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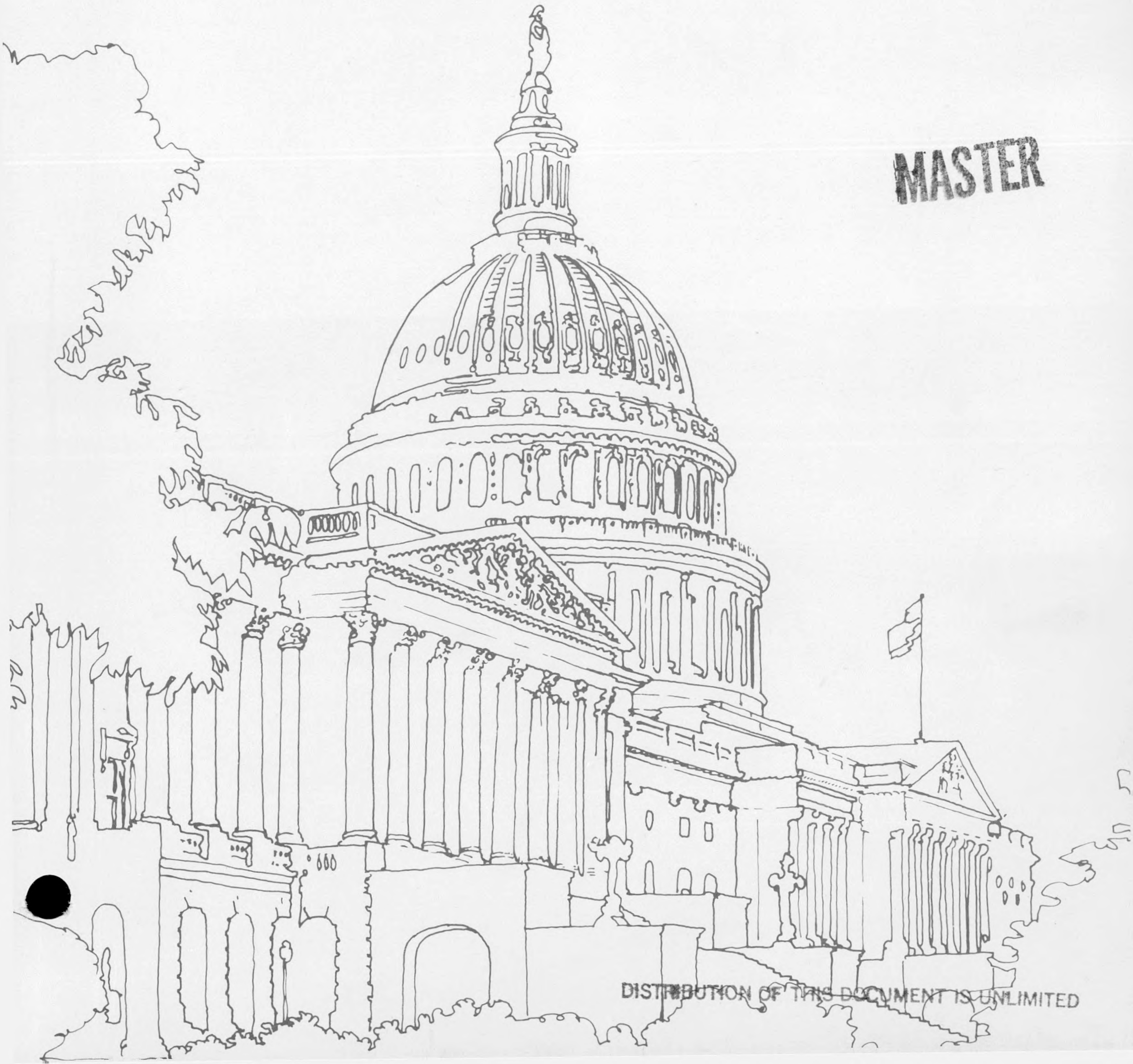

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Session 11

OPERATING WITH LEAN MIXTURES - A SOLUTION FOR THE FUTURE ?

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ABSTRACT

In the future the development of the combustion engine is determined by energy and economy requirements. The operation in the lean mixture range offers ideal conditions for the reduction of the fuel consumption and the exhaust emissions.
It is of increasing importance that the following combustion systems guarantee a good operating in the range of lean mixtures :
the lean burn Otto engine, the stratified charge engine, and the Diesel engine.

INTRODUCTION

For many years, the working range of the Otto and Diesel engines has been determined by the formation and the ignition of the combustion mixture. The high specific power output of the Otto engine is to be attributed among others to the range of rich mixture in which it operates and where the flame propagation velocity reaches its maximum values.
The forced operation of the Diesel engine in the lean mixture range as well as its possibilities to burn these mixtures also under part load assure the Diesel engine a high degree of efficiency, which, however, comes along with a relatively low specific power output.

The advantages of a lean mixture operation were recognized at a very early stage of the combustion engine development already. Extensive thermo-dynamical analyses were carried through, showing that an increase of the air/fuel ratio results into more favourable values as far as the thermal efficiency η_{th}, the indicated efficiency η_i, and the quality coefficient η_g are concerned.
With view to the Otto engine these advantages are of merely theoretical nature, as the condi-

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tions for ignition and flame propagation essentially deteriorate already with a mixture slightly leaner than stoichiometric.

In the course of the efforts undertaken to comply with the exhaust emission regulations on one hand and to improve the fuel economy on the other, the range of lean mixtures has proved to offer ideal possibilities for the fulfillment of the aimed-at objectives. The development of modern combustion engines has partially turned into a competition between the different combustion systems, permitting the extension of an efficient engine operation into the range of lean mixtures. This requirement is being met primarily by the Diesel, the stratified-charge, and the so-called lean burn Otto engine.

The analysis of the factors influencing the effective characteristic engine data leads to the conclusion that the leaning-out of the mixture alone can never be considered as ultimate objective of a development. It is only useful up to the point where the engine reaches its optimum characteristic data.

The decrease of the mechanical efficiency η_m with increased leaning out of the mixture results into maximum effective efficiency η_e at air/fuel ratios in the range of $\lambda = 1.3 \div 1.5$, despite the improvement of the thermal and indicated efficiency (Fig. 1). Under steady-state operation, the range of these air/fuel-ratios offer optimum conditions for favourable fuel consumption and exhaust emission, and this regardless of the combustion process involved. With view to the transient operational conditions, however, the ignition of the air/fuel mixture at the highest possible air/fuel-ratio must be assured, in order to avoid misfiring in any operating condition.

THE OTTO ENGINE.

During recent years, the possibilities of the Otto engine to be operated in the lean mixture ranges have been examined in detail (1-8)+.

The extent to which the mixture may be leaned-out is determined by the shape of the combustion chamber, the compression ratio, and the ignition system. The

+ "Numbers in brackets designate References at end of paper " ideal solution, i.e. to optimize the combustion chamber with simultaneously minimum fuel consumption, low exhaust emission under part load, as high as possible a maximum power output and a considerable distance from the knock limit cannot be realized with the conventional Otto engine. A low fuel consumption and a high specific efficiency necessitate a rapid complete combustion, which is realized by means of a compact combustion chamber, short flame propagation paths, and an intense turbulence of the mixture.

Low emission values can be obtained among others by retarding the combustion with low maximum temperatures and relatively high exhaust gas temperatures, i.e., by means of a combustion with low efficiency.

The optimization of the shape of the combustion chamber requires a compromise between the fuel consumption, exhaust emission and power output.

A single cylinder test engine served to examine the optimum location of the quench zones in a combustion chamber with fixed position of the spark plug and the valves. Ionisation probes served to control the flame propagation in different directions. Figure 2 shows the mean flame propagation velocity Vm to the remotest probe and the mean maximum pressure P_{max} in function of the ignition point. The big quench area B II situated opposite of the spark plug, accelerates the combustion process, which favourably influences the efficiency.

The shape of the combustion chamber does not only influence the combustion itself, but is also reflected in the fuel consumption and the composition of the exhaust gas (Fig. 3). The lowest fuel consumption was realized with combustion chamber B II whereas B IV with two big quench zones resulted into the highest overall consumption. As expected, the most rapid and completest combustion of chamber B II brought the highest NO_x-emission with a relatively small amount of unburnt hydrocarbons.

The increase of the compression ratio constitutes a well-known measure to improve the efficiency. For years, the compression ratio has been increased only to raise the specific power output and was limited by the knock rating of the fuel. With increase of the compression ratio it is possible to obtain better fuel consumption, but with higher HC- and NO-emission values in the range of minimum fuel consumption (Fig. 4). The lean mixture limit distinctly shifts towards higher air/fuel ratios.

Figure 5 illustrates the possibilities of engine operation in the range of lean mixtures. In case the shape of the combustion chamber does not permit an intense turbulence of the mixture, the operation in the lean-mixture range is practically impossible. At low compression ratios, the conditions for a flame propagation at $\lambda > 1,0$ are disadvantageous, whereas with higher compression ratios knocking combustion occurs. There is only a very small margin between the lean mixture limit and the knock limit. The realisation of a leanburn operation under such condition requires either the precise tuning of all influencing factors without the tolerances usually allowed in series production, or else the retarding of the ignition point, which would result into a considerable impairment of economy and operational stability. The combined effect of an intense mixture turbulence and a high compression ratio shifts the air/fuel-ratio limit into the lean-mixture range (Fig. 5, right). At the same time, the highly intense turbulence favourably influences the knock limit which is shifted towards higher compression ratios. These are the indispensable preconditions for an impeccable engine operation in the lean-mixture range.

The engine operation in the lean-mixture range can be improved to a certain degree by the adequate choice of the ignition system, the optimization of the spark-plug arrangement in the combustion chamber etc. Of special importance for the future development of the Otto engine is the optimization of the ignition point under consideration of fuel consumption and exhaust emission and the maintenance of optimum values under all operational conditions. The ignition of lean mixtures can be further improved by increasing the ignition energy, prolonging the spark duration and increasing the spark gap.

Figure 6 shows the influence of the ignition energy - represented as the ignition current in the secondary circuit - on the characteristic data of the Otto engine. Under hot running conditions the presently used characteristic data of the ignition system are quite sufficient to reach favourable emission and fuel consumption values. The operation of the engine under stringent conditions such as cold starting or overrun requires an increased ignition energy or a prolonged spark duration. According to our investigations, the optimum characteristic data of the ignition system are the following :

Ignition current	100 mA
Spark duration	10 - 25 ^{O}CA
Spark gap	1.5 mm

With the Otto engine, it is possible to realize the advantages of a lean-burn operation - such as reduction of the exhaust emission and improvement of the economical aspects - if all available measures are being scrupulously applied.

Figure 7 shows the ECE-test results obtained with two PORSCHE engines operated with lean mixtures under part load (λ = 1.2 - 1.4). The tested units were a 924 4-cylinder water-cooled engine and a 911 6-cylinder air-cooled engine.

The reduction of the CO and NO_x emissions by 40 to 60 p.c. as well as an improvement of the fuel consumption by 8 - 10 p.c. can be obtained without a major modification of the HC emission. The chances for a general utilisation of the "lean-burn engine" depend on future exhaust emission regulations, mostly as concerns the limitation of the NO_x-emission. The ECE test values of less than 5-6 g/Test NO_x respectively the '75 CVS test values of less than 1.5 - 2 g/mile can be obtained in a lean-burn engine only with simultaneous deterioration of the HC-emission, the fuel consumption and the driveability. The required extremely low CO and HC emission values can be obtained only by using an exhaust gas aftertreatment.

THE STRATIFIED CHARGE ENGINE.

Another possibility of realizing a stable engine operation with a high air/fuel ratio is the use of the so-called stratified charge or torch ignition. The stratified charge engine is one of two alternative systems which are likely to realize the '75 CVS-test values of $NO_x < 1$ g/mile without impairing the fuel consumption and the driveability characteristics.

Best suited for a smooth combustion throughout a wide load-speed range are those stratified charge engines disposing of a divided combustion chamber, with an auxiliary chamber volume of less than 20 p.c. of the overall compression volume (12) . With the known stratified charge engines, two systems of fuel supply to the auxiliary chamber are prevailing :

- An additional mechanically actuated pre-chamber valve, or
- Fuel injection.

Both variants were examined under identical conditions in a single cylinder test engine (Fig. 8). The fuel injection into the auxiliary chamber is highly advantageous with view to NO_x-emission, fuel consumption and lean-mixture limit. The stratified engine with scavenged auxiliary chamber obtains lower minimum HC and CO emissions. As PORSCHE considers the reduction of the NO_x-emission as the main objective of the development of the stratified engine, decision was made in favour of the fuel-injection variant, putting up with a deterioration of the HC and CO emissions.

The stipulation of the dimension and the shape of the orifice connecting the auxiliary chamber and the main combustion chamber has essential effects on the flow pattern in the auxiliary chamber, on the mixture formation as well as on the combustion itself. Moreover, the orifice influences the fuel consumption and the exhaust emission (Fig. 9) . The use of small orifice cross-sections leads to a visible reduction of fuel consumption as well as of the HC and NO emissions under partial load. These improvements are accompanied by reduction of the engine power ΔP.

Favourable emission values and low fuel consumption can be obtained from the stratified charge engine if no high specific output is required. Due to its rather special combustion the stratified charge engine is relatively insensible to modifications of the fuel quality. Tests carried through with a PORSCHE 911 6-cylinder SKS engine in accordance with the CRC-test procedure have shown that with identical compression the SKS engine requires a less high octane rating than the Otto engine (Fig. 10).
Extensive tests to which the PORSCHE 911 was submitted have proved a good driveability with low NO_x-emissions. It would be possible to realize a lean-mixture value of $\lambda_{overall}$ >2.5, this, however, does not seem reasonable with view to the exhaust emission and the fuel consumption. NO_x-values of less than 0.4 g/mile (CVS-test) require air-fuel ratios of 1.4 - 1.6 in the part-load-range. Minimum HC emissions are obtained at $\lambda_{overall} = 1.2 - 1.3$. The NO_x and CO values, in force in the USA starting from 1978, can be complied without using any exhaust gas aftertreatment system. Low HC-emission values as required by law cannot be realized without exhaust gas aftertreatment at the actual development stage of the SKS engine. Figure 11 shows the CVS '75 and the ECE test results obtained with the 911 SKS engine with different basic engine tunings. The HC-values were obtained using an oxydation catalyst.

THE DIESEL ENGINE.

The high thermal efficiency and the low fuel consumption of the diesel engine have always been a stimulus to increase the use of this engine version in passenger cars. In former times, diesel engines had been mostly installed into vehicles with high mileages per year. From 1971 on, the share of diesel-engine vehicles in the overall vehicle population is continuously increasing. Under consideration of the CO, HC and NO emissions alone the diesel engine shows essential advantages as compared to the conventional Otto engine. The fuel consumption, too, is better with the diesel engine than with the Otto engine, at least as far as driving in the city is concerned. This is due to the combustion process which enables the diesel engine to operate with a high excess air rate, so that there is always a sufficiently great amount of oxygen available to assure an almost complete oxidation of carbon monoxide and unburnt hydrocarbons.

The high soot emission and the great annoyance by odours and noise inhibit the introduction of the diesel system on a broad basis. Also with view to the fuel supply the diesel engine does not constitute an acceptable alternative to the Otto engine, as a complete changing-over of the existing car population to diesel operation is not possible. According to various forecasts, 12 to 30 p.c. of the Otto engines will be replaced by diesel engines, so that more than 70 p.c. of the remaining vehicles will continue to be driven by gasoline engines.

In comparison to the Otto engine the specific power output of the diesel engine is by 40 to 50 p.c. lower. Moreover the diesel engine shows an unfavourable dynamic behaviour when installed into a passenger car.

If the Otto engine disposed of the speed and load ranges of a diesel engine (n_{max} = 4,500-5,000 rpm; P_e = 18 - 22 kW/l), it would also permit low fuel consumption in city traffic and low exhaust emission values, by simply optimizing the compression ratio, the shape of the combustion chamber and the valve timing. The technical know-how is available, it would be necessary, however, to make customers change their habits completely.

CONCLUSIONS.

Already today, the future development of the passenger car engine is decisively determined by energy and ecology requirements. The reduction of the energy consumption being of absolute priority, the optimization of the engine aiming at a minimum fuel consumption is of increasing importance. These efforts must be accompanied by measures taken to reduce the exhaust emission. Both objectives can be easily complied with at

an operational air/fuel ratio $\lambda > 1.0$.
In the future, chances will be good for those combustion systems guaranteeing an impeccable operation within this air/fuel-mixture range.

The Otto engine with a lean air/fuel-mixture is an efficient evolution of its traditional version. It offers good possibilities to reach fuel consumption values, as low as those known from the diesel engine, with a decisive improvement of the exhaust emission and a slight increase of the manufacturing costs. The compliance with extremely low HC and CO-emission values, however, necessitates the use of exhaust gas aftertreatment devices. The further development of the leanburn Otto engine might be handicapped by the eventual introduction of severe NO_x-emission requirements.

The alternative would be the stratified charge engine, permitting the compliance with severe NO_x-requirements and offering favourable fuel consumption and good driveability at the same time. Due to the production cost increase involved, this system will have no chances to be introduced on a broad basis as long as the emission requirements can be met with less expensive measures.

The use of the diesel engine in passenger cars will be increasing in the future. Its mass introduction, however, will be impaired by the question of fuel availability on one hand the yet unsolved environmental problems of annoyance by soot, odour and noise emissions on the other.

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Captions for all illustrations.

Fig. 1 : Efficiency Curves of the Otto Engine.

Fig. 2 : Influence of the Shape of the Combustion Chamber on the Flame Propagation Velocity and the Maximum Pressure.

Fig. 3 : Influence of the Shape of the Combustion Chamber on Exhaust Emission and Fuel Consumption.

Fig. 4 : Influence of the Compression Ratio on Fuel Consumption, Exhaust Emission, and Lean Mixture Limit.

Fig. 5 : Influence of the Combustion Chamber Shape and Compression Ratio on the Lean Mixture Limit and the Knock Limit.

Fig. 6 : Influence of the Ignition Energy on the Operation Parameters of the Otto Engine.

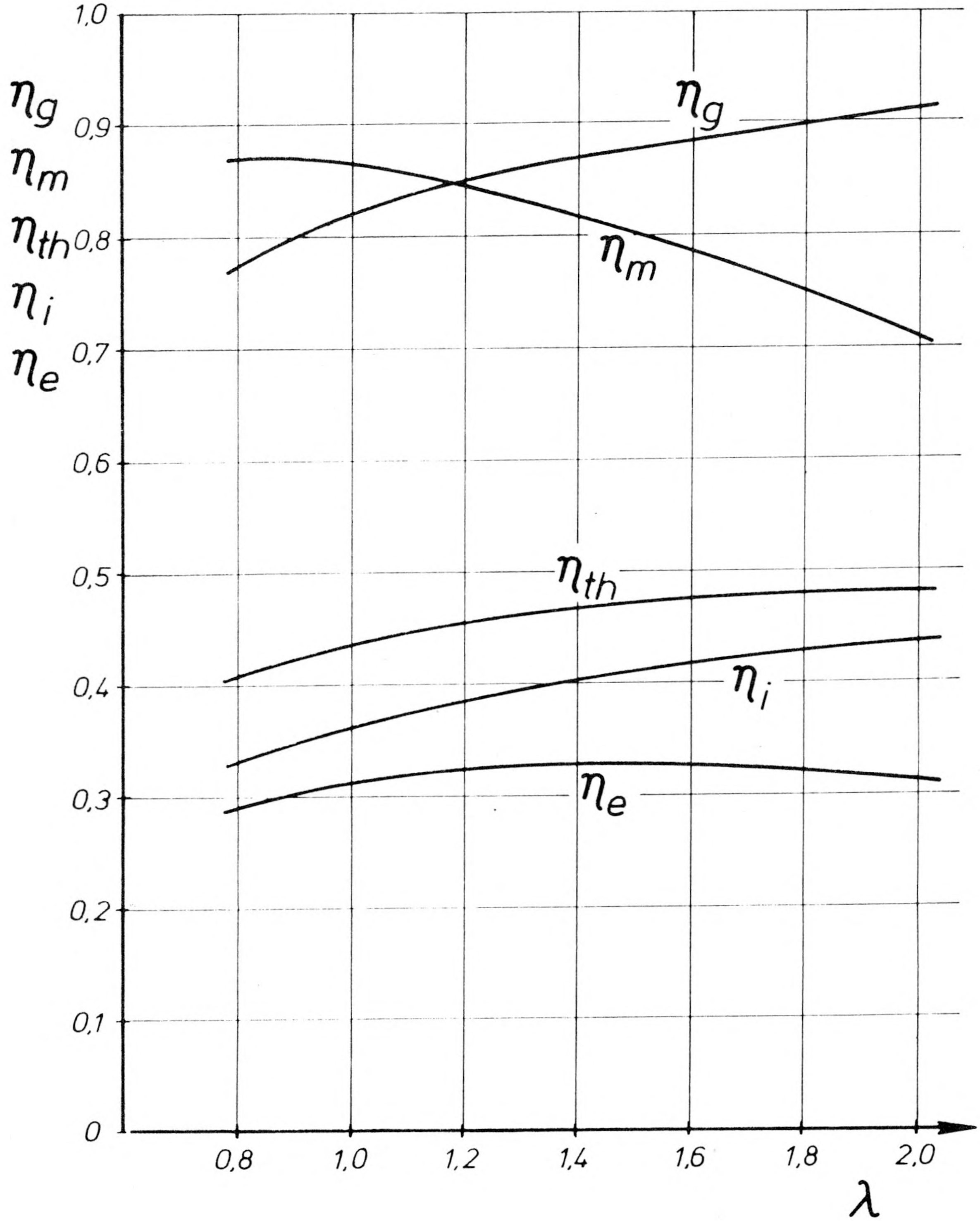
Fig. 7 : Exhaust Emission and Fuel Consumption with PORSCHE Vehicles.

Fig. 8 : Comparison of the Combustion Systems.

Fig. 9 : Influence of Orifice to Auxiliary Chamber Volume.

Fig. 10 : Octane Rating of the PORSCHE 911.

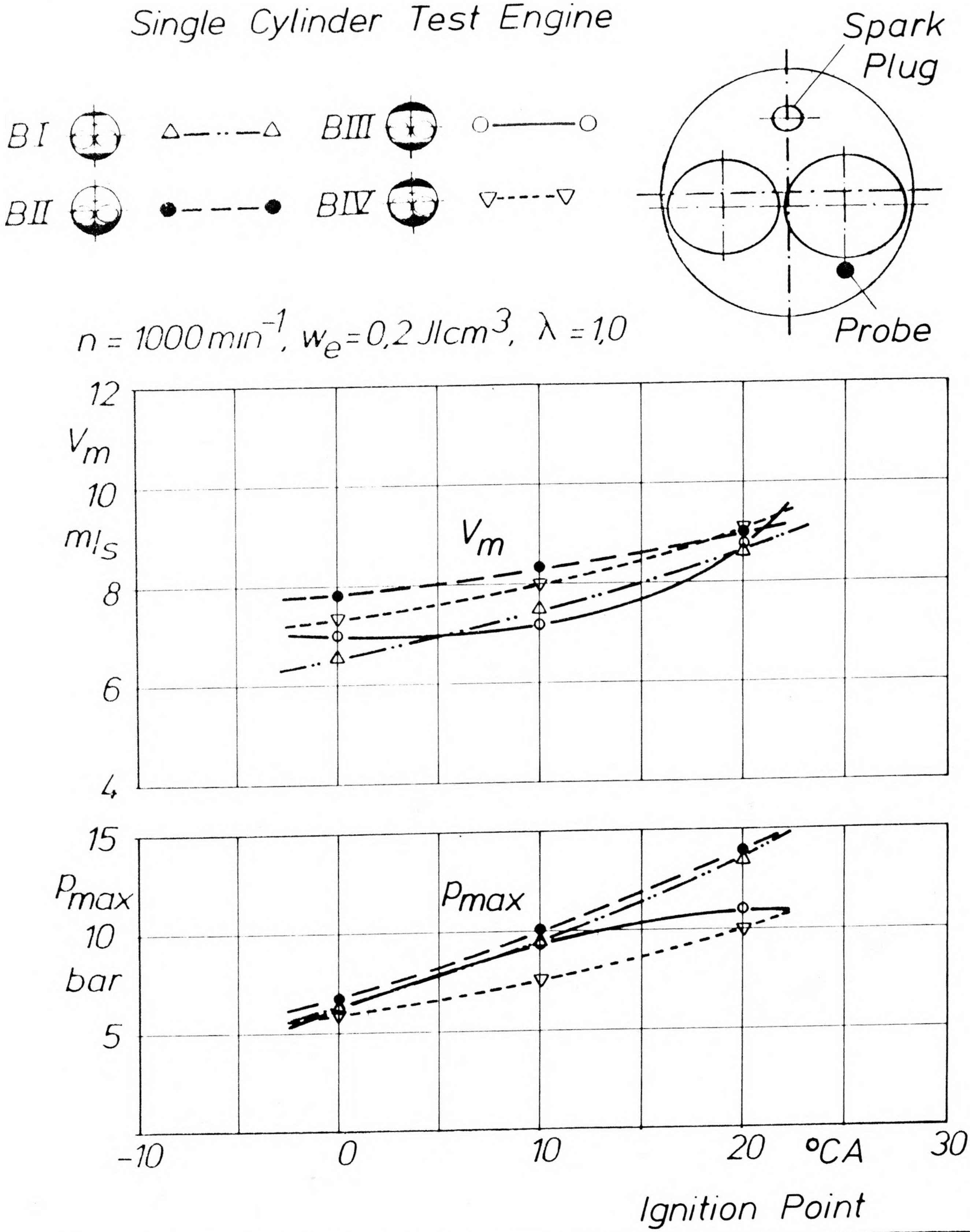
Fig. 11 : CVS and ECE Test Results obtained with a PORSCHE 911 SKS Engine.



PORSCHE

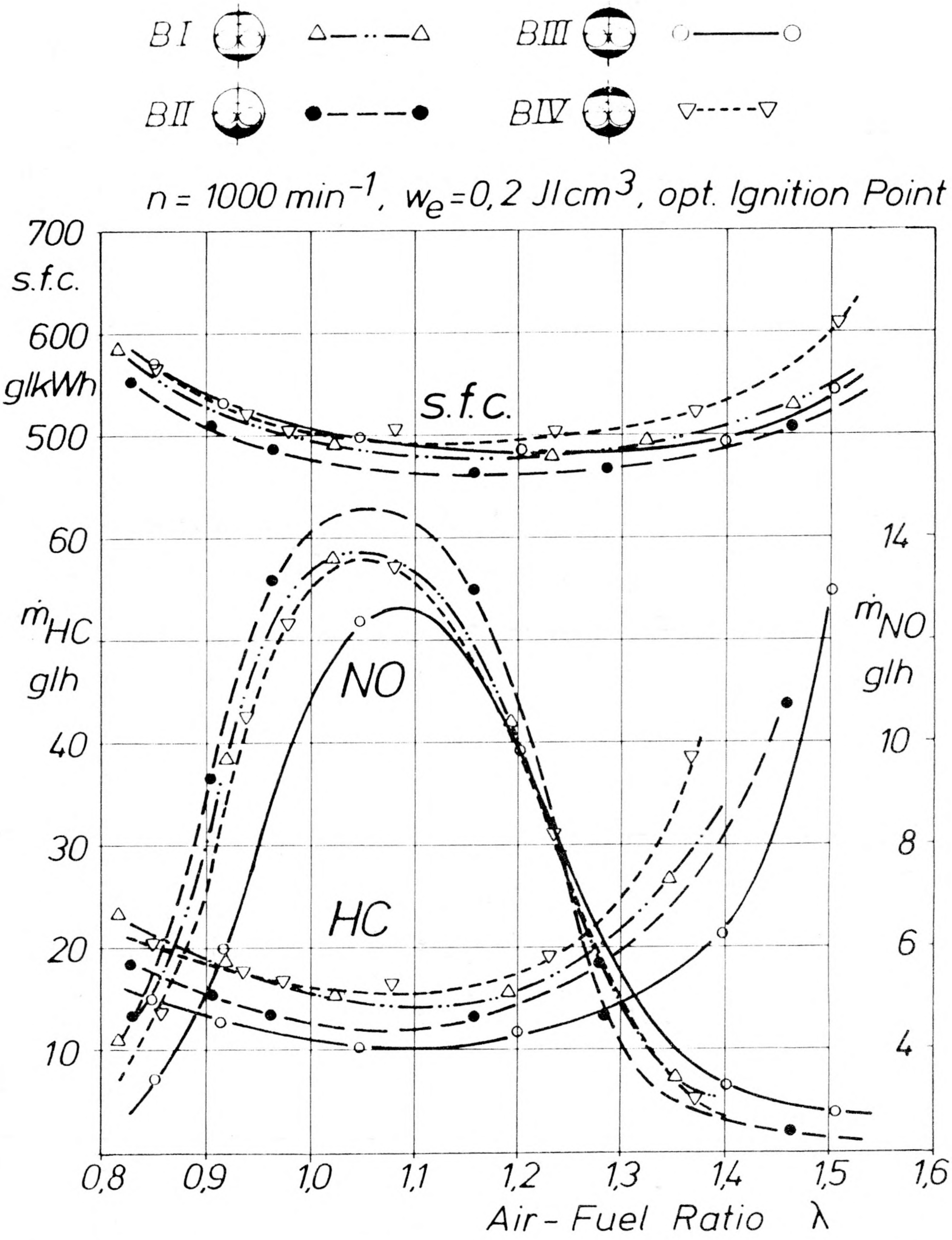
Efficiency Curves of the Otto Engine

1

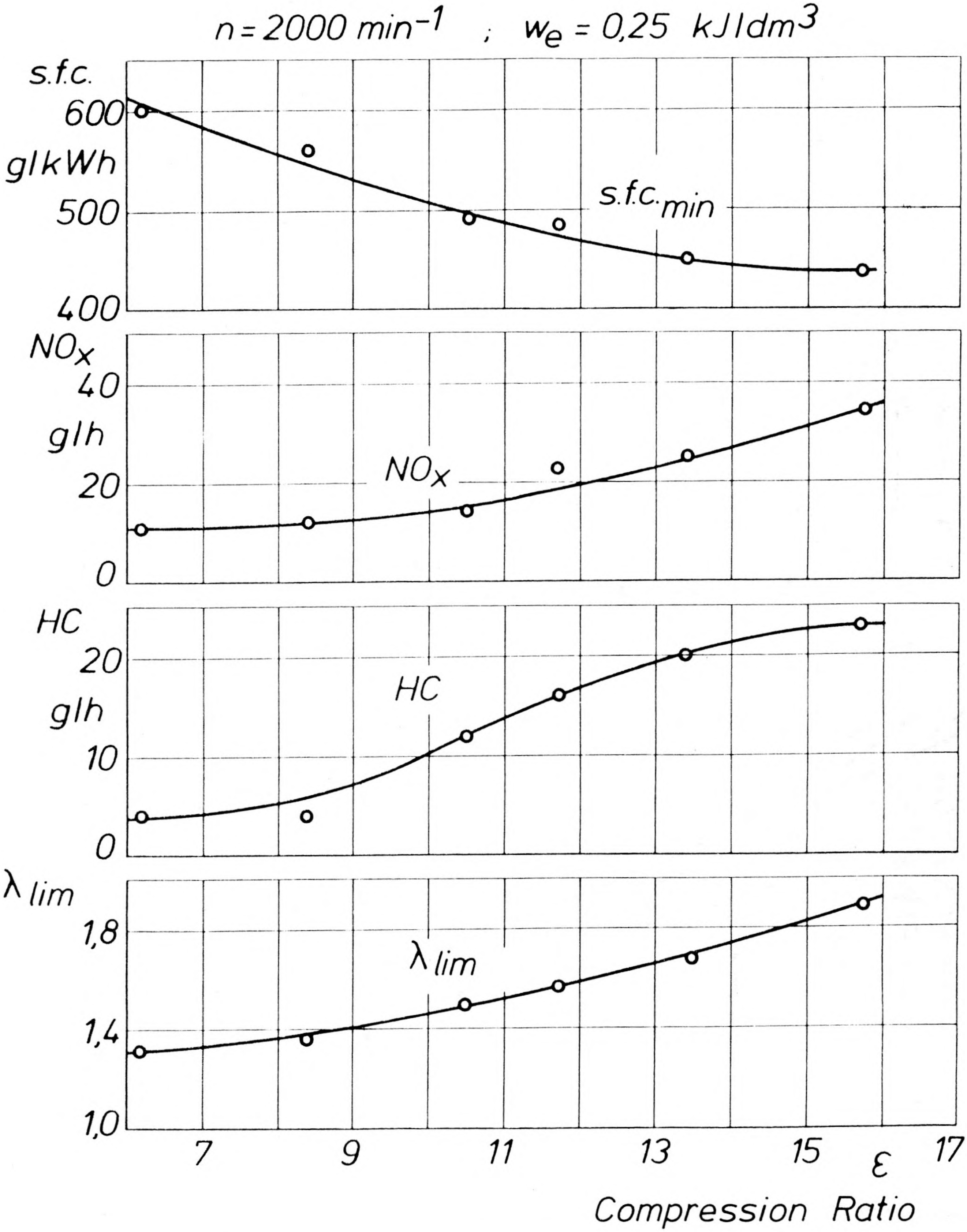


Influence of the Shape of the Combustion Chamber on the Flame Propagation Velocity and the Maximum Pressure

2

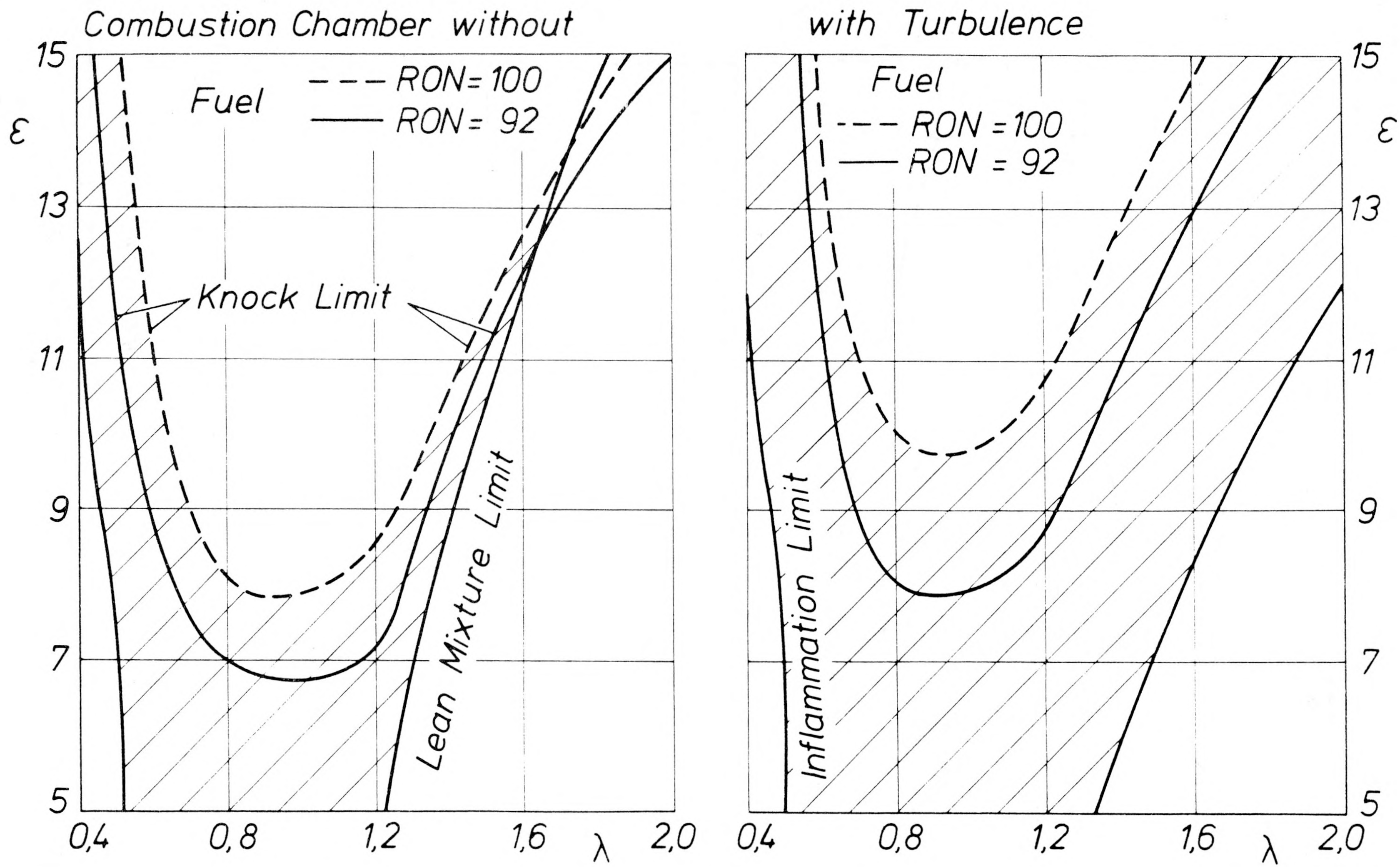


PORSCHE	Influence of the Shape of the Combustion Chamber on Exhaust Emission and Fuel Consumption	3



Influence of the Compression Ratio on Exhaust Emission, Fuel Consumption, and Lean Mixture Limit

4



Influence of the Combustion Chamber Shape and Compression Ratio on the Lean Mixture Limit and the Knock Limit

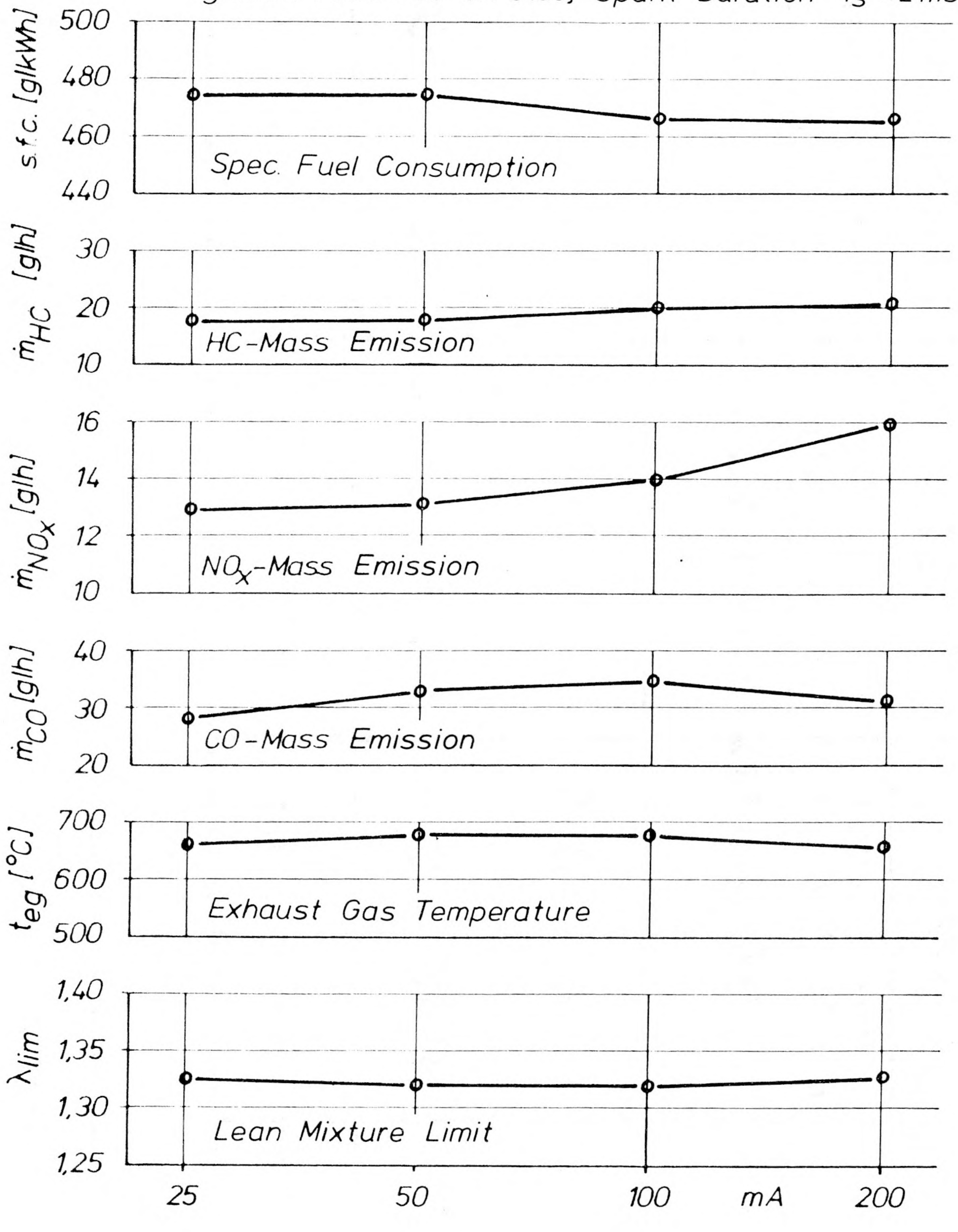
5

Single Cylinder Test Engine

$n = 2000\ min^{-1}$ $\varepsilon = 10{,}5$

$w_e = 0{,}24\ J/cm^3$ $\lambda = 1{,}2$

Ignition Point = 30° CA btdc, Spark Duration $T_S = 2\ ms$

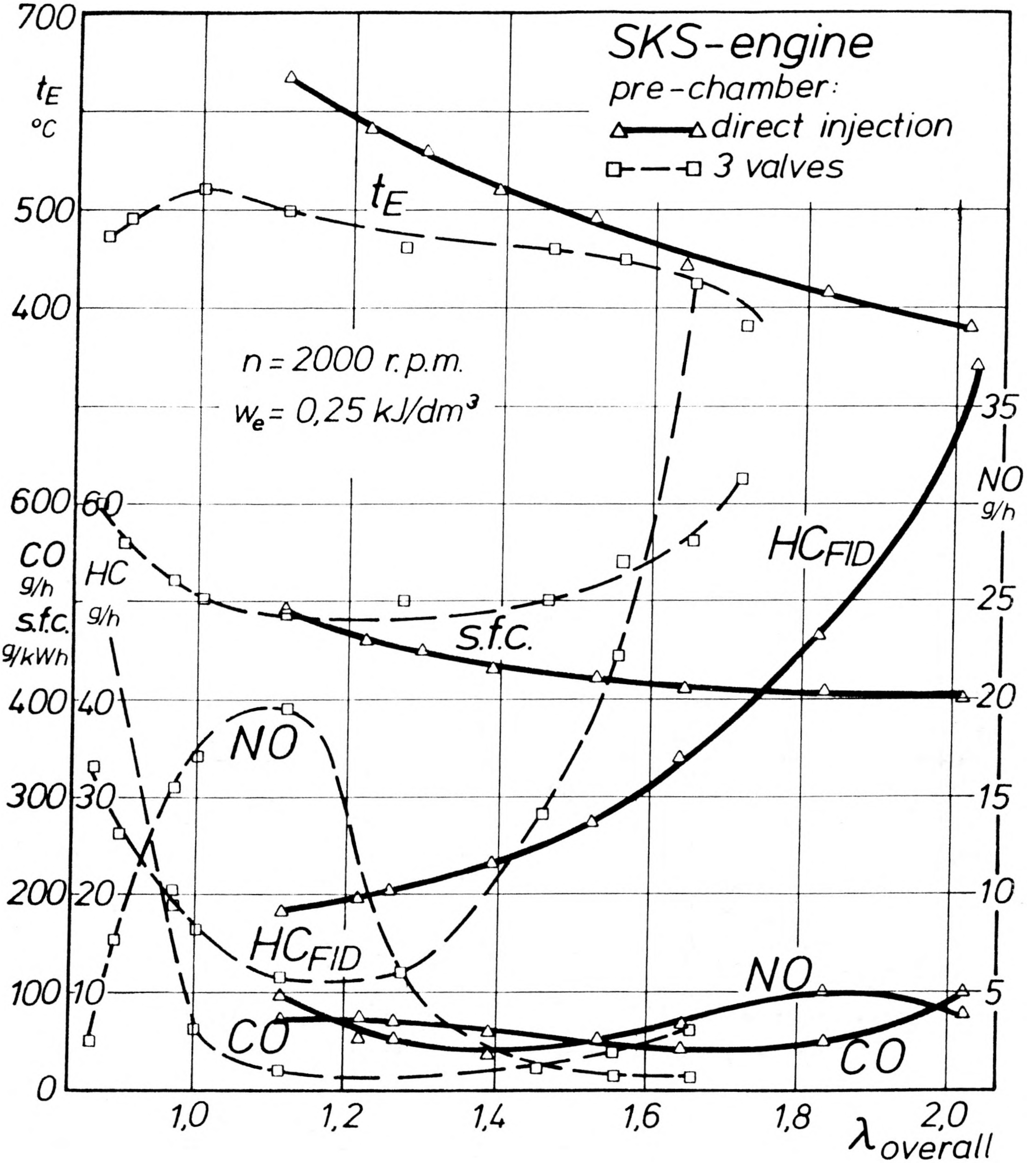


Influence of the Ignition Energy on the Operation Parameters of the Otto Engine

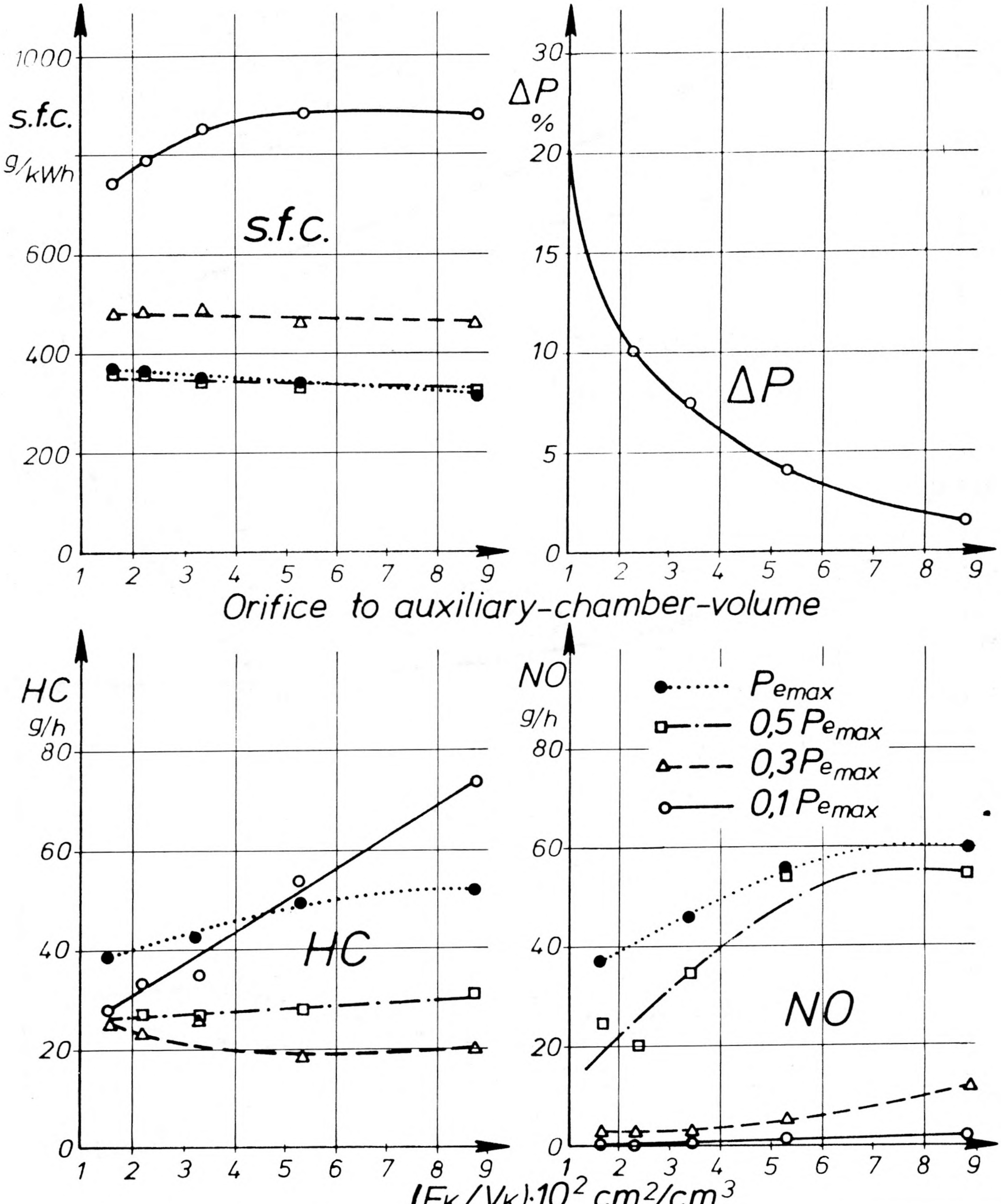
6

	924		911		ECE-Limits	
	Production	Lean-Burn	Production	Lean-Burn	1976	1980
CO g / Test	60-95	20-50	60-90	20-50	107	70
HC g / Test	2,5-4	3-6	2-3	4-7	8	6
NO_x g / Test	8-10	3-4	4,5-6	2,5-4	12	10
Fuel Consumption* l/100 km	8,4	7,8	10,2	9,2		

(* EPA - Composite Fuel Economy)

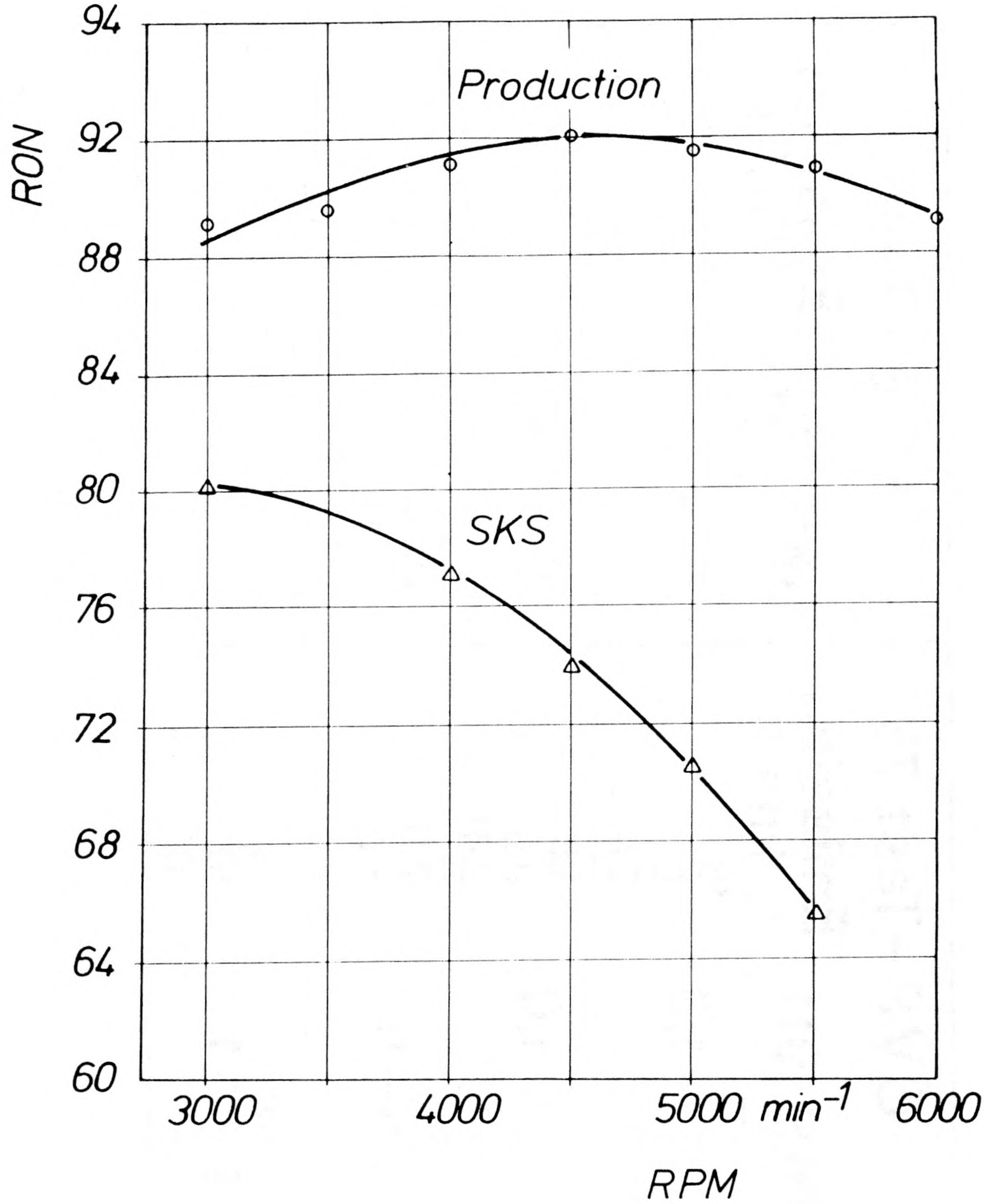


Comparison of the combustion processes



Influence of orifice to auxiliary-chamber-volume

9



Octane Requirement of the Porsche 911

	CVS-Test 75		ECE-Test	
	SKS-911	Emission limits	SKS-911	Emission limits
CO	0,2 – 1,3 g / mile	15 1976 9 1976 Cal 3,4 1979	0 ÷ 5,0 g/Test	107 1976 70 1980 32 1980
HC	0,4 ÷ 1,0 g / mile	1,5 1976 0,9 1976 Cal 0,41 1979	0,9 ÷ 5,5 g / Test	8 1976 6 1980 1,8 1980
NO_x	0,28 ÷ 0,7 g / mile	2,0 1976 1,0 1981 0,4	0,75 ÷ 2,0 g / Test	12 1976 10 1980 1.2 1980
B	Composite 19 ÷ 21 MPG	18 1978 19 1979 20 1980		

Development of a Compact Gas Generator for Fuel Gasification aboard a Motor-Vehicle

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Abstract

A gas generator suitable for installation in motor vehicles has been developed which converts liquid to gaseous fuel by partial oxidation with air (A/F < 1.5).

The reaction proceeds nearly soot free and the reaction temperature is self stabilizing.

Test bed measurements of an engine gas generator combination under low and medium load conditions show that in comparison with a gasoline fueled engine the specific fuel consumption is reduced by 15 % and is accompanied by a decrease of the NO_x emission by 93 %.

A cost benefit analysis revealed that for an NO_x limit of 0.4 g/mile the gas generator/engine combination will become superior to the stratified charge engine, the diesel engine and the three way catalyst system above 5000 km/year.

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1. Introduction

The use of gaseous fuels in internal combustion engines represents a suitable means to reduce the specific pollutant emission and the specific fuel consumption when compared with conventional carburetor operation. Gas engines can be operated far into the lean region due to the homogeneous miscibility of fuel gas and air. The fuel gas can be stored either in liquid form or in compressed gas cylinders. This gives rise to supply problems and makes it difficult to apply this technique to motor-vehicles. It is more favourable to generate the low molecular fuel gas from easily storable liquid hydrocarbons within a gas generator aboard the motor-vehicle. If the motor is operated solely with gas, then the primary fuel may have a low octane rating and may be lead-free, since as far as the engine is concerned only the high octane-rating of the gaseous fuel produced is relevant. Consequently processing steps in the refinery to increase the octane-rating can be omitted and result in a crude oil saving amounting to about 15 %.

A gas generator for operation aboard a motor-vehicle should comply with the following requirements:

- adiabatic operating mode with a view to rapid load changes
- easy control even in the presence of rapid load changes
- high efficiency
- low volume as well as weight

- independence from external water supply
- catalyst life time > 300 hours
- operation at atmospheric pressure
- soot-free operation

These requirements may be realized through the catalytic partial oxidation of hydrocarbons with air at low air/fuel mass ratios (A/F) and not in thermodynamic equilibrium. A suitable catalytic process, operating with an A/F ratio of 0.7 to 1.7 at about 820 °C, has been developed /1-3/.

2. Gasification Process

The schematic representation of the gasification process is shown in Figure 1. Gasoline and air in an A/F ratio of about 1.4 are heated from ambient temperature in a heat exchanger by the sensible heat of the product gas. They are then fed onto the catalyst bed within the reactor. At the inlet of the catalyst bed oxidation processes occur primarily and result in a temperature determined by the exothermal heat of reaction. Due to endothermal secondary reactions of the remaining hydrocarbons with the oxidation products the temperature then drops in the direction of the outlet side of the catalyst bed. Because of the short residence time of the reactant gases in the reactor thermodynamic equilibrium is not established and the reaction proceeds to a large extent soot-free. Decreasing the load of the reactor increases the time the reactants are in contact with the catalyst. Thermodynamic equilibrium is approached more closely in this case and at a constant low A/F ratio the danger of soot-formation is given. Soot-formation can, however, be eliminated by a simple load-dependent regulation of the A/F ratio at the reactor inlet.

The product gas leaving the reactor is cooled further in the heat exchangers, is then mixed with the air required for combustion in a separate mixer to be sucked into the intake manifold of the combustion engine.

3. Gas Generator

The construction of the gas generator is shown in Figure 2. A reactor of 10 cm diameter and lined with Al_2O_3 is filled with 300 cm^3 of catalyst. The catalyst in pellet form is rigidly fixed by two porous Al_2O_3 plates. Behind the reactor inlet port ceramic distribution vanes homogenize the fuel/air-mixture and then feed it uniformly onto the catalyst bed. In order to be able to realize good thermal insulation, the gaseous reaction products are recirculated twice over the exterior jacket of the reactor; as a consequence the heat losses through the jacket are negligible in comparison to the heat of reaction.

With a maximum reaction temperature of 820 °C and an A/F ratio of 1.35 a throughput of up to 26 l/h of straight run gasoline is possible. This corresponds to a loading of the catalyst bed with an hourly volume space velocity (HVSV) of 86 l/l h and a maximum residence time of reactants in the catalyst bed of about 10 ms. The efficiency of gasification under these conditions amounts to about 95 % and the load-dependent conversion to about 75 %.

Figure 3 shows the temperature profile of the catalyst bed under a load of 21 l/l h and for an A/F ratio of 1.35. The temperature of the boundary regions lies only minimally below that of the reaction zone.

Figure 4 shows the composition of the gaseous products as a function of the preheating temperature of the air/fuel mixture for a catalyst loading of 15 l/l h and an A/F ratio of 1.45.

The concept of a gas generator with integrated starter, exchangeable catalyst filling and gasoline/air preheating stages, which is suited for production purposes, is shown in Figure 5.

4. Reactor Stability

A gas generator suitable for operation with internal combustion engines must be able to follow rapid load changes. This is only possible if the reactor possesses self-stabilizing characteristics, which, for the adiabatic partial oxidation of gasoline with air, is fortunately the case below an A/F ratio of 3.

In an adiabatic reactor the heat of reaction generated per unit time is carried off by the stream of gaseous products. In an exothermal reaction the heat of reaction is proportional to the conversion, increasing initially exponentially with the temperature to reach saturation at elevated temperatures. The heat of reaction curve as a function of temperature thus processes an S-shape. In the process of oxidatively cracking gasoline, endothermal cracking processes and also endothermal reactions between hydrocarbons and the primary oxidation products CO_2 and H_2O follow with increasing temperature. This causes the heat of reaction to pass through a maximum and then drop. This behaviour is shown in Figure 6. Since the endothermal processes taking place do also increase exponentially with temperature, the drop is very rapid /4/.

The loss of heat from the reactor is a nearly linear function of the reaction temperature. If the slope of the heat of the reaction curve at the inflection point is steeper than that of the straight line for the heat losses, then three points of intersection result for a certain range of inlet temperatures. One of these points of intersection corresponds to the "on" state of the reactor with high conversion and one corresponds to the "off" state with negligible conversion. The central mode of operation is not a stable one.

For the gasification of straight run gasoline (boiling range 80 °C to 200 °C) with air at an A/F ratio of about 1.45, the working range lies within the region of steep descent of the heat of reaction, provided the reactor inlet temperature ranges from 200 °C to 350 °C. Within a certain range the reactor temperature is practically independent of fluctuations in the inlet temperature, as might occur, for instance, during load changes as a result of the sluggishness of the exchange of heat in the product gas coolers. The reactor is self-stabilizing. The danger of soot formation sets in only with approach to the minimum of the heat of reaction curve.

The mathematical model-description of reactor stability and composition of crack products starts with the assumption that essentially two factors will determine the progress of the cracking process. These factors are the cracking of C-C bonds of the hydrocarbons and the liberation of hydrogen /5/. The catalyst effects the liberation of about one hydrogen atom per C-C bond cleaved. After all C-C bonds have been broken, the system reaches absolute equilibrium by splitting off further H-atoms. Assuming now that the path of the reaction will proceed along the direction of steepest descent of the free enthalpy surface, while obeying, of course, the boundary conditions mentioned above, the oxidative cracking of heptane, for instance, according to the reaction

$$C_7H_{16} + 0.88\ (O_2 + 4\ N_2) \rightarrow \text{gaseous products}$$

will result in a dependence of the gas composition on the degree of cracking a_{cc}, as shown in Figure 7 for a temperature of 800 ^oC. (a_{cc} designates the number of cracked C-C bonds per heptane molecule). The paraffins C_2H_6 to C_6H_{14} are collectively referred to as C_nH_{2n+2}, the olefins and diolefins as C_nH_{2n}. In addition the ratio C_s/C_{tot} of soot to total carbon content is shown.

A closer consideration allows three stages of the reaction process to be distinguished:

1. Oxidation of part of the available hydrocarbon to yield CO_2 and H_2O; the heat of reaction increases during this stage.

2. Endothermal follow-up reactions (chain cleaving and platforming processes); the heat of reaction decreases during this stage.

3. Soot formation; the heat of reaction again rises in this phase.

The corresponding dependence of the heat of reaction on the instantaneous degree of cracking is shown in Figure 8. The independent variable a_{cc} as abscissa represents the number of C-C bonds cracked and consequently the time of reaction also. Since the degree of cracking at constant time increases with temperature, Figure 8 shows likewise, at least in principle, the behaviour of the heat of reaction as a function of temperature at constant resident time.

One thus arrives again at Figure 6.

5. Test Bench Measurements

The combination of a gas generator and an internal combustion engine was tested on a stationary test bench, utilizing a 1.6 l VW engine.

5.1 Constant Load Operations

It was possible to show that under constant load conditions, especially in the lower and medium power range, the specific fuel consumption and the specific pollutant emission can be considerably reduced in comparison with operation using normal carburetion. For a medium power value a comparison is given in Table 1.

Table 1: Specific fuel consumption and specific pollutant emission of a 1.6 l VW engine at 2000 rpm, operating with a normal carburetor and with a gas generator.

	A/F	IT °BTDC	mep bar	P kW	bsfc g/kWh	NO_x g/kWh	CO g/kWh	HC g/kWh
Engine with normal carburetor	16.7	35	5.0	13.3	319	12.4	27.9	2.0
Engine with gas generator	22.5	40	5.0	13.3	276	0.74	3.6	1.5

Figures 9 and 10 show a comparison of the specific fuel consumption and the specific NO_x emission as a function of the A/F ratio. Figure 11 shows the relationship between the relative specific fuel consumption, the ignition timing, and the A/F ratio. It is easily recognized, that during operation with crack gases the fuel consumption is not so dependent on ignition timing as for normal carburetor operation. Especially for small ignition angles, that is for low specific NO_x emission, the operation with gas is superior.

5.2 Load Changes

In order to investigate the behaviour of the combination gas generator/combustion engine under load changes, the input to the gas generator, a preheated gasoline/air mixture at 200 °C, was increased instantaneously and the torque of the motor at constant r.p.m. registered. Figure 12 shows the behaviour of the torque of the engine and the temperature of the catalyst bed in the gas generator. The torque reaches its new end value within 1.75 sec after changing the input from 5.4 to 9.3 l/h instantaneously while simultaneously reducing the A/F ratio of the engine from 25 to 14. Consequently the load change behaviour of the gas driven engine being accelerated out of the lean range is at least as good as that of a normal engine operating in the rich range.

As was to be expected the temperature of the catalyst within the gas generator changed only slightly during the load change.

6. Costs/Benefit Analysis

From the viewpoint of pollutant emission the only engines competing with the motor fuelled by a gas generator are the Diesel engine, the stratified charge engine, and the motor with a catalytic aftertreatment of exhaust gases. Within the complex "aftertreatment" a reactor with a bi-functional catalyst for NO_x reduction and simultaneous HC and CO oxidation is considered to be the most promising solution.

6.1 Pollutant Emission and Fuel Consumption

In an internal combustion engine the NO_x emission increases and the CO and HC emission decreases with increasing efficiency. The relationship between NO_x emission and specific fuel consumption during a CVS-test is shown in Figure 13. The representation is based in part on results taken from a 1974 report of the National Academy of Sciences /6/ and referring to vehicles with a displacement of 1.5 - 2.0 l and a weight of 2500 - 3000 lbs.

Since for the gas generator driven motor only stationary test bench measurements are available, these results were converted to CVS-values by empirical means. The presently available data indicate, that the HC limit of 0.41 g/mile can only be realized with an aftertreatment of exhaust gases, at least as long as no differentiation is made between smog-inert, non-poisonous paraffins and the remaining hydrocarbons.

6.2 Benefits

An advantage is to be seen in the low pollutant emission of the competing systems. In this respect the competing systems are nearly equivalent, with the exception of the Diesel engine.

6.3 Costs

The benefits have to be weighted against the specific mileage costs. They are determined by the costs for maintenance, lubricants, fuel per unit distance, as well as the capital costs (interest, amortization) referred to the yearly mileage covered; all other costs such as taxes, insurance etc. are not considered.

6.4 Calculation of Costs

The data required for the calculation are given in Table 2, namely data for a conventional motor vehicle and for the competing systems.

The specific maintenance costs for the competing systems are estimates; they include repair costs.

The prices for lead-containing regular and premium gasoline are those paid at German service stations in December 1975. The prices for lead-free (RON 92) and for straight run gasoline are estimates based on literature data /7/.

The fuel consumptions are average values according to Figure 13. The specific mileage costs were calculated with the following assumptions:

- Interest 8 % p.a.
- Amortization period 8 a
- Yearly mileage 12.500 km
- Residual book value of the gas generator motor at the end of the amortization period = 50 % of purchase price of an exchange motor.

6.5 Specific Mileage Costs

For a fixed NO_x limit of 0.4 g/mile Figure 14 shows the absolute specific mileage cost and Figure 15 the difference to the specific mileage cost of a conventional motor vehicle, both as a function of the yearly mileage.

The diagrams show that, with the assumptions made, the gas generator driven vehicle has the lowest specific mileage costs of the competing systems for a yearly mileage of 5000 km and above. Of course the differences are relatively small. The higher purchase price for the gas generator motor is overcompensated, however, by a lower fuel consumption and the use of cheaper fuel.

7. Conclusions

With the assumptions made, Figure 13 shows that the gas generator motor covers the capability range of the Diesel engine in the region of higher NO_x emission (> 0.6 g/mile) and the capability range of catalytic aftertreatment in the region of low NO_x emission (< 0.6 g/mile).

The concept of gasification presents an attractive solution to the problem of pollutant emission because of the low specific fuel consumption in connection with the modest requirements placed on the anti-knocking properties of the fuel (reduction of refining costs)

Table 2: Motor vehicle data required for cost/benefit analysis

Motor			Vehicle	Maintenance	Lubricants		Fuel	
No.	Type	Power	Purchase price incl. taxes	Costs incl. taxes	Costs incl. taxes	Price incl. taxes	$NO_x = 0.4$ g/mi	
							Consumption	Costs
		kW	DM	DM/km	DM/km	DM/dm^3	dm^3/100 km	DM/km
1	Normal carburetor	70	16 705	0.079	0.005	0.89	11.0 for $NO_x = 2.7$ g/mi	0.098 for $NO_x = 2.7$ g/mi
2	Stratified charged engine	70	17 315	0.081	0.005	0.82	13.0	0.107
3	Three way catalyst	70	17 357	0.083	0.005	0.86	11.0	0.095
4	Gas generator	70	18 620	0.083	0.004	0.75	10.5	0.079

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Figure Captions

Figure 1 Schematic representation of the autothermal process of fuel gasification by partial oxidation

Figure 2 Construction of a gas generator

Figure 3 Temperature profile of a gas generator as shown in Figure 2

Figure 4 Gas composition as function of the preheating temperature
A/F = 1.45; catalyst loading 15 $l \cdot l^{-1} \cdot h^{-1}$

Figure 5 Design of a gas generator suitable for production

Figure 6 Heat of reaction Q_R and heat requirements Q_A of an adiabatic gas generator

Figure 7 Change of the relative gas composition $\frac{a_i}{a}$ with continuing cracking of n-heptane

Figure 8 Heat of reaction Q_R as a function of the progress in n-heptane cracking

Figure 9 Specific fuel consumption of a 1.6 l VW engine as a function of the A/F ratio for operation with a stock carburetor and with a gas generator

Figure 10 NO_x emission of a 1.6 l VW engine as a function of the A/F ratio for operation with a stock carburetor and with a gas generator

Figure 11 Relative specific fuel consumption of a 1.6 l VW engine as a function of the A/F ratio for operation with a stock carburetor and with a gas generator

Figure 12 Torque of a 1.6 l VW engine with gas generator operation for an instantaneous change in the fuel supply

Figure 13 NO_x emission of the competing systems as a function of the fuel consumption under CVS-test conditions

Figure 14 Specific mileage costs of the competing systems as a function of the yearly mileage (CVS-test) for an NO_x limit of 0.4 g/mile

Figure 15 Difference of the specific mileage costs of the competing systems to those of a conventional vehicle as a function of the yearly mileage (CVS-test) for an NO_x limit of 0.4 g/mile

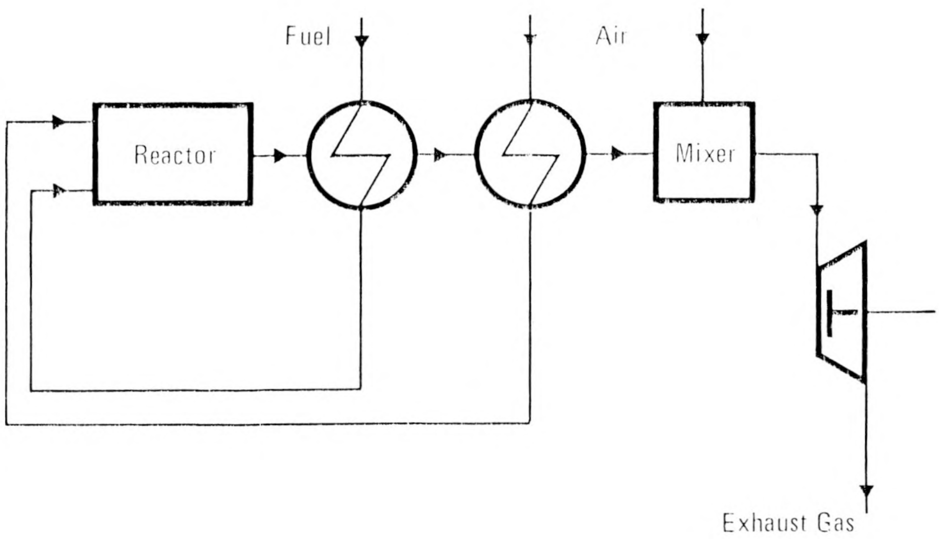

Figure 1: Schematic representation of the autothermal process of fuel gasification by partial oxidation

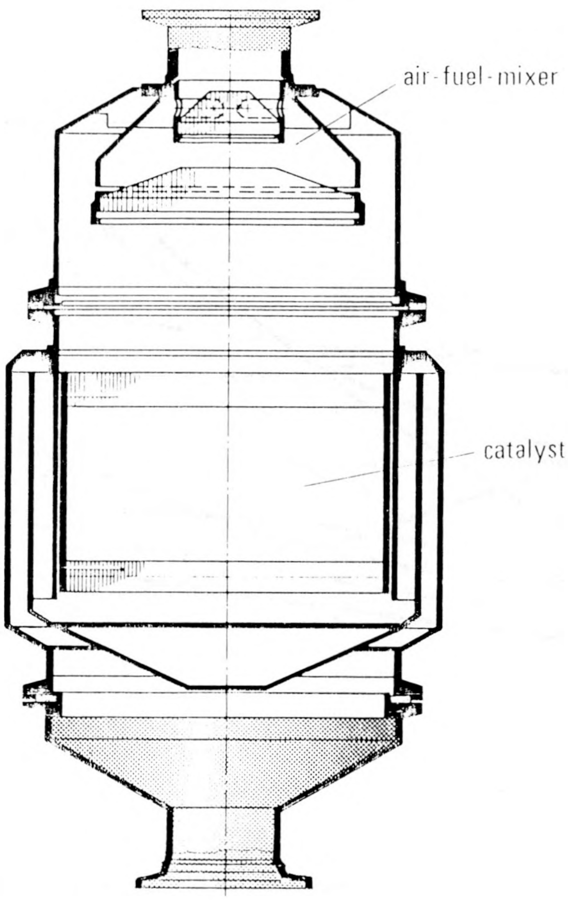

Figure 2: Construction of a gas generator

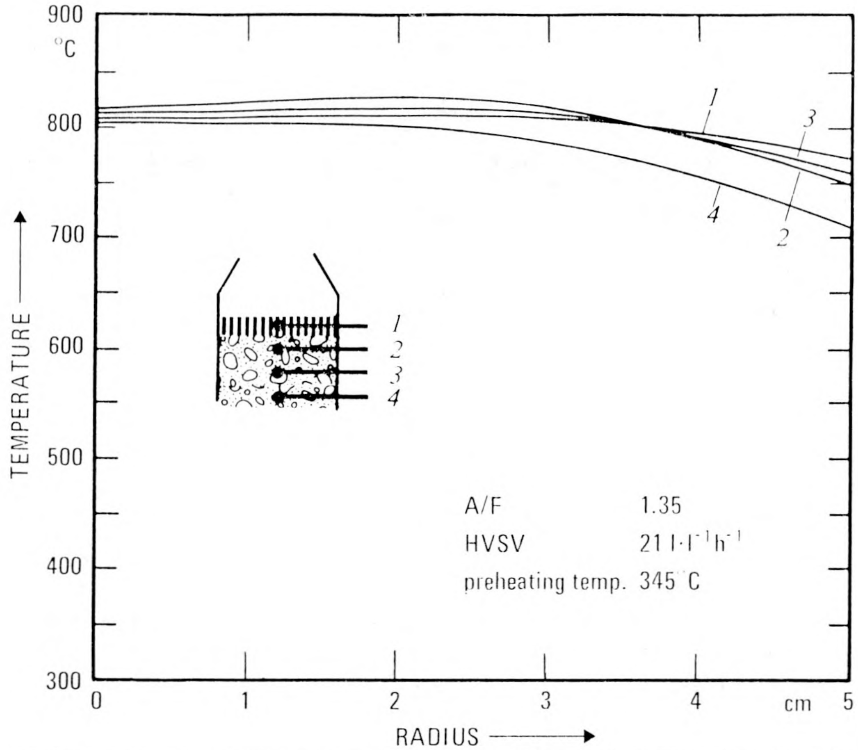

Figure 3: Temperature profile of a gas generator as shown in figure 2

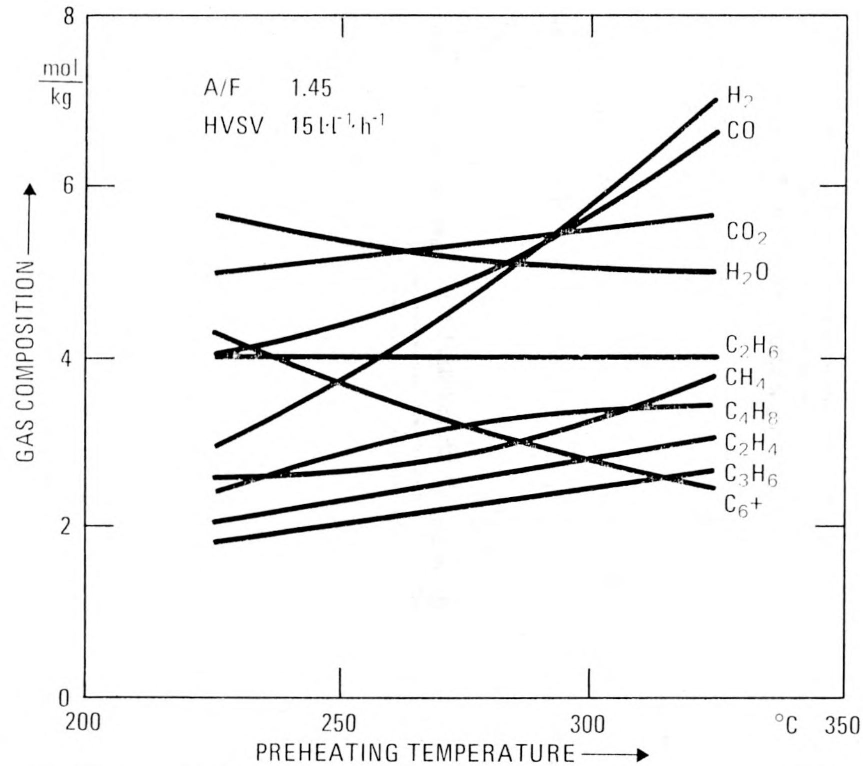

Figure 4: Gas composition as function of the preheating temperature

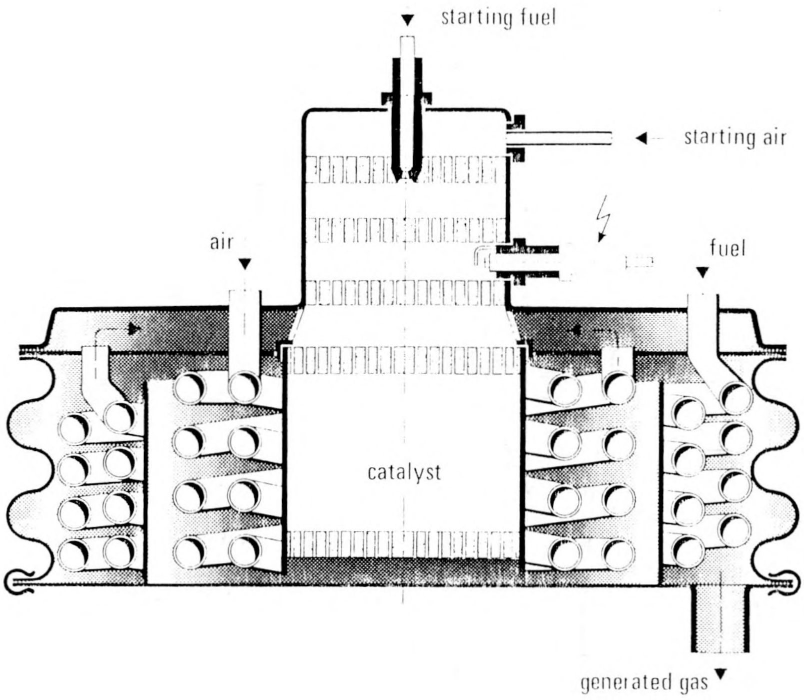

Figure 5: Design of a gas generator suitable for production

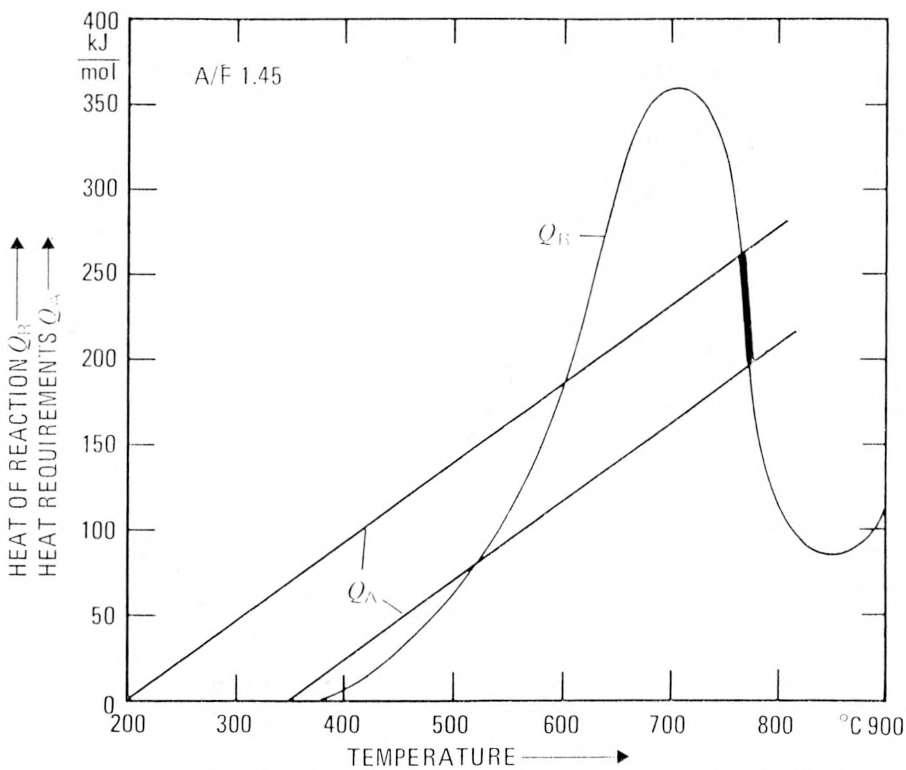

Figure 6: Heat of reaction Q_R and heat requirements Q_A of an adiabatic gas generator

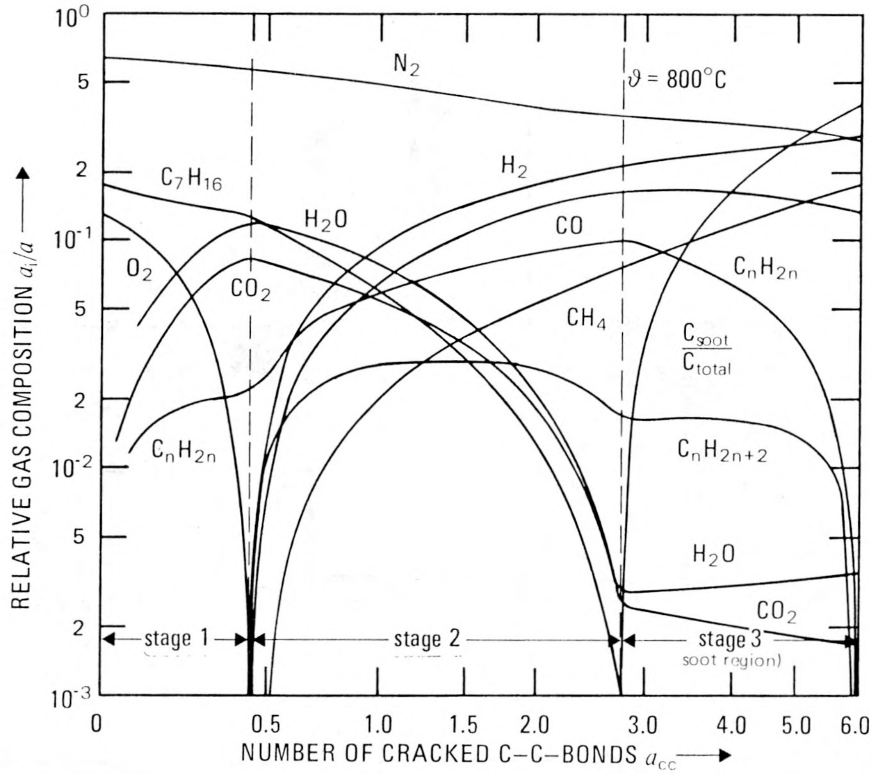

Figure 7: Change of the relative gas composition a_i/a with continuing cracking of n-heptane

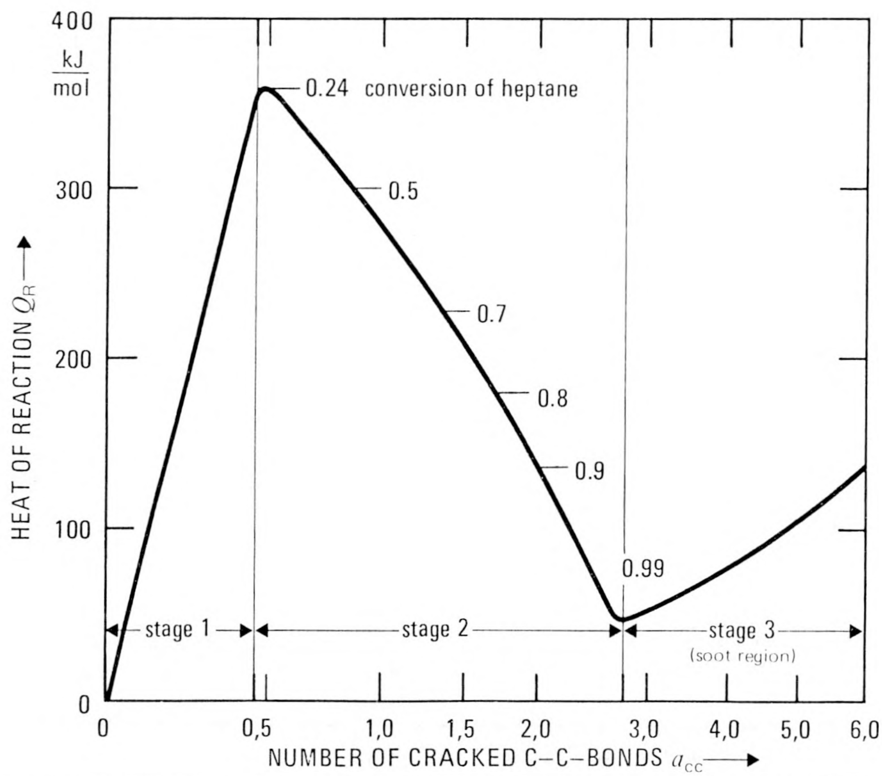

Figure 8: Heat of reaction Q_R as a function of the progress in n-heptane cracking

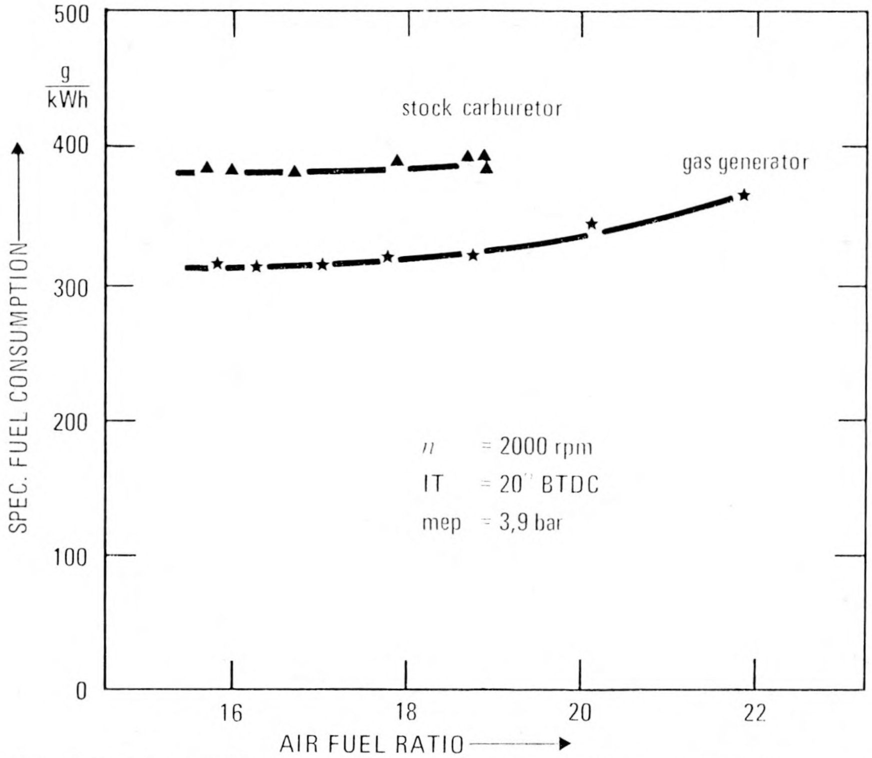

Figure 9: Specific fuel consumption of a 1.6 l VW engine as a function of the A/F ratio for operation with a normal carburetor and with a gas generator

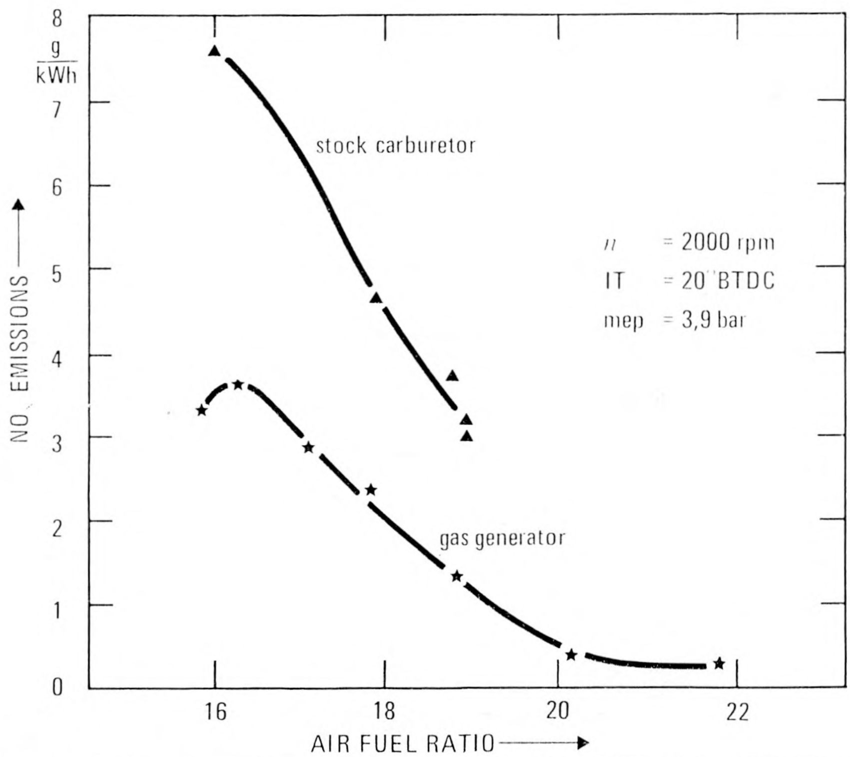

Figure 10: NO_x emission of a 1.6 l VW engine as a function of the A/F ratio for operation with a stock carburetor and with a gas generator

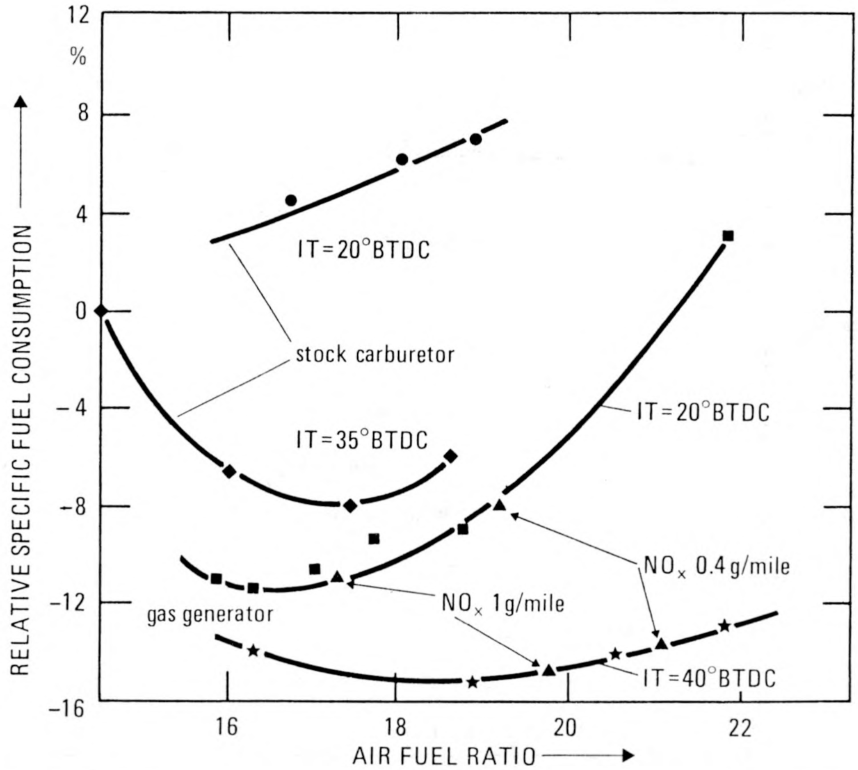

Figure 11: Relative specific fuel consumption of a 1.6 l VW engine as a function of the A/F ratio for operation with a stock carburetor and with a gas generator

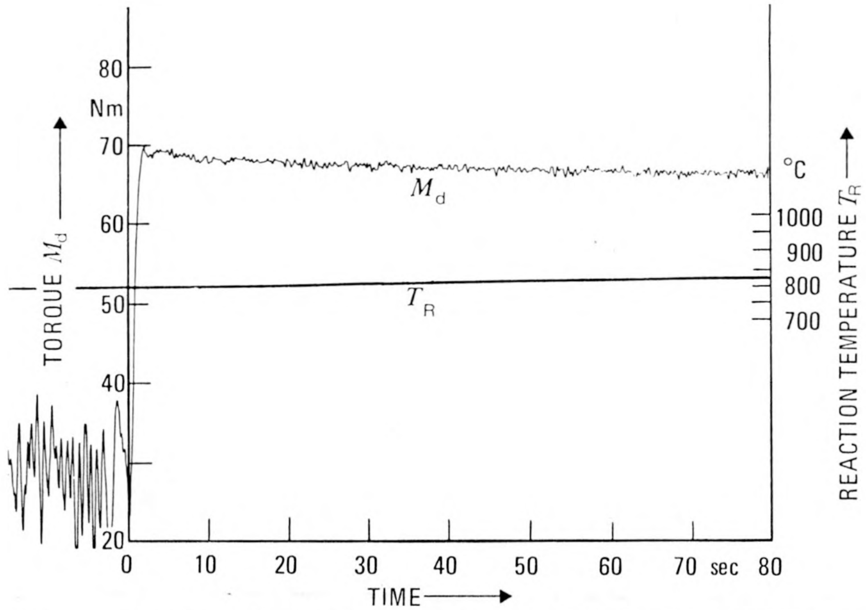

Figure 12: Torque M_d of a 1.6 l VW engine with gas generator operation and reaction temperature T_R for an instantaneous change in the fuel supply

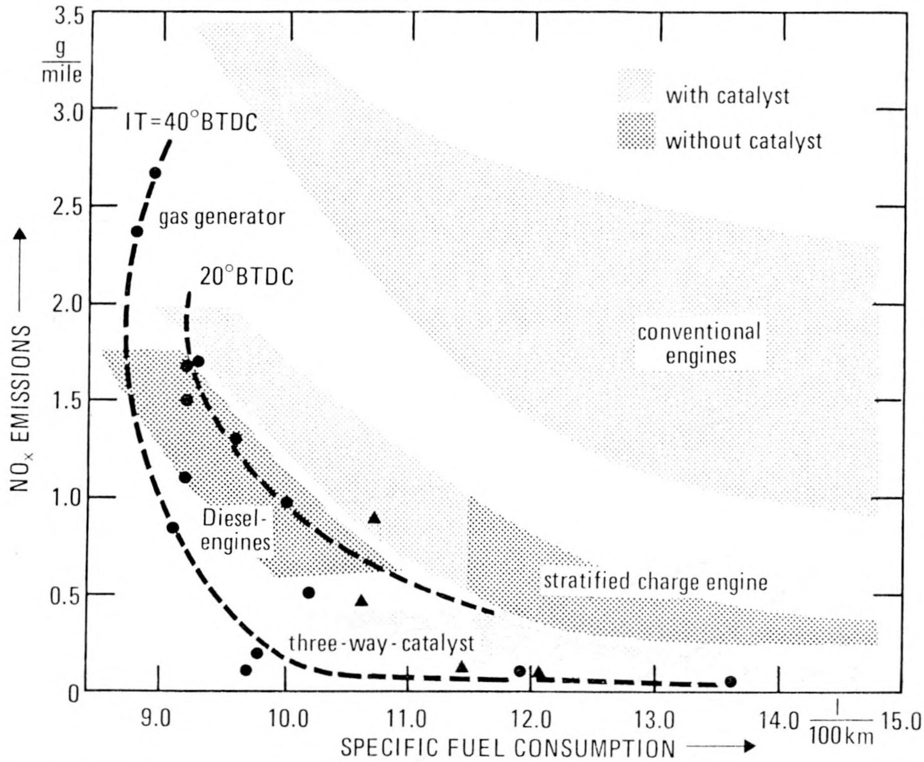

Figure 13: NO_x emission of the competing systems as a function of the fuel consumption under CVS-test conditions

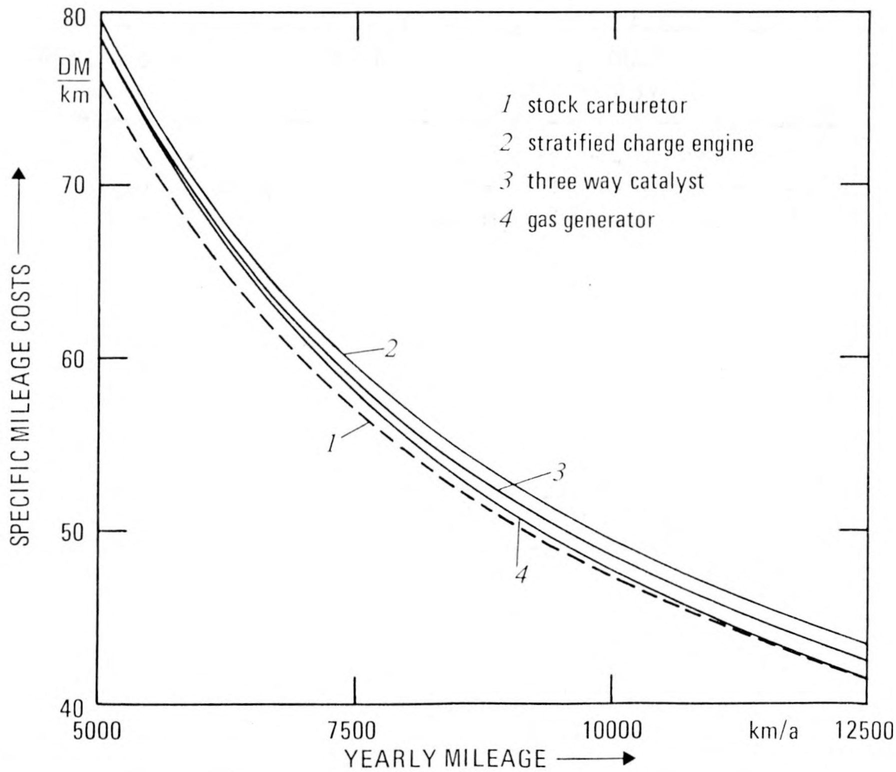

Figure 14: Specific mileage costs of the competing systems as a function of the yearly mileage (CVS-test) for an NO_x limit of 0.4 g/mile

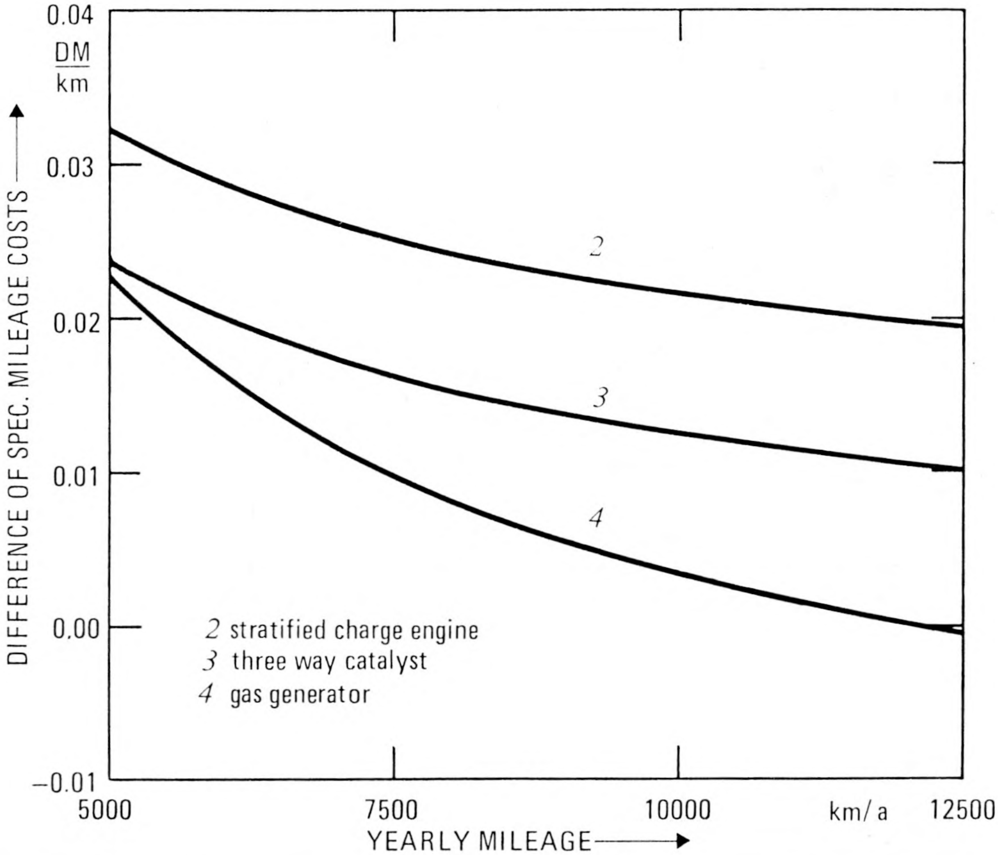

Figure 15: Difference of the specific mileage costs of the competing systems to those of a conventional vehicle as a function of the yearly mileage (CVS-test) for an NO_x limit of 0.4 g/mile

THE EFFECT OF VALVE SIZES ON FUEL CONSUMPTION AND EXHAUST EMISSIONS IN GASOLINE ENGINES.

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Abstract:

It has been found that, for a range of internal combustion engines, the use of exhaust valves significantly larger in area than inlet valves, can produce a reduction in fuel consumption together with a reduction of exhaust emissions with no loss of power or performance.

Introduction:

It has been the accepted practice of engine designers to use inlet valves with areas larger than (or at least equal to) the exhaust valves in order to optimise gas flow conditions in the engine. This assumption may have been based upon the reasoning that, whereas the exhaust stroke of the piston produces a positive pumping action, which can eject the combustion products efficiently through a relatively small aperture (the exhaust port), intake of air/fuel mixture into each cylinder, being generally brought about only by the depression in pressure caused by the movement of the piston away from the cylinder head, requires a relatively large aperture (the inlet port) for optimum operation of the engine.

It seems to have been accepted as a feature of generally overriding importance, in determining the sizes of inlet and exhaust valves, that the inlet valve flow should be made relatively large with a view to obtaining maximum volumetric efficiency (defined as the mass of fresh mixture which passes into the cylinder in one suction stroke, divided by the mass of the mixture which would fill the piston displacement at inlet density).

Many references to valve sizes appear to have been repeated by rote, there are however, some original source references such as those of Eppes et al[1] and Barnes-Moss[2]. The former concluded that the optimum ratio of exhaust to inlet valve diameters was 0.83 (this corresponds to an area ratio of 0.69). The latter concluded that the inlet valve should have a diameter between 0.48 - 0.50 D and the exhaust valve should have a diameter between 0.41 - 0.43D, where D is the diameter of the cylinder. These data give an exhaust to inlet diameter ratio of between 0.82 to 0.90 (corresponding to an area ratio of between 0.67 and 0.81).

It should be noted that Eppes et al, in their experimental work, did not actually increase the valve diameters but obtained their results by changes in valve lift that they assumed to be equivalent to their postulated changes in valve diameter. The conclusions of Barnes-Moss were derived from gas velocity calculations.

Accordingly, it has become generally accepted practice, as a normal design criterion for conventional production line engines to make the exhaust valves smaller in diameter than the inlet valves although occasional use has been made of equal sized valves in order to reduce the number of engine components. It should be noted that until 1971 the main emphasis was placed on the production of engines giving power and performance with fuel consumption a secondary factor except in compact and small vehicles. Over the period 1968-1973, engine design was modified by the demands of emission control regulations. Since the oil crisis of 1973 there has been much more interest in the production of efficient engines and it was to this end that the work reported in this paper was pursued.

Results:

Initial road experiments using a vehicle with a Ford 1200 c.c (72 cu.in) engine indicated that the use of exhaust valves greater in area than the inlet valves produced improvements in fuel consumption; it was also possible to run the engine on a lower octane fuel than was previously possible; no loss of power was observed, in fact in most cases a power gain was obtained.

The experiments were repeated with a Ford 1300 c.c (79 cu.in) engine using an engine dynamometer at the British Internal Combustion Engine Research Association (BICERA). The results are summarised in Figure 1. The standard ratio of exhaust to inlet valve areas was 0.79, combinations giving ratios of 1.07 and 1.26 were tested. In both cases all other variables were held constant and it was found that, in general, torque, power and fuel economy increased with increasing exhaust valve size. The 26% increase in exhaust valve area was the maximum permitted by the geometry of the cylinder head.

A wide range of experiments have since been carried out on the British Leyland 1000 c.c (60 cu.in) OHV A Series engines, on Ford 2000 c.c (120 cu.in) and 3000 c.c (180 cu.in) Vee and 1600 c.c (96 cu.in) OHC engines and General Motors Vauxhall 1800 c.c (108 cu.in) OHC engines.

These engines have differing combustion chamber designs but in all cases there have been observed improvements in fuel economy as well as reductions in exhaust emissions and the ability to run on low octane fuel with no loss of power or performance.

The majority of experiments have been carried out using the British Leyland 1000 c.c (60 cu.in) Mini engine. An exhaust to inlet valve area ratio of 1.39 (standard 0.83) was chosen to give exhaust valves significantly larger than the inlet valves. The results obtained may be summarised as follows:

Fuel Consumption:

Fuel consumption tests were made on the road by driving two vehicles, one standard, one modified,in convoy over a 377 mile route including urban, country and motorway sections. The order of driving and the drivers were changed at intervals. Fuel consumption was measured by refilling the fuel tank using a burette. For example, in a series of six convoy runs on different days in different traffic and weather conditions, the modified vehicle (which when standard, consumed 2% more fuel over the test route than the standard vehicle) was in all cases more economical than the standard vehicle and consumed on average 7.5% less fuel than the standard vehicle. This corresponded to an 8% reduction in fuel consumption following modification.

The most outstanding improvements were observed in a 39 mile section of the route through London and its suburbs where the modified vehicle used on average 19% less fuel than the standard vehicle.

It was found possible to run the vehicles on a lower octane fuel without making any adjustments to engine timing, fuel mixture etc. Indeed a modified British Leyland Mini was run on 92 octane gasoline free of lead and any other additive with no detectable detonation or 'knock'.

Power:

In the case of the British Leyland Mini, the BHP as measured using a chassis dynamometer, was increased by about 12% by increasing the exhaust to inlet valve ratio from standard to 1.39 without altering any other variables. A similar increase in power was observed in the other engines tested, in most cases it was possible to effect a further improvement in fuel consumption by using a weaker mixture and sacrificing the increased power.

Exhaust Emissions:

It was found that, in most cases, the modified engines showed reductions in two or more of the major exhaust emissions of carbon monoxide, hydrocarbons and oxides of nitrogen.

A 4.2 litre (252 cu.in) XJ6 Jaguar was tested by the makers before and after modification, the results for an ECE cycle were as follows:

	Standard (grams)	Modified (grams)
Carbon monoxide	35.4	35.6
Hydrocarbons	1.15	0.63
Oxides of nitrogen	2.26	1.04

In some cases, the amounts of all three major pollutants were reduced, for example, an ECE cycle carried out on a British Leyland Mini 1000 before and after modification by Ricardo & Co. Engineers (1927) Ltd., produced the following results:

	Standard (grams)	Modified (grams)
Carbon monoxide	40.4	33.4
Hydrocarbons	6.5	5.2
Oxides of nitrogen	5.5	4.1

As well as a reduction of exhaust emissions, it was found that after several thousand miles, the combustion chamber remained extremely clean. In the case of engines using leaded fuel there was a white deposit of lead oxide; when lead-free fuel was used little or no deposits were observed.

Conclusions:

The use of exhaust valves significantly larger than the inlet valves in an internal combustion engine has been shown, contrary to normally accepted teaching, to produce the following effects:

1. A reduction in fuel consumption.

2. A reduction of exhaust emissions.

3. No loss of power or performance.

4. The ability to run on a lower than normal octane fuel.

The mechanism is thought to depend upon an improved removal of combustion products from the combustion chamber leaving a lower concentration of residuals during the compression stroke and an improved intake of air-fuel mixture which in turn produces a better combustion efficiency and a lower level of exhaust emissions. It is also possible that the large exhaust valves introduce turbulence into the combustion chambers which influences the subsequent induction and combustion.

It is thought that too much emphasis has been placed to date on the design inlet systems and too little work has been done on optimising the design of the exhaust parameters.

This simple modification to the internal combustion engine offers motor manufacturers a relatively inexpensive way of producing more economical power units with the added advantages of reduced emissions and fuel octane requirement without introducing design complexity or altering the performance characteristics.

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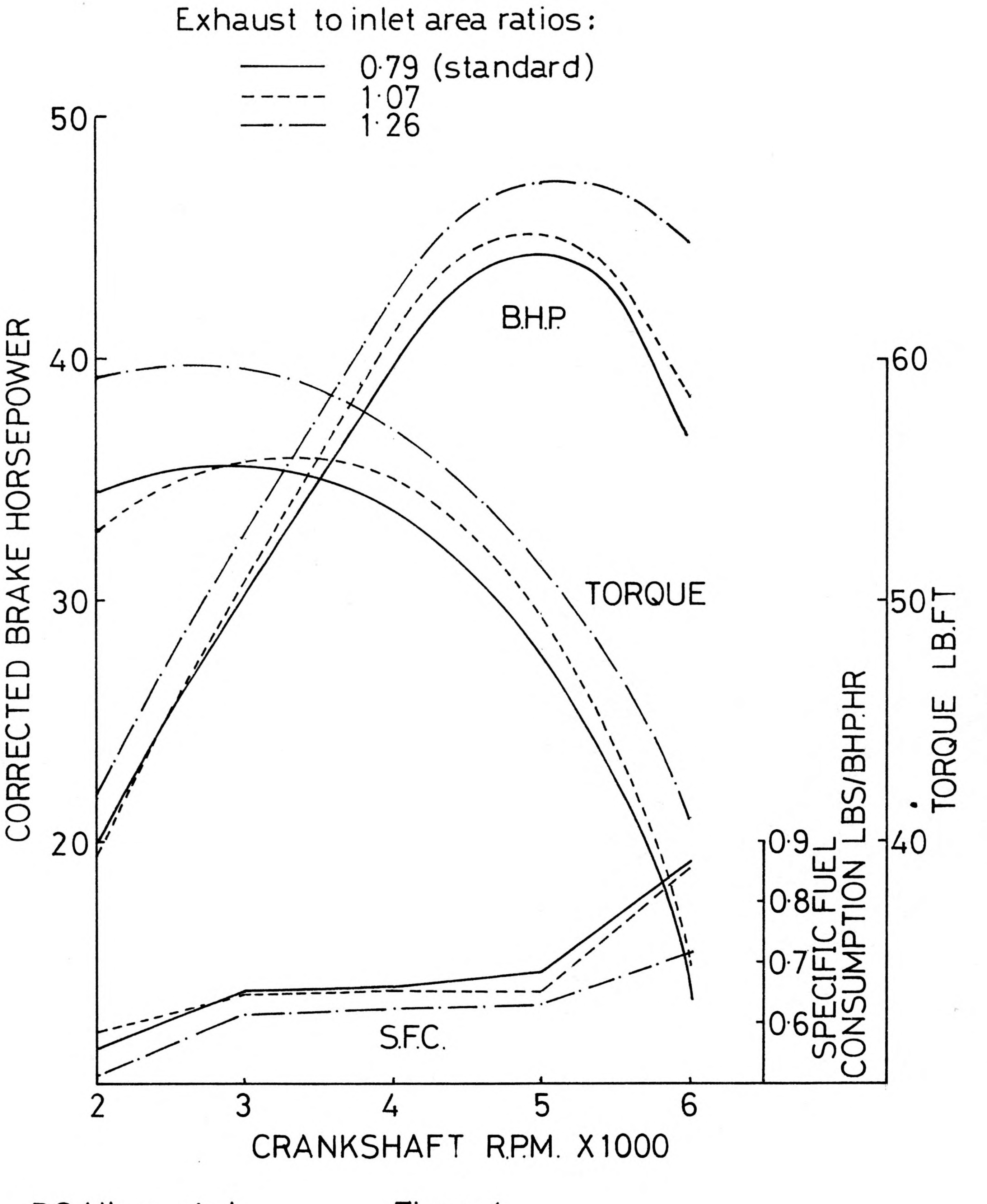
2. Barnes-Moss H.W. "A Designer's Viewpoint", Conference Publication 19, pp 133-147, Institute of Mechanical Engineers, London, 1973.



DSAllam et al

Figure 1

Title: <u>The Effect of Valve Sizes on Fuel Consumption and Exhaust Emissions in Gasoline Engines.</u>

Authors: D.S. Allam et al.

<u>Caption to Figure 1.</u>

Torque, BHP and fuel consumption curves at various speeds for a Ford 1300 c.c (79 cu.in) engine with exhaust to inlet valve area ratios of 0.79, 1.07 and 1.26.

Conversion factors:

BHP = 0.746 kW

lb.ft = 0.138 kg.m.

lb/BHP hr = 0.608 kg $(kW.h)^{-1}$

CLOSED LOOP CONTROL OF SPARK ADVANCE

by

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CLOSED LOOP CONTROL OF SPARK ADVANCE

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J.D. Powell‡

ABSTRACT

The objective of a closed loop spark advance control technique is to provide a schedule which is closer to optimum in the presence of changing engine and environmental conditions. This paper shows the benefits of closed loop spark control based on cylinder pressure information, and describes the ability of the system to adapt when the engine operating state is disturbed. The controller is evaluated at spark advances nominally set for best torque and at retarded conditions for exhaust emissions reduction. Furthermore, a feature of the controller that causes it to adaptively avoid detonation is described. An inexpensive pressure transducer has been fabricated and is shown to be suitable as a feedback sensor.

INTRODUCTION

One of the important inputs to a spark ignition engine which affects nearly all engine outputs is spark advance. It is well-known that the optimum spark advance (spark advance for maximum torque) for a given engine design varies with speed and percent throttle. Less well-known, but understandable based on the physics of combustion, is the fact that the optimum spark advance also varies with equivalence ratio, ϕ, (fuel-

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air ratio normalized by the stoichiometric fuel-air ratio), as well as with atmospheric humidity, fuel composition, and compression ratio.

In most present-day engines, spark advance is determined in an open loop manner by the distributor mechanism based on two measured variables: engine speed, and intake manifold vacuum. Because in most carburetion systems, equivalence ratio is determined by speed and percent throttle, a separate adjustment for equivalence ratio is not included; i.e., the optimum spark setting is the optimum for that speed and load, and the equivalence ratio which the carburetor nominally produces at that speed and load. Thus, errors in carburetors or other fuel metering schemes which cause the equivalence ratio to deviate from nominal also cause the optimum spark setting to change.

It is the purpose of this paper to present an approach to spark advance scheduling which accomplishes the functions of a conventional distributor (i.e., provides the required spark advance as a function of engine speed and load) while at the same time offering the advantage of insensitivity to variations in parameters which affect the value of optimum spark. The concept has been discussed earlier [Ref. 1]; this paper reviews the concept and presents current results from the research.

The sensitivity of MBT (minimum advance for best torque) spark advance to certain quantifiable effects is shown in Table 1. In actual

Table 1

OPTIMUM SPARK ADVANCE SENSITIVITY TO PARAMETER VARIATIONS

Parameter Variation	Change in Optimum Spark Advance (deg)
Speed increases 100 rpm	+ 0.5
Percent throttle decreases 10 percent	+ 2.5
Equivalence ratio decreases 10 percent	+ 4.0
Altitude increases 10,000 ft	+ 3 to 8
Daily barometric pressure fluctuates ± 2.5 percent	± 0.7
Intake humidity at 100°F and 90% relative humidity	+ 7

practice, these effects will be added to production tolerances, design compromises for manufacturing, and degradations with time due to engine aging and spark advance drift. It seems reasonable that spark advance errors of 10° or more may occur which is shown in Fig. 1 to represent a 6% loss in fuel economy.

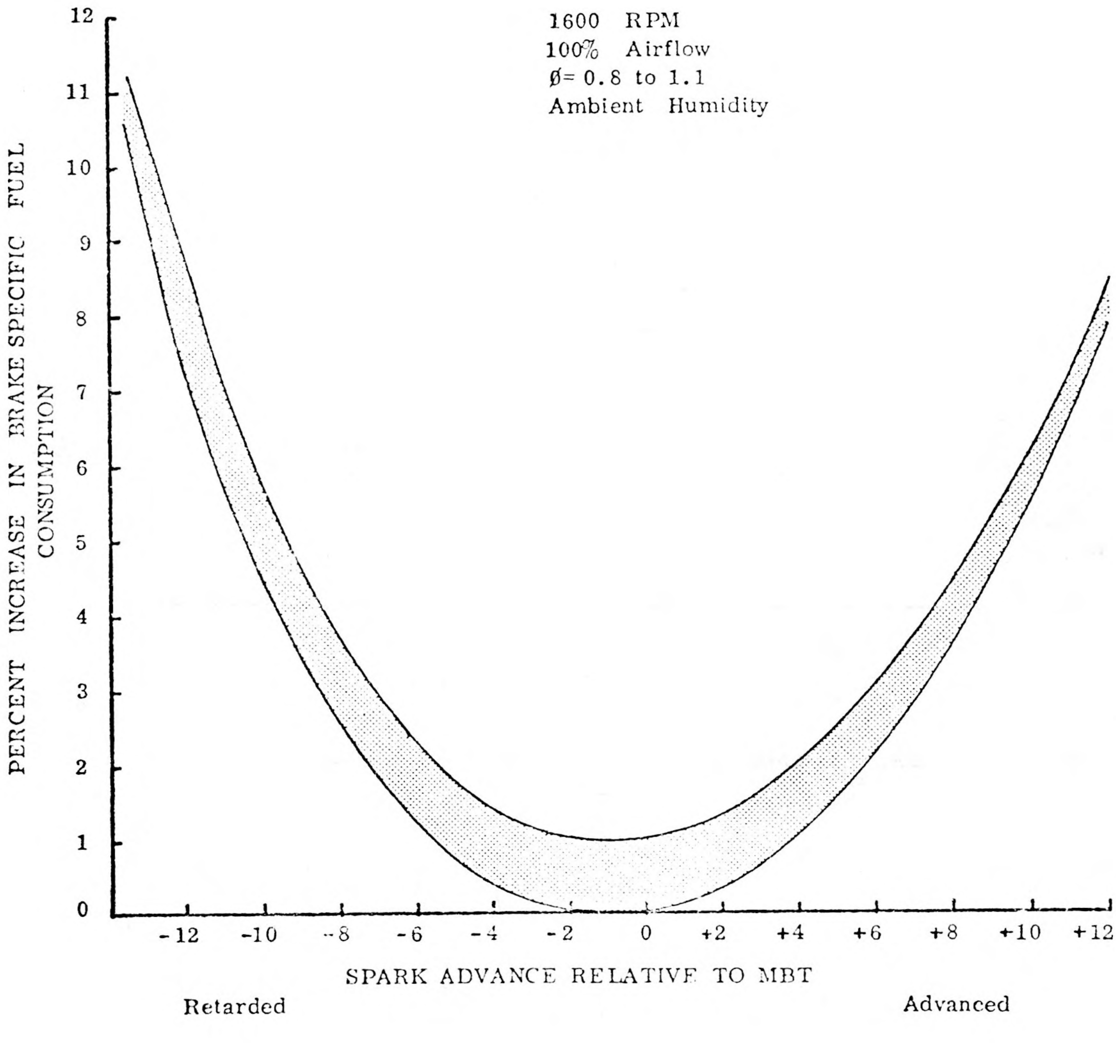

FIG. 1 LOSS IN ENGINE EFFICIENCY WITH OFF-MBT SPARK TIMING

A. Implementation

The basis for this closed loop spark advance control scheme is founded on the experimental fact that MBT timing results in a peak cylinder pressure position (in crankshaft degrees) that is uncorrelated with all engine variables, see Fig. 2. The data cover the range of engine* rpm from 600 to 1800, of equivalence ratio from 0.8 to 1.1, of throttle from 50%

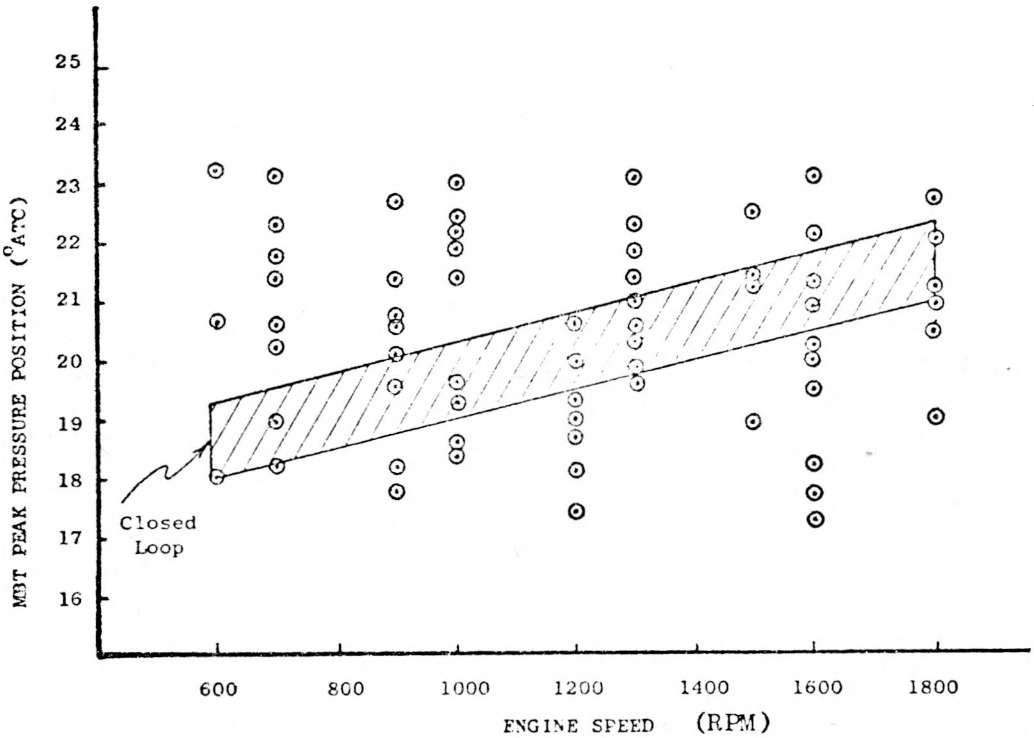

FIG. 2 PEAK CYLINDER PRESSURE POSITION AT MBT SPARK TIMING

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* The engine used for this study is a single cylinder Waukesha Cooperative Fuels Research (CFR) engine.

to 100%, and of humidity ratio from 10 to 30 g water vapor/kg dry air.

Therefore the Closed-Loop control law is to position the peak cylinder pressure at approximately 20° ATC (after top center) by adjusting the spark initiation point. The resulting spark advance is thus the MBT timing. An electronic spark advance controller which implemented this control law followed the shaded region in Fig. 2. The implementation is depicted schematically in Fig. 3 and is described in more detail in Ref. 1. The controller also lends itself to microprocessor implementations, which have been done and are described in detail in Ref. 2.

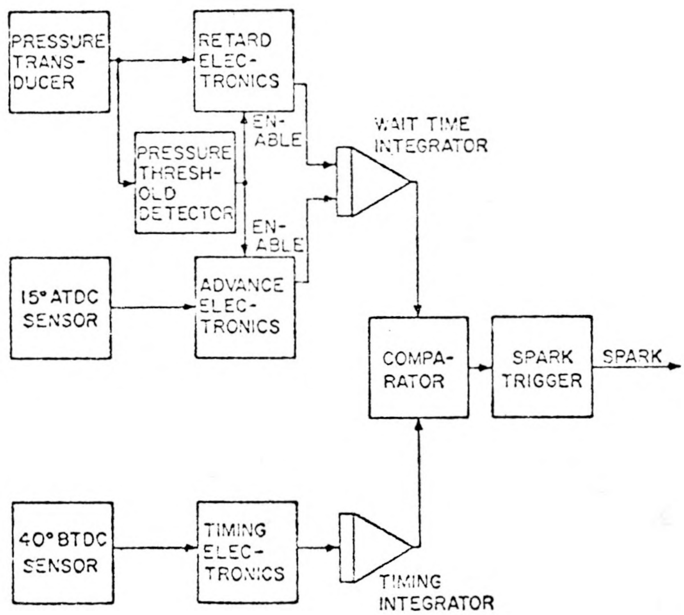

FIG. 3 LOGIC FLOW DIAGRAM FOR CLOSED LOOP SPARK ADVANCE CONTROLLER [from Ref. 1].

