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Thomas R. Stuecken

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Masters degree in Agric. Engr. Tech.

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THE DEVELOPMENT AND ANALYSIS OF A PORT FUEL INJECTED
TWO-STROKE CYCLE GASOLINE ENGINE WITH POPPET VALVE EXHAUST

By

THOMAS R. STUECKEN

A THESIS

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ABSTRACT

THE DEVELOPMENT AND ANALYSIS OF A PORT FUEL INJECTED
TWO-STROKE CYCLE GASOLINE ENGINE WITH POPPET VALVE EXHAUST

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A major problem with homogeneously charged two-stroke cycle gasoline engines is that a portion of the unburned air/fuel mixture is short circuited in the scavenging process because of the exhaust ports. The combustion of lubrication oil reduces engine performance and increases hydrocarbon emissions.

For this study, a gasoline two-stroke cycle engine was designed and built with poppet valve exhaust and electronic port fuel injection. The lubrication system is similar to that of a conventional four stroke engine. The cylinder was externally supercharged. The engine was built to fire but motoring tests studying the cylinder pressure and air flow characteristics at different cam timings were conducted for this thesis.

Variations in the cam timing had a large impact on the characteristics of air flow and peak cylinder pressures given a constant boost. The data shows that advancing the cam timing with the same duration reduces air consumption and increases the peak cylinder pressures. Results also show that a variable boost supply would be needed to match the air supply demands of the engine.

ACKNOWLEDGMENTS

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CHAPTER 1

INTRODUCTION

Two-stroke cycle gasoline engines are power dense and relatively simple compared to engines with poppet valves and high pressure fuel injectors. Controlling the combustion accurately over various engine speeds and loads is very difficult. Combustion inefficiencies are the reason that no two-stroke gasoline automotive engine has passed emission standards for street use in the United States (Scott, 1987). The main reason for the low combustion efficiencies is that 25-40% of the fuel is lost due to short circuiting to the exhaust of some unburned fresh charge (Blair and Douglas, 1982). This short circuiting is due to the fact that the exhaust port is open when the inlet transfer port is open and closes after the inlet transfer port is closed. This loss of unburned fuel through the exhaust system increases fuel consumption and results in high hydrocarbon emissions. The other factor that lowers the engine's efficiency is the burning of lubrication oil in the combustion process. This oil is mixed with the gasoline or the air-fuel mixture to lubricate the bearings, cylinder and compression rings. Oil also lowers the octane rating of the fuel and once it's burned, it will increase the hydrocarbon emissions.

Automotive companies are striving for smaller light weight engines that produce the same horsepower of existing engines with lower hydrocarbon emissions. Two-stroke cycle gasoline engines fit these criteria well other than for their efficiency and emissions. To reduce these problems the short circuiting of fuel must be eliminated and the combustion of lubrication oil must be reduced or eliminated. To prevent the fuel from short circuiting, the fuel must be introduced into the combustion chamber after the exhaust ports are closed or introduced through an inlet transfer port with a fresh air charge preceding it for scavenging (Grasas-Alsian et al., 1986).

Most of the research that has been done with controlling fuel losses in two-stroke cycle gasoline engines has incorporated low pressure electronic Bosch injectors for direct fuel injection into the combustion chamber. Direct injection of gasoline, with Bosch injectors, after the exhaust ports have been closed doesn't allow the proper amount of time to atomize gasoline properly for efficient combustion. The gasoline needs to be injected earlier in the cycle or into a turbulent swirl chamber to atomize properly.

A pneumatic gasoline injector used for direct injection has been tested in Perth, Australia by the Orbital Engine Company (OEC). This pneumatic injector is mounted in the engine head and directly injects gasoline into the cylinder after the exhaust ports are closed. OEC claims that their

injector has an exceptionally fine fuel atomization, about $1/20$ the droplet size of a conventional electronic injector which droplet size was not found. The major automotive companies in the United States are looking into this injector for possible use in two-stroke cycle gasoline engines for automobiles (Albrecht, 1987).

The injection of fuel through an inlet transfer port is another approach to fuel injection. In order to inject fuel efficiently at this location, there needs to be a delay from the time that the piston uncovers the inlet transfer port until arrival of the first droplet of injected fuel. This delay will allow a fresh air charge to precede the fuel and scavenge the cylinder thus minimizing the short circuiting of gasoline out of the exhaust port. With this arrangement, the fuel will have a longer time to be atomized, because it is injected earlier in the cycle. If a turbulent swirl is present, created from properly angled inlet transfer ports, the fuel will be mixed more thoroughly and evenly throughout the cylinder.

In order to inject fuel directly through an inlet transfer port, the injector has to have a short response time. For example, if an engine running at 5000 rpm has a 90 degree duration on the inlet transfer port opening time there would be less than three milliseconds to inject the proper amount of fuel into the cylinder. In order to get a shorter response time from electronic fuel injectors that are conventionally used in four-stroke automotive engines, a

higher voltage given with a shorter duration needs to be supplied to the injectors in order to reach the opening current sooner. When the response time of an injector is shortened, injection timing at different engine speeds can be controlled more easily and more accurately.

Another problem that standard two-stroke cycle engines have with losing fuel by short circuiting is that the exhaust port is closed after the inlet transfer port closes. If the exhaust port were closed before the inlet transfer port is closed, extra air and fuel could be forced into the cylinder. By using a poppet valve exhaust system, similar to Detroit Diesel engines, the efficiency problems could be partially eliminated. If a poppet valve system were used, the engine speed would be limited because of the valve train mass and valve float. If the engine ran more efficiently by being able to charge itself with more air and fuel, which would have been lost by short circuiting, the engine wouldn't need to run at high speeds to achieve the same power. Another benefit of lower engine speed is the slower piston speed. By reducing the piston speed, the engine's life can be greatly increased because of the reduced wear.

In order to control the burning of lubrication oil, the oil can't be allowed to enter the combustion chamber. The way that Detroit Diesel engines control the oil is to externally supercharge the engine by forcing air into the cylinders using a Roots blower. The crankcase is set up

like a standard four-stroke cycle engine for lubrication. With this arrangement, oil would not mix with the incoming air unless the oil control rings on the lower end of the piston skirt failed to seal the oil from the inlet transfer ports. Another benefit from supercharging compared with crankcase charging is that the air will no longer absorb the crankcase heat, thus allowing the air to be more dense.

Considering the results from studies on two-stroke cycle engines, developing a supercharged two-stroke cycle gasoline engine with inlet transfer port fuel injection and a poppet valve exhaust system appears promising.

CHAPTER 2

OBJECTIVES

The overall objective of this research was to investigate possible means of increasing the efficiency of the two-stroke gasoline engine. To obtain this overall objective the following specific objectives were set.

1. To develop a method to close the exhaust transfer ports before the inlet transfer ports are closed.
2. To develop a method to supply the engine with atomized fuel in a controlled manner at various speeds and loads.
3. To develop a method to reduce the burning of lubrication oil in the combustion process.
4. To fabricate an engine that met the first three objectives.
5. To study the cylinder pressure versus the crankshaft angle under different motoring speeds and exhaust transfer port openings.
6. To test the air flow characteristics of the engine at different motoring speeds and different exhaust transfer port openings.

CHAPTER 3

LITERATURE REVIEW

Much work has been done to improve the combustion efficiencies of two-stroke cycle gasoline engines. Most of this research consisted of placing fuel injectors in different engine locations on recent production engines. No literature was found on research done on a fuel injected gasoline two-stroke engine with poppet exhaust valves other than a computer simulation model developed by Carpenter and Ramos (1986). This model simulated a turbocharged two-stroke cycle gasoline engine that had direct fuel injection near the engine head. The gasoline was injected directly into the combustion chamber starting at 30 degrees before TDC until TDC. A poppet valve was used for the exhaust port located in the center of the head. This valve started opening at 72 degrees before BDC and finished closing at 30 degrees after BDC. The inlet transfer ports were circumferentially located on the cylinder wall and were covered and uncovered by the piston. The inlet transfer port starts to open at 36 degrees before BDC and fully closes at 36 degrees after BDC.

This simulation model was developed to find the scavenging efficiency, flow field and heat losses for this

simulated engine. Their computer model predicted a 94% scavenging efficiency with the 6% loss being supercharged air and injected fuel remaining in the cylinder after the scavenging process. From their results they concluded that the fuel could be burned more efficiently if it were injected earlier in the engine cycle or if large scale convective motions were present.

Taylor (1960) described a production two-stroke cycle gasoline engine with an engine inlet and exhaust porting table. This was a low speed engine with a large bore of 40.64 cm (16 inches) and its exhaust ports were controlled by a poppet valve exhaust system. The exhaust valves opened at 75 degrees before BDC and closed at 40 degrees after BDC. The inlet transfer port opened and closed 45 degrees from BDC. The type of fuel system was not mentioned. They also stated that poppet valves in two-stroke cycle engines provide excellent performance as well as high scavenging efficiencies. The only drawback from poppet valve exhaust is that it limits the engine speed because of valve float and the valve train mass. Taylor (1985) stated that the valves should be as large as feasible, because a rapid blowdown is essential for this type of engine. He also stated that the use of four valves, with the same flow area as one valve, would reduce the total valve mass to half of one large valve.

Grasas-Alsina et al. (1986) studied fuel injection on a single cylinder two-stroke Motesa Crono engine. This 350 cc

loop scavenged engine had a 10:1 total compression ratio. It had the standard two-stroke crankcase used for the scavenge pump. The exhaust ports and inlet transfer ports were covered and uncovered by the piston. The exhaust ports opened and closed 90 degrees from BDC while the inlet transfer ports started opening and were fully closed 64 degrees from BDC. They used an electronic Bosch fuel injector that was supplied with 300 kPa (43.5 psi) fuel pressure. This injector was located in the inlet transfer port, spraying fuel directly into the cylinder.

From the tests that Grasas-Alsina et al. ran they concluded that an earlier injection of gasoline into the cylinder was more desirable in order to increase the time available for the fuel to vaporize and mix with the air. They also stated that increased turbulence in the cylinder was necessary to achieve a more efficient combustion. This is consistent with the results which Carpenter and Ramos (1986) found in their research. Grasas-Alsina et al. also found that proper injection timing was critical in order to reduce the short circuiting through the exhaust port. They also found that scavenging with only fresh air is desirable, but the fuel must be introduced immediately behind the fresh air to assure proper mixing. Increasing the engine speed increased the air velocities in the cylinder and created more turbulence. High turbulence within the cylinder would permit a longer fuel free scavenging without subsequent inefficiency in the pre-combustion processes. This

injection delay from the time the inlet transfer port opens until injection of the first fuel droplet was 9 to 14 degrees at 3000 rpm, 19 degrees at 4000 rpm and 34 degrees at 5000 rpm.

Blair and Douglas (1982) fuel injected a 96.8 cc single cylinder two-stroke Yamaha YB100 motor cycle engine. This engine had a trapped compression ratio of 6.7:1 and a bore and stroke of 52 and 45.6 mm, respectively. Fuel injection was tested in four locations in the engine with Bosch electronic fuel injectors supplied with 280 kPa (40.6 psi) fuel pressure. The injector located in the inlet to the crankcase showed the best results over the other locations which included injection into the cylinder, swirl chamber and rear transfer duct. They found overall that there is very little improvement in fuel economy without a considerable loss in power. At partial engine throttle, however, it was determined that there was a power increase with fuel injection but only small improvements in fuel economy were achieved at low engine speeds.

Kuentscher (1986) studied lean combustion systems with a three cylinder gasoline loop scavenged two-stroke cycle engine. The exhaust and inlet transfer ports were covered and uncovered by the piston. The exhaust port started to open at 82 degrees after TDC and the inlet transfer port started to open at 55 degrees before BDC. Both ports were fully open when the piston reached BDC. In one of their tests an electronic Bosch fuel injector was located in the

engine head. Their objective was to inject gasoline directly into the cylinder after the exhaust port was closed to eliminate short circuiting. The test results showed that for minimum fuel consumption, the optimum ending of injection was found to be at 10 degrees before BDC. They also found that the crankshaft rotation from the static ending of injection until ignition timing must be 140-150 degrees to achieve the lowest specific fuel consumption. They concluded that direct injection from the cylinder head didn't adequately reduce fuel consumption and hydrocarbon emissions as compared to carburetion.

