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West Virginia Univ.

International Congress and Exposition
Detroit, Michigan
February 23-27, 1987

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Printed in U.S.A.

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The Stiller-Smith Engine: Floating Gear Analysis

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ABSTRACT

The Stiller-Smith Engine employs a non-standard gear train and as such requires a closer examination of the design and sizing of the gears. To accomplish this the motion of the Stiller-Smith gear train will be compared to more familiar arrangements. The results of a kinematic and dynamic analysis will introduce the irregular forces that the gears are subjected to.

The "floating" or "trammel" gear will be examined more closely, first stochastically and then with finite element analysis. This will pinpoint high stress concentrations on the gear and where they occur during the engine cycle. The configuration considered will be one with: an output shaft, negligible idler gear forces, and floating gear pins that are part of the connecting rods rather than the floating gear. Various loading techniques will be discussed with possible ramifications of each.

THE UNCONVENTIONAL BEHAVIOR of the gears in the Stiller-Smith Engine is soon apparent when compared against standard centrally mounted gears. Gears must meet a "no-slip" condition to operate properly. For round centrally mounted gears this is simply $r_1 \theta_1 = r_2 \theta_2$ which says the arc length ($r\theta$) measured from any contact point of position one to any other contact point of position two must match for each gear. This may seem trivial but for elliptical gears, or any other nonstandard shape the mathematics is far from simple.

Nonstandard gears are often used for cases where a design requires a varying angular output differing from the angular input. This is often accomplished with eccentrically shaped gears. However due to the cost of these types of gears, eccentrically mounted round gears are normally used even though in an ideal sense they don't obey the fundamental laws of gearing. For example, if perfectly meshed, with matching arc lengths, the radius from the pitch point to the shafts where the gears are mounted must maintain a fixed distance. But for an eccentric case this radius will change thus altering the radius and the no-slip conditions at the teeth. This problem is often corrected by either altering the teeth of the gears, to allow more play, or by mounting the gears with greater backlash, thereby working the tips of the teeth harder.

The Stiller-Smith Mechanism, on the other hand, successfully employs eccentrically mounted gears with large offset axials. The meshing problem is alleviated due to the motion of the floating gear which provides a movement of the axial points that re-establishes the $r\theta$ equals a constant condition.

MOTION DESCRIPTION

To understand the Stiller-Smith Mechanism first consider the motion of the double-cross slider. Fig. 1 is a pictorial representation illustrating that the center point of the

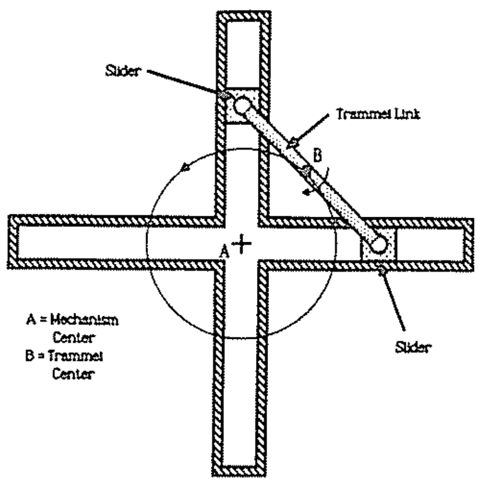

Fig. 1 The Double Cross-Slider

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cross-slider link travels in a circular path. Previous attemps to utilize a double-cross slider took advantage of this translational component. Fig. 2 further indicates how the area around point B rotates in a direction counter to that of the translation. It is this rotational phenomenon which is harnessed by the Stiller-Smith Mechanism. Note that all other points on the connecting "bar" travel in an ellipse.

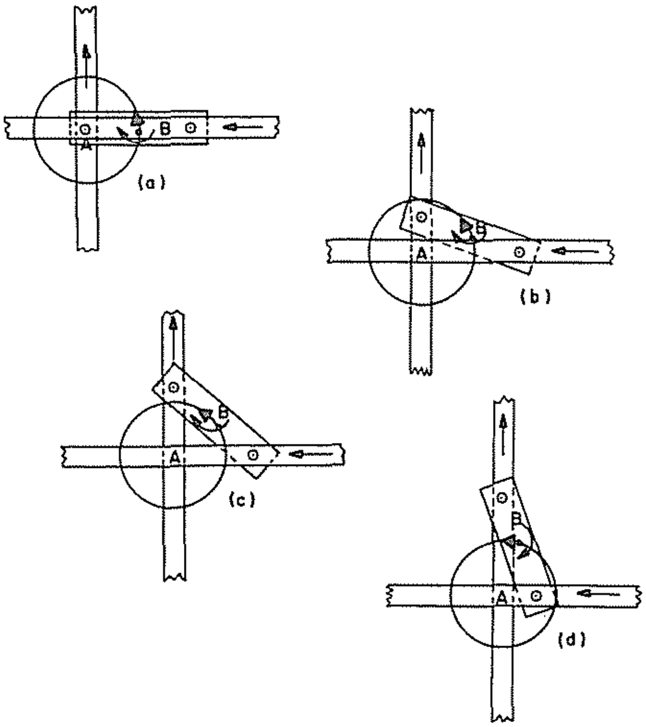

Fig. 2 The Double Cross-Slider Motion

The "bar" is now replaced with a twin axle floating gear. The center of this gear travels in a circle while the gear rotates in the opposite direction. Both of these motions occur at the same angular velocity which permits meshing with an offset gear while maintaining the no-slip condition. Figure 3 illustrates how the trammel may be meshed with two output gears the second of which is used as an idler gear for balancing purposes. The angular output of this train is constant, unlike an eccentric-gear/eccentric-gear coupling.

