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WATER INJECTION IN THE NORMALLY-ASPIRATED PISTON ENGINE

by

E. M. GOODGER

NOTE NO.78
March, 1957.

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Water Injection in the Normally-Aspirated Piston Engine*

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SUMMARY

The injection of water into a spark-ignition piston engine tends to reduce both the power and the tendency to knock. Because of the latter effect, the conditions of operation can be made more severe in order to realise overall gains in power and economy.

Injection of water at different points in a single-cylinder engine shows the most practicable method to be fine atomization into the inlet manifold. Water flow requirements are found to be directly proportional to the manifold absolute pressure, and a commercial type of water injection unit designed on this basis is described and road-tested.

An attempt has been made to estimate the distribution of water to individual cylinders of a multi-cylinder engine, and the effect of water up on engine components is examined.

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* This note includes results from a thesis submitted by Lt. J. H. Dunphy, R.N., in June 1955 (ref.7), as part of the requirements for the award of the Diploma of the College of Aeronautics. Results are also included from additional work by J. Hillsdon (ref.8), the research being under the supervision of the author.

Introduction.

The use of water injection in internal-combustion engineering dates back to 1913, when Professor Hopkinson injected water into a gas engine in order to cool the chamber walls (ref.1). A considerable reduction in thermodynamic efficiency was found, which indicated that the water had vaporised before reaching the walls and had reduced the activity of the burning gases. The object of water injection today is, in fact, to cool the end-gas portion of the charge, rather than the chamber walls, but similar adverse effects upon performance may result unless the system is applied correctly.

The attraction of water injection lies in its suppression of knock. Under normal conditions of combustion, the spark-ignited flame propagates throughout the chamber producing a smooth rise in pressure. During this process, the unburned portion of the charge experiences the same pressure rise and, in consequence, an adiabatic rise in temperature. Under severe conditions of operation, the temperature of the unburned portion reaches the level for spontaneous ignition, and the end-gases ignite and burn at a very rapid rate. This phenomenon is termed knock, and its occurrence depends largely upon the temperature level within the chamber, and the spontaneous-ignition characteristics of the charge at the normal peak chamber pressure. Improved cooling of the end-gas zone of the chamber, either externally or internally, widens the gap between the two temperature levels, and so delays the onset of knock.

A recognised anti-knock practice in spark-ignition piston engines is to use the fuel itself as an internal coolant, by employing an over-rich mixture under extreme power conditions. Water appears more attractive as a knock suppressant in view of its negligible cost and higher thermal capacity. The reduction in power occasioned by the injection of water must be offset, and outweighed, by an increase in the severity of operating conditions, so that overall gains in performance result.

An additional advantage claimed for water injection is the more complete combustion of the carbon content of the fuel, the reaction between hot carbon and water resulting in gaseous carbon monoxide and hydrogen. This tends to minimise the formation of carbon deposits, and extend the periods between top overhauls.

2. Effects of Water Upon Engine Performance.

2.1. Humidity and Air Temperature.

Some water vapour is normally present in atmospheric air, and the amount of water capable of existence in the vapour phase depends upon both temperature and pressure. Relative humidity is expressed on a percentage basis, derived from the mass ratio of water vapour actually present in unit volume of moist air to that which would be present at full saturation. In hot, moist, climates, a maximum relative humidity of about 90% is found at an air temperature of about 40°C., which represents a water vapour content of 0.044 lb./lb. dry air. Water vapour displaces dry air, so that a given volume of this moist air contains only 93.4% of the mass of dry air occupying the same volume. (See refs. 2 and 3).

In general, the power output of a spark-ignition engine is directly dependent upon the mass throughput of dry air. Since the swept volume is fixed, operation at constant temperature and variable humidity can be assumed to incur a constant volume throughput of moist air. The value 93.4%, therefore, represents the reduced power available due to a relative humidity of 90% at 40°C., and fig. 1 shows how the power varies with water vapour mass concentration.

In a similar way, assuming a constant volume throughput, the power varies inversely with the intake air absolute temperature, as shown in fig. 2.

2.2. Induction and Combustion Processes.

The individual effects of water introduced into the induction manifold of a spark-ignition piston engine are illustrated in the following table :-

TABLE 1.

Stage	Process	Effect on Performance
Induction	Vaporisation cools mixture until air saturated.	Power increased by cooling until saturation, then decreased by air displacement.
Compression Stroke	Further vaporisation cools charge and reduces work of compression.	Power increased.
Power Stroke (main flame)	Vaporisation, and presence of vapour, both reduce flame speed, and hence rate of pressure rise.	Power reduced.
Power Stroke (end-gas zone)	Vaporisation, and vapour, cool end gases and raise slightly their spontaneous-ignition temperature.	Knock suppressed.

Inspection of the above table suggests that, with low rates of water flow, the two initial effects would produce a slight overall increment in power output. At higher rates of water flow, the reduced flame speed could be expected to reverse the first beneficial effect and outweigh the second, so that a progressive loss of power would result.

3. Engine Bench Tests.

The introduction of water at different points in the engine airflow system should help to clarify the relative magnitude of the effects shown in Table 1. The Fedden single-cylinder sleeve-valve engine (see Appendix) was tested at constant conditions of 2,500 r.p.m., full throttle, 15° ignition advance, and maximum power mixture (approx. 12/1 A/F), using an Argus carburettor. Water was injected using four different methods, viz:-

A. Continuous injection upstream of carburettor }
B. Timed injection into inlet port } Inlet Manifold

C. Injection during compression stroke }
D. Injection during power stroke } Combustion Chamber

Additional tests were then conducted to determine the effect of injected water upon knock-limited power.

3.1. Continuous Injection Upstream of Carburettor.

Water from an overhead supply tank was passed through an edge filter into a Siemens gear-type pump running at 800 r.p.m. In order to obtain a finely-divided water spray in the manifold, the water was fed through an upriser pipe into a 'scent-spray' atomiser, using an atomising air pressure of 10 p.s.i.g. (Figs. 3 and 4). Interchangeable atomiser caps of different orifice diameters were available in order to obtain a wide range of water flow (i.e. to suit both a single- and a multi-cylinder engine) for a given range of water pressure from zero to 250 p.s.i.g. A 50 ml. bulb was used to determine the water flow rate. The variations with water flow in power, specific fuel consumption, and specific liquid consumption are shown in Figs. 5, 6, 7 and 8.

The shape of the power curve in Fig. 5 confirms the conclusions in para. 2.2 regarding the relative significance of the individual effects of water on the processes of induction and combustion.

The lower curve in Fig. 5 illustrates the reduction in power caused by an increase in humidity, as outlined in para. 2.1.

3.2. Timed Injection, into Inlet Port.

The Fedden engines were designed to operate with an Excello low-pressure (30 p.s.i.g.) timed fuel injection system, and one of the Excello spring-loaded injectors was used to spray water into the central inlet port of the Fedden single-cylinder engine. An Argus carburettor was used for the fuel supply. The water was pumped on a timed basis by means of an engine-driven Bryce single-element diesel-type fuel pump. In order to minimise corrosion in the system due to the passage of water, an oil tank and two-way cock were incorporated in the water system, and an Ensis type water-repelling oil was fed into the pump at the beginning and end of each test. The sleeve inlet port opened from 15° before top dead centre to 55° after bottom dead centre, and injection of water was timed to commence at 20° after top dead centre on the induction stroke. Results are included in Figs. 6, 7 and 8.

3.3. Injection During Compression Stroke.

The same Bryce pump and water system were employed, feeding to a B.M.W. 801 injector, operating at 500 p.s.i.g., fitted into the cylinder head. Injection commenced at 55° after bottom dead centre, i.e. at the instant of closure of the inlet port. Results are shown in Figs. 6, 7 and 8.

3.4. Injection During Power Stroke.

Using the same equipment, the water injection timing was retarded to commence at top dead centre on the power stroke. Results are included in Figs. 6, 7 and 8.

3.5. Comparison of Results.

The power curves in Fig. 6 show clearly that the most effective system of water injection is method B, in which the water is injected through the inlet port on a timed basis. Although less time is available for a 'supercharging' effect due to vaporisation and charge cooling, the injected water is introduced into the chamber in a positive manner, and does not displace the manifold air.

Corresponding variations are seen to occur in the specific fuel consumption (Fig. 7). The curves in Fig. 8 indicate the extent of the price to be paid, in terms of the total consumption of liquids, for the advantages gained when using water injection.

Despite these optimum results for method B, it can be concluded that the continuous manifold injection system of method A is more suitable, since this does not require the provision of a high-precision timed injection system, nor of oil-addition to prevent corrosion of the pump or injector. The slight increase in power at low water flows is acceptable, but not of sufficient magnitude in itself to warrant the additional complication of a water injection system.

