PERFORMANCE TESTING A 30 CUBIC INCH BOURKE ENGINE

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AMONG ALL OF the interesting and unusual internal combustion engines invented in the last 70 years, the engines designed and built by Russell Bourke must rank as some of the most controversial. Operating on the same mechanical principles, all the engines Bourke built during his lifetime were, according to Bourke, capable of remarkable horsepower outputs and thermal efficiencies. Unfortunately, Bourke died in 1968, having left behind no documented evidence of his engines' amazing performance capabilities. However, a compilation of his writings were published in a book entitled **Bourke Engine Documentary**; but while this serves to heighten interest, it does not provide for any detailed information concerning engine design, construction or performance. Of the many devotees and enthusiasts who have obtained some of the original engines or constructed their own Bourke engines, few have claimed to duplicate Bourke's stated performance and running characteristics, and to this day there is no known case of any of these individuals publishing any information on their findings.

My interest in the Bourke engine began in the spring of 1972 after reading an article about Russell Bourke in an EAA publication entitled **Engines, Volume II.** Some of the more interesting points discussed in the article that really stirred my curiosity was the engines' claimed ability to develop a great deal of horsepower with excellent fuel economy, while operating on lean air/fuel ratios on low octane fuels. Such claims prompted me to ask myself why wasn't such a remarkable engine in more widespread use and could it be possible that no one recognized the virtues of this discovery made by Bourke? I was a first-year engineering student at this time and since my interest pertained to engines of all types, especially two-strokes, the Bourke seemed to offer itself as an excellent college research project.

At the 1972 EAA Annual Fly-In at Oshkosh, Wisconsin, I was fortunate enough to meet a gentleman named Nicholas Fucille who informed me that Mr. Bourke was survived by his wife, Lois, and from her I would be able to obtain a copy of the **Bourke Engine Documentary**. I was also informed of the existence of a small manufacturing firm in Boulder City, Nevada by the name of Vaux Enterprises that would build a 30 cubic inch Bourke engine replica on special order. That evening I wrote to Vaux Enterprises requesting information and prices on their engine, and ordered the documentary from Lois Bourke.

It was through many hours of studying this documentary that I became acquainted with the Bourke theory of combustion and the principles by which he designed all of his engines. According to Bourke, an instantaneous explosion, or detonation of the air/fuel mixture in a suitably designed spark-ignition engine would release far more usable energy than progressively burning the mixture as is the practice in the normal combustion process. The detonation of a lean air/fuel mixture was, so Bourke thought, the key to developing high power outputs with low fuel consumption. Consequently, it was Bourke's firm belief that detonation should be encouraged and not surpressed as is now done by doping fuels with anti-knock additives. Thus, Bourke's sole aim was to design an engine that would run on the detonation of a low-grade, hydrocarbon fuel. In order to promote this combustion phenomena, Bourke designed his engines to operate with high compression ratios, low grade, low octane fuels, lean air/fuel ratios, considerable mixture preheating, and highly advanced spark timings. It was a well-known fact, even in Bourke's day, that a conventional spark-ignition engine will tolerate detonation only for brief periods of time without incurring severe mechanical damage to pistons and valves. This damage is caused by the high heat and pressures associated with the detonation phenomena. Bourke's solution to this problem was to eliminate the articulating connecting rod used in conventional engines by employing a modified Scotch-yoke crank mechanism coupled with the two-stroke engine cycle. The result was a lightweight and structurally strong engine configuration. Bourke's modification of the Scotch-yoke consisted of replacing the traditional crankpin slider block with a roller assembly, which greatly decreased friction and made this crank mechanism applicable to a high-speed engine.

Figure 1 shows the basic arrangement of the Bourke 30 cubic inch engine, which consists of two horizontally opposed cylinders attached to a common crankcase. Conventional piston controlled inlet, transfer, and exhaust ports are employed, as used in standard two-stroke prac-

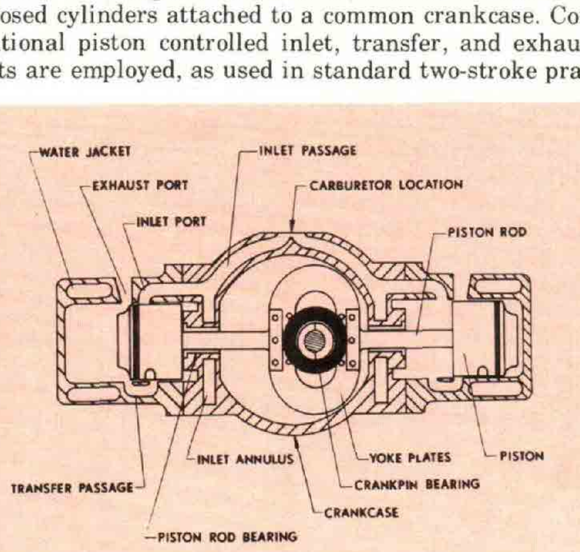

FIG. 1

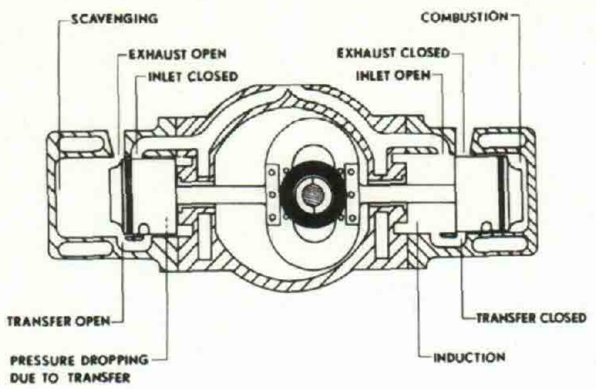

FIG. 2

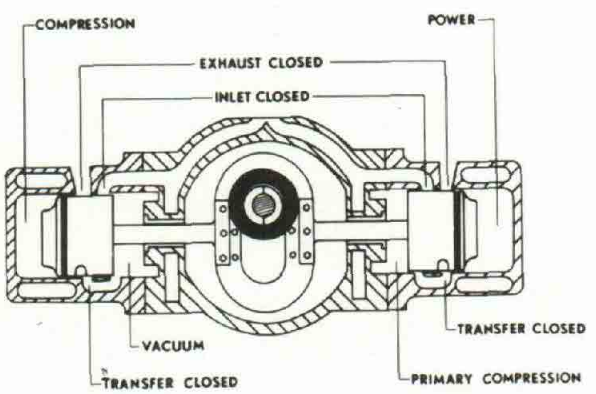

FIG. 3

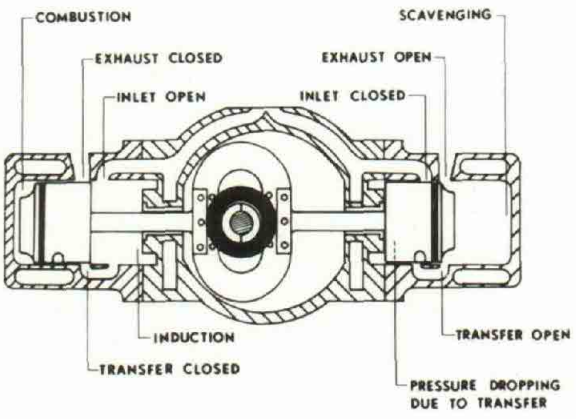

FIG. 4

tice. The cylinders are cross-scavenged (necessitating deflector-type pistons), water cooled and are isolated from the crankcase interior. Hence, the crankcase in this design is not used as a scavenge pump. The crankcase assembly contains integral inlet passages which lead from a conventional adjustable jet carburetor to each cylinder. To ignite the mixture at the appropriate time, a conventional spark ignition system is employed.

