INTERNAL CIRCULATION

F.R.I. 47

VERSLAG NR. 36 REPORT NO. 36

VAN 1978 OF 1978

FRI 36/1978

UI/E/5/5

BRANDSTOFNAVORSINGSINSTITUUT
VAN SUID-AFRIKA

FUEL RESEARCH INSTITUTE
OF SOUTH AFRICA

ONDERWERP:
SUBJECT: A REPORT ON INVESTIGATIONS INTO THE SINGLE-FUEL, DUAL-FUEL AND TRIPLE-FUEL OPERATION OF A DIESEL ENGINE

AFDELING:
DIVISION: ENGINEERING

NAAM VAN AMPTENAAR:
NAME OF OFFICER: F D HEIM; D CLARK

Coetzee 7751—L48

AUTHOR	:	F D HEIM, D CLARK
LEADER OF PROJECT	:	F D HEIM
TITLE	:	A REPORT ON INVESTIGATIONS INTO THE SINGLE-FUEL, DUAL-FUEL AND TRIPLE-FUEL OPERATION OF A DIESEL ENGINE
CO-WORKER	:	A JOUBERT
ENQUIRIES TO	:	F D HEIM
DIVISION	:	ENGINEERING

FUEL RESEARCH INSTITUTE OF SOUTH AFRICA

REPORT NO. 36 OF 1978

INTERNAL

A REPORT ON INVESTIGATIONS INTO THE SINGLE-FUEL, DUAL-FUEL AND TRIPLE-FUEL OPERATION OF A DIESEL ENGINE

1. INTRODUCTION

In the past, when operating a diesel engine, the primary goal was often to achieve maximum output. Now, however, low fuel consumption and clean exhaust gases have become more important considerations. In fact, the two are connected, for good combustion means low smoke emission and high fuel efficiency.

Traditionally, the diesel engine has a bad reputation for smoke emission, but a good one for fuel efficiency. Hence, the present investigation, which was aimed at improvement in fuel efficiency as a means of reducing smoke emission. It is, however, impossible to beat nature, and savings can only be made where previously there was waste. To move a given mass at a given speed, requires a set amount of energy, and provided that an engine is in good condition and properly tuned, there should be no margin for fuel saving. If fuel consumption is reduced in such circumstances, the vehicle will give a lower standard of performance. This is illustrated by the following data which have been taken from "Bosch" Tabellenbuch. The main figures refer to an Otto engine and those in brackets to a diesel engine.

	Otto	Diesel
Input via fuel	100%	100%
Loss via exhaust gases	36%	29%
Loss via cooling water	33%	32%
Loss by radiation	7%	7%
Total loss in engine	76%	68%
Useable energy at flywheel	24%	32%
Deductions for dynamo-fan, gearbox, transmission and other moving parts, also roll resistance and wind resistance	20%	20%
Thus Useful energy for load movement	4%	12%

The above figures take no account of smoke emission. When smoke is formed, fuel leaves the engine without doing work, or in other words, is wasted. In order to understand why smoke is produced, it is necessary to examine the diesel principle. In the diesel engine, the dieseline is sprayed into compressed air at about 600°C. Further, in order to achieve good combustion, this injection must be correctly timed within the cycle of the engine. (If injection occurs too late bluish white exhaust gas is formed with loss of power; or if too early, black smoke is emitted and the engine runs rough and very loud). Dieseline does not mix easily with air, and does not evaporate as easily as petrol. Hence, the sprayed fuel consists mainly of small droplets, not vapour. Furthermore, the spray is not distributed evenly over the whole of the combustion space, but is sprayed directionally. This creates a tendency to fuel rich and fuel lean zones. Fuel lean zones in particular occur outside the fuel rays; at the end the rays fuel can condensate against cold cylinder walls. In short, mixing is not always good, and neither all of the air nor the fuel is properly used. Many ways of solving this problem have been tried, with varied degrees of success.

The Fuel Research Institute choose to work on dual-fuel operation. This is not a new technique, and was originally used to run stationary diesel engines using gas (mainly towns-gas) as the secondary fuel. Again, in Germany in 1940, producer gas was used as the secondary fuel, and gas generators were mounted on the vehicles. The aim here was the saving of dieseline. Experience, however, showed that it was better to convert the diesel engine to an Otto engine, and to feed only producer gas. The main reason for this was that an engine can only inhale a certain amount of air; and this air had to be divided between combustion and gasification requirements. This caused the firebed of the gas generator to operate at too low a temperature, thus creating a gas of high tar content - which, in turn, created problems in the gas filter, and on occasions in the engine.

3/...

In the early work at the Institute, both liquefied petroleum gas (L.P.G.) and methane were used as secondary fuels, but proved inconvenient due to the necessity to refill the gas cylinders, and particularly in the case of methane, due to the low energy capacity in the bottles. Engine output and exhaust gas cleanliness were, however, good for both gases.

In order to overcome the above difficulties, petrol was then used as the secondary fuel, and pre-ignition became a problem. Alcohols gave trouble due to misfiring. This report describes problems which were found in the use of a diesel engine operated on a dual- and triple-fuel basis with a variety of secondary and tertiary fuels, and discusses the advantages and disadvantages of such systems.

2. METHODS OF TEST AND OBSERVATIONS

The engine used in the tests was a 3,6 litre diesel of specification:-

Type:	Ford Thames Trader, 4 cylinder
Bore:	100 mm
Stroke:	115 mm
Capacity:	3611 cm^3
Brake Horsepower:	64 or 47 KW (at sea level and 2500 r.p.m.)
Compression ratio:	16/1 - direct injection

The following modes of test were used:-

1. Normal diesel operation.
2. Derated diesel operation with petrol (lean).
3. Derated diesel operation with petrol (rich).
4. Derated diesel operation with ethanol.
5. Derated diesel operation with methanol.
6. Derated diesel operation with petrol and ethanol.
7. Derated diesel operation with petrol and methanol.
8. Derated diesel operation with a blend of petrol and ethanol.
9. Derated diesel operation with a blend of benzene and methanol.

