GASEOUS MOTOR FUELS. AN ASSESSMENT OF THE CURRENT AND FUTURE STATUS.
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The Engine Fuels Subcommittee of the API Committee of Environmental Affairs undertook an assessment of published information on the potential impact of adoption of gaseous fuels to reduce automotive emissions from existing and future motor vehicles as a means of achieving improved air quality in the 1975-80 period. The major considerations of vehicle emissions, fuel and equipment costs and availability, and overall impact on emissions from all sources provided the basis for the report. This study found that with favorable costs for the gaseous fuels and favorable fuel tax consideration coupled with fuel availability, and inherently lower maintenance, converted fleets can show an economic advantage. Conversion of older models to gaseous fuels can result in reduced emissions for those vehicles, but will not realize significant benefit for the current (1973) model vehicles, nor those anticipated to meet the 1975-76 Federal standards. Therefore, the impact of such conversion on air quality is expected to have marginal impact in the 1975-80 period. If the gasoline engine can meet or even approach the 1975-76 emissions limits, conversion of earlier models to gaseous fuels will not provide improved air quality during the next decade except under very special environmental and economic conditions.

FUEL VOLATILITY AS AN ADJUNCT TO AUTO EMISSION CONTROL. R. W. Hurn, Dennis B. Eccleston, and Barton H. Eccleston, U.S. Department of Interior, Bureau of Mines, P.O. Box 1398, Bartlesville, Okla. 74003

Late-model vehicles were used in an experimental study of the interaction of fuel volatility with emissions and associated fuel economy. Volatility characteristics of the test fuels ranged between 7 and 14 pounds Reid vapor pressure; between 130° and 240° F 50% point; and between 190° and 370° F 90% point. Choke settings of each vehicle were adjusted as needed for choke action appropriate to each fuel's volatility.

Midrange and back-end volatility were found to influence emissions significantly. The principal influence is upon emissions during cold start and warmup. Results show that, in general, hydrocarbon and carbon monoxide emissions are reduced by increasing either, or both, midrange and back-end volatility. Fuel economy during starting and warmup also was improved by increasing volatility in the midrange and back-end portion of the boiling curve. Within the vapor pressure limits traditional of U.S. fuels, vapor pressure and fuel front-end volatility were found to have only slight effect upon either emissions or fuel economy.

Pre-engine Converter

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I. INTRODUCTION

During the past decade, efforts to reduce vehicular pollutant emission have included suggestions for the removal of lead from gasoline or for use of alternative fuels such as H_2 and low molecular weight hydrocarbons which are known to have high octane values and good burning characteristics (1), (2). Lead removal, which is already being implemented, raises the octane requirement of the fuel. The increased severity required in refining to produce such high octane gasoline decreases the gasoline yield per barrel of crude and, therefore, increases crude oil consumption and demands more refining capacity in the face of an impending energy crisis. Use of low molecular weight hydrocarbons is difficult to implement because of safety hazards and lack of nationwide storage and distribution systems.

The concept of attaching a catalytic reactor to an internal combustion engine converting liquid hydrocarbons to gaseous fuel was disclosed in a patent issued to Cook (3) in 1940. Recently, it has received some renewed attention. Newkirk _et al_ (2) described their concept of an on-board production of CO_2/H_2 mixture by steam reforming of gasoline fuel. A U.S. patent was issued to W. R. Grace Company in 1972 (4) on a mobile catalytic cracking unit in conjunction with a mobile internal combustion engine. In 1973, Siemens Company (5) of Germany announced a "splitting carburetor" which breaks up gasoline and related fuels into burnable gases. The jet propulsion laboratory of NASA (6) is investigating the concept of the generation of hydrogen for use as an additive to gasoline in internal combustion engines.

While little technical data are available, these developments appear to represent different approaches of adapting established industrial catalytic processes designed for a narrow range of operating conditions to moving vehicles which must operate from idling to full throttle.

To design a reactor system capable of operating satisfactorily under full throttle conditions requires either a large reactor or an unusually active and efficient catalyst. A standard 300 cu. in. automotive engine at full throttle consumes fuel at a rate of about 10 cc/sec. If the reactor were operating at 1-2 LHSV (vol/vol/hr.), i.e., at the throughput of an average industrial reactor, the engine would require a catalyst bed volume of 18-36 liters (4.8 - 9.5 gallons) - far larger than the carburetor it replaces. The necessity of a multi-reactor system for continuous operation plus accessory devices including the fuel preheater, etc., would make the system impractically bulky and too slow to warm up. Therefore, a workable system clearly depended on the discovery of new catalysts of high activity. To reduce the size of the reactor to that of a carburetor, an increase in catalytic activity by a factor of at least 50-100 is necessary.

In addition to the problem of the catalytic reactor volume, the life of the catalyst is also of critical importance. Most industrial catalysts require periodic oxidative regeneration in a matter of minutes after operation to maintain their effectiveness. A catalytic cracking catalyst, such as that proposed in the patent issued to Grace (4), requires frequent regeneration. An example described in this patent states that with a zeolite-containing catalyst, 2% of the fuel was converted to coke in the catalytic converter. We estimate that under the proposed operating conditions, each volume of catalyst could process no more than 10 volumes of fuel before sufficient coke (20%) would have deposited on the catalyst to deactivate it. Thus, even if the catalyst were active enough to operate at 100 LHSV, no more than 6 minutes of continuous operation between oxidative regenerations would be feasible.

In this paper we present experimental studies with a catalyst that overcomes many of these limitations.

II. EXPERIMENTAL

1. Equipment

To demonstrate the performance of the new catalyst system, the catalytic reactor was attached to a standard motor knock Test Engine, Method IP44/60 (7), bypassing the carburetor. Figure 1 shows a schematic diagram of the catalytic unit. The reactor consisted of two 3/4 inch O.D. x 18 inches stainless steel cylinders mounted vertically, one on top of the other, and connected in series. The top chamber serving as the preheater contained 82 cc of 3 mm diameter glass beads; the catalyst bed (5 inches long) in the bottom chamber consisted of 24 cc of 20/30 mesh catalyst mixed with 12 cc of 8/14 mesh Vycor chips. During thermal runs, Vycor chips were substituted for the catalyst.

Both cylinders were electrically heated. Liquid fuel flow was metered with a rotameter. Air-fuel ratio was monitored by Orsat analysis of the exhaust.

2. Test Fuels

Two types of feedstocks were used: (1) an 86 research octane (R+O) and 79.5 motor octane (M+O) reformate obtained from Mobil's Paulsboro Refinery containing: 23.4 wt. % n-paraffins, 33.9% branched paraffins, 1.2% olefins, 1.0% naphthenes and 40.5% aromatics, and (2) a Kuwait naphtha of 40.5 clear motor octane (M+O).

III. RESULTS AND DISCUSSIONS

(1) Upgrading of a C_5-400°F reformate

The experiments were carried out by charging the liquid fuel stored in a pressurized reservoir (4500 cc) at 38 cc/min. continuously for about 2 hours through the catalytic converter during which time the motor octane number of the reactor effluent was determined every 30 minutes. At the end of 2 hours, the reactor was cooled to 800°F with purge nitrogen while the fuel reservoir was being refilled. The experiment was then repeated. Two catalysts were examined, viz., a new stable zeolite catalyst (12 gms) and a commercial zeolite cracking catalyst (16 gms), which had previously been aged for 2 hours in a test described later. The feed rate corresponds to a weight hourly space velocity of 140 and 93, respectively. The reactor was maintained at between 910 and 920°F. Octane rating of the reactor effluent as a function of the cumulative on-stream time is shown in Figure 2. During the first 2 hours, the stable zeolite raised the octane number from the thermal value of 79.6 to 85 M+O. The octane dropped 2 numbers during the next two hours and maintained above 82 M+O for the next seven hours. The aged commercial zeolite catalyst, on the other hand, produced no appreciable conversion under the experimental conditions, i.e., at this high space velocity.

(2) Upgrading of a C_6-350 Kuwait naphtha

After 12 hours of operation without regeneration using reformate as the feed, the fuel was switched to the low octane virgin naphtha and the test continued over the aged stable zeolite catalyst for an additional two 2-hours runs before the experiment was terminated due to a mechanical malfunction. The result of the naphtha test is summarized in Figure 3. Shown in the same figure are the results over a fresh Durabead 8 catalyst and the thermal run. It is interesting to note that a boost of 22 motor octane numbers was registered by the aged stable zeolite catalyst while the fresh commercial zeolite cracking catalyst and the thermal run recorded a gain of only 10 and 6 numbers, respectively.

(3) Shape Selective Cracking

In addition to their excellent burning qualities, i.e., non-polluting combustion, light hydrocarbons have volume blending octane ratings ranged between 100 and 150 research clear numbers (R+O). Thus low octane liquids such as virgin naphtha and mildly reformed reformate can be upgraded by partially converting them to light hydrocarbons in the pre-engine converter, and feeding the entire reactor effluent directly into the engine.

A typical distribution of reaction products is shown in Table I for three samples collected when a blend of C_6 hydrocarbons was passed over the stable zeolite catalyst at 1 atm. and 900°F. The results clearly show that the catalyst exhibited preferential shape selective cracking in the order of n- > monomethyl- > dimethyl-paraffins. Thus isomers having the lower octane ratings are preferentially cracked. The C_4 minus cracked products are highly olefinic and some C_7^+ aromatics are formed by secondary reactions.

The added advantage of shape selective cracking in the order of octane rating is illustrated by the cracking of a 61 research octane Udex raffinate, a low octane product from the solvent extraction of aromatics from a reformate. In Figure 4, curve 1 shows the calculated octane number of the reactor effluent vs. wt. % liquid cracked to C_4^- light hydrocarbons. To produce a 91 R+O fuel, about 49% of the liquid is cracked. Examination of the composition of the raffinate shows that the straight chain paraffins having an average octane rating of 17 R+O represent 27% of the liquid. Curve II shows that when these n-paraffins are selectively cracked, the octane rating of the fuel can be boosted to $\sim$ 90 R+O with only 27% conversion. An ideal shape selective cracking would yield curve III which represents the most efficient route of upgrading. The octane rating of the fuel is boosted to 100 R+O with less than 40% conversion.

(4) Catalyst life and stability toward oxidative regeneration

Preliminary data obtained in bench scale micro-reactor (8) studies using the reformate over both the fresh catalysts and the regenerated catalysts showed that the catalyst was stable toward air regeneration and catalyst activity was restored after regeneration. At 100 WHSV, the catalyst appeared to have a useful cycle life of about 7 hours, corresponding to processing 700 pounds of fuel per pound of catalyst. At lower space velocities, the cycle life appeared to be much longer than 7 hours, although the amount of fuel processed over the same length of operating hours was less than that at 100 WHSV.

IV. CONCLUSIONS

A highly active, stable and shape selective zeolite cracking catalyst overcomes a major problem in the application of the concept of pre-engine conversion to a moving vehicle. The catalyst is active enough to operate at above 100 LHSV and 900°F. The volume of a catalyst bed for a 300 cu. in. engine capable of operating satisfactorily at full throttle would be less than 360 cc - a manageable volume from both size and warm-up considerations. The catalyst has the capacity of processing more than 700 volumes of fuel per volume of catalyst. For a 360 cc catalyst bed, this corresponds to processing 66.5 gallons of fuel or about a driving range of 800 to 1000 miles before air regeneration would be necessary. The catalyst appears stable toward oxidative regeneration and its catalytic activity can be fully restored. Since the required volume of catalyst bed is small enough, segmented or multiple reactors could be used to accomplish cracking operation and regeneration at all times.

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TABLE I

Product Distribution at 900°F

Wt. %	Feed	WHSV			% Conversion		
		55	100	200	55	100	200
Methane	--	0.6	0.4	0.2			
Ethane, Ethene	--	3.3	2.5	0.8			
Propane	--	9.7	6.1	1.1			
Propene	--	7.0	5.8	3.6			
Butanes	--	2.1	1.5	0.3			
Butenes	--	3.1	2.6	2.0			
2,2-Dimethylbutane	9.1	8.6	8.6	8.6	5.5	5.5	5.5
2,3-Dimethylbutane	5.4	5.4	5.4	5.4	0.0	0.0	0.0
2-Methylpentane	13.5	8.5	10.7	11.7	37.0	20.7	13.3
Hexane, 1-hexene	24.5	6.9	10.4	16.1	71.8	57.6	34.3
Benzene	47.5	38.1	40.8	44.4	19.8	14.1	6.5
C_7^+ Aromatics	--	6.7	5.1	5.6			
R+O	77.1	96.8	92.0	84.5			
ΔON	--	19.7	14.9	7.4			
C_4^-% Conversion	--	25.8	18.9	8.0			
ΔON/ % Conversion	--	0.76	0.79	0.93			