The Scotch-yoke mechanism is formed by integral heels on the piston rod ends, the ends being joined together by yoke plates with shoulder bolts fastening all parts into a very rigid assembly. The crankpin roller, which consists of three concentric sleeve bearing rings, rolls against the rod heels and is free to move up and down in the yoke as the crankshaft rotates. Pistons are attached to piston rods with conventional wrist pins; however, no articulation between these parts occurs due to the purely linear motion of the rods. The whole reciprocating piston-yoke assembly is supported by the piston rod bearings which have oil and pressure seals. Hence, the volume of the cylinder beneath the piston is used as the scavenge pump. The piston skirts are cam ground and slotted with a diameter such that a slight preload exists between the skirt and cylinder wall. This feature, according to Bourke, greatly assists heat transfer from the piston to the coolant enabling the use of high compression ratios and correspondingly high combustion temperatures without associated piston failure. Figures 2 through 5 show a vertical cutaway of the engine with the crankshaft at different positions and the thermodynamic processes which occur.

There are several very significant advantages that I saw in this type of engine design:

1. By isolating cylinders from crankcase:

a. Oil is contained in the crankcase and thus a fuel and oil mix is not necessary (cylinder oiling is accomplished by small oil passages which lead from the crankcase to cylinders).

2. Piston side thrust is eliminated due to piston rod bearings absorbing the reaction forces.

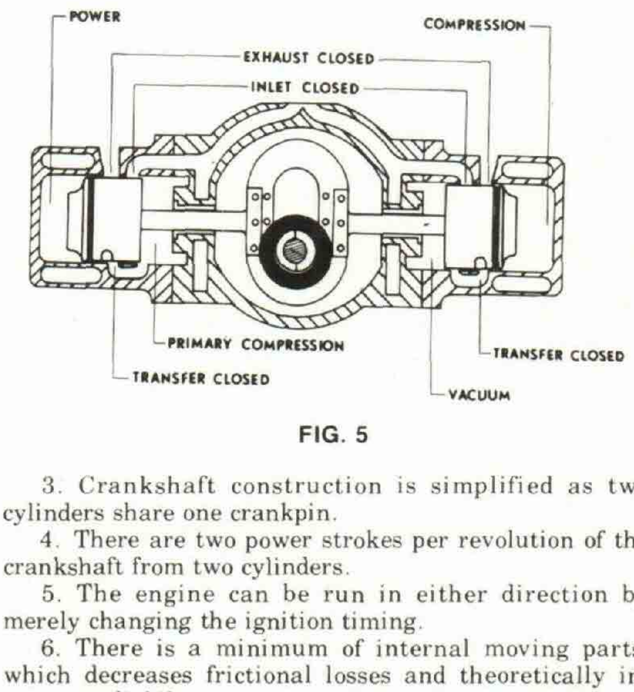

FIG. 5

3. Crankshaft construction is simplified as two cylinders share one crankpin.

4. There are two power strokes per revolution of the crankshaft from two cylinders.

5. The engine can be run in either direction by merely changing the ignition timing.

6. There is a minimum of internal moving parts, which decreases frictional losses and theoretically increases reliability.

7. No special machinery is required to construct this engine.

The major mechanical disadvantage that is apparent in this 2-cylinder engine configuration is that it is not in primary balance. In all of Bourke's original engines, the balance mass on the crankshaft balanced only the crankpin and roller with no attempt to cancel the primary out-of-balance force of the piston-yoke assembly (i.e., 100% rotating balance). In spite of this, these engines were reported to run smoothly under load at high speeds indicating that at higher frequencies, little gross motion of stationary engine components resulted due to the ratio of reciprocating-to-stationary mass.

From the early 1930's through the 1950's, Bourke designed and built six different engine configurations all of which employed the Scotch-yoke crank system. Bourke's first engine was constructed in 1932 and known as the "Silver Eagle." It was a four-cylinder radial design using air-cooled Maytag engine cylinders. This engine had a displacement of 20 cubic inches, weighed 45 pounds, and produced approximately 25 horsepower. Intended for aircraft propulsion, another four-cylinder radial design utilizing air-cooled cylinders was constructed some time between 1934-1937. This engine displaced 140 cubic inches, had a bore and stroke of 3.875 by 3.0 inches respectively, and weighed 125 pounds. Using direct drive to the propeller, 120 horsepower was produced at 3200 rpm. Patent drawings of this engine are shown in Figures 6 and 7.

During 1937, Bourke constructed a third four-cylinder radial design, this one being water-cooled and intended for an outboard motor application. Displacement was 60 cubic inches and at 6500 rpm, this engine reportedly produced 90 horsepower. At this time, there was no outboard lower unit available that would withstand this high horsepower. Thus, two cylinders were removed and the engine operated as a twin. In this configuration, Bourke reportedly logged over 2000 operational hours on this engine with no major mechanical problems occurring.

The fourth design, and probably the one that could be regarded as the most clever, was built in 1939. This was a flat four-cylinder, water-cooled, aircraft engine of 60 cubic inches. Known as the Model H, this engine consisted of two 30 cubic inch twin cylinder units with crankshafts placed vertically, both of which sat beneath and drove a horizontally placed propeller shaft via 2:1 reduction bevel gears. The engine was said to produce 60 horsepower at 5150 rpm (2575 rpm at the propeller shaft), weighed 95 pounds dry, and ran a total of 1100 hours without any mechanical problems occurring. This arrangement as shown in Figure 8 represented the first use of the 30 cubic inch units clustered in a modular engine concept.

Bourke's most famous engine, and one that he actually manufactured in limited quantities, was the 30 cubic inch twin. With a bore and stroke of 2.75 by 2.5 inches respectively and weighing only 38 pounds, this engine reportedly produced 76 horsepower at 10,000 rpm, 114 horsepower at 15,000 rpm, and would turn in excess of 20,000 rpm using glow plug ignition. Retailing for \$350.00 each, a total of 14 engines were sold during 1954 through 1955. Bourke personally gave every engine a 20 hour run-in, disassembled it for inspection, then reassembled it for a test run. Articles appeared in various magazines during this time concerning this particular engine and its superior performance.