The following remarks are made:

a) Injector Pump Timing

It was not possible to observe factory instructions in full. This was due to the altitude of Pretoria (1500 m above sea level). Hence the timing used was brought forward to 22° BTDC. This setting was used for all the tests described in the present report.

b) Dieseline Quantity

Again, factory instructions were not able to be observed, as the altitude of Pretoria for normal diesel operation, required a reduction from 12,3 cm^3 to 11,5 cm^3 for 200 strokes/min (i.e. equivalent to a control rod travel of 11,4 mm).

As a result of the necessary smoke free operation, various other changes in the injector pump setting had to be made. When the engine was operated with petrol as the secondary fuel, the setting had to be reduced to 7,5 mm. With methanol or ethanol as the secondary fuel, the setting was able to be increased to 8,2 mm. Here, the increased dieseline supply helped to compensate for the lower calorific value of the alcohols, but increased the fuel consumption.

c) Manner of Introduction of Secondary Fuel

The secondary fuel was injected into the inlet air by means of an electric pump, and the injection rate was measured by a calibrated jet. In order to control the secondary fuel supply more precisely, an adjustable conic needle was included in the fuel supply system. Adjustments to secondary fuel supply rate were aimed at optimum performance with respect to output, smoke emission, fuel consumption and engine noise.

d) Effect of Secondary Fuel Properties

When petrol or alcohol was used as the secondary fuel, the quantity able to be fed to the engine was decided by the properties of the fuel.

The compression ratio of a diesel engine has to be at least 15/1, and in order to guarantee a good standard of ignition, is usually higher. For the normal petrol engine, the compression ratio is 7/1 (9/1 for super grade). If it is higher, then pre-ignition occurs. In order to avoid this, the air-to- fuel ratio must be outside the lower ignition boundary (i.e. the fuel mass must be approximately 1,5% of the air mass). If the percentage is higher, pre-ignition occurs; or if lower, a lower output is obtained. In order to facilitate this, a new petrol feed system was developed, and gave clean exhaust gas with better fuel efficiency and minimum loss of output. (The increase of fuel efficiency was about 15 - 20% at full load, but varied over the speed range. The brake efficiency improved by 5%, from 25 to 30%, and compared to a normal diesel engine, the exhaust temperature fell. In effect, dieseline delivery to the engine was restricted to a level where no smoke occurred (Bosch Smoke No. 4). What would have been a large power loss, was avoided by the introduction of the right amount of petrol, which was maintained just below the pre-ignition level).

In the cases of methanol and ethanol, the limiting factor was misfiring caused by the high latent heat of the fuels, and this determined the maximum alcohol supply rate. In fact, the fuel setting had to be made while operating at high revolutions in order to avoid misfiring when later operating at such speeds. This, however, gave too-lean-air-to-fuel-mixtures, over the whole speed range downwards, (at low speeds nearly the double amount of alcohol could be applied). The explanation for this seemingly abnormal behaviour would appear to be that the time permitted for the alcohol to evaporate is too short at high revolutions, hence the greater the secondary fuel feed rate, the lower the r.p.m. level at which misfiring occurs.

e) Triple Fuel Operation

The properties of petrol and alcohols are in many ways contradictory, hence some uncertainty was felt as to the effect of introducing the two simultaneously. In effect, would the octane value of the mixture be increased over that of petrol and would the latent heat be decreased? If this were correct, then both secondary fuel injection rates and output could be increased. Essentially, this was the reason for the investigations into triple-fuel operation. A number of problems, however, emerged. It was found that petrol and ethanol would mix adequately, but petrol and methanol would not. Hence petrol and methanol were introduced separately and simultaneously via separate carburettors into the inlet air stream. It is appreciated that such a complex system is of little practical value, but it did enable engine performance to be evaluated, and precise fuel injection rates to be established.

When running the triple-fuel tests, the following start-up sequence was used:-

1. Normal engine start-up using dieseline in excess.
2. Engine running derated.
3. Secondary fuel added.
4. Tertiary fuel added.

Using this sequence, it was then possible to increase the output, thus answering the questions posed earlier. Some difficulties still occurred, however, when attempting to restart the engine at the determined optimum setting. Pre-ignition occurred when petrol was introduced first, and misfiring occurred when alcohol was introduced first. Unfortunately, the test-bed layout was not such as to permit the simultaneous introduction of the two secondary fuels.

f) BLENDING

Due to the desire to further investigate blends, and the problems encountered in blending methanol and petrol, a mixture of benzene and methanol, which gave no mixing problems, and facilitated triple-fuel operation with a single carburettor, was also used.

g) TEST RESTRICTIONS

In Table 1 (see Appendix), the 2500 r.p.m. highest revolutions setting was not used, and lower levels (2300 - 2400 r.p.m.), were substituted. This was because in some tests carried out at 2500 r.p.m., the governor operated to close the dieseline supply due to a high vacuum. This was due to differences in the air intake manifold vacuum in the different supplementary fuel systems. Operation at the lower r.p.m. level avoided this.

h) IMPURITIES IN METHANOL

It was evident that the methanol used in the investigation was not pure. No effect was noted early in tests, but later a deposit was found on the manifold. It looked like an incrustation and was not soluble in either petrol or water.

i) TIME OF CONSUMPTION OF STANDARD QUANTITY OF SECONDARY FUEL

The time required to consume the standard quantity of secondary fuel (200 cm^3), varied with the particular secondary fuel, despite identical jet and needle positions. Further, the authors are unsure whether or not this was due to the deposit described in h). When the jet was checked, it was always found to be clean, but the needle could not be checked as dismantling would dislodge any small quantity of deposit. Hence when considering the results, too much stress should not be placed on jet sizes and needle positions, instead, fuel rates, which constitute accurate measured values, should be considered.

