STUDY OF PERFORMANCE PARAMETERS AND EXHAUST GAS POLLUTION OF NATURAL GAS FUELED SPARK IGNITION ENGINE

BY

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ABSTRACT

Among many alternative fuels for internal combustion engine, natural gas can be considered as the single largest alternative fuel which can substitute totally or partially the liquid fossil fuel (gasoline and diesel).

This research work was intended to study the performance characteristics and exhaust gas emissions (CO, HC, CO_2, O_2) of a two cylinder diesel engine converted to spark ignition engine to run on 100 percent compressed natural gas (CNG).

In this experiment, Tartarini gas regulator of Itally, gas air mixer, electronic ignition system, spark plugs, etc. were fitted. Compressed natural gas (CNG) cylinder containing gas at 200-210 kg/cm^2 was used as a fuel source.

From the experimental result, it is seen that natural gas fueled spark ignition engine can deliver nearly equal output of its diesel counterpart (unconverted engine).

The natural gas fueled spark ignition engine was found to be more efficient at the ignition timing of 15° BTDC (before top dead centre) when the Brake Specific Energy Consumption (BSEC) of 13.22 Mj/kwhr and Brake Specific Fuel Consumption (BSFC) of 293.82 gm/kwhr and Brake Thermal Efficiency (BThEff) of 27.23% were found at full load operation.

The exhaust gas temperature of natural gas fueled spark ignition engine is found to be $120^\circ C$ to $180^\circ C$ above than that of a diesel engine and a petrol engine of comparative power range.

During the test CO, HC, CO_2 and O_2 were measured. NO_x is usually measured for diesel engine's exhaust gas pollutants where the formation of NO_x is high due to high combustion temperature and high compression ratio.

The harmful exhaust gas pollutant like carbon monoxide(CO) which restricts oxygen transfer in blood hemoglobin is found nil at the air fuel ratio above 17.48:1 for natural gas fueled spark ignition engine indicating that natural gas is a clean fuel.

It is also found that at air fuel ratio below 17.48:1 the exhaust gas pollutant (CO) is traceable in considerable amount depending upon load condition and richness of the air fuel mixture.

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NOTATION

1.	α	-	Power adjustment factor.
2.	β	-	Fuel consumption adjustment factor.
3.	BP	-	Brake Power.
4.	BSEC	-	Brake Specific Energy Consumption.
5.	BSFC	-	Brake Specific Fuel Consumption.
6.	BThEff	-	Brake Thermal Efficiency.
7.	%CO	-	Percentage of carbon monoxide v/v.
8.	$\%CO_2$	-	Percentage of carbon dioxide v/v in exhaust gas.
9.	Gm/gm	-	Gram.
10.	H	-	Humidity.
11.	j	-	Joules.
12.	k	-	Ratio of indicated horse power.
13.	K	-	Kelvine.
14.	Kw	-	Kilo Watt.
15.	Kg	-	Kilo gram.
16.	LNG	-	Liquified Natural Gas.
17.	LPG	-	Liquified Petroleum Gas.
18.	mg	-	Milligram.
19.	Mj	-	Mega joules.
20.	mm	-	Millimeter.
21.	N	-	Engine speed in revolution per minute (rpm).
22.	°BTDC	-	Degree Before Top Dead Centre.

23.	P_a	-	Atmospheric pressure.
24.	P_1	-	Exhaust pressure in mmH_2O
25.	P_2	-	Inlet air depression in mmH_2O.
26.	P_3	-	Inlet manifold vacuum in mmH_2O.
27.	P_4	-	Pressure across the air orifice meter in mmH_2O.
28.	RPM	-	Revolution per minute.
29.	R	-	Gas constant.
30.	SFC	-	Specific Fuel Consumption.
31.	t	-	Temperature, in °C.
32.	T	-	Temperature in °K.
33.	t_1	-	Temperature of intake air (air box) in degree centigrade.
34.	t_2	-	Temperature of lubricating oil(sump) "
35.	t_3	-	Temperature of air fuel mixture at intake "
36.	t_4	-	Temperature of exhaust gas of cylinder no.-1 "
37.	t_5	-	Temperature of exhaust gas of cylinder no.-2 "
38.	t_6	-	Temperature of cylinder head, "
39.	t_7	-	Temperature of outlet water of Hyd.dynamometer "
40.	ts	-	Standard temperature in degree centigrade (27°C).
41.	tx	-	Temperature of site condition in degree centgrade.
42.	τ	-	Observation time in seconds.
43.	w_{fs}	-	Standard fuel flow rate gm/sec.
44.	w_1	-	Initial weight of gas cylinder before experimental observation.
45.	w_2	-	Weight of gas cylinder (service) at the end of observation.
46.	ϕ_x	-	Relative Humidity.

CHAPTER -1

INTRODUCTION :

Fuel crisis and its subsequent effect was acutely felt in Western countries like U.K., West Germany, France, Italy, U.S.A. along with other developed countries in early seventies when the price of liquid fuel rose abnormally high. Since then developed countries took keen interest in finding out the alternative in the major fuel depending sectors. The necessity of alternative fuel in S.I (spark ignition) and C.I (compression ignition) engine also got importance in many countries. Subsequently lot of research and success have been achieved in countries like Italy, U.S.S.R., New Zealand, Canada, Denmark, U.S.A., Japan, France, etc. Tartarini of Italy has developed different type of gas regulators, mixer, cylinder and other gas controlling units which are commercially used world wide. Japan and France have developed gas engine of different ranges upto 750 Kw.

The alternative fuels so far sorted out are Compressed Natural Gas (CNG),Liquified Petroleum Gas (LPG),Methanol,Ethanol, Bio-mas, etc. Each alternative fuel has got advantages, disadvantages and own economics, but compressed natural gas has top position among them for using in I.C. engine because of its more advantages and less disadvantage like-good combustion with less engine noises and vibrations, less exhaust gas pollutants and satisfactory engine performances.

In Bangladesh, research on natural gas fueled S.I. engine is limited with Petro Bangla, BUET, and Bangladesh Diesel plant Ltd. CNG Plant under Petro Bangla converted a few number of gasoline fueled S.I. engine to natural gas fueled S.I. engine in collaboration with Tartarini of Italy and TNO of Netharland. TNO is working as a consultant and Tartarini as the supplier of CNG conversion kits. CNG plant under Petro Bangla is simply taking up the work of converting gasoline fueled S.I. vehicle to natural gas fueled S.I. vehicle with the conversion kit imported from Tartarini of Italy but no experimental data are available there.

Since Bangladesh has got sufficient gas reserve against 100% dependence on imported petroleum fuel, so the use of CNG in S.I. engine, in C.I. engine and converted S.I. engine can play a vital role towards the saving of foreign courrency and dependence on foreign coutnries for fuel.

Bangladesh Diesel Plant (BDP) Ltd manufactures 2 cylinder diesel engine locally. If this diesel engines can be converted to dedicated gas, it will generate economic and social values and develope experties locally.

The present research work has the following objectives -

(i) To study the effect of ignition timing on the performance of natural gas fueled spark ignition engine at different loads.

(ii) To study the overall performance characteristics of the engine at a constant speed of 1500 rpm with variable load test.

(iii) To study the exhaust gas emissions (CO, HC, CO_2 and O_2) of natural gas fueled spark ignition engine for correlating the exhaust gas pollution / emissions with engine performance characteristics.

CHAPTER-2

LITERATURE REVIEW

Nagosh and Samaga (20) optimised a biogas fueled S.I. Engine. In their test result it is reported that engine performance in general more satisfactory at advanced ignition (35°BTDC) and at larger spark plug gap (0.45mm). The tested engine showed a minimum requirement of biogas at the rate of 0.7 m^3 /bhp-hr. The biogas contained 59% methane and other combustible and 41% CO_2.

Karim and Ali (16) tested a single cylinder spark ignition research engine fueled with natural gas . Their result showed that in the exhaust, oxides of nitrogen (NOx) always decreased with the retarded spark ignition timing. The emission of CO increased with decreasing mixture temperature and also with increasing compression ratio.

Cambell (6) conducted experiments on the use of compressed natural gas (CNG) in spark ignition engine and in dual fuel engine . He converted a single cylinder diesel engine (Duetz 210 D, 6.7 Kw , air cooled, comp. ratio 17:1) to a natural gas fueled S.I. engine reducing compression ratio to 12:1 . From his test result of single cylinder S.I. engine, he reported that exhaust tempertature decreased ignificantly with ignition timimg advanced. An increase in ignition advance from 2° BTDC to 15° BTDC resulted in decrease in exhaust temperature from 702°C to 620°C at 2000 rpm at rated load . It was also mentioned that exhaust temperature in natural gas fueled engine was overall high compared to diesel fuel operation . Exhaust emission was found to follow typical spark engine trends.

Obert (23) explained the performance characteristics for a constant speed load test of I.C. engine . It is mentioned that with the increase of load , brake specific fuel consumption decreased due to increase of mechanical efficiency at high load.

Lyon. Howland and Loom (17) proposed the technique of controlling smoke and gaseous emissions from a diesel engine, using liquid petroleum gas as alternative fuel . They suggested that direct injection of diesel responded excellently to dual fueling and this method could be used to control smoke.

Baluswamy (2) worked on the LPG dual fuel engine using 70% diesel and 30% LPG and compared this result with 100% diesel operation. He concluded from his experiment that LPG dual fuel operation always led to lower S.F.C. values and also found that smoke was greatly reduced. From an examination of the various heat release record , they suggestd that dual fuel com·bustion generally appeared to undergo two distinct phases, the first phase was associated mainly with the combustion of pilot and associated gaseous fuel, while the second phase was dependent on the concentration and quality of the gaseous fuel employed. The heat release analysis of lean mixture operation showed agreement with other experimental evidence that comperatively inefficient combustion under such condition was due to mainly the inability of the gaseous charges to supplement effectively the heat release of the first phase . The use of larger pilots or higher initial temperature could reduce this tendency.

Murty (22) investigated kinematic NO (oxide of nitrogen) inside the cylinder of S.I. engine working on gasoline water mixture . He mentioned that formation of NO in the engine cylinder depended not only on the various parameters but also on the character of combustion process which in term depended on the design and other factors. In the investigation result it it is seen that Nitric Oxide (NO) increases from 250 ppm to 400 ppm for natural aspirated engine but Nitric Oxide (NO) increases from 500 ppm to 650 ppm to supercharged hydrogen engine when inlet temperature increased from 290 deg K to 320 deg K. It was further observed that as a result of lean air fuel mixture and high peak combustion temperature NO emission was high.

Newton, Steeds and Garretts (21) mentioned that unburnt hydrocarbon in the exhaust originated from the quench layer on the walls of combustion chamber, a few hundredths millimeter thick in which the flame was extinguished by cooling. Therefore surface /volume ratio must be kept small as possible and design should be free from crevices and corners where quench might occur.

Beg (4). studied the effect of mixing on the performance of a single cylinder dual-fuel diesel engine. The experiment was conducted with different shape of air-gas mixers, entry lengths, and different percentage of diesel gas consunption. From the experimental result it is reported that for a particular load, as the percentage of diesel consumption decreased, the equivalent Specific Fuel Consumption increased and the rate of increase was found to be higher at lower percentage of diesel consumption. This increase was more prominent at lower loads than higher loads. At higher loads, equivalent SFC was very close to 100 percent diesel operation though the gas flow rate increased. From the comparison of equivalent SFC for different air-gas mixers and entry length, it was reported that with 60 percent diesel consumption , equivalent SFC decreased by 3.8 percent for cross-shape, 3.7 percent for L-shape, and 3.5 percent for L-shape (direction of L opposite to air flow) mixer. This was due to better mixing of air and gas before entry of the engine.

Bari (3) also conducted an experiment on the study of the effect of dual fuel on some performance parameters of a single cylinder small diesel engine. It is reported that though the rated power of the original diesel engine was 5 Hp but the same engine gave maximum 3.75 Hp with dual fuel operation. Three types of mixers (Direct, L-shape and Cross-shape) were used in the experiment and among those the Cross-shape mixer was found to be more efficient. It was also reported that at 75% of full load and 60% diesel consumption, the SFC was about 7% less than the 100% diesel fuel operation. It was concluded that the combustion in dual fuel engine mainly depended on three factors were homogenity of the mixture, flame speed and load.

CHAPTER - 3

METHODOLOGY :

Diesel engine does not need much modification for running it with dual fuel mode. In dual fuel operation of a diesel engine, the CNG or natural gas is supplied at low pressure in the intake manifold of the diesel engine and diesel is injected in limited quantity as a pilot fuel for starting the combustion of the air fuel mixture in side the cylinder.

But the same diesel engine needs a lot of modification for operating it by hundred percent dedicated CNG fuel. The engine has to be fitted with ignition system, fuel induction system along with the reduction of the compression ratio. In this experiment the compression of the diesel engine were also reduced from 17:1 to 12:1 and the compression ignition system was replaced by the installation of electronic ignition system.

The details steps and procedure for modification of the diesel engine to hundred precent CNG fueled spark ignition engine is described in appendix A-18..

CHAPTER - 4

DESCRIPTION OF THE EXPERIMENTAL SET-UP :

A schematic diagram of the experimental set-up has been shown in fig.1.1.1. The experiment was set up with the following equipments and measuring devices etc.

a. A two cylinder Duetz Diesel Engine (12.68Kw) converted it to CNG fueled S.I. Engine. The details of the engine specifications are listed in the appendix table A-2.

b. Hydraulic Dynamometer for measuring engine load during experiment.

c. Air supply system including its air measuring and controlling devices as shown in schematic figure no. 1.2.1

d. Fuel (CNG) supply system including its gas weighing device (electronic weighing scale), pressure regulator, gas flow control valves, pressure pipes etc. as shown in shcematic figure no. 1.2.1..

e. 270 Crypton (NDIR) Emission (CO, HC, CO_2, O_2) Analyser with an additional hose pipe, water separator etc.

f. Manometers group for measuring different pressures (Orifice pressure, exhaust gas depression, intake manifold pressure, air inlet depression).

g. Digital thermocouple groups for measuring different temperatures (air box temperature, lube oil temperature, air fuel mixture temperature at the intake manifold, exhaust gas temperature, cylinder head temperature, dynamometer exit water temperature).

h. Mixer (gas carburator) for proper mixing of fuel and air.

i. Digital tachometer for measuring engine rpm.

j. A sling phychrometer for measuring dry bulb temperature and wet bulb temperature.

k. A Barometer for measuring atmospheric pressure.

The diesel engine was modified to S.I. Engine by reducing its compression ratio from 17:1 to 12:1 and fitting an electronic ignition system on it. Photographs of the engine fitted on the test bench are shown in fig. 2.1.1. and 2.1.2. The important technical details of the original diesel engine and converted CNG fueled S.I. Engine are listed in appendix table A-1 and A-2.

The schematic diagram of air flow system is shown in fig. 1.2.1. The original air filter was removed from the engine and an air flow measuring device was fitted with the intake manifold of engine. The air flow and its measuring device consisted of an air drum (actually two nos. empty drums fabricated and welded to single drum), an orifice meter with corner tapping, air hose pipe (dia 12.7cm), air shut-off valve inclined manometer, as shown in fig. 1.1.2. The air shut-off valve was installed before air hose (i.e. at the exit of air from drum) for controlling intake manifold air pressure and its flow as and when necessary.

The gas (CNG) supply system consisted of two nos. CNG cylinders connected in parallel by pressure pipe and gas control valves, so that gas (CNG) could be supplied to engine from any one of these two cylinders as and when necessary. One CNG cylinder was for supply and storage of CNG gas, called storage cylinder. The other CNG cylinder called service cylinder was fitted on a weighing electronic scale under the test bench which supplied CNG during the period of the experimental work. This weighing or service cylinder was refilled from the CNG of storage cylinder through control valves before starting the experiment.

Electronic weighing scale was used for taking weight of service cylinder before and after the experimental observation from where fuel flow rate was calculated. Digital stop watch has been used for measuring observation period during the engine test.

High pressure gas (CNG) passed from CNG cylinder to gas regulator (pressure reducer) where high pressure CNG (200-210 bars) was reduced to its line pressure of approximately 1 kg/cm^2 from where the gas passed into the mixer and it is then mixed with air to feed a homogenious mixture of gas and air to the engine cylinder.

The engine was installed on an engine test bench and its out put shaft was coupled to a hydraulic dynamometer with necessary fittings and couplings. The fittings were fabricated in the machine shop and welding shop of BUET.

The exhaust gas from the exhaust silencer was passed outside the engine room through a fabricated G.I. pipe (10.16 cm). Provission was also made for collection of exhaust gas sample from exhaust line to gas analyser. A separate pot or water separator was also fabricated and installed on the line of exhaust gas sample, so that it could separate water from the exhaust sample before its entry into the analyser.

A group of manometers were installed to measure different pressures like exhaust gas depression, intake air pressure, manifold vacuum, air orifice pressure etc.

A sling physhrometer was installed for measuring dry bulb and wet bulb temperature near the engine.

CHAPTER - 5

PROCEDURE

The experiment was conducted in the following steps:

5.1 Checking of CNG Reserve and Pressure in CNG Cylinders :

Before starting the experiment, it should be confirmed that CNG stock and its pressure is sufficient for taking at least one complete set of observation. As such before starting the engine, CNG pressure in the storage cylinder and in the service cylinder was checked individually by opening the gas control valve and observing the corresponding gas pressure from the pressure gauge fitted at the inlet line of the gas regulator.

In our case, 100 kg/cm^2 pressure of natural gas in storage cylinder and in service cylinder was sufficient for taking a complete set of observation.

5.2 Warming up and making the Crypton 270 Emission Analyser Ready for use:

It takes nearly 10 minutes to make the emission analyser ready for use. As such before starting the engine, the analyser should be brought up into operation mode as per instructions of operating manual of 270 Crypton Emission Analyser. The filter was particularly inspected for water deposit as water in the filter gives wrong reading. In rainy season, water deposit in the water seperator was found in so much quantity that at every time of engine starting, sufficient quantity of water needed to be removed.

After those preliminary checkings the power was supplied from the switch board to the gas analyser through a voltage stabliser (1 Kw). Then the emission analyser was allowed for its self calibration for bringing it into operation mode.

5.3 Checking Engine and Dynamometer :

Before starting the engine, oil level and its leakage were checked up thoroughly and oil was added or refilled as and when necessary. Moreover battery connections were also checked up and tightened if necessary.

Oil level in dynamometer bearings were also checked up and oil was added if necessary.

Obstructions to engine rotation and dynamometer rotation were also checked up and removed if any.

More over different measuring devices were thoroughly inspected before starting the engine.

5.4 Engine Operation and Experimental Observation :

After ensuring the natural gas supply from the storage CNG cylinder, the engine was started with the help of starting key fitted on the switch board fixed on the engine mounting frame/chasis.

Engine was allowed to warm up for 5-10 minutes and then load and speed were increased slowly. Engine was fixed at 1500 rpm and dynamometer's load was increased to the value for maximum load carrying capacity of the engine. This was done by simultaneously increasing the water supply to the dynamometer through the water regulating valve from the storage water tank to dynamometer inlet and increasing the entry of air fuel mixture through the throttle valve under the gas carburator, keeping the engine rpm at 1500 rpm.

Experimental data in this position was recorded . The dynamometer's load on the engine was reduced by 1kg step from the maximum load carrying capaity to its half value and in each step data were recorded without changing the ignition timing.

During experiment the engine was set at the ignition timing of 15°BTDC and at the constant speed of 1500 rpm. The engine load was varied at the step of 1 Kg load (approximately) and at each step experimental data for Engine Brake Power, Air Flow Rate, Fuel Flow Rate, Temperatures and Pressures for study of engine performance characteristics and exhaust gas emissions (CO, HC, CO_2, O_2) were also recorded.

Later on, the ignition timing was varied (reduced) at the interval of 1.50°. At each ignition timing interval, engine load was varied as mentioned above at a constant speed of 1500 rpm.

The emission analyser gave the direct reading of % CO v/v, % CO_2 v/v, % O_2 v/v and HC level in ppm.

Power, rated power , specific fuel consumption, specific energy consumption, thermal efficiency, equivalence ratio, etc.were calculated from the experimental data. Sample calculation is shown in Appendix A-17.

CHAPTER - 6

RESULT AND DISCUSSION

6.1 GENERAL :

The diesel engine (2 cylinders) converted to hundred percent compresssed natural gas fueled spark ignition engine was tested at a constant speed of 1500 rpm with variable loads, for the study of the performance characteristics and exhaust gas emissions at different ignition timings from 6°BTDC (before top dead centre) to 15°BTDC (before top dead centre) at 1.5° interval.

The fig. no. 3.1 to fig. no. 3.4 are drawn from the test data for the ignition timings of 15°BTDC, 12°BTDC. 9°BTDC and 6°BTDC for study of the performance characteristics at different ignition timings. Each parameter (BSEC, BThEff, Exht Temp.) of the performance characteristics at different ignition timings (15°BTDC, 12°BTDC,9°BTDC and 6°BTDC) has been drawn and compared to fig. no. 3.5, 3.6 and fig. no. 3.7. Moreover fig. no. 3.8 and fig. no. 3.9 show variations of maximum brake power and exhaust gas temperature at different ignition timings. Also fig. no. 3.10 to fig. no. 3.15 are presented from the test data for the study of exhaust gas emissions.

6.2 INVESTIGATION OF PERFORMANCE CHARACTERISTICS :

From the test result and figures, the observations regarding brake specific energy consumption, brake thermal efficiency, minimum brake specific energy consumption, exhaust gas temperature, maximum brake power, the effect of air fuel ratio on the engine brake thermal efficiency, percentage of carbon monoxide (CO) emission, carbon dioxide (CO_2) emission, oxygen (O_2) emission and level of hydrocarbon (HC) emission are presented here.

6.2.1 Brake Specific Energy Consumption :

It is seen from the fig. no. 3.1 to 3.4 that the brake specific energy consumption decreases with the increase of load upto 65% to 75% of the full load operation and again the same (BSEC) increases with the increase of loads till its full load operations in all the tests under different ignition timings from 6°BTDC to 15°BTDC.

6.2.2 Brake Thermal Efficiency :

It is also found that brake thermal efficiency increases with the increase of loads till its peak value and then falls again in all the different ignitions timings as shown in fig. no. 3.1 to 3.4. Again the brake thermal efficiency of different ignition timings has been compared in fig. no. 3.6 from where it is seen that the engine is more efficient at the ignition timing of 15°BTDC.

6.2.3 Minimum Brake Specific Energy Consumption (Mj/kw-hr):

Brake specific energy consumptions at different ignition timings from 6°BTDC to 15°BTDC have been compared in fig. no. 3.5 from where it is observed that the minimum specific energy consumption decreases with the advance of the ignition timing.

The Minimum Brake Specific Energy Consumption decreases from 14.97 Mj/kw-hr to 10.92 Mj/kw-hr with the corresponding increase of Maximum Brake Thermal Efficiency from 24.04% to 32.94% when the ignition timing is advanced from 6°BTDC to 15°BTDC. This result is compared with the previous test result (6) in the table below.

	Item	Test result for two cylinder engine (Natural gas fueled , spark ignition engine)	Test report (6)
i.	Minimum Brake Specific Energy Consumption.	10.92 Mj/kw-hr	12.2 Mj/kw-hr
ii.	Maximum Brake Thermal Efficiency.	32.94%	28.1%

6.2.4 Exhaust Temperature :

It is also observed from fig. no. 3.7 that exhaust temperature decreases with the advance of ignition timing. It is further observed from fig. no. 3.8 that maximum exhaust temperature decreases from 698°c to 554°c when the ignition timing was advanced from 6°BTDC to 15°BTDC.

But while comparing this exhaust temperature with that of a typical diesel engine or a gasoline engine, it is seen that exhaust gas temperature of CNG fueled engine is higher than that of diesel and gasoline fueled engine.

In general the exhaust temperature of diesel engine and gasoline engine is within the temperature range of 370°c to 410°c at full load operation but the exhaust temperature can be reduced to 584 °c by adjusting (advancing) the ignition timing for CNG fueled spark ignition engine at full load operation.

Item	Tested engine (2 cylinder CNG fueled spark ignition engine)	General range of exhaust at full load operation.	Reported (6)
i. Maximum exhaust temp. at full load operation	563°c-584°c at 1500 rpm.	370°c-410°c	550°c at 1500 rpm. & 700°c at 3000 rpm.

6.2.5 Maximum Brake Power :

It is observed from fig. 3.9 that the maximum brake power increased with the advance of ignition timing. The brake power increased from 10.09 Kw to 12.68 Kw when ignition timing was advanced from 6°BTDC to 15°BTDC. The maximum power is obtained at the ignition timing of 15°BTDC.

6.2.6 The Effect of Air Fuel Ratio on the Engine Brake Thermal Efficiency :

The effect of air fuel ratio on the engine brake thermal efficiency showed typical relation. Substantial increase of engine brake thermal efficiency could be obtained by proper air fuel mixure, as found from fig. no. 3.17. The complete burning at proper air fuel mixutre causes the increase of brake tharmal efficiency.

6.3 INVESTIGATION OF THE EXHAUST GAS EMISSIONS :

Though the exhaust gas pollutants are CO, NO_x, SO_2, HC particles but our investigation was limited to analyse the percentage of CO, CO_2, O_2 and HC level for natural gas fueled S.I. Engine. There were no facility for measuring NO_x in our test and moreover NO_x meter is generally used for measuring emission of diesel engine where the formation of the oxides of nitrogen (NO_x) is likely to be high due to high combustion temperature inside the cylinder for its high compressor ratio.