The last and largest engine that Bourke constructed was a 400 cubic inch, four-cylinder, horizontally opposed model. It was water cooled and intended for heavy duty industrial applications. The engine was very similar in basic design to the 30 cubic inch twin except enlarged some 6 times volumetrically. The compression ratio was a high 24:1. This is the only engine that has a small amount of performance data given in the **Bourke Engine Documentary.** From this data I calculated the brake specific fuel consumption to be .26 lb/BHP hr and thermal efficiency values to be 51.2%. Typical brake thermal efficiencies of the best diesel engines seldom exceed 37% with corresponding brake specific fuel consumption values of .36 lb/BHP hr. Thus, if the figures that Bourke quotes are valid, his engine is far superior to any existing internal combustion engine in that it converts a larger amount of the fuel's available heat energy into usable work. But if this is true, then how could this engine be so superior to existing types? The inventor stated it was due to the ability of the engine to harness an unusual detonation-type of combustion providing more energy, but this was contrary to the findings of such eminent researchers as Sir Harry Ricardo.

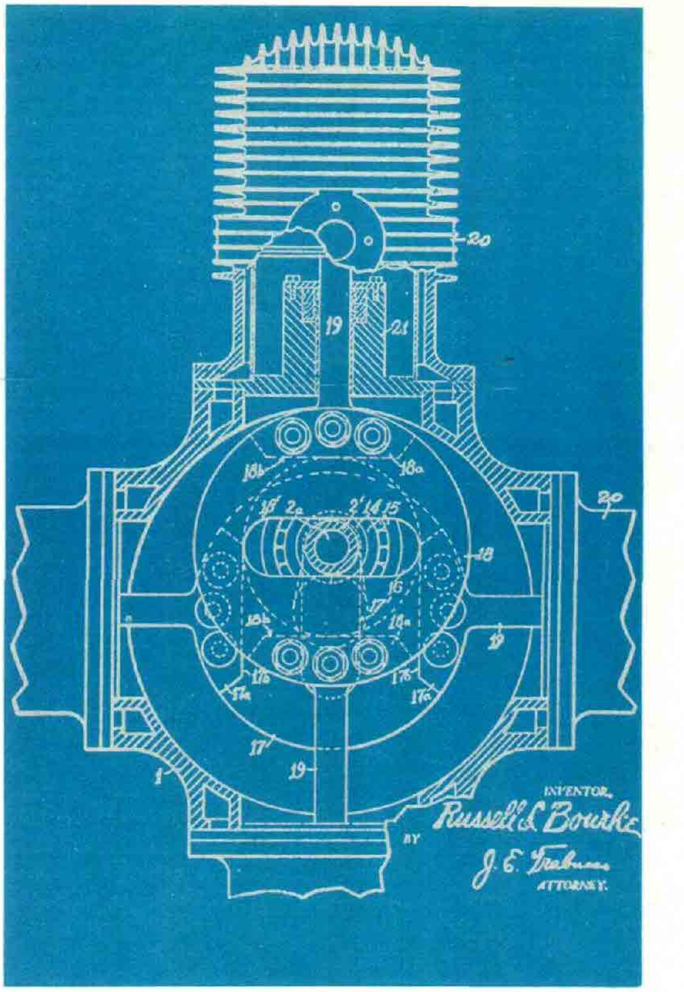

FIG. 6

Such discrepancies concerning the Bourke engine have never been satisfactorily answered, thus providing additional challenges to this research project.

From the onset of this endeavor, I decided that to accurately evaluate the potentials of the Bourke engine, an original engine should be used for testing. Unfortunately, this was to be an impossibility since owners of original engines were unknown to me at this time. As a result, the only alternative was to purchase a replica from Vaux Enterprises. I soon discovered that this company was producing more than a replica since their engine was constructed from the original patterns, jigs, and fixtures that Bourke used to produce his 30 cubic inch twin. I also learned that the owner, Melvin Vaux, worked with Russell Bourke during the 1950's and had obtained all of the original equipment from Bourke necessary to produce the engine in small, made-to-order quantities.

Sold as an "experimental engine," the Vaux-replica cost \$650.00 in 1972, minus carburetion and ignition (see Figure 9). After placing my order, the completed engine was shipped some 17 months later. After the crated engine finally arrived, and after several days of looking and admiring, I carefully disassembled it for examination. Comparing the Vaux engine to the drawings made by Donald Smail of the original pre-production 30 cubic inch Bourke engine, I discovered several changes that Vaux had made from the original design in order to facilitate his manufacturing. The most noticeable modification was to the crankshaft. On the Vaux engine, the shaft was changed from a fully machined part with two removable counterweights to a single-piece steel casting with one integral counterweight. Main bearing journal diameter had been en-

