DIRECT WATER INJECTION COOLING FOR MILITARY ENGINES AND EFFECTS ON THE DIESEL CYCLE

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A study was conducted on the feasibility of totally cooling a single-cylinder diesel engine by direct injection of water into the combustion chamber. The term "total cooling" can be taken to mean stabilized cooling at all loads and speeds so as to eliminate need for conventional cooling jackets, cooling fins, or oil spray jets. The engine used was a CLR Direct Injection Diesel with 42.5 cubic inch displacement and a compression ratio of 16:1. Most of the running was at 1800 rpm and 92 psi IMEP.

Separate measurements were made of heat rejection to the cylinder head, liner, and crankcase oil to determine more accurately where the cooling effect was being applied. Water injection was by means of a Bosch pump and various pencil-type nozzles installed, adjacent to the fuel injector in the cylinder head. Port injection and port induction were also briefly investigated. A five-hole, 90° included angle nozzle was used, as was a three-hole, 30° included angle unit. For comparison, a nozzle directing one spray obliquely at the cylinder wall was also tested. Firing pressure was monitored using a piezo-electric transducer; both pressure-time and pressure-volume (indicator) records were obtained. In order to determine timing of both fuel and water injection, needle lift was monitored using a differential transformer pickup.

The results of this study indicate:

Optimum total engine cooling by direct water injection was accomplished over a wide range of water injection timings (from 450 to 720 CA degrees after TDC power stroke) at water/fuel ratios of 2.9 to 3.7 with output power and brake specific fuel consumption improved 5 to 20%, respectively, over that with the standard jacket-cooled CLR engine.

Emissions are affected in an expected manner by the presence of water: NO_x is decreased, sometimes substantially, while the other emissions (HC, CO) tend to increase.

When cooling the exhaust, the condensate becomes an effective scrubber of sulfur oxides. NO_x was not significantly reduced by scrubbing, but if the condensate is made sufficiently alkaline (pH >8), CO_2 was unintentionally scrubbed out.

The quality of the uncondensed exhaust for turbocharging is attractive. A theoretical gain of about 17.5% in available exhaust energy due to generation of steam was calculated, along with a temperature decrease of several hundred degrees Fahrenheit.

Water contamination of the lubricating oil varies from negligible to extreme, depending on injection quantity, timing, and spray pattern. By not directing water at the liner wall, and by keeping the oil above 212°F, one can maintain the oil in a dry condition.

Based on this work, several pertinent recommendations have been made: (1) utilize water injection for short-duration, very high-output operation which would otherwise be destructive due to thermal overload; (2) use water induction cooling in event of loss of conventional liquid coolant; (3) utilize exhaust scrubbing in stationary applications to permit burning of high-sulfur fuels without producing sulfur oxide emissions; nitrogen oxides could likewise be reduced by the injection of small amounts of water; and (4) since 2-stroke-cycle engines are an important category of diesel engines, some work similar to this effort should be done to this engine type; prospects are good for success, but conditions are apt to be more restrictive.

Introduction

Internal combustion engines employed in military vehicles are often subjected to very severe duty cycle under environmental extremes. Engines with high power-to-weight ratios are required for operation in the Arctic at −60°F and in the desert up to 125°F. One of the major problems, particularly for armored combat vehicles, is the extremely limited space in which to install the power package. Cooling airflow path is seldom ideal, affected by necessary use of armored grilles, brush screens, guards, and baffles to direct the air for satisfactory cooling. Military engines operate under conditions including radiator or heat exchanger damage, scaling and plugging, low oil and water level and deterioration which causes reduction of cooling system performance. Thermal stresses imposed on engine components and on the engine oil can result in reduced engine life or early failure.

Questions addressed in this study of total cooling by water injection feasibility are as follows:

(1) Can operation of the diesel engine be achieved using direct water injection for total cooling?

(2) What conditions of timing spray and quantity of water are required?

(3) Can direct water injection be used for total cooling with 100% recovery of cooling water?

(4) What is the impact of direct water injection on emissions, i.e., NO, NO_x, hydrocarbons, carbon monoxide, and oxides of sulfur?

(5) When the exhaust is cooled to recover the water, does the condensate function as a scrubber for some emissions, i.e., sulfur oxides, NO_x or CO?

(6) Does water injection cooling have potential for stationary engine applications?

(7) Can water injection be utilized for short duration under high-output operation which would otherwise be destructive due to thermal overload?

(8) What is the impact of water injection on lubricant quality, engine deposits and wear of engine parts?

