Title
COMBUSTION OF RESIDUAL FUELS IN A CFR DIESEL ENGINE

Permalink
https://escholarship.org/uc/item/76t6q4zp

Authors
Ruzicka, D.J.
Robben, F.
Sawyer, R.F.

Publication Date
1983-11-01

LBL-17013
Preprint

Lawrence Berkeley Laboratory
UNIVERSITY OF CALIFORNIA

APPLIED SCIENCE DIVISION

To be submitted for publication

COMBUSTION OF RESIDUAL FUELS IN A CFR DIESEL ENGINE

D.J. Ruzicka, F. Robben, and R.F. Sawyer

November 1983

APPLIED SCIENCE DIVISION

Prepared for the U.S. Department of Energy under Contract DE-AC03-76SF00098

DISCLAIMER

COMBUSTION OF RESIDUAL FUELS IN A CFR DIESEL ENGINE

D. J. Ruzicka*, F. Robben and R. F. Sawyer
Lawrence Berkeley Laboratory
University of California, Berkeley

ABSTRACT

The correlation of diesel combustion quality with the characteristics of various residual fuels was carried out using a Cooperative Fuel Research diesel engine. This engine was slightly modified to obtain satisfactory operation on residual fuels with viscosities up to 500 cSt at 50°C. This engine, which has a displacement of 0.6 liter, was operated at 900 RPM in an attempt to partially simulate combustion conditions in larger engines. The heavy fuels employed were based on straight run, vacuum and visbroken residua. A modified straight run residual fuel and two commercial fuels were also examined. The following engine parameters were determined: thermal efficiency, specific fuel consumption and ignition delay. Particulates, hydrocarbons and carbon monoxide emissions were measured. The engine performance parameters (except ignition delay), hydrocarbon emissions and particulate emissions correlated best with the Conradson Carbon Residue values for the different fuels. The ignition delay correlated well with the C/H ratio. In addition, two combustion improvers were tested using diesel No. 2 and Bunker 6 fuels. Slight improvement in the engine performance and in emissions was observed with both fuels. However, the improvement observed was smaller than that reported from field engines. An explanation of this phenomenon is offered.

---

*Permanent address: Central Institute for Industrial Research, Oslo, Norway.

TABLE OF CONTENTS

1. INTRODUCTION

The marine shipping industry has traditionally used residual fuels for propulsion. The larger diesel engines used in these ships, both low and medium speed, have been engineered so as to be capable of operating on these fuels. Prior to the oil crisis (about 10 years ago) marine diesel engines were primarily operated on straight run residual fuels (residua after atmospheric distillation). These were well suited for diesel engine operation, since they consisted of components with more or less a continuous range of boiling points and often were of quite aliphatic nature. A few parameters were sufficient for adequate description of such fuels. These mainly classified the fuels in terms of their handling characteristics such as the density, viscosity, pour point, flash point, water, etc. Viscosity was the most important parameter, since the atomization characteristics of the injector depends on the viscosity of the fuel. The price of bunker fuels has always been related to the viscosity.

The principal consequence of the oil crisis has been a steep increase in the cost of crudes and therefore of marine fuels. To compensate somewhat for the increased prices, and to meet the increasing demand for distillate fuels, oil refineries have increased the yield of the lighter hydrocarbon fractions by intensifying the refining. The refining processes include vacuum distillation, visbreaking and fluidized bed catalytic cracking (FCC).

The residua from these processes are consequently of a lower quality. The lower quality is first of all reflected in considerably increased levels of impurities such as metals, sulfur, ash and catalytic fines. Secondly, the viscosity is higher and there is a higher proportion of chemical species which are not sufficiently reactive to make good diesel fuels.

The viscosity of a "modern" residuum can be reduced to an acceptable level by adding a relatively small amount of a low viscosity distillate and by heating. However, because the bulk of such bunker fuel is degraded by increased amounts of aromatic, polar and olefinic constituents, numerous problems arise, both prior to and during the combustion process(1,2,4). Typical problems are cylinder wall corrosion, deposits, mechanical wear, burned valves, stuck rings and smoke emission. Sometimes complete breakdowns resulted.

The use of these refining methods (especially visbreaking and FCC) is expected to increase with time, thus the problems associated with the use of bunker fuels will be increasing. Reference 3 gives a review of the expected situation in the 80's and includes a number of references on the use of bunker fuels in diesel engines.

The problems associated with the utilization of heavy bunker fuels in marine engines can be related partly to the increased impurity concentrations and partly to the undesirable chemical composition of the fuels. The two sets of problems can to a great extent be separated. The problems due to impurities are understood and various remedies have been employed, more or less successfully. The influence of the various chemical species on the diesel combustion process is less clear and there is also less agreement on the measures to be taken. Berryman(4) approximated the quality of combustion by the emission of particulates. However, no correlation was found between the particulates emission and the asphaltene content. Van der Horst et al.(5) found a correlation between asphaltene and piston deposits in a Bolnes 3 DNL engine, and a correlation between Conradson Carbon Residue and piston deposits in a Caterpillar engine. The correlations found were thus not general.

At the Central Institute for Industrial Research in Oslo chemical characterization of heavy fuels has been carried out for a number of years employing analytical techniques such as gas chromatography, high performance liquid chromatography and gel permeation chromatography(6). Satisfactory correlation with field engine performance has not been possible. The difficulty seems to be that, while the analytical data are accumulated in one laboratory using the same instrumentation, the field engine data originate from different types of engines run at different conditions.

There is thus a need for finding suitable fuel parameters which will correlate with engine combustion performance, so that ignition and combustion problems may be predicted.

The aim of the present project is as follows:

1. Measure the ignition and combustion properties of various residual fuels in a diesel engine;

2. Correlate the engine performance with fuel characteristics;

3. Test possible effects of combustion improvers. A cooperative fuel research (CFR) engine was chosen to carry out the combustion experiments. This engine is rather different in size, combustion chamber geometry and fuel injector design than modern medium-speed diesel engines. However, it was available, offers a good deal of flexibility in the operational parameters, and is fairly inexpensive to operate.

2. EXPERIMENTAL

2.1 CFR Engine

A detailed description of the Cooperation Fuel Research (CFR) engine is given in References 7 and 8. Some important data are:

Cylinder bore:	8.26 cm
Cylinder stroke:	11.40 cm
Displacement:	612 cm^3
Compression ratio:	8-23.5
Injection pump:	Bosch with variable port closing
Injector:	Bosch ADN-30S-3/1
Engine speed:	940 r.p.m.
Lubrication oil:	Chevron DLO SAE 30, kept at ~ 70°C

The engine was connected to an Eaton eddy-current dynamometer with manual torque adjustment. Readings of torque were taken every 5 minutes and averaged over a time interval, usually

1 hour. Pressures in the combustion chamber and in the fuel line were monitored by means of piezoelectric pressure transducers and recorded by a dual beam oscilloscope. The following instrumentation was used:

Combustion chamber:

Kistler Model 601 B21 X/N C3803 transducer

Kistler Charge Amplifier S/N 3312, Calibration factor: 1.11

Sensitivity: 100 psi/V (689 Pa/V)

Fuel line:

Kistler Model 609 transducer

Kistler Charge Amplifier, S/N 0628, Calibration factor: 2.19

Sensitivity: 308 psi/V (2.12 k Pa/V)

Oscilloscope: Tektronix 565 Dual Beam

The flow of charge air was monitored by measuring the pressure drop across an orifice kept at approximately 35°C. The following formula was used for calculating the flow:

$$\dot{M}_{air} = 43\ d^2\ \sqrt{P\Delta P/T}\ \text{g/min}$$

d = diameter of orifice in cm = 0.953 cm

P = pressure in mm Hg = 2838 mm Hg

ΔP = pressure drop in inches H_2O = 4.7 in. H_2O

T = temperature in K = 308 K

The engine was not equipped with a needle lift detector for measurement of injection timing and duration. Neither was any special instrumentation for the detection of ignition available. To obtain some measure of the injection timing and the ignition delay the pressure traces from the fuel line and the cylinder were used.

The time scale of the oscilloscope was calibrated as follows. Two trigger markings were placed on the flywheel, one at 15° before TDC and one at TDC. The time scale of the oscilloscope was then adjusted to indicate a suitable number of crank angle degrees per division. The maximum in the fuel line pressure was taken as the injection start and the point of departure of the cylinder pressure trace from the compression trace was taken as the ignition. Figure 1 shows two examples taken while running on Bunker 6. The time difference between the two pictures was about 5 minutes and random cycles were made in each photograph. It is seen that the repeatibility is good. An average ignition delay of 1.1 msec was found in both cases.

2.2 Modification of CFR Engine for Heavy Fuels

The CFR engine was designed to operate on diesel No. 2 type fuels, which have a viscosity in the 2-4 cSt. range (ASTM D445). Because its displacement is small and the speed fairly high, this type of engine is not ideal for testing heavy residual fuels. The small displacement leads to relatively high heat and frictional losses and the high speed makes the time available for combustion short. Further, this engine has a rather small precombustion chamber, with high turbulence level, and a pintle type injector with relatively fine spray characteristics. The diesel spray is also likely to impinge on the walls.

However, the CFR engine offers a great deal of flexibility in the operational parameters. Further, the consumption of fuel is low, so that the running expenses are very modest. Thus, if the operational parameters are well chosen, conditions in a large engine may be approximated to some extent.

To facilitate a sufficient rate of evaporation of the fuel after injection it was considered imperative to increase the wall temperature of the engine. This was achieved by using ethylene glycol in the coolant jacket of the cylinder, resulting in a coolant temperature of 166°C. This is a bit lower than the boiling point of glycol, probably due to the presence of some water. The intake air temperature was kept at 121°C (250°F) in all comparative runs and it was varied from 135°C (275°F) to 52°C (126°F) in the case of fuel IBF-180 D-3096. The water cooling of the injector nozzle was also replaced by glycol cooling to raise the temperature. The glycol was circulated through a bath containing a water cooling coil. By controlling the cooling water flow with a temperature activated solenoid valve, the temperature of the coolant could be set at any point between 40 and 130°C. The temperature was measured by means of a thermocouple placed directly in the glycol line and could be controlled to within ± 1°C.

Two separate fuel supply systems were connected to the fuel pump. One was diesel No. 2 (for the purpose of warming up the engine to operating temperature) and the other contained the fuel to be tested. To obtain a reasonably quick change-over between the fuels it was found necessary to fit one fuel filter (10 μm) to each line. It would have been desirable to have two separate fuel pumps as well.

The fuel to be tested was originally placed in a 500 ml burette and the fuel flow-rate was determined volumetrically. Later it was found more accurate to use a container placed on an electronic balance (Mettler PC16). The accuracy of this balance was satisfactory, provided the weight loss was observed over a period of three minutes or longer. Usually the fuel consumption was measured over a considerably longer time.

The heavy fuel container (burette or can), the filter and the fuel line to the pump were heated by heating tapes controlled by variable transformers. The temperature of the fuel was kept at about 70-80° C and the container was covered to reduce the evaporation of the more volatile components. The fuel pump and the fuel line to the injector were also wrapped with heating tapes and the temperature was controlled by a proportional controller. The temperature was measured by a thermocouple directly in the fuel line.

The temperatures of the different fuels were adjusted so that the viscosity in the fuel line was approximately 10cSt. This implied heating the fuels to the following temperatures:

Fuel:	Temperature in fuel line:
D-3096 IBF 180	130°C
D-3144 Vac. Res.	130°C
D-3143 Visbr. Res.	130°C
D-3031 Straight Run Res.	129°C
D-3031 SRR + 12% Me-naphthalene	103°C
D-3031 SRR + 23% Me-naphthalene	88°C
D-3150 Bunker No. 6	150°C

Viscosity data determined by Chevron at 100°C (see Table 3) were extrapolated to the desired values. The viscosities of IBF 180 D-3161 (used for preliminary test runs) and of IBF 180 D-3096 were measured as a function of temperature using a Brookfield LVT viscometer. The two sets of values agreed quite well (Fig. 2). The viscosities of the mixtures of SRR and α-Me-naphthalene were determined at 50° and 100°C and extrapolated.

	50°C	100°C
SRR + 12% MN	61.4 cSt	10.5 cSt
SRR + 23% MN	30.9 cSt	7.2 cSt

When performing runs with the different fuels an effort was made to keep the volumetric flow rate approximately constant so that the injection duration was approximately constant. It appeared that small adjustments had to be made on the fuel pump to achieve this. The resulting fuel rates were approximately 12-13 g/min, which is slightly above the values recommended by ASTM for the engine.

Since the charge air was heated, its pressure was increased slightly so that the amount of air corresponded approximately to natural aspiration. The equivalence ratio for fuel D-3096 was 0.68. The air charge was not changed for the other fuels in order not to change the turbulence conditions in the engine. It was assumed that this was more important than keeping the exact equivalence ratio constant.

2.3 Emissions Analyses

In addition to engine performance data it was desirable to obtain data on the emissions for the different fuels. Instrumentation for measuring the concentrations of hydrocarbons, carbon monoxide and particulates was available. These constituents of the exhaust give a picture of how much of the fuel hydrocarbons has not been utilized in the combustion process.

2.3.1 Hydrocarbon analysis

A Beckman Hydrocarbon Analyzer, Model 402, with a flame ionization detector was used for the measurement of hydrocarbon concentration in the exhaust gas. The sample was drawn by an internal bellows pump through a 3 m long heated sampling line. The probe was a 1/4 in. tubing with two holes of 0.07 in. diameter placed across the exhaust pipe (I.D. = 1 3/8 in.) with the holes downstream. The instrument was calibrated by internal admission of a calibration gas (N_2) containing 414 ppm propane.

Quantitative measurements could not be obtained unless the sample of the exhaust gas was filtered. A heated filter of 11 cm diameter (Whatman glass fiber 934AH) was placed upstream of the sampling line. It was changed daily. The sampling line was furthermore purged by drawing laboratory air through it as often as possible. To check whether the particulates in the filter interfered with the HC analysis, the calibration gas was admitted through the filter and the sampling line under normal operating conditions. Good agreement was obtained with the calibration procedure normally used.

During engine runs the hydrocarbon analyzer was used continuously. It responded well to changes in the operational conditions of the engine. The uncertainty in the hydrocarbon determinations in the exhaust is estimated at ± 10%.

2.3.2 Carbon monoxide analysis

Carbon monoxide was measured in the exhaust gas using a Beckman analyzer Model No. 315 L.P. based on IR absorption. Two analysis cells were provided and were calibrated using CO concentrations of 0.16% and 8.7% in nitrogen, respectively. Exhaust samples were withdrawn through the same probe and filter as the samples for hydrocarbon analysis. The interference of water with the CO determination virtually was eliminated by letting the gaseous sample pass through a glass condenser cooled with ice water*. Because of this arrangement analyses of CO were only taken intermittently. The accuracy of CO measurements is estimated at better than ± 10%.

2.3.3 Measurements of particulates

For collecting the particulates the exhaust gases were diluted 7.3 times with filtered air. The temperature of the gases was thus reduced to approximately 50° C and the concentration of particulates to a reasonable level for detection. The sampling was done through a 1/4 in. tube placed across the exhaust pipe (I.D. = 2 in.). The probe was closed at the end and had seven holes of 0.086 in. arranged in two rows and facing upstream, in order to provide a mean sample of the exhaust.

The samples were withdrawn at a constant rate of 10 ℓ/min for 10 seconds in order not to overload the filter. A Sierra pump, model 110 DK-SP, was used. The filters were made of teflon and had a porosity of 1 μm. In most cases five samples were collected for each operating point. The probe was purged prior to each sample by opening the sampling valve to the atmosphere.

The mass of the particulate samples was obtained by measuring the difference between the attenuation of β-radiation by the filter containing the sample and filter alone. This automated method was developed at The Lawrence Berkeley Laboratory and is described in Ref. 9. The uncertainty of each measurement was estimated at ± 20%, but the final

---

[*] 3.5% H_2O corresponds to 5 ppm CO; the saturation water content at 0° C is 0.6%.

uncertainty was reduced by taking several parallel samples.

2.4 Fuels Employed

The residual fuels for these experiments were kindly provided by the Chevron Research Company, Richmond, California. D-2 Diesel Control Fuel, purchased from Phillips Chemical Company, was used as a reference fuel. α-Me-naphthalene was used as a C/H modifier and was obtained from Kodak. Table 1 gives a description of the residual fuels. In addition to the above fuels Chevron provided us with a larger quantity of production intermediate bunker fuel IBF180 D-3161 for the purpose of optimizing the engine operational parameters.

Properties of the D-2 Diesel Control Fuel are shown in Table 2. Some important properties of the residual fuels are listed in Table 3. Table 4 gives the heats of combustion of the residual fuels, α-Me-naphthalene and D-2 Diesel Control Fuel. Data in Tables 3 and 4 on the residual fuels were kindly provided by Chevron.

2.5 Optimization of the CFR Engine for Residual Fuels

There are several operational parameter that can be adjusted on a diesel engine and some of them are interdependent. In order to adjust a diesel engine optimally one ought to make several rounds of adjustments of the individual parameters. Such fine adjustment was not possible to achieve within the scope of this work. It was therefore decided to adjust the parameters for a typical fuel among the tested ones in one round. Intermediate Bunker Fuel 180 D-3161 was chosen for this purpose.

Preliminary runs showed that the engine could barely be run on a fuel mixture of 25% IBF 180 and 75% Diesel #2 under the following conditions:

Cylinder coolant: 166°C (332°F)
Charge air: 110°C (232°F)
Injector pressure: 10.3 MPa (1500 p.s.i.)
Injection timing: 18° BTDC
Compression ratio: 23.5:1
Injector coolant: 63°C

The engine would not run continuously at all on 100% IBF 180, or on 50% IBF 180 and 50% Diesel #2, with the injector coolant temperature at 92°C or at 130°C.