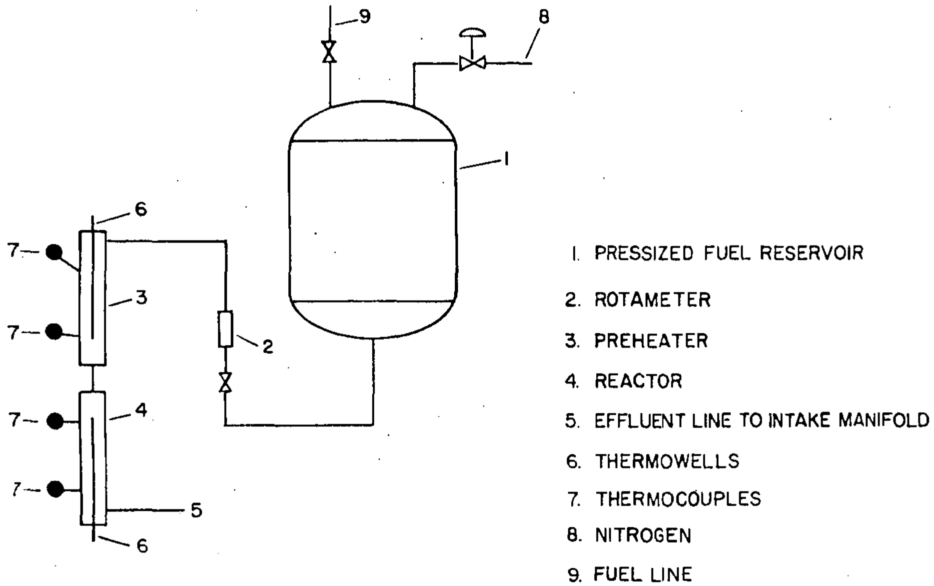

Fig. 1 SCHEMATIC DIAGRAM OF UNIT

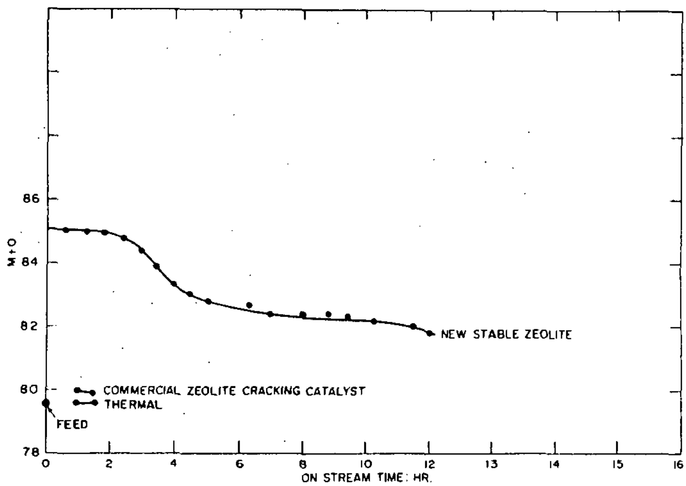

Fig 2 EFFECT OF ON STREAM TIME ON OCTANE RATING - KNOCK TEST ENGINE

79.5 M+O C_5-400 REFORMATE

915 °F 95 LHSV

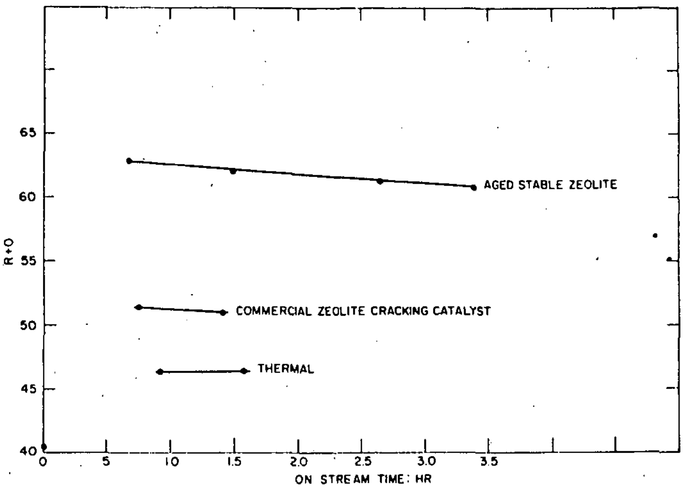

Fig 3 EFFECT OF ON STREAM TIME ON OCTANE RATING
405 M+O KUWAIT NAPHTHA
915 °F 95 LHSV

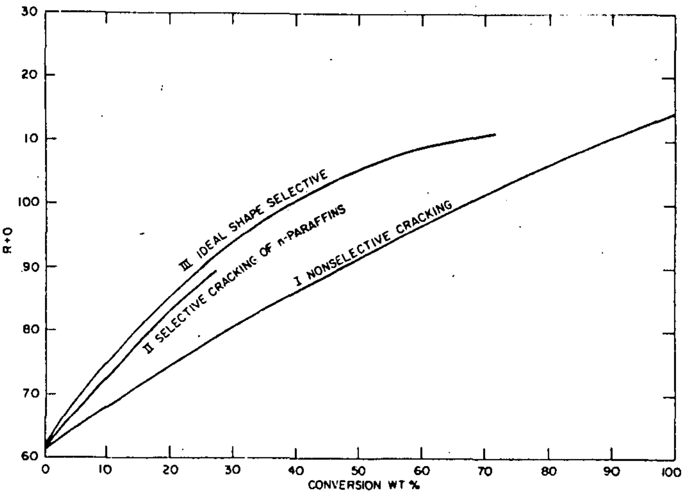

Fig 4 OCTANE RATING vs. CONVERSION LEVEL

Emission Control and Fuel Economy

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Extended Abstract

For at least the near term future, the conventional piston engine will continue to be the dominant automobile power plant. The two major factors to which it must respond are emission standards and fuel economy. Since these two factors are closely linked, we have made a theoretical and experimental study of fuel economy as a function of emission standards for a variety of catalytic and thermal control systems applied to the piston engine.

Theoretical Considerations

Factors affecting fuel economy and emissions can be divided into those external to the engine and those internal. For this paper, all external factors will be assumed constant, with vehicle weight, the most important of these, held at 4000 lbs.

Internal factors having significant effects include air-fuel ratio, compression ratio, spark timing, exhaust gas recycle, and load factor. Each will be discussed in turn and the entire discussion summarized by relating fuel economy to emission standards for several emission control systems.

Peak fuel economy is obtained at air-fuel ratios slightly leaner than the stoichiometric value. The region of 16-16.5 lbs. of air per lb. of fuel generally is the optimum. Richer mixtures release less of the fuel's available heat of combustion while leaner mixtures are increasingly difficult to burn at the optimum time. Additionally, dilution with fuel or air lowers peak flame temperature. Maximum production of nitrogen oxides occurs at about the same air-fuel ratio as maximum fuel economy, since both are functions of peak flame temperature. Carbon monoxide and unburned hydrocarbon emissions decrease with increasing air-fuel ratio, although a practical limit is reached with conventional systems at about 18, where mis-fire begins and hydrocarbon emissions turn up again.

Increasing compression ratio allows more efficient use to be made of the heat energy in fuel. For example, at constant performance, an increase in C. R. from 8:1 to 9:1 would improve fuel economy 5 - 6 %. However, the higher peak flame temperatures associated with this change would also produce an increase in nitrogen oxide production.

Engine load is another important parameter affecting fuel economy. The greatest relative economy is obtained at wide open throttle operation. At any reduced load (reduced intake manifold pressure) the engine must work harder to pump the air-fuel charge into the cylinders. These pumping losses are a minimum at full-load operation. In practice, maximum fuel economy is not obtained at a vehicle's top speed, since increased wind resistance and road friction losses more than compensate for increased engine efficiency. However, for a given vehicle weight at a given speed, a small engine operating closer to full load will have better fuel economy than a large engine throttled back.

Exhaust gas recycle is commonly used to reduce nitrogen oxide formation. It functions by lowering peak flame temperatures and thus might be expected to harm fuel economy. However, because EGR requires an increased intake manifold pressure (wider throttle opening) to maintain constant power output, the engine has less pumping and throttling losses and can compensate for most of the efficiency lost by lower peak flame temperatures. In order to take maximum advantage of this trade-off, it must be borne in mind that EGR also decreases flame speed. Therefore, spark timing must be advanced to allow proper combustion time. Spark timing must also be adjusted for changes in air-fuel ratio as flame speed also changes with this parameter.

The foregoing has discussed peak flame temperature as related to fuel economy and nitrogen oxide formation. It is also necessary to consider a related parameter, exhaust gas temperature, and its relation to emission control. Generally, the higher the peak flame temperature, the more heat energy which can be extracted from the combustion chamber, hence the lower the exhaust gas temperature. However, in order to control emissions by homogeneous or heterogeneous reactions outside of the engine, high temperatures are desirable. Thus we must examine the balance between temperature and emission control.

Thermal reactors require temperatures in excess of 1500°F to achieve satisfactory homogeneous control of carbon monoxide and hydrocarbons to the most stringent statutory levels. Normal exhaust gas temperatures are in the 1000°F range. Therefore, a substantial increase in exhaust temperature or in available heat of combustion in the exhaust is required for these devices. The most fuel economical method of supplying the needed heat is to richen the air-fuel ratio. This will supply excess carbon monoxide and hydrocarbons, which, when combusted in the reactor, will maintain it at its operating temperature.

Practical considerations militate against using this approach solely, so a combination of enrichment and spark retard, which also increases exhaust temperature, but at a greater fuel economy penalty, is necessary. A third method, lowering the compression ratio, imposes a still higher fuel penalty. Fuel economy debits of 20 - 25% compared to uncontrolled cars are typical for thermal reactors controlling emissions to the stringent statutory levels of 3.4 g/mi. of CO and 0.41 g/mi. of HC.

On the other hand, catalytic oxidation of carbon monoxide and hydrocarbons proceeds efficiently at temperatures associated with normal exhaust temperatures. Thus fuel economy debits of the type associated with thermal reactors are not necessary. The engine can be tuned for maximum operating efficiency without regard to exhaust temperatures. Therefore, oxidation catalysts allow decoupling of emission control from engine operation.

Unfortunately, catalytic reduction of nitrogen oxide is not as independent of engine operation as is catalytic oxidation of carbon monoxide and hydrocarbons. The reduction catalyst requires a reducing atmosphere, hence the engine must be run at an air-fuel ratio richer than stoichiometric. This means a fuel penalty will be incurred compared to an uncontrolled car even if all other engine parameters are optimized. In addition, most reduction catalysts require operating temperatures in excess of normal exhaust levels, so further inefficiencies would be necessary. It would be desirable to have a reduction catalyst capable of efficient conversion at normal exhaust gas temperatures. Ruthenium-containing catalysts have this potential, but to date neither they nor their high temperature base metal counterparts have exhibited satisfactory durability.

In summarizing all of these considerations, we can compare a pre-control, 1967, 4000 lb. vehicle in fuel economy with that predicted for vehicles equipped with thermal or catalytic control systems to meet several emission standards. First, a 1974 vehicle relying on engine modifications only, including a compression ratio of 8.2:1, to meet this year's standards shows approximately a 14% debit in fuel economy. Thermal reactor vehicles, which can tolerate leaded fuel and therefore operate at compression ratios of 10:1, could meet the 1974 standards with about a 6% debit and the 1975 United States interim standards for carbon monoxide and hydrocarbons with about a 12% debit. In meeting the more stringent California interim and future U. S. standards, rich thermal reactors are required and the debit should rise to the 20 - 22% level. Finally, if the statutory 1977 nitrogen oxide level of 0.4 g/mi. is to be achieved with a thermal system, the debit should reach approximately 25%.

Catalytic systems on the other hand, cannot use leaded fuels. They will therefore be designed with compression ratios in the range of 8:1 to accomodate lower octane unleaded fuels. Even so, their lower operating temperatures should result in better fuel economy. Thus the 1975 interim standards for carbon monoxide and hydrocarbons should be achievable at a fuel economy debit of only about 6% from pre-controlled levels. The more stringent 1976 standards should cause a rise to only about 8%, and even the 1977 standard for nitrogen oxide will produce only about a 12% debit.

Experimental Results

The relationship between fuel economy and exhaust emissions has been studied with two types of systems. The first uses a noble metal monolithic oxidation catalyst to control hydrocarbon and carbon monoxide emissions and exhaust gas recycle to limit nitrogen oxide emissions. The second system is a dual catalyst configuration, with a reduction catalyst for nitrogen oxide control followed by the oxidation catalyst. Air is injected between the two catalysts to convert the exhaust gas to a net oxidizing composition.

The oxidation catalyst-EGR system was mounted on a 1973 vehicle with a 350 in^3 displacement engine. The stock vehicle, equipped with a non-proportional EGR system, gave emissions, in g/vehicle mile as tested on the 1975 Federal Test Procedure, of 21.4 CO, 1.3 HC, and 3.3 NO_x. Its fuel economy over the same test cycle was 10.40 miles per gallon. As modified with oxidation catalysts and a proportional EGR system, that is one responding directly to the exhaust gas flow rate, the test vehicle easily met the 1976 statutory CO and HC standards of 3.4 and 0.41 g/mile respectively. With the timing advanced for good fuel economy, not only was the stock NO_x emission level matched, but a 7% gain in fuel economy was achieved. Retarding the timing lowered NO_x emissions further, but at some cost in fuel economy. Work is continuing in an effort to optimize the factors influencing the NO_x emission-fuel economy trade-off with this system.

The dual catalyst system was mounted in a 1973 vehicle similar to the base car described above. In this case, no EGR was used on the modified car. The reduction catalyst was the GEM reinforced Ni-Cu material made by Gould, Inc. With the dual catalyst configuration, as described earlier, catalyst temperature is the primary determinant of fuel economy and NO_x emissions. The temperature was varied by a combination of air-fuel ratio and spark timing control. The statutory 1976 standards for CO and HC emissions were met at all temperatures, but NO_x was dependent on catalyst temperature. At an average catalyst temperature around 1100°F,, an emission level of 1.7 g/mile was achieved, with fuel economy comparable to the unmodified vehicle. At 1200°F., NO_x emissions were controlled to 0.9 g/mile, but a fuel economy debit of 4% was incurred. Finally, the 1977 statutory standard of 0.4 g/mile was reached at 1300°F., with a fuel economy debit of 10%.

IMPACT OF AUTOMOTIVE TRENDS AND EMISSIONS REGULATIONS ON GASOLINE DEMAND. Dayton H. Clewell, Mobil Oil Corporation, 150 East 42nd Street, New York, N. Y., 10017. William J. Koehl, Mobil Research and Development Corporation, Research Department, Paulsboro Laboratory, Paulsboro, N. J., 08066.

Gasoline demand has increased steadily in recent years because of growth in vehicle registrations and miles traveled and because of trends in vehicle designs and equipment, among which emission controls are most notable. Through 1985, gasoline demand is projected to increase about 50%, and maybe substantially more depending on the emission controls required. In view of increasing demand and tightening supplies for energy in all forms, four alternatives are explored for moderating the growth in demand for gasoline. These alternatives are: (1) optimizing the energy cost of vehicle emission standards against the emissions reduction needed to achieve the ambient air quality standards; (2) increasing the use of smaller, more economical cars; (3) using more efficient engines; and (4) increasing the use of public transportation. Each can contribute to energy conservation; no one is the whole answer. The benefits of optimum standards can be assured by prompt government action. The trend toward smaller cars is already growing. Introduction of alternate engines requires a long lead time. Mass transportation could be most beneficial in metropolitan areas.

INFLUENCE OF FUEL COMPOSITION ON TOTAL ENERGY RESOURCES

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Combustion fuels are essentially limited to compounds of carbon, hydrogen and oxygen by the requirement that combustion products be non-offensive and biodegradable. Properties of these compounds are determined by their chemical structure, but can often be related in a gross way to their hydrogen to carbon ratio. The quality of fuels for continuous combustion decreases with the hydrogen to carbon ratio, and the hydrogen content of a crude petroleum limits the amounts of preferred fuel oils that can be made from it. Also, the removal of impurities such as sulfur, nitrogen, etc., from fuels at present requires the use of hydrogen. Other potential raw materials - shale, coal, etc. - are poorer in hydrogen, and thus future fuel manufacture will require the manufacture of this element in relatively pure form.

The production of hydrogen from water, the most probable source, requires the expenditure of energy, and thus improving fuel quality reduces the total energy resources availability. For various specific situations, e.g., coal conversion, the magnitude of this effect can be calculated, and these calculations may aid in emphasizing areas of desirable compromises among fuel quality, combustor or engine design, and emission regulations.

Low Emissions Combustion Engines for Motor Vehicles

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During the past 10-15 years, very significant advances in controlling exhaust emissions from automobile power plants have been made. Initially, emissions reductions were achieved through careful readjustment and control of engine operating conditions (1). More recently, highly effective exhaust treatment devices requiring a minimum of basic modification to the already highly developed internal combustion engine have been demonstrated. These are based on thermal and/or catalytic oxidation of hydrocarbons (HC) and carbon monoxide (CO) in the engine exhaust system (2,3,4). Nitrogen oxide (NO_x) emissions have been reduced to some extent through a combination of retarded ignition timing and exhaust gas recirculation (EGR), both factors serving to diminish severity of the combustion process temperature-time history without substantially altering design of the basic engine (5).