3.6. Knock-Limited Power Tests.

The real value of water injection lies in its suppression of knock, and in the possibility of operating at severe conditions which otherwise would be prohibited. Increases in severity, in the form of greater ignition advance or higher compression ratio, lead to improved power and economy. Tests were conducted, using the continuous manifold injection system (method A) in the F.K.F.S. single-cylinder variable-compression engine, with the ignition advanced to incipient knock at various rates of water flow. The knock-limited power curve is shown in Fig. 9A together with various falling power curves obtained at constant values of ignition advance. The figure shows an increase of 10.6% power to be obtainable by means of an advance in ignition from 8 to 41° before top dead centre, with a water/fuel mass ratio of about 1.5. Water/fuel mass ratios in excess of 1.5 led to unstable operation.

Further tests were conducted with the compression ratio increased to incipient knock at various rates of water flow. The results in

Fig. 9B show an increase of 4.6% power obtained from an increase in compression ratio from 6.4/1 to 8/1, with a water/fuel mass ratio of about 1.5.* The cruves in Fig. 9C show similar reductions in specific fuel consumption (of approx. 10%) when ignition timing and compression ratio are increased separately to maintain an incipient knock condition with the injection of water. These compare with the increase in specific fuel consumption of over 40% that resulted when water was injected without advancing ignition to incipient knock. Increases in specific liquid consumption of about 100% were incurred in each case, which compare with an increase of about 300% when ignition advance remained constant.

4. Practical Systems.

The greatest tendencies to knock are found at full throttle and low engine speeds, where the chamber pressure is high, and the time available for spontaneous ignition is long. A change in engine speed at constant throttle position gives rise to a change in the manifold pressure. A system of water injection into the intake manifold, therefore, should be controlled so that the water flow is directly proportional to the manifold absolute pressure, i.e. maximum water supplied at low manifold depression. It should also ensure that no water flow is possible when the engine is shut down, i.e. zero water supplied at zero manifold depression (Fig. 10). These two requirements almost conflict, and two control devices may be necessary.

In one commercially-developed system, (ref. 4), the water is introduced into the manifold, on the engine side of the throttle, under the action of the manifold depression (Fig. 11). The inverse relationship between manifold depression and water flow is obtained

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* Intercomparison between the two sets of results (Figs. 9A and 9B) may not be made directly since some engine maintenance work effected between the two tests was found to have influenced the performance level.

by allowing the manifold pressure to act upon a spring-loaded diaphragm and jet device. Reduced depression allows the spring to push the jet away from a fixed "inverse-flow" needle, and so increase the flow. The manifold pressure reaches a mximum value, equal to atmospheric pressure, when the engine is shut down. Since the inverse-flow jet is then fully open, a second needle is incorporated in order to cut off the supply of water. The cut-off needle is also connected to the diaphragm, and is arranged to close by the action of the diaphragm spring. Manual controls are fitted so that the water flow may be adjusted to suit the engine and driving conditions.

In an alternative system, developed in America, (Ref. 5), the injection device is virtually a carburettor, complete with float chamber (Fig. 12). The leaded alcohol-water fluid is delivered into the venturi of the main carburettor, under the action of the venturi depression. Since the venturi pressure bears an inverse relationship with the manifold pressure, changes in throttle position at constant engine speed cause the fluid flow to vary directly with the venturi depression, and no inverse-flow needle is required. However, this gives a less sensitive control of fluid for engine speed at constant throttle position, and a passage (shown dotted) is led from the diaphragm chamber to the venturi tube in order to improve this sensitivity.

4.1. Water Distribution to Individual Cylinders.

The successful application of any system in which fluid is introduced into the manifold of a multi-cylinder engine depends upon the uniformity of distribution of the fluid to the individual cylinders. Performance suffers markedly if the fuel is badly distributed, and similar problems arise when water or other manifold-injected fluids are employed.

Since the mean temperature of the chamber gases reduces progressively with increase in water flow, this temperature was investigated with a view to its use as a quantitative indication of water flow rate to the cylinder. The Fedden single-cylinder engine was used in order to obtain the temperature-flow relationship, and this curve was then employed as a calibration of water flow rate to the individual cylinders of the Fedden flat-six engine. The six cylinders of the latter engine are of the same design and dimensions as that fitted to the single-cylinder engine.

A thermocouple is incorporated in the head of the Fedden single-cylinder engine as a standard fitment, and the variations in temperature with water flow are shown in Fig. 13. The curves follow the general trends shown by the power curves in Fig. 6. In order to increase sensitivity, and hence accuracy, a thermo-couple-type sparking plug was fitted and the mean temperature at the tip of the central electrode was measured. The temperature-flow calibration curve obtained using injection method A , is shown in Fig. 14. After an initial slight curvature, the relationship is almost linear, and the calibration appears to be satisfactory for application to the flat-six engine.

The 'scent-spray' water atomiser unit was fitted onto the inlet of the Argus carburettor feeding the manifold of the flat-six engine, as shown in Figs. 15A and B. The Argus carburettor had been fitted for earlier tests on fuel distribution but, for the water distribution tests, the fuel was supplied by means of the standard Excello port-injection system, and the carburettor was not used. The engine was run at 2500 r.p.m. and 15° ignition advance, with a constant fuel consumption. Individual cylinder temperatures were measured by means of thermocouple sparking plugs. The engine consists essentially of two three-cylinder units mounted back to back, and the water distribution patterns were to be based on the two banks separately, i.e. cylinders 1, 3, and 5, and cylinders 2, 4 and 6.

The engine performance was not quite up to standard when the initial water distribution tests were conducted, due mainly to faults in the fuel system and in the thermocouple leads from cylinders 1 and 2, but the results are presented as an indication of the order of difference between individual flows. Figs. 16 and 17 show the temperature variation and the deduced water flows respectively, for cylinders 1, 3 and 5 only. If cylinder 1 is disregarded, in view of the thermocouple lead faults, the water appears to have traversed to the end of the straight rake manifold, and to have supplied cylinder 5 with a greater share. However, the results are too meagre for definite conclusions to be drawn.

4.2. Effect of Water on Engine Components.

On completion of the water injection tests in the single-cylinder engine, the engine was stripped and examined for signs of corrosion and deposits. In order to minimise the possibilities of corrosion, the engine had been run for five minutes at the end of each test after the injection of water had ceased, during which time the Bryce pump was fed with Ensis oil. The total running time of 400 hours included 20 hours with water injection in operation. The condition of the engine components shown in Fig. 18 was considered to be normal, and no signs of corrosion were apparent. The carbon deposits were as expected, and the injected water did not appear to have had any great effect upon the existing carbon which must have built up before the tests were commenced. The Bryce pump components are shown in Fig. 19. No evidence of corrosion was apparent, nor any damage to the precision-ground plunger and barrel.

4.3. Engine Road Tests.

Some preliminary road tests were conducted in a pre-war car with a four-cylinder 10 h.p. engine fitted with the commercial water injection unit of Fig. 11. An on-off cock was incorporated in the water delivery line so that the system could be isolated. A vacuum gauge was also fitted, and a maximum manifold depression of 8 p.s.i.g. was noted under idling conditions.

With current (1955/56) motor fuels, a range of octane rating (motor method) of about 72 to 82 was considered to be available, from regular to premium grades. The difference in severity between optimum conditions for the two fuels is not as wide as that represented in Fig. 9, and a water/fuel volume ratio of about 0.4 was estimated as the probable maximum. Since, in practice, knocking conditions are encountered for a fraction only of driving time, and since this would represent no more than about 20% of the total fuel consumption, the overall water/fuel volume ratio on the road would be expected not to exceed about 0.08, i.e. approximately 1 gallon of water to 12 gallons of gasoline, or a consumption of 1 pint of water per 50 miles.

The engine was operated initially with premium-grade gasoline at optimum ignition and mixture. Regular-grade gasoline was used next, at the same engine conditions, and knock was experienced under acceleration and hill-climbing. Water was introduced, and the flow rate adjusted to eliminate knock under the same conditions of acceleration and hill-climbing. A water consumption of 1 pint per 50 miles was found to represent a fair average at this setting. Increased water flow gave a noticeable power loss, and a reduced water flow led to trace knock.

The fuel mixture was not weakened when water injection was in operation, but some slight improvement in economy was noted. Fuel consumption was recorded during consecutive periods with and without water, the total mileage in each case being about 800 miles. Average values of 25.0 and 23.0 m.p.g. respectively were obtained, i.e. an improvement in economy of 8.7%. A top overhaul at the conclusion of the tests revealed a marked reduction in the extent of carbon deposits. Normal mains water was used throughout the tests, and a tendency to jet blockage was noted occasionally.