3. RESULTS

Table 1 shows engine settings together with the various fuel rates for all the quoted tests. Table 2 gives a comparison of outputs, fuel consumptions, brake efficiencies and smoke emission data for the tests. Table 3 compares exhaust tempertures. Both Table 2 and Table 3 give calculated losses and gains for the multiple fuel tests, using normal dieseline operation (test No. 17) as the basis for assessment.

Figure 1 shows the layout of the control system used for dual-fuel operation of the diesel engine, and Figure 2 shows the system used for triple-fuel operation. Figure 3 comprises two photographs of the triple-fuel system. Also appended, are examples of logsheets - test No. 14 DPM (petrol/methanol) and test No. 35 DB (Petrol/ ethanol blend), together with graphs for all the tests carried out.

4. DISCUSSION

The best performance data are shown by those tests which were operated on a triple-fuel basis. Further, generally ethanol gave better results than methanol, due to the higher calorific value of the former. Total fuel consumption was, in all cases involving multiple fuels, better than with dieseline alone. The lowest consumption of all, is shown by the dual-fuel test using petrol as the secondary fuel, but in this test the normal dieseline output could not be reached. Smoke emission data for all the multiple fuel tests are better than for dieseline alone, in fact the most smoke was only just visible. It is known from earlier work that the carbon-monoxide level in the exhaust gases of a diesel engine is very low. Addition of secondary fuel, (either petrol or alcohol), increases this by only a fraction of 1 per cent.

Due to the lack of a suitable instrument, the NO_x content of the exhaust gases could not be measured. However, it is considered reasonable to assume that the level of NO_x in the exhaust gases would be lower for the multiple fuels used, than for dieseline alone. NO_x formation is promoted by high combustion temperatures and long combustion times - both are reduced with multiple fuel operation.

The particular diesel engine used in the investigation, is fitted with direct injection. This type has the advantage of giving 8 to 10% better output, with 8 to 10% lower fuel consumption than the precombustion chamber type of diesel engine. The operation of this engine has been improved by multiple fuel operation in the following ways.

(i) About the same output.
(ii) Less smoke emission.
(iii) Lower fuel consumption.
(iv) Higher efficiency.
(v) Lower exhaust gas temperature.

These gains are thought to emanate from a number of individual small improvements which collectively are important.

(i) Finer adjustment of fuel flow rates.
(ii) Combustion was preconditioned by the presence of an air-fuel-mixture before injection. This appeared to give greater turbulence, and hence better mixing, of fuel and air in the combustion chamber before ignition.
(iii) The lower smoke emission and lower exhaust gas temperature shows that less potential and sensible heat was lost from the engine.
(iv) The latent heat of evaporation of the petrol or alcohol has the effect of cooling the inlet air; hence more air is admitted to the engine.
(v) Compared with hydrocarbon fuels, the use of alcohol involves the feeding of additional combined oxygen to the engine. This is considered to be an aid to combustion.
(vi) When petrol and alcohol are introduced to the engine simultaneously, the effective octane value of the mixture is higher than that of petrol, and the effective latent heat of evaporation of the mixture is lower than that of alcohol. These two effects act as a compensation for the lower calorific value of the alcohol.

5. CONCLUSIONS

The present investigation into the dual- and triple-fuel operation of a diesel engine have shown improvement in operation mainly due to cleaner exhaust gases and lower fuel consumption.

Certain inherent disadvantages of the multiple fuel system, if applied on a general basis, should not be overlocked. These are concerned with installation of an additional fuel system comprising tank, fuel feed pump, carburettor, and modifications to the existing fuel feed mechanism. Further, the current price of both of the alcohols used is higher than that of petrol or dieseline. From the economic angle, at present, therefore, the system would be uneconomic. Strategic necessity, the probable future increase in crude oil prices and the possibility of producing both of the alcohols from indigenous resources, could possibly reverse this disadvantage.

