AN ESTIMATION OF THE PERFORMANCE OF SPARK IGNITION METHANOL ENGINES

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I, ANDREW YATES, submit this thesis in fulfilment of the requirements for the degree of Master of Science in Engineering. I claim that this is my original work and that it has not been submitted in this or in a similar form for a degree at any University.

ABSTRACT

The possibility of using methanol as an alternative fuel for petrol and diesel engines has been investigated at the University of Cape Town in recent years by the Energy Research Institute. Engines, taken from existing road vehicles, were converted to methanol operation and evaluated. During the planning phase of each conversion, it was apparent that there was a need for a method of estimating the probable performance of the converted engine. The objective of this thesis was therefore to develop a generally applicable formula for predicting the rated performance of a methanol-fueled, spark-ignition engine given only the bore, stroke, number of cylinders and maximum rated engine speed.

The prediction was based on an estimate of the indicated efficiency, the volumetric efficiency and the frictional losses of a methanol engine, from which the shaft power and overall efficiency could be determined.

The performance of three engines, which were converted to methanol operation, was used to test the theory. The conversions were each of a very different nature:-

a) An automotive petrol engine of which only the carburettor was modified.

b) An automotive petrol engine that was modified to take advantage of the properties of methanol to improve the performance.

c) A truck diesel engine that was converted to spark ignition for operation on methanol.

The predicted results were found to compare favourably with the experimental results, with the exception of one spurious reading. An analysis of the range of error in the theoretical estimate was used as the basis for comparison:

Range of error in predicted overall efficiency $\pm$ 6%
Range of error in predicted shaft power $\pm$ 10%

The work of this thesis would be applicable in situations where a quick answer was required and few details were available. With the aid of a programmable calculator, the anticipated performance map could be obtained in a few minutes.

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NOMENCLATURE

SYMBOL	DESCRIPTION	UNITS
AFER	Air-fuel equivalence ratio: $\frac{\text{Actual air-fuel mass ratio}}{\text{Stoichiometric air-fuel ratio}}$	
BMEP	Brake mean effective pressure	Bar
C	Number of cylinders	
D	Bore diameter	mm
e_i	Indicated efficiency	%
e_o	Overall efficiency	%
e_s	Indicated efficiency at AFER 1,0	%
e_v	Volumetric efficiency	%
FMEP	Frictional mean effective pressure	Bar
F_s	Stoichiometric air fuel mass ratio	
IMEP	Indicated mean effective pressure	Bar
k	Ratio of specific heats for a gas	
m_a	Air mass flow rate	kg/s
m_f	Fuel mass flow rate	kg/s
N	Engine speed	Rev/min
N_r	Rated maximum engine speed	Rev/min
P	Shaft power	kW
P_i	Indicated power	kW
P_a	Air pressure at engine inlet	Pa
P_c	Pressure in the cylinder at the end of the induction stroke	Pa
Q_c	Fuel heat of combustion per unit mass	kJ/kg
r	Compression ratio	
S	Stroke	mm
T_a	Air temperature at engine inlet	°C
T_c	Temperature in the cylinder at the end of the induction stroke	°C
V_d	Swept Volume	l
v_a	Specific volume of air at engine inlet	m^3/kg

GLOSSARY

AIR-FUEL EQUIVALENCE RATIO : The ratio of the actual air-fuel ratio to the stoichiometric air-fuel ratio.

AIR-FUEL RATIO : The ratio of the mass flow of air to the mass flow of fuel.

BRAKE (MEAN EFFECTIVE PRESSURE, EFFICIENCY, POWER): Refers to the actual performance measured by an engine brake.

DETONATION : A phenomena of combustion associated with a very sudden, and uncontrolled, acceleration of the flame front, giving rise to shock waves within the combustion volume.

INDICATED (MEAN EFFECTIVE PRESSURE, EFFICIENCY, POWER): Refers to the potential performance indicated by the actual pressure-volume diagram.

MEAN EFFECTIVE PRESSURE : The height of a rectangle on the pressure-volume diagram having the same length and area as the cycle. It can be regarded as that constant pressure which, by acting on the piston over one stroke, can produce the net work of the cycle.

OVERALL EFFICIENCY : The ratio of the brake power to the product of fuel mass flow and fuel heat of combustion.

PREIGNITION : Ignition of the charge before the ignition spark occurs.

RATING : The limit of operation specified by the manufacturer.

RESEARCH OCTANE NUMBER : A number representing a measure of the fuel tendency to detonate. The number scale is determined as the ratio of two reference fuels which when mixed will produce a fuel that has the same resistance to detonation as the test fuel in a standard reference engine.

STOICHIOMETRIC AIR-FUEL RATIO : The air-fuel ratio such that there is theoretically just sufficient oxygen to burn all the combustible elements in the fuel completely.

CHAPTER ONE

INTRODUCTION

1.1 BACKGROUND

The long range forecast for the world oil reserves indicates that the inevitable shortfall in production is imminent. Increased awareness of the need to find alternative sources of energy has motivated a worldwide research movement, with particular emphasis on liquid transport fuel alternatives.

Investigations of fuels for the internal combustion engine are usually defined by three objectives: To improve the efficiency and performance of engines; to widen the availability of natural resources for fuel production; and to reduce pollutants in engine exhaust gases. Most recent research in alternate fuels for the internal combustion engine is aimed at attaining all three objectives.

Hydrogen and electricity are perhaps the ultimate transport power sources, that would meet the above objectives. However, the complete lack of a hydrogen-use infrastructure, from production to fuelling stations, together with the problems of storage in the vehicle, and the necessary technological development of the power unit, precludes the general use of hydrogen as a fuel for at least several decades. The electric vehicle has thus far failed to compete with the internal combustion engine, except where range and speed are of secondary importance. This situation is unlikely to change without a dramatic break-through in battery technology. Reference 1 (1) gives a comprehensive analysis of the future transport fuel requirements, and reveals an urgent need for a short to medium term (approximately 50 years) supplement for petrol and diesel derived from crude oil.

Investigation of previously neglected alternative fuels, or fuels used only in crisis situations, is now one of the most pressing tasks in assuring the short term fuel supply. Alcohol fuels are rapidly emerging as promising alternatives. In particular methanol, produced from coal or natural gas, would appear to be capable of replacing petrol and diesel as an internal combustion engine fuel (2). Another possibility, that of producing petrol and diesel from coal or natural gas warrants consideration. However, the economic estimates indicate that methanol yields the highest overall conversion efficiency in terms of energy content of the coal or gas to energy content of the liquid fuel product (3).

1.2 METHANOL RESEARCH AT U.C.T.

Research into the practical problems of powering engines with methanol has been initiated at the University of Cape Town by the Energy Research Institute.

Although a variety of different methods for utilizing methanol in internal combustion engines has been reported in the literature, it was speculated that the introduction of methanol as an alternative transport fuel would be based on the already proven technology of the conventional spark-ignition engine. An analysis of the possible alternative methanol engine designs which leads to this conclusion is presented in Appendix A.

A number of methanol engines were built using as many standard petrol and diesel engine components as possible. The converted engines were then tested and evaluated in the laboratory and on the road.

Throughout the course of this experimental programme it was found that there was a need for a method of anticipating the methanol engine performance prior to conversion and testing. This need arises from the fact that maximum compression ratios and lean burn air/fuel ratio limits for methanol are significantly higher than for petrol, and therefore petrol engine performance data is not comparable.

A methanol performance estimate would be especially useful when planning the conversion of a diesel engine to spark ignition, since the alterations are more extensive than for a petrol engine. Also the power and overall efficiency of a truck engine are principal considerations to the operator.

1.3 THESIS OBJECTIVES

The objective of this thesis is to establish a general formula for predicting the performance of a conventional spark-ignition engine operating on methanol of which bore, stroke, number of cylinders and rated engine revolutions are the only known parameters.

The definition of performance is taken as the rated wide open throttle power and efficiency, according to DIN 70020 Part 6 (4). An estimation of the part load efficiency is not included.

A side issue that forms an integral part of the performance calculations, is an estimation of the limits of compression ratio for the given engine.

Inevitably a prediction of this nature must be approximate, and therefore some consideration of the extent of the error is required. On the other hand, an approximate result can be justified if the results are quickly obtained. To this end, all the relationships necessary for the calculation of performance are expressed analytically in a format suitable for the use of a programable calculator.

CHAPTER TWO

ENGINE PERFORMANCE PREDICTION METHODS

A survey of the literature, relevant to the analysis of the performance of a given engine revealed a number of different techniques. Each technique had been developed to suit certain applications, some of which were relevant to the objectives of this thesis.

Four main approaches to the subject of engine performance prediction were identified.

a) The use of historical trends in engine parameters and fuel properties to predict future trends.

b) The use of similitude in engine design ratios to relate the performance of one known engine to an unknown engine of about the same size and output.

c) Engine modelling by means of numerical analysis with the aid of computer technology.

d) The application of simple theoretical relationships between engine parameters and performance which are modified empirically by means of laboratory test results.

These prediction techniques were analysed in order to determine their suitability as a tool for the general estimation of the performance of a given engine operating on methanol.

2.1 HISTORICAL TRENDS

The conventional petrol and diesel engine has been with us for so long that past experience is a good method of estimating the performance of such an engine. Blackmore and Thomas (5) found that the road fuel economy of gasoline powered cars could be related to the vehicle weight and the year of production with a fair degree of correlation. Examples of the type of results that were obtained are shown in figures 2.1 and 2.2. (The effect of the 1973 crude oil price increase is reflected in figure 2.2 by a sharp rise in fuel economy).

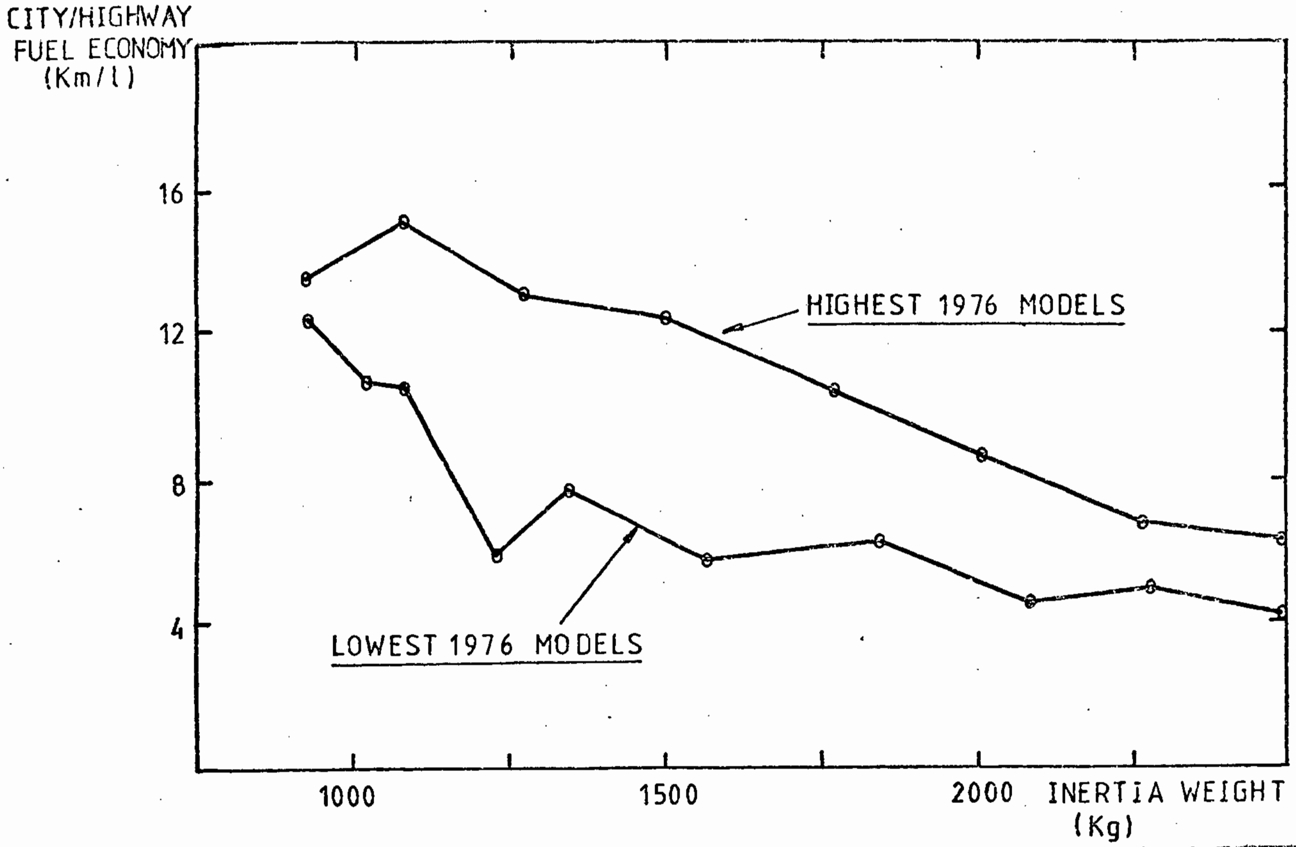
A more detailed analysis of historical trends in fuel properties and trends in engine parameters was given by Taylor (8, 9). Trends in average compression ratio and average fuel research octane number are compared. Also the trends of U.S. passenger automobile engine characteristics are presented, such as average compression ratio, brake power, displacement, rated engine speeds, and bore to stroke ratio as a function of year of production.

Unfortunately this information is insufficient to enable a prediction of the performance of any particular engine. Taylor does not give the trend in overall efficiency, neither does he give any indication of the statistical deviation on the average values.

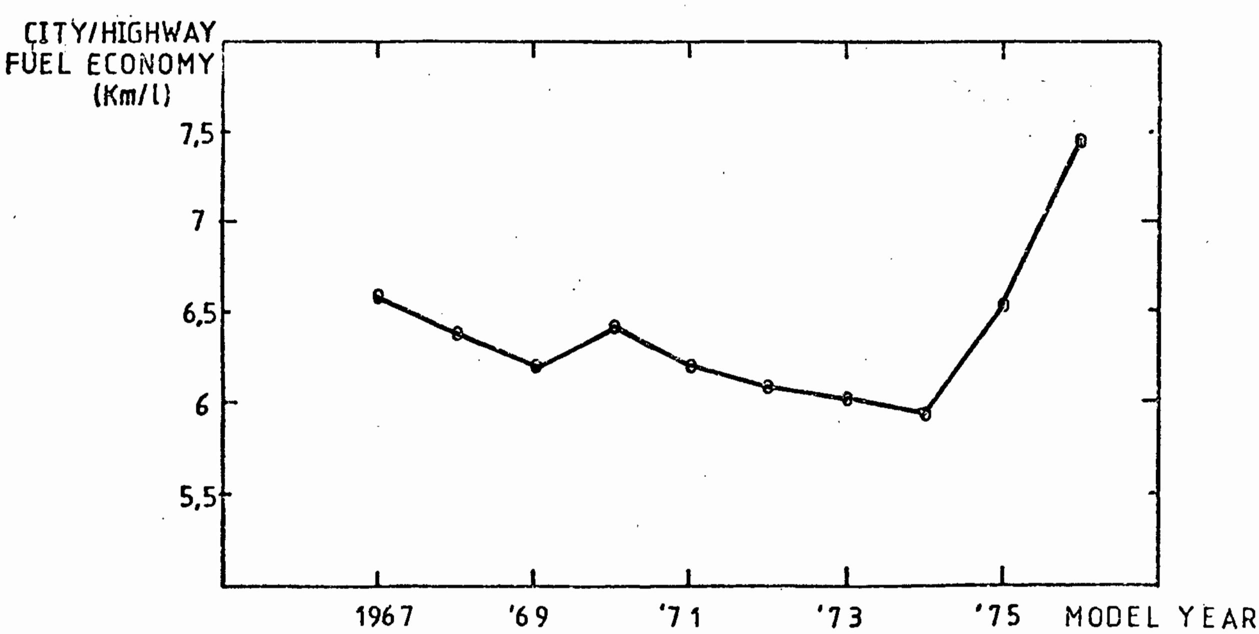
Another aspect concerning the suitability of this form of analysis for alternative fuels was raised by Stebar et al (10). The estimation of the fuel utilization efficiency of engines should be based on data for comparably sized engines using the same fuel and having similar performance at the same exhaust emission levels. This data is not available for most of the alternatively fuelled automotive powerplants.

2.2 SIMILITUDE MODELLING

The general application of similitude modelling in the content of engine performance was demonstrated by way of an extreme example by Taylor (11), comparing a toy model aircraft engine and a ship engine of 5211 litre displacement. Comparison of the brake mean effective pressure (BMEP), piston speed, power/piston area and weight/displacement revealed a maximum difference of 34% for the small engine to the large engine.



2.1 FUEL ECONOMY VERSUS INERTIA WEIGHT FOR 1976 GASOLINE POWERED CARS. Data based on Ref 6



2.2 SALES WEIGHTED FUEL ECONOMY TRENDS FOR 1967-76 Data based on Ref 7

The principles of similitude related to engine performance are given in most text books on internal combustion engines. Maleev (12) for example gives a concise account. In order that the rules of similarity should apply, engines being compared should have geometric, kinetic and dynamic similarity. This means that the engines should have

a) the same stroke-bore ratio

b) the same speed factor, defined as the product of piston speed and engine speed

c) the same brake mean effective pressure.

In practice all engines do not comply with these requirements and a refinement of the similarity principle is necessary. One approach which was used by Taylor (13) was dimensional analysis, where the engine parameters were grouped into suitable dimensionless functions and design ratios. In this example he used the technique to calculate volumetric efficiency. However he found that the volumetric efficiency, when enpressed in a mathematical form, required such intimate knowledge of the gas properties and temperatures, as well as the engine geometry, that some simplifying assumptions were necessary.

The problems of modelling by means of similitude were discussed by Spalding (14). He too concluded that the requirements of similarity theory were so numerous that only partial modelling was practicable. However his conclusions cast some doubt on his methods, since he concluded that often the modelling technique which most flagrantly flouts the similarity rules was found to be the most useful one in practice.

It is evident that the use of similitude as a method of engine performance prediction requires a complete dimensional definition of the engine under consideration. By the constraints of Chapter 1, only the most fundamental dimensions may be assumed to be defined, and therefore an analysis by means of simiiitude could only be somewhat approximate.

2.3 COMPUTER MODELLING

Perhaps the most important development in engine research techniques that has occurred in the last decade is the use of the digital computer to simulate various aspects of engine performance, including the performance of complete engines. As in many other fields, the computer has made possible the solution to complex relations that involved too much labour to be attempted by older methods of calculation. In particular, our understanding of the combustion process, exhaust emission formation, thermodynamic cycle efficiency, and the gas exchange process has been greatly refined as a result of computer modelling.

A complete analysis of the numerical method of analysing the complete Otto cycle (spark-ignition cycle) has been published in several text books (15, 16) and papers (17, 18, 19). The calculation is usually based on first principle theory, considering small time increments during which properties are considered to be constant.

The exact worth of computer modelling techniques is subject to some differences of opinion. For example, Benson (16) describes his calculated pressure/volume cycle as equivalent to the real cycle. However, one of his assumptions was that the flame front was thin, which is questionable in the light of flame modelling work conducted by Westbrook et al, (20) who stated that almost any definition of flame thickness was somewhat arbitrary.

As a general summary, the computation of the complete engine performance requires the input of a vast number of variables, many of which have to be assumed or omitted due to lack of basic data. The quality of the program thus depends heavily on the skill of the operator. By adjusting the assumptions so that the results agree with the measured results of one particular engine, the performance of similar engines can be fairly accurately predicted.

Only one example of the use of a computer model to predict the performance of a methanol fuelled engine was found (21). The computer model included the thermodynamic and chemical kinetic considerations to evaluate the indicated performance and exhaust emissions. The results were compared to those obtained by experiment. The trend in power, thermal efficiency and emissions as a function of air-fuel ratio was found to agree well, but the actual predicted results were more than 6% too high in some instances.

It would appear that in spite of its limitations, computer technology is a valuable tool more for the indication of trends in engine performance than for the accurate prediction of absolute values.

2.4 LABORATORY TESTS

The number of references pertaining to the results of engine studies is enormous. The Transactions of the Society of Automotive Engineers, which date back to the earliest published work on engines contain a wealth of test data. Text books have been written which effectively represent a literature survey of the field, of which perhaps the best known is that of Taylor (8, 11) who lists more than 2,500 references.

On the subject of methanol there are also a considerable number of publications as a result of four international symposia and some basic engine studies published by the Society of Automotive Engineers.

The research publications pertaining to the study of spark ignition engines can be broadly categorized into the following two areas.

2.4.1 The Thermodynamic Cycle

Starting with the laws of thermodynamics applied to an idealized constant volume cycle, the compression, ignition, expansion and gas exchange phases of the four stroke Otto cycle have been studied, and continue to be studied to better understand the reasons for the differences between theory and reality. The effect of changes of operating conditions on the cycle behaviour is also the subject of extensive research.

