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Design of a Two-Stroke Cycle Spark Ignition Engine Employing a Scotch-Yoke Crankshaft Mechanism

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ABSTRACT

This paper discusses the design aspects of a two-stroke cycle, spark ignition engine intended for light aircraft propulsion. A modified form of scotch-yoke crank mechanism is employed, and by utilization of two such reciprocating assemblies set at 90 degress to each other, a compact 4-cylinder radial engine configuration results. The unique mechanical design considerations of the scotch-yoke combined with the two-stroke engine cycle are examined in detail. The design offers an inherently balanced, low vibration machine employing a full circulatory lubrication system while retaining basic two-stroke engine simplicity. A prototype has been constructed which meets all initial design criteria and test results are presented.

THE DEMAND FOR A SMALL DISPLACEMENT, LIGHT AIRCRAFT ENGINE in the 15-22 kW range has been steadily increasing over the past several years. Advances in lightweight structural materials and aerodynamic designs have enabled single place sport aircraft to obtain rather remarkable performance figures on low power engines, and interest in this size of airplane in the experimental homebuilt category is growing. Referred to as an Air Recreational Vehicle, or ARV, this type of aircraft offers cruise speeds of approximately 160 km/h while obtaining 26 to 34 km/L fuel economy. With inherently low operational costs and reasonable prices for plans and kits, predictably this type of airplane is becoming increasingly popular. Another form of aircraft which utilizes an engine of approximately 18 kW is the Remotely Piloted Vehicle, or RPV. Initially developed by the Israeli Air Force, this aircraft is intended primarily for battlefield surveillance, which the Israelis have used with success. The United States government has become interested in developing an RPV, and several noted aircraft manufacturers are pursuing development of this interesting machine.

To meet the requirement for small aircraft engines, aircraft designers and manufacturers have been forced to utilize converted snowmobile, motorcycle, chain saw, and various forms of industrial engines of both 2 and 4 stroke-cycle. Complaints from pilots of ARV's indicate that the majority of these existing powerplants suffer from poor reliability, overheating, short operational life, excessive weight/power, and extreme vibration (in some cases severe enough to damage instruments in a short period of time). More recently, a few specialty manufactured engines have appeared for the purpose of small aircraft propulsion, but most of these designs are compromised to some extent and do not totally alleviate the aforementioned faults. It was quite obvious that a true aircraft quality engine that posessed none of the above-mentioned deficiencies would be highly desirable for both the ARV and RPV application. This formed the impetus for the project, which was to arrive at a design that was free of typical small engine drawbacks, yet was mechanically simple and low in cost, with relatively high specific output combined with good reliability.

DESIGN EVALUATION

From the onset, it was decided that the two-stroke engine cycle was desirable for the following reasons:

1) Lighter weight and smaller package volume for developed power as compared to a 4-stroke engine.
2) Inherent simplicity of design meaning fewer component parts and potentially lower costs to manufacture as compared to the 4-stroke.
3) Utilization of exhaust tuning on the 2-stroke, enables brake specific fuel consumption at cruise power to be approximately equal to a 4-stroke engine, especially in this relatively small

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displacement size of 600 cc's or less.

With the virtues of the 2-stroke, there also exists serious vices. It is generally accepted that this engine cycle is more thermally limited than the 4-stroke and this can be observed in practice. Sudden and catastrophic piston failures with no prior warning are much more common with the 2-stroke engine than on the 4-stroke. This in practice seems to be the main cause of unreliability. In addition, crank pin bearing failures due to high temperatures and inadequate lubrication further contribute to engine failures. The general notion of unreliability added to the fact that most of the engines in use are lubricated by oil mixed with fuel, can be considered serious disadvantages when compared to the 4-stroke. In order to make the 2-stroke engine dependable enough for aircraft applications, the aforementioned deficiencies would have to be minimized or preferably eliminated.

An important initial design consideration is the type of cooling system which is to be utilized. With a thermally limited engine, greater power outputs may be extracted reliably with liquid cooling as compared to air. However, engineering trade-offs enter. The liquid cooling system has additional weight, complexity, and cost penalties when compared to air cooling. It was decided that the slight sacrifice in ultimate power output with air cooling is acceptable in favor of simplicity and lighter weight. Use of a multi-cylinder engine with small bore cylinders and ample finning appeared to be the best compromise, in that large cooling fin area/bore area ratios could be maintained. This would ensure adequate cooling and therefore acceptable reliability.

With these design parameters decided, the next step was to arrive at a suitable mechanical configuration. The author has long been attracted to the scotch-yoke crank mechanism, 2-stroke cycle engine combination and examining patented prior art shows many other individuals have been intrigued by the concept as well (1-10)*. One of the more famous engines of this type was constructed by Russel Bourke who received a good deal of publicity during the early 1950's. Testing one of Bourke's engines (11) revealed obvious design flaws, mainly in the air handling capabilities and thermodynamic limitations, both contributing to the poor specific outputs obtained during the test. Mechanically, however, the engine functioned well with no measureable wear occuring on any stressed components over some ten hours of dynamometer running. Of greatest importance was Bourke's mechanical design contribution to this form of engine, and of specific interest is the construction of the rod-yoke assembly. This consisted of two identical tubular piston rods with integral heels, joined by dowel bolts and yoke plates which locate and space the rod heels, thus forming the yoke working faces. The complete assembly results in probably the lowest mass rod-yoke design consistent with adequate strength and rigidity. Bourke's designs utilized a crankpin mounted yoke roller made from an aluminum-bronze alloy of high strength and wearability. It has been rumored that major engine manufacturers have experimented with scotch-yoke engines, but to the author's knowledge, no reports were ever made public nor engines produced (12). Recently, a prominent refrigeration compressor manufacturer introduced an automotive air conditioning compressor utilizing a 4 cylinder, single plane, radial scotch-yoke configuration (13). The unit has more compact dimensions and is reportedly smoother in operation when compared to conventional piston type compressor designs.

ADVANTAGES OF THE SCOTCH-YOKE 2-STROKE ENGINE - The single most important advantage of the scotch-yoke compared to the conventional slider crank mechanism is that the former, when utilized with the two-stroke engine cycle, allows the crankcase to be totally isolated from the cylinder scavenge pumps. Since the tubular piston rods reciprocate without articulation, a seal surrounding the rod and fixed in a suitable aperture in the crankcase effectively isolates the crankcase interior from the cylinders. This feature allows a straight forward circulatory lubrication system to be employed. Hence, the main and crankpin bearings receive improved lubrication and cooling effects offered by an undiluted oil system, as exists in 4-stroke machines. The scavenge pumps are formed by the lower portion of each cylinder and corresponding crankcase face, with the underside of the piston forming the pumping element. The basic mechanical simplicity of the two-stroke engine is thereby maintained. It may be argued that a similar mechanical arrangement is possible with the slider crank by utilizing a cross-head with a tubular piston rod sealed in the same fashion, but simplicity and compactness are sacrificed when compared to the scotch-yoke mechanism.

Other advantages to be realized are better potential for complete engine balance due to pure harmonic motion (assuming constant angular velocity of the crankshaft) of the scotch-yoke reciprocating assemblies. There are no secondary or higher order forces to contend with as are induced by the articulating connecting rod of the slider crank, therefore all primary reciprocating forces may be totally resolved by appropriate counterweighting. A simplified, single throw crankshaft may be utilized by two cylinders of an opposed twin, or four cylinders (2 rod-yoke assemblies) of a radial configuration.

The disadvantage of the scotch-yoke system is slightly higher costs to manufacture than the conventional connecting rod slider crank. The rod-yoke assembly must be held to tight radial runout tolerances along the entire length, as well as exacting perpendicularity and parallelism tolerances on the rod heels (working faces of the yoke). A smooth, ground finish must exist on the piston rods to ensure low seal wear. These costs are partly offset in the

*Numbers in parenthesis designate references at end of paper complete engine by the previously mentioned potential for a simple, single throw crankshaft design, as well as lack of wrist pins, wrist pin bearings, and associated machining operations during manufacture. Due to lack of articulation of the piston rods, pistons may be rigidly affixed to the rods by simple low cost means.

ENGINEERING CONSIDERATIONS AND EVALUATION

Of the conventionally designed two-stroke engines presently being used for light aircraft propulsion, two major schools of design methodology exist. These are as follows:

1) Run a small displacement, highly tuned engine at high operational speeds (i.e. 7-9000 rpm), obtaining maximum power/displacement and power/weight ratios, with resultingly shorter operational life and less reliability.
2) Run larger displacement, moderately tuned engine at lower operational speeds (i.e. 4-5000 rpm), obtaining lower power/displacement and power/weight ratios, thus obtaining longer operating life and higher reliability.

Unfortunately, many engines are designed by the first method, relying on the use of speed reduction systems to obtain reasonable propeller efficiency. The reduction unit adds to overall powerplant weight and cost, thus rendering this design even more unattractive. For an aircraft engine, the lower speed design is much more preferable and was the adopted design scheme for this project.