B. Steady State Behavior

Figure 4 shows that the closed loop controller in steady state provides the functions of a conventional distributor by advancing the spark as rpm increases or as airflow drops. In addition, however, the closed loop controller will track equivalence ratio excursion as demonstrated in Fig. 5, and humidity variations as shown in Fig. 6. A

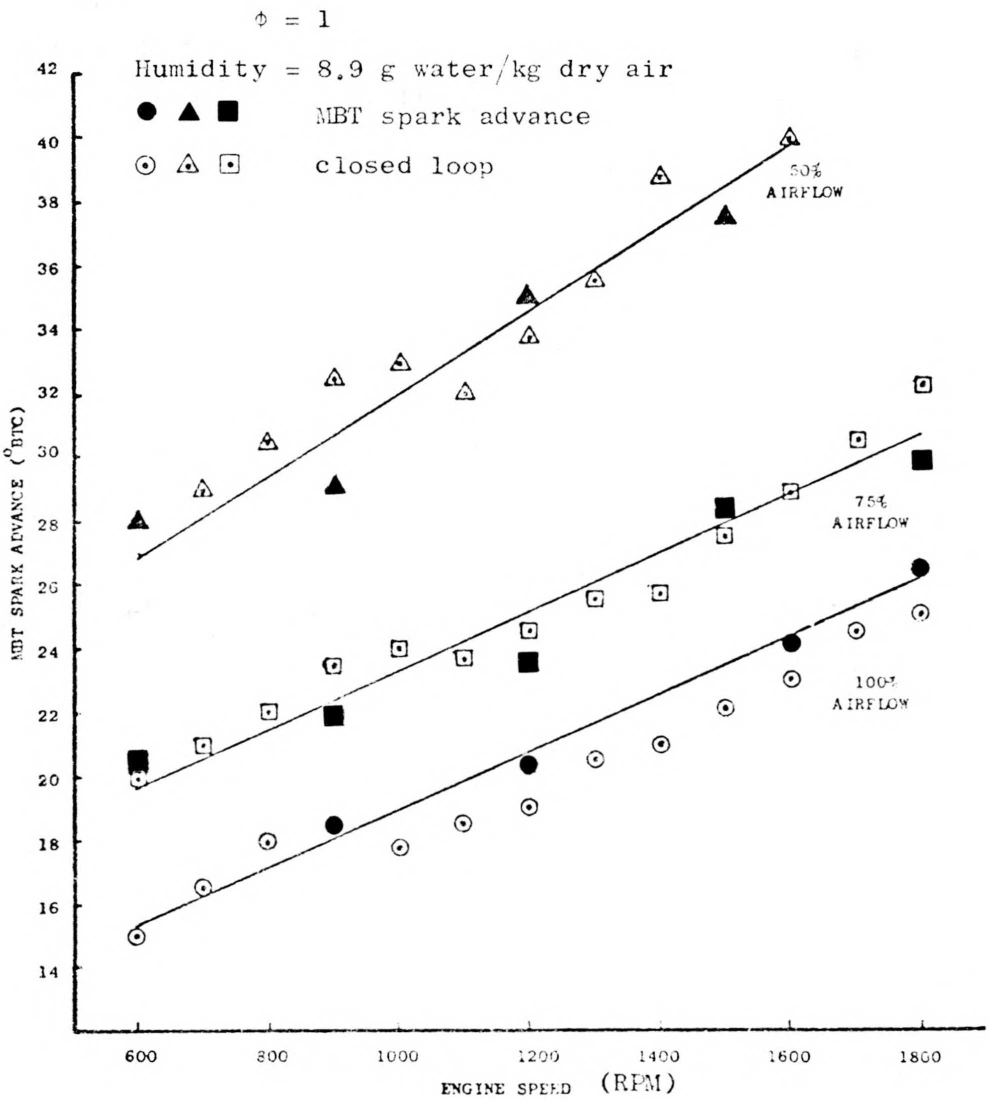

FIG. 4 PERFORMANCE OF CLOSED LOOP SPARK CONTROLLER–THROTTLE EFFECTS.

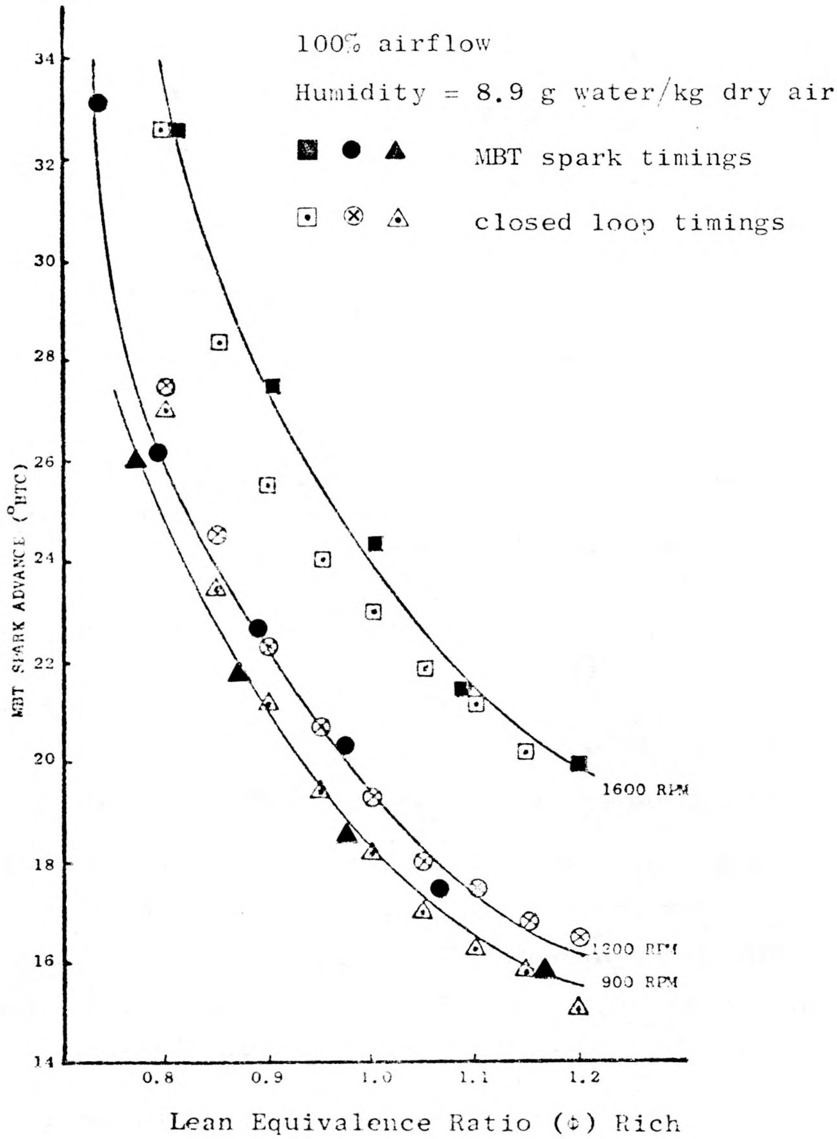

FIG. 5 PERFORMANCE OF CLOSED LOOP SPARK CONTROLLER-EQUIVALENCE RATIO EFFECTS.

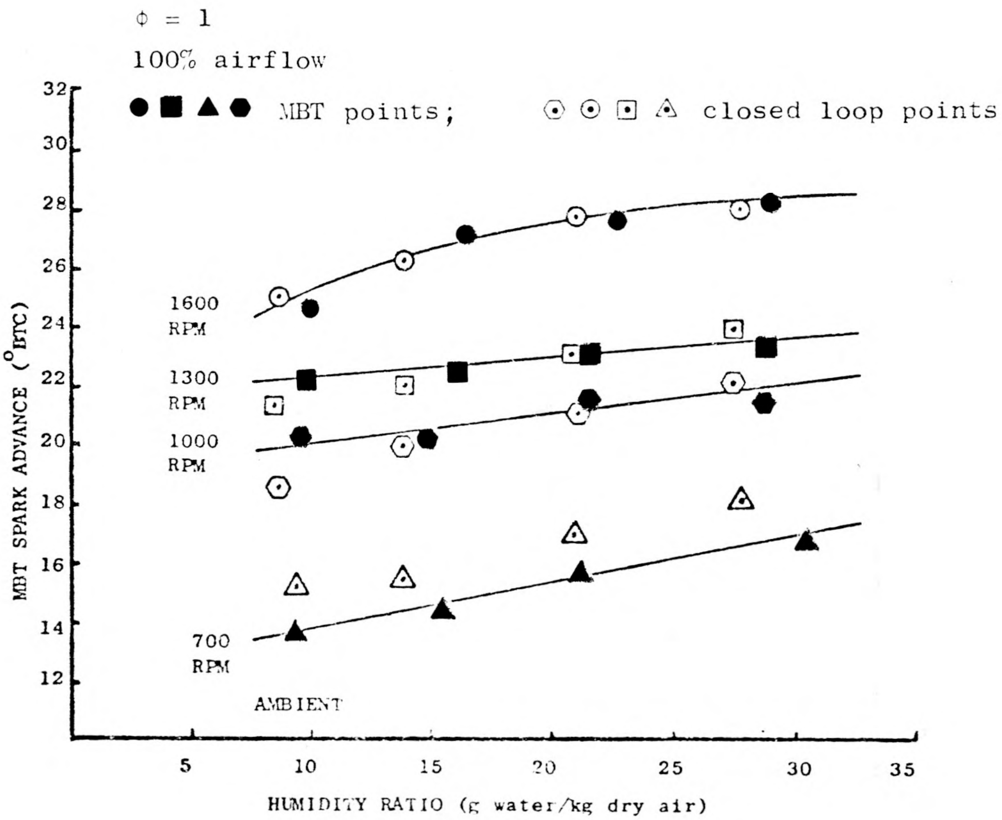

FIG. 6 PERFORMANCE OF CLOSED LOOP APARK CONTROLLER–HUMIDITY EFFECTS

conventional distributor is blind to these effects.

When designing engine controllers for automobiles with emission constraints, the spark timing is typically set at a position which is retarded from MBT timing by 10° or 20°. In the closed loop controller, this is implemented by retarding the "target" peak cylinder pressure position an amount approximately equal to the desired spark retard.

To establish the benefits of a closed loop controller under retarded timing conditions, a constant 10° of retard was programmed by setting the controlled peak pressure position at 30° ATC. Figures 7 through 10 show the results using intake air humidity as the disturbance to a closed loop system (constant peak pressure position), and an open loop system (constant spark advance set to ambient humidity MBT and MBT–10°).

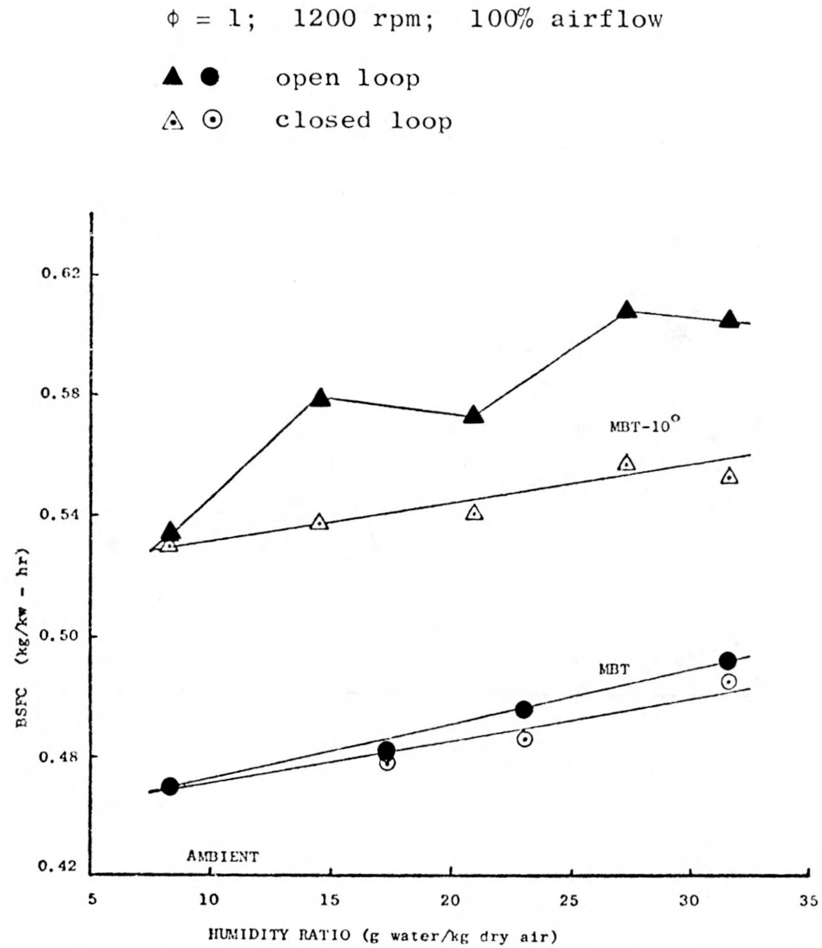

FIG. 7 SENSITIVITY OF BRAKE SPECIFIC FUEL CONSUMPTION TO HUMIDITY

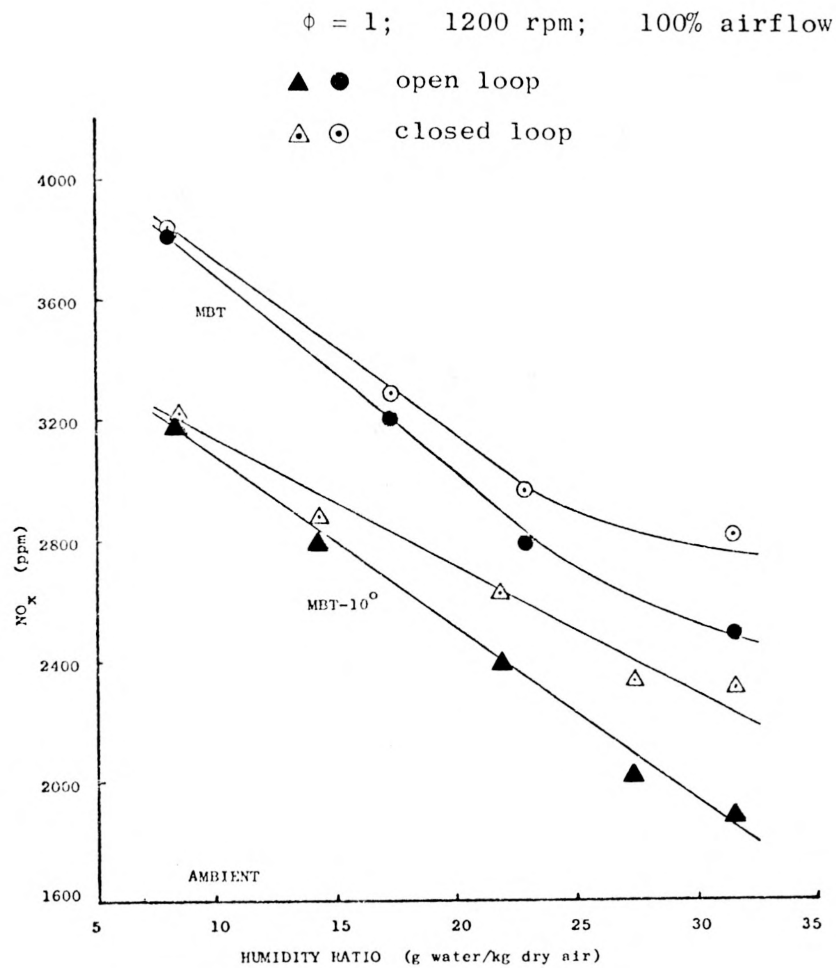

FIG. 8 SENSITIVE OF NITROGEN OXIDE EMISSIONS TO HUMIDITY.

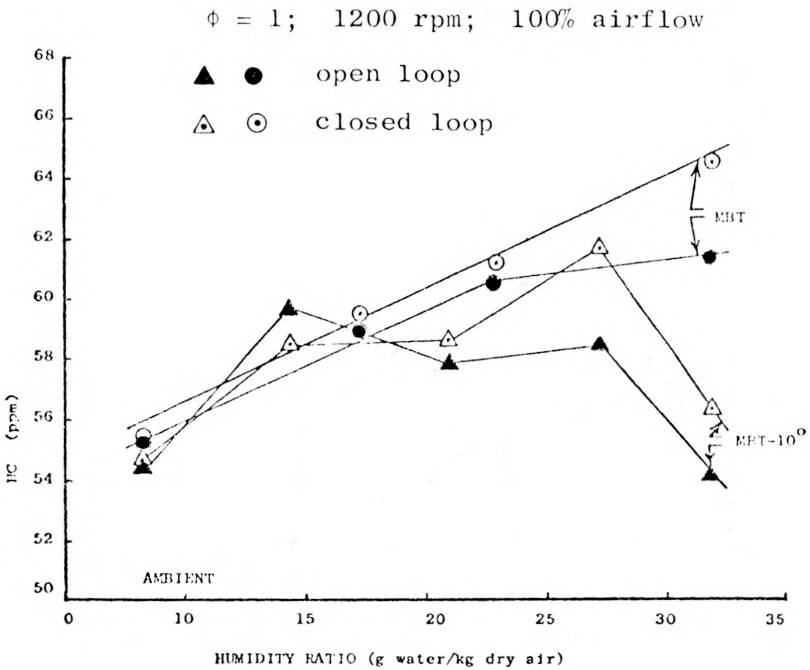

FIG. 9 SENSITIVITY OF HYDROCARBON EMISSIONS TO HUMIDITY

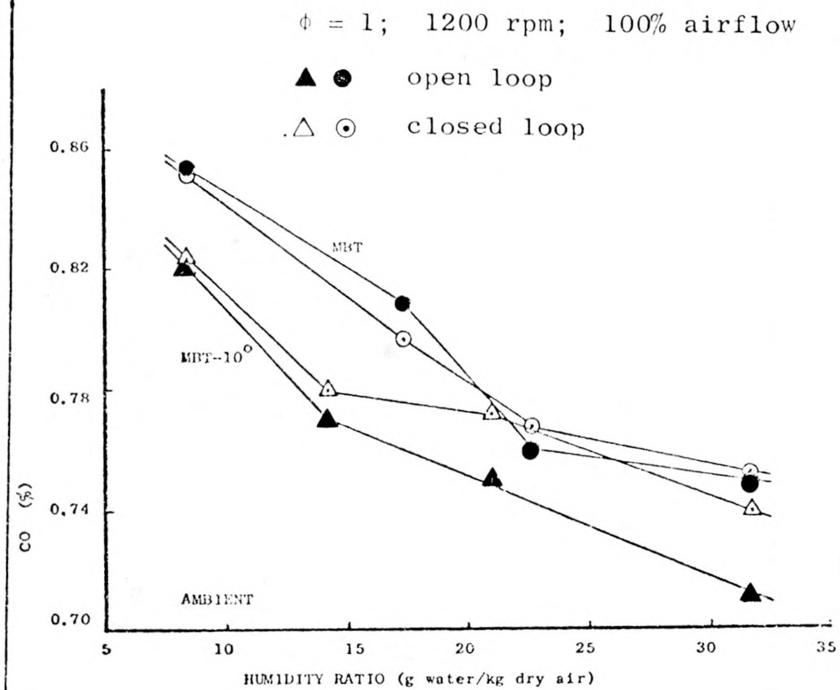

FIG. 10 SENSITIVITY OF CARBON MONOXIDE EMISSIONS TO HUMIDITY

Figure 7 shows that the closed loop feature improves BSFC with increasing humidity, especially at the retarded timing condition. This result is explained by Fig. 1 which showed the much greater sensitivity of BSFC to timing errors at the retarded condition. Figures 8, 9, and 10 show the effect on emissions. For NO_x and CO (Figs. 8 and 10), the level of emissions decrease with humidity; therefore the slight detrimental effect of the closed loop system is of no consequence. For HC, the level of emissions increase with humidity and the closed loop system appears to slightly aggravate the situation; however, the amount of degradation (about 6% at the highest humidity) in HC is judged to be significantly less important than the gain in efficiency (about 9% at the highest humidity).

C. Transient Behavior

Figure 11 shows the controller's transient response to an initial condition error. The particular gain used shows a time constant of about 0.5 sec. A greater gain results in a faster settling time ($\approx$ 0.1 sec) as Fig. 12 reveals. However, the controller in steady state is more sensitive to cycle-to-cycle variations in peak pressure due to combustion variations. This noise appears to be random, and no stability problems were encountered.

Figure 13 shows that the closed loop controller supplies the necessary spark timing response under a change in equivalence ratio transient.

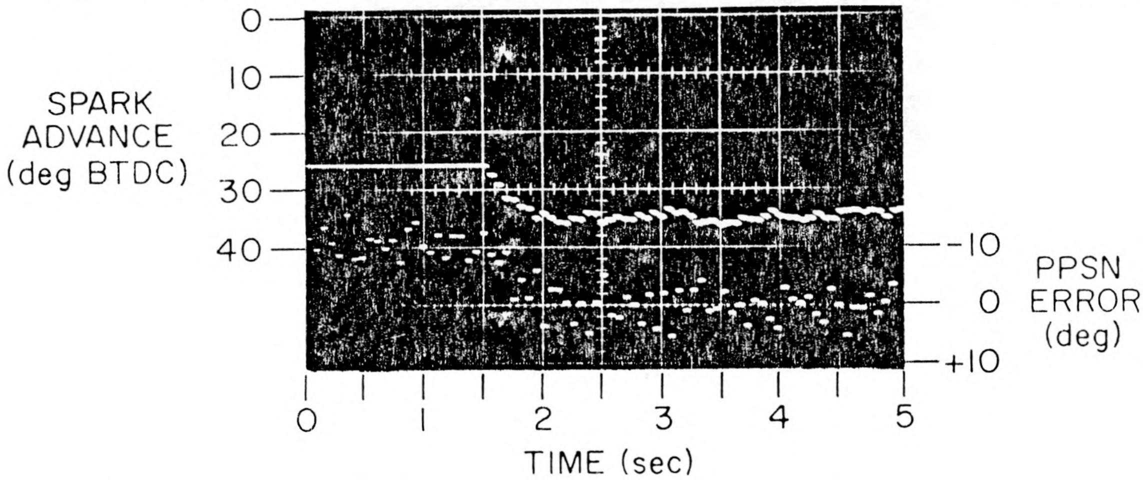

FIG. 11 CLOSED LOOP SPARK CONTROLLER RESPONSE TO INITIAL CONDITION ERROR AT WIDE OPEN THROTTLE, STOICHIOMETRIC EQUIVALENCE RATIO, 6.1 COMPRESSION RATIO (1800 rpm)

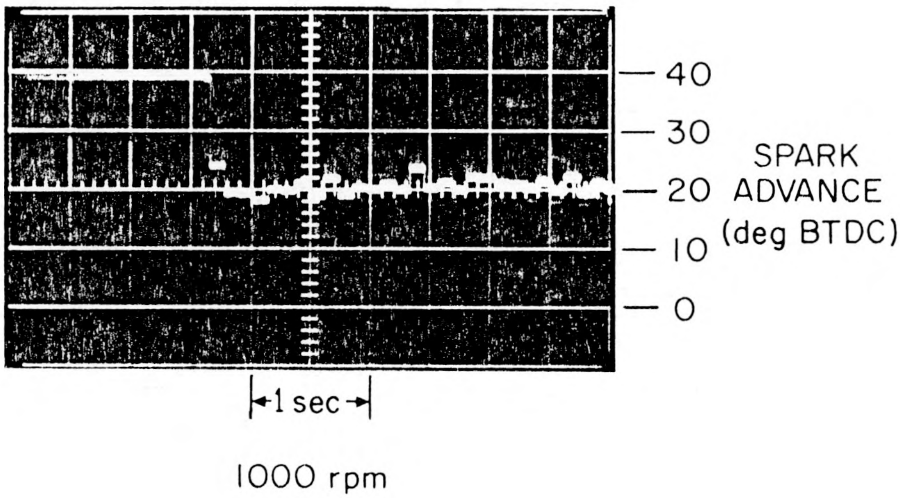

FIG. 12 FAST CONTROLLER RESPONSE TO INITIAL CONDITION ERROR AT WIDE OPEN THROTTLE, STOICHIOMETRIC EQUIVALENCE RATIO, 6.1 COMPRESSION RATIO.

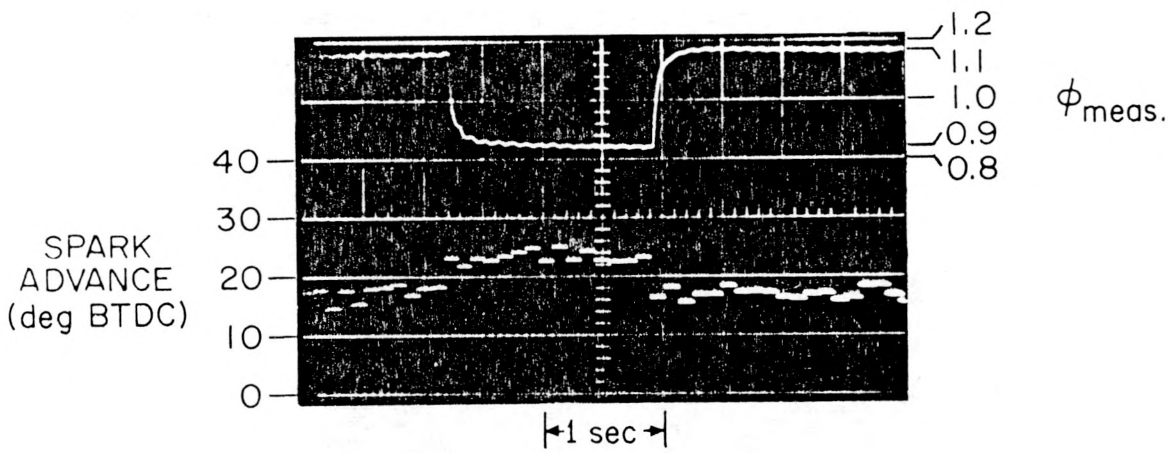

FIG. 13 CLOSED LOOP APARK CONTROLLER RESPONSE TO A CHANGE IN EQUIVALENCE RATIO AT WIDE OPEN THROTTLE, 1000 RPM, 6.1 COMPRESSION RATIO.

CLOSED LOOP TIMING FOR DETONATION CONTROL

The motivation for a spark advance controller which adapts the timing so as to always avoid detonation stems from the recent reductions in octane number in gasoline, and the desire to design engines to be as efficient as possible. The onset of detonation is dependent on many variables, for example, inlet temperature and engine deposits. It would therefore be difficult if not impossible to build an open loop controller which depended on a measurement of all these variables.

The closed loop approach bases its control solely on the measurement of incipient detonation. This measurement is based on the characteristic high frequency (~ 5000 Hz) component of the cylinder pressure signal. The cylinder pressure signal (from the same sensor used for the peak pressure determination) is passed through a filter with a passband around 5000 Hz, the amplitude of the resulting a-c signal is determined, and this amplitude is used as an indication of detonation.

Investigations on the best control algorithm to make use of this information are still in progress; however, the initial attempt consisted simply of retarding the spark 10° when detonation was encountered and slowly advancing the spark until the desired peak pressure position is reached if no further detonation is encountered. Figure 14 compares performance of the controller with and without the detonation retard feature.

PZT CYLINDER PRESSURE SENSOR

In order for the closed loop spark control strategy to be practical on a mass production basis, an inexpensive reliable pressure transducer must be available. Towards this end, a piezoelectric pressure transducer has been designed and fabricated based on the sensor of Kondo [3]. The transducer is a ring located beneath the spark plug, see Fig. 15, and detects strain variation related to the cylinder pressure. Figure 16 shows construction details which allow the sensor to survive continuous operating temperatures of 250°C.

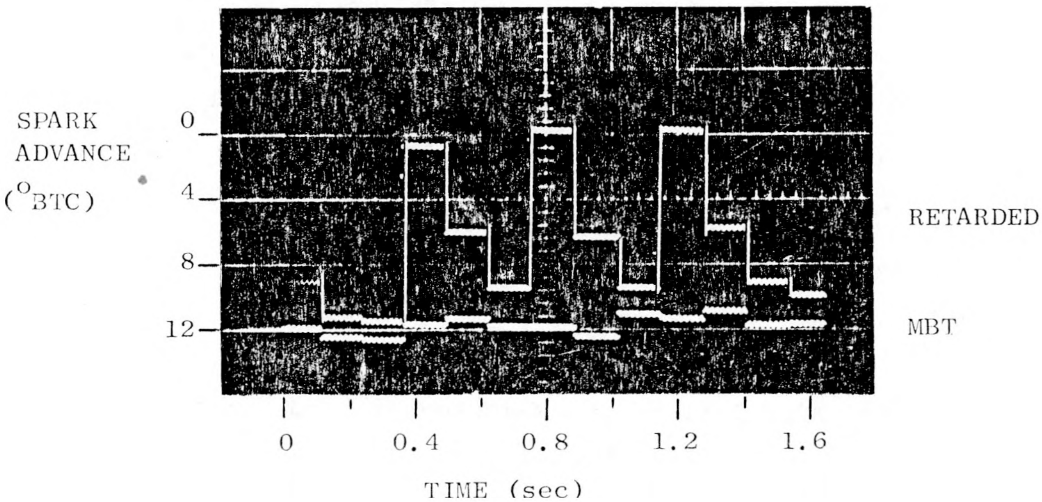

FIG. 14 STEADY STATE BEHAVIOR OF SPARK CONTROLLER WITH KNOCK RETARD MECHANIZATION.

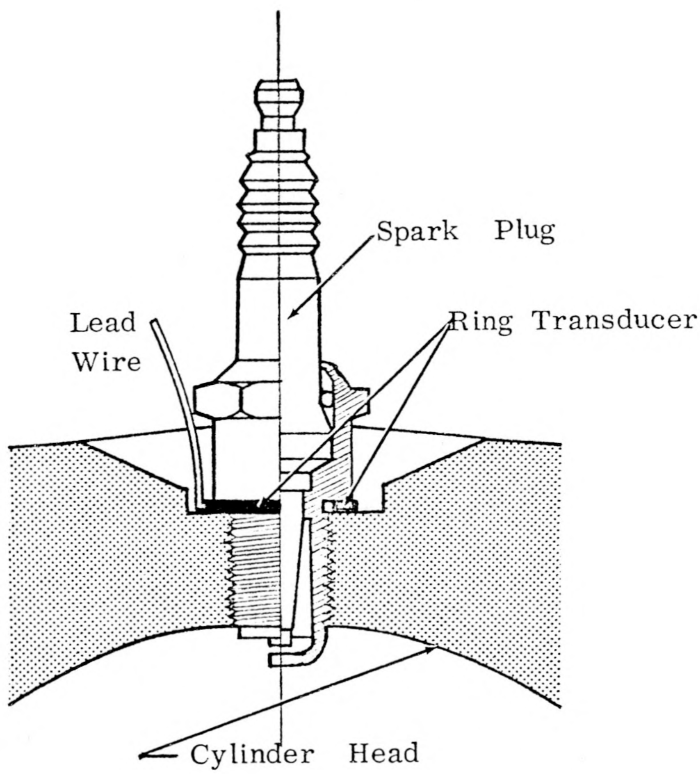

FIG. 15 PIEZOELECTRIC RING TRANSDUCER LOCATED BENEATH SPARK PLUG [from Ref. 3]

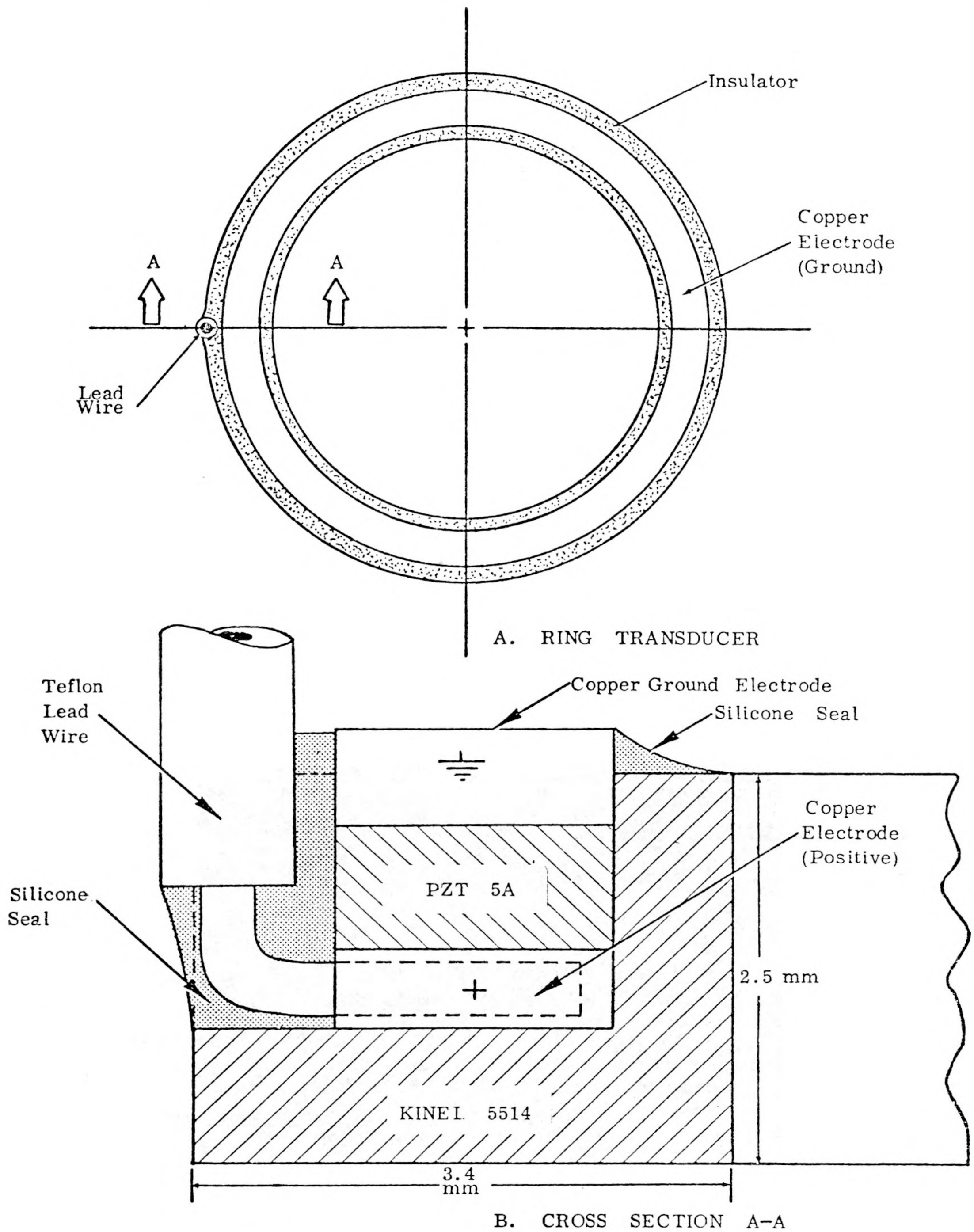

FIG. 16 PIEZOELECTRIC CYLINDER PRESSURE TRANSDUCER

Although no detailed cost estimate of the sensor in production quantities has been made, there are no expensive materials nor any precision machining involved; therefore we speculate that it will cost in the neighborhood of $1.00 if not less. It is important to note that there is absolutely no requirement for an accurate scale factor or null stability; but it must possess good bandwidth for detonation detection and the output should be roughly linear so that the cylinder pressure peak position is accurately determined.

Figure 17 shows the transducer output compared to that from a laboratory grade Kistler quartz piezoelectric pressure transducer. The signal from the piezoelectric ring transducer shows the location of a peak cylinder pressure clearly; shows no large peak pressure delay, and shows a bandwidth sufficiently wide to detect the characteristic knock frequencies. The transducer did display a large temperature sensitivity due to the degradation of the piezoelectric properties with temperature. However, this sensitivity did not compromise the spark control strategy.

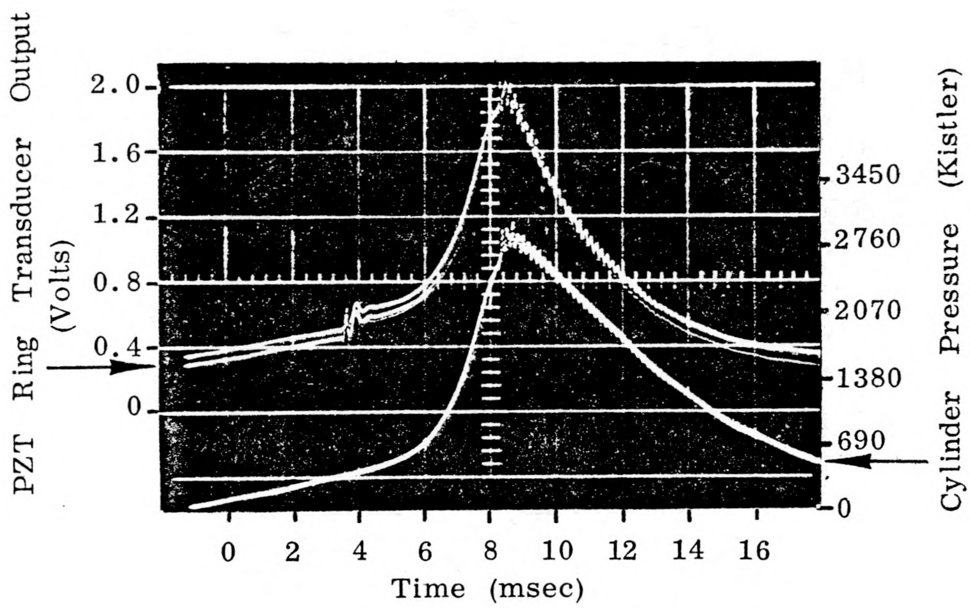

FIG. 17 COMPARISON OF PZT TRANSDUCER VOLTAGE OUTPUT TO CYLINDER PRESSURE (KISTLER TRANSDUCER). 1000 rpm, $\phi = 1$, WOT, CR = 6, SA = 16

CONCLUSIONS

A closed loop control technique for spark advance control has been presented. It has been shown to provide the necessary spark advance as a function of speed and load while simultaneously taking into account the changes in optimum spark advance caused by changes in humidity and equivalence ratio. This scheme was experimentally found to give performance within one percent of the performance at MBT spark.

The mechanization of the controller was described and its transient response characterized. Closed loop spark advance control was found to be advantageous both at MBT timing and a nominal retarded timing.

An adaptive knock control algorithm was presented which would allow the spark controller to be as advanced as warrented under the environmental and engine conditions, and would result in reduced brake specific fuel consumption.

An inexpensive piezoelectric pressure transducer was designed, fabricated, and shown to be satisfactory for the spark control schemes described in this research.

ACKNOWLEDGMENT

This research was supported by the Department of Transportation under Contract DOT-OS-30111, and DOT-OS-60151.

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Session 12

VEHICLE TEST AND EVALUATION PROGRAM

OF THE U. S. POSTAL SERVICE

BY

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U. S. POSTAL SERVICE

ABSTRACT

The U. S. Postal Service has a vested interest in engines and vehicle systems which, when used in the delivery and transportation of mail, will minimize the use of and dependence upon oil based fuels and will offer minimum life cycle costs.

In the light duty internal combustion engine area, diesels, stratified charge and hydrogen fueled engines, along with a gasoline/electric hybrid propulsion system were scheduled for engineering evaluation. Test results on the diesel and stratified charge engines show a significant increase in fuel economy over the presently used gasoline engine. Tests on the gasoline engine/electric hybrid system indicate the need for major component design before achieving operational data. Analysis and laboratory tests conducted thus far on the hydrogen fueled engine indicate potential for higher efficiency. Comparable small, low performance gasoline engines will be evaluated.

Vehicle Test and Evaluation Program
of the U. S. Postal Service

T. A. Norman
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The U. S. Postal Service has a program in which we look at improved vehicles for use in its fleet in order to obtain reduced vehicle life cycle costs, as well as, to reduce the use of and dependence on oil based fuels. As part of this program, we are testing and evaluating a number of internal combustion engines and hybrid propulsion systems. The purpose of this presentation is to describe the internal combustion engines and the hybrid propulsion system under evaluation and the test results that have been obtained thus far. Before discussing the program, it may be helpful to provide to you an appreciation for the magnitude of the Postal Service fleet and for the operational mission of the vehicles.

VEHICLE FLEET AND MISSION

The Postal Service operates one of the largest vehicle fleets in the country. It owns approximately 122,000 vehicles and leases an additional 22,000. In this fleet there are approximately 8,000 1 ton (1016 kg) vehicles, 26,000 1/2 ton (508 kg) vehicles and 75,000 1/4 ton (254 kg) vehicles. Additionally, there are 2 1/2 (2540 kg) and 5 ton (5080 kg) vehicles, as well as, tractors and trailers.

During FY 76, the fleet traveled approximately 690,000 miles, (1,110,900,000 km) used 88,000,000 gallons (33,000,000 lit) of fuel at a cost of $43,000,000.

As you will note, the majority of the vehicles are in the 1/4 (254 kg) to 1 ton (1016 kg) class. The 1 ton (1016 kg) vehicles are used primarily for collection of mail from the corner mail-boxes. The 1/2 (508 kg) and 1/4 ton (254 kg) vehicles are used for delivery of mail to individual customers.

There are basically two types of delivery routes. First, the park and loop route which consists of driving the vehicle to pre-established locations from which the mail is delivered, on foot, to the individual customers. Secondly, the mounted delivery route. This type of route consists of driving the vehicle to each customer's house or office to deliver the mail.

The characteristics of the many delivery routes vary widely, and, as such, it would be difficult to define a typical route. However, we can, in general, define the range of these characteristics. The distances of the routes vary from seven to 60 miles (11.27 to 96.6 kn). The number of stops may range from 50 to 600. It is important to note that the vehicle engine may idle up to 60% of the time during a delivery route cycle of up to six hours. The routes may be level or very hilly, and the temperature may range from very hot to very cold (50 -30°C).

Because of the variation of these routes, the vehicles performance may vary from four to 13 miles per gallon (1.7 to 5.5 km per lit). The average, fleet wide, is approximately 7.84 miles per gallon (3.33 km per lit).

ENGINE TEST AND EVALUATION PROCEDURES

We are presently in the process of testing and evaluating light duty diesel engines, stratified charge engines, hydrogen fueled engines and a gasoline-electric hybrid system. Each of the above engines or propulsion systems will be installed in a 1/4 ton delivery vehicle. Vehicle performance tests such as acceleration, speed and gradeability will be performed. A simulated delivery route test will then be conducted. The test route established is not necessarily intended to represent an average route. Instead, it represents, in general, the type of stopping and starting that is encountered on delivery routes. It is also designed for ease of duplication. The test route consists of 450 stops, approximately 150 ft (45.75 m) apart, over a 32 mile (51.52 km) distance. The actual route is laid out on a closed loop track with grades up to 4%.

Based on the results of the vehicle performance and the simulated route tests, decisions are made to conduct operational tests. Usually two test vehicles will be assigned to a selected location. Each of these vehicles, along with a control gasoline engine vehicle is to be operated, in turn, on selected delivery routes over a designated period of time. The results of the test vehicles are compared directly with the control gasoline engine vehicle and evaluations are made of its potential performance on postal delivery routes.

ENGINES AND PROPULSION SYSTEMS

Figure 1 shows a comparison of the diesel, stratified charge and hydrogen fueled engines with the gasoline used in the existing 1/4 ton (254 kg) delivery vehicles. Figure 2 shows a general schematic of the gasoline-electric hybrid propulsion system. A small 25 h.p. (18.65 kw) engine which operates continuously between 2500 to 3000 rpm, drives an electric motor/generator and a hydraulic pump/motor. Speed and direction of the vehicle are controlled by a tilt plate in the hydraulic pump. At low loads and during idle, the motor/generator acts as a generator and charges the battery. At high loads such as during acceleration and hill climbing, the motor/generator with energy from the battery works as a motor and assists the gasoline engine in driving the vehicle. All components were basically off-the-shelf, not specifically designed for this particular application. In theory, the system should be quite efficient since the engine is very small and operates in its most efficient rpm range. Additionally, excess energy developed by the gasoline engine during idling and light loads can be stored in the battery and used when required.

STATUS OF DEVELOPMENT AND TESTING

The light duty diesel engine has been installed in four 1/4 ton vehicles. Vehicle performance, simulated route and seven months of operational tests have been completed. Two each of these vehicles are operating in Merrifield, VA and Salt Lake City, UT. Salt Lake City was selected as one of the locations to determine if high altitudes had any significant affect on diesel engine performance.

The 183 cid (1299 ccd) stratified charge engine has been installed in two vehicles. Two additional 163 cid (2651 ccd) engines are scheduled for future installation. The stratified charge engine can use either diesel, kerosene or gasoline as fuel. The simulated route tests, using diesel fuel, have been completed. The vehicle performance tests are presently being conducted.

A hybrid propulsion system was installed in one vehicle. Vehicle performance and simulated route tests were completed.

The hydrogen engine is still under development and testing has not been scheduled.

VEHICLE PERFORMANCE TEST RESULTS

Figure 3 shows the results of the vehicle performance tests. As can be seen, the presently used gasoline out performs the other engines. This is probably because it is the larger engine, 3802 ccd. However, the performance of the diesel is considered suitable for postal delivery operation. The hybrid system was quite unsatisfactory, therefore, no operational testing was performed. Tests on the stratified charge engine are not complete. The hydrogen engine is still under development and not yet available for test.

FUEL CONSUMPTION RESULTS

Figure 4 shows the results of fuel consumption for these engines during idle, simulated route testing and under operational conditions. Both the diesel and the stratified charge engines, when operating on diesel fuel, appear to consume about the same amount of fuel. The gasoline engine consumes considerably more fuel than the diesel or stratified engine.

The hybrid system consumed the greatest amount of fuel. The fuel consumption for the diesel and gasoline engines under operational conditions are average figures for different types of delivery routes.

Figure 5 shows the miles per gallon (mk per lit) for the diesel and gasoline engines for the park and loop, curbside, and special delivery services. The range in miles for each of these routes is also shown. It should be noted that special delivery service routes were included in the average miles per gallon (km per lit) numbers shown in Figure 4. However, the number of of special delivery routes compared to the park and loop curbside delivery routes are very small.

CONCLUSION

The diesel engine obtained about 122% more miles per gallon (km per lit) during operational use than the gasoline engine under the same conditions. It is considered suitable for postal delivery mission. However, additional information on maintenance and life cycle cost must be obtained before its full potential for operational use can be established. The gasoline engine tested, however, was a larger engine than the diesel engine. Test of a comparable low performance gasoline engine is planned. Data from this test will result in a more appropriate comparison.

The stratified charge engine, based on preliminary test data, appears to exhibit the same level of fuel efficiency when using diesel fuel as the diesel engine. From the performance standpoint, it should be suitable for postal delivery missions. Cost and maintenance problems have yet to be determined.

The hybrid propulsion system consumed more fuel than the gasoline engine. It demonstrated limited operational performance with excessive engine and transmission noise. The design and the components used were not considered suitable for operational testing.

The hydrogen fueled engine has the potential of high efficiency at low loads. Oil based fuel is not required and very little exhaust pollution is produced. Necessary modifications to the gasoline engine to operate on hydrogen fuel is still under development. Test results are not available.

ENGINE DESCRIPTION

	DIESEL	STRATIFIED CHARGE	HYDROGEN	GASOLINE
C.I.D.	165	183	232	232
(CCD)	(2671)	(2999)	(3802)	(3802)
H.P.	70	85		100
Cylinders	4	4	6	6

Figure 1

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IC/ELECTRIC HYBRID

DESCRIPTION

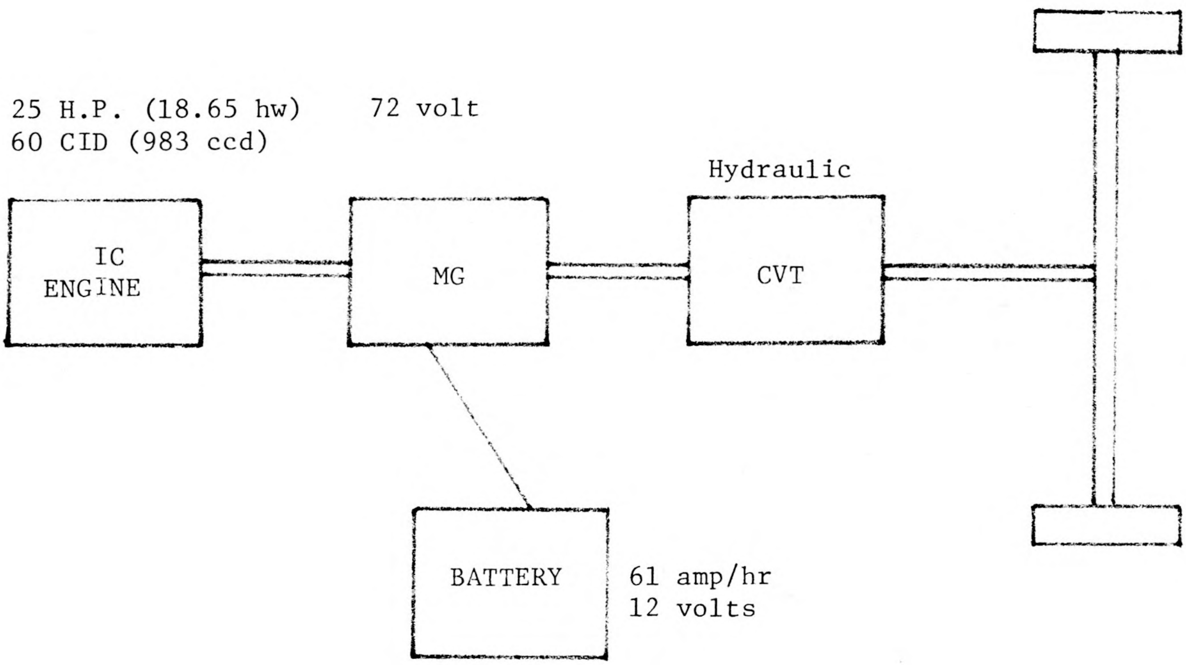

Figure 2

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VEHICLE PERFORMANCE

	ACCELERATION 0-20 (sec)	MAX GRADE (%)	MAX SPEED (MPH)	(kmph)
Gasoline	3.4	58	80	(128.8)
Diesel	5.3	36	68	(1114.5)
Stratified Charge				
Gasoline	5.2	42	67	(1098.1)
Diesel	-	42	67	(1098.1)
Hybrid	9	11	47	(770.3)
Hydrogen	-	-	-	

Figure 3

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VEHICLE FUEL CONSUMPTION

	IDLE	SIMULATED Route	Operational
	(GPH) (lit ph)	(MPG) (km p lit)	(MPG) (km p lit)
Gasoline	0.498 (1.885)	9.47 (4.03)	8.33 (3.54)
Diesel	0.270 (1.022)	15.50 (6.59)	18.48 (7.85)
Stratified Charge			
Gasoline	0.432 (1.635)	-	-
Diesel	0.263 (0.995)	15.50 (6.59)	-
Hybrid	1.40 (5.299)	6.09 (2.59)	-
Hydrogen	-	-	-

Figure 4

VEHICLE FUEL CONSUMPTION

(MPG) (km p. lit)

	Gasoline	Diesel	% Difference
Park and Loop	6.86 (2.92)	18.05 (7.67)	163
Curbside	7.49 (3.18)	17.16 (7.29)	129
Special Delivery	15.00 (6.38)	24.39 (10.37)	63

DELIVERY ROUTE RANGE

(Miles) (km)

Park and Loop	4 to 20	(6.4 to 32.2)
Curbside	4 to 38	(6.4 to 61.2)
Special Delivery	31 to 124	(49.9 to 125.6)

Figure 5

T. A. Norman

THE DEVELOPMENT OF CATALYSTS FOR EXHAUST EMISSION CONTROL

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Abstract

This paper discusses the development of noble metal monolithic catalysts for the control of exhaust emissions from gasoline fueled internal combustion engines.

The first section reviews the catalysts developed for use in model years 1975-77 in the US market. The way in which catalysts have helped to produce better fuel economy to be achieved is stated and discussed.

The second section looks to new catalysts currently being evaluated by the automobile industry.

1. The use of lower metal loadings which when coupled with improved catalyst formulations gives cost benefits. The possibilities of operating catalysts with higher lead and phospherus levels in fuels than that currently used in the USA is briefly considered.
2. The development of a new generation of catalysts based on a metal rather than the usual ceramic supports. The selection of an alloy with the necessary high temperature corrosion resistance is discussed. The metal supported catalyst can be more easily engineered into a vehicle because of its superior mechanical and thermal strength and through the application of a suitable catalyst coating it is shown to have at least the activity and life of the ceramic supported catalyst.

INTRODUCTION

Noble metal catalysts for the control of exhaust emissions are now an established feature on cars currently sold in the USA and Japan, and it is expected that catalysts will continue to be used for that purpose for the foreseable future.

This paper reviews the contribution made by catalysts so far and illustrates ways in which catalyst developments currently taking place will enable catalysts to be more effective in the future. Such improved catalysts can play a role in controlling exhaust emissions to meet future emission legislation in the USA, Japan and in other territories, such as Europe and Australia.

BACKGROUND

Catalysts have been fitted onto many of the cars sold in the USA since the start of model year 1975 in order that those cars might meet the more severe emission standards that came into force under the Clean Air Act. Full introduction of the standards originally set for model year 1975 was delayed following applications by some manufacturers for a suspension of those standards. The result of these applications, following public hearings in March 1973, was that for 1975 catalysts were needed only on vehicles sold in California, although many manufacturers elected to fit them on all vehicles sold in the USA. The catalysts fitted onto cars during the model years 1975-77 have been two distict types. Both types use the noble metals of the Platinum Group as the catalytically active component, but they differ in the form of the support onto which they are deposited.

Type 1 - Comprises of noble metals deposited onto pellets or granules of alumina; catalysts of these types were used by companies including General Motors, American Motors, Fiat and Alfa Romeo.

Type 2 - Comprises of noble metals deposited onto a ceramic monolith support, being a single piece of ceramic with a multitude of gas passages or cells extending longitudinally through the piece. The ceramic is generally a composition containing

Type 2 contd. - cordierite ($2MgO-2Al_2O_3-5SiO_2$) as the major phase (1,2). Before deposition of the noble metal active component the surface area of the ceramic is increased by coating with a thin washcoat of high surface area alumina.

The noble metals used in both the above catalyst types include Platinum, Palladium and Rhodium. Typical formulations are binary alloys such as Palladium/Platinum (1:2 by weight) and Rhodium/Platinum (1:9 by weight), and Platinum alone.

The Johnson Matthey Group of Companies is a major supplier of noble metal automotive exhaust emission control catalysts through its operating Companies Matthey Bishop Inc. of Malvern, Pennsylvania, USA and Johnson Matthey Chemicals Limited. Both Companies manufacture the monolithic type of catalyst (type 2) and supply Ford Motor Company (Matthey Bishop Inc.) and Volkswagenwerk AG, Leyland Cars (Jaguar, Triumph, and MG), Rolls Royce Motors Limited and Aston-Martin-Lagonda Limited (Johnson Matthey Chemicals Limited).

CATALYSTS AND FUEL ECONOMY

Although automobile manufacturers voiced many reservations during the period 1971-75 on the widespread adoption of catalysts for emission control, their introduction has caused few problems. It is perhaps indicative of the success of catalysts that many manufacturers have fitted catalysts to cars destined for sale in the 49 states (not including Calfornia) during the period 1975-77 rather than rely on engine controls such as lean carburrettion or manifold air oxidation.

One of the reasons for the widespread use of catalysts is that fuel economy is now a major issue, and manufacturers have been able to calibrate their engines for fuel economy rather than low emissions and relying on the catalyst for exhaust clean-up.

The leading US auto manufacturer reported (3) an overall gain of 28 percent in fuel economy with its 1975 models and 38 percent in its 1976 models, compared with 1974. The catalytic convertor was described by the auto company as "a key factor in achieving improvements".

Fuel economy trends across the US auto industry are summarised in Figure 1 (4,5).

Future legislation in the USA calls for more strict controls on exhaust emissions. There have been concerns that these stricter controls may impose a fuel penalty. This need not be so and catalysts will help in achieving this goal in the most economical way. Eric Stork, EPA Administrator, recently stated (6) that "EPA's estimate was that it is possible to meet either level II (0.41g/mile HC, 3.4g/mile CO and 2.0g/mile NOx) or level III (0.41g/mile HC, 3.4g/mile CO and 1.0 g/mile NOx) emission standards with negligible, if any, fuel economy penalty".

DEVELOPMENT OF IMPROVED CATALYSTS

During the period 1975-77 catalyst improvements have been made and introduced through intensive catalyst development work. This is necessary to meet the auto industry's needs for catalysts that:

1. meet the more stringent exhaust emission limits in California from model year 1977,
2. are more cost effective.

Some of the developments made by the Johnson Matthey Group have been used in the next section to illustrate how the catalyst industry has responded to this need. One important area of development, that of 3-way catalysts, has not been covered having been recently reported elsewhere. (7).

LOW METAL LOADING CATALYSTS

The catalysts, based on monolithic supports that were produced for 1975 model year cars generally had noble metal contents equivalent to between 40 and 50 g/ft^{-3} of monolith. This level of noble metal enabled the auto companies to obtain acceptable deterioration factors in 50,000 mile certification trials. The subsequent further reduction of lead levels in commercial gasoline has led to several automobile manufacturers achieving deterioration factors close to 1.0 on certification vehicles. In addition during the period 1975-76 most catalyst manufacturers worked towards producing modified or improved oxidation catalysts.

These two factors should enable automobile manufacturers to use more cost-effective catalysts provided that this allows them sufficient confidence that the deterioration factors will be adequate to meet the required limits with their particular vehicles.

The Johnson Matthey Group carried out optimisation studies during 1974-75 using a platinum oxidation catalyst of type 12C. A catalyst

In order to increase the savings that can be made, improved oxidation catalysts have been developed and introduced. These improved catalysts are capable of meeting more severe standards that may be imposed in the future.

These improved systems are designated by the letter G. A platinum based catalyst corresponding to 12C, type 20G, has been developed by the Johnson Matthey Group, and this shows improved catalytic performance when compared to the type 12C system under the test conditions described above. Data for the two systems after 300 hours of aging are compared in Figure 5 (CO) and Figure 6 (HC). From this comparison it can be seen that while the improvement is small at high metal loadings, an important difference is apparent when the metal loading is decreased. Thus for CO oxidation the sharp decline in performance at 30g Pt. ft^{-3} that is observed with type 12C is not observed with type 20G even at 25g Pt. ft^{-3}, and the CVS result for CO at this level is almost identical to that for the type 12C catalyst at 60g Pt. ft^{-3}. Correspondingly, for HC oxidation the steady decline in activity as metal loading is decreased that was observed with type 12C is not seen with type 20G catalyst, and again the activity at 25g Pt. ft^{-3} type 20G is at least as good as for 60g Pt. ft^{-3}type 12C.

Using the same basic method of assessment, broadly similar results have been obtained with platinum/palladium and platinum/rhodium oxidation systems. Figure 7 shows the results for a 92.5% Pt. - 7.5% Rh. catalyst, type 23G, with metal loadings in the range 15-40g PM ft^{-3}. It should be noted that the difference in performance of the lowest and highest metal loadings after 300 hours of aging is only marginal for both HC and CO

The series of experiments described above are comparable, having all been obtained using the same system of aging and testing, and shows that lower metal loading catalysts can be used to achieve current emission targets. In practice, it is necessary to consider how relevant these procedures are to the way the catalyst is fitted to the automobile, for example other considerations may dictate the catalyst location in the vehicle and this, together with the baseline emissions from the vehicle in question, has a major influence on the thermal aging of the catalyst. When investigating precious metal loadings below 25g ft^{-3} the aging cycle selected has an important effect on catalyst activity. Table 1 illustrates of this type was fitted to a Ford Capri car and achieved 50,000 miles within the statutory limits of 0.41g/mile HC and 3.4g/mile CO.

For the optimisation studies now reported, the catalyst was applied to a cylindrical Corning M20 monolith, 200 cells in.$^{-2}$, with a volume of 75.4 in^{-3} and a diameter of 4 in. The catalysts were aged on a 1.8 litre Triumph Dolomite static engine, on a cycle with 1 minute at idle followed by 1 minute at 3600 rev/min (equivalent to 70 mph). This procedure gives inlet temperatures in the range 550-800°C and CO contents ranging from 1 to 4 per cent. The fuel used contained 0.03 - 0.04g lead and 0.005 g phosphorus per US gallon, and 0.03 per cent sulphur. After the aging treatment the activity of the catalysts was assessed by running CVS tests on a 1.5 litre Avenger car.

The results show that with high (60g/ft^{-3}) and medium (40g/ft^{-3}) metal loadings the residual CO levels after the catalyst fell steadily with time of aging from zero to 300 hours. Nevertheless even at 300 hours CO conversions are still in excess of 90 per cent. This is in contrast to the low metal loading (30g/ft^{-3}) catalyst in this series, which shows a marked deterioration in CO activity after 300 hours, giving only 72 per cent conversion. The activity of this series of catalysts for hydrocarbon conversion at zero hours shows no dependence on metal loading in the range studied, but as the period of aging increases so the metal loading of the catalyst affects the observed performance. The results of this set of experiments are summarised in Figures 2 and 3.

Another way in which automobile manufacturers can achieve cost benefits is to change the volume of catalyst used while maintaining constant precious metal loading. Using the type 12C catalyst at 40g/ft^{-3} and the aging conditions described above the results obtained indicate that CO activity for both fresh and aged catalyst is at a maximum for the 75.4 in^{-3} catalyst, and that a steady decline in then observed with decreasing volume, the diameter being maintained at 4 in. A similar pattern with hydrocarbon activity is observed both at zero hours and after 300 hours aging. These results are shown in Figure 4. The results of these programmes with the 12C catalyst indicated that some savings were possible in the total cost of the catalyst, but that this would depend on the emission limits which are to be met.

this for a Johnson Matthey platinum/palladium catalyst type 26G. This work was done on a multibed reactor system and compares the effect of the engine aging cycle (cycle 1) described above (1 minute idle alternated with 1 minute equivalent to 70 mph) and a lower temperature cycle devised by Ford Motor Company (cycle 2). In both cases the fuel used contained 0.024g lead per US gallon but in the Ford cycle the average inlet temperature during the test is about 430°C, and the maximum temperature is around 760°C, compared to inlet temperatures running between 550°C and 850°C on the more severe cycle 1. In Table 1 the results shown in both cases are expressed as conversion efficiencies after 300 hours aging.

The implications of this set of results is that, in the context of thermal aging using low lead content fuels, lower precious metal loadings might be used where the catalyst is in a cool location than when a catalyst runs hot. Nevertheless even in the latter case savings in precious metal can be achieved compared to the loading used in the first generation of catalysts fitted to automobiles during model years 1975 and 1976, while still attaining the same degree of emission control. The advantages of low metal loadings result from the low levels of catalyst poisons obtained in currently available US gasoline. It has long been known that lead, phosphorous and sulhpur compounds can adversely affect the performance of both precious metal and base metal catalysts. In the specific case of automobile catalysts, it is found that the degree of activity loss is dependent on the concentration of poison in the exhaust gas, its composition, the form of the catalyst and its temperature. Further, the adsorption of lead by the catalyst is to a certain degree reversible. This can be shown both by direct analysis, and by the partial restoration in activity which occurs when a catalyst exposed to leaded fuel is switched to sterile fuel. The extent to which lead is retained by the catalyst is very dependent on temperature, reflecting the greater volatility of lead compounds at elevated temperatures (8).

Two different solutions can be considered for use with fuels having higher levels of lead than currently encountered in present US gasoline:

(i) Removal of the lead compounds prior to the catalyst. For this system to work the lead trap must be efficient in retaining lead for a long period of time. Since most systems of this type currently in development rely on alumina as the retaining element, which for maximum efficiency need to run in a cool location, the catalyst itself must be designed to operate efficiently at low temperatures.

(ii) Operation of the catalyst at high temperature, such that the level of lead retained by the catalyst is low enough not to affect the catalytic activity. In this case the catalyst must be designed to be resistant to thermal degredation by stabilisation of the catalyst components towards sintering.

METAL SUPPORTED CATALYSTS

The limitations imposed by the processes used to extrude ceramic monoliths and by the inherent properties of the ceramic materials that are suitable for automotive applications result in minimum cell wall thickness of the order of 2.51×10^{-3} cm. (0.010 in.) for currently available products.

This limitation has a significant effect on factors influencing the total system performance such as conversion, since it limits the surface available for mass transfer. In addition this factor may have a considerable effect on an aspect of growing importance; fuel economy. Thus, the thickness of the walls of ceramic substrates severely reduces the open area of a support having the cell density necessary to achieve the conversion levels required. This, in turn, adversely affects the power losses and therefore fuel economy.

The only solution to this problem is a drastic improvement in the properties of the material used to make the support so as to enable the manufacture of catalyst supports having very thin walls of about 0.5×10^{-3} cm. (0.002 in.). Only metals possess the requisite properties at such thicknesses. The potential advantages of such systems, in terms of substrate properties of prime importance, are summarised in Table 2. To exploit the advantage requires the identification of an alloy that is capable of quantity production, has sufficient corrosion resistance to withstand extended exposure to exhaust gases at elevated temperatures and has a surface capable of accepting a catalyst coating preferably of type already developed for ceramic systems.

Identification of suitable alloys.

The constraints of the application demand resistance to corrosion in exhaust gas at temperatures around 1000°C. A further requirement is a lack of interaction with the catalyst coating, which is normally alumina based, a factor which has focussed attention on alloys which passivate by formation of alumina layer, such as Armco 18SR, Kanthal DSD etc.

Recent work by the Harwell laboratory of the United Kingdom Atomic Energy Authority (UKAEA) has resulted in the development of Fecralloy* steels. These alloys of chromium (15-20%), alumina (4-6%), Yttrium (O-0.5%) and iron were developed initially for nuclear applications and exhibit high temperature corrosion and oxidation resistance. Comparative tests on Fecralloy* steel and other potential candidates conducted in hot exhaust gas from a spark ignition engine are summarised in Table 3.

The technical advantages of Fecralloy* steel over other materials is atributed to its ability to form a self healing film of alumina. Following an extensive development programme Fecralloy* steels are now commercially available in the United Kingdom and their manufacture in the USA is being actively investigated.

Catalyst manufacture.

There are two stages:

(i) Conversion of sheet steel into a rigid mechanical support

(ii) deposition of an adherent layer of an active and durable catalyst system.

Fabrication of metal support.

A number of methods have been proposed for producing multicellular thin walled metal supports capable of offering high surface area with low impedance to gas flow.

A simple method used in the tests described later involves corrugation of one strip of metal and coiling it under tension together with a

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* The registered trade mark of the UKAEA.

plain interleaf. This process is similar in many respects to techniques used in paper manufacture and has also been used for making early versions of ceramic supports. This method enables manufacture of cylindrical units the cell density and cell profile of which can be varied by changing the size and shape of the teeth on the corrugating rolls.

A typical catalyst together with a convertor is shown in Figure 8.

The catalysts evaluated in the study were all based on metal support manufactured at a fixed length of 3.5 in. which was chosen on the basis of an outline mathematical model. Variations in volume were achieved by changes in the diameter. The cell profile was similar in all the catalysts examined.

Catalyst

The catalysts used in this programme are identified in Table 4. The metal supports were made compatible with a proprietary oxidation catalyst by a novel pretreatment. In each case the weight of alumina based washcoat and precious metal catalyst per unit volume of catalyst were kept constant.

As a basis for comparison, the identical catalyst was deposited on a commercially available 75 cu.in. ceramic monolith at presently accepted metal loadings.

The catalysts were inserted into convertors using an interference fit and a restraint bar for the tests. A convertor of similar design has been successfully tested in a 50,000 mile trial conducted by Harwell.

Zero hours parametric study

The performance of the various catalysts at zero hours was evaluated on a 1.8L Leyland Marina test vehicle. Overall conversions in the CVS test were used as the basis for comparison to eliminate base line variations. In virtually every test tail pipe mass emissions are well within the statutory limits. During the CVS test the pressure drop across the catalyst was continuously monitored by means of a pressure transducer.

The data for a proprietary platinum/rhodium oxidation catalyst system deposited on the metal support together with that for the 75 cu.in. ceramic supported system is summarised in Figure 9.

The following conclusions may be made:

(i) The activity of metal supported catalysts for the oxidation of hydrocarbons is more dependent on catalyst volume than for the oxidation of carbon monoxide.

(ii) At cell densities in excess of 400/sq.in. the overall activities of the relatively low volume metal supported catalysts are equivalent to the ceramic system. Furthermore, the activity is, within the range evaluated, sensibly independent of catalyst volume.

(iii) Although at 400 cells/sq.in. performance at the highest volume ($33in^3$) metal supported is identical to that of the ceramic ($75in^3$), hydrocarbon performance declines markedly as the volume is decreased.

The maximum pressures during the CVS test are compared in Figure 10. The base for comparison is the peak back pressure i.e. that during the second power mode. The corresponding data for the total exhaust system excluding the catalyst and for the 75 cu.in. ceramic system are also included.

It may be seen that:

(i) by close attention to catalyst design pressure drops and therefore power losses are lower or the same as ceramic catalysts at lower catalyst volumes and higher cell densities.

(ii) pressure losses increase as cell density is increased.

(iii) at any cell density the pressure drop increases as the volume decreases. The effect is not directly related to the volume, indicating significant flow maldistribution at higher volumes.

Since the mass flow during the second power mode is not known accurately the pressure loss data has been augmented by a comparison of the pressure drop over the catalysts in a static fan blown test apparatus.

The raw data for 400 and 500 cell units are summarised in Figures 11 and 12 respectively, together with that for the 75 in^3 ceramic supported system. This data is in good qualitative agreement with the data obtained during the CVS test.

Extended durability testing has been conducted using a static engine test stand in conjunction with a slave vehicle for CVS testing. This test procedure, using a Triumph Dolomite 1.8L overhead camshaft engine fitted with an air pump delivering secondary air immediately prior to the convertor as described in an earlier part of this paper relating to the aging of variable metal loading catalysts (cycle 1). Catalyst activity was checked at regular intervals in a Chrysler Avenger 1.5L slave vehicle. Durability data for a proprietary platinum/palladium catalyst supported on metal and ceramic supports are summarised in terms of mass emissions in Figure 13 and residual emissions in Figure 14. The data covers two types of metallic support each of $33in^3$ volume. The first type is the interleaf type already described, the second identified as the helical type is fabricated from two corrugated sheets of metal each having corrugations at a small and opposite angle to the axis of the final unit. Each of the metallic units and the $75in^3$ ceramic unit used as the basis for comparison is tested with the same absolute weight of platinum/palladium catalyst. The ceramic and the two metallic concepts show a similar degree of activity and durability.

Similar data has been obtained using platinum/rhodium catalyst deposited on the two types of metallic supports.

ACKNOWLEDGEMENT

The authors acknowledge the assistance of the UKAEA for providing the Fecralloy* steel sheet for this test programme, and the contributions of other members of the Johnson Matthey Group Research and Development Departments to the work described in this paper is also gratefully acknowledged.

* The registered trademark of the UKAEA.

CONCLUSIONS

The developments described in this paper show how catalysts are being devised that can help the automobile engineer to produce systems that control emissions to low levels with good fuel economy. The authors are optimistic that catalysts are being developed that can be adopted to meet future legislated emission standards in other parts of the world.

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Table 1

Noble Metal loading, $g.ft.^{-3}$	Conversion Efficiency, %			
	Cycle 1		Cycle 2	
	CO	HC	CO	HC
25	92	60	100	84
15	88	51	100	78
10	55	48	-	-

Table 2

Ceramic-Metal Support Comparison

Typical Property	Ceramic Cell/in^2		Metal Cell/in^2		
	200	300	400	500	600
Cell area (cm^2)	0.023	0.143	0.015	0.011	0.0097
Surface to volume ratio (cm^{-1}) (ft^{-1})	18.9 576	22.05 672	32.3 987	35.8 1090	39.4 1197
Open area	70	60	89	86	83
Wall thickness (cm) (in)	0.028 0.011	0.030 0.012	0.005 0.002	0.005 0.002	0.005 0.002

Table 3

Engine Exhaust Corrosion Tests

Exposure Temp. °C.	ARMCO 18SR	AISI 310	KANTHAL DSD	FECRALLOY
1085	Failed	Failed	Failed	OK at 185 hrs.
1050	Failed at 30 hrs.	Deformed at 30 hrs.	OK at 30 hrs.	OK at 30 hrs.
1025	Failed at 95 hrs.	Failed at 95 hrs.	OK at 95 hrs.	OK at 95 hrs.
1000	Failed at 215 hrs.	Highly oxidised	OK at 420 hrs.	OK at 420 hrs.
950	Failed at 390 hrs.	Failed at 625 hrs.	OK at 625 hrs.	OK at 625 hrs.

Table 4

Matrix of Property Variables Evaluated During Parametric and Durability Programmes

Type	Volume (in^{-3})	Cell Density (in^{-3})	Active Species	Test Programme
Ceramic	75.4[1]	300	7.5% Rh/Pt[1]	Parametric/ Durability
Ceramic	75.4[2]	300	2:1 Pd/Pt	"
Metallic	33.7[1]) 26.9) 20.2) 16.8)	400, 500, 600	7.5% Rh/Pt	Parametric
Metallic	33.7[1]	400	7.5% Rh/Pt[1]	Durability
	33.7[2]	400	2:1 Pd/Pt	"

N.B. 1. Units contain same absolute weight of precious metal equivalent to 40 gm per cubic foot of ceramic volume.

2. Units as above but contain metal equivalent to 25 gm Pd/Pt per cubic foot of ceramic volume.

Captions for Figures

Fig. 1	Fuel economy trends for US vehicles on a sales-weighted basis from 1967 to 1976.
Fig. 2	The effect of platinum metal loading on residual CO emissions after 0.200 and 300 hours engine aging catalyst: Type 12C on 4 in. diameter x 6 in. long M20/200 cells in^{-2} substrate.
Fig. 3	The effect of platinum metal loading on residual HC emissions after 0.200 and 300 hours engin aging catalyst: as for Fig. 2.
Fig. 4	The effect of catalyst length at 4 in. diameter on residual CVS emissions. Catalyst:Type 12C on M20/200 cells m^{-2} substrate, aged 300 hours.
Fig. 5	Comparison of the effect of metal loading on residual CO emissions, after 300 hours aging, for Type 12C and Type 20G catalysts on 4 in. diameter x 6 in. long M20/200 cell $in.^{-2}$ substrate.
Fig. 6	Comparison of the effect of metal loading on residual HC emissions, after 300 hours aging, for Type 12C and 20G catalysts.
Fig. 7	Platinum/rhodium catalyst Type 23G. Effect of metal loading on residual emissions after 300 hours aging. Substrate: M20/200 cells in^{-2}.
Fig. 8	Catalyst on Fecralloy steel support together with typical convertor.
Fig. 9	Catalyst activity of metal supported catalysts at zero hours. Sum weighted hydrocarbon (upper) and carbon monoxide (lower) conversions for a platinum/rhodium catalyst ▽ 400 cells/sq.in. (CPSI), △ 500 CPSI, ○ 600 CPSI Metallic, ● 300 CPSI Ceramic. All metal supports contained 90g/cu.ft. PGM, ceramic 40g/cu.ft.
Fig. 10	Maximum exhaust pressure during the CVS test (Power Mode 2) for metallic and ceramic supported catalysts. × 400 CPSI, △ 500 CPSI, ○ 600 CPSI metallic, ● 300 CPSI ceramic. (Baseline is 50 mm Hg.)
Fig. 11	Pressure drop over 400 CPSI metal and 300 CPSI ceramic supported catalysts. 1-33.7 in^3, 2 26.9 in^3, 3 20.2 in^3, 4 16.8 in^3 metal, all 3.5 in. long. 5-75 in^3 (6 in. long) square cell ceramic catalyst.

Fig. 12 Pressure drop over 500 CPSI metal and 300 CPSI ceramic supported catalysts. Legend as in Fig. 11.

Fig. 13 Durability data for metal and ceramic supported platinum/palladium catalysts each containing 0.68g precious metal. Data are sum weighted emissions in the CVS test after static engine aging. Upper sum weighted hydrocarbon, lower sum weighted carbon monoxide; O 400 CPSI interleaf concept 33.7 in^3; □ 400 CPSI helical concept 33.7 in^3; 300 CPSI ceramic concept 75 in^3.

Fig. 14 Durability data for metal and ceramic supported platinum/palladium catalysts as in Fig. 13. Data are residual sum weighted emissions of HC (upper) and CO (lower) during the CVS test i.e. (100 - conversion)%. Legend as in Fig. 13.

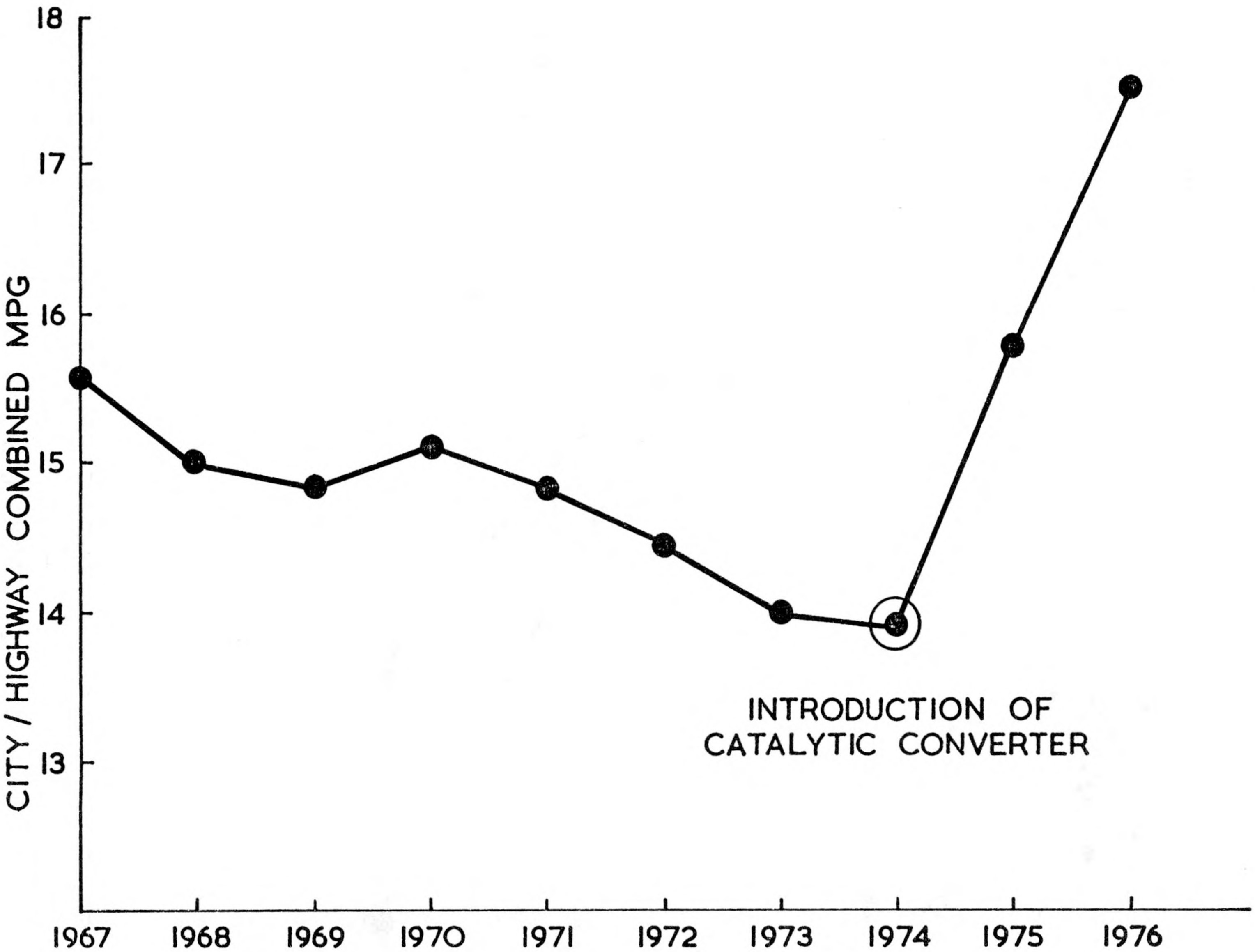
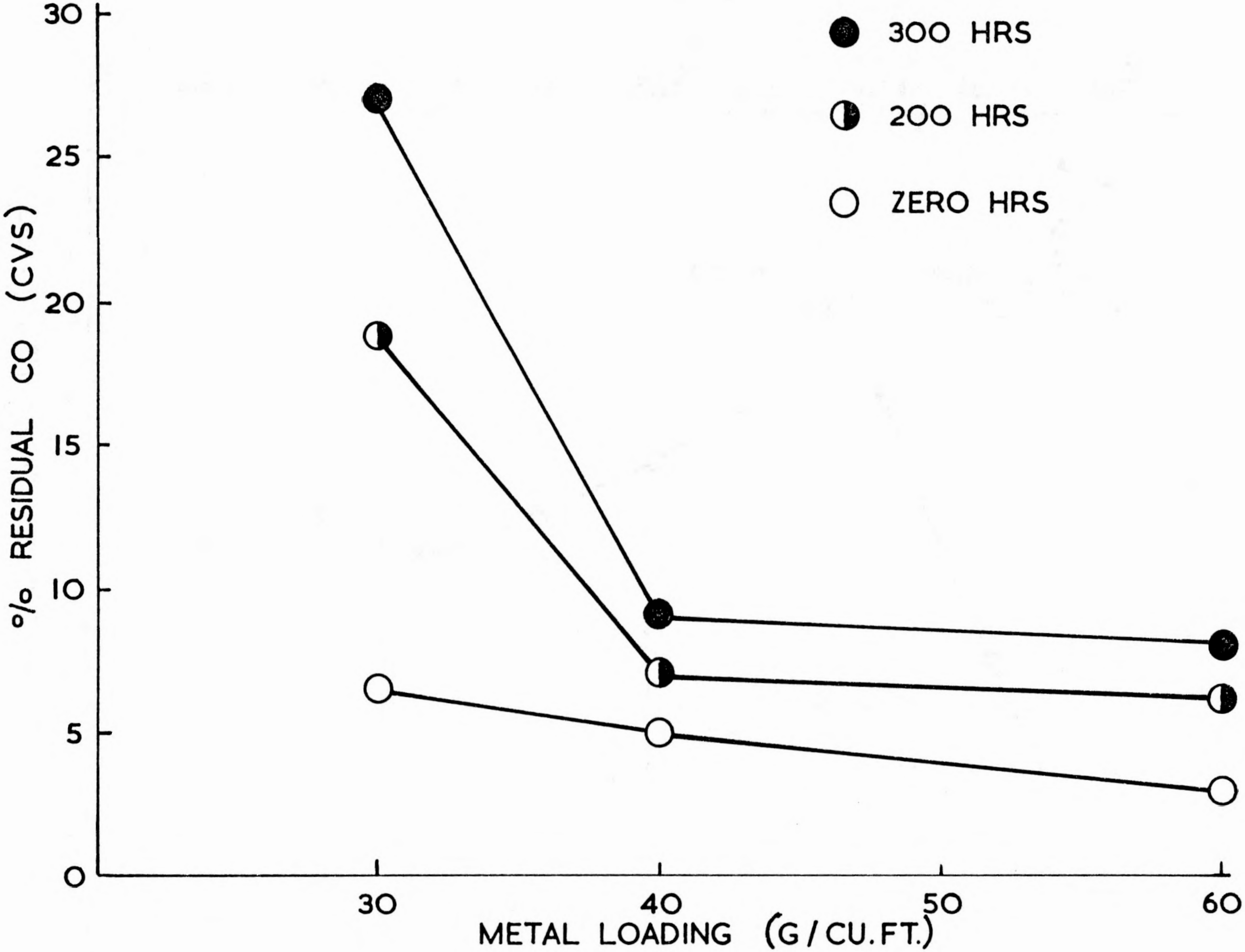

18
17
16
15
14
13
CITY / HIGHWAY COMBINED MPG
INTRODUCTION OF
CATALYTIC CONVERTER
1967
1968
1969
1970
1971
1972
1973
1974
1975
1976

Fig 1.



300 HRS
200 HRS
ZERO HRS
% RESIDUAL CO (CVS)
0
5
10
15
20
25
30
30
40
50
60
METAL LOADING (G / CU. FT.)

Fig 2

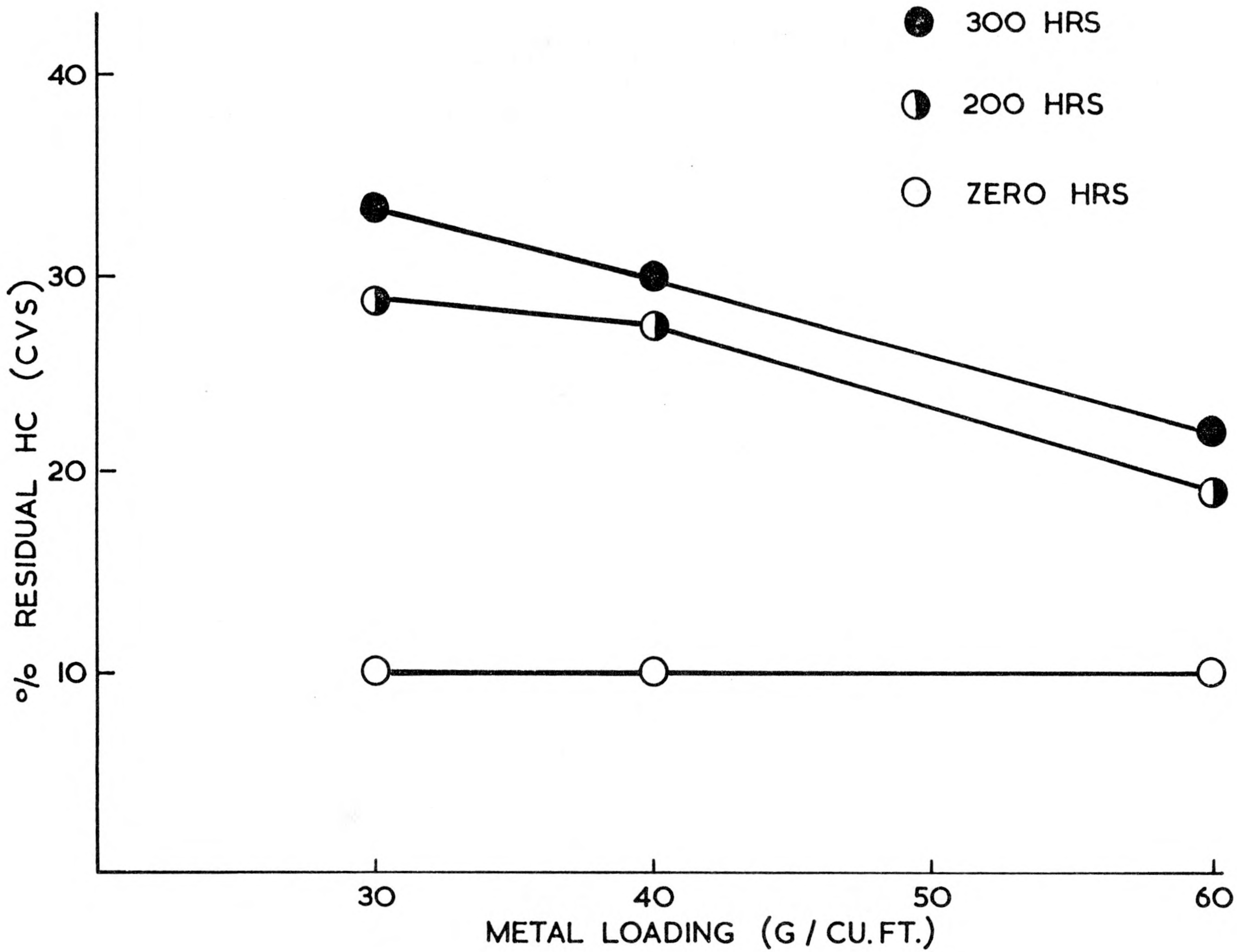

300 HRS
200 HRS
ZERO HRS
40
30
20
10
% RESIDUAL HC (CVS)
30
40
50
60
METAL LOADING (G / CU. FT.)

Fig 3

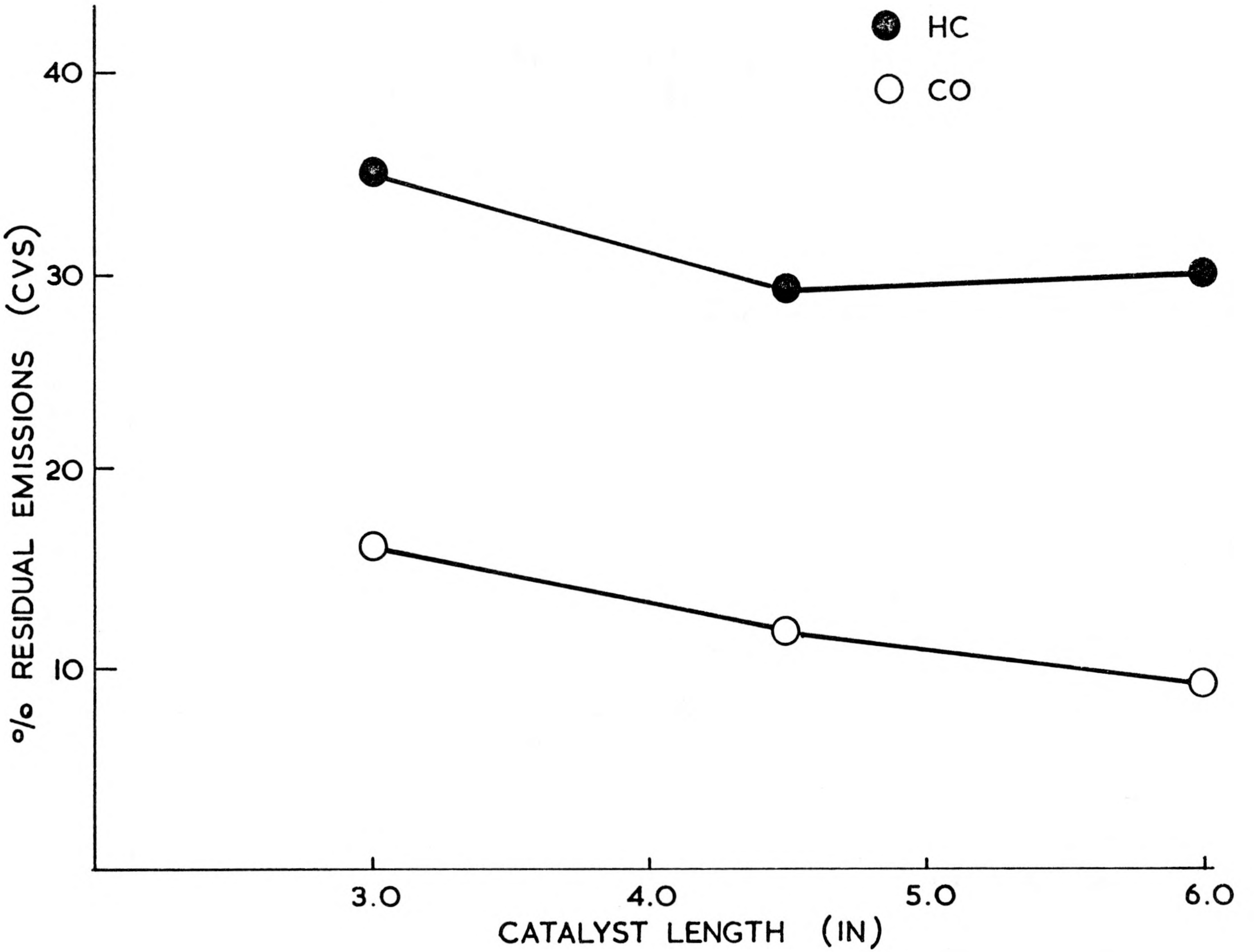
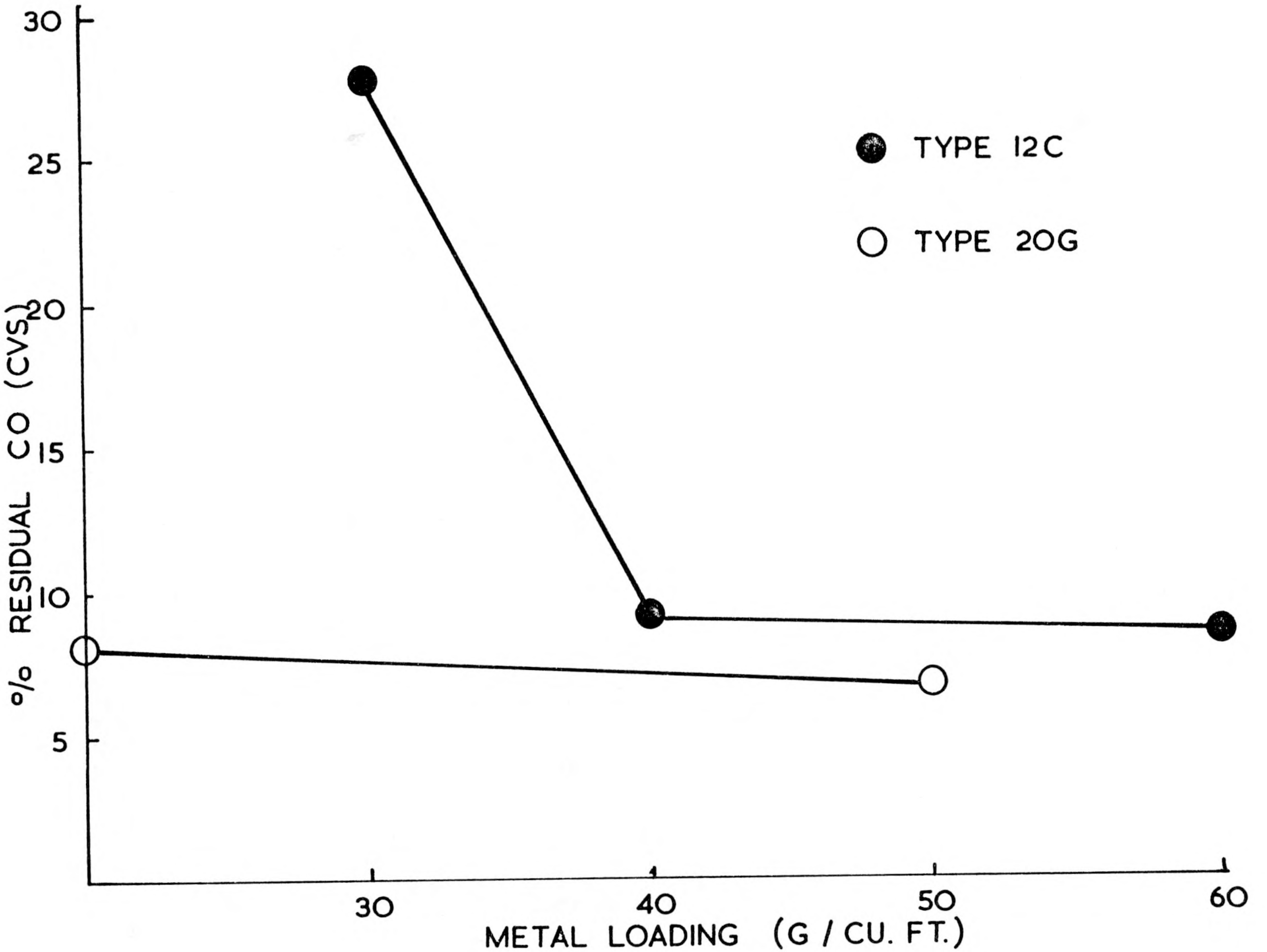

HC
CO
40
30
20
10
% RESIDUAL EMISSIONS (CVS)
3.0
4.0
5.0
6.0
CATALYST LENGTH (IN)

Fig 4



TYPE 12C
TYPE 20G
% RESIDUAL CO (CVS)
30
25
20
15
10
5
30
40
50
60
METAL LOADING (G / CU. FT.)

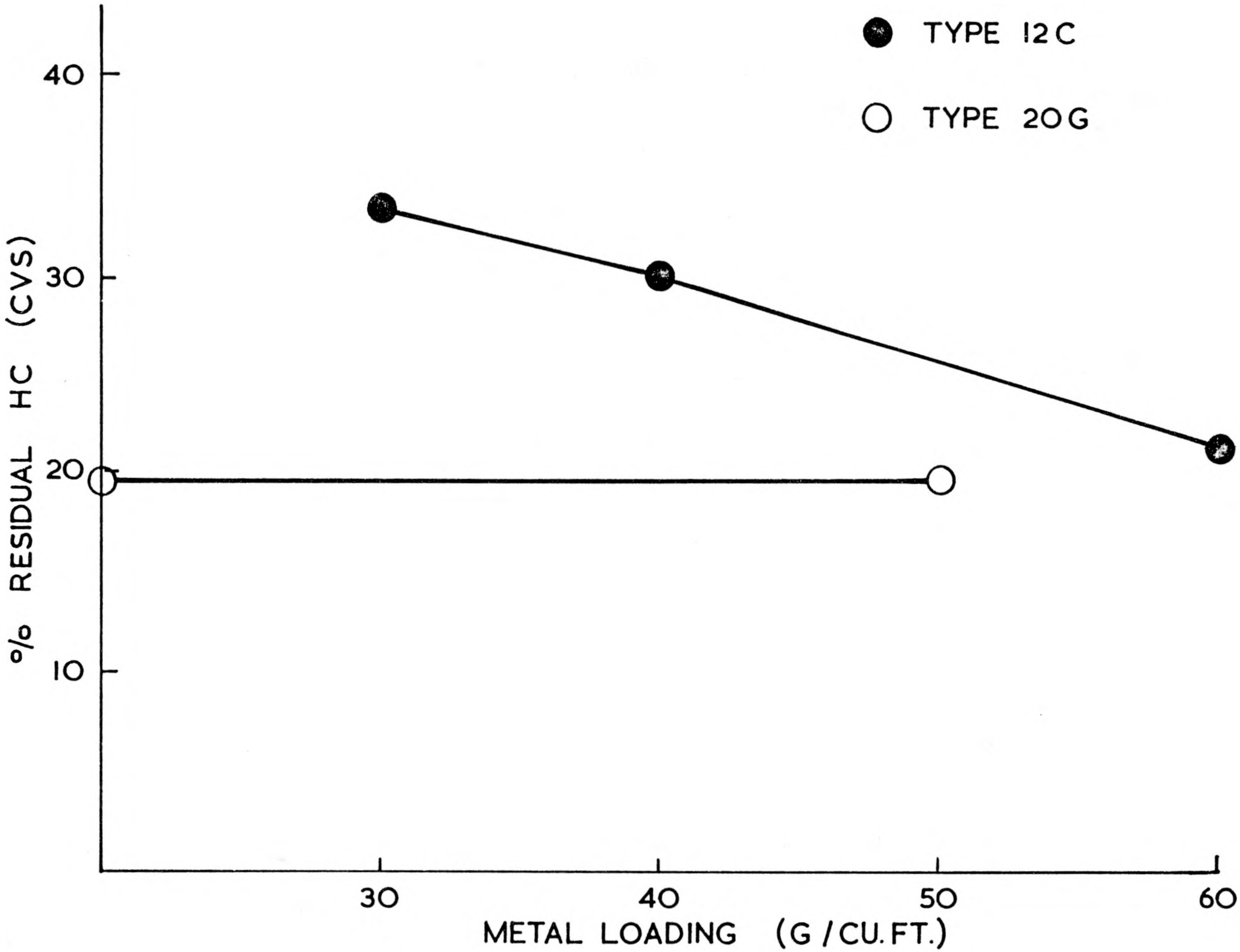

TYPE 12C
TYPE 20G
40
30
20
10
% RESIDUAL HC (CVS)
30
40
50
60
METAL LOADING (G / CU. FT.)

Fig 6

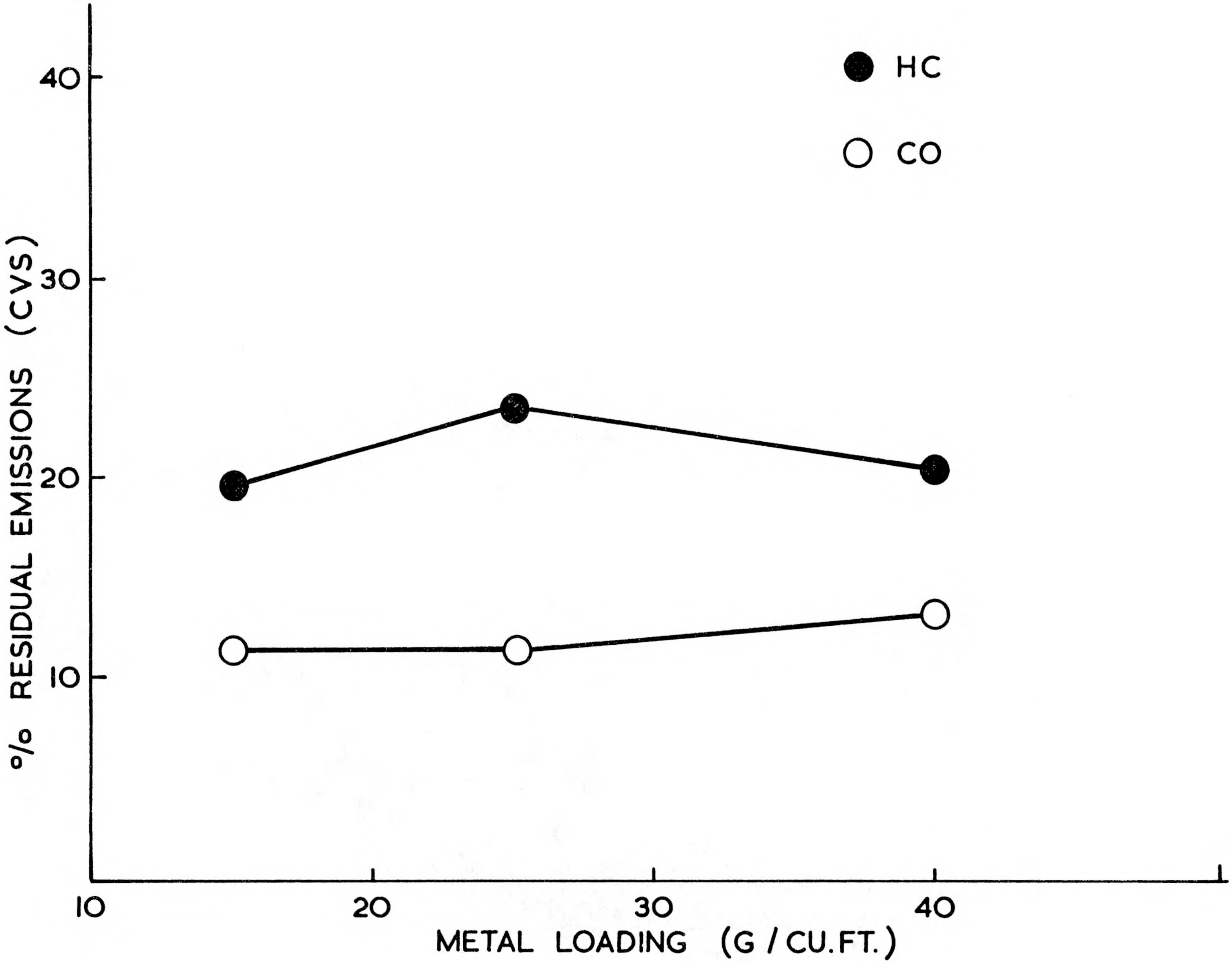

HC
CO
40
30
20
10
% RESIDUAL EMISSIONS (CVS)
10
20
30
40
METAL LOADING (G / CU. FT.)

Fig 7


T.I.
LTD

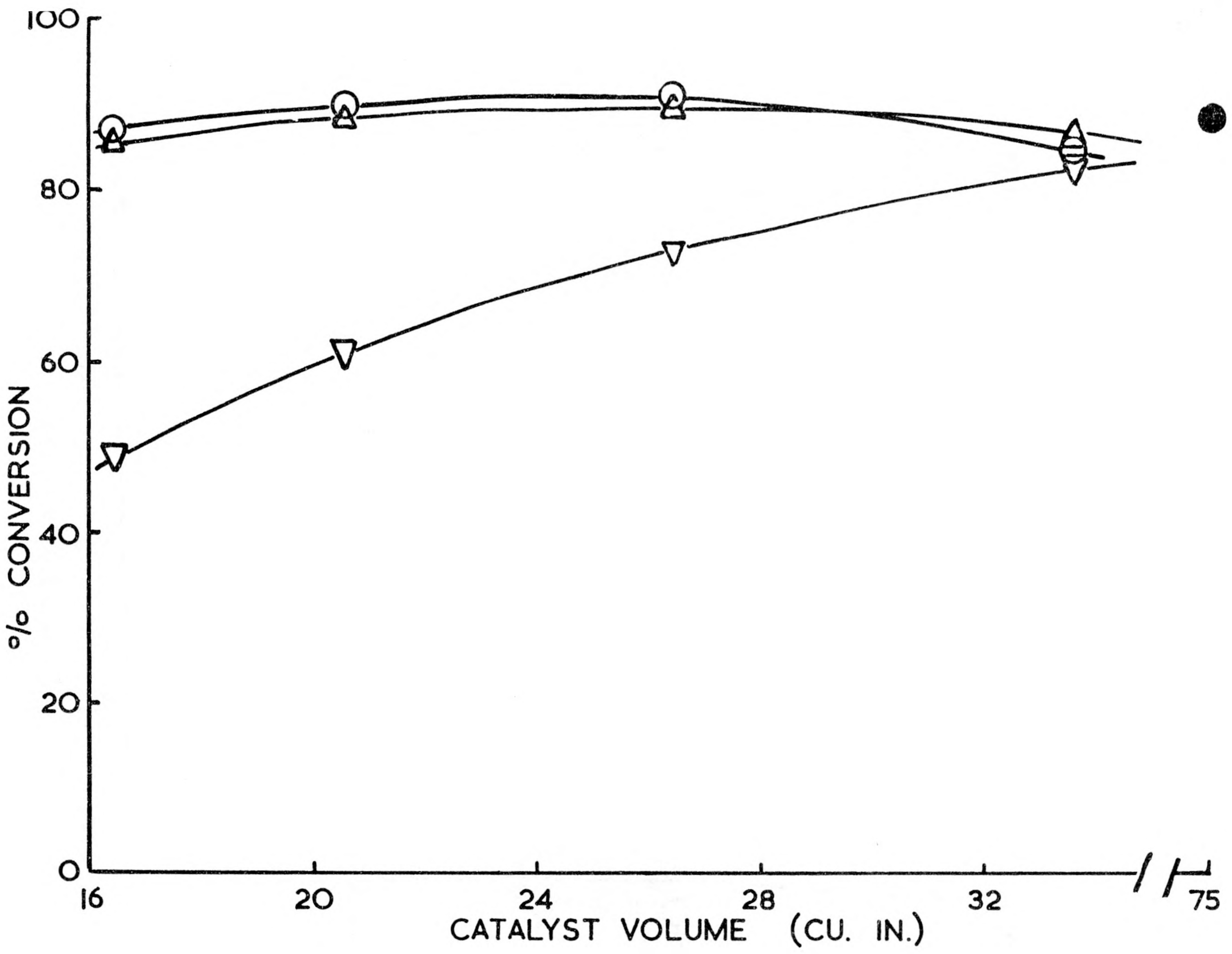
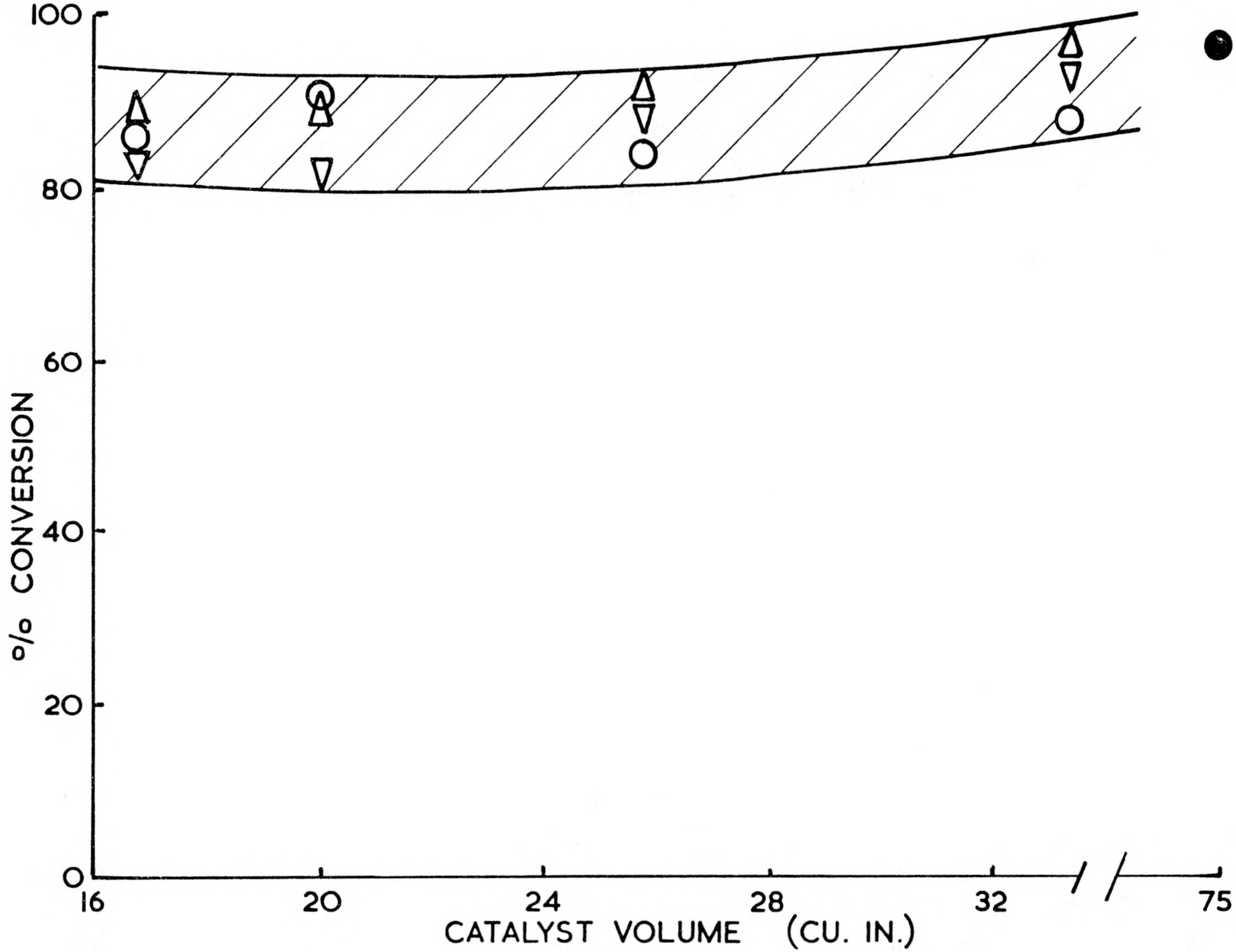
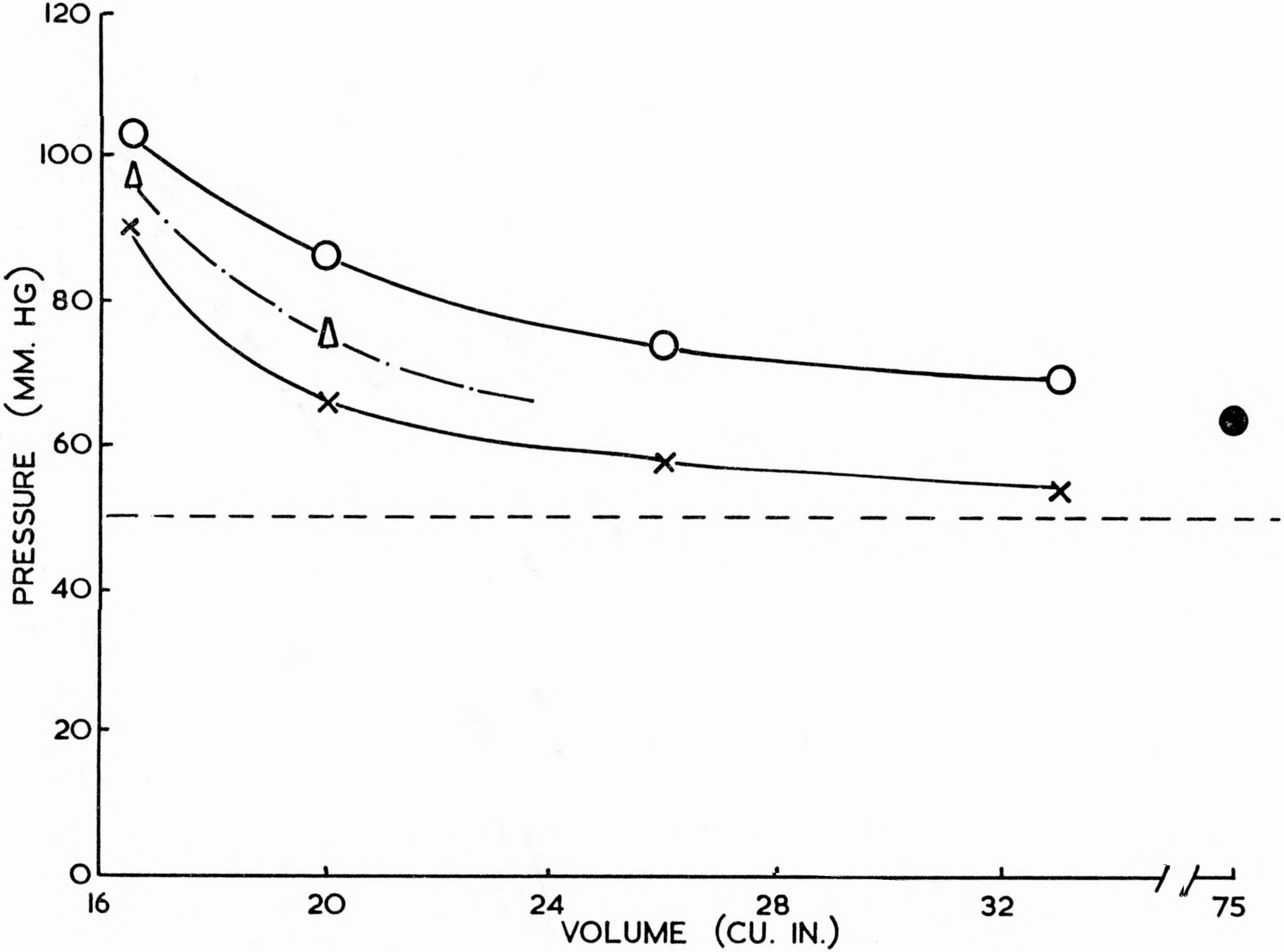

100
80
60
40
20
0
% CONVERSION
16
20
24
28
32
75
CATALYST VOLUME (CU. IN.)

Fig 9 upper



100
80
60
40
20
0
% CONVERSION
16
20
24
28
32
75
CATALYST VOLUME (CU. IN.)

Fig 9 Lower



120
100
80
60
40
20
0
PRESSURE (MM. HG)
16
20
24
28
32
75
VOLUME (CU. IN.)

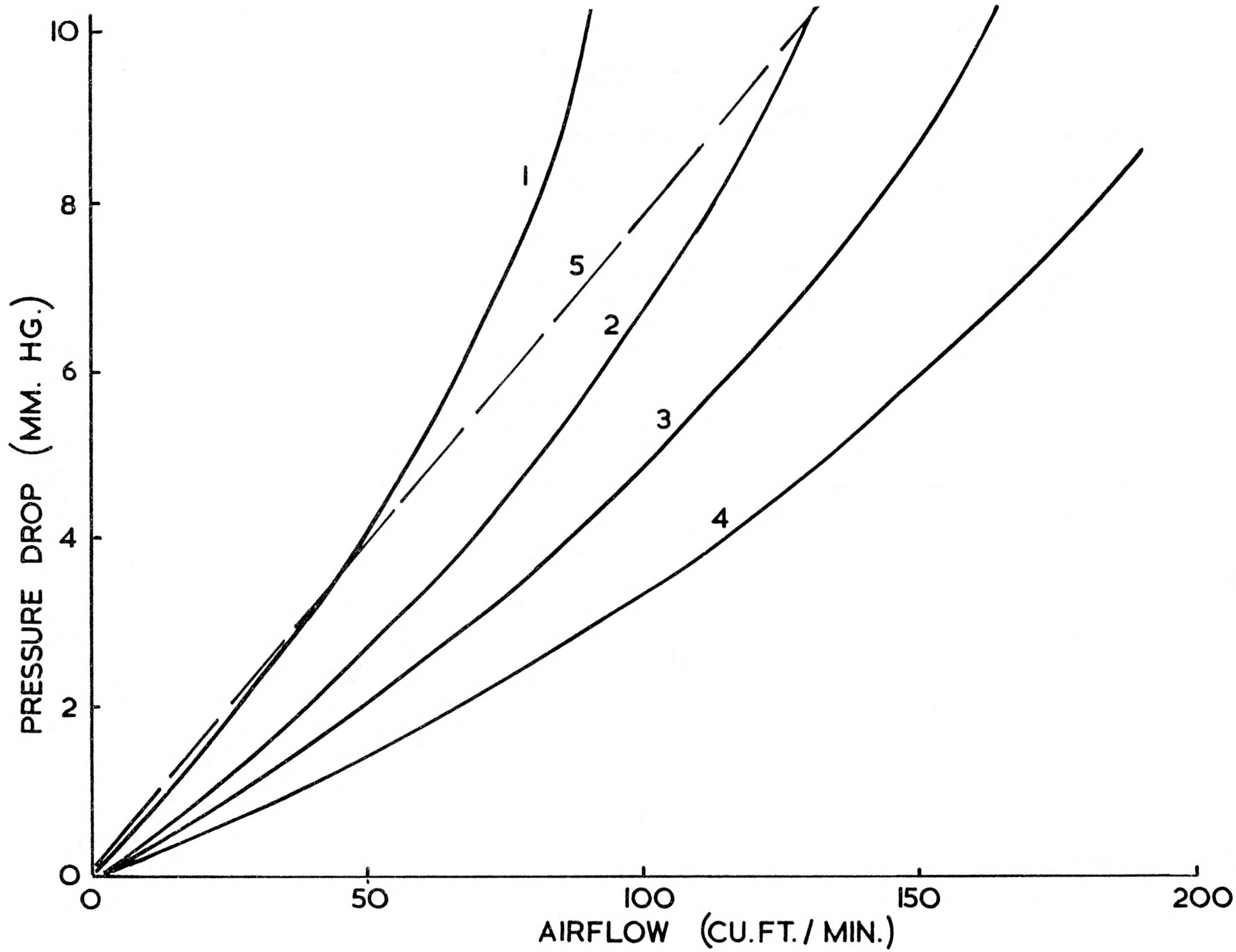

PRESSURE DROP (MM. HG.)
10
8
6
4
2
0
0
50
100
150
200
AIRFLOW (CU.FT. / MIN.)
1
5
2
3
4

Fig 11

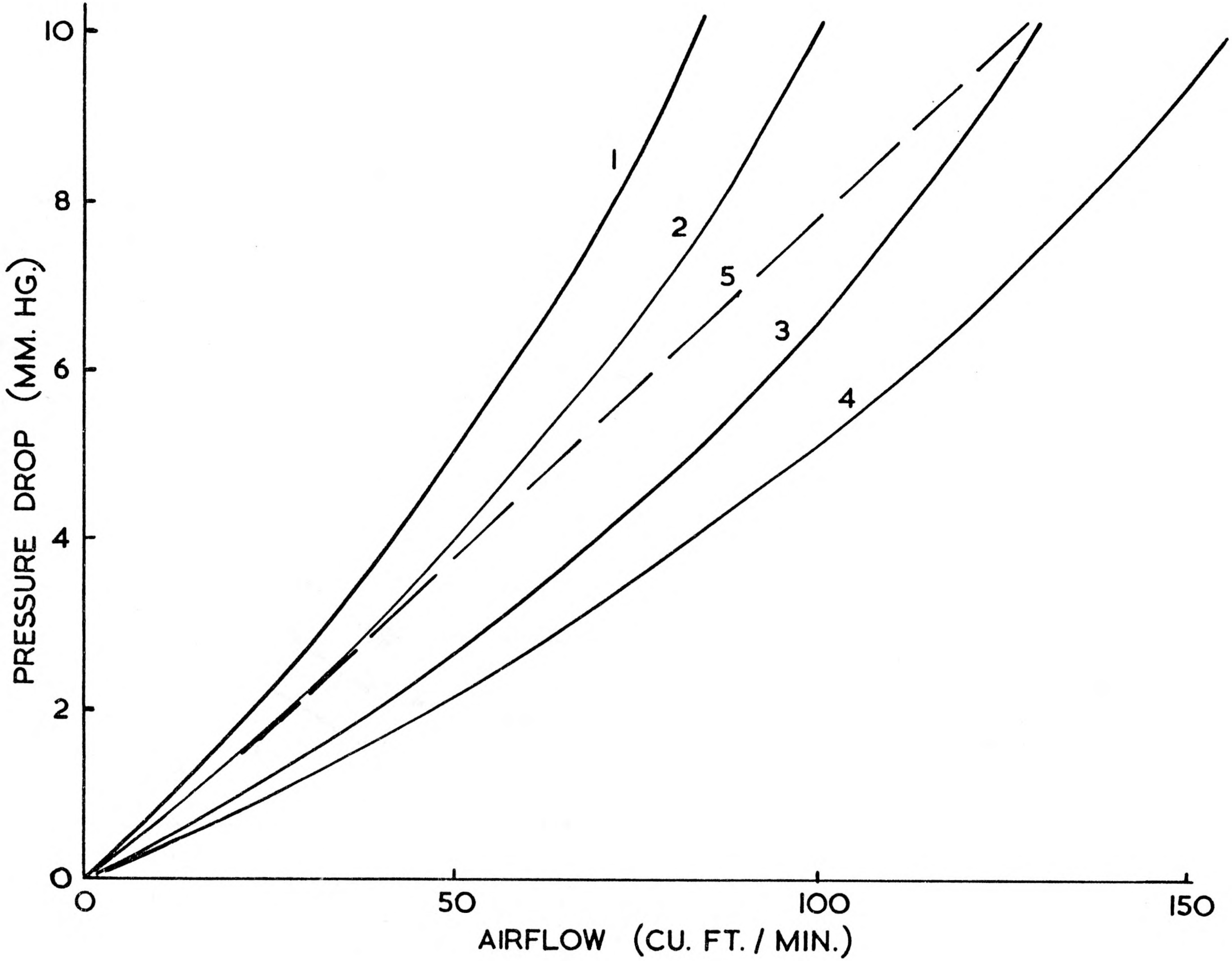

PRESSURE DROP (MM. HG.)
AIRFLOW (CU. FT. / MIN.)
0
2
4
6
8
10
0
50
100
150
1
2
3
4
5

Fig 12

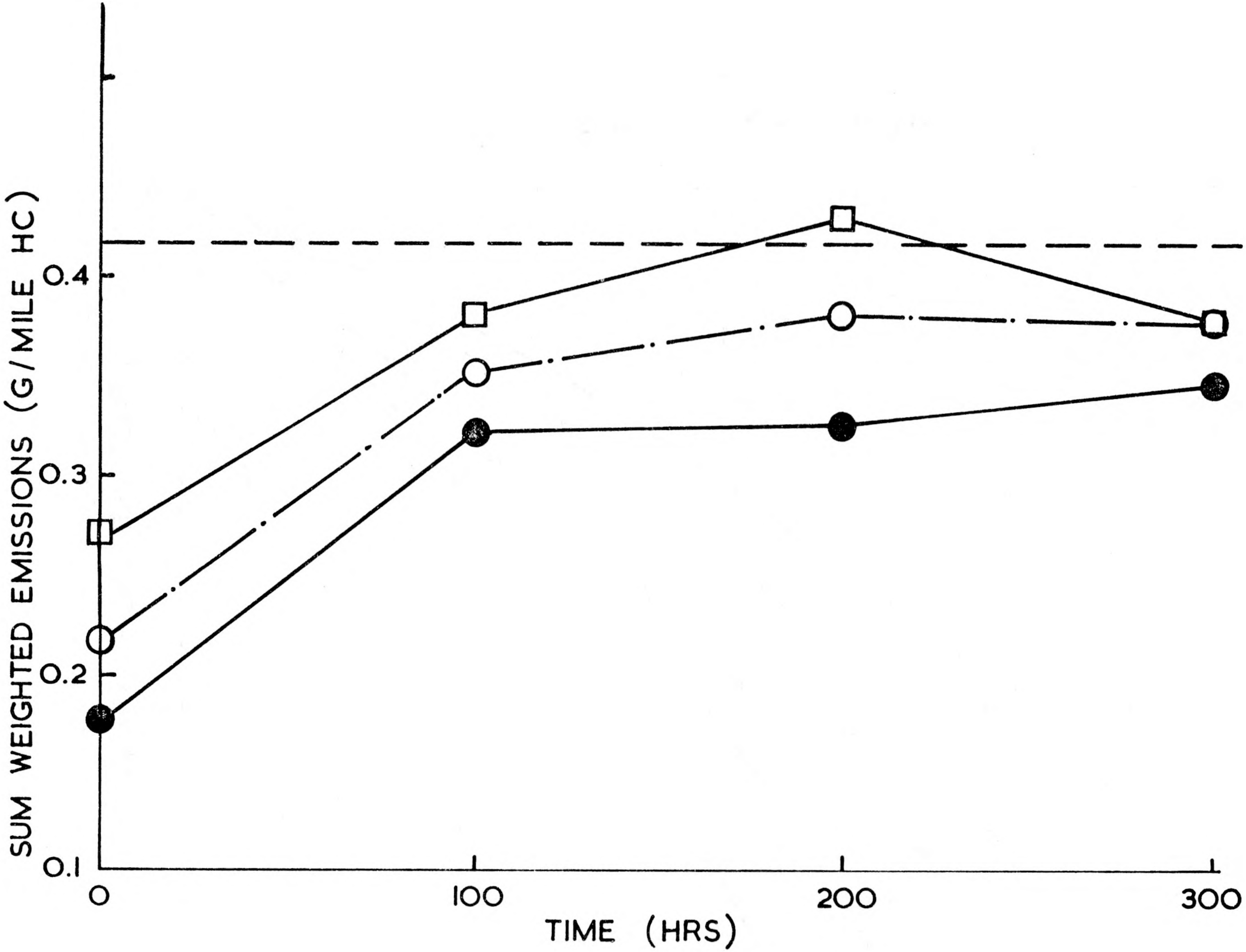


Fig 13 [redacted]

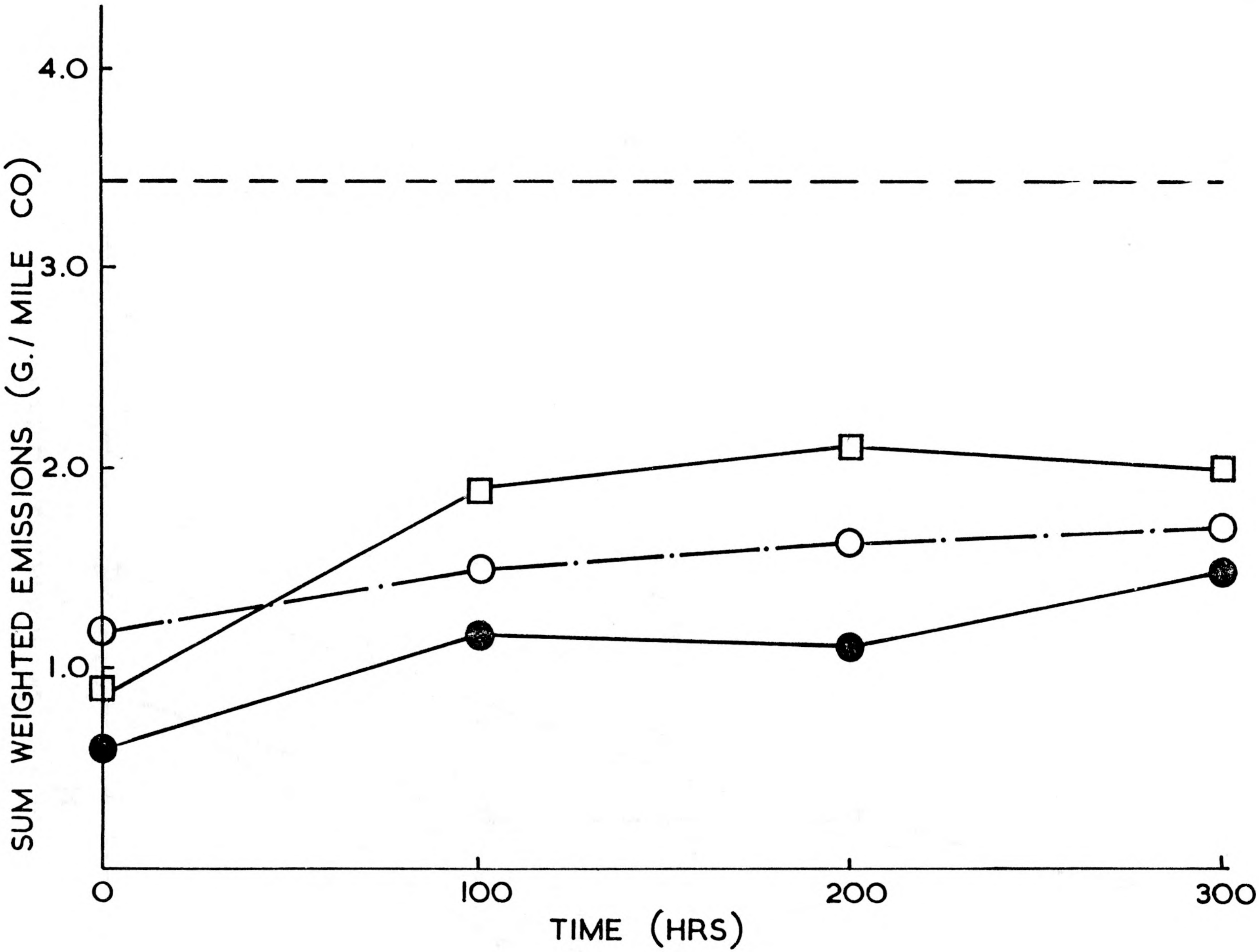

SUM WEIGHTED EMISSIONS (G./MILE CO)
4.0
3.0
2.0
1.0
0
100
200
300
TIME (HRS)

Fig 13 Lower

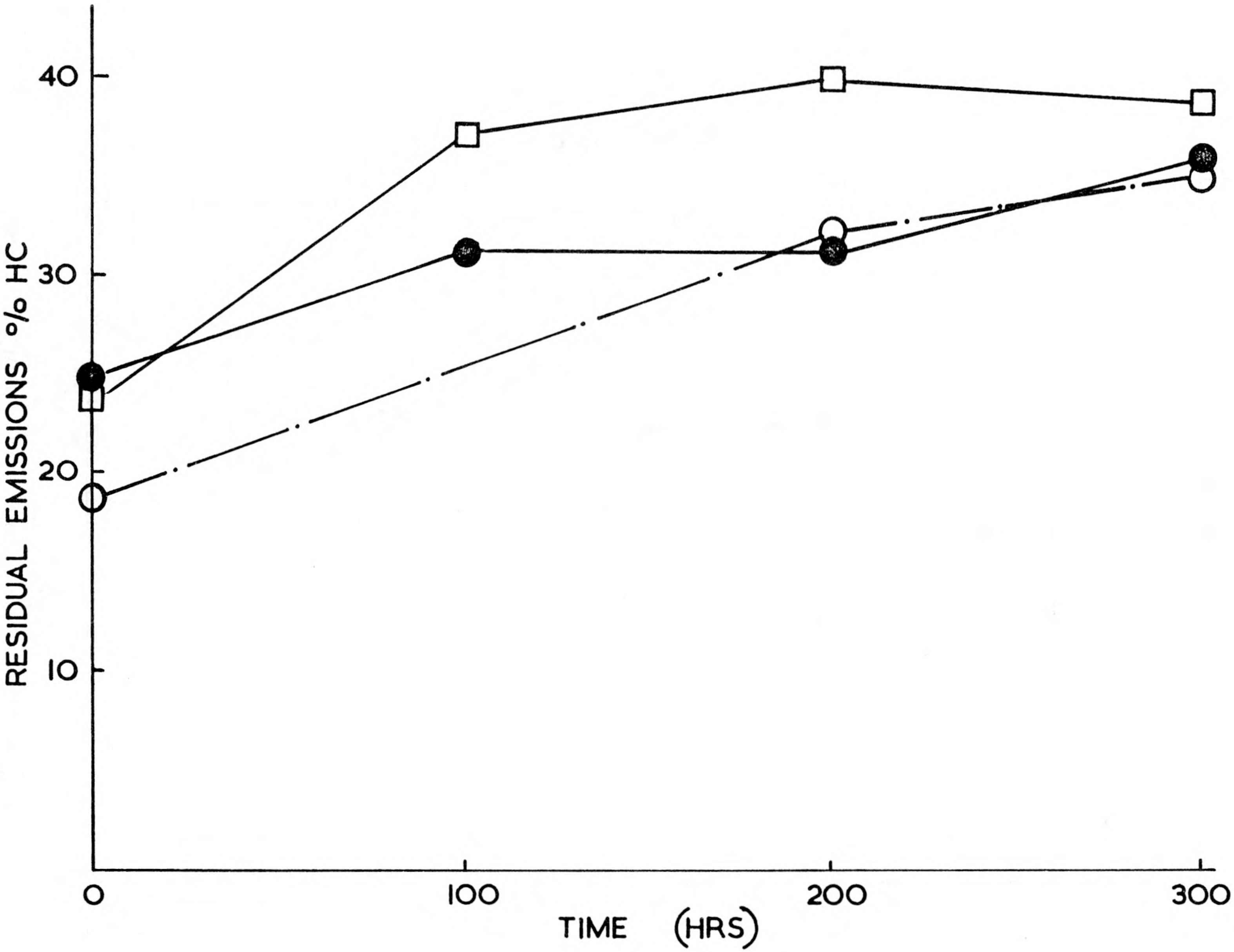


Fig 14 PER

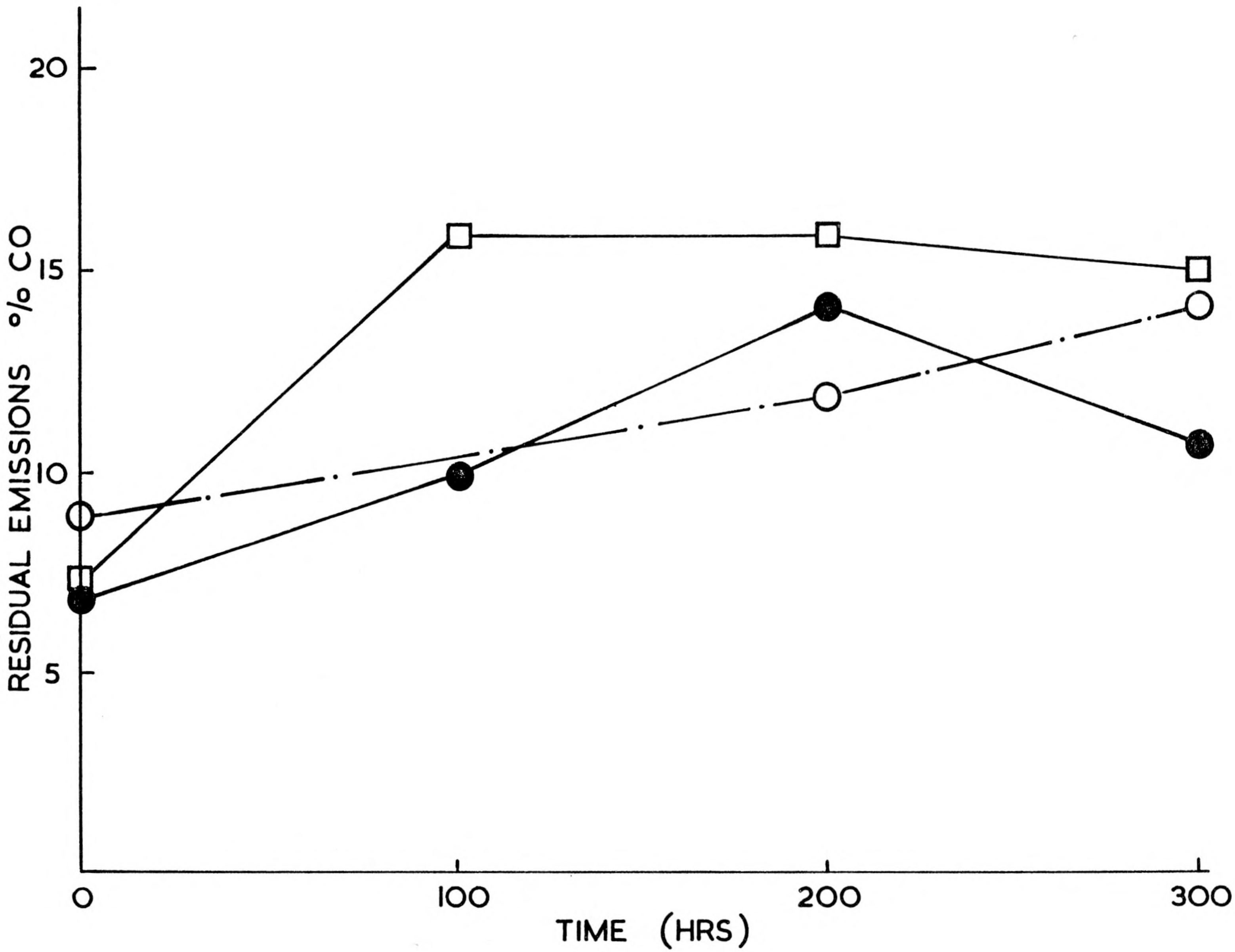


Fig 14 Lower

EMISSION CONTROL OF V-ENGINES WITH CATALYSTS AND OXYGEN SENSOR

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ABSTRACT

The various possible arrangements for application of closed loop mixture control with oxygen sensor for fuel-injected V-engines with catalytic emission control are described.