Paterson and Sung (1982) studied the effects of air flow through a Detroit Diesel 6V-92 two-stroke cycle engine cylinder. The engine's cylinders are set up with 18 inlet transfer ports equally spaced on the periphery of the cylinder and the ports are angled at 25 degrees relative to a radial line from the center of the cylinder. The exhaust exited through four poppet valves located in the cylinder head. A blower is used to scavenge the cylinders. Their objective was to find a different style of exhaust porting that would cause a uniform axial flow of air through the cylinder. The results of their experimentation showed that the swirl induced by the angled inlet ports caused the axial velocity profile to be non-uniform. These non-uniformities still remained with different styles of exhaust porting, even if the cylinder was open ended the non-uniformities remained. Carpenter and Ramos (1986) found the same results

in their research and concluded that the turbulence levels in the cylinder are controlled by the characteristics of the intake flow.

An article by Wooldridge (1987) describes the design of a Mercury marine two-stroke cycle boat engine's combustion chambers. These chambers generated more turbulence resulting in a more uniform mixture of fuel in the cylinder, which created more complete combustion than older designs.

Scott (1987) stated that lubricating oil in two-stroke cycle engines inevitably mixes with the fuel as the mixture passes through the crankcase. The burning oil also becomes a source of further hydrocarbon emissions. MacInnes (1978) stated in his book that oil has a detrimental effect on the octane rating of the fuel. Turbocharging a two-stroke engine with oil in the fuel would lower the boost capacity because of detonation caused by the lowering of the octane number.

Brunt and Lucas (1982) studied the effect of combustion chamber shape on the rate of combustion for a spark ignition engine. Lucas found that the dual ignition designs had the highest combustion rate with about 2-4 degrees less ignition advance required to reach the same point of peak combustion pressures of single ignition cases. Suzuki ran dual ignition in some of their later model two-stroke cycle boat engines resulting in a reported improvement in efficiency (Wooldridge, 1987). Taylor and Taylor (1966) stated that dual spark plugs, usually at

opposite sides of the combustion chamber, will somewhat reduce the tendency to detonate when the appropriate spark timing is used. With two spark plugs firing, if one plug fails to fire, the tendency to detonate may be reduced because the effective ignition timing is retarded.

Most of the research done with two-stroke cycle gasoline has been the testing of fuel injection in different locations in standard engines. No work was found involving the fuel injection of a two-stroke cycle gasoline engine with poppet valve exhaust other than for the computer simulation model by Carpenter and Ramos (1986). With the recent trend of automotive manufacturers building high performance four-stroke cycle fuel injected gasoline engines with four valves per cylinder, it appears that two-stroke cycle fuel injected gasoline engines with poppet valve exhaust would be a configuration which offers the possibility of high performance and controlled emissions.

CHAPTER 4

ENGINE COMPONENTS AND SYSTEMS

4.1 Engine components.

A single cylinder turbocharged two-stroke gasoline engine was designed and constructed that has a poppet valve exhaust system, direct fuel injection through one of the eight angled inlet transfer ports, and dual ignition. This engine follows some of the same principles of the General Motors Detroit Diesel engines. The Detroit Diesel engines use poppet valves for the exhaust ports with a Roots blower to supply a fresh air charge into the cylinder. The diesel fuel is injected through the head like standard diesel engines.

There were no University funds available for engine fabrication, so many parts for this experimental engine were scavenged from old engines and machinery. The base of the engine consisted of a 1975 four-stroke cycle Honda XL350 block (Figure 1) and crankshaft that were modified to house the new components.¹ Since this engine is unique in design, many of the components had to be machined. Any

¹ Trade names are used in this paper solely to provide specific information. Mention of a product name does not constitute an endorsement of the product by the author to the exclusion of other products not mentioned.

parts that could be purchased were used to save time and allow for easy replacement in case of a malfunction. The engine specifications are shown in table 1.

4.1.1 Overhead cam assembly.

Figure 2 shows a breakdown of the overhead cam assembly. This consists of the camshaft (1), which was machined from a camshaft out of a Chevrolet 5.74 liter (350 ci.) V-8 engine. One lobe was used and the rest of the cam was machined to hold the two bearing races which were pressed onto the camshaft. The camshaft rotates in two needle roller bearings which have a pressed fit in the steel bearing blocks (3). The bearings have one seal on their outer edge which prevents the oil from leaking from the assembly. Lubrication oil is allowed to flow out of the inner side of the bearings into the valve compartment. Each bearing block has an alignment end plate that is screwed onto each block. These plates keep the cam centered into its proper position.

The camshaft is powered by the hub and sprocket assembly (4). The hub has a slip fit over the camshaft and is held in place by a keyway and set screw. The sprocket, a cam sprocket from a Honda XL350, is fastened to the hub by two socket head cap screws. This sprocket is driven by a chain from an identically sized sprocket on the crankshaft.

The cam drives the rocker arm (2). This rocker arm, machined from 6061 T-6 aluminum, rotates on a hardened steel

Table 1. Engine specifications.

Stroke	7.099 cm (2.795 in.)
Bore	7.699 cm (3.031 in.)
Displacement	330.49 cc (20.17 ci.)
Compression ratio	
Total	7.78:1
Trapped	6.56:1
Inlet transfer ports (8)	
Duration open	110.7 degrees
Piston even with top edge	55.34 degrees from BDC
Height	1.283 cm (.505 in.)
Peripheral width	2.057 cm (.810 in.)
Area (all ports)	21.109 cm ² (3.272 in ²)
Exhaust ports (4 valves)	
Duration open	120 degrees
Valve lift	.828 cm (.326 in.)
Valve seat I.D.	2.388 cm (.940 in.)
Area (all ports)	17.910 cm ² (2.776 in ²)
Valve clearance	.203 mm (.008 in.)
Compression	1034.2 kPa (150 psi) motored at 600 rpm
Spark plug	NGK D8AE
Gap	.813 mm (.032 in)

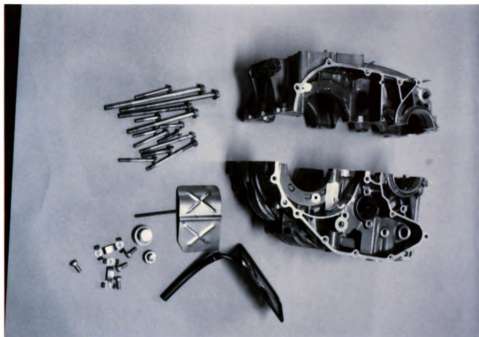


Figure 1. Honda XL350 block with bolts and crankcase breather.

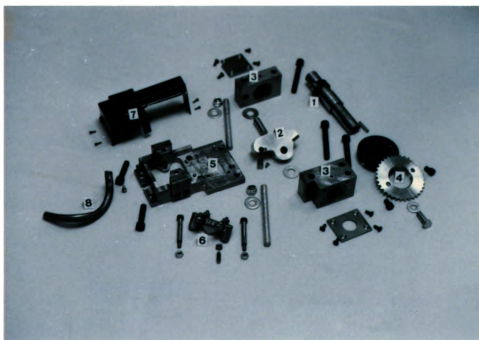


Figure 2. Overhead cam assembly.

pin that is supported on either end by the bearing blocks (3). The cam end of the rocker arm houses a hardened steel wear plate that was constructed from a grade eight bolt head that was hardened by heating and oil quenching. This plate rides on the cam lobe. On the opposite end of the rocker arm a hardened steel pin has a slip fit into the rocker arm and is allowed to rotate freely. Two small round head slotted machine screws fit into the rocker arm and the bottom edge of their heads hold the pin from sliding out.

The rocker arm roller rides on a hardened steel plate that is inlaid in the steel valve bridge (6). The valve bridge rides on two bridge guides which lock the valve bridge into position to assure that four valves will open simultaneously. The two bridge guides are 1/4-20 shoulder bolts that screw into the base plate (5) and are locked in place by two jam nuts. The valve bridge also has an adjustment screw and lock nut for each one of the four valves. These adjustment screws and nuts are from the Honda XL350.

The steel base plate (5) houses the components for the cam assembly. This plate is bolted onto the engine head by two 3/8-16 socket head cap screws and two 7/16-14 studs with nylon lock nuts. The bearing blocks are fastened to the base plate by the same two 7/16-14 studs and three 3/8-16 socket head cap screws. Holding the position of the bearing blocks and base plate with respect to the engine head is very critical. In order to accomplish this, the two bearing

blocks slip onto four locating pins mounted in the base plate. The base plate slips onto two locating pins mounted in the engine head. A support fastens onto the back edge of the base plate by two of the three socket head cap screws that hold the bearing blocks down. This support helps to stabilize the head and cylinder assembly to the engine block.

The overhead cam assembly is lubricated with motor oil that is distributed to the various parts by brake line tubing. The points of lubrication are the two needle roller bearings, the cam lobe, and the bridge guides. The oil exits the cam assembly by gravity flow through the oil tube (8). This tube mounts onto the lower edge of the base plate and is held in position with a machine screw. A hose is connected to the tube and allows the oil to return to the external oil reservoir. A cover (7) encloses the entire assembly and is sealed with silicon. The cover also acts as an attachment base for the oil lines. Figure 3 shows the overhead cam assembly assembled.

4.1.2 Engine head and cylinder assembly.

Figure 4 shows the engine head and cylinder assembly. The valve assembly (9) is the standard exhaust valve and spring system used in the Honda XL350. These valves use a double spring set up with keepers and a valve spring retainer to hold the valve/spring assembly in place. New valve guides were pressed into the head and their ends were

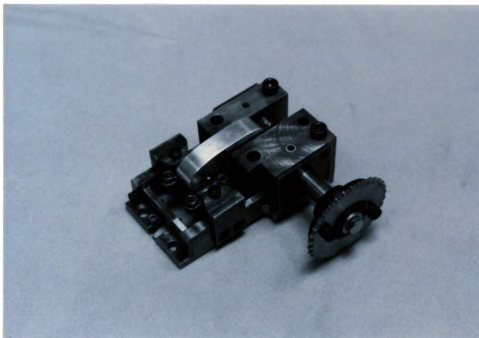


Figure 3. Overhead cam assembly assembled.

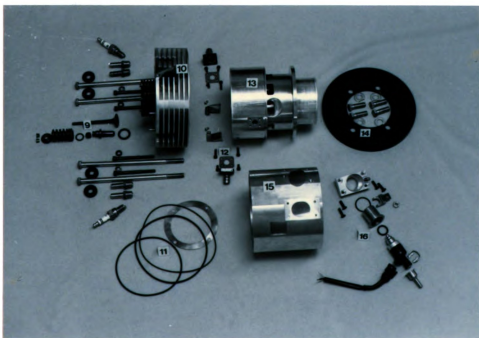


Figure 4. Engine head and cylinder assembly.

ground to fit the contour of the exhaust ports. The valve seats were machined from stainless steel, and then pressed into the head. The valves and valve guides were then ground to form a proper seal.

The engine head (10) was machined out of 2024 T-351 aluminum 2 1/2 inch plate stock. The head houses two sparkplugs, four exhaust valves, two exhaust ports, cooling fins, four water cooling ports, and a base for the overhead cam assembly. Figure 5 shows the top of the engine head. The spark plugs are the standard plugs for the Honda XL350. The plugs are located at the opposite sides of the head, 90 degrees from either exhaust port. The four exhaust valves work simultaneously with each other with two valves feeding each exhaust port. A four bolt flange with a flange gasket seals the exhaust port to the exhaust manifold.

The total flow area of the valve seats is 17.91 cm^2 (2.78 in^2) which gives a port-area to piston-area ratio of 0.38. This ratio is at the upper level of the exhaust port areas for two-stroke cycle engines with a piston speed of 366 meters/minute (1200 ft/min) as shown in a port area figure by Taylor (1985).

The head is cooled by water, oil and air. The head has a channel cut into the bottom side which extends the cylinder's water jacket by more than 1/2 inch. Figure 6 shows the bottom side of the engine head. Water circulates through the channel and exits at the four ports which are located at either side of the exhaust ports. Hose fittings

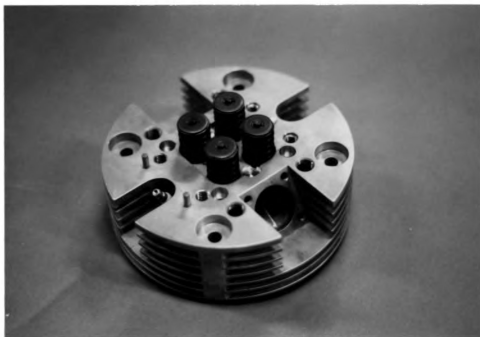


Figure 5. Engine head (top view).

