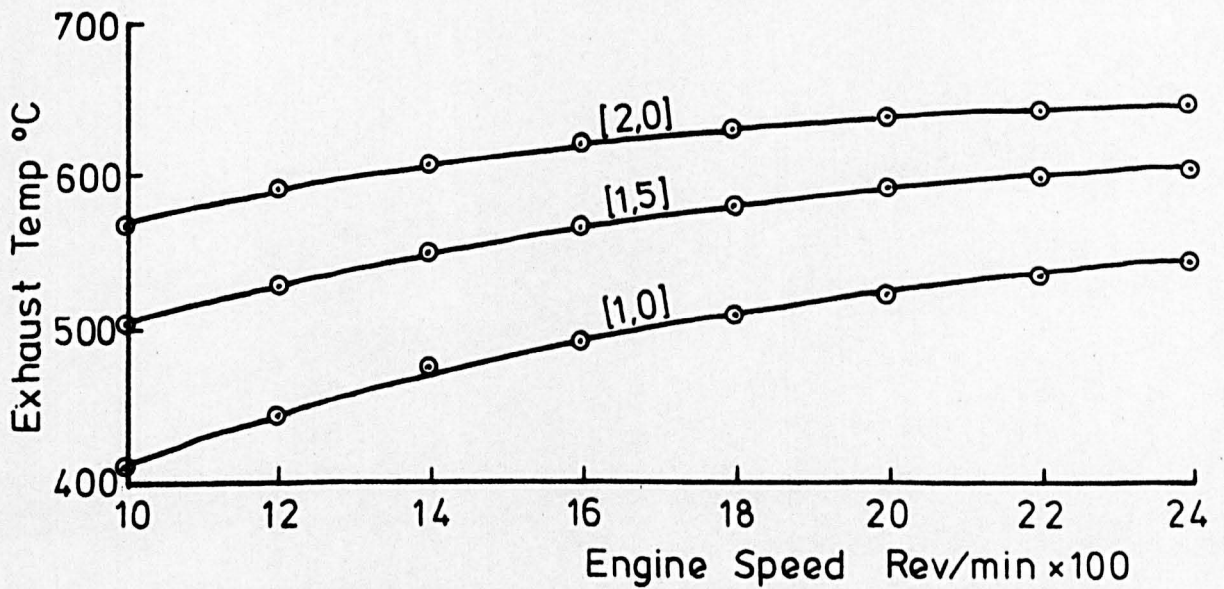
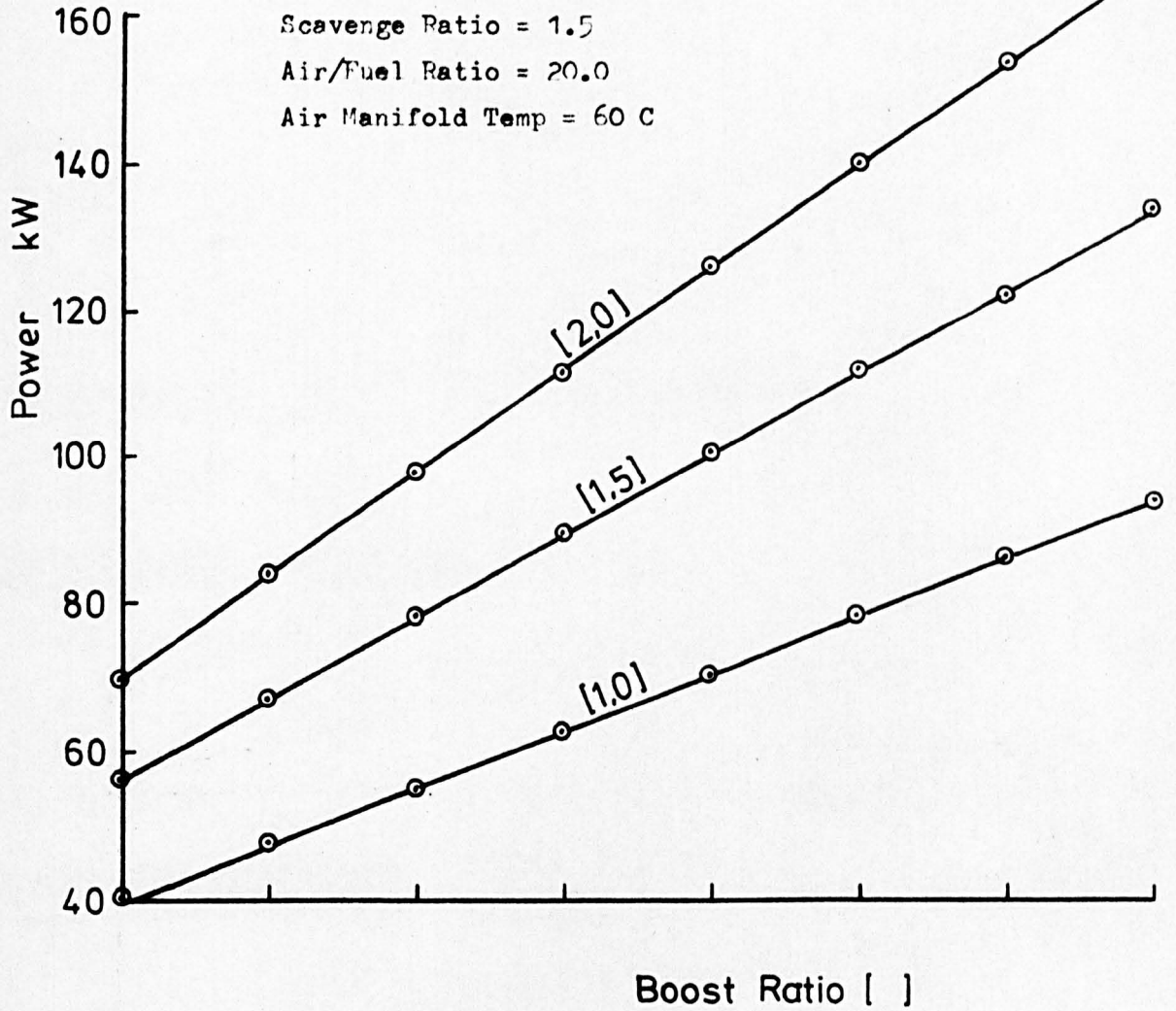
Scavenge Ratio ()



THEORETICAL PREDICTIONS

(Standard Engine Program)

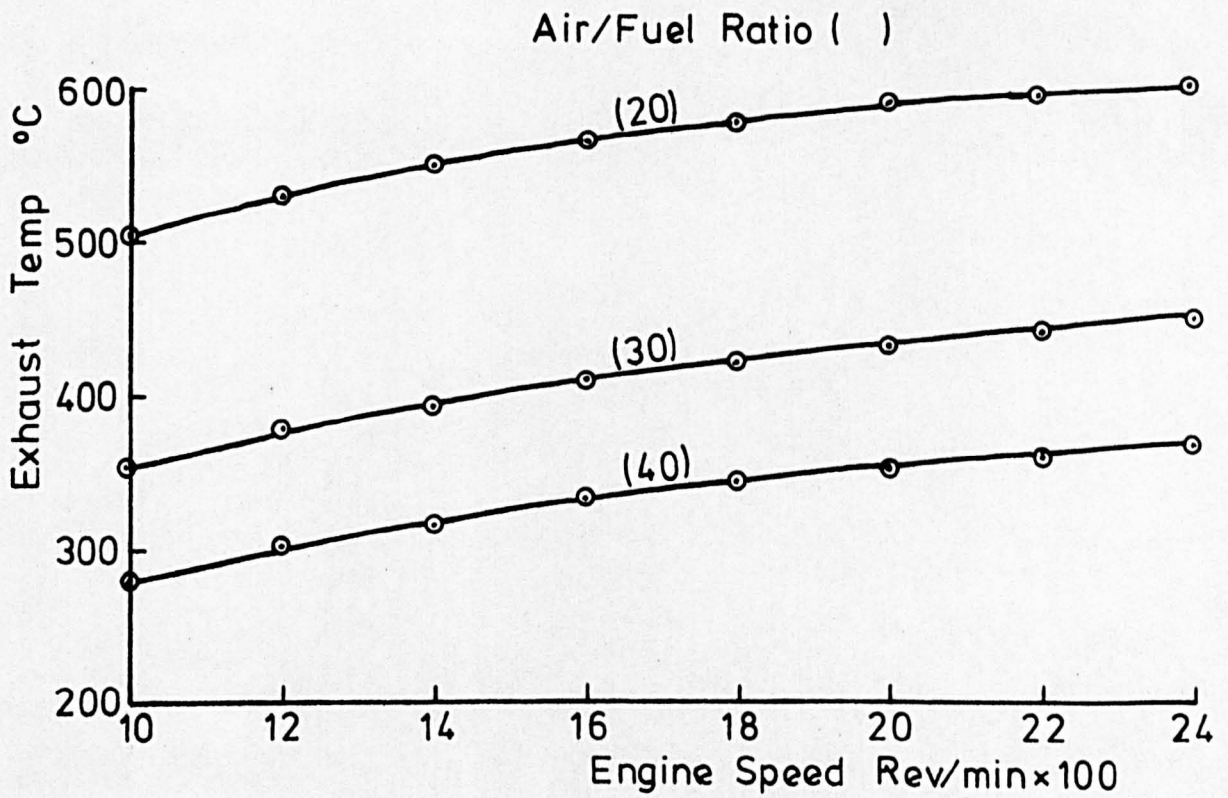
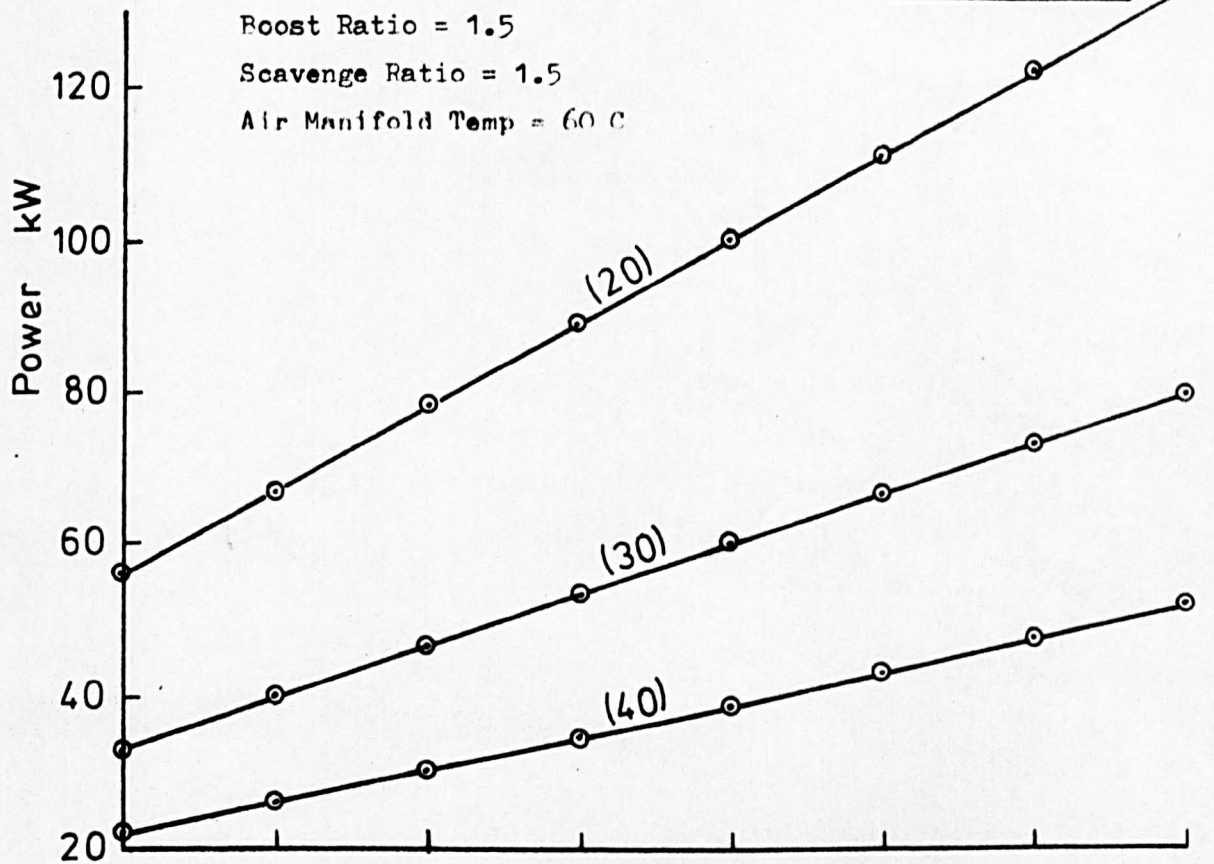
FIGURE 8.35



THEORETICAL PREDICTIONS

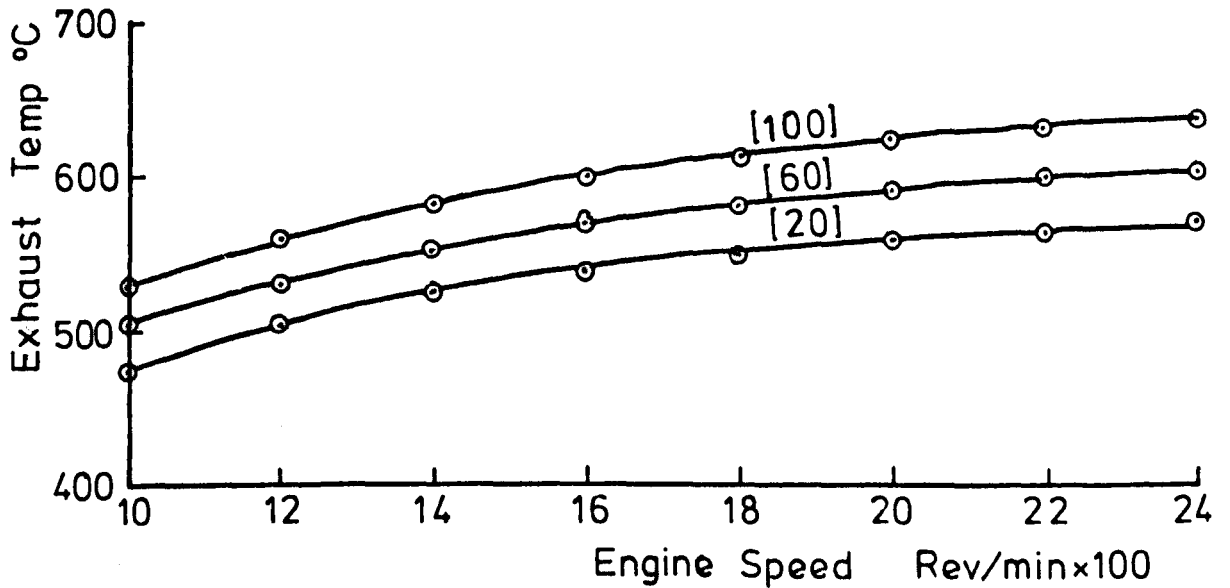
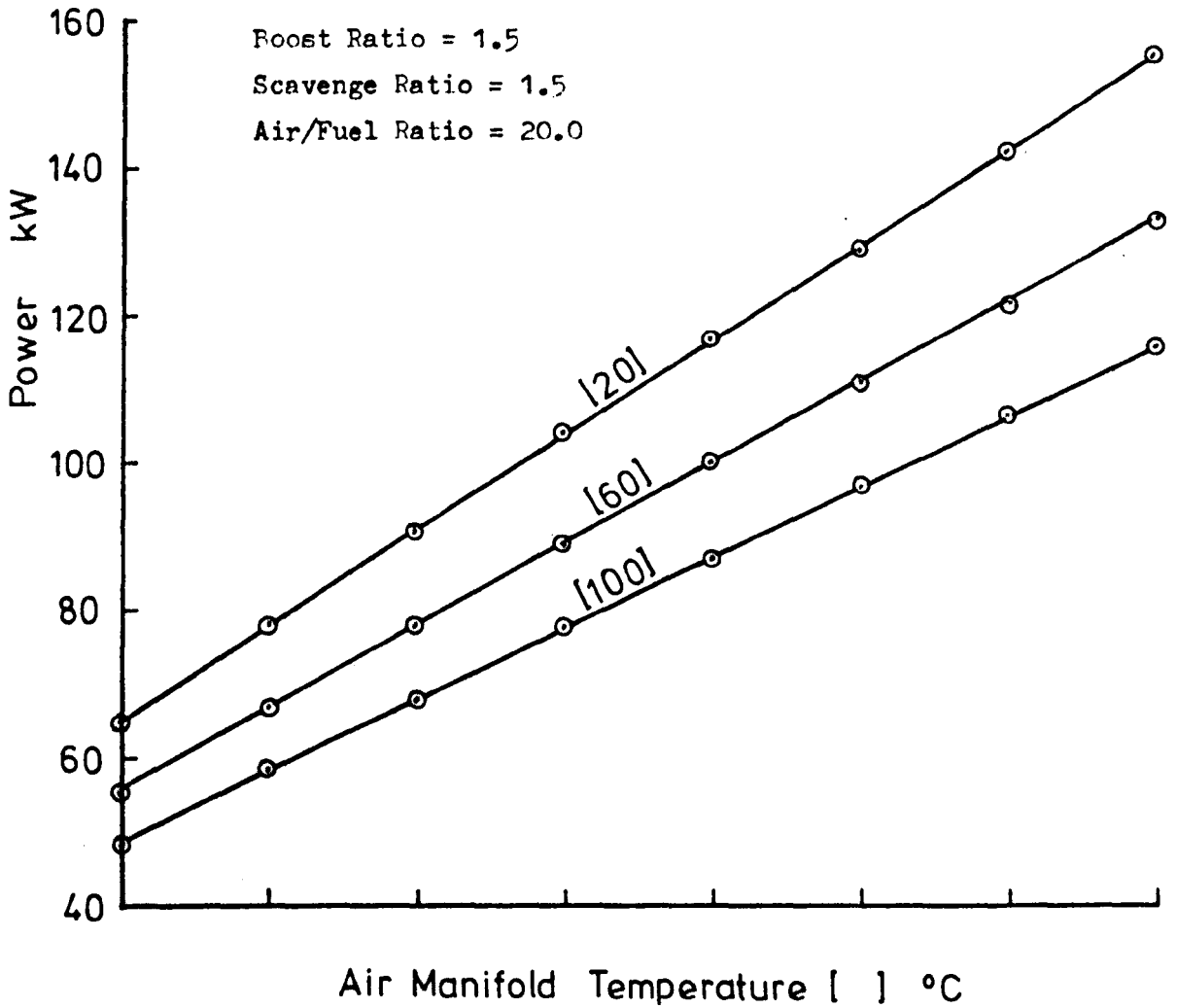
(Standard Engine Program)

FIGURE 8,36



THEORETICAL PREDICTIONS
(Standard Engine Program)

FIGURE 8.37



THEORETICAL PREDICTIONS
(Standard Engine Program)

FIGURE 8.38

Air/Fuel Ratio = 20.0

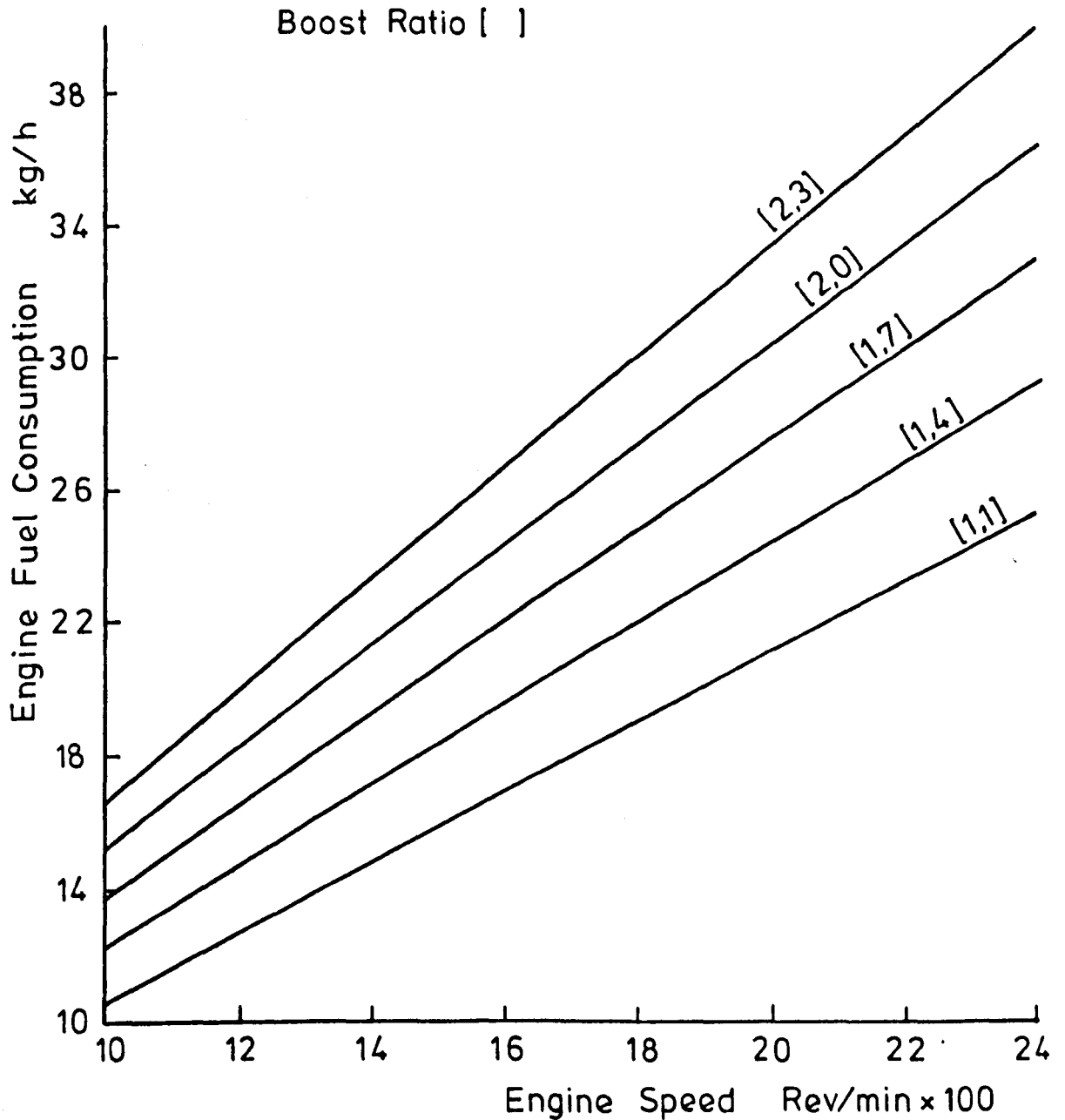
$\eta_r = 60\%$ $\eta_c = 70\%$

Series Turbocharged System

No Auxiliary Combustion

No Scavenge Blower

No Charge Cooling



THEORETICAL PREDICTIONS

(Preliminary Matching Program)

FIGURE 8.39

Air/Fuel Ratio = 20.0

Scavenge Ratio = 1.6

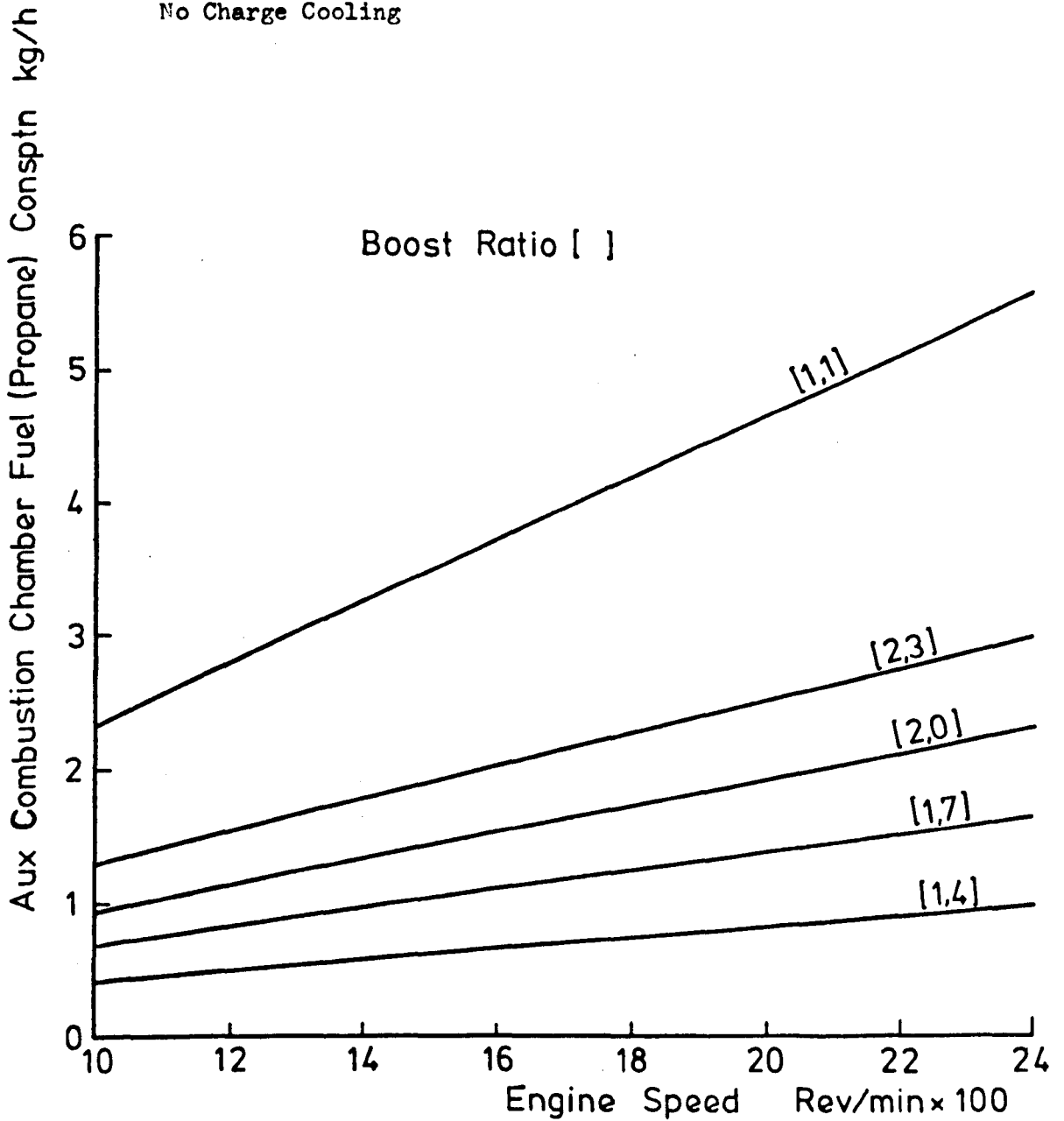
Blowdown Temp = 1000 C

$\eta_r = 60\%$ $\eta_c = 70\%$

Series Turbocharged System With Auxiliary Combustion

No Scavenge blower

No Charge Cooling

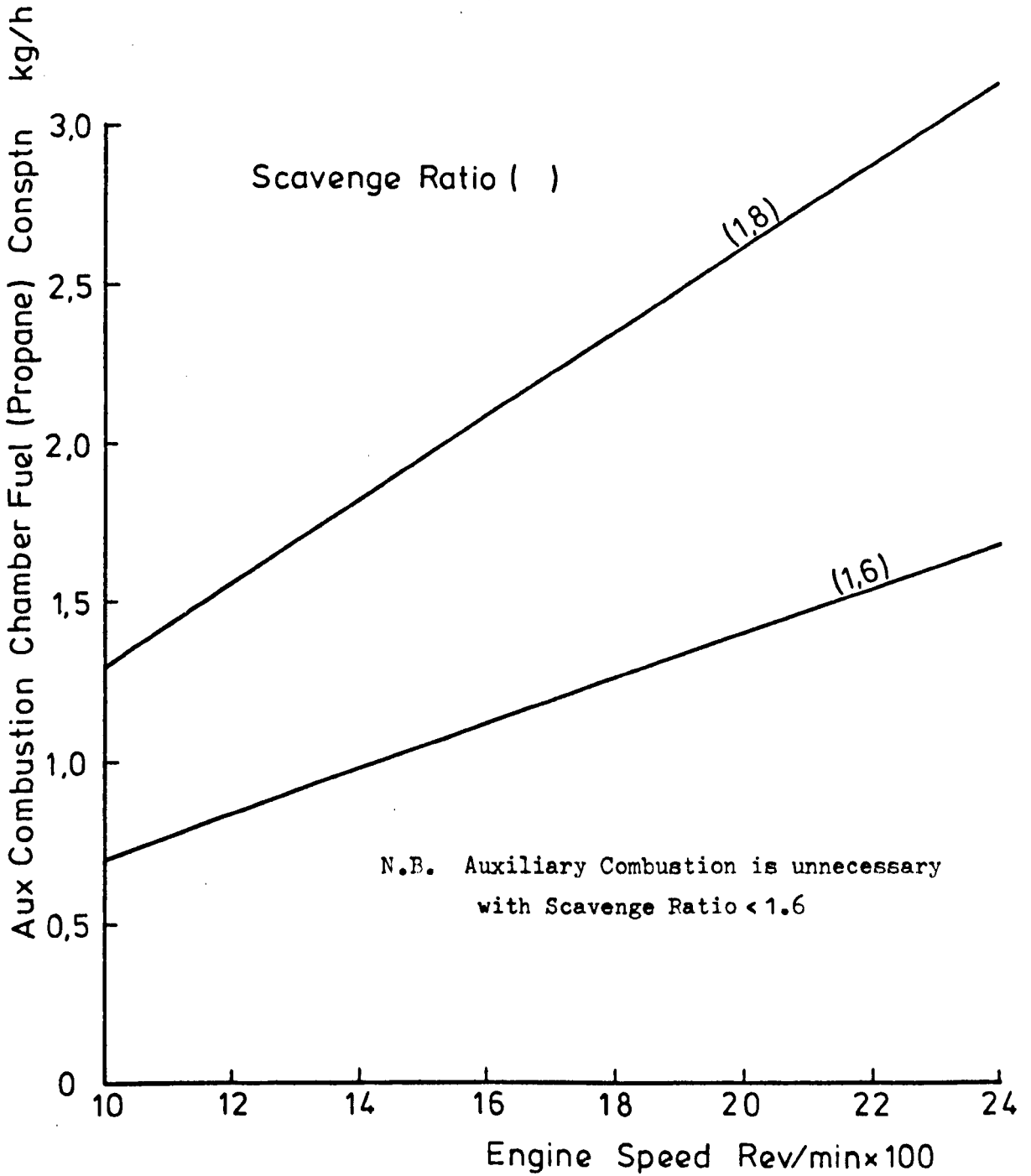


THEORETICAL PREDICTIONS

(Preliminary Matching Program)

FIGURE 8.40

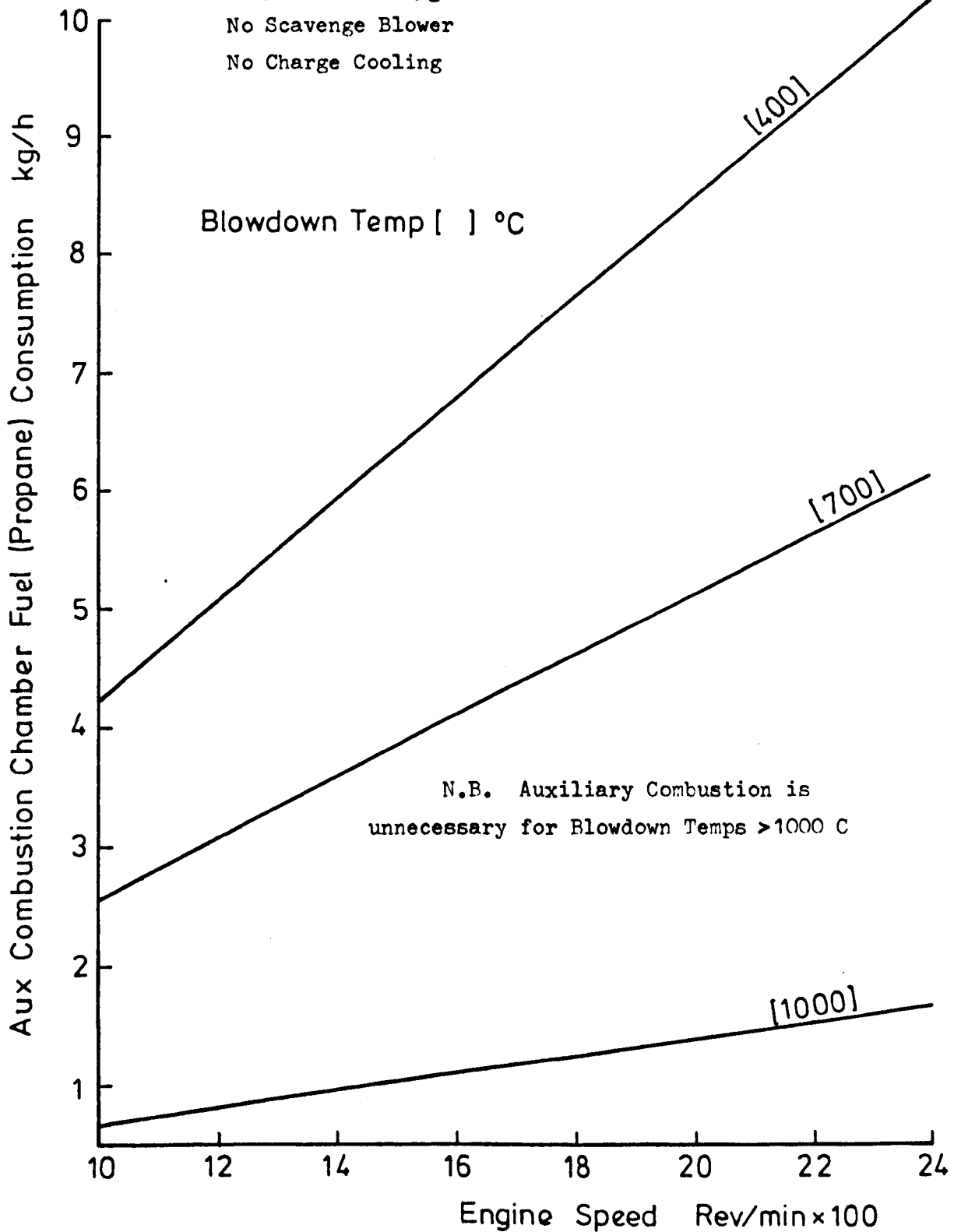
Boost Ratio = 1.7
Air/Fuel Ratio = 20
Blowdown Temp = 1000 C
 $\eta_r = 0.60$ $\eta_c = 0.70$
No Scavenge Blower
No Charge Cooling



THEORETICAL PREDICTIONS
(Preliminary Matching Program)

FIGURE 8,41

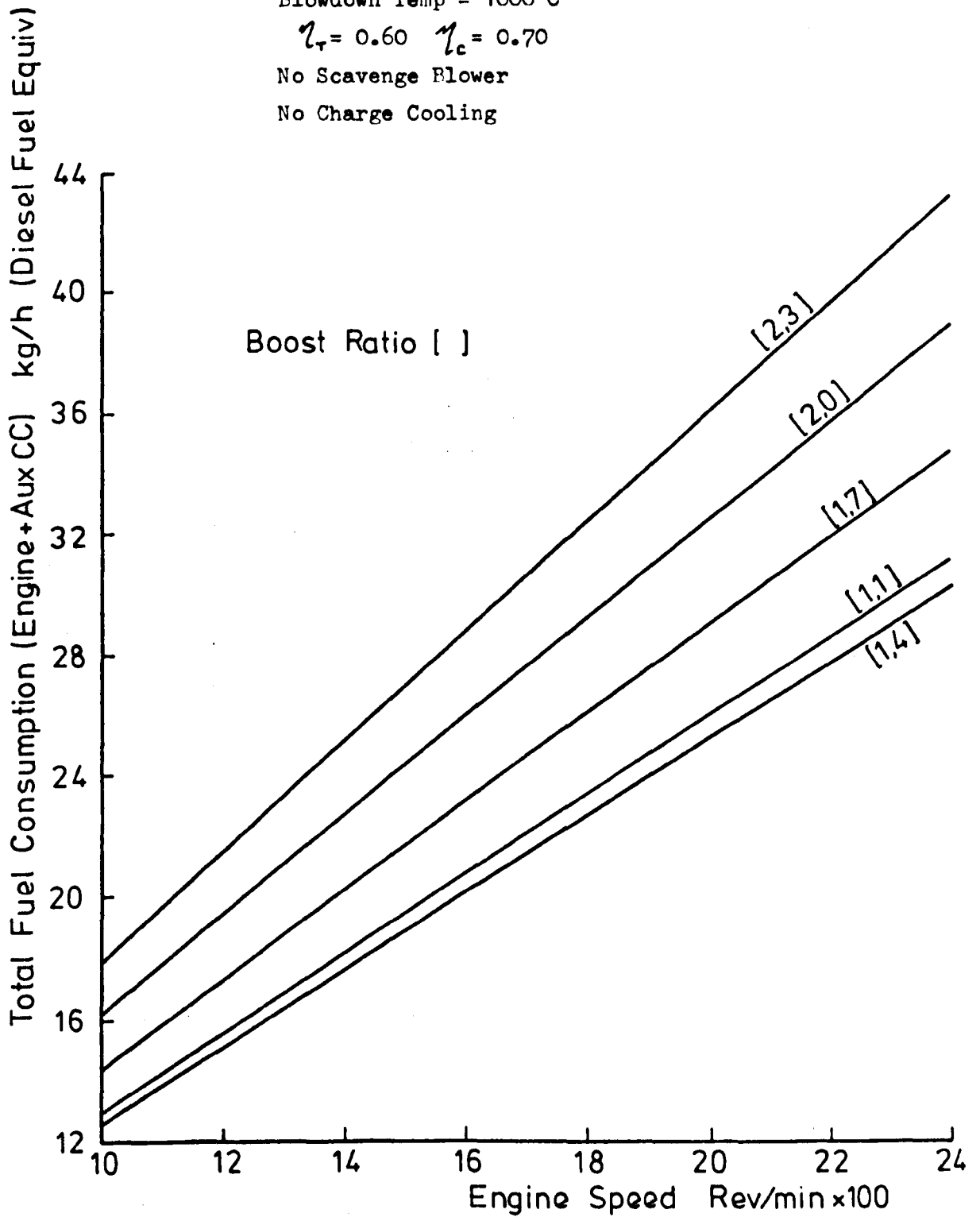
Boost Ratio = 1.7
Air/Fuel Ratio = 20
Scavenge Ratio = 1.6
 $\eta_r = 0.60$ $\eta_c = 0.70$
No Scavenge Blower
No Charge Cooling



THEORETICAL PREDICTIONS
(Preliminary Matching Program)

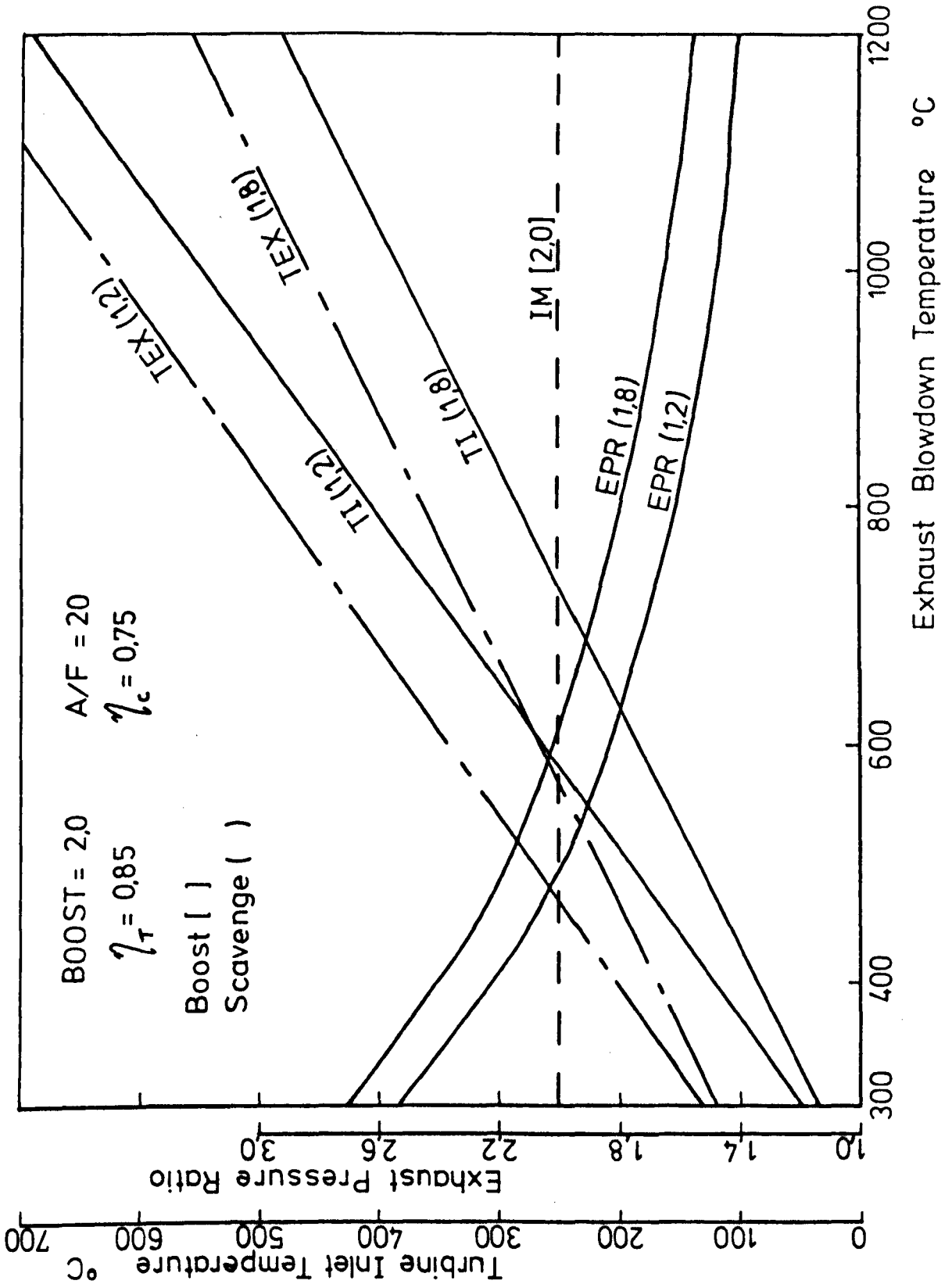
FIGURE 8.42

Air/Fuel Ratio = 20
Scavenge Ratio = 1.6
Blowdown Temp = 1000 C
 $\eta_r = 0.60$ $\eta_c = 0.70$
No Scavenge Blower
No Charge Cooling



THEORETICAL PREDICTIONS
(Preliminary Matching Program)

FIGURE 8.43



PRELIMINARY MATCHING PROGRAM

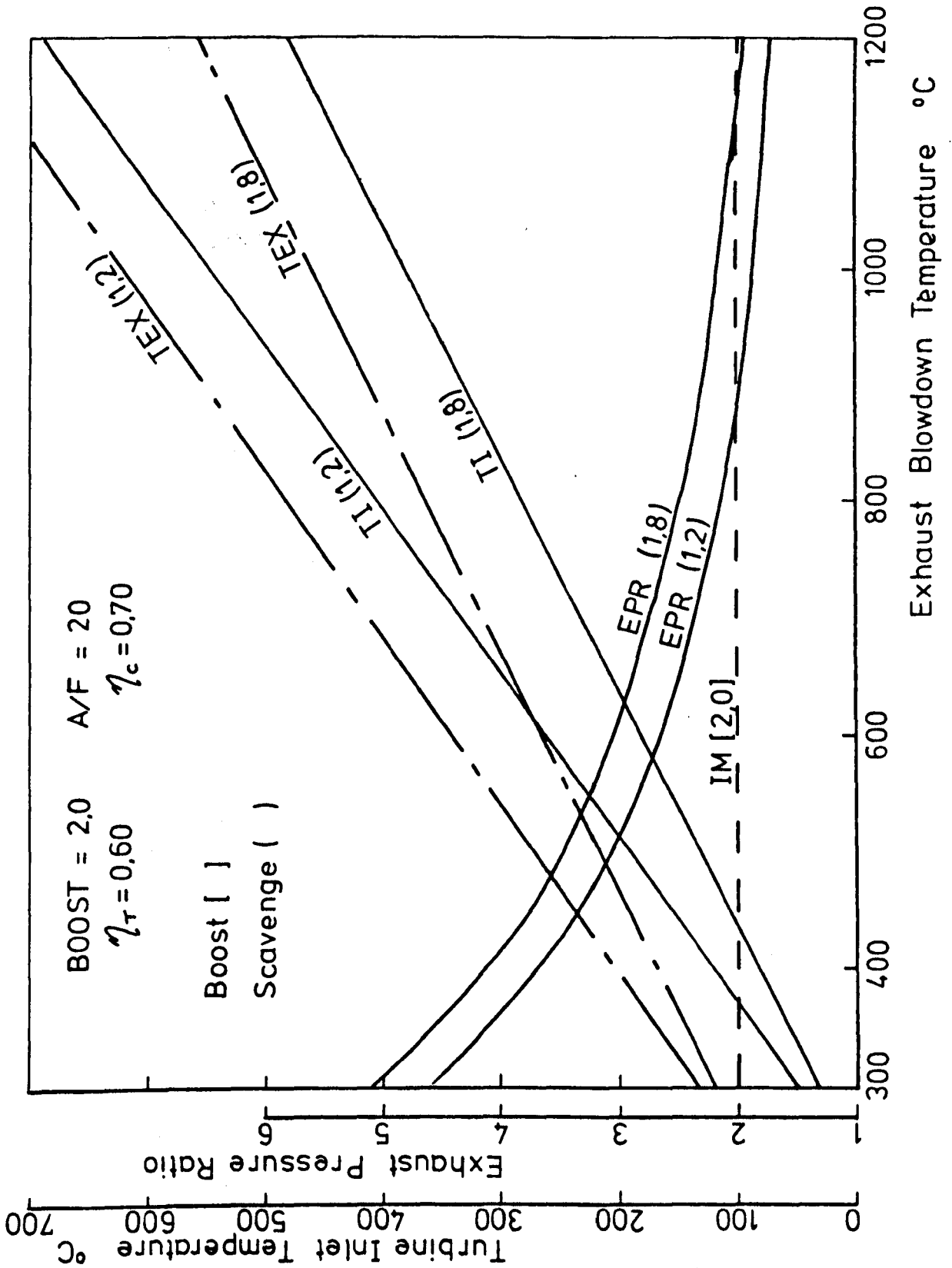
(Aid to Matching)

BOOST = 2.0

$\eta_T = 0.85$

$\eta_c = 0.75$

FIGURE 8.44



PRELIMINARY MATCHING PROGRAM

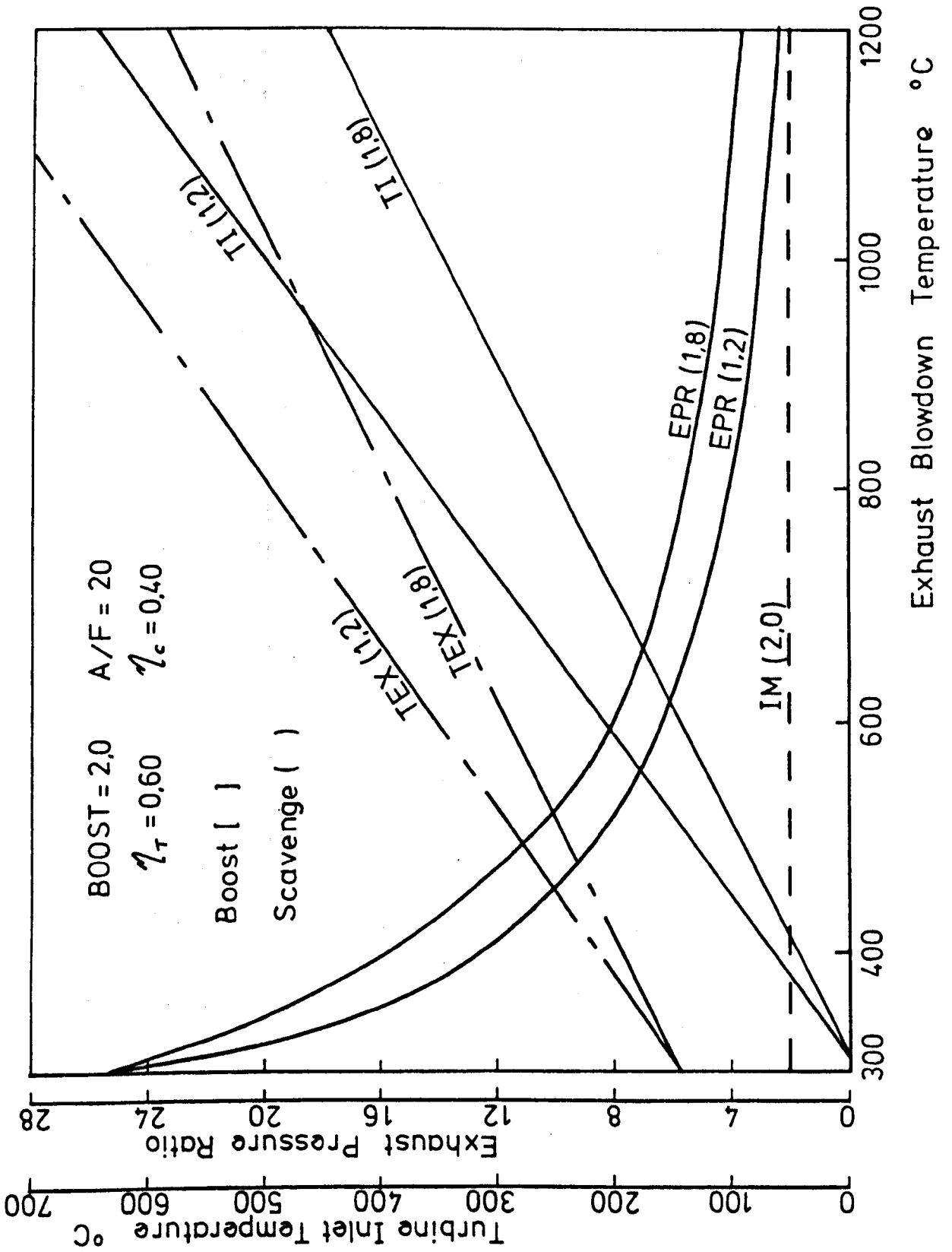
(Aid to Matching)

BOOST = 2.0

$\eta_{\tau} = 0.60$

$\eta_c = 0.70$

FIGURE 8.45



PRELIMINARY MATCHING PROGRAM

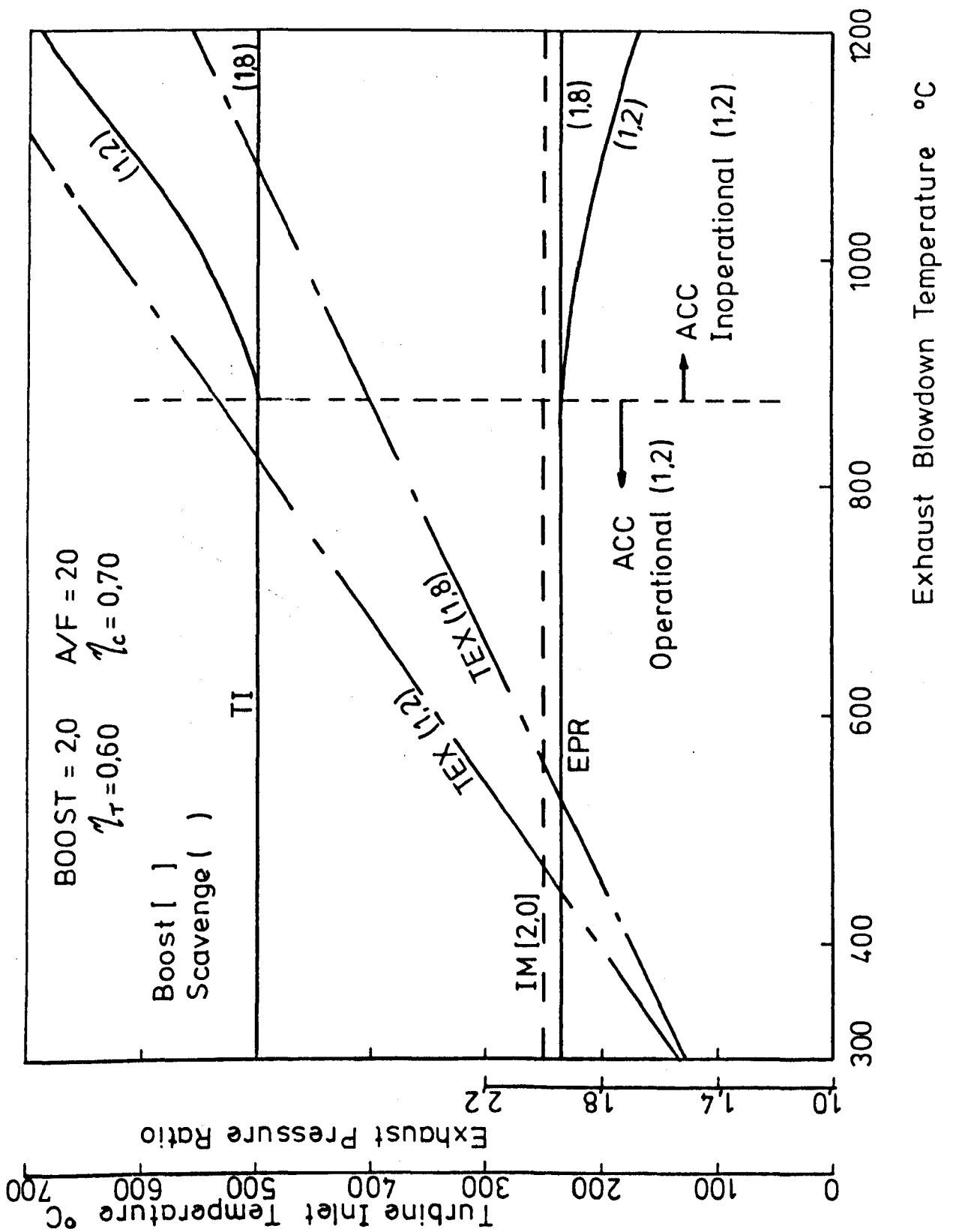
(Aid to Matching)

BOOST = 2.0

$\eta_T = 0.60$

$\eta_c = 0.40$

FIGURE 8.46



PRELIMINARY MATCHING PROGRAM

(Aid to Matching)

BOOST = 2.0

$\eta_T = 0.60$

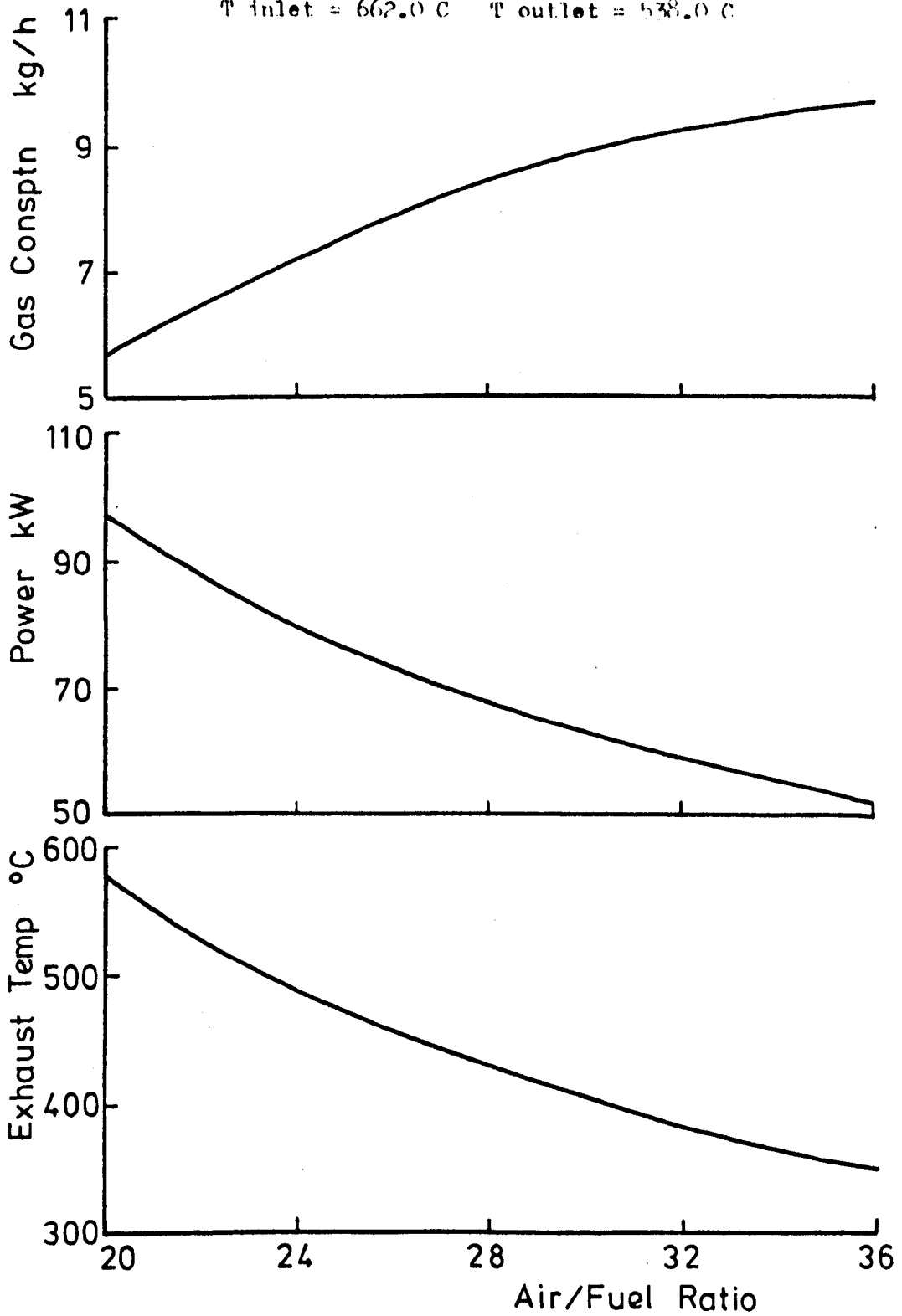
$\eta_c = 0.70$

FIGURE 8.47

Boost = 1.91 Delp = 0.06 bar

Scavenge = 1.4 Prest = 41.0 mm H₂O

T inlet = 662.0 C T outlet = 538.0 C



THEORETICAL PREDICTIONS

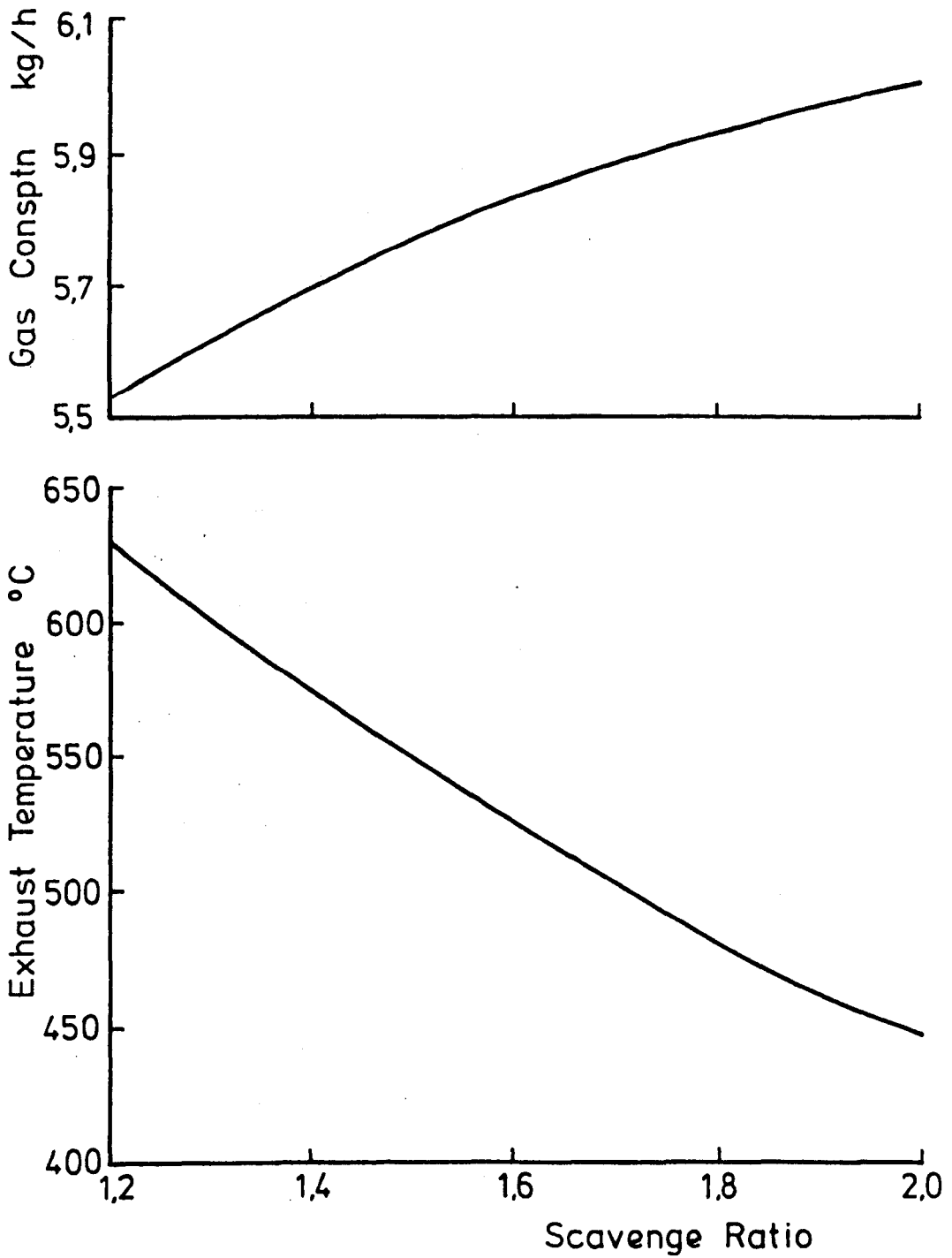
(Diesel Gas Turbine Program)

FIGURE 8,48

A/F = 20 Boost = 1.91

Delp = 0.06 bar P_{rest} = 41.0 mm H₂O

T_{inlet} = 662.0 C T_{outlet} = 538.0 C



THEORETICAL PREDICTIONS

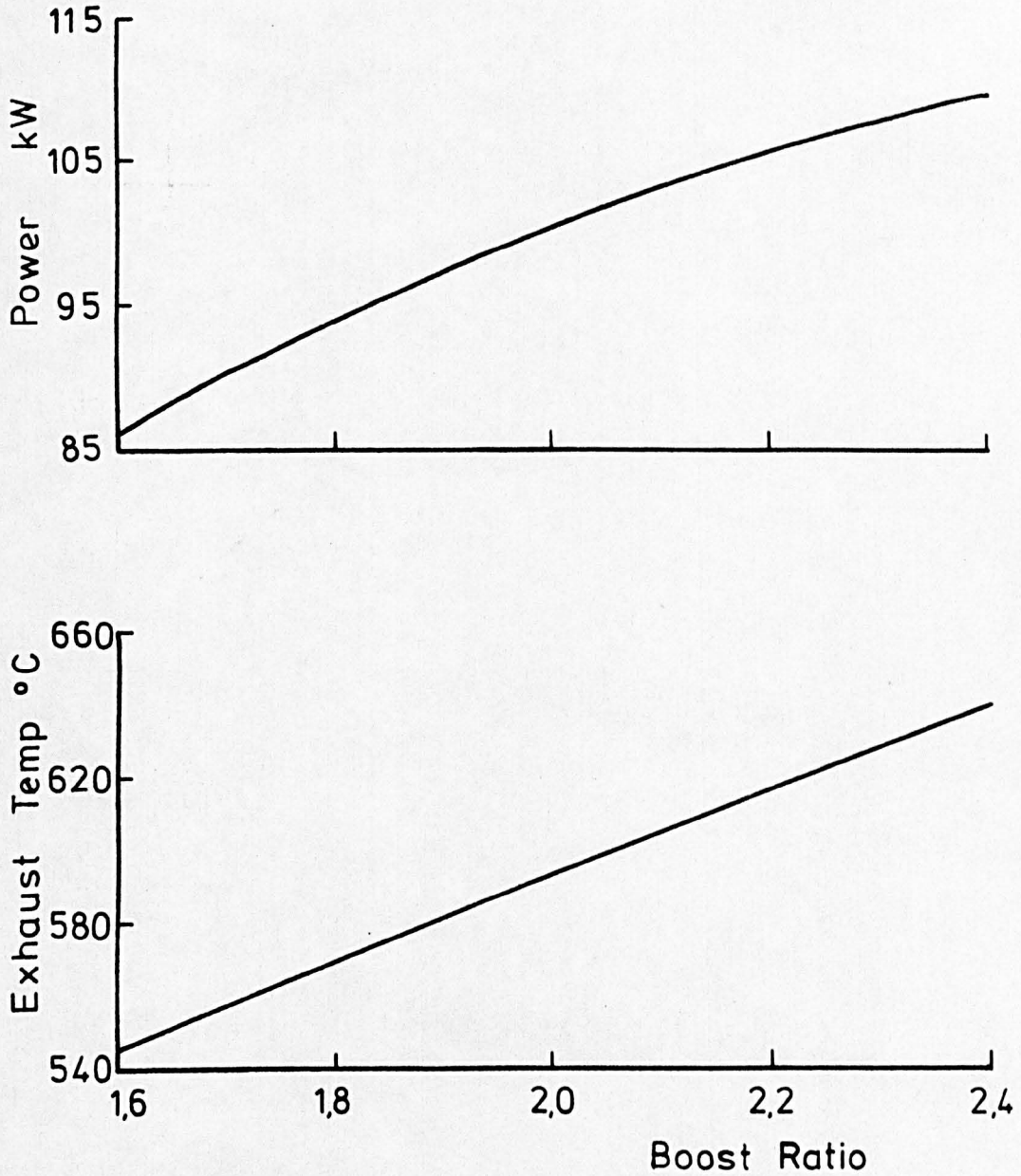
(Diesel Gas Turbine Program)

FIGURE 8.49

A/F = 20 Scavenge = 1.4

$P_{elp} = 0.06 \text{ bar}$ $P_{rest} = 41.0 \text{ mm H}_2\text{O}$

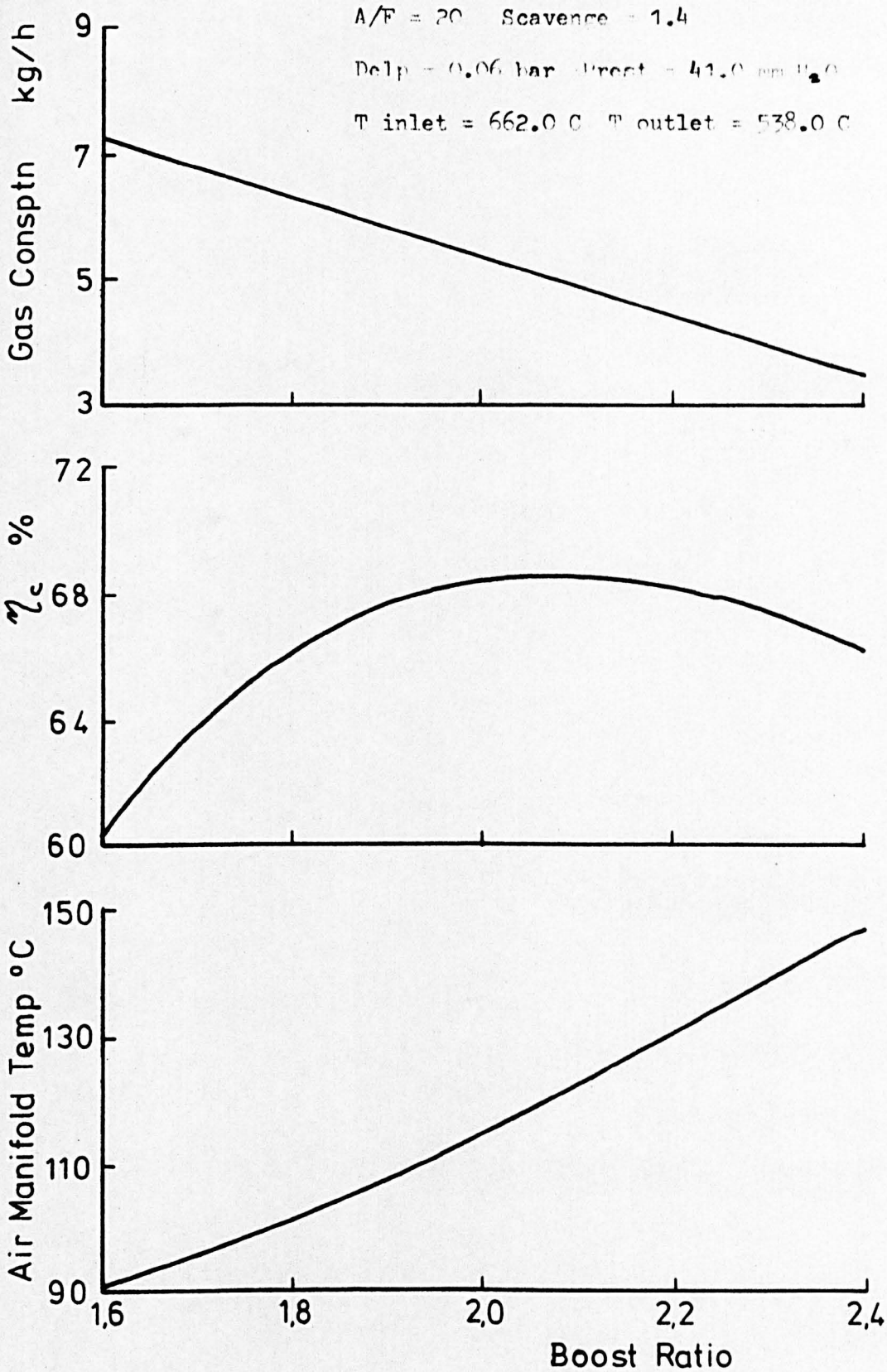
$T_{inlet} = 662.0 \text{ }^\circ\text{C}$ $T_{outlet} = 538.0 \text{ }^\circ\text{C}$



THEORETICAL PREDICTIONS

(Diesel Gas Turbine Program)

FIGURE 8.50



THEORETICAL PREDICTIONS

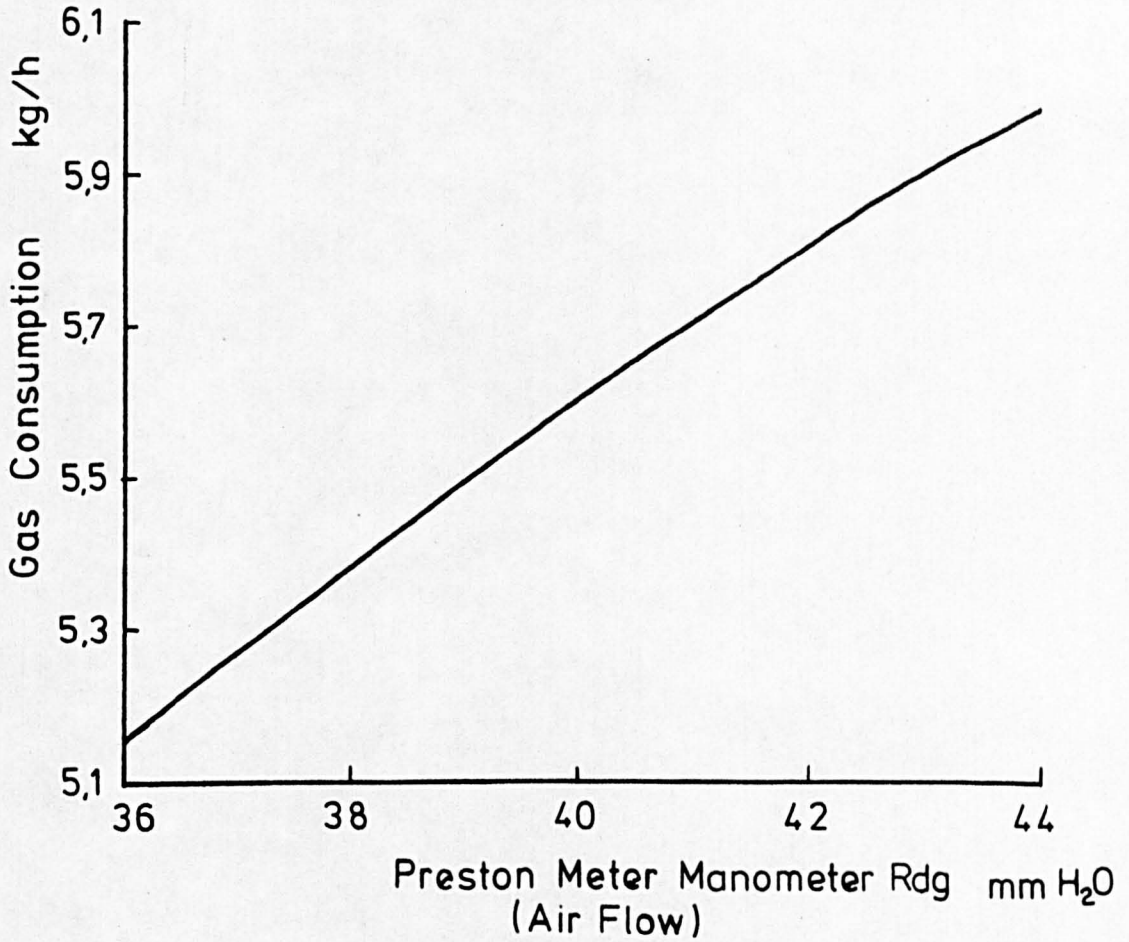
(Diesel Gas Turbine Program)

FIGURE 8.51

A/F = 20 Boost = 1.91

Scavenge = 1.4 Delp = 0.06 bar

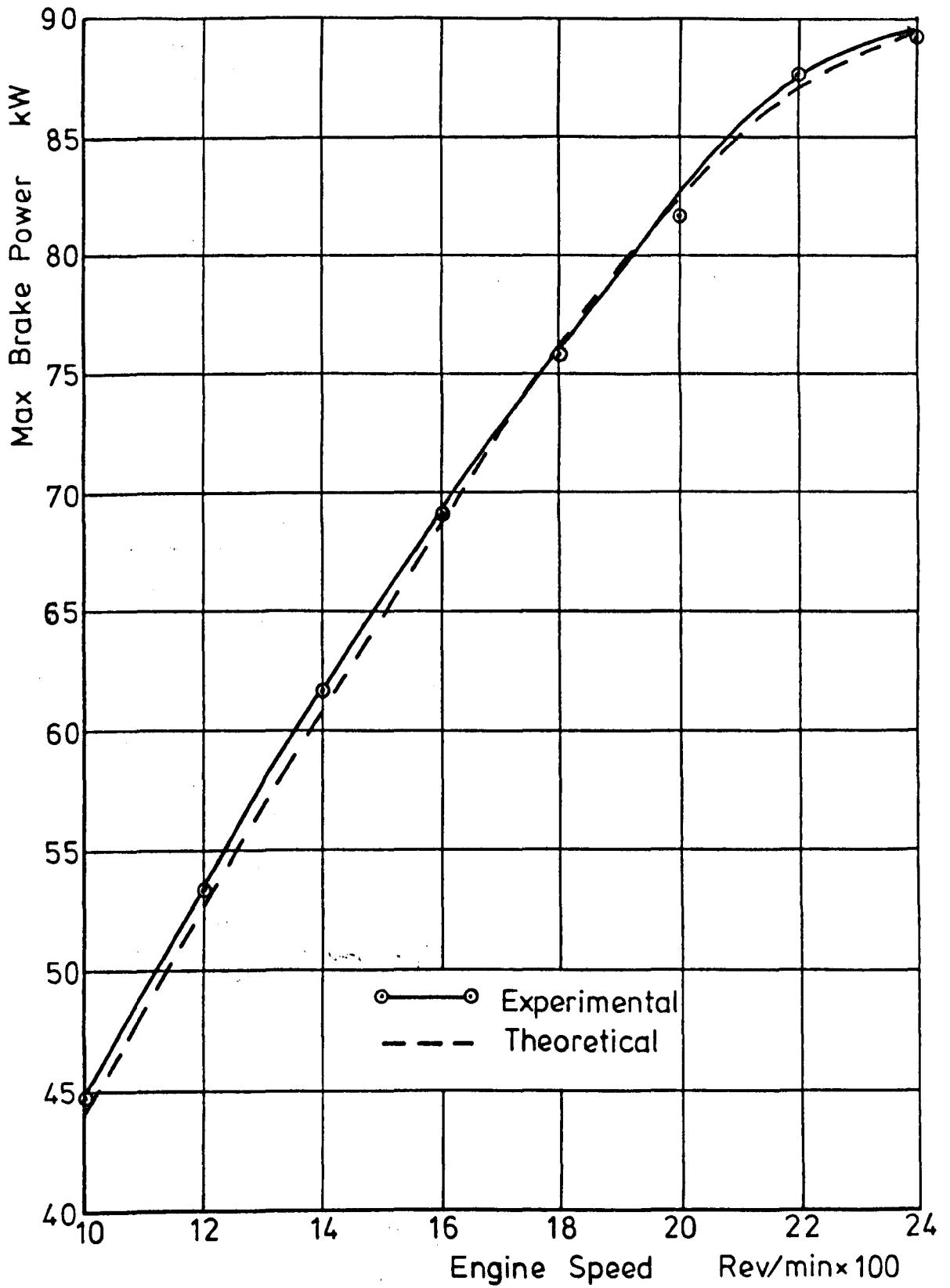
T inlet = 662.0 C T outlet = 538.0 C



THEORETICAL PREDICTIONS

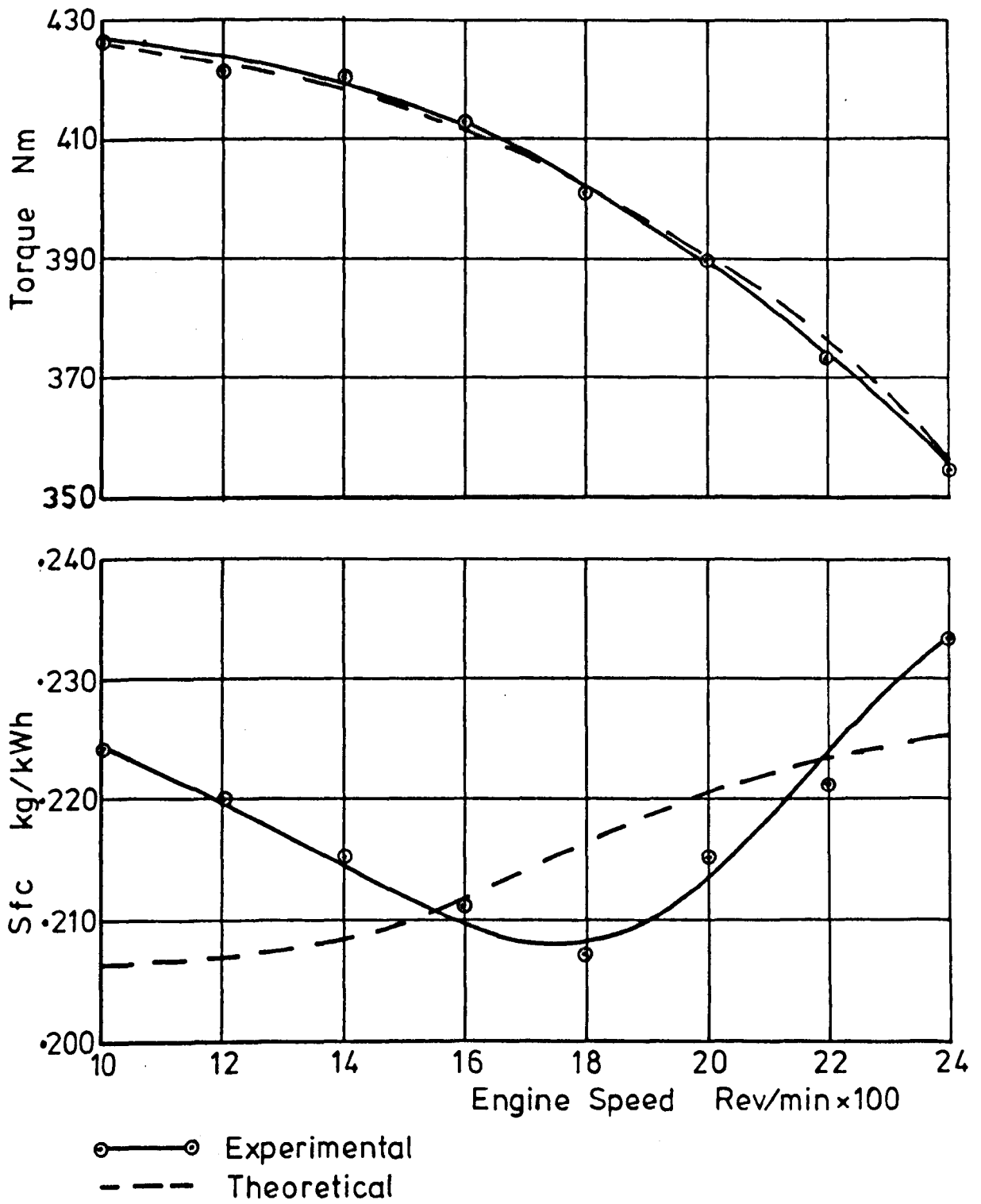
(Diesel Gas Turbine Program)