The decisive criteria such as fuel injection system, lay-out of exhaust system, characteristics of control, positioning and operation data of oxygen sensor and catalysts, which define the concept, are being discussed.

In addition, the conversion rate of three-way-catalysts in quasistoichiometric operation is elaborated. The influence of nobel metal composition and quantity, as well as aging of catalysts on the size of the "air/fuel ratio-window" are explained, also the possibility for optimizing the closed loop system. Finally CVS test results of such systems are presented.

EMISSION CONTROL OF V-ENGINES WITH THREE-WAY-CATALYSTS AND OXYGEN SENSOR

Progressive tightening up of exhaust emission limits for motor vehicles powered by internal combustion engines is increasingly calling for the use of catalytic control systems. For optimum conversion not only of the oxidizing pollutants carbon monoxide and unburned hydrocarbons but also for reduction of nitrogen oxides, at the same time ensuring the best possible fuel economy and driveability, the use of so-called three-way catalysts in conjunction with a gasoline engine mixture control system based on stoichiometry has of late gained wide acceptance.

Fig. 1 shows that in the range below stoichiometric with a shortage of oxygen, nitrogen oxide emission is extremely low due to the catalytically-induced reduction with CO, whereas in the oxygen surplus range the emission of carbon monoxide and hydrocarbons downstream of the catalyst is at a minimum. With a stoichiometric mixture on the other hand, carbon monoxide, unburned hydrocarbons and nitrogen oxide together have a low emission rate. The undesired by-products, such as ammonia (NH_3), hydrogen cyanide (HCN) and sulfates (SO_4), are likewise at a minimum or cannot be quantitatively detected using the best analytical means downstream of a catalyst under stoichiometric operation.

A prerequisite for efficient conversion, which is the same for all pollutants whether covered or not by legal regulations, is, in addition to selecting an appropriately effective catalyst, that the eingine should be operated with a virtually exact stoichiometric air-fuel mixture. The lambda range to be observed is so narrow that it cannot be kept by means of a mixture control system alone. It requires an additional lambda control system, i.e. the mixture composition must be measured and compared with the desired value as the actual control value. The difference between the two values then serves as a correction signal for fuel metering. The mixture composition is detected on the exhaust side by the oxygen sensor, which measures the oxygen concentration of the exhaust gas with great accuracy in the immediate vicinity of the stoichiometric mixture.(1)

The oxygen sensor shown in Fig. 2 is a galvanic oxygen concentration cell with a solid electrolyte, whose voltage is given by the Nernst equation:

$$EMF = \frac{R \times T}{n \times F} \times T \times \ln \frac{P_{O_2} (air)}{P_{O_2} (exhaust)}$$

where R = gas constant (8.30 J/mol x K)

T = absolute temperature

n = number of charge equivalents per molecule of oxygen

F = Faraday constant (97.0 J/mol x mV)

P_{O_2} air = oxygen partial pressure, air

P_{O_2} exhaust = oxygen partial pressure, exhaust

(1) Zechnall, MTZ 34 / 73

Yttria-stabilized zirconium dioxide serves as the solid, gastight electrolyte. It is a virtually pure oxygen conductor in the 350 to 900°C temperature range applicable to engines. The oxygen ion conductivity results from the substitution of the tetravalent zirconium ion in the crystal lattice by cations of low valence. For reasons of electroneutrality vacancies are created within the oxygen lattice sites; the oxygen ions can be transported via these vacancies. The solid electrolyte is provided with thin, porous electrodes, one side of which is exposed to exhaust gas, the other to atmospheric air. The electrode extending into the exhaust gas flow is, at the same time, catalytically active, so adjustment of the reaction equilibrium of the exhaust emission components is facilitated on the surface. In this way the characteristic step from approx. 100 mV to approx. 850 mV, as shown in Fig. 1, is reached within a narrow range around stoichiometric mixture, i.e. as the oxygen concentration changes from slight excess to slight deficiency. (2)

This voltage step is interpreted in control technology such that by changing the fuel supply in general a stochiometric mixture ensues on average at the engine or catalyst.

To enable better understanding, a brief look will be taken at the control characteristics of gasoline engines with continuous fuel injection.

(2) Duecker, SAE-Paper 75 0223

The analog sensor signal is compared in a comparator with a fixed reference voltage. The comparator output provides information as to whether the value of the mixture is smaller (leaner) or greater (richer) than that of the reference voltage. Since it has proven worthwhile in practice not to equate the reference voltage with the theoretical $\lambda = 1$ point but rather with the most stable point for the characteristic (in our experience between 200 and 300 mV), electronic correction toward $\lambda = 1$ is required. This is done as shown in Fig. 3 by means of so-called lambda shifting. This action ensures that the integrator downstream of the comparator runs longer in the rich mixture direction thanks to a delay time t_V. On the other hand, information from the sensor about a lean mixture is transmitted without any delay to the integrator. The output voltage of the integrator is fed to another comparator, whose second input is connected to a saw-tooth generator with a frequency of 70 Hz. At the output of this second comparator there is thus a pulse sequence whose duty cycle is dependent on the magnitude of the integrator voltage. These pulses are fed via an amplification stage to a solenoid valve which, in the case of mechanical fuel injection, modifies the differential pressure in the fuel distributor, and hence the injected fuel quantity.

In addition to the electronic interconnections described, further actions are required in practice, such as sensor recognition, sensor monitoring, fullload enrichment and acceleration enrichment, but these will not be dealt with in more detail.

As regards coordinating a closed-loop system with oxygen sensor and fuel injection, the respective system dead time or the corresponding control frequency is generally of decisive significance. The dead time is made up of the delay time of the sensor, electronic unit and fuel-injection system, and of the gas passage (running) times through the engine and in the exhaust system as far as the sensor. While the delay times of the electronic unit and fuel metering system are negligible, the dead times are chiefly determined by the gas passage times and, at higher engine speeds, the response behavior of the oxygen sensor itself (Fig. 4). The engineer thus has possibilities of modifying the dead time, primarily in selecting where the oxygen sensor is to be installed. In doing so, other criteria must, of course, be borne in mind, such as operating temperature and starting behavior or cut-in of the control system.

Measurements have shown (Fig. 5) that the solid electrolyte generates an exploitable EMF at a sensor temperature of 250°C or more. If the starting temperature of the sensor is related to the exhaust gas temperature, the comparable EMF only occurs at about 400°C in thermally unsteady operation on account of the thermal capacity of the ceramic and the metal cage surrounding it. On the other hand, the heat stored keeps the sensor active for a longer period of time even down to lower exhaust gas temperatures. This hysteresis enables control in the low temperature range for brief periods.

As already explained at the beginning, the sensor signal is compared by control techniques with an appropriately fixed reference voltage. Since, however, the catalyst only reaches optimum conversion in a narrow mixture range, the sensor signal must lie extremely precise around this comparative value. The greater the slope of the sensor characteristic is in this range, the less will be the effect of a change in EMF on the air-fuel ratio. Fig. 6 shows the EMF scatter range (spread) for 69 sensors of one type. It can be seen that the scattering is relatively large and allows proper control only conditionally. The difference becomes far more serious when the sensors are subjected to aging. Fig. 7 contains the characteristics of a sensor when new and aged. It was aged thermally and mechanically on a vibration test stand using burner gas at 900°C. The characteristic of the sensor itself was measured at a gas temperature of 600°C. If, on the basis of the new oxygen sensor together with a downstream three-way catalyst, the optimum control voltage has been fixed at approx. 600 mV, there results after as little as 20 hours aging a reduction in the CO conversion rate from above 95 % initially to roughly 80 % with the same catalyst activities merely as a result of the change in the characteristic. Fig. 8 shows that this aging test is correlated by use in vehicles. Here the leveling of the sensor characteristic can been seen after 15 000 miles of continuous operation in comparison to the new sensor. The severe aging of the oxygen sensor under high pulsating stress, as illustrated in Figs. 7 and 8, is not so pronounced at a low sensor voltage, so for stability reasons a lower reference voltage with appropriately adapted λ shifting will yield a more stable control behavior in durability runs.

Aging of the oxygen sensor also has a deteriorating effect on starting behavior (Fig. 9). In addition to the sensor characteristic being displaced upward by roughly 25°C, the maximum EMF generated is considerably reduced.

Both changes, however, have only a limited effect on the control system if it is ensured in practical operation that the inlet temperature will not drop. It must, however, be expected that the control system will cut in later after could starting. As regards the choke for fuel enrichment versus temperature relationship for proper engine performance on cold start must be determined.

On the other hand, lower temperature activity of the oxygen sensor may also enable installation away from the engine. Irrespective of the starting temperature, the operating temperature should not be reduced excessively, even when the sensor is in good working condition. Measurements have shown that at an operating temperature of less than 400°C, the soot that has accumulated on the ceramic or external electrode during the cold-starting phase or any unburned fuel particles cannot be removed or burned off and may result in a signal being given by the sensor. In this case the sensor continuously signals a shortage of oxygen, regardless of the actual mixture composition, leading to a lean correction or a so-called "lean drift" of the control system.

By way of contrast, the changes occurring during customary aging lead to a distinct control system drift towards the "rich" zone. This can be explained, on the one hand, by the considerable leveling above 400 mV - 500 mV the sensor characteristic (as shown in Figs. 7 and 8). On the other hand, the "rich drift" is also determined by the changing response time. The response time is the time which elapses before the sensor records a change in the mixture composition at the sensor or how quickly the sensor moves from "rich" to "lean", or vice versa (Fig. 10). A uniform increase in the response time for both directions, i.e. "rich-lean" and "lean-rich", does not give rise to any change in control behavior. Aging investigations showed, however, that the response time form "lean" to "rich" is subject to severe deterioration. The result is that the sensor records a lean mixture for a longer period of time and in this way displaces the control system into the "rich" zone.

The changes in the sensor characteristic that have occured to date in practical continuous operation clearly point to a so-called "rich-drift". However, the magnitude of this drift is not the same for all vehicle types and is highly dependent on the respective load in road tests. The deviation is, however, so great that even with the "wandering" of the sensor characteristic proper control with the appropriate optimum pollutant reduction is not guaranteed. It is to be noted that requirements in terms of control accuracy and corresponding efficiency of the catalysts are far higher in vehicles with high volume exhaust emission than in comparable smaller vehicles.

To achieve proper and adequate emission control by way of the oxygen sensor and three-way catalyst, several trade-offs are to be made in the interrelationship between the control system and catalyst and must be kept within narrow limits. The first trade-off in the closed-loop system is assignment of the sensor control voltage to the air-fuel mixture, as described in the previous section. The second trade-off is assignment of a specific amount of fuel to a control signal in such a way that an air-fuel ratio with a narrow tolerance around stoichiometric is available at the catalyst, irrespective of the operating state of the engine.

Observance of this so-called "lambda window" at the catalyst is an important prerequisite for proper emission control of all three pollutants. A further prerequisite gains in importance above all when attaining high degrees of control. This is the constant "oscillation" of the air-fuel mixture in the above-mentioned lambda window around the stoichiometric point. With uniform alternation between surplus and shortage of oxygen, the so-called oxygen storage of three-way catalysts has an effect. It is thus ensured that in the "rich" mixture, in addition to the residual oxygen, further stored oxygen is made available by the catalyst for oxidizing the unburned components which, assuming equal conversion rates, practically leads to an expansion in the operating range in the "rich" direction. On the other hand, carbon monoxide is also stored by the catalyst in the event of a shortage of oxygen and "transported to the lean zone", where it contributes slightly to the production of NO_x even with a surplus of oxygen. This in turn leads to an albeit lesser expansion of the operating range in the "lean" direction. Fig. 11 illustrates the differences in the "lambda window" for dynamic and static measurement for a conversion rate of at least 90 %. At the same time it can be recognized that the operating range limits in the lean zone and rich zone are determined by NO_x and CO conversion respectively. A major influence on the oxygen/carbon monoxide storage by the catalysts is their dwell time in the individual gas phases. Measurements have shown that the three-way catalysts with a Pt/Rh base available at present exhibit optimum storage capacity at a control frequency of 1 to 2 Hz. Since the control frequency corresponds to the system dead time, the frequency can be readily varied only within narrow limits (chiefly the position of the sensor). This can, however, lead to other catalyst configurations appearing optimal for underbody systems with regard to oxygen storage capacity as compared with systems close to the engine.

Oxygen storage of three-way catalysts takes place as a synergetic effect of contact material and components in the wash coat. After chemisorption the oxygen is split and stored in atomic form or as oxygen radicals on the precious metal.

This process is assisted by special promoters, e.g. special metal oxides, and stabilized by other substances in the wash coat, e.g. cerium oxide. By varying the appropriate promoters and stabilizers, the catalyst industry is endeavoring to further improve the dynamic behavior of the catalysts by way of this storage capacity. This can also mean improving the efficiency of the catalysts generally by adding less expensive basic substances or maintaining their efficiency, simultaneously reducing the quantities of more expensive precious metals.

Measurements with aged catalysts have shown (Fig. 12) that to achieve an appropriate conversion rate, a minimum rhodium and platinum content cannot be dropped below, a total precious metal quantity of 50 g/cft for roughly 20 % rhodium yielding a reasonable compromise between price and effect. A rhodium content of approx. 5 % corresponding to its natural exploitation does not, however, appear at present to be adequate, especially for vehicles which require an extremely high conversion rate on account of their high mass emission.

The mixture control system is ready for operation roughly 15 to 30 seconds after a cold start depending on the position of the oxygen sensor in the CVS test. This means that the catalysts must likewise exhibit a reasonable conversion rate after roughly the same time. Consequently, the starting behavior of three-way catalysts is equally as important as that of oxidizing catalysts (Fig. 13). Optimization must, however, relate chiefly to the starting behavior in the oxygen shortage range since in most applications additional air is not used for the starting phase. Fig. 14 shows the starting behavior for NO_x assuming an operating point of $\lambda = 0.95$. The values correspond roughly to the starting temperatures of the oxygen sensors.

In the evaluation of three-way catalysts their oxygen storage capacity is of decisive significance. It is, therefore, important when measuring also to consider the respective control behavior of the system. Testing the catalysts on the actual engine for the subsequent application on the basis of position and control behavior provides more information than when testing with synthetic gas.

Figs. 15 and 16 show the comparison between two different three-way catalysts measured on a V-8 engine under the same conditions. In addition to the standard oxygen sensor for control purposes, this engine has two further oxygen sensors upstream and downstream of the catalyst as pure measuring sensors. By upsetting the control system it is ensured that an exhaust gas mixture that is slightly different in its composition is applied to the catalyst, whereby it is checked by the sensor signal of the oxygen sensor upstream of the catalyst whether the control system constantly maintains the mixture between rich and lean. The scatter range of this measurement signal reveals that this is guaranteed at a signal span of roughly 100 to 850 mV, a slight tendency towards "leaning" generally being achieved by modifying the control system.

The emission of untreated exhaust gas and the conversion rate of both catalyst groups differ only slightly, while major differences can be determined in the sensor signal downstream of the catalyst. The scatter of catalyst type A is extremely slight over its entire range, i.e. both in the slightly rich field at high control frequency and in the slightly lean field at low control frequency. This means that this catalyst can fully perform oxygen exchange at high control speed. The storage capacity of catalyst A is quite adequate in terms of both the quantity stored and storage speed, i.e. reaction rate in chemisorption.

With the second catalyst B, however, oxygen particles also accur in the exhaust gas in the high control speed range downstream of the catalyst; this can be seen from the wide spread of the sensor signal. The oxygen periodically supplied in the exhaust gas to the catalyst cannot be completely stored by the catalyst and thus supplied for conversion. This can also be seen in the slightly worse conversion rates of this catalyst. Additional lab tests have shown that the catalyst B has a high storage capacity but also long dwell times, i.e. the ability of this catalyst to absorb and release the oxygen as quickly as possible is not yet sufficiently pronounced.

Adapting an oxygen sensor control system to a V-engine is highly dependent on the design of the vehicle engine. Fig. 17 shows the differing adaption variations with different exhaust routing for a V-engine with continuous fuel injection. The most favorable possibility for adaption is the central installation of the sensor where the individual exhaust pipes meet. The advantage of this is that the exhaust gas of all 6 oder 8 cylinders is detected by the sensor and can thus be controlled. However, such application presupposes that a temperature of approx. 400°C required for the sensor to ensure proper control is not dropped below during operation at the installation site or that the starting temperature of the sensor is reached after a reasonable time in terms of exhaust emission. In this case the cost of the complete system in controlling a V-engine is comparable with that for an in-line engine. Additional measures required for the controlled engine are the installation of an oxygen sensor, an electronic control unit, and modification to the fuel metering system. The additional possibilities outlined at the beginning, such as acceleration and full-load enrichment, do of course require additional consideration, which tends to increase the cost involved.

Modulation of fuel metering is achieved in continuous injection (K-Jetronic) by changing the fuel differential pressure across fuel metering slots in the fuel distributor of the K-Jetronic system thus affecting fuel delivery. By modulating only. This means, however, that during control by the sensor close to the engine, measurement is only made of the oxygen content of the exhaust gas of one cylinder bank. As this additional control during the CVS test takes effect within only roughly 50 seconds, it is a matter of effectiveness as to the extent to which this additional expenditure is justified. This decision is highly dependent on the cold-start emission of the design at issue or on the improvement possible with the second oxygen sensor. With an appropriate electronic interconnection this additional oxygen sensor can also be used to assist the control system at operating points of lower exhaust gas temperatures. Reductions in the accuracy of the central system may have to be accepted for cost or V-8 engine adaptation reasons. For instance, in the case of a V-8 engine exhaust pipe configuration where the y-connection of one pipe is close to the engine and the other is quite removed from the engine, the O_2-sensor has to be located in the highest temperature pipe, and since underbody catalyst may have to be used, these factors add up to a loss in accuracy of control and less efficient catalytic conversion. Usually there is a deviation between the mean air fuel ratio in each of the two cylinder banks. Then, if the exhaust gas of the "richer" cylinder bank is used for control purposes, this means that the catalyst receives an altogether lean mixture, so that then the oxidizing pollutants are well converted, but nitrogen oxides converted only slightly. The behavior is reversed when the "lean" cylinder bank is used for control.

the differential pressure a $\pm$ 20 % change in the air-fuel mixture is possible.

This range is sufficient to level out fluctuations in the intake air and intake pressure (altitude correction) to $\lambda = 1$, in addition to the airfuel mixtures differing from cylinder to cylinder, cylinder bank to cylinder bank and engine to engine.

Further measures are required to achieve proper control with a vehicle/engine design in which the exhaust pipes can only meet away from the engine, i.e. in a lower temperature area. If the temperature of the area where the exhaust pipes meet is high enough to install the oxygen sensor with the engine at operating temperature, these measures are required during the cold-starting phase only. The oxygen sensor can, for instance,be heated externally during this phase so as to obtain a usable sensor signal as quickly as possible, i.e. after approx. 20 seconds, if the engine can be operated with a stoichiometric mixture. It must, however, be borne in mind that during the cold-starting phase heat is constantly removed from the sensor by the cold exhaust gas, so an appropriate quantity of heat must be provided. Tests conducted to date have not yielded satisfactory results. One measure that can currently be applied, however, is the installation of an additional sensor close to the engine, whose signal is used for control purposes during the cold-starting phase

Fig. 18 shows the differences in exhaust emissions for the various types of control and with simultaneous variation of the reference voltage.

When a one-side mono control system is applied, exhaust control efficiency can be improved by using a downstream oxidizing catalyst. In this case the control system is trimmed slightly into the "rich" zone and the ensuing lower conversion of carbon monoxide and hydrocarbons in the first catalyst is compensated in the downstream oxidizing catalyst with air intake or air injection. A downstream oxidation stage is vital for designing a V-engine with separate exhaust pipes and catalysts when a one-side mono control system is employed. This leads to a controlled dual-bed catalyst system. As a result greater requirements in terms of accuracy are to be met by the mixture tolerance of both cylinder banks and the basic engine setting, coupled with simultaneous cuts in degrees of emission control and fuel economy. Measurements have shown that with downstream oxidizing catalysts, however, the conversion of nitrogen oxides is simultaneously reduced by up to 20 %. This is particularly serious when the actual three-way catalyst and oxidizing catalyst are not physically separated and the air must be fed in the narrow area between the two catalysts. In this case the result is back diffusion of the oxygen into the three-way catalyst, and hence oxygen oversaturation of the catalyst in certain areas bordering on the air feed zone, which leads to a reduction in the nitrogen oxide conversion rate. On the other hand, physical separation of the catalysts results in a greater temperature drop upstream of the oxidizing catalyst, which in turn may rule out proper conversion of the oxidizing substances. Furthermore, the ammonia produced in the three-way catalyst is oxidized in a downstream oxidizing catalyst to form nitrogen oxide.

The cost involved in controlling a V-engine increases sharply when design or thermal reasons make it impossible to install a central oxygen sensor and a downstream single-flow catalyst. In this case a V-8 engine has to be split into two L-4 engines insofar as control is involved. Each cylinder bank is then to be considered separately from the control technology angle. This means that for each cylinder line there is one oxygen sensor, one electronic closed-loop system and one fuel distribution control system acting on a single cylinder bank. In addition the cost of separate fuel distribution control systems increases further since the design of the fuel distributor as a control element for fuel metering has to be revised, and the existing control system cannot be used. The efficiency of such a stereo control system is comparable with control using a central oxygen sensor.

The variations for adapting the oxygen sensor control system are shown once again in Fig. 19.

Results of durability runs with the one-side mono control system show that mileage-dependent changes in the mixture composition can be leveled out only to a limited degree. Table 1 lists the results obtained with a vehicle powered by a V-8 engine (V_E = 4.5 l, V_{cat} = 3.0 l, 4000 lbs) before and after an endurance run of 40 000 kilometers. During this test, however, a subsequent oxidation stage was not used.

The oxygen sensors were changed after the 15 000 miles guaranteed by the manufacturer. All CVS tests listed were conducted with new sensors. By way of comparison the same endurance run vehicle was then measured with a central oxygen sensor (control of all cylinders). The relatively severe deterioration in the emission values, even in the case of central control, is probably due above all to the low catalyst volume, and hence the high space velocity.

In Fig. 20 various catalyst volumes have been measured on the same engine type under steady conditions on the engine test bench with regard to their conversion with a central oxygen sensor control system as a function of the reference voltage (as the control variable). It can be seen that a minimum catalyst volume is required, in particular for the conversion of carbon monoxide at the reference value, normally to be set, of 600 mV without lambda shifting, i.e. $\lambda = 1$. As already mentioned when dealing with the three-way catalyst, CO conversion forms the limit for the control tolerance in the "rich" mixture range. In the oxygen shortage field, however, CO conversion can only be achieved by improving the oxygen storage capacity of the catalyst. On account of this relationship it can be concluded that the space velocities for three-way catalyst, which are lower than for the oxidizing catalyst, are determined by the dwell times required for oxygen storage.

Oxygen storage can also be increased by adapting the control characteristics. For this purpose the control unit is optimized such that a reasonably long dwell is achieved in the oxygen surplus phase. This optimization possibility is in direct relation to the compromise to be made from the control frequency and control amplitude, both steady and transient operation having to be considered. As already explained, however, design and control requirements in principle limit freedom when it comes to specifying the control frequency.

Fig. 21 shows the dependence of exhaust emission on the control speed for the steady operating point of a centrally controlled V-8 engine. A low control time constant signifies a high control speed, and vice versa. It can be seen that at a low control speed (high control time constant) the oxygen storage is increased and as a result CO conversion is improved in the "rich" range and NO_x conversion reduced. Although the control system is on average around $\lambda = 1$, regardless of the control speed, the differing efficiency of oxygen storage at low control speeds means that the effect of displacement into the "lean" zone has practically been achieved. It must, however, be stressed that a low control speed yields optimum values for steady operating points only.

In contrast to the "quasi drift" into the "lean" zone, which can be obtained through the low control speed due to the catalyst oxygen storage capacity, the mean value of the control system is displaced by so-called lambda shifting into the oxygen shortage range. This displacement is primarily used, as already explained, to correct the air-fuel mixture to $\lambda = 1$ for a reference value of lambda > 1 due to asymmetric control characteristics.

Fig. 22 shows the dependence of exhaust emission on lambda shifting at the steady operating point of a V-8 engine and in Fig. 23 as mass emission in the CVS test (3rd bag). The behavior to be expected shows that as lambda shifting increases, i.e. enrichment increases slowly, NO_x conversion improves while CO conversion deteriorates.

As already explained when outlining the three-way catalyst, the HC emission from controlled engines is least sensitive to changes in the control characteristics. This can also be seen above all in Fig. 23.

In addition to limited exhaust emissions, secondary emissions, so-called "unregulated pollution", are of interest. As already illustrated in Fig.1, the substances currently under discussion, such as hydrogen cyanide, sulfates and ammonia, are of subordinate importance in terms of volume emission in the case of proper control. All these values are at a minimum when $\lambda = 1$. Even with a deliberately occasioned malfunctioning of the system, the values of hydrogen cyanide, for instance, are below the internal standard issued by theEPA. Regardless of this, there is the possibility of detecting electronically certain irregularities or malfunction within the closed loop which cause unregulated emissions for instance and correcting them. Errors that cannot be corrected may be indicated by an alarm so that possible secondary emissions occur only briefly and have no environmental impact.

Regardless of the above-listed problems involved in adapting the oxygen sensor control systems to V-engines, the best efficiency can be achieved with this emission control concept. Since, in contrast to other control systems, fuel economy also exhibits near optimum values in operation at $\lambda = 1$, it is predicated that this system will be used in production over a broad range of vehicles. However, certain requirements are still to be met as regards improving the system. In vehicles with high mass emission the good emission values refer exclusively to low-mileage values. The high volumetric degrees of control required for these vehicles are also to be maintained over higher mileages, but for at least 50 000 miles.

For this purpose on oxygen sensor characteristic with narrower tolerances, a low response time uniform throughout continuous operation, and hence a far lower sensor drift are among the requirements.

Furthermore, the oxygen storage capacity of the catalysts is to be improved not only in terms of quantity but also reaction rate. At the same time endeavors to reduce the high rhodium content of these catalysts required to date with an improved conversion rate and to reduce stability in continuous operation by using lower-grade materials are to be stepped up. The interaction of the control system and catalyst is further to be analyzed and optimized.

Summary

1. The oxygen sensor control system, in conjunction with three-way catalysts even for light-duty vehicles powered by V-engines, constitutes the emission control system with which the exhaust emission values specified by the Clean Air Act can best be met.

2. A prerequisite for a high degree of emission control is that the exhaust gas of all cylinders should be detected using control techniques. This is easiest achieved by combining the exhaust pipes and a central oxygen sensor. Use of a stereo control system with which each cylinder bank is controlled independently involves increased cost.

3. In view of the high mass emission of these vehicles the scattering of the control system is to be reduced and the efficiency of the catalysts improved.

4. The stability of the control system and catalyst is to be substantially improved in continuous operation if future exhaust limits are to be complied with.

LIST OF FIGURES

EMISSION CONTROL OF V-ENGINES WITH CATALYSTS AND OXYGEN-SENSOR

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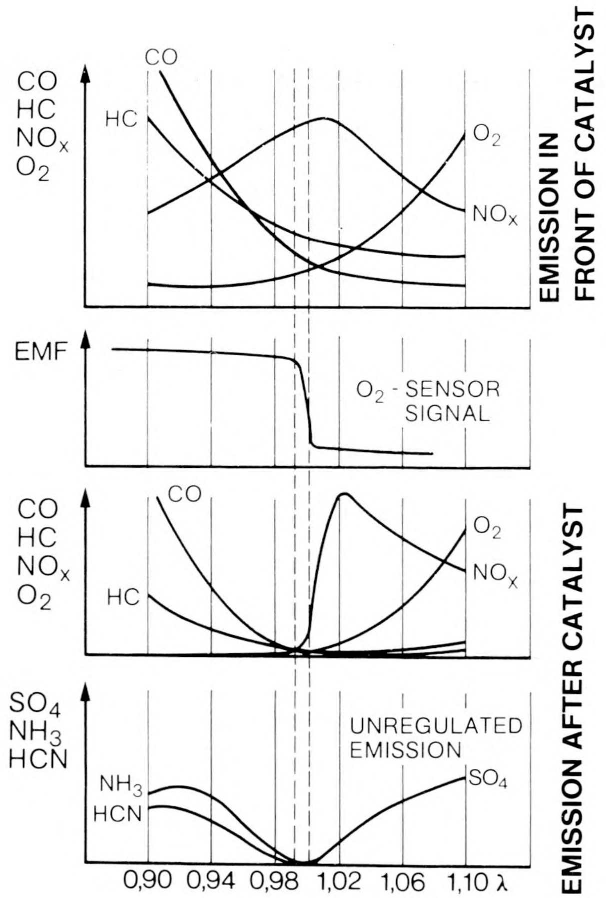
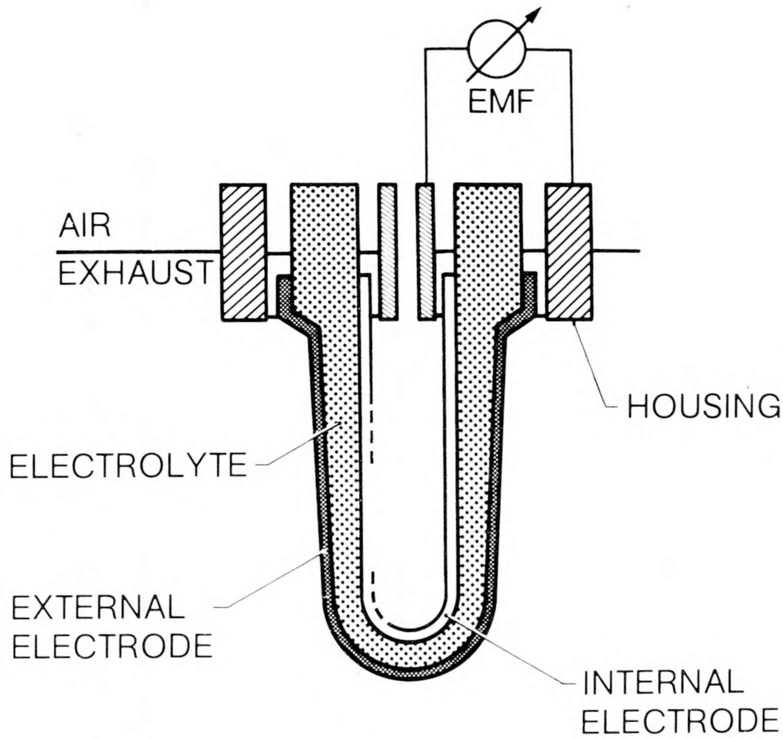


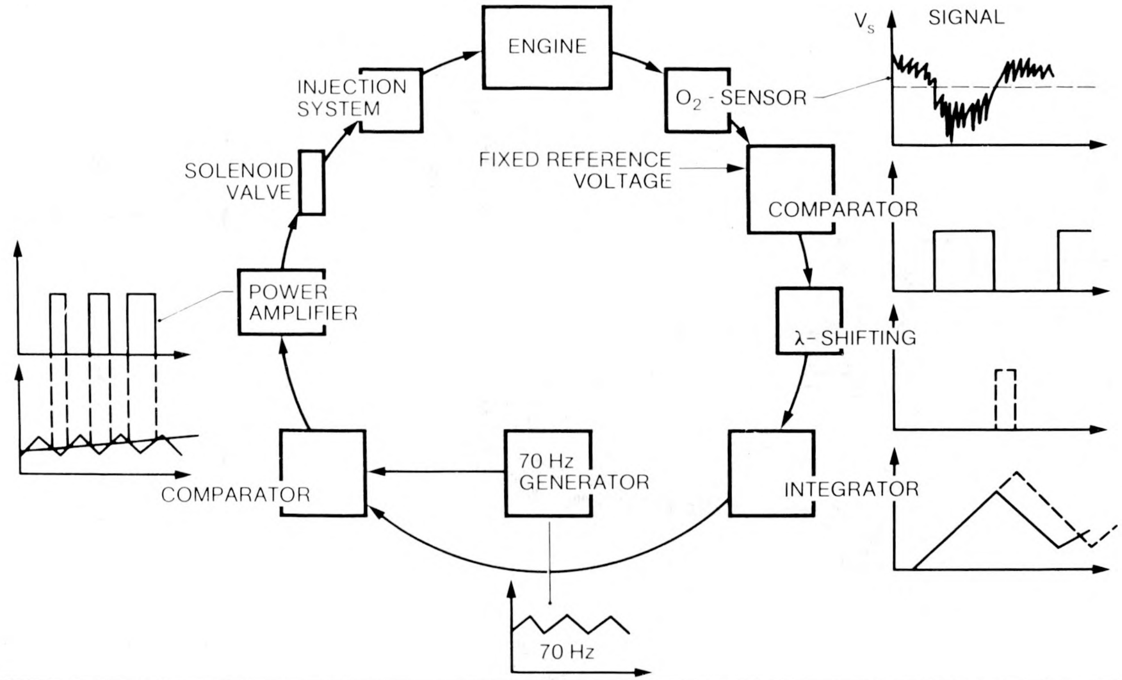
	EMISSION CONTROL WITH THREE WAY CATALYST	V1MK 1276 07 104

Fig 1



	O_2 - SENSOR	**V1MK** 1276 07 108

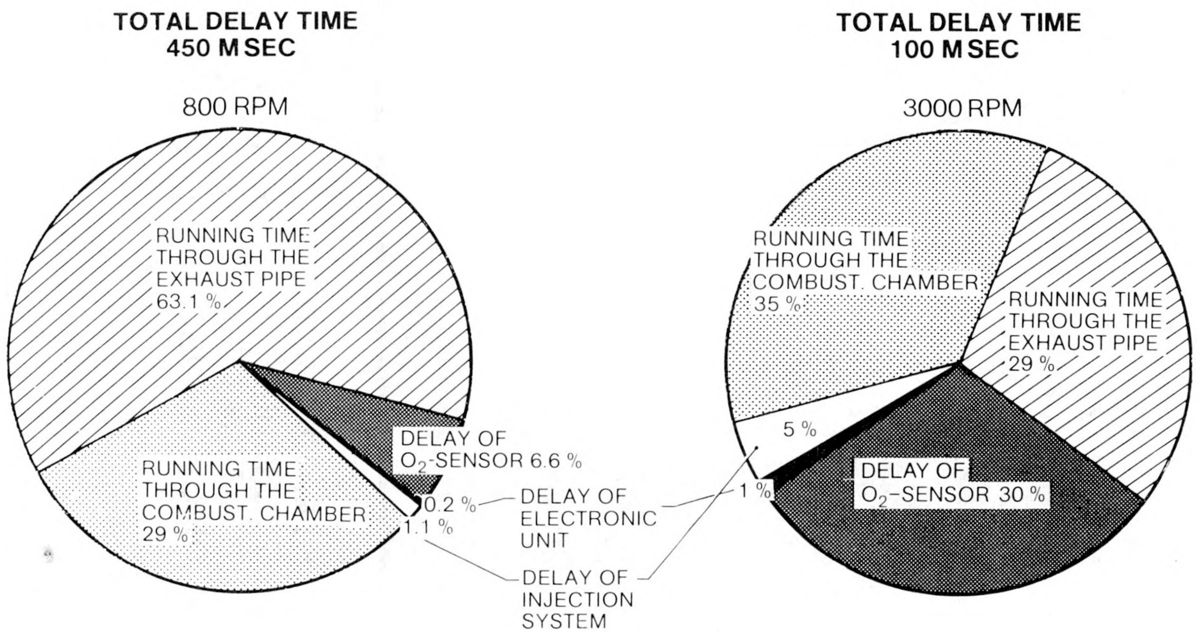
7.2



ELECTRONIC CLOSED LOOP SYSTEM

V1MK 1276
07 105

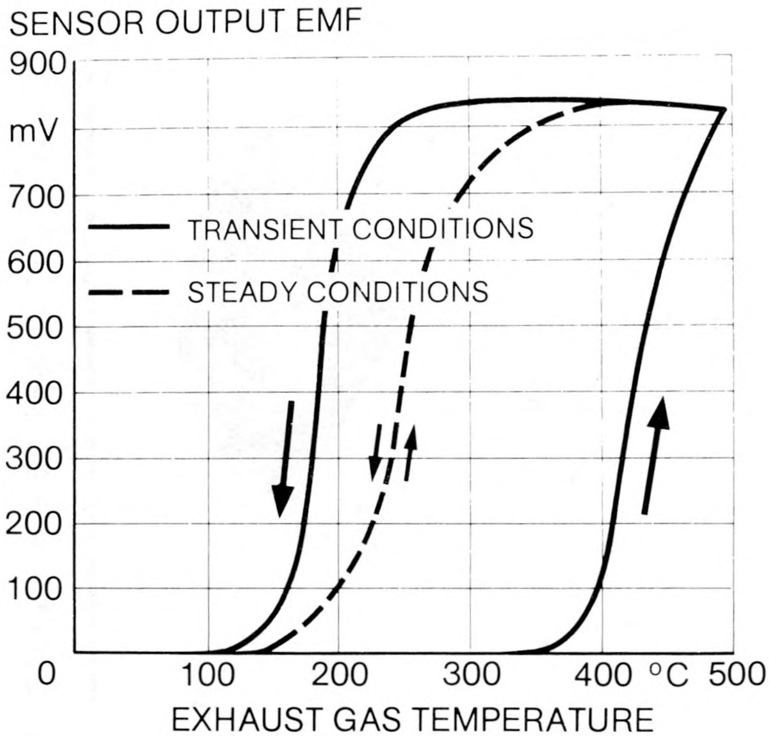
Fig 3



DELAY TIME OF A CLOSED LOOP SYSTEM

V1MK 1276
07 106

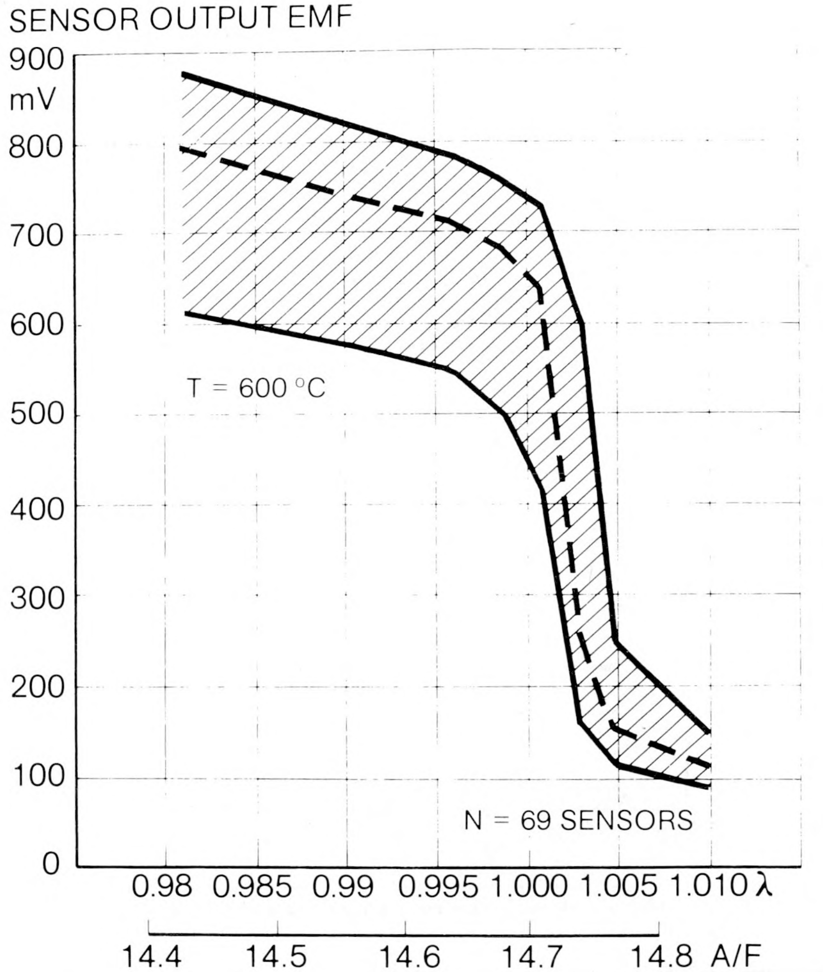
Fig 4



STARTING AND QUENCHING TEMPERATURE OF O_2 - SENSOR

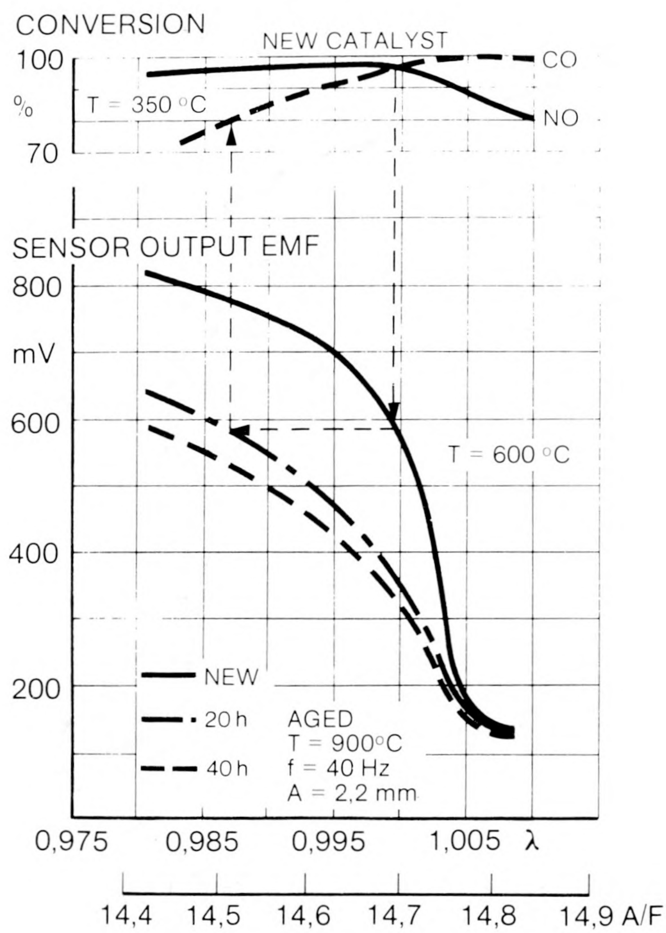
V1MK 1276
07 113

Fig 5



SPREAD OF
O_2 - SENSOR
SIGNAL

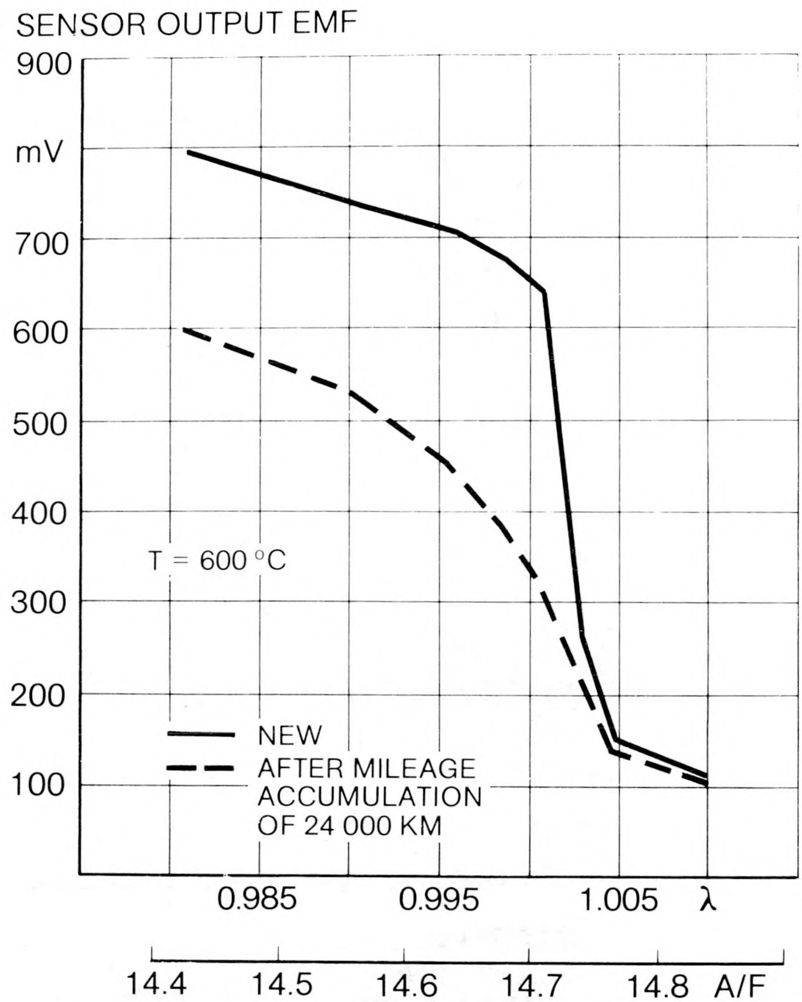
V1MK 1276
07 110



O_2-SENSOR CHARACTERISTIC AFTER AGING ON A VIBRATION TEST STAND

V1MK 1276
07 112

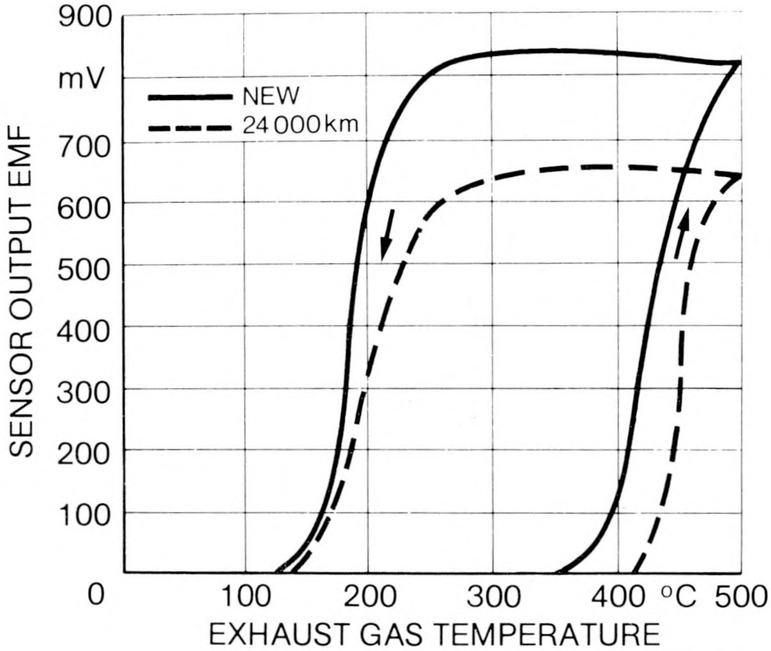
Fig 7



O_2 - SENSOR CHARACTERISTIC AFTER MILEAGE ACCUMULATION

V1MK 1276 07 111

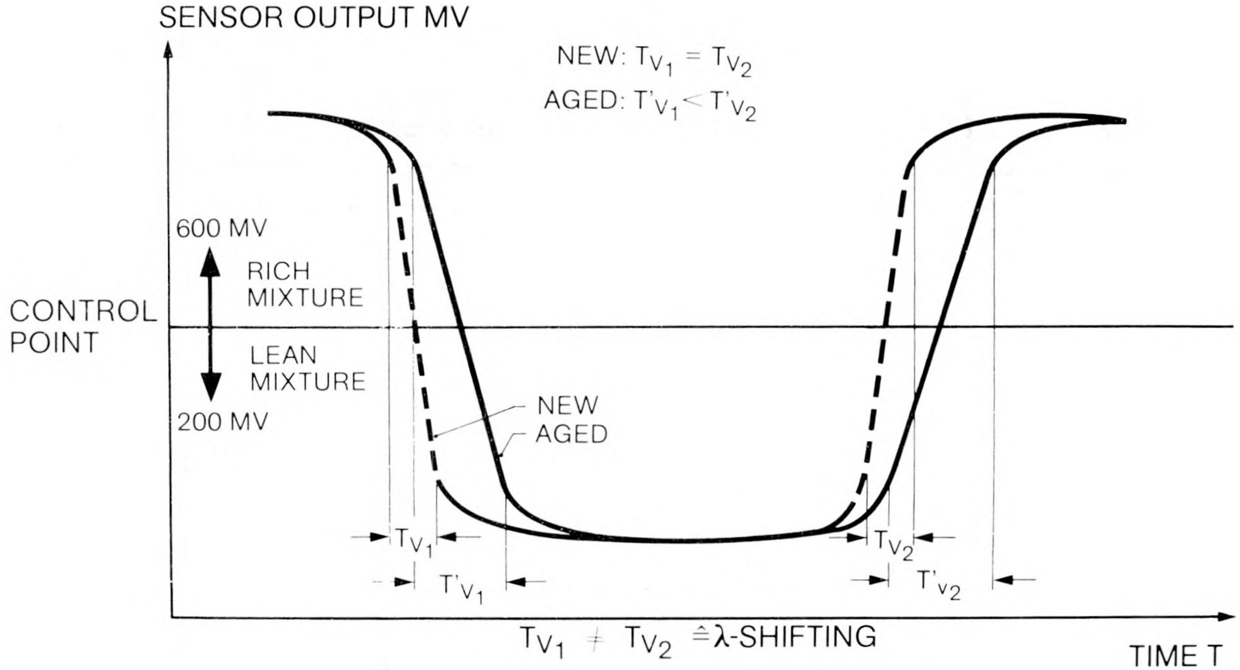
Fig 8



STARTING AND QUENCHING TEMPERATURE AFTER MILEAGE ACCUMULATION

V1MK 1276
07 114

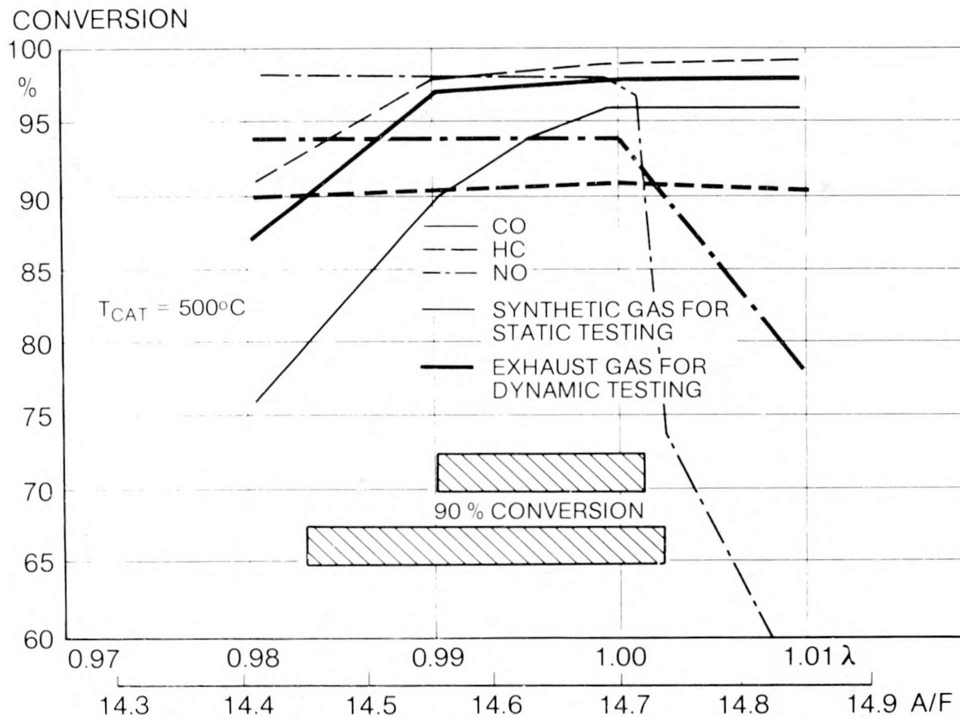
Fig 9



RESPONSE TIME OF O_2 – SENSOR

V1MK 1276
07 123

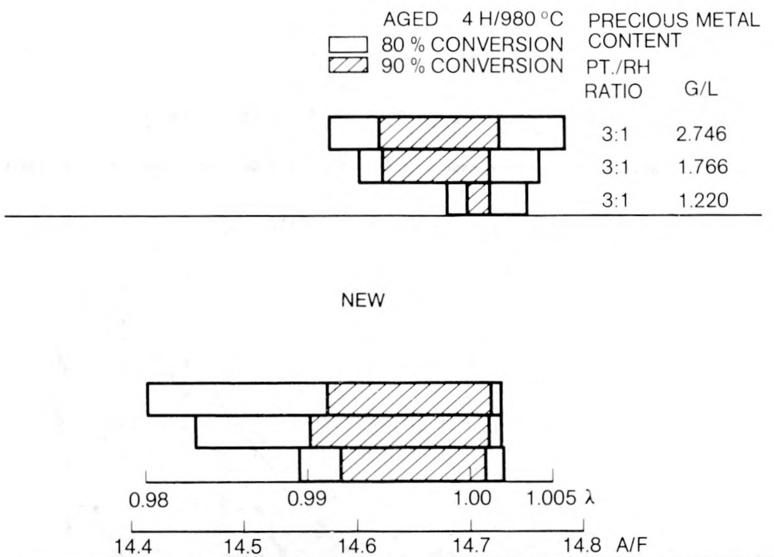
Fig 40



COMPARISON OF STATIC AND DYNAMIC TESTING OF THREE-WAY-CATALYSTS

V1MK 1276
07 116

Fig 11



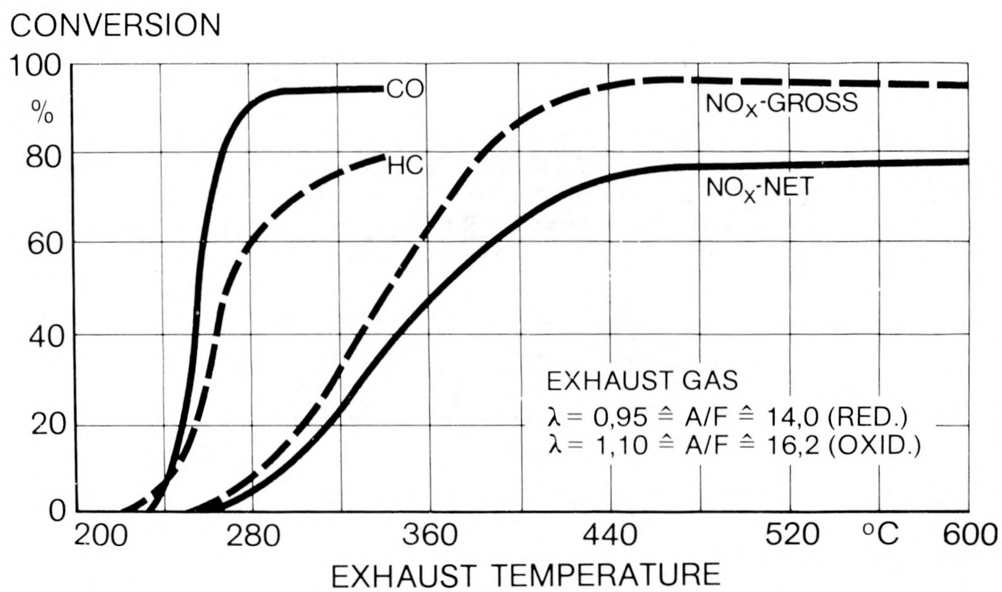
INFLUENCE OF PRECIOUS METAL CONTENT ON CONVERSION RATE MEASURED WITH SYNTHETIC GAS (STEADY CONDITION)

V1MK 1276
07 115

TG16508

18. 1. 77 Dom

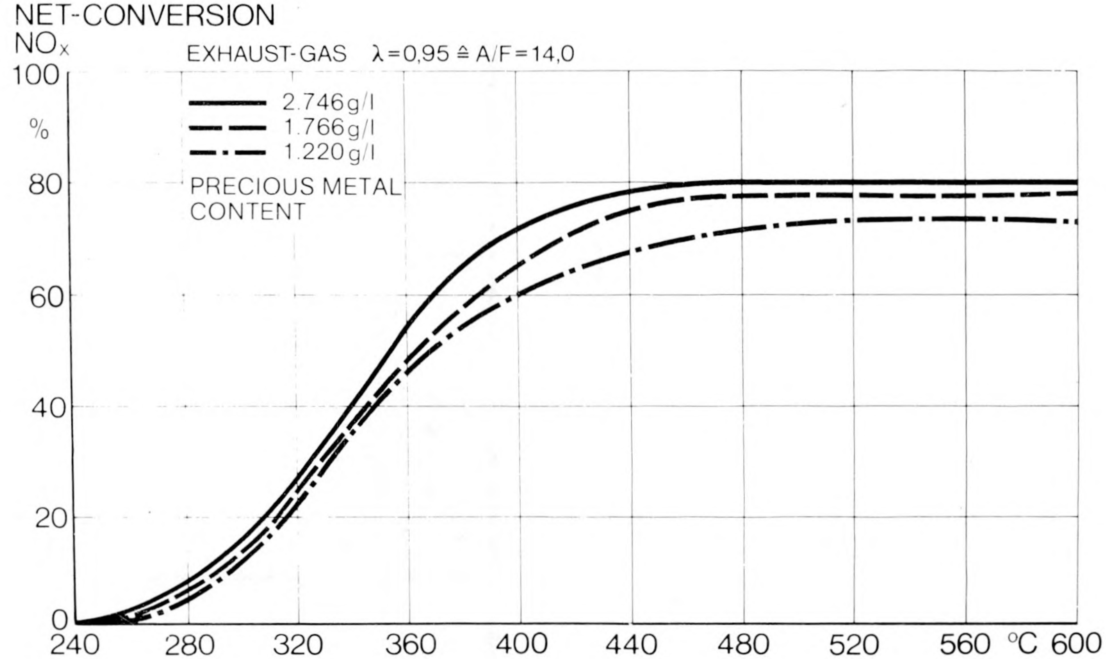
Fig 12



LIGHT OFF TEMPERATURE OF THREE-WAY-CATALYSTS

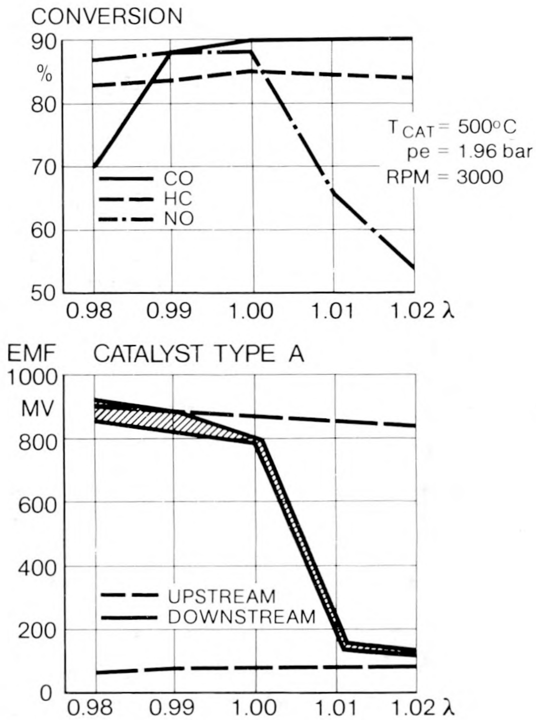
V1MK 1276
07 117

Fig 13



LIGHT OFF TEMPERATURE FOR NO_X-CONVERSION WITH DIFFERENT PRECIOUS METAL CONTENT

V1MK 1276
07 177

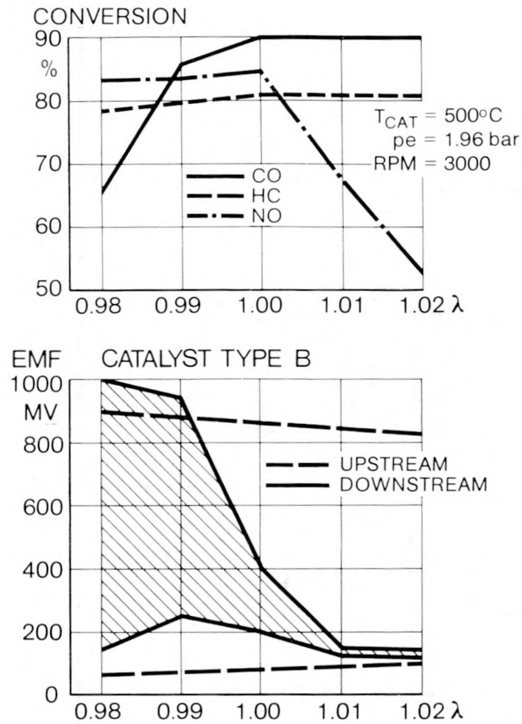


CATALYTIC CONVERSION WITH CLOSED LOOP OPERATION ON AN ENGINE DYNO

V1MK 1276
07 121

18.1.77 Dom

Fig 15



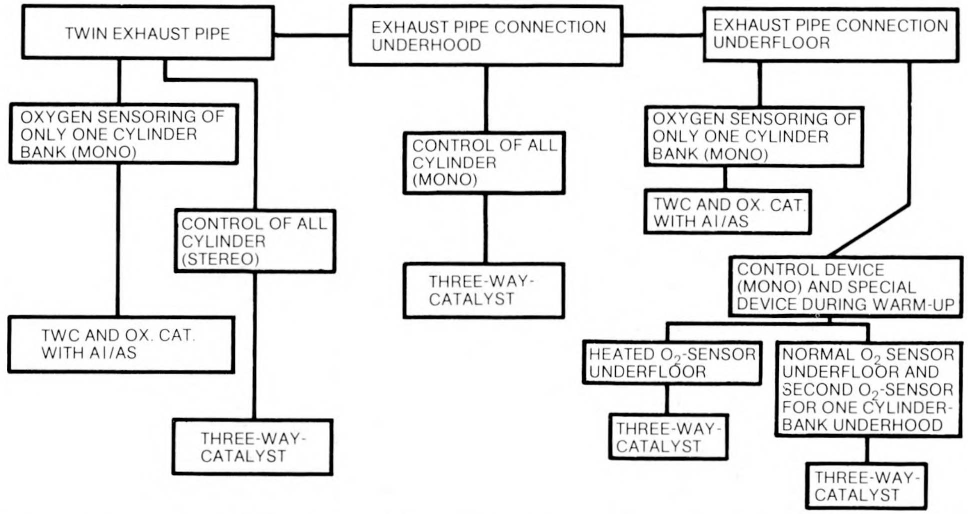
CATALYTIC CONVERSION WITH CLOSED LOOP OPERATION ON AN ENGINE DYNO

V1MK 1276
07 122

18.1.77 Dom

TG 16509

Fig 16



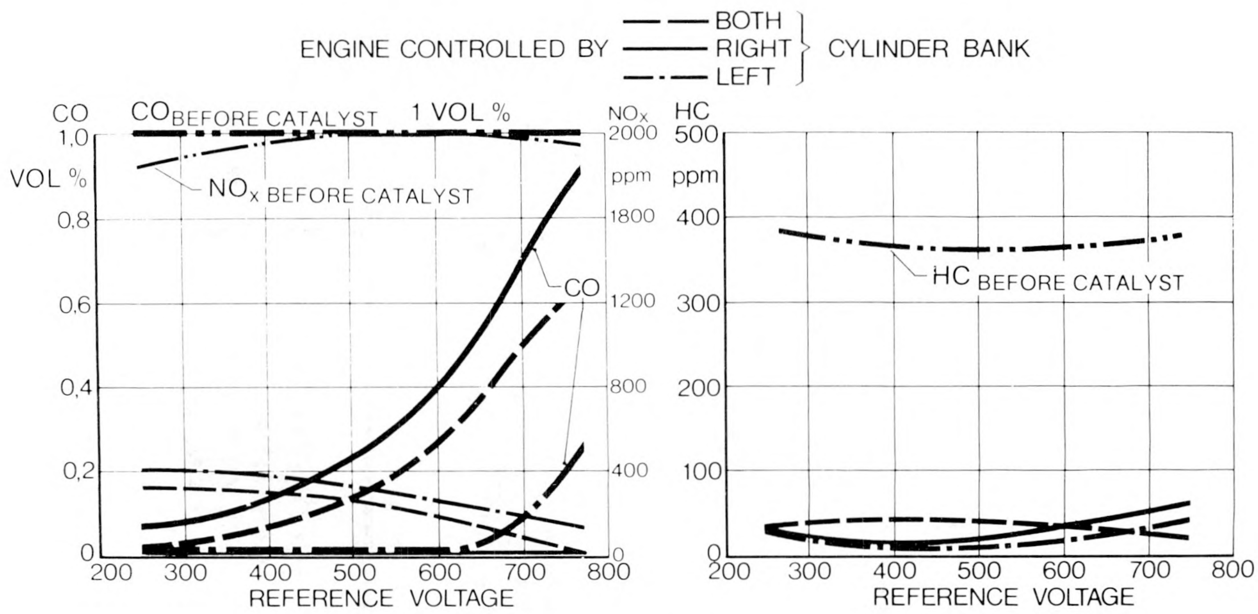
O_2-SENSOR CONTROL DEVICE FOR V-ENGINES

V1MK 1276
07 127

TG16508

18.1.77 Dom

Fig 17

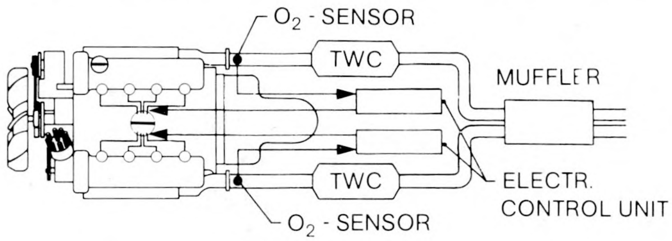


EXHAUST EMISSION OF A V-8-ENGINE WITH DIFFERENT CLOSED LOOP SYSTEMS

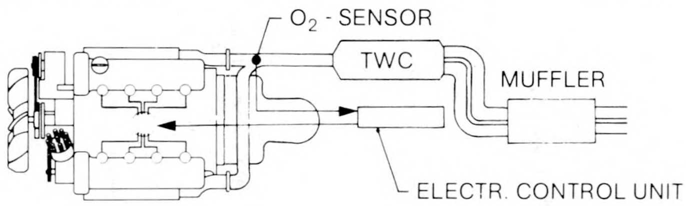
V1MK 1276
07 119

'45 18

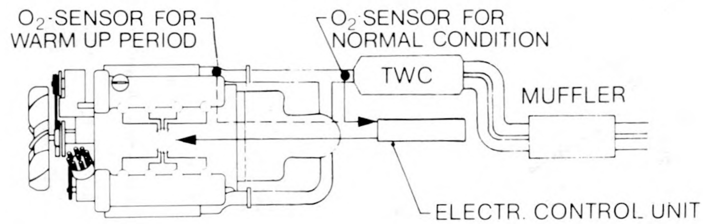
2 O_2 - SENSORS | 2 CONTROL LOOPS | STEREO



1 O_2 - SENSOR | 1 CONTROL LOOP | MONO



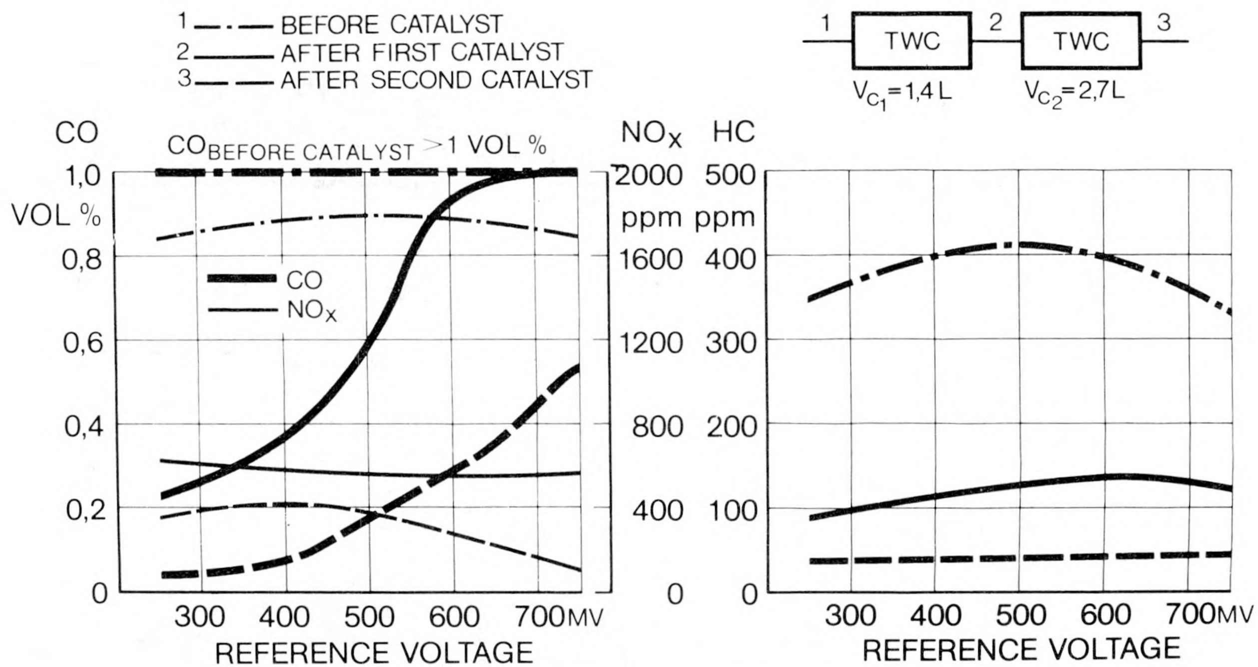
2 O_2 - SENORS | 1 CONTROL LOOP | MONO



CLOSED LOOP CONTROL DEVICES FOR A V-ENGINE

V1MK 1276
07 109

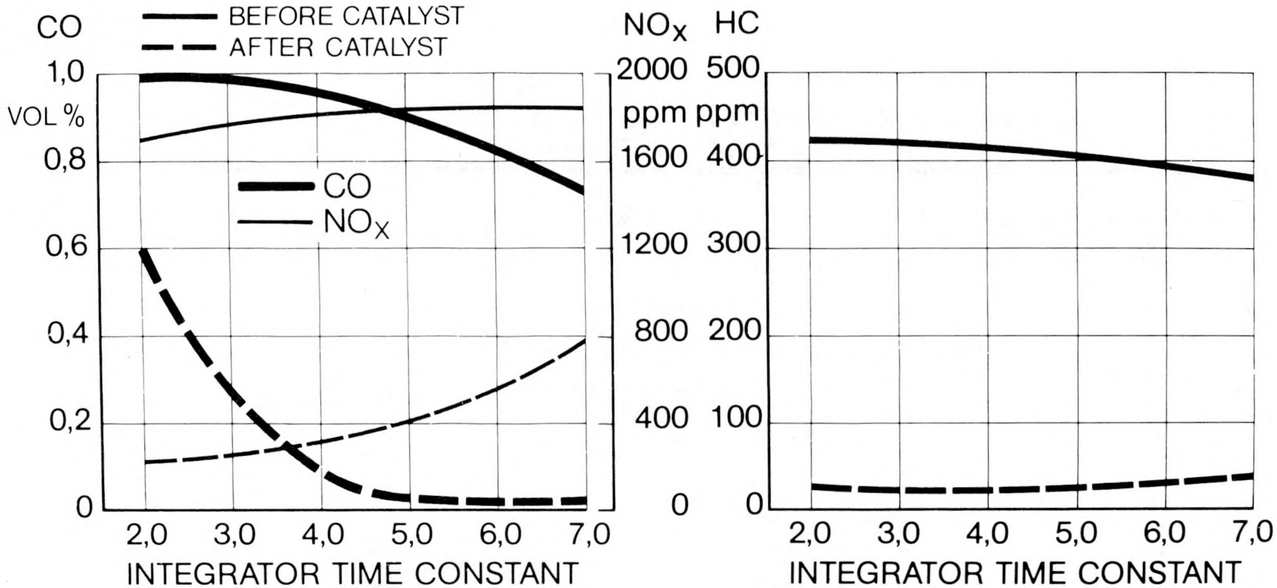
Fig 19



INFLUENCE OF CATALYST VOLUME ON THE EMISSION OF A CLOSED LOOP CONTROLLED V-8-ENGINE

V1MK 1276
07 120

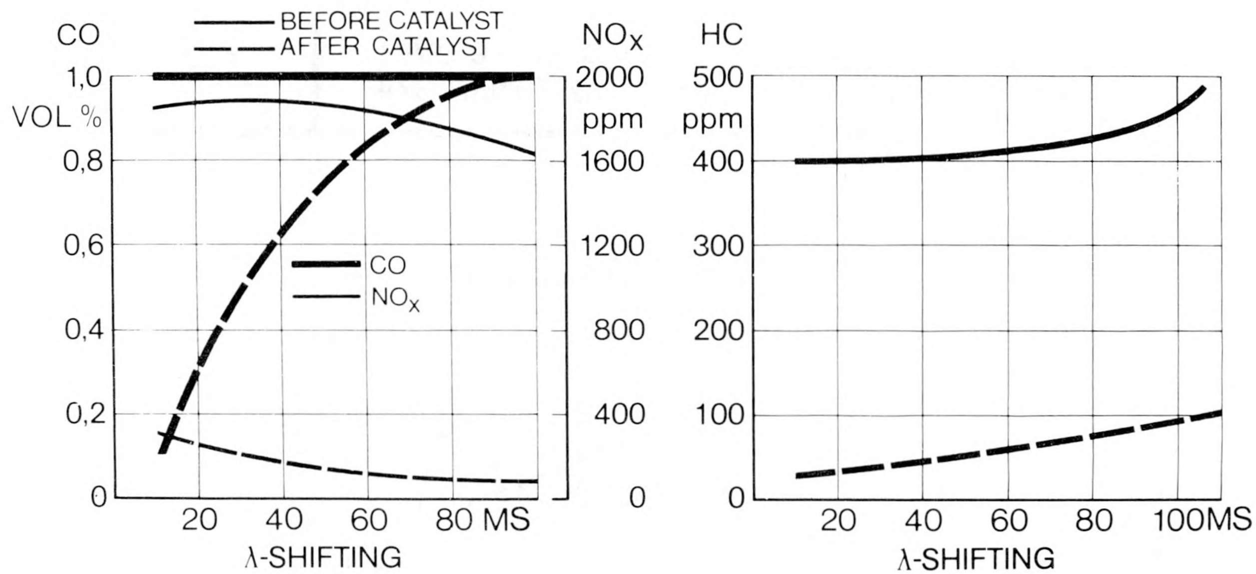
Fig 20



INFLUENCE OF INTEGRATOR TIME ON THE EMISSION OF A CLOSED LOOP CONTROLLED V-8-ENGINE

V1MK 1276
07 125

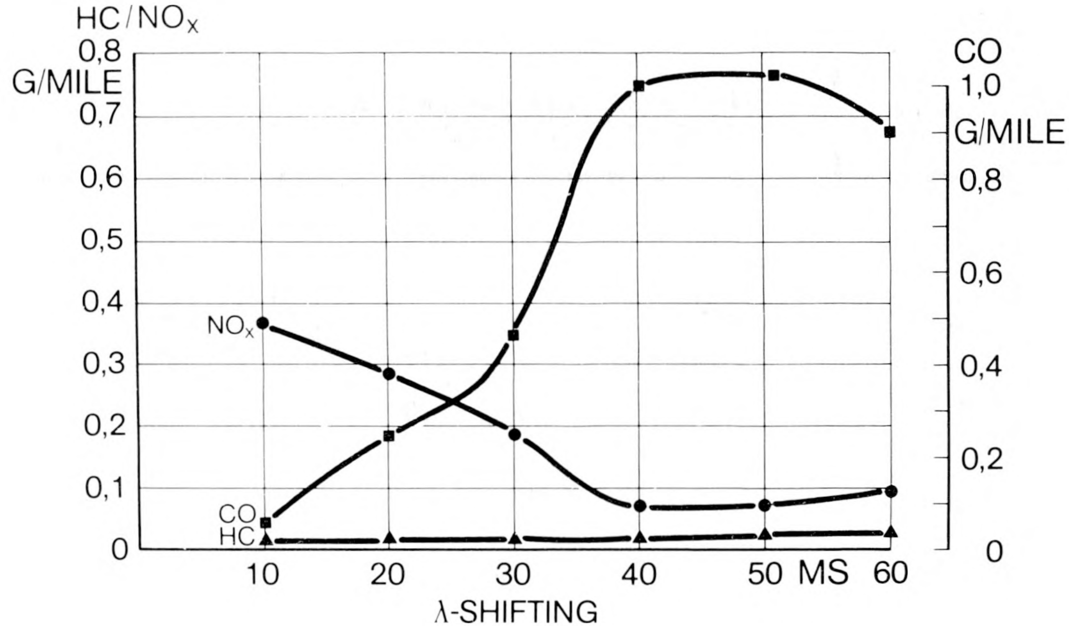
Fig 21



INFLUENCE OF λ-SHIFTING ON THE EMISSION OF A CLOSED LOOP CONTROLLED V-8-ENGINE

V1MK 1276
07 118

Fig 22



INFLUENCE OF λ-SHIFTING ON THE EMISSION (BAG 3 CVS-TEST)

V1MK 1276
07 124

Fig 23

SYSTEM	HC G/MILE	CO G/MILE	NO_x G/MILE
CLOSED LOOP OF ONLY ONE CYLINDER BANK AT LOW MILEAGE	0.25	2.4	0.51
CLOSED LOOP OF ONLY ONE CYLINDER BANK AT 40 000 KM MILEAGE ACCUMULATION	0.60	5.1	1.2
CHANGING TO A CONTROL OF BOTH CYLINDER BANKS AFTER 40 000 MILEAGE ACCUMULATION	0.33	3.0	0.90
CLOSED LOOP OF BOTH CYLINDER BANKS AT LOW MILEAGE	0.28	2.5	0.28

CVS-TEST RESULTS OF CLOSED LOOP CONTROL SYSTEMS ON A V8-ENGINE

V1MK 1276
07 126

Tab. 1

CRITICAL SURVEY of TEST PROCEDURES for AUTOMOTIVE EMISSIONS

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Research Division
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Wolfsburg, West Germany

Our experimental and theoretical investigations, which are now partly sponsored by the German Secretary of the Interior and assisted by the German "Umweltbundesamt", have shown that the test procedures prescribed by regulations in the USA and Europe are in many respects insufficient. They have to be improved in order to avoid unnecessary costs which ultimately must be borne by the consumer and thus by the public.

The different exhaust emission assembly line test procedures and certification test procedures, including the "Selective Enforcement Auditing" are surveyed. A probability analysis was carried out. It can be shown that there is an important relationship between the certification and assembly line procedures. Proposals for the improvement and standardization of existing exhaust emission procedures are deduced.

The regulation in the USA, that the exhaust emissions of e a c h vehicle have to be comply with the applicable standards, is unrealistic from the statistical viewpoint especially for assembly line testing. For the ambient air concentrations of the pollutants only the mean value is important.

INTRODUCTION

The current Exhaust Emission Regulations can be more or less categorized into three areas:

- Certification of prototype vehicles,
- Assembly Line Test of new vehicles,
- In-Use-Compliance and Inspection/Maintenance Program of in-use-compliance vehicles.

Each exhaust emission test in these categories is affected by the measurement uncertainties or test procedure variabilities.

Essentially, these errors and the evaluation methods determine the probability for passing the official tests.

The
measuring and evaluation procedures vary from country to country.

A worldwide standardization of test procedures is of course a most important item for governments and manufacturers. This standardization must include the driving cycle, the sampling and analytical equipment, as well as the evaluation of the measured data. Standardized procedures would save the manufacturers a waste of time and the customers a waste of money.

An experimental and theoretical investigation had to be carried out to propose better and standardized test procedures.

1. EXHAUST EMISSIONS

MEASUREMENT OF EXHAUST GAS MASS EMISSIONS - The European and the USA procedures for testing exhaust emissions differ not only in the driving cycles but also in the sampling and the analytical equipment. The vehicle is driven on a dyno. In Europe, the total exhaust emissions are collected undiluted in a bag while in the USA the exhaust emission gas is diluted with ambient air and gas quantities corresponding to the different phases of the driving cycle are directed into different small bags. In Europe, HC is measured with a NDIRA [1]; in the USA it is measured with a FID [2].

EXHAUST EMISSIONS VARIABILITY - In practice, all emission measurements will be subject to some degree of error. This results in uncertainties of the individual mass emissions consisting of random (or statistical) and systematic errors. The main part of the statistical exhaust emission variability, however, is the variability caused by the unavoidable influence of the vehicle (with the engine). Thus vehicle variability is defined as the spread in data caused by the instability of the vehicle itself without dyno and driver.

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[1] NDIRA = Non-Dispersive Infra-Red Analyzer

[2] FID = Flame Ionisation Detector

STATISTICAL ERRORS - Fig. 1 illustrates the results of the calculation of the statistical errors. The diagram shows the overall uncertainty of mass emissions ($s_{r\ total}$) as a function of the vehicle variability without measurement uncertainty ($s_{r\ vehicle}$) and the pure measurement uncertainty ($s_{r\ measurement}$), calculated as the square root of the squares of the individual error components. $s_{r\ measurement}$ was calculated to be about 6 %. The biggest part is caused by the driver (5 %) under the assumption that all measuring systems are very carefully calibrated. The usual vehicle variability is about 15 %. In this case, the contribution of the measurement uncertainty to the total variability is small compared to the contribution of the vehicle variability.

Fig. 2 shows as the results of repetitive measurements, relative standard deviations for several cars (model year US 76 and model year Europe 76) tested according to the US 75-procedure or European procedure respectively. Prior to the measurements the test equipment was carefully checked and calibrated using the new VW-torque calibration procedure. The cars were carefully conditoned and maintained including a 50...100 km run on the road after each test. The lowest achieved relative standard deviation $s_{r\ total}$ is 6 % for HC and for the US-vehicle no. 3 due to its fuel injection system. It represents a very stable car with regard to the HC-emissions. The highest value for HC in this group of US-vehicles is s_r = 19 %. It can be assumed that the pure measurement uncertainties were the same for the different US-vehicles. Thus, the difference (following the Gaussian law) between the 19 % and 6 % of 18 % is approximately the vehicle variability of the US-vehicle no. 1 and the pure measurement uncertainty is less or at most equal to the 6 % of vehicle no. 3. In this way, it is proven that the calculated value of 6 % of the pure measurement uncertainty can, in practice, be kept in this order of magnitude or even lower.

SYSTEMATIC ERRORS - Besides the above mentioned statistical errors there are systematic errors, especially unrecognized systematic errors.

Deviations in the test results between different cells or different laboratories represent typical unrecognized systematic errors. Fig. 3 shows the results of correlation tests. The emission mean values were computed on the basis of all test results per emission test cell (dynamometer) for three different vehicles. A distinct deviation of only one of the three curves indicates a vehicle-related effect. This condition indicates, in those cases where similar deviations occur in two curves, a systematic deviation (e. g. test cell no. 7 which had to be shut down). Systematic deviations of up to 40 % were even measured between two different test cells of EPA.

DISTRIBUTION FUNCTIONS - Most calculations on motor exhaust emissions assume a normal distribution of the measured values. Two more distribution functions were investigated:

the log-normal distribution and

the Weibull-distribution.

The applicability of these three distribution functions was tested by means of a chi-square test. For normal and log-normal distribution, a test of the third and fourth moments was also performed.

Fig. 4 shows as an example the results of repetitive exhaust emission measurements (US 75-cold start, FTP) for one and the same car in form of histograms and fitted distribution functions.

For this repetitive testing of one and the same car, the normal distribution was found to be a sufficient approximation for the US- und European cars.

SAFETY MARGIN - The true emissions of a vehicle can only be approximated by the mean emissions from a number of measurements, the approximation increasing with the number of measurements.

The consequence is that a manufacturer, when submitting for certification a vehicle with true emissions below the standards, has only a certain probability of passing the certification test. To make this probability reasonably high, the true emissions of the vehicle must be kept lower than the standards by a margin which increases with the increasing uncertainties of measurement at the official laboratory.

Conversely, the official laboratory runs a certain risk of accepting a vehicle with true emissions exceeding the standards.

The situation for one test is schematically shown in fig. 5. To ensure a high probability of meeting the standard, the true mean value of the mass emissions should have a certain safety margin which consists of the measuring uncertainty 2σ and a possibly unrecognized systematic error F.

CERTIFICATION TEST PROCEDURES

EXISTING PROCEDURES - The certification procedures of Europe and those of the USA do not only differ in the driving cycles and in the analytical equipment, but also in the evaluation of the measured data.

The evaluation method mainly determines the probability for passing or failing the certification.

Fig. 6 shows the European certification evaluation while fig. 7 shows the US-certification evaluation, both of which are in the form of flow charts.

In addition, fig. 8 and fig. 9 show two other flow charts for certification procedures; one is proposed by Volkswagen and the other is proposed by the European manufacturers (CCMC)*) following another proposal of Volkswagen.

PROBABILITY OF PASSING A CERTIFICATION - The calculations of probabilities were based on the assumption that the distribution of exhaust emission values of repetitive tests on one and the same vehicle follows a normal distribution which is proven by the investigations.

Fig. 10 shows the calculated operating characteristics for several certification procedures which give the probability of passing the certification test as a function of the true emission of the vehicle submitted. The true emissions are expressed as a percentage of the standard for a given uncertainty of measurement at the official laboratory.

The calculations have been performed for one pollutant, assuming that the standard deviation of the tests at the official laboratory is 20 % of the mean value.

*) CCMC = Committee of Common Market Automobile Constructors

The effect of changing the standard deviation is only a change in the slopes of the curves, which increase as the standard deviation decreases.

However, the relative position of the curves does not change and the discussion below remains valid. The main assumption is that the manufacturer knows the true emission before a certification test is performed.

The step curve 1 shows the theoretically ideal case for a certification, where the mean value is calculated from a very large number n ($n \rightarrow \infty$) of measurements. In this idealized case, the probability of passing a certification is 100 % as long as the true mass emissions are less or equal to the standard. The probability is zero, if the true mass emissions exeed the standard.

A good approximation is the mean value of 10 single measurements (curve 3). This case can be considered as the "practically ideal case" to which the other cases can be compared.

Curve 4 shows the undesirable case of only one measurement.

Curve 5 shows the case of the current European certification which deviates from the "practically ideal case" (curve 3) even more than the undesirable case of only one measurement (curve 4).

Curve 6 is the proposal by CCMC to the GRPA*) for a future European certification originally proposed to CCMC by Volkswagen.

*)GRPA = Group de Rapporteurs sur la Pollution Atmosphéric

This is a better approximation to the practically ideal case than the current European certification. Curve 7, another proposal by Volkswagen, includes an additional condition namely, an upper limitation of the first result. The corresponding operating characteristic shows only a little deviation in the lower part of the curve in comparison to curve 6.

Curve 2, which is valid for the US-certification, shows a better agreement with the "practically ideal case" at least in the vicinity of the standard and below the standard.

Consequently, the European procedure requires a much higher safety margin and is thus more stringent than the US procedure assuming the same standards. It should be improved in the direction of the US procedure (or at least in that of the CCMC or VW-proposals).

EXPERIMENTAL VERIFICATION OF PROBABILITY CALCULATIONS - The theoretical probability calculations have been well proven by the results of certification testing. Fig. 11 shows as an example the values of HC submitted and the values of HC measured at certification for the 1974 models in the USA. Similar results can be shown for the 75/76 model years and for the other components as well.

This figure illustrates that the theoretically calculated engineering goals are in good agreement with the results obtained in practice. Seven values exeed the "engineering goal for 1974 without systematic errors".

Of these, four (1, 2, 3 and 4) are critical with regard to compliance for certification. The differences between the values submitted and the values measured at certification are in many cases of the same order of magnitude as the safety margin, or even greater. There are additional discrepancies which are obviously due to systematic errors. This is proven by the different mean values of mass emissions of all vehicles measured at EPA and VW.

ASSEMBLY LINE TEST PROCEDURES

EXISTING PROCEDURES - For assembly line testing in Europe the measurements are performed in the same way as for certification. However, the standards to be fulfilled are less stringent than with certification. The standard for CO is 20 %; for HC it is 30 % and for NO_x it is 15 % higher. If a vehicle fails this test, the manufacturer can demand measurements on a large sample of production vehicles. The manufacturer determines the size of the sample.

The mean values of these measurements are decisive. They are compared to the standards under consideration of test variability.

In California, assembly line testing is required since April 1, 1970. Today the procedure is based on a sample size of 2 % of the vehicles sold. Only 10 % of the tested vehicles may in their mass emissions exceed the standards.

With model year 1977 another kind of evaluation of the 2 % - assembly line test data can be chosen as an option: the mean of the measured exhaust values of an engine family has to be less or equal to the standard which is however, multiplied by the deterioration factor.

For the USA (except California) a socalled SEA procedure (Selective Enforcement Auditing) is valid beginning with January 1977. SEA (similar to the Californian option which allows 10 % of the cars to exceed in their mass emissions the standards) is based on the attributive method.

For an attributive method, the tested units are classified into "pass" or "fail" classes depending on whether or not they satisfy a certain criterion.

For SEA this classification depends upon whether the exhaust emission values of a vehicle do or do not exceed the standards for all three limited components. The legal background is the "Clean Air Act", which is interpreted in such a way that for each vehicle the mass emissions must not exceed the standards. This is, of course, not the appropriate method from a statistical point of view.

The percentage of cars allowed to exceed the standards in their mass emissions is now 40 % for SEA.

Fig. 12 shows schematically the calculated safety margins for the mean of the emission results for assembly line testing and for the European- and US-procedures.

DISTRIBUTION FUNCTIONS - The results of assembly line test series for different models and makes (model year 75 of US- and

European cars) were investigated with regard to their distributions. Here, the normal distribution was found to be insufficient, log-normal and Weibull-distributions being better approcimations. For NO_x-mass emissions a normal distribution is also possible (see fig. 13). Generally the log-normal distribution can be applied.

Relative standard deviations for the assembly line test data of US-vehicles and for European vehicles were about 20 % to 50 % for HC and CO, and about 10 % to 25 % for NO_x.

The relative standard deviations for assembly line test data are higher by about a factor of two compared to the standard deviations of the repetitive test data.

CRITICAL COMMENTS TO SEA - According to EPA the sampling plan of SEA is selected in such a way that the batch reflects statistically the emission behavior of all the produced cars of a certain engine family. Consequently this sampling plan an AQL of 40 % allow in the extreme case up to 40 % "failures" of the produced vehicles, i. e. in this extreme case, 40 % of all produced vehicles are allowed to have emissions higher than the standards as long as they are not tested and identified and thereby can still be sold.

However, in the SEA proposal it is required that each individual vehicle which is tested by the EPA according to SEA and which exceeds the applicable standards is not allowed to be sold. That means that those vehicles which "fail" the test have to be adjusted and/or repaired and retested until they "pass" the test (e. g. out of a batch of max. 60 vehicles tested, 24 vehicles might be involved).

This request is illogical and cannot in principle be accomplished because in the case of individual vehicle testing (which is similar to certification testing) the uncertainty of a "single measurement" has to be considered. In the cases where emissions of individual vehicles during SEA testing exceed the standards, the repairement of these vehicles has to include the safety margin of up to 50 % to achieve a good enough probability to pass the single test (see fig. 5). Therefore, it might become necessary to install completely new parts, which would be considered to be a violation of the regulations. Consequently if EPA requests to adjust and/or to repair failed vehicles, the manufacturer would try to avoid additional testing because this testing might result in identifying failures.

According to regulations, the selling of failing vehicles can be used against the manufacturer. Additional testing, however, is necessary to achieve a good quality control. Thus SEA is restricting the manufacturer's activities to improve his own quality control program.

INTERRELATION BETWEEN CERTIFICATION AND ASSEMBLY LINE TESTING

Following the requirements of the Federal Register, certification vehicles and production vehicles must be identical with regard to their engine parts which influence exhaust emissions. This leads to the hypothesis that the mean of the emission test results of production vehicles must be nearly equal to the true emission value of the corresponding certification vehicle. We actually found this hypothesis confirmed for a number of engine families.

Fig. 14 therefore shows that the mean of emission results of production vehicles has a distance to the standard (with deterioration factor included) which is of the same order of magnitude as the safety margin for certification vehicles.

PROPOSALS FOR EXHAUST EMISSION TEST PROCEDURES

GENERAL - The purpose for establishing emission standards should be to assure a certain ambient air quality.

The ambient air quality is influenced by the total amount of mass emission of all vehicles in traffic. This total amount is given by the mean of the mass emissions of all vehicles multiplied by the number of vehicles in traffic.

$$x_{total} = \sum_{j=1}^{N} x_j = \bar{x} . N$$

x_j = mass emission of vehicle j

N = no. of vehicles

$\bar{x}$ = mean value

It follows that the total emission does not depend on the distribution of the individual values x_j.

The uncertainties of the means are small ($\sim 1/\sqrt{n}$, n = sample size); the means always follow approximately a normal distribution (central theorem of statistics).

The European procedure and the option of the California Air Resources Board of taking the mean (or average) as the deciding quantity are better with regard to ambient air quality. More important for ambient air quality is the assembly line testing; however, a certification is still useful to assure that the manufacturer is able to produce vehicles with emissions below the standards. For a certification procedure not only do the variability and the probability for passing or failing have to be considered, but also the interrelation between certification and assembly line testing as well.

The "police"-paper work of certification certainly is unnecessary.

PROPOSED STANDARDIZED TESTING PROCEDURES FOR ALL COUNTRIES

DRIVING CYCLE - As a standardized driving cycle we propose the US 72 or the main pattern of this cycle. Our investigations of the different current cycles and of urban driving behavior

have shown that the US 72 simulates urban driving behavior also in European cities.

SAMPLING EQUIPMENT - The exhaust sampling while the vehicle is being driven on a dyno should be performed like the current CVS method; however, the sampling tubes to the CVS-unit should be heated. For the purpose of analyzing the individual exhaust components we propose the current US-equipment including the FID.

CALIBRATION OF TEST CELLS - The total dyno road load characteristic should be defined not just in one point. The dyno should also be dynamically tested using a torque measurement procedure.

A standardardized vehicle with

- mechanized driver,
- "golden" engine with emission variability smaller or equal to $\pm$ 2 % and which is practically insensitive to influence of ambient air pressure, temperature and humidity,
- torque measuring equipment for dyno calibration

should be used for comparing and calibrating the different test cells in order to avoid systematic deviations.

STANDARDIZED EVALUATION PROCEDURES

CERTIFICATION - The certification test should be performed at the manufacturer as a general principle. For these tests always the same driver should be used to minimize his influence. However, we propose multiple measurements in connection with an averaging method:

The manufacturer performs at least 6 tests (maximum 10 tests) in the presence of personnel from the government.

Certification is passed if

$$\bar{x} + \frac{t \cdot s}{\sqrt{n}} \leq x_{standard} .$$

As shown in this paper, normal distribution can be assumed for the repetitive tests. Furthermore, the uncertainties of the mean always follow approxemately a normal distribution.

To minimize the expenditure, several engine families should be grouped together. Deterioration factors are inapplicable due to the known reasons for their questionable determination.

In exceptional cases, when the presence of people from the government at certification testing is not possible a control test could be performed at a governmental test facility which is calibrated beforehand with a standardized vehicle. In this case, the application of a retest as in the USA can be used. This retest is the best approximation to averaging (see Fig. 10).

ASSEMBLY LINE TESTING - In principle, it is assumed that the manufacturer is performing a quality audit test by himself using a continuous sampling plan. His criterion will be:

$$\bar{x} + \frac{t \cdot s}{\sqrt{n}} \leq x_{standard}.$$

Deterioration factors should not be used as in the case of certification.

If the Government performs a control testing it should be carried out by means of a batch sampling once a year (like SEA).

But in this case, the criterion for compliance should be (like in California - option 2):

$$\bar{x} \leq x_{standard}$$

IN USE COMPLIANCE - For in-use-compliance the idea of averaging should, of course, also be applied. For individual vehicles the variability of the production cars have then to be considered. The individual car has to be rechecked only if the emissions are higher than a certain control limit ($\bar{x}$ + 3s).

If a short test is used the correlation between this short test and the FTP has to be established first.

DISCUSSION OF PROPOSALS - Fig. 15 shows the consequences of the proposals. First of all it is very important that a systematic deviation need not be considered for the safety margin if a standardized vehicle is used for correlating the different test cells.

Because of the interrelation between assembly line testing and certification, the proposed certification procedure will still result in a small safety margin for the production mean thereby giving a protection against certain production faults. The control testing by the government is sufficient to assure a good air quality standard.

Fig. 1 - Total measurement variability ($s_{r\ total}$) as function of the vehicle variability ($s_{r\ vehicle}$) and the pure measurement uncertainty ($s_{r\ measurement}$).
$s_r = s/\bar{x} \cdot 100$ in %.

Fig. 2 - Relative standard deviations from 50 test results of exhaust emissions and of the calculated fuel consumptions for three typical US-vehicles, model year 75, and for typical European vehicles, model year 76. (Ambient air pressure variation ± 0,6 %; humidity variation ± 19 %, temperature variation ± 0,3 %).

Fig. 3 - Results of a correlation test for 10 test cells and 3 vehicles (o , □ , Δ). Mean values of 30 through 70 tests for each test cell. Test performed within 5 months.

Fig. 4 - Histograms and fitted distribution functions of exhaust emissions for 50 test results on one and the same vehicle.

Fig. 5 - Safety margin ($F + 2\sigma$) resulting from random (2σ) and systematic errors (F).

Fig. 6 - Flow-chart of European Certification Test.

Fig. 7 - Flow-chart of US-Certification Test.

Fig. 8 - Flow-chart of a CCMC-proposal (improved European Certification Test) originally proposed by Volkswagen.

Fig. 9 - Flow-chart of another VW-proposal (improved European Certification Test).

Fig. 10 - Operation characteristics.
Probability (risk) as functions of the true mean value μ_r, calculated with the "Monte Carlo-Method" for different test procedures and different numbers of single measurements. A relative standard deviation of 20 % is assumed.

Fig. 11 - Values submitted (o) and values measured at EPA-certification testing (x) of HC-mass emissions for vehicles, model year 1974, and safety margins.

Fig. 12 - Illustration of quality audit tests for the different countries and for HC as an example.

EU_1 : Europe, 1 vehicle, $x_i \leq x_{i\ standard}$

EU_2 : Europe, n vehicles, $(\bar{x}_i + k \cdot s) \leq x_{i\ standard}$, $k = f(n)$

CAL_1 : California Option 1: maximum 10 % of the tested vehicles with $x_i > x_{i\ standard}$

CAL_2 : California Option 2: $\bar{x}_i \leq x_{i\ standard}/DF$.
For both options CAL_1 and CAL_2;
sample size 2 % of the production,
$x_{HC} = x_{HC\ total} - x_{Methane}$

SEA : Selective Enforcement Auditing:
maximum 40 % auf the tested vehicles with
$x_i \geq x_{i\ standard}/DF$,
sample size, maximum 60 vehicles.

Fig. 13 - Histograms and fitted distribution functions of exhaust emissions for 101 catalyst equipped US-production vehicles, model year 75 from assembly line.

Fig. 14 - Results of certification testing and mean values of production vehicles relative to the standard for 9 engine families, and safety margins.

Fig. 15 - Illustration of the relationships of the different proposed exhaust emission test procedures.