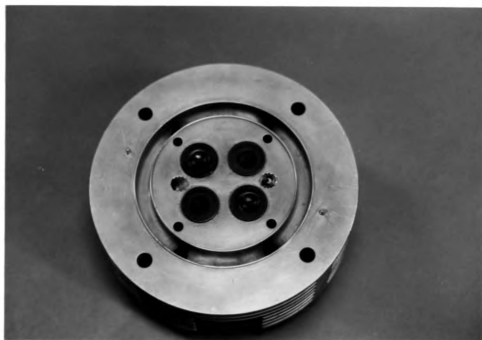


Figure 6. Engine head (bottom view).

screw into the four water outlet ports providing a place for 1/4 inch hose attachments (12, Figure 4). Oil cooling is provided by the oil flow through the valve train. This oil comes in contact with the top of the head around the valves. Air provides the third means of cooling to the head by cooling fins that were machined into the head along the four sides.

The valve plate fastens directly to the head by means of two 7/16-14 studs and two 3/8-16 socket head cap screws shown in figure 2. Two locating pins, pressed into the head, hold the position of the valve plate. Silicon seals the valve plate to the head to prevent oil leakage. The engine head is fastened to the cylinder (13), Figure 4 by four 1/4-20 socket head cap screws and fastened to the engine block by four 3/8-16 hex head cap screws. Hardened steel washers are used on all eight screws to prevent the distortion of the washers and aluminum.

The cylinder (13) was also machined from 2040 T-351 aluminum. A 1/8 inch wall cast iron sleeve was pressed into the cylinder to form the cylinder wall. Eight angled inlet transfer ports, one of which houses the fuel injector, were milled into the periphery of the cylinder wall. These ports are fully open when the piston is at bottom dead center. The total area of the inlet transfer ports is 21.11 cm² (3.27 in²) which gave a port-area to piston-area of 0.45. This ratio is within the range of small spark ignition two-stroke cycle engines and is on the high side

for diesel two-strokes that are shown in a port area figure by Taylor (1985).

The top third of the cylinder has a water jacket surrounding the combustion chamber. This water jacket is fed by two water inlet ports which are constructed out of a 1/4 inch hose fitting mounted to a plate with four mounting holes (12). Four socket head cap screws fasten the hose barb to the cylinder with a gasket used as a seal. On the inside of the water jacket two flow diverting plates are mounted over the water outlets. These two plates cause the water to swirl around the cylinder in the jacket. This swirling action will cause the cylinder wall to be cooled more uniformly and prevent two "cold spots" where the water enters into the jacket.

Figure 7 shows the piston and cylinder arrangement with the two flow diverting plates at the bottom of the water jacket. On the top of the cylinder the four threaded holes shown are for the 1/4-20 head bolts, and the o-ring groove to seal the water jacket is shown on the outer edge of the water jacket. The cylinder is mounted on the engine block with the piston at BDC.

The adapter plate (14) is machined out of 1/2 inch plate steel. Four hex head cap screws fasten the plate onto the engine block. The cylinder slides into the plate and is held in place by locating pins. The four long 3/8-16 hex head cap screws compress the cylinder between the head and adapter plate.

The shell (15), constructed from 6061 T3 aluminum, slides over the cylinder assembly. This shell provides three functions. The main function of the shell is to form the intake manifold for the cylinder. The volume that is created between the inside diameter of the cylinder through the inlet transfer ports to the inside edge of the shell is the same as the engine displacement volume. The reason for the plenum in the intake manifold is to act as an accumulator to store an air charge for the following intake cycle when the ports are closed.

Two O-rings seal the shell to form the manifold. One O-ring (14) fits into a groove above the top of the inlet ports on the outer edge of the cylinder (figure 8). The other O-ring is compressed between the bottom inside edge of the shell, the adapter plate, and the bottom edge of the lower lip of the cylinder base. The other two functions that the shell provides is a mounting base for the fuel injector assembly (16, Figure 4), and a place for the intake manifold tube to attach. Figure 8 shows the shell sliding onto the cylinder with the O-rings in position to seal the intake manifold. The intake manifold and water jacket are sealed by O-rings (11) Figure 4. The .127 mm (.005 inch) thick copper head gasket (11) which was made from copper sheet material is also shown.

The fuel injection assembly (16, Figure 4) is made up of a General Motors Multec solenoid operated fuel injector which fits into a steel cup. This cup fits into the holding

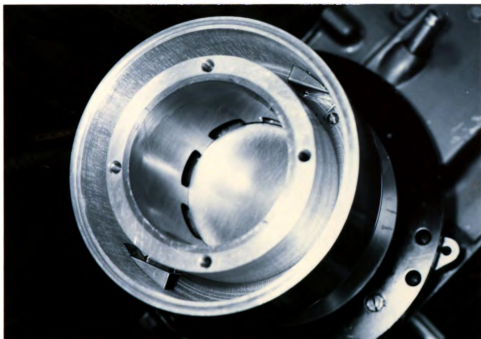


Figure 7. Piston/cylinder arrangement with diverting plates.

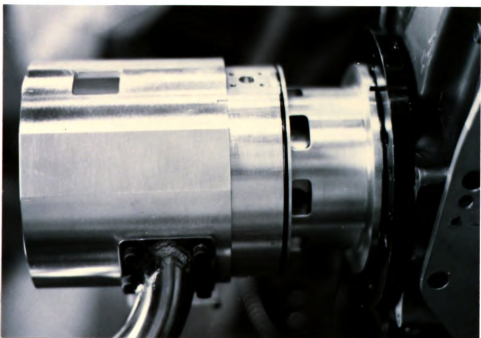


Figure 8. Shell sliding onto cylinder to create intake manifold.

plate which holds the injector so that the tip is angled at a 20 degree incline upward from perpendicular to the cylinder center line. This injector position allows the fuel to be injected upward into the combustion chamber. An O-ring seals the steel cup to the aluminum holding plate. Four #10 flat head screws fasten the injector holding plate onto the shell that is sealed with silicon. After the injector assembly is in place, a retaining bracket slides over a 1/4-20 stud and is held in place by a nut. Tightening this nut compresses the O-rings to seal the injector assembly. A two wire cord plugs into the injector from the computer where it is controlled. The fuel line slides onto a hose barb that has a cupped end on it. A hose clamp holds the hose on the barb. This cup fits over the end of the fuel injector and is sealed by an O-ring that is standard with the injector. A retainer clip holds the cup on the injector.

4.1.3 Intake and exhaust system.

Figure 9 shows the intake and exhaust system. Since the piston isn't acting as a charge pump another pump was needed, so a turbocharger from a 1982 Honda CX500 was used. This turbocharger is equipped with an internal wastegate pre-set to open when an intake charge of 102 Kpa (17.1 psi) is reached. An oil delivery of around 517 kPa (75 psi) through their oil fitting orifices is required to lubricate the full floating rotor bearings and cool the turbocharger

housing.

A throttle chamber (21) was machined out of 6061 aluminum. This houses a brass throttle plate and shaft that is rotated by a stepping motor under computer control. This motor bolts onto the throttle housing and has a 3:1 gear reduction to the throttle plate which causes the plate to rotate about 1 degree per step of the motor. The computer controls the stepping motor through a potentiometer on an instrument panel. The plate has the freedom to rotate from fully open to fully closed.

An air cleaner unit fastens onto the intake end of the throttle chamber. The throttle chamber has a slip fit over the intake of the compressor housing (20) and a set screw to hold the throttle chamber in place. An O-ring mounted on the inside of the throttle chamber provides an airtight seal. The intake manifold (22), made from 1 1/2 inch exhaust pipe, joins the compressor outlet to the shell. The compressor outlet connection is made by a hose and two hose clamps. The shell connection is made by four 1/4-20 socket head cap screws with silicon to provide the seal.

The exhaust manifold (19), was made from 1 1/2 inch exhaust pipe, and is fastened onto the head by eight 1/4-20 socket head cap screws. The shorter screws shown are used to clear the concave side of the manifold. The turbine end of the manifold has a three-bolt flange that slips over the three 7mm studs and is sealed with a copper gasket. Three nuts hold the turbine inlet to the manifold. The intake and

exhaust manifolds provide the main supports for the turbocharger.

A pyrometer probe (18) is mounted at the union of the two exhaust manifolds and is centered in the exhaust stream at the entrance of the turbine inlet. This probe is used to get an accurate temperature reading and to minimize the restriction in the exhaust flow. The exhaust pipe (17) slips over four studs and is held by four nuts. A copper gasket seals the exhaust pipe. The smaller pipe that leads into the main exhaust passage on the exhaust pipe is for the by-passed exhaust from the wastegate of the turbocharger.

4.1.4 Piston and crankshaft assembly.

Figure 10 shows the piston and crankshaft assembly for the engine. The three piece crankshaft (26) is from a Honda XL 350. The two crankshaft halves are fastened together by the piston rod journal. This is a hardened pin that has a pressed fit into the crank halves. The crankshaft rides on two large bearings that are supported by the engine block.

The original connecting rod was removed from the crankshaft and a new one was machined to meet the new cylinder design which called for a rod that was about 3.81 cm (1 1/2 inches) longer than the original one. The additional length was required because of the clearance between the bottom edge of the piston skirt and crankshaft counter weights. The new connecting rod was machined out of

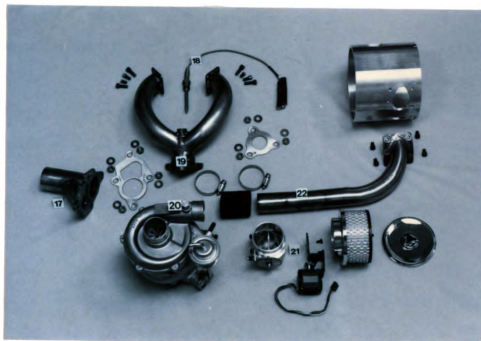


Figure 9. Intake and exhaust components.

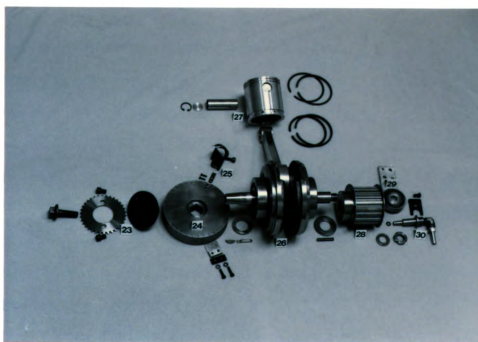


Figure 10. Piston and crankshaft assembly.

2024 T351 aluminum plate stock. The original rod bearings are made up of needle roller bearings that roll in pairs in a bearing retainer with the rod acting as the outer bearing race. No needle bearings or bearing race could be found to replace the originals so the rod journal was machined out of the old rod and pressed into the new aluminum rod. The original needle bearings were reused. The crankshaft was then pressed back together with the new connecting rod and original bearings. The connecting rod has a rectangular cross section and was sized as large as possible given the clearance constraints of the crankcase. The wrist pin boss has a close .005 mm (.0002 inch) clearance fit with the wrist pin to allow for the expansion of the wrist pin boss due to heat.

The piston (27) was also machined out of 2024 T351 aluminum. The top of the piston is flat which helps to reduce the piston mass and made the machining process much easier. The piston was machined to fit the bore of 7.698 cm (3.031 inches). After the piston was roughed out it was placed in a oven at around 550 degrees Fahrenheit to find the expansion. The diameter of the piston expanded .457 mm (.018 inches) keeping in mind that a hot piston in a cylinder needs a few thousandths of an inch clearance to run properly. The top portion of the piston, the compression ring area that gets the hottest, was machined .558 mm (.022 inches) smaller than the bore. The lower portion was machined with less clearance. Engine pistons are slightly

thinner on the wrist pin side to allow the piston to expand on the wrist pin. By doing this, the cold piston won't slap in the cylinder because the sides perpendicular to the wrist pin axis still have the tight tolerances while the piston heats up and expands on the wrist pin. Out of roundness was machined into the piston by off setting it .152 mm (.006 inches) in a four way chuck on a lathe. This was done on both sides of the piston to create .304 mm (.012 inches) of out of roundness.

The wrist pin is from the Honda XL350. It is held in the piston by two aluminum end plates and two snap rings. The end plates have a pressed fit into the piston which will isolate the boost air charge from entering the crankcase and will prevent the lubrication oil from entering into the combustion chamber through the wrist pin boss. The snap ring fits into a groove just outside of the end plates and locks the wrist pin into place.

The piston has two compression rings located at the very top edge of the piston to allow the wrist pin to be as close to the piston head as possible. This closeness helps to reduce the piston weight and also reduces torsion of the piston against the cylinder wall because of a smaller moment arm from the center of the wrist pin to the top of the piston.

