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A Critical Evaluation of the Geared Hypocycloid Mechanism for Internal Combustion Engine Application

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ABSTRACT

The geared hypocycloid mechanism, a kinematic arrangement that provides a straight-line motion, can be used as the basis for an internal combustion engine. Such an engine would have a number of advantages: Perfect balance can be achieved with any number of cylinders. The straight-line motion eliminates the need for a wrist pin bearing, further allowing a very short piston to be used without danger of cocking. Piston side load is virtually eliminated, and "piston slap" will not occur even with a large piston/cylinder clearance. These features make it particularly attractive for small single cylinder engine applications where vibration is undesirable, and also for the uncooled "adiabatic engines", in which piston cylinder lubrication and friction are major concerns.

The paper discusses the history of the concept, the limited engineering development that it has received, limitations and potential weak points of the design, and a presentation of design configurations that appear particularly attractive.

IN SPITE OF THE CLAIMS of potentially high fuel efficiency of alternative combustion engines such as the gas turbine and Stirling engine, the internal combustion piston engine will undoubtedly remain the predominant prime mover for passenger cars, trucks and buses for many years. It is even possible that, with further research and development, the efficiency of the internal combustion engine may be improved to such an extent that the alternatives never catch up. Electric cars are attractive from the point of view of reducing our dependency on petroleum fuel, but there are serious doubts as to whether an electric car would ever be satisfactory for general purpose use. Improved designs of piston engines can therefore be very important in reducing our national energy consumption well into the next century.

The topic of this paper is a kinematic mechanism that would replace the crankshaft-connecting rod system of a conventional engine. The term "hypocycloid" has been chosen, because it describes the basic kinematic principle involved. "A hypocycloid is a curve generated by a point on the circumference of a circle rolling internally on the directing circle"(1)*. If the smaller circle is 1/2 the diameter of the larger one, then the hypocycloid motion is a straight-line. Another important feature is that the motion is *simple harmonic*. If the "circles" are in actuality an internal gear and a mating pinion, with a diameter ratio of 2:1, then the resulting hypocycloid straight-line mechanism provides the basis of a possible replacement for the crankshaft-connecting rod system of a conventional piston engine. The term cardanic motion has also been used for this type of mechanism (2).

The basic hypocycloid mechanism as it might be applied to an engine is illustrated in Fig. 1. The "arm" is rigidly attached to the pinion gear, and that assembly is attached through a bearing to crank pin C. As the piston rod bearing D1 moves in its straight-line motion, the pinion rolls around the fixed internal gear, and the crank rotates in the opposite direction as the pinion-arm assembly. Note that the stroke of the piston is *four times* the crank radius. it is also important to note that D2 has a hypocycloidal straight-line motion perpendicular to that of D1.

The mechanism provides several features that make it potentially attractive for engine application. First, it allows perfect balance with any number of cylinders (including single-cylinder engines). The straight-line motion completely eliminates piston side thrust (except for the small amount caused by bearing friction). Such thrust occurs in a conventional engine from both gas and inertia forces due to the angle of the connecting rod, and contributes significantly to engine friction. The absence of piston side thrust also allows greater piston/cylinder clearance without danger of "piston slap". The straight-line motion eliminates the need for a wrist pin bearing. The piston can be made very short, since, with the piston and piston rod being one rigid assembly, there is no need for a piston skirt to prevent cocking. A short piston will weigh less and reduces the required cylinder length.

There are potential advantages in terms of engine friction and therefore efficiency. A recent paper (3) provides data on engine losses attributable to various factors, based on a passenger car being driven over the Federal Urban Driving Cycle. Approximately three percent of the input energy is lost to piston friction over the cycle. The average engine efficiency over the cycle is shown to be 25%, so that

*Numbers in parentheses designate references at end of paper.

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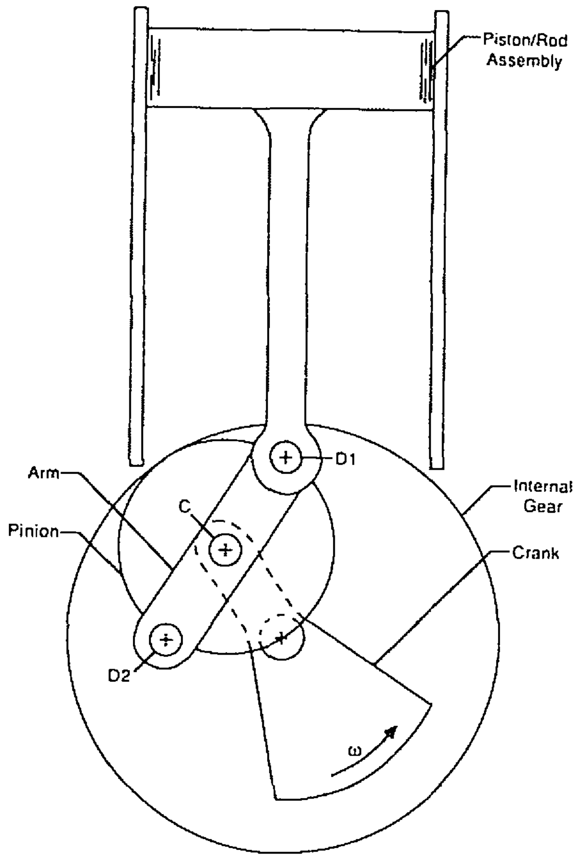


Fig. 1 A basic hypocycloid engine.

complete elimination of all piston friction would result in a 12% improvement in fuel economy. It is generally thought that about half of the piston friction is due to the rings, and half to the skirt. The ring friction is difficult to reduce appreciably without compromising the gas sealing function. If the skirt friction can be virtually eliminated (as should be possible in a hypocycloid engine with large piston/cylinder clearance, short piston length, and no side thrust), a passenger car with such an engine could be expected to have about a 6% increase in fuel economy.

It is thought by many that a significant reduction in friction can be achieved by replacing the journal bearings with ball or roller bearings. The hypocycloid engine concept allows for convenient application of such bearings. One important factor is the relative motion over the cycle between the inner and outer races. For a constant engine rpm, there would be a fixed relative speed between inner and outer races for the main bearings of all engines, but, with a connecting rod of finite length, a relative velocity between the two big-end races that varies significantly over the cycle. To quote from (4), "The rolling-element bearing is inherently unsatisfactory for a big end because of the angular accelerations to which the rollers are subjected, and one might prefer plain big ends for this reason, but no such objections apply to the main bearings where rollers would reduce the demands on oil pumping and thus minimize the losses that the designer is so keen to avoid." The hypocycloid engine, with no connecting rods, would eliminate the "angular accelerations" of the rollers, and therefore the above objection would not apply.

Besides the basic improvement in fuel economy that occurs with reduced mechanical friction, there are other potential fuel-saving advantages not quite so apparent at first glance. One of the main limitations to *maximum power* in current engines is the friction from piston side loading due to inertia forces which increase with the square of engine speed. If this component of friction could be removed, or at least dramatically reduced, then (with attention to breathing and mechanical strength) the maximum engine power should be significantly increased, occurring at a higher speed. This would allow a smaller engine displacement which would have the effect of improving efficiency at low levels of power, in addition to the advantages of lower weight and cost.

