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930683

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International Congress and Exposition
Detroit, Michigan
March 1 - 5, 1993

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ABSTRACT

A spark-assisted, direct-injected family of small rotary engines with a rotor displacement of 400cc was designed, manufactured and optimized for the use of heavy fuels, especially diesel fuel.

The development focused on the modification of an existing gasoline rotary engine to accommodate heavy fuel mixture formation and combustion. Off-the-shelf, low-cost fuel injection, and ignition systems were modified to match the peculiarities of the Wankel engine.

The performance optimization work was of an experimental nature. A multitude of factors which govern the air-fuel mixture ignition and combustion in the small, spark assisted, direct-injected rotary engines employing a single injector and an open combustion chamber were explored.

Satisfactory results were obtained for a defined speed range at various loads. Specific fuel consumption lower than 330 g/kW-hr (0.55 lb/hp-hr) was obtained at various selected regimes with power levels between 5-12 kW and speeds between 3000-4000 RPM.

In addition to diesel fuel, good results were obtained on a variety of jet fuels.

While the single-injector, open combustion chamber arrangement developed for this application did not yet match the efficiency and operating flexibility already obtained in medium and large dual-injector, stratified-charge Wankel engines, it offers a convenient solution for small-displacement rotary engines where the space available for the combustion chamber is limited and simplicity and cost play important roles.

Future work will focus on improving the engine power density, fuel consumption and flexibility of operation in a broader range of speeds and loads.

INTRODUCTION

A fuel flexible engine, which is an engine able to operate on a broad variety of distillate fuels, has been a very attractive alternative to the high cost of producing fuels with stringent composition requirements (cetane or octane numbers) and costly engine design features such as variable compression ratio. Heavy fuels, especially diesel fuel, are of particular interest for a variety of stationary and mobile applications.

The stratified-charge rotary engine has shown the potential for achieving this goal. Good performance has been obtained in medium and large rotary engines with rotor displacement greater than 500cc using a high-pressure dual-injector configuration with pilot-main injection sequence [1, 2]*. The system assures efficient operation for a broad range of engine loads and speeds.

The dual-injector, sequential injection system arrangement is, however, not suitable for a small rotary engine due to space limitation. For the purpose of this discussion, a small rotary engine is considered to be an engine with rotor displacement less than 500cc and a generating radius smaller than 100mm. Cost also plays an important role in the fuel system selection process. The single-injector two-stage injection system recently evaluated could represent a good solution for the fuel injection systems of small rotary engines if the system reaches off-the-shelf status.

Because of the high cost of developing a custom-made injection system and of the difficulty of securing manufacturer support for unglamorous production projections, the simplest approach is to rely on off-the-shelf components.

The single-injector, high-pressure injection system with spark-assisted combustion could be a good

* Numbers in brackets designate references at end of paper

alternative, specially for application with a pre-established operating regime (load and speed).

Previous investigations shed some light on the air and fuel motion, mixture formation and combustion of the direct-injected heavy fuel rotary engine [3]. These studies have been conducted on engines with a large rotor displacement and divided combustion chamber.

Work performed recently added valuable information on the fuel-air mixing process in a medium-size, direct-injection rotary engine with open chamber [4,5]. The engine having a generating radius of 105mm was motored at speeds up to 2000 RPM. Jet-type fuel was injected through a single hole nozzle at high injection pressure.

For small rotary engines operating on diesel fuel at high speed (3000 to 6000 RPM) and employing a multi-hole nozzle pattern, this type of study is not available. The extrapolation of data obtained from medium and large rotary engines has to be carefully analyzed, since the fuel-air dynamic in the small engine can be quite different. For example, for a typical high-pressure injection system, fuel jet's contact with the combustion chamber walls could not be avoided in practice in the case of a small rotary engine.

Some of the specifics of the air-fuel motion and mixture formation in a high pressure, fuel injected, small displacement rotary engine are as follows:

- the small combustion chamber, which is partially contained in the rotor recess, is moving with respect to the rotor housing which hosts the injector and the spark plug, positioned much closer one to another than in the case of the large engine. In this case the local fuel to air ratio around the spark plug is much more dependent on the engine load (quantity of fuel injected) at a constant speed. In addition, the fuel spray blast on the spark plug electrodes at close range can affect spark plug life.

- the air swirl induced during the intake is substantially extinguished during the compression process. This observation may not be valid for the medium and large rotary engines, but was observed in small displacement engines at Wankel R & D Laboratories by motored engine visualization using colored gases. As rotor approaches TC (Top Center) position, the induced swirl is extinguished.

- the engine speed (rotor speed) plays a more distinct role in the air-fuel motion in the small displacement engines because of confined spaces available for the combustion chamber.

- high mixture velocity induced by the rotor motion and the shape of the combustion chamber is drastically changed by the transfer between combustion chamber leading and trailing pockets. This transfer alters the entire spectrum of air-fuel mixture speed and local mixture composition.

- spark plug characteristics (gap geometry, energy, orientation, position relative to the spray and spark timing) play a more significant role in the combustion process. The spark plug electrode's orientation relative to the fuel spray is important for mixture ignitability.

- the injector spray pattern influenced by the engine load and speed also plays a major role in combustion stability. At the same speed, the fuel-to-air ratio varies largely with the engine load. The inability to reach the proper mixture composition in the immediate vicinity of the spark plug electrodes at the ignition time at low load causes misfiring which has a negative impact on engine emissions, efficiency and durability. In the case of a dual injector system used in larger rotary engines, the pilot injection event is much less dependent of the engine load.

In the absence of in situ visualization of the air and fuel movement, mixture formation and combustion conditions for small, high-speed rotary engines, experimentation is the best method for determining the optimization of these factors in order to achieve efficient combustion process.

EARLY EXPERIMENTS had been conducted at Wankel R&D on a 530cc single-rotor engine. The engine was fitted with an in-line, high-pressure mechanical fuel injection pump. The experiments have offered a first assessment of the impact of various injection and ignition parameters on the combustion process, and the feasibility of using various off-the-shelf hardware for the application.

Various parameters, such as injector nozzle geometry and position, spark plug position, combustion recess geometry, injection and ignition characteristics have been studied. Figure 1 illustrates the injector and spark plug positions tried out during the experiments. Different small size, off-the-shelf injectors and spark plugs with different electrode configurations have been evaluated always in the single injector and single spark plug arrangement. Comparative tests have been conducted for a specific spray configuration oriented in the direction of the rotor movement or in the opposite direction. The influence of the spark plug recess, acting as a combustion prechamber on the mixture ignition was evaluated.