Considerable data on the Otto cycle behaviour of methanol has been published, although in some cases the test conditions under which the results were obtained is not given. One particular example of meticulous research reporting is that of Ebersole and Manning, (22) who thoroughly investigated the performance and emissions of a single cylinder engine operating on methanol and compared the results to those obtained with iso octane. On the other hand Menrad et al, (23) describing their single cylinder tests which led to the development of a pure methanol fuel car, reported the highest brake thermal efficiency for methanol operation that could be found in all the available literature, but neglected to state inlet air conditions. He also neglected to state whether the corresponding power was indicated or brake output.

2.4.2 Physical Losses

The useful shaft power of an engine, in practice, is usually far less than the potential power indicated by the thermodynamic cycle. This difference is due to physical losses such as fluid friction during the gas exchange process, mechanical friction, and the necessary power to drive auxiliaries.

Text book methods of computing these losses are generally accepted as being sufficiently accurate for most applications. Again the work of Taylor (24) is about the most comprehensive publication on the subject, being itself a summary of the findings of many other researchers.

2.5 OTHER METHODS

One rather novel approach to modelling a spark ignition engine that should be mentioned is that presented by Marshall and Oversby (25). Using electrical control theory they analysed the engine performance and exhaust pollutants, considering the engine as a transfer function, the form of which they attempted to determine by observing its time response to a step input.

Their results were sufficiently encouraging to warrant further work, but not sufficiently accurate to justify the general use of the theory in its present form.

2.6 SUMMARY AND CONCLUSIONS

The use of historical trends in petrol and diesel engines cannot be related to the design of engines for other fuels with confidence.

The use of similitude as a tool for the estimation of the performance of any engine would require a full dimensional definition of the engine, as well as extensive knowledge of the properties of the working medium. Where this information was not available, simplification of the theory, and therefore uncertainty in the results would be inevitable.

The digital computer is undoubtedly a valuable prediction tool, and one that could suit the requirements of this thesis. As with similitude, the assumptions that would be required where detailed data was lacking would render the final result approximate.

However, the bulk of the published literature concerning methanol relates to the results of laboratory tests. The marriage of the reported thermodynamic cycle behaviour of methanol to the empirical relationships governing the physical losses in conventional engines seemed to offer the best method of meeting the thesis objectives. An additional factor in favour of this approach was that some feeling for the probable error in the final answer could also be determined.

CHAPTER THREE

THEORETICAL DERIVATION OF PERFORMANCE ESTIMATES

The derivation of the approximate performance of a spark-ignition methanol engine, given only the bore, stroke, rated engine speed and number of cylinders is presented in this chapter. Although details of the engine design cannot be specific if the theory is to remain generally applicable, it is nevertheless necessary to make broad assumptions concerning the engine design. In order for such assumptions to be as realistic as possible, reference is made to a discussion on "The Fuel Related Properties of Methanol : Their Influence On Engine design", given in Appendix B.

3.1 THEORY

The formulae relevant to this chapter are

$$P = N\, V_d\, (IMEP-FMEP)/2$$

$$e_o = e_i\, (1-FMEP/IMEP)$$

$$\text{where}\quad IMEP = e_v\, Q_c\, e_i/v_a\, F_s\, AFER$$

The meaning of the symbols is given in the nomenclature. The derivation of these formulae from first principles is given in Appendix C.

The terms can be considered as either independent variables, dependent variables or constants. Each term must therefore be examined in turn.

3.2 DETERMINATION OF INDICATED EFFICIENCY

The theoretical Otto cycle efficiency can be derived from the equation

$$e_i = 1 - (1/r)^{k-1} \qquad (26)$$

Although this equation is an over-simplification of what occurs in practice, it is nonetheless useful to indicate the logrithmic dependance of efficiency on compression ratio. The value of "k" differs for the compression and expansion cycle, but is primarily dependent on the air-fuel ratio.

An approach to a closer approximation of efficiency by means of numerical analysis was discussed in Section 2.3. This form of analysis would require much information about the particular engine which by the constraints of Section 1.3 is not available.

An alternative method of determining the indicated efficiency by suitably combining all the reported data of methanol engine studies was adopted. Since indicated efficiency is primarily a function of compression ratio and air-fuel ratio (the air-fuel ratio determines the values of k for the compression and expansion phases of the cycle), and is almost independent of engine load (35) provided the ignition timing is optimal (36), this practical approach is justified.

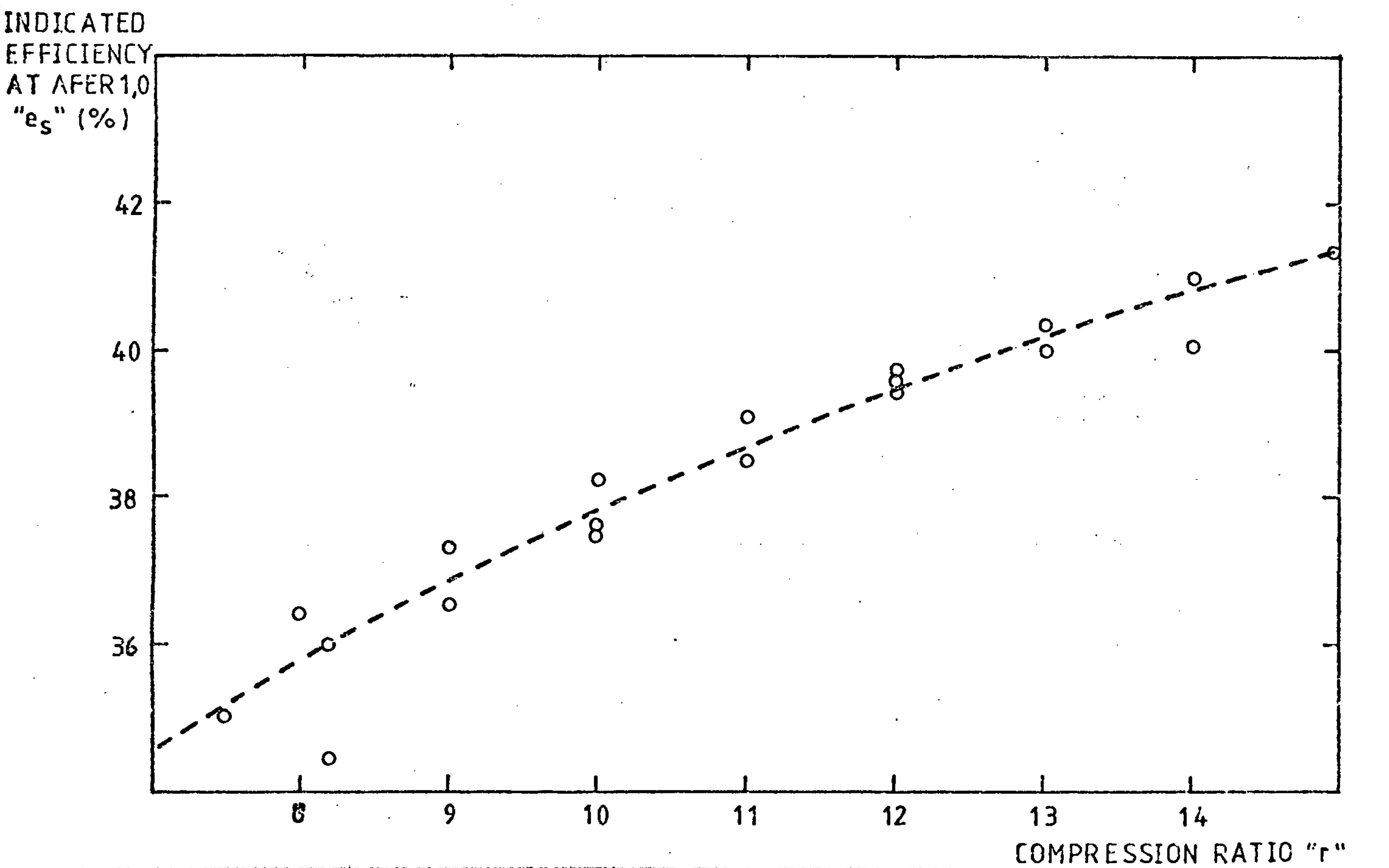
All the available data that could be found in the literature pertaining to the indicated efficiency of methanol engines was extracted and tabulated in Tables 3.1 and 3.2. Figure 3.1 shows the measured indicated efficiency for methanol at air-fuel equivalence ratio (AFER) 1,0 as a function of compression ratio. The best fitting curve was calculated assuming the theoretical logrithmic relation to hold true.

A correction curve for AFER other than 1,0 was then obtained by dividing the actual efficiency by the stoichiometric efficiency, shown in figure 3.2. When expressed in this form the correction is independent of compression ratio (35). Least squares methods were used to fit a curve through the points.

An estimate of the indicated efficiency for any methanol engine can therefore be made if the compression ratio and air-fuel ratio are known. Both variables are independent but subject to limits.

3.2.1 Compression Ratio Limits

The maximum compression ratio of a spark ignition engine is limited by detonation. There are a great many factors that influence



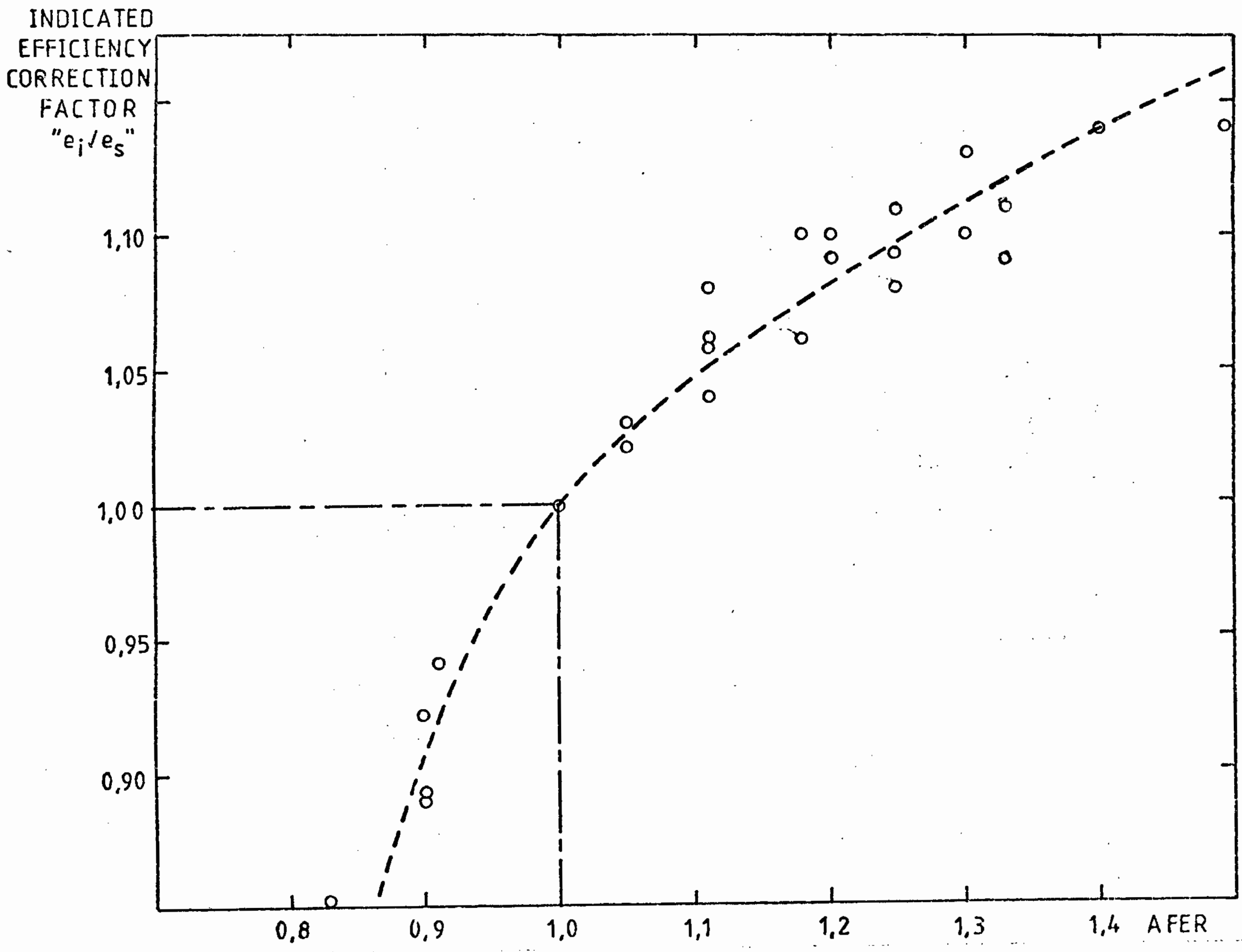
3.1 INDICATED EFFICIENCY FOR METHANOL AT AFER 1,0

Data based on Table 3.1 (Ref. 27,28,29)

Equation of Line

$$e_s = 100 - \mathrm{Log}^{-1} (-0{,}1454 \ \mathrm{Log}\ (r) + 1{,}939)$$

Standard error of the estimate : 0,479 Percentage Units



3.2 INDICATED EFFICIENCY CORRECTION FACTOR FOR METHANOL

Data based on Table 3.2 (Ref. 30, 31, 32, 33, 34)

Equation of line :

For AFER less than 1,0

$$e_i = e_s\,(-3{,}76(AFER)^2 + 8{,}105\ AFER - 3{,}345)$$

For AFER greater than 1,0

$$e_i = e_s\,(0{,}875 + (29{,}7\ AFER - 26{,}25)^{0{,}5}/14{,}85)$$

Standard error of the estimate : $2{,}05 \times 10^{-2}$

detonation, but the fuel octane rating, the engine compression ratio and the cylinder size are perhaps the dominant parameters. The Research Octane Number (RON) of methanol (discussed in Appendix B, Section 5) is taken as 110, which is considerably higher than that of premium grade petrol (RON 98).

For the purpose of giving a guide only, the relationship between maximum compression ratio and bore size for three geometrically similar engines was combined and extrapolated with the relationship between maximum compression ratio and octane rating for two engines of fixed bore size (see Appendix D). From this data it was possible to produce Figure 3.3, relating the maximum compression ratio of a methanol engine to the bore size.

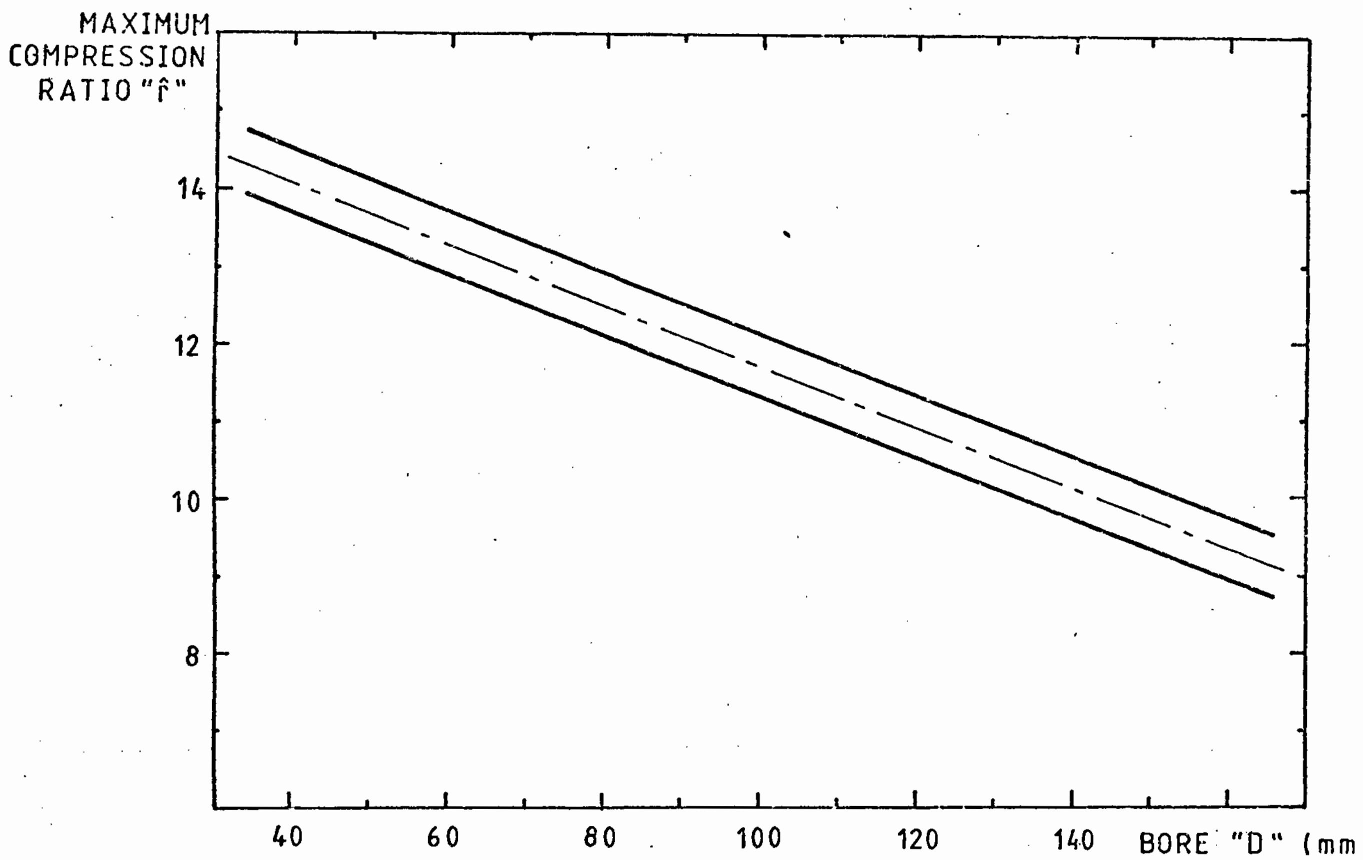
It must be stressed that Figure 3.3 does not define an exact relationship, since the onset of detonation is governed by many factors other than bore size. To emphasize this point, the relationship depicted in Figure 3.3 is not represented as a single valued function. A more detailed discussion of this derivation is given in Appendix D.

3.2.2 Air-Fuel Ratio Limits

The ignitability of methanol and air is discussed in Appendix B, Section 6. The lean limit of ignition which is generally of greatest interest, is leaner than that of petrol. Values of AFER 1,7 have been obtained in practice, under ideal conditions of mixture preparation, compared to values for petrol of about AFER 1,25.

3.3 DETERMINATION OF VOLUMETRIC EFFICIENCY

The volumetric efficiency of a four stroke engine is defined as the ratio of the mass of air that is drawn into the cylinder during the induction stroke to the mass of air that could ideally fill the cylinder during the induction stroke at the inlet temperature and pressure. Methods of predicting this could be empirical (37) or theoretical (38). The full theoretical treatment would normally be used to determine inlet valve sizes and camshaft profiles and is not readily applicable to a generalized formulation.



3.3 APPROXIMATE RELATIONSHIP BETWEEN MAXIMUM COMPRESSION RATIO AND BORE DIAMETER FOR METHANOL

Data based on Figure D4

Equation of average line : $r = 3{,}98 \times 10^{-2} D + 15{,}7$

Range in r $\pm$ 0,4

An attempt to generalize the volumetric efficiency of all engines could be open to considerable inaccuracy. However, in practice, the inaccuracies are diminished because the design objectives and constraints governing volumetric efficiency are similar for all engines. The designer would wish to maximize volumetric efficiency, but the maximum valve size is limited by water jacket requirements; the maximum valve lift is limited by the strength of the operating mechanism; the valve timing is restricted by the design of the inlet and exhaust manifolding; etc.