For the foreseeable future, conservation and environmental pressures will cause all researchers to consider fuel availability, fuel economy and engine gaseous emissions. In addition to the reduction in nitrogen oxides (NO_x) reported by others,[1–4] the probability existed that oxides of sulfur (SO_x) might also be removed from engine exhaust during operation with water injection. If so, this could make fuels with relatively high sulfur content look attractive, at least, perhaps for stationary applications. Earlier work[2] conducted at our laboratory demonstrated the feasibility of totally cooling an air-cooled spark ignition engine by direct water injection. While the concept of injecting (or inducting) water into an internal combustion engine (SI or CI) is not new,[5–9] it is nevertheless considered that the idea of assuming the full burden of handling rejected heat via direct water injection and reaping the advantages is an approach which may contribute significantly to the potential of internal-combustion engines. In some cases, this is because of the new types of engines (high-speed and high-output) that have come along; in other cases it is because the problems facing the internal combustion engine designer are somewhat different from those of times past (nitrogen and sulfur oxides emissions, particularly, and the fuel situation).

The current work[10] was undertaken to determine feasibility of total cooling diesel engines by direct water injection and determine effects on diesel cycle operation and combustion. A schematic illustration of a proposed water-injection condensate recycle concept is shown in Fig. 1. One of the features of such an engine is the potential of providing for increased water injection with increased severity of cooling load. As illustrated by Fig. 1, liquid coolant is injected directly into the combustion chamber during compression, expansion, and/or immediately

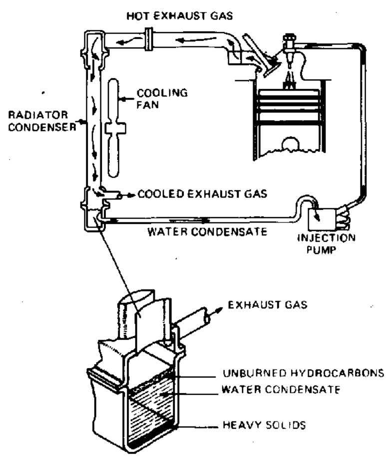

FIG. 1. Schematic illustration of water-injection condensate recycle concept.

following the "effective work" portion of the combustion cycle. When injected, the coolant absorbs heat from the combustion chamber, vaporizes, and is expelled during the exhaust cycle along with products of combustion. As the combustion and coolant products flow into the heat exchanger, the coolant condenses into a sump and the cooled exhaust gases are expelled into the air. Condensate from the sump is then recycled to the injection pump.

Theoretical Considerations

Early work[5] successfully applied direct water injection total cooling to slow-speed, stationary gas engines. In this work, water was sprayed at all surfaces exposed to hot gas near top center on the firing stroke. It was important that the water did not vaporize from the hot combustion gases, but rather, it was permitted to reach the hot surfaces from which it would remove heat by formation of steam. The injected water required was in close agreement with the normal engine heat rejection divided by the heat of vaporization water.

Utilizing the concept of vaporization at solid surfaces as the basis for cooling effectiveness, it is possible to examine some of the implications of that "model." In the absence of any known cogent model built to explain the direct effects of water injection on the combustion gases, the simplest analysis, based on the conservation of energy, can be applied. For example, assume a diesel engine has a thermal efficiency of 33.3%, and is operating with a fuel having a lower heating value (Q_S) of 18 000 Btu/lb. Distribution of supplied heat, Q_S, is normally divided equally between the exhaust gas (6000 Btu/lb), the jacket cooling load (6000 Btu/lb, including lubricating oil), and shaft work (6000 Btu/lb). Then theoretical water required (W_J) to pick up the jacket heat load, Q_J, at total cooling would be

$$W_J = Q_J/\Delta H = 0.333(18\,000)/1100$$

$$= 5.5 \text{ lb water/lb fuel}$$

where $\Delta H = 1100$ Btu/lb = average heat of vaporization for water between 200°F and 400°F.

A possible contribution of the current work to the theoretical understanding of water injection cooling is recognition of the effect of pressure in the combustion chamber on the cooling effect of water.[11] During a significant part of the diesel cycle (compression and power strokes), one cannot expect boiling film cooling to occur because ambient pressure exceeds saturation pressure. If typical metal temperature is, say, 300°F, then no boiling will occur when the pressure exceeds 67 psia. Above this pressure, water will simply wet the walls, warm up, and await the decline of cylinder pressure to the point that boiling can occur. Thus, vaporization cooling can occur only during a portion of the compression stroke and the latter part of the power stroke. The only benefit obtained from injection during the early power stroke comes from the fact that the piston is close to the head during that time and the injection targeting may therefore be easier. This part of the cycle (power stroke) has high pressure; high injection pressure is required in consequence, and there will be a tendency for excessively atomized water to vaporize in the gas phase, rather than at the walls. As a result, excessive amounts of cooling water will be needed, while the charge cooling will cause increased ignition delay or substantial power loss just after combustion. None of these effects is desirable.

