were the same with respect to speed and load, with NOx output raised at either increased load factor or higher engine speed. There was however a significant difference in the order of magnitude between fuels with gasoline producing significantly higher NOx emissions than diesel fuel at equal engine power.

For CO emissions, the trends between fuels were identical with speed but different with regard to load. With gasoline fuel, lowest CO emissions at a given speed were apparent at the lowest load factor tested. For diesel fuel, CO emissions tended to be at a minimum for the intermediate load tested. CO emissions were also much lower with gasoline fuel than diesel at equivalent power output.

HC emissions were markedly lower with gasoline than diesel fuel at equal power. With diesel fuel, HC emissions were minimum at intermediate load with a marked increase evident at either full load or light load. In the case of gasoline, HC emissions were lowest at full load and progressively rose with reduction in power output.

Although the L-163-S engine represents a multi-fuel development, these data clearly indicate major fuel differences in respect of emissions and tend to suggest improved combustion efficiency with the gasoline fuel judged by the lower HC and CO emissions and higher NOx output compared with diesel fuel. This view is supported by the specific fuel consumption results, Figure 64. In comparison with the IDI diesel engine, the absolute levels of these emission rates are high on both fuels, particularly with respect to HC and reflect in high values of simulated FTP results (111).

Other studies conducted by Bartlesville with a similar multi-fuel TCCS engine suggest that a 50/50 blend of gasoline and diesel, representing a broadcut fuel, yielded intermediate emission results.

In obtaining the aforementioned results from the L-163-S engine, the injection and spark timing were left coincident and the same for both fuels as per the multi-fuel philosophy of the TCCS system. The scope for potential emissions improvement with single fuel optimisation by changing for example injection and spark timing, either individually or together, exists. To a limited extent, such possibilities have been examined by Bartlesville (112) and results are shown in Figure 67.

In these experiments, the LIS-183 TCCS multi-fuel engine was utilised with gasoline, broadcut and diesel fuels. Initially, the influence of spark retard relative to injection timing for a range of injection timings was evaluated at part load. For a given spark/injection timing relationship, advancing the injection timing results generally in higher NOx and CO emissions for all fuels. HC emissions also tended to increase although the trends were relatively insignificant with the broadcut and diesel fuels.

At fixed injection setting, maintaining coincident spark timing returned either no penalty or minimum levels with respect to HC and CO emissions for all fuels. For these emissions, spark retard showed a marked adverse influence with the more volatile fuels, especially gasoline, presumably due to the ability to formulate lean mixtures incapable of efficiently supporting combustion. With regard to NOx emissions, spark retard appeared to have little effect with the broadcut and diesel fuels but tended to reduce emissions with gasoline fuel, this being in accord with the observations for HC and CO emissions.

These data suggest that maintaining coincident relatively retarded events will generally offer the best emissions results.

For this engine, at a given injection timing with coincident spark, emissions recorded from several part load conditions revealed that gasoline gave the highest HC emissions and diesel fuel the lowest, with broadcut essentially intermediate. CO emissions were generally lowest with gasoline with similar levels being emitted by diesel and broadcut fuels. NOx emissions were relatively unaffected by fuel type although at the higher load factors and speeds tested there was a tendency for NOx emissions to increase with diesel fuel.

Under full load conditions, dictated by the smoke limited torque on each fuel, NOx emissions were significantly lower with diesel fuel, this being commensurate with the lower smoke limited output. HC emissions were also in accord with full load combustion efficiency by being highest with diesel fuel. CO emissions were generally lowest for gasoline.

These data indicate as per the previously reported Bartlesville study that the emissions response of the multi-fuel TCCS concept can be appreciably different between fuels. It is interesting to make comparisons between the LIS-183 and L-163-S TCCS data. It can be noted that the L-163-S engine tended to give much lower HC and CO emissions for gasoline compared with diesel fuel, whilst NOx emissions were higher. In respect of the LIS-183 engine, the converse was generally applicable for HC and NOx emissions. In addition, there appears to be a large difference in the magnitude of HC and CO emissions between the two engines. These comparisons either suggest a significant sensitivity of the TCCS system between engines or simply reflect standards of development. When considering these data, it should be borne in mind that Texaco* regard the particular L-163-S engine used by Bartlesville (111) as not being representative of TCCS performance owing to a low energy ignition system and incorrectly set timings.

5.2 Transient Gaseous Exhaust Emissions

A selection of 1975 FTP gaseous emission results are shown in Table 1 attached to this Appendix for various vehicles fitted with TCCS multi-fuel engines. It is not proposed in this report to deal with the development history of TCCS emission reduction programmes as the main interest is the tolerance to different fuels. These results are only dealt with briefly therefore.

Table 1 shows that untreated emissions from the TCCS system (Line Nos. 1, 2, 3, 11) are significantly higher than current IDI diesel vehicles as regards HC and CO and are nearer to non-catalyst equipped gasoline vehicles. For NOx emissions, L-141 engined vehicles in both N/A and T/C forms can achieve levels of 1.5 - 2.0 g/mile whilst the L-163-S applications require EGR to meet similar standards.

Texaco have taken advantage of the tolerance of the TCCS system to EGR (see Section 6) and have achieved very low levels of NOx in both Jeep and Cricket vehicles.

These data show NOx control to below 0.41 g/mile whilst maintaining HC and CO levels below 1981 Federal light duty requirements. These levels have

* Comment extracted from correspondence between Texaco and DoE November 1980, submitted following Texaco's examination of the original draft of this report.

been held for 50,000 miles with the N/A L-141 Jeep vehicle over a constant speed dynamometer schedule (see Line Nos. 7,8). Such results have been achieved by the use of hang-on devices in addition to EGR and combustion retard (see Section 6). Hang-on devices include throttling (see Section 6), exhaust back pressure, 2 catalysts and, in the case of the N/A L-141, a catalytic swirl reactor in place of the standard exhaust manifold. The penalty of meeting these combined low emission levels is approximately 20-30% loss in economy and 30% power reduction. Relaxing NOx control to 1-1.5 g/mile by eliminating EGR reduces these economy and power penalties to approximately 10-15% and 15% respectively, whilst still maintaining HC and CO control closely in accord with 1981 requirements.

Of more direct interest in the context of this report is the emission response to various fuels. Suitable data in this respect are shown in Table 2.

In the case of vehicles equipped without catalysts (Line Nos. 3 and 4) there appears to be a definite tendency for gasoline to return higher HC emissions than JP-4, broadcut and diesel fuels whilst diesel fuel gave the lowest HC emissions. Texaco (118) state that these trends are due to the lower self-ignition temperature of the heavier fuels. CO emissions were a factor of 2 higher with diesel fuel compared with gasoline whilst JP-4 and broadcut emitted similar levels nearer to gasoline. The higher CO emissions with the heavier fuels were attributed to the onset of overfuelling (118). This implies that the vehicle power/weight ratio was such that full throttle was required to meet certain conditions of the test cycle. The trends of of these emission results between fuels are in broad agreement with the Bartlesville (112) test bed data reported earlier in this section.

The addition of a catalyst (Line Nos. 1 and 2) reduces the magnitude of the emitted pollutants but the trends between fuels are still evident with gasoline giving higher HC emissions than diesel fuel and lower CO. In this case, broadcut fuels appear to return the lowest HC output while the CO response is mixed.

These data again reveal that, although the TCCS system is multi-fuel, emissions can be influenced by fuel specifications suggesting scope for improvement by single fuel optimisation. This assumes that a fuel/combustion process relationship exists and that the trends observed are not fully accounted for by the arguments previously stated.

Regarding NOx emissions, clear fuel trends do not seem apparent from these data, although changing from gasoline to diesel fuel appears to have little effect.

6. RESPONSE TO GASEOUS EXHAUST EMISSIONS CONTROLS

The FTP results obtained from TCCS equipped vehicles have already been presented and it has been reported that various emission controls are utilised. The following results indicate the response of the TCCS engine to some of these control techniques and have been obtained from single and multi-cylinder L-141 test bed engines using gasoline as fuel (116).

These results demonstrate that at light load, the application of intake throttling appreciably reduces HC and NOx emissions by approximately 40-50% for example at a load of 30 psi imep at 1500 rpm with a fuel consumption penalty of 10%. Light load throttling eliminates the formation of very lean mixtures which can pose combustion problems and lead to misfire and in addition significantly elevates exhaust temperatures. Both of these factors aid HC control whilst the latter assists in the more efficient application of catalysts and thermal reactors. CO emissions were little affected by throttling. Data were not presented to illustrate whether throttling was similarly advantageous for controlling emissions with other fuels.

The TCCS system behaves like diesel and gasoline engines to retarded combustion. In the TCCS engine, retarding the coincident spark and injection timing at 1500 rpm reduces NOx emissions at high load factor but has little effect at low load factor. At high load, NOx emissions can be reduced by 40% for a 5% fuel economy penalty. HC and CO emissions are not seriously affected by retard, no doubt aided by the elevated exhaust temperatures. Data are again not submitted to show the influence of retard with other fuels.

The TCCS system also reflects the classic emission economy trade-off with exhaust gas recycle (EGR). The data reported for the L-141 multi-cylinder TCCS engine operating at 2000 rpm, mid-load, demonstrate a marked NOx sensitivity to relatively small degrees of EGR compared with Ricardo IDI diesel experience. With the application of 10% EGR, NOx emissions were suppressed by approximately 60% for little change in either HC or CO emissions. Fuel consumption was penalised by approximately 6% however. Raising EGR rates above 10% incurred progressively increasing HC, CO and fuel economy penalties whilst NOx levels continued to be suppressed at a lower rate. These data show that the TCCS system has a high tolerance to EGR with large NOx reductions attainable with moderate rates of EGR without incurring severe penalties in other areas. Similar data were unfortunately not available for other fuels.

7. EXHAUST PARTICULATE EMISSIONS

Bartlesville (120) have recorded 1975 FTP particulate emissions for a Gremlin vehicle equipped with a L-163-S TCCS engine. This vehicle is emission controlled with EGR and catalysts to NOx and HC levels of approximately 1.5-2.0 and 1.0 g/mile respectively with unleaded gasoline as fuels. Under these conditions, particulates ranging between 0.06 and 0.16 g/mile with a mean of 0.09 g/mile were recorded.

Blending No. 2 diesel fuel with gasoline increased mean particulate emissions to 0.18 and 0.3 g/mile for 25% and 35% diesel added by volume respectively. Particulate output with shale derived gasoline was significantly higher than regular gasoline and averaged 0.22 g/mile.

Without access to further data, the trends of the particulate results are difficult to explain. The higher particulate emissions obtained with the diesel blends in comparison with regular gasoline leads one to speculate that reduced fuel volatility may have been responsible. This can largely be dismissed however owing to the higher particulate output observed with shale gasoline. One potential contributor to higher particulate emissions in the case of the diesel blends could be sulphate generation by the catalyst upon the higher fuel sulphur levels induced by diesel blending. This may also be the case with the shale gasoline although additional impurities could also be responsible.

The generation of sulphate emissions to this extent assumes that exhaust temperatures are generally above approximately 500^{o}F throughout the cycle, in order for the catalyst to act efficiently as regards sulphur dioxide conversion. Such levels, if apparent, are in excess of typical IDI diesel mean cycle exhaust temperatures. It is understood* that this particular TCCS installation does not employ intake throttling which may have given rise to higher cycle exhaust temperatures with respect to the IDI diesel. It may therefore be speculated that higher exhaust temperatures with the TCCS powered Gremlin are due to lower power/weight ratio in comparison with the IDI diesel vehicle. This view may be substantiated by recent Texaco comments* relating to induced overfuelling with more dense fuels i.e. diesel/gasoline blends as being responsible for increased particulate emissions. This implies power/weight ratios dictating the frequent use of full throttle during the cycle, this not being typical of IDI diesel vehicles tested by Ricardo in this inertia class.

The mean particulate level recorded with the regular gasoline is approximately 60% lower than what might be expected from a catalyst equipped IDI diesel vehicle of similar inertia. The particulate levels from the other fuels reflect little improvement relative to catalyst equipped IDI diesel vehicles of similar inertia.

8. EXHAUST ODOUR

Odour intensity readings by Turk panel have been obtained for the turbocharged TCCS Cricket. With gasoline as fuel, the TCCS engine is somewhat better than the average IDI diesel engine. On diesel fuel, the TCCS is comparable with the IDI diesel engine.

9. NOISE

Exterior sound level measurements made by SWRI (110) comparing the turbocharged L-141 TCCS Cricket with IDI diesel passenger vehicles are shown in Figure 68. A typical European gasoline vehicle has also

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* Comments extracted from correspondence between Texaco and DoE, November 1980 ,submitted following Texaco's examination of the original draft of this report.

been included. Under acceleration drive-by conditions, the Cricket is quieter than either the diesel or gasoline vehicles. There are a number of factors governing noise under these conditions i.e. engine bore size, engine speed and type of engine structure. An empirical Ricardo relationship between these factors has been derived and correlated with anechoic measurements. Application of this relationship to the TCCS Cricket suggests that the lower noise levels of this vehicle during the acceleration drive-by test are associated with the restricted engine speed range (rated at 3600 rpm).

During the 30mph drive-by, the TCCS Cricket is noisier than the gasoline vehicle and somewhat quieter than the diesel. At idle, the TCCS Cricket had a very similar noise level to the mean of the various IDI diesels evaluated and both were noisier than the gasoline vehicle.

Significantly, noise level with the TCCS Cricket is indistinguishable between diesel and gasoline fuels except at idle, when gasoline operation was quieter. This is obviously a distinct advantage of the TCCS process.

10. COLD STARTING CHARACTERISTICS

With spark ignition, the TCCS combustion system appears to have no difficulty in starting. General Motors evaluated the cold starting characteristics of the turbocharged L-141 TCCS Cricket using a broadcut fuel (50/50 diesel/gasoline blend). Following overnight soaking at $-20^{o}F$, the engine started in 70 seconds total cranking time, idled smoothly, did not stall and demonstrated immediate drive away (110). Successful starting with the naturally aspirated L-141 engine has been demonstrated at $-25^{o}F$ with gasoline, CITE and winter grade diesel fuel (115). For all fuels, a start with sustained idle was achieved in 10 seconds without the use of external aids. Furthermore following such starts, the ability to accelerate immediately without hesitation is also reported.

Emission results with various combinations of vehicles and TCCS engines

TABLE 1

TABLE LINE NUMBER	REF SOURCE	ENGINE - VEHICLE	EMISSION CONTROLS OR OTHER SETTINGS	INERTIA LBS	FUEL	1975 FTP RESULTS G/MILE			ECONOMY MPG (US)	COMMENTS
						HC	CO	NOx		
1	121	T/C L-141 M-151 Jeep	None - Max Economy Settings	2750	Gasoline - results confirmed with No.2 JP-4 and broadcut	3.13	7.00	1.46	24.3	-
2	115	"	None	3000	Gasoline	3.85-4.58	9.08-9.62	1.52-1.74	- -	- -
3	113	N/A L-141 M-151 Jeep	None - Max Economy Settings	2750	Gasoline	4.24	7.28	1.43	-	-
4	121	T/C L-141 M-151 Jeep	5^{o} Retard No EGR 2 Catalysts	2750	Gasoline	0.30	1.07	1.40	20.9	-
5	121	"	8^{o} Retard Medium EGR 2 Cata-alysts	2750	Gasoline	0.33	1.05	0.61	19.7	10% power loss from line 1 build.

TABLE 1 continued

TABLE LINE NUMBER	REF SOURCE	ENGINE-VEHICLE	EMISSION CONTROLS OR OTHER SETTINGS	INERTIA LBS	FUEL	1975 FTP RESULTS G/MILE			ECONOMY MPG (US)	COMMENTS
						HC	CO	NOx		
6	121	"	13^{o} Retard High EGR 2 Catalysts	2750	Gasoline - results confirmed with No.2 JP-4 and broadcut	0.35	1.41	0.35	16.2	28% power loss from line 1 build
7	121	N/A L-141 M-151 Jeep	Retard EGR 3 Catalysts	2750	Gasoline	0.37	0.24	0.31	15.8	Low mileage
8	121	"	"	2750	Gasoline	0.30	0.67	0.34	15.6	50,000 miles
9	113	N/A L-163-S M-151 Jeep	EGR No Catalyst	2750	Diesel	2.2	11.4	1.3	30	SWRI results
10	113	"	EGR Catalyst	2750	Diesel	1.6	2.1	1.5	27.8	SWRI results
11	121	N/A L-141 Cricket	None	2500	Gasoline	2.22	7.11	1.99	-	-

TABLE 1 continued

TABLE LINE NUMBER	REF SOURCE	ENGINE-VEHICLE	EMISSION CONTROLS OR OTHER SETTINGS	INERTIA LBS	FUEL	1975 FTP RESULTS G/MILE			ECONOMY MPG (US)	COMMENTS
						HC	CO	NOx		
12	117	N/A L-141 Cricket	None	2500	Gasoline	1.07	0.84	1.89	25.3	-
13	117	"	Catalysts + Retard + Exhaust Back Pressure	2500	Gasoline	0.61	0.85	0.99	22.5	14% power loss from line 12 build
14	117	"	Catalysts + Retard + Throttl-ing + Exhaust Back Pressure + High Rate EGR	2500	Gasoline	0.36	1.15	0.38	20.0	30% power loss from line 12 build
15	117	T/C L-141 Cricket	Catalysts	2500	Gasoline	1.37	0.50	1.84	28.0	-
16	120	L-163-S Gremlin	Catalysts EGR	2750	Gasoline	1.11	2.5	1.8	27.9	Bartlesville data mean of
17	120	"	New Catalyst EGR	2750	Gasoline	0.5	0.77	2.2		several results

TABLE 2 The influence of fuel specification upon the exhaust emissions of TCCS equipped vehicles

TABLE LINE NUMBER	REF SOURCE	ENGINE - VEHICLE	EMISSION CONTROLS OR OTHER SETTINGS	INERTIA LBS	FUEL	1975 FTP RESULTS G/MILE			ECONOMY MPG (US)	COMMENTS
						HC	CO	NOx		
1	117	T/C L-141 Cricket	Catalyst No change in engine settings for fuels	2500	Gasoline (a)	1.37	0.50	1.84	28.0	
					Broad-cut (50/50 a/b)	0.92	1.08	1.63	29.2	EPA TEST RESULTS
					No. 2 Diesel (b)	1.01	1.88	1.91	30.2	
2	118	T/C L-141 M-151 Jeep	8^{o}Retard EGR 2 Catalysts No change in engine settings for fuels	2750	Gasoline	0.33	1.04	0.61	19.7	-
					JP-4	0.26	1.09	0.50	20.2	
					Broad-cut	0.14	0.72	0.59	21.3	
					No. 2 Diesel	0.27	1.14	0.60	23.0	
3	118	"	8^{o}Retard EGR No change in engine settings for fuels	2750	Gasoline	3.60	6.69	0.84	-	-
					JP-4	2.68	8.23	0.69	-	
					Broad-cut	2.45	8.82	0.82	-	
						2.26	12.21	0.78	-	
4	118	"	8^{o}Retard No change in engine settings for fuels	2750	Gasoline	3.04	5.58	1.29	-	1972 Hot Start Tests
					JP-4	2.35	6.29	1.10	-	
					Broad-cut	3.02	5.54	1.34	-	
					No.2 Diesel	1.75	10.47	1.26	-	

APPENDIX 11

THE M.A.N. FM COMBUSTION SYSTEM

1. HISTORY AND BRIEF DESCRIPTION OF THE SYSTEM DESIGN AND COMBUSTION PROCESS

It has already been observed elsewhere in this report that the M.A.N. M system has been successfully utilised for multi-fuel applications. The M system is successful in this respect owing to the unique M.A.N. design philosophy of spraying the fuel directly onto the walls of the combustion chamber as a thin film. The rate at which fuel evaporates from the wall is controlled by piston temperature and air swirl. This enables rates of pressure rise to be moderated with low cetane, high octane gasoline fuels by only permitting relatively small controlled quantities of fuel to be pre-mixed during the inevitably longer ignition delay period.

To reduce ignition delay with high octane gasolines, MAN raised compression ratios to beyond 20:1 resulting in unacceptable mechanical stresses. To alleviate this problem whilst still retaining multi-fuel capability, a sparking plug was added to the combustion chamber of the M system and the compression ratio suitably reduced. The spark ignited development is designated the FM.

The principal of the FM system is shown in Figure 69. Figure 69 shows the injector and the sparking plug disposed by 180^{o}. In more recent FM developments, the sparking plug is located adjacent to the injector. This has been done to minimise the risk of misfire at light load owing to the large distance between injector and plug. Furthermore, because the fuel spray is angled down into the spherical combustion bowl, extended electrode sparking plugs are necessary with the disposed design together with the attendant durability implications.

In the FM system shown in Figure 69, high intensity air swirl is induced by a helical inlet port and the spherical combustion chamber within the piston crown. During compression, fuel is sprayed via a single hole nozzle onto the wall of the combustion bowl. The high levels of air swirl assist with spreading the fuel over the chamber wall as a thin film thus maintaining excellent charge stratification. Heat supplied to the fuel film by the piston and air charge causes evaporation. The evaporated fuel is then mixed with air and carried to the sparking plug where ignition occurs or into the established flame path following ignition. Long duration sparks are required with this system to allow sufficient time for the ignitable mixture to pass through the plug gap.

In the case of the more recent developments with the adjacent sparking plug, the principle is essentially the same except that the "tail" of the fuel film is thought to be ignited initially.

As per the diesel engine, the maximum torque output of the FM system is smoke limited.

To date, the majority of FM applications have been to heavy duty size engines of approximately 4-5" bore and 61-98 CID per cylinder with rated speeds of 2200-3200 rpm. Some applications to light duty displacements of approximately 38-40 CID per cylinder have however been made. Ricardo have gained recent experience with high speed FM engines of cylinder sizes within the range 25-30 CID.

Multi-fuel FM engines operate with compression ratios of typically 15.5-17:1, the higher level being of value in enabling the system to operate as a conventional diesel M system without spark should diesel fuel be available. Gasoline (80-100 octane), JP-4, diesel fuel and diesel gasoline mixtures have all been successfully used with such compression ratios.

For optimised operation with 91 octane gasoline, Ricardo experience with small high speed FM engines revealed that compression ratios of 13-14 were required to avoid full load detonation. This value closely approaches the optimum balance between indicated thermal efficiency and mechanical friction.

2. COMBUSTION CHARACTERISTICS

Available cylinder pressure diagrams for the FM combustion system show that smooth stable combustion is achieved with most fuels. The exceptions are with the lower octane gasolines, c.80, when knocking can occur with the higher compression ratios. Low octane fuels such as Jet and diesel fuel do not knock however. This is probably due to differences in the manner of mixture formation and combustion with the lighter, more volatile gasolines being more capable of producing pre-mixed zones prior to ignition in which knock will occur. With gasolines of over c.90 octane, thermally efficient high compression ratios can be utilised without knock.

Typical naturally aspirated peak cylinder pressures of the FM system for a variety of fuels ranging from diesel to high octane gasoline range between approximately 800 and 1000 psi. This range is somewhat higher than typical gasoline engines at 800-900 psi but does not reach the IDI diesel with peak cylinder pressures of around 1100 psi.

3. PERFORMANCE

Smoke limited bmep curves for two high speed, naturally aspirated, light duty FM engines are shown in Figure 70, and compared with typical developed light duty IDI diesel data.

The multi-fuel L9204FMV engine curve represents the diesel smoke limited value. The torque curve is low at the lower speeds compared with the IDI diesel but matches the diesel at higher speed. By altering the fuel injection pump rack stop, this same torque curve is achieved with JP-4 and 100 octane gasoline. With these fuels, the smoke output at this rating is significantly lower in comparison with diesel fuel, especially with gasoline. The potential for raising the rating by increasing fuelling levels with the lighter fuels to match the original diesel smoke limit is therefore available.

This is demonstrated by the Ricardo data shown in Figure 70. This curve shows that optimisation with 91 octane gasoline achieved a favourable comparison with the IDI diesel at low speed and excelled the diesel performance at higher speeds. These comparisons are made with equivalent smoke limits.

Other data are available to show that the FM engine smokes less with gasoline fuels than diesel at high load factor (122). Maximum smoke limited ratings with the FM engine will therefore be favoured by high octane, volatile fuels.

In the context of smoke, M system diesels tend to emit blue/white smoke under light load conditions when fuel evaporation from the chamber wall is not very efficient due to low piston temperatures.

This characteristic also occurs with FM engines running on diesel fuel but has not been observed by Ricardo with gasoline fuelled FM engines. MAN are exploring combustion chamber insulation to remove this problem with diesel fuel.

The various data applicable to larger, heavy duty, FM engines reveal that acceptable performance standards are retained with increased cylinder sizes on either diesel or gasoline fuel (123). Maximum bmep for naturally aspirated engines ranges between 96 and 122 psi. These results are for gasoline fuel (93-100 octane) and are competitive with the DI diesel engine predominantly utilised for heavy duty applications. Turbocharged, heavy duty FM engines return peak bmep within the range 135-180 psi dependent upon boost levels with 93 octane gasoline. Smoke levels at these conditions are very low.

4. FUEL ECONOMY

4.1 Steady State Fuel Economy

Recorded test bed fuel consumption data for high speed, light duty FM engines are shown in Figure 71, in comparison with typical IDI diesel and gasoline envelopes. In comparison with the light duty Comet diesel engine, the FM is seen to compare very well with all data lying below, or in the lower half of the diesel envelope. In the case of the multi-fuel L9204 engine, there is generally little difference between the fuel consumption for diesel and gasoline fuels, except at 2000 rpm when diesel fuel imparts some improvement. It can also be observed that the gasoline optimised FM provides better fuel economy than the gasoline fuelled multi-fuel engine at 2000 rpm but that the levels are similar at 3000 rpm. The prototype status of the optimised engine should however be observed.

Heavy duty FM applications (123) also retain similar, if not superior, low fuel consumption characteristics. High speed FM engines appear to have 5-10% worse fuel consumption at the lower speeds than the narrower speed range heavy duty engines. This implies that difficulty is experienced with maintaining optimum fuel consumption over a wide speed range.

In the case of turbocharged engines where rating has been increased by approximately 25%, light load fuel consumption is similar to naturally aspirated engines. Minimum consumption is also similar but continues further up the load range. Higher rates of turbocharge appear to penalise light load fuel economy although minimum consumption is similar to the other engines.

Minimum fuel consumption for these FM heavy duty engines is generally comparable with contemporary DI diesel engines.

The low fuel consumption of the FM engine in relation to the light duty IDI diesel engine can be attributed to the combination of good cycle efficiency resulting from moderate compression ratio and lower friction. Friction data for FM engines (122) indicate that levels are intermediate to typical IDI diesel and gasoline engines and are achieved by the combination of open chamber design and lower compression ratio.

4.2 Transient Fuel Economy

Transient fuel consumption data for FM installations are limited. Mention of road testing is made for a light duty truck fitted with the multi-fuel L9204 engine in comparison with a conventional gasoline engine (124). In these tests, the FM equipped vehicle returned 30% lower fuel consumption. These results are entirely coherent judged by the test bed data.

Tentative 1975 FTP economy results have been acquired by Ricardo with a small, high speed FM engine running on gasoline (125). These results indicate that the fuel efficiency of the diesel engine is approached but not excelled, as judged by comparing with gasoline equivalent volumetric consumptions for 1978/79 model year certification IDI diesel cars (119).

The preliminary nature of these results should be borne in mind however, since no development for transient operation was carried out.

5. GASEOUS EXHAUST EMISSIONS

5.1 Steady State Gaseous Exhaust Emissions

Emission data for the FM system are limited. Some typical data for small high speed FM engines running on gasoline are shown in Figure 72, in comparison with developed Comet diesel engines.

From Figure 72 it can be noted that HC emissions are worse by a factor of 5-10 in comparison with the diesel engine. HC emissions are particularly high at light load. These HC emissions are thought to manifest primarily from the fuel which is swept away from the fringe of the spray creating lean pockets which are incapable of supporting combustion. In addition, the FM engine operates with a single hole, needle valve, DI type nozzle incorporating a sac volume. This uncontrolled volume in combination with volatile gasoline will aggravate HC emissions. NOx emissions are higher than the IDI diesel at the higher load settings but comparable at low load factors. CO emissions are somewhat higher at light load but under some conditions are lower towards full load compared with the diesel engine.

Limited emissions data published by MAN (126) are basically in accord with the aforementioned results, particularly with respect to the higher HC levels. Other MAN data (122) indicate that NOx emissions may be increased with FM multi-fuel engines running on diesel fuel in comparison with gasoline towards the higher load factors.

5.2 Transient Gaseous Exhaust Emissions

Vehicle emission studies with FM engines appear to be very limited and restricted to some preliminary 1975 FTP results obtained by Ricardo (125) utilising a small, high speed FM engine fitted to a passenger car with gasoline as fuel. The results obtained without emission controls in comparison with IDI diesel cars were in agreement with the test bed observations. HC emissions were an order of magnitude greater whilst NOx and CO emissions were only marginally higher than typical diesel levels. HC emissions were also approximately 2-3 times higher than typical untreated gasoline vehicles.

6. RESPONSE TO GASEOUS EMISSIONS CONTROLS

The reduction of HC emissions by intake throttling has been investigated (123). With 50% throttling, 25-30% reduction of HC emissions can be achieved whilst fuel consumption penalties range between 10% and 15%. Such throttling also reduces NOx emissions by 30-50% whilst CO emissions inevitably increase by approximately 10-15%.

7. EXHAUST ODOUR

Limited available odour data for FM engines indicate that light load and idling odour appears to be very noticeable and irritant. This problem is amplified by the high HC emissions and an improvement in this area may achieve some odour reduction.

8. NOISE

Subjective Ricardo noise evaluations of FM equipped passenger cars indicate that noise levels at idle are similar to an IDI diesel vehicle, although the familiar diesel knock is not prevalent. The smooth pressure diagrams of the FM engine and the lower rates of pressure rise (30-45 psi/^{o} crank FM - 80-100 psi/^{o} crank IDI diesel) would suggest that combustion noise should be lower for the FM system. Potential combustion noise benefits with the FM are, however, thought to be over-shadowed in the subjective Ricardo assessments due to unthrottled intake noise and mechanical noise emanating from the fuel injection equipment.

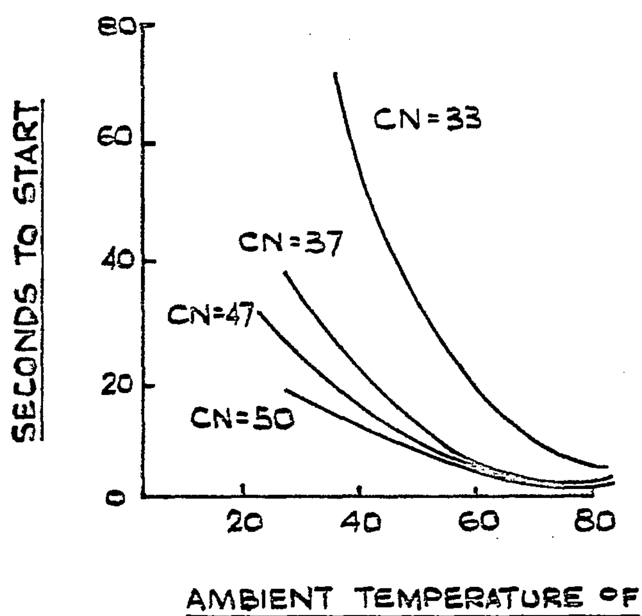
9. COLD STARTING CHARACTERISTICS

For the MAN L9204 FM multi-fuel engine, an intake manifold flame heater is provided for cold starting with diesel fuel. With gasoline fuel, this cold start aid is not required. These statements indicate, as might be anticipated, an influence of fuel volatility upon cold starting due to the necessity to evaporate the fuel film from the relatively cold chamber surfaces. No data regarding the cold start performance or limitations are available.

Ricardo FM experience of small engines fuelled with gasoline has shown instant starting characteristics at 32^{o}F. Starts have not been made at lower temperatures. Following cold start, these engines emitted white smoke of a density and duration typical of an IDI diesel engine.

FIG. No.1 [REF. 3]
Drg. No. S 8063
Date SEPT. '80

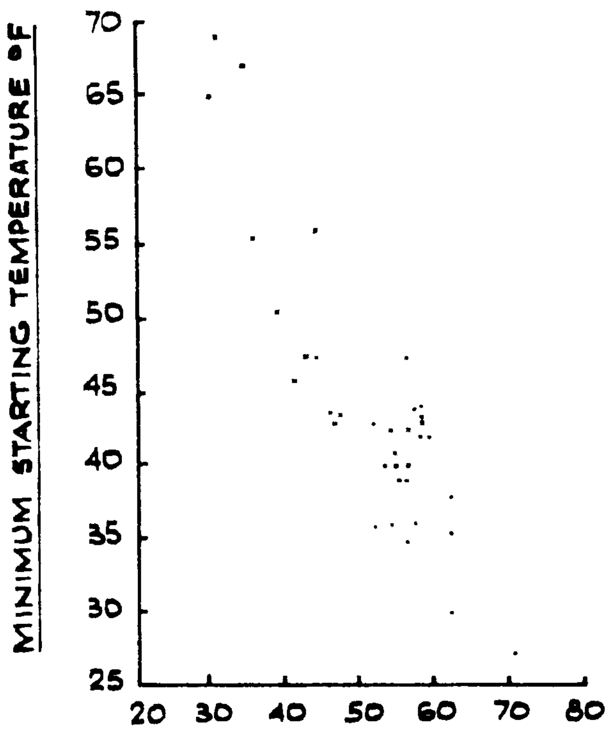
EFFECT OF FUEL CETANE NUMBER ON TIME REQUIRED FOR STARTING AT VARIOUS AMBIENT TEMPERATURES.



ENGINE MULTI-CYLINDER DIRECT INJECTION
[OTHER DETAILS NOT STATED]

FIG. No. 2 [REF 4]
Drg. No. S 8064
Date SEPT. '80

MINIMUM STARTING TEMPERATURE VERSUS CETANE NUMBER.



ENGINE:- SINGLE CYLINDER 4.25" BORE X 6" STROKE
DIRECT INJECTION
COMPRESSION RATIO 13:1
CRANKING SPEED 160 R.P.M.

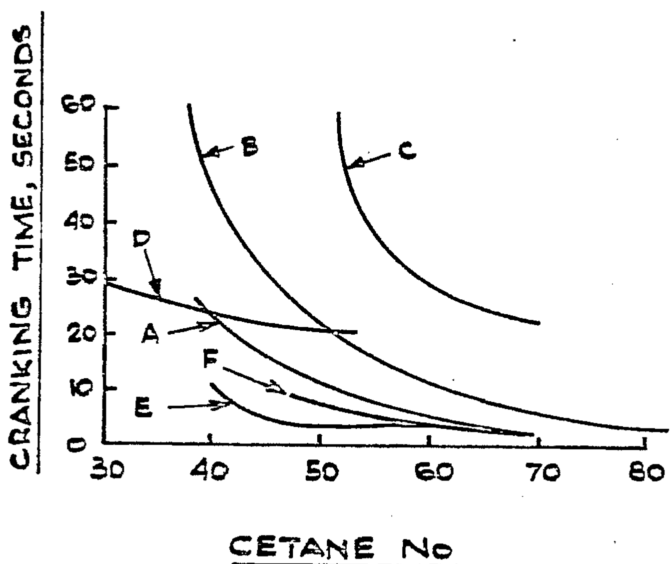
FUELS:-

DISTILLATION	10%	50%	90%
RANGE °F	381-527	424-640	489-734
MEAN °F	459	538	630

CETANE No. RANGE 30 - 70·5

FIG. No. 3 [REF. 4]
Drg. No. S 8065
Date SEPT.'80

COMPARATIVE EFFECT OF CETANE NUMBER ON THE STARTING PERFORMANCE AT 20°F AMBIENT TEMPERATURE OF SIX MULTI-CYLINDER ENGINES.



ENGINE A - DIRECT INJECTION
ENGINE B - DIRECT INJECTION
ENGINE C - INDIRECT INJECTION
ENGINE D - INDIRECT INJECTION
ENGINE E - INDIRECT INJECTION
ENGINE F - DIRECT INJECTION

ENGINES: A - F, RANGE OF LEADING VARIABLES

BORE: 3.5 - 5.0 in
STROKE: 4.0 - 6·7 in
CYLINDER CAPACITY 0·67 - 2·10 LITRES
COMPRESSION RATIO 15 - 19:1

FUELS:

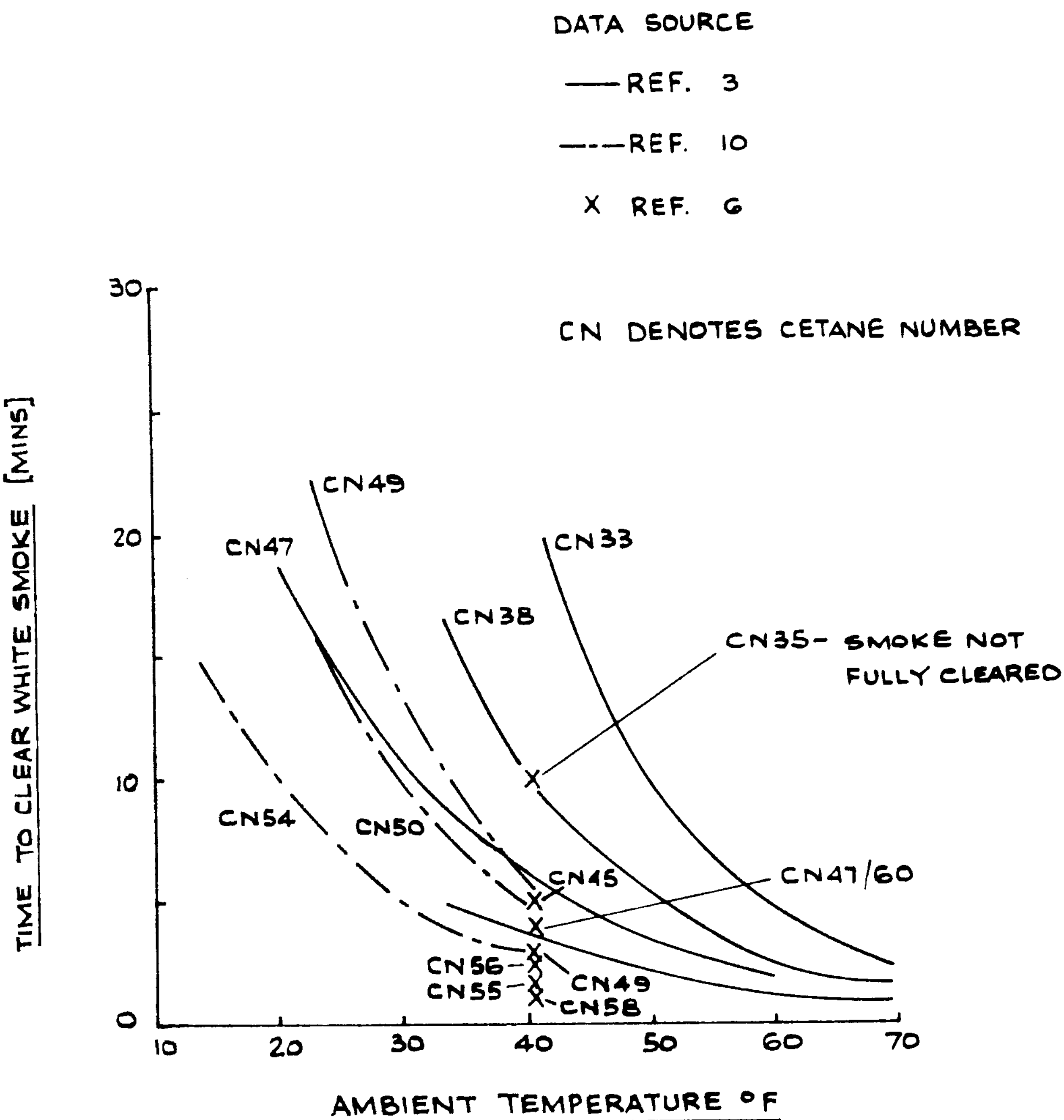
DISTILLATION	10%	50%	90%
RANGE °F	329-478	408-556	491 - 682
MEAN °F	430	511	621

CETANE NUMBER RANGE 29.5 - 82

3309KT

FIG. No. 4
Drg. No. S 8066
Date SEPT '80

INFLUENCE OF CETANE NUMBER AT VARIOUS AMBIENT TEMPERATURES ON THE TIME TO CLEAR WHITE SMOKE AT IDLE FOLLOWING COLD STARTING - HEAVY DUTY DIRECT INJECTION ENGINES.