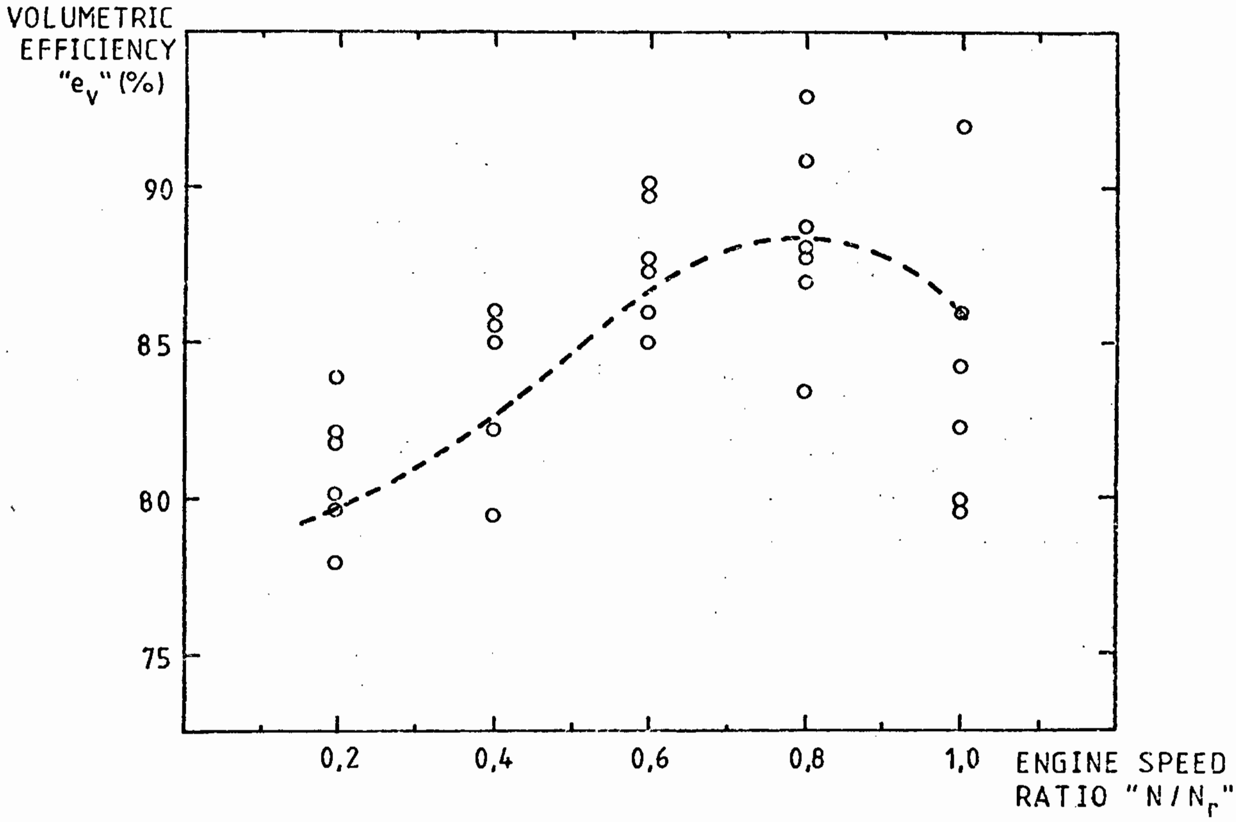
The volumetric efficiency is related to the nominal Mach Number of the air flowing between the valve and valve seat (39). A significant drop in volumetric efficiency occurs as the Mach Index approaches 0,6. Assuming that the valve design of the "typical" automotive engine is such as to maximize volumetric efficiency over the operating speed range, the mach index at the rated engine speed would be about 0,5. (37, 39). The volumetric efficiency as a function of mach index for several different petrol engines can then be related to engine speed as is shown in Figure 3.4. The form of the curve representing the best fit, was chosen as a cubic polynomial, since in practice the volumetric efficiency typically exhibits a peak near the maximum engine speed and a point of inflection at low engine speeds.

The effect of mixture cooling, which is a significant feature of methanol (Appendix B, Section 3) can be estimated from an expression for volumetric efficiency derived by Benson et al (40).

$$e_v = P_c T_a / P_a T_c$$

Measurements of mixture temperatures indicate that at wide open throttle, the equilibrium temperature for methanol is about 4 ^{o}C whilst that for petrol is about 15 ^{o}C (41). Similar findings were reported by Richards (42). Provided that the manifold walls remain wet with fuel, the effect of manifold heating can be expected to be small, since very little sensible heat can be supplied to the mixture until the walls are dry.

Assuming the ratio of these temperatures to hold true for the "in cylinder" conditions during the induction stroke, the



3.4 VOLUMETRIC EFFICIENCY OF 7 AUTOMOTIVE PETROL ENGINES

Data based on Ref 43

Equation of line :

$$e_v = -55{,}4\,(N/N_r)^3 + 74{,}7\,(N/N_r)^2 - 13{,}2\,(N/N_r) + 79{,}5$$

Standard Error of the estimate : 3,037 Percentage Units improvement in volumetric efficiency for methanol over petrol would be about 4%.

Applying this factor to the analytical equation of Figure 3.4, an expression for the volumetric efficiency of a methanol engine can be determined:

$$e_v = -57,6\ (N/N_r)^3 + 77,7\ (N/N_r)^2 - 13,7\ (N/N_r) + 82,7$$

with a standard error of the estimate of 3,0 percentage units.

It is evident that some simplifications have been made to obtain this result. These are discussed more fully in later sections.

3.4 DETERMINATION OF FRICTIONAL LOSSES

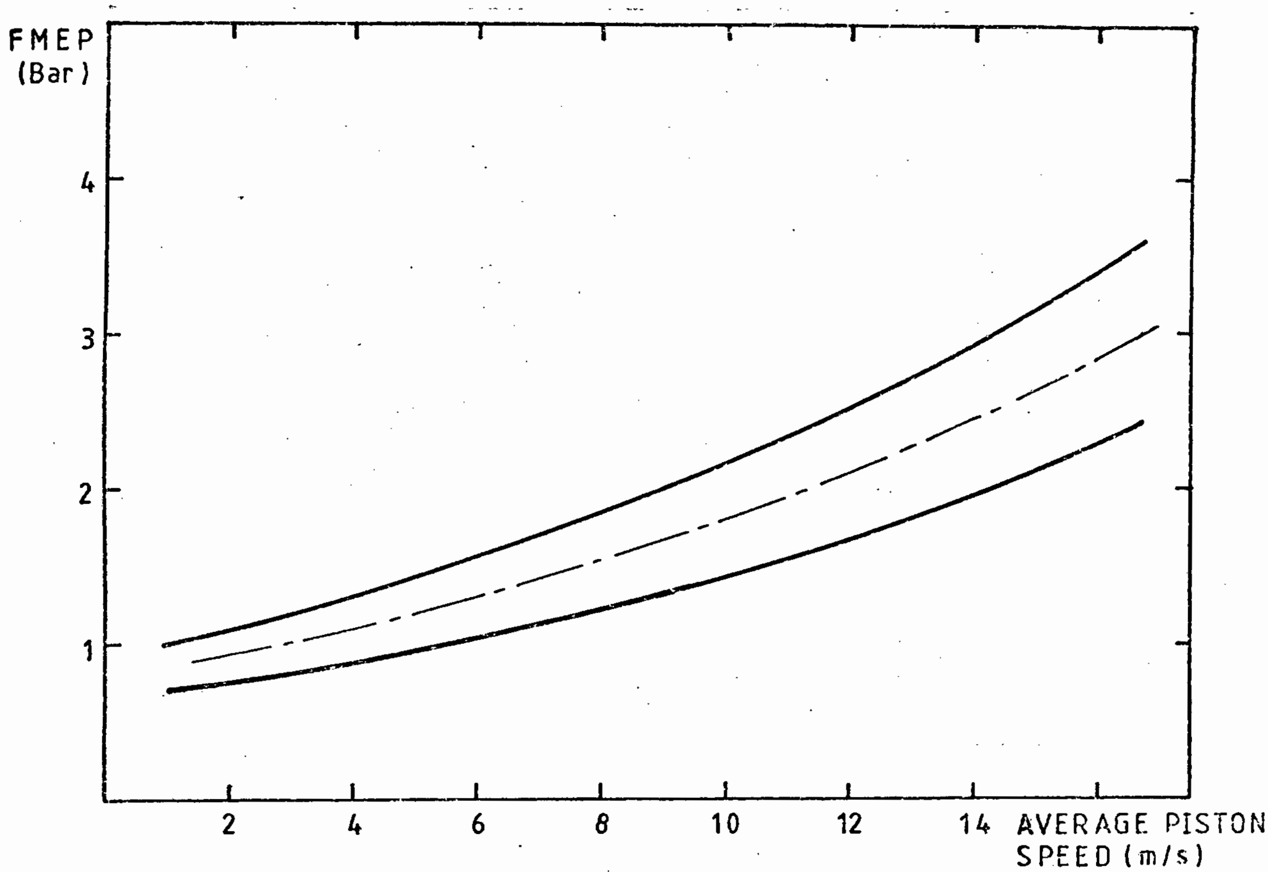
The macroscopic approach to frictional losses in a four-stroke, spark-ignition engine is well understood. Friction losses are not influenced by the type of fuel used because the method of calculation is based on the results obtained from motoring tests (i.e. engines being driven without firing). The accuracy of this method is generally accepted (44).

The estimation of frictional losses for the purposes of this thesis is based on methods given by Taylor (45). Auxiliary losses are included in the estimate.

It was found that frictional losses, when expressed as a loss in mean effective pressure, are independent of engine size, because friction is proportional to the cylinder wall area and bearing areas. The effect of compression ratio is small enough to be neglected (44, 46).

Figure 3.5 shows this relation for the average automotive spark-ignition engine. The uncertainty is given by Taylor as ± 20% (47).

The additional loss due to auxiliaries, when expressed as a mean effective pressure, could be expected to be proportional to the square of some speed function. Figure 3.6 shows auxiliary losses as a function of piston speed. This implies that the



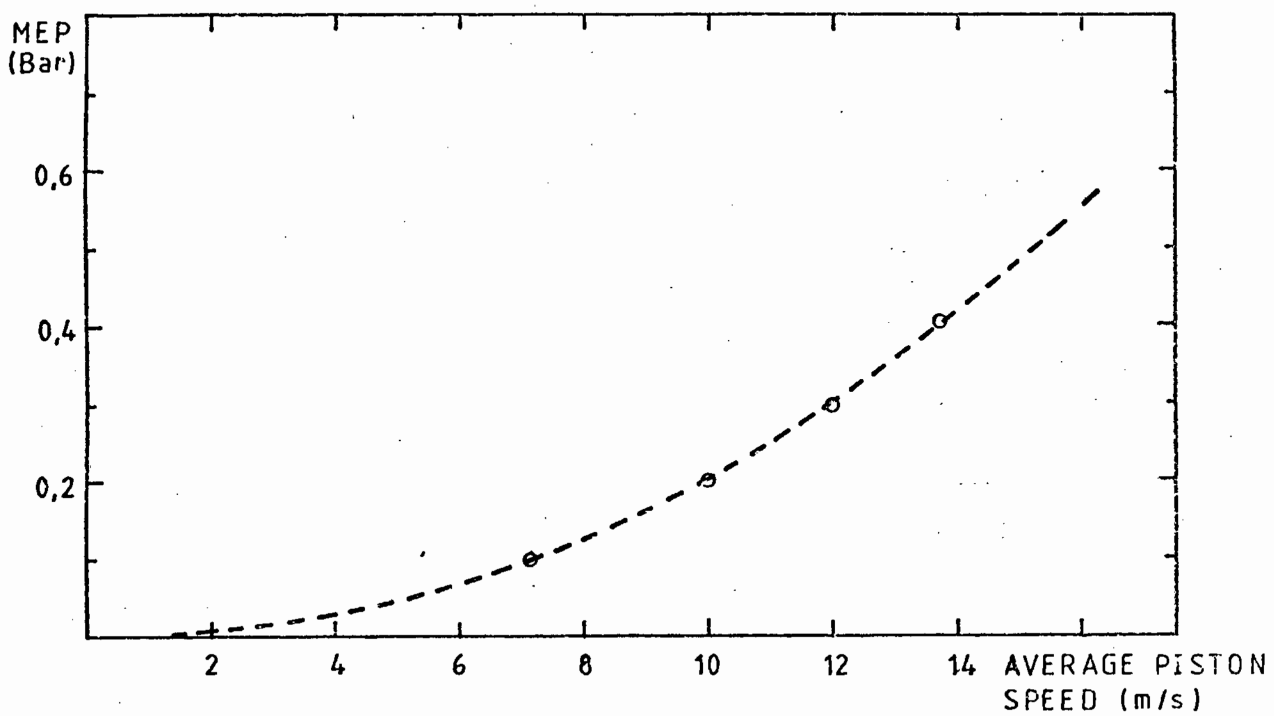
3.5 TYPICAL FRICTION LOSS FOR AUTOMOTIVE SPARK IGNITION ENGINES

Data based on Ref 47

Equation of line:

$$FMEP = 5,82 \times 10^{-12} (NS)^2 + 1,25 \times 10^{-6} (NS) + 0,91$$

Range in FMEP $\pm$ 20%



3.6 AUXILIARY FRICTION LOSS

Data based on Ref 48

Equation of line:

$$MEP = 2,22 \times 10^{-6} (NS)^2$$

auxiliary friction is a function of both engine speed and engine size which is reasonable since the operating speed generally decreases as size increases whilst the auxiliary friction mean effective pressure remains approximately constant.

Combining Figures 3.5 and 3.6:

$$FMEP = 8{,}04 \times 10^{-12} \, N^2S^2 + 1{,}25 \times 10^{-6} \, NS + 0{,}91$$

with a range of $\pm$ 20%.

3.5 CONSTANTS

Apart from the required input constants, bore, stroke, number of cylinders and rated engine speed the values of the following constants are required:

3.5.1 Heat of Combustion "Qc"

The heat of combustion of methanol is taken as 19925 KJ/kg (Appendix B, Section 2). This is the value for liquid fuel to gaseous products at 25°C.

3.5.2 Inlet Air Specific Volume "v_a"

The inlet air specific volume is defined by the ambient conditions. As was stated in Section 1.3, the estimated performance rating is according to DIN 70020 Part 6 (4), which stipulates the ambient pressure as 1,013 Bar and ambient temperature as 20°C. This defines the inlet specific volume as 0,830 m^3/Kg.

3.5.3 Stoichiometric Air-fuel Ratio "Fs"

It is convenient to discuss air-fuel ratios in terms of air fuel equivalence ratio (AFER). The actual air-fuel ratio is then computed from the product of AFER and the stoichiometric air-fuel ratio given as 6,55 for methanol, (Appendix B, Section 1).

3.6 ERROR DUE TO SIMPLIFYING ASSUMPTIONS

At this stage sufficient information has been gathered to calculate the shaft power output and overall efficiency. However, the influence of factors such as ignition timing, combustion chamber design etc., has been ignored. A short discussion of the factors that might significantly affect performance follows.

3.6.1 Ignition Timing

The data from which the indicated efficiency for methanol was extracted (Table 3.1 and 3.2) was based on minimum advance for best torque (MBT) timing. In practice it is sometimes necessary to slightly retard the ignition timing to suppress detonation or reduce exhaust pollutants. The effect of small retard angles is usually insignificant to indicated efficiency. In one instance retarding the ignition timing 10° from MBT was found to reduce the methanol indicated efficiency from 37% to 36% (28). It is unlikely that the actual timing of a methanol engine would need to deviate from MBT by as much as 10°.

3.6.2 Combustion Chamber Shape

Based on the trends observed with petrol one would expect that the combustion chamber shape would have a small effect on the absolute value of indicated efficiency and a significant effect on the detonation-limited maximum compression ratio.

A report on an investigation with two very different combustion chamber designs using methanol was published by Pefley et al (29). It was found that the indicated efficiency of a cylindrical flat head design was 36% whilst that of an inverted top hat "squish" design was 36,5%.

No reports on the effect of combustion chamber shape on compression ratio for methanol were found. However there is no reason why the findings for petrol should differ from those of methanol, in that swirl or "squish" both improve the rate of flame propagation (independent of flame speed) and thus reduce the tendency for detonation. (49)

3.6.3 Mixture Preparation

The problems of efficiently mixing methanol and air are discussed in Appendix B, Section 4.

The effects of poor mixture preparation are incomplete combustion due to the slow burning of large droplets of liquid fuel (50), and unequal cylinder to cylinder air-fuel ratio (51). A drop in indicated efficiency could be expected as a result, but a quantitative assessment applicable to all engines is almost impossible, since the extent of the problem depends entirely on the design of the particular carburettor and inlet manifold.

It is probable that the effects of poor mixture preparation would be most apparent at wide open throttle and low engine revolutions, since the manifold vacuum is minimal (see Appendix B, Section 4) and the air speed is relatively low.

3.7 SUMMARY AND ERROR ANALYSIS

The calculation of shaft power and overall efficiency of a methanol engine is summarized below.

Constants

D, S, C, N_r, Q_c, v_a, F_s

Independent Variables

N : upper limit N_r

r : upper limit $r = f(D)$ (function of D) (Section 3.2.1)

AFER : upper limit 1,7 (Section 3.2.2)

Dependent Variables

e_i = f (r, AFER) (Section 3.2)

e_v = f (N. N_r) (Section 3.3)

FMEP = f (N,S) (Section 3.4)

From this, Power "P" and overall efficiency "e_o" can be calculated. (Section 3.1).

The analytical formulae are given collectively in Appendix E, with correction factors for the units given in the Nomenclature.

An analysis of the error in these calculations was carried out. It was necessary to therefore establish the range of probable error in each variable.

3.7.1 Indicated efficiency

The indicated efficiency is computed from Figures 3.1 and 3.2. The standard error of the estimate was determined for each curve. Assuming the range of probable error to be sufficiently spanned by four standard errors of the estimate, (actually 90% of the total range would be theoretically spanned), the error in indicated efficiency was calculated as:

Error in indicated efficiency: $\pm$ 1,86 percentage units

A further source of probable error was estimated from the discussion in Section 3.6.

Thus the total error in indicated efficiency: Approximately $\pm$ 6%.

3.7.2 Volumetric Efficiency

The standard error of the estimate for volumetric efficiency was computed as 3,04 percentage units. Making the same assumptions as in Section 3.7.1,

The error in volumetric efficiency : $\pm$ 7%.

3.7.3 Frictional Losses

The frictional losses were determined in Section 3.4 as $\pm$ 20%.

3.7.4 Error Analysis

From the above data, the error in IMEP can be calculated as $\pm$ 9%.

In order to express the error in power and overall efficiency as a percentage, it was necessary to assume that

$$\text{FMEP} \doteqdot 0{,}1\ \text{IMEP}$$

Then the error in Power : $\pm$ 10%.

The error in overall efficiency : $\pm$ 6%.

CHAPTER FOUR

EXPERIMENTAL WORK

Three engines, two from passenger cars, and one from a truck, were converted to operate on methanol and bench tested. The range of size represented by these engines falls within the probable commercial size range suitable for operation with methanol by spark ignition. The engines were converted from petrol or diesel usage and although the conversion itself is incidental to this thesis, it was a necessary step in the experimental work, and a brief description is therefore included.

4.1 EXPERIMENTAL METHOD

Each engine was tested on a Heenen Froude eddy current bench dynamometer. The dynamometer was rated at a maximum torque of 890 Nm below 2400 rev/min, and a maximum Power of 224 kw above this speed. The engine speed was held constant during testing to within [illegible] 20 rev/min. The error in the measurement of torque reaction of the dynamometer was less than 2%.

The fuel consumption was measured by observing the time taken for a known volume of fuel to flow through a burette. The experimental error was minimized by using a burette of sufficient volume so that the time measurement was [illegible] thirty seconds or longer.

The air mass flow was calculated from the gauge pressure measured at the vena contracta of a plain, thin wall pipe intake. The coefficient of discharge is related to Reynolds number by means of a calibration curve given by Dall (52). Accuracy of measurement could be maintained by the suitable choice of entrance pipe diameter. The effect of engine pulsations on the measurement was assumed to be negligible. This assumption was checked by comparing the air-fuel ratio calculated from this air flow measurement to the air-fuel ratio determined from the exhaust analysis. The two methods agreed to within 3%.

In addition to the measurement of engine speed, torque, fuel and air flow, the engine oil pressure and coolant temperature was monitored.

It was proposed to conform to DIN 70020 Part 6 (4). However for the sake of safety, the engine fan was not fitted, since necessary adjustments had often to be made on the engine whilst running. This deviation was not considered to be a serious source of error since the tests were conducted at wide open throttle, under which conditions the fan power represents about 1% of the total power output (53). The specification was followed in all other respects.

4.2 ENGINE SELECTION

The basic specification of each engine, and a brief description of the conversion and the objectives of the conversion, is given here. A more detailed account of the engine conversions is given in Appendix F.

4.2.1 Ford Cortina 2000 OHC

The modifications to this engine were limited to carburettor jet changes and adjustments to the ignition timing. Although it was intended to match the methanol AFER to the original petrol AFER, this could not be done because the volume flow rate of methanol was greater than that of petrol to such an extent that the drilled carburettor passages were controlling the flow.

The relevant engine specifications are:

Engine Type	:	4 in a line
Bore	:	90,82 mm
Stroke	:	76,95 mm
Max. Rated Revs.	:	5200 rev/min
Compression Ratio	:	9,2:1

4.2.2 Volkswagen Passat 1.6 l.

The objective of this conversion was to explore the benefits associated with the use of higher compression ratios which are possible with methanol operation. It was intended to adjust the carburettor air-fuel ratio so as to match the converted engine output to the rated petrol output, but this was not completely successful. Although the methanol AFER was set as lean as possible with the carburettor supplied, the converted engine power output was slightly greater than planned (see Appendix F, Section 2.2).

The distributor mechanical advance mechanism and ignition static timing were optimized. A modified inlet manifold with extra engine water jacketing was also fitted to improve the engine response to transient load demands (refer Appendix B, Section 4.1).

The relevant engine specifications are:

Engine Type	:	4 in a line
Bore	:	79,5 mm
Stroke	:	80 mm
Max. Rated Revs.	:	5600 rev/min
Compression Ratio	:	12:1 (originally 8,2:1)

4.2.3 Mercedes Benz OM 355

Originally a diesel engine, this conversion necessitated fitting spark plugs in place of the diesel injectors, an ignition distributor in place of the injector pump, a custom built fuel supply system and a throttle butterfly valve. The compression ratio was also lowered. It was possible to match the converted engine output almost exactly to the original diesel rated power output but the overall efficiency was lowered as a result of the conversion.