Each slider is now expanded to be a connecting rod with a piston on each end. Forces on the trammel gear can be seen to be: the pressure forces of the combustion process, the inertia of the connecting rod assemblies, the inertia of the gear itself, the tooth forces and the reaction forces of each component interacting with the others. Fig. 4 illustrates the use of the Stiller-Smith Mechanism in an actual prototype. It was in the design of this prototype that attention was drawn to the nonstandard nature of the floating-gear design.

GEARING OVERVIEW

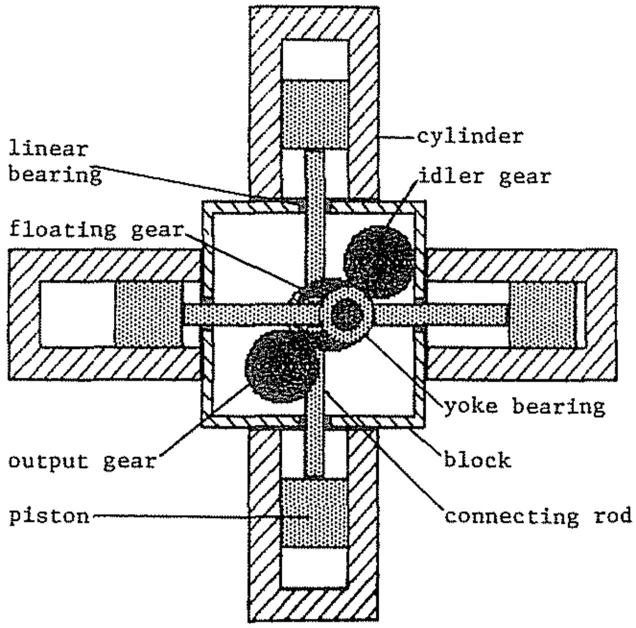

Fig. 3 Gear Train of the Stiller-Smith Mechaism

Forces are reacted on the floating gear primarily at three locations: the two pin positions and the teeth meshed with the output gear. Stochastically it can be seen that there are two areas of concern for stress concentrations. Area A, (Fig. 5) defined as the area between the tooth force and the pins, is seen to be critical when the gear is positioned such that the distance between the tooth force and either of the pins is at a minumum. This occurs at two places during gear rotation, one for each pin. Also of concern is area B which

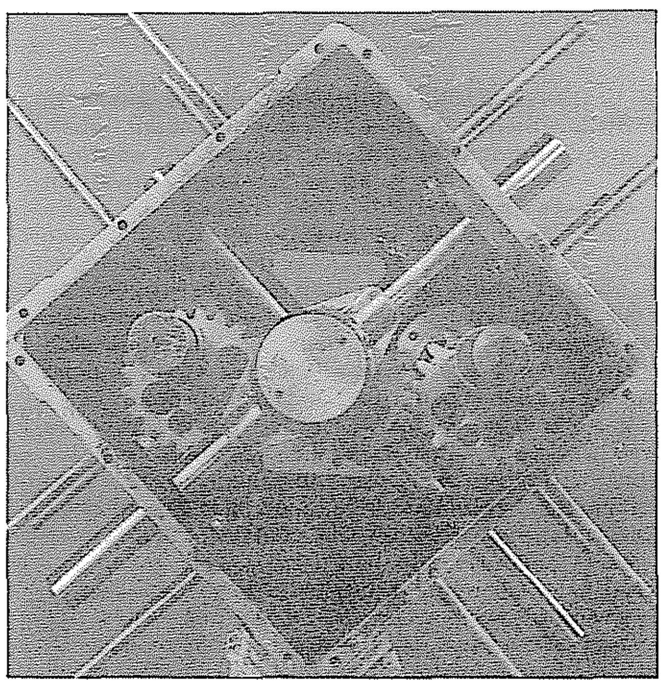

Fig. 4 Prototype of the Stiller-Smith Engine

is defined as the area between the pins. High stresses will occur here whenever the two pin forces are opposite in direction either directly opposed or skewed in nature. Due to the nature of the loadings, section B will not be in direct tension when used as an engine, however other applications of the mechanism may alter this situation. Analysis reported here will confine itself primarily to illustrations of one or both of these high stress cases.

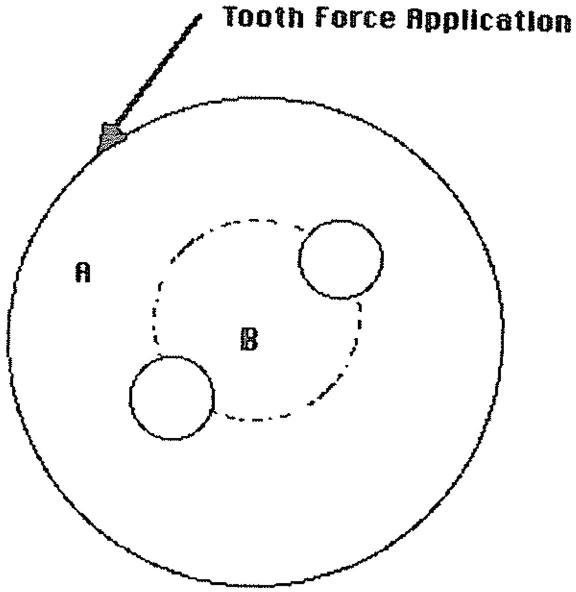

Fig 5 Forces Overview of the Trammel Gear

When evaluating the gear train it can be considered as a mechanism in general or as an engine in particular. As a mechanism, constant loads would be applied at the pins and teeth in various positions of the mechanism. This would indicate the high stress areas of the gear, and where over a cycle they occur. When considered specifically as an engine the actual combustion pressure forces should be used which would result in different stresses due to the cyclic nature of the pressure profile. This would be especially true because of the pronounced pressure spike near top dead center of a piston.