Keeping the cylinder coolant constant at 166°C, each of the other parameters were then optimized, one at a time. The optimum value obtained was then used when optimizing the next parameter. However, in some cases the injection timing had to be retarded when optimizing the next parameter because of engine knock. Table 5 summarizes the five series of experiments carried out. Figures 3 to 7 show the variations in the torque and in the hydrocarbon emissions as a function of the parameter being optimized.

For the other test fuels the injection timing had to be retarded further to 11° BTDC and the conditions used for comparative measurements are those listed in bottom line of Table 5.

2.6 Runs with Various Residual Fuels

Engine runs with the various fuels were carried out using the operational parameters given at the bottom of Table 5. The fuels in the fuel line were heated to the temperatures listed in Section 2.2. The injector coolant temperature had to be decreased from 54°C to 40°C after three runs because of deposit formation around the injector, presumably due to pyrolysis. This change should not have a serious bearing on the results, according to Figure 7 which shows that the engine output and HC emissions are constant in this temperature range. The injector coolant temperatures for the different fuels were as follows:

Fuel:	Injector coolant T:
IBF 180 D-3096	54°C
Vac. R.	54°C
Vis. R.	54°C
SRR	40°C
SRR + α-Me-N	40°C
D-2 Diesel Control Fuel	40°C
Bunker 6	40°C
Bunker 6 + combustion improvers	40°C
D-2 D.C.F. + combustion improvers	40°C

In addition to these comparative runs the influence of the variation of injection timing and of charge air temperature on the combustion of IBF 180 D-3096 was investigated. The timing was varied from 3 to 21° before TDC and the air temperature from 135° to 52°C.

Stabilizing the charge air temperature at discrete values proved difficult. The heater was therefore switched off at the highest temperature (135°C) and the air temperature allowed to drop. A plot of the air temperature vs. time was made, and the engine performance data (recorded as a function of time) could then be related to charge air temperature.

2.7 Runs with Combustion Improvers

There have been a great number of combustion improvers on the market but reports of their effect on the engine performance vary a great deal. We have tested the effect of two such additives:

Improver I - to be used in the ratio 1.25 ℓ to 1 metric ton. It is claimed to have all-round beneficial effects on engine combustion resulting in an improvement of fuel economy in excess of 10%.

Improver II - to be used in the volume ratio 1:2500. This additive is supposed to reduce deposit formation through combustion improvement.

We have measured the effect of these improvers with D-2 Diesel Control Fuel and with the heaviest of our fuels--Bunker 6. In the case of D-2 D.C.F. 100% load conditions were established and then the effect of the additives was measured at 76% and at 97% of full load. The estimated load in the experiments with Bunker 6 was in excess of 76%, since the fuel pump setting was the same as for D-2 D.C.F.

2.8 Internal Friction of the Engine

In the engine runs the brake output power was measured for the different fuels. It was, however, considered better to base the comparison between the different fuels on the total power output, including the internal frictional losses.

There were no easy means available for measuring the indicated power. However, the frictional power loss may be considered nearly constant, since the lubrication oil temperature was kept constant. The frictional loss for operations on one fuel was determined and used as a constant for the other fuels.

By careful measurement of the cylinder pressure trace for D-2 Diesel Control Fuel it was possible to construct a P-V diagram. Integration of this, taking into account the geometry of the piston motion, then gave the indicated power. The measured brake power was subtracted, yielding the frictional power loss. The results were the following:

Indicated power:	3.28 kW
Brake power:	2.04 kW
Frictional power:	1.24 kW

3. RESULTS AND DISCUSSION

3.1 Comparison between Different Fuels

Table 6 gives the data on ignition delays, engine performance and emissions for the different fuels. For an easier comparison the fuels have been ranked from 1 (best) to 6 in Table 7, for each of the measured engine performance parameters. Obiously, D-2 Diesel Control Fuel has the best rating for all the performance parameters. Using the rating figures, and giving the different performance parameters the same weight, the following average fuel performances were obtained:

Best performance:	SRR and Vac. R.
Medium performance:	IBF 180
Worst performance:	Bunker 6 and Vis. R.

Bunker 6 is the heaviest of the investigated fuels. Visbroken Residuum is the fuel that has undergone the greatest chemical changes among the tested fuels, e.g. suffered the greatest increase in unsaturated hydrocarbons. The latter possess a low cetane number.

Figure 8 shows gas chromatograms of the residual fuels, except for Bunker 6*. The abscissa shows the boiling temperature in °C. Figure 8a shows that the b.p. distribution for SRR is continuous - thus distinctly different from the other fuels. This explains the good overall combustion properties and a short ignition delay in particular (see Tables 6 and 7). The gap in b.p. distribution in the case of Vac. R. (Fig. 8b) is not so favorable for the evaporation of the fuel after atomization. There is a considerable ignition delay as compared to SRR (Table 6).

---

*The data were kindly provided by G. Østvold of the Central Institute for Industrial Research, Norway. Bunker 6 was not available at the time of analysis.

Figure 8d shows the gas chromatogram of Vis.R. It shows first of all that the b.p. range of the light fraction only covers 200-350°C and that the amount is relatively small. This results in a long ignition delay. The large gap that exists between the light and heavy fraction is unfavorable, since the evaporation of fuel is not so monotonous as when the gap is smaller. Since the heavy fraction is also of low quality (shown for example by a high asphaltene content in the blended fuel: 12.2%) the overall combustion properties are poor.

3.2 Combustion Quality vs. Fuel Properties

Various engine performance parameters and emissions of products of incomplete combustion have been plotted against the important fuel parameters. The aim was to find out whether a single fuel parameter would correlate with engine performance and emissions. Engine performance and emissions were expressed in terms of indicated parameters except for sfc which was also expressed by the brake quantity.

The results are summarized in approximate terms in Table 8. Three grades were chosen to assess the degree of correlation, as indicated in Table 8. It has been difficult to make exact judgments of the correlation, since there are few points on each curve and statistics cannot be applied. However, even so, some interesting information has been obtained. When considering plots with C/H ratio as a parameter on the one hand and asphaltene on the other, it is noted that the results are opposite. Ignition delay, HC and CO emissions correlate with C/H, while imep, isfc, bsfc and particulates correlate with asphaltenes and CCR.

The good correlation between engine performance and asphaltenes is understandable. When running on a blended residual fuel the performance may be limited by the worst components of the fuel--the asphaltenes. Asphaltenes, being highly aromatic and polar in nature, are also prone to produce carbon-rich particulates. No correlation was obtained between the ignition delay and asphaltenes (Table 8). Under our conditions of running the engine fairly warm (166°C in the coolant) the ignition delay depends mainly on the dominating lighter fractions of the fuel, which do not contain asphaltenes. Since the concentration of asphaltenes is relatively small (see Table 3), the average C/H ratio of these fuels are mainly determined by the other components. There is therefore a correlation between the ignition delay and the average C/H of the fuels.

The emissions of hydrocarbons and of carbon monoxide appears to depend on the average composition of the fuels (C/H) and are independent of asphaltenes. Asphaltene molecules (the rings in their structure in particular) are slower in combustion and are largely responsible for the formation of carbon-rich particulates. Thus the correlation between particulates and asphaltenes concentration is reasonable.

It is seen from the last column of Table 8 that when aromatics and polars are added to the asphaltenes, the correlation becomes worse. This confirms that asphaltenes (which we found correlate with CCR) are the principal limiting factor in combustion of these fuels.

The plots where good correlations exist, and those where some correlation exists, are shown in Figs. 9-23. They should be studied in conjunction with Table 8. Attention will be drawn to several figures.

It is seen in Figs. 9 and 11 how increase in the C/H ratio, by adding α-Me-naphthalene, affects the I.D. and the HC emissions. The I.D. increases but the points do not fall on the curve (at least not the second point). The deviation from the curve is even more pronounced in Fig. 11 (HC vs. C/H). This means that the average molecules causing the C/H to vary from fuel to fuel are of lower combustion quality than α-Me-naphthalene, which has a cetane number zero.

It is characteristic of Figs. 22 and 23 that data for Vis.R. fall far outside the existing curve for the other fuels. This has also been observed in other plots where partial correlation existed. The reason for the different behavior of Visbroken Residiuum is that visbreaking is a chemical process producing molecules which are rather different from the original ones. As mentioned earlier, a great deal of unsaturation and aromatization results during visbreaking.

When comparing the correlation plots with asphaltenes and CCR as parameters it is seen that the CCR gives somewhat better correlation than asphaltenes. For practical purposes it might therefore be desirable to use CCR figures as approximate measures of the suitability of residual oils as diesel fuels. It is, however, felt that the accepted procedure for measuring the CCR (which is a measure of cokability and oxidizability) needs some improvement. Lenda(13) has in fact been able to correlate CCR values with data obtained by a modern analytical method: Gel Permeation Chromatography.

3.3 Optimization of Engine with IBF 180 D-3096

As mentioned in an earlier section, the performance of the CFR engine with the above fuel was examined further as a function of injection timing and charge air temperature.

3.3.1 Injection Timing

The results for torque and hydrocarbons emission vs. injection timing are shown in Fig. 24. It is seen that optimum performance is reached at an injection timing of about 16° before TDC. Ignition delays were measured for three injection timings:

Injection Timing (deg bTDC)	Injection Delay (m sec)
11	0.89
16	0.93
21	0.93

The ignition delay at 11° bTDC agrees well with the one previously found (Table 6). The longer ignition delay at 16° and 21° bTDC is due to the lower temperature and pressure at the time of injection.

3.3.2 Charge Air Temperature

The temperature in the cylinder of a diesel engine if very important for the ignition and combustion. Wolfer(10) proposed the following type of formula for the ignition delay:

$$t = \frac{A\ e^{E/R_oT}}{p^n} ,$$

where T and p are the absolute temperature and pressure at the ignition point, the other symbols are constants (approximately). Henein and Elias(11) set T equal to the average temperature of the ignition delay period. The two temperatures may not be too different, so that this has no serious bearing on the result, or at least not on the value obtained for E.

In our experiment, in which the charge temperature was varied, we obtained the results listed in Table 9. It is seen that only the ignition delay varied with the charge air temperature. The slight increase in torque that was observed was probably due to the slightly larger amount of charge air at the lower temperatures.

An attempt was made to construct an Arrhenius plot from the obtained results. To calculate the temperature of ignition for the different charge air temperatures isentropic compression was assumed and the following expression used:(12)

$$T = T_o \left(\frac{V_o}{V}\right)^{K-1}$$

T_o is the initial temperature (charge air temperature)
V_o is the initial volume
V is the volume corresponding to the piston position at ignition
$K = C_p/C_v$

The temperature is very sensitive in changes in K. To use as real a value for K as possible it was determined from the cylinder pressure trace data using the expression

$$P = P_o \left(\frac{V_o}{V}\right)^{K} ,$$

where P_o is the pressure of the charge air (1.07 atm). The value that gave best agreement between the above pressure formula and the experimental pressure data was K = 1.22. The following temperatures for the ignition were then obtained:

Charge air temperature	Temperature of ignition
135° C	767° K
104° C	741° K
83° C	670° K
63° C	634° K
52° C	614 K

The plot of log (ignition delay) vs. 1/T is shown in Fig. 25. No straight line was obtained and a complete levelling off is observed at the two highest temperatures.

The physical processes affecting the ignition delay, such as mixing, are expected to have a low activation energy. These processes obviously are rate determining in our case at the highest temperatures, where the apparent activation energy is approximately zero. At the lower temperatures the chemical processes will be slower (because of their higher activation energy) and play a more important role in the total process. Disregarding the highest temperature, an activation energy $E_a \sim 1.4$ kcal/mole is obtained. This value is quite low, indicating that pure chemical processes do not fully dominate the total process. The overall conclusion from this is that we have operated the engine at rather favorable conditions.

3.4 Effect of Combustion Improvers

The effect of combustion improvers was investigated with D-2 Diesel Control Fuel and Bunker 6 (see also Section 2.7). The results are shown in Table 10. For an easier comparison the relative values of the data are shown in Table 11. The different cases are discussed in more detail in the following.

D-2 Diesel Control Fuel at 76% Load

The use of both of the improvers at these conditions resulted in a decrease in the ignition delay and the emission of CO. There was also a slight increase in the particulate emission. This may be due to some constituents of the improvers which may coke more easily, relatively to the high quality D-2 Diesel Control Fuel at the low load conditions.

D-2 Diesel Control Fuel at 97% Load

The fuel consumption and emissions decreased with both improvers. The decrease in emissions was greater for Improver II than for Improver I. Also, the Mep increased somewhat with both improvers. The ignition delay was not measured. By comparison of the two sets of results, it is clear that the effect of the improvers increased as the load was increased.

Bunker 6 at an Estimated 80% Load

There was a measurable effect of both improvers on all the parameters. It is seen that the effects were greater in all cases with Improver II. The fuel consumption was reduced by 2% with Improver I and by 4% with Improver II.

The effect of these combustion improvers appears to be dependent upon the load of the engine and upon the type of fuel. Improver I is claimed to reduce fuel consumption by 10% or more. Under laboratory conditions, with a well adjusted engine, this was not found to be true. We believe our results are valid for larger engines of the type used in smaller marine vessels. Perhaps if an engine is poorly adjusted or defective in some manner, and is as a consequence not operating well, the combustion improver could result in a larger decrease in fuel consumption.

Improver II is claimed to reduce deposits. We have not been able to measure deposit formation, but particulates emissions should correlate with deposits. The particulates and also the other emissions were lower for Improver II than Improver I in all cases.

4. CONCLUSIONS

The main conclusions from this study are the following:

1. Even a small high speed diesel engine can be operated on heavy residual fuels if the operating conditions are chosen properly. Corrosion aspects have, however, been disregarded.
2. Fuel based on visbroken residuum showed the poorest combustion characteristics among the tested fuels.
3. Engine performance (mean effective pressure, specific fuel consumption) and particulates emissions could be correlated with the Conradson Carbon Number, and to a lesser degree, with the asphaltene content.
4. The combustion improvers which were tested showed a positive effect. This was dependent upon the engine load and the type of fuel.

5. ACKNOWLEDGEMENTS

This work was supported by the U.S. Department of Energy, LBL Director's Research Fund, Contract No. DE-AC-03-76 SF00098. Support for D. J. Ruzicka was also provided by the Central Institute for Industrial Research, Oslo, the Lawrence Berkeley Laboratory, and the Royal Norwegian Council for Scientific and Industrial Research.

We would like to express our thanks to J. A. Bert and Jim Clerc of the Chevron Research Company, Richmond, CA, for providing us with the various types of fuels with extensive data and for a pleasant cooperation throughout this work. Thanks are also due to N. A. Henein of Wayne State University for useful discussions and suggestions, and to Hitoshi Takagi of the University of California, Berkeley, for his assistance in calculations and in plotting the figures. The assistance of the technical staff of the Department of Mechanical Engineering, UCB, was also greatly appreciated.

6. REFERENCES

1. Sjöberg, Harry: Heavy Fuel Characteristics and their Influence on Diesel Engines; Wärtsilä Vasa Factory, Finland, 1980.

2. Lake, V. M. et al., Trans. I. Mar. E. 92, Conference No. 5, paper C40 (1979).

3. "Bunker Qualities" in the 1980's and their challenge to Marine Diesel Engines built in the 60's and 70's," International Power Engineering AS, Denmark.

4. Berryman, T. J., Trans. I. Mar. E., 92, Conference No. 5, paper C41 (1979).

5. Van der Horst, G. W. et al., CIMAC 1981, Helsinki.

6. Lichtenthaler, R. G. and Østvold, G., Report No. 780614-3, 1980, Centr. I. Ind. Res., Oslo.

7. ASTM Manual for Rating Diesel Fuels by the Cetane Method, ASTM 1963.

8. Sczomak, D. P. and Henein, N. A., SAE Paper No. 790924, 1979.

9. Brehob, D. D. et al., LBL report No. 14987, 1982.

10. Wolfer, H. H., VDI Forschung, 392 (1938).

11. Henein, N. A. and Elias, N. Y., SAE paper No. 780640, 1978.

12. Benson, R. S. and Whitehouse, N. D., Internal Combustion Engines, Vol. 1, p. 55, Pergamon Press, 1979.

13. Lenda, K., J. Liq. Chrom., 5 (4), 605 (1982).

TABLE 1

Description of residual fuels

No.	Sample Identification	Identification used	Description
1	D-3031	SRR	Atmospheric residuum from Alaska North Slope crude
2	D-3066	FCC	Heavy cycle oil from a fluid catalytic cracker (FCC). This is the highest b.p. fraction from FCC. Contains catalyst fines
3	D-3096	IBF 180	Production intermediate bunker fuel. Residuum from a solvent deasphalt unit and light cycle oil cutter stock
4	D-3143	Vis.R.	Approx. 74% visbroken residuum and 26% light cycle oil
5	D-3144	Vac.R.	Approx. 60% vacuum residuum and 40% production No. 2 diesel. Crudes: Californian, Alaskan and foreign
6	D-3150	Bunker 6	Production No. 6 bunker fuel. Residuum from a solvent deasphalt unit and light cycle oil

TABLE 2

Properties of D-2 Diesel Control Fuel

	Results	EPA Specification*	Test Method
Cetane Number	47.5	42-50	D 613
Distillation Range			
IBP, °F	386	340-400	D 86
10% Point, °F	430	400-460	
50% Point, °F	506	470-540	
90% Point, °F	576	550-610	
End Point, °F	610	580-660	
Gravity, °API	35.8	33-37	D 287
Total Sulfur, wt. %	0.22	0.2-0.5	D 3120
Aromatics (FIA), vol. %	29.1	27 min.	D 1319
Kinematic Viscosity (cs)	2.5	2.0-3.2	D 445
Flash Point (PM), °F	157	130° min.	D 93
Particulate Matter	2.39	-	
Cloud Point, °F	-2	-	D 2500
Elemental Analysis, wt. %		-	
C	86.85		Chromatography
H	13.00		Chromatography
N	0.01		Chemiluminescence
O	0.574		Neutron Activation
C/H	6.68		Calculated

10.0 ptb of DuPont FOA #11 antioxidant enhances the stability of this fuel.