Basic combustion process modification as an alternative means for emissions control has received less attention than the foregoing techniques, though it has been demonstrated that certain modified combustion systems can in principle yield significant pollutant reductions without need for exhaust treatment devices external to the engine. Additionally, it has been demonstrated that when compared with conventional engines controlled to low emissions levels, modified combustion processes can offer improved fuel economy.

Nearly all such modifications involve engine designs permitting combustion of fuel-air mixtures lean beyond normal ignition limits. As will be shown, decreased mean combustion temperatures associated with extremely lean combustion tend to limit the rate of nitric oxide (NO) formation and, hence, the emission of NO_x. At the same time, the relatively high oxygen content of lean mixture combustion products tends to promote complete oxidation of unburned HC and CO provided that combustion gas temperatures are sufficiently high during late portions of the engine cycle.

The purpose of this paper is to present an overall review of the underlying concepts and current status of unconventional engines employing modified combustion as a means for emissions control. Detailed findings related to specific power plants or to specific applications will be treated by the papers which follow.

Throughout the paper exhaust emissions will be compared with emissions standards legislated for the years 1975 and 1976. As a result of Environmental Protection Agency (EPA) actions suspending the 1975 HC and CO standards and the 1976 NO_x standard, several sets of values exist. These are listed in Table I and in the text will be referenced either as statutory (original standards as set by the Clean Air Act Amendment of 1970) or as interim standards as set by the EPA.

Theoretical Basis for
Combustion Modification

Figure 1 has been derived from experimental measurements (6) of the rate of NO formation in combustion processes under conditions typical of engine operation. This figure demonstrates two major points related to control of NO_x emissions: First, the slow rate of NO formation relative to the rates of major combustion reactions responsible for heat release and, second, the strong influence of fuel-air equivalence ratio on the rate of NO formation.

Experimental combustion studies (7) employing "well-stirred reactors" have shown that hydrocarbon-air combustion rates can be correlated by an expression of the form

$$\frac{N}{V\, p^{1.8}} = 48\ \frac{\text{Gram-Moles/Liter-Second}}{\text{Atm}^{1.8}}$$

where:

N = moles reactants consumed per second
V = combustion volume
p = total pressure

For conditions typical of engine operation, this expression yields a time of approximately 0.1 ms for completion of major heat release reactions following ignition of a localized parcel of fuel-air mixture within the combustion chamber. Comparison with Figure 1 shows that the time required for formation of significant amounts of NO in combustion gases is at least a factor of 10 greater. Thus, in principle, energy conversion can be effected in times much shorter than required for NO formation. In the conventional spark ignition engine, the relatively lengthy flame travel process permits combustion products to remain at high temperatures sufficiently long that considerable NO formation occurs.

Figure 2, which consolidates the data of Figure 1, indicates that maximum rates of NO formation are observed at fuel-air equivalence ratios around 0.9 (fuel lean). For richer mixtures, the concentrations of atomic and diatomic oxygen, which participate as reactants in the formation of NO in combustion gases, decrease. On the other hand, for mixtures leaner than approximately 0.9 equivalence ratio, decreasing combustion temperatures result in lower NO formation rates.

Figure 2 serves as a basis for combustion process modification. Operation with extremely rich fuel-air mixtures (Point A of Figure 2), of course, results in low NO_x emissions since the maximum chemical equilibrium NO level is greatly reduced under such conditions. However, the resultant penalties in terms of impaired fuel economy and excessive HC and CO emissions are well known. An alternative is operation with extremely lean mixtures (Point B), lean beyond normal ignition limits. Combustion under such conditions can lead to low NO_x emissions while at the same time providing an excess of oxygen for complete combustion of CO and HC.

Operation of internal combustion engines with extremely lean overall fuel-air ratios has been achieved in several ways, employing a number of differing combustion chamber configurations. One approach involves ignition of a very small and localized quantity of fuel-rich and ignitable mixture (Point A of Figure 2), which in turn serves to inflame a much larger quantity of surrounding fuel-air mixture too lean for ignition under normal circumstances. The bulk or average fuel-air ratio for the process corresponds to Point B of Figure 2; and, as a consequence, reduced exhaust emissions should result.

A second approach involves timed staging of the combustion process. An initial rich mixture stage in which major combustion reactions are carried out is followed by extremely rapid mixing of rich mixture combustion products with dilution air. The transition from initial Point A to final Point B in Figure 2 is, in principle, sufficiently rapid that little opportunity for NO formation exists. Implicit here is utilization of the concept that the heat release reactions involved in the transition from Point A to Point B can be carried out so rapidly that time is not available for formation of significant amounts of NO.

Reciprocating spark ignition engines designed to exploit the foregoing ideas are usually called stratified charge engines, a term generally applied to a large number of designs encompassing a wide spectrum of basic combustion processes.

Open-Chamber Stratified Charge Engines

Stratified charge engines can be conveniently divided into two types: open-chamber and dual-chamber engines. The open-chamber stratified charge engine has a long history of research interest. Those engines reaching the most advanced stages of development are probably the Ford-programmed combustion process (PROCO) (8) and Texaco's controlled combustion process (TCCS) (9). Both engines employ a combination of inlet air swirl and direct timed combustion chamber fuel injection to achieve a local fuel-rich ignitable mixture near the point of ignition. The overall mixture ratio under most operating conditions is fuel lean.

The Texaco TCCS engine is illustrated schematically in Figure 3. During the engine inlet stroke, an unthrottled supply of air enters the cylinder through an inlet port oriented to promote a specified level of air swirl within the cylinder and combustion chamber. As the subsequent compression stroke nears completion, fuel is injected into and mixes with an element of swirling air charge. This initial fuel-air mixture is spark ignited, and a flame zone is established downstream from the nozzle. As injection continues, fuel-air mixture is continuously swept into the flame zone. The total quantity of fuel consumed per cycle and, hence, engine power output, are controlled by varying the duration of fuel injection. Under nearly all engine operating conditions, the total quantity of fuel injected is on the lean side of stoichiometric. The TCCS system has been under development by Texaco since the 1940's. To date, this work has involved application of the process to a wide variety of engine configurations.

Like the TCCS engine, the PROCO system (Figure 4) employs timed combustion chamber fuel injection. However, in contrast to the TCCS system, the PROCO system is based on formation of a premixed fuel-air mixture prior to ignition. Fuel injection and inlet air swirl are coordinated to provide a small portion of rich mixture near the point of ignition surrounded by a large region of increasingly fuel-lean mixture. Flame propagation proceeds outward from the point of ignition through the leaner portions of the combustion chamber.

Both the TCCS and PROCO engines are inherently low emitters of CO, primarily a result of lean mixture combustion. Unburned HC and NO_x emissions have been found to be lower than those typical of uncontrolled conventional engines, but it appears that additional control measures are required to meet statutory 1976 Federal emissions standards.

The U.S. Army Tank Automotive Command has sponsored development of low emissions TCCS and PROCO power plants for light-duty Military vehicles. These power plants have been based on conversion of the 4-cylinder, 70-hp L-141 Jeep engine. The vehicles in which these engines were placed were equipped with oxidizing catalysts for added control of HC and CO emissions, and EGR was used as an additional measure for control of NO_x.

Results of emissions tests on Military Jeep vehicles equipped with TCCS and PROCO engines are listed in Table II (10). At low mileage these vehicles met the statutory 1976 emissions standards. Deterioration problems related to HC emission would be expected to be similar to those of conventional engines equipped with oxidizing catalysts. This is evidenced by the increase in HC emissions with mileage shown by Table II. NO_x and CO emissions appear to have remained below 1976 levels with mileage accumulation.

Table III presents emissions data at low mileage for several passenger car vehicles equipped with PROCO engine conversions (10). These installations included noble metal catalysts and EGR for added control of HC and NO_x emissions, respectively. All vehicles met the statutory 1976 standards at low mileage. Fuel consumption data, as shown in Table III, appear favorable when contrasted with the fuel economy for current production vehicles of similar weight.

Fuel requirements for the TCCS and PROCO engines differ substantially. The TCCS concept was initially developed for multifuel capability; as a consequence, this engine does not have a significant octane requirement and is flexible with regard to fuel requirements. In the PROCO engine combustion chamber, an end gas region does exist prior to completion of combustion; and, as a consequence, this engine has a finite octane requirement.

Prechamber Stratified Charge Engines

A number of designs achieve charge stratification through division of the combustion region into two adjacent chambers. The emissions reduction potential for two types of dual-chamber engines has been demonstrated. First, in a design traditionally called the "prechamber engine," a small auxiliary or ignition chamber equipped with a spark plug communicates with the much larger main combustion chamber located in the space above the piston (Figure 5). The prechamber typically contains 5-15% of the total combustion volume. In operation of this type of engine, the prechamber is supplied with a small quantity of fuel-rich ignitable fuel-air mixture while a very lean and normally unignitable mixture is supplied to the main chamber above the piston. Expansion of high temperature flame products from the prechamber leads to ignition and burning of the lean main chamber fuel-air charge.

The prechamber stratified charge engine has existed in various forms for many years. Early work by Ricardo (11) indicated that the engine could perform very efficiently within a limited range of carefully controlled operating conditions. Both fuel-injected and carbureted prechamber engines have been built. A fuel-injected design initially conceived by Brodersen (12) was the subject of extensive study at the University of Rochester for nearly a decade (13,14). Unfortunately, the University of Rochester work was undertaken prior to widespread recognition of the automobile emissions problem; and, as a consequence, emissions characteristics of the Brodersen engine were not determined. Another prechamber engine receiving attention in the early 1960's is that conceived by R. M. Heintz (15). The objectives of this design were reduced HC emissions, increased fuel economy, and more flexible fuel requirements.

Initial experiments with a prechamber engine design called "the torch ignition engine" were reported in the U.S.S.R. by Nilov (16) and later by Kerimov and Mekhtier (17). This designation refers to the torchlike jet of hot combustion gases issuing from the precombustion chamber upon ignition. In the Russian designs, the orifice between prechamber and main chamber is sized to produce a high velocity jet of combustion gases. In a recent publication (18), Varshaoski et al. have presented emissions data obtained with a torch engine system. These data show significant pollutant reductions relative to conventional engines; however, their interpretation in terms of requirements based on the Federal emissions test procedure is not clear.

A carbureted three-valve prechamber engine, the Honda Compound Vortex-Controlled Combustion (CVCC) system, has received considerable recent publicity as a potential low emissions power plant (19). This system is illustrated schematically in Figure 6. Honda's current design employs a conventional engine block and piston assembly. Only the cylinder head and fuel inlet system differ from current automotive practice. Each cylinder is equipped with a small precombustion chamber communicating by means of an orifice with the main combustion chamber situated above the piston. A small inlet valve is located in each prechamber. Larger inlet and exhaust valves typical of conventional automotive practice are located in the main combustion chamber. Proper proportioning of fuel-air mixture between prechamber and main chamber is achieved by a combination of throttle control and appropriate inlet valve timing. Inlet ports and valves are oriented to provide specific levels of air swirl and turbulence in the combustion chamber. In this way, a relatively slow and uniform burning process giving rise to elevated combustion temperatures late in the expansion stroke and during the exhaust process is achieved. High temperatures in this part of the engine cycle are necessary to promote complete oxidation of HC and CO. It should be noted that these elevated temperatures are necessarily obtained at the expense of a fuel economy penalty.

Results of emissions tests with the Honda engine have been very promising. The emissions levels shown in Table IV for a number of lightweight Honda Civic vehicles are typical and demonstrate that the Honda engine can meet statutory 1975-1976 HC and CO standards and can approach the statutory 1976 NO_x standard (10). Of particular importance, durability of this system appears excellent as evidenced by the high mileage emissions levels reported in Table IV. The noted deterioration of emissions after 30,000-50,000 miles of engine operation was slight and apparently insignificant.

Recently, the EPA has tested a larger vehicle converted to the Honda system (20). This vehicle, a 1973 Chevrolet Impala with a 350-CID V-8 engine, was equipped with cylinder heads and induction system of Honda manufacture. Test results are presented in Table V for low vehicle mileage. The vehicle met the present 1976 interim Federal emissions standards though NO_x levels were substantially higher than for the much lighter weight Honda Civic vehicles.

Fuel economy data indicate that efficiency of the Honda engine, when operated at low emissions levels, is somewhat poorer than that typical of well-designed conventional engines operated without emissions controls. However, EPA data for the Chevrolet Impala conversion show that efficiency of the CVCC engine meeting 1975-1976 interim standards was comparable to or slightly better than that of 1973 production engines of similar size operating in vehicles of comparable weight. It has been stated by automobile manufacturers that use of exhaust oxidation catalysts beginning in 1975 will result in improved fuel economy relative to 1973 production vehicles. In this event fuel economy of catalyst-equipped conventional engines should be at least as good as that of the CVCC system.

The apparent effect of vehicle size (more precisely the ratio of vehicle weight to engine cubic inch displacement) on NO_x emissions from the Honda engine conversions demonstrates the generally expected response of NO_x emissions to increased specific power demand from this type of engine. For a given engine cubic inch displacement, maximum power output can be achieved only by enriching the overall fuel-air mixture ratio to nearly stoichiometric proportions and at the same time advancing ignition timing to the MBT point. Both factors give rise to increased NO_x emissions. This behavior is evidenced by Table VI, which presents steady state emissions data for the Honda conversion of the Chevrolet Impala (20). At light loads, NO_x emissions are below or roughly comparable to emissions from a conventionally powered 1973 Impala. This stock vehicle employs EGR to meet the 1973 NO_x standard. It is noted in Table VI that for the heaviest load condition reported, the 60-mph cruise, NO_x emissions from the Honda conversion approached twice the level of emissions from the stock vehicles. This points to the fact that in sizing engines for a specific vehicle application, the decreased air utilization (and hence specific power output) of the prechamber engine when operated under low emissions conditions must be taken into consideration.

Divided-Chamber Staged Combustion Engine

Dual-chamber engines of another type, often called "divided-chamber" or "large-volume prechamber" engines, employ a two-stage combustion process. Here initial rich mixture combustion and heat release (first stage of combustion) are followed by rapid dilution of combustion products with relatively low temperature air (second stage of combustion). In terms of the concepts previously developed, this process is initiated in the vicinity of Point A of Figure 2. Subsequent mixing of combustion products with air is represented by a transition from Point A to Point B. The object of this engine design is to effect the transition from Point A to Point B with sufficient speed that time is not available for formation of significant quantities of NO. During the second low temperature stage of combustion (Point B), oxidation of HC and CO goes to completion.