FIGURE 8.52



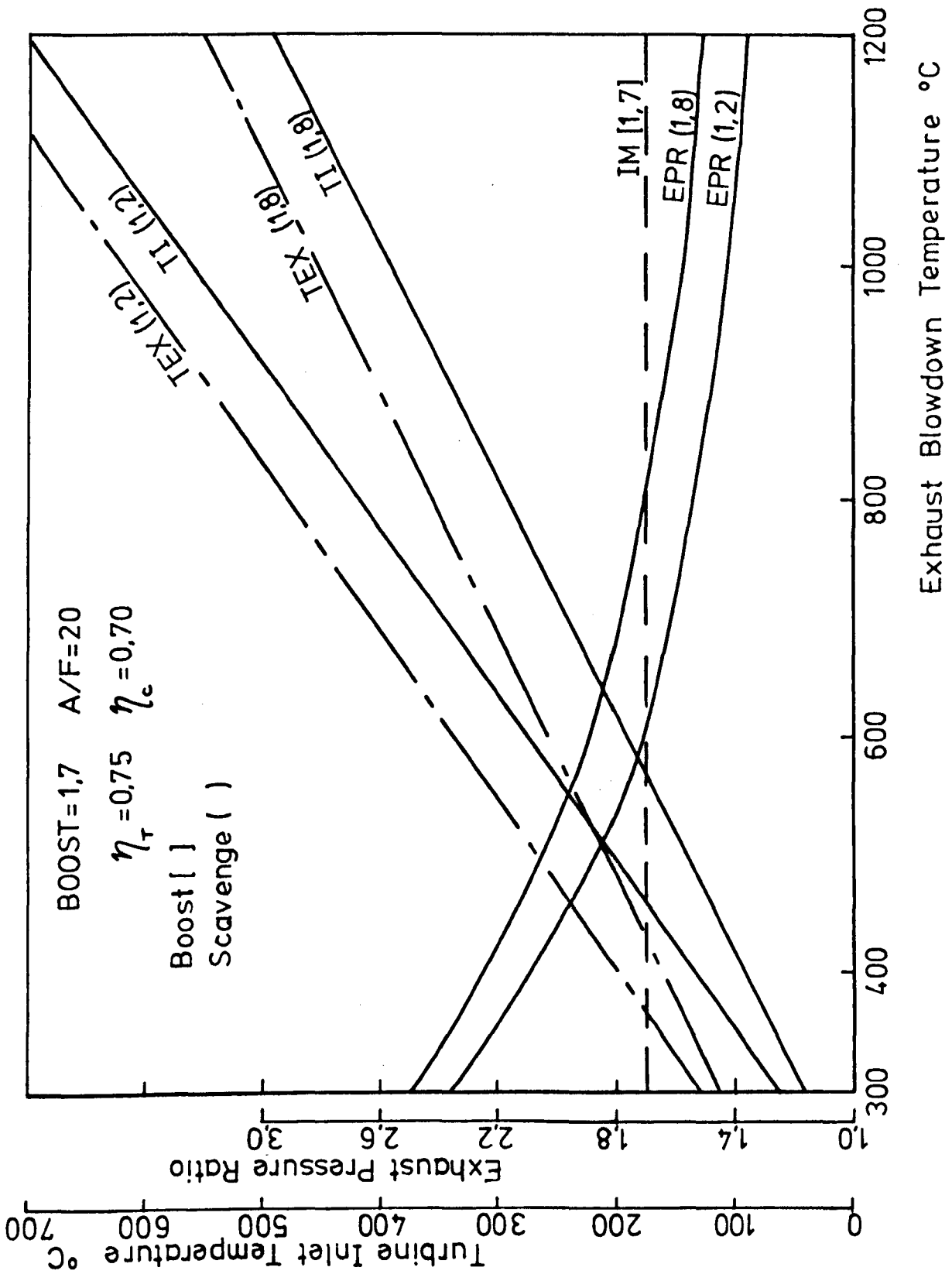
STANDARD ENGINE PERFORMANCE

FIGURE 8.53



STANDARD ENGINE PERFORMANCE

FIGURE 8.54



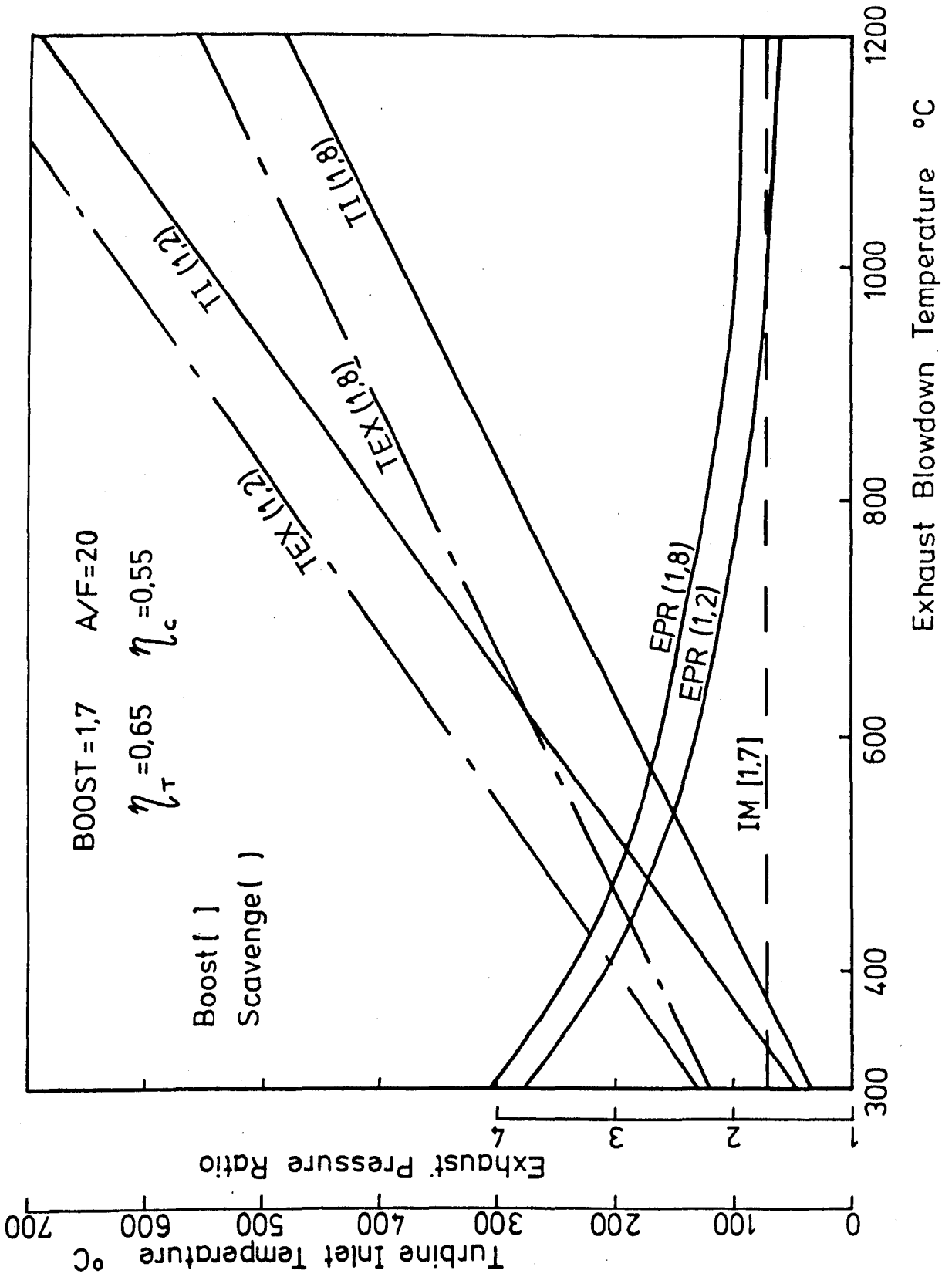
PRELIMINARY MATCHING PROGRAM

BOOST = 1,7

$\eta_{\tau} = 0,75$

$\eta_c = 0,70$

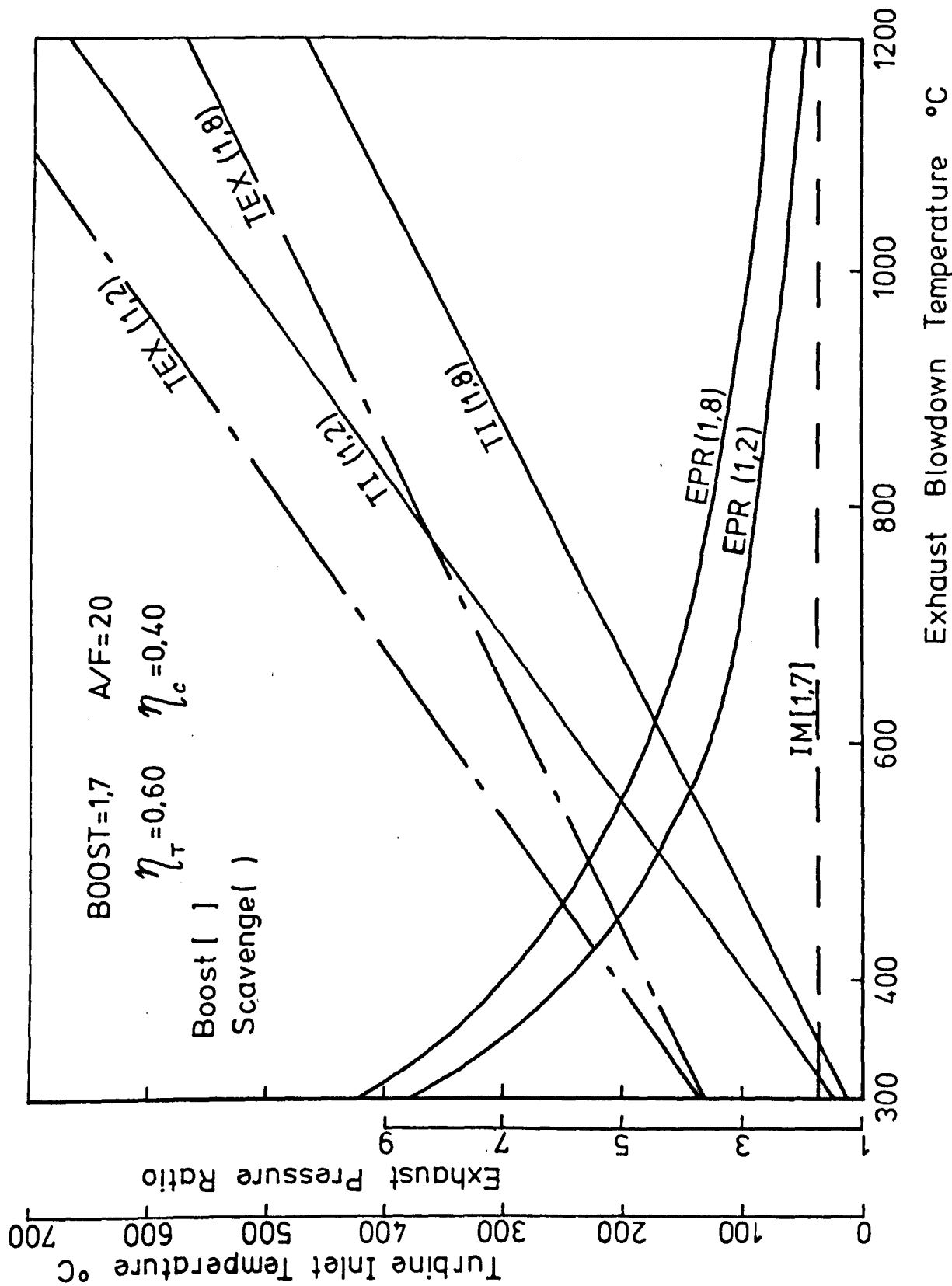
FIGURE 8.55



PRELIMINARY MATCHING PROGRAM

BOOST=1,7
 $\eta_{\tau} = 0,65$
 $\eta_c = 0,55$

FIGURE 8,56



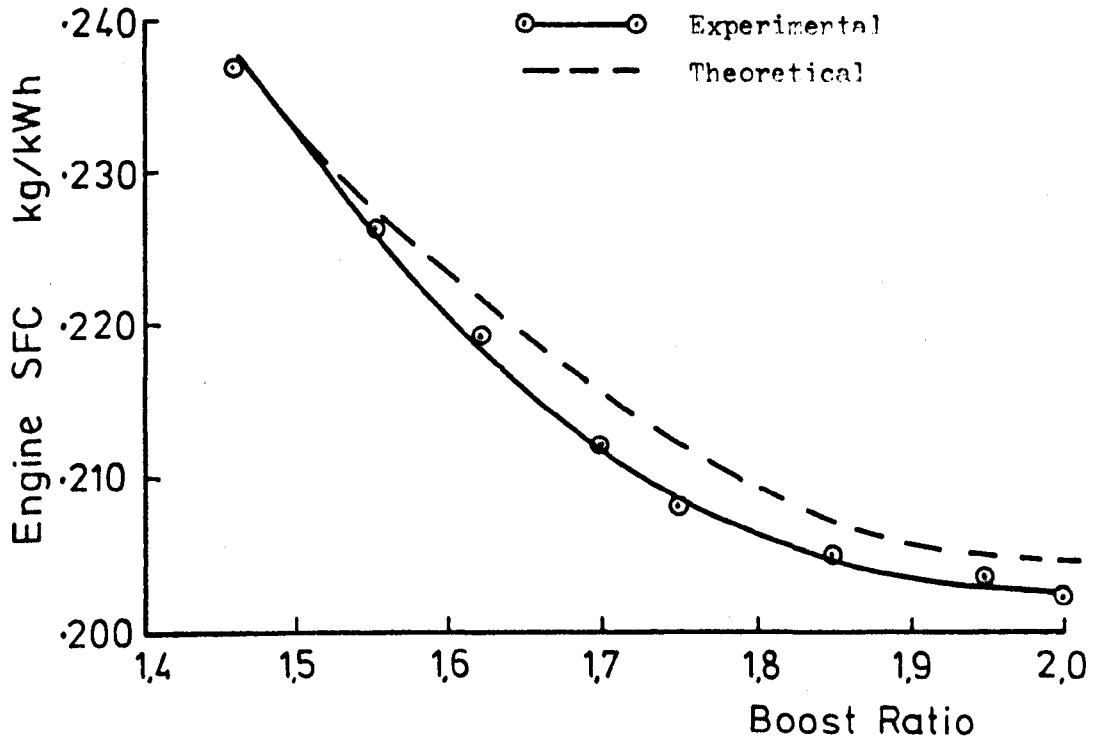
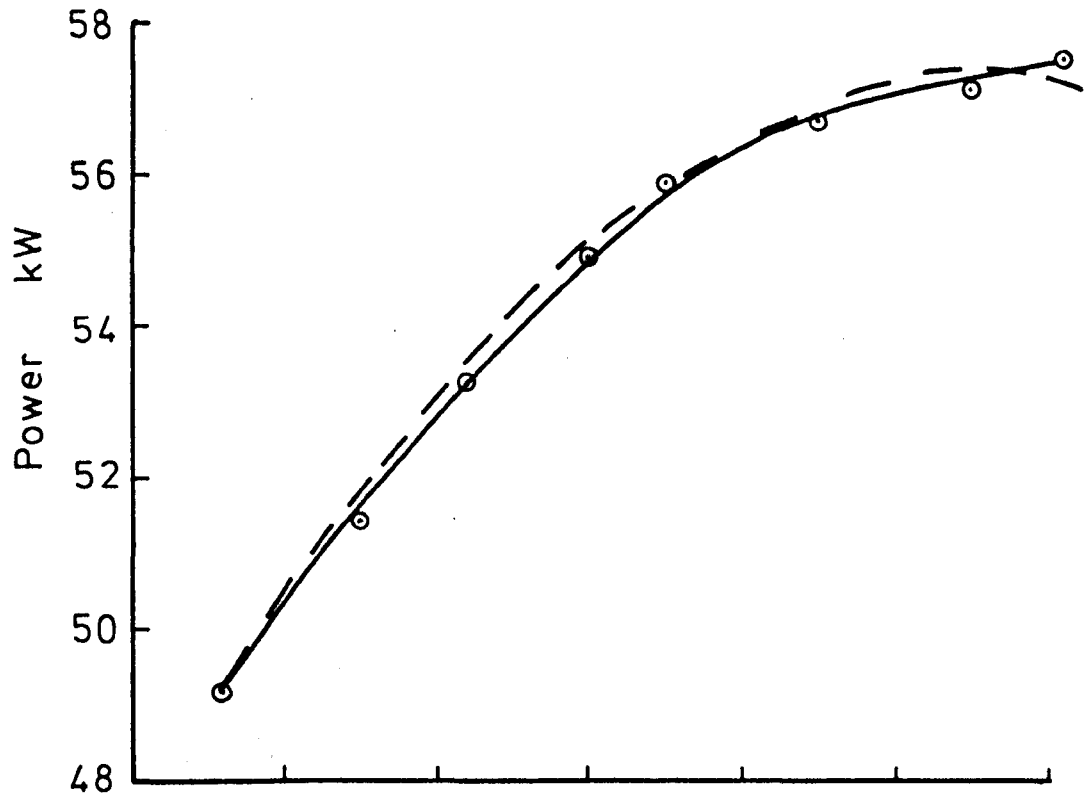
PRELIMINARY MATCHING PROGRAM

BOOST = 1,7

$\eta_{\tau} = 0,60$

$\eta_c = 0,40$

FIGURE 8.57

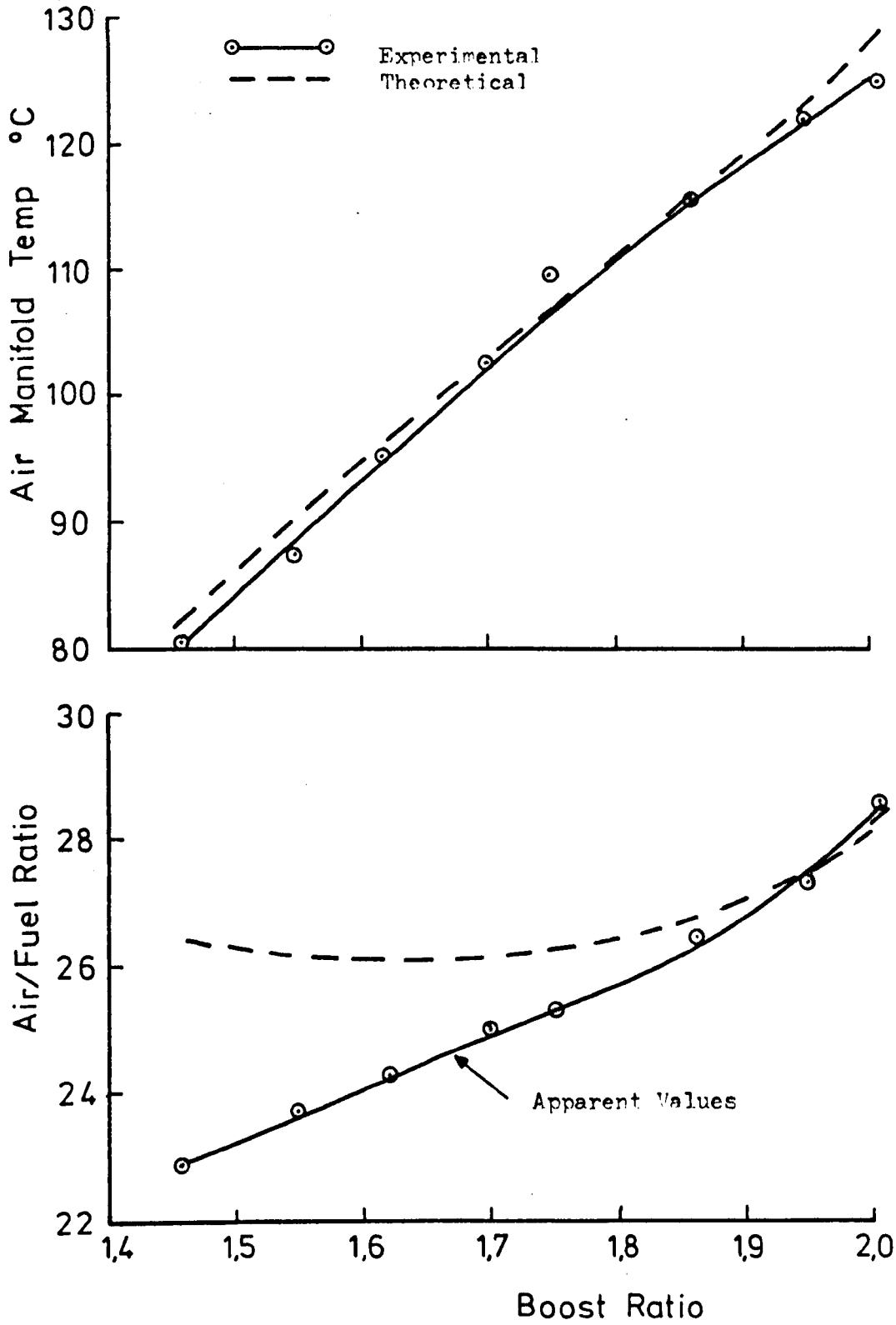


EXPERIMENTAL & THEORETICAL COMPARISON

HOLSET 3LD1 1.57

(Engine Speed=1200 Rev/min)

FIGURE 8,58

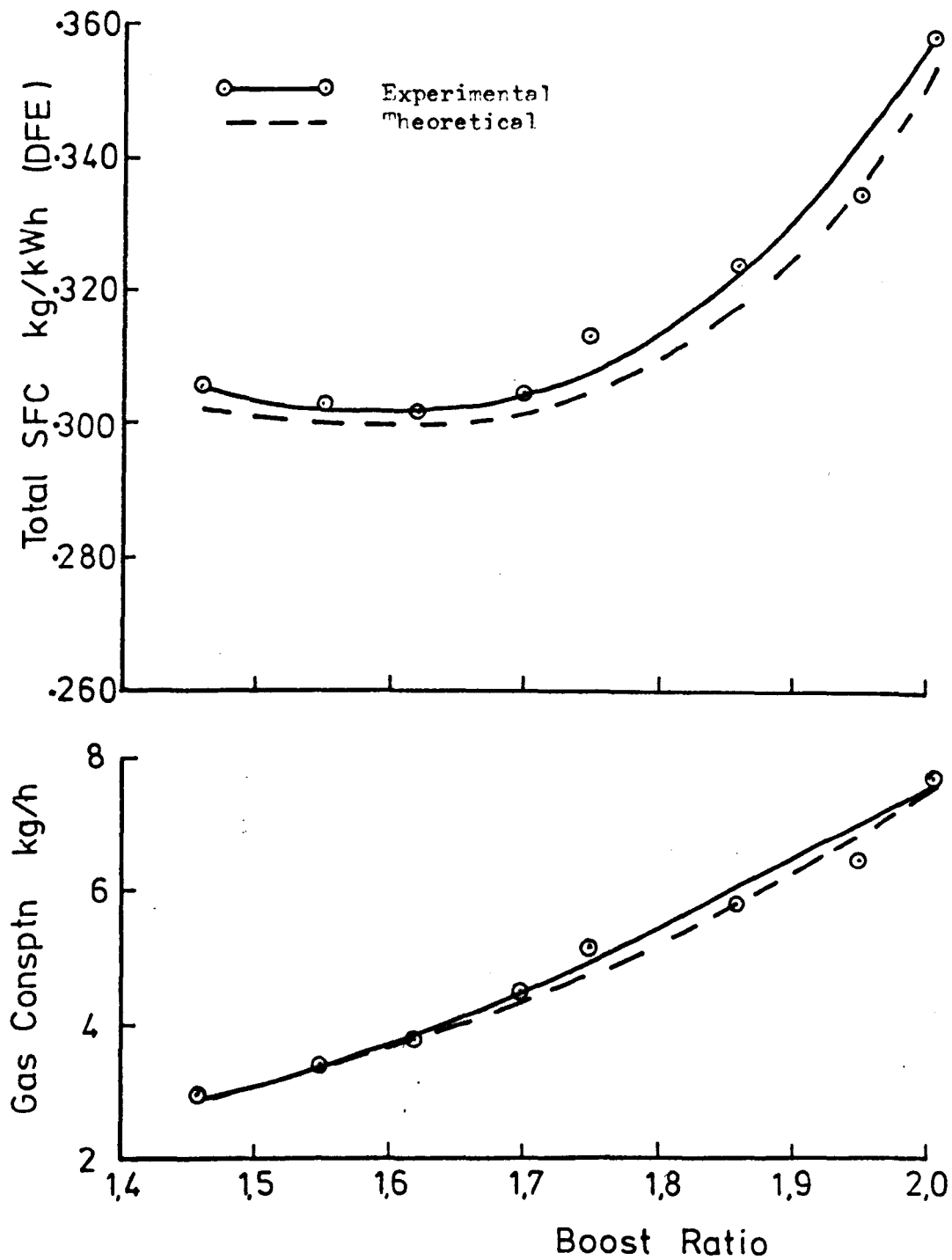


EXPERIMENTAL & THEORETICAL COMPARISON

HOLSET 3LD1 1,57

(Engine Speed=1200 Rev/min)

FIGURE 8.59

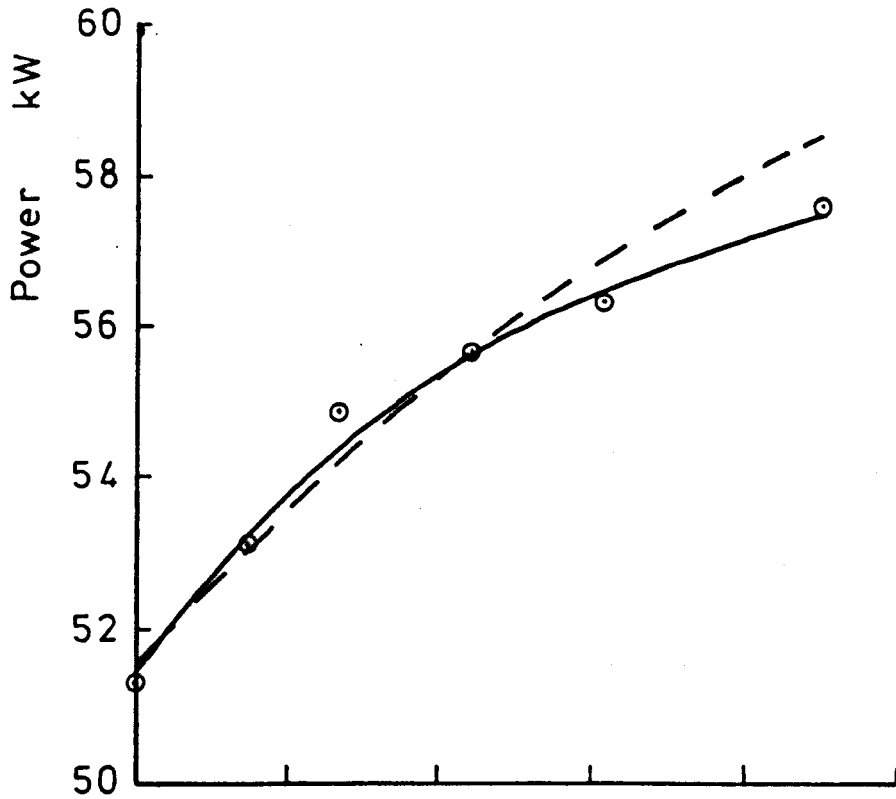


EXPERIMENTAL & THEORETICAL COMPARISON

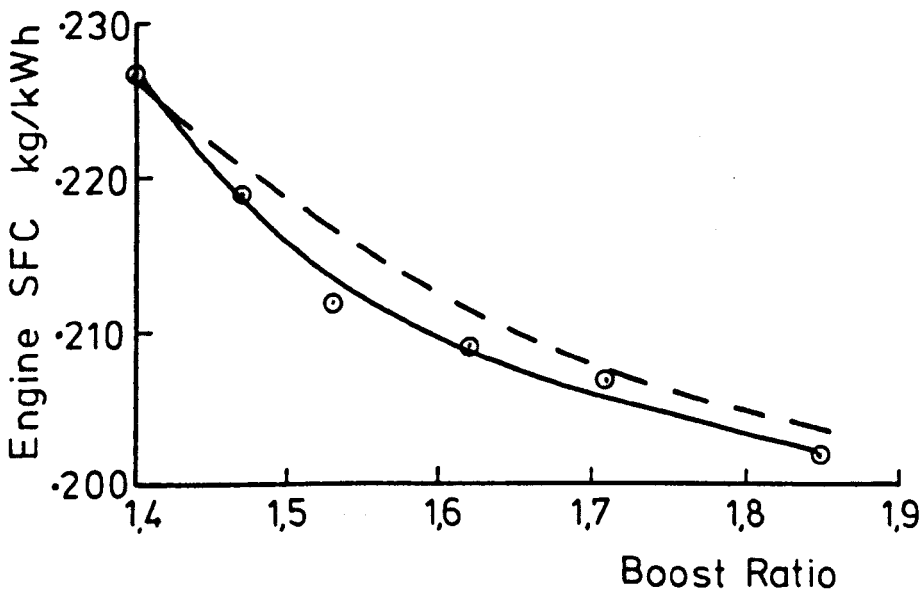
HOLSET 3LD1 157

(Engine Speed=1200 Rev/min)

FIGURE 8.60



○—○ Experimental
- - - Theoretical

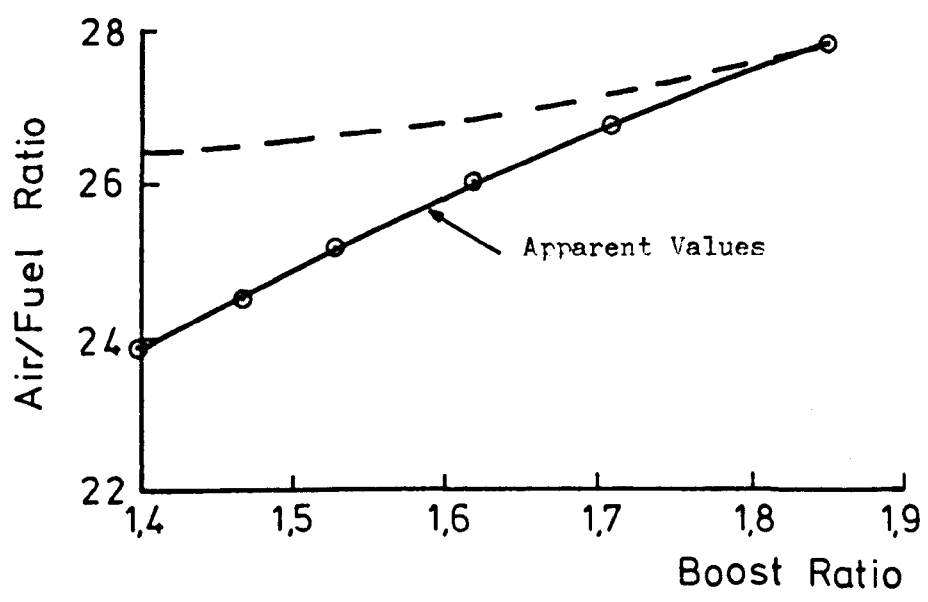
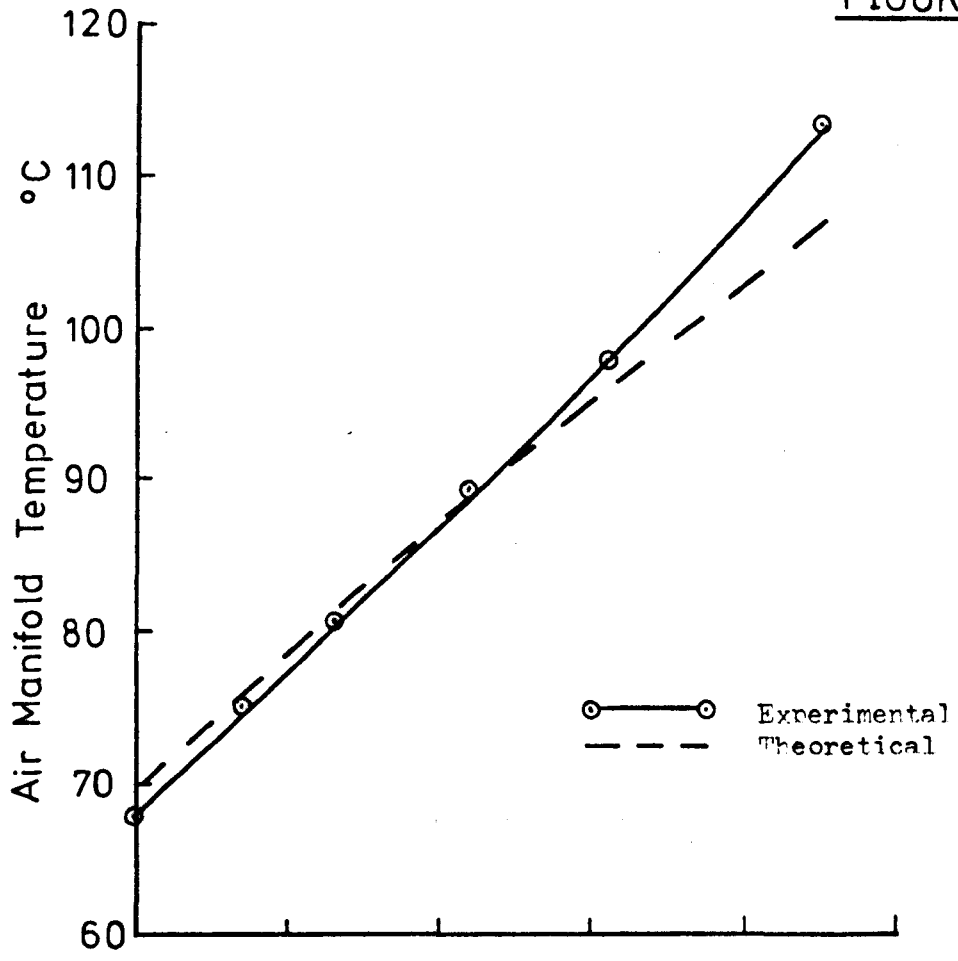


EXPERIMENTAL & THEORETICAL COMPARISON

HOLSET 3LD1 2.5

(Engine Speed=1200 Rev/min)

FIGURE 8.61

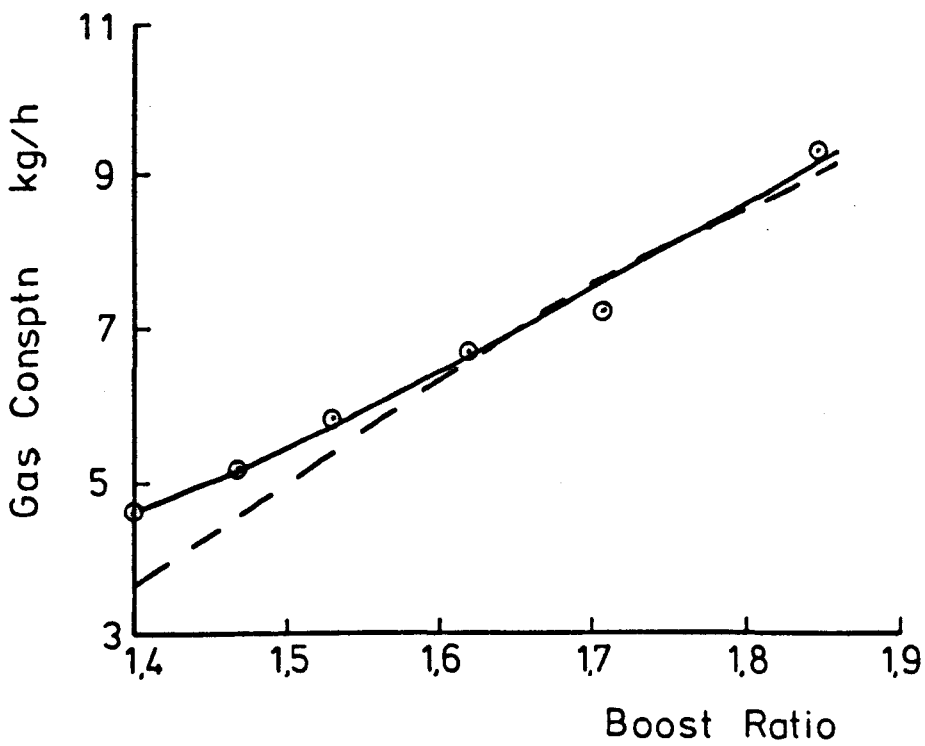
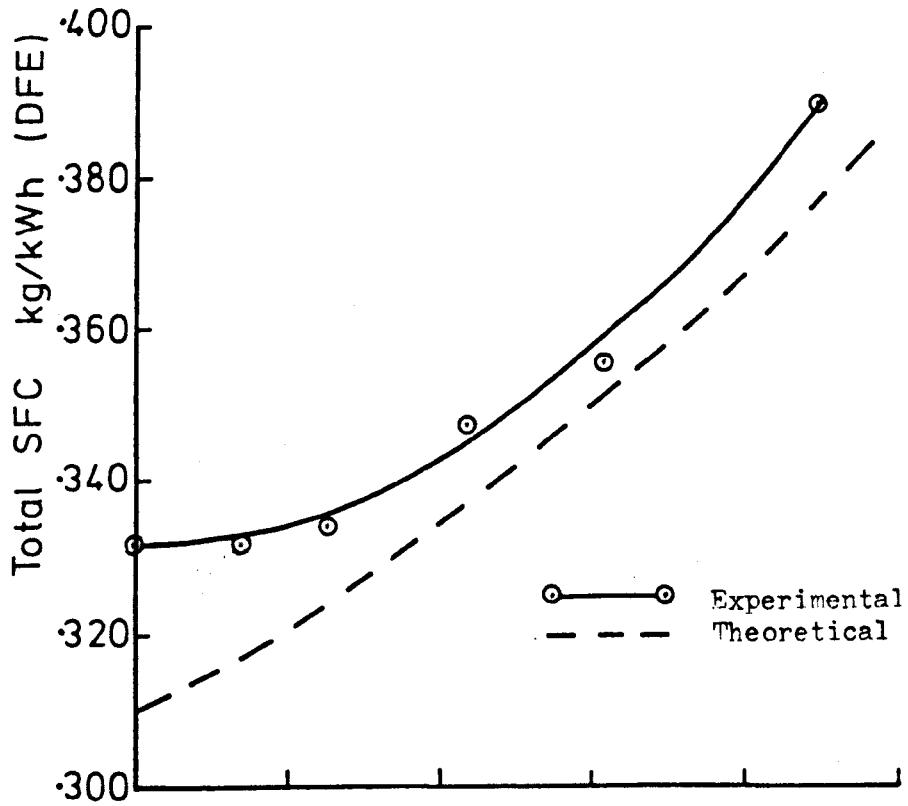


EXPERIMENTAL & THEORETICAL COMPARISON

HOLSET 3LD1 2.5

(Engine Speed = 1200 Rev/min)

FIGURE 8.62

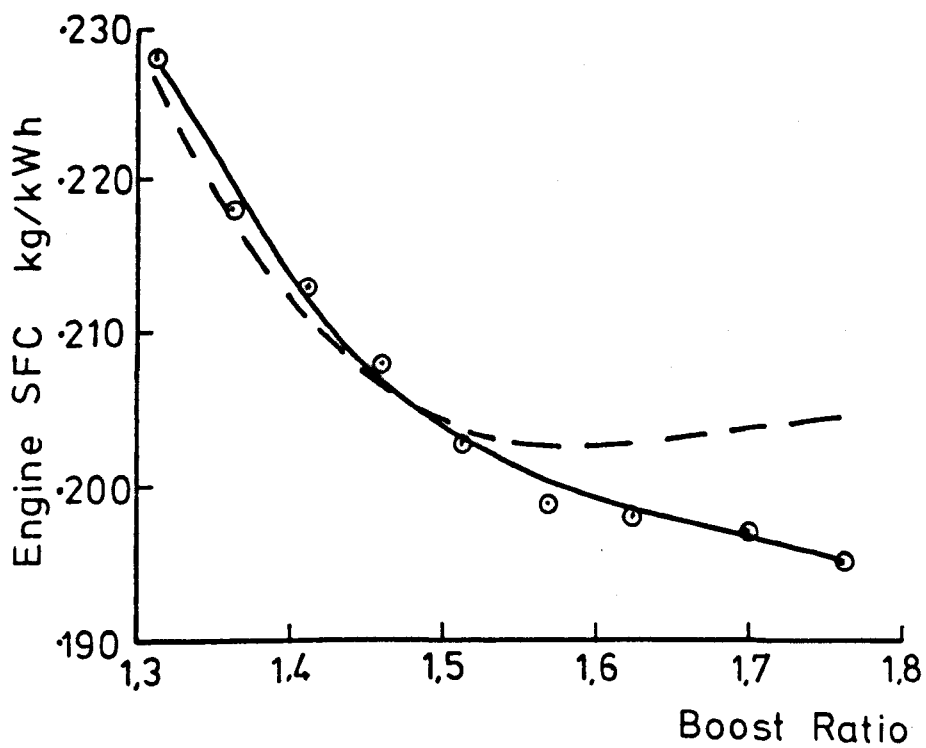
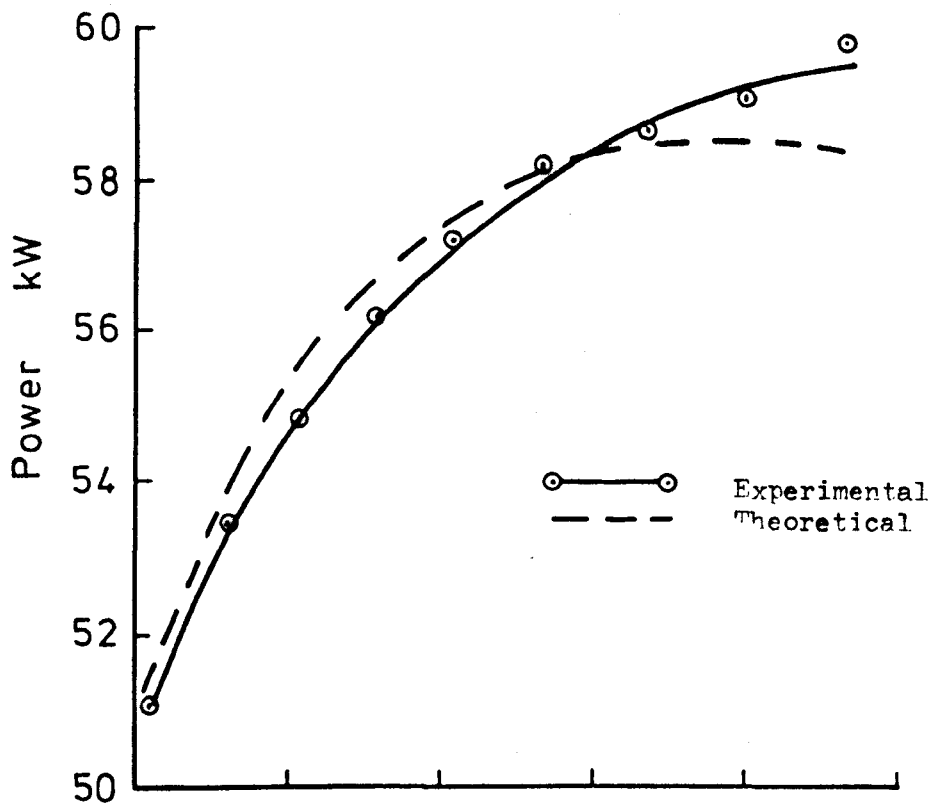


EXPERIMENTAL & THEORETICAL COMPARISON

HOLSET 3LD1 2.5

(Engine Speed = 1200 Rev/min)

FIGURE 8.63

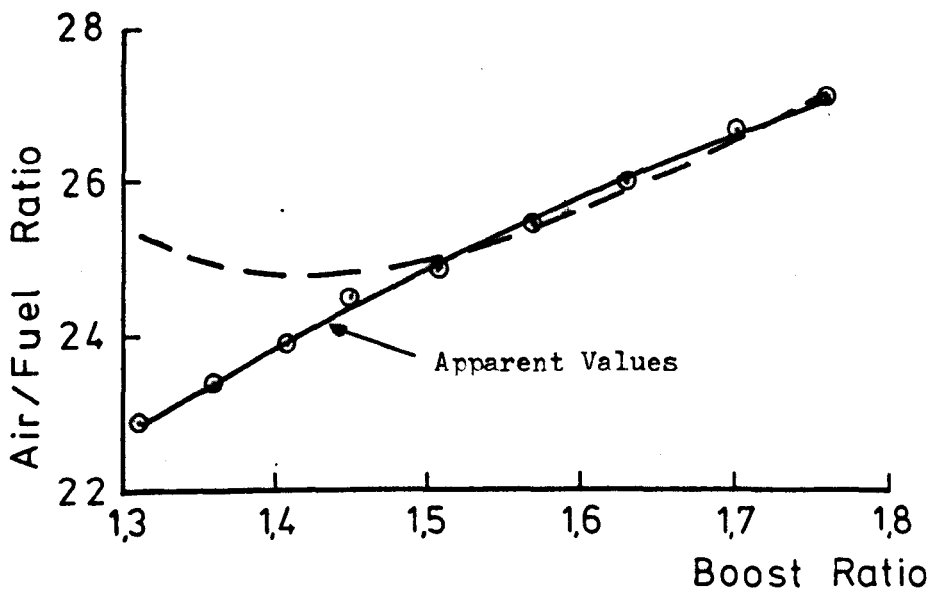
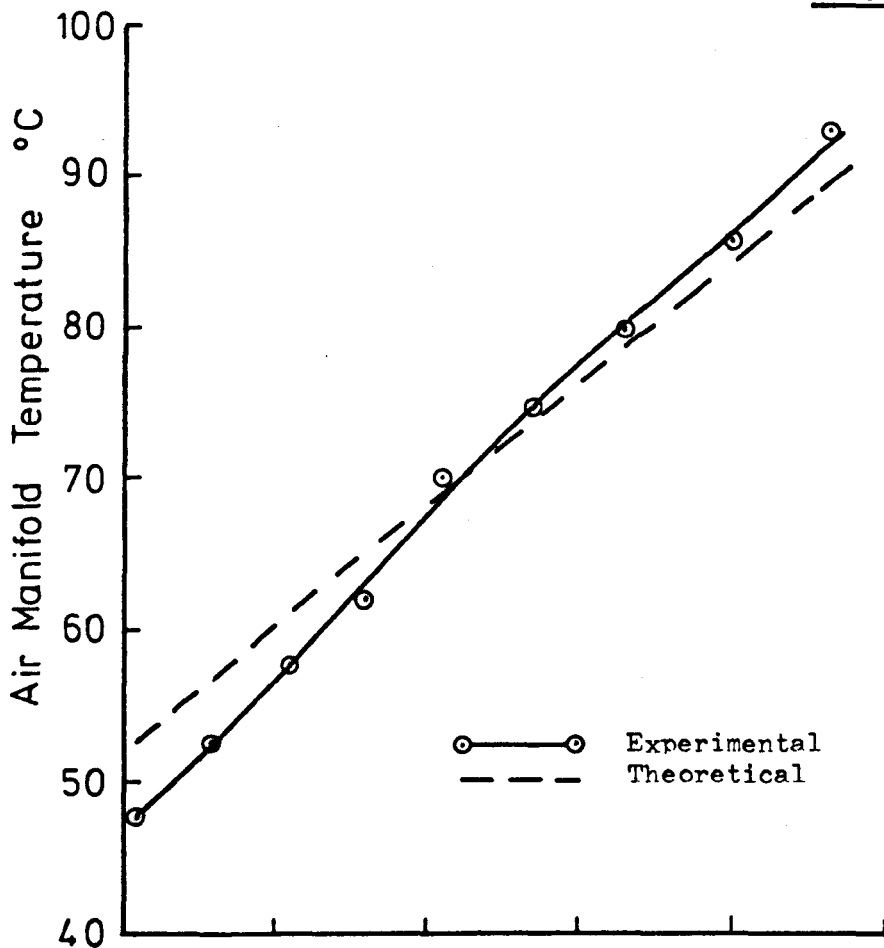


EXPERIMENTAL & THEORETICAL COMPARISON

HOLSET 4LGK 2.6T2

(Engine Speed = 1200 Rev/min)

FIGURE 8,64

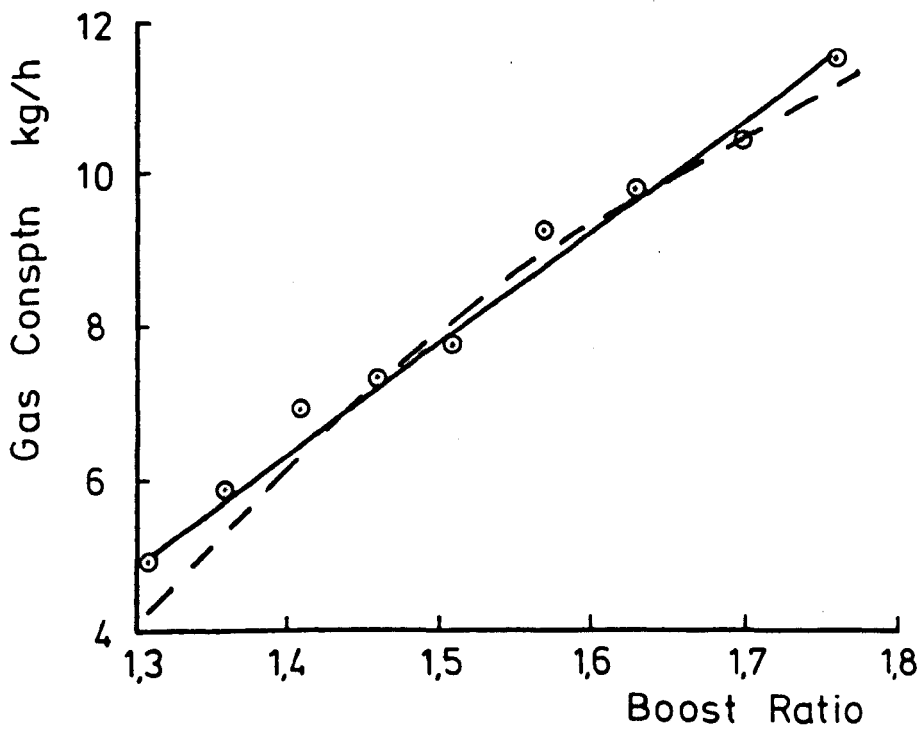
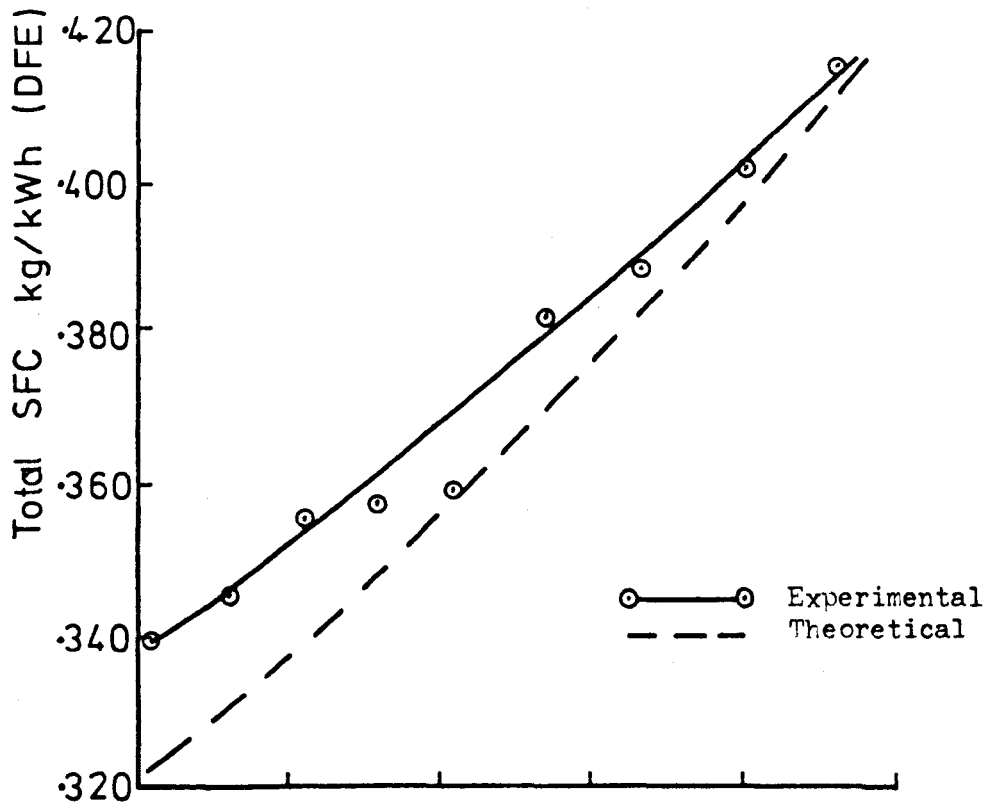


EXPERIMENTAL & THEORETICAL COMPARISON

HOLSET 4LGK 2,6T2

(Engine Speed=1200 Rev/min)

FIGURE 8.65

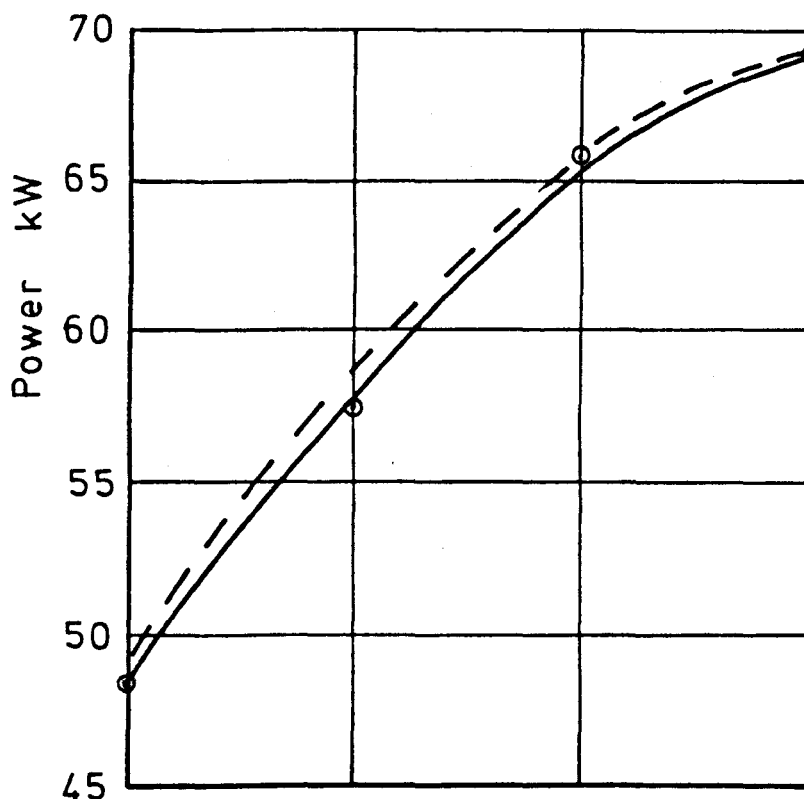


EXPERIMENTAL & THEORETICAL COMPARISON

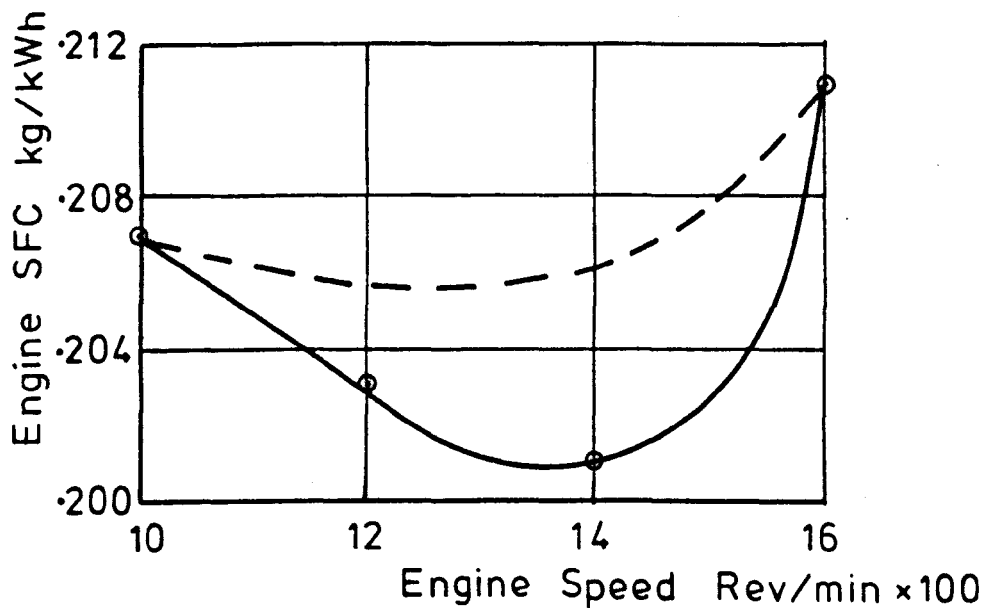
HOLSET 4LGK 2.6T2

(Engine Speed=1200 Rev/min)

FIGURE 8.66

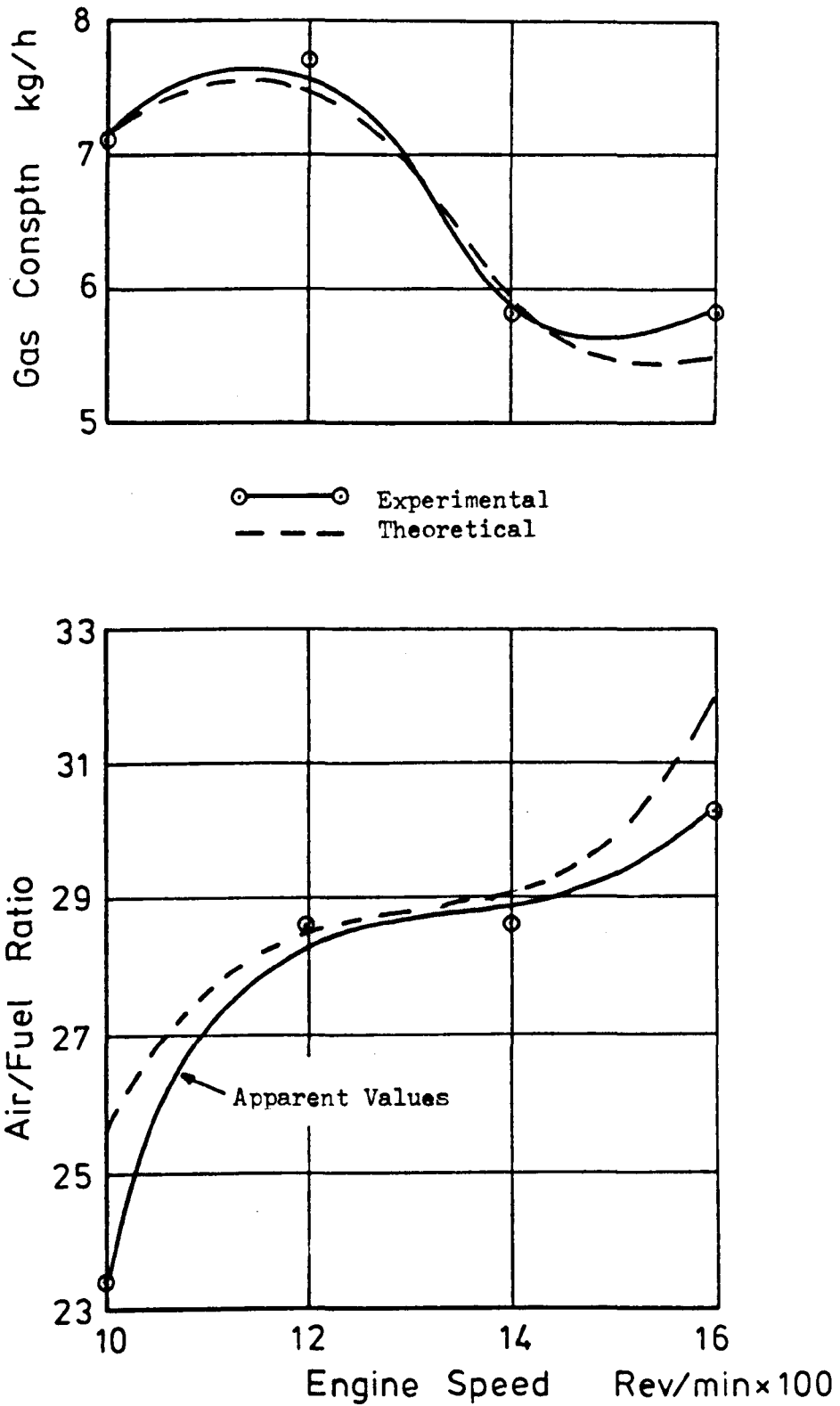


○—○ Experimental
- - - Theoretical



EXPERIMENTAL & THEORETICAL COMPARISON

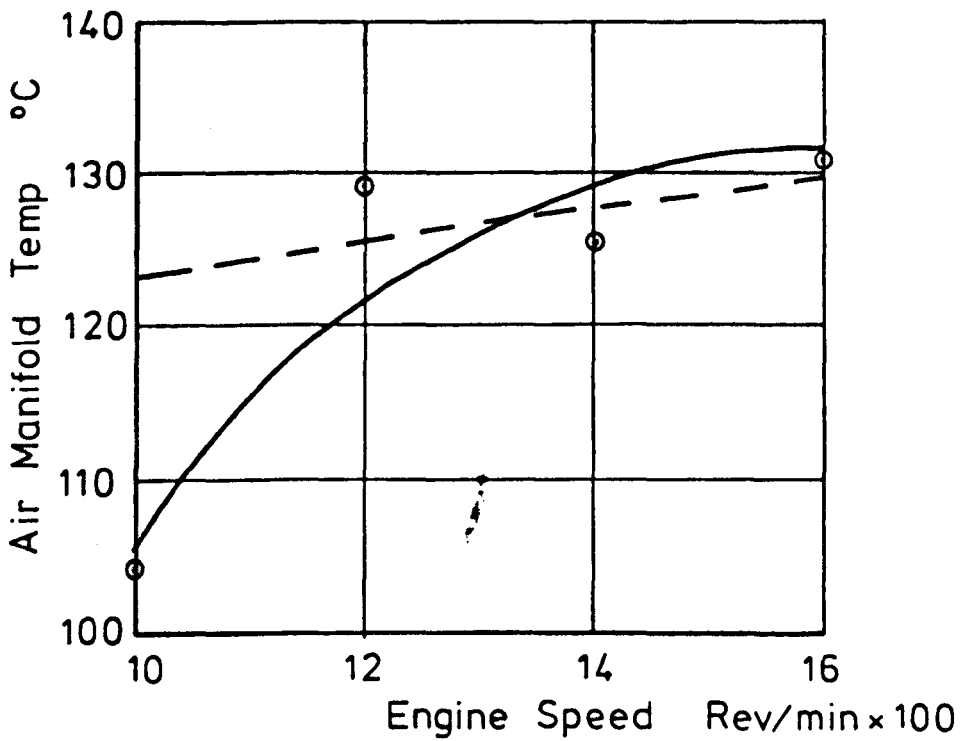
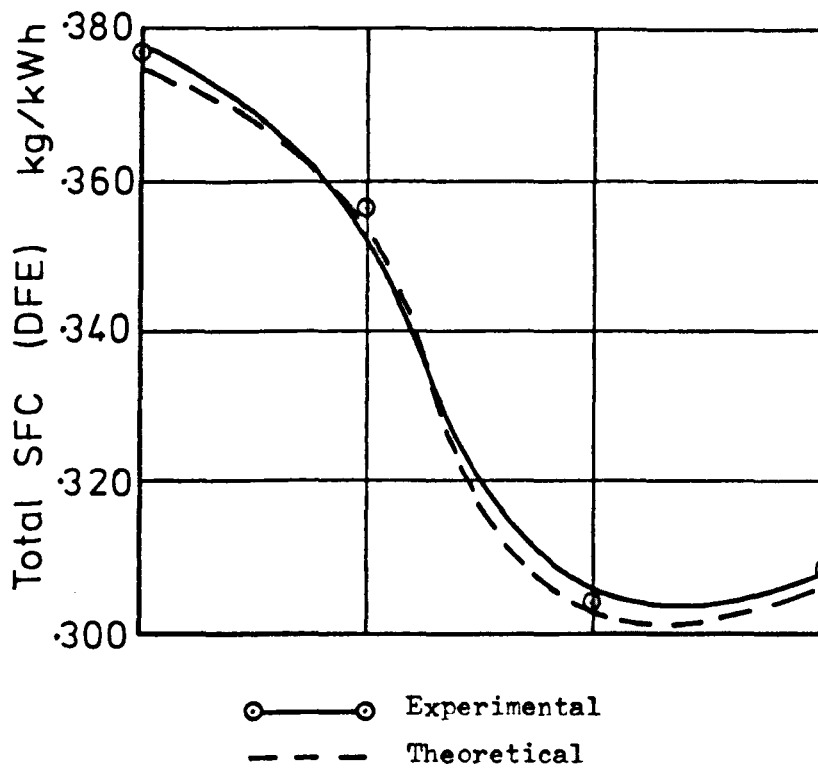
HOLSET 3LD1 1.57



EXPERIMENTAL & THEORETICAL COMPARISON

HOLSET 3LD1 1,57

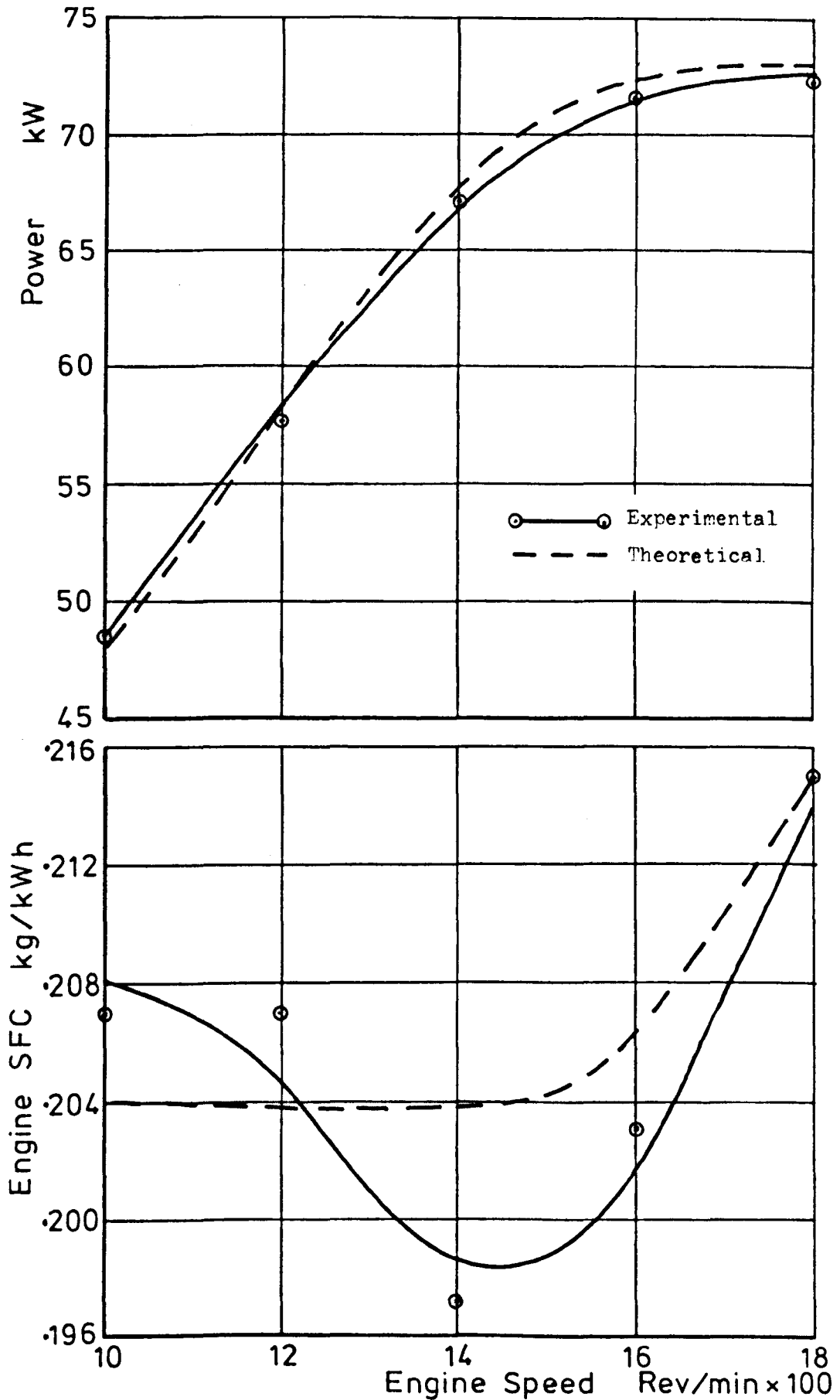
FIGURE 8.68



EXPERIMENTAL & THEORETICAL COMPARISON

HOLSET 3LD1 1,57

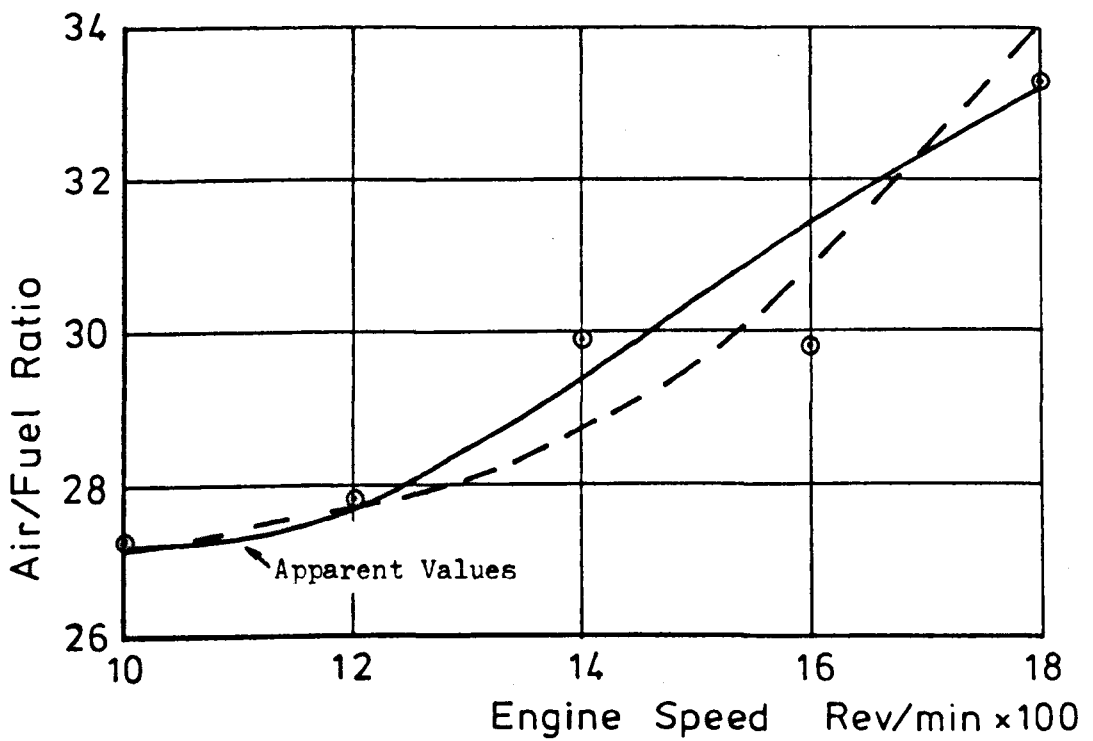
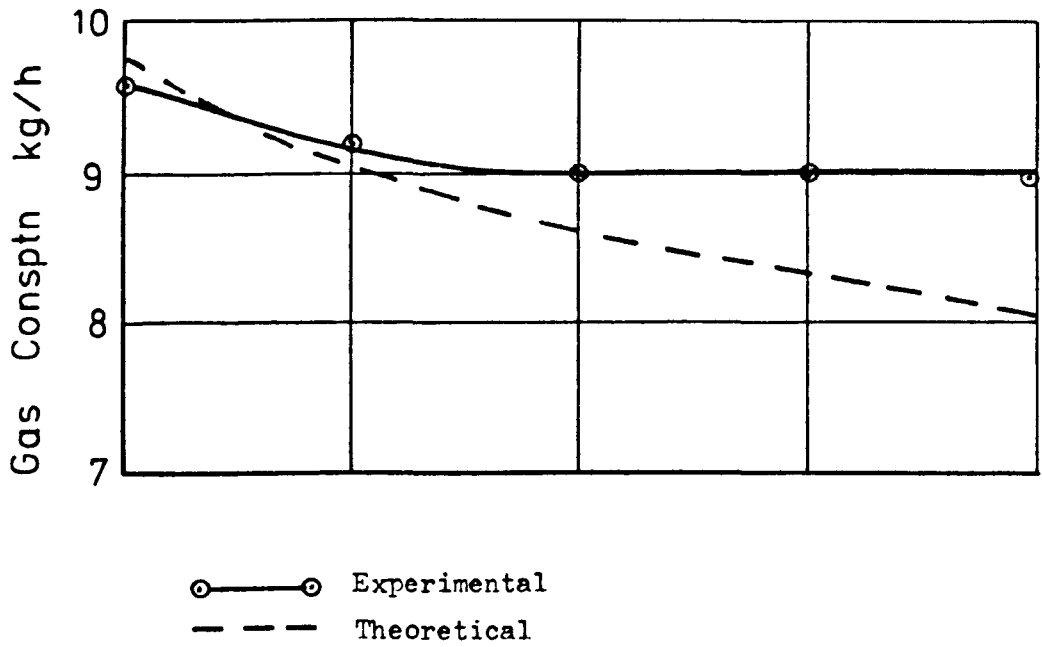
FIGURE 8.69



EXPERIMENTAL & THEORETICAL COMPARISON

HOLSET 3LD1 2.5

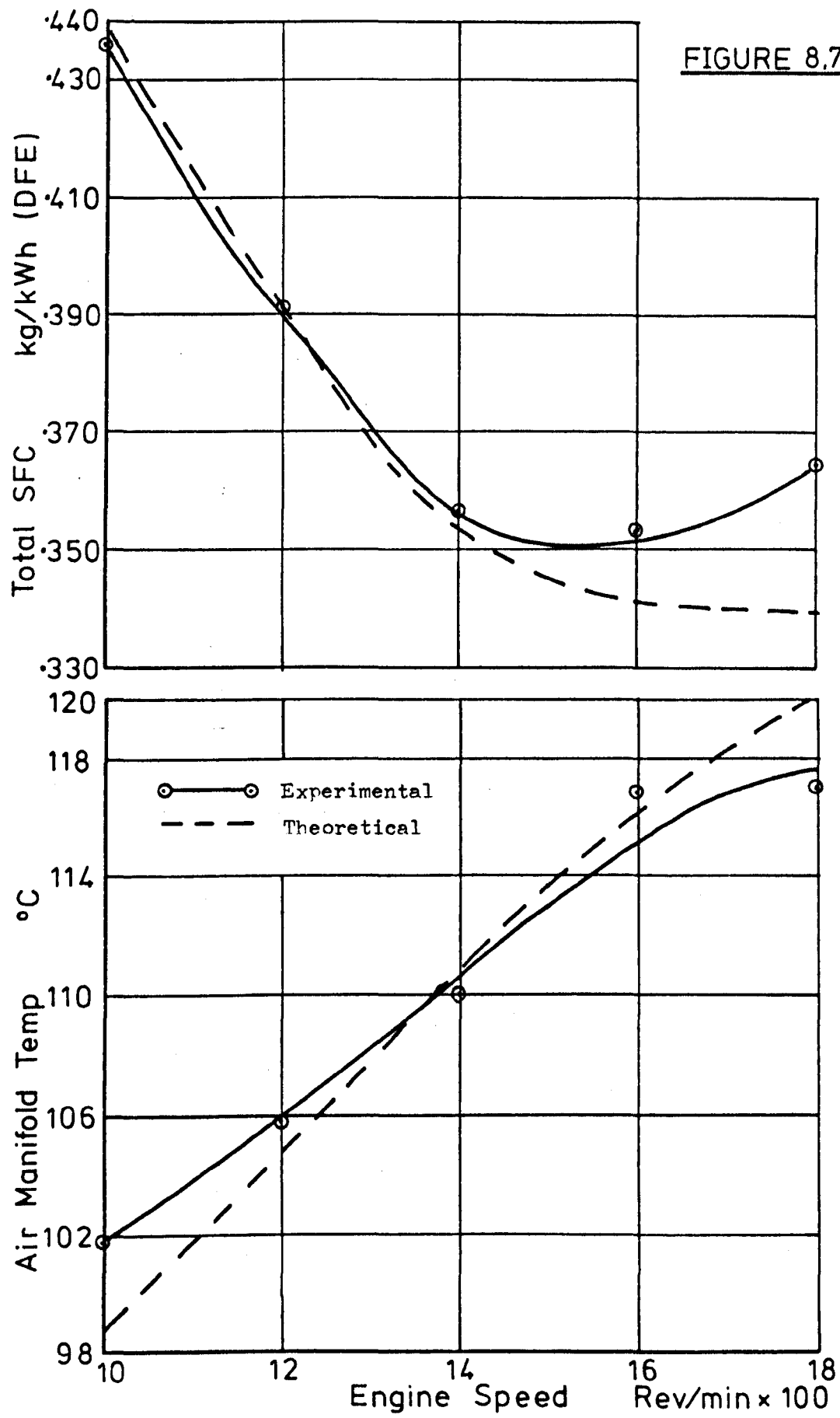
FIGURE 8.70



EXPERIMENTAL & THEORETICAL COMPARISON

HOLSET 3LD1 2,5

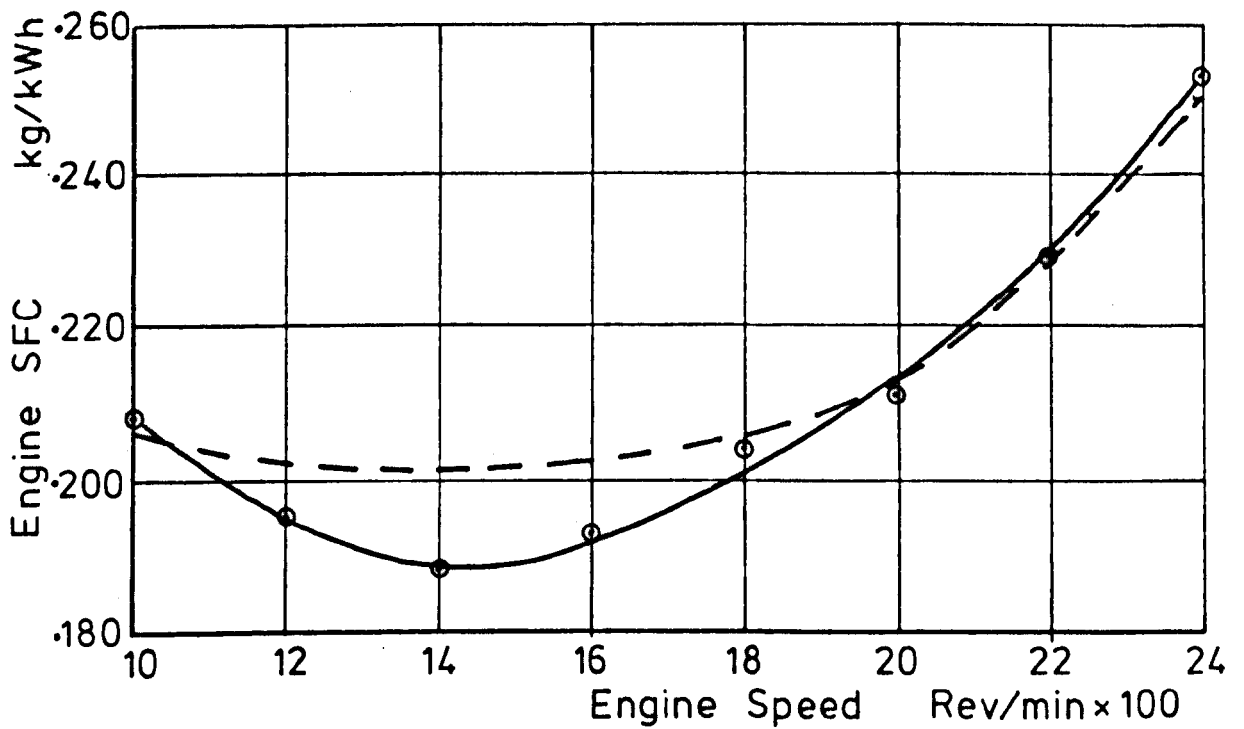
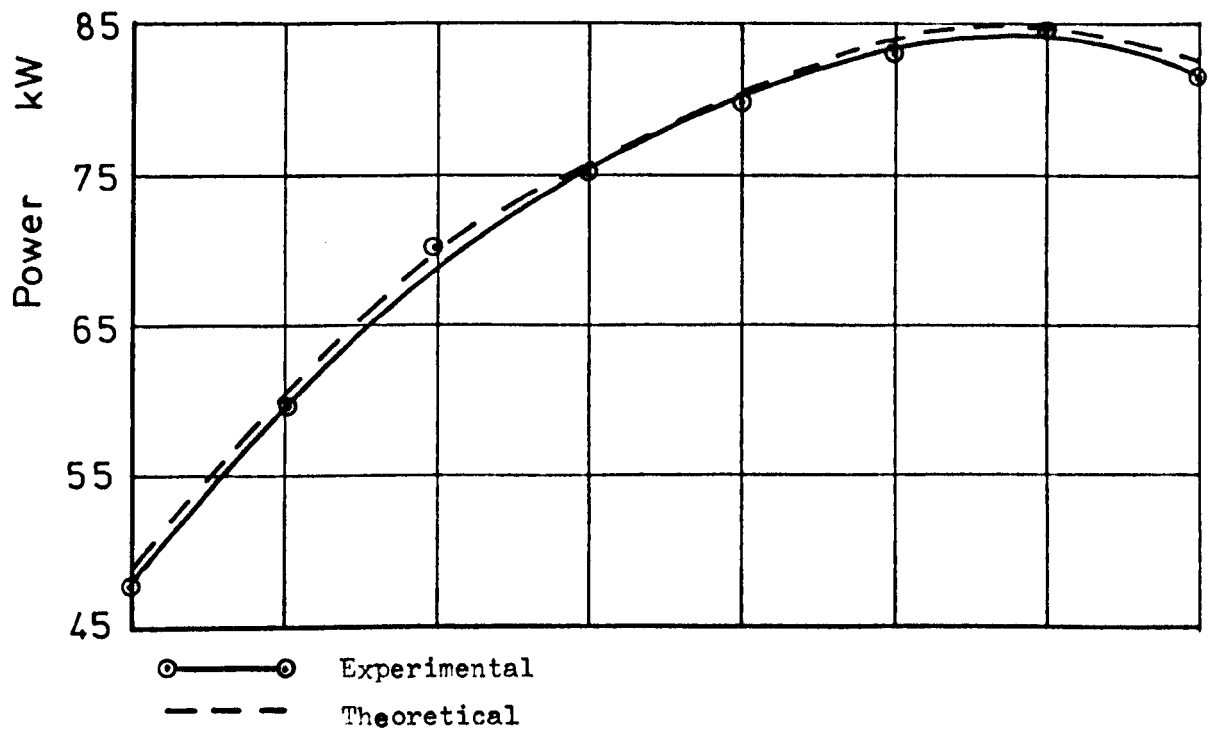
FIGURE 8.71