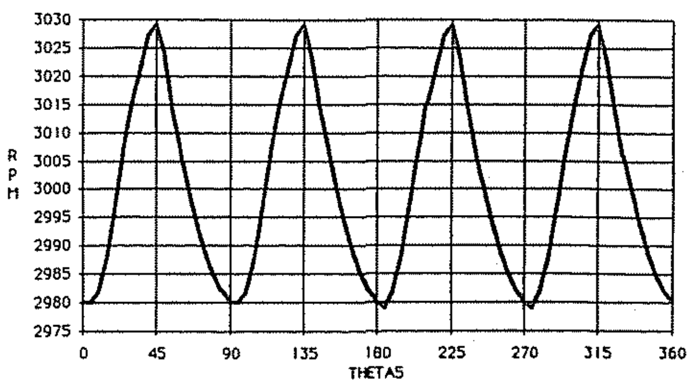

Fig. 6 Steady-State Engine Speed/Balanced Stiller-Smith Engine.

To meet the immediate design needs of this project the latter engine case was of primary interest. This requires that reasonable force data be available for use in loading the finite element models. Kinematic[1] and Dynamic[2] analyses were conducted and their results incorporated into a computer simulation model. This model did not assume constant angular velocity but instead let the simulation reach a steady state condition between pressure (driving) force and torque (retarding). Fig. 6 demonstrates this steady state condition for a computer run at about 3000 rpm.

As this is a typical run at normal operating conditions it will be used further for this analysis. Further data for this computer run are shown in Table 1. These numbers are representative of one of the early prototypes and as such are not optimal.

TABLE 1
STILLER-SMITH ENGINE PROTOTYPE-BALANCED CONFIGURATION

1.	Friction Coefficient	0.05
2.	Engine Center to Linear Bearing	4.5 in
3.	Engine Stroke	3.38 in
4.	Gear Weight	2.5 lbs
5.	Piston Assembly Weight	7.6 lbs
6.	Output Assembly Weight	45.0 lbs
7.	Output Assembly Inertia	5.892 x 10^{-3}slug-ft^2
8.	Position of Idler Assembly	225 deg
9.	Idler Assembly Weight	17.0 lbs
10.	Position of Idler Assembly	3.764 x 10^{-3}slug-ft^2
11.	Position of Idler Assembly	45 deg
12.	Gear Pressure Angle	20 deg
13.	Output Counterweight Distance	.13051 in
14.	Output Counterweight Angle	180 deg
15.	Idler Counterweight Distance	.5872 in
16.	Idler Counterweight Angle	180 deg
17.	Gear Radius	1.75 in
18.	Engine Block Weight	81.0 lbs
19.	Engine Position	Horizontal
20.	Piston Displacement	500 cc
21.	Trammel Gear Inertia	9.614 x 10^{-4}slug-ft^2
22.	Distance Between Piston Planes	.42 in

Before examining the force outputs from the computer run, first examine Fig. 7 which demonstrates the erratic nature of the pressure inputs. Theoretical pressure points were provided in 10 degree increments and linear interpolation between points was used. The discontinuity of this curvature will yield similar discontinuities in the reaction forces that are outputed. These curves are for two-cycle operation and the opening of the exhaust port is also observable as a discontinuity in the curvature.

Figs. 8, 9, and 10 represent the reaction forces seen by the floating gear. There is a large force loading at about 225° of the imaginary crank angle. This reference vector defines the center of the floating gear with

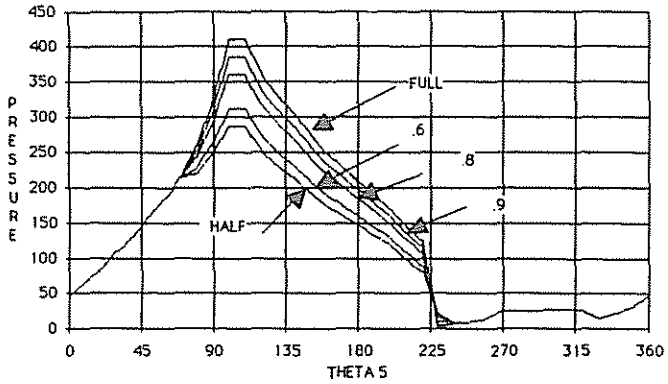

Fig. 7 Pressure Curves at Various Throttle Settings

respect to the center of the engine. Figure 11 shows the floating gear in this position and the direction of the three forces acting upon it. This is an especially good position to model as it matches the stochastic criteria.