6.3.1 CO Emission :

At air fuel ratio above 18.84:1, CO emission is found nil in all the different ignition timing from 6°BTDC to 15°BTDC as apparent from fig. no. 3.14.

The carbon monoxide (CO) emission is found to increase with rich fuel operation and it is true for all the different ignition timing. The percentage of CO is found to decrease with the lean air fuel mixture as seen from the above figures no. 3.14. Percentage of Carbon monoxide emission found nil even at 85% of full load operation and it raises to 0.6% v/v when the output (kw) is 93% of its full load and percentage of CO further rises to 1.9% at its full load operation. While comparing the CO emission of CNG fueled spark ignition engine with that of admissible limits of CO pollutants in USA legislative or accepted limit of exhaust gas pollutants in E.E.C. and Japan, it is seen that the harmful CO level is in safe level and much lower than that of gasoline or diesel fuel operation as shown in the table below :

Emission	CNG fueled engine(at constant speed and variable load test)	Accepted limit of US legislative 1978. (22)	Accepted limit of Toyota vehicle, Japan. (27)	Accepted limit in England & in European country (8)
a. %CO	1.9% by v/v	3.4 *	1 to 2.5% v/v	-
b. HC	160 ppm	-	-	400 ppm
c. % NO_x	-	0.39*	-	-

(Noted : * - g/vehicle mile ; 275 ppm = 2.2 g/vehicle mile)

So it found that the harmful CO emission is well below the accepted limits of USA,Japan,and EEC.Moreover CO emission can be reduced to zero by leaning air fuel ratio above 17.48:1 , at 15° BTDC when the load carryng capacity is about 85% of full load. The test procedure of exhaust gas pollutants in USA is different from laboratory test, since in USA, exhaust gas pollutants are measured by collecting them in bag while the vehicle is operated inside and outside of town at different speed and modes on a chassis mounted dynamometer. But Toyota Company recommended to measure CO when engine run at hot condition while the vehicle is kept stationary for its test.

6.3.2 HC Emission :

HC is the unburnt hydrocarbon originating from the quench layer on the walls of combustion, a few hundredths millimeter in which the flame is extinguished by cooling. The other cause of the presence of hydrocarbon in the exhaust is the misfiring when the unburnt hydrocarbon of the full cylinder is escaped into the exhaust gas. The level of hydrocarbon is found high when the fuel in the mixture is high as apparent from fig. no. 3.10 to 3.1 . At too lean mixture the level of hydrocarbon is found to rise again. From the experimental data, the fig. no. 3.10 shows that the level of HC is traced even at lean mixture. This is mainly due to insufficient fuel in the mixture to start firing.

Through there are several proposals of the HC level limit in U.S.A. from 1966 to 1975 but no level limit of HC is available for world wide or for their national level application. But the manufacturer of Crypton 270 Emission Analyser of U.K. has recommended the level of HC upto 400 ppm as normal. But in our case it is 160 ppm at full load.

So it is found that HC level in CNG fueled S.I. Engine is also in safe side and comparatively low and it only 40% of the accepted limit.

6.3.3 CO_2 Emission :

CO_2 in the exhaust gas indicate the combustion efficiency and it is the equilibrium product of combustion. CO_2-curve in the fig. no. 3.9 to 3.13 is found to follow typical CO_2- curve trends of gasoling engine. The level of CO_2 is found to raise at its peak value at the air fuel ratio arround 17.48 : 1 and comes down again. At optimum air fuel ratio (17.48:1),the percentage of CO_2 becomes high because of complete combustion of the air fuel mixture. At rich mixture, percentage of CO_2 is lower because of incomplete combustion of air fuel mixture. Again at too lean air fuel mixture, percentage of CO_2 is also low because of the fact that excess air containing only 0.4% CO_2 bring the overall percentage of CO_2 in the exhaust gas to a low level.

While comparing the percentage of CO_2 in the experimental work with that of the value of the ultimate analysis of natural gas in the table A-4 , it is found that the percentage of the CO_2 of the experimental result is slightly less than that of the theoritical value as shown in the table below :

Exhaust emission	Test result with CNG fuel.	Theoritical result from ultimate analysis (Table A.4)
%CO_2 for ignition timing of 15°BTDC and at 1500 rpm.	9%	9.52%

The above 0.52% discrepency in the percentage of CO_2 is most probably for the leakage of air through the exhaust gas manifold joints.

6.3.4 O_2- Emission :

Level of O_2 emission in the exhaust gas has been shown in fig. no. 3.10 to 3.13 for different ignition timing. Moreover percentage of oxygen(O_2) with respect to air fuel ratio in the exhaust gas emission is shown in fig.no. 3.16 for the ignition timing of 15 degree before top dead centre and it is found that with rich mixture, percentage of O_2 is low because of the fact that most of the oxygen is burnt in the combustion process. Again percentage of oxygen is found to decrease with leaner fuel-air mixture because most of the oxygen remains unburnt in leaner mixture.

CHAPTER - 7

CONCLUSIONS

7.1 General Conclusion :

From the experimental result it is seen that the overall performance characteristics are competitive and quite satisfactory in natural gas fueled spark ignition engine. Moreover the exhaust gas pollutants (CO/HC) are also very low and below the admissible limits of the exhaust gas pollutants for natural gas fueled spark ignition engine.

The formation of the oxides of nitrogen(NO_x) in natural gas fueled vehicle can also be controlled by installing exhaust gas recirculation system (EGR) with natural gas fueled vehicles/engines if necessary. Duetz 2FL 912 diesel engine can be adopted to run on dedicated CNG with appropriate modifications.

7.2 Future Recommendation :

Though our research was limited with variable load test at constant speed, in future research, the natural gas fueled spark ignition engine should be optimized at a variable speed load test along with the study of the side effects like the durability of engine oil viscosity and wear and tear of engine parts like piston rings, cylinder liners, bearings etc. for modified 2FL912 engine.

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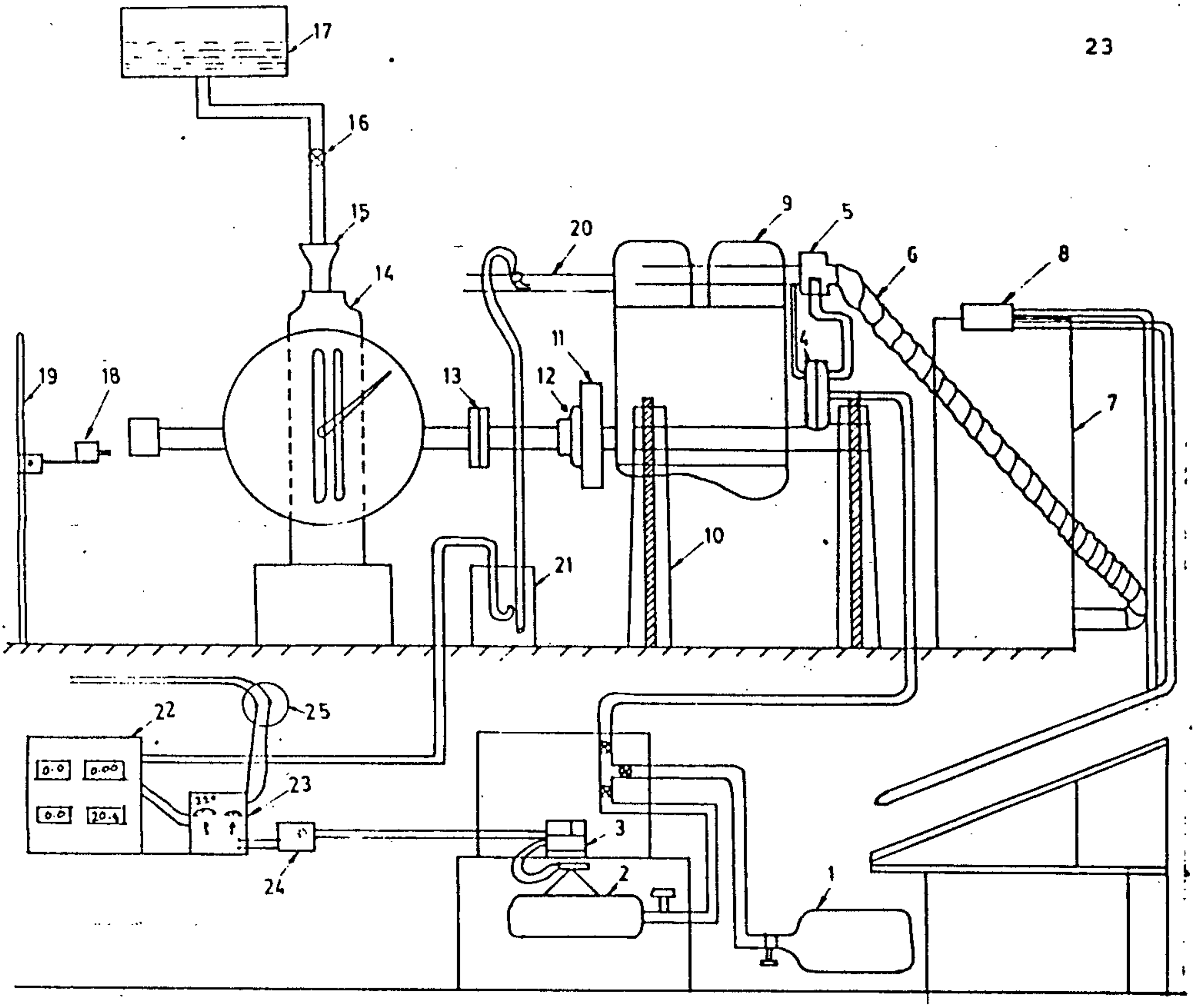


Fig. 1.1.1 Schematic Diagram of The Experimental Set-up.

1. CNG cylinder (storage)
2. CNG cylinder (service)
3. Electronic weighining scale
4. Gas regulator
5. Mixer
6. Air hose
7. Air drum
8. Air orifice meter
9. Engine unit
10. Engine test bench
11. Engine fly wheel
12. Flexible coupling
13. Shaft flange
14. Hydraulic dynamometer
15. Funnel
16. Water regulating valve
17. Reserve water tank
18. Digital tachometer
19. Digital tachometer stand
20. Exhaust pipe
21. Water seperator
22. Crypton 270 emission analyser
23. Voltage stablizer
24. Step down tranformer
25. Power supply board

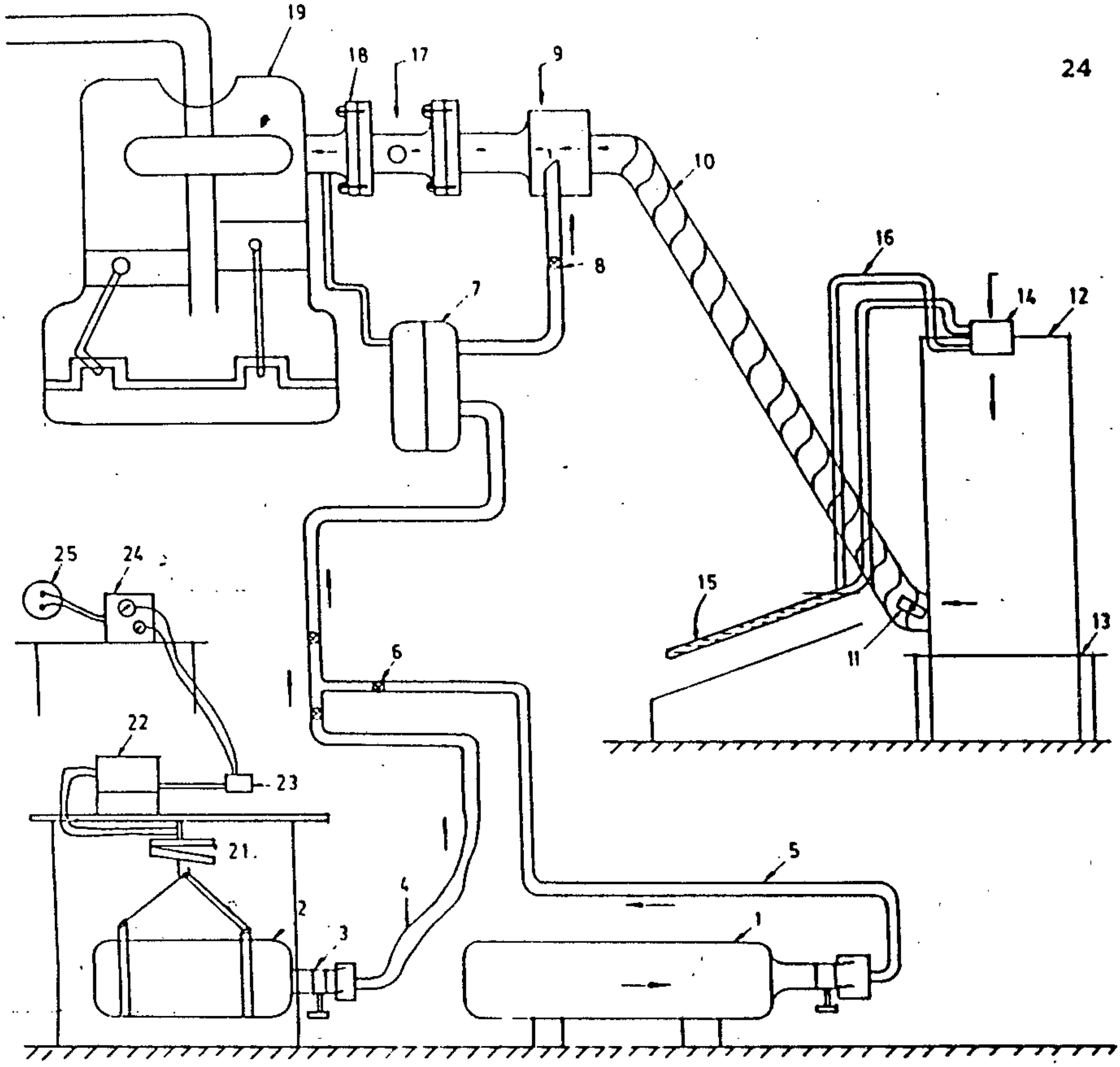


Fig. 1.1.2 Schematic Diagram of Air and Fuel Flow System .

1. CNG cylinder (storage)
2. CNG cylinder (service)
3. CNG cylinder shut-off valve
4. High pressure hose
5. High pressure pipe
6. Gas shut-off valve
7. Pressure regulator
8. Power/load valve
9. Mixer (gas carburator)
10 Air hose
11. Shutter
12. Air drum
13. Air drum stand
14. Air orifice
15. Inclined manometer
16. Flexible plastic pipe
17. Throttle valve
18. Flange with nut- bolt
19. Two cylinder engine unit
20. Piston
21. Sensor with hanger of weightronic scale
22. Weightronic scale
23. Adopter
24. Voltage stabliser
25. Power plug of pannel board

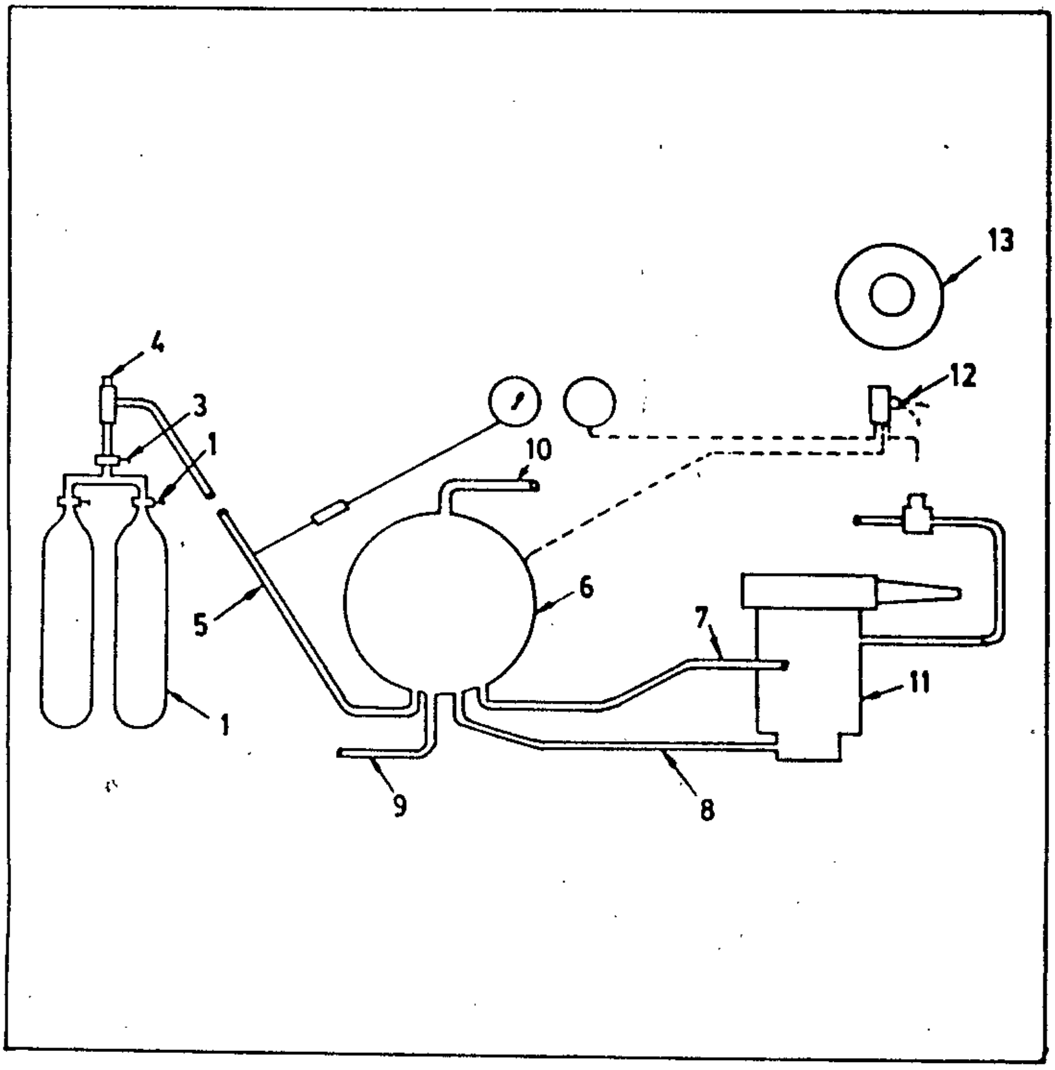


Fig. 1.2 Schematic Diagram of Fuel Flow System.

1. Fuel cylinders
2. Cylider shut-off valve
3. Master shut-off
4. Refilling point
5. High pressure line
6. Regulator
7. Gas line
8. Vacumm line
9. Water from radiator
10. Water return line
11. Carburator
12. Petrol shut-off valve
13. Changeover switch

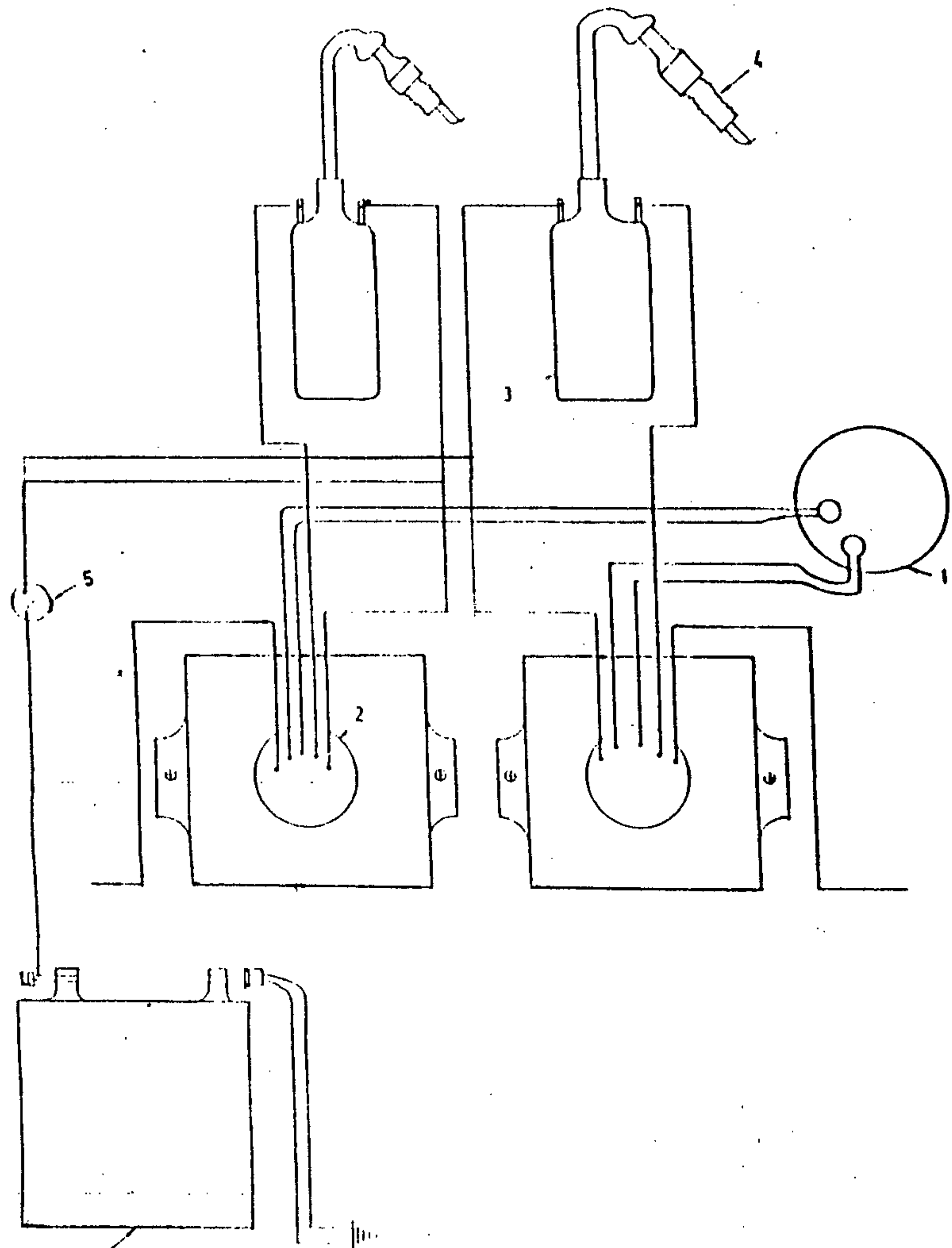


Fig. 1.3 Schematic Diagram of Electronic Ignition System.

1. Proximity switch
2. Ignition control unit
3. Ignition coil
4. Spark plug
5. Switch
6. Battery (12 V)

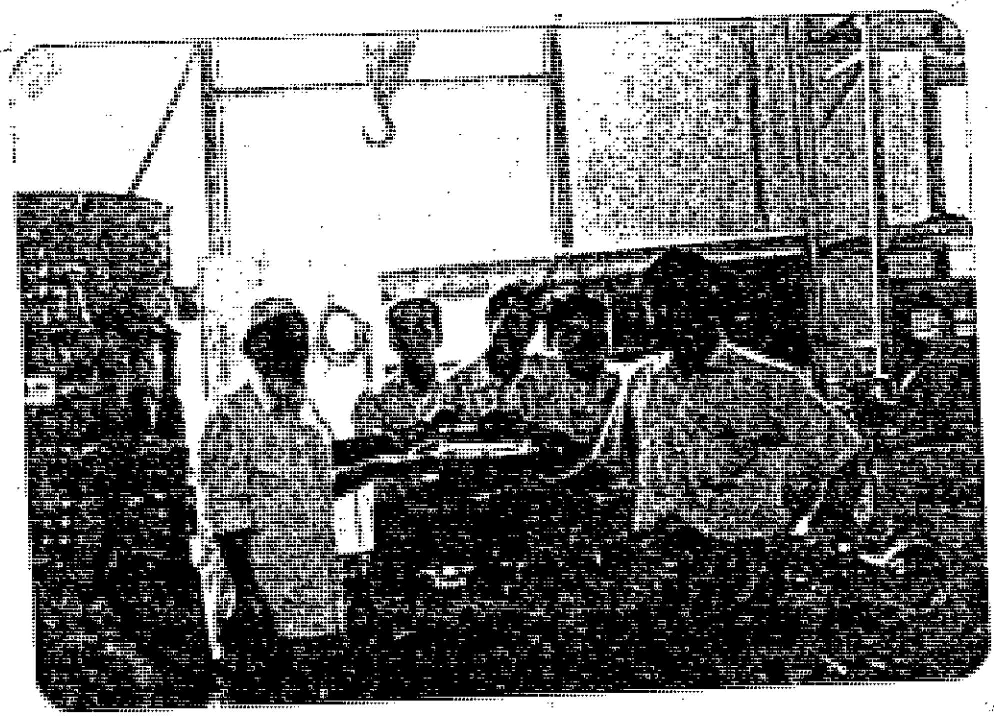


Fig. 2.1.1 Photograph of the converted CNG Fueled Spark Ignition Engine at the time of its installation on the engine test bench.