THE INFLUENCE OF CETANE NUMBER UPON EXHAUST SMOKE – DIRECT INJECTION ENGINES.

FIG. No. 5 [REF. 5,6,11]
Drg. No. S 8067
Date Sept '80

FUELS:

X——X NATURALLY VARYING CETANE No. MID-POINT 372–383°F

△ FUEL A – NATURAL CETANE No. } ASTM MID-POINT 392°F
⊡ FUEL A + IGNITION IMPROVER

▽ FUEL B – NATURAL CETANE No. — ASTM MID-POINT 396°F

—·— AROMATIC/PARAFFINIC FUELS ——— MID-POINT 491°F

– – – AROMATIC/PARAFFINIC FUELS ——— MID-POINT 383°F

—···— PARAFFINIC FUELS ——— MID-POINTS 383 AND 491°F

–·– GAS OIL – NATURAL CETANE No. } ASTM MID-POINT 565°F
¦ GAS OIL + IGNITION IMPROVER

◇ TRACTOR VAPORIZING OIL – NATURAL CETANE No. } ASTM MID-POINT 356°F
⊙ TRACTOR VAPORIZING OIL + IGNITION IMPROVER

ENGINE TYPE/OPERATING CONDITIONS AS STATED

▽ △ ⊡ X——X 85 CID SINGLE CYL. DIRECT INJECTION – 95 PSI BMEP

– – – / —·— / —···— } DIRECT INJECTION – CONSTANT LOAD

◇ ⊙ –¦– MULTI-CYLINDER DIRECT INJECTION 1250 RPM 100 PSI BMEP

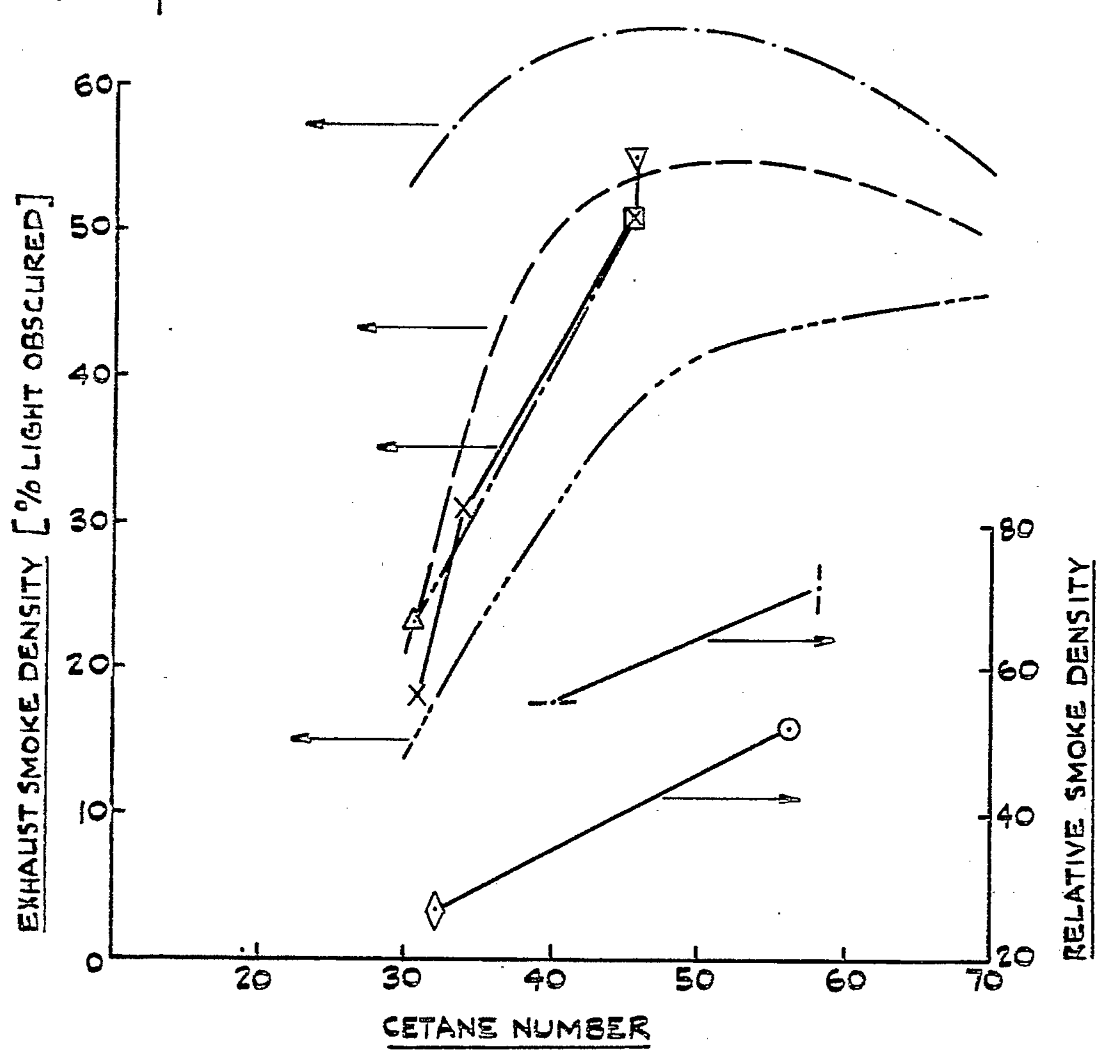
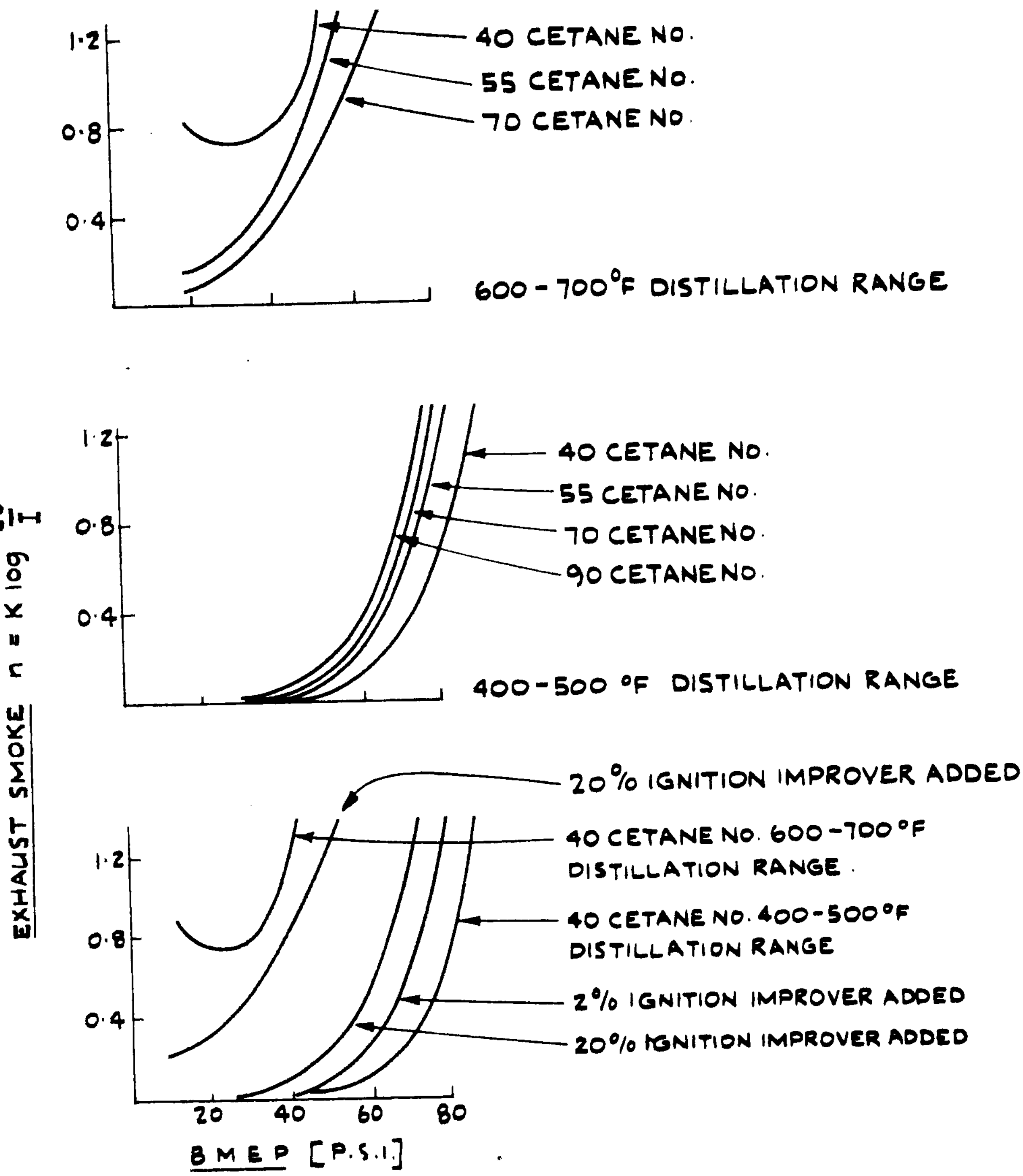


FIG. No. 6 [REF. 13]
Drg. No. S 8068
Date SEPT. '80

THE INFLUENCE OF CETANE NUMBER UPON EXHAUST SMOKE AT 1200 R.P.M.

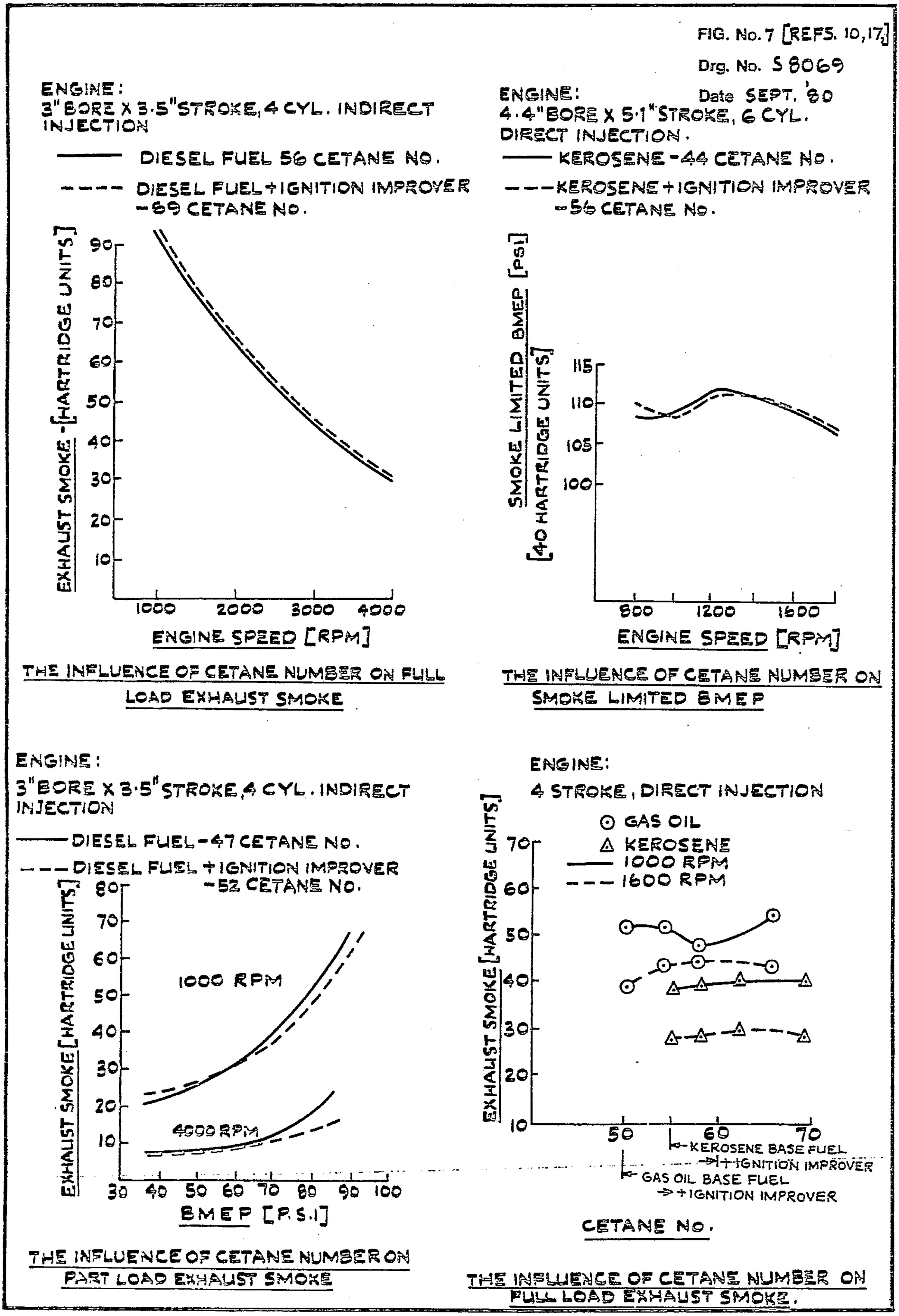


ENGINE:

3·75" BORE X 5" STROKE, SINGLE CYLINDER COMPRESSION RATIO 12:1

DIRECT INJECTION

6983LT/5431



THE INFLUENCE OF CETANE NUMBER ON FULL LOAD EXHAUST SMOKE

THE INFLUENCE OF CETANE NUMBER ON SMOKE LIMITED BMEP

THE INFLUENCE OF CETANE NUMBER ON PART LOAD EXHAUST SMOKE

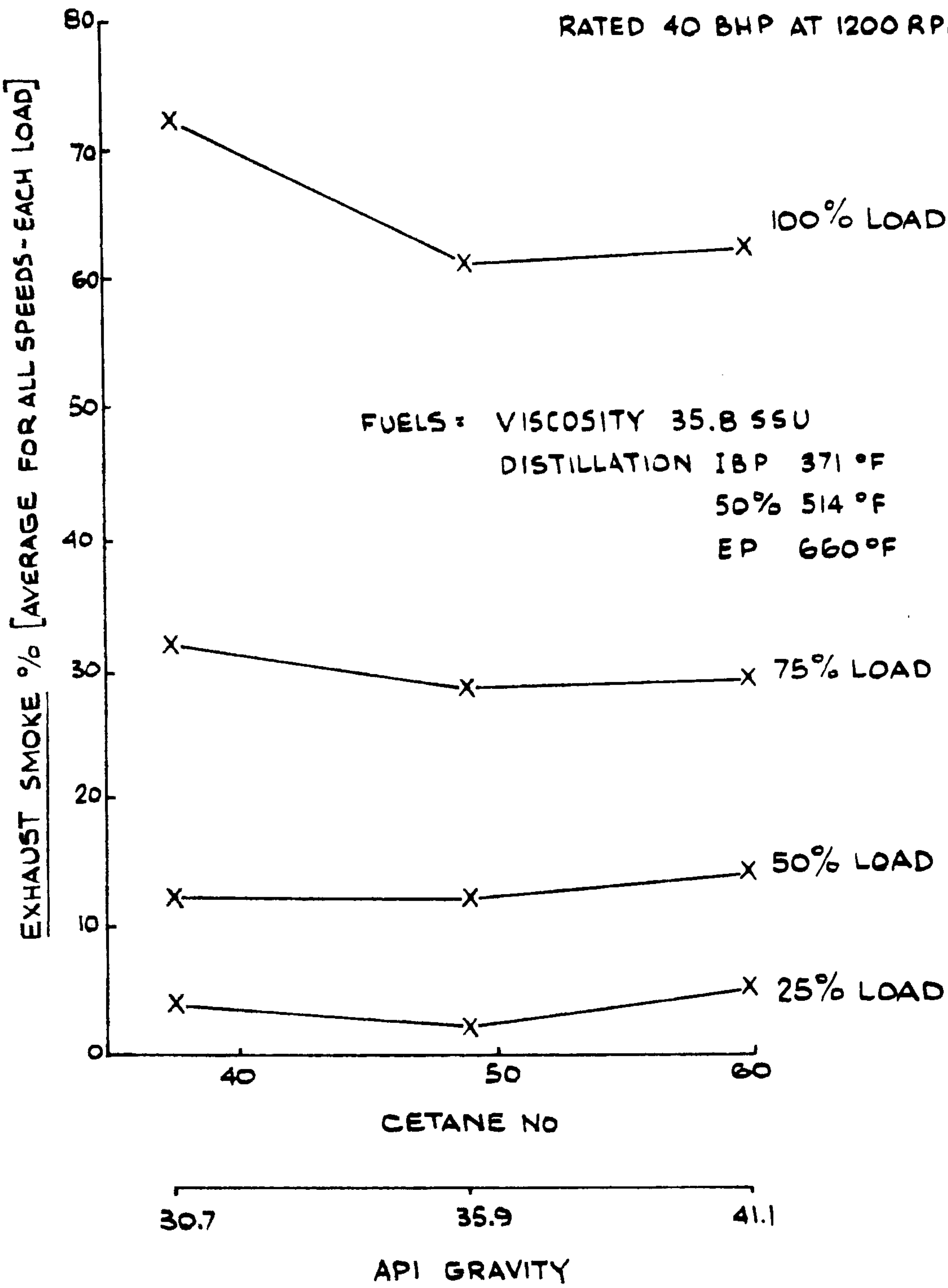
THE INFLUENCE OF CETANE NUMBER ON FULL LOAD EXHAUST SMOKE.

3309KT

FIG. No. 8 [REF. 8]
Drg. No. S 8070
Date SEPT. '80

THE INFLUENCE OF CETANE NUMBER ON EXHAUST SMOKE OVER THE LOAD RANGE FOR FUELS OF EQUAL VISCOSITY & VOLATILITY

ENGINE: FAIRBANKS MORSE 36A 4¼ - 4.25" BORE x 6" STROKE x 4 CYL. - 4 STROKE INDIRECT INJECTION - CR 15.2 : 1
RATED 40 BHP AT 1200 RPM.



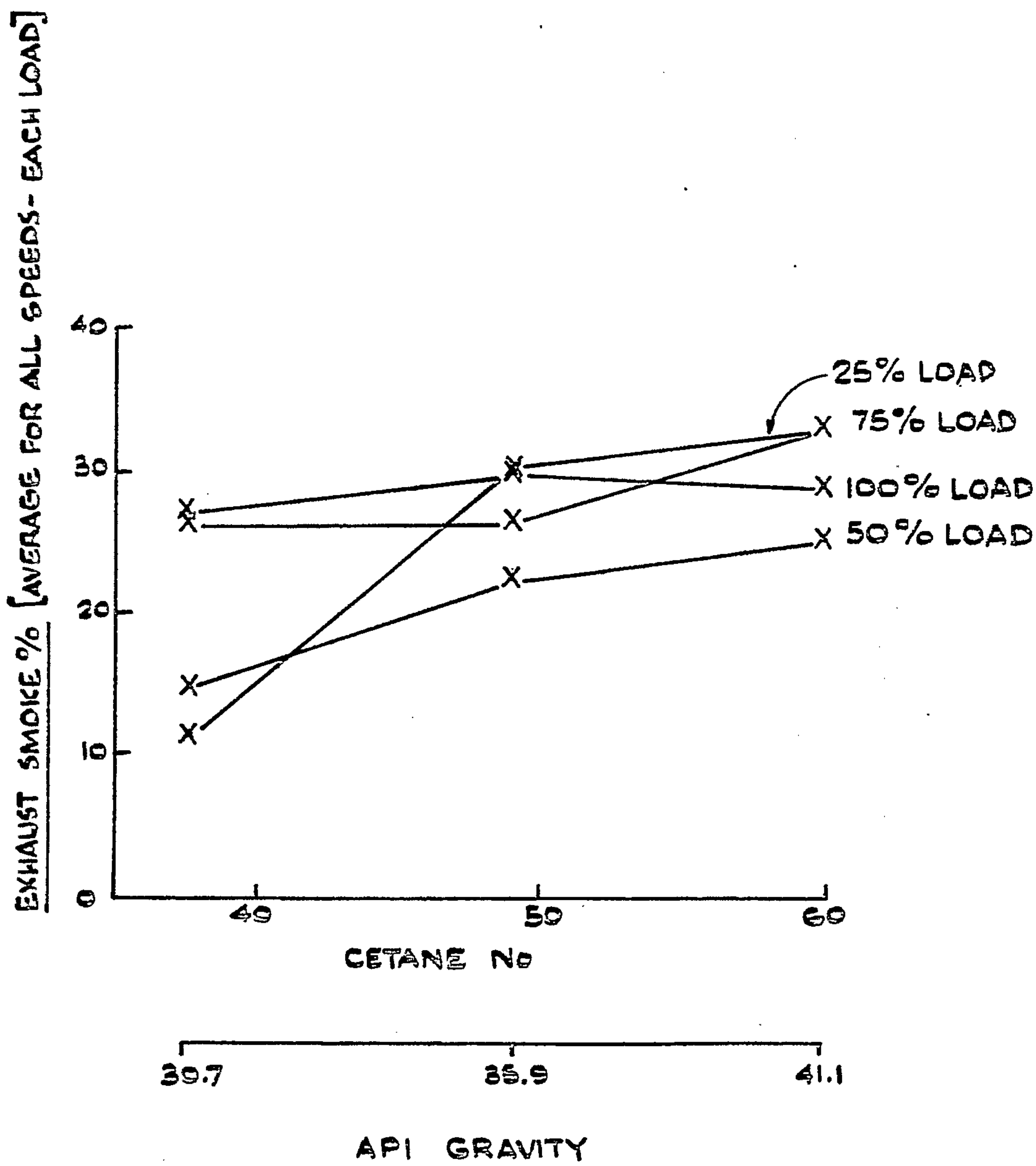
3309KT

FIG. No. 9 [REF. 8]
Drg. No. S 8071
Date SEPT. '80

THE INFLUENCE OF CETANE NUMBER ON EXHAUST SMOKE OVER THE LOAD RANGE FOR FUELS OF EQUAL VISCOSITY AND VOLATILITY.

ENGINE: GM 371 - 4.25" BORE × 5" STROKE × 3 CYL. - 2 STROKE - DIRECT INJECTION - CR 19:1
RATED 83 BHP AT 2000 RPM

FUELS: VISCOSITY 35.8 SSU
DISTILLATION IPB 371°F
50% 514°F
EP 660°F



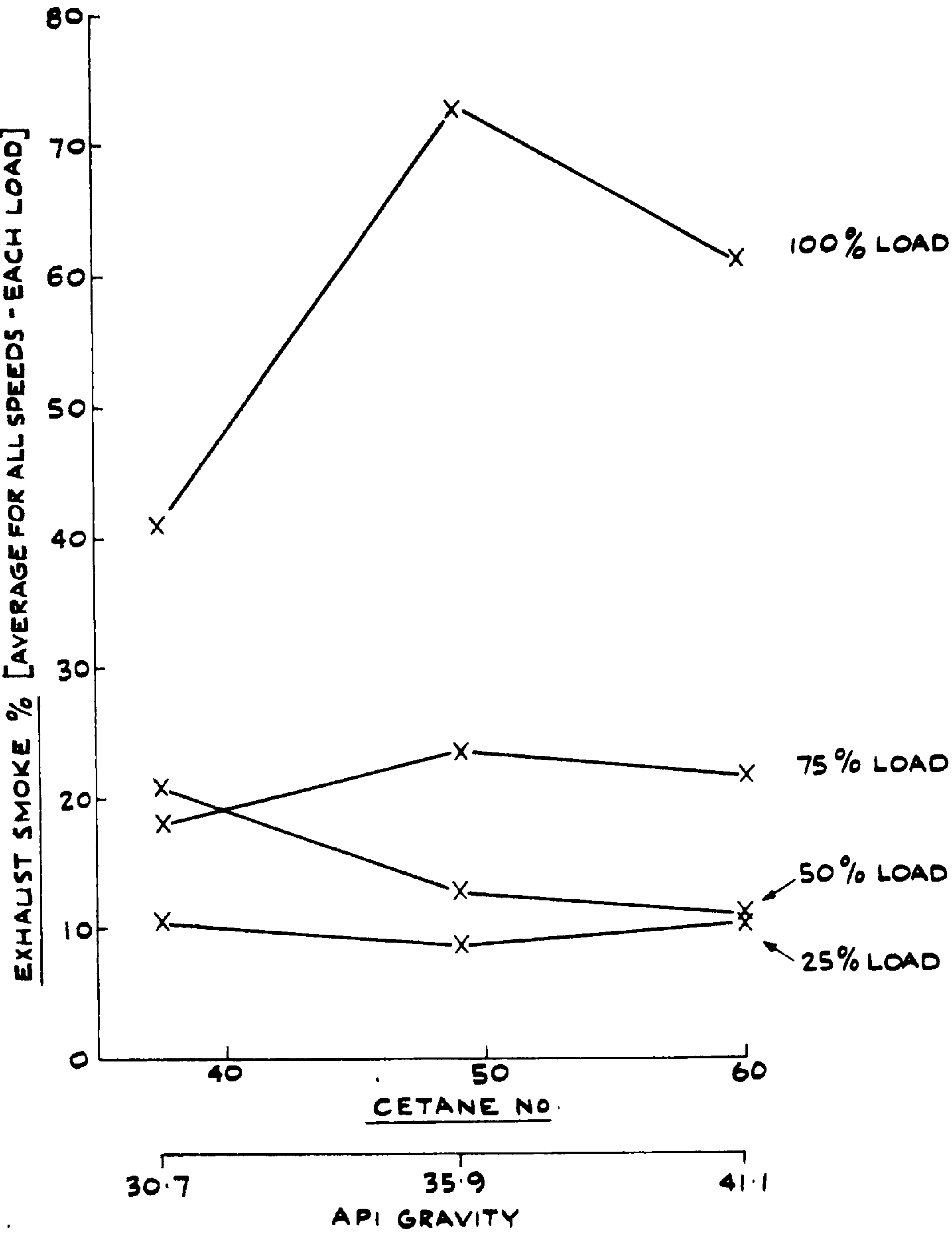
3309KT

FIG. No. 10 [REF. 8]

Drg. No. S 8072

Date SEPT. '80

THE INFLUENCE OF CETANE NUMBER ON EXHAUST SMOKE OVER THE LOAD RANGE FOR FUELS OF EQUAL VISCOSITY AND VOLATILITY.



ENGINE:

MACK END 457 4·25" BORE x 5·38" STROKE x 6 CYL.
4 STROKE LANOVA CELL
CR 14·6:1
RATED 122 BHP AT 2200 RPM.

FUELS:

VISCOSITY 35·8 S.S.U.

DISTILLATION IBP 371°F
50% 514°F
EP 660°F

FIG. No. 11 [REF. 8]
Drg. No. S 8073
Date SEPT. '80

THE INFLUENCE OF CETANE NUMBER ON EXHAUST SMOKE OVER THE LOAD RANGE FOR FUELS OF EQUAL VISCOSITY AND VOLATILITY

ENGINE: HERCULES DRXB - 4·38" BORE x 5·25" STROKE x 6 CYL. 4 STROKE
INDIRECT INJECTION · CR 14·5:1 RATED 104 BHP AT 2000 RPM

FUELS: VISCOSITY 35·8 SSU
DISTILLATION: IBP 371°F
50% 514°F
EP 660°F

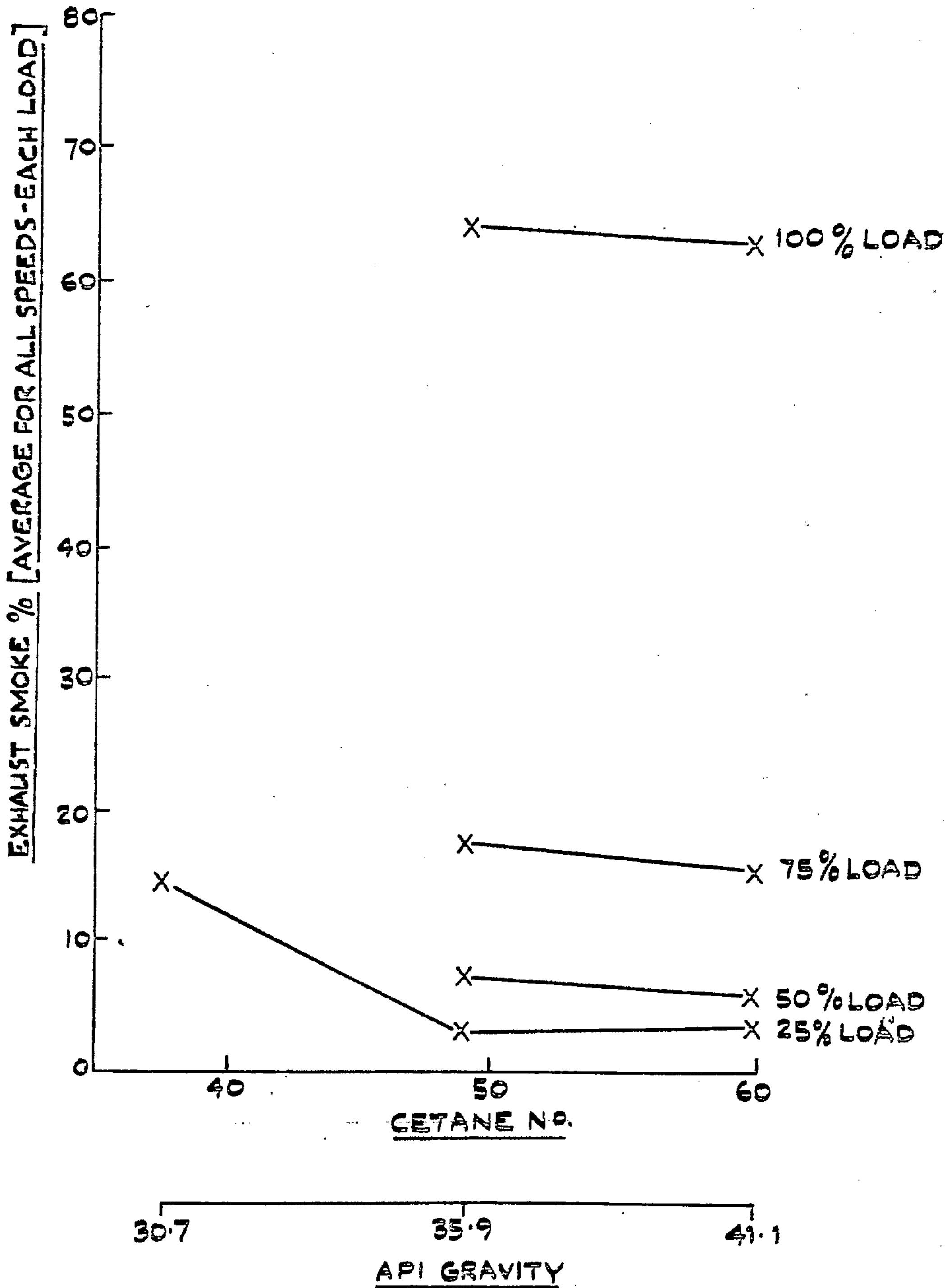


FIG. No. 12a,b,c, [REF. 21]
Drg. No. S 8074
Date SEPT. '80

FIG. 12a.

THE INFLUENCE OF LOAD ON EXHAUST SMOKE AT 1400 RPM

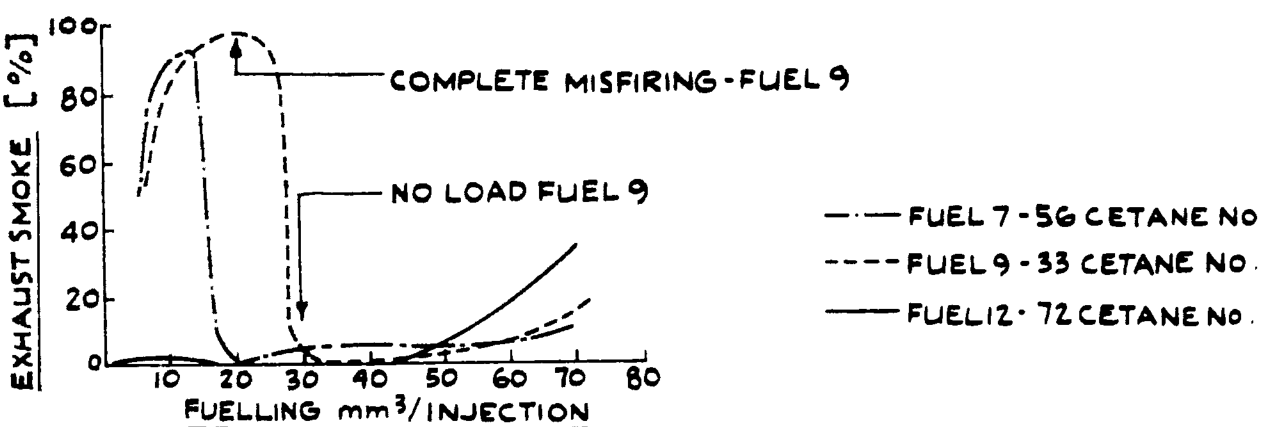


FIG. 12b.

THE INFLUENCE OF CETANE NUMBER ON LIGHT LOAD WHITE SMOKE AND MISFIRING

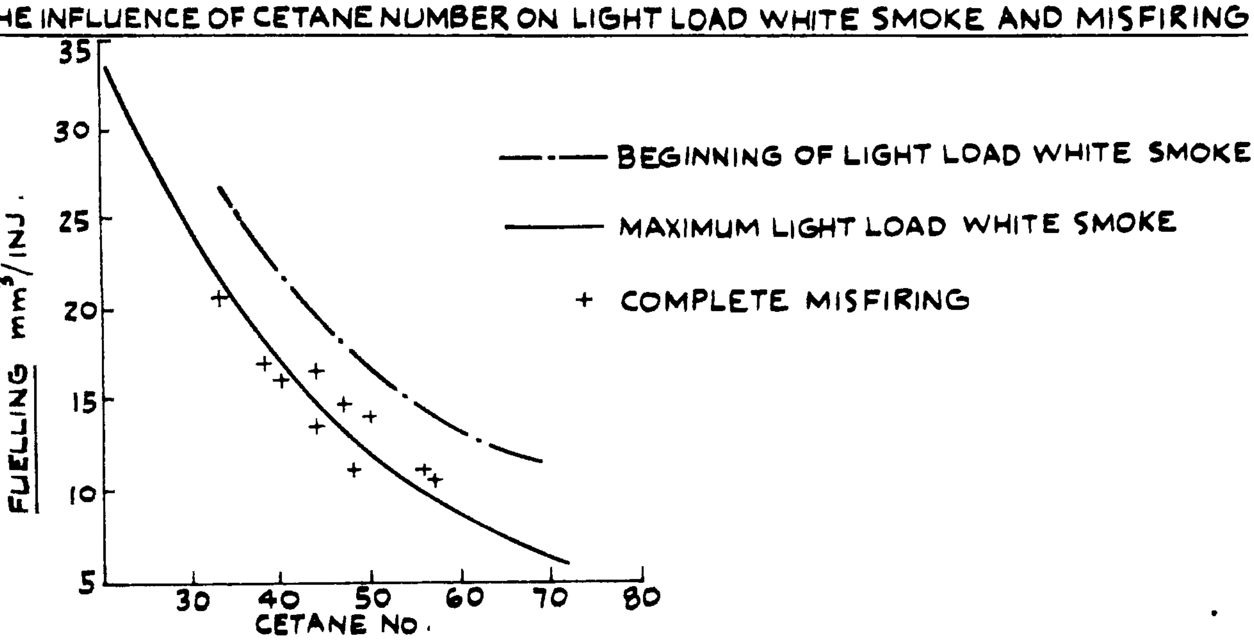
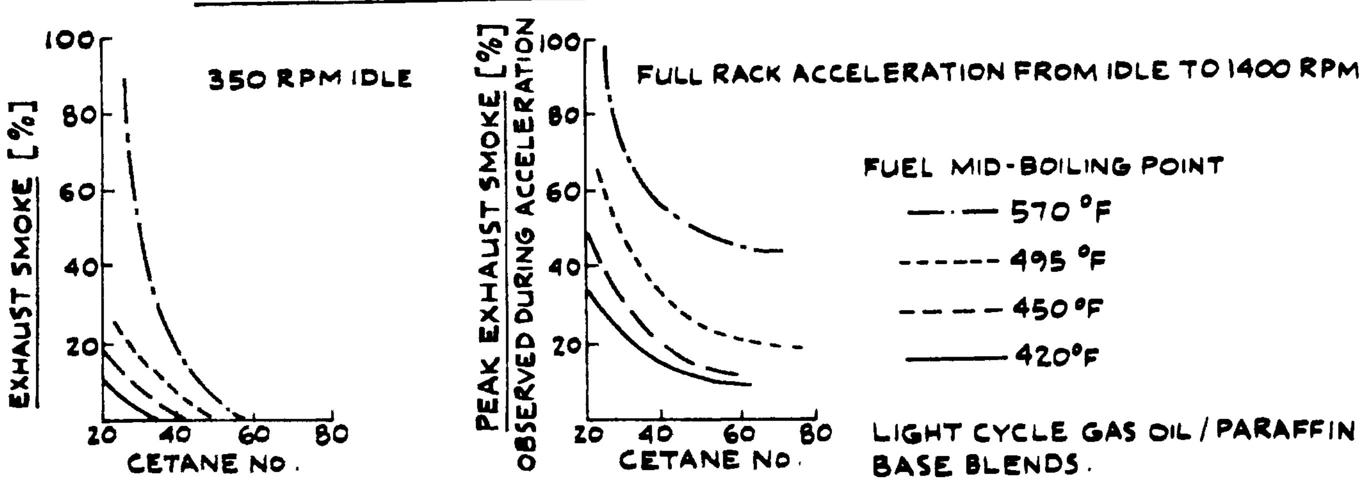


FIG. 12c.

THE INFLUENCE OF CETANE NUMBER ON SMOKE OUTPUT AT IDLE AND DURING FULL RACK ACCELERATION FROM IDLE TO 1400 RPM



THE INFLUENCE OF CETANE NUMBER ON IGNITION DELAY ACCORDING TO SEVERAL SOURCES.