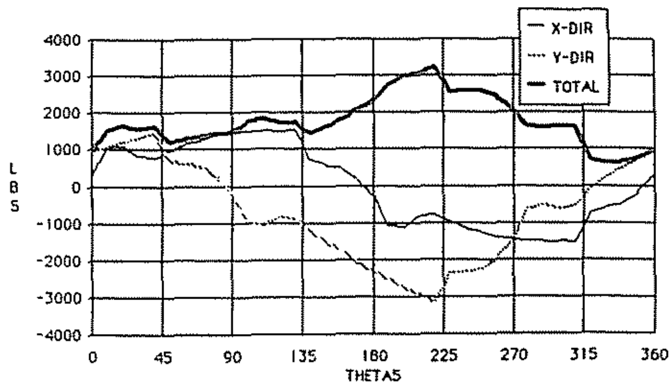

Fig. 8 X-Direction Floating Gear Bearing Forces for a Balanced Engine

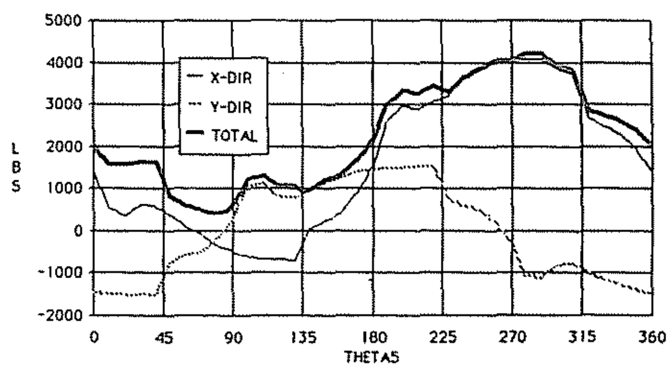

Fig. 9 Y-Direction Floating Gear Bearing Forces for a Balanced Engine

FINITE ELEMENT MODELLING

Supertab by SDRC, which is part of their I-DEAS package, was the program used for analyzing the floating gear. Many modelling options needed to be considered to produce valid results. Geometry, meshing type, and load/restraint sets were primary issues.

The choices of geometries to represent the gear are numerous and many are valid depending on the areas of concern. The worst case for engine work was considered to be when the pins were part of the connecting

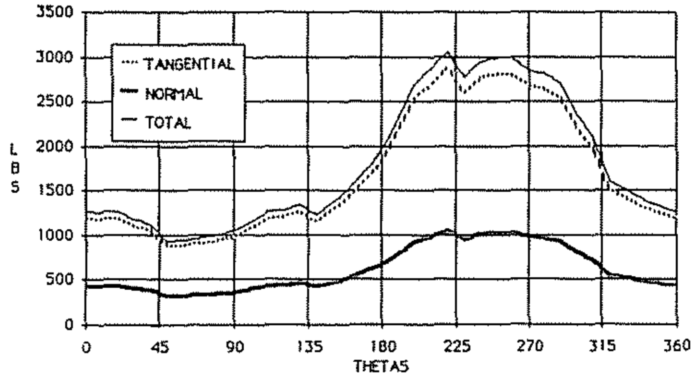

Fig. 10 Output Gear Teeth Forces for a Balanced Engine

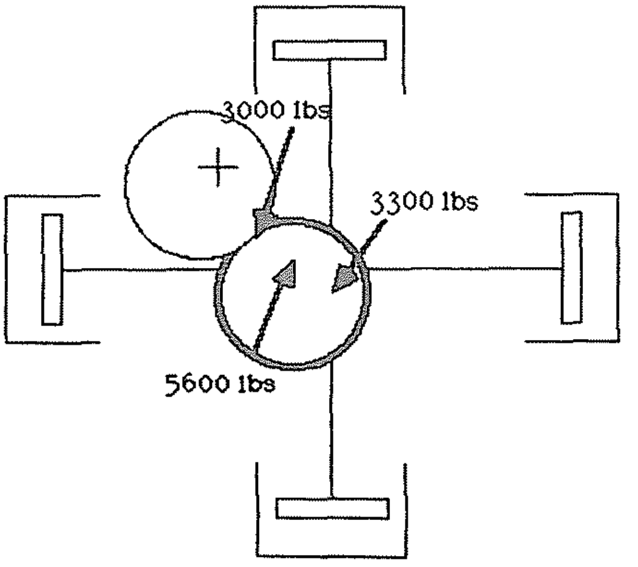

Fig. 11 Representation of Forces on Floating Gear at 225°

rods and therefore there were "holes" in the floating gear. As the area between the holes (Area B) is important a simple point model for the holes was deemed inadequate. Tooth representation was another key geometric decision. First approximaions represented the gear as a circle but this does not fully illustrate the problem between deep tooth cuts and the close proximity of either of the holes. It is also believed that stresses in the teeth will be different from standard gear teeth due to the predominate three force nature of the floating gear.

With this in mind it was decided to give special attention to the teeth under load to determine the extent of the change in stress profiles as compared with standard centrally mounted gears. The final geometry consisted of trapezoidal teeth and pin holes of a size found in the first prototype.