Each compression ring is restrained from moving around the piston. A pin, with a diameter equal to the thickness of the ring, is pressed into each piston ring groove

perpendicular to the piston axis and protrudes half of the ring depth. The rings ends were slotted from the inside diameter half way out to the outside diameter to fit over this pin. The pin keeps the ring from moving and the piston ring keeps the pin from working out of the piston. The reason for fixing the position of the piston rings is to keep the ends of the ring from passing across an inlet transfer port. The ends of the ring could expand into the port, because of not being supported on the ends, and cause severe damage to the rings, piston and the cylinder wall.

In the ring groove, at the bottom of the piston skirt, an oil control ring with a compression ring mounted above it are used. The compression ring is used to seal the intake charge from entering into the crankcase and help prevent oil from entering into the inlet transfer ports. These rings never pass over the inlet transfer ports but meet the bottom edge of the ports. Oil passages were drilled into the oil control groove to allow the oil to escape from behind the rings.

The original flywheel/magneto assembly was discarded and a new flywheel (24) was machined to take its place. This modification was done for two reasons: One was that the cam chain had to run external to the engine block in order to line up with the cam gear. The original cover wouldn't allow this positioning so a flat steel plate (shown in Figures 12 and 13) replaced the cover and was bolted onto the side of the block. The new cover reduced the space for

the enclosed flywheel so a narrower flywheel was built to fit behind the new cover. The second reason for the design change was that timing marks were needed to allow the calibration of the computer controlling system. Timing marks, which provide about a $1/2$ degree of accuracy, were stamped onto the new flywheel. These timing marks extend for 360 degrees and they can be observed through a clear Plexiglas plate that mounts onto the cover plate. Figure 11 shows the flywheel with the magnetic pickup and steel timing tab.

The flywheel houses the steel tab that passes through the magnetic pick-up (25, Figure 10) that is held onto the block by a screw. This magnetic pick-up is out of a Ford four cylinder distributor and was modified to fit into the block. Three wires come out of the magnetic pick-up which are fed through the plate and hook up to the computer. When the tab passes through the magnetic pick-up a signal is sent to the computer. This signal gives a reference point for the timing of spark, fuel injection and a timing light that has 360 degrees of adjustment and is discussed further in section 4.3.

The flywheel has a slip fit over the end of the crankshaft and is locked in place by a keyway and the cam sprocket hub (23). The hub is outside of the block and is sealed by a shaft oil seal that is pressed into the .635 cm ($1/4$ inch) steel plate cover. The cam chain sprocket is to the one that is mounted on the cam shaft for a 1:1 drive



Figure 11. Flywheel with magnetic pick-up and steel timing tab.

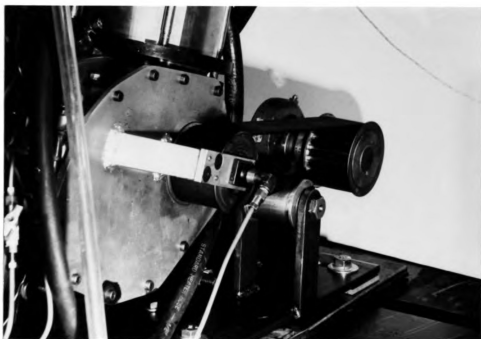


Figure 12. Crankshaft support and oil feed into crankshaft.

ratio and it is fastened to the hub by two 5/15-24 socket head cap screws. The hub is fastened to the crankshaft by a tapered fit, keyway and a bolt that screws into the end of the crankshaft.

The drive end of the crankshaft was machined with a keyway to mount a 19 grove, 1/2 inch pitch gear belt pulley on with a keyway (28) that is held on by a nut that is threaded onto the end of the crank. A 5.08 cm (2 inch) wide H200 series gear belt rides on this gear pulley for motoring the engine or transferring power from the engine to a dynamometer. A .635 cm (1/4 inch) steel plate was also used as a cover on this side of the engine (Figure 12). An oil seal was pressed into the cover plate and its lip rides on the edge of the pulley to control the oil leakage. Since there is tension on the belt, extra support was needed to reduce the side load on the crankshaft. An arm extends from the cover plate to the outside of the pulley. The inside race of a bearing has a pressed fit onto the arm (29, Figure 10). The outer edge of this bearing has a slip fit into the gear pulley and is locked with Loc-tite. This setup will support the crankshaft from side loads as shown in figure 12.

The oil source for the piston needle bearings is through the center of the crankshaft on the pulley side of the crankshaft. The oil travels through the crankshaft to the crank journal and exits through a hole directly into the needle bearings. The oil tube (30, Figure 10) protrudes

into the end of the crankshaft, through the center of the support bearing and is sealed with a small o-ring on the tip of the tube. This is held in place by a retainer clamp and screw. The oil line slips over the .635 cm (1/4 inch) hose barb that is connected to the tube and is held on by a hose clamp.

4.1.5 Variable cam timing.

As much of this engine's combustion controlling components were made as variable as possible because it is not known where the optimum timing of these components should occur because of its originality. There is total control over the throttle position, fuel injection timing and duration, spark (can even control which spark plug is firing by the computer), and the timing of the camshaft. The camshaft has 20 degrees of adjustment with respect to the crankshaft position. If more than 20 degrees of adjustment is required for the cam the chain would be moved over one tooth which would advance or retard the timing by 10.6 degrees. The crankshaft rotates counterclockwise with the left side of the chain being under tension as shown in figure 13. To make a timing adjustment the length of chain on the tension side is changed. Lengthening the amount of chain on the tension side will advance the cam timing with respect to the crankshaft and shortening the chain length will retard it. This adjustment is achieved by a threaded take-up unit (Figure 13). An idler sprocket is mounted on

the take-up unit which slides on the take-up unit frame and is adjusted by turning the adjustment knob.

On the slack side of the chain a spring loaded idler arm holds the chain taut. The arm has a Teflon slider pad which rides on the chain. An adjustable backstop supports the back of the spring loaded idler arm. At low engine speeds the cam would temporarily drive the chain when the cam lobe passed center on the rocker arm causing the slack side to become taut overcoming the spring tension on the idler arm. The adjustable backstop prevents any chain whipping or jumping on the sprockets. A steel cover encloses the cam chain assembly and is fastened in place by the four 1/4-20 studs on the timing adjuster and by the nearest 3/8-16 socket head cap screw that holds the bearing block in place.

The timing of the cam was set by rotating the cam until the cam lobe made contact with the rocker arm. Pointed timing tabs were mounted onto the bearing block (Figure 14). An indentation mark was made, in line with the timing tabs, on the cam when the cam started to engage the rocker arm. This made a known reference point of when the valves would start to open. With the mark in line with the tabs the timing marks on the fly wheel could be checked to see what the cam timing was in degrees. The cam timing could also be adjusted with the engine running and the timing checked with the adjustable timing light. Checking the timing would be accomplished by shining the timing light onto the timing

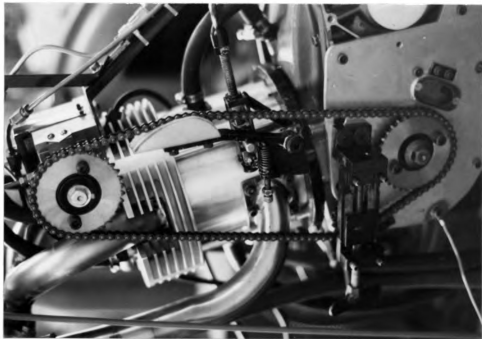


Figure 13. Cam chain with timing adjustment system.

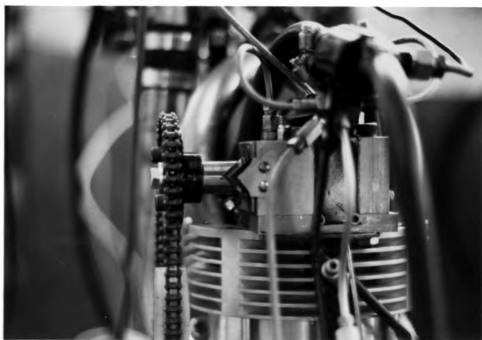


Figure 14. Cam timing mark and indicating tabs.

tabs and adjusting the timing light flash until the cam mark lined up with the points on the tabs. The timing light would then be aimed at the timing marks on the flywheel and the degrees shown would give the accurate timing of the valves starting to open.

4.2 Engine systems.

4.2.1 Lubrication system.

The engine is lubricated by an external oil source that supplies 483 kPa (70 psi) oil pressure. turbocharger. The oil source is made up of a steel container which houses a submerged gear pump that is driven by an electric motor. The pressure line from the pump feeds a manifold which is connected to six oil lines from the engine. These oil lines are made from 3/16 inch automotive brakeline tubing. An oil line supplies: 1) The turbocharger which requires a 483 kpa (70 psi) oil supply to the orifices that fit into the turbocharger body. 2) The two main cam needle bearings whose flow is regulated by a needle valve (Figure 15). 3) The cam lobe and bridge plate guide whose flow is regulated by a needle valve. 4) The two main crankshaft bearings (Figure 16). 5) The bottom side of the piston and connecting rod through a needle valve for flow control (Figure 17). 6) The pulley end of the crankshaft that has a port leading to the connecting rod bearings. The oil is fed back by gravity into the reservoir through three drain lines. One line drains the turbocharger, another drains the

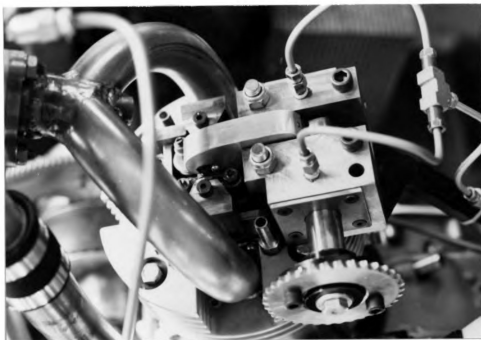


Figure 15. Oil lines supplying the the cam needle bearings.

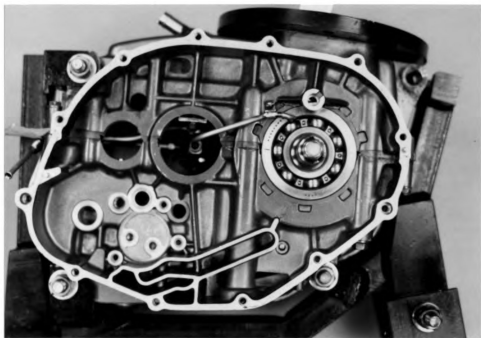


Figure 16. Lubrication of crankshaft bearings (pulley side).

camshaft assembly, and the third line drains the engine block.

4.2.2 Cooling system.

The cooling system is a closed system with a small variable speed electric water pump which circulates the water. The coolant is fed into the water jacket by two inlet ports around the cylinder. The coolant then circulates up to the engine head and exits out the four coolant ports back to the radiator. The radiator is from a Chevrolet Vega and has a variable speed range fan behind it for cooling.

4.2.3 Fuel system.

A General Motors Multec electronic port fuel injector was used in the engine. This recently developed injector sprays a fine swirled conical spray pattern created from six small angled orifices in the injector tip. The injector is supplied with 379.2 kPa (55psi) fuel pressure from a Ford, 12 volt DC, in-tank electric fuel pump. The fuel pressure is regulated by a Bosch pressure regulator from a Chrysler 2.2 liter fuel injected engine. This regulator monitors the intake manifold pressure and maintains a fuel pressure of 379.2 kPa (55 psi) above the intake manifold pressure by controlling the amount of bleed off returned to the fuel storage tank from the main fuel line.

4.3 Engine computer control system.

A Cromemco Z-80 computer running C-DOS is used to control and monitor the engine. This computer was highly modified by the addition of new circuit boards that were built to make the transition between CDOS and the actual hardware that controlled the engine components. The computer provided two main functions. The first function is the controlling and displaying of the engine parameters which include the fuel injection timing and duration, spark timing, timing light and the throttle position. The controlling of these parameters (except for the throttle position) is based on the magnetic pick-up that senses a metal tab that is mounted on the flywheel. When this metal tab passes through the magnetic pick-up, a signal is sent to the computer. The computer calculates the length of time that it takes the tab to make a complete cycle. This cycle time is set equivalent to 360 degrees, meaning that each degree is equal to a certain amount of time at that given engine speed. This measured cycle time is used to compute the location of crankshaft after the timing tab passes through magnetic pick-up until it passes through again.