Propeller diameter is the major factor in determining limiting engine speeds, and it has been shown that tip velocities above .8 Mach contribute to decreased propulsive efficiency and noise generation. This speed limitation, combined with the fact that most popular propellers produced for small direct drive engines range from 91 to 100 cm in diameter, results in maximum rotational speeds calculated to be 5700 and 5250 rpm respectively. Thus, a maximum engine speed of 5000 rpm was chosen to be the peak power rpm where it was desired to produce a conservative 18.6 kW (25 horsepower). From experience, it was realized that a BMEP of 550 kPa would be realistic for a mildly exhaust-tuned engine running at 5000 rpm. From these parameters, a displacement can be calculated which resulted in 405.6 cc swept volume. With engine speed and displacement now loosely defined, a suitable mechanical configuration utilizing the scotch-yoke was next to be examined. This study revealed that a 4-cylinder, X configuration radial layout utilizing 2 scotch-yoke reciprocating assemblies disposed at 90 degress to each other and operating on 1 crankpin, would be completely balanced by implementation of suitable crankshaft counterweighting. This is shown in the following, basic derivation.

Differentiating the position equation twice yields acceleration for the scotch-yoke, in both x and y axis orientations:

$$a_x = \omega^2 r \cos\Theta \tag{1}$$

$$a_y = \omega^2 r \sin\Theta \tag{2}$$

where:
a = Acceleration
ω = Crankshaft Angular Velocity
r = Crankthrow Radius
θ = Crankshaft Angular Position from tdc

Noting that

$$F = m\ a$$

We may write:

$$F_x = m_x \omega^2 r \cos\Theta \tag{3}$$

$$F_y = m_y \omega^2 r \sin\Theta \tag{4}$$

where:
F = Force generated by 1 reciprocating component
m = Mass of 1 reciprocating component

From the Pythagorean relationship, the resultant force may be determined:

$$F_{res} = \left[(m_x \omega^2 r \cos\Theta)^2 + (m_y \omega^2 r \sin\Theta)^2\right]^{\frac{1}{2}} \tag{5}$$

For any practical engine, the 2 reciprocating masses will be equal:

$$m_x = m_y = m \tag{6}$$

From trigonometry

$$\sin^2\Theta + \cos^2\Theta = 1$$

Thus:

$$F_{res} = \left[m^2 \omega^4 r^2 (\cos^2\Theta + \sin^2\Theta)\right]^{\frac{1}{2}} = m\omega^2 r \tag{7}$$

It will be readily apparent that for a given engine speed, the resultant force is a constant magnitude vector, rotating in the same direction as the crankthrow. Thus, for total resolution of reciprocating forces, crankshaft counterweights with a moment equal to the product of crank throw radius and 1 reciprocating mass is desired.

In order to achieve total reciprocating force resolution by counterweighting, all cylinders, and therefore all reciprocating components must lie in the same plane. In an internal combustion engine where high component structural strength is required, it is advantageous to axially offset the rod-yoke assemblies from each other with each assembly utilizing its own crankpin bearing (a necessity when using a roller crankpin bearing in the yoke). This allows for a more structurally strong design for the rod-yoke assemblies, at the expense of introducing a dynamic couple. In practice, however, this offset is minimal and the resulting couple is small.

In addition to desirable balance characteristics, the 4-cylinder X-configuration allows evenly spaced firing intervals of 90 degrees to be realized, thus reducing the amplitude of torsionally induced vibration. Futhermore, in a two-stroke engine of 3 or more cylinders with overlapping exhaust phases, it is possible to utilize inter-cylinder exhaust tuning to increase both performance and thermal efficiency (14). The physical exhaust system consists of predetermined, equal length branch pipes which all merge in a central collector, whereby a single exhaust stack or silencer may be fitted.

Designing a radial engine to be oversquare (bore/stroke > 1) decreases the overall diameter/displacement ratio, thus reducing the frontal area. Additionally, oversquare geometry greatly reduces reciprocating forces, and therefore, counterweight requirement resulting in a savings in total engine mass. This reciprocating force reduction is accomplished not only by the shorter stroke and therefore lower acceleration values, but also the scotch-yoke components become physically smaller and less massive when compared to a square design. These initial observations prompted a detailed parametric study of internal component mass, inertia forces, and various other design features all of which are functions of the bore/stroke ratio. In generating this model, equations for piston rod length, rod heel length, piston skirt length, and yoke plate dimensions were written to be functions of the stroke. Parameters held constant were cylinder displacement, engine speed, piston rod diameters and material wall thickness. Piston diameter was then calculated from the predetermined stroke and the displacement constant. The mechanical design of the rod-yoke assembly is a modification of the Bourke configuration and is shown in Figure 1. The derivation and descriptions of equations are given in Appendix A. By knowing the dimensional limits plus density of materials to be used, it was possible to estimate volumes and therefore masses of each reciprocating component as a function of bore/stroke ratio. Since it seemed impractical to design with a bore/stroke ratio less than unity (undersquare), the square design (bore/stroke = 1) was considered the lower limit baseline to which comparisons by dimensionless ratios were all based. Figure 2 summarizes the results of this study. The detrimental aspects of an oversquare design are

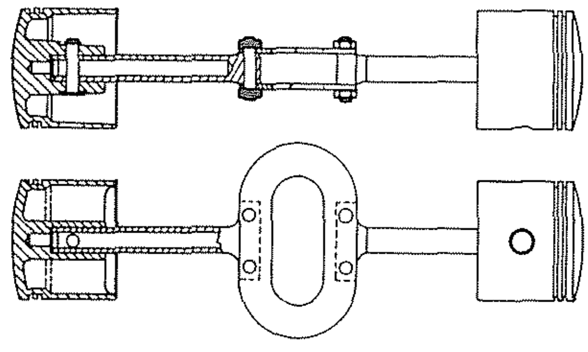

Fig. 1. - Configuration of reciprocating assembly

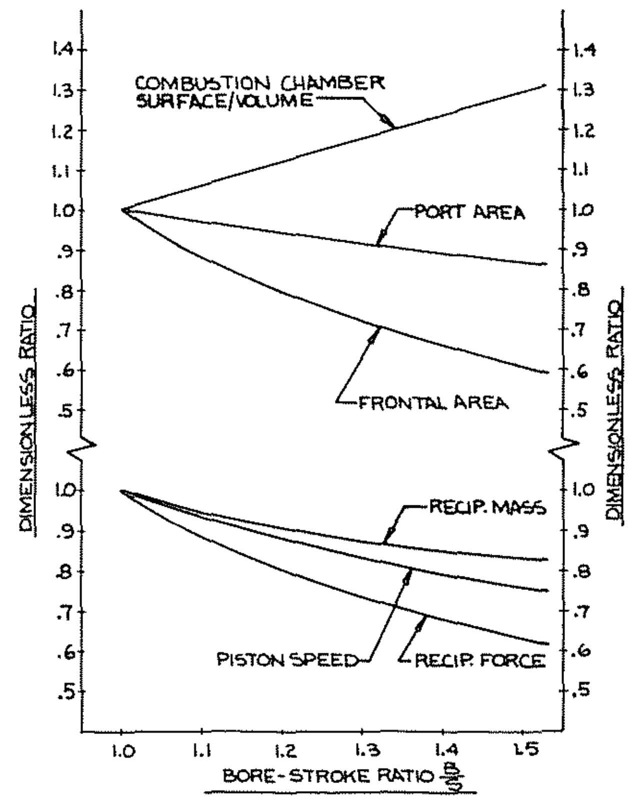

Fig. 2. - Bore/Stroke ratio study

an increase in surface/volume ratio (leading to poorer combustion efficiency and higher thermal losses), and in the two-stroke engine, a reduction in available port area. Figure 2 shows the trends in this regard and it was decided, therefore, to limit the bore/stroke ratio to an upper value of approximately 1.5. The lowermost curves of Figure 2 show the desirable and significant reduction in reciprocating mass and force ratio, plus a dramatic reduction in frontal area ratio as bore/stroke ratios are increased. Utilizing goals of bore/stroke = 1.53, and displacement requirement of 101.4 cc's per cylinder yielded a bore of 58.7 mm and stroke of 38.1 mm. This resulted in a final bore/stroke value of 1.54

and a displacement value of 103.2 cc's per cylinder. Component masses could now be approximated and a kinematic model was next to be constructed.

KINEMATIC MODEL - In studying the prior art, it can be seen that many constructional variations of the scotch-yoke have been applied to proposed internal combustion engine designs (1-10). Inventors have also been torn between use of the traditional yoke slider block or substituting this for a roller (Figure 3). For this design, the crankpin-mounted yoke roller was chosen due to simplicity, especially in regards to lubrication requirements. The roller however, appeared to offer some challenging design problems that would require a detailed study in order to fully understand the dynamic loading and velocity relationships imposed. It can be seen that as loads reverse on the reciprocating components, the roller "swaps sides" in the yoke. If this occurs at a crankshaft position other than exactly mid stroke and assuming the roller was not slipping against the yoke face, the swap will involve skidding of the roller due to the opposing motion of the adjacent yoke face. It can also be seen that during crankshaft rotation, the velocity of linear motion of the roller relative to the yoke faces is changing sinusoidally therefore imparting continual angular acceleration forces to the roller contact face. This combined with the opposing inertia indicated that a combination of rolling and sliding contact would occur under actual operation. Thus, a knowledge of the imposed loads and resultant Hertzian contact stresses would be necessary to adequately size components and to allow selection of a suitable roller material for such conditions.