5. Alternative Injection Fluids.

For operation under cold weather conditions, some form of frost protection is required for the water injection system, and the effects on the engine of the anti-freeze agents must be considered. The properties of some possible anti-freeze materials are presented in Table 2, together with the properties of water and of a typical motor gasoline. The effectiveness of water as an internal coolant in comparison with gasoline, is brought out by the relative values of latent heat. In addition, the specific heat of water is roughly twice that of all the other liquids shown. The conventional anti-freeze material, ethylene glycol, is only about one third as effective a heat absorber as water, whereas methanol is about one half as effective as water. In addition, the methanol contained in a water-methanol blend will act as a blending agent in the fuel, and give a slight trend towards improved anti-knock quality. Alcohols are, therefore, the recommended anti-freeze agents for water injection systems, and the freezing points of water-alcohol blends are shown in Fig. 20. Domestic methylated spirits contain approximately 97% of mixed alcohols.

6. Conclusions.

1. Water vapour displaces dry air when it enters the atmosphere, so that a reduction occurs in the mass flow rate of dry air when an engine operates with humid air. Increased humidity, therefore, leads to a reduction in engine power.

2. Increased air temperature reduces the mass flow rate of dry air, and leads to a reduction in engine power.

3. Injection of water into the manifold air of a spark-ignition piston engine leads to an initial slight rise in power due to the cooling effect on the charge resulting from vaporisation, and to the reduction in the work of compression.

4. When the manifold air has become saturated, the engine power drops with increased ra tes of water flow due to displacement of dry

air by the water vapour, and to the inhibiting effect of the water upon the speed of flame propagation.

5. A maximum water/fuel mass ratio of about 1.5 is obtainable before the onset of unstable operation due to 'drown out' of the sparking plugs.

6. The reduced power output resulting from extensive water injection leads to a lower chamber temperature and a reduced tendency to knock.

7. The cooling effect of water injection upon the end gases, and the slight increase in their spontaneous-ignition temperature due to the presence of water vapour, lead to a reduced tendency to knock at more severe conditions of operation, so that an overall power improvement can be realised.

8. By maintaining a condition of incipient knock, an improvement in power of about 10% is obtainable with ignition advanced over a wide range, and of about 6% with compression ratio increased from 6.4/1 to 8/1.

9. By maintaining a condition of incipient knock, fuel economy can be improved by about 10%.

10. The presence in the chamber of additional water vapour tends to inhibit further carbon deposition, but not to affect existing deposits.

11. Alcohols are considered to be suitable anti-freeze agents for water injection systems.

TABLE 2. - PROPERTIES OF ALTERNATIVE INJECTION FLUIDS.

MATERIAL	CHARGE COOLING PROPERTIES		ANTI-KNOCK PROPERTIES	
	BOILING POINT °C	LATENT HEAT Cal./gm.	AUTOGENOUS-IGNITION Temp. °C. (A .S.T.M. D286-30)	OCTANE NO. MOTOR METHOD
Typical Motor Gasoline	35 to 200	70	315 *	73
Water	100	539.3	-	-
Ethylene Glycol	197.5	191	436 *	-
Methanol	64.5	263	489 *	98
Ethanol	78.3	201	439 *	99

* Ref. 6

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APPENDIX

ENGINE DATA

ITEM	FEDDEN SINGLE	FEDDEN FLAT SIX	F.K.F.S.
Bore m.m.	101.6	101.6	100
Stroke m.m.	95.3	95.3	130
C.R.	8/1	8/1	4.5/1 to 25/1
Swept Volume c.c.	773	4638	1020
Ignition	Magneto Two Plugs	Magneto Two plugs/cyl.	Magneto One plug
Fuel Metering	Argus Carburettor or Excello port injection	A rgus Carburettor or Excello port injection	Port Injection