F O HEIM
SENIOR TECHNICIAN

D CLARK
CHIEF RESEARCH OFFICER

PRETORIA
1978/09/14
FOH/DC/ug

APPENDIX

TABLE 1

FUEL RATES

Test No. Setting	RPM	Dieseline		Petrol		Ethanol		Methanol		Blend		Benzole		Total Fuel
		Kg/min	%	Kg/min	%	Kg/min	%	Kg/min	%	Kg/min	%	Kg/min	%	Kg/min
17 D 11,4 mm	1112 2327	0,117 0,225	100 100	- -	- -	- -	- -	- -	- -	- -	- -	- -	- -	0,117 0,225
19 DP 7,5 -40 - 4,8	1091 2300	0,061 0,140	68,5 81,4	0,028 0,032	31,5 18,6	- -	- -	- -	- -	- -	- -	- -	- -	0,090 0,172
25 DP 8,2 - 40 - 5,0	1102 2310	0,073 0,152	75,3 84,4	0,024 0,028	24,7 15,6	- -	- -	- -	- -	- -	- -	- -	- -	0,098 0,180
31 DE 8,2 - 100 - 5,2	1103 2301	0,071 0,153	66,4 81,0	- -	- -	0,036 0,036	33,6 19,9	- -	- -	- -	- -	- -	- -	0,107 0,189
21 DM 8,2 - 80 - 2,7	1104 2330	0,071 0,155	59,7 73,8	- -	- -	- -	- -	0,048 0,055	40,3 26,2	- -	- -	- -	- -	0,119 0,210
24 DPE 8,2 - 80 - 2,6 Petrol 40 - 4,6	1099 2322	0,072 0,152	61,0 73,8	0,020 0,023	16,9 11,2	0,026 0,031	22,0 10,7	- -	- -	- -	- -	- -	- -	0,118 0,206
14 DPM 8,2 - 80 - 2,5 Petrol 35 - 4,4	1073 2337	0,069 0,154	62,2 76,2	0,017 0,021	15,3 10,4	- -	- -	0,025 0,027	22,5 14,3	- -	- -	- -	- -	0,111 0,202
35 D-Blend 50% Petrol 50% Ethanol 8,0 - 100 - 5,5	1081 2330	0,063 0,150	63,0 75,4	0,019 0,024	17,6 11,9	0,021 0,025	19,4 12,8	- -	- -	0,040 0,049	37,0 24,6	- -	- -	0,100 0,199
33 D - Blend 40% Benzene 60% Methanol 8,2 - 100 - 5,2	1107 2353	0,073 0,157	65,8 77,0	- -	- -	- -	- -	0,022 0,027	19,8 13,3	0,033 0,047	34,2 23,0	0,016 0,020	14,4 9,7	0,111 0,204

TABLE 2

COMPARISON OF ENGINE OPERATION DATA

Test No.		RPM	Output %	Output KW	Consumption %	Consumption gr/KWh	Smoke %	Smoke No.	Brake Efficiency	Output % *	Consumption % * SAVING	Smoke % *	Brake Eff. % *
17 D	normal dieseline operated engine	1112 2327	100 100	21,9 40,0	100 100	321 330	100 100	6,6 7,6	24,6 23,4	100	-	100	real values 24,6 23,8
19 DP lean	derated engine + petrol	1091 2308	94,1 93,5	20,6 37,4	81,3 81,4	261 275	39,8 33,2	3,5 2,9	30,1 28,5	- 5,9 - 6,5	18,7 18,6	60,2 61,8	+ 5,5 + 5,1
25 DP rich	derated engine + petrol	1102 2318	100,5 98,0	22,0 39,2	82,9 78,7	266 276	56,8 53,9	5,0 4,1	29,5 28,5	+ 0,5 - 2,0	17,1 21,3	43,2 46,1	+ 4,9 + 5,1
31 DE	derated engine + ethanol	1103 2301	105,9 99,0	23,2 39,6	86,6 84,9	278 287	31,8 43,4	2,8 3,3	32,4 29,6	+ 5,9 - 1,0	13,4 15,1	63,2 56,6	+ 7,8 + 6,2
21 DM	derated engine + methanol	1104 2330	103,7 99,5	22,7 39,8	98,1 93,2	315 316	36,4 44,7	3,2 3,4	31,7 23,9	+ 3,7 - 0,5	1,9 6,8	63,6 55,3	+ 7,1 + 5,5
24 DPE	derated engine + petrol + ethanol	1099 2322	112,8 105,0	24,7 42,0	89,4 84,9	287 294	52,3 59,2	4,6 4,5	29,9 28,3	+ 12,8 + 5,0	10,6 15,1	47,7 40,8	+ 5,3 + 4,9
14 DPM	derated engine + petrol + methanol	1073 2337	100,9 104,0	22,1 41,6	93,5 88,8	300 292	42,0 52,6	3,7 4,0	29,7 29,0	+ 0,9 + 4,0	6,5 11,2	58,0 47,4	+ 5,1 + 5,6
35 D Blend	derated engine + blend of 50% petrol + 50% ethanol	[illegible] 2330	105,9 103,0	23,2 41,2	85,9 82,5	279 289	45,5 44,7	4,0 3,4	30,2 28,5	+ 5,9 + 3,0	13,1 17,5	54,5 55,3	+ 5,6 + 5,1
33 D Blend	derated engine + blend of 40% benzene + 60% methanol	1107 [illegible]	103,2 101,3	22,6 40,5	91,6 87,0	294 302	37,5 39,5	3,3 3,2	30,4 28,3	+ 3,2 + 1,3	8,4 13,0	62,5 60,5	+ 5,8 + 4,9

* Normal dieseline operation (test No. 17 D used as standard).

TABLE 3

COMPARISON OF EXHAUST TEMPERATURES

(with different fuels and settings using test 17 D as standard)

17 D 11,4 mm - -			19 DP (lean) 7,5 mm - 40 - 4,8			25 DP (rich) 8,2 mm - 40 - 5,0			31 DE 8,2 mm - 100 - 5,2			21 DM 8,2 mm - 80 - 2,7			24 DPE - 40 P - 4,4 8,2 mm - 80 E - 2,6		
Output KW	RPM	Exhaust °C	Output KW	RPM	Exhaust °C	Output KW	RPM	Exhaust °C	Output KW	RPM	Exhaust °C	Output KW	RPM	Exhaust °C	Output KW	RPM	Exhaust °C
21,9	1112	495	20,6	1091	420	22,0	1102	470	23,2	1103	460	22,7	1104	450	24,7	1099	490
25,7	1311	505	24,5	1303	440	25,5	1293	470	26,4	1295	460	26,6	1304	460	28,7	1311	500
31,4	1593	550	29,4	1570	470	30,3	1561	495	32,6	1572	490	31,3	1558	495	33,6	1569	540
35,6	1874	620	34,1	1864	515	35,2	1844	550	35,7	1847	540	35,6	1834	540	37,8	1837	590
38,1	2114	670	36,2	2095	540	37,2	2065	580	38,4	2062	570	37,6	2065	570	40,8	2082	620
40,0	2327	705	37,4	2308	550	39,2	2318	590	39,6	2301	580	39,8	2330	585	42,0	2322	640
42,6	2628	740	38,3	2536	560	40,0	2564	600	39,6	2570	600	40,5	2566	610	42,6	2538	640