Figure 2 shows the influence of different injector opening pressure values on the engine maximum obtainable power for a particular hardware arrangement. In order to properly assess this parameter's influence on the engine performance, all other related geometrical

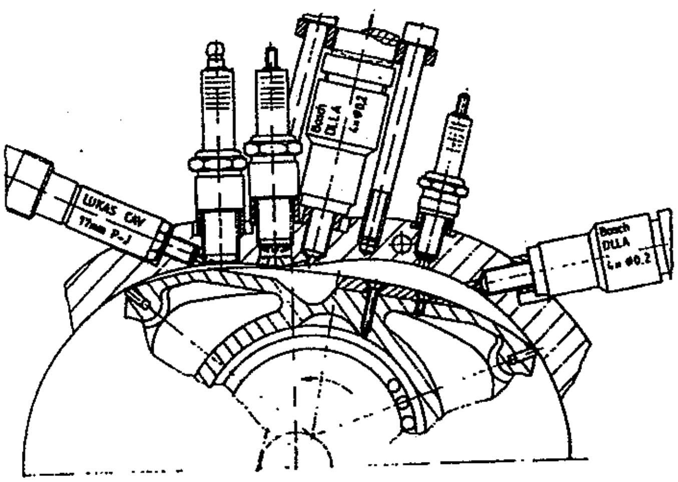


Figure 1 - Various injector and spark plugs locations tested on a small rotary engine

and functional characteristics were kept unchanged. It was found that a certain injector opening pressure yields optimum results for a given combustion chamber configuration and interrelated injection and ignition characteristics.

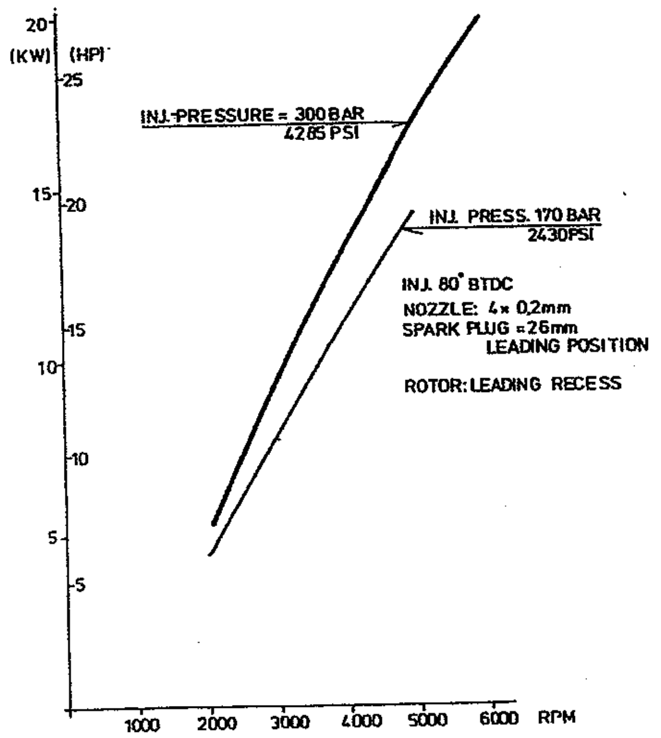


Figure 2 - The influence of the injector opening pressure on engine power performance

ENGINE DESIGN

An existing family of rotary engines [6] developed in the single-rotor and twin-rotor versions for gasoline applications was modified for diesel fuel applications. Two types of high-pressure injection systems have been evaluated for use in conjunction with heavy fuels; a mechanical injection system and an electronically controlled injection system.

Figure 3 represents the layout concept of the LCCR 400 SD engine; the single-rotor, charge-cooled, 400cc displacement, diesel fuel engine. Table I summarizes the engine main characteristics. The layout concept of the LCCR 800 TD engine, the twin-rotor version of the diesel fueled family of engines, is shown in Figure 4.

Figure 5 shows the actual LCCR 400 SD engine. The LCCR 800 SD engine is shown in Figure 6.

In order to accommodate the mechanical injection system, a camshaft and its driving gears have been incorporated in the accessories housing. The camshaft is

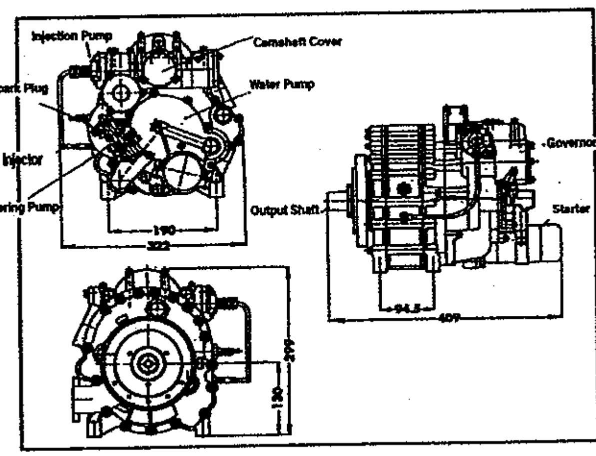


Figure 3 - LCCR 400 SD single rotor engine, layout and dimensions

driven by the engine crankshaft at 1:1 ratio through a pair of gears. The individual single-plunger injection pump is imbedded in the accessories housing.

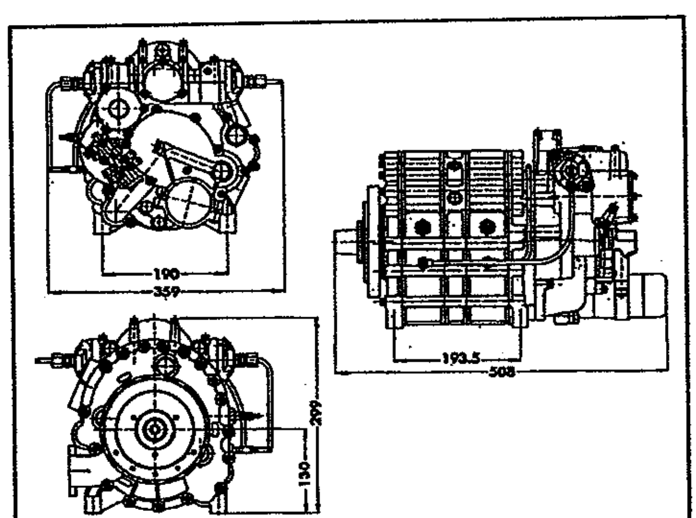


Figure 4 - LCCR 800 TD twin engine, layout and dimensions

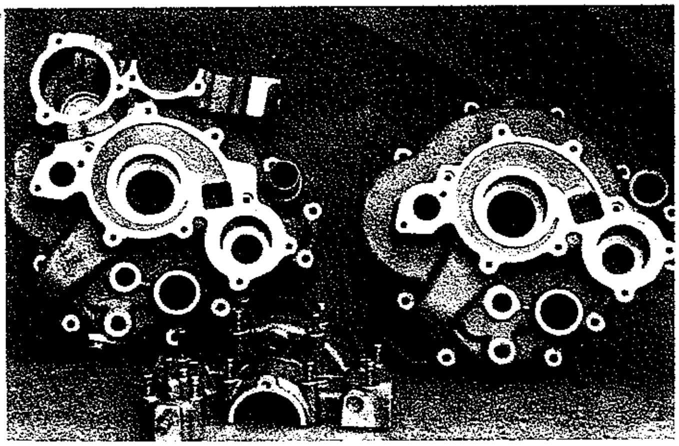


Figure 7 - Accessories housings for the 400 series of engines;
Left side: accessories housing and camshaft assembly cover for heavy fuel engine
Right side: accessories housing for gasoline engine

