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James E. Smith, Robert Craven Aubra D. McKisic and John C. Smith

West Virginia University
Morgantown, WV

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ABSTRACT

The Scotch yoke in its various forms and inversions has received considerable attention as possible alternatives to the slider-crank for internal combustion engine use. As a recent entry, the Stiller-Smith Mechanism has shown promise as being a viable and strong option. In this study emphasis was placed on comparing the number and similarity of mechanism components and the balancing aspects of these components, implications of component and linkage motions, the severity of loading experienced by similar bearing surfaces within the engines, and some of the friction losses associated with these new motions. It was found that the Stiller-Smith Engine has significantly fewer moving parts. It was also found that journal bearings in the slider-crank engine were more severely loaded than those in the Stiller-Smith Engine. The linear reciprocating bearings in the Stiller-Smith Engine were more heavily loaded than the slider-crank piston skirts.

THE MOST COMMON motion conversion mechanism used in internal-combustion engines is the slider-crank. Kinematically the slider-crank is modeled as a planar four-bar linkage. The crank throw serves as the input link, the connecting rod is the coupler, and the piston is the output link. All members are linked by revolute joints (journal bearings) except for the pistons and frame (block) which are joined by prismatic joints. Multi-cylinder engines can be considered as multiple slider-cranks sharing a common frame and coupled by a common crank. The kinematic behavior of the slider-crank, as in any other four-bar linkage, is easily described.

Of the possible alternatives to the slider-crank, the scotch-yoke and its various kinematic inversions has been the the object of several investigations. Its use in internal combustion engines has been patented by Hunter [1], Bourke [2], Reitz [3], and Flinn [4]. The Geisel engine [5] has also shown promise.

The Stiller-Smith Mechanism represents a different approach to the motion conversion objective. The mechanism was originated at West Virginia University [6] and presently holds two U. S. patents [7, 8]. Detailed descriptions of the mechanism can be found in these and other sources [9-11] so only a cursory introduction is included here. The mechanism is in the form of a double cross-slider, or elliptic trammel. The trammel link shown in Fig.1 is replaced by a gear whose center is located midway between the pins which constrain its motion with respect to the "connecting rods." These are not connecting rods as in a slider-crank engine because they are rigidly connected, that is without a wrist-pin, to the pistons located at the opposite ends. As depicted in Fig. 1 as the connecting rods reciprocate linearly, the center of the trammel gear translates in a circular fashion about an axis which is located at the intersection of the connecting rods and is perpendicular to both rods. As the center of the gear translates in a circle, the entire gear rotates about its geometric center in the opposite