An experimental divided-chamber engine design that has been built and tested is represented schematically in Figure 7 (21,22). A dividing orifice (3) separates the primary combustion chamber (1) from the secondary combustion chamber (2), which includes the cylinder volume above the piston top. A fuel injector (4) supplies fuel to the primary chamber only. Injection timing is arranged such that fuel continuously mixes with air entering the primary chamber during the compression stroke. At the end of compression, as the piston nears its top center position, the primary chamber contains an ignitable fuel-air mixture while the secondary chamber adjacent to the piston top contains only air. Following ignition of the primary chamber mixture by a spark plug (6) located near the dividing orifice, high temperature rich mixture combustion products expand rapidly into and mix with the relatively cool air contained in the secondary chamber. The resulting dilution of combustion products with attendant temperature reduction rapidly suppresses formation of NO. At the same time, the presence of excess air in the secondary chamber tends to promote complete oxidation of HC and CO.

Results of limited research conducted both by university and industrial laboratories indicate that NO_x reductions of as much as 80-95% relative to conventional engines are possible with the divided-chamber staged combustion process. Typical experimentally determined NO_x emissions levels are presented in Figure 8 (23). Here NO_x emissions for two different divided-chamber configurations are compared with typical emissions levels for conventional uncontrolled automobile engines. The volume ratio, β, appearing as a parameter in Figure 8, represents the fraction of total combustion volume contained in the primary chamber. For β values approaching 0.5 or lower, NO_x emissions reach extremely low levels. However, maximum power output capability for a given engine size decreases with decreasing β values. Optimum primary chamber volume must ultimately represent a compromise between low emissions levels and desired maximum power output.

HC and particularly CO emissions from the divided-chamber engine are substantially lower than conventional engine levels. However, further detailed work with combustion chamber geometries and fuel injection systems will be necessary to fully evaluate the potential for reduction of these emissions. Table VII presents results of tests cited by the National Academy of Sciences (10).

Emissions from the divided-chamber engine are compared with those from a laboratory PROCO stratified charge engine, the comparison being made at equal levels of NO_x emissions. NO_x emissions were controlled to specific levels by addition of EGR to the PROCO engine and by adjustment of operating parameters for the divided-chamber engine. These data indicate that the divided-chamber engine is capable of achieving very low NO_x emissions with relatively low HC and CO emissions.

As shown by Table VII, fuel economy of the divided-chamber staged combustion engine is comparable to that of conventional piston engines without emissions controls. When compared with conventional piston engines controlled to equivalent low NO_x emissions levels, the divided-chamber engine is superior in terms of fuel economy.

The Diesel Engine

The diesel engine can be viewed as a highly developed form of stratified charge engine. Combustion is initiated by compression ignition of a small quantity of fuel-air mixture formed just after the beginning of fuel injection. Subsequently, injected fuel is burned in a heterogeneous diffusion flame. Overall fuel-air ratios in diesel engine operation are usually extremely fuel lean. However, major combustion reactions occur locally in combustion chamber regions containing fuel-air mixtures in the vicinity of stoichiometric proportions.

The conventional diesel engine is characterized by low levels of CO and light HC emissions, a result of lean mixture operation. On a unit power output basis, NO_x emissions from diesel engines are typically lower than those of uncontrolled gasoline engines, a combined result of diffusion combustion and, in an approximate sense, low mean combustion temperatures. Work devoted to mathematical simulation of diesel combustion has shown that NO formation occurs primarily in combustion products formed early in the combustion process, with the later portions of diffusion-controlled combustion contributing substantially less (24).

Table VIII presents emissions levels for three diesel-powered passenger cars as reported by the EPA (25). These vehicles, a Mercedes 220D, Opel Rekord 2100D, and Peugeot 504D, were powered by 4-cylinder engines ranging in size from 126-134 CID with power ratings ranging from 65-68 bhp. Two of the diesel-powered vehicles were capable of meeting the 1975 statutory emissions standards. NO_x emissions were in excess of the original Federal 1976 standards but were well within present interim standards.

The preceding data do not include information on particulate and odorant emissions, both of which could be important problems with widespread diesel engine use in automobiles. Complete assessment of the environmental potential for the diesel engine would have to include consideration of these factors as well as emission of polynuclear aromatic hydrocarbons. All are the subject of ongoing research.

Fuel economy data referred to both 1972 and 1975 Federal test procedures are presented in Table VIII. As expected, diesel engine fuel economies are excellent when compared with gasoline engine values. However, a more accurate appraisal would probably require comparison at equal vehicle performance levels. Power-to-weight ratios and, hence, acceleration times and top speeds for the diesel vehicles cited above are inferior to values expected in typical gasoline-powered vehicles.

Gas Turbine, Stirling Cycle, and Rankine Cycle Engines

Gas turbine, Stirling cycle, and Rankine engines all employ steady flow or continuous combustion processes operated with fuel-lean overall mixture ratios. In a strict sense, the gas turbine is an internal combustion engine since high temperature combustion products serve as the cycle working fluid. Rankine and Stirling engines are external combustion devices with heat exchanged between high temperature combustion gases and the enclosed cycle working fluid.

In contrast to the situation with conventional spark ignition piston engines, the major obstacles related to use of continuous combustion power plants are in the areas of manufacturing costs, durability, vehicle performance, and fuel economy. The problem of exhaust emissions, which involves primarily the combustion process, has been less significant than the foregoing items.

As a consequence of lean combustion, these continuous combustion power plants are characterized by low HC and CO emissions. Several investigators have reported data indicating that existing combustion systems are capable of approaching or meeting statutory 1975 and 1976 vehicle emissions standards for HC and CO (26,27).

For a given power output, NO_x emissions appear to be lower than those of conventional uncontrolled gasoline engines. However, it has been shown that existing combustors probably will not meet the statutory 1976 NO_x standard when installed in motor vehicles (26).

The formation of NO_x in continuous-flow combustors has been found to result from the presence of high temperature zones with local fuel-air ratios in the vicinity of stoichiometric conditions. Approaches suggested for minimizing NO_x formation have involved reduction of these localized peak temperatures through such techniques as radiation cooling, water injection, and primary zone air injection. Other approaches include lean mixture primary zone combustion such that local maximum temperatures fall below levels required for significant NO formation. Laboratory gas turbine combustors employing several of these approaches have demonstrated the potential for meeting the 1976 standards (28). With a laboratory Stirling engine combustor, Philips has measured simulated Federal vehicle test procedure emissions levels well below 1976 statutory levels (29).

Conclusion

As an alternative to the conventional internal combustion engine equipped with exhaust treatment devices, modified combustion engines can, in principle, yield large reductions in vehicle exhaust emissions. Such modifications include stratified charge engines of both open and dual chamber design. On an experimental basis, prototype stratified charge engines have achieved low exhaust emissions with fuel economy superior to that of conventional engines controlled to similar emissions levels.

The diesel engine is capable of achieving low levels of light HC, CO, and NO_x emissions with excellent fuel economy. Potential problems associated with widespread diesel use in light-duty vehicles are initial cost, large engine size and weight for a given power output, the possibility of excessive particulate and odorant emissions, and excessive engine noise.

Several power plants based on continuous combustion processes have the potential for very low exhaust emissions. These include the gas turbine, the Rankine engine, and the Stirling engine. However, at the present time major problems in the areas of manufacturing costs, reliability, durability, vehicle performance, and fuel economy must be overcome. As a consequence, these systems must be viewed as relatively long range alternatives to the piston engine.

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Table I

Federal Exhaust Emissions Standards
Emissions, Grams/Mile[1]

	1975			1976			1977
		Interim			Interim		
	Statutory	U.S.	California	Statutory	U.S.	California	Statutory
HC	0.41	1.5	0.9	0.41	0.41	0.41	0.41
CO	3.4	15	9.0	3.4	3.4	3.4	3.4
NO_x	3.0	3.1	2.0	0.4	2.0	2.0	0.4

[1]As measured using 1975 CVS C-H procedure.

Table II

Average Emissions from Military Jeep Vehicles with Stratified Engine Conversions (Reference 10)

Engine	Miles	Emissions, g/Mile[2]			CVS Fuel Economy, mpg
		HC	CO	NO_x	
L-141 Ford[1] PROCO	Low 17,123	0.37 0.64	0.93 0.46	0.33 0.38	18.5-23
L-141 Texaco[1] TCCS	Low 10,000	0.37 0.77	0.23 1.90	0.31 0.38	16-22

[1]Engines equipped with oxidation catalysts and exhaust gas recirculation.

[2]1975 CVS C-H test procedure.

Table III

Average Low Mileage Emissions Levels - Ford PROCO Conversions (Reference 10)

	Emissions,[2] g/Mile			CVS Fuel Economy, mpg	Inertia Weight, Lb
	HC	CO	NO_x		
PROCO 141-CID[1] Capri Vehicles	0.12	0.46	0.32	20.4	2500
	0.13	0.18	0.33	25.1	
	0.11	0.27	0.32	22.3	
PROCO 351-CID[1] Torino Vehicle	0.30	0.37	0.37	14.4	4500
PROCO 351-CID[1] Montego Vehicles	0.36	0.13	0.63	-	-
	0.36	1.08	0.39	12.8	-

[1]All vehicles employed noble metal exhaust oxidation catalysts and exhaust gas recirculation.

[2]1975 CVS C-H test procedure.

Table IV

Honda Compound Vortex-Controlled Combustion-Powered Vehicle[1] Emissions (Reference 19)

	Emissions,[2] g/Mile			Fuel Economy, mpg	
	HC	CO	NO_x	1975 FTP	1972 FTP
Low Mileage Car[3] No. 3652	0.18	2.12	0.89	22.1	21.0
50,000-Mile Car[4] No. 2034	0.24	1.75	0.65	21.3	19.8

[1]Honda Civic vehicles.

[2]1975 CVS C-H procedure with 2000-lb inertia weight.

[3]Average of five tests.

[4]Average of four tests.

Table V

Emissions from Honda Compound Vortex-Controlled Combustion Conversion of 350-CID Chevrolet Impala (Reference 20)

Test	Emissions,[1] g/Mile			Fuel Economy, mpg
	HC	CO	NO_x	
1	0.27	2.88	1.72	10.5
2[2]	0.23	5.01	1.95	11.2
3[3]	0.80	2.64	1.51	10.8
4	0.32	2.79	1.68	10.2

[1]1975 CVS C-H procedure, 5000-lb inertia weight.

[2]Carburetor float valve malfunctioning.

[3]Engine stalled on hot start cycle.

Table VI

Steady State Emissions from Honda Compound Vortex-Controlled Combustion Conversion of 350-CID Chevrolet Impala (Reference 20)

Vehicle Speed, mph	Emissions, g/Mile					
	HC		CO		NO_x	
	350 CVCC	350 Stock	350 CVCC	350 Stock	350 CVCC	350 Stock
15	0.15	0.60	3.30	7.26	0.37	0.52
30	0.00	1.22	0.65	9.98	0.53	0.37
45	0.00	0.51	0.19	4.71	1.00	0.93
60	0.01	0.32	0.53	2.48	3.00	1.78

Table VII

Single-Cylinder Divided Combustion Chamber Engine Emissions Tests (Reference 10)

Engine	NO_x Reduction Method	Emissions, g/ihp-hr			Fuel Economy, Lb/ihp-hr
		NO_x	HC	CO	
PROCO	EGR	1.0	3.0	13.0	0.377
Divided Chamber	None	1.0	0.4	2.5	0.378
PROCO	EGR	0.5	4.0	14.0	0.383
Divided Chamber	None	0.5	0.75	3.3	0.377

Table VIII

Automotive Diesel Engine Emissions
(Reference 25)

Vehicle	Emissions, g/Mile				Inertia Weight, Lb	Fuel Economy, mpg	
	HC (Cold Bag)	HC (Hot FID)	CO	NO_x		1975 FTP	1972 FTP
Mercedes 220DD	0.17	0.34	1.42	1.43	3500	23.6	23.3
Mercedes 220D (Modified)	0.13	0.28	1.08	1.48	3500	24.6	23.6
Opel Rekord 2100D	0.16	0.40	1.16	1.34	3000	23.8	23.2
Peugeot 504D	1.30	3.53	3.34	1.04	3000	25.2	24.2

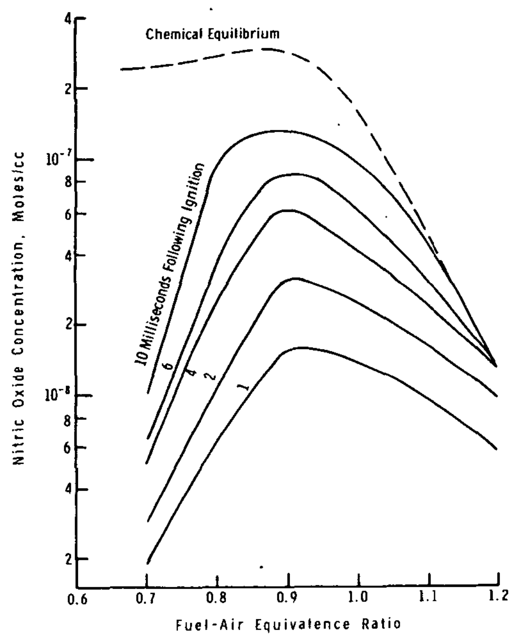

Figure 1: Rate of Nitric Oxide Formation in Engine Combustion Gases (Reference 6)

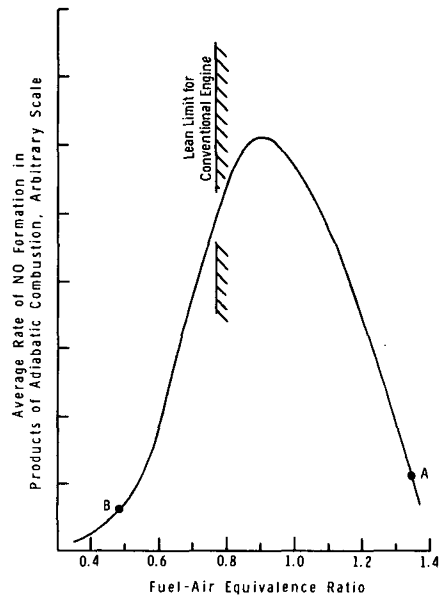

Figure 2: Influence of Fuel-Air Ratio on Rate of Nitric Oxide Formation

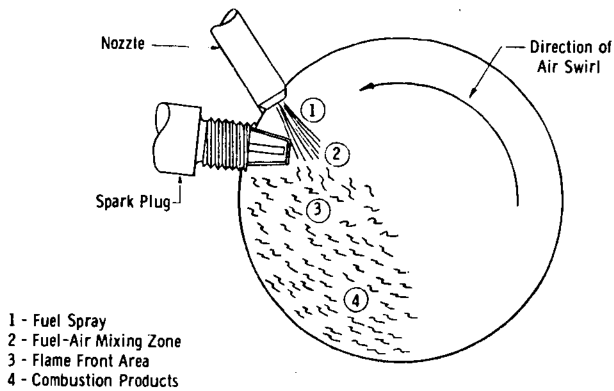

Figure 3: Texaco-Controlled Combustion System (TCCS)

