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Design and preliminary testing of a rotary internal combustion engine

Frank Darrell Statkus

DESIGN AND PRELIMINARY TESTING

OF A

ROTARY INTERNAL COMBUSTION ENGINE

BY

FRANK DARRELL STATKUS, 1945-

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ABSTRACT

With the advent of legislation against pollution, and rising wage, manufacturing, and materials costs, the prime mover industry is presently intent on developing new economical, non-polluting methods of power. It was the hope of the author that the engine described herein would provide an advancement towards meeting the new requirements.

The design and resulting prototype is an eight cylinder rotary internal combustion engine, weighing approximately sixty pounds, and operating on the four stroke principle. Theoretical calculations were made to determine ideal horsepower, mean effective pressure, and efficiency. Ideal horsepower ranged from 38.7 to 62.8. A dynamic analysis of the pistons indicates that, neglecting external loads and friction forces, they caused no net work on the system. Initial testing of the prototype with combustion occurring indicated conceptual feasibility and good cycle characteristics, although poor sealing caused the compression ratio to be too low for sustained operation. With proper seals, the engine's performance should approach the calculated values.

ACKNOWLEDGEMENTS

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CHAPTER I

INTRODUCTION

In 1876 Otto, a German, built an engine which operated on the principles set forth by Beau de Rochas in 1862, (see reference 1). Most of the piston combustion engines built today operate in the same general manner as did Otto's gas engine, and consequently the term Otto cycle is used for the series of events that make up the cycle in today's four-stroke-cycle engines.

Since the turn of the century, the Otto cycle has been used extensively by reciprocating internal combustion engines. Over the years, this type of engine has become the nucleus around which a great deal of engineering and testing has quickly evolved. As a result, the internal reciprocating engine of today faces a dilemma brought about by the pace of progress of its own development. Yesterday, size and horsepower were the foremost thoughts of automotive engine designers. Today though, while these ideas certainly still play an important role in the combustion engine analysis programs, they are not the only problems which must be dealt with. New ecological terms such as noise control, exhaust emissions, economy, size and weight have indicated that a reevaluation of the future acceptability of today's internal combustion engine is in order.

Certainly, it must be agreed that, while reciprocating engines have not yet reached their design limit, they are fast approaching it. In addition, inherent problems such as bulk, weight, an extensive number of moving components, costs, maintenance, and emissions contribute to the somewhat bleak outlook for the reciprocating combustion engine as a future source of power.

This does not mean to say, however, that the Otto cycle cannot still be utilized in its thermodynamic sense; but instead might indicate that a new, simpler, mechanical mechanism for power might be designed around this cycle. These were the feelings of Felix Wankel, (see reference 2), when he developed an internal rotary combustion engine employing the Otto cycle in 1954. The advantages of such an engine become immediately apparent. Using a rotating piston and cam arrangement, approximately 50% of all reciprocating parts were eliminated, resulting in reduced friction and a lighter engine. Moreover, due to symmetry and simplicity of design, limitations such as maintenance, vibration, wear, and noise were considerably reduced.

In comparison, there would seem to be very distinct advantages of rotary over reciprocating engines as far as future power systems are concerned. Since the Wankel Rotary, there have been many new rotary combustion engine designs, (see references 3-10), each exhibiting certain advantages over the others, but few enjoying any measure of success, due to the complexity of inherent problems in the basic design.

Hopefully, as a step in the right direction, a new rotary internal four-cycle combustion engine has been designed. With much forethought in mind, and a little subject knowledge, it is felt that an engine of this type might succeed where others have failed.

With the planning of the Statkus four-cycle combustion engine, an attempt has been made to initiate the development of a small, lightweight, economical, long-life engine with reasonable efficiency.

In addition, consideration has been given to easy-access maintenance, interchangeable and multi-operational parts, and light machining operations with special emphasis on cast versus machined components.

The future for an engine such as this is certainly well based. The applications would indeed be numerous as a small, economical, lightweight and efficient prime mover. Its applications are compounded considerably with the thought that the engine itself need not be small, and with the idea that any number of these units may be "stacked" together to produce increased torque and horsepower. In addition, an engine having few moving parts lends new light to industrial terms such as quality control, effectiveness and working life.

CHAPTER II

ENGINE DESCRIPTION

A. General Description

The Statkus Engine is an eight-cylinder, swinging piston, rotary combustion engine. The unit consists of thirteen primary pieces, ten of which are moving members while the remaining three are fixed components. Of the moving members, eight are pistons, attached to a star-shaped wheel which also contains the output shaft. The fixed components make up the housing of the engine and consist of an annular hoop and two circular cover plates. The following articles describe these prime components.

B. Arc Piston

The piston is defined in this manner as its face is a fixed arc whose radius is the distance from the center of the engine shaft to the inside of its outer circular rim, (see figure 1). There are eight such pistons within the engine all of which are alike. During operation, the motion of these pistons provides for engine breathing, while the piston itself assumes the load-bearing responsibilities during the power strokes. One end of the piston is attached to the rotor using a hinge pin leaving the other end free to swing toward and away from the center of the rotor through a cam-follower and cam-follower guide mechanism explained below in section E.

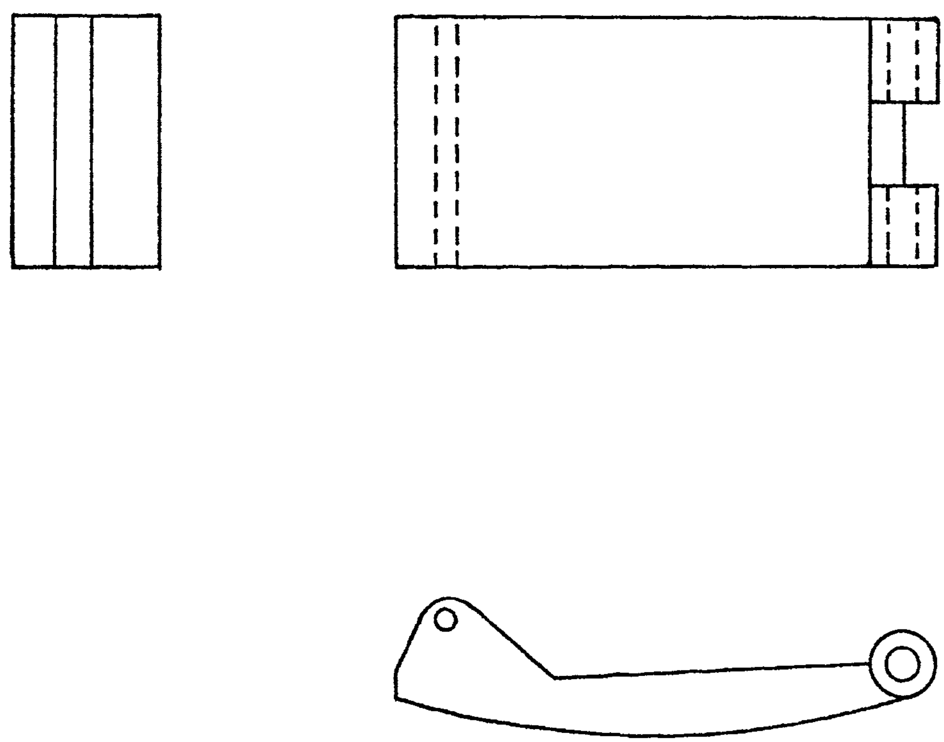


Figure 1 Piston Design

C. Rotor

The rotor is the largest moving component of the engine, (see figure 2). The mass of this piece, together with its axial symmetry about the drive shaft enables it to serve as the engine flywheel, in addition to its primary load transmitting duty.

The primary function of this component is to impart rotary motion to the shaft by transferring the forces on the pistons during the power strokes through its mass to the keyed output shaft at its center.

D. Hoop

The hoop has an annular geometry, the inside diameter being slightly larger than the diameter of the rotor, (see figure 3). The breathing ports and spark plug holes are located around the circumference of the hoop. The unit has five rows of cooling fins around its periphery except where a port or plug position occurs. Position of these breathing ports and spark plug holes is critical with regard to the engine timing, and therefore these ports and holes are generally spaced every 45 degrees about the circumference of the hoop.

E. Cover Plates

As implied, the cover plates, (see figures 4 and 5), represent the left and right lids of the engine and in fact, seal off all cylinders from each other and the