14 DPM - 35 P - 4,4 8,2 mm - 80 M - 2,5			35 D - Blend 50% Petrol 8,0 mm - 80 - 5,5 50% Ethanol			33 D - Blend 40% Benzole 8,2 mm - 100 - 5,2 60% Methanol		
Output KW	RPM	Exhaust °C	Output KW	RPM	Exhaust °C	Output KW	RPM	Exhaust °C
22,1	1073	450	23,2	1081	480	22,6	1107	470
26,7	1296	480	28,0	1308	490	26,3	1303	470
32,0	1582	510	32,9	1559	520	31,4	1576	500
36,9	1865	560	37,2	1844	570	36,0	1858	555
39,1	2079	590	38,9	2049	580	38,4	2085	580
41,6	2337	610	41,2	2330	615	40,5	2353	610
42,3	2578	620	42,7	2587	645	40,5	2548	610

BENCH TEST RESULTS.

TEST NO. 35 DB DATE 8.5.78

Engine Ford Diesel Vol. Vo. 3611 cm^3

Fuel Shell dieseline Gross C.V. 45640 KJ/kg

8,0 mm injector pump stop - 100 jet for blend
5,5 needle position - one carburettor fitted

+ Blend: 50% Mobil Prem. petrol + 50% Ethanol } = 37670 KJ/kg

Actual Speed rpm.	Air Rate m^3/min.	Air Rate kg/min.	Fuel Rate kg/min.		A/F Ratio		Input kW		Torque N.m.	Torque p	Output kW	Efficiency Brake %	Efficiency Volume %	Consumption gr/KWh	Total Fuel Rate kg/min.	Total Air: Fuel Ratio	Total Input kW	Bosch Smoke No.
n	Va	1,03 Ma	0,834 M D	0,759 M B	α D	α B	Q D	Q B	T	p	L	η	τ	υ				
1081	1,75	1,80	0,068	0,040	24,6	45,2	52,0	21,5	205,5	21,5	23,2	30,2	89,6	279	0,108	16,7	77,0	4,0
1308	2,07	2,13	,085	,044	25,0	48,9	64,8	27,4	204,6	21,4	28,0	30,4	87,6	276	,129	16,6	92,2	3,8
1559	2,45	2,52	,104	,046	24,2	54,6	79,4	29,0	201,7	21,1	32,9	30,4	87,0	275	,151	16,8	108,4	4,0
1844	2,83	2,91	,126	,047	23,1	61,9	95,8	29,6	193,1	20,2	37,2	29,7	85,0	279	,173	16,8	125,4	4,0
2049	3,10	3,19	,136	,045	23,5	70,5	103,5	28,4	181,6	19,0	38,9	29,5	83,8	280	,181	17,6	132,0	3,0
2330	3,44	3,54	,150	,049	23,6	72,9	114,1	30,5	196,2	17,7	41,2	28,5	81,8	289	,199	17,8	144,6	3,4
2587	3,75	3,86	,162	,051	23,8	74,9	123,4	32,4	157,7	16,5	42,7	27,4	80,3	301	,214	18,1	155,8	3,0

Air rate Va and Ma in first instance from $V = k\sqrt{\Delta p}$ (when $\Delta p >$ 2 mm H_2O).
This is correct for air density $p = 1 kg/m^3$. Density correction
$Va = V: \sqrt{p}$ $Ma = V\sqrt{p}$ $Q = M.CV$, $L = p.n.10^{-3}$, $= Ma/M$,
$= 100.L/Q$, $= 200.Va/n.Vo$, $n = \frac{N}{t}$

d = 75,4 mm	k = 0,88
67,9	0,68
60,7	0,51

BENCH TEST RECORD TEST NO. 35 DB DATE 8.5.78

Engine Ford Diesel Fuel Shell dieseline + Blend of 50% Mobil Prem. petrol + 50% Ethanol

Auxiliaries Used: Airfilter/Air Tank/ Fan/Generator Barometer 656

Ignition or Injection Timing 22 ^{o}BTDC (Static), Humidity 42

Compression Ratio 16:1

Other Remarks on Engine Adjustments Injector pump stop 8,0 mm + 100 jet for blend - 5,5 needle position

Air Orifice Dia 75,4 mm Measuring Vessel 200 cm^3 (grams)

Speed rpm	Torque Nm	p	Power (Meter) kw	Consumption Time S	Revs. No.	Temperatures Amb. ^oC	Exh. ^oC	Water	Diesel	Diff. mmH_2O	Smoke Bosch Nr.	Consumption Time of Blend
n^1	T	p	L^1	t	N	ta	te	tw	^oC	ΔP		200 cm^3
1081	205,5	21,5	23,2	146,5	2640		480	32		3,9	4,0	228,2 sec.
1308	204,6	21,4	28,0	117,4	2590		490	33		5,5	3,8	208,7
1559	201,7	21,1	32,9	95,8	2490	25,1	520	35	21,4	7,6	4,0	197,0
1844	193,1	20,2	37,2	79,4	2440		570	38		10,2	4,0	193,2
2049	181,6	19,0	38,9	73,5	2510		580	40		12,2	3,8	200,8
2330	169,2	17,7	41,2	66,7	2590		615	43		15,0	3,4	187,4
2587	157,7	16,5	42,7	61,7	2660		645	46		17,8	3,0	176,5

BENCH TEST RESULTS   **TEST NO.** 14 DPM   **DATE** 15.2.78.