*Diesel Fuel as described in Chapter One - Environmental Protection Agency, Subsection 86.113-78, of the Code of Federal Regulations.

TABLE 3

Properties of Residual Fuels

Test	Test Method	SRR	FCC	IBF180	Vis.R.	Vac.R.	Bunker 6
		Sample Identification No.					
		D 3031	D 3066	D 3096	D 3143	D 3144	D 3150
Density, kg/dm^3, 20°C	Digital Density Meter	0.9477	1.0418	0.9678	1.0003	0.9492	0.9910
Viscosity, cSt at 50°C	D 445	150	99.9	180.5	184.1	182.5	495.5
at 100°C		20.98	10.05	21.79	23.66	22.68	39.15
Conradson Carbon Residue, %	D 524*	8.6	6.3	12.0	21.1	3.3	14.7
Asphaltenes, %	Hot Heptane	1.9	0.9	3.6	12.2	3.3	5.6
Simulated Distillation, °F	Gas Chromatograph						
IBP		315	387	311	291	342	289
5		522	528	431	399	415	405
10		588	618	465	427	447	437
20		664	698	525	479	501	489
30		745	748	578	549	547	549
40		826	785	624	795	596	623
50		920	821	702	1045	647	1021
60		1045	861	1058		932	
70			908				
80			975				
Maximum		1045	1041	1058	1045	1042	1052
% Recovered at Maximum		60.7	87.0	60.1	50.7	65.8	51.2
C/H/N, %	Carlo Erba						
Carbon		85.99	90.22	87.33	85.23	86.02	86.10
Hydrogen		11.14	8.72	10.91	9.92	11.51	9.31
Nitrogen		0.42	0.44	0.95	0.33	0.73	1.13
Aliphatic/Aromatic/Polar, %	D 2007						
Aliphatic		36.3		22.4	11.1	31.0	15.6
Aromatic		29.7	Not Applicable	19.9	22.2	19.3	23.0
Polar		27.0		38.9	26.7	32.6	41.9
Insolubles		5.4		8.9	18.0	8.8	15.2
Recovered		98.4		90.1	78.0	91.7	95.7

*Carbon residue measured by D 524 and converted to Conradson Carbon Residue (D 189).

Data provided by Chevron Research Company

TABLE 4

Heats of Combustion

D-3031 SRR	43.68 MJ/kg
D-3066 FCC	41.77 MJ/kg
D-3096 IBF 180	42.94 MJ/kg
D-3143 Visbr. R.	41.87 MJ/kg
D-3144 Vac. R.	43.59 MJ/kg
D-3150 Bunker 6	42.29 MJ/kg
α-Me-naphthalene	38.84 MJ/kg
D-2 Diesel Control Fuel	45.35 MJ/kg

TABLE 5

Optimization of CFR Engine for Intermediate Bunker Fuel 180 D-3161

Optimization of:	Fig. No.	Other parameters				
		Inj.P M Pa	Inj. timing ° bTDC	C.R.	Air T °C	Inj. coolant T° C
Injection Pressure	3	-	18	23.5	110	55
Injection Timing	4	12.7	-	23.5	110	55
Compr. Ratio	5	12.7	21	-	110	55
Air Temp.	6	12.7	16	19.0	-	55
Injector Coolant	7	12.7	16	19.0	121	-
Parameters used in comparative studies		12.7	11	19.0	121	54→40

TABLE 6

Results for Different Fuels

		Ignition Delay	Engine performance (indicated)		Emissions (indicated)		
Fuel Type	C/H	msec	Sfc g/MJ	Mep kPa	Particulates g/MJ	HC g/MJ	CO g/MJ
D-3096 IBF180	8.00	0.89	69.6	626	2.0	0.35	1.3
D-3144 Vac.R.	7.47	0.93	67.8	622	1.6	0.32	1.1
D-3143 Visbr.R.	8.59	0.96	78.6	584	3.5	0.44	1.3
D-3031 SRR	7.24*	0.67	69.2	628	1.9	0.28	0.83
D-3031 SRR +12% -MN	7.94*	0.80	68.9	636	1.9	0.28	0.96
D-3031 SRR +23% -MN	8.59*	0.83	69.2	641	1.9	0.33	1.0
D-3150 Bunker 6	9.25	1.1	72.6	607	2.5	0.53	2.4
D-2 Diesel Control Fuel	6.68	0.53	63.2	651	0.83	0.14	0.37

*average of Chevron and U.C. Berkeley values

TABLE 7

Ranking of Fuels

Rating of fuel	I.D.	Sfc	Mep	Particulates	HC	CO
1	D-2	D-2	D-2	D-2	D-2	D-2
2	SRR	Vac. R.	SRR	Vac. R.	SRR	SRR
3	IBF 180	SRR	IBF 180	SRR	Vac. R.	Vac. R.
4	Vac. R.	IBF 180	Vac. R.	IBF 180	IBF 180	IBF 180
5	Vis. R.	Bunker 6	Bunker 6	Bunker 6	Vis. R.	Vis. R.
6	Bunker 6	Vis. R.	Vis. R.	Vis. R.	Bunker 6	Bunker 6

TABLE 8

Combustion Quality vs. Fuel Properties

○ = good correlation ◑ = some correlation ● = poor correlation

Engine Performance Parameters	FUEL PARAMETERS					
	C/H	aliphatics	aromatics	asphaltenes	CCR	Ashpaltenes +aromatics +polars
I.D.	○	◑	●	●	●	●
imep	●			○	○	◑
isfc	◑			○	○	◑
bsfc	●			○	○	◑
<u>Emissions</u>: (indicated)						
HC	○			●	◑	
CO	○			●	●	
Particulates	●			○	○	

TABLE 9

Effect of Charge Air Temperature on the Performance with D-3096

Charge air temperature °C	I.D. msec	Torque	HC emissions	Particulates emission
135	0.89	slight increase	constant	constant
104	0.89			
83	0.94			
63	1.00			
52	1.08			

TABLE 10

Effect of Combustion Improvers

Fuel Type	Injection timing °BTDC	Ignition Delay msec	Engine Performance (indicated)		Emissions (indicated)		
			Sfc g/MJ	Mep kPa	Particulates g/MJ	HC g/MJ	OC g/MJ
D-3150 Bunker 6	11	1.1	72.6	607	2.5	0.53	2.4
D-3150 Bunker 6 + Improver I	13.5	0.96	71.1	622	1.7	0.40	1.5
D-3150 Bunker 6 + Improver II	13.5	0.93	69.6	628	1.3	0.33	1.1
76% load:							
D-2 Diesel Contr. F	11	0.76	64.4	647	1.2	0.13	0.32
D-2 Diesel Contr. F + Improver I	11	0.63	64.4	653	1.5	0.13	0.27
D-2 Diesel Contr. F + Improver II	11	0.63	64.8	645	1.3	0.11	0.27
97% load:							
D-2 Diesel Contr. F	11	-	75.2	703	3.4	0.46	1.9
D-2 Diesel Contr. F + Improver I	11	-	73.3	726	3.2	0.29	1.5
D-2 Diesel Contr. F + Improver II	11	-	72.6	726	1.9	0.26	0.96

TABLE 11

Effect of Combustion Improvers

Relative values listed: $\frac{\text{fuel with improver}}{\text{neat fuel}}$

Fuel Type	Improver I						Improver II					
	I.D.	Sfc	Mep	Part	HC	CO	I.D.	Sfc	Mep	Part	HC	CO
Bunker 6	0.87	0.979	1.02	0.68	0.75	0.62	0.85	0.959	1.03	0.52	0.62	0.46
D-2 Diesel Contr. F. 76% load	0.83	1.00	1.01	1.25	1.00	0.84	0.83	1.00	1.00	1.08	0.85	0.84
D-2 Diesel Contr. F. 97% load	-	0.974	1.03	0.94	0.63	0.79	-	0.965	1.03	0.56	0.57	0.51

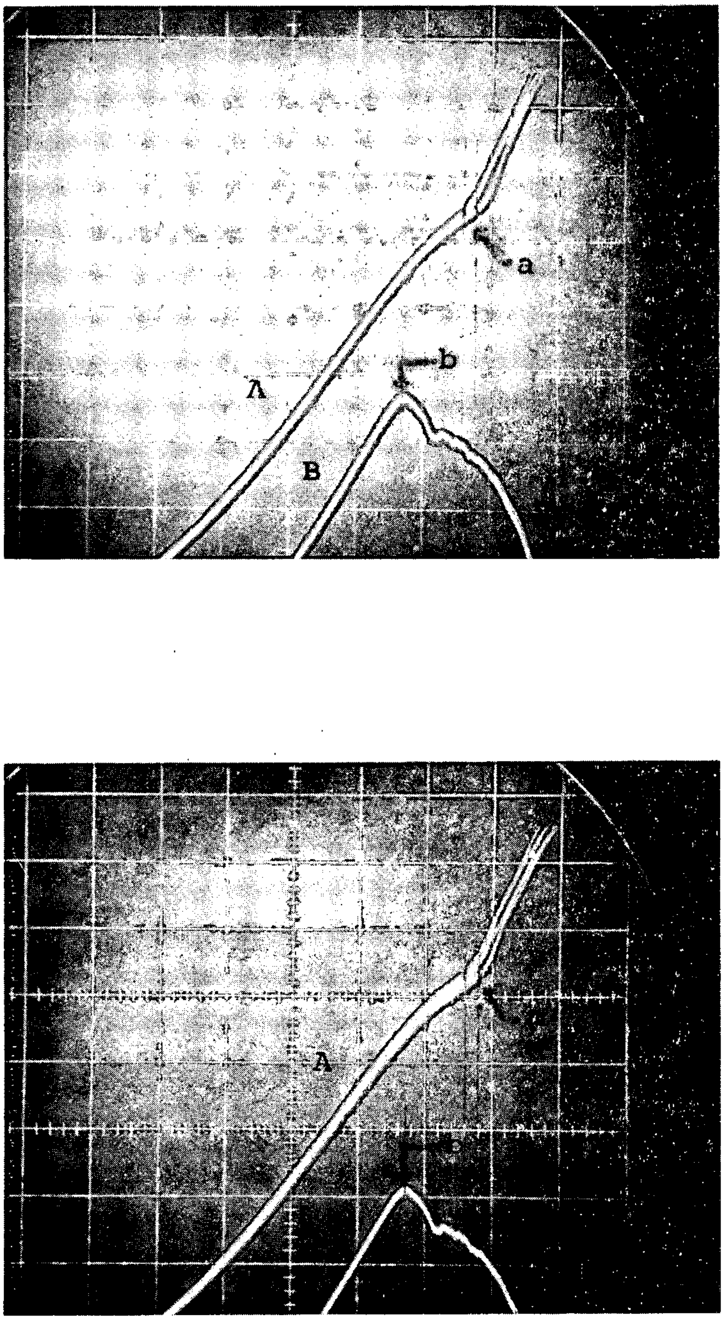

FIG. 1 BUNKER 6, IGNITION DELAY 1.0-1.2 MSEC.
TIME SCALE: 6° PER DIVISION.
A-COMBUSTION CHAMBER PRESSURE; a-IGNITION;
B-INJECTOR PRESSURE; b-BEGINNING OF INJECTION.

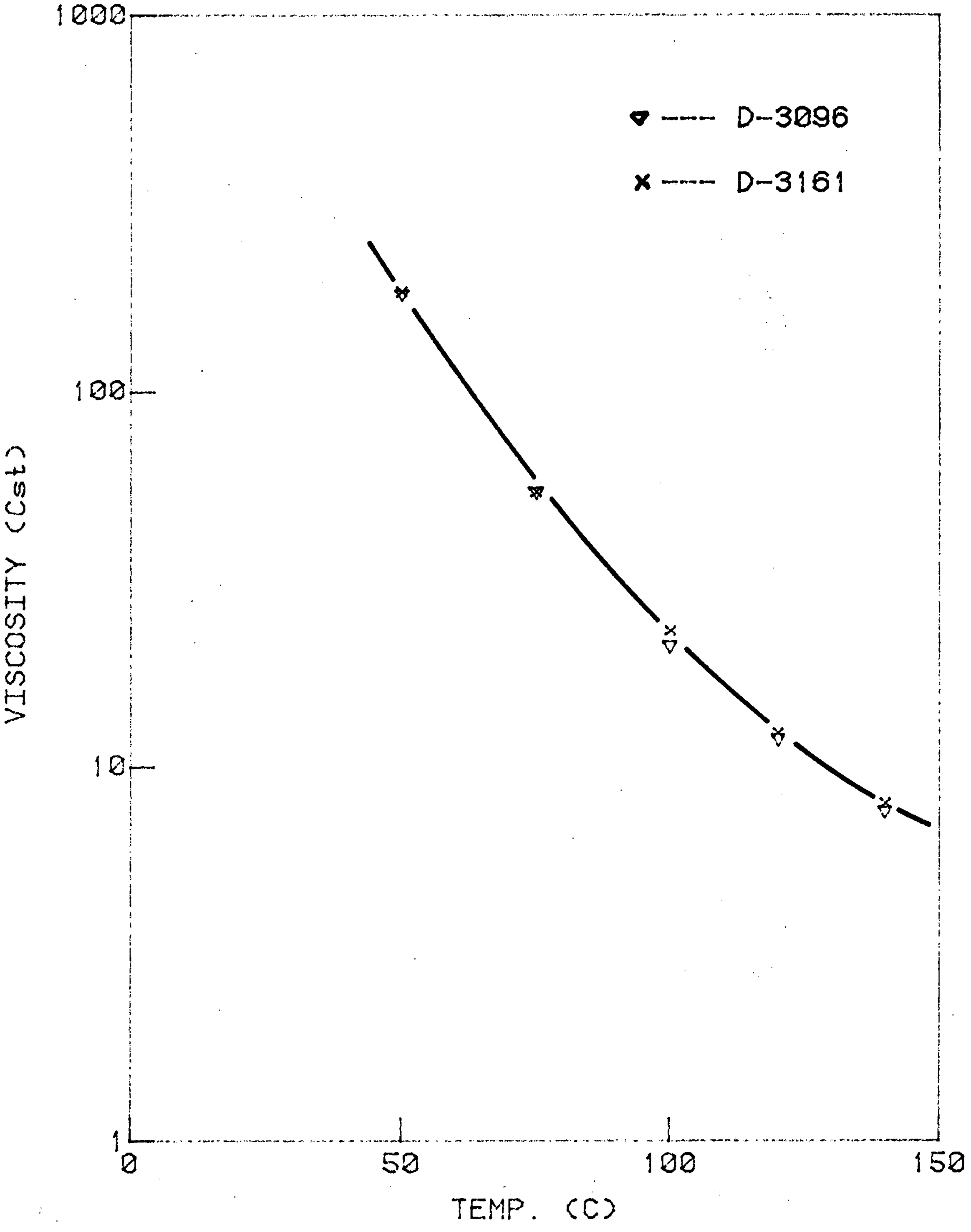

FIG. 2 VISCOSITIES OF TWO COMMERCIAL FUELS (IBF 180).