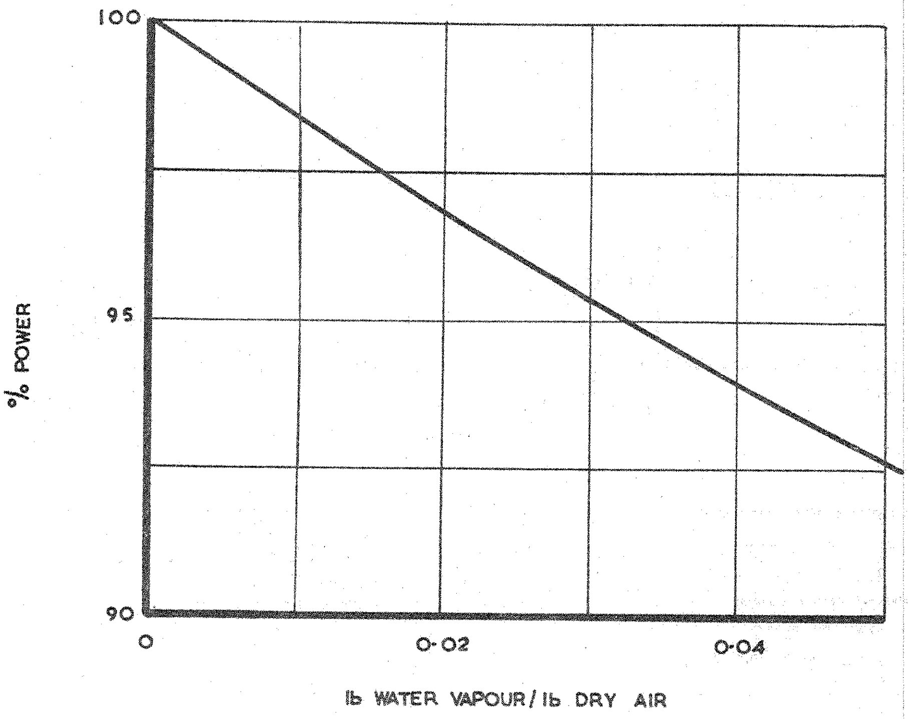


FIG. 1. EFFECT OF HUMIDITY ON ENGINE POWER

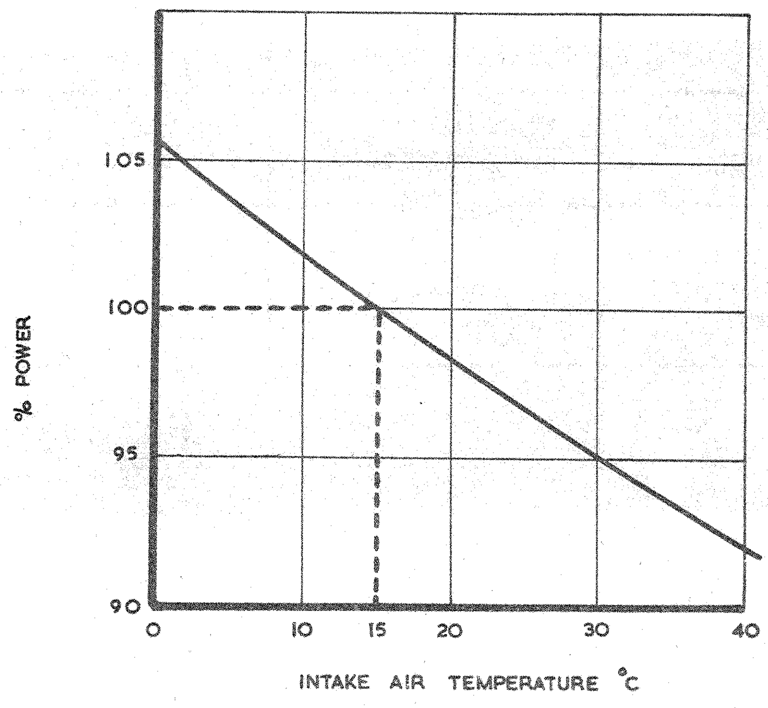


FIG. 2. EFFECT OF INTAKE AIR TEMPERATURE ON ENGINE POWER

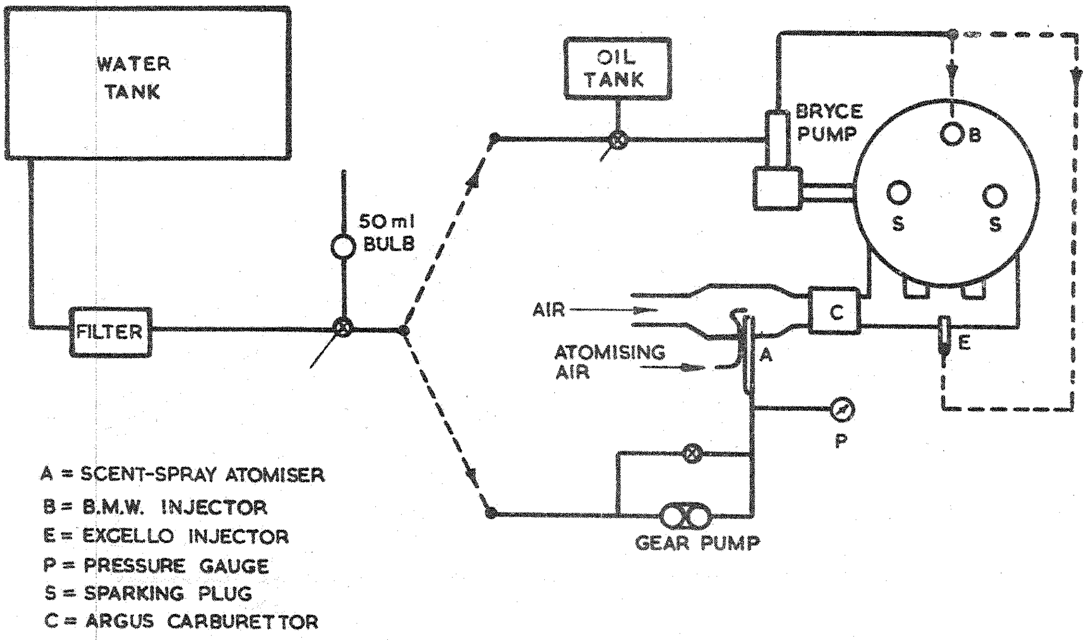


FIG. 3. WATER-INJECTION SYSTEMS USED ON FEDDEN SINGLE-CYLINDER ENGINE

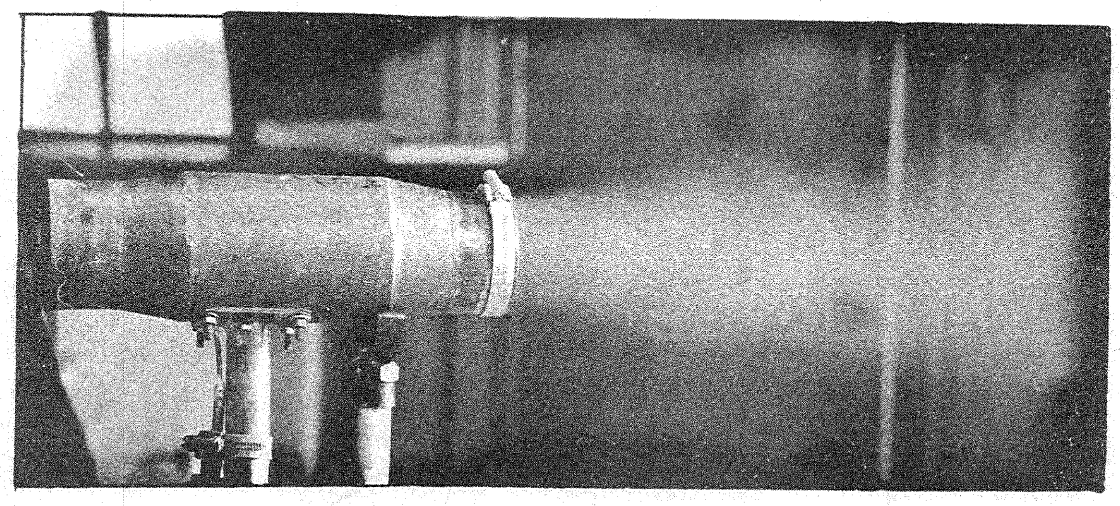
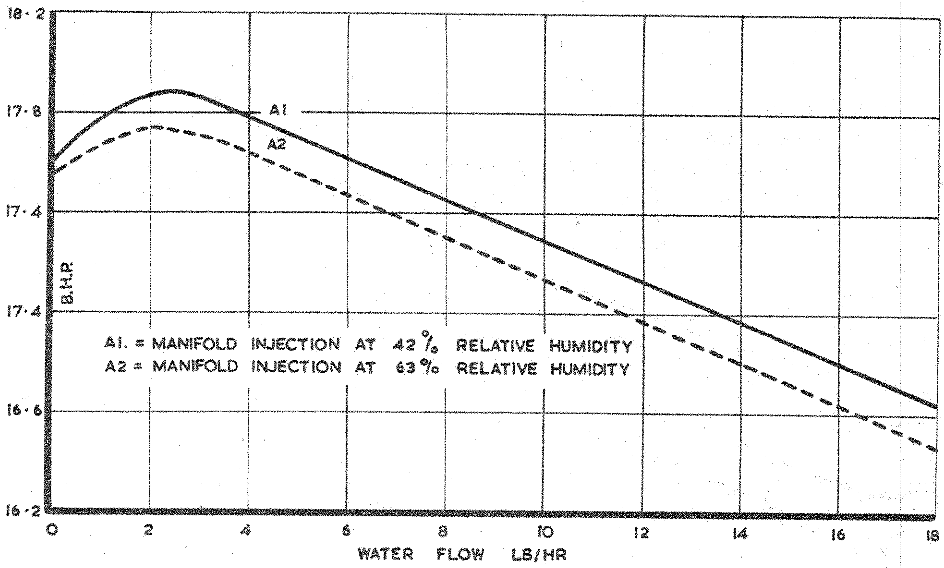
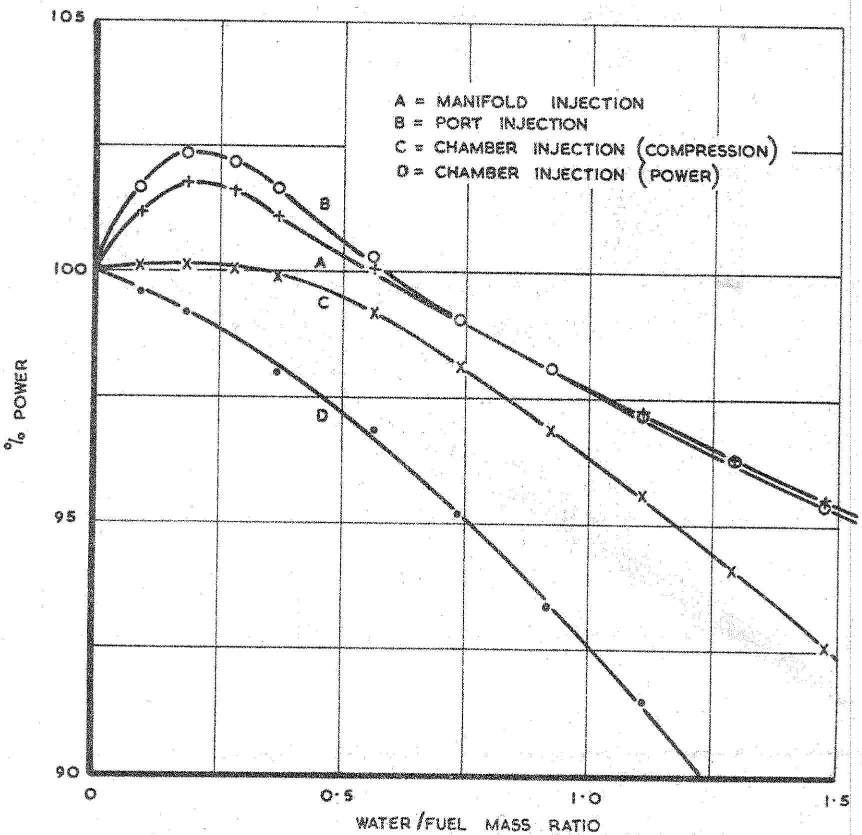


FIG. 4. FINELY-DIVIDED SPRAY PRODUCED BY SCENT-SPRAY ATOMISER AT 100 p.s.i.g. WATER PRESSURE AND 10 p.s.i.g. AIR PRESSURE.