FIG. No. 13

Drg. No. S 8075

Date SEPT. '80

KEY	DETAILS OF TEST CONDITIONS AND ENGINES.
---- / ——	RICARDO DATA [ref. 26] - VARIOUS FUEL DISTILLATION CHARACTERISTICS - FULL LOAD OPERATION - VARIOUS SPEEDS - 92 CID - 4 CYL. - 4 STROKE - SWIRL CHAMBER INDIRECT INJECTION.
⊙——⊙	SHELL DATA [ref. 26] - FUELS OF SIMILAR DISTILLATION CHARACTERISTICS - 80 psi BMEP LOAD AT 3000 RPM - INDIRECT INJECTION.
□——□ / △——△	FIAT DATA [ref. 27] - VARIOUS FUEL DISTILLATION CHARACTERISTICS & CHEMICAL COMPOSITIONS - 2/3 FULL LOAD OPERATION AT 2500 RPM, 116 CID, 4 CYL. 4 STROKE SWIRL CHAMBER INDIRECT INJECTION
CR 12·0 / CR 15·0	LANDEN [ref. 13] 55 CID SINGLE CYLINDER SWIRLING, DIRECT INJECTION 95 p.s.i. B.M.E.P. LOAD AT 1800 RPM
X----X 1000 rpm min. load / ▽——▽ 2200 rpm full load	OLSON ET AL DATA [ref. 28] VARIOUS FUEL DISTILLATION CHARACTERISTICS AND CHEMICAL COMPOSITIONS. THREE ENGINE AVERAGE - INDIRECT AND DIRECT INJECTION INCLUDING 'M' SYSTEM.
•——• 600 rpm / □——■ 900 rpm	TSAO ET AL DATA [ref. 29] 37 CID CFR ENGINE - CONSTANT LOAD FUELS AS MIXTURES OF CETANE AND α METHYL NAPTHALENE.

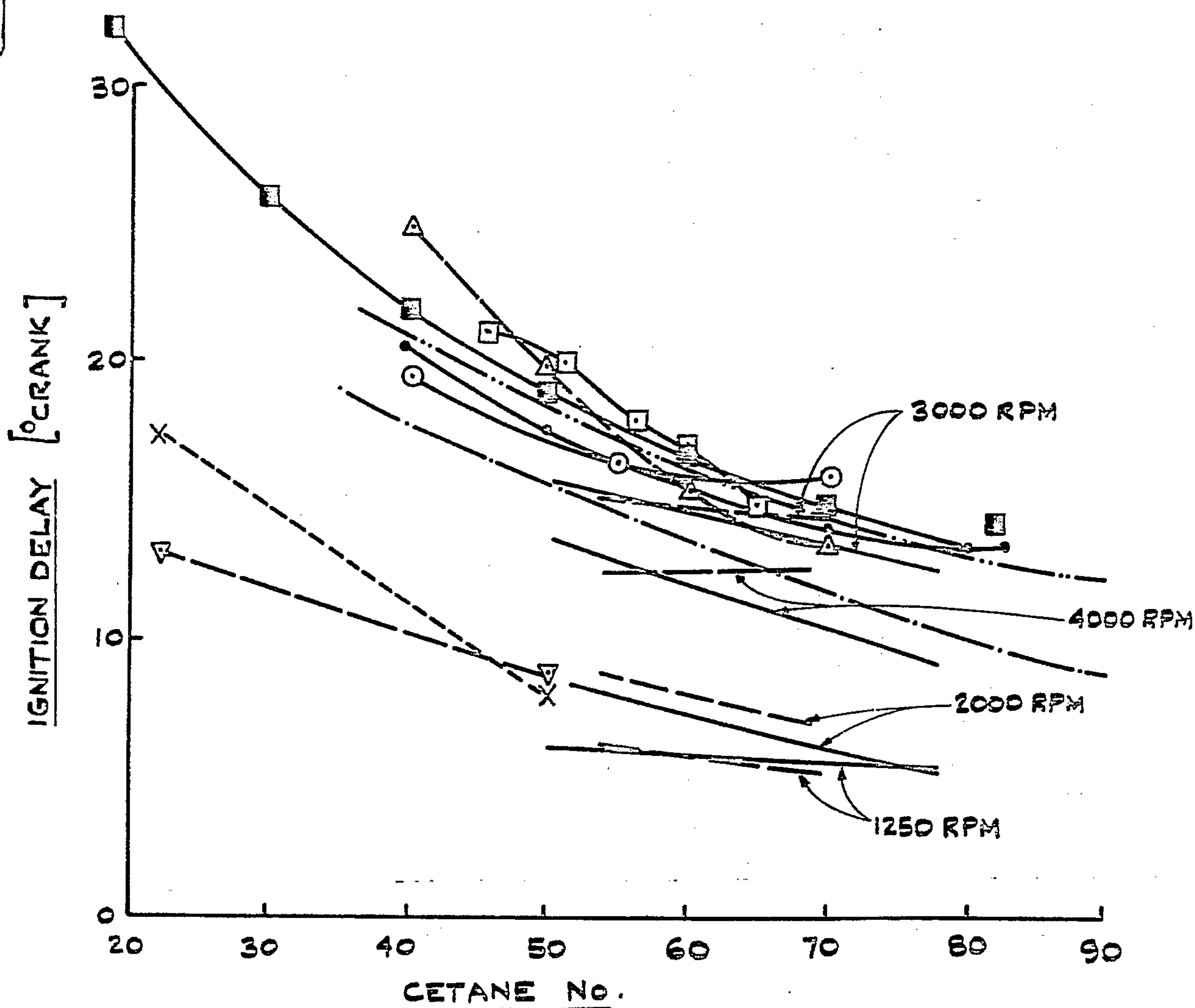
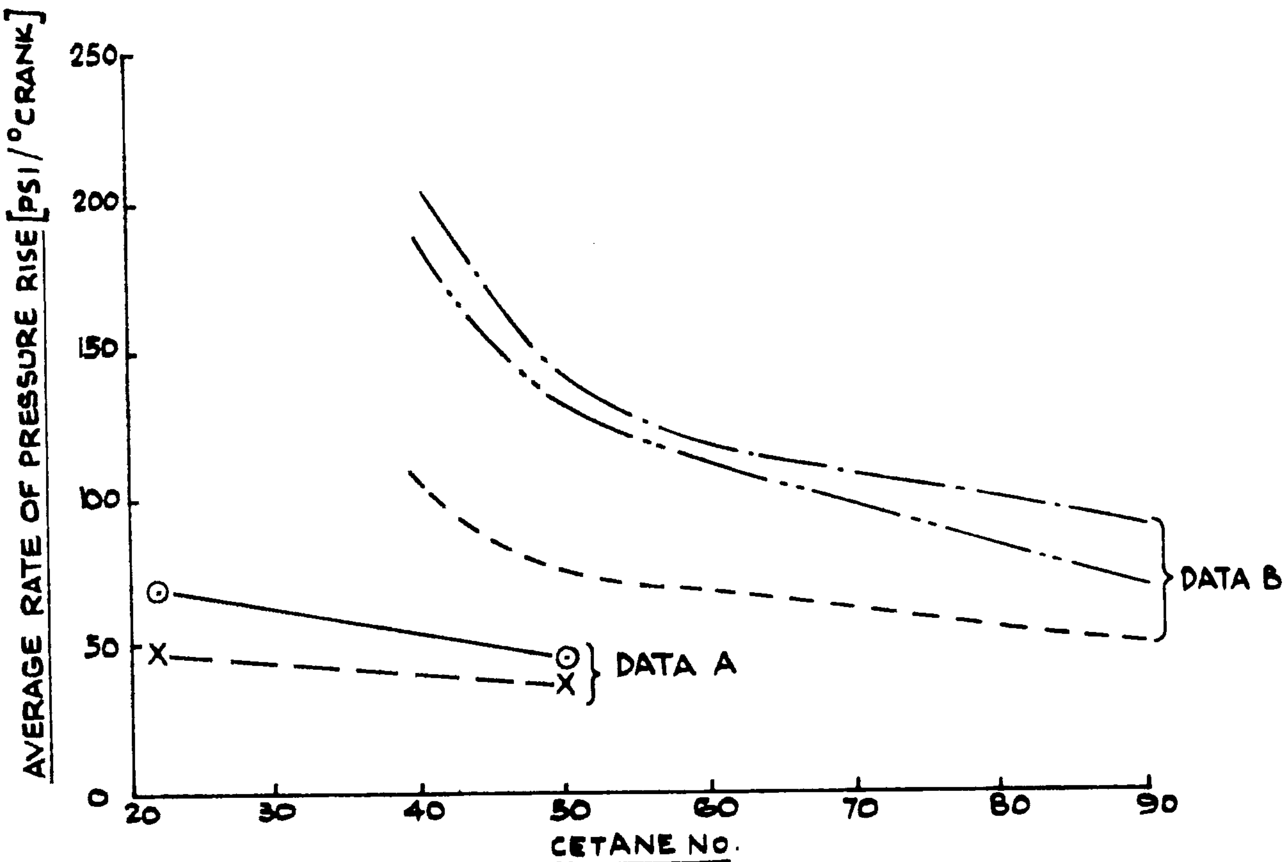


FIG. No. 14

Drg. No. S 8076

Date SEPT. '80

THE INFLUENCE OF CETANE NUMBER UPON RATE OF PRESSURE RISE.



KEY

DETAILS OF TEST FUELS, CONDITIONS AND ENGINES

DATA A

X——X 1000 RPM MINIMUM LOAD

⊙——⊙ 2200 RPM FULL LOAD

VARIOUS FUEL DISTILLATION CHARACTERISTICS AND CHEMICAL COMPOSITIONS. THREE ENGINE AVERAGE - INDIRECT AND DIRECT INJECTION INCLUDING 'M' SYSTEM DATA ACCORDING TO OLSON et al [REF. 28]

DATA B

—·— 15 CR, —··— 12 CR: INJECTION TIMING 29° BTDC

- - - - - 15 CR: INJECTION TIMING 19° BTDC

55 CID SINGLE CYLINDER SWIRLING, DIRECT INJECTION - 96 PSI BMEP LOAD AT 1800 RPM - DATA ACCORDING TO LANDEN [REF 13]

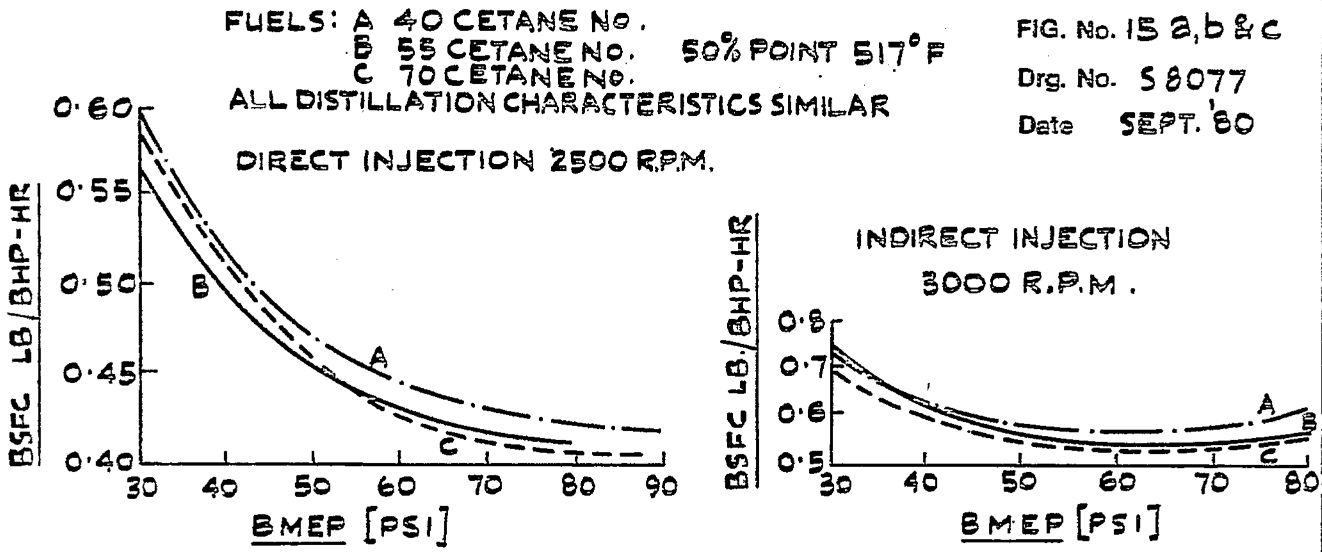


FIG.15 a [REF. 26] THE INFLUENCE OF CETANE NUMBER UPON BSFC.

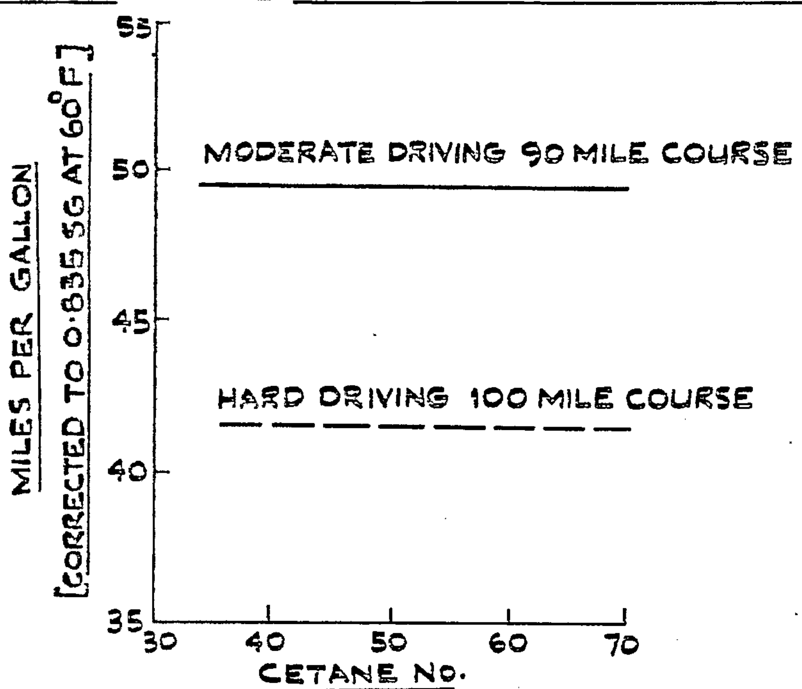


FIG.15 b. [REF. 26] THE INFLUENCE OF CETANE NUMBER ON FUEL CONSUMPTION OF A HIGH SPEED DIESEL ENGINED CAR.

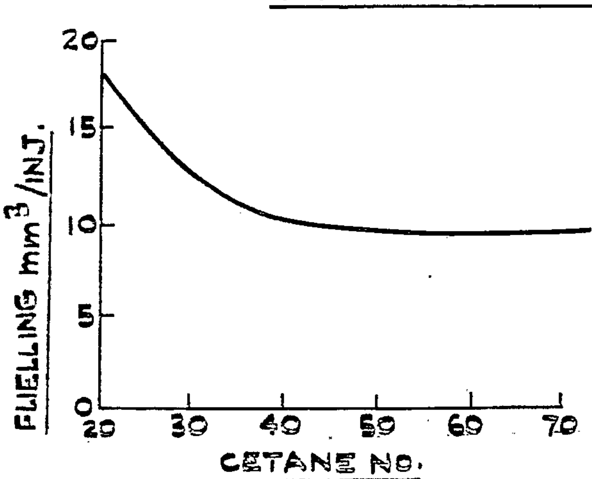
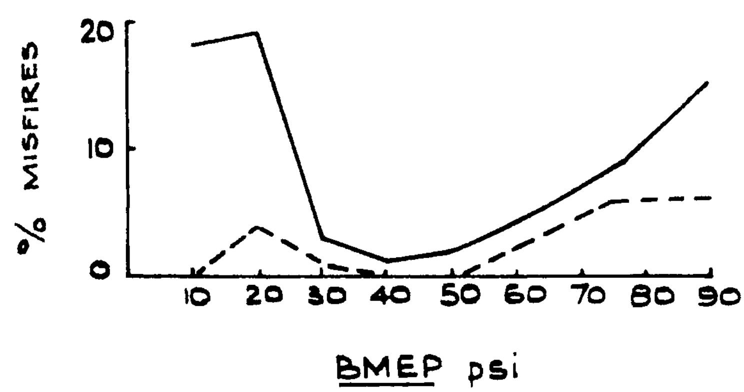


FIG.15 c [REF. 21] THE INFLUENCE OF CETANE NUMBER ON FUEL CONSUMPTION AT 350 RPM IDLE

3309KT

FIG. No. 16 [REF. 5, 11]
Drg. No. S 8078
Date SEPT. '80

THE INFLUENCE OF CETANE NUMBER ON MISFIRING TENDENCY - INDIRECT INJECTION ENGINE AT 3000 RPM

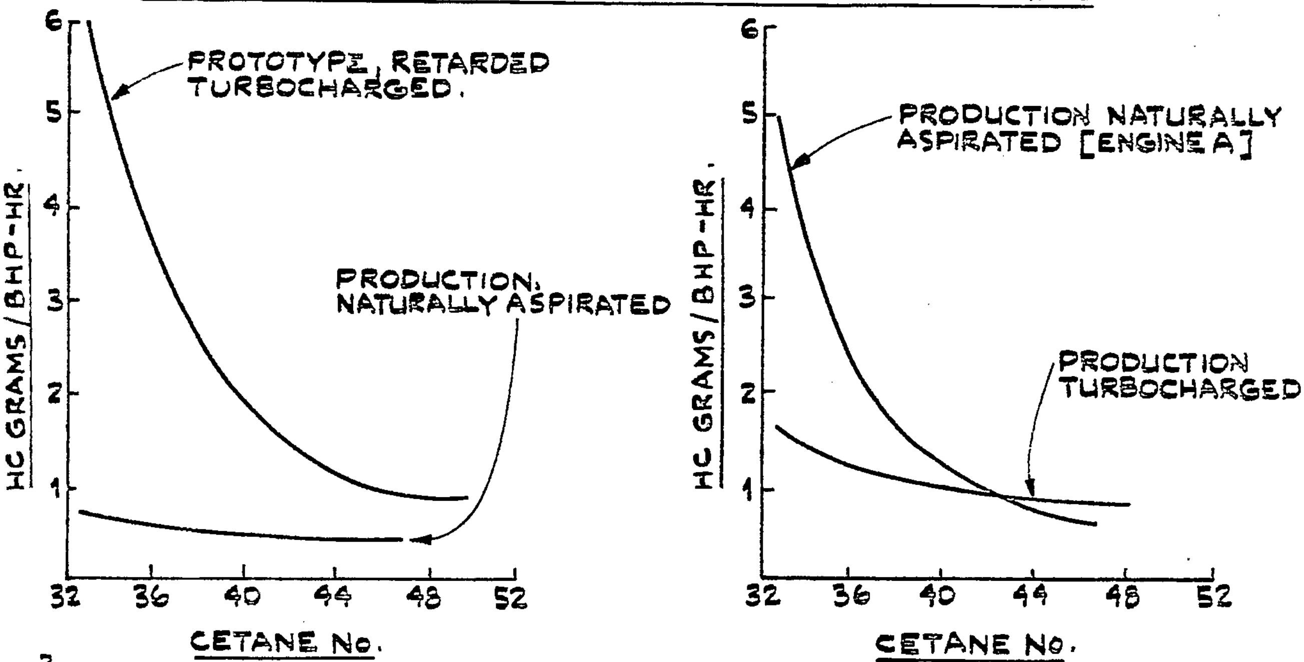


——— 46 CETANE No.
528°F 50% POINT

- - - 51 CETANE No.
518°F 50% POINT

FIG. No. 17a, b, c, [ref. 3]
Drg. No. S 8079
Date SEPT. '80

FIG. 17a THE INFLUENCE OF CETANE NUMBER ON 13 MODE CYCLE HC EMISSIONS – QUIESCENT CHAMBER DIRECT INJECTION ENGINES



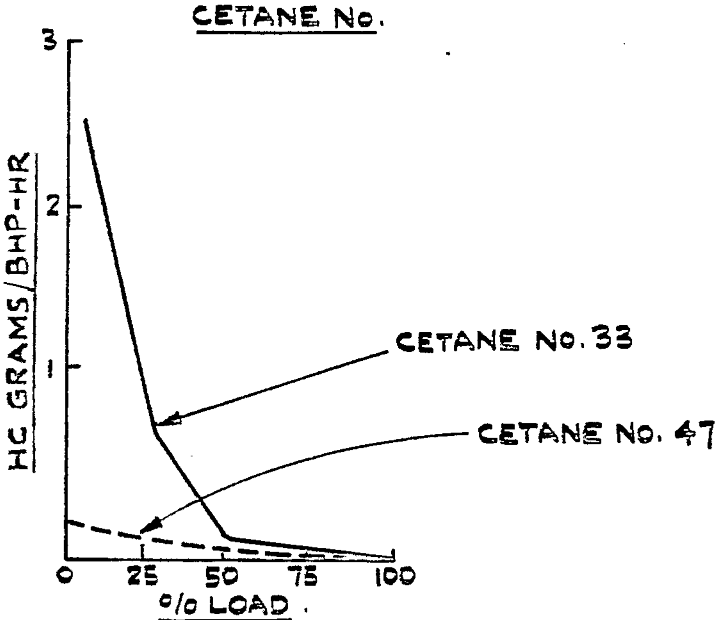


FIG. 17b. THE INFLUENCE OF CETANE NUMBER ON HC EMISSIONS OVER THE LOAD RANGE – QUIESCENT CHAMBER DIRECT INJECTION ENGINE AT RATED SPEED [ENGINE A – FIG. 17a]

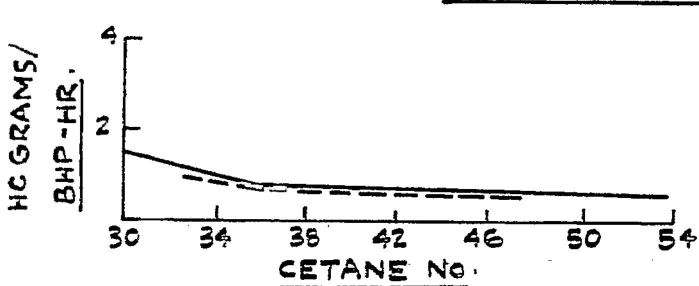


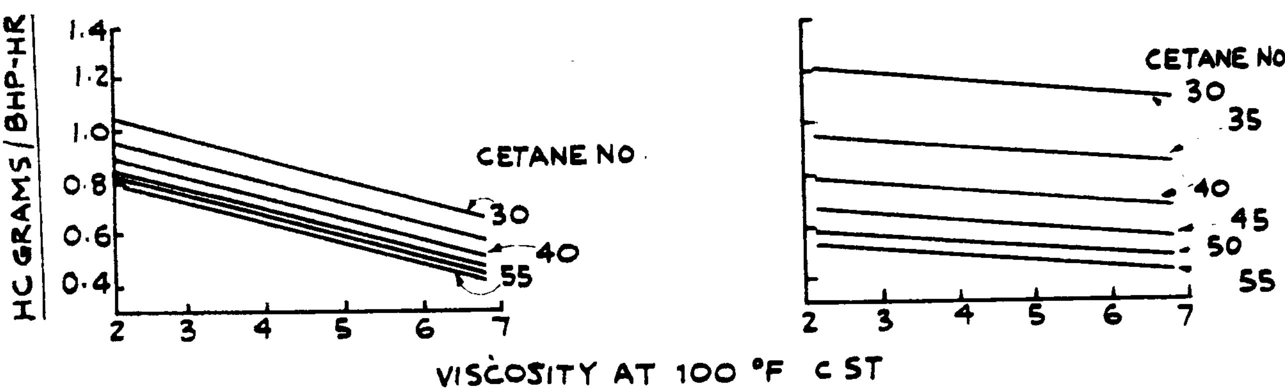
FIG. 17c. THE INFLUENCE OF CETANE NUMBER ON 13-MODE CYCLE HC EMISSIONS – TURBOCHARGED, SWIRLING DIRECT INJECTION ENGINES

FIG. No. 18 [REF. 25]

Drg. No. S 8080

Date SEPT. '80

THE INFLUENCE OF CETANE NUMBER ON 13 MODE CYCLE HC EMISSIONS - DIRECT INJECTION ENGINES.



NATURALLY ASPIRATED	TURBO CHARGED
2 STROKE	4 STROKE
CR 18·7	CR 15·0
RATED SPEED 2100 RPM	RATED SPEED 2100 RPM

FIG. No. 19 a. & b

Drg. No. 58081

Date SEPT. '80

FIG. 19 a [REF. 25]

THE INFLUENCE OF CETANE NUMBER ON 13 MODE CYCLE CO EMISSIONS - DIRECT INJECTION ENGINES.

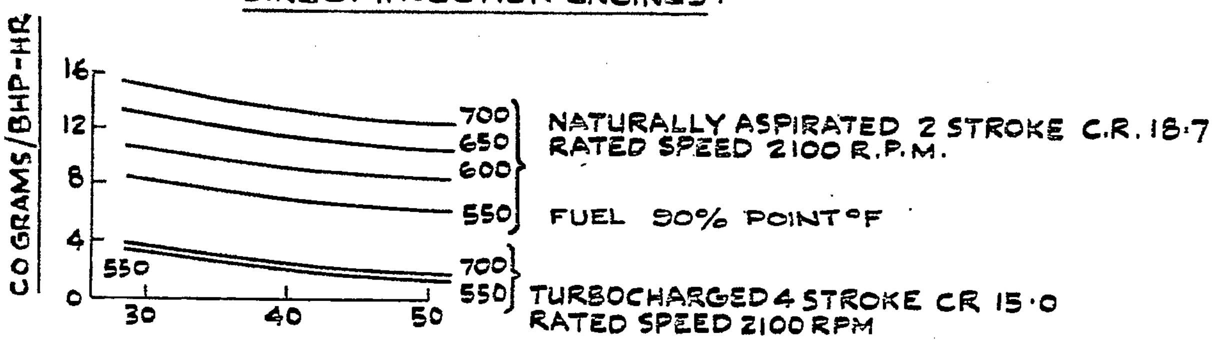
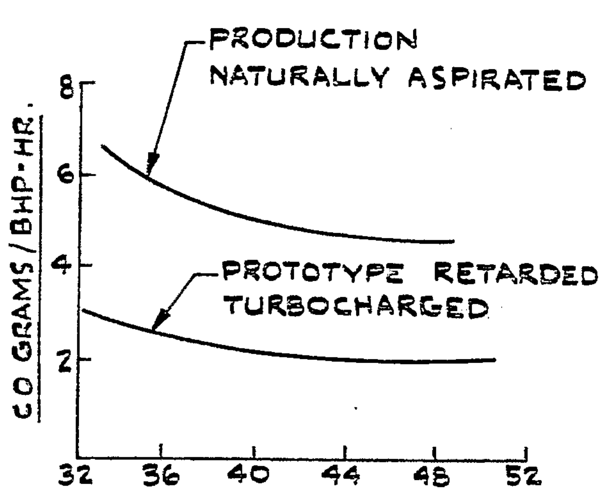
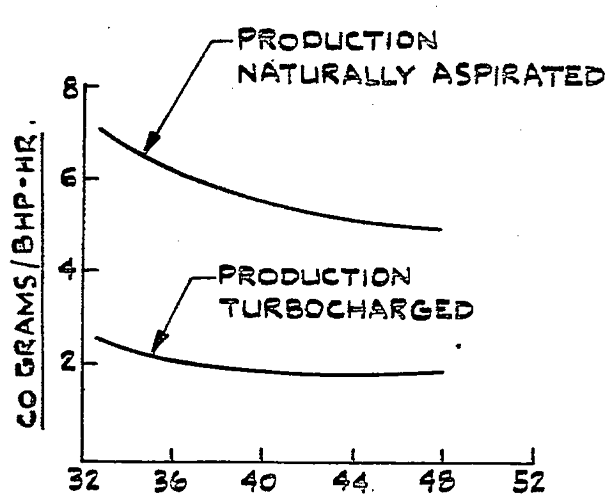


FIG. 19b. [REF. 3]

THE INFLUENCE OF CETANE NUMBER ON 13 MODE CYCLE CO EMISSIONS - QUIESCENT CHAMBER DIRECT INJECTION ENGINES.





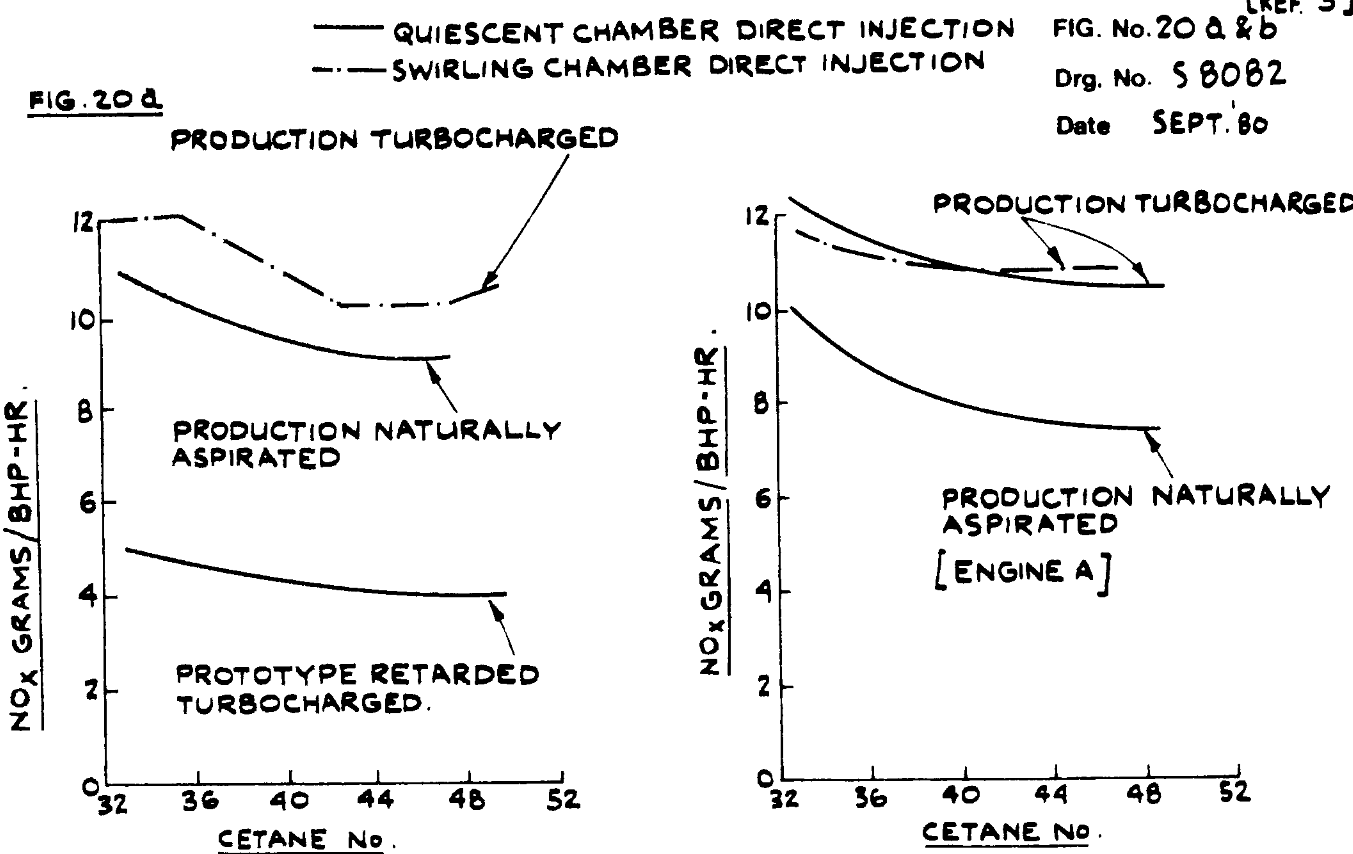
[REF. 3]

FIG. No. 20 a & b

Drg. No. S 8082

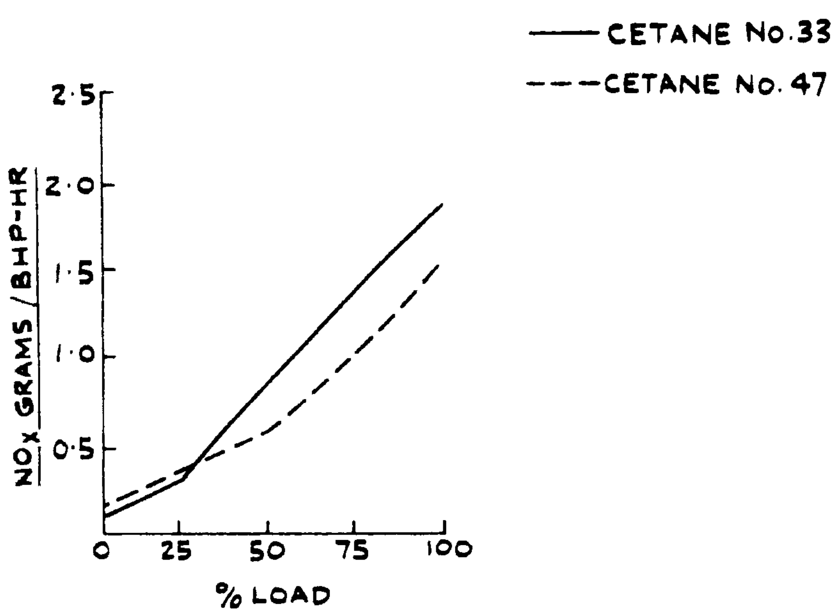
Date SEPT. '80

FIG. 20a



THE INFLUENCE OF CETANE NUMBER ON 13 MODE CYCLE NO_x EMISSIONS – QUIESCENT AND SWIRLING CHAMBER DIRECT INJECTION ENGINES.

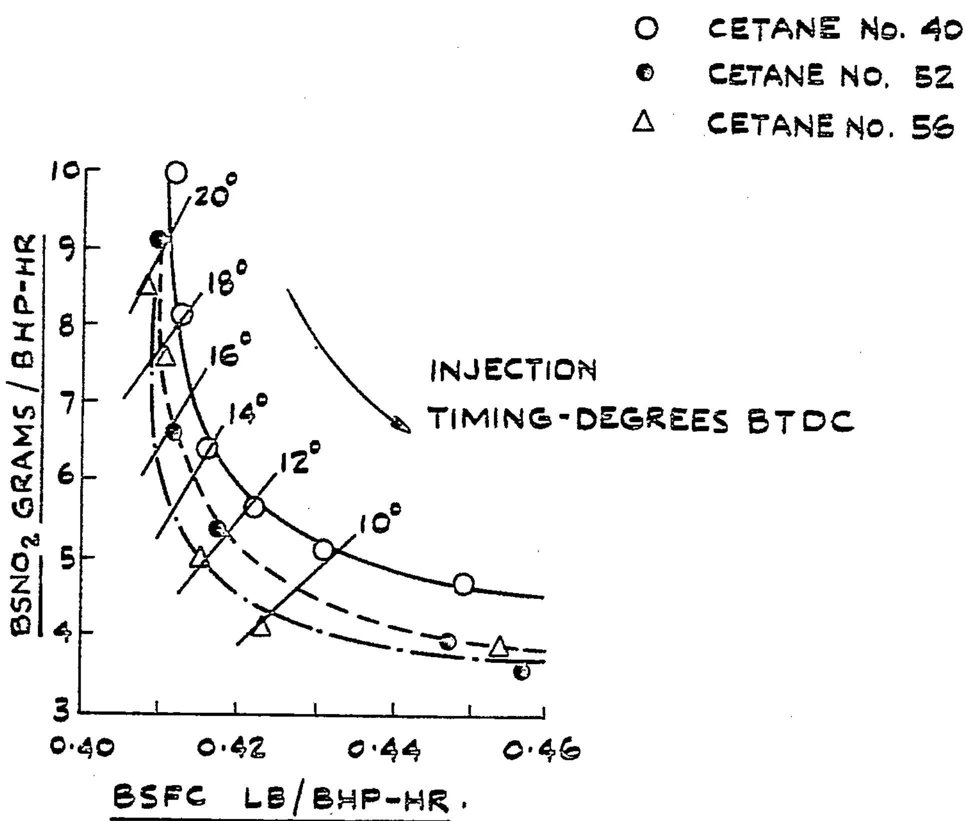
FIG. 20b.



THE INFLUENCE OF CETANE NUMBER ON NO_x EMISSIONS OVER THE LOAD RANGE – QUIESCENT CHAMBER DIRECT INJECTION ENGINE AT RATED SPEED [ENGINE A – FIG. 20a]

FIG. No. 21 [REF. 33]
Drg. No. S 8083
Date SEPT. '80

THE INFLUENCE OF CETANE NUMBER UPON BSNO$_2$ - BSFC TRADE-OFF AT CONSTANT SPEED AND LOAD.

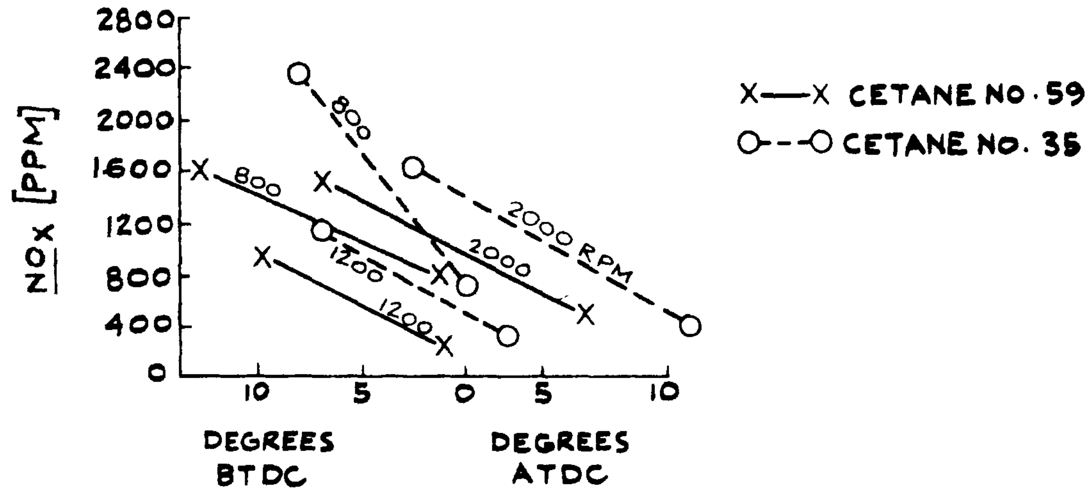


ENGINE :

SINGLE CYLINDER, 4 STROKE, DIRECT INJECTION
4·75" BORE X 4·75" STROKE CR 14·2:1

FIG. No. 22 [REF. 10/40]
Drg. No. S 8084
Date SEPT. '80

THE INFLUENCE OF CETANE NUMBER ON NO_X EMISSIONS



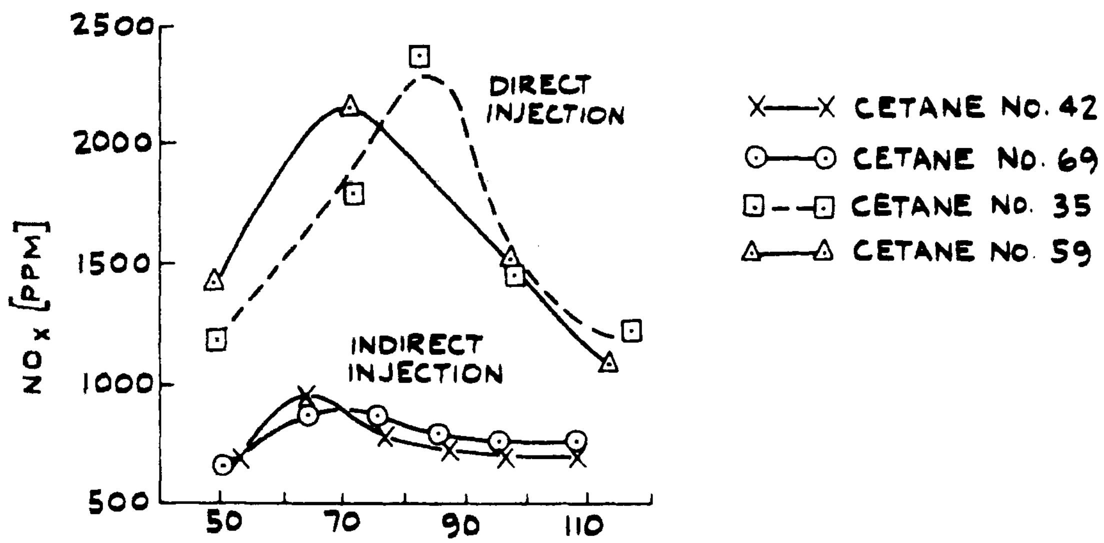
POSITION OF START OF COMBUSTION

ENGINE:

4·4" BORE X 5·1" STROKE, 6 CYLINDER 4 STROKE DIRECT INJECTION

INJECTION TIMING VARIED FOR COMPARING EQUIVALENT START OF COMBUSTION

THE INFLUENCE OF CETANE No. ON NO_X EMISSIONS OVER THE LOAD RANGE.