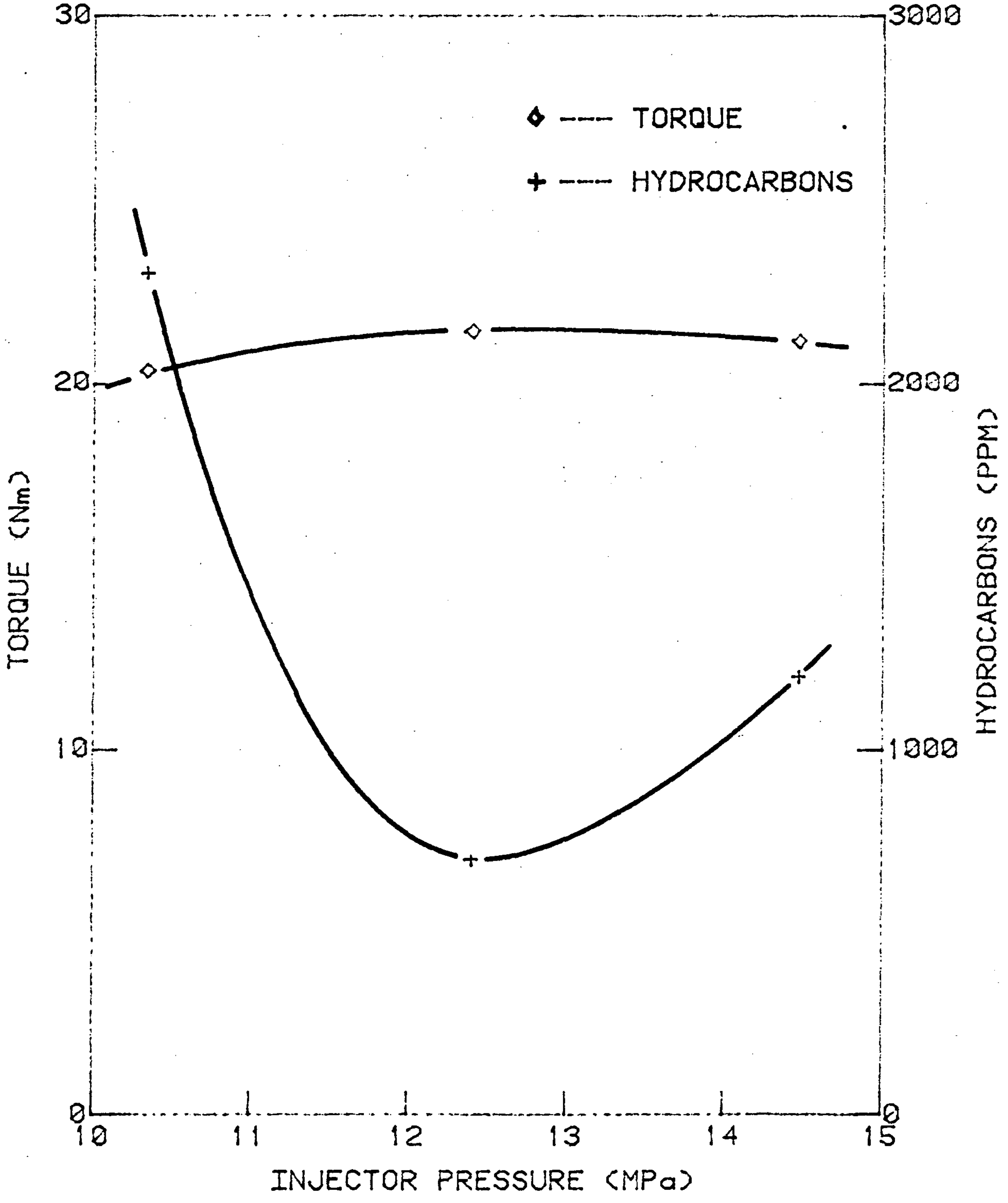

FIG. 3 OPTIMIZATION OF THE INJECTOR PRESSURE.

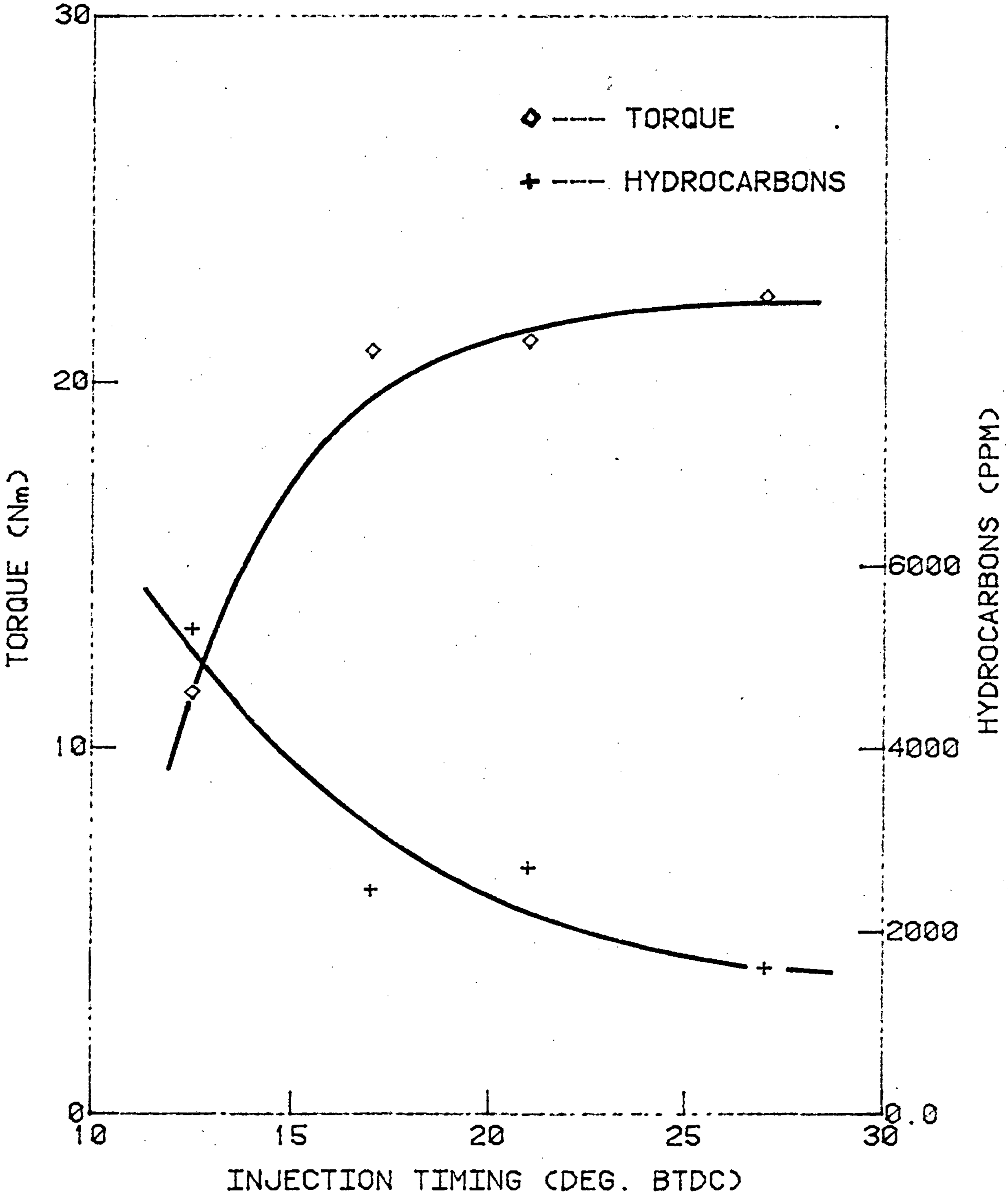

FIG. 4 OPTIMIZATION OF THE INJECTION TIMING.

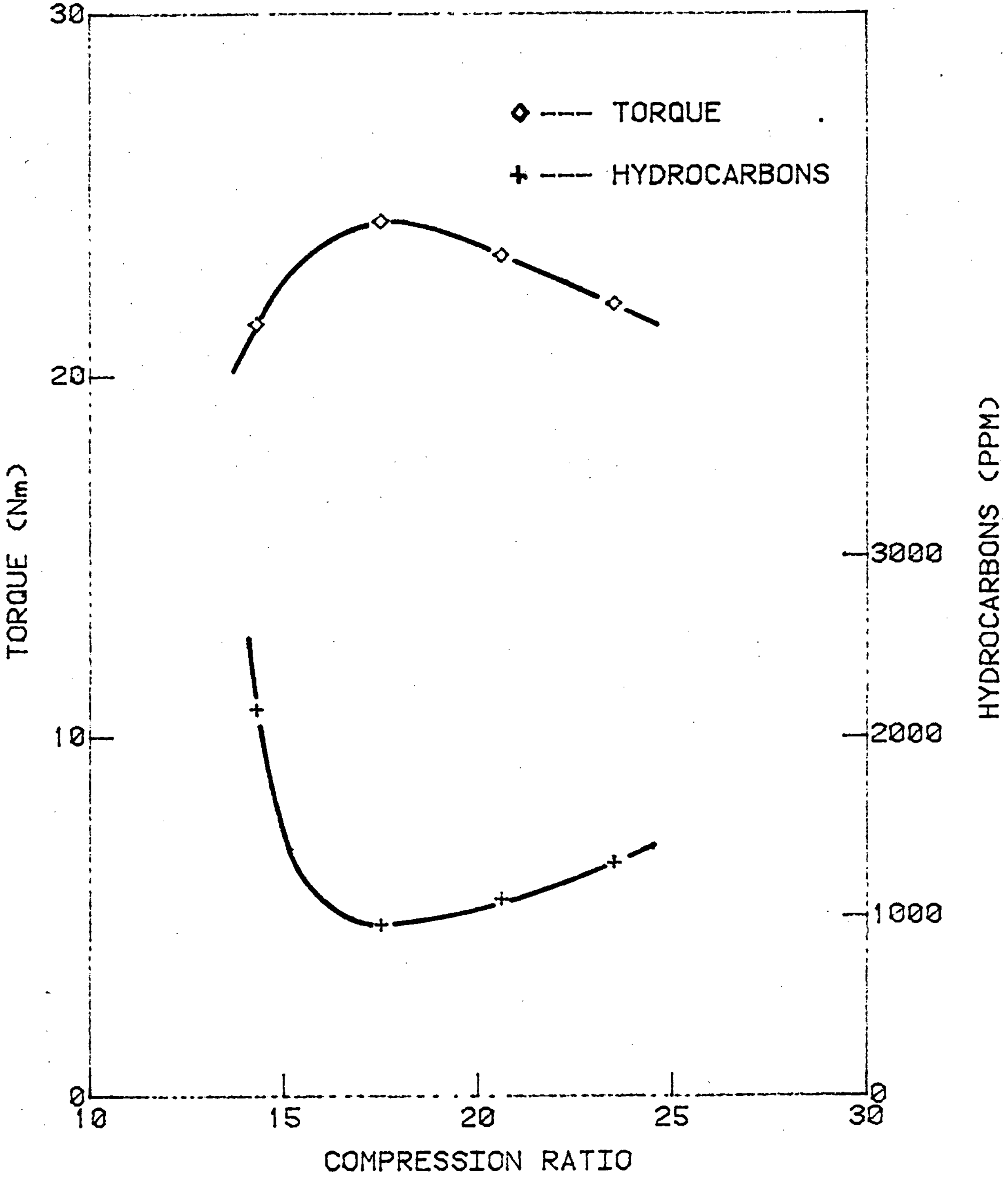

FIG. 5 OPTIMIZATION OF THE COMPRESSION RATIO.

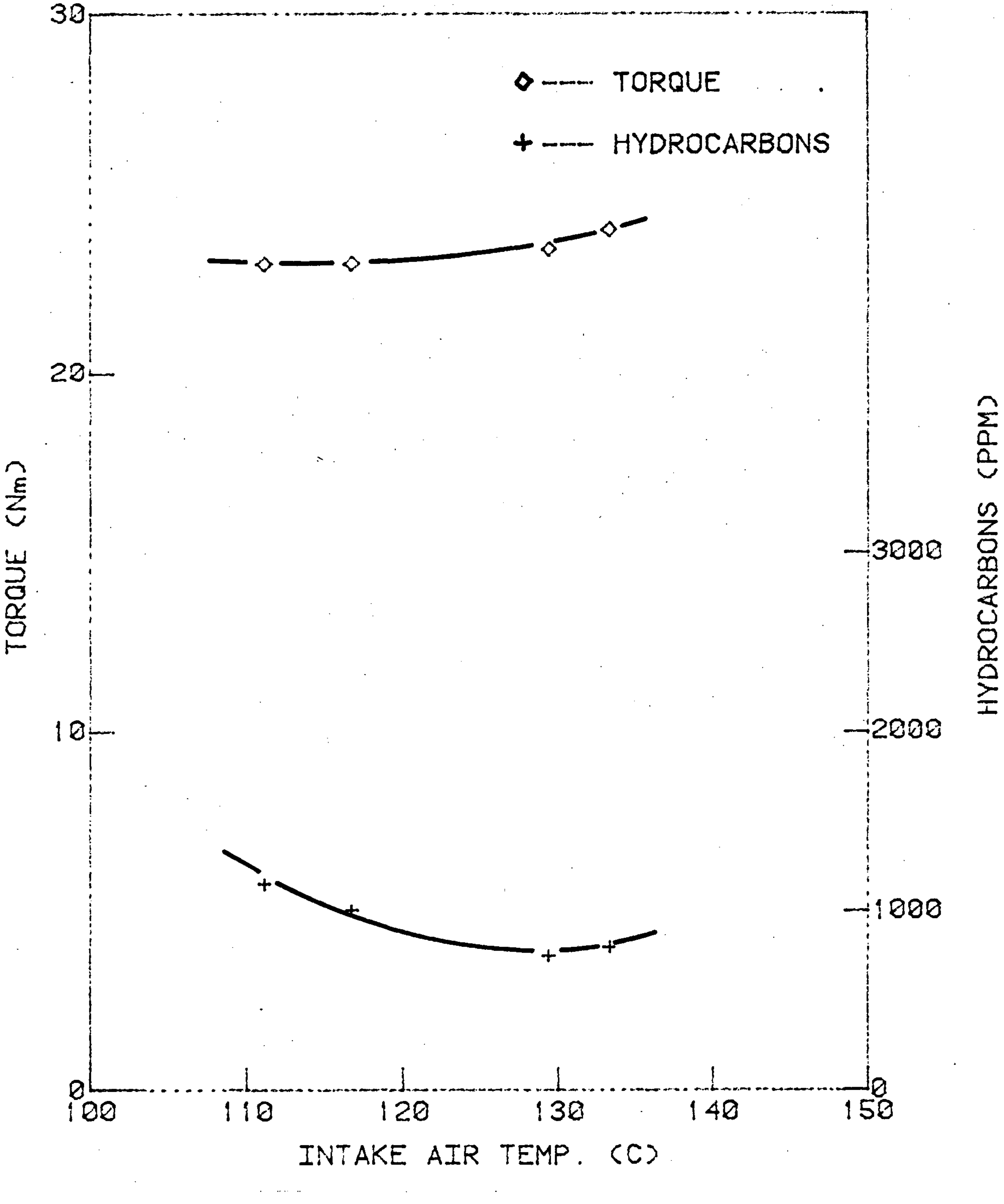

FIG. 6 OPTIMIZATION OF THE INTAKE AIR TEMPERATURE.

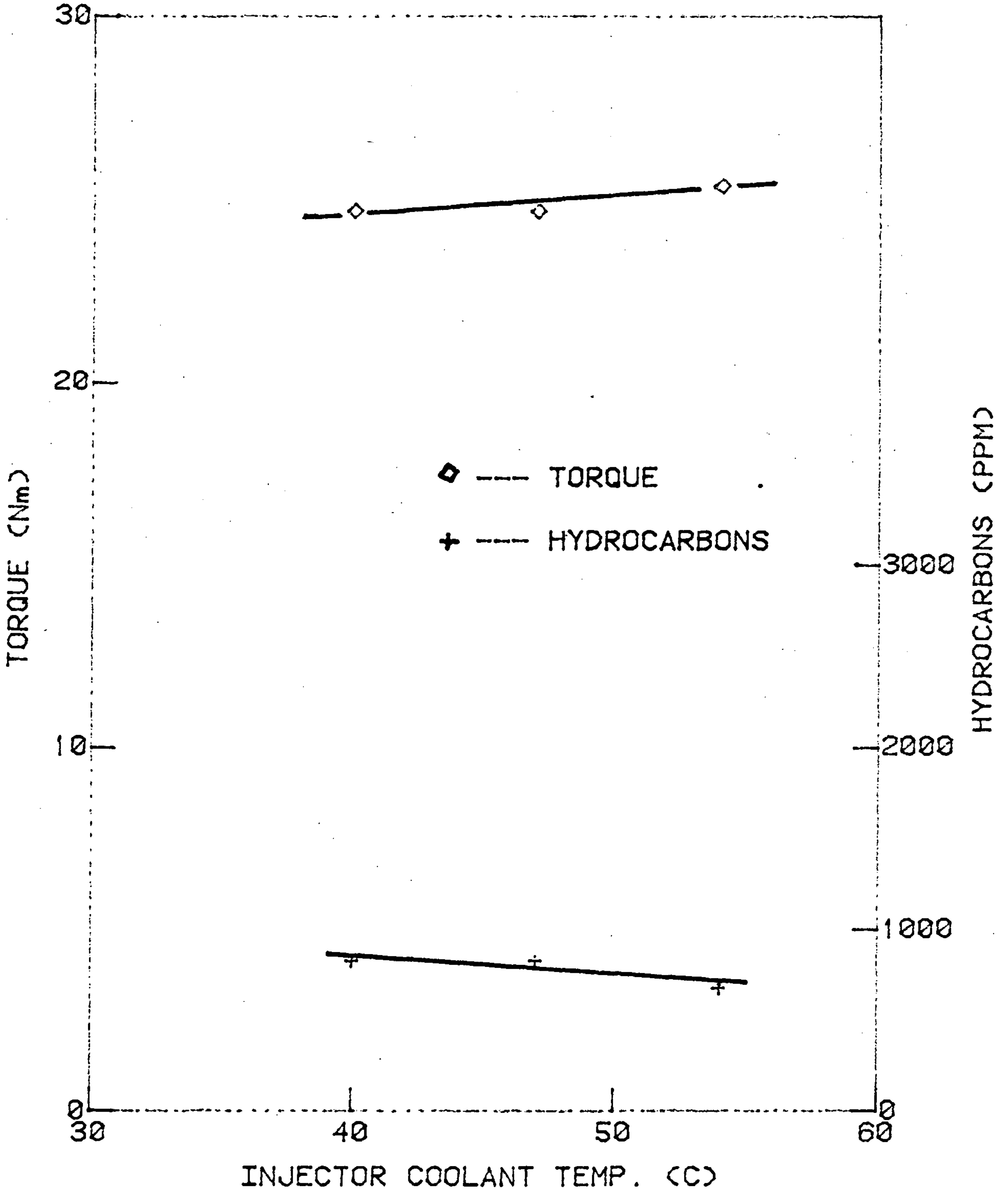

FIG. 7 INFLUENCE OF THE INJECTOR COOLANT TEMPERATURE ON TORQUE AND HC EMISSIONS.