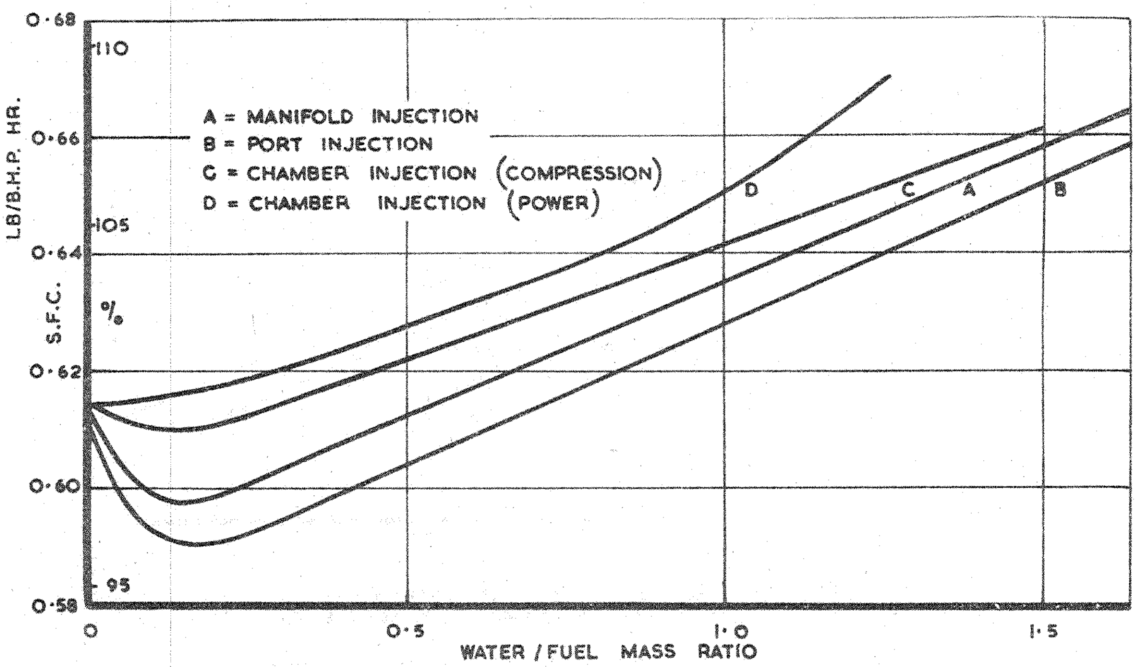
FEDDEN SINGLE-CYLINDER ENGINE, FULL THROTTLE, 2500 R.P.M., MAXIMUM POWER MIXTURE, 15° IGNITION ADVANCE, 73 AVGAS. (REF. 7.)

FIG. 5. EFFECT OF MANIFOLD WATER INJECTION AND HUMIDITY ON ENGINE POWER.



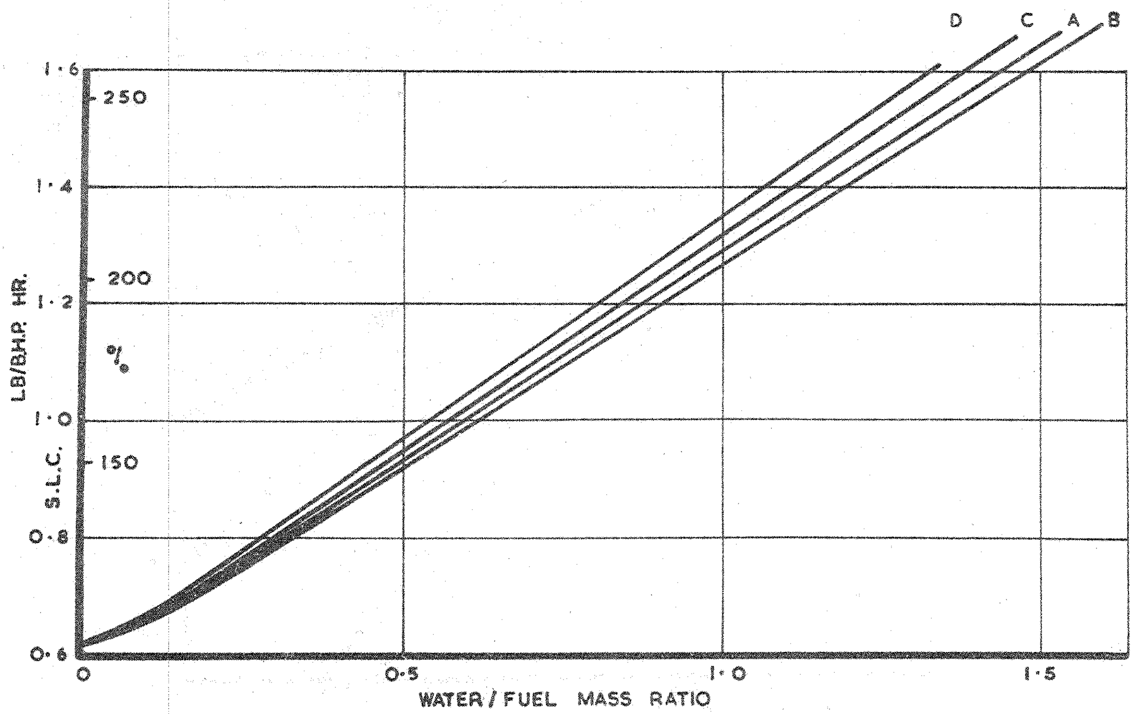
FEDDEN SINGLE-CYLINDER ENGINE CONDITIONS AS IN FIG. 5. (REF. 7.)

FIG. 6. EFFECT OF WATER INJECTION METHOD ON ENGINE POWER



FEDDEN SINGLE CYLINDER ENGINE CONDITIONS AS IN FIG. 5. (REF. 7.)

FIG. 7. EFFECT OF WATER INJECTION METHOD ON SPECIFIC FUEL CONSUMPTION



FEDDEN SINGLE-CYLINDER ENGINE CONDITIONS AS IN FIG. 5. (REF. 7.)

FIG. 8. EFFECT OF WATER INJECTION METHOD ON SPECIFIC LIQUID CONSUMPTION.

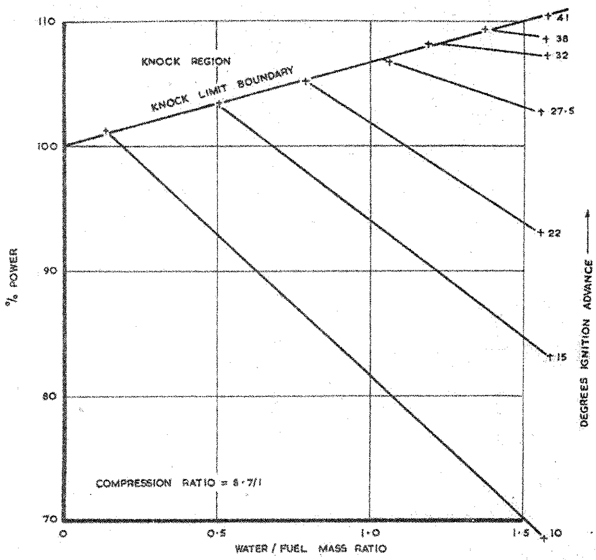


FIG. 9a. VARIABLE IGNITION TIMING

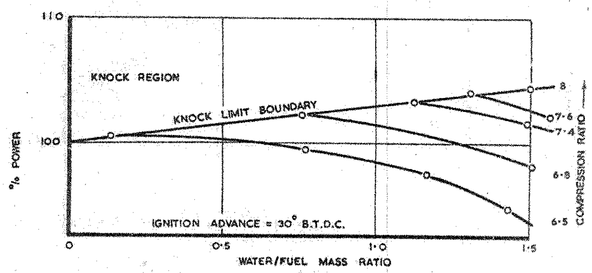
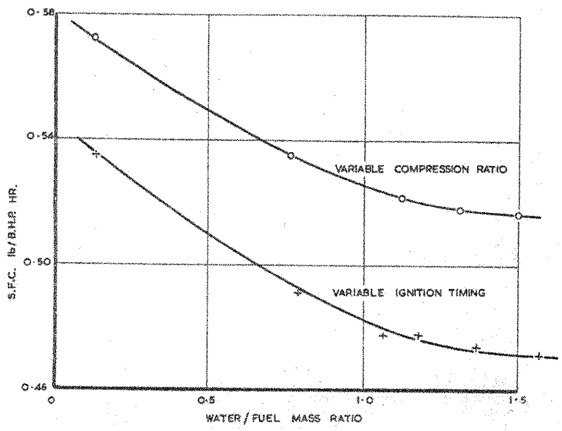


FIG. 9b. VARIABLE COMPRESSION RATIO



F.K.F.S. ENGINE, FULL THROTTLE, 1800 R.P.M., MAXIMUM POWER MIXTURE, 73 AVGAS, WATER INJECTION METHOD A. (REF. 8)

FIG. 9c ECONOMY CURVES

FIG. 9. EFFECT OF WATER INJECTION ON POWER & ECONOMY WITH VARIABLE IGNITION TIMING AND COMPRESSION RATIO.