A question may be asked whether additional work is produced by the generation of steam during the power stroke. It appears from analysis of this situation that the pressure gain from steam formation is about one-sixth of the pressure loss due to charge cooling; thus, any vaporization in the gas phase during the power stroke constitutes a pure loss.

The other way to get work is to have steam formed by vaporization of water at the walls; this steam is "free" and is potentially a source of increased power. A calculation indicates that one can expect a 3-psi increase in mean effective pressure (assuming about 75 psi IMEP) if it were possible to get steam all formed halfway through the power stroke. Recalling the difficulty previously explained where boiling is suppressed by firing pressure, it is not probable that more than half of the expansion stroke can be utilized. However, it will be shown that when fuel injection timing is advanced to compensate for increased ignition delay (during total-cooling water injection operation), the combined diesel cycle effects actually can produce a significant increase in power output.

Apparatus

The engine* used in this work was the standard CLR (with the standard direct-injection diesel conversion package) coupled to an eddy-current-type dynamometer. The fuel system used a

* Bore and stroke = 3.80 in. × 3.75 in. = 42.5 CID.

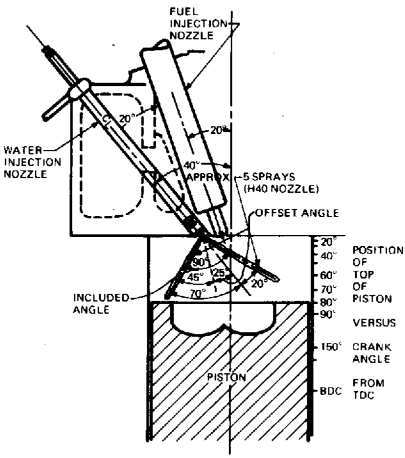

FIG. 2. Engine head and water injector location.

four-hole fuel injector (4×0.010-in. hole openings, equally spaced, 150° included angle, with 20° offset angle), having a 3000-psi crack pressure. A Bosch APE-1B fuel injection pump was used with a 6-mm delivery valve and 10-mm pump plunger. Fuel was supplied from an overhead burette with delivery rates taken in minutes for 100 ml of fuel consumed and reported in lb/min. The water-injection system used several slender, "pencil"-type (0.375 in. diam) nozzles manufactured to desired specifications. A Bosch unit pump (APE-1B) identical to the fuel injection pump was used to drive the water injector. The geometry of the water injectors used in this work is shown as follows:

Designation	Hole size, in.	Number of holes	Offset angle, deg	Included angle, deg
H40	0.012	5	25	90
L150, H150	0.010, 0.008	3, 4	30	30
"A"	0.051	1	0	0

The numbers in the designation represent the crank angle from TDC at which the spray is expected to be directed into the piston cup.

Figure 2 is a section through the cylinder head showing the location of fuel injector and water injector in relation to the piston cup. The exhaust system was modified as required to facilitate recovery, recycle, scrubbing, or analysis of the exhaust water, whatever the case being investigated.

Calorimetry System

When this program was still in the planning and setup stages, it was expected that total cooling might be difficult to accomplish and that some means of quantitatively determining the extent of "partial" cooling would contribute guidance to the difficult optimization process that was anticipated to be necessary to achieve total cooling. The logical indicator of partial cooling effectiveness is a calorimetric determination of absolute heat rejection. It was decided to obtain separate heat flux measurements for the head coolant, liner coolant, and lubricating oil. Separate systems were developed for each of these based on appropriate measurements of fluid flow rates, temperature changes, etc. so that rigorous heat balances could be made. Observed differences in heat-rejection measurements made during engine operation were found to be reasonable and repeatable. Diethylene glycol was used for coolant temperatures up to 230°F, and a vegetable oil was used at 325°F and above. Engine metal temperatures were monitored at the top of the cylinder liner and in the fire deck of the cylinder head.

Pressure and Timing Measurements

Pressure-time, pressure-volume, and fuel and water injection timing data were taken with conventional electro-mechanical devices and monitored on an oscilloscope.