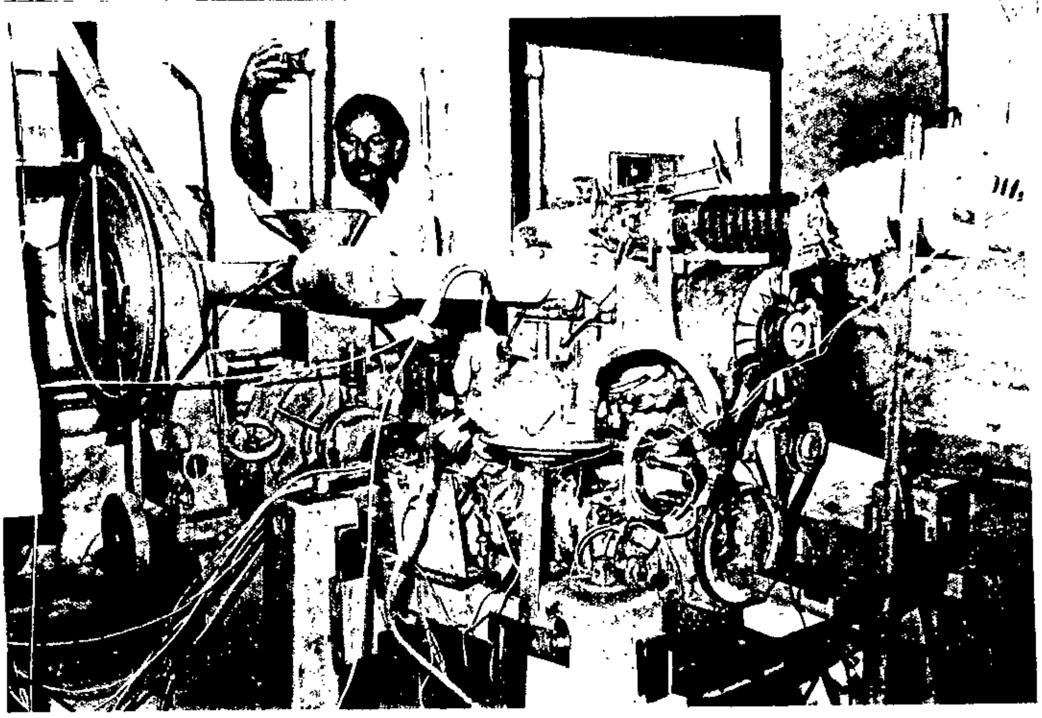


Fig. 2.1.2 Potograph of the Engine, Dynamometer, Engine Test Bench along with other accessories where the engine load is varied by controlling the water regulating valve for supply of water from storage water tank to dynamometer.

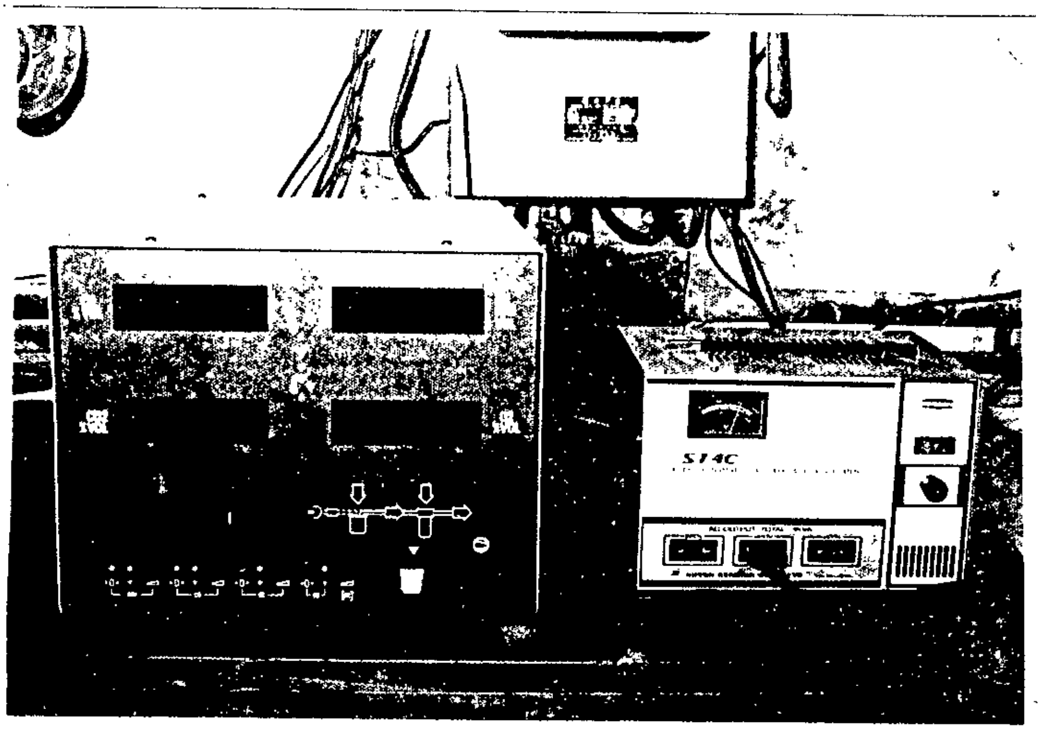


Fig. 2-2 Photograph of the Crypton 270 Emission Analyser, 1Kw Stac Voltage Stablizer and Power Distribution Box.

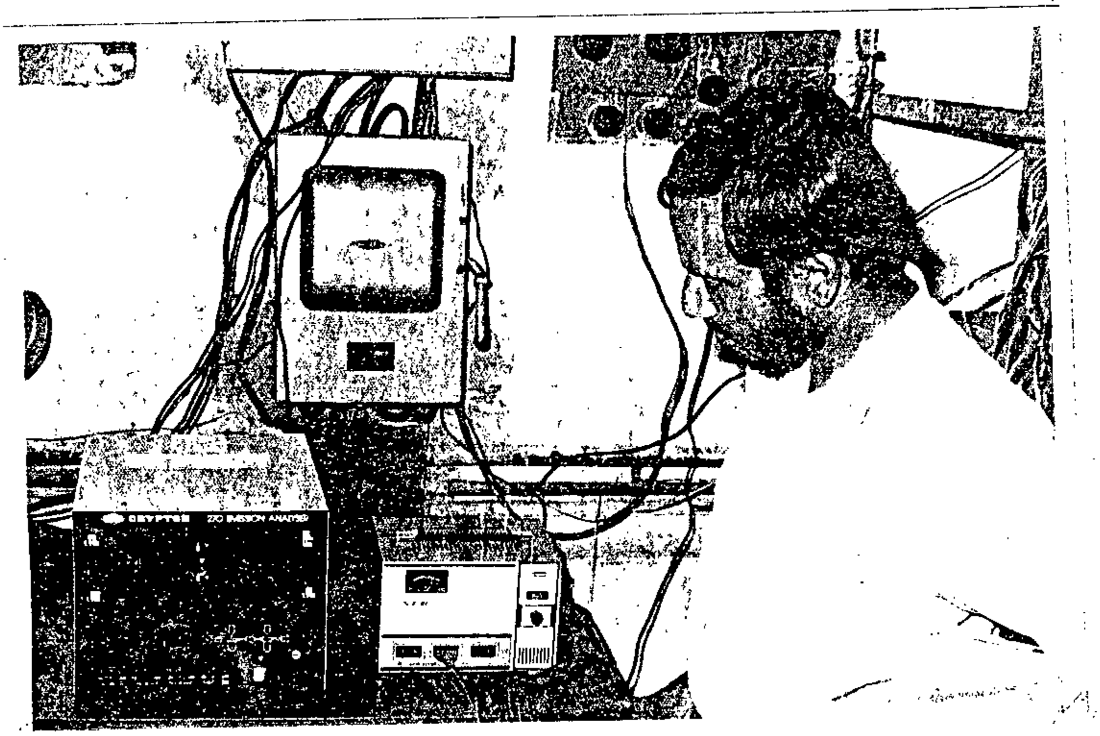


Fig. 2.3 Photograph of Crypton 270 Emission Analyser and Stac Voltage Stablizer during warm up period.

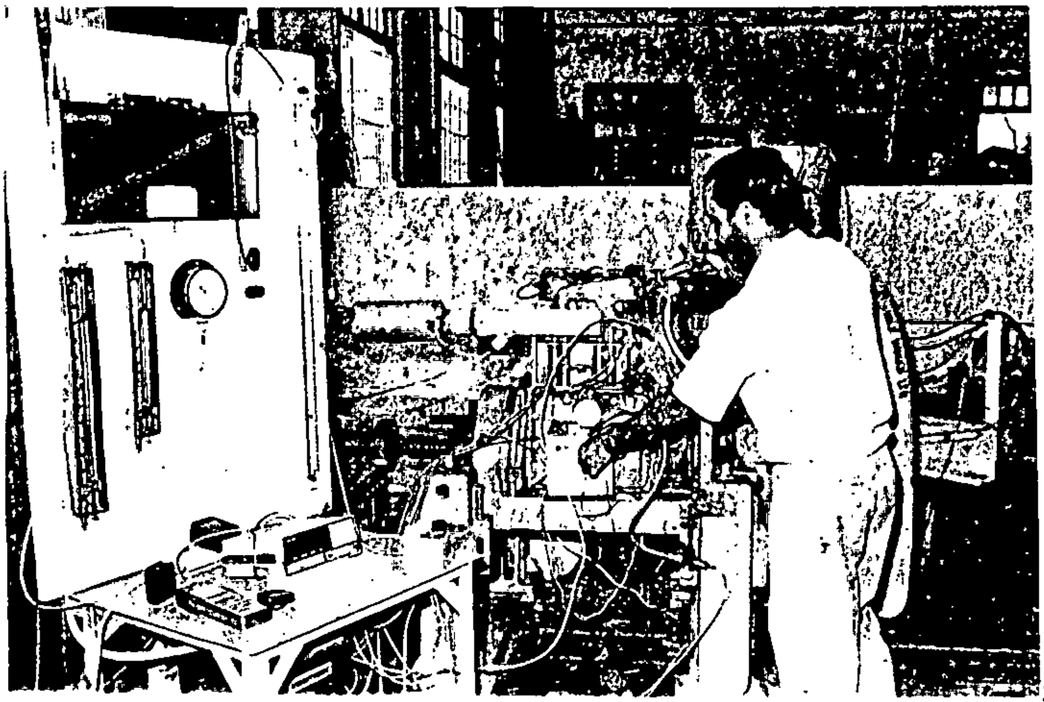


Fig. 2.4 Photograph of the CNG Fueled S.I. Engine showing switch board for starting the engine.

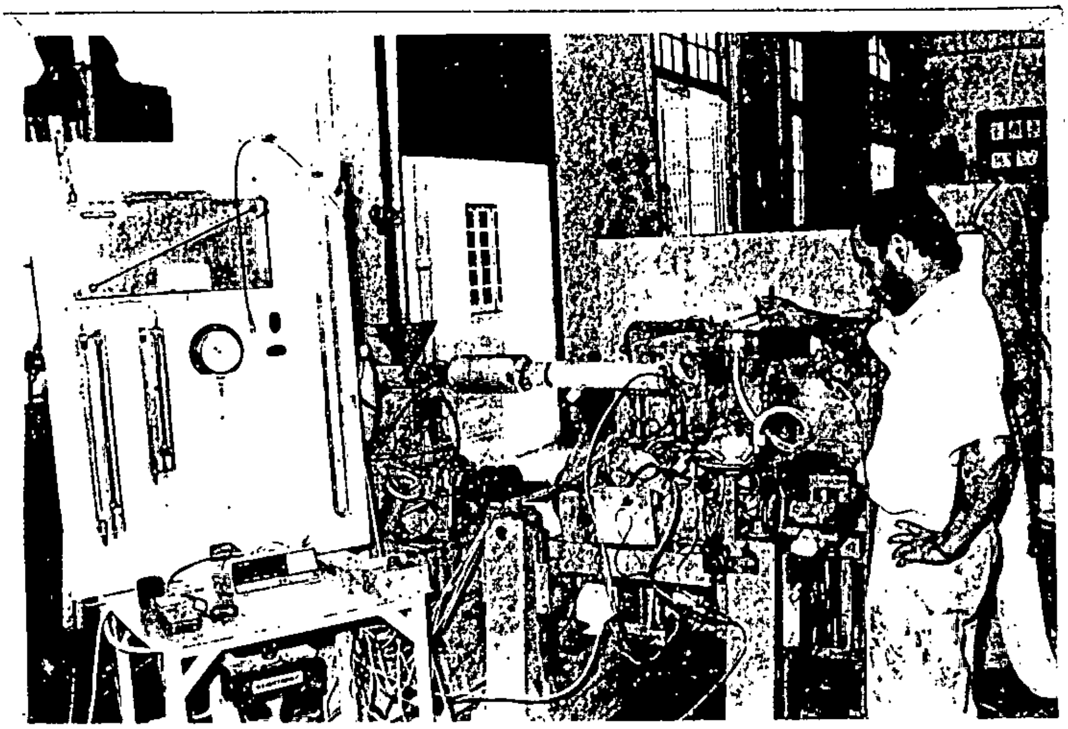


Fig. 2.5 Photograph of the CNG Fueled S.I. Engine showing its speed adjustment.

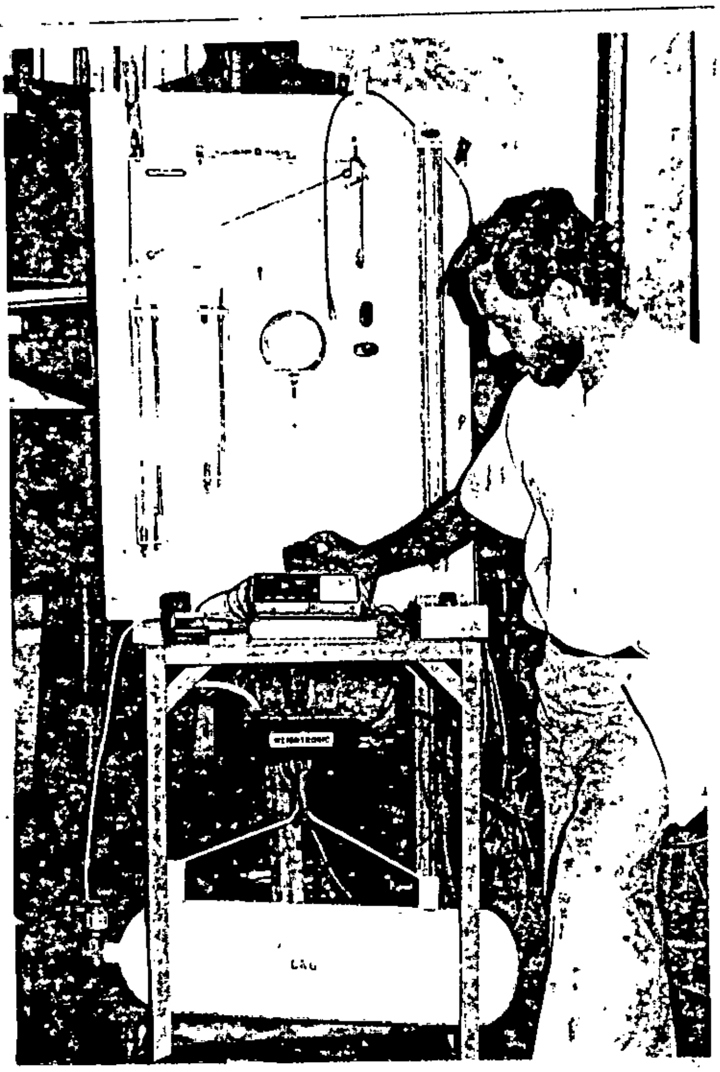


Fig. 2.6 Photograph of Weightronic Scale, Stop Watch, Adopter, CNG Service Cylinder and Pressure Gauges.

Fig. 2.7 Photograph of Inclined Manometer for measuring intake air.

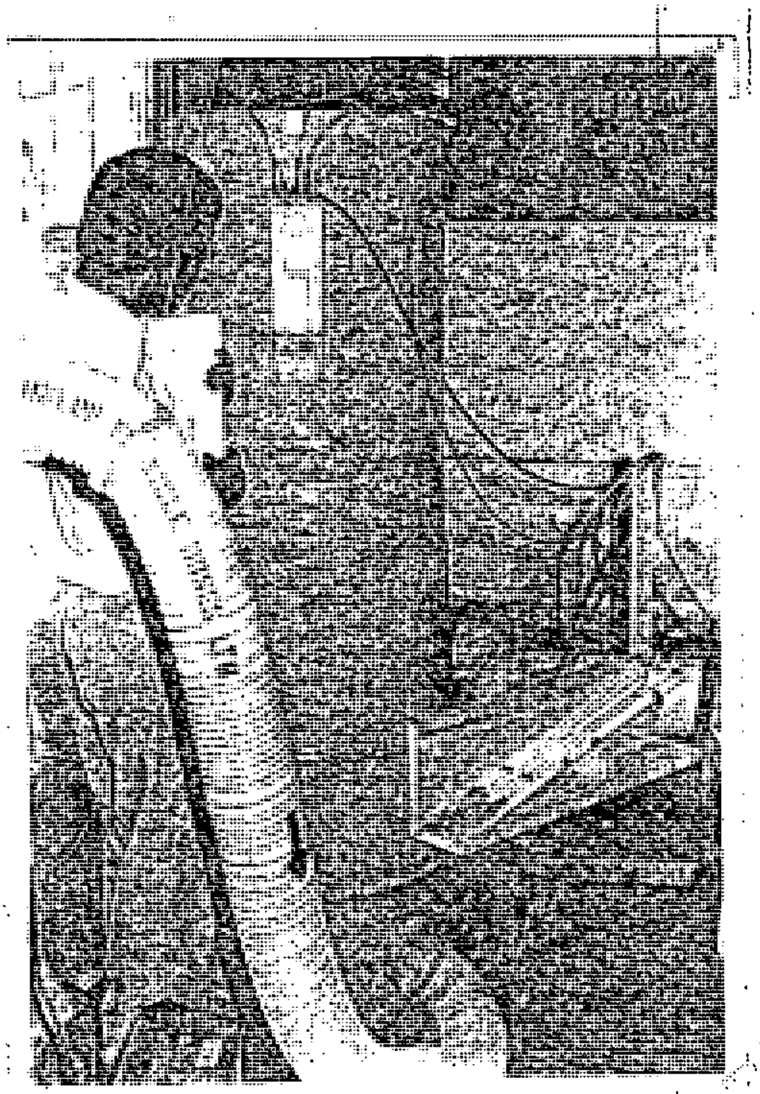
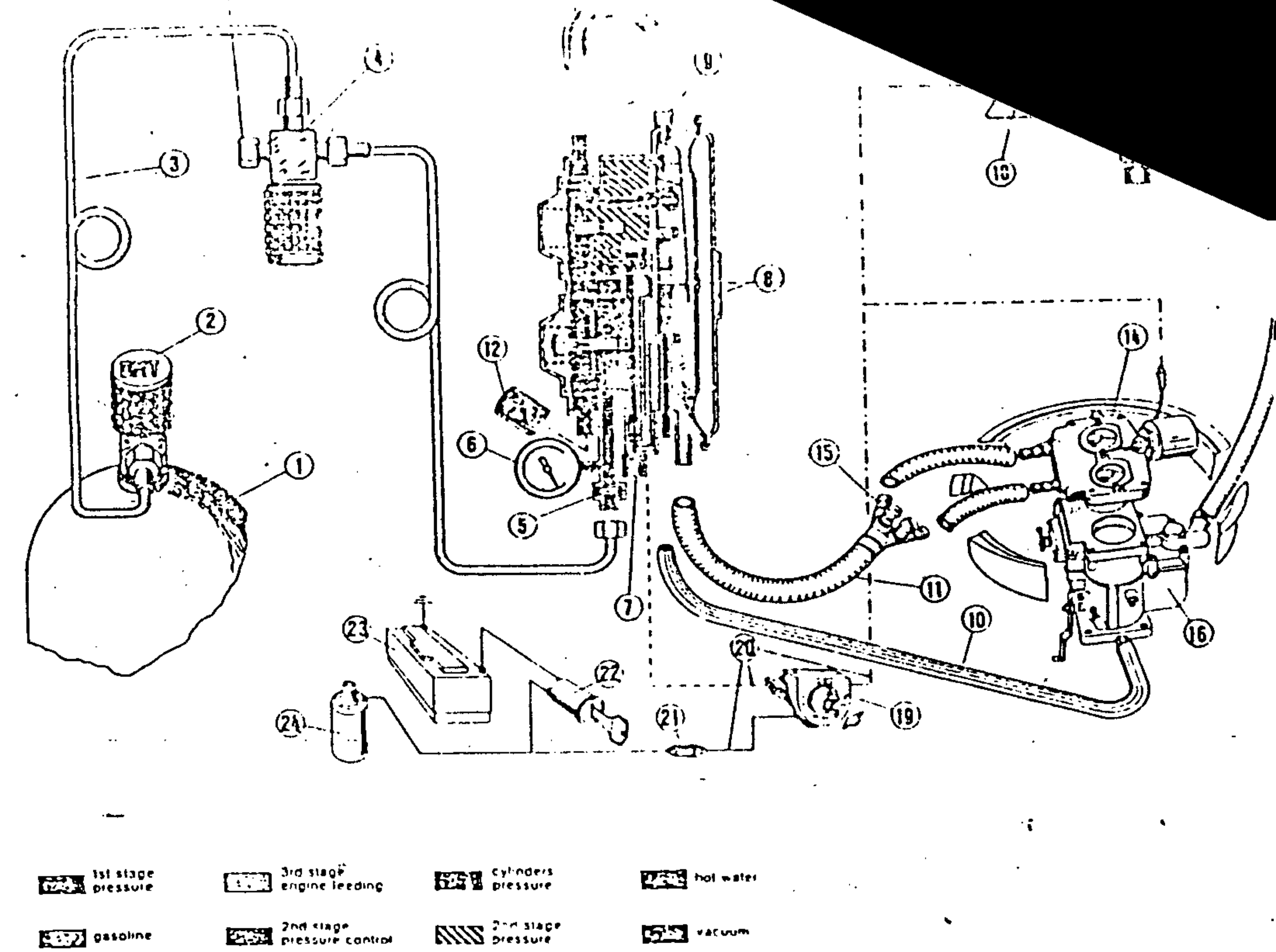


Fig. 2.8 Photograph of Sling Psychrometer.



KEY TO DIAGRAM

1 C.N.G. cylinders, connected to the main pipe-line. Inside these cylinders the gas pressure reaches 2800 PSI.
2. Cylinder stop valves.
3 Main feed pipe.
4 Filling valve, which allows the filling of the cylinders without having to move them.
5 Filter for the natural gas to the pressure regulator.
6 Pressure gauge to control the natural gas pressure from the cylinders.
7 C.N.G. solenoid valve with electric primer. This device carries out three important functions: stops the gas from the 1st stage pressure regulator, every time the ignition key is withdrawn from the dash board, thus avoiding any risk of gas leakage. It also stops the flow of natural gas when the vehicle is running on gasoline. Moreover it allows an additional enrichment (primer) at any start, thus facilitating starting when the engine is cold.
8 Pressure regulator. Decompresses the natural gas from the pressure 2800 PSI in the cylinders to its operating level and passes the gas to the mixer in the quantity required by the engine.
9 Slow-running adjusting screw.
10 Pipe, connecting the vacuum to the intake manifold.
11 Pipe, connecting the pressure regulator to the mixer.
12 Pipe, circulating water to the pressure regulator.
13 Pipe, circulating water from the pressure regulator.
14 Mixer, fixed on the carburator.
15 Maximum flow adjusting screw.
16 Carburator
17 Gasoline solenoid valve. Stops the flow of fuel when the engine is fuelled by natural gas.
18 Pipe, connecting the gasoline solenoid valve to the gasoline pump
19 Switch, from gasoline to natural gas and vice-versa. By pushing the button, the primer opens a valve permitting the gas to flow without engine demand
20 Electric connections between the switch, the solenoid valves and the primer
21 Fuse
22 Ignition key on dashboard
23 Battery
24 Coil

Fig. 2.9 Photograph of CNG Pressure Regulator and CNG Flow System including it Electrical Wiring and other accessories.