Supertab provides three basic means of creating elements: enter the actual model positions in by hand (and subsequent copying, reflecting, and/or interpolating), mapped mesh generation, and triquamesh or free mesh generation. Fig. 12 is an example of free meshed, linear, quadrilatteral, shell elements. It was felt that a better control over the mesh was permitted using mapped mesh generation and was therefore used in the more detailed models (Fig. 13)

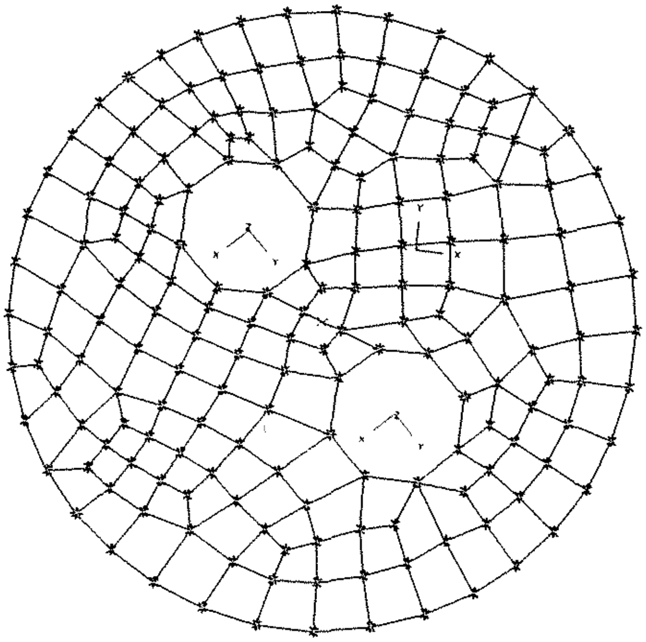

Fig. 12 Linear Element Freemesh of "Round" Gear.

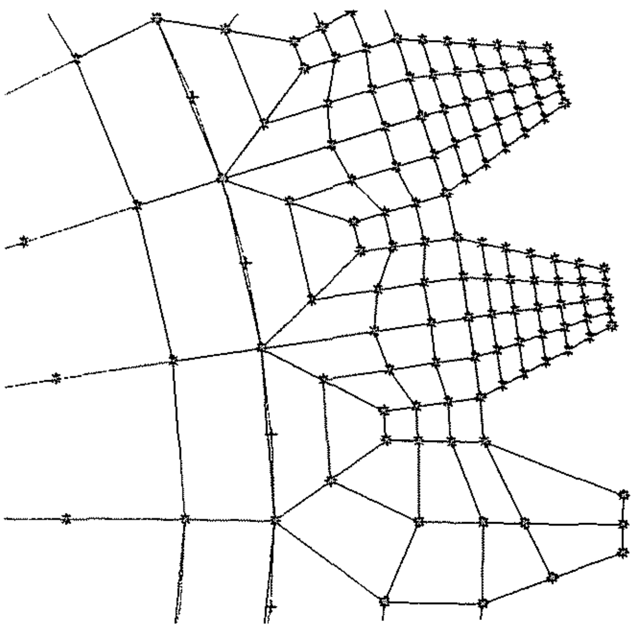

Fig. 13 Mapped Mesh Representation of a Gear with Trapezoidal Teeth.

Applying loads and restraint sets which adequately modelled the gear, proved to be the most difficult task. It was first thought to restrain the holes in various ways while applying inertial and pressure forces and letting the analysis solution compute the reaction loads. But this would restrict the model from deforming in the area between the holes. Restraints of the tooth are acceptable but not enough degrees of freedom can be specified without altering the actual deformations occuring by restricting rotation. Final load sets required the use of kinematic degrees of freedom constraints which restrict rigid body motions but not deformation.

PROBLEM ANALYSIS

The purpose of this analysis is to locate the high stress regions and when they occur during a gear rotation. It is suspected that region-A will endure high stresses at a crank position of 225°. It is not immediately seen, however, where the highest region-B stresses will occur. To give a quantative understanding of where the high stresses occur, a simplified model will be subjected to the computed reaction forces at eight different positions. A simple model may be used because of the desire only to locate the areas and/or trends of the primary concern for a later more detailed analysis. The 45° intervals were believed to be of sufficient resolution for this purpose.

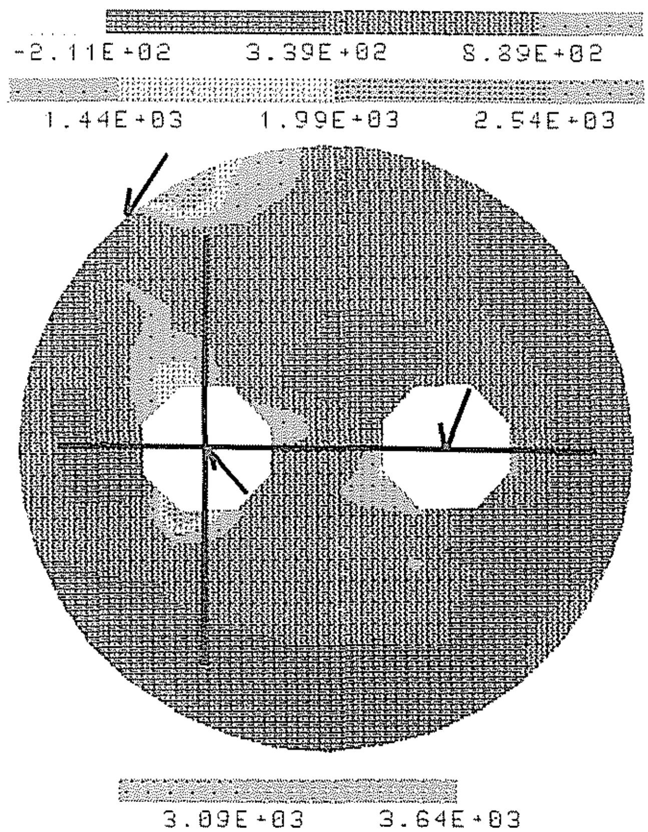

Fig. 14: Finite Element Analysis of Round Freemeshed Gear as Stress Contours for "Crank" Angle 0°.