Exhaust Emissions Measurements

In all cases, bag sampling was used for basic emissions measurements. The instruments used on this bag sample were as follows:

NO	Chemiluminescence analyzer
NO_x	Chemiluminescence analyzer
CO	NDIR
CO_2	NDIR
O_2	Polarigraphic-type analyzer
HC	Flame type ionization detector apparatus
Smoke	Bosch paper filter type with photo-optical reading unit

Since sulfur content is an important factor in the fuel industry and since sulfate showed up in the water recovered from the exhaust, it was decided to conduct exhaust scrubbing tests to remove oxides of sulfur (SO_x). Wet-chemical analysis was used since no direct reading instrument was available.

For a more thorough description of test apparatus, the reader is referred to Ref. 10.

Experimental Results and Discussion

Cooling Via Direct Water Injection

Utilizing the apparatus as previously described, the engine was operated under the following fixed conditions, unless specified otherwise:

Speed: 1800 rpm

Fuel rate: Set to maintain constant brake load of 19.2 lb with no water injection (5.2 Bhp at 1800 rpm) (Approx 0.056 lb/min). Fuel rate with water injection was also equal to 0.056 lb/min. Only fuel injection timing was varied and *not* fuel delivery rate. Timing advance required to maintain TDC as the end of the ignition delay period ranged from 6 to 18 CA degrees over that of the standard jacket-cooled engine.

Water injection

Rate: 0 to 0.40 lb/min

Timing: Varied by 90° CA increments

Duration: Varied from 30° to 90° CA degrees

Spray characteristics: Nozzles were chosen to provide spray angles which would be targeted for full spray entering the pistion cup at 40° to 150° from TDC utilizing cone angles of 30° and 90°. In addition, one single spray nozzle of 0° cone angle was used.

Since three different heatflow measurements were obtained and since the cooling effect was not equally divided among head, liner, and oil systems, their *sum* was used to define total cooling. When net heat rejection from the engine equaled zero total absolute cooling was achieved. In practice, net heat flow had to be computed after-the-fact and the total-cooling condition estimated by linear interpolation because all tests were conducted using predetermined water injection rates which might have been slightly more or less than that required for total cooling. Space does not permit discussion of the individual heat rejection data acquired for the cylinder head, cylinder liner, and sump. Figure 3 shows *total engine* heat rejection for the five-hole nozzle versus water/fuel ratio for the full range of injection timings. Total cooling water/fuel ratio is that ratio at which the net heat rejection is zero. Interestingly, the range in water/fuel ratio

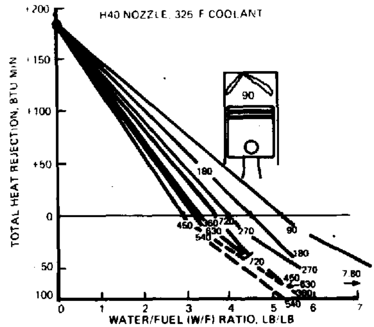

Fig. 3. Total Btu rejected versus water/fuel ratio—Numbers on curves are the end of water-injection timing, CA degrees from TDC power stroke.

is from about three to somewhat over five. Maximum water/fuel ratio of 5 is required for total cooling when injecting during the first 90 deg of the power stroke. A minimum water/fuel ratio of 2.9 results for injection during the last half of the intake stroke. In all probability, the former behavior results from unnecessary cooling of dense, hot gas (there is a marked power loss), while the latter is associated with spraying water directly onto the liner (which will be seen to cause a great deal of water to enter the crankcase).

Figure 4 shows the interpolated water/fuel ratios on a weight basis for total cooling plotted versus the approximate end of the water injection event in degrees ATC (power stroke) for the H40 and L150 water injection nozzles. Water injection timing has an important effect on required water/fuel ratio for total cooling. Total cooling is possible at *any* injection timing tried here. Since injection duration is a significant part of 90 deg, it may be said that no part of the cycle has been skipped, and further, that there is a slight sensitivity to water injection spray geometry.