counterpart, the rotor has also been redesigned. The thin wall, sintered casting used for gasoline engines has been modified in order to allow a much higher design flexibility in the rotor recess shape, and its structure has been reinforced in order to withstand the higher combustion pressure, the steeper $\Delta p/\Delta t$ rate and higher heat release always associated with the heavy fuel combustion process. Reinforcement was added to the rotor bearing hub structure. The rotor wall structure was modified in order to facilitate the increase in heat absorption by the rotor and its transfer to engine housings and crankshaft.

The rotor and crankshaft bearing structures and lubrication have been modified to support the diesel fuel application. In the case of a diesel fuel rotary engine, the rotor bearing cooling and lubrication are critical. The single and twin rotor diesel fueled engines are unthrottled, but the fresh charge passing the rotor in this case contains only air without the beneficial cooling effect of the fuel vaporization in the fuel-air mixture as in most charge-cooled rotor, gasoline rotary engines.

The special engine cooling circuit employing a hollow crankshaft used in the LCCR concept [6] proved to be very helpful for the heavy fuel application in which the rotor and crankshaft bearing have to withstand higher mechanical and thermal loads. In the LCCR solution (Liquid and Charge Cooled Rotor) the hollow engine shaft accommodates a liquid cooling circuit running in parallel with the housings cooling circuit. The shaft cooling reduces the thermal load of the rotor bearing.

Figure 8 illustrates the structural differences of the rotor and crankshaft between the heavy fuel engine (left hand side) and gasoline engine (right hand side).

Rotor and crankshaft bearing lubrication by oil dispersed in the charge mixture can not be applied in an unthrottled, diesel-fueled rotary engine. In the absence of fuel droplets to carry the oil to the lubrication points, the oil pulverization in the air flow would require a high-pressure metering pump.

Instead, the lubrication system was modified in order to deliver the lubrication oil directly to the critical points under a slight pressure created by the oil metering pump. Figure 9 shows a back view of the engine with accessories housing identical for the single and twin rotor engines. The accessories housing contains the injection pump, water pump, starter motor, oil metering pump, and the mechanical governor. The four metering pump outlets distribute the oil to the shaft and rotor bearings and rotor seals. The diesel fueled engine crankshaft shown in Figure 8 on the left hand side incorporates an oil supply ring fed by an oil channel in the side housing. One of the metering pump outlets delivers a metered amount of oil to this channel. The second outlet supplies oil to the rotor housing for the apex seals lubrication and the remaining two outlets lubricate both side housing surfaces on which the rotor side seals coast.

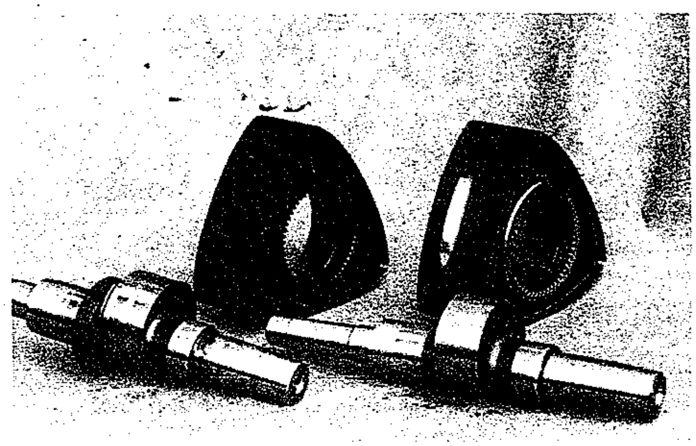


Figure 8 - Rotors and crankshafts for the single rotor, heavy fuel (left) and gasoline engines (right)

The same arrangement is used for the lubrication of the twin rotor engine but in this case each oil metering pump outlet feeds a split line connected for a total of eight lubrication points.

HIGH PRESSURE MECHANICAL INJECTION SYSTEM - The mechanical injection system for the single rotor engine shown in Figure 10 is comprised of an off-the-shelf integrated pump with non-return and relief valves, and an automotive type injector with a multi-hole nozzle. The injection pump plunger is driven by a roller tappet from the camshaft.

A mechanical governor of centrifugal force-load type, also driven at engine speed, controls the injection pump rack.

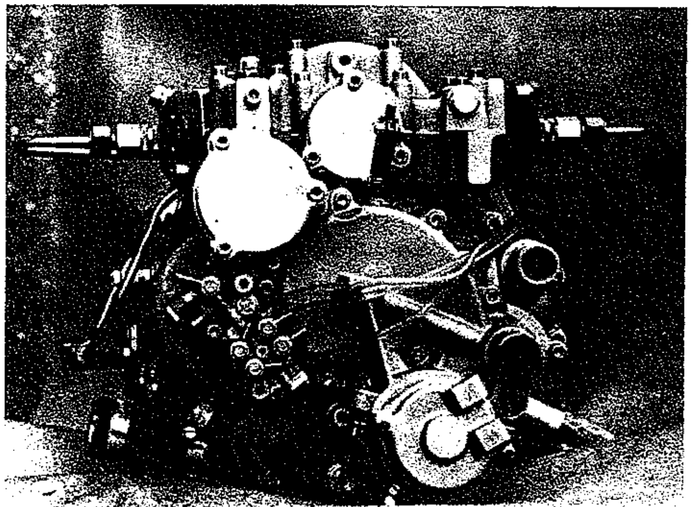


Figure 9 - View of accessories housing for the heavy fuel and gasoline 400 series engines

In the case of a rotary engine there is one injection per shaft revolution; therefore the pump plunger operates at twice the frequency when compared to the four-stroke piston engine. Special attention has been given to the injection cam profile, since the regular cams designed for four-stroke piston engines are operating at speeds below 3000 RPM.

Specific cam profiles have been developed for the rotary engine application. Two cam profiles, modified eccentric and tangential, were selected for actual engine testing. Both profiles were designed to produce faster injection pressure rise times and shorter injection events compared to regular cams.

HIGH PRESSURE ELECTRONIC INJECTION SYSTEM - The electronic injection system selected as an alternative to the mechanical system is a customized rotary engine version of the Servojet fuel injection system [7].