BMEP [P.S.I]

ENGINE:

4·4" BORE X 5·1" STROKE, 6 CYLINDER, 4 STROKE FITTED WITH EITHER DIRECT OR INDIRECT INJECTION COMBUSTION SYSTEMS - FIXED INJECTION TIMING.

FIG. No. 23 a & b
Drg. No. S 8085
Date SEPT. 80

FIG. 23 a. [REF. 3]

THE INFLUENCE OF CETANE NUMBER ON ENGINE NOISE - QUIESCENT CHAMBER DIRECT INJECTION ENGINES

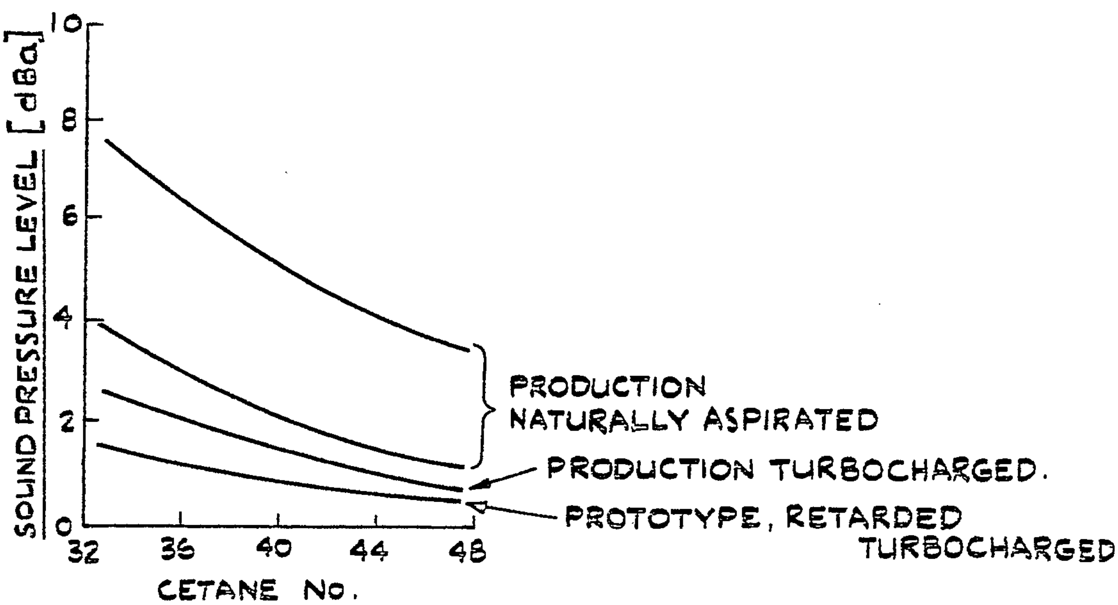


FIG. 23b. [REF. 5]

THE INFLUENCE OF CETANE NUMBER ON ENGINE NOISE

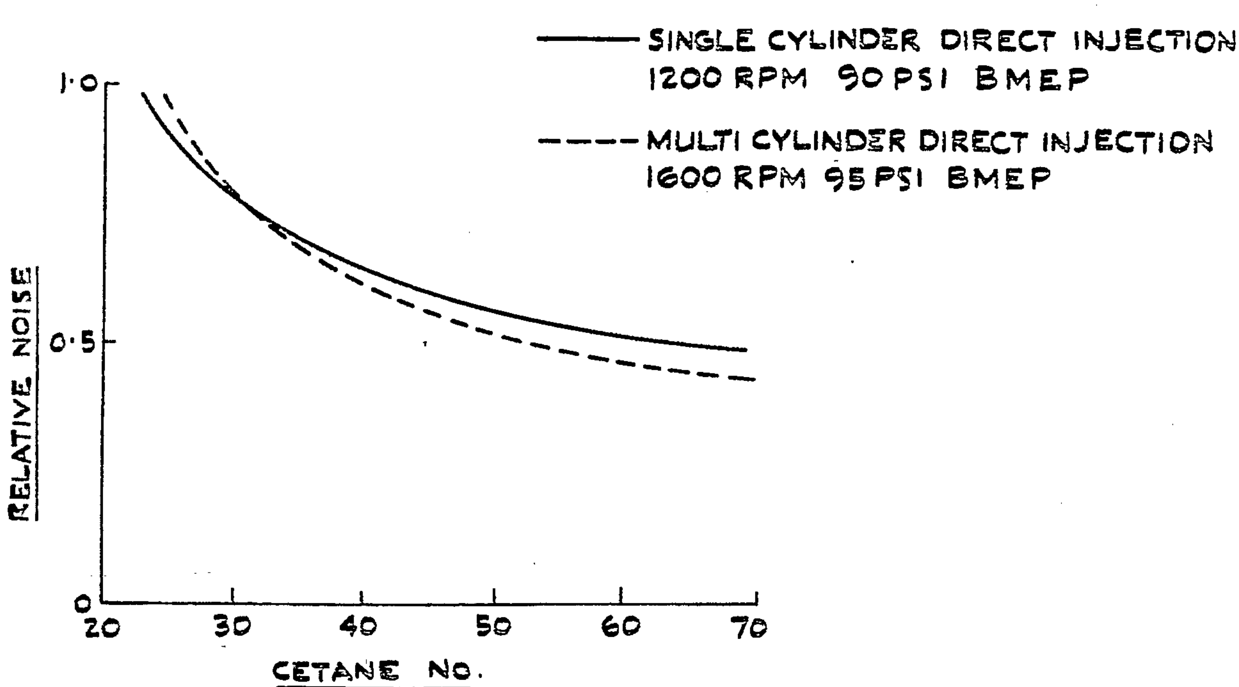
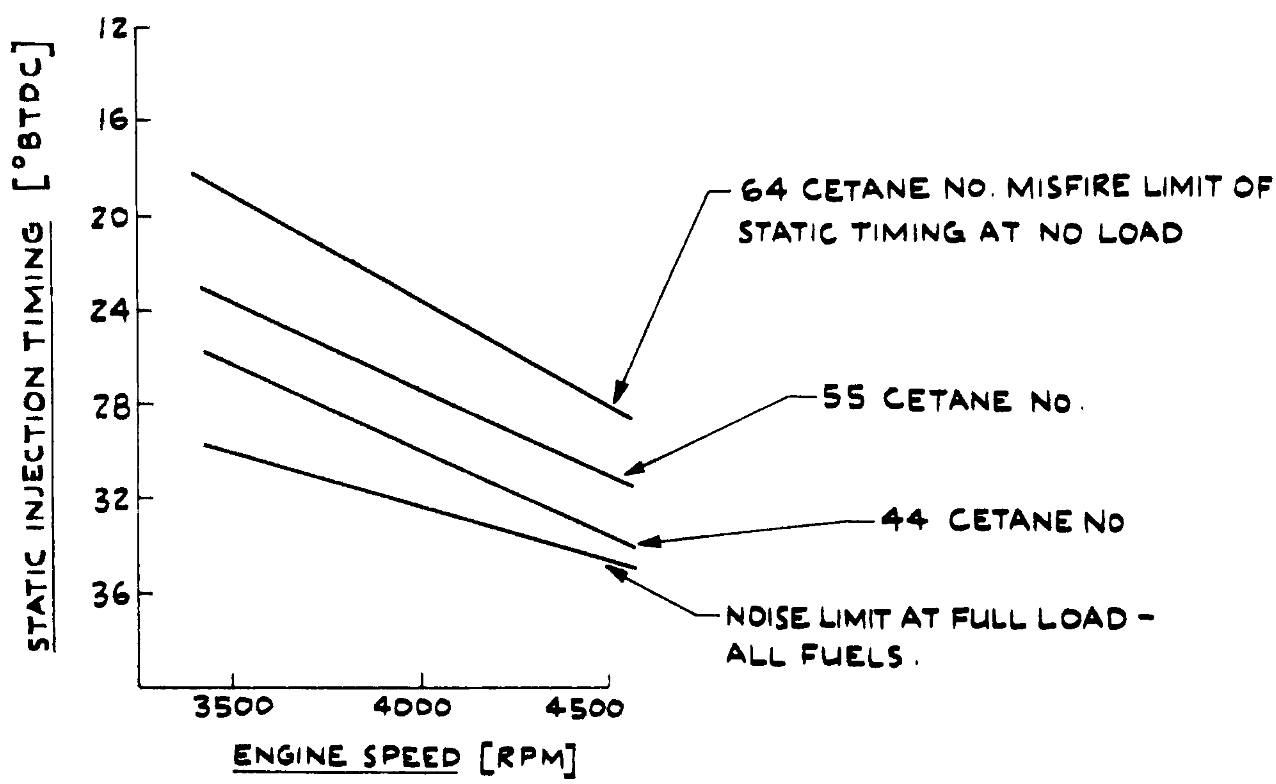


FIG. No. 24 [REF 34]
Drg. No. S 8086
Date SEPT. 80

THE INFLUENCE OF CETANE NUMBER ON INJECTION TIMING REQUIREMENTS TO CLEAR LIGHT LOAD MISFIRE - INDIRECT INJECTION ENGINE.



[REF. 21]

FIG. No. 25 a, b & c

Drg. No. 58087

Date SEPT. '80

FIG. 25 a.

THE INFLUENCE OF LOAD ON ALDEHYDE EMISSIONS AT 1400 R.P.M.

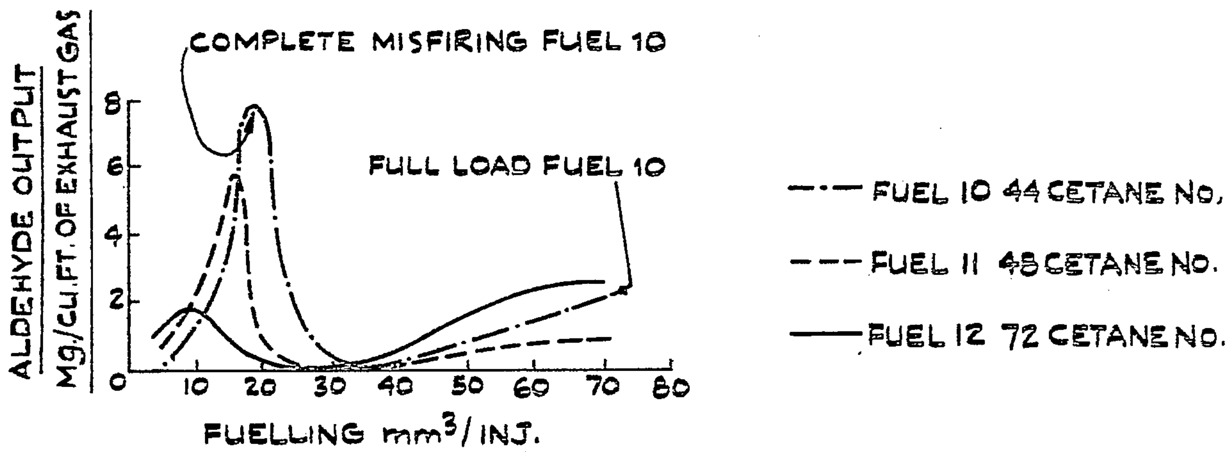


FIG. 25 b.

THE INFLUENCE OF CETANE NUMBER ON LIGHT LOAD PEAK ALDEHYDE EMISSIONS AND MISFIRING

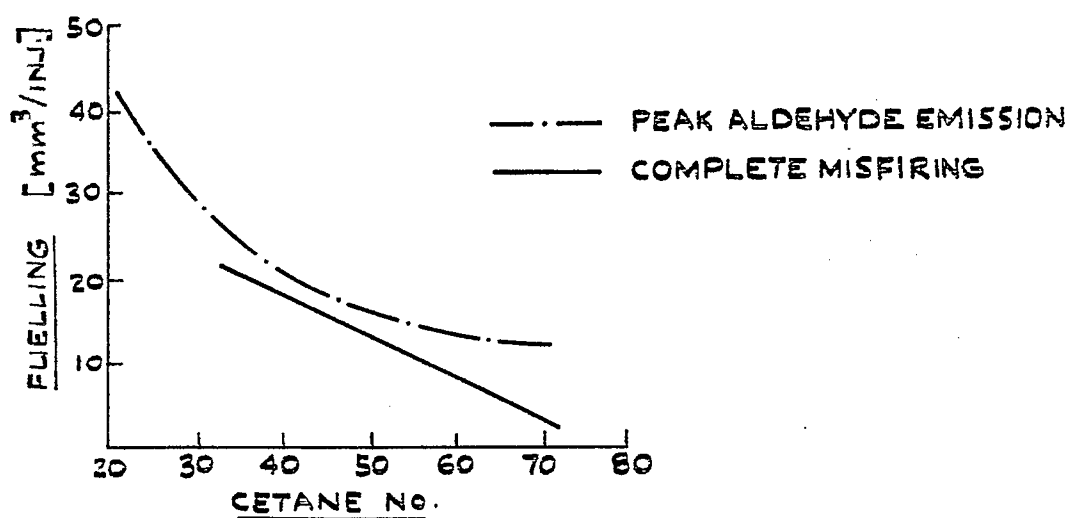


FIG. 25 c.

THE INFLUENCE OF CETANE NUMBER ON ALDEHYDE EMISSIONS AT 350 R.P.M. IDLE

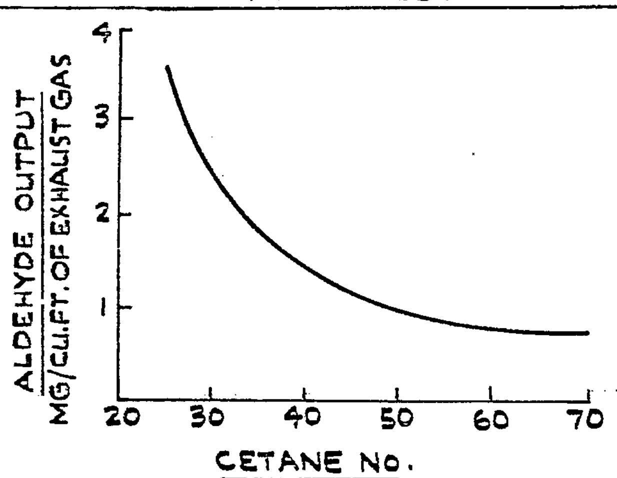
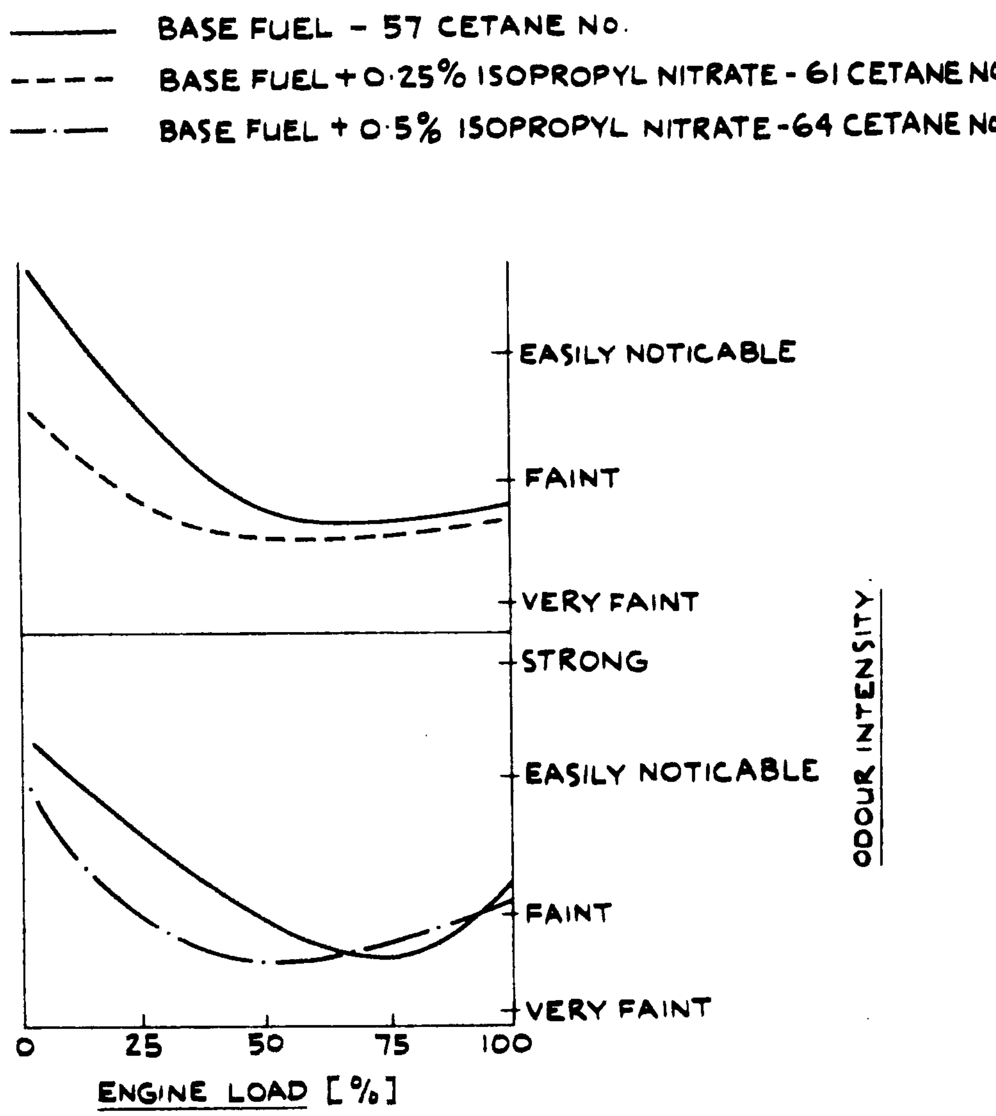


FIG. No. 26 [REF. 10]
Drg. No. S 8088
Date SEPT. '80

THE INFLUENCE OF CETANE NUMBER ON EXHAUST ODOUR OVER THE LOAD RANGE AT 1200 R.P.M.

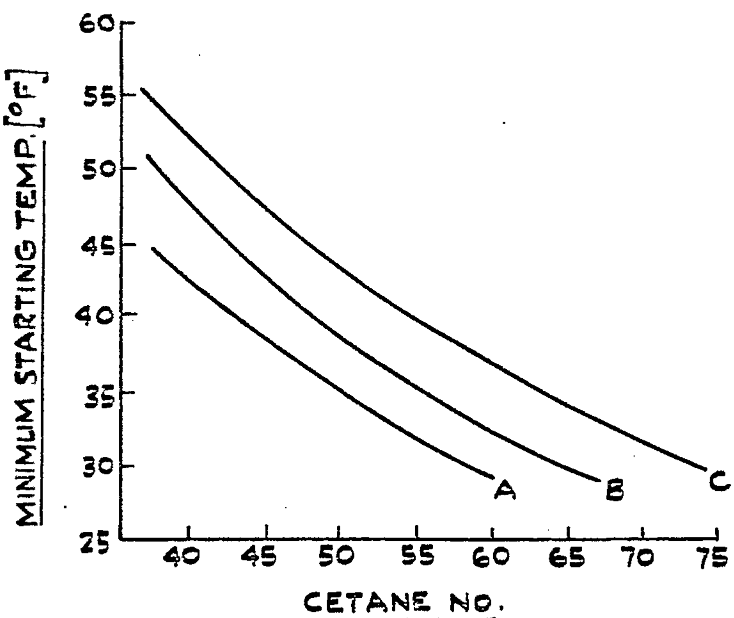


ENGINE:

MULTICYLINDER, 4·25" BORE x 6" STROKE, DIRECT INJECTION.

FIG. No. 27 [REF. 4]
Drg. No. 5 8092
Date SEPT. '80

THE INFLUENCE OF FUEL VOLATILITY ON MINIMUM STARTING TEMPERATURE FOR A RANGE OF CETANE NUMBERS



ENGINE:

SINGLE CYLINDER, 4·25" BORE x 6" STROKE, DIRECT INJECTION

COMPRESSION RATIO 13:1 CRANKING SPEED 160 R.P.M.

FUEL DISTILLATION RANGE °F

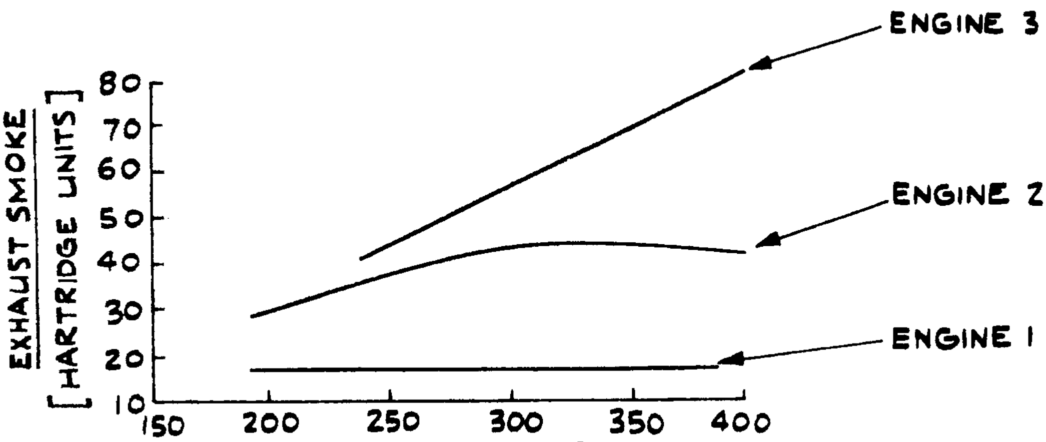
A 351 - 450

B 450 - 550

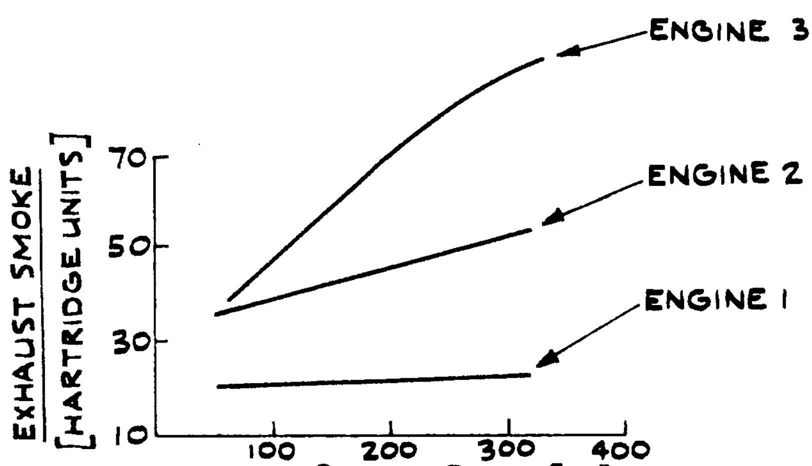
C 550 - 649

FIG. No. 28 [REF. 42]
Drg. No. S 8093
Date SEPT. 80

THE INFLUENCE OF FUEL VOLATILITY UPON EXHAUST SMOKE



CETANE NUMBER CONSTANT - 45   AROMATIC CONTENT CONSTANT - 15% VOL.
INITIAL BOILING POINT CONSTANT - 302°F



CETANE NUMBER CONSTANT - 45   AROMATIC CONTENT CONSTANT - 15% VOL.
FINAL BOILING POINT CONSTANT - 662°F

ENGINE	TEST CONDITIONS R.P.M.	BMEP [P.S.I]
1. PRODUCTION 4 CYL. DIRECT INJECTION	1550	97
2. SINGLE CYLINDER, EXPERIMENTAL 'M' SYSTEM DIRECT INJECTION	1200	94
3. PRODUCTION 4 CYL. INDIRECT INJECTION	4500	75

3309KT

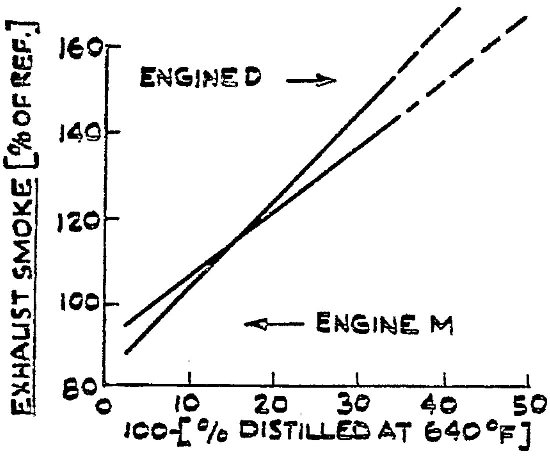
FIG. No. 29 a,b & c

Drg. No. 5 8094

Date SEPT. '80

FIG. 29a [REF. 25]

THE INFLUENCE OF FUEL VOLATILITY UPON EXHAUST SMOKE.



ENGINE D:

AIR SCAVENGED 426 CID 2 STROKE
DIRECT INJECTION - CR 18·7:1
RATED 228 BHP AT 2100 RPM

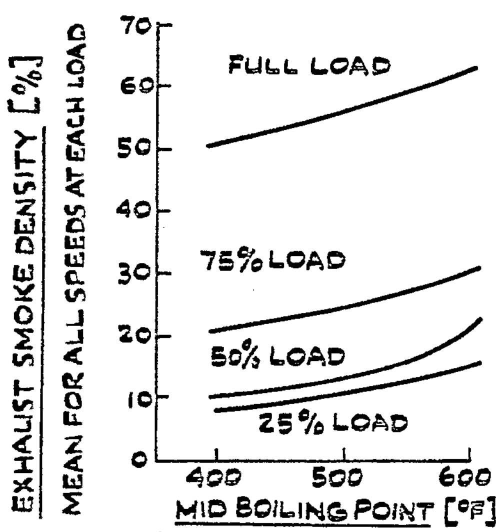
ENGINE M:

TURBOCHARGED 672 CID 4 STROKE
DIRECT INJECTION CR 15·0:1
RATED 283 BHP AT 2100 RPM

TEST CONDITIONS:
ENGINE D - MEAN SMOKE RECORDED OVER FULL LOAD SPEED RANGE.
ENGINE M. - MEAN SMOKE LEVEL DURING FEDERAL SMOKE CYCLE.

FIG. 29b [REF. 8]

INFLUENCE OF FUEL MID BOILING POINT ON EXHAUST SMOKE FOR FUELS OF FIXED IGNITION QUALITY - MEAN FOR A FOUR ENGINE STUDY.



ENGINES [HEAVY DUTY]

a) FAIRBANKS-MORSE 36A4¼ 4 STROKE, INDIRECT INJECTION.

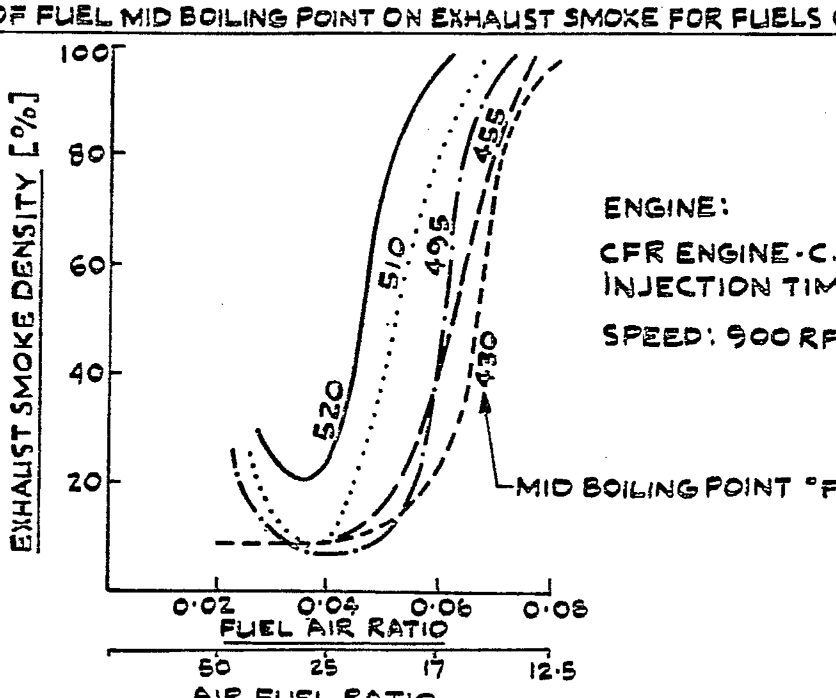
b) GM 371 - 2 STROKE, DIRECT INJECTION

c) HERCULES DRXB - 4 STROKE INDIRECT INJECTION.

d) MACK END 457, 4 STROKE, LANOVA CELL

FIG. 29c [REF. 24]

EFFECT OF FUEL MID BOILING POINT ON EXHAUST SMOKE FOR FUELS OF 45 CETANE NUMBER.



ENGINE:

CFR ENGINE - C.R. 16·1:1,
INJECTION TIMING 17° BTDC

SPEED: 900 RPM

FIG. No. 30 [REF. 21]
Drg. No. S8095
Date SEPT. '80

THE INFLUENCE OF FUEL MID BOILING POINT ON THE MAXIMUM EXHAUST SMOKE OBSERVED AT 1400 RPM. 75 % FULL LOAD FOLLOWING 25 MINUTES IDLING AT 350 R.P.M. - FUELS OF SIMILAR CETANE NUMBER - CN.

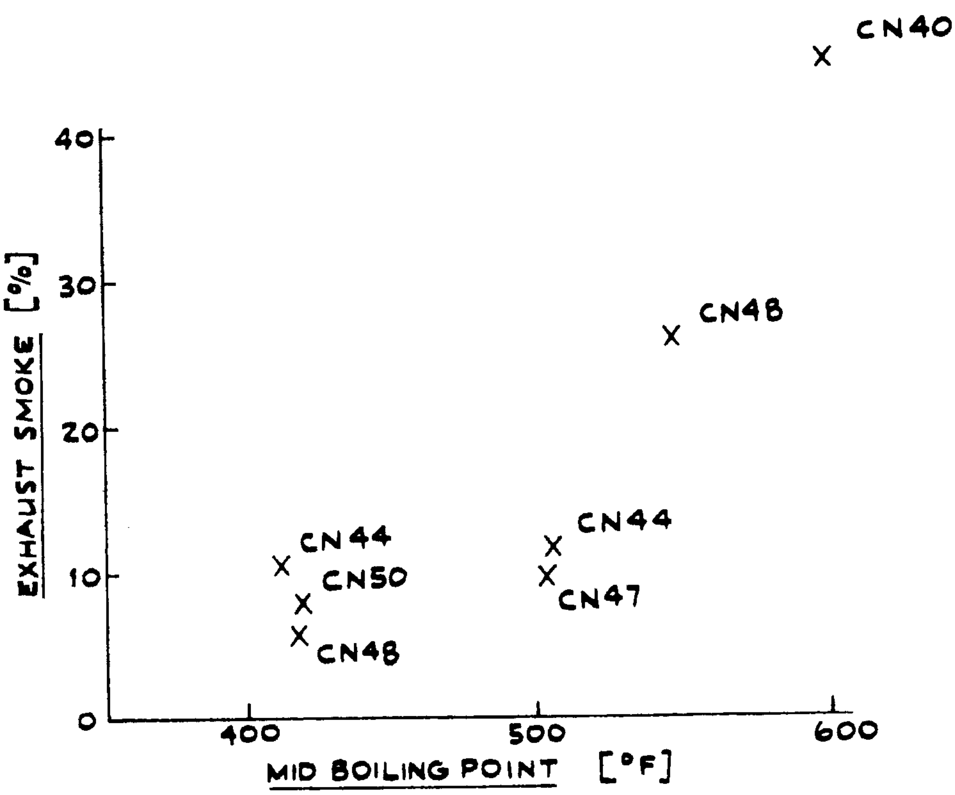
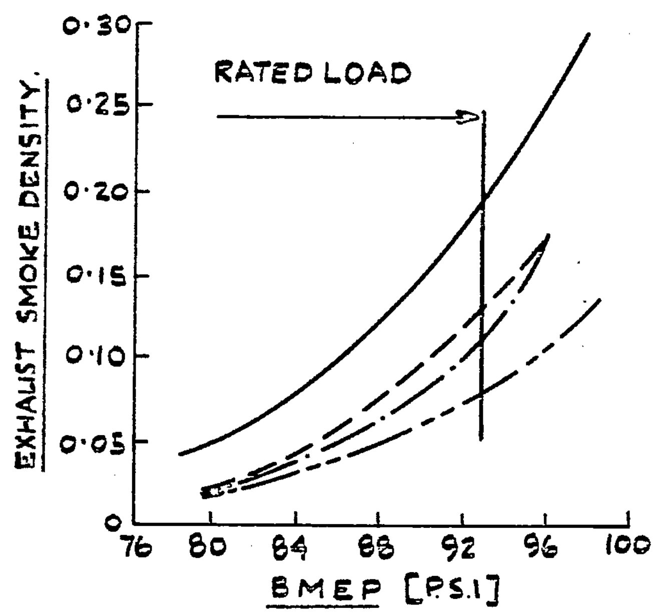


FIG. No. 31 [REF. 15]
Drg. No. 5 8096
Date SEPT. '80

THE INFLUENCE OF FUEL MID BOILING POINT ON EXHAUST SMOKE FOR FUELS OF CONSTANT CETANE NUMBER - DIRECT INJECTION ENGINE AT 1800 R.P.M.



——— CETANE No. 55, 50% POINT 446°F

– – – CETANE No. 55, 50% POINT 538°F

–·– CETANE No. 40. 50% POINT 408°F

–··– CETANE No. 40 50% POINT 529°F

FIG. No. 32

Drg. No. S 8097

Date SEPT. '80

THE INFLUENCE OF FUEL MID BOILING POINT ON 13 MODE CYCLE HC EMISSIONS.

ENGINE A	4·5" BORE X 5" STROKE X 8 CYL. - 4 STROKE NATURALLY ASPIRATED DIRECT INJECTION.	REF. 37
ENGINE C	4·25" BORE X 5" STROKE X 6 CYL. - 2 STROKE AIR SCAVENGED DIRECT INJECTION.	
ENGINE D	4·78" BORE X 6" STROKE X 6 CYL. - 4 STROKE TURBOCHARGED DIRECT INJECTION.	
ENGINE A'	AVL 520 SINGLE CYL. DIRECT INJECTION	REF. 12.
ENGINE B'	PEUGEOT 504 4 CYL. SWIRL CHAMBER INDIRECT INJECTION.	
ENGINE C'	SAVIEM 720-08 4 CYL. 'M' SYSTEM DIRECT INJECTION	

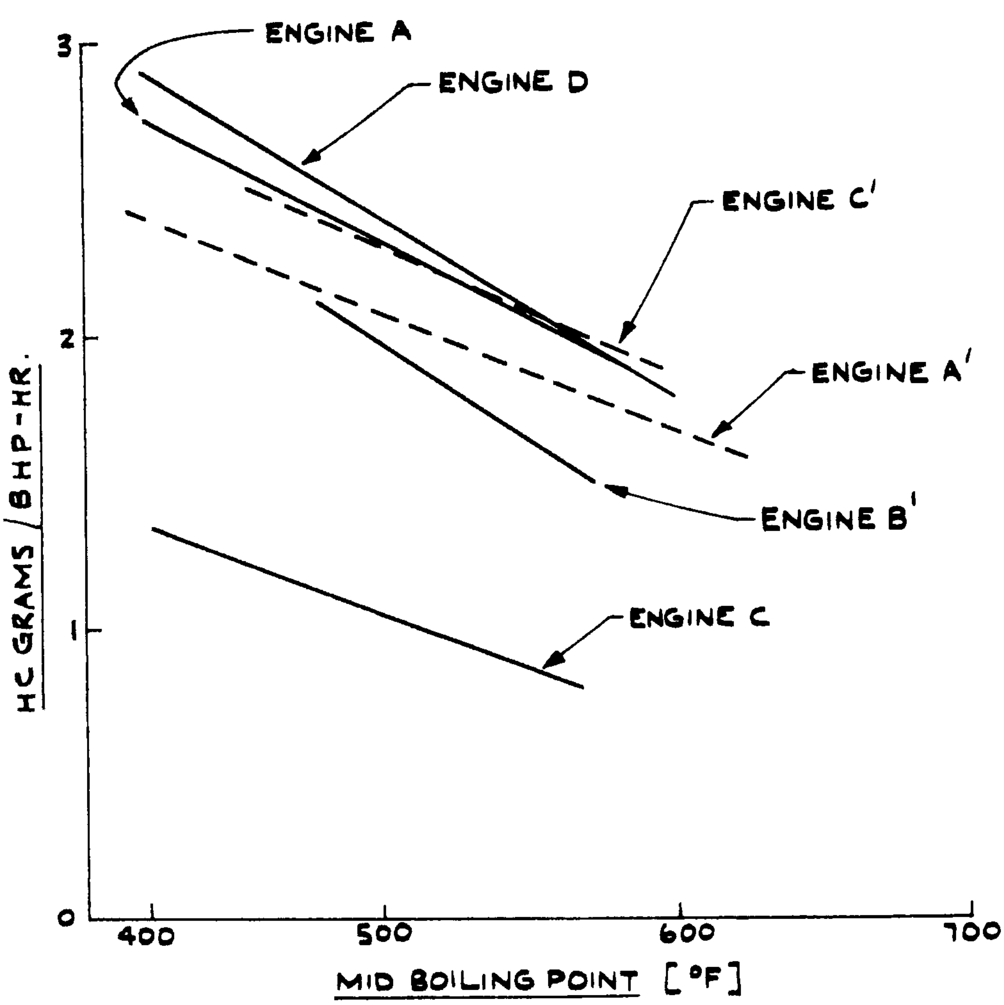
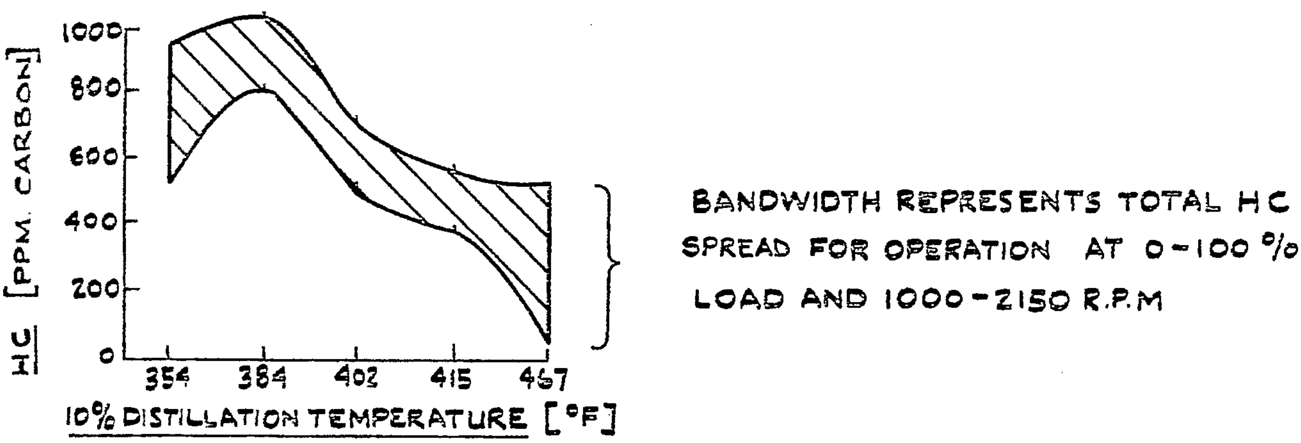


FIG. No. 33 [REF. 47]
Drg. No. S 8098
Date SEPT. '80

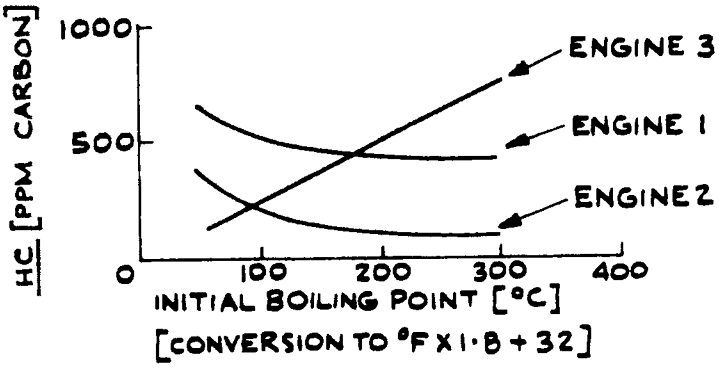
THE INFLUENCE OF 10% DISTILLATION TEMPERATURE ON HC EMISSIONS



ENGINE: GM 6V-71N, STANDARD SAC VOLUME [3·5 mm³] NEEDLE VALVE INJECTORS.