The relevant engine specifications are:

Engine Type	:	6 in a line
Bore	:	128 mm
Stroke	:	150 mm
Max. Rated Revs.	:	2200 rev/min
Compression Ratio	:	10:1 (originally 16:1)

CHAPTER FIVE

EXPERIMENTAL RESULTS

The results of the engine tests are presented graphically in figures 5.1 to 5.4. The tabulated results are listed in Tables 5.1 to 5.3.

5.1 FORD CORTINA 2000 O.H.C.

The engine performance curves are shown in figure 5.1. Evidence of the fuel restriction at the "top end" (Section 4.2.1) is apparent in the levelling-off of power. This can also be seen in the air-fuel ratio curve shown in figure 5.4, where the AFER becomes steadily leaner as the engine speed increases.

Although the shape of the efficiency curve is quite normal for a conventional engine it is perhaps surprising in view of the unusual air-fuel ratio. Presumably an increase in indicated efficiency which would result from the lean air-fuel ratio at high engine speeds was offset by the reduction in mechanical efficiency caused by the ratio of rising frictional losses to the declining I.M.E.P. as engine speed increases.

5.2 VOLKSWAGEN PASSAT 1,6 l.

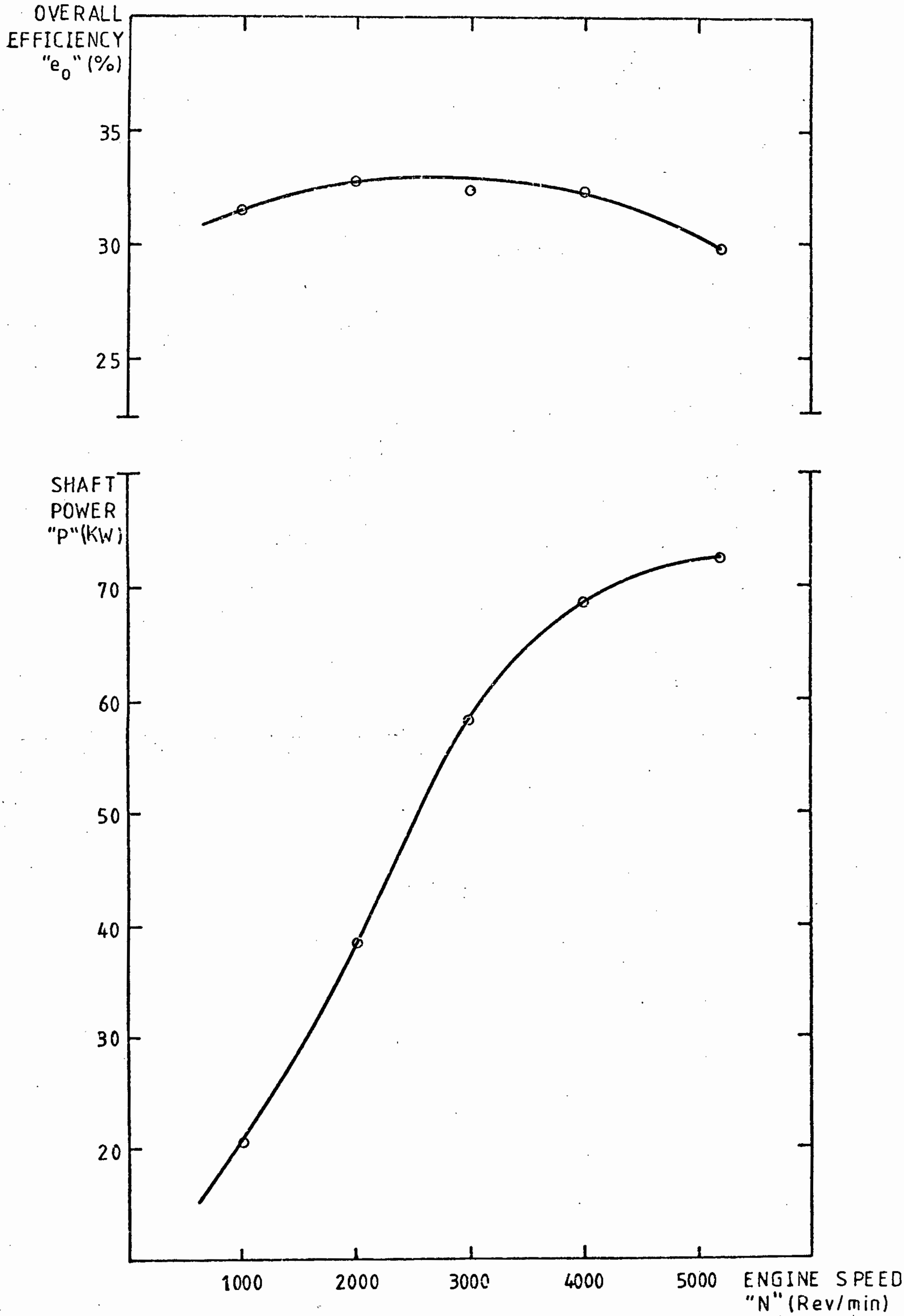
The performance curves for the Passat engine are shown in figure 5.2 Over the entire speed range the output was about 2 to 3 kw greater than the rated petrol output. This difference arose from the difficulty that was experienced in controlling the carburettor air-fuel ratio under all conditions (Section 4.2.2).

The air-fuel ratio curve is shown in figure 5.4. Although the mixture was only slightly lean from stoichiometric over the engine speed range, the overall efficiency was found to be high; the maximum recorded efficiency was 34,3% which compares to an equivalent figure of 31% for a standard petrol engine which was reported from Volkswagen S.A. Thus at 3000 rev/min the methanol engine was producing 8% more power than the petrol engine at an 11% improvement in energy consumption. The volumetric fuel consumption of the methanol engine would be about twice that of the petrol engine because of the greater specific heat of combustion of petrol over methanol. (Appendix B, Section 2).

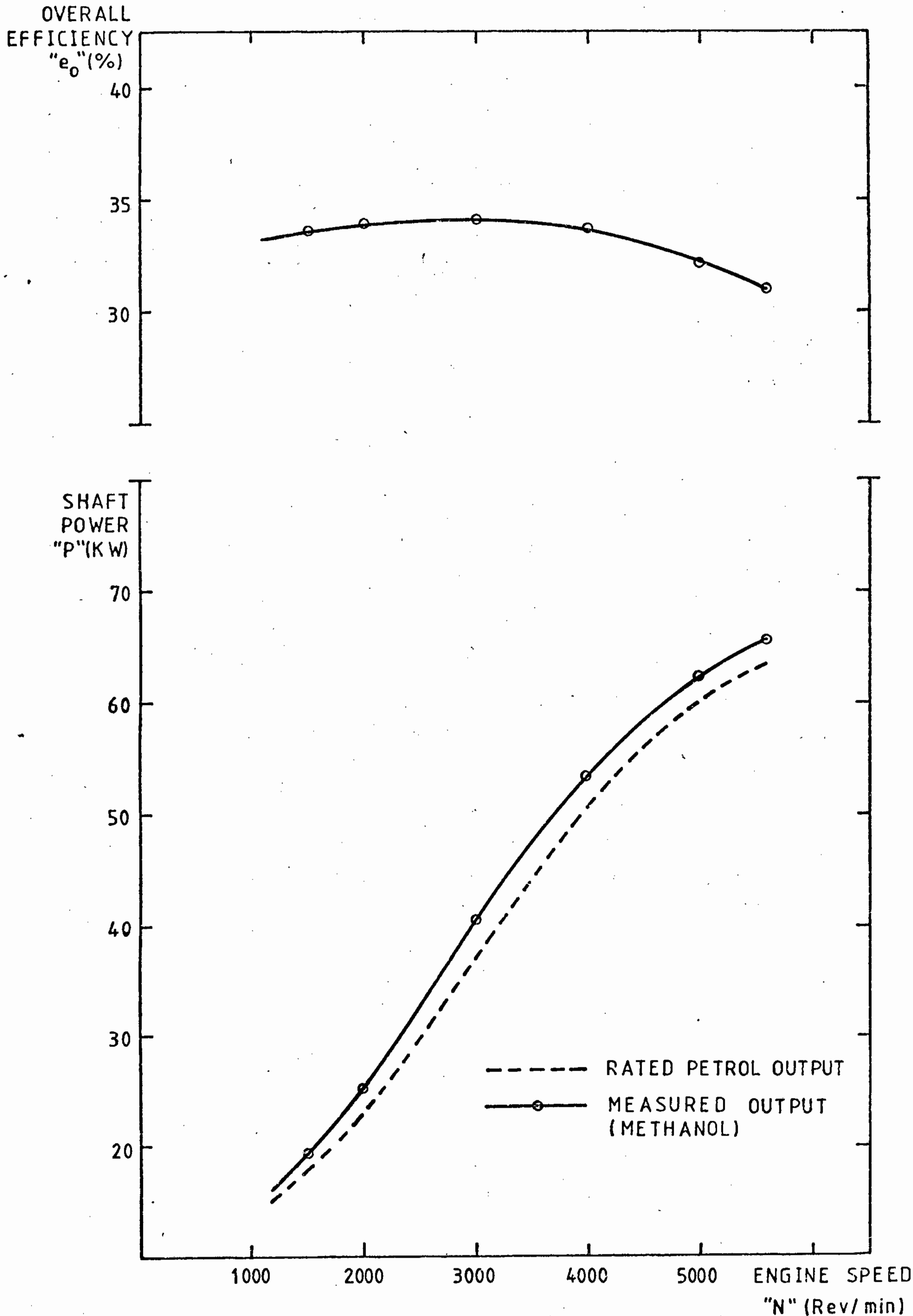
5.3 MERCEDES BENZ OM 355

The performance curves for the converted OM 355 engine are shown in figure 5.3. It can be seen that the engine output was closely matched to the rated diesel output over the whole engine speed range.

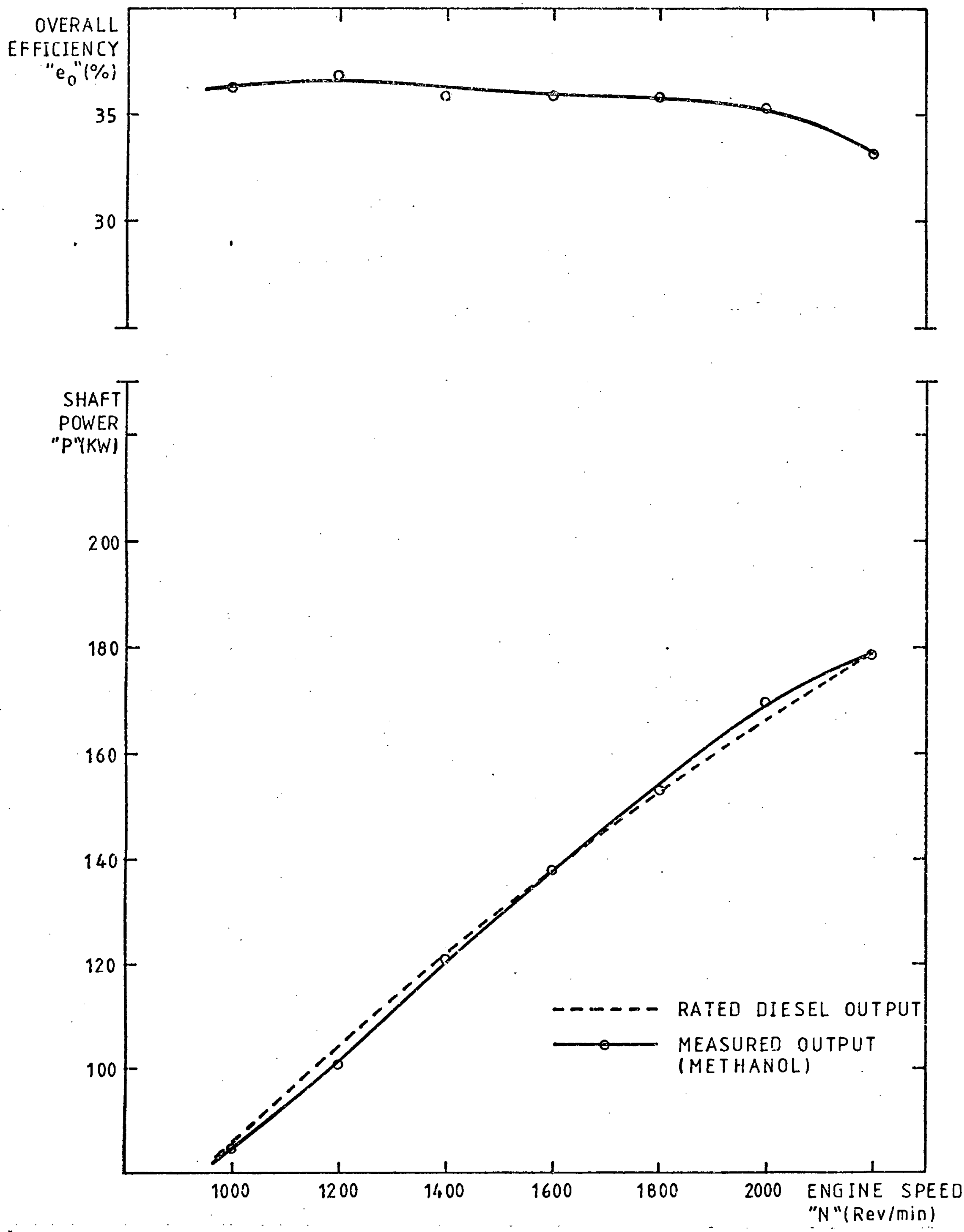
The air fuel ratio curve, shown in figure 5.4 was found to be approximately constant at about AFER 1,2. This gave a maximum overall efficiency of 36,7% at about 1200 rev/min.



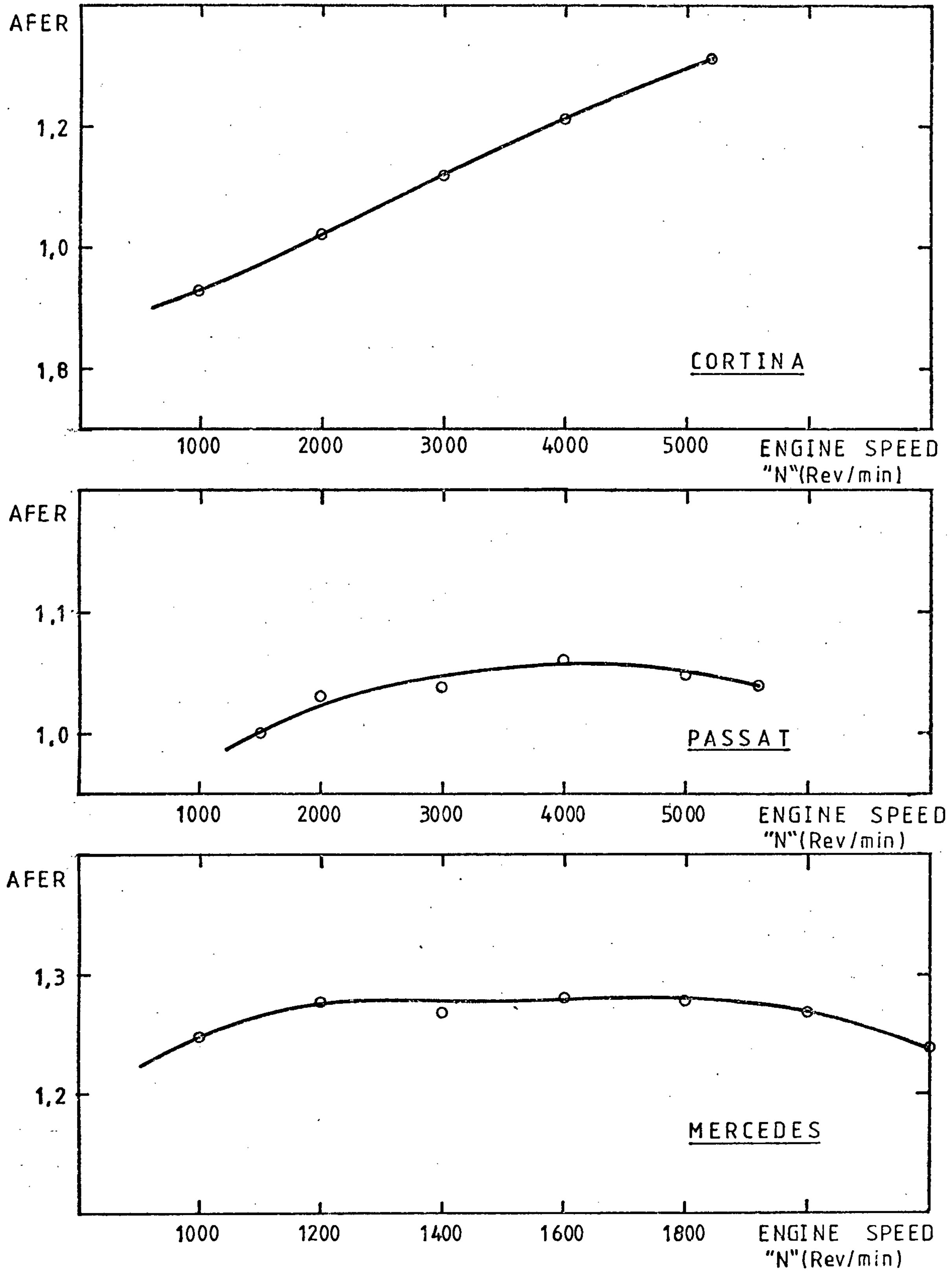
5.1 CORTINA PERFORMANCE AT WIDE OPEN THROTTLE USING METHANOL. Compression Ratio 9,2:1; Ignition Timing MBT; Test Spec. DIN 70020 Part 6 (without engine fan)



5.2 PASSAT PERFORMANCE AT WIDE OPEN THROTTLE USING METHANOL COMPARED TO PETROL. Compression ratio : 8,2:1 Petrol, 12:1 Methanol; Ignition Timing MBT; Test Spec. DIN 70020 Part 6 Rated Data obtained from Volkswagen (SA)



5.3 MERCEDES PERFORMANCE AT WIDE OPEN THROTTLE USING METHANOL COMPARED TO DIESEL. Compression Ratio : 16:1 Diesel, 10:1 Methanol; Ignition timing MBT; Test Spec. DIN 70020 Part 6 (without engine fan)



5.4 AIR-FUEL RATIO FOR THE CONVERTED ENGINES AT WIDE OPEN THROTTLE

CHAPTER SIX

DISCUSSION OF PREDICTED AND MEASURED PERFORMANCE

In this chapter, the measured power and efficiency of the three test engines is compared to the predicted performance. The prediction method presented in this report is also discussed against the background of other methods.

6.1 MAXIMUM COMPRESSION RATIO

It was stated in Section 3.3.1 and discussed in Appendix D that the maximum compression ratio of a methanol engine could be higher than that of a conventional petrol engine. The approximate relationship between octane rating, bore size and maximum compression ratio shown in figure 3.3 could not be verified by the results of only three test engines. However no detonation was experienced at the compression ratios that were used. In the case of the Passat and the Mercedes engines, these were slightly below the estimated maximum limit as can be seen from the table below:

	Estimated Maximum	Actual Compression Ratio
Cortina	11,6 to 12,4:1	9,2:1
Passat	12,1 to 12,9:1	12,0:1
Mercedes	10,2 to 11,0:1	10,0:1

6.2 PREDICTED PERFORMANCE MAPS

As was shown in Section 3.7, the three independent variables upon which the power and efficiency are dependent are engine speed (N), compression ratio (r) and air-fuel ratio (AFER). Since compression ratio is constant for a particular engine, the performance can be depicted graphically as a function of engine speed and air-fuel ratio.

Figures 6.1, 6.2 and 6.3 represent the performance maps for the Cortina, the Passat and the Mercedes engine respectively. The curves of shaft power and overall efficiency are plotted as a function of engine speed at various constant air-fuel ratios. The compression ratio used for each calculation was the same as that used for each engine experiment.

It can be readily seen that as the engine speed increases, the power tends to level off and the efficiency tends to droop in every instance. This trend is due to the combined effect of increasing frictional losses with increasing speed, and a peak in volumetric efficiency at about 80% of the rated engine speed.