EXPERIMENTAL & THEORETICAL COMPARISON

HOLSET 3LD1 2.5

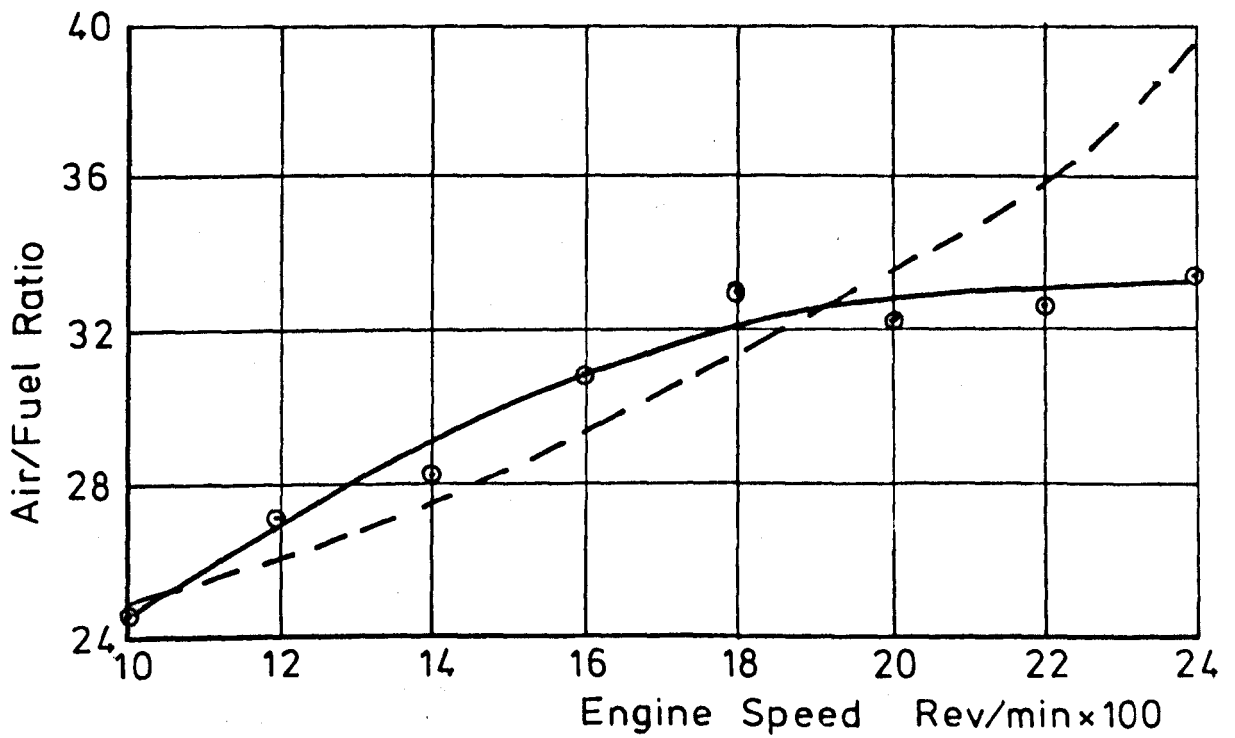
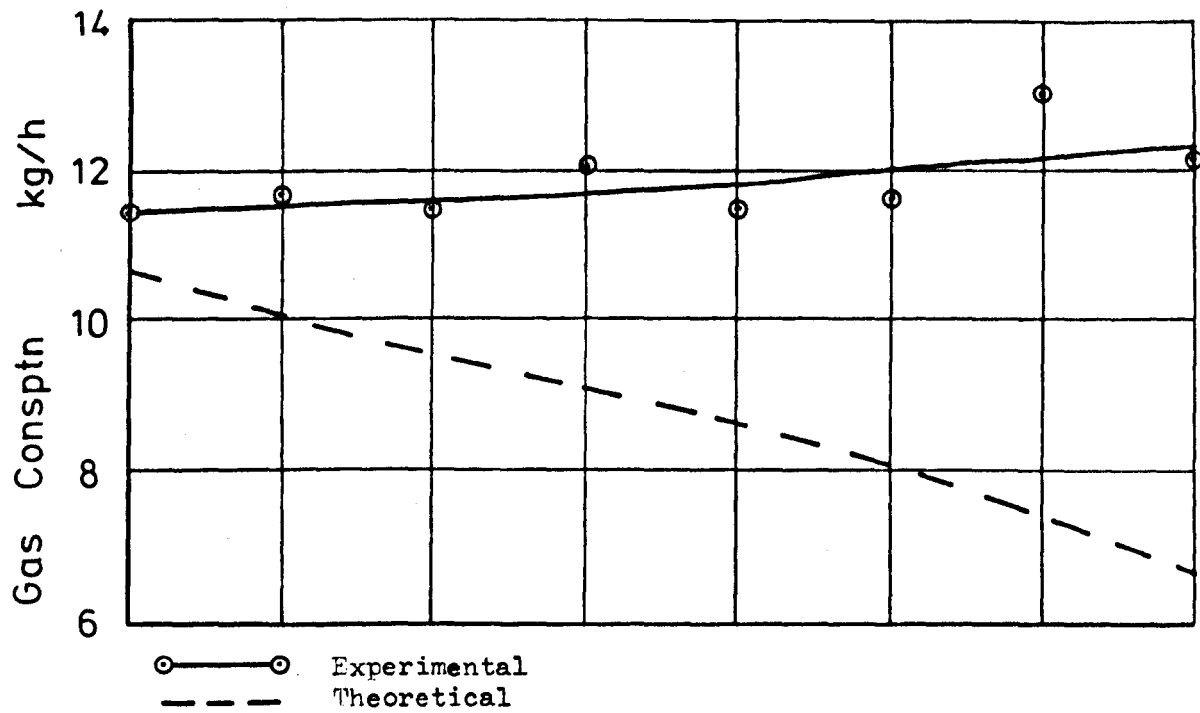
FIGURE 8.72



EXPERIMENTAL & THEORETICAL COMPARISON

HOLSET 4LGK 2.6 T2

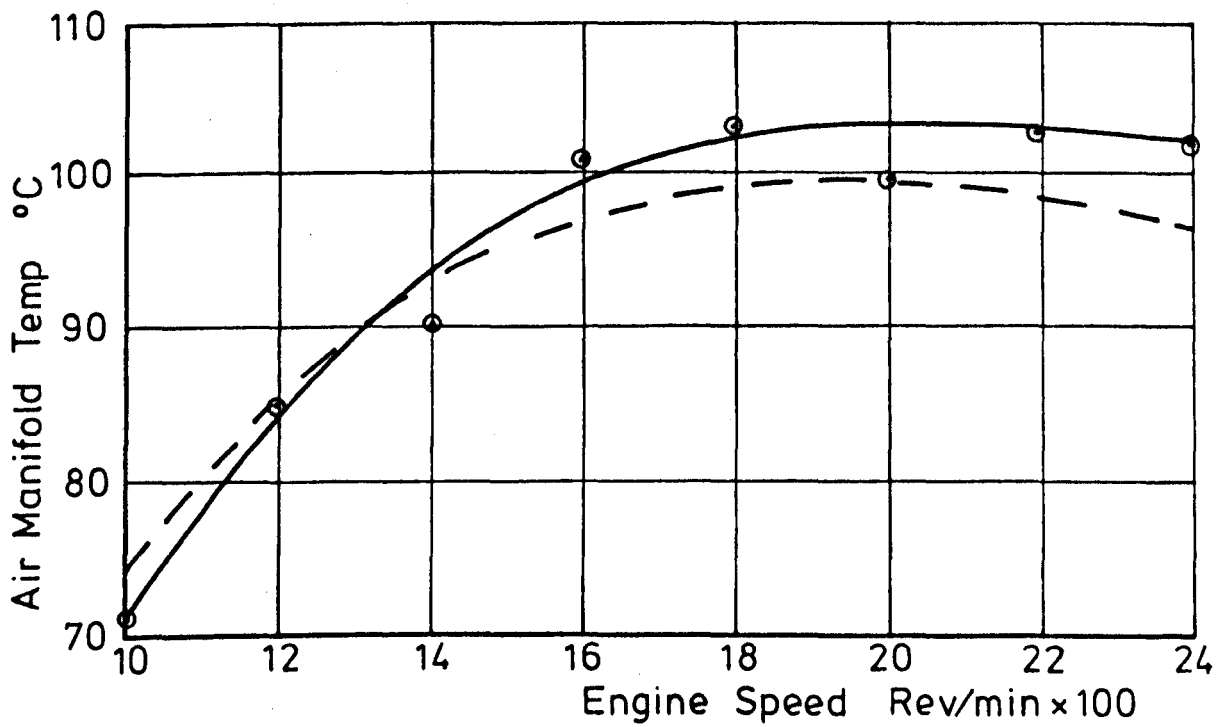
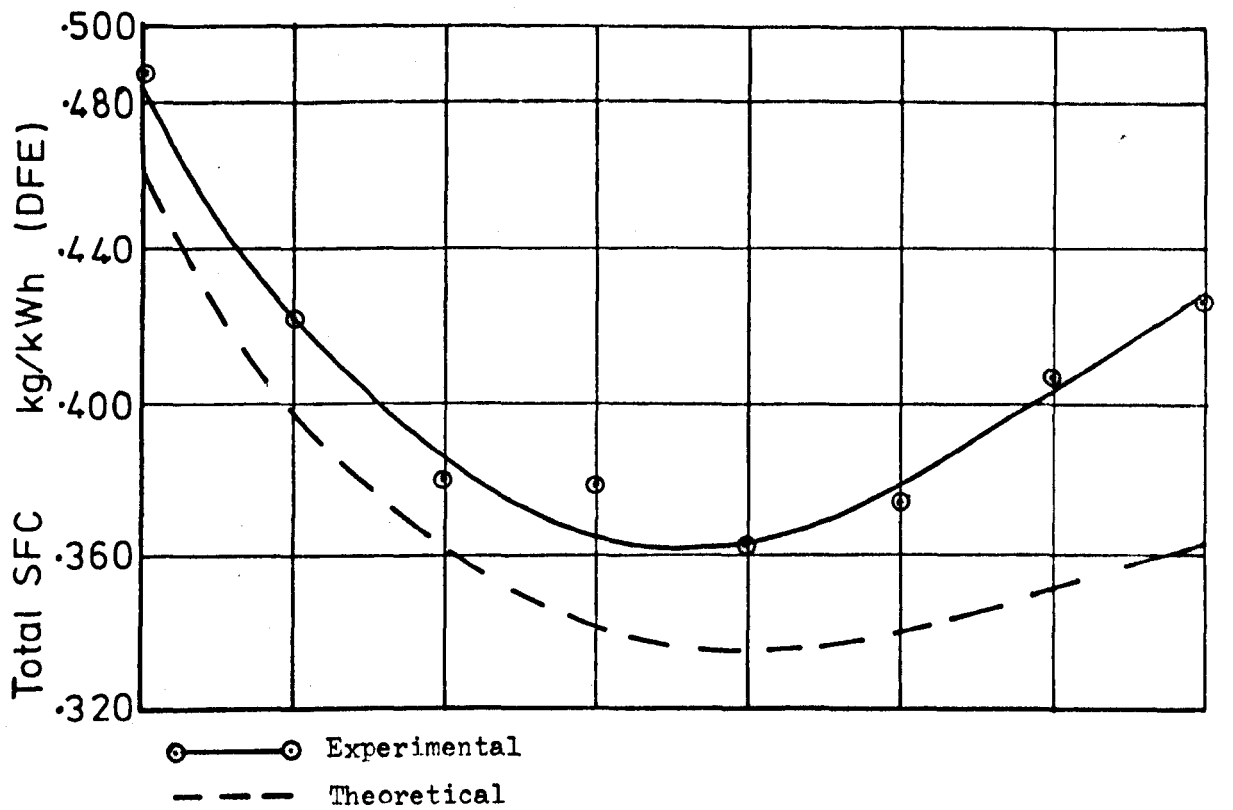
FIGURE 8.73



EXPERIMENTAL & THEORETICAL COMPARISON

HOLSET 4LGK 2.6 T2

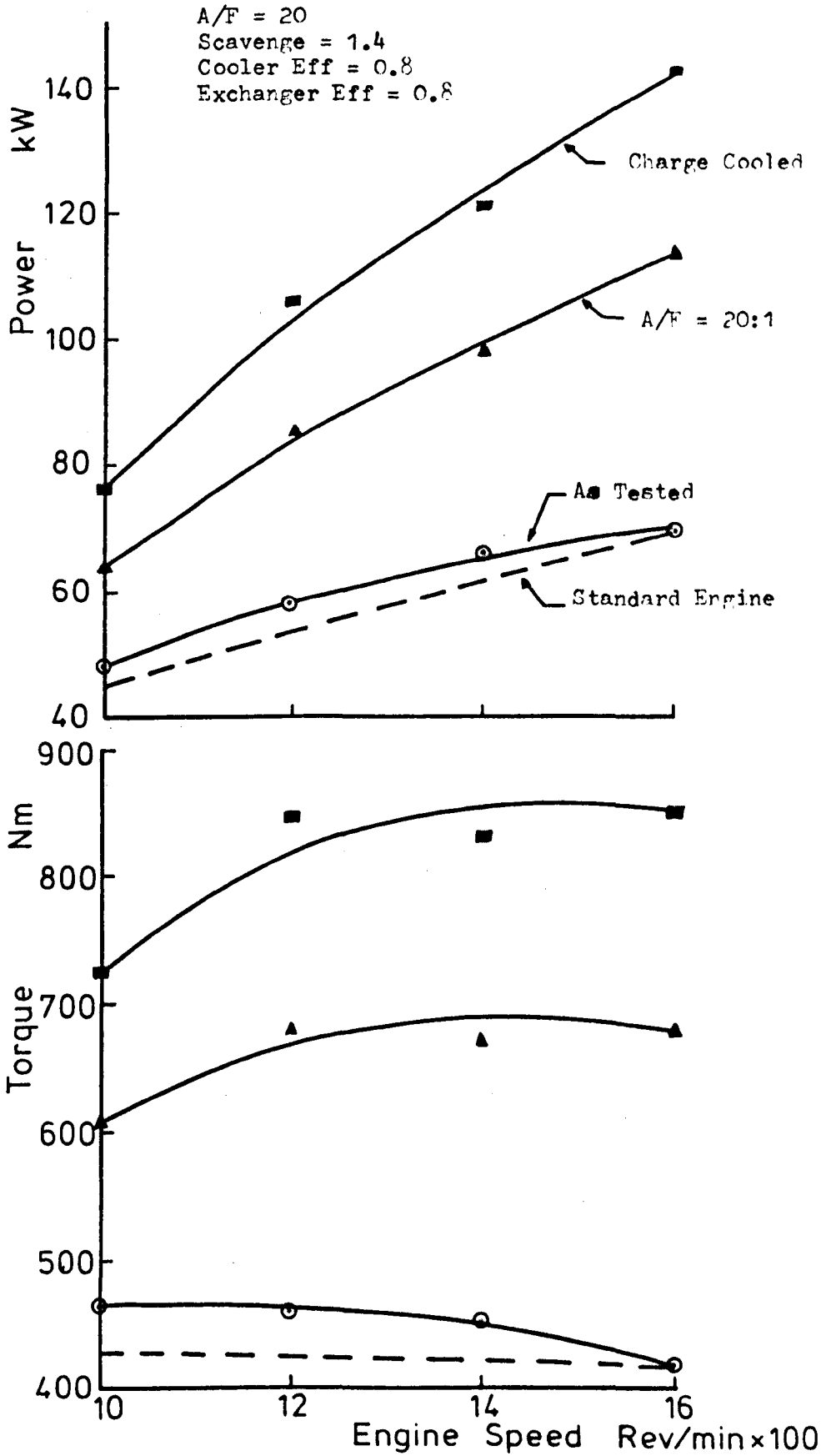
FIGURE 8.74



EXPERIMENTAL & THEORETICAL COMPARISON

HOLSET 4LGK 2.6 T2

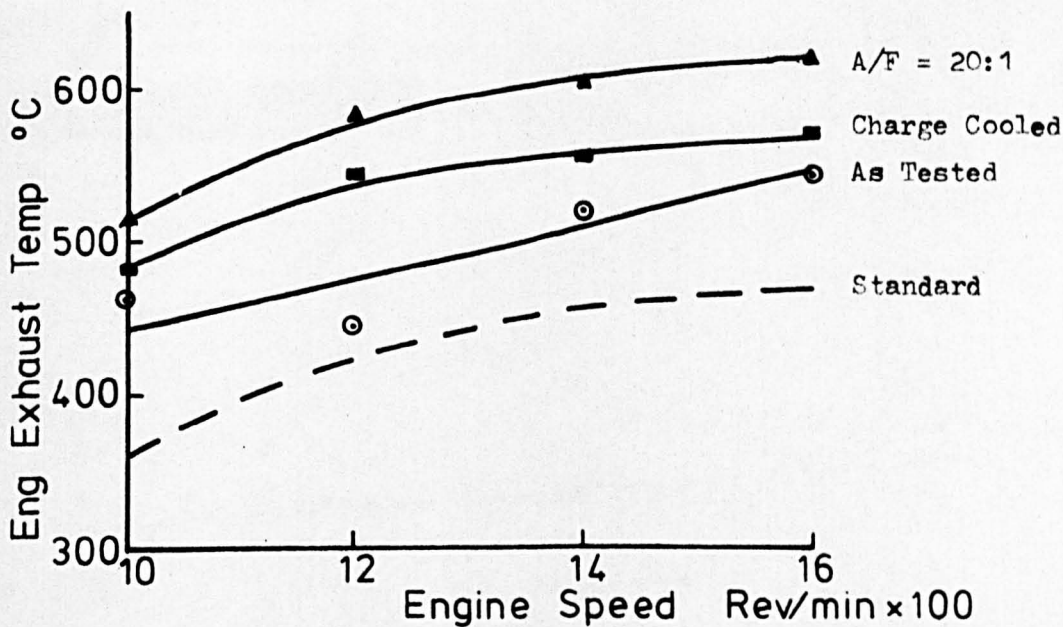
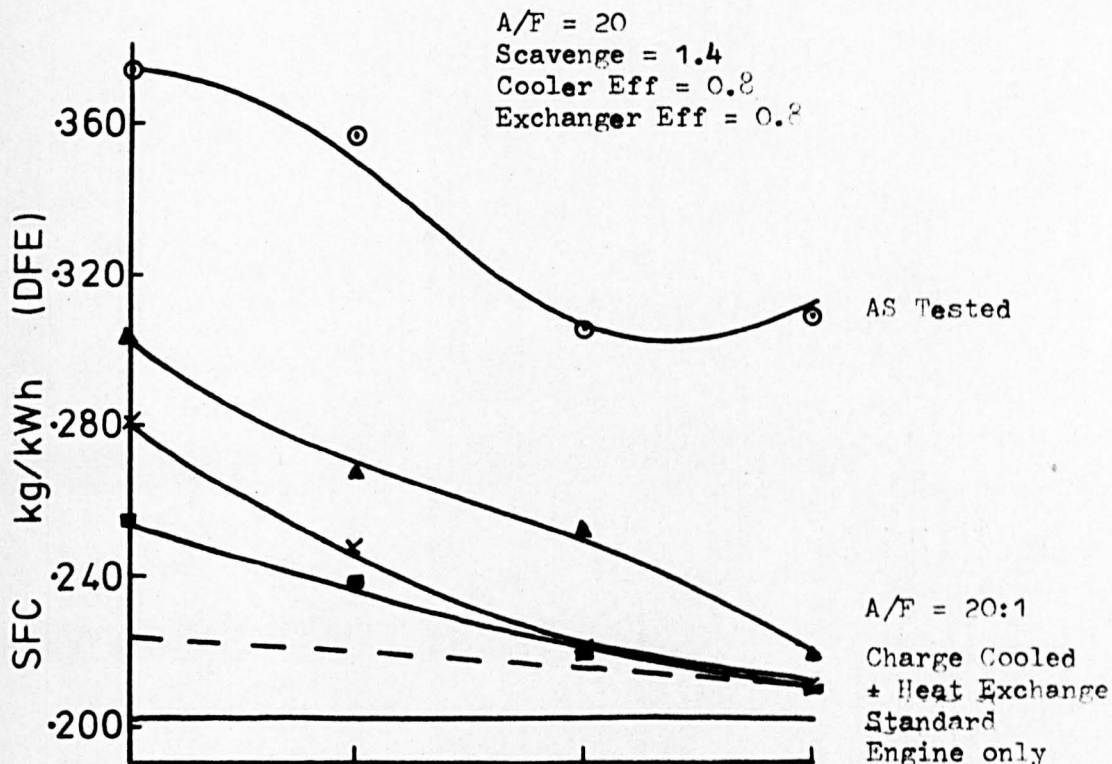
FIGURE 8.75



EXTENSION OF EXPTL WORK BY PREDICTION

HOLSET 3LD1 1,57

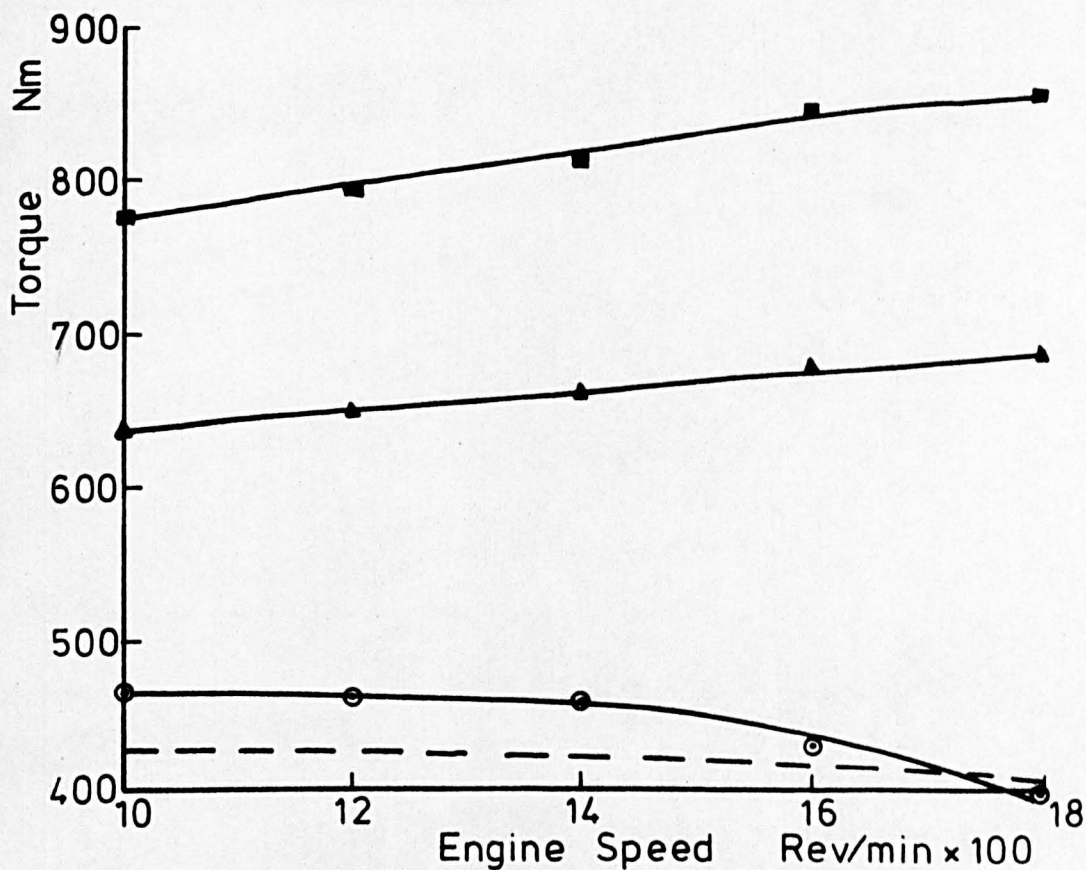
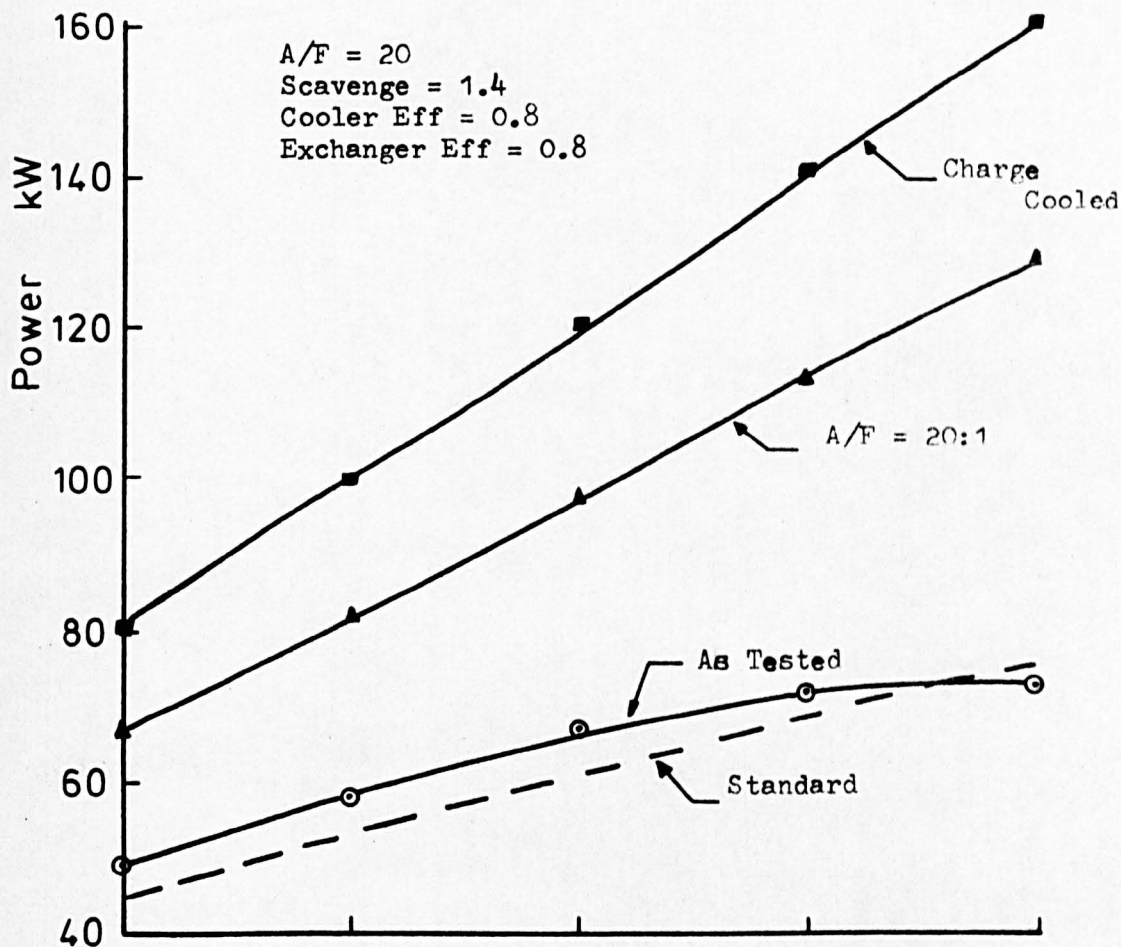
FIGURE 8,76



EXTENSION OF EXPTL WORK BY PREDICTION

HOLSET 3LD1 1,57

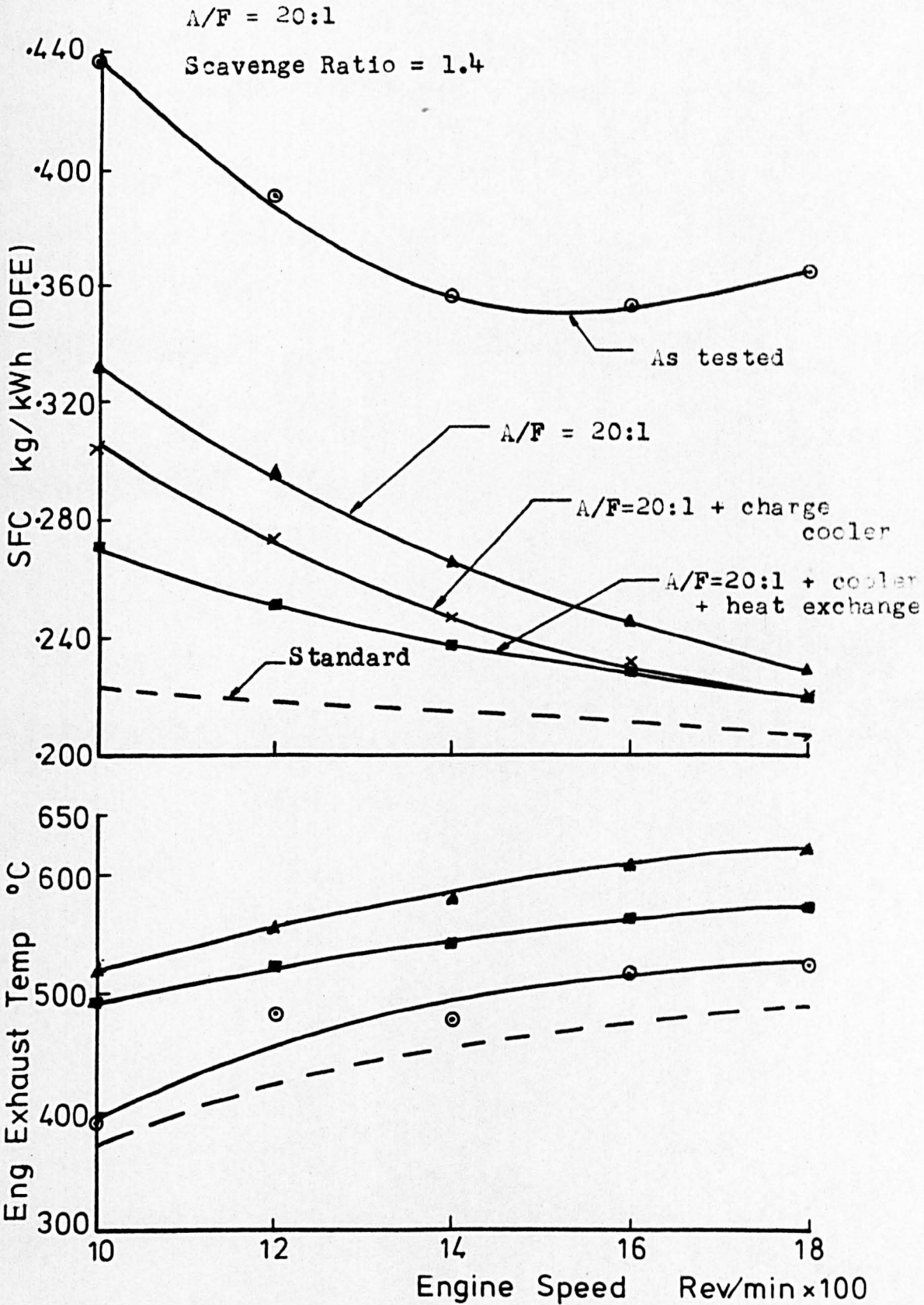
FIGURE 8.77



EXTENSION OF EXPTL WORK BY PREDICTION

HOLSET 3LD1 2.5

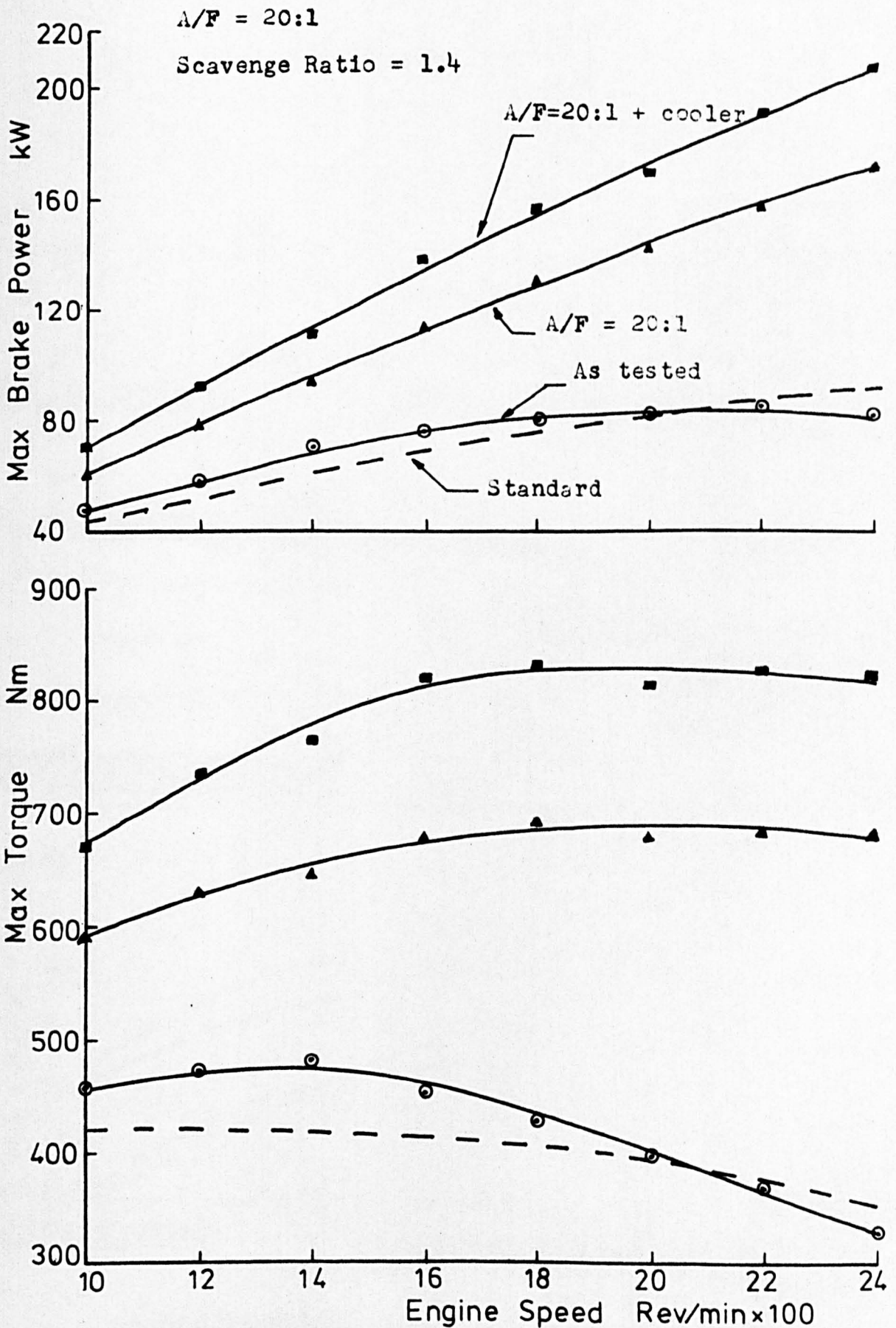
FIGURE 8.78



EXTENSION OF EXPTL WORK BY PREDICTION

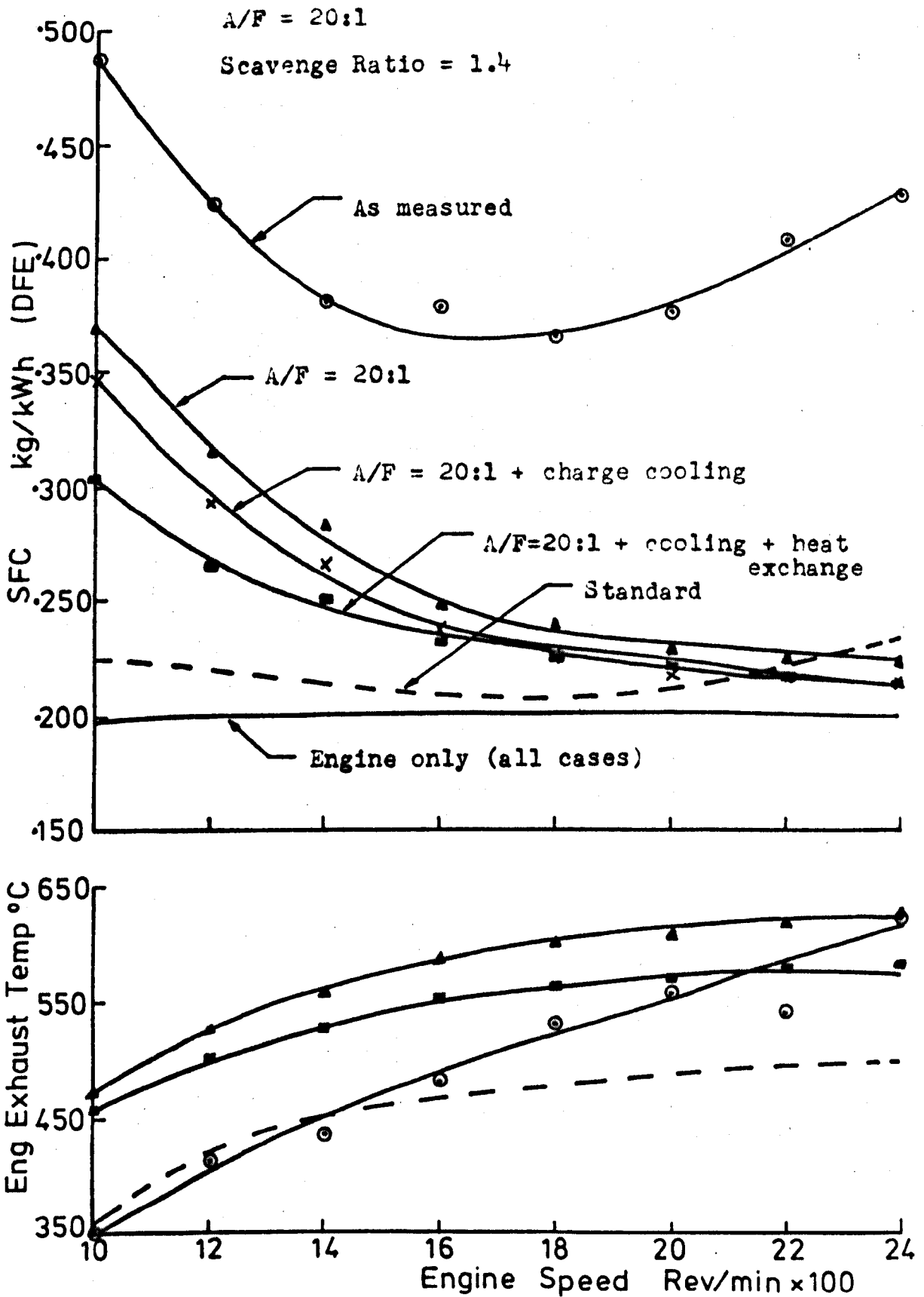
HOLSET 3LD1 2.5

FIGURE 8,79



EXTENSION OF EXPTL WORK BY PREDICTION
HOLSET 4LGK 2.6 T2

FIGURE 8,80



EXTENSION OF EXPTL WORK BY PREDICTION

HOLSET 4LGK 2,6 T2

FIGURE 8.81

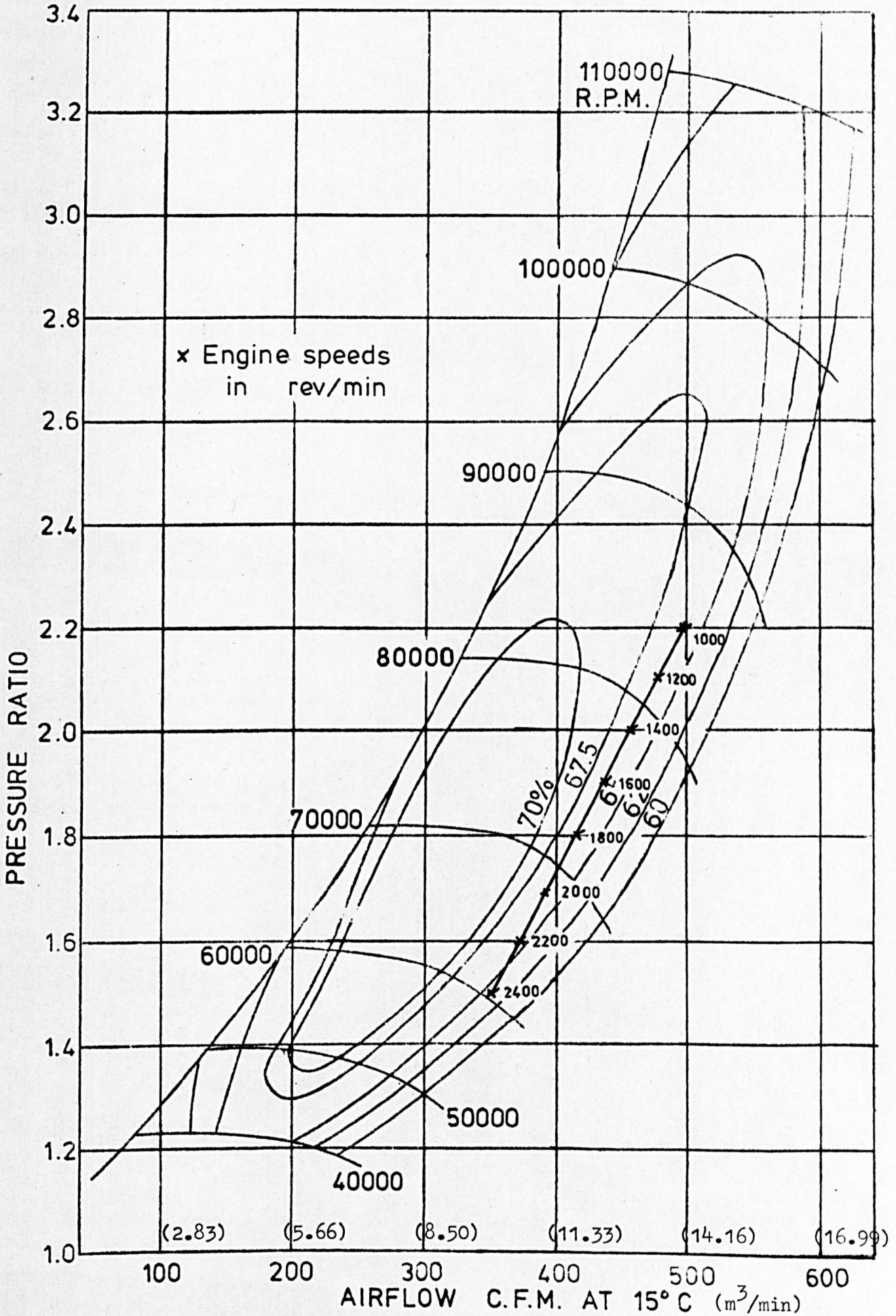
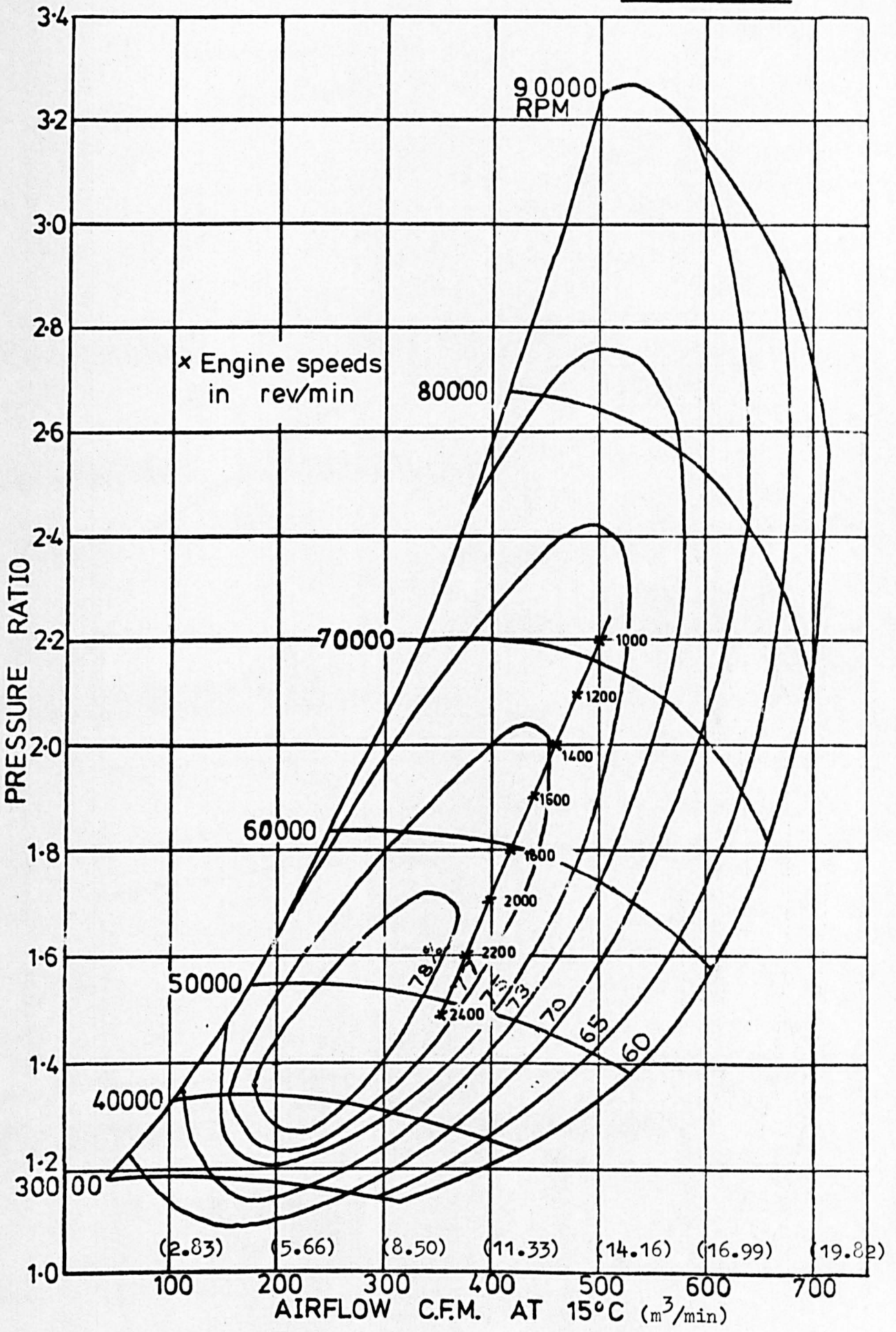
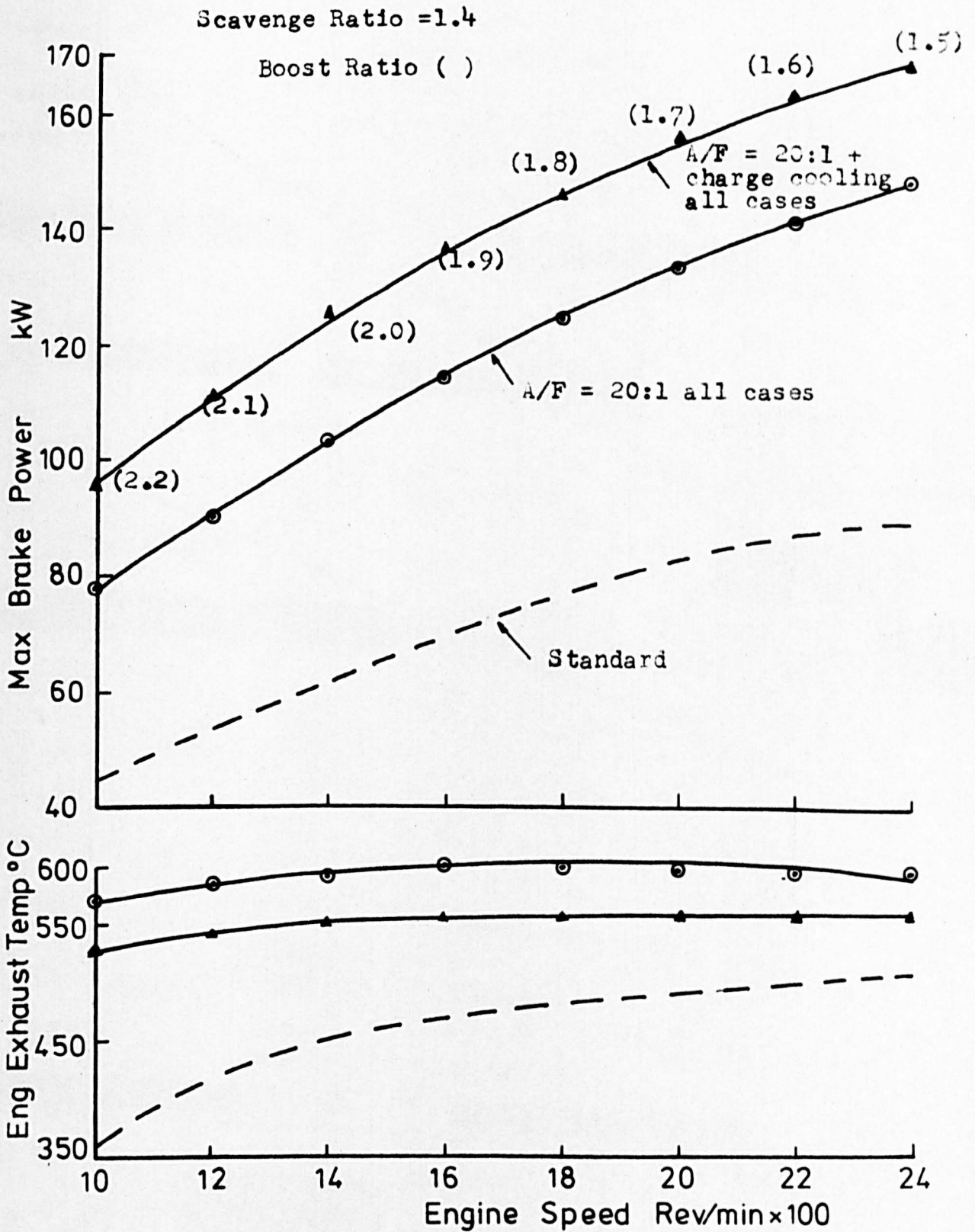


FIGURE 8.82



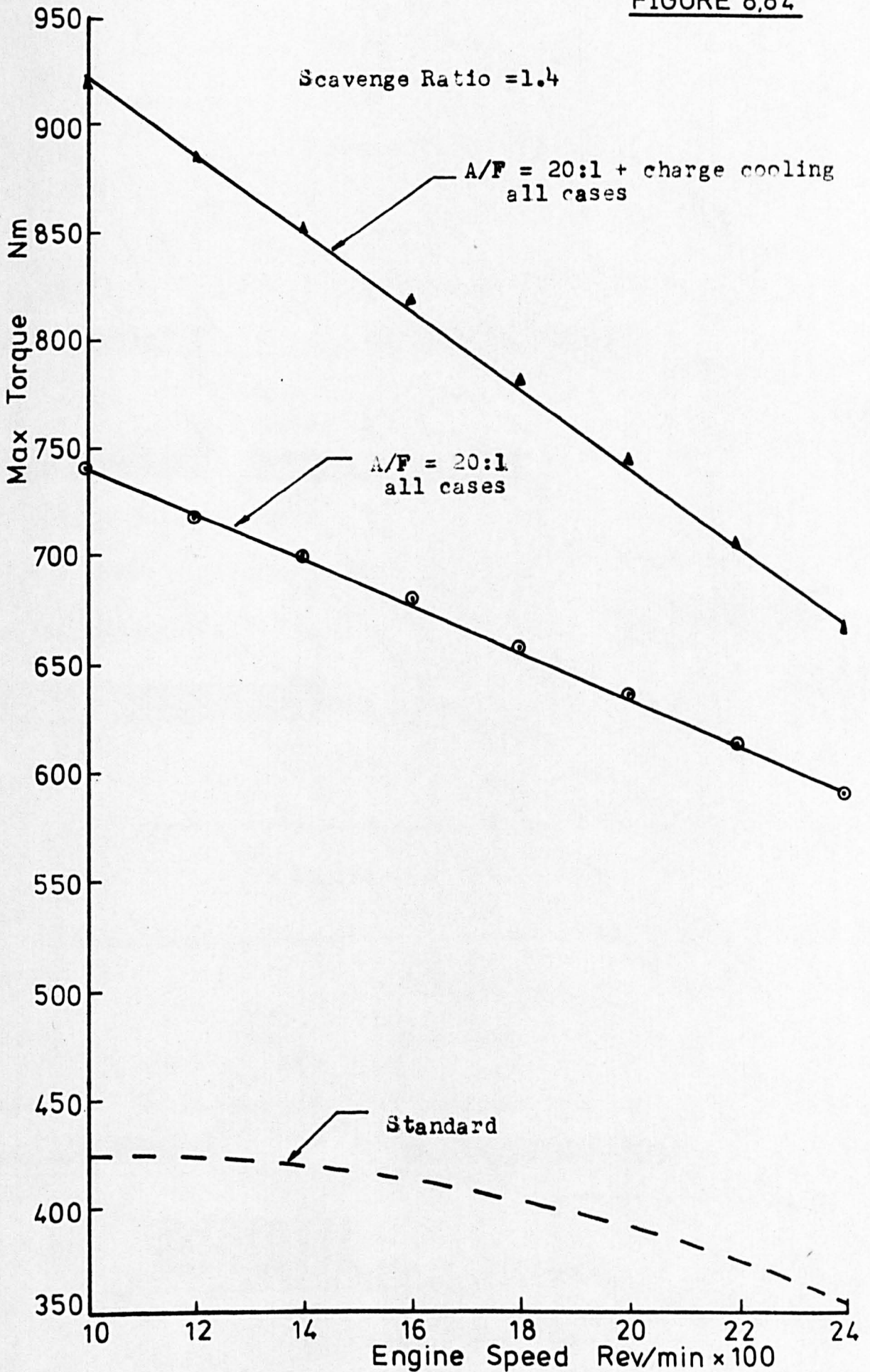
TORQUE TAILORING HOLSET 4LGK

FIGURE 8.83



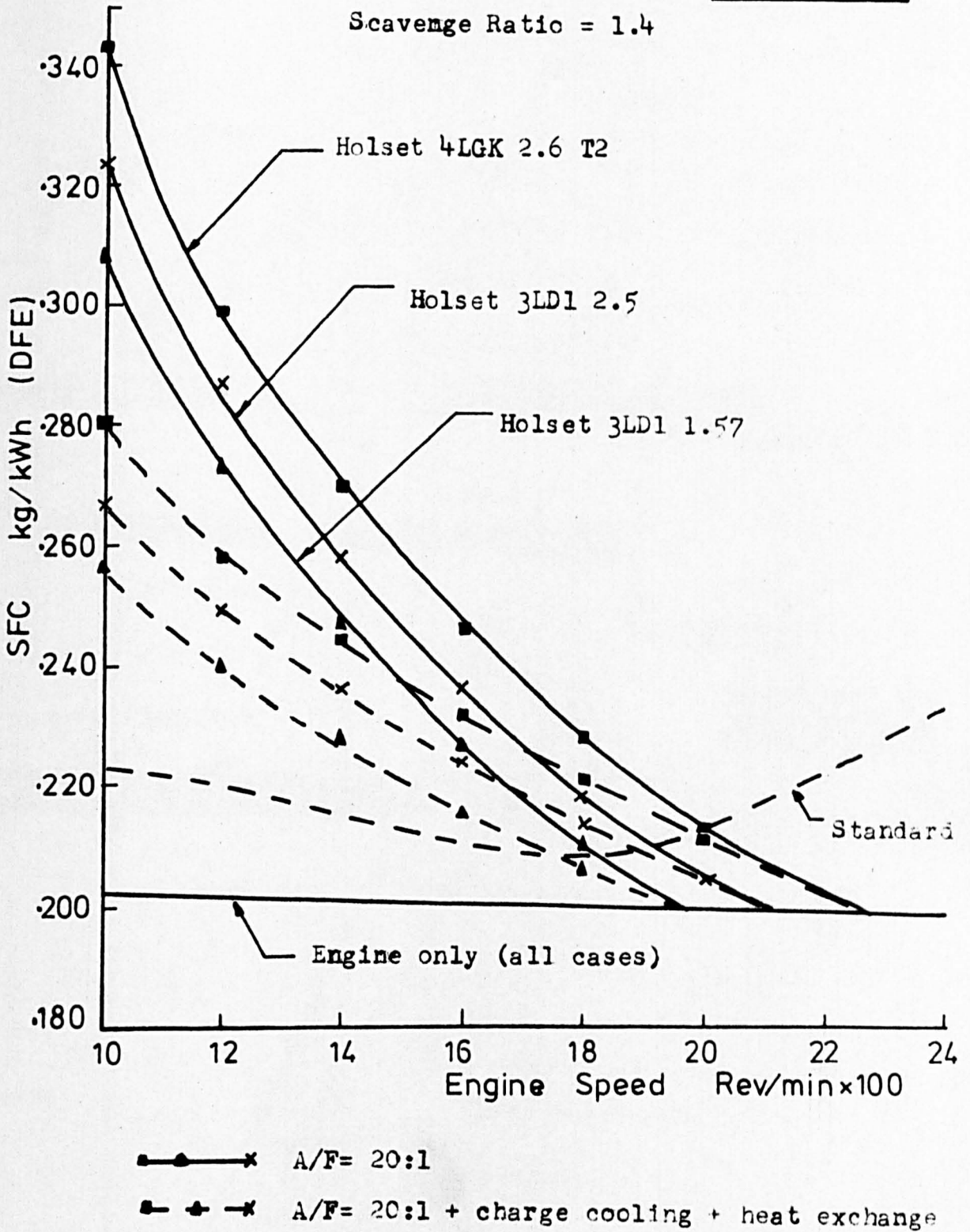
THEORETICAL PREDICTIONS FOR
TORQUE TAILORING

FIGURE 8.84



THEORETICAL PREDICTIONS FOR
TORQUE TAILORING

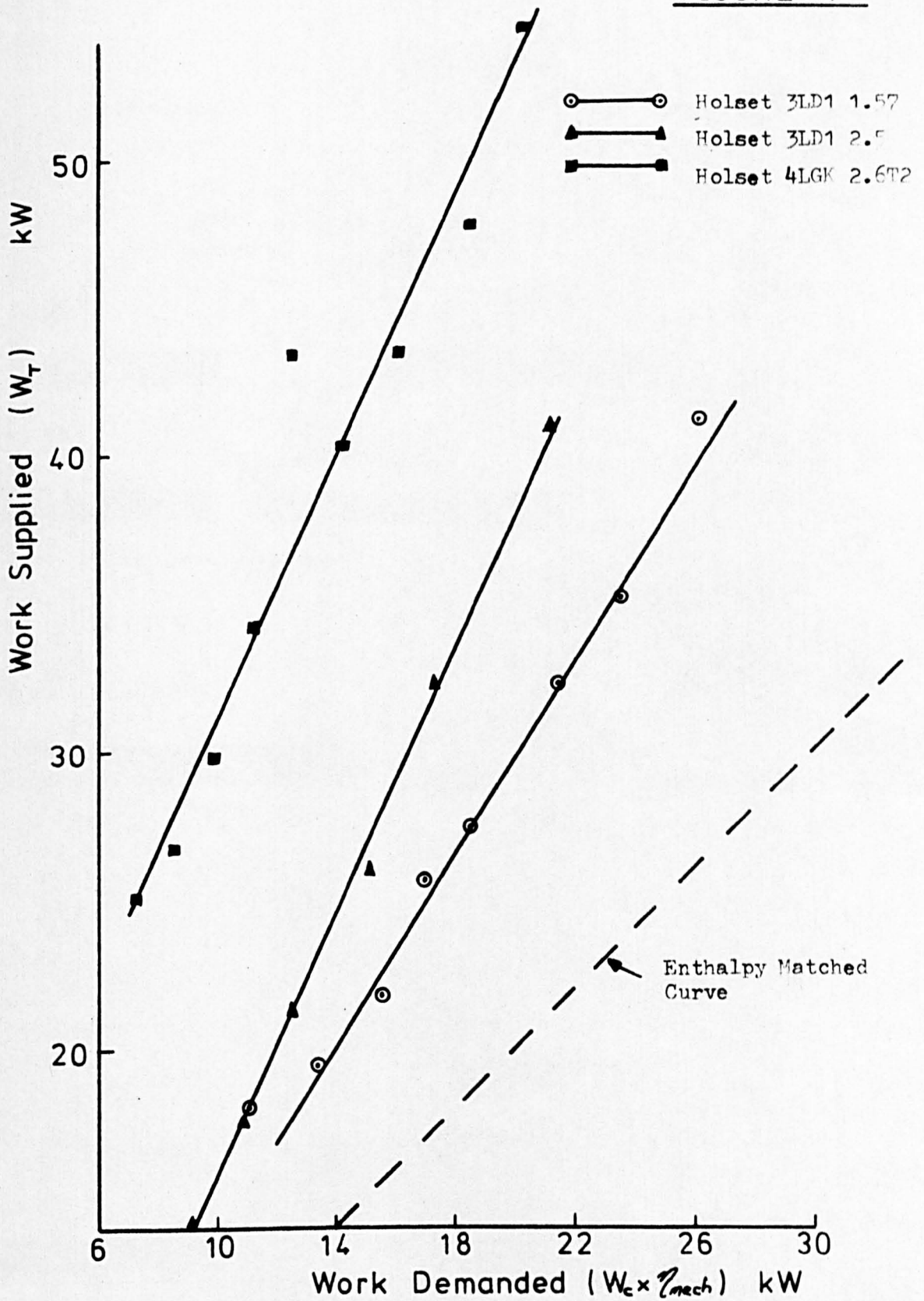
FIGURE 8,85



THEORETICAL PREDICTIONS FOR

TORQUE TAILORING

FIGURE 9.1

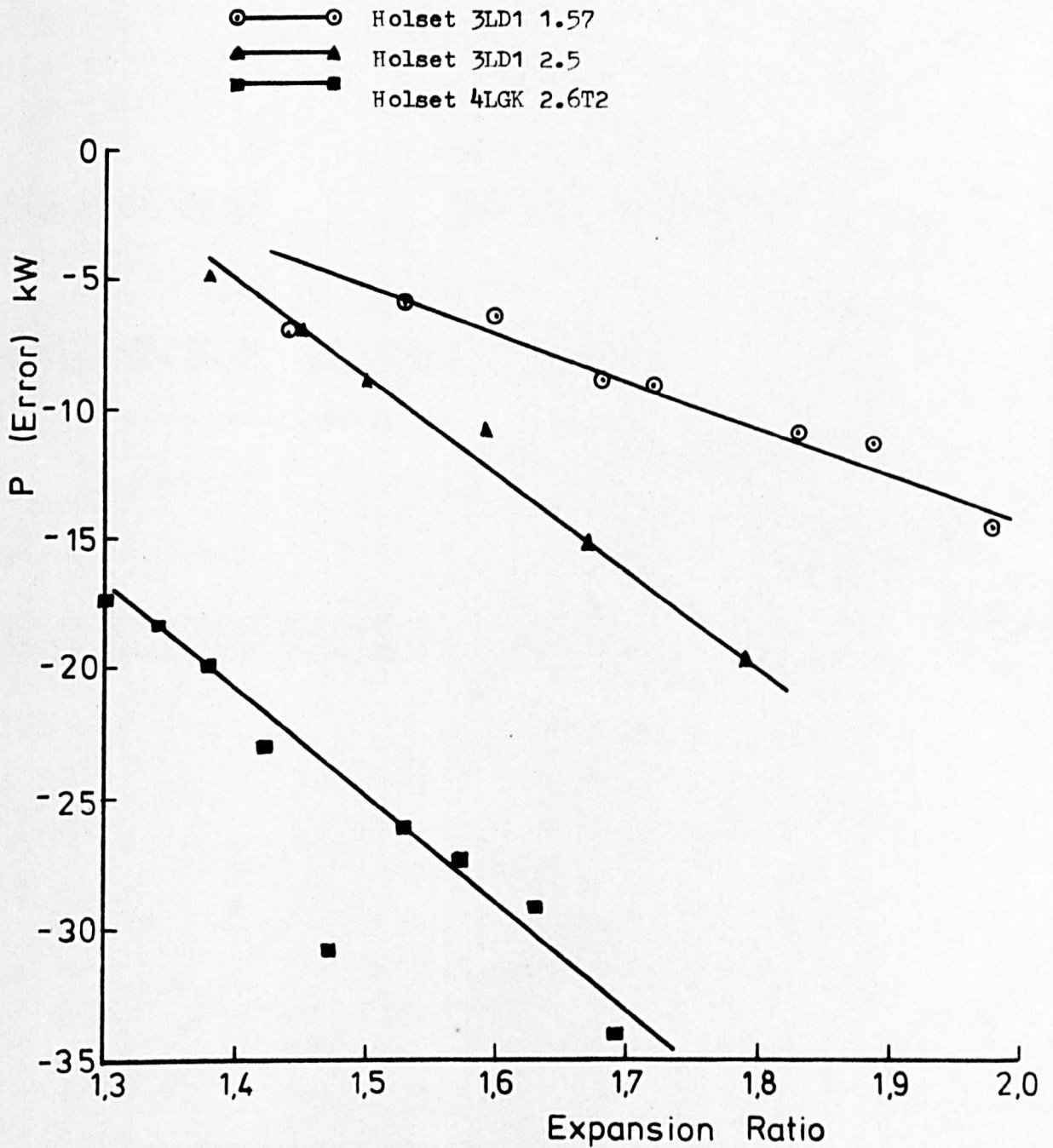


PREDICTED TURBOCHARGER PERFORMANCE

(Diesel Gas Turbine Program)

(Engine Speed = 1200 Rev/min)

FIGURE 9.2

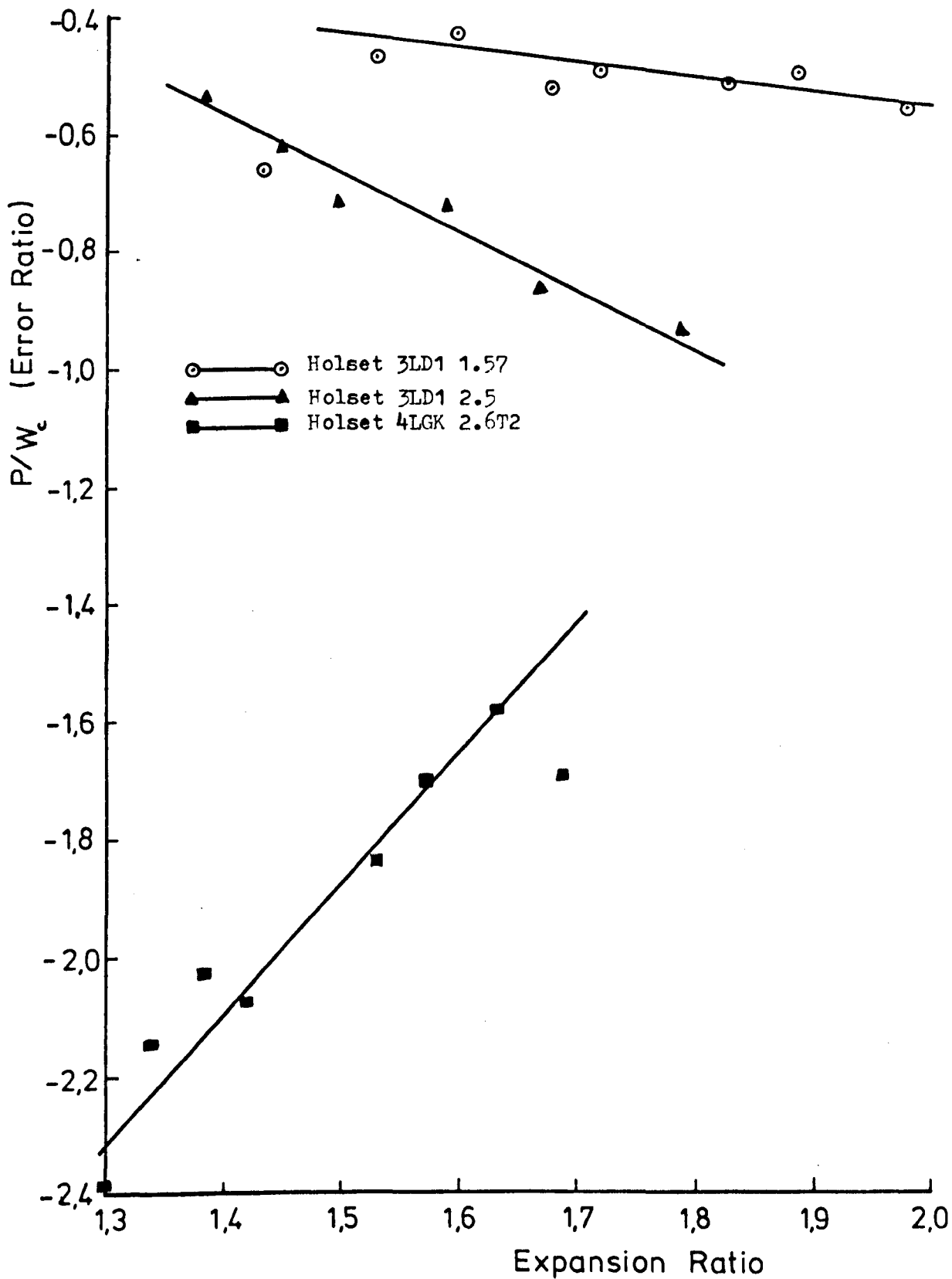


PREDICTED TURBOCHARGER PERFORMANCE

(Diesel Gas Turbine Program)

(Engine Speed=1200 Rev/min)

FIGURE 9.3

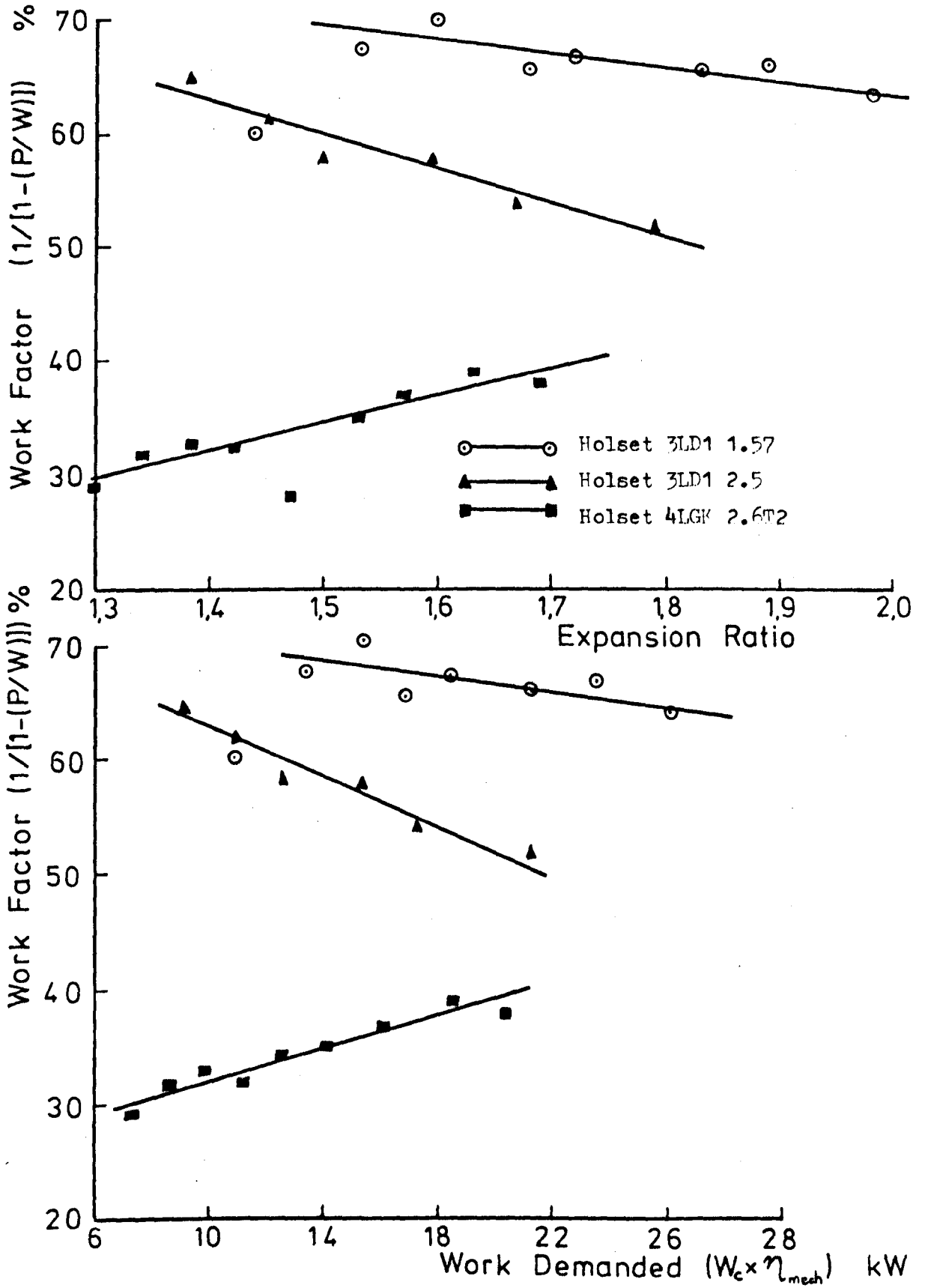


PREDICTED TURBOCHARGER PERFORMANCE

(Diesel Gas Turbine Program)

(Engine Speed = 1200 Rev/min)

FIGURE 9.4

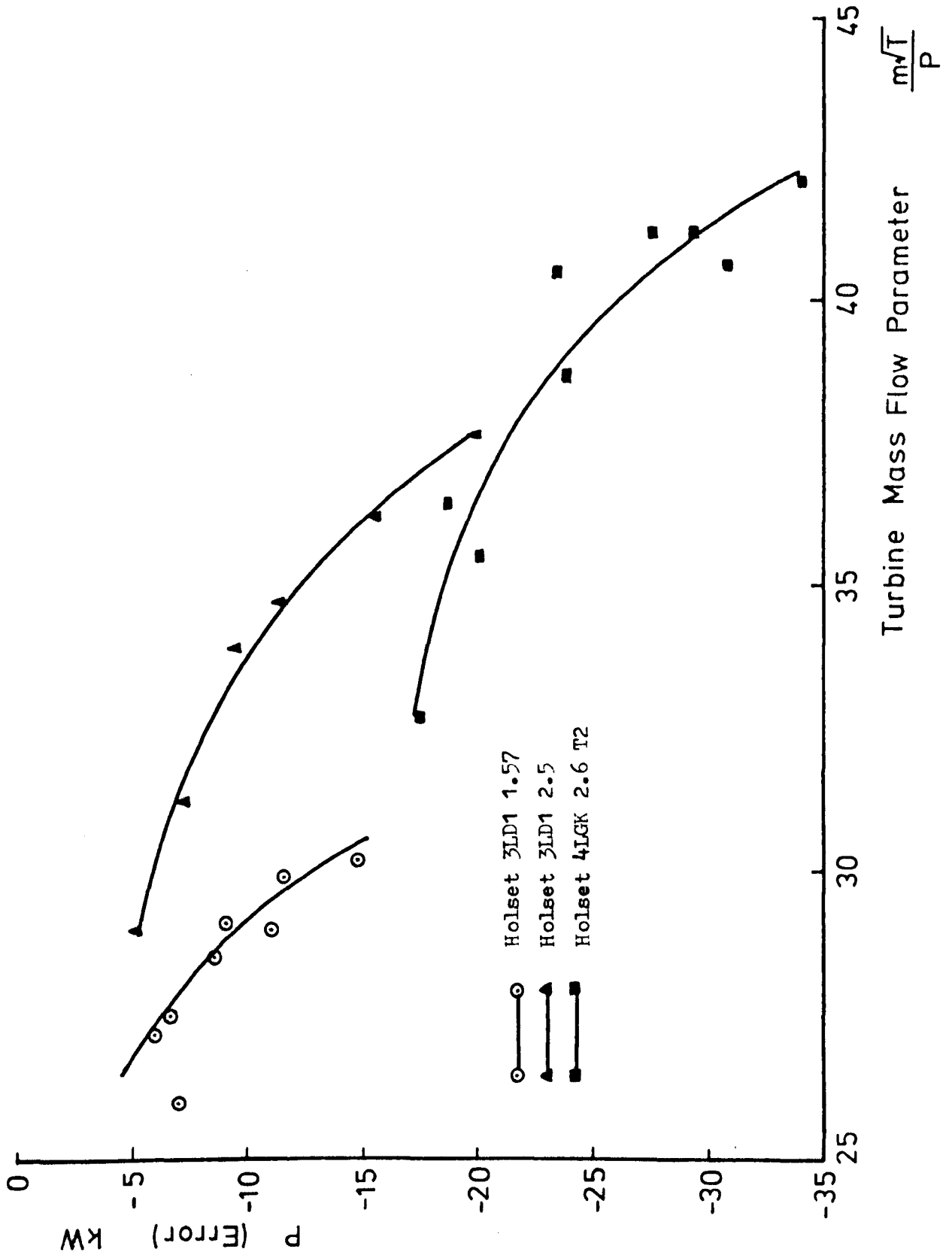


PREDICTED TURBOCHARGER PERFORMANCE

(Diesel Gas Turbine Program)

(Engine Speed = 1200 Rev/min)

FIGURE 9.5



PREDICTED TURBOCHARGER PERFORMANCE

(Diesel Gas Turbine Program)

(Engine Speed = 1200 Rev/min)

APPENDICES

Contents

- A1 Computer programs
- * A2 Fuel flow rate measuring system
- * A3 Voltage load readout for an hydraulic dynamometer
- * A4 The measurement of liquid petroleum gas flow rate for I.C. engine application
- * A5 Turbocharging the automotive two stroke diesel engine
- * A6 The diesel gas turbine
- * A7 The two stroke diesel gas turbine, promises and problems
- A8 Self sustained gas turbine tests and pulse investigation
- A9 Diesel gas turbine starting procedure

- * Publications

APPENDIX 1

COMPUTER PROGRAMS

This appendix contains the full listings of the performance prediction programs described in Chapter 5 and the data acquisition and control programs described in Chapter 6. The programs are listed in order as follows :-

- 1.1 Standard Engine Program
- 1.2 Preliminary Matching Program
- 1.3 Diesel Gas Turbine Program
- 1.4 Real Time Logging Program
- 1.5 Crankshaft Triggered Logging Program
- 1.6 Control Programs

The notation for each program is given and the performance prediction programs show cross references to equation numbers in Chapter 5.

Each main program is listed first and any subroutines are listed separately. Comment lines have been included to illustrate the working of the programs.

A1.1 Standard Engine Program

The program used for predicting the performance of the standard engine was developed by Wallace (4) and is based on a simple cycle analysis (see section 5.1).

The program makes use of several subroutines. The program notation and a complete listing follows.