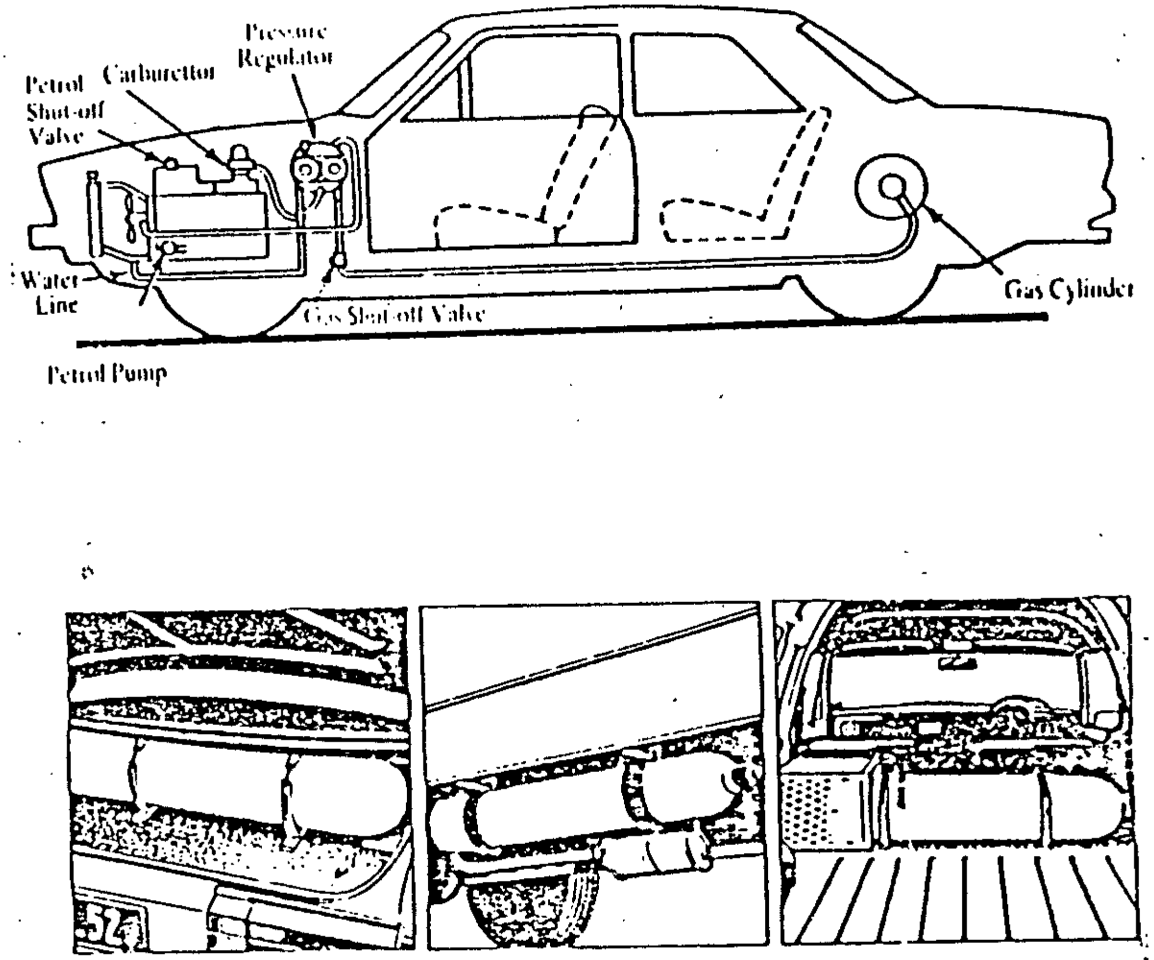
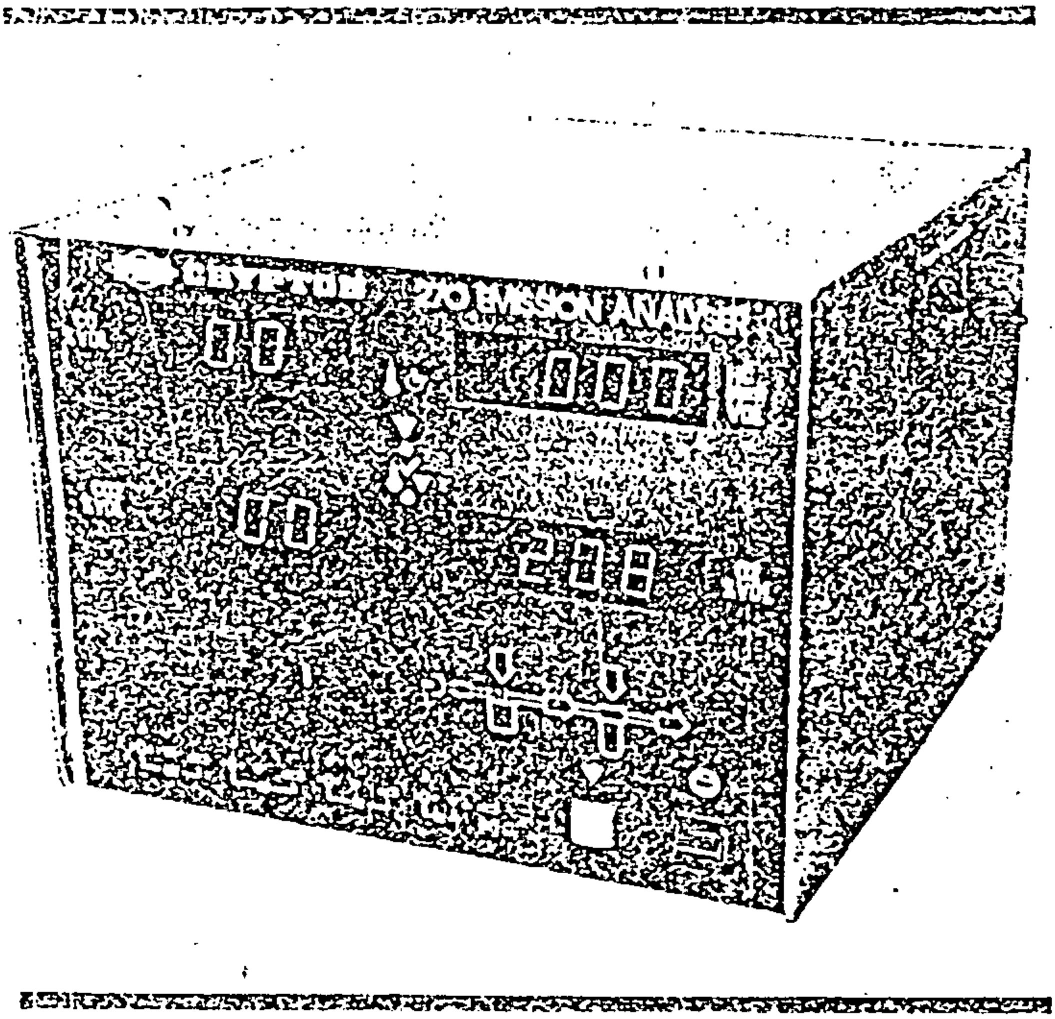


Fig. 10 Photograph of CNG Conversion Kits Installation in Gasoline Fueled Vehicles.



267 CO/HC Analyser
270 CO/HC/CO_2/O_2 Analyser

Fig. 2.11 Photograph of 270 Crypton Emissions(CO/HC/CO_2/O_2) Analyser.

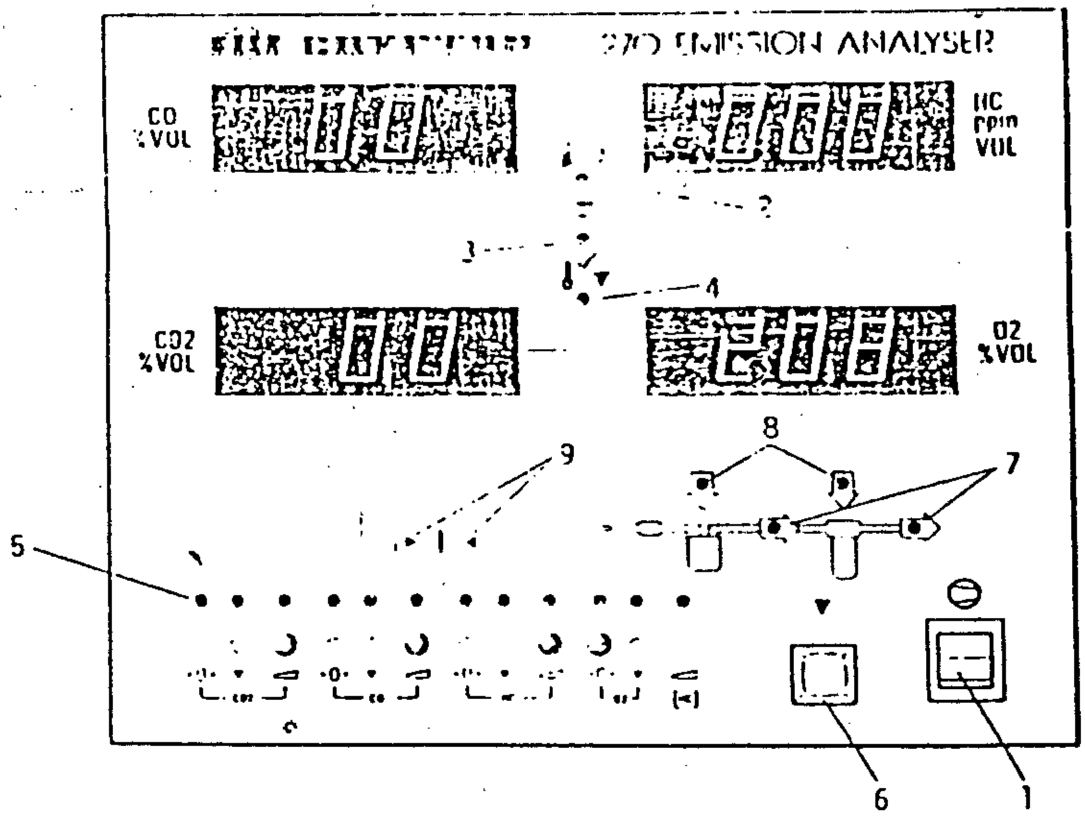


Fig. 2.12 Photograph of Front Pannel of 270 Crypton Emissions Analyser.

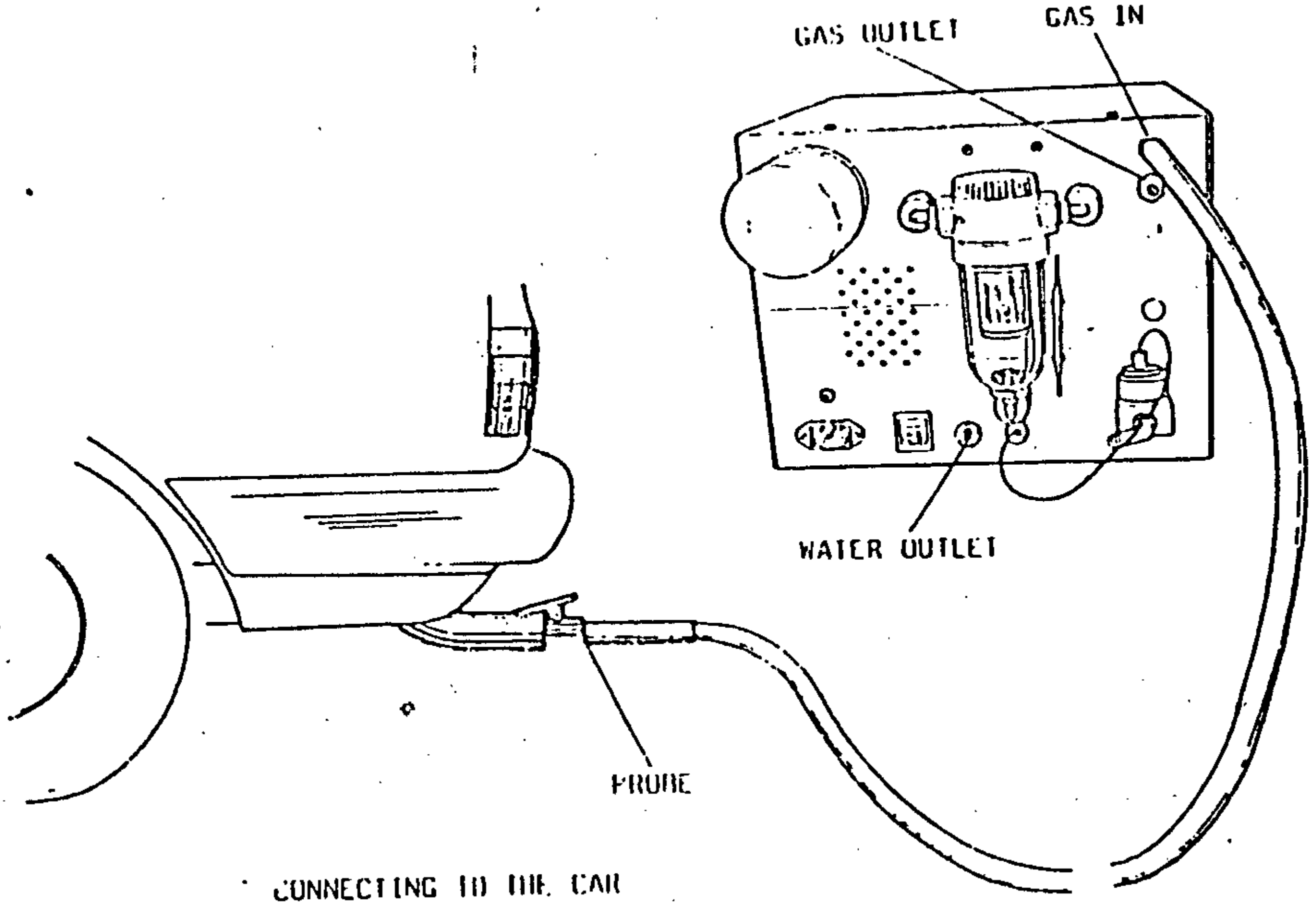


Fig. 2.13 Photograph of Emission Analyser fitted with vehicle. during observation of emissions from running vehicle.

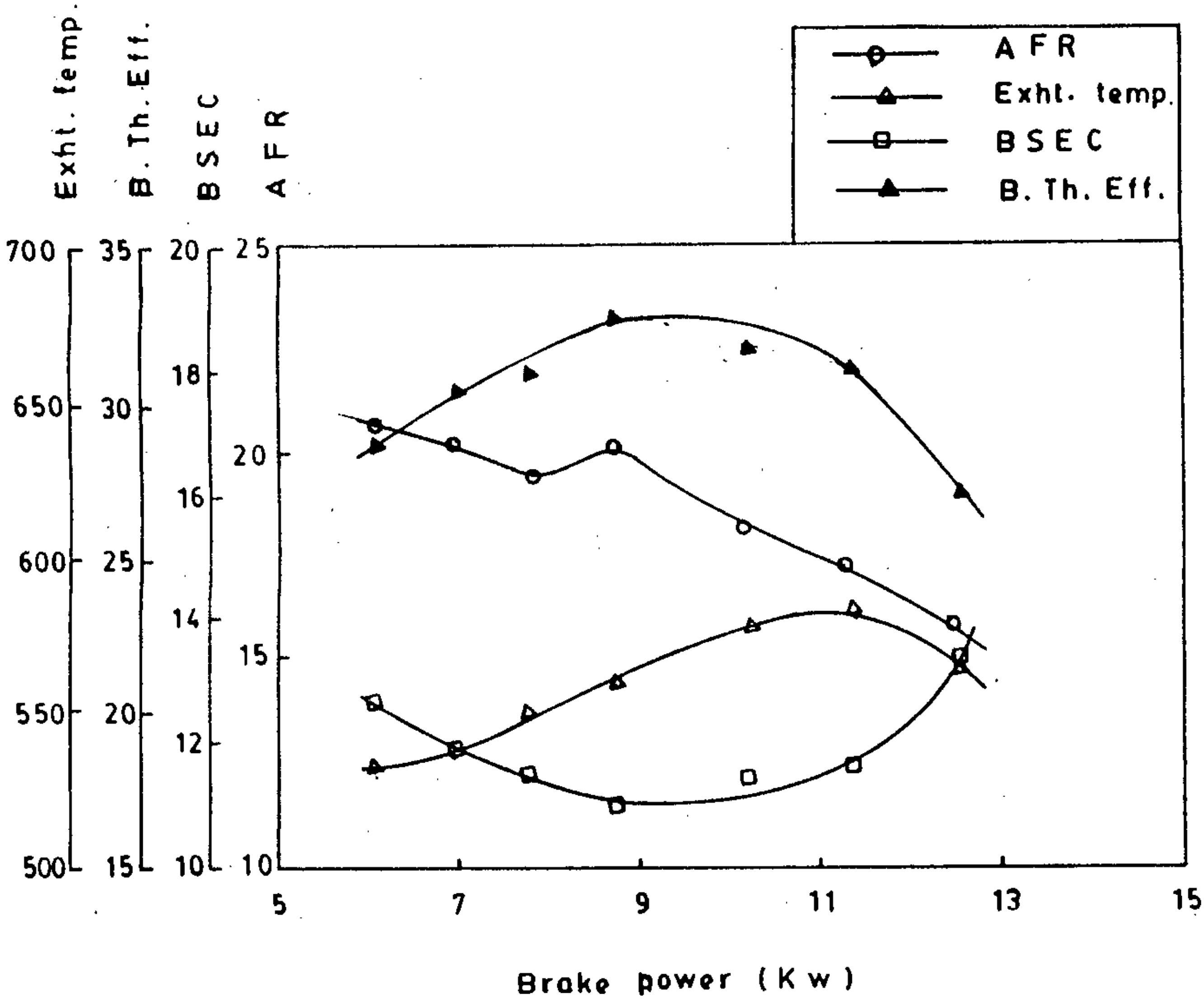


Fig. 3.1 The variation of brake specific energy consumption (BSEC), brake thermal efficiency (B.Th.Eff), exhaust gas temperature (Exht.temp.) and air fuel ratio (AFR) with respect to brake power (Kw) at a constant speed of 1500 rpm and at an ignition timing of 15° BTDC.

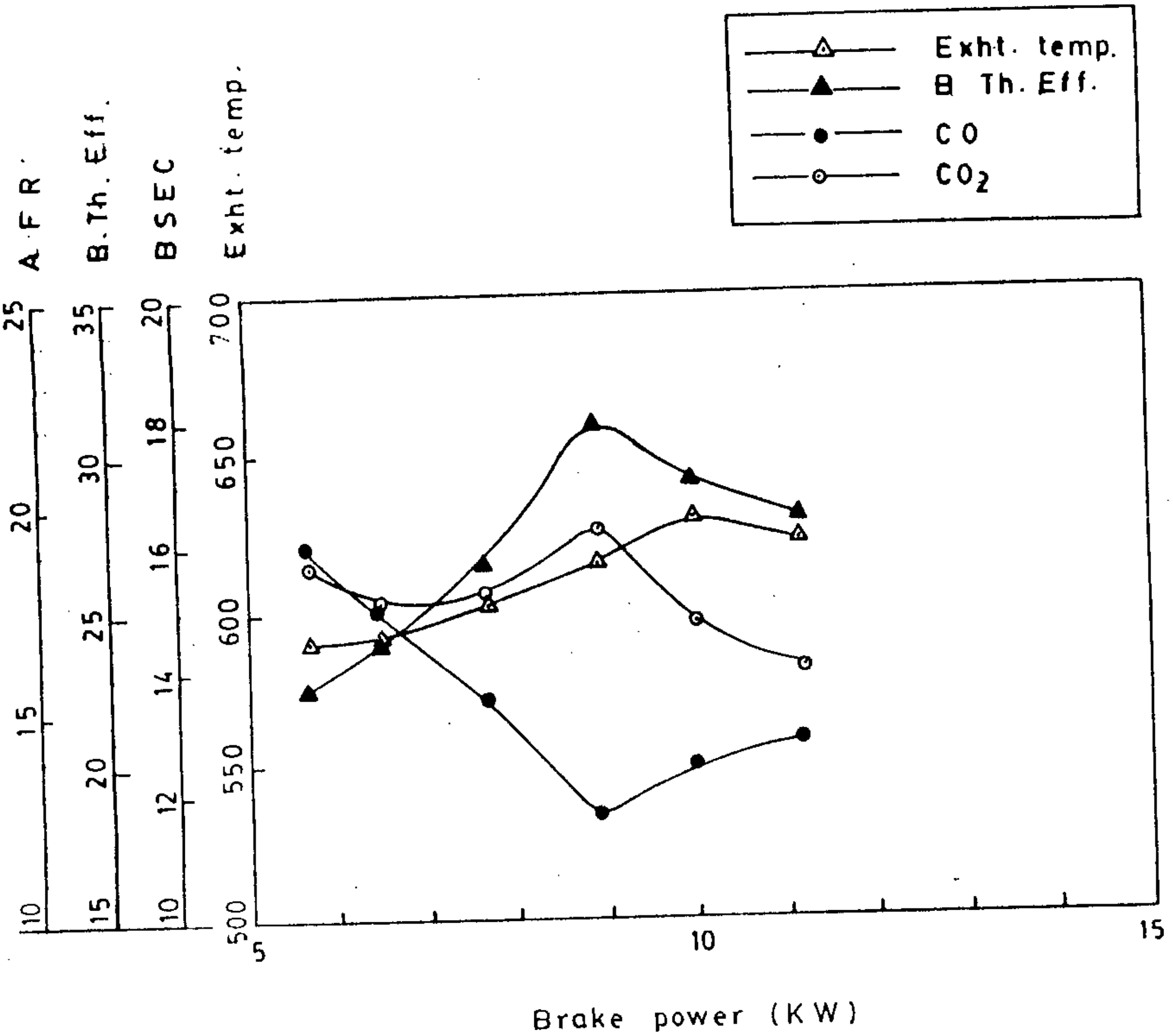


Fig. 3.2 The variation of brake specific energy consumption (BSEC), brake thermal fficiency (B.Th.Eff), exhaust gas temperature (Exht.temp.) and air fuel ratio (AFR) with respect to brake power (Kw) at a constant speed of 1500 rpm and at an ignition timing of 12° BTDC.

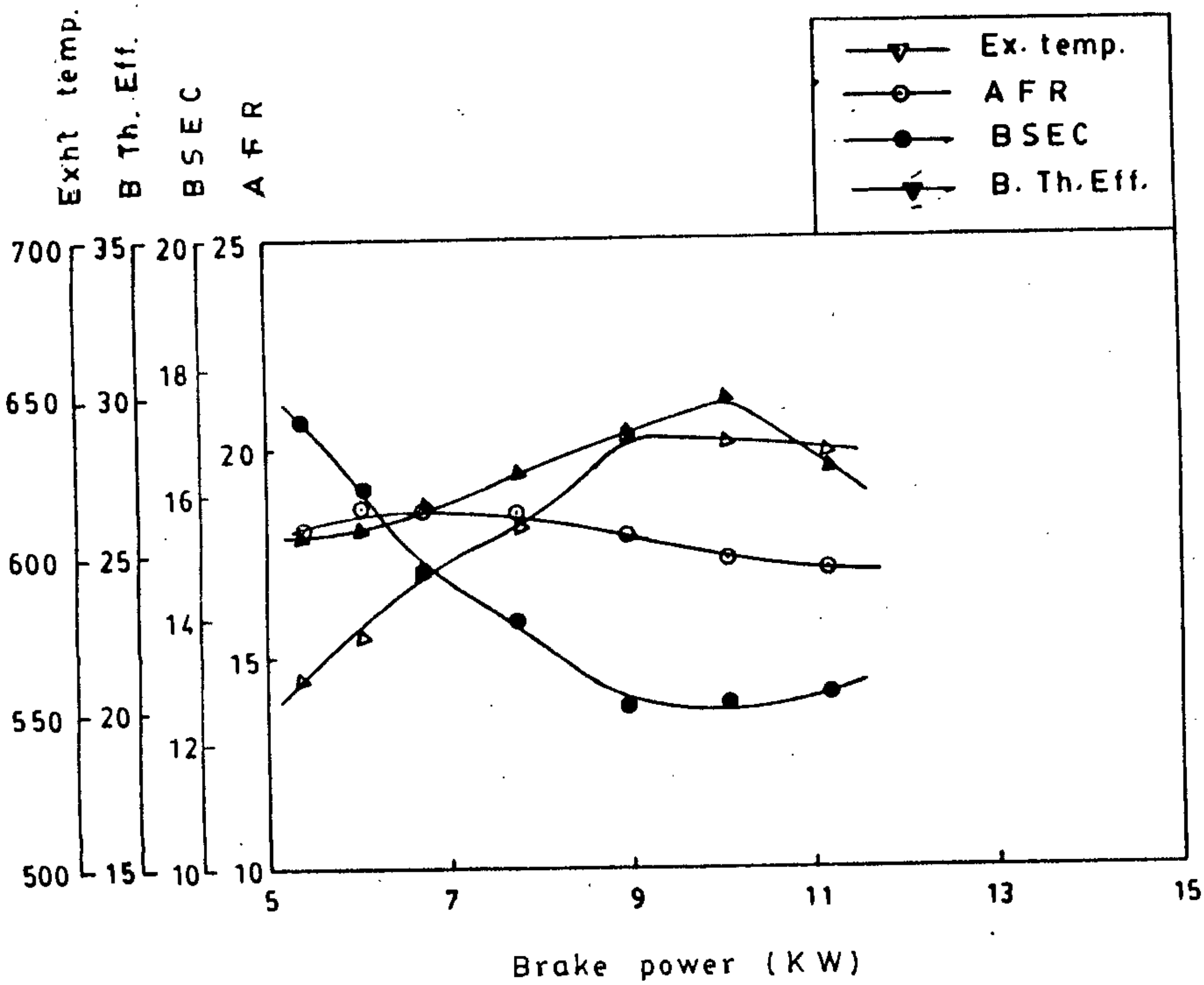


Fig. 3.3 The variation of brake specific energy consumption (BSEC), brake thermal efficiency (B.Th.Eff), exhaust gas temperature (Exht. temp.) and air fuel ratio (AFR) with respect to brake power (Kw) at a constant speed of 1500 rpm and at an ignition timing of 9° BTDC.