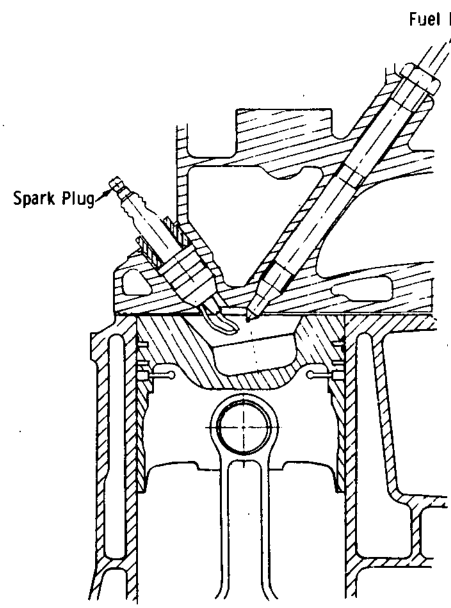

Figure 4: Ford-Programmed Combustion (PROCO) System

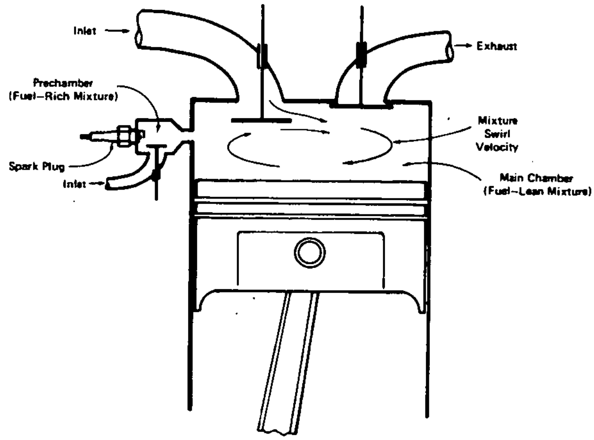

Figure 5: Schematic Representation of Prechamber Stratified Charge Engine

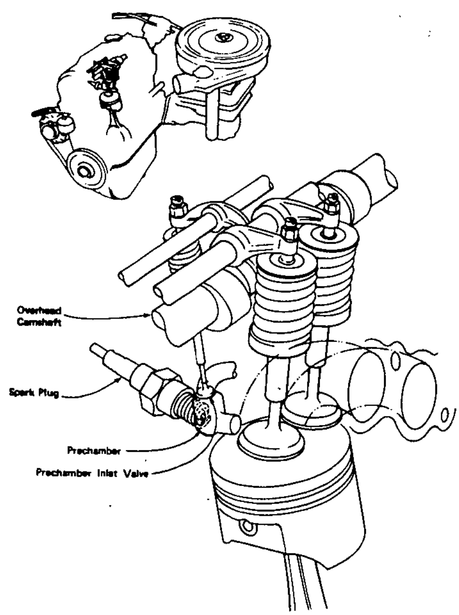

Figure 6: Honda CVCC Engine (Reference 19)

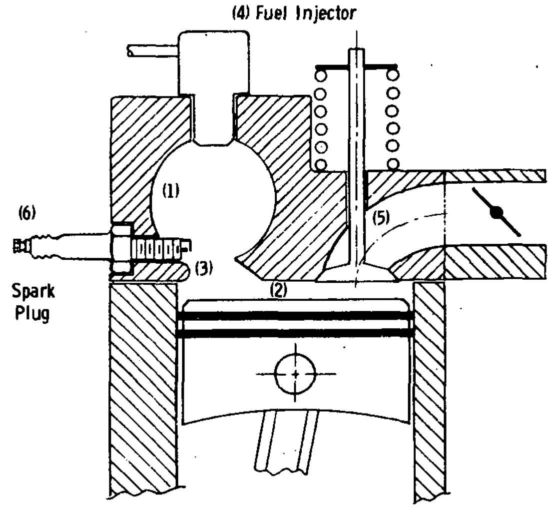

Figure 7: Schematic Representation of Divided Chamber Engine (Reference 21)

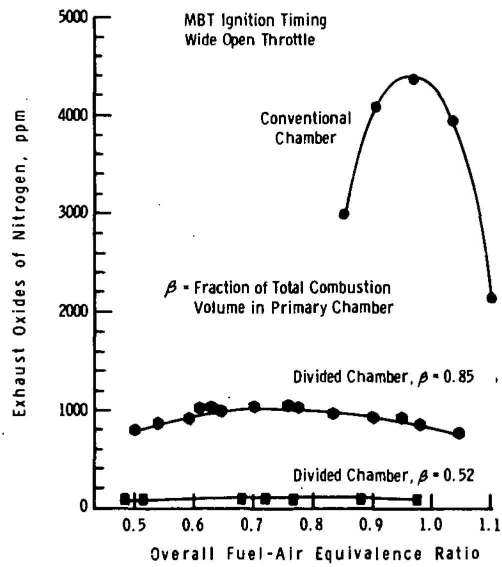

Figure 8: Comparison of Conventional and Divided Combustion Chamber NO_x Emissions (Reference 23)

TWO CURRENT APPROACHES TO AUTOMOTIVE EMISSION CONTROL. I. N. Bishop and J. H. Jones, Ford Motor Company, Dearborn, Michigan 48121

With the ever tightening requirements for automotive emission control, as especially regards the oxides of nitrogen, and the more recent increased need for improved fuel economy, two unique variants of the spark ignited internal combustion engine have been investigated for their potential in meeting these most important objectives. These engines are:

·Fast Burn - A homogeneous charge mixture cycle engine which utilizes maximum charge dilution for NO_x control while maintaining the combustion rate and thus the engine efficiency (fuel economy) through an increased level of chamber turbulence.

·PROCO (programmed Combustion) - A direct cylinder, fuel injected, stratified charge engine which utilizes the rich/lean combustion stratification scheme for both NO_x control and improved fuel economy.

The investigations of these alternate power systems have included math modeling for prediction of NO_x levels, basic engine configuration and operating parameter studies conducted on an engine dynamometer, vehicle evaluations of low mileage emission control capabilities, fuel economy, performance and driveability and system durability when subjected to 25,000 miles of the EPA mileage accumulation schedule. The results of these investigations have led to the conclusion that low NO_x levels can be achieved with good driveability and a definite improvement in fuel economy over conventional engine designs when calibrated to the same emission levels. However, the hydrocarbon and carbon monoxide levels are extremely high and were not able to be contained even with double the nominal catalyst volume.

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Title: Alternate Automotive Emission Control Systems

ABSTRACT

Automotive emission control systems have been developed to meet current and future exhaust emission standards with optimum fuel economy.

The 1973-1974 U.S. vehicle emission standards were easily met with full size 1970 model sedans which were modified by changing combustion chamber, piston head, spark and valve timing, carburetion, and increasing the engine compression ratio. The acceleration performance and city/suburban fuel economy were improved over that of unmodified 1970 cars and were markedly better than comparable 1974 model vehicles.

A 1971, 1.6 liter Pinto was equipped with the Du Pont Total Emission Control System (TECS) and driven 100,000 miles on leaded gasoline. It easily met interim Federal emission standards in effect for California for 1975. This emission control system used exhaust manifold thermal reactors, exhaust gas recirculation (EGR), and carburetor and spark timing modifications to control gaseous emissions. In road tests the Pinto low emission car gave 6 percent better fuel economy than comparable 1973 models which met less stringent emission standards. This low emission vehicle was equipped with a muffler lead trap which reduced the total lead emissions by 84% without deterioration in efficiency over 100,000 miles. This emission control system has been used on standard sized vehicles equipped with V-8 engines with similar results.

Both large and small vehicles have been equipped with catalytic exhaust emission control systems. The fuel economy of these vehicles designed to meet a range of emission standards have been determined. Potential advantages and disadvantages of the various systems with respect to fuel consumption are discussed.

Automotive Engines for the 1980's

Robert W. Richardson

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The reciprocating piston engine has dominated the automotive scene for more than 60 years and until very recently, at least to most realists, seemed unlikely to ever be displaced. Although the piston engine has served its users well for many years and is likely to continue to do so for some time to come, it does have a number of shortcomings which are becoming more serious as ever greater numbers come into use and as we become enlightened on social values. It is a major contributor to air and noise pollution. It is also relatively inefficient and has a narrow fuel tolerance consuming large amounts of highly refined petroleum.

The early phases of an expected long-term energy crisis are now upon us. The era of abundant low cost energy is over. Much higher prices are certain and rationing likely cannot be avoided. The need to greatly increase efficiency rather than trade off efficiency for emission control is therefore becoming more obvious. Before the end of this century (which is closer than the end of World War II) petroleum must likely be replaced as the dominant fuel for mobile powerplants.

Although much progress has been made in reducing automotive emissions, it has been achieved at the price of increased fuel consumption. Much further reductions in emissions are needed to meet the requirements of the Clean Air Act of 1970. Growing, but of somewhat lesser importance is the issue of noise pollution.

Wankel, Stirling, turbine, stratified charge and diesel engines are the most serious contenders to replace or supplement today's piston engines. Electric vehicles are not considered serious contenders because of grossly inadequate technology and steam engines have too low an efficiency.

In addition to the three social parameters discussed previously, there are seven other major engine selection parameters - flexibility (torque-speed characteristics and driveability), smoothness, cost, weight, size, maintenance requirements and durability. Figure 1 lists these parameters in order of importance for passenger cars as of 1973. Arrows show the importance of noise and especially, fuel consumption rising to late 1970's (and perhaps Mid-1970's) values. The five contenders are compared on these ten parameters with the 4-cycle gasoline piston engine.

The Wankel, despite much recent fanfare, has little to offer in the three important social areas and uses substantially more fuel. It is also a more costly and less flexible engine and has poorer durability characteristics.

The Wankel is smaller and lighter, but nowhere near as much as often claimed. These advantages are not readily convertible into major reductions in vehicle size and weight. Design studies indicate that several of the most compact cars using transverse piston engines would have to increase in length if a Wankel engine were substituted.

The turbine engine is quieter and can have very low emission but has higher fuel consumption. It is lighter, smoother and more flexible, should require less maintenance, but is costly and its durability has not been proven (automotive application). The turbine requires considerable additional development before it could enter volume production.

The Stirling engine has the lowest fuel consumption, lowest emissions, and the lowest noise of any known engine. It is potentially capable of burning any fuel since it is an external combustion engine. It is becoming increasingly apparent that we must supplement or begin to replace petroleum consuming mobile powerplants within the next 10 to 20 years. The Stirling engine also has flexibility, smoothness, maintenance and durability advantages, but tends to be somewhat bulky and costly.

The Stirling engine is in an early state of development. Introduction in high volume production is not likely until at least the early to Mid-1980's.

Stratified charge engines could be introduced relatively quickly into production as it is a variation of today's piston engine. The stratified charge engine provides a better trade off between fuel consumption and exhaust emissions; the engine appears to be capable of meeting the interim 1975 and 1976 emissions standards while equalling or bettering today's engines' fuel economy.

Stratified charge engines have a disadvantage in that their specific power output is somewhat less than conventional engines, resulting in lower performance cars or an increase in engine size. Ultimately this disadvantage may be overcome by turbocharging but at least the first generation of stratified charge engines are not likely to use turbochargers.

Diesel engines have low fuel consumption and low emissions of controlled pollutants but high emission of smoke,odor and noise. They require less maintenance and have a long life but are at a disadvantage in all other characteristics.

On balance therefore, the stratified charge reciprocating engine appears to be the leading near-term challenger and the Stirling engine, the leading long-term contender.

Figure 2 is a composite chart showing our estimated range of probable market penetration of each engine type through 1985. The lower dark shaded band is for the Wankel. The maximum probable is about 13% by 1980 and 23% by 1985. The minimum probable rises to 3% in the late 1970's gradually fading away in the early-1980's. Second, for the turbine and Stirling engines - penetration again, from none up to 8%. The balance of the market, the reciprocating piston engine is obtained by subtracting the sum of turbine and Wankel minimum and maximum penetrations from 100. It would have a market share of at least 69% and could conceivably take the whole market in 1985. The maximum piston engine market share in 1980 is 97% due to the forecast minimum Wankel penetration. The number of catalyst-controlled reciprocating engines will be substantially lower than shown if the 1975 standards are liberalized. The picture for 1976 and beyond is still very unsettled.

The catalyst curve shows an early decline as the stratified charge engine comes into use. The stratified charge engine may indeed prove sufficiently attractive to not only take over this whole reciprocating engine segment, at least 69% of the total, but to even recapture the small segment lost to the Wankel in the mid- and late-1970's. By 1985 the stratified charge engine could be the only engine in production.

In conclusion:

1. Reciprocating piston engines will remain dominant well into the 1980's.

2. Vehicle and engine manufacturers continue to approach change with caution and will follow conservative introduction and commercialization strategies.

3. Economics will continue to be the dominant influencing factor.

4. But social requirements, especially fuel consumption, will become more significant in influencing change to different engines.

The overall conclusion, therefore, is that there still is considerable uncertainty as to the choice and rate of commercialization of specific <u>new</u> engines, but no revolutions are likely in the near future.

This summary is based on a complete report by the same title published by the Eaton Corporation.

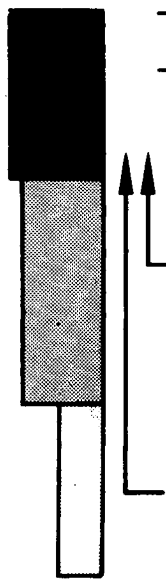
Major inputs for the report were obtained from over 60 in-depth interviews worldwide. These included car and truck manufacturers; heavy duty and small engine producers; developers of new engines; materials, parts, fuels and lubricants suppliers; machine tool builders; government agencies; trade associations; independent research institutes and consultants. These inputs were combined with business, technical and historical analyses and an evaluation of the social, political and economic forces that cause change.

Primary emphasis was placed on the Wankel engine and on those factors which will have the greatest bearing on its (degree and rate of) commercialization. Priority was placed on passenger car application followed closely by heavy duty markets with a relatively modest effort in the small engine area.