Notation

AME	Engine mass flow (total)
AMF	Mass of fuel
AMT	Trapped air mass
B	Volume ratio (Beta)
CP	Specific heat at constant pressure
CR	Compression ratio
CV	Calorific value
E	Brake thermal efficiency
ED	Diagram efficiency
EP	Engine power
ED	Engine speed
GC	Gamma for compression process
GE	Gamma for expansion process
HC	Heat lost to coolant
NES	Number of engine speeds
PA	Atmospheric pressure
PHI	Cylinder filling efficiency
PMAX	Peak cylinder pressure
PMAXL	Limiting value of peak cylinder pressure
PT	Exhaust pressure
RC	Boost ratio
RT	Air/fuel ratio
RTU	Exhaust pressure ratio
S	Effective stroke ratio
SFC	Specific fuel consumption
SR	Scavenge ratio
TD	Temperature rise across engine

TM	Air manifold temperature
TMC	Mean temperature of Compression
TME	Mean temperature of expansion
TT	Exhaust temperature
VS	Swept volume

```

100 $RESET FREE
200 C
300 C PROGRAM FOR STANDARD ENGINE PERFORMANCE PREDICTIONS
400 C*****
500 C USING EXPERIMENTAL DATA
600 C*****
700 C
800 C
900 FILE 3(KIND=PRINTER)
1000 FILE 11(KIND=DISK,TITLE="KEEP/TASDAT",FILETYPE=7)
1100 COMMON IC(17,6),ID(23,2),IE(17,5)
1200 C SET UP CONSTANTS USED IN PROGRAM
1300 NES=8
1400 ET=1.0;VCR=0.0;S=0.78;VS=0.126;CRA=16.0;CRB=0.0;PMAXL=1800.0
1500 READ(11,/)PA
1600 C CONVERT BAR TO PSI
1700 PA=PA*14.5
1800 CV=18060.0;CA=144.0;CB=0.008;WHC=778.0;HPC=33000.0;HPCA=42.42
1900 PDC=0.86;R=53.3;TLA=5.0;TLB=5.0;TLC=5.0;TLD=50.0
2000 C OUTPUT PROGRAM PARAMETERS
2100 WRITE(3,5)ET,VCR,S,VS,CRA,CRB,PMAXL,PA,CV,CA,CB,WHC,HPC,HPCA,
2200 *PDC,R,TLA,TLB,TLC,TLD
2300 5 FORMAT(13H1 INPUT DATA //
2400 *7H ET = ,F10.3/7H VCR = ,F10.3/7H S = ,F10.3/7H VS = \
\F10.3/
2500 *7H CRA = ,F10.3/7H CRB = ,F10.3/9H PMAXL = ,F10.3/7H PA = \
\F10.3/
2600 *7H CV = ,F10.3/7H CA = ,F10.3/7H CB = ,F10.3/7H WHC = \
\F10.3/
2700 *7H HPC = ,F10.3/8H HPCA = ,F10.3/7H PDC = ,F10.3/7H R = \
\F10.3/
2800 *7H TLA = ,F10.3/7H TLB = ,F10.3/7H TLC = ,F10.3/7H TLD = \
\F10.3/
2900 *7H1 TRACE )
3000 C RUN PROGRAM FOR THE SET NO OF ENGINE SPEEDS
3100 DO 180 KA=1,NES
3200 I=0
3300 C INPUT ENGINE CONDITION DATA
3400 READ(11,/)ES,TM,ST,RT,RC
3500 C PROGRAM RESTRAINT A/F RATIO 20-40
3600 IF (RT.LT.20.0)RT=20.0
3700 C CONVERT DEG C TO DEG R
3800 TM=((TM*1.8)+459.67+32.0)
3900 RC=RC-0.1
4000 C ERROR CYCLE OF BOOST
4100 DO 180 ITIN= 1,6
4200 RC=RC+0.025
4300 SR=ST-0.4
4400 C ERROR CYCLE OF SCAVENGE RATIO
4500 DO 180 ITIM=1,4

```

Standard Engine Program

```

4600      SR=SR+0.1
4700      I=I+1
4800      PHI=0.9
4900      ED=0.72
5000 C WRITE/NOT WRITE HEADING
5100      IF(I.EQ.1) GO TO 11
5200      12 DTK=ED*PHI*WHC
5300 C INITIAL GAMMA SETTINGS
5400      GE=1.3
5500      GC=1.35
5600 C ROUTING FOR VARIABLE COMPRESSION RATIO
5700      IF(VCR.NE.0.0) GO TO 30
5800      CR=CRA
5900 C *****
6000 C ITERATE FOR PMAX TOLERANCE(TLA)
6100 C
6200 10  PMAX =(GE-1.0)*RC*PA*CR/(1.0-1.0/CR**(GE-1.0))*((DTK*CV/R/RT\
      \sqrt{TM*(
6300      *1.0-1.0/CR**0.4))-1.0/(GC-1.0)*(1.0-CR**(GC-1.0))) [4]
6400      TME=TM*PMAX*(1.0+1.0/CR**(GE-1.0))/RC/PA/CR/2.0 [6]
6500 C FIND GAMMA EXPANSION
6600      CALL GG (RT,TME,GE)
6700      TMC=TM*(CR**(GC-1.0)+1.0)/2.0 [7]
6800 C FIND GAMMA COMPRESSION
6900      CALL GG (0.0,TMC,GC)
7000      PMAXA=(GE-1.0)*RC*PA*CR/(1.0-1.0/CR**(GE-1.0))*((DTK*CV/R/RT\
      \sqrt{TM*(
7100      *1.0-1.0/CR**0.4))-1.0/(GC-1.0)*(1.0-CR**(GC-1.0)))
7200      IF(ABS(PMAX-PMAXA).GT.TLA) GO TO 10
7300 C
7400 C *****
7500 C ROUTING FOR DUAL CYCLE COMBUSTION
7600      IF(PMAX.GT.PMAXL) GO TO 20
7700      90 AMT=RC*PA*S*PHI*VS*ES*CA/R/TM/(1.0-1.0/CR)/ET [2]
7800      EP=ED*AMT*CV*WHC*(1.0-1.0/CR**0.4)/RT/HPC
7900      GO TO 40
8000 C *****
8100 C SECTION FOR DUAL CYCLE COMBUSTION
8200 C CONST VOLUME THEN CONST PRESSURE
8300      20 B=0.0
8400      BS=0.0
8500      BD=0.0
8600 C FIND VOLUME RATIO(BETA)
8700      CALL SOL (0.5,B,BS,BD)
8800      B = 1.01
8900 C *****
9000 C ITERATE FOR PMAX TOLERANCE(TLB)
9100 C
9200      M=0
9300      50 M=M+1

```

Standard Engine Program

```

9400 PMAX=PMAXL
9500 A=PMAX/RC/PA/CR**GC
9600 P=1.0-(A*B**1.4-1.0)/(A-1.0+1.4*A*(B-1.0))/CR**0.4
9700 TME=TM*PMAX*B*(1.0+(B/CR)**(GE-1.0))/RC/PA/CR/2.0 [6]
9800 CALL GG (RT,TME,GE)
9900 IF(GE.LT.0.01) GOTO 170
10000 TMC = TM * (CR**(GC - 1.0) + 1.0) / 2.0 [7]
10100 CALL GG (0.0,TMC,GC)
10200 IF(GC.LT.0.01)GOTO170
10300 PMAX=(P*(DTK*CV/R/RT/TM-(1.0-CR**(GC-1.0))/P/(GC-1.0)))/(B\
\CR*(1.0
10400 *(B/CR)**(GE-1.0))/(GE-1.0)/RC/PA+(B-1.0)/RC/PA/CR) [5]
10500 DA=PMAX-PMAXL
10600 IF(ABS(DA).LT.TLB) GO TO 60
10700 C
10800 C *****
10900 CALL SOL (DA,B,BS,BD)
11000 C ARRANGE WARNING FOR EXCESSIVE LOOPING
11100 IF(MA.EQ.40) GO TO 15
11200 GO TO 17
11300 15 WRITE(3,16)
11400 16 FORMAT(37H EXCESSIVE LOOPING PMAX(TLB) AS BELOW)
11500 GO TO 60
11600 17 GO TO 50
11700 C ***** [2]
11800 60 AMT=RC*PA*CA*S*PHI*VS*ES/R/TM/(1.0-1.0/CR)/ET
11900 C 60 IS RE-ENTRY POINT FOR DUAL CYCLE
12000 EP=ED*AMT*CV*WHC*(1.0-((A*B**1.4-1.0)*1.0/CR**0.4)/(A-1.0+1\
\4*A*
12100 *(B-1.0)))/RT/HPC [9]
12200 GO TO 40
12300 C
12400 C *****
12500 C SECTION FOR VARIABLE CR
12600 30 MA=0
12700 CR=CRA-((CRA-CRB)/2.0)
12800 PMAX=PMAXL
12900 C *****
13000 C ITERATE FOR PMAX TOLERANCE(TLC)
13100 70 MA=MA+1
13200 TME=TM*PMAX*(1.0+1.0/CR**(GE-1.0))/RC/PA/CR/2.0
13300 CALL GG(RT,TME,GE)
13400 IF(GE.LT.0.01) GO TO 170
13500 TMC=TM*(CR**(GC-1.0)+1.0)/2.0
13600 CALLGG (0.0,TMC,GC)
13700 IF(GC.LT.0.01) GO TO 170
13800 PMAX=(GE-1.0)*RC*PA*CR/(1.0-1.0/CR**(GE-1.0))*((DTK*CV/R/RT\
\TM*(
13900 *1.0-1.0/CR**0.4))-1.0/(GC-1.0)*(1.0-CR**(GC-1.0)))
14000 DB=PMAX-PMAXL

```

Standard Engine Program

```

14100      IF (ABS(DB).LT.TLC) GO TO 80
14200      IF (DB.GT.0.0) CR=CR-0.2
14300      IF (DB.LT.0.0) CR=CR+0.2
14400      IF (MA.GT.40) GO TO 41
14500      GO TO 42
14600 C ARRANGE WARNING FOR EXCESSIVE LOOPING
14700      41 WRITE(3,99)
14800      99 FORMAT(31H EXCESSIVE VCR LOOPING AS BELOW)
14900      GO TO 80
15000      42 GO TO 70
15100 C
15200 C*****
15300      80 IF (CR.LE.CRA) GO TO 85
15400      CR=CRA
15500      GO TO 20
15600      85 IF (CR.GE.CRB) GO TO 90
15700      CR=CRB
15800      GO TO 20
15900 C*****
16000 C FRICTIONAL HORSE POWER CORRECTION
16100      40 EP=EP-(CB*ES/0.7457) [8]
16200 C 40 IS RE ENTRY POINT
16300 C SCAVENGE BLOWER HORSE POWER CORRECTION
16400      EP=EP-(ES*1.8*0.00060*((PA*RC)-PA)) [13]
16500      AMF=AMT/RT [15]
16600      SFC=AMF*60.0/EP [17]
16700      E=EP*HPCA/CV/AMF
16800      FF = AMF * 100.0 / 60.0 / VS
16900 C FIND HEAT LOSS TO COOLANT
17000      CALL H (FF,HC)
17100      IF (HC.LT.0.01) GOTO 170
17200      CP = 0.3
17300 C DELTA T FOR ENGINE(TEMP RISE)
17400 C*****
17500 C ITERATE FOR ENG TEMP RISE TOLERANCE (TLD)
17600 C
17700      200 TD = (CV * (1.0-E-HC)) / (CP * (SR * RT + 1.0)) [16]
17800      TMD = TM + TD / 2.0
17900 C OUTPUT TO DENOTE FAILURE IN THIS LOOP
18000      IF (TMD.GT.3600.0) WRITE (3,977) TM,TD,CV,E,HC,FF,CP,SR,RT\
      \,EP,
      * CB,ES,AMF,AMT,HPCA,CV
18100
18200      977 FORMAT (6H TM = ,F12.6,6H TD = ,F12.6,6H CV = ,F12.6,6H E \
      \= ,
18300      *F12.6/6H HC = ,F12.6,6H FF = ,F12.6,6H CP = ,F12.6,6H SR = ,
18400      *F12.6/6H RT = ,F12.6,6H EP = ,F12.6,6H CB = ,F12.6,6H ES = ,
18500      *F12.6/6H AMF =,F12.6,6H AMT = ,F12.6,6H HPCA =,F12.6,6H CV = \
      \,F12.6)
18600 C FIND SPECIFIC HEAT AT CONSTANT PRESSURE(CP)
18700      CALL C (RT,TMD,CP)

```

Standard Engine Program

```

18800      IF(CP.LT.0.01)GOTO 170
18900      TDA = (CV * (1.0-E-HC)) / (CP * (SR * RT + 1.0))
19000      IF (ABS(TD-TDA).GT.TLD) GOTO 200
19100 C
19200 C*****
19300      120 PT=PDC*RC*PA
19400      130 RTU=PT/PA
19500      TT=TM+TD
19600      AME=SR*AMT+AMF
19700      GO TO 170
19800      11 WRITE(3,150)
19900      GO TO 12
20000      150 FORMAT(116H1ENG SPD   BOOST   TR A/F   SCAV R   MAN\
      \ TEMP   ENG
20100      *POW   BRK EFF   EXH TEMP   EXH RAT   PMAX   COMP RAT   \
      \ SFC/
20200      *42X,5HDEG.C,6X,3HKW ,16X,5HDEG.C,16X,4H BAR,16X,11H KG/KW\
      \.HR   )
20300 C*****
20400 C S. I. UNIT CONVERSION
20500 C
20600      170 PZ=EP*0.7457
20700      PY=(TM-459.67-32.0)/1.8
20800      SFP=(SFC*0.4536)/0.7457
20900      PX=(TT-459.67-32.0)/1.8
21000      PMAT=PMAX/14.5
21100 C
21200 C*****
21300 C OUTPUT
21400 C ENG SPD,BST,CR,A/F,SCAV,MAN TEMP,POWER,BRK EFF,EX TEMP,
21500 C EX RAT,PMAX,COMP RAT,SFC
21600      WRITE(3,160) ES,RC,RT,SR,PY,PZ,E,PX,RTU,PMAT,CR,SFP
21700      160 FORMAT(1H ,F7.1,3X,F5.2,5X,F5.1,4X,F5.2,4X,F7.1,3X,
21800      *F7.2,3X,F7.4,3X;F7.1,3X,F6.3,6X,F7.2,3X,F7.2,5X,F6.4,3X)
21900      180 CONTINUE
22000      STOP
22100      END

```

Standard Engine Program

```

22200      SUBROUTINE SOL (DAB,B,BS,BD)
22300 C      *****
22400 C
22500 C DETERMINES VOLUME RATIO(BETA)
22600 C
22700      IF (B.LE.0.01) BD = 1.0
22800      BS = 0.2
22900      IF (ABS(DAB).LT.1200.1) BS = 0.1
23000      IF (ABS(DAB).LT.700.1) BS = 0.05
23100      IF (ABS(DAB).LT.200.1) BS = 0.01
23200      IF (DAB.GE.0.0) GOTO 10
23300      BD = BD / 2.0
23400      BS =-BS
23500 10    B = B + BS * BD
23600      RETURN
23700      END
23800 C
23900 C
24000      SUBROUTINE C (RT,TMD,CP)
24100 C      *****
24200 C
24300 C DETERMINES VALUE FOR SPECIFIC HEAT(CP)
24400 C
24500 C ARRANGE FLAG IF A/F RATIO OR MEAN TEMP
24600 C OUTSIDE BLOCK DATA LIMITS
24700 C TEMP=400-3600 DEG R
24800 C AIR/FUEL RATIO=20-40
24900      COMMON ICD(17,6), IDD(23,2), IC(17,5)
25000      I = RT / 5.0 - 3.0
25100      A = TMD / 200.0 - 1.0
25200      J = A
25300      IF (I.GT.5)GOTO 10
25400      IF (J.LT.1.OR.J.GT.17) GOTO 10
25500      IF (J.EQ.17)GOTO 30
25600 C LINEAR INTERPOLATION FROM BLOCK DATA
25700      AI = FLOAT(IC(J,I)) / 10000.0
25800      AI1 =FLOAT(IC(J+1,I)) / 10000.
25900      CP = AI + (AI - AI1) * (FLOAT(J) - A)
26000      RETURN
26100 30    CP = FLOAT(IC(J,I))/10000.
26200      RETURN
26300 10    WRITE (3,20) RT, TMD
26400 20    FORMAT (16H INVALID REQUEST / 6H.RT = ,E12.6,8H TMD = \
      \,E12.6)
26500      CP = 0.0
26600      RETURN
26700      END

```

Standard Engine Program

```

26800      SUBROUTINE H (FF,HC)
26900 C      *****
27000 C
27100 C DETERMINES HEAT LOSS TO COOLANT
27200 C
27300 C ARRANGE FLAG IF FUEL FLOW OUTSIDE BLOCK DATA LIMITS
27400 C FUEL FLOW=2.5-65.0 LB/HR
27500      COMMON ICD(17,6),IC(23,2),IED(17,5)
27600      IF (FF.LT.2.5.OR.FF.GT.65.0) GOTO 30
27700      IJ = FF * 100.0
27800      DO 10 J = 1,23
27900      IF(IJ.LT.IC(J,1))GOTO 20
28000 10      CONTINUE
28100      GOTO 30
28200      20 J=J-1
28300 C LINEAR INTERPOLATION FROM BLOCK DATA
28400      AI=FLOAT(IC(J,2)) /1000.0
28500      AI1 = FLOAT(IC(J+1,2)) / 1000.0
28600      HC=AI+(AI-AI1) * (FLOAT(IC(J,1))-100.*FF)/FLOAT(IC(J+1,1))\
      \-IC(J,1))
28700      RETURN
28800      30 WRITE(3,40)FF
28900      40 FORMAT (16H INVALID REQUEST/ 6H FF = ,E12.6)
29000      HC = 0.0
29100      RETURN
29200      END

```

Standard Engine Program


```

29300      SUBROUTINE GG (RT, TM, G)
29400 C      *****
29500 C
29600 C DETERMINES VALUE FOR GAMMA
29700 C
29800      COMMON IC(17,6), IDD(23,2), IED(17,5)
29900 C ARRANGE FLAG IF A/F RATIO OR MEAN TEMP
30000 C OUTSIDE BLOCK DATA LIMITS
30100 C TEMP=400-3600 DEG R
30200 C AIR/FUEL RATIO=20-40
30300      I = RT / 5.0 - 2.0
30400      IF (RT.LE.0.01) I = 1
30500      IF (I.GT.6) GOTO 10
30600      IF (J.EQ.17) GOTO 30
30700      A = TM / 200.0 - 1.0
30800      J = A
30900      IF (J.LT.1.OR.J.GT.17) GOTO 10
31000 C LINEAR INTERPOLATION FROM BLOCK DATA
31100      AI = FLOAT(IC(J,I)) / 10000.0
31200      AI1 = FLOAT(IC(J+1,I)) / 10000.0
31300      G = AI + (AI - AI1) * (FLOAT(J) - A)
31400      RETURN
31500 30 G = FLOAT(IC(J,I)) / 10000.
31600      RETURN
31700 10 WRITE (3,20) RT, TM
31800 20 FORMAT (16H INVALID REQUEST/ 6H RT = ,E12.6/ 6H TM = ,E12.6)
31900      G = 0.0
32000      RETURN
32100      END

```

Standard Engine Program

```

32200          BLOCK DATA
32300 C          *****
32400 C
32500 C IE(17,5)=CP ID(23,2)=HC IC(17,6)=GAMMA
32600 C TAKEN FROM WALLACE CHARACTERISTICS
32700 C
32800          COMMON IC(17,6), ID(23,2), IE(17,5)
32900          DATA(IE(1,1), I=1, 17)/2470, 2518, 2580, 2657, 2737, 2804, 2880,
33000          *2940, 3010, 3070, 3120, 3165, 3205, 3230, 3255, 3280, 3300/
33100          DATA(IE(1,2), I=1, 17)/2460, 2497, 2553, 2625, 2700, 2773, 2842,
33200          *2901, 2960, 3018, 3070, 3110, 3147, 3170, 3200, 3225, 3235/
33300          DATA(IE(1,3), I=1, 17)/2445, 2480, 2535, 2600, 2673, 2750, 2810,
33400          *2873, 2927, 2982, 3030, 3070, 3105, 3130, 3153, 3170, 3190/
33500          DATA(IE(1,4), I=1, 17)/2438, 2468, 2520, 2580, 2655, 2730, 2790,
33600          *2852, 2908, 2955, 3008, 3050, 3080, 3105, 3125, 3148, 3155/
33700          DATA(IE(1,5), I=1, 17)/2430, 2463, 2513, 2573, 2645, 2715, 2780,
33800          *2840, 2895, 2943, 2990, 3030, 3070, 3090, 3112, 3135, 3140/
33900          DATA(ID(1,1), I=1, 23)/250, 375, 500, 625, 750, 875, 1000, 1125, 1250,
34000          *1375, 1500, 1750, 2000, 2250, 2500, 3000, 3500, 4000, 4500, 5000, 5500,
34100          *6000, 6500/
34200          DATA(ID(1,2), I=1, 23)/400, 330, 283, 255, 232, 214, 200, 190, 182,
34300          *175, 170, 160, 152, 147, 140, 130, 125, 119, 116, 114, 112, 109, 109/
34400          DATA (IC(1,1), I=1, 17)/14030, 13975, 13900, 13805, 13700, 13565,
34500          *13450, 13360, 13285, 13220, 13165, 13110, 13070, 13045, 13015,
34600          *12990, 12975/
34700          DATA (IC(1,2), I=1, 17)/13840, 13745, 13625, 13480, 13345, 13225\
\, 13120,
34800          *13030, 12945, 12875, 12810, 12760, 12720, 12685, 12660, 12635, 12610/
34900          DATA (IC(1,3), I=1, 17)/13870, 13875, 13675, 13540, 13395, 13280\
\, 13175,
35000          *13085, 13000, 12940, 12880, 12825, 12785, 12750, 12725, 12700, 12685/
35100          DATA (IC(1,4), I=1, 17)/13895, 13810, 13710, 13580, 13445, 13330\
\, 13215,
35200          *13125, 13050, 12980, 12920, 12875, 12830, 12800, 12775, 12775, 12735/
35300          DATA (IC(1,5), I=1, 17)/13910, 13840, 13735, 13610, 13480, 13360\
\, 13250,
35400          *13160, 13070, 13010, 12950, 12900, 12860, 12835, 12805, 12780, 12770/
35500          DATA (IC(1,6), I=1, 17)/13930, 13855, 13750, 13630, 13500, 13380\
\, 13270,
35600          *13170, 13090, 13025, 12970, 12925, 12875, 12845, 12820, 12795, 12780/
35700          END

```

Standard Engine Program

```

100 C DATA FILE TADAT
200 C .....
300 C CONTAINS DATA FOR STANDARD ENGINE PROGRAM
400 C .....
500 C (KEEP/MILO)
600 C ATMOS PRESSURE
700   0.986
800 C ENG SPD,MAN TEMP,SCAV,A/F RATIO,BOOST
900   1000.0,34.59,1.53,19.45,1.1
1000  1200.0,40.20,1.57,19.75,1.1
1100  1400.0,46.54,1.61,19.99,1.1
1200  1600.0,53.12,1.65,20.18,1.1
1300  1800.0,62.15,1.63,22.02,1.18
1400  2000.0,69.95,1.52,22.28,1.23
1500  2200.0,79.22,1.56,22.89,1.3
1600  2400.0,88.73,1.48,24.52,1.41

```

Data for Standard Engine Program

A1.2 Preliminary Matching Program

The program used for indicating likely matched areas of turbocharged operation was developed by Blencoe (17).

The program is based on the calculation of the internal energy of the cylinder contents at blowdown. The program has the facility to predict the effects of auxiliary combustion (as a simple series system) and makes use of several subroutines. The program notation and a complete listing follows.

Notation

AE	Charge cooler effectiveness
AMT	Trapped air mass
CE	Turbo compressor efficiency
CP	Compressor power
CW	Compressor work
FX	Fraction of theoretical air
HB	Specific enthalpy
HIS	Enthalpy after isentropic process
PA	Ambient pressure
PC	Pressure at compressor outlet
PDC	Pressure drop coefficient
PE	Exhaust pressure
PM	Air manifold pressure
RC	Compressor pressure ratio
RCC	Overall boost ratio
REXT	Turbine pressure ratio
RM	Limiting exhaust pressure ratio
RS	Scavenge blower pressure ratio
SE	Scavenge blower efficiency
TA	Ambient temperature
TE	Turbine efficiency
TEX	Exhaust temperature
TG	Exhaust blowdown temperature
UGB	Internal energy
VEC	Cylinder volume (exhaust port closing)
VEO	Cylinder volume (exhaust port opening)
VIC	Cylinder volume (inlet port closing)

```

100 $RESET FREE
200 C PRELIMINARY MATCHING PROGRAM
300 C *****
400 C
500 FILE 3=OUT,UNIT=PRINTER
600 FILE 11(KIND=DISK,TITLE="PREDAT",FILETYPE=7)
700 COMMON TAB1(30,3),TAB7(35,3),ITAB1,ITAB7,IP,D,CRL,S,R
800 IC = 11
900 IP = 3
1000 C INPUT DATA FOR TABLES
1100 C FROM "GAS TABLES" BY KEENAN AND KAY
1200 READ(IC,/)ITAB1
1300 DO 10 I = 1,ITAB1
1400 10 READ(IC,/)TAB1(1,1),TAB1(1,2),TAB1(1,3)
1500 READ(IC,/)ITAB7
1600 DO 20 I = 1,ITAB7
1700 20 READ(IC,/)TAB7(1,1),TAB7(1,2),TAB7(1,3)
1800 C INPUT DATA FOR ENGINE CONDITIONS
1900 C AND PROGRAM PARAMETERS
2000 READ(IC,/)ESA,ESB,PA,TA,CE,TE,AE
2100 READ(IC,/)RCA,RCB,SRA,SRB,TGA,TGB
2200 READ(IC,/)CV,DAL
2300 READ(IC,/)CRL,S,D,R
2400 READ(IC,/)THETEO,THETEC,THETIC
2500 READ(IC,/)VEO,VEC,VIC
2600 READ(IC,/)IY,NES,NRC,NSR,NTG,ISCC
2700 C OUTPUT PARAMETERS OF PROGRAM
2800 WRITE(IP,7)ESA,ESB,PA,TA,CE,TE,AE,RCA,RCB,SRA,SRB,TGA,TGB,CV\
\,CRL,
2900 1S,D,R,THETEO,THETEC,THETIC,VEO,VEC,VIC,IY,NES,NRC,NSR,NTG\
\,DAL,ISCC
3000 7 FORMAT(1H1,15H INPUT DATA//4X,6HESA = ,F10.2/4X,6HESB = \
\F10.2
3100 1/4X,6H PA = ,F10.2/4X,6H TA = ,F10.2/4X,6H CE = ,F10.2/4X,6H\
\ TE =
3200 2,F10.2/4X,6H AE = ,F10.2/4X,6HRCA = ,F10.2/4X,6HRCB = ,F10.2\
\ /4X,6H
3300 3SRA = ,F10.2/4X,6HSRB = ,F10.2/4X,6HTGA = ,F10.2/4X,6HTGB = \
\F10.2
3400 4/4X,6H CV = ,F10.2/4X,6HCRL = ,F10.2/4X,6H S = ,F10.2/4X,6H\
\ D =
3500 5,F10.2/4X,6H R = ,F10.2/10H THETEO = ,F10.2/10H THETEC = \
\F10.2/
3600 610H THETIC = ,F10.2/4X,6HVEO = ,F10.2/4X,6HVEC = ,F10.2/4X\
\ ,6HVIC =
3700 7 ,F10.2/4X,6H IY = ,12/4X,6HNES = ,12/4X,6HNRC = ,12/4X\
\ ,6HNSR = ,
3800 812/4X,6HNTG = ,12/4X,6HDAL = ,F10.3/4X,7HISCC = ,12/)
3900 IFLAG=1
4000 K=0

```

Preliminary Matching Program

```

4100      BD=1.0
4200 C INPUT AIR/FUEL RATIO,SCAVENGE BLOWER PRESSURE RATIO
4300 C AND SCAVENGE BLOWER EFFICIENCY
4400      50 READ(10,/)RT,RS,SE
4500      IF(RT.EQ.0.0)CALL EXIT
4600      J=0
4700      ES=ESA-ESB
4800 C CYCLE ENGINE SPEED
4900      DO 1000 KA=1,NES
5000      ES=ES+ESB
5100      RC=RCA-RCB
5200 C CYCLE AIR/FUEL RATIO
5300      DO 1000 KB=1,NRC
5400      RC=RC+RCB
5500      SR=SRA-SRB
5600 C CYCLE SCAVENGE RATIO
5700      DO 1000 KC=1,NSR
5800      SR=SR+SRB
5900      TG=TGA+TGB
6000 C CYCLE BLOWDOWN TEMPERATURE
6100      DO 1000 KD=1,NTG
6200      TG=TG-TGB
6300      J=J+1
6400 C WRITE/NOT WRITE MAIN HEADING
6500      IF(J.EQ.1) WRITE(10,5)ES
6600      5  FORMAT(1H1,15X,6H ES = ,F7.1///5X,2HPA,6X,2HPM,6X,2HPE,6X\
        \,4HREXT,
        \,14X,2HTA,5X,2HTG,6X,3HTEX,6X,2HT6,7X,2HSR,5X,2HRT,7X,2HRC,6X\
        \,2HRS,
        \,24X,2HEM,4X,2HEM/)
6800
6900 C DETERMINE THE FRACTION OF THEORETICAL AIR
7000      FX = 14.5/RT
7100      T  =(TG -32.0)/180.0
7200 C CALCULATE INTERNAL ENERGY
7300 C OF EXHAUST GAS AT BLOWDOWN
7400      UGB = ((-(0.1755+ 0.0873*FX**0.75))*T
7500      1      +(13.98 + 6.048*FX**0.8))*T
7600      2      +(881.3 + 83.52*FX**0.93))*T + 2282.0
7700 C CALCULATES CYLINDER VOLUME AT GIVEN CRANK ANGLE
7800      IF(VEC.EQ.0.0)VEC=FUNC(THETEC)
7900      IF(VIC.EQ.0.0)VIC=FUNC(THETIC)
8000      IF(VEO.EQ.0.0)VEO=FUNC(THETEO)
8100 C FIND AIR MANIFOLD CONDITIONS
8200      T1 = TA +460.0
8300      CALL TABL1(T1,H1,PREL1,1)
8400      PREL2 = PREL1 * RC
8500 C ROUTING FOR SCAVENGE BLOWER SECTION
8600      IF(RS.NE.1.0)GO TO 500
8700      CALL TABL1(T2,HIS2,PREL2,3)
8800      H2 = (HIS2 - H1)/CE +H1

```

[18]

[19]

[20]

[21]

Preliminary Matching Program

```

8900 C MODIFY MANIFOLD CONDITIONS
9000 C IF AIR COOLER FITTED
9100     H3 = H2 - AE*(H2-H1) [22]
9200     PM=PA*RC [23]
9300     CALL TABL1(T3,H3,PTEM,2)
9400 C *****
9500 C
9600 C SECTION FOR SCAVENGE BLOWER CALCULATIONS
9700 C (IF FITTED)
9800     GO TO 600
9900     500 CALL TABL1(TIS2,HIS2,PREL2,3)
10000     T2=(TIS2-T1)/CE+T1
10100     CALL TABL1(T2,H2,PREL22,1)
10200     PREL3=PREL22*RS
10300     CALL TABL1(TIS3,HIS3,PREL3,3)
10400     T3=(TIS3-T2)/SE+T2
10500     T3=T3-AE*(T3-T1)
10600     PC=PA*RC
10700     PM=PC*RS
10800 C *****
10900 C
11000     600 CONTINUE
11100     AMT =(PM*VIC)/(53.3*T3*12.0) [24]
11200 C CALCULATE ENGINE FUELLING AND COMPRESSOR WORK [25]
11300     EM=3.0*AMT*ES/RT
11400     CW = AMT * SR*(H2-H1)
11500 C EXHAUST MANIFOLD CONDITIONS FOUND BY COMBINING
11600 C ENTHALPIES OF EXHAUST GAS AT BLOWDOWN AND
11700 C SCAVENGE AIR
11800 C DIFFERENT EQTNS FOR 2 OR 4 STROKE
11900     IF(IY.EQ.2)GOTO 30
12000     H5 = ((SR-1.0)*H3 + UGB*(1.0+1.0/RT))/23.97
12100     1 + PM*(VEO-VEC)/(AMT*12.0*778.0))/(SR +1.0/RT) [29]
12200     GOTO 40
12300     30 H5 = ((SR-1.0)*H3 +UGB*(1.0+ 1.0/RT))/28.97 [28]
12400     1/(SR+1.0/RT)
12500 C TURBINE INLET + OUTLET CONDITIONS USING TABLE 7
12600     40 H6 = H5 - CW/(AMT*(SR + 1.0/RT)) [30]
12700     HB5 = H5 * 28.97
12800     HB6 = H6 * 28.97
12900     HB7 = HB5 - (HB5-HB6)/TE
13000     CALL TABL7(TEX,HB5,PREL5)
13100     CALL TABL7(T6 ,HB6,PREL6)
13200     CALL TABL7(T7 ,HB7,PREL7)
13300     REXT = PREL5/PREL7
13400     PE = REXT*PA
13500     CP = CW*ES*788.0/33000.0
13600     IF(IY.EQ.4)CP = CP/2.0
13700     TEX=TEX-460.0
13800     T6 = T6 -460.0

```

Preliminary Matching Program


```

13900      RCC=PM/PA
14000      RM=0.97*RCC
14100 C SKIP SECTION IF NO AUXILIARY COMBUSTION
14200      IF(ISCC.EQ.0)GO TO 60
14300      IF(K.NE.0)GO TO 110
14400 C TEST FOR POSSIBLE MATCHING, IF LIKELY PRINT OUTPUT
14500 C IF NOT INCREASE TURB INLET ENTHALPY BY USE OF
14600 C ACC IN EXHAUST LINE
14700      IF(REXT.LE.RM)GO TO 60
14800      TEXTS=TEX
14900      110 IF(IFLAG.EQ.0)GO TO 70
15000      H5S=H5
15100      IFLAG=0
15200      70 DA=REXT-RM
15300 C IF MATCHING ATTAINED, CALCULATE+PRINT OUTPUT
15400 C IF NOT INCREASE TURB INLET ENTHALPY
15500 C AND RE-CALCULATE
15600      IF(ABS(DA).LE.DAL)GO TO 80
15700      A=100.0
15800      IF(ABS(DA).LT.0.50) A=75.0
15900      IF(ABS(DA).LT.0.30) A=50.0
16000      IF(ABS(DA).LT.0.20) A=25.0
16100      IF(ABS(DA).LT.0.10) A=10.0
16200      IF(DA.GE.0.0)GO TO 90
16300      BD=BD/2.0
16400      A=-A
16500 C ARRANGE WARNING FOR EXCESSIVE LOOPING
16600      IF(K.EQ.40)GO TO 99
16700      GO TO 90
16800      99 WRITE(IP,2000)
16900      2000 FORMAT(22H CANNOT MATCH AS BELOW)
17000      GO TO 80
17100      90 H5=H5+A*BD
17200      K=K+1
17300      GO TO 40
17400      80 AM=ANT*(SR+1.0/RT)
17500      Q=(H5-H5S)*3.0*AM*ES
17600      FM=Q/CV
17700 C *****
17800 C
17900 C SECTION FOR SI UNIT CONVERSION AND OUTPUT
18000 C IF ACC OPERATIVE
18100      PZ=PA/14.5
18200      PM=PM/14.5
18300      PE=PE/14.5
18400      TZ=(TA-32.0)/1.8
18500      TY=(TG-32.0)/1.8
18600      TEXTS=(TEXTS-32.0)/1.8
18700      T6=(T6-32.0)/1.8
18800      EM=EM*27.273

```

[31]

[32]

Preliminary Matching Program

```

18900      FM=FM*27.273
19000      WRITE(IP,5)PZ,PM,PE,REXT,TZ,TY,TEXS,T6,SR,RT,RC,RS,EM,FM
19100 C *****
19200 C
19300      GO TO 100
19400 C *****
19500 C
19600 C SECTION FOR SI UNIT CONVERSION+OUTPUT
19700 C IF ACC INOPERATIVE
19800      60 PZ=PA/14.5
19900      PM=PM/14.5
20000      PE=PE/14.5
20100      TZ=(TA-32.0)/1.8
20200      TY=(TG-32.0)/1.8
20300      TEX=(TEX-32.0)/1.8
20400      T6=(T6-32.0)/1.8
20500      EM=EM*27.273
20600      WRITE(IP,6)PZ,PM,PE,REXT,TZ,TY,TEX,T6,SR,RT,RC,RS,EM
20700 C *****
20800 C
20900      6 FORMAT(1H ,12F8.2,2F6.3)
21000      100 CONTINUE
21100      K=0
21200      IFLAG=1
21300      BD=1.0
21400 C START NEW PAGE FOR NEXT RUN THROUGH PROGRAM
21500      IF(KD.EQ.NTG)WRITE(IP,9)
21600      9 FORMAT(1H ,//)
21700      IF(J.LT.200)GO TO 1000
21800      J=0
21900      1000 CONTINUE
22000      GO TO 50
22100      END

```

Preliminary Matching Program

```

22200      SUBROUTINE TAB1(T,H,P,N)
22300 C      *****
22400 C
22500 C      INTERPOLATING ROUTINE FOR AIR MANIFOLD PARAMETERS
22600 C      KEENAN AND KAY GAS TABLES (NO 1)
22700 C      INCORPORATES WARNINGS FOR VALUES
22800 C      OUTSIDE DATA FILE LIMITS
22900 C
23000      COMMON TAB1(30,3),TAB7(35,3),ITAB1,ITAB7,IP,D,CRL,S,R
23100      IF(N.EQ.3)GOTO 80
23200      IF(N.EQ.2)GOTO 10
23300      IF(T.LE.TAB1(1,1))GOTO 1001
23400      DO 20 J = 2,ITAB1
23500 20      IF(T.LE.TAB1(J,1))GOTO 30
23600      J=ITAB1
23700      GOTO 100
23800 30      RATIO = (T - TAB1(J-1,1))/(TAB1(J,1)-TAB1(J-1,1))
23900      H = TAB1(J-1,2) +(TAB1(J,2) - TAB1(J-1,2))*RATIO
24000      P = TAB1(J-1,3) +(TAB1(J,3) - TAB1(J-1,3))*RATIO
24100 70      RETURN
24200 10      IF(H.LE.TAB1(1,2))GOTO1001
24300      DO 50 J = 2,ITAB1
24400 50      IF(H.LE.TAB1(J,2))GOTO60
24500      J=ITAB1
24600      GOTO 100
24700 60      RATIO = (H - TAB1(J-1,2))/(TAB1(J,2)-TAB1(J-1,2))
24800 120     T = TAB1(J-1,1) +(TAB1(J,1) - TAB1(J-1,1))*RATIO
24900      GOTO 70
25000 80      IF(P.LE.TAB1(1,3))GOTO 1001
25100      DO 90 J = 2,ITAB1
25200 90      IF( P.LE.TAB1(J,3))GOTO 110
25300      J=ITAB1
25400      GOTO 100
25500 110     RATIO =(P - TAB1(J-1,3))/(TAB1(J,3)-TAB1(J-1,3))
25600      H = TAB1(J-1,2) +(TAB1(J,2)- TAB1(J-1,2))*RATIO
25700      GOTO 120
25800 1001    J=2
25900 100     GOTO(1,2,3),N
26000 1      WRITE(IP,11) T
26100 11     FORMAT(26H T VALUE OUTSIDE TABLE T =,F15.4)
26200      GOTO30
26300 2      WRITE(IP,12) H
26400 12     FORMAT(26H H VALUE OUTSIDE TABLE H =,F15.4)
26500      GOTO60
26600 3      WRITE(IP,13) P
26700 13     FORMAT(26H P VALUE OUTSIDE TABLE P =,F15.4)
26800      GOTO110
26900      END

```

Preliminary Matching Program

```

27000      SUBROUTINE TABL7(T,H,P)
27100 C      *****
27200 C
27300 C INTERPOLATING ROUTINE FOR EXHAUST PARAMETERS
27400 C KEENAN AND KAY GAS TABLES (NO 7)
27500 C INCORPORATES WARNINGS FOR VALUES
27600 C OUTSIDE DATA FILE LIMITS
27700 C
27800      COMMON TAB1(30,3),TAB7(35,3),ITAB1,ITAB7,IP,D,CRL,S,R
27900      IF(H.LE.TAB7(1,2))GOTO 1001
28000      DO 10 J = 2,ITAB7
28100 10     IF(H.LE.TAB7(J,2))GOTO 20
28200      J=ITAB7
28300      GOTO 100
28400 20     RATIO = (H- TAB7(J-1,2))/(TAB7(J,2)-TAB7(J-1,2))
28500      T = TAB7(J-1,1) +(TAB7(J,1)-TAB7(J-1,1))*RATIO
28600      P = TAB7(J-1,3) +(TAB7(J,3)-TAB7(J-1,3))*RATIO
28700      RETURN
28800 1001   J=2
28900 100   WRITE(IP,1)H
29000 1     FORMAT(30H H VALUE OUTSIDE TABLE 7  H = ,F15.4)
29100      GOTO20
29200      END
29300 C
29400 C
29500      FUNCTION FUNC(TH)
29600 C      *****
29700 C
29800 C CYLINDER VOLUME FUNCTION
29900 C CALCULATES CYLINDER VOLUME AT GIVEN
30000 C CRANK ANGLE FROM STROKE,BORE
30100 C CONN ROD LENGTH AND COMPRESSION RATIO
30200 C
30300      COMMON TAB1(30,3),TAB7(35,3),ITAB1,ITAB7,IP,D,CRL,S,R
30400      THC = COS(TH* 3.141592/180.)
30500      FUNC= .785398*D*D*(CRL+ S*(1.0-THC))/2.0
30600      1 - SQRT(CRL*CRL + S*(THC*THC-1.0)/4.0)+S/R)
30700      RETURN
30800      END

```

Preliminary Matching Program

```

100 C DATA FILE PREDAT
200 C *****
300 C CONTAINS DATA FOR PRELIMINARY MATCHING PROGRAM
400 C *****
500 C
600 C NUMBER OF DATA LINES FOR TABLE 1
700 C PROPERTIES OF AIR AT LOW PRESSURE
800 C FOR ONE POUND (KEENAN AND KAY)
900 16
1000 C DATA LINES FOR TABLE 1
1100 C TEMP, ENTHALPY, REL PRESSURE
1200 450.0, 107.5, 0.7329
1300 500.0, 119.48, 1.059
1400 550.0, 131.46, 1.4779
1500 600.0, 143.47, 2.005
1600 650.0, 155.5, 2.655
1700 700.0, 167.56, 3.446
1800 750.0, 179.66, 4.396
1900 800.0, 191.81, 5.526
2000 850.0, 204.01, 6.856
2100 900.0, 216.26, 8.411
2200 950.0, 228.58, 10.216
2300 1000.0, 240.98, 12.298
2400 1050.0, 253.45, 14.686
2500 1100.0, 265.99, 17.413
2600 1150.0, 278.61, 20.51
2700 1200.0, 291.3, 24.01
2800 C NUMBER OF DATA LINES FOR TABLE 7
2900 C PROPERTIES OF PRODUCTS OF COMBUSTION
3000 C 200 PER CENT THEORETICAL AIR
3100 C FOR ONE POUND MOLE (KEENAN AND KAY)
3200 35
3300 C DATA LINES FOR TABLE 7
3400 C TEMP, ENTHALPY, REL PRESSURE
3500 300.0, 2096.7, 0.1677
3600 350.0, 2448.5, 0.2896
3700 400.0, 2801.4, 0.4655
3800 450.0, 3155.6, 0.7085
3900 500.0, 3511.2, 1.033
4000 550.0, 3867.9, 1.455
4100 600.0, 4226.3, 1.992
4200 650.0, 4586.1, 2.662
4300 700.0, 4947.7, 3.487
4400 750.0, 5310.9, 4.487
4500 800.0, 5676.3, 5.690
4600 850.0, 6043.6, 7.120
4700 900.0, 6413.0, 8.808
4800 950.0, 6784.9, 10.787
4900 1000.0, 7159.8, 13.089
5000 1050.0, 7536.8, 15.754

```

Data for Preliminary Matching Program

5100 1100.0, 7916.4, 18.822
5200 1150.0, 8298.7, 22.34
5300 1200.0, 8683.6, 26.34
5400 1250.0, 9071.4, 30.9
5500 1300.0, 9461.7, 36.05
5600 1350.0, 9854.8, 41.86
5700 1400.0, 10250.7, 48.38
5800 1450.0, 10649.2, 55.7
5900 1500.0, 11050.2, 63.88
6000 1550.0, 11453.6, 72.98
6100 1600.0, 11859.6, 83.10
6200 1650.0, 12268.0, 94.3
6300 1700.0, 12678.6, 106.7
6400 1750.0, 13091.7, 120.38
6500 1800.0, 13507.0, 135.43
6600 1850.0, 13924.4, 151.96
6700 1900.0, 14344.1, 170.09
6800 1950.0, 14765.9, 189.95
6900 2000.0, 15189.3, 211.60
7000 C ENGINE AND PROGRAM DETAILS
7100 C ESA,ESB,PA,TA,ETA C,ETA T,AE
7200 1000.0,200.0,14.7,65.0,0.75,0.85,0.0
7300 C RCA,RCB,SRA,SRB,TGA,TGB
7400 1.1,0.3,1.2,0.2,2192.0,180.0
7500 C CAL VALUE,DAL
7600 19500.0,0.005
7700 C CRL,S,D,R
7800 0.0,0.0,0.0,0.0,
7900 C THETEO,THETEC,THETIC
8000 0.0,0.0,0.0
8100 C VEO,VEC,VIC
8200 0.01,0.01,55.9
8300 C IY,NES,NRC,NSR,NTG,ISCC
8400 2,1,5,4,10,0
8500 C A/F RATIO,SB P RATIO, SB EFFICIENCY
8600 20.0, 1.3, 0.60
8700 20.0, 1.00, 1.00
8800 25.0, 1.00, 1.00
8900 30.0, 1.00, 1.00
9000 35.0, 1.00, 1.00
9100 40.0, 1.00, 1.00
9200 45.0, 1.00, 1.00
9300 50.0, 1.00, 1.00
9400 55.0, 1.00, 1.00
9500 60.0, 1.00, 1.00
9600 C SET TO ZERO TO END PROGRAM RUN
9700 0.0,0.0,0.0

Data for Preliminary Matching Program

A1.3 Diesel Gas Turbine Program

The program was used to predict the performance of the diesel gas turbine. A modified version of the program shown in Appendix 1.1 was used to predict engine performance and subroutines were added to predict the performance of the other components in the system.

The notation is the same as that given in Appendix 1.1 (standard engine program) with the following additions.

Notation

AFEN	Engine air/fuel ratio
DELP	Pressure drop across engine
DEMT	Work demanded from turbine
EE	Charge cooler effectiveness
EF	Turbo compressor efficiency
EHT	Heat exchanger effectiveness
EMEC	Turbocharger mechanical efficiency
ETAT	Turbine efficiency
ETT	Turbine work factor
FACT	Error work factor
HCIN	Enthalpy at auxiliary combustion chamber inlet
HENG	Enthalpy of engine exhaust
HIN	Enthalpy at turbine inlet
HO	Enthalpy at turbine exhaust
HPC	Compressor power requirement
HPT	Turbine power
P	Pulse work
PREST	Preston meter manometer reading (air flow)
QCC	Heat addition during auxiliary combustion
RST	Turbine expansion ratio
SFD	Total system specific fuel consumption
SUPT	Work supplied by turbine
TCCIN	Auxiliary combustion chamber inlet temperature
TCOOL	Temperature of coolant to charge cooler
TSUP	Temperature of air at compressor delivery
TT	Engine exhaust temperature
TTIN	Turbine inlet temperature

TTUE	Turbine outlet temperature
WAP	Compressor air flow
WENG	Mass flow through engine
WFACC	Auxiliary fuel flow
WGAS	Auxiliary fuel consumption
WT	Mass flow through turbine

```

100 $RESET FREE
200 $SET AUTOBIND
300 $BIND = FROM (L)B00261/A, (L)B00261/B, (L)B00632/=
400 C
500 C PROGRAM DGT
600 C *****
700 C
800 C PREDICTS DIESEL GAS TURBINE PERFORMANCE
900 FILE 1(KIND=REMOTE)
1000 FILE 3(KIND=PRINTER,MAXRECSIZE=22)
1100 FILE 4(KIND=PRINTER,MAXRECSIZE=22)
1200 FILE 9(KIND=DISK,FILETYPE=7,TITLE="ENGIN2.")
1300 FILE 12(KIND=DISK,TITLE="PLOTENG")
1400 FILE 10(KIND=DISK,FILETYPE=7,TITLE="ENGIN1.")
1500 FILE 11(KIND=DISK,TITLE="SUPDAT",FILETYPE=7)
1600 COMMON IC(17,6), ID(23,2), IE(17,5)
1700 COMMON /C/TA,PA,RS,EF,EE,TCOOL,ES,ST,RT1,DELP,ETAT,EHT
1800 *,WAF,SPD,WA,TSUP,HPC,TM,RC,WENG,RST,HPT,WT,THA,IFP
1900 *,P3,TTUR,TTUN,ET,TTUE,WCC,CPE,CPC,COCN,TCCIN,WFACC
2000 *,WGAS,SFC
2100 C SET UP CONSTANTS USED IN PROGRAM
2200 READ(11,/)NES,IFP
2300 ET=1.0;VCR=0.0;S=0.78;VS=0.126;CRA=16.0;CRB=0.0;PMAXL=1800.0
2400 READ(11,/)PA,TA,EE,TCOOL,EHT,MAPN,THA,IGRAPH
2500 C READ IN COMPRESSOR DATA
2600 CALL INTAR(MAPN)
2700 C CONVERT BAR TO PSI
2800 PA=PA*14.5
2900 C CONVERT DEG C TO DEG R
3000 TA=((TA*1.8)+459.67+32.0)
3100 TCOOL=((TCOOL*1.8)+459.67+32.0)
3200 CV=18060.0;CA=144.0;CB=0.0065;WHC=778.0;HPD=33000.0;HPCA=42\
    \.42
3300 PDC=0.86;R=53.3;TLA=5.0;TLB=5.0;TLC=5.0;TLD=50.0
3400 C OUTPUT PROGRAM PARAMETERS
3500 WRITE(3,5)ET,VCR,S,VS,CRA,CRB,PMAXL,PA,CV,CA,CB,WHC,HPD,HPCA,
3600 *PDC,R,TLA,TLB,TLC,TLD
3700 WRITE(3,6)TA,EE,TCOOL,EHT,MAPN,THA,IGRAPH
3800 5 FORMAT(13H1 INPUT DATA //
3900 *7H ET = ,F10.3/7H VCR = ,F10.3/7H S = ,F10.3/7H VS = \
    \,F10.3/
4000 *7H CRA = ,F10.3/7H CRB = ,F10.3/9H PMAXL = ,F10.3/7H PA = \
    \,F10.3/
4100 *7H CV = ,F10.3/7H CA = ,F10.3/7H CB = ,F10.3/7H WHC = \
    \,F10.3/
4200 *7H HPC = ,F10.3/8H HPCA = ,F10.3/7H PDC = ,F10.3/7H R = \
    \,F10.3/
4300 *7H TLA = ,F10.3/7H TLB = ,F10.3/7H TLC = ,F10.3/7H TLD = \
    \,F10.3)
4400 6 FORMAT(7H TA = ,F10.3/7H EE = ,F10.3/9H TCOOL = ,F10.3/

```

Diesel Gas Turbine Program

```

4500      *7H EHT = ,F10.3/9H  MAPN = ,13/7H  THA = ,F10.3/10H  IGRAPH = \
         \,13/
4600      *7H1 TRACE      )
4700      IF (MAPN.EQ.2) GO TO 53
4800      WRITE(3,52)THA
4900      WRITE(1,52)THA
5000      52 FORMAT(/19H TURBOCHARGER 3LD1 ,F10.3)
5100      GO TO 55-
5200      53 WRITE(3,54)THA
5300      WRITE(1,54)THA
5400      54 FORMAT(/19H TURBOCHARGER 4LGK ,F10.3)
5500 C RUN PROGRAM FOR THE SET NO OF ENGINE SPEEDS
5600      55 DO 180 KA=1,NES
5700      I=0-
5800 C INPUT ENGINE CONDITION DATA
5900      READ(11,/)ES,ST,RT1,RC,PREST,DELP,TURN,TTUE1
6000      149 DELP=DELP*14.5
6100 C CONVERT DEG C TO DEG R
6200      TTUE1=(TTUE1*1.8)+491.67
6300      TURN=(TURN*1.8)+491.67
6400 C FIND AIR FLOW FROM PRESTON METER MANOMETER RDG
6500      CALL PMETER(PREST,WAP)
6600 C PROGRAM RESTRAINT A/F RATIO 20-40
6700      IF (RT.LT.20.0)RT=20.0
6800 C SET INITIAL ETA C
6900      EF=60.0
7000      ICO=0
7100 C FIND ETA C AND AIRFLOW FROM COMP TABLES
7200 C ITERATE UNTIL A SUITABLE VALUE IS FOUND
7300 C *****
7400      78 CALL COMPRE
7500      ICO=ICO+1
7600      IF (ABS(WAP-WA).LT.0.03) GO TO 79
7700      IF (MAPN.EQ.2) GO TO 81
7800      IF (EF.GT.69.5) GO TO 79
7900      81 IF (EF.GT.77.5) GO TO 79
8000      IF ((WAP-WA).LT.0.0) EF=EF+0.5
8100      IF ((WAP-WA).GT.0.0) GO TO 79
8200      IF (ICO.GT.40) GO TO 79
8300      HPC1=HPC
8400      EF1=EF
8500      TM1=TM
8600      GO TO 78
8700      79 CONTINUE
8800      WA=WAP
8900      HPC=HPC1
9000      EF=EF1
9100      TM=TM1
9200      SR=ST
9300      RT=RT1
9400 C *****

```

Diesel Gas Turbine Program

```

9500 C IF SCAVENGE TOO LOW FOR GOOD COMBUSTION,
9600 C ITERATE WITH PROGRESSIVE A/F INCREASE
9700     IF(SR.GT.1.220) GO TO 19
9800     RT=RT-0.5
9900     DO 180 ITIO=1,15
10000    RT=RT+0.5
10100    19 I=I+1
10200    PHI=0.9
10300    ED=0.72
10400 C WRITE/NOT WRITE HEADING
10500    IF(1.EQ.1) GO TO 11
10600    12 DTK=ED*PHI*WHC
10700    GE=1.3
10800    GC=1.35
10900 C ROUTING FOR VARIABLE COMPRESSION RATIO
11000    IF(VCR.NE.0.0) GO TO 30
11100    CR=CRA
11200 C *****
11300 C ITERATE FOR PMAX TOLERANCE (TLA)
11400 C
11500 10 PMAX =(GE-1.0)*RC*PA*CR/(1.0-1.0/CR**(GE-1.0))*((DTK*CV/R\
- \sqrt{RT/TM}*
11600    *1.0-1.0/CR**0.4))-1.0/(GC-1.0)*(1.0-CR**(GC-1.0))) [4]
11700    TME=TM*PMAX*(1.0+1.0/CR**(GE-1.0))/RC/PA/CR/2.0 [6]
11800 C FIND GAMMA EXPANSION
11900    CALL GG (RT,TME,GE)
12000    TMC=TM*(CR**(GC-1.0)+1.0)/2.0 [7]
12100 C FIND GAMMA COMPRESSION
12200    CALL GG (0.0,TMC,GC)
12300    PMAXA=(GE-1.0)*RC*PA*CR/(1.0-1.0/CR**(GE-1.0))*((DTK*CV/R\
\sqrt{RT/TM}*
12400    *1.0-1.0/CR**0.4))-1.0/(GC-1.0)*(1.0-CR**(GC-1.0)))
12500    IF (ABS(PMAX-PMAXA).GT.TLA) GO TO 10
12600 C
12700 C *****
12800 C ROUTING FOR DUAL CYCLE COMBUSTION
12900    IF (PMAX.GT.PMAXL) GO TO 20
13000    90 AMT=RC*PA*S*PHI*VS*ES*CA/R/TM/(1.0-1.0/CR)/ET [2]
13100    EP=ED*AMT*CV*WHC*(1.0-1.0/CR**0.4)/RT/HPD [9]
13200    GO TO 40
13300 C *****
13400 C SECTION FOR DUAL CYCLE COMBUSTION
13500 C CONST VOLUME THEN CONST PRESSURE
13600    20 B=0.0
13700    BS=0.0
13800 C FIND VOLUME RATIO (BETA)
13900    CALL SOL (0.5,B,BS,BD)
14000    B = 1.01
14100 C *****
14200 C ITERATE FOR PMAX TOLERANCE (TLB)

```

Diesel Gas Turbine Program

```

14300 C
14400 M=0
14500 50 M=M+1
14600 PMAX=PMAXL
14700 A=PMAX/RC/PA/CR**GC
14800 P=1.0-(A*B**1.4-1.0)/(A-1.0+1.4*A*(B-1.0))/CR**0.4
14900 TME=TM*PMAX*B*(1.0+(B/CR)**(GE-1.0))/RC/PA/CR/2.0 [6]
15000 CALL GG (RT,TME,GE)
15100 IF (GE.LT.0.01) GOTO 170
15200 TMC = TM * (CR**(GC - 1.0) + 1.0) / 2.0 [7]
15300 CALL GG (0.0,TMC,GC)
15400 IF (GC.LT.0.01)GOTO170
15500 PMAX=(P*(DTK*CV/R/RT/TM-(1.0-CR**(GC-1.0))/P/(GC-1.0)))/(B\
\CR*(1.0
15600 *-(B/CR)**(GE-1.0))/(GE-1.0)/RC/PA+(B-1.0)/RC/PA/CR) [5]
15700 DA=PMAX-PMAXL
15800 IF (ABS(DA).LT.TLB) GO TO 60
15900 C
16000 C *****
16100 CALL SOL (DA,B,BS,BD)
16200 C ARRANGE WARNING FOR EXCESSIVE LOOPING
16300 IF (MA.EQ.40) GO TO 15
16400 GO TO 17
16500 15 WRITE(3,16)
16600 16 FORMAT(37H EXCESSIVE LOOPING PMAX(TLB) AS BELOW)
16700 GO TO 60
16800 17 GO TO 50
16900 C *****
17000 60 AMT=RC*PA*CA*S*PHI*VS*ES/R/TM/(1.0-1.0/CR)/ET [2]
17100 C 60 IS RE-ENTRY POINT FOR DUAL CYCLE
17200 AA1=(A*(B**1.4))-1.0
17300 AA2=A-1.0
17400 AA3=1.4*A*(B-1.0)
17500 AA4=1.0/(CR**0.4)
17600 A5=(1.0-(AA1*AA4/(AA2+AA3)))
17700 EP=ED*AMT*CV*WHC*A5/(RT*HPD) [8]
17800 GO TO 40
17900 C
18000 C *****
18100 C SECTION FOR VARIABLE CR
18200 30 MA=0
18300 CR=CRA-((CRA-CRB)/2.0)
18400 PMAX=PMAXL
18500 C *****
18600 C ITERATE FOR PMAX TOLERANCE(TLC)
18700 70 MA=MA+1
18800 TME=TM*PMAX*(1.0+1.0/CR**(GE-1.0))/RC/PA/CR/2.0 [6]
18900 CALL GG(RT,TME,GE)
19000 IF (GE.LT.0.01) GO TO 170
19100 TMC=TM*(CR**(GC-1.0)+1.0)/2.0 [7]
19200 CALLGG (0.0,TMC,GC)

```

Diesel Gas Turbine Program

```

19300      IF (GC.LT.0.01) GO TO 170
19400      PMAX=(GE-1.0)*RC*PA*CR/(1.0-1.0/CR***(GE-1.0))*((DTK*CV/R/RT\
\ /TM*(
19500      *1.0-1.0/CR**0.4))-1.0/(GC-1.0)*(1.0-CR***(GC-1.0))) [4]
19600      DB=PMAX-PMAXL
19700      IF (ABS(DB).LT.TLC) GO TO 80
19800      IF (DB.GT.0.0) CR=CR-0.2
19900      IF (DB.LT.0.0) CR=CR+0.2
20000      IF (MA.GT.40) GO TO 41
20100      GO TO 42
20200 C ARRANGE WARNING FOR EXCESSIVE LOOPING
20300      41 WRITE(3,99)
20400      99 FORMAT(31H EXCESSIVE VCR LOOPING AS BELOW)
20500      GO TO 80
20600      42 GO TO 70
20700 C
20800 C *****
20900      80 IF (CR.LE.CRA) GO TO 85
21000      CR=CRA
21100      GO TO 20
21200      85 IF (CR.GE.CRB) GO TO 90
21300      CR=CRB
21400      GO TO 20
21500 C *****
21600 C FRICTIONAL HORSE POWER CORRECTION
21700      40 EP=EP-(CB*ES/0.7457) [8]
21800 C 40 IS RE ENTRY POINT [13]
21900      AMF=AMT/RT1 [15]
22000      SFC=AMF*60.0/EP
22100      E=EP*HPCA/CV/AMF
22200      FF = AMF * 100.0 / 60.0 / VS [17]
22300 C FIND HEAT LOSS TO COOLANT
22400      CALL H (FF,HC)
22500      CP = 0.3
22600 C DELTA T FOR ENGINE (TEMP RISE)
22700 C *****
22800 C ITERATE FOR ENG TEMP RISE TOLERANCE (TLD)
22900 C
23000      200 TD = (CV * (1.0-E-HC)) / (CP * (SR * RT + 1.0)) [16]
23100      TMD = TM + TD / 2.0
23200 C OUTPUT TO DENOTE FAILURE IN THIS LOOP
23300      IF (TMD.GT.3600.0) WRITE (3,977) TM,TD,CV,E,HC,FF,CP,SR,RT\
\ ,EP,
23400      * CB,ES,AMF,AMT,HPCA,CV
23500      977 FORMAT (6H TM = ,F12.6,6H TD = ,F12.6,6H CV = ,F12.6,6H E \
\ = ,
23600      *F12.6/6H HC = ,F12.6,6H FF = ,F12.6,6H CP = ,F12.6,6H SR = ,
23700      *F12.6/6H RT = ,F12.6,6H EP = ,F12.6,6H CB = ,F12.6,6H ES = ,
23800      *F12.6/6H AMF = ,F12.6,6HAMT = ,F12.6,6HHPCA = ,F12.6,6H CV = \
\ ,F12.6)
23900 C FIND SPECIFIC HEAT AT CONSTANT PRESSURE(CP)

```

Diesel Gas Turbine Program

```

24000 CALL C (RT,TMD,CP)
24100 IF (CP.LT.0.01)GOTO 170
24200 TDA = (CV * (1.0-E-HC)) / (CP * (SR * RT + 1.0))
24300 IF (ABS(TD-TDA).GT.TLD) GOTO 200
24400 C
24500 C*****
24600 120 PT=PDC*RC*PA
24700 130 RTU=PT/PA
24800 TT=TM+TD
24900 AME=SR*AMT+AMF
25000 GO TO 170
25100 11 WRITE(3,152)
25200 WRITE(3,151)
25300 WRITE(3,150)
25400 WRITE(1,/)ES
25500 GO TO 12
25600 150 FORMAT(9H ENG SPD,1X,6H BOOST,2X,8H A/F RAT,4X,5H SCAV,3X,
25700 *8H MAN TEM,3X,6H POWER,3X,
25800 *8H BRK EFF,3X,8H EXH TEM,3X,9H SFC(ENG),/,6H RPM,32X,6H\
\ DEG C,
25900 *6X,3H KW,16X,6H DEG C,5X,9H KG/KW.HR,///)
26000 151 FORMAT(10H TURB SPD,1X,9H COMP TEM,1X,5H ETAC,3X,9H SFC\
\ (DFE),3X,
26100 *8H TURB IN,3X,8H TURB EX,3X,5H ETAT,5X,4H GAS,5X,
26200 *4H RST,5X,6H BR/EF,3X,5H ETAM,/,5H RPM,5X,
26300 *6H DEG C,12X,9H KG/KW.HR,5X,6H DEG C,5X,6H DEG C,15X,6H KG\
\ /HR,)
26400 152 FORMAT(8H1 ERROR,4X,5H HDEM,4X,5H HSUP,5X,4H P/W,6X,
26500 *6H WFACT,4X,6H TOUT,10X,12H (KW, DEG C),)
26600 C*****
26700 C S. I. UNIT CONVERSION
26800 C
26900 170 PZ=EP*0.7457
27000 PY=(TM-459.67-32.0)/1.8
27100 SFP=(SFC*0.4536)/0.7457
27200 PX=(TT-459.67-32.0)/1.8
27300 PMAT=PMA/14.5
27400 C CALCULATE TURBINE OPERATING CONDITIONS
27500 C AND AUXILIARY FUELLING RATE
27600 CALL ENG(PZ,AMT,SR,AMF,TT,MAPN,TTUE1,TURN)
27700 C
27800 C*****
27900 995 WRITE(3,160)ES,RC,RT,SR,PY,PZ,E,PX,SFP
28000 160 FORMAT(1H ,F7.1,3X,F5.2,5X,F5.1,4X,F5.2,4X,F7.1,3X,
28100 *F7.2,3X,F7.4,3X,F7.1,3X,F6.4,3X,/)
28200 180 CONTINUE
28300 C GRAPHICAL OUTPUT OF COMPRESSOR MAP (IF REQUIRED)
28400 IF (IGRAPH.EQ.1) WRITE(1,998)
28500 IF (IGRAPH.EQ.1) CALL GPLOT(MAPN)
28600 998 FORMAT(30H TYPE RUN *SYSTEM/PLOT PLOTENG)
28700 STOP
28800 END
28900 C
29000 C

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Diesel Gas Turbine Program

```

29100      SUBROUTINE SOL (DAB,B,BS,BD)
29200 C      *****
29300 C
29400 C DETERMINES VOLUME RATIO(BETA)
29500 C
29600      IF (B.LE.0.01) BD = 1.0
29700      BS = 0.2
29800      IF (ABS(DAB).LT.1200.1) BS = 0.1
29900      IF (ABS(DAB).LT.700.1) BS = 0.05
30000      IF (ABS(DAB).LT.200.1) BS = 0.01
30100      IF (DAB.GE.0.0) GOTO 10
30200      BD = BD / 2.0
30300      BS =-BS
30400 10    B = B + BS * BD
30500      RETURN
30600      END
30700 C
30800 C
30900      SUBROUTINE C (RT,TMD,CP)
31000 C      *****
31100 C
31200 C DETERMINES VALUE FOR SPECIFIC HEAT(CP)
31300 C
31400 C ARRANGE FLAG IF A/F RATIO OR MEAN TEMP
31500 C OUTSIDE BLOCK DATA LIMITS
31600 C TEMP=400-3600 DEG R
31700 C AIR/FUEL RATIO=20-40
31800      COMMON ICD(17,6),IDD(23,2),IC(17,5)
31900      I = RT / 5.0 - 3.0
32000      A = TMD / 200.0 - 1.0
32100      J = A
32200      IF (I.GT.5)GOTO 10
32300      IF (J.LT.1.OR.J.GT.17) GOTO 10
32400      IF (J.EQ.17)GOTO 30
32500 C LINEAR INTERPOLATION FROM BLOCK DATA
32600      AI = FLOAT(IC(J,I)) / 10000.0
32700      AI1 =FLOAT(IC(J+1,I)) / 10000.
32800      CP = AI + (AI - AI1) * (FLOAT(J) - A)
32900      RETURN
33000 30    CP = FLOAT(IC(J,I))/10000.
33100      RETURN
33200 10    CONTINUE
33300      CP=0.317
33400      IF (J.LT.1.0) CP=0.244
33500      RETURN
33600      END
33700 C
33800 C

```

Diesel Gas Turbine Program


```

33900      SUBROUTINE H (FF,HC)
34000 C      *****
34100 C
34200 C DETERMINES HEAT LOSS TO COOLANT
34300 C
34400 C ARRANGE FLAG IF FUEL FLOW OUTSIDE BLOCK DATA LIMITS
34500 C FUEL FLOW=2.5-65.0 LB/HR
34600      COMMON ICD(17,6),IC(23,2),IED(17,5)
34700      IF (FF.LT.2.5.OR.FF.GT.65.0) GOTO 30
34800      IJ = FF * 100.0
34900      DO 10 J = 1,23
35000      IF(IJ.LT.IC(J,1))GOTO 20
35100 10     CONTINUE
35200      GOTO 30
35300 20     J=J-1
35400 C LINEAR INTERPOLATION FROM BLOCK DATA
35500      AI=FLOAT(IC(J,2)) /1000.0
35600      A11 = FLOAT(IC(J+1,2)) / 1000.0
35700      HC=AI+(AI-A11) * (FLOAT(IC(J,1))-100.*FF)/FLOAT(IC(J+1,1))\
      \-IC(J,1))
35800      RETURN
35900 30     WRITE(3,40)FF
36000      WRITE(1,40)FF
36100 40     FORMAT (16H INVALID REQUEST/ 6H FF = ,E12.6)
36200      HC = 0.0
36300      RETURN
36400      END
36500 C
36600 C

```

Diesel Gas Turbine Program

```

36700      SUBROUTINE GG (RT, TM, G)
36800 C      *****
36900 C
37000 C DETERMINES VALUE FOR GAMMA
37100 C
37200      COMMON IC(17,6), IDD(23,2), IED(17,5)
37300 C ARRANGE FLAG IF A/F RATIO OR MEAN TEMP
37400 C OUTSIDE BLOCK DATA LIMITS
37500 C TEMP=400-3600 DEG R
37600 C AIR/FUEL RATIO=20-40
37700      I = RT / 5.0 - 2.0
37800      IF (RT.LE.0.01) I = 1
37900      IF (I.GT.6)GOTO 10
38000      IF (J.EQ.17)GOTO 30
38100      A = TM /200.0 - 1.0
38200      J = A
38300      IF (J.LT.1.0R.J.GT.17) GOTO 10
38400 C LINEAR INTERPOLATION FROM BLOCK DATA
38500      A1 = FLOAT(IC(J,1)) /10000.0
38600      A11 = FLOAT(IC(J+1,1)) / 10000.0
38700      G = A1 + (A1 - A11) * (FLOAT(J) - A)
38800      RETURN
38900      30 G = FLOAT(IC(J,1))/10000.
39000      RETURN
39100      10 WRITE (3,20) RT, TM
39200      WRITE(1,20)RT, TM
39300      20 FORMAT (16H INVALID REQUEST/ 6H RT = ,E12.6/ 6H TM = ,E12.6)
39400      G=1.276
39500      IF (J.LT.1.0) G=1.39
39600      RETURN
39700      END
39800 C
39900 C

```

Diesel Gas Turbine Program

```

40000      BLOCK DATA
40100 C      *****
40200 C
40300 C      IE(17,5)=CP ID(23,2)=HC IC(17,6)=GAMMA
40400 C      TAKEN FROM WALLACE CHARACTERISTICS
40500 C
40600      COMMON IC(17,6), ID(23,2), IE(17,5)
40700      DATA(IE(1,1), I=1, 17)/2470, 2518, 2580, 2657, 2737, 2804, 2880,
40800      *2940, 3010, 3070, 3120, 3165, 3205, 3230, 3255, 3280, 3300/
40900      DATA(IE(1,2), I=1, 17)/2460, 2497, 2553, 2625, 2700, 2773, 2842,
41000      *2901, 2960, 3018, 3070, 3110, 3147, 3170, 3200, 3225, 3235/
41100      DATA(IE(1,3), I=1, 17)/2445, 2480, 2535, 2600, 2673, 2750, 2810,
41200      *2873, 2927, 2982, 3030, 3070, 3105, 3130, 3153, 3170, 3190/
41300      DATA(IE(1,4), I=1, 17)/2438, 2468, 2520, 2580, 2655, 2730, 2790,
41400      *2852, 2908, 2955, 3008, 3050, 3080, 3105, 3125, 3148, 3155/
41500      DATA(IE(1,5), I=1, 17)/2430, 2463, 2513, 2573, 2645, 2715, 2780,
41600      *2840, 2895, 2943, 2990, 3030, 3070, 3090, 3112, 3135, 3140/
41700      DATA(ID(1,1), I=1, 23)/250, 375, 500, 625, 750, 875, 1000, 1125, 1250,
41800      *1375, 1500, 1750, 2000, 2250, 2500, 3000, 3500, 4000, 4500, 5000, 5500,
41900      *6000, 6500/
42000      DATA(ID(1,2), I=1, 23)/400, 330, 283, 255, 232, 214, 200, 190, 182,
42100      *175, 170, 160, 152, 147, 140, 130, 125, 119, 116, 114, 112, 109, 109/
42200      DATA(IC(1,1), I=1, 17)/14030, 13975, 13900, 13805, 13700, 13565,
42300      *13450, 13360, 13285, 13220, 13165, 13110, 13070, 13045, 13015,
42400      *12990, 12975/
42500      DATA(IC(1,2), I=1, 17)/13840, 13745, 13625, 13480, 13345, 13225\
\, 13120,
42600      *13030, 12945, 12875, 12810, 12760, 12720, 12685, 12660, 12635, 12610/
42700      DATA(IC(1,3), I=1, 17)/13870, 13875, 13675, 13540, 13395, 13280\
\, 13175,
42800      *13085, 13000, 12940, 12880, 12825, 12785, 12750, 12725, 12700, 12685/
42900      DATA(IC(1,4), I=1, 17)/13895, 13810, 13710, 13580, 13445, 13330\
\, 13215,
43000      *13125, 13050, 12980, 12920, 12875, 12830, 12800, 12775, 12775, 12735/
43100      DATA(IC(1,5), I=1, 17)/13910, 13840, 13735, 13610, 13480, 13360\
\, 13250,
43200      *13160, 13070, 13010, 12950, 12900, 12860, 12835, 12805, 12780, 12770/
43300      DATA(IC(1,6), I=1, 17)/13930, 13855, 13750, 13630, 13500, 13380\
\, 13270,
43400      *13170, 13090, 13025, 12970, 12925, 12875, 12845, 12820, 12795, 12780/
43500      END

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Diesel Gas Turbine Program

```

43600      SUBROUTINE COMPRE
43700 C      *****
43800 C      COMPRESSOR
43900 C      CARRIES OUT COMPRESSOR CALCULATIONS
44000      COMMON /C/TA,PA,RS,EF,EE,TCOOL,ES,ST,RT1,DELP,ETAT,EHT
44100      *,WAF,SPD,WA,TSUP,HPC,TM,RC,WENG,RST,HPT,WT,THA,IFP
44200      *,P3,TTUR,TTUN,ET,TTUE,WCC,CPE,CPCC,TCN,TCCIN,WACC
44300      *,WGAS,SFC
44400      COMMON IC(17,6),ID(23,2),IE(17,5)
44500      RS=RC
44600 C      FIND GAMMA (AIR)
44700      CALL GG(0.0,TA,G)
44800      CPA=0.241
44900 C      FIND SPEED (SPD) AND AIR FLOW FROM TABLES
45000 C      USE LOWER VALUE OF AIR FLOW IF AVAILABLE
45100      IF=1
45200 C      LOOK UP COMPRESSOR OPERATING POINT
45300 C      AND ESCAPE IF DATA IS INVALID
45400      CALL COMTAB(RS,EF,WAF,SPD,IF)
45500      IF(IF.EQ.0.0)GOTO 10
45600      IF=2
45700      CALL COMTAB(RS,EF,WAF,SPD,IF)
45800      IF(IF.EQ.0)GOTO 10
45900 C      WARNING FOR EXCEEDING DATA LIMITS
46000      WRITE(3,1)RS,EF
46100      WRITE(1,1)RS,EF
46200 1      FORMAT(" OUT OF RANGE OF COMPRESSOR TABLE"/
46300      *" RS = ",F15.2," EF = ",F15.2)
46400      STOP
46500 C      CONVERT CFM TO LB/MIN
46600 10      WA=WAF*.0741
46700      RSG=(RS**((G-1)/G)-1)/(EF/100.0)
46800      TSUP=TA*(1+RSG)
46900      HPC=WA*CPA*TA*RSG*.023575758
47000 C      MAN TEMP USING COOLER (IF FITTED)
47100      TM=TSUP-EE*(TSUP-TCOOL)
47200      RC=RS
47300      RETURN
47400      END

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Diesel Gas Turbine Program

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47500      SUBROUTINE ENG(PZ,AMT,SR,AMF,TT,MAPN,TTUE1,TURN)
47600 C      *****
47700 C
47800 C ROUTINE FOR ASSESSING TURBINE OPERATING CONDITIONS
47900 C
48000      COMMON /C/TA,PA,RS,EF,EE,TCOOL,ES,ST,RT1,DELP,ETAT,EHT
48100      *,WAF,SPD,WA,TSUP,HPC,TM,RC,WENG,RST,HPT,WT,THA,IFP
48200      *,P3,TTUR,TTUN,ET,TTUE,WCC,CPE,CPCC,TCCN,TCCIN,WFACC
48300      *,WGAS,SFC
48400      COMMON IC(17,6),ID(23,2),IE(17,5)
48500 C CORRECT ENGINE EXHAUST HEAT LOSS (EMPIRICAL)
48600      TT=TT-70.0
48700      TTIN=TURN
48800      RST=((RS*PA)-DELP)/PA
48900 C WARNING FOR LACK OF BYPASS AIR
49000      IF((WA-(AMT*(SR+0.1))).LT.0.1) WRITE(3,87)
49100      87 FORMAT(29H NO BYPASS - IGNORE GAS BELOW,)
49200 C ENGINE MASS FLOW RATE
49300      WENG=(AMT*(SR+0.1))+AMF
49400 C SPEED + BEARING LOSS CORRECTION FOR TURBO
49500      IF(MAPN.EQ.1) SPD1=SPD+5000.0
49600      IF(MAPN.EQ.2) SPD1=SPD+2500.0
49700      IF(MAPN.EQ.1) FALL=1.7584*(1.0E-10)*SPD1*SPD1
49800      IF(MAPN.EQ.2) FALL=5.6455*(1.0E-10)*SPD1*SPD1
49900      HPT=HPC+FALL
50000      EMEC=HPC/HPT
50100 C INITIAL VALUE FOR ACC FUEL
50200      WFACC1=0.15
50300      ICO1=0
50400 C
50500 C *****
50600 C ITERATE FOR STABLE ACC FUELLING
50700      20 WT=WA+AMF+WFACC1
50800 C AUX COMB CHAMBER MASS FLOW
50900      WCC=WA-(AMT*(SR+0.1))+WFACC1
51000 C ACC INLET TEMP WITH HEAT EXCHANGE (IF FITTED)
51100 C WITH EMPIRICAL HEAT LOSS CORRECTION
51200      TCCIN=(TSUP-10.0)+(EHT*(TTUE1-(TSUP-10.0)))
51300 C FIND CP + CORRECT FOR A/F RATIO
51400      CALL C(40.0,TTUE1,CPTO1)
51500      AFTO=WA/(AMF+WFACC1)
51600      CPTO=CPTO1-((0.008*(AFTO-40.0))/50.0)
51700      AFEN=AMT/AMF
51800 C FIND CP FOR ENG EXHAUST
51900      CALL C(AFEN,TT,CPE)
52000 C ENTHALPY AT TURBINE OUTLET
52100      HO=WT*CPTO*TTUE1
52200 C FIND CP + CORRECT FOR A/F RATIO
52300      CALL C(40.0,TTIN,CPTIN1)
52400      CPTIN=CPTIN1-((0.008*(AFTO-40.0))/50.0)

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Diesel Gas Turbine Program

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52500 C ENTHALPIES FOR TURB INLET. ENG EXH + ACC INLET [43]
52600     HIN=WT*CPTIN*TTIN [44]
52700     HENG=WENG*CPE*TT [45]
52800     HCIN=(WA-(AMT*(SR+0.1)))*0.242*TCCIN
52900 C ENTHALPY INPUT TO ACC [46]
53000     QCC=(HIN-HENG)-HCIN [47]
53100     WFACC=QCC/21000.0
53200     IF((WFACC-WFACC1)/WFACC).LT.0.02) GO TO 41
53300     WFACC1=WFACC
53400     GO TO 20
53500 C *****
53600 C
53700 C
53800 C TURBINE WORK ERROR
53900     41 P=HO+(HPT*42.416)-HIN [48]
54000     ICO1=ICO1+1
54100     IF(ICO1.NE.1) GO TO 9
54200     IF(P.LT.0.0) ILP=0
54300     IF(P.GE.0.0) ILP=1
54400 C FIND GAMMA + CORRECT FOR A/F RATIO
54500     9 CALL GG(40.0,TTIN,GAMT1)
54600     GAMT=GAMT1+((0.008*(AFTO-40.0))/50.0)
54700     TOUT=((1.0/RST)**((GAMT-1.0)/GAMT))*TTIN
54800     ETAT=((TTIN-TTUE1)/(TTIN-TOUT)) [49]
54900     DENT=HPT*42.416
55000 C ERROR WORK FACTOR
55100     FACT=P/DEMT [50]
55200     SUPT=HIN-HO
55300 C TURBINE WORK FACTOR
55400     ETT=(1.0/(1.0-FACT))*100.0 [51]
55500 C *****
55600 C SECTION TO EVALUATE CONST PRESS TURBOCHARGING CONDITIONS
55700     IF(IFP.EQ.1) GO TO 40
55800     IF(ILP.EQ.0) GO TO 10
55900     IF(ILP.EQ.1) GO TO 11
56000     10 IF(SUPT.LE.DENT) GO TO 40
56100     IF((SUPT-DEMT).LT.60.0) GO TO 42
56200     TTIN=TTIN-10.0
56300     GO TO 20
56400     42 TTIN=TTIN-2.0
56500     GO TO 20
56600     11 IF(SUPT.GE.DENT) GO TO 40
56700     IF((DEMT-SUPT).LT.60.0) GO TO 12
56800     TTIN=TTIN+10.0
56900     GO TO 20
57000     12 TTIN=TTIN+2.0
57100     GO TO 20
57200 C *****
57300 C SI UNIT CONVERSIONS
57400     40 WGAS=WFACC*27.272727

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Diesel Gas Turbine Program

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57500      SFD=(((WFACC*60.0)/2.2)*1.163)+((AMF*60.0)/2.2)/PZ
57600      E1=(PZ/0.7457)*42.42)/(18060.0*(AMF+(WFACC*1.163)))
57700      TSUP1=(TSUP-491.67)/1.8
57800      TTUR1=(TTIN-491.67)/1.8
57900      TTUE2=(TTUE1-491.67)/1.8
58000      ETAT1=ETAT*100.0
58100      TOUT1=(TOUT-491.67)/1.8
58200      EMEC=EMEC*100.0
58300      P=P*0.01758
58400      DENT=DEMT*0.01758
58500      SUPT=SUPT*0.01758
58600 C OUTPUT REQUIRED PARAMETERS
58700      WRITE(3,159)P,DEMT,SUPT,FACT,ETT,TOUT1
58800      159 FORMAT(1H,6F10.3,)
58900      WRITE(3,160)SPD,TSUP1,EF,SFD,TTUR1,TTUE2,ETAT1,WGAS,RST,E1\
        \,EMEC
59000      160 FORMAT(1H, ,F7.1,10F10.3,)
59100      RETURN
59200      END

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Diesel Gas Turbine Program

```

59300          SUBROUTINE COMTAB(RS,EF, WAF, SPD, IF)
59400 C          *****
59500 C
59600 C COMPRESSOR TABLE LOOK UP.
59700 C FINDS VALUES FOR AIR FLOW AND SPEED
59800 C FOR GIVEN EFFICIENCY AND PRESSURE RATIO
59900 C IF = 1 FOR LOWER VALUES
60000 C      = 2 FOR UPPER VALUES
60100 C      ON RETURN SET TO 1 IF NO VALUES AVAILABLE
60200 C          0 IF O.K.
60300 C
60400          COMMON/COMP/C(15,10,4),RSV(15),RAL(15),REL(15),EFV(10),MAXR\
          \,MAXE
60500 C ALLOCATE STORAGE
60600          DIMENSION A(2)
60700          DIMENSION AA(3,3),AC(3),AD(3),ACC(3),TEM(3,3),W01(3)
60800          DIMENSION W02(3)
60900          IND=IF
61000          IN=0
61100          IF=1
61200          WAF=0
61300          SPD=0
61400          IF(IND.LE.0.OR.IND.GE.3)RETURN
61500          IF(RS-RSV(1).LT.-.001)RETURN
61600          DO 20 JR=2,MAXR
61700 20          IF(RS-RSV(JR).LT.-.001)GOTO 30
61800          RETURN
61900 30          JRK=JR
62000          IF(EF-EFV(1).LT.-.001)RETURN
62100          DO 40 JE=2,MAXE
62200 40          IF(EF-EFV(JE).LT.0.001)GOTO 50
62300          RETURN
62400 C CHECK FOR LEGAL VALUES.
62500 50          JRL=JR-1
62600          IF(REL(JR).EQ.0.OR.REL(JRL).EQ.0)GOTO 51
62700          EUL=(RS-RSV(JRL))*(REL(JR)-REL(JRL))/(RSV(JR)-RSV(JRL))
62800          * +REL(JRL)
62900          IF(EUL-EF.LT.-.01)RETURN
63000 51          CONTINUE
63100 70          JEL=JE-1
63200          E1=(EF-EFV(JEL))/(EFV(JE)-EFV(JEL))
63300 C IF ALL POINTS DEFINED DO 2 AXIS PARALLEL LIN. INTS.
63400          IF(C(JRL,JEL,IND).NE.0.AND.C(JRL,JE,IND).NE.0
63500          *.AND.C(JR,JEL,IND).NE.0.AND.C(JR,JE,IND).NE.0)GOTO100
63600          KIF=0
63700          KI=1
63800          DO 991 JEC=JEL,JE
63900          KIF=KI
64000          DO 991 JRC=JRL,JR
64100          IF(C(JRC,JEC,IND).EQ.0)GOTO991