The features listed above indicate that there may be several important applications in which the hypocycloid engine might have significant advantages over a conventional engine. First of all, there are many single cylinder engine applications (e.g., lawn mowers, and engine-driven generator sets) in which vibration caused by the inherent unbalance of a conventional single-cylinder engine can be quite annoying. Conventional four-cylinder engines are also not able to be perfectly balanced, and application of the hypocycloid design could completely eliminate engine vibration in cars, tractors, and airplanes having 4-cylinder engines. Another potentially attractive application would be for an uncooled "adiabatic engine", in which the high temperature has been found to cause serious lubrication and wear problems, especially at the piston/cylinder interface where high side thrust occurs with conventional engine designs. Finally, the prospect of higher efficiency would make the hypocycloid concept attractive for virtually all motor vehicles.

The above advantages are obviously attractive, but the mechanism will be useful only if it can be shown to be capable of handling the loads and speeds required for a modern internal combustion engine, and able to be packaged within a reasonable envelope at a reasonable cost. These practical aspects will be emphasized in this paper.

Figure 2, taken from (5), graphically illustrates the balancing principles for a single-cylinder engine. Part (a) of the figure shows the condition at top dead center (TDC), (b) is for a crank rotation of 30°, part (c) for 60°, and (d) for 90° rotation. Extension of the principles involved for the full 360° rotation should be obvious. The mass M_{D1} of the piston and its rod can be considered to be concentrated at the journal pin D1. An equal mass must be placed at center D2, which moves through a straight-line motion that is perpendicular to that of D1. Note that the motions of D1 and D2 are simple harmonic. The inertia force of D1 is therefore maximum at TDC, and equal to that value times the cosine of the crank angle θ for all other positions. The inertia force at D2 is the same, except that it acts at right angles to that of D1, and is 90° out of phase. These inertia forces, and their vector sum, are illustrated in Fig. 2 for the four crank angles. The vector sum has a fixed value, $M_{D1}r\omega^2$, rotates at the crank rotational speed, and always acts through the crank centerline. It can be balanced by a balance weight E attached to the crank as shown. With the symmetry shown, the center of gravity of the pinion and arm should be located at the pinion centerline, and therefore able to be balanced by additional weight at E. The above will achieve balance in the plane illustrated. By properly distributing the weights along the crank axis, there will be no unbalance moments, and perfect balance can therefore be achieved.

Since a single-cylinder engine can be perfectly balanced, it is apparent that engines with any number of cylinders can also be perfectly balanced.

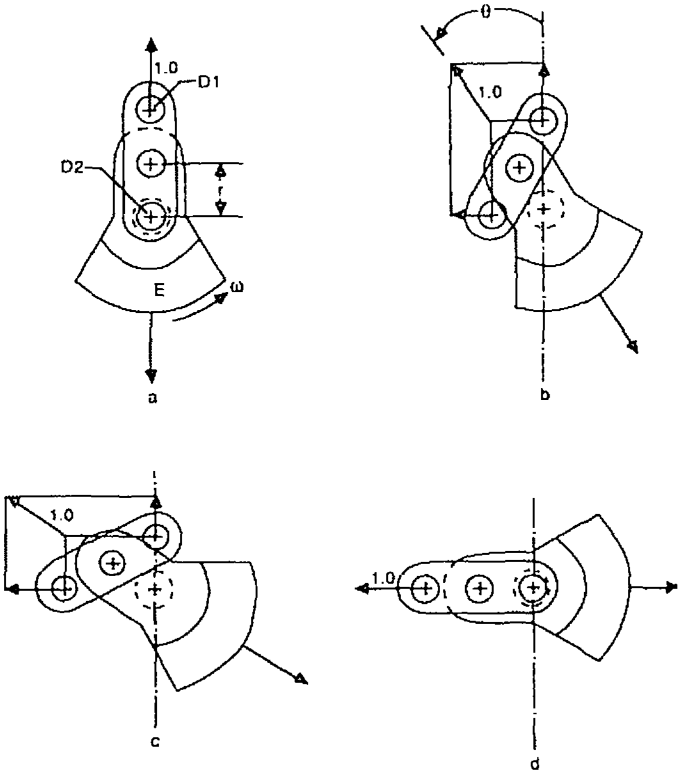


Fig. 2 Illustration of hypocycloid engine balance principles (5).

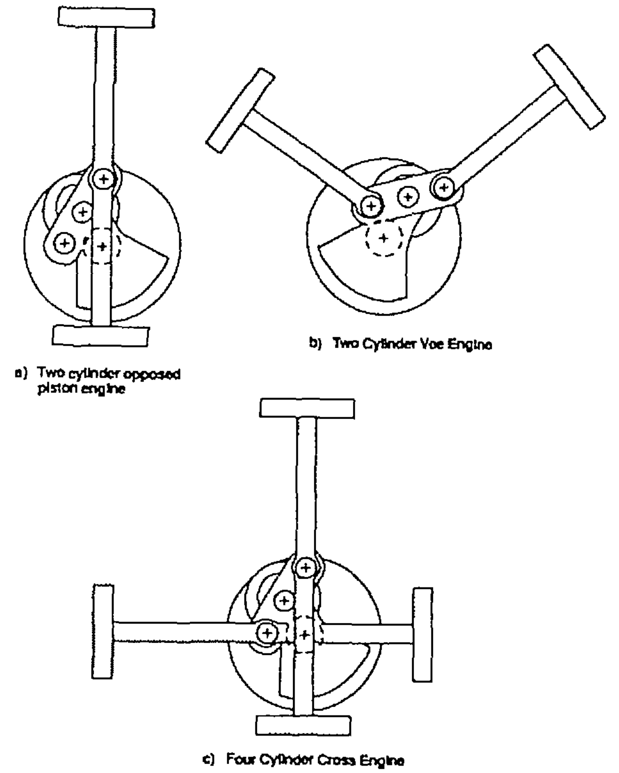


Fig. 3 Alternative configurations for hypocycloid engines.

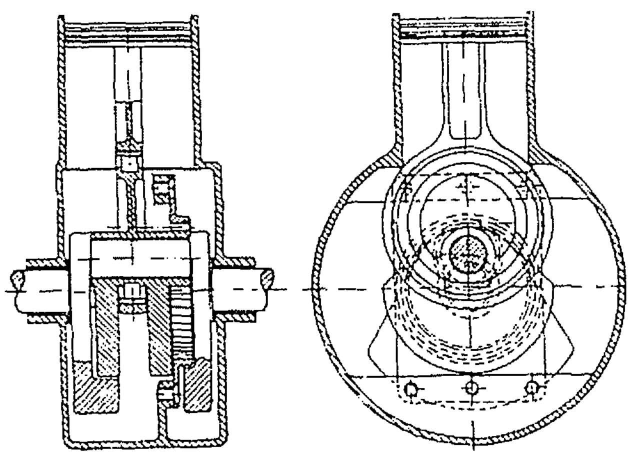


Fig. 4 Hypocycloid single-cylinder engine design by Ifield (5).