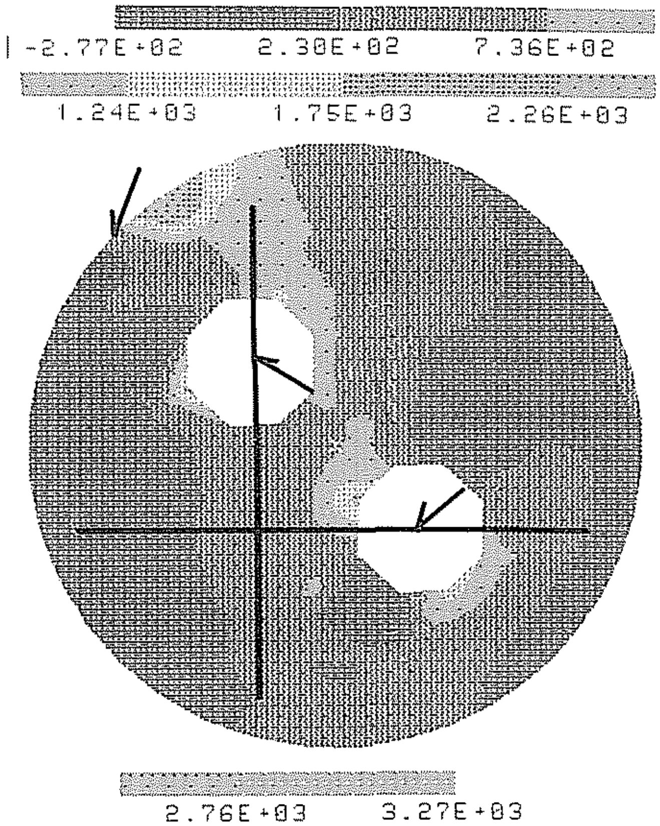

Fig. 15: Finite Element Analysis of Round Freemeshed Gear as Stress Contours for "Crank" Angle 45°.

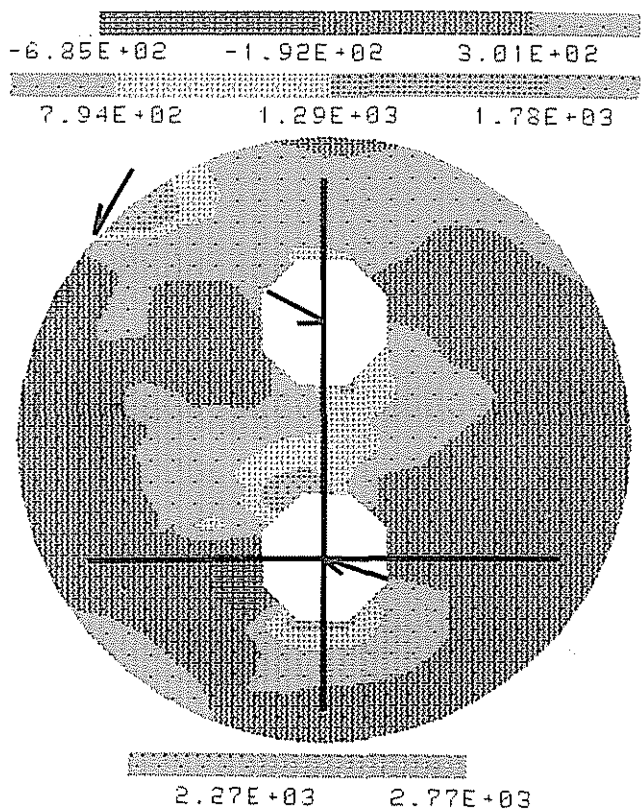

Fig. 16: Finite Element Analysis of Round Freemeshed Gear as Stress Contours for "Crank" Angle 90°.

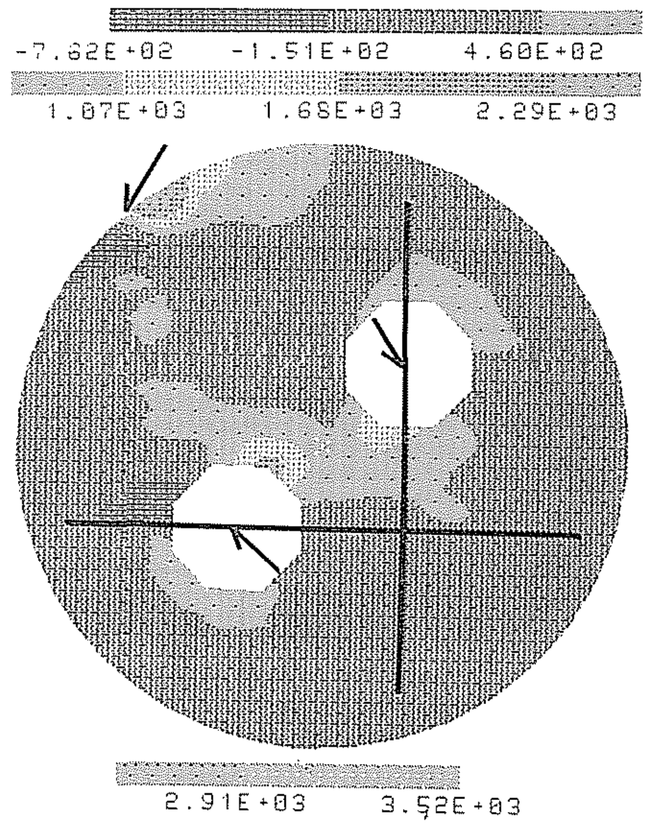

Fig. 17: Finite Element Analysis of Round Freemeshed Gear as Stress Contours for "Crank" Angle 135°.