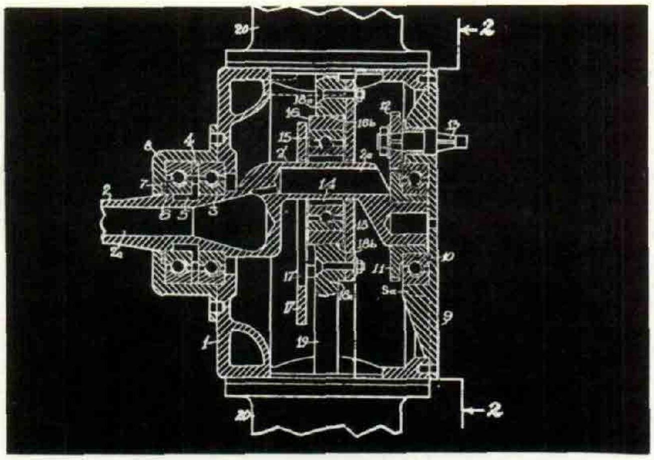

FIG. 7

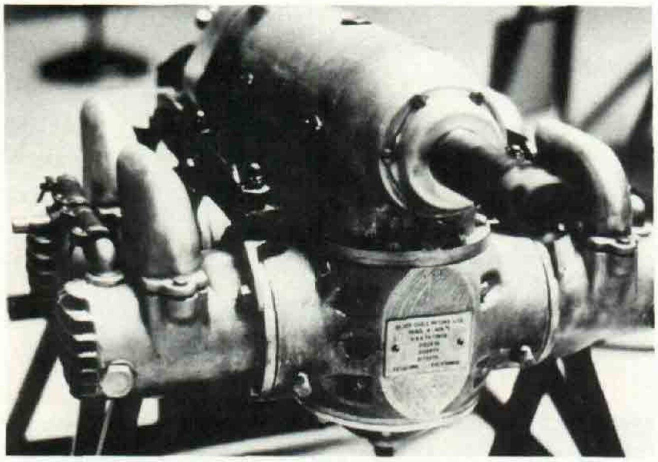

FIG. 8

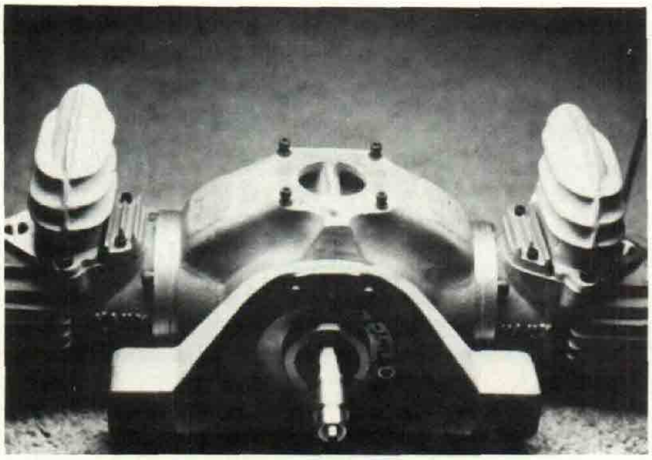

FIG. 9

larged from the original 1.181 to 1.377 inches. Single-row, large diameter ball main bearings had been substituted for double-row type as used on the original design. Noticeable changes were also made to the rod and yoke plate assembly. These parts had been changed from fully machined components in original to cast steel components in the Vaux engine. The cast rods were not rifled-drilled and the yoke plates were thicker than the originals, thus adding significant mass to the reciprocating components. Changes to the crankcase and cylinders consisted of eliminating the bronze piston rod bushings in the crankcase thereby allowing the rods to run directly on aluminum case bores. Oil drillways providing lubrication to main bearings and the transfer side of cylinders were eliminated. However, drillways to the exhaust side of cylinders were retained for piston lubrication. Inlet port timing was reduced slightly on the Vaux engine in an apparent effort to improve low-speed running characteristics. Three 14mm spark plug holes were provided per cylinder on the Vaux engine versus 2, 18mm spark plugs on the original. Crankcase end plates were also changed from two-piece to single-piece assemblies.

In evaluating these changes made by Vaux to the Bourke design, it was apparent that the single counterweight crankshaft served as a manufacturing simplification at the expense of inducing a rather severe dynamic force couple due to the single balance weight located to one side of the crankpin. The double-row ball main bearings were seemingly unnecessary on the original engine so the change to single row type could be considered adequate and beneficial. The cast rod and yoke assembly appeared much too heavy for high-speed operation due to high reciprocating forces that would be generated. Casting the rods and yoke plates results in an inherently weaker assembly structurally than an assembly made from bar and flat stock.

Elimination of the bronze rod bushings in the crankcase is not considered a disadvantage, except for serviceability. The elimination of oil drillways to main bearings could also be considered a useful manufacturing simplification as these appeared to be unnecessary on the original engine for satisfactory lubrication. Also, elimination of transfer-side oil passages and oil ports in the cylinders is considered an advantage as cylinder and piston lubrication has proven to be adequate with the remaining exhaust-side oil ports. The reduction in inlet port timing is minor and would appear to be beneficial as compared to original (this is confirmed on a running engine as carburetor spit-back is severe at speeds under 5000 rpm, even with the mild timing).

My initial thoughts were to correct the changes that Vaux had made and return the engine closer to the original design. However, I was certain that the engine would function in its existing form; and as a student, I was not able to afford the expensive and elaborate machining operations required. I, thus, decided to initially run and test the engine as Vaux had built it. Fortunately, acting in my favor was the fact that as an engineering student, I had access to the University's machine shop and the equipment therein. Thus, I began the task of fabricating an engine mount, carburetion and ignition components. A Lake injector was recommended by Vaux as this slide-valve carburetor can be adjusted for a wide variety of mixture ratios throughout the operating range.

An adaptor was required to attach the carburetor to the integral crankcase manifold, so I made a pattern, cast and machined several adaptors using the available equipment. The ignition system that I selected was constructed from two Delta Mark 10 capacitive discharge units, triggered by two Allison photoelectric triggering units, with each independent system serving an individual cylinder. While using a separate system per cylinder proved expensive, overall simplification resulted, as no distributor and associated hardware was required. The photocells were mounted on a movable plate such that ignition timing was fully variable, and a rotor was made to trigger the photocells at the proper 180 degree interval. Each Delta unit discharged through 2 coils connected in parallel which in turn fired two spark plugs per cylinder. I rationalized that dual ignition would be advantageous on this engine as two ignition sources more reliably and consistently ignite lean mixtures. An engine mount was then fabricated and an electrically-driven water pump was assembled to provide circulation of water for cooling. An eight pound steel flywheel was turned and fitted to the crankshaft and a crank starter from a Briggs and Stratton industrial engine was adapted. The completed engine is shown in Figure 10.

The day finally arrived for the initial run of the engine which took place in my backyard. All was not successful, however; with the crank starter, I could only pull the engine through one compression stroke at a time, and the Lake injector either flooded the engine quickly or was too lean. After several hours of cranking and adjusting, the engine finally started and ran briefly. Large quantities of blue smoke poured from the #2 cylinder, obviously oil being burned, which fouled the spark plugs in short order. Later, after disassembly and much head scratching, porosity was discovered in the inlet passages of the crankcase allowing oil to escape into the air/fuel mixture. This small flaw was patched with Devcon aluminum epoxy, and the engine reassembled. A Solex automotive carburetor with an adjustable main jet was installed in hopes of alleviating the flooding problem.

On the next attempt, the engine started somewhat easier and did not smoke profusely; however, vibration was indeed severe. The engine shuffled back and forth so violently that carburetor adjustments were virtually impossible as the whole device was just a blur. By idling it down to under 1000 rpm, the shaking subsided somewhat so adjustments could be made. With mixture leaned and spark advanced to around 30 degrees, the idle was very regular and steady, sounding much like a four-stroke. With more throttle opening and higher spark advance, the engine would run steadily at higher speeds but with severe vibration. The exhaust was amazingly cooler than any two-stroke engine I had previously worked with, the gasses feeling just warm to the hand when held close against the exhaust stack. After several more runs were made under a no-load condition for breaking-in purposes, the engine was mounted on a dynamometer in the University's thermodynamics lab. But, as could have been predicted, vibration was so severe that anchor bolts which held the engine mount were pulled from the concrete floor. At this time, it was decided to construct a lightened rod-yoke assembly to reduce the large reciprocating forces. Before proceeding, however, I thought it best to review once again the theoretical aspects of this engine.

Bourke claimed that his engine operated efficiently on detonation of the fuel mixture due to what he believed was a "hydrogen-oxygen" reaction occurring (Bourke cycle), as opposed to a "carbon-oxygen" reaction as he believed occurred during "normal" combustion. "The hydrogen-oxygen reaction releases far more energy per pound of hydrocarbon fuel than the carbon-oxygen reaction," states Bourke. However, the contrary has been established by combustion bomb experiments where it has been demonstrated that all available hydrogen and oxygen in a hydrocarbon fuel is utilized in combustion as predicted by the stoichiometric mixture ratios (i.e., 15.2:1 air/fuel ratio using C_8H_{18} as a fuel).

Over the last 50 years, considerable research has gone into analysis of the detonation phenomena. Such research has shown that the normal combustion cycle is characterized by a smooth pressure change whereas the detonating cycle shows severe, vibratory-like fluctuations in pressure. Evidence indicates that these pressure peaks can attain quite high values, but since the **average** pressure in the cylinder is approximately the same as with normal combustion, there is little or no change in the power output. In view of this information, it was difficult for me to believe that detonation of the fuel mixture in the Bourke engine provided more usable energy than the conventional combustion process.

I felt that a more realistic answer was that high cylinder pressures were developed through a form of controlled preignition of the fuel charge. Moreover, Bourke himself states that his engines can run with a spark advance of up to 90 degrees before tdc and that the exhaust products are under 200 degrees F. Recent experiments into abnormal combustion in two-stroke engines has shown that preignition of a relatively lean air/fuel mixture almost doubles the combustion pressure at tdc while proportionally increasing the work of the compression process. From Figure 11, it can be demonstrated that the overall mean effective pressure would be increased as compared to normal combustion with a predictable increase in the pressure during the compression process.