Basically, the computer model consisted of the accelerative forces generated by the pre-determined mass of a complete scotch-yoke assembly (hereafter referred to as a reciprocating component), over which gas

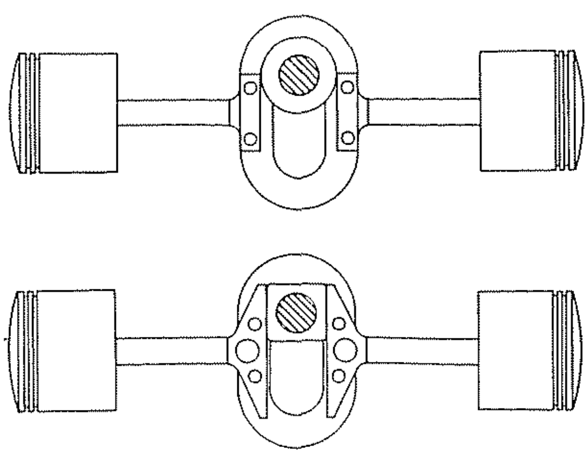

Fig. 3. - Yoke roller (top) and yoke slider block designs utilizing similarly configured reciprocating assemblies

pressure forces are superimposed. Cylinder firing gas pressure values were generated for 10 degree crankangle increments using techniques given in (15) and (16), for various IMEP values. An estimated exhaust port open angle of 100 degrees atdc was utilized, and cylinder pressure values adjusted to atmospheric values after 20 degrees of exhaust opening, simulating blowdown. This "indicator diagram" was stored in a separate file such that appropriate gas pressure values for any portion of the cycle could be selected by the main program as a function of crank angle. Since each opposing piston of a reciprocating assembly has a 180 degree phase relationship, it was a simple matter to select gas pressures at a given crank angle θ, and opposing gas pressures selected at θ + 180 degrees. For this study, sliding frictional forces caused by pistons against cylinder walls were deemed unnecessary and were therefore not included. The piston side load thrust vectors were of interest, however, mainly for comparative purposes to a conventional slider crank system. The side load, or reaction forces, are the result of the moment of the reciprocating assembly about the crankpin, and therefore are applied to the engine frame as reactive torque. The force applied to the crankpin roller through the rod heels is inherently uniaxial. Working through the crankthrow radius, this force imparts a turning moment on the crankshaft. This was also of interest in determining a turning effort diagram for both comparative purposes and crankshaft torsional stress analysis. By summation of torque values for each crank increment for a complete cycle at a given rpm, a convenient method existed for checking agreement between shaft output versus "indicator" IMEP. The roller was modeled as being rotationally frictionless against the crankpin, but suitable frictional coefficients were chosen for the roller-to-yoke contact interface, based on recommended or accepted values for the roller materials considered. Knowledge of forces on the roller and roller velocity are necessary to determine stresses, amount of slippage, and roller to crankpin relative speeds. As previously mentioned, roller inertia was considered to cause resistance to roller rotational acceleration thereby causing slip to occur. The general trends and conclusions derived from this model will be subsequently discussed in detail. All pertinent equations concerning the complete model are given in Appendix B.

Another consideration is the selection of a suitable roller diameter. A small diameter roller is attractive due to the fact that rod heels may be brought closer together thereby reducing the engine external dimensions. This is in conflict however, with the Hertzian stress equations as presented in (17), as these stresses as well as rotational velocity become unacceptably high as roller diameter is reduced. Therefore, a "starting point" roller diameter equal to stroke length was selected, as this appeared visually acceptable for engine size constraints. At this point, the remainder of physical dimensions and masses were input and the model debugged and run for engine speeds between 4000 and 5000 rpm, and IMEP values between 414 and 724 kPa. For comparison purposes, the same gas pressure history was run on a model of a conventional slider-crank engine, where bore and stroke were identical to the scotch-yoke model. Connecting rod length/crank throw radius (L/r ratio) was selected to be 4.0, which is relatively common in practice. Additionally, this resulted in similar dimensions from crank centerline to piston dome such that both models represented physically similar external engine dimensions. Figures 4 through 6 show the results and comparisons between the two crank systems, which will be discussed in detail.

Figure 4 shows piston side thrust comparisons through 360 degrees of crankshaft rotation, at 5000 rpm, 724 kPa IMEP. It will be noted that side thrust on the scotch-yoke piston is slightly more uniform and on one side of the piston skirt, where the slider crank piston loadings undergo large side force excursions in magnitude plus reversals, due primarily to connecting rod angularity. With the scotch-yoke, 2 pistons share the load created by the reaction moment against the crankpin, thus inherently reducing peak loading magnitudes. Due to inertia-gas pressure superposition during the first 90 degrees of rotation from dead center position, gas pressure predominates, maintaining undirectional piston side forces. From the 90 degrees midstroke to dead center of crankshaft rotation, inertia forces predominate over gas pressures, such that piston side thrust remains in the previous same direction. The average piston side thrust for a complete rotational cycle is 50% higher on the slider crank, and the peak side thrust loads are higher by 350% when compared to the scotch-yoke. For a relatively constant speed high duty cycle application, such as encountered with an aircraft engine, the scotch-yoke geometry would seem to offer greatly reduced piston and

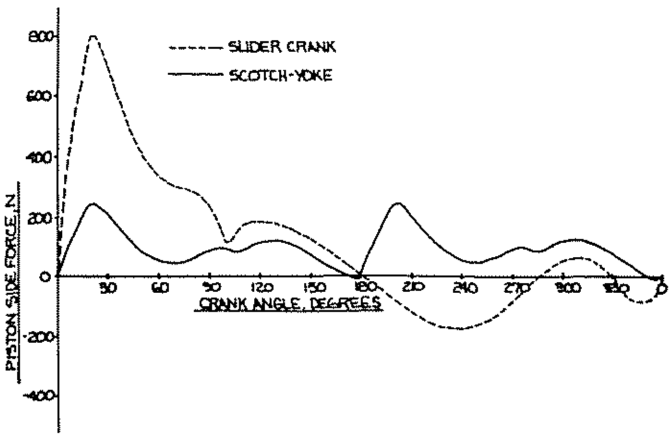

Fig. 4. - Piston side thrust comparison at 5000 rpm, 724 kPa IMEP

cylinder friction and wear. Piston slap would appear to be greatly reduced or eliminated.

Since the magnitude and direction of the force applied to the crankpin is known for both cases, it is now possible to construct a turning effort diagram for each system. This is shown in figure 5 for one complete engine revolution at 5000 rpm, and 724 kPa IMEP. The average turning efforts are virtually identical for two cylinders of a scotch-yoke engine versus a twin cylinder, 180 degree firing slider crank engine. This is, of course, to be expected since indicated work is the same for both models considered. Of more important interest is the more uniform turning effort diagram of the scotch-yoke mechanism, with peak instantaneous torque values some 28% less than the slider crank. This reduction is again primarily due to lack of rod angularity. Figure 6 shows a turning effort diagram comparison for four cylinder engines. A slight improvement in torque uniformity of the scotch-yoke as compared to the slider crank engine is evident.