Engine Ford Diesel Vol. Vo. 3611 cm^3 Fuel Shell dieseline Gross C.V. 45640 KJ/kg

8,2 mm dieseline stop + 35 petrol jet - 4,4 needle   + Mobil Prem. petrol 46500 KJ/kg

80 methanol jet - 2,5 needle position   + Methanol 21720 KJ/kg

Actual Speed rpm.	Air Rate m^3/min.	Air Rate kg/min.	Fuel Rate kg/min.	A/F Ratio	Input kW	Torque N.m.	Torque p	Output kW	Efficiency Brake %	Efficiency Volume %	gr/KWh	Bosch Smoke
n	Va	1,01 Ma	D 0,834 P 0,727 M 0,884 M	α	Q	T	p	L	η	τ	υ	No.
1073	1,70	1,72	0,111	15,5	74,5	196,9	20,6	22,1	29,7	87,7	300	3,7
1296	2,02	2,04	0,129	15,8	88,5	196,9	20,6	26,7	30,2	86,3	290	4,0
1583	2,43	2,45	0,152	16,2	105,8	193,1	20,2	32,0	30,2	85,0	285	4,2
1865	2,78	2,81	0,177	15,9	124,2	189,3	19,8	36,9	29,7	82,6	287	4,8
2080	2,04	3,07	0,191	16,1	134,9	179,7	18,8	39,1	29,0	81,0	293	4,3
2337	3,34	3,37	0,202	16,7	143,4	170,2	17,8	41,6	29,0	79,2	292	4,0
2578	3,67	3,71	0,212	17,5	150,4	156,8	16,4	42,3	28,1	78,8	300	3,6
				Total								

Continued on separate sheet

Air rate Va and Ma in first instance from $V = k\sqrt{\Delta p}$ (when $\Delta p > 2$ mm H_2O).

This is correct for air density $p = 1 kg/m^3$. Density correction

$Va = V: \sqrt{p}$ $Ma = V\sqrt{p}$ $Q = M.CV$, $L = p.n.10^{-3}$, $= Ma/M$, $= 100.L/Q$,

$= 200.Va/n.Vo$, $n = \frac{N}{t}$

d = 75,4 mm	k = 0,88
67,9	0,68
60,7	0,51

CONTINUED BENCH TEST RESULTS

TEST 14 DPM of 15.2.78.

	dieseline	petrol	methanol
Fuel Rate kg/min.	0,069	0,017	0,025
	,086	,018	,025
	,108	,019	,025
	,130	,020	,026
	,143	,021	,027
	,154	,021	,027
	,164	,021	,028
Air:Fuel Ratio	24,8	103,1	69,2
	23,7	112,7	82,5
	22,7	130,6	99,1
	21,6	138,7	106,9
	21,5	143,7	113,9
	21,8	164,8	124,2
	22,7	178,7	134,9
Input KW	52,6	12,9	9,0
	65,5	14,0	9,0
	82,3	14,6	9,0
	99,0	15,7	9,5
	108,6	16,6	9,8
	117,7	15,9	9,8
	124,4	16,1	9,9

BENCH TEST RECORD TEST NO. 14 DPM DATE 15.2.78

Engine Ford Diesel Fuel Shell dieseline + Mobil Prem. petrol + Methyl alcohol

Auxiliaries Used: Airfilter/Air Tank/ Fan/ Generator Barometer 654

Ignition or Injection Timing 22 °BTDC (Static), Humidity 56

Compression Ratio 16:1

Other Remarks on Engine Adjustments 8,2 mm injector pump stop - 35 petrol jet - 4,4 needle - 80 methanol jet - 2,5 needle position

Air Orifice Dia 75,4 mm Measuring Vessel 200 cm^3 (grams)

Speed (Tacho) rpm	Torque Nm	p	Power (Meter) Kw	Consumption Time S Diesel	Revs. No.	Temperatures Amb. °C	Exh. °C	Water	Diesel °C	Diff. mmH_2O	Smoke Bosch Nr.	Petrol Consumption Time	Methanol Time
n^1	T	p	L^1	t	N	ta	te	tw	°C	ΔP		sec.	sec.
1073	196,9	20,6	22,1	144,8	2590		450	40		3,7	3,7	523,6	427,8
1296	196,9	20,6	26,7	116,2	2510		480	50		5,2	4,0	481,8	428,7
1582	193,1	20,2	32,0	92,5	2440	29,8	510	54	27,9	7,5	4,2	464,1	428,4
1865	189,3	19,8	36,9	76,9	2390		560	58		9,8	4,8	430,8	403,8
2079	179,7	18,8	39,1	70,1	2430		590	61		11,7	4,3	408,2	393,4
2337	170,2	17,8	41,6	64,7	2520		610	65		14,1	4,0	426,2	390,7
2578	156,8	16,4	42,3	61,2	2630		620	68		16,8	3,6	420,6	386,0

NOTE:

The nine graphs are given separate, to enable you to lay a graph on top of the other.

By holding them against the light the gains and losses can easily be studied.

As reference line the KW line can be used.
The normal dieseline Test No. 17 should be taken as standard.