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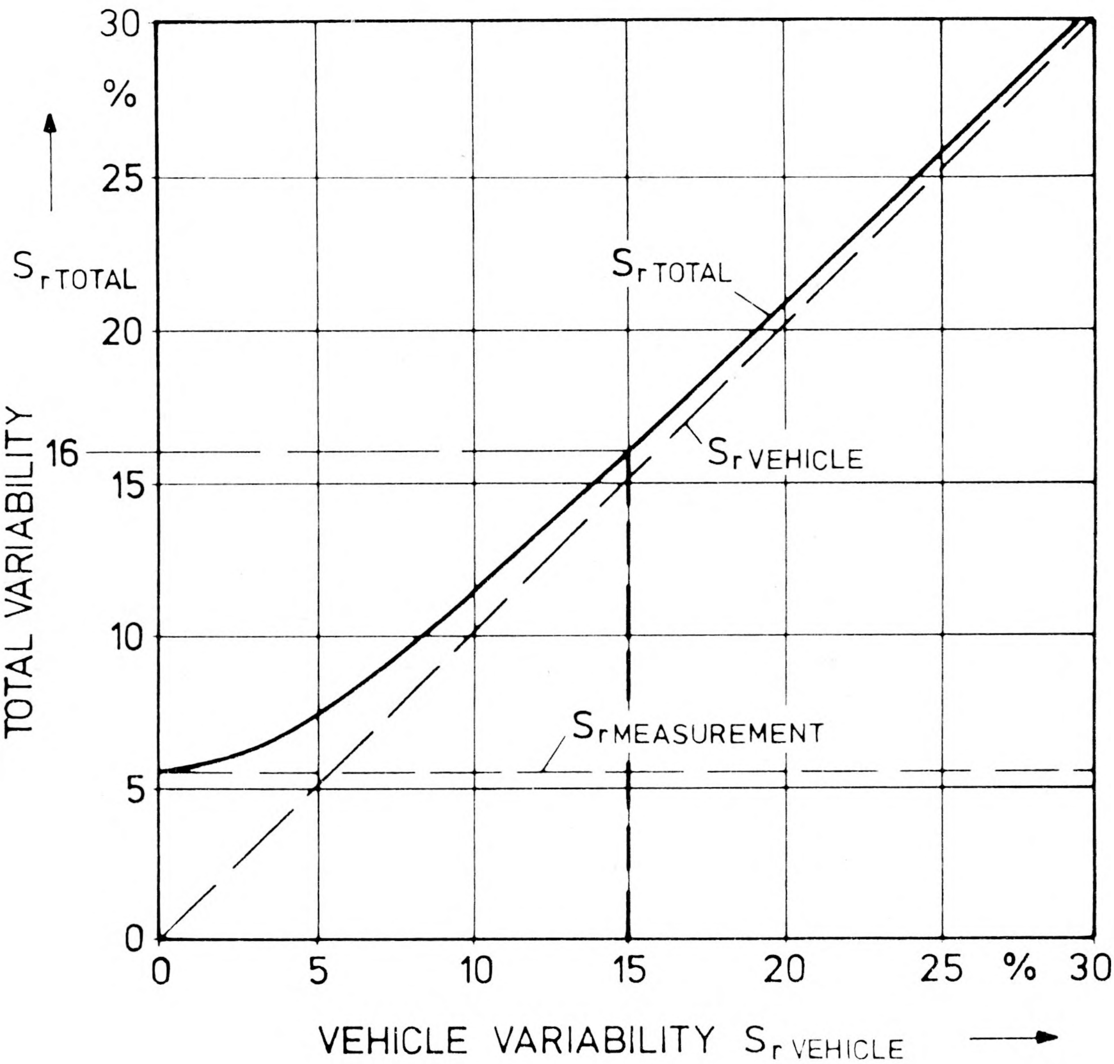
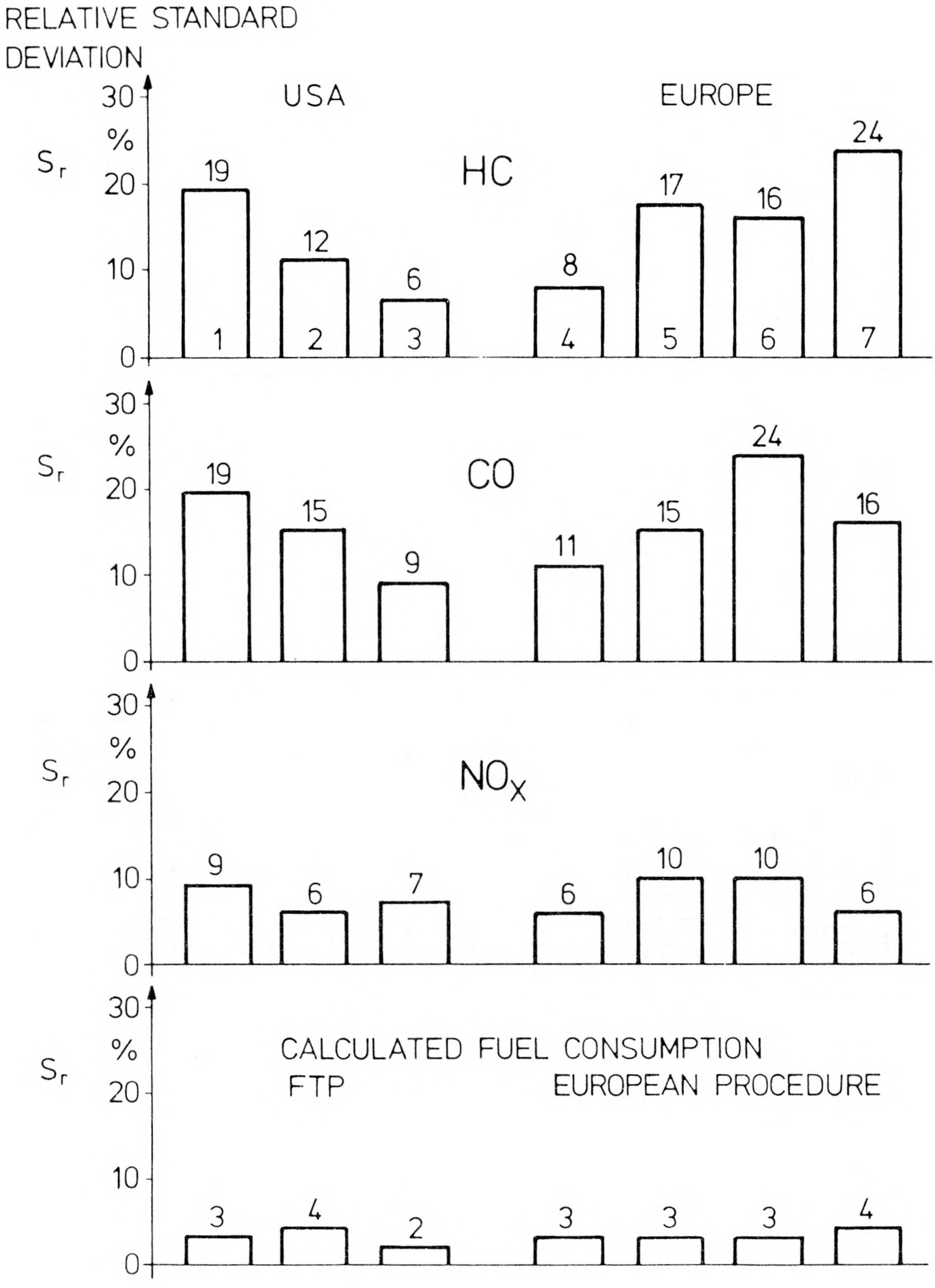
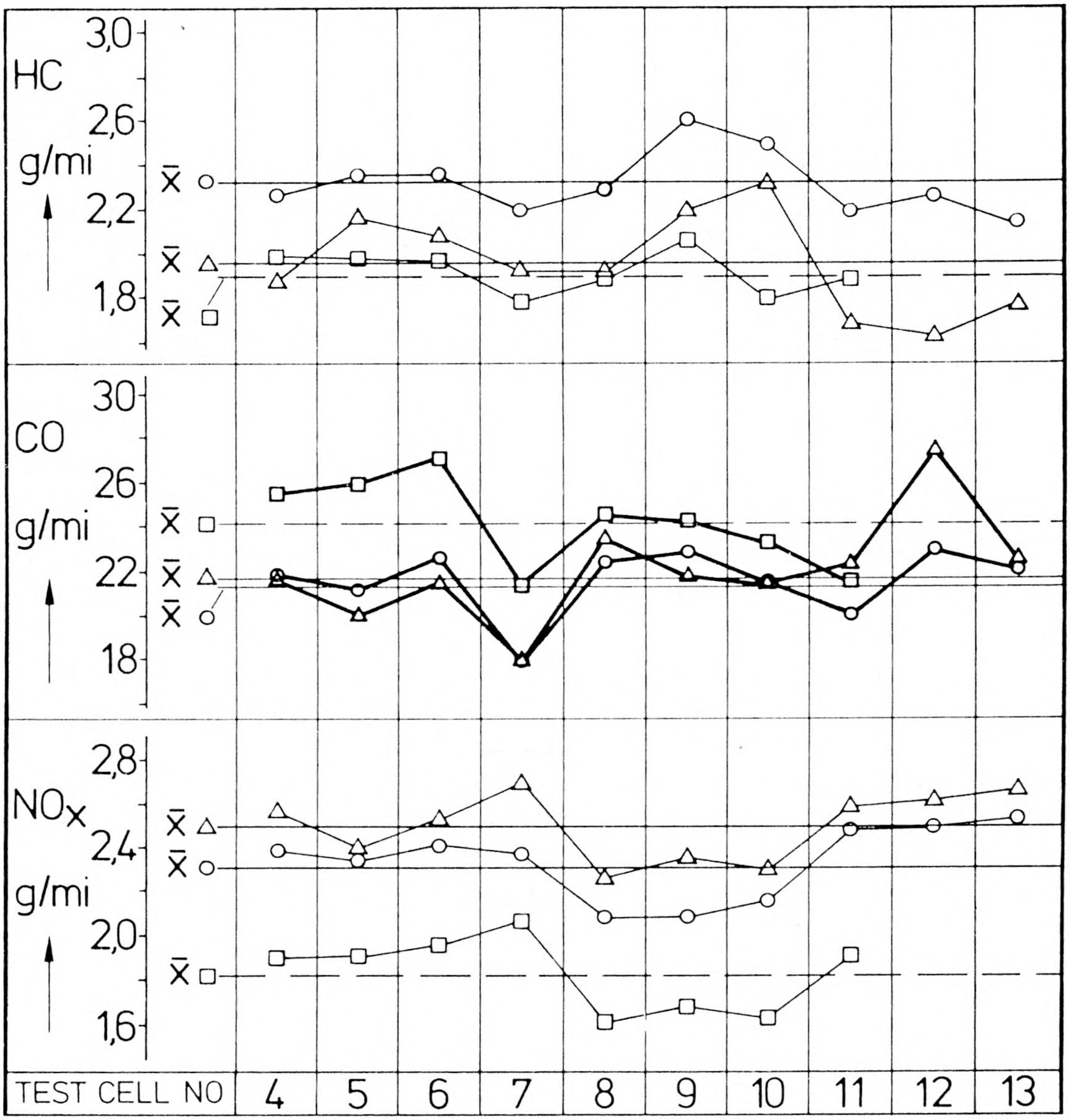


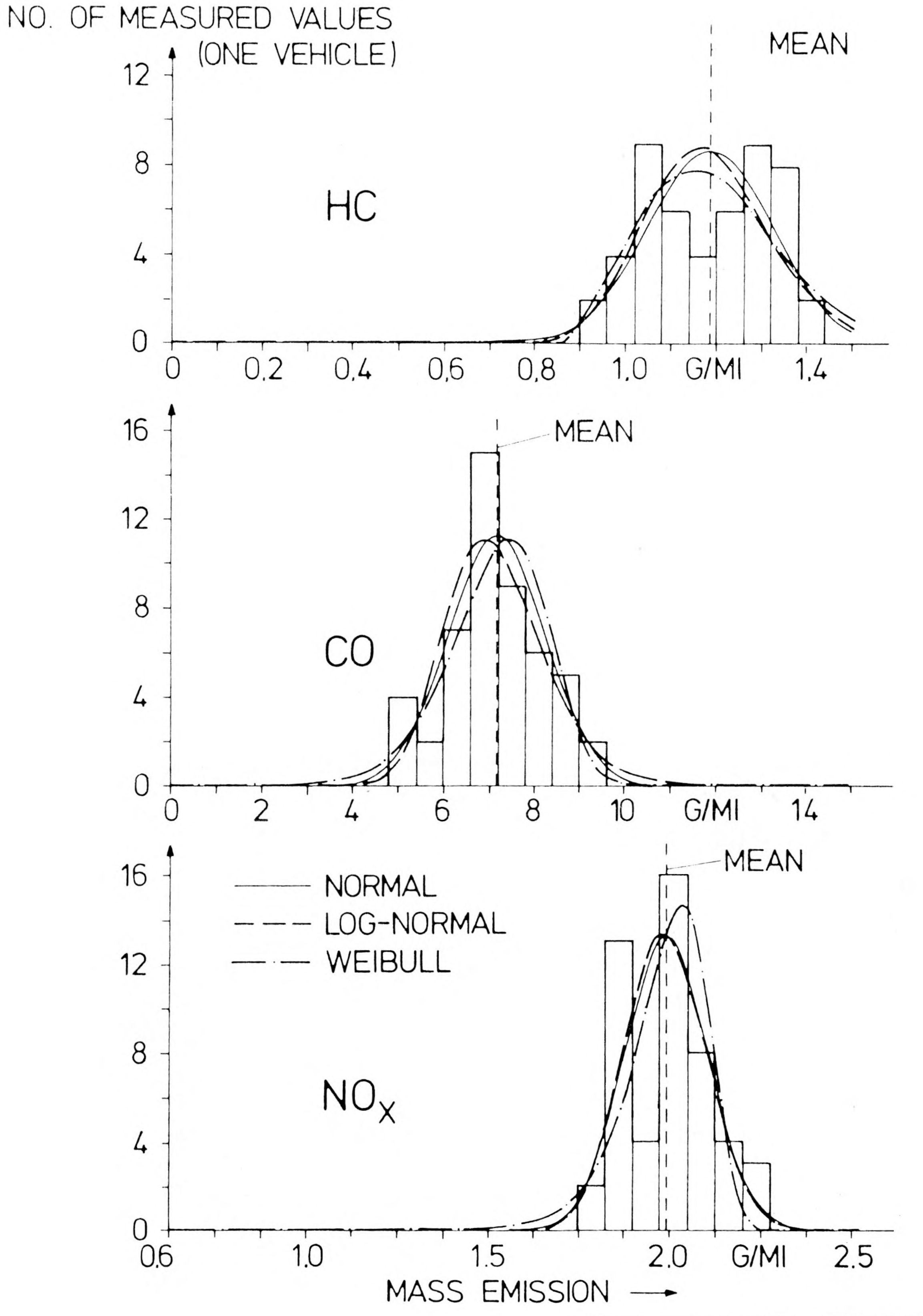
30
%
25
$S_{r\,TOTAL}$
20
16
15
10
5
0
TOTAL VARIABILITY
$S_{r\,TOTAL}$
$S_{r\,VEHICLE}$
$S_{r\,MEASUREMENT}$
0
5
10
15
20
25
%
30
VEHICLE VARIABILITY $S_{r\,VEHICLE}$



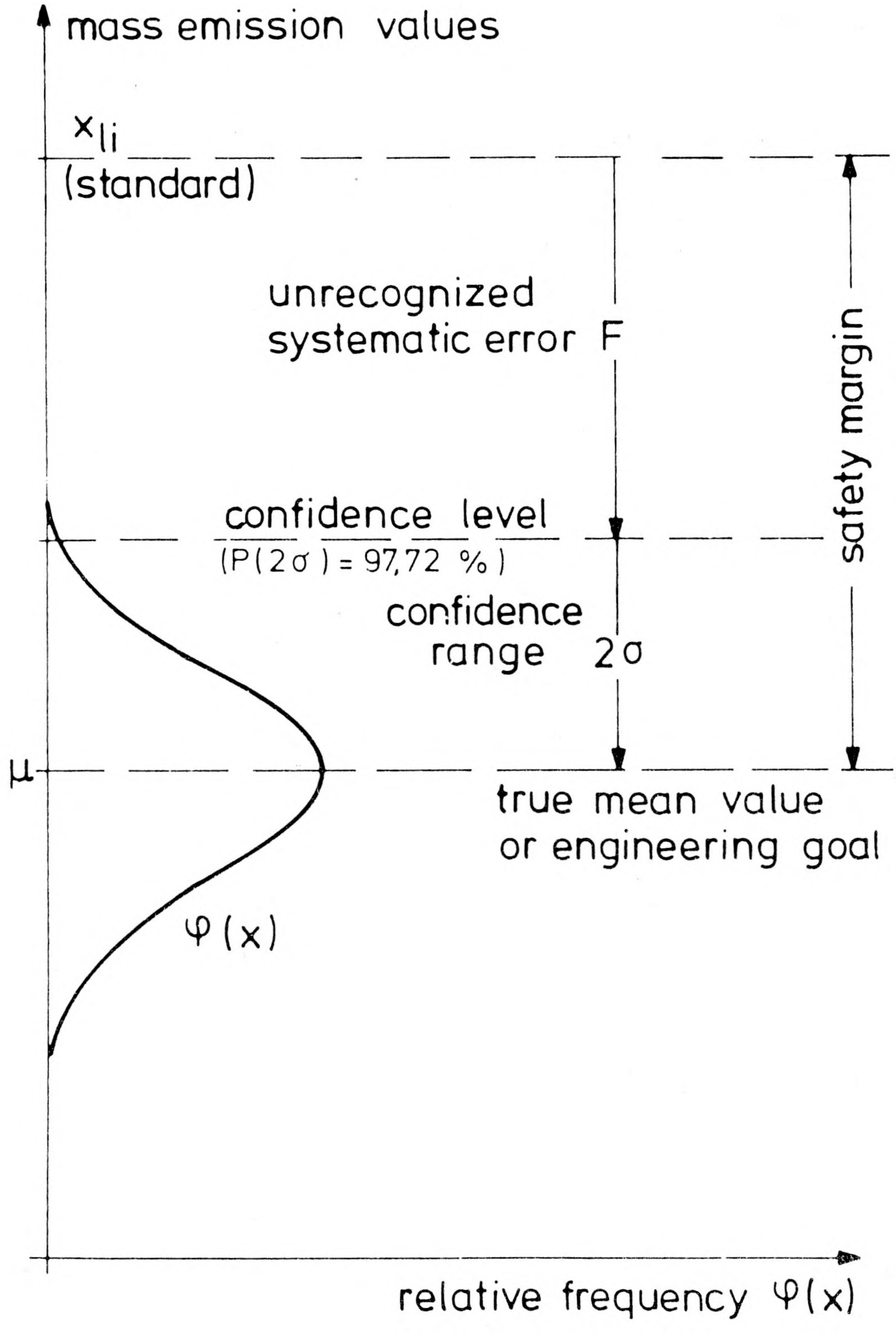
RELATIVE STANDARD DEVIATION
S_r
%
USA
EUROPE
HC
19 12 6 8 17 16 24
1 2 3 4 5 6 7
CO
19 15 9 11 15 24 16
NO_X
9 6 7 6 10 10 6
CALCULATED FUEL CONSUMPTION
FTP
EUROPEAN PROCEDURE
3 4 2 3 3 3 4
0 10 20 30

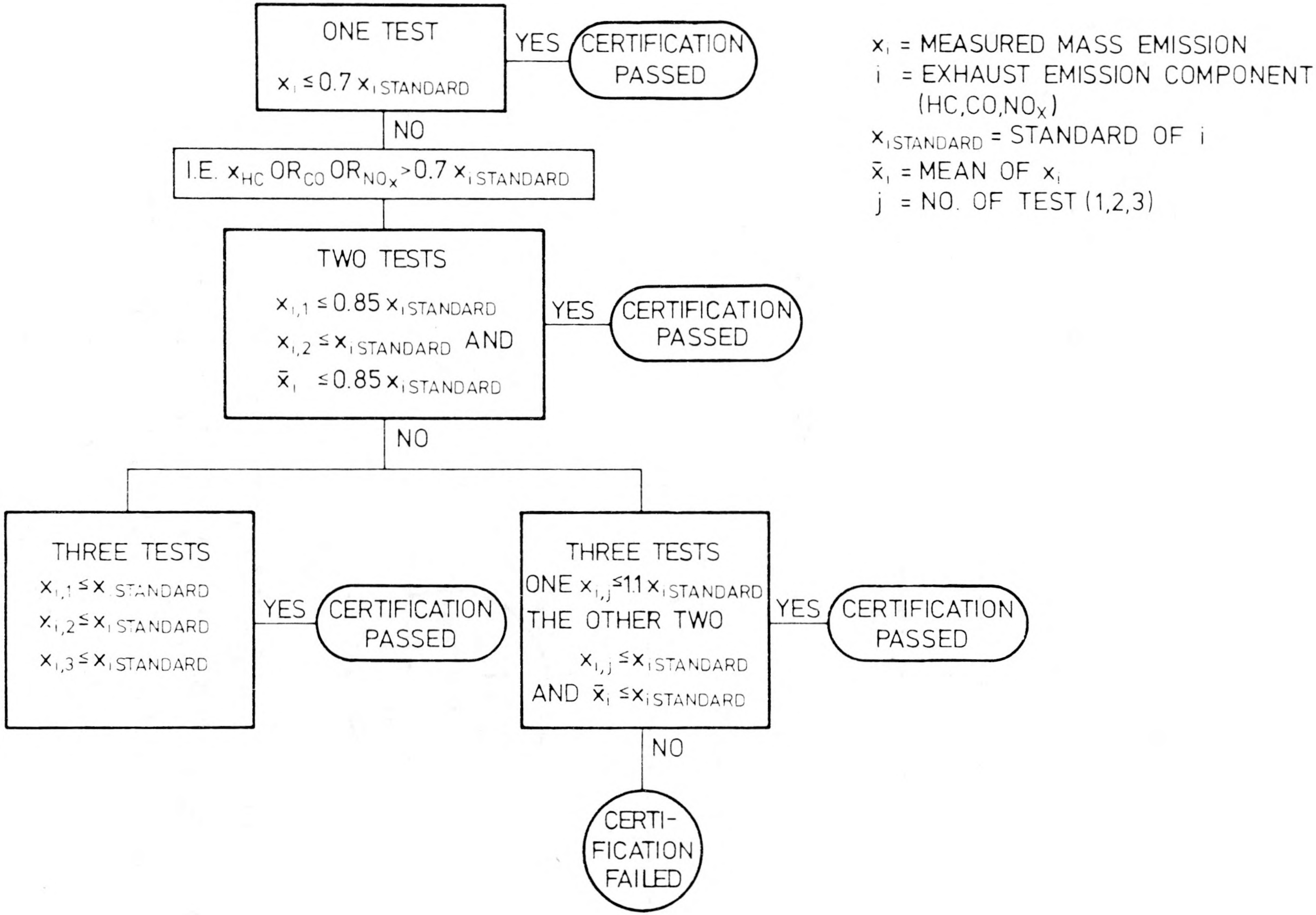


HC
g/mi
3,0
2,6
2,2
1,8
x̄ ○
x̄ △
x̄ □
CO
g/mi
30
26
22
18
x̄ □
x̄ △
x̄ ○
NOx
g/mi
2,8
2,4
2,0
1,6
x̄ △
x̄ ○
x̄ □
TEST CELL NO
4
5
6
7
8
9
10
11
12
13

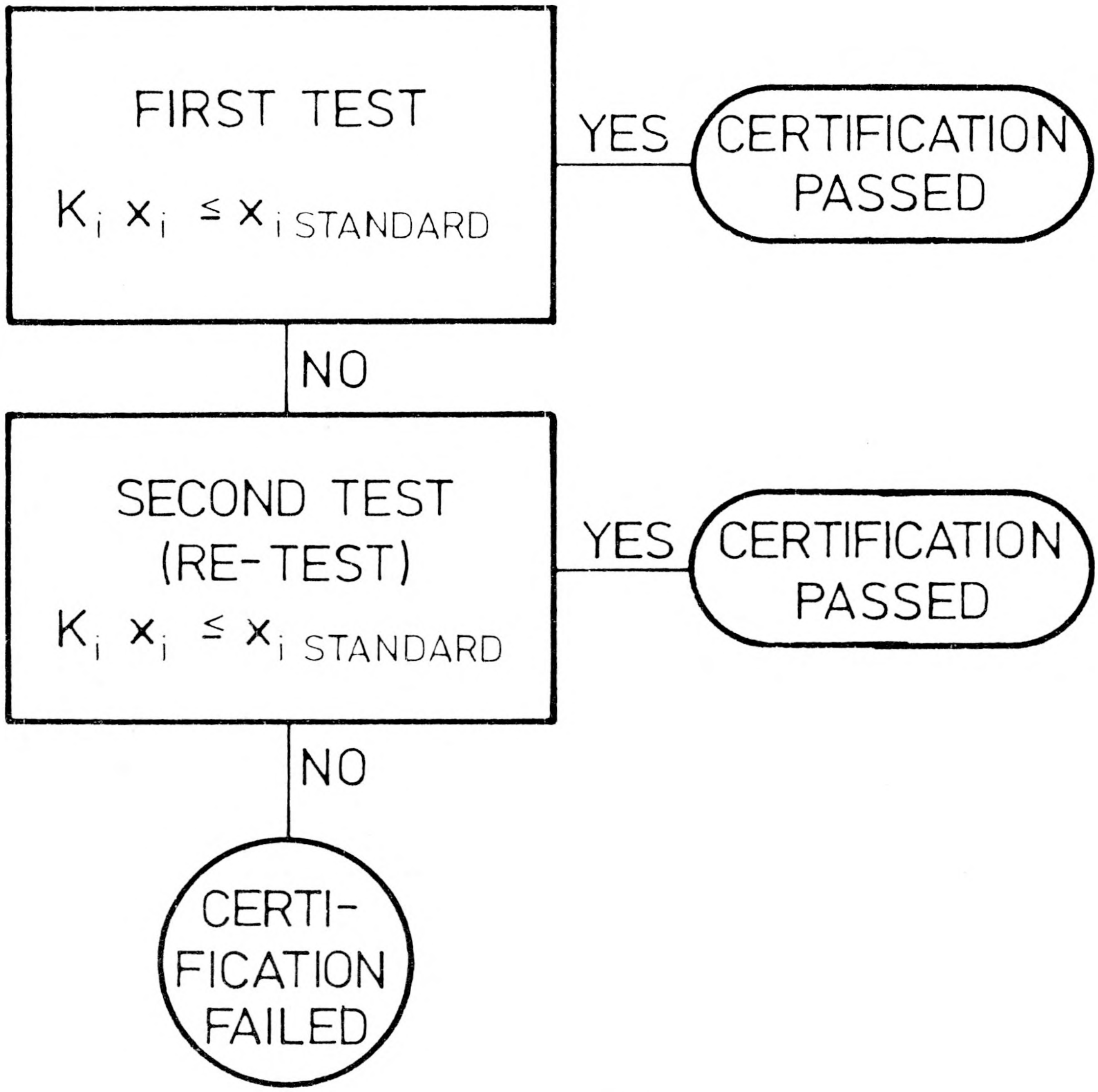


NO. OF MEASURED VALUES
(ONE VEHICLE)
MEAN
12
8
4
0
HC
0
0,2
0,4
0,6
0,8
1,0
G/MI
1,4
16
12
8
4
0
MEAN
CO
0
2
4
6
8
10
G/MI
14
16
12
8
4
0
MEAN
NORMAL
LOG-NORMAL
WEIBULL
NO
X
0,6
1,0
1,5
2,0
G/MI
2,5
MASS EMISSION


mass emission values
x_{li}
(standard)
unrecognized
systematic error F
safety margin
confidence level
(P(2σ) = 97,72 %)
confidence
range 2σ
μ
true mean value
or engineering goal
φ(x)
relative frequency φ(x)



ONE TEST
x_i ≤ 0.7 x_i STANDARD
YES
CERTIFICATION PASSED
NO
I.E. x_HC OR_CO OR_NOx > 0.7 x_i STANDARD
TWO TESTS
x_i,1 ≤ 0.85 x_i STANDARD
x_i,2 ≤ x_i STANDARD AND
x̄_i ≤ 0.85 x_i STANDARD
YES
CERTIFICATION PASSED
NO
THREE TESTS
x_i,1 ≤ x_i STANDARD
x_i,2 ≤ x_i STANDARD
x_i,3 ≤ x_i STANDARD
YES
CERTIFICATION PASSED
THREE TESTS
ONE x_i,j ≤ 1.1 x_i STANDARD
THE OTHER TWO
x_i,j ≤ x_i STANDARD
AND x̄_i ≤ x_i STANDARD
YES
CERTIFICATION PASSED
NO
CERTI-FICATION FAILED
x_i = MEASURED MASS EMISSION
i = EXHAUST EMISSION COMPONENT (HC,CO,NO_X)
x_i STANDARD = STANDARD OF i
x̄_i = MEAN OF x_i
j = NO. OF TEST (1,2,3)

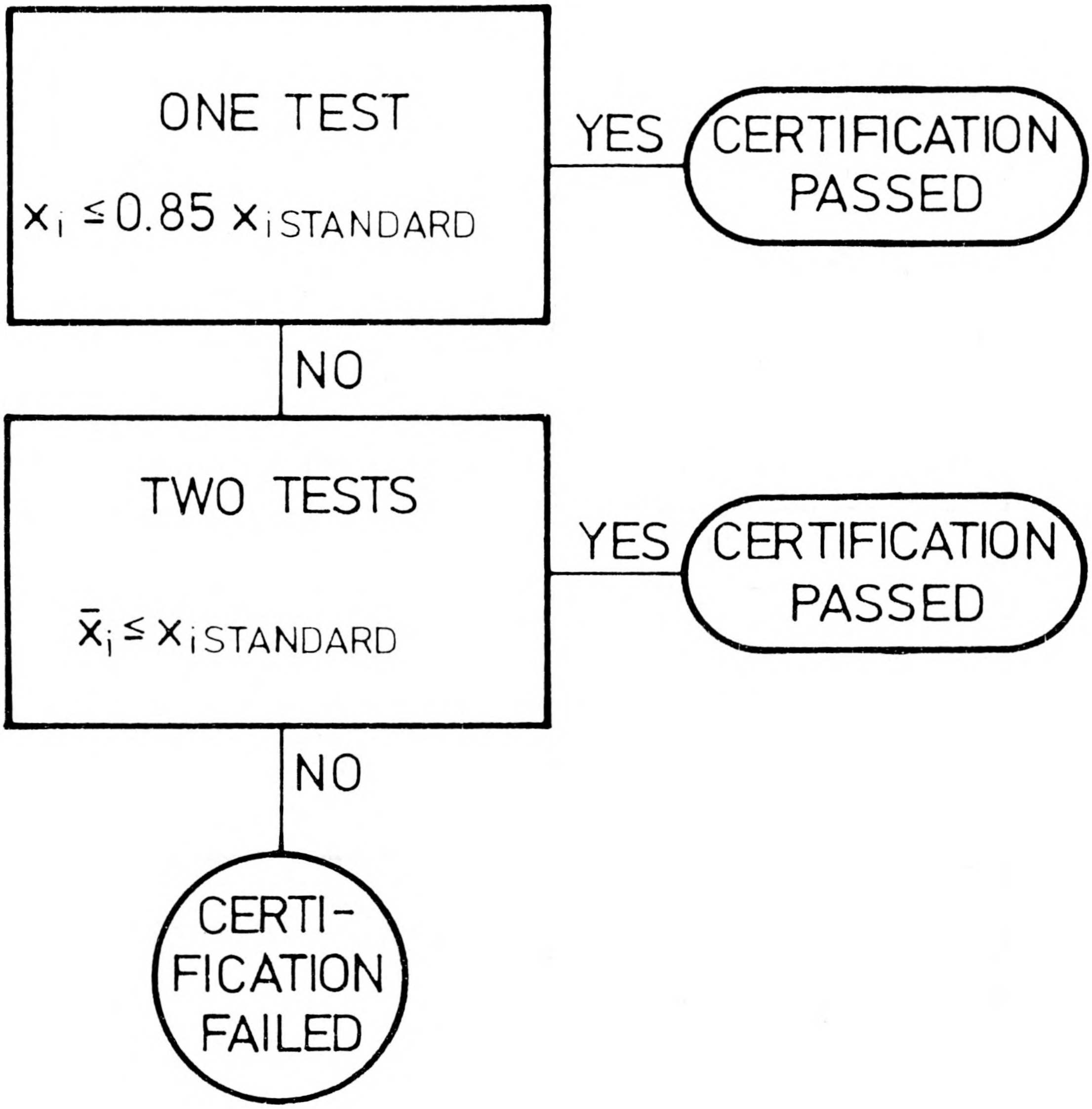


K_i = DETERIORATION FACTOR

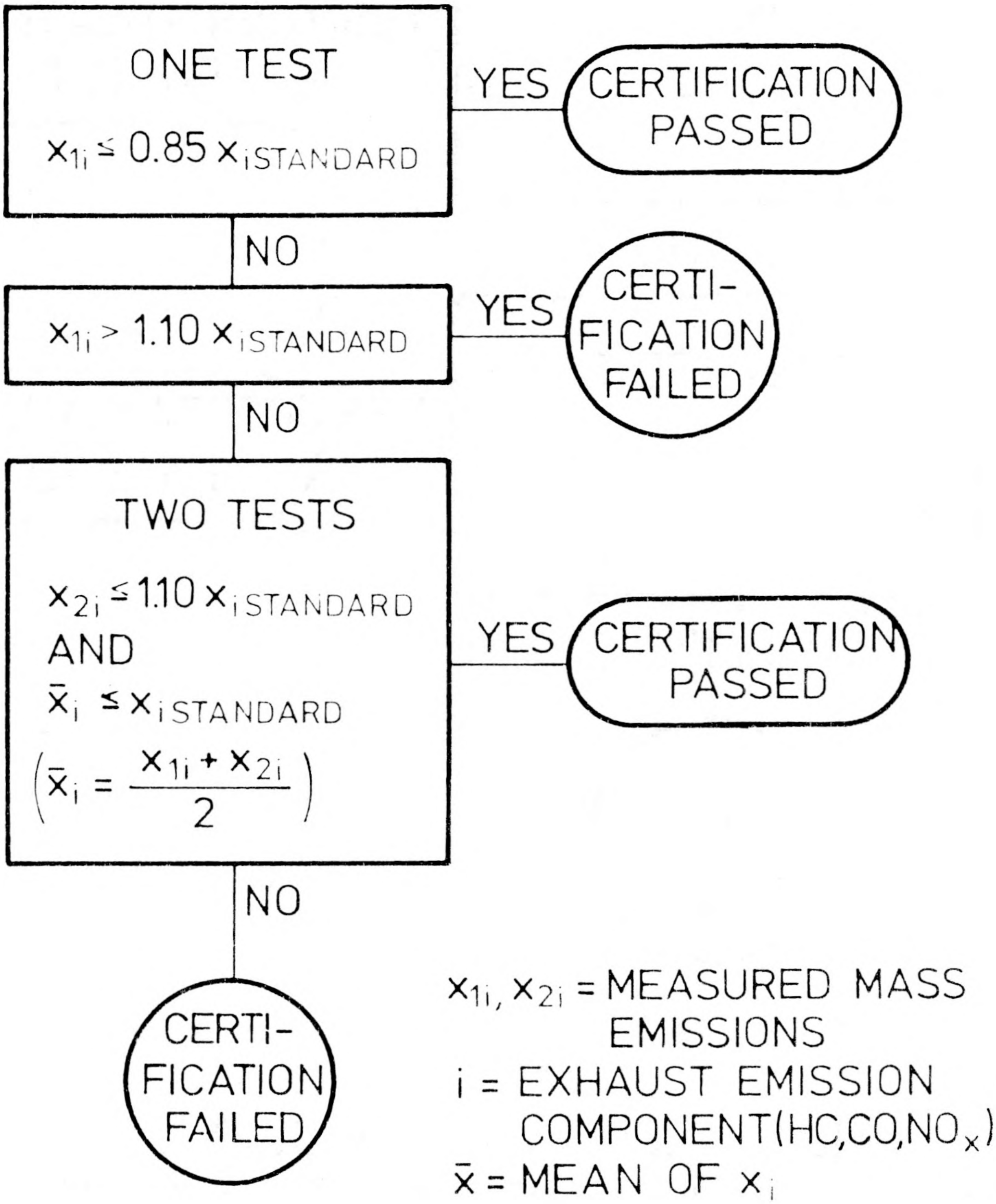
x_i = MEASURED MASS EMISSION

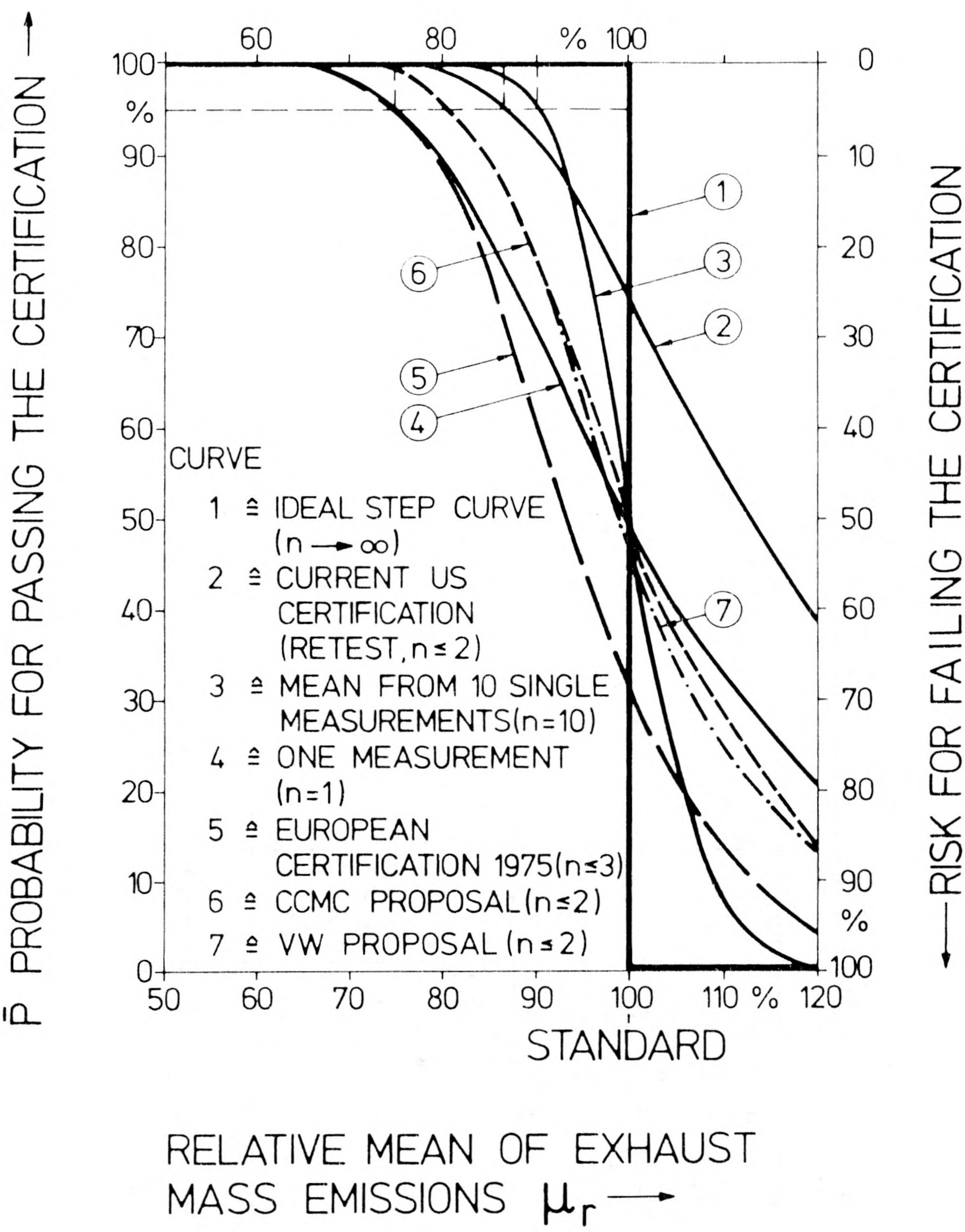
i = EXHAUST EMISSION COMPONENT (HC, CO, NO_X)

$x_{i\ STANDARD}$ = STANDARD OF i

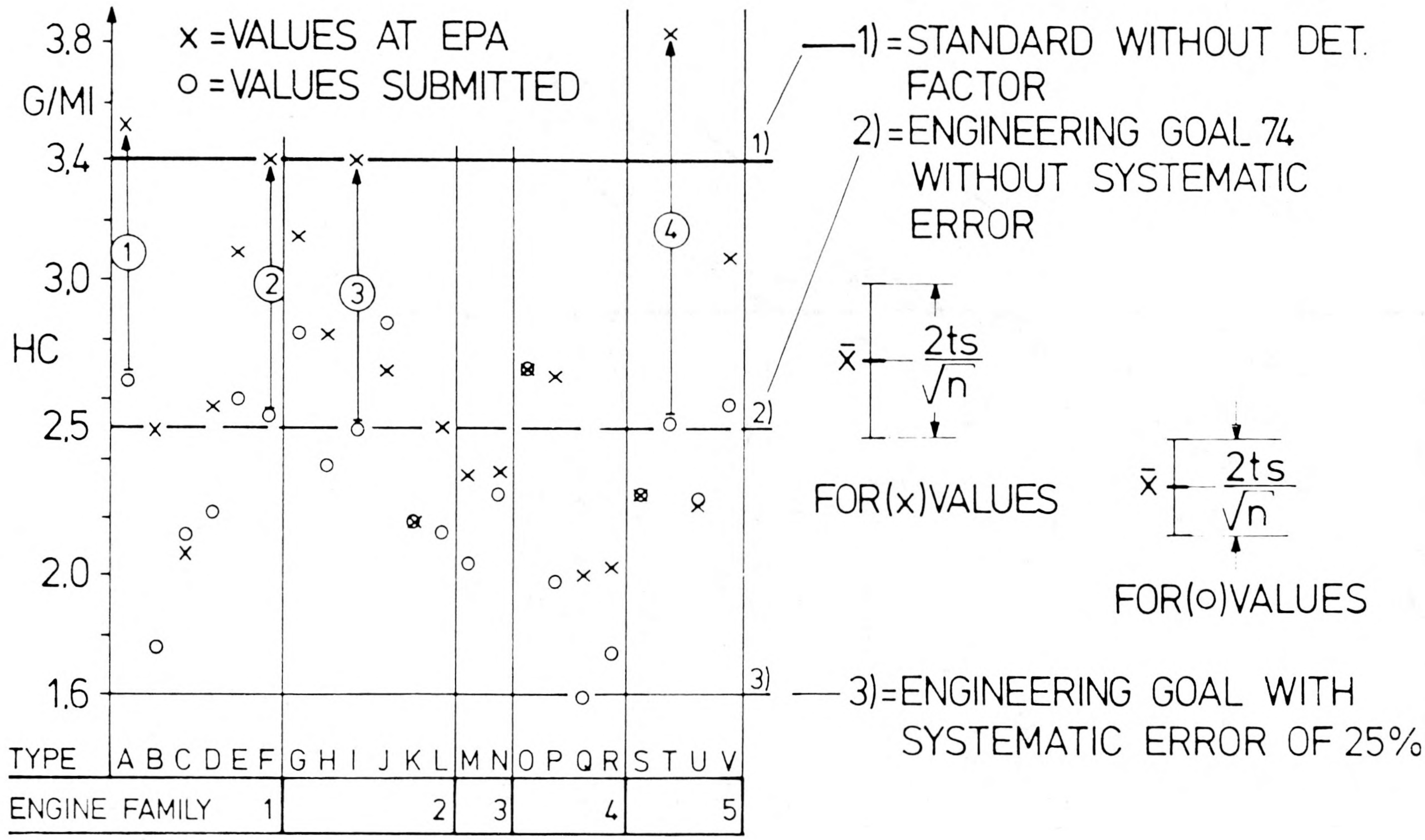


x_i = MEASURED MASS EMISSION
i = EXHAUST EMISSION COMPONENT (HC, CO, NO_X)
$\bar{x}_i$ = MEAN OF x_i

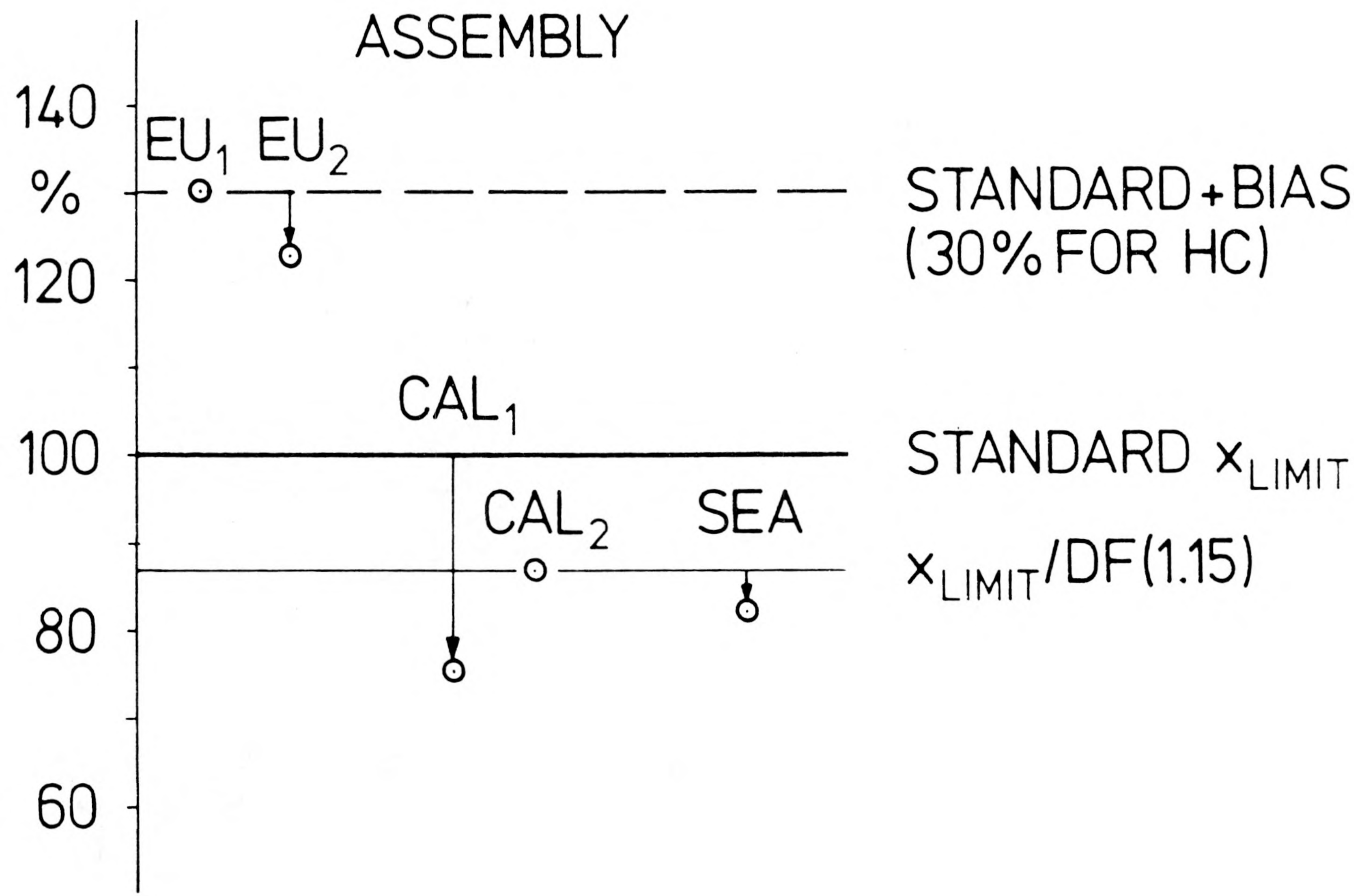

ONE TEST
$x_{1i} \leq 0.85\, x_{i\,STANDARD}$
YES
CERTIFICATION PASSED
NO
$x_{1i} > 1.10\, x_{i\,STANDARD}$
YES
CERTI-FICATION FAILED
NO
TWO TESTS
$x_{2i} \leq 1.10\, x_{i\,STANDARD}$
AND
$\bar{x}_i \leq x_{i\,STANDARD}$
$\left(\bar{x}_i = \frac{x_{1i} + x_{2i}}{2}\right)$
YES
CERTIFICATION PASSED
NO
CERTI-FICATION FAILED
x_{1i}, x_{2i} = MEASURED MASS EMISSIONS
i = EXHAUST EMISSION COMPONENT(HC,CO,NO_x)
$\bar{x}$ = MEAN OF x_i



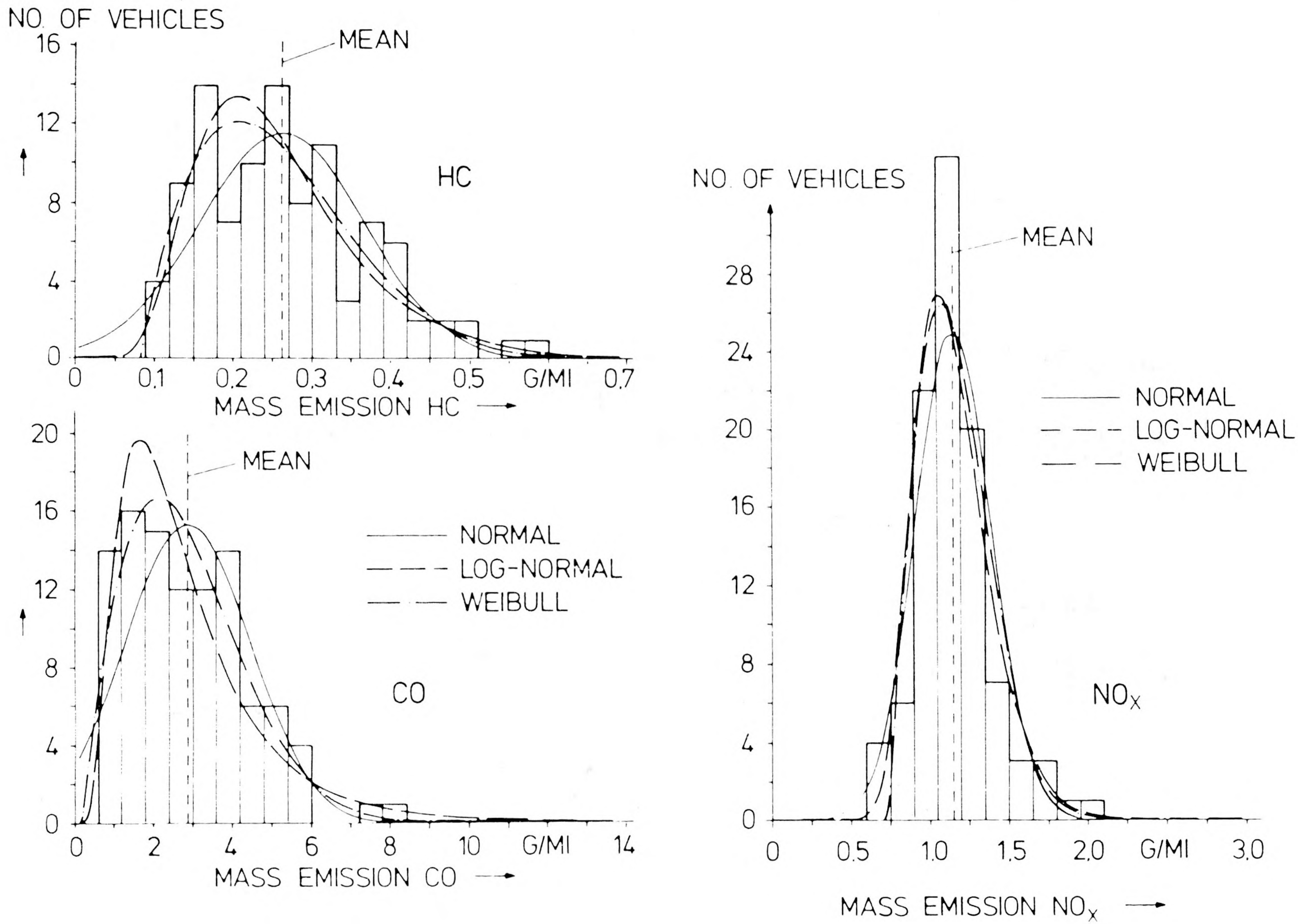
(BASED ON THE STANDARD) DETERMINED BY THE MANUFACTURER FROM MANY MEASUREMENTS, I.E. $\mu_r \approx$ TRUE MEAN VALUE.
ABSOLUTE ACCURACY FOR $\bar{P} = \pm 2\%$

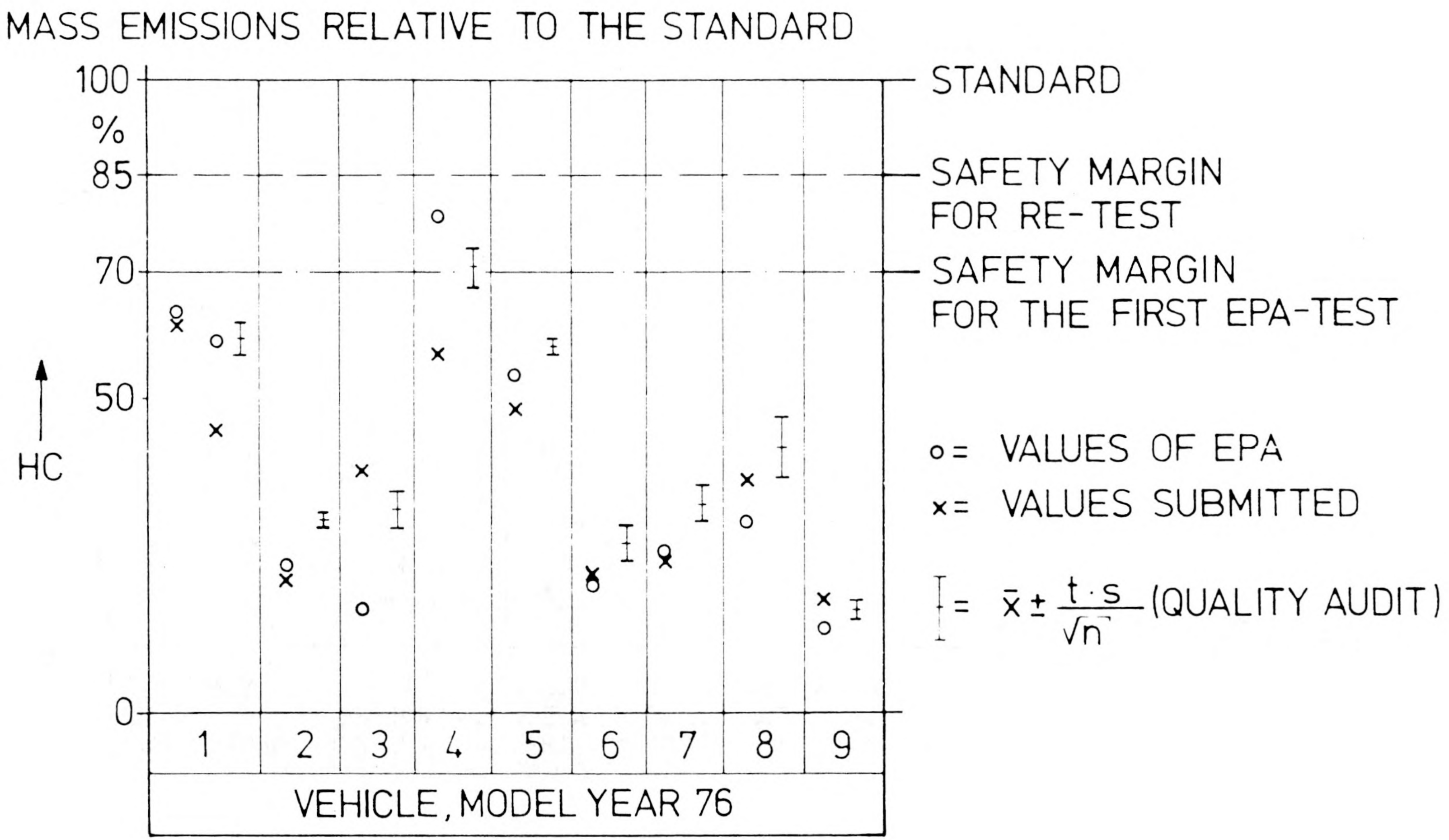


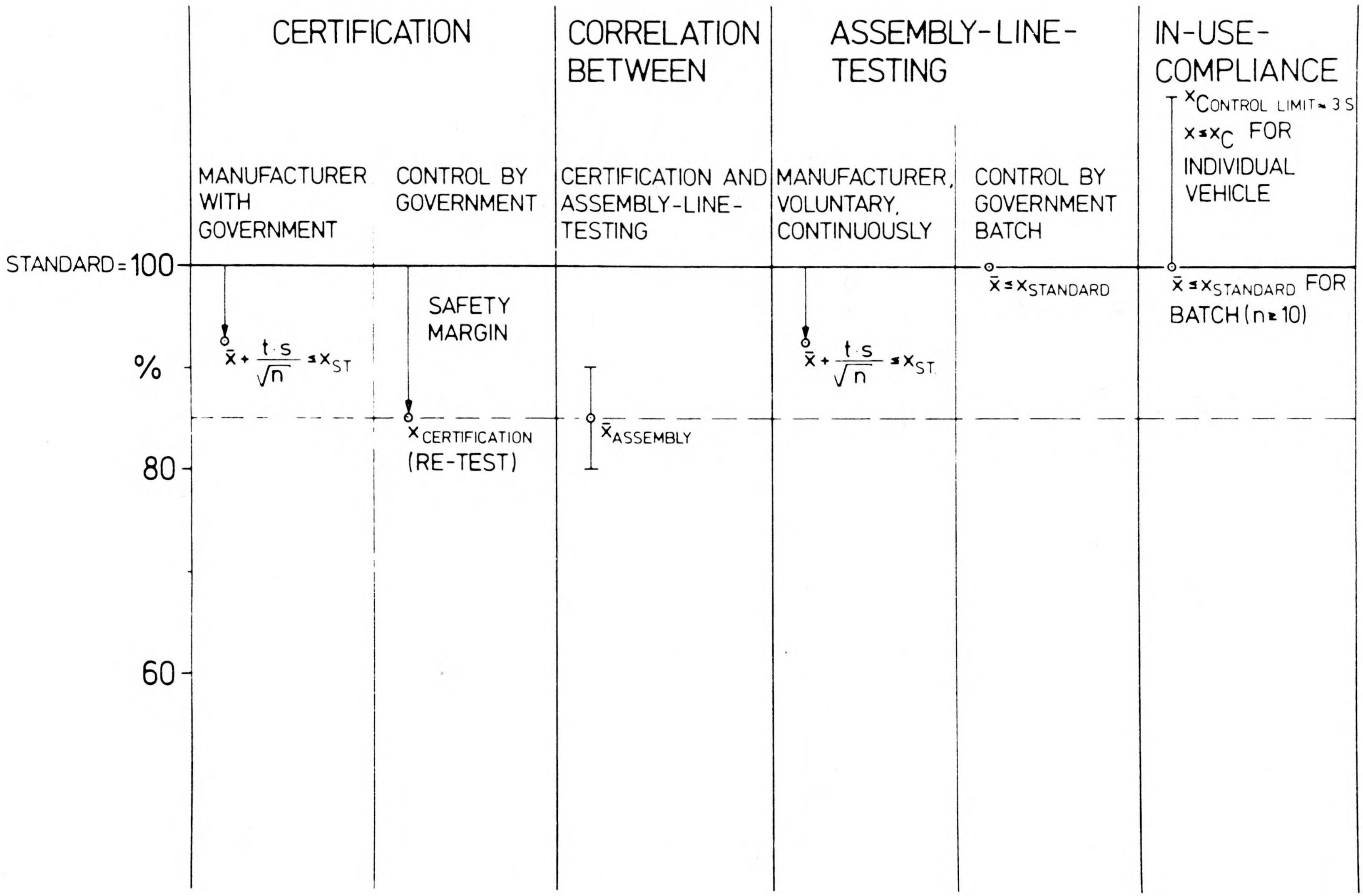
3.8
G/MI
3.4
3.0
HC
2.5
2.0
1.6
x = VALUES AT EPA
o = VALUES SUBMITTED
1)
2)
3)
1
2
3
4
TYPE
A B C D E F G H I J K L M N O P Q R S T U V
ENGINE FAMILY 1 2 3 4 5
1) = STANDARD WITHOUT DET. FACTOR
2) = ENGINEERING GOAL 74 WITHOUT SYSTEMATIC ERROR
$\bar{x}$ $\frac{2ts}{\sqrt{n}}$
FOR (x) VALUES
$\bar{x}$ $\frac{2ts}{\sqrt{n}}$
FOR (o) VALUES
3) = ENGINEERING GOAL WITH SYSTEMATIC ERROR OF 25%



ASSEMBLY
140
%
120
100
80
60
EU1 EU2
CAL1
CAL2 SEA
STANDARD + BIAS
(30% FOR HC)
STANDARD xLIMIT
xLIMIT/DF(1.15)


NO. OF VEHICLES
MEAN
HC
0 0,1 0,2 0,3 0,4 0,5 G/MI 0,7
MASS EMISSION HC
MEAN
NORMAL
LOG-NORMAL
WEIBULL
CO
0 2 4 6 8 10 G/MI 14
MASS EMISSION CO
NO. OF VEHICLES
MEAN
NORMAL
LOG-NORMAL
WEIBULL
NO_X
0 0,5 1,0 1,5 2,0 G/MI 3,0
MASS EMISSION NO_X


MASS EMISSIONS RELATIVE TO THE STANDARD
100
%
85
70
50
0
HC
STANDARD
SAFETY MARGIN FOR RE-TEST
SAFETY MARGIN FOR THE FIRST EPA-TEST
o = VALUES OF EPA
x = VALUES SUBMITTED
= x̄ ± t·s/√n (QUALITY AUDIT)
1 2 3 4 5 6 7 8 9
VEHICLE, MODEL YEAR 76


CERTIFICATION
CORRELATION BETWEEN
ASSEMBLY-LINE-TESTING
IN-USE-COMPLIANCE
MANUFACTURER WITH GOVERNMENT
CONTROL BY GOVERNMENT
CERTIFICATION AND ASSEMBLY-LINE-TESTING
MANUFACTURER, VOLUNTARY, CONTINUOUSLY
CONTROL BY GOVERNMENT BATCH
$x_{CONTROL\ LIMIT} \approx 3S$
$x \leq x_C$ FOR INDIVIDUAL VEHICLE
STANDARD = 100
%
80
60
$\bar{x} + \frac{t \cdot s}{\sqrt{n}} \leq x_{ST}$
SAFETY MARGIN
$x_{CERTIFICATION}$ (RE-TEST)
$\bar{x}_{ASSEMBLY}$
$\bar{x} + \frac{t \cdot s}{\sqrt{n}} \leq x_{ST}$
$\bar{x} \leq x_{STANDARD}$
$\bar{x} \leq x_{STANDARD}$ FOR BATCH $(n \geq 10)$

INSTITUT FRANCAIS DU PETROLE

Division "APPLICATIONS"

Département "Combustion dans les Moteurs Thermiques"

Mars 1977 - Rapport I.F.P. n° 24 939

Projet : B 61/73 019

DIESEL EXHAUST ODOR MEASUREMENTS

Fourth International Symposium on Automotive Propulsion Systems/Low Pollution Power Systems Development

Washington, D.C. USA April 18-22, 1977

P. DEGOBERT

A B S T R A C T

INSTITUT FRANCAIS DU PETROLE, under sponsorship of INSTITUT DE RECHERCHE DES TRANSPORTS, has developed a sensory analysis system able to get simultaneous responses of a selected nose panel of 12 people, sniffing engine exhaust diluted with purified air in known proportions-Measurements are made by matching the unknown exhaust odors with pyridine standards, choosen as better than standard more closely related to genuine exhaust odors, because of different individual preferences for certain Diesel exhaust odor types.

Simultaneously odorous pollutants are analysed by physico-chemical methods, FID for unburned hydrocarbons, MBTH for aldehydes, liquid chromatography on trapped exhaust following Arthur A. LITTLE DOAS procedure.

First measurements have been made on Peugeot and Saviem high and medium speed Diesel engines. The first results show that ADL DOAS liquid chromatographic data do not fit better but not worse with sensory olfactory results than unburned hydrocarbons and aliphatic aldehydes.

INTRODUCTION

In the United States, Diesel exhaust odors have been studied since 1962, and the Coordinative Research Council Cooperative Project CAPE-7-68 has supported since 1968 the different investigations started under sponsorship of the U.S. Department of Health, Education and Welfare.

CRC CAP-7-68 efforts have been focused towards designing an analytical instrument, manufactured by Arthur D. Little laboratories, and called DOAS (Diesel Odor Analytical System). By a liquid chromatographic technique, that instrument measures specific components of exhaust, previously trapped, and being correlated with the olfactory responses given on the same Diesel engine by a human nose panel.

Small Diesel passenger cars and small Diesel delivery trucks are already numerous in European cities, particularly in France, and their number is to be progressively increased. For they present the advantage to be more efficient in the conditions of city traffic, and to use Diesel fuel, which for the moment in France is sold at a lower price that gasoline. Therefore Diesel odor level is due to increase in European cities, although already high in some particular cities as Lisbon, Madrid, Athens or London, and in 1973, French Authorities and French Diesel car manufacturers decided to start an investigation program relative to Diesel engines and cars responsible for transportation malodors.

The Arthur D. Little instrument was just available at that time. It has been developped with an American Diesel engine, fueled with an American Diesel fuel. It was not sure that the correlations developped in such American conditions would apply directly to smaller European Diesel engines, fueled with European fuels and it was decided both to start sensory evaluation on Diesel odors and to purchase an Arthur D. Little DOAS instrument, to check if its analytical figures would fit directly with the French Diesel conditions.

For that reason, INSTITUT FRANCAIS DU PETROLE, has been asked by the French INSTITUT DE RECHERCHE DES TRANSPORTS and by French Diesel vehicle manufacturers PEUGEOT, and SAVIEM, RENAULT's subsidiary, to start an investigation program on exhaust odors.

The object of that research program is:

- To establish sensory evaluation methods to measure odors in reliable conditions ;
- To establish relationships between odor figures given by the sensory panel and some analytical features, properly choosen, ADL analytical figures for instance, in order to substitute, in the future, physical instrument for expensive sensory evaluation ;
- To study engine and fuel improvement to decrease exhaust malodor level and develop Diesel cars number without further nuisance for the population.

For the moment our work is still in its earlier stage. This paper will describe successively the sensory analytical system, which has been choosen, and built at I.F.P., the selection and training of the sensory panel and the sensory method we have developped for Diesel Odor evaluation. We will discuss the analytical methods and among them the ADL liquid chromatographic technique.

Our first measurements deal with odor exhaust measurement of a Peugeot 504 D engine on particular conditions, and allow us some provisionnal conclusions on the comparison between sensory responses and analytical features. We will end with the improvements of our method foreseeable in the near future and our future investigation program.

DESIGN OF THE ODOR PRESENTATION SYSTEM

Diesel exhaust has already been evaluated by sensory methods. Some reviews have been published on that subject (1).

Among the existing systems to deliver exhaust odors to the human nose, we may essentially distinguish 3 types of set-up.

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(1) Numbers in brackets designate References at the end of Paper

1) The so-called "camera inodorata", which is a cabin where the subject bathe completely in the odorous atmosphere. Exhaust is injected in the air, which subjects breathe. Such system is used by Arthur D. Little laboratory (2), and by Ricardo consulting Engineers (3).

2) The sniff-box technique, which consists in a facial mask, or a hole at his face dimension, in which the subject plunges his face, for the time of sniffing. The hole is set on a duct fed with the odorous mixture to be evaluated. Such systems were used by Scott laboratories (4) by the Karolinska Institutet (5), by General Motors (6) and by US Bureau of Mines (7).

3)The funnel-type odor presentation system as existing at Southwest Research Institute (8) (9) ; we will describe it now, as choosen also by our Institute.

Our decision to build an odor presentation cabin very similar to that installed at Southwest Research Institute in San Antonio was based on the following reasons.

To take into account the discrepancies of the verbal answers of each people taking part to the nose panel, it is necessary to repete each evaluation a sufficient number of times. In other words, we have the choice between repeting the same evaluation many times with a small panel, repeting the same points a few times with a greater panel, or making the entire evaluation at one time with a numerous population. From an economical point of view, the optimal number of panalists lies around a number of ten. The "camera inodorata" or the single sniff-port or sniff-mask technique demands successive evaluations of each panelist one after another. This needs to let the engine run in the same conditions during the whole time, when people pass in procession before the sniff-port, or enter successively the sniff-cabin, and as an engine is not always easy to keep stable, it is not sure that everybody evaluates the same odor. Moreover that procedure needs manpower to drive the engine and to monitor the panel for a longer time, and may demand a lot of special fuels not always easily available. On the contrary, the SWRi-type cabin allows the whole panel to sniff at the same time the same engine exhaust, during short sessions, demanding finally shorter occupationnal time for everybody.

For a question of location, and to get some mobility for future investigations, the presentation cabin is laid on a trailer. The walls, floor and roof of that specially made cabin were manufactured from odorless resin coated multiply wood, as used for meat or fish refrigerated delivery vans.

Figure n° 1 is an interior view of the panel room. It can house up to twelve panelists. The cabin is air-conditionned, at a temperature around 20°C, moisture content about 50%, with an air recirculation rate allowing complete renewal of the atmosphere every 3 to 5 minutes. Air is recirculated from the rectangular ducts hanging from the roof over the panelists' table.

Figure n° 2 shows the detailed appearance of a panelist'sbooth. The presentation funnel delivers diluted exhaust picked in parallel on the main duct located under the table. When people are not sniffing, an inversed funnel covers the inferior funnel, to suck out odorous air and avoid any pollution of the room. Moreover a butterfly valve at the end of the main delivery duct brings a constant positive overpressure VS the room, and leads to a constant flow in the funnel at the time of sniffing, flow choosen as the most convenient for sensory evaluation. The butterfly valve is opened and the flow suppressed during intervals between evaluations.

Flow sheet of the air is represented on figure n° 3. Air is used both to condition the cabin and to dilute the exhaust gases for presentation to the noses. Flow of preconditionned air is separated in two streams, one for conditionning cabin atmosphere, a part of which is recirculated, a second to dilute exhaust gases sucked from the exhaust manifold of the engine. That second stream is purified by a charcoal canister just before mixing with the exhaust gases.

Dilution ratio of the exhaust gases is measured and regulated by orifices and remote control pneumatic valves installed both on the air stream and in the exhaust gas stream before dilution. A control panel standing in the evaluation room allows to choose the proper ratio air vs exhaust gas, which is automatically regulated. The exhaust line between the picking-up point on the exhaust manifold and the dilution point is about one meter long with an inside diameter of 15 mm. In order to avoid any condensation of the exhaust components, before quenching by the dilution air, the whole line with its metering orifice and its pneumatic valve is kept between 180°C and 200°C. Flow of picked-up exhaust lies between 1 to 2 cubic meter per hour (about 0.18 to 0.36 SCFM). Flow of dilution air varies between 50 and 1 000 cubic meter per hour (9 to 180 SCFM). Therefore the possible dilution range lies between 1:25 and 1:1,000.

Figure n° 4 shows an overall niew of the system. In front of the presentation cabin stands the air purification unit, containing the filters, the blower, the heaters, the refrigerating unit and the first charcoal canister. Purified air is for one part injected in the nose panel cabin for air conditioning, the other part is pushed next to the engine test bench through the pipes visible on the foreground and one of the three possible metering orifices. One pipe is used for fresh air, the other to return diluted exhaust gases to the cabin.

The dilution point is visible on the next figure n° 5. Close to the engine test bench on the right, stands the charcoal canister in which enter simultaneously fresh air from the large pipe and exhaust gases through the isolated and heated small diameter pipe at the lower part o the canister. The foreground blower is used to suck out exhaust from the exhaust manifold and to mix it properly with the air before pushing the mixture towards the evaluation cabin. Diameter of the air pipes is about 300 mm (about one foot).

DESCRIPTION OF THE SENSORY METHODS

Nose panel has been selected among I.F.P. staff, following methods previously developped in the U.S.A. by Amos Turk (10). We chiefly used methods suitable to detect people's hability to compare odor intensities of the same character as screening procedure, such as triangle test and intensity rating test, but we discarded the multicomponent odor identification test. Our aim is not to give an odor profile of the Diesel exhaust, but is to measure odor intensities. We also look for getting a measure of the hedonistic value of the sniffed gases by asking panel people not only to give an answer concerning the intensity of the odor but also an answer about the nastiness they feel when submitted to the same odor.

During the training session of the panel, we looked for finding the most suitable intensity standard to be compared with the actual exhaust and the best nastiness standard. As standards, we discarded the complex mixtures proposed in the TURK kit, because odorous balance between the different components of such complex mixtures may change with their different dilution ratios, and we restricted our choice inside pure chemical compounds, more likely to keep their genuine odor with dilution.

The standards we tested were similar to Amos Turk's but pure chemicals:

- heptaldehyde for aldehydic-type odors
- carvacrol for smoky-burnt type odors
- crotonic acid for pungent type odors.

Unluckily we never found any nasty pure chemical, not fragrant, to represent oily-type odor and we added to the first three standards, pyridine as proposed standard. For pyridine is reputed to be recognized as bad by everybody.

We evaluated the reliability with which selected sensory judges were appreciating about ten different bad odors, taken in the odor-type represented in Diesel exhaust (aldehydic, smoky-burnt, oily, pungent). Two methods were used, an estimation method close to the category scaling used in psychophysics, and a paired comparison method or matching method. By comparing the different evaluations of the judges, referred to one of the four proposed standards, we saw that evaluations versus pyridine were the most stable for the majority of the nose panel tested, and we decided to choose pyridine as the best standard available to evaluate odor intensities.

To evaluate the nastiness of the same range of bad odors, by the same methods (estimation or paired comparisons methods), we were unable to find a suitable standard common for the majority of the panel. For the sensation of objectionnability of somebody towards odors, varies from one to another, and no general tendancy may be found. So we let people give their impression of nastiness towards diluted exhaust without any term of reference, on their self category scale.

After this training period, in the mean time, the sensory installation was ready for the first actual engine olfactory evaluation. Therefore for our first experiments, we choose a matching method to evaluate intensities, and the category scale for nastiness.

To evaluate intensity of the exhaust presented in the funnels, people had to compare it with a set of ten sniff bottles containing pyridine diluted with pure water. Those bottles were arranged in geometrical progression, i.e. one bottle had half the concentration of its right neighbor and a double concentration of its left neighbor. Judges were asked to give the number of the sniff bottle, giving the odor the closest to the odor of the funnel.

To evaluate nastiness of the exhaust odors, panelists are asked to express their objectionnability on a five point category scale as follows.

1	2	3	4	5
NOT AT ALL	A LITTLE	MEANLY	VERY	EXTREMELY

All responses from panelists were written on standard IBM computer punching cards, easier for further treatment, such as decoding, variance analysis, etc.

Sensory evaluation sessions are always programmed in the early morning, before people start their regular jobs, and before they get olfactory fatigue. Duration of a test session is about one hour. That times allows to present about 12 different odors at 5 minutes interval. That interval is necessary to reset new engine operating conditions, and get stable operation of the engine. It avoids also olfactory adaptation, which may alter the responses, if the olfactory mucosa had not the time to "forget" the precedent odor.

All engine operating conditions are repeated or presented in randomized order, and for the first evaluation sessions, 12 repetitions of each conditions were provided.

PHYSICO-CHEMICAL ANALYSIS OF THE EXHAUST GASES

Following analytical methods are applied to the exhaust gases:

- Non dispersive infrared spectroscopy for carbon monoxide and carbon dioxide
- Chemiluminescence for nitrogen monoxide
- Flame ionisation detector for unburned hydrocarbons

- MBTH colorimetry for aliphatic aldehydes
- Liquid chromatography for LCO and LCA fractions, following Arthur D. Little type modified procedure.

In the case of unburned hydrocarbons evaluation, up to now, transfer of exhaust gases from the exhaust manifold to the analyser has occurred through the commercial hot line sold with the FID-instrument. Temperature of that line does not reach the 180°C-200°C temperature range and analytical results may be suspected.

Aliphatic aldehydes are measured by the conventional MBTH colorimetric method. But to avoid any loss of aldehydes through absorption in the water condensed from the exhaust in the transfer line to the wet traps, a glass capacity easy to rince before analysis is placed close to the exhaust pipe, at a distance less than 20 mm. Rincing solution is added to the absorbing solutions from the traps. That procedure affords a great reliability in the measurements.

Aromatic and aromatic oxygenates fractions are determined by a liquid chromatographic technique, which is a modification of the original method proposed by Arthur D. Little, developped in its DOAS, Diesel Odor Analytical System. A DOAS instrument was ordered by our laboratory at the beginning of the research program, but awaiting its delivery, our liquid chromatography laboratory, under leading of Mrs Petroff, began to test the analytical method as published in ADL reports (11). We rapidly establish that in the original procedure the sudden injection of the eluent isopropanol in the cyclohexane chromatographic loop, suddenly displaces the oxygenate impurities always present in cyclohexane, even as pure as possible. That sudden displacement of those oxygenates previously adsorbed on the chromatographic column leads to erratic blank values, almost of the same magnitude as the unknown values to be evaluated. When our ordered DOAS arrived in our laboratory, we noticed the same phenomenon, as well as the other European users of DOAS instruments (12).

Meanwhile, our analytical laboratory had developped a method avoiding to suddenly disturb the steady conditions existing in the chromatographic column, relative to the adsorbed oxygenates coming from the impurities. That method consists in filling the chromatographic loop at the beginning of the experience with cyclohexane containing already a small amount of isopropanol (about 0.1%). That low content of isopropanol displaces continuously the oxygenate impurities from the cyclohexane which tend to adsorb on the chromatographic column ; the lower permanent UV response due to those impurities is taken into account in the corresponding blank value. Moreover that low content of isopropanol, once adjusted, is sufficient to afford a proper separation of the two peaks, one for LCA, the other for LCO, appearing successively in the detector, after injection of the unknown sample in the chromatographic loop. In the genuine ADL procedure the second peak, LCO peak, appears only after injection of isopropanol in the loop.

That modified procedure gave us better results than using the original ADL procedure. For in that case to avoid erratic blank values, due to cyclohexane sensitivity to air oxygen, it is necessary to inject isopropanol at constant intervals, what is not provided in the automatic instrument as delivered by Arthur D. Little. Moreover some drift in the electronic circuitry of the instrument, specially in the integrator, made its use very inconvenient and we decided to keep on using our own modified procedure.

The other European users of DOAS instruments met the same kind of troubles. RICARDO changed the electronic circuitry and added to its instruments an automatic valve which injects isopropanol at constant timing to get stable blank values. The same trick has been choosen by PERKINS-DAIMLER-BENZ starts from ultropure cyclohexane, and cleans it again thoroughly just before beginning the liquid chromatographic analysis by percolation on special ultrapure aluminium oxide. All those tedious operations are by-passed in our method.

To collect the samples to be submitted to liquid chromatographic analysis, we use the same procedure as Arthur D. Little. As shown on Figure n° 6, exhaust gases are first filtered on a large area glassfiber filter (about one sq.ft) at the back of the sampling device. That filter is heated at 180°C-200°C and is followed by a metal-bellows pump properly isolated to keep hot the gases, before entering the traps put in parallel on the foreground. The sampling train ends with water-separator and wet gas meters. The two traps contain each 10 grams of chromosorb 102, collect each 500 liter of exhaust and allow to get two simultaneous analytical results.

After collection, active components adsorbed in the traps are eluted by pentane or cyclohexane and the resulting solutions are analysed following previously described method.

APPLICATION OF THE SYSTEM TO THE OLFACTORY ANALYSIS OF DIESEL EXHAUST

Up to now, the system developped at I.F.P. has been used to test the olfactory level of a PEUGEOT 504 D engine in certain steady conditions. The study of a SAVIEM 720 engine generally installed on small delivery vans, has just begun.

The aim of that first set of determinations, was not only to test the olfactory level of the engine, but also to test the ability of our selected nose panel to perform correctly in sensory evaluation of Diesel exhaust, by giving reliable answers about presented exhaust odors. A second purpose was to get some ideas about the possible relationship between olfactory responses and analytical results about exhaust gases.

TEST PLAN AND PROCEDURE

The first engine studied, PEUGEOT 504 D, is a 2.1 liter 4 cylinder Diesel engine, compression ratio 22.2, fitted with a Ricardo-type prechamber of 11.4 ml. This engine is mostly used for taxi cabs.

We selected eight characteristic operating conditions, to be submitted both to olfactory evaluation and to physico chemical analysis. Selection of those eight conditions, all steady state conditions, is based either on extreme values of concentrations of typical compounds present in exhaust, or on typical operating conditions to be met in normal operation of the engine.

They were all steady state conditions, for as yet our system is not ready to evaluate transient conditions deceleration or acceleration.

Table 1 describes those eight operating conditions tested - 16 sensory assessors were judging the exhaust, which was presented with three dilution ratios (1:100 - 1:200 - 1:400). The dilution ratio was held constant during each session, and the eight conditions were presented in random order with alternative repetitions of three of them. For the whole test plan, each operating condition at a given dilution ratio has been presented 12 times. Overall we collected 192 answers (12 repetitions x 16 people) pertaining both to intensity and to nastiness.

RESULTS AND DISCUSSION

Panel answers were decoded and variance analysis applied both on individual and on global panel responses. Some other treatments, such as elimination of some too variable assessors, recentering of responses by changing origin of notation, transform of exponential scale to linear scale were also applied. Results of ranking of the engine operating conditions are given on Table 2. That table presents on a linear scale the ranking of the arithmetic means of the panel responses corresponding to the perceived intensity of each operating conditions. Answers were got by matching with a pyridine scale and results are based on 192 repetitions. Arrows on the figure indicate means not statistically different at a 5% confidence level.

The first thing to remark is that the ranking order is practically the same at the three dilution ratios. The magnitude of odorous intensity felt by the judges increases with the concentration of the exhaust gases, but the gap is greater between 1:100 and 1:200 than between 1:200 and 1:400. This fact was not obvious, as all operating conditions has been presented in randomized order. It is also to be remarked that the range of the intensity answers is relatively narrow, there is just a factor three between the less odorant and the most odorant operating conditions.

The most intense exhaust odor in steady state conditions are to be found at high speed , high load just before partial load and low speed. Minima of intensity correspond to full load, low speed and half load, low speed.

Overall PEUGEOT 504 D is not a very odorant engine, nor very nasty as shown on Table 3.

On Table 3 are represented arithmetic means of all the nastiness responses corresponding to the eight operating conditions. They were calculated by assuming number 2 to A LITTE and number 3 to MEANLY before calculating the means. Even if that calculation is not rigourous, it shows that all the responses vary between SLIGHTLY and MEANLY objectionnable. Although our panel was trained to distinguish in a presentation the odorous intensity from the hedonistic character, ranking appears to be the same as for intensity. Nastiness increases with the concentration of the exhaust, but at greater dilution, the gap between the two dilution ratio becomes negligeable.

That leads to remark that the engine we tested, with a Ricardo type prechamber does not offer great variations in odor intensity and odor quality, in any steady state operating conditions. In those conditions, people are led to confuse the odorous intensity and the nastiness, as shown by the close correlations between rankings.

RELATIONSHIPS BETWEEN ODOR INTENSITY AND CHEMICAL ANALYSIS

The relative importance of several analytical data (unburned hydrocarbons, SO_2 , NO, aliphatic aldehydes, LCO and LCA) is illustrated on Table 4, compared with the mean intensity responses for the three dilution ratios. The importance of each feature corresponds to the relative dot area for all the diagrams brake mean effective pressure BMEP in kg/cm^2 versus engine speed in RPM. Unit corresponding to each feature is given at the right lower angle of each diagram.

Table 4 shows no relationships between odor and SO_2 or NO levels. Both SO_2 and NO maxima correspond to high load at any speed, but in high load conditions odors is more intense at high speed.

Smoke and odor follow opposite tendancies: when we compare operating conditions 2 and 3, odor minimum corresponds to smoke maximum and conversely.

Comparison between odor and the other analytical data is more controversial.

Odor and unburned hydrocarbons seem to be closely related, put point number 8 (2,000 RPM, 1:10 load) with the highest hydrocarbon level is not the most odorous.

The relationship between odor and aliphatic aldehydes seems to be fairly good, but here again point 8, with a maximum in aldehydes is not the most odorous and point 1, second for aldehydes is felt just as moderatily odorous.

The discrepancies between odor and LCO level and LCA level are even greater. Here again, point 8 heads the list for LCO and LCA but not for odor. Point 2 presenting the more intense odor gives just moderate responses in odor give distinct values for LCO and LCA. It is also to be remarked that LCO and LCA are themselves closely related.

The correlation coefficients between odor intensities and the last four analytical data have been calculated and are given at the head of Table 5. Unburned hydrocarbons give for the present time the "less bad" relationships with the odor intensities, anyway not worse than the relationship between LCO and odor as applied in ADL DOAS liquid chromatographic system. K.J. SPRINGER as early as 1972 (13) had already shown, when ADL DOAS system was not yet available that among all odor predictors, HC exhaust level was the best, but that it was not possible to rely on one single emission measurement , to give an exact pattern of the odorous emissions.

Starting from data given by K.J. SPRINGER (14), who later submitted four European and Japonese cars, to sensory evaluation, to physico-chemical analysis and to liquid chromatographic analysis through ADL DOAS instrument, we calculated the same correlation coefficients, given at the end of Table 5.

Table 5 gives correlation coefficients for chemical determinations and odor intensity as expressed by "D rating", i.e the matching with the exponential scale of the TURK Kit (10), consisting in the complex mixture of all component to simulate Diesel odor*. We also calculated correlation coefficient between analytical data and individual odor notes as reported by K.J. SPRINGER (Smoky-burnt, Oily, Aldehydic, Pungent), not given in Table 5.

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*Like our own sensory data "D" rating have been first put on a linear scale, as they were expressed on an exponential scale corresponding to sniff-bottle in geometric progression.

Surprisingly neither aliphatic aldehydes nor LCO are correlated at all with the A rating, corresponding in principle with the aldehydic note.

Considering the D ratings given in Table 5, we see that, except for the Mercedes 220 D, the less bad relationships existing between sensory data and analytical data concern unburned hydrocarbons. LCO is poorly related to odor, still worse than in the case of our own results. Mercedes 220 D is an exception, correlation with hydrocarbons does not exist, but the correlation with LCO is fairly good. When mixing all data concerning the five cars, here again HC show the best relation with odor, but LCO shows the worse.

CONCLUSION AND FUTURE PROSPECTS

The first sensory determinations with the equipment installed in our laboratory show the correct operation both of the sensory set-up and of the selected nose panel.

The first engine tested PEUGEOT 504 D is, in the explored experimental conditions, neither very odorant, nor very nasty.

For the present time, the Arthur D. Little DOAS is not a reliable "odormeter", nor any other analytical determination of the exhaust gases.

Still further investigations are needed to find empirical correlations to apply to a certain number of analytical data concerning Diesel exhaust gases, to be able to predict odor in all the operating conditions of an engine.

So sensory testing has to be continued in the future awaiting development of a perfect "odormeter". In parallel with the type of investigations described in that paper and applied to other Diesel engines and other operating conditions, our hope is to use in the future reliable olfactometer such as invented and developped by Dr. P. MAC LEOD (15). That device, now in developpment stage, will next be tested in our laboratory on Diesel engines and compared with a conventionnal nose panel. By using a physiological property pertaining to the differences of olfactory sensations from the two nostrils, it would have the advantage for a panelist to answer un ambignously about the odor presented. That would avoid repetitions or numerous panels until now necessary to get reliable olfactory evaluation, and the cost of such a system, operated by one single nose expert, would advantageously compete with any "odormeter" based on chemical determinations.

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- Mr. PASQUEREAU for exploiting the results
- The I.F.P. staff who lent their noses and took part with great sincerity to those very often tedious tests.

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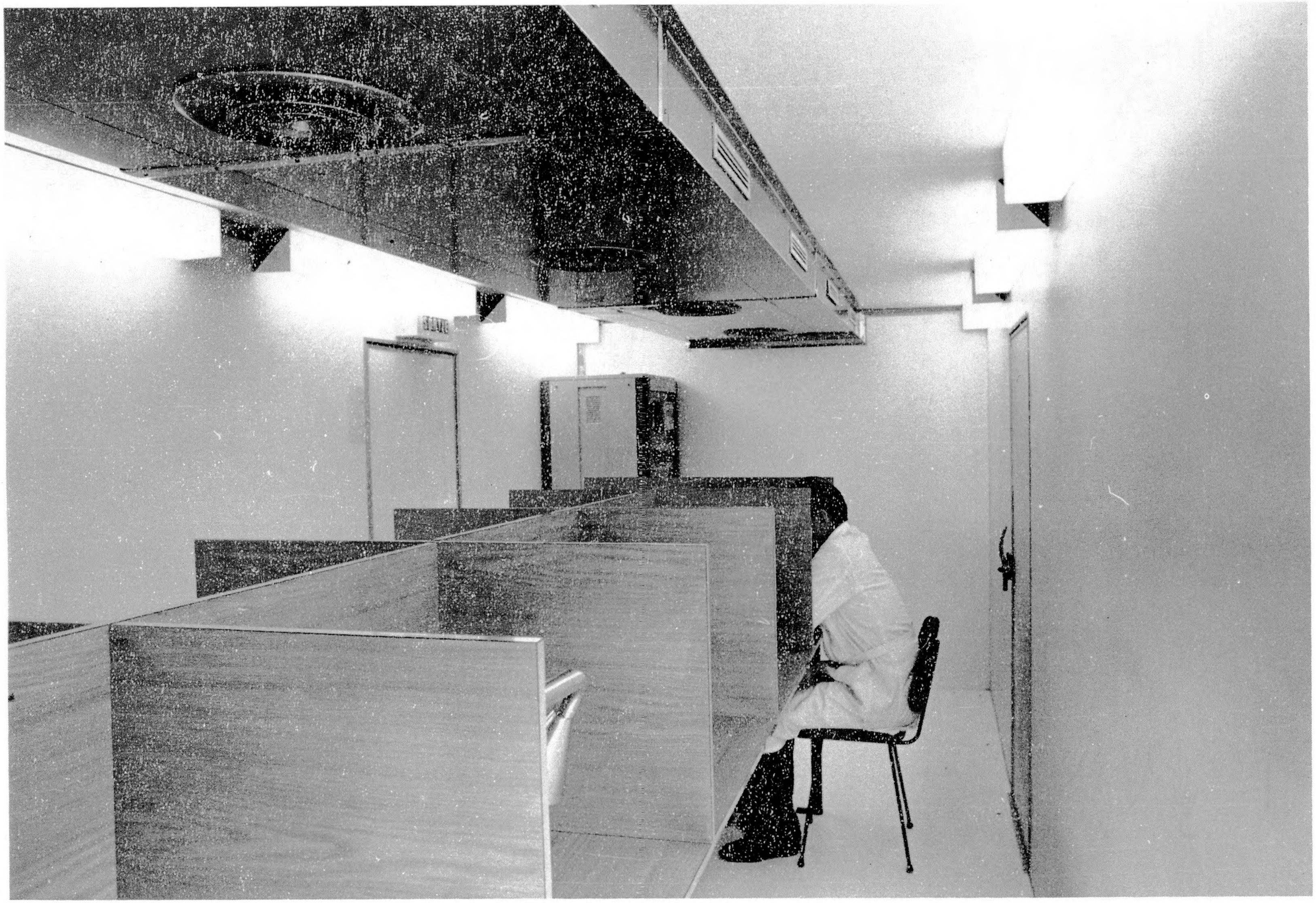
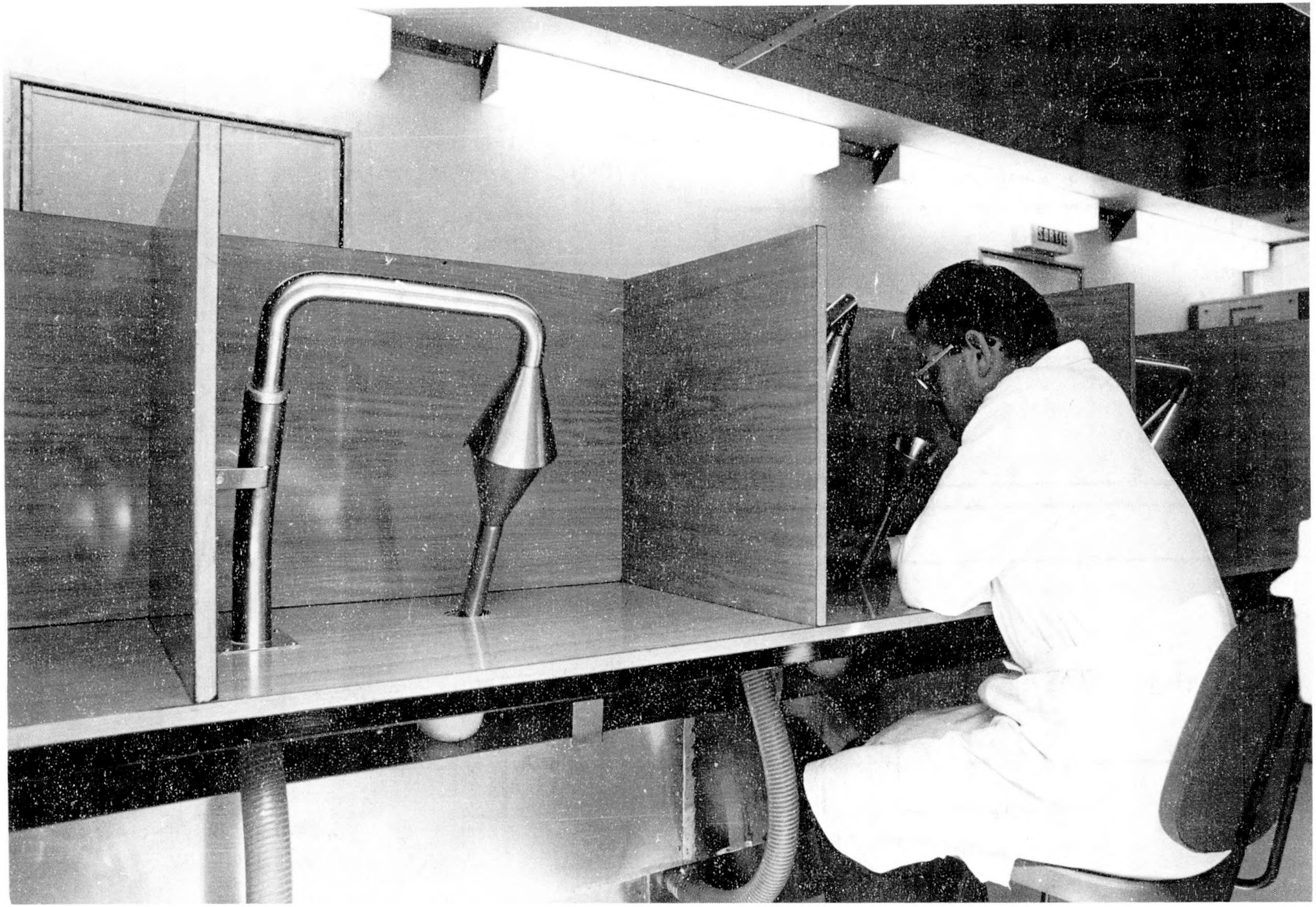
LIST OF FIGURES AND TABLES





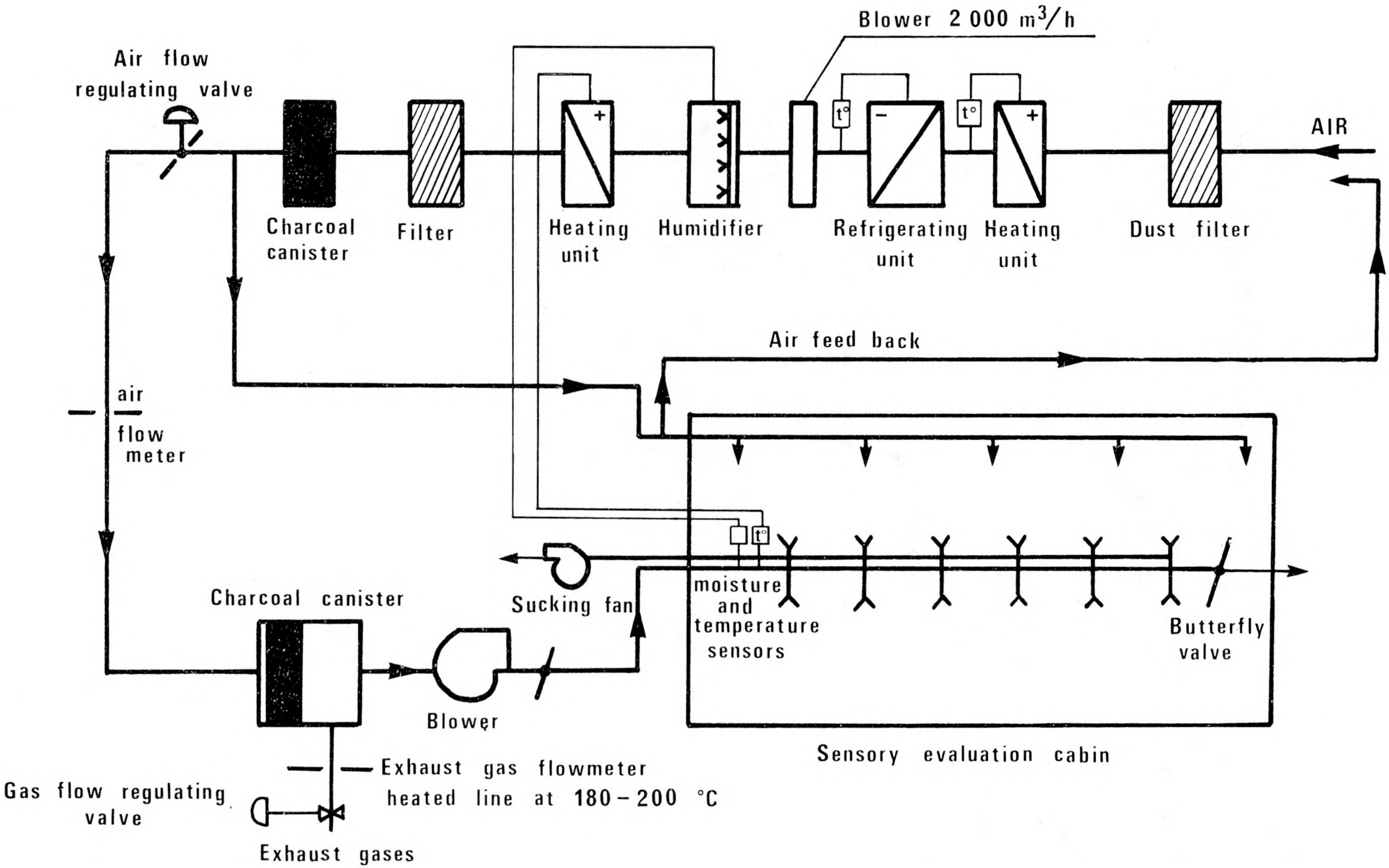
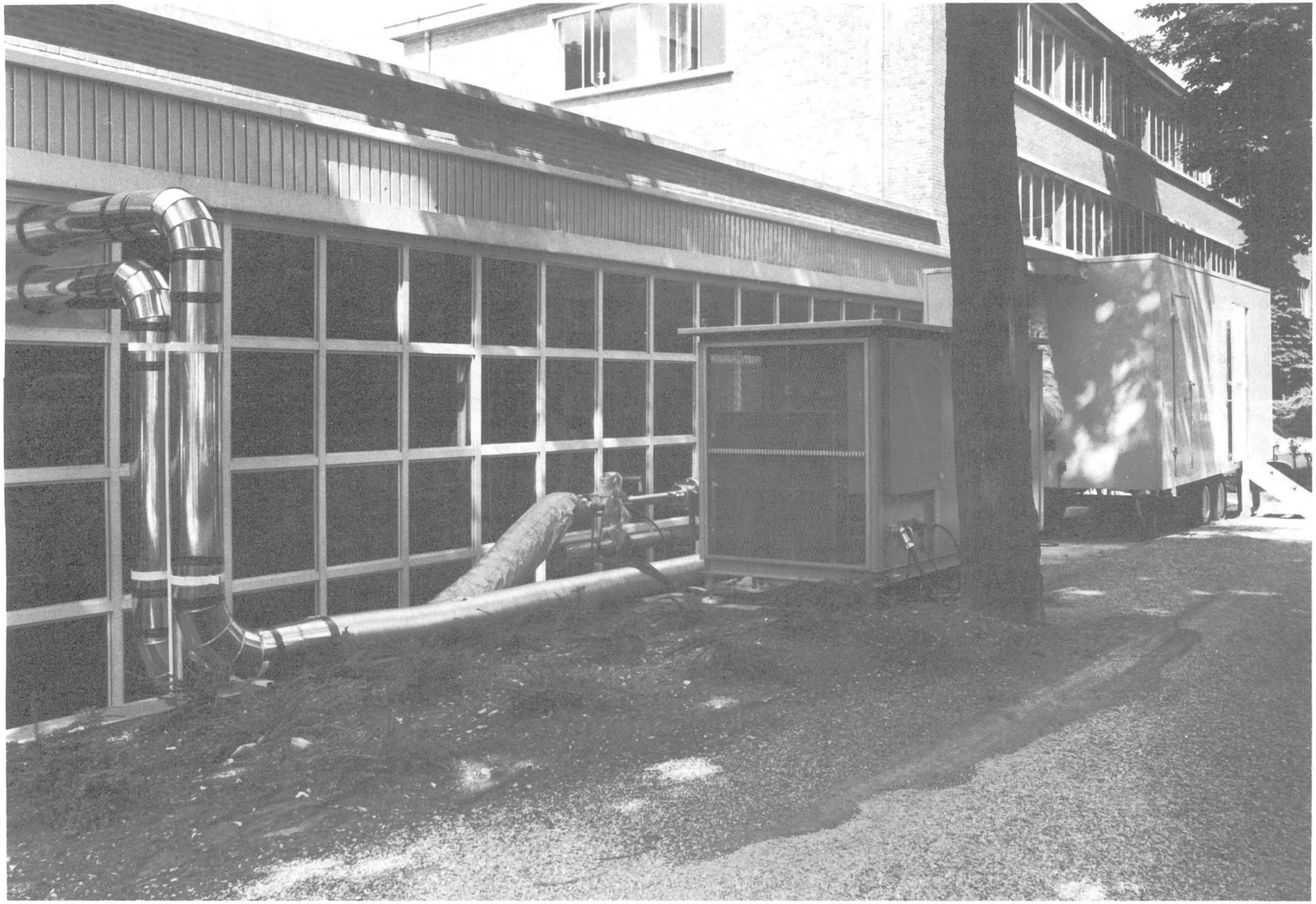
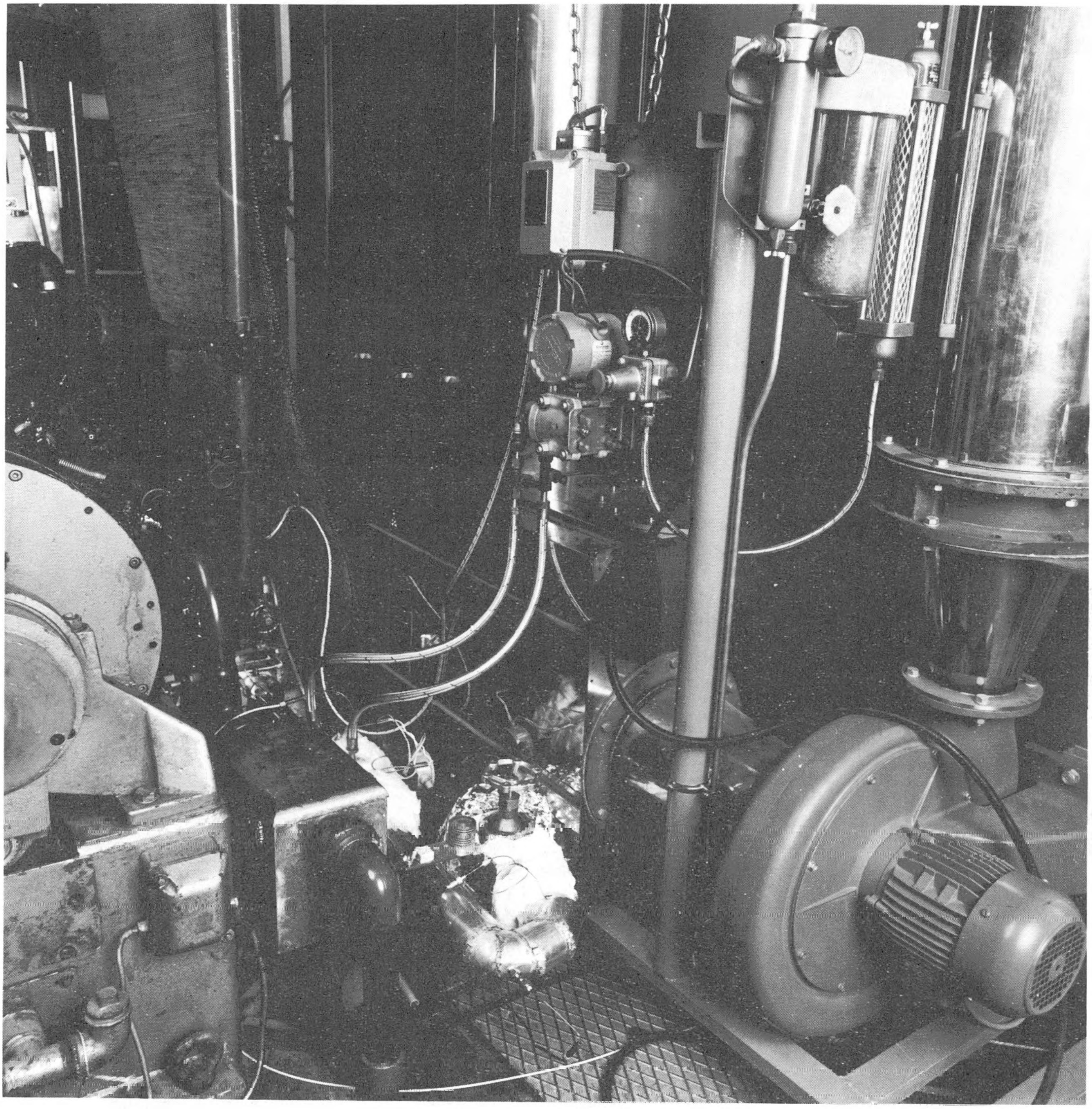

Blower 2 000 m3/h
Air flow
regulating valve
AIR
Charcoal
canister
Filter
Heating
unit
Humidifier
Refrigerating
unit
Heating
unit
Dust filter
Air feed back
air
flow
meter
Charcoal canister
Sucking fan
moisture
and
temperature
sensors
Butterfly
valve
Blower
Sensory evaluation cabin
Exhaust gas flowmeter
heated line at 180 – 200 °C
Gas flow regulating
valve
Exhaust gases

Figure 3





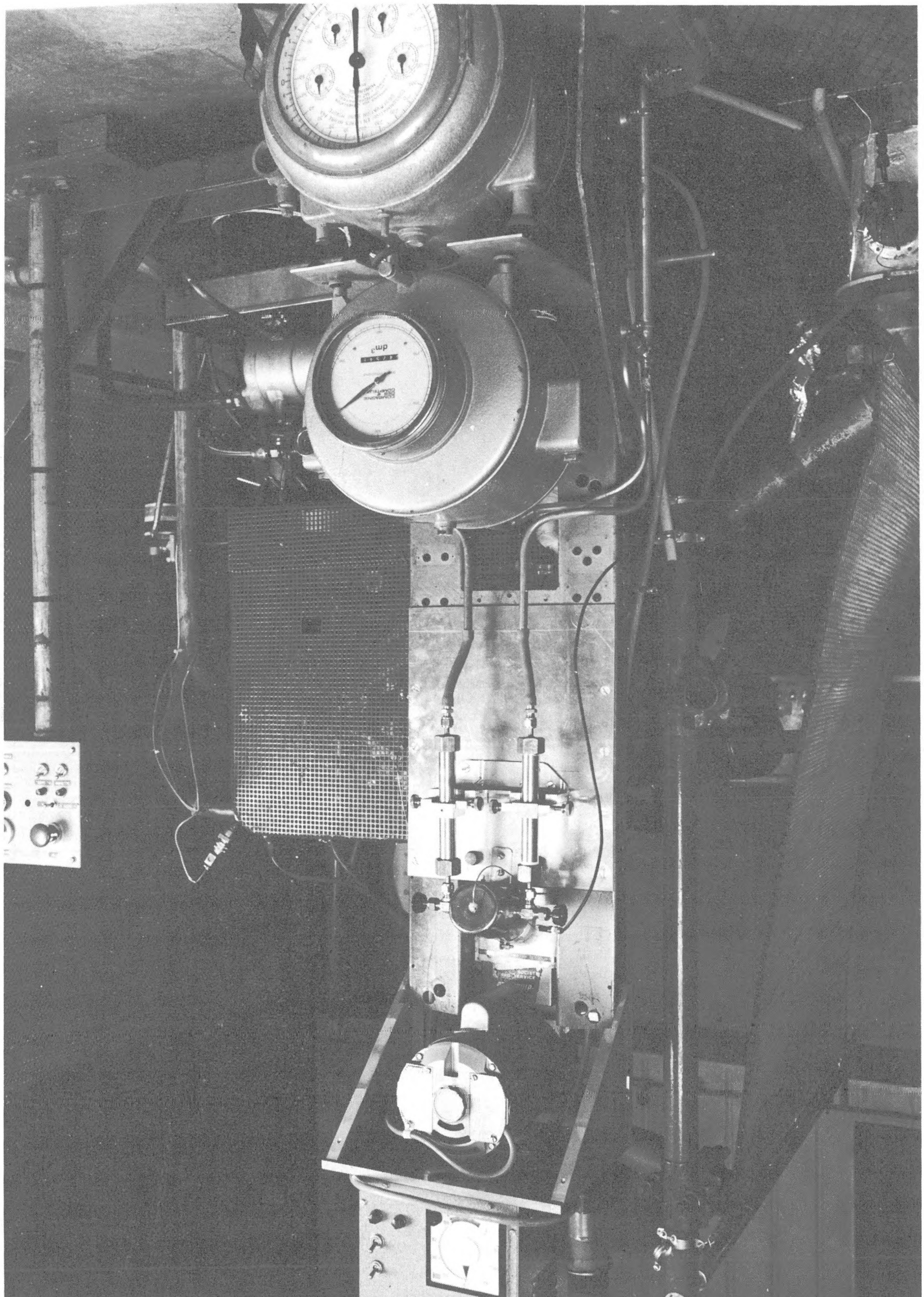


TABLE 1 -

CODE	ENGINE OPERATING CONDITIONS				
	SPEED RPM	LOAD	POWER HP	BEPM kg/cm^2	NOTES
①	1,000	0	0	0	∽ IDLE
②	1,500	1 : 4	6	1.7	MIN. SMOKE
③	1,000	4 : 4	12	5.11	MAX. SMOKE
④	3,000	1 : 4	12	1.7	MIN. SMOKE
⑤	3,000	4 : 4	42	5.11	ACCEL. GENTLE SLOPE
⑥	4,500	4 : 4	54	5.11	FULL LOAD MAX. HC
⑦	1,000	1 : 4	6	2.56	MIN. HC
⑧	2,000	1 : 10	3	0.64	RUNNING ON THE FLAT

TABLE 2 - **RANKING OF THE ODOROUS INTENSITIES**

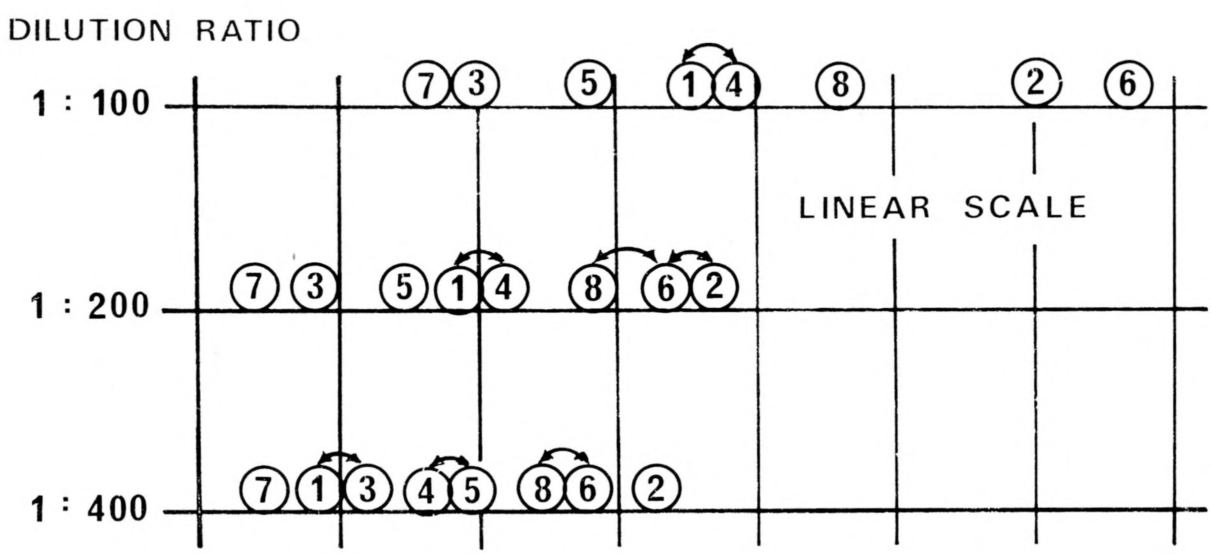
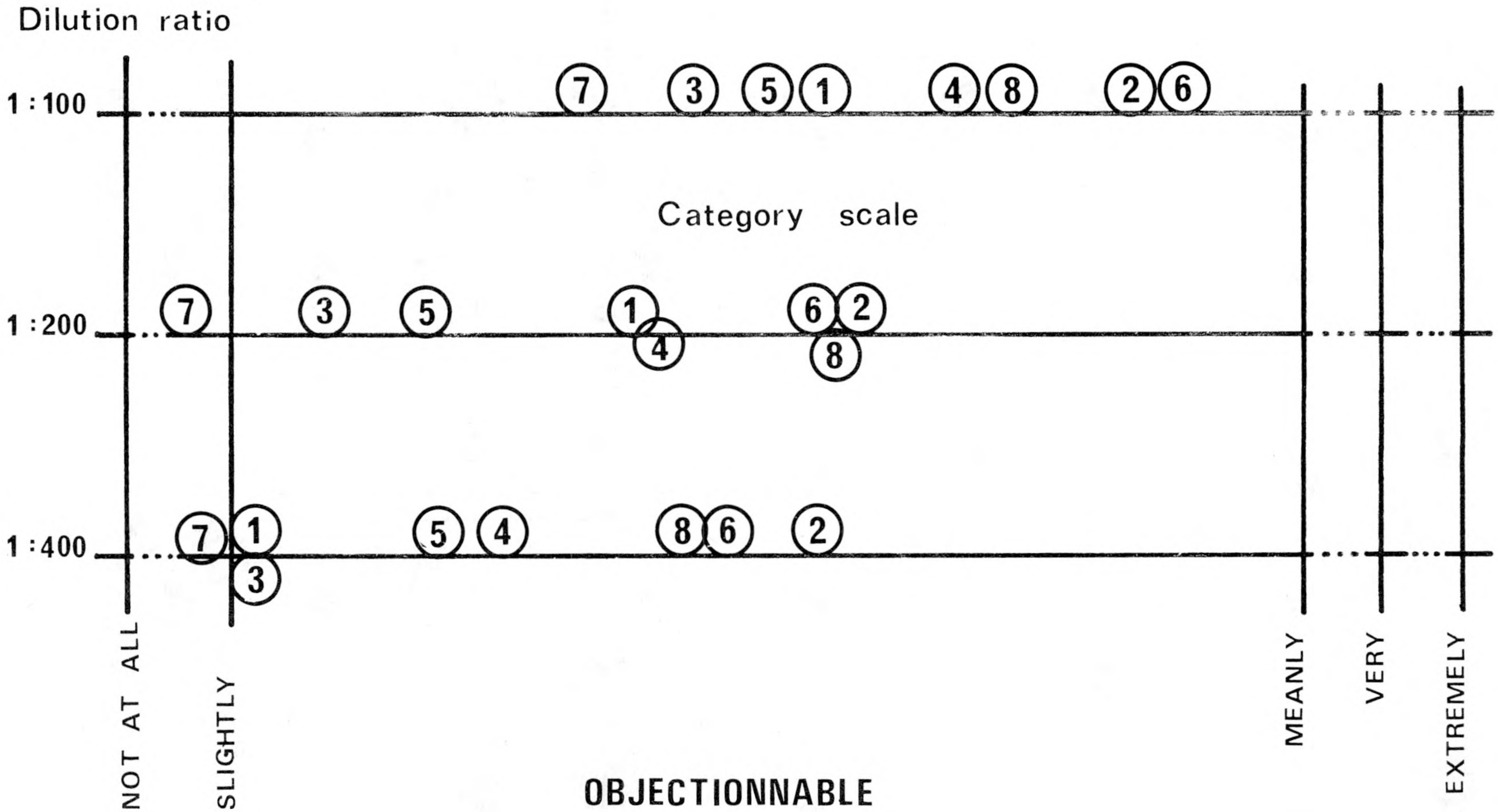


TABLE 3 – RANKING OF NASTINESS OF THE EIGHT OPERATING CONDITIONS



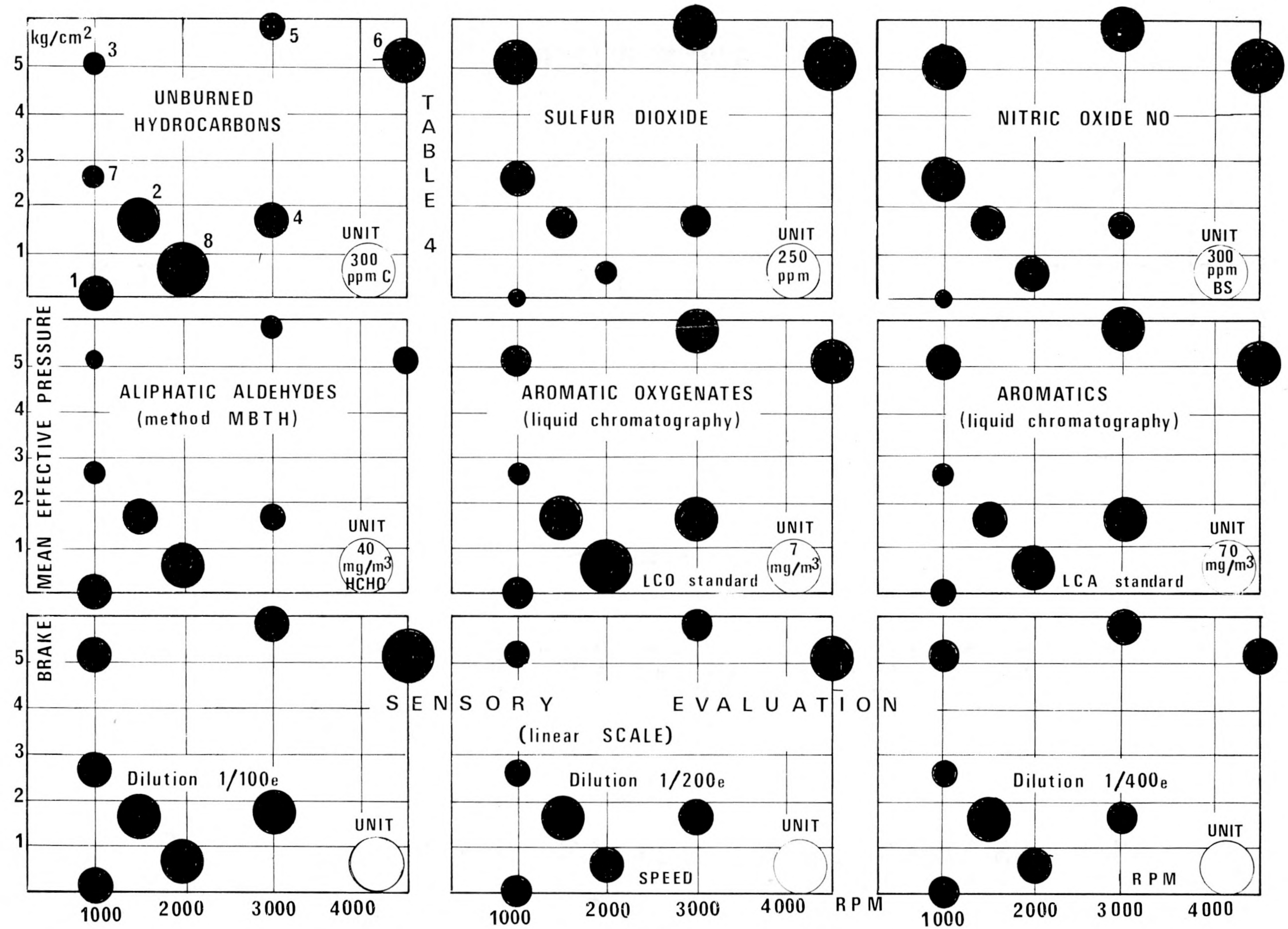

kg/cm2
UNBURNED HYDROCARBONS
UNIT 300 ppm C
SULFUR DIOXIDE
UNIT 250 ppm
NITRIC OXIDE NO
UNIT 300 ppm BS
ALIPHATIC ALDEHYDES (method MBTH)
UNIT 40 mg/m3 HCHO
AROMATIC OXYGENATES (liquid chromatography)
LCO standard
UNIT 7 mg/m3
AROMATICS (liquid chromatography)
LCA standard
UNIT 70 mg/m3
BRAKE MEAN EFFECTIVE PRESSURE
SENSORY EVALUATION (linear SCALE)
Dilution 1/100e
Dilution 1/200e
Dilution 1/400e
UNIT
SPEED RPM
1000 2000 3000 4000

TABLE 4

PROGRAMMABLE CONTROLS -

A CONTROL ENGINEERS DREAM

A.N. THATCHER
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LIST OF SYMBOLS AND ABBREVIATIONS

α	Accelerator Pedal Angle
εNgg	Gas Generator Governor Error
F.M.V.	Fuel Metering Valve
G.C.A.	Gain Controlled Amplifier
I.G.V.	Inlet Guide Vane
Ngg	Speed, Gas Generator
Nos	Speed, Output Shaft
PAMB	Ambient Pressure
Ro	1st Special Purpose Store
R1	2nd Special Purpose Store
R.O.M.	Read Only Memory
T1	Intake Temperature
T5	Compressor Turbine Inlet Temperature
T8	Regenerator Inlet Temperature
TC1	Thermocouple 1
TC2	Thermocouple 2
TC3	Thermocouple 3
TAMB	Ambient Temperature
Vv	Vehicle Velocity

ABSTRACT

Avoiding the use of specialist programmers in engine development can materially reduce development time scales. The Programmable Analog Controller, PAC, is readily programmable by non specialists due to its high level language. The programming language and operating principles of the production PAC 500 are discussed. Applications to the Chrysler - ERDA automobile gas turbine engine program and to a diesel engine demonstrator for low emissions are discussed. Other applications are mentioned which demonstrates the wide capabilities of the PAC with little change to production hardware. It is concluded that the use of the controller provides significant benefits especially in situations where the total quantity of units is not large.

INTRODUCTION

It is becoming more obvious that Engine Control tasks are now too complex for mechanical controls to handle on their own. Therefore control of a wide range of Industrial and Aerospace prime movers by the use of dedicated analog electronic units is now an established fact. Apart from the ability to perform complex control tasks with improved accuracy and speed of response, their use in a development role has resulted in a significant reduction in system development time. The reduction is difficult to quantify over mechanical systems, but is nevertheless universally recognised.

Changes occurring in engine and system development may generally be divided into three broad areas for electronic controls.

1. Trim adjustments, usually by means of potentiometers, which can often be done on-line.

2. More complex modifications which involve the physical replacement of circuit components, yet still retaining the same basic control loop.

3. Changes in control philosophy which often require major rework or even complete redesign of a functional module.

Actions 2 and 3 can be implemented rapidly, when compared with similar changes for a hydro-mechanical system, but are still time consuming. Further, it should be recognised that "interference" of this nature with electronic equipment has an adverse effect on reliability.

The use of a programmable system minimises the above problems in that hardware changes are not required. The time required to implement system changes by program depends to a large extent on the type of machine selected the level of programming language and the programming aids available.

The majority of programmable systems involve digital machines and have perforce, been programmed by experts in this field. It is most unlikely that such people will be expert in the totally unrelated field of prime mover performance and a committee approach to the solution of control problems is generally inevitable. A problem with this approach of course is that the committee members do not understand each others language.

The specialist programmer however, becomes superfluous if the programming language can be simplified so that it can be readily applied by the engine performance engineer. This is not yet the case for the digital machine unless specialised and expensive programming devices are employed.

These considerations were instrumental in the development of the Programmable Analog Controller (P.A.C.), which is now production hardware, and has proved to be an extremely powerful development tool. Among the major advantages of the P.A.C. are:-

. Its high level language which is readily understood by engineers of having unrelated disciplines.

. Its use of conventional block diagrams to define control action.

. The ability to monitor the internal performance of the machine and obtain meaningful information. This arises because signal information is present in analog form. Similar monitoring on a digital machine is meaningless to the majority of people.

A detailed description of the operation of a development PAC was presented in footnote 1. Continued development resulted in a more powerful machine aimed specifically at automotive applications. This machine, and its use in a specific requirement, is described in footnote 2. The need for greater program length and increased integration capability has now led to the production version of the controller, the PAC 500.

Footnote 1 Dent, J.R., Bergman, G.D., "Developments in Programmable Analog Control Systems" ASME Paper No. 74-GT-117

Footnote 2 Dent, J.R., LeFevre, H.P., "Advancements and Applications of Programmable Analog Control Technology" ASME Paper No. 76-GT-122

BASIS OF CONTROLLER OPERATION

A simplified block diagram of such a controller is shown in Fig.1. Signals from analog sensors are conditioned to provide D.C. voltages at a suitable level for the computing system. These voltages, together with signals representing input switch states and other quantities obtained from storage elements of varying kinds, are supplied to an input multiplexer. This is effectively a multi-position switch in which it is possible to output any one of the input signals in response to information from the program. The output of the multiplexer feeds the analog arithmetic unit which is, in reality, a gain controlled operational amplifier in which the gain is determined by an input resistor network and selector switches which are also controlled by the program. The output of the arithmetic unit supplies an output multiplexer, identical to that used for the input function, which directs the signal, by program, either to suitable power output stages or storage elements. The storage elements are of three types and provide:-

a) Working storage (capacitors) in which the results of intermediate calculations are held.

b) Integrating stores in which the stored level is the time integral of the calculation.

c) "Greatest-of" store in which the quantity stored is the highest of a number of sequentially applied signals.

It will be seen that all signals are processed in their analog form, however, the signal to be examined, the type of calculation and the routing of the answer are determined by the program, which is held in digital information to control the computation activities. The program is itself under the control of a clock and program counter. Each clock pulse advances the program one step and at each step, input and output multiplexer positions are defined, the amplifier gain is set and the type of calculation to be performed is determined. The duration of a program step, or "instruction word", is determined by the time taken for the various circuit components to provide an accurate answer and is approximately 25 microseconds. Thus, several hundred instruction words can be processed in the 13 milliseconds period chosen as the program execution time. Because of the transitory nature of the calculation result, the output stages contain storage capacitors which are updated once in every program cycle.

A facility not shown in Fig.1 is the jump capability which allows specific program addresses to be selected in response to calculation results. The unit can thus make decisions and so provide alternative actions.

The arrangement of an instruction word is shown in Fig.2, the complete word consisting of four groups of binary information, and contains 32 digits in the case of the PAC 500.

Group A determines the input address selected and has sufficient digits to enable the required number of input channels to be selected. For instance, 6 binary digits provides capacity for 64 input channels.

Group B determines the amplifier gain and as the desired accuracy of the controller is 0.1%, it must be possible to specify the gain required to better than 1 part in 1000. The use of 10 binary digits provides a resolution of 1 part in 1023.

Group C determines the output address and the remarks of Group A apply.

Group D are modifier bits, used singly, which determine the type of calculation to be performed. Ten modifier bits are utilised in the PAC 500.

The program to provide control action consists of a sequence of instruction words, the number of words being determined by the complexity of the control system desired. The program may be implemented in a number of ways depending on the particular use of the controller. For control system development purposes, the use of a diode-pin matrix in which the binary digits are entered by insertion of diode pins is the preferred approach. This allows program changes to be made in a short time by repositioning of the diode pins, often while the controlled engine is running. Once the program is reasonably firm, it may be transferred to an electrically programmed R.O.M. of the Ultra Violet light erasable type which is capable of being re-programmed.

For production systems, when program changes are not required, the program may be stored in either fusible link or mask-programmable R.O.M.s, depending on the quantities involved.

PRODUCTION UNIT TYPE PAC 500

Fig.3 is a photograph of a PAC 500 in its vehicular configuration which is specially packaged for under-hood installation. The ROM working program is contained within the unit although it is still possible to use the pin-program panel if so desired.

A detailed block diagram of the computer section of the controller is shown in Fig.4 Operation is as follows:-

Instructions

A clock circuit with a period of 26 us advances a "program counter" successively so that the instructions stored in a "Read Only Memory" (R.O.M.) are output in sequence. Each instruction is stored as a 32-bit digital word. The format of a digital word is shown in Fig.5. In hardware terms, bits are set to '1' by inserting pins into the pin-program panel (where this is being used); holes without pins represent '0'.

The first 6 bits of an instruction word define an input address, the next 10 bits define a gain-control setting, the next 6 bits define an output address, and the final 10 bits are special-purpose modifier bits (switches).

A control program consists of a sequence of 32-bit instructions of this type, up to a maximum of 512 instructions.

During the 26 us that is taken to execute an instruction, all 32 bits of the instruction word are simultaneously available.

The first 6 bits are connected to the input multiplexer.

With 6 bits it is possible to select one out of 64 possible inputs.

The next 10 bits are connected to the gain-controlled amplifier (GCA) which is the central element of the "arithmetic unit" of the PAC. These bits allow the GCA gain to be set between zero and 10.23 in 0.01 steps (i.e. to better than 1 part in 1000).

The next 6 bits are connected to the output multiplexer, giving 64 possible output channel selections.

Of the final 10 bits, the first 9 are used as special purpose modifier bits, designated M.1 to M.9, the functions of which will be described later. The last bit is either unused or used for checking parity.

Typically, during one instruction, the input multiplexer will be connected to a particular sensor conditioning circuit. The output from this circuit will be multiplied by a desired constant by means of the GCA, and the result will be routed to a particular storage location by means of the output multiplexer. The storage is directly of analog voltages and is achieved by means of low leakage capacitors.

This basic response to any PAC instruction can be modified considerably by use of the Modifier Bits.

HARDWARE FUNCTIONS OF THE MODIFIER BITS

M1 and 2

Two of the storage locations are special purpose stores, designated Ro and R1. These are organized so that their contents can be subtracted from the signal being processed by the gain-controlled amplifier during any instruction. Voltages are routed into Ro and R1 in the normal way by means of the output multiplexer. The contents of Ro are subtracted from the gain-controlled amplifier output during a particular instruction by setting the first modifier bit (M1) to 1. Similarly, if M2 is set to 1, the contents of R1 are subtracted from the output of the gain-controlled amplifier. With both modifier bits set to 1, both the contents of Ro and the contents of R1 are subtracted from the output of the GCA.

M3

M3 controls the polarity of the gain-controlled amplifier.

Set to 0, the gain is positive; set to 1, the gain is negative.

M4

One of the output multiplexer outputs is connected to a special-purpose store, the "Greatest Of" or G.O. Store, which performs a maximization operation. This is a simple but very powerful function in a complex control system. If a number of voltages are routed in turn to the "Greatest Of" store, the final voltage stored is the greatest of these input voltages. Setting the fourth modifier bit M4 to 1 performs the operation of setting the maximization store to the first input voltage in preparation for the maximization operation.

M5

The two special purpose stores, Ro and R1 are permanently connected to the inputs of an analog multiplier. The output of this multiplier is equal to 0.1 Ro R1, and is connected to one of the input multiplexer lines. M5, when set to 1 changes the multiplier function to a divide function, and the output is then equal to 10Ro/R1.

M6

So far, the computer has been described as running sequentially through the instructions stored in the R.O.M. Much greater power in programming is conferred by use of the PAC conditional jump operation. That is, at a given instruction, the program either goes on to execute the next instruction in sequence, or 'jumps' to some other instruction, depending upon whether a condition occurring at the output of the GCA is "true" or "false". The Conditional Jump operation has been implemented in PAC in the following way. When M6 is set to 1, the jump circuits are brought into operation, and, if the output from the GCA is negative, no jump takes place and the next instruction in sequence is performed. If, however, the net output from the gain-controlled amplifier is positive, a jump takes place, to an instruction by using the bits normally used for GCA gain setting.

M7, 8 and 9

For real time operation, it has been found useful to provide a number of integrating stores and normally, eight such integrators are provided. The first three can be individually reset to zero by setting modifier bits 7, 8 or 9 to '1' thus allowing them to be used for timing functions.

M10

Is spare but may be used for parity checking or diagnostic use.

INTERNAL STORAGE

In addition to the special registers, Ro and R1, the greatest of store and the integrators, eight working stores for analog quantities are provided. Additionally, eight flag stores for event marking are available.

A result to be stored can be directed to one of these circuits simply by specifying in a program instruction the appropriate 'output' multiplexer address. The stored value can then be accessed as required by use of the corresponding 'input' multiplexer address.

INTERFACING CONSIDERATIONS

The computing section described is a common element in all PAC units. The differences necessary to enable control of a wide range of diverse applications is confined entirely to the types of sensors and actuators employed and their signal conditioning circuits, of which a standard range is available.

30 input and 28 output channels of the 64 of each available are required by the computing section of the unit, the remainder are available as machine inputs and outputs for either analog or switch signals. In practice, vehicular control units do not require this capacity and are therefore provided with 24 input and 20 output channels. A typical Input - Output list is shown in Table 1. It is, in fact that used for the PAC units supplied for use on the Chrysler - ERDA gas turbine car,the control of which is discussed later.

PROGRAMMING

The format for the machine instruction word has previously been discussed, as have the pin program panel and PROM means of program storage. Entry of data for each instruction word is by means of binary digits. To avoid conversion from one arithmetic base to another, conversion errors and reduce programming time, a new programming unit is being developed.

This is based on the use of a Random Access Memory and a key-board similar to that used in Pocket Calculators. Program read-out will be available in decimal form and the desirable facility of program changes "on-line" will be retained.

Programs are in existence for use on main-frame computers to allow PAC programs to be written on a standard computer terminal. Paper tape records are produced which may be loaded to PROM either directly, if the computer interface hardware is available, or via a standard PROM programmer.

APPLICATIONS

By its nature, the PAC is a general purpose computer which is particularised for a specific application by the interfaces and working program employed. The same basic unit is utilised for a wide variety of tasks as typified by the following sample of applications.

Automobile Gas Turbine Engine

A number of PAC 500 equipments are now in operation on the Chrysler Baseline and Upgraded engines being developed against an E.R.D.A. contract. At present, the baseline engines are being used to test and develop control system concepts which are then utilised on the upgraded engine in both test cell and vehicular installations. Details of the upgraded engine are contained in footnote 3. A block diagram of the engine control system is shown in Fig.6.

Footnote 3 Ball, G.A., Gumaer, J.I., Sebestyen, T.M. The ERDA/Chrysler Upgraded Gas Turbine Engine, Objectives and Design. SAE Paper 760279

Four engine variables are controlled, viz.

Engine Fuel Flow

Variable Power Turbine Nozzles

Variable Inlet Guide Vanes

Water Injection Power Augmentation

Fuel flow is controlled by a closed loop system containing an electronic drive stage, Fuel Metering Valve and a flow measuring feedback sensor. Development is in progress to use closed loop acceleration temperature control and so avoid the need for the flow sensor. The metering system reference drive level is obtained from a "low fuel wins" logic arrangement which accepts signals from the gas generator governor, output shaft speed and compressor turbine entry temperature limiters and the Start/Acceleration fuel flow schedule.

Speed governing is a composite function featuring a corrected speed range speed governor, responding to the electrically sensed position of the accelerator pedal,and a mechanical speed topping governor. An output shaft speed signal provides for an increased gas generator idle speed to maintain output shaft speed when heavy accessory loads are present. A 75% maximum speed demand is also enabled when the transmission selector is in the netural or park positions. Governing control features proportional plus reset action.

The output shaft speed and compressor turbine inlet temperature limiters are proportional control functions which reduce fuel in the event the measured parameter exceeds a defined reference level. A time delay is incorporated in the temperature control limiter to prevent its action for normal engine transients.

Start and acceleration fuel flow levels are scheduled as a function of corrected gas generator speed. The function is non-linear and is approximated by a number of straight line segments. The schedule is modified in the start region by regenerator inlet temperature to compensate for its equivalent fuel flow.

Power turbine nozzle control is effected by a closed loop trim positioning arrangement which incorporates a power drive stage, trim actuator and a position sensor. The trim actuator operates over a restricted range in both power and braking modes of engine operation. The braking mode is by means of a braking solenoid controlled by pedal position, vehicle velocity and output shaft speed via suitable logic.

Trim actuator position demand signals are supplied by open nozzle wins logic having reference position and temperature control inputs. Nozzle start and acceleration position requirements are determined by gas generator shaft speed and speed governor error respectively.

Regenerator temperature control is in accordance with a schedule generated as a function of corrected gas generator speed, suitably modified by a further function of ambient temperature.

The signal polarity of the temperature control loop is inverted in the braking mode to take account of the inversion occurring in the engine nozzle - temperature gain in this mode of operation. Further features of the nozzle control loop include:

- Proportional plus reset temperature control
- Nozzle velocity limit in the increase temperature direction only.
- Inclusion of an A.C. dither component in the actuator drive current to minimise stiction and hysterisis effects.

Control of the variable inlet guide vanes is by means of a proportional actuator whereby the magnitude of the electrical signal determines the position of the vanes. The basic schedule for defining vane position is generated as a function of accelerator pedal position and is enabled under certain specific conditions.

Power augmentation by water injection is under the control of solenoid valve driven by a logic function. Injection occurs at near rated speed and then only when ambient temperature is within a certain band.

In addition to the control functions described, a comprehensive start/stop and safety sequence is incorporated. A number of output channels suitable for driving meters, recorders or oscilloscopes are available and programming is arranged so that virtually all computed parameters of interest are available for diagnostic purposes.

The working program for the complete system consists of approximately 350 instruction words, that is, about two thirds of the available memory. Adequate capacity for further control development is thus available.

DIESEL ENGINE

Fig.7 is a block diagram of a possible control system, implemented by a PAC 500, for the investigation of Diesel engine emissions. The controlled variables are the rack position of a conventional pump and line fuel injector system and the time of commencement of fuel injection. The rack actuator is a proportional solenoid featuring rack position feedback as a linearising means. The timing actuator is an electro-hydraulic device which is capable of varying the pump drive shaft relative to the engine crankshaft. Timing feedback is obtained by phase angle measurement of the pump drive shaft and crankshaft.

The fuel rack is positioned in response to signals arising from a low fuel wins logic arrangement which accepts input signals from the speed governor, rack limiter and fuel/air ratio control. The range speed governor reacts to a sensor attached to the accelerator pedal and provides proportional control action. The gain of the governor and hence system droop, is variable and enables change in high idle relative to rated speed.

The Rack Limiter restricts the maximum rack position and is scheduled as a function of engine speed. Adjustable torque rise can thus be obtained in lug operation. Fuel air ratio control is employed primarily for acceleration smoke suppression. The defined ratio is a complex function of engine speed and is also modified by the transient errors occurring in injection timing control. Manifold pressure, modified by manifold air temperature, is used as a multiplying factor on fuel air ratio, so taking account of turbo-charger effects and resulting in a desired fuel quantity.

Two injection timing schedules are provided for use in steady state and transient modes of engine operation. The transient timing schedule is a function of engine speed and is implemented whenever the speed governor is not the controlling factor. The steady state schedule is implemented at all other times and is a function of both engine speed and fuel quantity as determined by the pump rack position. Selection logic determines which of the timing schedules should be in effect, and passes the relevant signal to the timing actuator position loop.

Start/Stop sequencing and safety, together with diagnostic monitoring signals are also provided. The complete control program occupies about 230 lines of instructions.

MISCELLANEOUS GAS TURBINE PROGRAMS

The PAC is also in use on a number of other turbine programs. applications are concerned with controls for:-

- Three further automobile gas turbine engines, one of which is a novel approach featuring a 3-shaft concept.
- A gas turbine engined heavy transport truck for on highway use.
- Gas Turbine engine for use in tanks and other forms of military vehicles.

The PAC is also packaged as an Industrial control unit and may then be used in conjunction with a Programmable Sequencer, see Fig.8. The Sequencer was designed for use in applications where a large amount of plant sequencing was required, in addition to the engine control function, and operates on the same broad principles as the PAC except that the input/output signals are essentially digital in nature.

The industrial PAC/Sequencer combination is currently in use on 3 applications for both aero-derivative and heavy industrial gas turbine generator sets.

OTHER APPLICATIONS

Although the design of the PAC was originally focussed on Gas Turbine Engine control requirements, its general purpose nature has allowed its application on a number of unrelated programs, changes being confined exclusively to interface circuits of which a standard range is available. These applications include:-

- Stirling cycle engine and control system development.
- Control of a complex steam valve arrangement for submarine purposes.
- Projected use as a control element in an "Auto-Pilot" system to enable accurate positional control of oil-rig support vessels in bad weather situations. This system would use up to 7 PAC's in a distributed system under the control of a central digital computer.
- Inclusion of an engine simulation program in the same PAC as the control program. Complete system performance can thus be obtained with one unit for demonstration and training purposes.

CONCLUSIONS

Practical experience in the automotive gas turbine engine field has demonstrated the ease of application and flexibility of Programmable Analog controls, footnote 4. Other uses have proved the extremely wide range of control problems which benefit from the programmable approach with only minimum change to proven production hardware.

Because of the high level programming language, and consequent non-reliance on specialist programmers, the PAC is an Engineers Tool and provides the facility for minimising system development time. Further, because the PAC is production hardware there is no bill to meet for electronic circuit development.

In situations where the total quantity of the product is not large, significant cost benefit can be shown over the microprocessor approach because the software development bill is drastically reduced. The use of Large Scale Integration techniques

Footnote 4 Angell, Peter R., Golec, Thomas, Upgrading Automotive Gas Turbine Technology - An Experimental Evaluation of Improvement Concepts. SAE Paper 760280 for the PAC circuits can be expected to show cost reductions for larger volume production. The PAC then enhances its viability as a canditate for consideration.

New application areas are manifesting themselves, particularly in the field of distributed control, and the use of PAC's in such control systems under the supervision of a coordinating digital machine is one exciting avenue of the near future.

ILLUSTRATIONS

DENT J.R.