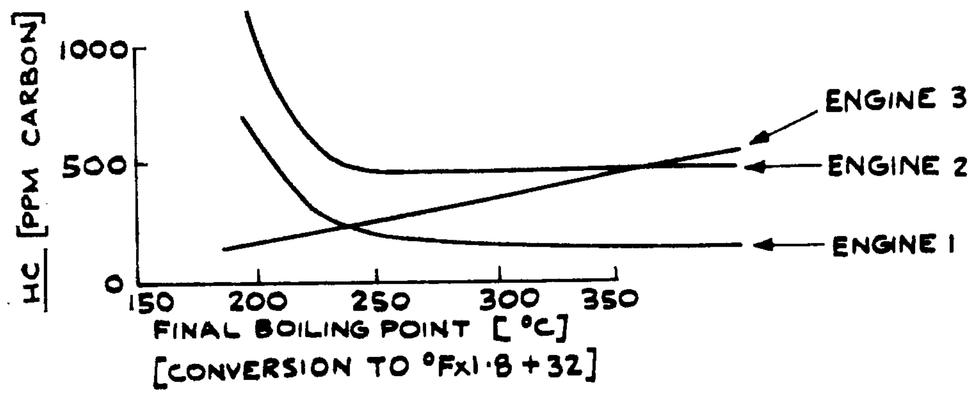
FIG. No. 34 (REF. 42)
Drg. No. S 8099
Date SEPT '80

THE INFLUENCE OF INITIAL AND FINAL BOILING POINTS ON HC EMISSIONS



CETANE NUMBER CONSTANT - 45 AROMATIC CONTENT CONSTANT - 15 % VOL.

FINAL BOILING POINT CONSTANT - 662 °F



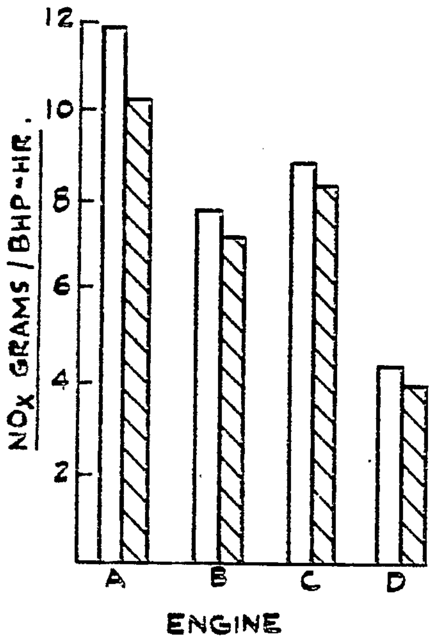
CETANE NUMBER CONSTANT - 45 AROMATIC CONTENT CONSTANT - 15 % VOL.

INITIAL BOILING POINT CONSTANT - 302 °F

ENGINE	TEST CONDITIONS R.P.M.	B.M.E.P [P.S.I]
1. PRODUCTION, 4 CYL. DIRECT INJECTION	1550	97
2. SINGLE CYLINDER EXPERIMENTAL 'M' SYSTEM DIRECT INJECTION	1200	94
3. PRODUCTION 4 CYLINDER, INDIRECT INJECTION	4500	75

FIG. 35a [REF. 3]

THE INFLUENCE OF FUEL VOLATILITY ON 13 MODE CYCLE NO_X EMISSIONS - QUIESCENT CHAMBER, DIRECT INJECTION ENGINES



FUELS:	□	▧
CETANE NUMBER	46·5	47
INITIAL BOILING POINT °F	362	416
50% °F	504	503
FINAL BOILING POINT °F	632	700
API GRAVITY	34·4	35·2

ENGINE TYPE:

QUIESCENT CHAMBER, DIRECT INJECTION 4 STROKE.

A. PRODUCTION TURBOCHARGED.

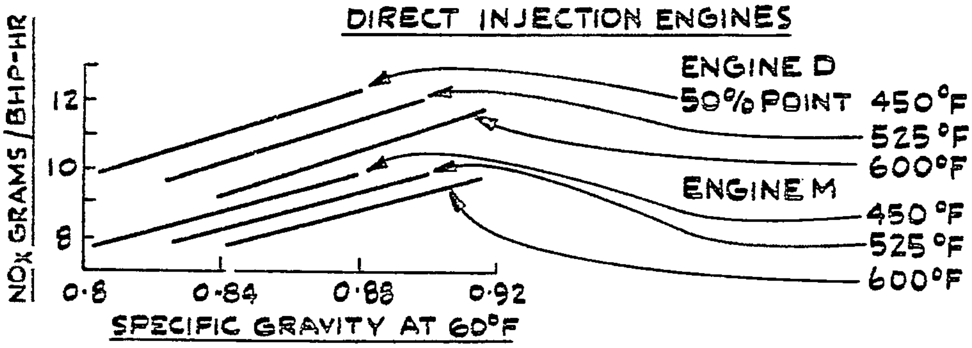
B. PRODUCTION NATURALLY ASPIRATED.

C. PRODUCTION NATURALLY ASPIRATED.

D. PROTOTYPE RETARDED TURBO CHARGED.

FIG. 35b [REF. 25]

THE INFLUENCE OF FUEL 50% POINT ON 13 MODE CYCLE NO_X EMISSIONS DIRECT INJECTION ENGINES



ENGINE D

AIR SCAVENGED 426 CID 2 STROKE DIRECT INJECTION - CR 18·7:1

RATED 228 BHP AT 2100 RPM

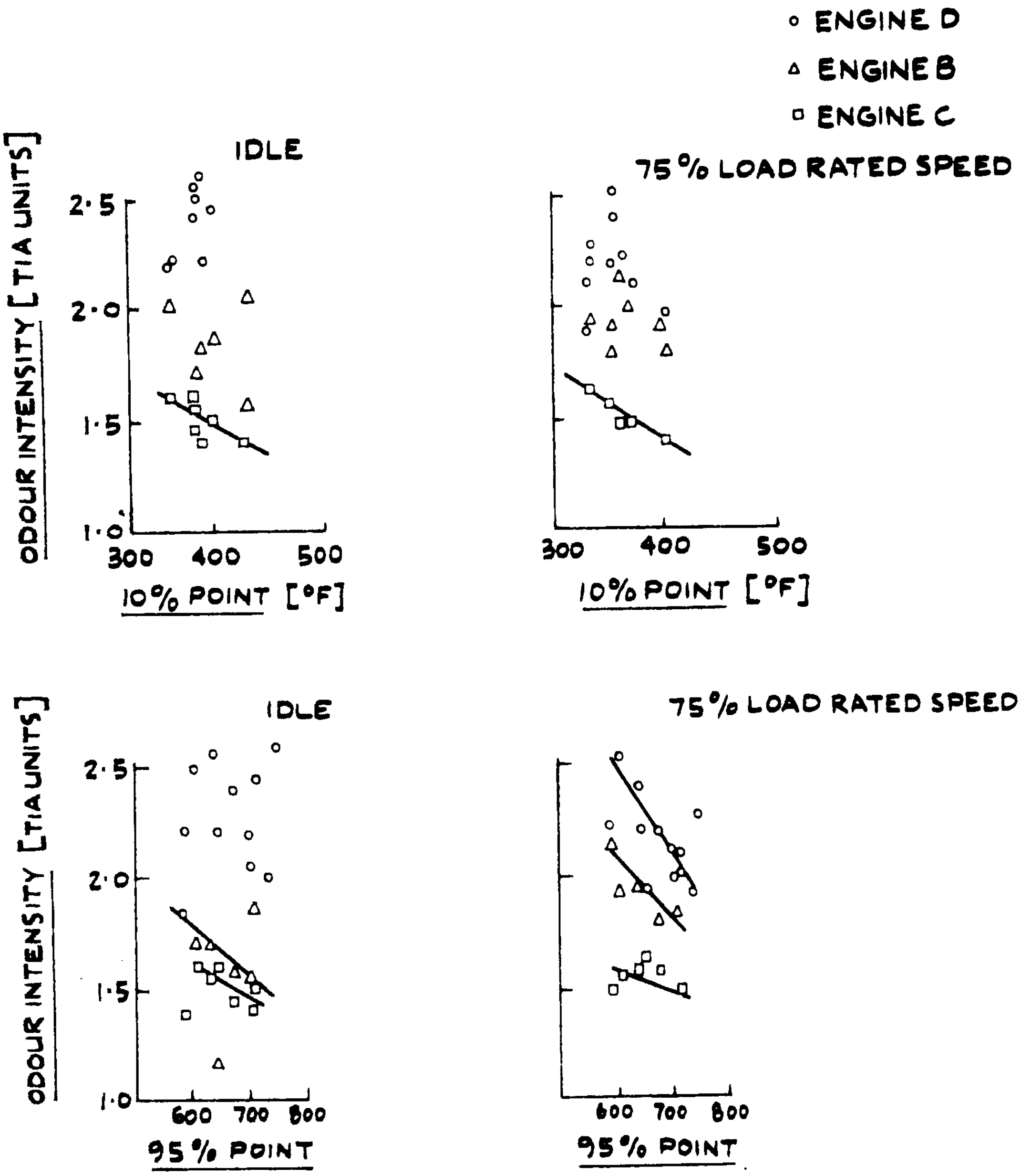
ENGINE M

TURBOCHARGED 672 CID 4 STROKE DIRECT INJECTION CR 15·0:1

RATED 263 BHP AT 2100 RPM

FIG. No. 3G [REF. 37]
Drg. No. S 8101
Date SEPT. '80

THE INFLUENCE OF FUEL 10% AND 95% POINTS ON EXHAUST ODOUR AT IDLE AND PART LOAD



ENGINES:

ENGINE B — 5·5" BORE X 6" STROKE X 6 CYL. 4 STROKE, NATURALLY ASPIRATED, DIRECT INJECTION, RATED 230 HP AT 2100 RPM

ENGINE C — 4·25" BORE X 5" STROKE X 6 CYL. 2 STROKE, AIR SCAVANGED, DIRECT INJECTION, RATED 228 HP. AT 2100 R.P.M.

ENGINE D — 4·78" BORE X 6" STROKE X 6 CYL. 4 STROKE TURBOCHARGED, DIRECT INJECTION, RATED 237 HP. AT 1800 R.P.M.

FIG. No. 37

Drg. No. S 8089

Date SEPT. '80

THE INFLUENCE OF FUEL AROMATIC CONTENT ON EXHAUST SMOKE

KEY	DETAILS OF TEST CONDITIONS AND ENGINES		
X——X	2100 RPM - FULL LOAD - 6 CYL. 5·5" BORE X 6" STROKE		[REF 37]
X- - -X	1500 RPM - FULL LOAD - 4 STROKE - NATURALLY ASPIRATED - DIRECT INJECTION		[REF 37]
⊙——⊙	2800 RPM - FULL LOAD - 8 CYL. 4·5" BORE X 5" STROKE		[REF 37]
⊙- - -⊙	1400 RPM - FULL LOAD - 4 STROKE - NATURALLY ASPIRATED - DIRECT INJECTION		[REF 37]
△——△	2600 RPM EQUAL POWER, FULL LOAD	ENGINE DETAILS NOT STATED	[REF 45]
▽——▽	1200 RPM EQUAL POWER, FULL LOAD	ENGINE DETAILS NOT STATED	[REF 45]
•——•	1550 RPM, 97 PSI BMEP — 4 CYL. DIRECT INJECTION	CONSTANT FOR ALL FUELS	[REF 42]
◧——◧	1200 RPM, 94 PSI BMEP — SINGLE CYL. DIRECT INJECTION 'M' SYSTEM	CONSTANT FOR ALL FUELS	[REF 42]
▲——▲	4500 RPM, 75 PSI BMEP — 4 CYL INDIRECT INJECTION	CONSTANT FOR ALL FUELS	[REF 42]
+——+	1200 RPM, 101 PSI BMEP — 6 CYL. 4·8" BORE X 5·5" STROKE, 4 STROKE DIRECT INJECTION	CONSTANT FOR ALL FUELS	[REF. 17]

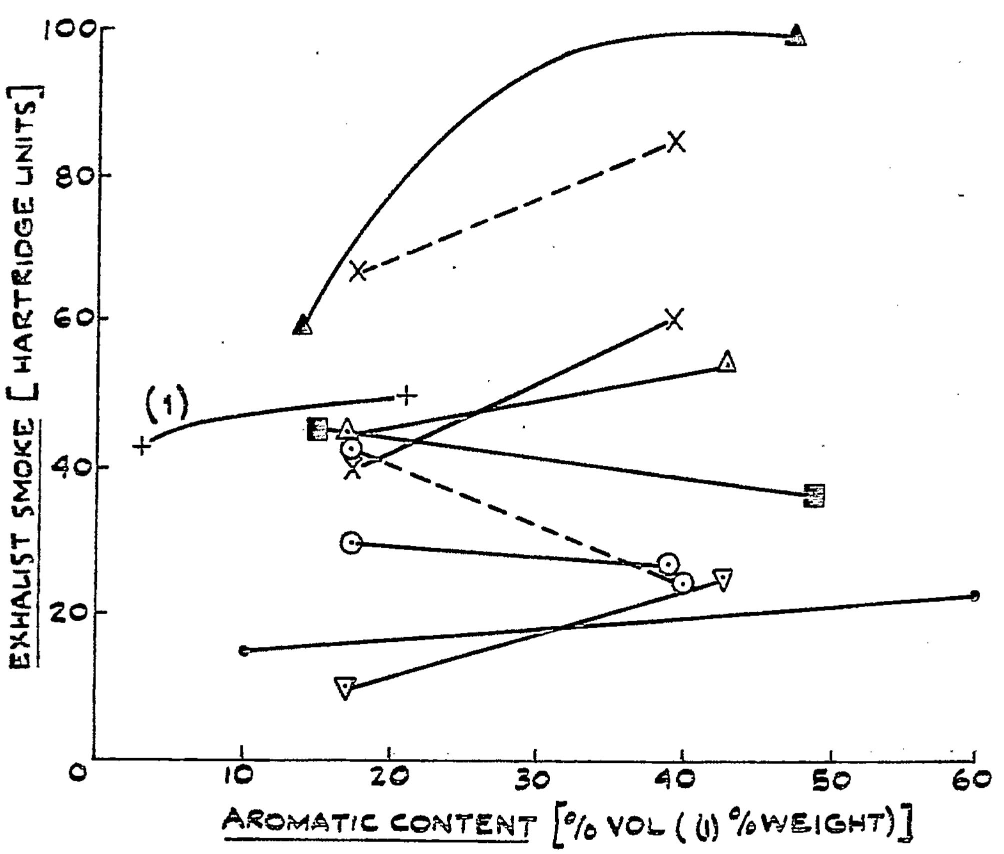
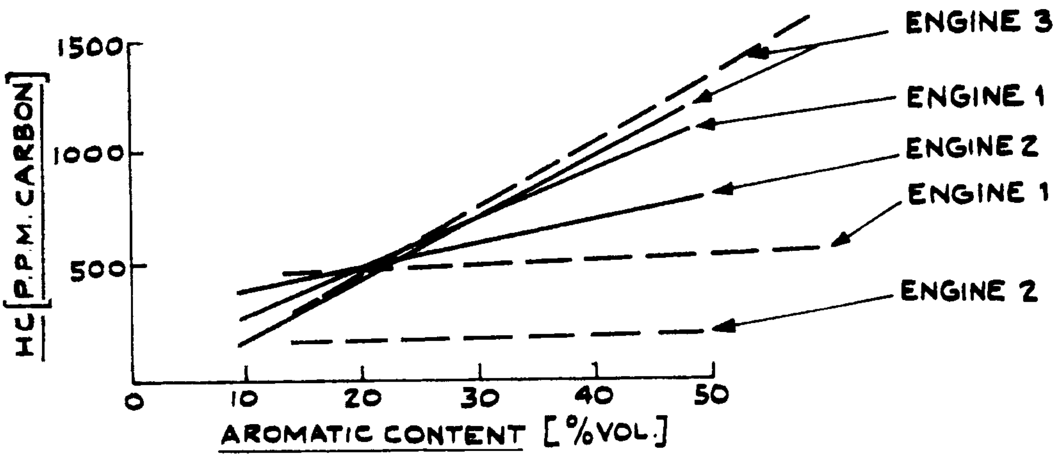


FIG. No. 38 [ref.42]

Drg. No. S 8090

Date SEPT. '80

THE INFLUENCE OF FUEL AROMATIC CONTENT ON HC EMISSIONS AT COLD IDLE AND NORMAL TEMPERATURE, HIGH LOAD OPERATING CONDITIONS - FUELS OF FIXED BOILING RANGE [302 - 662°F] AND CETANE NUMBER [45]



ENGINES:

ENGINE 1 4 CYLINDER, PRODUCTION DIRECT INJECTION

ENGINE 2 SINGLE CYLINDER, EXPERIMENTAL 'M' SYSTEM, DIRECT INJECTION

ENGINE 3 4 CYLINDER, PRODUCTION, INDIRECT INJECTION

TEST CONDITIONS [SPEED AND LOAD CONSTANT FOR ALL FUELS]

INJECTION TIMING FIXED ON ALL FUELS

——— COLD RUNNING - NO LOAD, 800 R.P.M. IDLE

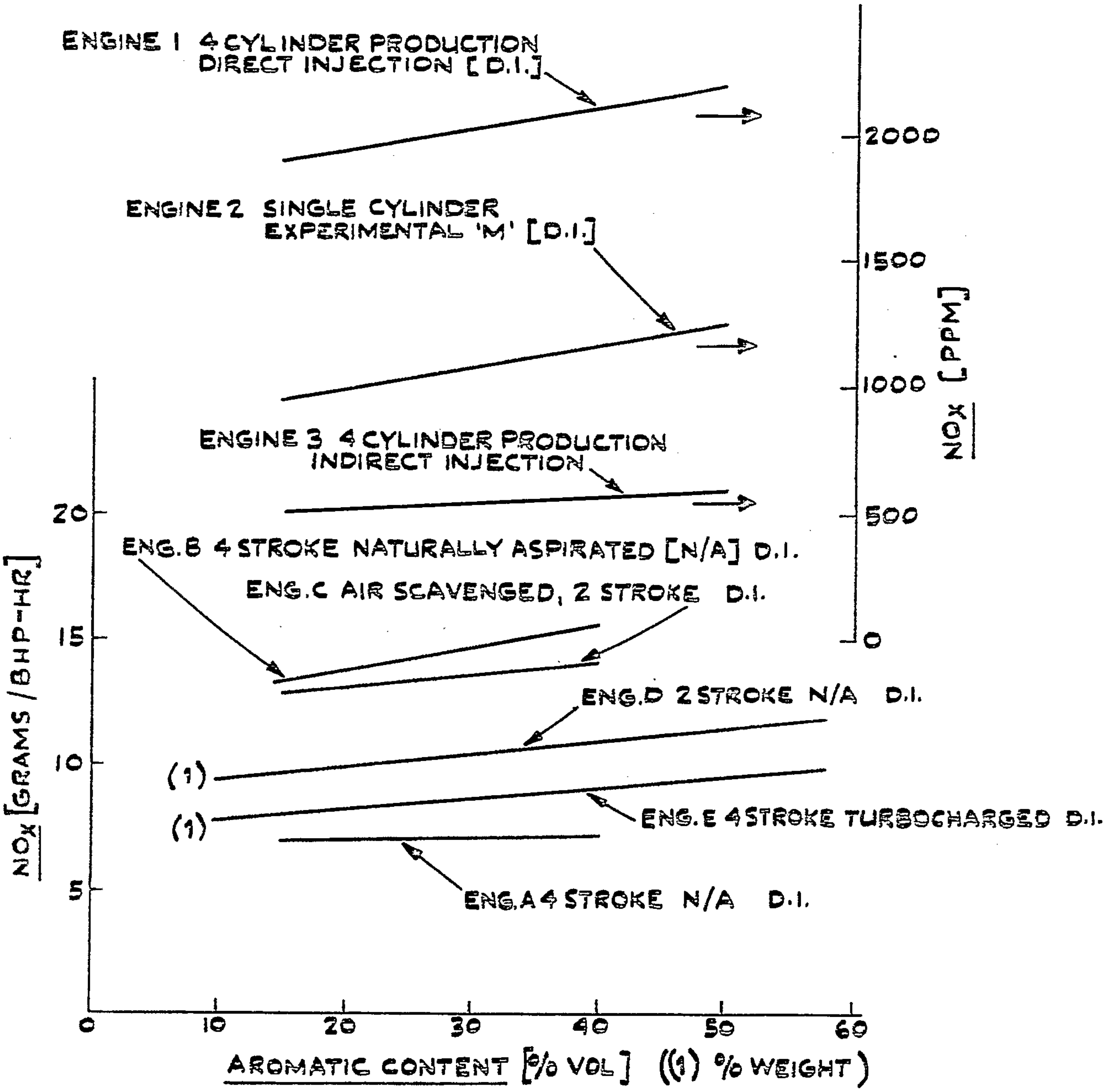
CYLINDER HEAD TEMPERATURE MAINTAINED AT 23°F

– – – NORMAL TEMPERATURE OPERATION

	RPM	BMEP [P.S.I.]	AIR RATIO [WITH REFERENCE FUEL]
ENGINE 1	1550	97	1·8
ENGINE 2	1200	94	1·4
ENGINE 3	4500	75	0·98

FIG. No.39 [REF. 25,36,42]
Drg. No. S 8091
Date SEPT. 80

THE INFLUENCE OF FUEL AROMATIC CONTENT ON NO_x EMISSIONS



TEST CONDITIONS

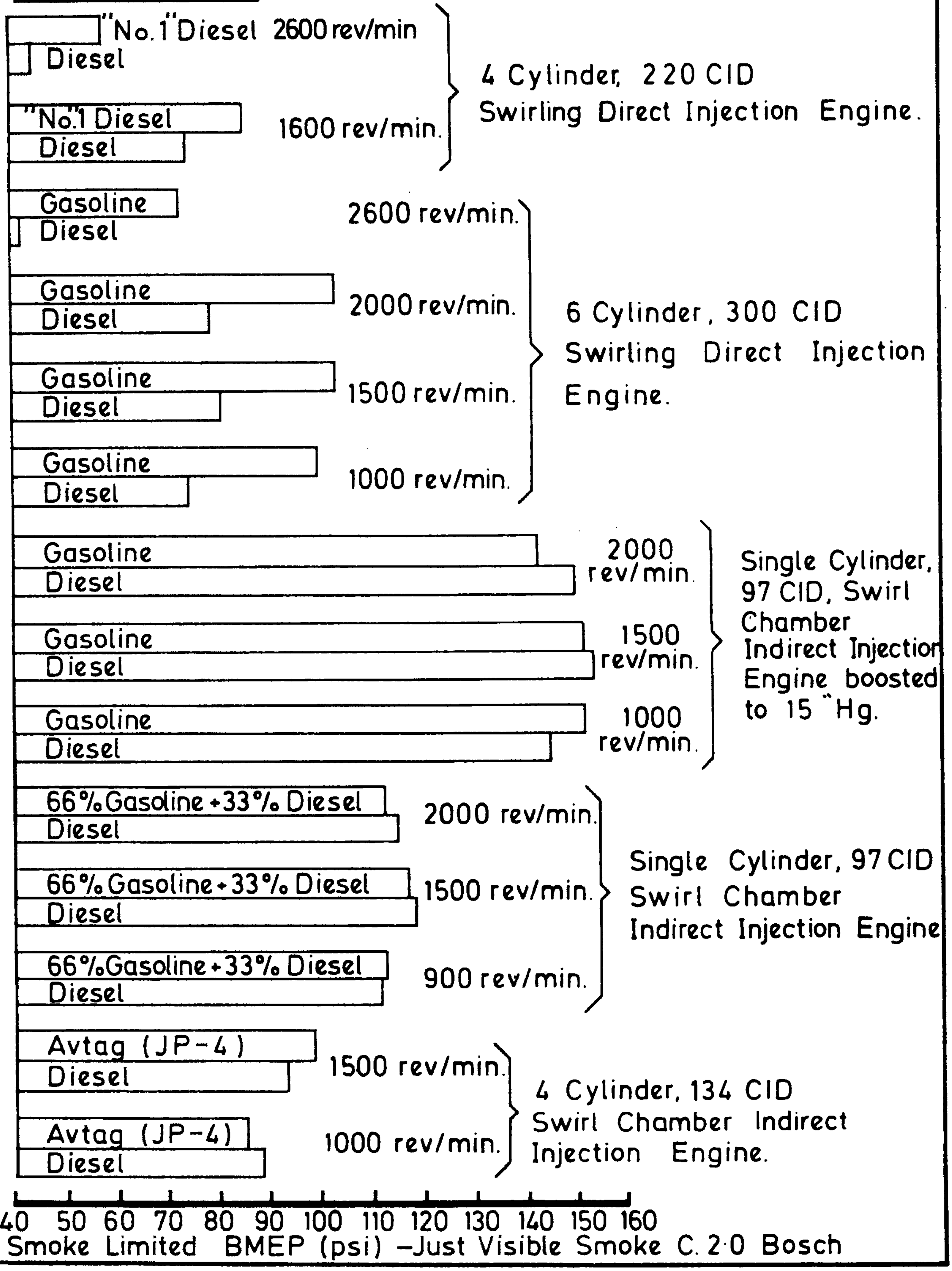
ENGINE 1	1550 R.P.M.	97 P.S.I. B.M.E.P.	CONSTANT SPEED AND LOAD FOR ALL TEST FUELS
ENGINE 2	1200 R.P.M.	94 P.S.I. B.M.E.P.	
ENGINE 3	4500 R.P.M.	75 P.S.I. B.M.E.P.	

ENGINES A, B, C, D AND E – 13 MODE CYLCE

3309KT

FIG. No. 40
Drg. No. S.8037
Date Sept. 80

The Influence of Alternative Fuels on Smoke Limited BMEP



The Performance of a Swirl Chamber IDI Engine when Operating Under Simulated Turbocharged Conditions on Various Heavy Fuel Oils.

ENGINE Single cylinder, "5 Bore x 5·5" stroke, 4 stroke, Swirl chamber indirect injection.

FUELS

Optimum injection °E 500 rev/min.	Optimum injection °E 1250 rev/min.	Fuel	Cetane No.	Distillation-Recovery % at 572°F	Distillation-Recovery % at 662°F	S.G. 60/60°F
14·0	15·5 ————	1. Pool gas oil.	–	–	–	–
16·0	15·5 - - - - - - -	2. Light industrial 'A' diesel fuel.	40	40·5	80	0·893
17·0	16·0 ············	3. Balik' Papan diesel fuel.	34	50	74	0·916
16·0	16·0 -·-·——	4. Venezuelan 'B' grade diesel fuel.	38	27	51	0·915
120	16·5 — — — —	5. Iranian marine diesel fuel.	43	48	72	0·873
14·0	16·5 —·—·—	6. Venezuelan Admiralty fuel oil.	–	–	–	0·945

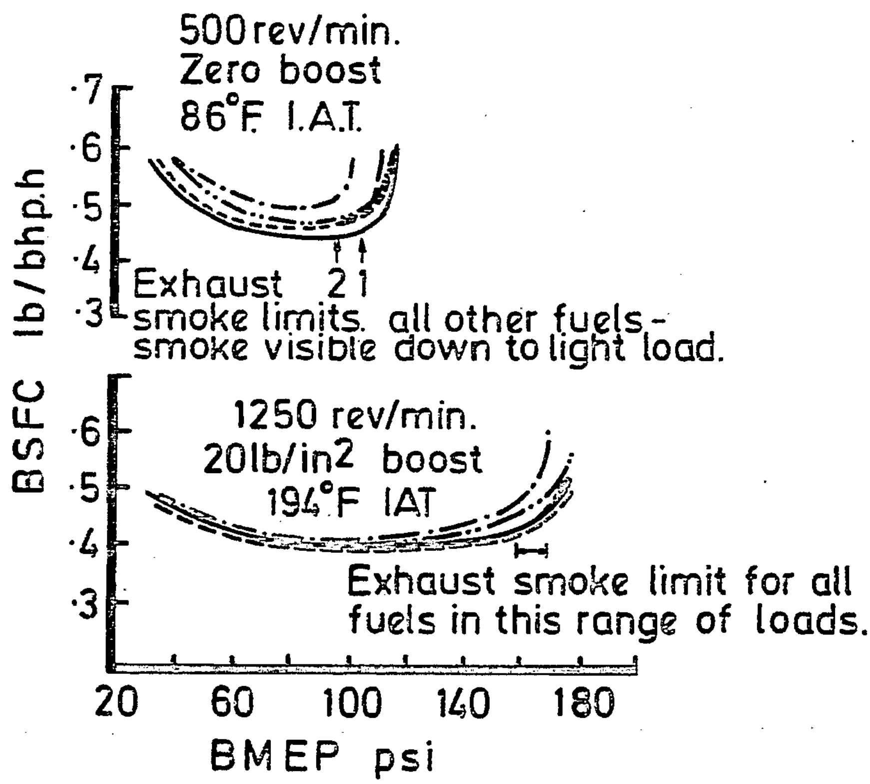


FIG. No. 42
Drg. No. S. 8131
Date Sept. 80

Comparative Performance of a 4-Cylinder 236 CID Swirling Direct Injection Engine on Kerosene and Diesel Fuel, Injection Timing Fixed, – Rack Adjusted for Equal Full Load Performance.

FUELS	Cetane Index	Distillation Range °F	Aromatic Content %vol.	S.G. 60/60°F
———— Diesel	55	370 – 688	28·3	0·838
- - - - - - Kerosene	56	315 – 493	7·7	0·782

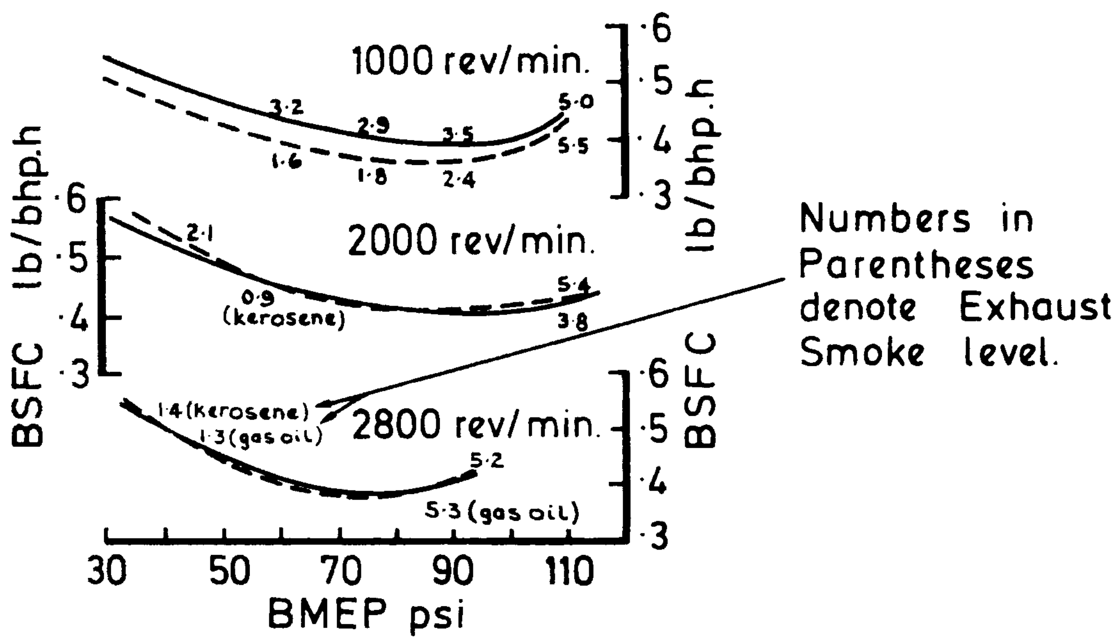
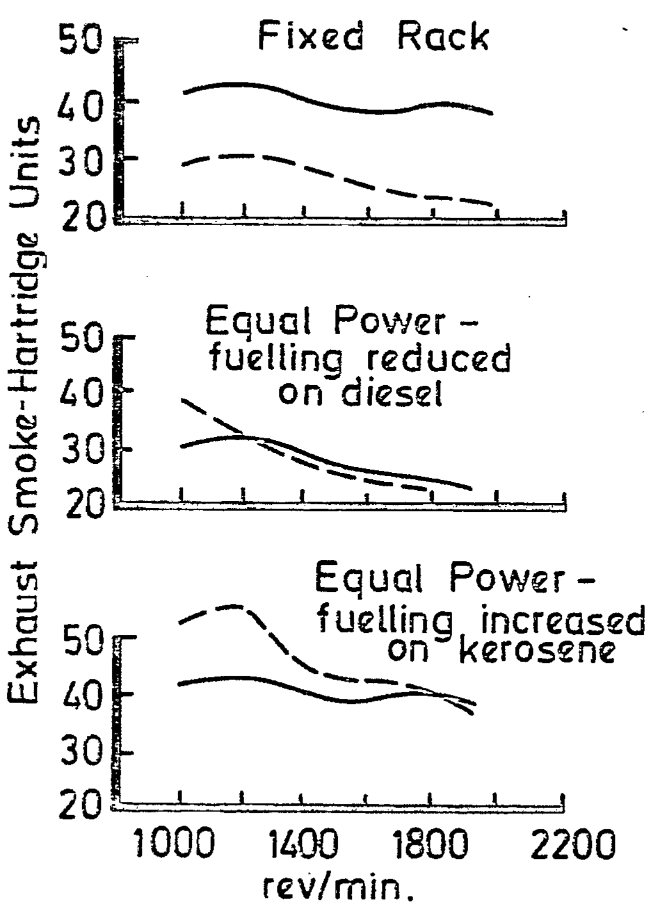


FIG. No. 43 (ref 10)
Drg. No. S.8038
Date Sept. 80

Comparison of Full Load Exhaust Smoke when using Kerosene and Diesel Fuels in a Direct Injection Engine.



ENGINE

6 Cylinder, 673 CID
Direct Injection Engine.

FUELS

	Diesel (———)	Kerosene (- - - - -)
Cetane No.	52	44
A.P.I. gravity	36·6	47·7
Distillation Range °F	389-650	317-468

Comparison of Exhaust Smoke over the Load Range when using Gasoline and Diesel Fuels in Two Direct Injection Engines.

——— Diesel Fuel. - - - - - 80 M.O.N. Gasoline.