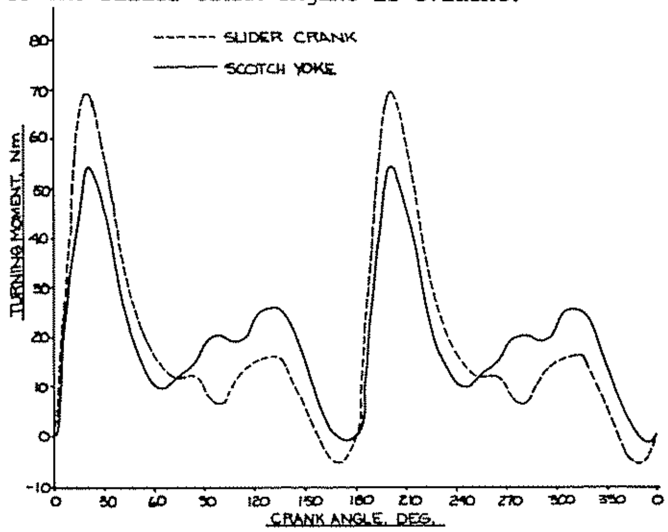

Fig. 5. - 2 cylinder turning effort diagram at 5000 rpm, 724 kPa IMEP

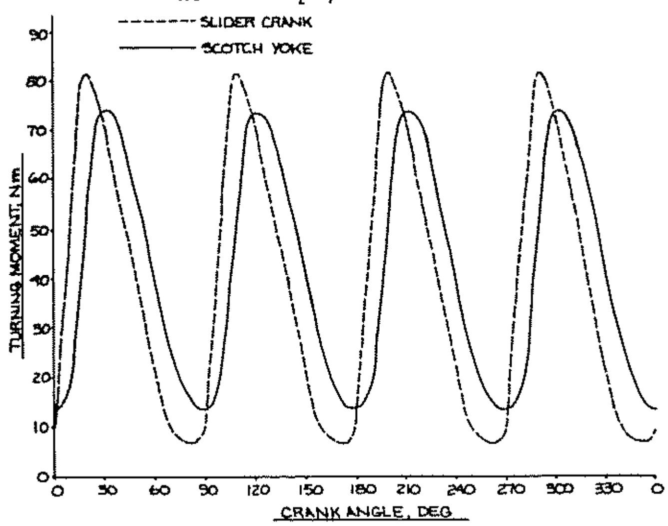

Fig. 6. - 4 cylinder turning effort diagram at 5000 rpm, 724 kPa IMEP

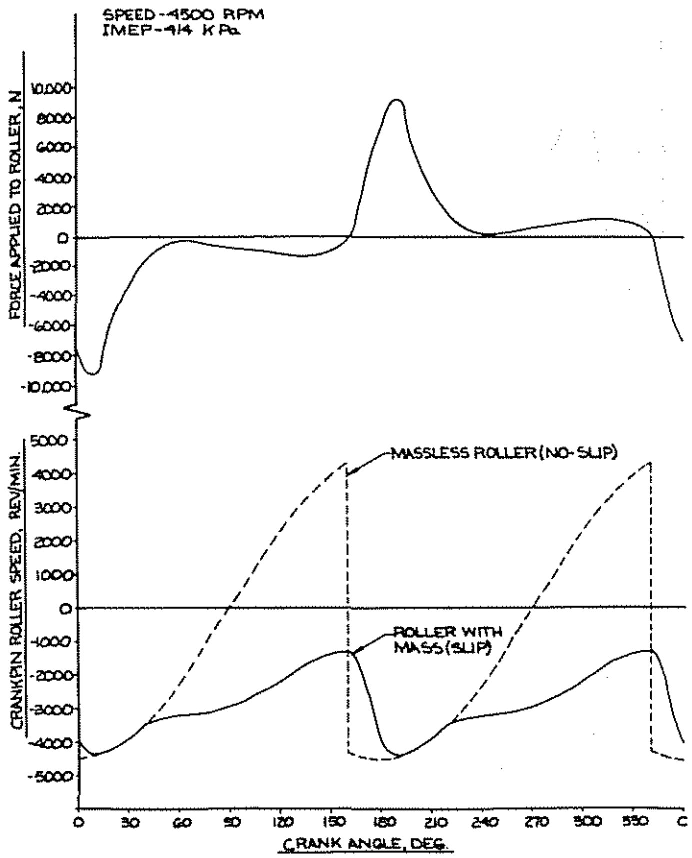

Fig. 7. - Force applied to roller and slip - no-slip roller velocities as a function of crankshaft position at 4500 rpm, 414 kPa IMEP

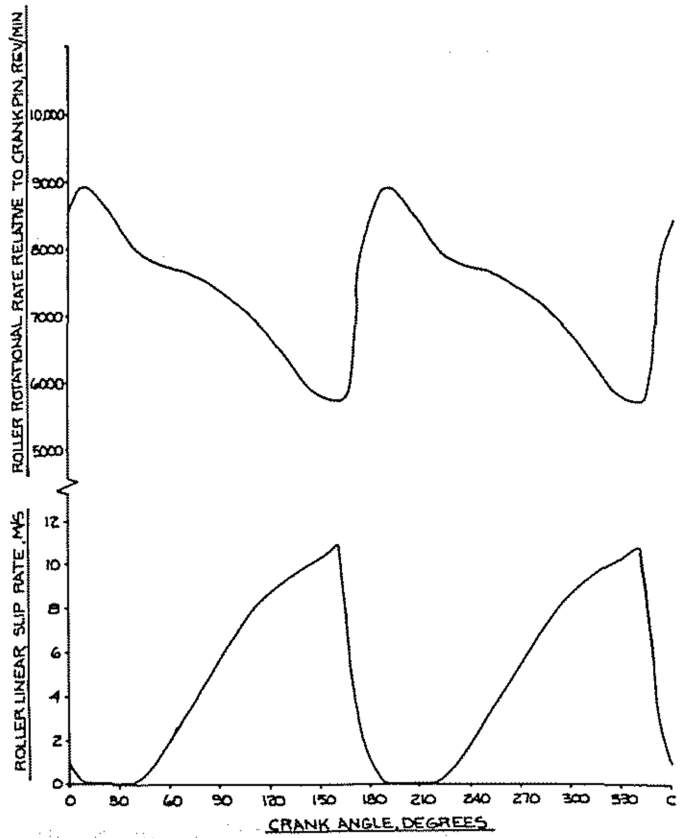

Fig. 8. - Roller angular velocity with respect to crankpin and roller linear slip rate against yoke face at 4500 rpm, 414 kPa IMEP

Of greatest interest however, was the behavior of the crankpin roller under various speed and load conditions. Again, variations in rpm and IMEP were run in the model to ascertain the force applied to the roller by the reciprocating component, the theoretical (no-slip) and actual (slip) roller velocities at a given crank angle, and relative angular velocity of the roller with respect to the crankpin. Figure 7 shows the load history for 360 degrees of revolution (the cycle repeats every 180 crankangle degrees), for 4500 rpm, 414 kPa IMEP. Figure 7 also shows the rotational velocity of a massless roller with no-slip versus the mass roller with an assumed coefficient of dynamic friction of .08. Shortly after the tdc position when gas pressure is greatest, the applied roller load is then greatest and accelerates the roller to no-slip speed. As the crankshaft turns through 40 degrees atdc, the applied force is decreasing, roller inertia overcomes frictional forces and slip begins to occur. At the 90 degree crankangle, or mid stroke position, the theoretical roller velocity is zero due to no relative motion of the roller across the yoke faces. However, inertia overcomes the frictional load and the roller continues to rotate, as is true throughout the remainder of the cycle. At 170 degrees atdc, the reciprocating force direction changes sign, due to commencement of combusion in the opposing cylinder. This force reversal causes a discontinuity in the no-slip roller velocity due to the "swap sides" effect, and the roller is again accelerated to no-slip velocity. Thus the roller does not reverse direction but continues to rotate unidirectionally, with slip occurring throughout the majority of the cycle. Figure 8 shows slip as linear velocity of the roller face against the yoke, calculated from the difference between the no-slip versus slip angular velocities. Figure 8 also shows the angular velocity of the roller with respect to the crankpin. This velocity is approximately double that of engine rpm, due to the crankpin rotating in the opposite direction with respect to the roller. Figures 9 and 10 show similar behavior when the roller is subject to 724 kPa IMEP and 5000 rpm engine speed.

Hertzian contact stresses were analyzed to determine a suitable material combination which would function under the line contact loading conditions imposed. Since the piston rods and integral yoke working faces were to be made of steel, it was desirable to use a dissimilar material for the yoke roller for good wear properties under sliding conditions. Aluminum

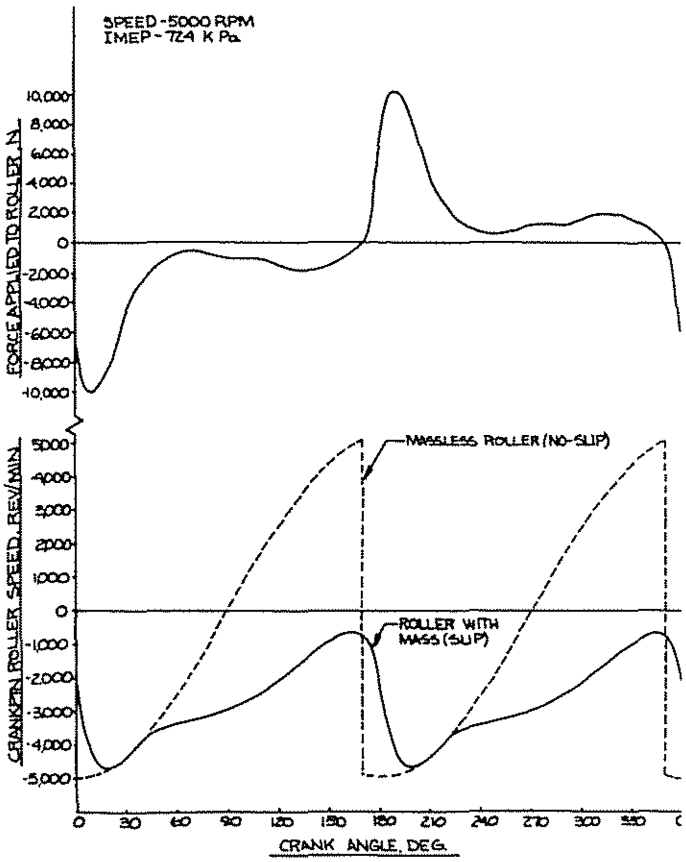

Fig. 9. - Force applied to roller and slip - no-slip roller velocities as a function of crankshaft position at 5000 rpm, 724 kPa IMEP