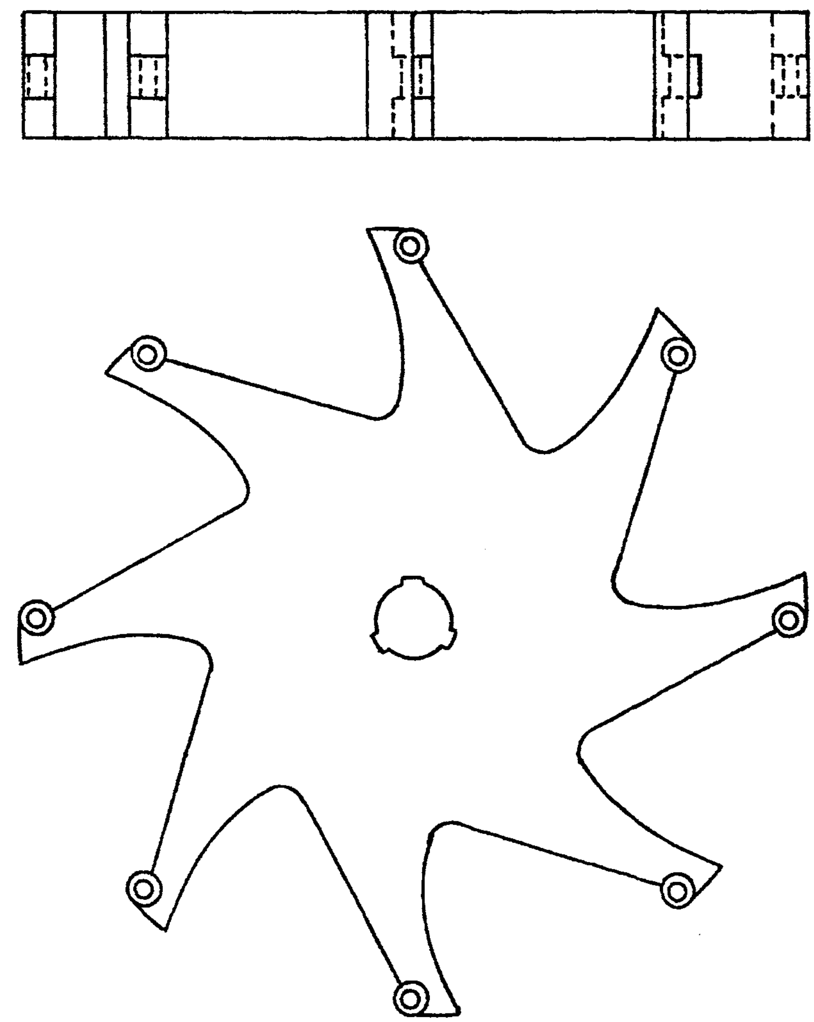


Figure 2 Rotor Design

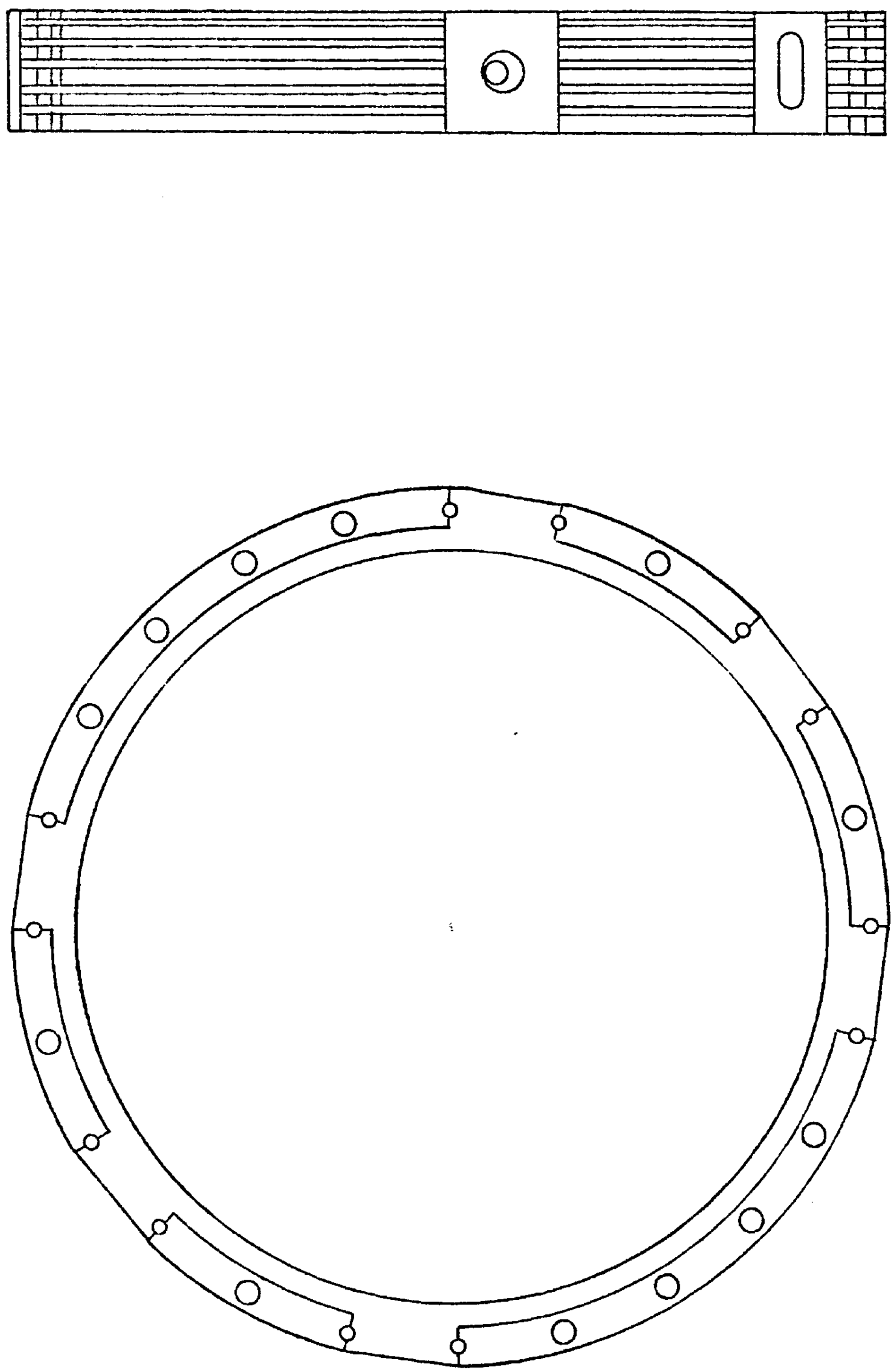


Figure 3 Hoop Design

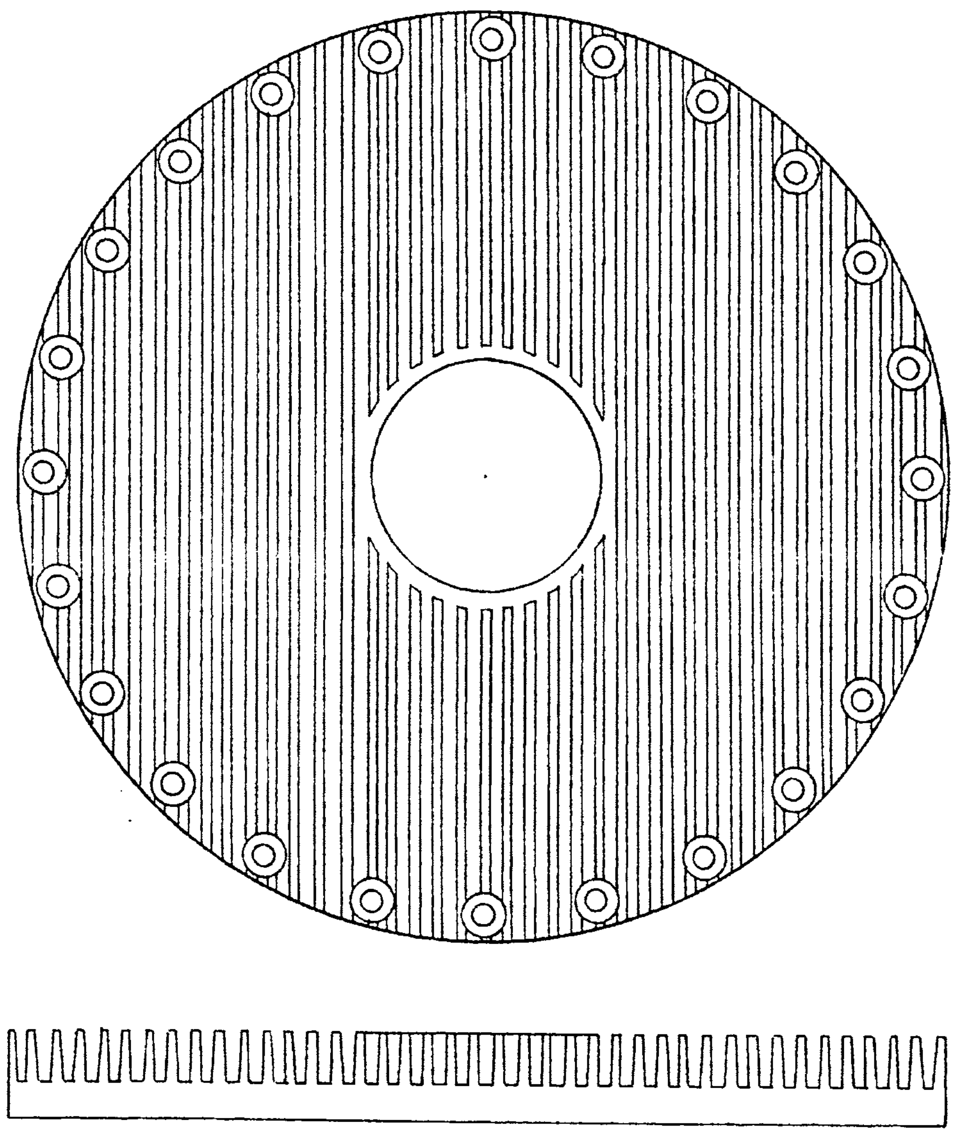


Figure 4 Cover Plate Design

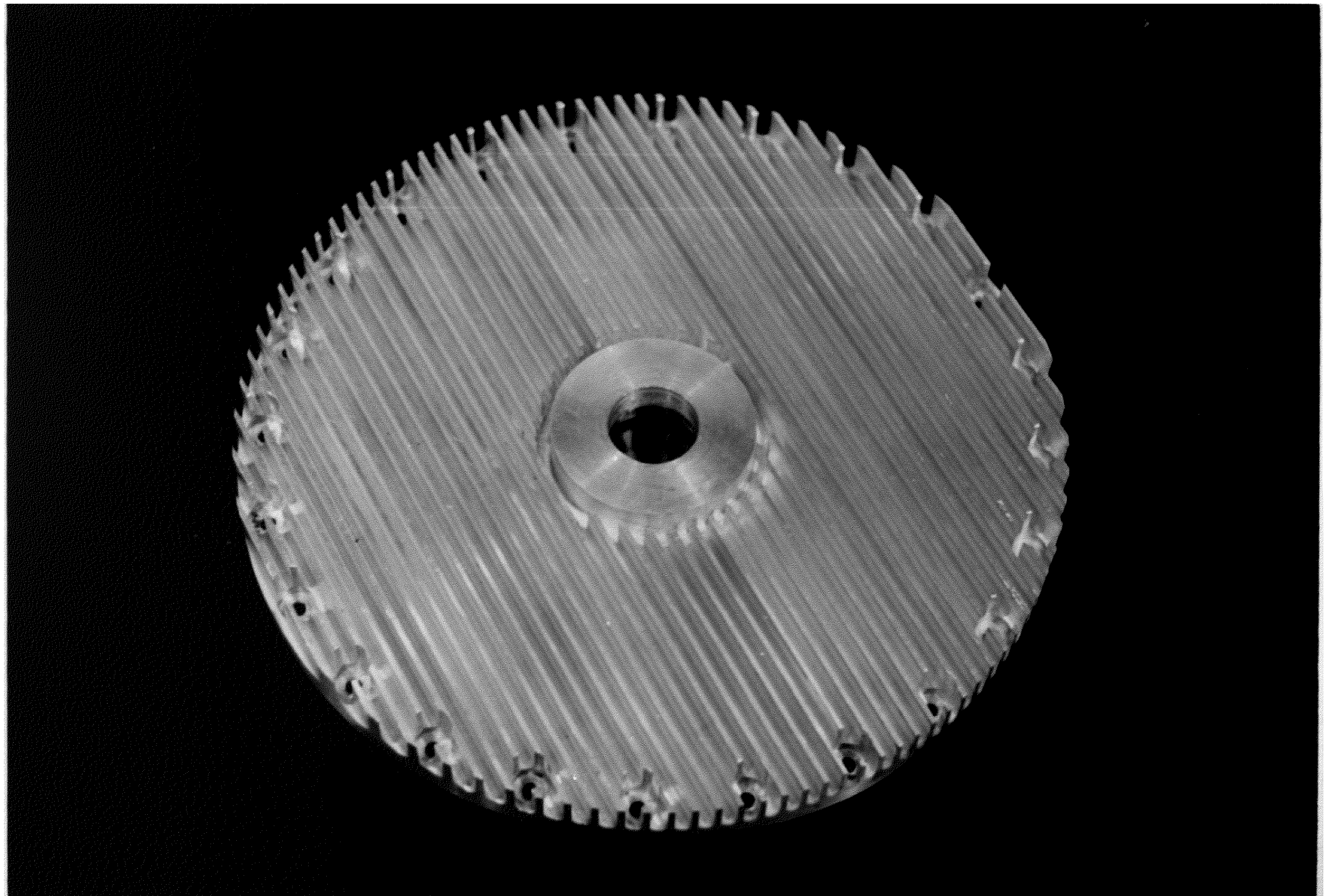


Figure 5 Prototype Cover Plate

rest of the engine. The outer faces of both plates are finned while the inner surfaces carry a symmetric, somewhat sinusoidal machined cam-follower groove. Since the groove in one plate is a mirror image of the other, the plates are not interchangeable. The groove is the tracking mechanism for the cam-follower of each arc piston and the geometry of this groove provides for the operation, timing, and breathing of each cycle as it is occurring. In addition, the groove is the reactionary member which causes the forces on the piston to be transmitted through the hinge pin to the rotor generating the rotational moment of the engine.

CHAPTER III

ENGINE OPERATION

A. General Description

One of the primary considerations of the design of this engine was simplicity. Basically, there are only ten moving parts; the rotor, the output shaft, and eight pistons. The wheel is keyed directly to the output shaft, which in turn is aligned at either end by tapered roller thrust bearings seated in their respective cover plates. Each piston is hinged at one of its ends to the rotor, and since there are eight pistons, a hinge appears every 45 degrees around the circumference of the rotor. Since the pistons are connected directly to the rotor, connecting rods are not required. Breakdown views of the engine are shown in figures 6-8.

B. Cycle

This engine is an application of a modified Otto cycle and thus completes four "strokes" during one cycle. Although one end of a piston does move toward and away from the center of the engine during rotation, the motion of the piston could not properly be called reciprocating, and thus the word "stroke" can be more clearly understood as a "period" of motion of the piston. This idea gives rise to a four-period cycle of operation instead of the typical four-stroke cycle. The periods do, however,

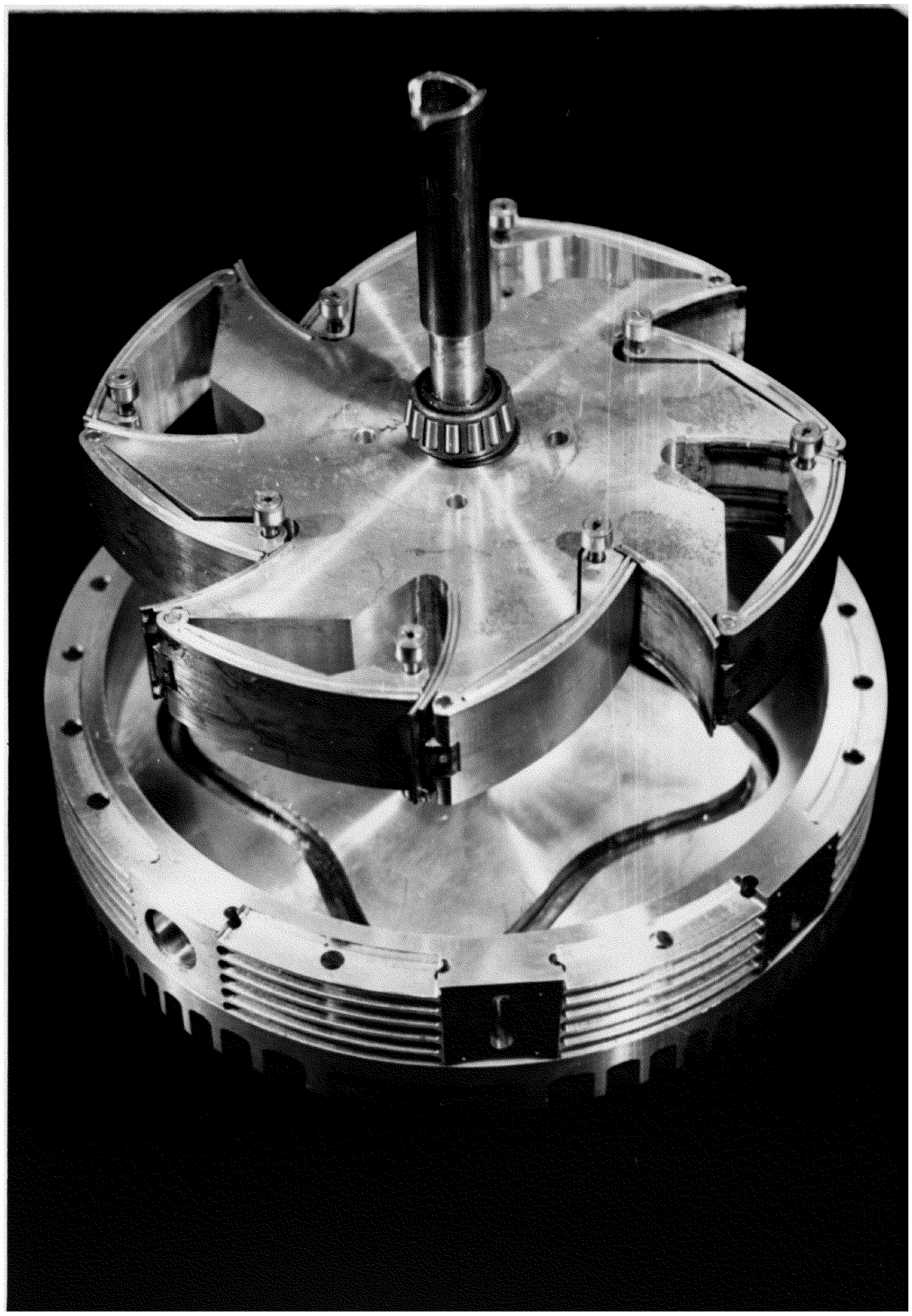


Figure 6 Prototype Assembly

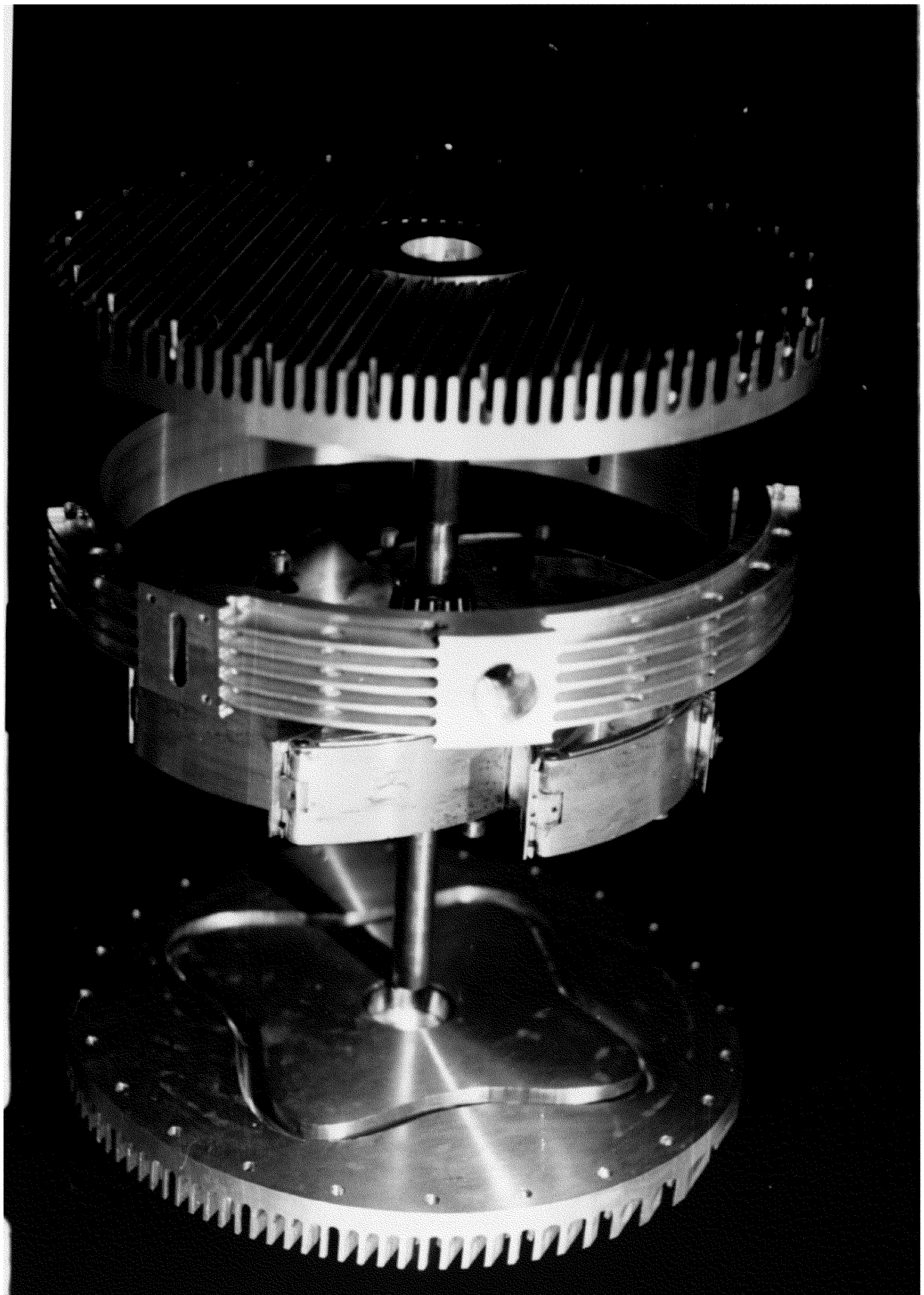


Figure 8 Prototype Exploded View

correspond to the four strokes of a reciprocating engine and are intake, compression, power or expansion, and exhaust. Figures 9-12 illustrate these periods for one half of a revolution of the rotor. Since there are eight pistons around the circumference of the engine, and only four periods per cycle of operation, it is clear that there must be two cycles completed per rotation of the rotor or output shaft. This sequence requires the start of a new period every 45 degrees of rotation; hence, each piston begins and completes eight periods or two cycles per revolution of the engine. Since the pistons, rotor, and output shaft are interconnected and because each of the eight pistons are in two power periods per revolution, there are sixteen power periods per revolution of the output shaft.