The timing of the spark and fuel injection would be set at the desired amount of degrees after the timing tab. The duration of the fuel injection is run by a similar timer as the magnetic pick-up system but would be triggered by the turning on of the injector. Due to the acceleration and deceleration of the crankshaft speed in one cycle, because

of the compression stroke, the time based system could be off a few degrees depending on the crankshaft location. To solve for this inaccuracy a third fully controllable ignition system was used that had a timing light attached to it. The timing light could be adjusted to flash at the same time as the spark or fuel injection occurred. By observing the timing marks the spark or fuel injection timing could be set to occur exactly at specified time. The timing light can also be used to time the camshaft which was mentioned in section 4.1.5.

Potentiometers that are connected to the computer are used to control the fuel injection timing, spark timing, fuel injection duration, throttle position, and the timing light operation. The potentiometers are mounted on a control panel (Figure 18) with the other instruments. The position of the potentiometers will be displayed in degrees on the computer monitor. The position of the throttle would be shown in the number of increments that it was open because a stepping motor controls the throttle plate's movement.

The second function of the computer is its capability to store information. Information is stored on eight inch floppy discs that can store one Mega byte. The information that can be stored includes: 1) The fuel injection timing and duration. 2) The spark timing. 3) Which spark plug is firing. 4) The throttle position. 5) The engine's speed in rpm. 6) Torque on the dynamometer by the use of a LVT



Figure 17. Line to spray oil on the connecting rod and piston bottom.



Figure 18. Computer system with control panel.

position sensor. 7) The intake manifold pressure. 8) The exhaust temperature. Other information is stored by the addition of comments in the program which consist of the date, cam timing, fuel and oil pressure, and the oil and water temperature. All of this information is stored once each second so a disc would be full after four hours of use.

The computer system is capable of controlling and testing the engine by itself, other than for the cam timing, if a program is added to the software that would systematically control the parameters.

CHAPTER 5

TEST SETUP AND PROCEDURE

5.1 Test setup.

A series of motoring tests were conducted for this research. Firing operation of the engine will occur at a later date. The tests involved measuring the air flow characteristics of the engine at different motoring speeds and cam timings given a constant inlet boost pressure. The tests consisted of finding the cylinder pressure versus the crankshaft angle at a known air flow rate through the motored engine.

The tests involved motored operation at three different cam timings with the engine motored at four speeds. The camshaft timing was based on the degrees crankshaft angle after TDC where the exhaust valves started to open. The duration of the cam lobe lift was constant at 120 degrees. The camshaft timing was set at for valve openings of 95, 103 and 110 degrees after TDC. Since the duration of the valve opening is fixed at 120 degrees the only way to test different exhaust port openings was to change the cam timing. The three valve timings chosen were within the valve opening and closing timings shown by Taylor (1960) in a two-stroke inlet and exhaust porting table for various

two-stroke cycle engines. The valve opening timings were chosen at the advanced and retarded side of the two-stroke cycle exhaust port openings with the third valve opening chosen between the two. Table 2 shows the three camshaft timings with respect to the intake port timing and crankshaft timing for the three different cam positions.

The engine was motored by a 7.46 kW (10 HP) variable speed electric motor controlled by a Louis Allis speed controller. The electric motor drove the gear pulley on the crankshaft through a 5.08 cm (2 inch) wide gearbelt. The engine was motored at speeds of 420, 840, 1200 and 1620 rpm. The maximum speed of 1620 rpm was chosen because the motor and controller could maintain a very stable speed up to this rpm. The speed was incremented in about 400 rpm increments down to 420 rpm.

5.1.1 Air flow system.

In order to calculate the boost air that flows through the engine at standard conditions of 21.1 degrees centigrade and 76 cm of mercury absolute (70 degrees F and 29.92 inches Hg) specific information about the air flow was required. An air flow measurement system was set up to display the flow rate of air in cubic feet per minute (cfm), the temperature in Celsius, and the pressure of the boost air in inches of water as shown in figure 19.

Boost air was supplied by a Spencer 2.34 kW (3 HP) four stage centrifugal blower that produced about 10.34 kPa (1.5

Table 2. Camshaft timing in respect to the intake ports and crankshaft angle.

<u>Camshaft and intake timing in relation to BDC.</u>				
Degrees exhaust valve opening after TDC.				
	<u>95</u>	<u>103</u>	<u>110</u>	<u>Intake timing</u>
Degrees open before BDC.	85	77	70	55
Degrees closed after BDC.	35	43	50	55
 <u>Camshaft timing with respect to intake ports.</u>				
Degrees exhaust valve opening after TDC.				
	<u>95</u>	<u>103</u>	<u>110</u>	
Degrees exhaust valves open before intake port opening.	30	22	15	
Degrees exhaust valves close before intake port closes.	20	12	5	

psi) of filtered boost air to the engine through the air sensing system. The volume of air in standard cubic meters per minute (standard cfm) that the engine received under motoring conditions was calculated with data collected by using input from a laminar air flow element, two manometers, a barometer, and a temperature sensor.

A Meriam laminar flow element, model 50 MC2-4 rated at 11.33 scmm (400 scfm) and 20.32 cm (8 inches) of water differential was mounted between two 1.02 m (40 inch) long 10.16 cm (4 inch) O.D. pieces of aluminum tubing. This tubing was fastened upstream and downstream of the element to produce a laminar flow through the element. Meriam required that this straight tubing have a length which is at least ten times the diameter of the element ends. A hose fitting on either side of the element allowed hoses to be connected to a Meriam inclined tube manometer which displayed the pressure differential across the laminar air flow element. The inclined tube manometer used was model 40HE35WM with a maximum capacity 20.32 cm (8 inches) of water differential which would allow for 11.33 cmm (400 cfm) to flow through the element. Both the air flow element and inclined manometer tube were calibrated at 21.1 degrees Celsius (70 degrees F) and 76 cm (29.92 inches) of mercury absolute.

The pressure of the boost air was measured at the upstream side of the laminar flow element and at the engine's intake manifold with a Meriam well type manometer

model 30EB25 with a 88.9 cm (35 inch) scale. The manometer well was filled with a Meriam 175 blue indicating fluid with a specific gravity of 1.75. A hose connected the well of the manometer was to a three way valve which could be switched to either the engine's intake manifold or to the laminar flow element.

The temperature of the boost air upstream of the laminar flow element was measured with a wire thermocouple (ANSI type T copper/constantan wire) that was mounted in the aluminum tube just before the laminar flow element. The thermocouple was connected to an OMEGA DP80 series digital monitor.

A 0.148 cubic meter (39 gallon) plenum was mounted in the air stream between the engine and the end of the straight aluminum tube downstream from the laminar flow element. This plenum acted as an accumulator to dampen out any flow pulses caused from the motored engine so that the accuracy of reading the manometers was improved.

In order to convert the information on the air flow to scmm (scfm) the use of correction tables supplied by Meriam were required. Both the laminar flow absolute element inlet pressure and temperature were needed to convert the cubic meters per minute (cmm) (cubic feet per minute (cfm)) to the standard conditions of the calibrated air flow equipment by the use of the correction factors in the tables. The laminar flow element inlet pressure was added to the barometric pressure of that day, and then converted with a

correction factor. Both the temperature correction factor and the laminar flow element inlet absolute pressure correction factor were then multiplied by the cmm (cfm) read from the inclined manometer to achieve the standard cmm (cfm).

5.1.2 Pressure and crank angle sensing system.

A Kistler quartz pressure transducer model 6121A2 was inserted into one of the sparkplug holes through an adapter. A low noise coaxial (insulation resistance of 10^{14} Ohms) cable connected the pressure transducer to a Kister dual mode amplifier model 5004 where the calibrated ($-.972$ pico Coulombs / 6.895 kPa ($-.972$ pC / psi) at 0 to 2413 kPa (0 to 350 psi) range) pressure transducer charge was converted to a 344.73 kPa / volt (50 psi / volt) signal through the charge amplifier.

The crankshaft angle was sensed by a BEI model H25D shaft angle encoder which was directly coupled to the crankshaft. The encoder was used to calculate the engine speed and to assign a crank angle to the pressure reading. This encoder has a resolution of 1024 pulses per revolution with a once per revolution reference pulse. The reference pulse was set to occur with the piston at TDC. The signal from the pressure amplifier was stored on an eight inch floppy disc every eight counts of the encoder which is equivalent to 2.81 degrees of the crankshaft rotation.

5.2 Test procedure.

The following procedure was used to run the engine tests. First the pressure transducer was calibrated and the signal monitored with the computer system. The pressure amplifier was turned on about an hour before the tests were run to allow the electronic components to warm up and stabilize. When the amplifier was warmed up, the engine was rotated until the exhaust valves were opened to achieve atmospheric conditions in the cylinder. With the engine held in this position the pressure transducer cable was then grounded and reconnected to the amplifier in the reset position. This eliminated any stored charge in the pressure transducer and provided a known reference.

With the encoder disconnected from the crankshaft the computer was turned on and set to record data. The encoder was rotated to simulate the engine's being motored with the cylinder pressure constant at atmospheric pressure. The computer displayed and stored the cylinder pressure, a value of 0 to 255, after every eight counts of the rotating encoder. With the encoder being rotated with the exhaust valves open, a stable value near zero was displayed. The value was very low and stable. This value was recorded as zero kPa (psi). This process was repeated except that the amplifier was disconnected from the computer and the voltage supplied by the computer was read along with the value displayed from rotating the encoder. This displayed

value was very high. By knowing the voltage from the computer with the amplifier disconnected, the pressure at that voltage was computed by multiplying the voltage by 344.73 kPa (50 psi) as the amplifier was set to produce an output signal that was equivalent to 344.73 kPa / volt (50psi / volt). An equation to convert the stored test data was formulated by knowing that zero kPa (psi) equaled a certain display value and that the disconnected voltage equaled a pressure value equivalent to a greater display value. To convert the test data 0 - 255 value into actual pressure the stored test data are then subtracted from the zero display value and then multiplied by the constant (pressure of the disconnected voltage divided by the disconnected display value).

The next step was to remount the encoder onto the crankshaft and set the once-per-revolution pulse to occur at TDC. This was achieved by setting the engine timing marks at TDC and rotating the encoder shaft with the coupling loose. The voltage from the encoder, supplied by the computer, read zero until the once-per-revolution pulse was located by observing the voltage shift to 5 volts. With the encoder outputting 5 volts the coupling was tightened to lock the encoder to the crankshaft.

Changing the timing of the camshaft followed if it was necessary. To change the cam timing the idler sprocket, on the tension side of the cam chain was moved simultaneously with the slack side adjustment. To advance the cam timing,

the chain was lengthened on the tension side and retarding the timing occurred from shortening the length. The chain was shifted until the timing mark on the camshaft lined up with the timing tabs (figure 14) and the degrees on the flywheel read the cam timing desired. The teflon slack take-up was then adjusted to eliminate any chain slack. The engine was then rotated several times and the cam timing rechecked to see if the adjustment was accurate.

The last step was to return the engine to the position where the exhaust valves were open and to rezero the pressure transducer. After these procedures, a test was ready to begin. A new data file was opened in the computer and data collection was started when the encoder was rotated. At this point the blower, oil system and engine coolant were turned on. Just before the engine was motored the ground switch on the charge amplifier is switched to the operate position. The entire engine system, except for the blower system, is shown in figure 20.

The engine was motored at each of the four set speeds until the speed was stable. At a stable speed the air flow data was collected. After the data was collected for the four speeds, the electric motor, blower and oil supply were turned off. The cam timing was then changed and the pressure transducer was reset before the next test. A total of twelve tests were run with each of the three cam timings being motored at the four selected speeds.

The pressure transducer charge amplifier drifts with

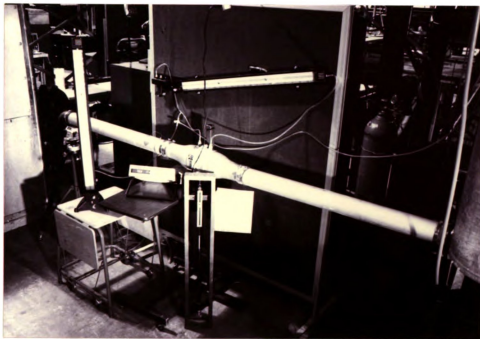


Figure 19. Boost air supply and monitoring system.