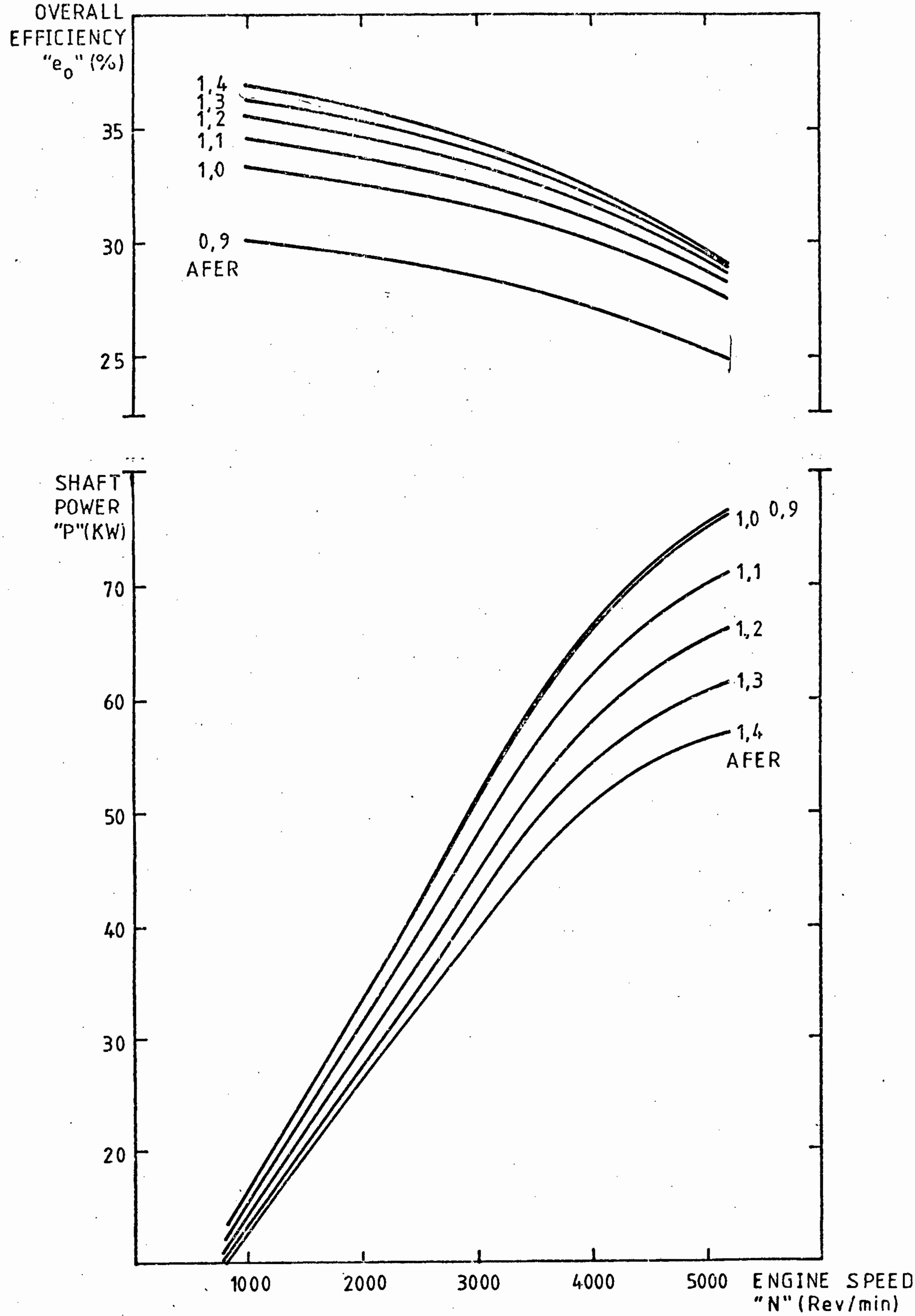
It is also interesting to note that there is a very significant drop in efficiency as the air-fuel ratio becomes richer than stoichiometric. The peak power occurs between AFER 0,95 and AFER 0,94. The reason that peak power does not occur at AFER 1,0 is due, in practice, to factors such as poor mixture preparation and distribution in each cylinder, with the result that a slight surplus of fuel is required in order to consume all the available oxygen. This is reflected in the theoretical analysis because the theory is based on actual measured data (Tables 3.1 and 3.2).

It is also evident that the relative droop in overall efficiency with engine speed is not as pronounced for the Mercedes engine as it is for the smaller engines. This is because the operating range of engine speed relative to the rated engine speed is less for Mercedes than for the automobile engines.

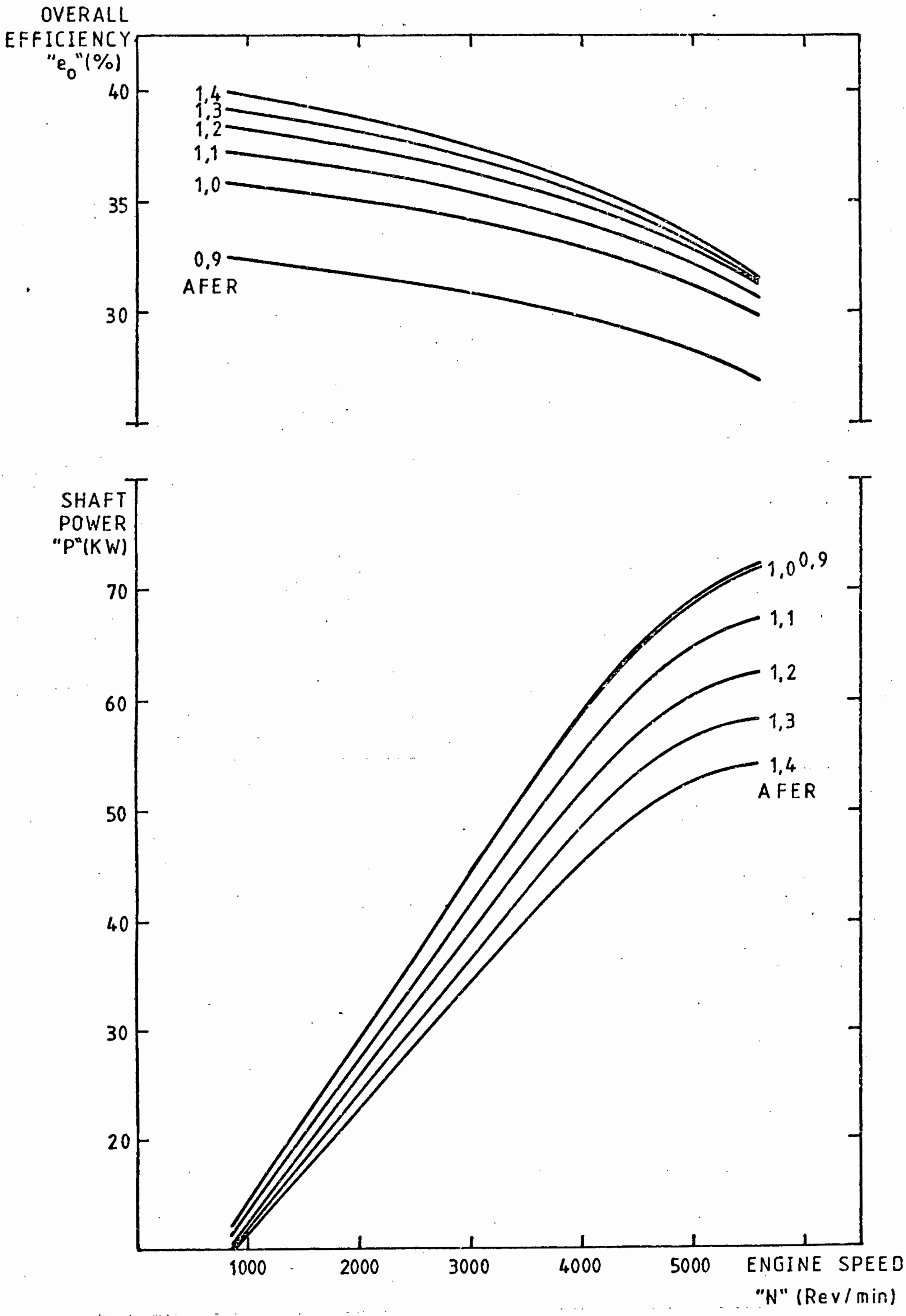
6.2.1 Application

The manner in which the prediction theory could be applied to a given situation would depend on the problem requirements.

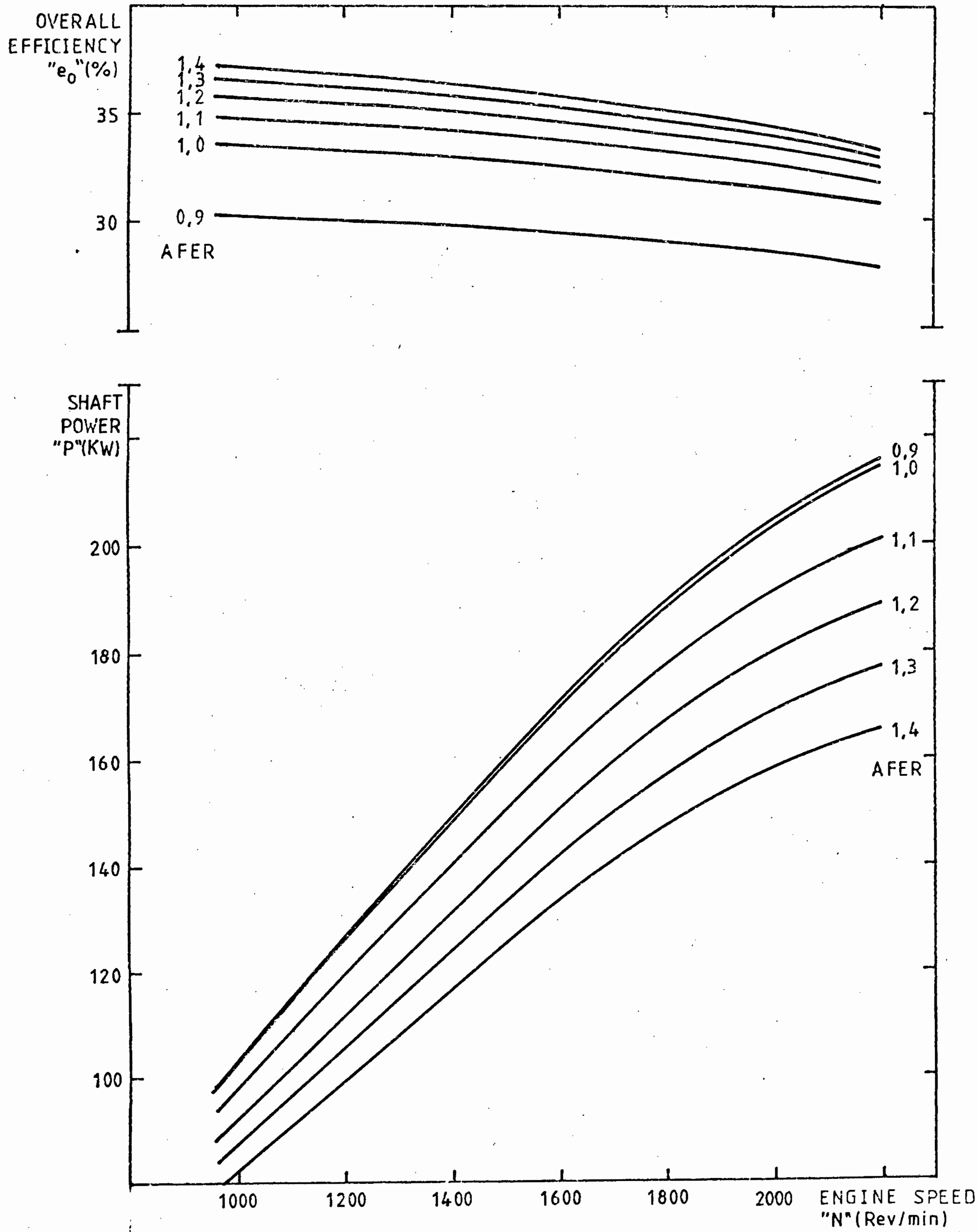
Assuming that it was desired to match the converted engine output to the rated diesel/petrol output, a map similar to that shown in Figures 6.1 to 6.3 would be required. The desired power versus engine speed could then be superimposed on the predicted power graph and the required air-fuel ratio could be



6.1 CORTINA : PREDICTED PERFORMANCE MAP - WIDE OPEN THROTTLE. Compression Ratio 9,2:1



6.2 PASSAT : PREDICTED PERFORMANCE MAP – WIDE OPEN THROTTLE. Compression Ratio 12:1



6.3 MERCEDES : PREDICTED PERFORMANCE MAP - WIDE OPEN THROTTLE. Compression Ratio 10:1

determined, and hence the overall efficiency.

If required, the maximum power and corresponding efficiency could be calculated directly at about AFER 0,945.

The maximum efficiency would depend on the leanest air-fuel ratio that could be tolerated by the engine without misfiring. This would have to be estimated. The mixture preparation and manifold distribution (Appendix B, Section 4) are factors that would significantly affect the lean-burn limit. From this estimate, the power corresponding to maximum efficiency could also be calculated.

6.3 PREDICTED AND MEASURED PERFORMANCE

For the purpose of comparing the experimental results for each engine to the predicted results, the measured air-fuel ratios for each engine, given in Figure 5.4 were used as a basis for the theoretical calculation. The results, theoretical and experimental, are compared on the same axes of power and efficiency versus engine speed in Figures 6.4, 6.5 and 6.6 for the Cortina, Passat and Mercedes engines respectively.

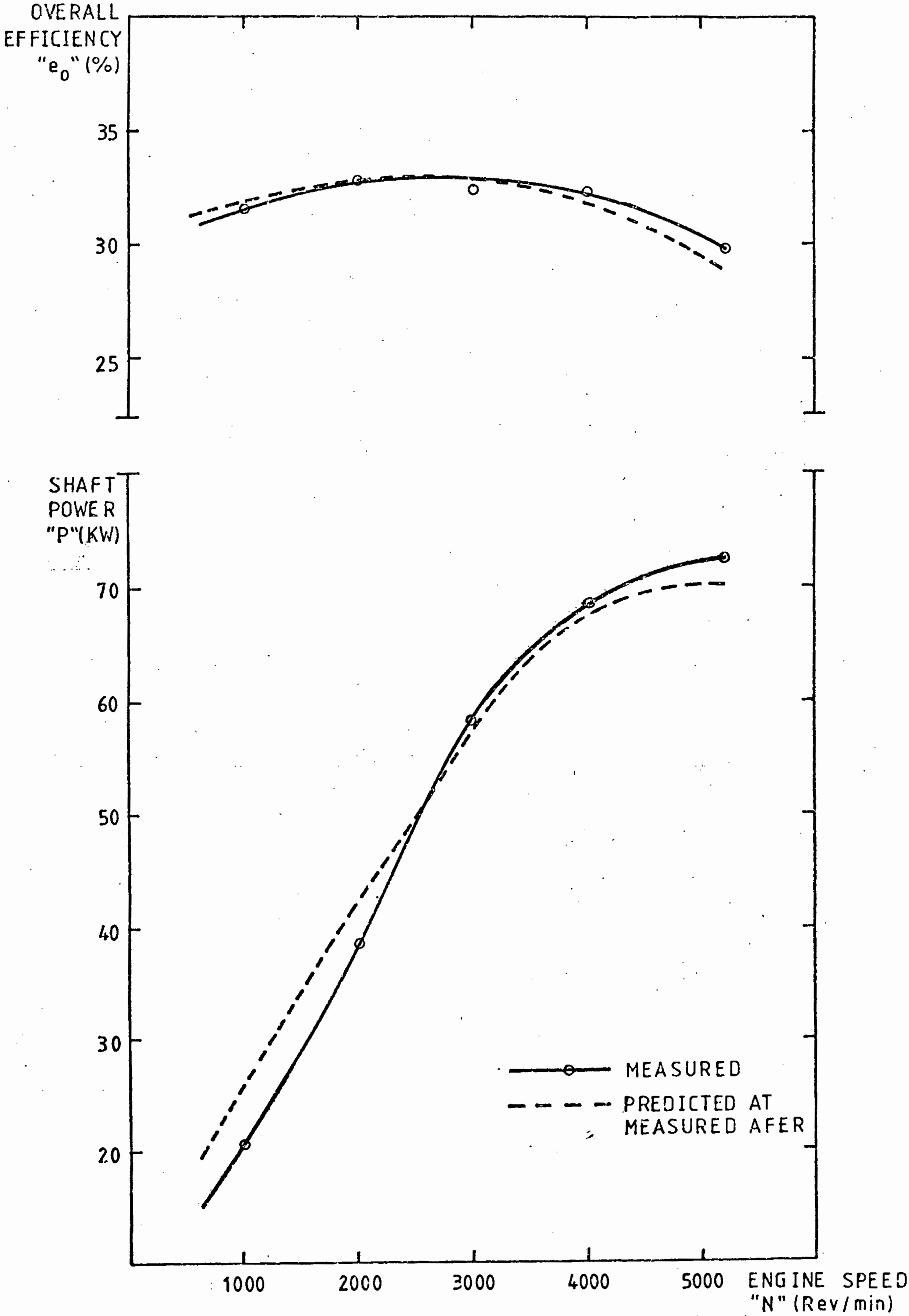
Although the measured and predicted results do not match exactly, the general trend in both power and efficiency was found to agree. In order to compare the pairs of curves more effectively, the difference between the theoretical and the measured values was expressed as a fraction of the theoretical value and plotted against the engine speed, expressed as a fraction of the rated speed. Overall efficiency is compared in this manner in Figure 6.7, and shaft power in Figure 6.8.

6.3.1 Efficiency Comparison

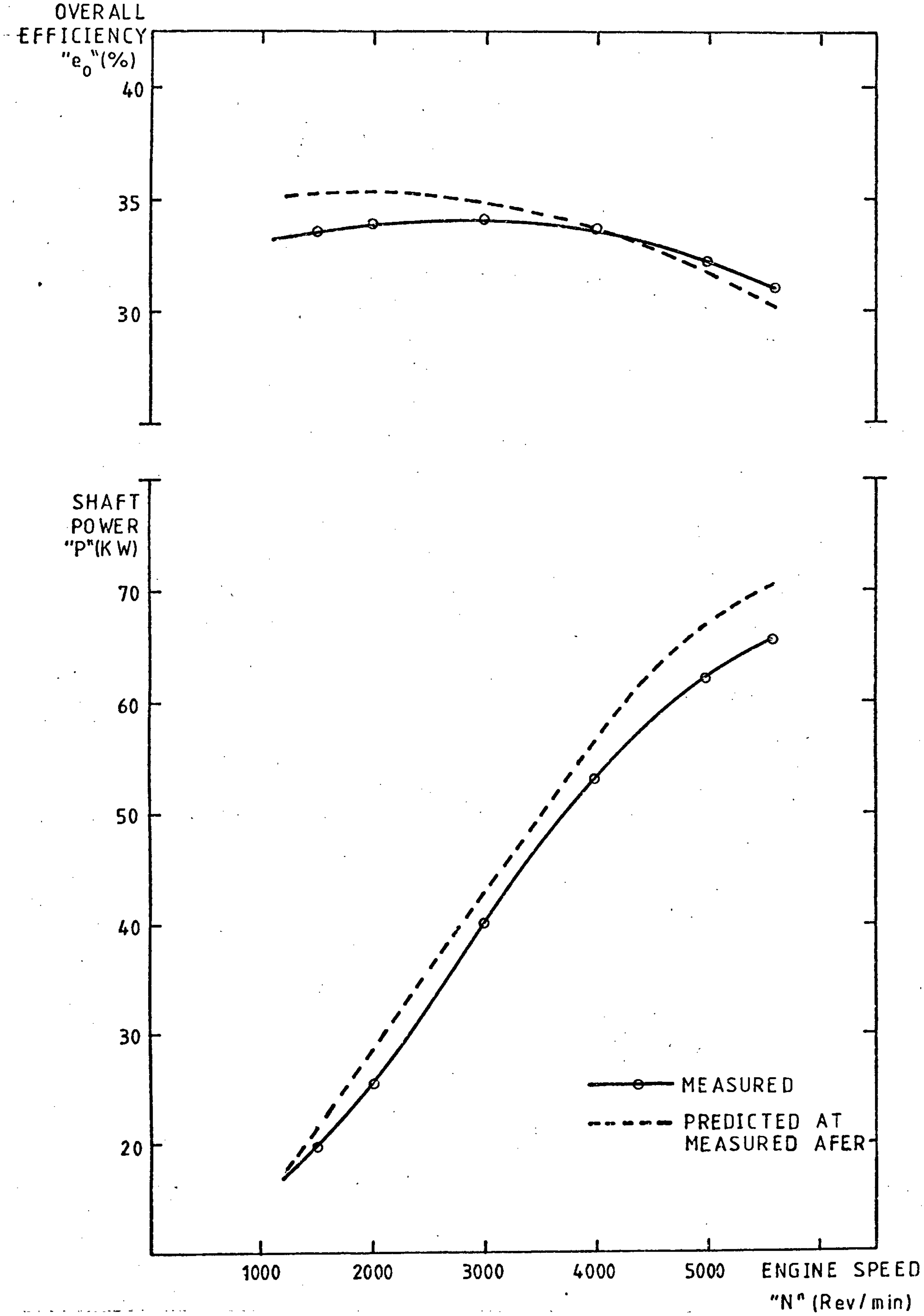
The anticipated error in the prediction of efficiency was calculated in Section 3.7 as :

Error in predicted efficiency $\pm$ 6%.