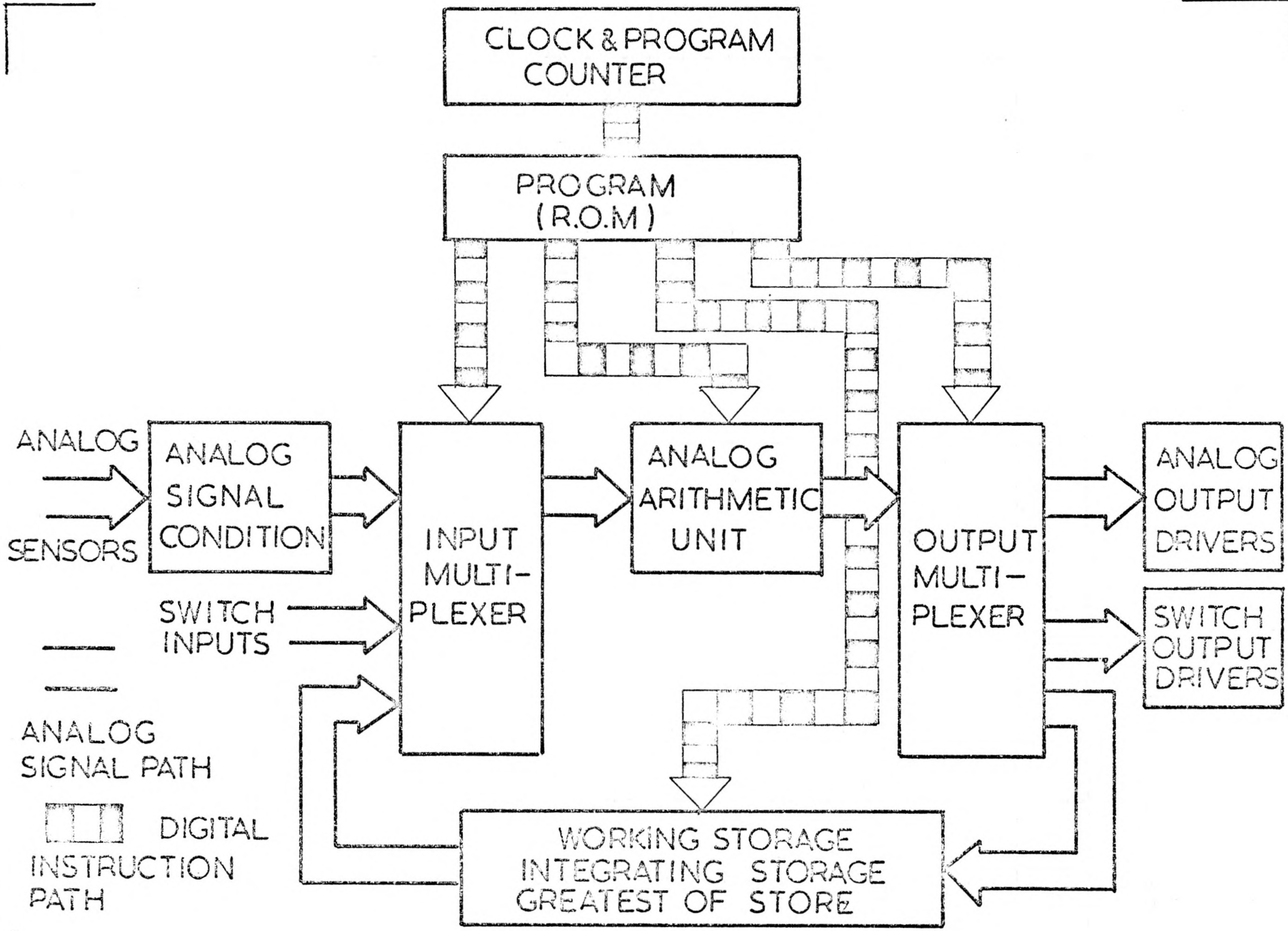
TABLE 1 - TYPICAL INPUT/OUTPUT LIST

ADDRESS	INPUT	OUTPUT
0	Ro	
1	R1	
2	Greatest of Store	
3	Mult/Div.	IGV Angle
4	1 Volt Ref.	IGV Drive
5	Fuel Flow	Nozzle Drive
6	-	FMV Drive
7	-	-
24	Flag	
25	Flag & Switch (LED) O/P	
26	Flag	
27	Flag	
28	Flag	
29	Flag	
30	Flag & Acceleration Solenoid	
31	Flag & Rev. IGV Current Switch	
32	Nos. Speed	T8 Indicator
33	TAMB	NGG Indicator
34	Trim 2	NGG Demand
35	Trim 3	Accel. WF Demand
36	NGG Speed	NGG O/Speed Gov. Demand
37	TC.3	Nozzle Angle Demand
38	TC.2	$1/\sqrt{\theta}$ Computed
39	TC.1	NGG/$\sqrt{\theta}$ Computed
40	Trim 5	-

TABLE 1 (Continued)

ADDRESS	INPUT	OUTPUT
41	Nozzle F/B	-
42	PAMB	-
43	Abort Start Timeout	-
44	-	-
45	α	-
46	Trim 1	-
47	Trim 6	-
48	-	Brake Sol. SW.
49	-	'Start Cycle' Ind.
50	-	'No Start' Ind.
51	-	Starter Sol. SW.
52	-	Water Inj. Sol. SW.
53	-	Fuel Enable Sig.
54	-	Power Boost Sig.
55	-	Start Sig.
56	NGG Safety SW.	-
57	Accelerator SW.	-
58	Start/Park SW.	-
59	Nos. Safety SW.	-
60	Spare Switch	-
61	Start SW.	-

Addresses 8 to 23 are working stores and integrators.



DENT. J.R.

FIG.1 BASIC PROGRAMMABLE ANALOG CONTROLLER

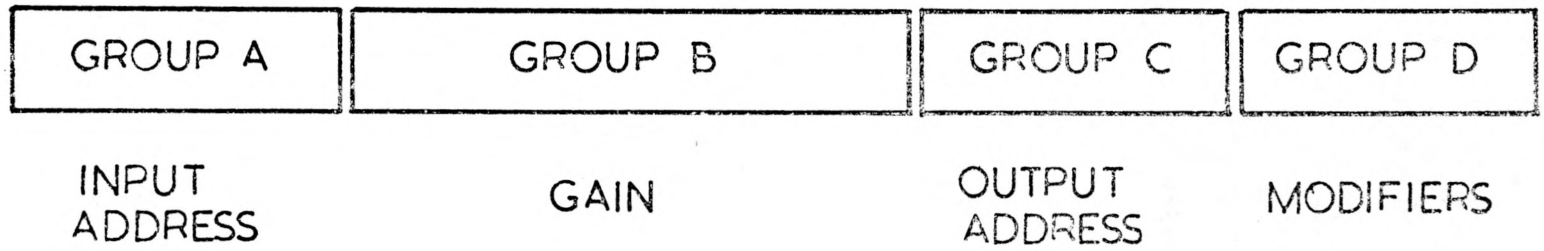
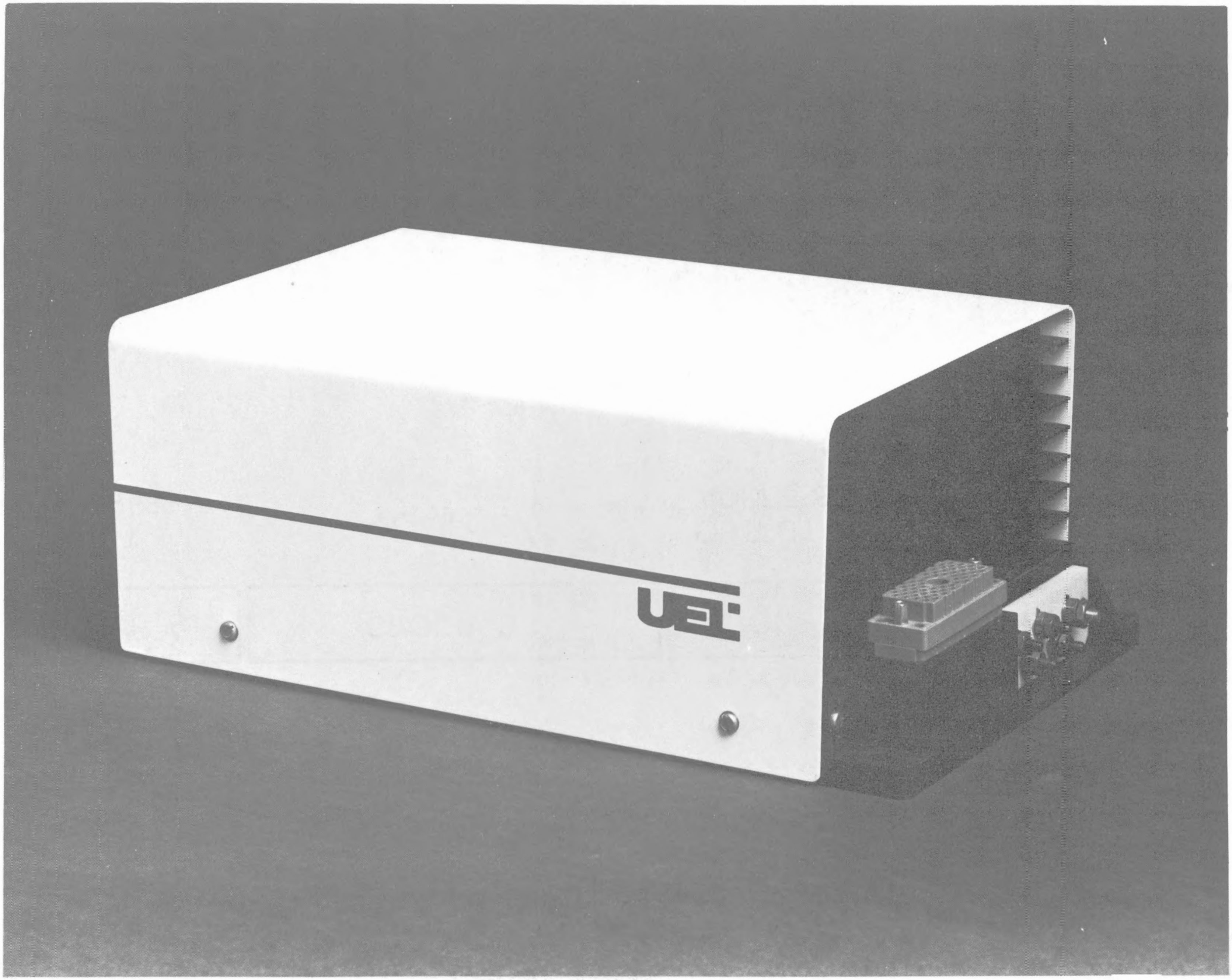


FIG.2 BASIC INSTRUCTION WORD

DENT.J.R.


UEI

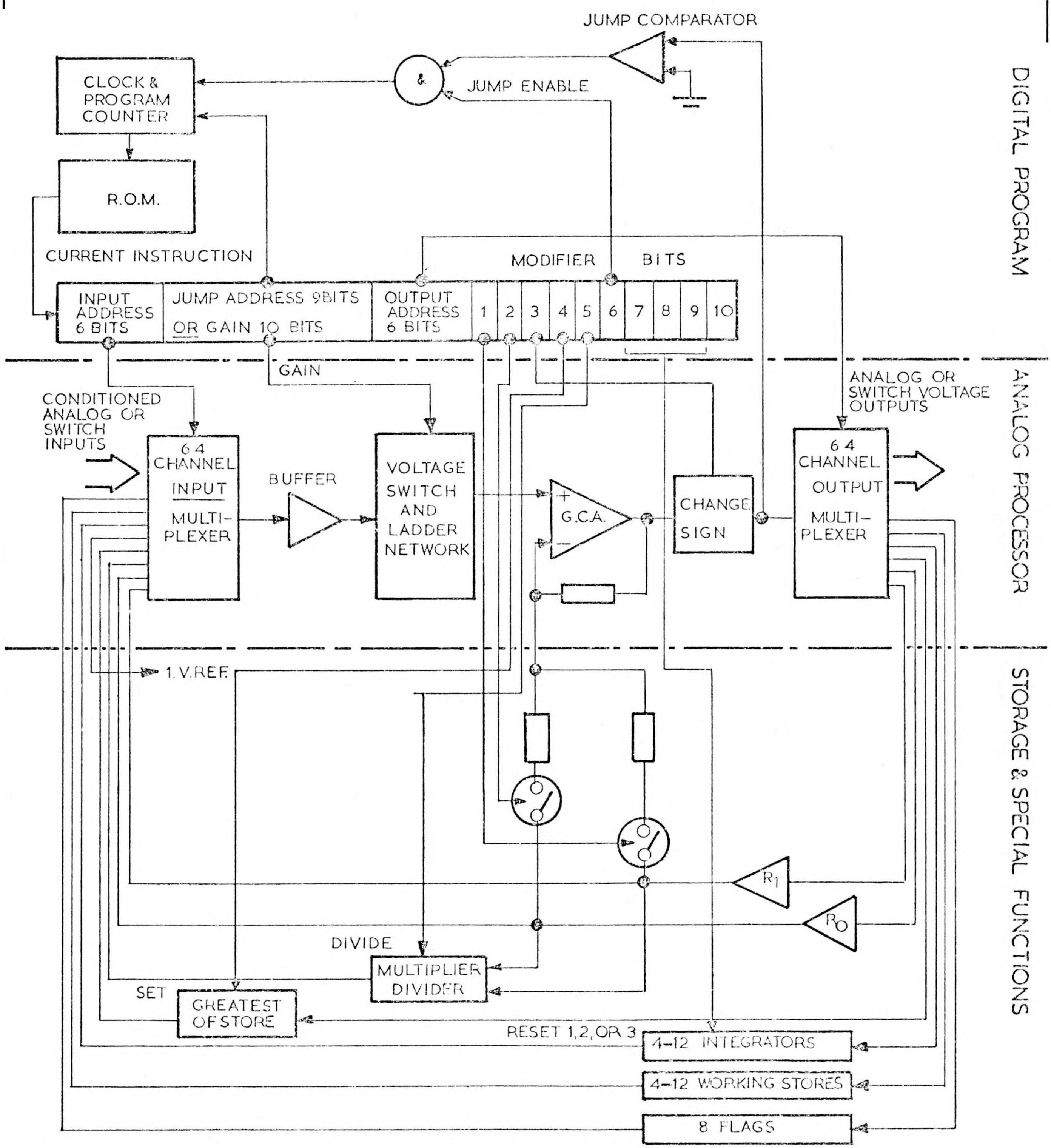


FIG.4. BLOCK DIAGRAM OF P.A.C. INTERNAL STRUCTURE

DENT.J.R.

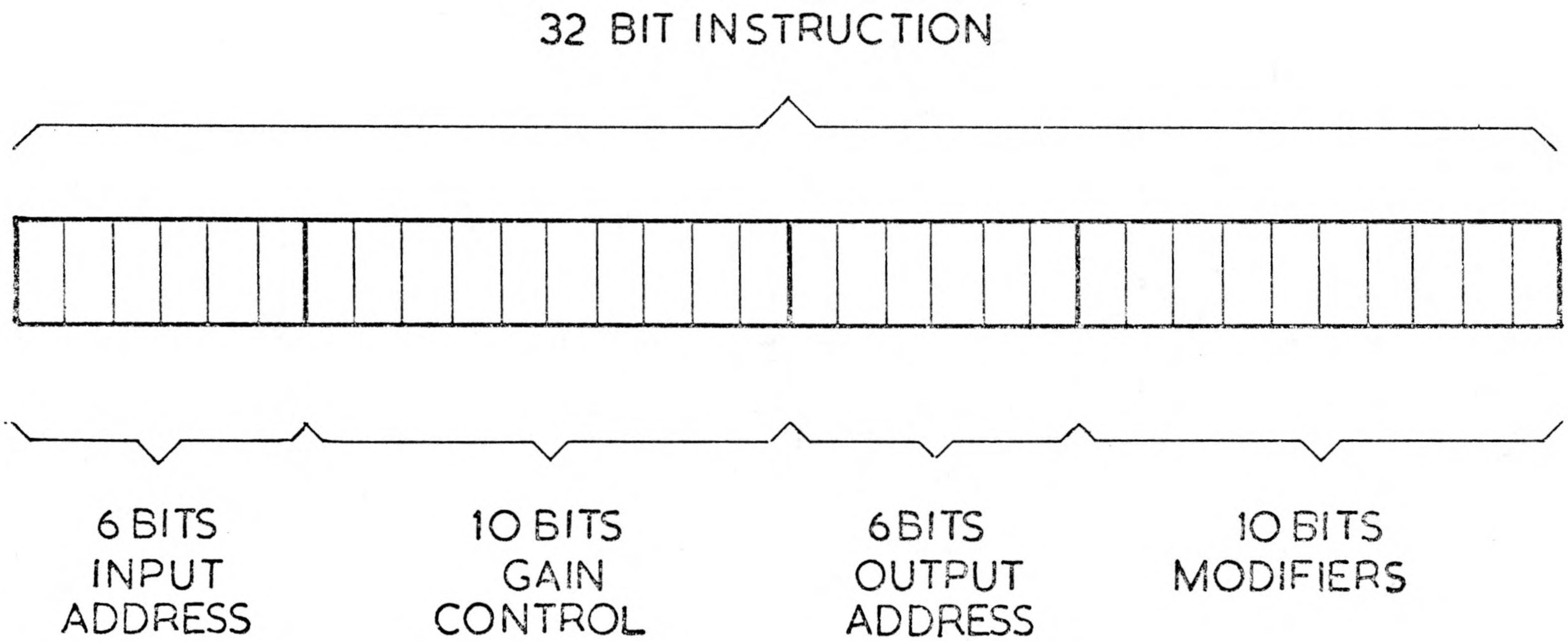


FIG.5 INSTRUCTION WORD FORMAT

DENT.J.R

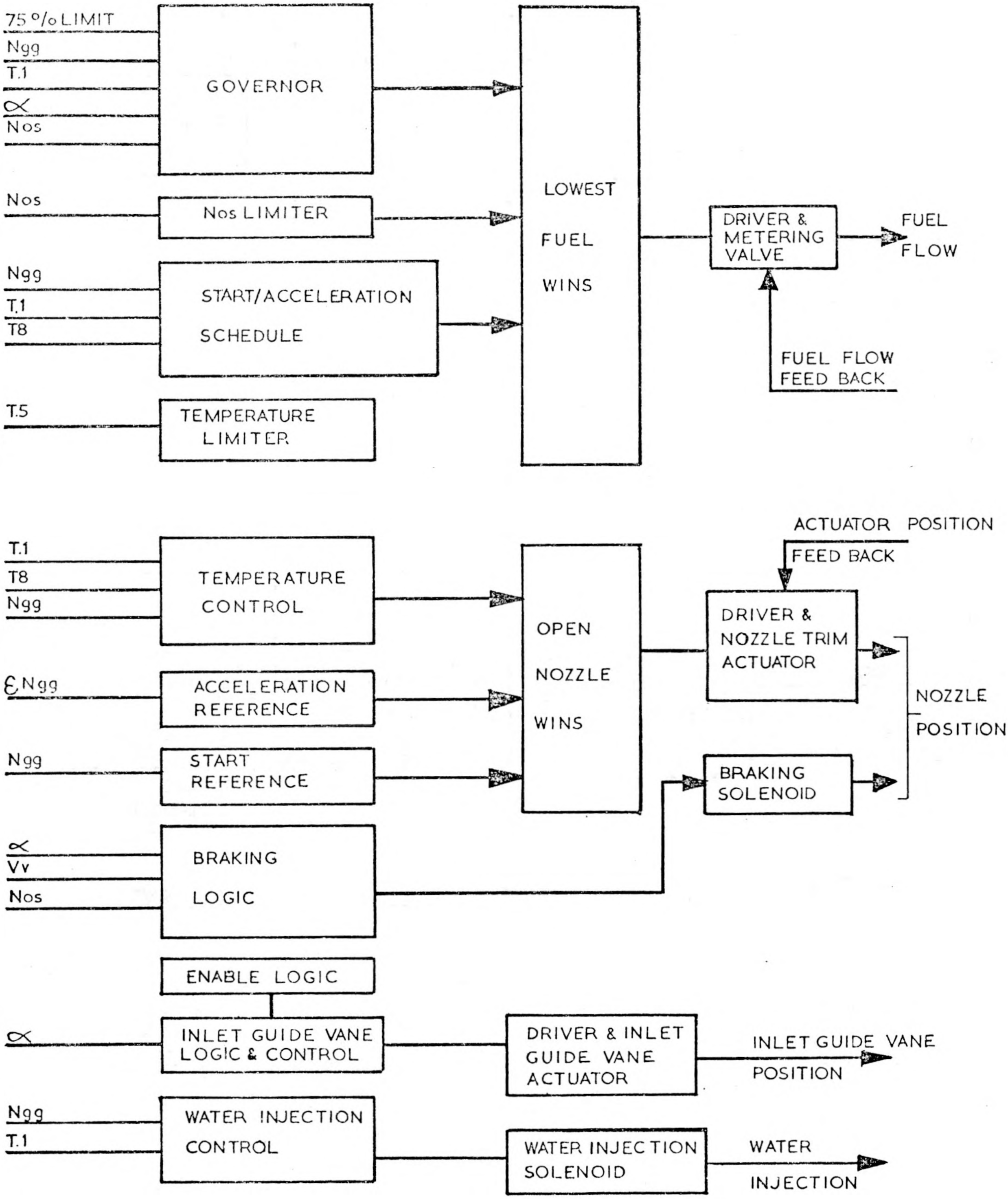


FIG.6 AUTOMOTIVE GAS TURBINE ENGINE CONTROL

DENT.J.R.

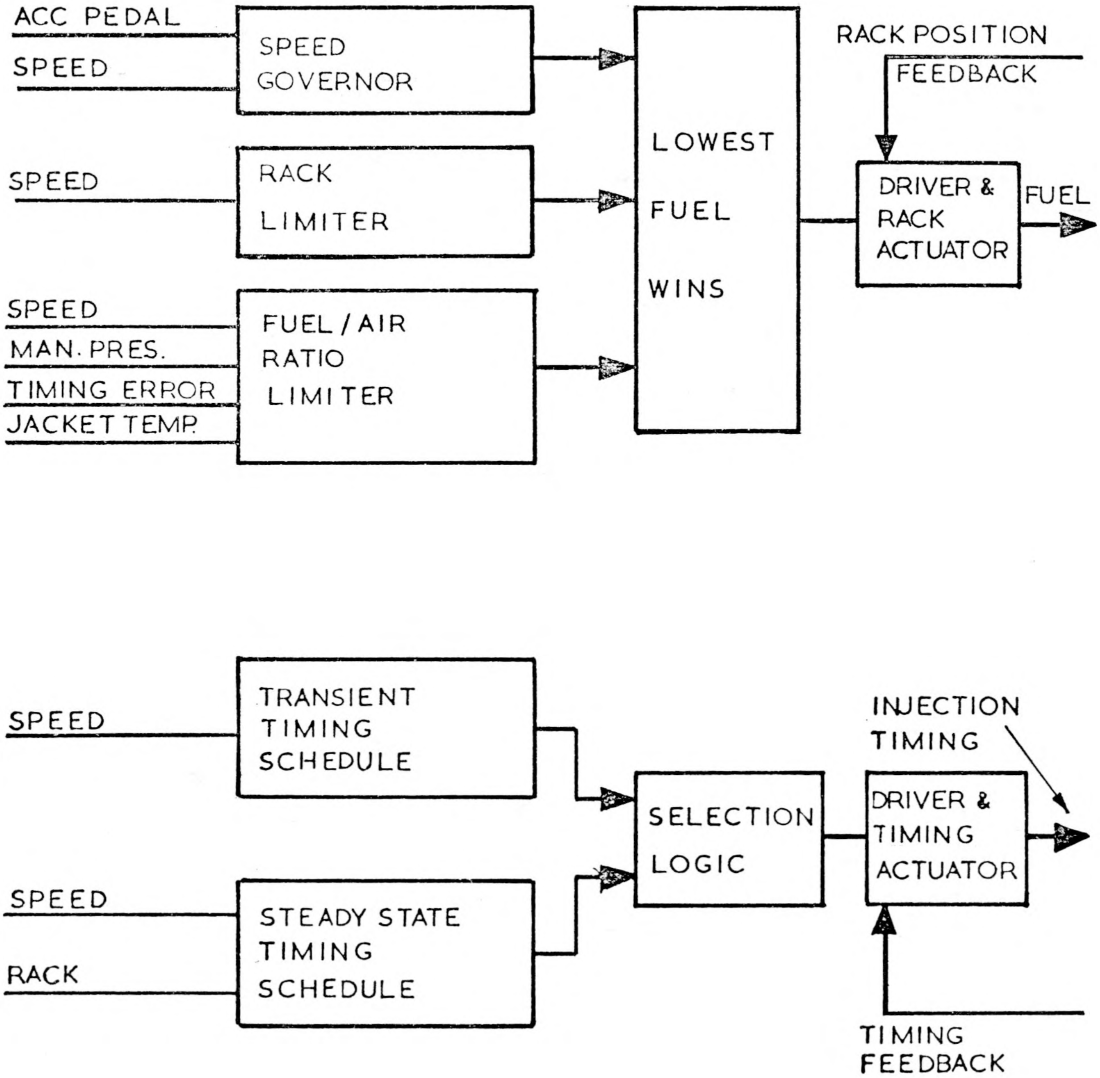
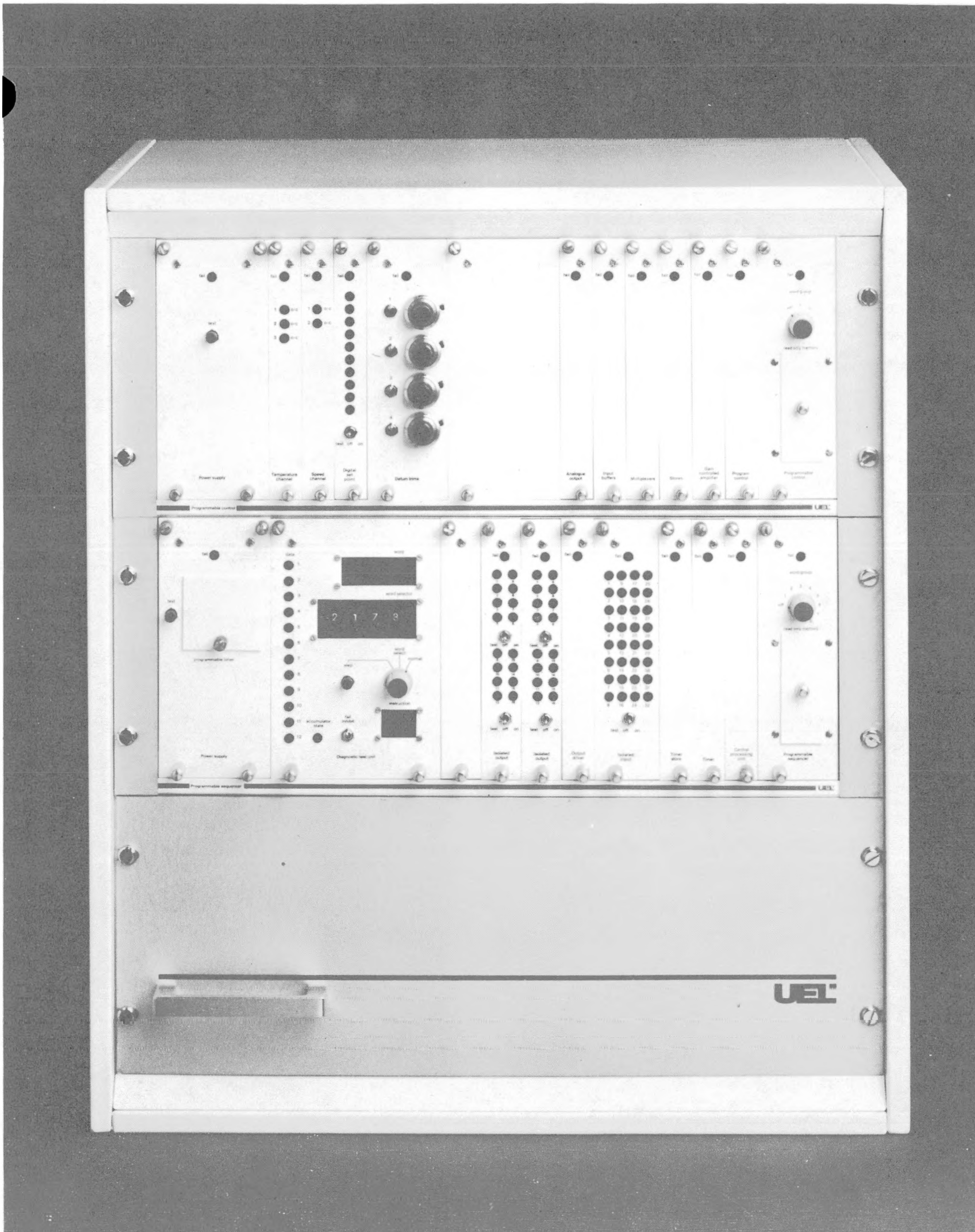


FIG.7 PROPOSED DIESEL ENGINE CONTROL SYSTEM

DENT.J.R.


UEL

AN ANALYTICAL STUDY OF TRANSMISSION MODIFICATIONS AS RELATED TO VEHICLE PERFORMANCE AND ECONOMY

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AN ANALYTICAL STUDY OF TRANSMISSION MODIFICATIONS AS RELATED TO VEHICLE PERFORMANCE & ECONOMY

ABSTRACT:

A method of vehicle performance measurement has been developed so that selection of optimum fuel-economy-performance trade-offs can be made for a vehicle having various powertrain components. This method was utilized in an analytical study of drivetrain component features such as--overall ratio range, number of ratio steps, locked converters, continuously variable drives, etc. Both manual and automatic type transmissions are considered. Indications are that ratio range is an important consideration in the selection of transmission design parameters and also conventional transmission concepts can be competitive with the more exotic continuously variable type units.

INTRODUCTION:

The material discussed here was presented during the February 28-March 4, 1977 Society of Automotive Engineers, International Automotive Engineering Congress and has been published in the SAE Technical Paper Number 770418. With todays increased emphasis on fuel economy and the rigid emission requirements for the vehicles produced by the automotive industry, new methods have to be used to help guide the selection of the future powertrains to provide the best overall compromise of size, weight, and cost for optimum performance and economy. Time does not allow for all the various powertrain combinations to be designed, fabricated, developed and evaluated with the actual vehicles in which they would be used. There is also difficulty encountered in measuring small changes in fuel-economy and performance with road or chassis dynamometer tests. These factors and the availability of a previously developed vehicle simulation program led us to set-up an analytical process to evaluate the various combinations to indicate their potential in meeting these goals.

The transmissions used today are primarily of two types, manually shifted type and hydrodynamic drive automatics. They utilize a varied number of geared ratio steps and overall ratio spreads. To determine the maximum potential of these conventional transmissions, various parameters of these designs were evaluated to show any improvements that could be made to further their contribution to the overall performance and economy of a vehicle. For the manual transmission, the effect of the number of geared ratios and the overall ratio spread was examined to find the optimum combination for best vehicle performance and economy. In addition to these two parameters, the automatic transmission also considered the efficiency of the hydraulic torque converter.

After the potential of these optimized conventional drivelines had been determined, an evaluation of the continuously variable transmission was made and compared to determine any advantages that it might have. This comparison was made since several continuously variable transmissions are under development and there is need to establish the benefits of such a transmission.

The information presented here is directionally and qualitatively correct; however, their magnitudes are only representative and subject to qualifications. The simulation uses only normal operating temperature engine data while the EPA car test requires a cold start. There also is no allowance for any engine adjustments that may be required to meet driveability criteria and emission standards. Since the drivetrain concepts force the engine to operate at different load-speed conditions, emissions will be affected.[1] Only a single set of vehicle parameters other than the powertrain was evaluated. Other vehicle parameter changes could alter the magnitudes of the differences shown in the study.

[1]Marks, Craig; Niepoth, George, "Car Design for Economy and Emissions" SAE Paper 750954 presented at Automotive Engineering and Manufacturing Meeting, Detroit, Michigan, October 1975.

THE METHOD:

The method used to precisely evaluate the effect of drivetrain variations on vehicle performance and economy is called the "Optimum Performance Versus Economy Line" Technique. Such lines are developed and used as follows.

The 0-60 MPH time (or other performance parameter) and composite EPA fuel economy (or other) of a specified vehicle can be computed, obtained by test, or obtained by a combination of both methods. The result is one discrete point on a plot of 0-60 MPH time versus EPA composite MPG as shown in Figure 1. Changing the final drive ratio of the vehicle produces another discrete point. Obtaining the points over a range of final drive ratios produces a curve which depicts the trade-off of performance and economy caused by final drive ratio changes. This is illustrated in Figure 2.

Repeating the above process on the specified vehicle for a range of engine sizes will produce a series of final drive performance-economy trade-off curves. Constructing a tangent line to this series of curves produces a curve which represents the best fuel economy at a given performance level that can be obtained from the specified vehicle through changes of engine size and final drive ratio. This is shown in Figure 3. Such a tangent curve is referred to as an "Optimum Performance vs. Economy" curve. It should be noted that it is not possible to specify an engine final drive combination that will produce a point to the right of the optimum curve. All combinations but the optimum will fall to the left.

Constructing such "Optimum" curves for driveline changes in the specified vehicle provides a means of precisely evaluating the effect of the changes. Figure 4 shows the "Optimum" curves for two different transmissions in the specified vehicle. Note the ease of determining the economy difference at a given performance level.

The "Optimum" lines technique overcomes several difficulties in evaluating drivetrain changes. Other methods often compare the economy gains at different performance levels for the changes. This makes assessments difficult. They may also compare at less than optimum conditions or at different levels of optimization for the changes, both of which may produce less accurate conclusions.

The "Optimum Line" was used in the following analysis of powertrains.

COMPUTER PROGRAM USED FOR ANALYTICAL STUDY

The analytical tool used to evaluate the various parameters in this study was a computer program referred to as GPSIM[2] (A General Purpose Automotive Vehicle Performance and Economy Simulator). The GPSIM computer program is an automotive vehicle simulator used to evaluate the performance and economy of a vehicle by computing the operating conditions of the engine and driveline as the vehicle is directed through a prescribed operating schedule. The program will handle transient and steady state vehicle operating conditions. This provides the capability to simulate any performance criteria or to evaluate fuel economy for many driving schedules.

The data required by the program consists of data tables needed to describe the various components of the vehicle and powertrain and to provide adequate information for the simulated vehicle to operate normally. The data tables used to describe the engine, torque converter and accessory and spin losses are taken from test data measured at stabilized conditions. Information describing other driveline and vehicle parameters such as the transmission, shift schedule, driveline inertias, road load power requirement, etc., are taken from design information and test results. In addition to this data, information for the desired operation of the vehicle must be provided. This information could be a simple wide open throttle performance request or a more complex set of performance instructions to enable a vehicle to follow a desired economy driving schedule.

[2]Waters, William C., "General Purpose Automotive Vehicle Performance and Economy Simulator." SAE Paper 720043 presented at the Automotive Engineering Congress, Detroit, Michigan, January 1972.

Data from various existing and experimental vehicles and powertrain components were used for this analytical study. The output data generated for each vehicle powertrain combination was the fuel economy for the EPA Urban and EPA Highway driving cycles and the 0-60 MPH wide open throttle performance time. Fuel consumption for these driving cycles were used to determine the EPA Composite (55%-45%) Fuel Economy. It must be remembered that exact simulation of the EPA Urban cycle cannot be made since only normal operating temperature engine data is used. To obtain the optimum values for fuel economy and performance for each of these combinations, an optimum shift schedule was used. This schedule chooses the best gear of a specific transmission to be used to satisfy a particular vehicle operating condition. For the wide open throttle condition, the ratio allowing the best wheel power is always the operating gear; for part throttle conditions, the ratio allowing the best fuel economy to meet a specific driving schedule requirement becomes the operating gear.

The flexibility and convenience provided by this program made it possible to determine the optimum performance vs. economy lines for the transmissions considered in this study.

TRANSMISSIONS INVESTIGATED AND SIMULATION ASSUMPTIONS:

As mentioned earlier, both manual and automatic transmissions were studied. The basic manual was of 4-speed design while the basic automatic had 3-speed Simpson type gearing with a conventional hydraulic torque converter.

For the manual transmission comparison, units having wider ratio range and more speeds were investigated. Three 4-speed units and one 5-speed unit were included in the study and the basic schematic for the transmission is shown in Figure 5. All 4-speed transmissions used similar spin losses relative to input speed. These losses were determined from tests of a transmission having a similar capacity. A percentage loss was also added for each active gear mesh. When the 5-speed unit was investigated, its input shaft losses were adjusted in relation to the number of additional gear tooth meshes required to provide the additional geared ratio. All transmissions were sized to handle the same engine torque and vehicle weight.

The basic automatic transmission, shown in Figure 6, consisted of a unit designed to meet the same capacity requirements as the manual. The losses associated with the gear box portion of the transmission consisted of the front pump losses running at converter input speed and the gear box losses (including clutch losses) running at gear box input shaft speed. The losses applied to these characteristics were obtained from test data of units having a similar capacity. Once again, a percentage loss was added for each active gear mesh. The torque converter in front of the gear box was of conventional construction and sized to give the same stall speed for all engines used. The basic unit used an open converter; however, in several of the comparisons, lock-up clutch converters were also used. As ratio spread was increased, losses were assumed to stay the same relative to input shaft speed. A 4-speed automatic type transmission was also investigated. This transmission was of conventional construction utilizing a torque converter ahead of the gear box. The torque converter characteristics were similar to those in the 3-speed units. The gear box component changes were of such a nature that the spin losses could be assumed to be similar to that of the 3-speed automatic. Active gear mesh losses were included similar to the other automatic units.

The simulation used to represent a continuously variable transmission was an automatic unit having 12 equally stepped geared ratios. This model, Figure 7, included a torque converter with a lock-up clutch directly behind the engine and ahead of the simulated gear box. The gear box was assigned a constant efficiency for all operating conditions and spin losses and pump losses were assumed to be the same as the basic 3-speed automatic unit. No additional losses for increased pump capacity or pressure requirements were added. It was assumed that this unit would provide the same degree of automatic driving convenience that the conventional automatic transmission provides. To further represent the conditions of a continuously variable transmission, shift time and energy loss during the shifts were assumed to be zero.

Several assumptions were made for the computer simulation. Geared ratio shifts occurred at the optimum schedule to provide maximum fuel economy on the driving schedules and did not have to be in sequence. They were also selected to provide maximum acceleration for performance comparisons. Shift time for manuals was 0.5 seconds per up-shift with a zero energy transfer during the shift. Conventional automatics had a 0.6 second shift time; however, energy transfer during the shift was 80%. As mentioned previously the simulation of the continuously variable transmissions assumed a zero shift time with a 100% energy transfer during the shift. The converter was locked in all gears except its highest numerical ratio gear where it could be either locked or open. In this highest ratio gear the converter was selected to be locked or open and switched in mode at the appropriate instance to provide either maximum fuel economy or performance depending on the schedule in which it was operating.

The various manual transmission combinations evaluated in this study are tabulated in Figure 8 and the automatics and continuously variable units in Figure 9.

RESULTS:

MANUAL TRANSMISSIONS

The manual transmissions were evaluated using the technique described in the previous sections. The four manuals simulated in this study have been described in Figure 8. The first manual investigated, labeled "A", which we will use as a baseline, was one having an overall ratio spread of 3.56:1. The "Optimum Performance vs. Economy" curve for this unit is shown in Figure 10. It must be remembered that for a given vehicle, the engine displacement must be adjusted in order to develop the performance and fuel economy shown by the "Optimum Performance vs. Economy" curve.

Four-speed transmissions having overall ratio spreads of 4.72:1 and 5.94:1 were also investigated. These transmissions are identified as units B & C respectively. "Optimum Performance vs. Economy" curves were determined for these units and are shown in Figure 11. For the purpose of this discussion a performance level of 13.5 seconds to accelerate from 0-60 MPH will be assumed as being the vehicle requirement. This requirement was set forth by the Office of Air Programs of the Environmental Protection Agency in 1972 for several powertrain investigations. At this level of performance, unit B will provide an improvement in fuel economy over the base of 3.8% while unit C provides a 4.4% increase.

To further study improvements that could be made in manual type transmissions, a 5-speed unit was also investigated. This unit is identified in Figure 8 as unit D and had an overall ratio range of 4.72:1--the same as 4-speed unit B.

The "Optimum Performance vs. Economy" curve for this transmission is shown in Figure 12 and the improvement in fuel economy over the base manual at the required performance level is 4.1% or only 0.3% better than its 4-speed counterpart.

AUTOMATIC & CONTINUOUSLY VARIABLE TRANSMISSIONS:

The automatic and continuously variable drive transmissions were evaluated in a similar manner as the manuals. The base three-speed (unit E, Figure 9) was operated with an open converter as well as with a lock-up converter--the lock-up clutch being applied in 3rd gear only. The optimum lines for these transmissions are shown in Figure 13 with the improvement at the selected 13.5 second 0-60 MPH performance time being 6.7%. It is not the intent of this paper to imply that a lock-up clutch could be satisfactorily adapted to today's transmissions since the driveability and emission targets must also be met and this study does not take these factors into consideration.

A 4-speed transmission which is designated unit F having an overall ratio range of 4.22:1 was mathematically modeled and operated with a converter lock-up clutch applied in both 3rd and 4th gears. It was also operated with the lock-up clutch applied at the most beneficial time in first gear and fully applied in 2nd, 3rd, and 4th gears. The optimum lines for these units are shown in Figure 14. It should be noted that the economy improvement is 14.3% when the converter lock-up clutch was applied in 3rd and 4th gears and 18.0% when utilized in all gears. The overall ratio of the 4-speed type transmission was then expanded to 6.00:1. This transmission is designated unit G on Figure 9. The optimum line for this unit, also shown on Figure 14, indicated an economy improvement of 19.7% when the lock-up clutch was applied in all gears.

The continuously variable units used the 12-speed simulation described earlier. In order to give these transmissions maximum benefit, the converter lock-up clutch was engaged at the optimum time in 1st gear and remained engaged in the remaining 11 gears at all times. The geared ratio ranges investigated were the same as the 4-speed conventional automatics--that is, 4.22:1 and 6.00:1. These units are identified as H & I in Figure 9. The optimum lines generated by these units when the continuously variable portion of the transmission was assumed to be 95% efficient are shown in Figure 15. Improvement for the narrower ratio unit over the base 3-speed is 18.0% and the wider ratio unit 21.6%. This same data for the 6.00:1 overall ratio range transmission was generated for a unit having 90% efficiency in the continuously variable portion of the gear box and is also shown in Figure 15. Improvement here was only 15.7%.

In order to assure that the 12-speed gear box adequately simulated the continuously variable transmission, a transmission having 20 speeds with equal geared steps and a 6.00:1 overall ratio was developed. The optimum line for this unit indicated an error of only 0.16% for the 12-speed unit. It was, therefore, concluded that the 12-speed units resulted in a satisfactory simulation.

SUMMARY:

It is important that drivetrains be evaluated at their optimum performance--fuel economy capability since it is by this means that the true potential of a transmission can be determined. This study is a step in that direction. Erroneous results can be obtained by investigating transmissions in vehicles having other drivetrain parameters such as fixed axle N/V. Estimates of fuel economy benefits of wide ratio, overdrive, and continuously variable units can be forecast unusually high if the comparison is based on only a high axle N/V vehicle. The system established here eliminates this possibility and compares the transmissions on an equal basis.

Figure 16 is a tabulation which compares the type of improvement that was predicted for manual transmissions having wider ratios and increased number of geared speeds (from 4 to 5) over a base manual 4-speed. It should be noted that increased ratio range has almost equal effects whether done in a 4- or 5-speed design configuration. Figure 17 is a similar tabulation for the automatic transmissions and the continuously variable units. Here it can be noted that automatic transmissions utilizing conventional design techniques could perform as well as the continuously variable units and that the continuously variable ratio portion of such a transmission must have high operating efficiency to remain competitive. The magnitudes of the changes indicated are subject to qualifications which have been described previously.

Since emissions and driveability have not been considered in this study, the benefits indicated may not be totally achievable; however, the most promising areas of gain are indicated and it remains for the engineer to resolve the problems.

None of these areas can be solved independently since they are all interrelated with the powertrain selection and the final product offered the public must meet the emission requirement levels and fuel economy targets for that model year as well as driveability and acceptability standards.

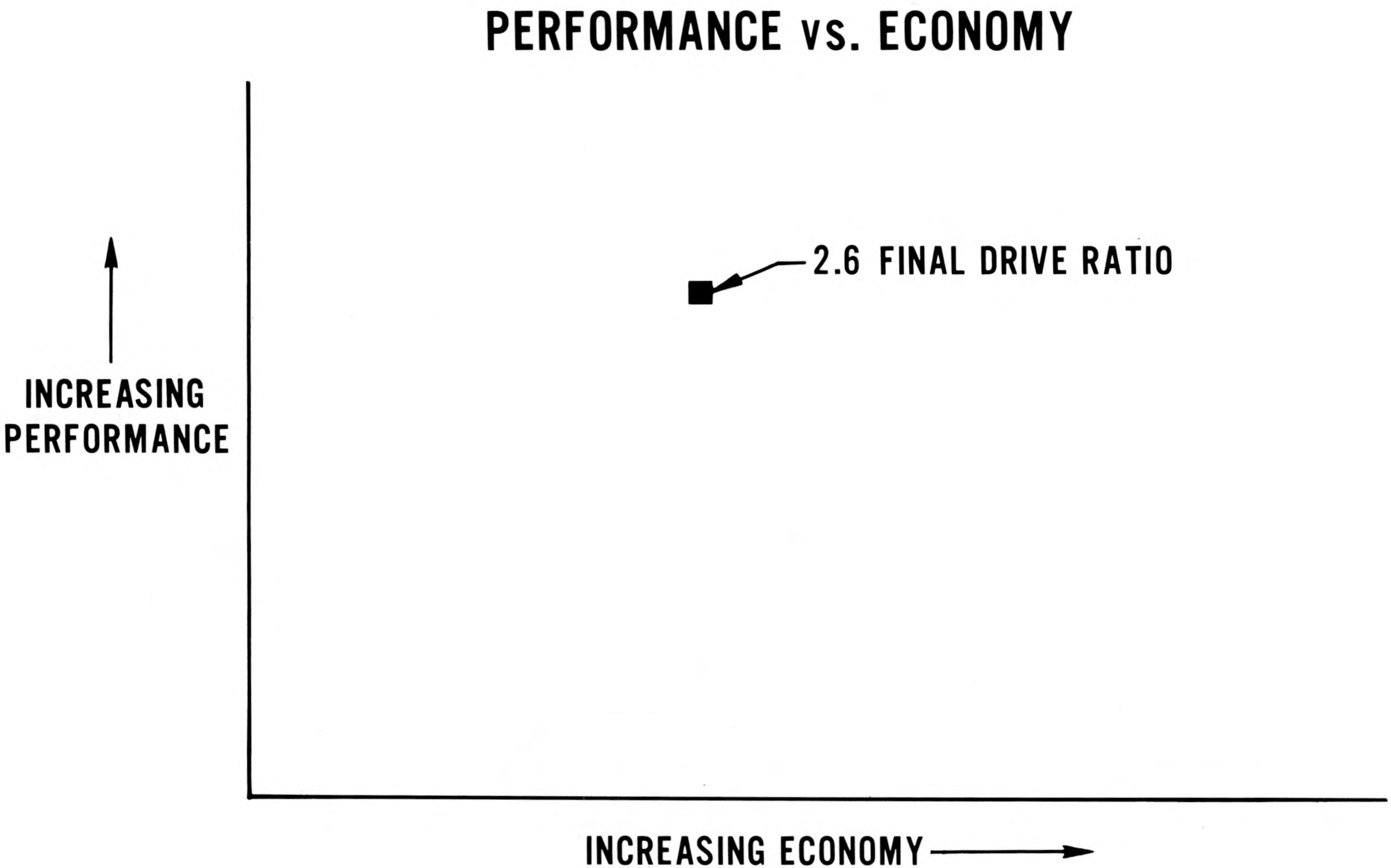
PERFORMANCE vs. ECONOMY
2.6 FINAL DRIVE RATIO
INCREASING PERFORMANCE
INCREASING ECONOMY

FIGURE 1

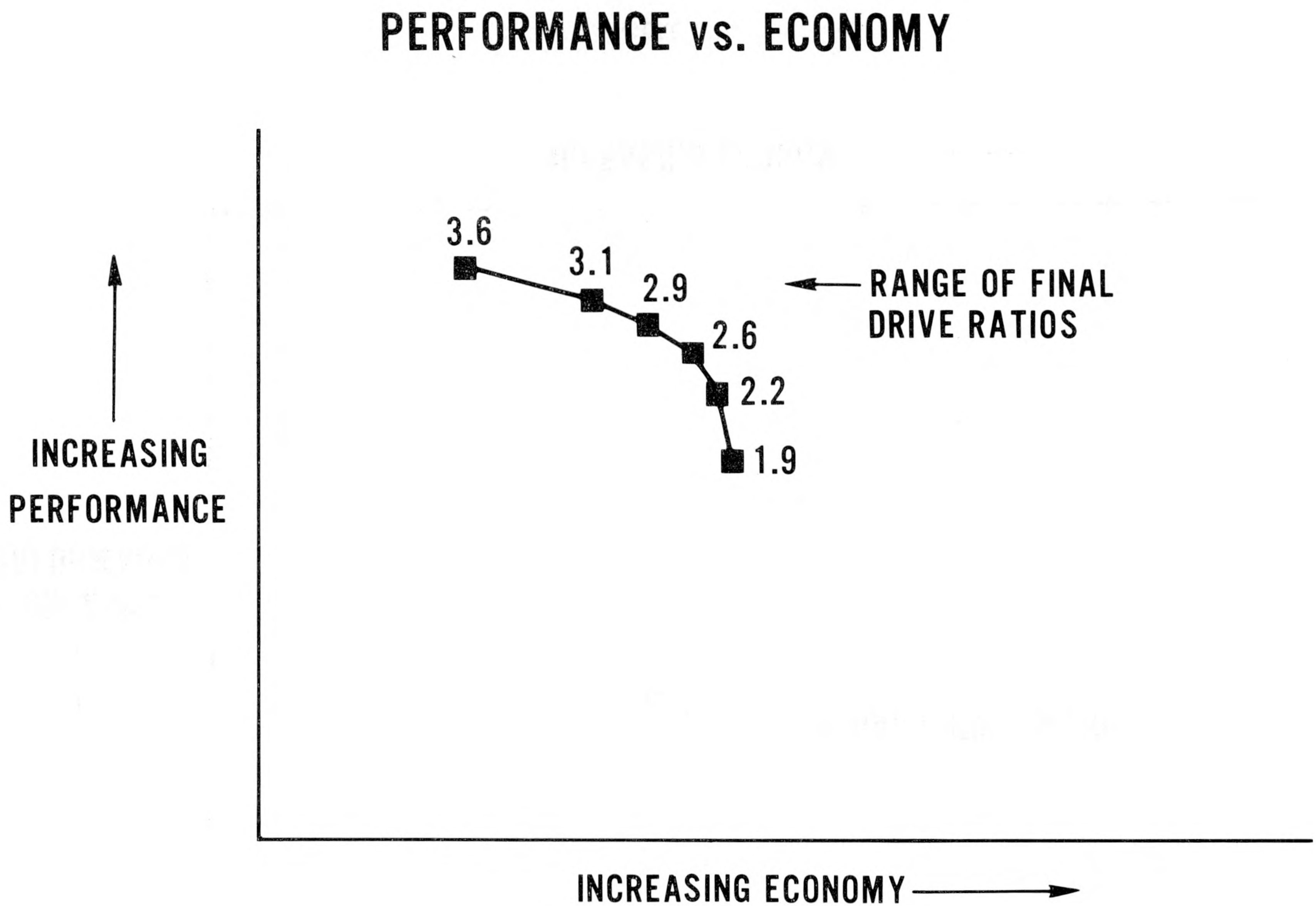
PERFORMANCE vs. ECONOMY
3.6
3.1
2.9
2.6
2.2
1.9
RANGE OF FINAL
DRIVE RATIOS
INCREASING
PERFORMANCE
INCREASING ECONOMY

FIGURE 2

PERFORMANCE VS. ECOMOMY

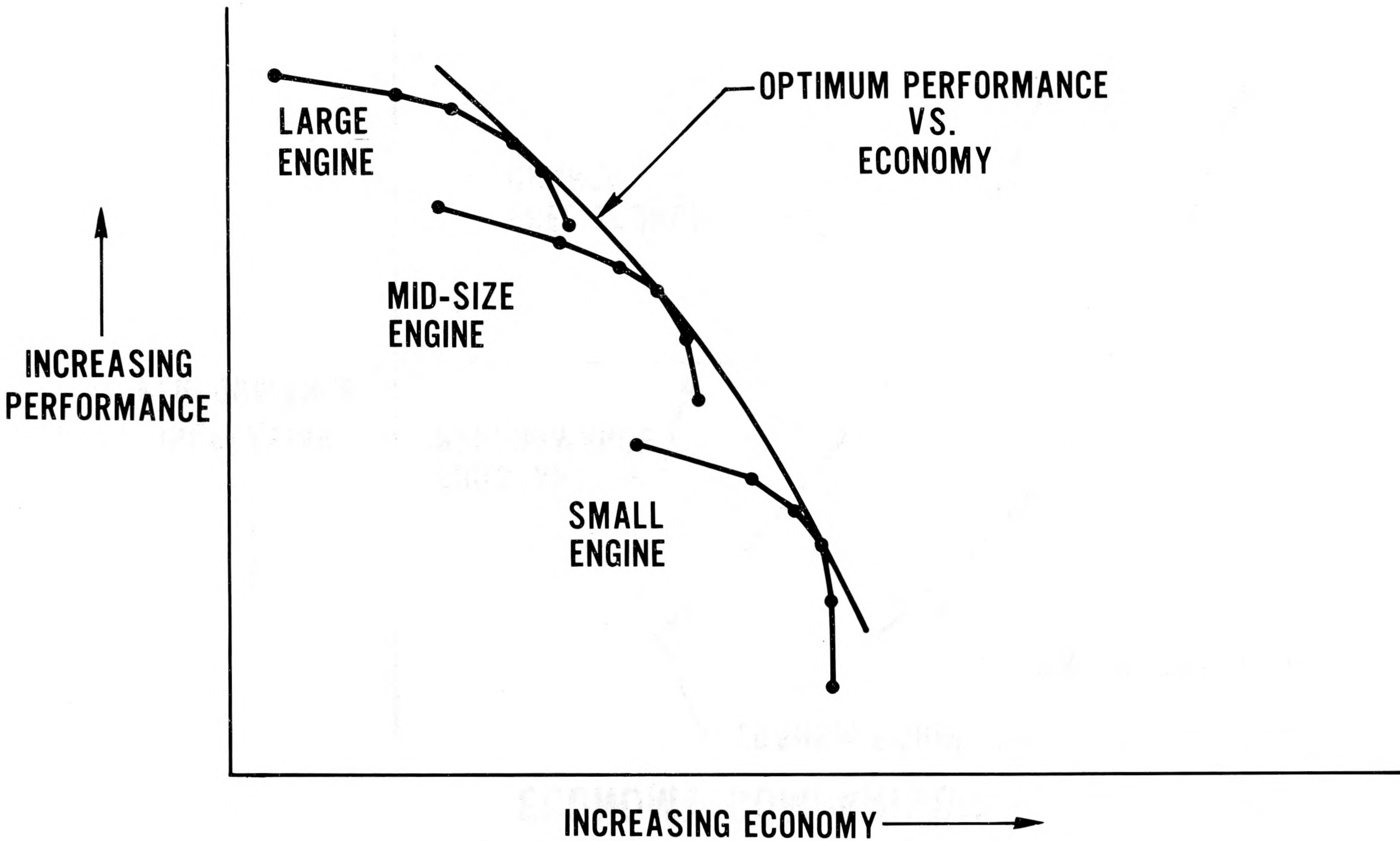

FIGURE 3

OPTIMUM PERFORMANCE vs. ECONOMY COMPARISON

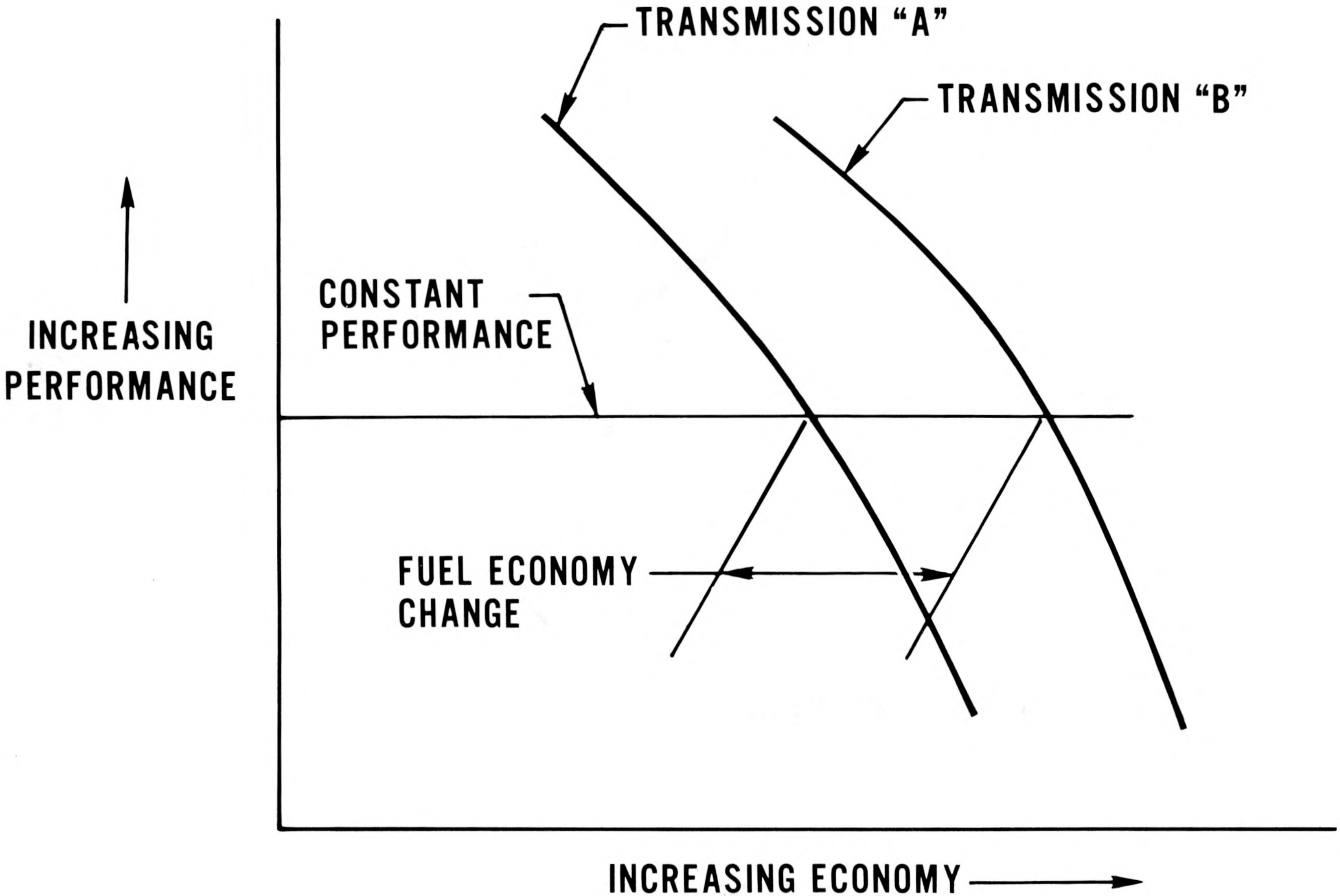

FIGURE 4

CONVENTIONAL MANUAL TRANSMISSION

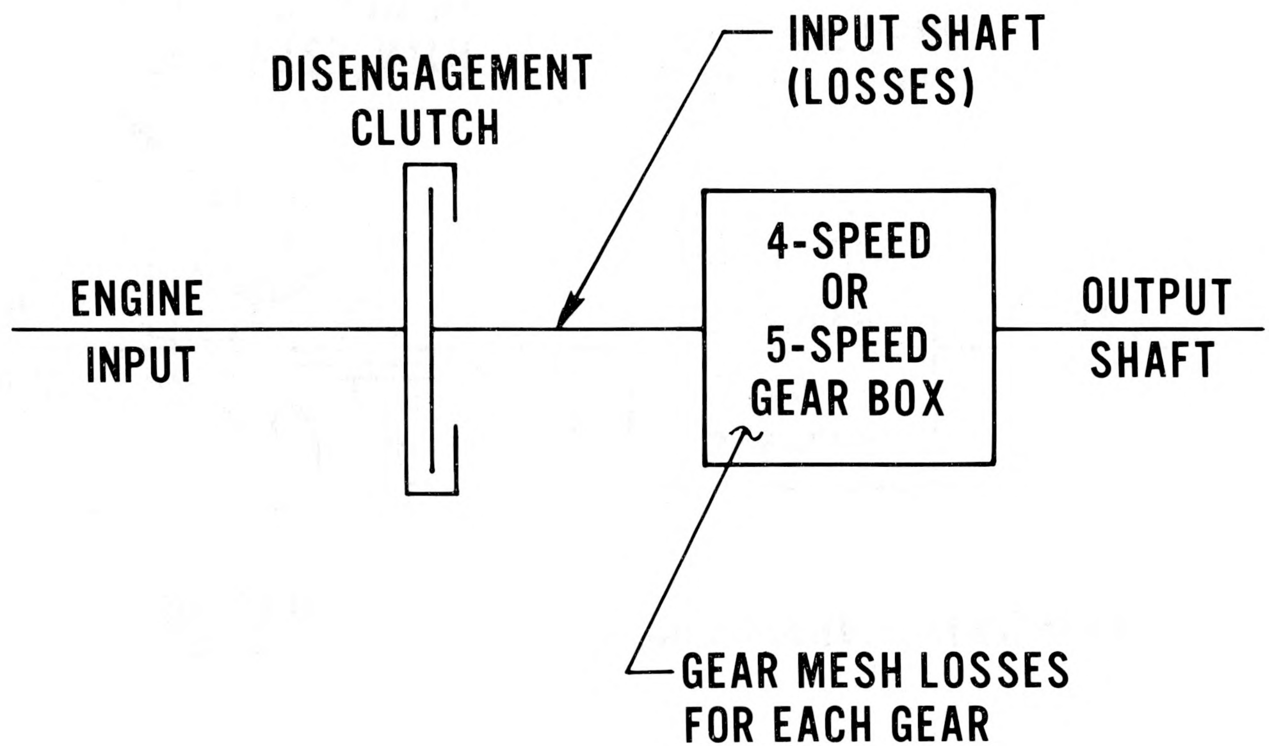

FIGURE 5

CONVENTIONAL AUTOMATIC TRANSMISSION

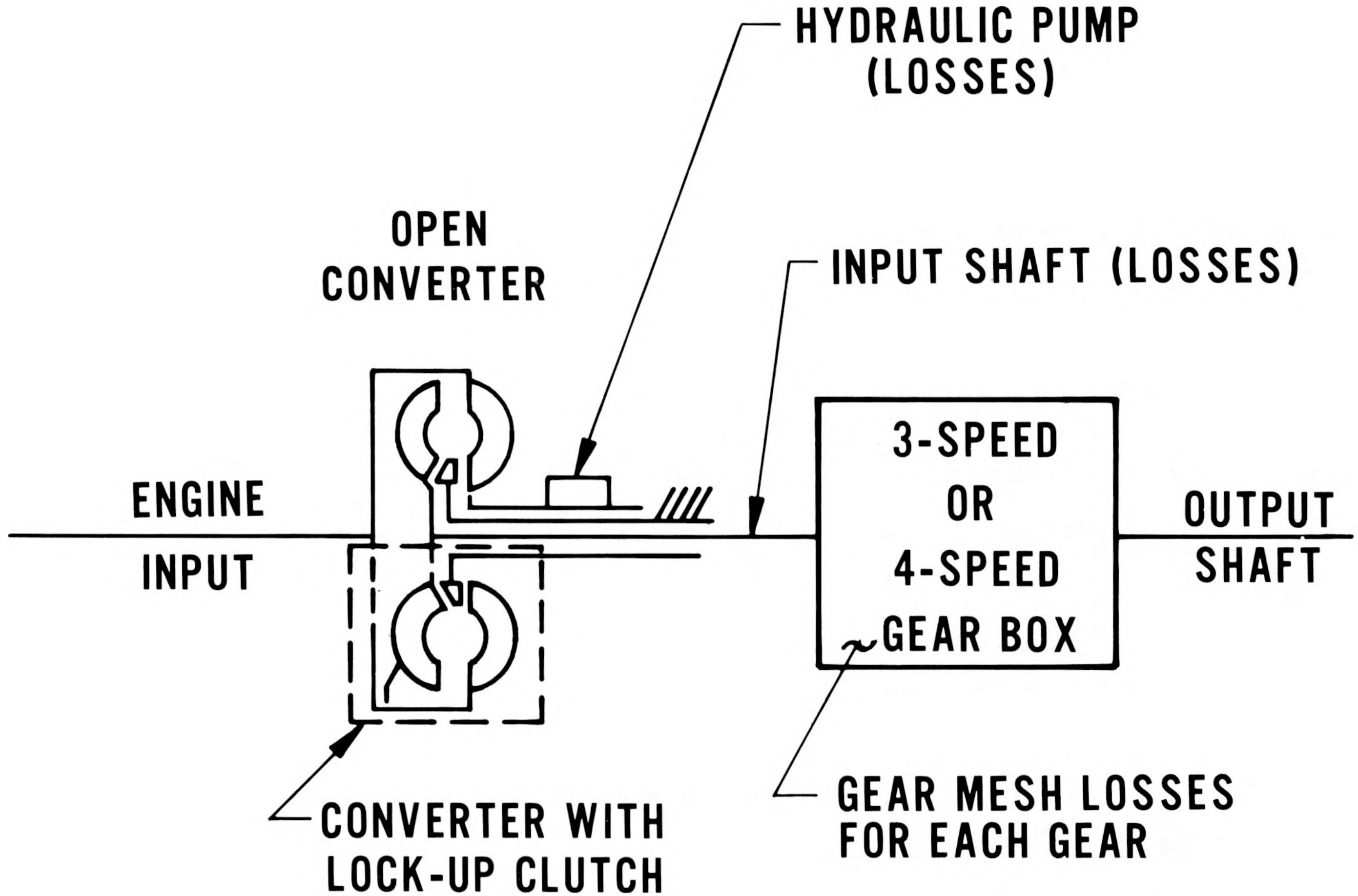

FIGURE 6

CONTINUOUSLY VARIABLE TRANSMISSION (SIMULATION)

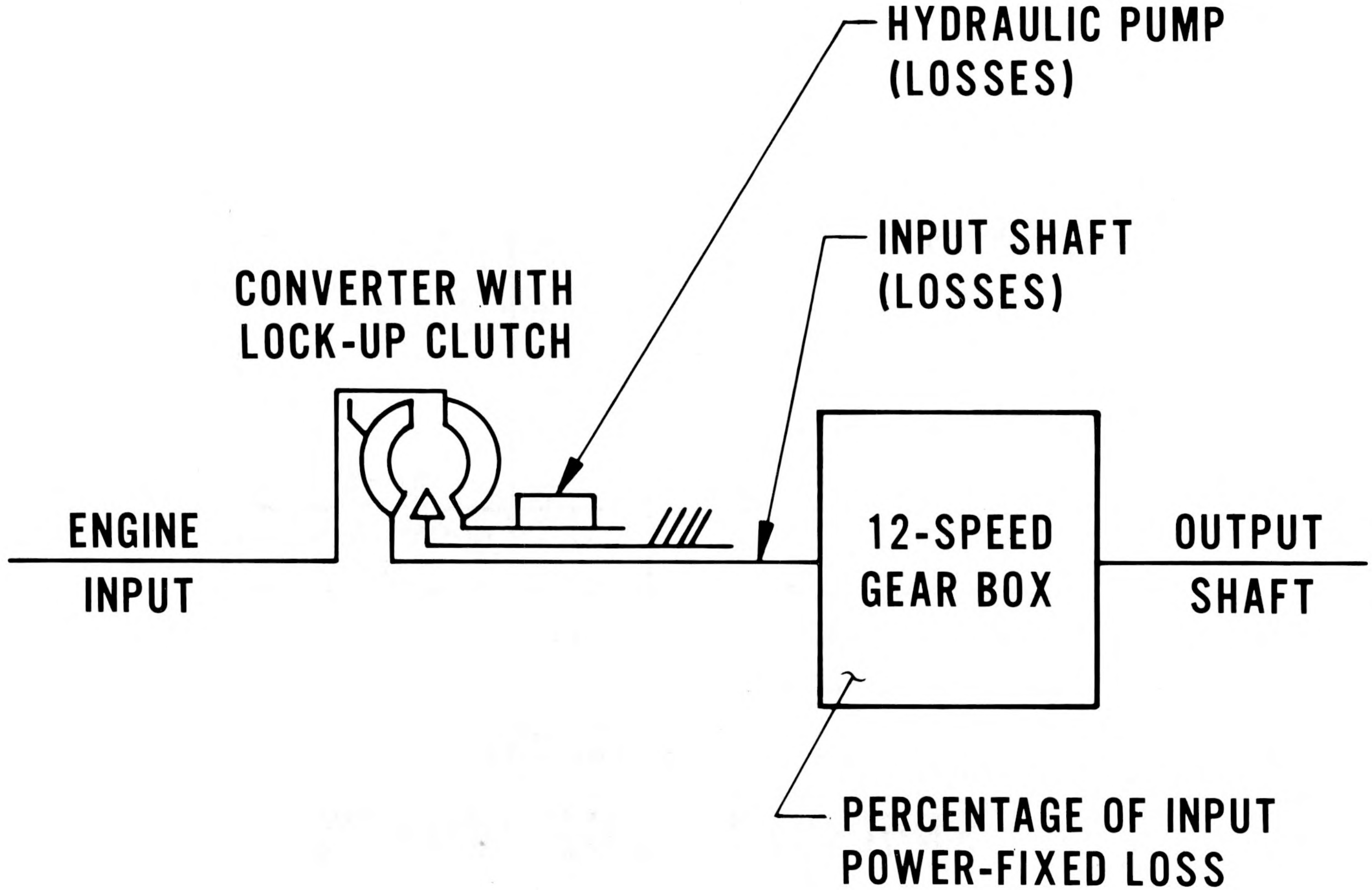

FIGURE 7

MANUAL TRANSMISSIONS
Geared Ratios

OVERALL RATIO	4-SPEED			5-SPEED
	3.56	4.72	5.94	4.72
UNIT IDENTIFICATION	A	B	C	D
GEAR NO.				
1	3.45	3.45	3.45	3.45
2	1.94	1.94	1.79	1.98
3	1.37	1.03	.97	1.42
4	.97	.73	.58	1.02
5	–	–	–	.73

FIGURE 8

AUTOMATIC TRANSMISSIONS

Geared Ratios

	3-SPEED	4-SPEED		C.V. SIMULATION	
OVERALL RATIO	2.84	4.22	6.00	4.22	6.00
UNIT IDENTIFICATION	E	F	G	H	I
GEAR NO.					
1	2.84	2.87	3.42	2.87	3.42
2	1.60	1.60	1.78	2.52	2.91
3	1.00	1.00	1.00	2.21	2.47
4		.68	.57	1.94	2.10
5				1.70	1.78
6				1.49	1.52
7				1.31	1.29
8				1.15	1.09
9				1.01	.93
10				.88	.79
11				.78	.67
12				.68	.57

FIGURE 9

OPTIMUM PERFORMANCE vs. EPA COMPOSITE ECONOMY

Base Manual Transmission

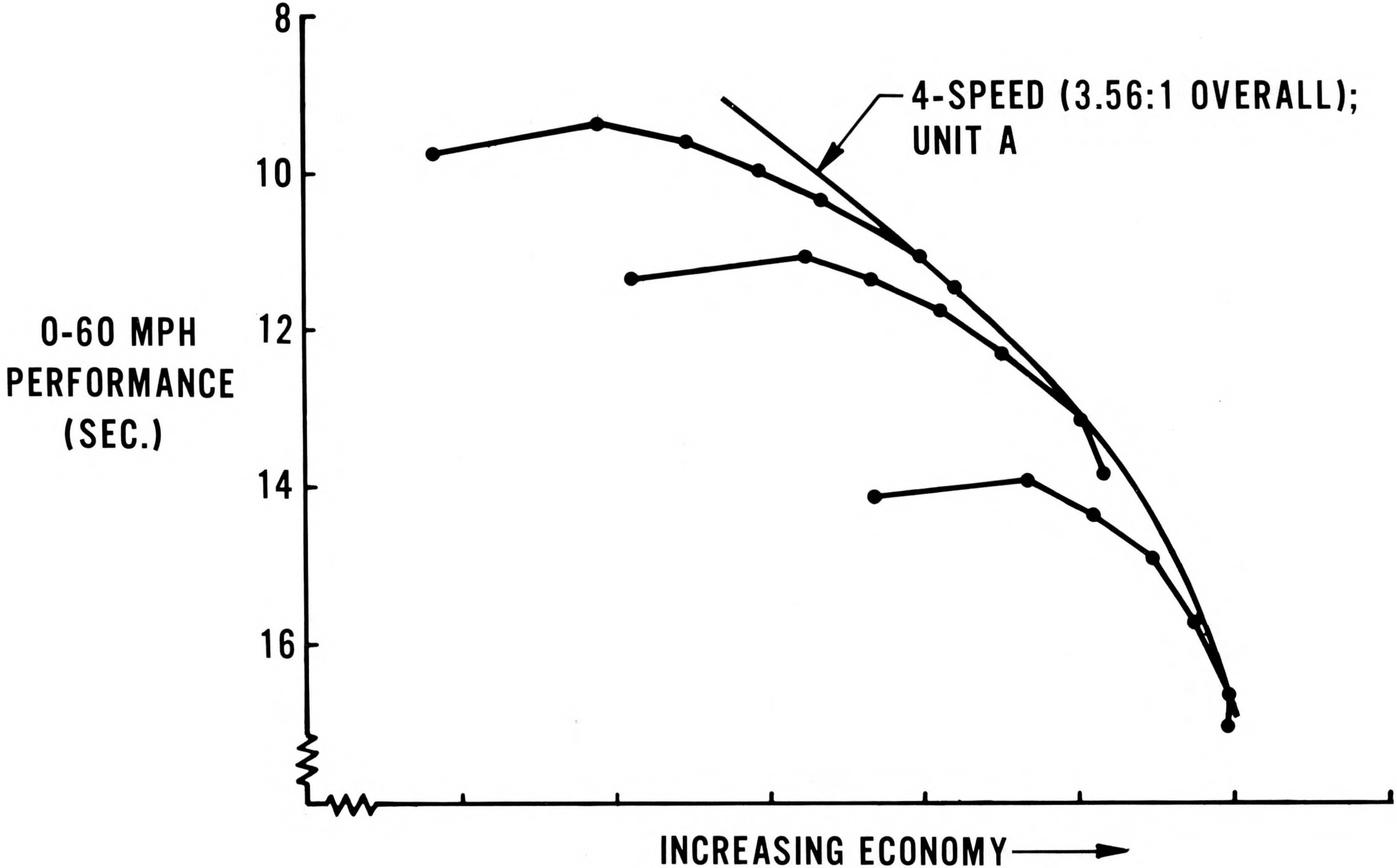

FIGURE 10

EPA COMPOSITE ECONOMY

Manual Transmissions

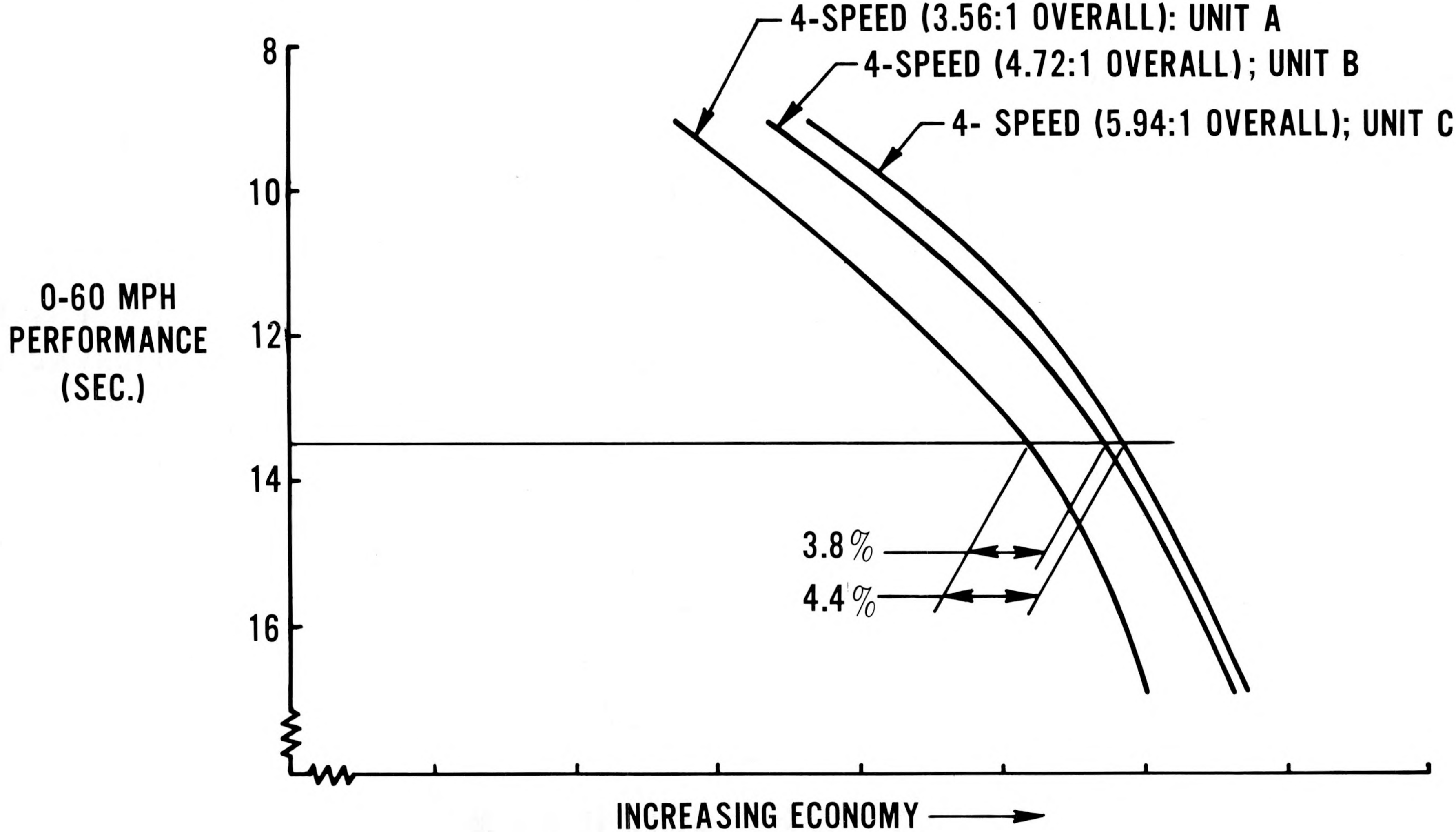

FIGURE 11

EPA COMPOSITE ECONOMY
Manual Transmissions

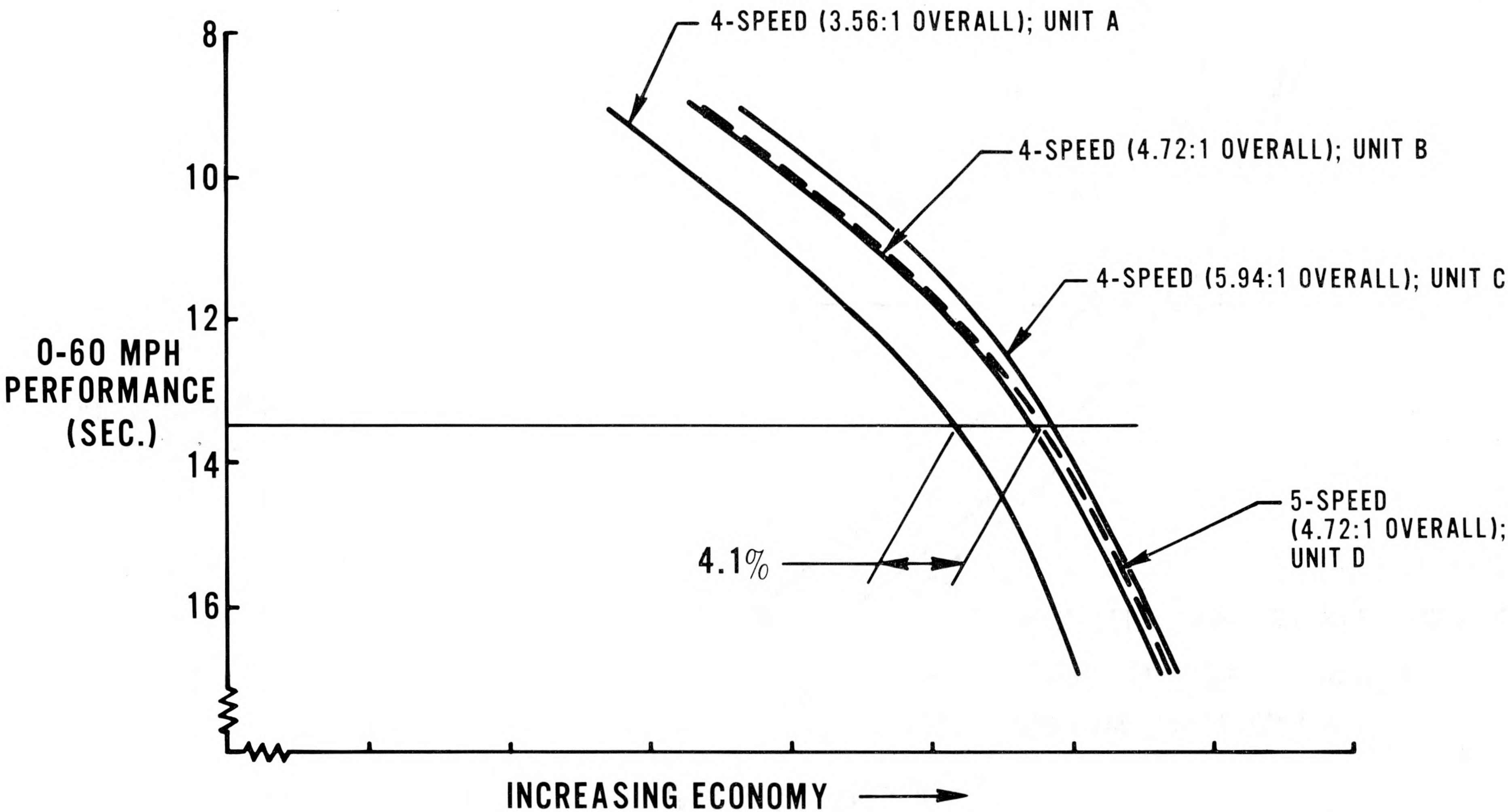

FIGURE 12

EPA COMPOSITE ECONOMY
Automatic Transmissions

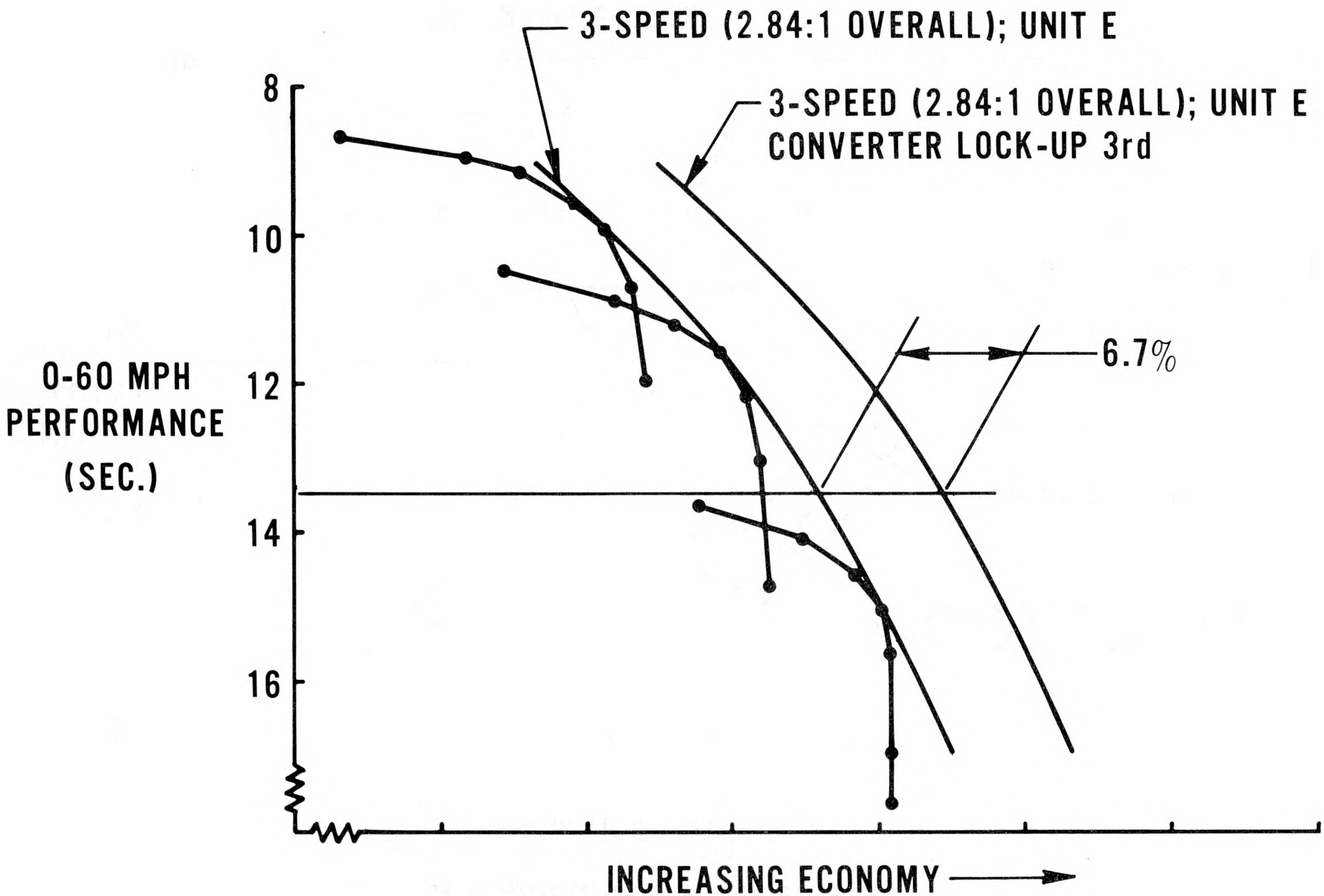

FIGURE 13

EPA COMPOSITE ECONOMY

Automatic Transmissions

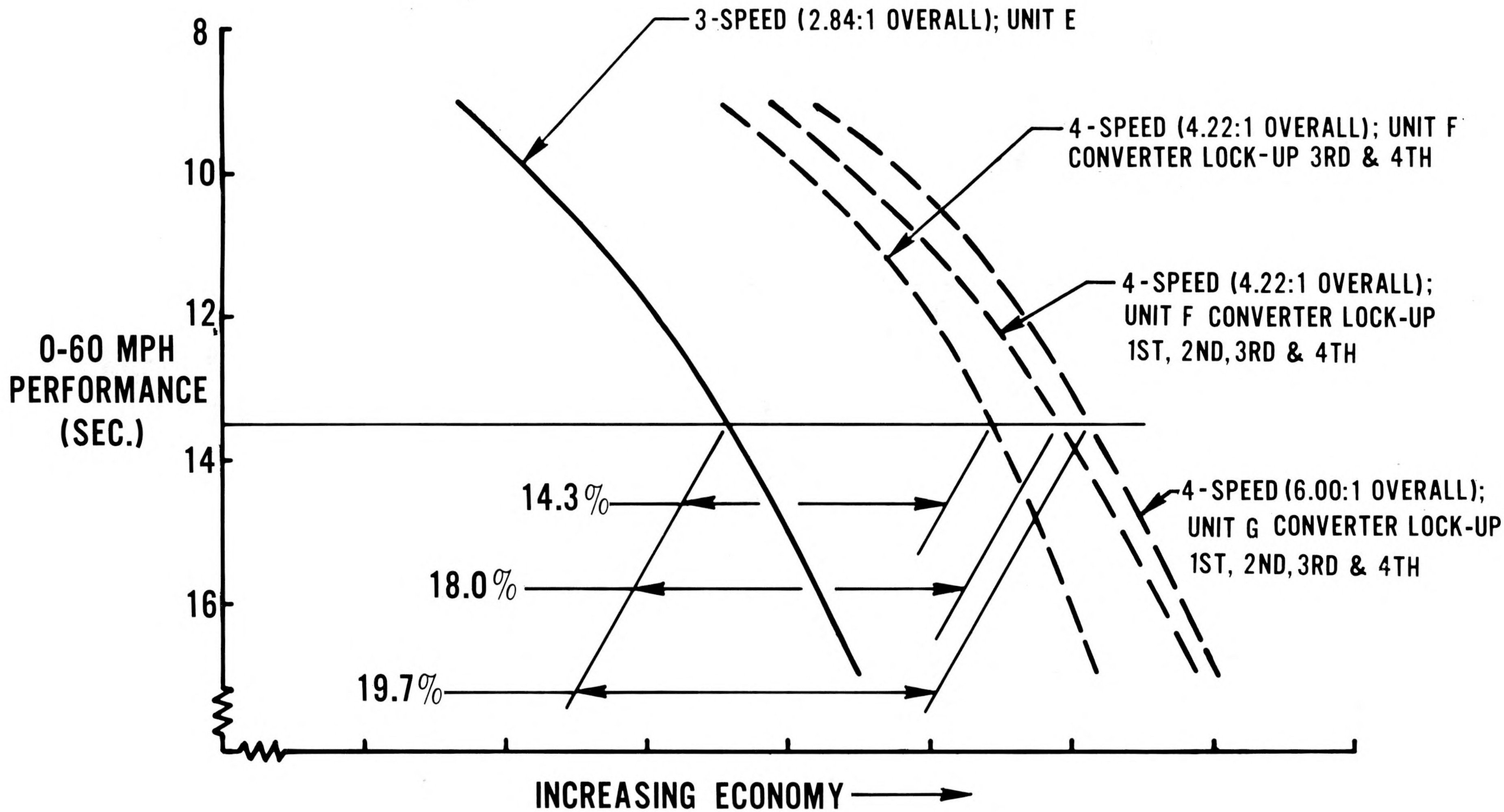

FIGURE 14

EPA COMPOSITE ECONOMY
Automatic & Continuously Variable Transmissions

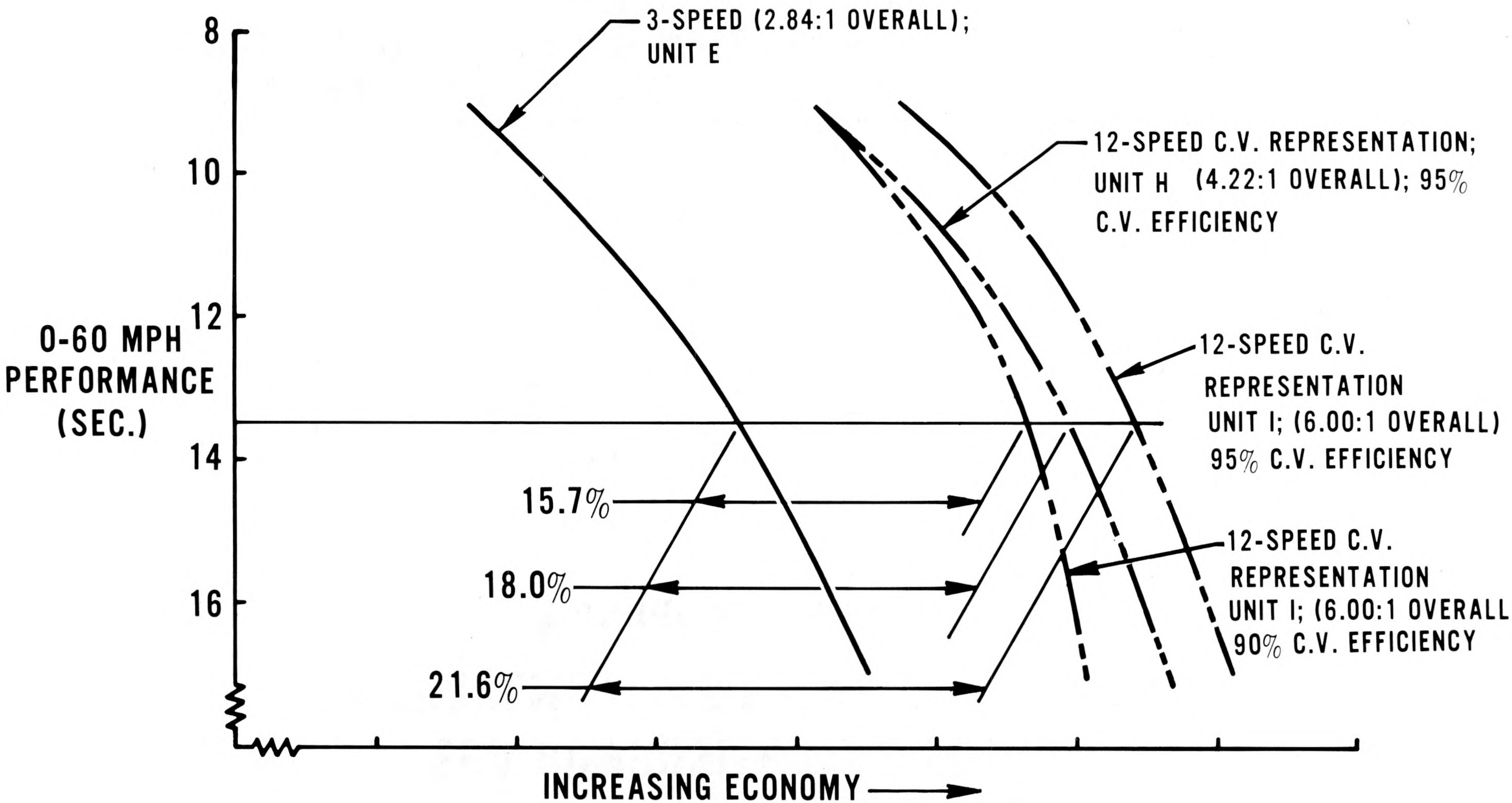

FIGURE 15

EPA COMPOSITE FUEL ECONOMY IMPROVEMENT AT EQUAL PERFORMANCE

Manual Transmissions

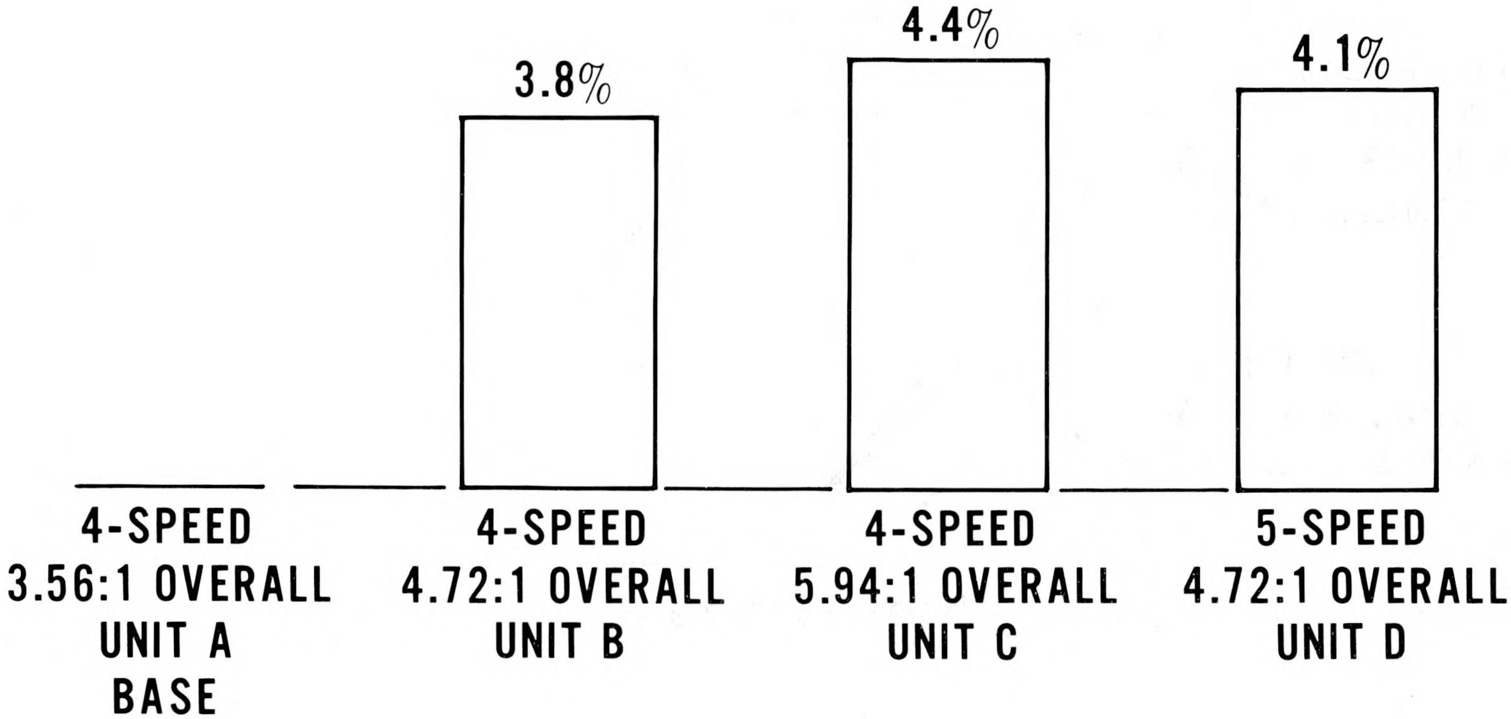

FIGURE 16

EPA COMPOSITE FUEL ECONOMY IMPROVEMENT AT EQUAL PERFORMANCE
Automatic Transmissions

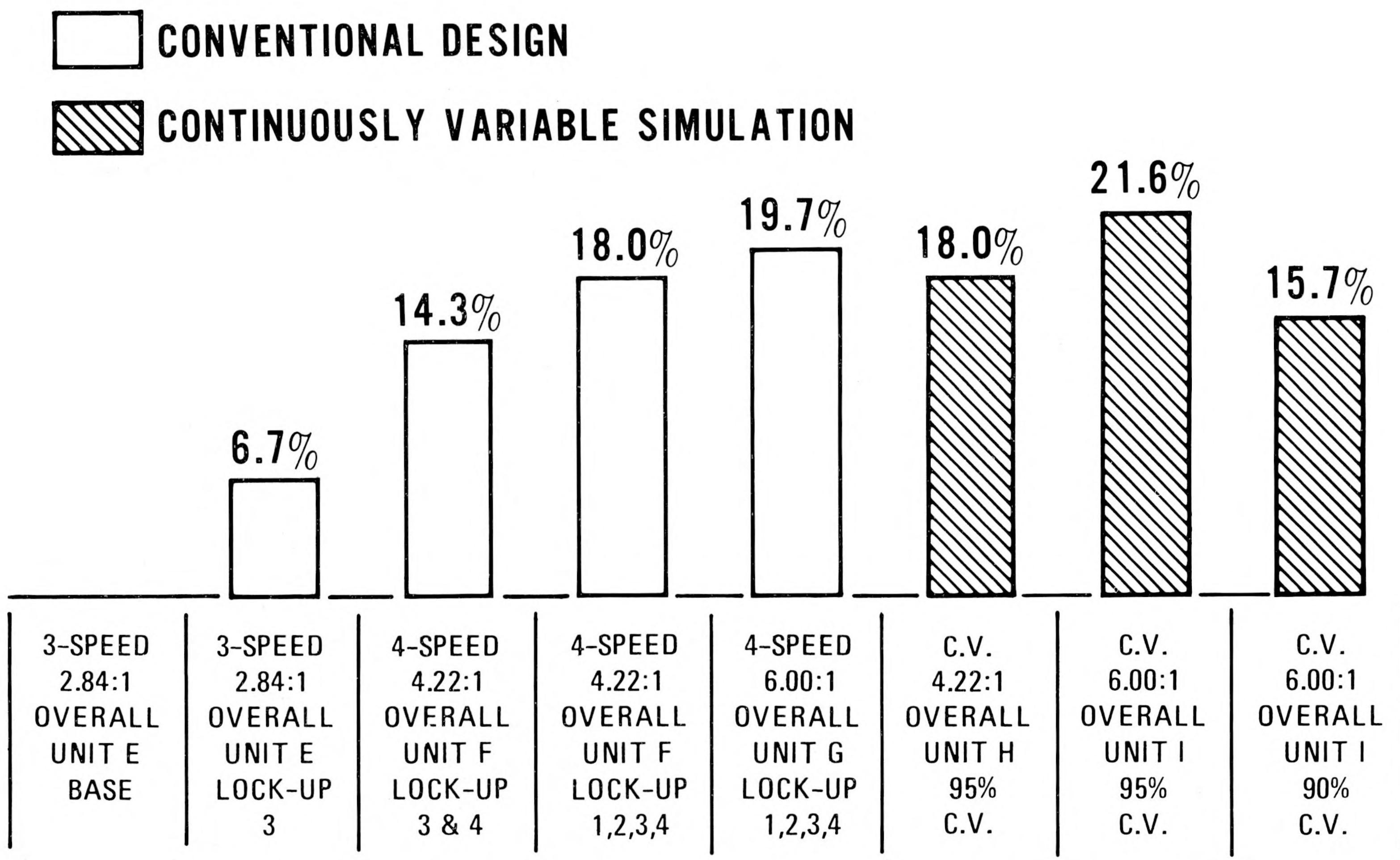

FIGURE 17

TABLE 5 – CALCULATED CORRELATION COEFFICIENTS BETWEEN SENSORY AND ANALYTICAL DATA

ENGINE INVESTIGATED		DILUTION	ANALYTICAL DATA			
			UNBURNED HC	ALIPH. ALDEHYDES	LCO	LCA
PEUGEOT 504 D IFP DATA		1 : 100	0.679	0.439	0.540	0.516
		1 : 200	0.735	0.536	0.638	0.559
		1 : 400	0.642	0.454	0.667	0.598
DATA SWRi (14) "D" RATINGS	PEUGEOT 504 D	1 : 100	0.828	0.431	0.276	0.477
	NISSAN DATSUN (Winter)		0.778	– 0.185	– 0.106	0.250
	NISSAN DATSUN (Summer)		0.643	– 0.358	– 0.371	0.314
	MERCEDES 220 D		0.383	– 0.464	0.764	0.701
	OPEL REKORD		0.654	– 0.221	0.568	0.799
	5 CARS ALL TOGETHER		0.941	0.757	0.720	0.814

HYDROGEN ENGINE NO_x CONTROL BY WATER INDUCTION

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Provo, Utah

Water induction in a hydrogen engine is an effective means to reduce production of oxides of nitrogen and also to suppress flashback. Two methods of inducting the water are compared: (1) spraying into the air-hydrogen mixture in the intake manifold and (2) spraying into the hydrogen stream prior to mixing with air. The second method is considerably more effective. Power and efficiency are essentially unaffected by water addition. This makes possible a very simple on/off design that is switched by the engine throttle.

INTRODUCTION

A hydrogen engine may be operated at low brake mean-effective-pressure (bmep) without difficulty provided that the timing is properly set for the particular mixture used. At higher bmep levels an intermittant problem of ignition in the intake manifold (flashback) is encountered. Without some means of flashback suppression the engine cannot be made to run continuously at the higher torque settings. The formation of oxides of nitrogen (NO_x) accompanies an increase in bmep.

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[1]Associate Member ASME

The amount of NO_x formed and the severity of the flashback tendency depend on many engine parameters associated with the design of the combustion chamber and the ignition system. A number of proposed "causes" of flashback are discussed by Woolley and Henriksen [1][2]. Also included in this earlier paper is a discussion of probable reasons why water induction is an effective means of controlling NO_x and suppressing flashback, together with supporting data.

In the earlier report[1], water was sprayed into the air stream just below the mixer/throttle body attachment to the intake manifold. In more recent tests we have chosen to spray the water into the hydrogen stream just before entering the mixer. The latter method is shown to be considerably more effective and has made possible simplifications in the system.

We will summarize below the conclusions reached previously with regard to water induction[1] since these conclusions still apply qualitatively. We will then show new data supporting the conclusion that induction of water into the hydrogen stream is superior and also briefly describe the water control system to be used in the next prototype.

ENGINES

The data reported herein was taken on two engines. Therefore, some differences exist because the compression ratios and intake systems are not identical.

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[2]Numbers in brackets designate References at end of paper.

A Dodge 440 CID engine was set up with twin spray nozzles to induct water into the air/hydrogen mixture in the intake manifold. The nozzles were located just below the throttle body on the manifold sidewalls. The engine was modified to have a 12:1 compression ratio.

The design incorporating water induction into the hydrogen stream was applied to a Chevrolet 454 CID engine with compression ratio of 12.9:1. A single spray nozzle was used to add water to the hydrogen stream just before entering the IMPCO 425 mixer (gaseous carburetor designed for propane and natural gas). This mixer is the same as that used on the Dodge engine.

A comparison of the engine characteristics is given in Figure 1. Customary U.S. units are used in this figure in order to make the information easily comparable to earlier data. Conversion factors to SI units are provided. As expected, the higher compression-ratio engine has better thermal efficiency, and the larger displacement engine has greater power. The amount of the difference is made larger by the increase in equivalence ratio used with the Chevrolet 454.

Changes in the shape of the curves are due to modifications in the intake systems. The Dodge engine has a stock manifold whereas the Chevrolet engine was equiped with an Edelbrock manifold tuned for optimum power near 3500 rpm.

WATER INDUCTED INTO AIRSTREAM

Data in this section applies to the Dodge 440 engine as described above.

Before a system can be operated effectively it is first essential to properly adjust the spark timing. Timing in a hydrogen engine is strongly a function of equivalence ratio (ϕ) with more advance being required at lower equivalence ratio. Timing is nearly independent of rpm and amount of inducted water. The best method found for determining the proper setting utilizes NO_x measurement.

Figure 2 shows that NO_x increases exponentially with spark advance without water induction for a given equivalence ratio. This is also found to be true with water induction.

Maximum power and efficiency occur at about the same spark advance and have a fairly wide peak. Hence the following procedure for determining proper timing is suggested: (1) Set the equivalence ratio at the desired value and hold constant. (2) Find the spark timing for best torque. (3) Retard the timing until there is a noticeable drop in torque while monitoring NO_x. There may also be a slight drop in thermal efficiency. The selected setting is a compromise of reduced torque for lower NO_x. Since water induction is such an effective means of NO_x control, we choose to select settings only slightly on the retard side of best torque (retarded best torque setting).

An increase in equivalence ratio also produces an exponential increase in NO_x. This is illustrated in Figure 3 for wide open throttle conditions. Equivalence ratios greater than $\phi = 0.6$ are not easily obtained under driving conditions without water induction or some other means of flashback suppression.

Equivalence ratio, NO_x, and temperature were measured just downstream of the exhaust valve of each cylinder (Figure 3). Mixing of the hydrogen with air is not quite uniform after leaving the mixer. The stock manifold has runners that prevents further mixing. For these reasons, the variation in equivalence ratio between cylinders was found to be around 10%. However, since NO_x is an exponential function of equivalence ratio, this slight nonuniformity is manifested as an order of magnitude difference in NO_x from cylinder to cylinder. The change was not due to irregularities in the distribution of water since the same pattern was observed for the no water case.

When a manifold without runners was substituted the fuel distribution improved but was still not completely uniform. This prompted the change in design to add water to the hydrogen stream in order to have a similar nonuniform water distribution (data shown in next section).

Exhaust temperature is nearly the same with or without water addition but correlates with equivalence ratio (Figure 4). It should be rembered that a difference in peak temperature is diminished by the thermodynamics of the gas expansion. Therefore, slight cooling can result in large reductions in NO_x. Reduction in flame speed with water addition is also believed to be important.

At constant rpm and equivalence ratio, NO_x is exponentially reduced as the water to hydrogen flow-rate ratio is increased (Figure 5).

Figure 5 also reveals that NO_X production is greater at low rpm. This is one reason why an increase in the overall gear-ratio of the vehicle is appropriate for a hydrogen engine (the other being increased power output at low road speed).

One might expect the addition of water to a hydrogen engine to cause a drop in thermal efficiency because of the energy required to vaporize the water. The data of Figure 6 shows that both efficiency and power are essentially constant over the range of practical interest. A drop in power and effieicncy is not seen until the water rate is much greater than that required for NO_X control.

WATER INDUCTED INTO HYDROGEN STREAM

After observing that the distribution of hydrogen to the various cylinders was slightly nonuniform, it was decided to add a water spray to the hydrogen stream as the hydrogen entered the mixer. In this way it was hoped that the water distribution would also be nonuniform in the same way. It is also possible, though unproven, that there is inadequate mixing to produce uniform composition within the cylinder. If so, having greater water concentrations where there is a greater fuel concentration would be advantageous.

The fineness of the spray depends upon the water pressure for a given nozzle. Once in the cylinder, coarse sprays appear to be almost as beneficial as fine sprays. Fine sprays, however, follow the hydrogen flow better to the cylinder. The induction of water ahead of the mixer allows the droplets to be further broken up as they pass through the venturi in the mixer. There is also more time for cooling of the gas stream before entering the cylinder.

This means of water induction was used on a Chevrolet 454 CID engine with a conventional throttle body to control power output. Efficiency and power for this engine with throttled governing are given in Figure 7. A slight rpm effect is observed.

The extent of the improvement in this system over the prior method of water induction into the airstream is shown in Figure 8. The line designated, "water inducted into air/H_2 stream," is an interpolation of the data of Figure 5 at an equivalence ratio of 0.73 and 3000 rpm. The data of Figure 8 was taken on the combined exhaust stream of all eight cylinders.

This data may be interpreted from two points of view: (1) For a specified water to hydrogen mass-flow ratio, oxides of nitrogen are reduced by an order of magnitude with the new technique (reduction factor = 4 at 3 water to hydrogen, 40 at 8 water to hydrogen). (2) Less than half as much water is required to reduce NO_x to a given level (40% as much).

Another benefit of the more effective system is that it permits use of a higher equivalence ratio. This provides greater power and also greater efficiency (Figure 1) since the efficiency peak for equivalence ratio is around $\phi = 0.7$.

WATER INDUCTION SYSTEM

A simple on/off water induction system is possible since (a) water is not required at low load and (b) excess water is not detrimental to power and efficiency. Earlier designs attempted to maintain a constant ratio of water to hydrogen flow rates [2,3,4,5].

At idle and low load a solenoid valve between the spray nozzle and pump is closed. NO_x production is as shown in Figure 9 for the "no water" case. Power and NO_x increase as the throttle is opened. With an equivalence ratio of 0.73, intermittant flashback is encountered before reaching wide open throttle. In order to supress flashback and reduce NO_x, the water solenoid is opened at a preset point of travel of the throttle linkage. At all higher throttle settings the water flow rate remains constant. Hence, the ratio of water to hydrogen flow rates is greatest when the valve first opens.

Shown in Figure 9 are two water flow rates, both initiated at approximately the same throttle position. The water to hydrogen ratio when the valve first opens is 5.0 for the middle line and 8.5 for the lower line. Dashed lines indicate the approximate position of the lines of constant ratio of water to hydrogen mass flow rates.

Two settings are required: (1) the switch position on the throttle linkage and (2) the supply pressure of the pump (corresponding to water flow rate). The amount of NO_x produced over an emissions test cycle depends on these two choices.

CONCLUSIONS

(1) Water induction is an effective means of both suppressing flashback and reducing NO_x in a hydrogen engine.

(2) Adding the water to the hydrogen stream is more effective than adding it to the air/hydrogen mixture.

(3) The level of NO_x production is an exponential function of several parameters: spark advance, equivalence ratio, and ratio of water to hydrogen mass-flow rates.

(4) Excessive water is not detrimental to engine power and efficiency in the range of interest.

(5) A simple on/off design of the water system activated by the throttle linkage is adequate.

ACKNOWLEDGEMENTS

The authors are indebted to D.L. Henriksen and E.H. Davis for their assistance in converting and testing the Dodge 440 and Chevrolet 454 engines respectively.

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[3] Henriksen, D.L., Mackay, D.B., and Anderson, V.R., "Prototype Hydrogen Automobile Using a Metal Hydride," 1st World Hydrogen Energy Conference, Vol, 3, 7C-1, Miami Beach, Florida, March 1976.

[4] Billings, R.E., "A Hydrogen Powered Mass Transit System," 1st World Hydrogen Energy Conference, Vol. 3, 7C-27, Miami Beach, Florida, March 1976.

[5] Woolley, R.L., "Performance of a Hydrogen-Powered Transit Vehicle," Eleventh IECEC, Vol. 1, 769174, Stateline, Nevada, Sept. 1976, pp. 992-996.

FIGURE CAPTIONS

Figure 1. Engine maps for Dodge 440 CID and Chevrolet 454 CID engines with hydrogen fuel. (0.7457 KW/Hp = 1, 1.3558 $N \cdot m/ft \cdot lb_f = 1$)

Figure 2. Effect of spark advance on NO_x, power, and efficiency.

Figure 3. Oxides of Nitrogen vs. equivalence ratio measured at each exhaust valve.

Figure 4. Temperature of manifold exhaust gas at each valve.

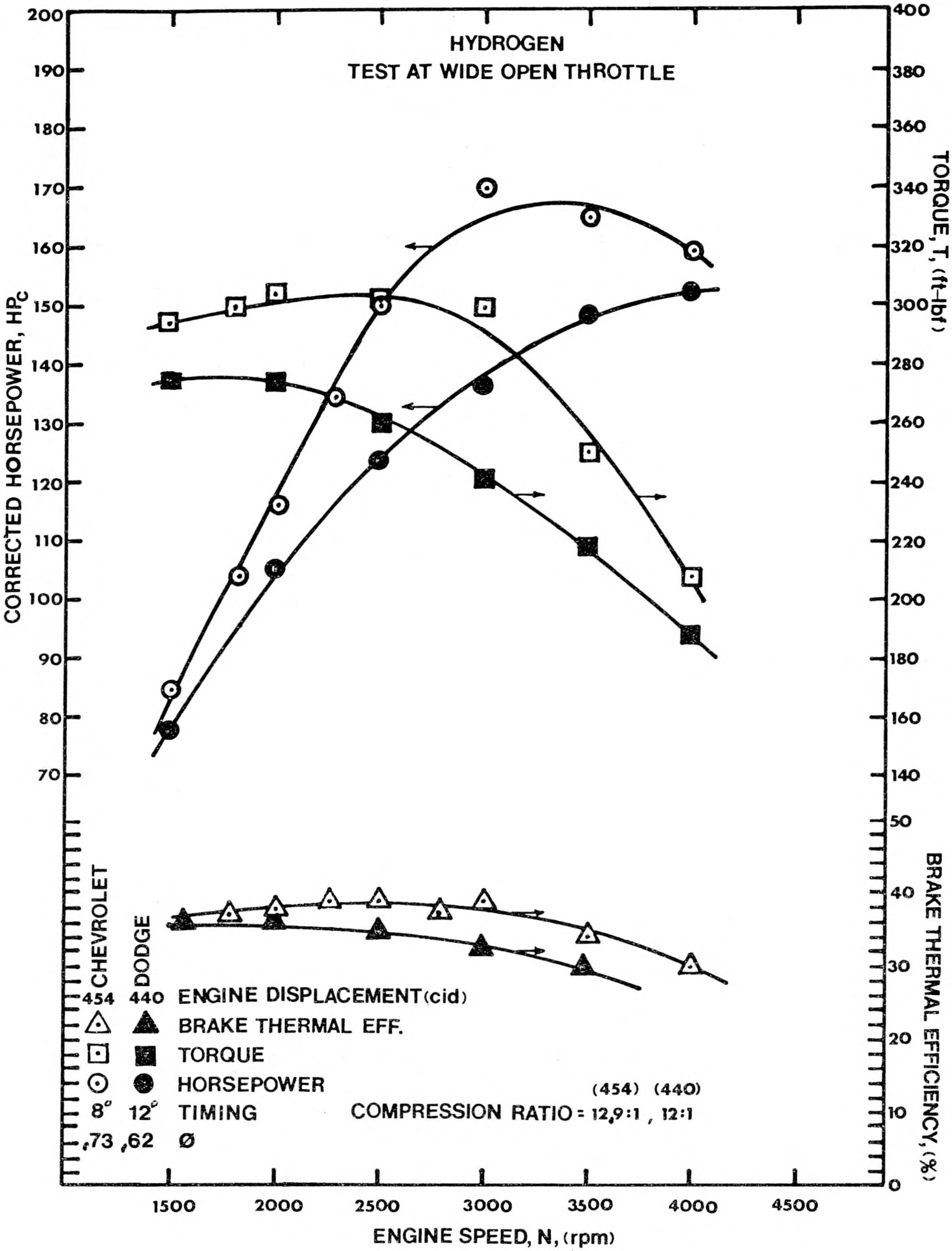
Figure 5. Effect of water, equivalence ratio, and engine speed on NO_x formation.

Figure 6. Effect of water on efficiency and power.

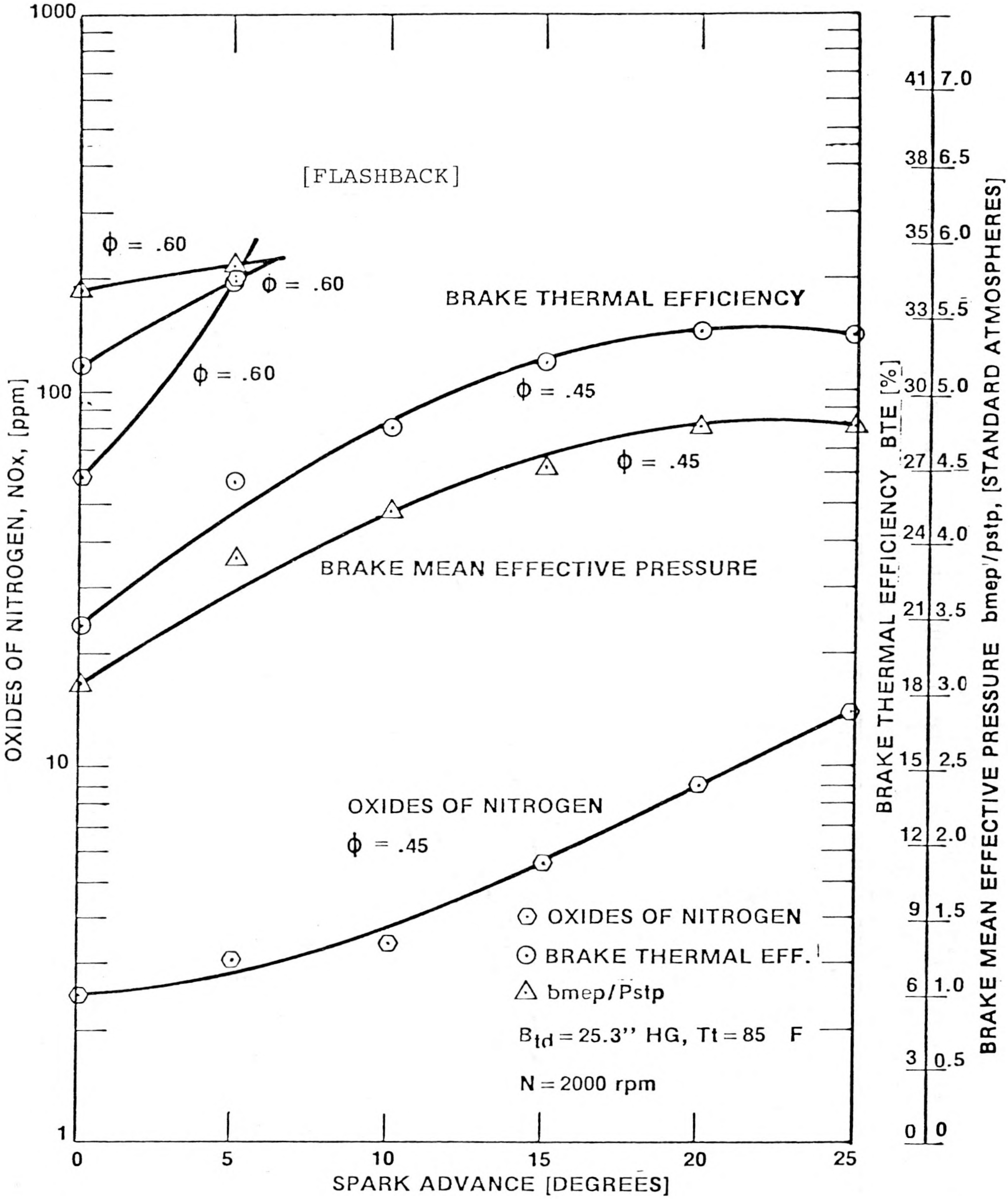
Figure 7. Efficiency vs. power of a throttled hydrogen engine.

Figure 8. Comparison of two methods for inducting water into a hydrogen engine.

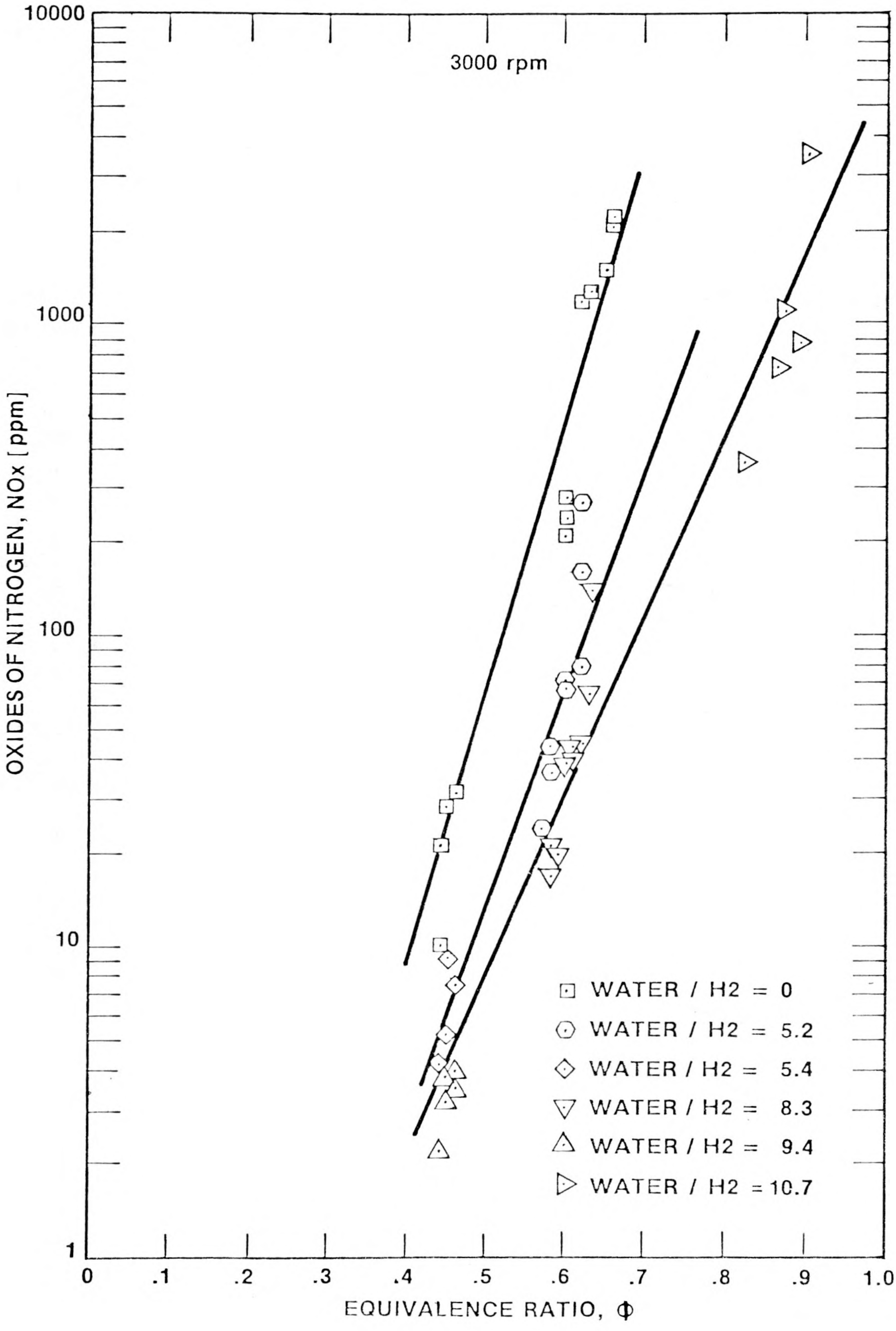
Figure 9. Performance of on/off water system.