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Diesel Gas Turbine Program


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64200      AA(KI,1)=RSV(JRC)
64300      AA(KI,2)=EFV(JEC)
64400      AC(KI)=C(JRC,JEC,IND)
64500      ACC(KI)=C(JRC,JEC,IND+2)
64600      AA(KI,3)=1
64700      KI=KI+1
64800      991  CONTINUE
64900      C  IF THREE POINTS AVAILABLE , PLANE SURFACE FIT.
65000      1001 IF (KI.EQ.4)GOTO1000
65100      C  IF ONLY 1 POINT AVAILBLE ASSUME FUNCTION IS UNDEFINED.
65200          IF (KI.LE.2)GOTO200
65300          IF (KIF.EQ.2)GOTO200
65400      C  IF 2 POINTS AVAILABLE WITH SAME EF VALUE TRY 1 SHIFT.
65500          IF (IN.EQ.1)GOTO200
65600          IN=1
65700          IF (KIF.EQ.1)GOTO1003
65800          IF (JEL.EQ.1)GOTO200
65900          JE=JE-1
66000          GOTO70
66100      1003 IF (JE.EQ.MAXE)GOTO200
66200          JE=JE+1
66300          GOTO70
66400      1000 IFF=1
66500      C-CALL LIBRARY SOLUTION ROUTINES
66600          CALL F04ATF(AA,3,AC,3,AD,TEM,3,W01,W02,IFF)
66700          IF (IFF.NE.0)GOTO200
66800      1010 SPD=AD(1)*RS+AD(2)*EF +AD(3)
66900          IFF=1
67000          CALL F04ATF(AA,3,ACC,3,AD,TEM,3,W01,W02,IFF)
67100          IF (IFF.NE.0)GOTO200
67200          WAF=AD(1)*RS+AD(2)*EF+AD(3)
67300          GOTO 120
67400      100  R1=(RS-RSV(JRL))/(RSV(JR)-RSV(JRL))
67500          DO 110 JJ=1,2
67600              II=IND+(JJ-1)*2
67700              A1=(C(JR,JEL,II)-C(JRL,JEL,II))*R1
67800              *  +C(JRL,JEL,II)
67900              A2=(C(JR,JE,II)-C(JRL,JE,II))*R1 +
68000              *  C(JRL,JE,II)
68100      110  A(JJ)=(A2-A1)*E1+A1
68200          WAF=A(2)
68300          SPD=A(1)
68400      120  WAF1=(RAL(JRK)-RAL(JRK-1))*(RS-RSV(JRK-1))/
68500          *  ((RSV(JRK)-RSV(JRK-1))) +RAL(JRK-1)
68600          IF (WAF1-WAF.LT.1.0)IE=0
68700          RETURN
68800      200  CONTINUE
68900          RETURN
69000          END

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Diesel Gas Turbine Program

```

69100      SUBROUTINE INTAR(MAPN)
69200 C      *****
69300 C
69400 C      COMPRESSOR TABLES READ FROM DISK FILE 9 OR 10.
69500 C      FORMAT      NO. OF RS VALUES, NO. OF EFF VALUES (2110)
69600 C      RS1,EF1,AIRFLOW LOWER,AIRFLOW UPPER,SPEED LOWER,SPEED UPPER \
        \ (8F10.0)
69700 C      RS1,EF2,      ::::::::::      ::::::::::      ::::::::::      ::::::::::
69800 C      ::::::::::
69900 C      RS1, LAST EF
70000 C      RS2,EF1      ETC.
70100 C      UNAVAILABLE VALUES IN THE RECTANGULAR ARRAY SPECIFY AS ZERO.
70200 C      ON EACH LINE BEGINING A NEW RS VALUE TWO EXTRA VALUES ARE REQ.
70300 C      AIRFLOW VALUE AT CUTOFF LINE
70400 C      MAXIMUM EFFICIENCY AT THIS RS VALUE. (0 IF NOT APPLIC.)
70500 C      MAPN=1 FOR 3LD1 COMP = 2 FOR 4LGK COMP
70600 C
70700      COMMON/COMP/C(15,10,4),RSV(15),RAL(15),REL(15),EFV(10)\
        \,MAXR,MAXE
70800      IF (MAPN.EQ.1) READ(10,1)MAXR,MAXE
70900      IF (MAPN.EQ.2) READ(9,1)MAXR,MAXE
71000      IF (MAXR.GT.15.OR.MAXR.LT.1.OR.MAXE.GT.10.OR.MAXE.LT.1\
        \)GOTO100
71100      1  FORMAT(2110)
71200 C      INPUT DATA FROM ON LINE FILES
71300      DO 20 J=1,MAXR
71400      DO 20 JJ=1,MAXE
71500      IF (MAPN.EQ.1)READ(10,2)R,E,A1,A2,S1,S2,RA,RE
71600      IF (MAPN.EQ.2)READ(9,2)R,E,A1,A2,S1,S2,RA,RE
71700      2  FORMAT(8F10.0)
71800      IF (JJ.NE.1)GOTO10
71900      RSV(J)=R
72000      IF (RA.EQ.0)GOTO 100
72100      RAL(J)=RA
72200      REL(J)=RE
72300      10  CONTINUE
72400      IF (J.EQ.1)EFV(JJ)=E
72500      IF (RSV(J).NE.R)GOTO100
72600      IF (EFV(JJ).NE.E)GOTO100
72700      C(J,JJ,3)=A1
72800      C(J,JJ,4)=A2
72900      C(J,JJ,1)=S1
73000      C(J,JJ,2)=S2
73100      20  CONTINUE
73200      RETURN
73300 C      STOP PROGRAM IF ERRORS OCCUR IN DATA FILES
73400      100 WRITE(3,3)
73500      WRITE(1,3)
73600      3  FORMAT(" COMPRESSOR INPUT ERROR")
73700      STOP
73800      END

```

Diesel Gas Turbine Program

```

73900      SUBROUTINE GPLOT(MAPN)
74000 C      *****
74100 C
74200 C      SUBROUTINE TO PLOT GRAPH OF COMPUTED COMPRESSOR TABLES.
74300 C      MAPN = 1 FOR 3LD1 COMPRESSOR
74400 C      = 2 FOR 4LRK COMPRESSOR
74500 C      ROUTINE INTAR MUST HAVE BEEN CALLED BEFORE GPLOT.
74500 C
74700      COMMON/COMP/C(15,10,4),RSV(15),RAL(15),REL(15),EFV(10)\
\,MAXR,MAXE
74800      RSS=RSV(1)
74900      RSF=RSV(MAXR)+.02
75000      RSI=.02
75100      EFS=EFV(1)
75200      EFF=EFV(MAXE)
75300      EFI=1
75400      AS=5
75500      IF(MAPN.EQ.1)AS=4
75600 C UTILISES LIBRARY PROGRAM ROUTINES
75700      CALL PLOTIN
75800      CALL DEVICE(12,100)
75900      CALL PLOT(30,180,-3)
76000      CALL SYMBOL(195.0,-30.0,4,"COMPUTED COMPRESSOR MAP",270,23)
76100      IF(MAPN.EQ.1)CALL SYMBOL(185,-30,4,"FOR HOLDSET 3LD1",270,15)
76200      IF(MAPN.EQ.2)CALL SYMBOL(185,-30,4,"FOR HOLDSET 4LRK",270,15)
76300      CALL AXIS(0,0,"CFM AT 15 DEG. C",-15,170,270,0,AS,.4)
76400      CALL AXIS(0,0,"(PRESSURE RATIO)",16,200,0,1,.01,.5)
76500      CALL OFFSET(1,.01,0,-AS)
76600      IFF=1
76700      50 EF=EFS
76800 C      WRITE(3,4)EF,IFF
76900      40 RS=RSS
77000      PF=13
77100      20 IF=IFF
77200      CALL CONTAL(RS,EF,MAF,SPD,IF)
77300      IF(IF.EQ.1)GOTO 6
77400      CALL PLOT(RS,MAF,PF)
77500      PF=12
77600      GOTO 7
77700      6 PF=13
77800      7 CONTINUE
77900 C      WRITE(3,3)RS,EF,MAF,SPD,IF
78000 C      3 FORMAT(5F15.5)
78100      RS=RS+RSI
78200      IF(RS.LE.RSF)GOTO 20
78300      EF=EF+EFI
78400      IF(EF.GT.EFF)GOTO 30
78500 C      WRITE(3,4)EF,IFF
78600 C      4 FORMAT("1 EFFICIENCY VALUE ",F15.5," PART ",I2/
78700 C      * ,10X,"RS",13X,"EF",13X,"MAF",12X,"SPD",12X,"IF"//)

```

Diesel Gas Turbine Program

```

78800      GOTO 40
78900      30  IF (IFF.EQ.2)GOTO8
79000      IFF=2
79100      GOTO 50
79200      6  CALL DEVEND
79300      RETURN
79400      END
79500      SUBROUTINE PHETER(PREST,WAP)
79600      C      *****
79700      C
79800      C  CALCULATES COMPRESSOR AIR FLOW FROM PRESTON METER
79900      C  MANOMETER PRESSURE DROP(INN H2O)
80000      C
80100      C  SCALING FACTOR
80200      PR=PREST/10.0
80300      C  COEFFS OF POLYNOMIAL (FROM LIBRARY PROG $(L)B00273)
80400      A1=0.1298450
80500      A2=30.0576770*PR
80600      A3=-24.8359529*(PR**2)
80700      A4=12.3399425*(PR**3)
80800      A5=-3.2256519*(PR**4)
80900      A6=0.4176552*(PR**5)
81000      A7=-0.0212015*(PR**6)
81100      Y=A1+A2+A3+A4+A5+A6+A7
81200      WAP=(Y*30.378*2.2)/50.0
81300      RETURN
81400      END

```

Diesel Gas Turbine Program

```

100 C DATA FILE ENGIN1
200 C *****
300 C
400 C CONTAINS COMPRESSOR DATA FOR USE IN PROGRAM DAT
500 C
600 C NO OF PRESSURE RATIO POINTS AND EFFICIENCY POINTS
700      11      5
800 C PRESS RATIO,EFFICIENCY,MIN AIR FLOW,MAX AIR FLOW,MIN SPEED,
900 C MAX SPEED,AIR FLOW VALUE AT CUT OFF LINE,EFF AT THAT VALUE
1000 C ALL DATA AREAS LEFT BLANK ARE TAKEN TO BE ZERO
1100  1.4      60.0      0.0      346.0      0.0      57500.0      140.0
1200  1.4      62.5      136.0     325.0     50400.0     54800.0
1300  1.4      65.0      154.0     300.0     50400.0     53500.0
1400  1.4      67.5      192.0     272.0     50400.0     52000.0
1500  1.4      70.0      200.0     253.0     50400.0     51600.0
1600  1.5      60.0      0.0      390.0      0.0      62500.0      170.0
1700  1.5      62.5      144.0     368.0     56500.0     61500.0
1800  1.5      65.0      180.0     344.0     56500.0     60000.0
1900  1.5      67.5      212.0     312.0     56000.0     58500.0
2000  1.5      70.0      220.0     298.0     56000.0     58000.0
2100  1.6      60.0      0.0      425.0      0.0      68400.0      200.0
2200  1.6      62.5      0.0      402.0      0.0      66000.0
2300  1.6      65.0      196.0     380.0     60200.0     65000.0
2400  1.6      67.5      232.0     347.0     60800.0     63300.0
2500  1.6      70.0      240.0     336.0     61000.0     62500.0
2600  1.7      60.0      0.0      452.0      0.0      72200.0      235.0
2700  1.7      62.5      0.0      432.0      0.0      71000.0
2800  1.7      65.0      0.0      410.0      0.0      69800.0
2900  1.7      67.5      248.0     376.0     65000.0     67400.0
3000  1.7      70.0      256.0     363.0     65000.0     66600.0
3100  1.8      60.0      0.0      476.0      0.0      76000.0      254.0
3200  1.8      62.5      0.0      456.0      0.0      74500.0
3300  1.8      65.0      230.0     434.0     69300.0     73300.0
3400  1.8      67.5      262.0     402.0     69300.0     71500.0
3500  1.8      70.0      276.0     386.0     69300.0     71000.0
3600  2.0      60.0      0.0      516.0      0.0      82400.0      300.0
3700  2.0      62.5      0.0      498.0      0.0      81000.0
3800  2.0      65.0      0.0      476.0      0.0      79500.0
3900  2.0      67.5      292.0     440.0     75500.0     77200.0
4000  2.0      70.0      316.0     414.0     75500.0     76500.0
4100  2.2      60.0      0.0      548.0      0.0      89200.0      340.0 \
      \      70.1
4200  2.2      62.5      0.0      530.0      0.0      87000.0
4300  2.2      65.0      0.0      514.0      0.0      85500.0
4400  2.2      67.5      332.0     472.0     81300.0     84000.0
4500  2.2      70.0      380.0     412.0     81500.0     81500.0
4600  2.3      60.0      0.0      564.0      0.0      91800.0      356.0 \
      \      69.5
4700  2.3      62.5      0.0      546.0      0.0      90000.0
4800  2.3      65.0      310.0     528.0     84500.0     88200.0

```

Data for Diesel Gas Turbine Program

4900	2.3	67.5	364.0	486.0	84500.0	86600.0	
5000	2.3	70.0	440.0	412.0	85000.0	84600.0	
5100	2.4	60.0		578.0		94000.0	375.0 \
		68.9					
5200	2.4	62.5		560.0		92300.0	
5300	2.4	65.0	344.0	540.0	87300.0	91200.0	
5400	2.4	67.5	396.0	498.0	87300.0	89000.0	
5500	2.4	70.0	480.0	420.0	88500.0	87400.0	
5600	2.5	60.0	0.0	588.0	0.0	95500.0	390.0 \
		68.4					
5700	2.5	62.5	0.0	568.0	0.0	94500.0	
5800	2.5	65.0	376.0	548.0	90000.0	93000.0	
5900	2.5	67.5	432.0	508.0	90000.0	91500.0	
6000	2.5	70.0	519.0	428.0	92200.0	90000.0	
6100	2.6	60.0		598.0		98000.0	404.0 \
		67.8					
6200	2.6	62.5		578.0		97500.0	
6300	2.6	65.0	412.0	556.0	92500.0	95800.0	
6400	2.6	67.5	468.0	516.0	93200.0	94500.0	
6500	2.6	70.0	556.0	435.0	95500.0	92500.0	

```

100 C DATA FILE SUPDAT
200 C *****
300 C
400 C CONTAINS ENGINE DATA TO RUN PROGRAM DGT
500 C NO OF ENGINE SPDS,PULSE OR CONST PRESS SYSTEM
600 8,1
700 C ATMOS P + T,COOLER EFFECTIVENESS,COOLANT TEMP,HEAT EXCHGR\
    \ EFFECTIVENESS
800 C COMPRESSOR MAP NO,TURBINE HSG AREA,PLOTTING INSTRUCTION
900 1.0,21.0,0.0,21.0,0.0,2,2.6,0
1000 C ENG SPD,SCAV RATIO,A/F RATIO,BOOST,PRESTON MTR READING,
1100 C ENG PRESSURE DROP,TURB INLET TEMP,TURB OUTLET TEMP
1200 1000.0,1.65,24.53,1.52,34.0,0.01,695.0,546.0
1300 1200.0,1.45,26.0,1.69,42.0,0.07,676.0,541.0
1400 1400.0,1.4,26.5,1.76,47.0,0.07,673.0,529.0
1500 1600.0,1.4,29.8,1.91,44.0,0.10,677.0,530.0
1600 1800.0,1.0,31.5,1.94,51.0,0.13,682.0,528.0
1700 2000.0,1.0,32.14,1.89,50.0,0.13,694.0,524.0
1800 2200.0,1.2,32.63,1.93,54.0,0.12,732.0,553.0
1900 2400.0,1.0,33.41,1.92,53.0,0.11,691.0,537.0

```

Data for Diesel Gas Turbine Program

A1.4 Real Time Logging Program

This program was used for logging auxiliary fuel consumption, from a bottled gas supply, and several temperatures.

Notation

C	Data point number
D	Day
G(I)	Successive gas bottle weight readings
H	Sample time (duration)
K	Year
M	Month
T1(J)	Temperatures (up to J=9)
T8	Ambient temperature
W1	One minute average of gas flow
W2	Gas flow based on total sample period

```

5 REM PROGRAM VARILOG(RP)
10 REM LOGGING PROGRAM FOR GAS FLOW AND NINE TEMPERATURES
20 DIMG(15),T(8,15),T1(8),P(15),Q(15),R(15)
25 PRINT " SAMPLE TIME IN MINS (MAX=15) ?" \ INPUT H
26 IF H>15 GO TO 28
27 GO TO 30
28 PRINT " NOT VALID, RETYPE" \ GO TO 25
30 PRINT " INPUT DAY,MONTH,YEAR" \ INPUT D,M,K
40 PRINT " INPUT AMBIENT TEMP" \ INPUT T8
45 PRINT " PAPER TAPE DATA OPTION? (1=YES,0=NO)"
46 INPUT F
50 PRINT " INPUT DATA POINT 0 TO STOP; 1,2,3 ETC FOR RUN"
51 IF F=0 GO TO 55 \ PRINT #2:H,CHR$(13,10),D,CHR$(13,10)
52 PRINT #2: M,CHR$(13,10),K,CHR$(13,10),T8,CHR$(13,10)
55 PRINT " DATA POINT?"
60 INPUT C
62 STIME(0,0,0)
65 IF C=0 GO TO 290
70 FOR I=0 TO H
80 SETDVM(2,3)
85 MEASURE(G(1),0)
86 RTIME(P(1),Q(1),R(1))
90 SETDVM(4,3)
92 LET V=0
95 SCAN(T(0,1),1,9, LINE 100 )
96 IF V=0 GO TO 96
99 GO TO 105
100 LET V=1 \ RETURN
105 IF I=H GO TO 120
107 LET U=0
110 DELAY(59, LINE 115 )
111 IF U=0 GO TO 111
112 GO TO 120
115 LET U=1 \ RETURN
120 NEXT I

```

PDP 11/05 BASAC IV LOGGING PROGRAM (1)


```

140 PRINT D;"/";M;"/";K;" DATA POINT";C
141 IF F=0 GO TO 145 \ PRINT #2:C,CHR$(13,10)
145 PRINT "*****" \ PRINT
150 LET Z=0
155 FOR I=1 TO H
160 LET Y=(G(I-1)-G(I))
161 LET Y1=(Q(I)+(R(I)/60))-(Q(I-1)+(R(I-1)/60))
162 LET Z=Z+(Y/Y1)
170 NEXT I
180 LET W1=(Z/H)*10*60
190 PRINT " GAS FLOW 1 MIN AVG=";W1;"KG/HR"
191 IF F=0 GO TO 200 \ PRINT #2:W1,CHR$(13,10)
200 LET W2=G(0)-G(H)
201 LET W3=(Q(H)+(R(H)/60))-(Q(0)+(R(0)/60))
202 LET W2=(W2/W3)*10*60
210 PRINT " GAS FLOW FULL TIME=";W2;"KG/HR" \ PRINT
211 IF F=0 GO TO 220 \ PRINT #2:W2,CHR$(13,10)
220 FOR J=0 TO 8 \ LET T1(J)=0
230 FOR I=0 TO H \ LET T1(J)=T1(J)+T(J,I) \ NEXT I
240 LET T1(J)=(((T1(J))/((H+1)*4.10000E-03))*100)+T8
250 PRINT " T";J+1;"=";T1(J);"DEG C"
255 IF F=0 GO TO 256 \ PRINT #2:T1(J),CHR$(13,10)
256 NEXT J
260 PRINT \ PRINT \ PRINT \ PRINT
280 GO TO 55
290 IF F=0 GO TO 295 \ PRINT #2:C,CHR$(13,10)
295 STOP
300 END

```

DATA POINT?

16

28 / 2 / 79 DATA POINT 16

GAS FLOW 1 MIN AVG= 5.87023 KG/HR

GAS FLOW FULL TIME= 5.87023 KG/HR

T 1 = 52.6504 DEG C

T 2 = 53.0894 DEG C

T 3 = 510.211 DEG C

T 4 = 468.463 DEG C

T 5 = 51.2114 DEG C

T 6 = 387.959 DEG C

T 7 = 671.642 DEG C

T 8 = 551.236 DEG C

T 9 = 518.919 DEG C

DATA POINT?

17

28 / 2 / 79 DATA POINT 17

GAS FLOW 1 MIN AVG= 4.90801 KG/HR

GAS FLOW FULL TIME= 4.90776 KG/HR

T 1 = 47.7724 DEG C

T 2 = 47.8862 DEG C

T 3 = 535.285 DEG C

T 4 = 490.561 DEG C

T 5 = 47.3089 DEG C

T 6 = 385.333 DEG C

T 7 = 672.618 DEG C

T 8 = 550.301 DEG C

T 9 = 509.024 DEG C

TYPICAL EXAMPLE OF PRINTOUT FROM PDP 11/05 LOGGING PROGRAM

A1.5 Microprocessor Logging Program

This program was used for crankshaft triggered data acquisition. A listing follows written in assembly language for the Quarndon Microcomputer System (QMS) Based on the Intel 8080 processor.

;PROGRAM NAME - MCLP

;PROGRAMMER - MJM

;DATE - 1 5 78

;MULTI-CHANNEL LOGGING PROGRAM

```

0000          MCLP:
0000      CD3306      CALL INIT      ;INITIALISE TTY I/F.
0003      CD3B06      CALL NEWL      ;CLRF
0006      AF          XRA A
0007      D380        OUT 80H        ;RESET 'IN' TO 0.
0009      CD2405      CALL TSET      ;LOAD THE LOOK-UP TABLE
000C      210002      LXI H,TABP     ;INITIALISE H+L TO START
                                       ;ADDRESS OF RESULTS TAELE.

000F      1600        MVI D,00H
0011      1E04        MVI E,04H     ;(E) IS SCAN COUNTER

0013          LOPA:
0013      DB81        IN 81H
0015      1F          RAR
0016      DA1300      JC LOPA        ;WAIT FOR DAS RESET.

0019          LOPB:
0019      DB81        IN 81H
001B      1F          RAR
001C      D21900      JNC LOPB      ;WAIT FOR CTRL TO BE 1

001F          LOPC:
001F      DB81        IN 81H
0021      17          RAL
0022      D21F00      JNC LOPC      ;WAIT FOR ADC TO FINISH
0025      DB80        IN 80H        ;GET VALUE
0027      77          MOV M,A       ;STORE IN RESULTS TABLE
0028      23          INX H         ;BUMP RESULTS TABLE POINTER
0029      DB81        IN 81H
002B      1F          RAR
002C      D23B00      JNC YCON      ;JUMP IF 'GO' IS NOW 0
002F      CD2B06      CALL PULS     ;OTHERWISE GENERATE AN 'IN'
0032      C31F00      JMP LOPC      ;AND JUMP BACK
0035      00          NOP
0036      00          NOP
0037      00          NOP          ;THE NEXT PART OF THE PROGRAM
0038      00          NOP          ;IS EXECUTED AFTER EACH SCAN.
0039      00          NOP
003A      00          NOP
003B          YCON:
003B      1D          DCR E         ;DECREMENT SCAN COUNT
003C      C24300      JNZ RETS      ;JUMP IF MORE SCANS REQUIRED
003F      CDEC05      CALL PRIN     ;IF FINISHED PRINT RESULTS
0042      76          HLT
0043          RETS:
0043      7B          MOV A,E       ;GET SCAN COUNT
0044      FE03        CPI 03        ;03=SCAN COUNT-1

```

Microprocessor Logging Program

```

0046 C21900 JNZ LOPB ;JUMP BACK IF NOT END OF FIRST SC
0049 7D MOV A,L ;OTHERWISE ISOLATE CHANNEL/
;COLUMN COUNT
004A 321601 STA XCCT ;AND STORE.
004D C31900 JMP LOPE ;NOW JUMP BACK
    
```

;EXTERNAL ADDRESS REFERENCES

```

063B NEWL EQU 063EH
0633 INIT EQU 0633H
0524 TSET EQU 0524H
0200 TABP EQU 0200H
062B PULS EQU 062EH
05EC PRIN EQU 05ECH
0116 XCCT EQU 0116H
    
```

END

NO PROGRAM ERRORS

SYMBOL TABLE

* 01

A	0007	B	0000	C	0001	D	0002
E	0003	H	0004	INIT	0633	L	0005
LOPA	0013	LOPE	0019	LOPC	001F	M	0006
MCLP	0000 *	NEWL	063B	PRIN	05EC	PSW	000C
PULS	062B	RETS	0043	SP	0006	TABP	0200
TSET	0524	XCCT	0116	YCON	003E		

#ET=7.08.2 PT=9.9 IO=1.0

Microprocessor Logging Program

;SUEFOUTINE NAME - PFIN

;PPOGFAMMEP - MJM

;DATE - 1 5 78

;USED TO PRINT DECIMAL EQUIVALENT OF
;BINARY VALUES READ FROM THE DAS.

```

05EC          OFG  05ECH
05EC          PFIN:
05EC      3A1601      LDA  XCCT          ;PUT COLUMN COUNTER
05EF      4F          MOV  C,A          ;INTO REGISTER C
05F0      3E04        MVI  A,04H        ;STORE SCAN COUNTER
05F2      321701      STA  LINF        ;IN ADDRESS LINF.
05F5      210002      LXI  H,TABL       ;GET STARTING ADDRESS OF
                                          ;RESULTS TABLE TO H AND L.

05F8          XLOP:
05F8      CD5D05      CALL XSET          ;CLEAR TOTL, INITIALISE XSET
                                          ;TO START OF LOOK-UP TABLE.
05FB      7E          MOV  A,M          ;GET FIRST VALUE.
05FC      CD6C05      CALL XADC          ;CONVERT TO BCD.
05FF      CDF305      CALL FDEC          ;PRINT BCD CHARACTER.
0602      23          INX  H            ;BUMP RESULTS TABLE POINTER.
0603      0F          DCF  C            ;DECREMENT COLUMN COUNT
0604      CA0D06      JZ   XNEW          ;IF ALL COLUMNS DONE, JUMP.
0607      CD2206      CALL SPAS          ;OTHERWISE PRINT 2 SPACES.
060A      C3F805      JMF  XLOP          ;AND GO BACK FOR NEXT VALUE
060D          XNEW:
060D      CD3B06      CALL NEWL          ;CRLF
0610      00          NOP
0611      00          NOP
0612      00          NOP
0613      3A1701      LDA  LINF
0616      3D          DCF  A            ;DECREMENT SCAN COUNT
0617      0B          RZ                ;RETURN IF ALL RESULTS PRINTED
0618      321701      STA  LINF        ;OTHERWISE SAVE
061B      3A1601      LDA  XCCT        ;RE-INITIALISE REGISTER C
061E      4F          MOV  C,A          ;WITH COLUMN COUNT
061F      C3F805      JMF  XLOP        ;AND JUMP BACK FOR NEXT VALUE

```

;EXTERNAL ADDRESS REFERENCES

```

0116          XCCT EQU  0116H
0117          LINF EQU  0117H
0200          TABL EQU  0200H
055D          XSET EQU  055DH
056C          XADC EQU  056CH
05B3          FDEC EQU  05B3H
0622          SPAS EQU  0622H
063B          NEWL EQU  063EH
          END

```

NO PROGRAM ERRORS

Microprocessor Logging Program

;SUBROUTINE NAME - XSET

;PROGRAMMER - MJM

;DATE - 1 5 78

;USED TO INITIALISE TOTL TO ZERO, AND
;XPTP TO START ADDRESS OF LOOK-UP TABLE

```

055D          ORG 055DH
055D          XSET:
055D          E5          PUSH H
055E          AF          XPA A          ;CLEAP A
055F          211201      LXI H, TOTL      ;PUT 0112 IN H AND L
0562          77          MOV M, A          ;PUT ZERO IN 0112
0563          23          INX H            ;POINT AT 0113
0564          77          MOV M, A          ;PUT ZERO IN 0113 AS WELL
0565          23          INX H            ;POINT AT 0114
0566          77          MOV M, A          ;ZERO INTO 0114
0567          23          INX H            ;POINT AT 0115
0568          3601      MVI M, 01H        ;01 INTO 0115
                                ;START ADDRESS OF LOOK-UP
                                ;TABLE IS 0100

056A          E1          POP H
056E          C9          RET

```

;EXTERNAL ADDRESS REFERENCES

0112 TOTL EQU 0112H

END

NO PROGRAM ERRORS

SYMBOL TABLE

* 01

A	0007	B	0000	C	0001	D	0002
E	0003	H	0004	L	0005	M	0006
PSW	0006	SF	0006	TOTL	0112	XSET	055D *

#ET=3:35.2 PT=4.8 IO=0.8

;SUBROUTINE NAME - TSET

;PROGRAMMER - MJM

;DATE - 1 5 78

;USED TO INITIALISE THE LOOK-UP TABLE
;AND THE STARTING ADDRESS OF THE
;RESULTS TABLE IN 0110 AND 0111.

```

0524          ORG 0524H
0524          TSET:
0524 210001    LXI H,0100H
0527 3600     MVI M,00H
0529 23       INX H           ;M=0101
052A 3639     MVI M,39H
052C 23       INX H           ;M=0102
052D 3600     MVI M,00H
052F 23       INX H           ;M=0103
0530 3678     MVI M,78H
0532 23       INX H           ;M=0104
0533 3601     MVI M,01H
0535 23       INX H           ;M=0105
0536 3656     MVI M,56H
0538 23       INX H           ;M=0106
0539 3603     MVI M,03H
053B 23       INX H           ;M=0107
053C 3613     MVI M,13H
053E 23       INX H           ;M=0108
053F 3606     MVI M,06H
0541 23       INX H           ;M=0109
0542 3625     MVI M,25H
0544 23       INX H           ;M=010A
0545 3612     MVI M,12H
0547 23       INX H           ;M=010F
0548 3650     MVI M,50H
054A 23       INX H           ;M=010C
054P 3625     MVI M,25H
054D 23       INX H           ;M=010D
054E 3600     MVI M,00H
0550 23       INX H           ;M=010E
0551 3650     MVI M,50H
0553 23       INX H           ;M=010F
0554 3600     MVI M,00H

```

;TABLE COMPLETE

```

0556 23       INX H           ;M=0110
0557 3600     MVI M,00
0559 23       INX H           ;M=0111
055A 3602     MVI M,02H

```

Microprocessor Logging Program

;SUBROUTINE NAME - XADC

;PROGRAMMER - MJM

;DATE - 1 5 78

;USED TO CONVERT 8-BIT BINARY IN
;A REGISTER TO 4 BCD CHARACTERS
;IN THE ADDRESSES LABELLED 'TOTL'.

```

056C          ORG 056CH
056C          XADC:
056C          E5          PUSH H          ;SAVE H AND L
056D          C5          PUSH B          ;SAVE B AND C
056E          00          NOP
056F          2A1401      LHLD XPTR      ;POINTER TO LOOK-UP TABLE
0572          STRT:
0572          1F          RAR            ;ISOLATE LSB IN CY.
0573          F5          PUSH PSW      ;SAVE SHIFTED BINARY
0574          DA7F05      JC XCON       ;JUMP IF ISOLATED BIT=1
0577          23          INX H         ;OTHERWISE PUMP LOOK-UP
0578          DAA305      JC GOON       ;TABLE POINTER AND GO ON.
057B          00          NOP
057C          00          NOP
057D          00          NOP
057E          00          NOP
057F          XCON:
057F          56          MOV D,M       ;GET MS TABLE ENTRY TO D
0580          23          INX H
0581          5E          MOV E,M
0582          221401      SHLD XPTR     ;PUT BACK TABLE POINTER
0585          00          NOP
0586          00          NOP
0587          00          NOP
0588          2A1201      LHLD TOTL     ;PUNNING TOTAL TO H AND L
058B          7D          MOV A,L
058C          83          ADD E
058D          27          DAA          ;PUT LS HALF IN A
                                ;ADD LS TABLE ENTRY
                                ;DECIMAL ADJUST RESULT
                                ;(CONVERT TO BCD.)
058E          6F          MOV L,A
                                ;PUT BACK INTO L
058F          D29605      JNC NEXT     ;JUMP IF NO CY FROM DAA
0592          24          INR H
                                ;OTHERWISE ADD IN CY TO H
0593          00          NOP
0594          00          NOP
0595          00          NOP
0596          NEXT:
0596          7C          MOV A,H
                                ;PUT MS HALF INTO A
0597          82          ADD D
                                ;ADD MS TABLE ENTRY
0598          27          DAA
                                ;AGAIN CONVERT TO BCD
0599          67          MOV H,A
                                ;PUT RESULT BACK INTO H
059A          221201      SHLD TOTL
                                ;PUT NEW RUNNING TOTAL BACK
059D          00          NOP

```

Microprocessor Logging Program

```

059E 00      NOP
059F 00      NOP
05A0 2A1401  LHLD XPTR      ;GET BACK TABLE POINTER
05A3      GOON:
05A3 23      INX  H      ;BUMP IT TO START OF NEXT
                        ;COMPLETE TABLE ENTRY
05A4 7D      MOV  A,L      ;LS BYTE OF POINTER TO A
05A5 FE10    CPI  10H     ;IS POINTER BEYOND TABLE?
05A7 CAAE05  JZ   XFIN      ;JUMP IF YES
05AA F1      POP  PSW     ;OTHERWISE RESTORE PSW
05AE C37205  JMP  STRT      ;AND JUMP BACK
05AE      XFIN:
05AE F1      POP  PSW     ;ALL FINISHED NOW, RESTORE
05AF 1F      RAR      ;PSW AND SHIFT NINTH TIME
05B0 C1      POP  B      ;RESTORE B
05B1 E1      POP  H      ;AND H
05B2 C9      RET      ;AND RETURN TO MAIN
    
```

;EXTERNAL ADDRESS REFERENCES

```

0114      XPTR EQU 0114H
0112      TOTL EQU 0112H
    
```

END

NO PROGRAM ERRORS

SYMBOL TABLE

* 01

A	0007	B	0000	C	0001	D	0002
E	0003	GOON	05A3	H	0004	L	0005
M	0006	NEXT	0596	PSW	0006	SP	0006
STRT	0572	TOTL	0112	XADC	056C *	XCON	057F
XFIN	05AE	XPTR	0114				

#ET=7:45.6 PT=10.8 IO=0.9

Microprocessor Logging Program

; SUBROUTINE NAME - PTEC

; PROGRAMMER - MJM

; DATE - 1 5 78

; USED TO PRINT THE FOUR BCD CHARACTERS
; HELD IN ADDRESSES TOTL AND TOTL+1

```

05E3          OFC  05E3H
05E3          PTEC:
05E3          E5          PUSH H          ;SAVE EXISTING H AND L
05E4          2A1201      LHLD TOTL      ;GET TOTAL TO H AND L
05E7          FE          PUSH PSW       ;SAVE A
05E8          E5          PUSH H          ;SAVE TOTAL ON STACK
05E9          7C          MOV  A,H        ;PUT MS CODES IN A
05EA          CDD605      CALL FONE     ;PRINT THE FIRST ONE
05ED          062E        MVI  E,2EH     ;ASCII '.'
05EF          CD0005      CALL PUTC   ;PRINT IT!
05C2          E1          POP  H
05C3          7C          MOV  A,H        ;GET MS CODES BACK
05C4          E5          PUSH H
05C5          CDE305      CALL FTWO     ;PRINT THE SECOND CODE
05C8          E1          POP  H
05C9          7D          MOV  A,L        ;GET LS CODES TO A
05CA          E5          PUSH H
05CB          CDD605      CALL FONE     ;PRINT FIRST OF LS CODES
05CE          E1          POP  H
05CF          7D          MOV  A,L        ;GET LS CODES BACK
05D0          CDE305      CALL FTWO     ;PRINT SECOND
05D3          F1          POP  PSW
05D4          E1          POP  H
05D5          C9          RET
05D6          FONE:
05D6          E6F0        ANI  0FH      ;MASK OFF LS 4-BITS
05D8          1F          RAR
05D9          1F          RAR          ;SHIFT FIRST CODE OF 161F
05DA          1F          RAR          ;TO LS END OF A REGISTER
05DE          1F          RAR
05DC          C630        ADI  3EH      ;CONVERT TO ASCII
05DE          47          MOV  E,A        ;PUT ASCII IN E REGISTER
05DF          CD0005      CALL PUTC   ;PRINT IT!
05E2          C9          RET          ;AND RETURN
05E3          PTWO:
05E3          E60F        ANI  0FH      ;MASK OFF UPPER 4-BITS
05E5          C630        ADI  3EH      ;CONVERT TO ASCII
05E7          47          MOV  E,A        ;MOVE INTO E REGISTER
05E8          CD0005      CALL PUTC   ;PRINT
05EF          C9          RET          ;AND RETURN

```

; EXTERNAL ADDRESS REFERENCES

Microprocessor Logging Program

0112 TOTL EQU 0112H
0500 PUTC EQU 0500H

END

NO PROGRAM ERRORS

SYMBOL TABLE

* 01

A	0007	E	0000	C	0001	D	0002
E	0003	H	0004	L	0005	M	0006
FDEC	05E3 *	PONE	05D6	PSW	0006	PTWO	05E3
PUTC	0500	SF	0006	TOTL	0112		

#ET=6:16.5 FT=8.0 IO=0.6

Microprocessor Logging Program

;MISCELLANEOUS SUBROUTINES

;PROGRAMMER - MJM

;DATE - 1 5 73

;1. SUBROUTINE INIT

; USED TO INITIALISE THE TTY INTERFACE

```

0633          OFG  0633H
0633          INIT:
0633  2102FF    LXI  H,0FF02H          ;CONTROL REG. ADDRESS
0636  3643     MVI  M,43H
0638  3651     MVI  M,51H
063A  C9      RET

```

;2. SUBROUTINE PUTC

; USED TO PRINT CHARACTER AT THE TTY

; CHARACTER CODE MUST BE IN REG.B

```

0500          OFG  0500H
0500          PUTC:
0500  F5       PUSH PSW                ;SAVE PSW
0501  E5       PUSH H                  ;AND H
0502  2102FF    LXI  H,0FF02H          ;CONTROL REG. ADDRESS
0505          NFDY:
0505  7E       MOV  A,M                ;PUT WORD IN A
0506  E602     ANI  02H                ;CHECK D1 OF WORD
0508  CA0505   JZ   NFDY              ;JUMP IF NOT READY
050B  23       INX  H                  ;OTHERWISE POINT TO
                                        ;TTY BUFFER
050C  70       MOV  M,B                ;CHAR. CODE INTO BUFFER
050D  2E       DCX  H                  ;BACK TO CONTROL REG.
050E          NFIN:
050E  7E       MOV  A,M                ;CTPL WORD INTO A
050F  E602     ANI  02H                ;TEST AGAIN
0511  CA0E05   JZ   NFIN              ;WHEN PRINTED
0514  E1       POP  H                  ;DESTOPE H AND L
0515  F1       POP  PSW                ;AND PSW
0516  C9      RET                      ;AND RETURN

```

;3. SUBROUTINE NEWL

; USED TO PRINT CARriage RETURN/

; LINE FEED.

```

063F          OFG  063EH
063F          NEWL:
063E  C5       PUSH B
063C  060A     MVI  B,0AH                ;LINE FEED
063E  CD0005   CALL PUTC

```

Microprocessor Logging Program

```

0641 060D MVI E,0DH ;CAPSPACE RETURN
0643 CD0005 CALL PUTC
0646 0600 MVI E,00H ;NULL
0648 CD0005 CALL PUTC
064B C1 POP E
064C C9 RET

```

```

;4. SUBROUTINE SFAS
; USED TO PRINT TWO SPACES BETWEEN
; SUCCESSIVE COLUMNS ON THE PRINTOUT

```

```

0622 OFC 0622H
0622 SFAS:
0622 0620 MVI E,20H ;ASCII SPACE
0624 CD0005 CALL PUTC
0627 CD0005 CALL PUTC
062A C9 RET

```

```

;5. SUBROUTINE FULS
; USED TO GENERATE A PULSE ON THE
; 'IN' LINE TO THE DAS LOGIC.

```

```

062B OFC 062EH
062B FULS:
062E 3E01 MVI A,01H ;MAKE D0 A 1
062D D380 OUT 80H ;SEND TO PORT 80
062F 3D DCF A ;D0 = 0
0630 D380 OUT 80H ;SEND AGAIN
0632 C9 RET

```

```

END

```

```

NO PROGRAM ERRORS

```

Microprocessor Logging Program

A1.6 Control Programs

These programs were used to effect microprocessor control of the stepper motors. Two programs are shown; one for speed control, the other for positional control. The programs are written in assembly language for the Texas 9900 microprocessor.

PROGRAM STP1

F000	02E0	LWP1 > F100	Start
F002	F100		
F004	2E42	XOP 2,9	Input 'N'
F006	F004	+ > F004	Null entry return
F008	F004	+ > F004	Error entry return
F00A	020C	LI 12,0	CRU address 0
F00C	0000		
F00E	C0C2	MOV 2,3	Copy N into R3
F010	1E00	SBZ 0	O/P 0 at 0
F012	0603	DEC 3	N = N-1
F014	1BFE	JH > F010	Loop for delay
F016	C0C2	MOV 2,3	N = > R3
F018	1D00	SBO 0	O/P 1 at 0
F01A	0603	DEC 3	N = N-1
F01C	1BFE	JH > F018	Loop for delay
F01E	10F7	JMP > F00C	Loop to O/P 0

Program to drive stepping motor at varying speed dependent upon 'N'.

PROGRAM IDT STP2

F000	02E0	LWP1 > F100	Choose work space
F002	F100		
F004	2E42	XOP R2,9	Input D
F006	F004	+ > F004	Null entry return
F008	F004	+ > F004	Error entry return
F00A	020C	LI 12, >120	CRU base address PO
F00C	0120		
F00E	0203	LI 3, > 80	} Motor speed selection
F010	0080		
F012	C003	MOV 3,4	} Output one pulse
F014	1E00	SBZ 0	
F016	0604	DEC 4	
F018	1BFD	JH > F014	
F01A	C103	MOV 3,4	
F01C	1D00	SBO 0	
F01E	0604	DEC 4	
F020	1BFD	JH > F01C	
F022	0602	DEC 2	D = D-1
F024	1BF6	JH > F012	Loop if D > 0
F026	2FA0	XOP @ > F02C,14	Write message
F028	F02C		
F02A	10EC	JMP > F004	Return to start
F02C	2020		Message begins
F02E	2020		
F030	2020		
F032	4649	F I	
F034	4E49	N I	
F036	5348	S H	
F038	4544	E D	
F03A	0A0D	CR LF	
F03C	0000		Message ends

Program for open loop position control of stepper motor.
Position selected by number of steps 'D'.

APPENDIX 2

FUEL FLOW RATE MEASURING SYSTEM

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Fuel Flow Rate Measuring System

The problems of fuel flow measurement for computerised data logging are well known. Medium size, automotive, engine work means low flow rates giving low frequency outputs (in the region of 50 Hz max) leading to difficulty in reliable frequency to dc conversion. In this paper a system is described giving a directly proportional analogue voltage output suitable for steady state conditions.

A system suitable for data acquisition is based on a turbine flow meter giving an analogue voltage output but such systems can cost in the region of £1,600. The system under consideration may be built for a fraction of this cost.

Using, on line, computerised data logging in engine performance tests it is desirable to have the parameters to be measured in the form of directly proportional analogue voltage outputs. Most data acquisition systems deal with analogue voltages.

Fuel flow rates have previously been measured by timing a fixed amount of fuel (50 or 100 cc) flowing from a pipette. It is now possible to get automatic versions using optical and electronic devices. These are not suitable for use with voltage logging systems.

In the course of a research project on highly rated diesel engines, a need was discovered for measuring the steady state fuel flow. The output was required in the form of an analogue voltage (max 10v) for compatibility with the logging system.

Design Details

The system is based on a Litre Meter LM25GN rotor type flow transducer. This requires an input voltage of 10-20v d.c. at under 10mA and delivers a 2.5v peak to peak square wave output whose frequency is directly proportional to flow rate.

The output is suitable to drive (the 74 series) TTL (Transistor-Transistor Logic) so the readout unit was based on this system and is capable of giving either analogue voltage or parallel binary outputs or both if desired. A schematic circuit is given in fig. 1.

The output from the Litre Meter is filtered to remove a d.c. component (fig. 2a.) and fed through an RS 741 operational amplifier used as a unity gain, high impedance, buffer (1) to isolate the input and passed through a switch to protect the TTL circuitry from mains 'switch on' transients.

The signal is then passed to an RS 555 multivibrator (fig. 2a.) used in a monostable mode (2). (This proved necessary to avoid spurious triggering of the counter.) The multivibrator delivers TTL compatible signals which are inverted and fed to two cascaded SN 74161 synchronous 4 bit binary counters (3). The counters sample the input pulses for a period of approx. 4 secs. The counter binary outputs are fed through an SN 74100 8 bit latch (3) to an RSZN 425 E Digital to Analogue Converter (4). The latch is used so that a constant voltage output can be maintained. The output is updated at the end of each sampling period. The converter output is fed through an RS 741 operational amplifier used in the non-inverting mode (1) with offset null adjustment for setting zero and variable gain to set full scale output.

At the end of the sampling period the counters and the latch require resetting such that the latch transmits the updated frequency counts and the counters are reset to zero. This is facilitated (fig. 2b.) by using an RS 555 multivibrator in an astable mode (2) giving an adjustable sample period of up to 4 secs. This signal triggers an SN 74121 monostable multivibrator (3) giving a reset pulse of 35 ns duration. This pulse is inverted by an SN 7400 'Nand' gate (3) and used to update the latch. The same signal is fed through a further 5 SN 7400 (3) inverting 'Nand' gates and used to reset the counters to zero. The 5 gates give a time delay of 50 ns. If a time delay is not used the counters reset prior to data transfer across the latch.

The usual precautions are taken with regard to capacitor decoupling using 74 series logic. The system requires power supplies of \pm 15v for the operational amplifiers and + 5v for the logic circuits. The Litre Meter is supplied with + 12.6v smoothed d.c. taken from the 5v supply circuit just prior to the voltage regulator.

Discussion

The sampling period is adjustable; a longer period gives increased accuracy. The gain of the output amplifier is also adjustable and can be set to give a desired output at maximum flow prior to calibration.

To facilitate 'in situ' calibration the mechanical fuel lift pump was replaced with an S.U. electric fuel pump. During calibration it was found that the flow transducer was extremely sensitive to variation in delivery pressure head. This necessitated the construction of a recirculatory constant head fuel delivery system. The flow transducer was placed in such a position as to measure nett fuel flow, ignoring returns such as spill and recirculatory fuel flow for lubrication and cooling of the D.P.A. injector pump on the Diesel engine (fig. 3.).

The system proved to be insensitive to fluctuations in ambient temperature and pressure (range of 20 mm Hg and 7°C) which occurred during calibration.

Fuel was collected for 10-30 mins depending upon flow rate and was weighed on an Avery Type 3205 ABA scale to 1 oz (28.4 gm) and timed to 1 sec using a Smith's stop clock.

The calibration (fig. 4.) proved linear down to 2.5 kg/hr. It was not possible to calibrate at lower flow rates as the flow rate transducer rotor proved unreliable. A smaller transducer would overcome this problem.

This readout system is versatile in that the output can be scaled to the user requirements up to a maximum output of 13.5v. Litre Meter Ltd supply a range of flow transducers to suit most applications.

The readout system, or slight variations would be suitable for any frequency based system that will deliver square wave output of 2v peak to peak such as engine or turbocharger speed measurement using relevant transducers.

Conclusion

The fluid flow rate measuring system described is a versatile system capable of measuring a wide range of fluid flows with an accuracy better than $\pm 1.5\%$ of full scale, giving a linear relationship.

The whole flow system can be built for under £120. This figure neglects the cost of conversion to a constant head fuel system. This represents a significant saving over available systems, whilst having an accuracy that is acceptable for a wide range of applications.

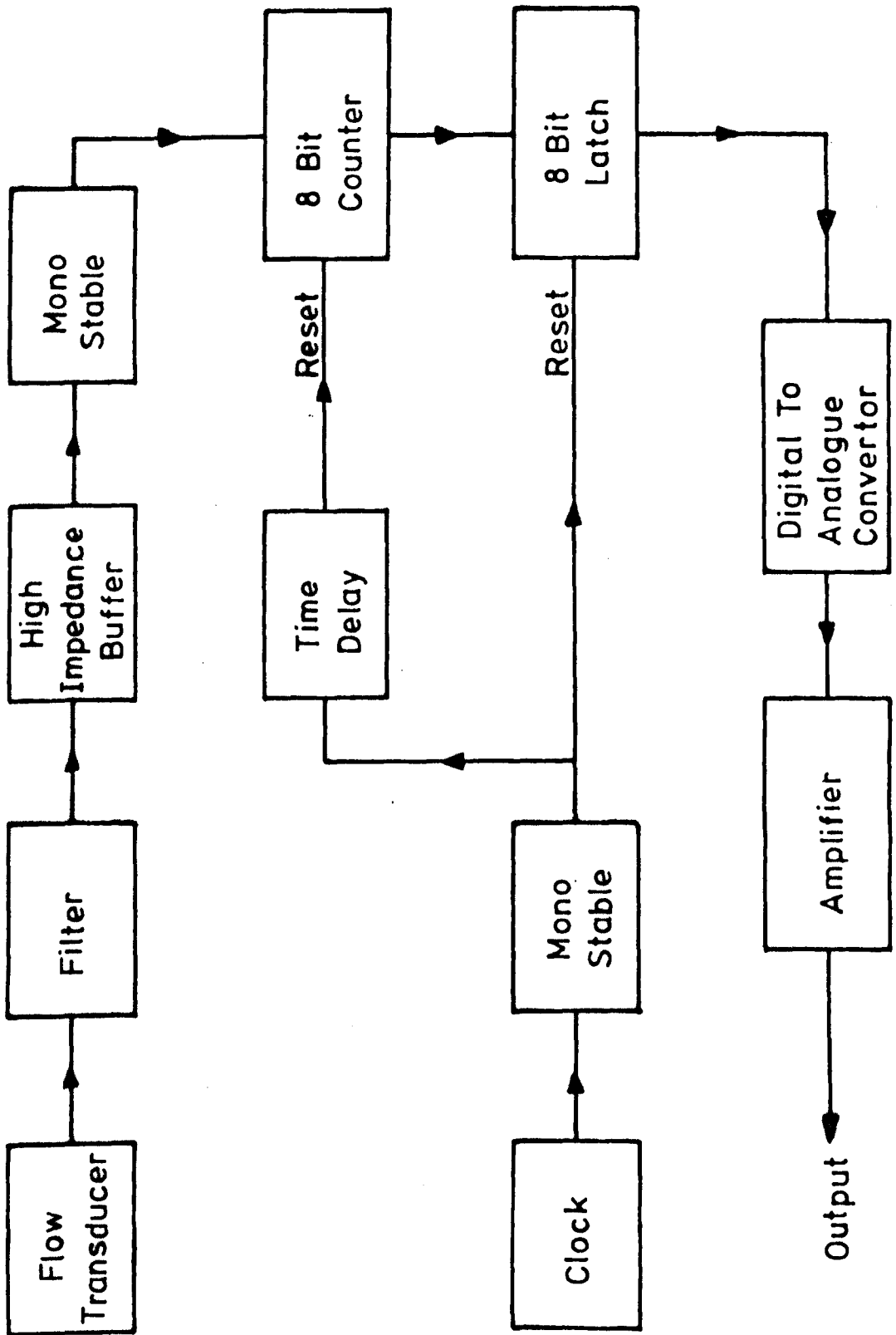
Acknowledgements

Permission to publish this paper has been given by:

Litre Meter Ltd.,
Ryefield Crescent,
Northwood,
Middlesex.
HA5 1NN

References

- 1) Radio Spares Data Sheet R/2602
- 2) " " " " R/2113
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Texas Instruments Ltd.,
Manton Lane,
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- 4) Radio Spares Data Sheet R/2911.



SCHEMATIC DIAGRAM OF FUEL FLOW MEASUREMENT SYSTEM READOUT UNIT

**FUEL FLOW READOUT UNIT
CIRCUIT DIAGRAM**

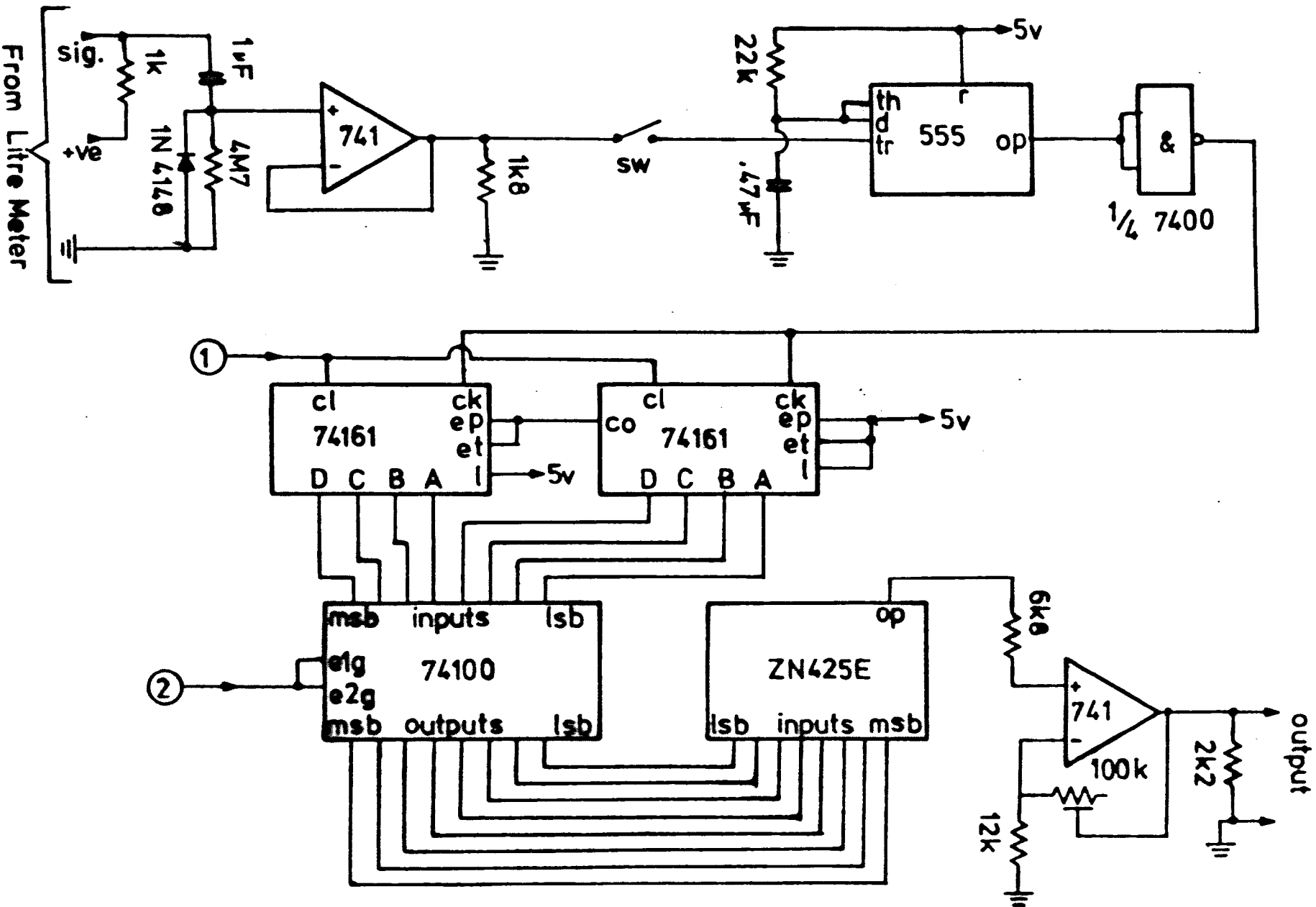
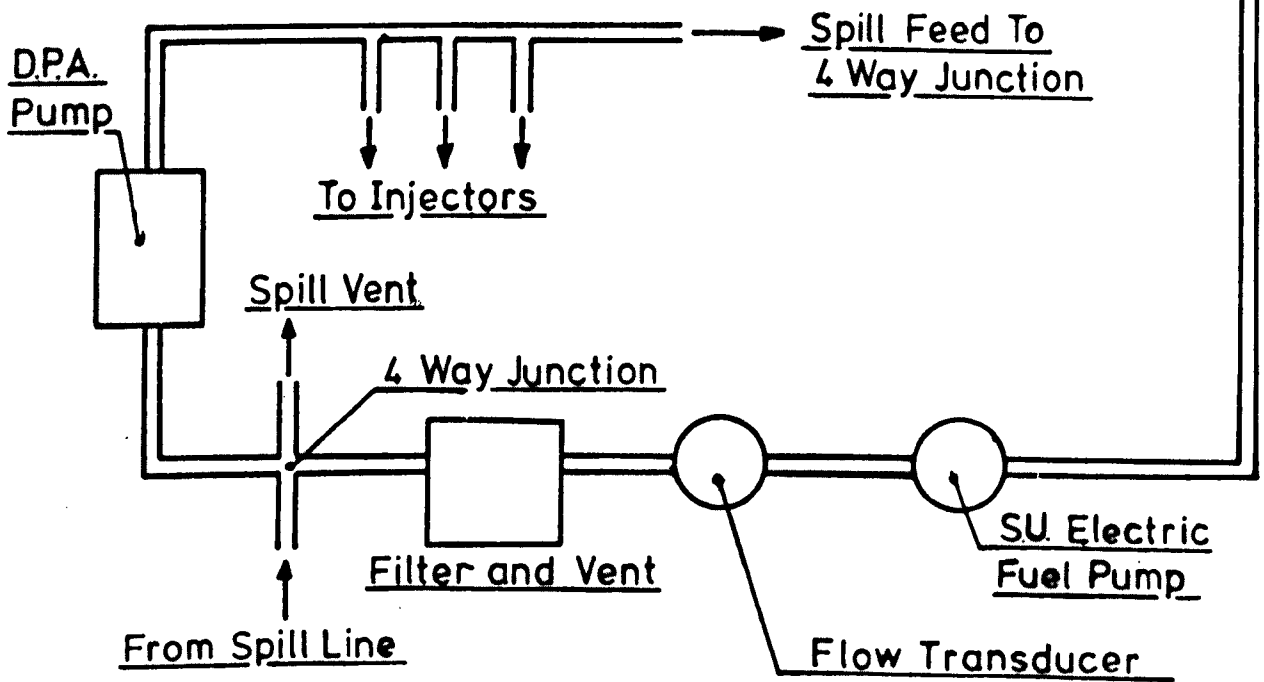
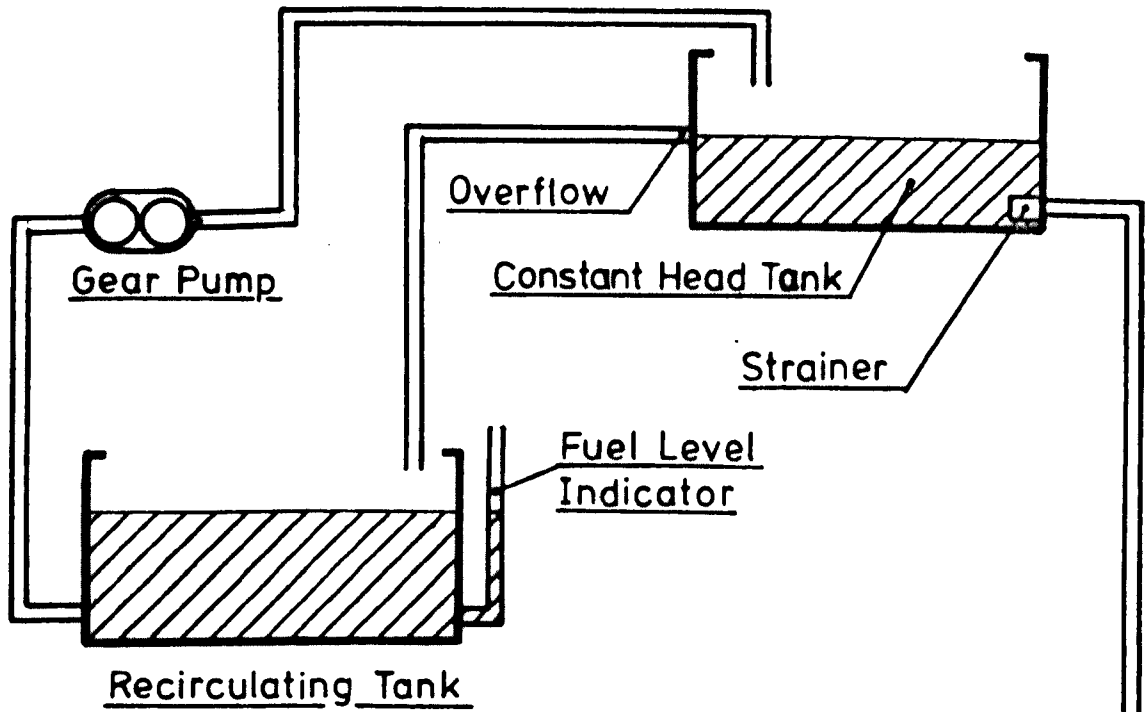


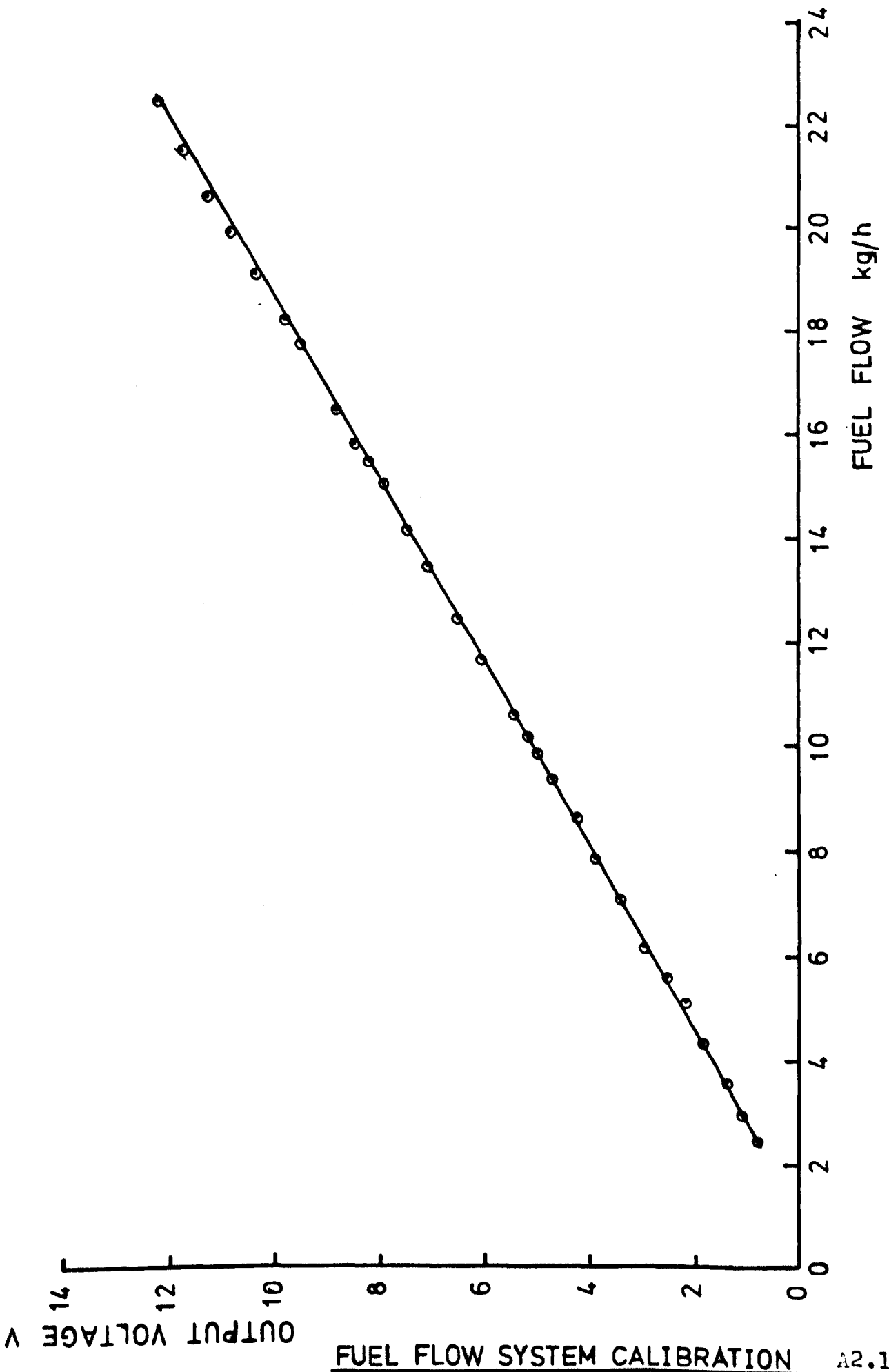
FIGURE 2a

FIGURE 3



ENGINE FUEL OIL SYSTEM

FIGURE 4



APPENDIX 3

VOLTAGE LOAD READOUT FOR AN HYDRAULIC DYNAMOMETER

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Voltage Load Readout for an Hydraulic Dynamometer

Introduction

Modern engine test requirements make on line computerised data acquisition a necessity. This requirement must also apply to the dynamometer used to load the engine. Many dynamometers are now made with this criterion in mind and electrical dynamometers, particularly cooled Eddy Current Dynamometers are easily adapted to give analogue voltage outputs.

However the commonest form of dynamometer still in use is the standard hydraulic dynamometer where the output torque is measured by a spring balance in conjunction with weights applied to one side of the swinging stator assembly. This report describes a successful attempt to measure the output torque by means of a strain gauged cantilever, giving a linear analogue voltage output.

The method has proved simple, reliable and cheap and may well be beneficial in other applications.

General Principles

It was decided to use a strain gauged cantilever system to convert the dynamometer to give an analogue voltage readout of applied load. This would be connected to the dynamometer lever arm and bolted to the base plate.

In the associated work on turbocharged diesel engines (1) a Redman Heenan Froude DP X 4 hydraulic dynamometer is used. A maximum applied load of 1.5 kN (337 lbf) will be reached. For this reason the strain gauged cantilever was designed to have an output of 500 microstrain at an applied load of 1.5 kN (see Appendix).

The cantilever was manufactured from nominal 12.7 mm x 76.2 mm ($\frac{1}{2}$ in x 3 in) mild steel stock. In practice it was found necessary to grind the surfaces of the cantilever to remove blemishes and provide a uniform surface on which to mount the strain gauges. The finished dimensions of the cantilever were 11.938 mm x 73.355 mm (.470 in x 2.888 in).

Strain gauges were then mounted on the cantilever; two on the compression face and two on the tension face. The gauges were placed 13 mm ($\frac{1}{2}$ in) from the 'built in' end of the cantilever. All four strain gauges were selected from the same batch to ensure that they were matched and connected in a 'full bridge' configuration to minimise thermal drift. The cantilever was cured at a temperature of 75°C for 4 hours to minimise adhesive creep effects in service.

The gauges were painted with polyurethane varnish and then sealed with a silicone-rubber compound to exclude environmental moisture. The whole cantilever system (fig.1.) was then painted with a plastic based paint to prevent corrosion.

The cantilever was bolted to the dynamometer base plate, the 'free end' being connected to the dynamometer lever arm by a tie incorporating two Rose Joints (fig.2.). The Rose Joints give some degree of flexibility which ensures that small misalignments do not impose bending moments in the cantilever.

Discussion

The cantilever was calibrated up to full load (fig.3.) using a Dartec M1501 Tensile Test Machine with an X - Y Plotter and a Sangamo Weston C56 Transducer Amplifier. It is, of course, possible to use simpler and cheaper amplification systems depending upon the accuracy required and the amplifier drift that can be tolerated. The cantilever proved linear throughout the full range of applied load. The amplifier output was set at 10v for full load conditions prior to calibration to be compatible with the acquisition system in use.

The cantilever system was attached to the dynamometer and 'worked'. The actuating link is adjustable to permit balancing of the dynamometer lever arm. It is desirable to 'work' the cantilever and then recalibrate as the initial calibration will alter, stabilising after a few hours use. It was possible to calibrate the cantilever up to 1/3 full load 'in situ' on the dynamometer (fig.4.). This was effected by connecting the cantilever actuating link to the dynamometer lever arm and the spring of the spring balance. The cantilever was then loaded using the spring balance.

A discrepancy of 3% was found due to the variation in cantilever alignment in the two calibration situations. The latter calibration method allows amplifier adjustments to be made such that the final calibration covers the output range required and is known to be accurate.

The dashpot on the dynamometer becomes redundant, as the cantilever system is very rigid, and may be removed (prior to calibration). The necessity of adding weights and rebalancing the dynamometer has been eliminated; the lever arm only requiring balancing when the cantilever system is first fitted. This is effected by adjusting the cantilever actuating link.

Output voltages may be read to 0.01 v (0.1% of full scale) and the variation found when using the dynamometer with a diesel engine gives an accuracy of $\pm 0.2\%$ of full scale. No thermal drift problems, and no problems associated with strain gauge adhesive creep, have been experienced. The repeatability of the system is better than 0.2%.

Conclusion

The cantilever system described converts an hydraulic dynamometer to an analogue voltage output which is directly proportional to applied load. The output signal is suitable for computerised data acquisition.

The converted dynamometer is much simpler to use than the standard system and leads to a large reduction in the time taken to perform engine tests.

The calibration is linear and the system has an accuracy of $\pm 0.2\%$.

The cantilever system can be built for a small cost and may be used with a wide range of transducer amplifiers.

References

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Acknowledgements

Bryan and Sullivan Ltd.,
504, Aylestone Road,
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APPENDIX

A design point of 500 microstrain at maximum applied load was chosen as mild steel yields at approximately 1000 microstrain. This allows us to work in the linear range of the material.

Reasonably good amplifier systems have a resolution of better than 1 microstrain thus giving a potential resolution better than 0.2% of full scale.

From Beam Theory:

$$\frac{M}{I} = \frac{\sigma}{y} = \frac{E}{R}$$

where,

M - bending moment
 σ - stress
E - Young's modular
R - Radius of curvature
y - depth to neutral axis
I - second moment of area

For a rectangular cross section:

$$I = \frac{bd^3}{12}$$

where,

b - breadth of section
d - depth of section

Also,

$$E = \frac{\sigma}{\epsilon}$$

$$\sigma = E\epsilon$$

where,

ϵ - strain

For mild steel:

$$E = 207 \text{ GN/m}^2 (30 \times 10^6 \text{ lbf/in}^2)$$

For a section 12.7 mm x 76.2 mm ($\frac{1}{2}$ in x 3 in) with an effective cantilever length of 144.5 mm (5 $\frac{11}{16}$ in) and a design load of 1.5 kN(337 lbf).

$$\frac{M}{I} = \frac{\sigma}{y} \quad \text{and} \quad \sigma = \epsilon E$$

$$\epsilon = \frac{MY}{IE}$$

$$M = 1.5 \times 10^3 \times 144.5 \times 10^{-3} = 216.75 \text{ Nm}(1918 \text{ lbf in})$$

$$y = \frac{d}{2} = 6.35 \times 10^{-3} \text{ m (0.25 in)}$$

$$I = \frac{bd^3}{12} = \frac{76.2 \times (12.7)^3}{12} = 13007.231 \times 10^{-12} \text{ m}^4 \text{ (0.03125 in}^4\text{)}$$

$$\epsilon = \frac{216.75 \times 6.35 \times 10^{-3}}{13007.231 \times 10^{-12} \times 207 \times 10^9} = 511 \mu\epsilon$$

Thus a mild steel section of 12.7 mm x 76.2 mm ($\frac{1}{2}$ in x 3 in) will produce an acceptable cantilever.

After grinding to remove surface defects the cantilever section was found to be 11.938 mm x 73.355 mm (0.470 in x 2.888 in).

Using these values:

$$I = \frac{73.355 \times (11.938)^3}{12} = 10400.235 \times 10^{-12} \text{ m}^4 \text{ (0.025 in}^4\text{)}$$

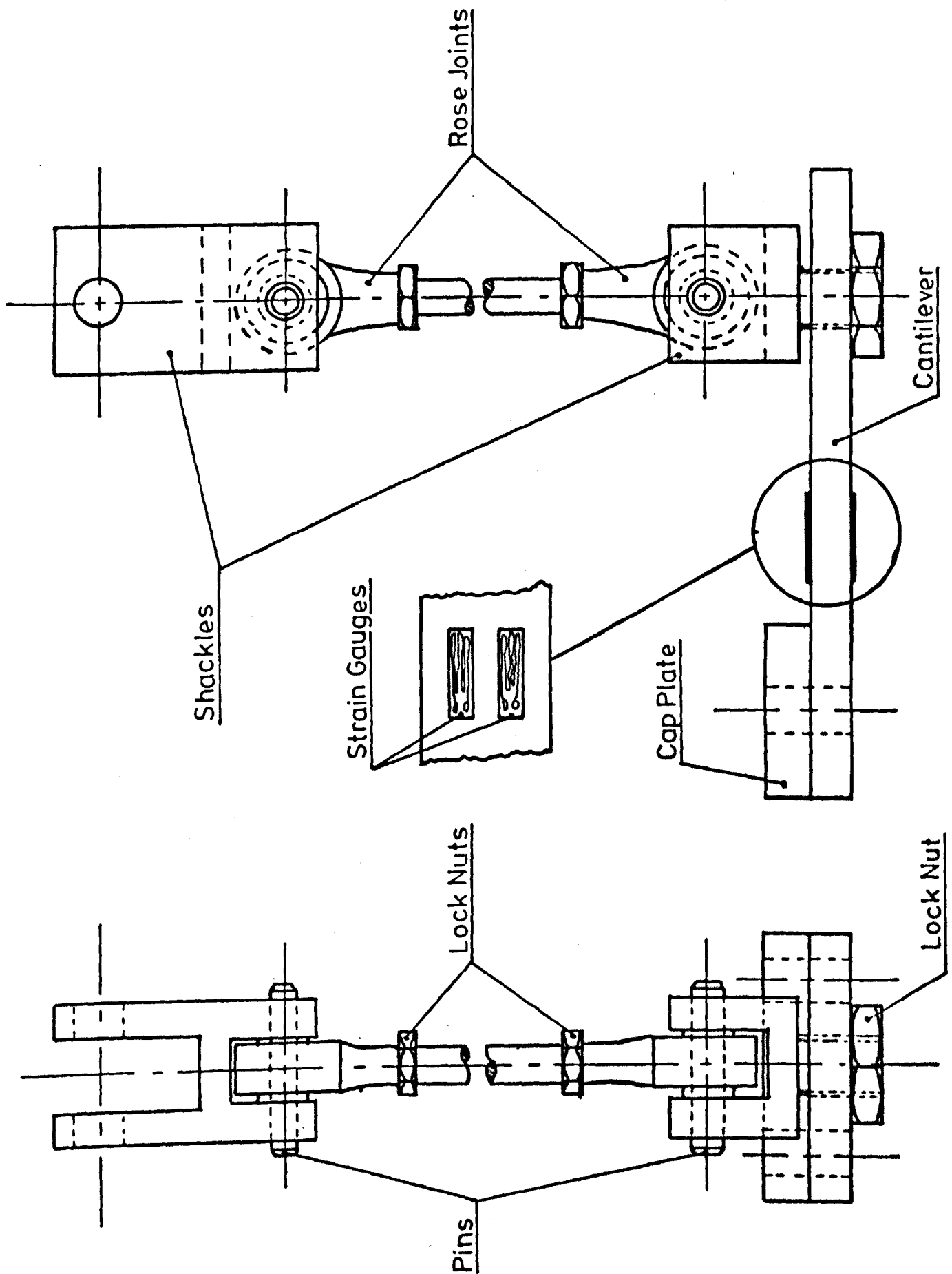
$$y = 5.969 \text{ mm (0.235 in)}$$

Thus,

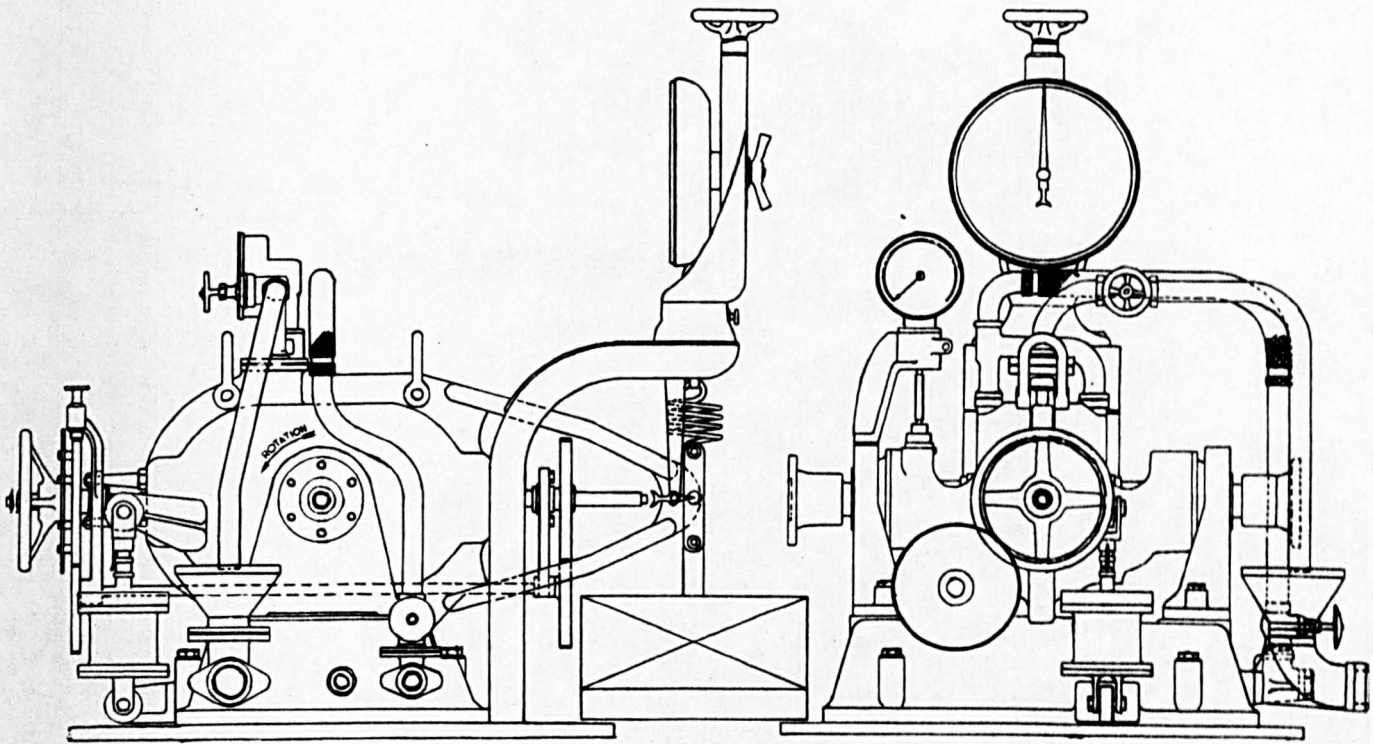
$$\epsilon = \frac{216.75 \times 5.969 \times 10^{-3}}{10400.235 \times 10^{-12} \times 207 \times 10^9} = 601 \mu\epsilon$$

This gives a more sensitive cantilever with better resolution and is still safely within the linear working range of mild steel.