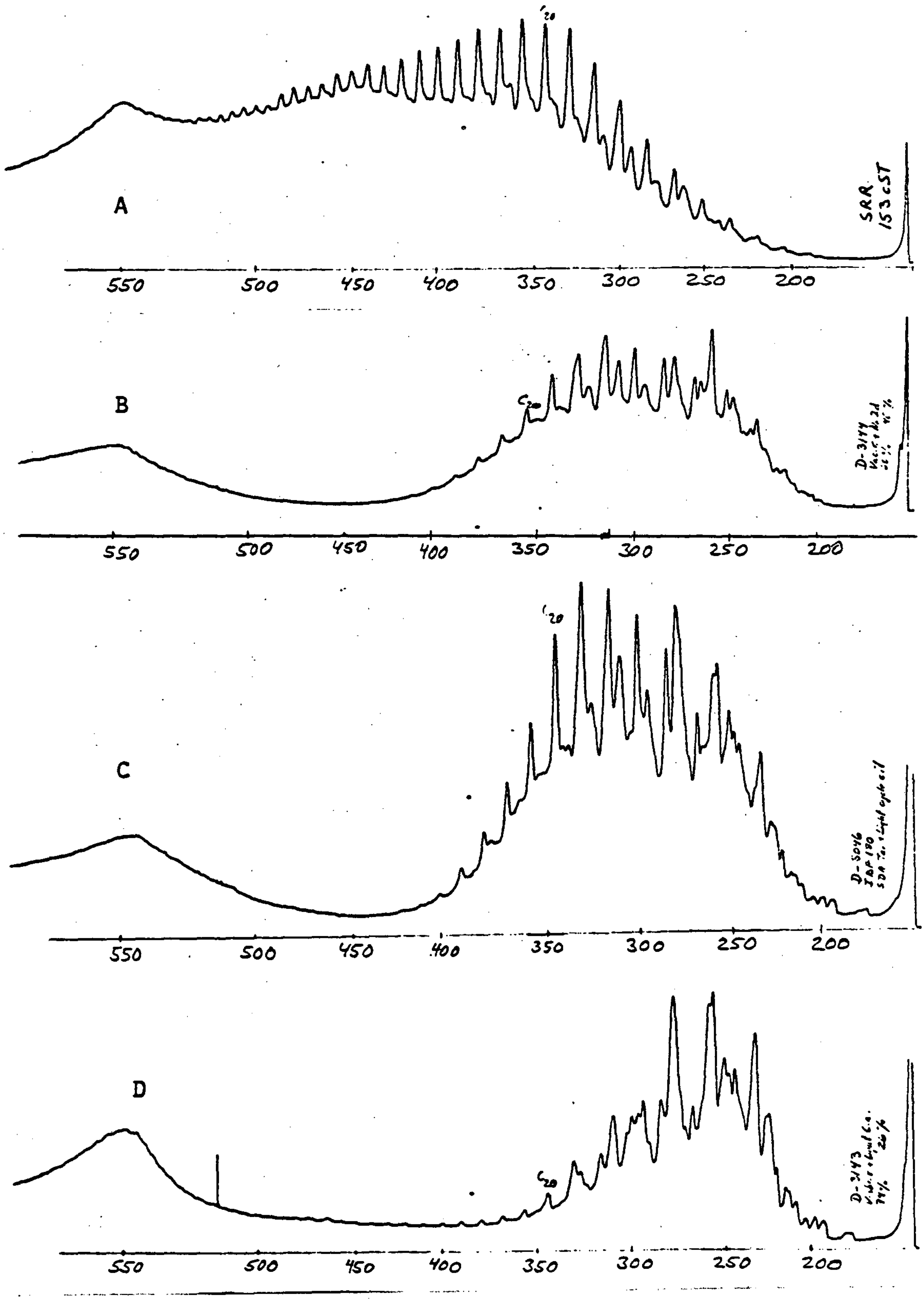

FIG. 8 GAS CHROMATOGRAM BOILING POINT DISTRIBUTIONS:
A. SRR, B. VAC.R., C. IBF 180, D. VIS.R.

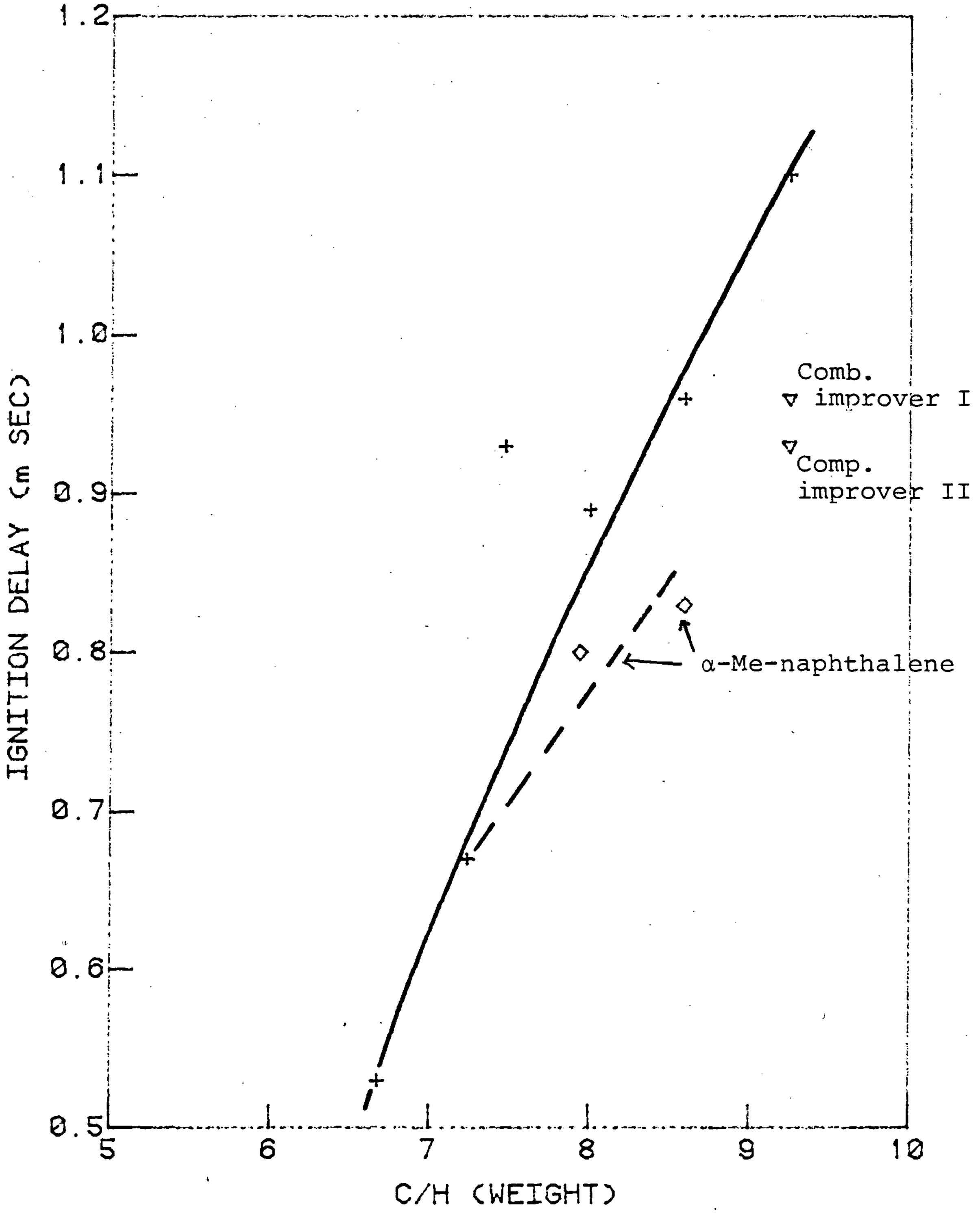

FIG. 9 IGNITION DELAY VS. C/H.

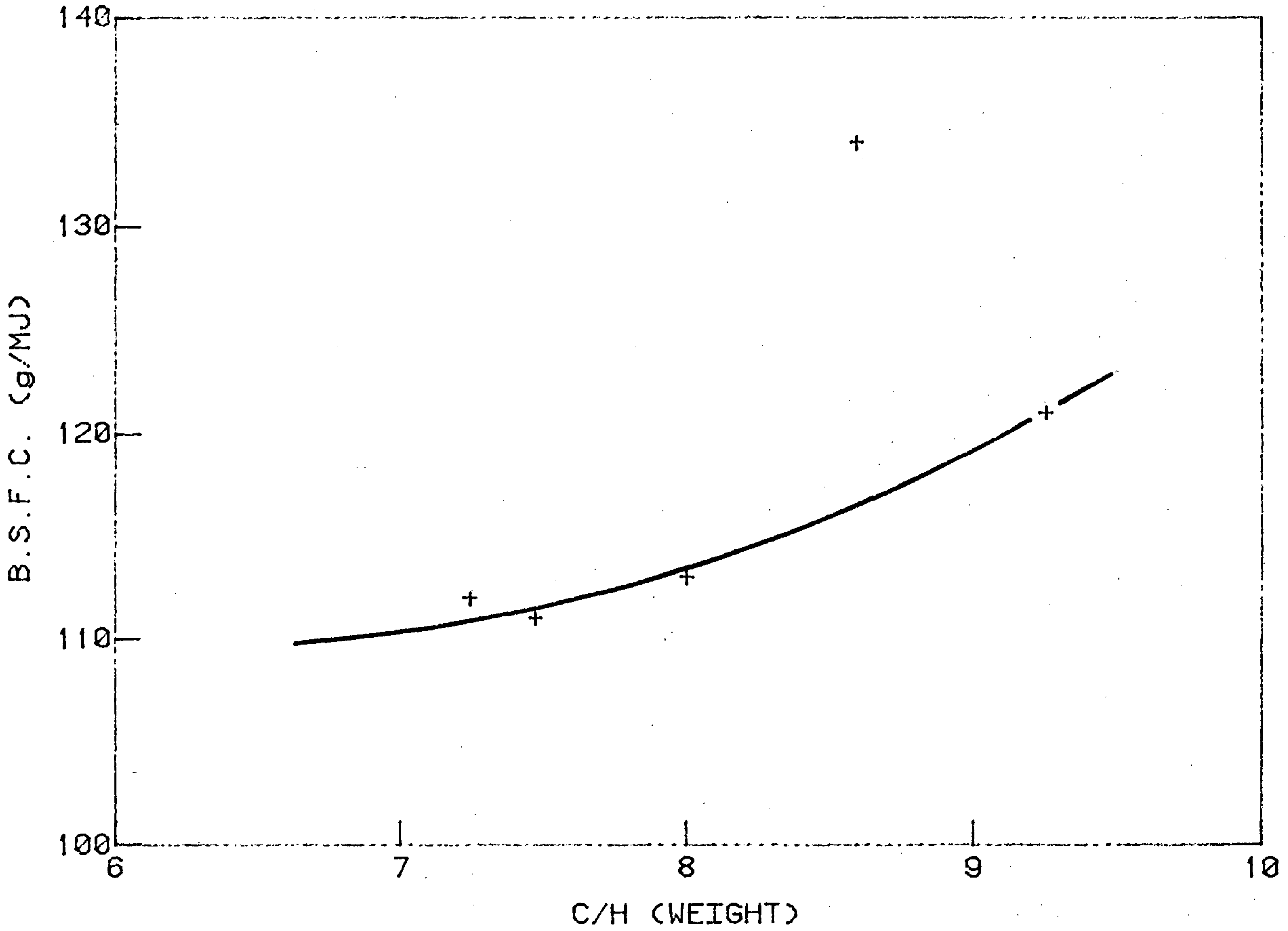

FIG. 10 BRAKE SPECIFIC FUEL CONSUMPTION VS. C/H.

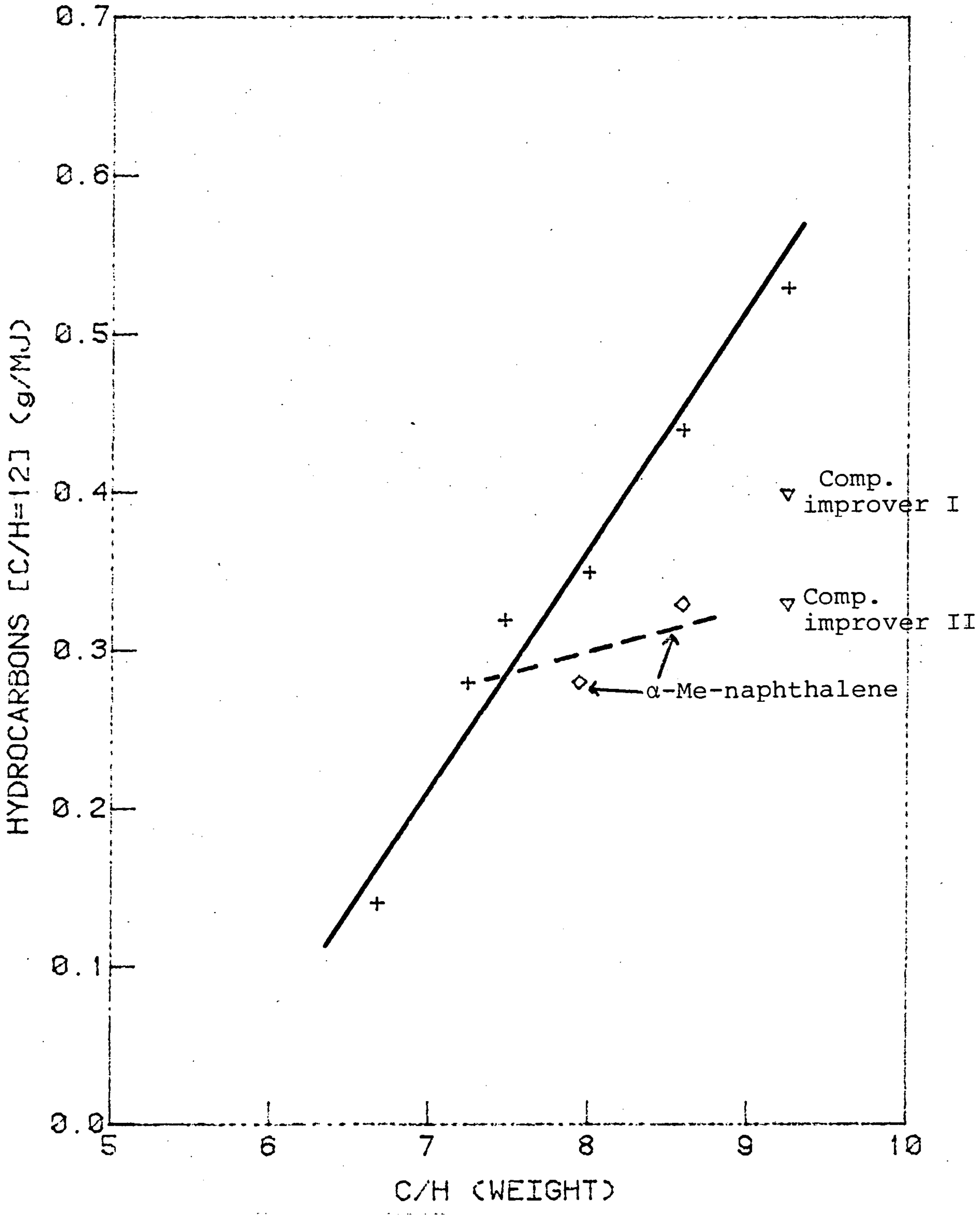

FIG. 11 HYDROCARBONS' EMISSION VS. C/H.

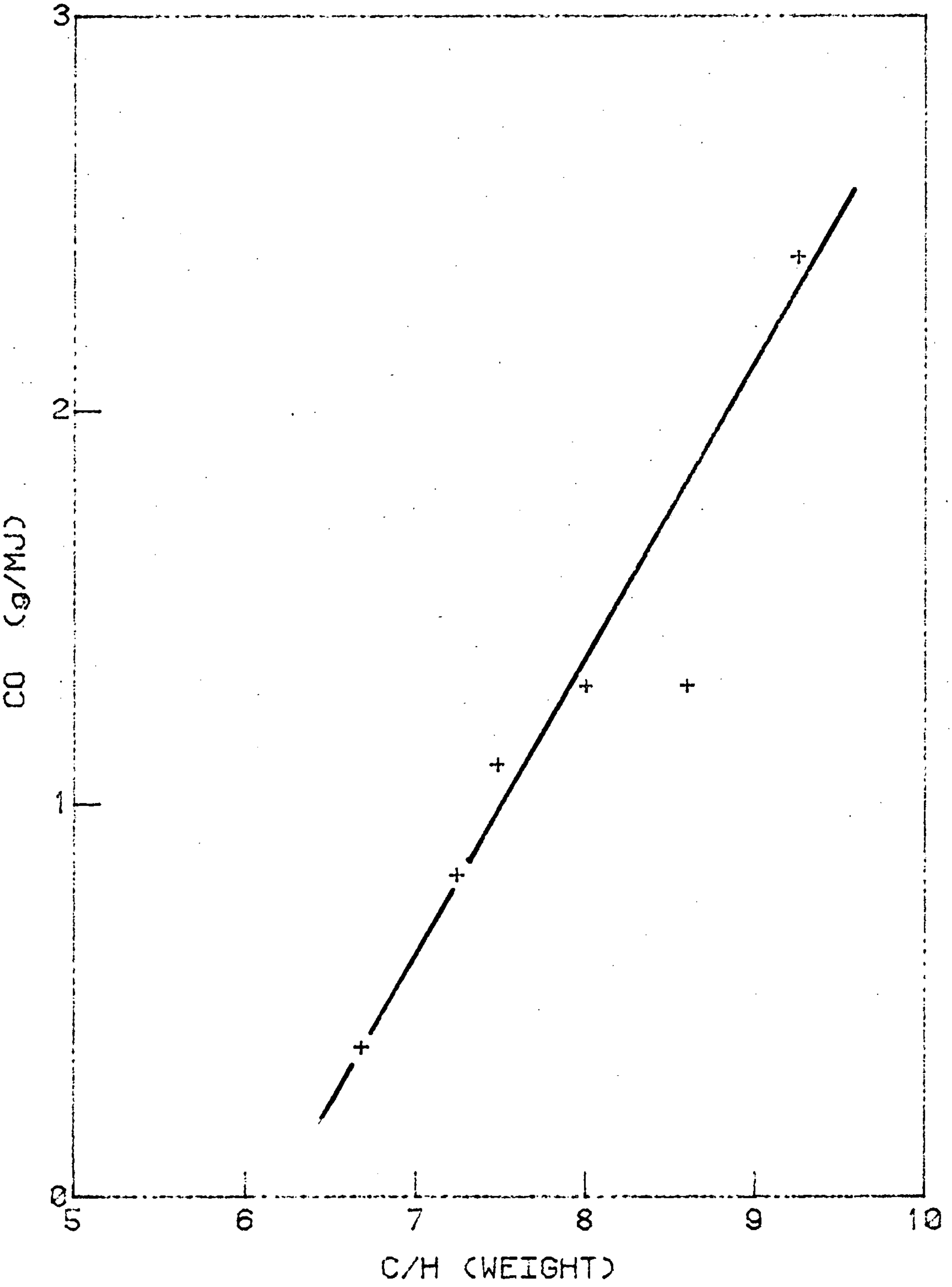

FIG. 12 CO EMISSION VS. C/H.