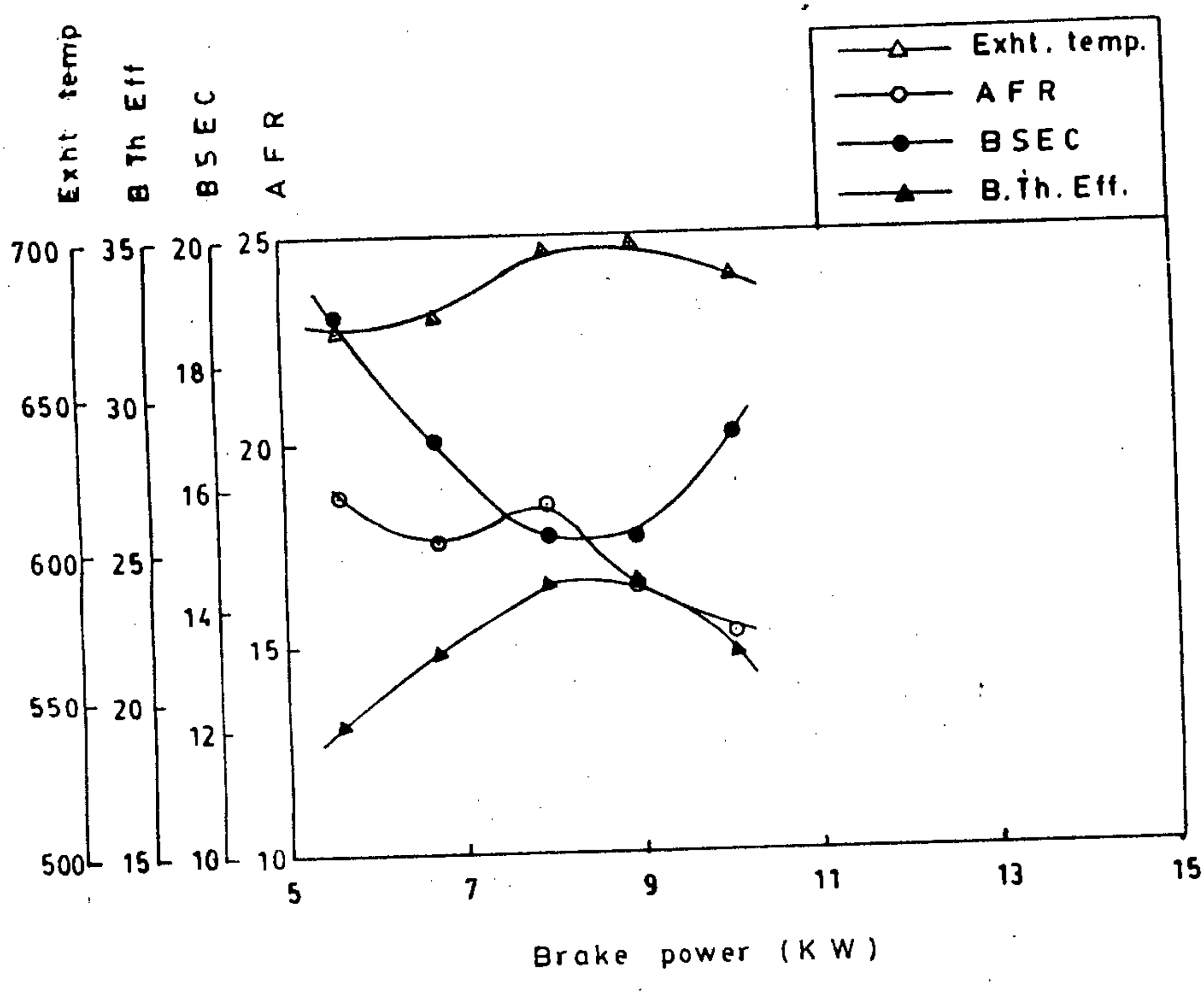


Fig.3.4 The variation of brake specific energy consumption (BSEC), brake thermal efficiency (B.Th.Eff), exhaust gas temperature (Exht.temp.) and air fuel ratio (AFR) with respect to brake power (Kw) at a constant speed of 1500 rpm and at an ignition timing of 6° BTDC.

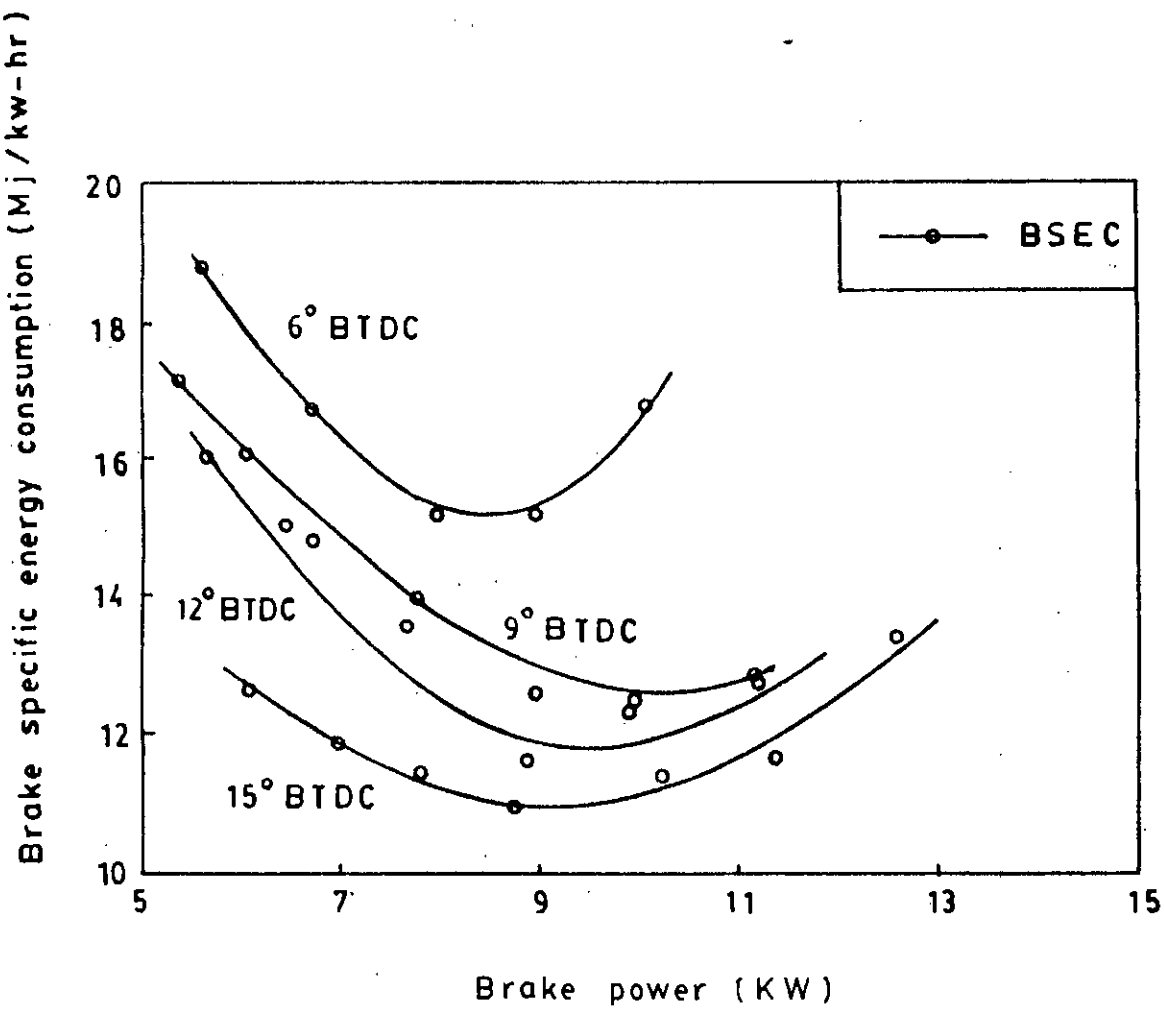


Fig. 3.5 The variation of brake specific energy consumption (BSEC) with respect to brake power (Kw) at different ignition timing from 6°BTDC to 15°BTDC at a constant speed of 1500 rpm.

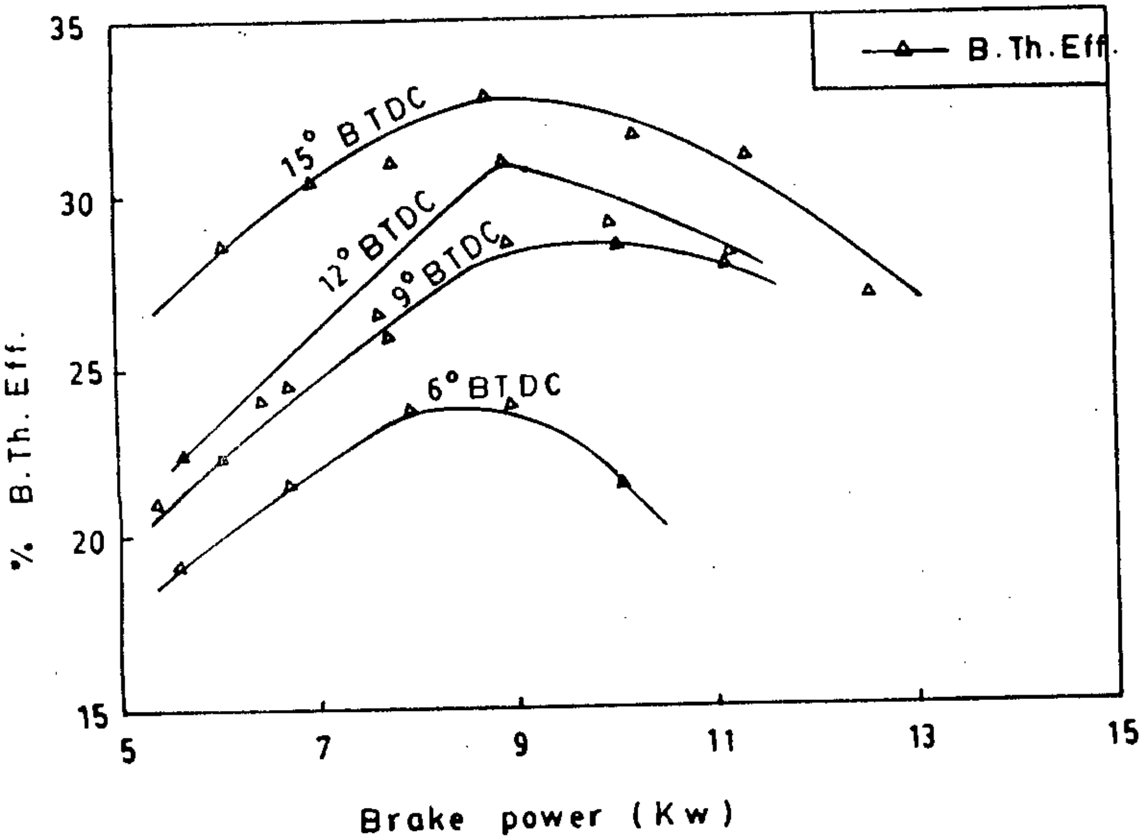


Fig. 3.6 The variation of brake thermal efficiency (B.Th.Eff) with respect to brake power (Kw) at different ignition from 6° BTDC to 15°BTDC at a constant of speed of 1500 rpm.

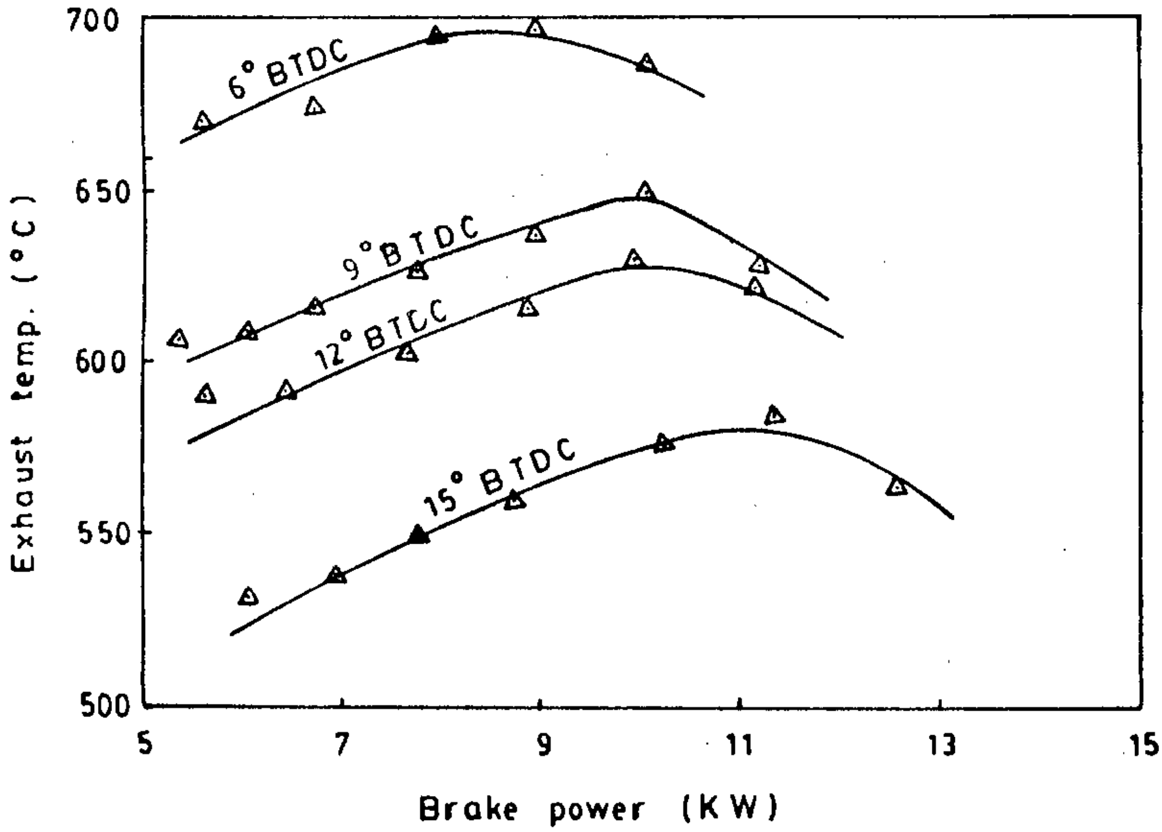


Fig. 3.7 The variation of exhaust gas temperature with respect to brake power(Kw) at different ignition timing from 6°BTDC to 15°BDTC at a constant speed of 1500 rpm.

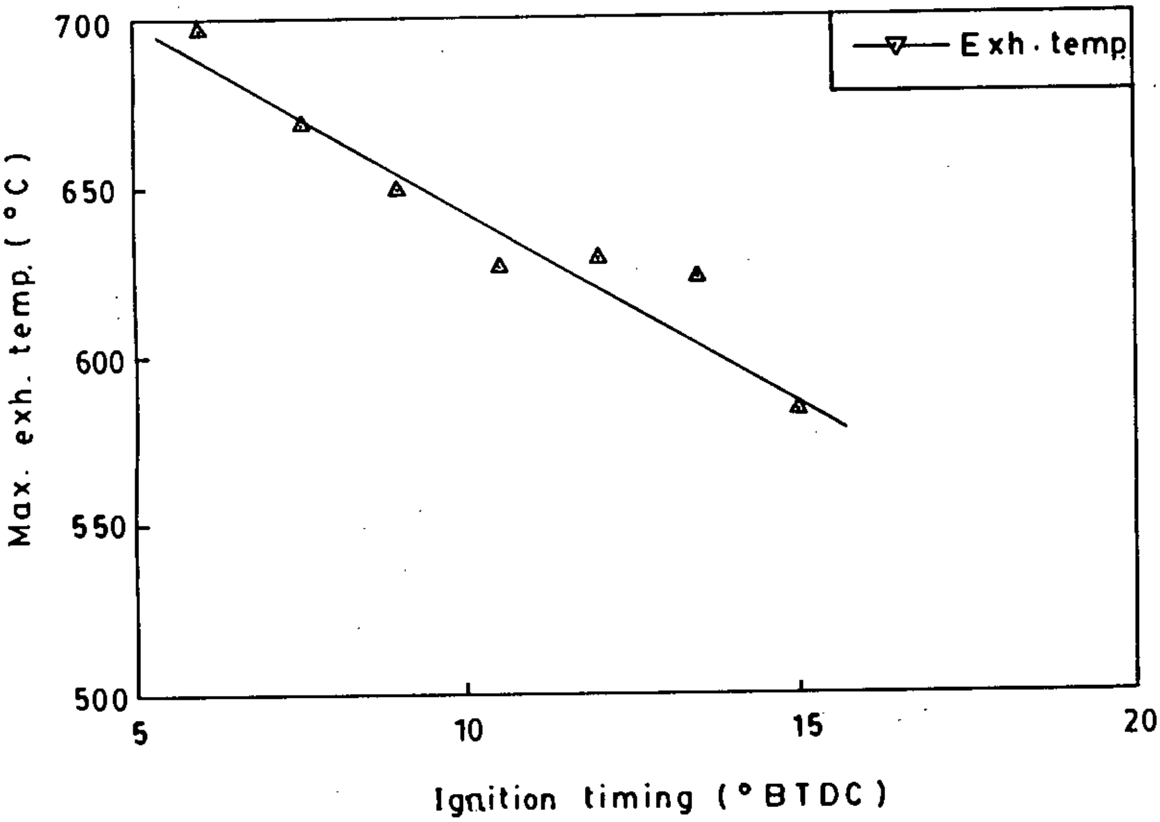


Fig.3.8 The variation of maximum exhaust gas temperature with respect to the different ignition timings from 6°BTDC to 15° BTDC at constant speed of 1500 rpm.

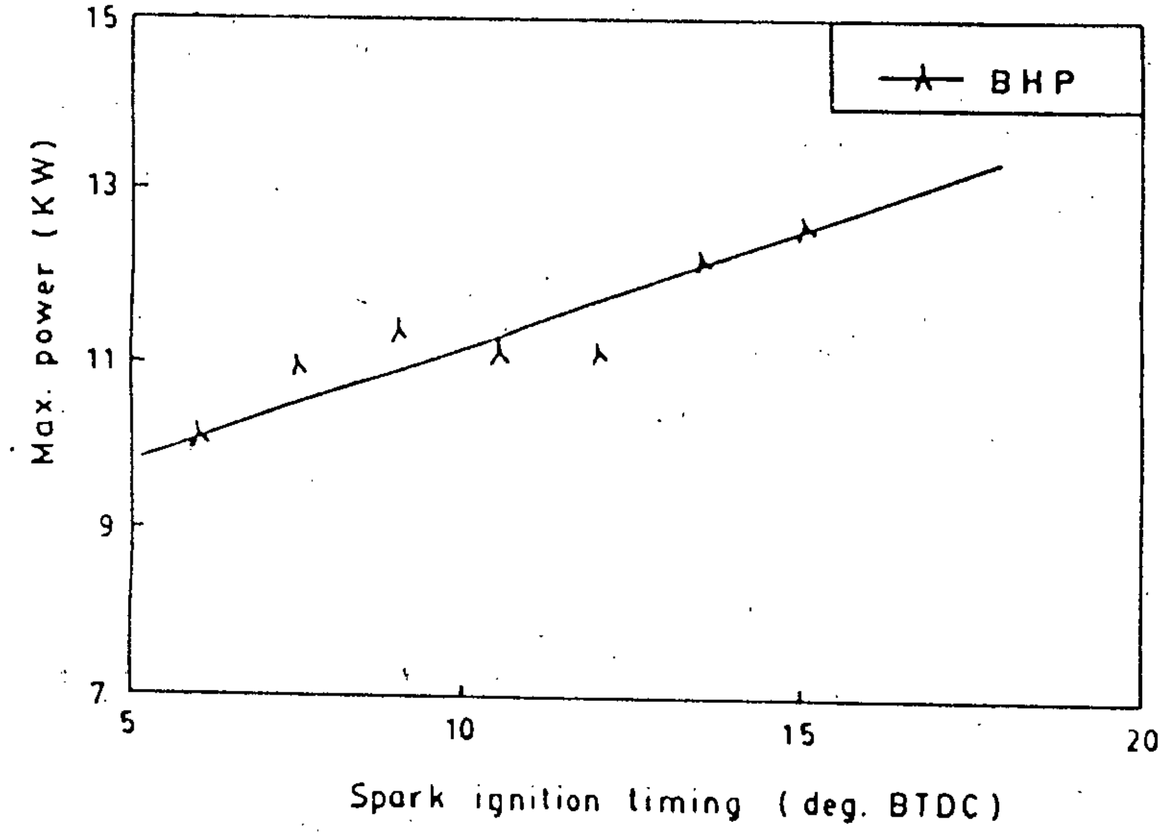


Fig.3.9 The variation of maximum load carrying capacity of the engine (Kw) with respect to the different ignition timings from 6°BTDC to 15°BTDC at a constant speed of 1500 rpm.

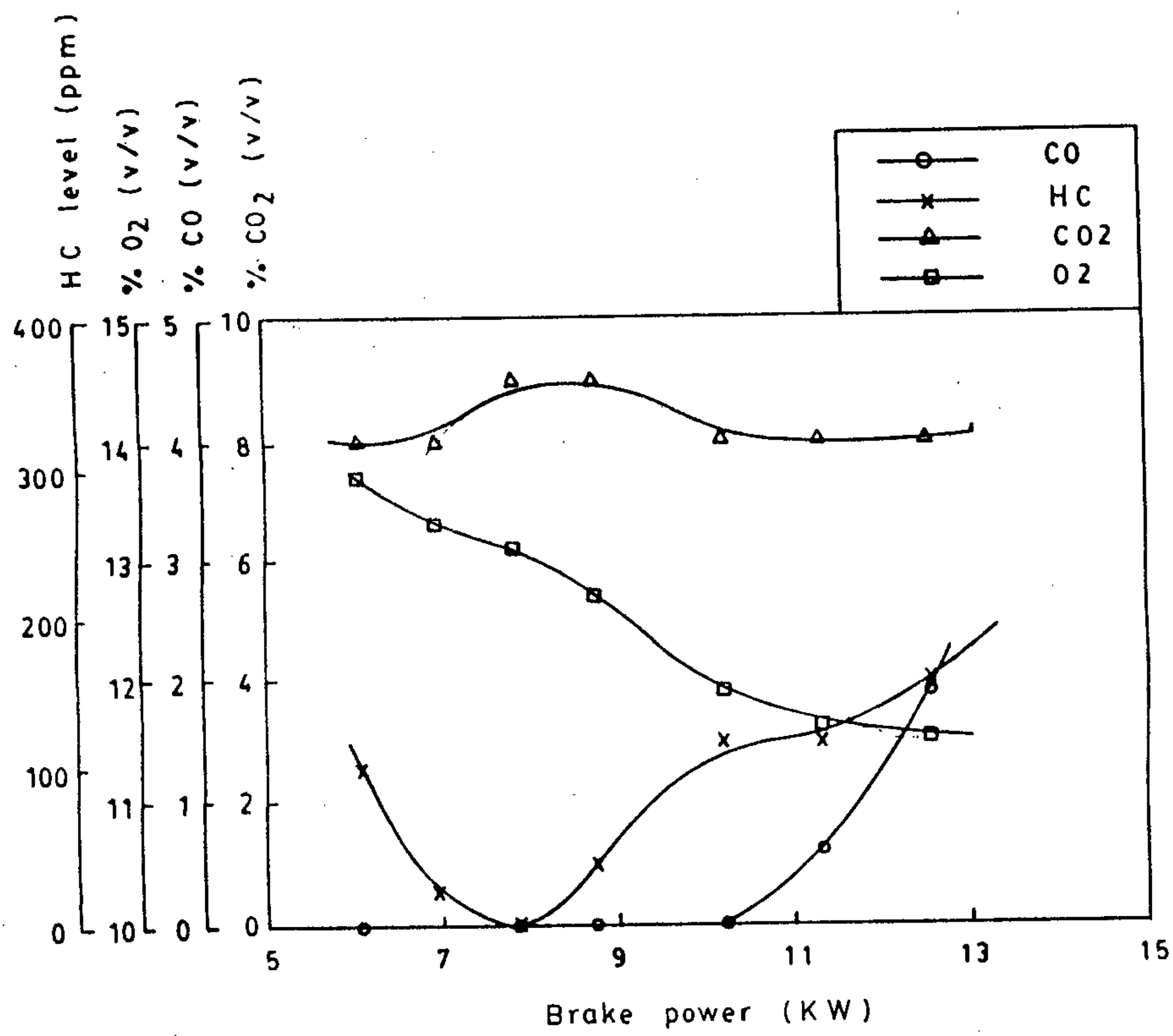


Fig. 3.10 The variation of exhaust gas emissions (CO, HC, CO_2 and O_2) with respect to brake power (Kw) at a constant speed of 1500 rpm and at an ignition timing of 15°BTDC.