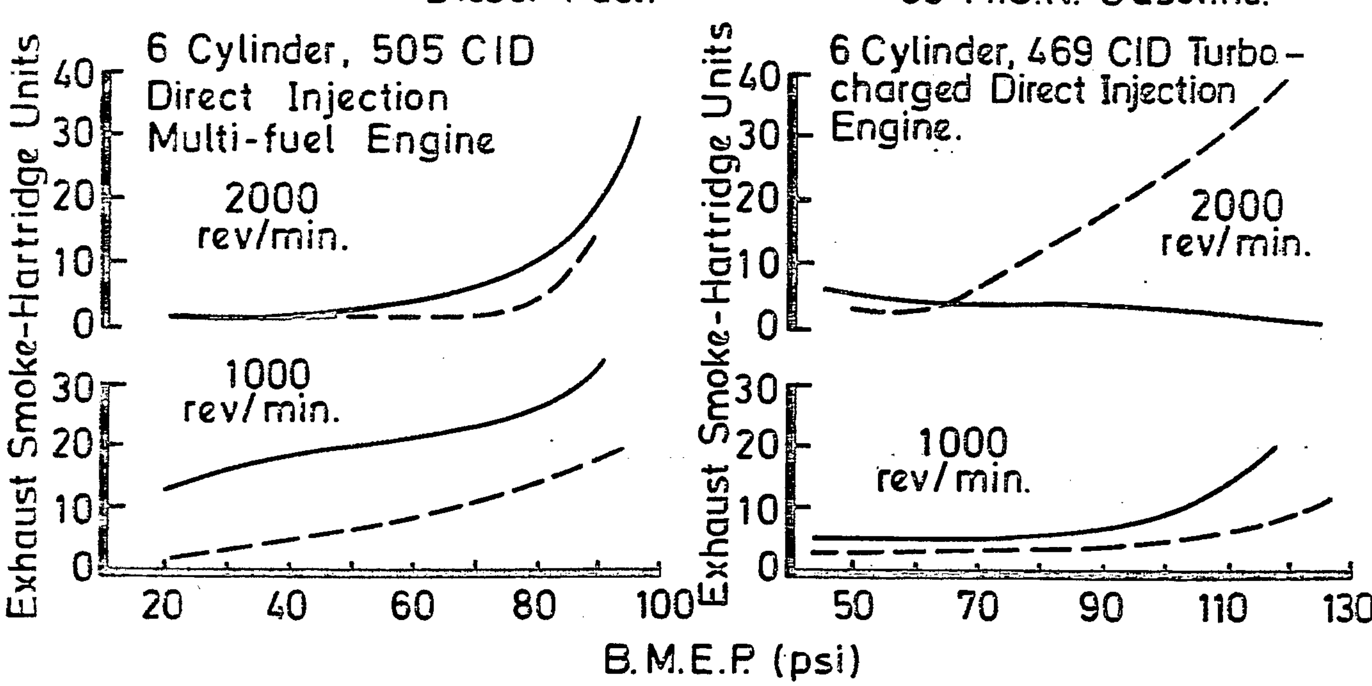
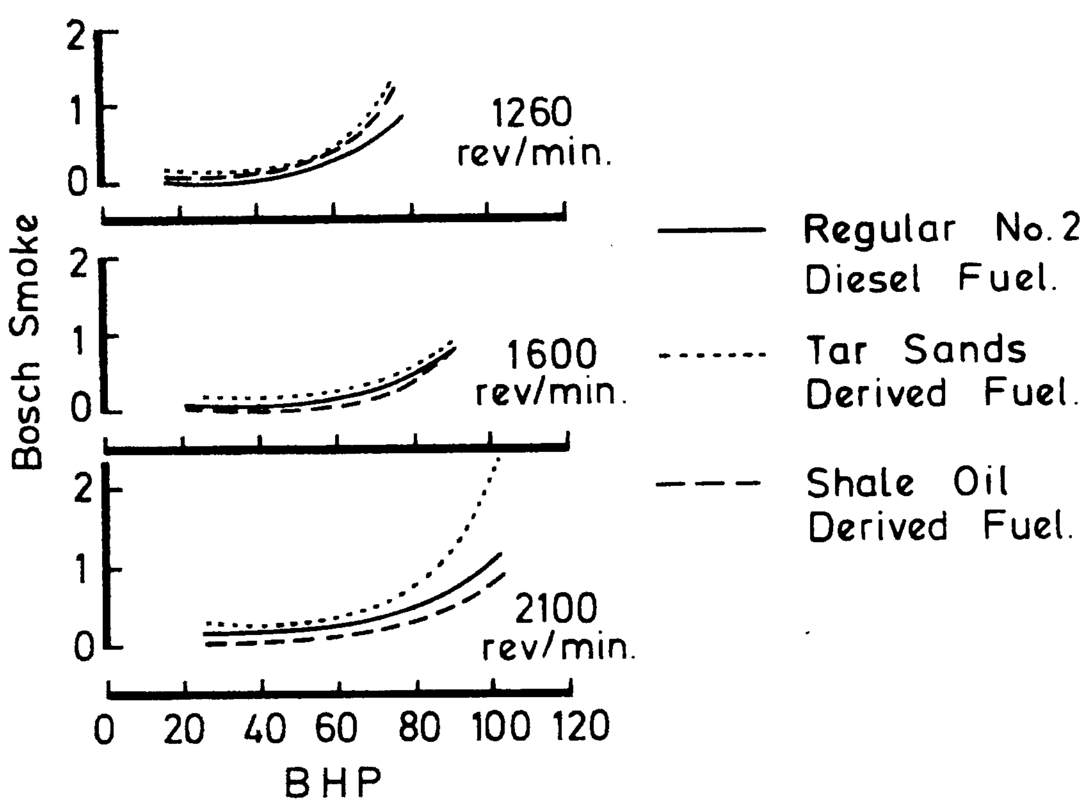


FIG. No. 44 (ref 53)
Drg. No. S. 8039
Date Sept. 80

Exhaust Smoke over the Load Range Comparing Shale and Tar Sands Derived Diesel Fuels with Regular Diesel Fuel.

ENGINE Detroit Diesel, 213 CID 3-Cylinder, 2 Cycle, Naturally Aspirated, Direct Injection, C.R. 18·7:1, N-60 Needle Valve Injectors, Rated 100 HP at 2100 rev/min. Fixed start of injection-18·6° Crank BTDC.

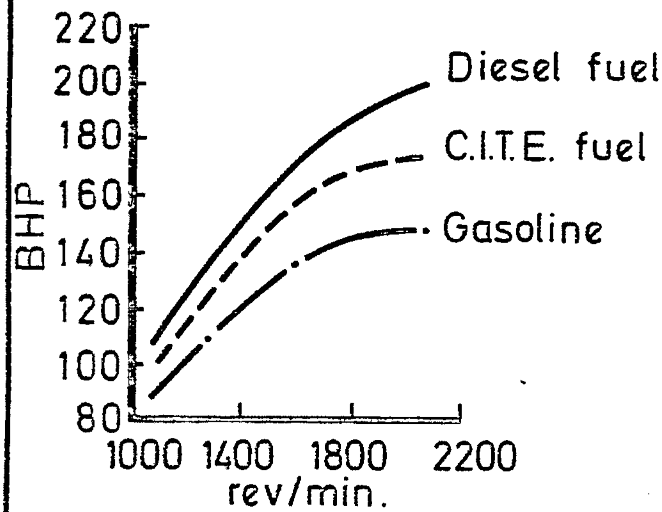
FUELS	Tar Sands Derived	No. 2	Shale Oil Derived (Marine)
Cetane No.	36·8	43	52·2
Distillation °F			
10%	446	425	533
50%	555	509	594
90%	630	605	656
Viscosity cS at 100 °F.	4·35	2·50	5·58
Aromatics % wt	43·6	42·6	33·7
°API	28·9	34·1	32·9



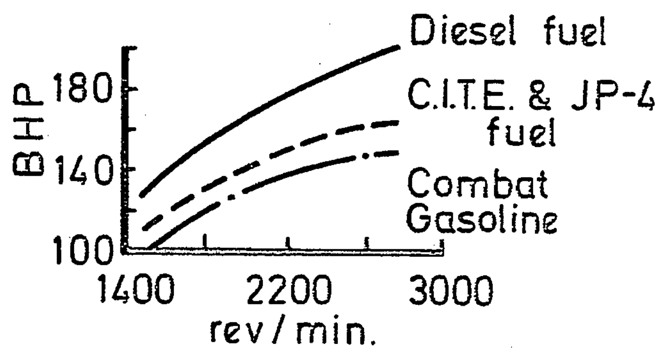
6983LT/8431

FIG. No. 45
Drg. No. S. 8029
Date Sept. 80

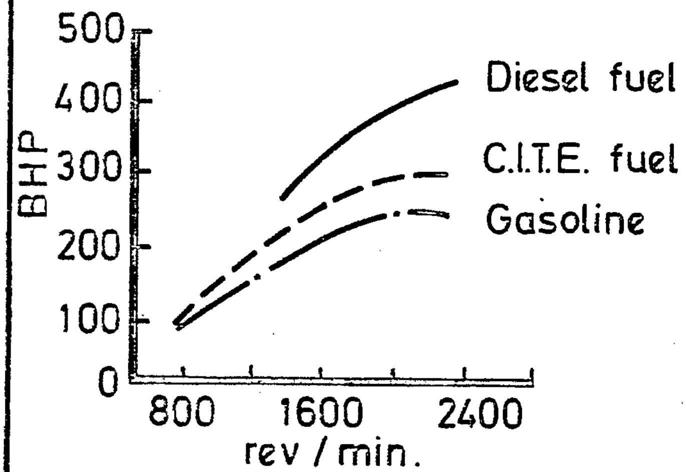
The Influence of Fuel Type on Full Load Power at Fixed Rack Position for Four Engines.



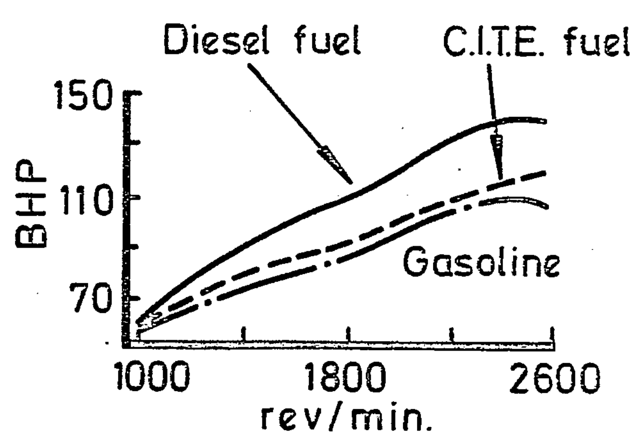
Mack ENDT 673
Turbocharged Direct
Injection – Swirling



G.M. 6V-53
Direct Injection – Quiescent



Caterpillar LDS-750
Turbocharged Pre-chamber



Continental LDS-427
Direct Injection 'M' System

FIG. No. 46
Drg. No. S.8032
Date Sept. 80

Comparative Performance of a 6-Cylinder 300 CID Swirling Direct Injection Engine on Gasoline and Diesel Fuel.

——————— Diesel Fuel, C.50-55 Cetane No. Injection timing 23° BTDC (standard diesel)

—— · —— 82 Octane Gasoline, C.18-24 Cetane No. Injection timing 32° BTDC.

- - - - - - 86 Octane Gasoline, C.15-22 Cetane No. Injection timing 32° BTDC.

JV – Just visible smoke.

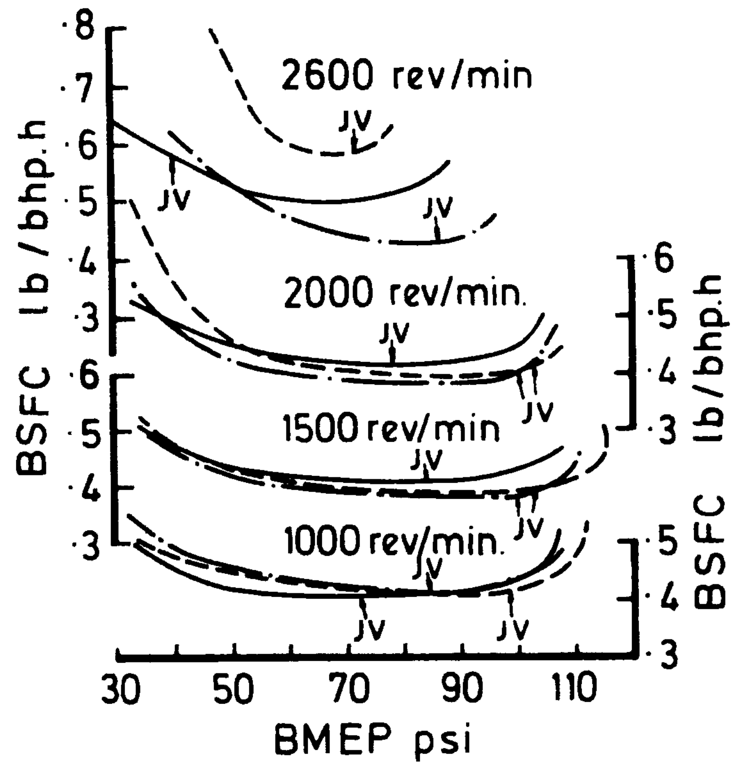
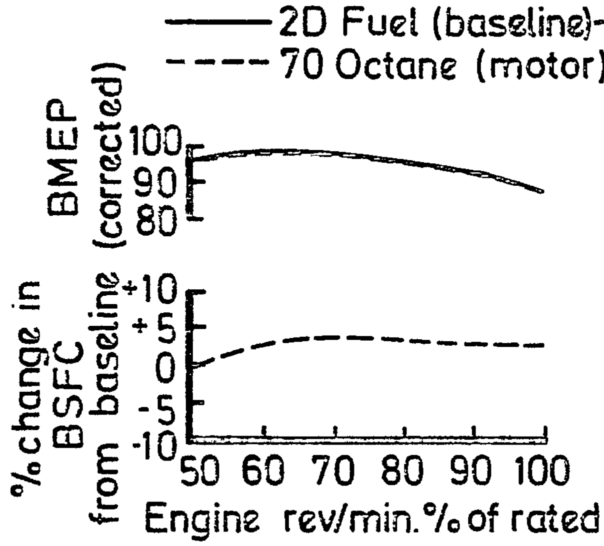
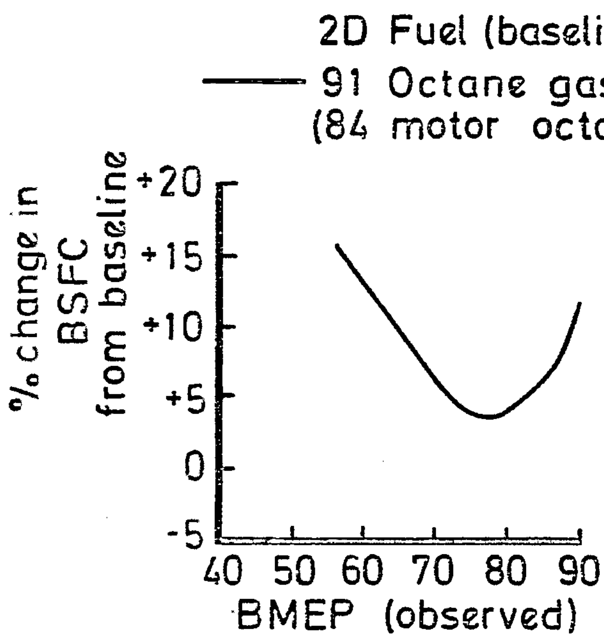


FIG. No. 47 (ref 66)
Drg. No. S. 8033
Date Sept. 80



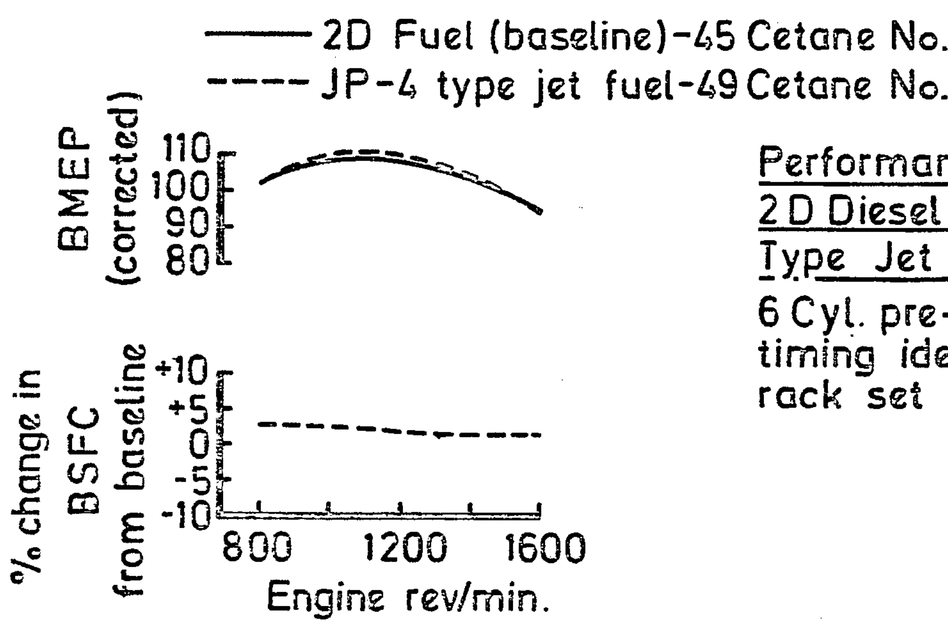
Performance Comparison 2D Diesel Fuel VS 70 Octane Gasoline.

21:1 Compression ratio, pre-combustion chamber, injection timing identical for each fuel, engine fully equipped, rack set to specified HP curve on each fuel.



Variable Load Fuel Consumption Comparison 2D Diesel Fuel VS 91 Octane Gasoline.

6 Cyl. pre-combustion chamber 1600 rev/min. 15:1 compression ratio Injection timing advanced 22° crank for gasoline.



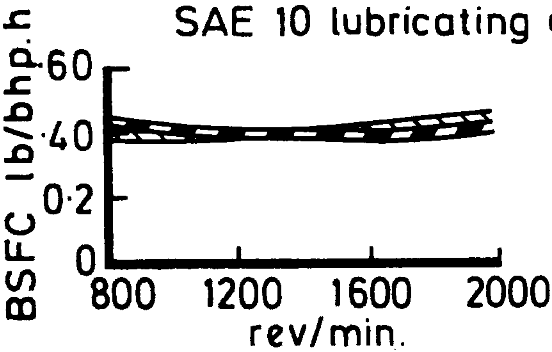
Performance Comparison 2D Diesel Fuel VS JP-4 Type Jet Fuel.

6 Cyl. pre-combustion chamber timing identical for each fuel rack set for equal power.

FIG. No. 48
Drg. No. S. 80 34
Date Sept. 80

Scatter of performance for Specifically Developed Multi- Fuel Engines. Automatic Compensation of Rack Stop for Fuel Density / Viscosity

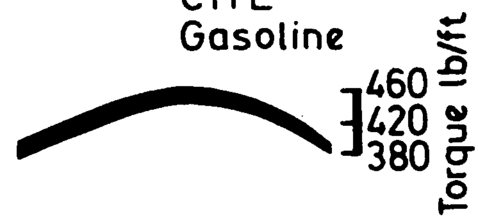
FUELS Diesel fuel
86-96 octane gasoline
JP 3/4
Kerosene
SAE 10 lubricating oil.

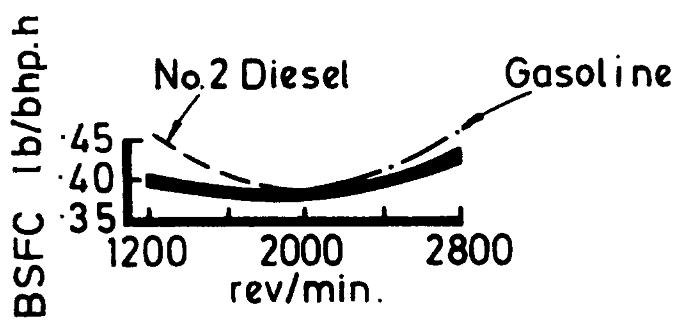


Mercedes - Benz
Pre-chamber Indirect Injection
C.R. 20+:1

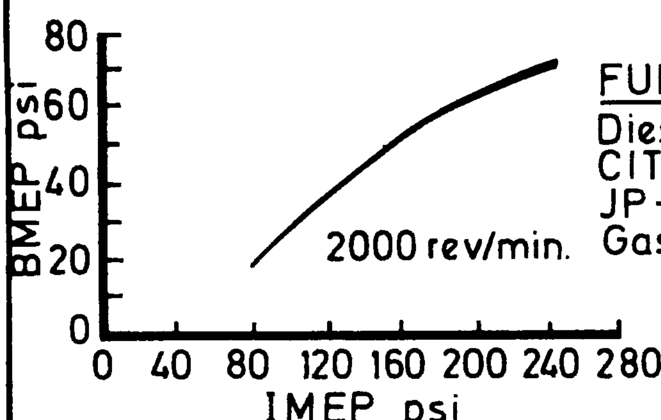
MAN D1246
Direct Injection ˋMˊ System C.R. 20+:1

FUELS No.2 Diesel, No.1 Diesel
CITE
Gasoline



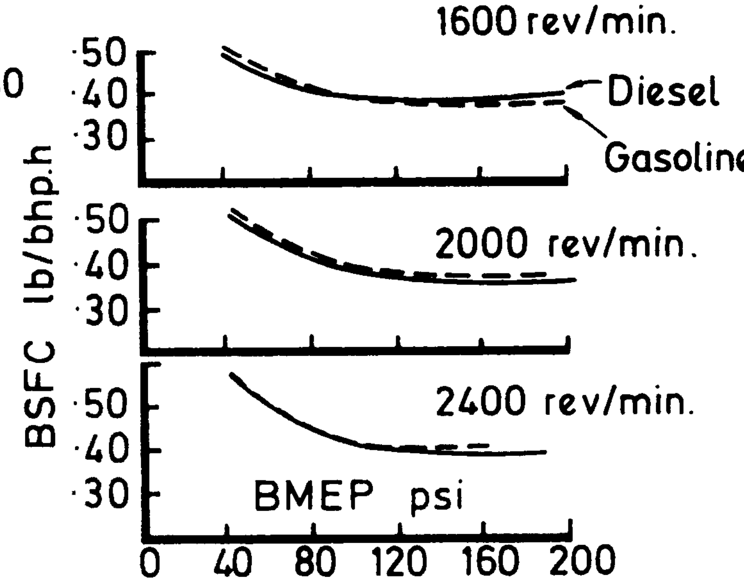


Continental - LDS 465
Direct Injection ˋMˊ System
C.R. 20:1



FUELS
Diesel
CITE
JP-4
Gasoline

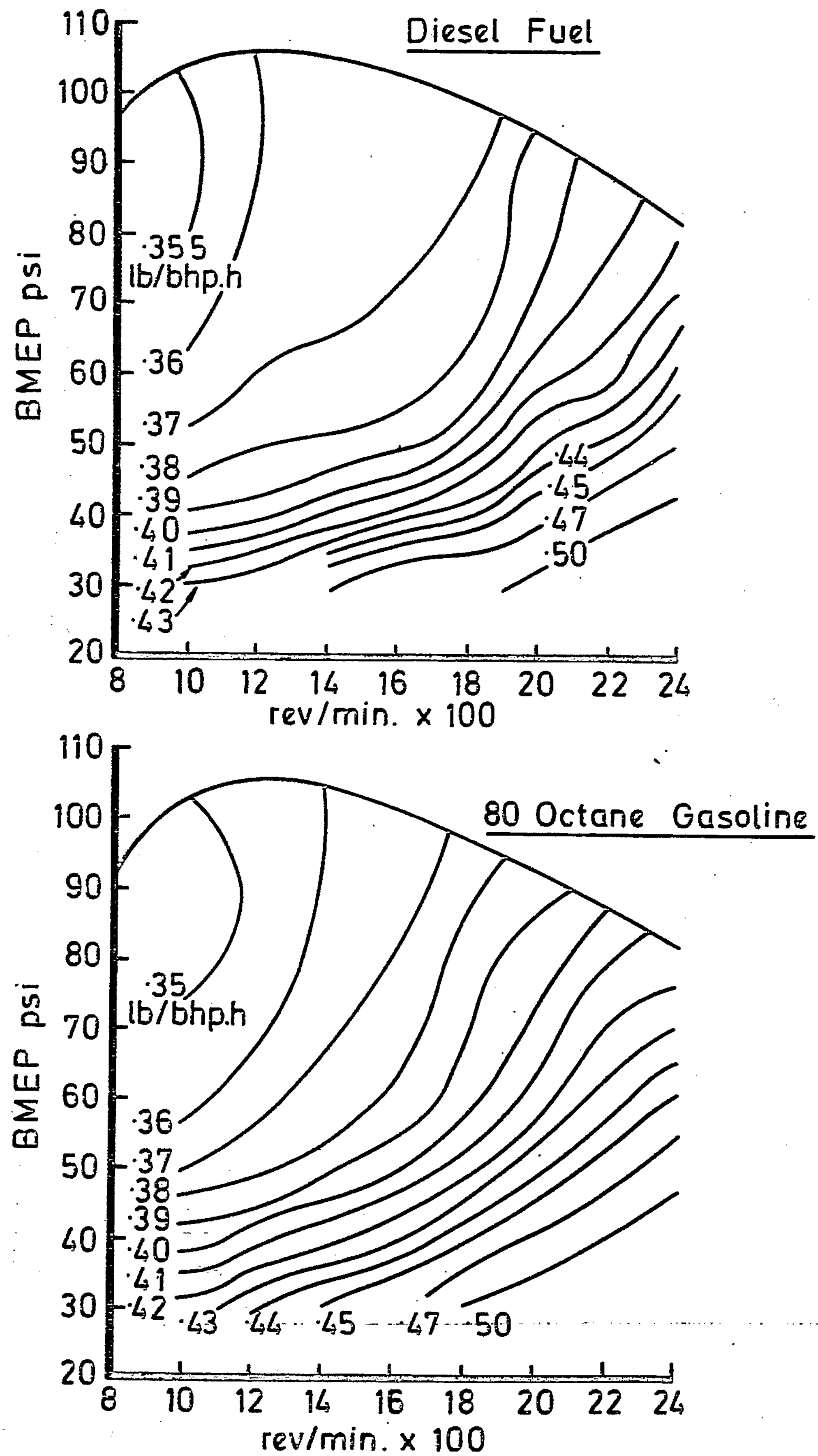
Lycoming S & M AVM 310
Direct Injection C.R. 18·7:1



Caterpillar LDS - 750
Pre-chamber Indirect
Injection (turbocharged)
C.R. 19·5:1

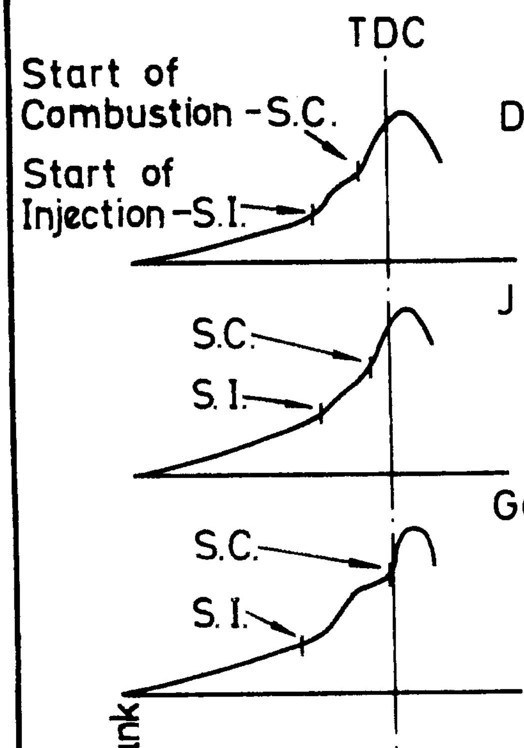
FIG. No. 49 (ref 89)
Drg. No. S. 8040
Date Sept. 80

Fuel Consumption Maps - Rootes 199 CID Opposed Piston 2-Stroke Engine



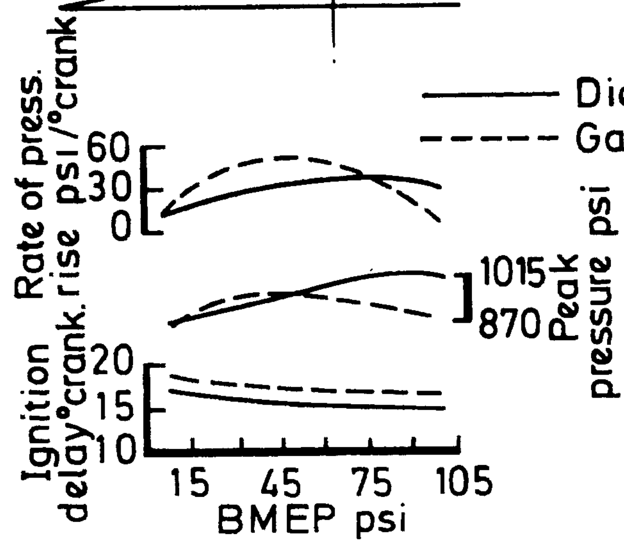
COMBUSTION CHARACTERISTICS WITH DIFFERENT FUELS

FIG. No. 50
Drg. No. S. 8036
Date Sept. 80



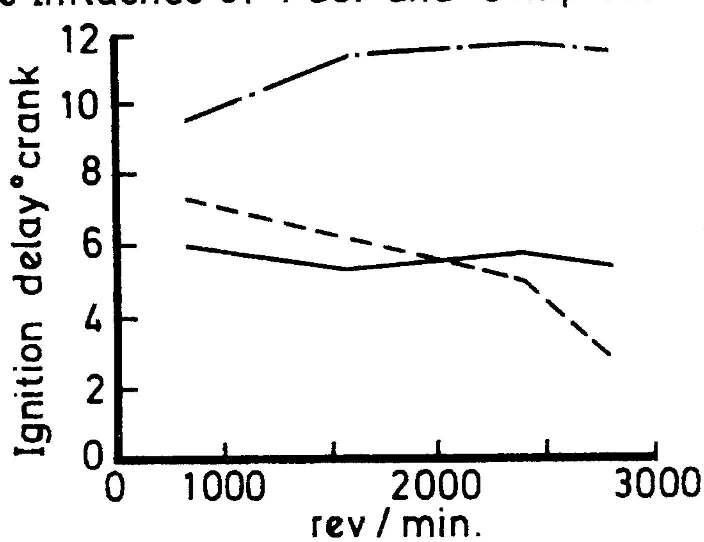
	Ignition Delay °Crank	Rate of Pressure Rise psi / °Crank	Peak Pressure psi
Diesel Fuel	16·4	29·1	727
JP-4	18·0	38·2	747
Gasoline	23·5	97	716

Deutz Pre-Chamber Indirect Injection Multi-Fuel Engine - Combustion Details for Three Fuels Running at Idle.



Mitsubishi Pre-Chamber Indirect Injection Engine - Combustion Details for Diesel Fuel and Gasoline over the Load Range at 2000 rev/min.
Fixed Injection Timing.

The Influence of Fuel and Compression Ratio on Ignition Delay



Normal Compression Ratio - Diesel Fuel.

Normal Compression Ratio - Combat Gasoline.

Raised Compression Ratio - Combat Gasoline.

6983LT/S431

FIG. No. 51 (ref 53)
Drg. No. S. 8035
Date Sept. 80

Relative BSFC with Shale and Tar Sands Derived Diesel Fuels with Respect to Regular Diesel Fuel. (No. 2 Diesel Fuel = 1·0)

ENGINE Detroit Diesel - 213 CID 3 Cylinder, 2 cycle, Naturally Aspirated, Direct Injection, C.R. 18·7 : 1 N-60 Needle Valve Injectors, Rated 100 HP at 2100 rev/min. Fixed start of injection - 18·6° Crank BTDC.

FUELS	Tar Sands Derived	No. 2.	Shale Oil Derived (Marine)
Cetane No.	36·8	43	52·2
Distillation °F			
10%	446	425	533
50%	555	509	594
90%	630	605	656
Viscosity cS at 100°F	4·35	2·50	5·58
Aromatics %wt	43·6	42·6	33·7
°API	28·9	34·1	32·9

---- Tar Sands Derived Fuel. —— Shale Oil Derived Fuel.

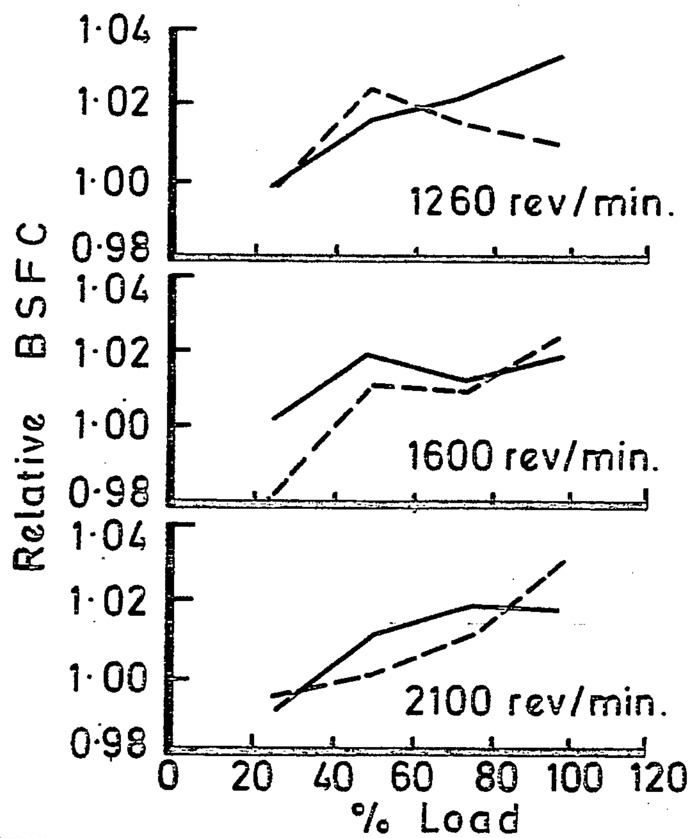
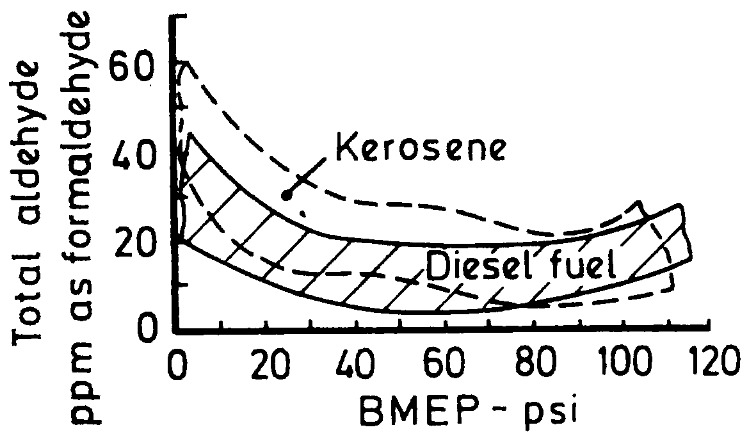
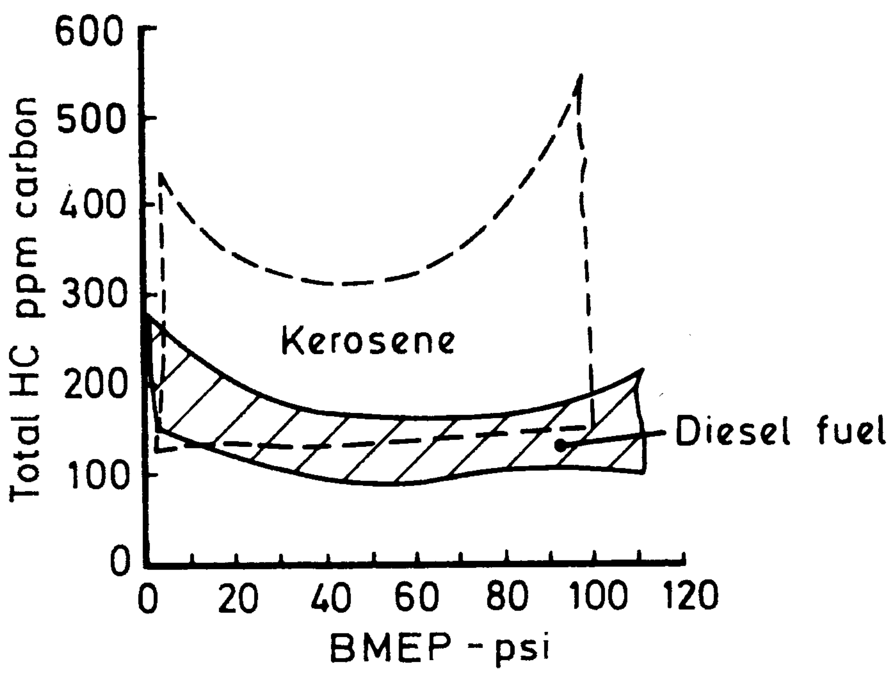


FIG. No. 52 (ref 17)
Drg. No. S.8026
Date Sept. 80

Aldehyde and Total H C Emissions Measured when using Kerosene and Diesel Fuels — Extremes of Results on Three, Four Stroke Direct Injection Engines — Speed Range 1000 rev/min. to Rated Speed.



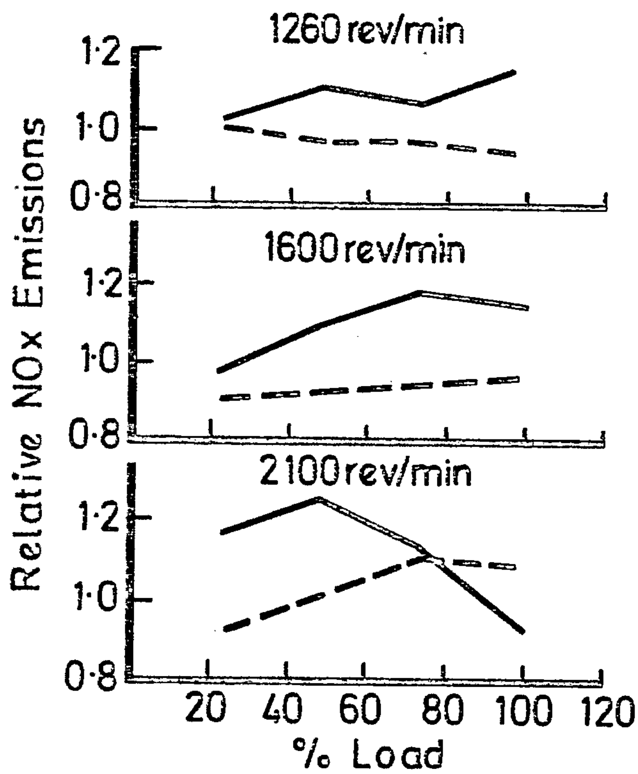


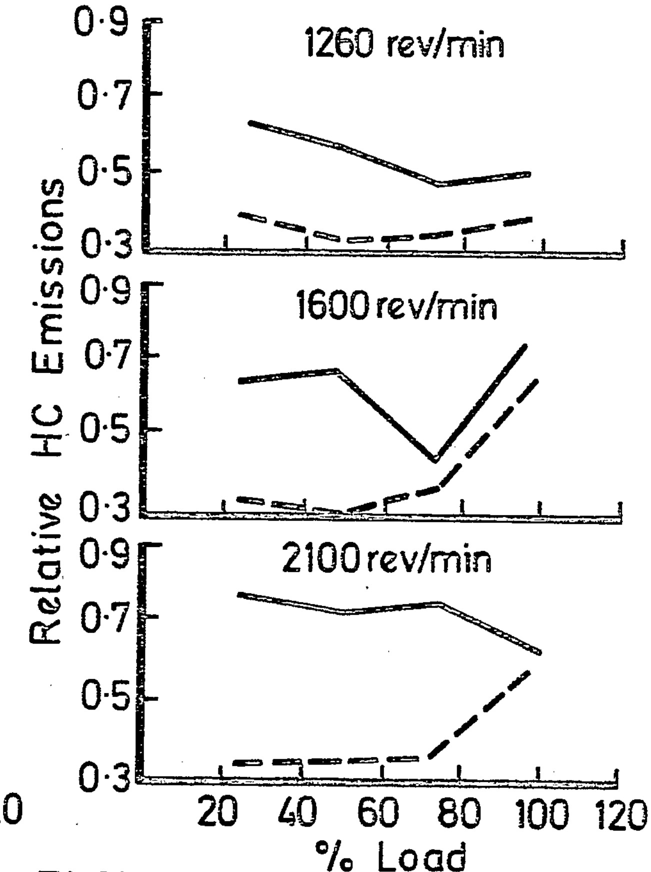
6983LT/8431

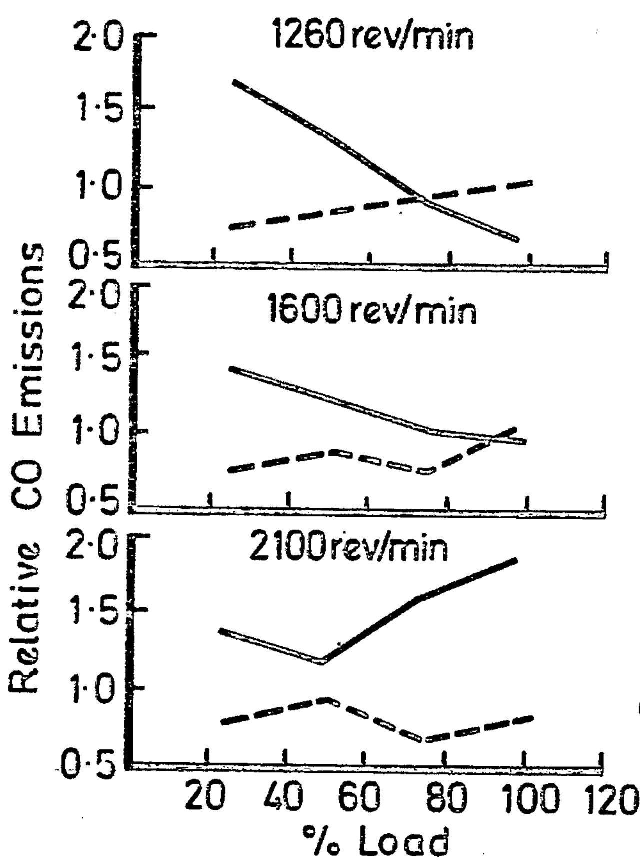
FIG. No. 53 (ref 53)
Drg. No. S.8027
Date Sept. 80

Relative Gaseous Emissions with Shale and Tar Sands Derived Diesel Fuels with Respect to Regular Diesel Fuel. (No.2. Diesel Fuel = 1·0)

——— Tar sands derived fuel.
- - - - Shale oil derived fuel.