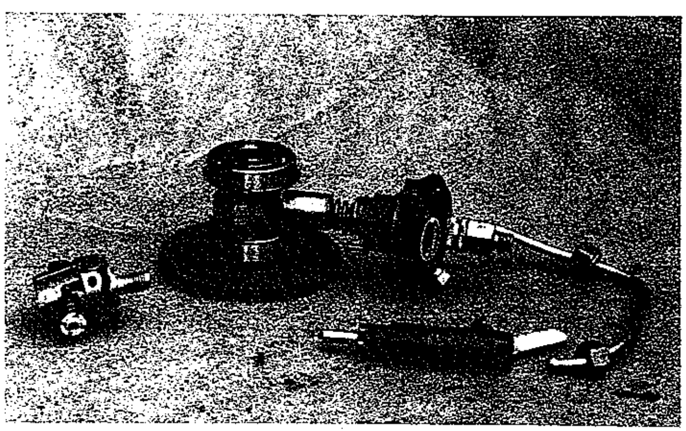


Figure 10 - Mechanical fuel injection system for the single rotor engine

The system depicted in Figure 11 is based on a common rail fuel supply with electronic control of the fue injection. By utilizing a high-pressure accumulator injecto and a large portion of the engine cycle to charge thi: accumulator, dynamic loads, acceleration and componen sizes are decreased allowing high injection pressures an very fast, repeatable response. The accumulator nozzl allows the injection energy to be stored and released in a very short interval. A laboratory version of this system wit the capability of adjusting incrementally the injectio timing, injection pressure, injection duration and ignitio timing was developed. The influence of variou parameters of injection and ignition systems on the engin combustion efficiency was studied.

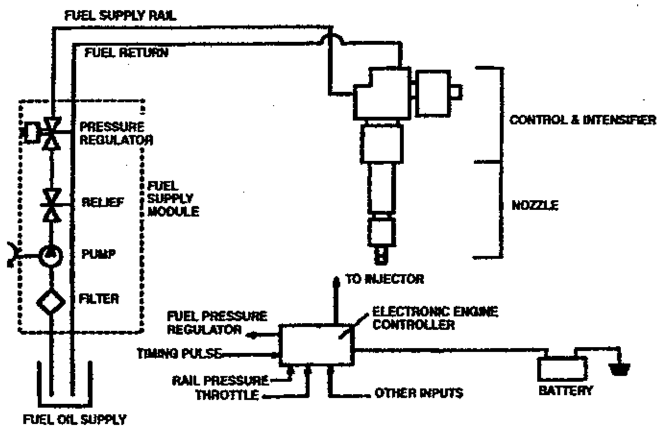


Figure 11 - Electronic fuel injection system for the single rotor engine

RESULTS

The development work, mainly of experimental nature focused on the injection and ignition systems, combustio optimization and development of necessary test rigs t support this work.

NOZZLE SPRAY STRUCTURE VISUALIZATION - Figure 12 shows a nozzle spray visualization rig with rapi photography capability. This installation was used for th visualization of the spray structure and its penetration i respect to the combustion chamber for various nozzle hol geometries and injector opening pressures. A "mule" roto housing was employed to assess the fluid motion relativ to the rotor housing and its timed evolution with respect t the injector and spark plug location. Figure 13-a and 13-b represent the spray structure evolution for 1 ms tim interval. By correlating the rotor position at differen degree of shaft rotation with the above images, a evaluation of spray impingement and deflection on th combustion chamber's walls was made, allowing selectio of the arrangements with minimum unwanted effect.

ENGINE WORK CYCLE - In order to measure, recor and analyze the working agent pressure evolutio throughout the engine cycle, an engine was instrumented with multiple piezoelectric transducers. The arrangement made possible the continuous recording of the gas pressure in all the engine chambers. A minicomputer based data acquisition system able to acquire, store and display various engine signals was developed. A data acquisition board assures the compatibility of signals between various transducers, their amplifiers and the time base. Data acquisition and post-processing software was customized for small rotary engine requirements.

HIGH SPEED FUEL INJECTION OPTIMIZATION - In a regular homogeneous spark-ignited engine, a highly volatile fuel such as gasoline is mixed with air before

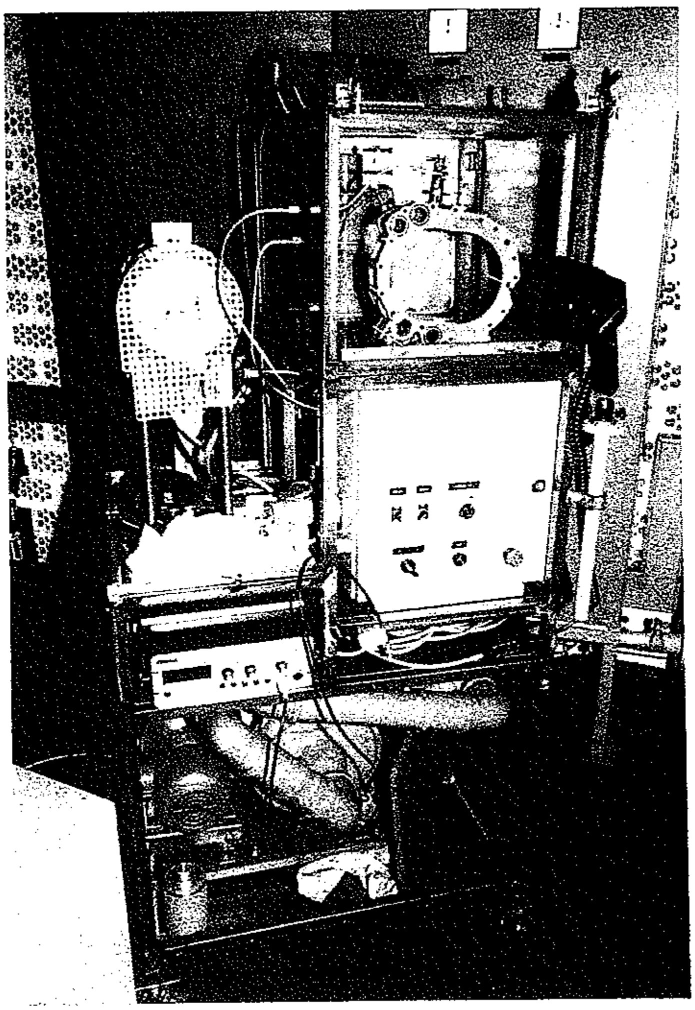
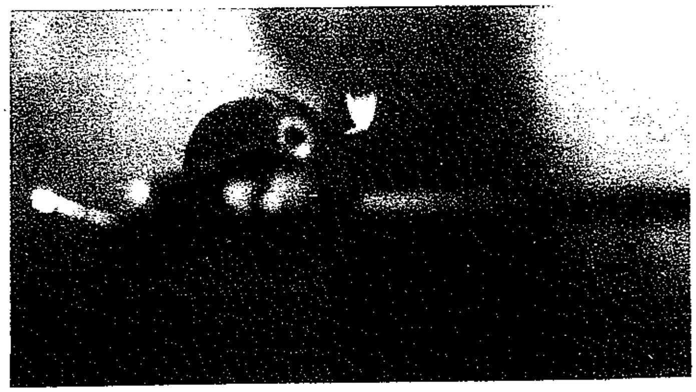
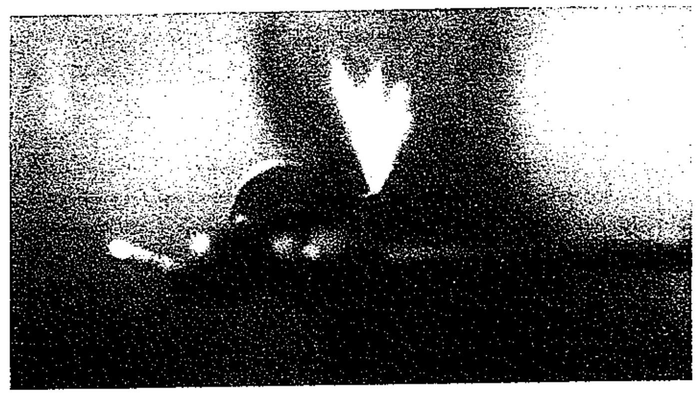