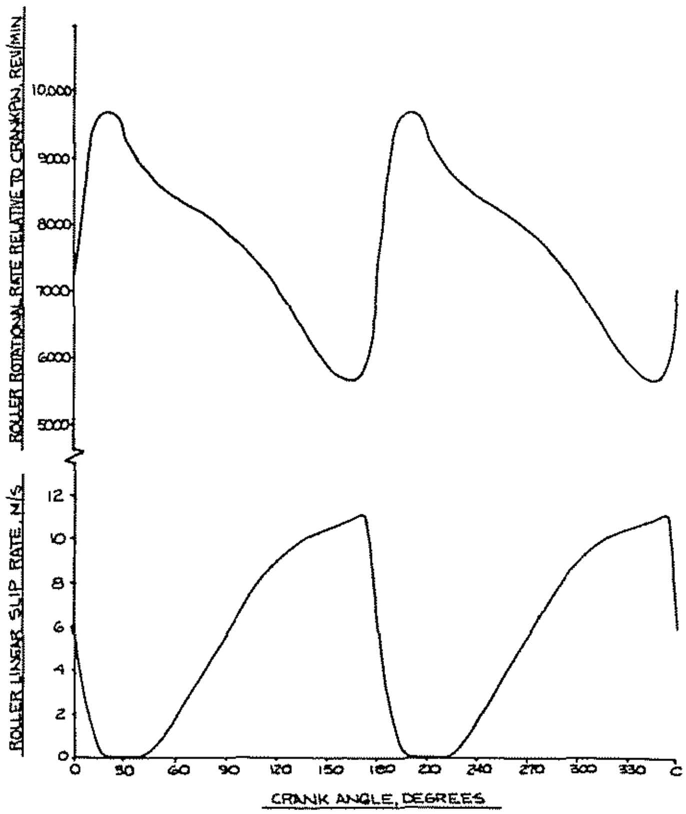

Fig. 10. - Roller angular velocity with respect to crankpin and roller linear slip rate against yoke face at 5000 rpm, 724 kPa IMEP

of a 7075 T-6 alloy was initially considered as a roller material with a hard anodized working surface, but the material properties were not adequate for high contact stresses as predicted by the Hertz equations. The properties of aluminum-bronze alloy were found to be suitable from a stress standpoint when suitable roller dimensions were selected. The alloy chosen for the roller was sold by the trade name of Ampco 21W, which offers a high compressive strength with excellent shock loading resistance. Further numerical analysis of the dynamic stress state of the roller utilizing the Smith-Liu equations (18) indicated that the largest compressive contact stress, occuring at the surface of the roller, is within a safe range with regard to physical properties of the material. The Ampco alloy is also suited as a journal bearing alloy and therefore can be used as the crankpin bearing surface, running directly against a hardened steel crankpin. Therefore, each roller assembly becomes a relatively simple, single piece component made entirely of aluminum-bronze.

The kinematic model was further expanded to include main bearing loads. This was based on the known gas pressure plus inertia loads imposed on the crankpin by the reciprocating assemblies and the rotating masses of the yoke rollers, crankpin, and counterweights. The resulting main bearing loads were surprisingly uniform in magnitude and low enough such that medium duty ball bearing assemblies were more than adequate for this task. Thus, suitable capacity bearings were selected and sized for a minimum B-10 life of 2000 hours at the highest anticipated load conditions. Next, bending loads and stresses were examined in the rod-yoke assembly, and parts sized accordingly to have unlimited life under the cyclic loading values as predicted by the model. Since it was desirable to use the smallest outside diameter tubular piston rod consistent with adequate strength, a 12.7 mm diameter was initially selected. Bending loads indicated that the rods could safely have an inside diameter of 8 mm, thus yielding a high section modulus and low mass. Stresses in the yoke plates were examined by using classical curved beam equations (17) as well as column strength loading, and it was found that the latter was most critical. To adequately resist deflection under the predicted forces for compressive loading, the yoke plates were necessarily made 2.3 mm thick each. This provided ample fatigue strength under cyclic loading conditions. At this point, the kinematic analysis of the stressed internal components showed the design to be feasable and workable. With this reassuring information obtained, the mechanical design was next begun.

MECHANICAL DESIGN

GENERAL - In studying the patented prior art concerning previous attempts at scotch-yoke two-stroke engines, several features were independently incorporated into the majority of the designs which seemed to be their detriment. One of these features, which seems to be irresistible to inventors, is to make the scavenge pump volume as small as possible, thereby obtaining high primary compression ratios. Since the crankcase is not part of the scavenge pump, these small volumes are easily obtained with this type of engine. The author's experience with testing an engine of this type (11) indicated that the high primary compression ratio, in this case of almost 3:1, apparently causes a choked flow condition to exist at the transfer ports during the initial scavenging portion of the cycle. The resulting sonic discharge of mixture into the cylinder apparently upsets the normal scavenging process by inducing direct short-circuiting of the mixture into the exhaust, as was indicated by the low trapping efficiency as measured by a free oxygen sampling probe placed in the exhaust system. The poor scavenging seemingly caused by the high scavenge pressure ratio, was the major cause of poor performance from this particular engine. The current state-of-the-art regarding primary compression ratios of crankcase pumped engines indicated that highest performance is obtained using ratios of 1.2 to 1.5:1. For an engine without radical exhaust tuning, experience shows that primary compression ratios of 1.5:1 are the highest that is normally required. Thus a ratio of between 1.4 and 1.5:1 was a reasonable value to strive for on the scotch-yoke engine.

A second, seemingly unnecessary feature which inventors find attractive is to support and axially guide the reciprocating assemblies by bushings located in the crankcase, through which the circular cross-section piston rods must pass. The bushings therefore absorb all torque producing reaction forces of the rod-yoke assembly, allowing the pistons to be free of side loads. There are several disadvantages to this arrangement. The major disadvantage is that high unit pressures (from reaction forces) developed over a small contact area (piston rod diameter) may create a good deal of friction and potential wear. The remedy is to eliminate these bushings, rigidly affix the piston to the rods, and allow the piston skirts to take the thrust loads directly. From the previous discussion, it was shown that the average loads are half that of a geometrically similar slider crank engine, and the larger contact surface of the piston (as compared to the piston rod) yield lower unit pressures. This simpler and more practical design was that used on this new engine.

A distinction should be made here between the conventional crankcase scavenged two-stroke, versus the scotch-yoke two-stroke engine utilizing the piston underside as the scavenge pump and tubular piston rod sealed at the crankcase face. The conventional engine, with the entire crank mechanism contained in the scavenge pump volume (this being the crankcase), has a pumped/cylinder swept volume ratio of 1, assuming of course that no double diameter or differential pistons are used. In the scotch-yoke engine the crank mechanism is isolated from the scavenge pump with the mechanical connection being the tubular piston rod, the presence of which reduces the effective area of the underside of the piston. This in turn makes the pumped/cylinder volume ratio less than unity. With a highly oversquare engine geometry, this effect is lessened due to a relatively large piston diameter relative to the piston rod. For an engine where high output at WOT is desired, this volume ratio should be as close to unity as possible to promote the highest possible delivery ratio. This is best obtained in the scotch-yoke engine by designing with an oversquare geometry combined with the use of the smallest diameter piston rods consistent with adequate strength. For the subject engine using a 58.7 mm bore with a 12.7 mm diameter piston rod, the pumped/cylinder swept volume ratio is .953. For determination of scavenge pump delivery ratio, the reduced swept volume should be used in the calculations rather than cylinder swept volume, as in conventional practice.

CYLINDERS - It was decided that a Schneurle loop scavenging system was to be employed. This system allows for a favorable combustion chamber shape and high specific power outputs to be developed. For the sake of simplicity, a piston controlled inlet port was utilized. In most traditional two-stroke designs, the inlet port is located diametrically opposite of the exhaust port, greatly complicating the implementation of back transfer ports. Experience indicates the desirability of using back transfer ports due to their contribution to overall transfer time area plus possible improvements in scavenging efficiency. Placement of the inlet port directly under the exhaust port allows implementation of back transfer ports and unrestrictive passages. The resulting cylinder port development is shown in figure 11. With the close proximity of inlet and exhaust passages, adequate heat break slots were provided in the casting design to minimize heat transfer between these passages. With the radial configuration two-stroke engine, there exists no limitations to transfer passage design which can occur in inline cylinder bank arrangements due to close cylinder spacing. Thus, full advantage was taken to make the transfer ducts unrestrictive and aerodynamically correct, with generous turning radii and slight but consistent cross sectional area convergence. The shape of the duct is rectangular in cross section, which is generally thought to be superior than the skewed or rhomboidally shaped passage commonly used.

The cylinder head was designed to be

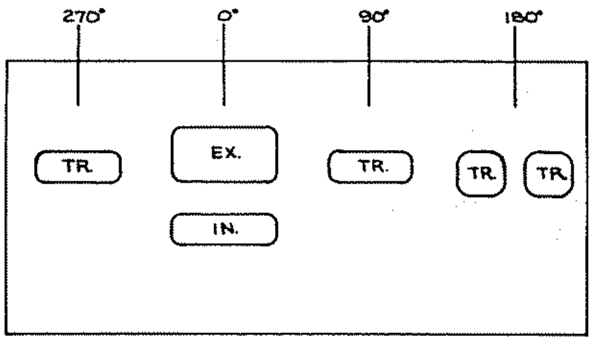

Fig. 11. - Cylinder development

integral with the cylinder casting for reasons of simplicity and reliability, in that no head gasket is necessary. This form of construction also allows for better heat transfer between the cylinder and head. The actual cooling fin area totals 925 cm^2/kW or about 2.7 times the recommended value (19), and while this adds some additional mass and bulk to the engine, the tradeoff was beneficial for the application. A removable iron alloy sleeve is used, being a .13 mm interference fit in the aluminum cylinder casting. The sleeve projects from the cylinder mounting face to pilot the cylinder to the crankcase, thereby ensuring alignment with the piston rod axis.

With regard to port timings and time-area values, certain differences arise in these parameters with a scotch-yoke crank mechanism versus the slider crank. The pure harmonic piston motion afforded by the scotch-yoke allows equal piston dwell periods at both top and bottom dead center positions, whereas the slider crank has smaller dwell values around tdc and larger values near the bdc positions. A comparison is shown in table 1 for the exhaust port and can be summarized as follows: for transfer and exhaust ports that occupy the same percentage of stroke in otherwise identical engines, the scotch-yoke engine will have lower time-area values than the slider crank equivalent. For piston controlled inlet ports that occupy the same percentage of stroke in otherwise identical engines, the scotch yoke engine will have higher time-area values. Suitable time-areas for all ports were calculated by the methods outlined in (20), and are given in table 2.