Because of the straight-line motion, the piston rod can be extended beyond the bearing D1 and have a second piston attached to it, forming an opposed piston engine (Fig. 3a). This does not require an additional bearing, so it is a quite attractive alternative, even though the balance weights would have to be increased (but not doubled). In a similar manner, the balance weight at D2 could be in actuality a piston-piston rod assembly, traveling at right angles to the first piston, forming a 90° Vee engine (Fig. 3b). If opposed pistons are used at both D1 and D2, then the result is a four-cylinder "cross" engine (Fig. 3c). The practical mechanical details of these and other configurations will be discussed later. Note that the straight-line motion also allows the possibility of double-acting pistons; i.e., with a piston rod seal, there could be combustion chambers on both sides of each piston. This would allow higher power density, but could introduce additional problems.

There are two major variations of the hypocycloid engine that use the internal gear/pinion pair. We have used the descriptive terms "big bearing engine" and "built-up shaft engine" because of the predominant characteristics of the two.

The engine of Fig. 4 is an example of a "big bearing engine" (5). It can be seen that the pinion is an integral part of a piece which turns on a long journal bearing. The rod is connected to this member via a large diameter bearing which must span the journal, hence the name "big bearing". The crankshaft of this engine is essentially the same as a conventional one-cylinder crankshaft, except that the center journal is quite long. It is likely to be made of three pieces assembled with press fits to facilitate engine assembly.

The engine of Fig. 5 is an example of the "built-up shaft" version of a single cylinder engine (P34). Here the pinion is fixed on the end of a member which has the appearance of a crankshaft. The piston rod is attached to the throw in the normal manner. The ends of this shaft are connected to the end pieces by bearings. The end pieces rotate around a fixed center. Power is taken off of the end pieces since the "crankshaft" or gear-shaft, is performing planetary motion. An opposed two-cylinder engine can be made using the same built up shaft by extending the piston rod and adding a second piston at the opposite end of the first.

It is important to have a countershaft connecting the two end pieces of the built-up shaft, even though this has not been realized by many inventors/investigators. If the two end pieces are not maintained in phase by a rigid countershaft system, the alignment will be achieved (and not accurately at that) only through undesirable twisting moments on the bearings connecting the three shaft pieces. This point is considered in more detail later.

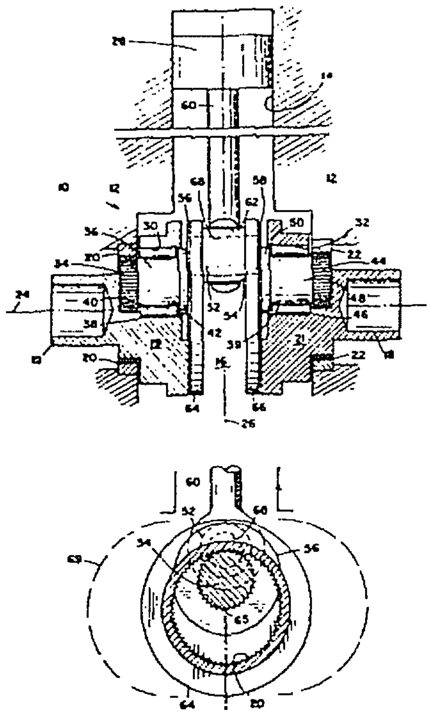


Fig. 5 Cherry one-cylinder engine: example of a "built-up shaft" (P34).

HISTORY

The hypocycloid engine concept is quite old. According to Freudenstein (2), the basic hypocycloid ("cardonic") motion has been known since the middle ages, with mechanical applications predating the industrial revolution. Over the years there have been many inventions and patents pertaining to its application to steam engines, internal combustion engines, and similar devices. The references at the end of the paper provide a fairly comprehensive summary of the design variations that have appeared over the years, and the limited development work that has been done. There is little likelihood, however, that we have uncovered all the pertinent articles, patents, and inventions.

Some of the hypocycloid engine designs have not used gearing. The concept will work without gears, but this introduces a high piston side load unless there is some other kinematic alternative to the gearing.

In (P3), a 1895 patent, the inventor mentions that the basic mechanism "has long been known and used". This particular patent involves having bearings on both sides of the working parts, and includes a countershaft, so that the device would be more suited for heavy work. The design was apparently used in a "pumping windmill", i.e., it converted rotation into reciprocating motion.

In looking over the various patents on the device, it appears that the principle was reinvented by a number of persons. There seem to be little difference in the claims made and apparently allowed in many of the patents.

One interesting variation is a design in which the cylinders (and therefore pistons) rotate as well as the crankshaft. This was used by Parsons in England in about 1877 for a steam engine (P1), and also by the Burlat Brothers of France in about 1905 for a rotary spark ignition engine (P5). Figure 6, taken from the Burlat patent, illustrates that design. The cylinder assembly rotates at half the speed of the crankshaft. No gearing is indicated, which would imply that the pistons have considerable side loads.

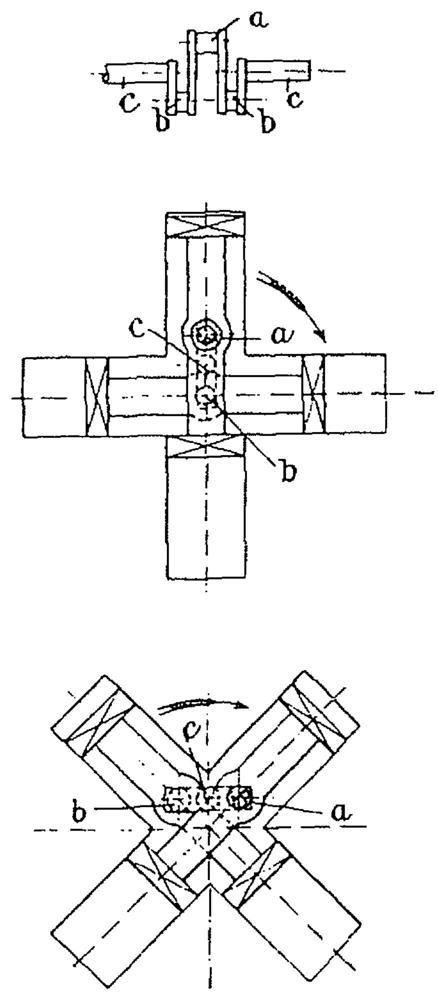


Fig. 6 Rotary hypocycloid engine design by the Burlat brothers (P5).

The invention of Huf (P35) uses a double eccentric system to provide the hypocycloidal straight-line motion. The Huf design eliminates the internal/pinion gear pair (even though other gearing is required), but tends to reduce the length of stroke that is practical without very large bearings for the eccentrics. A prototype engine based on the Huf patents was built a few years ago by Limbach Motorenbau of Konigswinter, West Germany, but there has apparently been no commercialization as of yet.