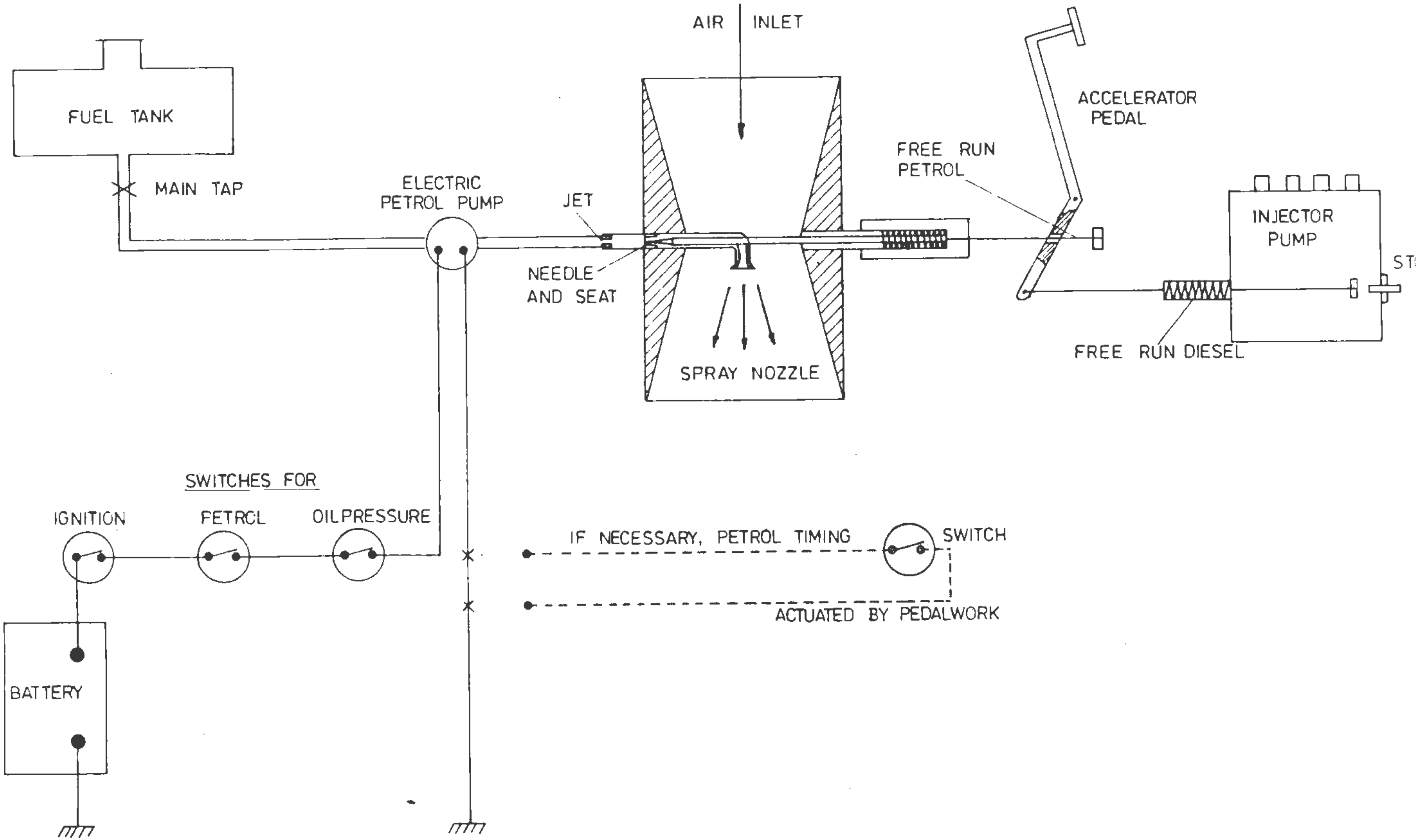
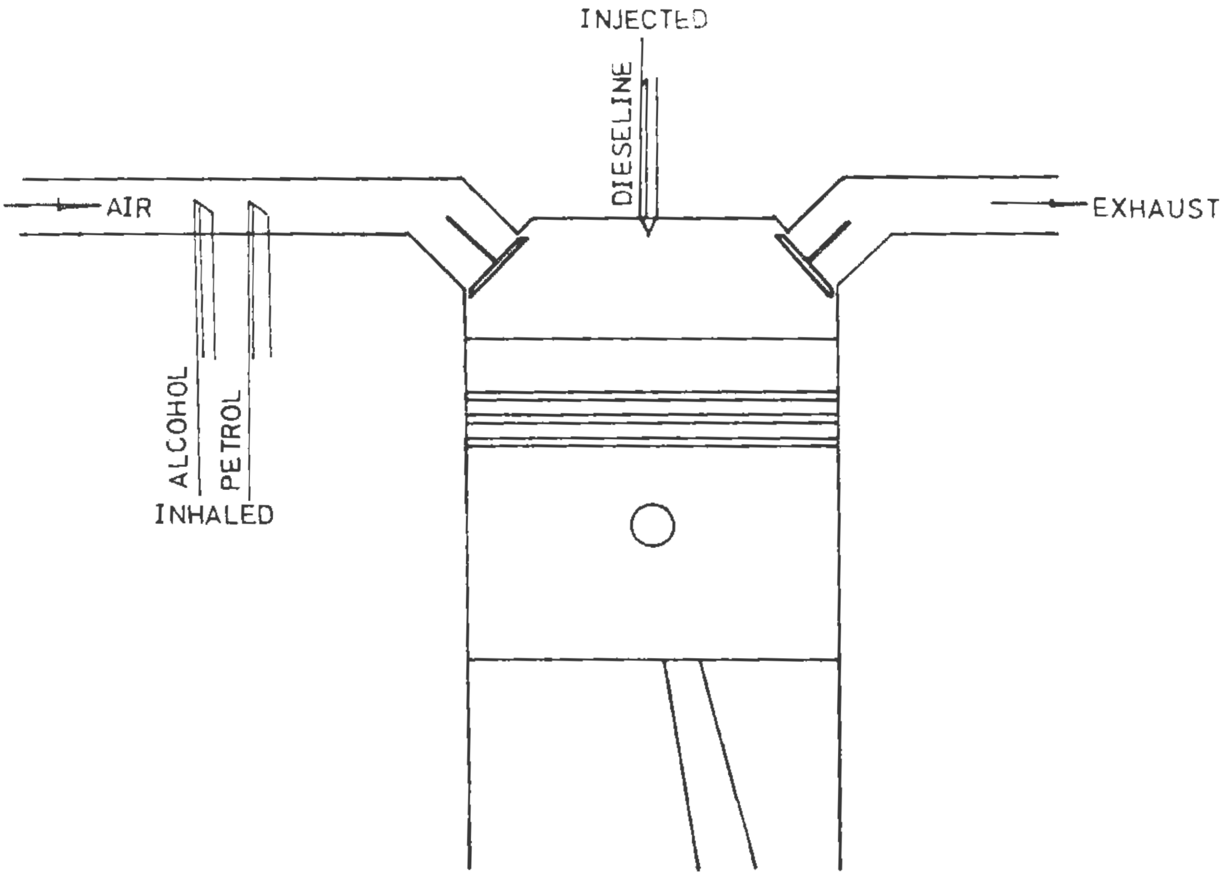
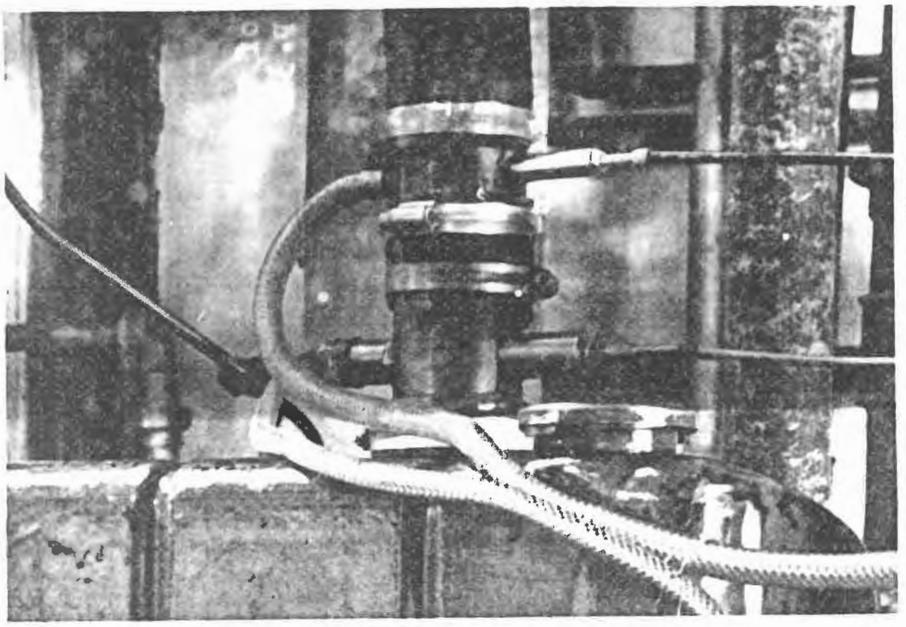