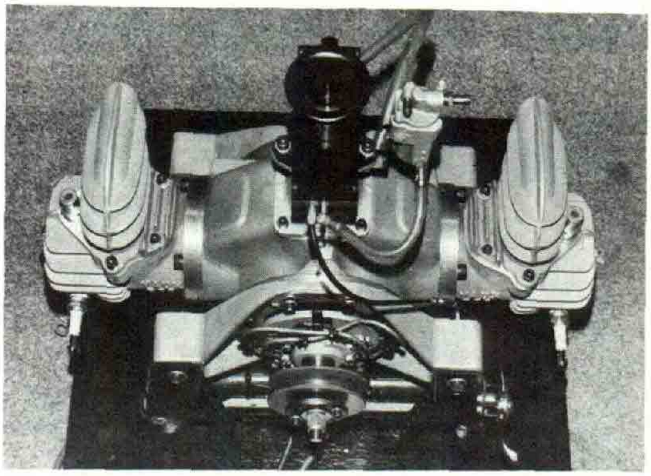

FIG. 10

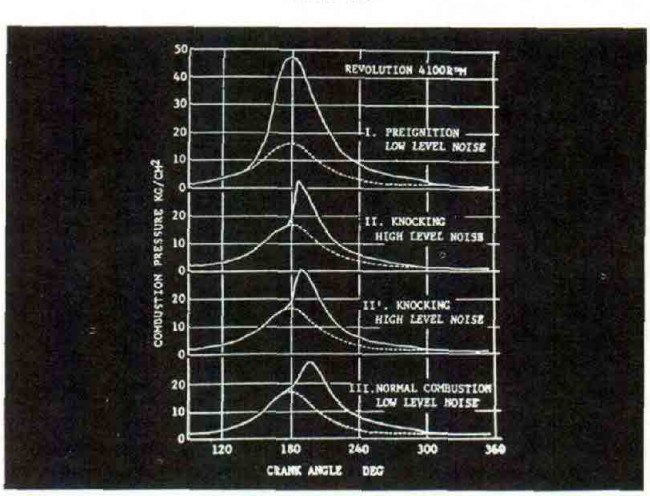

FIG. 11

In any spark-ignited internal combustion engine, preignition tends to increase heat loss to the coolant, subject the engine to excessive pressures and increase the work of the compression stroke. However, the Bourke engine, with its inherent lack of connecting rod angularity, is supposedly able to withstand these high pressures. Controlled preignition in this engine would seem to be of greatest benefit in stopping and reversing the reciprocating mass at high speeds, primarily by gas pressure alone. This would greatly relieve the loads imposed on the crankshaft and reduce tensile stresses in the reciprocating parts.

Considering the mechanical aspects of this engine, a very interesting attribute of the Bourke design is that high primary compression ratios are obtainable due to the mechanical configuration. Since the scavenge pump comprises only the swept volume of the cylinder beneath the piston, the free volume that exists may be considerably reduced as compared to the conventional two-stroke engine with a crankcase scavenge pump. For most conventional high performance two-stroke engines, a primary compression ratio of 1.5:1 has been shown to offer the greatest delivery ratio without introducing excessive pumping losses. However, in all of Bourke's designs, ratios of up to 3:1 are used. It can be proven from thermodynamics that doubling the primary compression ratio increases the delivery pressure ratio by a factor of 2.64. The higher pressure ratio theoretically means that when the transfer ports open, the mixture will transfer into the cylinder at a higher velocity, with correspondingly better directional control and at the same time assuring a more thorough vaporization of the mixture. It should also be noted that the higher primary compression ratio causes a greater depression in the scavenge pump prior to inlet port opening; hence, the velocity and resultant vaporization of the entering mixture should be increased as compared to the lower compression ratio. The combination of a high pressure ratio pump using a piston controlled inlet port induction system would seem to assure a high degree of atomization of the air/fuel mixture. Therefore, the increase in pumping work with the higher ratio could be offset by a more efficient combustion process. With the high primary compression ratio available in the Bourke design, relatively mild port timings with small area ports appear to be all that is necessary to effectively move the same quantity of mixture versus that moved by large ports and lower pressure ratios.

Another interesting feature inherent in the Bourke engine is the fact that the scavenge pump swept volume is less than the cylinder swept volume. This is due to the volume displaced by the cylindrical piston rod which reduces the effective area of the underside of the piston by the rod's cross-sectional area. In the 30 cubic inch engine with a piston rod diameter of .625 inch, one cylinder's swept volume is 14.85 cubic inches, while the scavenge pump swept volume is 14.08 cubic inches. This results in a scavenge pump/cylinder swept volume ratio of .948. Scavenging the cylinder with this slightly smaller volume of mixture indicates that chances of overscavenging is somewhat reduced at the expense of a larger exhaust residual remaining. It should be noted in making air consumption or volumetric efficiency calculations to use the scavenge pump swept volume on this type of engine. This in turn may be related to cylinder swept volume to obtain an overall volumetric efficiency (example: if at a given speed of the 30 cubic inch engine, the scavenge pump volumetric efficiency is 80%, the overall volumetric efficiency is .80 x .948 = .758 or 75.8%). Any increase in fuel economy due to this volume ratio being less than 1.0 is in reality very slight as some mixing of the fresh scavenging charge with the exhaust products and resulting charge loss to exhaust is bound to occur.

In considering differences in cranktrain design, the Scotch-yoke crank mechanism with its inherent pure sinusoidal reciprocating motion, has equal dwell periods at both dead center positions in comparison to the conventional crank-connecting rod system which has a correspondingly shorter dwell time at tdc, and longer dwell time at bdc. According to Bourke, the extended dwell time around tdc with the Scotch-yoke, plus initiation of the combustion process while the lean air/fuel ratio is being compressed, results in completion of the combustion reaction prior to expansion. However, it can be observed in Figure 12 that the motion of the Scotch-yoke piston assembly is very similar to the more conventional crank-connecting rod mechanism, and the slightly longer dwell time of the former would not appear to significantly alter the combustion process to any great extent.

At this stage of the project, I devised a mathematical model of the engine by which all stresses and loads could be calculated in critical areas. This model was based on Bourke's claims of 76 horsepower at 10,000 rpm, which gave a brake-mean-effective pressure of 100 psi. Working out an idealized cycle with this average pressure, gas pressures could be predicted at any point in the cycle. The gas loading combined with inertia forces constituted the basic model by which many interesting things became apparent.

In concentrating on the stresses in the yoke plates, it was found that these parts could be made some .060 thinner than what Bourke specified reducing several ounces of mass from the reciprocating assembly. The crankpin slot was also widened saving more weight. These changes would not structurally weaken the plates to any extent, and a large safety factor still existed as long as top quality materials were used. However, lightening the reciprocating components this small amount would not significantly reduce the vibration problem to any great extent as calculations showed. The obvious compromise was to counterweight the crankshaft as is done in any other reciprocating engine in order to reduce the forces generated.