C. Geometry

The engine is circular around its periphery with round disc-shaped cover plates as the upper and lower lids. Machined in the inside of each cover plate is the cam follower track which, due to its geometry and positioning with respect to the peripheral ports, provides for the valving action of the pistons, the timing of the engine, and determines the compression ratio of the unit. The geometry of the groove corresponds somewhat to an axisymmetric sinusoidal curve (see figure 13). Actually the groove is warped at certain positions from the curve

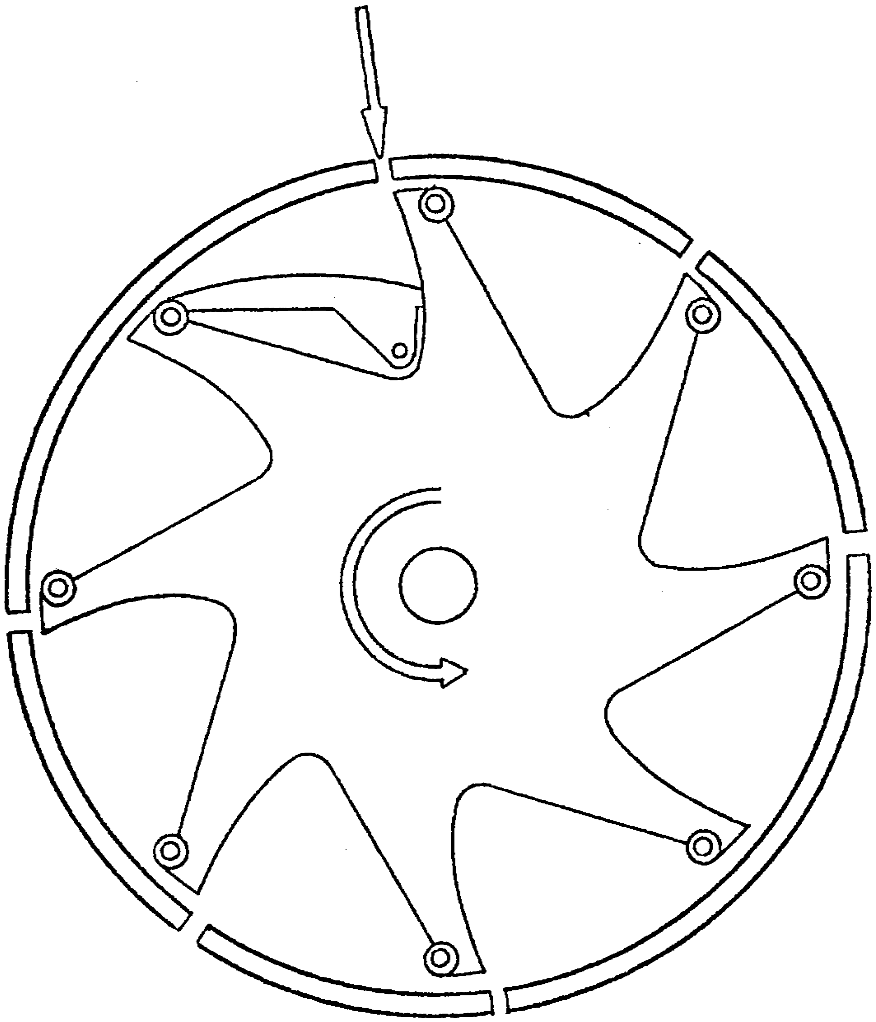


Figure 9 Intake

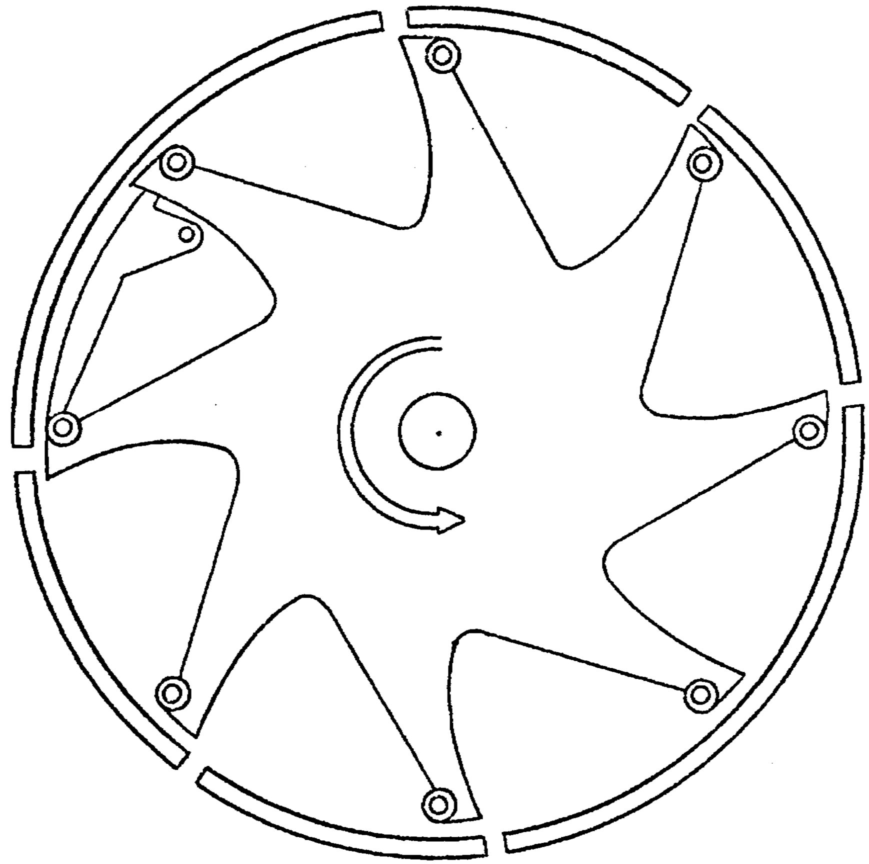


Figure 10 Compression

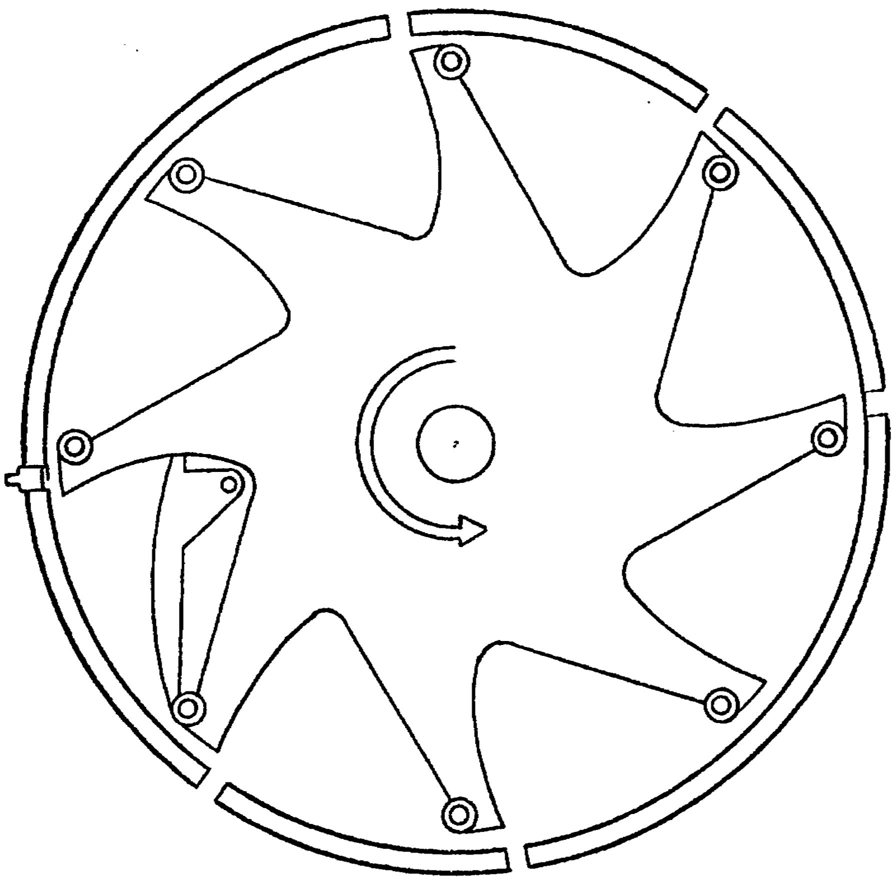


Figure 11 Expansion

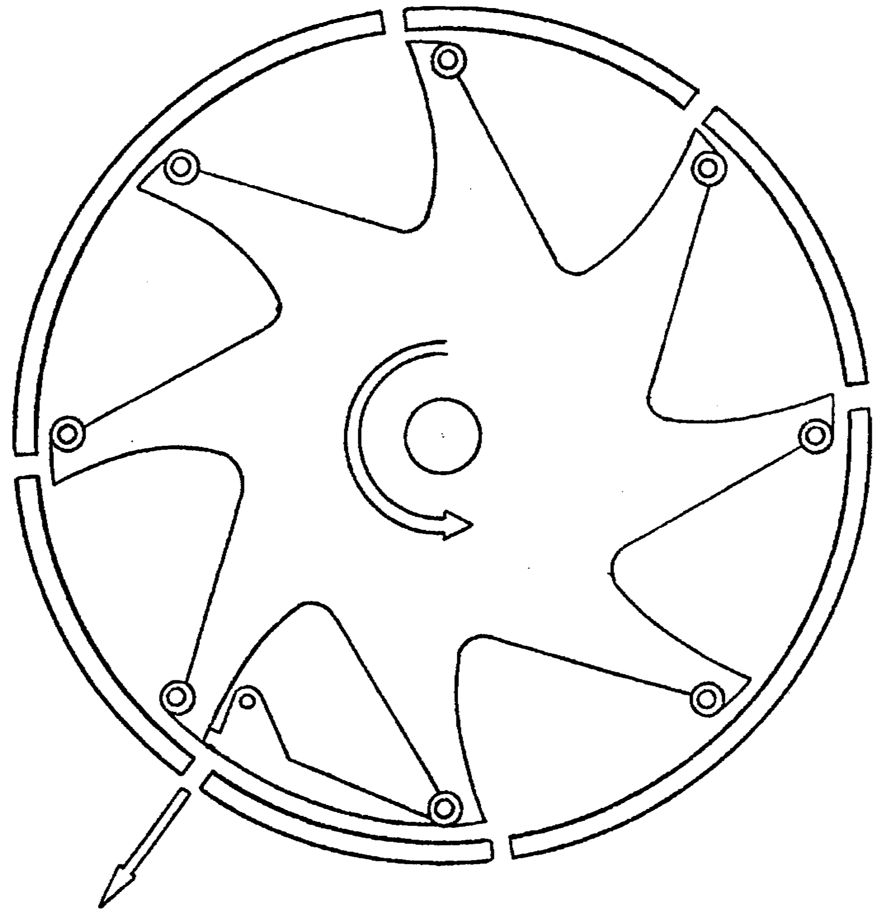


Figure 12 Exhaust

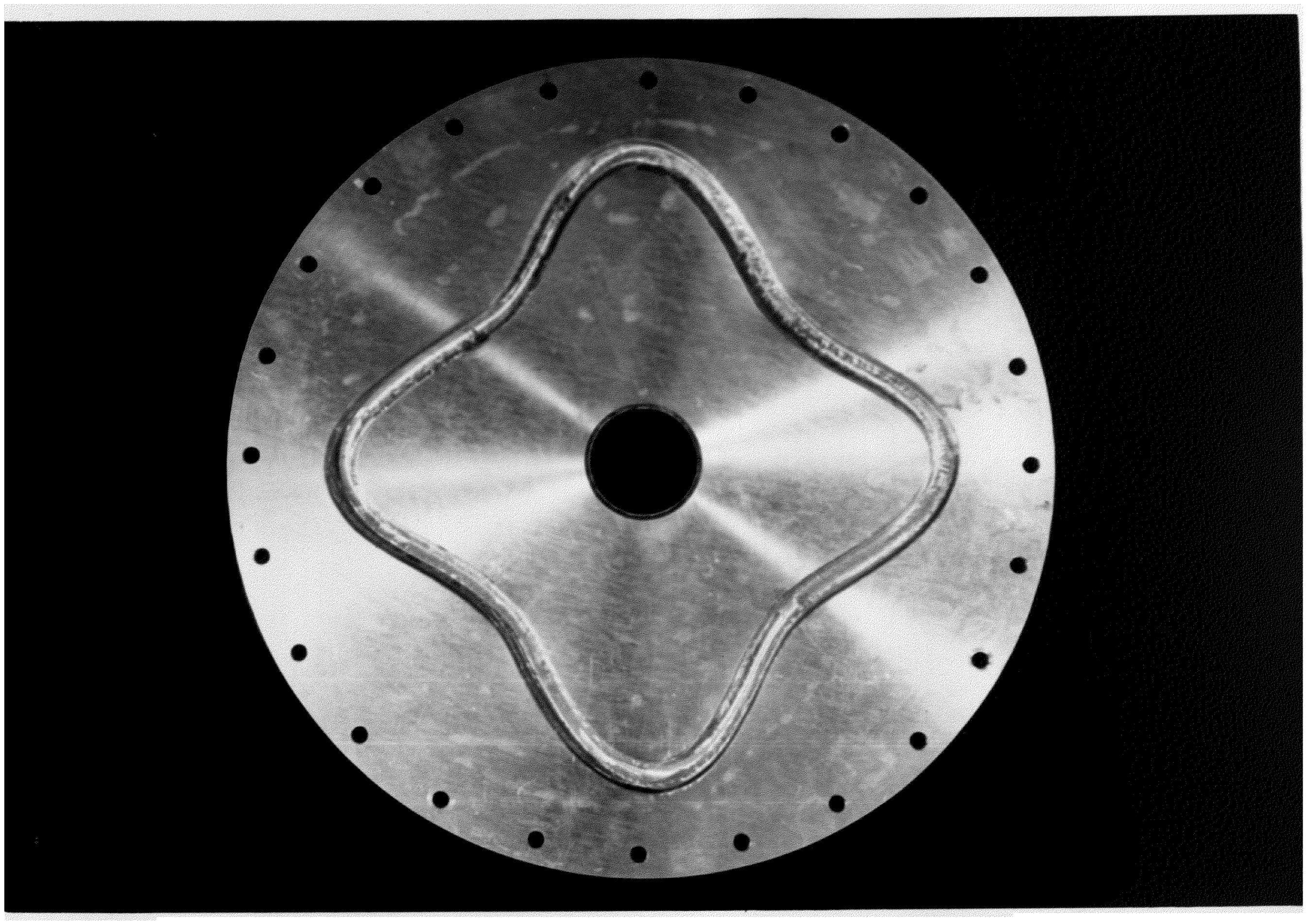


Figure 13 Cam Follower Groove Geometry

described in order to accelerate or decelerate the pistons during certain periods. For instance, at the beginning of the compression period, the piston is in its full intake position which corresponds to the perigee of the groove. In the ideal case, the following compression period should be accomplished instantaneously in order that losses of the fuel-air mixture due to blowby past the rings or seals be eliminated. To approach this case in the real engine, the perigee of the groove at this point in the cycle was designed to remain in dwell for a few degrees after which there is a sharp incline toward the apogee where the piston attains full compression. In doing this, the period of compression remains 45 degrees, while the actual change in pressure in the cylinder during compression takes place in a considerably shorter period, thereby minimizing the blowby losses. In addition, the geometry of the groove was such as to eliminate the clearance volume at the end of the exhaust stroke. The significance of this arrangement can be seen in the Ideal Otto Engine process analysis in the appendix.

As another example, consider the exhaust period. Since the pistons of this engine move by the breathing ports instead of to and from them as in a reciprocating engine, exhausting can be accomplished more efficiently by pressurizing the exhaust volume as early as possible in the period, promoting a high velocity, high pressure discharge during the remainder of the period. In order to do this, the perigee dwell of the groove at the beginning of the exhaust period was shortened, followed by a sharp incline to the apogee position of the groove early in the period, afterwhich, near dwell remains to the end of the period.

At this point, consideration was given to the distances from the center of the plate to the various apogee and perigee positions of the groove. Since all identical periods take place at only two positions around the periphery of the engine, there is no reason to consider any period at a particular peripheral location other than the period which will take place there. As an example, there is no need to maintain a clearance volume at full exhaust, since the clearance volume normally represents the compressed volume and is solely a cause for a different period which would never occur at an exhaust peripheral position. In other words, the groove apogee position corresponding to full exhaust may be lengthened to eliminate those residue exhaust gases which are inherent in the typical reciprocating engine. Furthermore, by varying the perigee and apogee distances on either side of the compression period, the compression ratio of the engine can be changed.

D. Motion

As mentioned earlier, the motion of this engine is largely rotational from an internal point of view, even though one end of the piston would actually be seen to reciprocate by a person with a vantage point on the rotating rotor.

Due to the symmetry of the engine, rotation can theoretically be reversed by "flipping" the hoop 180 degrees, thereby reversing the cycle, and then by rotating the two cover plates containing the grooves through an angle of 15 degrees, hence, setting the engine timing.

E. Cooling

The prototype engine was aircooled. Water or oil-bath methods of cooling, although more efficient, would have increased considerably the machining costs of the first engine and were therefore unacceptable. The methods of heat transfer were conduction and free and forced convection.

During the cycle, the temperature of the gases in the cylinder is anticipated to vary from 100 degrees F or less to as high as 5000 degrees F assuming combustion, but no detonation occurs. The mean gas temperature in the cylinder may be as high as 1200 degrees F but drops considerably through the boundary layer of stagnant gases adjacent to the cylinder wall, (see figure 14).

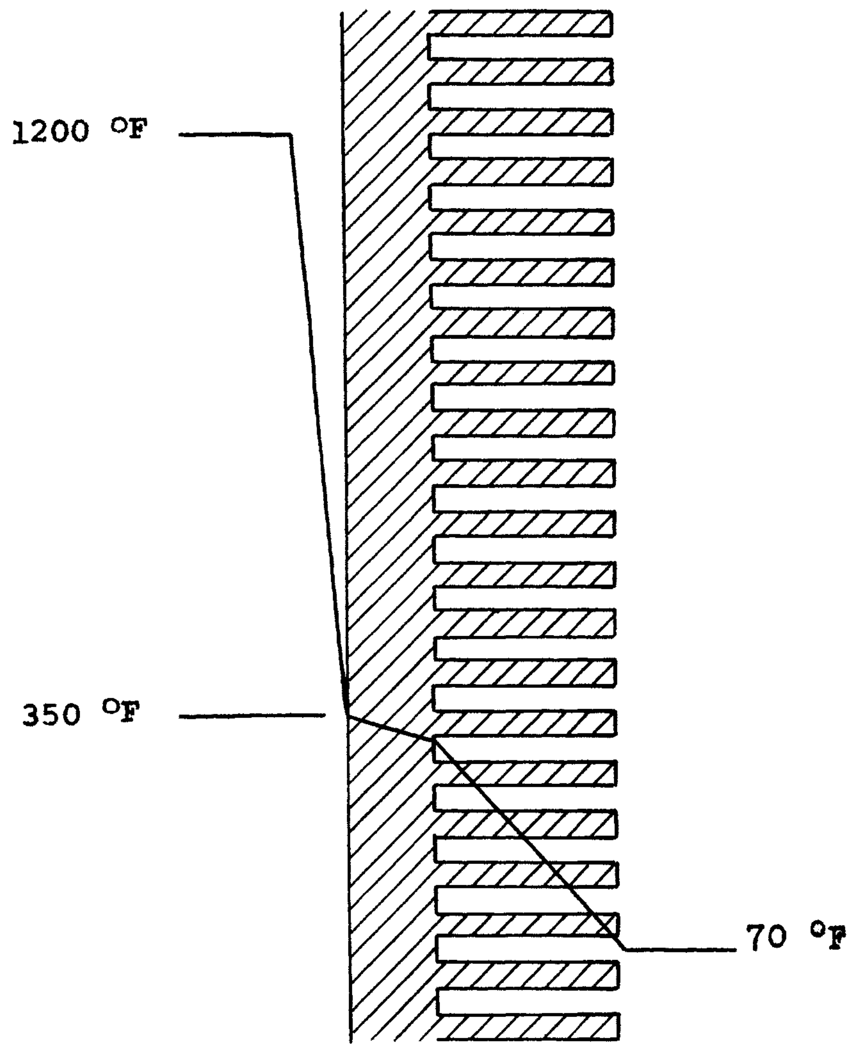


Figure 14 Boundary Layer Temperature Profile

Heat is then conducted through the wall to the external fins where free convection to ambient takes place. On the prototype, fins were machined circumferentially around the hoop and across both cover plates.