ENGINE.

Detroit Diesel 213CID, 3 cylinder, 2 cycle, naturally aspirated, direct injection C.R. 18·7:1. N-60 needle valve injectors - rated 100 HP at 2100 rev/min. Fixed start of injection - 18·6° Crank BTDC.

FUELS.

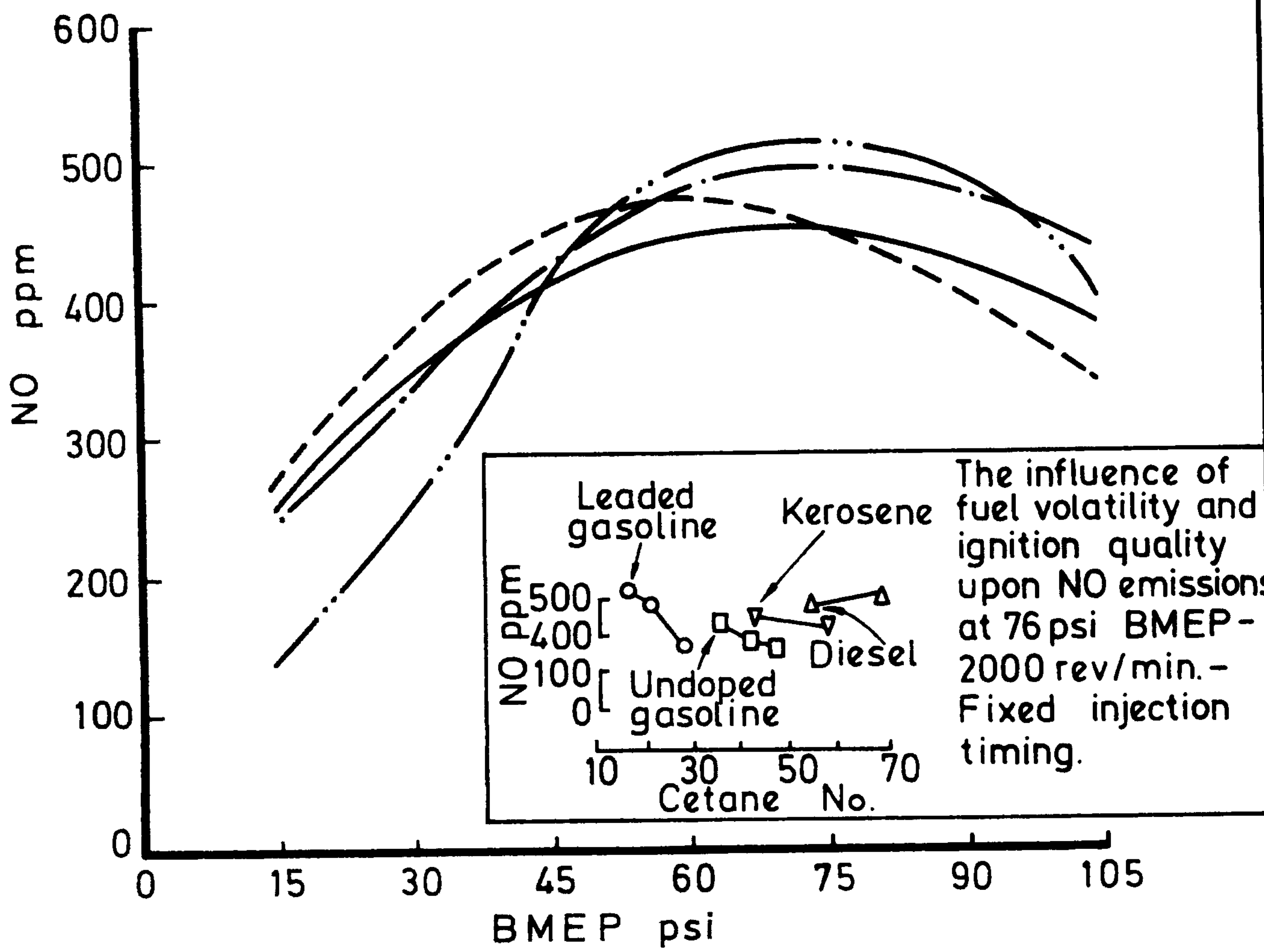
	Tar sands derived	No.2.	Shale oil derived (marine)
Cetane No.	36·8	43	52·2
Distillation °F			
10%	446	425	533
50%	555	509	594
90%	630	605	656
Viscosity cS at 100°F	4·35	2·50	5·58
Aromatics %wt	43·6	42·6	33·7
°API	28·9	34·1	32·9

FIG. No. 54 (ref 39)
Drg. No. S.8028
Date Sept. 80

The Influence of using Gasoline and Kerosene Fuels on NO Emissions over the Load Range at 2000 rev/min. - Pre-Chamber Engine.

ENGINE Single cylinder Mitsubishi DV-4 Pre-chamber
46 CID - 3·7" Bore x 4·33" stroke - C.R. 19:1
Injection timing 14° Crank BTDC.

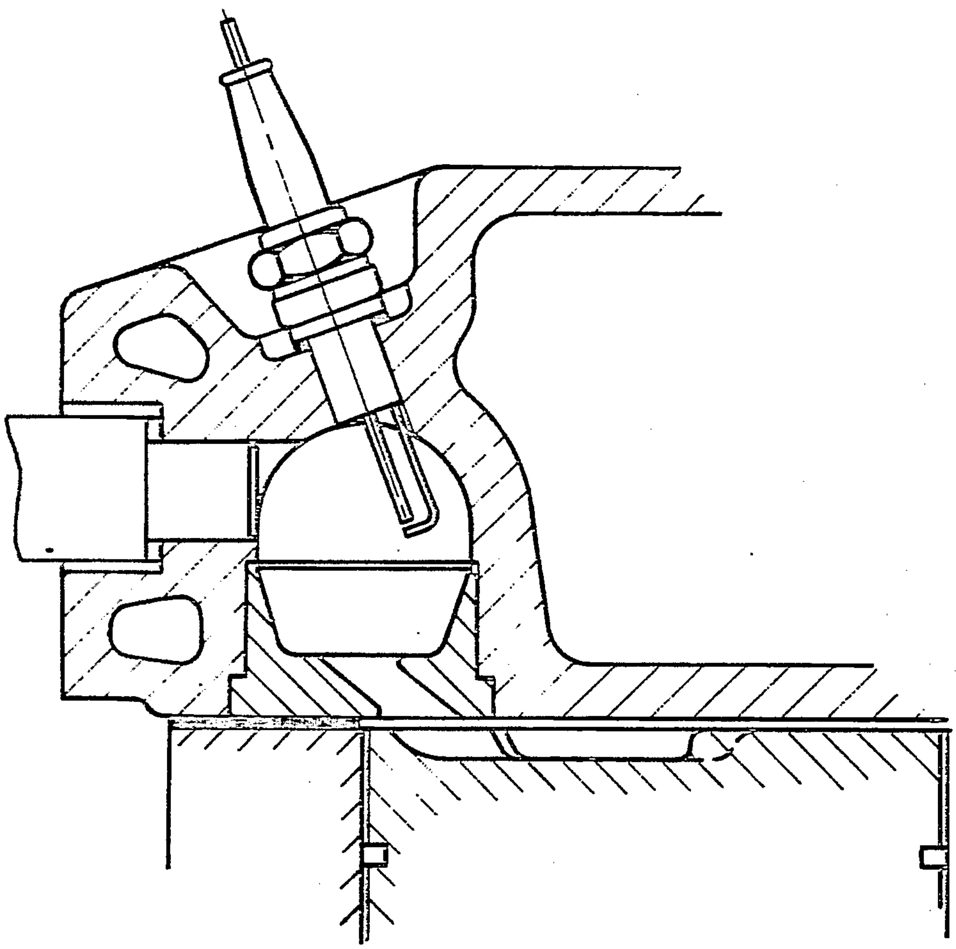
FUELS	— · — Diesel	——— Kerosene	- - - - Undoped Gasoline	—— ·· —— Leaded Gasoline + 5% heavy oil
Cetane No.	55	43	35	16
Distillation Range °F.	347-604	298-435	109-298	100 - 365 100 - 702 with 5% heavy oil added.



The influence of fuel volatility and ignition quality upon NO emissions at 76 psi BMEP - 2000 rev/min. - Fixed injection timing.

FIG. No. 55 (ref 108)
Drg. No. S 8421
Date APRIL '81

ARRANGEMENT OF COMBUSTION CHAMBER
SPARK IGNITED COMET V



6930LT/3491

FIG. No. 56 (ref 108)
Drg. No. D 44066
Date SEPT '80

Single Cylinder Comparisons – Indicated Fuel Consumption

ENGINE	Test speeds rev/min. Low	Medium	High
A. Spark Ignited Comet	1000	2000	4000
B. Gasoline Engine	1200	1800	4200
C. Comet Diesel	1000	2000	3000

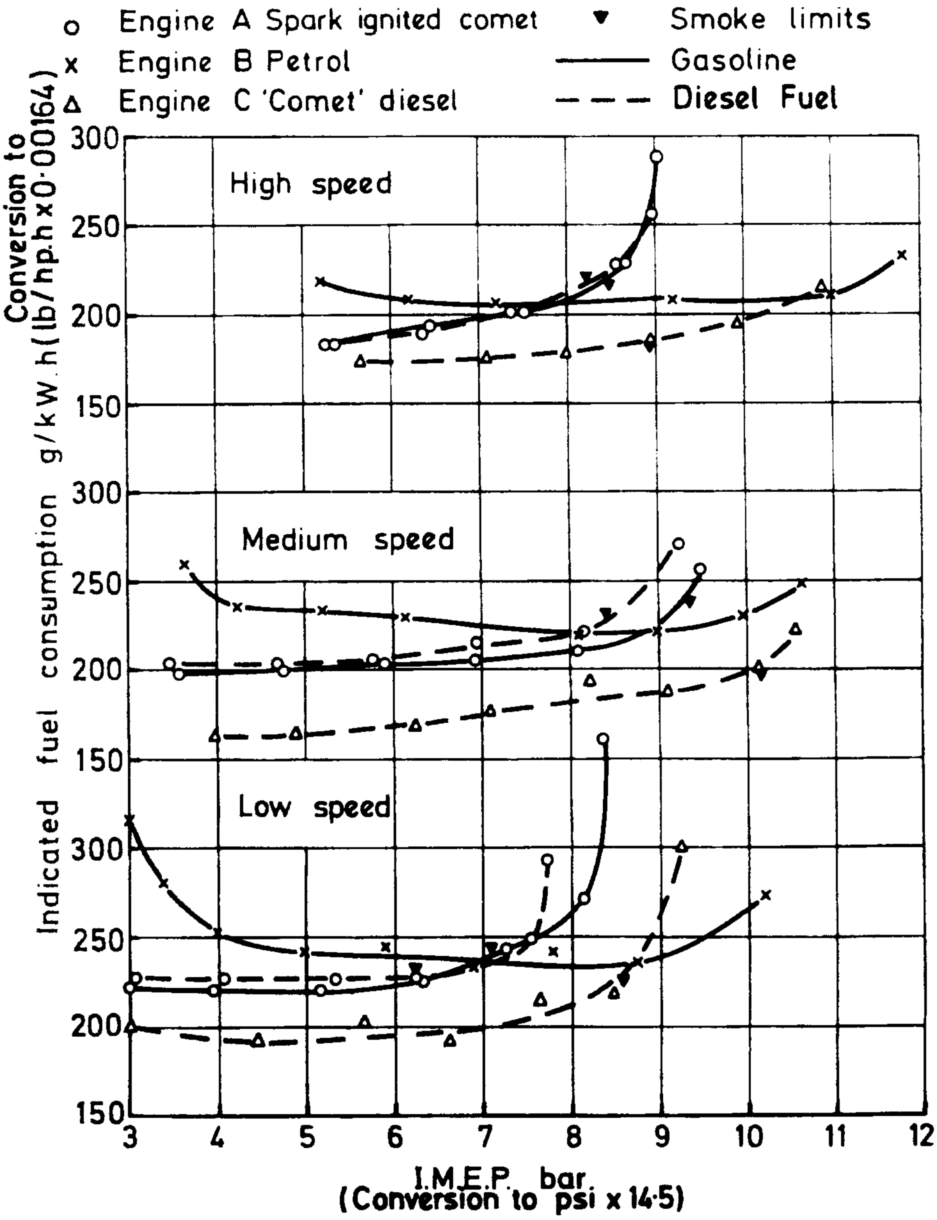


FIG. No. 57(ref 108)
Drg. No. D44068
Date SEPT 1980

Single Cylinder Comparisons — Indicated HC Emissions

ENGINE	Test speeds rev/min. Low	Medium	High
A. Spark Ignited Comet	1000	2000	4000
B Gasoline Engine	1200	1800	4200
C. Comet Diesel	1000	2000	3000

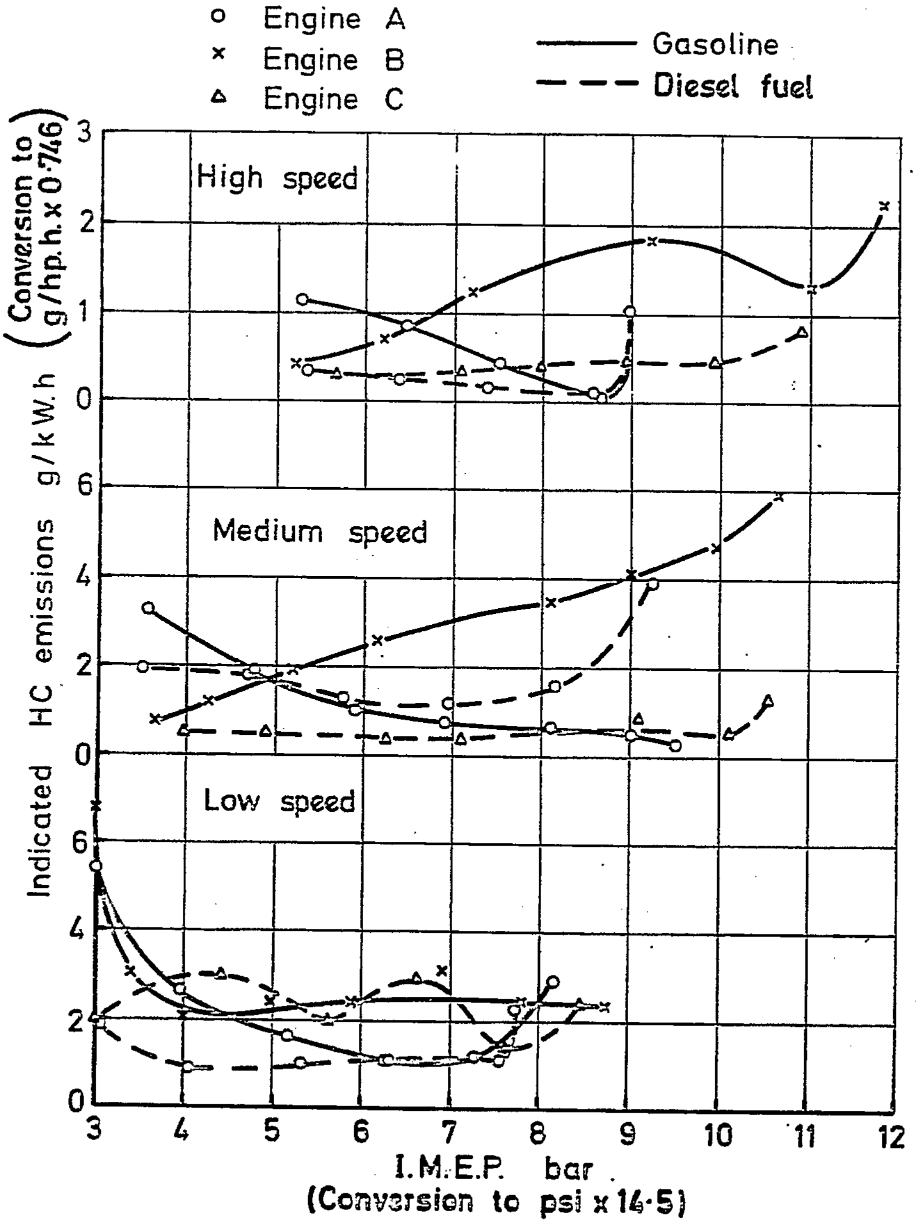


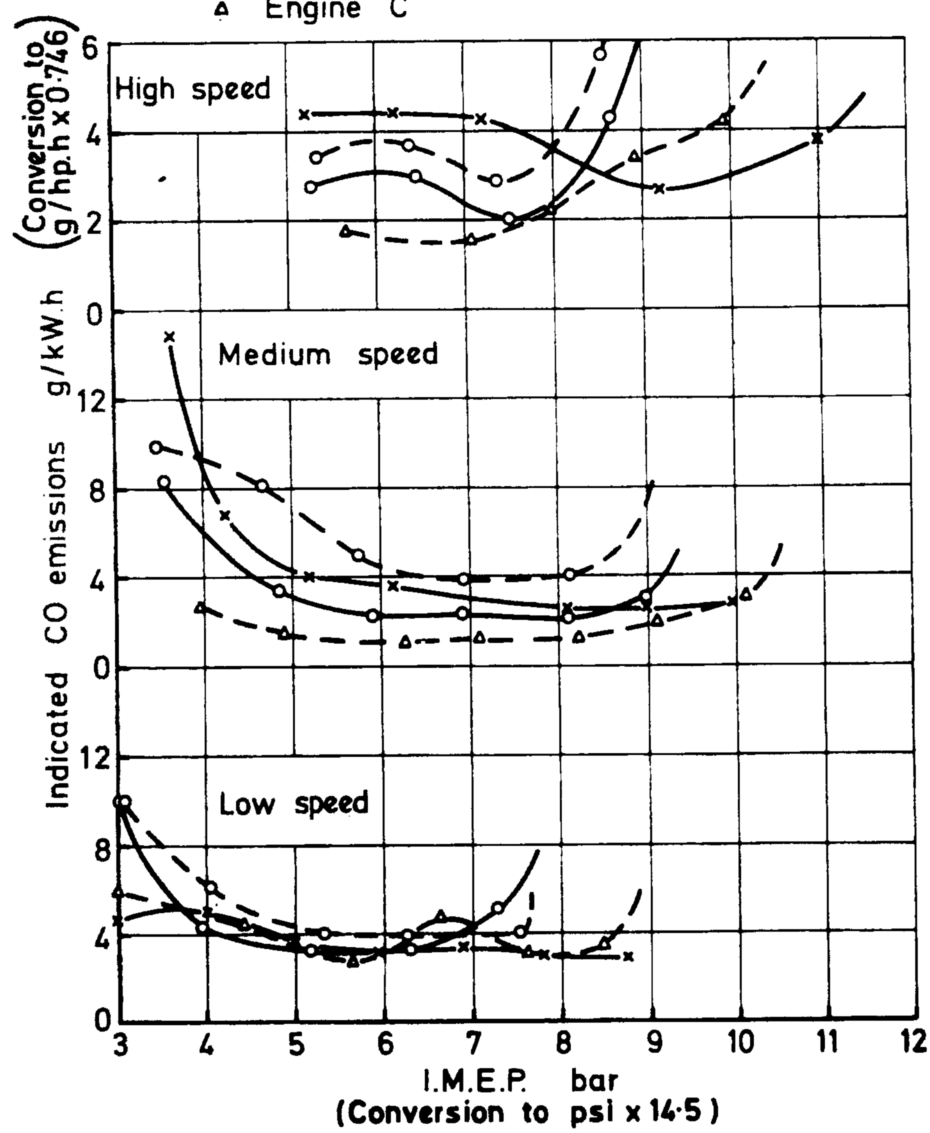
FIG. No. 58(ref 108)
Drg. No. D44070
Date SEPT '80

Single Cylinder Comparisons — Indicated CO Emissions

ENGINE	Test speeds rev/min. Low	Medium	High
A. Spark Ignited Comet	1000	2000	4000
B. Gasoline Engine	1200	1800	4200
C. Comet Diesel	1000	2000	3000

o Engine A
× Engine B
Δ Engine C

——— Gasoline
– – – Diesel Fuel



6963LT/8431

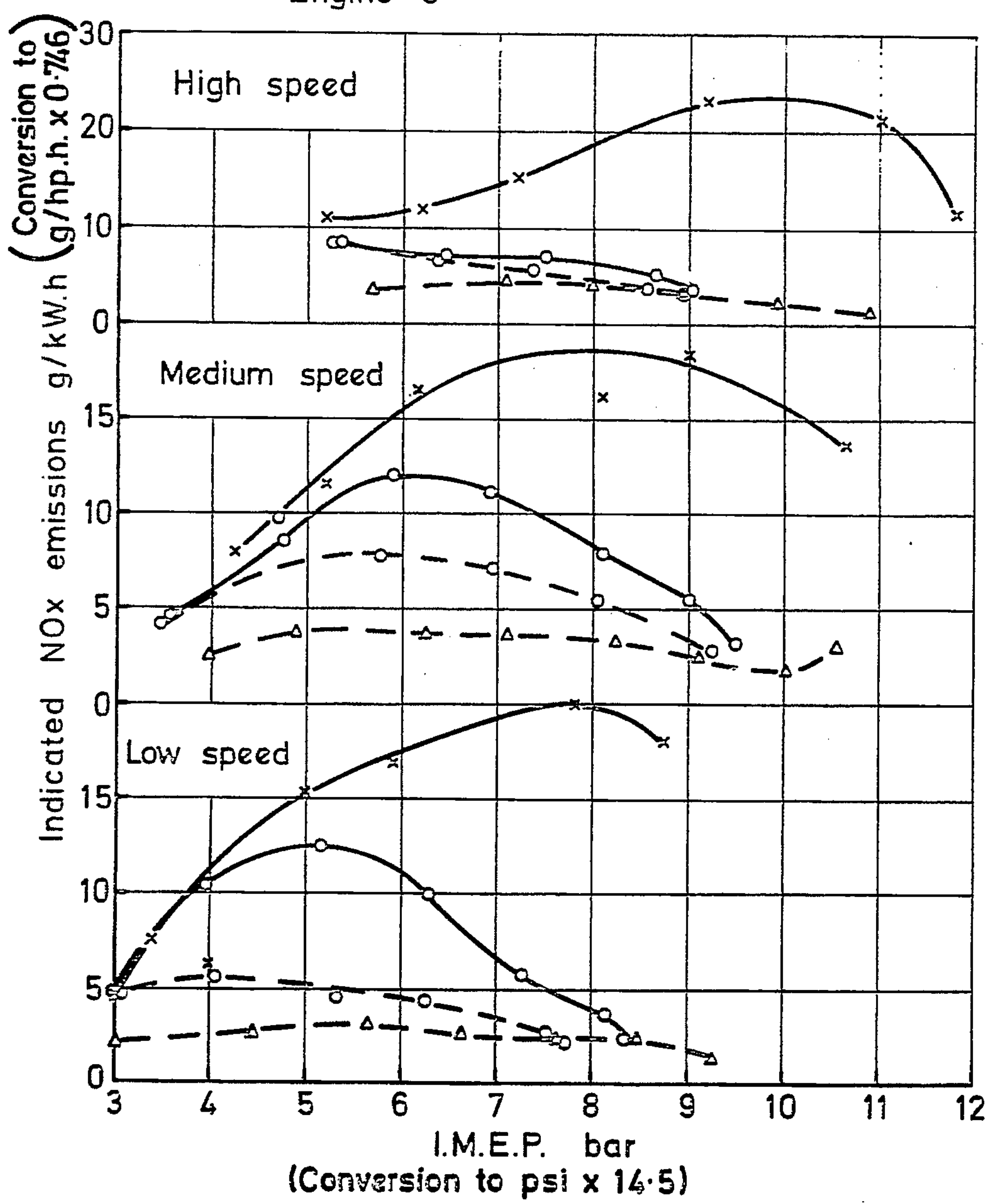
FIG. No. 59 (ref 108)
Drg. No. S8429
Date SEPT '80

Single Cylinder Comparisons – Indicated NOx Emissions

ENGINE	Test speeds rev/min. Low	Medium	High
A. Spark Ignited Comet	1000	2000	4000
B. Gasoline Engine	1200	1800	4200
C. Comet Diesel	1000	2000	3000

o Engine A
× Engine B
△ Engine C

——— Gasoline
– – – Diesel fuel



6993LT/3491

FIG. No. 60 (ref 108)
Drg. No. D44071
Date SEPT '80

Spark Ignited Comet – Response of Emissions and Fuel Economy to Retard, Exhaust Gas Recycle and Throttling.

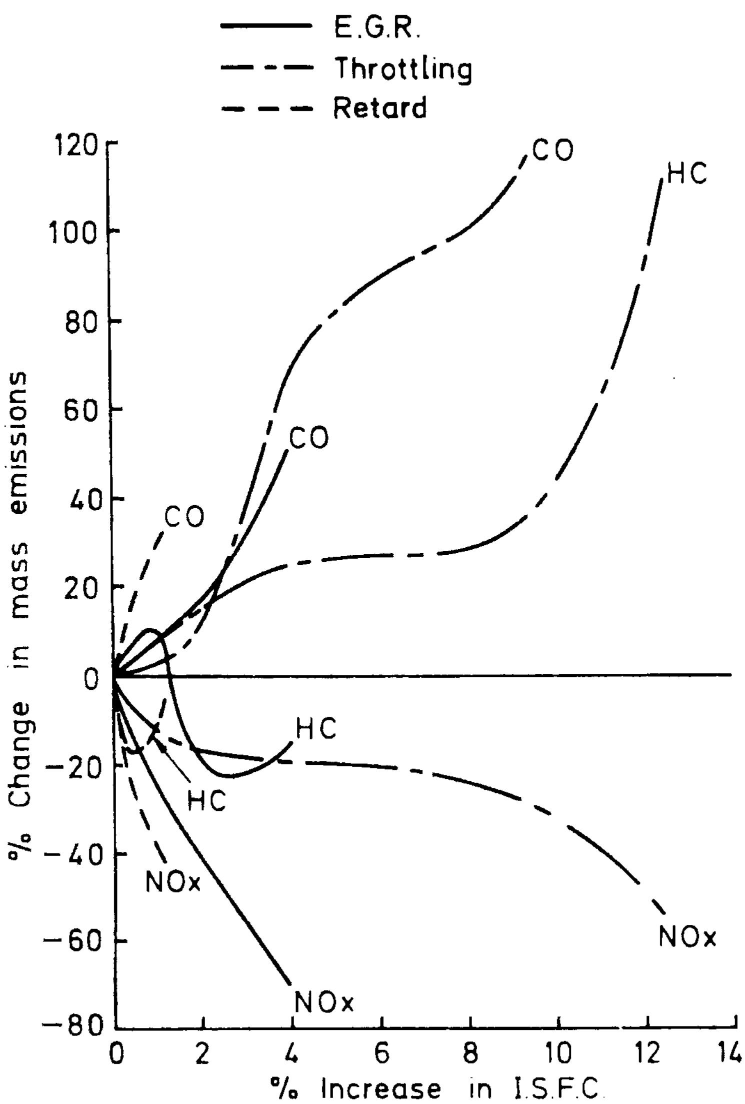


FIG. No. 61 (ref 108)
Drg. No. D44069
Date SEPT '80

Comparison of Combustion Noise

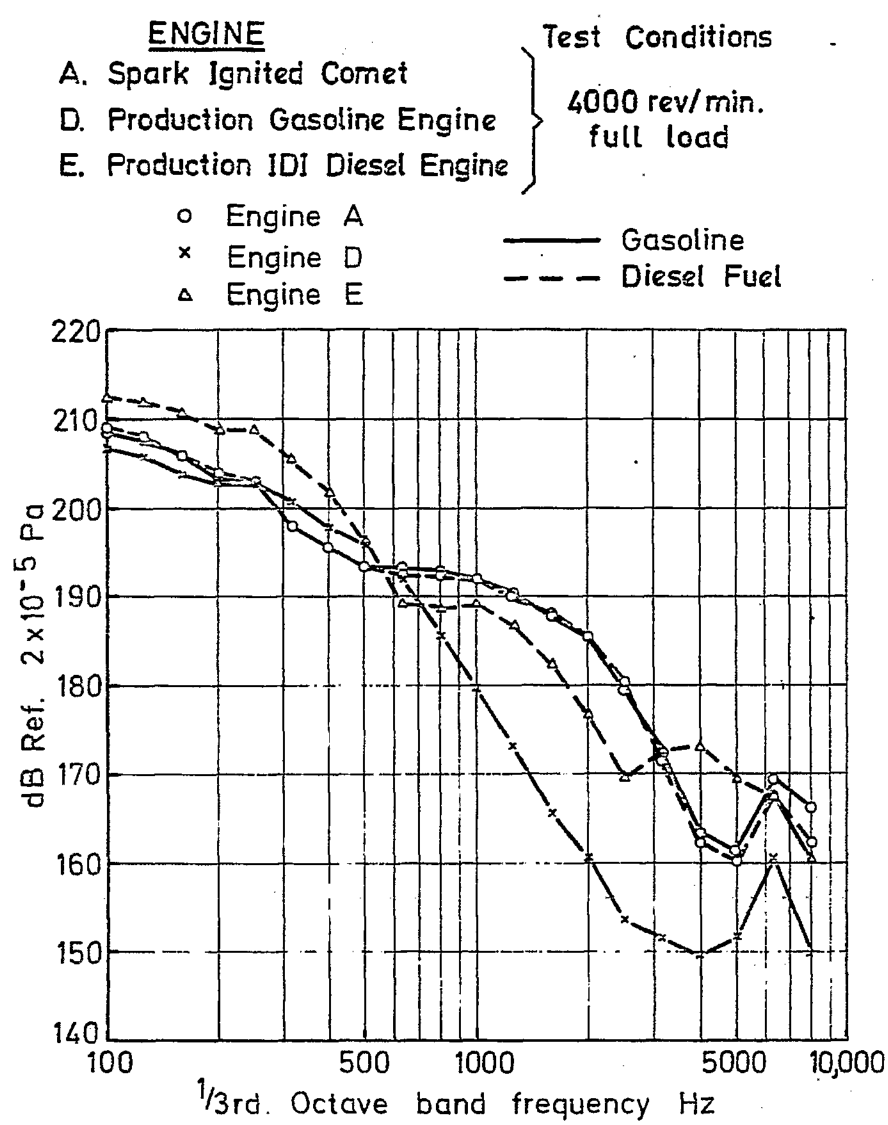
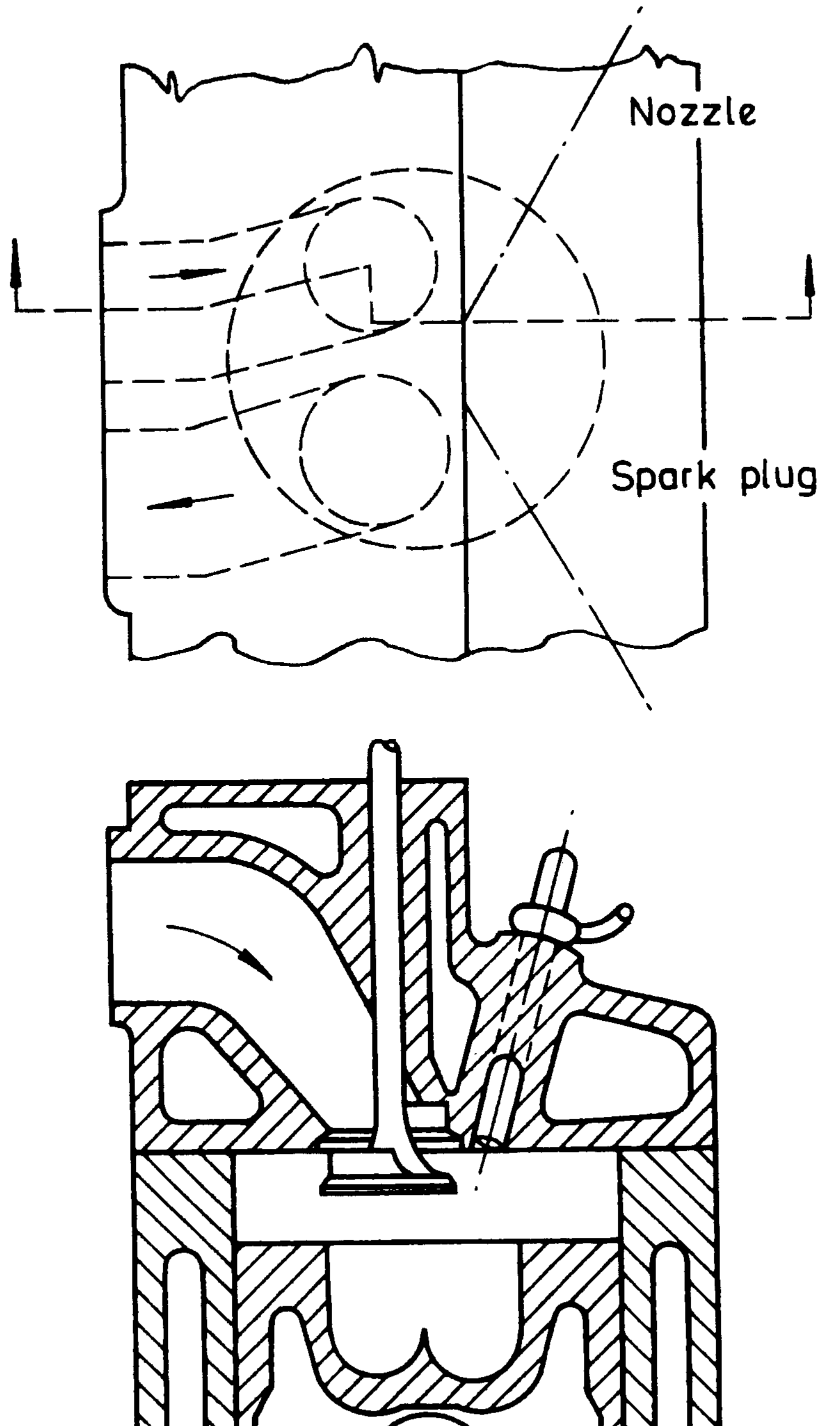


FIG. No. 62
Drg. No. S.8023
Date Sept. 80



Schematic Diagram of the TCCS Combustion Chamber

FIG. No. 63
Drg. No. S. 8024
Date Sept. 80

TCCS – Torque Curves

* ⊙ 147 Engine – Gasoline Optimised.

\+ Unleaded Gasoline
Δ 50/50 Gasoline/Diesel
X Diesel Fuel
} LIS-183 Engine - Multi-fuel (part developed) Smoke limit -10% opacity ≈ 2·3 Bosch

⊡ Diesel Fuel - L-163-S Engine - Multi-fuel
Smoke limit - 20% opacity ≈ 4 Bosch.

* • 144-HCC Engine – typical curve for gasoline, diesel or jet fuel.

* ◧ Gasoline - L-141 Engine - Multi-fuel with optimisation bias towards gasoline.

* ▲ Gasoline - UPS/TCCS 292 - Multi-fuel with optimisation (estimated performance) bias towards gasoline.

\- - - Envelope for developed, swirl chamber IDI light duty diesel engines – smoke limit, 2–3 Bosch low speed, 1·5–2 Bosch mid and high speed.

*Smoke limit just visible — Texaco standard.