PISTONS - Since no wrist pin is necessary, a rigid attachment of piston to rod is possible via a boss cast integral with the piston dome. This boss receives the tubular piston rod and the piston is retained by a threaded fastener located perpendicularly to the rod axis. This design offers a direct heat flow path from the center of the piston dome area into the piston rod, with the heat ultimately being transfered into the crankcase lubricating oil. This feature would seem to ensure a piston that is less prone to thermal failures.

The piston skirt is taper turned and carries two rectangular cast iron rings of 1.6

Bore	- 58.7 mm
Stroke	- 38.1 mm
L/r, Slider Crank	- 4.0

Exhaust Port Specifications

Width	- 31.6 mm
Height	- 17.4 mm
Corner Radius	- 3.2 mm
Number of Ports	- 1

	Scotch-Yoke	Slider Crank
E.O., deg ATDC	95.0	87.7
Specific Angle Area, Deg. cm^2/cm^3	5.71	6.35

TABLE 1 - Exhaust Port Time-Area Comparison

mm thickness. The rings are pinned to prevent rotation in their grooves. The dome is fully machined on a radius.

CRANKCASE - A single piece, barrel type crankcase of 356 T-6 aluminum alloy was designed, around which the remainder of the engine is based. Since perpendicularity and parallelism of adjacent and opposing mounting faces is critical on this type of engine, emphasis was put into a configuration that could be relatively easy to machine in order to hold these tolerances. The crankcase is essentially machined as a 6 sided cube, with all internally machined bores referenced from principle surfaces.

Externally, the four surfaces that receive the cylinders are identical. Each face has a suitable depression from the machined surface, and when the cylinder is fitted, the free volume contained therein allows the primary compression ratio to be at the desired value of 1.4:1. Since a dry sump lubrication system was intended, the crankcase interior is provided with a small oil sump where oil is collected and gravity fed to an external tank.

BEARING HOUSINGS - These 356 T-6 aluminum alloy components attach to opposing crankcase faces by threaded fasteners. The housings are located radially by pilot diameters which register in the crankcase, with sealing provided by O-rings. The front housing contains an outwardly positioned ball bearing which functions as a crankshaft locator and thrust bearing. An inwardly mounted ball bearing functions as the forward crankshaft main bearing.

The rear bearing housing also contains two similarly sized ball bearing assemblies. The inwardly placed bearing forms the rear

	INLET	MAIN TRANS.	BOOST TRANS.	EXHAUST	BLOWDOWN
Port Width, mm	31.6	24.8	15.2	31.6	31.6
Port Height, mm	10.3	9.5	9.5	17.4	7.9
Top R., mm	3.2	3.2	3.2	3.2	3.2
Bottom R., mm	3.2	3.2	---	3.2	---
# Ports	1	2	2	1	1
Port Opens, Deg.	63 BTDC	120 ATDC	120 ATDC	95 ATDC	95 ATDC
Total Open Angle, Deg.	126	120	120	170	25*
Total Open Area, cm^2	3.2	3.4	1.4	5.4	2.4
Mean Open Area, cm^2	2.1	2.2	.9	3.5	1.2
Specific Angle Area, $\frac{Deg.\ cm^2}{cm^3}$	2.5	2.6	1.1	5.7	.3

* From E.O. to T.O.

TABLE 2 - Engine Porting Specifications

crankshaft main bearing, with the smaller outwardly placed bearing contributing to accesory drive support. To the outside of the bearing cavity is a pocket containing a crankshaft driven oil pump. Additionally, the housing contains an annular induction manifold and carburetor mounting. A separate plate covers both the manifold and oil pump. This plate contains drilled passages for the oil system plus bosses for ignition system and engine mounting.

CRANKSHAFT - The single throw crankshaft is of the built-up type, using press fits. Each identical crankthrow-counterweight is a 4140 alloy steel casting, with heat treated alloy steel shafts pressed in and welded to each crankthrow, thus forming the main journals. The case hardened crankpin is pressed into each throw and retained by an interference fit. Yoke plates and crankpin rollers are placed on the crankpin prior to the final pressing operation.

A unique situation arises when a built up crankshaft is designed for a short stroke engine such as this. With adequate diameters selected for journal and crankpin, these bores tend to overlap in the crankthrow, eliminating the web of material that normally exists between the bores. In order to take the place of this material and to provide adequate strength for a press fit, a high strength threaded fastener is used, passing through the crankthrow-main journal area as shown in figure 12. This fastener is initially installed and tightened prior to final machining of the crankpin bore, and is never removed or loosened thereafter. The fastener provides the necessary clamping force to assure a uniform press fit of the crankpin into the throw.

The propeller hub is joined directly to the crankshaft by a locking taper and draw bolt. At the accessory end of the engine, a keyway is provided in the crankshaft journal to engage the oil pump; a drive for the ignition system is provided adjacent to this.

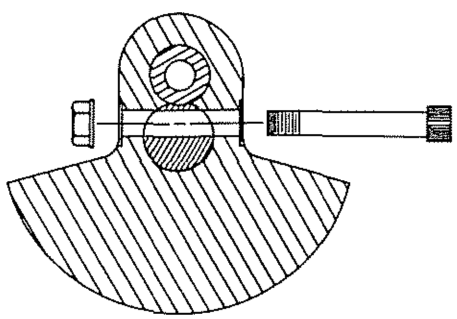

Fig. 12. - Crankthrow showing fastener

ROD-YOKE ASSEMBLY - Piston rods are machined from billet 4140 steel and heat treated. Yoke plates are made from tool steel plate due to this material's resistance to warpage during surface grinding, and are heat treated prior to machining. The completed assembly is shown in Figure 1.

SEALS - The alternating pressures of each scavenge pump chamber must be sealed from the crankcase, and conversely oil mist in the crankcase prevented from reaching the cylinders in quantity. Seal materials must be able to withstand the relatively high temperature of the piston rods, and must have a low dynamic coefficient of friction. In addition, seal material must be resistant to wear under reciprocating service, and must be highly resilient such that sealing is maintained under slight radial displacements of the piston rods due to necessary running clearances. A Teflon-graphite matrix, spring assisted lip seal has proven to be well suited for this application. The seal is augmented by a relatively close fitting oil baffle surrounding the piston rod and facing into the crankcase. This baffle deflects the majority of oil away from the seal and has worked well in testing.

LUBRICATION SYSTEM - A single oil pump is employed, which draws oil from an external tank. The lubricant is routed to the top of the crankcase via an external line, through an orifice, and directed onto the rod-yoke and crankpin roller assemblies; the crankshaft ball main bearings are in turn lubricated by splash. The oil collects in the sump portion of the crankcase and returns to the oil tank by gravity, the tank being located below the crankcase. Since the cylinders and pistons are effectively isolated from the crankcase interior, a means must be provided for piston and ring lubrication. External tubes that communicate with the crankcase interior are routed to each cylinder via a check valve and sized orifice. In turn, a drillway leads to the cylinder wall surface, and is positioned such that the skirt of the piston uncovers the drillway at tdc position, allowing scavenge pump vacuum to draw in a small quantity of oil from the crankcase. This, plus the slight amount of oil that gets by the rod seals, has proven sufficient to adequately lubricate pistons and rings throughout the operational range of the engine. Naturally this is a total loss type of system, but due to the small lubrication requirement of the pistons and rings, oil consumption has proven comparable with conventional oil injected two-stroke engines.

ACCESSORIES - The engine is presently configured to utilize a single magneto, driven directly from the accessory end of the crankshaft. Carburetion is provided by a slide throttle valve carburetor, spigot mounted in an updraft position. The exhaust system consists of welded steel tubing with equal length branches communicating into a common merge, where a single exhaust stack or muffler may be fitted. The branch lengths are sized such that the system "tunes" at the desired 4500-5000 rpm operational range.

Front and side internal views of the engine assembly are shown in Figures 13 and 14, and the prototype shown in Figures 15 and 16.

TESTING - Due to the economical nature of this project, dynamometer procurement or rental was not feasable. Instead, a test stand was designed and constructed so as to measure torque transmitted by the engine frame with an aircraft propeller acting as the load device. This has the advantage of supplying cooling air to the engine while applying thrust loads that would be encountered during actual aircraft operation. However, since the propeller load and engine load curves intersect at one point, this