Figure 12 - Fuel spray visualization rig

entering the combustion chamber. Heavy fuels, a generic name used for various diesel and jet fuels, have much lower volatility and knock tolerances than gasoline. This means that fuel cannot be mixed with air before the compression stroke without risking detonation. Instead, some sort of injection system must be used to atomize fuel directly in the engine's combustion chamber.



[a]



[b]

Figure 13 - Fuel spray timed evolution

The method employed to insure adequate fuel/air mixing, and subsequent spark ignition, was to use high pressure to atomize the fuel in the immediate vicinity of the spark plug. A predetermined amount of fuel was injected and substantial fuel vaporization and mixing took place before being ignited by the spark plug. As a practical matter, this means that the injection event, fuel atomization and mixing, spark ignition, and initial combustion all must occur in a very compressed time period.

__Combustion Efficiency vs. Injection Duration__ - It was found that in order to obtain efficient engine operation, the required volume of fuel has to be injected in a certain time period. If the fuel injection event is too long, and fuel is still being injected well after combustion has started, the fuel spray contributes to a quenching effect which in the case of small rotary engine, is more predominant because of the small space available for the combustion chamber. This quenching extinguishes the flame front in the vicinity of the injection nozzle and leads to incomplete combustion, and unburned fuel is transferred to the exhaust. Obviously, combustion that takes place in the exhaust stroke doesn't contribute to the engine's output or fuel efficiency.

Ideally, the injection event has been completed before the rotor reaches TC and the spark ignites the air-fuel mixture. This insures that any cooling caused by fuel vaporization will be complete before combustion begins and that any tendency to quench the flame front will be minimized. Independent of an injection event, the combustion chamber pressure (and temperature) should show a steady rise as the rotor approaches TC. This is desired with the injection also but was not always the situation as will be shown.

Injection Duration - Elapsed Time vs. Crank Angle - As injection system development progressed, it became apparent that generating a short injection pulse in terms of elapsed time was not the only challenge. There are at least two other important factors that complicate the effort to achieve optimum injection system performance at high rpm: 1) the varying duration of the injection event in terms of crank angle as a function of RPM, and 2) a shift in injection start timing caused by fluid effects at high RPM.

Of these two factors, the first has greater operational impact. As engine RPM increases, the amount of rotation that occurs during a fixed time period increases. Therefore, for a fixed injection duration time, the relative rotor placement at the start and at the end of the event changes as a function of RPM. Consider the injection pressure and injector needle position traces illustrated in Figure 14. Note that although the overall duration of the injection event remains nearly constant in terms of time elapsed (approximately 1.5 ms), the effective duration as a function of crank angle doubles as engine speed increases from 3000 to 6000 RPM (from 27 to 54 degrees of crank angle).

At slow speeds (below 3000 RPM) and high load, the start of the injection event can be delayed until as late as 20 degrees BTC to avoid auto ignition. The injection event will then be complete before spark ignition around TC. But as RPM increases, and the effective duration of the injection event increases in terms of crank angle, the injection event must be started earlier in order to be completed before TC. This presents the engine designer with a dilemma. For a 1.5 ms injection event it is impossible to advance the start of injection sufficiently to complete the event before TC without also causing auto ignition. Note that in Figure 15 at 6000 RPM, fuel is still being injected until approximately 20 ATC (After Top Center) although the start of injection occurs some 34 degrees BTC (Before Top Center).

It was believed that a short enough injection event would allow injection to be started late enough to avoid auto ignition and completed early enough to achieve satisfactory combustion. Much of the injection system research and modification at the beginning had centered on shortening the injection duration. From this particular point of view it was believed that the high pressure electronic ignition system has an advantage. However, a very short injection duration did not yield the expected results. It was found that the injection duration has an optimum value which depends on the combustion chamber

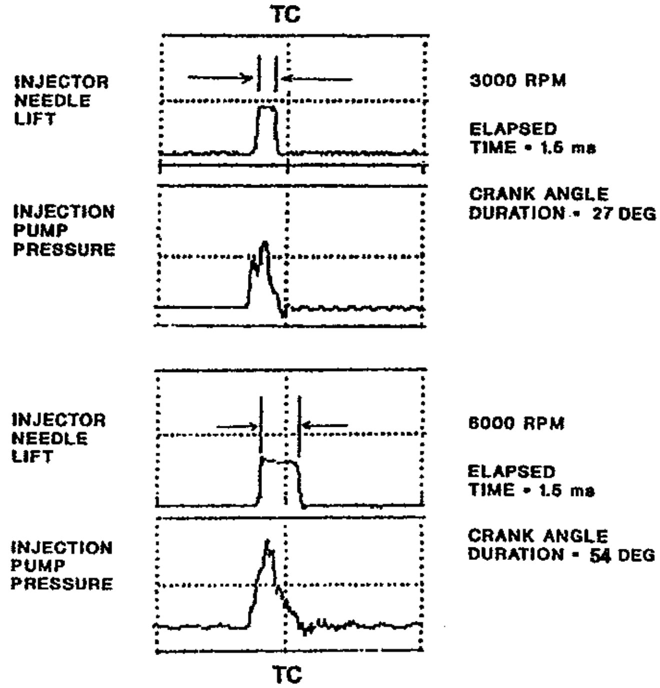


Figure 14 - Fuel injection events at medium and high speed

volume and other factors related to engine operation. It was also found that the pressure value at the end of injection is also a critical parameter for the combustion process in the spark-assisted heavy fuel rotary engine.