FIGURE 1 LAYOUT OF CONTROL SYSTEM FOR DUAL FUEL OPERATION OF A DIESEL ENGINE

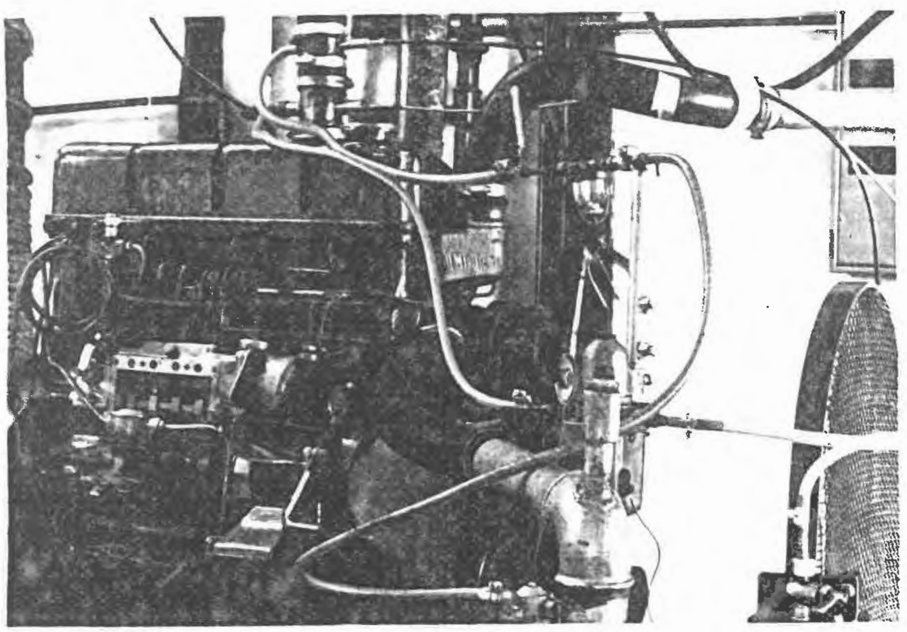


TRIPLE FUEL SUPPLY TO ENGINE

FIGURE 2



2 CARBURETTORS FITTED FOR TRIPLE FUEL OPERATION



ENGINE SET UP FOR TRIPLE FUEL OPERATION