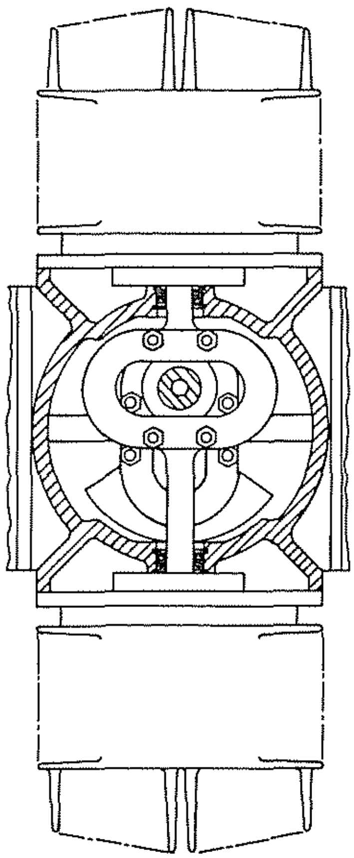

Fig. 13. - Cutaway of complete engine, front view

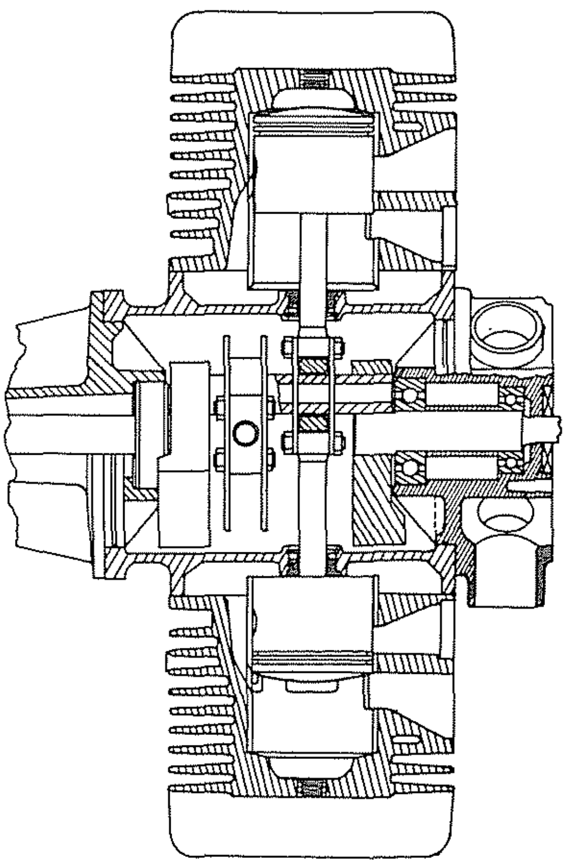

Fig. 14. - Cutaway of complete engine, side view

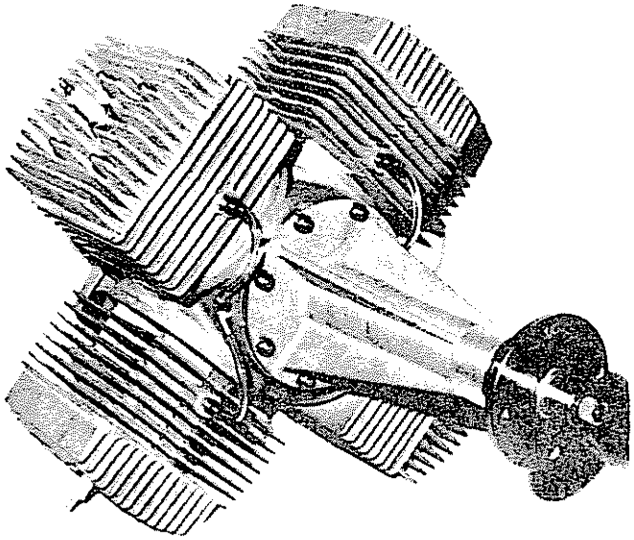

Fig. 15. - Front view, first prototype

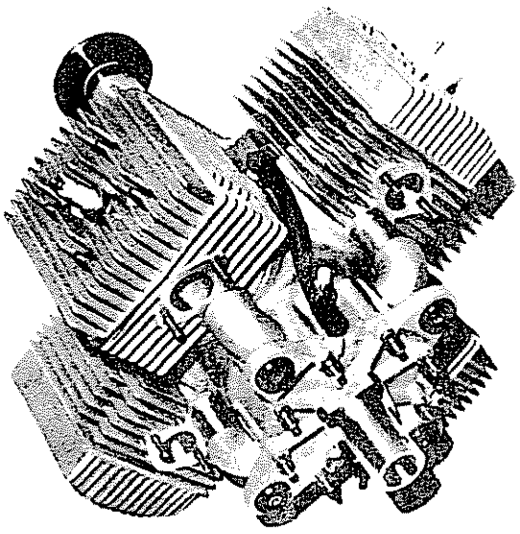

Fig. 16. - Rear view, first prototype less ignition and exhaust system

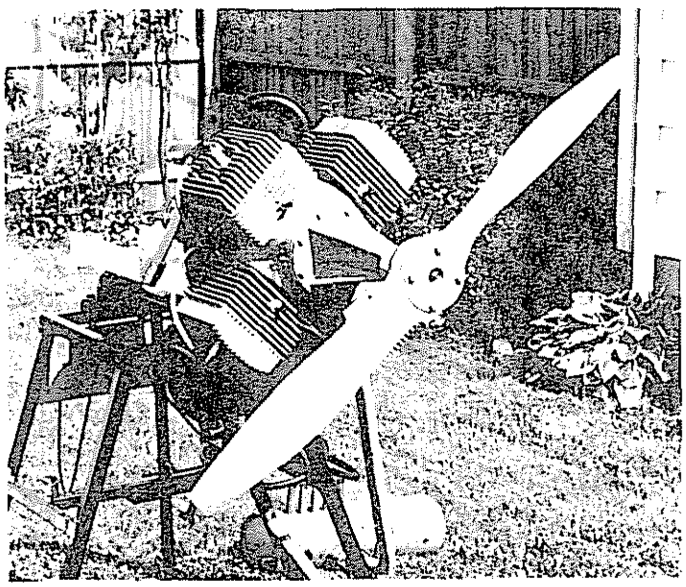

Fig. 17. - Engine on test stand, front view

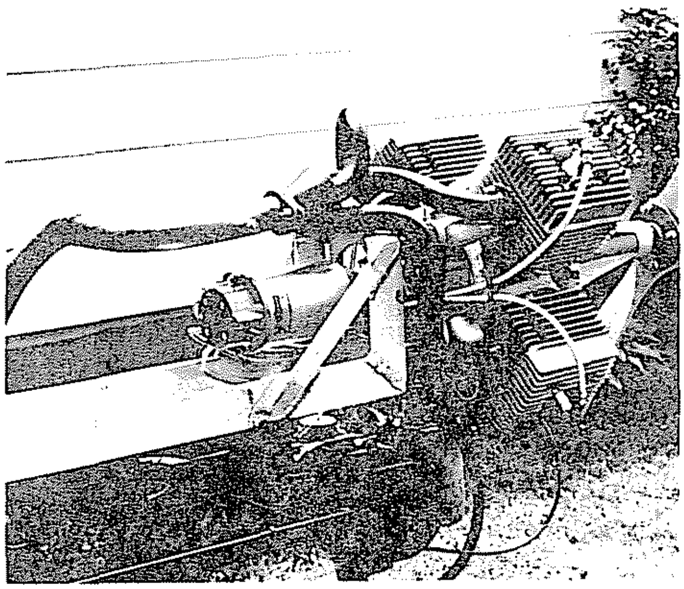

Fig. 18. - Rear view of engine on test stand showing magneto and exhaust system

necessitates using a selection of various pitched propellers in order to run a full, wide open throttle power curve. This disadvantage is offset by the test stand's ability to double as an endurance test fixture. The stand with engine mounted, is shown in Figures 17 through 19.

Several hours of part throttle break-in were performed, at which time a propeller selection was made to load the engine in the desirable rpm range at wide open throttle. After further optimization of carburetion and ignition settings, a reading of 19.8 kW observed power at 4800 rpm was recorded on the torque stand yielding a BMEP of 598 kPa, thus slightly exceeding the target design performance. At this output the engine used 25 cc's of fuel in 8.0 seconds, for a WOT BSFC of 409.4 g/(kWh). For a 75% power setting simulating cruise, this translates to 14.8 kW which the given propeller absorbs at 4400 rpm. At this speed, the engine used 25 cc's of fuel in 11.9 seconds for a cruise BSFC of 368.6 g/(kWh).

After several more hours of full throttle operation, the spark plugs were examined for coloration, indicating mixture distribution quality. All four spark plugs showed very uniform coloration, indicating that the piston port induced inlet dynamics occuring in the inlet manifold are sufficiently high in amplitude to maintain even mixture distribution.

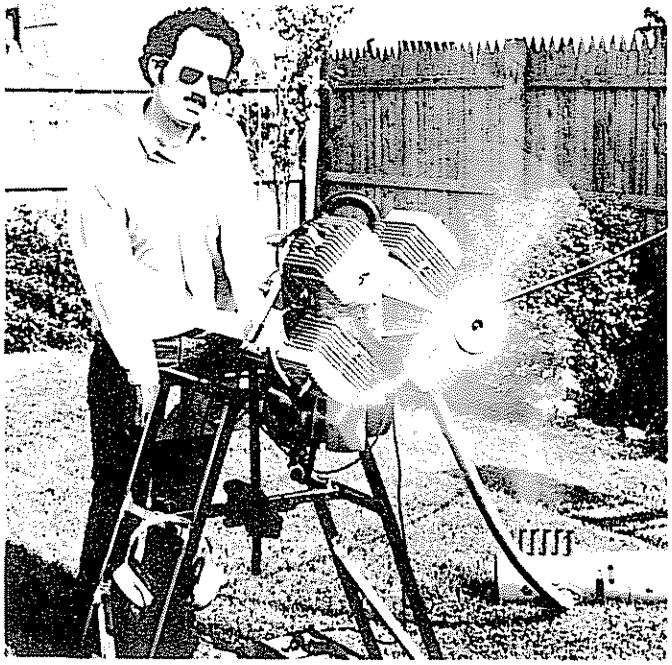

Fig. 19. - Engine running on test stand

At present, the prototype engine has now acrued 30 hours of mostly full throttle operation with no degradation in performance. Overall running qualities are very acceptable, with good throttle response and rapid acceleration, and low vibration levels likened more to a rotary rather than a reciprocating engine.