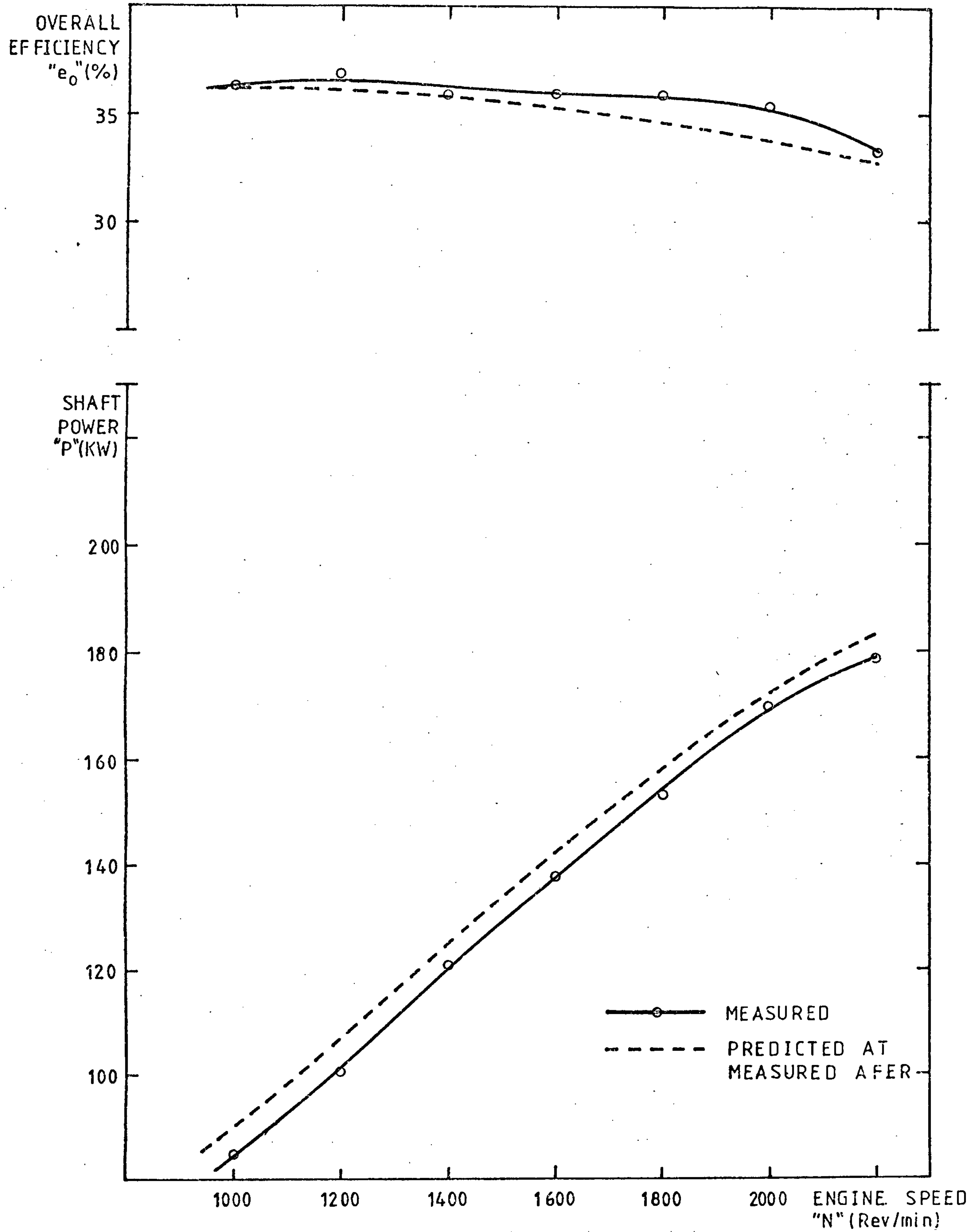
Referring to Figure 6.7 it can be seen that the experimental results lie within this predicted range.



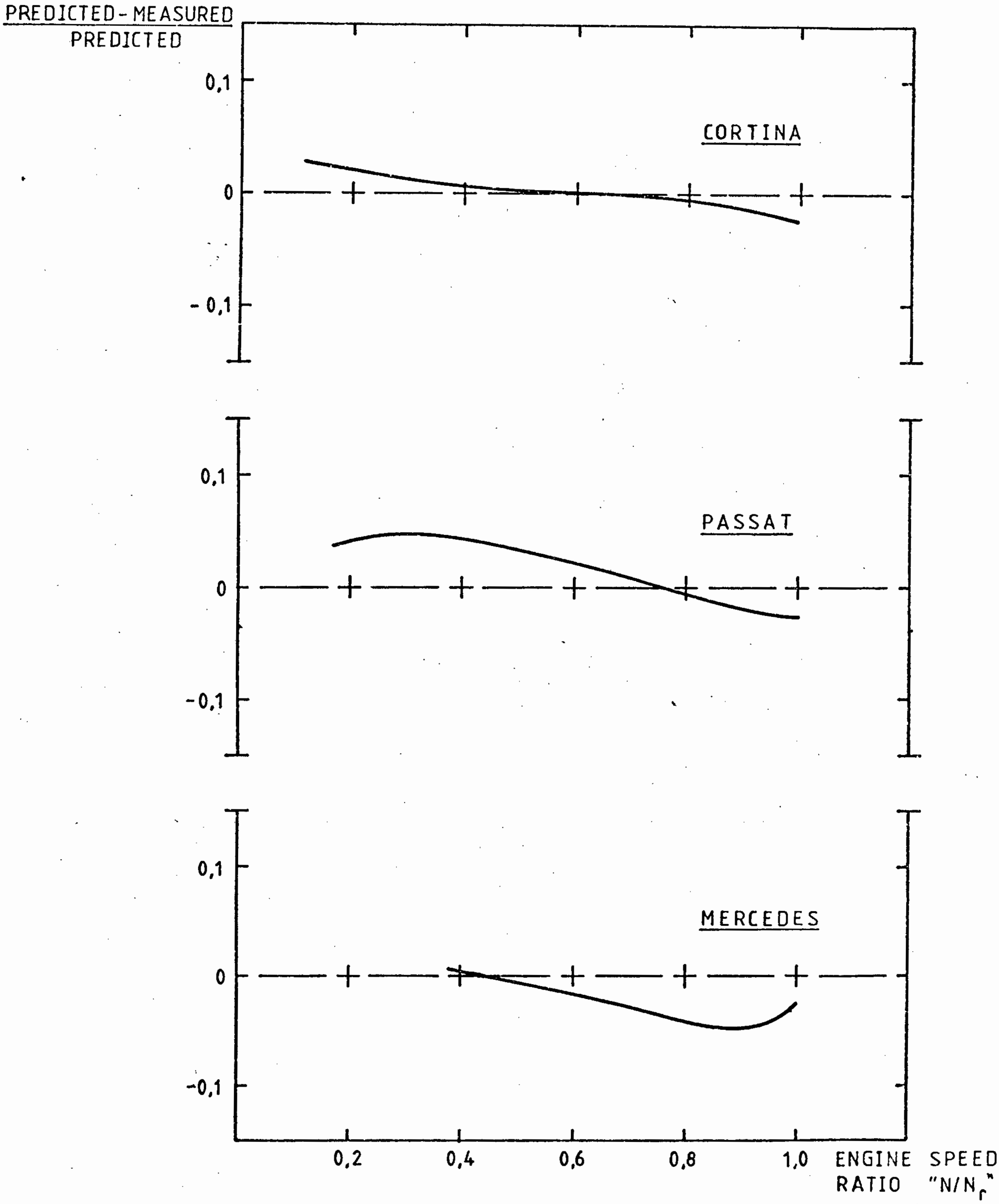
6.4 CORTINA : MEASURED AND PREDICTED PERFORMANCE AT THE SAME AFER AND COMPRESSION RATIO



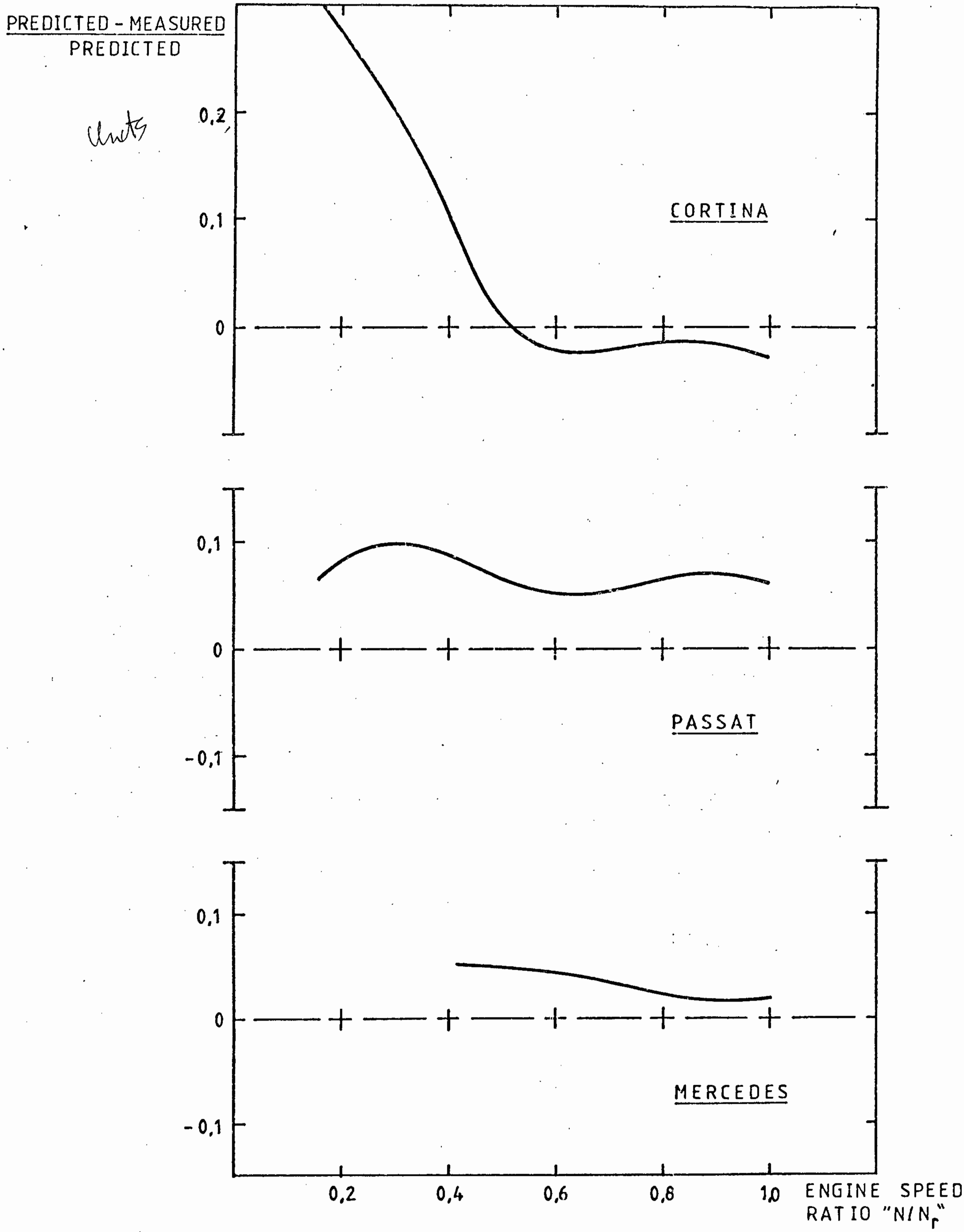
6.5 PASSAT : MEASURED AND PREDICTED PERFORMANCE AT THE SAME AFER AND COMPRESSION RATIO



6.6 MERCEDES : MEASURED AND PREDICTED PERFORMANCE AT THE SAME AFER AND COMPRESSION RATIO



6.7 MEASURED AND PREDICTED COMPARISON FOR OVERALL EFFICIENCY. Data based on Figures 6.4 to 6.6



6.8 MEASURED AND PREDICTED COMPARISON FOR SHAFT POWER. Data based on Figures 6.4 to 6.6

The comparison curves for the three engines show a similar trend. The predicted efficiency appears to over-estimate at low engine speeds and to under-estimate at high engine speeds. This trend was suggested in Section 3.6.3 as being caused by poor mixture preparation, the effect being most pronounced at low engine revolutions and wide open throttle. Under these conditions the manifold vacuum, which would aid fuel evaporation, is minimal and the air speed, which would aid fuel droplet fragmentation, is relatively slow.

It is probable then, that the mixture preparation is significantly affected by the engine speed. There is a temptation to generalize the results of Figure 6.7 into an indicated efficiency correction factor, but it was felt that the results of only three engine tests were insufficient to establish a quantitative relationship that was generally applicable.

6.3.2 Power Comparison

The anticipated error in the prediction of shaft power was calculated in Section 3.7 as:

Error in predicted Power $\pm$ 10%.

Referring to Figure 6.8, it is evident that at low engine speeds, the anticipated error range for the Cortina was exceeded by more than 30 %. The results for the Cortina at other engine speeds, and for the Passat and the Mercedes were found to lie within the estimated error range.

The trend that was noted for the overall efficiency, to over-estimate at low speeds and to under-estimate at high speeds, is reflected in the power comparison curves as a general negative slope. Both overall efficiency and shaft power are functions of indicated efficiency (see Section 3.7), although the relationship between power and indicated efficiency could be somewhat masked by the possible error in volumetric efficiency.

The significant difference between the measured and predicted power values for the Cortina at low engine speeds was not completely investigated. Unfortunately the equations of

Chapter 3 were only developed after the engine experiments were completed and thus the engine was already reinstalled into the vehicle chassis by the time the problem was identified. An analysis of the experimental data did however pinpoint the problem area.

The fact that the difference between the measured and predicted overall efficiency was about 2% at 1000 rev/min indicated that the actual measurements of power were probably correct, since the overall efficiency is determined by the ratio of shaft power to fuel flow, and it is unlikely that an error in power measurement would coincide with a proportionately similar error in fuel flow measurement. However, it was noticed whilst computing the error analysis of Section 3.7, that shaft power is directly proportional to volumetric efficiency whereas the overall efficiency is relatively insensitive to changes in volumetric efficiency. The volumetric efficiency was therefore calculated from the experimental data and found to be 55% at 1000 rev/min. The predicted volumetric efficiency at this engine speed was 82%.

This significant difference between theory and experiment at one engine speed was possibly caused by a resonant condition in the engine intake system. The plain intake air flow meter which formed a part of the intake system would have certainly been instrumental in inducing such a condition.

Another possible explanation, that the throttle valve was accidentally not fully open, is unlikely since this would be accompanied by a drop in overall efficiency due to the associated pumping losses.

6.4 OTHER PREDICTION METHODS

There is very little reference in the literature to comparable predictions of the performance of a methanol engine.

The use of a digital computer by Browning et al (21) was found to produce good agreement in trends of indicated power, indicated thermal efficiency and emissions versus equivalence ratio when compared to experimental results. The absolute values of their predictions were however found to differ from the measured values by as much as 6%. This is no better than the estimated range of error in indicated efficiency (discussed in Section 3.7.1) of this thesis. (Subsequent improvements in the computer model gave valuable insight into the mechanism of pre-ignition and the formation of exhaust emissions (54), thereby demonstrating the type of analysis that is well suited to computer techniques.)

The method used in this thesis for the calculation of power and efficiency was in accordance with the conventional text-book approach for estimating the performance of petrol and diesel engines. However the method of estimating the indicated efficiency on the basis of reported results from other methanol engines, and the approximate estimate of the detonation limited compression ratio for methanol spark-ignition engines was not encountered anywhere in the available literature.

CHAPTER SEVEN

CONCLUSIONS:

7.1 SUMMARY OF THE THEORETICAL DERIVATION

A formula was derived whereby the shaft power and overall efficiency of a conventional spark-ignition engine operating on methanol could be predicted. The derivation was based on a generalization of indicated efficiency for methanol engines in terms of compression ratio and air-fuel ratio; of volumetric efficiency in terms of the ratio of engine speed to rated speed; and of mechanical friction in terms of average piston speed. A certain degree of approximation was inevitable as a result of the generalization, but could be quantified:

Probable error in the prediction of overall efficiency $\pm$ 6%

Probable error in the prediction of shaft power $\pm$ 10%

An estimation of the detonation limited compression ratio of a methanol engine as a function of bore size was also determined.

7.2 SUMMARY OF THE EXPERIMENTAL RESULTS

Three engines were converted to operate on methanol by spark ignition and bench tested. In terms of bore size, they represent a cross-section of the likely commercial engine size range:

Passat	79,5 mm bore
Cortina	90,8 mm bore
Mercedes	128,0 mm bore

The measured shaft power and overall efficiency at wide open throttle was compared to that predicted from theory. In general, the measured results were found to agree with the predicted values within the anticipated error margin. However in one instance, a significant difference between theory and experiment was noted. This was traced to an unusually low measured volumetric efficiency which was thought to be caused by resonance in the air intake system.

7.3 THESIS OBJECTIVES

In all respects the objectives of the thesis were met:- The estimation of rated power and efficiency of a methanol engine, given only the fundamental engine parameters, was possible. A map of the predicted engine performance could be produced in a few minutes with the aid of a programable calculator.

It could be argued that the approximation which is inherent in the prediction formulation renders the results somewhat useless. This argument is especially convincing when a very much more detailed analysis could be undertaken with computer modelling techniques if the constraint limiting foreknowledge of the engine design details was lifted. However, there are situations where a quick answer is needed and few details are available, and under such "unscientific" circumstances an approximate answer is acceptable, especially if the estimate is based on sound theory.

7.4 SCOPE FOR FURTHER WORK

It was the objective of this thesis to estimate the rated performance of a methanol engine. The rated performance was defined as the wide-open-throttle power and efficiency according to DIN 70020 Part 6 (4). In most automotive situations, engines operate only occasionally at wide open throttle. There is therefore scope for a useful extension to this thesis in the form of a prediction of part-load performance. With this information, the complete engine operational map could be produced.

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APPENDIX A

THE METHANOL ENGINE - POSSIBLE ALTERNATIVES

The manner in which methanol is utilized as a fuel for the internal combustion engine need not necessarily conform to that of petrol or diesel, that is, using the same methods of introducing the fuel to the combustion engine and igniting it. However, the motor car and truck industry is one of the largest single industries in the world, and changes must inevitably be gradual. It is probable that in the initial stages at least, the methanol engine will be a compromise between making the optimum use of the fuel and making as little change to the traditional petrol or diesel engine as possible.

The chemical properties of methanol relate more closely to petrol than to diesel. The conventional petrol engine can run on methanol with the only change being carburettor jet sizes, (material incompatabilities could also occur). However the potential fuel economy and low exhaust emissions are not fully realized by this simple modification, and driveability is almost certainly impaired. A great variety of improved methods for utilizing methanol in the internal combustion engine have been investigated and reported:

a) Use of methanol in conventional spark ignition engines. Improvements in economy and exhaust emissions are due to raised compression ratios and improved air/fuel mixture preparation. (55)

b) Spark ignition of methanol with hydrogen enrichment. The hydrogen is obtained by dissociating a small fraction of the methanol into carbon monoxide and hydrogen. (56)

c) Flash boiling injection of methanol into the cylinder prior to ignition, the object being to create a homogeneous air/fuel mixture. (57)

d) Electrostatic carburation of methanol in spark ignition engines, the object being to improve air/fuel mixture preparation. (58)

e) Operation of spark ignition engines on dissociated methanol. The fuel is dissociated into hydrogen and carbon monoxide using exhaust heat. The gas mixture is a clean burning fuel suitable for automotive engines or large stationary engines. (59)

f) Use of methanol as a fuel for stratified-charge engines. The use of this type of engine permits very lean, and therefore economical, operation at part load. A second benefit is that large displacement cylinders may be used at high compression ratios without detonation or knocking. (60)

g) Diesel-methanol dual fuel operation. The methanol is either aspirated together with the inlet air (61) or injected into the combustion chamber after diesel injection (62). The object of both systems is to reduce the diesel consumption of large engines by partially replacing the diesel with methanol.

h) Alteration of the ignition properties of the alcohols by the addition of cetane improvers to produce a fuel that could be used as a straight diesel fuel. This approach has been more successful with ethyl alcohol (ethanol) than with methanol. (63)

i) Emulsification of methanol with diesel to form a mixture that could be used on existing diesel engines, thereby reducing the diesel consumption. (64)

It is apparent when examining this list that research is concentrating on two aspects of methanol operation.

1. <u>Air Fuel Mixture Preparation</u>

This topic accentuates perhaps the main practical difference between methanol and petrol used in spark ignition engines. The combined effect of vapour pressure and high heat of vapourization of methanol has a negative influence on mixture preparation when used in conventional petrol inlet manifolds.

Two solution options are open; either the manifold must be changed to suit the fuel, or the fuel must be changed to suit the manifold. The literature indicates that both approaches have been investigated, and in some cases, very advanced technology has been employed. It is probable however, that the introduction of methanol as an alternative transport fuel will be based on already proven technology and that some degree of potential efficiency sacrifice will be made in the initial stages in order to utilize the present generation of spark ignition engines.