Relative Importance of Selection Parameter

Passenger Cars

Compared with 4-Cycle Spark Ignition Piston Engine



	Wankel	*Turbine	Stirling	Stratified Charge	Diesel
Flexibility	–	+	+	0	–
Smoothness	+	++	++	0	–
Emissions	0	+	++	+	+
Cost	–	–	–	?	–
Noise	0	+	++	0	–
Weight	+	+	0	–	–
Size	+	0	–	–	–
Maintenance	0	+	+	0	+
Fuel Consumption	–	–	++	+	++
Durability	–	?	+	0	+

Advantage (+) or Disadvantage (–) *Two-Shaft Regenerative 1900 F Turbine Inlet Temperature

FIGURE 1

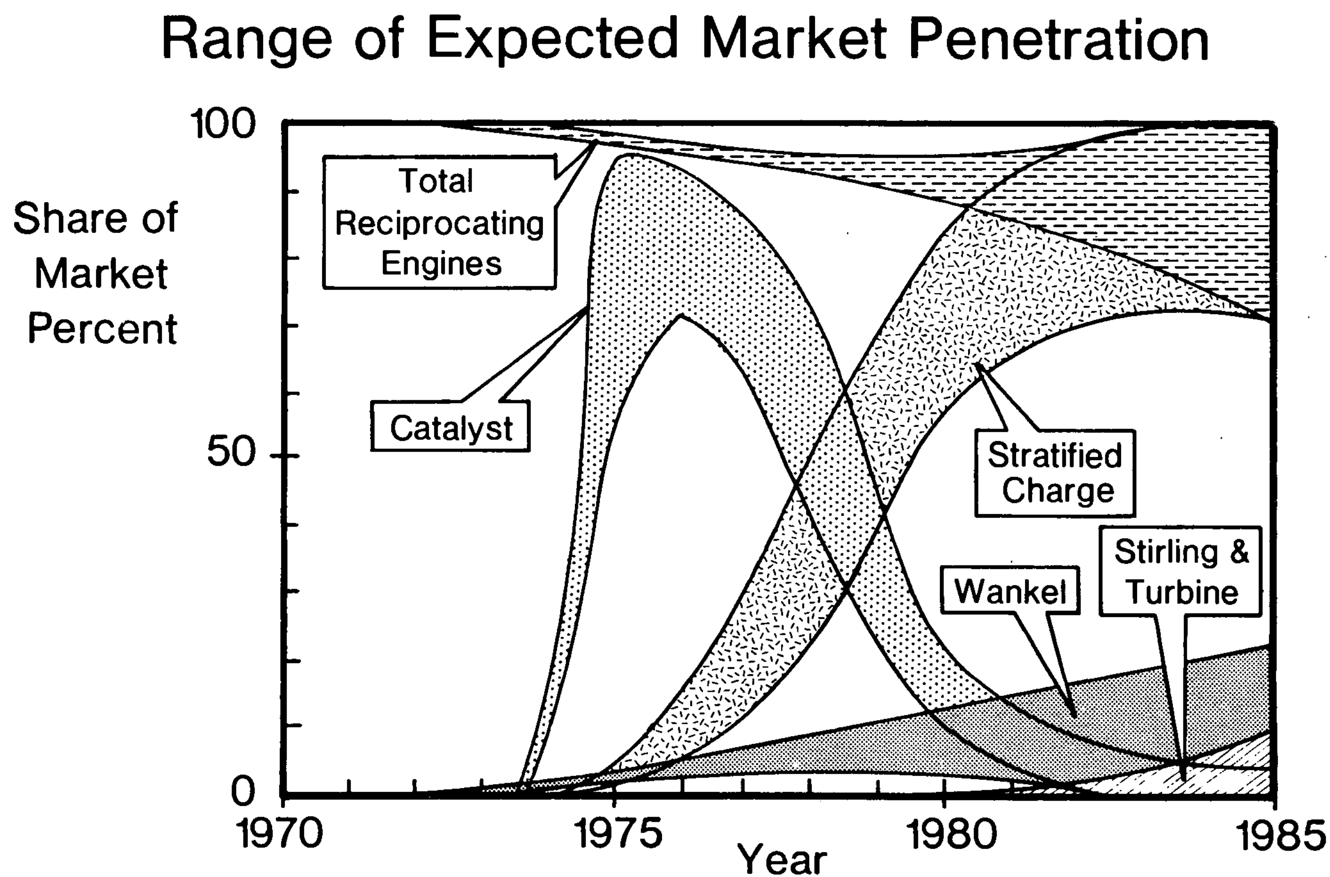

Range of Expected Market Penetration
Share of Market Percent
100
50
0
1970
1975
1980
1985
Year
Total Reciprocating Engines
Catalyst
Stratified Charge
Wankel
Stirling & Turbine

FIGURE 2

The Application of the High Speed Diesel Engine as a Light Duty Power Plant in Europe

C.J. Hind

Perkins Engines Company, Peterborough, England

The fact that the diesel engine has been considered and used as a saloon car power unit for some 40 years may come as a surprise to some people. They may admit that this is so but will come back with the reply that it has not made very much progress through the years. The diesel engine succeeded in getting a name very early on, and quite rightly so in some cases, as a dour thumping engine that plods on for ever, and not so flatteringly as a smelly, noisy,and rather smoky power unit. Very few of us would disagree with this description up to say 30 years ago, but great strides have been made since the mid-forties which have elevated the small diesel engine into a much more acceptable automotive power unit. The days when only an enthusiast or an eccentric would drive a diesel powered car are now passing and the wisdom and foresight of those early engineers is now bearing fruit. The design and combustion features of the diesel engine are showing to be more compatible with the strict legislative demands that are being thrust upon us and more people are now looking for a vehicle with good reliability, long life and maximum fuel economy. The words "fuel resources and energy crisis" are becoming commonplace these days and so it is worth remembering that the great redeeming feature of the diesel engine is its excellent fuel economy and low running costs.

But when did it all begin and why?

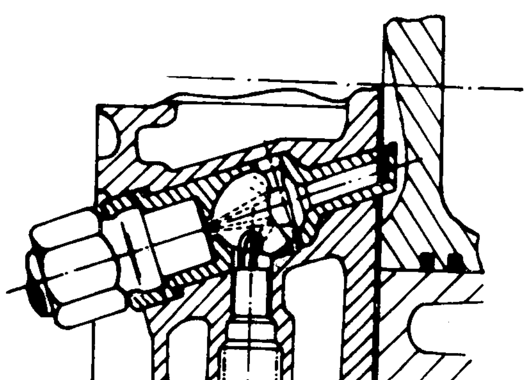
The beginning in Europe.

The early 1930's really saw the first production high speed diesel engines, and these required a whole new philosophy to be applied. The first diesel engines had been very heavy and bulky industrial and marine units with a maximum speed of around 1000 RPM, which made them unsuitable for vehicle applications.

Eventually the fuel economy shown by these engines, along with the attractive low fuel costs, made their progression into the commercial vehicle market a natural move. The rated speeds were raised to around 2000 RPM, although some of them remained below 2000 RPM, and in fact Gardner engines to this day still keep their rated speed in that same speed range.

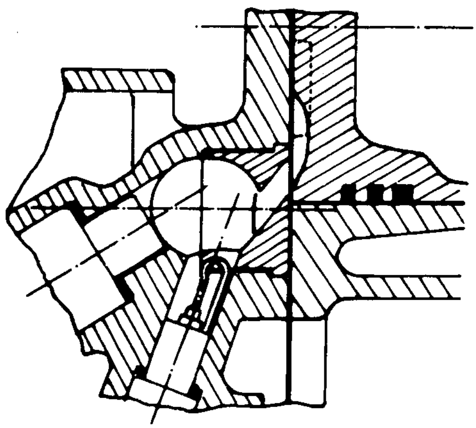
The rapid development of these engines from the mid-1920's to the mid-1930's was very impressive and the commercial vehicle operators attracted by the lower operating costs very soon saw the advantages of the diesel engined vehicle and helped this market to rapidly expand. Various companies, mainly in Great Britain and Germany, were developing these engines, whilst most of the French engines were being built under licence, excluding Peugeot who had extended their very successful petrol engine experience into the diesel engine field in 1928. Those early marine and industrial engines were made even more bulky by the fact that an air compressor was required to help atomise the fuel and provide the necessary air movement for good mixing. With the advent of the Bosch fuel injection equipment in Germany and later when C.A. Vandervell took up the manufacture of Bosch equipment in England, real strides were taken in the development process.

The high speed diesel engine, with rated speeds of 3000 RPM plus came to be used in the light truck market by two different roads. The company who manufactured both trucks and diesel engines saw the high speed engine as a natural extension of his engines in his trucks. The other approach was being made by the



MERCEDES - BENZ

FIG.1.



RICARDO COMET

FIG.2.

diesel engine manufacturer who offered to replace an existing gasolene engine in another company's truck. In the former case a vast amount of experience had been gained in the designing of the diesel engines for the bigger commercial vehicles and from this a large amount of knowledge was drawn which assisted in the development of the smaller units. In many cases the smaller engine was a scaled down version of its bigger brother, and the basic design and combustion principles were very similar.

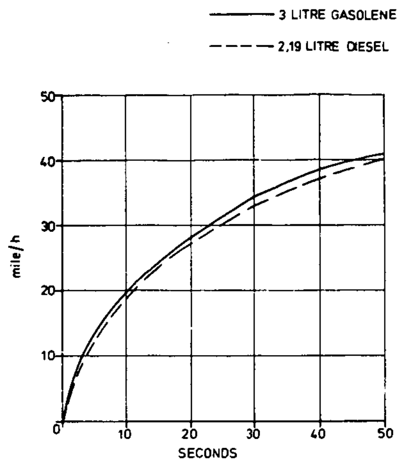
In the latter cases where an existing gasolene engine was being replaced by a diesel engine, a whole new design philosophy had to be applied, because interchangeability was a key factor and the diesel engine had to fit into the space vacated by the gasolene engine. As the transmission of the trucks was again designed for the displaced gasolene engine, this meant that the equivalent diesel engine had to have a similar speed and torque range. All this was a considerable break away from the traditional diesel requirement, and a large amount of design and development work was required.

It was realised early on in the development of the high speed diesel engine that cylinder pressures and engine breathing were going to be prime reliability and performance parameters.

The adoption of an indirect chamber engine allowed the intake port to be concerned only with inducing as high a mass of air as possible, and the swirl properties required for efficient combustion were provided by the air movement into and out of the chamber. Many designs of chambers were evolved during this time, each with its own theory and optimistic efficiency put forward by its inventor. One of the earliest and most successful designs was the Benz, later Mercedes Benz of course, pre-chamber or pepper pot design. This type of chamber has certainly stood the test of time as it is still widely used today and in many sizes of engines. This chamber was first used in the bigger design of engine, as was the well-known Ricardo Comet combustion chamber, which again underwent a smooth transition into the high speed engine, where it is still very widely used. See Fig. 1 and 2.

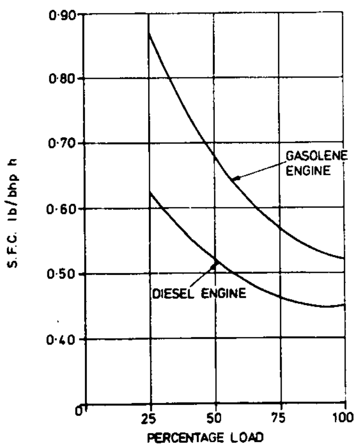
My own Company, Perkins Engines Company, was formed in 1932 specifically to manufacture high speed diesel engines for the lighter class of vehicle. As previously mentioned, interchangeability with the gasolene engine wherever possible was the primary aim. Fig. 3 shows comparative acceleration data from a road test of 4.2 GVW ton truck when fitted with its original 3 litre, six cylinder gasolene engine, and a 2.19 litre, four cylinder diesel engine. Both trucks had the standard gasolene transmission. The similarity between the two curves was very encouraging at the time, especially when the fuel consumption of 15 mpg for the gasolene engine and 25 mpg for the diesel was also considered. The rated speed of 3000 RPM was the same for both types of engine, and it was said that the diesel engine had run smoothly at 4000 RPM. It should be added that the engine was run ungoverned. The savings due to the substantially better fuel economy of the diesel engine were even more enhanced when one considers that gasolene in Great Britain in 1933 cost the equivalent of 17 cents per gallon, whereas the diesel fuel cost only 5 cents per gallon. The main reason for the difference was because the gasolene fuel tax was some eight times higher than that on the diesel fuel. In France diesel oil cost about half of the gasolene price, and in Germany an even greater differential of approx. 70% was seen.

Fig. 4 shows a comparative set of running costs that were issued in 1933 by the Commercial Motor. The considerably lower fuel costs are an obvious point, but the lower maintenance costs, even though the diesel engine was a new type of power unit, shows that one of the other virtues of the diesel engine, was born in those early development days. The diesel engined vehicle had a 20% lower maintenance cost than the gasolene engine.



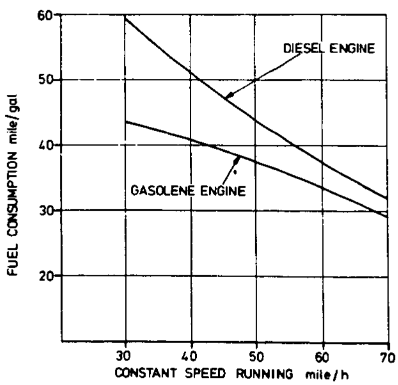
4·2 G.V.W. VEHICLE ACCELERATION WHEN FITTED WITH A GASOLENE AND DIESEL ENGINE

Fig. 3



CONSTANT SPEED FUEL CONSUMPTIONS OF SAME CAPACITY ENGINES IN DIESEL AND GASOLENE FORM

Fig. 6



CONSTANT ROAD SPEED FUEL CONSUMPTION

Fig. 7

FIG. 4. RUNNING COSTS (PENCE PER MILE) IN 1933 IN GREAT BRITAIN.

	Petrol Engined Vehicles			
	2 Ton	3 Ton	4 Ton	5 Ton
Fuel	1.33	1.80	2.10	2.63
Lubricants	0.06	0.07	0.09	0.09
Tyres	0.28	0.35	0.44	0.49
Maintenance	1.23	1.42	1.57	1.70
Depreciation	0.54	0.66	0.93	1.05
Total :	3.44	4.33	5.13	5.96

	Diesel Engined Vehicles		
	3 Ton	4 Ton	5 Ton
Fuel	0.38	0.44	0.55
Lubricants	0.12	0.16	0.16
Tyres	0.56	0.74	0.84
Maintenance	1.15	1.26	1.35
Depreciation	0.80	1.10	1.27
Total :	3.01	3.70	4.17

FIG. 5. CHANGES IN THE VEHICLE ROAD TAX IN GREAT BRITAIN IN 1934

Weight Unladen	Gasolene Pneumatic Tyres		Diesel Pneumatic Tyres		Diesel Solid Tyres •
	1933	From 1.1.34	1933	From 1.1.34	From 1 Jan '34
Under 12 cwt.	£10	£10	£10	£35	£46
12 cwt - 1 ton	£15	£15	£15	£35	£46
1 - 1½ ton	£20	£20	£20	£35	£46
1½ - 2 ton	£25	£25	£25	£35	£46
2 - 2½ ton	£28	£30	£28	£35	£46

• For gasolene engined vehicles with solid tyres the road tax remained unchanged at the same rate as the present pneumatic tyre tax.

This Utopia for the diesel engine vehicle could not last, and in Great Britain in 1934, they were penalised against the equivalent gasolene engine by a higher road tax. See Fig. 5. The new tax could be offset to some extent by the conversion from solid tyres to pneumatic tyres, and thus a saving of £11 per annum was possible. So this showed that technology was not altogether being retarded by the new laws.