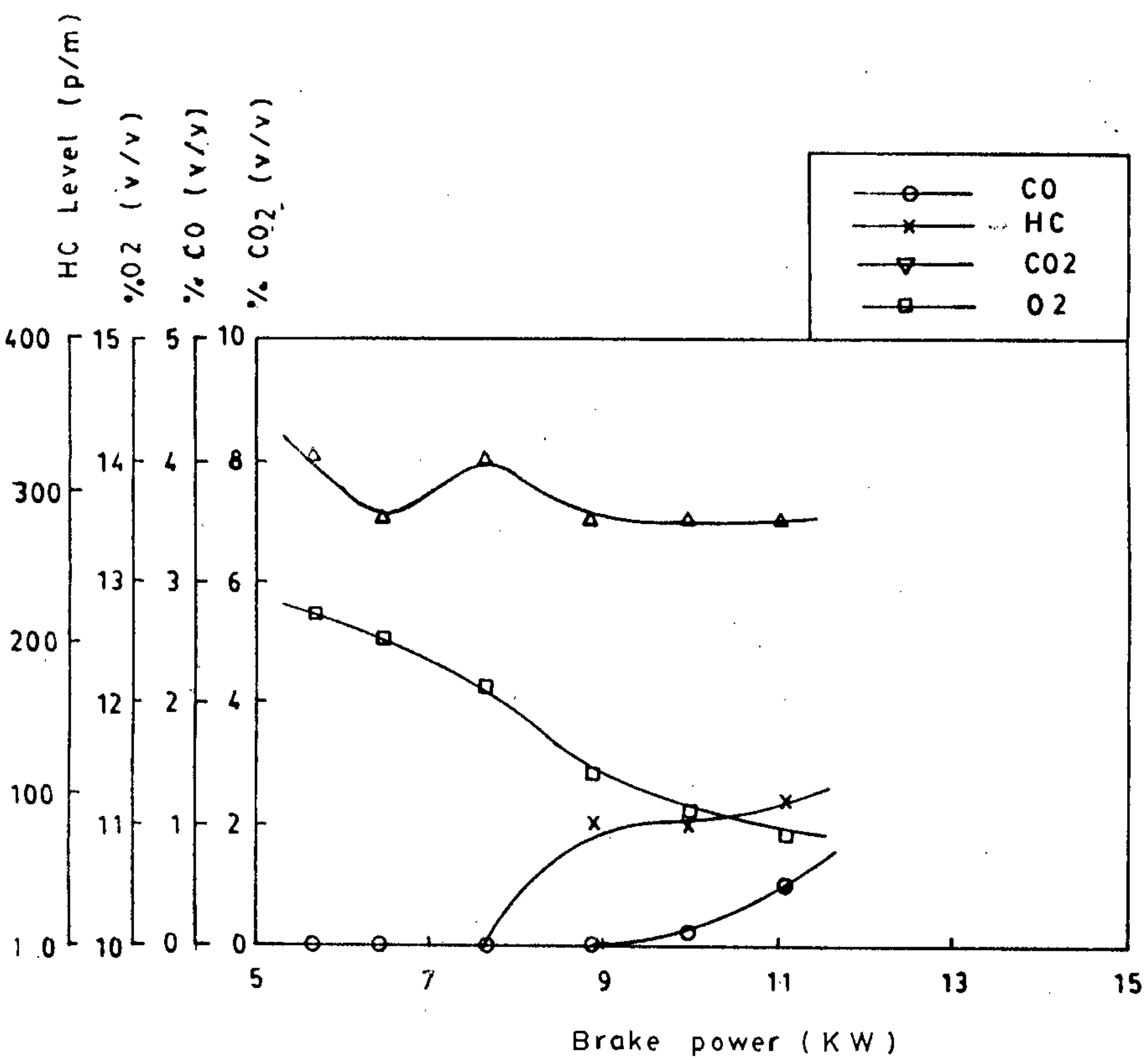


Fig. 3.11 The variation of exhaust gas emissions (CO, HC, CO_2 and O_2) with respect to brake power (Kw) at a constant speed of 1500 rpm and at an ignition timing of 12°BTDC.

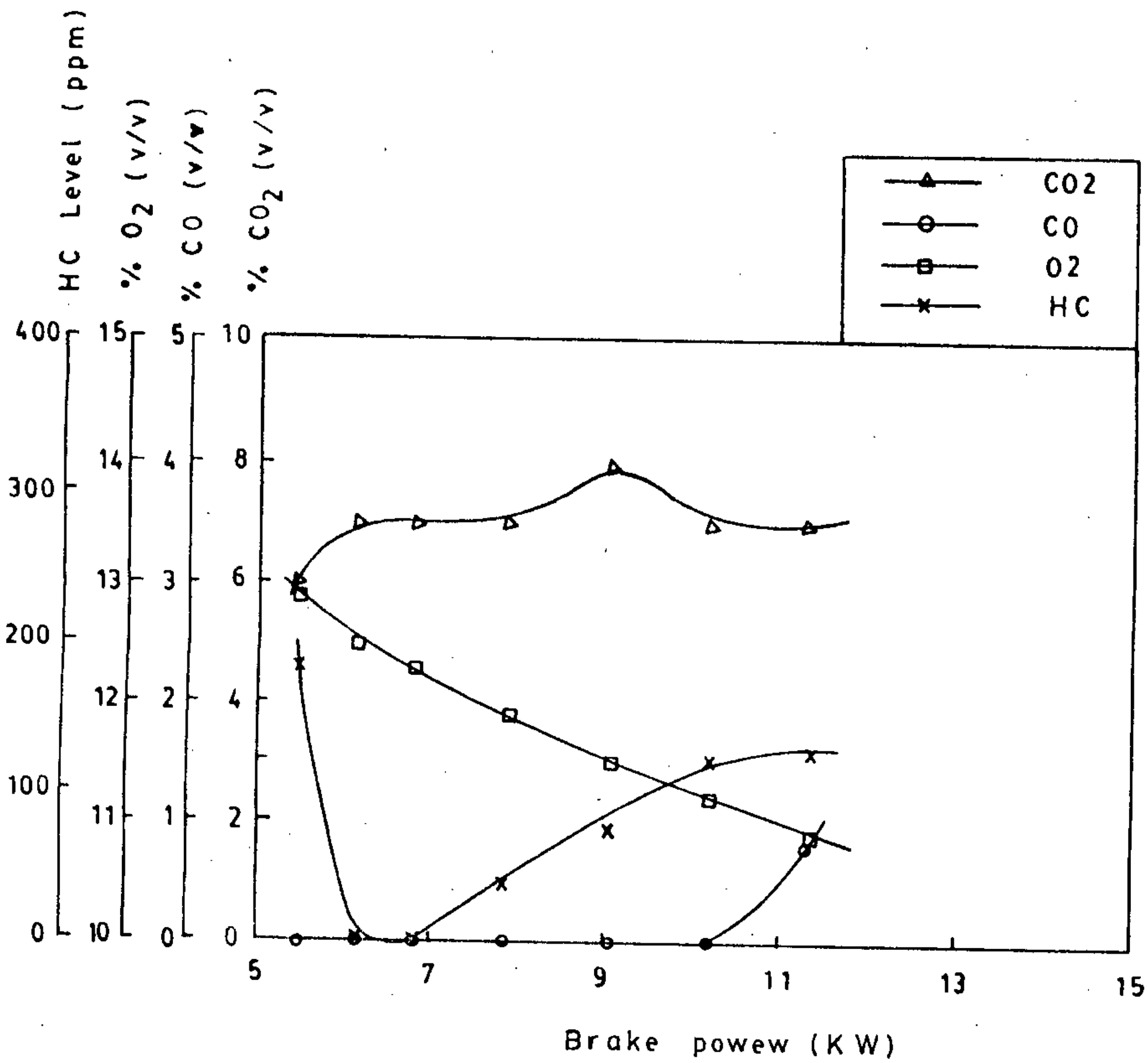


Fig. 3.12 The variation of exhaust gas emissions (CO,HC,CO_2 and O_2) with respect to brake power(Kw) at a constant speed of 1500 rpm and at an ignition timing of 9°BTDC.

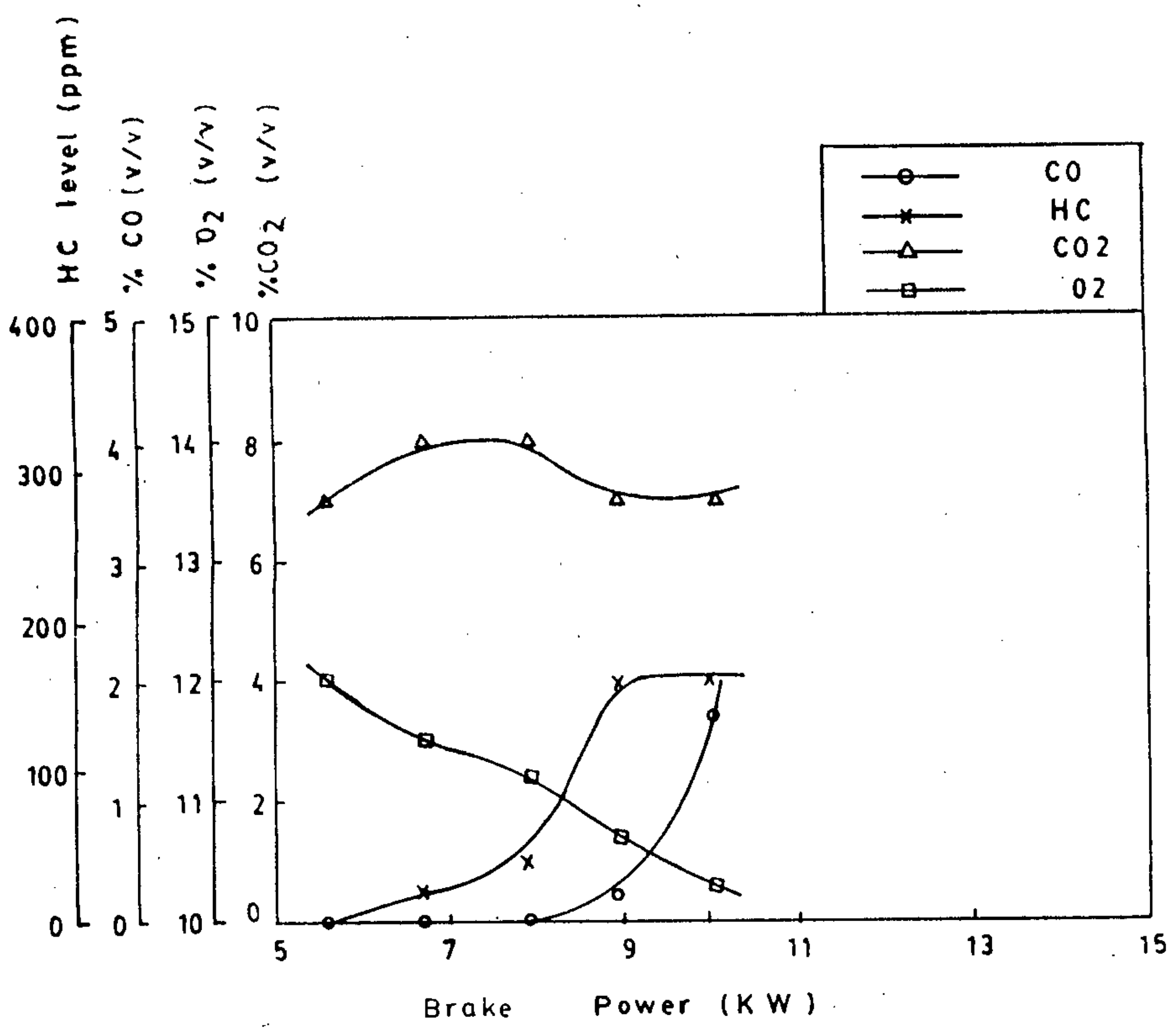


Fig. 3.13 The variation of exhaust gas emissions (CO,HC,CO_2 and O_2) with respect to brake power(Kw) at a constant speed of 1500 rpm and at an ignition timing of 6°BTDC.

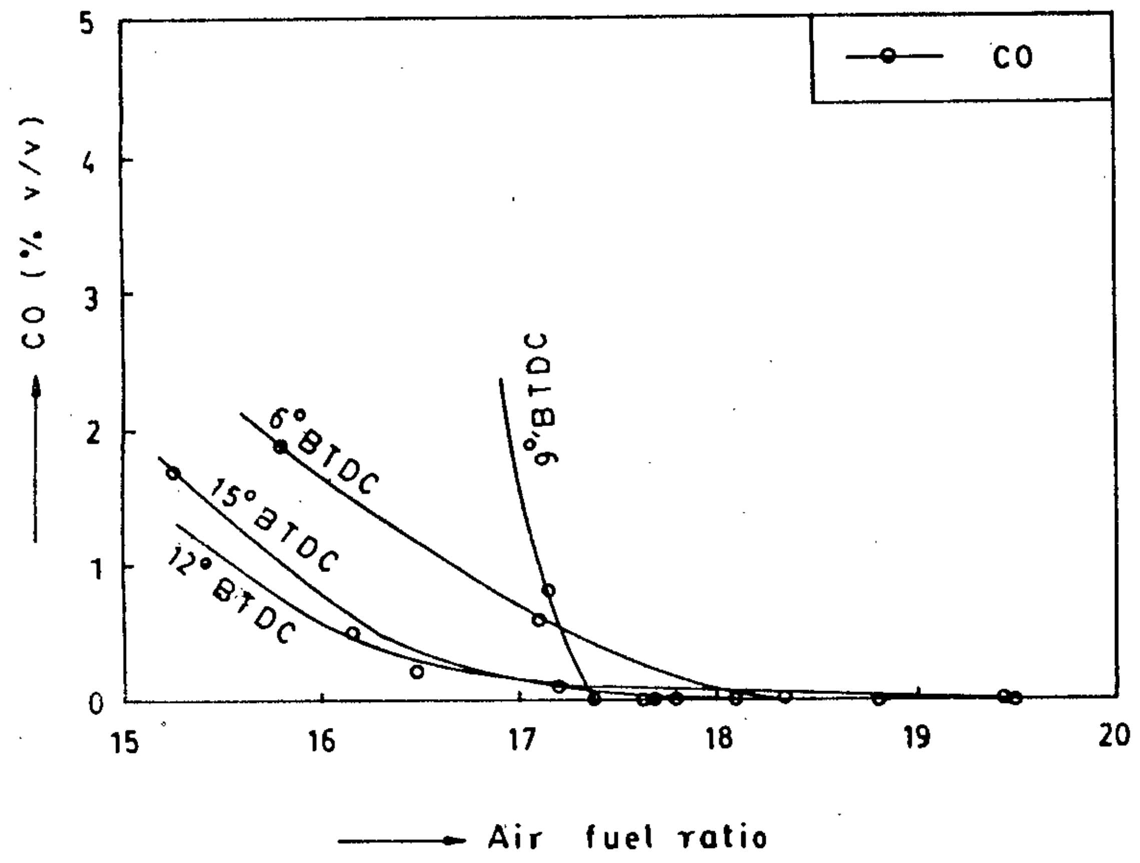


Fig. 3.14 The variation of the percentage of carbon monoxide pollution (%CO v/v) with respect to different air fuel ratio at a constant speed of 1500 rpm.

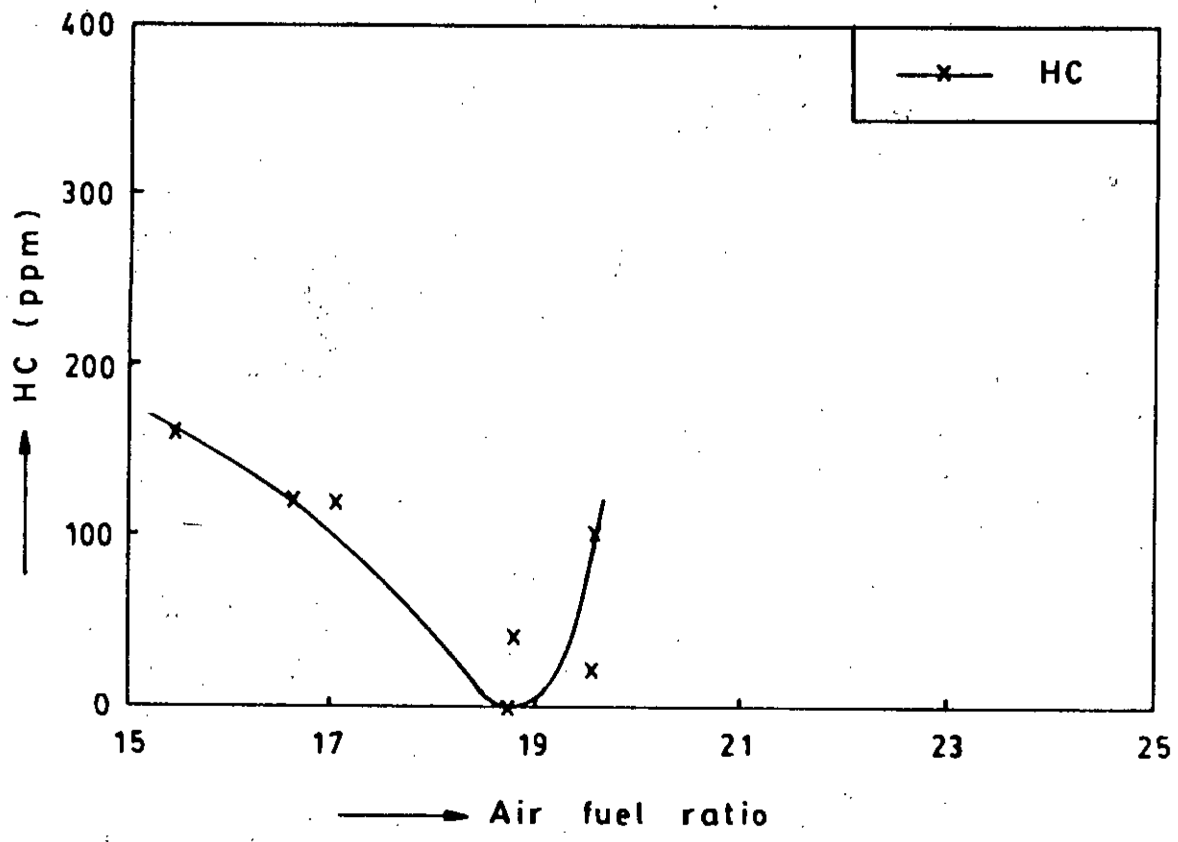


Fig. 3.15 The variation of hydrocarbon pollution (HC level in ppm) with respect to brake power (Kw) at different ignition timing from 6°BTDC to 15°BTDC at a constant speed of 1500 rpm.

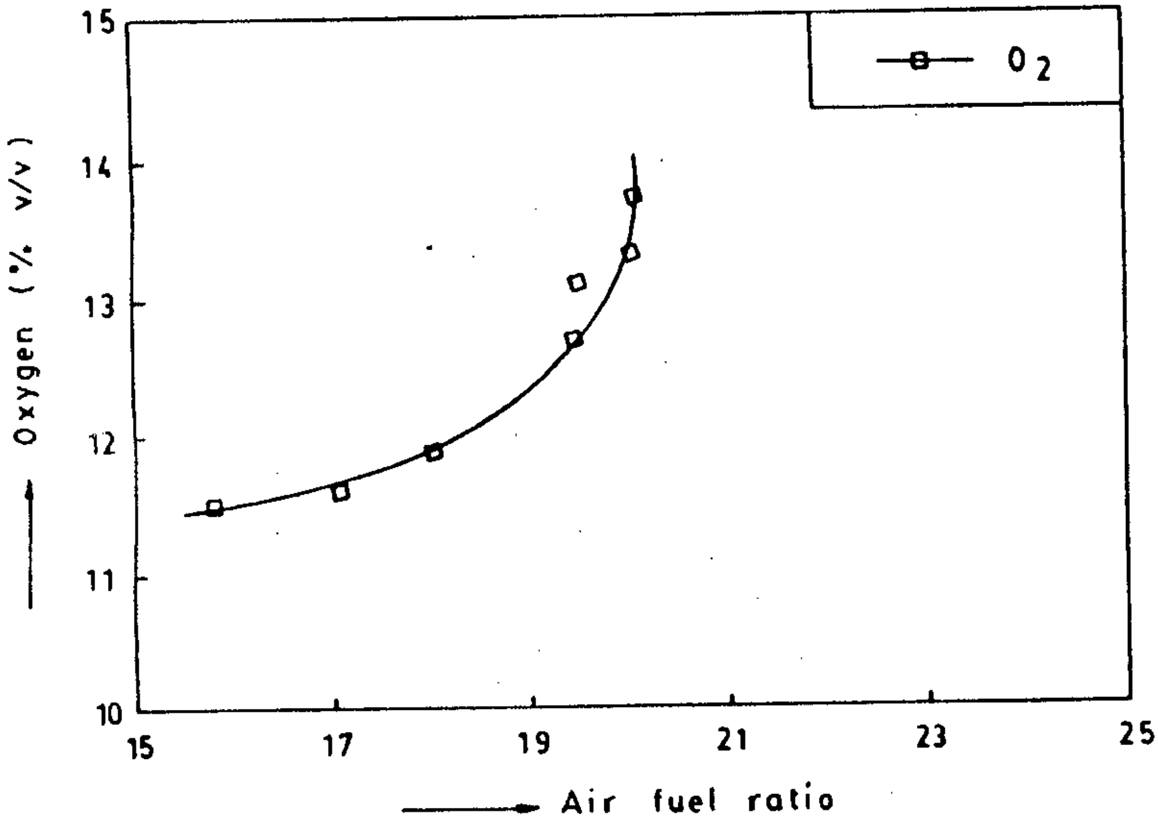


Fig. 3.16 The variation of the percentage of Oxygen emission (% O_2 v/v) with respect to different air fuel ratio at the ignition timing of 15°BTDC and at a constant speed of 1500 rpm.

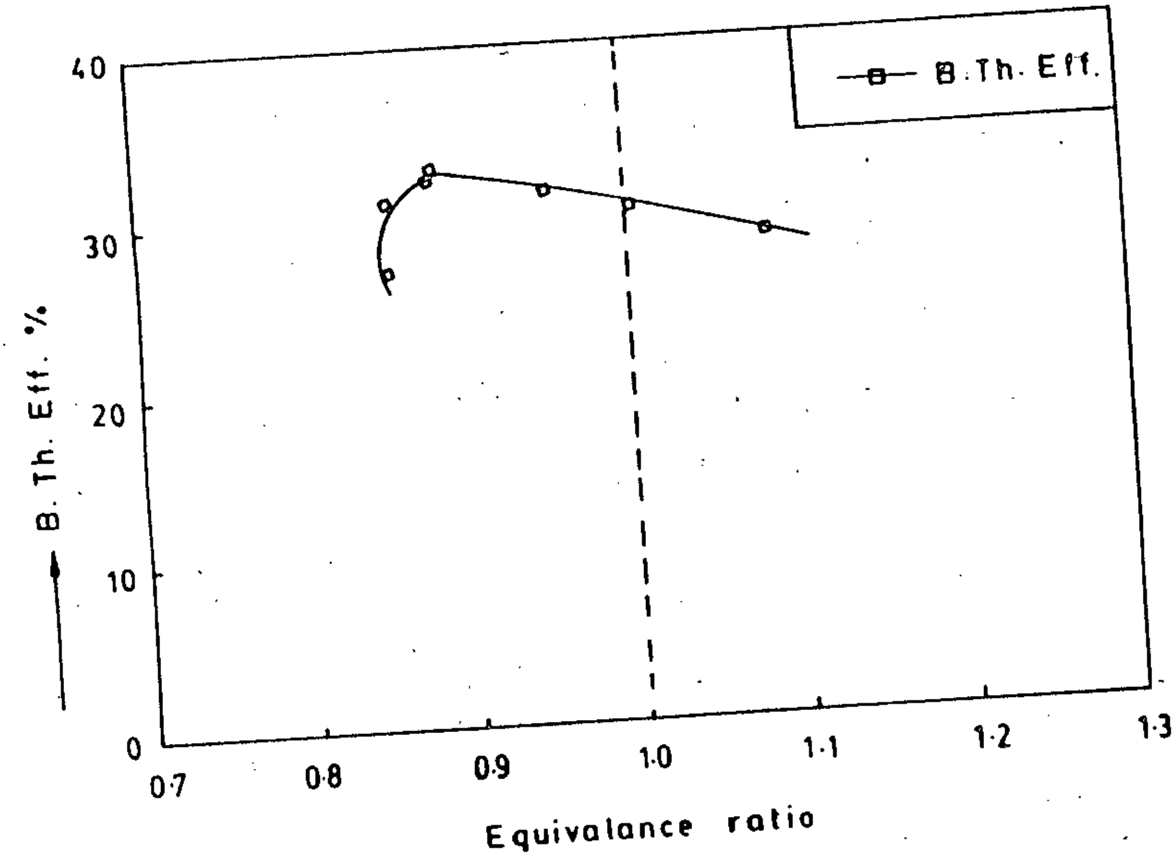


Fig. 3.17 The variation of brake thermal efficiency with respect to equivalence ratio at the ignition timing of 15°BTDC and at a constant speed of 1500 rpm.

TEST RESULT AT THE IGNITION TIMING OF 15°BTDC.

Sl. No.	Item No.	Obs.1	Obs.2	Obs.3	Obs.4	Obs.5	Obs.6	Obs.7	Unit of item.
1	Power(S)*	12.68	11.43	10.29	8.82	7.93	7.03	6.12	KW
2	W_{fs}	1.035	0.813	0.718	0.595	0.555	0.512	0.479	gm/sec.
3	W_a	16.306	13.860	12.597	11.770	10.790	10.220	9.500	gm/sec.
4	AFR	15.71	16.99	17.48	19.73	19.38	19.91	19.94	
5	BSEC	13.22	11.52	11.30	10.92	11.24	11.80	12.70	Mj/Kwhr
6	BThEff	27.23	31.23	31.85	32.94	31.75	30.53	28.63	%
7	Ex.Tem.	563	584	576	558	549	537	531	o^{C}
8	E.R.	1.09	1.01	0.98	1.15	0.89	0.86	0.86	
9	% CO	1.9	0.6	0.0	0.0	0.0	0.0	0.0	v/v
10	HC	160	120	120	40	0.0	20	101-1	ppm
11	$\%CO_2$	8.0	8.0	9.0	9.0	8.0	8.0	8.0	v/v
12	$\%O_2$	11.5	11.6	11.9	12.7	13.1	13.3	13.7	v/v
13	$\emptyset_x$	71.17	71.17	71.17	71.17	70.29	69.4	69.4	%
14	P_a				752.18				mmHg.

Power(S)*- Power at Standard condition, W_{fs}-Standard fuel flow rate.