Using the total cooling water/fuel ratio, Fig. 5 is obtained showing brake load and specific fuel consumption plotted against water injection timing. Note that the power output was obtained at constant fuel rate (0.056 lb/min), and therefore increases in engine brake power result in decreases in brake specific fuel consumption. Notice that the instant water is injected into the hot, dense gas (middle of power stroke) there is a drastic decrease in power and fuel economy. This demonstrates that injection

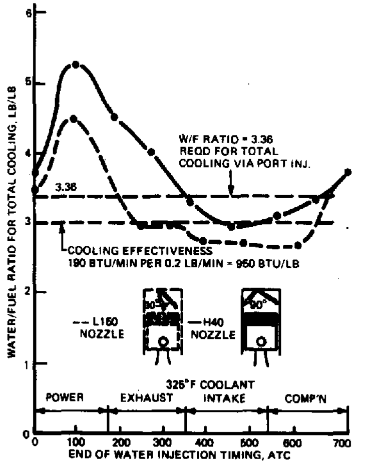

FIG. 4. Water/fuel ratio for total cooling versus water injection timing.

into the hot gases causes charge cooling and has a detrimental effect on the cycle. However, if water injection is confined to the balance of the cycle—exhaust, intake, or compression strokes—power and economy increases ranging from 5.2%

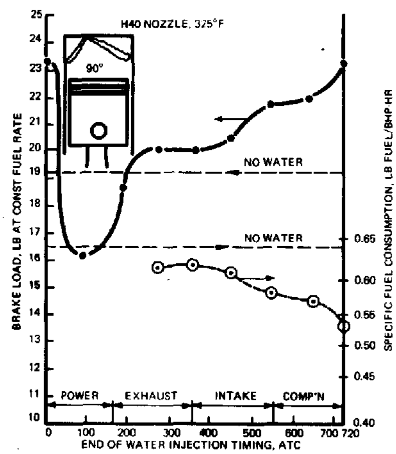

FIG. 5. Brake load and BSFC at total cooling (interpolation of Fig. 3) versus water injection timing.

for injection during the exhaust stroke to 20.8% for injection at the end of the compression stroke are realized, compared with the standard jacket-cooled engine. This shows that evaporative cooling of the metal surfaces is a more efficient process and produces beneficial effects in the cycle. Also note that the fuel consumption improves from 0.65 to 0.54 lb/Bhp/hr for the best water injection condition.

Figure 6 shows how water collected from the crankcase "blowby" (using a copper coil condenser) varies with water injection timing at the total-cooling water/fuel ratio. Since oil temperature exceeds 212°F, water bypassing the rings will not accumulate but will be evaporated and expelled. As can be seen, the quantity of water

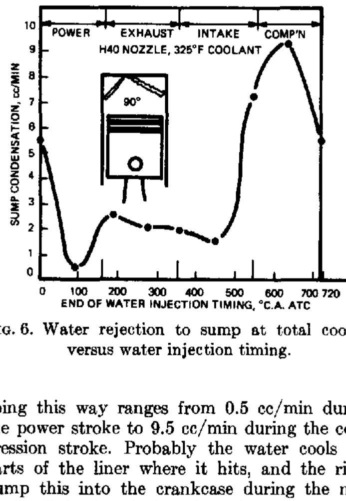

FIG. 6. Water rejection to sump at total cooling versus water injection timing.

going this way ranges from 0.5 cc/min during the power stroke to 9.5 cc/min during the compression stroke. Probably the water cools the parts of the liner where it hits, and the rings pump this into the crankcase during the next stroke. Injection during the exhaust stroke has less tendency to flood the crankcase than intake-compression injection. The 30° included angle nozzle (L150) produced essentially the same results (data not shown), which is somewhat remarkable, considering how much the two nozzles differ.

Cooling via port injection and port induction were also studied on a limited scale. Figure 4 and Table I show the water/fuel ratio of 3.36 (required for total cooling with port injection) superimposed over the direct injection water/fuel ratio requirements. In this instance, water was injected into the inlet port during the last of the compression stroke, and thus was all on the port walls by the time of the next intake stroke. It appears that port injection cooling is about equivalent to direct injection cooling but simpler to achieve for practical purposes. Of equal significance, there was a 10% increase in brake power and fuel economy, and less than 1 cc/min of water found its way to the engine sump.

The experiment with port induction was conducted to determine the amount of water the engine could tolerate. At a water/fuel ratio equal to 3.5, there was a 10% increase in power and fuel economy. Further increases in water input caused very rapid rates of pressure rise to occur at water/fuel ratio equal to 5, with power down 15% at water/fuel ratio equal to 6, and a 50% power loss at water/fuel ratio equal to 9. Neither calorimetry nor emissions were taken during this test.

Following tests to determine requirements for total cooling by water injection which were conducted with the calorimetry system, it was desired to demonstrate total cooling with direct water injection without coolant flow in the head and block. This was done and the results are shown in Fig. 7. During these tests, the water/fuel ratio was varied and the effects were determined. The data in Fig. 7 suggest that lower water/fuel ratios are possible as critical temperatures are being raised.