A recent project to dramatically increase the fuel economy of passenger cars was apparently based on the use of a geared hypocycloid engine (6). The claims made (145 mpg!) were never substantiated by independent testing, and we have not seen any recent reports on the project.

Ishida has patented several hypocycloid variations (P30, P31, P32). One of these uses an eccentric gear (a round gear with the shaft offset from its centerline), combined with other external gearing. He has done what is probably the most extensive developmental work to date on this concept, with experimental engines having been built and tested (7, 8, 9, 10, 11). He has used a fairly elaborate mathematical proof to demonstrate the capability of perfect balance (as was demonstrated graphically by Ifield (5), Fig. 2), and provided balance data for actual engines. Ishida uses the terms "crankshaft rotary motion" and "crankshaft planetary motion" to describe the two major variations that we refer to respectively by the terms "big bearing" and "built-up shaft".

Freudenstein (2) discusses a number of kinematic devices that provide motion equivalent to the hypocycloid mechanism (or cardanic motion, as he prefers to call it).

R.J. Ifield, who independently discovered the hypocycloid principle in the late 1930's, designed several versions of what he called the "harmonic crankshaft engine". He has shown remarkable ingenuity in handling the practical details, and in using the features of the concept for compact packaging. Two of the Ifield designs are illustrated in Figs. 4 and 7. The engine of Fig. 4 is a single-cylinder "big bearing" engine, and illustrates how a practical engine might be packaged. The engine of Fig. 7 might be called a "cube eight". It might also be considered to be a pair of flat four engines joined by a single countershaft. It is based on the "built-up shaft" principle, with two such crankshaft assemblies. Power can be taken off of either one of the crankshafts, or the countershaft, the latter option allowing a choice of effective engine output speeds by means of the countershaft gear dimensions. The inspiration provided by the preliminary designs of Ifield has been largely responsible for our interest in hypocycloid engines.

Working plans for a small hypocycloid steam engine model are given by (12).

EVALUATION OF THE HYPOCYCLOID ENGINE FOR PRACTICAL APPLICATION

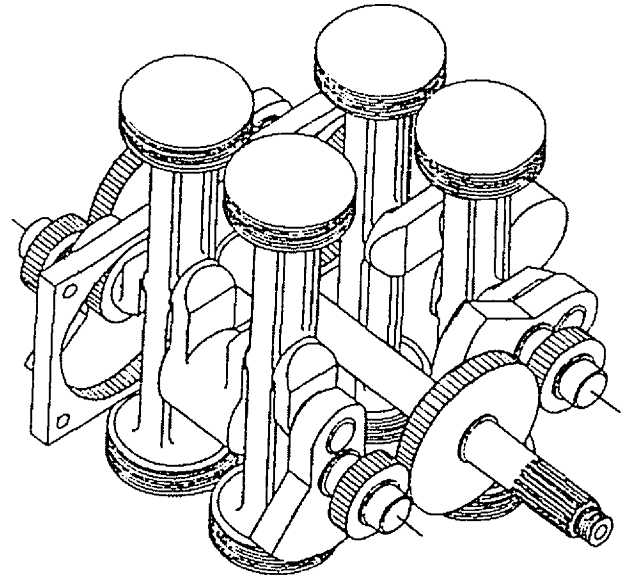
Most papers and patents concerning the hypocycloid engine have tended to not discuss the problems that might limit its practical commercial application. Here we will try to consider the potential weak points, and how they can be strengthened, or the design changed to minimize problems.

The discussion of this section will be limited to geared hypocycloid engine designs with the pinion/internal gear pair as illustrated in Fig. 1. Both major versions, the "big bearing" and "built-up shaft" designs will be considered.

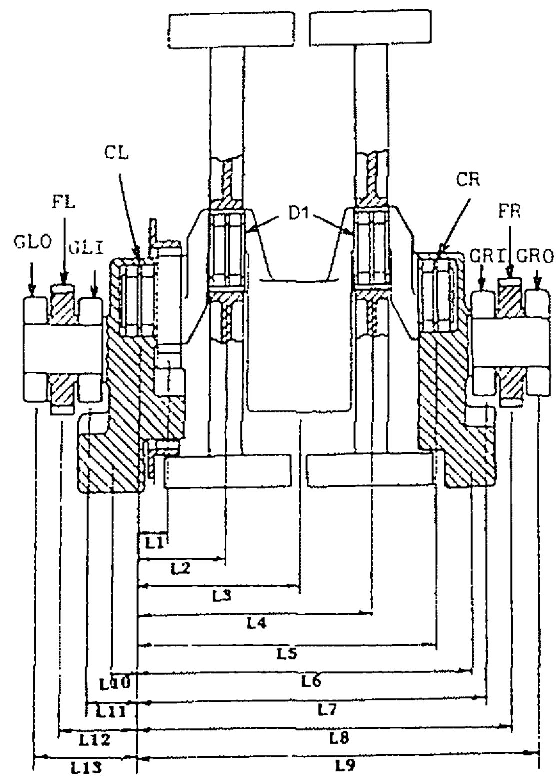
BREATHING AND COMBUSTION CONSIDERATIONS– The pure sinusoidal piston motion (assuming constant flywheel rpm) is different than that of a conventional engine with a connecting rod of finite length. These differences will have some effect on the breathing and combustion characteristics. The hypocycloid engine has a longer "dwell" period at TDC that tends to provide a little more time for combustion before the expansion stroke begins. At Bottom Dead Center there is less of a "dwell" period. These are factors worth considering, but probably of no great significance since the fuel system and/or spark timing and the valve timing need to be adjusted for any specific engine to match its characteristics. Note that engines are generally expected to run satisfactorily over a wide range of speeds, and speed variation is a more significant factor than piston "dwell" characteristics.

LOAD ANALYSIS – ONE -CYLINDER FOUR-STROKE SI ENGINE – In order to evaluate the practicality of the hypocycloid engine, it is important to obtain realistic values of forces on the bearings and the gear pair. These will provide information on the requirements for these components, and also provide the basis for stress analysis of the crankshaft system.

A single-cylinder four-stroke spark ignition hypocycloid engine of the "big bearing" type is considered first. A fairly detailed preliminary layout drawing of such an engine has been used to provide realistic dimensions and masses (Fig. 8). This engine has a stroke of 4 inches, and a bore of 3 inches. The weights of the basic components were calculated from scaling the drawing and assuming all parts to be steel except for the aluminum piston. The values of the balancing weights were determined by the balance requirements. There are two main bearings, and the journal bearing, rather than acting over the full length of the journal, is assumed to consist of two short bearings at each end, as illustrated. (Even if a full length journal bearing is to be used, it is still useful for analysis to assume two point bearing loads rather than a load distribution over the full-length bearing.)