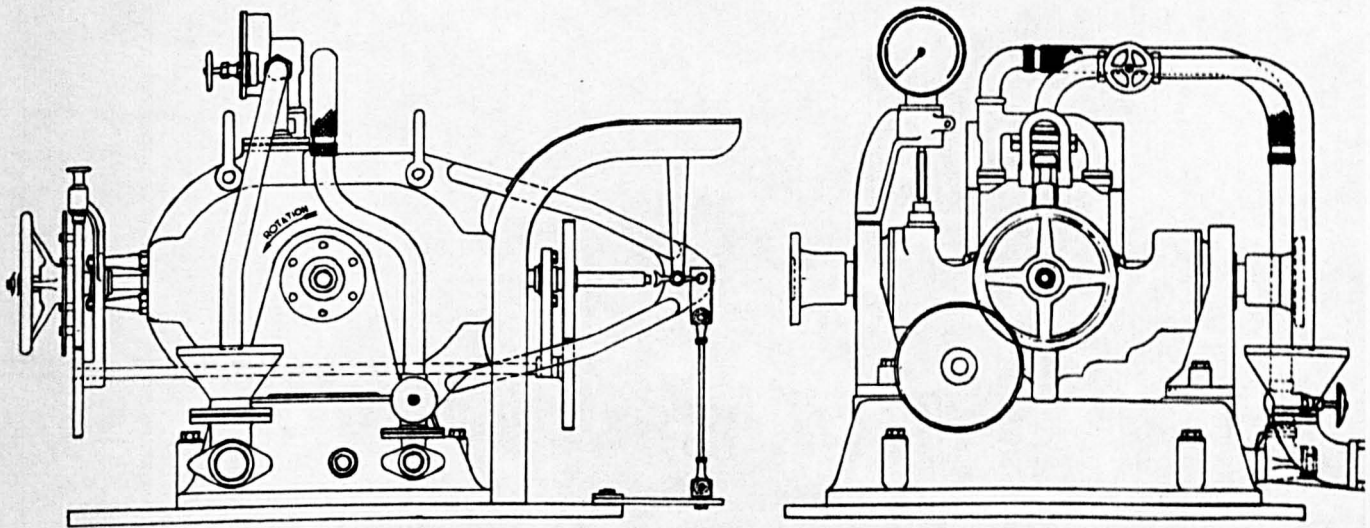
FIGURE 1



STRAIN GAUGED CANTILEVER SYSTEM



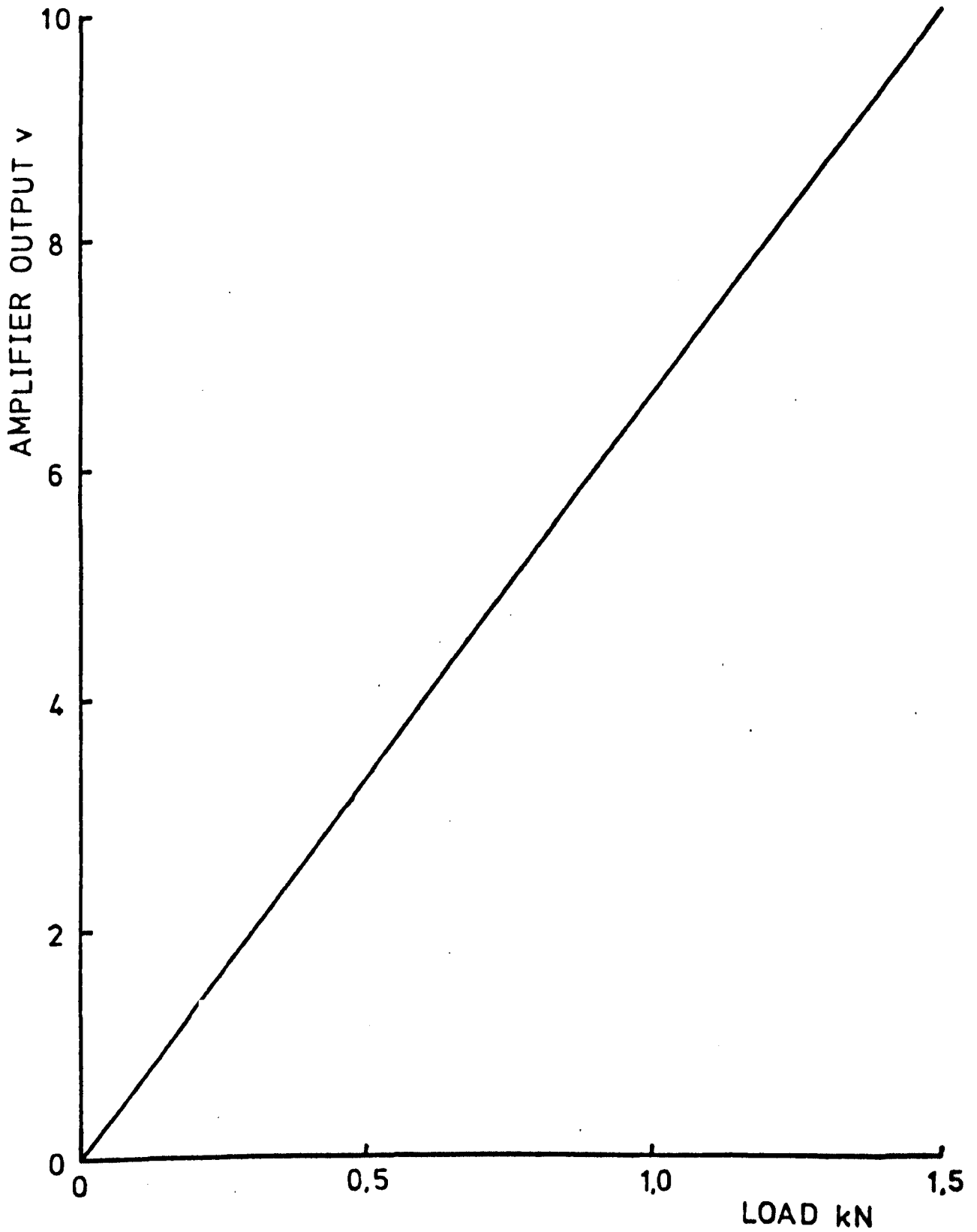
STANDARD DYNAMOMETER



ADAPTED DYNAMOMETER

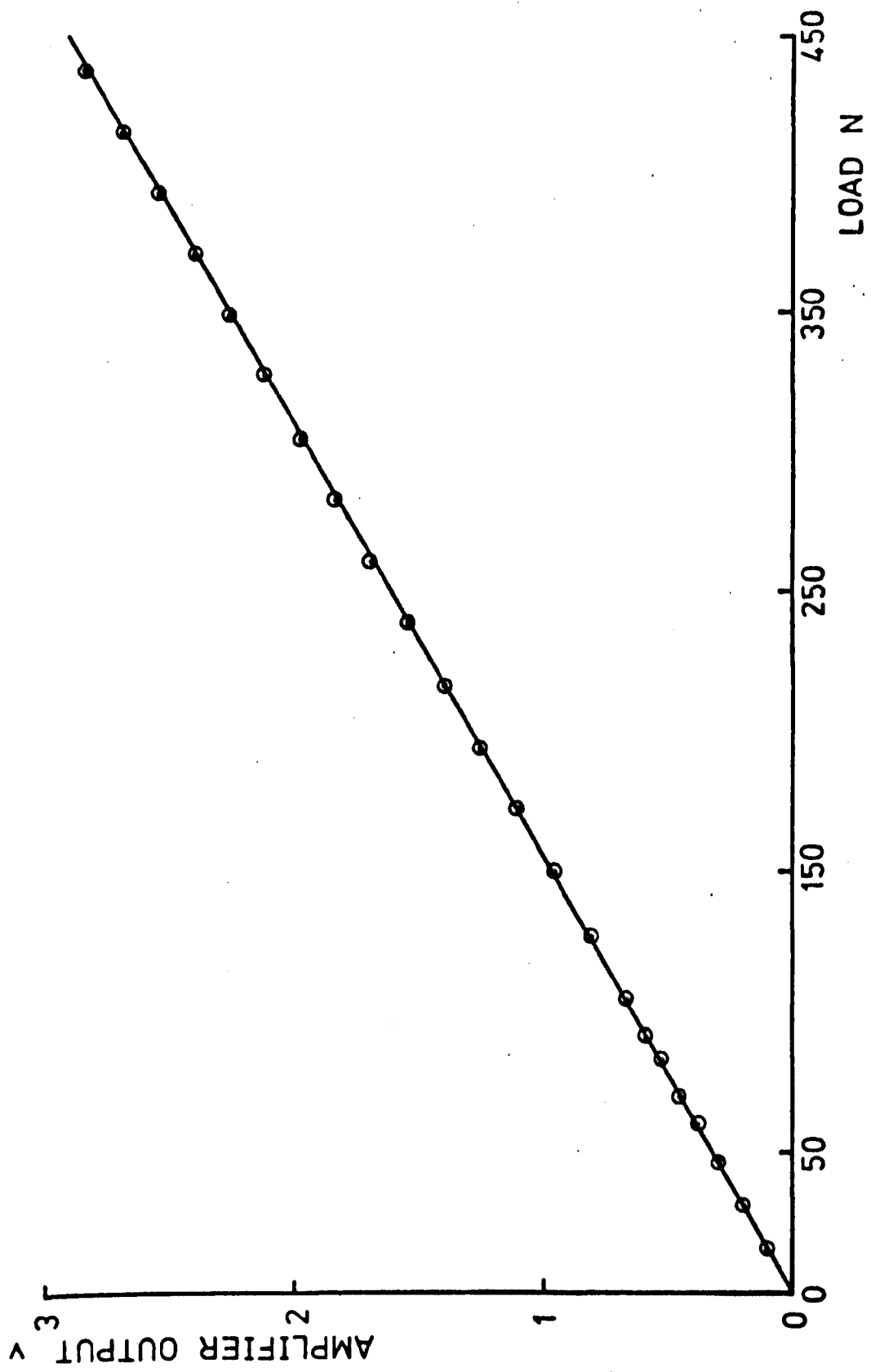
STANDARD AND ADAPTED
DPX 4 HYDRAULIC DYNAMOMETER

FIGURE 3



STRAIN GAUGED CANTILEVER CALIBRATION
[Tensile Test Machine and X-Y Plotter]

FIGURE 4



'IN SITU' STRAIN GAUGED CANTILEVER CALIBRATION
[Loaded by Spring Balance]

APPENDIX 4

THE MEASUREMENT OF LIQUID
PETROLEUM GAS FLOW RATE FOR
I.C. ENGINE APPLICATION

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(I.J.M.E.E.)

THE MEASUREMENT OF L.P. GAS FLOW RATE FOR

I.C. ENGINE APPLICATION.

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Abstract

This paper examines a variety of methods for evaluating the flow rates of liquefied petroleum gas. The two most promising methods appeared to be volumetric measurement by means of a vane driven motor/voltage generator system and direct gravimetric measurement. The paper shows that there are serious limitations to the former method and that the gravimetric method gives reliable results and can be made more universal by weighing with a strain gauged proving ring which enables data processing to be applied directly.

Introduction

During a research project on a Diesel engine associated with a gas turbine a need arose for the measurement of propane flow rate to a turbine combustion chamber. A data acquisition system was used on the test bed requiring test data to be presented in the form of analogue voltages in the range 0-10v.

Generally gas flow rates may be evaluated by either volumetric or gravimetric techniques.

Volumetric Methods

For satisfactory volumetric measurement the gas properties must remain constant. This criterion would be met in a temperature controlled process (heater or vaporiser), or for measurement of mains gas flow, in which pressure and temperature could be considered to remain constant. Volumetric methods are not suited to two-phase flow.

Suitable devices for volumetric measurement of flow rates are:-

- a) Rotameter. This does not provide a voltage output and is thus unsuitable for applications using data acquisition systems.
- b) Turbine or Rotor devices.

The rotor type flowmeter used for these tests provides a frequency output proportional to flow rate which is readily convertible to analogue voltages.

Gravimetric Methods

Gravimetric measurements of flow rates involve the weighing of the fluid at known intervals of time. This may be achieved by measuring the time taken for a given weight loss or by the weight loss in a given time. Such methods are suitable for gas flows (or two-phase flows) in an enclosed system.

A programme of work was carried out to assess the use of a rotor type flowmeter for gas flow measurement and to develop a suitable system for measuring propane flow rate from a bottled supply.

Tests with a Volumetric, Rotor Type, Flowmeter.

The device used proved unsuitable for use with bottled propane due to the fluctuations in delivery temperature and pressure. The occasional passage of liquid droplets also poses a problem. No results of this test are presented as they displayed wide scatter with no repeatability i.e. no correlation of frequency output with flow rate.

Investigations were carried out to assess conditions under which rotor type flowmeters would successfully measure gas flow. The device was tested on a mains natural gas supply to a single cylinder spark ignition engine. The mains supply may be considered as a large single phase reservoir at constant pressure and temperature. Calibration of the rotor type flowmeter was referenced to a Standard Rotameter (G.E.C.Elliott, Rotameter series 2000, 14A Tube, Float Type A). Two tests were carried out within 10 mins of each other. The results are given in Figure 1.

A further set of tests were carried out using air as the medium. Air was taken from a large receiver at a constant temperature and pressure, and was fed to the single cylinder engine in place of the mains gas supply previously mentioned, the engine obviously not running. The carburettor acted as a nozzle exhausting the air to the atmosphere. The calibration (Figure 2) proved linear and repeatable.

Gaseous L.P.G Flow measurement

As previously stated the fluctuations of temperature and pressure (and the occasional passage of liquid) render volumetric methods unsuitable, leaving the option of a gravimetric system. Bottled gas supplies present problems as the gas supplied has a net weight which is usually of the same order as the cylinder. For the propane in use at this establishment this results in a gross weight of approximately 42kg (19kg net propane).

For reasonable resolution it is desirable to be able to detect a weight loss of the order of 1g. It is not usually possible to detect such changes on a balance having a maximum range of, say, 50kg. This normally leaves the option of taking measurements over a protracted period of time, which can be expensive (1). This method does not yield a direct reading.

A system was adopted using a load cell (a strain gauged proving ring) with a suitable amplifier, giving analogue voltages in the range 0-10v. Details of strain gauge procedure have been previously reported (2). The proving ring was designed for a strain of approximately 700 $\mu\epsilon$ at an applied load of 50kg (Figure 3). Linearity was checked on a tensile test machine and was found to be satisfactory. Calibration was carried out using deadweights and a Sangamo Weston C56 transducer amplifier giving an analogue voltage proportional to applied load. The calibration showed excellent linearity and repeatability (Figure 4).

Voltage output was logged on a Solartron Data Logger controlled by a PDP 11/05 mini computer. Upon first reading the voltage, the internal clock of the computer was set to zero. The voltage was re-measured and stored at one minute intervals (timed to 0.01 secs).

Calculations were then carried out to yield a gas flow in kg/h, taken as an average of the one minute values and as a value based on the first and last readings.

The sampling period is chosen by the operator, with a maximum of 15 minutes. The system was tested for 'zero drift' using a sampling time of 5 minutes. There was a zero drift of 0.05 kg/h after a 'warm up' period of 30 minutes, falling to 0.02 kg/h after 45 minutes. Experience has shown that transmitted vibrations can cause high readings of gas flow.

The arrangement of the gas supply system is given in Figure 5. A listing of the computer program, written in BASAC IV, to log gas flow rate and nine temperatures (from thermocouples) is shown in the appendix. A typical example of program output is also given.

Discussion

The tests with a volumetric, rotor type, flowmeter have shown that it is not suitable for measuring flow rates of gases with varying temperatures and pressures, and possible two-phase flow (bottled gas).

Tests on a single cylinder spark ignition engine using natural gas have also shown it to be unreliable. Although the supply conditions are at a constant temperature and pressure, the outlet pressure fluctuates due to the intermittent charging of the four stroke cycle engine. These fluctuations would be reduced on a multi-cylinder engine, although it is still unlikely that the rotor type flowmeter would prove suitable. Such pressure fluctuations have not been found to have an adverse effect when using such a device for liquid flow measurement (3), although a constant head supply system was found to be necessary.

It is proposed that the inaccuracy is due to the compressibility of the gas causing rotor instability.

The tests using air as a medium, with constant pressure and temperature inlet, and the outlet exhausted to atmosphere through a carburettor, gave good results with flow rates in excess of 8 litres/min. Non-linearity below this level is due to rotor friction. This is normal on this type of transducer, as a certain level of flow is required to overcome rotor friction. Linearity could be extended below 8 litres/min by the use of a smaller transducer. A rotor type flowmeter is thus suitable for measuring gas flow rates in applications having constant pressure conditions at both the inlet and outlet of the transducer.

A suitable method for measuring bottled gas flow rates is provided by a weighing system, based on a load cell giving an analogue output, coupled to a data acquisition system with accurate interval timing. The method yields good results with minimal error and good repeatability. Accuracy is governed by instrumentation 'zero drift', which may be minimised by allowing adequate 'warm up' times. The system should be well isolated from vibration as this results in reading errors. In certain situations (high flow rates) gas bottle 'icing' may occur. This will adversely affect readings, and may be overcome by partially immersing the bottle in a tank of water to act as a heat source. Providing that the water level is carefully chosen and maintained constant, a negligible error is introduced. The measurement system described is suggested for flow rates in excess of 1 kg/h. The system has demonstrated an accuracy of better than 0.1% for a flow of 10kg/h.

Conclusion

A volumetric rotor type flowmeter may be used for the measurement of gas flows providing that the supply is of a constant temperature and pressure, and the gas flow is exhausted into a constant pressure. With these constraints the rotor type flowmeter provides a reliable reading with good accuracy and repeatability.

Propane flow rates from a bottled gas supply are best measured using a gravimetric system. The system described yields a good accuracy (\pm 0.05 kg/h) and repeatability, giving a direct output.

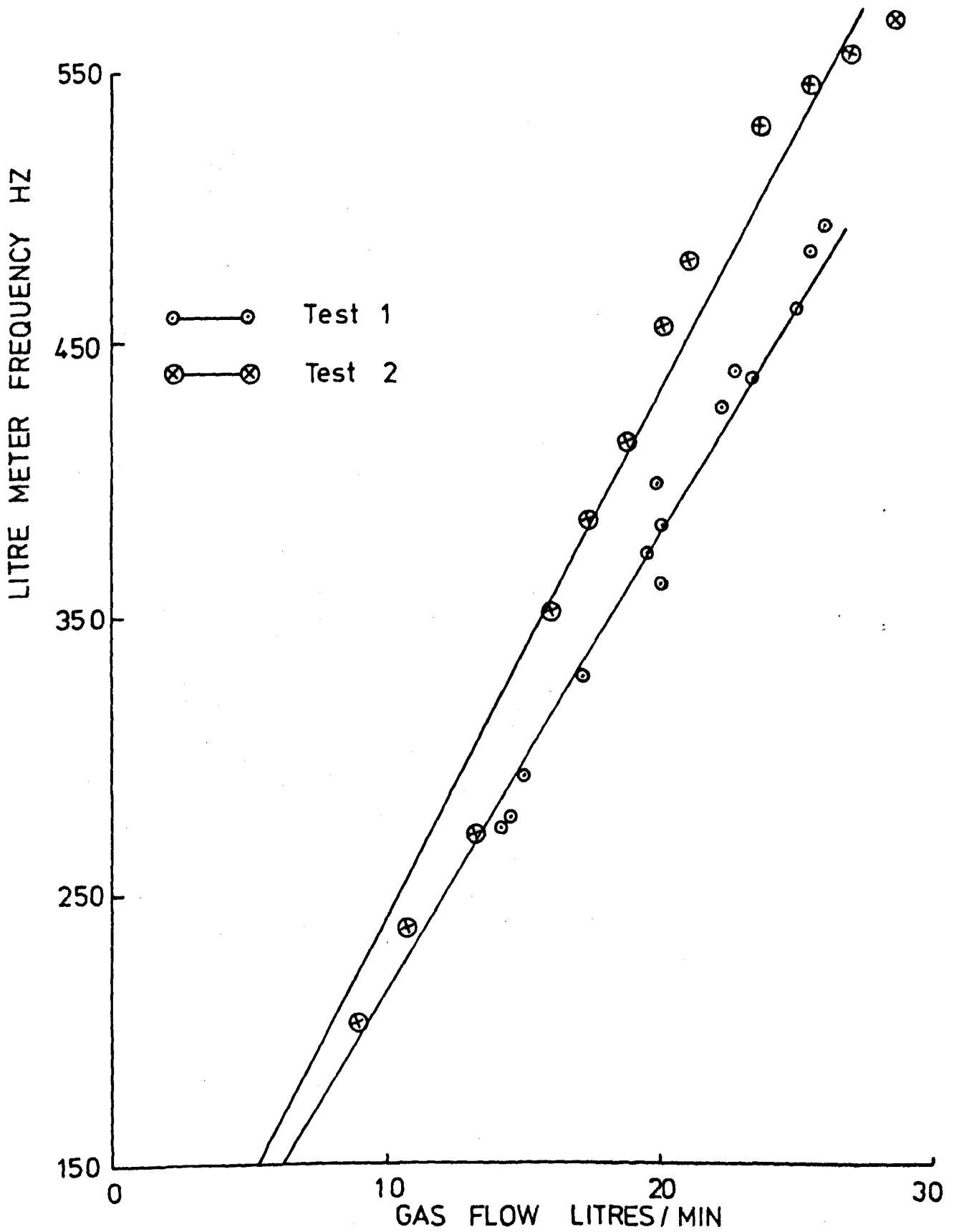
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- 2) R. Parkin, P.C. Few, D.J. Picken.
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I.J.M.E.E. Vol. 7, No. 1, Feb 1979.
- 3) R. Parkin, P.C. Few, D.J. Picken.
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I.J.M.E.E. Vol. 6, No. 3, July 1978.

Acknowledgements

The volumetric, rotor type, flowmeter (LH 220 GN) used for this work was supplied by Litre Meter Ltd., Unit 9, Park Street Industrial Estate, Aylesbury, Bucks, HP20 IET, who also gave permission to publish this paper.

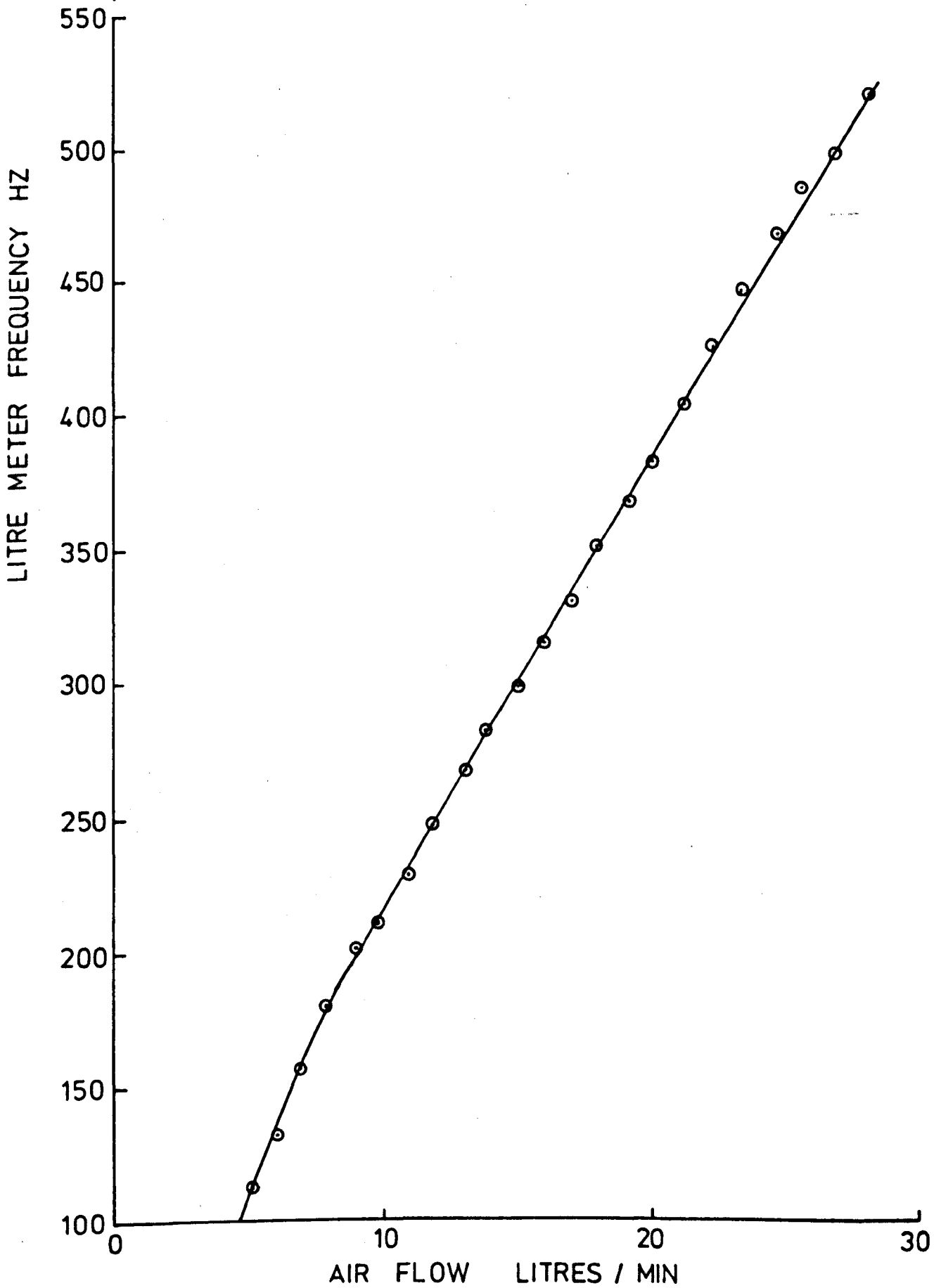
FIGURE 1



LITRE METER WITH MAINS NATURAL GAS ON A SINGLE
CYLINDER S.I. ENGINE

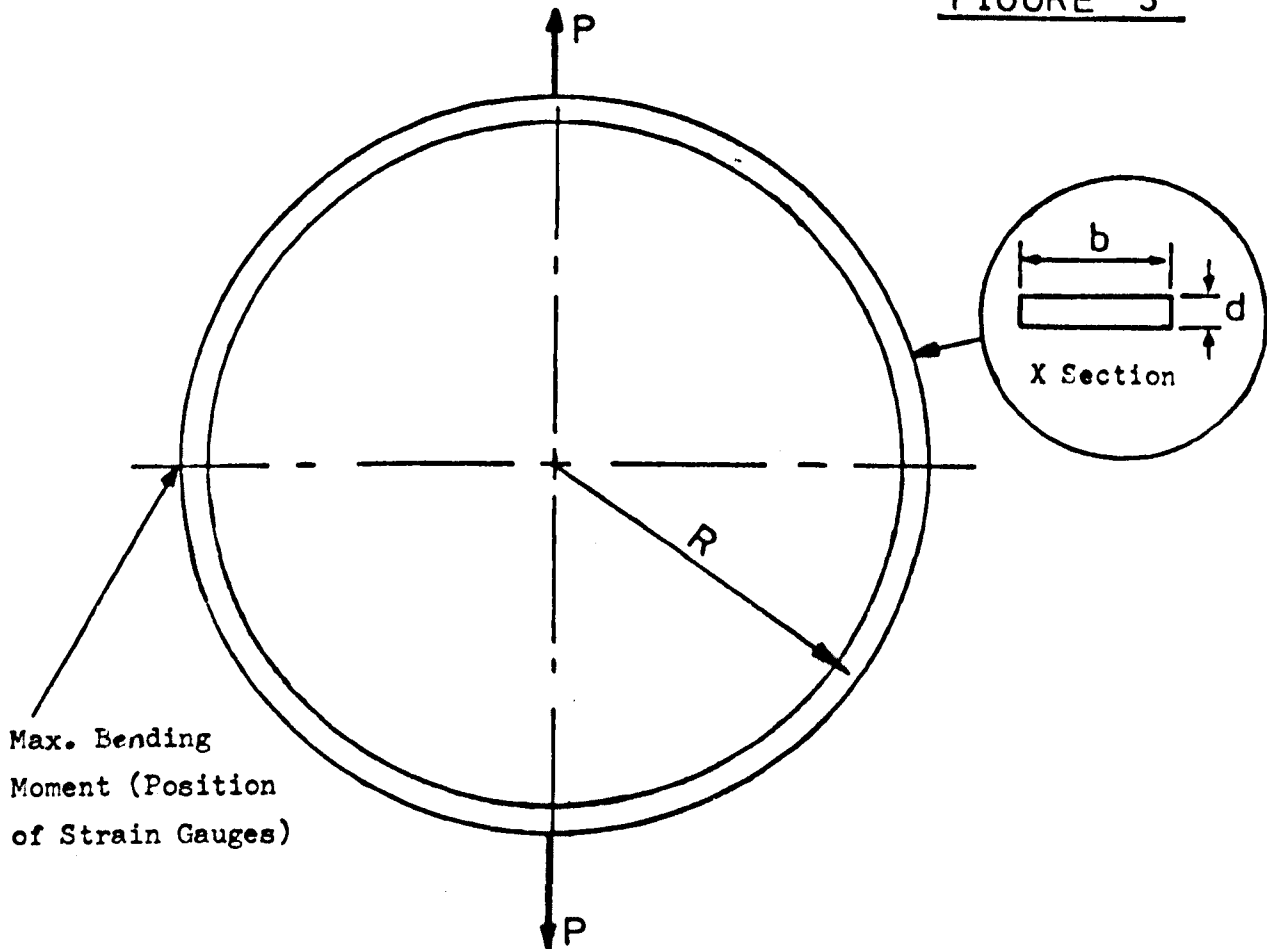
(Two tests under identical conditions)

FIGURE 2



CALIBRATION OF LITRE METER USING AIR
(constant T & P inlet, exhaust to atmosphere)

FIGURE 3



$$\text{Maximum Bending Moment} = \frac{PR}{\pi}$$

$$\therefore \sigma = \frac{PR}{\pi} \cdot \frac{d}{2} \cdot \frac{12}{bd^3} = \frac{6PR}{\pi bd^2}$$

$$\epsilon = \frac{\sigma}{E} \quad \therefore \epsilon = \frac{6PR}{\pi bd^2 E}$$

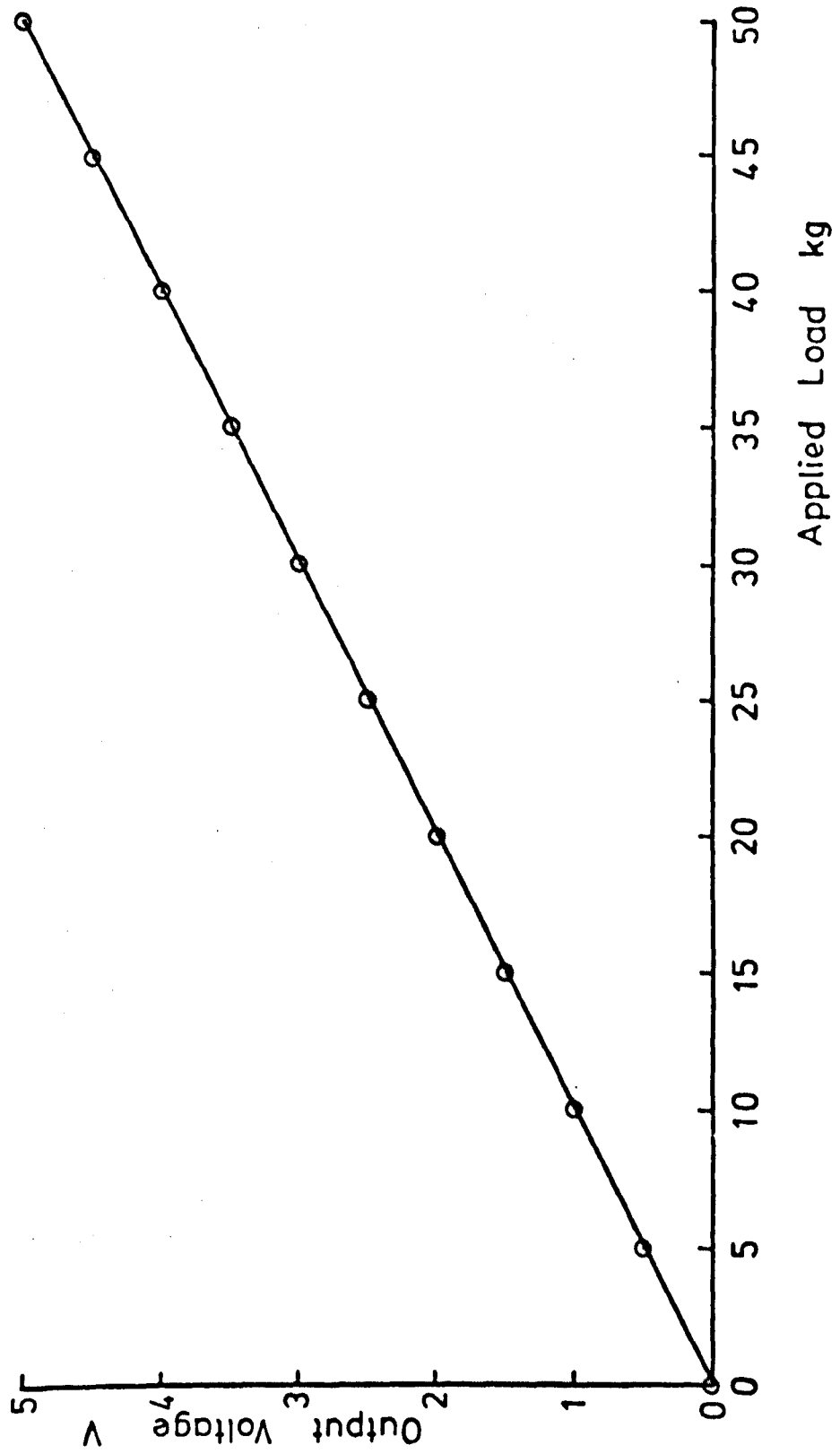
With :- $P = 50 \text{ kg (490.5 N)}$ $R = 50 \text{ mm}$ $b = 25 \text{ mm}$
 $d = 3.5 \text{ mm}$ $E = 207 \text{ GN/m}^2$

$$= \frac{6 \times 490.5 \times 50 \times 10^{-3}}{\pi \times 25 \times 10^{-3} \times 3.5^2 \times 10^{-6} \times 207 \times 10^9}$$

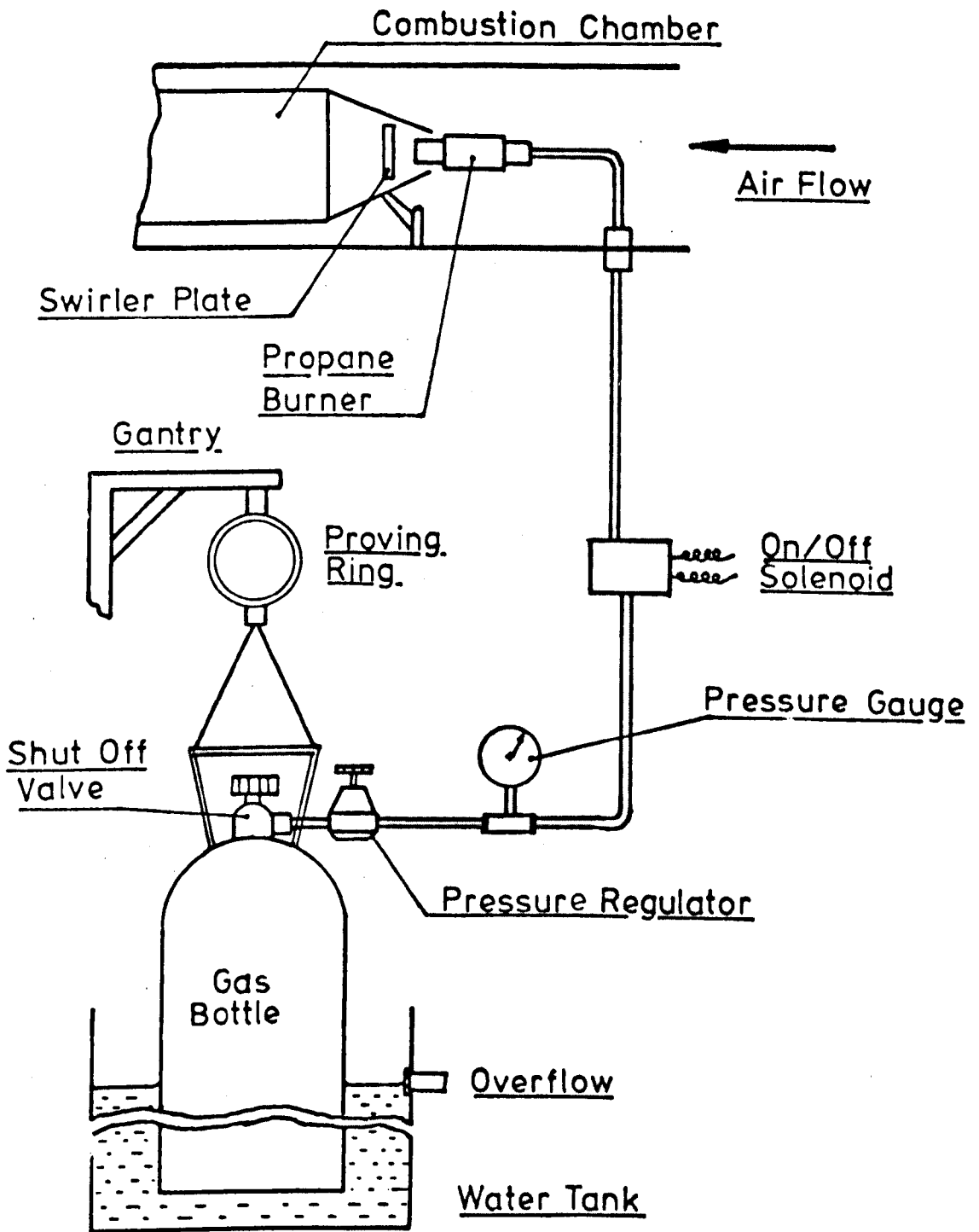
$$\epsilon = 739 \mu\epsilon$$

DETAILS OF PROVING RING

FIGURE 4



CALIBRATION OF PROVING RING



COMBUSTION CHAMBER FUEL SYSTEM

APPENDIX

```
5 REM PROGRAM VARILOG(RP)
10 REM LOGGING PROGRAM FOR GAS FLOW AND NINE TEMPERATURES
20 DIMG(15),T(8,15),T1(8),P(15),Q(15),R(15)
25 PRINT " SAMPLE TIME IN MINS (MAX=15) ?" \ INPUT H
26 IF H>15 GO TO 28
27 GO TO 30
28 PRINT " NOT VALID, RETYPE" \ GO TO 25
30 PRINT " INPUT DAY,MONTH,YEAR" \ INPUT D,M,K
40 PRINT " INPUT AMBIENT TEMP" \ INPUT T8
45 PRINT " PAPER TAPE DATA OPTION? (1=YES,0=NO)"
46 INPUT F
50 PRINT " INPUT DATA POINT 0 TO STOP; 1,2,3 ETC FOR RUN"
51 IF F=0 GO TO 55 \ PRINT #2:H,CHR$(13,10),D,CHR$(13,10)
52 PRINT #2: M,CHR$(13,10),K,CHR$(13,10),T8,CHR$(13,10)
55 PRINT " DATA POINT?"
60 INPUT C
62 STIME(0,0,0)
65 IF C=0 GO TO 290
70 FOR I=0 TO H
80 SETDVM(2,3)
85 MEASURE(G(I),0)
86 RTIME(P(I),Q(I),R(I))
90 SETDVM(4,3)
92 LET V=0
95 SCAN(T(0,1),1,9, LINE 100 )
96 IF V=0 GO TO 96
99 GO TO 105
100 LET V=1 \ RETURN
105 IF I=H GO TO 120
107 LET U=0
110 DELAY(59, LINE 115 )
111 IF U=0 GO TO 111
112 GO TO 120
115 LET U=1 \ RETURN
120 NEXT I
```

```

140 PRINT D;"/";M;"/";K;" DATA POINT";C
141 IF F=0 GO TO 145 \ PRINT #2:C,CHR$(13,10)
145 PRINT "*****" \ PRINT
150 LET Z=0
155 FOR I=1 TO H
160 LET Y=(G(I-1)-G(I))
161 LET Y1=(Q(I)+(R(I)/60))-(Q(I-1)+(R(I-1)/60))
162 LET Z=Z+(Y/Y1)
17 \ NEXT I
180 LET W1=(Z/H)*10*60
190 PRINT " GAS FLOW 1 MIN AVG=";W1;"KG/HR"
191 IF F=0 GO TO 200 \ PRINT #2:W1,CHR$(13,10)
200 LET W2=G(0)-G(H)
201 LET W3=(Q(H)+(R(H)/60))-(Q(0)+(R(0)/60))
202 LET W2=(W2/W3)*10*60
210 PRINT " GAS FLOW FULL TIME=";W2;"KG/HR" \ PRINT
211 IF F=0 GO TO 220 \ PRINT #2:W2,CHR$(13,10)
220 FOR J=0 TO 8 \ LET T1(J)=0
230 FOR I=0 TO H \ LET T1(J)=T1(J)+T(J,I) \ NEXT I
240 LET T1(J)=(((T1(J)))/(H+1)*4.10000E-03))*100)+T8
250 PRINT " T";J+1;"=";T1(J);"DEG C"
255 IF F=0 GO TO 256 \ PRINT #2:T1(J),CHR$(13,10)
256 NEXT J
260 PRINT \ PRINT \ PRINT \ PRINT \ PRINT
280 GO TO 55
290 IF F=0 GO TO 295 \ PRINT #2:C,CHR$(13,10)
295 STOP
300 END

```

DATA POINT?

16

28 / 2 / 79 DATA POINT 16

GAS FLOW 1 MIN AVG= 5.87023 KG/HR

GAS FLOW FULL TIME= 5.87023 KG/HR

T 1 = 52.6504 DEG C

T 2 = 53.0894 DEG C

T 3 = 510.211 DEG C

T 4 = 468.463 DEG C

T 5 = 51.2114 DEG C

T 6 = 387.959 DEG C

T 7 = 671.642 DEG C

T 8 = 551.236 DEG C

T 9 = 518.919 DEG C

DATA POINT?

17

28 / 2 / 79 DATA POINT 17

GAS FLOW 1 MIN AVG= 4.90801 KG/HR

GAS FLOW FULL TIME= 4.90776 KG/HR

T 1 = 47.7724 DEG C

T 2 = 47.8862 DEG C

T 3 = 535.285 DEG C

T 4 = 490.561 DEG C

T 5 = 47.3089 DEG C

T 6 = 385.333 DEG C

T 7 = 672.618 DEG C

T 8 = 550.311 DEG C

T 9 = 509.024 DEG C

TYPICAL EXAMPLE OF PRINTOUT FROM PDP 11/05 LOGGING PROGRAM

APPENDIX 5

TURBOCHARGING THE AUTOMOTIVE
TWO STROKE DIESEL ENGINE

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Submitted for publication to S.A.E. July 1978

Summary

This paper reports the theoretical and experimental investigation of different combinations of automotive, two stroke, diesel engine with scavenge blower, turbocharger and auxiliary combustion chamber.

The various configurations are compared on a basis of trapped air flow, charge density and power output.

It is shown that freely turbocharged operation of a high speed, two stroke, engine is limited by the low overall efficiency of turbochargers in the size range recommended, and methods of alleviating this problem are described.

List of Symbols used in Figures

ACC	Auxiliary Combustion Chamber
A/F	Trapped Air/Fuel Ratio
BMEP	Brake Mean Effective Pressure
C	Turbo-Compressor
EPR	Engine Exhaust Pressure Ratio
Hsg	Turbine Housing Size
IM	Engine Air Inlet Manifold Boost Ratio
SB	Scavenge Blower
T	Turbine
TC	Turbocharger
TEX	Engine Exhaust Temperature
TI	Turbine Inlet Temperature
η	Efficiency
[]	Boost Ratio
()	Scavenge Ratio

Subscripts

C	Compressor
T	Turbine

Introduction

Turbocharging of two stroke automotive diesel engines is less obviously advantageous than for four stroke engines because a scavenge blower is needed for starting and low speed running.

It was proposed in previous work (1) to use a turbocharger to supply air to the engine for the medium to high speed range and to fit an auxiliary combustion chamber to augment the exhaust gas enthalpy, thus increasing turbocharger efficiency, for low speed operation. The necessity for a scavenge blower for low speed engine operation is eliminated and the 'back up' torque of the engine is improved.

The engine used in this work was a horizontally opposed piston, two stroke (Tab 1).

Programme

The power output of the standard engine was established using sophisticated instrumentation which included an analogue system for fuel flow measurement (2) and an hydraulic dynamometer with analogue voltage readout of applied load (3).

A computer program for engine performance prediction was developed from previous work (4,5). Results of the standard engine performance tests are shown in figs 1 and 2 .

The engine system was then developed in 3 stages.

Stage I Development

This system (fig 3) incorporated a turbocharger in series with the original scavenge blower. The auxiliary combustion chamber was included in the system but was not fuelled, thus having no effect on engine performance.

Turbocharger matching was carried out with reference to a turbocharger manufacturer. Matching was constrained to give acceptable freely turbocharged performance (ie no scavenge blower assistance) in the medium to high speed range. A computer program was used to indicate areas in which freely

turbocharged operation would be attainable with given values of compressor and turbine efficiency, air/fuel ratio and boost level (fig 4).

The performance of the engine over the whole speed range was first established with the scavenge blower and turbocharger in series. The engine was then set at about 3/4 full speed and the scavenge blower output line to the turbo-compressor removed. Freely turbocharged operation was then investigated.

It was found that freely turbocharged running was impossible whilst working within engine exhaust temperature constraints set at 600°C, for this engine, by Timoney (6).

Changing the turbine size had no significant effect on this result, confirming that the limitation is a function of turbocharger overall efficiency and exhaust temperature limitations. However if the engine exhaust constraint was ignored and the exhaust temperature was allowed to rise to 700°C (only permitted for short periods of time) then freely turbocharged operation could be attained over a limited range of engine speed.

The high exhaust temperature was then simulated by fuelling and igniting the auxiliary combustion chamber. This proved to be difficult because stable combustion in the auxiliary chamber could not be achieved, even at very high engine air/fuel ratios (no load condition). This is believed to be due to two primary causes:-

a) The gas flow from the turbocharger (a free running machine) is being adversely affected by the engine (a partially volumetric machine).

b) The engine exhaust pulses transmitted through the auxiliary combustion chamber tend to make the flame unstable.

Stage II Development

The engine system was then altered (fig 5) to a parallel turbocharging system in which the turbocompressor outlet air flow is divided between the engine and the auxiliary combustion chamber. This permits the turbocharger to run independently of the engine and thus allows the turbocharger to be

operated at optimum efficiency. The effect of this has been predicted using the matching program and is shown in fig 6.

A throttling valve is incorporated in the engine air bypass line to provide a positive pressure drop which is necessary to provide an adequate scavenge air flow for satisfactory engine operation.

In this configuration a mixture of exhaust gas and clean air is passed through the auxiliary combustion chamber. It was anticipated that the constant bypass air pressure would alleviate the effects of engine exhaust pulsation on flame stability.

This system is similar in principle to the Hyperbar system (6) but differs in the amount of modification required to the basic engine. A comparison of some of the major design parameters is given in Table 2.

The requirement for a throttled air bypass for a two stroke system is mentioned briefly in a later Hyperbar publication (7) although no two stroke installation is reported.

The stage II system has permitted starting in two modes:-

- 1) Starting the engine, with scavenge blower and turbocharger in series, and then igniting and fuelling the auxiliary combustion chamber.

- 2) Starting the turbocharger using an auxiliary air supply (or an electric motor geared to the turbocompressor, as in the Hyperbar system) necessitating an auxiliary oil pump to supply turbocharger oil until the engine oil system takes over. This procedure is carried out with the scavenge blower disconnected. Once the gas turbine is self sustaining (approx 10 - 15 secs from start up) the engine may be started in the normal manner.

The second starting procedure has the advantage that the scavenge blower becomes redundant. The removal of the scavenge blower has the advantage of reducing noise levels (Tryhorn, Pullen and Grover (8) have demonstrated that the roots blower makes a significant contribution to engine noise). There is also the possibility of recouping approximately 10KW of output power at full rated engine speed which is taken in driving the scavenge blower.

Engine tests performed with the stage II system have shown that the engine runs satisfactorily and that auxiliary combustion can be maintained up to approximately one third of full load conditions. Loading the engine above this value causes flame instability. This is believed to be due to increased levels of exhaust pulses as loading is increased, or possibly the decreasing oxygen content available for auxiliary combustion.

The experience gained from these tests has led to a third stage of development presently undergoing testing.

Stage III Development

This system (fig 7) is essentially similar to the Stage II development except that the engine exhaust gases do not pass through the auxiliary combustion chamber but are mixed with heated bypass gases after the combustion chamber, just prior to the turbine inlet. As the auxiliary combustion chamber is operating with clean, constant pressure air there should be no problems associated with flame instability. The scavenge blower has been physically removed and replaced with a fabricated air manifold.

The engine system is started as in Stage II, method 2. Performance tests will be carried out using different turbocharger builds and frame sizes.

Discussion

Results from the preliminary matching program (fig 4) have shown that turbocharged operation of the engine is feasible. The manufacturers original match has proved unsuitable, displaying the condition shown in fig 8. Later efforts at improving matching by the use of a smaller turbine housing have had only limited success (figs 9,10) ie turbocharged operation has been demonstrated over a restricted range of engine speed, only if engine exhaust temperature is allowed to rise to 650-700°C. This condition is displayed by fig 11.

Thus turbocharged operation of this engine with the low turbocharger efficiency in this size range demands a high turbine inlet temperature. This is made possible by the incorporation of the auxiliary combustion chamber.

The Stage II Development offers a system which features reburning of the engine exhaust yielding lower unburnt hydrocarbon emissions (but with a probable penalty regarding NO_x emissions). This development has also shown that pulsed auxiliary combustion is more easily maintained when constant pressure air is mixed with the engine exhaust, although the performance is still unacceptable.

The engine used is fitted with a DPA type injector pump which will deliver a fixed amount of fuel at full load, depending on engine speed. It is not possible to alter the fuel delivery at the present time, thus it is not possible to show improved power curves. The improvements shown in figs 9,10 are due to the fact that the scavenge blower delivers air at a lower pressure than on the standard engine because of the turbocharger 'partially unloading' the scavenge blower. Hence the scavenge blower requires less power to run it, the difference being available at the engine output shaft.

Improvements in charge air density and trapped air flow are shown in figs 12,13. In all cases (except standard engine) these curves show developments I and II using two stage compression (turbocharger in series with scavenge blower). The improvements of the smaller turbine housing are shown, as well as the effect of fuelling the auxiliary combustion chamber.

Performance results (figs 14,15,16) have been calculated for Stage I and II Developments using actual trapped air mass results obtained and assuming full load conditions. These calculations were carried out using the air/fuel ratio and brake thermal efficiency pertaining to each engine speed as demonstrated on the standard engine.

These figures show that power increases in excess of 40% are possible. Further development of Stage III and changes of the injector pump should allow these results to be realised.

An advantage of the use of the auxiliary combustion chamber and throttled bypass system is that a wide range of turbocharger sizes may be used at optimum efficiency thus eliminating the need for complicated matching procedures. Some form of matching will be necessary to optimise the fuel consumption of the auxiliary combustion chamber.

Conclusion

The work carried out so far has demonstrated that the engine used cannot be conventionally turbocharged with the engine exhaust temperature limit of 600°C .

If this limit can be raised to 700°C then a classically turbocharged system incorporating some form of low speed boost assistance is feasible.

The developments detailed in this paper have shown some measure of success in overcoming this problem and have resulted in a hybrid system allowing the removal of the scavenge blower. With further development the system should demonstrate useful improvements in power output.

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S.A.E. Paper No 750840 from S.A.E. Diesel Engine Noise Conference SP397 Aug 1975.

Table 1Chrysler TS3

Two stroke, horizontal three cylinder with opposed pistons, direct injection diesel engine.

Type 3D215

Bore 3.375 in (85.7mm)
 Stroke 4.00 in (101.6mm)
 Effective Compression Ratio 16:1
 Capacity 215 cu. in (3520cc)
 B.H.P 125 (93.2 kw) at 2400 rev/min.
 No of cylinders Three
 Firing order 1,2,3
 Max. Torque 313 lb ft (425 Nm) at 1200 rev/min
 Blower Make Wade 5R034 No, 2679
 Type 'Roots' with two 3 lobe rotors

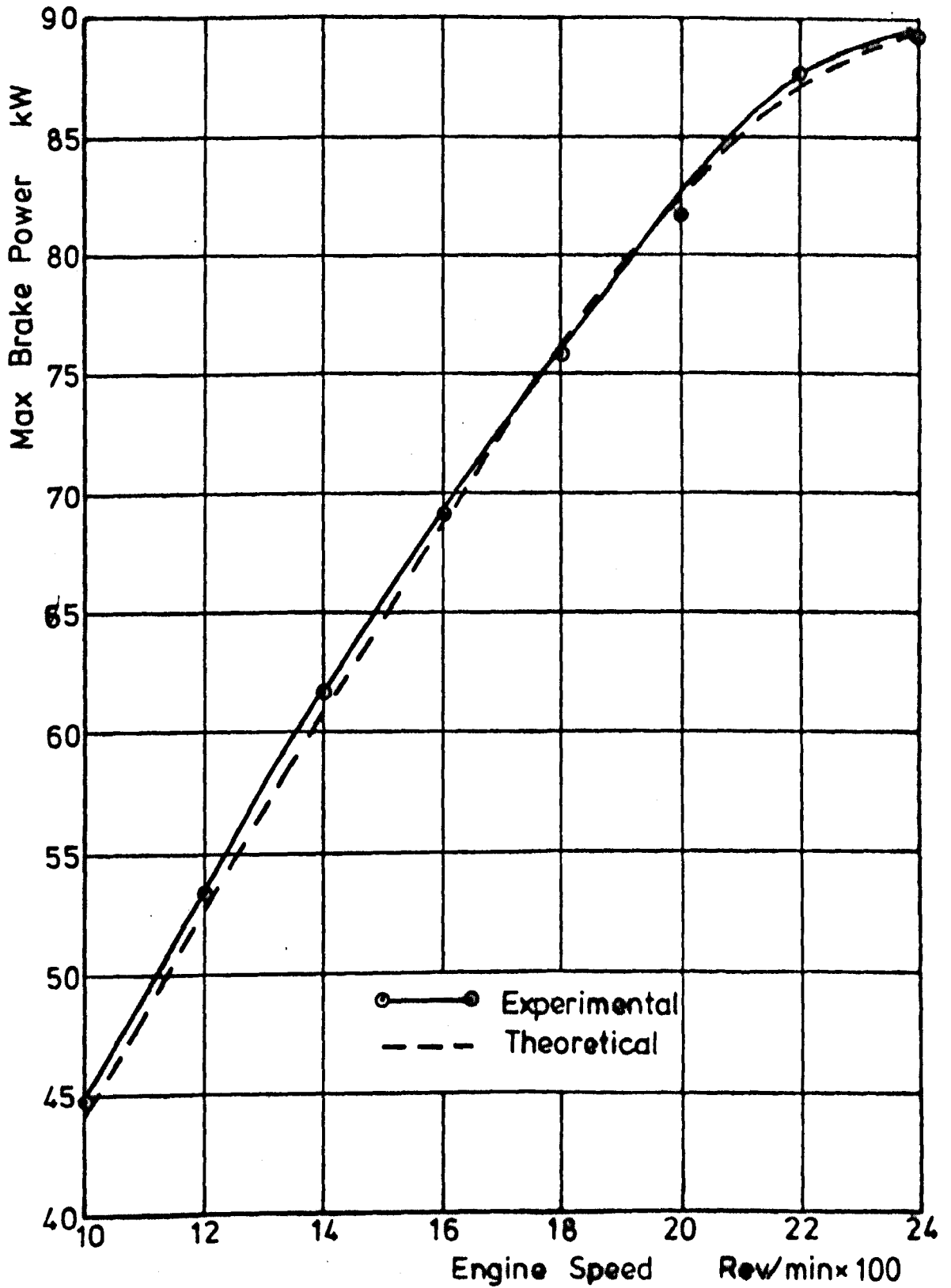
Table 2

	Leicester Polytechnic System	Hyperbar System
Boost Ratio	2.3:1	9:1*
Compression Ratio	16:1 ⁺	5:1
Minimum Air/Fuel Ratio	22:1 (proposed)	30-35:1

* The Hyperbar system utilises two stage turbocharging

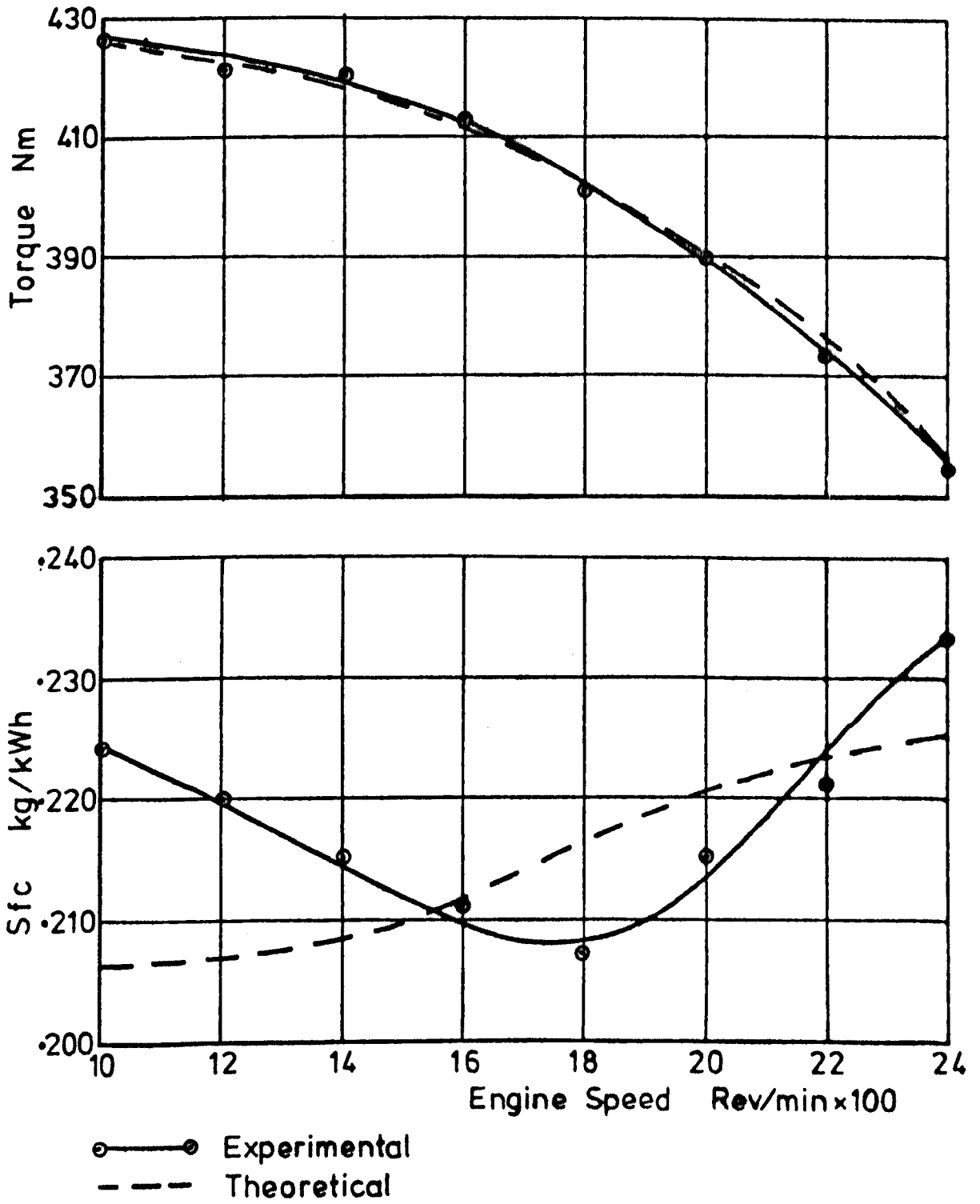
+ Standard engine compression ratio

FIGURE 1



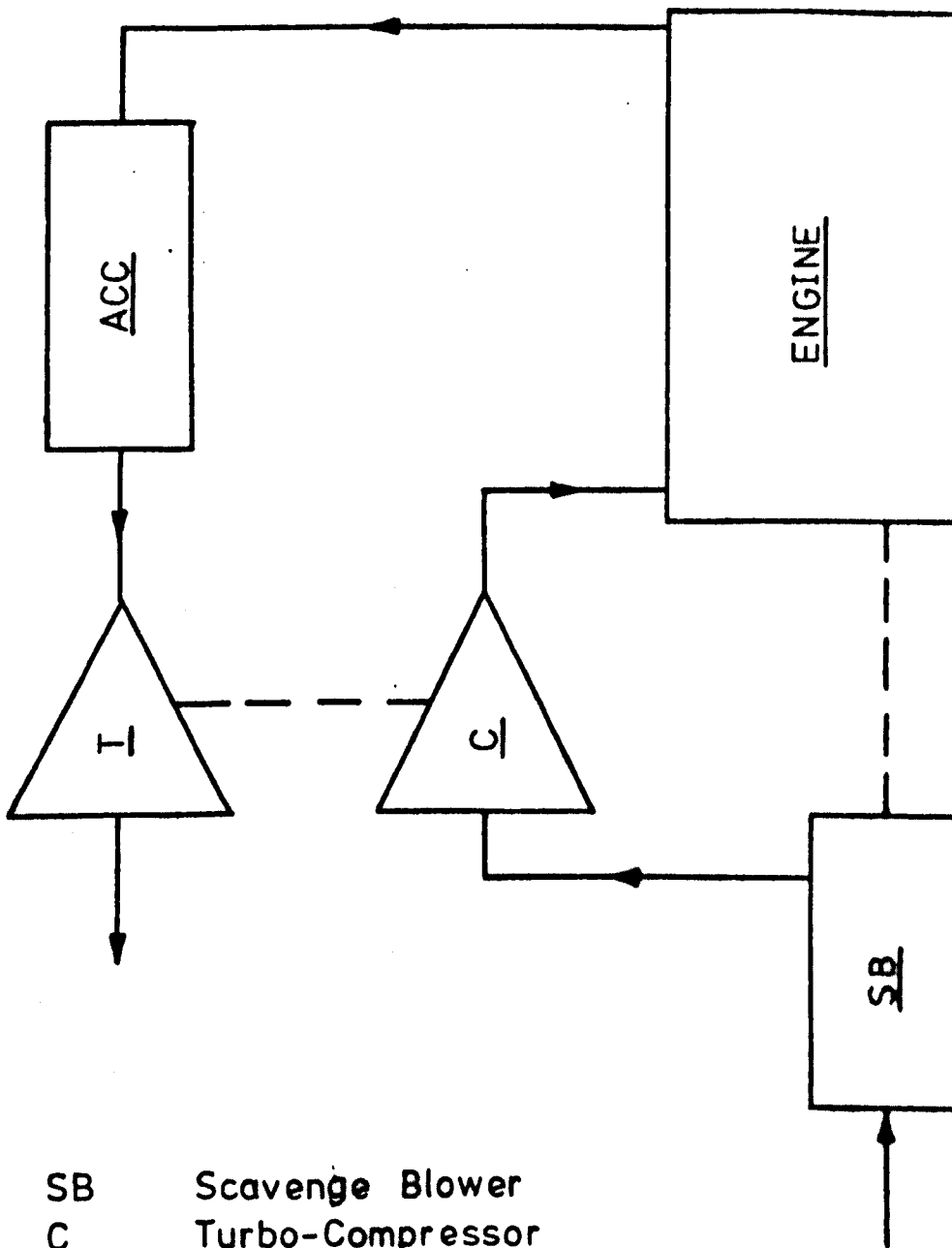
STANDARD ENGINE PERFORMANCE

FIGURE 2



STANDARD ENGINE PERFORMANCE

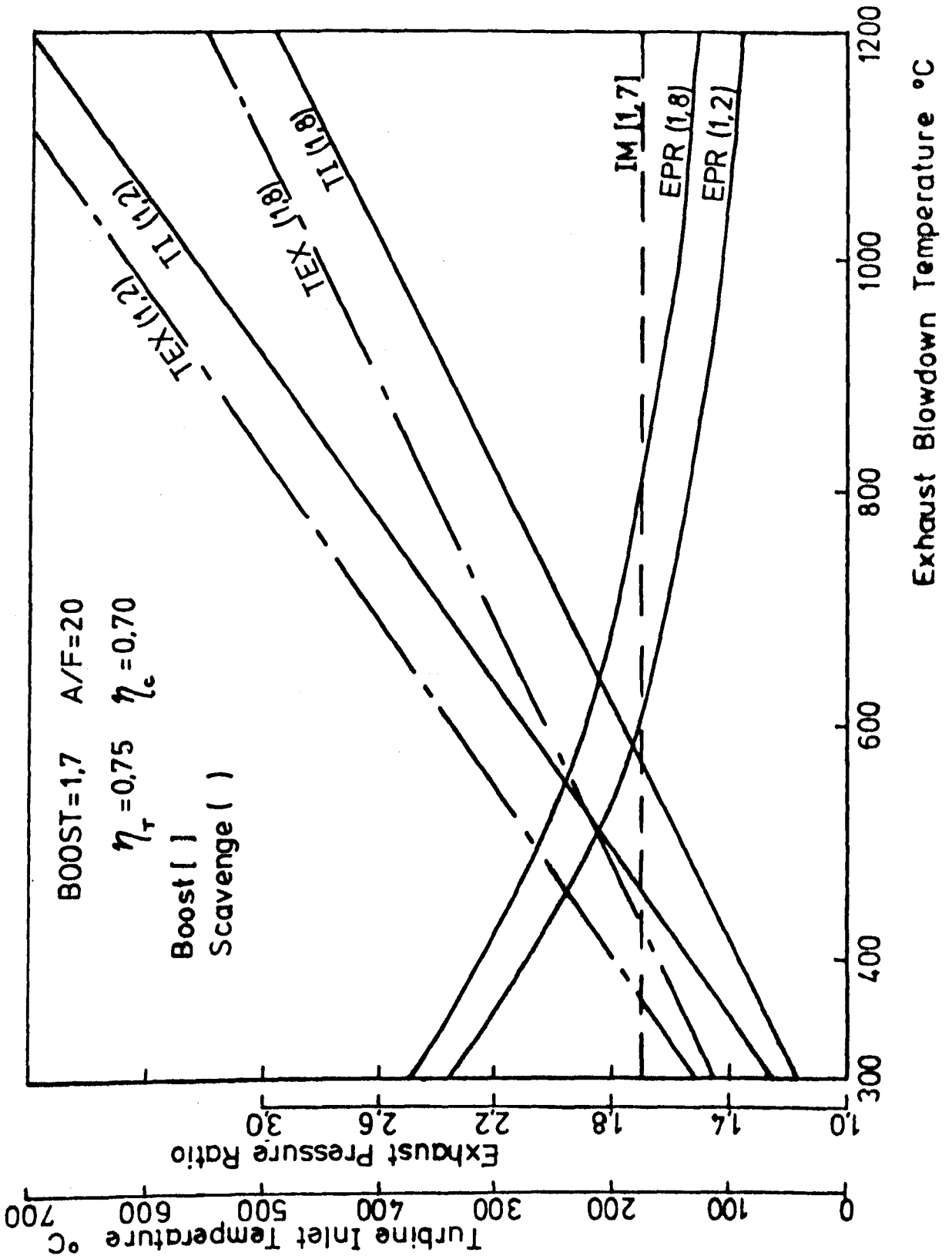
FIGURE 3



- SB Scavenge Blower
- C Turbo-Compressor
- ACC Auxiliary Combustion Chamber
- T Turbine
- - - Mechanical Link
- > Thermodynamic Link

ENGINE SYSTEM: STAGE I DEVELOPMENT

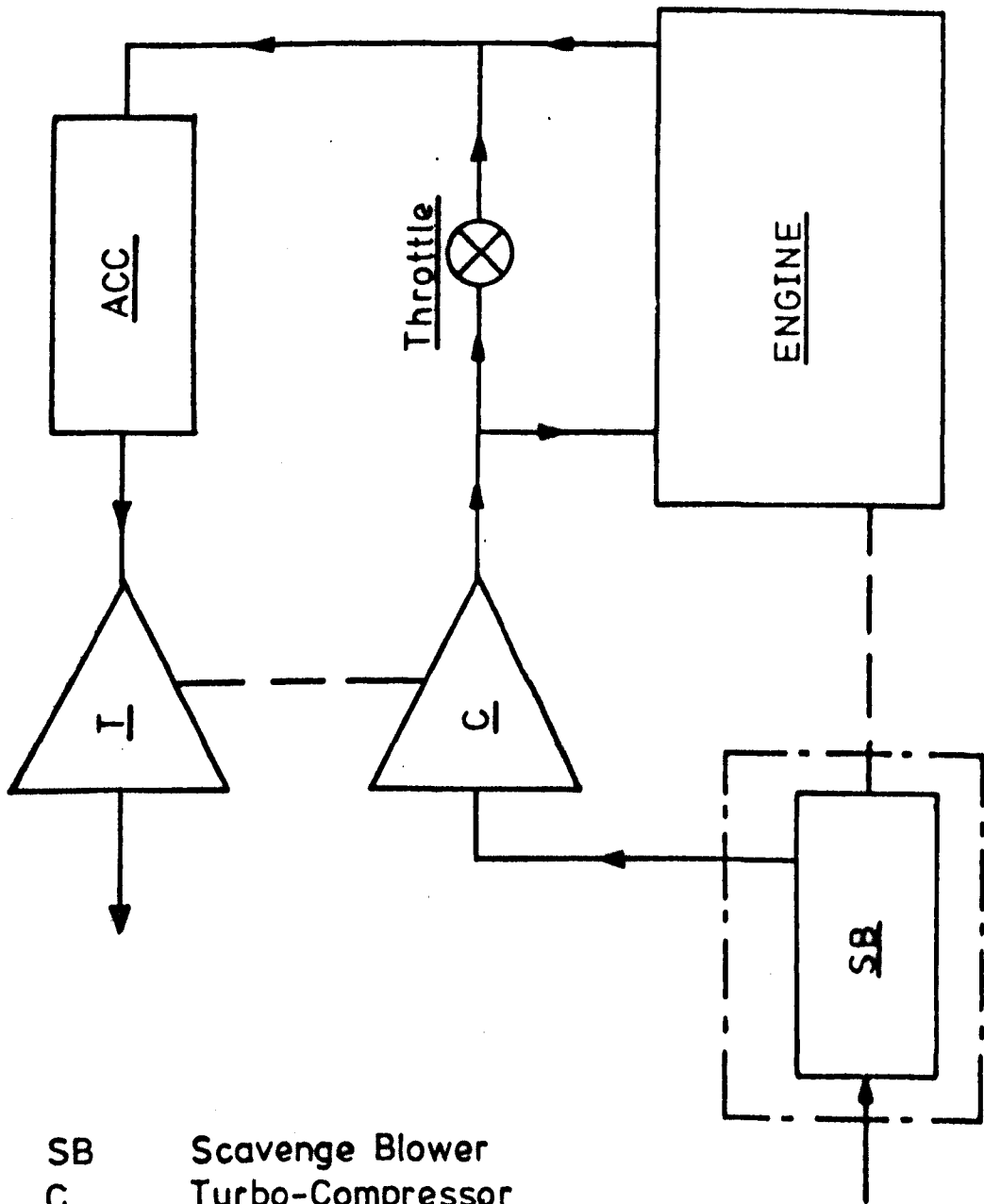
FIGURE 4



PRELIMINARY MATCHING PROGRAM

BOOST = 1,7
 $\eta_{\tau} = 0,75$
 $\eta_c = 0,70$

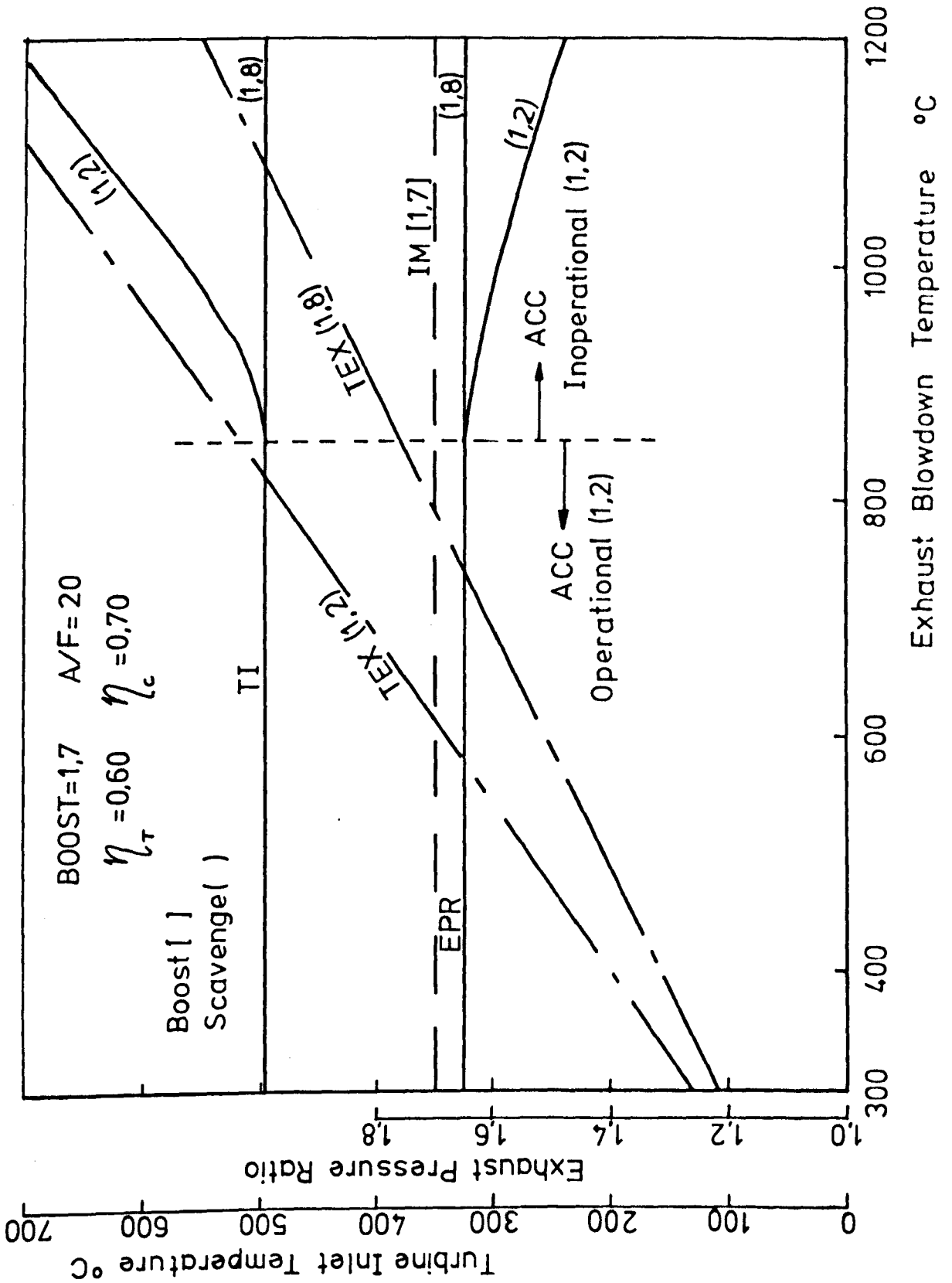
FIGURE 5



- SB Scavenge Blower
- C Turbo-Compressor
- ACC Auxiliary Combustion Chamber
- T Turbine
- — Mechanical Link
- > Thermodynamic Link

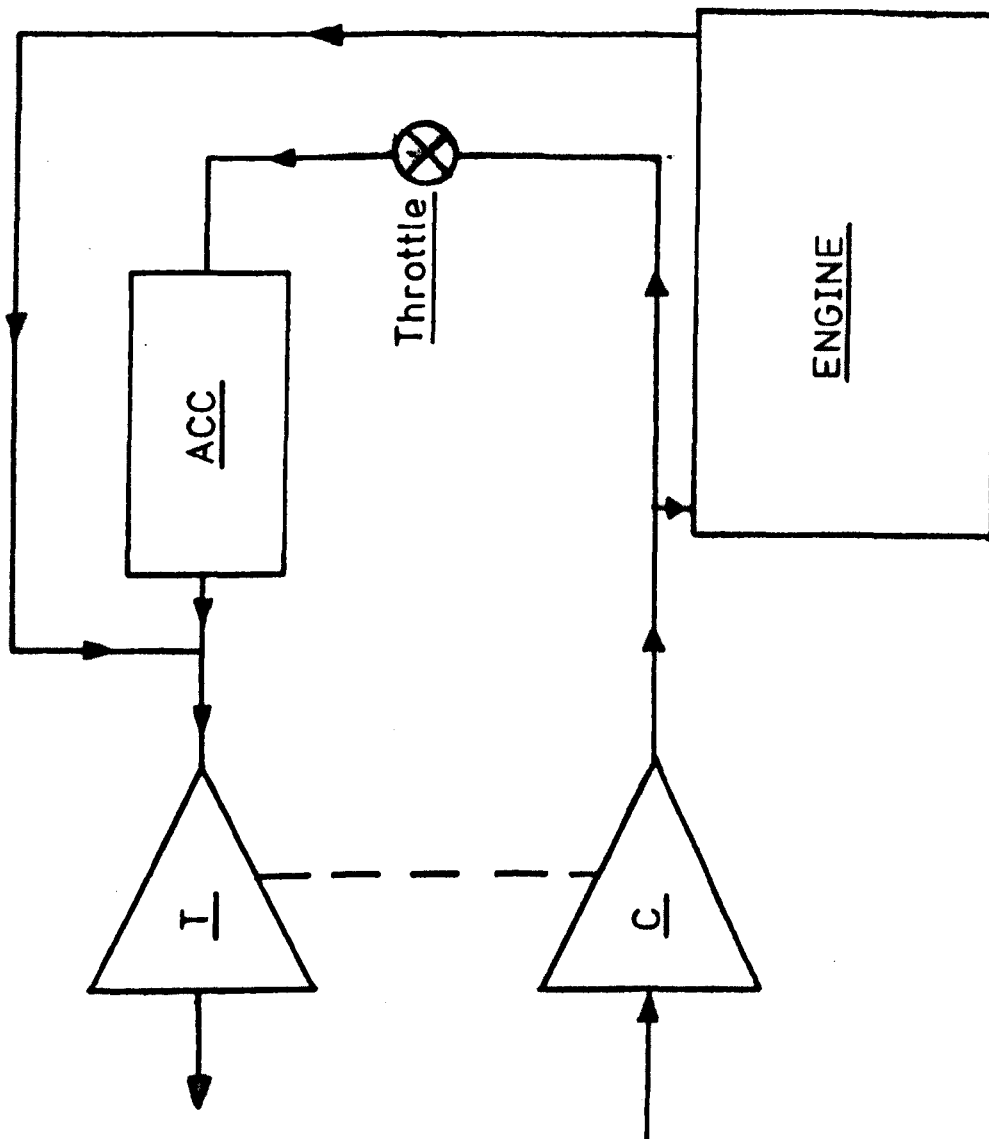
ENGINE SYSTEM: STAGE II DEVELOPMENT

FIGURE 6



PRELIMINARY MATCHING PROGRAM
[Combustion Chamber Operational]

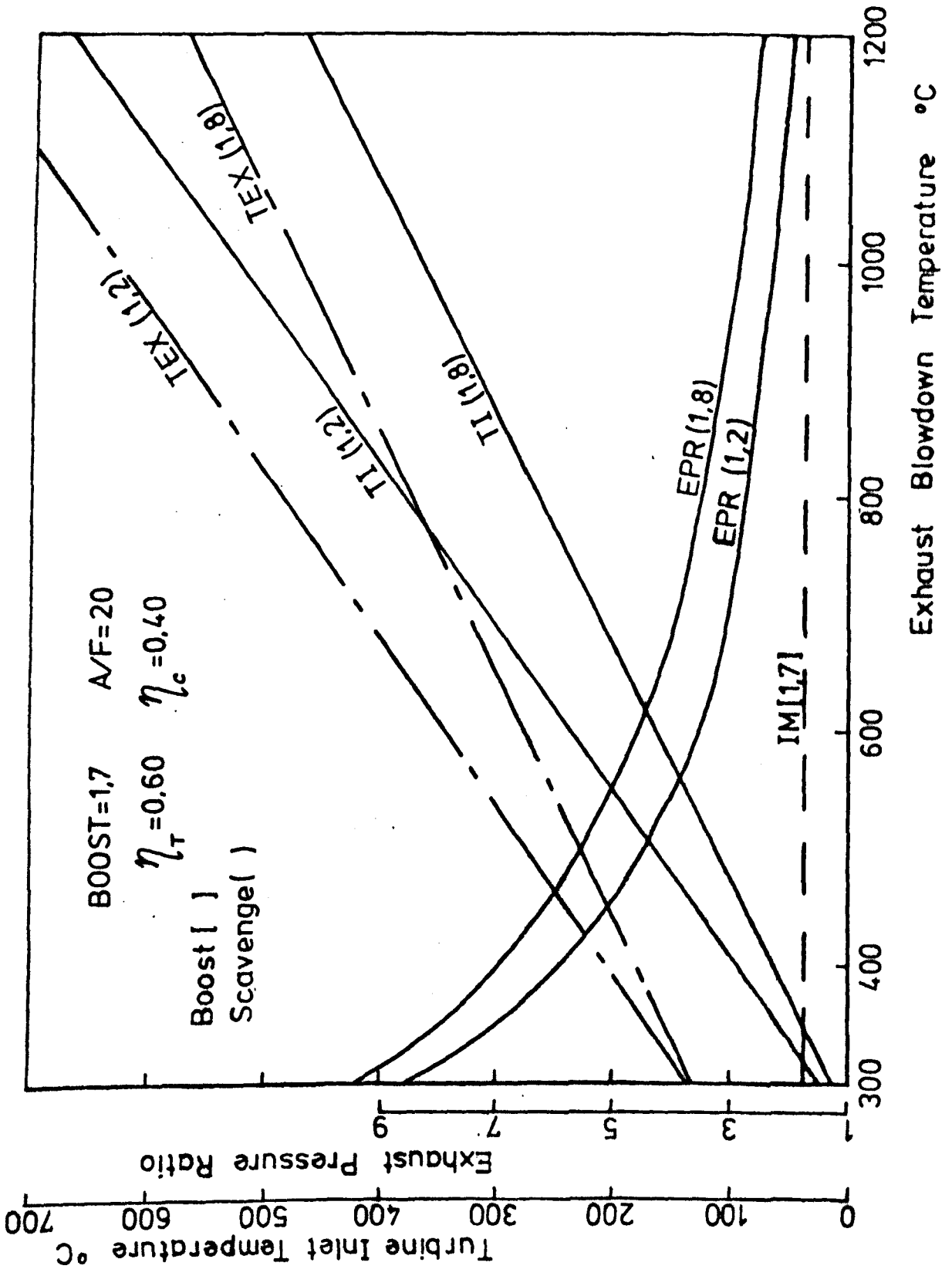
BOOST=1,7
 $\eta_{\tau}=0,60$
 $\eta_c=0,70$



- C Turbo-Compressor
- ACC Auxiliary Combustion Chamber
- T Turbine
- — Mechanical Link
- > Thermodynamic Link