Sustained Operation At 6000 RPM - Tests performed on the 400cc engine with regular off-the-shelf cams were not satisfactory. Significant pressure wave effects and

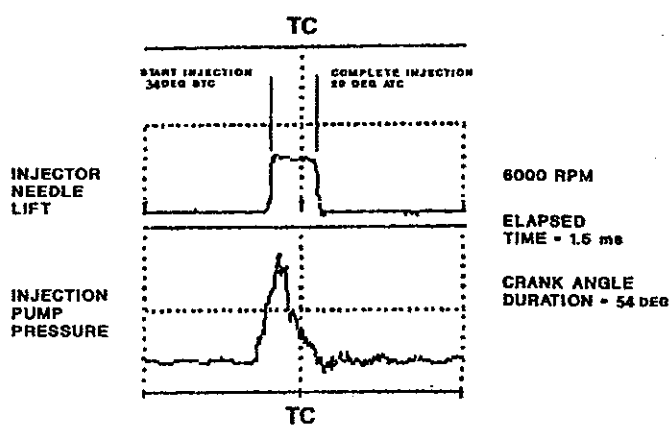


Figure 15 - Injector needle lift and injection pump pressure at high engine speed

severe injector chatter occurred at speeds in excess of 3000 RPM. Operation with both 5.5mm and 7mm diameter pump plungers did little to alleviate the problem. Figure 16 contains traces of injection system pressure and injector needle position obtained with the off-the-shelf cams. Notice the large magnitude of fluctuations in pressure and needle lift position compared to the relatively smooth pressure rise and steady needle lift position displayed in Figure 15. Previous experience has shown that sustained operation with such fluctuations will cause rapid injector wear.

From a multitude of factors with direct influence on the injection system pressure waves, the rate of the pressure rise and nozzle hole lengths had the most notable effects in the selected arrangement. Sustained 6000 RPM operation was achieved using special modified eccentric cam profiles. Although overall engine performance has not been optimized at this RPM, the injection system performance was steady and free from large-scale destructive pressure wave effects.

Shortening Injection Duration - It was initially believed that a tangential cam profile would improve engine performance by creating a shorter injection event. In actual operation, however, this was not the case. Figure 17 illustrates the injection events for both the eccentric and tangential cam profiles at 3000 RPM. Note that the rate of pressure rise is much faster with the tangential cam. But the duration of the injection event, as determined by injector needle position, remains approximately the same. Thus no realizable gain in combustion efficiency was

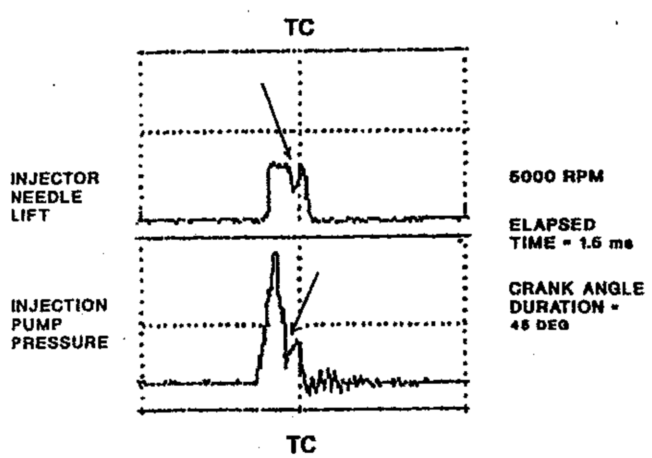


Figure 16 - Pressure waves and needle chatter at high engine speed

attained during the engine test. It is theorized that this nozzle configuration is effectively operating in a choked condition; a further increase in supply pressure does not result in a greater mass flow rate through the injector.

Increasing the diameter and number of holes did not necessarily improve the engine performance. The degree of fuel atomization and subsequent mixing is a function of many variables including injector pressure, flow rate, and hole diameter, number and orientation. Nozzles with two, three and four holes, hole diameters between 0.12mm and 0.25mm and various spraying angles were tested. Injection pumps with plunger diameters of 5, 5.5, 6 and 7mm have been evaluated. Increasing the hole diameter beyond 0.20mm to shorten the duration of the high RPM injection event caused poor fuel atomization, especially at low RPM. This was particularly critical for the selected fuel injection system in which the frequency of the injection pump plunger covers a broad spectrum.

IGNITION SYSTEM EVALUATION - An ignition test rig was developed in order to evaluate various off-the-shelf ignition systems for the heavy fuel rotary application. More

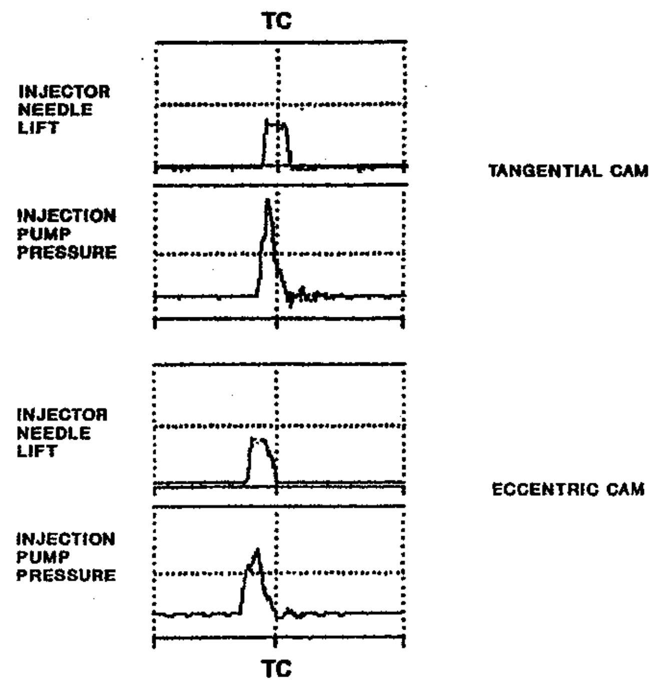


Figure 17 - Injection event for modified tangential and eccentric cams at 3000 RPM

than ten ignition system of various design have been tested. The system design differed in respect to the type of the pick-up sensor (Hall-effect or inductive) and coil design. The ignitions have been evaluated in respect to spark duration, spark energy, primary coil voltage and stored energy. The test rig data were correlated with hot tests on the 400cc engines running on heavy fuels. The most promising system was then modified in order to provide optimum performance in conjunction with heavy fuel application. A high performance transistor incorporated in the system generates a voltage peak higher than 30 kV in order to improve the heavy fuel-air mixture ignitability. The ignition timing control is achieved through an optical sensor activated by a toothed ring.