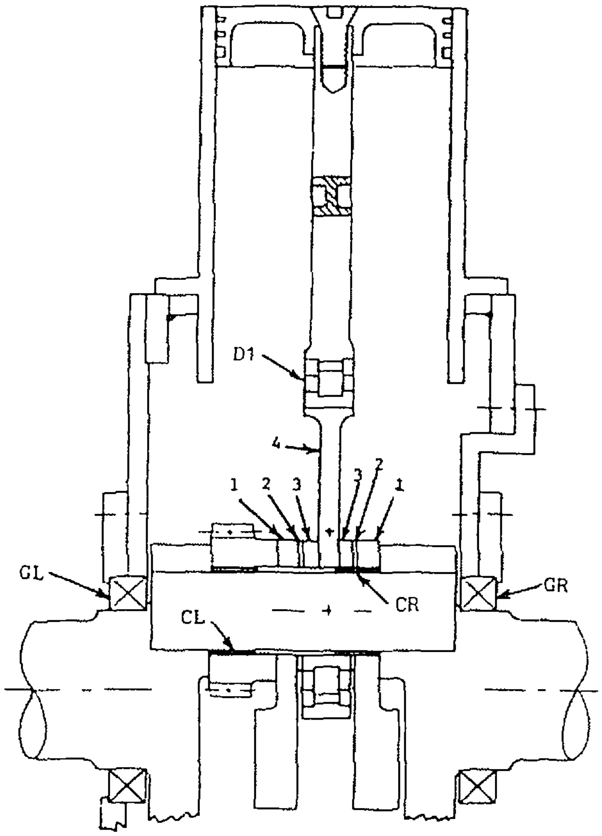


(a) General Layout.

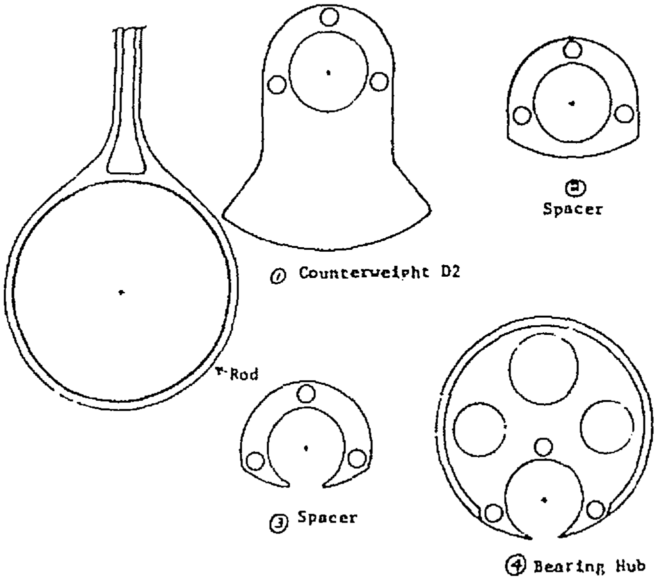


(b) Side view of hypocycloid crankshaft assembly, with significant dimensions.

Fig. 7 The Ifield eight-cylinder hypocycloid engine (5).



(a) Side view, to scale.



(b) Details of engine parts.

Fig. 8 One-cylinder engine design used as basis for load analysis.

Operation is assumed to take place at wide-open throttle and a speed of 6,000 rpm. The assumed gas loading for this operation is given by Fig. 9, taken from (13).

The load on the big bearing is due to the combination of gas loading (pressure times piston area) and the inertia load. Figure 10 gives the load as a function of crank angle.

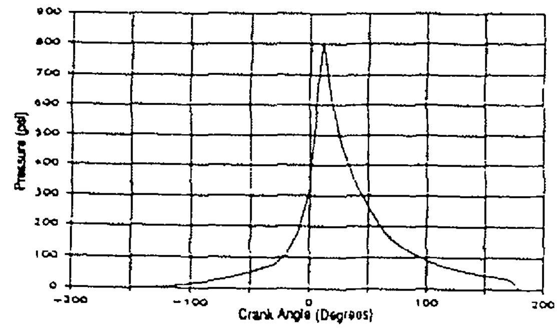


Fig. 9 Assumed wide-open-throttle combustion gas pressure for four-stroke spark ignition engine.

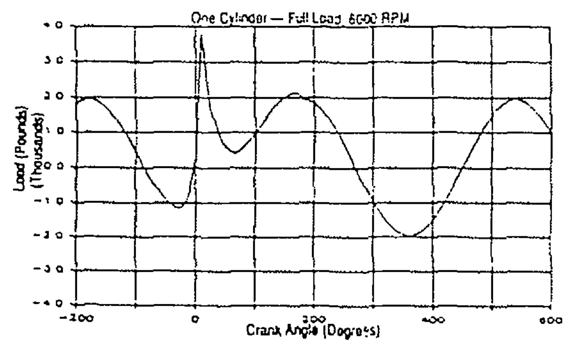


Fig. 10 Load at bearing D1.

The tangential gear tooth load is determined from the force and moment balance requirements of the center assembly that includes the "big bearing" and the pinion gear (14). If the engine is balanced (i.e., the effective masses at D1 and D2 are equal), we find

$$F_t = F_g \sin \theta$$

where

F_t = tangential gear tooth force
F_g = force from cylinder pressure
θ = crank angle

This result is an important one. The inertia loads cancel, thus *the gear load is independent of mass and speed, being a function of gas load only.* Even though the bearing load will increase with engine speed, the gear loading will not. The tangential gear load is plotted versus crank angle in Fig. 11. The load remains zero for the majority of the cycle, and a peak load of 1550 pounds is reached about 30° after TDC. Note that the same tooth is required to carry this load for every cycle. The gear load reverses just before TDC, but the magnitude of this reversal is not considered to be significant unless backlash is excessive. Note that the peak gear load does not occur when the gas load is a maximum, but somewhat later, due to the kinematics of the system.

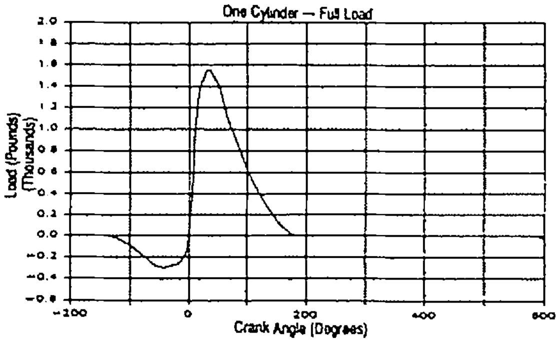
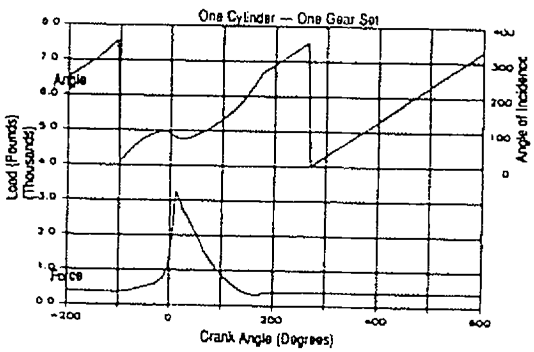
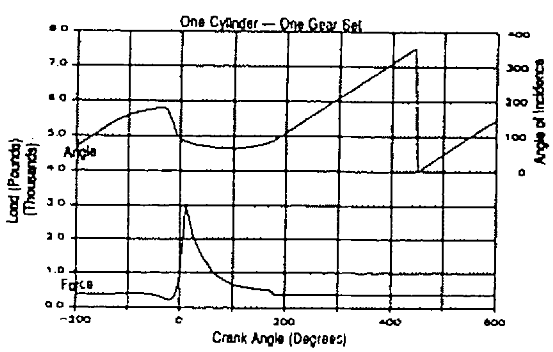


Fig. 11 Tangential gear load.