ENGINE SYSTEM: STAGE III DEVELOPMENT

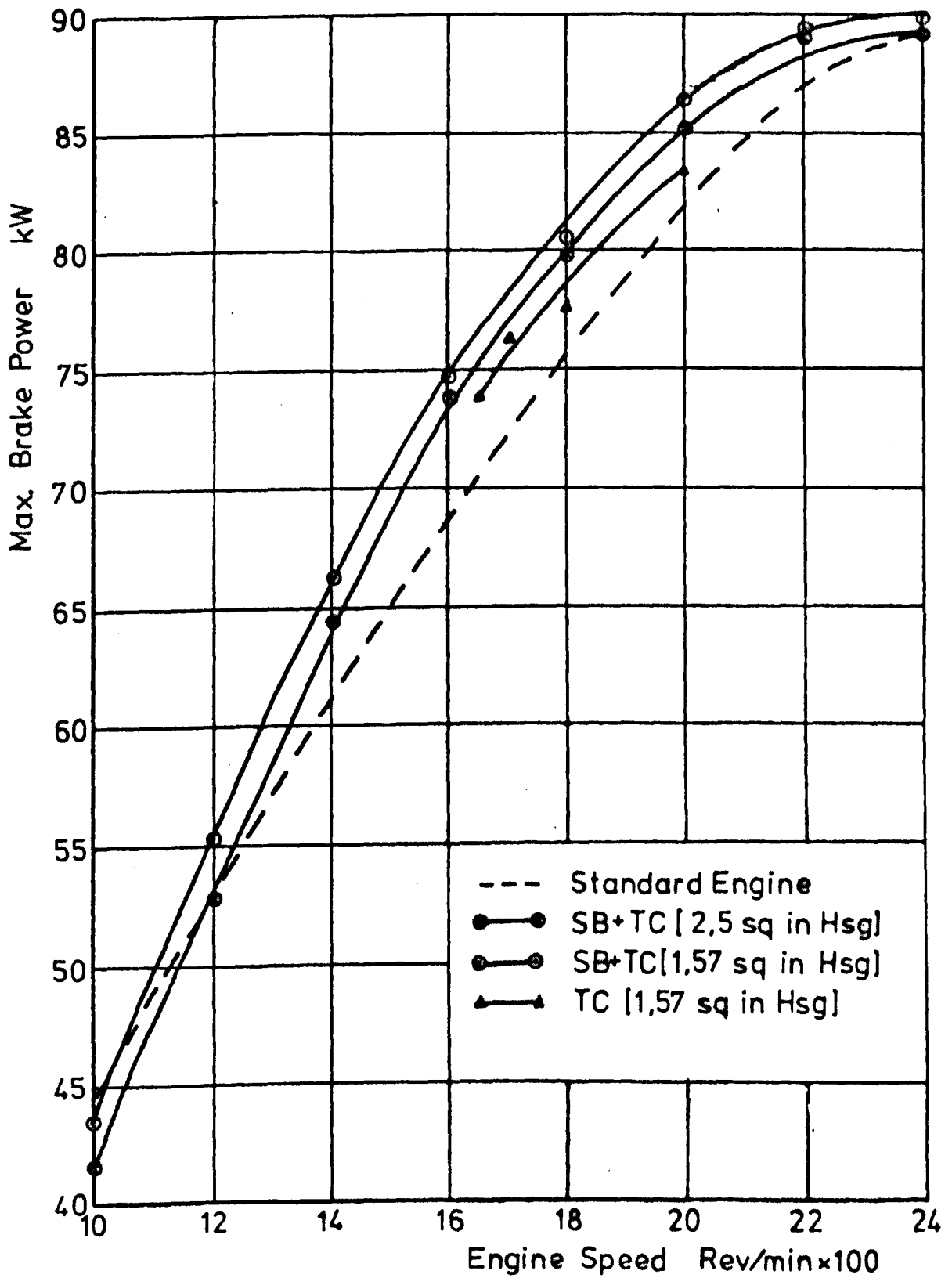
FIGURE 8



PRELIMINARY MATCHING PROGRAM

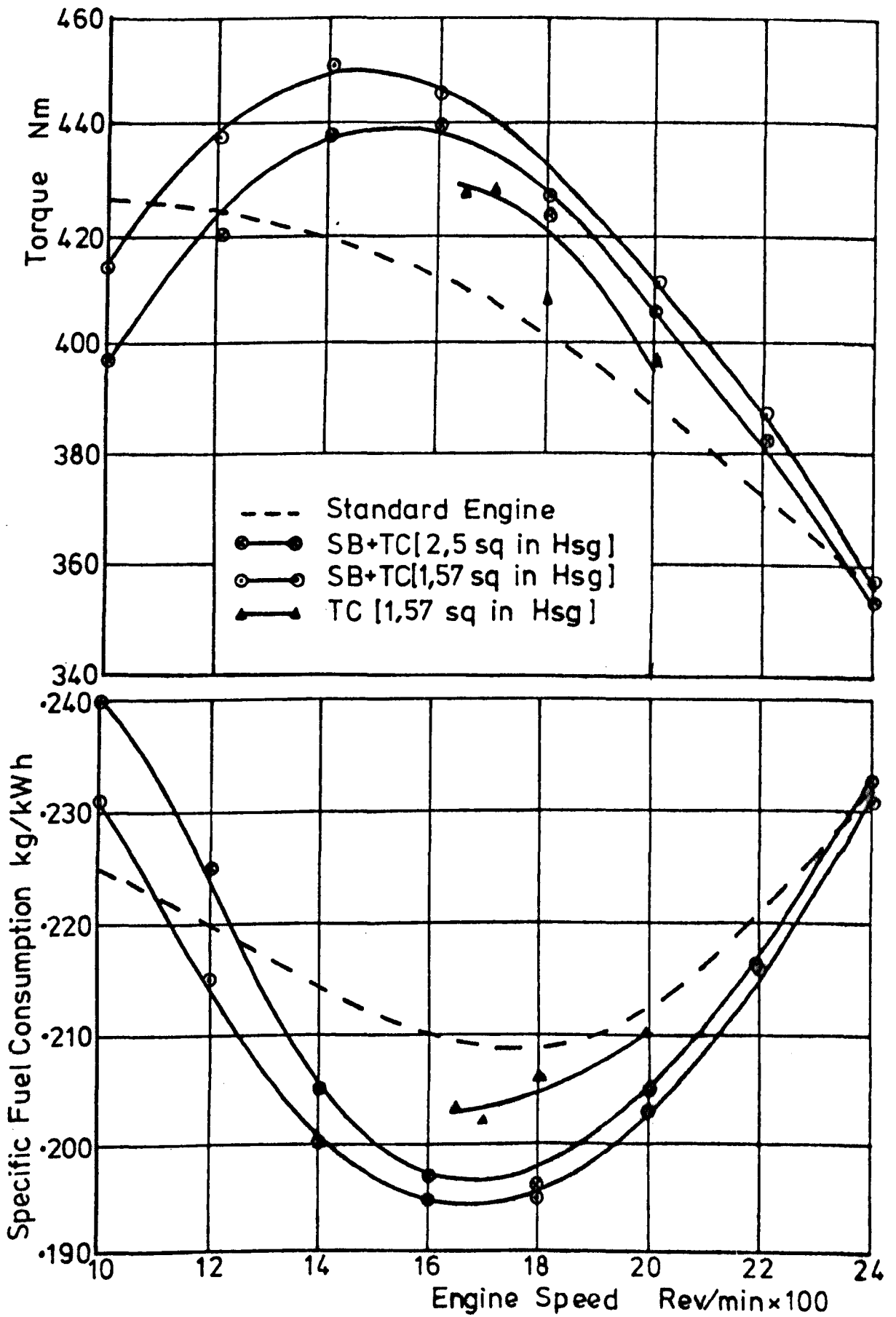
BOOST = 1.7
 $\eta_t = 0.60$
 $\eta_c = 0.40$

FIGURE 9



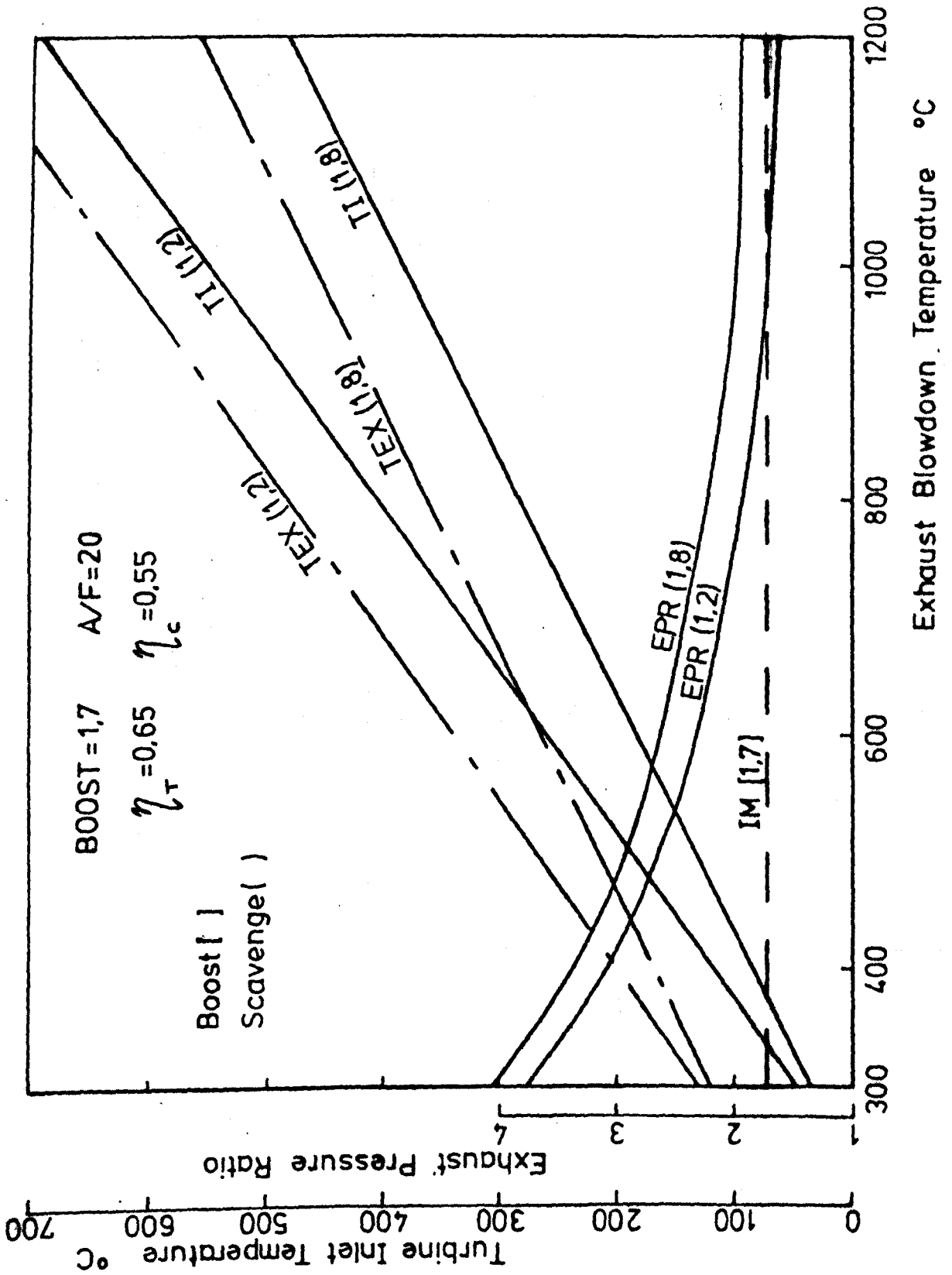
ENGINE SYSTEM PERFORMANCE
STAGE I DEVELOPMENT

FIGURE 10



ENGINE SYSTEM PERFORMANCE STAGE I DEVELOPMENT

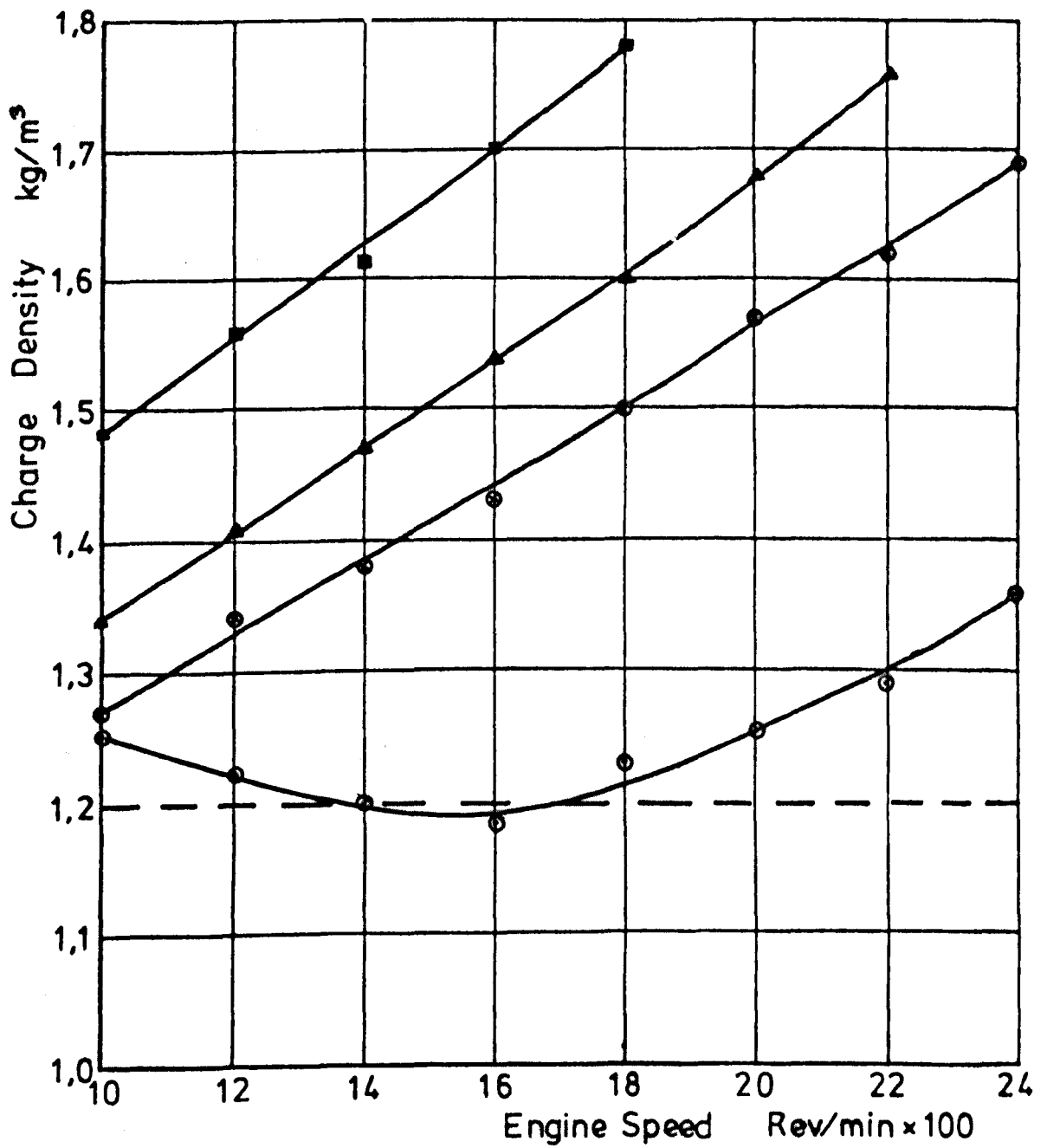
FIGURE 11



PRELIMINARY MATCHING PROGRAM

BOOST = 1.7
 $\eta_{\tau} = 0.65$
 $\eta_c = 0.55$

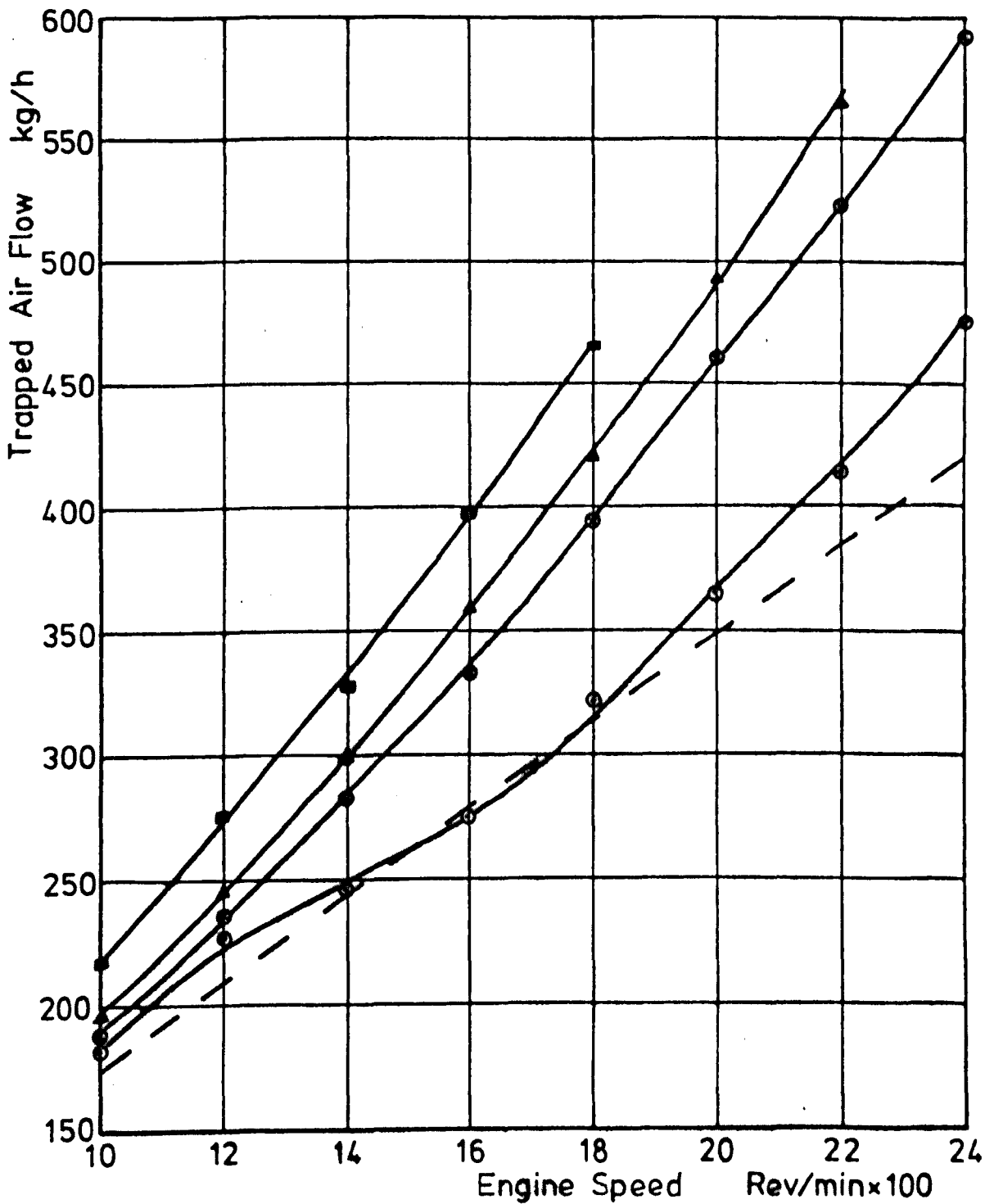
FIGURE 12



- Standard Engine
- SB+TC [2,5 sq in Hsg]
- ▲—▲ SB+TC [1,57 sq in Hsg]
- SB+TC+ACC [1,57 sq in Hsg]
- - - Naturally Aspirated Conditions

CHARGE AIR DENSITY
DEVELOPMENTS I AND II

FIGURE 13

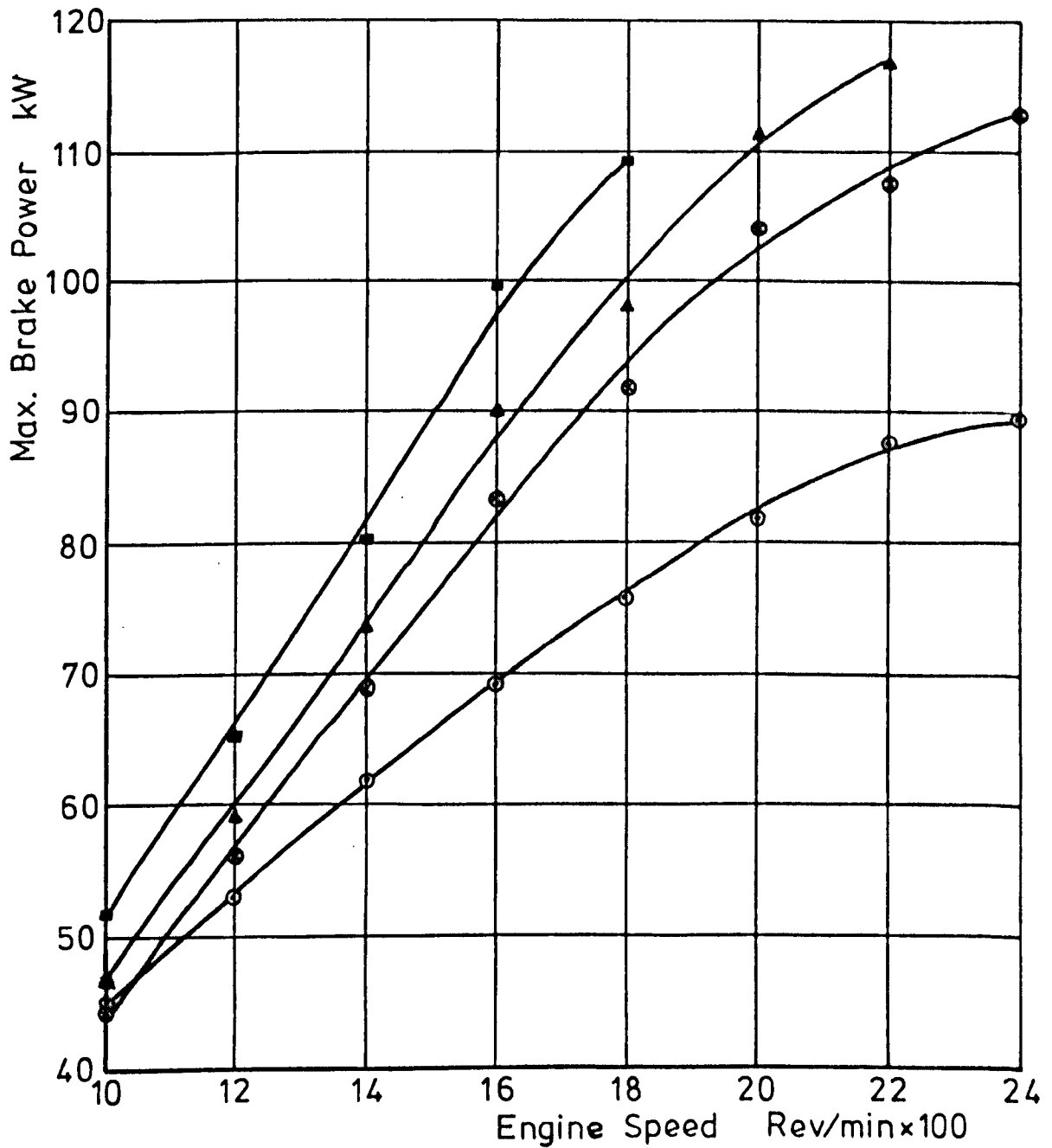


- Standard Engine
- SB+TC [2,5 sq in Hsg]
- ▲—▲ SB+TC [1,57 sq in Hsg]
- SB+TC+ACC [1,57 sq in Hsg]
- - - Naturally Aspirated Conditions

TRAPPED AIR FLOW

DEVELOPMENTS I AND II

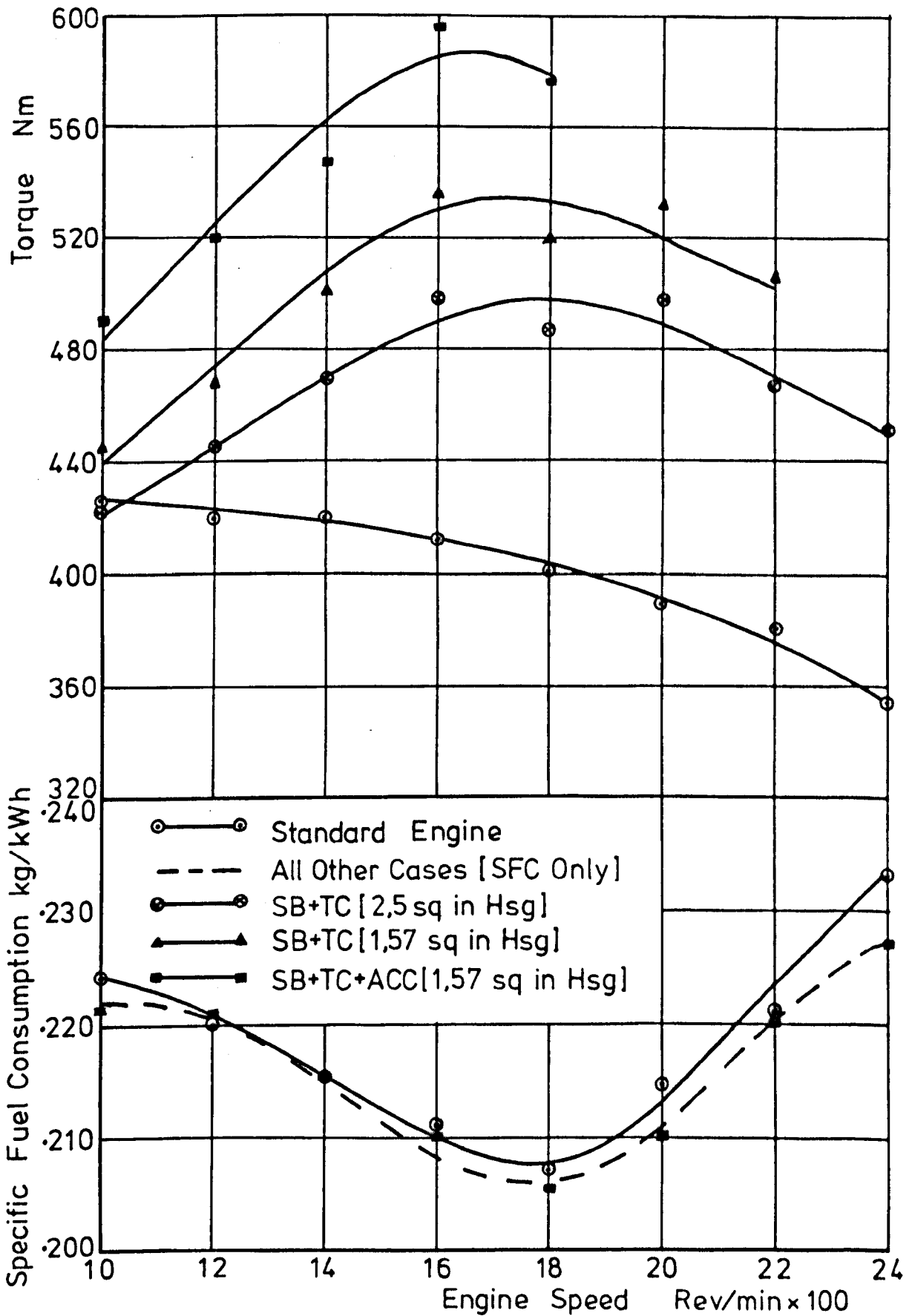
FIGURE 14



- Standard Engine
- ⊗—⊗ SB+TC [2,5 sq in Hsg]
- ▲—▲ SB+TC [1,57sq in Hsg]
- SB+TC+ACC [1,57 sq in Hsg]

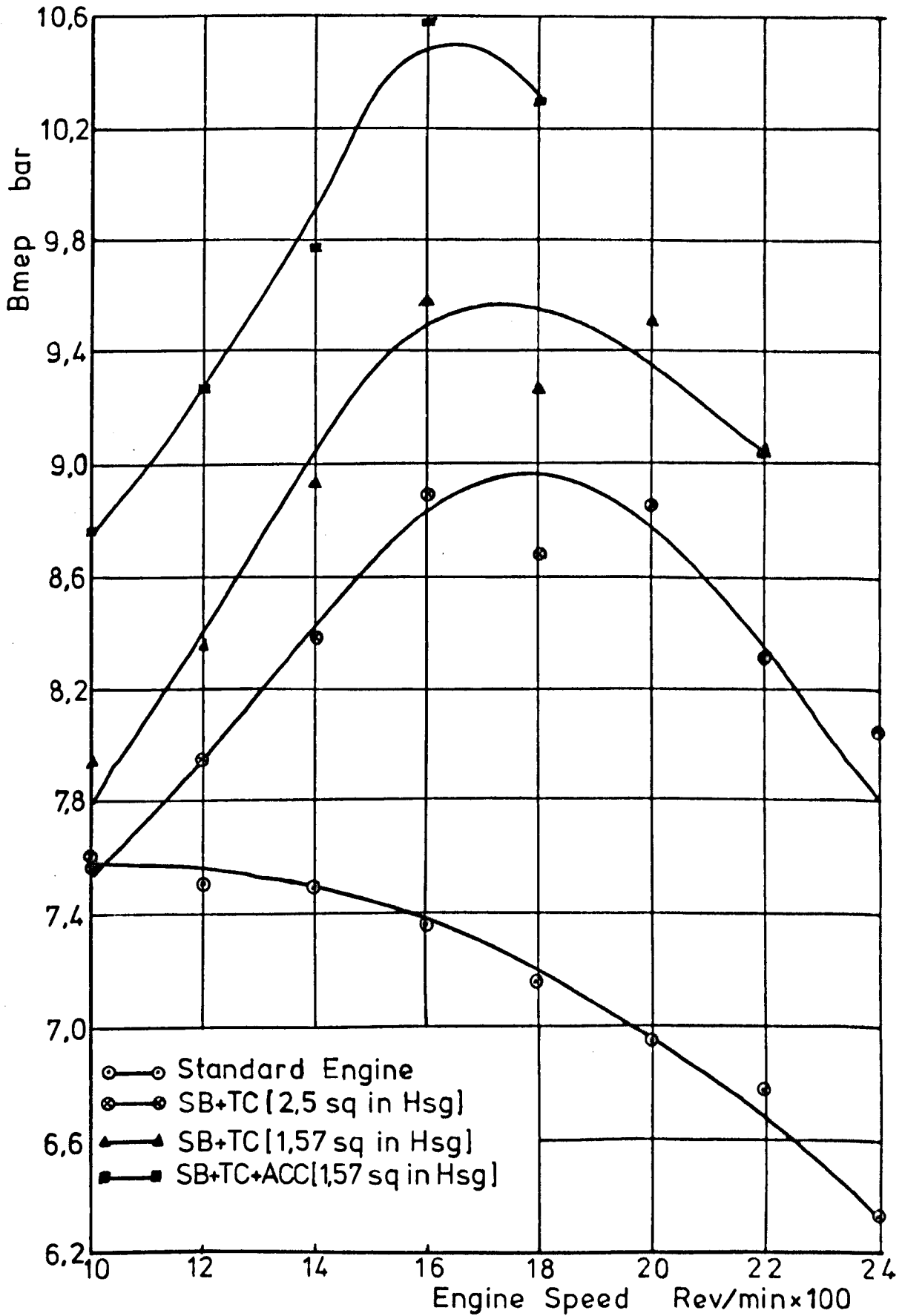
ENGINE PERFORMANCE DEVELOPMENTS I AND II
[CALCULATED TO COMPENSATE FOR A/F RATIO]

FIGURE 15



ENGINE PERFORMANCE DEVELOPMENTS I AND II
[CALCULATED TO COMPENSATE FOR A/F RATIO]

FIGURE 16



BMEP DEVELOPMENTS I AND II
[CALCULATED TO COMPENSATE FOR A/F RATIO]

APPENDIX 6

THE DIESEL GAS TURBINE

Mostanbo

Submitted for the Henry R Worthington
award, administered by the Von Karmann
Institute, Brussels. January 1979

MOSTANBO

THE DIESEL GAS TURBINE

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D.W. Tryhorn, H.L. Pullen, E.C. Grover.
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August, 1975.

THE DIESEL GAS TURBINE

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Summary:-

This paper investigates a diesel gas turbine comprising a two-stroke diesel engine, turbocharger and auxiliary combustion chamber.

The system is investigated using a single turbocompressor in conjunction with a number of different turbine housings, in order to isolate the important parameters of system operation.

It was found that the system shows great promise for traction applications giving $\approx 10\%$ improvement in back-up torque. There are, however, penalties incurred in specific fuel consumption and in system complexity.

List of Symbols used in Figures.

ACC	Auxiliary Combustion Chamber.
A/F	Trapped Air/Fuel Ratio.
BMEP	Brake Mean Effective Pressure.
C	Turbo Compressor.
DFE	Diesel Fuel Equivalent.
DGT	Diesel Gas Turbine.
Hsg	Turbine Housing Size (Quantified by throat area e.g. 1.41 sq.in, 1.57 sq.in, 2.5 sq.in.)
T	Turbine.
ΔP_E	Pressure drop across the engine.
ρ_A	Charge air density.

Introduction

The turbocharging of diesel engines for increased power output has been common practice for some years. Prime movers for automotive heavy transport are predominantly either naturally aspirated or turbocharged four stroke diesels.

The trend towards more compact, lighter power plants favours the two stroke engine as it can, in theory, deliver roughly twice the power output as a four stroke of corresponding bulk at the same engine speed.

There are certain drawbacks in turbocharging automotive two stroke engines that must work over a wide range of engine speeds. The main difficulty is that of producing adequate scavenge at low engine speed when turbocharger boost is low or non-existent. This paper describes an attractive combination of two stroke engine, turbocharger and auxiliary combustion chamber which overcomes these difficulties.

History

The engine used for this programme of work was chosen because its characteristics have been well established (1,2). The engine has also been the basis of comparisons of turbocharged, compound and gas generator systems (3,4,5) and novel hybrid power plants (6,7,8,9). The engine has proved to be amenable to systems having an increased power output within certain limitations. These limitations involve the necessity to use some form of scavenge blower to provide low speed boost in turbocharged form and the mechanical complexities of the compound system, involving mechanical links between turbine, compressor, engine and scavenge blower.

Present work

The present work is a continuation of some earlier work (10), which showed that combinations of engine, turbocharger, scavenge blower and auxiliary combustion chamber could produce a feasible prime mover.

The continuation of this work was originally intended as an optimisation programme, but it was discovered that the system had serious drawbacks. A new investigation was undertaken to examine the diesel gas turbine and to define the important parameters of its operation.

Improvements were made in instrumentation (11) and dynamometer operation (12). Investigation of turbocharged operation of the engine was carried out both with and without scavenge blower assistance, together with various combinations of auxiliary combustion chamber. The work was carried out with reference to certain engine exhaust temperature limits and peak cylinder pressure limits set at 650°C and 115 bar respectively by earlier workers (7,9). The results of this investigation are detailed elsewhere (13), and led to the proposal of a system overcoming the earlier disadvantages and working within peak cylinder pressure and engine exhaust temperature constraints.

Details of the engine used are given in Table I.

System details

The Diesel Gas Turbine system (fig.1.) consists of a turbocharger, two stroke opposed piston diesel engine and an auxiliary combustion chamber.

The turbocompressor air delivery is split between the engine and the auxiliary combustion chamber. The engine supply incorporates a valve that is used to cut off the supply for system start-up. The supply to the auxiliary combustion chamber may be throttled to ensure a positive pressure drop across the engine and thus adequate scavenging. The engine exhaust is mixed with the hot gas from the auxiliary combustion chamber just prior to the turbine inlet. The auxiliary combustion chamber is one section from an aircraft gas turbine combustion system. It is fuelled with propane and ignited using a conventional automotive spark plug. The spark is produced by a modified, proprietary, capacitive discharge, electronic, ignition unit.

The system has some similarities with the Hyperbar System (14) although the Hyperbar utilises four stroke engines. However, the necessity of a throttled air bypass for two stroke application is mentioned in a later Hyperbar publication (15), but no such installation is reported. The main differences between this system and the Hyperbar system are detailed in Table 2.

There are several advantages in the diesel gas turbine and the particular configuration reported here.

- 1) The gas turbine section is self-sustaining, thus boost levels and air flow rates can be varied irrespective of engine speed. However, this includes penalties in auxiliary combustion chamber fuel consumption.
- 2) The system may be operated at optimum turbine and compressor efficiencies.
- 3) Since the turbine operation is independent of engine speed, there is no necessity for a scavenge blower. This has benefits in terms of more power being available at the engine output shaft with a reduced noise level. (The Roots type blower used with this engine has been shown to make a major contribution to engine noise (16)).
- 4) The engine itself is not subject to drastic modifications i.e. it has the same compression ratio, ignition timing etc. The only modifications are the removal of the scavenge blower and alterations to the exhaust ducting.
- 5) The present system has advantages over previous designs (13), in that the auxiliary combustion chamber operates with clean, constant pressure air thus avoiding problems of flame instability due to pulsating combustion. Since the engine exhaust gas is admitted just before the turbine inlet, the engine exhaust pulsations are utilised to good effect in the turbine.

This system has been investigated using a turbocharger match recommended for this engine by a turbocharger manufacturer. The same turbocompressor was used throughout in combination with three different turbine housings (classified by throat area) in order to investigate the effect of turbine size. This meant that the boost ratio must not exceed 2.2 to remain within the peak cylinder pressure constraint of 115 bar. The DPA fuel pump fitted to this engine delivers a fixed and limited amount of fuel at full load conditions for a given engine speed. For this reason the air/fuel ratio could not be maintained at the same value as on the standard engine.

Experimental Procedure

Starting of the diesel gas turbine is effected as follows:-

- 1) The engine air supply valve is fully closed, thus eliminating the engine from the system.
- 2) The air bypass valve is fully opened. This leaves the system as a free running gas turbine.
- 3) An auxiliary oil pump is started so that oil may be delivered to the turbocharger.
- 4) The auxiliary combustion chamber igniter is activated.
- 5) A slave air supply is connected to the auxiliary combustion chamber and fuelling is initiated.
- 6) When the flame is stabilised (viewed through a quartz observation window in the auxiliary combustion chamber), the slave air flow is increased until the gas turbine is self-sustained (about 15 secs. from first fuelling).
- 7) The slave air supply is disconnected and the engine air supply valve is fully opened.
- 8) The air bypass valve and auxiliary combustion chamber fuelling are adjusted until there is a boost ratio of 1.5 and a pressure drop across the engine of 0.01 bar.

- 9) The engine is then started in the conventional manner.
- 10) The auxiliary oil pump is turned off, the supply of turbocharger oil being taken over by the engine oil circuit.

Load may then be applied to the engine. Tests were carried out at various engine speeds, the speed being increased in 200 rev/min steps until the power output falls to that of the corresponding standard engine power, or below. Results are shown in Figs 2-6.

Tests were also carried out at a constant engine speed of 1200 rev/min by varying auxiliary combustion chamber fuelling and hence boost. These tests give a good comparison of performance obtained with different turbocharger builds (in this case different turbine housings). Results of these tests are given in Figs 7-10.

Discussion

From the tests at various engine speeds it can be seen that there is a useful power increase at low to medium engine speed (Fig 2). This increase (9%) is greater than can be explained by the recouping of power from the removal of the scavenge blower. It is probably due to better fuel vaporisation (there is no charge cooler) and hence more efficient combustion. There is a useful torque 'back-up'. Engine specific fuel consumption is much improved (Fig 3), as is charge air density, trapped and total air/flow and b.m.e.p. (Figs 4,5,6).

At low engine speed there is little to choose between turbine housings, but as speed increases the larger housing permits extended operation. The use of a larger housing has penalties in the fuel needed for the auxiliary combustion chamber. The specific fuel consumption of the total system expressed as a diesel fuel equivalent shows the expected penalties of the diesel gas turbine system (Fig 6).

If the air/fuel ratio was decreased from $\approx 28:1$ to $20:1$, thus burning more fuel in the engine cylinders, the specific fuel consumption of the system would be improved as it would not be necessary to increase the fuelling of the auxiliary combustion chamber. This could yield power increases of $\approx 60\%$ for the larger turbine housing as compared with the standard engine.

It can be seen (Fig 5) that the pressure drop across the engine falls as speed increases, due to turbine 'choking', with a corresponding increase in smoke emission, although the trapped air/fuel ratio is still $28 - 30:1$ (Fig 4). Turbine 'choking' causes the fall in engine pressure drop thus reducing scavenge to unacceptable levels. Thus the trapped air/fuel ratio at this point should only be regarded as an apparent value. Ideally the engine pressure drop should rise with engine speed.

Tests conducted at constant engine speed with varying charge pressures again show the larger housing to be more effective. Power does not deteriorate so rapidly as charge pressure is reduced (Fig 7). This is due to the fact that the engine pressure drop does not fall off as rapidly with the larger housing and thus adequate scavenging is more easily maintained. Again a fuel consumption penalty is found with a larger housing (Fig 10).

No results were obtained with the smallest housing (1.41 sq.in.) due to the system persistently going into surge conditions on engine start up. It was also impossible to obtain a suitable pressure drop across the engine prior to engine start up. It was thus concluded that the turbine housing was far too small.

It would appear that the diesel gas turbine operates best with large turbine housings such that an adequate engine pressure drop, hence acceptable scavenge, can be maintained. However a penalty is incurred in specific fuel consumption.

Fuel economy could be improved by attention to the whole system. Charge cooling, of the engine air supply branch only, would increase charge density without significantly reducing charge pressure. This would enable more fuel to be burnt in the engine cylinders, leading to improvements in specific fuel consumption. Further improvements could be made by heat exchange between the turbine exhaust and bypass air supply to the auxiliary combustion chamber, thus reducing auxiliary fuelling. However this would significantly increase mechanical complexity.

Further work is proceeding with the investigation of turbochargers of increased frame size and efficiency.

Conclusion

The diesel gas turbine provides a possible method of increasing power output and 'back-up' torque of a two stroke diesel engine.

It is necessary to use relatively large turbine housings to maintain a sufficient pressure drop across the engine for adequate scavenge. This pressure drop should increase with engine speed.

The diesel gas turbine incurs some penalties in specific fuel consumption but this could be improved at the expense of system complexity.

TABLE 1Chrysler TS3

Two stroke, horizontal three cylinder with opposed pistons
direct injection diesel engine.

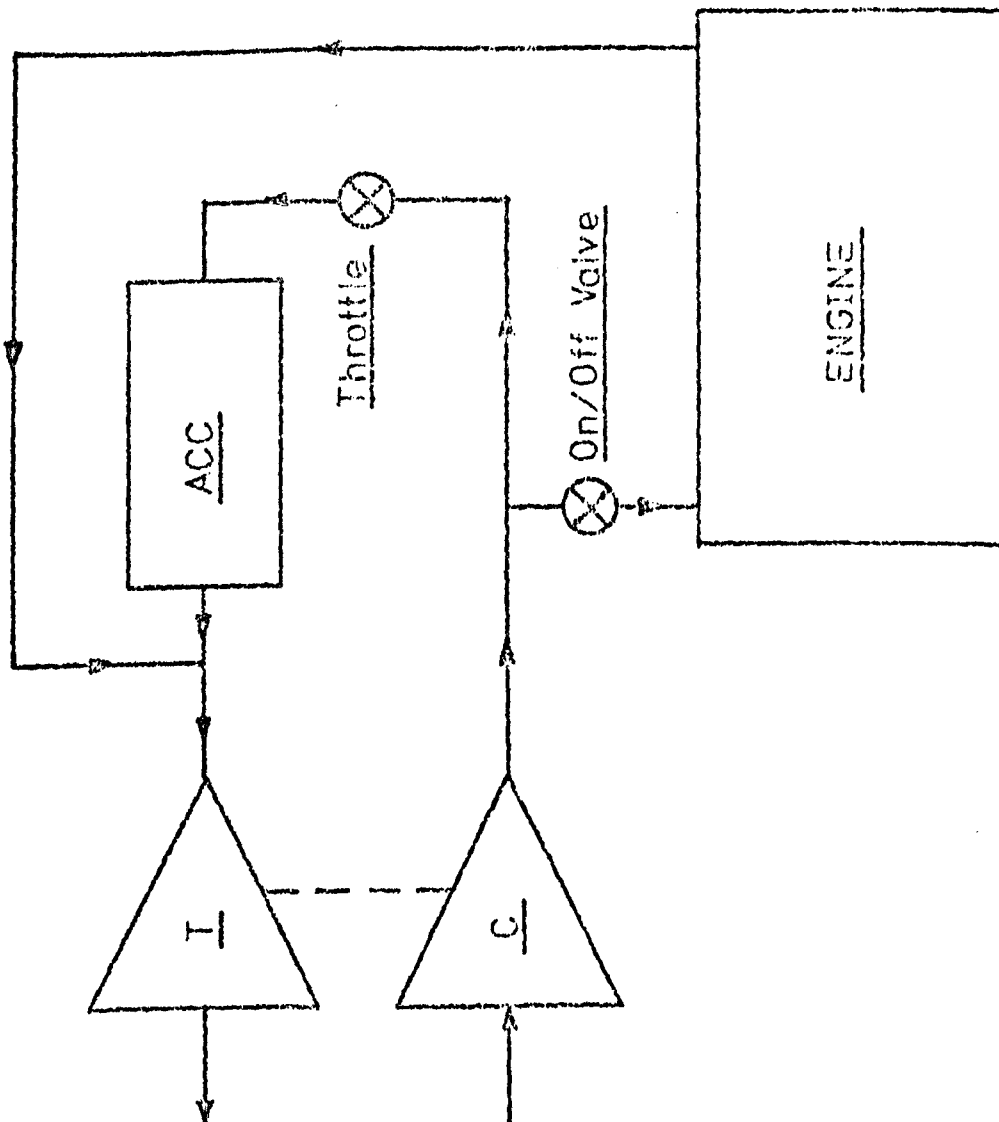
Type 3D215
Bore 3.375 in (85.7 mm)
Stroke 4.00 in (101.6 mm)
Effective Compression Ratio 16:1
Capacity 215 cu.in. (3520 cc)
B.H.P. 125 (93.2 kw) at 2400 rev/min
No. of cylinders - three
Firing order 1,2,3
Max. torque 313 lb.ft (425 Nm) at 1200 rev/min
Blower make Wade 5R034 No.2679
Type 'Roots' with two 3 lobe rotors.

TABLE 2

	Present System	Hyperbar System
Boost Ratio	2.3 : 1	8 : 1*
Compression Ratio	16 : 1 ⁺	5 : 1
Engine	2 stroke	4 stroke
Minimum Air/Fuel Ratio	22 : 1 (proposed)	30-35 : 1
Auxiliary Combustion Chamber	Bypass clean air supply	Engine exhaust supply

* The hyperbar system utilises two stage turbocharging.

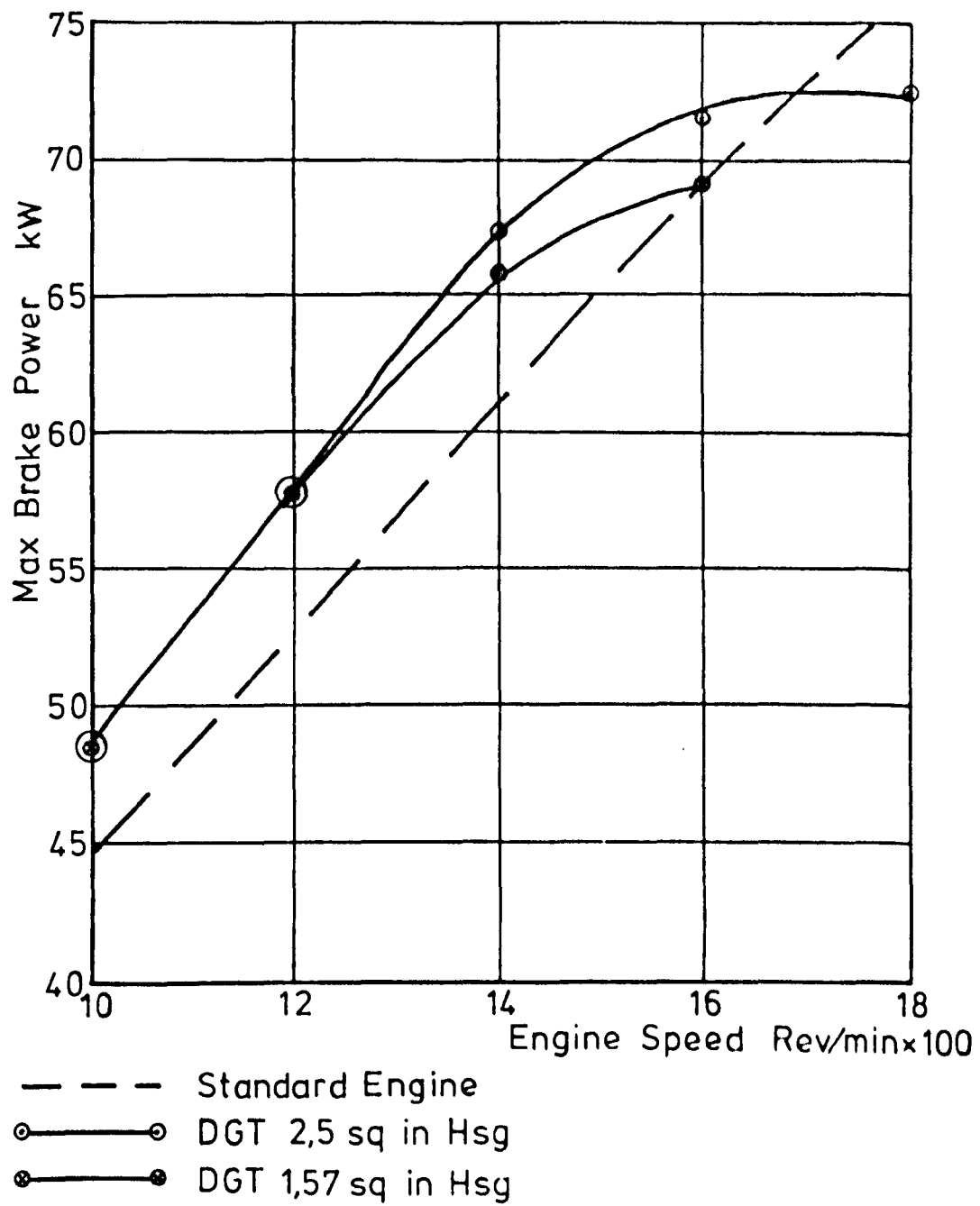
+ Standard engine compression ratio.



- C Turbo Compressor
- ACC Auxiliary Combustion Chamber
- T Turbine
- Mechanical Link
- Thermodynamic Link

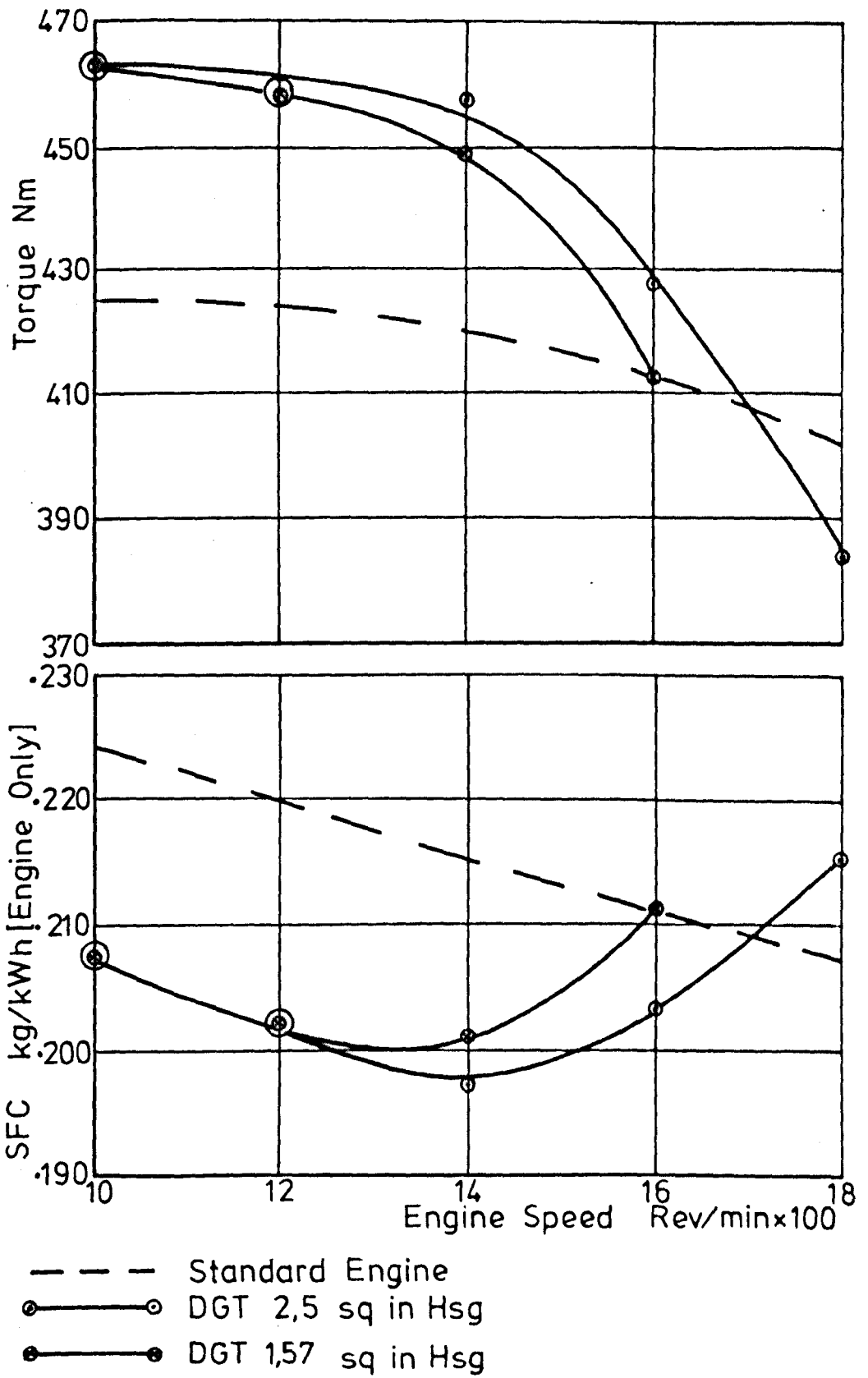
DIESEL GAS TURBINE (DGT) SYSTEM

FIGURE 2



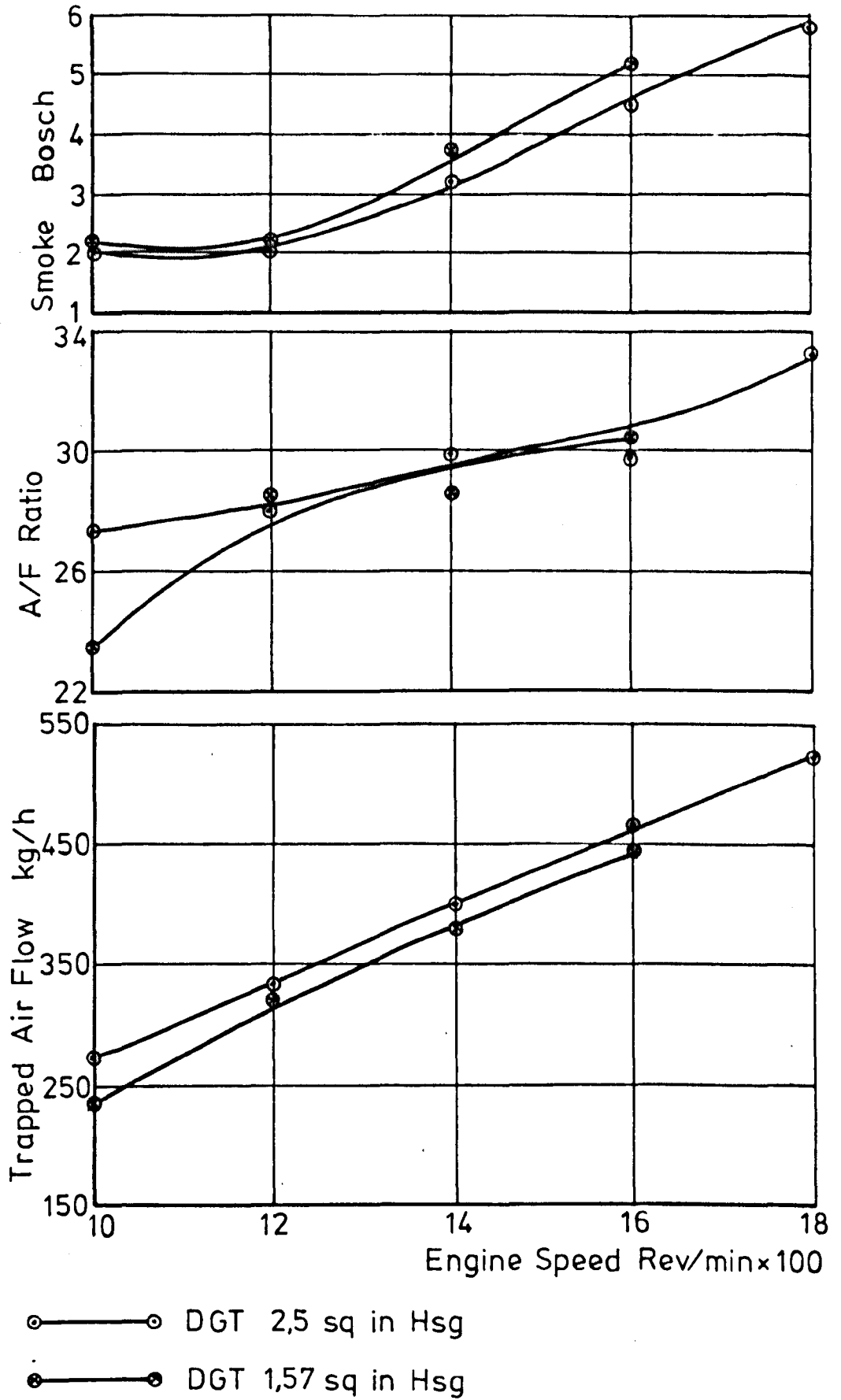
DIESEL GAS TURBINE [DGT] PERFORMANCE

FIGURE 3



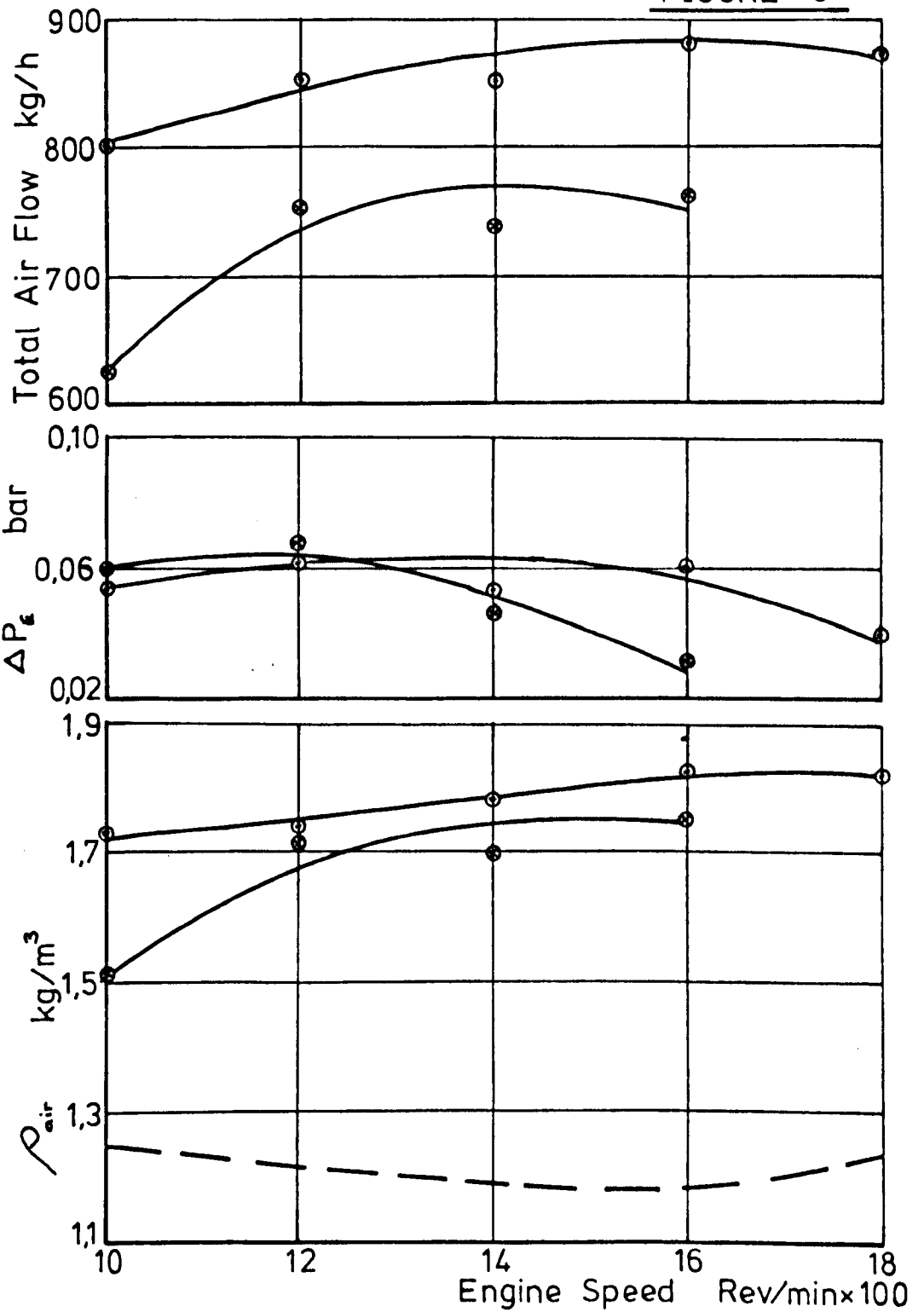
DIESEL GAS TURBINE [DGT] PERFORMANCE

FIGURE 4



DIESEL GAS TURBINE [DGT] PERFORMANCE

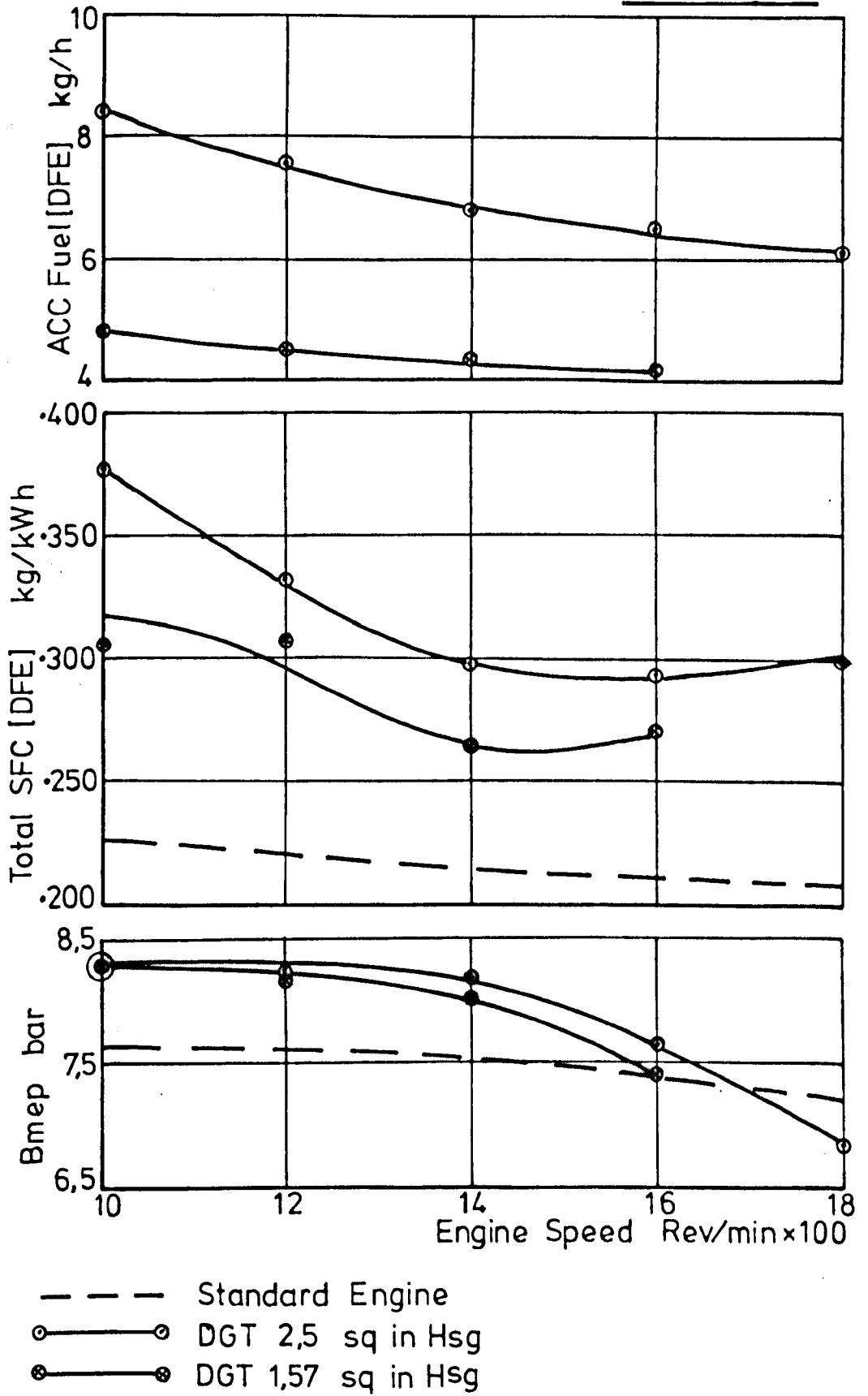
FIGURE 5



- Standard Engine
- DGT 2,5 sq in Hsg
- DGT 1,57 sq in Hsg

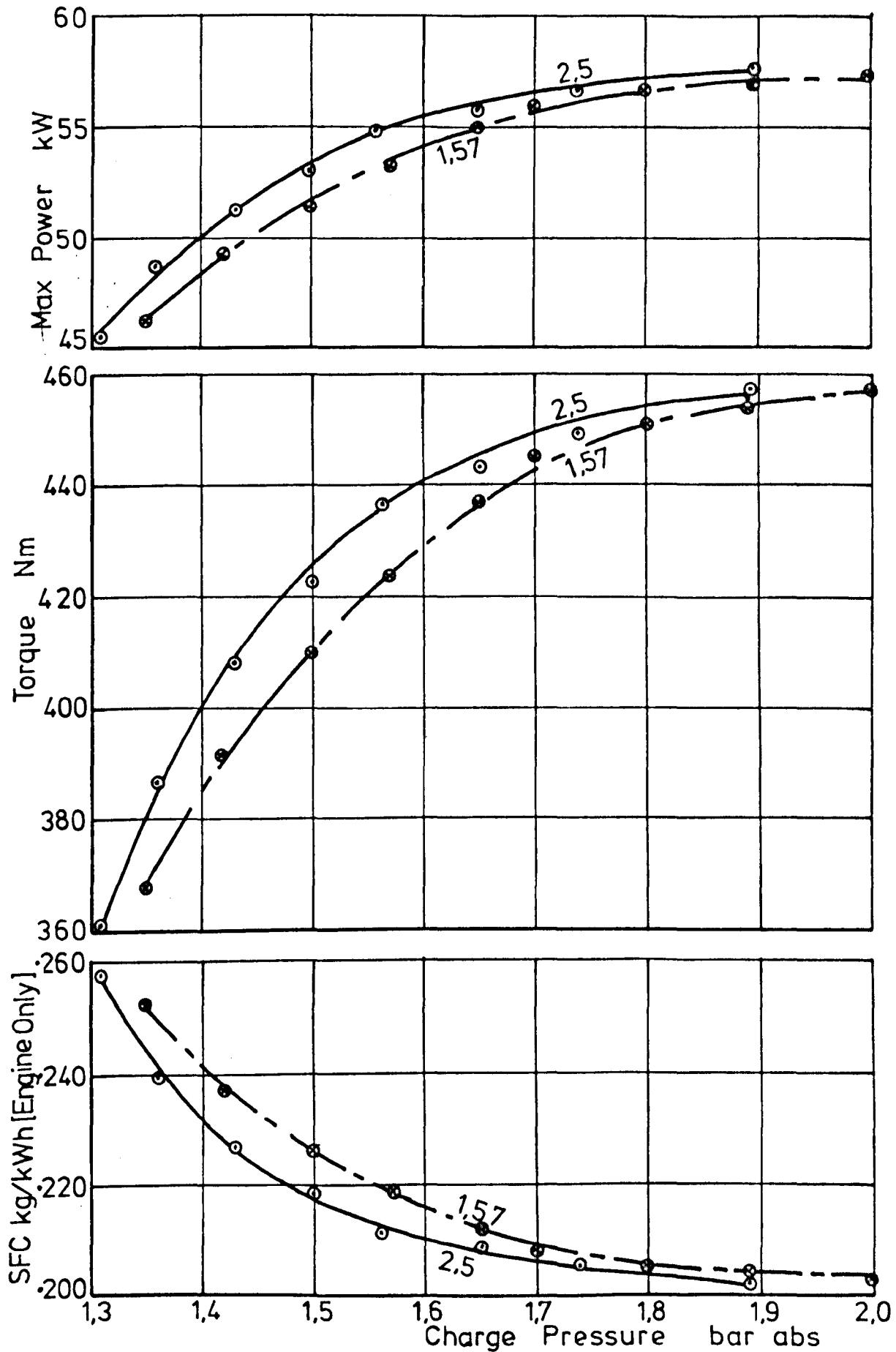
DIESEL GAS TURBINE [DGT] PERFORMANCE

FIGURE 6



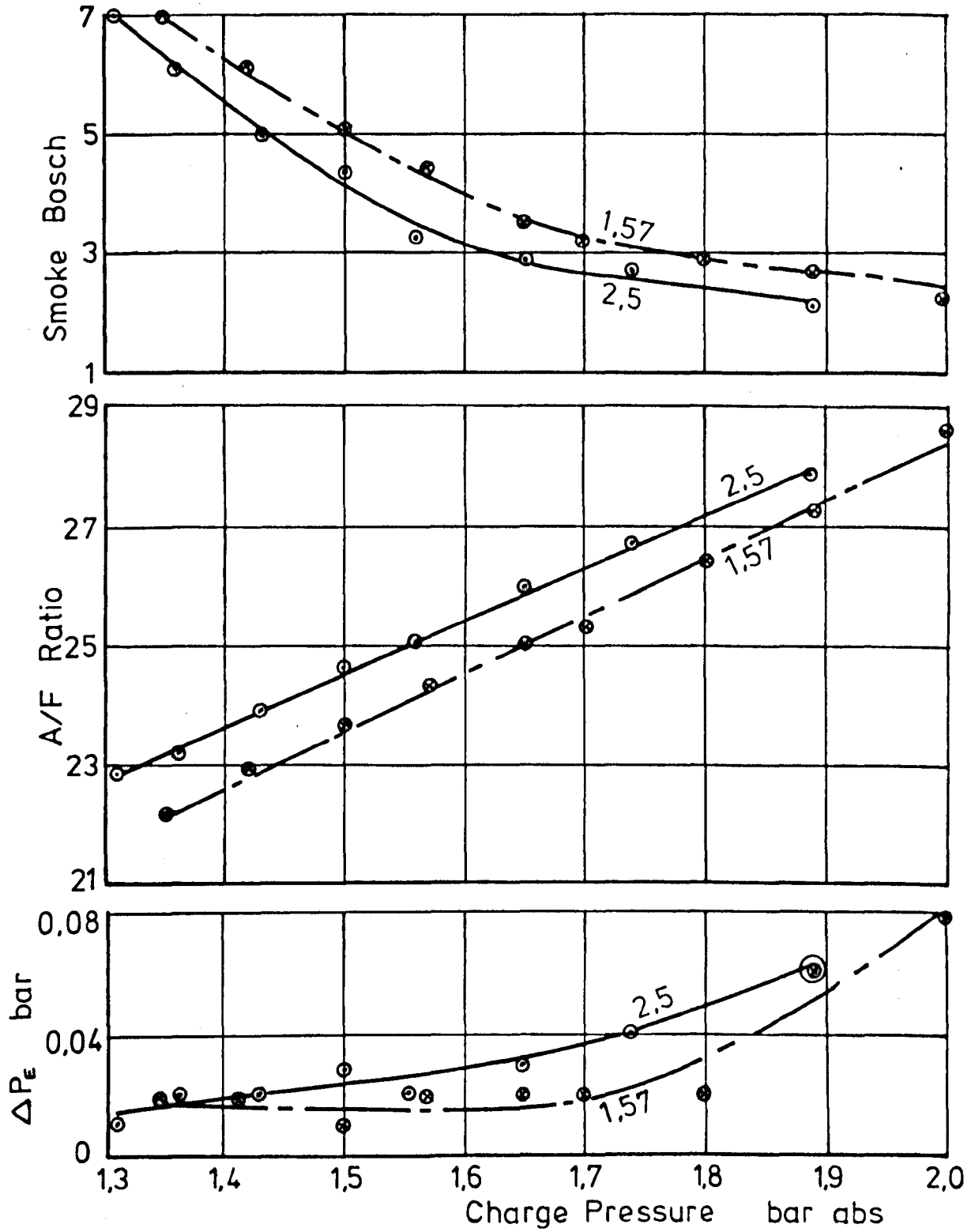
DIESEL GAS TURBINE [DGT] PERFORMANCE

FIGURE 7



DIESEL GAS TURBINE [DGT] PERFORMANCE
[Engine Speed 1200 Rev/min]

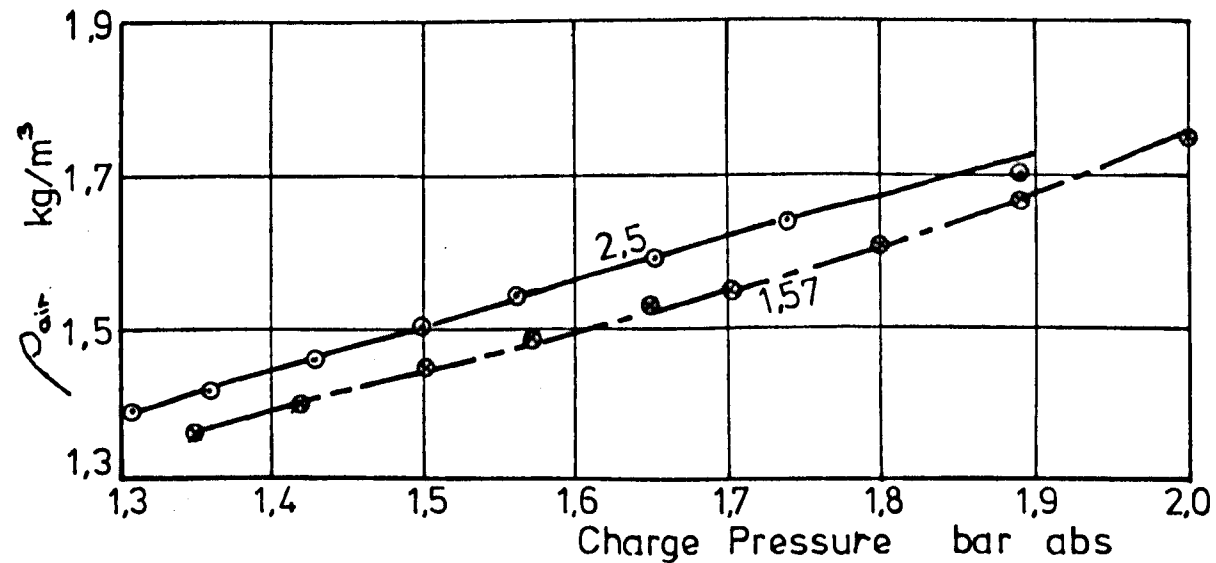
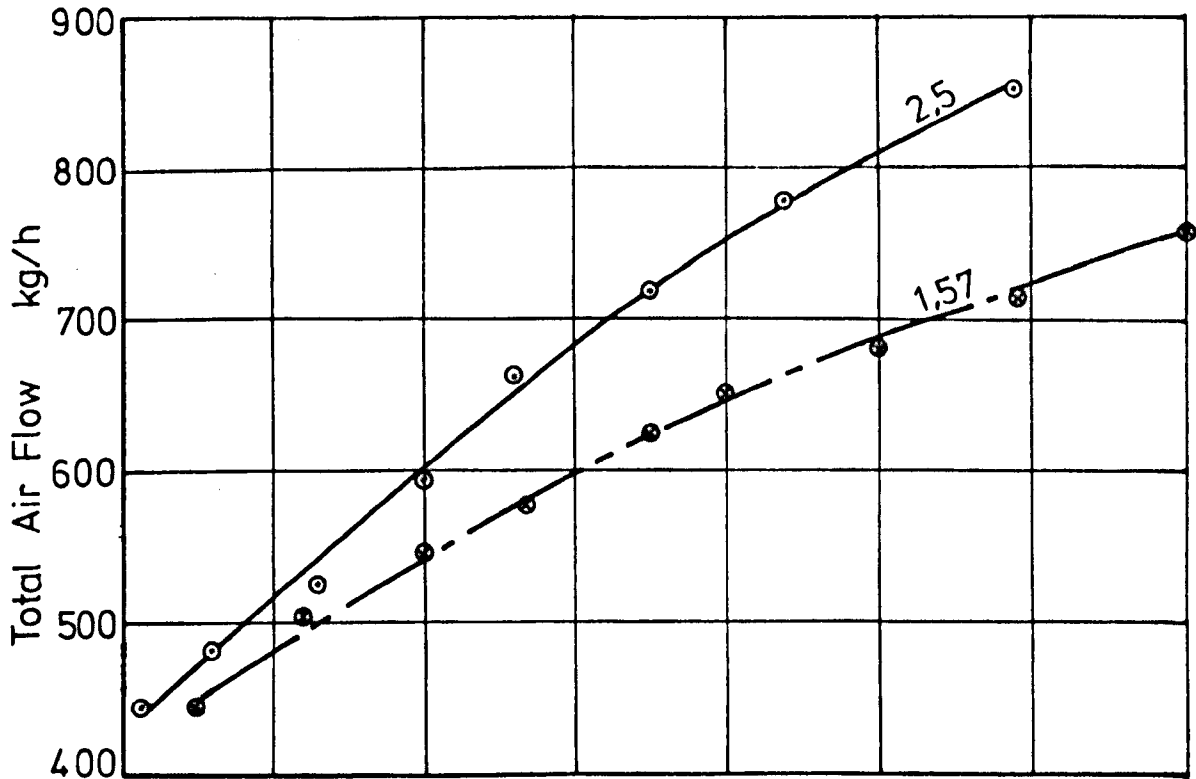
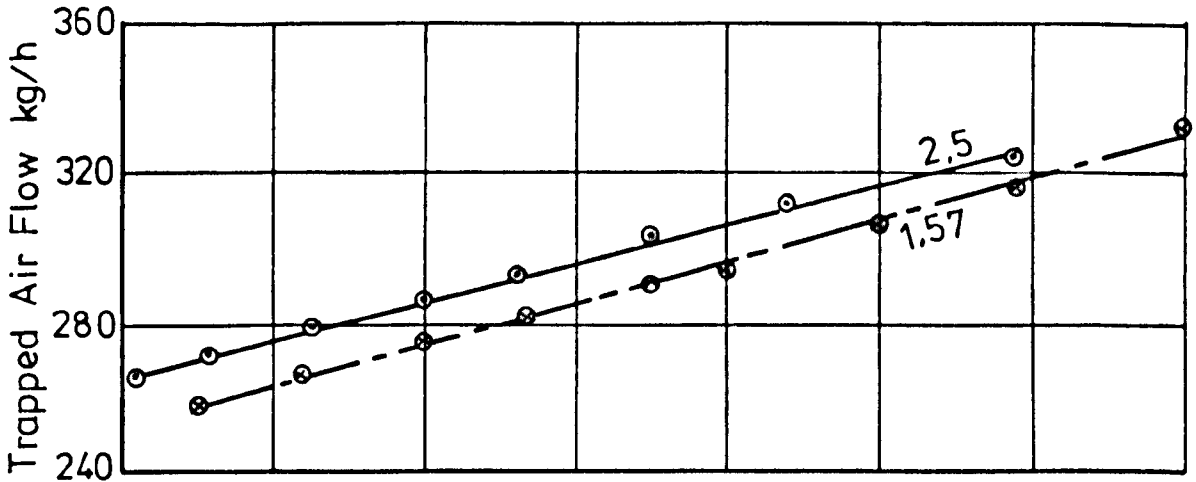
FIGURE 8



DIESEL GAS TURBINE [DGT] PERFORMANCE

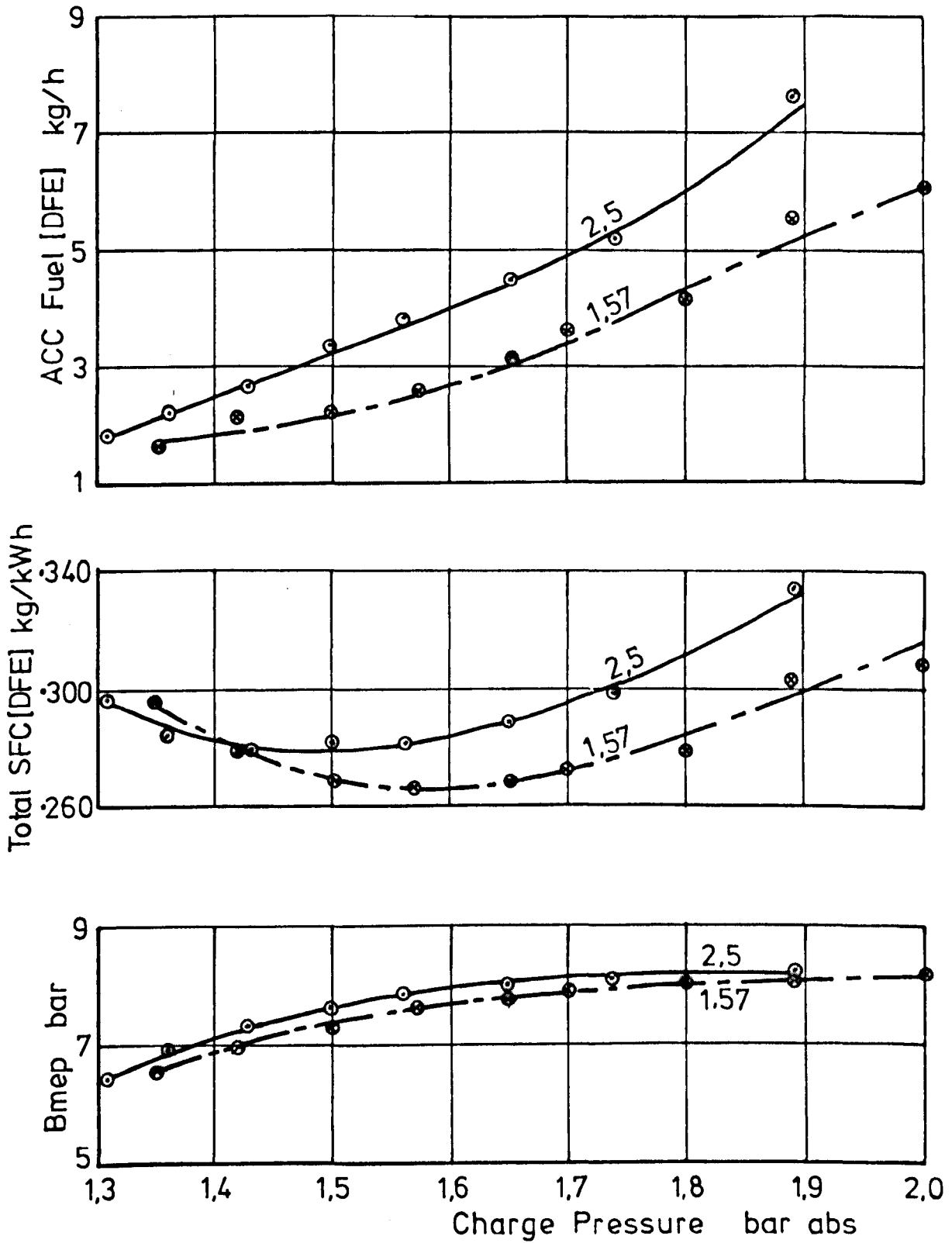
[Engine Speed 1200 Rev/min]

FIGURE 9



DIESEL GAS TURBINE [DGT] PERFORMANCE
[Engine Speed 1200 Rev/min]

FIGURE 10



DIESEL GAS TURBINE [DGT] PERFORMANCE

[Engine Speed 1200 Rev/min]

APPENDIX 7

THE TWO STROKE DIESEL GAS TURBINE
PROMISES AND PROBLEMS

R. Farkin B.Sc.

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Symposium on Thermodynamics and Heat
Transfer. November 1979, Centre Hotel,
Leicester.

THE TWO STROKE DIESEL GAS TURBINE
PROMISES AND PROBLEMS

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ABSTRACT

In this paper the performance of a Diesel Gas Turbine (D.G.T.) is investigated. The reasons for choosing a two-stroke operating system are outlined, and differences from the Hyperbar system are discussed.

The D.G.T. is shown to be a viable automotive traction package with good torque 'back up', improved power output, and good acceleration prospects. Some problems remain to be solved before the system would be commercially viable. These include inadequate scavenge air at high engine speeds and high specific fuel consumption. Development of a more sophisticated control system is necessary before the D.G.T. could be used by an operator.