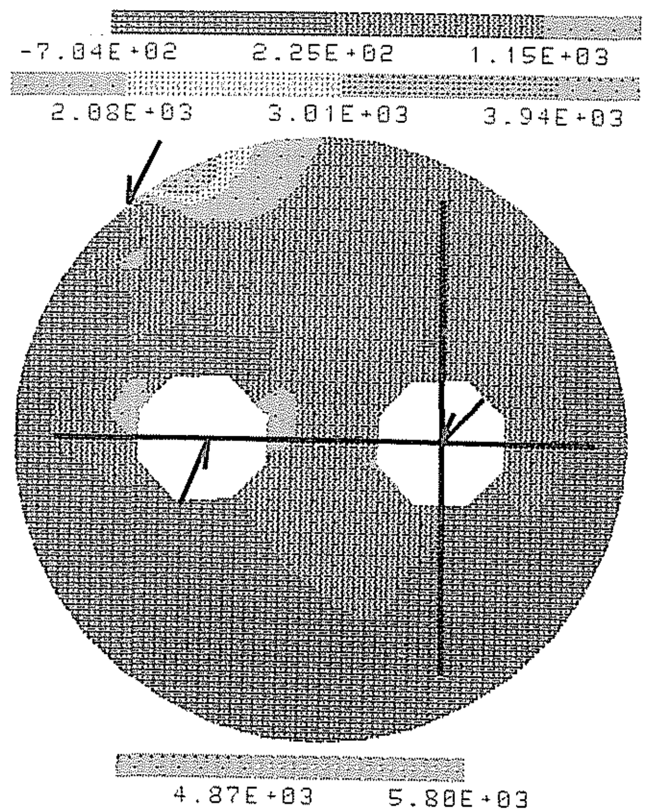

Fig. 18: Finite Element Analysis of Round Freemeshed Gear as Stress Contours for "Crank" Angle 180°.

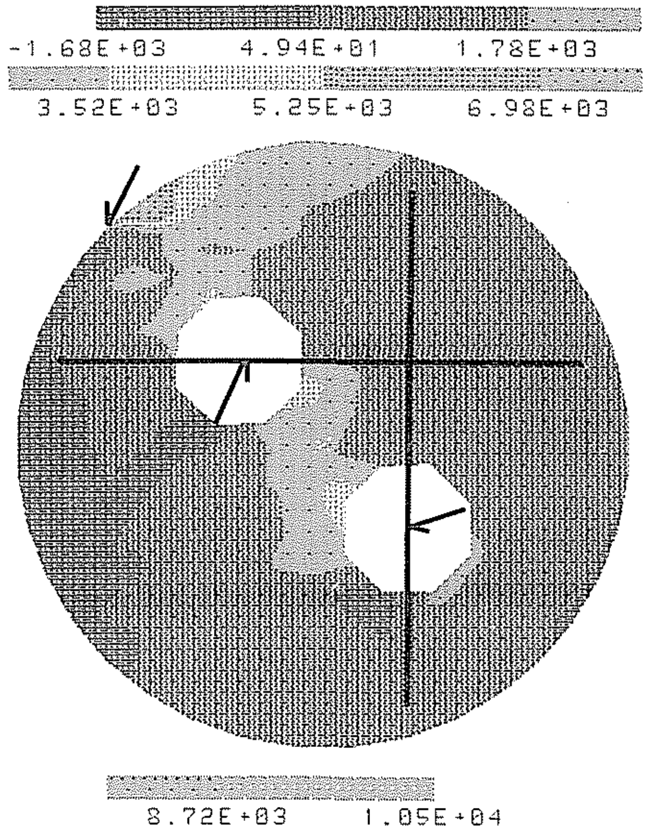

Fig. 19: Finite Element Analysis of Round Freemeshed Gear as Stress Contours for "Crank" Angle 225°.

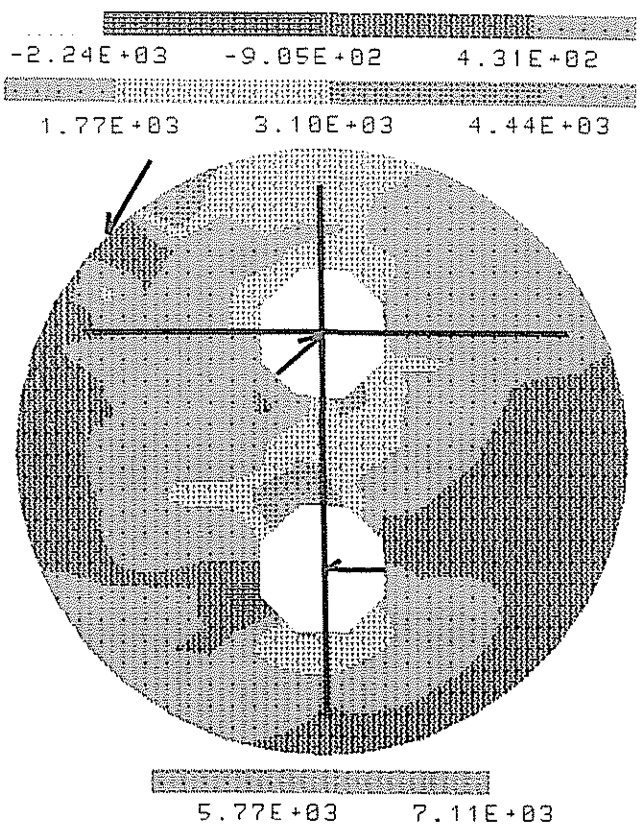

Fig. 20: Finite Element Analysis of Round Freemeshed Gear as Stress Contours for "Crank" Angle 270°.