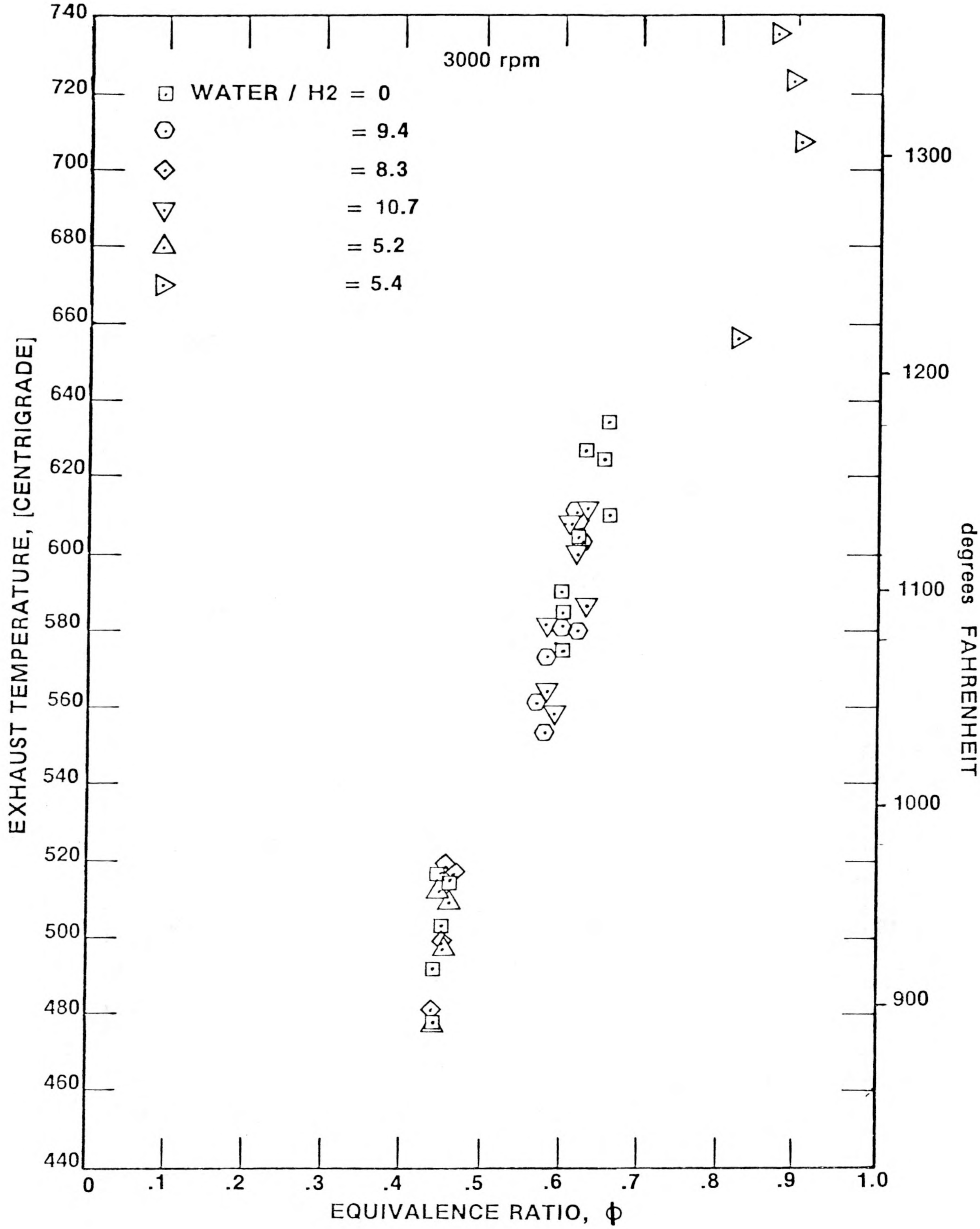
HYDROGEN
TEST AT WIDE OPEN THROTTLE
CORRECTED HORSEPOWER, HP_C
TORQUE, T, (ft-lbf)
BRAKE THERMAL EFFICIENCY, (%)
ENGINE SPEED, N, (rpm)
CHEVROLET
DODGE
454 440 ENGINE DISPLACEMENT (cid)
BRAKE THERMAL EFF.
TORQUE
HORSEPOWER
8° 12° TIMING
.73 .62 Ø
COMPRESSION RATIO = (454) 12.9:1, (440) 12:1
200
190
180
170
160
150
140
130
120
110
100
90
80
70
400
380
360
340
320
300
280
260
240
220
200
180
160
140
50
40
30
20
10
0
1500
2000
2500
3000
3500
4000
4500

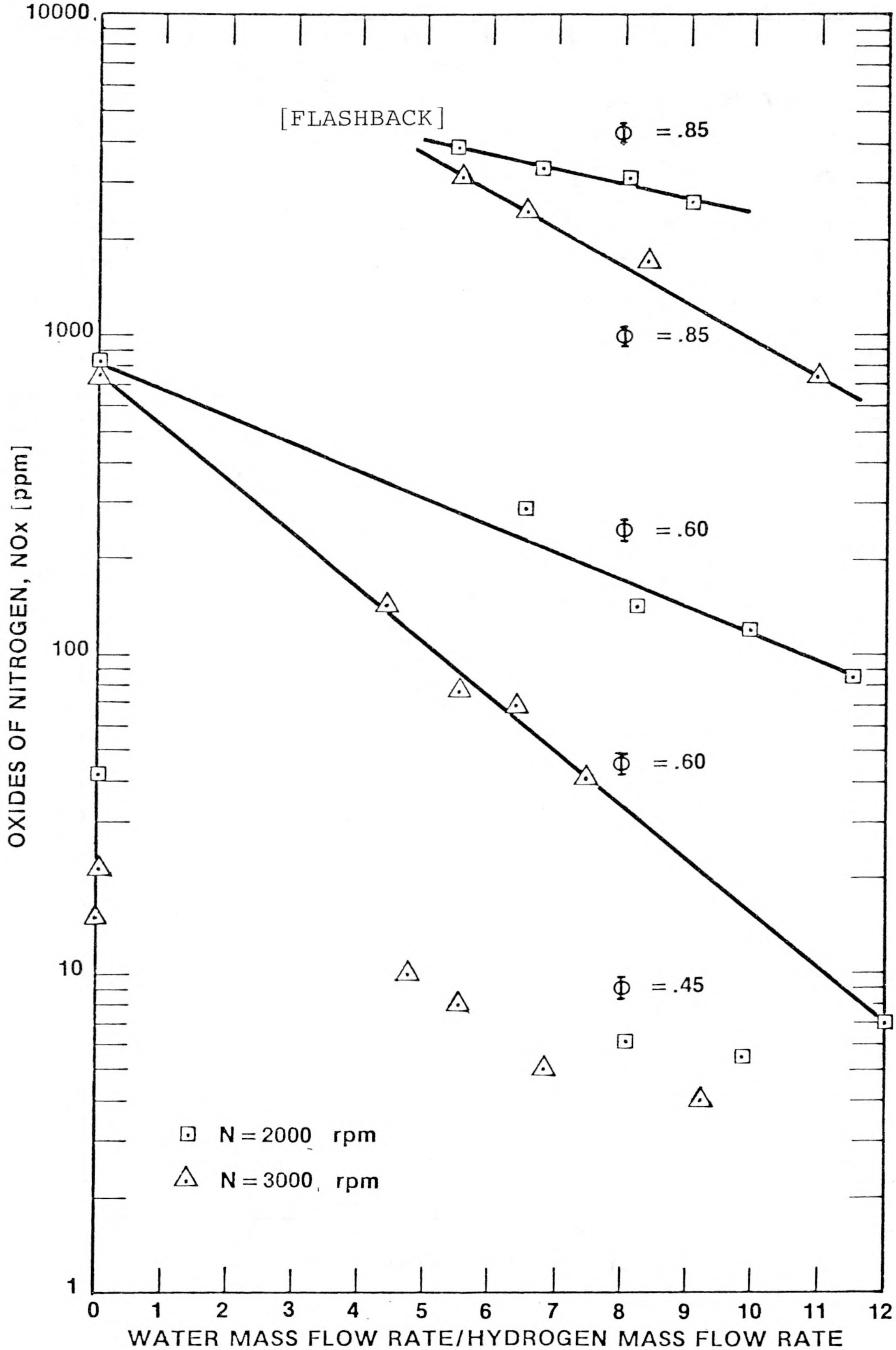


[FLASHBACK]
ϕ = .60
ϕ = .60
ϕ = .60
BRAKE THERMAL EFFICIENCY
ϕ = .45
ϕ = .45
BRAKE MEAN EFFECTIVE PRESSURE
OXIDES OF NITROGEN
ϕ = .45
OXIDES OF NITROGEN
BRAKE THERMAL EFF.
bmep/Pstp
B_td = 25.3'' HG, Tt = 85 F
N = 2000 rpm
OXIDES OF NITROGEN, NOx, [ppm]
SPARK ADVANCE [DEGREES]
BRAKE THERMAL EFFICIENCY BTE [%]
BRAKE MEAN EFFECTIVE PRESSURE bmep'/pstp, [STANDARD ATMOSPHERES]

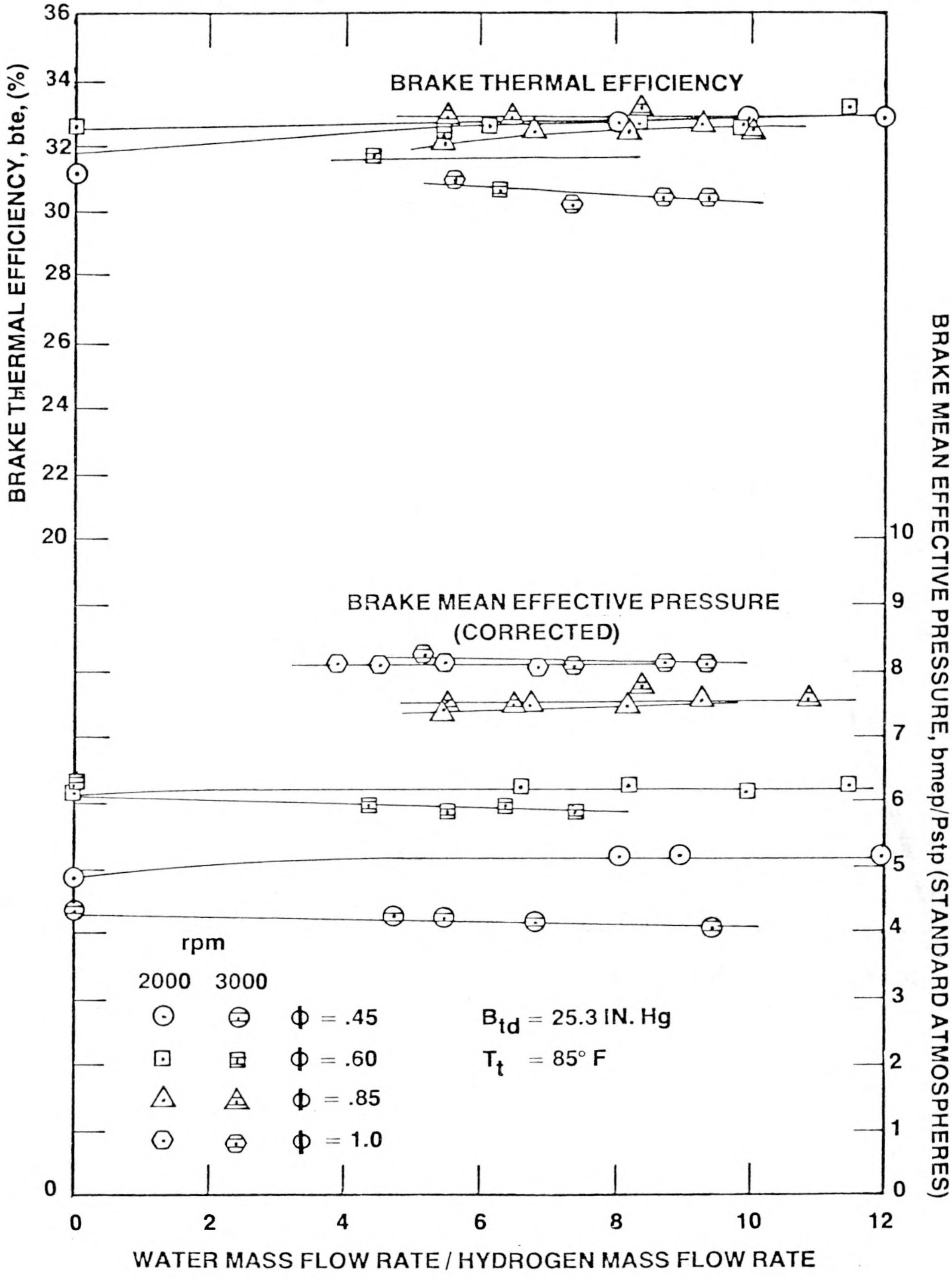


3000 rpm
OXIDES OF NITROGEN, NOx [ppm]
10000
1000
100
10
1
0
.1
.2
.3
.4
.5
.6
.7
.8
.9
1.0
EQUIVALENCE RATIO, Φ
WATER / H2 = 0
WATER / H2 = 5.2
WATER / H2 = 5.4
WATER / H2 = 8.3
WATER / H2 = 9.4
WATER / H2 = 10.7

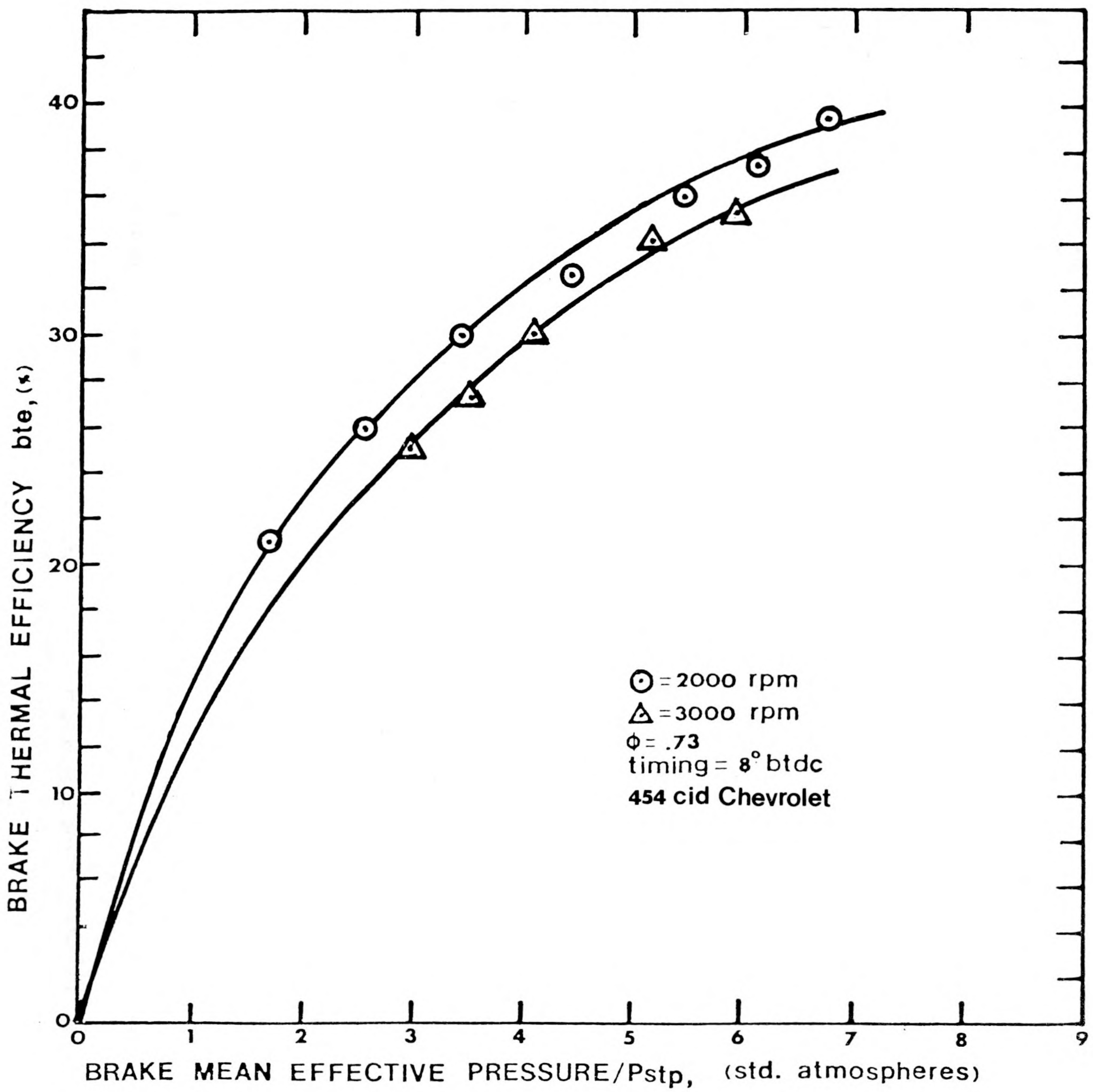

3000 rpm
WATER / H2 = 0
= 9.4
= 8.3
= 10.7
= 5.2
= 5.4
EXHAUST TEMPERATURE, [CENTRIGRADE]
740
720
700
680
660
640
620
600
580
560
540
520
500
480
460
440
degrees FAHRENHEIT
1300
1200
1100
1000
900
0
.1
.2
.3
.4
.5
.6
.7
.8
.9
1.0
EQUIVALENCE RATIO, φ

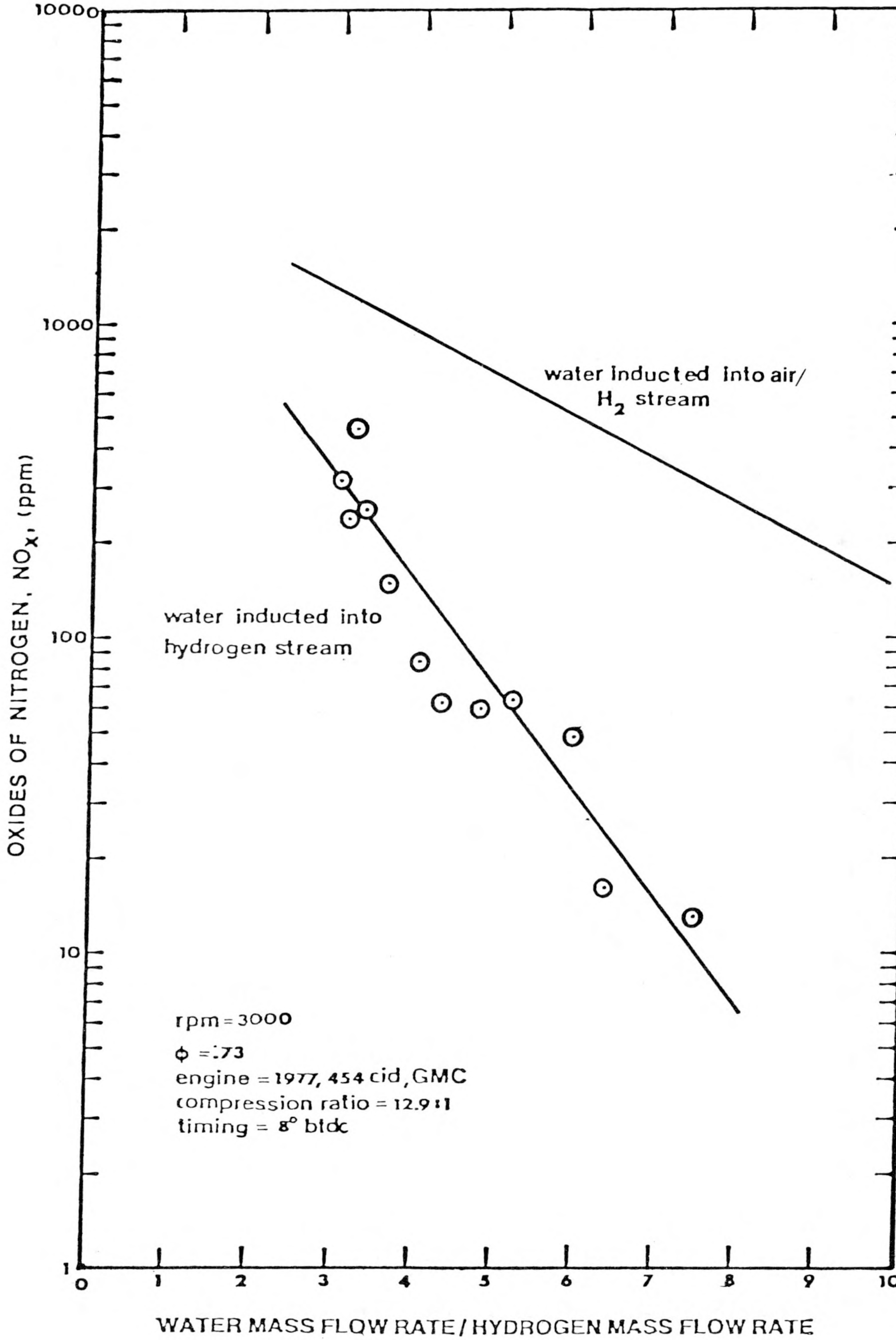


10000
1000
100
10
1
[FLASHBACK]
Φ = .85
Φ = .85
Φ = .60
Φ = .60
Φ = .45
N = 2000 rpm
N = 3000 rpm
0 1 2 3 4 5 6 7 8 9 10 11 12
OXIDES OF NITROGEN, NOx [ppm]
WATER MASS FLOW RATE/HYDROGEN MASS FLOW RATE

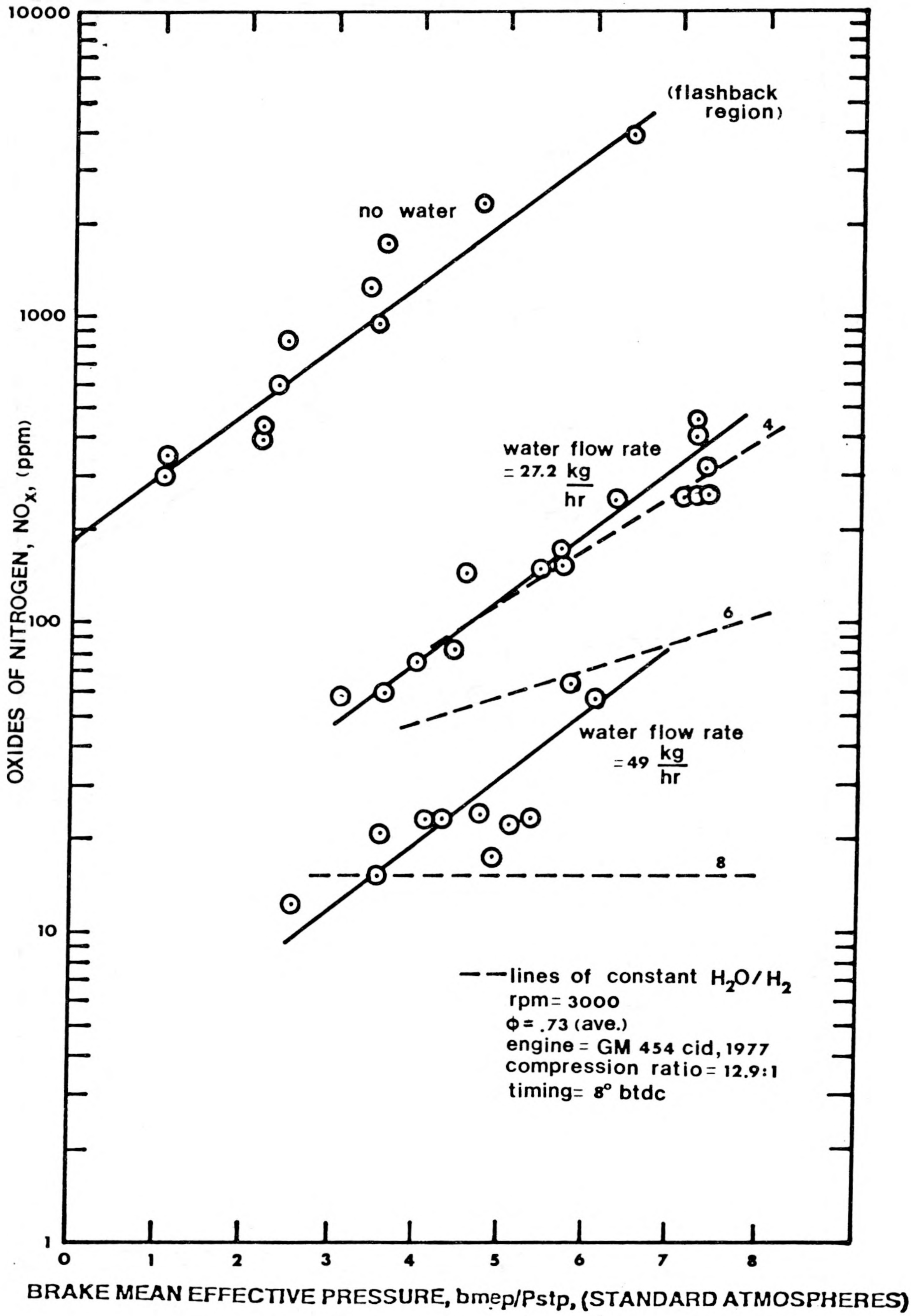


BRAKE THERMAL EFFICIENCY
BRAKE MEAN EFFECTIVE PRESSURE (CORRECTED)
BRAKE THERMAL EFFICIENCY, bte, (%)
BRAKE MEAN EFFECTIVE PRESSURE, bmep/Pstp (STANDARD ATMOSPHERES)
WATER MASS FLOW RATE / HYDROGEN MASS FLOW RATE
rpm
2000 3000
Φ = .45
Φ = .60
Φ = .85
Φ = 1.0
B_td = 25.3 IN. Hg
T_t = 85° F


40
30
20
10
0
BRAKE THERMAL EFFICIENCY bte, (%)
0
1
2
3
4
5
6
7
8
9
BRAKE MEAN EFFECTIVE PRESSURE/Pstp, (std. atmospheres)
⊙ = 2000 rpm
△ = 3000 rpm
ϕ = .73
timing = 8° btdc
454 cid Chevrolet



10000
1000
100
10
1
0
1
2
3
4
5
6
7
8
9
10
OXIDES OF NITROGEN, NO_x, (ppm)
WATER MASS FLOW RATE / HYDROGEN MASS FLOW RATE
water inducted into air/ H_2 stream
water inducted into hydrogen stream
rpm = 3000
φ = .73
engine = 1977, 454 cid, GMC
compression ratio = 12.9:1
timing = 8° btdc



10000
1000
100
10
1
0
1
2
3
4
5
6
7
8
OXIDES OF NITROGEN, NO_x, (ppm)
BRAKE MEAN EFFECTIVE PRESSURE, bmep/Pstp, (STANDARD ATMOSPHERES)
(flashback region)
no water
water flow rate = 27.2 kg/hr
4
6
8
water flow rate = 49 kg/hr
lines of constant H_2O/H_2
rpm = 3000
Φ = .73 (ave.)
engine = GM 454 cid, 1977
compression ratio = 12.9:1
timing = 8° btdc

Dr. -Ing. E. Plassmann, Dr. -Ing. H. Waldeyer, Dipl. -Ing. D. Hassel

Exhaust Gas Emission Factors in the Motor Traffic Field - Their Impact on Air Pollution Control

The number of motor vehicles in the Federal Republic of Germany more than decupled in the past 25 years as it increased from about 2 million vehicles in 1950 to more than 21 million vehicles in 1975 /1/. In addition to industry, trade and household, therefore, road traffic has become a factor considerably contributing to air pollution particularly by:

- substances other than air constituents
- noise emissions
- impairment of subsoil water and air by fuel lost during vehicle refuelling operations
- scrap and waste materials.

So-called cadastral emission surveys can be used to show the extent to which the most important emission groups - in terms of quantity and quality, location and time-contribute to air pollution, i. e. to total emission and the air pollution resulting therefrom. Since the German Federal Air Pollution Control Act took effect in 1974 the competent state authorities in the Federal Republic of Germany have been liable by law to provide such cadastral surveys for selected areas /2/.

Already in 1970, the Technischer Überwachungs-Verein Rheinland in the course of development of an emission cadastre model managed to establish a system for determination of motor traffic emissions which for the first time was tested in the "large area of Cologne" with good results and which in the meantime was applied to other air polluted areas in the Federal Republic of Germany /3, 4, 5, 6, 7/.

Systematics Adopted to Determine Motor Vehicle Exhaust Gas Emission Factors

Determination of exhaust emissions from in use motor vehicles by measurement of actual emissions is not practicable for a number of reasons whereas the total emission can be assessed from traffic specific data and such information that depicts the mean emission behavior of a state's motor vehicle fleet. Several reports have been presented on this method /8, 9/.

Extensive investigations revealed that the mean exhaust emission from a given fleet of motor vehicles substantially depends on two parameter categories:

- the composition of the motor vehicle fleet

- the mode of operation or performance of the vehicle fleet.

The emission-related composition of the motor vehicle fleet in the Federal Republic of Germany can be determined by means of criteria such as type of vehicle, cylinder displacement, horsepower, year of first registration etc. forming part of the annual statistics as published by the competent authorities. This results in a quasi-representative cross-section of the existing motor vehicle fleet.

The mode of operation of this motor vehicle sample depends on traffic-specific conditions such as type of road, road routing and road state, traffic density etc. Investigations have shown that the plurality of potential traffic situations and their characterizing parameters such as percentage shares of time of standstill, acceleration, deceleration and constant speed can be represented by the mean driving speed as characteristic value.

The traffic situations frequently encountered in urban traffic are described in Table 1.

Driving Mode	Velocity Range (km/h)	Traffic Situation
1	>65	Fast-Moving Traffic on Highways
2	55 – 65	Traffic on Outward-Bound Roads
3	30 – 55	Fast-Moving Through-Traffic
4	22 – 30	Fast-Moving City Traffic
0	17 – 22	Comparsion Cycle to the European Test
5	10 – 17	Slow-Moving City Traffic
6	2 – 10	Jammed Roads
7	0 – 2	Traffic Jams
8	0	Cold Starting, Heating up to the Engine

Table 1: Driving Modes for Description of Road Traffic (Passenger Cars)

They are correlated to specific driving cycles which are also referred to as driving modes. Each driving mode is a sequence of idling, acceleration, constant speed and deceleration as shown in Fig. 1 and compared with the European Test Cycle /10/. These driving modes are simulated on a chassis dynamometer to determine the pertinent exhaust emission according to the measurement method prescribed for the European Test Procedure. As a result of these determinations we obtain for each type of vehicle the mean exhaust emission as a function of mean driving speed. The emissions correlated to each of the various driving modes are referred to as emission factors. Consistent with the purpose of the test, namely to determine the total pollutants emitted by the road traffic, mean emission factors are formed from the plurality of individual

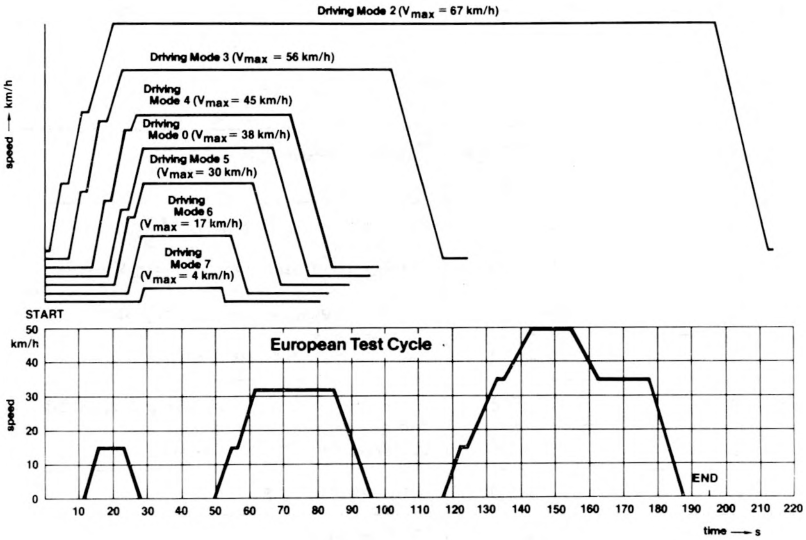


Fig. 1: Driving Modes for Exhaust Emission Testing for Passenger Cars

measurements: the total emissions then can be calculated on the basis of these mean emission factors in conjunction with the traffic-specific data (traffic density and mean driving speed) as applicable.

Test Performance

The investigation presently under way by order of the German Federal Ministry of the Interior respectively the German Federal Office for Environmental Affairs is performed on a selection of 100 passenger cars. This selection represents the fleet of passenger cars and caravans in the Federal Republic of Germany for the 1975 reference year adequately in respect of emission-specific parameters /11, 12/.

The test program comprises the previously described driving mode tests and the European Test Cycle as per ECE R 15 for comparison purposes.

Each vehicle is tested in accordance with this program first in its as-received state and then after maintaining and engine tuning. This way it is possible to determine the influence of the vehicle's maintenance state on exhaust emissions and fuel consumption.

Maintenance state in this context does not mean the outer appearance of the vehicle (good order condition) and compliance with manufacturer's recommended inspection intervals, but its state after maintaining of ignition and carburettor only as essential emission-relevant variables. Comparison of these adjustment data of the as-received vehicles with manufacturer's prescribed values permits to characterize the maintenance state of a vehicle insofar as emission behaviour is concerned. The likewise emission-relevant valve clearance adjustment is not checked for reasons of time and costs.

The emissions of such components as carbon monoxide, unburnt hydrocarbons and nitrogen oxides are determined by the collecting bag method both for the driving mode tests and the European Test Cycle. Toxic substance concentrations are determined with the aid of an infrared analyzer (NDIR) for the CO component and by means of both an n-hexane sensibilized infrared analyzer and a flame ionization detector (FID) for unburnt hydrocarbons. Determination of nitrogen oxides is effected with the aid of a chemiluminescence detector (CLD).

Test Results

Exhaust emission factors determined by the systematics described before are basic data for emission cadastral surveys to be established on for road traffic. Their availability is moreover a condition to be satisfied for atmospheric dispersion calculations established to determine air pollution situations due to motor vehicle emissions. The variation with time of the emission factors reflects the influence of statutory restrictions and the technical progress. The efficiency of statutory regulations can be checked on by corresponding evaluations and comparisons.

Exhaust Gas Emission Factors for the 1975 Reference Year

The exhaust emissions and fuel consumption related to the driven distance as a function of mean driving speed are shown in Fig. 2 for the 1975 motor vehicle fleet of the Federal Republic of Germany. These results have been obtained from measurements performed on a total of 84 vehicles. They affirm the known tendency that exhaust emissions and fuel consumption heavily increase as the mean driving speed reduces. It is solely the NO_x emission that is absolutely minimum at very low speeds (Driving Mode No. 7) and that increases at growing speed. The obvious conclusion that exhaust emissions may be diminished by increasing the mean driving speed, i. e. by improving traffic flow, is generally correct except for the NO_x component, but there is to be taken into account that the scale of the respective traffic-specific measures liable to result in traffic flow improvement are already exhausted in many cases. Thus the only possibility left is to reduce emissions at their source already by improvement of the power system.

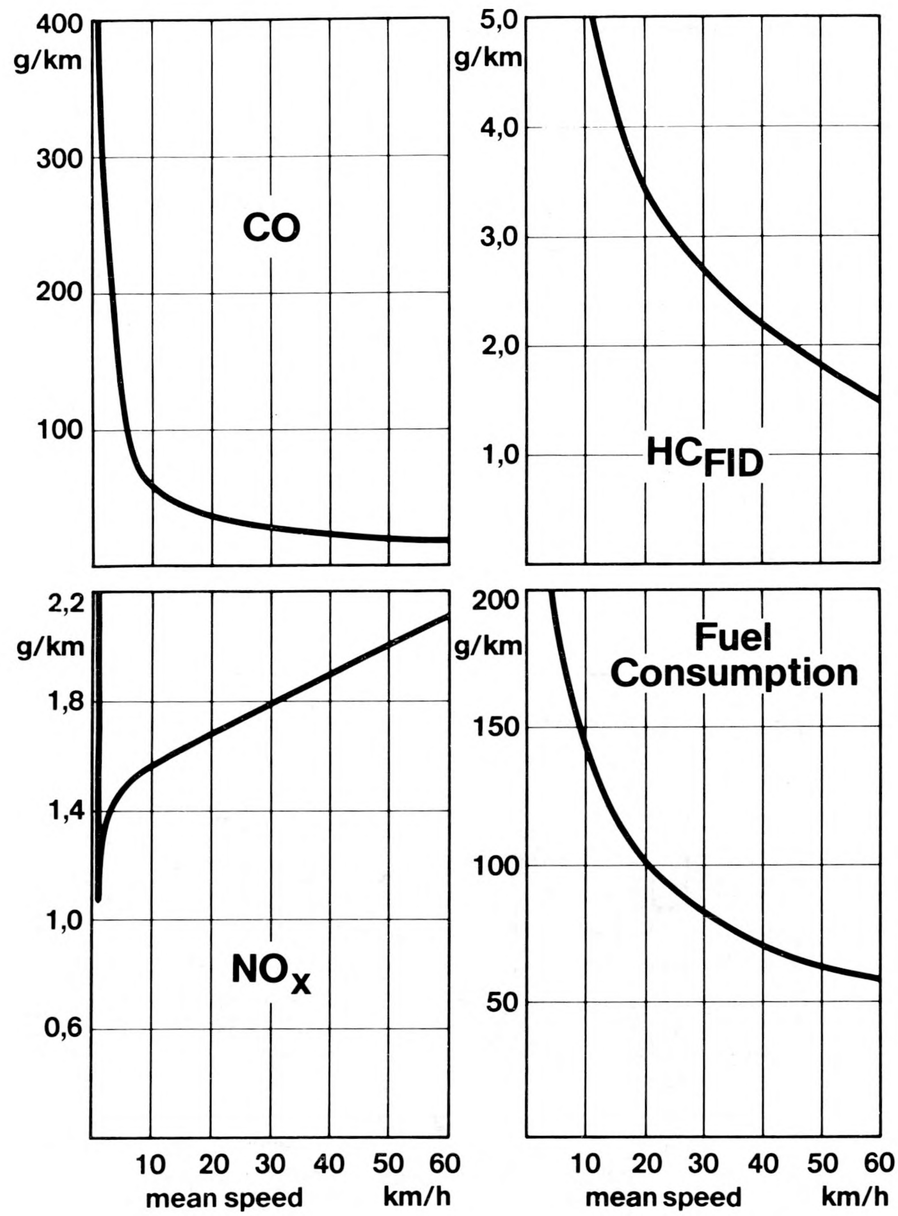
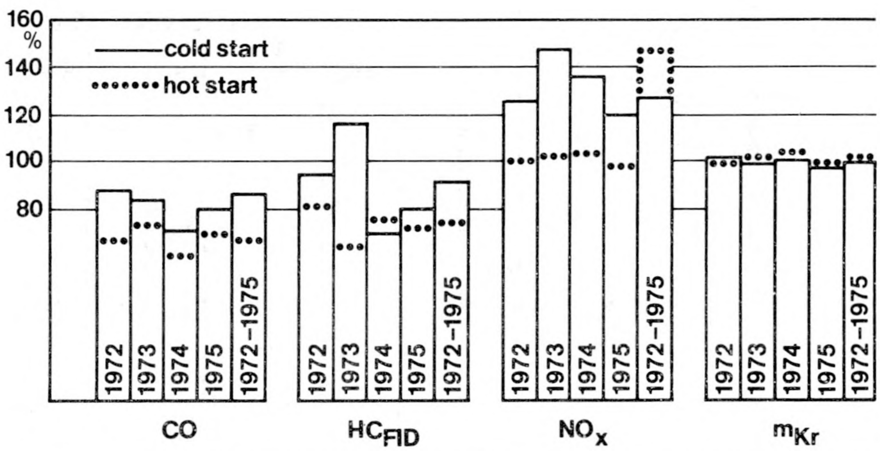


Fig. 2: Emissions and Fuel Consumption as a Function of Mean Speed (Based on Driving Modes of TÜV Rheinland)

Impact of Emission Regulation

The representative vehicle selection of the 1975 reference year which serves as a basis for these investigations includes such vehicles as are subject to the first stage of the European Emission Regulation (year of first registration 1972 - 1975) and such vehicles as were admitted to traffic prior to the effective date of said Emission Regulation (registrated first in 1970 and prior years). Thus it is possible to determine the effects of the emission control on emission behaviour by comparing these two vehicle categories. Fig. 3 shows such a comparison on the basis of European Test Cycle results obtained on a total of 43 vehicles registrated first between 1972 and 1975.



year	number of cars	mean displacement [cm³]	mean inertia weight [kg]	mean odometer reading [km]
1972	13	1430	1069	65.500
1973	11	1562	1106	48.500
1974	11	1524	1097	39.300
1975	8	1376	992	18.300
1972-1975	43	1478	1071	45.600

Fig. 3: Emissions of In-use Cars in the European Test Cycle as a Function of Registration Year (Registration Year 1970 and earlier ≙ 100 %)

The results for the 1970 and older vehicles are taken as 100 %. The various columns conform to test results according to ECE R 15 which means that in compliance with the requirements of this regulation the vehicles had been exposed to an ambient temperature of 20 - 30 ^{o} C for 6 hours (cold starting). The dotted columns represent test results where this vehicle conditioning had been omitted (hot starting). The table in Fig. 3 moreover indicates the mean displacement, the mean inertia weight and the mean odometer reading for the respektive sample.

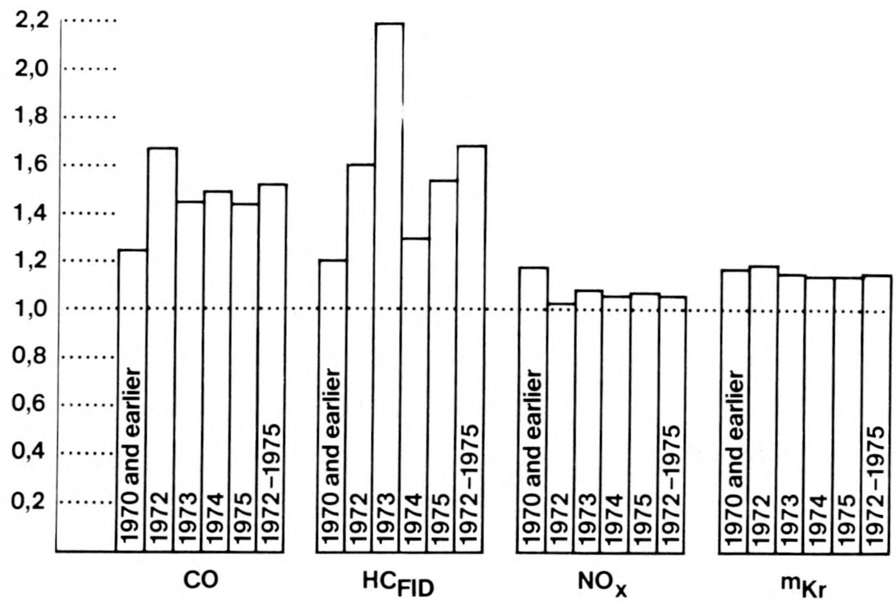
It has been found that a statistically safe influence of driven distance on the emission behaviour and the fuel consumption of the vehicles is not recognizable either for the cold or the hot test.

The mean values of CO emissions for the 1972 - 1975 year of first registration category referred to the average of 1970 and earlier registration years reduce totally by about 14 % (cold test) respectively 33 % (hot test). Comparison with the HC emissions determined by a FID shows a reduction of 9 % for the cold started European Test and 26 % for the hot test. In contrast to CO and HC emissions, the NO_x emissions from vehicles subject to the first stage of the Emission Regulation increase by 26 % (cold test) respectively 46 % (hot test) while fuel consumption remains substantially unchanged.

This implies that from a quality point of view the emission regulation has influenced the real emission behaviour of motor vehicles as expected. The results clearly reflect that the reduction of CO and HC emissions as involved in the cold started European Test also is primarily attributable to an improvement of the emission behaviour of working-temperature engines.

- 10 -

The differences in emission behaviour of motor vehicles subjected to cold started European Tests and of those tested under hot starting conditions can be better judged when the test results for individual registration years are directly correlated (Fig. 4). For vehicles registrated first in or before 1970 there is a ratio of 1.24 obtained for carbon monoxide while the corresponding ratio for vehicles admitted



year	number of cars	mean displacement [cm³]	mean inertia weight [kg]	mean odometer reading [km]
1970 and earlier	29	1340	991	95.800
1972	13	1430	1069	65.500
1973	11	1562	1106	48.500
1974	11	1524	1097	39.300
1975	8	1376	992	18.300
1972–1975	43	1478	1071	45.600

Fig. 4: Ratio of Emissions and Fuel Consumption in the European Test Cycle-Hot Started Versus Warm Started for Several Registration Years

to traffic after coming into force of the first stage of the European Emission Regulation is 1. 51. The ratios are similar for unburnt hydrocarbons while the impact of the engine's warm-up time on nitrogen oxides is considerably less important. Comparison of the vehicle categories shows that the influence of the warm-up phase is smaller in case of the older vehicles.

Irrespective of the year of first registration the fuel consumption of the vehicles subjected to a cold started European Test is about 14 % higher than the consumption determined during hot start tests.

Fig. 5 shows a comparison of exhaust emissions and fuel consumption as determined on the basis of the earlier described driving modes for the same vehicle categories (models 1972 - 1975 and 1970 and earlier models) as a function of mean driving speed. This representation is based on measurements after engine tuning, in order to establish for both random tests a basis of comparison that precludes the influence of the different-quality maintenance state that cannot be exactly determined of individual vehicles. There ist clearly to be seen from the illustration

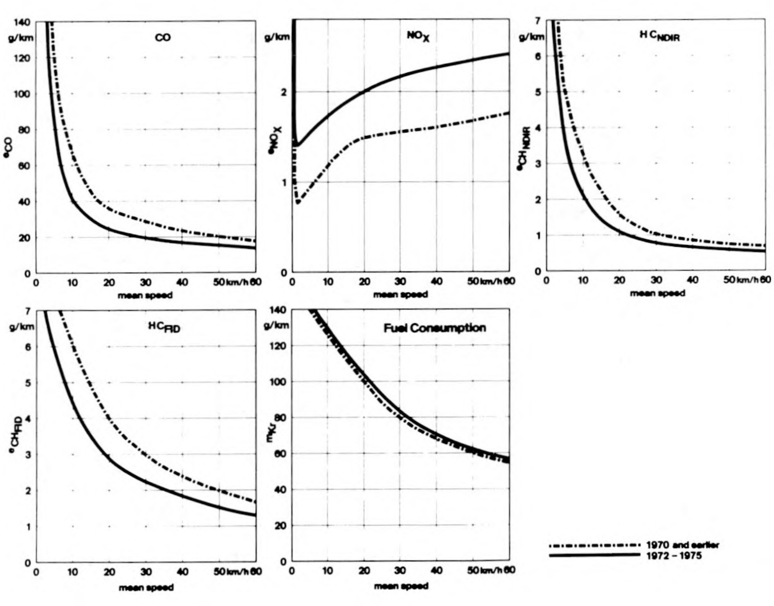


Fig. 5: Emissions and Fuel Consumption After Tuning as a Function of Mean Speed and Registration Year

that the CO and HC emissions of recent-year vehicles have reduced throughout the entire speed range under consideration as compared with those of older models. In line with what has been expected the nitrogen oxide emissions from recent-year vehicles are higher due to the "leaner" tuning of their engines. Fuel consumption, too, has slightly increased even though the reduction in CO und HC indicate that an improvement of efficiency may be expected. As the measured absolute consumptions reveal is this improvement of the specific consumption more than outweighed by the influence of the inertia weight, which is increased about 80 kg on an average.

In order to assess compliance or noncompliance with the emission standards called for under the European Emission Regulation, the number of vehicles that satisfy and that fail to satisfy the requirements applicable to the individual vehicles in respect of these standards are determined for the various vehicle age categories.

While the values established for 30 % already of the 1970 and older vehicles are lower than the stardards for production vehicles although they are not subject to these standards, only 50 % of the vehicles whose exhaust emissions are subject to statutory restrictions are within the respective limit values.

Irrespective of whether the findings made within the scope of the tests and investigations so far materialized can be considered representative of the total fleet of vehicles or not (the study is not yet complete), the emission situation cannot be judged solely on the basis of the specified percentage of vehicles of an age category that have satisfactorily passed the European Test Procedure. This is the reason why against the various year categories shown in Fig. 3 the allowable emissions as per the emission standards stipulated in the Emission Regulation for the various vehicles have been additionally cummed up. The 1977 planned standards for the European Community were taken as a basis for the NO_x component. The total value for a given toxic component as obtained for a defined vehicle random test characterizes the exhaust emission that all the vehicles together are allowed to emit as maximum in case of compliance with the ECE R 15 standards for production vehicles. This total is compared with the sum of actual emissions of the respective vehicle random test.

Table 2 indicates that the allowable total emission is substantially met for the CO component, the excess being only 1.6 %. HC and NO_x emissions for the random test are by far less than the allowable values. The total of unburnt hydrocarbons real emissions is by 52 % and the total of nitrogen oxide emissions by approximately 49 % lower than the allowable sum of standards as determined for subject selection of vehicles.

	Σ CO [g]	ΣHC_{NDIR} [g]	ΣNO_x [g]
allowable emissions in compliance with emission standards	**6626**	**511,8**	**586,8**
real emissions	**6736**	**247,5**	**301,2**
deviation	**+1,6%**	**-52%**	**-48,7%**

Table 2: Allowable Emissions in Accordance with ECE R 15 Standards Versus Real Emissions for 43 Cars (Year of First Registration: 1972 - 1975)

Influence of Maintenance State

In the course of investigations conducted to determine the emission behaviour of in-use vehicles the question arises as to what extent the vehicles' emission behaviour is influenced by the engine's maintenance state. This question is of particular importance in the European area because the legislator is not calling for any durability test (i. e. a test similar to the 50. 000 miles test in the USA) to be performed for approval of a vehicle type. Testing of the 43 vehicles put into circulation during the period from 1972 to 1975, for emission-relevant settings brought the results summarized in Table 3.

		number	mean deviation
ignition point	too early	14	4,2 °KW
	too late	15	2,3 °KW
	no objection	14	--
dwell angle	no objection	23	--
idle CO in accordance to manufacturers' prescriptions	greater	18	2,3 Vol.%
	less and equal	25	2,1 Vol.%

Table 3: Adjustment of 43 Cars as Received
(Year of First Registration: 1972 - 1975)

Fig. 6 shows a comparison of mean emissions as determined on the basis of the European Test Cycle for working-temperature engines of in-use motor vehicles in their as-received state and after engine tuning for 3 vehicle random tests of which two are subject to the first stage of the Emission Regulation and differ merely by their displacement range and of which one comprises only such vehicles that are not yet subject to said regulation. The mean CO emissions of the older vehicles in maintenance state are substantially equal to those of recent-year vehicles in their state as received. This implies that

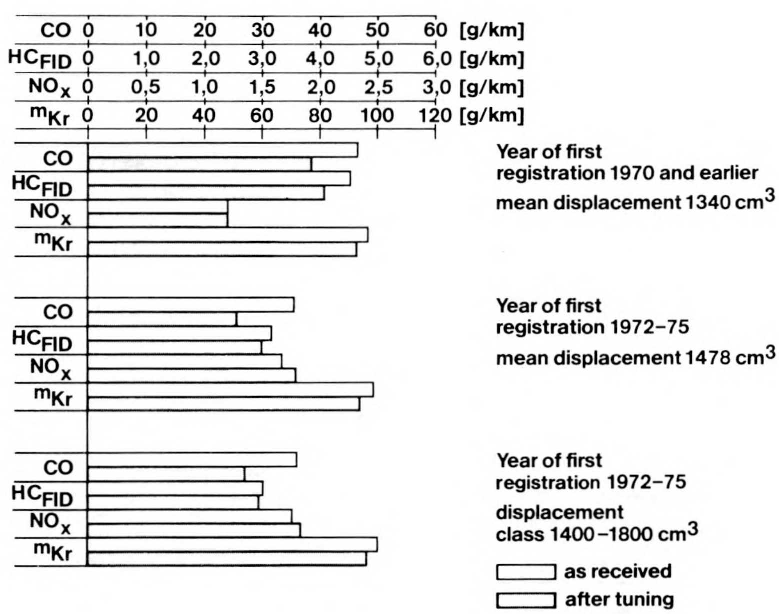


Fig. 6: Emissions of 84 In-use Cars for Several Registration Years (Based on the European Test Cycle, hot started)

careful adjustment and maintenance of ignition system and carburettor results in a reduction of CO and HC emissions for all three categories. The impact on NO_x emissions is a minor one. Engine maintenance and tuning results in a reduction of CO emissions by about 27 % for 1972 - 1975 vehicles and of 12 % only for the vehicles older than those. The reduction in HC emissions as measured by means of a FID is comparatively high for the older vehicles (11 %) while a slight reduction of said HC emissions only could be observed for the 1972-1975 vehicles. NO_x emissions on an average are slightly higher (up to 8 %) for the 1972-1975 vehicles, but no change is observed after maintenance and setting in case of the older vehicles. Fig. 7 supplementarily shows the impact of tuning and

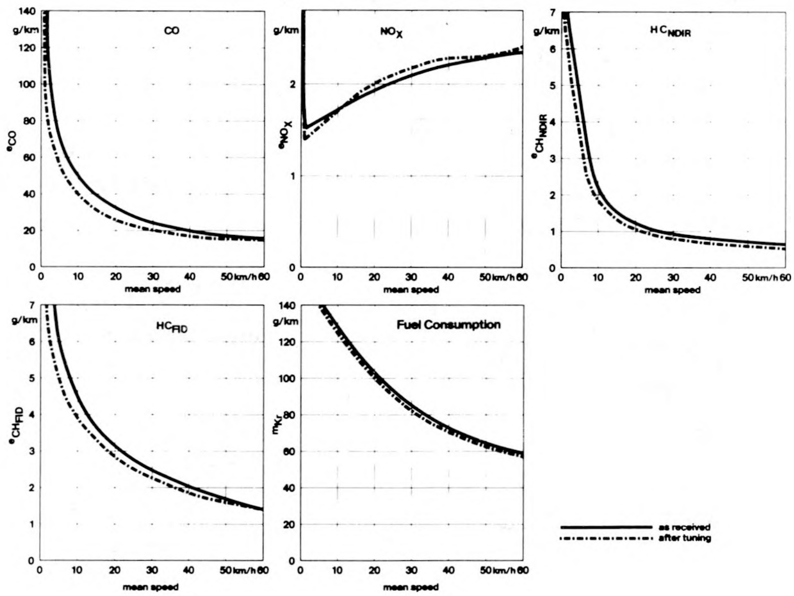


Fig. 7: Effect of Tuning on Emissions
and Fuel Consumption
(Year of First Registration: 1972 - 1975)

maintenance of carburettor and ignition system for the entire mean speed range involved in urban traffic. There is a reduction throughout the range for CO and HC emissions while NO_x emissions in the mean speed range show a slight increase after engine maintenance and tuning as compared with the as-received state. A reduction in NO_x emissions is observed as against the as-received state in the low speed range.

Influence of Vehicle Mass

For predominantly nonstationary performance and mean driving speeds such as involved in urban traffic the vehicle mass is an important parameter that influences fuel consumption and emission behaviour. While the dependence of emissions and fuel consumption on such drive efficiency-influencing engine-specific characteristics as displacement, compression ratio, stroke to bore ratio, combustion chamber geometry etc. cannot be determined separately from other influencing factors derived from the results of tests performed on a selection of vehicles, the impact of vehicle mass on emissions and fuel consumption can be defined with absolute distinctness. Fig. 8 shows the mean CO emission, the sum of NDIR measured HC and NO_x emissions and the fuel consumption as a function of inertia weight, with the dotted lines representing the current standards for production vehicles. The straight lines are the result of a regression analysis based on 84 vehicles, part of which are not subject to the first stage of the European Emission Regulation. It can be noticed that the sum of HC and NO_x emissions remains significantly under the sum of the corresponding standards for production vehicles in the whole range. The carbon monoxide standard for production vehicles is exceeded by more than 30 % on an average. The different dependence of emissions and fuel consumption on the vehicle mass for instance reflects the character of the European Driving Cycle on which these present investigations are based, in the engine characteristic field. Doubling up the inertia weight would for instance result in an increase of CO emissions by about 30 % and of the sum of unburnt hydrocarbon and nitrogen oxide emissions by 25 %. The fuel consumption rises approximately by 50 %.

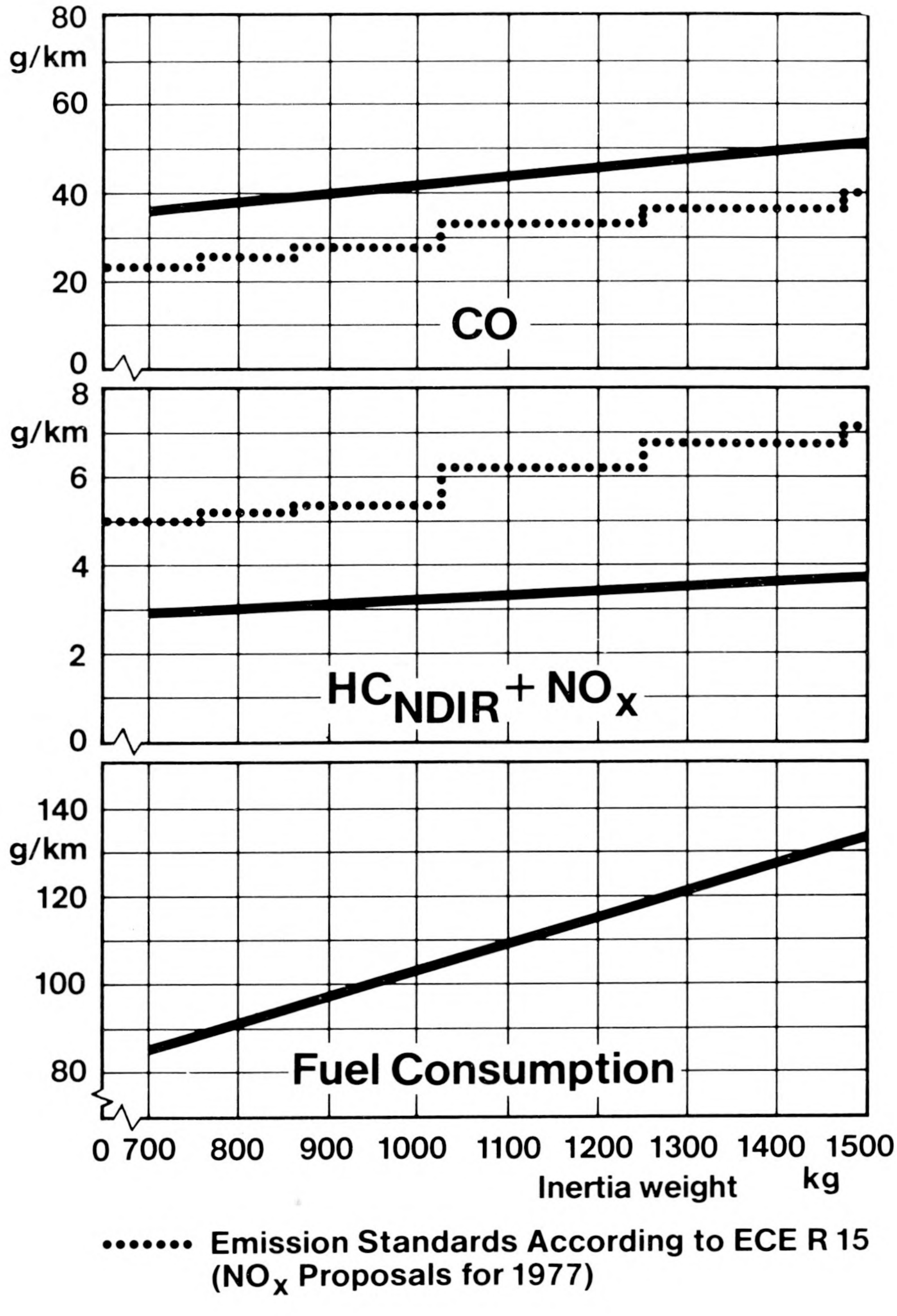


Fig. 8: Emissions and Fuel Consumption as a Function of Inertia Weight (Based on the European Test Cycle, cold started)

After all it can be stated that the graduation of the emission standards in depencence of inertia weight prescribed by the legislator are verified by the gradient of the regression lines.

Conclusion

Exhaust emission factors are an important requirement to investigate the emissions of the road traffic. The systematic to establish emission factors has been described in detail. It could be demonstrated that emission factors represent a good basis to judge the impact of Emission Regulation and technological development on exhaust behaviour of a specific vehicle fleet.

As expected the first stage of the German Emission Regulation has led to a significant decrease of emissions for the components CO and HC in the whole range of mean driving speed. These emission reductions caused by lean air-fuel ratios result naturally in a distinct rise of NO_x emissions. The increased fuel consumption of vehicles registrated first between 1972 and 1975 are due to the higher vehicle mass. This influence could not be compensated by the leaner setting of air-fuel ratio, which generally leads to higher efficiency.

The impact of the maintenance state of carburettor and ignition system has been investigated by measuring the vehicles as received and after tuning in accordance to manufacturer's prescription. This comparison shows that improvements of emission behaviour and fuel consumption can be reached by correct tuning. The quantitative assessment of the influence of maintenance determined on the basis of this random test cannot be generalized since there are no statistical data characterizing the maintenance state of the vehicle fleet of the Federal Republic of Germany.

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Session 13

OPTIMIZING THE ENGINE-TRANSMISSION SYSTEM BY MEANS OF AN ELECTRONICALLY-CONTROLLED GEARBOX

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ABSTRACT

The scope of this research is the study of an experimental engine-transmission system control affording minimum fuel consumption without jeopardizing the desired vehicle performance level. As the I.C. engine can furnish the same output at different RPM rates and throttle positions, when maximum performance is not a requirement the engine can be so controlled as to supply the same acceleration at a given RPM through different gear ratios of which only one gives maximum fuel economy.

Efforts have been made in an attempt to solve the problem using an experimental automatic stepped transmission incorporating special controls.

1. INTRODUCTION

The target set for this research is the assessment of the extent to which it is possible to save fuel in driving an automobile over a given route by operating exclusively on the engine utilization mode without in any way altering engine design characteristics. With a spark-ignited internal combustion engine (such as the unit considered here) the specific fuel consumption varies within wide limits, depending on operation conditions. Upon observing the fuel consumption isometric chart for an I.C. engine (Fig.1) it becomes readily apparent why it is possible to obtain from the engine the same power at differing consumptions, depending on RPM rates.

Given a correct use of the transmission, it would theoretically be possible to optimize consumption rates but under the very conditions of traffic (urban, suburban) in which said correct usage would have importance the driver has to concentrate his attention on exterior events and, therefore, cannot deal with this additional problem.

Because of this and other reasons an attempt was made to obtain the set target through an experimental automated transmission designed to fit a small European car whose reference weight is 2000 lbs.

2. BEHAVIORAL ANALYSIS OF CAR WITH DIFFERENT TYPES OF TRANSMISSION

The Federal Driving Cycle and Highway Driving Cycle speed pattern are considered in this research as typical examples of car utilization in urban and suburban traffic.

Quoted in the reasoning that follows are the results from a mathematical model, by which it is possible to compute the fuel consumption of a vehicle over a route plotted by point allocations, once the engine consumption isometric chart and vehicle specifications are known.

With this model, the instantaneous power required by the operation cycle at a given time is paired up with a consumption value obtained by interpolating between the actual values recorded under steady state conditions on the dynamometer.

Hence, this procedure neglects the effects on consumption of fuel air mixture ratio variations occurring in both the transient and warm-up stages.

By accepting this approximation the resulting consumption figures are always underestimated by about 10 - 20%, depending on the case in question, but experience proves they are nevertheless useful for comparative and trend detection purposes.

Simulation was run using the transmission according to the standards normally adopted for dynamometer cycles.

Referring to the car considered here, Fig. 2 shows the histogram of the power demanded of the engine during the Federal Driving Cycle; sampling frequency is 1 Hz.

Immediately apparent is that the prevailing power requirement levels are very low for this car, this justifying the statement that during operation the engine works in high specific consumption areas and at power levels at which marked consumption variations occur depending on engine RPM rates.

Within the limits of the proposed scope it is clear that the least consumption figure, for a given car and assigned cycle, may be obtained operating the engine along the minimum specific consumption line plotted in Fig. 1.

The mathematical model was modified to accommodate the simulation of the ideal case represented by a stepless transmission, affording the same

efficiency as that of a conventional unit and so controlled as to instantly vary its ratio to adequately match the minimum consumption speed rate.
In such ideal hypothesis, this transmission was assumed to provide gear ratios of infinite range.
Table 1 compares the fuel consumption ratings of the considered car, as originally equipped and when fitted with the ideal stepless transmission.
Depending on the test cycle, fuel savings vary in the 25 to 28% range.
We now turn to the histogram plotted for the drive ratios used by the ideal stepless transmission in urban operation (Fig. 3).
Also plotted here is the drive ratio range covered by current manual transmissions. The fuel consumption variation between stepless and conventional gearboxes is not due only to the possibility of choosing the speed ratios used, but also to the availability of lower ratios than presently used minimums. Clearly, this still does not provide adequate information on the importance of the two aspects of the problem.
The reasonings outlined for the F.D.C. apply equally well to the H.D.C., and with similar results.
However, they will not be detailed here for the sake of brevity. Judging from the results obtained so far, the stepless transmission is no doubt promising but its idealization does not take two facts into account:
- In any case, the range of a stepless transmission cannot be infinite.
- The efficiency of presently known stepless transmissions is lower than with conventional transmissions.

It was therefore decided to check the influence of reasonable limitations to the features idealized for this transmission by extending the investigation to a test 5 - speed unit automated as described earlier.
The approach for the choice of speed ratios for this transmission was to

provide the car with the same performance obtainable with the conventional unit. In practice, this meant the adoption of the same overall drive ratio in 1st gear (and, hence, the same maximum gradability) and of 1.5 as the maximum interval between ratios, constant for all gears. Shown in Table 2 is a comparison between the drive ratios chosen and the resulting performance obtained for the test car with conventional and automated transmission. Plotted in Fig. 3 is the ratio variation range covered by the experimental transmission.

Table 3 contains the simulation results of a small European car equipped with conventional (C.), stepless ideal (S.I.) and stepped automated (S.A.) transmissions.

The downgrading of consumption figures from the S.I. to the S.A. transmission is in no case very marked: 5% on F.D.C. and 7% on H.D.C. This leads us to consider, on one hand, that the very low ratios required by the S.I. transmission in the F.D.C. are utilized at consumption rates having little influence, percentagewise, on the cycle; on the other hand, for the same cycle, that the gear ratio choice criterion is of great importance.

This is not true for the H.D.C. where, because of infrequent accelerations, the lower ratios may be used to the best advantage. Here, the speed choice criterion is highly intuitive and, therefore, could even be done manually.

3. THE STEPPED AUTOMATED TRANSMISSION

In view of the results obtained with the simulation the scope of the research was extended to include a stepped automated, 5 - speed transmission having such characteristics as to permit transversal installation in a small FWD car.

Preliminary basic features for this experimental transmission are:

- High mechanical efficiency.
- Quick-shift possibilities.
- Additional devices permitting smooth synchronization of engine-drive units during gearshifts.

Let us now examine the control system requirements considered to be essential in ensuring the basic features sought.

Through the transmission, the engine is presently called upon to perform two functions:

- To provide, at road wheels, the power needed to move the vehicle in the conditions set by the driver.
- To provide, at road wheels the braking torque needed in decelerations or on downgrades, without frequent application of the brakes.

The strategy envisaged to give the least possible consumption meets the first function requirement as it does not impose any limitation on engine power output but acts merely on the engine operation conditions at which such power is supplied.

This same strategy accounts for the fact that the lower powers are supplied, practically, at minimum speed rate of engine smooth operation, namely, at those RPM at which the engine braking effect with closed throttle is minimal.

Therefore, the second function cannot be met when considering the case of negative power in the same perspective of the previously defined strategy. It is, in other words, necessary to introduce a discontinuity signal which will cause transmission to operate in a conventional way whenever the driver intends to slow down.

Moreover, with the type of operation imposed by the chosen strategy, it is possible to have a very high number of gearshifts: with the F.D.C., for in-

stance, the total number of gear changes for this car extends from 101 with the conventional transmission to 244 with the automated transmission.
For this reason, it becomes essential to extend the utilization ranges of the different gear ratios so that they will overlap thus preventing any instability.
Simulation was repeated, due account of this fact being taken, and the results obtained are shown in Table 3.

4. DESIGN CRITERIA

A survey conducted over a number of alternative solutions capable of overcoming the constraints described earlier has led us to believe that for the specific requirements in point the best possible compromise in terms of effectiveness and functionality can be obtained with a 5 - speed transmission and breakaway centrifugal clutch or a 4 - speed transmission and lock-up torque converter. The consumption figures resulting from the simulation are practically identical for both layouts provided the four speed ratios of the unit with converter coincide with the last four speed ratios of the unit with centrifugal clutch.
For the purpose of experimenting both concepts, the design was focused on the 5 - and the 4 - speed versions without introducing any major modification.
In order to permit installation on cars with power plant either longitudinal or transversal, also overall compactness (lengthwise, in particular) was set as a design requisite for the transmission package.
The twin-countershaft configuration with gear engagement by multidisc clutches was pinpointed as the type of transmission best meeting the set targets whereas the epicyclic arrangement was discarded as being too complex for 5 gear

ratios.

5. TRANSMISSION LAYOUT

Fig. 4 shows the transmission layout for the 5 - speed version with incorporated differential for installation on transverse engined cars. Through two pairs of spur gears the drive passes from the input shaft S_i to the two countershaft S_1 and S_2.

The latter shafts carry the 2nd - 3rd - 4th - 5th driving gears G_2 - G_3 - G_4 - G_5, respectively, which, in turn, mesh with gears G_6 and G_7 carried by the output shaft.

Cut in the end of the output shaft is the spur pinion transmitting the final drive to the differential and the road wheels.

The freely rotating gears G_2 - G_3 - G_4 - G_5 may be made integral with their countershaft by the locking action of the respective multi-disc clutches C_2 - C_3 - C_4 - C_5 to provide the 2nd - 3rd - 4th and 5th speeds.

The 1st speed is instead a freewheeling gear G_1 which is engaged by shifting sleeve SS (shown here in neutral position) to the left; when this same sleeve is shifted to the right, reverse is engaged.

The drive coupling between transmission and engine is provided by a break-away centrifugal-weights clutch characterized by the fact that, with engine at full power, in the clutch "applied" stage the driven plate slips up to 2000 - 2200 RPM whereas in the clutch "released" stage it begins to slip at 1000 RPM. This arrangement ensures smooth and progressive car break-aways while allowing the engine to operate at very low speed rates (up to around 1000 RPM) in conformity with the requirements set by the least consumption strategy.

The 4 -speed version is derived from the basic 5 - speed unit by simply eliminating the 1st gear pair, and relevant freewheel, and replacing the centrifugal clutch with a torque converter provided with lock-up. Quite evidently, the reason for the latter is to eliminate, above a certain car speed, the hydraulic losses resulting from the "slippage" of converter rotating elements.

The times and ways in which gearshifts must take place are governed by a computer (described below) operating on stored information memorized from engine test determinations and from periodical signal recordings of accelerator pedal position and vehicle road speed fed by sensors.

Also, gearshifts are actuated by multi-disc clutches C_1, C_2, C_3, C_4, each operated by a power cylinder through electrovalve E_1, E_2, E_3, E_4 as shown in the hydraulic circuit schematics of Fig. 5. Electrovalve E_B operates the transmission brake which provides upshifts synchronisation.

Based on the above, a gearbox was designed and built for research purposes. This 5 - speed unit is entirely automated, has a 5.38 to 1 range for input torque up to 10 Kgm and is intended for a FWD, transverse-engined car, as shown in Fig. 6.

In this transmission gearshifting occurs through an interruption of power. Engine and related parts synchronisation is obtained as follows:

- <u>upshifts</u>
- by bringing the throttle to idle position and simultaneously braking transmission input shaft;
- <u>downshifts</u>
- by suitably cracking the throttle.

6. THE CONTROL SYSTEM

This control consists of :

1) sensors

2) actuators

3) computer

The system diagram is shown in Fig. 7.

Sensors

The following signals: engine RPM, transmission input shaft RPM (past the centrifugal clutch), and car speed, are collected by sensors consisting of toothed wheels and electromagnetic pickups.

A potentiometer, on the accelerator pedal senses pedal position while another potentiometer, on the throttle spindle senses the throttle opening angle. The driver can overrun the operation logic through switches installed on the gearshift lever and on the brake, respectively. Furthermore, a panel can be used to manually select the desired gear ratio.

Actuators

The means used at present to control throttle position is a small electric motor provided with suitable gear reduction and coupled to the throttle spindle.

The engagement of the different gears and of the transmission brake is controlled by electrovalves contained in the hydraulic control circuit.

Computer

The computer is connected to sensors and actuators through one input and one output interface. The signals fed from the car are, practically, of two types: voltages proportional to throttle opening angle or to power demand (accelerator pedal); ON/OFF signals for the other sensors.

In the input interface all these signals are processed and converted into digital signals. In this way, RPM, angular positions or "states" are thus associated to "words" of 8 bits.

The output interface decodes the signals from the computer rendering them suitable for controlling the electrovalves (for the engagement of the different transmission gears, the synchronisation brake and the throttle positioning motor).

7. SYSTEM LOGIC

The electronic control unit is designed to perform three main functions:

a. Selection and control of throttle opening

b. Selection of the gear ratio to set up

c. Transmission synchronisation during transmission gear ratio changes.

Throttle Control

As said earlier, on the car in question the accelerator pedal is no longer mechanically linked to the throttle which, instead, is controlled through a much ampler loop including the computer.

The accelerator function itself has been modified inasmuch as a significant magnitude of the power demand by the driver has been associated with it. Based on the family of curves identifying the same throttle opening angle obtained from the engine isometric chart (Fig. 8), throttle angle ϕ_i , ideal for a given power, and RPM may be determined instant by instant. The above chart was divided into a number of parts and the values of throttle angle (corresponding to the corners of the rectangles into which the chart has been divided) have been stored in the computer. The determination of throttle angle ϕ_i relevant to power value P_i and engine RPM value ti_1 was obtained by causing the computer to perform a three dimension linear interpolation. The data obtained hence represents the throttle opening degree required for every running condition. Throttle positioning, performed by a small D.C., ON/OFF controlled motor, is monitored through entirely digital techniques.

Fig. 9 shows how the computer is used to monitor throttle position. Required and actual throttle opening examined at a constant rate, are summed algebraically to form error signal "e". Based on such signal, and on those of the preceding cycles, a new error signal "^{e}c" is built, made up of the algebraic sums of the input error functions which have, as "weights", coefficients expressing proportionally derived and integrative actions. The result of this processing is compared with a threshold called the "dead" or instability area; if the value is outside this area the micro-computer sends out a drive order to the motor.

The "dead" area and the proportional, integrative and derived networks are determined in the course of the study on the stability of the system under examination.

Selection of Gear Ratio

The engine RPM and the power demanded by the driver with a given degree of accelerator pedal depression, identify a given engine operation point. The chart of Fig. 1 reveals that three conditions may occur:

a) the point in question is within the area which represents the minimum consumption band; the gear engaged is the right one.

b) the point is to the left of the area; shifting to a higher ratio gear is required in order to obtain necessary power.

c) the point is to the right of the area; shifting to a lower ratio gear is required in order to move into the optimum band.

Boundary curves of the area obtained theoretically on the basis of the minimum consumption logic, are stored point by point in the micro-computer data memory. To obtain good accuracy of the output data processing is performed through a linear interpolation.

Synchronization for upshifts

When the above processing result calls for gearshifting, the computer memorizes the engine RPM and car speed data and calculates the engine RPM required after shifting.

The gear engaged at the time is then disengaged, a command is sent to engine brake to reduce gear shifting time to a minimum, and the throttle opening angle is set for the idle position.

During the time elapsing between these operations and gearshifting the processor modifies, if necessary, the values which will be obtained for synchronization in the case of speed variations.

When the optimum synchronization point is neared, the engine brake is released, a command is sent out for gearshifting, and throttle opening is set again at the angle established as shown in Fig. 8.

Synchronization for downshifts

Through a procedure similar to the one described above the actual gear is disengaged and a command is sent to the throttle so that it moves to an angle whose value, depending on engine RPM rate, has been determined experimentally.

With some advance with respect to the optimum synchronization value, the throttle opening angle is set back to the value dictated by Fig. 8. Subsequently, the command for gear engagement is sent out.

8. EXPERIMENTAL RESULTS

A car equipped with the experimental gears was installed on a dynamometer bench for experimental evaluation.

It was found possible to effect a saving in consumption of 17% on FDC, but the tests have not yet been completed.

Beyond the uncertainty of the consumption results there remain strong doubts as to the emission level and as to the drivability, but it is hoped that more complete data will be available for oral discussion.

It is, nevertheless, observed that the car used complies with European, but not the Federal, emission regulations.

Table 1. Simulation results of a small European car equipped with conventional transmission (C) and stepless ideal transmission (I.S.) on F.D.C. and H.D.C.

Transmission Type	F.D.C.		H.D.C.	
	l/100km	mpg	l/100km	mpg
C.	7.53	31.2	5.94	39.6
I.S.	5.41 (-28%	43.5 (+ 39%)	4.47 (-25%)	52.6 (+ 33%)

G.L. Falzoni, R. Guagliumi, L. Morello, "Optimizing the Engine-Transmission System by Means of an Electronically-Controlled Gearbox"

Table 3. Simulation results of a small European car equipped with conventional (C.), stepless ideal (S.I.) and stepped automated (S.A.) transmissions on F.D.C. and H.D.C.

Transmission Type	F.D.C.		H.D.C.	
	1/100Km	mpg	1/100Km	mpg
C.	7.53	31.2	5.94	39.6
S.I.	5.41 (-28%)	43.5 (+ 39%)	4.47 (-25%)	52.6 (+ 33%)
S.A. (1st Approximation)	5.76 (-23%)	40.8 (+ 31%)	4.87 (-18%)	48.3 (+ 22%)
S.A. (2nd Approximation)	5.78 (-23%)	40.7 (+ 30%)	4.91 (-17%)	47.9 (+ 21%)

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Table 2. Comparison between transmission ratios on performances obtained for conventional (C.) and stepped automated (S.A.) transmissions.

Transmission Type		C.	S.A.
Final transmission ratio		4.692	3.647
Gearbox ratios	1st	3.636	4.612
	2nd	2.055	2.914
	3rd	1.348	1.943
	4th	0.963	1.285
	5th	-	0.857
Top speed (Km/h)		135.9	135.0
Acceleration time to 100 Km/h (sec)		25.6	25.0

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CAPTIONS FOR ILLUSTRATIONS

Fig. 1 Isometric consumption chart; it is apparent why it is possible to obtain from the engine the same power at different consumptions, depending on RPM rates. Curve 1 is the minimum consumption line at the same power requirement; curve 2 is the right limit of the area where the motor is used by automated stepped transmission.

Fig. 2 Histogram of the power required during F.D.C.; sampling frequency is 1 Hz. Histogram does not include strong decelerations and stops.

Fig. 3 Histogram of ratios used by stepless ideal transmission during F.D.C.; sampling frequency is 1 Hz. Histogram does not include strong decelerations and stops.

Fig. 4 Automated 5 - speed transmission layout; input shaft S_i moves countershafts S_1 and S_2. 2nd, 3rd, 4th and 5th gears (G_2, G_3, G_4, G_5) can be engaged by multi-disc clutches (C_2, C_3, C_4, C_5); 1st gear G_1 is freewheeling. Shifting sleeve SS engages forward drive or reverse.

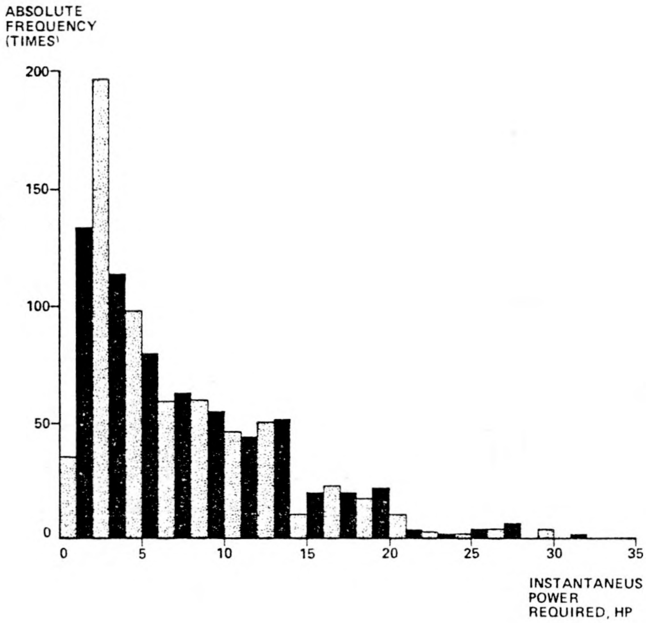
Fig. 5 Hydraulic circuit schematics; electrovalves E_1, E_2, E_3, E_4 operate multi-disc clutches C_1, C_2, C_3, C_4, electrovalve E_B operates transmission brake which provides upshifts-synchronization.

Fig. 6 Experimental 5 - speed automated transmission.

Fig. 7 Control system block-diagram.

Fig. 8 Isometric throttle opening chart; the chart is divided into a number of rectangles (A_i, B_i, C_i, D_i. The determination of correct throttle angle ϕ_i , relevant to power P_i and RPM n_i, is obtained by three-dimension linear interpolation. Vertices coordinates are stored in the data memory.

Fig. 9 Throttle position control loop.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(TIMES)
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0
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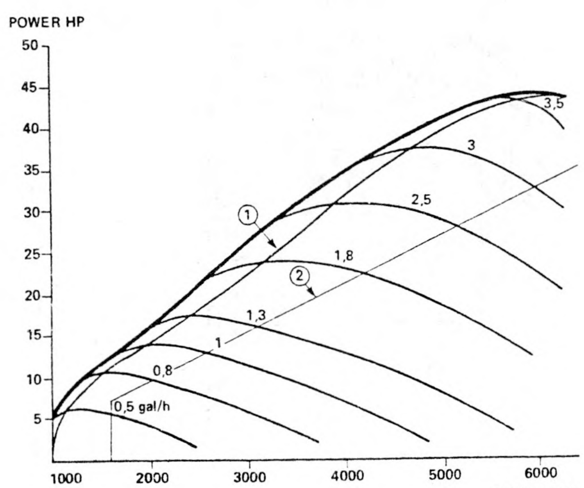
15
20
25
30
35
INSTANTANEUS
POWER
REQUIRED, HP

Fig 1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4000
5000
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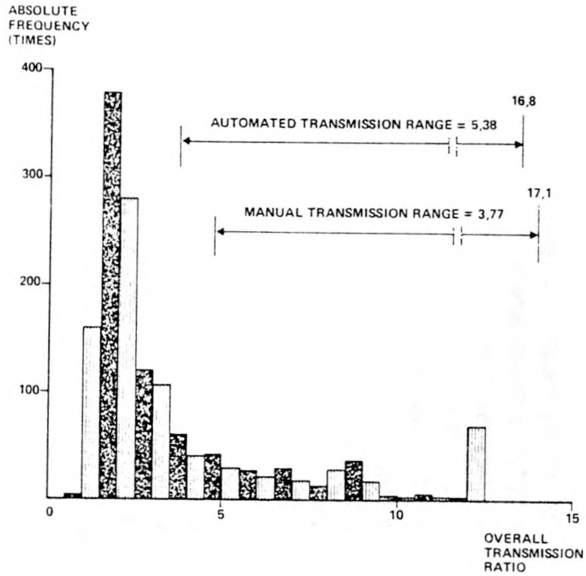
0,8
0,5 gal/h
1
2

Fig 2


ABSOLUTE
FREQUENCY
(TIMES)
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300
200
100
0
5
10
15
16,8
AUTOMATED TRANSMISSION RANGE = 5,38
17,1
MANUAL TRANSMISSION RANGE = 3,77
OVERALL
TRANSMISSION
RATIO

Fig 3

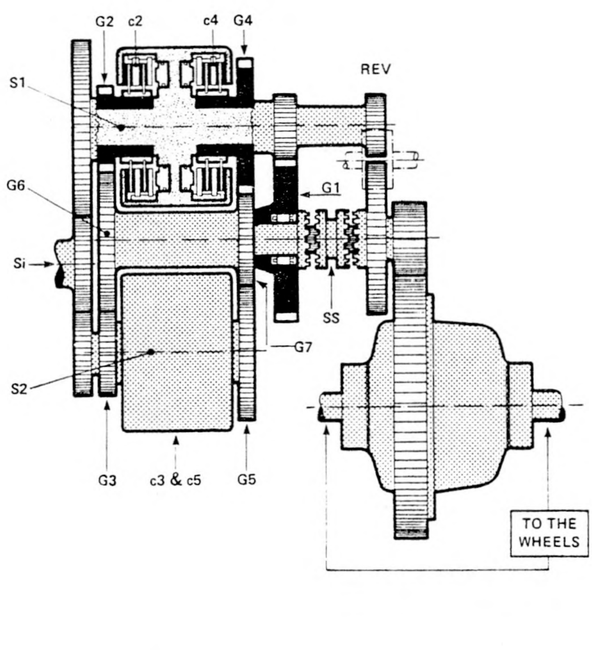
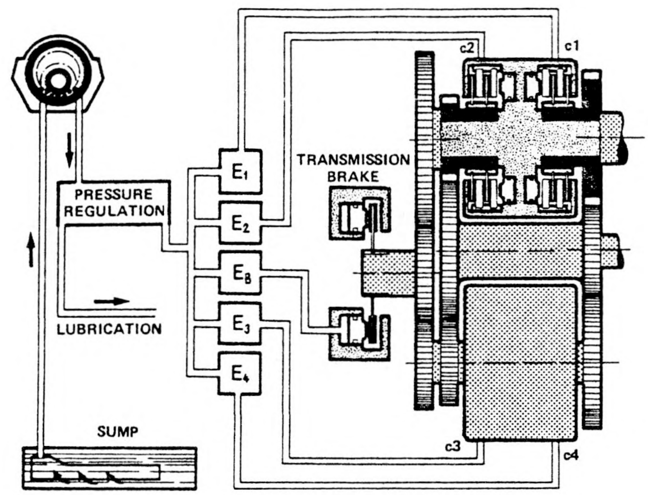

G2
c2
c4
G4
S1
REV
G6
G1
Si
SS
G7
S2
G3
c3 & c5
G5
TO THE
WHEELS

Fig 4


PRESSURE
REGULATION
LUBRICATION
SUMP
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BRAKE
E1
E2
EB
E3
E4
c1
c2
c3
c4

Fig 5

Fig 6

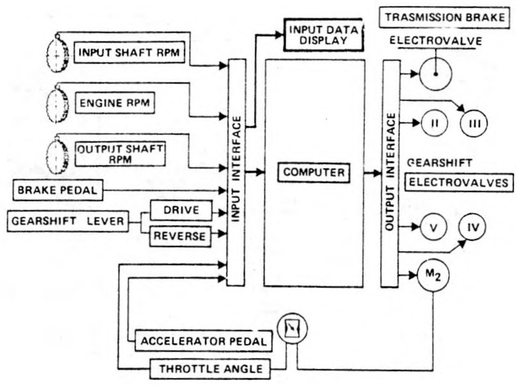

Fig 7

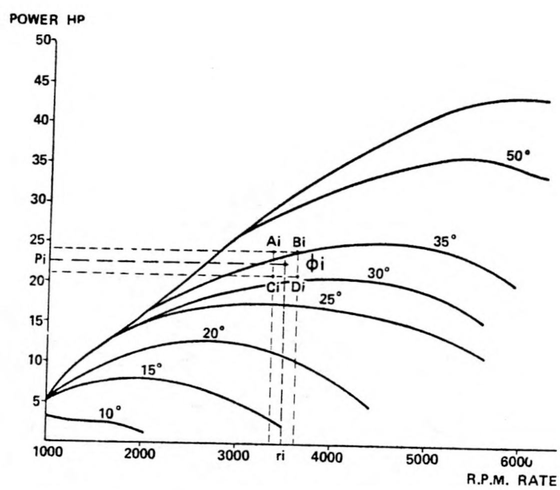

Fig 8

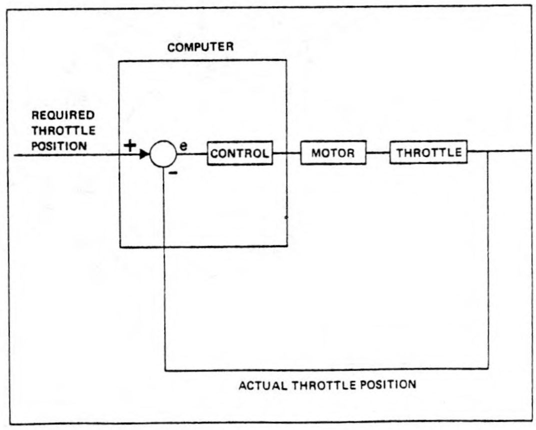

Fig 9

A Study to Validate Automobile Drivetrain Modifications that Improve Fuel Economy

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ABSTRACT

This study is being conducted primarily for the purpose of validating by hardware test, computer predictions of the EPA urban and highway fuel economy improvements that can be achieved through the use of wide-range three and four-speed automatic transmissions with and without torque converter lock-up. This study will also determine the impact of these transmissions on driveability, engine life, emissions, and performance.

Tests are being carred out on four different 1975 cars ranging in inertia weight from 2500 to 4000 pounds. Engine sizes in these cars range from 97 to 320 cubic inches.

The results to date indicate that significant improvements in fuel economy can be achieved with the wide-range transmissions provided that the torque converter is locked up. Emissions remain within the 1975 Federal Standards for which the test vehicles were designed. Fuel economy improvements are accompanied by significant decreases in nitrogen oxides emissions. The driveability and performance of these cars are not seriously degraded.

A Study to Validate Automobile Drivetrain Modifications that Improve Fuel Economy

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Introduction

The primary purpose of this study is to validate the potential of improved fuel economy by improving the matching of engine operation to road load*. The particular drivetrain modifications that were tested consisted of substituting a conventional automatic three-speed transmission with a wide-range three and four-speed automatic transmission with the capability of torque converter lock-up. Four representative 1975 compact and subcompact vehicles with engine sizes ranging from 320 to 97 CID were fitted with these modified transmissions.

The vehicles were tested to establish a baseline reference for fuel economy, exhaust emissions, driveability, and acceleration. The vehicles were again tested with the modified transmissions to establish the impact of the improved drivetrain. Test procedures include EPA dynamometer tests, driveability evaluation, and wide open throttle (WOT) acceleration test.

The results have been compared to computer simulation predictions of fuel economy.

The second volume is an illustrated history of the automatic transmission with particular emphasis on its development in the United States.

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*Another goal of this study, not discussed in this presentation, is to document the technical decision processes in drivetrain design, and to study the history of the automatic transmission in the United States. One volume of the two-volume report contains a description of the processes by which a drivetrain designer matches the engine characteristics to the desired vehicle performance requirements. The second volume is an illustrated history of the automatic transmission with particular emphasis on its development in the United States.

Vehicle and Transmission Description

The automatic multi-test transmission configurations with torque converter lock-up are modifications of production Chrysler Torque-flite units, and have characteristics shown in Table 1.

TABLE 1
DRIVETRAIN MODIFICATIONS

	Standard 3-Speed	Wide Range 3-Speed		Wide Range 4-Speed
Lock-up Converter	None	2,3 Or Not At All		2,3,4 Or Not At All
Gear Ratio			Optional 2nd Mode	
1st	2.45:1	2.45:1	2.45:1	2.45:1
2nd	1.45:1	1.45:1	1.13:1	1.45:1
3rd	1.00:1	.778:1	.778:1	1.00:1
4th	—	—	—	.778:1
Ratio Range Increase	1.00	1.28		1.28

Torque converter performance equals that of a production torque converter in all cases. The optional second mode was not tested, nor was lock-up in second gear of the four-speed configuration. Four identical transmissions, illustrated in Figures 1 and 2, were built by B&M Automotive Products, Chatsworth, California. Each of these units can perform as any one of the desired transmission configurations in Table 1. A change from one to the other can be accomplished from

TRANSMISSION MODIFICATION
TO STANDARD AUTOMATIC

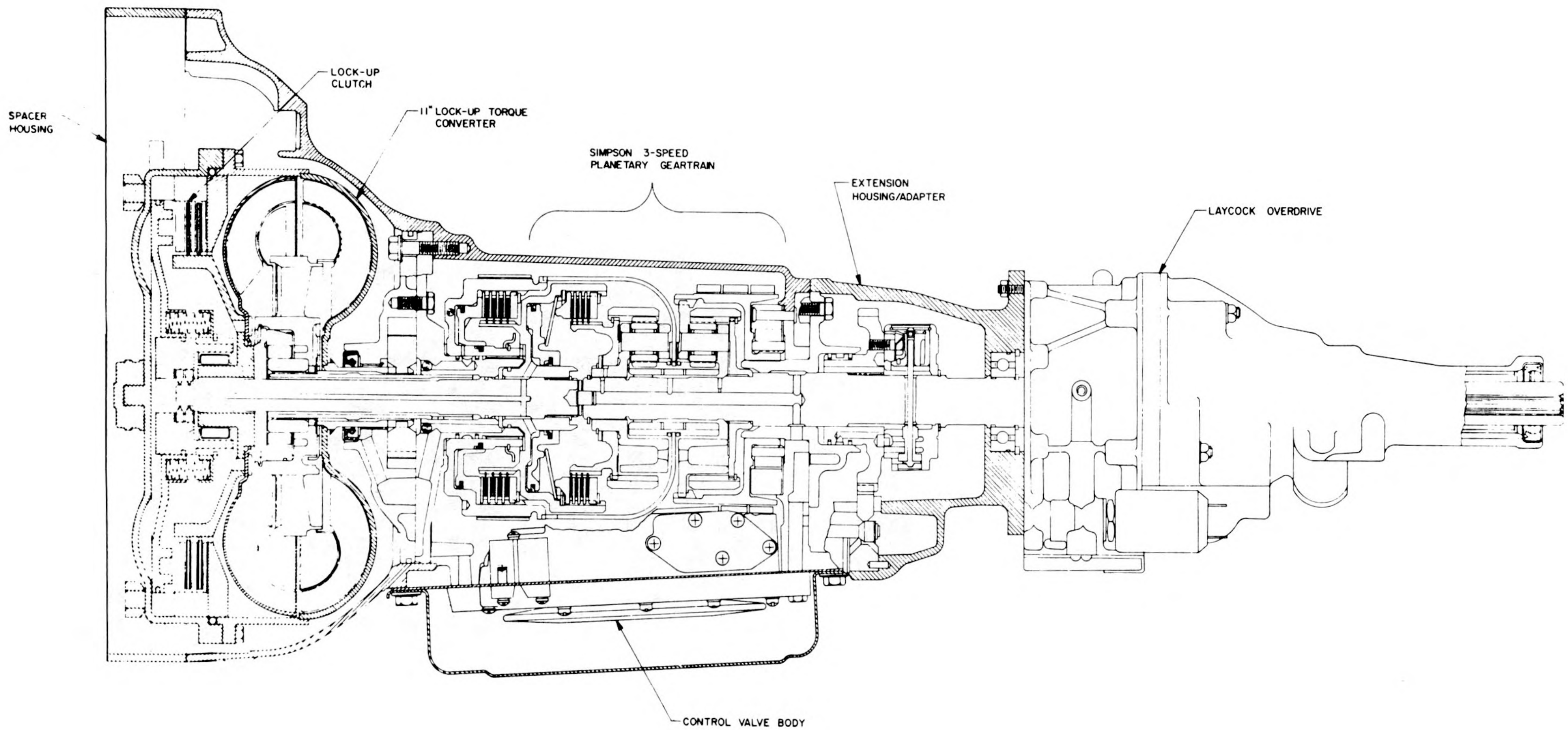

Figure 1

Arthur D Little, Inc.

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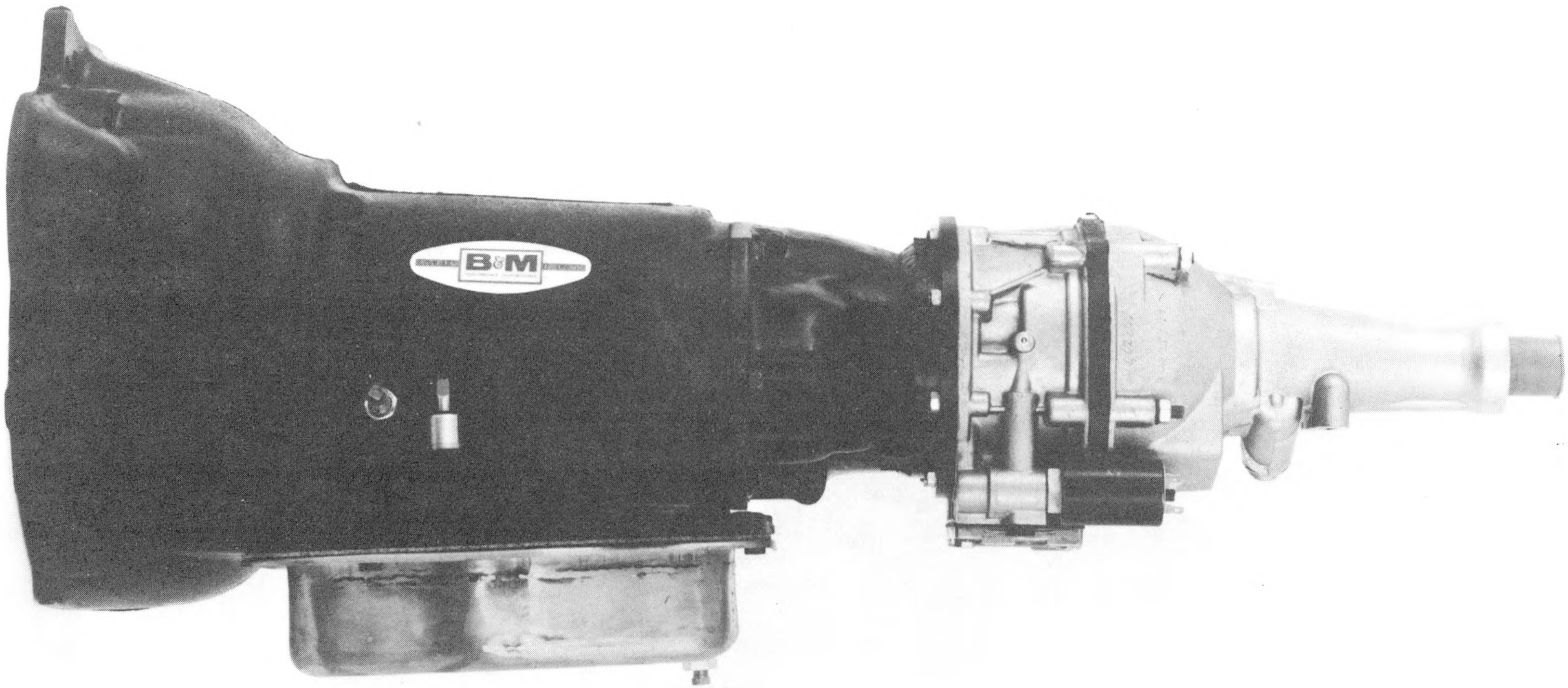

View of assembled A-904 transmission with overdrive unit and special pan.

Figure 2

the driver's seat through a unique electro-hydraulic control system. The lock-up torque converter is shown in Figures 3 and 4.

The vehicles used in this study are described in Table 2. These vehicles were chosen to represent a range of vehicles weights and engine sizes determined by the Department of Transportation to be typical of the 1980's. These cars however, were all 1975 models which met 1975 Federal Emission Standards.

Test Procedures

Three replicate 1975 EPA urban and highway tests of each of the following transmission configurations for each car were conducted in a random sequence:

- Wide-range three-speed
- Wide-range three-speed with lock-up in third gear
- Wide-range three speed with lock-up in second and third gear
- Wide-range four-speed
- Wide-range four-speed with lock-up in fourth gear
- Wide-range four-speed with lock-up in third and fourth gear

In addition, three baseline tests of the production configuration were tested before the start of the modified configurations to ensure that the vehicles met the 1975 Federal Emission Standards. Three additional baseline tests were conducted in a random sequence throughout the modified configuration tests. In all, 96 individual EPA urban and 96 individual EPA highway tests were conducted on the four cars. Fuel consumption was determined by both carbon balance and fuel weight measurement techniques. A total of 384 individual fuel economy data points were obtained in this study.

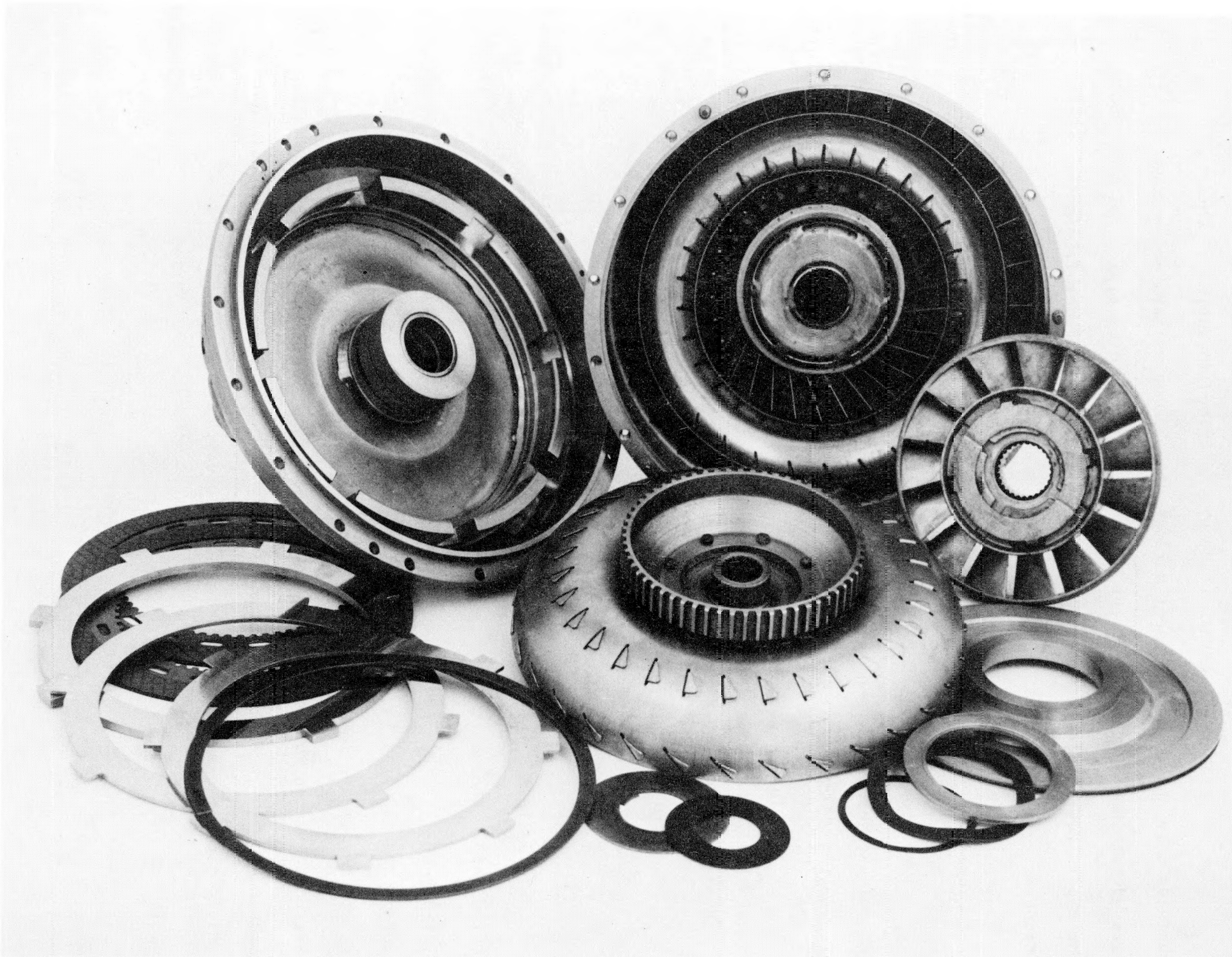

Figure 3

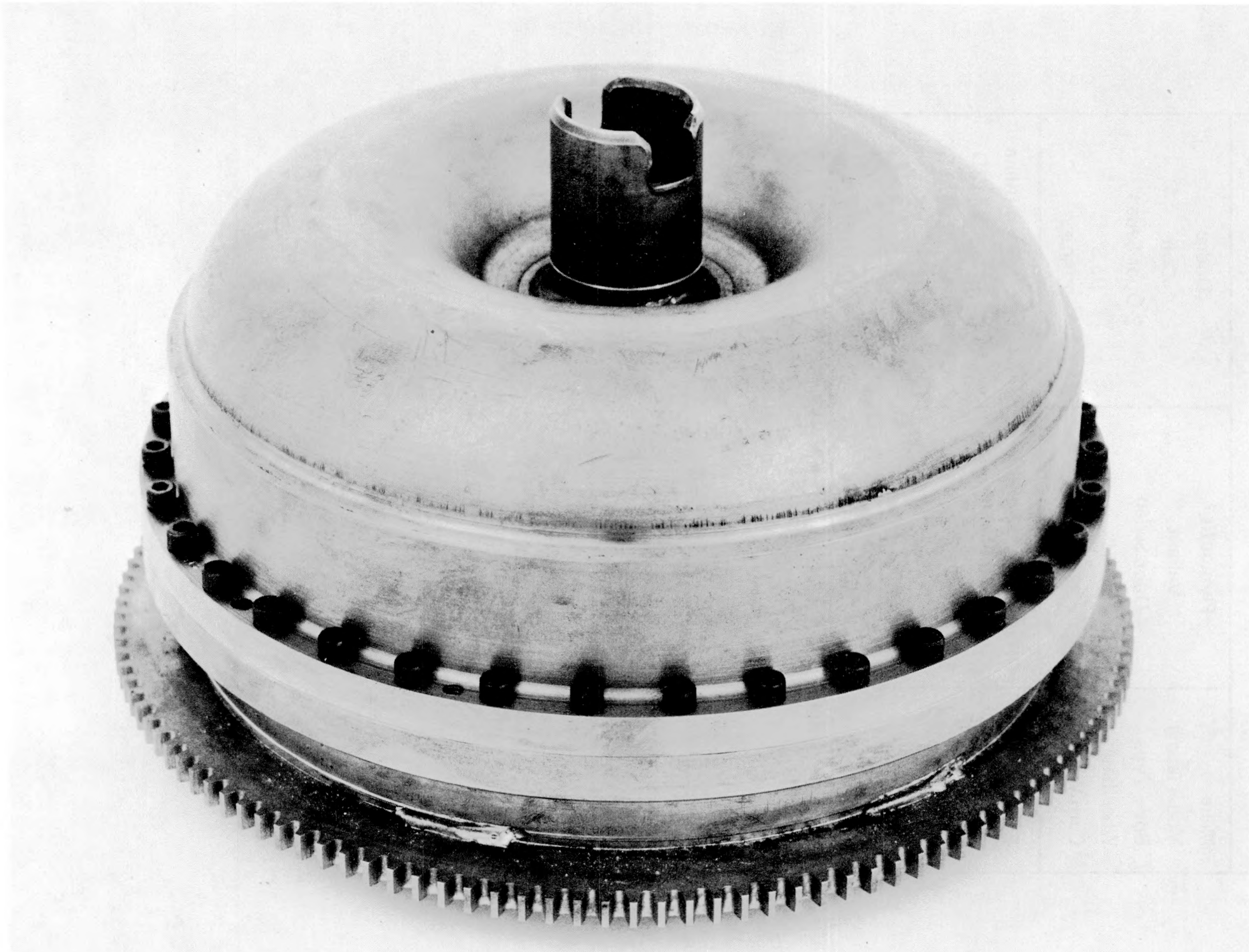

Figure 4 Assembled 10-3/4" diameter B&M lock-up converter.

TABLE 2
VEHICLE DESCRIPTION

Car Number	1	2	3	4
Make	Plymouth		Dodge	
Model Name	Valiant		Colt	
Body Type	4-Door Sedan		2-Door Sedan	
Wheelbase	111"		95.3"	
Curb Weight	3200#		2250#	
Engine Type	OHV–8	6 Inline	4 Inline	4 Inline
Displacement	318 CID	225 CID	122 CID	97 CID
	5213 CC	3688 CC	2000 CC	1600 CC
Horsepower	145–4000	95–3600	89–5200	79–5200
Unmodified Transmission	A727 Torque Flite		A904 Torque Flite	
1st Gear	2.45:1		2.45:1	
2	1.45:1		1.45:1	
3	1.00:1		1.00:1	
Standard Axle Ratio	2.45	2.76	3.55	3.89

The fuel economy results of these tests were statistically analyzed. Any set of three replicate tests of a given configuration which exhibited a coefficient of variance greater than 4.44% (95% confidence level) was rejected. A few retests of variant individual tests were necessary to meet this acceptance criteria.

The impact of these transmissions on vehicle driveability was determined by the MVMA Driveability Procedure as revised September 13, 1971. One of the vehicles is currently being modified for improved driveability as a result of these tests. This vehicle will be subjected to the Vehicle Acceptability Performance Evaluation Technique under development by Arthur D. Little, Inc.

Results

At the time of this writing all the test results are not available.* However, the general trends predicted by the computer simulation for the lock-up modes of the transmission appear valid. These are:

Vehicle		Percent Improvement	
Engine (CID)	Wide-Range Transmission	Urban (%)	Highway (%)
225	3-Speed, third lock-up	9	14
318	3-Speed, third lock-up	16	22
225	3-Speed, second and third lock-up	11	14
318	3-Speed, second and third lock-up	19	23
225	4-Speed, fourth lock-up	3	13
318	4-Speed, fourth lock-up	5	21
225	4-Speed, third and fourth lock-up	6	14
318	4-Speed, third and fourth lock-up	14	22

*An addendum of results after submission of this report (1/30/77) will be provided at the Fourth International Symposium on Automotive Propulsion Systems - April 18-22, 1977, Washington, D.C.

A slight decrease in fuel economy was experienced with the wide-range transmissions without lock-up due to increased slippage in the torque converter.

To date, emissions have generally stayed at or below the 1975 Federal regulatory levels. It is interesting to note that improvements in fuel economy have been accompanied by significant decreases of nitrogen oxide emissions with the wide-range three and four-speed transmissions locked up in the top two gears.

The driveability of the vehicles has not been significantly degraded. There are problems of engine lugging and minor torsional vibration at very low engine speeds. These problems are minor, however, and may be alleviated by changes in the downshift logic of the Chrysler automatic (production upshift and downshift points have been used throughout the test program) and a logic feature which prevents torque converter lock-up below 4-cylinder engine speeds of approximately 2500 rpm, 1300 rpm in the 6-cylinder, and V-8 speeds of 1000 rpm. We find that it is significant that this driveability performance has been achieved without the use of driveline torsional vibration dampers.

The wide spacing of the second to third gear shift in the wide-range three-speed transmission has not been found objectionable. This may be due to the fact that the shift time is on the order of 1-2 seconds, during which time there is no sudden change in driveline torque.

Conclusions

We can conclude from the preliminary results of this study that significant improvements in fuel economy can be obtained with wide-range three or four-speed automatic transmissions used in conjunction with lock-up torque converters. Such configurations may also enhance the reduction of nitrogen oxide emissions.

Furthermore, no significant degradation in vehicle driveability is inherent in the use of lock-up torque converters on wide-range transmissions. This last is not surprising, as the same conclusion has been evident ever since Oldsmobile first introduced the Automatic Safety Transmission in 1937, which power shifted without the benefit of a torque converter, and Packard and Studebaker marketed their Ultramatic and the Studebaker Automatic Transmission (Borg-Warner) respectively. Both of these early 1950's automatic transmissions utilized a lock-up feature in top gear.

The work performed to date indicates that computer simulation is a useful tool in predicting incremental changes in fuel economy. The predicted incremental changes in fuel economy were generally found to predict measured percentage changes in fuel economy within ± 5 percentage points.

Session 14

The Effect of Aerodynamic Drag on the
Fuel Economy of Passenger Cars

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ABSTRACT

The relationship between fuel economy and aerodynamics for full-size and subcompact cars is discussed. At first gasoline-powered and Diesel-powered synthesized cars were selected for an analysis using a computerized-simulation program to predict fuel economy and performance with and without incorporating of aerodynamic improvements. Secondly the technique for realizing the desired aerodynamic improvements on cars is shown.

By the so called "Detail Optimization Method" a drag coefficient of $c_D = 0.42$ within a usual styling concept is achievable. With such optimized gasoline-powered cars (values for Diesel-powered cars in brackets) maximum improvements in fuel economy referred to average '77 cars of 7 % (9 %) for full-size cars and 4 % (5 %) for subcompact cars are possible.

By aerodynamic modification with an as a minimum assumed drag coefficient of $c_D = 0.32$, which implies a styling concept strongly influenced by aerodynamics, improvements in fuel economy of 14 % (20 %) for full-size cars and 11 % (14 %) for subcompact cars are achievable.

Form FE 76a

The Effect of Aerodynamic Drag on the Fuel Economy of Passenger Cars

L.J. Janssen H.-J. Emmelmann

INTRODUCTION

The fuel economy of cars basically depends on three factors:

- the power and the SFC-characteristic of the engine
- the car's weight
- the aerodynamic drag

Within this paper mainly the last category is discussed: the relationship between fuel economy and aerodynamics for full-size and subcompact cars. The influence of aerodynamics on fuel economy for daily driving is often said to be negligible. In some cases this results because only City Driving and not the representative EPA Combined Driving is considered. In other cases only the gradient of a fuel economy increase by reducing the aerodynamic drag is discussed neglecting the big possible percentage potential of aerodynamic improvements compared with other influencing parameters, e. g. vehicle weight.

In this paper the fuel economy increase caused by aerodynamic improvements will be compared with those by reduction of weight and installed engine horsepower from the view of an auto maker having a considerable know-how in vehicle aerodynamics.

Because the effect of aerodynamic improvements is expected to be different for heavy standard-size cars and light subcompact cars or cars equipped with Diesel and gasoline engines and in order to cover the whole field of possible configurations the following investigations were done for a typical full-size car and a typical subcompact car both powered by Diesel as well gasoline engines. At first gasoline-powered and Diesel-powered synthesized cars were selected for an analysis using a computerized simulation program to predict fuel economy and performance with and without incorporation of aerodynamic improvements.

Secondly the way to obtain the desired aerodynamic improvements on cars is shown. The following items will be considered:

- Volkswagen's optimization method to reduce aerodynamic drag, practicability of body modifications
- Limitations of the optimization method
- Drag related potential of car aerodynamics and future prospects.

PROCEDURE - SIMULATION TECHNIQUE

All aerodynamic test results quoted in this paper were obtained in Volkswagenwerk's climatic wind tunnel. A drawing of the wind tunnel with its principal dimensions is given in Fig. 1. A detailed description of the wind tunnel including its measuring systems is given in (1, 2)*.

In the design stage of cars at the VW-Research Division a computer program is used to predict performance and fuel economy for steady-state operation as well as any given driving cycles. Usually the fuel economy values of EPA City Driving, EPA Highway Driving and EPA Combined Driving are computed taking into consideration the EPA regulations about shifting points.

The program determines the required engine output from instantaneous drag, roll and acceleration resistance as well as transmission efficiency data and takes the corresponding specific fuel consumption values out of SFC maps drawn from fuel economy bench tests, and is taking into account the transients.

Similarity analyses are used to produce engine maps for given engine families to determine the projected effects of power on vehicle fuel consumption and performance. The obtained values show good agreement with road tests.

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*Numbers in parentheses designate References at end of paper.

In order to make use of the full potential e. g. of a reduction of air drag, the transmission ratio must be adjusted, as shown in Fig. 2.

If the drag coefficient of a vehicle is lowered without alteration of the transmission ratio, maintaining the dynamic circumference of the drive wheels, or the engine data, the original load curve (Fig. 2a) is shifted towards a lower mean effective pressure PME at the same engine speed n. This new load point is usually associated with higher specific fuel consumption (Fig. 2b).

In view of the reduced power requirement, a drop in fuel consumption is achieved. The effect of this is, however, diminished by the increase in specific fuel consumption.
By means of the applied transmission ratio adjustment, the new load curve can be brought more or less into alignment with that of the vehicle in its original state (Fig. 2c), and in this case the same engine speed will be equivalent to higher road speed.

One normalized Diesel SFC map and one normalized gasoline SFC map are used as bases for the Diesel and gasoline engine families to exclude the influence of engine size on SFC characteristic.

In the following the effect of aerodynamic drag on fuel economy of two typical '77 cars, a full-size and a subcompact car, will be compared.

The full-size car was selected out of the redesigned '77 generation, which is already smaller, lighter and equipped with smaller engines than the '76 generation. The exterior dimensions and interior disposition of the typical subcompact in comparison to the full-size car is shown in Fig. 3.

Fig. 4 shows the well known definition of the aerodynamic drag coefficient c_D, including the definition of the reference area A.

The variance of drag among the cars is remarkable. As can be seen from Fig. 5, which is a revision of the figure published in (2) containing European cars only, the drag coefficient among them varies between $0.37 \leq c_D \leq 0.52$. The average drag coefficient is $\overline{c_D} = 0.46$.

Table 1 explains the investigated aerodynamic modifications and shows the variance of the parameters drag coefficient and frontal area starting from average '77 values (Aerodynamic Modification (A)). The frontal area is constant for all aerodynamic modifications of the subcompact cars while a reduction of frontal area for full-size cars even with constant interior dimensions to 2.15 m^2 seems to be possible (e. g. Mercedes-Benz 240 D/300 D/380 E: Area = 2.08 m^2, 450 SEL: Area = 2.19 m^2, new AUDI 100: Area = 1.96 m^2).

The possibility to realize the aerodynamic modifications according to (B) - (D) and the selected steps will be discussed later.

The effect of these aerodynamic modifications is analyzed for three synthesized car configurations. Table 2 shows the variance of the vehicle parameters weight, engine horsepower and power-to-weight ratio for the full-size and subcompact car families. The maximum possible weight reduction again starting from average '77 values, is assumed to be 10 % for the already light subcompact cars and 25 % for the full-size cars with their big potential for weight reduction. While design (1) and (2) show the same power-to-weight ratio and therefore the same performance, for design (3) the power-to-weight ratio is reduced 25 %. The acceleration of such cars is reduced too but is still in the range of accelerations of cars already on the road with an accepted performance (e. g. the Diesel-powered cars VW Rabbit-D, Mercedes-Benz 240 D, 300 D).

SIMULATION RESULTS IN FUEL ECONOMY

The potential in an increase of fuel economy given by means of reduction of aerodynamic drag is compared in Figs. 6 - 8 with the improvements by reductions of weight, engine horsepower, and a combination of weight and horsepower for the synthesized car design (1) (standard '77 design). For lower mean velocity and large amounts of idle and for acceleration and deceleration periods under city driving conditions, the effect of the aerodynamic drag reduction is less than that under highway driving conditions, where its potential in an increase of fuel economy is bigger compared with all the others.

Figs. 6 and 7 show the increase in fuel economy for EPA City Driving and Highway Driving. The following comment refers only to Fig. 8 where the effect on fuel economy for EPA Combined City and Highway Driving is plotted.

This figure shows for all cases that reducing only the weight increases the fuel economy less than by reducing the aerodynamic drag. On the other hand the most favourable influence for the full-size car results from a combined reduction of weight and installed engine horsepower (unchanged power-to-weight ratio). For a maximum possible reduction of both weight and engine power of 25 %, the fuel economy can be increased by 19 % for gasoline-powered full-size cars. For these cars the corresponding fuel economy increase is 13 % for a maximum possible reduction of aerodynamic drag of 40 %.
For the Diesel-powered full-size car the efficiency of a reduction of aerodynamic drag increases. In this case a 15 % improvement is possible compared with 17 % obtained by a reduction of weight and engine horsepower.

Opposite relations are observed for the subcompact car. By already realized reductions of weight; only a 10 % reduction of weight and engine horsepower for subcompacts seems to be possible. Therefore, the fuel economy improvement for the gasoline-powered car as well as for the Diesel-powered car of 7 % results. By a 30 % reduction of aerodynamic drag, assumed to be maximum possible for subcompact cars, the fuel economy increases 9 % for the gasoline-powered and 12 % for the Diesel-powered car.

It is obvious that the same reduction of weight and installed engine horsepower leads to a smaller improvement of fuel economy for the Diesel-powered car compared with the gasoline-powered car.

For the improvements caused by better aerodynamics a greater improvement for Diesel-powered cars compared with the gasoline-powered cars results.

These relations result from different characteristics of the used Diesel and gasoline SFC maps.

The percentage fuel economy improvements are summarized in Fig. 9. The potential of fuel economy increase from aerodynamics can be taken out of this figure for the synthesized standard configuration (Design ①) and improved configurations (Design ② and ③). Only the effects on the EPA Combined Driving shall be interpreted.

The percentage fuel economy improvements for Diesel-powered cars are in all cases slightly better than for gasoline-powered cars. The possible improvement for full-size cars is higher than for subcompacts. It is obvious that design ③ with the most favourable conditions for a high fuel economy has the best efficiency for aerodynamic improvement too.

For full-size cars aerodynamic modifications dependent on design yield maximum improvements of 13 % to 14 % for gasoline-powered cars and 15 % to 20 % for Diesel-powered cars.

For subcompact cars the corresponding increase is 9 % to 11 % and 13 % to 14 % for gasoline-powered and Diesel-powered cars respectively.

REALIZATION OF AERODYNAMIC MODIFICATIONS

Historical development of vehicle aerodynamics

The development of vehicle aerodynamics, considered here only in conjunction with body styling, took place in three main stages, although these can not be sharply distinguished from one another in terms on time. The major steps in this development process are illustrated in Fig. 10; the comments are restricted to consideration of passenger cars.

In the first phase, which might be considered as having started about the turn of the century, an attempt was made to adopt directly various streamlined shapes known in other engineering disciplines such as shipbuilding, sailing or flying. These shapes, however, did not succeed in gaining a foothold in automobile engineering.

The second phase was characterized by efforts to apply knowledge of fluid mechanics (obtained mostly from aircraft aerodynamics) systematically to the automobile and, as development progressed, to adapt such outlines effectively to the practice of automobile engineering. The aim was to build a streamlined car. Aircraft designers, in fact, produced a whole series of vehicle shapes, but scarcely any of them proved acceptable for road vehicle use, mainly on account of excessive length and a poor relationship between available internal space and overall dimensions. It was not until the work of W.E. Lay, E. Everling, and W. Kamm was published that the fact which permit acceptable road vehicle aerodynamics to be achieved became available for mass-production vehicles. The main results of the work referred to was what we know today as the "Kamm tail". This successfully fulfilled the requirement for a streamlined automobile without an excessively long, flowing tail.

The third phase is still in progress and differs in its approach from the preceding two. No further attempts are being made to develop streamlined shapes and to adapt them subsequently to the demands of the automobile engineering. Instead, numerous details of the overall shape of the vehicles are optimized in order to obtain good airflow characteristics without sacrificing the features of a particular styling.

The detail optimization method

The optimizing method employed by VW (3, 4, 5, 6) is based on the postulate that the styling concept of a vehicle must be accepted as it stands. Aerodynamic improvements can only be attempted in the form of detail changes. These must be executed in such a way as not to change the vehicle's appearance. An example for this procedure, taking into account the peripheral conditions referred to, is shown in Fig. 11 which summarizes the main results of shape optimization on a medium-sized automobile. The left part of the figure shows the shape changes which proved necessary in order to achieve the reductions in drag shown in the diagram on the right. The form modification proposals shown here constitute the optimum values which can be achieved, if we define as an optimum form modification one which produces the full aerodynamic effect with the minimum deviation from the original shape. In order to arrive at these forms, it is necessary to vary each individual detail of the form systematically within reasonable limits. From Fig. 11 it will be clear that the range of results which can be obtained through aerodynamic form optimization is relatively small. On the other hand, the illustrations show that full use can be made of the range, however small, if the detail work is performed with great care.

An example how the optimizing method is applied on a specific detail is shown in Fig. 12, where the development of the leading edge of VW's subcompact car Rabbit is demonstrated.

The original front end form (a) with fully-separated hood airflow - caused by the sharp-edged front end contour - yields a high drag coefficient of $c_D = 0.48$.

In order to establish what degree of drag reduction can be achieved as a maximum by means of optimum front end design, the airflow around the bow is initially improved by an add-on nose section produced in accordance with pure aerodynamic principles with no consideration given to questions of appearance.

By mounting the add-on optimum nose (Form (b)), a decrease of 17 % in drag was achieved. The optimum nose was removed and an attempt was made by a step-by-step method to meet the results obtained with the optimum nose.

In the case of form (c) , on which the flow around the front edge was improved by an aerodynamically sound molding, the air flow around the hood is partly separated. The drag coefficient is reduced only 4 %.

Finally, the rounded-off front edge as in form (d) yields a flow over the hood free from separation, associated with a drag 15 % lower than that of the sharp-edged original front end form. This shape with a drag coefficient near the optimum was still acceptable to the stylist and is used as front end contour of VW's subcompact car Rabbit.

Fig. 13 shows the streamlines past VW Rabbit, which was optimized by the method just described. It is clearly seen that there is no separation of the airflow over the entire front part of the car. The picture was taken with the aid of multiple exposures of a single smoke trail, the height of which was varied in increments.

Limitations of the optimization method - future prospects

It is of course impossible to quote a precise value for the drag coefficient limit which could be achieved by the use of this optimizing method. The figure depends to a large extent on the original styling concept. A series of optimizations performed on subcompact- and medium-sized automobiles has shown that the lower limit is $c_D \approx 0.42$ when the styling concept is not influenced (Aerodynamic Modification Ⓑ). A typical car of this category is VW Rabbit, as shown in Fig. 13 and VW's sports coupe Scirocco.

A drag coefficient of $c_D \approx 0.37$ is possible when the styling concept is aerodynamically influenced and a wind tunnel optimization process follows (Aerodynamic Modification Ⓒ). The streamlined cars Citroen ID 19 and NSU Ro 80 shown in Fig. 10 and the new Porsche 924, shown in Fig. 14 are typical cars of this category.

If it is desired to reduce the drag coefficient even further, the optimizing method will have to be abandoned. Instead, specific low-drag configurations will have to be developed and will be bound to involve a different styling approach. Since the majority of current mass produced automobiles can claim only a drag coefficient far worse than the limit value of $c_D = 0.37$ as achieved by optimization, it would be appropriate at this time to focus on the potential of the optimization method.

Fig. 15 represents an attempt to illustrate the trend in the development of automobile air drag. Reliable data are available only for vehicles of fairly recent date. Comprehensive data are similarly required if one is to be able to reconstruct an accurate picture of the development of drag coefficient. Such data, however, are not available. Therefore reference is made to the results published by MIRA (7) for some older vehicle shapes. Disregarding the drop in two stages in drag coefficient from $c_D = 0.8$ for cars of the Twenties to an average value of $c_D = 0.46$ for 1960 European automobiles (3), since that time no tendency of a systematic reduction in drag coefficient can be detected.

The drag coefficient value $c_D = 0.15$, which in Fig. 17 is given particular prominence, was achieved as early as 1922 by W. Klemperer (8) on a model as shown in the left sketch at the bottom of Fig. 15. Other shapes, for example that drawn on the right in Fig. 15 with overall dimensions of a modern subcompact car, have been used by Volkswagenwerk recently to obtain an equally good drag coefficient of 0.15 which can be regarded as the probable optimum value for an automobile. Even if this figure is not based on any form of natural law. The gap between this value and those achieved by modern mass-produced automobiles is considerable.

The lowest possible drag coefficient for a passenger car is assumed to be between the best car out of Fig. 3 with $c_D = 0.37$ and an aerodynamically ideal vehicle body (Fig. 15) with $c_D = 0.15$.

Based on a 10-years-experience in the field of aerodynamic optimization of full-scale vehicles we are of the opinion that special car shapes can be technically realized in the range of $c_D = 0.32$ to $c_D = 0.28$.

For the aerodynamic modification (D) such a favourable drag coefficient of $c_D = 0.32$ has been assumed. The realization of extremely low drag coefficients for mass-produced cars requires some presuppositions. By the aid of further basic windtunnel tests completed by theoretical investigations of the flow around cars the knowledge of drag mechanismen must be enlarged. Further the interference between styling and aerodynamics must take place in a different way. Contrary to the present stage the development of car shapes should start in the windtunnel to define the main vehicle contours which are necessary to achieve low drag coefficients. Subsequently the real styling phase should begin. In this period it must be secured by escorting wind tunnel tests that the aspired drag coefficient can be maintained.