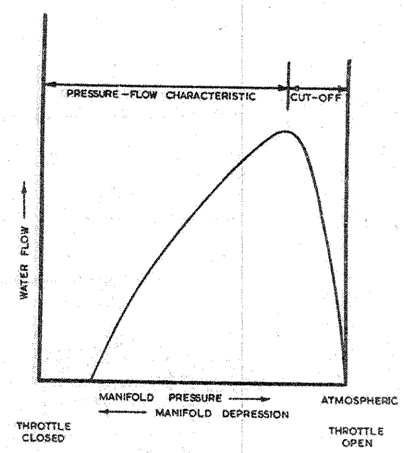


FIG. 10. WATER FLOW REQUIREMENTS

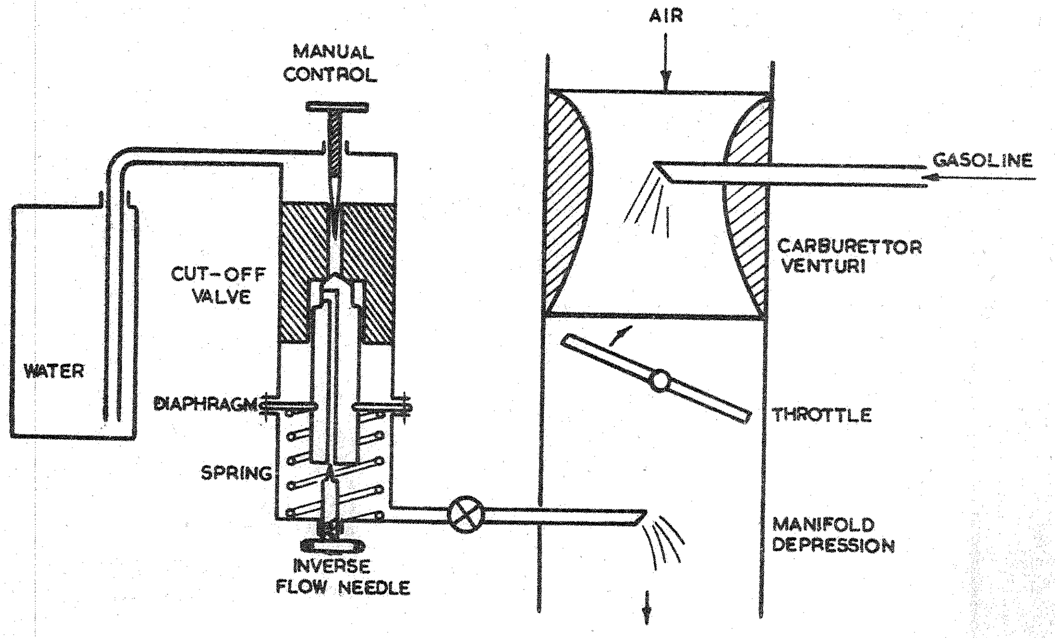


FIG. 11. DIAGRAMMATIC REPRESENTATION OF A COMMERCIAL WATER INJECTION UNIT

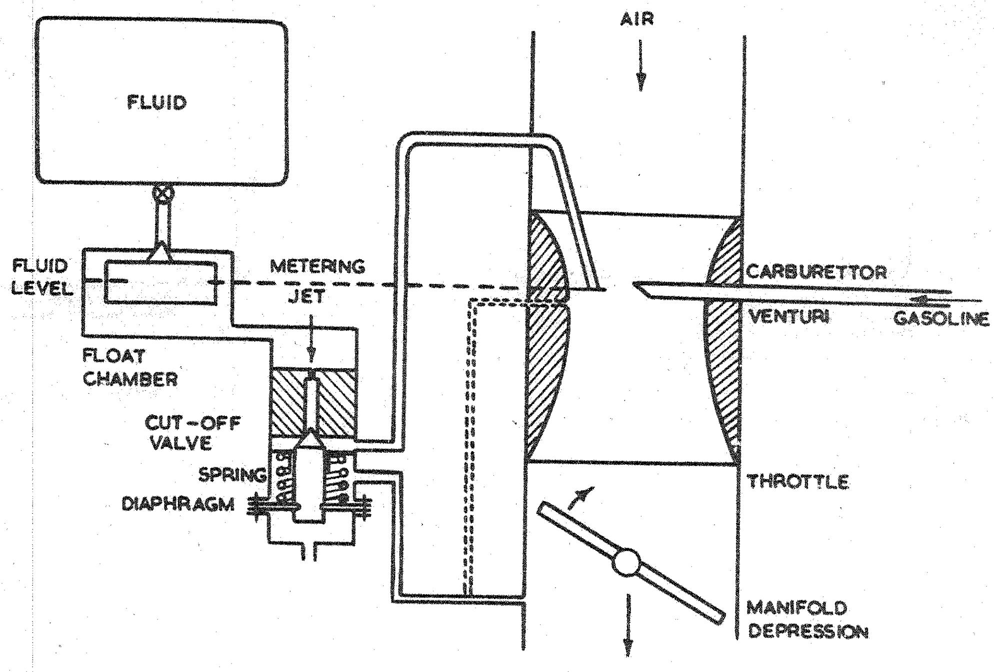
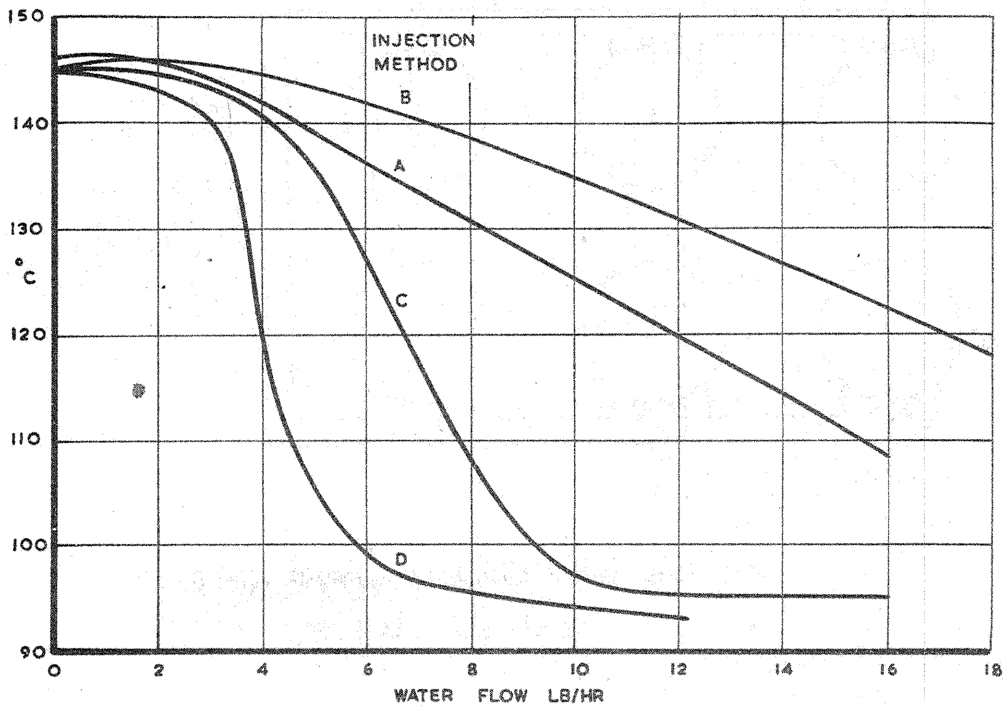
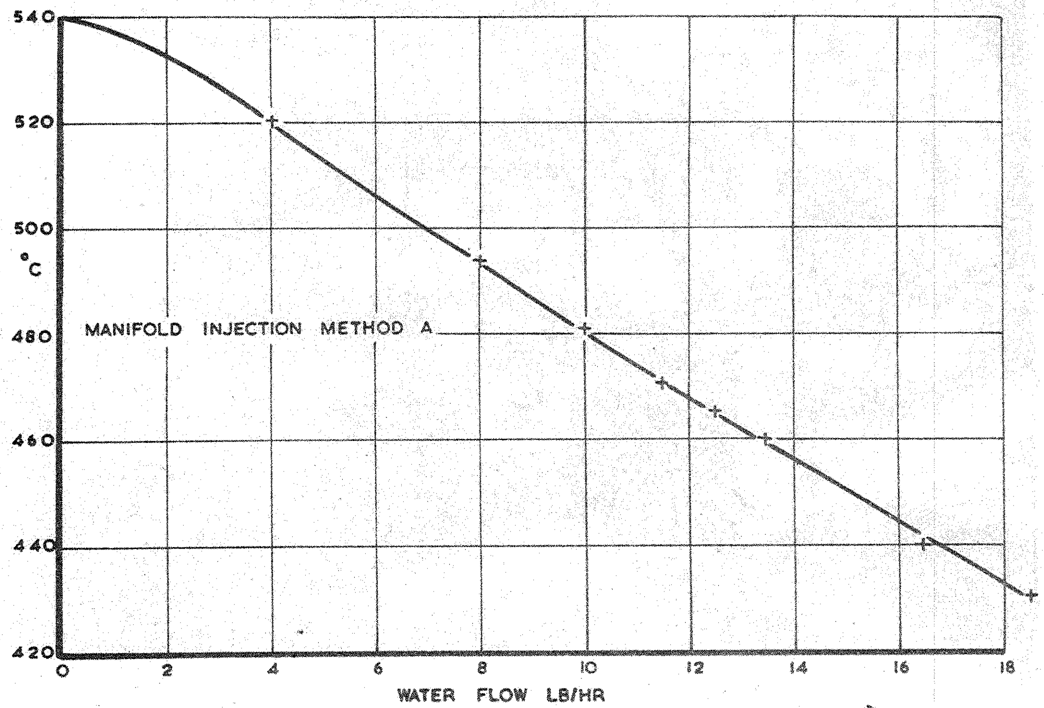


FIG. 12. DIAGRAMMATIC REPRESENTATION OF AN AMERICAN FLUID INJECTION UNIT.