One novel fact that was put forward was that the increased motor taxes could lead to more deaths. The reasoning behind this statement being that more people would now go back to horse driven carts, and these beasts attracted flies which killed more people by infection than did the motor vehicle by road accidents at that time.

Further pressure was applied to the diesel engine in 1935 when the British Government realised that there was a danger to its gasolene revenue, and so they increased the tax on the diesel fuel and made it equal to that on the gasolene.

A number of statements made at the time make interesting reading such as the Minister's statement that "The oil engine can do as much work on 1 gallon of fuel as the petrol can do on 1¼ gallons", and the pro-diesel faction who "believe that the oiler will continue to live and flourish but it must not be stunted in its youth", and the increase of tax even pleased some people as it would "encourage the steam vehicle trade". Times don't change that much do they?

This increase of tax was a considerable blow to all concerned in the diesel market, but work continued as the better fuel economy of the diesel was still worthwhile, but it now became even more essential that the first cost should be maintained as low as possible. This meant that the production principles and techniques that applied to the gasolene engine manufacturing industry, had also to be applied to the diesel engine wherever possible. This was especially essential for the smaller diesel engine, as it took that much longer to offset the first costs with the lower fuel consumption, simply because the total quantity of fuel consumed was small. The manufacturer who made both gasolene and diesel engines had an advantage in that he had many components at hand which he could design into both engines and maximise on rationalisation between the two types of engines.

The fuel injection equipment was, and still is, an expensive component in relation to the total engine first costs of a small diesel engine. This was, therefore, one of the main factors why the engine first costs were so high, and this coupled with customer inexperience of this type of equipment was a holding factor in the possibly even more rapid development of the smaller engine. Due to the commendable reliability of these first fuel pumps it was not long before most operators' doubts were dispelled and it soon became obvious that the reliability of the fuel pump was considerably better than that of the electric ignition equipment fitted to the gasolene engine. Consequently, the lower maintenance and down time costs were soon seen as a further bonus to the diesel engine vehicle operator.

The first diesel powered saloon cars.

It was obvious that the excellent fuel economy of the diesel engine would also prove attractive to the private motorists, and so the early 1930's saw parallel tests being run in both trucks and passenger cars.

The need for comparative size, weight, power and engine speed between the diesel engine and the gasolene engine became even more important when installation into a passenger car was considered. Further factors had also now to be considered such as noise, vibration and smell.

The first production diesel engined car was the Mercedes Benz "260D" which was powered by a four cylinder 2.6 litre engine which gave 45 HP at 3000 RPM. The car was normally fitted with a 2.3 litre gasolene engine. This diesel engine, the OM138, was a descendant of the pre-chamber truck engine and proved to be the very successful forerunner of a whole range of Mercedes diesel engines designed to suit the passenger car. The fuel consumption of 30 mpg and a maximum speed of 60 mph was very commendable, especially when the size and weight of the vehicle, which was really only a small transition from the light commercial vehicle, was considered. This car gave excellent service to many people, but of course the war years prevented any further development on these lines, and it was not until 1949 that a new model, the 170D, was seen.

The passenger car application was also being looked at in England in the early 1930's with an eye to Diesel conversion. In 1933, a 2.9 litre Perkins engine was installed in a gasolene production car and a creditable running cost of $\frac{1}{8}$ cent per mile was seen with equivalent performance to that given by the displaced gasolene engine.

Various capacity diesel engines were tested and one of the bigger conversions was a 3.8 litre Gardner engine rated at 83 BHP at 3200 RPM which replaced a 3.5 litre gasolene engine. This saloon car had a top speed of 83 mph and an overall fuel consumption of 44 mpg, which was considerably better than the 16 - 18 mpg achieved with the gasolene engine. A point of note was also that the conversion only added 100 lbs. to total vehicle weight.

The excellent fuel economy and reliability of these cars attracted people who had to cover very long distances, but even greater benefits were to be seen by the operators of stop start vehicles such as small delivery vans and taxis.

Further impetus to the development of the diesel engine was given by the political climate in Europe during the mid and late 1930's. Independence from imported fuels was aimed at, and so a variety of home produced fuels from coal and gas fuel were tested. As it was simpler to convert a diesel engine to operate on a variety of fuels rather than a petrol engine, it was generally the former which was the basic engine used for the development work.

The Second Era.

In 1949, Daimler-Benz produced the 170 Series of saloon cars. This model was the forerunner of a whole new series of passenger cars produced by this company, and has seen gasolene and diesel engines installed in parallel up to the present time.

The 1.76 litre diesel engine (OM 636) embodied much of the experience gained from the earlier 2.6 litre engine, and this enabled the smaller engine to have a rated speed of 3200 rpm and an output of 21.6 bhp/litre. The popularity of this vehicle is shown by the fact that 27,000 170D's were sold in the three years from 1949 to 1952. The first cost of the diesel engined car was only $185 more than the equivalent petrol model, and with a fuel consumption of 40 - 45 mpg, it took very little time before the diesel car was making a considerable saving.

This engine was developed further and in 1953 the 180D was introduced with the four cylinder engine now rated at 43 bhp at 3500 rpm, 24.4 bhp/litre, and a capability of 3800 rpm. These engines had a stroke/bore ratio of 1.33, but when a new 2 litre engine was introduced in 1959, it had a reduced ratio of 0.96, which allowed a higher operating speed of 4350 rpm and a specific output of 27.5 bhp/litre.

The European Continental countries still gave an extra boost to the development of the diesel engine in the early fifties by keeping the cost of diesel fuel well below the gasolene costs, whilst in Great Britain the difference in 1954 was only a little over 2.5 cents. There was also very little difference in fuel costs in the U.S.A. at this time and, so again, the incentive was low.

Various European Continental manufacturers now began producing diesel powered cars, such as Fiat in Italy and Borgward Hansa in Germany and eventually in 1954 the Standard Motor Company Limited began producing a saloon model in England. The essential point on first costs was pointedly shown by an automotive magazine at the time which stated that 61,200 miles was needed to be covered by this car before the high price differential of $640 was offset. This mileage was required on the basis of the diesel engined car giving 40 mpg as against 23 mpg of its equivalent gasolene engine.

The top speeds of the diesel car were generally some 10 - 20 mph lower than the gasolene, but even more frustrating was the poor acceleration. This generally was due to the prime essential of interchangeability. The specific output HP/litre of the diesel engine has always been lower than the gasolene and, as the engine bulk dimensions had to remain essentially the same for both engines, this meant that the diesel had a 10 - 15% lower power output, and a maximum engine speed between 1000 - 1500 rpm lower than the gasolene. In many cases the transmission ratios were not changed and so the vehicle performance suffered again from this. Sometimes an overdrive ratio was fitted which enabled a higher top speed, but the poor acceleration was generally seen as a big disadvantage to the average motorist.

The driver who covered very long distances and required a reasonable cruising speed with good reliability, found the diesel car to his liking.

An even more beneficial application was in the vehicle that used a stop start and low load factor type of operation.

The diesel engine has nominally a constant volumetric efficiency and compression ratio through the load range at a given speed, whereas the gasolene engine has to contend with falling values at part load due to the throttling of the air flow at these conditions. This difference is shown in the better part load economy of the diesel engine and so the stop start or part load applications show the diesel engine to considerable advantage.

Fig. 6 shows how the specific fuel consumption curves of the same capacity engine when tested in diesel and gasolene forms diverge at the part load condition. This feature when transferred to actual road running results shows that the light load running gives approximately three times the fuel saving seen at the high load factor running. See Fig. 7.

Various types of vehicles saw the economy of the diesel engine in this way in the mid 1950's, and the engine was used in applications varying from taxis to delivery vans and road sweepers.

The rapid increase in the use of the diesel engine for taxi applications was most spectacular in Great Britain. Fig. 8 shows how the first taxi was

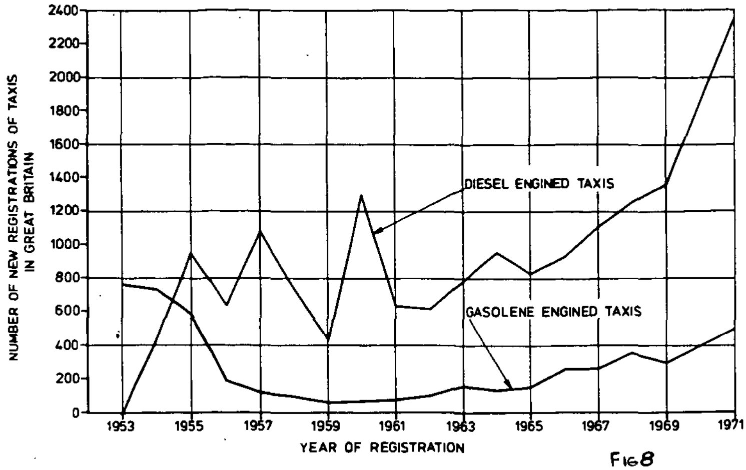


Fig 8

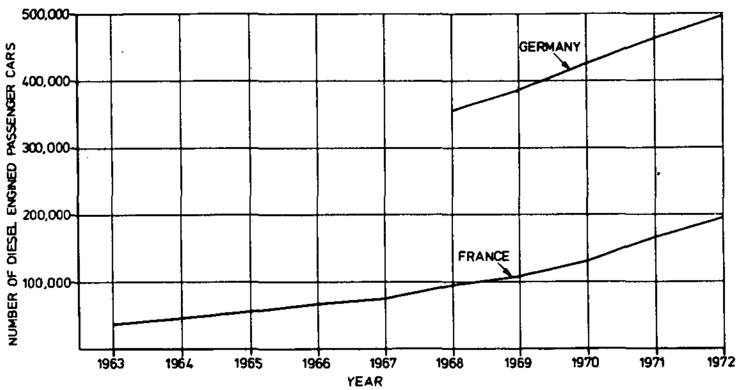


Fig 9

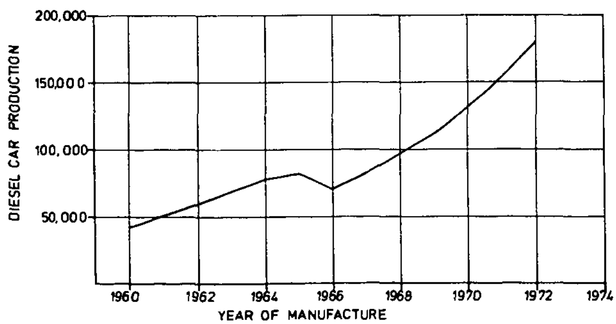


Fig 10

registered in 1953 and within 2 years the number of new registrations had overtaken that of the gasolene engined taxis. The late 1950's saw an erratic trend, possibly due to the economic climate at that time, but since 1961 the increase has shown a positive upwards swing. The rising trend of the gasolene taxi since 1961 is due to the number of smaller companies and individuals who are using their private cars in this market.

Fig. 9 shows how the German diesel passenger car market has always been the largest in the world, with an impressive figure of 0.5 million diesel cars being used in 1972. It is estimated that 45,000 taxis will be registered in Germany during 1972/73, and 80% of these will be diesel powered. This shows that the vast majority of diesel engined cars are being run by companies and the public for their private use and overall fuel consumption and reliability must be priority features as they are in this market in any country. The position in France since 1963 is also shown on Fig. 9, and although the actual numbers involved are much smaller, the trend shown from 1969 - 1972 is parallel to the German experience.

The owner of a motor car who travels above the average annual mileage, say 25,000 miles or more, will see the benefit of running a diesel car, and the auto-routes seen across the European Continent are ideal roads for this type of driving, as are the American freeways.

In Great Britain we do not have the road system, or even possibly the square mileage of country, to see the same usage of diesel engined cars as on the European Continent, and consequently the majority of these vehicles are used as taxis. As the fuel savings are so much greater at these part load running conditions, the mileage necessary to offset the higher first costs is much less. A typical difference in the fuel consumption for a London taxi cab type of duty would be 20 mpg for the gasolene engined taxi and 35 mpg for its diesel engined equivalent.

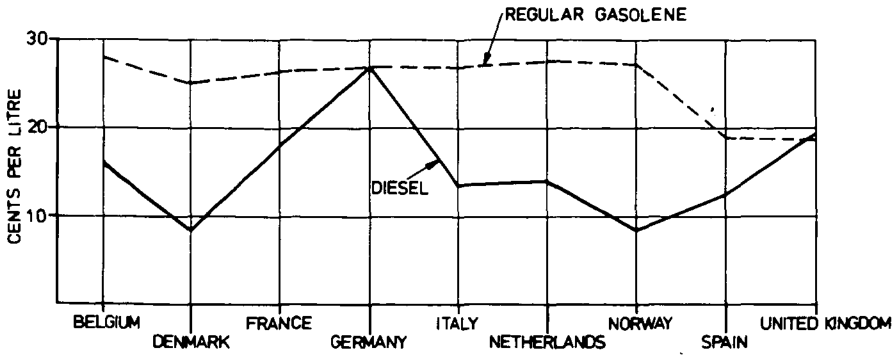
The 12cwt - 1 ton light van market is a very high quantity market, but as yet the diesel engine has made very few inroads. This again is essentially because of first costs, although to some extent the performance penalty is still felt in this low weight vehicle.

The 1 - 1½ ton vehicle market is also a very lucrative market, and the diesel engined vehicle is showing a steady rise here.

The light truck applications used by local Authorities for road cleaning, refuse disposal and other city work, see the advantages of the diesel in these applications. The part load economy again shows its benefit in these trucks, and the higher first costs can be offset in about 3 years. The reliability of these engines giving less 'down-time' and 'call out' problems is a further added bonus. These trucks give about 10 years' service before a major overhaul is necessary.

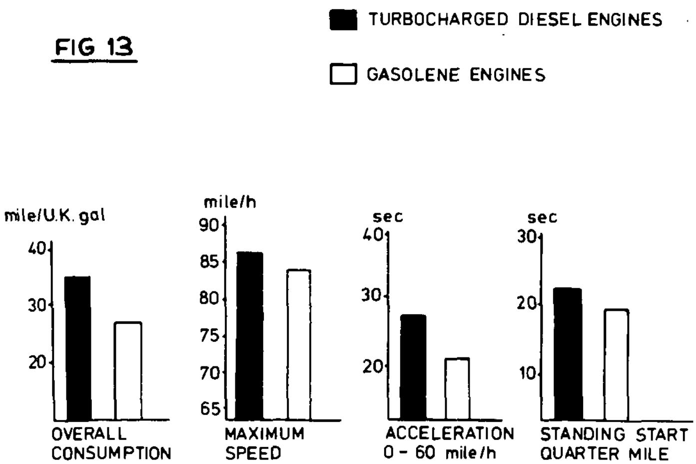
If we look at the production rate of the diesel engined car in Europe over the last 15 years, we see that there has been a positive increasing rate - See Fig. 10. The graph does not include conversions but only production line cars.

This trend proved attractive to more gasolene engine car manufacturers and today we have three major manufacturers who produced a total of 180,000 diesel engined saloon cars in 1972. Between them, these manufacturers - Mercedes Benz, Peugeot and Opel, produce a wide range of diesel engined vehicles ranging from a small saloon to an 8 seater limousine.