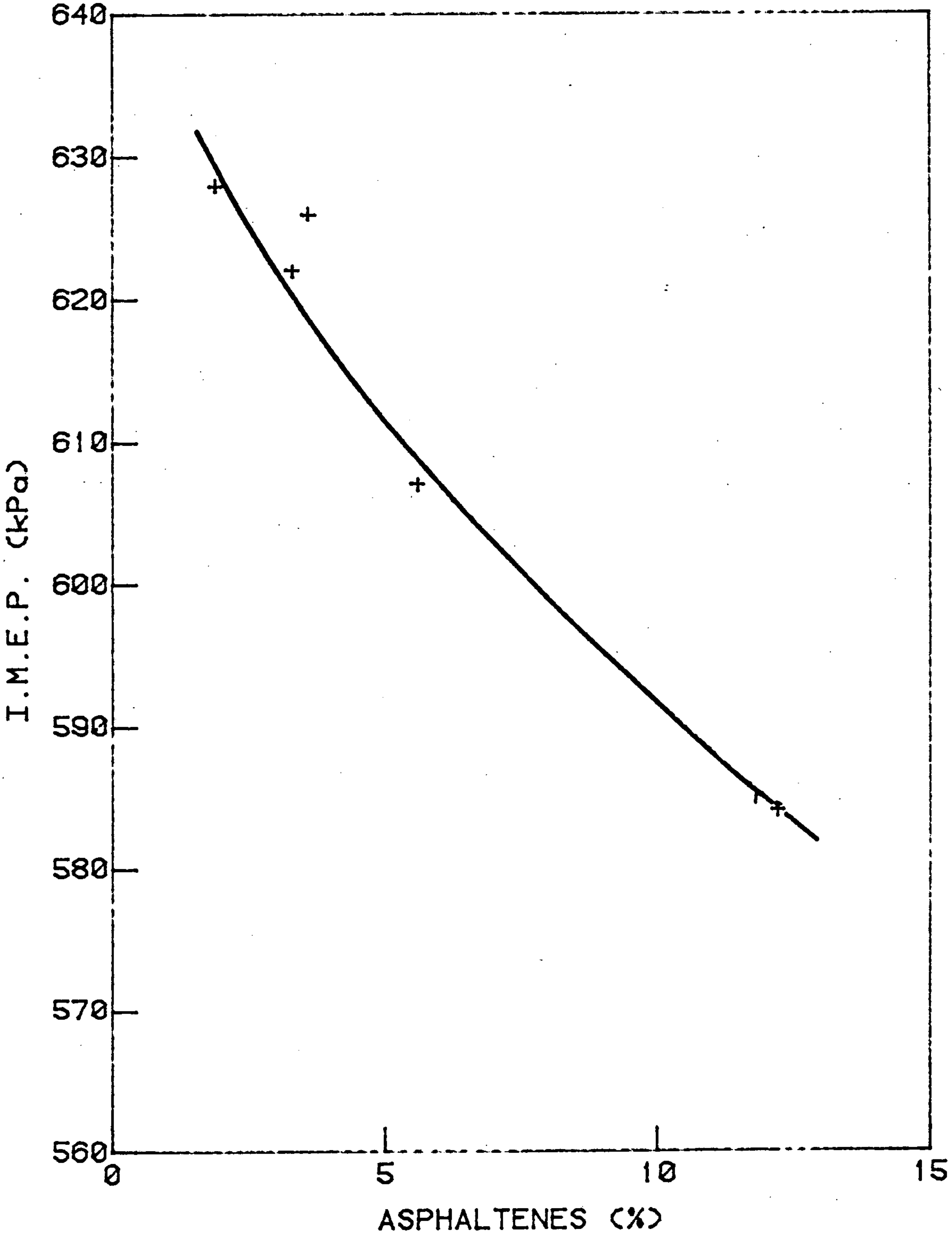

FIG. 13 INDICATED MEAN EFFECTIVE PRESSURE VS. ASPHALTENES.

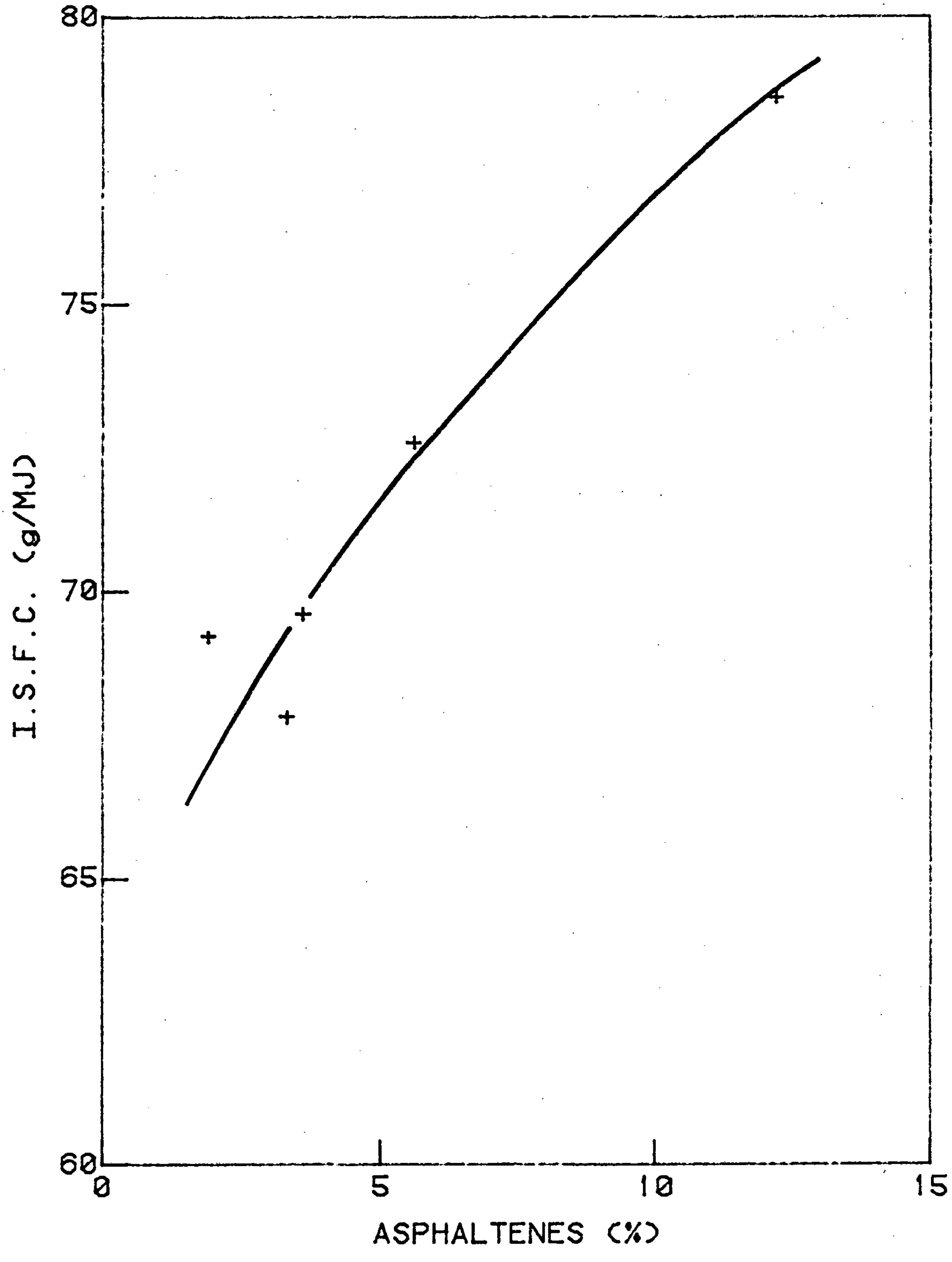

FIG. 14 INDICATED SPECIFIC FUEL CONSUMPTION VS. ASPHALTENES.

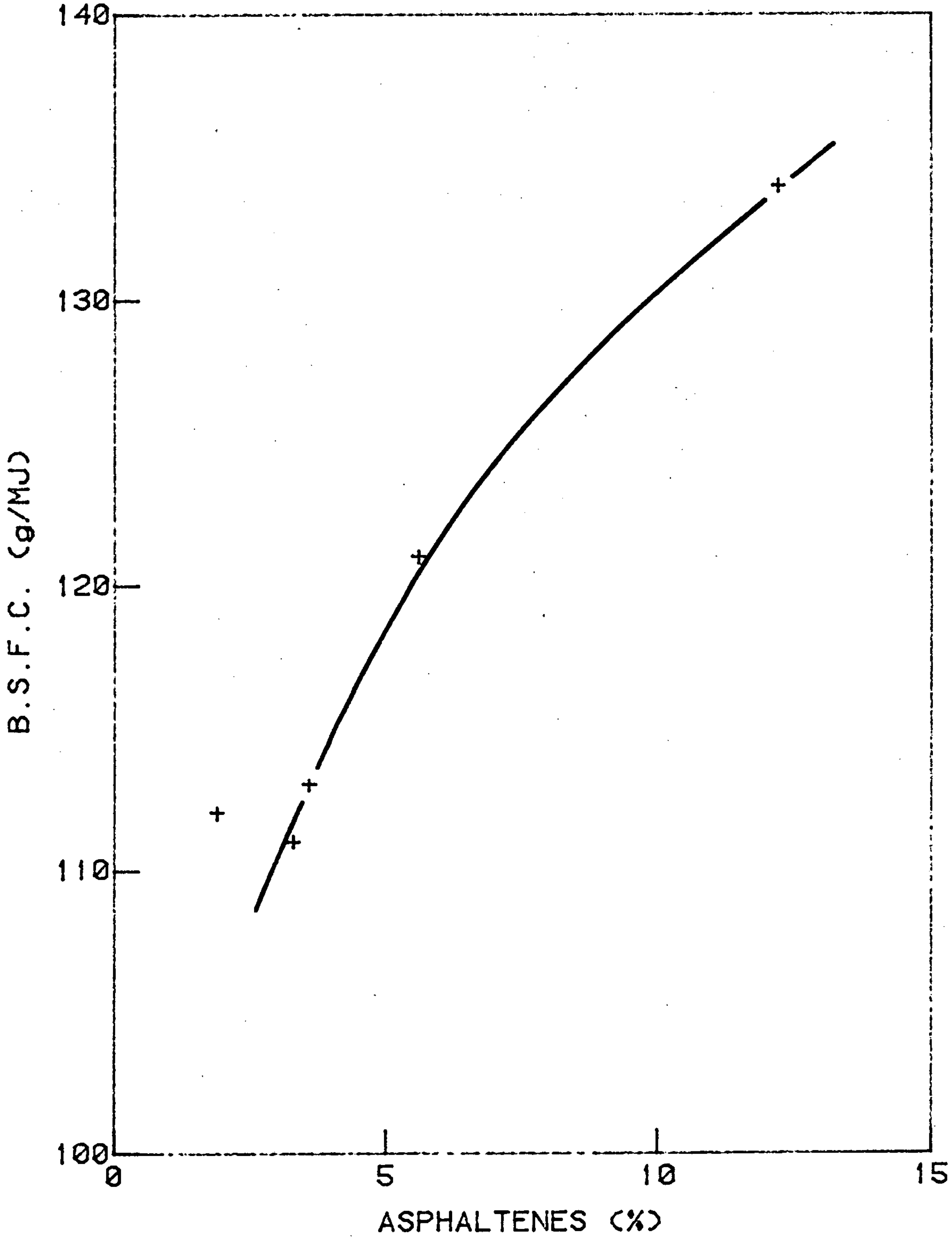

FIG. 15 BRAKE SPECIFIC FUEL CONSUMPTION VS. ASPHALTENES.

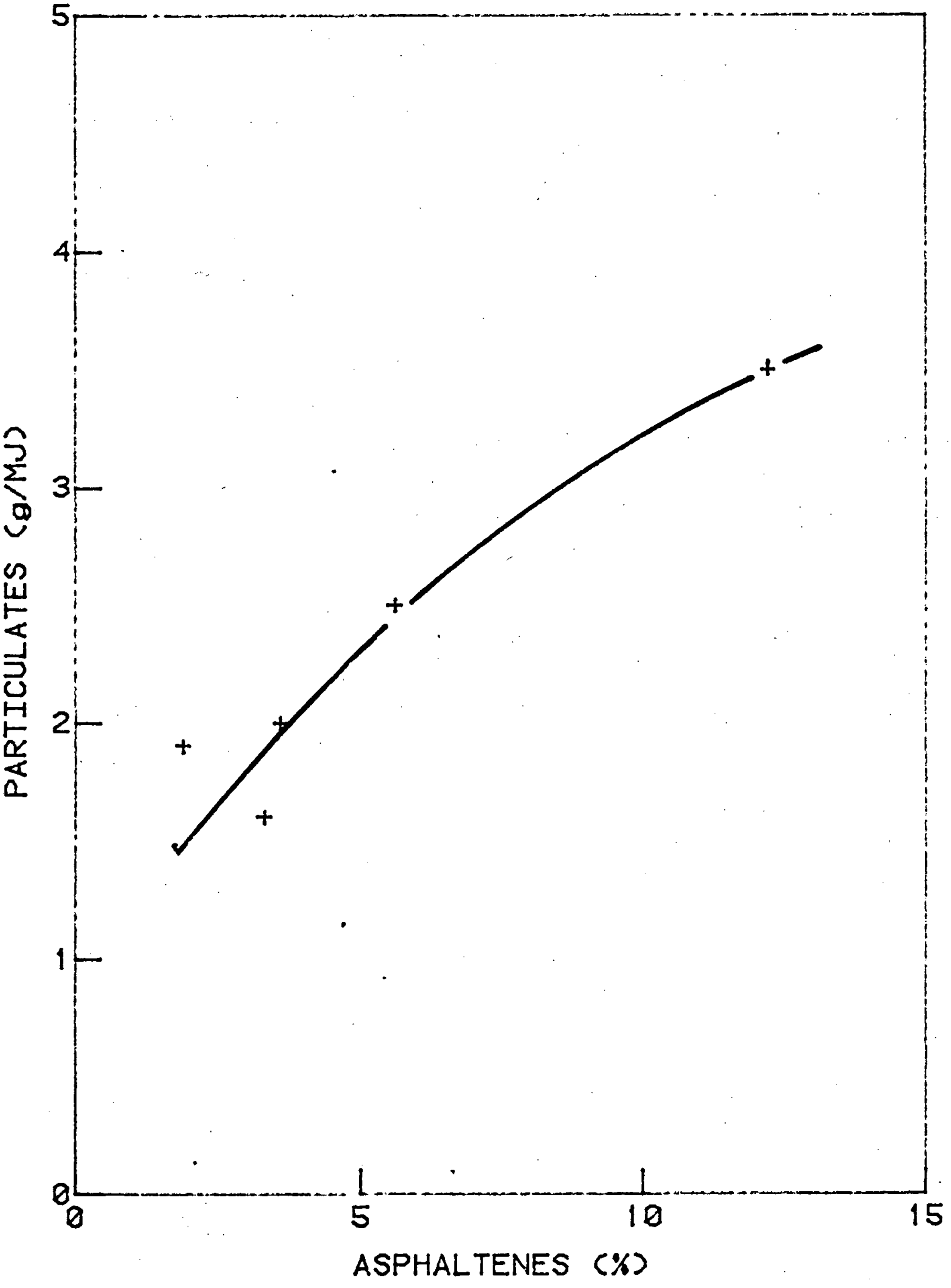

FIG. 16 PARTICULATES' EMISSIONS VS. ASPHALTENES.