Table II presents a summary of the total-cooling picture with emphasis placed on the more

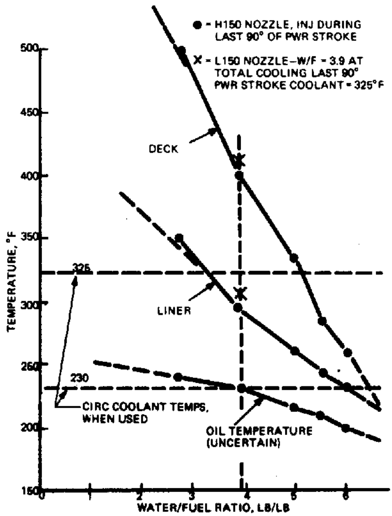

FIG. 7. Estimated equilibrium temperatures versus water/fuel ratio with head and liner coolant drained.

favorable or optimal times in the cycle for water injection. From this it is seen that optimum conditions range from 450 to 720 CA deg after TDC power stroke at water/fuel ratios of 2.9 to 3.7 with output power and fuel economy improved 5 to 20 percent, respectively, over that with the standard jacket-cooled CLR engine.

TABLE I

Port injection test summarized

Conditions:

1800 rpm
20 lb load without water
Water flow rate: 0.188 lb/Min
Fuel flow rate: 0.056 lb/Min
Water/fuel ratio: 3.36 lb/lb
Water injection during last of compression, into intake port
Circulating coolant and oil temperature: 325°F

Results:

Load: 22.0 lb with water, up 10%
Head heat flow: +6 Btu/Min
Liner heat flow: −3 Btu/Min
Sump heat flow: 0
Net heat flow: +3 Btu/Min, undercooled
Deck temperature: 456°F
Liner temperature: 349°F
Exhaust temperature: 570°F
Sump water flow: Less than 1 ml/min

Effects on the Diesel Cycle

The effects of direct water injection on the diesel work cycle were investigated at water- and fuel-injection timings believed likely to exhibit significant effects. For lack of space, the following discussion of cycle data concentrates on the most optimal time in the cycle for water injection—during late compression stroke—and summarizes the effects observed during water injection at two different timings during the power stroke. These conditions represent the extremes of strong combustion influence, i.e., charge cooling and no charge cooling effect:

(1) "TDC"—This puts water injection in the range of 85° to 30° BTC and therefore just prior to fuel injection. General results are a substantial increase in ignition delay, strongly

TABLE II

Optimum engine selection chart at total internal cooling conditions

End of water injection timing, °CA ATC power stroke	Percent increase in brake power and fuel economy	Water fuel ratio required	Sump water condensate cc/min
Direct injection			
450	5.2	2.9	1.6
540	12.5	3.1	7.2
630	13.5	3.3	9.5
720 (TDC)	20.8	3.7	5.5
Port injection 720 (TDC)	10	3.36	<1.0

effective cooling, and a power and BSFC gain, with this engine (Fig. 8).

(2) "90-deg ATC"—This involves injecting water into dense, hot gas over the range of 5° to 80° ATC and inevitably causes a substantial power loss. Much water is required for total cooling, and the increase in ignition delay is relatively small.

(3) "180-deg ATC"—This timing injects water during approximately the last 90 deg of the power stroke, where its presence is not felt so strongly. The total-cooling requirement is moderate, power loss is nil, and ignition delay increase is minor.

Figure 8 shows the situation relative to the no-water case for water injection late in the compression stroke, just preceding fuel injection. Shown in that figure are the whole indicator diagram, the magnified bottom of same, and the pressure-time situation (plus injection timings) in the vicinity of power stroke TDC. The full indicator diagram shows lower pressures to occur late in the compression stroke and early

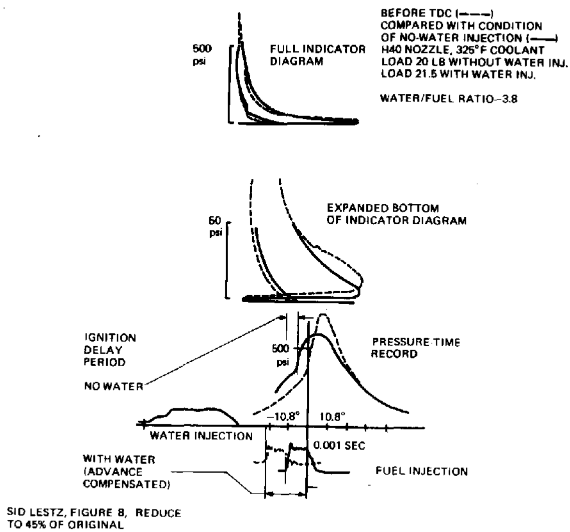

Fig. 8. Cycle data-water injection immediately prior to TDC.