2. Methanol in Large Engines

The cetane rating of methanol is too low for it to be used as a direct substitute for diesel fuel. The current techniques for evading this problem such as stratification of the charge or dual fuel injection have one common denominator - a high pressure methanol injection pump.

In the South African context, one of the desirable features of methanol is that it can be produced locally. This must apply to the engine hardware if the advantage is to be maintained. However, the likelihood of fuel injector pumps being produced in South Africa in the near future is remote, and for this reason, the possibilities of converting diesel engines to spark ignition have been investigated by the Energy Research Institute.

APPENDIX B

THE FUEL RELATED PROPERTIES OF METHANOL:

THEIR INFLUENCE ON ENGINE DESIGN

Methanol (CH_3OH) is a colourless and almost odourless liquid at normal ambient conditions. It is toxic and may cause blindness and death if injested in quantities. The hazards of handling and distribution have been estimated as being similar to that of petrol (65, 66, 67). The fuel related properties on the other hand are somewhat different from petrol and have considerable influence on engine design.

1. STOICHIOMETRY OF OXIDATION

Using iso octane to represent petrol for comparison, the equations for complete combustion are:

Methanol	:	$CH_3OH + 1,5\ O_2$	$CO_2 + 2H_2O$
Iso octane	:	$C_8H_{18} + 12,5\ O_2$	$8\ CO_2 + 9H_2O$

Assuming the air to be 50% humid the stoichiometric mass air to fuel ratio for methanol is 6,55 and 15,31 for iso octane. The comparable relation for diesel is meaningless since the diesel engine is limited to lean operation due to smoke emissions.

Assuming the density of methanol and iso octane to be 786 Kg/m^3 and 687 Kg/m^3 respectively at 25^oC (68), the stoichiometric oxidation of 1 Kg of air would require either 194 ml of methanol or 95 ml of iso octane.

There are considerable implications arising from this fact. If methanol were to replace petrol as a transport fuel, the quantities involved would be about twice that of petrol. This would affect storage tanks, distribution infrastructure, vehicle fuel tanks and control systems.

2. HEAT OF COMBUSTION

The heat of combustion for methanol and iso octane are

Methanol	:	19925 KJ/Kg
Iso octane	:	44381 KJ/Kg

at 25^{o}C with liquid fuel to gaseous products (68). The combined effect of stoichiometric air-fuel ratio and heat of combustion can be related to the energy density per unit mass of air:

Methanol and Air	:	3042 KJ/Kg Air
Iso octane and Air	:	2899 KJ/Kg Air

It is thus clear that although the volume requirements for methanol and petrol are very different, for the same displacement engine, the potential power output is of the same order for the two fuels.

3. HEAT OF VAPORIZATION

Methanol is a polar compound, similar to water, which gives it a latent heat of vaporization greater than that of iso octane:

At 25^{o}C	Methanol	:	-1170 KJ/Kg
	Iso Octane	:	- 308 KJ/Kg

Again relating this to the stoichiometric air-fuel mixture

Methanol and Air	:	178,6 KJ/Kg Air
Iso octane and Air	:	20,1 KJ/Kg Air

This difference in latent heat causes methanol to cool the air-fuel mixture to a considerably lower temperature than petrol. This increases the inlet air density which is desirable for improving the volumetric efficiency. For this reason, Methanol is commonly used as an automotive racing fuel (69, 70). Also aircraft sometimes employ methanol injection into the inlet manifold for maximum take-off power (71).

The effect of mixture cooling on engine performance and manifold design is discussed in conjunction with vapour pressure in the following section.

4. VAPOUR PRESSURE

The vapour pressure of methanol cannot be meaningfully compared to petrol. Petrol is a blend of hydrocarbons with various boiling points and partial pressures. The vapour pressure is thus defined by Raoult's law :

"Above a solution of normal liquids, the partial pressure exerted by any component is proportional to the product of the normal vapour pressure of that component at the existing temperature and its molecular concentration in the liquid." (72)

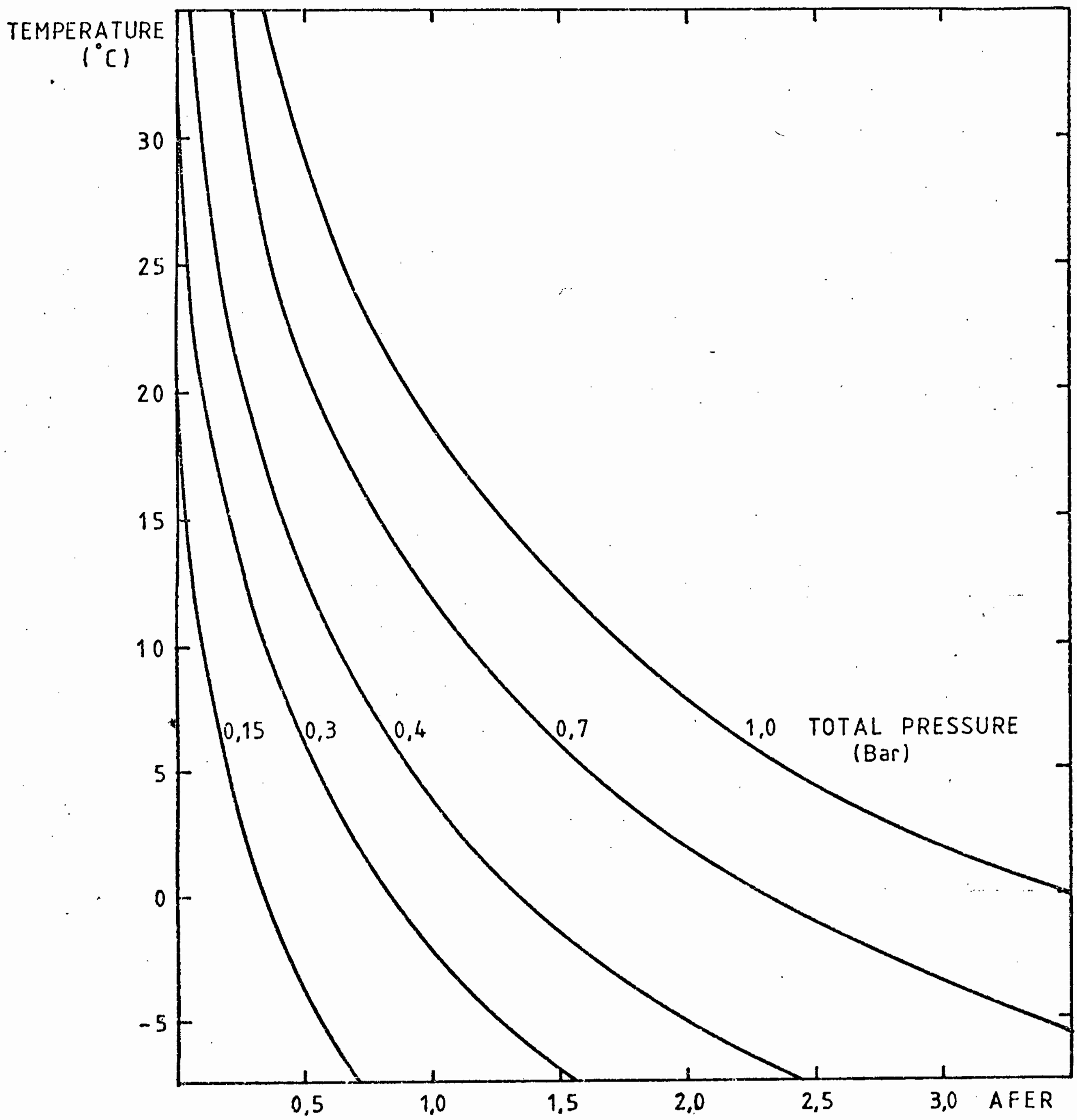
The vapour pressure of petrol can thus be conveniently adjusted by the addition of a small percentage of volatile hydrocarbon, usually termed "light end".

Methanol is a single constituent liquid and therefore the vapour pressure is a function of temperature only. Figure B.1 shows the equilibrium air fuel equivalence ratio (AFER) for methanol and air as a function of temperature and total pressure. Given the lean misfire limit of methanol as about AFER 1,7 (68) it is apparent that at atmospheric pressure, methanol vapour will not ignite below about $10^{\circ}C$. Ignition below $0^{\circ}C$ could only occur at total pressures below about 0,5 Bar.

Combining the effects of latent heat of evaporation and vapour pressure, a mixture of liquid fuel and air would cool rapidly as the liquid evaporates, until the equilibrium air-fuel ratio is reached, after which time no more fuel would evaporate unless the temperature of the mixture was raised by the addition of heat.

The situation differs for petrol in that the cooling effect of petrol is only 11% that of methanol (Appendix B, Section 3), and the equilibrium air-fuel ratio of petrol is a function of the quantity of liquid fuel present as well as temperature by Raoult Law.

This discussion bears relevance to the inlet manifold design and cold starting performance of a methanol engine.



B1 METHANOL VAPOUR - AIR EQUILIBRIUM CHART

4.1 Inlet Manifold Design - Carburettor Engine

It was mentioned in Appendix A that much research is being conducted to discover improved ways of preparing an air-methanol mixture. This stems from the fact that the conventional carburettor and inlet manifold system does not yield satisfactory results when used with methanol (51), especially under acceleration and deceleration conditions.

In carburettor engines using liquid fuel, the process of acceleration is complicated by the presence of unevaporated fuel in the inlet manifold. During normal steady state operation, the inlet manifold contains an amount of liquid fuel that clings to the walls and runs along them to the cylinders at a speed that is very low compared to that of the rest of the mixture which contains air, fuel vapour and entrained fuel droplets. The quantity of liquid fuel is proportional to the pressure and temperature in the manifold. This is apparent from figure (B1).

As the total pressure is increased so the equilibrium air-fuel vapour ratio becomes leaner. Assuming for the moment that the carburettor supplies a fixed air-fuel ratio, then the quantity of unevaporated fuel must increase. The air-fuel ratio reaching the cylinders is therefore leaned until the increased liquid fuel on the manifold walls reaches the cylinders.

This situation arises whenever the throttle is opened. If the throttle opening is sudden, the temporary leaning of the air-fuel ratio reaching the cylinders could be sufficient to cause misfiring.

This phenomenon is true for petrol to a lesser extent than for methanol. It is standard practice to fit acceleration enrichment devices to petrol carburettors and to heat the walls of the inlet manifold. It is apparent that for such measures to be effective with methanol, considerably more heat should be supplied to the inlet manifold. Also the carburettor enrichment device should deliver a greater volume of fuel to achieve sufficient enrichment.

When decelerating, the reverse situation applies. Lowering the manifold pressure causes the liquid fuel present to evaporate, thereby enriching the air-fuel ratio at the cylinders. This condition is both unwanted and uneconomical.

Heating the inlet manifold has an additional advantage in that the variation in air-fuel ratio reaching the different cylinders is improved. The liquid fuel which wets the cylinder walls tends to take the easiest path to the cylinders. It has been shown that the equivalence ratio for methanol in a four cylinder engine using a standard petrol manifold can vary from AFER 1,2 to AFER 0,75 in adjacent cylinders (73). Heating the walls was found to greatly improve the distribution by evaporating most of the liquid fuel.

Perhaps a logical conclusion to the above argument would be to evaporate all the fuel and power the engine on an air-gas mixture. However this could have a number of disadvantages.

a) It was stated in Appendix B section 3 that the cooling effect of methanol improved the volumetric efficiency by increasing the charge density. Not only would this desirable feature be lost by fully evaporating the fuel, but gaseous methanol would effectively throttle the air supply by occupying about 12% of the mixture volume.

b) The cooling effect of methanol on the inlet mixture raises the detonation limited compression ratio which in turn increases the overall efficiency and power output. This relationship is discussed further in Appendix B, Section 5.

4.2 Fuel Injection

An alternative solution to the problems of carburation is fuel injection. The absence of liquid fuel lining the inlet manifold removes all of the associated transient problems. In addition, equal distribution to all cylinders is maintained by the metered fuel nozzles at each inlet port. Full advantage can be taken of the cooling effect of methanol to both increase charge density and to suppress detonation.

4.3 Cold Starting Performance

The usual method of choking a petrol carbureted engine by means of an offset butterfly valve ahead of the venturi achieves three objectives.

a) The quantity of fuel that is drawn from the fuel bowl is increased by the pressure depression caused by closing the air passage.

b) The pressure depression increases the fuel evaporation rate.

c) The air flow is reduced.

Enrichment of the liquid petrol-air ratio increases the vapour-air ratio because petrol is a blend of hydrocarbons and therefore Raoult's law is applicable.

In the case of methanol, as is shown in figure B.1 the quantity of liquid present has no influence on the vapour-air equilibrium ratio. One solution that has been adopted by various research teams has been to add a small percentage of volatile component such as i-pentane (74) or butane (75).

An alternative solution, that of electrically heating a small quantity of methanol for starting, is not entirely satisfactory, because of the unacceptable engine performance immediately after starting (74). This applies too for the application of a separate starting fuel.

No reports could be found in which advantage of the change in equilibrium air-fuel ratio with total pressure, as is indicated in Figure B.1, is taken. It is conceivable that almost completely closing the throttle butterfly, and drastically retarding the ignition timing could yield low temperature ignition conditions. Once ignition is achieved the timing and throttle could be reset for normal running.

5. OCTANE RATING

Some uncertainty exists over the exact octane rating of methanol. The test procedure for establishing octane ratings has evolved primarily for hydro carbon fuels, and is not entirely suited for methanol. The non-homogeneous fuel mixture and combustion pressure pulses have led to reported research octane ratings of 106 to 114 for methanol (68).

Since the octane rating is established on the basis of the compression ratio of a test engine, it is interesting to note that Ebersole and Manning found that the maximum compression ratio for iso octane could be higher than that for methanol under certain operating conditions (76). In this particular instance the inlet mixture temperature was maintained at 52^{o}C. This differs from the standard research octane rating procedure where the inlet air temperature is maintained at 52^{o}C. It is known that the inlet mixture temperature has a very significant effect on maximum compression ratio (77) and it is therefore likely that it is the cooling effect of methanol that is partly responsible for the high octane rating.

In the practical situation, almost every engine parameter has a direct influence on the relation between maximum compression ratio and octane rating. A discussion on detonation and engine design is outside the scope of this thesis. However an attempt is made in Appendix D to approximately relate maximum compression ratio, engine size and octane rating for the "average" engine.

6. IGNITABILITY, FLAME SPEED AND MISFIRE LIMITS

The cetane rating of methanol is so low that it cannot be measured directly. Extrapolation of test results indicate that the cetane number of methanol is about 3 (68). Pure methanol is therefore not a suitable diesel fuel. As indicated in Appendix A it can be used in conjunction with diesel fuels in a dual fuel configuration.

Another measure of ignitability is the hot plate ignition temperature. At stoichometric air-fuel ratio, the hot plate temperature for preignition of methanol in a test engine was about 788°C whilst the plate temperature for iso octane was about 877°C (78). The implication of this result applied to engine design is that care must be exercised to avoid hot spots, especially at the spark plug which would cause preignition. This is especially true if high compression ratios are employed.

Measurements of flame speed indicate that methanol burns significantly faster than petrol, particularly in the lean region (68). This means that the lean misfire limit for methanol is correspondingly leaner than that of petrol. The limit is to some extent dependent on operating conditions such as mixture preparation and inlet temperatures. Typically a value of AFER 1,7 could be quoted (79). Values for the rich limit were not quoted, since this is usually of little interest once the point of maximum power has been reached.

7. EXHAUST EMISSIONS

The study of exhaust emissions is an entire subject in its own right. Considerable research has been reported and many of the references for this thesis include sections on the subject. To summarize; the emissions from methanol test engines compare similarly to those of petrol engines, with the exception of aldehyde emissions which are typically higher with methanol. The effects of air-fuel ratio, compression ratio and ignition timing on emissions shows similar trends for both fuels.

APPENDIX C

THE DERIVATION OF PERFORMANCE FORMULAE

By definition $e_i = P_i/m_f\ Q_c$

$e_v = 2\ m_a v_a/N\ V_d$ (4 stroke engines)

$IMEP = 2\ P_i/N\ V_d$ (4 stroke engines)

Noting that $m_f = m_a/F_s$ AFER

The equations can be combined to eliminate P_i and m_a

$$IMEP = e_v\ Q_c\ e_i/v_a\ F_s\ AFER$$

By definition $P = N\ V_d\ BMEP/2$

At wide open throttle $BMEP = IMEP - FMEP$

Thus $P = N\ V_d(IMEP - FMEP)/2$

By definition $e_o = P/m_f Q_c$

$= (P_i/m_f\ Q_c)\ (1 - (Pi - P)/Pi)$

But by definition $P_i = N\ V_d\ IMEP/2$

Thus $e_o = e_i\ (1 - FMEP/IMEP)$

APPENDIX D

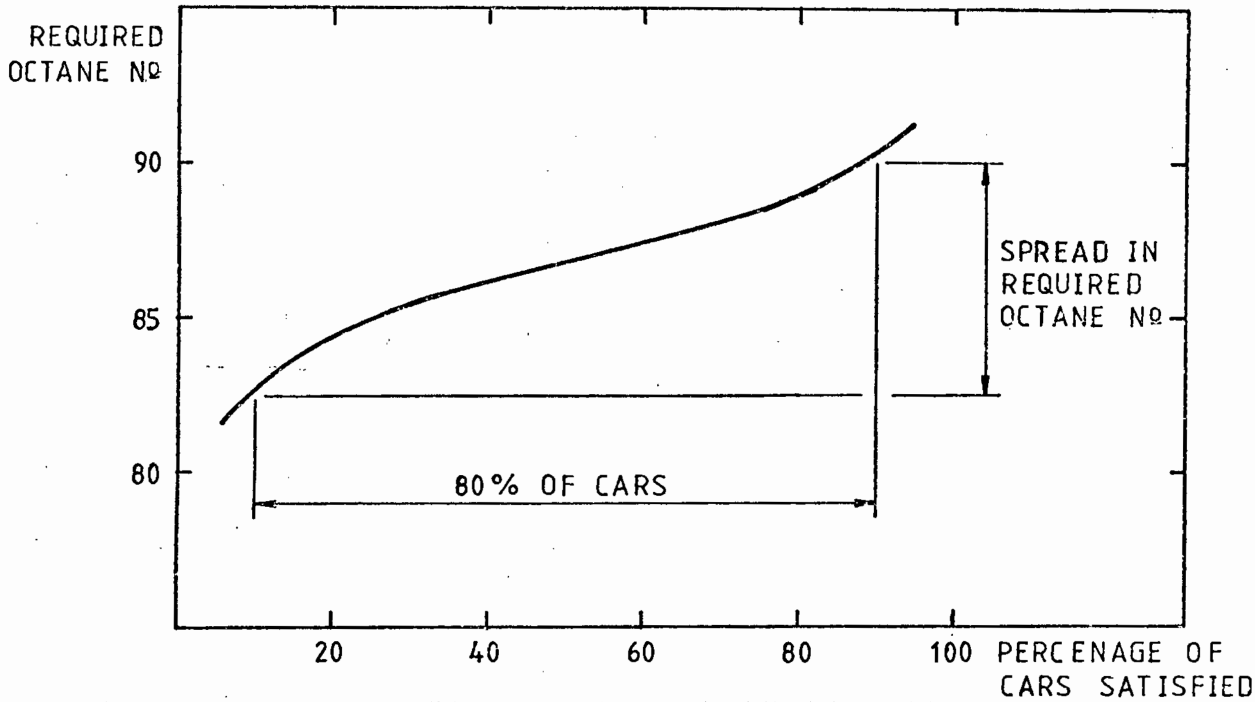
APPROXIMATE RELATIONSHIP BETWEEN ENGINE SIZE, MAXIMUM COMPRESSION RATIO AND OCTANE RATING

The effect of engine size on maximum compression ratio is well documented. However the fuel related properties of methanol are sufficiently different from petrol that engine operation in a regimé of compression ratios outside that of petrol is possible.