In any engine, whether rotary or reciprocating, it can be shown that gas pressure forces cancel themselves through the structure of the engine; i.e., pressure on a cylinder head which tends to push the engine structure in one direction, is balanced by the force on that cylinder's piston which show up as equilibrium forces on the crankshaft main bearings. In other words, when calculating vibrational forces, gas pressures can totally be ignored and only rotating and reciprocating forces need be contended with. Due to the nature of the Scotch-yoke's pure harmonic motion, maximum reciprocating forces at top and bottom dead center positions are approximately 20% less as compared to conventional crank-connecting rod mechanisms with identical reciprocating masses. Also, since no rod angularity exists with the Scotch-yoke, there are no secondary imbalance forces present.

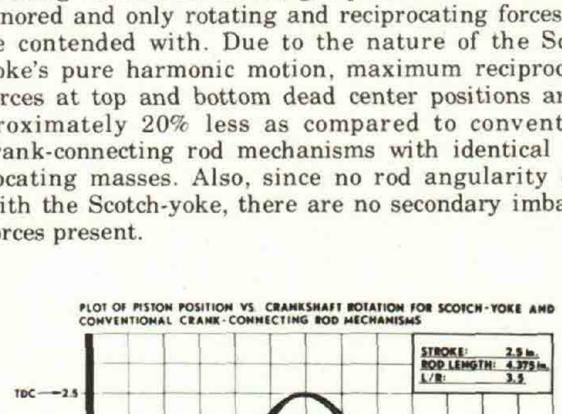

FIG. 12

In studying this mechanism, it became readily apparent that a 50% reciprocating balance factor added to the crankshaft counterweights is the best compromise in simplicity for the two-cylinder configuration without resorting to counter-rotating or reciprocating balance masses. Even though the latter mentioned devices would enable the engine to be perfectly balanced, the mechanical complication is undesirable. With the 50% balance factor, there is a resultant force vector which rotates opposite to, and at the same speed as the crankshaft. This force will, of course, tend to shake the engine, but since it is exactly half the magnitude of the unbalanced case, and since it is a rotating force, it is much easier to contend with as far as mounting systems, etc., are concerned.

With regard to the crankshaft, the increase in main bearing diameter to 1.377 inches, as Vaux had specified, was thought beneficial. It seems that the original Bourke 30 would occasionally break its 1.181 diameter shaft under certain conditions, as I learned from talking to some of Russ Bourke's acquaintances. It was also stated in Bourke's writings that propeller hub bolts, driveshaft keys, etc., could be sheared at will simply by opening the throttle too rapidly. Again relying on the mathematical model, I determined the forces applied on the crankpin from the reciprocating components and plotted a moment diagram for 180 degrees of crankshaft rotation (or one stroke) at speeds ranging from 3000 to 10,000 rpm. The results are shown in Figure 13. At a given rpm, say 8000, it will be noted that from 0 to about 85 degrees, the torque becomes positive indicating the shaft is now exerting this torque on the load. Note that the torque peaks are always of a higher value on the positive side indicating that the average torque over the 180 degree interval is positive, as would be necessary for the engine to produce power (the same values are obtained during the next 180 degrees of rotation, so one needs only to study the first 180 degree interval). Since these values are hypothetical, the actual values on a real running engine would be somewhat less. This does, however, indicate the high-magnitude second-order torque fluctuations which caused shaft and drive system breakage. The obvious remedy is to add rotational inertia to the crankshaft in the form of a flywheel, something Bourke, for rather obscure reasons, always seemed to avoid.

The Vaux crankshaft was adequately sized for strength, but with only one counter-balance weight, it was not considered suitable for reasons previously mentioned. It was apparent that a new crankshaft should be made to compliment the lightened rod-yoke assembly and allow desirable counterbalance masses to be located on both sides of the crankpin. I then designed a crankshaft utilizing the larger main bearings of the Vaux shaft with removable counterweights similar to that used on the original Bourke crankshaft.

During this time I graduated from Vanderbilt with a Bachelors degree in mechanical engineering and went to work for a major engine manufacturer. Being somewhat disappointed that I was not able to complete the Bourke engine evaluation while in college, I continued to work on the engine in my spare time at home.

My next step was to complete the drawings of the crankshaft and rod-yoke assembly and submit them to a machine shop. The first parts to be made comprised the rod-yoke assembly. Both rods and yoke plates were to be made of billet and flat stock 4140 alloy steel, hardened and drawn to Rockwell C-35. The piston rods were rifle-drilled for lightness and had a ground finish all over as did the yoke plates. Figure 14 shows these components prior to assembly. A crankshaft was next to be built and this too was made from 4140 steel, turned from billet stock, and heat treated to a hardness of Rockwell C-38. A new crankpin bearing was also made, the inner and outer rings consisting of Ampco 22 bronze alloy, which has a yield strength of 62,000 psi, slightly superior to Ampco 18-22 alloy, which Bourke recommended for this part of the engine. A hardened and ground alloy steel ring was made to serve as the intermediate bearing between the bronze rings, and this completed the triple-slipper bearing assembly. Figures 15 and 16 show the bearing, crankshaft, and counterweights in the disassembled and assembled states. At

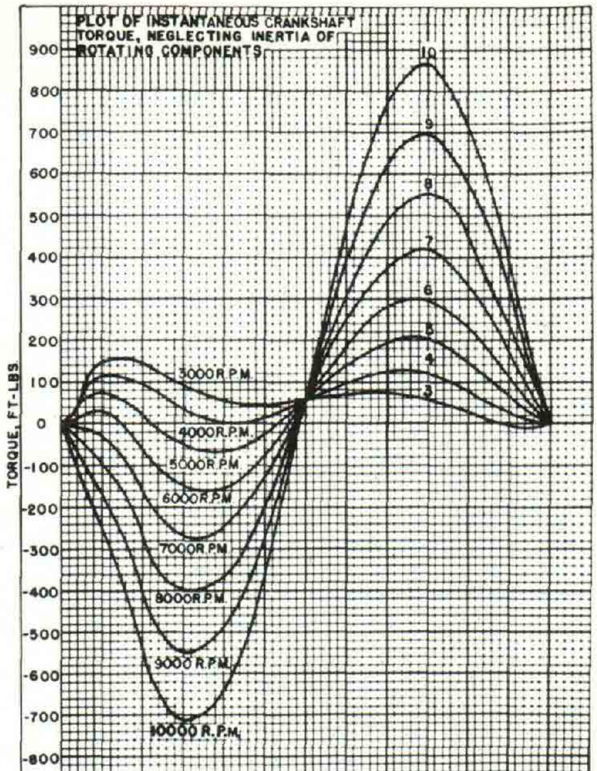

FIG. 13

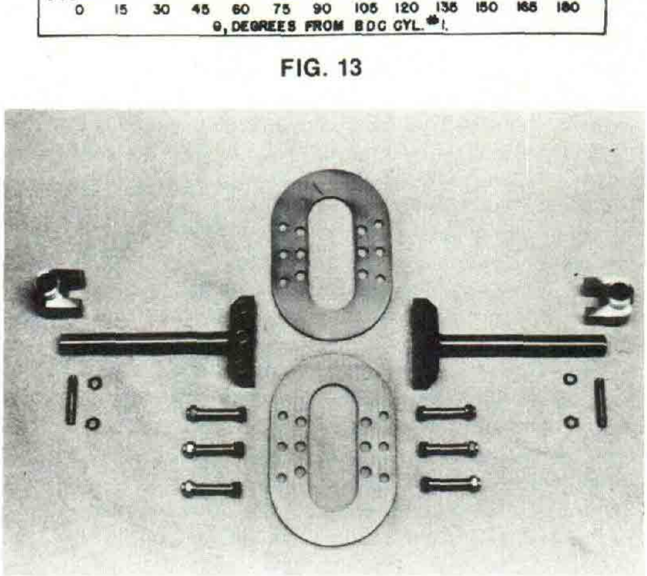

FIG. 14

this time, a number of other items were remade to bring all components up to the highest standard of precision. The crankcase faces were checked and resurfaced to ensure that all mounting surfaces would be square, and the piston rod bores were machined larger for bronze rod bushings, which were then turned and pressed into place.