INTRODUCTION

There is an ever increasing pressure in automotive circles to increase engine power output for a given bulk. One method used is exhaust gas turbocharging and this is now a common feature in automotive four-stroke cycle diesel engines. The Comprex system of Pressure Wave Supercharging (P.W.S.) is also receiving much attention (1,2).

The picture is different for two-stroke cycle engines. The two-stroke would be a natural choice for a high output/bulk ratio as it has twice the number of working strokes, hence, potentially double the power output for a given bulk. Large two-stroke cycle engines working over restricted speed range (e.g. shipping, power generation) have been successfully turbocharged. The automotive engine is much smaller, and must work over a wide range of speed and load for maximum flexibility.

The automotive two-stroke is not amenable to either classical turbocharging (3) or P.W.S. alone(4). The engine has peculiar problems in that it needs a positive pressure drop across the engine to provide adequate scavenge. This usually implies some form of crankshaft driven supercharger. Thus, turbocharging of automotive two-strokes has been attempted using a scavenge blower in series with the turbocharger (5).

Acceleration of a turbocharged engine is usually not as good as the acceleration of a corresponding naturally aspirated engine. This is due to the necessity of first accelerating the turbocharger to provide the necessary extra air in order to allow increased engine fuelling. This cannot be achieved without an increase in engine fuelling rates to provide more heat energy at the turbine inlet. The net effect is an 'acceleration lag' and increased exhaust smoke emission. Attempts have been made to overcome this by the use of compressed air injection (6), or by the use of Pelton Wheel/hydraulic oil assistance (7). Both these modifications add to the bulk and complexity of the engine system.

This paper describes another approach :- the diesel gas turbine.

SYSTEM DETAILS

The engine used was a Rootes TS3, horizontally opposed, two-stroke, direct injection, diesel engine. As the engine may be considered as part of an overall traction system, its choice is only constrained, for reasons already discussed, in that it should be a two-stroke. A four-stroke engine based system would also be viable.

This particular engine was chosen because its characteristics and limitations have been well established, and it has proved amenable to increased power output (8,9). Such an engine was already available to the author. Engine specifications are given in Table 1.

The engine was used in conjunction with a turbocharger and an Auxiliary Combustion Chamber (A.C.C.), designed to increase exhaust gas enthalpy at turbine inlet, and associated ducting. The Roots type scavenge blower fitted to the standard engine is removed and replaced with a blanking plate and a fabricated air manifold.

The air from the turbocharger is divided between the engine and the auxiliary combustion chamber. The engine supply incorporates a butterfly on/off valve for the purpose of isolating the engine during system start up. The air bypass to the A.C.C. includes a throttle to allow the engine pressure drop to be varied to permit adequate scavenge.

Engine exhaust gas is mixed with the hot gases from the A.C.C. just prior to the turbine inlet (shown schematically in Figure 1). The A.C.C. is fuelled with propane from a bottled gas supply. Propane ignition is achieved using an automotive spark plug, a proprietary automobile Capacitive Discharge ignition unit in conjunction with an automotive ignition coil and a trigger circuit. The ignition system gives a spark frequency of ≈ 10 Hz.

The D.G.T. described allows for system start up by employing the turbocharger and the A.C.C. as a gas turbine. When the turbine is operating in a self-sustained mode the engine is started in the normal manner.

ADVANTAGES OF THE D.G.T.

The D.G.T. system allows a two-stroke engine to be started and run without the use of a crankshaft driven supercharger. Acceleration of the engine is not a problem as the engine can take air from the compressor as required. A demand for more air reduces the bypass flow through the A.C.C. This will result in an increase in turbine inlet temperature and accelerate the turbocharger.

As the turbocharger is free running (i.e. not fully restricted by the volumetric nature of the engine) due to the parallel turbocharging system, the provision of air is governed by the excellent acceleration of the gas turbine. This feature should improve transient performance and reduce smoke emission. The major differences between the D.G.T. and the reported Hyperbar installations (10, 11) are detailed in Table 2.

AUXILIARY COMBUSTION

In the Hyperbar system, auxiliary combustion is undertaken using a mixture of engine exhaust gases and clean bypass air. Accordingly, some level of exhaust pulsation will be present. This may lead to flame instability, and satisfactory operation may only be possible with high air/fuel ratios. The effect would be minimised by the use of engines having a relatively large number of cylinders.

A similar auxiliary combustion arrangement was investigated on the D.G.T. and flame instability was found to result at loads in excess of 2/3 of the expected maximum. This was due to the increasing level of exhaust pulsation as loading was increased. The effect is exacerbated by the relatively small number of cylinders.

The flame instability problem was overcome by adopting an auxiliary combustion system which utilised only clean, constant pressure, bypass air. The engine exhaust was mixed with hot gases from the A.C.C. just prior to the turbine inlet.

The Hyperbar system may benefit from its auxiliary combustion system in that engine exhaust smoke is subject to a reburning process, thus reducing hydrocarbon and CO emissions. The D.G.T. system does not have this facility, although the very hot auxiliary gases may have some beneficial effect on engine exhaust emissions, when the two flows are mixed just after the A.C.C.

EXPERIMENTAL PROGRAMME

Test rig instrumentation was collected and modified (12, 13). The standard engine was tested over the full range of rated engine speed and a range of loads. This gave a 'base line' by which the diesel gas turbine could be evaluated. The test rig was then subjected to several stages of modification and development, culminating in the diesel gas turbine detailed in this paper (shown in Figure 2). The start up procedure of the D.G.T. is as follows :-

- 1) The butterfly valve for the air supply is closed, thus isolating the engine from the system.
- 2) The air bypass valve is fully opened. The system is effectively a gas turbine.
- 3) A slave oil supply is connected to the turbocharger.
- 4) The A.C.C. igniter is activated.
- 5) A slave air supply is connected to the A.C.C. and fuelling is initiated.
- 6) When the flame has stabilised (viewed through a quartz observation window in the A.C.C.), the slave air flow and fuelling rate are increased until the gas turbine is self-sustained. These conditions are normally reached within 10-15 secs. from first fuelling.
- 7) The slave air supply is disconnected and the engine air supply butterfly valve opened.
- 8) The air bypass throttling valve and A.C.C. fuelling are adjusted until there is a boost ratio of 1.5 and a pressure drop across the engine of 0.01 bar.
- 9) The engine is then started in the conventional manner.
- 10) The slave oil pump is turned off, the supply of turbocharger oil being taken over by the engine lubricating oil circuit.

Load may then be applied to the engine. The diesel gas turbine was tested over the full range of engine speed. Only full load tests, within the limitations of the injector pump delivery rate, were carried out. A set of tests were also conducted at a constant engine speed of 1200 rev./min., to assess the effect of varying charge inlet pressure. This was varied by adjusting the fuelling rate to the A.C.C.

RESULTS

The results of the standard engine tests are given in Figures 3-7. Curves obtained from a computer program are also shown. The computer program was matched to the experimental data for later use as a subsection of a D.G.T. performance prediction program.

The results from the D.G.T. tests are shown in Figures 8-10. The standard engine curves are superimposed for ease of comparison. The results obtained from the tests conducted at a constant engine speed are shown in Figures 11, 12.

A computer program to predict D.G.T. performance and exhaust pulse energy is being developed, and the results will be presented at a later date.

DISCUSSION

From the results shown it can be seen that the D.G.T. displays useful improvements in 'back up' torque and power output at speeds up to 2000 Rev/min. These improvements are obtained with an air/fuel ratio of 30 (standard engine air/fuel ratio 20). As such it holds great promise as a traction package. However, there are problems to be solved. The smoke curves show that the system has inadequate scavenge air at speeds of 1800 Rev/min and above, and this is reflected in the power output.

As scavenge efficiency decreases with engine speed, the necessary pressure drop across the engine must increase with engine speed. From the constant speed tests it is apparent that scavenge efficiency increases with increasing boost.

The D.G.T. displays scavenge problems at speeds in excess of 1800 Rev/min; thus the cylinders are not adequately purged and power output falls due to inefficient combustion. It was not possible to alleviate these inadequacies by further throttling of the bypass air. This would suggest that the pressure drop, achieved by the engine acting as an orifice, is not sufficient for acceptable scavenge at these speeds and boost levels.

One solution would be to substantially increase the boost level, which would result in a lowering of the compression ratio, and also the complexity of two stage turbocharging. Experience suggests that a solution could be achieved by using a turbine with a larger throat area, thus decreasing back pressure on the engine. This course of action would require increased fuelling rates to the A.C.C.

The high specific fuel consumption could be improved by using a 'total energy' design concept. As the D.G.T. is presently operating at air/fuel ratios of the order of 30, the specific fuel consumption could be improved by increasing engine fuelling rates. It could be further improved by fitting a charge cooler to the engine air supply circuit, and further increasing the engine fuelling rate. This would not improve the part load specific fuel consumption: this, and the full load specific fuel consumption, could be improved by utilising a heat exchanger to preheat the A.C.C. inlet air. The turbine exhaust could provide the heat source for such an exercise.

For use in automotive traction the D.G.T. would need an effective control system, to enable an operator to use it. The basics of such a system have been designed, and involve the use of a micro-processor and stepper motors. A substantial development programme would be necessary before the control system could be operational.

CONCLUSION

The D.G.T. constitutes a viable traction package, yielding substantial improvements in power output and 'back up' torque. However, certain problems remain to be solved. These include a suitable control system and measures to improve high speed scavenge. Steps must also be taken to reduce the specific fuel consumption.

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TABLE 1

Rootes TS3

Two-stroke, horizontal three cylinder with opposed pistons, direct injection diesel engine.

Type 3D215

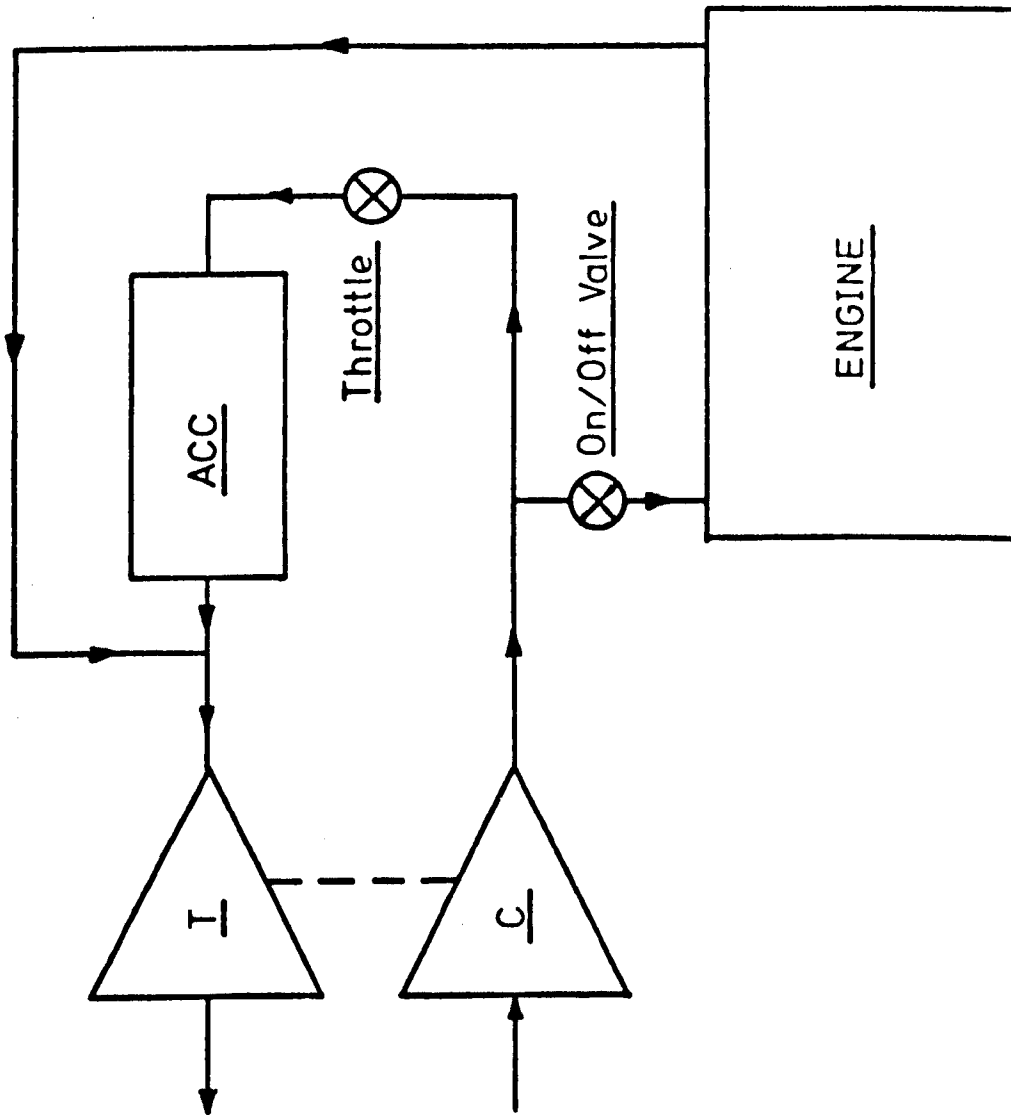
Bore 3.375in. (85.7mm)
Stroke 4.00 in. (101.6mm)
Effective Compression Ratio 16:1
Capacity 215 cu.in. (3520cc.)
B.H.P. 125 (93.2kw) at 2400 rev/min.
No. of cylinders Three
Firing order 1,2,3
Max.Torque 313lb ft (425 Nm) at 1200 rev/min.
Blower Make Wade 5R034 No. 2679
Type Roots with two 3 lobe rotors

TABLE 2

	D.G.T.	Hyperbar System
Boost Ratio	2.3 :1	9 :1*
Compression Ratio	16 :1 [†]	5 :1
Minimum Air/Fuel Ratio	22 :1 (proposed)	30-35 :1
Auxiliary Combustion	Clean constant pressure air	Pulsating engine exhaust with constant pressure air

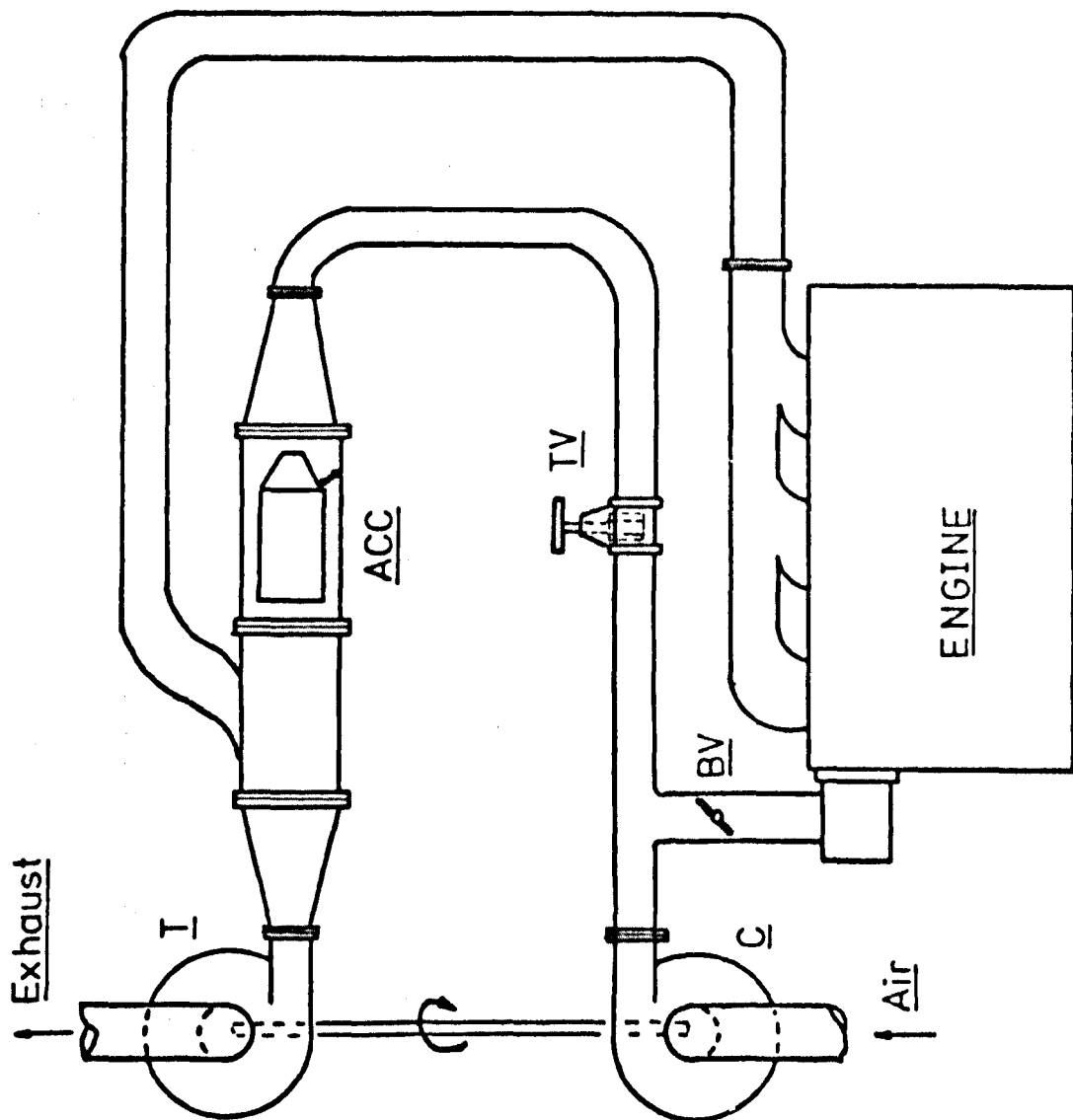
* The Hyperbar system utilises two stage turbocharging

† Standard engine compression ratio



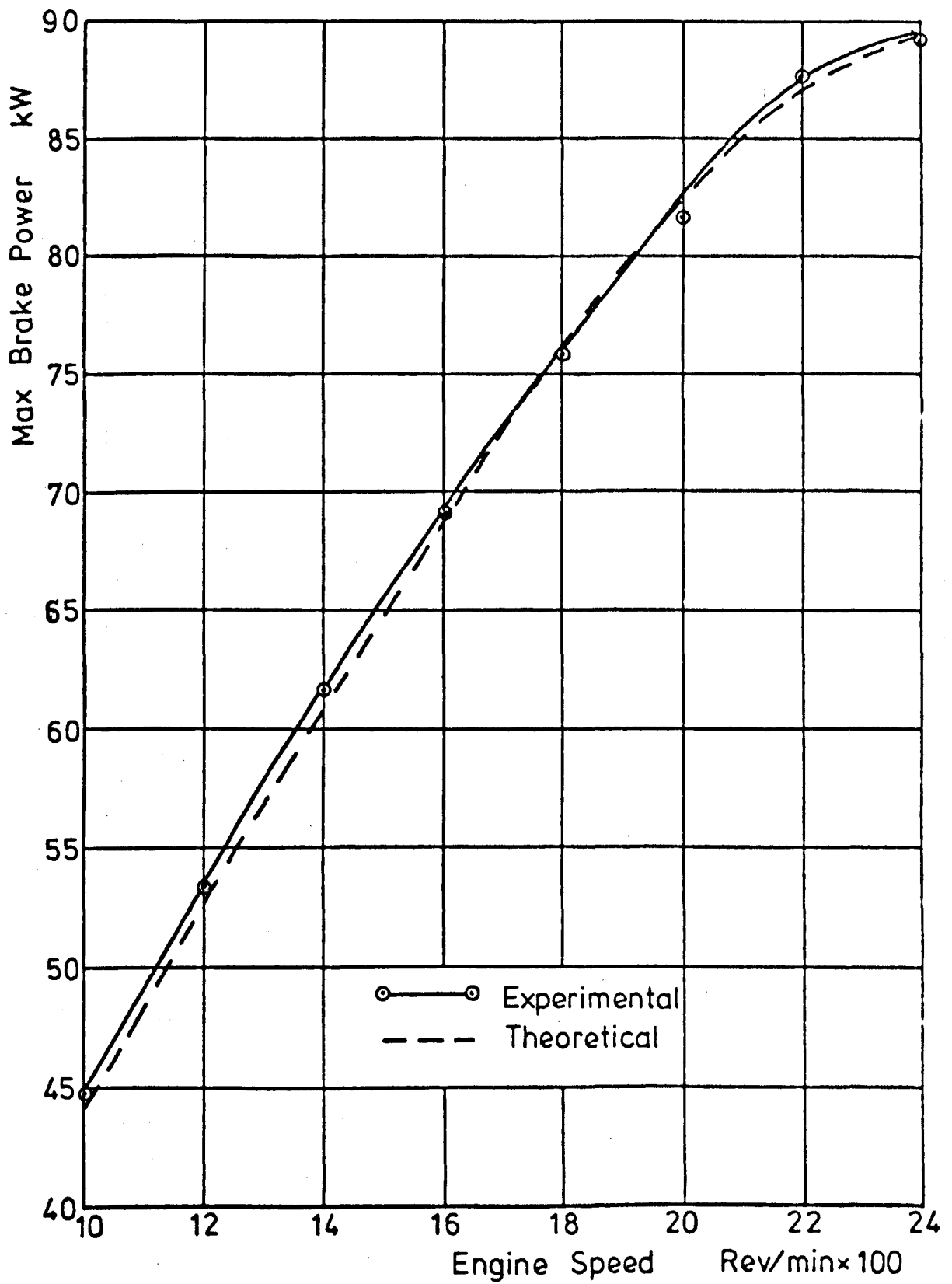
- C Turbo Compressor
- ACC Auxiliary Combustion Chamber
- T Turbine
- Mechanical Link
- Thermodynamic Link

DIESEL GAS TURBINE [DGT] SYSTEM (FIG1)

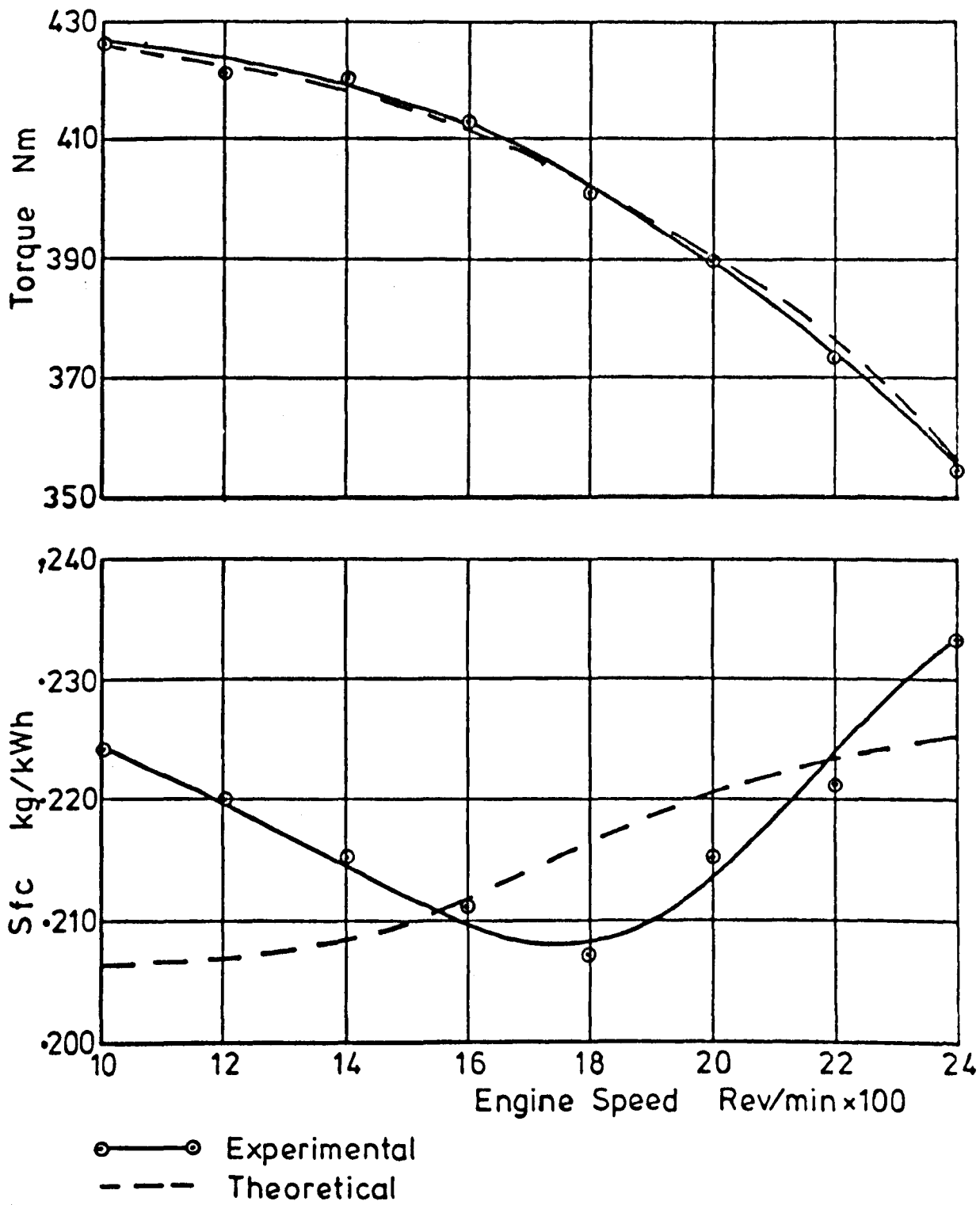


- C Compressor
- T Turbine
- ACC Auxiliary Combustion Chamber
- BV Butterfly Valve
- TV Throttling Valve

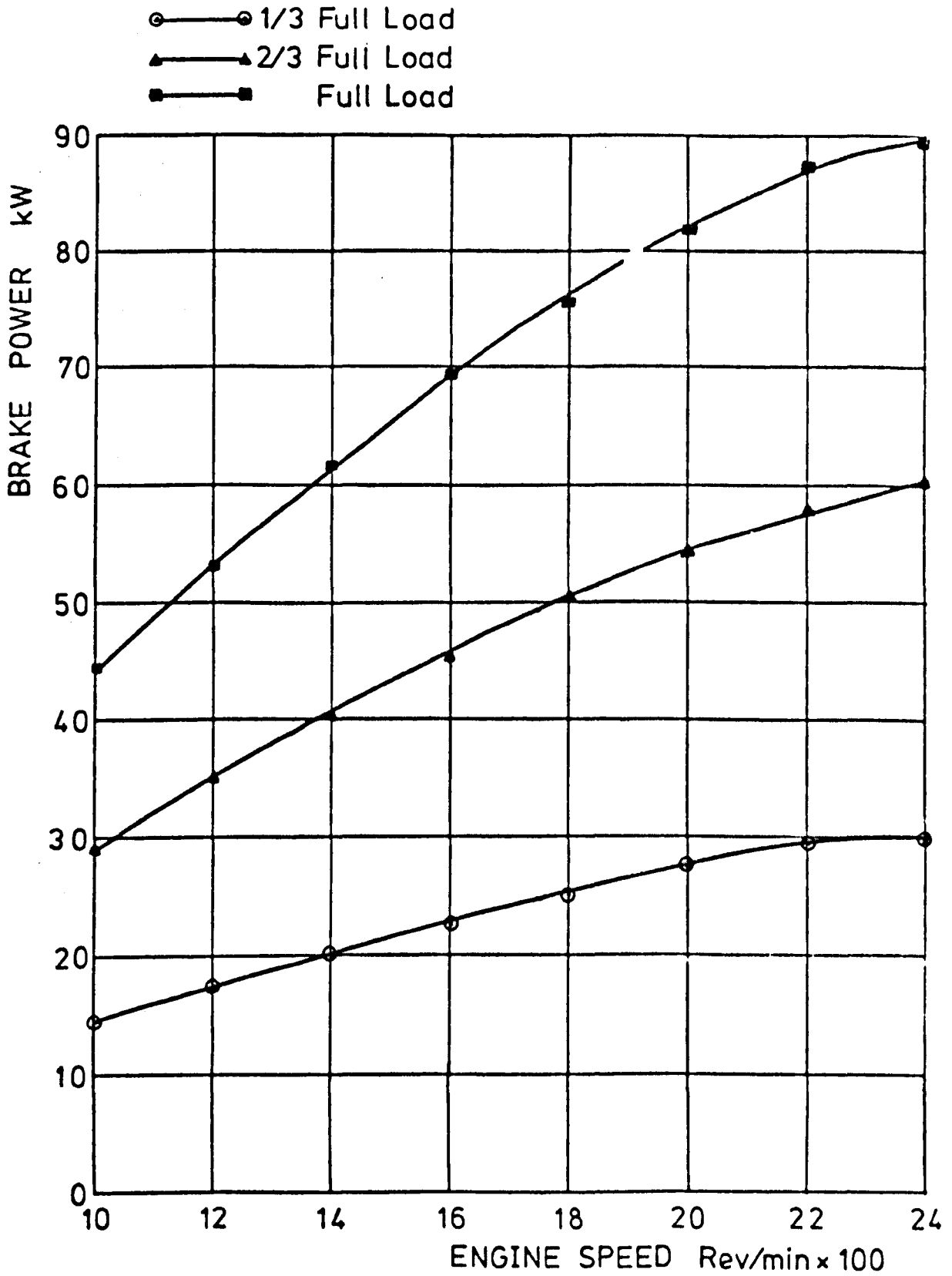
DETAIL OF DGT SYSTEM (FIG 2)



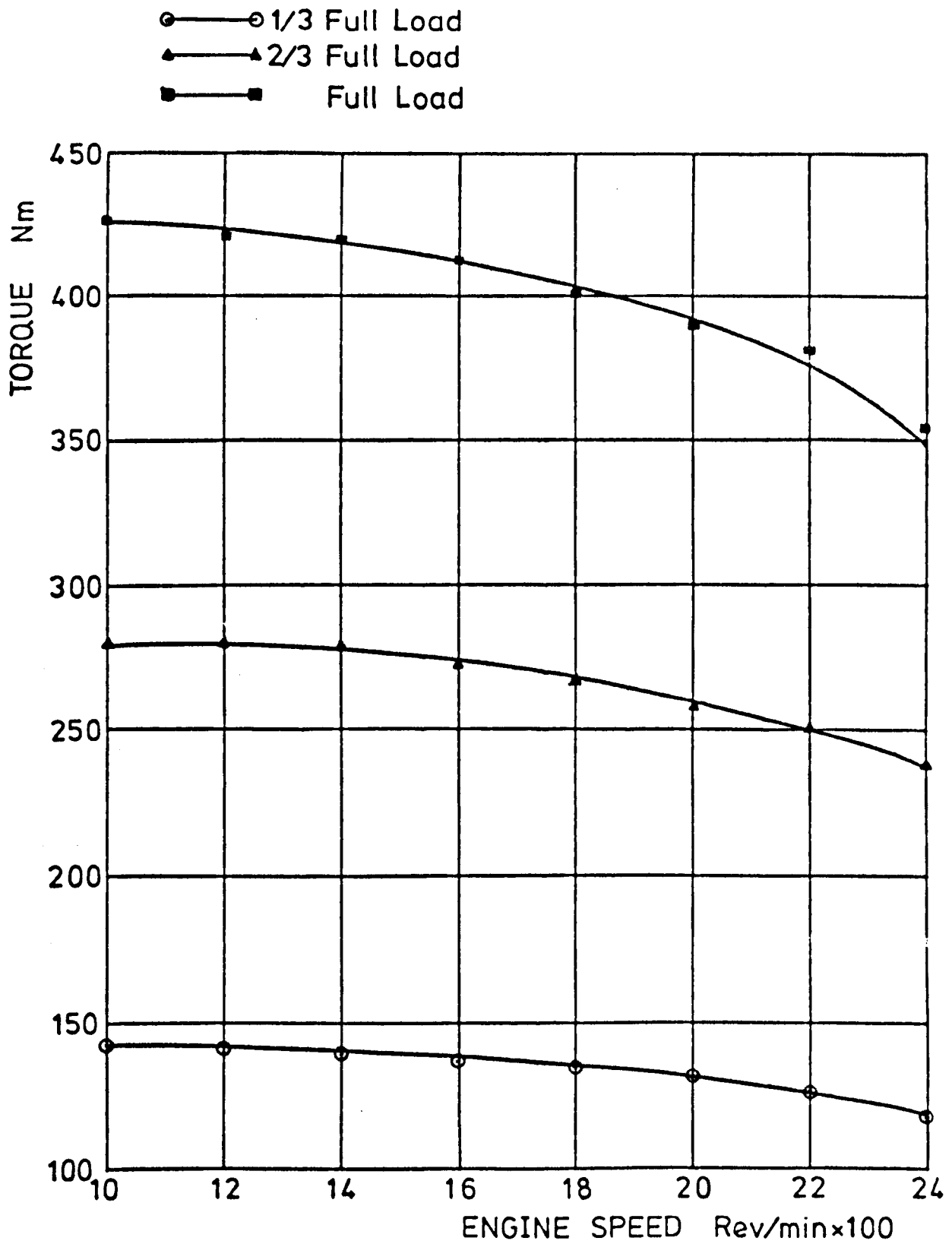
STANDARD ENGINE PERFORMANCE (FIG 3)



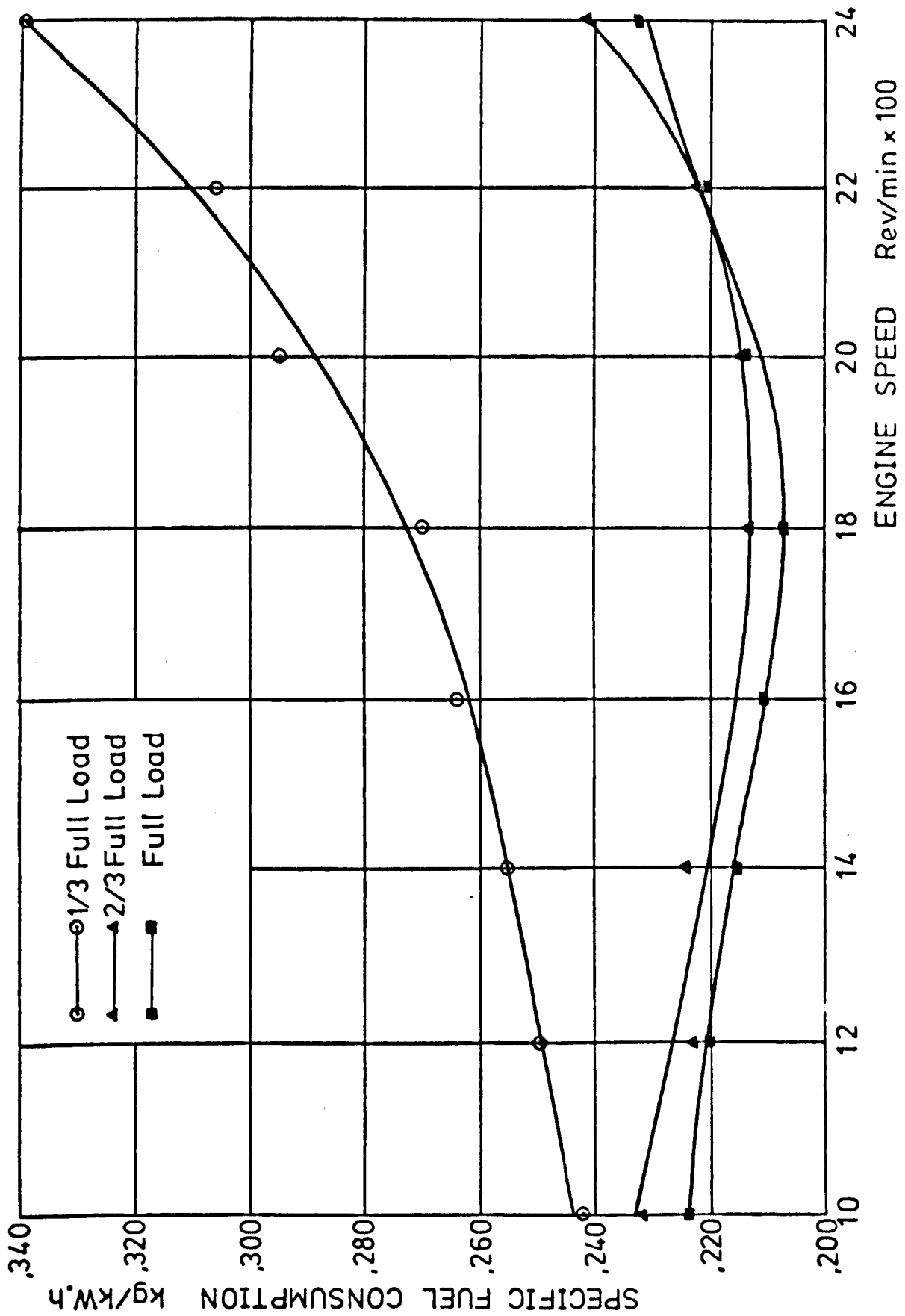
STANDARD ENGINE PERFORMANCE (FIG 4)



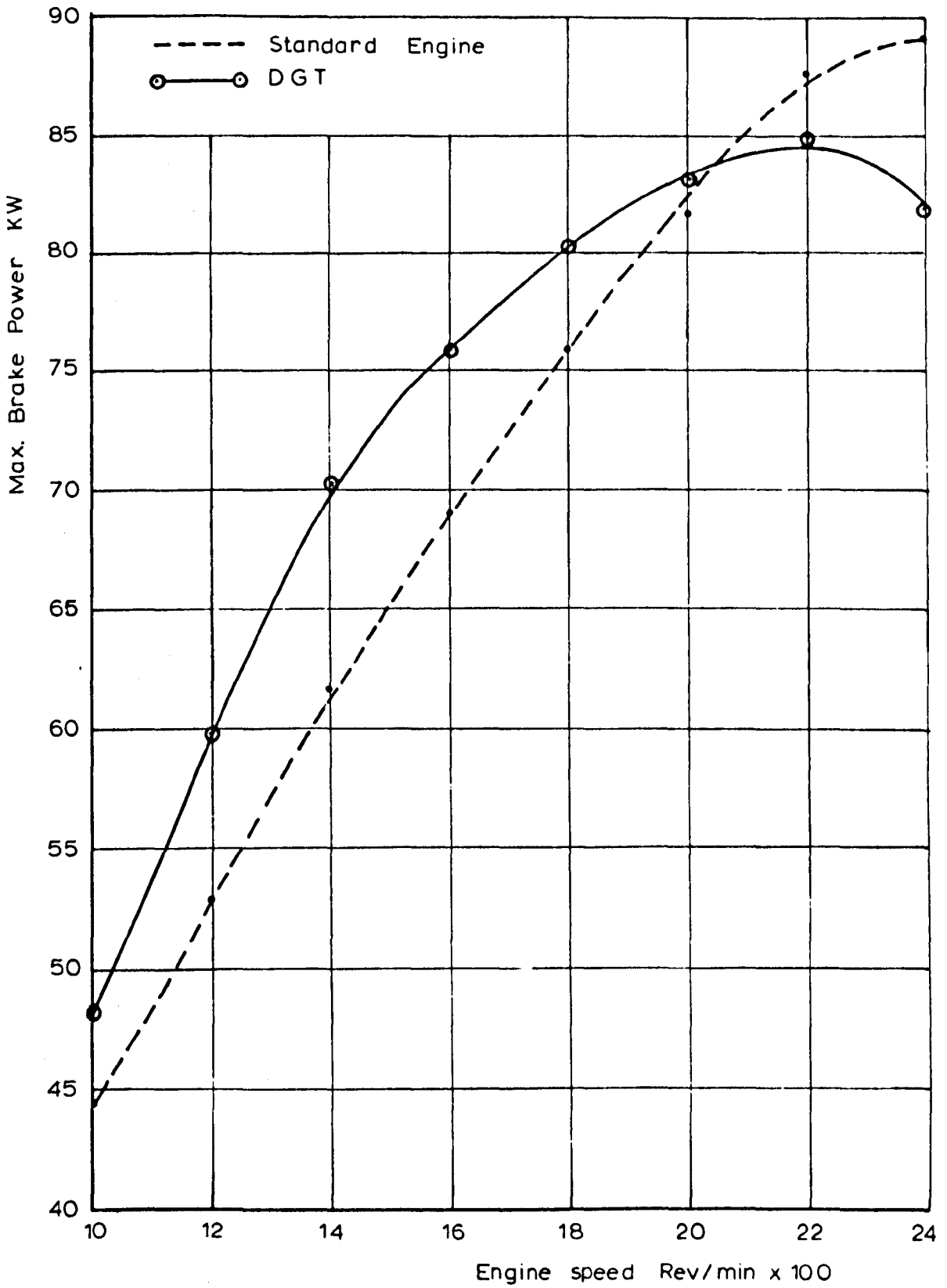
STANDARD ENGINE PERFORMANCE (FIG 5)



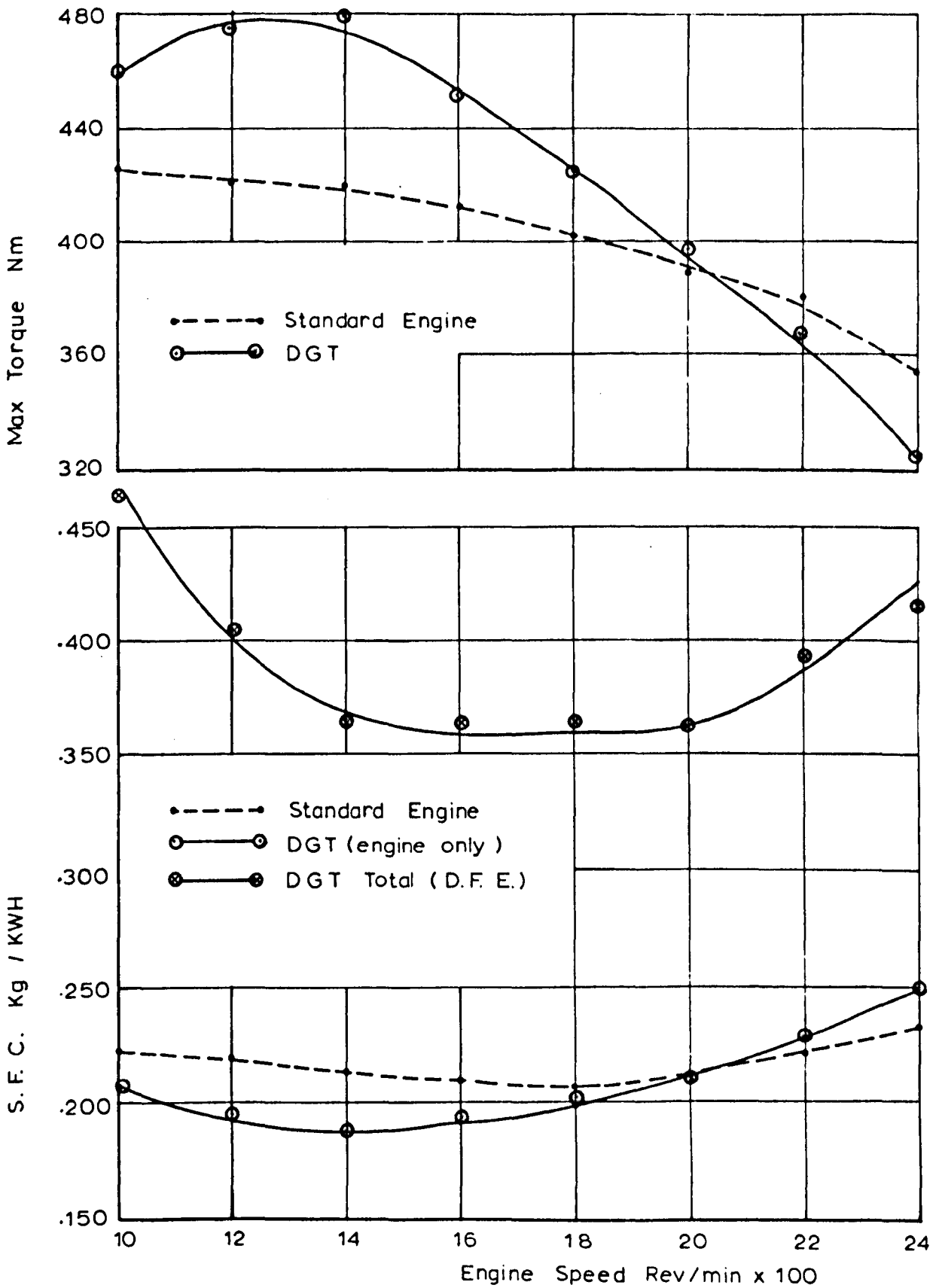
STANDARD ENGINE PERFORMANCE (FIG 6)



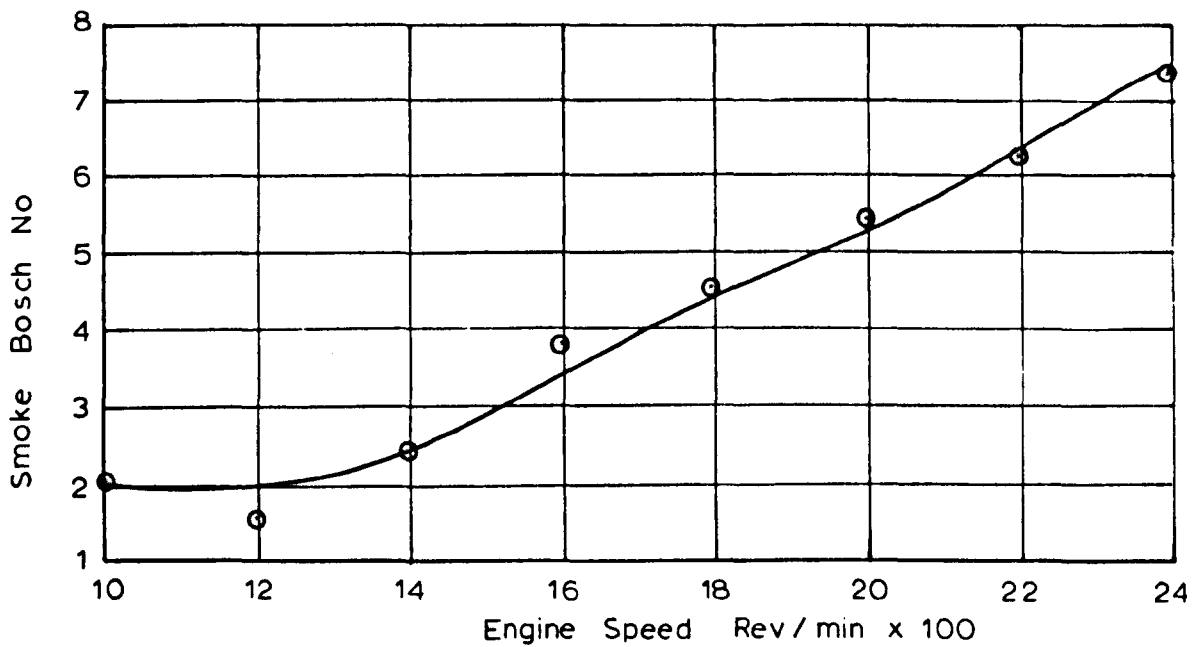
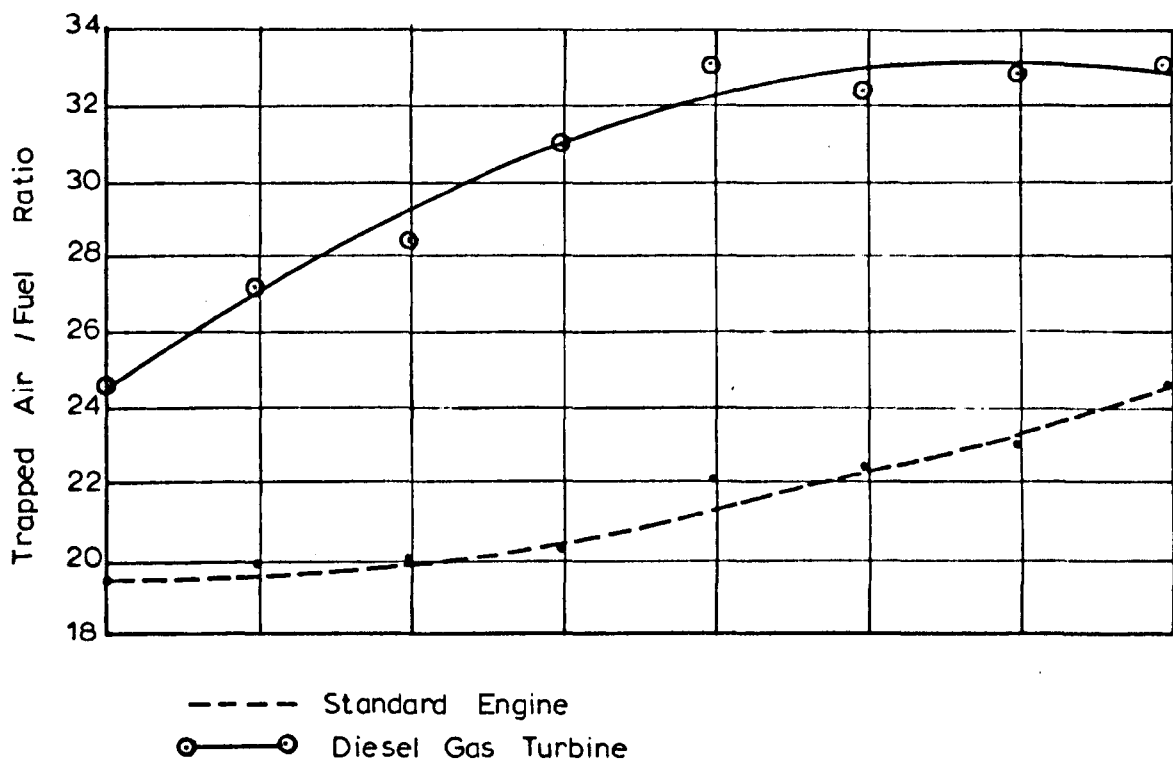
STANDARD ENGINE PERFORMANCE (FIG 7)



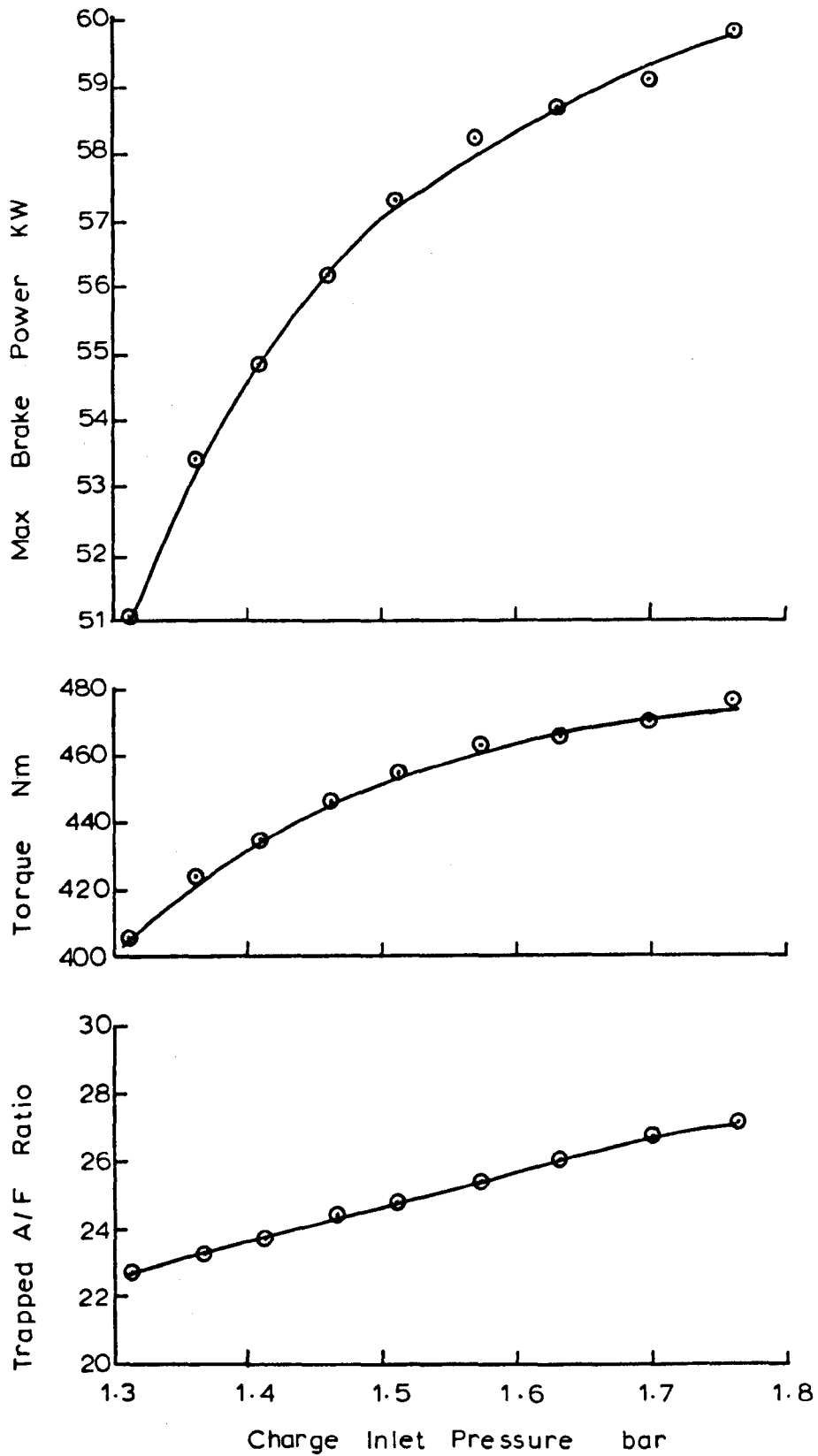
DIESEL GAS TURBINE PERFORMANCE (FIG. 8)



DIESEL GAS TURBINE PERFORMANCE (FIG. 9)

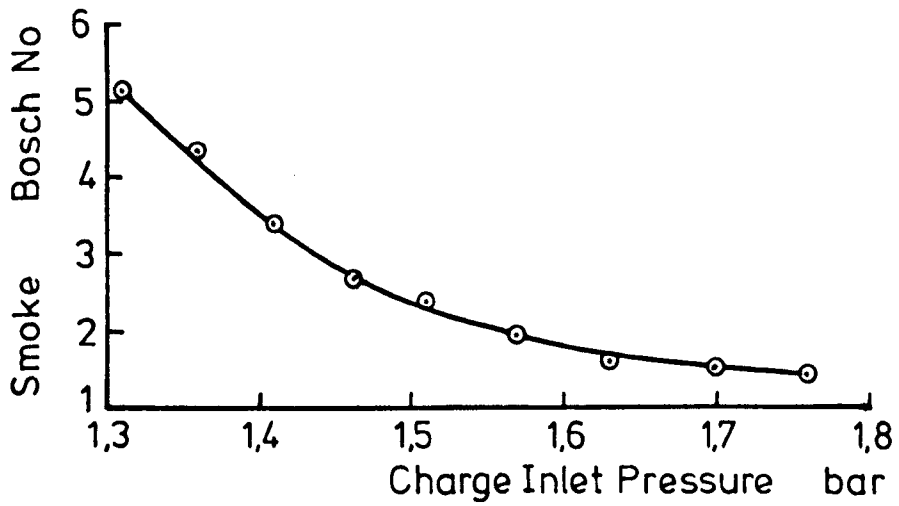
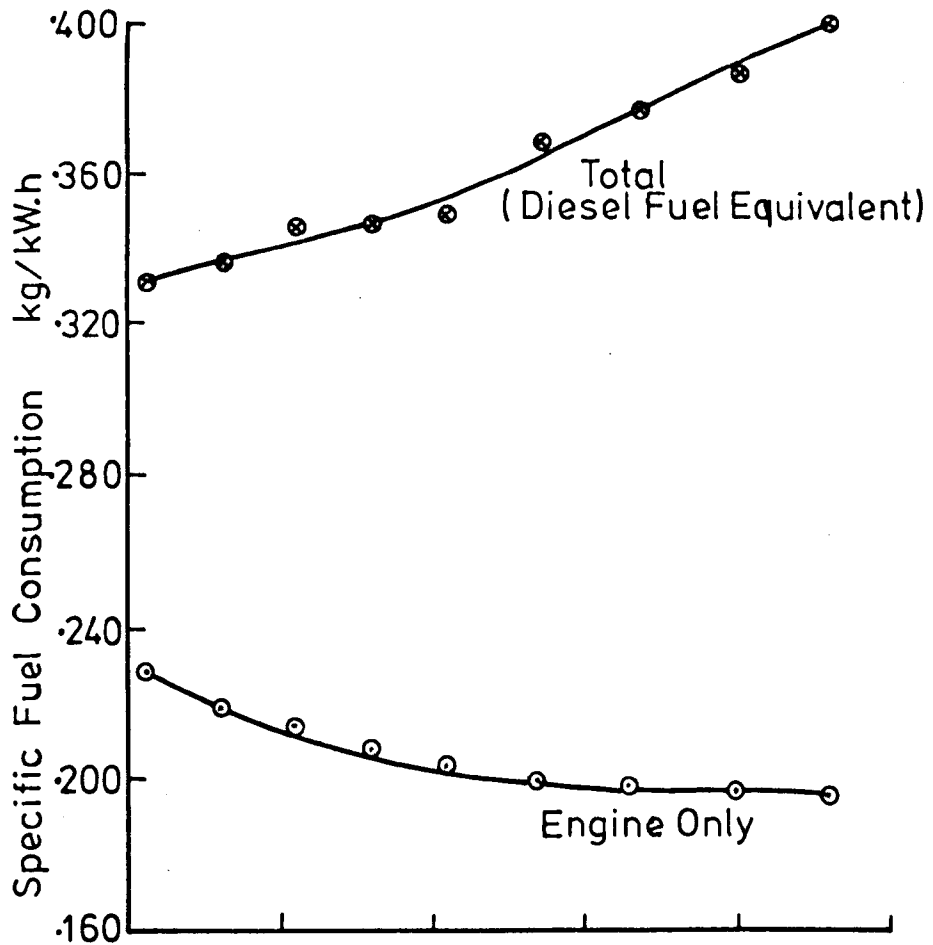


DIESEL GAS TURBINE PERFORMANCE (FIG 10)



CONSTANT ENGINE SPEED (1200 Rev/min)

DIESEL GAS TURBINE PERFORMANCE (FIG.11)



CONSTANT ENGINE SPEED (1200 Rev/min)
DIESEL GAS TURBINE PERFORMANCE (FIG 12)

APPENDIX 8

SELF SUSTAINED GAS TURBINE TESTS AND ENGINE EXHAUST PULSE INVESTIGATION

1. Self sustained gas turbine

A test rig was constructed, during the course of the project, to examine the behaviour of the turbocharger and auxiliary combustion chamber running as a self-sustained gas turbine. The rig (Fig. 1) was supplied with auxiliary air for starting. Lubricating oil was supplied from a reservoir by an electrically driven pump. The rig used bottled propane fuel which was ignited by a spark plug, the spark energy being supplied from an ignition unit as described in the thesis. The instrumentation has also been described in the main body of the thesis.

Tests were carried out with a Holset 3LD1 turbocharger using turbine housings of 1.57 and 2.5 sq. in. throat areas. A range of turbine pressure ratios was obtained by varying the fuelling rate to the auxiliary combustion chamber. The results are presented in Tables 1, 2 and 3, and Figures 2, 3 and 4. Figure 2 shows work supplied (turbine work) plotted against work demanded (compressor work). An enthalpy matched curve is also shown.

$$\text{Turbine work } (W_T) = \dot{m}C_p\Delta T_{TUR}$$

$$\text{Compressor work } (W_C) = \dot{m}C_p\Delta T_{\text{COMP}}$$

$$\text{Error} = W_T - W_C$$

The curves demonstrate that the turbine is apparently supplying more work than that demanded by the compressor. Figure 3 shows error plotted against turbine expansion ratio and demonstrates that the error increases with an increase in expansion ratio. Figure 4 shows that, similarly, the error increases with increasing turbine mass flow parameter.

In such steady state tests, the turbine work must be equal to the compressor work. The errors found experimentally indicate that the measurement of the relevant parameters must be at fault. Turbine operating parameters are notoriously difficult to measure and are most likely to be the cause of the discrepancies.

2. Investigation of engine exhaust pulsation

An investigation was carried out to assess the level of engine exhaust pulsation at the turbine inlet on the diesel gas turbine. A piezo-electric pressure transducer (as described in the thesis) was used. The environment was particularly hostile, due to the proximity of the auxiliary combustion chamber, and required that the water cooled transducer be mounted in a jacket which was also water

cooled. Tests were carried out at full load conditions with an engine speed of 1400 rev/min. The pressure traces were recorded using a tungsten light recorder and are shown in Figure 5. It can be seen that the large pulsations (exhaust) are followed by smaller pulses (scavenge). The frequency of these pulsations was 70 Hz which represents an engine speed of 1400 rev/min. The pressure traces do not demonstrate any pulse reflections or interference at the turbine inlet but these effects may be occurring nearer to the engine. The engine duct was of a large bore (100 mm) and comparatively long (approx. 1 m), thus the pulses at the turbine would be expected to show some degree of attenuation.

The exhaust pulses had a value of 0.14 bar peak to peak and the scavenge pulses 0.06 bar. When compared to the bourdon gauge reading of 1.6 bar absolute, these represent levels of 9% and 4% respectively. These levels of exhaust pulsation would not make a significant contribution to turbine work, but may be sufficient to cause auxiliary combustion problems in a system feeding engine exhaust gases through the auxiliary combustion chamber.

TABLE 1

Pressure Ratio	$\frac{m/T}{P}$	W_C kw	W_T kw	Error kw	Smoke Bosch	CO %	HC ppm
<u>Holset 3LD1 1.57</u>							
1.52	22.76	6.46	19.93	13.47	0.1		
1.41	20.96	5.23	13.80	8.57	0.2		
1.32	18.60	3.83	9.83	6.00	0.2		
1.24	16.59	2.65	7.69	5.04	0.3		
1.18	14.48	1.75	6.00	4.26	0.3		
1.12	12.36	1.13	4.53	3.40	0.2		
1.05	10.70	1.79	3.79	2.01	0.2		

TABLE 2

<u>Holset 3LD1 2.5 Test 1</u>							
1.42	28.50	8.37	13.18	4.45			
1.40	30.13	8.37	11.86	3.49			
1.27	27.98	5.81	6.93	1.12			
1.23	25.73	5.23	5.95	0.72			
1.21	24.74	3.44	5.35	1.91			
1.12	18.31	1.93	5.61	3.68			
1.01	12.44	1.39	3.08	1.69			

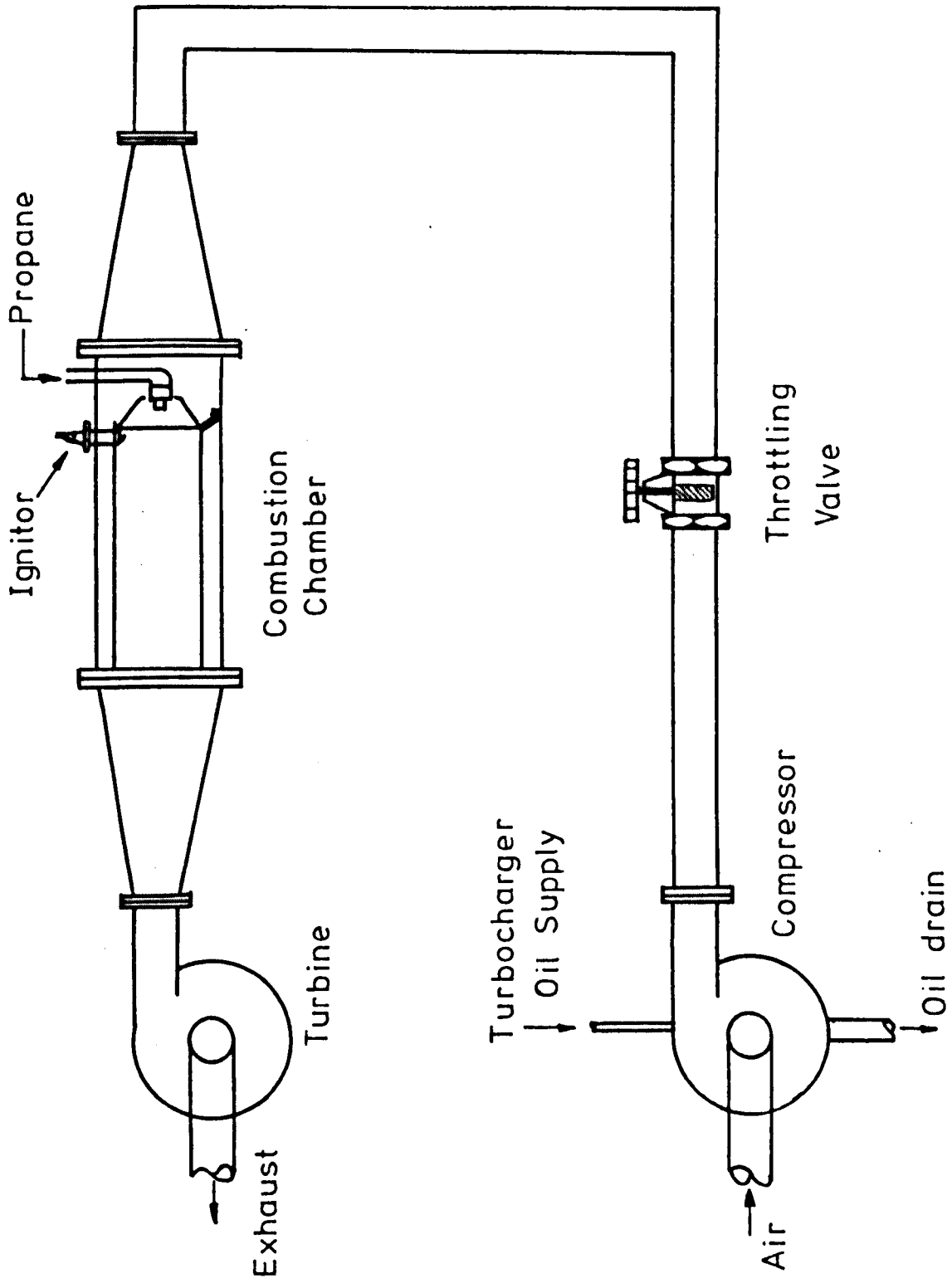
TABLE 3

<u>Holset 3LD1 2.5 Test 2</u>							
1.28	22.79	3.54	10.42	6.88	0.2	0.13	24
1.26	24.60	4.34	5.30	0.96	0.1	0.14	33
1.22	23.72	3.67	5.35	1.68	0.2	0.15	31
1.20	24.17	3.55	5.07	1.52	0.2	0.13	23
1.16	23.10	2.81	5.93	3.12	0.3	0.12	22
1.16	20.79	2.40	5.56	3.16	0.2	0.12	21
1.13	21.37	2.21	5.33	3.12	0.3	0.11	20
1.11	15.19	1.38	3.86	2.48	0.3	0.09	16
1.09	11.45	0.92	2.62	1.70	0.2	0.07	15
1.03	12.25	0.88	3.02	2.14	0.3	0.06	12

$\frac{m/T}{P}$ = Turbine Mass Flow Parameter

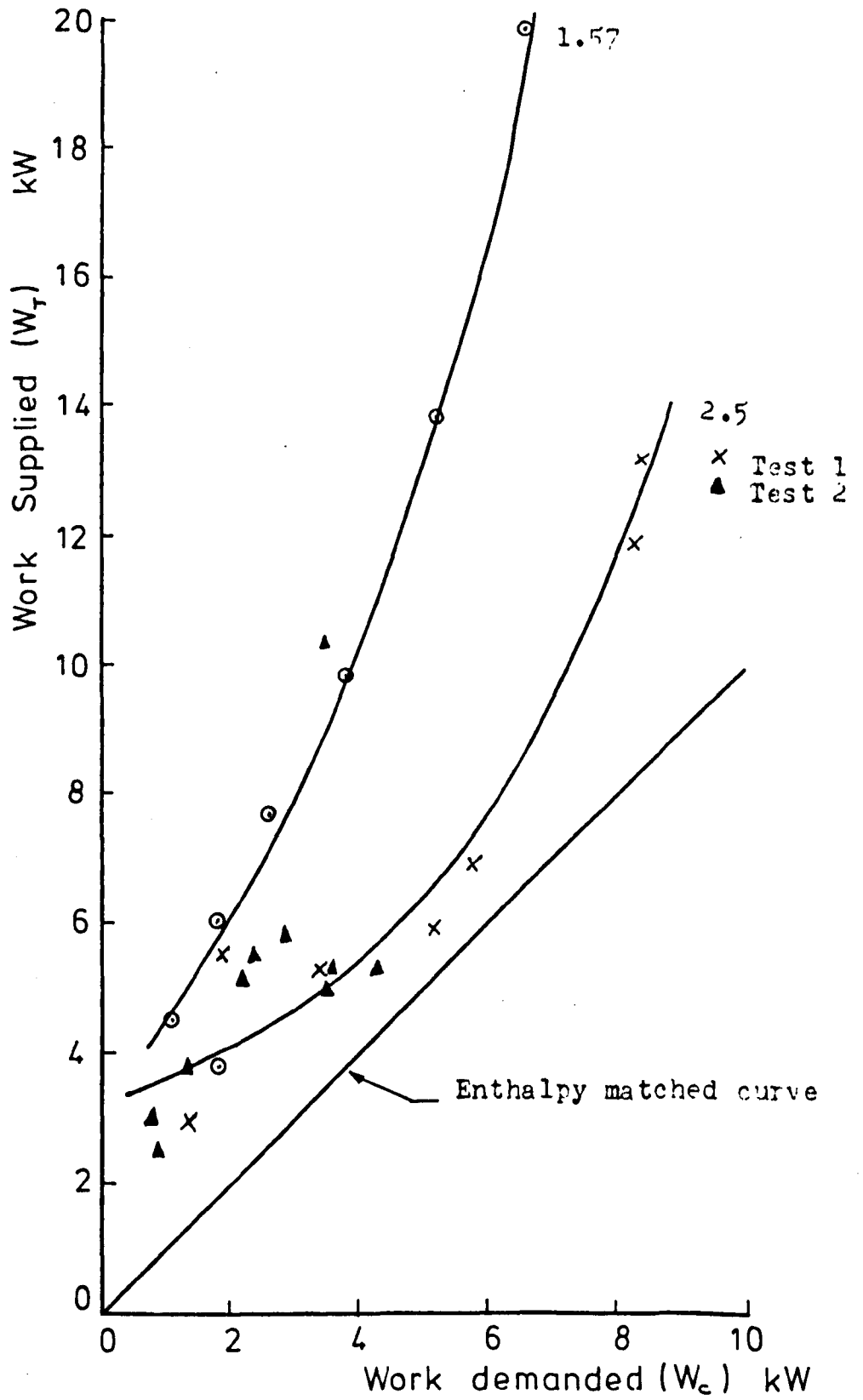
W_T = Turbine Work

W_C = Compressor Work



ARRANGEMENT OF SELF SUSTAINED
GAS TURBINE TEST RIG

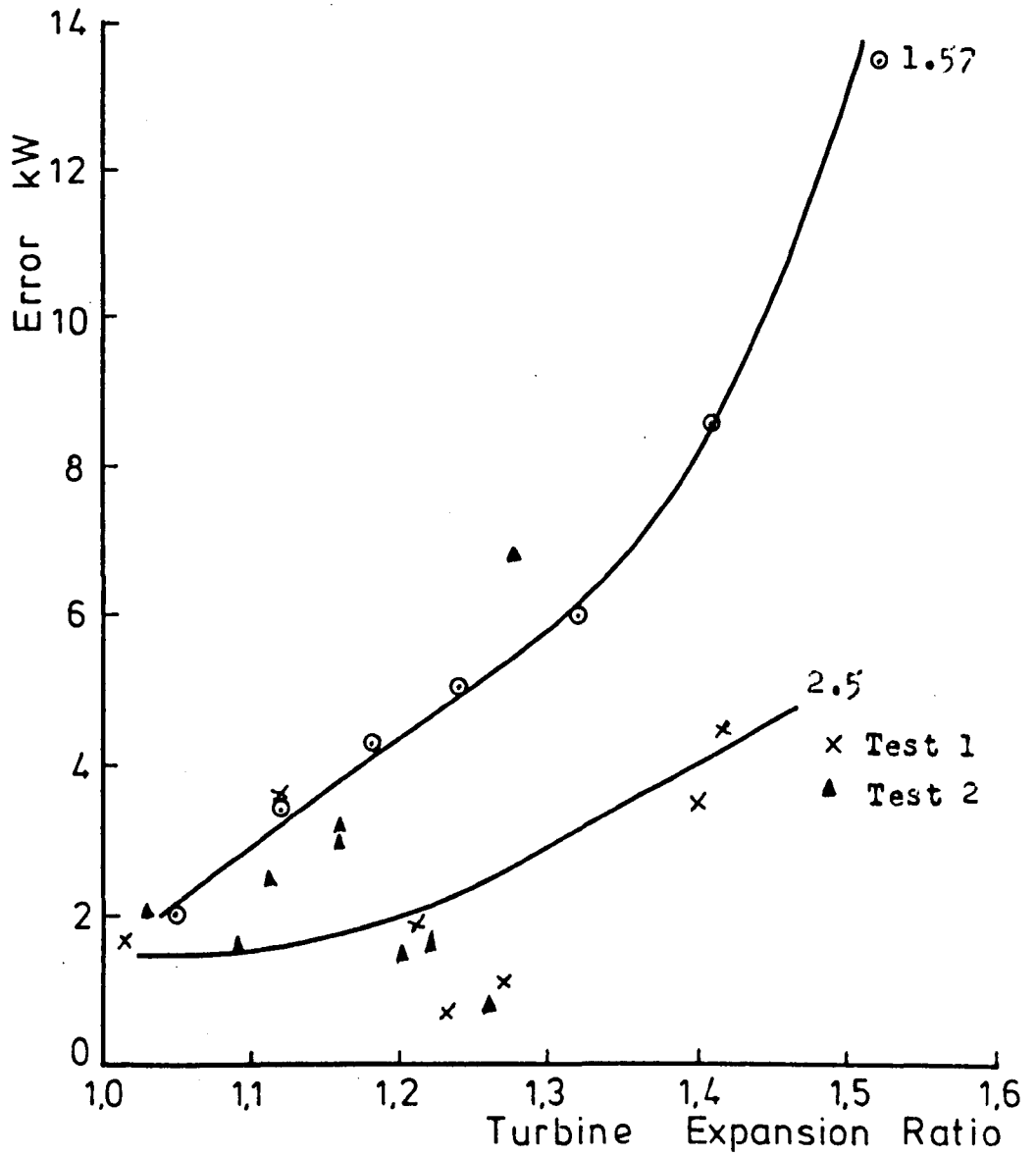
FIGURE 2



SELF SUSTAINED GAS TURBINE

HOLSET 3LD1

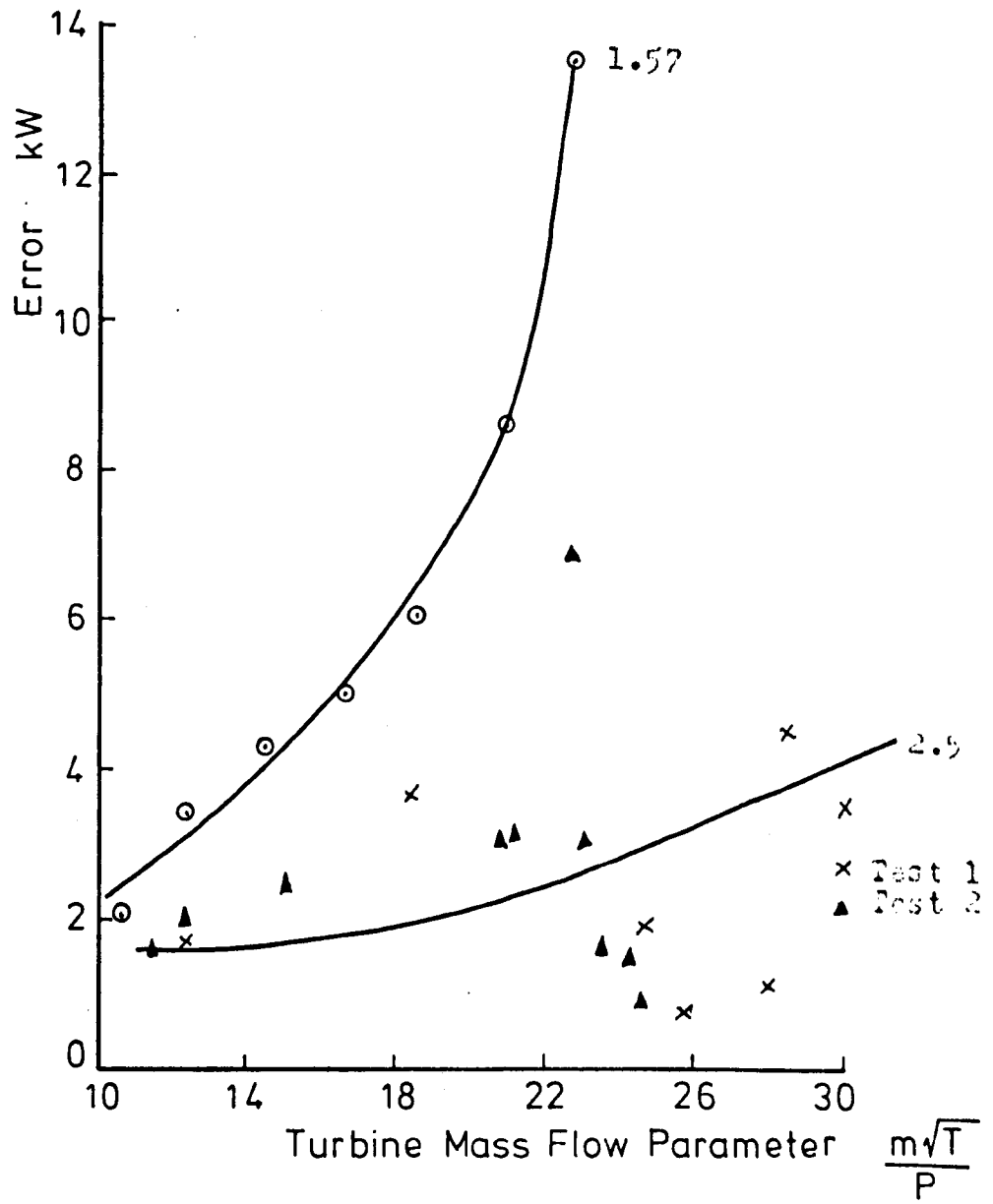
FIGURE 3



SELF SUSTAINED GAS TURBINE

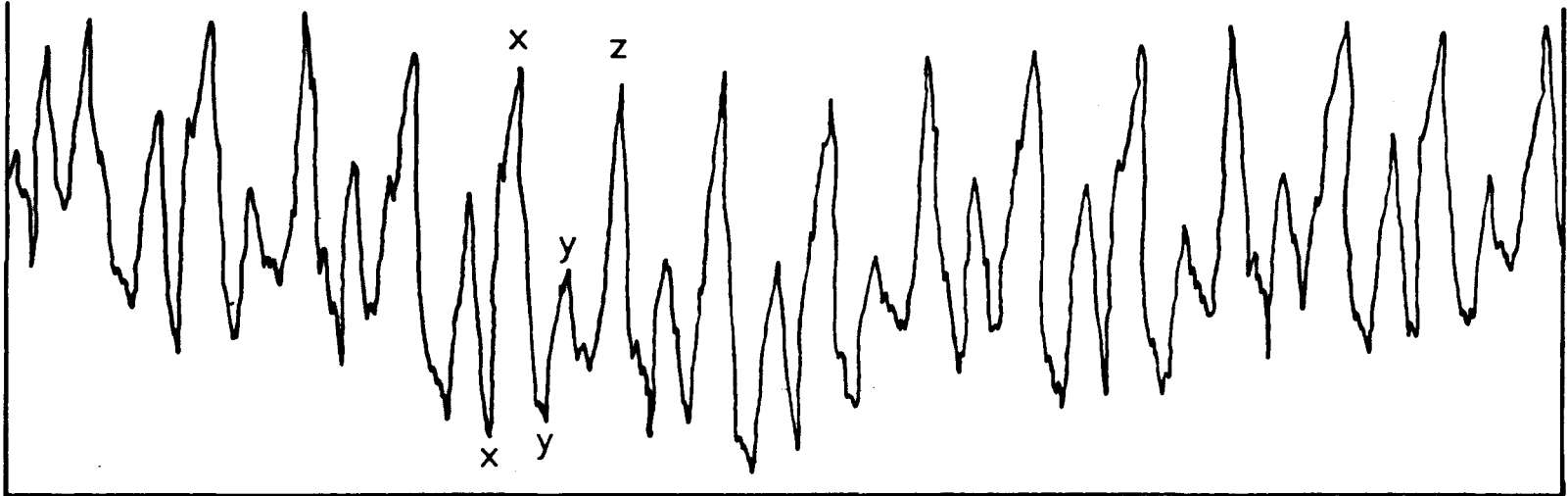
HOLSET 3LD1

FIGURE 4



SELF SUSTAINED GAS TURBINE

HOLSET 3LD1

EXHAUST PULSATIIONS AT TURBINE INLET

Magnitude of Exhaust Pulse $xx = 0.14$ bar ($\approx 9\%$)

Magnitude of Scavenge Pulse $yy = 0.06$ bar ($\approx 4\%$)

Frequency of Repetition $xz = 70$ Hz (1400 Rev/min)

Exhaust Pressure = 1.6 bar

APPENDIX 9. Diesel Gas Turbine Starting Procedure

- 1) All cooling water supplies were turned on.
- 2) Dynamometer water back pressure was set at 1.7 bar
(25 psi) gauge.
- 3) The slave oil supply was turned on.
- 4) The bypass air throttle was fully opened.
- 5) The engine air supply butterfly valve was closed.
- 6) The slave air supply valves were opened.
- 7) Auxiliary combustion chamber ignition was started.
- 8) Auxiliary fuelling was initiated.
- 9) When auxiliary combustion had commenced (viewed through a quartz 'flame window' in the combustion chamber), the slave air and auxiliary fuelling rates were increased until the gas turbine (formed by the turbocharger and auxiliary combustion chamber) reached a self-sustained condition. This usually took about 10 seconds from start-up.
- 10) The slave air valves were closed and the supply was disconnected.
- 11) The engine air butterfly valve was opened and the air bypass throttle was closed until a small, positive pressure drop was obtained across the engine.
- 12) The engine was then started, using the conventional starter motor.
- 13) The slave oil supply to the turbocharger was then turned off as the engine lube oil circuit had taken over the supply of turbocharger oil.
- 14) Engine speed was then increased and brake load applied.