FEDDEN SINGLE CYLINDER ENGINE CONDITIONS AS IN FIG. 5. (REF. 7.)

FIG. 13. VARIATION OF CYLINDER HEAD TEMPERATURE WITH WATER FLOW



FEDDEN SINGLE CYLINDER ENGINE CONDITIONS AS IN FIG. 5. (REF. 7.)

FIG. 14. VARIATION OF THERMOCOUPLE SPARKING – PLUG MEAN TEMPERATURE WITH WATER FLOW

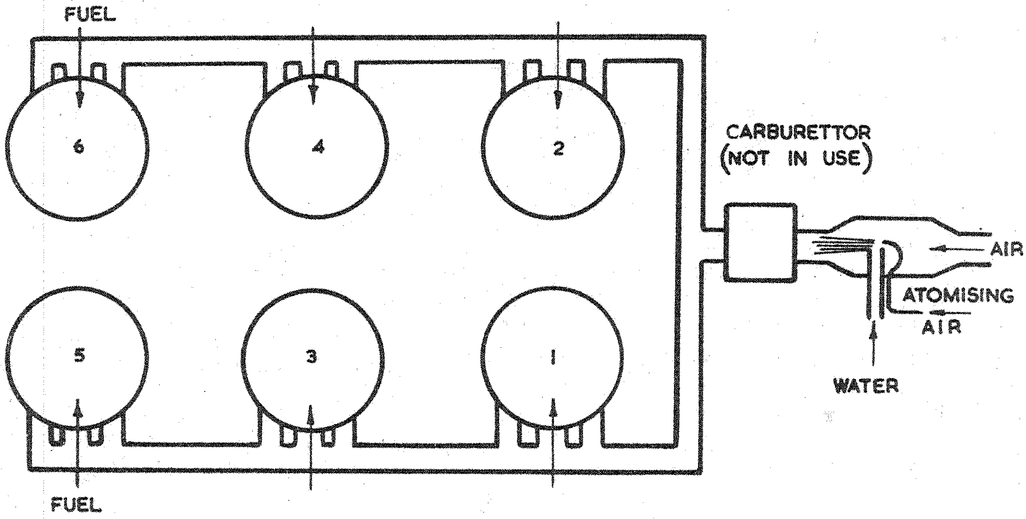


FIG. 15a DIAGRAMMATIC LAYOUT OF FEDDEN FLAT-SIX MANIFOLD

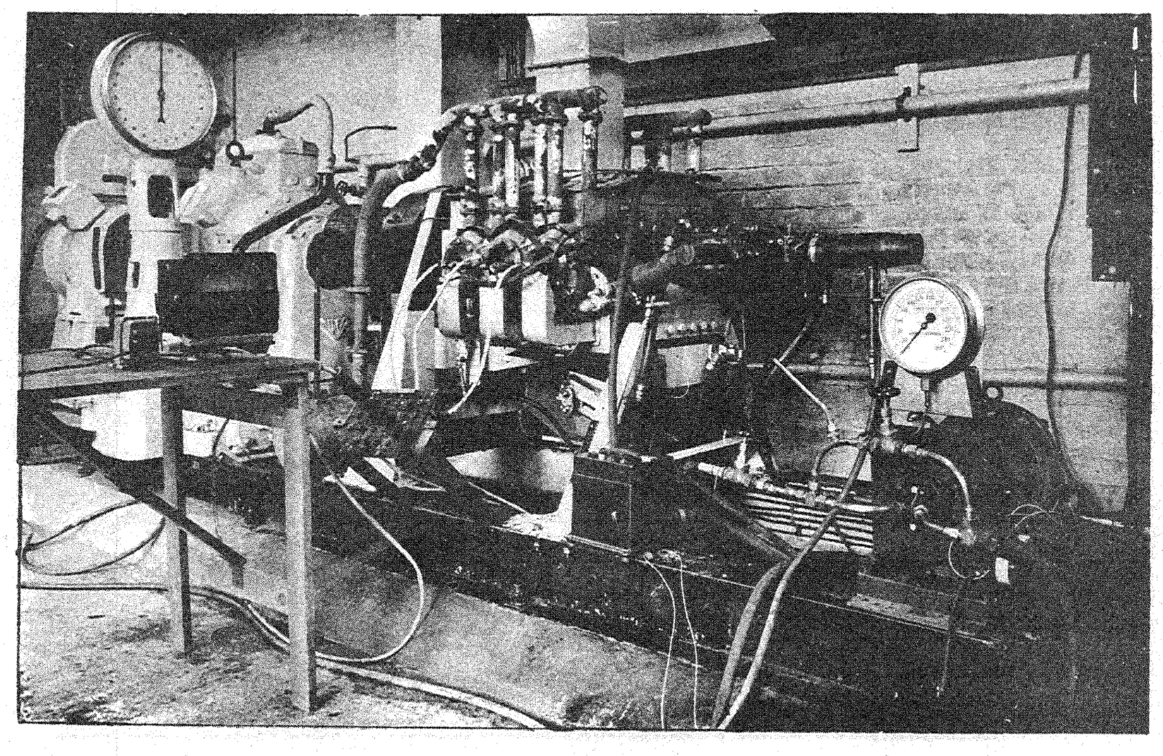
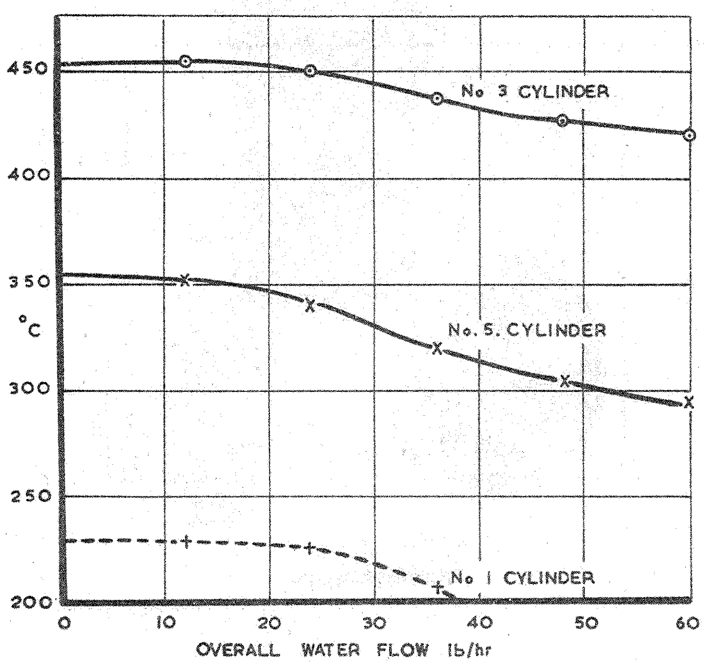
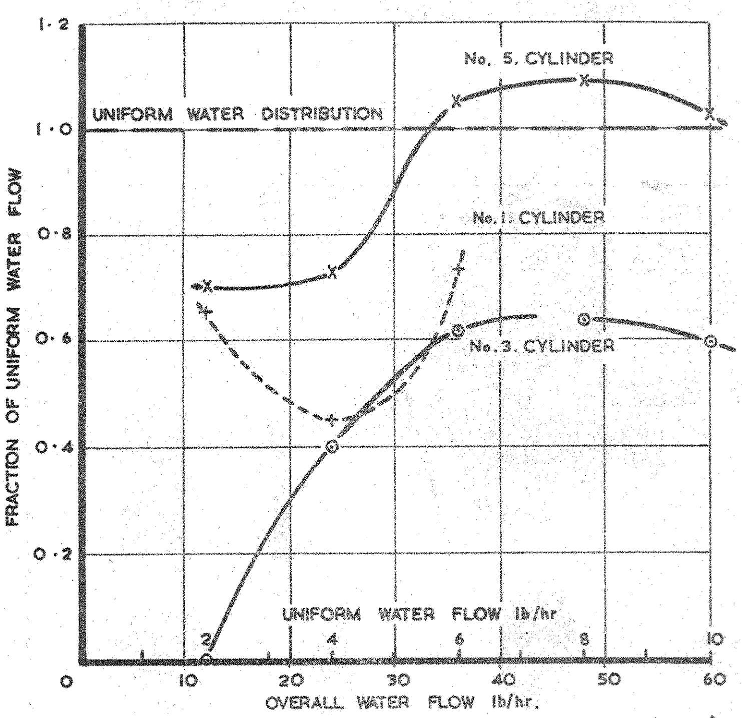