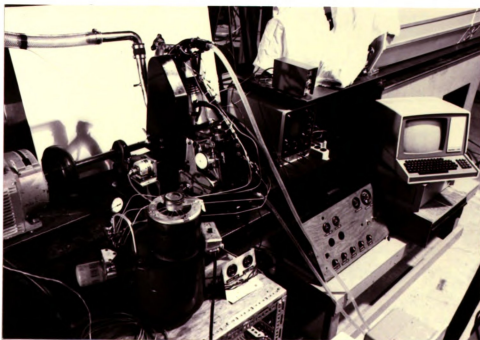


Figure 20. Engine controlling and monitoring system.

time. The pressure values were valid but all of the values were shifted an equal amount over a period of time. In order to correct this problem a once-per-engine revolution reset was required to reset the amplifier at a desired crankshaft angle. This device wasn't available so the data values were shifted to their proper pressures by a reference pressure in each cycle. When the piston is at BDC, the intake ports are fully open and the exhaust valves are starting to close. It was estimated that the cylinder pressure should be equal to the intake manifold pressure, a pressure around 6.89 kPa (1 psi), when the piston is at BDC. With the known intake manifold pressure all of the data points were then shifted until BDC equaled 6.89 kPa (1 psi), the boost pressure.

CHAPTER 6

RESULTS AND DISCUSSION

6.1 Pressure data.

The cylinder pressure data are shown in graph form by plotting the cylinder pressure on the y-axis and the crankshaft angle on the x-axis. The data shown in a single curve are for a single engine cycle. The cycle chosen was toward the end of the test for that particular testing speed. This gave the engine and air flow time to stabilize so the chosen cycle had consistent values compared to the cycles preceding and following the chosen cycle. Figures 21 through 23 represent a single cam timing motored at the four test speeds. Figures 24 through 27 compare the three different cam timings at the same motored speed.

The three different cam timings followed the same pattern for the peak cylinder pressures as shown in table 3. The lowest peak pressures occurred with the engine motored at 420 rpm. The peak pressure increased until a maximum was reached at 1200 rpm then it dropped slightly to a value close to the 840 rpm results at 1620 rpm. The reason for the pressure decline at 1620 rpm is because a smaller mass of air was trapped in the cylinder for compression at the higher motored speed.

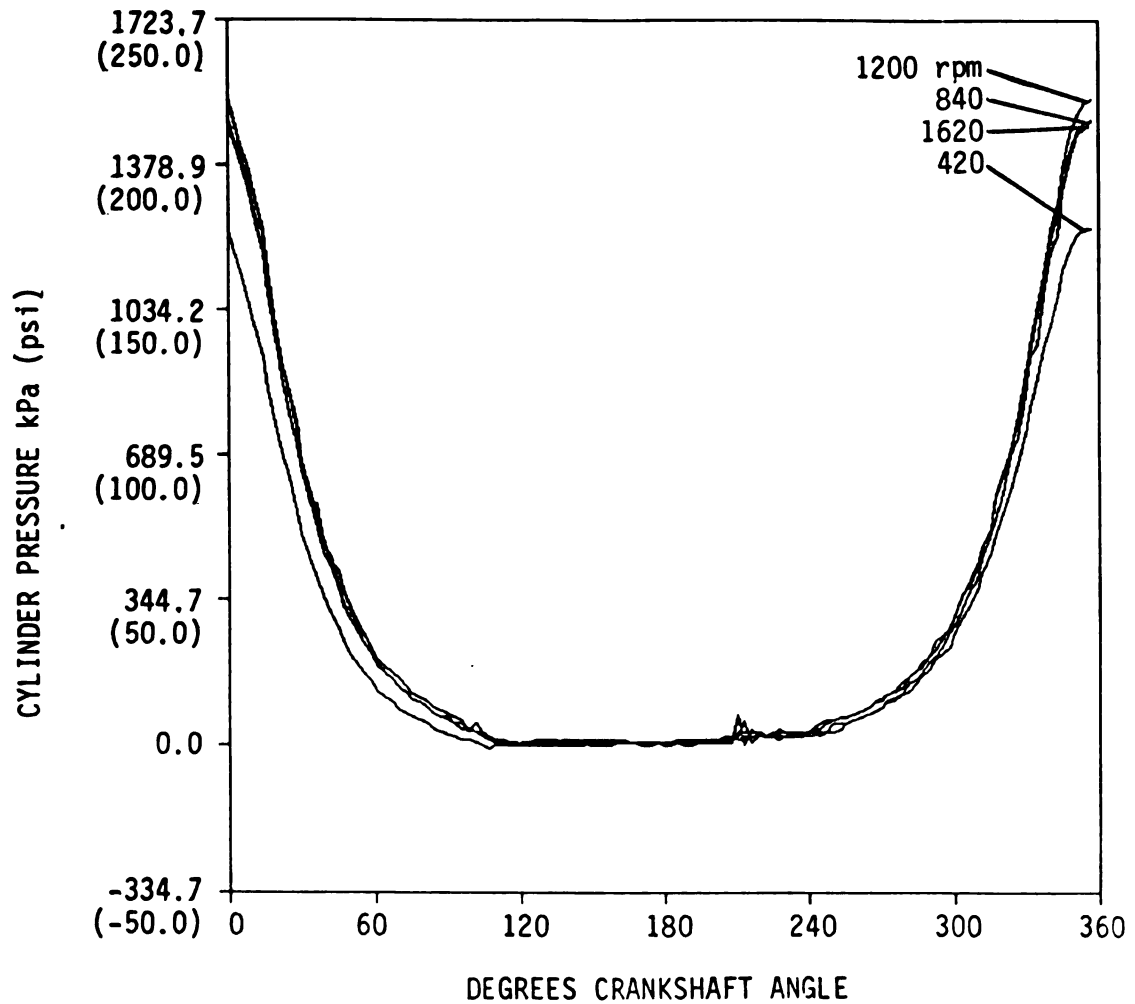


Figure 21. Cylinder pressure verses crankshaft angle at 95 degrees valve opening after TDC and motoring speeds of 420, 840, 1200, and 1620 rpm.

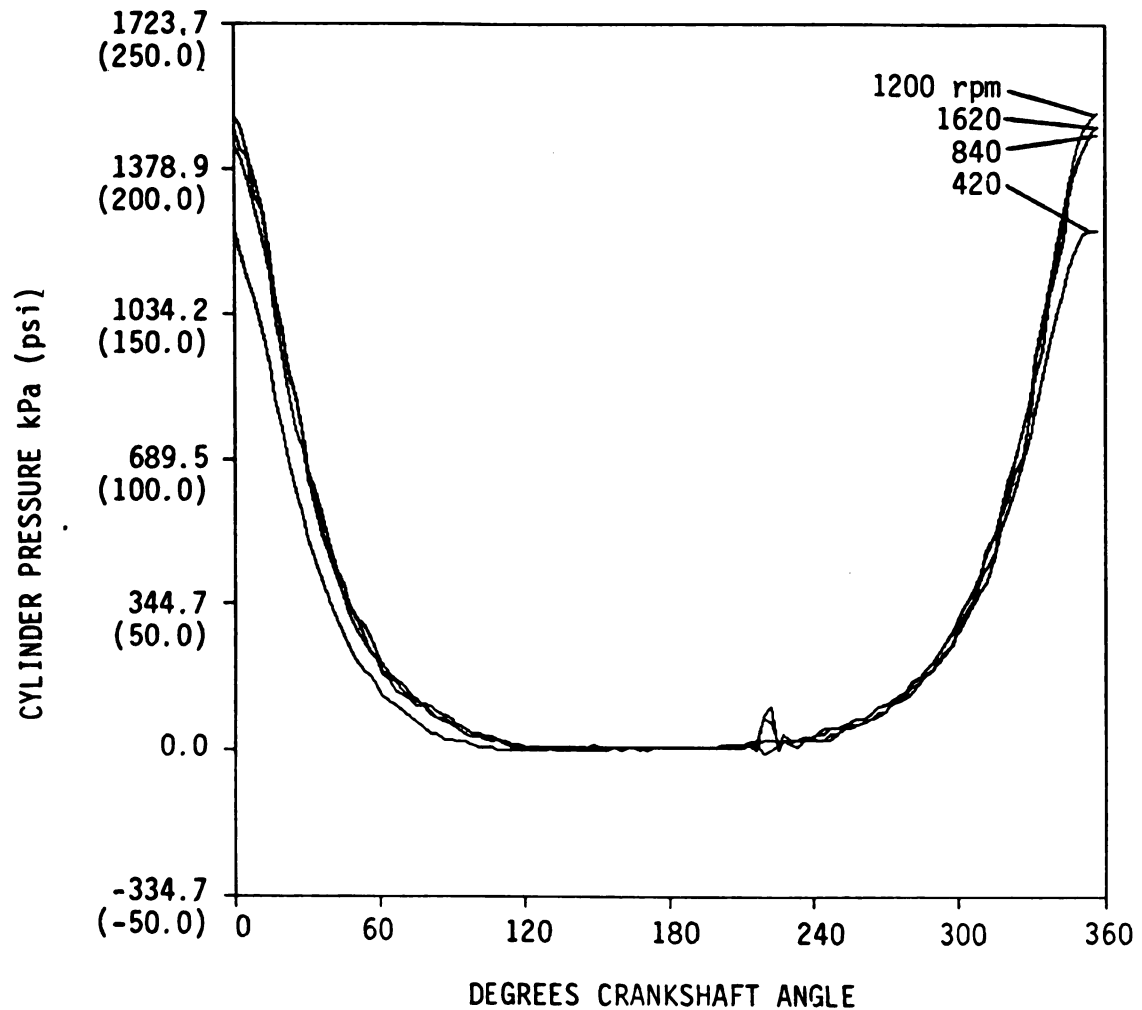


Figure 22. Cylinder pressure verses crankshaft angle at 103 degrees valve opening after TDC and motoring speeds of 420, 840, 1200, and 1620 rpm.

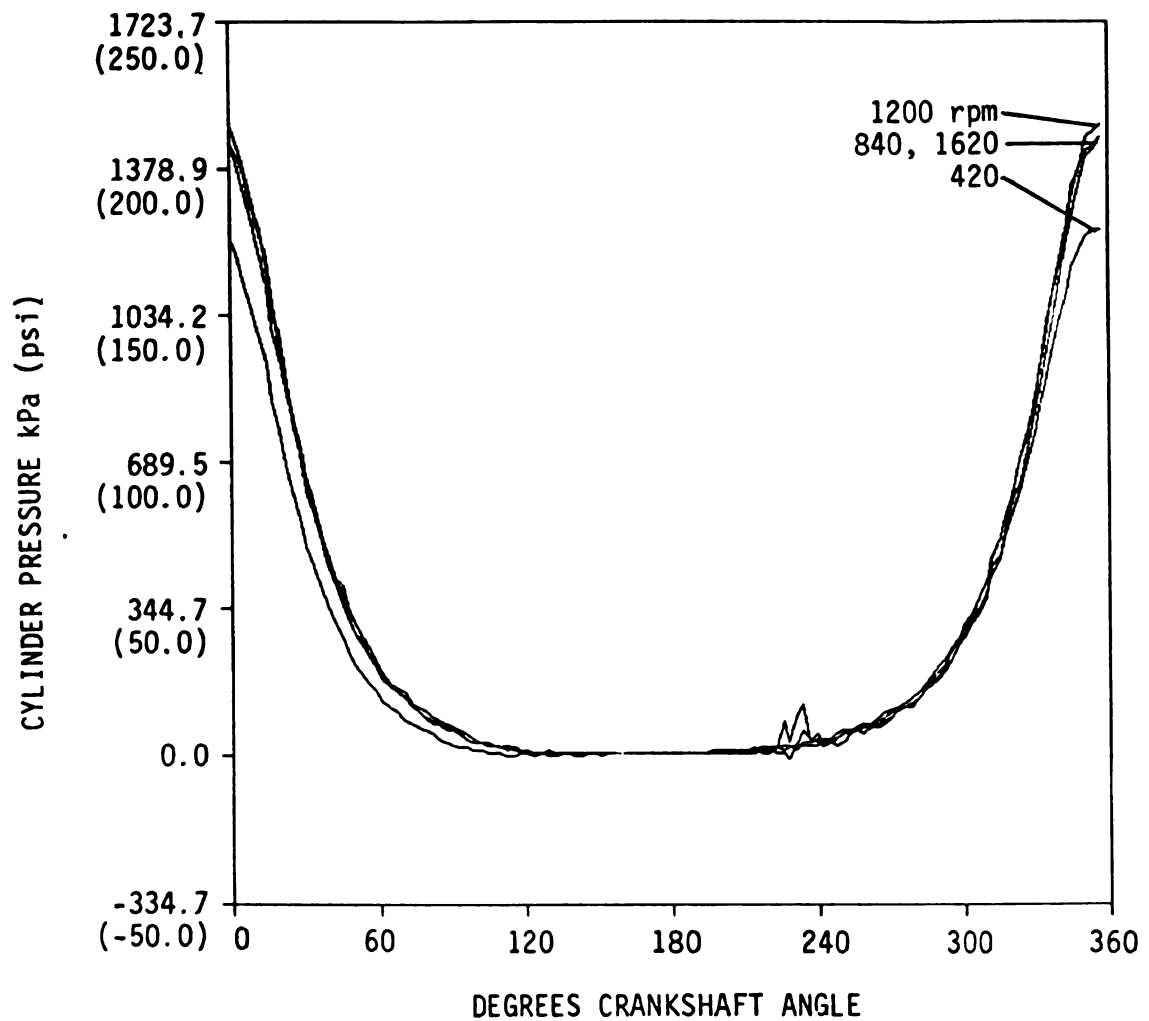


Figure 23. Cylinder pressure verses crankshaft angle at 110 degrees valve opening after TDC and motoring speeds of 420, 840, 1200, and 1620 rpm.