Table no. 1

TEST RESULT AT THE IGNITION TIMING OF 13.5° BTDC.

Sl. No.	Item No.	Obs.1	Obs.2	Obs.3	Obs.4	Obs.5	Obs.6	Obs.7	Unit of item.
1	Power(S)	12.14	11.13	9.90	8.72	7.50	5.79	4.50	KW
2	W_{fe}	1.11	0.84	0.75	0.70	0.61	0.58	0.50	gm/sec.
3	W_a	16.54	14.57	13.07	12.08	11.05	10.34	9.20	gm/sec.
4	AFR	14.86	17.29	17.38	17.21	18.06	17.78	18.36	
5	BSEC	14.81	12.23	12.27	13.01	13.18	16.28	18.00	Mj/Kwhr
6	BThEff	24.30	29.44	29.33	27.80	27.37	22.18	20.00	%
7	Ex.Tem.	617	619	634	621	603	594	587	o^C
8	E.R.	1.15	0.99	0.99	0.99	0.95	0.96	0.93	
9	%CO	2.4	0.9	0.0	0.0	0.0	0.0	0.0	v/v
10	HC	80.0	60.0	0.0	0.0	20.0	20.0	20.0	ppm
11	$\%CO_2$	6.0	7.0	8.0	8.0	6.0	6.0	5.0	v/v
12	$\%O_2$	11.2	11.5	11.8	12.2	12.6	13.1	13.5	v/v
13	$\emptyset_x$	75.0	75.0	75.0	72.0	72.0	72.0	72.0	%
14	P_a			749.58					mmHg

Table No. 1.2

TEST RESULT AT THE IGNITION TIMING OF 12° BTDC.

Sl. No.	Item No.	Obs.1	Obs.2	Obs.3	Obs.4	Obs.5	Obs.6	Unit of item.
1	Power(S)	11.11	10.01	8.89	7.67	6.45	5.67	KW
2	W_f.	0.89	0.77	0.64	0.64	0.60	0.56	gm/sec.
3	W_a	14.35	13.19	12.07	11.17	10.28	10.13	gm/sec.
4	AFR	16.08	17.11	18.84	17.40	17.08	18.04	
5	BSEC	12.98	12.46	11.66	13.52	15.07	16.00	Mj/Kwhr
6	BThEff	27.74	28.88	30.87	26.63	23.89	22.50	%
7	Ex.Tem	622	629	615	602	591	590	°C
8	E.R.	1.07	1.00	0.88	0.96	0.970	0.93	
9	%CO	0.5	0.1	0.0	0.0	0.0	0.0	v/v
10	HC	95	80	80	0.0	0.0	0.0	ppm
11	$\%CO_2$	8	8	9	8	7	6	v/v
12	$\%O_2$	10.9	11.1	11.4	12.1	12.5	12.7	v/v
13	$Ø_x$	78.2	71.2	71.2	71.2	71.2	71.2	%
14	P_a			750.53				mmHg.

Table no. 1.3

TEST RESULT AT THE IGNITION TIMING OF 10.5° BTDC.

Sl. No.	Item No.	Obs.1	Obs.2	Obs.3	Obs.4	Obs.5	Obs.6	Obs.7	Unit of item.
1	Power(S)	11.11	10.57	10.01	8.90	7.79	6.57	5.79	KW
2	W_{fa}	0.90	1.19	1.18	0.85	0.68	0.54	0.36	gm/sec.
3	W_a	14.24	19.43	19.33	13.99	11.57	10.08	8.94	gm/sec.
4	AFR	15.77	16.27	16.33	16.40	16.95	18.60	24.77	
5	BSEC	13.13	18.24	19.10	15.47	14.14	13.12	10.07	Mj/Kwhr
6	BThEff	27.43	19.97	19.02	23.28	25.46	27.04	35.74	%
7	Ex.Tem.	607	620	622	627	600	584	575	°C
8	E.R.	1.09	1.06	1.05	1.05	1.01	0.92	0.69	
9	%CO	3.80	3.50	2.60	0.40	0.00	0.00	0.00	v/v
10	HC	160	135	140	20	0.0	20.0	20.0	ppm
11	$\%CO_2$	6.0	6.0	6.0	7.0	8.0	6.0	6.0	v/v
12	$\%O_2$	10.4	10.4	10.6	10.9	11.8	12.6	12.9	v/v
13	$Ø_x$	72.5	69.0	69.0	69.0	69.0	69.0	69.0	%
14	P_a				752.33				mmHg

Table no. 1.4

TEST RESULT AT THE IGNITION TIMING OF 9° BTDC.

Sl. No.	Item No.	Obs.1	Obs.2	Obs.3	Obs.4	Obs.5	Obs.6	Obs.7	Unit of item.
1	Power(S)	11.34	10.21	9.08	7.88	6.81	6.13	5.45	KW
2	W_{fs}	0.88	0.78	0.69	0.67	0.61	0.60	0.57	gm/sec.
3	W_a	15.11	13.63	12.44	12.36	11.36	11.17	10.13	gm/sec.
4	AFR	17.11	17.40	17.98	18.39	18.62	18.56	17.71	
5	BSEC	12.57	12.38	12.31	13.77	14.51	15.86	16.94	Mj/Kwhr
6	BThEff	28.64	28.09	29.24	26.14	28.81	22.70	21.25	%
7	Ex.Tem	628	650	638	627	616	608	606	°C
8	E.R.	1.08	0.99	0.96	0.93	0.92	0.92	0.97	
9	%CO	0.8	0.0	0.0	0.0	0.0	0.0	0.0	v/v
10	HC	125	120	75	40	0.0	0.0	0.0	ppm
11	$\%CO_2$	7	7	8	7	7	7	6	v/v
12	$\%O_2$	10.9	11.2	11.5	11.9	12.3	12.5	12.9	v/v
13	$Ø_x$	70.54	63.1	63.1	63.1	63.1	63.1	63.1	%
14	P_a			752.63					mmHg

Table no 1.5

TEST RESULT AT THE IGNITION TIMING OF 7.5° BTDC.

Sl. No.	Item No.	Obs.1	Obs.2	Obs.3	Obs.4	Obs.5	Obs.6	Unit of item.
1	Power(S)	10.92	9.94	8.89	7.78	6.66	5.55	KW
2	W_{fo}	0.97	0.89	0.75	0.71	0.68	0.60	gm/sec.
3	W_a	15.50	15.38	13.75	12.28	11.84	11.00	gm/sec.
4	AFR	16.00	17.26	18.32	17.28	17.39	16.16	
5	BSEC	14.39	14.51	13.67	14.78	16.54	17.51	Mj/Kwhr
7	BThEff	25.02	28.82	26.34	22.42	21.76	20.37	%
8	Ex.Tem	644	669	667	655	636	633	°C
9	E.R.	1.11	0.99	0.94	0.98	0.99	0.94	
9	%CO	2.2	0.9	0.0	0.0	0.0	0.0	v/v
10	HC	150	140	140	80	20	0.0	ppm
11	$\%CO_2$	8	8	9	7	7	5	v/v
12	$\%O_2$	10.7	10.9	11.3	11.7	12.0	12.3	v/v
13	$Ø_x$	79	80.5	79.5	80	80	80	%
14	P_a				750.19			mmHg

Table no. 1.6

TEST RESULT AT THE IGNITION TIMING OF 6°BTDC.

Sl. No.	Item No.	Obs.1	Obs.2	Obs.3	Obs.4	Obs.5	Unit of item.
1	Power(S)	10.09	8.98	7.97	6.72	5.60	KW
2	W_{fs}	1.05	0.83	0.75	0.69	0.65	gm/sec.
3	W_a	15.97	13.72	13.78	12.21	12.20	gm/sec.
4	AFR	15.16	16.51	18.32	17.63	18.70	
5	BSEC	16.86	14.97	15.25	16.63	18.69	Mj/Kwhr
6	BThEff	21.35	24.04	23.61	21.64	19.15	%
7	Ex.Tem.	687	697	695	674	670	°C
8	E.R	1.04	1.04	0.94	0.97	0.92	
9	%CO	1.7	0.2	0.0	0.0	0.0	v/v
10	HC	160	160	40	20	0.0	ppm
11	$\%CO_2$	7	7	8	8	7	v/v
12	$\%O_2$	10.3	10.7	11.2	11.5	12.0	v/v
13	$\emptyset_x$	64.38	63.41	74.99	65.48	65.48	%
14	P_a			749.92			mmHg

Table no. 1.7

APPENDIX TABLE A-1

SPECIFICATION OF THE DIESEL ENGINE CONVERTED TO C.N.G. FUELED S.I. ENGINE.

Engine	Duetz
Model	F2L912
No. of cylinders	2 cylinders
Compression ratio	17:1
Bore/Stroke	100/120 mm
Rotational speed (rated rpm)	2500
Tested speed (rpm)	1500
Minimum iddle speed (rpm)	650 - 700
Injection	With two stage combustion
Specific fuel consumption	223 gm/Kw-hr
Starting system	Electric motor started(12 V)
Cooling system	Air cooled
Battery	12 V
Cylinder order	Inline Vertical
Power, Intermetent Rating (B" to DIN 6270), no overload.	
(a) Heavy duty	(i) 22 Kw (ii) 25 Kw
(b) Light duty	(i) 24 Kw (ii) 25 Kw
Continous rating (A" to DIN 6270), at 10% overload capacity.	(i) 14 Kw (ii) 15.5 Kw

(i) for Deutz pump
(ii) for Bosh pump

APPENDIX TABLE A-2

SPECIFICATION OF CONVERTED ENGINE (CNG FUELED S.I. ENGINE)

Engine	Duetz
No. of cylinder	2
Rpm (tested)	1500
Compression ratio	12:1
Ignition system	Electronic ignition system
Spark plug (type)	CR-9E (for NGK) G- 58 (for Champion)
Spark plug size	19mm
Spark plug gap	0.3 to 0.5 mm
Dual angle	75 degree

APPENDIX TABLE A-3

COMPOSITION OF NATURAL GAS BY WEIGHT (9)

Methane	Ethane	Propane	Butane	Nitrogen	Carbondioxide
94.3502226	2.56423	1.09133	1.08266	0.2545388	0.67257

APPENDIX TABLE A-4

PROPERTIES OF NATURAL GAS(24)

1. Chemical formula	CH_4 (mainly)
2. Ultimate analysis	
Carbon	12%
Oxygen	-
Sulpher	-
3. Specific heat	2.26
4. Auto ignition temp.	730°C
5. Theretical flame temp.	2100°C
6. Stoichiometric :	
(i) Air requirement (w/w)	16.5 Kg. air/Kg. fuel
(ii) Air requirement (v/v)	-
7. Stoichiometric combustion product :	
CO_2	9.52
H_2O	19
N_2	71.48
8. Octane number	120-127

APPENDIX TABLE A-5

TABLE FOR LIMIT OF FLAMIBILITY OF GASES IN AIR AT S.T.P. WITH UPWARD PROPOSAL (13)

Gas	Lower limit (v/v)%	Upper limit(v/v)%
Methane	5.0	15.0
Ethane	3.2	12.5
Propane	2.2	9.5
n-Butane	1.9	8.5
Hydrogen	4.0	75.0
Carbon monoxide & water vapor	12.5	74.0

APPENDIX TABLE A-6

IGNITION TEMPERATURE OF COMMON FUEL GAS IN AIR AT ATMOSPHERIC PRESSURE (13)

Gas	Ignition temperature, °C
Methane	825-945
Butane	800
Ethylene	820
Hydrogen	860
CO	95
Propone	820

APPENDIX TABLE A-7

TABLE OF MAXIMUM FLAME SPEED AND CORRESPONDING MIXTURE OF GAS IN AIR (13)

Gas	Mixture % vol	Flame speed (cm/sec)
Methane	9.96	33.8
Ethane	6.28	33.8
Propane	4.54	39
Butane	3.52	37.9
Ethylene	7.40	62.3
Hydrogen	42	252

APPENDIX TABLE A-8

SPECIFICATION OF CNG CYLINDER USED IN CNG FUELED VEHICLE IN BANGLADESH (*)

CNG cylinder's working pressure	200-210 Kg/cm^2
CNG cylinder's test pressure	300 Kg/cm^2
Cylinder content	10 m^3
Cylinder water capacity	40 litres
Cylinder empty weight	39 Kg
Length of the cylinder	950 mm
Diameter of the cylinder	256 mm
Cylinder weight when filled with gas	50 Kg
Fuel equivalent	2.2 Imp.gallon/ cylinder gas
Service cylinder's weight(empty)	23 Kg

* - CNG Ltd, Bangladesh

APPENDIX TABLE A-9

TRADE BALANCE AND PETROLEUM IMPORT OF BANGLADESH (24)
(In Million U.S. Dollar)

Year	Mercendise exports (FOB)	Mercendise imports (C&F)	Trade balance	Petroleum import bill	Petroleum import as percent of export	Quantity of petroleum product imported (million tons)
1975-76	610	-1,556	-946	126	20	1.25
1979-80	723	-2,372	-1,649	383	53	1.05
1980-81	711	-2,533	-1,822	503	70.1	1.58
1981-82	626	-2,572	-1,946	561	89.6	1.58
1982-83	686	-2,309	-1,623	456	66.5	1.30
1983-84	811	-2,553	-1,542	361	44.5	1.35
1984-85	943	-2,633	-1,690	371	39.3	1.51
1985-86	953	-2,500	-1,546	337	35.4	1.72

* -

APPENDIX TABLE A-10

(Natural gas reserve in Bangladesh)

Feilds	Chatak	Sylhet	Kailas tila	Rashid pur	Habi gonj	Titas	Bakhra bad	Semu tan	Kutub dia	Begum gonj	Feni	Beani bazar	Kamta
1. Proven reserve in tncf	0.04	0.43	0.60	1.06	1.28	2.25	3.70	0.03	1.0	0.125	0.177	1.10	0.095
2. Condensate recovery bbl/mmcf	Trace	3.4	10-13.1	0.30	0.3-9	1.5	2.00	Trace	Trace	0.290	-	20.00	0.15
3 Calorific value, Gross btu/cft	1007	1.52	1050	1014	1020	1036	1022	1043	1043	1064	-	-	-
4. Chemical composition													
Methane %	99.05	96.26	95.7	98.02	97.8	97.5	94.3	96.94	95.72	94.07	92.00	97.00	94.66
Ethane %	0.24	1.99	2.6	1.2	1.5	1.4	3.4	1.70	2.87	N.A	N.A	N.A	N.A
Propane %	-	0.14	0.9	0.2	-		0.4	0.8	0.14	0.67	*	*	*
Butane %	-	0.32	0.4	0.1	-	0.3	0.6	1.01	0.31	*	*	*	*
Nitrogen %	0.67	0.98	0.2	0.25	0.7	0.15	0.4	0.86	0.365	*	*	*	*
Carbon dioxide %	0.04	0.34	0.2	0.05	-	0.25	0.5	0.35	0.065	*		*	*
Year of discovery	1959	1955	1962	1060	1963	1962	1968	1968	1977	1980	1981	1982	1982

13 (thirteen) gas feilds have been discovered in the country upto now. Out of these gas feilds only Titas, Habigonj, Haripur (Sylhet) and Chatak have been devoloped and at present producing gas from 11 (eleven wells), five at Titas, two at Habigonj, two at Haripur, one at Chatak and one at Kailashtil, producing at an verage of 212 mmcft per day.

Note: tncf- Trillion Cubic Feet - 10^{12}
mmcf- 10^{6} cft, bbl- American barrel, btu- British Thermal Unit
Source- Petrobangla.

APPENDIX TABLE A-11
(FROM BS 5514 /ISO 3046)

Annex F

Determination of water vapour pressure

The water vapour pressure ($\varphi_r\, p_{sr}$) is given in the table below in units of kPa for different values of the air temperature t_r in degrees Celsius and relative humidity φ_r.

t_r (°C)	$\varphi_r\, p_{sr}$ (kPa) φ_r 1	0.8	0.6	0.4	0.2
−10	0.3	0.2	0.2	0.1	0.1
−5	0.4	0.3	0.2	0.2	0.1
0	0.6	0.5	0.4	0.2	0.1
5	0.9	0.7	0.5	0.4	0.2
10	1.2	1	0.7	0.5	0.2
15	1.7	1.4	1	0.7	0.5
20	2.3	1.9	1.4	0.9	0.5
25	3.2	2.5	1.9	1.3	0.6
27	3.6	2.9	2.1	1.4	0.7
30	4.2	3.4	2.5	1.7	0.9
32	4.8	3.8	2.9	1.9	1
34	5.3	4.3	3.2	2.1	1.1
36	6	4.8	3.6	2.5	1.2
38	6.6	5.3	4	2.7	1.3
40	7.4	5.9	4.4	3	1.5
42	8.2	6.6	4.9	3.3	1.6
44	9.1	7.3	5.5	3.6	1.8
46	10.1	8.1	6.1	4	2
48	11.2	8.9	6.7	4.5	2.2
50	12.3	9.9	7.4	4.9	2.5

APPENDIX TABLE A-12
(FROM BS 5514 / ISO 3046)

Annex E

Determination of dry air pressure ratio

he dry air pressure ratio $\left(\frac{p_x - a\varphi_x p_{sx}}{p_r - a\varphi_r p_{sr}}\right)$ used in formula (3) is given in the table below for the value of $a = 1$ of formula references A, E and G and for different values of total barometric pressure (p_x) and water vapour pressure ($\varphi_x p_{sx}$). If the water vapour pressure is not known it can be obtained from the air temperature and relative humidity by the use of annex F.

Altitude	Total barometric pressure p_x	$\frac{p_x - a\varphi_x p_{sx}}{p_r - a\varphi_r p_{sr}}$													
		$\varphi_x p_{sx}$ (kPa)													
m	kPa	0	1	2	3	4	5	6	7	8	9	10	11	12	13
0	101.3	1.04	1.02	1.01	1.00	0.99	0.98	0.97	0.96	0.95	0.94	0.93	0.92	0.91	0.90
100	100.0	1.02	1.01	1.00	0.99	0.98	0.97	0.96	0.95	0.94	0.93	0.92	0.91	0.90	0.89
200	98.9	1.01	1.00	0.99	0.98	0.97	0.96	0.95	0.94	0.93	0.92	0.91	0.90	0.89	0.88
400	96.7	0.99	0.98	0.97	0.96	0.95	0.94	0.93	0.92	0.91	0.90	0.89	0.88	0.87	0.86
600	94.4	0.96	0.95	0.94	0.93	0.92	0.91	0.90	0.89	0.88	0.87	0.86	0.85	0.84	0.83
800	92.1	0.94	0.93	0.92	0.91	0.90	0.89	0.88	0.87	0.86	0.85	0.84	0.83	0.82	0.81
1 000	89.9	0.92	0.91	0.90	0.89	0.88	0.87	0.86	0.85	0.84	0.83	0.82	0.81	0.80	0.79
1 200	87.7	0.90	0.89	0.88	0.87	0.86	0.85	0.84	0.82	0.81	0.80	0.79	0.78	0.77	0.76
1 400	85.6	0.87	0.86	0.85	0.84	0.83	0.82	0.81	0.80	0.79	0.78	0.77	0.75	0.75	0.74
1 600	83.5	0.85	0.84	0.83	0.82	0.81	0.80	0.79	0.78	0.77	0.76	0.75	0.74	0.73	0.72
1 800	81.5	0.83	0.82	0.81	0.80	0.79	0.78	0.77	0.76	0.75	0.74	0.73	0.72	0.71	0.70
2 000	79.5	0.81	0.80	0.79	0.78	0.77	0.76	0.75	0.74	0.73	0.72	0.71	0.70	0.69	0.68
2 200	77.5	0.79	0.78	0.77	0.76	0.75	0.74	0.73	0.72	0.71	0.70	0.69	0.68	0.67	0.66
2 400	75.6	0.77	0.76	0.75	0.74	0.73	0.72	0.71	0.70	0.69	0.68	0.67	0.66	0.65	0.64
2 600	73.7	0.75	0.74	0.73	0.72	0.71	0.70	0.69	0.68	0.67	0.66	0.65	0.64	0.63	0.62
2 800	71.9	0.73	0.72	0.71	0.70	0.69	0.68	0.67	0.66	0.65	0.64	0.63	0.62	0.61	0.60
3 000	70.1	0.72	0.71	0.70	0.69	0.68	0.67	0.66	0.64	0.63	0.62	0.61	0.60	0.59	0.58
3 200	68.4	0.70	0.69	0.68	0.67	0.66	0.65	0.64	0.63	0.62	0.61	0.60	0.59	0.58	0.57
3 400	66.7	0.68	0.67	0.66	0.65	0.64	0.63	0.62	0.61	0.60	0.59	0.58	0.57	0.56	0.55
3 600	64.9	0.66	0.65	0.64	0.63	0.62	0.61	0.60	0.59	0.58	0.57	0.56	0.55	0.54	0.53
3 800	63.2	0.65	0.64	0.63	0.62	0.60	0.59	0.58	0.57	0.56	0.55	0.54	0.53	0.52	0.51
4 000	61.5	0.63	0.62	0.61	0.60	0.59	0.58	0.57	0.56	0.55	0.54	0.53	0.52	0.51	0.50
4 200	60.1	0.61	0.60	0.59	0.58	0.57	0.56	0.55	0.54	0.53	0.52	0.51	0.50	0.49	0.48
4 400	58.5	0.60	0.59	0.58	0.57	0.56	0.55	0.54	0.53	0.52	0.51	0.50	0.49	0.48	0.47
4 600	56.9	0.58	0.57	0.56	0.55	0.54	0.53	0.52	0.51	0.50	0.49	0.48	0.47	0.46	0.45
4 800	55.3	0.57	0.55	0.54	0.53	0.52	0.51	0.50	0.49	0.48	0.47	0.46	0.45	0.44	0.43
5 000	54.1	0.55	0.54	0.53	0.52	0.51	0.50	0.49	0.48	0.47	0.46	0.45	0.44	0.43	0.42

Annex D

Determination of the ratio of indicated power (k)

Formula (3) or (4) can be written as : $k = (R_1)^{y_1} (R_2)^{y_2} (R_3)^{y_3}$

where $R_1 = \dfrac{p_r - a\phi_r p_{sr}}{p_x - a\phi_x p_{sx}}$ or $\dfrac{p_r}{p_x}$

$R_2 = \dfrac{T_x}{T_r}$ or $\dfrac{T_{cx}}{T_r}$

$R_3 = \dfrac{T_{cr}}{T_{cx}}$

and $y_1 = m$ $\quad y_2 = n$ $\quad y_3 = q$

The value of $R = \dfrac{p_r - a\phi_r p_{sr}}{p_x - a\phi_x p_{sx}}$ can be obtained from annex E and other values of R can be calculated.

The values of m, n, q are obtained from table 1.

The table below then gives values of R^y for known ratios R and known factors y

The value of k is then obtained by multiplying together the appropriate values of R^y.

R	R^y								
	0.5	0.55	0.57	0.7	0.75	0.86	1.2	1.75	2
0.60	0.775	0.755	0.747	0.699	0.682	0.615	0.542	0.409	0.360
0.62	0.787	0.769	0.762	0.716	0.699	0.663	0.564	0.433	0.384
0.64	0.800	0.782	0.775	0.732	0.716	0.681	0.585	0.458	0.410
0.66	0.812	0.796	0.789	0.748	0.732	0.700	0.607	0.483	0.436
0.68	0.825	0.809	0.803	0.763	0.749	0.718	0.630	0.509	0.462
0.70	0.837	0.822	0.816	0.779	0.765	0.736	0.652	0.536	0.490
0.72	0.849	0.835	0.829	0.795	0.782	0.754	0.674	0.563	0.518
0.74	0.860	0.847	0.842	0.810	0.798	0.772	0.697	0.590	0.548
0.76	0.872	0.860	0.855	0.825	0.814	0.790	0.719	0.619	0.578
0.78	0.883	0.872	0.868	0.840	0.830	0.808	0.742	0.647	0.608
0.80	0.894	0.885	0.881	0.855	0.846	0.825	0.765	0.677	0.640
0.82	0.905	0.897	0.893	0.870	0.862	0.843	0.788	0.707	0.672
0.84	0.917	0.909	0.905	0.885	0.877	0.861	0.811	0.737	0.706
0.86	0.927	0.920	0.918	0.900	0.893	0.878	0.834	0.768	0.740
0.88	0.938	0.932	0.930	0.914	0.909	0.896	0.858	0.800	0.774
0.90	0.949	0.944	0.942	0.929	0.924	0.913	0.881	0.832	0.810
0.92	0.959	0.955	0.954	0.943	0.939	0.931	0.905	0.864	0.846
0.94	0.970	0.967	0.965	0.958	0.955	0.948	0.928	0.897	0.884
0.96	0.980	0.978	0.977	0.972	0.970	0.966	0.952	0.931	0.922
0.98	0.990	0.989	0.989	0.986	0.985	0.983	0.976	0.965	0.960
1.00	1.000	1.000	1.000	1.000	1.000	1.000	1.000	1.000	1.000
1.02	1.010	1.011	1.011	1.014	1.015	1.017	1.074	1.035	1.040
1.04	1.020	1.022	1.023	1.028	1.030	1.034	1.048	1.071	1.082
1.06	1.030	1.033	1.034	1.042	1.045	1.051	1.072	1.107	1.124
1.08	1.039	1.043	1.045	1.055	1.059	1.068	1.097	1.144	1.166
1.10	1.049	1.054	1.056	1.069	1.074	1.085	1.121	1.182	1.210
1.12	1.058	1.064	1.067	1.083	1.089	1.102	1.146	1.219	1.254
1.14	1.068	1.075	1.078	1.096	1.103	1.119	1.170	1.258	1.300
1.16	1.077	1.085	1.088	1.110	1.118	1.136	1.195	1.297	1.346
1.18	1.086	1.095	1.099	1.123	1.132	1.153	1.220	1.335	1.392
1.20	1.095	1.105	1.110	1.135	1.147	1.170	1.245	1.376	1.440

APPENDIX A-14

Annex C

Determination of the fuel consumption adjustment factor (β)

The table below gives values of the fuel consumption adjustment factor (β) for known values of the ratio of indicated power (k) and mechanical efficiency (η_m).