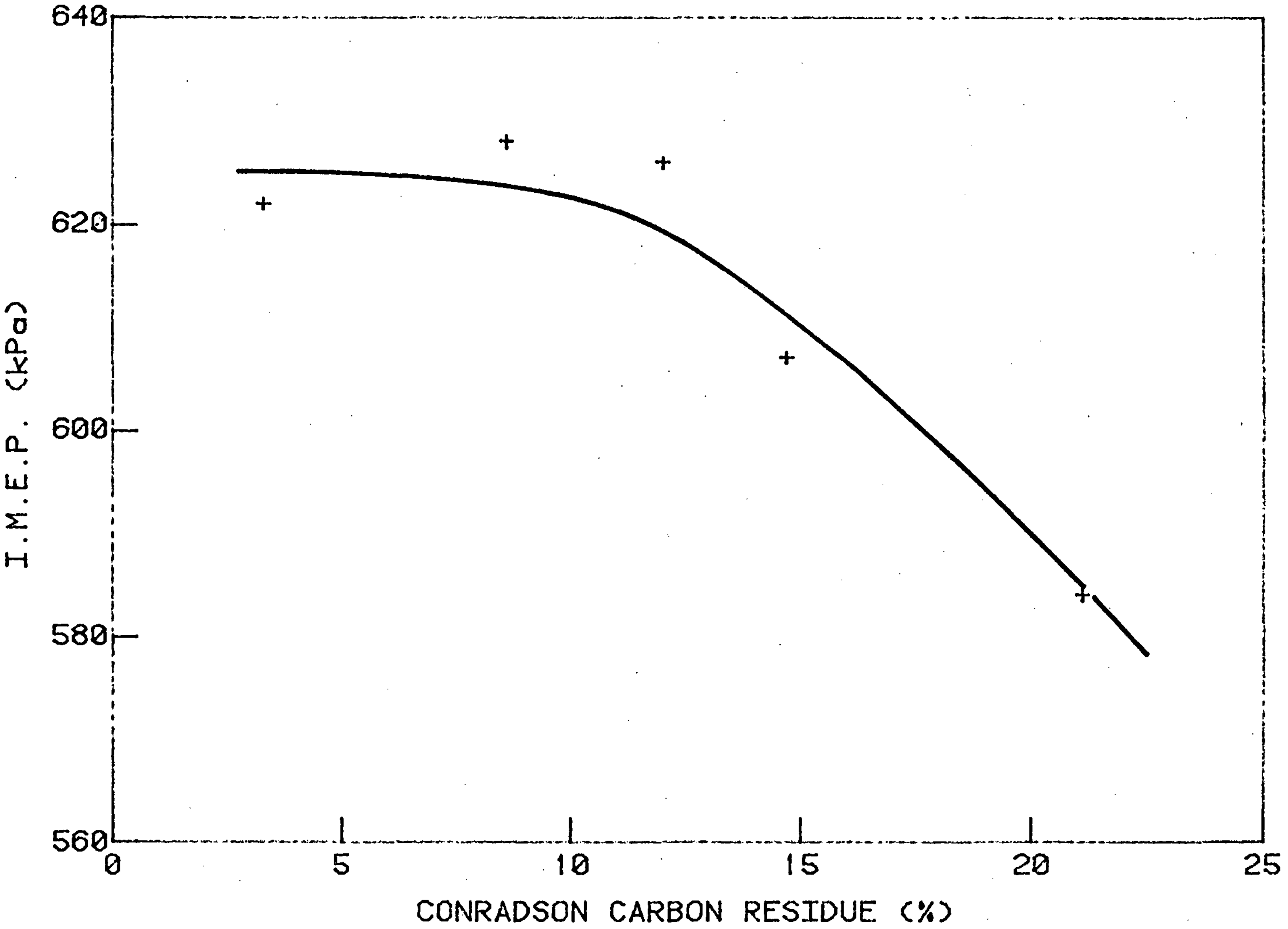

FIG. 17 INDICATED MEAN EFFECTIVE PRESSURE VS. CONRADSON CARBON RESIDUE.

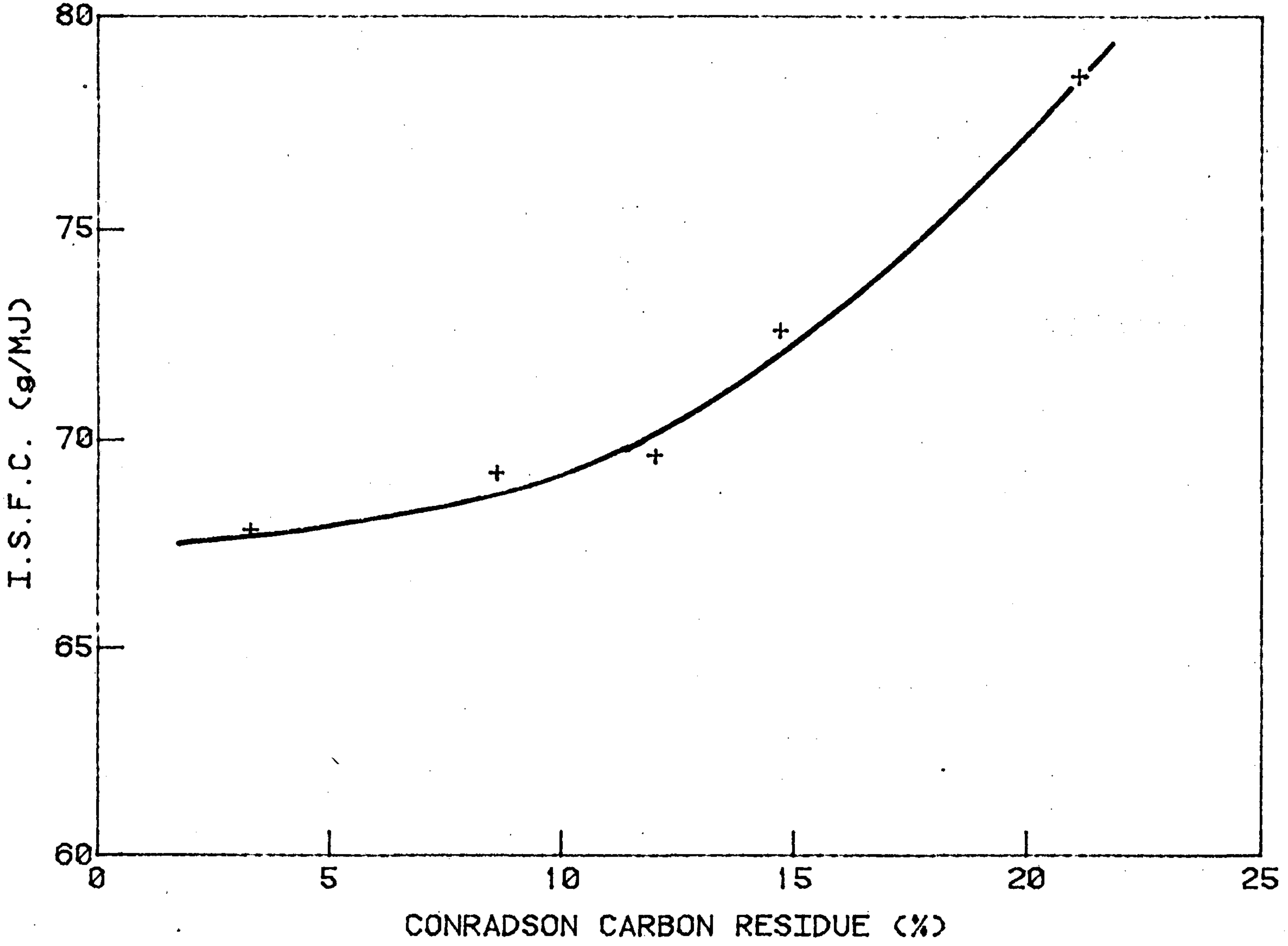

FIG. 18 INDICATED SPECIFIC FUEL CONSUMPTION VS. CONRADSON CARBON RESIDUE.

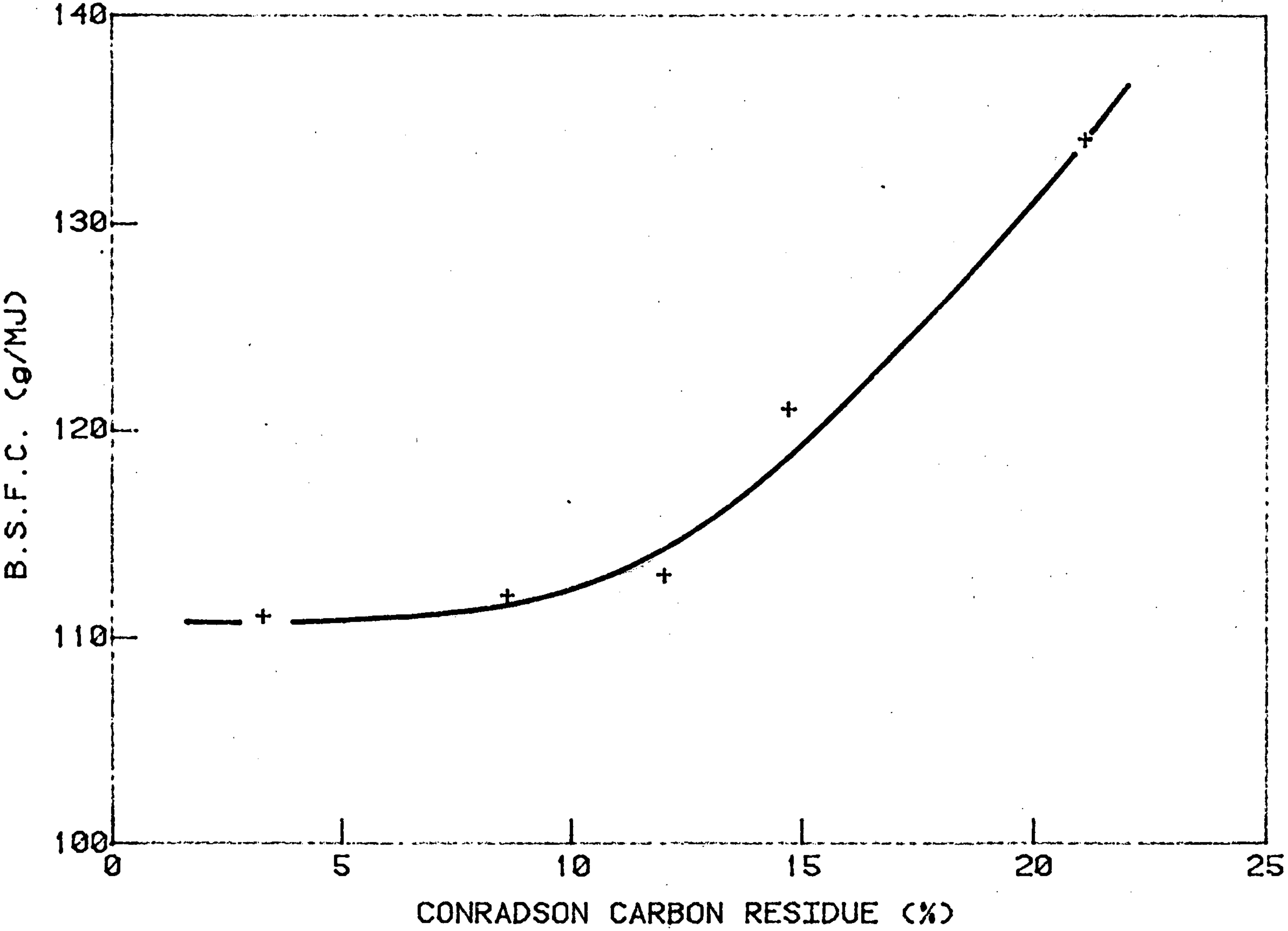

FIG. 19 BRAKE SPECIFIC FUEL CONSUMPTION VS. CONRADSON CARBON RESIDUE.

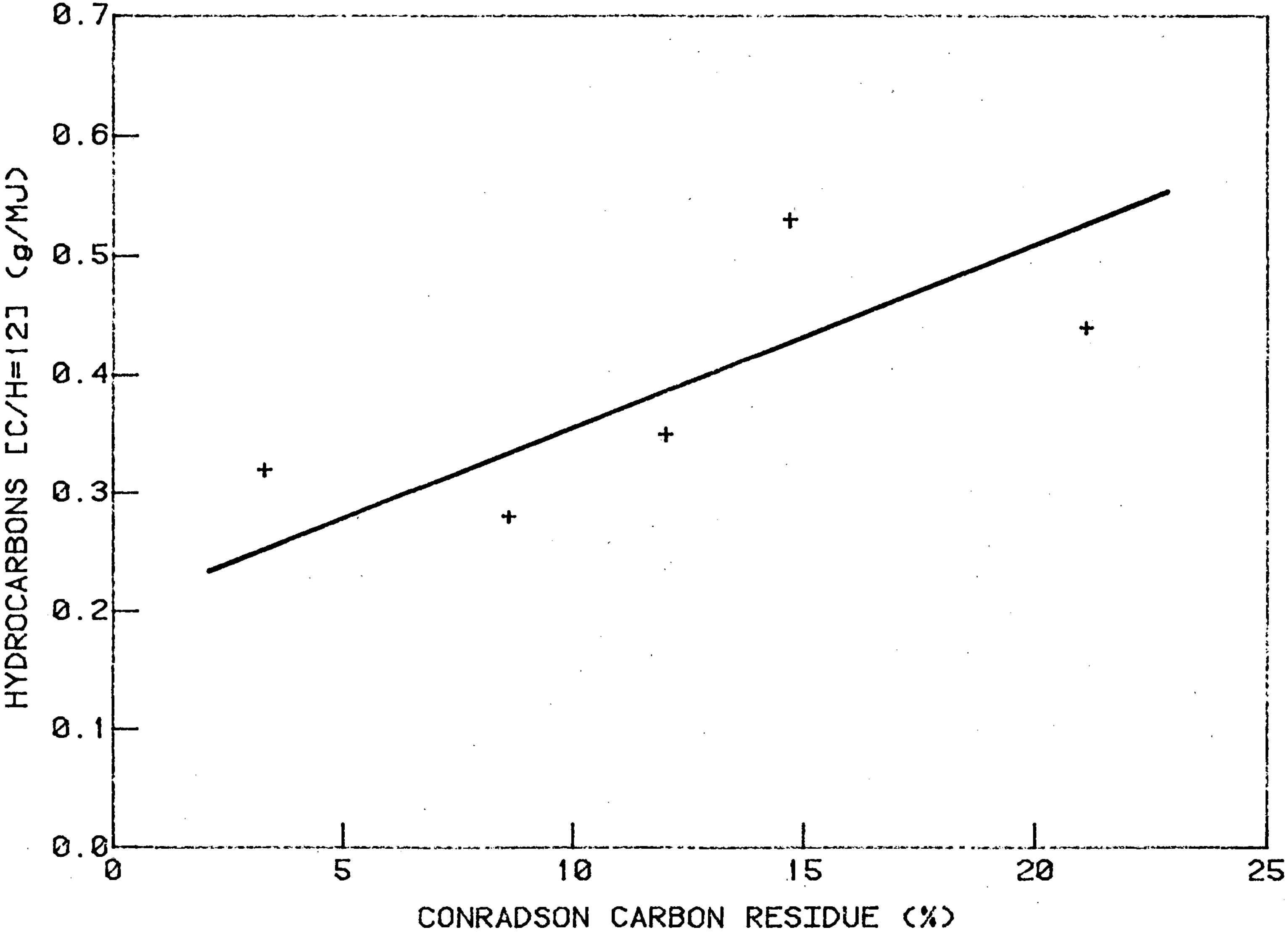

FIG. 20 Hydrocarbons' Emission vs. Conradson Carbon Residue.

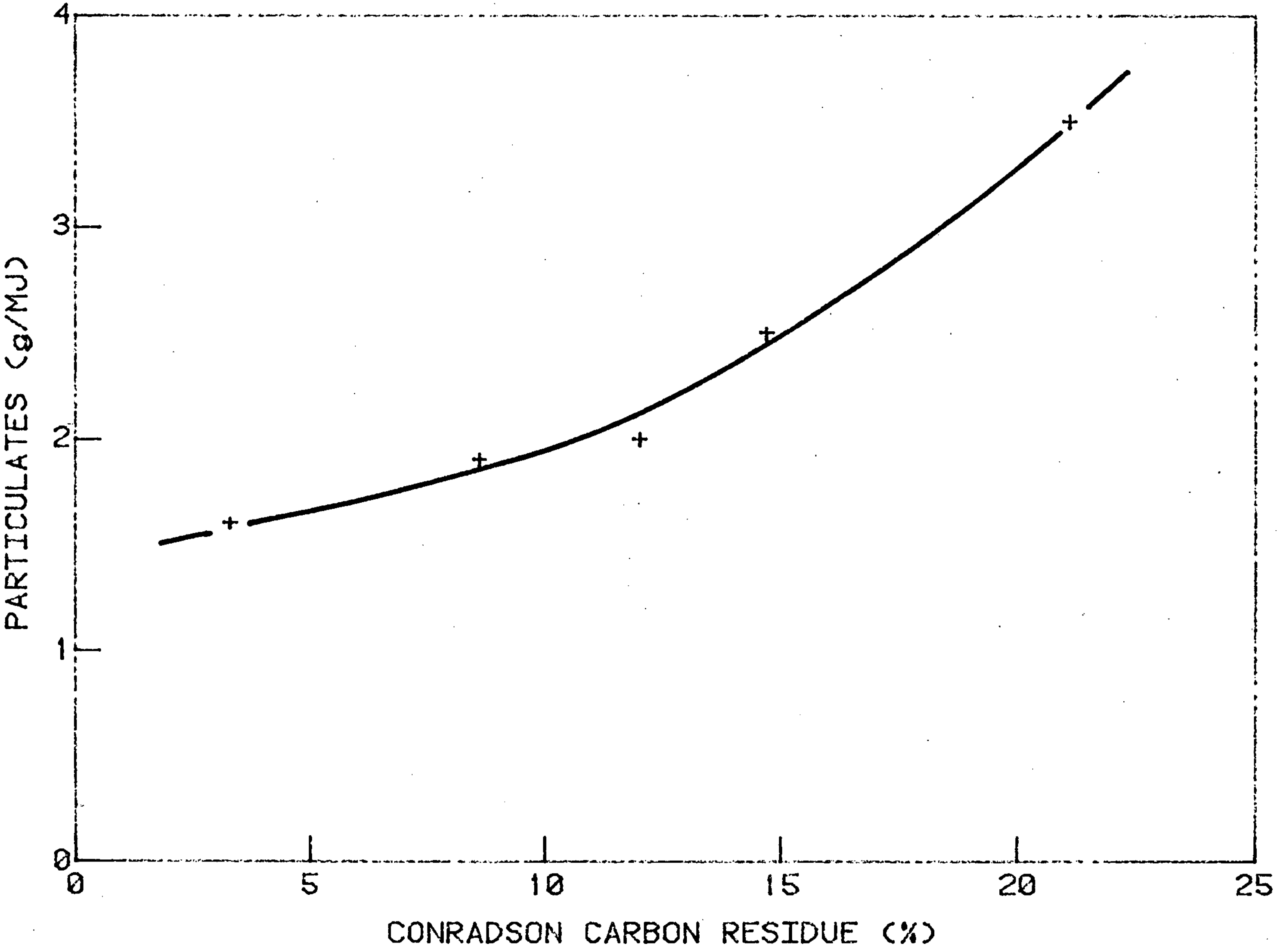

FIG. 21 PARTICULATES' EMISSION VS. CONRADSON CARBON RESIDUE.