Within the engine, heat transfer was designed, for the most part, to be through forced convection and conduction. During operation, the pistons not only open and close on the engine hoop to provide for cycle periods and breathing, but also open and close on the piston cavity in the rotor. When the piston is opening with respect to the hoop, as during the intake stroke, it is simultaneously closing on the rotor cavity behind it. This cavity closing causes the air trapped in that volume to be pressurized and eventually escape at high velocity to adjoining cylinder cavities having a lowered pressure due to a reversal period, and also to the clearance volume between the rotor and each cover plate. During rotation, due to the nature of the cycle, four of the eight pistons are in a period causing high cavity pressure while the remaining four are in a period causing low cavity pressure. The result is a constant flow of high velocity, turbulent air around the back of the pistons, and over the wheel, enabling heat to be transferred from these components to the finned cover plates where convection takes place.

F. Ignition

During combustion, the piston moves by the plug position instead of away from it as in a reciprocating engine. This sequence provides the ability to utilize a constant spark, in order to generate a continuous flame front and, thereby, more complete combustion. In addition, keeping in mind that each plug must fire eight times per revolution of the shaft, the firing of a spark plug would fast approach a state of continuous firing even at low engine speeds. For these reasons, in addition to simplicity of the mechanism, an ignition system with two continuously firing plugs was devised (see figure 15). Basically, the utilization of a spark plug was accomplished by replacing the coil, distributor, and points of the standard engine electrical system with a DC vibrator. In reality, the vibrator is all three of these components. The voltage from the primary of the vibrator is continuously pulsed to the secondary through the vibrating mechanism causing a near constant high voltage discharge at the plug gap. A separate vibrator is required for each plug.

At this point it is worth mentioning that a glow plug arrangement might prove to be a simpler and more efficient constant ignition mechanism than that described above.

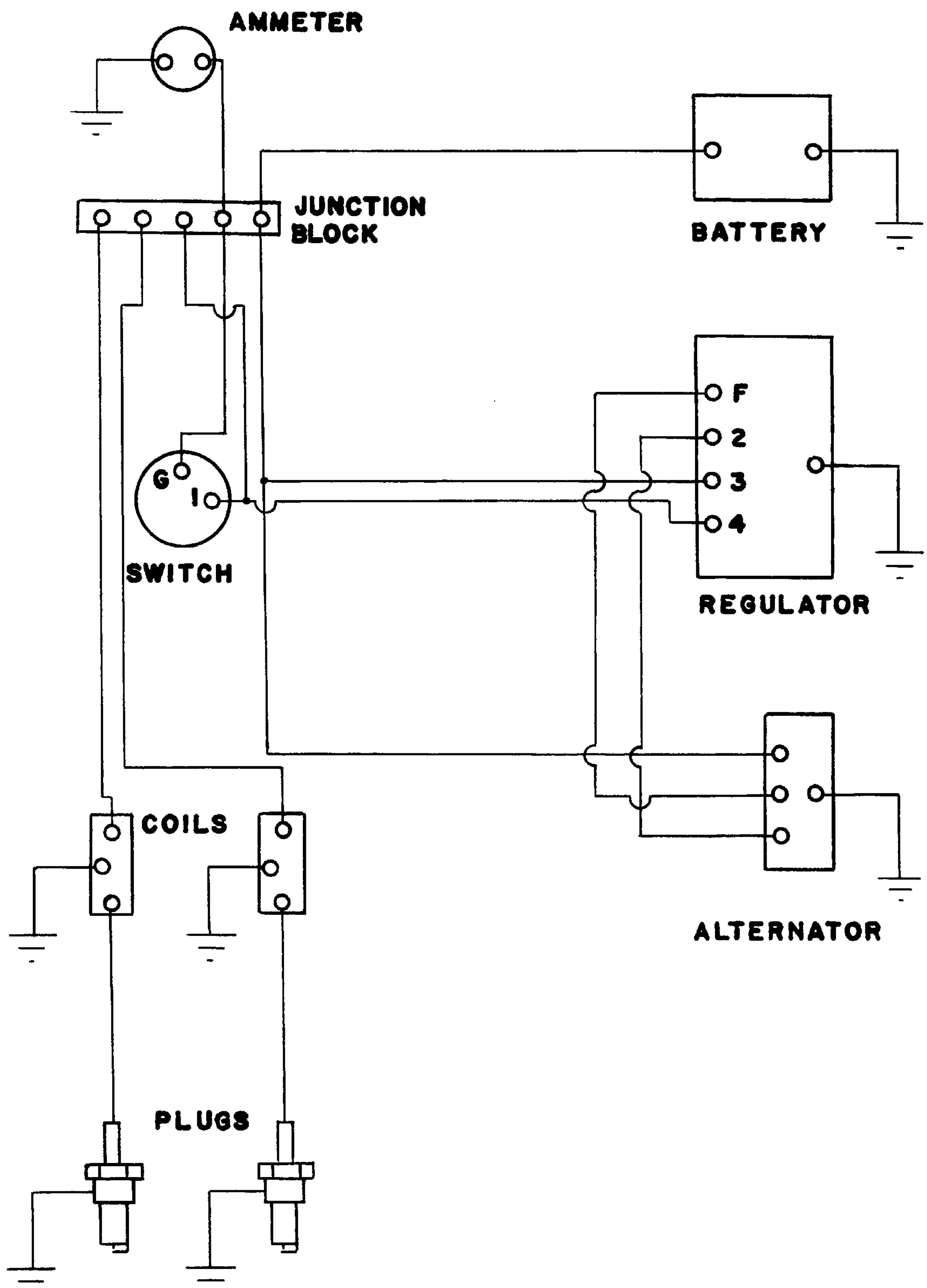


Figure 15 Ignition Schematic

G. Sealing

Chamber sealing represented one of the most difficult design problems of the engine. There are six separate seals per chamber and each fits into its respective slot on the rotor or piston. Three of the seals per cylinder are located on the piston (see figure 16), while the remainder of the seals are fitted to the rotor (see figure 17), and serve to separate the cylinders. Wave and coil springs are used to initiate contact of the seals with the facing surfaces. By design, all seals should be machined for a slip-fit in the seal grooves, tapered on their surfacing edge to cut down friction, and long enough to achieve an acceptable seal without binding. In addition, gas slots should be made at the base of the seal (see figure 18), so that, during combustion, high pressure gas would be admitted to the base of the seal in the clearance space to force the seal outward against its facing surface.

In the prototype, due to scarcity of funds, the seals were first machined in straight lengths having the proper design width and depth to permit a slip fit in the seal grooves. At this point, the seals were warped in a heat treating oven to approximate groove contour. The seals were then cut to length and the facing edge was tapered. From a monetary point of view, this method proved considerably cheaper over a totally machined seal,

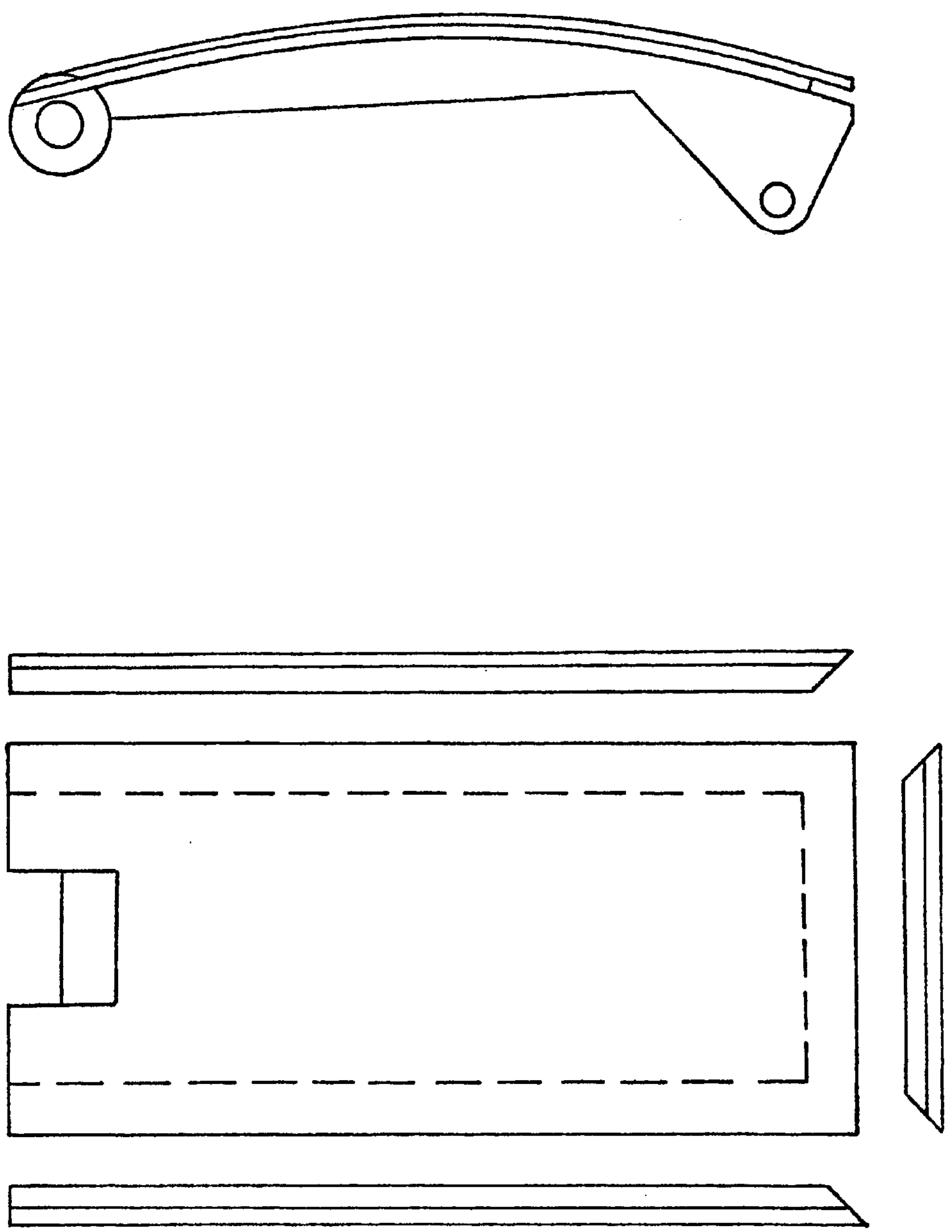


Figure 16 Piston Seal Positions

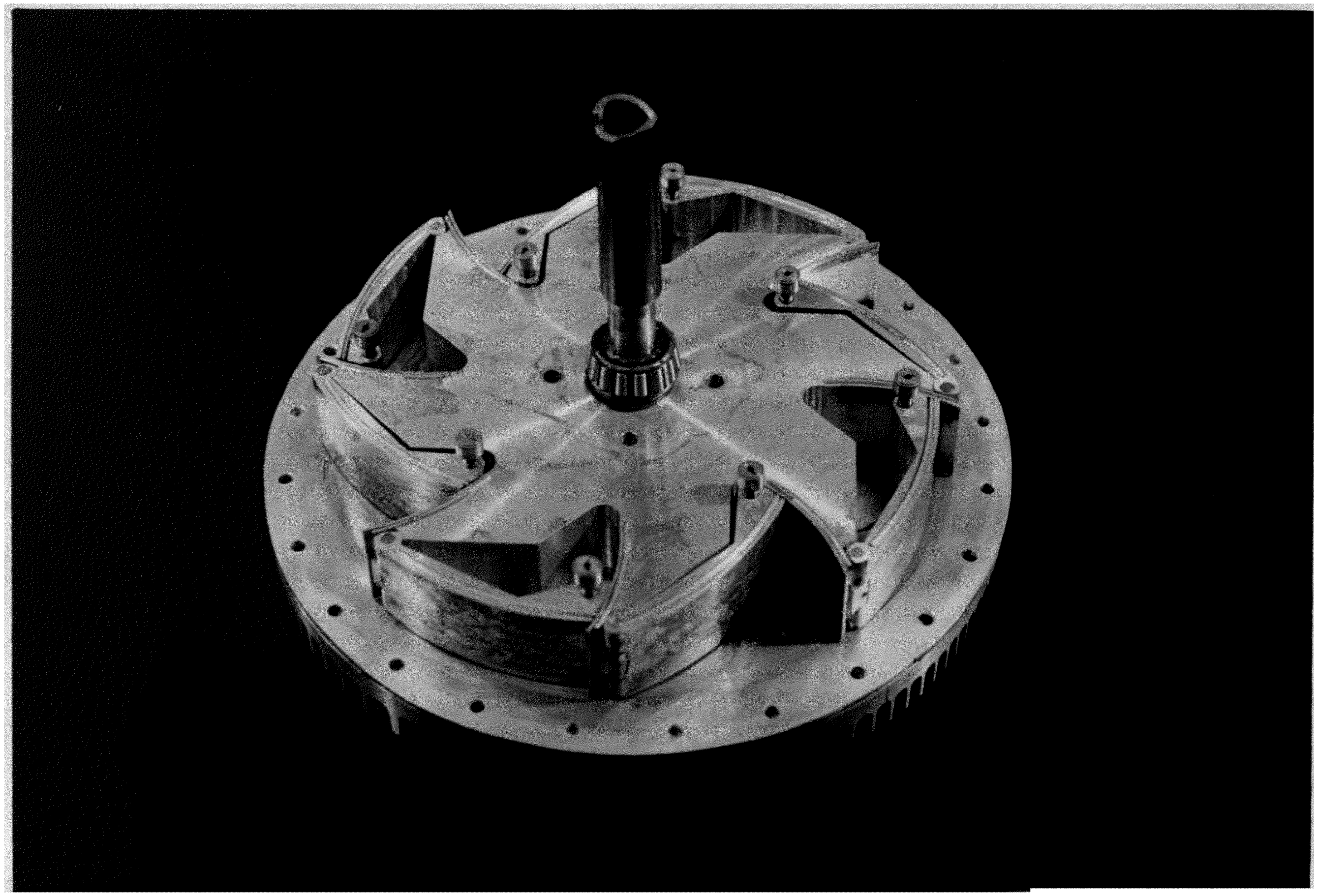


Figure 17 Wheel Cylinder Seal Positions

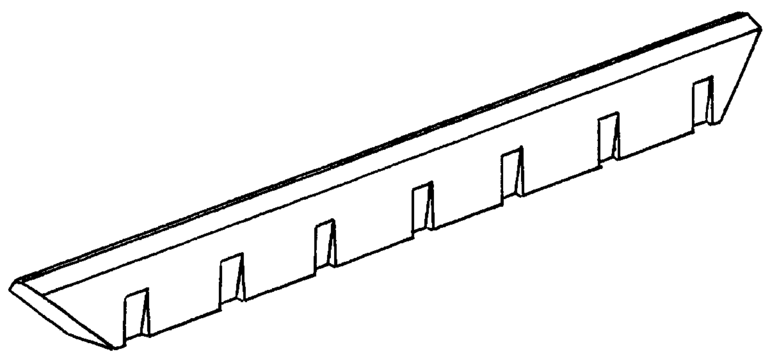


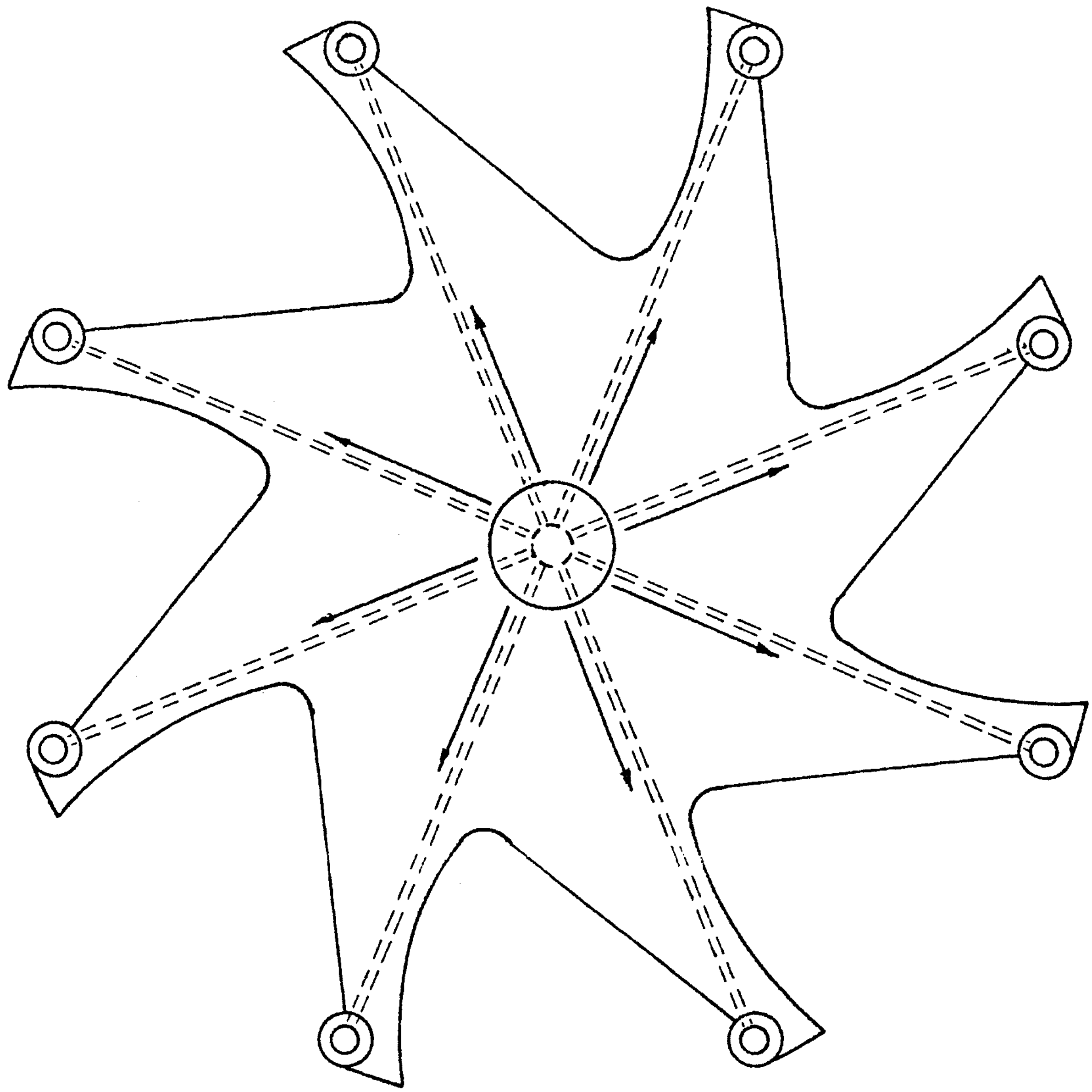
Figure 18 Typical Seal With Gas Pressure Slots