The value of k can be determined from annex D.

The value of η_m is stated by the manufacturer (see clause 10, note 4).

k	β					
	η_m					
	0,70	0,75	0,80	0,85	0,90	0,95
0,50	1,429	1,304	1,212	1,141	1,084	1,038
0,52	1,383	1,275	1,193	1,129	1,077	1,035
0,54	1,343	1,248	1,175	1,118	1,071	1,032
0,56	1,308	1,225	1,159	1,108	1,065	1,030
0,58	1,278	1,203	1,145	1,098	1,060	1,027
0,60	1,259	1,184	1,132	1,090	1,055	1,025
0,62	1,229	1,167	1,120	1,082	1,050	1,023
0,64	1,203	1,151	1,109	1,079	[illegible]	1,021
0,66	1,183	1,137	[illegible]	1,068	1,042	1,019
0,68	1,164	1,123	1,090	1,062	1,038	1,018
0,70	1,148	1,111	1,081	1,056	1,035	1,016
0,72	1,132	1,100	1,073	1,051	1,031	1,015
0,74	1,118	1,089	1,066	1,045	1,028	1,013
0,76	1,105	1,080	1,059	1,041	1,025	1,012
0,78	1,092	1,070	1,052	1,036	1,022	1,011
0,80	1,081	1,062	1,046	1,032	1,020	1,009
0,82	1,071	1,054	1,040	1,028	1,017	1,008
0,84	1,061	1,047	1,035	1,024	1,015	1,007
0,86	1,051	1,040	1,029	1,021	1,013	1,006
0,88	1,043	1,033	1,024	1,017	1,011	1,005
0,90	1,035	1,027	1,020	1,014	1,009	1,004
0,92	1,027	1,021	1,016	1,011	1,007	1,003
0,94	1,020	1,015	1,011	1,008	1,005	1,002
0,96	1,013	1,010	1,007	1,005	1,003	1,002
0,98	1,006	1,005	1,004	1,003	1,002	1,001
1,00	1,000	1,000	1,000	1,000	1,000	1,000
1,02	0,994	0,995	0,997	0,938	0,999	0,999
1,04	0,939	0,991	0,993	0,995	0,997	0,999
1,06	0,983	0,987	0,990	[illegible]	0,996	0,998
1,08	0,978	0,983	0,987	0,991	0,994	0,997
1,10	0,974	0,979	0,984	0,989	0,993	0,997
1,12	0,969	0,976	0,982	0,987	0,992	0,996
1,14	0,965	0,972	0,979	0,985	0,991	0,996
1,16	0,960	0,969	0,976	0,983	0,989	0,995
1,18	0,956	0,966	0,974	0,982	0,988	0,994
1,20	0,952	0,963	0,972	0,980	0,987	0,994

APPENDIX TABLE A-15
(From BS 5514 / ISO 3046)

Annex B

Determination of the power adjustment factor (α)

The table below gives values of the power adjustment factor (α) for known values of the ratio of indicated power (k) and mechanical efficiency (η_m).

The value of k can be determined from annex D.

The value of η_m is stated by the manufacturer (see clause 10, note 4).

k	α					
	η_m					
	0.70	0.75	0.80	0.85	0.90	0.95
0.50	0.350	0.383	0.413	0.438	0.461	0.482
0.52	0.375	0.408	0.435	0.461	0.483	0.502
0.54	0.427	0.433	0.460	0.483	0.504	0.523
0.56	0.428	0.457	0.483	0.506	0.525	0.544
0.58	0.454	0.482	0.507	0.528	0.547	0.565
0.60	0.480	0.507	0.530	0.551	0.569	0.585
0.62	0.506	0.531	0.554	0.573	0.590	0.606
0.64	0.532	0.556	0.577	0.595	0.612	0.627
0.66	0.558	0.581	0.601	0.618	0.634	0.648
0.68	0.584	0.605	0.624	0.641	0.655	0.668
0.70	0.610	0.630	0.648	0.663	0.677	0.689
0.72	0.636	0.655	0.671	0.685	0.698	0.710
0.74	0.662	0.679	0.695	0.708	0.720	0.730
0.76	0.688	0.704	0.718	0.730	0.741	0.751
0.78	0.714	0.729	0.742	0.753	0.763	0.772
0.80	0.740	0.753	0.765	0.775	0.784	0.793
0.82	0.766	0.778	0.789	0.798	0.806	0.813
0.84	0.792	0.803	0.812	0.820	0.828	0.834
0.86	0.818	0.827	0.836	0.843	0.849	0.855
0.88	0.844	0.852	0.859	0.865	0.871	0.876
0.90	0.870	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
0.92	[illegible]	[illegible]	[illegible]	0.910	0.914	0.917
0.94	[illegible]	[illegible]	[illegible]	[illegible]	0.935	0.938
0.96	0.949	[illegible]	[illegible]	[illegible]	0.957	0.959
0.98	0.974	[illegible]	[illegible]	[illegible]	0.978	0.979
1.00	[illegible]	[illegible]	[illegible]	[illegible]	1.000	[illegible]
1.02	1.026	[illegible]	1.024	[illegible]	1.022	1.021
1.04	1.052	[illegible]	1.047	[illegible]	1.043	1.042
1.06	1.078	[illegible]	1.071	[illegible]	1.065	1.062
1.08	1.104	[illegible]	1.094	[illegible]	1.086	1.083
1.10	1.130	[illegible]	1.118	1.112	1.108	1.104
1.12	1.156	[illegible]	1.141	1.135	1.129	1.124
1.14	1.182	1.173	1.165	1.157	1.151	1.145
1.16	1.208	1.197	1.188	1.180	1.172	1.166
1.18	1.234	1.222	1.212	1.202	1.194	1.187
1.20	1.260	1.247	1.235	1.225	1.216	1.207

APPENDIX TABLE A-16
(From BS 5514 / ISO 3046)

(Numerical values for power adjustment .)

Engine type	Condition		Formula reference	Factor	Exponents		
				c	m	n	q
Compression ignition oil engines and dual-fuel engines	Non-turbocharged	Power limited by air excess	A	1	1	0,75	0
		Power limited by thermal reasons	B	0	1	1	0
	Turbocharged without charge air cooling	Low and medium speed four-stroke engines	C	0	0,7	2	0
	Turbocharged with charge air cooling		D	0	0,7	1,2	1
Spark ignition engines using gaseous fuel	Non-turbocharged		E	1	0,86	0,55	0
	Turbocharged with charge air cooling	Low and medium speed four-stroke engines.	F	0	0,57	0,55	1,75
Spark ignition engines using liquid fuel	Naturally aspirated		G	1	1	0,5	0

NOTE — The factors and exponents given in table 1 have been established by tests on a number of engines to be generally representative and shall be used in the absence of any other specific information; for example in formula reference D, for an engine with the charge air cooled by engine jacket water, the value for exponent q could be zero. At present, they apply only to the types of engines specified but table 1 will be extended to include other types when sufficient data are available.

APPENDIX A-17

SAMPLE CALCULATION

A-17.1.1 AIR FLOW RATE CALCULATUION

Air flow rate is calculated using sample data of appendix table A-17.11. From B.S. 1042, air flow rate is calculated by the following formula-

$$\text{Air flow rate} = 0.01252\ CZe\ Ed^2\ (h_a\ \rho_a)^{1/2}$$

Where, W_a = Air flow rate in Kg/hr.
C = Constant of orifice.
d = Diameter of orifice.
h_a = Pressure difference across the orifice meter in mmH_2O
ρ_a = Density of the air at the orifice meter in Kg/m^3
Z, e and E -approximately equal to one.

$$W_a\ gs^{-1} = \frac{0.01252 \times 0.596 \times 1000 \times d^2 (h_a\ \rho_a)^{1/2}}{60 \times 60}$$

$$= 0.002073\ d^2 (h_a\ \rho_a)^{1/2}$$

$$\text{Again, } \rho_a = \frac{P_a}{RT_a} = \frac{752.18 \times 13.6 \times 62.4}{25.4 \times 12 \times 53.3 \times (460+91.4)}$$

$$= 0.0713\ lb/ft^3$$

$$= \frac{0.0713}{2.205} \times \frac{1}{0.0283}\ Kg/m^3$$

$$= 1.1419\ Kg/m^3 = 1.142\ Kg/m^3$$

Therefore ,

$$W_a = 0.002073 \times (30.02)^2\ (h_a\ \rho_a)^{1/2}$$

$$= 1.868188 (h_a\ \rho_a)^{1/2} = 16.306 \quad \text{for } \rho_a = 1.142\ Kg/m^3$$

$$h_a = 66.709\ mmH_2O$$

$$= 16.306\ gm/sec. \quad \text{and } t_a = 33\,^{\circ}C$$

A 17-1.2 Uncertainty of discharge coefficient

From BS 1042 : Sec 1.1 :1981, page-20, ß, D, Re_D, and k/D are assumed to be known without error, the percentage uncertainty of the value of C is equal to :

	Corner taps	Flangs taps	D and D/2 taps
ß < 0.6	0.6 %	0.6 %	0.6 %
0.6 < ß <0.8	ß %	-	-
0.6 < ß <0.75	-	ß %	ß %

Where ß is d/D

A-17.3 CALCULATION OF FUEL FLOW RATE

Initial wt. of service cylinder w_1 = 25.30 Kg, measured by weightronic scale.

Final weight of service cylinder w_2 = 24.880 Kg (at the end of observation)

Time of observation t = 337.25 sec., measured by digital tachometer

$$\text{Hence fuel flow rate } w_f = \frac{(25.30-24.880)\times 1000}{337.25}$$

$$= 1.038 \text{gm/sec}$$

A-17.3 AIR FUEL RATIO

$$\text{Air Fuel Ratio(AFR)} = \frac{\text{Air flow rate}}{\text{Fuel flow rate}} = \frac{16.306}{1.038} = 15.709$$

A-17.4 POWER CALCULATION

BHP= NLx0.001 for hydraulic dynamometer used in the experiment.

Where, BHP - Brake horse power (metric horse power)
N - Speed in rpm.
L - Indicated load on dynamometer scale in Kg.
0.001 - Constant of the dynamometer.

Hence BHP=1500x11.2x0.001 = 16.8 for N= 1500 rpm and L=11.2 Kg

Therefore, POWER = 16.8 x 0.736 =12.3648 Kw. (since 1 metric hp =0.736 Kw)

A-17.5 STANDARDIZATION OF BHP

Since the laboratory condition (site condition) were not the same as standard condition accepted by ISO ,so it needs standardization for finding out standard power and standard fuel consumption .

For standardization of BHP (brake horse power) and BSFC (brake specific fuel consumption), ISO-3046 with corresponding BS 5514 has been used.

From BS 5514 t_r = 27 °C, T_r =273+27 = 300°K and p_r = 100 Kpa are taken as standard temperature and pressure.

Where as our laboratory temperature varied from 27.5 °C to 35°C and pressure also varied . P_x=100.28 Kpa from sample data.

From the sample data (at the ignition timimg of 15 °BTDC),

t_x =	91.4	91.4	91.4	91.4	91.58	91.76	and 91.76°F
t_w =	83.3	83.3	83.3	83.3	83.3	83.3	and 83.3 °F
$Ø_x$ =	72.13,	71.13	71.13	71.13	71.14.	70.14	and 70.14 %

Now from table A-11 (Annex. F of BS 5514), Water vapor pressure ($Ø_x\ p_{sx}$) is found out for t_x=33°C (91.4 °F) and $Ø_x$ =0.7213, we get $Ø_x\ p_{sx}$ = 3.6565

Let us now find out Dry Air Pressure Ratio $R_1 = \left(\dfrac{p_x - aØ_x\ p_{sx}}{p_r - aØ_r\ p_{sr}}\right)$ from table A-12 (Annex. E of BS 5514). For the value of water vapor pressure $Ø_x\ p_{sx}$ =3.6565 and p_x =100.28 Kpa , we get

$$\text{Dry Air Pressure Ratio } (R_1) = \frac{p_x - \O_x\, p_{sx}}{p_r - \O_r p_{sr}} = 0.9869$$

Using standard temperature and room tenperature we get

$$R_2 = \frac{Tr}{Tx} = \frac{273+27}{273+33} = \frac{300\,^\circ K}{306\,^\circ K} = 0.98039$$

Again from table A-16 we get the numerical value of m=0.86, n=0.55 and p=0 for natural aspirated gas fueled spark ignition engine.

Now the ratio of indicated power (k) is found out using the formula

$$\begin{aligned} k &= (R_1)^m (R_2)^n\ (R_3)^p \\ &= (R_1)^m (R_3)^n \quad \text{since } p=0 \\ &= (0.9869)^{0.86} (0.98039)^{0.55} \\ &= 0.9785 \end{aligned}$$

The power adjustment factor (α) is now calculated from table A-15. For the value of k = 0.9785 and mechanical efficiency η_m =0.85, we get α= 0.976, which is the power adjustment factor.

The fuel consumption adjustment factor (β) is similarly found out by interploting the value of k= 0.9785 and η_m=0.85 in table A-14., from where we get the value of β = 1.00315.

Brake horse power is now adjusted by using the formula-

$P_x = \alpha P_r$ where P_r - Brake horse power at standard condition
P_x - Brake horse power at laboratory condition

Hence Standard Power $P_r = P_x/\alpha = (16.8 \times 0.736)/0.976 = 12.678$ Kw

A-17.6 CALCULATION OF BRAKE SPECIFIC FUEL CONSUMPTION.

Brake Specific Fuel Consumption at standard condition is now found out from the relation, $b_x = \beta . b_r$, where b_x is Specific Fuel Consumption at room condition and b_r is Specific Fuel Consumption at standard condition.

Now using sample data , $b_r = \frac{b_x}{\beta} = \frac{1.038\text{x}60\text{x}60}{12678 \text{ x}1.00315}$

b_r = 293.82 gm/Kw-hr

Hence Brake Specific Fuel Consumption at standard condition is b_r =293.82 gm/Kw-hr.

A-17.7 CALCULATION OF BRAKE SPECIFIC ENERGY CONSUMPTION.

It is calculated on the basis of Lower Calorific Heating Value = 45 Mj/Kg CNG , the value used by Liquid Fuel Management Group Ltd of Newzealand.

Brake Specific Energy Consumption (BSEC) = BSFCxFuel Value (kg/Kw-hr x Mj/Kg).

$$\text{Thus, BSEC} = \frac{293.82\text{x}45 \text{ (kg/Kw-hrxMj/kg)}}{1000}$$

= 13.22 Mj/Kw-hr

A-17.8 CALCULATION OF BRAKE THERMAL EFFICIENCY

$$\text{Brake Thermal Efficiency} = \frac{\text{Brake Thermal Output}}{\text{Thermal Imput}}$$

$$\text{B.Th.Eff.} = \frac{12.678\text{x}10^3\text{(Kwxjoules/sec-Kw)}}{1.0345\text{x}10^{-3}\text{x}45\text{x}10^6\text{(kg/sec x joules/kg)}}$$

= 0.2723 = 27.23%

A-17.9 CALCULATION OF EQUIVALENCE RATIO.

$$\text{Equivalence ratio} = \frac{\text{Theoritical value of air fuel ratio}}{\text{Actual value of air fuel ratio}}$$

Let us now find out the theoritical value of air fuel ratio from the combustion equation of natural gas.

The composition of natural gas is as follows as mentioned in appendix table A-3 .

Methane -	94.35022%	W/w
Ethane -	2.56423 %	"
Propane -	1.09133%	"
Butane & higher-higher	1.08266%	"
Nitrogen -	0.2545388%	"
Carbondioxide-	0.67257%	"

A 17.9.1 **Combustion of Methane (CH_4):**

$$CH_4 + 2O_2 = CO_2 + 2H_2O$$

Therefore, $(12+4)CH_4 + 2(32)O_2 = (12+32)\ CO_2 + 2(16+2)\ H_2$.

Hence , (12+4) Kg of Methane combines with 2x32 Kg of Oxygen for complete combustion to give (12+32) Kg of Carbon dioxide and 2x18 Kg of vapor.

Hence 16 Kg of CH_4 needs 64 Kg of O_2 for its complete combustion (theoritically).

Again from the gravimetric analysis, air has 77% N_2 by weight and 23% O_2 by weight. So 1 Kg of Oxygen is present in 100/23 Kg of air.

Therefore 16 Kg of Methane needs 64 Kg of Oxygen which is available in $64\text{x} \frac{100}{23}$ Kg of air.

Hence one Kg of methane require $\frac{64}{16} \text{ x } \frac{100}{23} = 17.39$ Kg of air.

Since the compressed natural gas contains 94.35022 % of CH_4. Therefore 1 Kg of compressed natural gas needs 17.39x0.9435022 Kg of air = 16.409078 Kg of air for complete combustion of its CH_4.

A 17.9.2 **Combustion of Ehtane (C_2H_6):**

$$2C_2H_6 + 7O_2 = 4\ CO_2 + 6\ H_2O$$

2(24+6)Kg of C_2H_6 + 7(32)Kg of O_2 gives 4(12+32) Kg of CO_2 + 2 (18) Kg of H_2O.

Hence 1 Kg of $C_2 H_6$ needs $\frac{100}{23} \times \frac{224}{60}$ Kg of air.

Therefore 1 Kg of natural gas needs $\frac{100}{23} \times \frac{224}{60} \times \frac{2.56423}{100}$

0.4162228 Kg of air for complete combustion of its C_2H_6

A 17.9.3 **Combustion of Propane** ($C_3 H_8$):

$$C_3H_8 + 5\ O_2 = 3CO_2 + 4H_2O$$

Hence 1 Kg of CNG having 0.42% of propane require

$$\frac{100}{23} \times \frac{1.09133}{100} \times \frac{(5x32\ )}{(36\ +8\ )} = 0.1725422 \text{ Kg of air.}$$

A 17.9.4 **Combustion of n-Butane** (C_4H_{10}):

$$2\ C_4\ H_{10} + 13\ O_2 = 8\ CO_2 + 10\ H_2O$$

Hence 1 Kg of CNG having 1.08266% of n-Butane require

$$\frac{100}{23} \times \frac{1.08266 \times 32 \times 13}{100 \times 2\ (24\ +48\ )} = 0.1688105 \text{ Kg of air.}$$

Thus 1 Kg of CNG needs 16.409078 + 0.4162228 + 0.1725422 + 0.1688105 =17.167356 =17.17 Kg of air.

Since the composition on natural gas may vary and more over there may be other form of n-Octane (C_8H_{18}),so the theoritical air requirement may also vary.

For our calculation we have taken the theoretical air fuel ratio 17.17 . Thus the Equivalence Ratio (E.R.) = 17.17/16.409695 =1.046 where 16.409695 is the actual air fuel ratio as found in the sample calculation and its data.

17.10 CALCULATION FOR REDUCTION OF ENGINE COMPRESSION RATIO.

Compression ratio is changed by increasing clearence volume by removing metal from piston head or piston cavity .If V_c is the clearence volume of the engine initially. V_s is the swept volume.

Then the compression ratio $C_r = \dfrac{V_c + V_s}{V_c}$

Now, let V_r be the volume of metal to be removed for reducing the compression ratio to a desired value, then final compression ratio becomes

$$(V_{rf}) = \frac{V_c + V_s + V_r}{V_c + V_r}$$

So choosing any compression ratio we may calculate the volume (V_r) to be removed from the piston.

APPENDIX-17. SAMPLE DATA AT THE IGNITION TIMIMNG OF 15°BTDC

Sl. No.	Item No.	Initial reading	Obs.1	Obs.2	Obs.3	Obs.4	Obs.5	Obs.6	Obs.7
1.	N	0	1500	1500	1500	1500	1500	1500	1500
2.	L	0	11.20	10.10	9.10	7.80	7.00	6.2	5.4
3.	P_1	0	51.82	41.43	34.54	20.73	10.36	6.91	6.91
4.	P_2	0	93.66	109.22	147.32	182.88	210.82	233.68	259.08
5.	P_3	0-0	84.14	56.20	44.12	37.15	30.29	27.31	24.76
6.	P_4	0-0	66.709	48.205	39.816	34.782	29.212	26.244	22.660
7.	SIgT.		15°BTDC	15°BTDC	15°BTDC	15 °BTDC	15°BTDC	15°BTDC	15°BTDC
8.	W_1	25.23	25.230	24.960	24.849	24.769	24.632	24.489	24.210
9.	W_2	-	24.880	24.750	24.640	24.590	24.465	24.335	24.040
10.	t	-	337.25	257.43	290.03	300.00	300.00	3000.00	356.78
11.	t_1	33	35	35	35	35	35	35	35
12.	t_2	33	60	66	68	68	70	71	72
13.	t_3	33	42	56	65	71	71	71	70
14.	t_4	33	563	584	576	556	549	537	531
15.	t_5	33	600	601	596	567	552	540	533
16.	t_6	33	101	105	101	96	92	89	86
17.	t_7	29.5	43	44	44.5	44	44	44	44
18.	%CO		1.9	0.6	0.0	0.0	0.0	0.0	0.0
19.	HC(ppm)		160	120	120	40	000	20	101-1
20.	$\%CO_2$		8	8	9	9	8	8	8
21.	$\%O_2$		11.5	11.6	11.9	12.7	13.1	13.3	13.7
22.	t_a		33	33	33	33	33.1	33.2	33.2
23.	t_w		28.5	28.5	28.5	28.5	28.5	28.5	28.5
24.	$\emptyset_x$		72.13	72.13	72.13	72.13	71.14	70.14	70.14%

APPENDIX A-18

A-18.1 REDUCTION OF ENGINE COMPRESSION RATIO:

The engine compression ratio was reduced in the following steps :

a. The diesel engine was properly placed on a support for dismantling of its cylinder head.

b. The tappet cover from the cylinder head and crank case were removed.

c. Cylinder head bolts were loosened and both the cylinder head were removed from the engine block.

d. Connecting rod's bolts were removed for opening the piston and connecting rod from the crank shaft.

e. After removing the connecting rod's hubs, the connecting rods with piston assemblies were pushed upward to bring out the pistons and connecting rods through the upper side of the cylinders.

f. The pistons and connecting rods assemblies were placed on a table.

g. Piston pin and its lock were removed and piston was seperated from its connecting rod. Piston rings were also removed from the piston.

h. To reduce compression ratio, either piston head is cut off or piston groove (precombustion chamber) is enlarged to increase clearence volume for reducing the compression ratio to desired value.

i. In our case the compression ratio was reduced from 17:1 to 12:1. This was done by increasing the clearence volume to the predtermined or calculated value. The calculation for volume to be enlarged has been shown in the sample calculation.

In our laboratory one single cylinder diesel engine (Duetz 210 D, 6.7 Kw) and a double cylinder diesel engine (Duetz 2FL 912) were modified to hundred percent CNG fueled operation. In the single cylinder diesel engine, the precombustion chamber's diameter (piston cavity diameter) was enlarged in the machine shop by removing metal from the piston on a lathe machine to the calculated diameter for reducing the compression ratio from 17:1 to 12;1. Here the piston head or length of the piston were not changed.

But in the two cylinder diesel engine, an alternative technique was adopted to reduce the compression ratio. In this case the piston head's metal was removed by facing it on a lathe machine to enlarge the clearance volume to the calculated value. The dome inside the precombustion chamber was also removed. The volume of the dome in the piston precombustion chamber was measured by filling the precombustion chamber with water before removing the dome and after removing the dome as measured by micro shringe.

After necessary fitting of the piston and connecting rod to the respective position of the crank shaft, the cylinder head and tapet cover were also fitted to the engine block as they were before.

A-18.2 MODIFICATION AND FITTING OF SPARK PLUGS ON THE CYLINDER HEAD:

The injectors of the diesel engine were removed from the cylinder heads. Two nos. spark plugs of the required specification (as mentioned in table A-2) are selected for fitting them in the place of injectors. If the hole of injector on the cylinder head is smaller than spark plug diameter, then the hole is enlarged in the same angle by drilling to the required diameter and keeping allowence for thread cutting and fitting spark plug to the hole on the cylinder head.

A-18.3 INSTALLATION OF ELECTRONIC IGNITION SYSTEM :

The diesel engine having compression ignition system was modified to spark ignition engine by installing an electronic ignition system with the engine. For doing so, two nos. electronic control units, two nos. induction coils and two nos. spark plugs were installed on the unit along with two other proximity switches as shown in the diagram no. 1.3. The necessary electrical wiring was done as per wiring diagram of electronic ignition system as shown in the schematic diagram no. 1.3.

A-18.4 INSTALLATION OF CNG FUEL SYSTEM BY REPLACING DIESEL FUEL INJECTION SYSTEM

The diesel fuel injection system of the original engine was replaced by CNG fuel system by the installation of pressure regulator, mixer(carburator), CNG cylinder etc as shown Fig. 1.1.1 and Fig. 1.1.2.