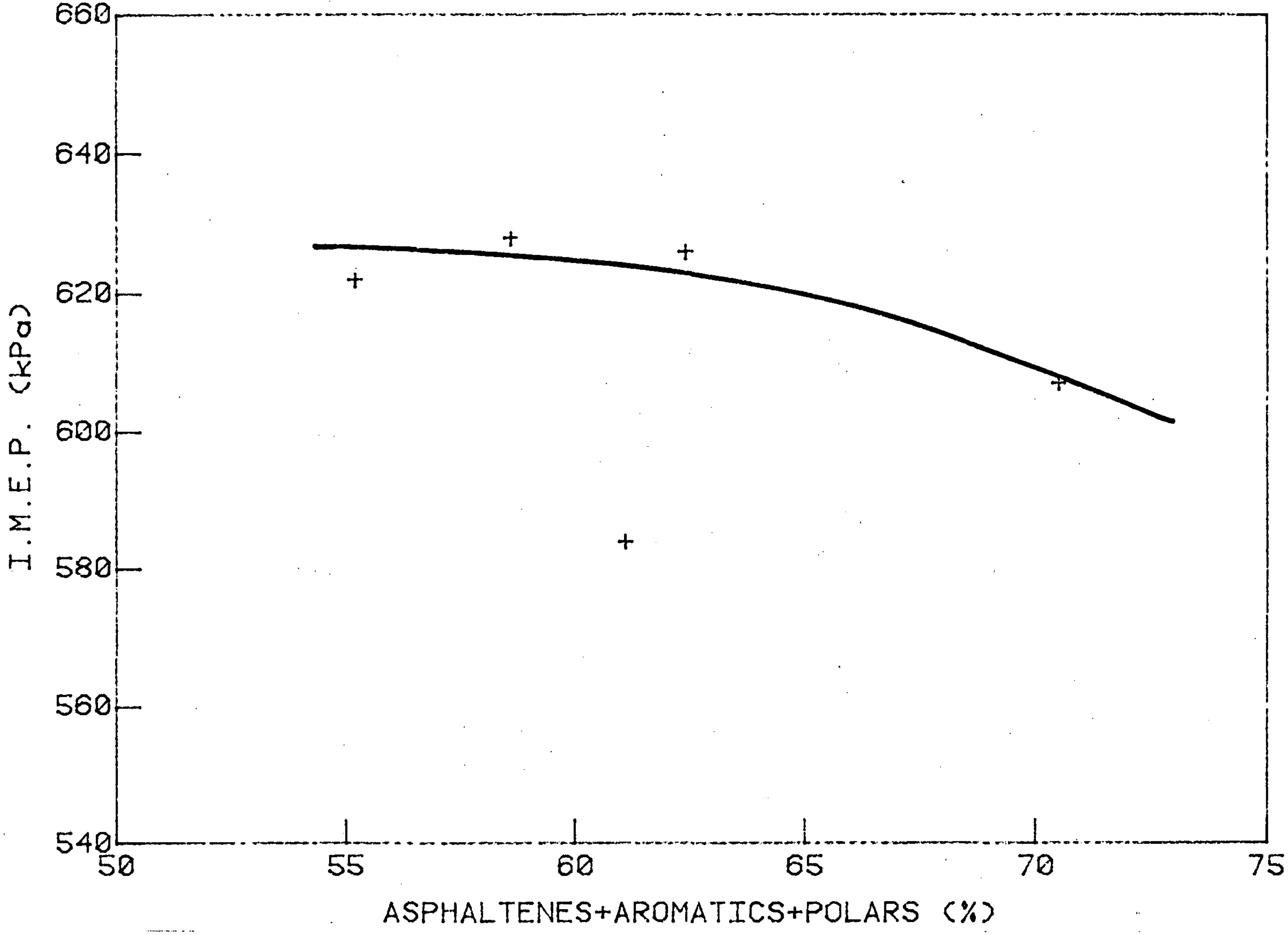

FIG. 22 INDICATED MEAN EFFECTIVE PRESSURE VS. INFERIOR CONSTITUENTS OF FUELS.

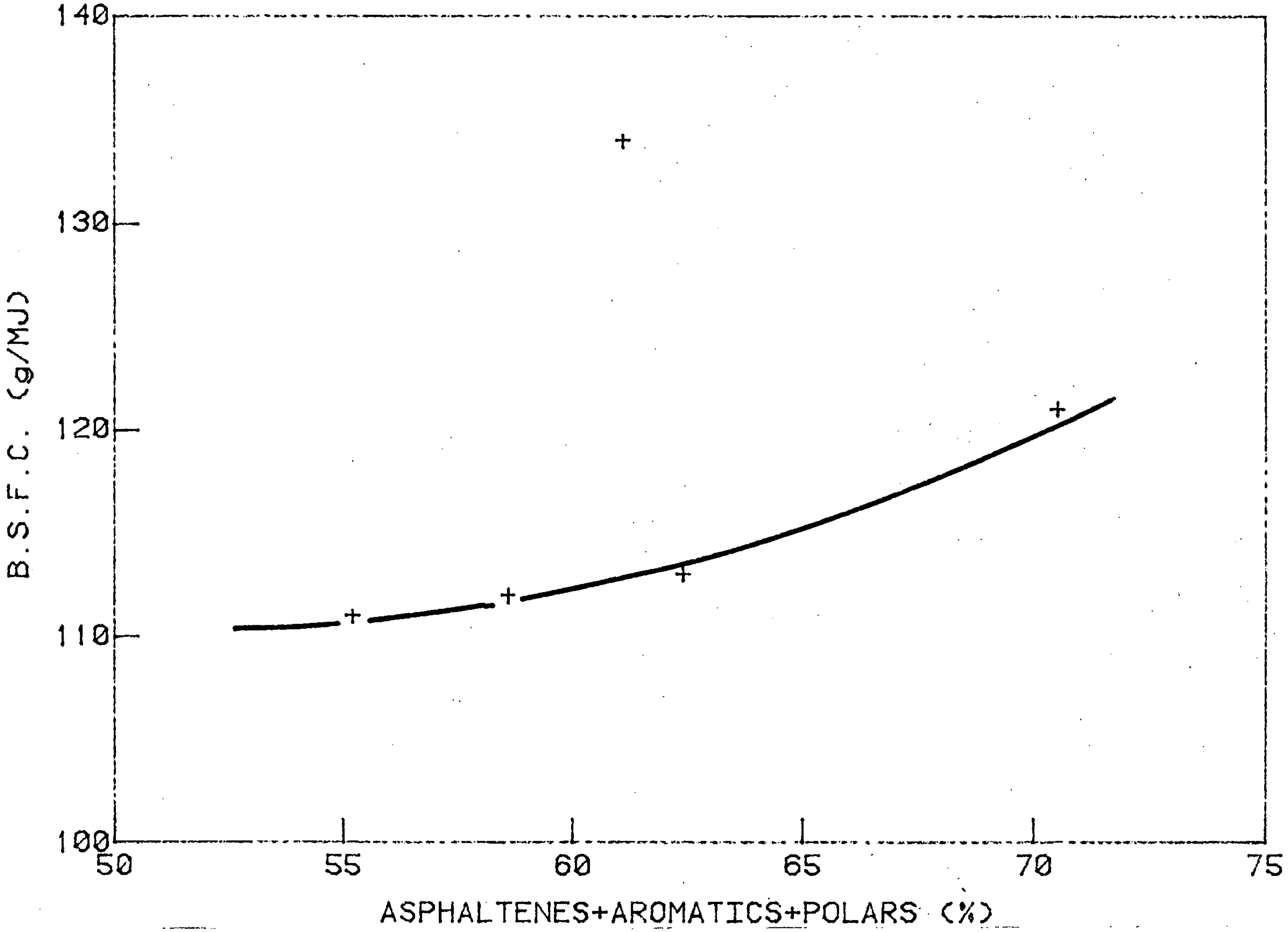

FIG. 23 BRAKE SPECIFIC FUEL CONSUMPTION VS. INFERIOR CONSTITUENTS OF FUELS.

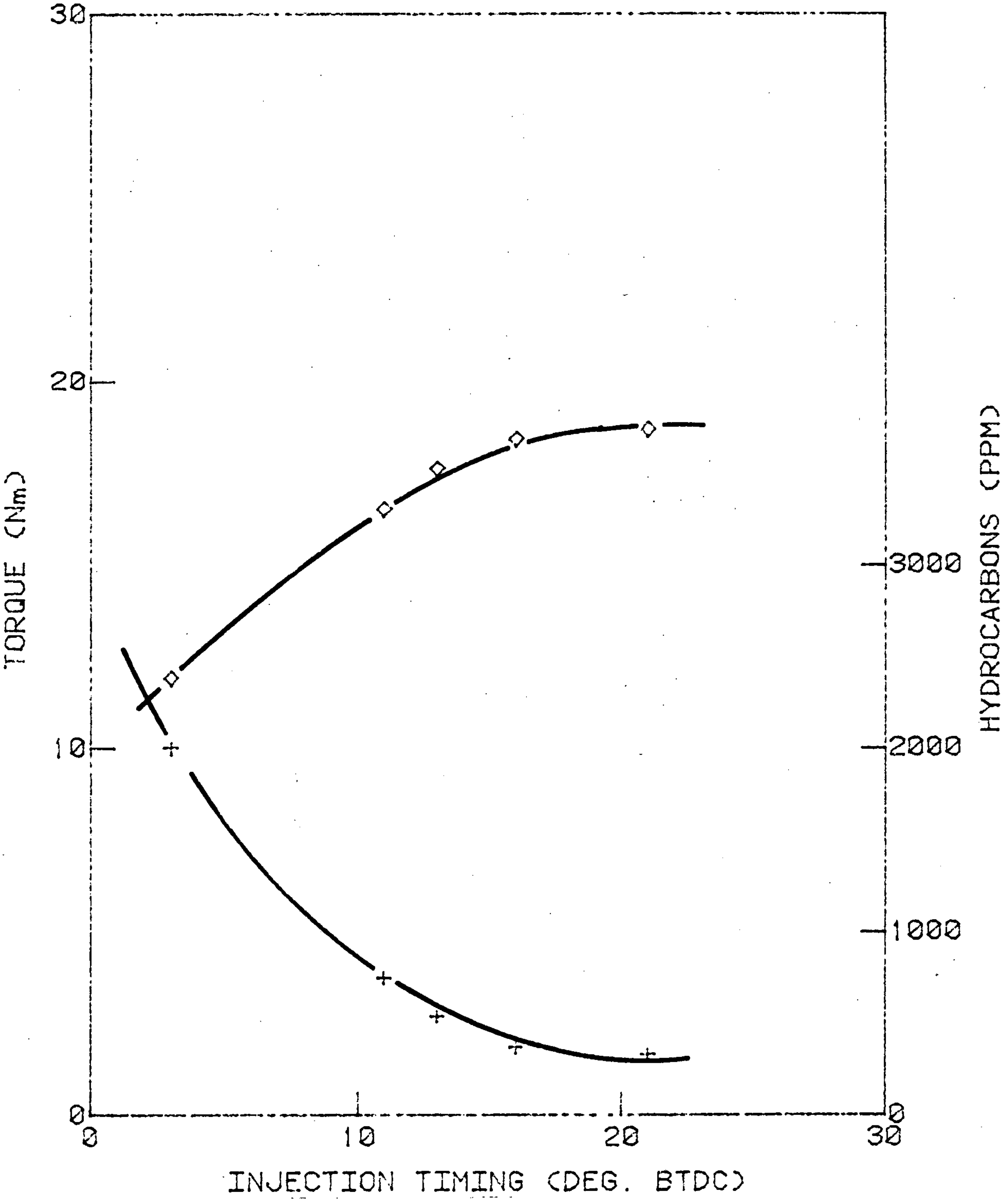

FIG. 24 VARIATION OF ENGINE PERFORMANCE WITH INJECTION TIMING.
FUEL: IBF 180 D-3096

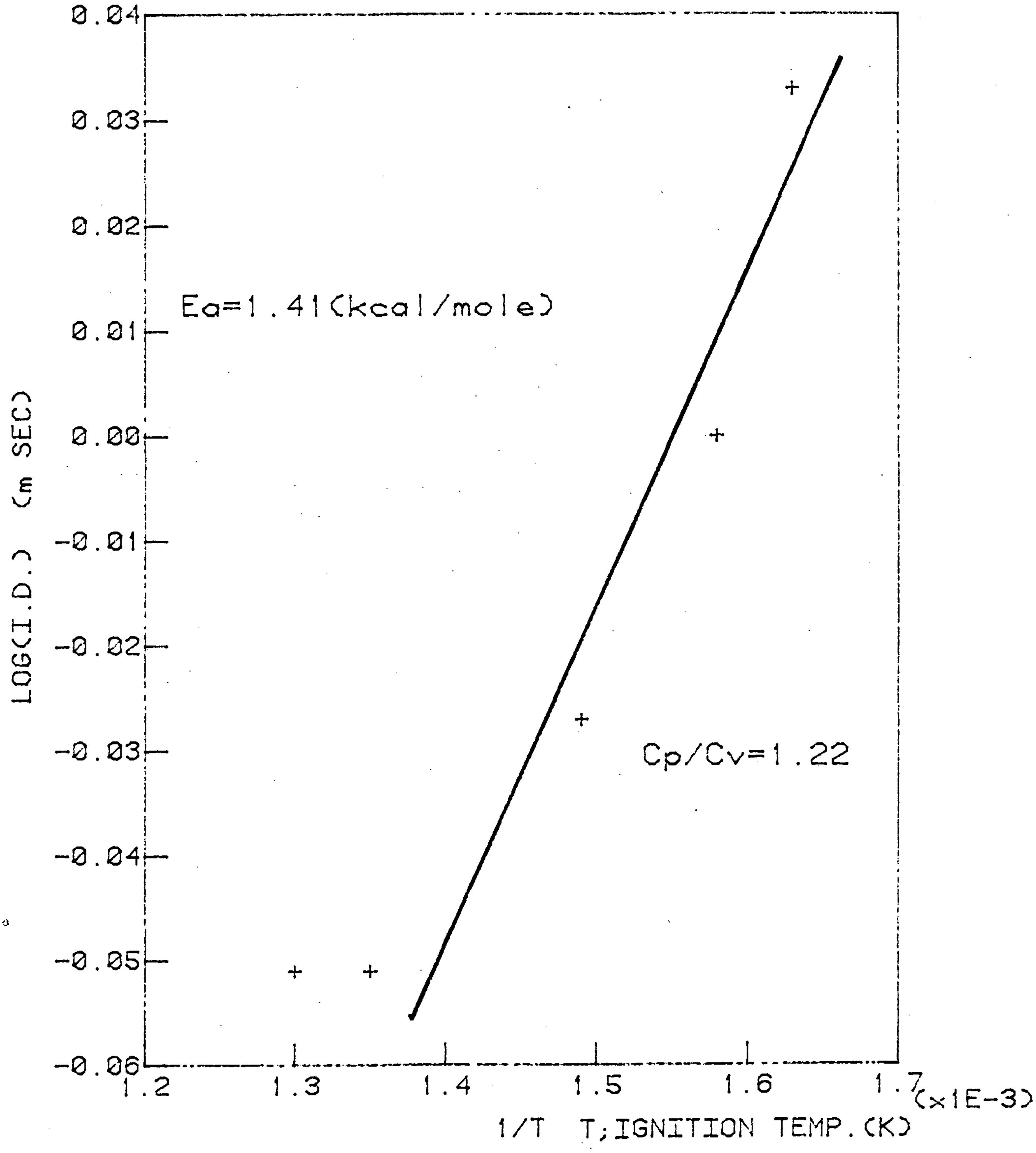

FIG. 25 ARRHENIUS PLOT OF IGNITION DELAY VS. IGNITION TEMPERATURE.

TECHNICAL INFORMATION DEPARTMENT
LAWRENCE BERKELEY LABORATORY
UNIVERSITY OF CALIFORNIA
BERKELEY, CALIFORNIA 94720