but in the end could not outweight the disadvantages of poor seal performance.

H. Lubrication

In design, lubrication of the engine was directed essentially at three major areas to sustain operation. Those areas were the cam roll bearings attached to the pistons, the hinge pins, and the two shaft thrust bearings. The tapered roller thrust bearings were designed to be greased by fittings located in the cover plates adjacent to the bearings. Each piston hinge pin and cam follower is centrifugally lubricated from an oil resevoir at the center of the rotor. Oil is pumped to the resevoir through the output shaft of the engine. During operation, lubrication is first fed to the piston hinge pins, (see figure 19), from the resevoir through the wheel. As the piston moves back and forth over the hinge pin, a metered amount of oil is bled to the cam follower pin through a channel in the piston. From here the oil flows through the pin to the base of the cam follower bearings at either end of the follower pin.

Due to machining costs of such a lubrication system, none of these methods was able to be implemented into the prototype, cutting possible length of run time considerably.



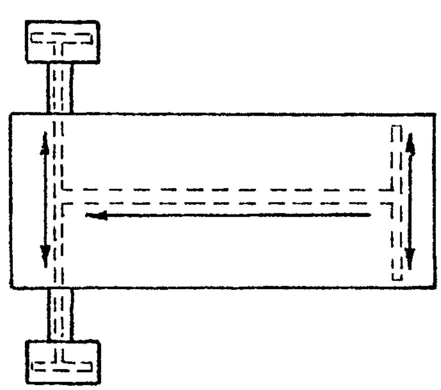


Figure 19 Centrifugal Lubrication System

I. Induction System and Fuels

The two-barrel carburator was chosen by comparing its maximum volumetric flow rate with the engine speed that would have to be attained in order that a 100% efficient induction system could injest that flow rate. Since there were only two intake ports on the engine, an induction manifold was designed in such a way, that the fuel-air mixture from each barrel was carried separately within identical lengths of pipe to the hoop intake ports. In this way, there was reasonable assurance that the mixture and flow rates to each port were similar and would then aid in keeping combustion pressures and temperatures at either side of the engine nearly equal.

Assuming that an acceptable cylinder pressure could be achieved prior to ignition, the engine was designed to run on locally available fuels, with initial testing to be done with a mixture of 80-octane gasoline and oil in an attempt to speed the cylinder glazing process. Hopefully, if an engine failure or breakage were to occur, scoring or scratching of the cylinder walls could be somewhat minimized.

J. Exhaust Systems and Emissions

Since there were two exhaust ports, a dual manifold system was designed to be bolted directly to the hoop, and contoured in such a way as to direct the hot exhaust away from the induction system. It was believed that an engine of this type could reduce the amount of unburned gases leaving the engine due to its method of operation. As an example, consider the motion of a piston during an expansion period. At high speed, the piston of a reciprocating engine moves away from the flame front at a speed such that the flame cannot catch up to the piston prior to the beginning of the exhaust stroke and subsequent flame quenching. These unburned gases are then blown out during exhausting of that cylinder. On the other hand, the pistons of the rotary engine move by the plug during expansion. This concept, together with the system's adaptability to a continuous spark ignition system, permits a constantly energized flame front to traverse the expanding cylinder as it moves by the plug. In this way, the flame front has only to move a short distance to the end of the cylinder and is, therefore, able to contribute to combustion of a larger percent of the mixture.

CHAPTER IV

ANALYTIC CALCULATIONS

A. Ideal Combustion Analysis

The cycle analysis used below is based upon simplifying assumptions and is an attempt at modeling the actual process. The actual combustion engine process deals with fuel, air or oxygen, or other energy-liberation reactants, and the fuel products of combustion or reaction. A chemical reaction occurs in the combustion chamber of the piston which results in products of combustion, principally CO_2, H_2O, and N_2. Variable specific heats and chemical non-equilibrium prevent the attainment of the high temperatures and pressures indicated by the ideal cycle analysis. The products are exhausted from the engine at a different temperature and pressure than that of the air and fuel entering the engine. Since the products are not transformed back to air and fuel in the engine or auxiliary apparatus at the supply conditions, the combustion engine process cannot be termed a cycle. Hence, the mechanism may go through a cycle, but the mediums do not.

In the ideal theoretical engine process analysis, it is assumed that no heat transfer to or from the media and no fluid-friction losses occur. In addition, engine breathing operations are assumed to occur with no flow restrictions, thereby eliminating fluid-friction effects.

Analyses assuming no heat transfer and fluid friction result in the maximum efficiencies attainable and serve as a basis for comparison with actual engine efficiencies, which can approach but never reach the ideal standard established by such an analysis.

In the Ideal Otto Engine process, the usual procedure is to assume both the induction temperature and the clearance gas weight fraction. The various parts of the engine process are then analyzed, using the foregoing assumptions. However, in constructing an ideal thermodynamic cycle for the Statkus engine, the clearance gas effects were neglected since there was no clearance volume in the cycle. The induction temperature was assumed to be 500 ^{o}R. The analysis was then separated into three processes: compression, combustion, and expansion. The exhaust and induction processes were neglected, since the work done on the engine piston during the induction process is equal to the work done on the exhaust gases by the piston during the exhaust stroke. The assumption made here is that the system is normally aspirated, or that the induction pressure equals the exhaust pressure.

Since the analysis has been reduced to graphical results employing tables for stoichiometric mixtures of iso-octane (gasoline) and air in reference 11, these tables were used in this analysis and will be referred to in the discussion.

1. <u>The Compression Process</u>

The fuel-air mixture inducted is compressed reversibly and adiabatically from 1 to 2, (see figure 20). Hence the entropy of the total mixture remains constant during the process, and for "n" moles of perfect gases,

$$\Delta S = 0 = n \int_{T_1}^{T_2} \frac{Cv\, dT}{T} - nR \ln V_1/V_2 \, , \qquad (1)$$

where, V_1/V_2 = compression ratio = r = 8.5:1 .
Letting Sv equal the integral quantity, equation (1) becomes: $n(Sv)_{T_2} - n(Sv)_{T_1} = nR \ln r$.

Chart B of reference 11 contained the plots of Sv for three mixtures above 400 °R. For any mixture ratio and value of T_1, the value of $n(Sv)T_1$ could be obtained, assuming that the curve on the chart is for the volume V_1. Adding this value to nR ln r resulted in the value of $n(Sv)T_2$, which indicated the value for T_2 on the Sv line for the same mixture ratio. This was also the temperature at the end of the compression at volume V_2. The internal energies U_1 and U_2 corresponding to T_1 and T_2 were found on the chart, and the difference between them was the work of compression, (see appendix A).

2. <u>The Combustion Process</u>

The combustion process was computed by two methods. First, an equilibrium combustion analysis assuming an adiabatic, constant specific heats, constant volume, process with no fluid friction occurring was calculated to arrive at point 3 in figure 20. At the beginning of combustion, the mixture contained both chemical energy "C" and internal energy "U" of the fuel and air. At the end of the process, the products of combustion (H_2O, CO, CO_2, O, N, NO, etc) all contain chemical and internal energy. Assuming no heat transfer, the energy equation for the process is:

$$n(U_2 + C_2) = n(U_3 + C_3). \qquad (3)$$

Using chart D from reference 11, which equates the thermodynamic properties of combustion for the mixture, to include the $(U_3 + C_3)$ term in equation 3, the properties of the mixture at the end of the constant volume combustion were determined, (see appendix A).

The second method assumes a nonadiabatic compression followed by a two step combustion process--a constant volume process followed by a constant pressure process.

The fraction of mixture reacting at constant volume and constant pressure combustion were separated into several percentage cases. A reasonable heat transfer loss from the mixture was calculated in order to determine its impact on this portion of the combustion calculation. The insignificance of the result, (see appendix C), when compared to the internal and chemical energies of the process was obvious and was therefore omitted from the calculation. The mixture percentages chosen were as follows:

	Constant Volume Combustion	Constant Pressure Combustion
1)	70%	30%
2)	60%	40%
3)	50%	50%

The constant volume portion of combustion was analyzed by considering equation 3, and the fact that only a fraction of the total mixture at state 2 would react completely to state 3. During this reaction, the remaining portion of the mixture would be carried along unaffected. The properties at 3 were then evaluated, using the charts, (see figure 21).

The energy equation for the constant pressure process had the form:

$$(H_A)_{T_3} + x(H + C)_3 = (H + C)_{3'} - Q \qquad (4)$$

where, H_A = enthalpy of air at T_3 in Btu/lbm

x = percent of mixture reacting

Q = heat loss to the mechanism in Btu/lbm

Letting $Q = 0$ as stated earlier, computing H_A at T_3, and knowing "x" and the quantity $(H + C)_3$ from the charts, the value of $(H + C)_4$ was found. This quantity, together with the knowledge that $P_3 = P_3$, isolated state 4 on the chart, from which the remaining properties at 4 were taken.

3. The Expansion Process

The gases do work on the piston during the expansion process, with a resulting decrease in internal energy and temperature. The decrease in temperature permits the recombination of some of the dissociated products throughout the entire process. This process is assumed to be adiabatic and reversible.

The work done during the adiabatic expansion process is equivalent to the difference between the internal and chemical energies at the beginning and end of the process. Thus,

$$\text{Work} = n(U + C)_3 - n(U + C)_4 \qquad (5)$$

At the end of expansion, $V_4 = V_1$; and since the process is assumed adiabatic and reversible, the entropy at 3 and 4 is also equal. In the expansion analysis, the intersection of the constant volume line V_4 with the constant entropy line S_4 defined the remaining properties at 4, to include the energy tern $(U + C)_4$ in equation 5. At this point, the work done on the piston by the change in energy during the expansion stroke was computed from equation 5.

Data obtained from the ideal combustion analysis for the end points of each stroke appear in table 1. Temperature and pressure-volume profiles for the processes are shown in figures 18-20.

B. Net Piston Work Analysis

Finally, as mentioned earlier, the work done on the engine piston during the induction process is equal to the work done on the exhaust gases by the piston during the exhaust stroke when the induction and exhaust pressures are assumed equal. Thus, the work values for only the compression and expansion processes were required for evaluation of the net work on the piston during one cycle of operation; and the equation became:

$$\text{net piston work} = \text{expansion work} - \text{compression work}\,. \qquad (6)$$

The results of equation 6 for each of the four calculations are found in table 3.

C. Static Force Analysis

An analysis of the static forces on the piston at the combustion stroke was made after it was determined

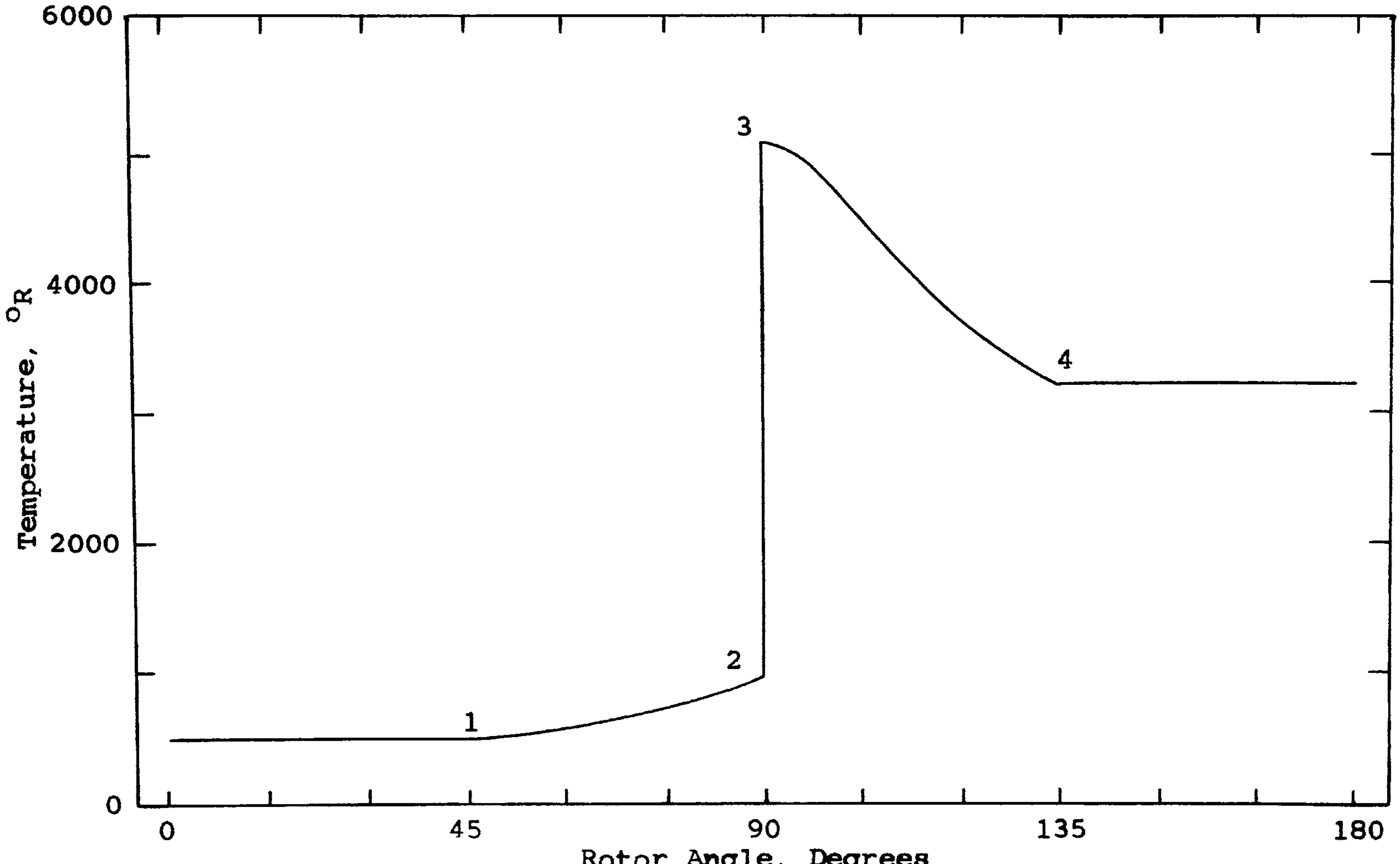


Figure 20 Gas Temperature Profile (Constant Volume Combustion)