I now possessed all the parts, made to my specifications, which would give a highly precision engine with which an accurate evaluation could now be performed. I carefully assembled the engine on a meticulously cleaned work bench, with all parts fitting together like the proverbial "Swiss watch." An electric starter from a snowmobile engine was now fitted and again, initial running was performed in my backyard under a no-load condition. The engine started on the first attempt and ran better than it ever had previously. Vibration levels were greatly reduced by the 50% balance factor, and the engine appeared to shake no more than a conventional

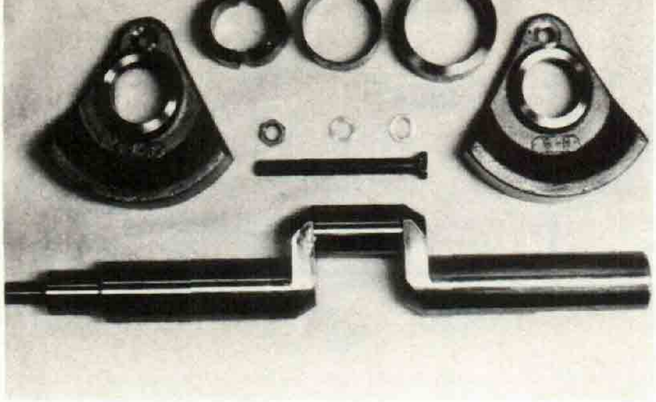

FIG. 15

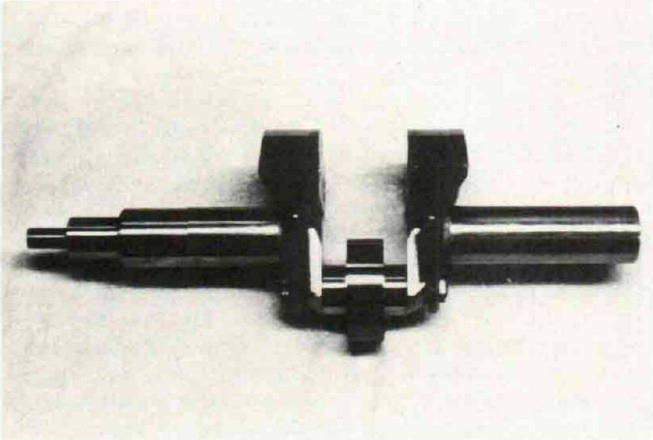

FIG. 16

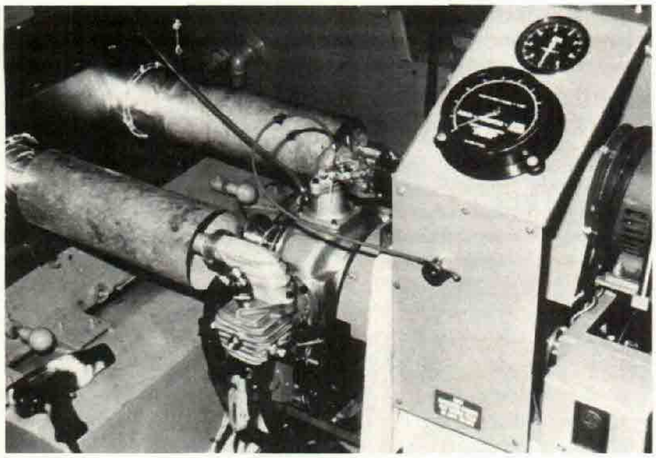

FIG. 17

single cylinder engine. The Lake injector carburetor now functioned well in conjunction with the electric starter, as the engine could now be turned over fast enough such that flooding on starting was no longer a problem. After several hours of running at various conservative speeds with no load, I felt the engine was ready for the dynamometer.

During this time, a dynamometer had been made available for my personal use. I thus began fabricating the necessary equipment to mount and properly couple the engine to the dynamometer. With the setup completed, as shown in Figure 17, a six-hour run-in was then conducted at varying speeds and moderate loads, during which minor problems were corrected and changes made. Things like engine mount changes, exhaust system ducting, crankcase breather systems, and other minor modifications all contributed into expanding the six-hour break-in into a four-month time span.

Now, with the break-in completed, a full testing program could begin. Previously the engine had only been run on regular leaded gasoline with oil mixed at a 1:24 ratio to aid in cylinder lubrication during break-in. Since I wanted to follow Bourke's recommendations as close as possible, a fuel mixture of 3 parts unleaded gasoline was mixed with 1 part stove oil (kerosene), with a small quantity of lubricating oil added in a 1:100 ratio. The crankcase oil used was Shell "Rotella" 30 weight. The dynamometer, which was a Stuska brand water brake capable of absorbing 90 horsepower at 10,000 rpm, was carefully checked for accuracy prior to testing. I had originally planned to run a power curve with wide-open throttle and optimized spark and fuel mixture in order to determine the most important parameters initially, these being the horsepower potential and fuel consumption. If these figures were anywhere in line with what Bourke had claimed, the engine would then be fully instrumented to record cylinder pressure histories and exhaust gas analysis in order to determine the combustion phenomena taking place.

With everything in order, mechanically and otherwise, the testing began on the morning of October 12, 1978, almost six years from my initial conception of the project. My procedure during this test was to run the engine from 3000 rpm, up in 500 rpm increments until the horsepower curve peaked (became its maximum value), and run 500 rpm over this value to verify that a peak had been reached. I felt that it would be useless to run extremely high rpm's and take a chance damaging the engine, especially if only small power outputs were being obtained at these speeds.

With the engine running and up to operating temperature, the throttle was eased wide open and rpm held down to 3000 by the dynamometer load. At this speed, fuel spit-back from the carburetor was excessive, as it is on most piston-ported two-strokes, so the throttle was closed slightly until the spit-back was reduced (I later found that optimizing throttle position was necessary at all speeds to avoid this problem). Next, the spark advance and fuel mixture were adjusted until the dyno scale load was at a maximum. Computing the horsepower outputs as I progressed up in rpm, it was apparent that the engine was not producing anywhere near the claimed values. Much time was spent in minute and careful adjustment of fuel mixture and spark timing at each speed, both simultaneously and individually, with no significant power increases noted. Thus far, the engine behaved as any other two-stroke; that is, advancing the spark timing over 40 degrees btdc would send the spark plug thermocouple readings over 250 degrees and correspondingly the power would decrease, just as leaning the mixture past the maximum power setting would cause a drop in power as well, or result in complete engine stoppage. Many hours were spent in adjusting these controls at various speeds, all with the same predictable results.