SPARK PLUG POSITION - The spark plug position relative to injector nozzle for a given combustion chamber design was evaluated on the development rig presented in Figure 12, and subsequently refined based on the performance obtained on the engine test bench. In order to determine the optimum spark depth position for engine startability, a spark plug with standard gasket was installed in the rotor housing of a baseline 400cc engine, and tightened with the required torque. A measurement was taken from the tip of the center electrode to the inside of the rotor housing surface. Similar measurements were taken to determine the spark plug depth with gaskets of varying thicknesses. Using the appropriate gasket, spark plugs were installed in the same engine at different depths and startability tests were run.

Figure 18 shows the engine startability as a function of spark plug depth for a certain combustion chamber configuration (rotor recess, nozzle geometry, injector and spark plug relative position, injection system characteristics, etc).

The start procedure allowed for a maximum of five consecutive start attempts with a waiting period of 20 seconds between two consecutive starts. The cranking speed was kept constant at 500 RPM. An arbitrary startability scale from 1 to 10 was determined based on the easiness of engine starting. The high number reflects good starting behavior.

ENGINE PERFORMANCE - One of the most critical factors which affect the combustion process in the single-injector open chamber spark- ignited small rotary engine is the ignition delay of the combustible mixture. The ignition delay in this particular case was considered as the time (or crankshaft angle degrees) between the start of the injection and the rapid combustion phase of the fuel-air mixture. It includes the time necessary for fuel-air mixture to reach the proper ratio around the spark plug electrodes, the spark discharge and flame kernel formation.

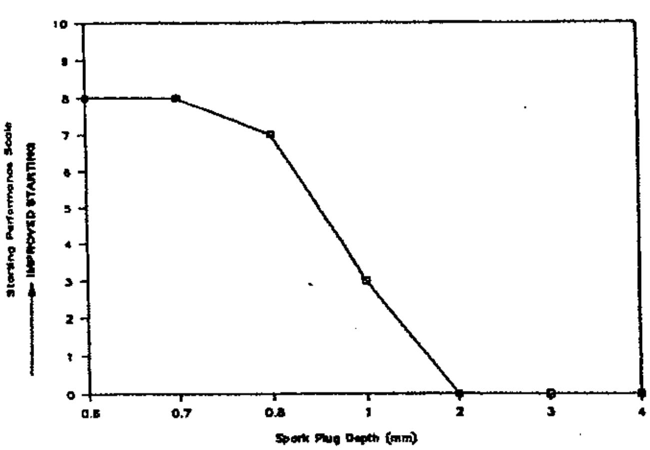


Figure 18 - The influence of spark plug depth position on engine startability

This delay is influenced by a multitude of desig factors, such as rotor recess, injector and spark plu relative position, injection process characteristics, ignitio system characteristics, etc. for a given fuel. The fuel - ai mixture process analysis, for example, should take int consideration that, in the case of small displacemer rotaries, impingement of the fuel spray on the combustio chamber wall is difficult to avoid. This impingement migh amplify the impact of fuel vaporization effect on th combustion, causing an increase in the ignition delay Figure 19 shows the ignition delay effect on the combustio chamber pressure for an unoptimized combustion proces with a strong fuel spray impingement effect at one of th engine functional points, 11kW at 3300 RPM.

Figure 20 reveals a substantially improved combustio process at the same functional point after all design factor enumerated above had been optimized, and the spray impingement on the combustion chamber walls wa modified.

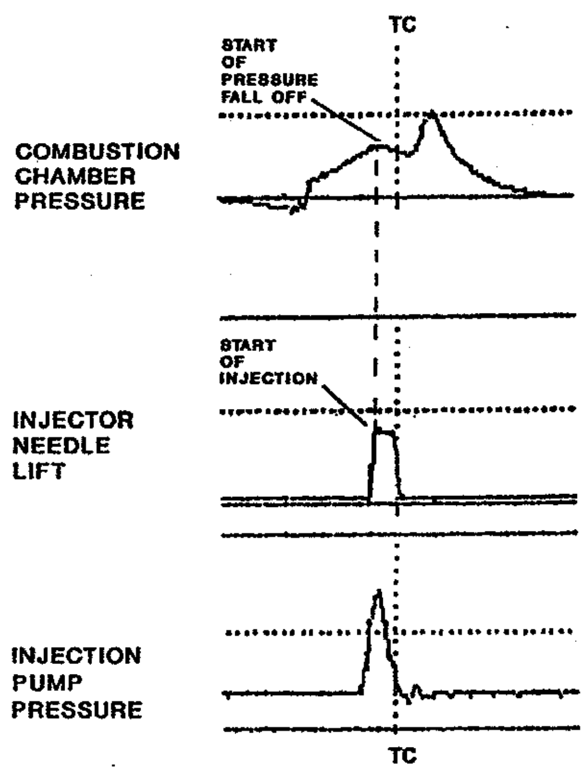


Figure 19 - Ignition delay effect on combustion chamber pressure evolution for an unoptimized combustion process at 11kW and 3300 RPM

Figure 21 illustrates the current engine performance for the governor set at 3300 RPM. For limited power stationary application, the engine speed can be set by the mechanical governor at 3000 RPM, 3300 RPM, or 360(RPM. The twin- rotor engine power performance is twice that of a single-rotor engine, while the specific fue consumption is slightly better.

The engine performance optimization continues. An advanced engine concept is also currently under development with increased power density (power/weight ratio). Based on the experience available at hand and planned engine modification primarily to increase the amount of air available for combustion, a specific fuel consumption lower than 300 g/kW-hr. (0.5 lb/HP-hr.) at the engine rated power is believed to be achievable.