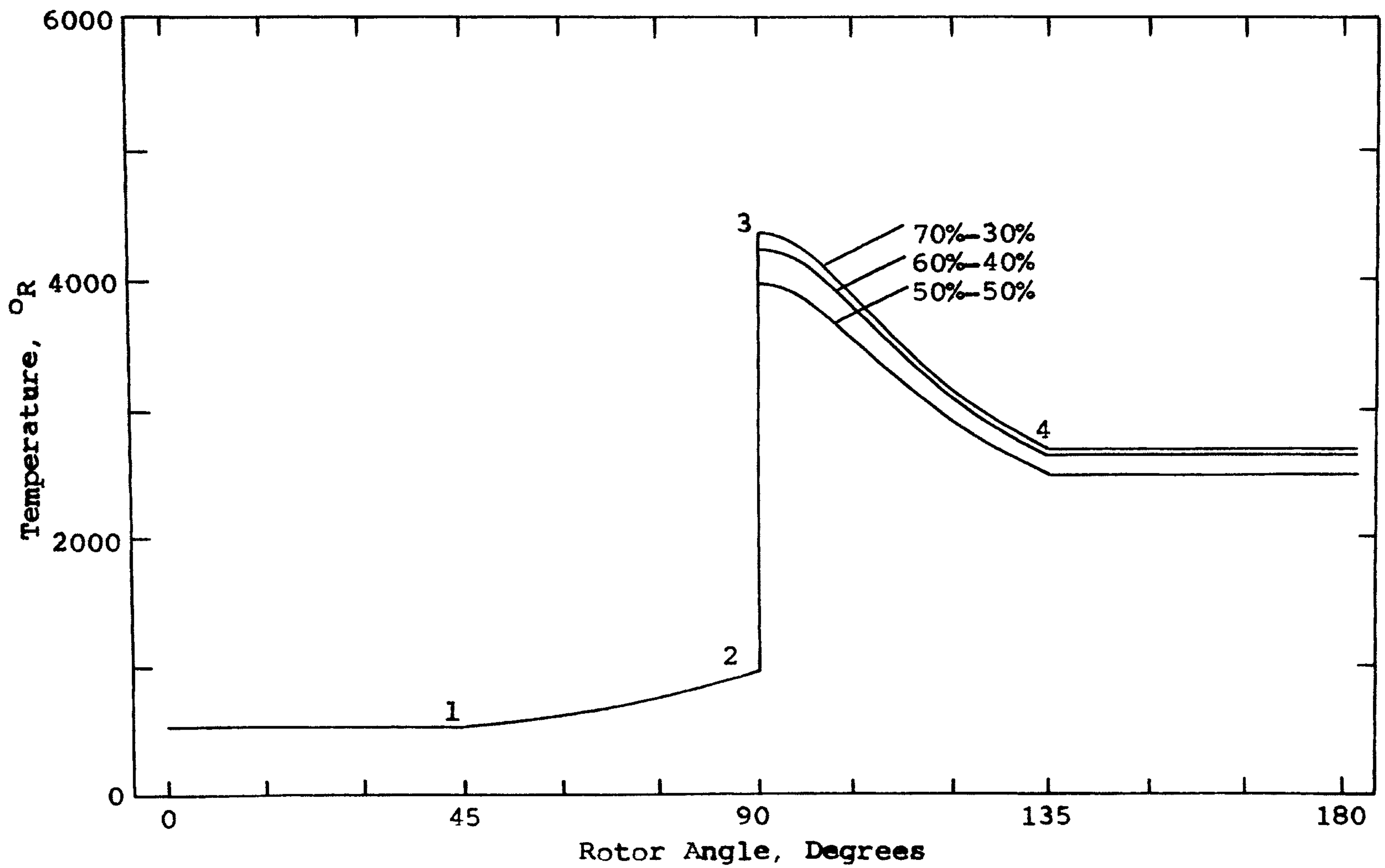


Figure 21 Gas Temperature Profile (Constant Volume-Constant Pressure Combustion

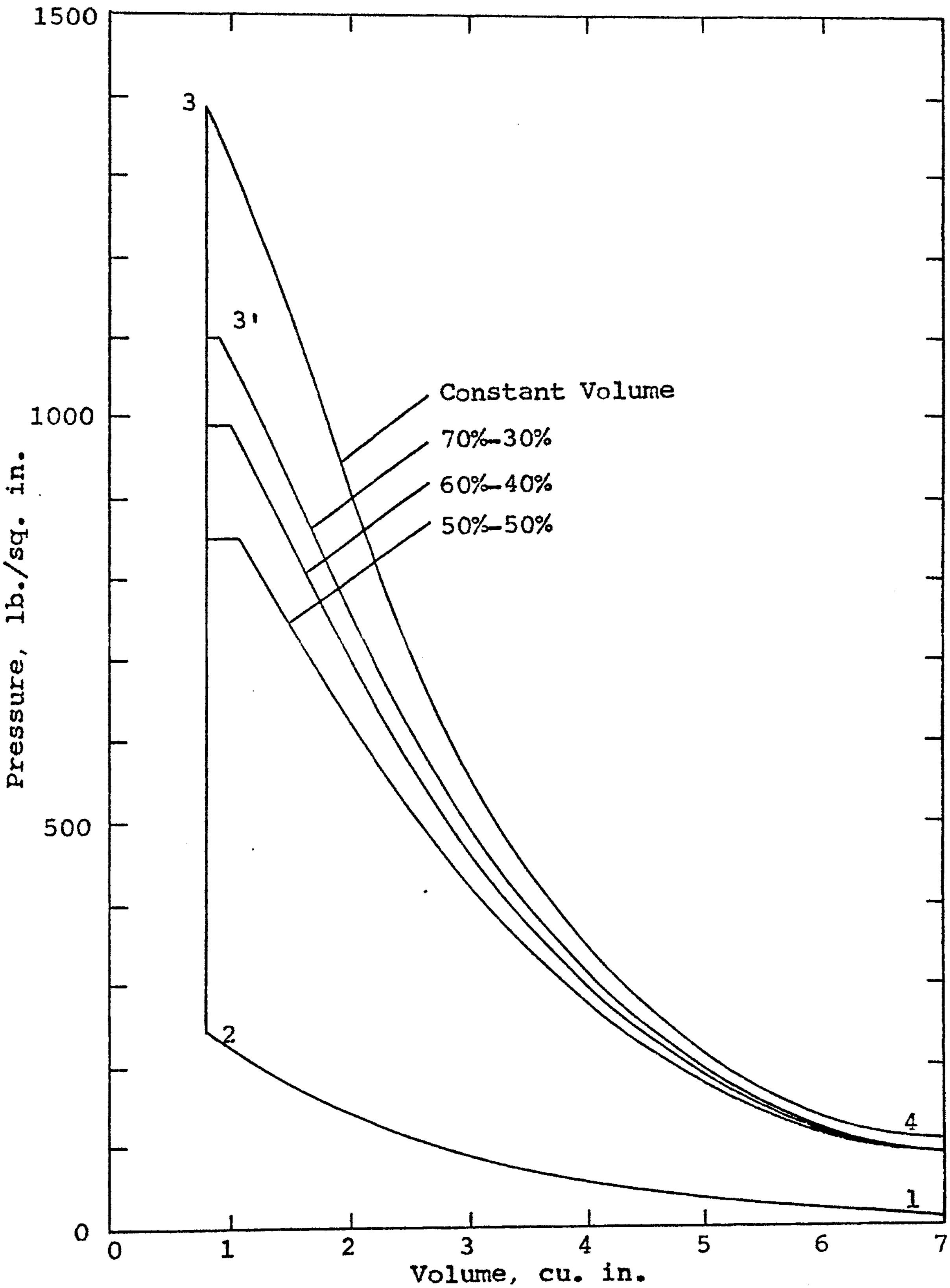


Figure 22 P-V Profile

TABLE I

Properties of the End States

Type of Combustion	Properties at 1	Properties at 2	Properties at 3	Properties at 3'	Properties at 4
1. Constant volume	P = 14.7 V = 12.88 T = 500°	P = 242 V = 1.516 T = 970°	P = 1380 V = 1.516 T = 5100°		P = 105 V = 12.88 T = 3320°
2. 70-30	P = 14.7 V = 12.88 T = 500°	P = 242 V = 1.516 T = 970°	P = 1100 V = 1.516 T = 4030°	P = 1100 V = 1.6 T = 4400°	P = 85 V = 12.88 T = 2700°
3. 60-40	P = 14.7 V = 12.88 T = 500°	P = 242 V = 1.516 T = 970°	P = 980 V = 1.516 T = 3620°	P = 980 V = 1.75 T = 4290°	P = 84 V = 12.88 T = 2670°
4. 50-50	P = 14.7 V = 12.88 T = 500°	P = 242 V = 1.516 T = 970°	P = 850 V = 1.516 T = 3080°	P = 850 V = 1.85 T = 3980°	P = 82 V = 12.88 T = 2500°

70-30 = 70% constant volume combustion followed by 30% constant pressure combustion

P (Psia)
V (ft^3)
T (°R)

that the inertial forces during rotation of one complete cycle cause no net work on the system. This computation was done in order to determine the magnitude and direction of the resultant force on the rotor that would cause rotation. If peak dynamic forces were desired for bearings, hinge pins, etc., then a complete dynamic analysis would have been required.

The assumption for this analysis was that the piston lay at the mid-point of its power stroke; and that the cam follower groove "U-U," shown in figure 23, was linear in nature from apogee to perigee in the stroke.

Letting the Mean Effective Pressure equal an unknown constant pressure P, and taking forces and moments equal to zero on the piston; the internal forces Fn, Fx, and Fy were found by analyzing a free body diagram of the piston, (see figure 23).

The free body diagram of the rotor, (see figure 24), was made showing the component forces at the hinge pin position and the magnitude, direction, and position of the MEP force acting on that portion of the rotor which forms a part of the cylinder wall.

Finding the sum of moments about the center of the rotor gave the net moment causing rotation to be:

$$\text{Sum of Moments} = .442 \text{ MEP (in-lbf)} . \qquad (7)$$

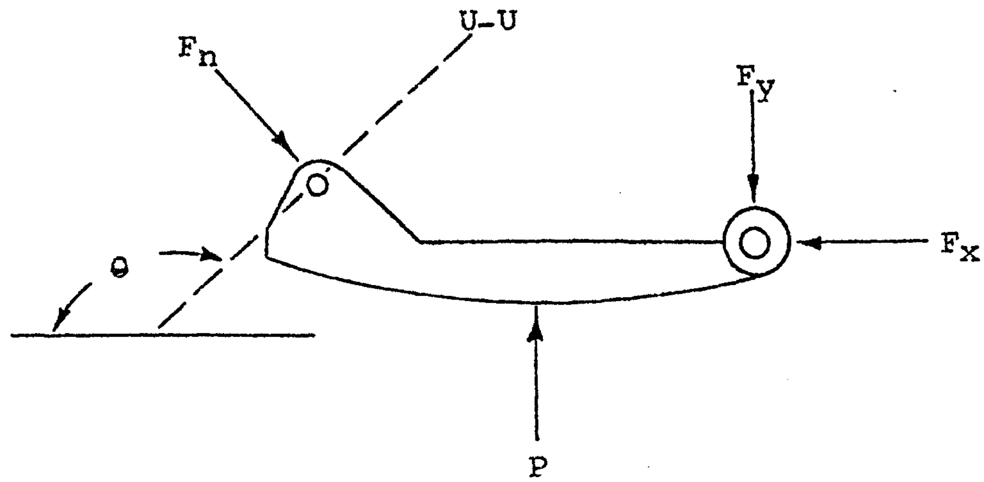


Figure 23 Static Forces (Piston)

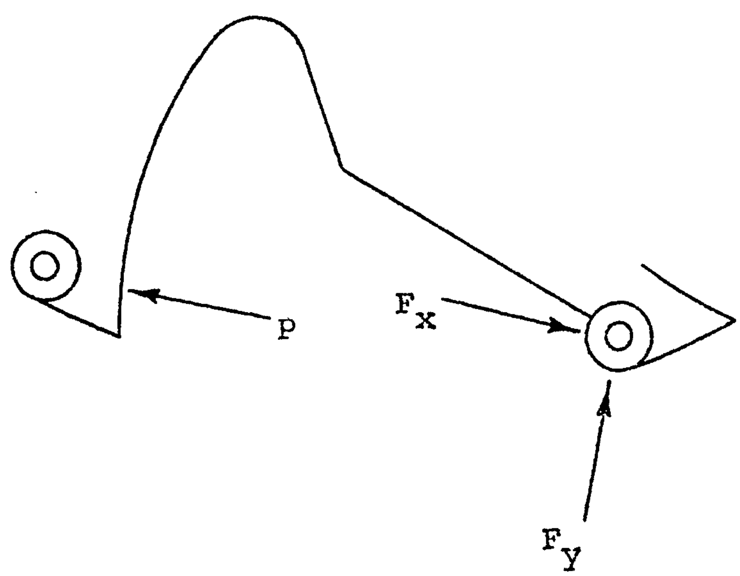


Figure 24 Static Forces (Rotor)

D. Kinematic Analysis

A dynamic analysis of the engine was made to determine the magnitude of the inertia forces on the piston during operation. The speed of the rotor was assumed to be a constant 1000 RPM. At this speed, it was found that the rotor moved through five degrees of rotation in .000833 seconds. The radial distance between the center of the rotor (output shaft) and the c.g. of the piston was measured for one full cycle or 180 degrees. These data are plotted as a function of rotor angle or rotation in figure 25. The change in radial distance of the piston from the rotor center was computed at five degree intervals and divided by the time to traverse that interval, to arrive at the average radial velocity of the piston as seen in figure 27. Since the tangential component of velocity of the piston was seen to vary womewhat constantly from 39.5 ft/sec to 47.5 ft/sec during nine time increments, or 45 degrees; this component was linearized to change by 1 ft/sec during each time interval. The average tangential acceleration of the piston was then found by dividing the incremental change in tangential velocity for one time interval by the time interval. The resulting tangential velocity profile is plotted in figure 26.

Examination of the velocity curves indicate that, the radial velocity of the piston, which is the normal component, begins at zero, increases to a maximum, and