One fact was paramount throughout the testing — there was audible spark knock. This occurred especially at speeds over 4000 rpm and was audible to all personnel witnessing the test. Rather severe detonation was indeed occurring, and the results were the same as when it occurs in a conventional engine, which is a significant rise in spark plug and coolant temperature with a slight drop in power. When the dual ignition system was operated on single ignition, the knock was even more severe, spark requirement was up around 50 degrees btdc, and power down some 20% over the dual ignition values. Quite apparent was the fact that the low octane value of the fuel mix was contributing to the severe detonation, so a switch to regular leaded gasoline was tried. This did decrease the knocking tendency somewhat, but power levels stayed approximately the same as with the other fuel at all speeds. At slightly over 5000 rpm, the engine was virtually at its volumetric limit, the throttle at this point being wide open and the dynamometer completely unloaded. I really could not believe it; here was an engine that supposedly would produce over 70 horsepower at 10,000 rpm, and the best I was getting was a mere 8.8 horsepower at 4000 rpm, with a maximum no-load speed of 5000 rpm. Something surely must be wrong.

The next day the engine was checked and more runs were made. Compression pressure was equal and high on both cylinders, everything was tight and in adjustment, and the dynamometer was rechecked for accuracy. High octane premium fuel was tried, but horsepower outputs remained unchanged at their low levels. A smaller diameter-venturi carburetor was also tried in place of the Lake injector, but no improvement in power resulted. Finally, during one run the oil temperature soared past 260 degrees F. and the engine slowed down abruptly, at which time I immediately turned off the ignition switch. After things cooled down a bit, I noted that the engine turned over rather stiffly by hand, indicating probable mechanical damage and termination of testing. Later upon disassembly, I discovered that the center steel ring of the crankpin triple slipper bearing had begun to seize against the yoke plates, obviously due to lack of lubrication by the small quantity of oil contained in the crankcase.

Figure 18 is plotted from the data collected during one run. The high fuel consumption was due mostly to blowback through the carburetor, with partial and incomplete scavenging probably contributing to the situation. The poor overall performance certainly indicated that something was inherently wrong with my particular engine as I felt that at least 30 horsepower would be obtainable.

But what could the problem be? All port sizes corresponded exactly to the dimensions given in the Donald Smail prints, drawn from the original Bourke cylinders. But careful measurements of port height relationships indicated something that I had ignorantly overlooked and taken for granted. In a two-stroke engine, the exhaust ports are always timed to open prior to the transfer ports, in order for cylinder pressure to "blow down" to near atmospheric pressure. This allows the transferring charge to enter the cylinder without resistance such that scavenging is more complete with a larger quantity of fresh charge available in the cylinder. The prints showed an exhaust lead (difference in height between exhaust and transfer ports) of .094 in., but the Vaux cylinders measured a lead of only .015 in. Clearly, this explained why power was down, as very little fresh charge was able to enter the cylinder at the higher speeds. This condition would also aggravate carburetor spit-back, which was observed during all tests.

It was noted during testing that the crankcase was running around 200 degrees F. when the cylinders were at 160 degrees; the high crankcase temperatures probably resulting from heat transfer from the piston rods into the rod bushings located in the crankcase casting. It was apparent that a large capacity, dry sump, pressure lubrication system would be the obvious remedy. Such a system would greatly assist in cooling the internal parts, plus offering more thorough and efficient lubrication as compared with the existing splash system. A Volkswagen oil pump, as used on the air cooled 1600 engine with automatic transmission, seemed to be perfectly suited to my application due to its size and construction. This gear pump is a divided type, in reality 2 gear pumps of different capacities placed back to back, and is intended to circulate engine oil as well as transmission fluid. Such a pump was then purchased and adapted to my engine such that the low capacity side served to provide oil from a tank to the engine, and the higher capacity side served as a scavenge pump return-

FUEL - 3 PARTS UNLEADED GASOLINE-1PART FUEL OIL.—CARBURETION-LAKE INJECTOR

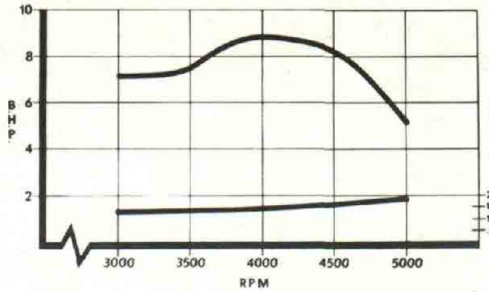

FIG. 18

ing the oil from the crankcase to the tank. A one gallon oil tank was then fitted along with a full-flow filter. The crankcase was modified by drilling an oil passage and providing a nozzle whereby a high pressure oil spray was directed onto the crankpin bearing and yoke assembly.

The cylinders were next to be modified, the ports being remachined to allow an exhaust lead as indicated on the Donald Smail prints. Piston rings and all bearings and seals were replaced to bring the engine back up to high standards. The engine was then carefully reassembled and remounted on the dynamometer.

After another break-in period, testing was then resumed. A most noticeable improvement was the reduction in crankcase temperature due to the circulatory oil system. However, I was surprised and disappointed to see that the horsepower output was virtually unchanged from the previous runs, in spite of the modifications made to the ports. With the engine now running reliably, many hours were spent in trying different ideas in order to increase performance. Many different sizes and types of carburetors were tried, without beneficial results. The best performance was obtained with a slide valve carburetor of my own design, which functioned in a similar fashion to the Lake injector. Exhaust baffle plates were tried in an attempt to vary back-pressure, but this reduced power as compared to an unrestricted exhaust. Variations in fuel mixtures and octane rating was explored further with all results showing only insignificant variations in power output.

Naturally, when one undertakes a project of this magnitude, impressive results are desired and are most rewarding. However, based upon the data I have acquired thus far, I must honestly conclude that this engine in its present form requires extensive development to even match a conventional two-stroke engine in power output and fuel consumption. Since the intent of this project was to verify Bourke's claims, no modifications were made to any of the gas flow passages, piston deflector, or combustion chamber shape. If Bourke's theory on combustion is valid, it should have been duplicated in this experiment resulting in acceptable horsepower and fuel consumption figures for this size of engine.

Some people state that Russell Bourke highly exaggerated his claims as to the performance of his engine. Others, however, swear that the engine was everything Russ said it was, and more. My feelings are mixed. It is nice to believe that one man invented a simple engine that runs almost contrary to the laws of physics and chemistry, with performance that far overshadows all other internal combustion machinery; but one must also be in touch with reality, in this case the science of the internal combustion engine and its over 80 years of development. Gaps do exist in our knowledge of the complex phenomena of combustion, however, and it is remotely possible that Bourke discovered something that all other researchers overlooked. Only continued interest, dedication and experimentation with this interesting engine will reveal its hidden secrets, potential and ultimate future.