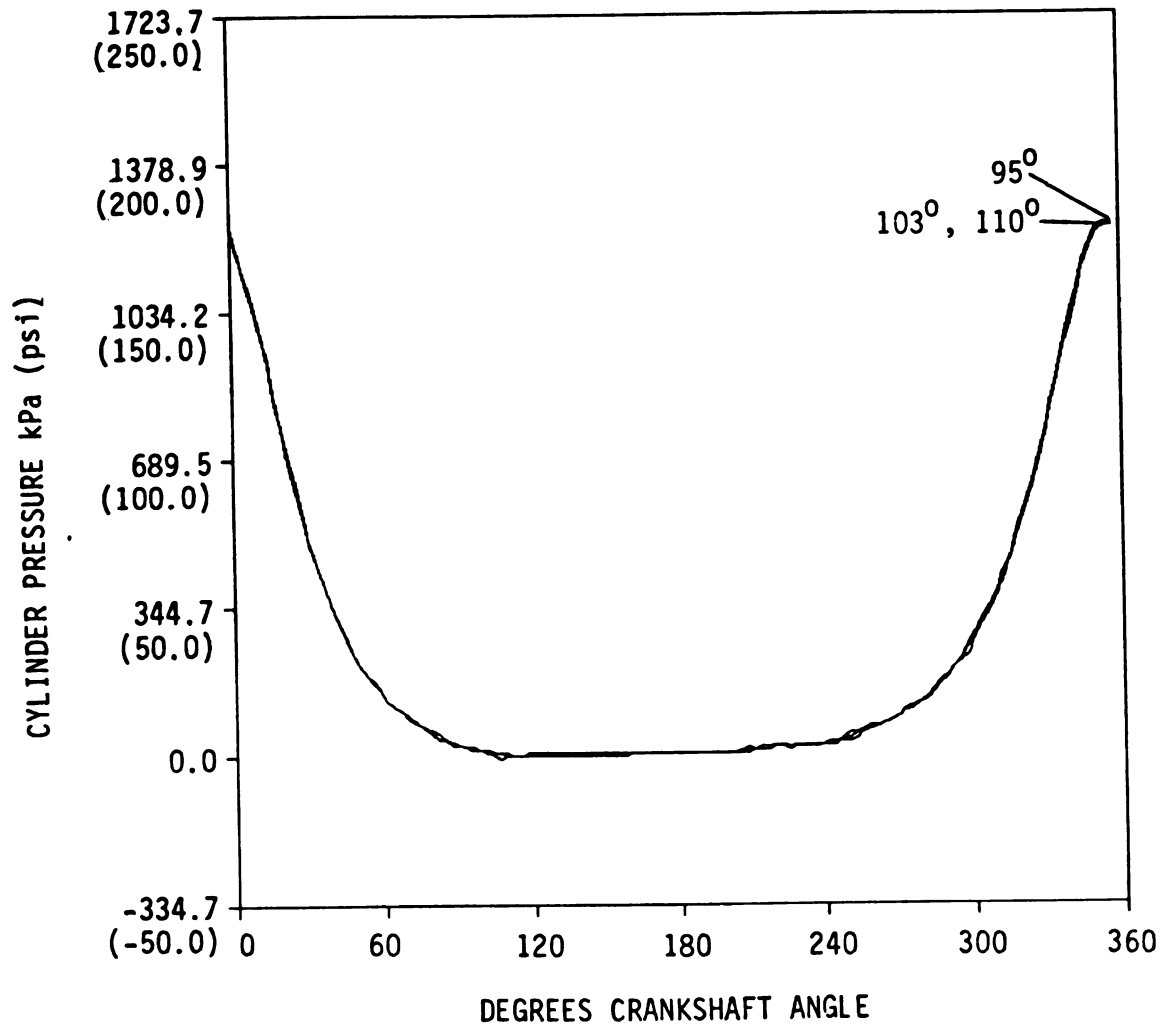


Figure 24. Cylinder pressure verses crankshaft angle with a motoring speed of 420 rpm and valve openings at 95, 103, and 110 degrees after TDC.

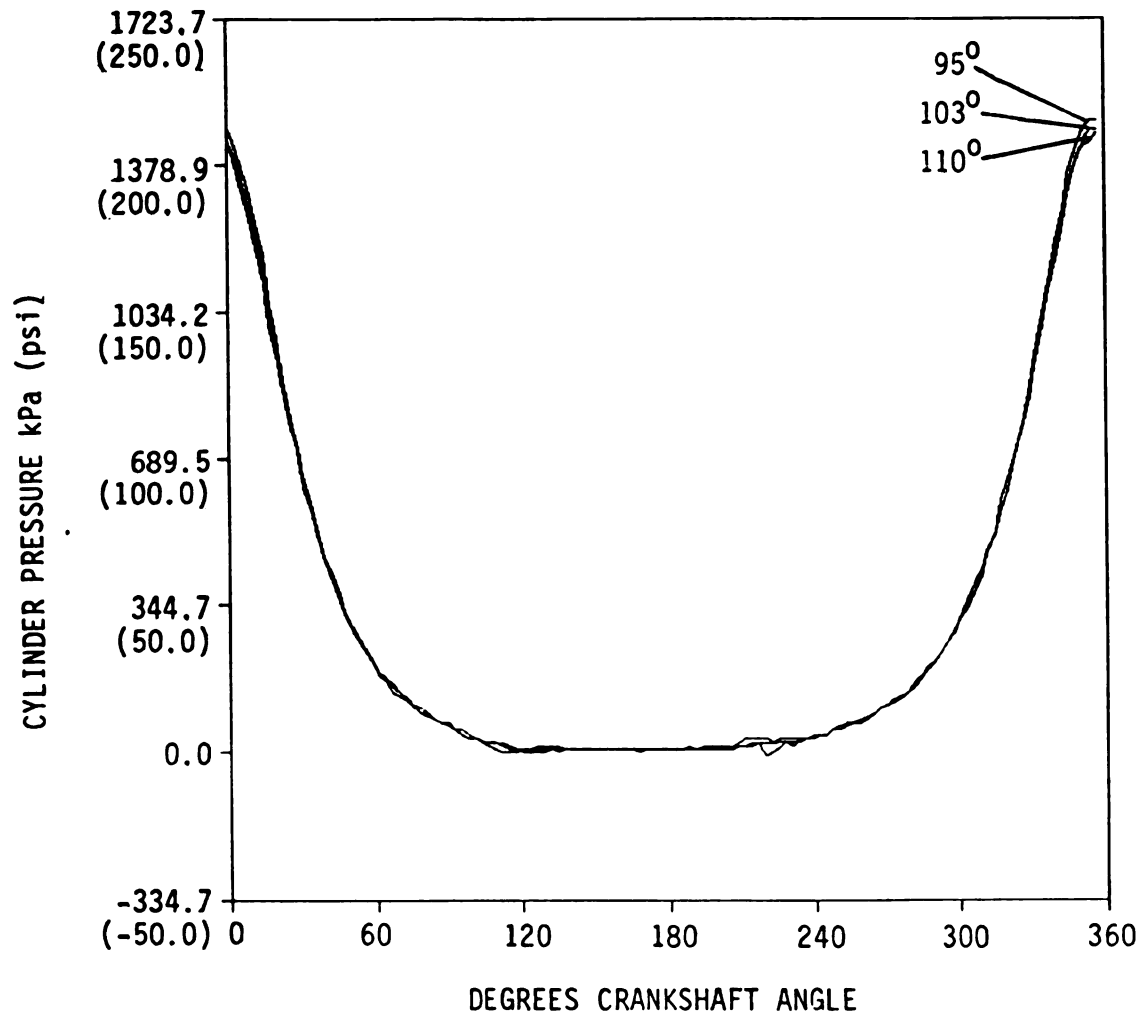


Figure 25. Cylinder pressure verses crankshaft angle with a motoring speed of 840 rpm and valve openings of 95, 103, and 110 degrees after TDC.

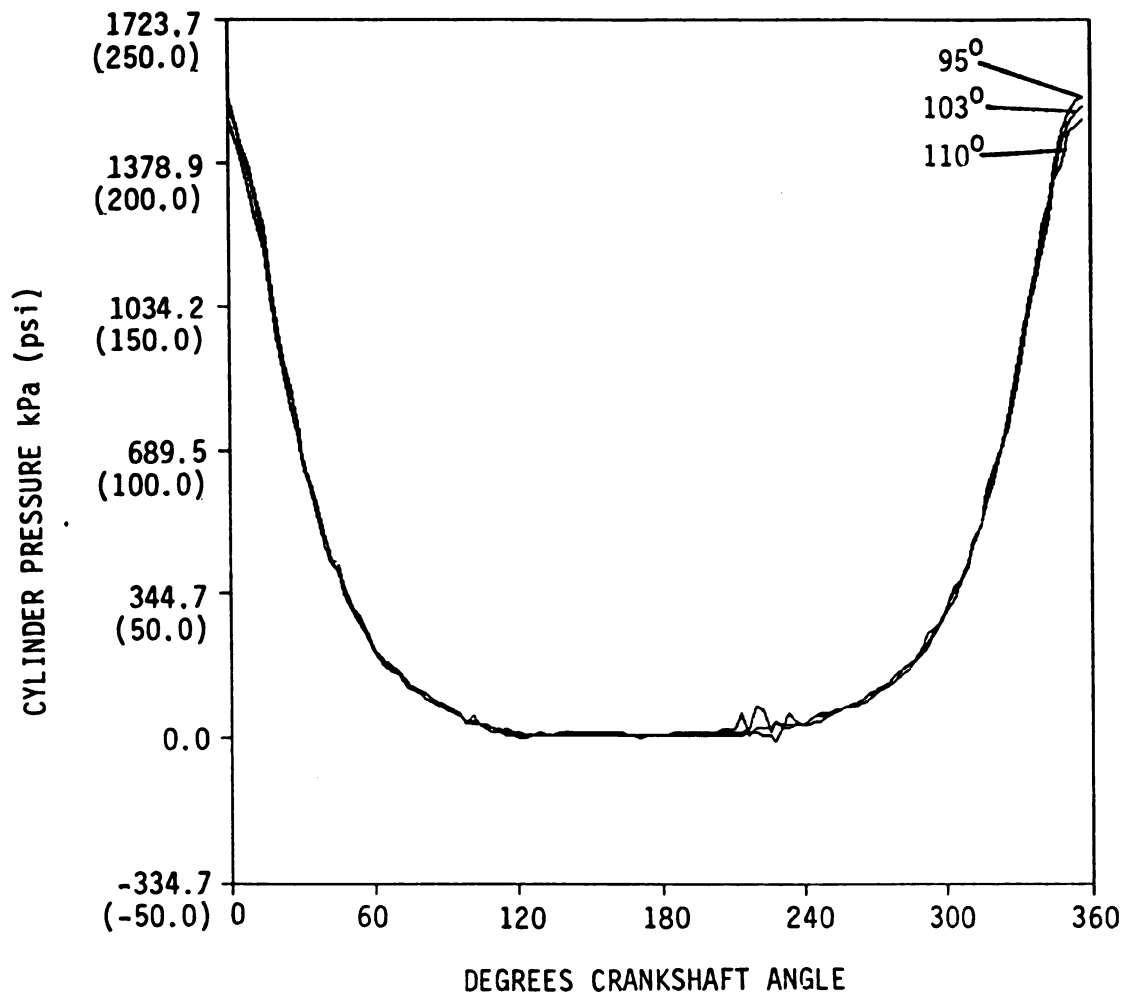


Figure 26. Cylinder pressure verses crankshaft angle with a motoring speed of 1200 rpm and valve openings of 95, 103, and 110 degrees after TDC.