MULTIFUEL OPERATION - In addition to various grades of diesel fuel, the engine was tested and optimized

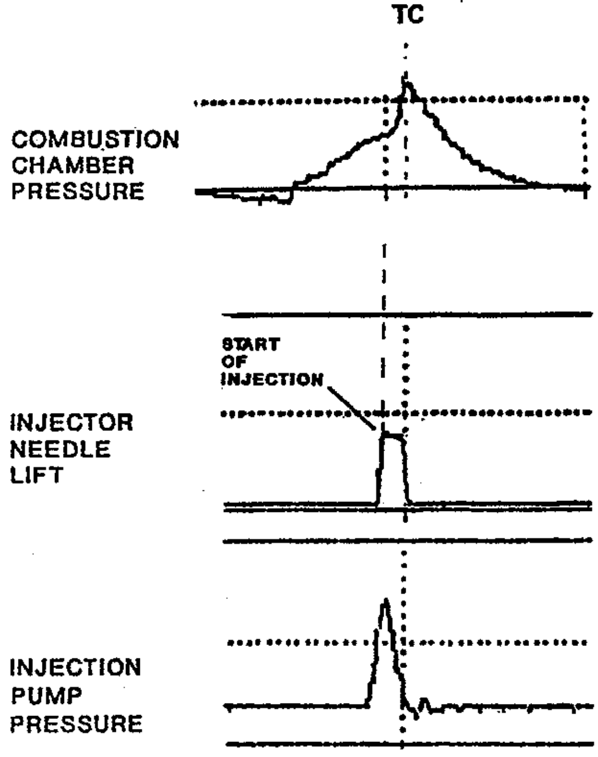


Figure 20 - Improved combustion process with reduced ignition delay at 11kW and 3300 RPM

on jet fuels, particularly JP-5 and JP-8 type of fuels. The 400cc series of engines shows good potential for multifuel application, being able to run on most common heavy fuels without adjustment to the engine or the fuel injection system. Figure 22 shows data obtained during a 300 hour test run at partial load on DF-2, JP-5 and JP-8 on the LCCR 400 SD engine. The slight variation of the engine BSFC recorded during the test for the runs using the same fuel (e.g. JP-8) was caused by variations of the nozzle hole geometry among different nozzles of a production lot. For the same nozzle the engine fuel economy recorded for the JP-5 and JP-8 fuels are slightly better than that obtained for the diesel fuel at the same engine load and speed. During the entire test all other engine functional

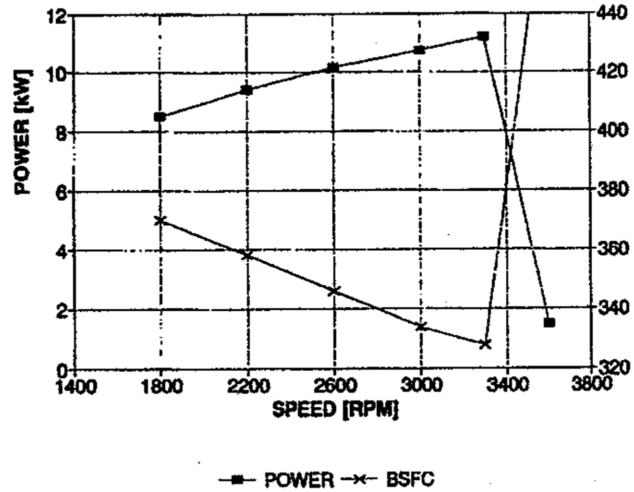


Figure 21 - LCCR 400 SD engine performance

parameters were monitored and maintained within the acceptable limits.

SUMMARY

APPROACH - With the assistance of an automotive type of ignition system and high pressure fuel injection system, a single injector, open chamber rotary engine was optimized to run satisfactorily on diesel and jet fuels. In the absence of in-depth studies on fluids motion (air and fuel) and combustion process in the small spark assisted heavy

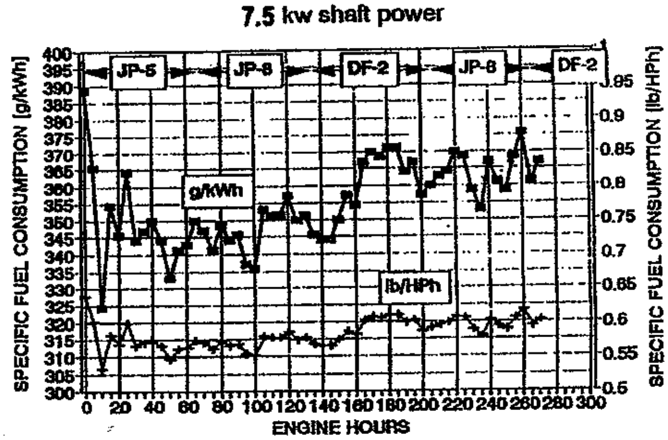


Figure 22 - LCCR 400 SD engine test with various heavy fuels

fuel rotary engines, the performance optimization work relied on experimental techniques. Significant modifications of an off-the-shelf fuel injection system were necessary in order to accommodate the operational peculiarities of the single-injector rotary engine, especially at speeds higher than 3000 RPM. The modification addressed the injection cam profile, injection pump fuel delivery mode and injector nozzle configuration.

A multitude of design inter-related parameters, such as rotor recess design, injection process average pressure injection pump plunger diameter and stroke, injector opening pressure, injection timing and duration, nozzle hole geometry, ignition timing, spark energy and duration, nozzle - spark plug relative position, etc. had to be considered during the combustion optimization process.

The particulars of the high speed operation on heavy fuels were investigated. The engine capability of operating on heavy fuel with a mechanical injection system at constant speed between 3000 and 6000 RPM was demonstrated.

Various factors which influence the ignition delay of the air-fuel mixture for different heavy fuels have been investigated. Improvement of the combustion process of the heavy fuels in a single-injector, open combustion chamber was obtained.

RESULTS - The engine demonstrated satisfactory results operating on heavy fuels at constant speed and various loads. A power rating of 11 kW at 3300 RPM was selected in order to increase durability. A BSFC (brake specific fuel consumption) of 330 gr./kW-hr (0.55 lb/hp hr) at rated point is consistent on different engines of the same configuration. Multifuel capability was demonstrated for a continuous run of 300 hours. A 1,000 hour durability evaluation test was performed on a single prototype engine with a scheduled engine tear down for technical evaluation at 500 and 1000 hour running time. The wear rate of the engine critical parts such as apex seal, rotor and shaft bearings and fuel injection pump had been documented. No structural problems have been identified with respect to engine design while running at rated conditions.

CONCLUSIONS - The single-injector, open combustion chamber arrangement could be a convenient solution for small-displacement rotary engines where the space available for the combustion chamber is limited and simplicity and cost play an important role.

FUTURE IMPROVEMENTS

Continuous performance enhancement is anticipated by further improvement of the fuel injection and fuel-air mixing process and by increasing the amount of air made available for combustion. An engine optimization program currently in progress for the single-rotor engine is considered to be representative for the technology of the entire 400cc family of engines. Design improvements are planned for engine cooling, gas sealing, combustion air supply and fuel injection systems.

In parallel, a high power density engine, with turbocharging capability based on the same engine geometry, is being optimized on heavy fuel for a broader range of speeds and loads.

ACKNOWLEDGEMENT

The following individuals contributed to the work presented in this paper: Josef Speiser, Wolfgang Sohler, Manfred Fischer, Rainer Linsenmeyer and Michael Schwarz at Wankel R & D; Ajay Patel, Bibi Ursu, Victor Nicolescu, Mike Pana and Ralph Curcio at RPI, Inc.; and Charles Preston of Aurora Flight Science Inc.

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