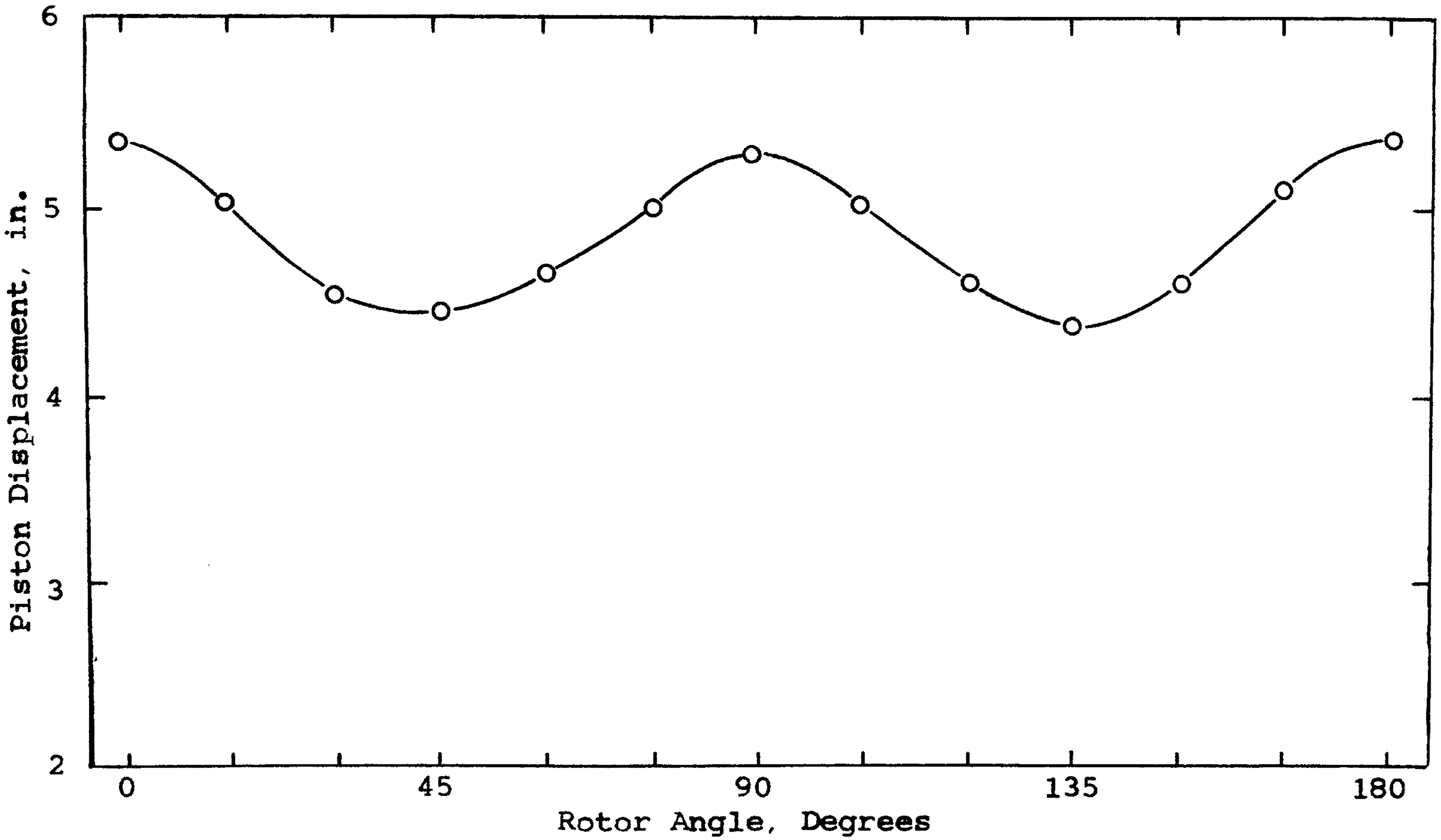


Figure 25 Piston C.G. Displacement

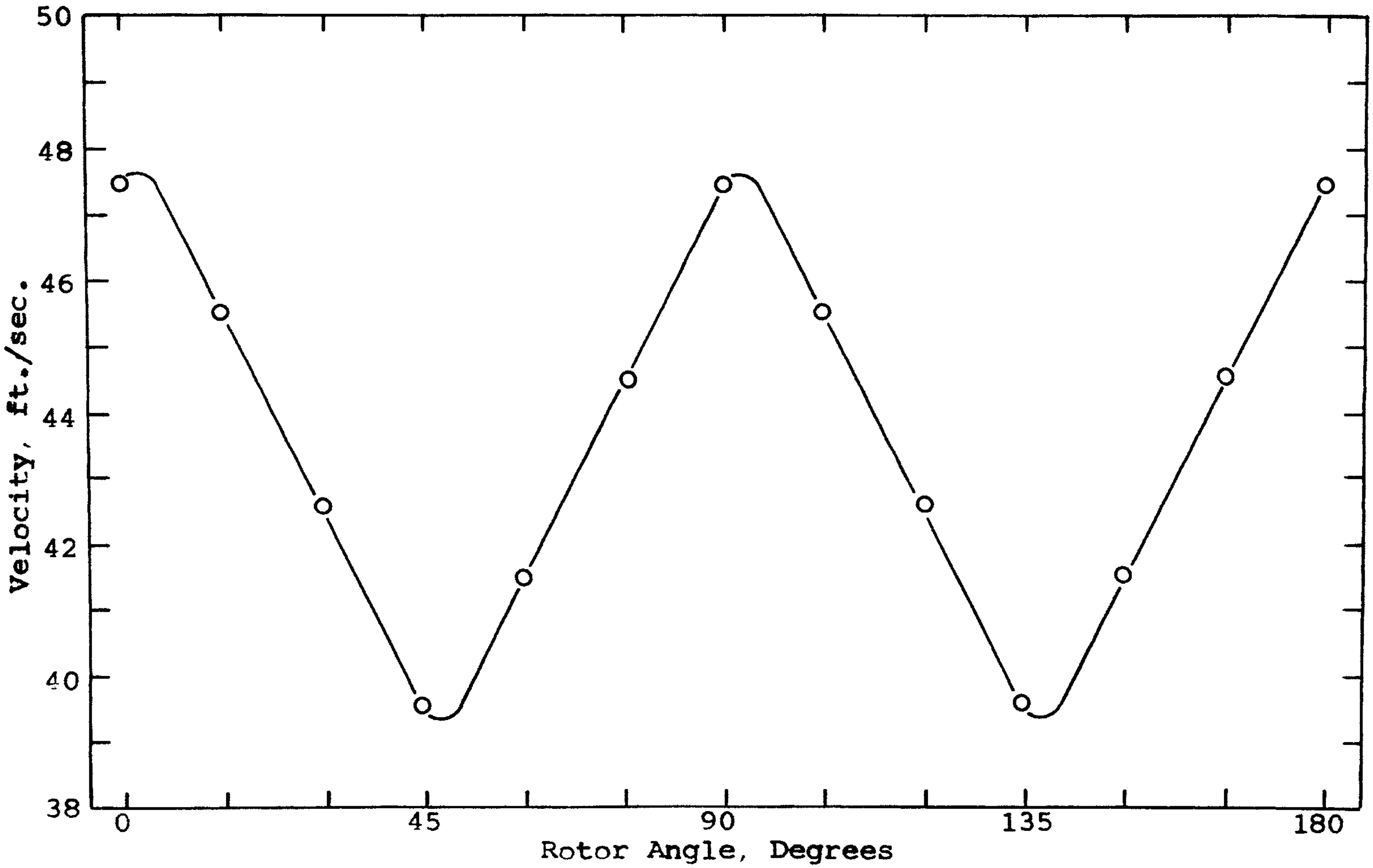


Figure 26 Piston C.G. Tangential Velocity

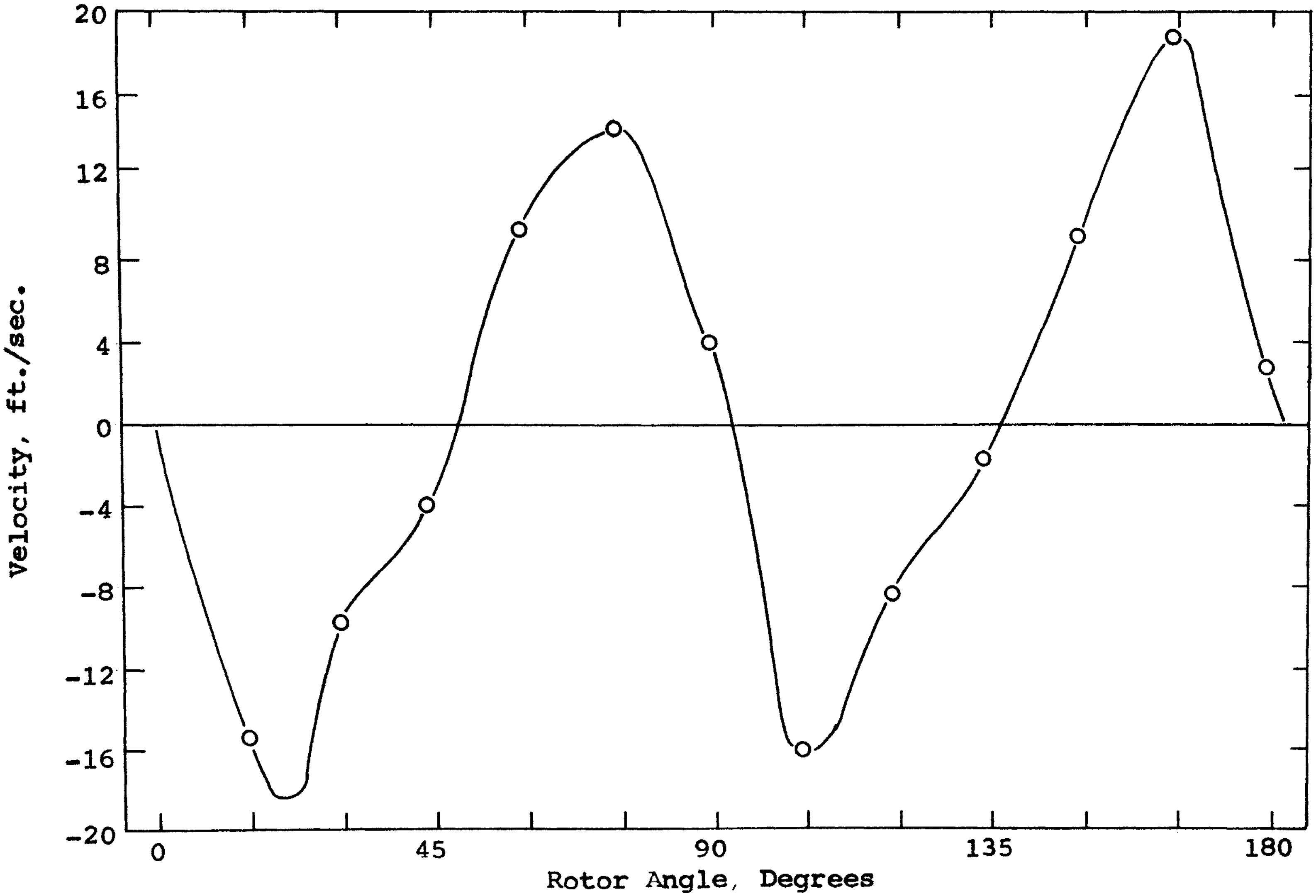


Figure 27 Piston C.G. Radial Velocity

finally decreases to zero during each stroke or period of the cycle. Further, the corresponding radial acceleration must assume positive zero, and negative values during each period. For this reason, the net acceleration on a piston during any stroke equals zero, which indicates that the inertial forces on the piston do not contribute to the net work of the system, (see appendix E). This is not to say that the acceleration forces have no effect on engine loads, since they do influence to some extent the friction forces.

The analysis above assumes that frictional forces are negligible, and that neither combustion nor shaft work by or on the engine is occurring.

Data for incremental displacement, velocity, and acceleration of a piston during a full cycle are found in table II.

E. Ideal Mean Effective Pressure

The net work of the engine process provides a means for calculating its mean effective pressure. This is the constant pressure acting on the engine piston for its work stroke that would result in the net work of the engine process. In equation form,

$$IMEP = (\text{net work})(J)/144(V_1 - V_2) \ \text{lbf/in}^2 \qquad (8)$$

TABLE II

Kinematic Piston Data for One Cycle

Rotor Angle (degrees)	Change in Radial Displacement (in)	Radial Velocity (ft/sec)	Tangential Velocity (ft/sec)	Radial Acceleration (ft/sec^2)	Tangential Acceleration (ft/sec^2)
5	0.063	6.250	47.5	-7500	-1200
10	0.125	12.500	46.5	-7500	-1200
15	0.156	15.625	45.5	-3750	-1200
20	0.188	18.750	44.5	-3750	-1200
25	0.125	12.500	43.5	7500	-1200
30	0.094	9.375	42.5	3750	-1200
35	0.063	6.250	41.5	3750	-1200
40	0.040	4.000	40.5	2700	-1200
45	0.020	2.000	39.5	2400	-1200
50	0.031	-3.125	39.5	6150	1200
55	0.063	-6.250	40.5	3750	1200
60	0.094	-9.375	41.5	3750	1200
65	0.120	-12.000	42.5	3150	1200
70	0.125	-12.500	43.5	600	1200
75	0.145	-14.500	44.5	-2400	1200
80	0.100	-10.000	45.5	-5400	1200
85	0.063	-6.250	46.5	-4500	1200
90	0.040	-4.000	47.5	-2700	1200

TABLE II (continued)

Kinematic Piston Data for One Cycle

Rotor Angle (degrees)	Change in Radial Displacement (in)	Radial Velocity (ft/sec)	Tangential Velocity (ft/sec)	Radial Acceleration (ft/sec^2)	Tangential Acceleration (ft/sec^2)
95	0.063	6.250	47.5	-12,300	-1200
100	0.100	10.000	46.5	-4500	-1200
105	0.156	15.625	45.5	-6750	-1200
110	0.125	12.500	44.5	3750	-1200
115	0.100	10.000	43.5	3000	-1200
120	0.083	8.250	42.5	2100	-1200
125	0.063	6.250	41.5	2400	-1200
130	0.031	3.125	40.5	3750	-1200
135	0.015	1.500	39.5	2010	-1200
140	0.031	-3.125	39.5	5550	1200
145	0.063	-6.250	40.5	3750	1200
150	0.094	-9.375	41.5	3750	1200
155	0.125	-12.500	42.5	3750	1200
160	0.156	-15.625	43.5	3750	1200
165	0.188	-18.750	44.5	-3750	1200
170	0.156	-15.625	45.5	-3750	1200
175	0.094	-9.375	46.5	-7500	1200
180	0.031	-3.125	47.5	-7500	1200

The ideal mean effective pressure corresponding to each combustion analysis can be found in table III.

F. Ideal Horsepower

Once the IMEP was found, the ideal horsepower was calculated using the net rotational moment from appendix B and,

$$\text{Horsepower} = 2(.442\ \text{IMEP})(2\pi)/33{,}000\ . \qquad (9)$$

The horsepower for 1000 RPM resulting from each combustion calculation is found in table III.

G. Ideal Otto Engine Efficiency

The efficiency of any engine process is the ratio of the net work obtained to the energy supplied. For the energy supplied, it is customary to use some arbitrary value, such as the heat value of the fuel at constant pressure, and some standard temperature, such as 77 °F.

The low heat value of liquid isooctane at standard conditions minus the latent heat value at standard conditions in equation form is:

$$(Q_p - H_{fg})_{77\ ^\circ F} \qquad (\text{Btu/mole}) \qquad (10)$$

Dividing the quantity in equation (10) by the product of the gram molecular weights of isooctane and air results in the low heat value of the mixture.

The Ideal Otto Engine efficiency is then equal to:

$$\frac{\text{net piston work}}{\text{low heat value}} ,$$

and this result for each combustion calculation also appears in table XII.

TABLE III

Calculated Performance at 1000 RPM

Type of Combustion Analysis	Net Piston Work (Btu/lbm air)	IMEP (lbf/in^2)	Horsepower	Ideal Efficiency %
1. Constant Volume	783	373	62.8	62.4
2. 70% Constant Volume 30% Constant Pressure	528	251	42.3	42.1
3. 60% Constant Volume 40% Constant Pressure	518	246	41.4	41.3
4. 50% Constant Volume 50% Constant Pressure	483	230	38.7	38.5

CHAPTER V

EXPERIMENTAL RESULTS

After machining was completed, the prototype was assembled. A test bench was built to serve as the engine mount, and contain the ignition harness and fuel system. Both fuel and electrical systems employed quick shut-down capability in case of an engine failure. An electric fuel pump was used in order to maintain constant fuel pressure at the carburator.

When attempting to start the engine with a large hand-held drill motor, combustion was seen to occur intermittently on both banks of the engine; but eventually subsided to intermittent combustion on a single bank. This was caused by a breakdown of the six-volt ignition system. During combustion it was noted that the engine was unable to sustain itself. Following a compression test using a transducer, amplifier, and oscilloscope, the compression ratio was found to be extremely low and almost non-existant, (see figures 28-30 and table IV). The low compression was caused by the poor sealing characteristics. Contouring the seal by heat warping to alleviate machining costs produced a very tight fit in the seal groove due to the fact that the seal curvature was not a constant arc as required. The poor fit between seal and groove would not permit the seal springs to force many of the seals against the facing surfaces giving rise to areas of large pressure loss during any given period in the cycle. Furthermore, there was an area over the hinge-pin mechanism of each piston which remained unsealed. Based on the foregoing problems, the prototype compression ratio fell considerably short of the design criteria of 8.5:1.

Although new and proper seals may indicate that the carburator and induction manifold were functioning properly during the initial testing, it appeared that the mixture was rich and unable to be leaned. Furthermore, although each of the exhaust manifolds are thirty inches long; in the attempts at running, neither were able to extinguish a flame within their length. The rich mixture may have accounted for this problem.

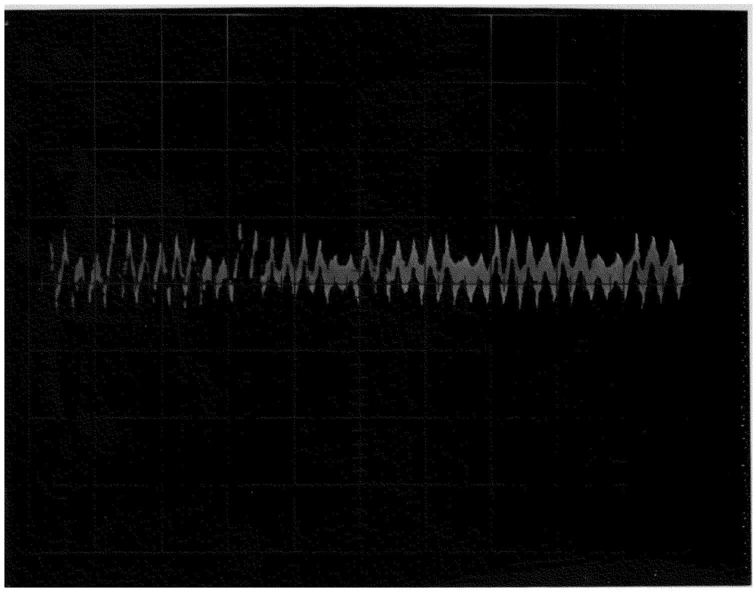


Figure 28 Pressure Profile 300 RPM

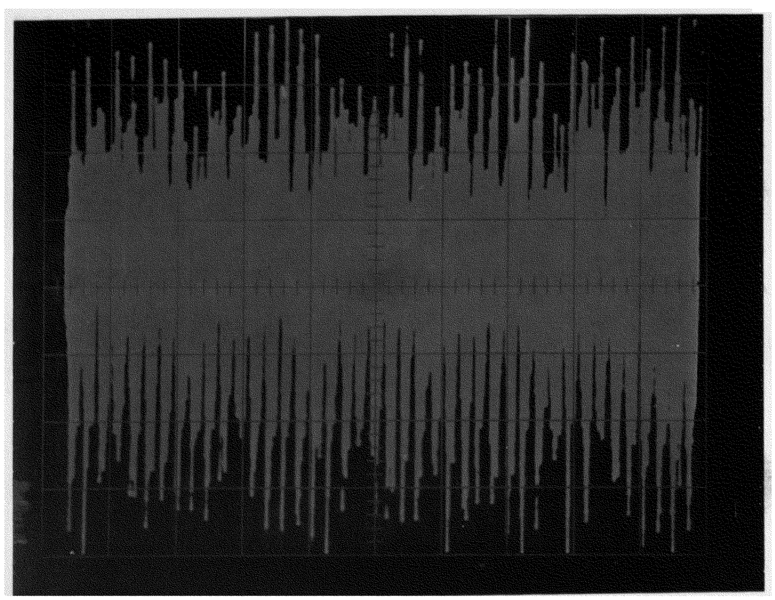


Figure 29 Pressure Profile 329 RPM

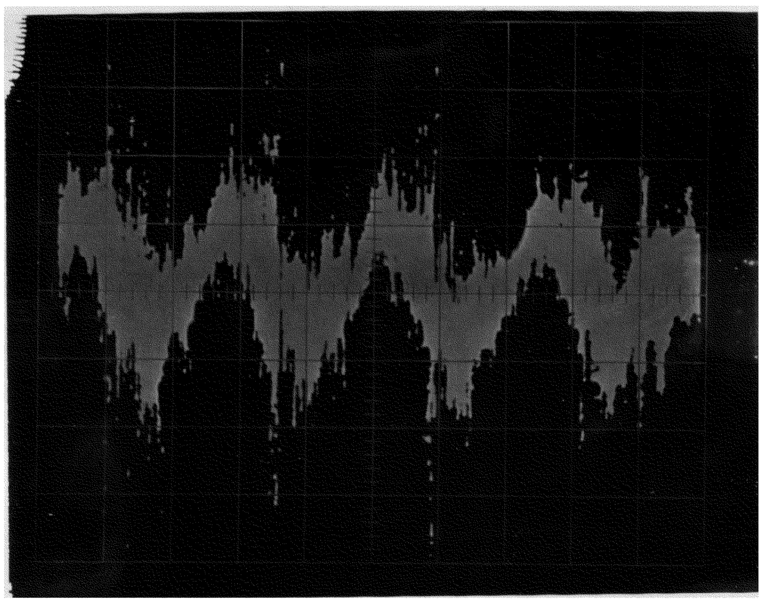


Figure 30 Pressure Profile 342 RPM

TABLE IV

Rotor Speed and Cylinder Head Compression

Run No.	Kiestler Output mv/psi	Oscilloscope Sweep Cal. sec/div	Oscilloscope Horizontal Cal. mv/div	Indicated RPM	Cylinder Head Pressure psi
1	50	.10	10	300	.20
2	100	.10	10	329	.35
3	100	.01	10	342	.45