Further dynamic force analysis based on computer simulation has been done to obtain values of loads on the two main bearings and the two short journal bearings at the ends of the long journal. Details of the analysis are given in (14). The results are given in Figs. 12 and 13. The magnitudes and directions of the loads are given for two engine revolutions, since it is a four-stroke engine.

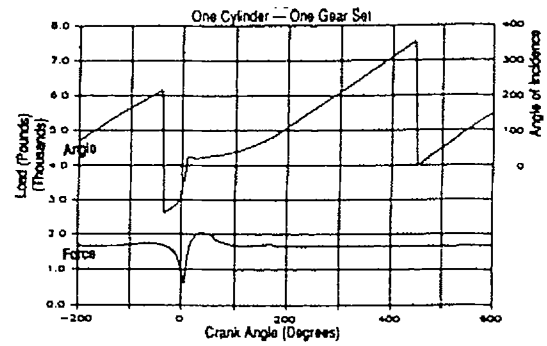


(a) Load on left main bearing GL.

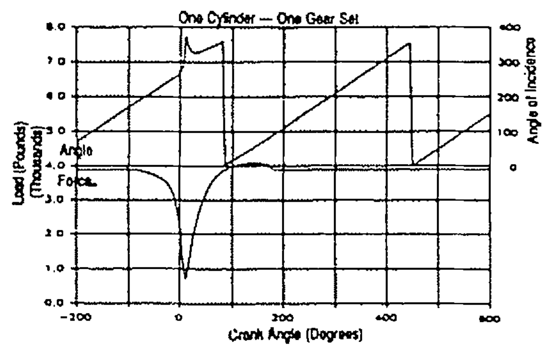


(b) Load on right main bearing GR.

Fig. 12 Main bearing loads.



(a) Load on left end bearing CL.



(b) Load on right end bearing CR.

Fig. 13 Journal bearing loads.

OTHER ENGINE CONFIGURATIONS – The loads on the gears and bearings have been studied for other engine configurations, with computer programs developed to handle all cases of interest. The computer program listings, in FORTRAN, are given in (14). The basic dimensions of the one-cylinder engine (Fig. 8) have been used as the guide for this; i.e., the multi-cylinder pistons are assumed to be the same size and weight, and the other components have weights as determined from a logical extension of the sizes and shapes of Fig. 8. The stroke is 4 inches and the bore 3 inches in all cases. (There would be some justification for increasing sizes and masses of components for diesel engines, based on their heavier loads, usual higher required reliability, etc. By keeping the weights the same, however, a useful direct comparison with the spark ignition engines is obtained.) All of the analyses have been based on wide-open throttle operation at 6,000 rpm. For the four-stroke SI engines, the same gas pressure curve is assumed for each cylinder (Fig. 9). For the two-stroke CI engines, a similar curve is used based on data for a typical diesel engine (15). The maximum pressure of this curve is 1275 psi, compared to 800 psi for the SI engines. The engines addressed in this phase of the study are listed in Table I.

TABLE I. Engine Configurations Simulated.

Big Bearing Engines

1. 4-stroke spark ignition engines
   a. 1 cylinder, 1 gear set
   b. 1 cylinder, 2 gear sets
   c. 2 cylinder opposed, 1 gear set
   d. 2 cylinder opposed, 2 gear sets
   e. 2 cylinder Vee, 1 gear set
   f. 2 cylinder Vee, 2 gear sets
   g. 4 cylinder cross, 1 gear set
   h. 4 cylinder cross, 2 gear sets.

2. 2-stroke compression ignition engines
   a. 1 cylinder, 1 gear set
   b. 1 cylinder, 2 gear sets
   c. 2 cylinder opposed, 1 gear set
   d. 2 cylinder opposed, 2 gear sets
   e. 2 cylinder Vee, 1 gear set
   f. 2 cylinder Vee, 2 gear sets
   g. 4 cylinder cross, 1 gear set
   h. 4 cylinder cross, 2 gear sets.

Built-up Shaft Engines — 4-stroke SI only

a. 2 cylinder in-line
b. 4 cylinder flat
c. 4 cylinder Vee
d. 8 cylinder flat
e. 8 cylinder double cross

One variation studied is the use of two internal /pinion gear sets, one on each end of the center piece. Here the assumption is made that they carry an equal load, i.e., distortion from metal compliance that would tend to vary gear tooth loading has not yet been considered. The use of two gear sets causes a slight change in spacing along the crank axis.

Note in Table I that the four-stroke compression ignition and the two-stroke spark ignition engines are not represented. In most cases, the general effect of changing from one pressure curve to the other should be fairly obvious when the results for the engines considered are compared.

The list of "built-up shaft" engines is shorter than that of "big bearing" ones primarily because the two-stroke cycle did not prove to be particularly advantageous. Also, the two gear set built up versions were not examined since the symmetry effect is well documented for the other engines.

The basic configurations of the two-cylinder opposed, two-cylinder Vee, and the four-cylinder cross engines are given in Fig. 3.

The results of the simulations for the "big bearing" four-stroke SI engines are summarized in Table II. Further data, including gear and bearing loads as functions of crank angle are provided in (14). It is interesting to note that adding more cylinders (up to 4) in the configurations studied has little effect on the gear and bearing loads, due to the various crank angles at which ignition occurs. Even though more gear teeth may be heavily loaded, the maximum load on each particular tooth is not increased until four cylinders are used, and then only by about 16%. This increase is because the four-cylinder four-stroke cross engine requires two cylinders to fire only 90° apart.

TABLE II. Peak Loads for "Big Bearing" Engines.

Four-Stroke Cycle, Spark Ignition

Engine	1 Cylinder	2 Opposed	2 Vee	4 Cross
One Gear Set:				
No. of Peaks on Same Tooth	1	2	1	2
Gear Load	1550	1550	1550	1800
Bearing D1	3800	3000	3800	3000
Bearing CL	2000	2200	2100	2500
Bearing CR	4000	4800	4200	4800
Bearing GL	3200	3300	3400	3500
Bearing GR	3000	3000	3300	3800
Two Gear Sets:				
No. of Peaks on Same Tooth	1	2	1	2
Gear Load	775	775	775	900
Bearing D1	3800	3000	3800	3000
Bearing CL	2900	3400	2900	3400
Bearing CR	2900	3400	2900	3400
Bearing GL	3100	3100	3300	3400
Bearing GR	3100	3100	3300	3400

The simulation results for the "big bearing" two-stroke CI engines are given in Table III. Note that gear and bearing loads are significantly increased due to the higher combustion pressures. One slight difference from the four-stroke SI engine is that the worst gear loading is for the two cylinder Vee engine rather than the four-cylinder cross engine, even though the difference is not great.