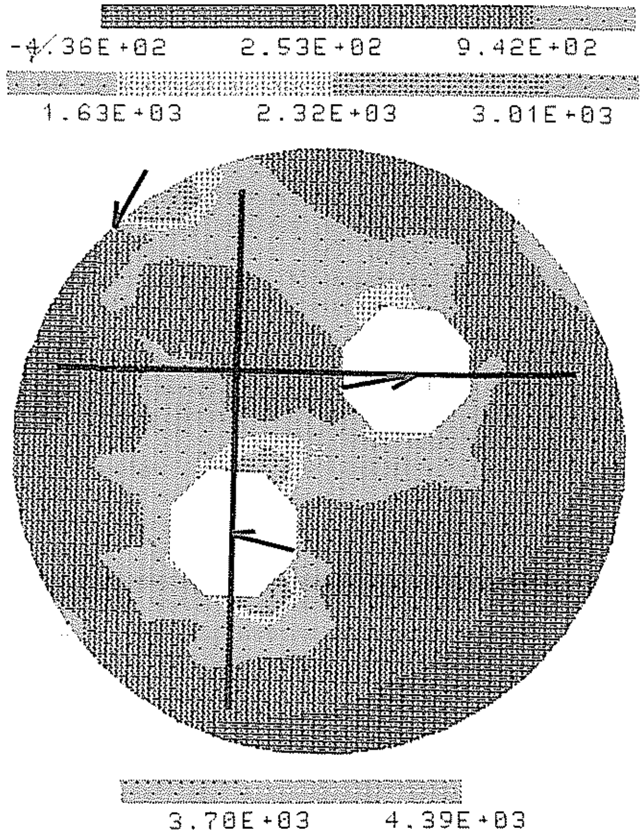

Fig. 21: Finite Element Analysis of Round Freemeshed Gear as Stress Contours for "Crank" Angle 315°.

A round gear with a sparse free mesh of elements was used. Global element size was about 0.4 inches. The material properties were chosen to be about that of steel and were evenly distributed over the gear so case hardening, etc. are not considered. The choice of a circular representation will give different stress values than if teeth were present but for now the actual stresses are not the primary concern.

The loadings at the pins were also of a simplified nature. They have been applied as distributed loads but each component (X and Y) of the forces was distributed separately and the term distributed is used loosely as it is over 2 nodes. A more refined mesh in future analysis will permit a more representative application of load.

Thin shell linear quadrilatteral elements were chosen with a uniform thickness of .75 inches. Of the supported elements in Supertab these most closely met our needs. Future models should employ parabolic elements to better fit the curvature of the holes or at least much more refined elements should be used local to the holes.

Figs. 14-21 illustrate the results of this analysis. The maximum and minimum stresses are 1.65 E 04 and -3.3 E 03 psi. The connecting rod placement is represented by the black lines while force direction is indicated by the arrows. It is interesting to note that the "tooth" undergoes maximum compressive stress as the trammel gear is travelling upward; relative to the page (crank angle = 0°). At this position the trammel gear is beginning its swing toward the output gear and its mass center motion is opposed to the direction of application of the tooth force. Conversely a "tooth" is subjected to maximum tensile stress at crank angle 225 where motion of the mass center is away from the output gear and perpendicular to the angle of application of the tooth force. This means certain teeth will always be subjected to compressive stress while others undergo tensile stresses in a pattern which will repeat each cycle. These extrema at 225° are those that were predicted in the overview analysis.

Area B, in addition to actual values of stresses, has an added importance in that the same area is subjected over an engine cycle and may be more susceptible to fatigue. It can be seen that there are small variations in the stress concentrations of this area over a cycle. There is a large maximum stress at 225° which will, over many cycles, represent a potential high fatigue area.

It is important to note that around the holes the same portion of the hole remains in either tension or compression throughout the cycle. The force application in a global coordinate system seems to spin around while in a coordinate system local to the gear it moves very little. The forces change in magnitude but are basically applied at the same point. This is a very important point since ultimately fatigue may be reduced. The portion of the hole boundary that is in compression initially remains in compression throughout the entire cycle and the same is true with the portions of the hole boundary in tension. This is a second important result which may be utilized in better gear design.

This phenomenon can possibly be better understood by first considering that in a two force member the forces would be directly opposed. The addition of a third force applied predominately as a rotational driving force tends to skew the opposed forces. Area B then tends to be twisted but the twist is constant in direction and will therefore tend to minimize fatigue.

GEAR DESIGN

This analysis has pointed out several areas which will enhance future designs of the floating gear assembly for the Stiller-Smith Engine. Identification of the areas around the teeth that are subjected to compression and those that are predominately under tension have been located. The portions of the gear in the vicinity of the holes under compression and tension have also been located. More importantly the intermediate areas of low stress are also now known. (Note: any local changes in gear thickness which may be implemented to take advantage of this knowledge must maintain a symmetry about the center of the gear to minimize balancing problems.) This information will help to focus future FEM work on new gear designs for this application, especially the area of boundary condition identification, which will ultimately allow for optimized eccentric gear designs for this new mechanism.

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