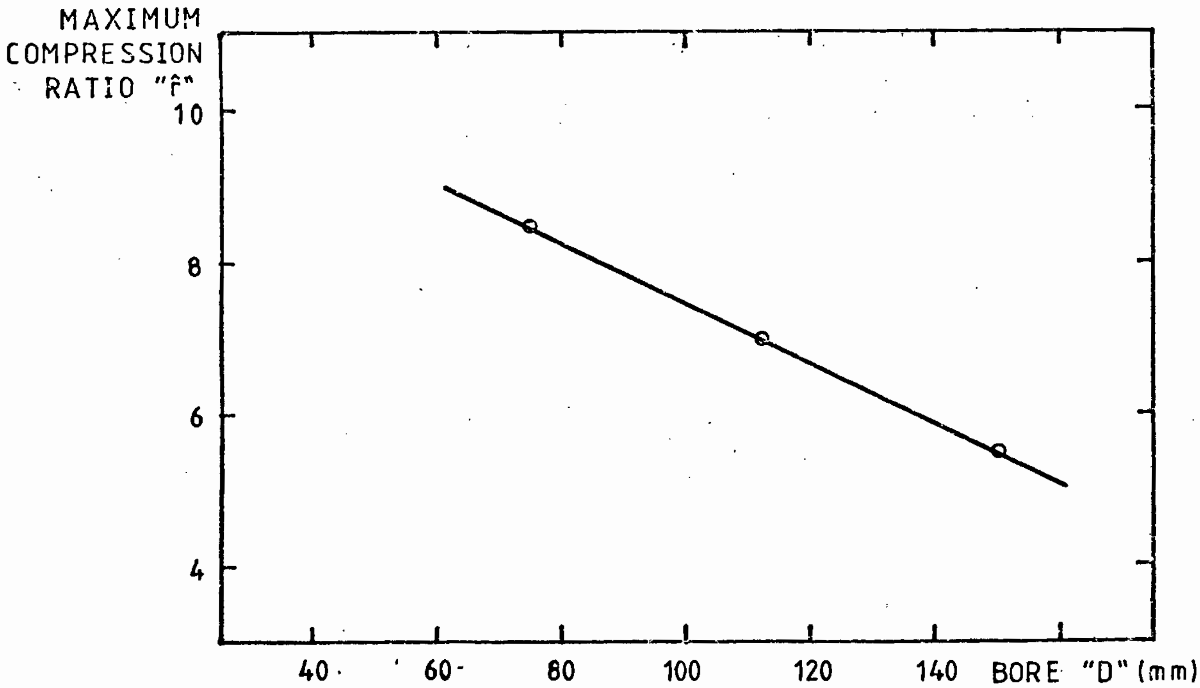
It is a general rule that as the cylinder bore of a spark ignition engine increases, the tendency to detonate increases, and therefore fuels of increasing octane number or reduced compression ratios are required (80). It has been shown in Figure 3.1 that reduced compression ratio has a negative effect on indicated efficiency. It is for this reason that, with the increased public awareness of the global energy situation in recent years, the use of large spark ignition engines operating on petrol has diminished.

As was discussed in Appendix B, Section 5, the research octane rating (RON) of methanol is about 110, which is about twelve numbers higher than premium grade petrol. By implication, methanol could be used as a fuel for larger spark ignition engines than are currently used with petrol, at an acceptable efficiency level.

The relation between maximum compression ratio and octane rating is not a single valued function, but is influenced by bore size, gas flow patterns in the combustion chamber, ignition timing, air-fuel ratio and many other factors. An indication of the sensitivity of octane rating to slight variations in engine parameters was demonstrated by a test of 29 cars of the same model, tuned to the manufacturers recommended ignition timing. A spread of seven numbers in the detonation-limited minimum required octane rating was found, see figure D.1 (81).



D1 TYPICAL SPREAD IN REQUIRED O.N. FOR ONE MAKE OF CAR. Make of car - Volkswagen; Ignition Timing set to makers specification; Date based on Ref 81



D2 RELATION BETWEEN MAXIMUM COMPRESSION RATIO AND BORE SIZE FOR THREE GEOMETRICALLY SIMILAR ENGINES Fuel RON 82,5; Constant inlet and exhaust conditions; Various piston speeds; Data based on Ref 82

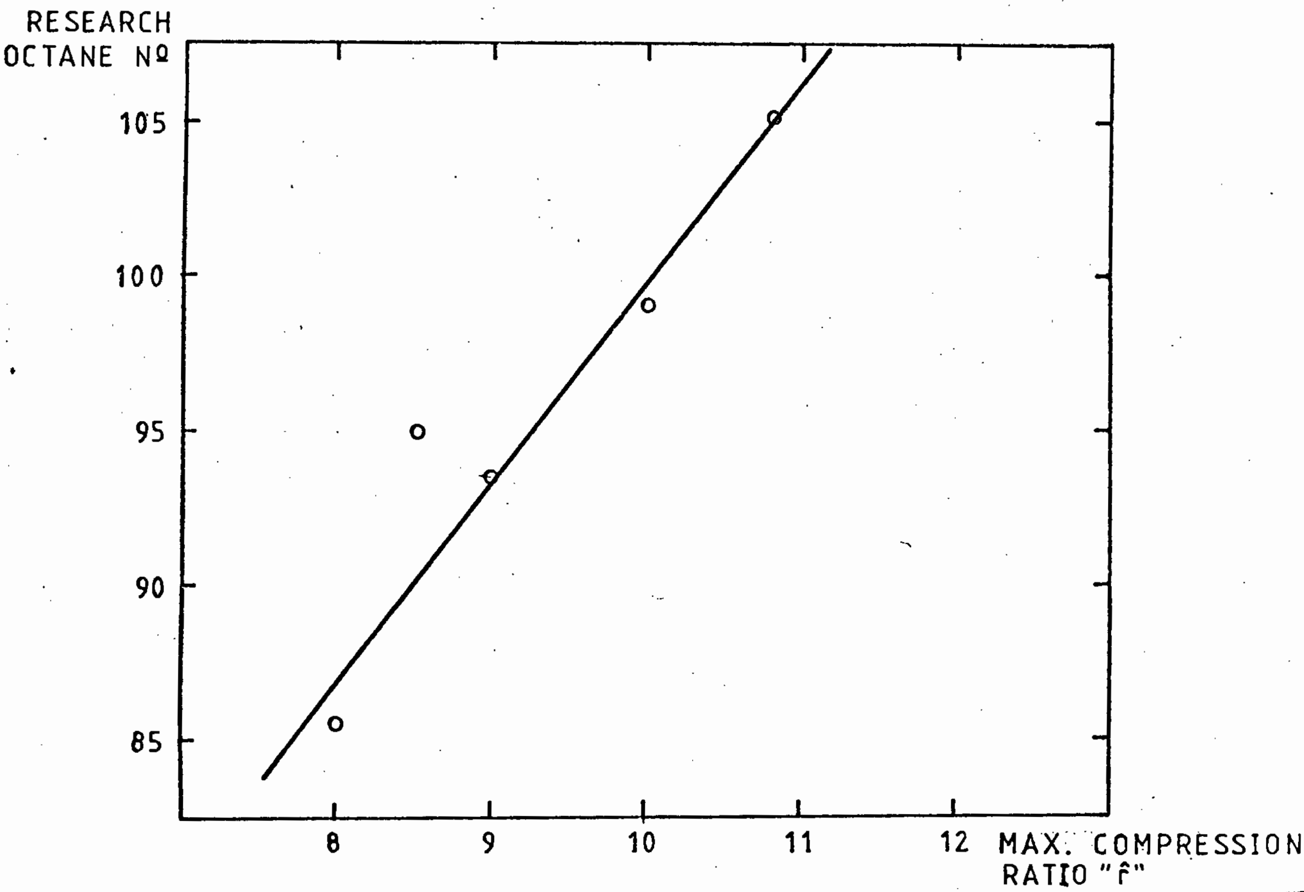
In the light of the above discussion, an analysis of required octane number in terms of compression ratio and bore size only must inevitably be very approximate. However, this can be acceptable as a rough and ready guide in certain circumstances.

Figure D.2 shows the relation between maximum compression ratio and bore size for three geometrically similar engines operating on 82,5 RON fuel at various piston speeds. Figure D.3 shows the relation between maximum compression ratio and octane number for two different engines having the same bore but at different compression ratios. Combining these two relations onto one graph and extrapolating the results, Figure D.4 was produced. The relationship between maximum compression ratio and bore size for methanol corresponds to RON 110.

This result is reproduced in Figure 3.3. However, in order to emphasise the approximate nature of this analysis as was illustrated in Figure D.1, the relationship is not represented as a single line, but rather as a band, the width of which was arbitarily chosen as representing five octane numbers.

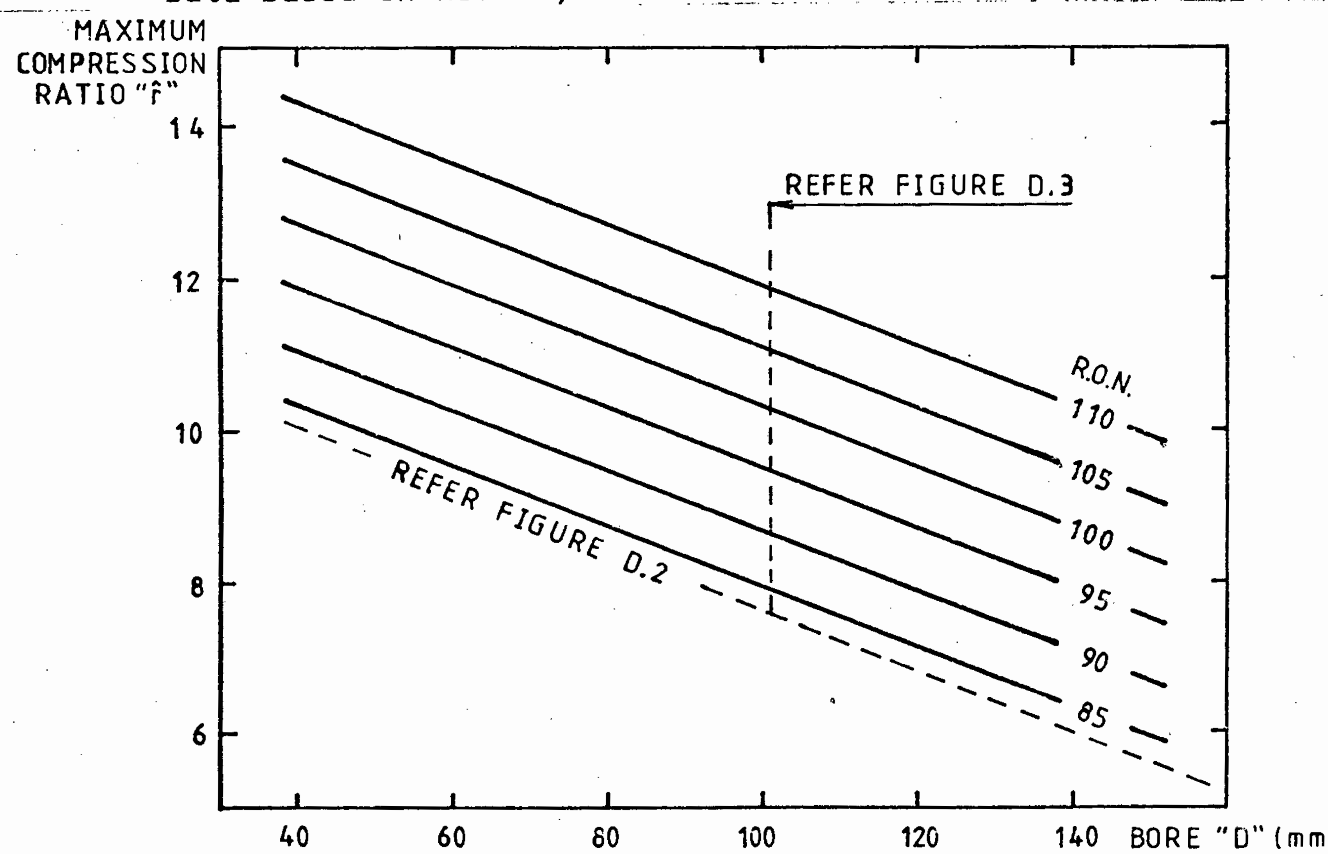
It must be stressed that Figure 3.3 is not an exact relation and is intended as a guide only. The trend does nevertheless serve to illustrate the potential for operating large engines on methanol.

For example: An engine with compression ratio 10:1, operating on petrol with RON 93 would be limited to a bore size of about 7,5 cm. If the fuel was methanol with RON 110, the bore size limitation becomes about 14,5 cm at the same compression ratio.



D3 RELATION BETWEEN RESEARCH OCTANE NUMBER AND COMPRESSION RATIO FOR TWO ENGINES WITH BORE 102 mm (4 INCHES)

Data based on Ref 83, 84.



D4 FIGURES D2 AND D3 COMBINED

APPENDIX E

FORMULAE FOR CALCULATING THE PREDICTED PERFORMANCE

(Corrected to the units given in the Nomenclature)

CONSTANTS

D, S, C, N_r

Q_c = 19,925 KJ/Kg (Section 3.5.1)

v_a = 0,830 m^3/Kg (Section 3.5.2)

F_s = 6,55 (Section 3.5.3)

$V_d = 7{,}854 \times 10^{-7}\ D^2\ S\ C$

MAXIMUM COMPRESSION RATIO

$r = -3{,}98 \times 10^{-2}\ D + 15{,}7$ (Figure 3.3)

VARIABLES

r, N, AFER

INDICATED EFFICIENCY

$e_s = 100 - \mathrm{Log}^{-1}\ (-0{,}1454\ \mathrm{Log}\ (r) + 1{,}939)$ (Figure 3.1)

For AFER greater than 1,0 :-

$e_i = e_s\ (0{,}875 + (29{,}7\ \mathrm{AFER} - 26{,}25)^{0,5}/14{,}85)$ (Figure 3.2)

For AFER less than 1,0 :-

$e_i = e_s\ (-3{,}76\ (\mathrm{AFER})^2 + 8{,}105\ \mathrm{AFER} - 3{,}345)$ (Figure 3.2)

VOLUMETRIC EFFICIENCY

$e_v = -57{,}6\ (N/N_r)^3 + 77{,}7\ (N/N_r)^2 - 13{,}7\ (N/N_r) + 82{,}7$

(Section 3.3)

FRICTIONAL LOSSES

$$FMEP = 8,04 \times 10^{-12} N^2 S^2 + 1,25 \times 10^{-6} NS + 0,91$$

(Section 3.4)

INDICATED MEAN EFFECTIVE PRESSURE

$$IMEP = 10^{-6} e_v Q_c e_i/v_a F_s AFER$$ (Section 3.1).

PERFORMANCE

$$P = 1,667 \times 10^{-3} N V_d (IMEP - FMEP)/2$$

$$e_o = e_i (1 - FMEP/IMEP)$$

APPENDIX F

ENGINE CONVERSIONS TO METHANOL

1. FORD CORTINA 2000 OHC

In Appendix B section 1 it was shown that in order to combust with the same amount of air, the volumetric flow rate of methanol must be more than double that of petrol. Thus the conversion of an existing petrol carburettor to methanol requires drastic jet changes.

It was intended originally to match the AFER of methanol to that of the petrol. However it was found that at the higher air-fuel flow rates, the carburettor passages themselves were constricting the fuel flow. As the engine speed was increased at wide open throttle, the air-fuel ratio was consequently leaned out. Nevertheless, a combustible mixture was maintained under all conditions and it was therefore considered unnecessary to enlarge the fuel passages.

The resulting AFER at wide open throttle is shown in figure 5.4.

The faster flame speed of methanol, (Appendix B, section 6) effectively alters the ignition timing. The static advance was therefore adjusted to give MBT timing at wide open throttle.

2. VOLKSWAGEN PASSAT 1.6

The object of this conversion was to take advantage of the higher compression ratios possible with methanol in order to improve the overall efficiency. An attempt was made to adjust the carburettor air-fuel ratio so as to match the rated petrol output, although this was not completely successful.

2.1 Compression Ratio

The compression ratio selected for the conversion was 12:1. This decision was made on the basis of tests made in Germany on a similar engine (85). Special high compression ratio pistons were obtained from Germany.

2.2 Air-Fuel Ratio

As was shown in Appendix B, Section 1, the carburettor jets had to be modified so as to supply about twice their designed fuel flow rate.

It was found that the methanol air-fuel ratio required to produce the rated petrol power could not be used because of a feature of the carburettor supplied with the engine. At wide open throttle the automatic enrichment valve was functioning. If the AFER was then adjusted to produce the rated power, it was found that at part throttle, when the enrichment valve closed, the mixture became too lean to ensure reliable combustion in all cylinders. In the end a compromised AFER, shown in Figure 5.4, was found to give reliable combustion under all circumstances whilst at the same time producing a slightly greater output than the original petrol rating.

2.3 Inlet Manifold Design

It was stressed in Appendix B, Section 4.1 that the inlet manifold of a methanol engine would ideally be very different to that of a conventional petrol engine.

A specially built manifold was available from Germany for the Passat. Almost the entire inlet passage was jacketed with engine coolant which was forced to make a double pass over the manifold and to form a hot spot directly beneath the down-draught carburettor. It was thought that provided all the inlet walls were maintained at a temperature above the boiling point of methanol, then the liquid fuel present would be minimized.

2.4 Ignition Timing

The ignition timing had to be altered for operation with methanol. The static timing was retarded about 5^o from the petrol setting to cater for the higher flame speed of methanol (Appendix B, Section 6). Furthermore, the increased compression ratio evidently gave rise to greater turbulence within the combustion chamber with the result that the mechanical advance movement had to be limited from 30^o (crankshaft) to 22^o to maintain MBT timing at high engine speeds.

3. MERCEDES BENZ OM 355

The object of this conversion was primarily to investigate the feasibilities of converting a diesel engine to spark ignition. It was considered important that the converted engine should match the diesel rated power output and should retain as many of the original diesel components as possible. Whilst overall efficiency was not of paramount importance, obviously the prototype should be optimised within the design constraints already imposed.

3.1 Compression Ratio

Experimental work on similar engines by the Energy Research Institute had shown that about 10,5:1 was about the highest safe ratio needed to avoid detonation.

10:1 Was in fact chosen as being slightly conservative. The desired compression ratio was achieved by machining the crown of the standard diesel piston. The resulting combustion chamber had the form of a dished disc with a diameter equal to that of the cylinder bore.

3.2 Ignition System

A conventional ignition distributor was mounted in the place of the diesel injector pump.

The aperture where the diesel injector was located was modified by means of suitable inserts to take a conventional automotive spark plug. The position was ideally situated in the diametral centre of the combustion chamber.

A capacitor discharge ignition system was used to ensure the optimum spark and to prolong the points life.

3.3 Fuel System

A continuous inlet port injection system was custom built for the engine. Standard aircraft fuel nozzles were positioned in one of the two intake ports of each cylinder, about 80 mm from the inlet valve.

The mixture control valve was situated on the inlet air ducting with a flat disc placed in the air stream to sense the air mass flow. Fuel pressure was achieved with a positive displacement gear pump which was belt driven from the engine.

The fuel system control was modified by trial and error to give an air-fuel ratio such that the methanol engine output was closely matched to the rated diesel output at wide open throttle.

3.4 Miscellaneous

The diesel engine controls power output by variation in the fuel supply alone. When converted to spark ignition, a butterfly valve had to be fitted to the intake ducting to control the air supply. The fuel control valve then automatically corrected the fuel supply to match the air flow.

Some additional features were also included to complete the conversion:

a) An overspeed protection system that cut first the fuel supply, and then the ignition was devised.

b) A choking device combined with an electric priming pump was required to facilitate cold starting.

APPENDIX G

TABLES

TABLE 3.1 Reported Indicated Efficiency As A Function Of Compression Ratio For Methanol At AFER 1,0

r	Ref 27	Ref 28	Ref 29
7		35,0	
8		36,4	
8,2	34,4		
9		37,3	36,5
10	37,5	38,2	37,5
11		39,1	38,5
12	39,6	39,7	39,5
13		40,4	40,0
14	40,1	41,0	
15		41,4	

TABLE 3.2 Reported Indicated Efficiency As A Function Of AFER For Methanol

AFER	Ref 30	AFER	Ref 31	AFER	Ref 32	AFER	Ref 33	AFER	Ref 34
						0,83	29,3		
		0,9	31,0	0,9	32,0	0,91	31,8	0,91	28,0
1,0	35,0	1,0	35,0	1,0	36,0	1,0	34,4	1,0	29,9
1,11	37,0	1,1	37,0	1,1	38,0	1,05	35,5	1,05	30,6
1,25	37,5	1,2	38,0	1,2	39,7	1,11	37,0	1,11	31,2
		1,3	38,5	1,3	40,7	1,18	37,8	1,18	31,8
				1,4	41,1	1,25	38,3	1,25	32,2
				1,5	41,1	1,33	38,3	1,33	32,6

TABLE 5.1 Measured AFER, Shaft Power, And Overall Efficiency At Wide Open Throttle Using Methanol : Cortina

Engine Speed rev/min	AFER	Power KW	Efficiency %
1000	0,95	10,8	31,5
2000	1,02	28,7	32,9
3000	1,12	48,1	32,4
4000	1,21	58,7	32,3
5200	1,33	62,1	29,8

TABLE 5.2 Measured AFER, Shaft Power, And Overall Efficiency At Wide Open Throttle Using Methanol : Passat

Engine Speed rev/min	AFER	Power KW	Efficiency %
1500	1,00	19,5	33,9
2000	1,03	25,2	34,0
3000	1,04	40,7	34,3
4000	1,06	53,9	33,8
5000	1,05	62,3	32,4
5600	1,04	65,9	31,1

TABLE 5.3 Measured AFER, Shaft Power, And Overall Efficiency At Wide Open Throttle Using Methanol : Mercedes

Engine Speed rev/min	AFER	Power KW	Efficiency %
1000	1,25	85	36,2
1200	1,28	101	36,9
1400	1,27	121	35,8
1600	1,28	138	35,8
1800	1,28	153	35,8
2000	1,27	170	35,4
2200	1,24	179	33,1