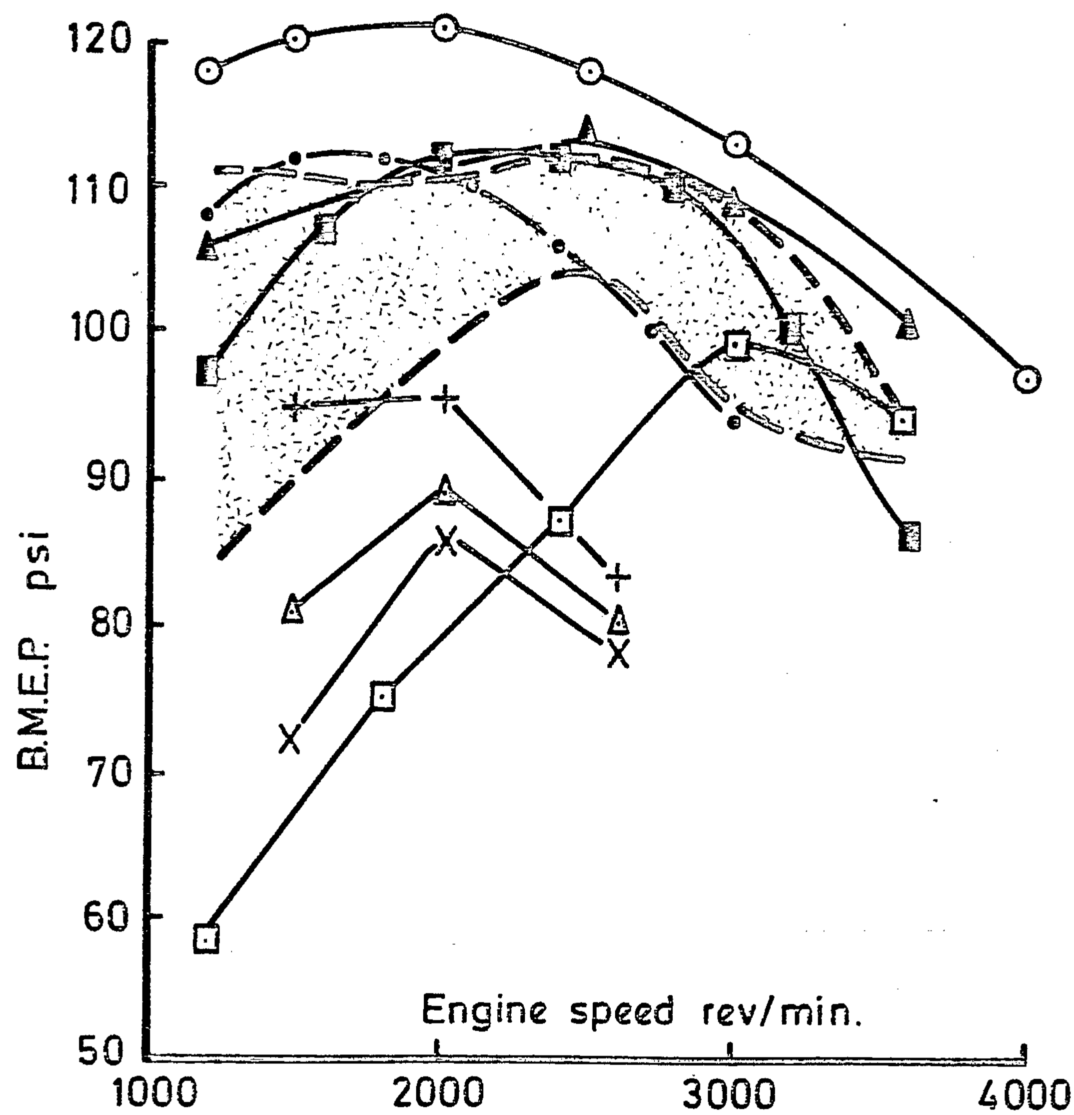


FIG. No. 64
Drg. No. S.8025
Date Sept. 80

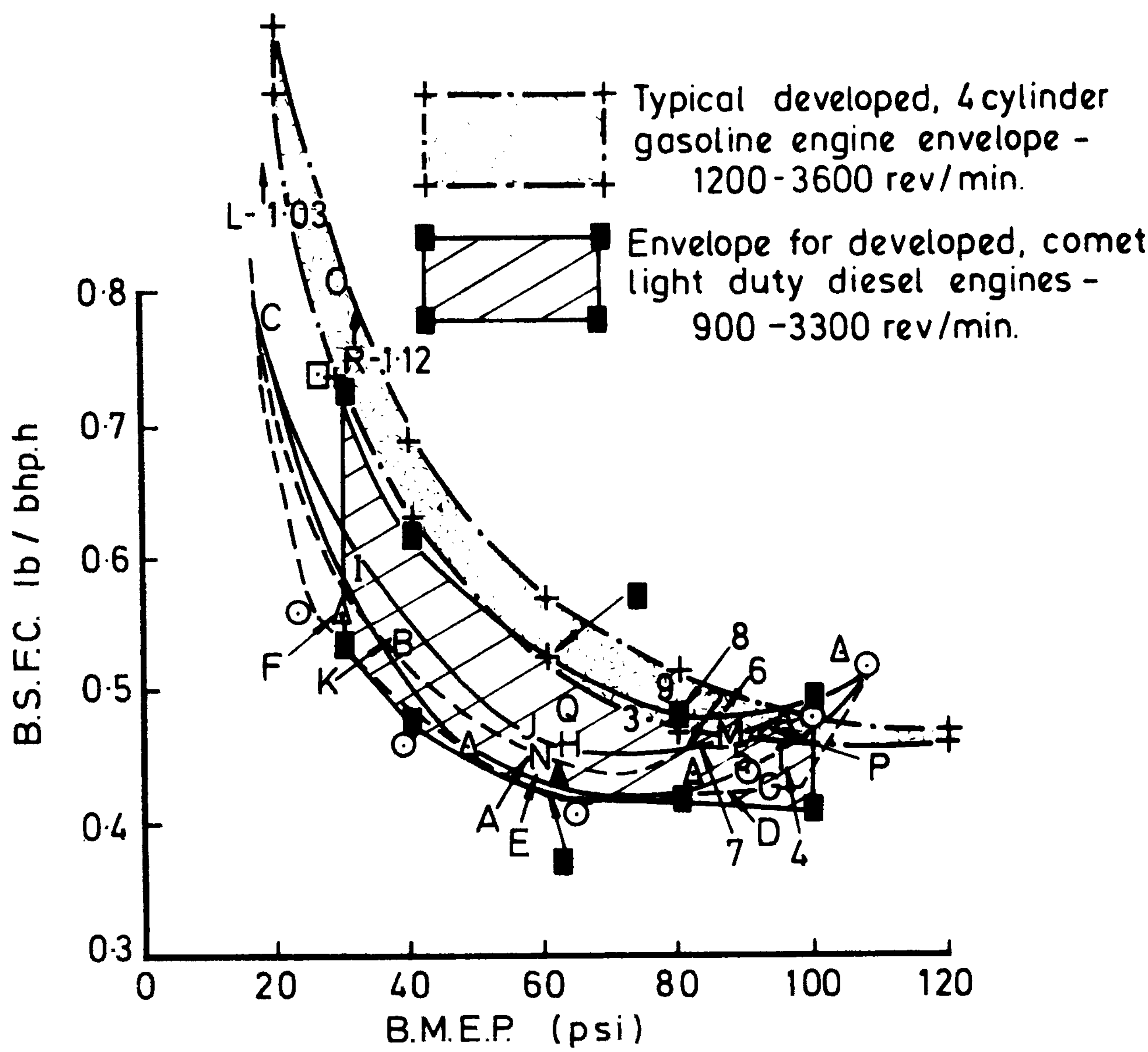
Fuel Consumption Comparisons for TCCS Multi-Fuel Engines, Diesel and Gasoline Engines

⊙ 1200 rev/min.
Δ 2400 rev/min.
} 144-HCC Engine – typical data for gasoline, diesel or jet fuel.

▭ 2800 rev/min.
⬚ 2000 rev/min.
} L-141 Engine – economy envelopes – all data gasoline, CITE and diesel fuels.

1500	2000	2600 rev/min.		
1	4	7	Unleaded Gasoline	LIS-183 (development not complete)
2	5	8	50 / 50 Gasoline / Diesel	
3	6	9	No. 2. Diesel Fuel.	

1200	2400	3600 rev/min.		
A,B,C.	D,E,F.	G,H,I.	Gasoline	L-163-S Engine
J,K,L.	M,N,O.	P,Q,R.	Diesel Fuel	



6983LT/8431

FIG. No. 65
Drg. No. S8428
Date SEPT '80

Urban Fuel Economy Comparisons For Gasoline Diesel and TCCS Vehicles TCCS Vehicles 1·5 – 2·0 g/mile NOx

Ref Source	Key	Vehicle	Engine	Fuel
117	◐	Jeep (M–151)	T/C L–141 TCCS	Gasoline
	+			Diesel
117	⊡	Cricket	T/C L–141 TCCS	Broadcut
	△			Gasoline
120	×	Gremlin	L–163–S TCCS	Gasoline
113	▽	Jeep (M–151)	L–163–S	Diesel
117	▽	Cricket	L–141 TCCS	Gasoline
119	(hatched box)	1978/79 IDI Diesel Vehicles		Certification Vehicles (49 States) Manual and Auto Transmissions
119	(stippled hatched box)	1979 Gasoline Vehicles		
	----	Diesel Lower Limit Expressed as MPG Gasoline Equivalent		

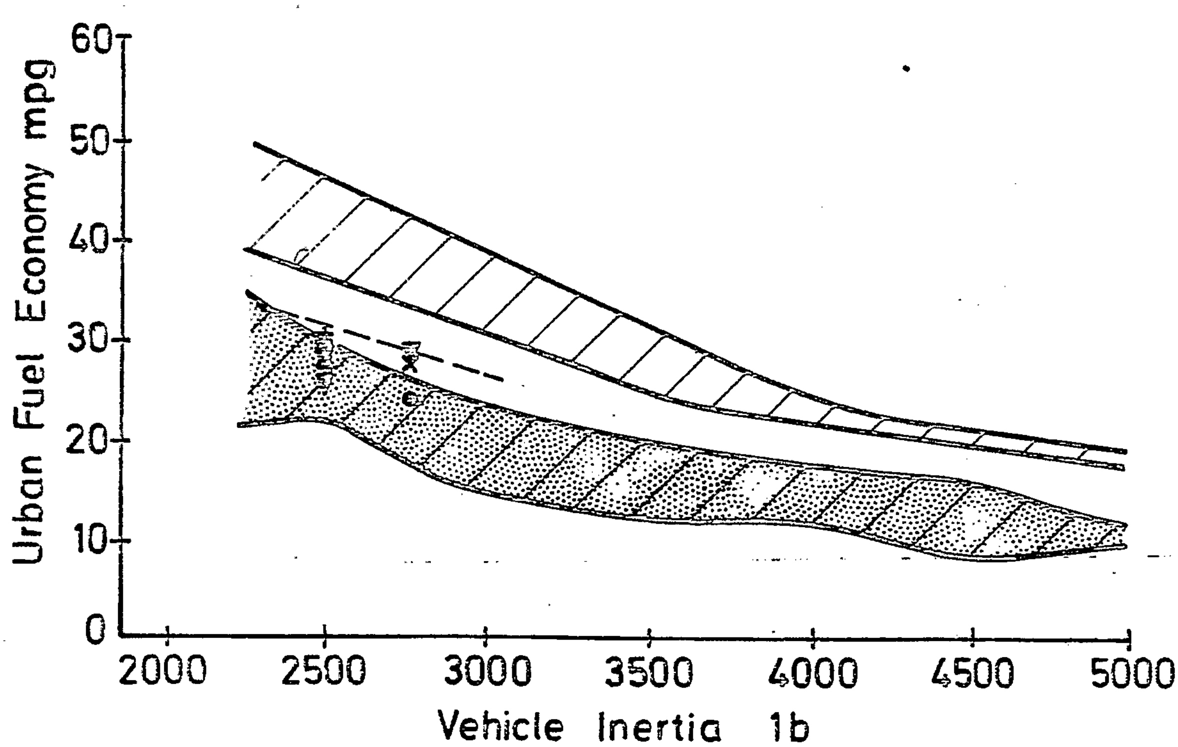
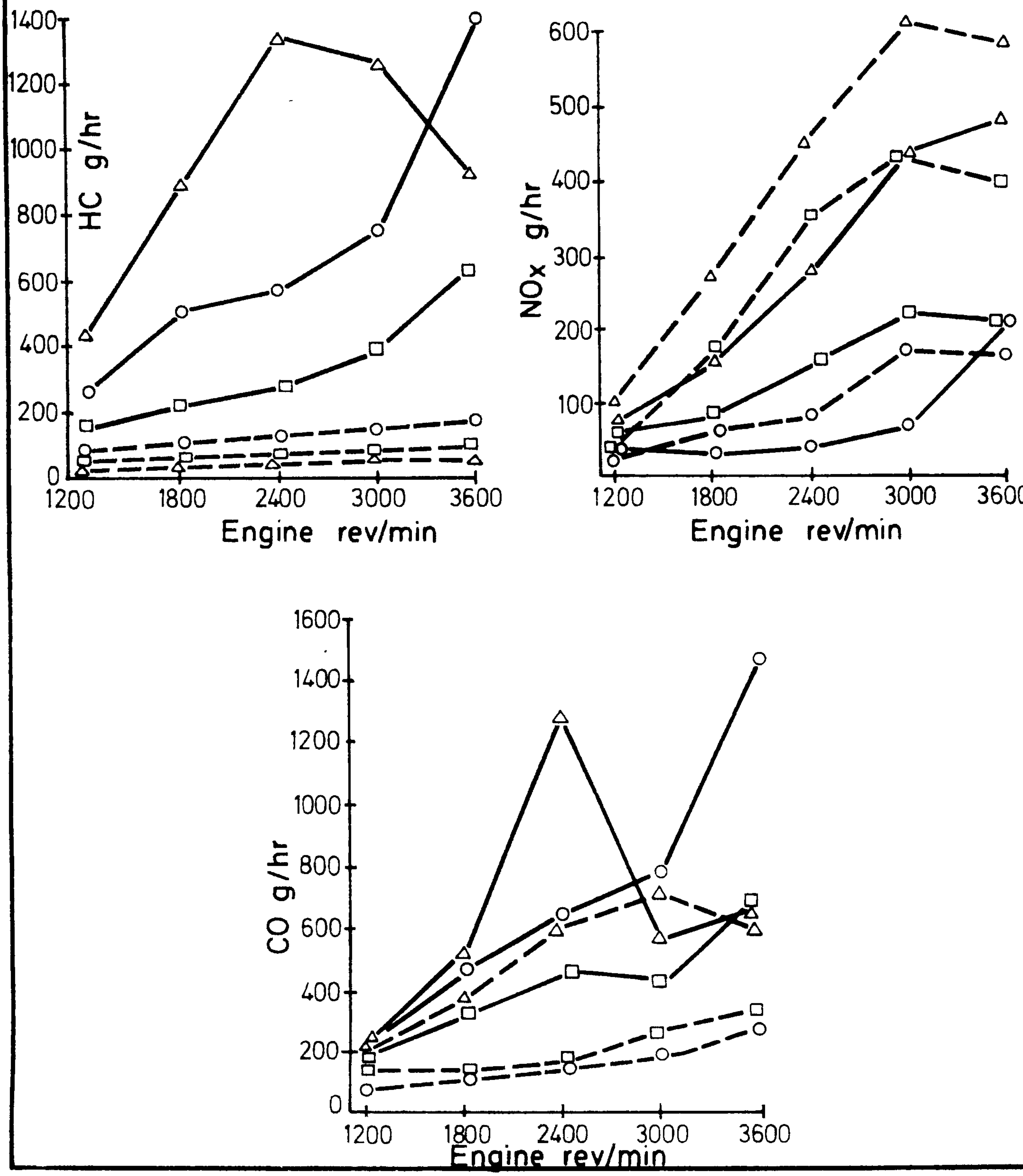


FIG. No.66 (ref 111)
Drg. No. S8427
Date SEPT '80

Steady State Gaseous Emissions
L-163-S TCCS Engine

—— N° 2 Diesel Fuel
– – Reference Indolene Gasolene

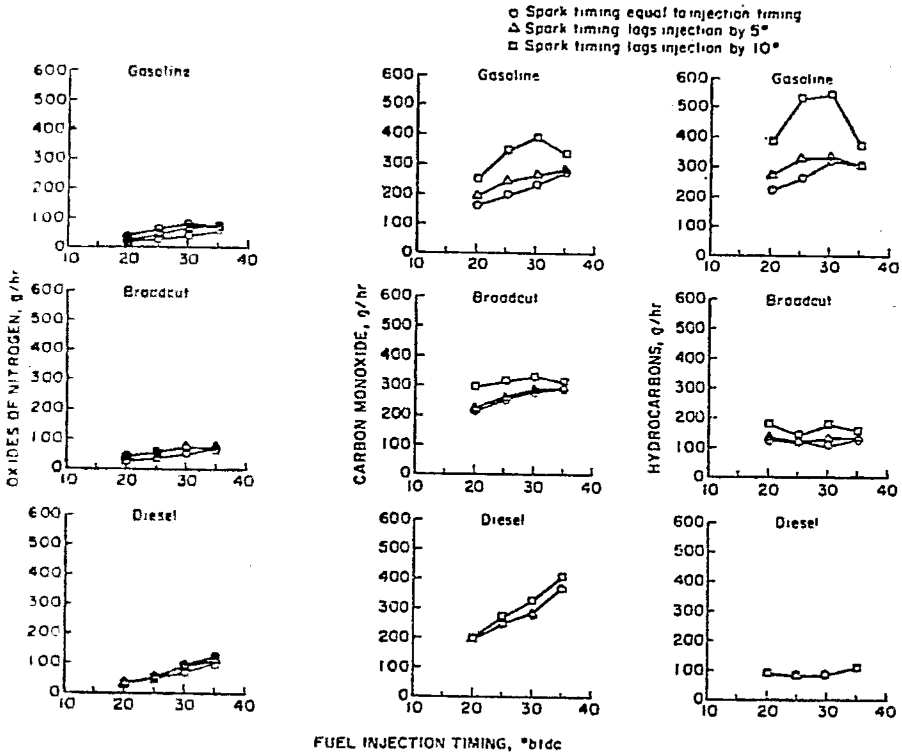
△ Full Load (equivalent for both fuels)
□ 2/3 Load
○ 1/3 Load



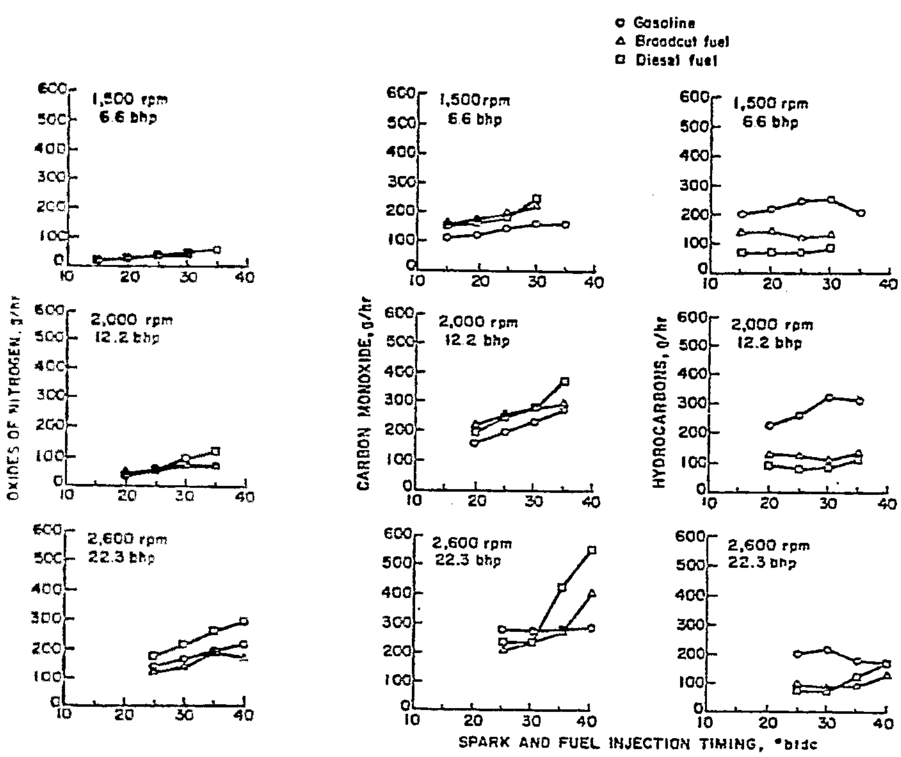
6983LT/5431

The Influence Of Spark Timing, Injection Timing And Fuel Type On NOx CO And HC Emissions LIS – 183 TCCS Engine

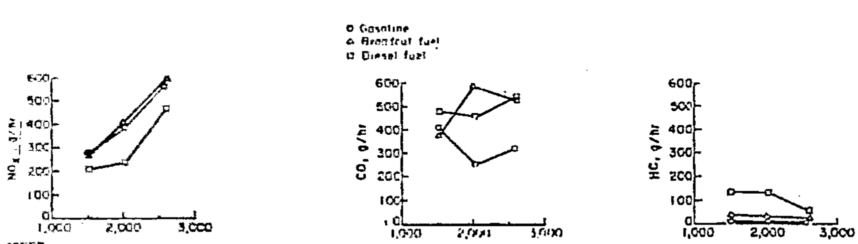
Fig N°67 (ref 112)
Drg. N° S8109
Date Oct 1980



NOx, CO and HC Emissions Versus Spark and Fuel Injection Timing 2000 RPM, 12·2BHP



NOx, CO and HC Emissions at Three Road Load Conditions With Coincident Spark and Injection Timing

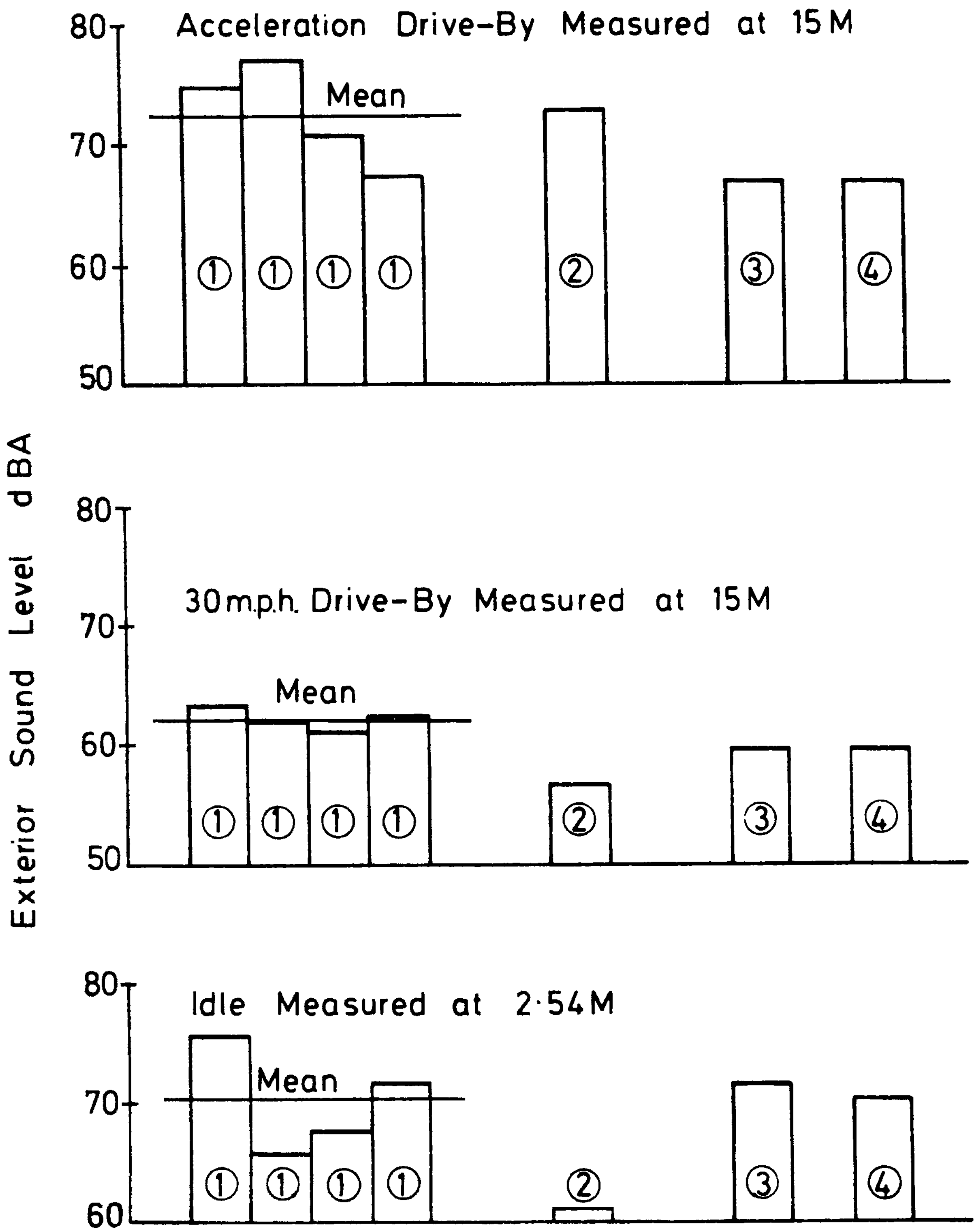


NOx, CO and HC Emissions at Full Load With Coincident Spark and Injection Timing

FIG. No. 68 (ref 110)
Drg. No. S8426
Date SEPT '80

Exterior Sound Level Comparisons
IDI Diesel, Gasoline and TCCS Powered Vehicles

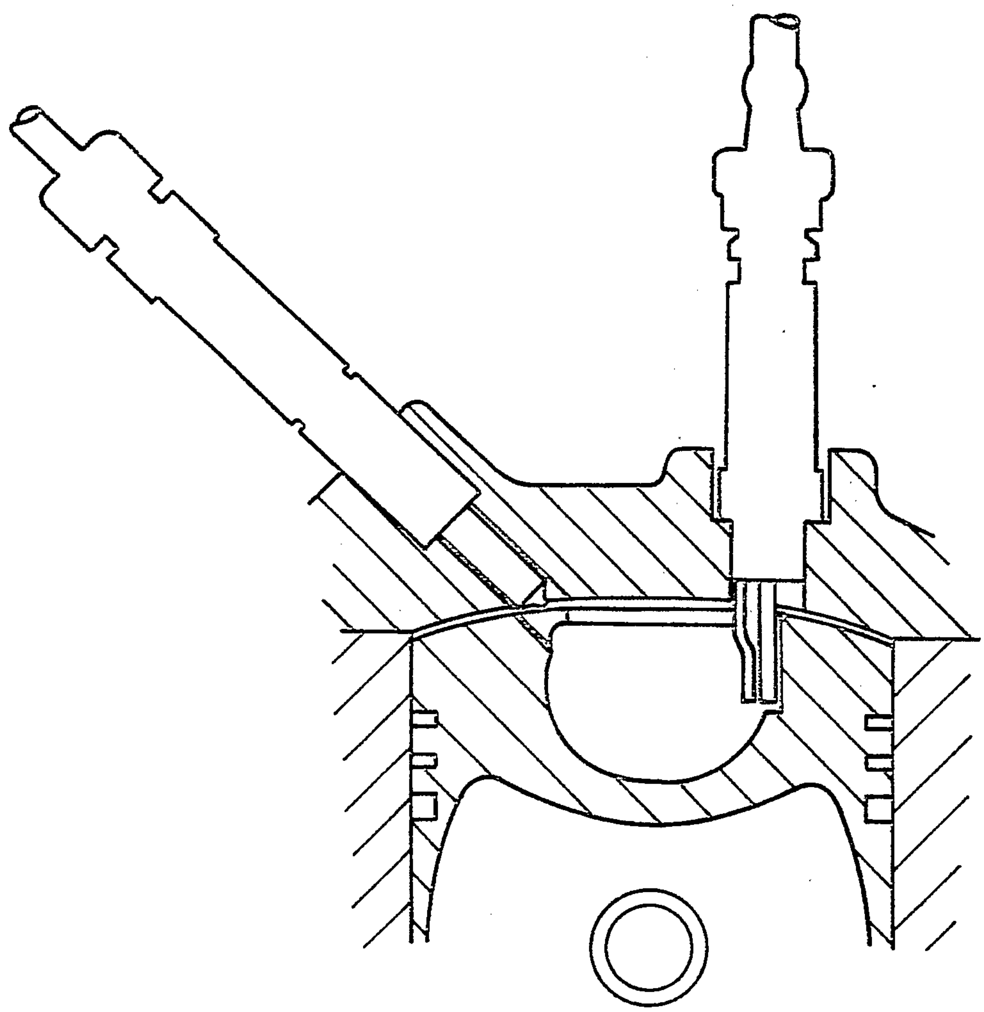
① IDI Diesel Passenger Vehicles
② Typical European Gasoline Passenger Vehicle — 122 CID
③ TCCS Cricket — Diesel Fuel
④ TCCS Cricket — Gasoline Fuel



RICARDO

FIG. No. 69
Drg. No. S8425
Date SEPT '80

Layout of the MAN. FM. Combustion System



FM Combustion Bowl Details

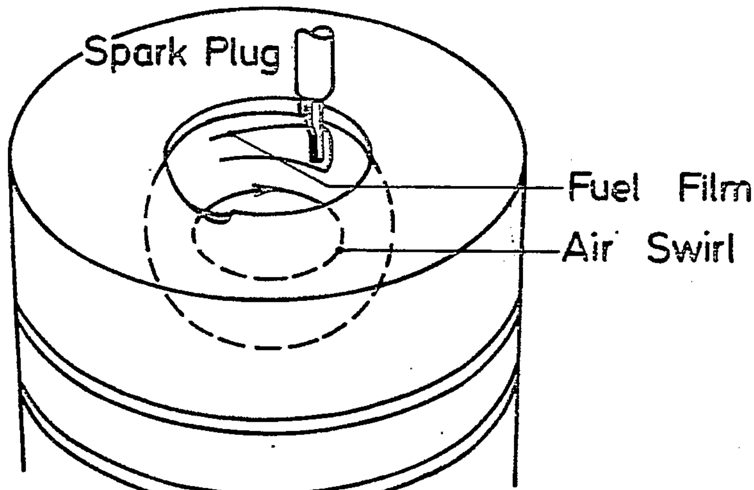


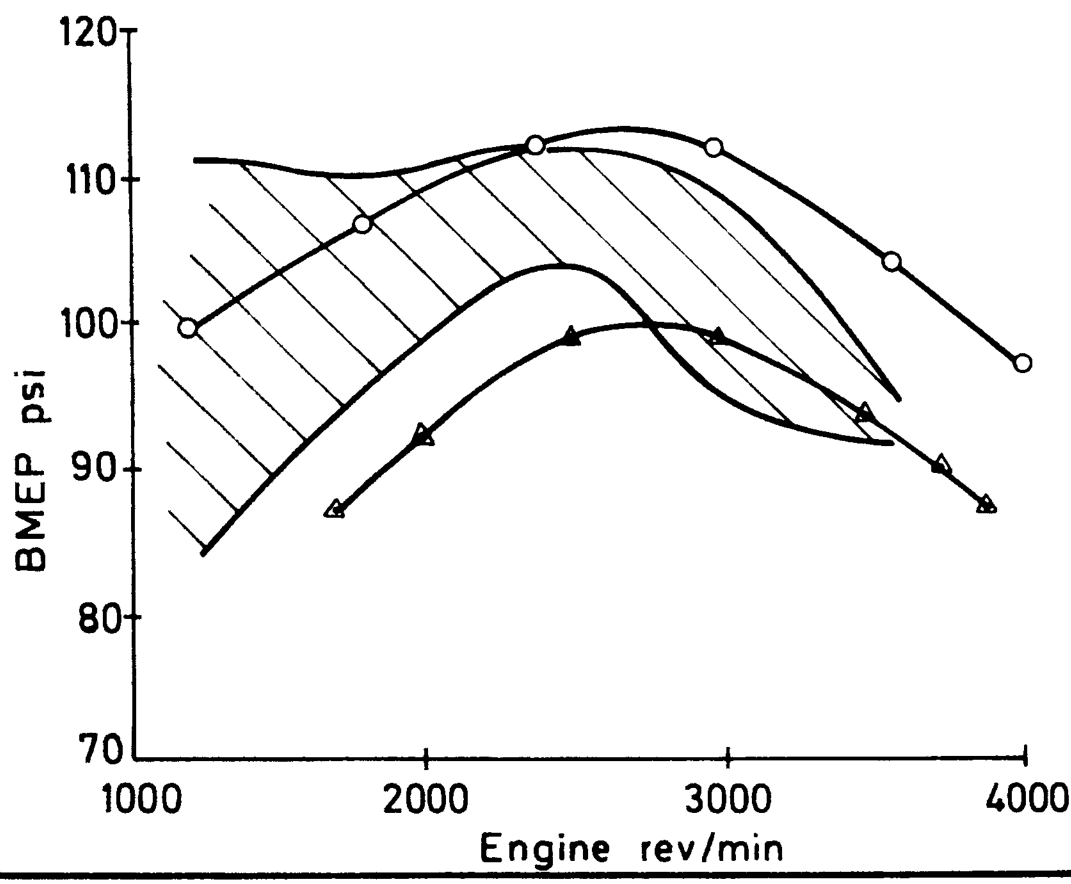
FIG. No. 70 (ref 125, 124)
Drg. No. S8424
Date SEPT '80

Smoke Limited BMEP Curves For Light Duty FM and IDI Diesel Engines

△ Multi-fuel MAN L9204 FMV–3·62″ Bore x 3·94″ stroke x 4 cyl. 162 CID. C.R. 16·5 : 1
Torque Curve for Diesel Fuel JP-4 and 100 Octane Gasoline – Smoke Output Typically 3, 2·5 and 1 Bosch For Diesel Fuel JP-4 and Gasoline Respectively

⊙ 91 Octane Gasoline Optimised – Typical Prototype Multi-Cylinder Curve for FM Engines of 25–30 CID per Cylinder C.R. 13–14 : 1
Smoke Limit Same as IDI Diesel Band

▨ Envelope for developed Swirl Chamber IDI Light Duty Diesel Engines – Smoke Limit, 2–3 Bosch Low Speed, 1·5–2 Bosch Mid and High Speed



6983LT/8431

FIG. No. 71 (ref 125,124)
Drg. No. S8423
Date SEPT '80

Fuel Consumption Comparison For FM Diesel and Gasoline Engines

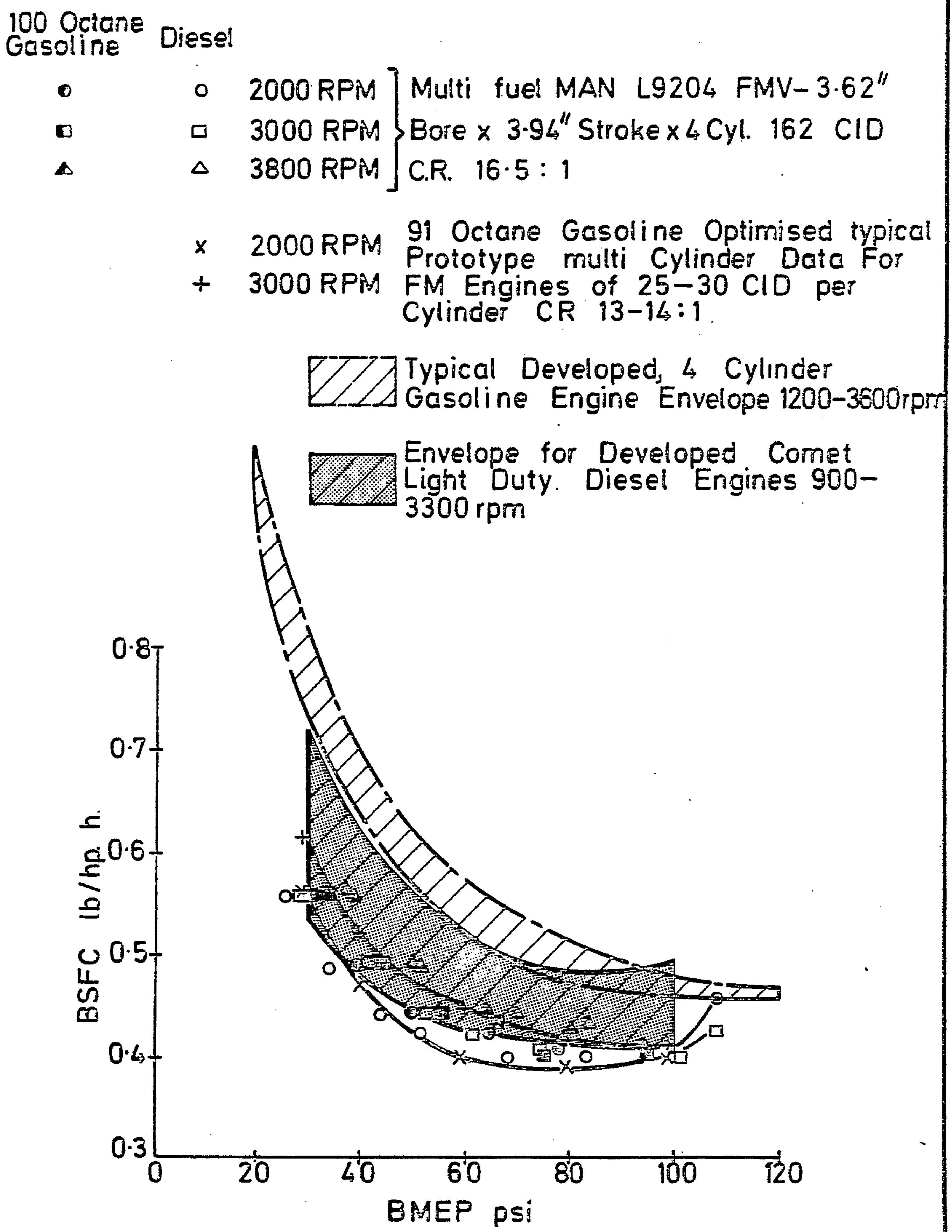
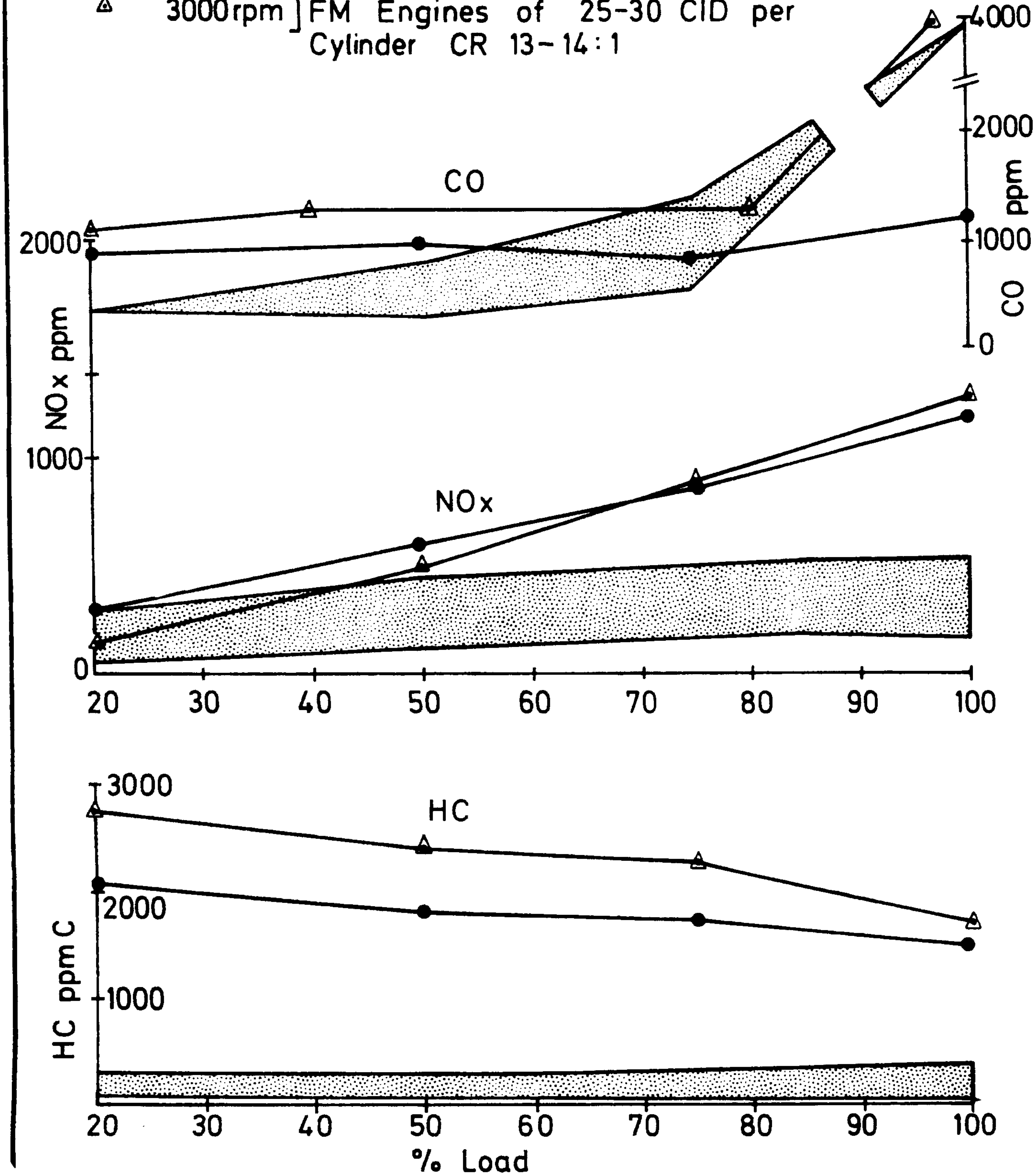


FIG. No. 72 (ref 125)
Drg. No. S8422
Date SEPT '80

Comparative HC CO and NOx Emissions for FM and Diesel Engines

Envelope for Developed Comet Light Duty Diesel Engines 1500 – 4400 rpm

● 2000rpm, △ 3000rpm: 91 Octane Gasoline Optimised Typical Prototype multi Cylinder Data for FM Engines of 25-30 CID per Cylinder CR 13-14:1



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