CHAPTER VI

DISCUSSION

In the design of this prototype engine, an attempt was made to initiate the development of a small, light-weight internal combustion engine exhibiting reasonable reliability. One preliminary aspect of the design criteria was that the resulting concept should be at least theoretically less pollutant than conventional engines of the day. It was believed that the combustion or power stroke could be improved with a design of this type since the dynamics of the system lended itself readily to the employment of continuously firing spark plugs. With this method of mixture ignition, it was believed that unburned hydrocarbons in the exhaust could be reduced considerably by eliminating "piston runaway" at higher engine speeds.

"Piston runaway" is a term coined by the industry to describe the idea that, under certain conditions of cylinder head temperature and pressure, RPM, mixture ratios and varying octane ratings; the piston tends to move faster than the flame front propogating from the spark plug. When this occurs, the piston begins its exhaust stroke prior to complete combustion, resulting in an amount of unburned fuel in the exhaust products. Certainly, it was not assumed at this point, that using a constant spark ignition system would completely eliminate unburned fuel in the exhaust gases. It was assumed that an ignition such as this, coupled with the piston expansion concept, would help to alleviate the problem. In addition, it was obvious that pollution control devices could be readily adapted to this engine in a manner similar to that of present engines.

Since there are no valves to provide for engine breathing, but instead fixed ports, elimination of a considerable number of moving parts was accomplished when compared to the reciprocating engine aspiration system.

Eight pistons seemed to be optimum for the design when consideration was given to the geometry of the cam groove. A non-multiple of four pistons was eliminated immediately since they could not perform when governed by a four-cycle process. The use of four pistons was ruled out due to pressure losses to the cam groove during the combustion process. Any multiple over eight pistons seemed to congest the system with too many moving parts.

It was decided to machine the prototype of aluminum and use aircooling to lower the cost. Standard induction and exhaust systems were chosen for the same reason.

Assembled engine weight was slightly under sixty pounds which included the manifolds. Generally, the engine is fifteen inches in diameter by approximately six inches deep, for the horsepower ranges reported herein.

Analytical results indicate the range of shaft horsepower at 1000 RPM to be between 30 and 45. In addition, elimination of the clearance gases prior to combustion appeared to increase resulting horsepower by eliminating the (1-f) multiplier term from the energy calculations.

Theoretical efficiencies are high because they only represent the efficiency of the work on the piston and do not account for losses by rotor work.

In the dynamic analysis of the system, it was found that inertial forces cause no net work on the system when friction forces are considered negligible.

Although preliminary testing has fallen far short of anticipated goals due mainly to poor sealing of the combustion chamber, the failure has not been with the design, but with the lack of funding necessary to produce the prototype in a manner necessary to arrive at the attainable results.

CHAPTER VII

CONCLUSION

A new rotary internal combustion engine has been designed which possesses the following potential:

1. High horsepower per pound. Theoretically, since the engine package weight was kept under sixty pounds, the combustion calculations indicate the engine is capable of 1/2 to 2/3 horsepower per pound.

2. Continuous combustion. There are two pistons in a power stroke at any time during engine operation.

3. Few moving parts. Neglecting bearings, the engine has ten moving parts, eight of which are pistons.

4. Symmetric geometry. Two combustion processes, and in fact all processes, take place simultaneously and continuously opposite each other around the engine, thereby eliminating the need for counterweights and flywheels.

5. Ease of fabrication. In production, with the exception of a two piece cast housing, all components could essentially be extruded and cut to width, followed by light machining.

6. Small size. The engine less manifolds is fifteen inches in diameter by approximately six inches deep.

7. Friction forces small. Due to fewness of moving parts, friction loads are reduced when compared to similar output engines.

8. Smooth operation. Eight pistons result in sixteen power strokes per revolution of the output shaft, in addition to cancellation of the inertia forces due to symmetry and rotational motion.

9. Assembly time. Owing ot its simplicity, the entire engine package can be torn down and rebuilt in a matter of minutes. This should improve maintenance ability of such engines.

10. Cycle timing eliminated. The timing process is a constant built into the design by the positioning of the cover plates containing the cam follower grooves.

11. Variable compression ratio. Compression ratios may be varied simply by replacing the two cover plates containing the cam groove.

APPENDIX A

Ideal Combustion Analysis

The charts referred to herein were taken from reference 11. Assuming a chemically correct mixture of iso-octane vapor and air, 1.0 F/A (stoich.), recall that:

$P_1 = 14.7$ psia

and, $T_1 = 500^{o}R$.

1. During the compression process,

$$n(Sv)_{T_2} - n(Sv)_{T_1} = n_{1.0}R \ln r \ . \qquad (A.1)$$

From chart B, $n_{1.0}R = 0.0701$ Btu/^{o}R lbm air,

$n(Sv)_{T_1} = 0.043$ Btu/^{o}R lbm air

and $U_1 = 93$ Btu/lbm air .

From equation A.1,

$n(Sv)_{T_2} = 0.178$ Btu/^{o}R lbm air .

Also from chart B, at the intersection of $n(Sv)_{T_2}$ and the constant 1.0 F/A line:

$T_2 = 970\ ^{o}R$

and $U_2 = 190$ Btu/lam air .

From the PVT relation: $V_1 = KT_1/P_1$ (A.2)

where $K_{1.0\ F/A} = 0.379$,

$$V_1 = .379(500)/14.7 = 12.88 \text{ ft}^3/\text{lbm air},$$

$$V_2 = V_1/r = 12.88/8.5 = 1.516 \text{ ft}^3/\text{lbm air},$$

and $$P_2 = KT_2/V_2 = .379(970)/1.516 = 242 \text{ psia.} \quad (A.3)$$

The work of compression is:

$$U_2 - U_1 = 190 - 93 = 97 \text{ Btu/lbm air} . \quad (A.4)$$

2. The first method of combustion assumes an adiabatic, constant volume process. The energy equation for this process is:

$$n(U + C)_2 = n(U + C)_3 \quad (A.5)$$

Using chart D, $C_2 = 1275(1-f)$, where "f" equals the clearance gas weight fraction which is zero in the case of this engine. Therefore,

$$C_2 = 1275 \text{ Btu/lbm air},$$

and $$(U + C)_2 = 1465 \text{ Btu/lbm air} .$$

The intersection of the $(U + C)_2$ and $V_2 = V_1$ lines on chart D determines the properties at state 3 to be:

$$T_3 = 5100 \text{ }^{\circ}\text{R}$$

$$P_3 = 1380 \text{ psia}$$

and $$S_3 = 2.185 \text{ Btu/}^{\circ}\text{R lbm air} .$$

3. For constant volume combustion followed by constant pressure combustion at a ratio of 70% and 30% of the mixture reacting respectively, the analysis begins at 2, or the end of the compression stroke.

For 70% of the mixture burning at constant volume,

$$(U + C)_2 = 0.070(U + C)3 \ . \qquad \text{(A.6)}$$

Since V_2 = 1.516 ft^3, from chart D:

$$T_3 = 4030 \ ^oR$$

$$P_3 = 1100 \ ^oR$$

$$S_3 = 2.093 \ Btu/^oR \ lbm \ air$$

$$(U + C)_2 = 1025 \ Btu/lbm \ air$$

and $$(H + C)_3 = 1320 \ Btu/lbm \ air \ .$$

For 30% of the remaining mixture burning at constant pressure, the energy equation becomes:

$$(H_A)_{T_3} + 0.30(H + C)_3 = (H + C)_{3'} - Q \qquad \text{(A.7)}$$

From appendix C, Q = 0, since it is approximately 1% of the total heat of combustion. For $P_3 = P_{3'}$, from reference 11,

$$(H_A)_{T_3} = 1104 \ Btu/lbm \ air$$

and $$(H + C)_{3'} = 1104 + 0.30(1320) = 1470 \ Btu/lbm \ air.$$

From chart D at $P_3 = P_{3'}$ = 1100 psia, and $(H + C)_{3'}$ = 1470,

$$V_{3'} = 1.6 \ ft^3/lbm \ air$$

$$S_{3'} = 2.125 \ Btu/lbm \ air$$

and $$T_{3'} = 4400 \ ^oR \ .$$

The resulting expansion process follows the same procedure as outlined below.

4. The energy equation for the expansion process analysis of constant volume combustion is:

$$\text{Work} = n(U + C)_3 - n(U + C)_4 \ . \qquad \text{(A.7)}$$

At the end of expansion,

$$V_4 = V_1 = 12.88 \ \text{ft}^3/\text{lbm air} \ .$$

Applying this data to chart D,

$$T_4 = 3320 \ ^{o}R$$

$$P_4 = 105 \ \text{psia}$$

and $$n(U + C)_4 = 800 \ \text{Btu/lbm air} \ .$$

The net piston work for the preceding process analysis is:

$$\text{Net Work} = \text{Expansion Work} - \text{Compression Work}, \qquad \text{(A.8)}$$

so,

$$W = 880 - 97 = 783 \ \text{Btu/lbm air} \ .$$

APPENDIX B

Static Force Analysis

In the following analysis,

Cylinder pressure = MEP = P,

and the engine cylinder has a unit depth. For the piston forces, see the free body diagram, (see figure 23).

Summing moments about "A" equal to zero gives:

$$Fn\ (.1875) - Fy\ (.625) = 0,$$

or $$Fy = .3\ Fn \qquad \text{(B.1)}$$

Summing forces in the X-direction equal to zero gives:

$$Fn \cos 47^{\circ} - Fx = 0,$$

or $$.682\ Fn - Fx = 0 \qquad \text{(B.2)}$$

Summing forces in the Y-direction equal to zero gives:

$$1.25\ P - Fy - Fn \sin 47^{\circ} = 0$$

or $$1.25\ P - Fy - .732\ Fn = 0 \qquad \text{(B.3)}$$

Equating B.1 and B.3 gives:

$$1.25\ P - .3\ Fn - .732\ Fn = 0$$

or $$Fn = 1.21\ P$$

Similarly,

$$Fy = .363\ P$$

and $$Fx = .825\ P\ .$$

Using a free body diagram of the rotor, (figure 24), with forces Fx, Fy and the pressure force "P" known, the resulting net moment causing rotation is:

$$\text{Net Moment} = D_2 d_2 - Fxd_3 + Fyd_4 \; . \tag{B.4}$$

Therefore:

$$.825\ P\ (1.633) + .363\ P\ (1.0) - .25\ P\ (1.75) = \text{net moment,}$$

or, the net moment causing rotation = .442 P (in-lbf).

APPENDIX C

Combustion Process Heat Losses to the System

Assuming the mean effective gas temperature, Tg, to be 1200 ^{O}F, the mean surface temperature, Tsg, of the cylinder on the gas side is approximately 350 ^{O}F, (see reference 11, page 192).

For the heat flow from the gas to the inside wall of the cylinder,

$$Q/A = hg\ (Tg - Tsg)\ . \qquad (C.1)$$

In order to evaluate hg, the Reynolds Number was found from reference 11 as:

$$Re = GL/vg\ , \qquad (C.2)$$

where; $G = 0.88$ lbm/sec ft^2 ,

$Vg = 22 \times 10^{-6}$ lbm/sec ft

and L = mean stroke = 1/12 ft .

Now, from equation C.2,

$$Re = 3330\ ,$$

and the corresponding Nusselt Number, Nu, for a fuel-air mixture of 1.0 (stoich.) is 4500 as taken from reference 11, page 194.

Further,

$$Nu = hgD/Kg\ , \qquad (C.3)$$

where $Kg = 8.4 \times 10^{-6}$ Btu/sec ^{O}F - ft for isooctane vapor and air.

From equation C.3,

$$hg = .1135 \text{ Btu/sec ft}^2 \, {}^{\circ}\text{F} ,$$

assuming the characteristic length "D" equals four inches.

Knowing hg and the temperature difference from equation C.1,

$$Q/A = 10.66 \text{ Btu/lbm} ,$$ or approximately 1% of the heat generated during combustion.

APPENDIX D

Experimental RPM and Cylinder Head Pressure

The sample data are taken from run 3, table IV,

where: Kiestler output: 100 mv/psi

Scope calibration:	sweep	horizontal
	.01 sec/div	10 mv/div

Figure 30 indicates 2.2 divisions per pressure pulse, therefore, (2.2 div/pulse)(.01 sec/div) = 45.5 pulses/sec. Now, (45.5 pulses/sec)(1 rev/8 pulses)(60 sec/min) = 342 RPM. For the head pressure at this RPM,

$$(10 \text{ mv/div})(4.5 \text{ div})(1/100 \text{ psi/mv}) = .45 \text{ psia.}$$

APPENDIX E

Kinematic Analysis

Letting the rotor, pistons, and cover plate grooves function as a mechanism, the net power of the mechanism may be written as the sum of:

$$\int_0^t T_n \cdot w_n \, dt + \int_0^t F_c \cdot V_c \, dt - \int_0^t M_n A_{Gn} \cdot V_n \, dt - \int_0^t I a_n \cdot w_n \, dt = 0 \tag{E.1}$$

where t = time of one stroke

V = velocity of the CG of the member

A_n = acceleration of the CG of the member

a_n = angular acceleration of the CG of the member n

w_n = angular velocity of the CG of the member n

term 1 = rate of external work done on or by the system

term 2 = friction and combustion forces

terms 3 and 4 = instantaneous values of power due to inertial forces and torque.

Each of these terms accounts for the rate of work done by external torques or forces and inertial torques and forces on each rigid member of the system, (see reference 12).

Assuming that friction forces are small, combustion is not occuring, and there is no external work being done on or by the system, equation E.1 reduces to:

$$\int_0^t M_n A_{Gn} \cdot V_n \, dt - \int_0^t I a_n \cdot w_n \, dt = 0 \ . \qquad (E.2)$$

Rewriting the first term as:

$$\frac{M_n V_{Gn}^2}{2}\Bigg|_0^t = \frac{M_n}{2}(V_{Gn_t}^2 - V_{Gn_0}^2) \ ,$$

and noting that the radial velocity at the beginning and end of each stroke is zero, this term equals zero.

Rewriting the second term of E.2 as:

$$\frac{I w_n^2}{2}\Bigg|_0^t = \frac{I}{2}(w_{nt}^2 - w_{n0}^2) \ ,$$

note that the angular velocity, w_n, must be the same at the beginning and end of any one stroke of the piston. Also, the angular velocity for the rotor and the output shaft is very nearly constant. Hence, the last term is also zero.

This method confirms the argument in Chapter IV, that the inertial forces cause no net work on the system when employing the preceding assumptions.

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VITA

Frank Darrell Statkus was born on April 16, 1945, in Lincoln, Nebraska. He received his primary and secondary education in Champaign, Illinois; Cincinnatti, Ohio; and Florissant, Missouri. He has received his college education from the University of Missouri-Rolla, in Rolla, Missouri, and received his Bachelor of Science degree in Aerospace Engineering from the University in August 1971.

He has been enrolled in the Graduate School of the University of Missouri-Rolla since June 1971.