TABLE III

Engine emissions with various water-injection timings

Date (1973)	NO, ppm	NO_x ppm	HC, ppm	CO, %	Bosch smoke No.	CO_2, %	O_2, %	Water injection nozzle type	Approximate water/fuel ratio	Comments
21 Nov.	450	560	610	0.80	—	8.5	9.0	None	0	Three runs average
26 Dec.	590	585	279	0.60	6.2	9.4	8.3	None	0	One run only
21 Nov.	75	115	700	0.41	—	7.1	12.0	A	4	Timing last part
26 Dec.	238	257	530	0.58	2.2	8.7	9.35	H150	3	Compression stroke
21 Nov.	115	160	753	1.01	—	7.6	11.0	A	4	Timing first part,
26 Dec.	377	403	194	0.30	5.8	8.9	9.15	H150	3	Power stroke
21 Nov.	200	240	710	1.26	—	9.6	7.0	A	4	Timing last part,
26 Dec.	412	428	182	0.54	6.4	9.35	8.1	H150	3	Power stroke
					5.2	Timing	360°	H150	3	(Smoke data only)
					6.2	Timing	450°	H150	3	(Smoke data only)
					6.9	Timing	540°	H150	3	(Smoke data only)

1800 rpm, 20-lb load, fuel rate about 0.05 lb/min temperature 325°F for oil and coolant (vegetable oil).

in the expansion stroke with some slight gain in peak pressure late in the expansion stroke. The indicator and pressure-time (P–t) diagram show a net increase in cycle work as a result of the following:

(1) Reduced compression work due to ongoing cooling of the working fluid. This cooling, however, also reduces early expansion work by a similar amount.

(2) Increase in end-of-cycle (late power stroke) work due to flash evaporation of water from the hot metal surfaces. Evaporation is suppressed until cycle pressure falls below the water saturation pressure corresponding to the cylinder metal temperature.

(3) The P–t record shows conditions near-TDC which are not evident from the indicator diagram. We see the reduced compression pressure noted above. Also, the increased firing pressure just after TDC augments cycle work. Ignition delay was increased by water injection, but was compensated for by advancing fuel injection timing. The ignition delay is attributed to reduced compression temperature as well as possible chemical effects. It is considered that with this engine, the increased ignition delay with water injection improves fuel mixing and combustion, accounting for a substantial part of the observed power increase.

Volumetric Efficiency

The effect of water injection (using the 5-hole H40, 3-hole L150, and single hole "A" nozzles) and port induction on volumetric efficiency was studied. In general, it was observed that water input caused a slight but sustained increase or decrease (2–3%) in volumetric efficiency. Increases occur due to charge and engine metal cooling, while decreases are due to the formation of steam on the intake stroke. However, in this study significant *transient effects* were observed in that commencement of water input often produced a much greater reduction in volumetric efficiency (10–12%) and cessation of water input typically produced a temporary boost (a few percent) in volumetric efficiency.

The Engine Exhaust

Emissions. Table III shows the exhaust emissions produced at standard speed and load by the CLR engine without or with water injection for three water-injection timings and two water-injection nozzles. Emissions data were not taken at any other times in the cycle. The following comments can be made about the data:

(1) The basic engine exhaust contains a great deal of oxygen; the volumetric efficiency is low and so is the utilization of what air is taken in. Smoke is high, as is CO.

(2) Injection of water during the last part of the compression stroke seems to cut NO_x substantially, quite in keeping with expectations. In one case shown, smoke declined sharply while CO was not changed. Why oxygen should increase is not known.

(3) Injection of water during the first part of the power stroke (into the hot high-pressure gas) produces a fair effect on NO_x. Whether this is due to water carry-over into the combustion phase is not known for certain, of course. Hydrocarbons went up for one water injector and down for another, while smoke was unaffected. The decrease in CO is unexplained; the increase is more expected.

(4) Injection of water late in the power stroke reduces NO_x while the effect on hydrocarbons parallels that of the earlier timing. Smoke was unaffected.

Direct water injection has the greatest effect on combustion when injected during the late compression stroke. This is partially due to the increase in ignition delay caused by charge cooling, allowing greater mixing *prior* to ignition. Also, the resulting combustion is influenced by the large amount of water present. For the other two water injection timings, the water is introduced after the combustion event, and therefore, does not affect the course of combustion. However, carry-over effects are likely to affect emissions. It appears both experimentally and theoretically that parts remain wet during the entire cycle, and that water vapor can get into the intake charge and chemically influence combustion rather than affecting it by cooling the compressed charge and causing increased delay as described above.