RETAIL PRICE OF DIESEL AND GASOLENE FIG.11

FIG 13



All the vehicles have four cylinder engines, even the biggest which has a 2.4 litre engine, and the highest rated speed is now a creditable 5350 rpm seen from the Peugeot cm^3 engine.

Whilst the diesel engine has been making considerable strides in its development with an eye on the saloon car market, the gasolene engine has, of course, equally been intent on further development and, consequently, it would be true to say that the performance gap has not decreased. The performance of the gasolene engined car has improved substantially since the end of the 2nd World War, and so in a direct comparison the gasolene car is still superior in acceleration and maximum speeds. But we must not let this overshadow the developments that have been seen in the diesel engine, where a 50% increase in rated speed has been achieved and specific outputs have nearly doubled. Without a doubt, the saloon car market has provided the stimulus for this development, and many people believe that the potential world market for the diesel engined car and light truck has yet to be exploited.

Today's gasolene engined car has on average still a 10 seconds advantage on a 0 - 60 mph acceleration test, and a top speed some 15 - 20 mph faster, but in these days of increasing legislation to reduce speed limits, the diesel car performance giving 75 - 85 mph is more than adequate.

We still have the old problem of first costs and the basic price differential varies from $250 to $750, but equally so we also still have the considerably better fuel consumption from the diesel car. Such adjectives as "astonishing", "tremendous" and "dramatic" are frequently used when people compare the fuel consumptions of these cars and, in general, they give 60 - 70% miles more per gallon than their gasolene engined counterparts.

We have seen how economy has always been a paramount factor in the sales of diesel cars, and this was undoubtedly helped by the beneficial differential in fuel costs seen in most European countries. It is possibly a demonstration of the insight and gratitude of the politician to see from Fig. 11 that Germany, who for so long has been the leader in the diesel car market, has now, along with the United Kingdom, the dubious honour of having no or even an adverse cost differential when compared with current gasolene prices. Extra strength is really given to the case for the diesel engine by this fact, as the fuel economy is still being seen as a worthwhile factor in purely mpg terms.

THE FUTURE:

If we now look into the future, how do we see the diesel engined saloon car in the light of legislative and fuel resource pressures.

The use of the I.D.I. combustion principle for the small diesel engine began, as I said before, at the very beginning of the development era of the diesel engine. Some people might call it foresight, fortuitous or just luck, that this type of engine is now proving to be a much better emission controlled engine than either the D.I. diesel engine or the gasolene engine. But really the fact that they were chosen because they had lower cylinder pressures, along with better breathing, is the reason why this combustion principle is now showing to advantage in these days of low NO and noise. Lowering the rates of pressure rise and peak cycle temperatures by retarding the injection is a well known principle and in the I.D.I. engine this also has the added benefit of reducing the exhaust smoke. This later timing also reduces the combustion noise levels and so we gradually have a situation where the previous disadvantages of the diesel engine are also being reduced. Taking the old problem of installation. If we can sufficiently decrease the rate of cylinder pressure rise and hence the combustion noise, at both high and low speeds, it may be possible to reduce the bulk and weight of the diesel engine and so reduce the installation problems, and at the same time reduce the first cost differential.

This principle has of course to be investigated in considerable detail and analysis, or the situation will arise where the reduction in engine bulk will allow more noise to be released.

By extensive analysis of the cylinder block loading and vibration it may be possible to design a block which can distribute the loading more effectively and so reduce the noise generating sources along with a reduction in engine bulk.

The diesel knock becomes more obtrusive in the car application at the lower engine speeds, and means of reducing ignition delay periods and smoothing out the rates of cylinder pressure rise seen at part load conditions will have to be found before the average motorist will be satisfied. His previous experience of such sounds with his gasolene engined car has usually given him visions of failing bearings and pistons, and possible some re-education is needed to convince him that the diesel engine is designed to withstand these loads.

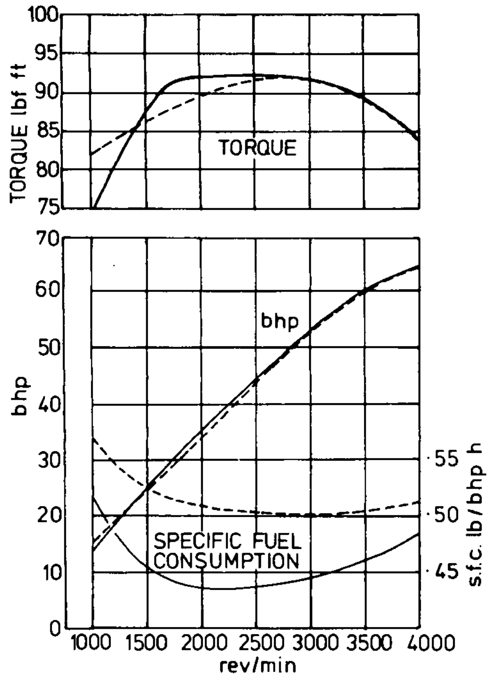
The U.S. legislation on g aseous emissions has caused enormous headaches for all engine manufacturers all over the world.

The manufacturers of the gasolene engine have been the hardest hit, but after all it was them who created the problem in the first place and are now experiencing the greatest difficulty in meeting the stringent requirements.

Many estimates and gloomy predictions have been made on the reduced power, increased fuel consumption, and increased first costs of the gasolene engined car that can meet the 1975/1976 and subsequent years' legislation. The Honda CVCC, and various rotary engine design concepts have been developed so as to meet the legislation, whilst the standard reciprocating gasolene engine has had to introduce many external innovations. The I.D.I. diesel engine has many of the required design and combustion features already built into it and any further modifications will generally come about by engine internal modifications. This means that the offending pollutants are not generated in the first place, and so the need for expensive corrective action is not required.

The number of engine modifications required by the diesel engine are relatively small if the 1975/76 Federal limits are to be met, and it is generally true to say that the stricter the limits the more able the diesel engine is to meet them. A very small power and SFC penalty is expected from the diesel engine if it has to meet the projected 1975 California legislation, and with only marginal increased cost. Very few figures are released by the gasolene engine manufacturer on the effects of tidying up his emissions problem, but considerable power derates, increases of vehicle weight, increases of first costs, and most critical of all increased fuel consumption, are all factors which will generally apply.

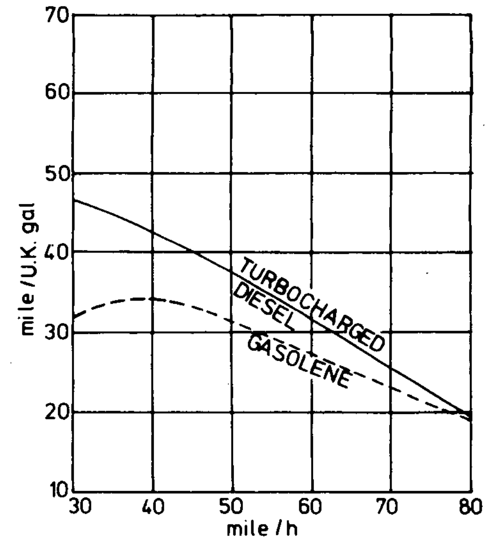
The lower specific output of the diesel engine has to be increased if it is to effectively compete on a performance basis with the gasolene engined saloon. This increase can come about by turbocharging and along with it, further improvements in fuel economy. The turbocharger will obviously increase the first costs, but these will be more than offset by the very



—— TURBOCHARGED DIESEL ENGINE.

----- GASOLENE ENGINE

FIG. 12



CONSTANT SPEED FUEL CONSUMPTION

FIG. 14

expensive catalytic converters required by the gasolene engine.

Fig. 12 shows a test bed comparison between a 4-cylinder 108 cu. in. turbocharged diesel engine and a 104 cu. in. gasolene engine. The gasolene engine was in standard, non-de-toxed condition. Since 4000 rev/min was the maximum speed of the diesel, the gasolene curve was also discontinued at this speed although not reaching a maximum until 4800 rev/min. The superior fuel consumption of the diesel is clearly shown.

Each engine was installed in a UK Ford passenger car and comparative road test data obtained. Histograms of fuel consumption, maximum speed and acceleration are shown in Fig. 13. The standing start acceleration of the diesel powered vehicle was slightly inferior to the gasolene car, due mainly to the higher rotating inertia of the diesel engine and heavier installed weight. Top gear acceleration above 40 mph was, however, better with the diesel engine, as was the top speed. Fuel consumption was considerably better with the diesel, particular at lower speeds. Fig. 14 shows the steady speed fuel consumption at various speeds.

So the turbocharger will give improved performance and fuel economy to the diesel engined vehicle, whilst its gasolene counterpart is subjected to reduced performance and economy.

Two more fundamental yet substantial changes may be required to the diesel saloon philosophy, which affect both engine and car manufacturer, if this type of vehicle is to be fully accepted.

First, engines of six cylinder configuration may be required, one manufacturer has split the difference and is working on a five cylinder engine, but if powers over 120 BHP are required then a turbocharged six cylinder will be the answer.

Second, the transmission should be designed for the diesel engine and, if the engine is turbocharged, then a torque converter should be matched to its torque curve.

I therefore foresee the role of the small high speed diesel engine increasing in the light duty market, and this market potential should provide a real stimulus to the diesel engine manufacturer to further improve his product and prove that the image of the diesel engined car is due for a well deserved brush-up.

The vehicle manufacturer has to accept that the transmission has to be developed around the diesel engine, and if a concerted effort was made by all parties concerned, the late 1970s and into the 1980s could see improvements in both environmental conditions and a substantial reduction in the rate of exhaustion of our valuable fuel resources.

CURRENT LIMITS FOR LIGHT DUTY DIESEL ENGINES:

1. Power - This is dependent of speed (rev/min) and brake mean effective pressure (b.m.e.p.).

2. Speed - For the size of engine considered, the limiting factor is usually mean piston speed. Problems arise if diesel engines are operated for sustained periods at piston speeds over 2500 ft/min. Some gasolene engines operate at up to 3500 ft/min. Fig. 15 shows the permissible stroke dimension for various maximum engine speeds.

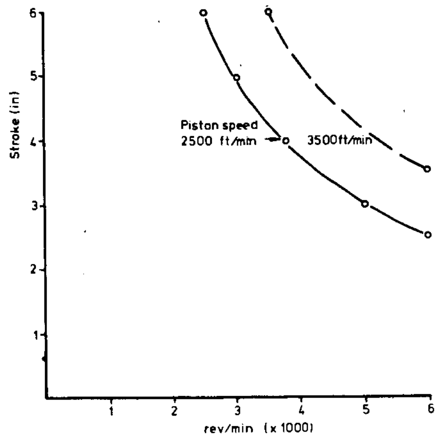


FIG.15

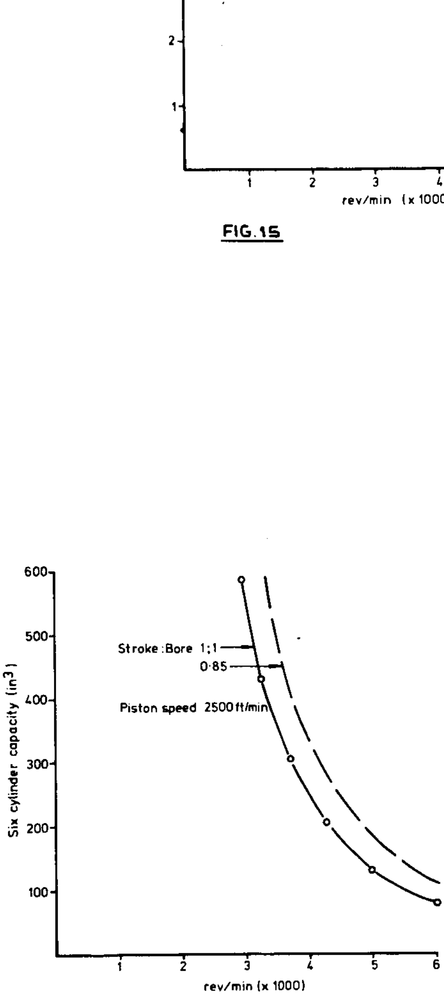


FIG. 16

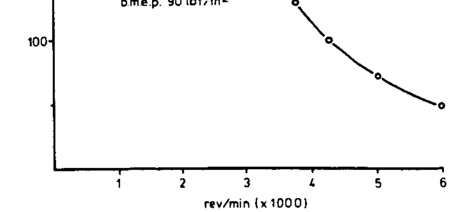


FIG 17

3. Stroke to Bore Ratio - For indirect injection diesels, a stroke/bore ratio of between 1.0 and 0.85 is possible. This therefore sets a limit on cylinder capacity for a given rev/min. and piston speed. Fig. 16 shows the permissible maximum speed of various capacity six cylinder engines.

4. B.M.E.P. - Normally aspirated diesel engines should produce 90 lbf/in^2 b.m.e.p. at maximum speed. Using this value, Fig. 17 shows the horsepower limit at various rated speeds for the six cylinder engine.

5. Supercharging - More power can be obtained by turbocharging, but limited by the temperature of pistons, rings, cylinder head face and valves, and cylinder pressure. By turbocharging, an increase in power of 30% may be expected.

6. Engine bulk - Diesel engines tend to be longer than gasolene engines due to water passages between bores, more robust crankshaft and bearings and heavy duty timing drive.

   Siamesed cylinders may be used for light duty applications, but problems due to cylinder distortion are likely.

   The height is usually greater than for an equivalent gasolene engine, due to longer stroke and thicker head. Carburettors, however, frequently add to the height of gasolene engines. Oil pans tend to be deep to hold a larger volume of oil.

   There is little difference in engine width, particularly in-line engines.

   The bulk of a diesel is likely to be up to 50% greater for a given cylinder capacity.

7. Engine weight - Where cast iron is used for the blocks and heads of both diesel and gasolene engines, the diesels are usually heavier. This can amount to 100% more for equal power, normally aspirated.

   Fig. 18 shows a comparison between a light commercial vehicle diesel engine and a typical compact car gasolene engine.

FIG. 18.

ENGINE COMPARISON

MAIN DIMENSIONS.

	Diesel Engine	Six Cylinder Gasolene Chrysler 225 ins^3 "RG" Inclined 30° from Vert.
	inches	inches
Cylinder block length	27.6	26.1
Length engine from rear face cylinder block to front of fan	36.6	31.0
Height of water pump	8.2	6.7
Depth of sump	10.6	8.6
Height above crankshaft	18.4	18.3
Overall height	29.0	26.9
Width L.H. Looking from drivers seat	10.7	9.0
Width R.H. Looking from drivers seat	12.5	13.8
Overall width	23.2	22.7
Engine weight (lbs.) (dry)	708 Flywheel plus backplate. Starter plus alternator plus fan.	475 Alternator plus air cleaner only. 555 lbs. if to equivalent specification