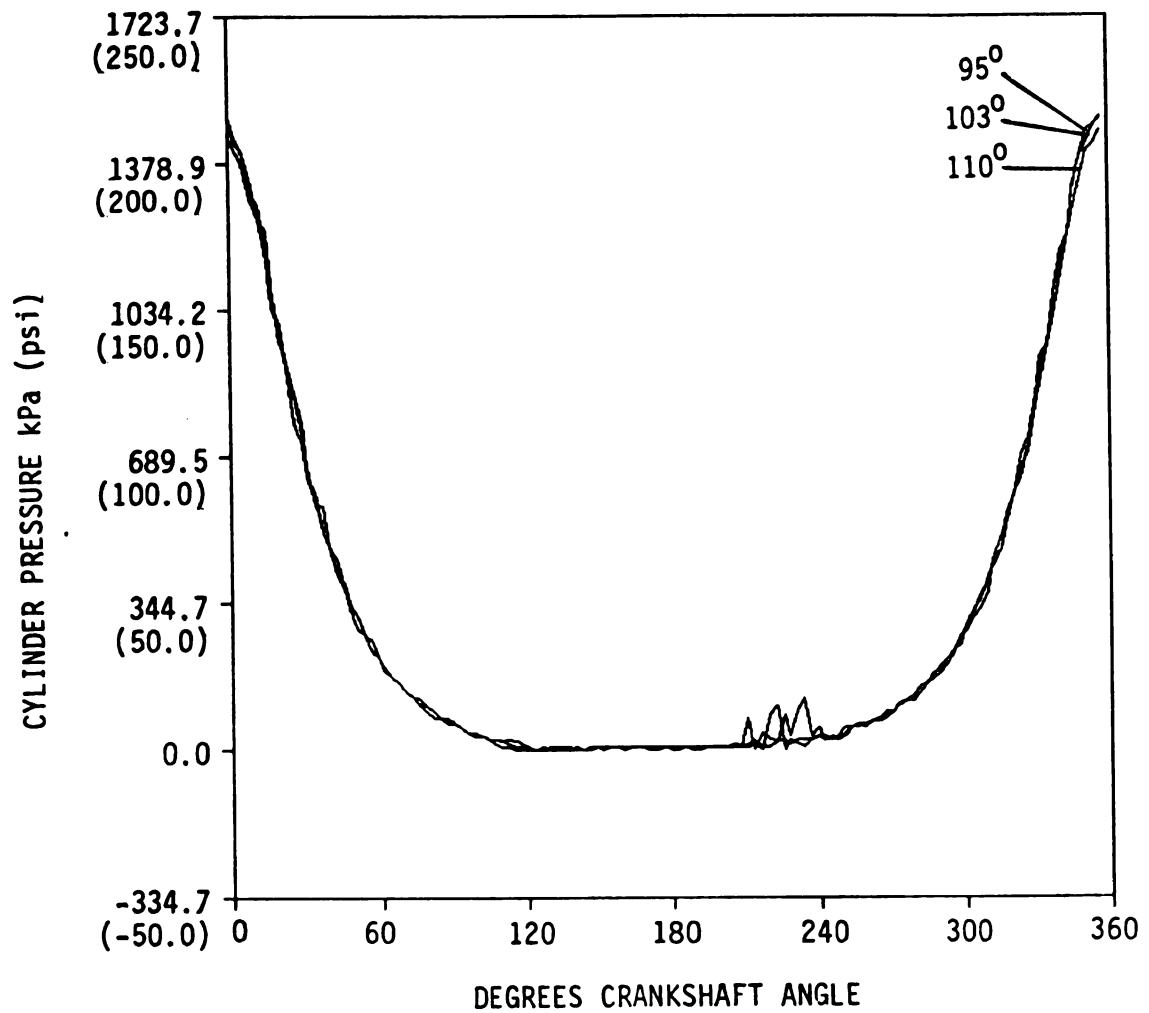


Figure 27. Cylinder pressure verses crankshaft angle with a motoring speed of 1620 rpm and valve openings of 95, 103, and 110 degrees after TDC.

A lower peak cylinder pressure is expected at lower engine speeds because at the slower compression rate the trapped air has more time to leak past the compression rings. The values of the peak cylinder pressures would change slightly after the engine was run for a period of time long enough to seat the rings.

The cam timing had a direct effect on the peak cylinder pressures. The peak cylinder pressures were the highest for the valve timing of 95 degrees after TDC and decreased with the retarding of the cam timing for all engine speeds. This shows that less air is trapped in the cylinder for compression when the cam timing is retarded. By closing the exhaust valves earlier the boost air can fill the cylinder more completely without passing through the exhaust ports. Table 2 shows the cam timing with respect to the intake port.

The data showed that each engine cycle had a spike in the cylinder pressure which increased with the engine speed at a point just before the exhaust valves close. It is not known what the cause is but it can only be one of two different phenomena. One possibility is that the pressure spike is caused from the valves slamming shut thus causing a sudden shock on the pressure transducer that disturbs the pressure signal. The fact that it occurs before the calculated closing time of the valves might possibly be from the cam chain slack take-up not working properly. If the cam drove the chain after the cam lobe passed center on the

rocker arm (caused by the force of the valve springs) and the slack take-up wasn't performing properly, the cam could advance itself with respect to the crankshaft. The second possibility is that the flow of air through the cylinder is being stopped abruptly when the exhaust valves close and a pressure wave is created in the cylinder.

It was also observed that there were torsional vibrations from the crankshaft/connecting rod assembly when motoring the engine near 800 rpm. Once the engine was motored past this speed the vibrations were minimal up to 1800 rpm which was the maximum motoring speed that the engine was brought to. The engine will be balanced before the actual running tests are performed.

6.2 Air flow data.

Table 3 shows the Liters per cycle (sq. inches per cycle) of air that the engine required at the set motoring speeds and boost for the three cam timings. The scmm (scfm) of boost air that the motored engine consumed ranged from 0.179 to 1.036 l/cycle (10.94 to 63.19 in²/cycle).

The valve timing had a large impact on the air flow characteristics. A valve opening of 95 degrees after TDC had the lowest air consumption and was the most beneficial for creating the highest peak cylinder pressures. By reducing the air consumption at the more advanced cam openings, the scavenging process would also be shortened which may or may not be beneficial for the running engine.

Table 3. Flow rate of boost air and peak cylinder pressures at different motoring speeds and cam timings.

95 degree cam timing after TDC.						
<u>RPM</u>	<u>Peak cylinder pressure</u>		<u>Air flow</u>		<u>Boost pressure</u>	
	<u>kPa</u>	<u>(psi)</u>	<u>l/cycle</u>	<u>(in³/cycle)</u>	<u>kPa</u>	<u>(psi)</u>
420	1227	(178)	0.847	(51.62)	9.17	(1.33)
840	1482	(215)	0.331	(20.20)	9.31	(1.35)
1200	1538	(223)	0.179	(10.94)	9.51	(1.38)
1620	1489	(216)	0.210	(12.79)	9.45	(1.37)
103 degree cam timing after TDC.						
420	1234	(179)	0.953	(58.13)	9.10	(1.32)
840	1462	(212)	0.386	(23.55)	9.17	(1.33)
1200	1517	(220)	0.229	(13.96)	9.45	(1.37)
1620	1482	(215)	0.231	(14.07)	9.17	(1.33)
110 degree cam timing after TDC.						
420	1234	(179)	1.036	(63.19)	9.10	(1.32)
840	1455	(211)	0.443	(27.03)	9.24	(1.34)
1200	1482	(215)	0.238	(14.53)	9.38	(1.36)
1620	1455	(211)	0.229	(14.00)	9.17	(1.33)
Calculated trapped cylinder volume.						
	<u>Liters/cycle</u>		<u>(in³/cycle)</u>			
	0.32		19.49			

The reduction of the air flow was the result of closing the exhaust valves earlier. This stopped the air flow through the exhaust and trapped a greater mass of air for compression before the inlet ports closed. The opposite case was true for retarding the cam timing. The valves remained open longer prolonging the scavenging process and resulting in less boost air being trapped in the cylinder for compression. By calculating the trapped cylinder volume of the engine from the top edge of the inlet transfer ports to the bottom side of the cylinder head the liters (cubic inches) of air required under perfect scavenging conditions could be solved for at a given rpm shown in table 3. When the engine displacement volume is compared to the actual air consumption, at different motoring speeds and cam timings, it is shown that at 420 rpm there is excessive scavenging taking place. As much as 2.5 to 3 times the trapped cylinder volume of air is being blown through the cylinder which shows that there is excessive scavenging. This is caused by the long period of time that the inlet ports are open at the low rpm with the constant boost. The ports are open for 44.7 milliseconds per cycle at this speed.

The air flow at 840 rpm seemed to match the displacement flow more closely, especially with the cam timing of 95 degrees after TDC. This shows that the boost pressure was too high for the 420 rpm speed to have efficient scavenging but well matched for the 840 rpm speed. At 1200 rpm the recorded air flow fell short of the

displacement volume. At this point the boost pressure was too low to supply enough air at the higher speed with a shorter inlet opening time of 15.3 milliseconds per cycle. The lowest air flow case for 1200 rpm was with the valve timing of 95 degrees after TDC where the scavenging timing was cut shorter. The engine's air consumption went up when the speed was increased to 1620 rpm. The inlet ports were only open for 11.3 milliseconds per cycle but the increased number of cycles per minute had to cause the air flow to increase because the boost remained the same. Again the lowest flow rate occurred at the 95 degree cam timing.

CHAPTER 7

SUMMARY AND CONCLUSIONS

Motoring tests were run on a single cylinder two-stroke cycle gasoline engine with poppet valve exhaust. The following are conclusions derived from designing and testing of the engine:

1. The closing of the exhaust transfer ports before the inlet ports closed was achieved by constructing a four poppet valve exhaust system in the engine head. With this system fine adjustments in the valve timing were possible which showed that a few degrees change in the cam timing significantly affects on the dynamic characteristics of the engine.
2. In order to supply atomized fuel to the engine in a controlled manner a General Motors Multec port fuel injector was mounted into one of the eight angled inlet ports in the cylinder wall. This injector sprays a spiraled conical fuel pattern out of six small orifices directly into the cylinder. When this injector is used with a timing circuit and a mass air flow sensor the fuel injection can be timed and metered to produce the desired equivalence ratio.
3. The oil entering the combustion chamber was minimized by having a standard four stroke cycle lubrication system

and supplying the boost air with a blower. Also a set of oil rings were placed on the piston skirt so that oil couldn't enter into the inlet transfer ports.

4. The peak cylinder pressures increased with the engine speed until it reached a maximum at 1200 rpm then dropped slightly at 1620 rpm. The higher speeds allowed less leakage past the piston rings and less time for heat absorption from the compressed air into the combustion chamber walls. At the higher motored speed however the boost time appeared to be too short to acquire the same trapped mass as in the 1200 rpm tests.

5. The peak cylinder pressures increased as the cam timing advanced from 110 to 95 degrees after TDC. This allowed a larger mass of air to be trapped in the combustion chamber.

6. Changing the valve timing had an important impact on the consumption of boost air through the engine. Less air was consumed when the cam timing was advanced to 95 degrees after TDC which closed the valves sooner. The valve timing also effects the scavenging of the engine because of the changing air consumption.

7. A variable boost supply would be needed to run the engine properly. The peak cylinder pressures and air flow indicates that the boost pressure should be minimal at low speeds but increase as the engine speed increases.

CHAPTER 8

RECOMMENDATIONS FOR FURTHER STUDIES

Before the engine is fired the following steps should be accomplished.

1. The engine's crankshaft and piston/connecting rod assembly should be balanced to eliminate the torsional vibrations at the engine speed of 800 rpm.
2. A mass air flow sensor needs to be mounted in the air inlet system and integrated with the computer and fuel injection system.
3. The fuel injector would have to be calibrated in order to supply the proper amount of fuel to the mass of air supplied to the engine.
4. A shaft angle encoder would need to be mounted on the camshaft along with a liner velocity displacement transducer mounted on a valve. With these two instruments it would be known what the exact location of the camshaft and valves would be in respect to the crankshaft angle in order to determine the cause of the cylinder pressure spike when the valves closed.
5. A variable boost source that would increase the air supply with an increase in engine speed would be required. This would reduce the air losses at low engine speeds and

supply the needed boost at the higher engine speeds in order to trap the same mass of air in the cylinder.

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