It is worth noting that the changes in smoke and CO were independent, instead of being closely linked as normally expected. And finally, engine emissions values were reasonable, allowing for poor volumetric efficiency and air utilization of the CLR engine.

Engine emissions as affected by injection of aqueous solutions. Several aqueous solutions were directly injected into the engine combustion chamber to compare their effects on exhaust emissions with the injection of plain water; results are in Ref. 10.

Exhaust scrubbing. An extensive study of exhaust gas scrubbing (after the exhaust port) was conducted, but space permits reporting only the highlights.

(1) None of the solutions attempted scrubbed NO and NO_x significantly. These solutions included sodium bicarbonate solution, sodium carbonate solution, strong caustic soda solution, weak peroxide solution, and neutral and acid solutions. The problem is that NO must be oxidized to NO_2 before it can hydrolyze, and even that process is inefficient.

(2) Scrubbing of exhaust sulfur seems to be very effective regardless of acidity or alkalinity of the scrubbing solution, although only about

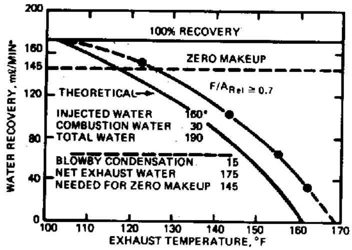

FIG. 9. Exhaust water recovery rate versus exhaust temperature (single experiment).

45% of theoretical sulfur has been found in the exhaust by wet chemical methods. It is possible the remainder of the sulfur was combined with particulate carbon in the exhaust smoke, and being in the form of inert carbon-sulfur complexes, CS_x (Refs. 12, 13, 14, and 15), it could not be scrubbed out with the gaseous sulfur compounds. Indications are that sulfur oxides will indeed be scrubbed out in almost any wet system, although our work did not result in a complete sulfur balance.

Other exhaust studies. Exhaust water recovery and the aspects of turbocharging with direct water injection were also examined. Again, space does not permit discussion of these, and the reader is referred to Ref. 10 for these details, which conclude that:

(1) Water can be recovered quantitatively from the exhaust in agreement with humidity table predictions; the exhaust must be cooled nearly to 100°F at atmospheric pressure for 100% recovery. This is shown graphically and compared with the theoretical case in Fig. 9.

(2) Turbocharged engine calculations show a theoretical gain of about 17.5% in available exhaust energy due to generation of steam; exhaust temperature will be reduced several hundred degrees at the same time.

Conclusions

1. Direct water injection cooling of diesel engines can be accomplished with increased power and better BSFC. Optimum total engine cooling by direct water injection was accomplished over a wide range of water injection timings (from 450 to 720 CA deg after TDC power stroke) at water/fuel ratios of 2.9 to 3.7 with output power and brake specific fuel consumption improved 5 to 20%, respectively, over that with the standard jacket-cooled CLR engine.

2. Total cooling of military diesel engines for ground mobile application is not feasible due to the required condensing temperature for full water recovery (approximately 100°F at atmospheric pressure). The totally cooled water injection engine is impractical as a military engine that must operate on a self-sustaining basis.

3. Emissions are affected in an expected manner by the presence of injected water: NO_x is decreased, while HC and CO tend to increase.

4. When the exhaust is cooled to recover the water, the condensate functions as a scrubber for some emissions. Sulfur oxides are effectively scrubbed, while NO_x and CO are not removed from the exhaust.

5. The possibility of control of sulfur oxides may justify use of a wet exhaust system for stationary applications to permit high-sulfur fuels in the future.

6. Short-duration direct-water injection (a more traditional role for water injection) may have potential use for spurt power or overload operation of military vehicles without water recovery.

7. Long-range effects of direct water injection on engine lubricant quality or hardware durability were not determined, but results show that adverse effects are minimized if excessive water application to the cylinder wall is avoided. It is also important to keep the oil above 212°F to maintain the engine oil in a dry condition.

Acknowledgments

This work was sponsored by the U.S. Army, Contract DAAD05-72-C-0053, under technical management of E. J. Rambie, USA TACOM. Contract monitor was C. F. Schwarz, USA MERDC. The authors also thank E. R. Lyons who provided the laboratory assistance, and other members of the U.S. Army Fuels and Lubricants Research Laboratory who provided many helpful suggestions.

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