FIG. 15b. THE FEDDEN FLAT-SIX ENGINE EQUIPPED FOR WATER DISTRIBUTION TESTS



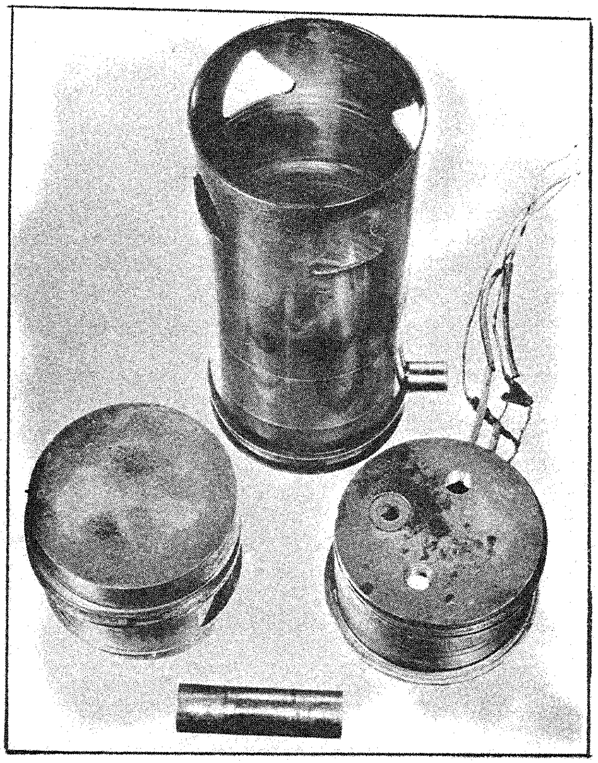
FEDDEN FLAT-SIX ENGINE CONDITIONS AS IN FIG. 5. (REF. 7.)

FIG. 16. VARIATION OF INDIVIDUAL CYLINDER TEMPERATURE WITH OVERALL WATER FLOW



FEDDEN FLAT-SIX ENGINE CONDITIONS AS IN FIG. 5. (REF. 7.)

FIG. 17. VARIATION OF WATER DISTRIBUTION WITH OVERALL WATER FLOW



FEDDEN SINGLE CYLINDER ENGINE

FIG. 18. ENGINE COMPONENTS AT CONCLUSION OF WATER-INJECTION TESTS.

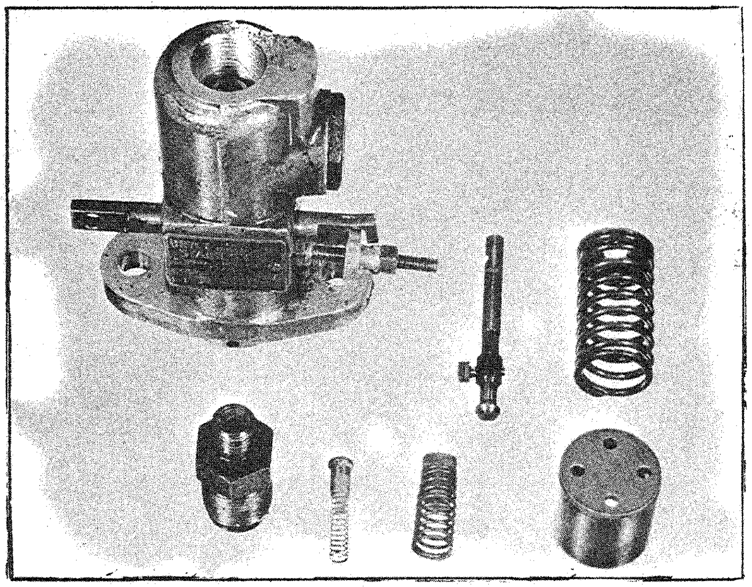


FIG. 19. FUEL-PUMP COMPONENTS AT CONCLUSION OF WATER INJECTION TESTS.

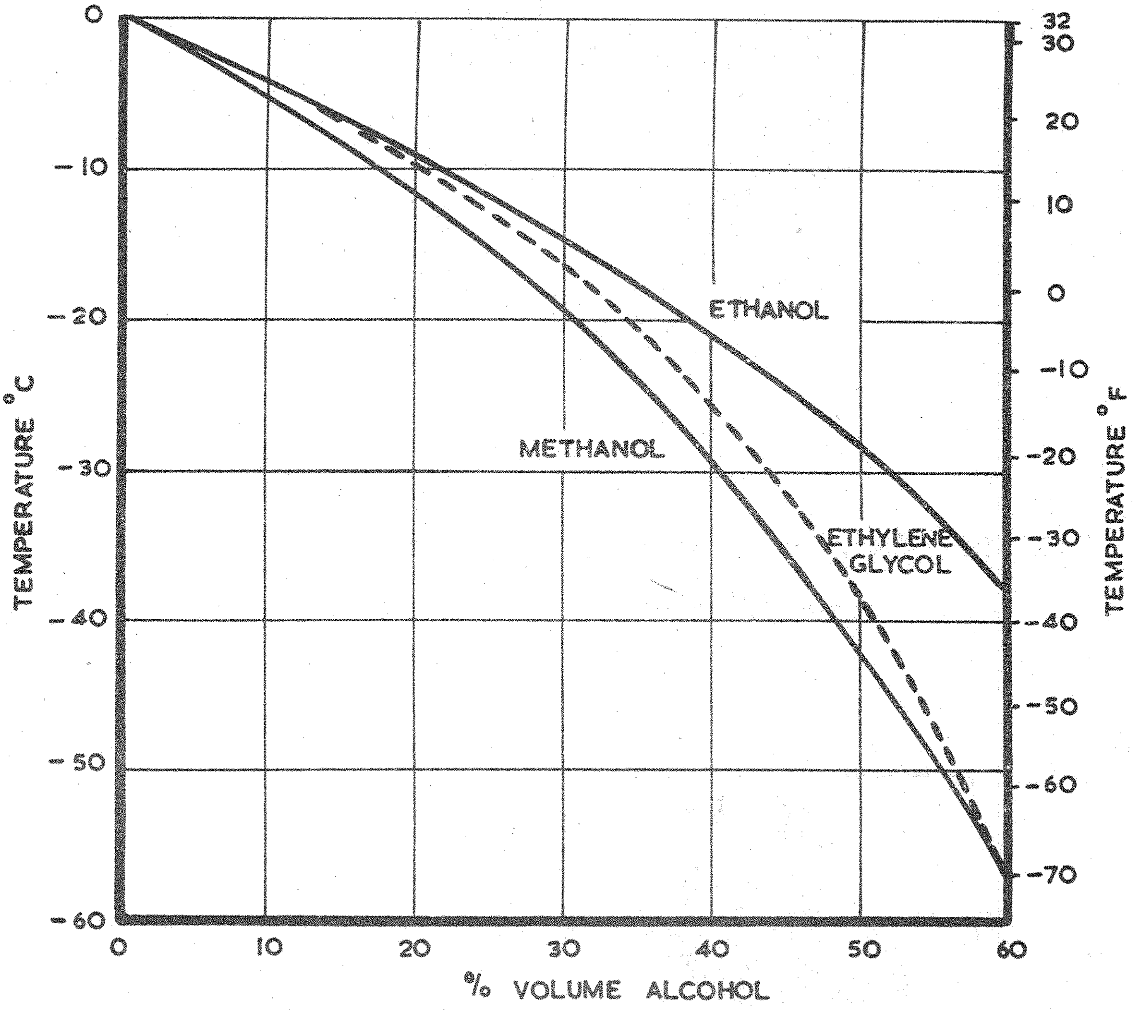


FIG. 20. FREEZING POINTS OF WATER ALCOHOL BLENDS