Checking the clearances between the crankpin roller and yoke indicate virtually no wear has occurred during this time. This clearance measurement, incidentally, is accomplished without disassembly of the engine by using a dial indicator fitted into a spark plug hole and contacting the piston dome. By rotating the crankshaft, the piston is brought to tdc and the indicator zeroed. By pressing on the opposing piston's dome with a wooden dowel, the clearances may be read directly.

Examination of pistons and rings at 25 acrued hours showed no measurable wear with most of the original machining marks still evident, thus substantiating the low side thrust values that the model predicted. Piston rod seals appeared in excellent condition with no deterioration in sealing qualities being noted.

CONCLUSION

An engineering design study has been presented concerning a two-stroke cycle, scotch-yoke, light aircraft engine. From this study, a prototype engine was constructed and is now undergoing endurance testing. The performance goals were met or exceeded, with the prototype demonstrating acceptable running qualities with low vibration. The two major areas of concern with this design, these being the durability of both the crankpin yoke roller bearing and piston rod seals, will be assessed as more endurance hours are obtained.

NOMENCLATURE

A	- area
a	- acceleration
B	- cylinder bore diameter
D	- material density
d	- diameter
F	- force
I	- inertia
k	- constant
L	- length
M	- mass
m	- moment
N	- rpm
P	- pressure
p	- position
r	- crank throw radius
S	- stroke = 2r
T	- time
t	- material thickness
V	- volume
v	- velocity
α	- angular acceleration
ω	- rotational velocity

SUBSCRIPTS

a	- aluminum
c	- crankshaft
f	- frontal
g	- gas pressure
I	- inertia
i	- inner
o	- outer
P	- port
p	- piston
R	- reciprocating
r	- rod
S	- side
s	- steel
y	- yoke plate
ϕ	- roller

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APPENDIX A

Consider various practical bore/stroke ratios and evaluate parameters.

<u>Piston Rod:</u>

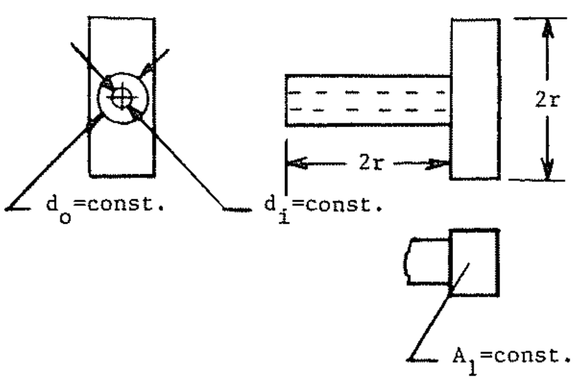

Fig. 20

$$V_r = 2R\left[\frac{\pi}{4}\left(d_o^2 - d_i^2\right) + A_1\right] \tag{8}$$

$$M_r = D_s V_r \tag{9}$$

<u>Piston:</u>

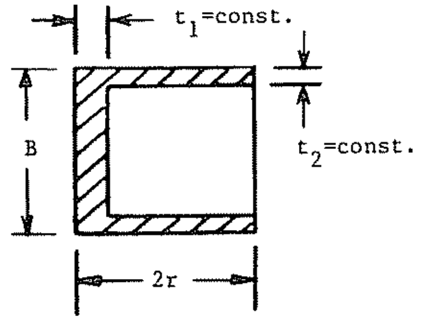

Fig. 21

Since cylinder displacement is held constant, cylinder bore becomes a function of the stroke.

$$B = \left[\frac{2\ V_{cyl}}{\pi r}\right]^{1/2} \tag{10}$$

Therefore, an expression may be written for material volume of the piston as a function of the stroke.

$$V_p = \frac{\pi}{4}\left[B^2 2r - \left(B-2t_2\right)^2 \left(2r-t_1\right)\right] \tag{11}$$

$$M_p = D_a V_p \tag{12}$$

<u>Yoke Plate:</u>

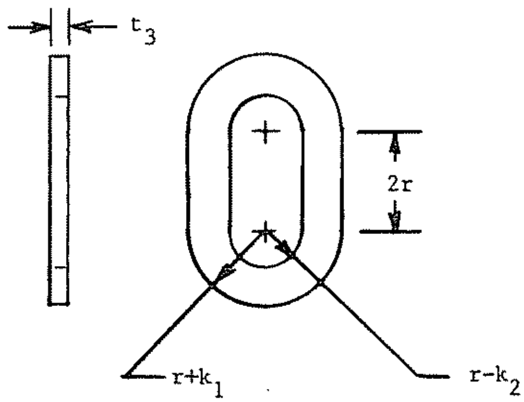

Fig. 22

$$V_y = \frac{\pi}{4}\left[\left(r+k_1\right)^2 - \left(r-k_2\right)^2\right]t_3 + 2\left[2r\left(k_1+k_2\right)\right]t_3 \tag{13}$$

$$M_y = D_s V_y \tag{14}$$

Total mass for one reciprocating component is

$$M_R = 2\left(M_r + M_p + M_y\right). \tag{15}$$

Force generated by this mass at a given crankshaft speed is given by

$$F_R = M_R \omega_c^2 r \cos\Theta \quad . \tag{16}$$

Average piston speed is given by

$$V_p = \frac{Nr}{3} \quad . \tag{17}$$

Frontal area of a cowled radial engine approximates a disc. Assume diameter to equal length of one reciprocating component, plus stroke, plus a constant.

$$A_f = \frac{\pi}{4}\left(12r+k_3\right)^2 \tag{18}$$

Port area is calculated by assuming port width is a fixed percentage of bore diameter, and port height is a fixed percentage of stroke.

$$A_p = k_4 B k_5 r$$

Substituting equation 10 for bore in terms of stroke.

$$A_p = \left[\frac{2\ V_{cyl}}{\pi r}\right]^{1/2} k_4 k_5 r \tag{19}$$

Since exhaust port height is held at a fixed percentage of stroke length, and compression ratio is constant for all cases considered, compressed volume is therefore constant.

$$V_2 = k_6$$

Considering a cylindrical volume of infinitesimal height, surface area is approximated by

$$A = \frac{\pi B^2}{2} \quad .$$

Therefore, combustion chamber surface area to volume ratio is given by

$$\frac{A}{V} = \frac{\pi B^2}{2V_2} \quad .$$

Substituting equation 10 for bore in terms of stroke and simplifying,

$$\frac{A}{V} = \frac{1}{V_2}\left[\frac{\pi V_{cyl}}{2r}\right]^{1/2} . \tag{20}$$

APPENDIX B

Roller acceleration modeled using D'Alembert's principle (16). With program steps of 10 degree increments and time units of seconds, the following is obtained:

$$T = \frac{1.667}{N} \tag{21}$$

Thus

$$\omega_2 = \omega_1 \pm \frac{1.667 r\mu \left|F_R\right|}{NI} \tag{22}$$

Roller inertia is given by

$$I_\phi = \frac{M_\phi\left(d_o^2 + d_i^2\right)}{8} \tag{23}$$

Reciprocating force is modeled by the following:

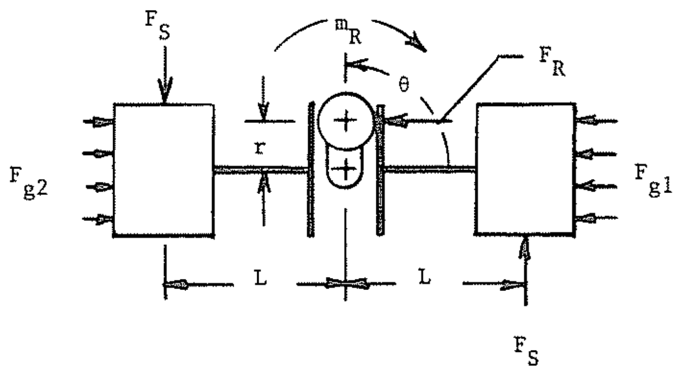

Fig. 23

$$F_R = F_{g2} - F_{g1} + F_I \tag{24}$$

$$F_S = \frac{F_R r \sin\Theta}{2L} \tag{25}$$

APPENDIX C

ENGINE SPECIFICATIONS

BORE	58.7 mm	2.3125 in.
STROKE	38.1 mm	1.500 in.
DISPLACEMENT	413.0 cm^3	25.2 $in.^3$
COMPRESSION RATIO	7.0:1	7.0:1
DIAMETER	413 mm	16.25 in.
LENGTH	533 mm	21.0 in.
FRONTAL AREA	944 cm^2	146.3 $in.^2$
WEIGHT, COMPLETE	27.2 kg	60 lbm
WEIGHT/POWER	1.46 kW/kg	2.4 lbm/hp
FUEL	Leaded Auto or Aviation Gasoline	
OUTPUT/DISPLACEMENT	45.3 kW/L	.992 hp/$in.^3$
OUTPUT/PISTON AREA	.043 kW/cm^2	.372 hp/$in.^2$
PISTON SPEED (MAX.)	6.35 M/sec	1250 ft/min.
BMEP (MAX.)	542 kPa	78.6 psi
OUTPUT (TAKE OFF)	18.7 kW @ 5000 rpm	25 bhp @ 5000 rpm
OUTPUT (RATED)	18.7 kW @ 5000 rpm	25 bhp @ 5000 rpm
OUTPUT (CRUISING)	15.6 kW @ 4700 rpm	21 bhp @ 4700 rpm