TABLE III. Peak Loads for "Big Bearing" Engines.

Two-Stroke Cycle, Compression Ignition

Engine	1 Cylinder	2 Opposed	2 Vee	4 Cross
One Gear Set:				
No. of Peaks on Same Tooth	1	2	1	4
Gear Load	2500	2500	2900	2500
Bearing D1	7000	6000	7000	6000
Bearing CL	3300	3300	4000	3300
Bearing CR	4100	4800	4100	2800
Bearing GL	4900	5000	5700	5400
Bearing GR	4900	4900	5300	5100
Two Gear Sets:				
No. of Peaks on Same Tooth	1	2	1	4
Gear Load	1250	1250	1450	1250
Bearing D1	7000	6000	7000	6000
Bearing CL	2900	3350	3200	2500
Bearing CR	2900	3350	3200	2500
Bearing GL	4900	4900	5400	5300
Bearing GR	4900	4900	5400	5300

The construction of the "built-up shaft" necessitates major changes in the load analysis, especially of the main bearings (which come in pairs). The Ifield two-throw shaft (5), illustrated to scale in Fig. 7b, was used as the basis for the dimensions needed for the analysis. Even though this shaft was designed for use in the eight-cylinder engine illustrated in Fig. 7a, it would be suitable for other versions with fairly obvious modifications.

The design of the end pieces deserves explanation. Typically, power is taken up at one end of an engine only. If that were done here, one of the end pieces would be essentially freewheeling in space because the two ends are not rigidly connected and the bearings do not absorb much energy. Therefore, a means must be provided for resisting torque on the opposite end. A countershaft is used as shown in the figure.

It should be further noted that the addition of a second internal gear set would not aid the end piece in resisting torque, as one might be inclined to think. In order to achieve hypocycloidal motion, the pinion would be mounted somewhere between the end bearings, C. In this position, the external load on the end pieces is not affected. It must be concluded that the countershaft is a necessary member in all the "built-up shaft" designs. For this reason, the one cylinder big bearing engine has fewer parts. The one-cylinder built-up shaft engine was not investigated.

The results of the simulations for the "built-up shaft" engines are presented in Table IV. Additional data are available in (14). Note that two "main bearings" (inner and outer) are required for each of the two end pieces. The placement of these bearings is left to the designer, but it should be noted that a small change in axial dimensions will have a significant effect on the load of the inner bearing. The load on the countershaft gears (FL and FR)is dependent on the gear pitch radius. This dimension was selected as 1.5 inches for this study. With this radius, the peak loads on the left gear are as high as 2400 lb., and are therefore not trivial.

TABLE IV. Peak Loads for "Built-up Shaft" Engines.

Four-Stroke Cycle, Spark Ignition

Engine	2 In-Line	4 Flat	4 Vee	8 Cross
No. of Peaks on Same Tooth	2	4	2	4
Gear Load	1550	1550	1900	1700
Bearing D1	3800	3000	3800	3000
Bearing CL	3700	4600	3700	4400
Bearing CR	4300	5300	4300	5000
Bearing GLI	8300	9800	9200	11400
Bearing GL0	4800	4800	4900	5800
Bearing GRI	7100	7500	7900	9000
Bearing GR0	3500	3400	3800	4000
Gear FL	2000	2000	2300	2400
Gear FR	1200	1300	1200	1300

EFFECTS OF SPEED AND LOAD – Since a single superior configuration for the hypocycloid engine did not surface in the studies as described above, it was decided to investigate the effects of speed and load on the one cylinder-engine, since this is the easiest version to visualize. Simulations were made at 1000, 3000 and 6000 rpm, using pressure traces developed from p-v diagrams (16) for part load and idle conditions. The gas pressure curves were assumed to be independent of speed and stroke.

This study produced no unexpected results. The hypocycloid gear loading is a function of gas pressure only (as explained earlier). At low speeds, the inertia of the moving parts decreases. This has a significant impact on the bearing loads, especially at D1 (and D2 where applicable). Recall that at high speed the inertia serves to decrease the load due to gas pressure. The loss of this effect means that the peak loads are highest at full load and low speed for bearing D1. This is also true in engines of conventional design.

With the exception of the bearing at D1, the most critical operation is at wide open throttle and maximum speed. For the most part, the bearing loads are similar to those found in conventional engines. The gear load is of greatest concern, and was the subject of further investigation.

GEAR DESIGN

Because of the required hypocycloid gear dimensions (diameter of the internal gear equal to the stroke, and the pinion diameter half this value), it is not easy to reduce the gear loads. For the engines studied (4 inch stroke, 3 inch bore), the loads are high enough for concern, especially with compression ignition diesels. it is therefore important to consider the gear loads in considerable detail.

There has been a great deal of work done over the years on determining allowable gear loads, and standard procedures have been developed. Unfortunately, little of this work has been applicable to internal gear/pinion sets. The most sophisticated set of procedures, the AGMA standards (17), is for spur and helical gears only.

Evidently, very little has been published on the design of nonstandard gears in the American Literature, and even less has been disclosed regarding internal gear sets. H.H. Mabi attributes this to the use of finite element methods in the gear industry (18). The same is apparently true of the international community. For example, at least two Japanese technical papers on the bending strength of internal gears rely entirely on the use of finite elements and photoelasticity (19,20). Unfortunately, such papers do not contribute much in the way of quantitative information.

Researchers working with the organization DIN and GOST have also published studies on stress in internal gear sets. These standards use different notation from American practice, making the paper difficult to understand. Moreover, the methods do not agree with each other, nor with the procedures used by the AGMA. Due to time restrictions, and the above reasons, these studies were not used, although they may be useful for future work.

In line with the time constraints on the study, our analysis was done in three stages (14):

1. A simplistic study using spur gear equations, to obtain a rough idea of the stresses involved.
2. A study of available modifications that would tend to equalize the strength of the internal gear and pinion (and therefore raise the strength of the weaker member – the pinion). Shifted profile techniques were emphasized.
3. A detailed study of alternative gear sets. After analyzing six alternatives, the design that provided the best results was chosen.

The detail study of the bending stresses was based on two approaches, with stresses calculated by both methods.

One approach was to determine "pseudo-Lewis" form factors by graphical means, and to use these in the classical analysis technique. The second approach was to use the non-dimensional stress data provided by Andersson from his finite element studies on internal gears (21).

The analysis of the contact stresses was somewhat simpler. Two procedures were again used, one based on the AGMA standards (17), and the other on the data of Andersson (21). The contact stresses were made less severe by the fact that the internal gear involute profile is concave.

During the analysis, it was found important to insure that there was no mechanical interference between the two gears for any of the trial sets of parameters.