This method requires more time for developement and essentially impeded task for the stylists. The question whether higher costs are combined with a low drag coefficient or not is only depending on the conversion from aerodynamically required shape contours to attractive car shapes, the aerodynamic requirements itself do not contribute to higher costs.

ACKNOWLEDGMENT

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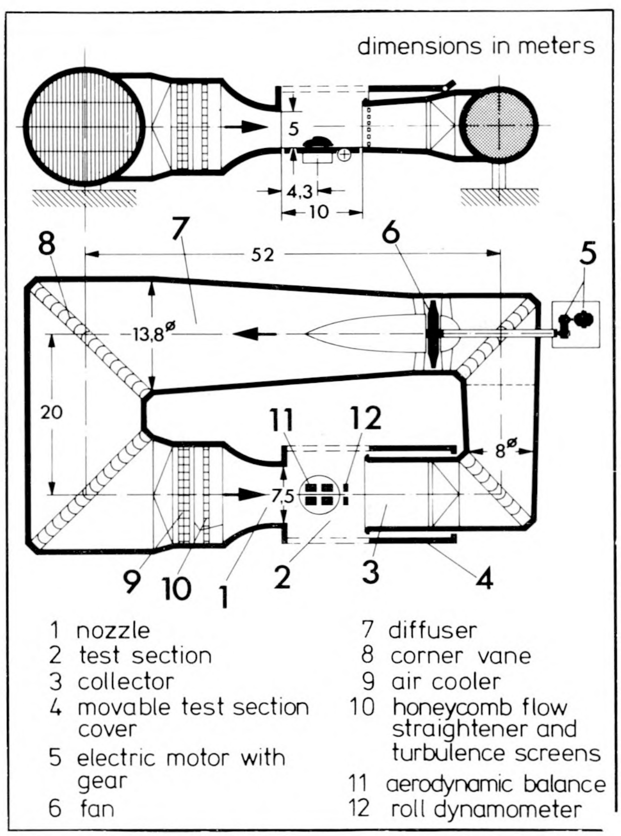

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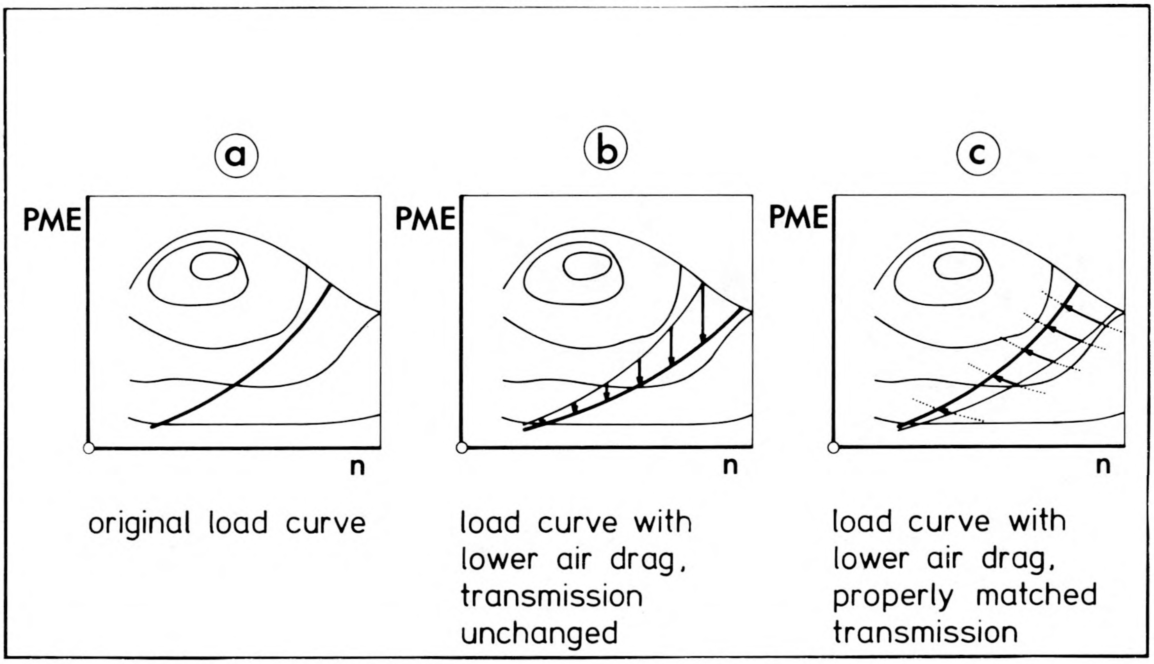

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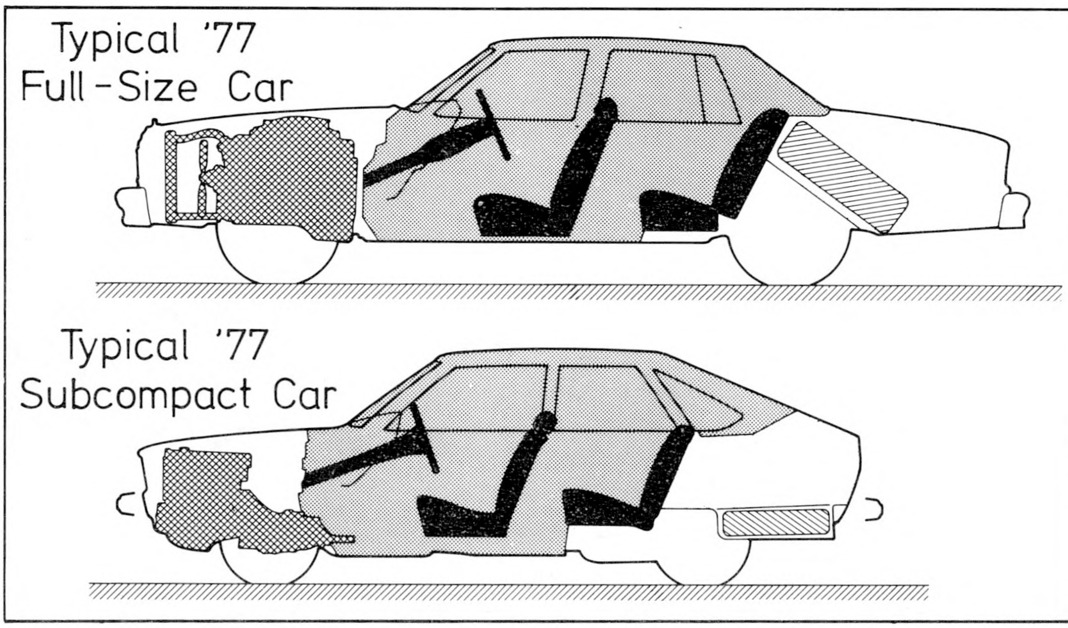

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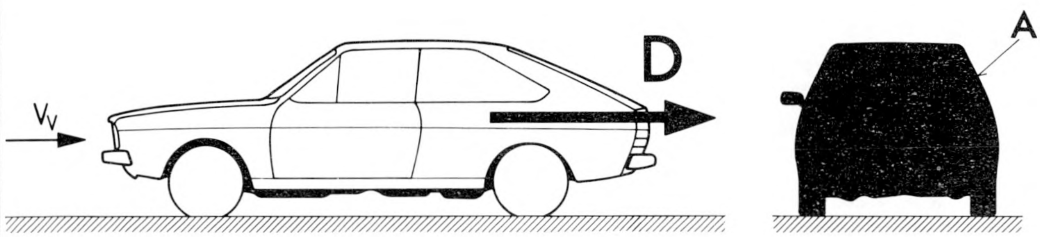

$$D = c_D \cdot A \cdot \frac{\varrho}{2} V_V^2$$

$$\boxed{c_D = \frac{D}{A \cdot \frac{\varrho}{2} V_V^2}}$$

$$P_D = D \cdot V_V = c_D \cdot A \cdot \frac{\varrho}{2} V_V^3$$

D = Aerodynamic Drag

c_D = Aerodynamic Drag Coefficient

A = Vehicle Projected Frontal Area

ϱ = Air Density

V_V = Vehicle Velocity

P_D = Aerodynamic Drag Power

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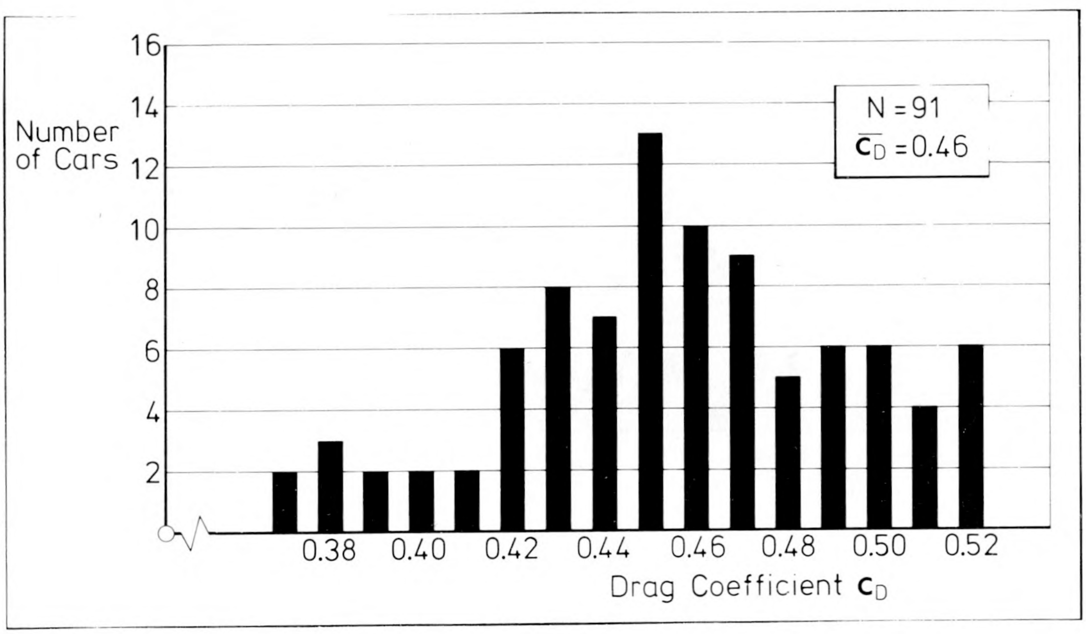

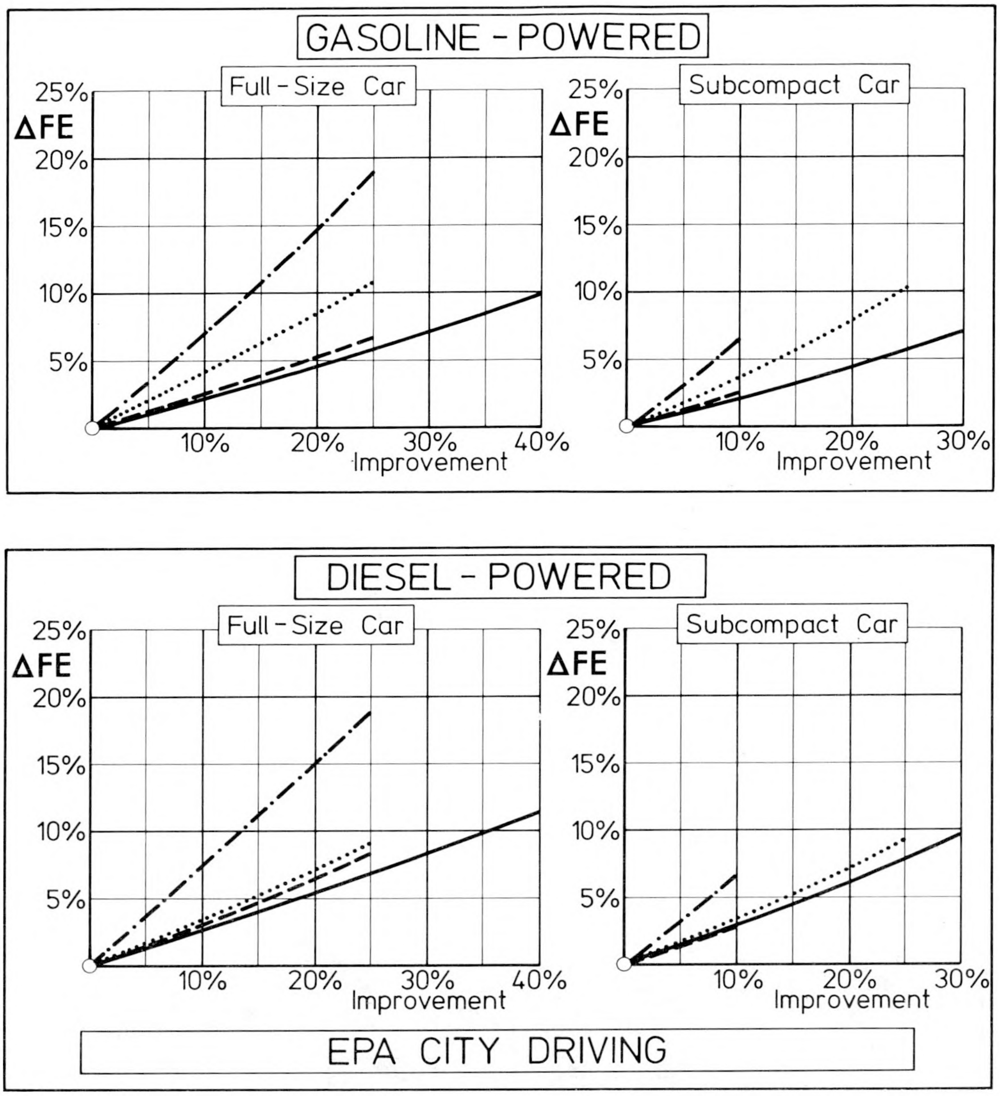

PERCENTAGE INCREASE FUEL ECONOMY **ΔFE**

- ——— by reduction of aerodynamic drag
- – – – – by reduction of vehicle weight
- ········· by reduction of engine horsepower
- –·–·– by reduction of both vehicle weight and engine horsepower (unchanged power - to - weight ratio)

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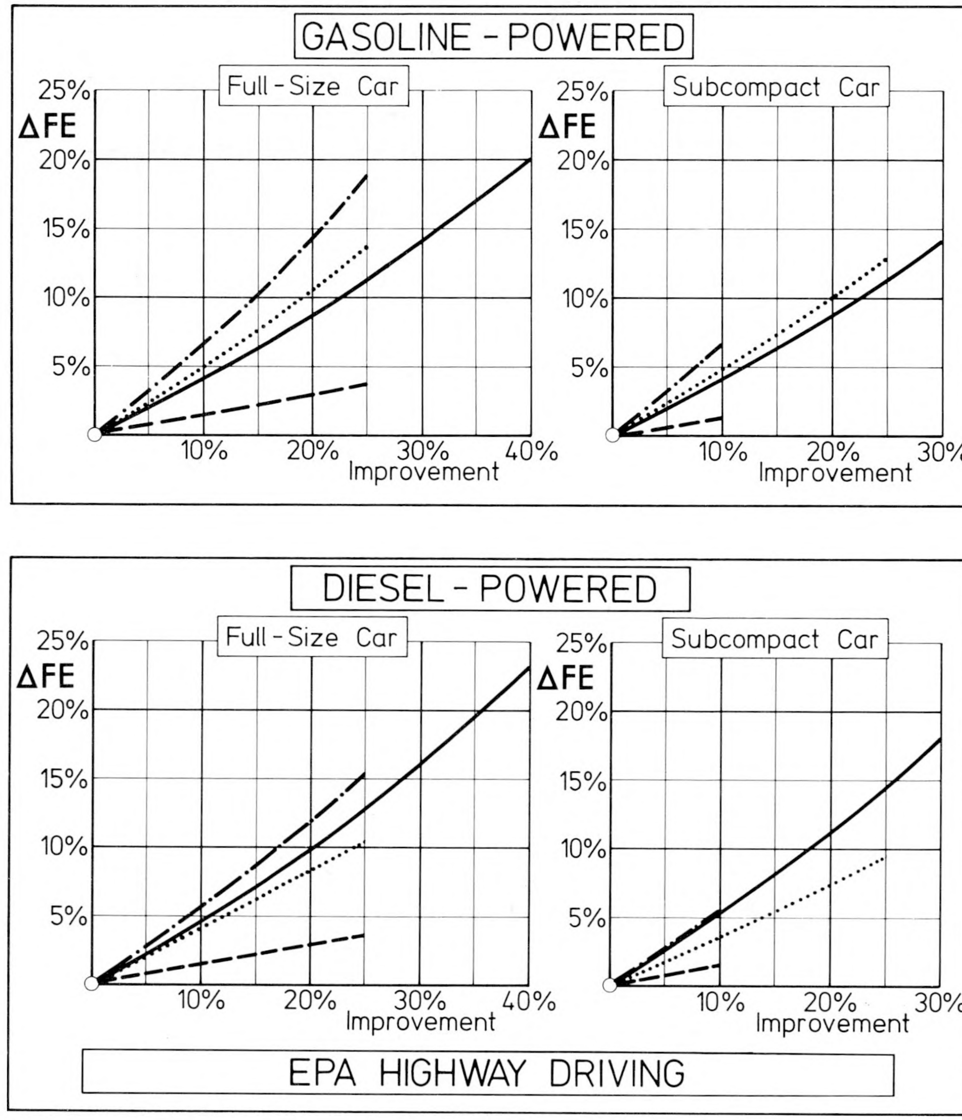

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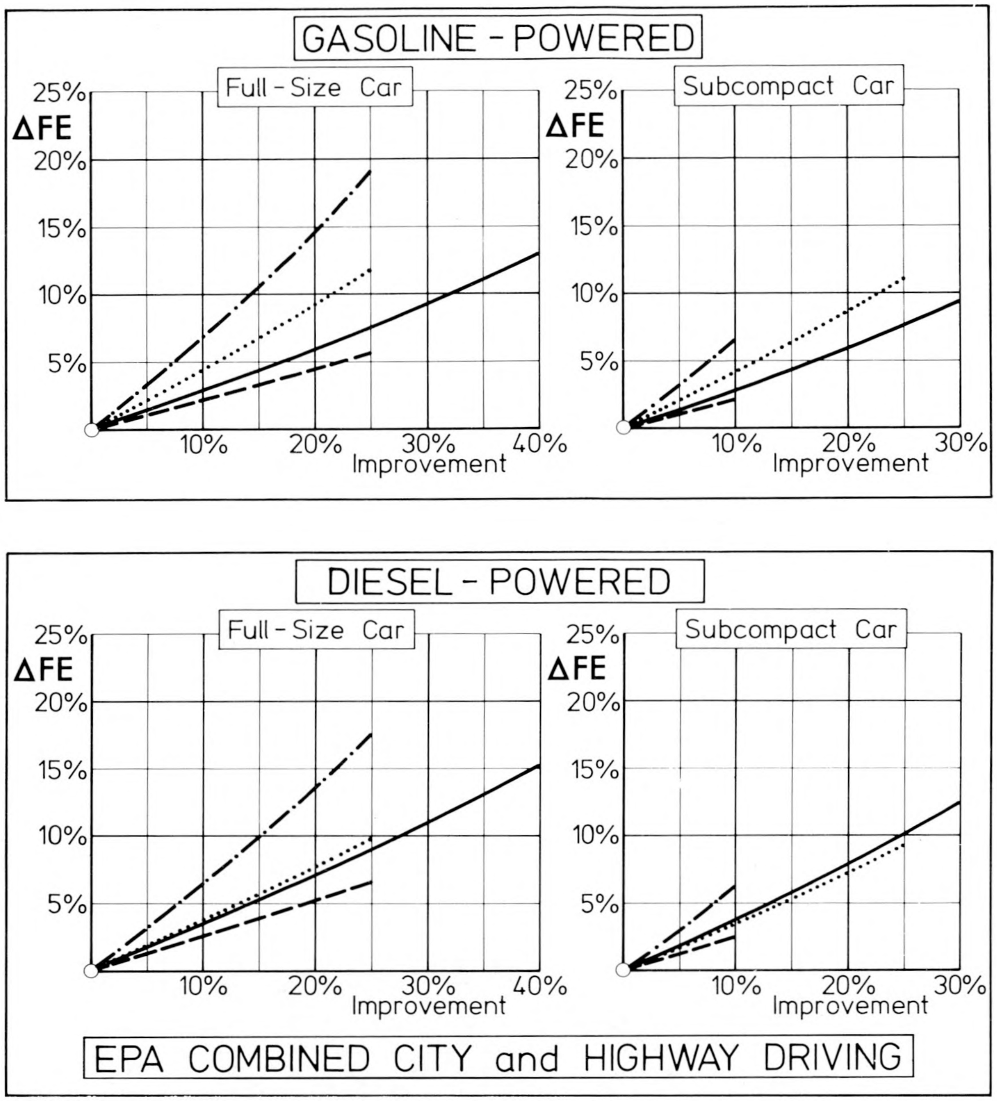

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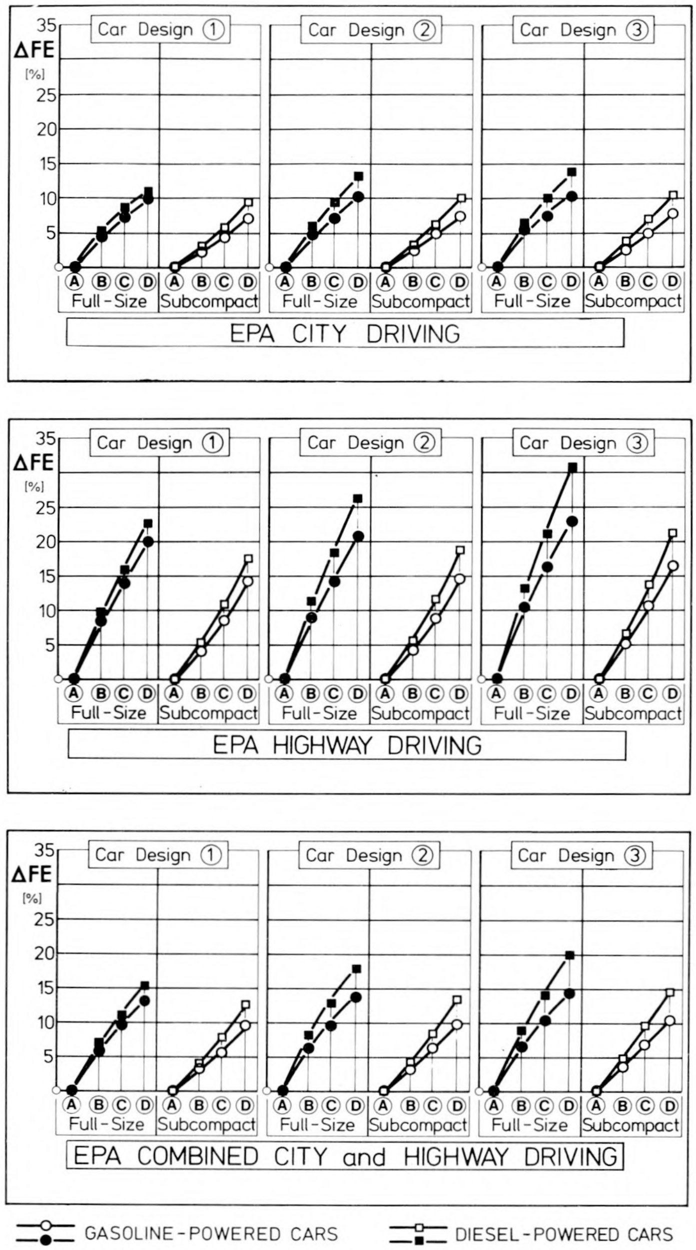

Aerodynamic Modifications (Details see Table 1)

Ⓐ Standard '77 aerodynamics

Ⓑ Reduced aerodynamic drag ($c_D = 0.42$), no influence on styling concept

Ⓒ Reduced aerodynamic drag ($c_D = 0.37$), styling concept aerodynamically influenced

Ⓓ Further reduced aerodynamic drag ($c_D = 0.32$), styling concept strongly influenced by aerodynamics

Synthesized Cars (Details see Table 2)

Design ①: Standard '77 design

Design ②: Maximum possible weight and power reduction, which maintains the same power-to-weight ratio as design ①

Design ③: Maximum possible weight reduction as for design ② with the power-to-weight ratio reduced 25% based on '77 design

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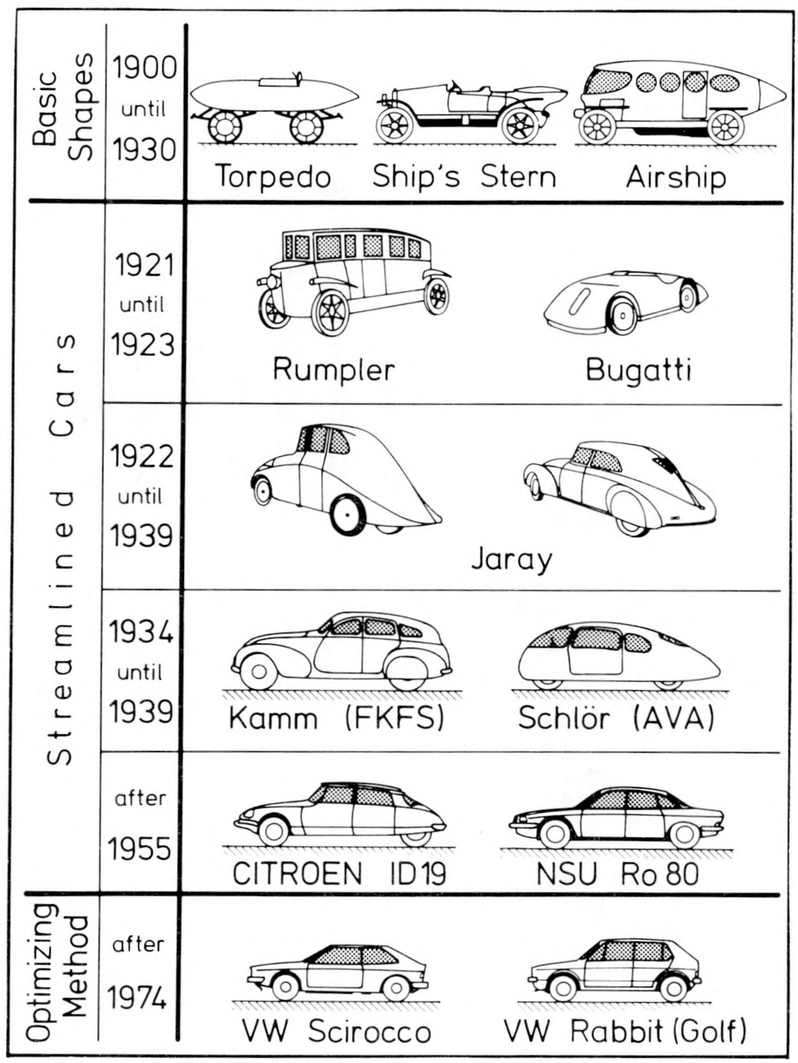

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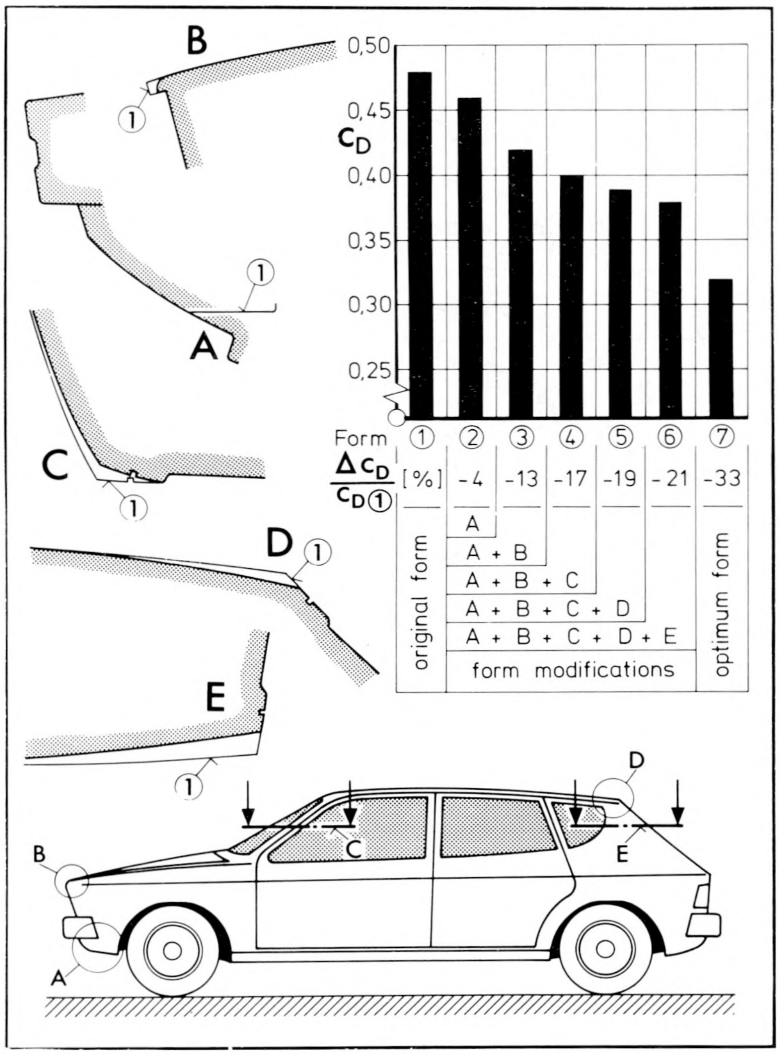

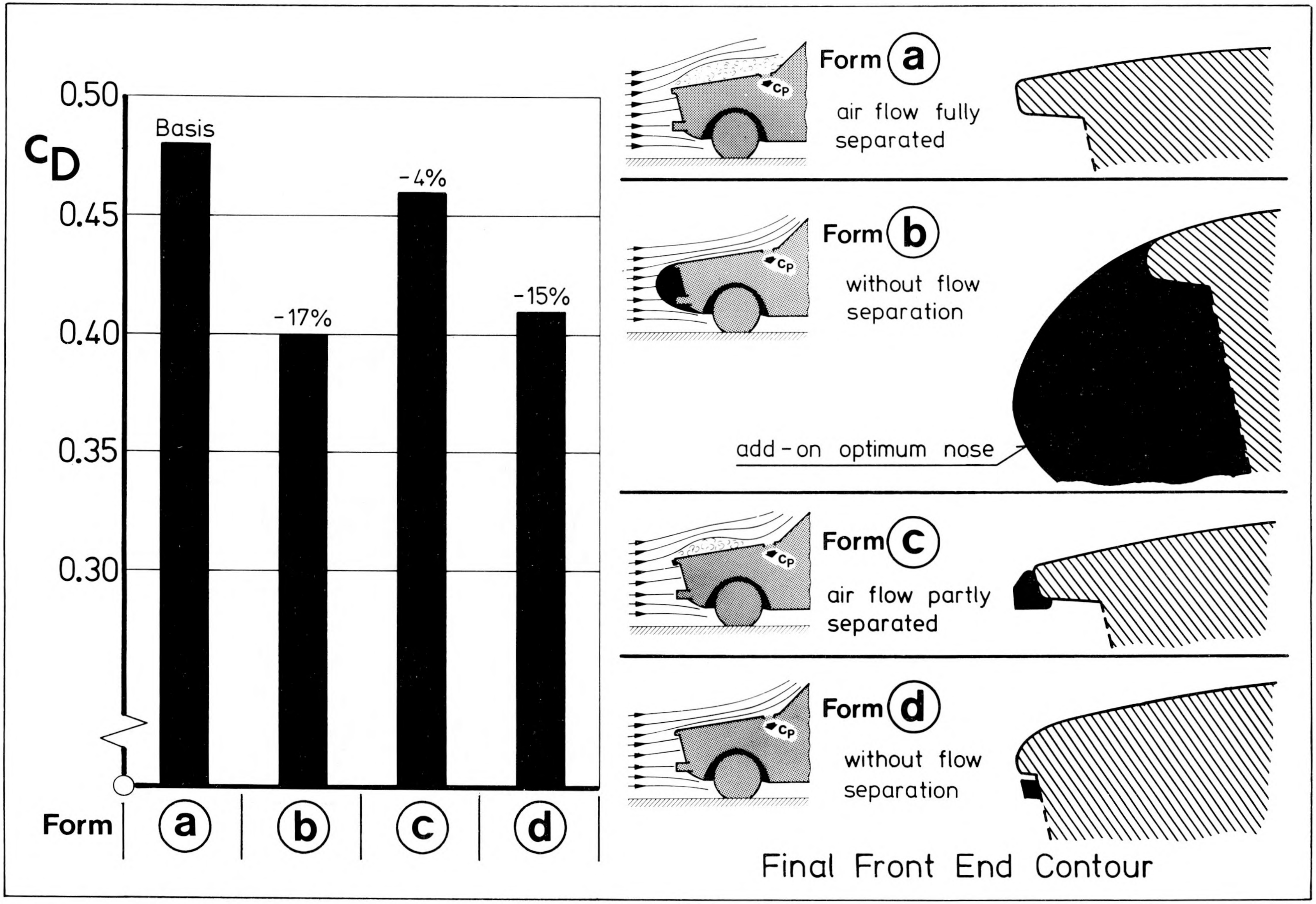

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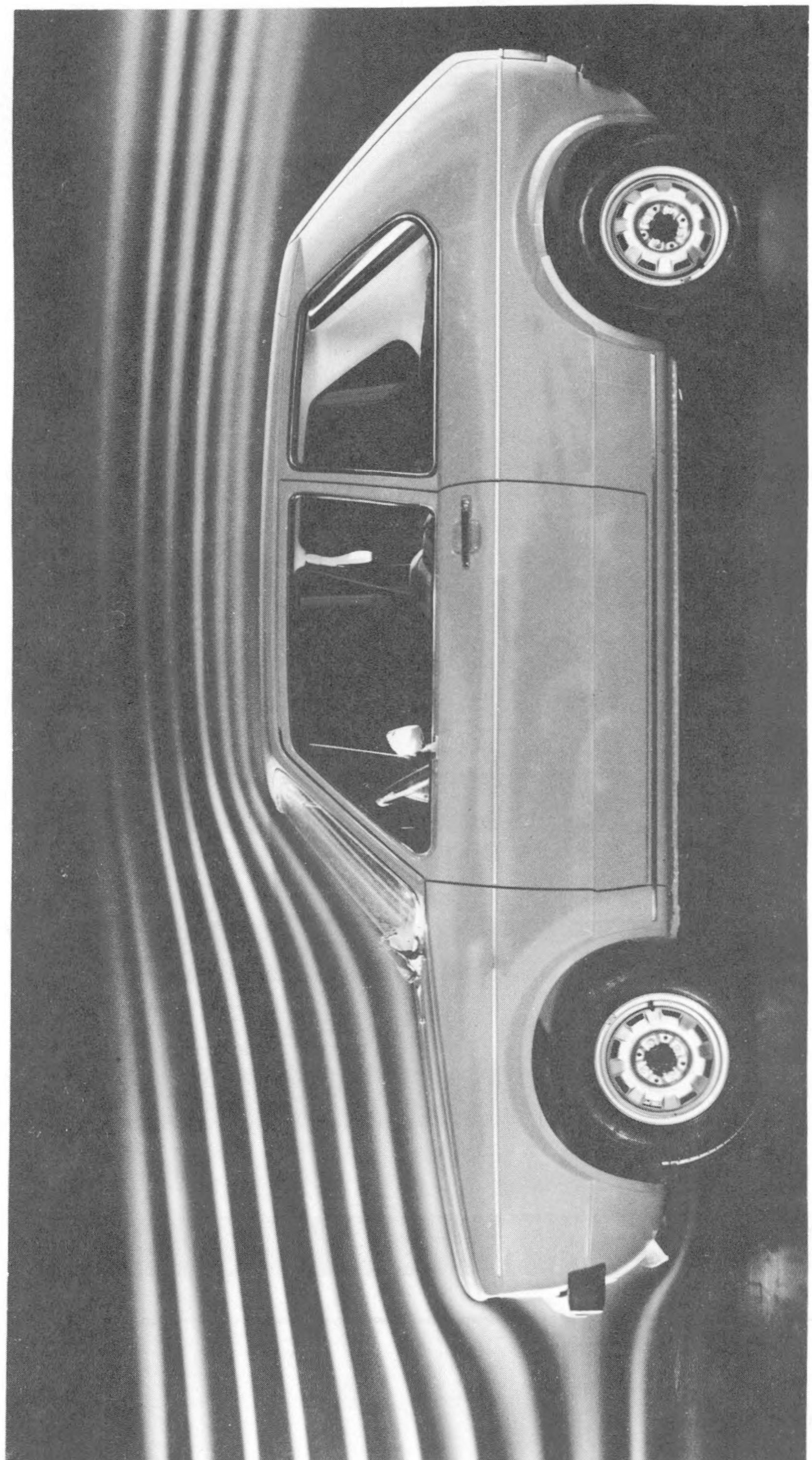

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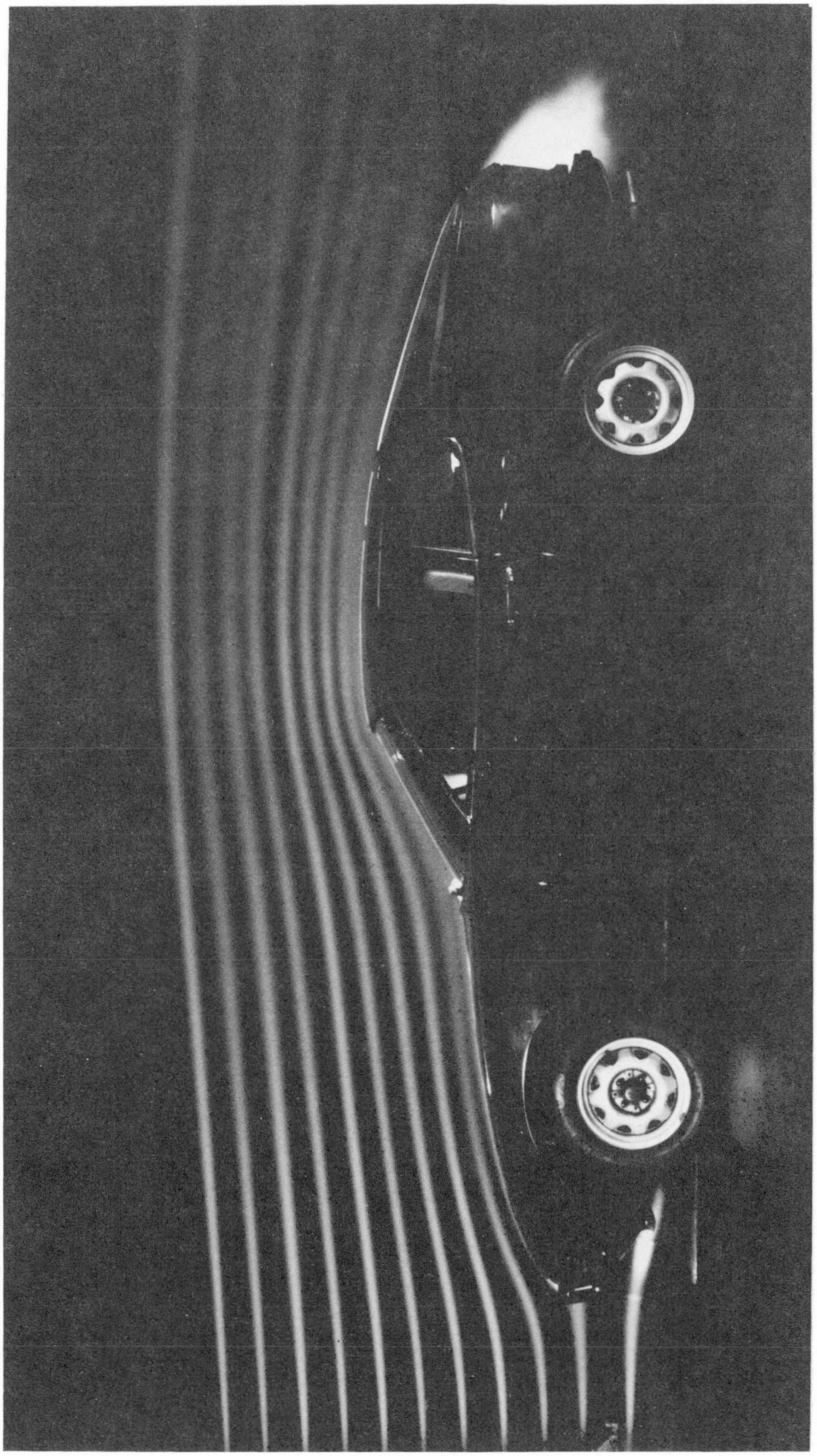

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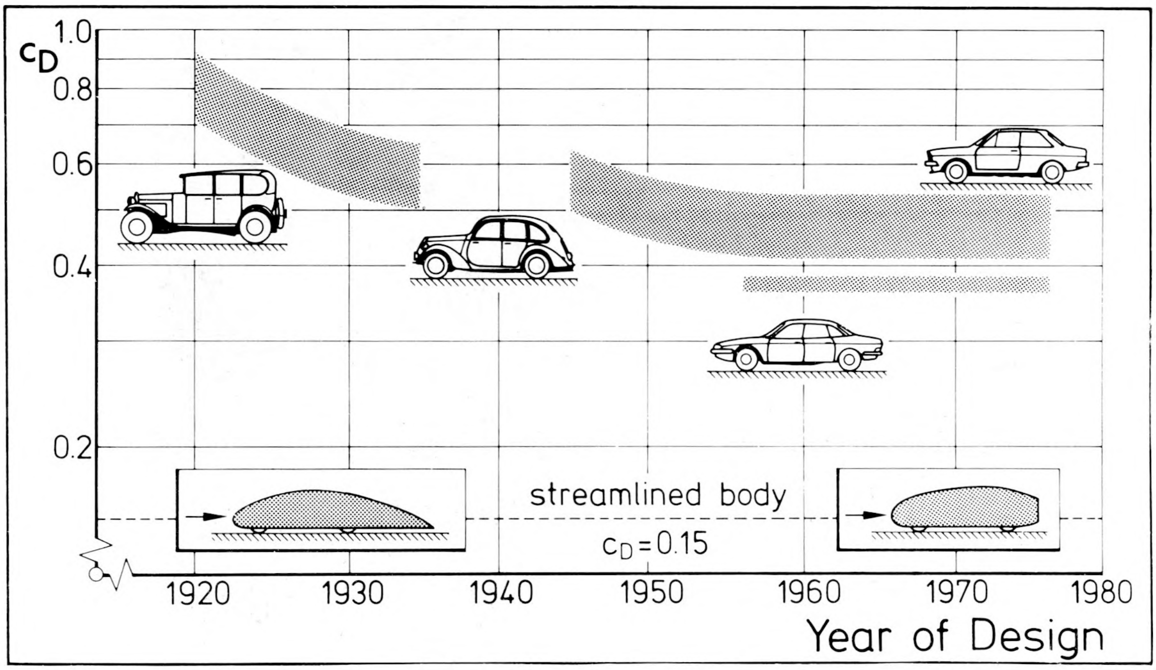

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Aerodynamic Modifications

Ⓐ Standard '77 aerodynamics

Ⓑ Reduced aerodynamic drag ($c_D = 0.42$) by „DETAIL OPTIMIZATION METHOD" - no influence on styling concept

Ⓒ Reduced aerodynamic drag ($c_D = 0.37$) by „DETAIL OPTIMIZATION METHOD" with the styling concept aerodynamically influenced

Ⓓ Further reduced aerodynamic drag ($c_D = 0.32$) with the styling concept strongly influenced by aerodynamics

	FULL-SIZE CAR Aero Modifications				SUBCOMPACT CAR Aero Modifications			
	Ⓐ	Ⓑ	Ⓒ	Ⓓ	Ⓐ	Ⓑ	Ⓒ	Ⓓ
Aerodynamic Drag Coefficient c_D [-]	0.50	0.42	0.37	0.32	0.46	0.42	0.37	0.32
Vehicle Projected Frontal Area A [m^2]	2.30	2.15	2.15	2.15	1.85	1.85	1.85	1.85
Product of c_D times A (Proportional Aerodynamic Drag) $c_D \cdot A$ [m^2]	1.15	0.90	0.80	0.69	0.85	0.77	0.69	0.59
	Basis	-21%	-31%	-40%	Basis	-9%	-20%	-30%

Synthesized Cars

Design ① : Standard '77 design

Design ② : Maximum possible weight and engine horsepower reduction, which maintains the same power-to-weight ratio as design①

Design ③ : Maximum possible weight reduction as for design ② with the power-to-weight ratio reduced 25% based on '77 design

All Designs : — Four speed manual transmission
— Steel belted radial ply tires

		Synthesized FULL-SIZE CARS			Synthesized SUBCOMPACT CARS		
		Design①	Design②	Design③	Design①	Design②	Design③
Loaded Vehicle Weight (IW)	[kg]	1812	1359	1359	1133	1019	1019
	[lbs]	4000	3000	3000	2500	2250	2250
		Basis	- 25%	- 25%	Basis	- 10%	- 10%
Engine Horsepower (P)	[kW]	91	68	51	57	51	38
	[hp]	122	91	68	75	68	51
		Basis	-25%	-44%	Basis	-10%	-32%
Power-to-Weight Ratio (P/IW)	[W/kg]	50	50	37.5	50	50	37.5
		Basis	± 0	- 25%	Basis	± 0	- 25%

SUBJECT TO REVISION PA313/77

ENERGY FOR ROAD TRANSPORT IN THE UNITED KINGDOM

by

J Porter and J W Fitchie
Transport Systems Department
Transport and Road Research Laboratory

Abstract

Looking at the short-to-medium term, examples are given to illustrate the use of computer simulation and instrumented cars to identify and evaluate potential energy conservation measures. For the longer term, when natural crude oil becomes scarce and expensive, the paper compares vehicles using synthetic gasoline from coal and battery vehicles, and examines the factors needed to give a primary energy advantage of one type of vehicle over the other. However, other considerations need to be assessed and it is suggested that the main need now is to continue the research and initial development needed to keep both options open.

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ENERGY FOR ROAD TRANSPORT IN THE UNITED KINGDOM

1. INTRODUCTION

This paper describes studies of some of the effects on road transport of changes in the price and availability of energy supplies. It deliberately concentrates on the United Kingdom, and considers both the short to medium term, when conservation measures are the most important practical steps, and the longer term, towards the first quarter of the next century, when natural oil may become scarce, and more fundamental changes are possible.

The remaining years of this century are likely to see the United Kingdom more than self sufficient in oil, because of the exploitation of the North Sea resources. Nevertheless, there is likely to be considerable economic incentive for energy conservation because, unlike the two decades from 1950 when oil was abundant and cheap, the price of oil is likely to increase in real terms as more expensive deposits have to be exploited. Thus scarcity and its effect on price will tend to reinforce the drive for energy conservation.

In the first quarter of the next century, it seems likely, on present estimates, that the United Kingdom will cease to be self sufficient in oil and will again become a net importer of energy, if indigenous reserves of coal, nuclear electricity, and the renewable resources of solar and wave power are insufficient. In this period, the kind of energy available for free moving road transport is considered in broad terms.

At the present time transport fuels account for about 15% of total United Kingdom energy demand, and on most projections this percentage does not increase above 25% in the future. About three quarters of primary fuel is used in the industrial and domestic sectors, and it is these sectors, and not transport, which are likely to dominate the overall pattern of energy demand and thence supply. Although transport fuel is a premium product, the transport sector is unlikely to drive energy demand nor will it dominate energy policy. Thus, both to establish a timescale for any potential changes in transport fuels and in order to map out the general features of the energy situation, it is necessary to make projections of total UK energy supply and demand. A preliminary study was completed in 1972, but a more extensive study involving the Energy Research Group at the Open University was completed in 1976. This latter study* provides a plausible energy supply and demand scenario against which the present analysis may be conducted. It examined primary fuel supply and demand for both high and low demand cases as illustrated in Figure 1. It suggested that in the remaining years of this century North Sea oil surpluses would stimulate GDP growth rates above the lower level, but that the energy deficit which followed would depress GDP growth. On this basis it concluded that only a medium growth rate (2½% to the year 2000 and 1½% thereafter) was consistent with the assumed availability of fuel supplies.

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* Chapman P, Charlesworth G, Baker M. Future Transport Fuels. Transport and Road Research Laboratory, SR251, 1976.

It is against this medium growth sceanrio that the transport fuel situation is now examined. The paper describes the use of computer simulations and instrumented cars as an aid to the assessment of energy conservation measures in the short to medium term. It then moves on to a discussion of the relative merits of liquid fuel from coal compared with electricity stored in batteries as alternative fuels for road transport for the next century.

2. THE IMPORTANCE OF CONSERVATION

Road transport is highly dependent on supplies of oil, and this is a dependence which is unlikely to be either easily or quickly altered. But by the more efficient operation and design of transport systems conservation of oil is possible without the large scale reductions in mobility which rationing (physically or by price) would bring about.

Energy, though important, is not the only resource used in transport, and the evaluation of transport improvements must also take into account time savings, labour costs, convenience, and environmental factors. Even at todays oil prices, a typical cost benefit analysis of a road improvement scheme might show direct energy savings of only about 10% of total benefits, while time savings were valued at 80%. This ratio of 8-1 in favour of time savings appears to indicate that energy conservation is not a major factor in present decisions. To some extent this is true, and it is a shortcoming of conventional cost benefit analysis which needs developments like energy analysis* to overcome. This technique can show the whole influence of energy on alternative options by making clear the feedback loops which amplify direct energy consumption. It can also emphasise the thermodynamic limits of processes, and the difficulty of substituting another factor of production for energy - unlike the substitution between labour and capital which is inherent in much economic analysis.

However the shortcomings of cost benefit analysis are mitigated by a fortunate chance. It so happens that in urban conditions both the time a journey takes and the fuel it uses are roughly inversely proportional to average journey speed. This ensures that measures to reduce congestion and to allow traffic to flow more freely are measures that will by and large save energy. Similarly improvements in public transport which encourage a move from private car to public transport are also potentially energy saving, though the cost benefit analysis might show that energy appears as only a small part of the total benefits. In these two important areas, it is helpful that the policies which have been pursued in the past, to reduce road congestion and encourage public transport, are policies which could make an important contribution to energy conservation even though they were not originally conceived for this purpose. It is only fair to say however that in other cases a more difficult balance has to be struck between energy conservation and other factors such as safety and environmental standards where desirable goals are not always compatible.

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* See for example, Confernece on Energy Accountancy, The Institution of Mechanical Engineers, London, December 1976.

The possibilities for conservation of energy by policies applied to the operation of transport, and by changes to the design and use of vehicles, cover a very wide range. Two examples have been selected to illustrate ways in which conservation measures can be evaluated; one is based on changes to vehicle design, and the other is based on alterations to traffic management strategy.

In order to investigate the potential for energy conservation in vehicle design, TRRL has developed computer simulation programs to model car and lorry performance, with a particular emphasis on fuel economy. This computer simulation work is supported by extensive road testing in normal traffic conditions with vehicles instrumented to measure instantaneous speed, fuel consumption and torque in the propeller shaft.*

In the car simulation, driver behaviour is a central feature, because the interaction between driver and car is one of the principal factors affecting fuel consumption. The model simulates journeys in which the car is driven over a predetermined route defined in terms of speed limits, bends, hills and intersections or other traffic hazards. The driver is modelled in terms of minimum and maximum engine speeds, determination, anticipation and gear change time. By incorporating a driver it permits a more realistic evaluation than might be obtained with a simple driving cycle simulation with its fixed speed-time profile and fixed gear change points; it also allows changes in driving technique to be evaluated. The data in

* EASINGWOOD-WILSON, D., NOWOTTNY, P M., PEARCE, T C. An Instrumented Car to Analyse Energy Consumption on the Road. Transport and Road Research Laboratory Supplementary Report, to be published.

Table 1 illustrates the use of the simulation to investigate the effect on fuel consumption of changes in overall gearing and driving cycle specification.

TABLE 1

THE EFFECT OF DRIVER BEHAVIOUR AND FINAL DRIVE GEAR RATIO ON FUEL CONSUMPTION

(Figures show the fuel consumed per unit distance driven relative to the standard case)

Final Drive Ratio / Driving Mode*	Normal (3.9 to 1 gearing)	20% higher gearing (3.25 to 1)	40% higher gearing (2.79 to 1)
1	100.0	93.1	91.5
2	90.2	92.1	93.0
3	95.6	87.3	89.4

* Driving modes:

1. ECE15 cycle with fixed speed time profile, fixed gearchange points.
2. ECE15 cycle with gearchange at driver's discretion.
3. ECE15 cycle maximum speeds specified, all else at the driver's discretion.

The different results obtained under the three driving cycle modes, with their different levels of constraint show the importance of evaluating a fuel economy measure such as a change in overall gearing under realistic conditions. The differing results highlight the possible shortcomings of dynamometer driving cycles as the only method of assessing fuel economy measures.

Instrumented vehicles have been used to help to calibrate and validate computer simulations of the kind described above. They can also be used to evaluate a range of conservation measures which are not readily amenable to simulation. As an example, instrumented cars have been used extensively to measure the relative fuel economy of different traffic management schemes. In Glasgow, a city with traffic lights linked to a central computer, the normal strategy was to program the traffic lights to minimise total delay time. As an alternative, it was suggested that fuel consumption might be reduced by a strategy aimed at minimising the number of stops. An instrumented car was driven over representative routes under the minimum delay strategy and the experiment was then repeated with the minimum number of stops strategy in operation. Under the latter conditions the fuel consumption determined over many separate journeys averaged 6% less than with the minimum delay strategy and there was an increase in journey speed of less than ½%. These results indicated a worthwhile net benefit (about £50 K per annum) over central Glasgow as a whole through what was a very minor change in traffic management.

Many other operational measures and changes in vehicle design can be assessed by computer simulation combined with experiments using instrumented vehicles. As conservation measures are likely to be a continuing feature of the transport energy scene, the development of these research tools is seen as a valuable contribution to the assessment of a wide variety of policies.

3. ANALYSIS OF ELECTRIC AND LIQUID FUEL CARS

In the longer term, the major choice in transport fuels is likely to be between internal combustion engine vehicles using a synthetic fuel made from coal, and an electric vehicle using batteries. The analysis which follows considers a time when there is assumed to be no natural crude oil available for road transport. The year is taken to be 2025 when it is estimated that there are 25 million cars and light goods vehicles in the United Kingdom, each covering 20 000 km annually.

The comparison between liquid fuel and battery vehicles is not straight-forward since the low energy densities at present achievable in batteries mean that a substantial fraction of the traction work performed by an electric vehicle is used to move its own batteries around. To compare primary fuel requirements of gasoline (natural or synthetic) and battery cars on a rational basis, it is necessary to derive an expression for the overall efficiency of converting primary energy to payload kilometres.

For a car of weight M carrying payload M_p the vehicle efficiency η_v is given by:-

$$\eta_v = \frac{\eta_2}{1 - \beta\eta_3} \cdot \frac{M_p}{M} \qquad (1)$$

where η_2 is the efficiency of converting fuel to rotation of the car's wheels

η_3 is the efficiency of regenerating a proportion β of the energy at the wheels.

The overall efficiency of converting primary energy η is then:-

$$\eta = \eta_1 \eta_v \qquad (2)$$

where η_1 is the efficiency of converting primary energy to fuel.

Equations 1 and 2 may be rewritten in terms of installed range (R):-

$$\eta = \frac{\eta_1 \eta_2 M_p}{M_u + M_p} \left\{ \frac{1}{(1 - \beta\eta_3)} - \frac{Z\,R}{\eta_2 E} \right\} \qquad (3)$$

where Z is the energy required at the wheels per unit weight and per unit distance.

M_u is the unladen weight of the car, ie excluding payload and battery or gasoline weight.

E is the energy storage capacity of the battery or liquid fuel per unit weight.

Battery or fuel weight M_f is given by:-

$$M_f = \frac{M_p \eta_1 ZR}{\eta E} \qquad (4)$$

Equations 3 and 4 define the basic performance when the car carries its full rated payload. Normally a car will carry only a fraction α of its rated payload. Under these circumstances the range will increase from the rated value R to R_α but the net transport efficiency will decrease from its rated value η to η_α :-

$$R_{\alpha} = \frac{MR}{M - (1-\alpha)M_p} \tag{5}$$

$$\eta_{\alpha} = \frac{\eta_1 \eta_2 \alpha M_p}{M_u + \alpha M_p} \left\{ \frac{1}{1-\beta\eta_3} - \frac{Z R_{\alpha}}{\eta_2 E} \right\} \tag{6}$$

Equation 6 provides the basis on which to compare different cars with specified rated ranges and allows the comparisons to extend to operation at part load.

To compare the overall efficiency of gasoline and battery cars representative values may be assigned to each of the variables in the equations 1 to 6 as specified in Table 2. Values of η_1 are taken for a typical conventional refinery, a synthetic fuel plant making gasoline and diesel from coal, and electricity generation. Vehicle efficiency η_2 is taken as representative of the efficiencies achieved in the best present day petrol and battery vehicles. Regeneration efficiency η_3 is fixed by reference to recycling of energy through the battery. The energy required at the wheels, Z, and the fraction available for regeneration are based on unpublished measurements made under typical urban and suburban traffic conditions; these values are obviously dependent on traffic conditions. The value of energy storage capacity of gasoline makes no separate allowance for the tank and associated structure. The values chosen for the advanced lead acid battery 150 MJ/tonne (40 Wh/kg) and the sodium sulphur battery 540 MJ/tonne (150 Wh/kg) represent very good battery characteristics with, again, no separate allowance for support structure.

TABLE 2

CHARACTERISTICS OF LIQUID FUEL AND

BATTERY CARS DESIGNED FOR 150 km RANGE

		GASOLINE FROM:		BATTERIES:	
		Natural Crude Oil*	Coal	Advanced Lead-Acid	Sodium-Sulphur
Basic Parameters					
η_1		0.88	0.55	0.24	0.24
η_2		0.15(0.11)	0.15	0.70	0.70
η_3		-	-	0.5	0.5
Z	MJ/t.km	0.35(0.39)	0.35	0.35	0.35
β		-	-	0.25	0.25
E	MJ/t	47000	47000	150	540
Vehicle Weights					
M_p	kg	400	400	400	400
M_u	kg	700(800)	700	700	700
M_f	kg	8	8	856	152
M	kg	1108	1108	1956	1252
η_α (100 kg payload)		0.016	0.010	0.012	0.020

* Figures in brackets are for today's car.

Table 2 shows that a 4 passenger sodium-sulphur battery car designed for 150 km range has a better overall efficiency with a 100 kg payload ($1\frac{1}{2}$ passengers) than either of the equivalent gasoline cars or the lead acid car. (η_α in Table 2).

The synthetic gasoline car can match the conventional car in every respect except overall efficiency where the lower efficiency of making gasoline from coal leads to a 60 percent higher primary energy requirement. A car powered by a low energy density battery like lead acid will always have some difficulty meeting range requirements, and in striving for sufficient range the weight of batteries is likely to detract from the basic conversion efficiency advantage of electric traction. A high energy density battery like sodium-sulphur is better placed to capitalise on the inherent conversion efficiency advantage of electric cars; the weight of battery necessary to provide a good range capability is unlikely to be so great as to remove the basic efficiency advantage compared with liquid fuel cars.

So far, only the direct use of energy to propel the car has been considered. It is also necessary to take some account of the energy needed to manufacture the battery, which is an extra component in the electric car, and to heat the car, because the gasoline car uses waste heat from the engine. In Table 3 these factors are listed for the cars designed to use the four types of fuel.

The heating requirement was computed on the basis of the annual variation of temperature in the UK and allows for three complete air changes per minute. For the sodium-sulphur battery, which has an operating temperature of 300°C, it is more difficult to assess the additional heating requirement. It may be that sufficient waste heat will be available from the battery when it is operating. On the other hand some extra energy may be required to keep the battery at its operating temperature when not in use, especially if the vehicle is used infrequently. Table 3 therefore gives a range of energy requirements, for heating and heat loss.

TABLE 3

ANNUAL DIRECT AND INDIRECT ENERGY REQUIREMENTS OF LIQUID FUEL AND BATTERY CARS IN 2025 (mtce/yr.)

(25.10^6 cars each covering 20000 km per year)

	GASOLINE FROM:		BATTERIES	
	Natural Crude Oil	Coal	Advanced lead-acid	Sodium-sulphur
Direct energy	40	64	56	32
1 kW average heating requirement	-	-	7	0-7
Battery manufacture (3 year life)	-	-	11	5
Car manufacture (10 year life)	6	6	6	6
Total	46	70	80	44-51

Battery manufacturing energy was calculated from a published paper.* The way in which battery manufacturing energy has to be amortized depends on battery life. Although the whole battery range of 150 kilometres may not be used every day it was thought that the battery would probably be charged each day and would thus approach 1000 cycles in three years of use. Fig 2 compares the total primary energy requirement of the four vehicle types and

* WILLIAMS K R. Energy Analysis and the Electric Car, SAE Paper No 760120, 1976.

includes data on today's car for reference. It can be seen that the advanced lead acid battery car uses appreciably more energy than the gasoline from coal car. The sodium-sulphur car uses less energy than the gasoline from coal car and even when the battery life is reduced to one year the energy requirement is no more than for the liquid fuel car.

It may be argued that the assumed performance of the sodium-sulphur battery (540 MJ per tonne) may never be realised in practice, but even if the achieved energy density were as low as 300 MJ per tonne (80 Wh per kg) the battery car would still have a primary energy requirement no worse than for the synthetic gasoline car. On the other hand, the difference in primary energy requirement is more sensitive to changes in the efficiency of the synthetic gasoline car. The successful development of a high efficiency powerplant for small cars, perhaps a diesel engine, might increase the vehicle efficiency sufficiently to reduce the primary energy requirement to a level comparable with the basic sodium-sulphur car. An increase in efficiency from 0.15 up to 0.20 is all that is required, or less if the improvement in vehicle efficiency were matched by an improvement in the efficiency of making synthetic gasoline. The sensitivity of the computed energy requirements to these and other perturbations is illustrated in Fig. 3 which shows the effect of simultaneous variation of all the main independent variables. The energy requirements of all the vehicles are sensitive to overall efficiency, gross vehicle weight and energy requirement at the wheels. The energy requirement of the lead acid battery car is particularly sensitive to these variables and also design range and battery energy density.

This analysis of the energy requirements for cars would be incomplete without some reference to the effect of other demands for primary energy. Some of the demands from the non-transport sector (eg industrial and domestic) particularly for heating can be met in different ways so that substitution is possible between gas from coal and electricity. This leads to the concept of maximum demand for coal to meet total energy requirements as one limit, and maximum demand for electricity as another. Most, but not all, electricity is assumed to be provided by nuclear or renewable resources. Table 4 shows these two limiting cases based on an estimate of demand for primary energy in the United Kingdom in the year 2025*.

It can be seen that transport accounts for about a quarter of total primary energy demand, and about half of that demand is generated by cars and light goods vehicles. Most of the rest is accounted for by heavy long distance lorries and aircraft, which are likely to continue to be dependent on liquid fuel. Thus the direct effect on primary energy demand of choosing battery electric or liquid fuel cars is relatively small - and within the uncertainties of the assumptions. The differences between the coal-based supply setting and the electricity-based one is more significant, and much more difficult to predict, depending as it does on questions of coal mining technology, nuclear power generation and the development of solar and wave power resources.

It is perhaps fortunate that the choice of source of energy for cars and light goods vehicles is not critically dependent on which future supplies scenario is most likely. Table 4 has illustrated two limiting cases of

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* CHAPMAN P, CHARLESWORTH G, BAKER M. Future Transport Fuels. Transport and Road Research Laboratory, SR251, 1976.

TABLE 4

TOTAL PRIMARY ENERGY DEMAND IN 2025

(a) Maximum demand for coal

mtce

	Synthetic gasoline	Advanced lead-acid battery	Sodium-sulphur battery
Transport Coal	150	80	80
Total Coal (inc 40 mtce electricity)	384	314	314
Transport electricity	7	87	58
Total electricity (ex 40 mtce from coal)	188	268	239
Total energy	572	582	553

(b) Maximum demand for electricity

mtce

	Synthetic gasoline	Advanced lead-acid battery	Sodium-sulphur battery
Transport Coal	150	80	80
Total Coal (inc 40 mtce electricity)	248	173	173
Transport electricity	7	87	58
Total electricity (ex 40 mtce from coal)	409	497	468
Total energy	657	670	641

energy supply: there are of course numerous others which could have been chosen, but the main conclusions would remain unchanged.

4. OTHER FACTORS IN THE CHOICE OF FUTURE TRANSPORT FUELS

The previous section has dealt with a fairly simple analysis of the energy use of battery electric and liquid fuel cars. There are however many other factors which will have an influence on the choice of transport fuels, and which cannot be handled by these techniques. This does not mean that they are unimportant, and indeed it may be that some of them will dominate the actions of consumers and producers. The factors which are discussed briefly below are:-

- consumer attitudes
- choices made by other countries
- taxation policy
- environmental aspects.

The attitude of consumers to the choice between liquid fuel and battery electric vehicles is likely to be significant. There may well be inertia in consumer markets which would favour the continuation of conventional vehicles and thus favour the liquid fuel choice. It is also likely that battery electric vehicles will experience difficulty in matching the characteristics of the luxury high performance car market. As liquid fuel of some kind is likely to be available for long distance lorries and aviation, it seems at least possible that there will always be a small market for luxury high performance liquid fuel cars.

If there were a substantial difference in operating cost between the liquid fuel and battery electric vehicles, this would help to indicate which way consumer choice might fall. However some illustrative estimates of capital and annual running costs given in Table 5 show that there is no firmly based and substantial difference in operating costs between gasoline from coal cars and advanced battery cars. Unless such a difference emerges in the future, cost is unlikely to be a driving force in favour of electric cars. Moreover, until a high energy density battery is available with cost and life attributes rather better than were assumed in this report, cost will tend to favour the syncrude car.

The feature of electric cars which is most likely to influence consumer attitudes adversely is their limited range. The fact that the electric car cannot easily and quickly be refuelled without recourse to an expensive battery exchange station is likely to have an important effect on consumer preferences, and it might be thought that in an electric car scenario personal mobility would be quite severely curtailed. One way in which this problem might be solved is to have a fleet of liquid fuel cars available for long journeys. Only about 5 per cent of car journeys are likely to exceed the capabilities of a battery vehicle. If an electric car fleet were supplemented by an additional say 10 per cent of synthetic gasoline cars which could be hired for long journeys then the combined fleet would be capable of meeting all of the demands for car transport. The energy penalty associated with manufacture of these additional cars would add only 2 or 3 per cent to the total energy requirement for cars, and there would be a similar operating cost penalty. More important the fuel requirement would shift from all-electric to something like 1 part coal to 3 parts electric.

TABLE 5

CAPITAL AND ANNUAL RUNNING COSTS OF ELECTRIC AND LIQUID FUEL CARS

(1975 £, excluding tax)

	LIQUID FUEL FROM:		BATTERIES:	
	Natural crude oil	Coal	Advanced lead acid	Sodium-sulphur
CAPITAL COSTS				
Vehicle (2)	1600	1600	1600	1600
Battery (3)	-	-	706	451
Total	1600	1600	2306	2051
ANNUAL RUNNING COSTS (4)				
Fuel (5)	85	136	32	19
Maintenance (6)	100	100	50	50
Insurance (7)	90	90	130	114
Annual capital charge (8)				
Vehicle	253	253	253	253
Battery	-	-	272	174
Total annual costs	528	579	737	610

Notes:

1. Vehicles as specified in Table 2.

2. The capital cost of all vehicles is assumed to be equal.

3. All batteries cost £20 per kWh (£5.5 per MJ).

4. Annual running costs based on 20 000 km.

5. Present gasoline cost is 36p per gallon excluding taxes; gasoline from coal costs 58p per gallon (proportional to process efficiency). Electricity is supplied at the off-peak rate of 0.8p per kWh. No allowance for heating the vehicle.

6. Maintenance of battery vehicles is half maintenance of liquid fuel vehicles.

7. Insurance is proportional to capital cost.

8. Annual capital charge is based on a vehicle life of ten years and a battery life of three years; interest rate is 10 per cent.

Probably the single most important industrial factor in a choice between electric and conventional cars is the decision made in other countries, because vehicle manufacturers would prefer to produce vehicles which could be sold in world markets as well as in the UK. In the USA the existence of abundant coal, the history of large cars and the very long transport distances would favour conventional vehicles. In contrast, Japan has negligible fossil fuel resources, is developing a large nuclear electricity programme, and has a high population density and fewer long range journey requirements. These factors would favour electric vehicles. Providing that the major industrialised nations do not all choose the same option then the UK will be left with a real choice. Although there may be some justifiable reluctance on the part of manufacturers to change from conventional vehicles, the 50 years involved in making a change is much longer than the lifetime of existing capital facilities and should give adequate time for radical changes by vehicle manufacturers if they are seen to be necessary.

Government policy might well play an important part in influencing the future choice of transport fuels. Control of North Sea oil depletion rates and investment in the coal industry are key issues. If the depletion of indigenous oil resources is tightly controlled then oil could be made available for transport for an extra 10 to 15 years, thereby pushing back the time when a choice has to be made. Substantial increases in coal production or imports of coal are needed to make the domestic syncrude option possible. The development of the coal industry could be markedly influenced by the choice between gasification (which is associated with battery vehicles) and liquefaction. In a free market this issue is likely to be influenced by whether indigenous oil or gas supplies run short first.

At the moment in the UK, battery vehicles enjoy a tax advantage over conventional vehicles, largely because they use untaxed electricity, often at off-peak marginal cost, while motor spirit is highly taxed. (In 1975 the price of motor spirit was 36p untaxed, but 74p per gallon including duty and VAT). The market penetration of electric vehicles could be encouraged, if desired, by keeping electricity for transport free of tax. Alternatively, if it were policy to discourage the introduction of electric vehicles, an equivalent or higher "motor spirit" tax would be imposed on electricity for transport purposes. Thus taxation could be used to influence consumer choice in a way which appeared desirable for policy purposes.

Turning to environmental aspects, it is clear that, since battery vehicles have no exhaust gases and have quiet engines, the local environmental effects will be less than those caused by conventional internal combustion engined vehicles. This advantage in lower noise and emission may be offset because, by their quietness, they may be more likely to be involved in accidents than conventional vehicles: the measures needed for safety in the event of an accident with a high energy battery like sodium sulphur also pose new, though not necessarily insuperable, problems.

Finally, the widespread use of electric cars could in the long run affect travel patterns. While electric cars with advanced batteries should have sufficient range for most urban journeys, they are not likely to be suitable for long trips. There could be a requirement for a fleet of liquid fuel vehicles available for hire (as mentioned earlier) or there could be a need to link major centres of population by rail and have a fleet of electric cars available for hire to handle the local collection and distribution journeys.

It is difficult to predict whether such reductions in the ability of the private car to make long journeys will be acceptable, but if they are they may have long term effects on the location of residence and work place.

5. CONCLUSIONS

Road transport is an important user of imported oil and in the short term the dependence of road transport on oil is unlikely to be either easily or quickly altered. Fortunately the urban and suburban transport policies to encourage public transport and relieve congestion in cities which were being pursued in the United Kingdom before the 1973 oil embargo are also effective means of energy conservation. It is likely that in the medium and longer term, energy will remain an important factor in shaping transport policy. Research tools like the instrumented cars and lorries, supported by computer simulation, are a valuable means of evaluating ways of saving energy.

In the first quarter of the next century, indigenous supplies of oil from the North Sea are unlikely to be sufficient to meet demand, and it will be necessary to look for an alternative fuel for road transport. The strongest candidates have been identified as either a synthetic liquid fuel derived from coal, or electricity (derived from various sources) and advanced batteries. Cars with high energy density batteries could have a marginally better overall energy efficiency for transport as well as environmental advantages. There are however a number of other factors, for example consumer acceptance, which make the choice of fuel less clear cut.

The choice is likely to be greatly influenced by the availability of coal within the United Kingdom and on the world market. If coal production could be expanded or if imports of coal were available, then there would be an incentive to the United Kingdom to make the fullest possible use of it in producing both liquid fuel and gas from coal. It is worth noting that modest improvements, beyond those assumed in this paper, in the production of synthetic gasoline and in conventional vehicle efficiencies would enable such vehicles to match battery electric cars on primary energy grounds.

It is not possible yet to choose between the long term prospects for liquid fuel cars and those for battery cars. Neither alternative should be ruled out and a mixed vehicle fleet is by no means an unlikely outcome. What is needed at this stage is further assessment work to examine the key problem areas:

- the likely developments in conventional vehicles and the efficiency of producing synthetic liquid fuels.
- the possible further expansion of planned UK coal production.
- the prospects of a world coal or syncrude market beyond say 1990.
- the effect of battery cars and light goods vehicles on mobility of people and freight and possibilities for optimising the transport system around battery vehicles or a mixed fleet of conventional and battery vehicles.
- the likely availability of off-peak electricity.
- the overseas market for electric vehicles.

This assessment work will have to be supported at appropriate times by R and D on coal mining and conversion technology and advanced batteries. Some of the assessment work and the R and D is, of course, already in progress.

6. ACKNOWLEDGEMENTS

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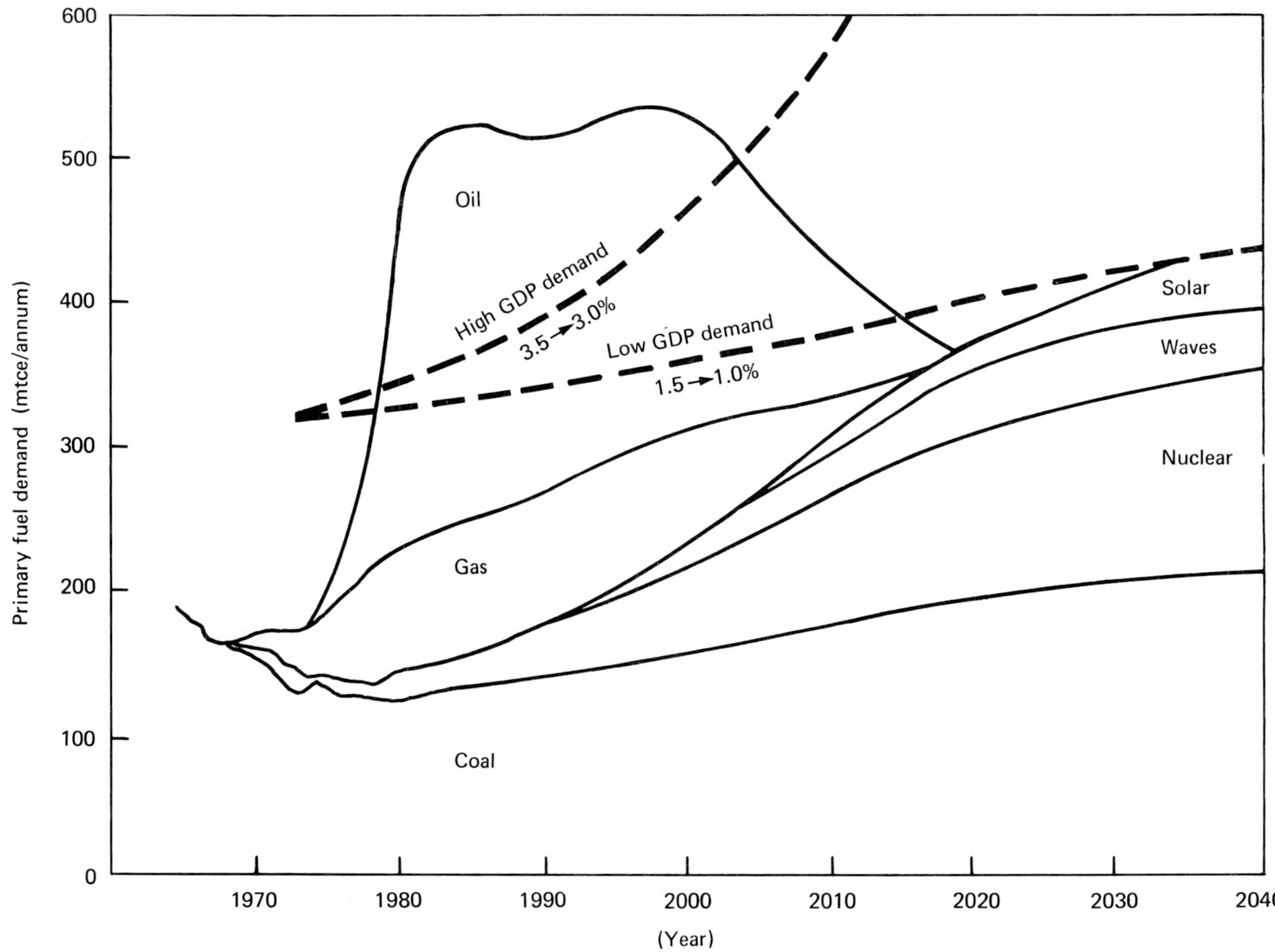

Fig. 1 A COMPARISON OF ENERGY SUPPLY AND DEMAND

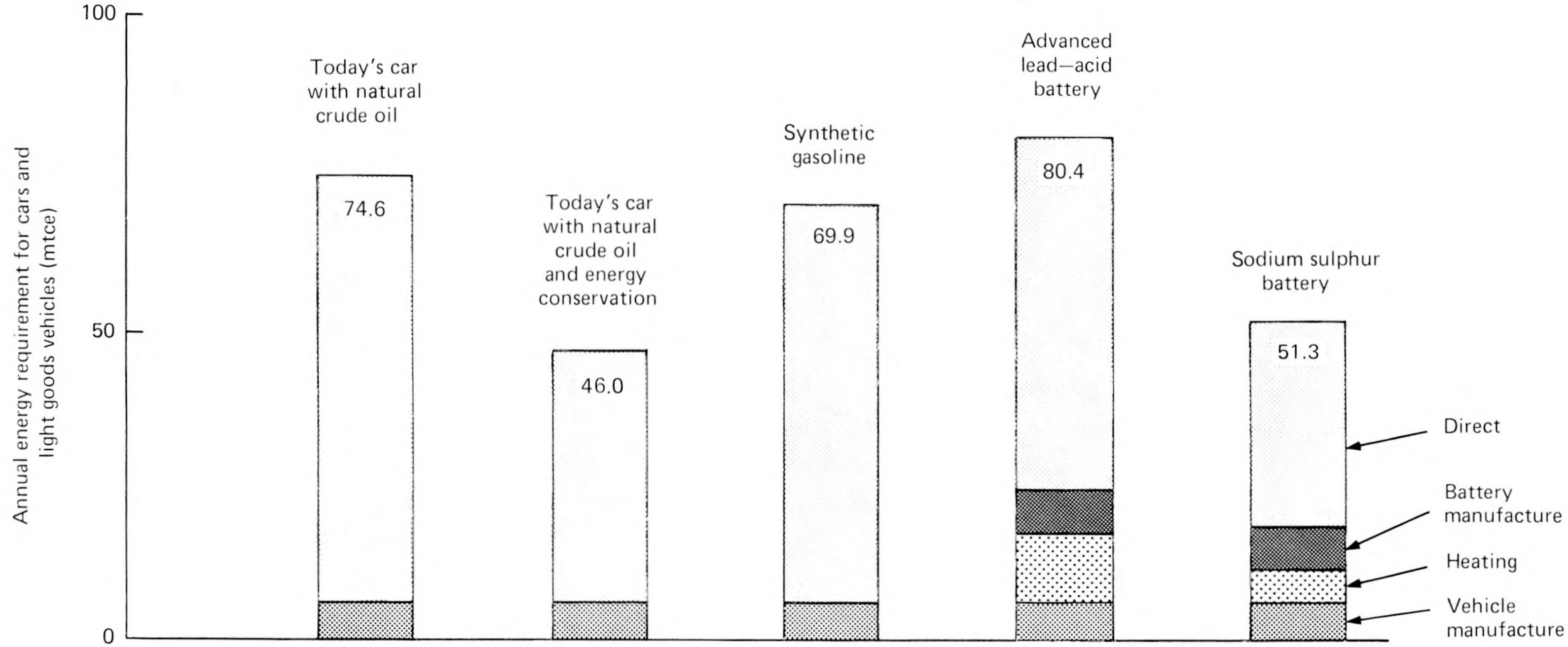

Fig. 2 ANNUAL ENERGY REQUIREMENT FOR CARS AND LIGHT GOODS VEHICLES IN 2025

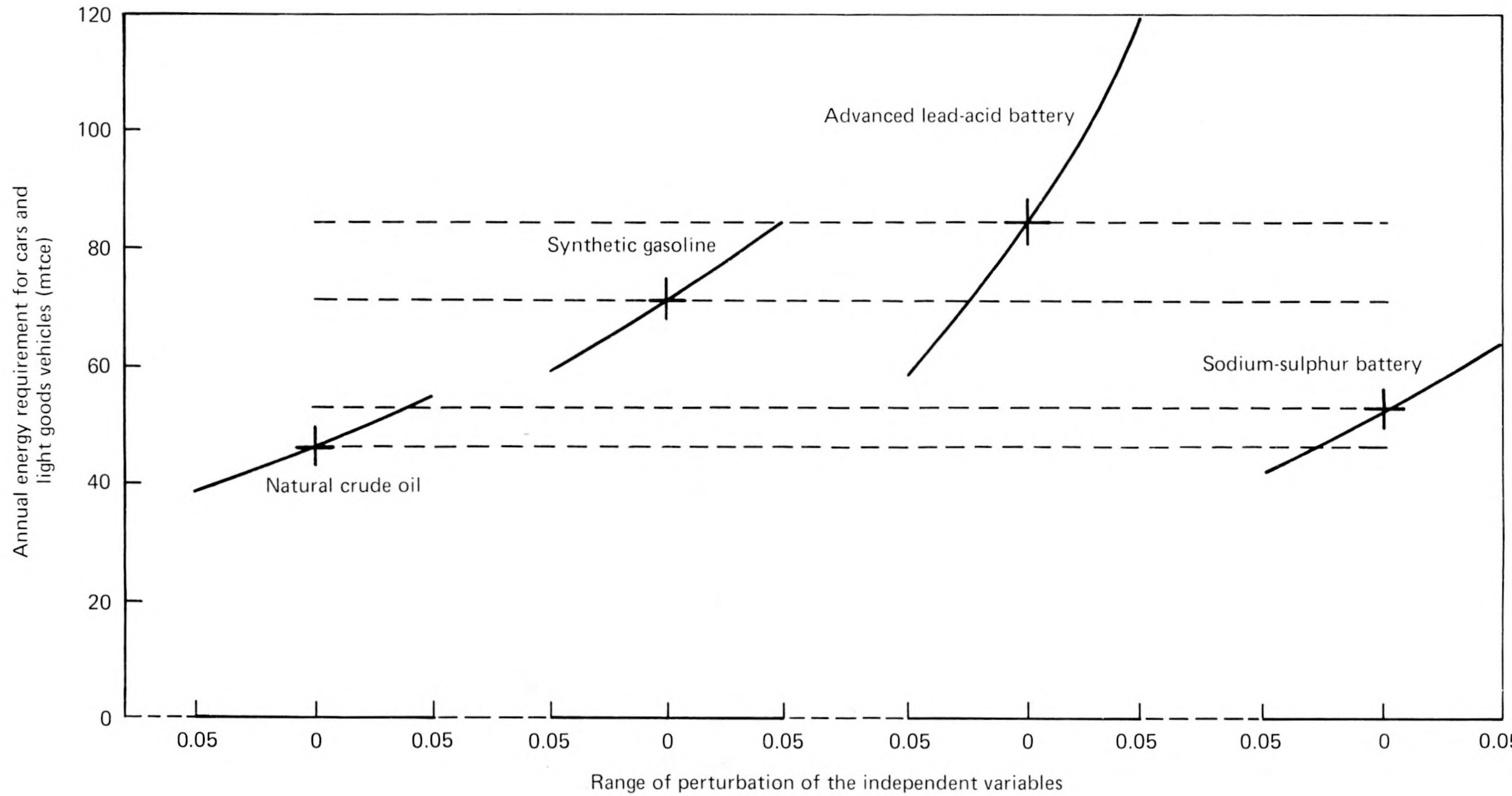

Fig. 3 THE EFFECT OF SIMULTANEOUS PERTURBATION OF THE MAIN INDEPENDENT VARIABLES WHICH SPECIFY THE ENERGY REQUIREMENT FOR CARS AND LIGHT GOODS VEHICLES

A Systems-Analysis of Alternative Drive Units for Future Heavy Duty Trucks

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ABSTRACT

Until recently the main criteria for the selection of automotive power units was operational economy. The shortage of resources and the increasing environmental pollution make it necessary to also consider viewpoints of political economy and ecology. It is necessary to balance the significance of the requirements, which are often contradictory. For the examination of different alternative power systems a method of systems-analysis has been developed, employing conventional techniques. Thereby, the requirements of the particular field of application and the characteristics of the possible alternatives are combined to establish a basis for decision. Future changes in the legal requirements and in the market situation are being taken into consideration as well as the advances in technology.
The systems-analytical method will be described.
As an example, an investigation based on this method, carried out for a heavy duty truck will also be discussed. The alternative drive units are :

- a conventional diesel engine predominantely employed in heavy duty trucks
- an unconventional automotive gas turbine

The results of this investigation will be presented and discussed in detail.

1. Introduction

The efforts to achieve better economy and the growing awareness of environmental problems resulted in more rigorous requirements to be met by technical products.

Therefore it is not surprising that the search for new technologies and their future application nowadays covers a wide range in research and development. In the line of internal combustion engines, these endeavours are aimed at less pollutant and less noisy engines of identical or, preferably, even better economy.

However, the financial resources for research and development for coping with these problems are limited by economical requirements. Therefore, the financial resources must be concentrated on developments with high probability of success. It will, therefore, be necessary, first to determine all alternative drive units that seem suitable for realizing the target set and then select those which are especially promising with regard to maximum value and have a good chance for realization. Further assessments made during the different stages of develoment work should guarantee the control of a success oriented research and development effort.

2. Method

Since the growing development of systems engineering during the second half of this century (1), a multitude of systems engineering approaches have been made and applied practically in various fields serving support and security of society (2, 3). In the area of drive units, too, systems engineering approaches are required because of the complex problems of decision making. With the increase of technical know how the number of possible variants and, therefore, the number of alternatives grow (4). The complex structure of the requirements to be met as well as the early stage of development of many alternatives further complicate the selection of the drive unit which will most satisfactorily meet future demands. However, in view of limited financial funds and staff available, this decision must be made as early as possible to minimize the expenditure on research and development in the case of less promising projects (5, 6, 7).

Fig. 1 shows a flow chart of the systems-analytical method. Starting from the field of application and from the alternatives, the system is defined which forms the framework for the investigation. The result of assessment is obtained by connecting the requirements of the field of application with the characteristics of the alternatives.

This result (fig. 2) contains comparative and quantitative statements on

- economy. The basis is a cost/benefit analysis. Benefit and relative costs are quoted separately. The significant value is arrived at by dividing benefit, by cost and is called product value.
- prospects of feasibility. The result is a measure for the probability with which the minimum requirements will be met.
- research and development expenditure. Anual expenditures are quoted as well as the total sum.

The following input data are considered in the analysis :

- the restrictions and the weights of the field of application with due consideration of the different viewpoints of manufacturers, government authorities and users
- the basic costs, i. e. the expenses for manufacture and operation of comparable conventional units
- the characteristics of the alternative drive units and
- the anticipated costs of the alternatives (manufacturing and operating expenses)

All input data and, consequently, the results of the decision basis are time - dependent functions because of the fact that the requirements change in the course of time and as a result of technical advance.

Any product to be analyzed must be viewed within a large system with many mutual interdependencies.
Fig. 3 is a schematic representation of the system of drive units within the environment to be considered. The sections shown are of different degree of importance which may also vary with time.

For the complex range of automotive propulsion systems a system of objectives has been established which makes possible an assessment of alternative drive units with regard to technical and ecological aspects. The engine alone is considered within the framework of a system where engine, gear box, power train, exhaust system, etc. are separate sub systems. This allows a statement of the extent to which an alternative engine will meet the specific requirements independent of other sub systems (for example the gear box). However, this limited definition does not include the effect of the engine on the other sub systems and on the vehicle.

The system of objectives shown in fig. 4 has been created for evaluating drive units. It is organized in the form of a hierarchy and takes into account the following requirements :

- the system of objectives shall be applicable for all combustion engines of the complete morphological system of drive units
- the system of objectives shall be applicable to all fields of application
- the objectives shall be definable by physical parameters
- the objectives shall be independent of each other.

Since the system of objectives shall be universally applicable it was prepared by taking a general view to the power units. Emphasis was not placed on the kind of power plant but on the result of energy conversion. This means that specific engine data, such as mean effective pressure, compression ratio, etc. are not entered into the analysis since they are irrelevant for the user. The objectives in the corresponding last hierarchical level describe the drive characteristics of interest for the user using physical parameters.

In order to enable a quantitative assessment, two limits have to be defined for these objectives in compliance with the requirements of the specific duty :

- limit of permissibility, indicating which value has to be attained in view of legislation (for example exhaust emission), design (for example dimension of installation), specifications, etc.
- limit of practicability, indicating the value with due consideration of physical, technical and economic aspects.

Specific fuel consumption is used as an example for explaining these limits (fig. 5). In accordance with the mechanical heat equivalent, the energy contained in 85 grams of diesel fuel corresponds to the work of 1 kWh with an overall efficiency of 1. This would be the maximum output achievable by a combustion engine (point P_1). The overall efficiency of combustion engines build today ranges from η = 0,3 to η = 0,45 which corresponds to a specific fuel consumption of

b = 285 g/kWh and b = 190 g/kWh (point P_2, P_3).
No legislative law for the maximum fuel consumption exists today. Therefore, the limit of permissibility has been fixed of about 10 % above the highest fuel consumption assumed in this example (point P_4). Because of physical principles the physical limit of 85 g/kWh cannot be reached by combustion engines. Therefore, technical and economical aspects are decisive for the limit of practicability. In this case the limit of practicability is fixed at about 15 % lower than the fuel consumption of the best engine considered.
Since influences of technical developments as well as future legislative measures are to be considered in this systems-analysis, the restrictions will have to be specified as functions of time because of possible changes of requirements.

In addition to these limits another criterion : the objective success rating for a quantitative assessment is necessary. The objective success rating is the function between the limits of permissibility and practicability. The course of the function for defining the objective success rating may vary for the different objectives and requirements. The functions may be represented by curves of different shape, for example linear type, s-type, as well as saturation type.

The objectives of the system have to be weighted against each other in accordance with their importance. This must be done in all three hierarchical levels. Time-dependent weighting functions are used for this classification.

Just like the restrictions, the characteristics of the different alternatives have to be established by the manufacturers in form of time-dependent functions. These characteristic functions describe the operational properties of the alternatives and depend on the research and development activities envisaged for future developments. They can be established only in form of scattering ranges which diverge, the more the forecast points into the future and the more conventional the technology is that has been used.

The benefit (B) of the alternatives is calculated from the restrictions (PE, PR), weightings (W) and characteristics (C). For this analysis, only the ecological-technical benefit is used.

$$B\ (t) = \sum \frac{PE_i\ (t) - C_i\ (t)}{PE_i\ (t) - PR_i\ (t)} \cdot \overline{W_i\ (t)}$$

The product value is then calculated by dividing the ecological-technical benefit by the costs. The costs include manufacturing and operational expenses and are related to the service life of the alternative.
To take into account violations of the limits of permissibility, if any, the prospects of feasibility will be calculated. The product value, combining benefit and costs, the prospects of feasibility, and the research and development expenditure, all time-dependent functions, together form the decision basis (fig. 2).

3. Comparative assessment : diesel engine - gas turbine

Using the method described above, a standard automotive diesel engine was compared with a vehicular gas turbine designed for a long distance truck. The assessment is made for the period from 1970 - 1990. Taking into account trends of recent years a forecast was made for the next 15 years.

3.1. Field of application

A European truck train is taken as an example. It's total weight equals 38 tons. According to the legislation valid in the Federal Republic of Germany from 1971 - 1976, a minimum power of 8 HP/ton was required. Therefore, the drive power was chosen to be 304 HP = 224 kW, although the Federal Republic of Germany since April 1976 has adopted a minimum engine rating of 6 HP/ton just like the other West European countries.

The six-wheel cab over engine vehicle has a total weight of 22 tons at a pay load of 13 tons. Maximum speed is 95 km/h. The engine is installed underneath the seat.

Manual transmission, as today commonly applied has been chosen. The gear steps have been matched to the specific engine.

Legislation exists for :

1. minimum drive power
2. brake power of the engine
3. exhaust emissions
4. noise emission

A data bank, which included the restrictions valid for the defined field of application (such as limits of permissibility and practicability) was not available.

Therefore, the restrictions have been ascertained by consulting specialists, applying the Delphi-method. Experts of science and legislation, manufacturers, and users were asked to assess the present situation and to give a forecast for the next 15 years. In general, the replies given for the limits of permissibility agree well. The replies about the targets to be aimed at (the limits of practicability) revealed differing opinions among the consulted groups.

The analysis was carried out using the average of the different replies.

The weighting of the objectives for the heavy truck shown in fig 6. was determined by the Delphi-method, too. Specialists agree that the impotance of the ecological aspects will steadily grow and the importance of raw-material preservation will grow disproportionately from 1980 onward. The tendency of the different statements is basically identical within a scattering range of ± 15 % except for environmental protection. This fact reveals a certain uncertainty with respect to the assessment of future legislation in the line of exhaust and noise emission.

Seen from a technical point of view, availibility and - subordinate to it - reliability rank first. In accordance with a customer's poll organized by a Swedish truck manufacturer, operational safety - reliability - heads the criteria for the choice of an engine, followed by power rating and flexibility, fuel consumption, service life and maintenance.

In this systems-analysis environmental protection, exhaust and noise emission are considered to be equally important.

3.2. Alternatives

The following drive units have been examined as alternatives and their suitability for driving heavy duty trucks has been compared :

1. A mass-produced, air cooled 4-stroke diesel engine of the KHD family FL 413

Technical data

number of cylinders	12	
power rating according to DIN	224 kW	
rated speed	2,650 rpm	
overall length	1,576 mm	without exhaust gas system, including cooling system
overall width	1,022 mm	
overall height	942 mm	
weight	1,090 kg	

The characteristic functions have been established for the standard engine. Future legislation for exhaust emission and noise as well as the anticipated technical requirements of the market and competition will necessitate a continued development in the design of the engine. The corresponding improvements have been considered in the forecast.

2. A vehicular gas turbine (designed as a two-spool engine) with heat exchanger (fig. 7)

Technical data

power rating according to DIN	250 kW	
rated speed	2,600 rpm	
length	1,250 mm	without exhaust gas system
width	700 mm	without exhaust gas system
height	900 mm	without exhaust gas system
weight	850 kg	without exhaust gas system

The characteristic functions have been forecast in compliance with the present know how and based on calculations, project studies as well as studies of the technical literature, considering additional success in development and requirements to be met by truck drive units until 1990. Some of the data have been quoted as scattering ranges of the uncertain technical development.

3.3. Results

As mentioned above the restrictions and weighting functions were ascertained by an expert poll. Applying them to the forecast characteristic functions, diesel engine and gas turbine have been evaluated by the method described with the aid of electronic data processing. The result of the analysis is the decision basis shown in fig. 8. The calculated benefit of the gas turbine exceeds that of the production diesel engine. This indicates that the gas turbine has properties that are advantageous for a vehicular drive unit. The scattering ranges of the ecological-technical benefit overlap in wide areas, especially for the statements made about the future. Both ranges show a downward trend which is due to the predicted severity of requirements, particularly for environmental protection. This reduction in benefit means that - although diesel engine and gas turbine will in general meet future requirements - some objectives may come very close to the limit of permissibility. For this reason legislative authorities have to regard very seriously the technical possibilities and the economic potential, when intensifying existing environmental laws or issuing new regulations. In order to examine more closely the reason why the ecological-technical benefit of the gas turbine is somewhat larger than of the diesel engine the differences in the partial benefits are shown in fig. 9. The diagram does not show how well each alternative will meet the specific requirements, but only the magnitude of the differences between the two alternatives. Considering only those objectives where the differences in the partial benefits exceeds 0,5 % only 15 decisive objectives remain out of the total of 45 objectives included in the investigation. In some of these the diesel engine is advantageous, in others the gas turbine (table 1).

3.3.1. Discussion of the main decisive objectives

Suitability for mass production

The principle of modular construction applied to piston engines, which allows a wide power range to be covered by varying the number of cylinders, cannot be realized for gas turbines. A certain power range can be achieved with gas turbines too, by way of a special modular design, but this power range will be more limited than in the case of diesel engine.

Oil (fuel consumption)

The generally high fuel consumption of the gas turbine, particularly in the lower power range, is one of the main reasons for the slow progress in the introduction of the automotive gas turbine. The results of this analysis which is based on a rating of appr. 240 kW (8 HP/ton minimum) for a heavy duty truck shows that the use of gas turbines made with present technology would imply an increase in fuel consumption. In higher power classes and thus with larger vehicle weight, too, the fuel consumption of the gas turbine improves and comes close to that or even better than the fuel consumption of the diesel engine.

Time performance

The two-spool gas turbine investigated here has a disadvantage when the vehicle is accelerated from zero speed. The gas generator turbine first has to be brought up to the required speed before sufficient power for acceleration is available. This results in a time lag while on the other hand the power of a diesel engine can be used immediately. The behaviour of the two spool-turbine can be improved by maintaining the gas generating unit at high speed which in turn will increase fuel consumption.

Power loss

The gas turbine rating is directly dependent on the mass flow, and therefore, on the density of the inlet air. The power loss can be limited by using a turbine with flat rating. This, however, has disadvantages with regard to other objectives, such as greater weight and volume. Higher ambient temperature and operation at higher geographic levels and thus lower ambient pressure will reduce the power of the gas turbine relative to the operation in standard conditions. This power loss will be more significant than in the case of the diesel engine.

Materials

With respect to ecology, the gas turbine shows a disadvantage in the need for scarce raw materials. High temperature of certain parts require high alloy metals. Thereby, large quantities of alloying constituents, such as cobalt, nickel and tungsten are needed. The use of gas turbines in quantities that correspond to those of reciprocating piston engines may therefore result in a bottleneck regarding the availibility of these raw materials.

Noise

In the experts' poll, ecological objectives and especially noise have received a high weighting. Therefore, the lower noise level expected from the gas turbine has a decisive influence on the result. An essential reduction of the noise level of diesel engines will hardly be possible without an influence on other objectives, for example, power-to-volume ratio. Solutions such as the enclosure of engines, increase weight, volume and costs and complicate maintenance. However, alternation of intake and exhaust noise may be more difficult for the gas turbine, as the air flow rate is several times as great as comparable diesel engines.

Flexibility

The torque at 60 - 70 % of full speed is 10 - 20 % higher than the torque at full speed for todays vehicular diesel engines. Theoretically, the stalled torque of gas turbines is about twice the torque at full speed. Gas turbines, therefore, enable a 100 % torque rise for speed range of about 100 %. In view of these torque characteristic transmissions with a low number of gear steps are considered adequate for gas turbines. This possibility will have to be proved practically.

Maintenance

An essential advantage of the gas turbine is that it requires little maintenance during a long service life. A time between overhauls(TBO) of more than 500,000 km is considered possible as well as maintenance intervals of 30,000 km to 50,000 km. This in combination with a low maintenance vehicle leads one to accept significant economic advantages since downtime is minimized.

Multifuel capability

The gas turbine allows to use a range of fuels with only small adaptions. Because of the existing infra-structure in West European countries there is no bottleneck in the supply of any sort of fuel at the moment. The advantage of multifuel capability of the gas turbine, therefore, cannot be utilized at the moment but could be interesting in case of future fuel shortage.

Pollutants

The continuous combustion applied in gas turbines allows adjustments for low emission. The very stringent legislative limits anticipated for the eighties, therefore, can be achieved more easily with the gas turbine than with the diesel engine. For diesel engines such low emission levels will cause increased cost and fuel consumption and lower output.

Weight per power ratio, volume per power ratio.

Important features of the gas turbine are the low weight per power ratio and the compact design. It must be noted that with any gain in weight of the drive unit an additional corresponding weight reduction of the vehicular structure can be realized. The total weight will either reduce fuel consumption or can be utilized for higher payload, both will result in better economy.

Engine braking power

Since the gas turbine considered is fitted with a power feed back the engine braking power is higher than that of the diesel engine. However, this is only a small advantage because the natural aspirated diesel engine meets the requirements well.

3.3.2. Other criteria in the decision basis

The greater ecological and technical benefit of the gas turbine has to be seen against high costs (fig. 9). The manufacturing cost for the gas turbine and the diesel engine have been assumed to be of the same level in view of future mass-production. The maintenance expenditure for the gas turbine is less than for the diesel engine, provided a corresponding infra-structure exists. The main portion of the costs is accounted for by the fuel. Depending on the type of engine fuel cost amount to 75 - 80 % of the total costs during service life. Because of the greater fuel consumption of the 240 kW gas turbine, total cost are correspondingly higher than for the diesel engine.

Consequently, this results in an identical product value for the two alternatives which means that the greater benefit of the gas turbine is offset by correspondingly higher costs.
The prospects of feasibility of the diesel engine are, of course, 100 % today, but will decline in the eighties because of the more stringent requirements anticipated, especially, with regard to the use of scarce raw materials (2). The gas turbine fails to meet the minimum requirements for various objectives, such as time performance (acceleration), power loss (susceptibility to altitude), suitability for mass production and the consumption of scarce raw materials.
In order to meet the minimum requirements, technical improvements are conceivable but will adversely effect the other objectives. Thereby, a vehicular gas turbine can be realized already today, but only with a somewhat reduced benefit and higher costs.
For gas turbines of higher output the technical features can be proved to be superior to a conventional diesel engine at an identical cost level.

From present know how, the vehicular gas turbine of higher power ratings is technically and economically superior to the conventional diesel engine. Those ratings are larger than the requirements laid down for heavy duty trucks in European countries. Unless significant technical advances are made that enable an improvement in the economy of gas turbines, the installation of gas turbines in an appreciable number of vehicles will be unlikely in the course of the next decade.

In the US, where higher power ratings are usual, where greater distances have to be covered at constant speed, the use of vehicular gas turbines is conceivable for specific applications already today without neglecting the economic aspects of the subject.

4. Conclusion

A method of systems-analysis for the investigation of alternative drive units has been developed to provide an unbiosed basis for ascertaining the best drive unit for a special field of application.
Thereby, a hierarchically organized system of objectives has been established which enables a critical examination of alternative drive units with due regard to technical and ecological aspects to be made. The requirements of application and the characteristics of the alternatives are connected, utilizing the electronic data processing. The result is a decision basis including a comparative and quantitative statement on the alternatives examined with regard to :

- economy (product value) considering separately ecological-technical benefit and relative costs
- prospects of feasibility
- research and development expenditure.

Using this method, an investigation for a European heavy duty truck of 38 tons has been carried out.

The alternative drive units were :
- a series-produced automotive diesel engine
- a vehicular gas turbine

Restrictions and weighting functions have been ascertained by using the Delphi-method. Experts in science and legislation, manufacturers and users were asked to assess the present situation and to give a forecast for the next 15 years.

Characteristic functions for the diesel engine have been ascertained on the basis of a mass-produced engine.
The characteristic functions for the gas turbine have been forecast in compliance with the present know how and based on calculations, project studies and the technical literature.

This systems-analytical investigation results in an ecological-technical benefit for the gas turbine that exceeds the one attained by the diesel engine. However, the diesel engine has lower total costs. Consequently, the product value for both alternatives is the same.
However, this result cannot be transferred to other applications, different restrictions and weightings may cause significant changes.
Systems-analytical investigations using the method described, result in a quantitative comparison of alternative drive units and an unbiosed statement of the advantages and disadvantages in the particular objectives. The utilization of this systems-analytical method during the different stages of the development work allows an early modification of the development program to be made and, therefore, it will guarantee a control of a success oriented research and development effort.

ACKNOWLEDGEMENTS

The work described herein has been carried out in co-operation with the Volkswagenwerk AG as a research program sponsored by the Ministry of Research and Technology (Bundesminister für Forschung und Technologie) of the Federal Republic of Germany.
The German Institution for Technical Supervision (Technischer Überwachungsverein TÜV) has been acting as a project advisor for this program.
The author would like to express his due thanks to all persons involved.

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Captions of Illustrations and Tables

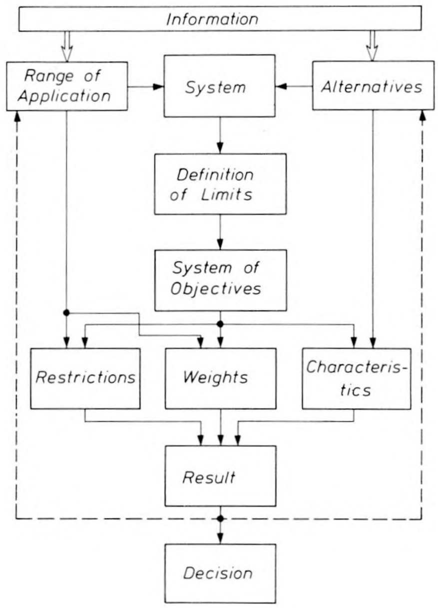

Fig. 1 : Systems-Analysis Flow Chart

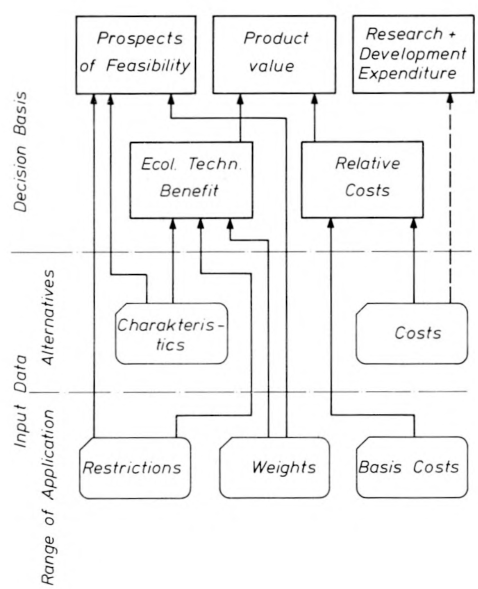

Fig. 2 : Logic of the Systems-Analytical Model

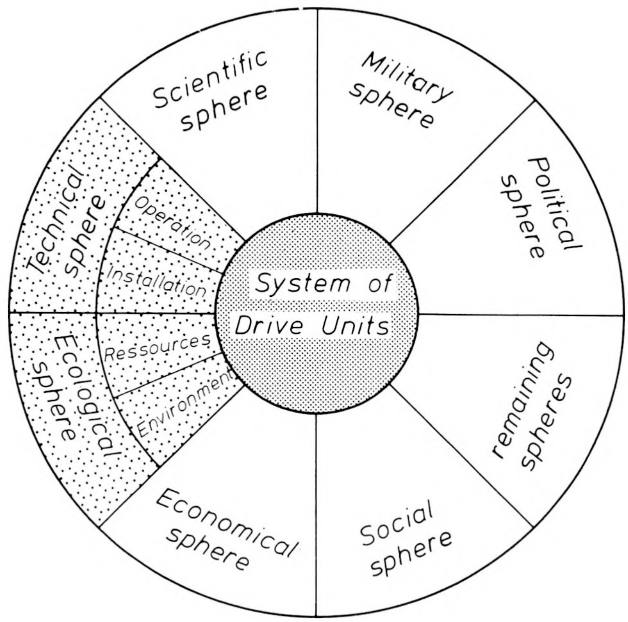

Fig. 3 : System of Drive Units and Important Spheres

1000 Function Quality	1100 Control Requirements	1101 Flexibility
		1102 Time Performance
		1103 Engine Braking Power
	1200 Availability	1201 Reliability
		1202 Maintenance
		1203 Service Life
		1204 Operating Range
	1300 Performance	1301 Startability
		1302 Power Decrease
		1303 Energy Available for Heating
	1400 Vibration	1401 Uniformity
		1402 Mass Balancing
	1500 Fuel Requirements	1501 Multi-Fuel Capability
		1502 Operating Danger
2000 Installation Suitability	2100 Dimensions	2101 Length
		2102 Width
		2103 Height
	2200 Volume per Power Ratio	
	2300 Weight per Power Ratio	
	2400 Suitability for Mass Production	
3000 Raw Material Conservation	3100 Energy Consumption	3101 Oil
		3102 Natural Gas
		3103 Coal
	3200 Material	3201 Aluminium
		3202 Chromium
		3203 Cobalt
		3204 Copper
		3205 Iron
		3206 Lead
		3207 Manganese
		3208 Molybdenum
		3209 Nickel
		3210 Platinum Group
		3211 Magnesium
		3212 Tungsten
		3213 Titan
		3214 Vanadium
		3215 Zinc
		3216 Tin
4000 Environmental Protection	4100 Pollutants	4101 CO
		4102 NO_x
		4103 $C_n H_m$
		4104 Particulates
	4200 Noise	
	4300 Waste	

Fig. 4 : System of Objectives

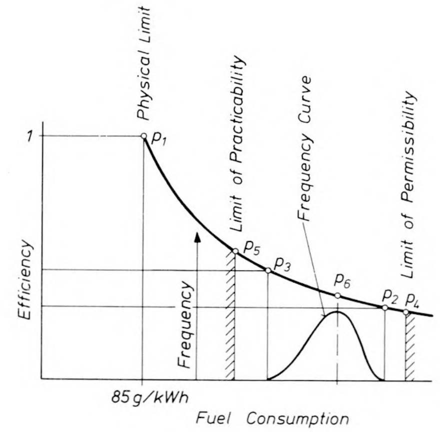

Fig. 5 : Objective Oil : Example for the Determination of Restrictions

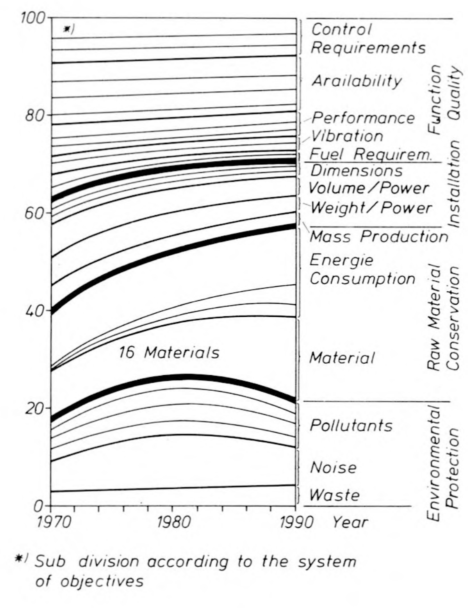

Fig. 6 : Weighting Functions for Heavy Duty Trucks

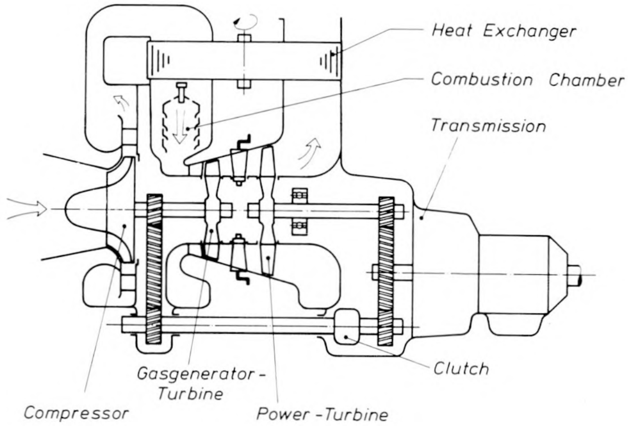

Fig. 7 : Automotive Gas Turbine
(Schematic)

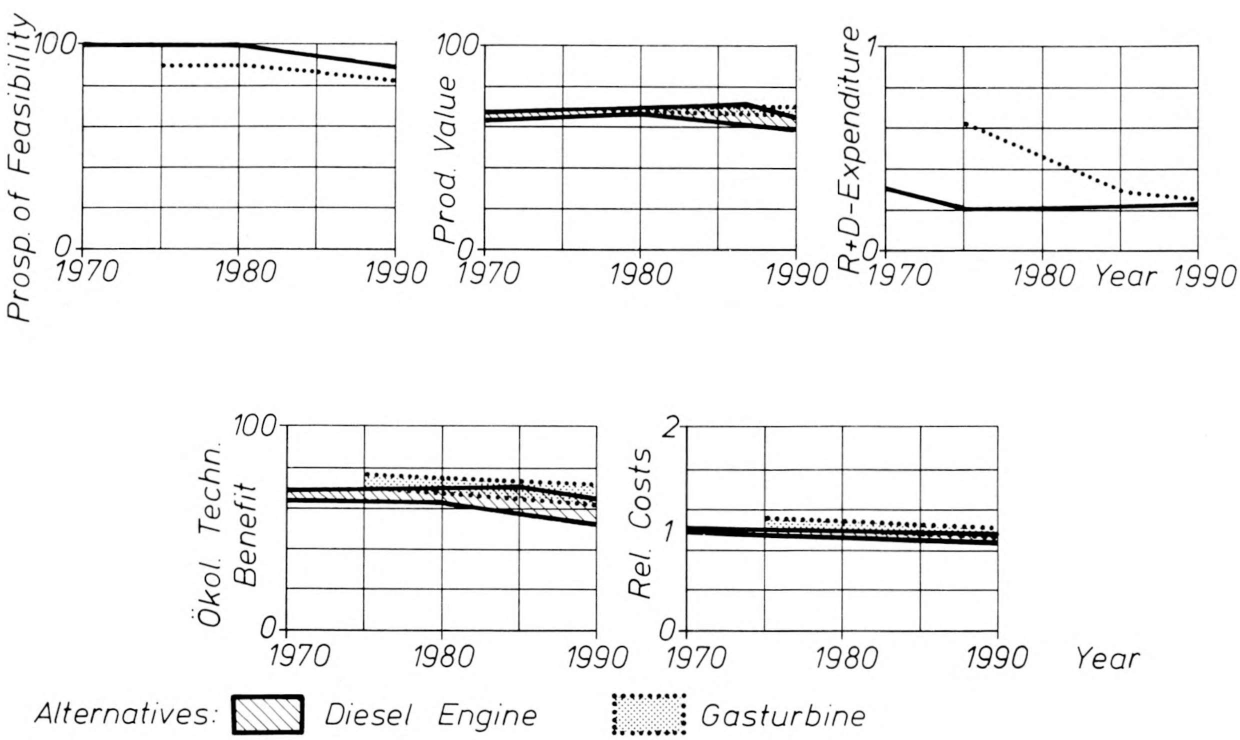

Fig. 8 : Decision Basis Heavy Duty Truck

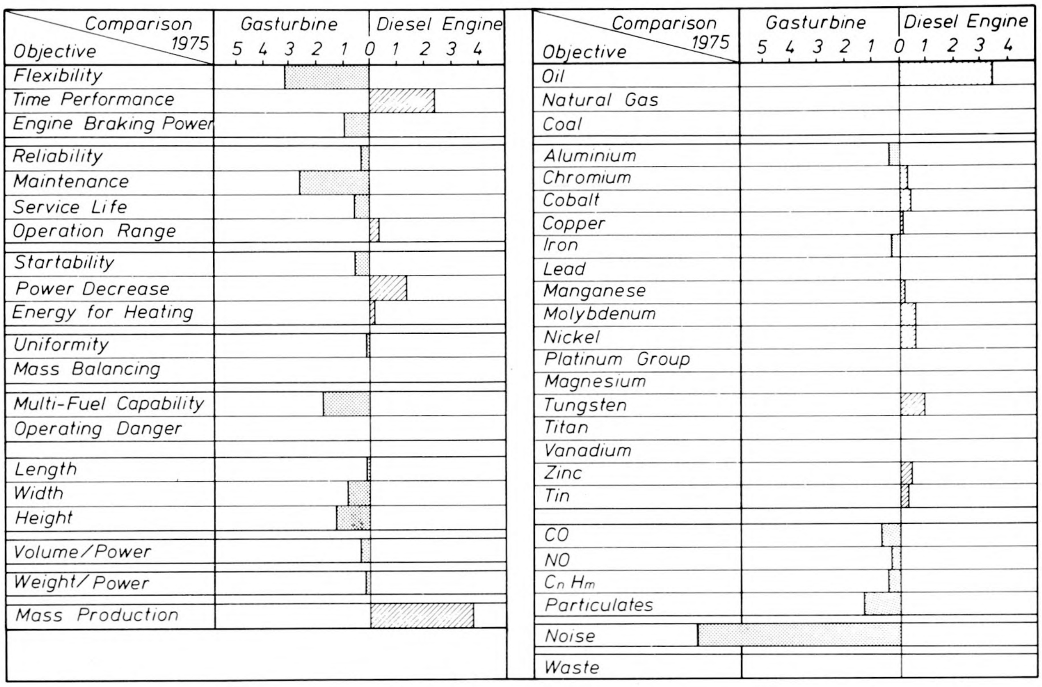

Fig. 9 : Comparison Diesel Engine - Gas Turbine for Heavy Duty Trucks

DIESEL ENGINE SUPERIOR TO GAS TURBINE

- *Suitability for Mass Production*
- *Oil (Fuel Consumption)*
- *Time Performance (Acceleration)*
- *Power Decrease (High Sentivity)*
- *Tungsten* } *Consumption of scarce*
- *Molybdenum* } *Raw Materials*
- *Nickel*

GAS TURBINE BETTER THAN DIESEL ENGINE

- *Noise*
- *Flexibility*
- *Maintenence Requirement*
- *Multi-Fuel Capability*
- *Pollutants ($C_n H_m$, CO, Particulates)*
- *Weight per Power*
- *Volume/Power (Compactness)*
- *Engine Braking Power*

Table 1 : Major Decisive Objectives Comparison Diesel Engine - Gas Turbine for Heavy Duty Trucks

Combining Engine Cooling with Heat-Driven Air Conditioning to Improve Automotive Fuel Economy

by

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ABSTRACT

Conventional automotive air conditioning equipment, with its highly excited belt-driven reciprocating compressor, is examined for its penalties to fuel economy. These penalties are shown to arise from both fixed weight contributions to road load as well as parasitic shaft load. Penalties in fuel consumption due to equippage and usage have been found to run as high as 20% under adverse weather and driving conditions with average annual fuel penalties of about 6% in the equipped automotive population as a whole. The A/C equippage rate of new automobiles sold in the United States is currently exceeding 75% and is projected to reach 95% by 1980. A format for evaluating the energy impact of automotive air conditioning and waste heat alternatives is presented. The conceptual design of one such alternative--an organic fluid engine cooling system combined with a novel jet-vapor-compression automotive air conditioning system--is described and evaluated in comparison with the conventional belt-driven vapor compression system added on to the contemporary water-cooled power plant. The salient features of this novel Rankine bottoming cycle approach and some of its characteristics are also described. The value of shifting from a prime shaft-driven system to a hypothetical light-weight waste heat recovery system is determined for a single model-year fleet. A potential reduction in the annual energy cost of automotive air conditioning usage of more than 70% is indicated, which is shown to be equivalent to the conservation of about 270 million barrels of crude oil over a ten-year period.

INTRODUCTION

Conventional automotive air conditioning systems employ a prime shaft-driven reciprocating compressor, belt driven from the engine (Figure 1). This highly excited accessory creates an additional shaft load on the engine as well as structural weight in the vehicle that has been shown to result in an average penalty of 10% in fuel economy whenever the air conditioner is operational, with instantaneous penalties going up as high as 20% under adverse weather and driving conditions [1-4].* The energy impact resulting from this accessory usage is thought to be significant.[4]. The fact that over three-fourths of the new cars produced in the U.S. are being equipped with air conditioning (a figure that is projected to reach 95% by 1980) further heightens this significance [5].

Past attempts to apply engine waste heat to the reduction of accessory penalties have not produced encouraging results [6-8]. Complicated heat recovery features, inadequate performance, excessive bulk and/or weight are some of the common disadvantages. The thermodynamic handicap of using the conventional jacket coolant as an intermediate heat transfer medium is formidable. Direct heating with the exhaust of a spark ignition engine involves some exceedingly complex heat transfer, materials and control requirements. Conventional absorption cycle and Rankine cycle refrigeration machines have been found inherently complex, bulky and expensive to manufacture for automotive applications.

A further deterrent to the progress of waste heat utilization for automotive accessories has been the uncertainty heretofore in the criteria determining the marginal benefits to be gained. This obstacle is yielding to the results of studies to determine the energy impact of automotive air conditioning in such a form as to permit evaluation of waste heat alternatives [9].

FUEL DEMANDS DUE TO THE CONVENTIONAL AUTOMOTIVE AIR CONDITIONER

The penalty in fuel economy due to the use of air conditioning equipment arises both from the shaft power required to run the compressor and

*Numbers in brackets designate references listed at the end of paper.

the dead-weight addition to vehicle road load. Table I gives the average fuel economy of the 1973 model year U.S.-produced automobiles with and without air conditioning, based on an urban-suburban driving cycle [4]. The fuel economy figures presented in Table I are based on the constant percentage dead weight and shaft power penalties given in Reference 4 which were derived from experience with the larger class vehicles. Specifically, a dead weight penalty of about 3% and a total penalty of about 12% were applied independently of the vehicle class in this study. This simplifies the analysis but understates the overall impact of air conditioning accessory penalties on energy consumption for the following reasons:

(1) Both dead weight and shaft power penalties on fuel consumption will definitely escalate inversely with curb weight and rated engine power;

(2) Air conditioner installed weight does not vary significantly with capacity for commercial auto-type systems;*

(3) Smaller class cars operate at a higher average fraction of their rated power and are more sensitive to changes in both road load and engine parasitic load;

(4) Larger class cars operate at such a small fraction of their rated power that an accessory base load might actually improve their thermal efficiency partially off-setting extra shaft load; and

(5) Smaller class cars are the fastest growing segment of automotive sales, and even though their air conditioner equippage rates are lower (see Table I), these rates are also growing rapidly.

The 1973 U.S.-produced new car distribution by market class and the corresponding air conditioner equippage rates are also shown in Table I [10, 11]. In order to utilize this data to compute the energy impact of air conditioning, an air conditioner use factor representing the fraction of yearly vehicle miles driven with the air conditioner in operation must

*York Borg-Warner 6 cu. in. compressors are identical in weight to their 10.3 cu. in. units. The compressor and its mounting and drive features represent about half of the system weight.

be known [9]. While this statistic is subject to wide regional variability, a use factor value of 0.3 is a reasonable and conservative approximation for the U.S., considering the estimated monthly distribution of vehicle miles driven over the period of a year [12].*

Based on the above simplifying assumptions, the estimated annual impact of automotive air conditioning on fuel consumption for the 1973 model year fleet of cars is shown in Table II [4, 9, 13]. Air conditioning of the 1973 models alone results in an additional annual consumption of 333 million gallons of gasoline requiring an additional 9.5 million barrels of crude oil at the refinery [14].** When measured on an individual vehicle basis, air conditioning equippage and usage requires a minimum of 33 gallons of additional gasoline consumption per year, or from 4 to 5% overall reduction in mileage for the 1973 model year. However, when this additional fuel consumption is projected for just the new car fleets for the 10-year period starting with the 1973 model year, and assuming a 10% annual retirement rate and a continuation of the 1973 rate of equippage and usage, the additional requirement for crude oil at the refinery will amount to about 365 million barrels. This represents about 1% of the 1975 proved national petroleum reserves [14].

AN ALTERNATIVE WASTE HEAT UTILIZATION APPROACH

The adverse impact on fuel consumption arising from using the conventional air conditioner can be considerably reduced if the prime shaft driven system is replaced by a light weight, thermally driven system and the benefit is proportionately greater for smaller cars. The waste heat from the engine cooling jacket and exhaust exceeds 75% of the heating value of the vehicle fuel consumption and represents a very attractive

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*According to the 1973 Federal Highway Statistics of the Department of Transportation, around 36% of the annual vehicle miles driven is during the months of June through September. The average miles travelled per vehicle in 1973 was 9992 miles.

**In computing the global energy penalty, allowance should be made for the by-products of petrochemicals and non-gasoline fuels which altogether represent about an equal fraction of the refining output as motor fuel [14]. When such allowance is made and all refining losses are charged against the prime product, the net crude equivalent of gasoline is approximately 1.2 barrels per barrel.

source for this purpose. Although utilization of the waste heat from internal combustion engines to operate an air conditioner has been investigated in the past and has even seen some limited application [6, 7], this study differs from the previous work in three key respects:

(1) It considers a jet-vapor-compression (ejector) system instead of an absorption system or a Rankine cycle shaft-driven conventional vapor compression system;

(2) It makes use of both jacket and exhaust heat where previously only the exhaust heat was considered as a thermal source; and

(3) It combines the function of the working fluid in the air conditioning cycle with that of the engine coolant.

Thus, the system contemplated herein is a combined engine cooling system and waste heat driven air conditioning system (see Figure 1).

The potential advantages suggested by the inventors of this system [15] include the following:

(1) Pure dielectric engine coolant boiling heat transfer:

- (a) Minimizes temperature gradients;
- (b) Maximizes temperature uniformity;
- (c) Minimizes structural distortion;
- (d) Eliminates corrosion and scaling; and
- (e) Favors engine weight reduction and reliability: light weight alloy construction; cooling jacket volume reduction; improved engine tolerances; reduced engine wear and fouling rates; reduced accessory mounting and drive weights.

(2) Rapid warm-up, higher jacket temperature, improved temperature control:

- (a) Lowers exhaust-emissions;
- (b) Reduces quenching and chamber fouling;
- (c) Improves lubrication; and
- (d) Reduces temperature losses and improves thermal and mechanical efficiencies.

(3) Light weight bottoming cycle accessory drive:

(a) Reduces vehicle fuel consumption; and

(b) Increases vehicle performance.

As depicted in Figure 1, the alternative system provides engine temperature control as well as compartment cooling and heating with a single working fluid and a simplified set of components. The three operational modes of the system and the various control features are described in detail in Appendix A and in Figure 2. The following presentation describes some of the major design considerations for automotive application.

The passenger compartment cooling performance of the waste heat driven jet-compression air conditioning system was evaluated under engine idling conditions with regard to engines typically used in full size, intermediate size, compact and sub-compact vehicles [9]. The characteristics of these engines were obtained from several sources [8, 11, 16-18]. The working fluid used in the evaluation was trichlorofluoromethane (CCl_3F)* and the design cycle operating conditions (see Table III) were selected so that the heat transfer from the engine cylinder walls to the coolant will occur under forced convection boiling conditions (flooded evaporator). Cycle conditions are consistant with experimental data derived from a similar system [19].

The principal design criterion of the condenser is the same as for a conventional automobile radiator; namely, adequate heat dissipation at intermittent, full speed wide open throttle engine operation at maximum ambient air temperature and flow conditions. The selection of the design maximum condensing temperature involves a trade-off between compartment cooling capacity and condenser (radiator) size which has not yet been examined in detail. However, the following considerations tend to support the conclusion that an adequate single condenser/radiator for the hypothetical system can have less surface and volume than the tandem radiator and condenser of the conventional system:

*Commonly known as Freon-11, which is a Trademark of the DuPont Company.