The chosen gear parameters (i.e., the parameters of the best pair found with a limited set of trials) are presented in Table V. The contact stress values are well below the typical working value of 200,000 psi used for high quality gears. The bending stress values, on the other hand were found to be slightly above the typical working value of 70,000 psi with the graphical analysis, but slightly below from the analysis based on the Andersson data. Therefore, to insure that the strength is adequate, it may be appropriate to increase the face width, or to increase the stroke/bore ratio.

TABLE V. Parameters of the Selected Internal/Pinion Gear Design (Dimensions in inches).

		Pinion	Gear
Number of teeth		20	40
Face width		0.5	0.5
Operating pitch radius		1.0	2.0
Base radius		0.989	1.978
Radius of curvature at point of calculation		1.024	2.050
Outside radius		1.264	2.291
Diametral pitch		9.5	9.5
Base pitch		0.311	0.311
Normal profile angle, standard cutter		20°	20°
Operating transverse pressure angle		8.5°	8.5°
Transverse contact ratio		1.273	1.273
Profile shift		0.980	0.600
Stresses (based on 1,500 lb tangential load):			
Contact Stress, psi	AGMA	102,920	---
	Hertz	88,600	---
Bending Stress, psi	Graphical	74,020	75,200
	Andersson	61,460	60,377

The above results should be considered preliminary. They can serve as the starting point for a detailed finite element analysis, which should include consideration of the effect of the high contact ratio inherent in an internal/pinion gear pair.

It would also be worthwhile to consider tooth profiles other than involute, e.g., the "Wildhaver-Novikov" gears.

EFFECTS OF BORE AND STROKE – If we assume a given combustion pressure curve, the gear loading will vary with the square of the bore. On the other hand, it can be shown to be independent of stroke. An increase in stroke, however, provides for gears of larger diameters, which, for the same geometry and numbers of teeth, will decrease the stress values. The gear strength characteristics might therefore be considered to be highly variable for different combinations of bores and strokes. Our analysis, however, has produced an interesting result: With logical and reasonable assumptions, if the face width, gear load, diametral pitch and tool parameters are all chosen as functions of the pitch radius, then gears of various sizes will have the same stresses for a given number of teeth. This can be extended to show that hypocycloid engines of similar design having a given bore/stroke ratio will have equal gear stresses. This means that there is no significant size factor in terms of gear stresses. Once the optimal geometry in terms of bending strength is determined for one engine size, the best tooth profile for any other stroke is known. The maximum capacity can then be tailored to the engine by changing the face width of the gear. If the face width is arbitrarily set equal to the pinion pitch radius, the maximum usable bore-to-stroke ratio is about 3:4, as was chosen for the one-cylinder design that has been analyzed.

SUMMARY AND CONCLUSIONS

The geared hypocycloid engine has been shown to be a promising concept for providing perfect balance and reducing mechanical friction. It appears particularly attractive for single-cylinder applications where balance is important, and for advanced engine concepts in which piston/cylinder lubrication and friction are matters of concern. More research and development are needed, but no new basic technology is required for the concept.

The load analysis has produced no optimal configuration. To the contrary, it has shown that there are a number of configurations that appear equally attractive load-wise, ranging from one to eight cylinders. The bearing loads appear to be in line with those of conventional engines, and therefore design of the hypocycloid equivalent of a crankshaft should not present any significant problems.

Preliminary analysis of the gear loading has shown this to be the critical part of the design concept. Although more study is called for, both theoretical and experimental, the tentative conclusion is that gear strength would probably be adequate for any size engine having a bore/stroke ratio of about 0.75 or less. Up to four cylinders can be used with a single internal gear/pinion set without significantly increasing the effective gear loading. It is also significant that the gear loading is independent of engine speed.

With continuing research, development, and resulting design refinements, it is likely that practical hypocycloid engines will result that are superior to conventional engines for a number of applications.

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PATENTS

	Patent No.	Name	Year	Subject
P1.	British 2344	Parsons	1877	Steam engine with rotating cylinders.
P2.	399,492	Burke	1889	Hypo mechanism using internal gear.
P3.	563,860	Ziegler	1895	Windmill pump; adds countershaft.
P4.	684,745	Carey	1901	Hypo concept using guides, not gears.
P5.	Br. 23,079	Burlat	1905	IC engine with rotating cylinders.
P6.	947,233	Hammond	1910	Device using eccentrics & guide blocks.
P7.	1,056,746	Pitts	1913	Internally geared hypo device; adds counter-weights; big bearings & 3-piece shafts.
P8.	1,073,656	Barker	1913	Uses Scotch Yoke crank; patent on gas passages.
P9.	1,090,647	Pitts	1914	Uses hypo device; patent on gas passages.
P10.	1,579,083	Collins	1926	Internally geared hypo device.
P11.	1,756,915	Short	1926	Geared hypo balancing device.
P12.	1,761,429	Dean	1926	Both hypo and epicyclic devices; patent covers orientation of wrist pin.
P13.	1,789,207	Williams	1931	Device using eccentrics and guide blocks with a geared countershaft.
P14.	1,867,981	Mudd	1932	Hypo type device using nesting bearings; gears or equivalent not shown.
P15.	1,983,901	Frampton	1933	3-piece shaft with guides; misses the countershaft.
P16.	2,132,595	Bancroft	1938	See Reissue 21,583
P17.	Re 21,583	Bancroft	1940	Combines piston and rotary engines; piston half uses hypo drive with guides; uses elliptical path.
P18.	2,223,100	Foster	1940	Engines using geared hypo and epicyclic drives.
P19.	2,271,766	Huebotter	1942	Engine using "well known" hypo concept; claims counterweights.
P20.	2,506,736	Oschwald	1950	Geared hypo mechanism for power tools.
P21.	2,650,543	Pauget	1953	Hypo device in radial piston pump.

P22.	2,775,128	Young	1956	Hypo device using guides; pin inside pitch circle limits stroke.
P23.	2,844,040	Bancroft	1958	Similar to Re 21,583
P24.	2,936,632	Palmer	1960	Device using eccentrics and guides.
P25.	3,175,544	Hughes	1965	Scotch yoke mechanism.
P26.	3,258,992	Hittell	1966	Engine using eccentrics, guides and counterweights.
P27.	3,277,743	Kell	1966	Mentions geared hypo with countershaft for use with double-acting pistons; claims guided version.
P28.	3,329,134	Llewellyn	1967	Geared hypo engine with counterweights plus claims on carburetion and charge pistons..
P29.	3,386,429	Trammell	1968	Engine with eccentrics and guides
P30.	3,528,319	Ishida	1970	Planetary motion system.
P31.	3,563,222	Ishida	1971	Rotary motion system.
P32.	3,563,223	Ishida	1971	Engines as above but with double-acting pistons.
P33.	3,626,786	Kinoshita	1971	Geared hypo mechanism; claims balancing method.
P34.	3,791,227	Cherry	1974	Geared hypo engine; claims special counterweights.
P35.	4,237,741	Huf	1980	Unusual geared hypo device aimed at eliminating overloaded gears.

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