(1) Upstream blockage and air preheating by the separate refrigerant condenser is eliminated;

(2) Air-side thermal conductance is low compared with that of the tube side condensing film of pure refrigerant, which in turn is higher than can be maintained with forced convection raw water in conventional cooling systems;

(3) The uniform temperature of the condensing coolant can obtain a higher average thermal potential for heat transfer than can the sensible heat effects available with conventional single phase liquid coolant; and

(4) Condensing temperature can be allowed to float upward at the expense of refrigeration cycle COP* and with the higher flow and heat rates available at high engine load, adequate compartment cooling can still be obtained.

The system heat input features conceptually provide for "flooded evaporator" heat transfer conditions in the engine cooling jacket followed by a separate "superheat/desuperheat" arrangement utilizing a jacketed exhaust manifold. Conventional refrigeration controls manage a combination of by-pass flow paths so as to accelerate cylinder warm-up, limit maximum cylinder temperature (saturation pressure) and maximum exhaust-jacket fluid temperature (outlet superheat). Gas-side film conductance in the exhaust jacket is too low to economically accomplish more than a small amount of superheating (about 8% of total heat input). But the superheating is valuable for good ejector performance. This characteristic is also valuable in preventing hot spots and limiting maximum refrigerant temperatures during full load operations. Typically, CCl_3F is first heated in the cooling jacket and exists as a mixture of saturated vapor and liquid at a temperature of 240°F and a pressure of 160 psia. It is then completely vaporized and superheated over the exhaust manifold to a temperature of 280°F and at nearly the same pressure, and enters the ejector under these conditions. For idle

---

*COP denotes the coefficient of performance, which is defined as the net refrigeration effect per unit of input energy.

cooling conditions, the temperature at the condenser is assumed to be 120°F and the condensing pressure is 33 psia.

Other than these heat rejection (condenser) and waste heat recovery (jacket) features, the only other components of any sizable consequence involved in this system are the recuperators and the receiver. Based on the idle condition design point requirement (Table III), the high-side recuperator must handle a maximum of about 4000 Btu/hr at about unity capacity rate ratio with a pure counterflow temperature difference of about 43°F. State-of-art compact heat exchanger practice [20] indicates that less than 2 sq. ft. of compact surface occupying less than 75 cu. in. in volume (less than 1/3 gallon) will be required for this component. The low-side recuperator must handle about 750 Btu/hr at about unity capacity rate ratio with a pure counterflow temperature difference of about 43°F. A similar compact unit for this component would require about one-half the surface area in less than 30 cu. in. (about a pint) of volume.

The receiver volume sizing will depend on the service interval objectives and sealing practices. The contemplated system is considered to be hermetic (unlike conventional systems with the open compressor). However, since elastomeric system connections may be justified in practice and service competence may be initially lacking, over-sized refrigerant inventories may be required. Allowing for the liquid hold up in the engine block, condenser, plumbing and recuperators, no more than a 6-lb. (1/2 gallon) receiver should be necessary. The refrigerant charge for a full sized automobile is estimated at about 36 lbs. (3 gallons), about $30 worth at current retail prices.

The refrigerant (coolant) pump must provide a maximum flow of 560 lbs/hr (about 0.8 gpm) under idle conditions with a pressure rise from 130 to 140 psi. This requires a shaft input of about an 1/8 HP. Various positive displacement pump types are available which can handle this requirement without the addition of lubricant to the liquid refrigerant. A true hermetic system would require a canned electric drive or a sealless magnetic coupling with a mechanical drive, either of which is feasible. In order to stay within the latent cooling range in the engine jacket, a flow rate variation of about 13 to 1 from idle to full load will be required. Since the engine speed range is only 7 to 10 to 1, some sort of flow or pump speed modulation may be required.

The results of the calculation of engine idling heat rejection recovery and ejector air conditions system cooling capacities are summarized in Figure 3. It is seen that full size, intermediate size and compact cars have good potential for utilizing their waste heat for air conditioning. In sub-compacts, adequate comfort cooling with the heat driven system requires a higher degree of heat recovery or reduction of vehicle cooling requirements at idle to about 6000 Btu/hr. Full sized class cars obtain adequate compartment cooling under idle static conditions with no more than 12,000 Btu/hr of cooling capacity. The cooling requirements of the smaller cars will certainly be less but have not been specifically determined. However, the cooling requirements for all classes of cars can be reduced through the choice of colors, finishes, insulation, ventillation, glass coatings, etc.

The operating characteristics and air conditioning system performance under engine idling conditions are shown in Table IV for two full size vehicles, assuming that one is equipped with the conventional air conditioner and the other with the waste heat driven air conditioner. Several differences between the two systems are apparent, most notably the heat rejection to the ambient. The heat rejected through the single condenser-radiator in the vehicle equipped with the ejector system is about 61,000 Btu/hr. In contrast, the vehicle equipped with the conventional system will reject only 18,000 Btu/hr through the condenser of the air conditioning system and must still reject the cooling system load of 57,500 Btu/hr through the radiator. Thus, although it appears that the condenser-radiator will have to be about three times as large as the condenser of the conventional air conditioner, the absence of the normal radiator should partly compensate for the size effects. The high refrigerant flow rate with the ejector system is due to the dual use of refrigerant as engine coolant.

FUEL REQUIREMENTS FOR THE WASTE HEAT DRIVEN JET-VAPOR-COMPRESSION AIR CONDITIONING SYSTEM

The fuel economy estimates for automobiles similar to those shown in Table I, but now considered to be equipped with the ejector air conditioning system are shown in Table V. The dead weight penalty of the ejector air conditioner has been assumed to be only 55% of that for the conventional system since in addition to its inherently lower weight it does not require the additional engine displacement needed to accomodate the conventional air conditioner and maintain the same driveability. Further, when combined with the refrigerant cooling technique and the advantages it offers in reducing engine and structural weight, it is conceivable that still further reductions in fuel consumption could be gained from vehicle weight reduction alone [21].

Annual automotive fuel consumption arising from the use of the hypothetical waste heat driven system is shown in Table VI for the 1973 model year cars and the air conditioner equippage rates given in Table I. These results are compared with the annual fuel consumption of the same vehicles equipped with the conventional air conditioning system. Substitution of the ejector system appears to have the potential for reducing the annual accessory fuel penalty from 333 million gallons of gasoline to 91 million gallons—a reduction of about 73%. This amounts to an annual reduction of refinery crude throughput of over 7 million barrels from this single model year fleet. If it is assumed that 10% of the 1973 new car fleet is retired annually, and just the equipped portion of this fleet has benefitted from the conservation potential of the Rankine ejector bottoming cycle heat recovery system, about 40 million barrels of crude oil might have been conserved. Projecting further, if each new model year fleet thereafter were so equipped, the cumulative reduction of refinery petroleum requirement over a ten-year period could be as much as 270 million barrels. This is about 0.75% of the 1975 proved U.S. petroleum reserves [14].

CONCLUSIONS

The emphasis of this paper has been to show that the adverse effect of conventional automotive air conditioners on fuel consumption can be substantially reduced if the engine driven system is replaced by a weight reducing waste heat driven Rankine ejector bottoming cycle system. In particular, the 1973 model year fleet of cars was used as an example to illustrate this and show that over 70% of the fuel consumption due to the use of the conventional air conditioner can be saved without attempting to capitalize on all of the weight/economy potential available.

Based on what we believe to be conservative estimates of fuel cost savings available to the public--some hundreds of million dollars in a relatively short time period--development and implementation of a vehicle weight reducing waste heat driven automotive air conditioning system appears to deserve priority treatment. We have looked at one possible approach and have identified some promising features for a practical vehicle adaptation. Unquestionably, there are many more considerations to weigh in the translation of the contemplated system into operational and commercial reality.

However, one more general factor should be mentioned. There is a lively controversy today over the widespread public use of volatile chlorinated hydrocarbons because of a conceivably adverse affect on the earth's ozone layer [22]. This controversy is still unresolved [23]. We chose CCl_3F as the working fluid in the system described herein to illustrate the characteristics of a conceptual approach. No advocacy of the wider use of chlorofluorocarbon materials is intended. It is our belief that in the event halocarbons are determined to have a serious environmental impact, other acceptable substitutes will be found which could also be applied to the combined engine cooling/ejector air conditioning system without affecting its feasibility.

APPENDIX A

EJECTOR SYSTEM OPERATION AND CONTROL

The three operational modes of the system are described as follows:

(1) Engine cooling only;

(2) Engine cooling and cooling of the passenger compartment; and

(3) Engine cooling and heating of the passenger compartment.

Reference is made to Figure 2 for the arrangement of valves, sensors, components, and conduits. The shaded portion of conduit represents the flow path for the refrigeration cycle.

The system operation in the first mode is one where neither cooling nor heating of the passenger compartment is called for and only engine heat needs to be rejected to the ambient. Solenoid valves A and B are closed so that only the left side of the system is operative. Initially, with a cool engine, refrigerant fluid driven by the pump passes through the high side recuperator and through the normally open thermostatic valve F bypassing the engine cooling jacket so as to accelerate engine warm-up. The valve F senses the engine temperature as indicated by line 'a' and closes after engine warm-up so that the refrigerant fluid then passes in series through the engine cooling jacket and jacketed exhaust manifold. The refrigerant fluid leaves the engine cooling jacket as a saturated vapor and liquid and is completely vaporized and superheated in the jacketed exhaust manifold. The superheated vapor then passes through the pressure bypass valve E. Valve E is responsive to the upstream pressure of the vapor as shown by line 'c'. Under conditions where valves A and B are closed, the pressure and temperature of the refrigerant vapor float upward to the pressure relief setting of valve E. When valve E opens, the refrigerant vapor is conveyed to the condenser through the high-side recuperator. At the condenser, the refrigerant is condensed to a liquid by heat transfer to ambient air and is then accumulated in the receiver. From the receiver, the refrigerant liquid is pumped back to the engine through the high-side recuperator to repeat the cycle. Control of the degree of superheat in the refrigerant vapor is effected through valve D which is responsive to the vapor temperature as shown by line 'b'.* At a preset superheat temperature, valve D opens to admit the relatively low temperature refrigerant liquid directly into the exhaust manifold jacket.

The second mode becomes operative when cooling of the passenger compartment is called for. In this mode of operation, solenoid valve A is opened to the high pressure side, solenoid valve B is open only between ports P2 and P3 (valve E will likely be closed). The high pressure refrigerant vapor stream enters the primary nozzle of the ejector and, as it expands, evacuates low pressure refrigerant vapor from the evaporator through the low side recuperator. The two streams mix and the mixture then diffuses to the condenser pressure level. In the high-side recuperator, prior to entering the condenser, some of the residual heat of compression from the vapor stream flowing out of the ejector is recovered by transfer to the engine feed liquid. At the condenser, the vapor stream is condensed to a liquid and collected in the receiver. The major portion of the condensed liquid is returned to the engine cooling jacket by the pump via the high- side recuperator. The remainder passes through the low-side recuperator for sub-cooling by interchange with vapor leaving the evaporator. It is then throttled to the evaporator pressure level to make available its latent heat of vaporization at the evaporator temperature to produce a refrigeration effect. The throttle valve G is controlled to increase delivery of refrigerant liquid to the evaporator based on superheat temperature increase sensed by line 'd' at the refrigerant exit. Modulation of the cooling capacity is effected by cycling valve A manually on demand or automatically by an air temperature thermostat (not shown).

The third mode of operation is required when the passenger compartment needs to be heated. In this mode of operation, valve A and valve E are closed to the high-side refrigerant vapor stream. Valve B is open only between ports P1 and P2. The refrigerant vapor stream from the engine is conveyed directly to the evaporator which now operates as a

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*Valve D is a conventional thermostatic refrigerant expansion valve with a vapor charge selected for the appropriate superheat and temperature range.

condenser. At the evaporator, the refrigerant produces a heating effect in giving up its latent heat of condensation to the air from the passenger compartment. The partially condensed refrigerant is bypassed around the throttle valve via check valve C directly to the condenser inlet via the high-side recuperator. The condensed refrigerant liquid is then returned to the engine by the pump. Modulation of the heating capacity is effected by cycling valve B with valve A closed.

The design of the system as described permits switching from a cooling mode or heating mode, or to neither cooling nor heating by actuating the solenoids controlling valves A and B. The refrigerant pump operates continuously while the engine is operating, and together with the pressure bypass valve E serves to limit the maximum engine jacket temperature to a predetermined level.

TABLE I

Characteristics of the 1973 Model Year

U.S. Domestic Automobile Fleet[1]

	MARKET CLASS			
	FULL SIZE	INTERMEDIATE	COMPACT	SUB-COMPACT
Gross Vehicle Weight, 1000 lbs. [14]	4.5-6.0	3.5-4.4	2.5-3.4	2.0-2.4
Fraction of New Car Production, % [10]	54.0	22.0	16.0	8.0
Air Conditioner Equippage Rate, % [11]	89.0	76.0	40.0	29.0
Fuel Economy, mpg:[2]				
Base Case (not equipped)[2]	14.5	14.9	16.9	20.3
A/C Equipped, Not Operating[3]	14.1	14.5	16.4	19.7
A/C Equipped, Operating[3]	12.8	13.5	14.9	17.9

NOTES:

[1]Total U.S. New Car Production Approximately 10 million [10].

[2]Based on an urban/suburban driving cycle [4].

[3]Fixed weight penalty of about 3% and total penalty of about 12% assumed [4, 9].

TABLE II

Estimated Effects of Air Conditioning on
Annual Fuel Consumption for 1973 Model Year [9]
Millions of Gallons of Gasoline

	BASE CASE[1]	AIR CONDITIONED[2]
Direct Consumption [9]	6,480	6,763
Indirect Consumption [4][3]	1,100	1,150
Total	7,580	7,913
Additional Consumption	---	333

NOTES:

[1]Assumes none of the 1973 cars is air conditioned.

[2]Equippage rates and fuel economies as given in Table I. Assumes a use factor of 0.3 [9].

[3]Accounts for energy spent in equivalent gallons of gasoline in producing, extracting, refining, transporting and distributing the direct gasoline requirements and represents about 17% of the direct consumption [4]. Other authorities quote higher figures [13]. Energy consumed in the manufacture of air conditioning equipment is neglected.

TABLE III

Design Equilibrium Cycle Operating Conditions
for the Ejector Air Conditioning System [9]

STATE POINT[1]	TEMPERATURE (°F)	PRESSURE[2] (psia)	ENTHALPY[3] (Btu/lb)	ENTROPY (Btu/lb/°R)
1	280.0	160	126.2	0.2027
2	82.0	7	102.5	0.2076
3	81.5	7	102.4	0.2075
4	212.0	33	120.2	0.2151
5	166.0	33	113.2	0.2042
6	120.0	33	32.9	0.0662
7	95.0	33	28.8	0.0568
8	40.0	7	28.8	0.0606
9	50.0	7	98.1	0.1993
10	80.0	7	102.2	0.2071
11	130.0	160	33.2	0.0662
12	162.0	160	40.2	0.0777

NOTES:

[1]Refer to Figure 2 for locations.

[2]Line and core pressure drops neglected.

[3]Design mass flow ratio of primary to secondary stream = 3.04.

TABLE IV

Comparison of Automotive Air Conditioning System Characteristics[1]

	CONVENTIONAL AIR-CONDITIONER	WASTE HEAT DRIVEN AIR-CONDITIONER
Cooling Capacity:	12000 Btu/Hour	12750 Btu/Hour
Refrigerant:	CCl_2F_2 (Freon-12)	CCl_3F (Freon-11)
Energy Input:		
Shaft	2.3 HP (Compressor)	0.125 HP (Pump[2])
Cooling Jacket	-	44,100 Btu/Hour (60% jacket heat utilization)
Exhaust	-	4000 Btu/Hour
Total Input	6000 Btu/Hour	48,100 Btu/Hour
Energy Rejected To Ambient:		
Refrigerant Condenser	18000 Btu/Hour	60850 Btu/Hour
Engine Radiator	57500 Btu/Hour	-
COP: [3]	2.0	0.265
Evaporator:		
Temperature	40°F	40°F
Pressure	52 psia	7 psia
Condenser:		
Temperature	141°F	120°F
Pressure	224 psia	33 psia
Flow Rate of Refrigerant:	190 lbs/hour (0.26gpm)	560 lbs/hour (0.8gpm)

NOTES:

[1] Comparison between the conventional and waste heat driven air conditioners in a full size vehicle under idling conditions.

[2] The conventional water pump is eliminated so there is actually a negative shaft power increment.

[3] Conventional system based on compressor HP only. A direct comparison requires that the thermal efficiency of the engine be taken into account. Obviously, at idle conditions engine thermal efficiency is nil and, thus, a comparable COP is also nil.

TABLE V

Estimated Fuel Economy for 1973 Automobiles
Equipped with the Ejector Air Conditioner

MARKET CLASS	FUEL ECONOMY, MILES PER GALLON: NOT AIR CONDITIONED[1]	AIR CONDITIONED: ACCESSORY NOT IN USE[2]	AIR CONDITIONED: ACCESSORY IN USE[3]
Full Size	14.5	14.28	14.28
Intermediate	14.9	14.67	14.67
Compact	16.9	16.64	16.64
Sub-Compact	20.3	20.00	20.00

NOTES:

[1]From Table I.

[2]Dead weight penalty in fuel economy assumed = 1.5% [9].

[3]Figures in column 2 and 3 are identical because, unlike in the engine driven system, the operation of the ejector system has negligible drain on engine output.

TABLE VI

Summary of Estimates of Annual Fuel Consumption

Millions of Gallons of Gasoline

	BASE CASE: NO AIR CONDITIONING[1]	WITH CONVENTIONAL AIR CONDITIONER[2]	WITH WASTE HEAT DRIVEN EJECTOR AIR CONDITIONER[2]
Direct Consumption	6,480	6,763	6,556
Indirect Consumption	1,100	1,150	1,115
Total Consumption	7,580	7,913	7,671
Additional Consumption	---	333	91
Savings	333	---	242

NOTES:

[1]See Table II for further conditions, assumptions and references.

[2]1973 U.S. model year fleet and a 0.3 use factor.

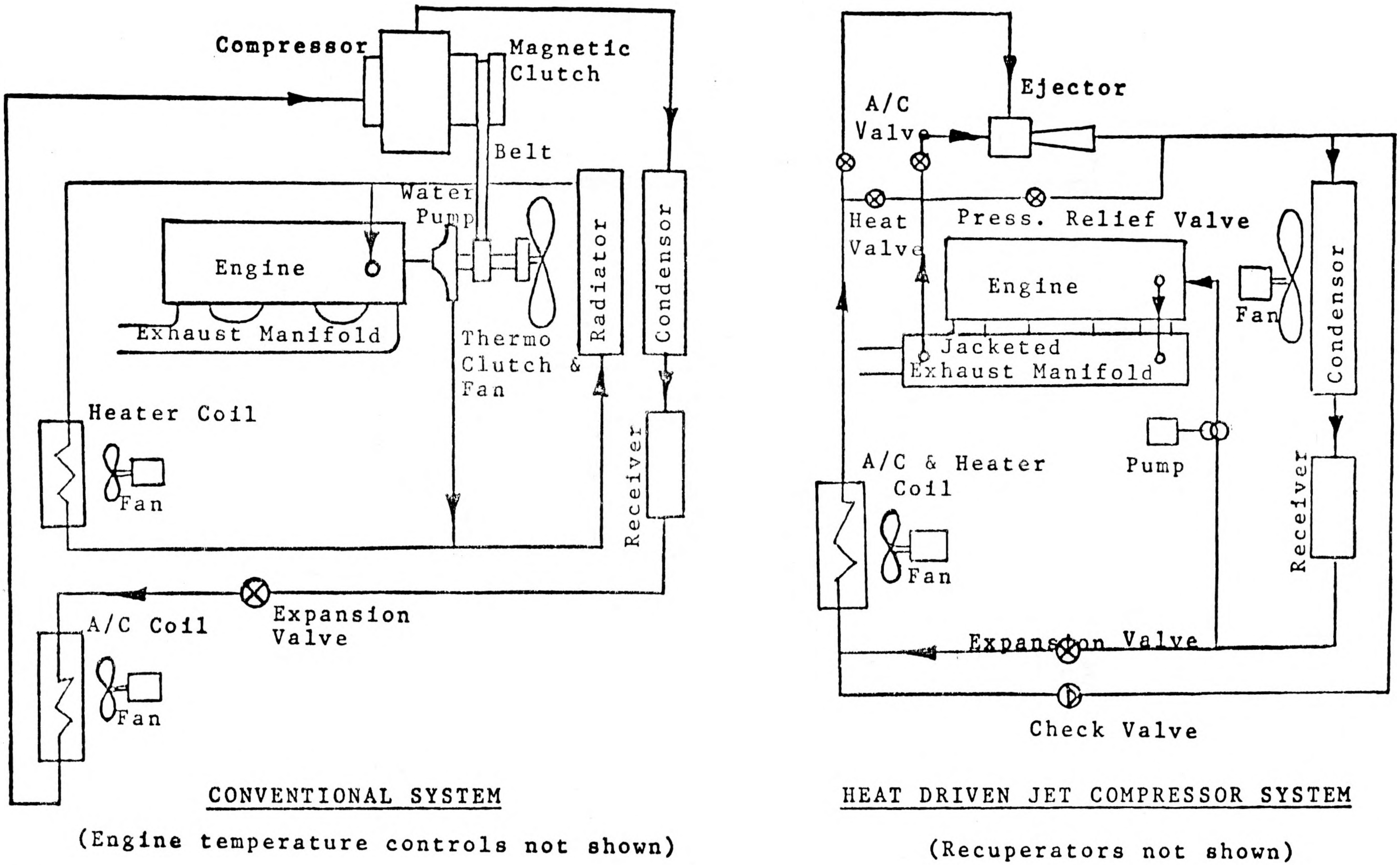

FIGURE 1: Schematic Comparison of Alternative Engine Cooling and Air-conditioning Systm

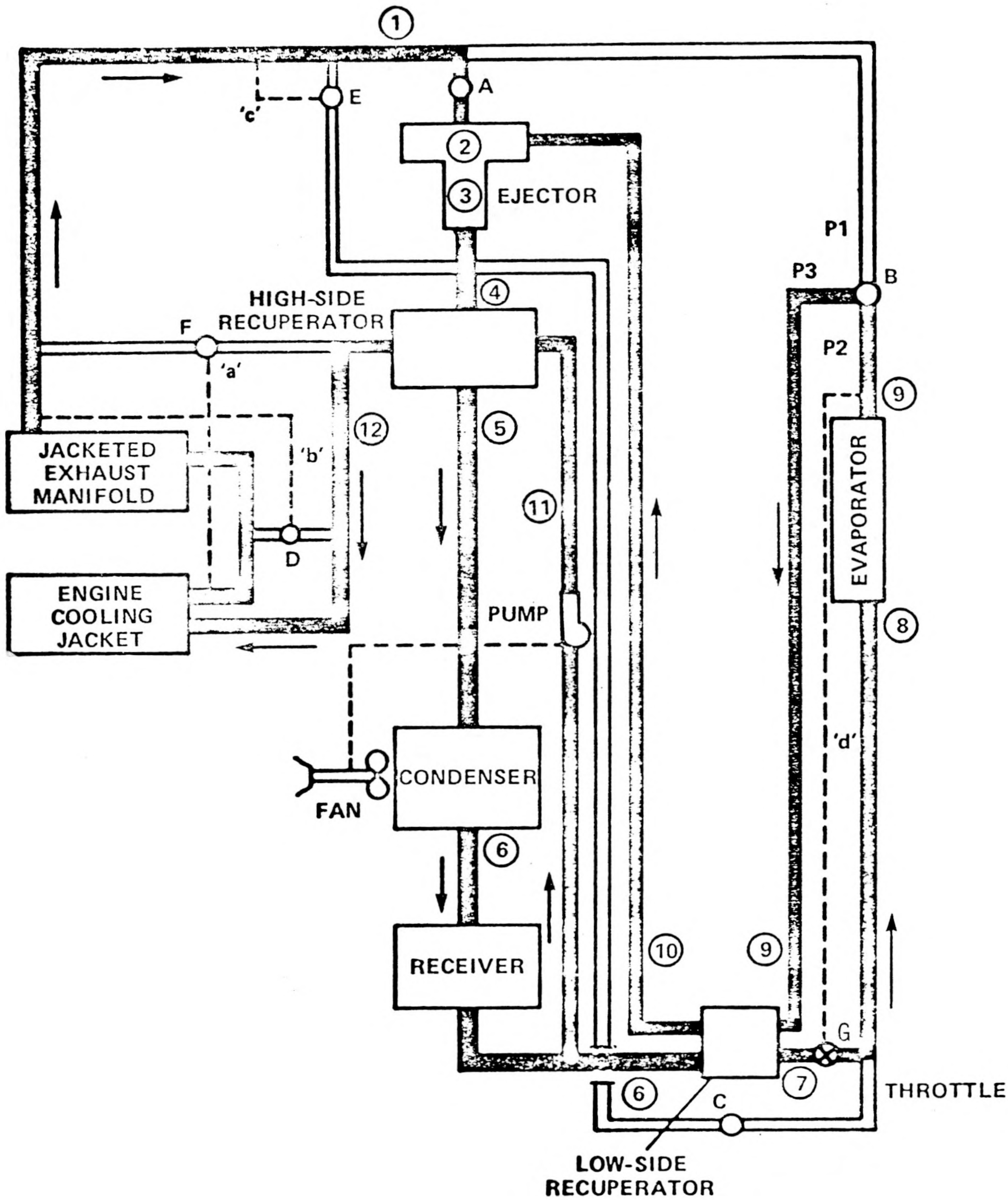

FIGURE 2. Block Diagram of Combined Engine Cooling, Compartment Heating and Jet-Driven Cooling System

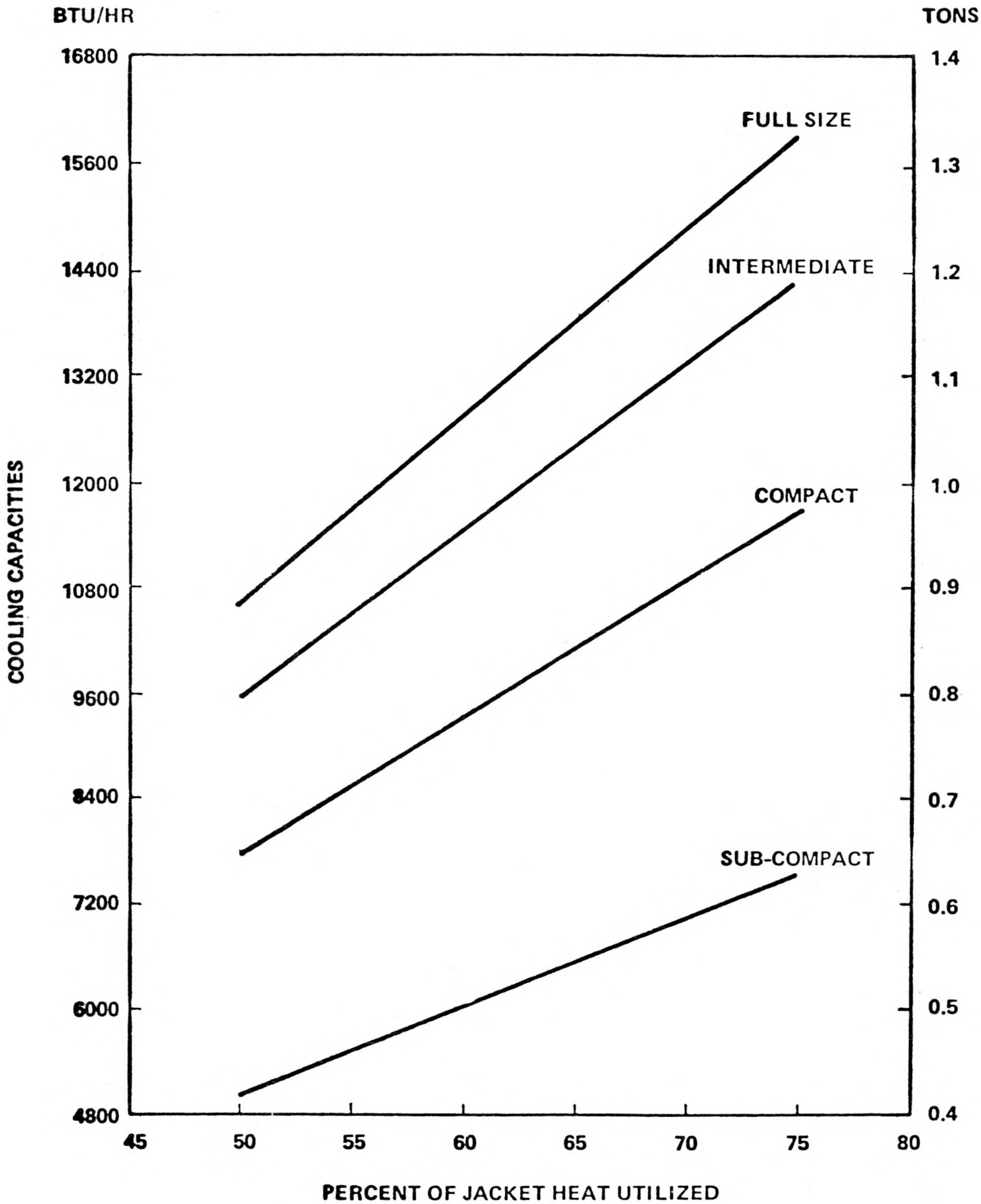

FIGURE 3. Estimated Engine Idle Cooling Capacity Characteristics of Ejector Air Conditioning System [9]

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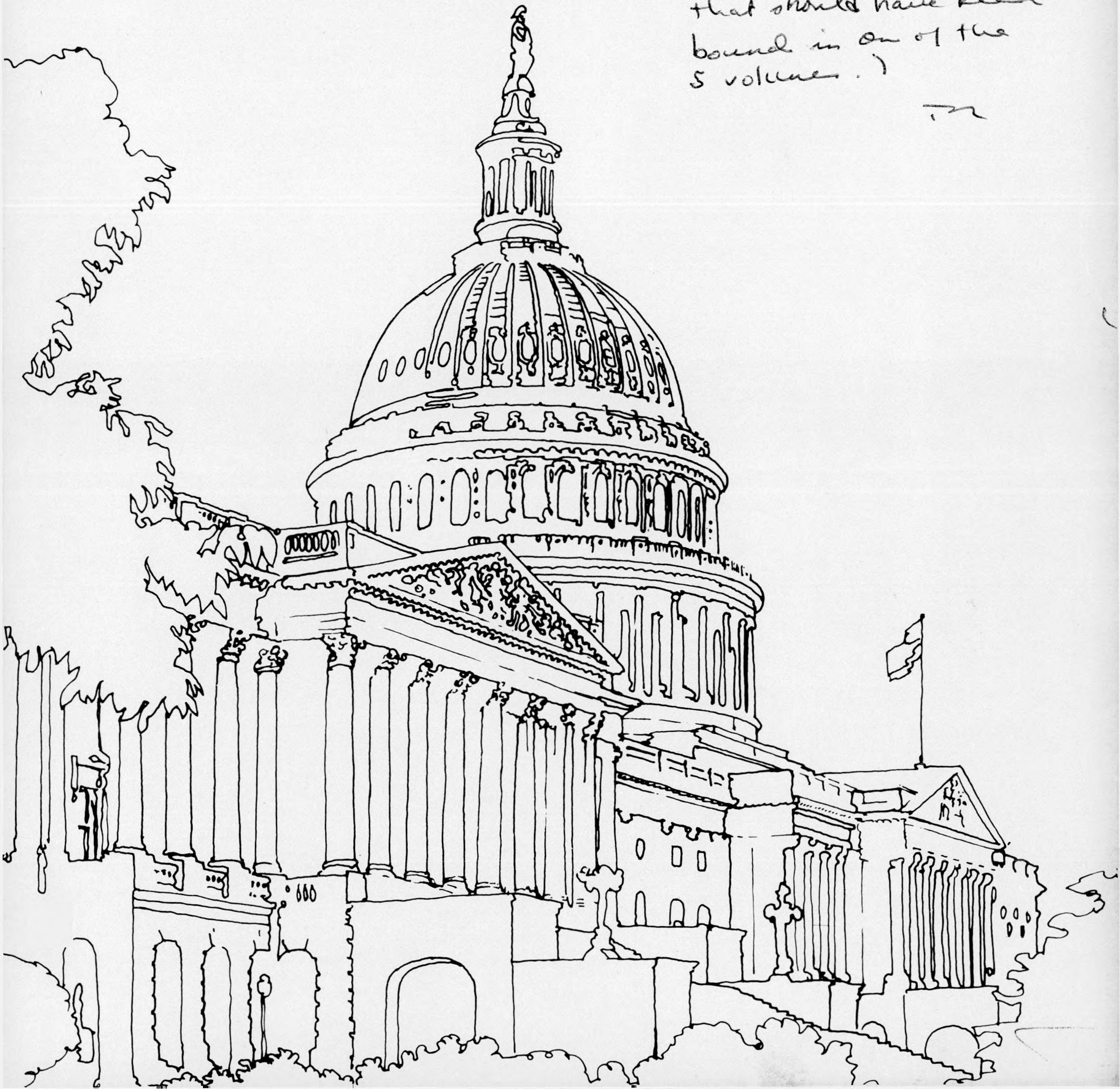

4th NATO - CCMS SYMPOSIUM

Design und Lay Out of Energy Efficient and Low Polluting Diesel Engines

The improvement in engineering objectives caused additional efforts in passenger car diesel engine development. Despite this current diesel engines are still inferior to modern designed gasoline engines in a few main points such as power to weight ratio and power to volume ratio are unfavorable when compared. Furthermore there are noise and smoke emission problems.

By modifying the combustion process the emissions and the noise can be influenced positively. Particular attention must be given to initation of combustion. Different possibilities of combustion initation especially for cold starting are being investigated.

Based on modern diesel engine designs different concepts for small highspeed diesel engines with a power to weight ratio and power to volume ratio similar to those of a gasoline engine are considered. Object is to retain the good fuel consumption and emissions typical of a diesel engine. Design and lay out of energy-efficient and low polluting future diesel engines are considered.

Design and Lay Out of Energy-Efficient and Low-Polluting Engines for Passenger Car Application

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Introduction

The scarcity of energy as well as ecological, economic, and safety considerations are responsible for an increasing amount of change in passenger car power plant standards. For this reason, it is necessary to optimize a power plant to meet a number of complex requirements and to find new trade offs between the most essential design goals (1). The following are 4 of the most important goals:

- Functional quality
- Installation Suitability
- Conservation of raw materials
- Environmental protection.

In recent years, Diesel engines have come to be increasingly used in passenger cars because of their low fuel consumption, and because their emissions (HC, CO, NO_x) adhere closely to legal regulations. However, there are disadvantages in that their horsepower-to-weight ratio, their size and their noise emissions are unfavorable (2). Table 1 shows the most essential aspects entering into a comparison between Diesel and spark ignition power plants.

ADVANTAGES:

Availability

Fuel requirements

Conservation of energy carrier

CO, NO_x, C_nH_m emission

Disadvantages:

Behavior under extreme conditons

Vibration level

Engine dimensions

Horsepower-to-volume ratio

Horsepower-to-weight ratio

Particle emissions

Noise

Table 1- Comparison Diesel Engine Versus Spark Ignition Engine

During the last few years developments concentrated on finding ways of alleviating the disadvantages of Diesel engines as compared to spark ignition engines: Consequentially, the design of an energy-efficient and low-polluting Diesel engine will have to meet the following demands:

- A combustion cycle must be developed which offers an optimum compromise between tendencies like fuel consumption, emissions, black smoke and power output, some of which are contradictory.
- An engine concept must be developed which combines a high specific power output and suitability for mass production.

Combustion Process

A comparison between the three existing basic combustion processes, i.e. pre-chamber, swirl chamber, and direct injection shows that the swirl chamber design is especially suitable to passenger car operation. Because of its efficient utilization of air and its wider range of engine speeds, its specific power output is higher than that obtained by the other two processes, which is one of the reasons why it is so well suited to passenger car operation, especially as the low specific power output of Diesel engines is mainly due to two facts:

a) All Diesel combustion processes require air surplus;

b) The engine speeds are lower than those of Otto engines.

Provided that 2-stage combustion is used, the emission level requirements of $CO < 3.4$, $HC < 0.41$, and $NO_x < 1.5$ (US City Cycle) can be met. If direct injection is used, meeting emission levels becomes extremely risky. Should the emission limits be relaxed, the use of direct injection will improve fuel economy by 10 to 15 %. If, however, evaluations are based on the limits given above, the significance of this advantage over swirl chamber engines is reduced to that of a slight improvement in fuel consumption, as the fuel economies of a direct injection engine geared to low emissions and of a swirl chamber engine are nearly the same. This slight advantage can be evened out easily, inexpensively, and without risk by simple gearbox modifications, such as adding a fifth gear.

Especially if the MAN-M-concept is used, the idling noise of a Diesel engine can be reduced to the level of a spark ignition engine, but the separate-working-chamber concept generates less noise under driving conditions, which we believe is of more importance.

Compared to the prechamber concept, the swirl chamber concept seems to be preferable. It is easier to meet conditions of satisfactory fuel consumption under all operating conditions, sufficient power output, and good black-smoke emission behavior using the swirl-chamber concept (3). Although there has been some success in the past in obtaining low emissions and good fuel economy from pre-chamber Diesel engines specially geard to operating conditions in mines (4). These engines suffered from the disadvantages of having a low power output and a low horsepower-to-weight ratio. Only the swirl chamber concept will permit rated engine speeds in excess of 83 rps (5,000 rpm) and a specific power output of 25 kW (34 BHP) per liter in naturally aspirated Diesel engines. Finally, tests at high altitudes prove conclusively that prechamber engines are far more delicate than swirl-chamber engines as far as controlling black-smoke emission is concerned.

As swirl chamber engines permit especially high rated speeds, engines of this concept will yield the same power when operated at lower mean effective pressures. Moreover, the very fact that the combustion chamber is divided contributes substantially towards reducing peak pressure.

Manufacturing considerations demand that a maximum possible number of parts in the Diesel and Otto engines should be equal or similar. This seems to be the only way to produce Diesel engines at comparatively favorable cost. This can be done only if the peak pressures and, depending on the crank angle, the pressure gradients in the Diesel engine do not exceed those of the spark ignition engine by a considerable margin. Fig. 1 compares the pressure curves of spark ignition and Diesel engines. Tests have shown that the loads caused by inertia forces exceed those caused by gas forces at engine speeds of 75 rps (4,500 rpm) and more. Assuming that these pressure curves are applicable, the major dimensions of the crank gears of some production spark ignition engines are sufficient for use in Diesel engines.

The most improtant factors influencing the optimization of combustion are:
Air fuel mixture formation; ignition; combustion of the mixture.
These factors, in turn, are determined mainly by the configuration of the combustion chamber, the movement of the air in the chamber and injection characteristics. In direct injection engines, the swirl generating inlet port is the most essential factor influencing combustion, whereas in 2-stage combustion engines the entire process of combustion is governed by the geometry of the swirl chamber and of the throat, where heat losses are high (flow speed). Throttling losses are of little importance as far as this aspect is concerned, but the swirl largely affects the process of combustion and the increase of pressure.

For each application, compression ratio, timing, swirl chamber geometry, the ratio between swirl chamber and dead space volume throat area and cavities, and injection specifications have to be optimized (5,6).

2-stage combustion Diesel engines for passenger car application have to have a high compression ratio ($\varepsilon > 22$) in order to ensure good cold-start behavior, low noise, and low HC emission under partial load. For this reason, it is necessary to trade-off between these goals and the incrase in the losses through friction, which amounts to an average 0.085 bar per unit of compression and is accompanied by a minimal increase in thermal efficiency, and manufacturing difficulties due to the increase in the compression ratio (tolerances).

It is, therefore, especially interesting to see which measures may be used to modify final compression temperature, which is greatly affected by losses of mass and temperature when starting as shown in Fig. 2. Temperatures can be raised by means of the following modifications and devices.

- Further increase in the compression ratio (increasing friction losses)
- Advancing the closing time of the inlet valve (restrictions imposed by kinematic conditions)
- Using gas-leak proof piston rings
- Using shrouded valves (lower volumetric efficiency at high engine speeds)
- Increasing back pressure (low volumetric efficiency at high speeds or functional problems with the exhaust flap)

- Variable compression ratio (involves large tolerance problem in small cylinder units)
- Using ceramic materials (difficult to adapt)

Within the limits of feasibility, these and other engineering measures have given only insufficient improvements up to now. Consequentially, it is necessary to have a starting aid, and the only solutions which seem feasible in a passenger car are the flame starting device and the glow plug. Because of their irregular combustion and the sometimes steep increase in pressure over a small crank angle range, chemical starting aids may be considered only in some special cases.

In a swirl chamber engine the advantage of the glow plug lies in the fact that it produces heat close to the jet of fuel injected. The width of the temperature range shown in Fig. 2 demonstrates that the temperature distribution in the swirl chamber is not homogenious However, ignition is assured as the jet of fuel is dispersed by swirling and will ignite as soon as it touches the heater plug, if not earlier. Once perfected, the rapid-action heater plugs undergoing development at the moment will bring the starting comfort of Diesel engines much closer to spark ignition engine standards.

Swirl Chamber Geometry

The influence of swirl chamber geometry variations, such as flat bottom, spherical chamber, and pockets (Fig. 3) has been discussed elsewhere (7). The intention is to influence the combustion process by modifying the flow of the air current. One successful step in the direction of reducing the noise of Diesel combustion was the introduction of swirl chambers fitted with ribs and pockets. It was thought that guiding the jet of fuel injected to a hot zone would shorten the delay between fuel injection, thus ensuring a soft pressure buildup. The way in which engine performance maps react to this measure is shown in Fig. 4 through 6. In this context, it is worthy of note that the injection profile of the experimental swirl chambers was not as highly developed as that of the production type. Enlarging the throat diameter will improve the peak power output and increase the maximum torque, but it will increase the noise at idle and in the lower engine speed ranges. It seems possible that equipping engines with ribbed swirl chambers will make for good performance and improved noise emissions.

Most passenger car Diesel engines are equipped with distributor pumps and throttle pintle nozzles. Distributor pumps are lighter, smaller, and cheaper to manufacture than in-line pumps. Moreover, they can be used at relatively high engine speeds.

Their low service life, and their small dimensions have not recommended Pintaux nozzles) for largescale production use.

Using pump injectors does make for more freedom in tailoring injection profiles because there is no injection line volume to be taken into account, but speed- und load control, the injection timing of each individual cylinder unit would have to be variable, and this is expensive.

When adapting an injection pump to passenger car use there are two points to be considered especially, namely:

- Matching the governor setting to the conditions of passenger car use (high horsepower-to-weight ratio compared to trucks)
- Cold-start and warm-up phases.

Power Plant

In spark ignition engines, load peaks occur at high engine speeds and are caused by inertia forces, whereas in Diesel engines they occur at low speeds and are caused by gas forces. Therefore, there are no outstanding differences in the major dimensions, weights and manufacturing requirements, such as casting tolerances, between modern highspeed Diesel engines and Otto engines (3,8). Today, Diesel engines designed strictly according to weightsaving priciples offer horsepower-to-weight ratios between 3.2 and 3.6 kg/kW over a wide performance range (Fig. 7). The influence of swept volume, and number of cylinders on specific weight is shown in Fig. 8.

Applying specific lightweight construction technologies (9) might make for additional weight saving, to the extent of approximately 20 % in, for instance, a 50 bhp 1.5 l Diesel engine.

Compared to a castiron cylinder block, approximately 18 kg could be saved in weight by using a cylinder block made of an aluminum silicone alloy which allows direct contact between the block and the piston. Using a coated exhaust manifold made of ceramic fibres is advantageous as far as weight and temperature insulation are concerned. Implementing all these measures would bring the horsepower-to-weight ratio up to 2.6 kg/kW or, in other words, on a level with that of present spark ignition engines. On the other hand, we have to take into account the increase in primary energy consumption (aluminum), and the scope of manufacturing expenses, and soundproofing which could become necessary.

In small cylinder units it is extremely difficult as well as extremely important to adhere closely to the pre-determined compression ratio. Passenger car Diesel engines are always beset by problems concerning cold starting, warmup noise and emissions, and idling noise with the engine cold as well as warm.

The disadvantage of high compression ratios is that they do not permit optimum fuel consumption, and that it becomes difficult to adhere to manufacturing tolerances as compression volumes decrease with increasing compression ratio.

From the fuel economy viewpoint, the optimum compression ratio for small swirl chamber engines ranges between $\varepsilon = 16$ and $\varepsilon = 18$. However, with the manufacturing procedures available today it is impossible to produce an engine with a compression ratio of less than 21 whose cold-start and idle behavior are acceptable to passenger car customers. The fuel consumption of small swirl chamber engines could be reduced and manufacturing conditions could be relaxed noticeably if the compression ratio could be lowered to a level guaranteeing optimum fuel economy.

Steel-strutted pistons with ring-carrier have done well in passenger car Diesel engines. Their advantages are: Low skirt clearances, high rigidity, and uniform and uninterrupted flow of heat from the head to the skirt. Even when installed at a low clearance this type of piston will work well over the entire engine load range. The clearance of smooth-skirted pistons, on the other hand, is matched to conditions pertaining at maximum power output, so that it is larger than necessary whenever the engine is cold or operating at part throttle.

The heat loading of swirl chamber engines is higher than that of direct-injection engines. Therefore, the thermal loading of the valve bridge and heater plug is of outstanding significance, as is the configuration of the lower cylinder head face and the coolant ducts. As far as these aspects are concerned, the measures which could be taken include introducing special guiding ribs (3) or ceramic heater plugs.

The following three objectives are of special significance in developing the charge cycle of low-capacity high-speed Diesel passenger car engines:

- High volumetric efficiency at rated speeds so as to ensure attaining the desired power output;
- High final compression temperature at starting speed 2-2.5 rps (120-150 rpm) and at idle 12-13.5 rps (720-810 rpm) to approximate spark ignition engine standards of idle noise and cold-start and warm-up comfort
- Muffling the air intake noise, (in spark ignition engines, the presence of a throttle eleminates this problem, especially at partial load).

In order to attain these goals it is necessary to design a special air intake system and to design very exactly the timing of valves, especially the inlet valve closing time.

The inlet valves of today's passenger car Diesel engines close effectively at 90° after BDC. Advancing the closing of the inlet valve 25° would permit lowering the compression ratio by 6 units while maintaining the final compression temperature or increasing final compression temperature by 40° C while maintaining the compression ratio.

The paramount goal in optimizing the inlet cam of a passenger car Diesel engine, therefore, is to advance inlet closing as far as possible. On the other hand, delivery at rated speed must be sufficiently high to ensure the desired rated power output. Fig. 9 shows the restrictions presented by power plant kinematics. As we are dealing with low capacity engines where the gap between piston and valve was minimized in order to have about half of the dead space volume available for accomodating the swirl chamber, the lift face of the inlet cam is circumscribed by the piston travel.

So we are dealing with a typical trade-off problem: We have to compromise between a 'narrow' cam, which means early inlet closing (good starting and idling behavior), and late inlet opening time (kinematic restrictions), and a 'broad' cam to ensure the maximum possible power output.

The structure of a compact passenger car Diesel engine of low horsepower-to-weight ratio has been described in detail in (3). Deformation and stress calculations performed with the aid of the finite element and photoelasticity methods have shown that certain major engine elements subjected to loading, such as cylinder block, cylinder head, and crankshaft can still be improved in their design and material thickness.

Valve Gear

The OHC design often used in spark ignition engines is advantageous in Diesel engines as well. If the camshaft is driven by toothed belts or chains there is more latitude available for designing the crankcase and especially the inlet and outlet ducts as well as the cylinder head geometry. The length of toothed belts today remains constant in operation, so that it is possible for a toothed belt to run for more than 1,000 hours without any significant change in its length. For this reason, it was possible to design modern Diesel engines with their injection pumps and camshafts driven by toothed belts. As in this case it is possible to do away with the wheel cover, the chain cover, the chain, and its tensioning elements, an appreciable amount of weight is saved, and engine noise is improved at the same time. In addition the design of oil pressure drillings and lines is easier.

As Diesel engines have a high compression ratio it is necessary to keep the distance between piston and cylinder head at TDC as low as possible. For this reason it is necessary to make the belt jump-proof.This was achieved by keeping the belt wrap especially around the crankschaft pulley, as great as possible, and to ensure that no foreign bodies will penetrate into the system by fitting deflectors and covers (Fig. 10).

Lastly, injection pumps may be driven by means of shaft and a helical gear drive, as shown under (10). In this case, extra care is to be taken in designing the helical gear and in matching the materials employed.

Turbocharging

As lightweight design concepts are applicable to large-scale production engines only in a limited way we investigated another alternative, that of turbocharging. Engines running with ungoverned,turbochargers can only be designed to work within a very narrow engine speed range, so that their torque delivery is not up to passenger car engine standards. However, turbochargers governed by exhaust gas discharge (Wastegate) do not suffer from this disadvantage.

Turbochargers of this kind will increase the engine torque substantially even at low engine speeds. They are designed to operate most efficiently in the lower engine speed ranges. Exhaust gas can be blown off directly, circumventing the turbine, by means of a blow off valve controlled either by the exhaust gas pressure before the turbine or by the charge pressure (11).

Using twin-chamber turbines does not seem sensible for small swept volumes because of the unfavorable surface-to-flow ratio pertaining to all small turbines.

Fig. 11 shows the engine fitted with a turbocharger. Its weight is 127 kg, which means that its horsepower-to-weight ratio is 2.4 kg/kW, which is up to spark ignition engine standards

Fig. 12 shows the performance, torque, and specific fuel consumption at full throttle of both the 50 bhp naturally-aspirated and the 70 bhp turbocharged Diesel engines as a function of engine speed. As the camshaft was modified, the torque of the turbocharged engine could be raised at 16.7 rps (1000 rpm). Over the entire engine speed range, the specific fuel consumption of the turbocharged engine is lower than that of the naturally aspirated engine.

Fig. 13 shows a comparison between a 70 bhp spark ignition engine and a 70 BHP turbocharged Diesel engine, the former reaching its maximum power output at 96.7 rps (5800 rpm), and the latter at 83.3 rps (5000 rpm). Taking into account the difference in the rated speeds of the two engines we may say that the two torque curves are nearly identical, barring the lower speed ranges.

Figures 14 and 15 show fuel economy maps comparing the naturally aspirated engine to the turbocharged version.

Fig. 16 shows the comparison of the turbocharged Diesel engine versus a gasoline engine of the same output. It points out that acceleration is identical.

If turbocharged engines are used, a given power range may be covered with a much smaller number of basic engine models. For instance, Fig. 17 shows a power output range of 45 to 110 bhp being covered by 4 naturally aspirated engines and by 2 basic engines supplemented by turbocharged versions.

Fig. 18 and Fig. 19 show the main dimensions of a 2.500 pounds car with two different power plants with the same output. The calculations were performed for three safety levels. The smaller dimensions for the car especially of the engine compartment and the increased deformation zone for the 4 cyl. turbocharged application versus the 5-cyl. naturally aspirated application are quite obvious. On the other hand the additional cost for a turbo-charged engine should not be neglected.

Ceramic Materials

By using ceramic materials, it is possible to modify the transfer of heat as well as wall temperatures. As they are highly resistant to erosion and temperature, these parts are especially well suited for use in parts subjected to high loadings, such as swirl chambers, throats, and pistons. However, even if ceramic materials with their low heat transfer coefficient are used the efficiency of the engine will improve by no more than 2 -3 % at full throttle (12), which in this case is of no more than secondary importance. What is more important is that combustion quality is improved, and that the emissions of HC and of particulates are reduced.

One of the problem being investigated at the moment is how to join ceramic swirl chambers to metallic cylinder heads, and ceramic pistons to metallic connecting rods. Fig. 20 shows how a ceramic swirl chamber is mounted by means of an intermediate ring. Experiments run with ceramic swirl chambers brought improvements in cold-start and warm-up noise. Moreover, there is a surprising reduction in HC emissions, which can be explained by the fact that air fuel mixture formation is improved because the temperature of the combustion chamber is higher. However, a study of all engine characteristics shows that the entire combustion system (injection system and swirl chamber geometry) will have to be redesigned.

Clamping seems to be the most obvious way to connect ceramic pistons to their pistons rods, as the grooves which normally receive the piston pin clips might raise problems in connection with the ceramic material. As the clearance of pistons made entirely of ceramic material increases during operation, it seems most advantageous to combine a ceramic piston head (low noise and HC emissions) and a metal guiding skirt (narrow clearance).

The amount of increase in the temperature of individual components is limited by the temperature resistance of lubricants due to the lower heat transfer coefficient of ceramic materials.

Noise

To a certain degree, the interior noise level of passenger cars depends on the power plant. The noise generated by the engine may be reduced within certain limits by means of primary measures, such as changing the combustion chamber configuration, piston tolerance, valve gear design, and lowering the rated engine speed.

Fig. 21 shows several ways in which engine noise can be reduced:

1. Silent auxiliaries
2. Partial encapsulation of the power plant
3. Complete encapsulation of the power plant.

The only way to reduce noise emissions extensively seems to encapsulate the power plant entirely.

This could be done in two ways:

1. If the engine is to be designed from scratch, it can be made to include the capsule as an integral part. A good opportunity is offered by the cylinder block and crankcase, which is reduced to a skeleton, and its U-shaped oil sump (13). However this would grow longer and would require more room for installation.
2. It is simpler to encapsulate completely a conventional engine. We assumed a complete capsule and analyzed the influence of casing, padding, air intake and exhaust silencers, and ventilation apertures, all dimensions being designed in accordance with the thermal conditions of the engine.

Fig. 22 shows a detailed design drawing of concept A 3b. The wall of the capsule, which is padded with absorbent material, is located between the engine and the body wall and connnected to the latter via rubber elements. We used parts of the body itself wherever scarcity of space did not permit any other solution.

The cover of the capsule is connected to the bonnet by rubber springs, so that the capsule opens widely as soon as the bonnet is opened (maintenance). The bottom of the capsule, which is detachable, is held in place by latches.

The cooling air flow could be guided in two different ways:

1. The capsule may be cooled by a separate unit. In this case, the air-to-water cooler is located outside the capsule itself, which will be cooled by an additional fan.

2. The entire flow of cooling air is guided through the capsule after leaving the radiator, which in this case would be located in the cooling air intake aperture of the capsule.

The last-named alternative was favored because of cost and installation space reasons. The intake and exhaust ducts for the cooling air are designed as silencers.

Emissions

Passenger cars powered by modern Diesel engines meet all statutory HC and CO emissions standards without aftertreatment (14). However, NO_x emissions are still problematic. Although they may be reduced by retarding the injection timing (lowering the peak temperature), there is little elbowroom here as this measure will make the HC and especially the smoke emissions rise, deteriorating fuel economy. Therefore a special significance attaches to improving air fuel mixture formation by modifying the swirl chamber geometry, the fuel delivery curve, and the injection profile. Moreover, to ensure continued good fuel economy injection timing would have to be considerably more accurate.

Turbocharging is of some advantage in this respect, as it lessens the effects of ignition delay. On the other hand, it is more difficult to control engine performance at low partial loads in engines fitted with boost controlled injection pumps, as the charge pressure is low, and quantities of fuel injected can vary more widely. Moreover, there must be more precise injection timing and there must not be secondary injections.

The emissions of NO_x may be reduced only within a certain range by retarding injection. Therefore additional devices are necessary. But one has to keep in mind the increased HC emissions caused by them. The three most important measures are:

- Using ceramic materials (to shorten the warm-up phase).
- Exhaust gas recirculation
- Water injection.

Variable compression ratio pistons (16) do offer some advantages when used in research prototypes, but they seem probably difficult to use in large-scale production especially of low-capacity passenger car engines. To get a favorable swirl chamber-combustion chamber ratio we kept the distance between piston head and cylinder head as small as possible (min. 0.5 mm). Chamber ratio, combustion process, injection profile, and injection timing vary with the compression ratio. This special type of piston used necessitates additional tolerances which either influence the combustion process (much wider chamber ratio variation), or else the tolerances of the other parts are narrowed down to extreme limits.

At present, ceramic materials cannot be used for large-scale engine production. Their main effect would be to lower HC emissions and noise.

Compared to that of a gasoline engine, exhaust gas recirculation requires much more expenditures as HC emissions, smoke emissions, and fuel economy are affected drastically. Moreover, the manifold vacuum is very low. In order to avoid a too-extensive performance drop, EGR must be timed, and exhaust gases must be cooled. The service life of these devices still is something of a problem.

Water injection permits improving NOx emissions considerably while retaining good performance and fuel economy. Fig. 23 shows a map of an engine fitted with water injection. It is especially effective to inject the water before the compressor of the turbocharger, as this improves the efficiency of the turbocharger itself. The worst problems to be overcome in applying water injection are those of practicability (two fuels) and of operation at temperatures below 0^{o} C, as any use of ant-freeze additives could immediately cause additional HC-emission trouble.

Conclusion

With modern high-speed swirl chamber Diesel engine designs (especially when turbocharged) it is possible to attain power-to-weight and power-to-volume ratios similar to those of present gasoline engines. Moreover, these engines feature good fuel economy and will meet emission level standards of HC/CO/NO_x 0.41/3.4/1.5 . Table 2 shows the fuel economy ratings which could be attained with those engines in cars of the 2,250 lbs and 2,500 lbs weight class.

Power Plant	Inertia Weight 2,250 lbs	Inertia Weight 2,500 lbs
4 Cyl. Diesel Engine Naturally Aspirated	44 mpg	40 mpg
4 Cyl. Diesel Engine Turbocharged	51 mpg	46 mpg
4 Cyl. Diesel Engine Naturally Aspirated Fuel Economy Optimized Drive Train	56 mpg	52 mpg
4 Cyl. Diesel Engine Turbocharged Fuel Economy Optimized Drive Train	60 mpg	56 mpg

Table 2- Composite Fuel Economy Ratings

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M. Willmann

Fig. 23- Fuel Economy Map 70 bhp Diesel Engine

Turbocharged with Water Injection

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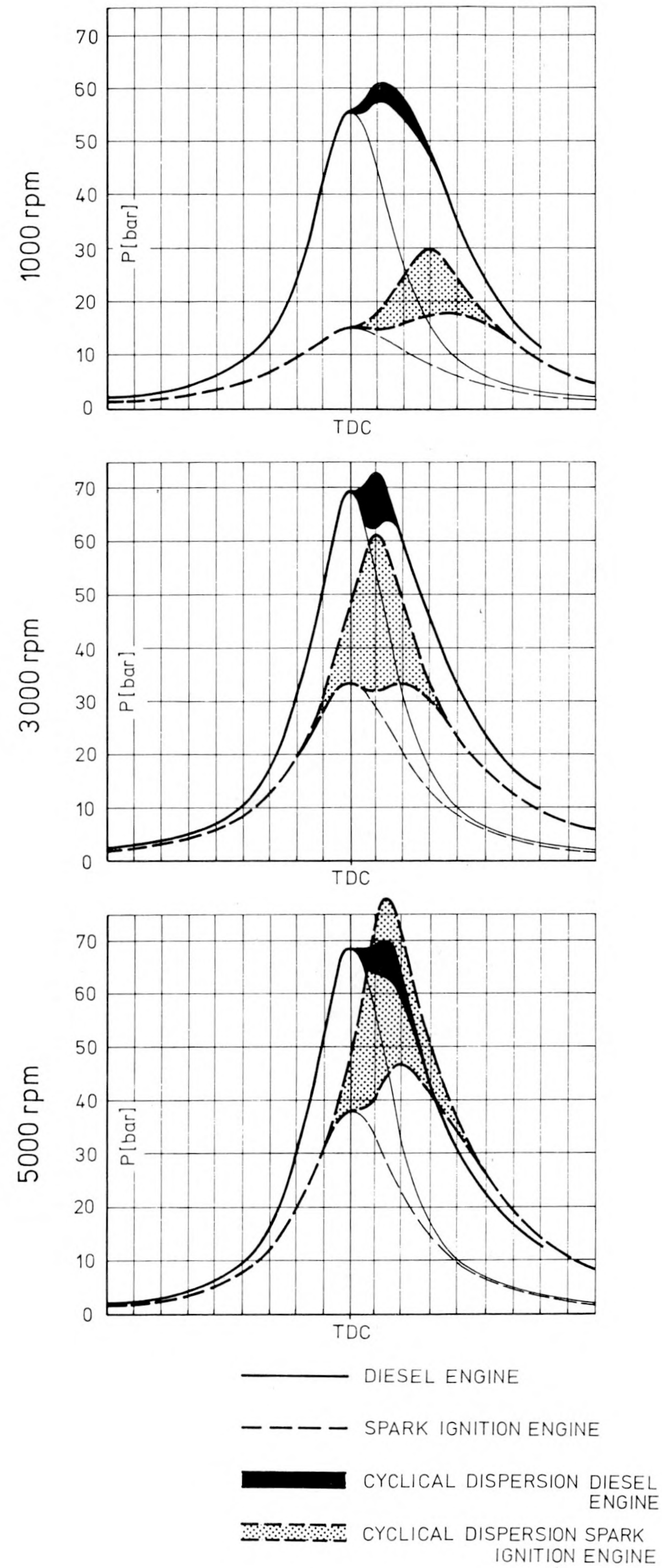


Fig. 1- Pressure Comparison of Spark Ignition and Diesel Using the Same Internal Engine Components

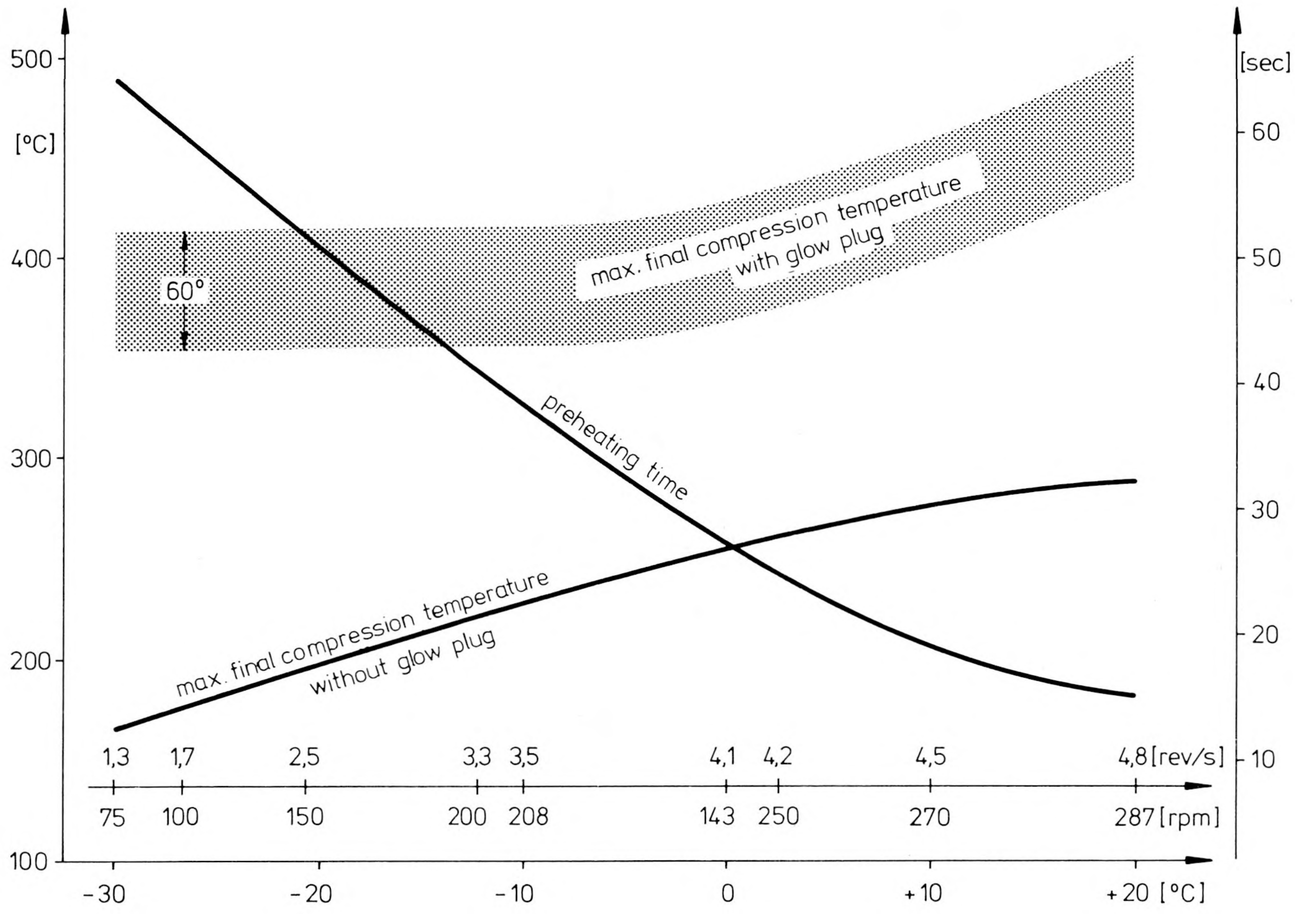


Fig. 2- Swirl Chamber Temperature at Starting

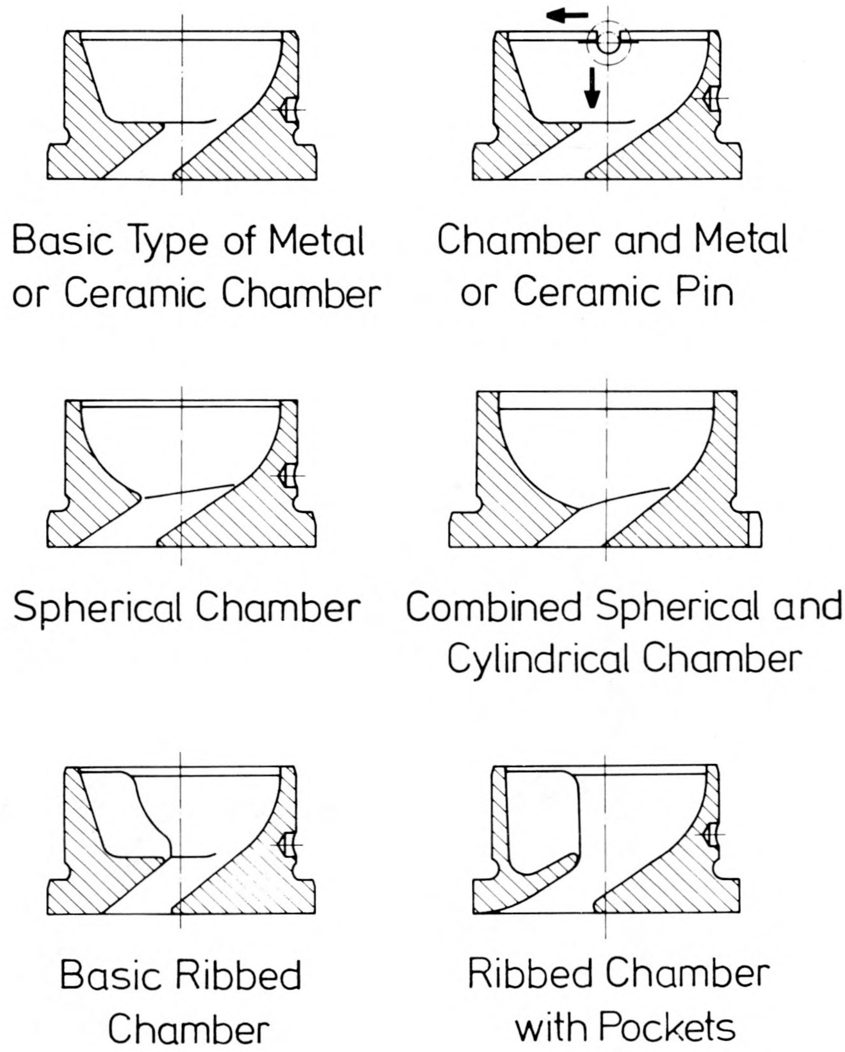


Fig. 3- Swirl Chamber Varations

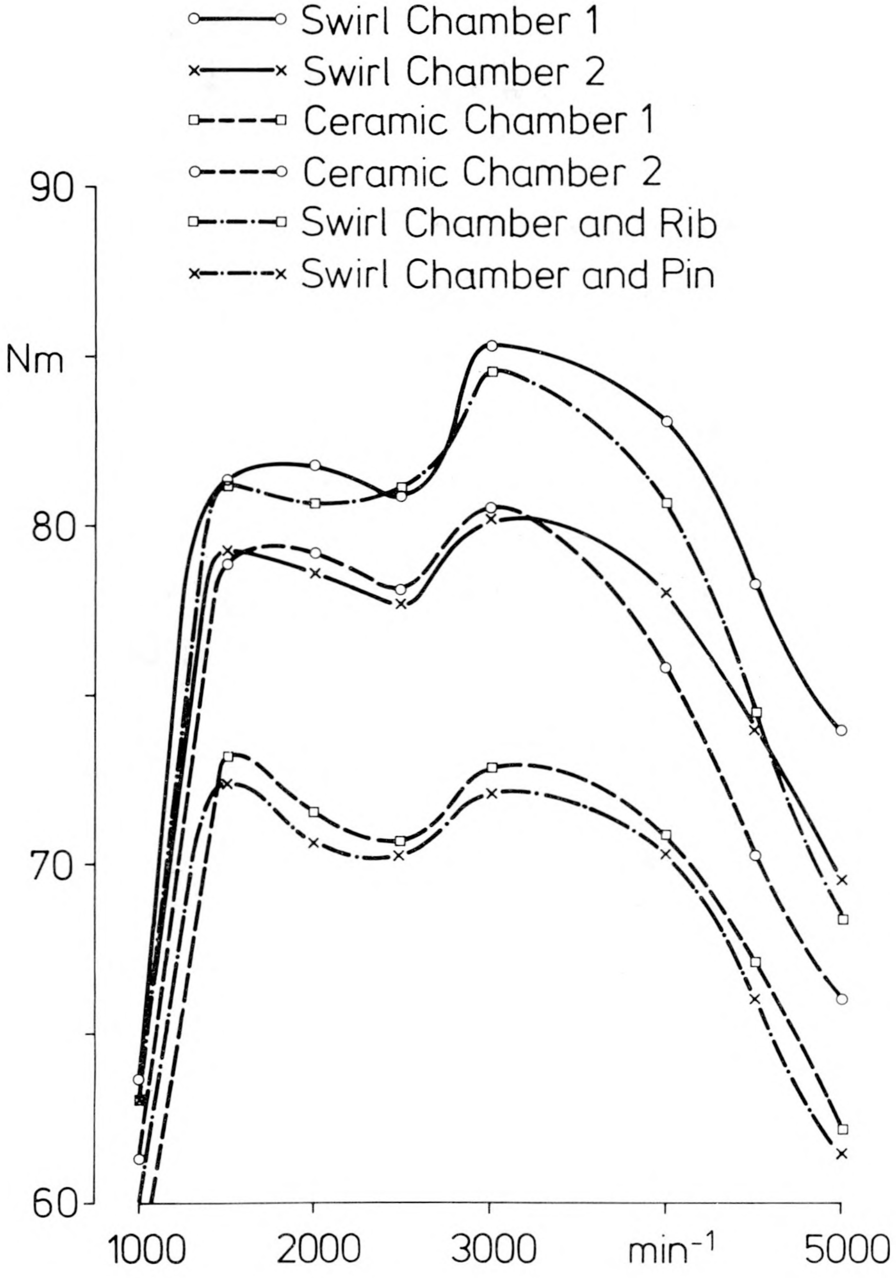


Fig. 4 - Torque Comparison of Swirl Chamber Variations

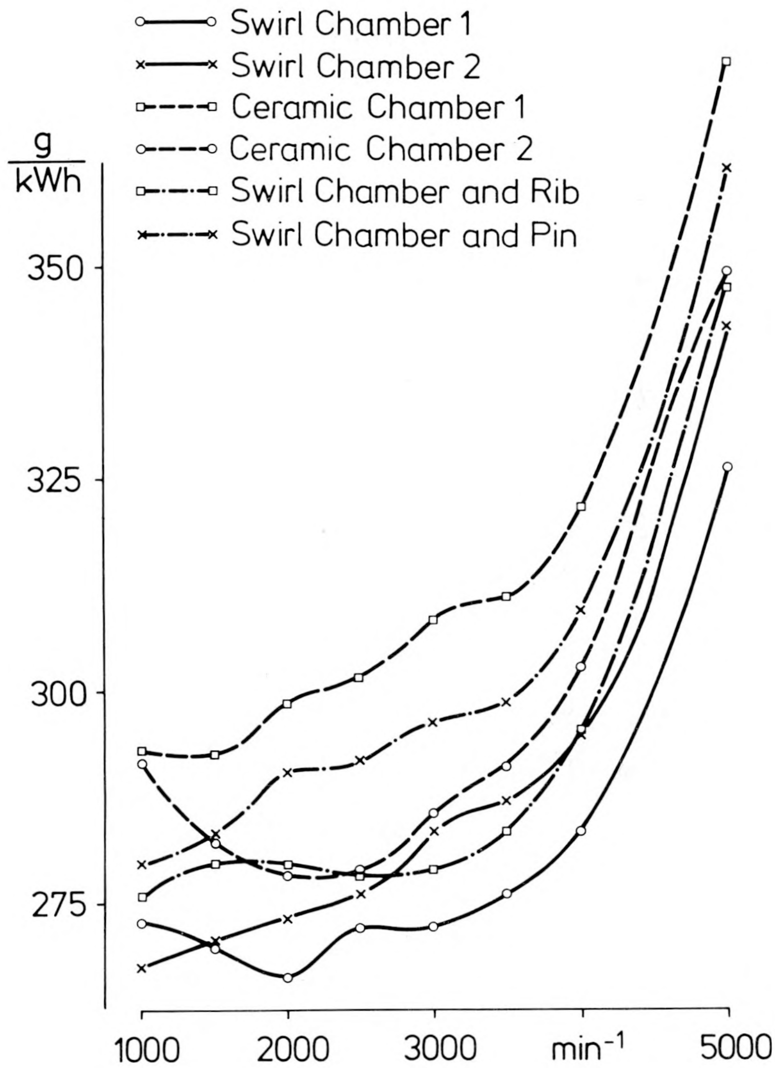


Fig. 5 - Fuel Consumption Comparison (Full Load)

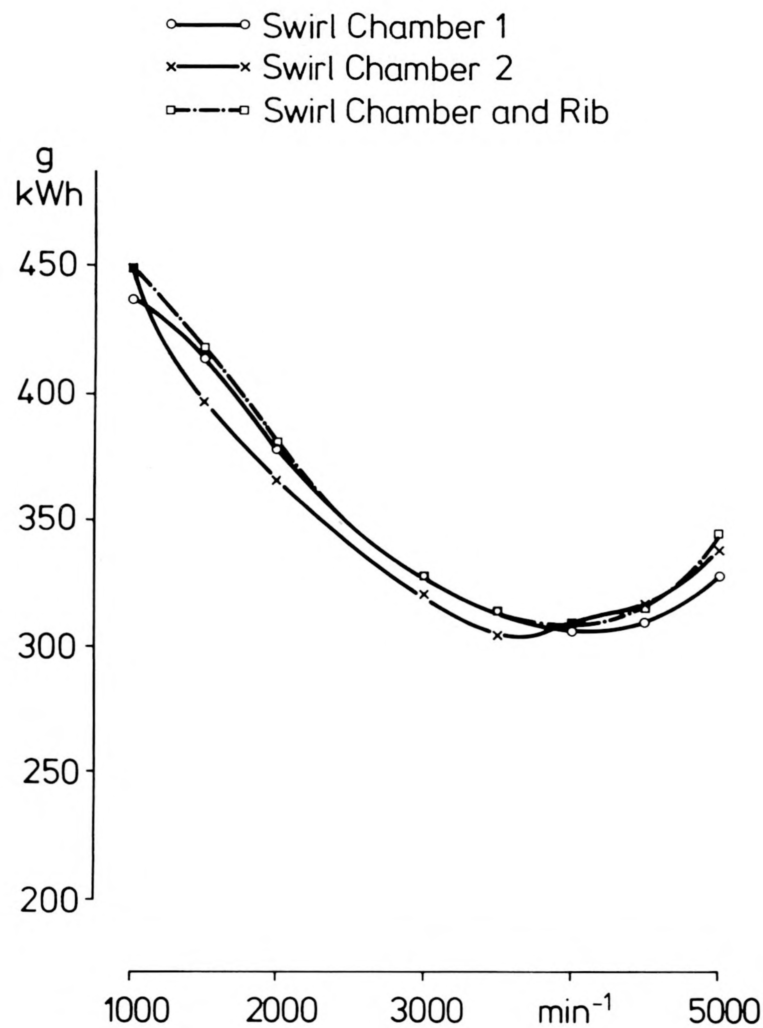


Fig. 6 - Fuel Consumption Comparison (Part Load)

Manufacturer Model	Displacement litre	Power kW	Speed rev/s	Weight kg	Weight to Power kg/kW	Power to Volume kW/litre
Peugeot XDP 4.88	1.946	44.7	75	170	3.8	23
Opel 2100 D	2.068	44.7	73.3	199	4.5	21.6
Nissan SD 22	2.164	48.4	66.7	186	3.8	22.4
Mercedes 240 D	2.404	48.4	70	202	4.2	20.1
Mercedes 240 3.0	3.005	59	67	234	4.0	19.6
Volkswagen 1,1 l* Spark Ignition	1.093	37	100	96	2.6	33.9
Volkswagen 1,6 l Spark Ignition	1.588	55	93.3	128	2.3	34.6
Volkswagen Rabbit D	1.471	37	83.3	120	3.2	25.2

Fig. 7-- Power to Weight Ratio of Engines for Passenger Car Application

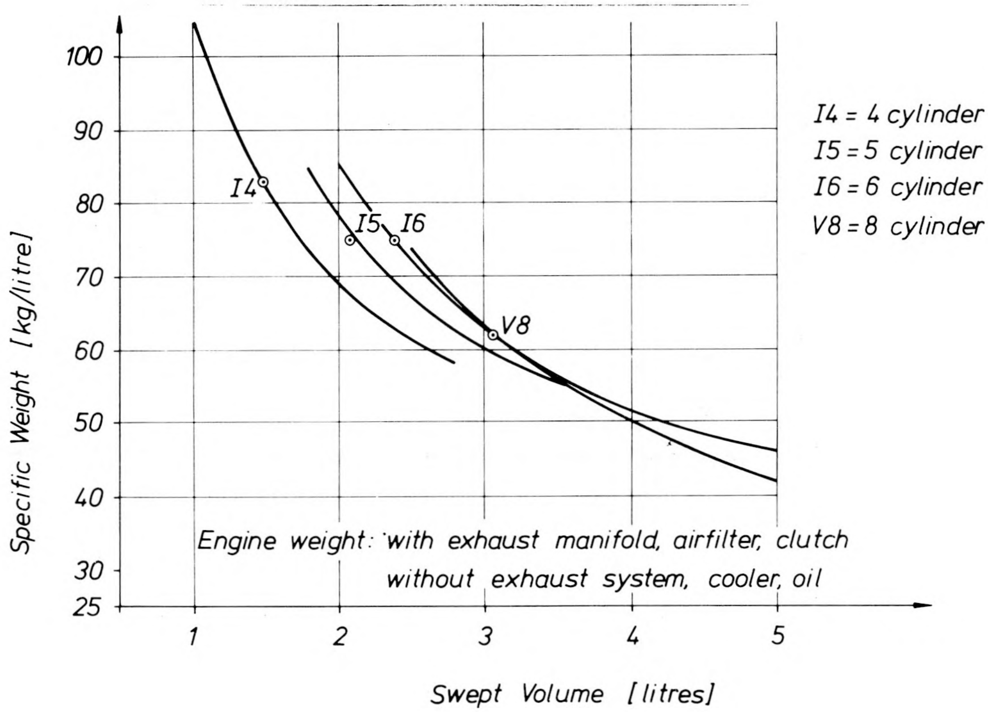


Fig. 8 - Specific Weight Versus Swept Volume, and Number of Cylinders

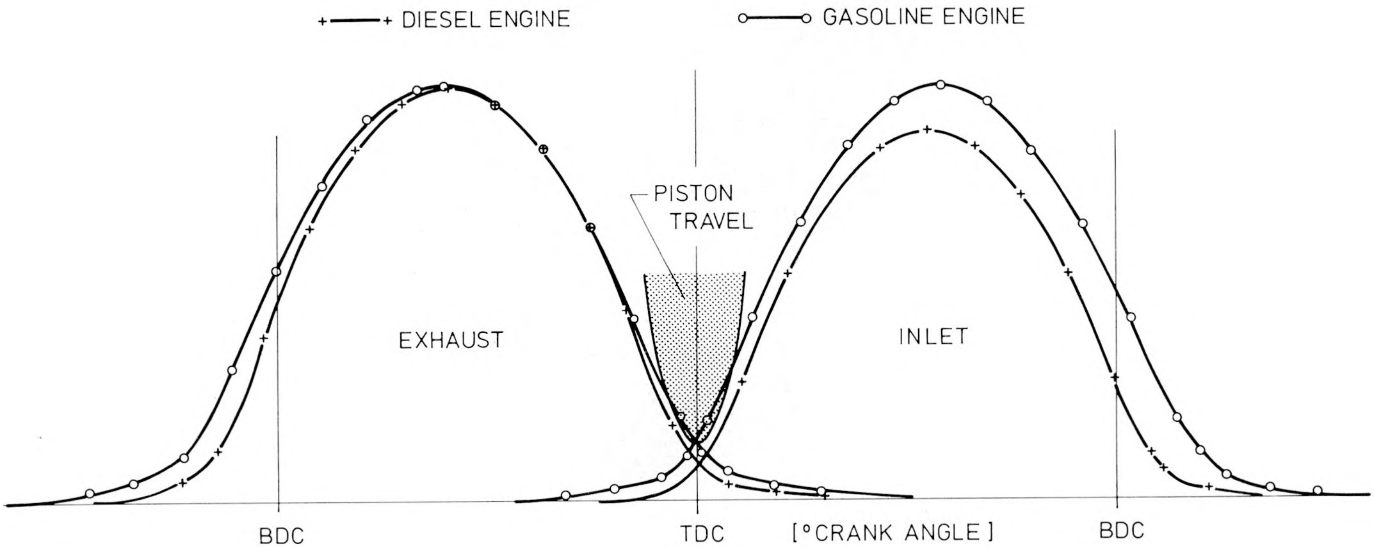


Fig. 9- Valve Travel Curve

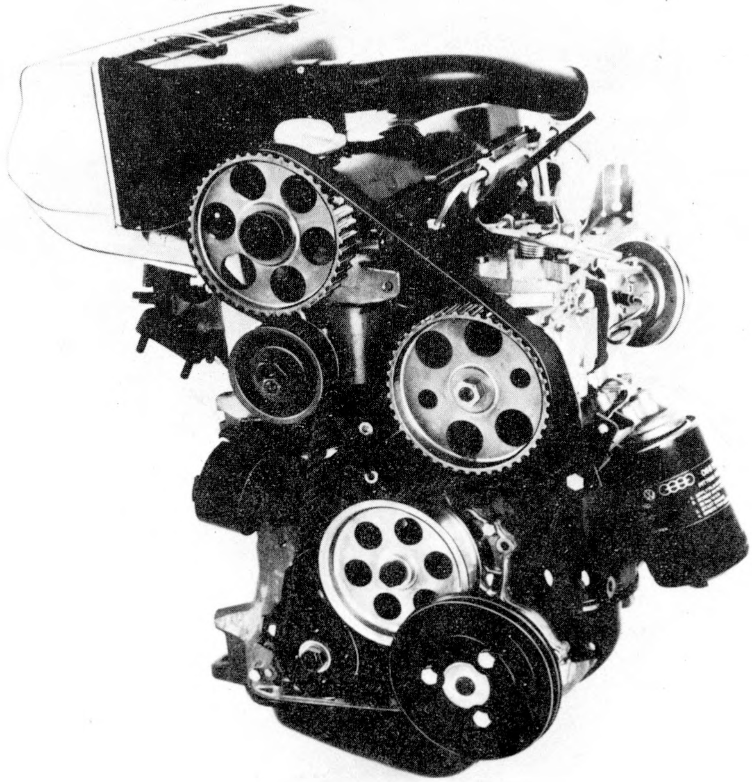
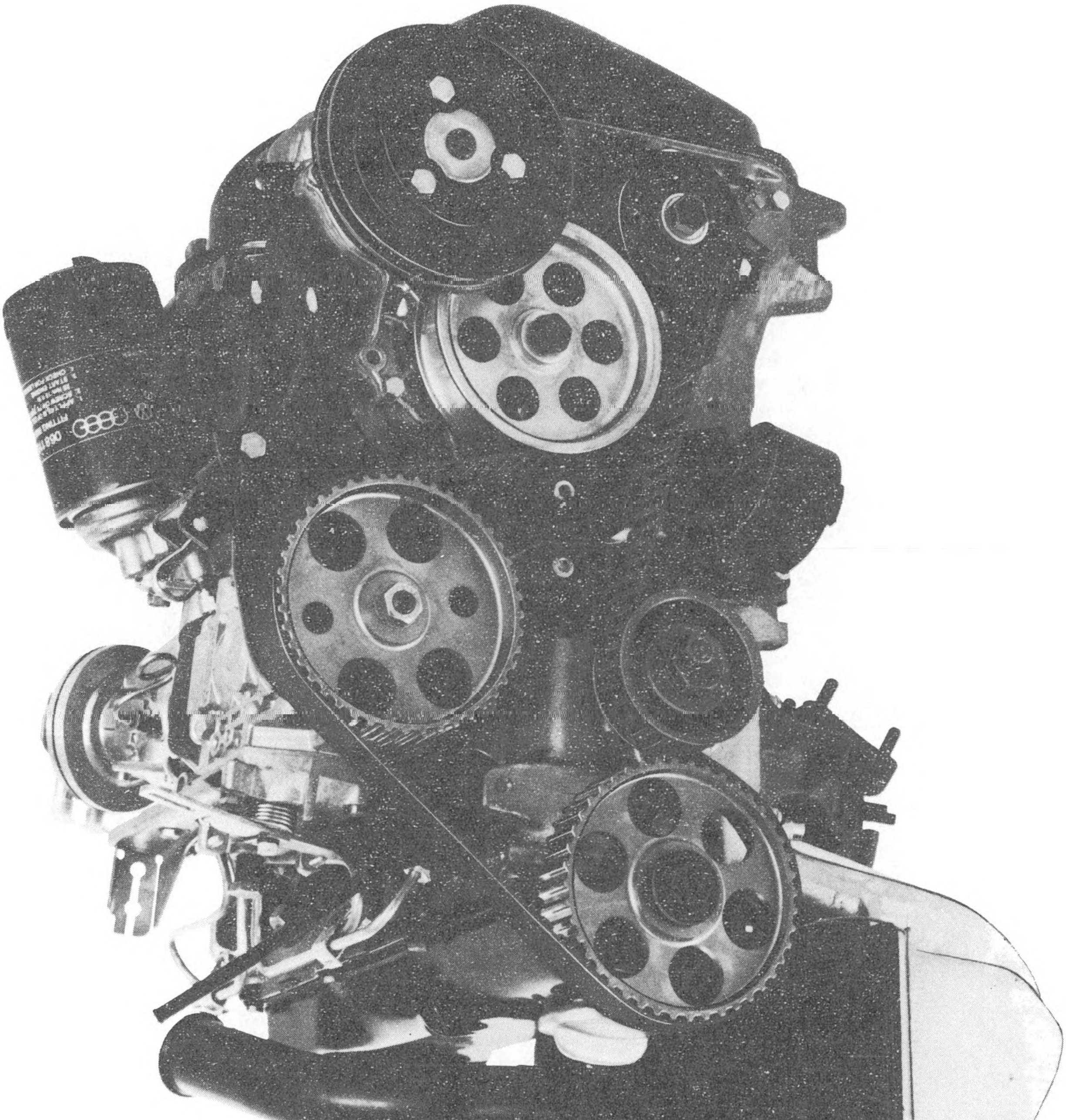


Fig. 10- Toothed Belt Drive



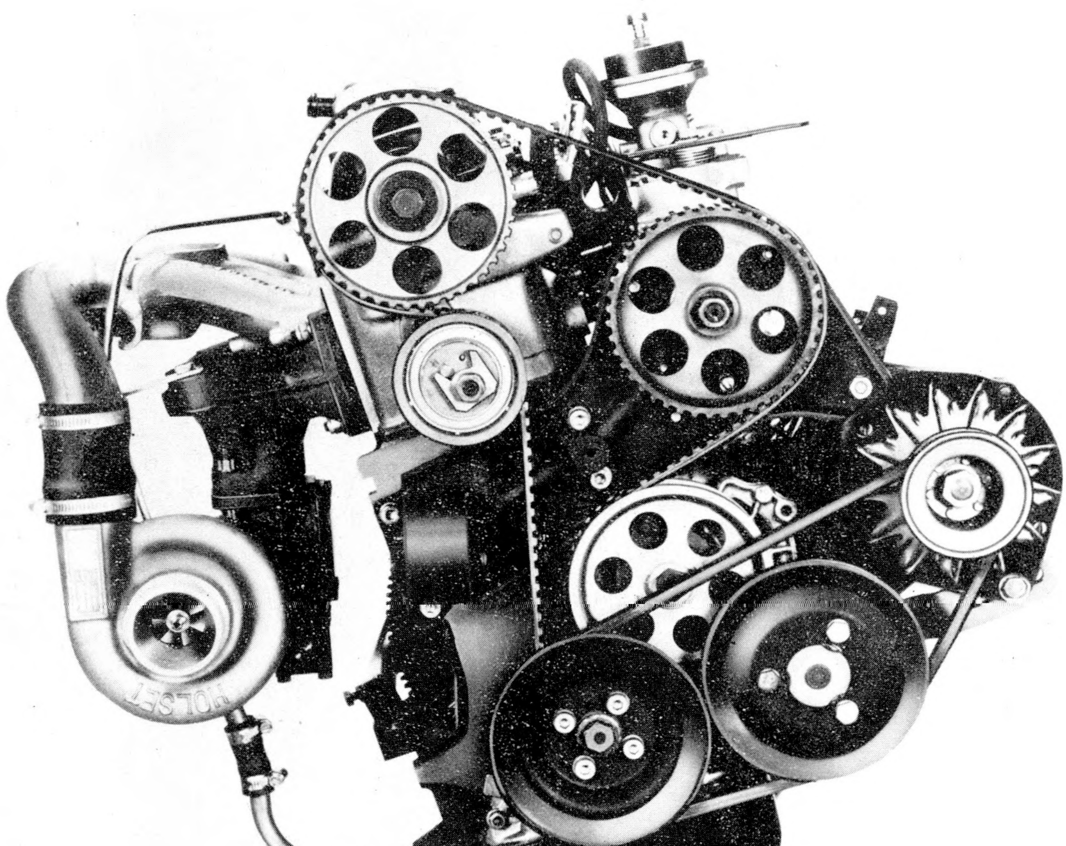
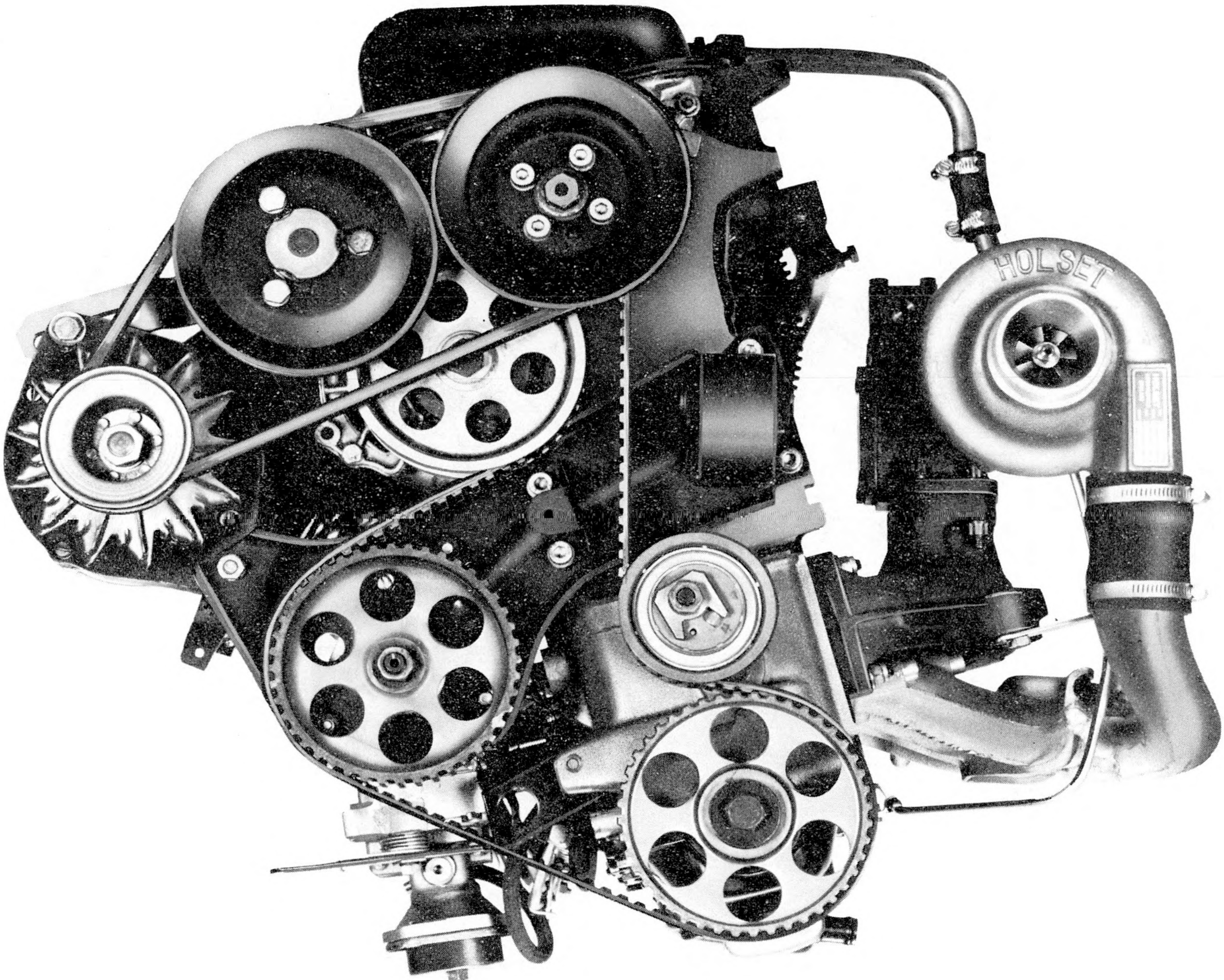


Fig. 11- Turbocharged 1,5 l Diesel Engine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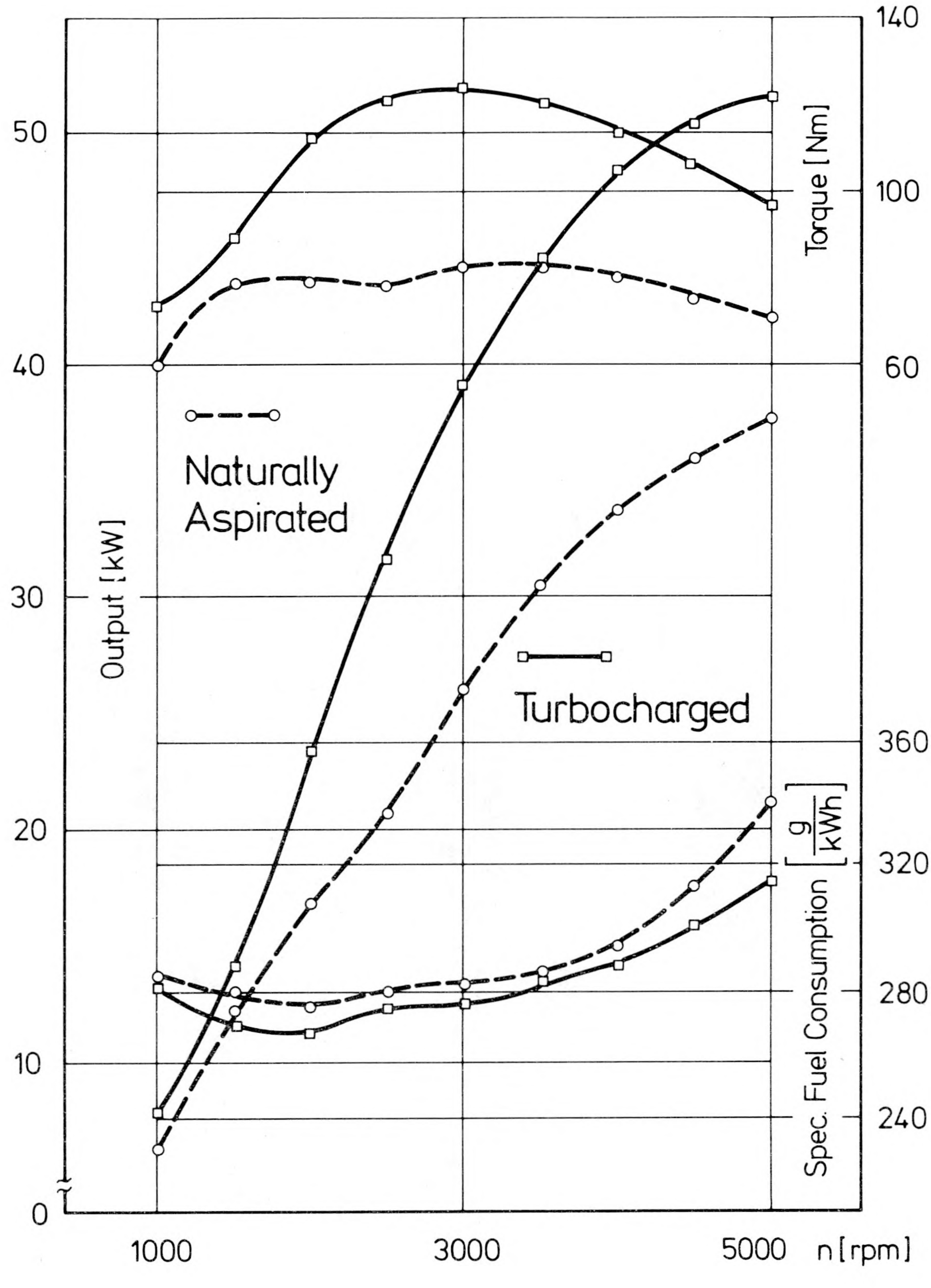

Fig. 12- Comparison Naturally Aspirated Engine versus Turbocharged Engine

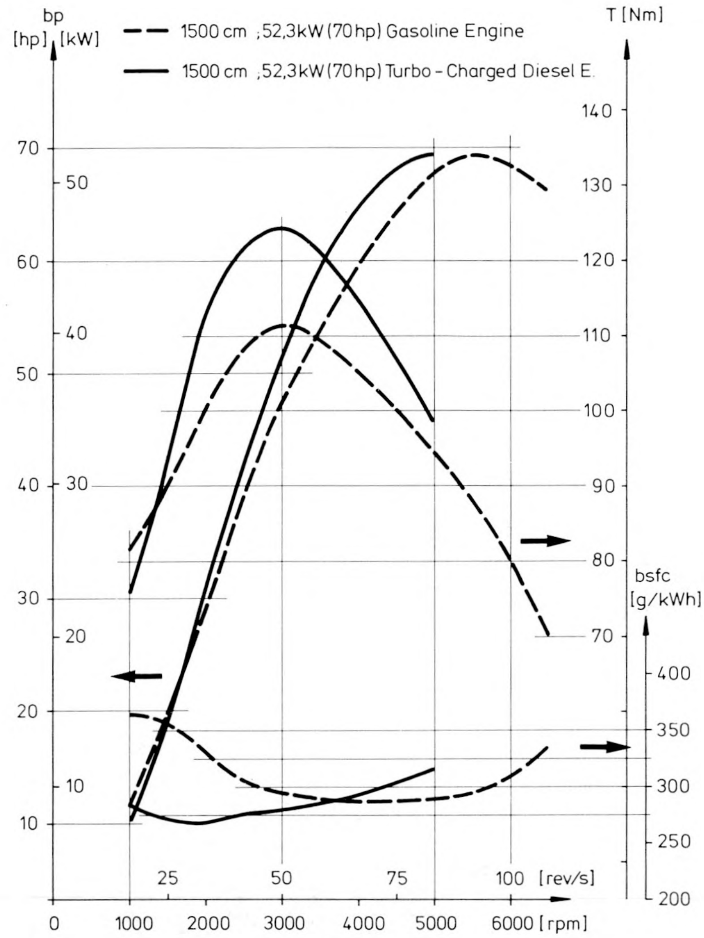

Fig. 13- Comparison 70 BHP Spark Ignition Engine Versus 70 BHP Turbocharged Diesel

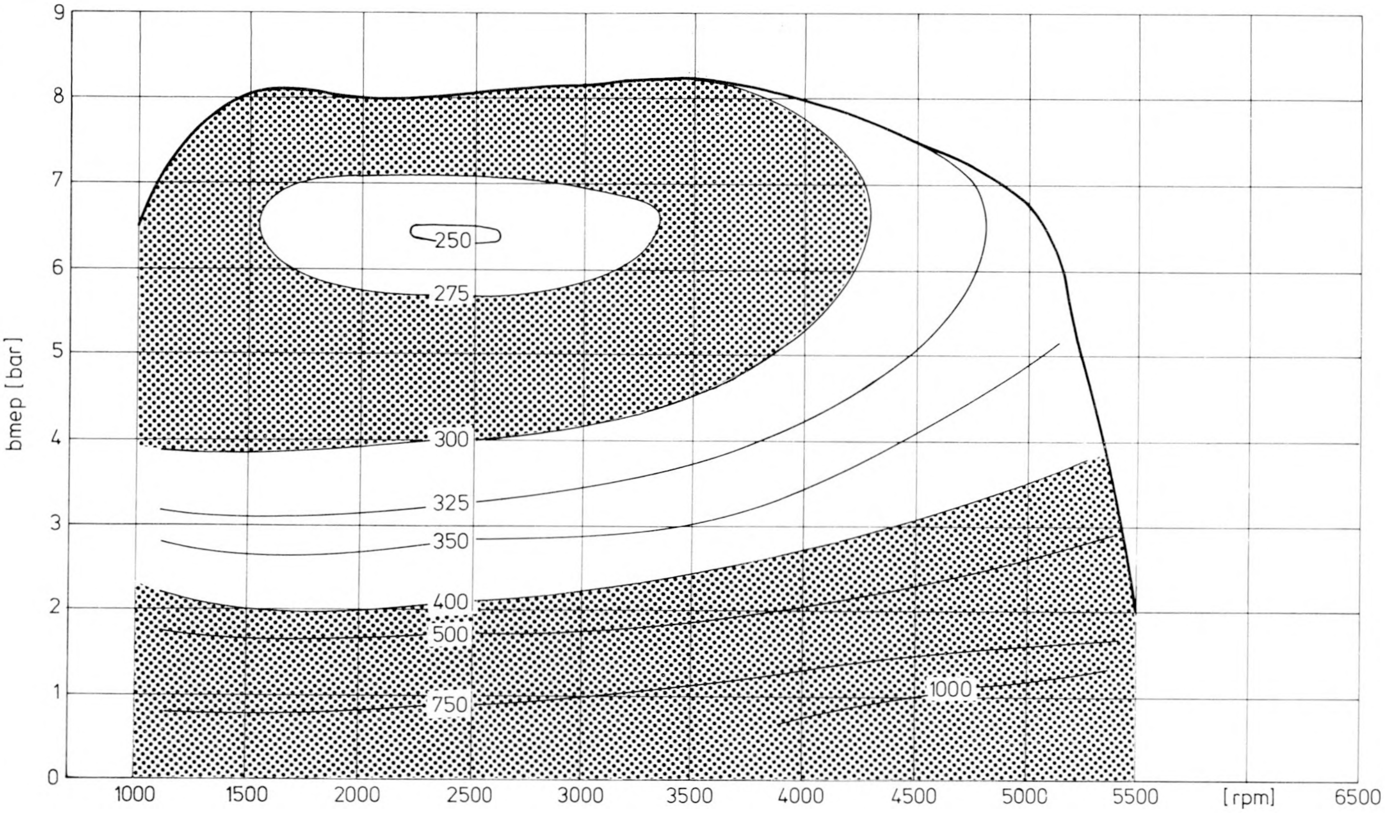

Fig. 14- Fuel Economy Map 50 BHP Diesel Engine Naturally Aspirated

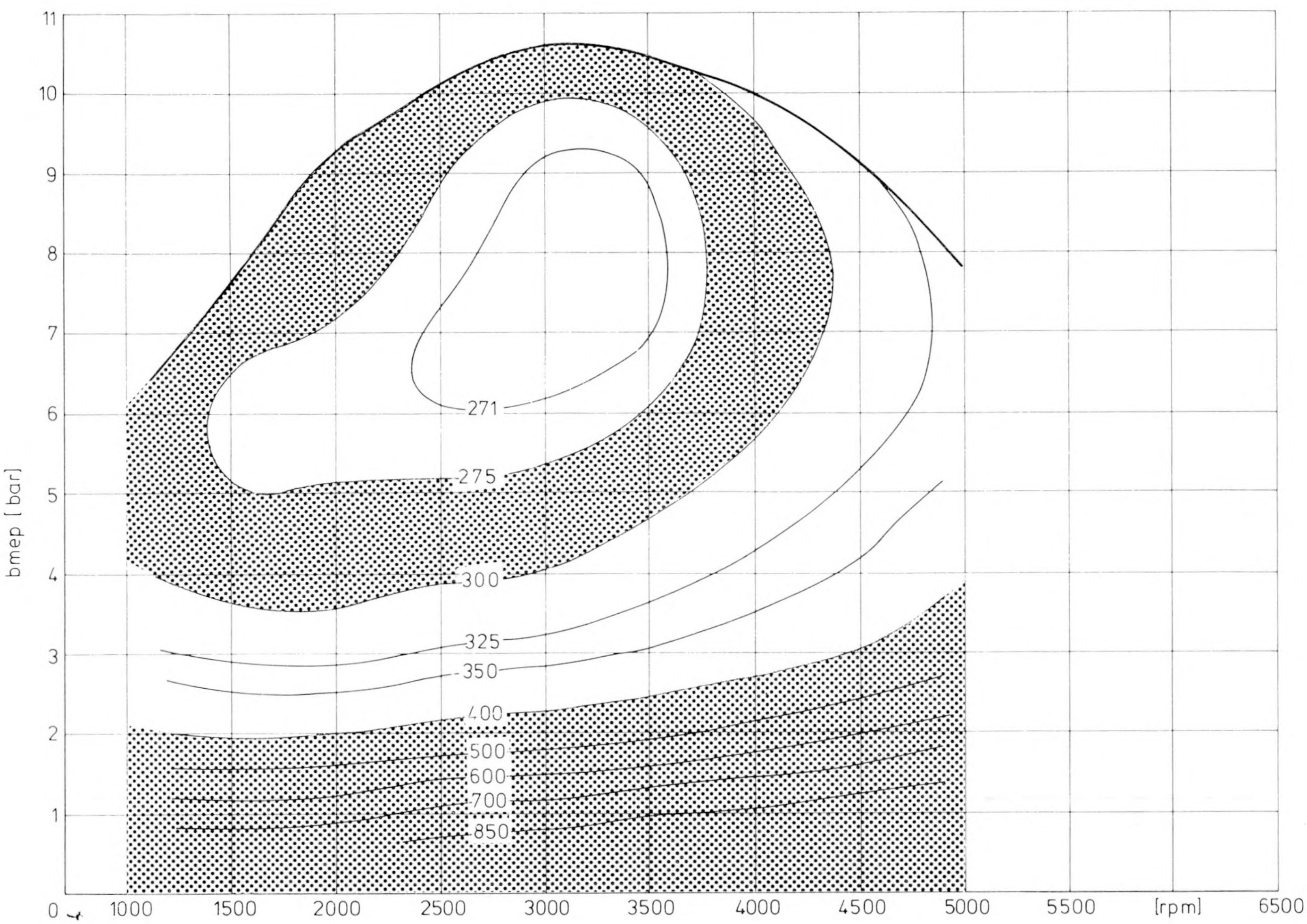

Fig. 15- Fuel Economy Map 70 BHP Diesel Engine Turbocharged

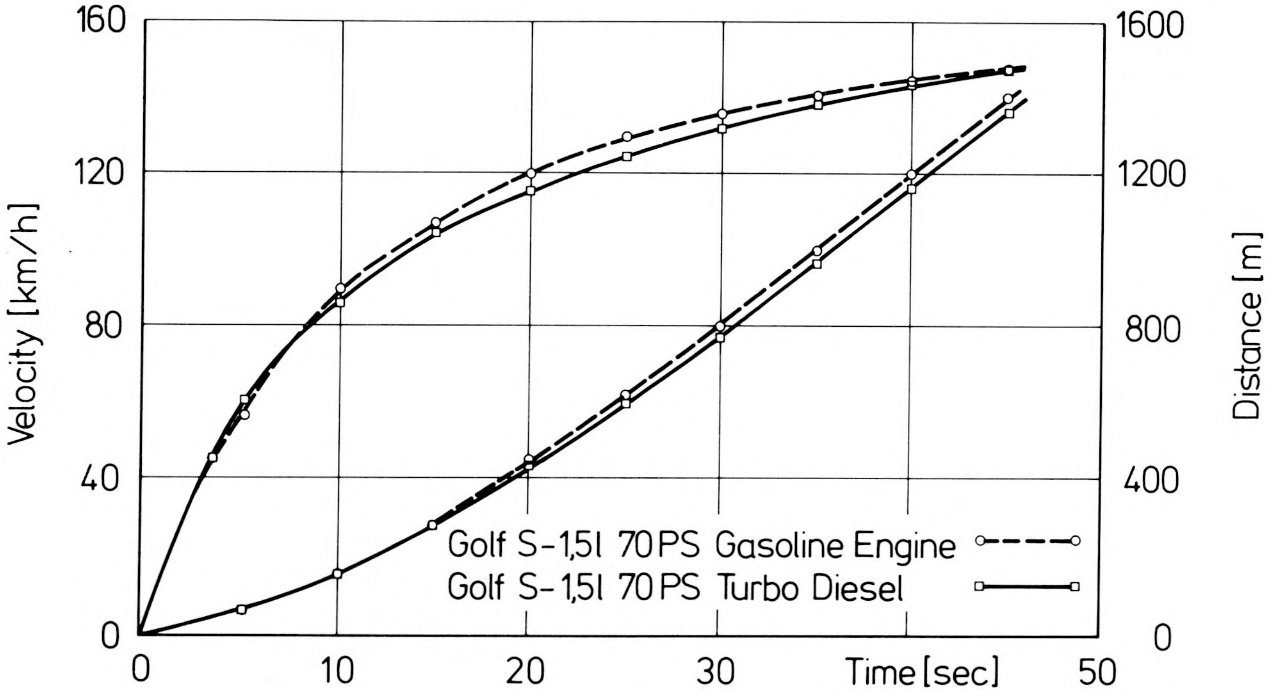

Fig. 16- Performance Comparison

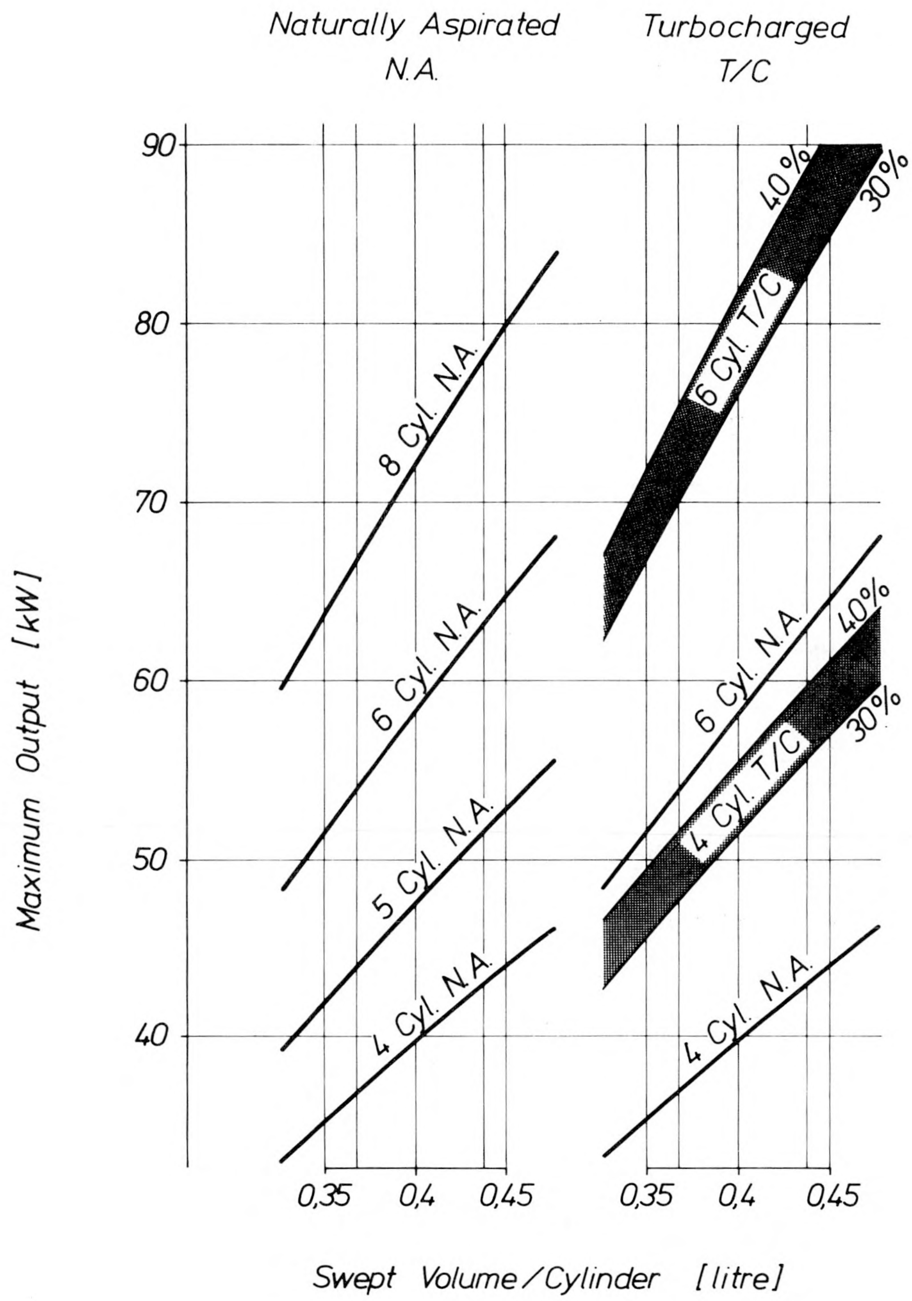

Fig. 17- Comparison of Two Engine Families

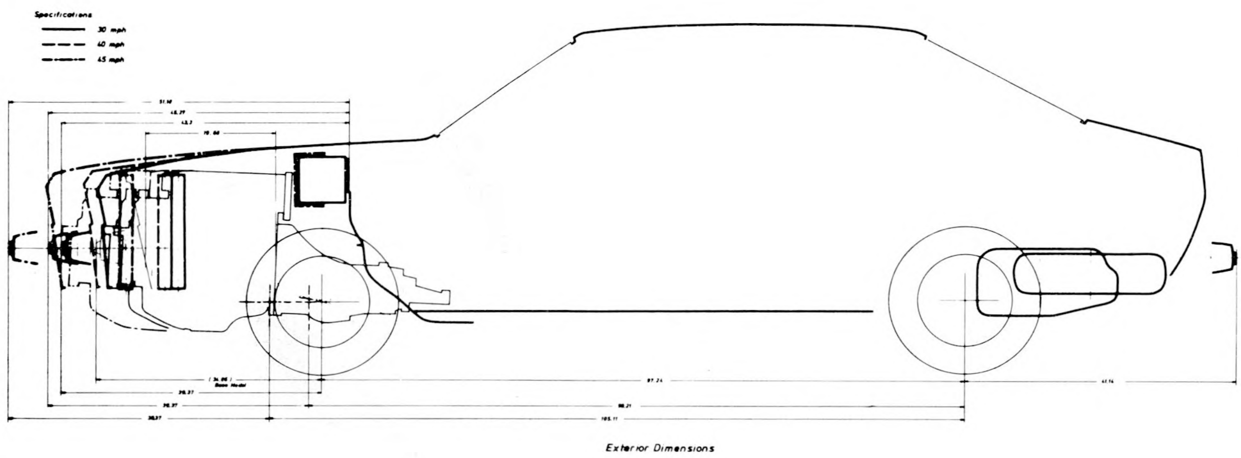

Fig. 18- Front Engine Longitudinally Installed,

Naturally Aspirated 5-Cylinder Engine

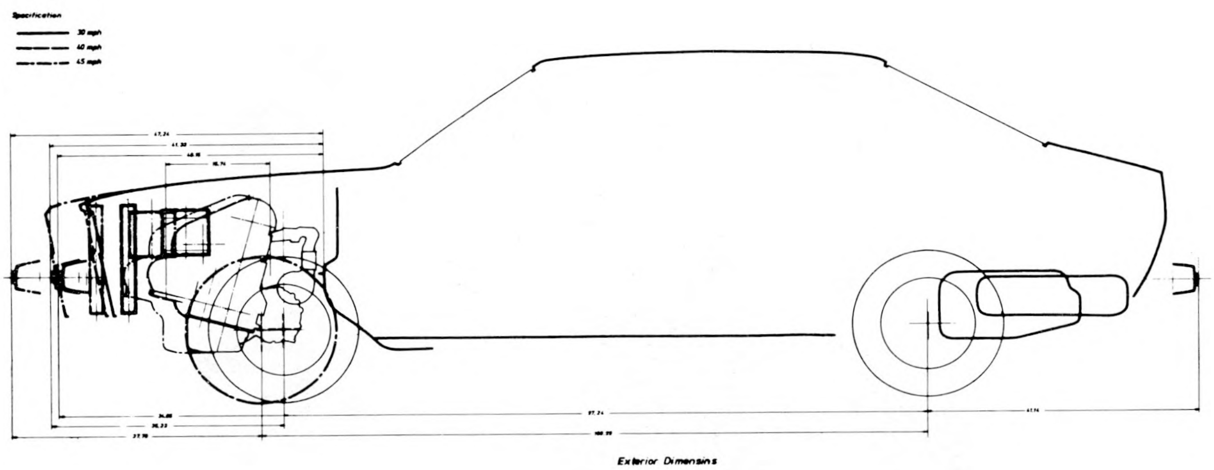

Fig. 19- Front Engine Transversally Installed

Turbocharged 4-Zylinder Engine

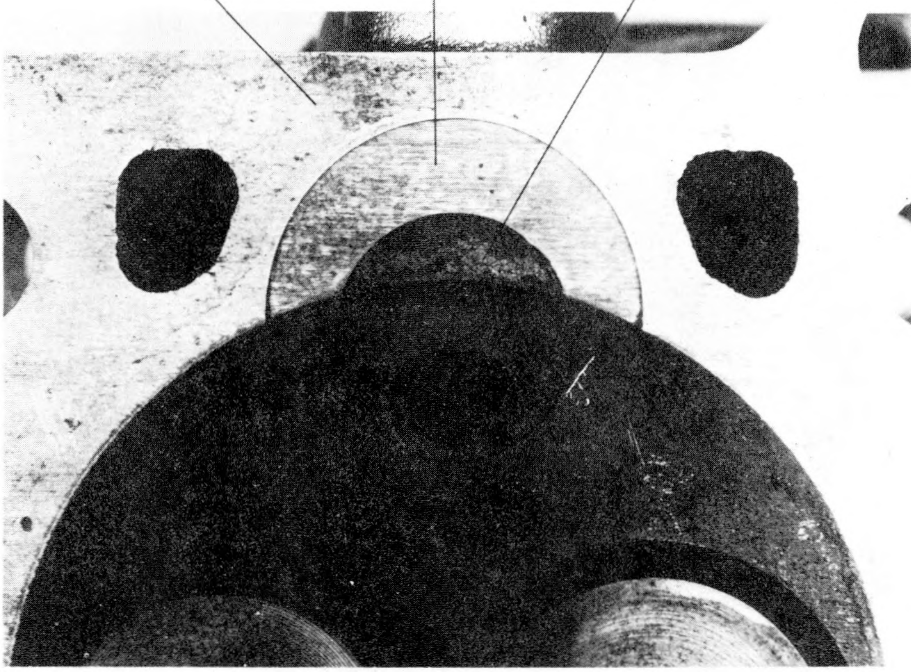

Fig. 20- Ceramic Swirl Chamber

Power Plant	Silencing Measure		
A Existing Engine + Modified Process	1. Silent Auxiliaries		
	2. Partly Encapsulated	a. Close to E.	
		b. Away from E.	
	3. Fully Encapsulated	a. Close to E.	
		b. Away from E.	
B Newly Designed Power Plant	1. Integrated Partial Capsula		
	2. Integrated Total Capsula		

Fig. 21- Engine Noise Reduction

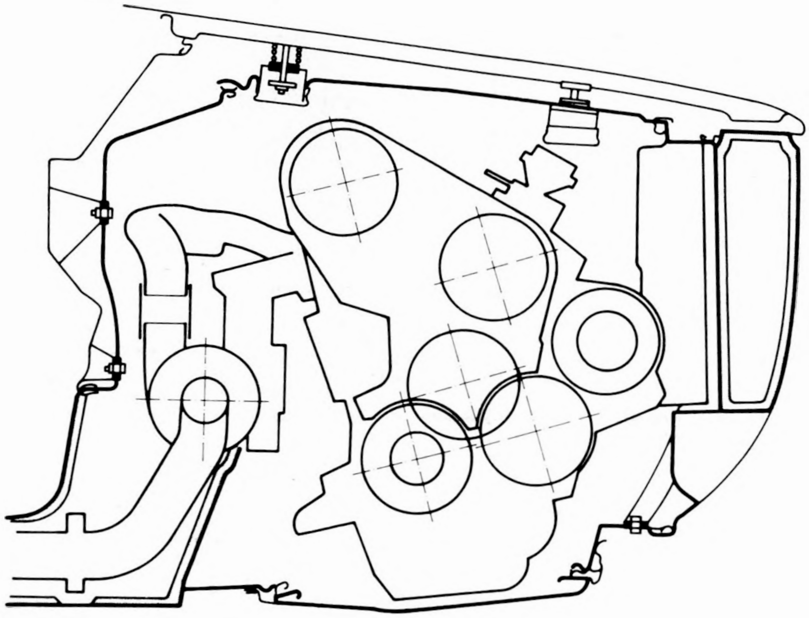

Fig. 22- Encapsulated Power Plant

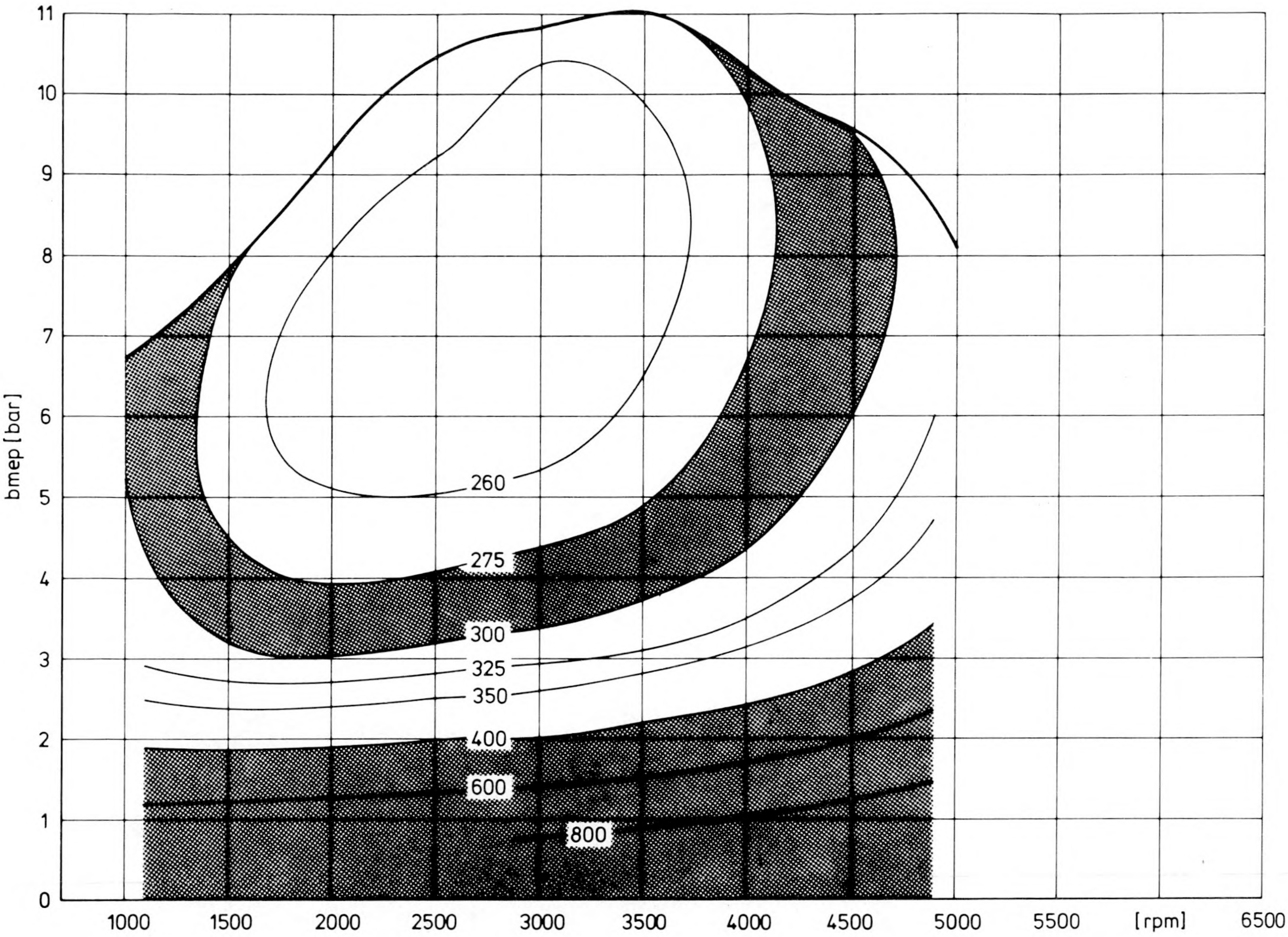

Fig. 23- Fuel Economy Map 70 BHP Diesel Engine
Turbocharged with Water Injection