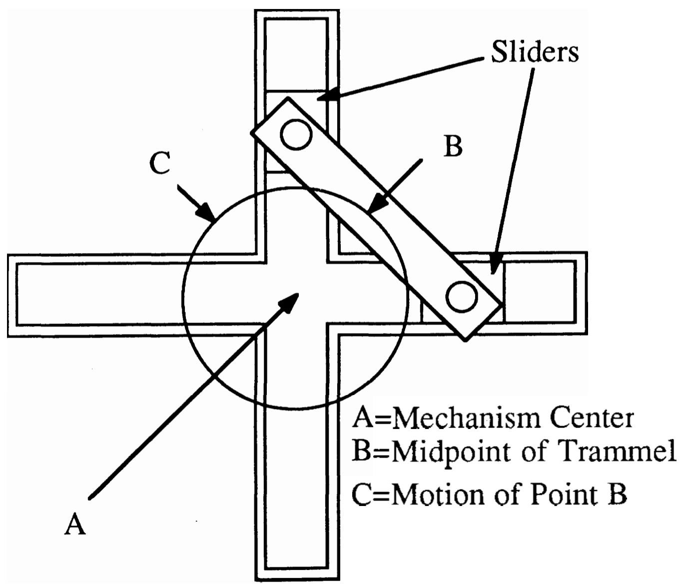

Fig. 1 The Double Cross-Slider

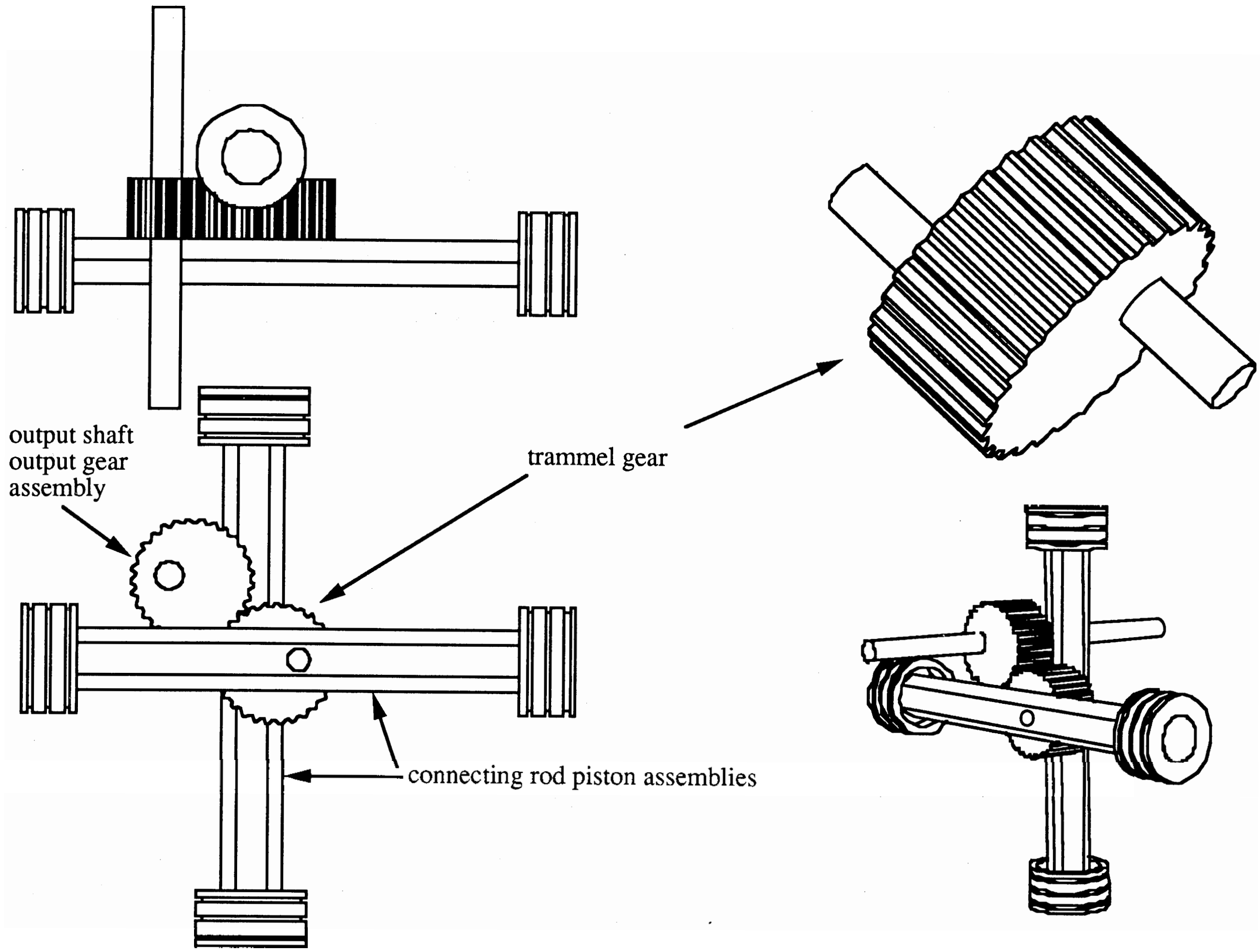

Fig. 2 The four moving part Stiller-Smith Mechanism configured for an internal engine application.

direction of the translation, Fig. 2. The magnitude of the angular velocity of this rotation is equal to that of its angular translation but with the opposite algebraic sign. The same is true for the angular acceleration [9]. Herein lies the principle difference between the Stiller-Smith Mechanism and other double cross-sliders. In the previous attempts to employ the cross-slider, the translation was harnessed by means of a crank which rotated about the axis located at the center of translation. To compensate for the trammel rotation, a bearing was required at the trammel center. The Stiller-Smith Mechanism utilizes this trammel rotation instead of eliminating it. This is accomplished by a trammel link in the form of a gear. This gear is in continuous mesh with one or more similar gears mounted eccentrically on one or more output shafts, as shown in Fig. 3.

The output gears are mounted eccentrically to compensate for the center translation. Otherwise a continuous mesh would not be possible. With this arrangement, as the trammel gear rotates, so does the output gear, but in the opposite direction. Therefore, the angular velocities and accelerations of the output shafts are all identical to those of the translation of the trammel gear center [9]. A feature inherent in the Stiller-Smith Mechanism, as in any double-cross-slider, is that the motions of the pistons are all described by single harmonic terms.

BALANCING

In the case of the slider-crank, the piston motion is often approximated as a two-term harmonic. This complex motion results in the need for complicated balancing schemes. Since these high order terms are absent from the equations of

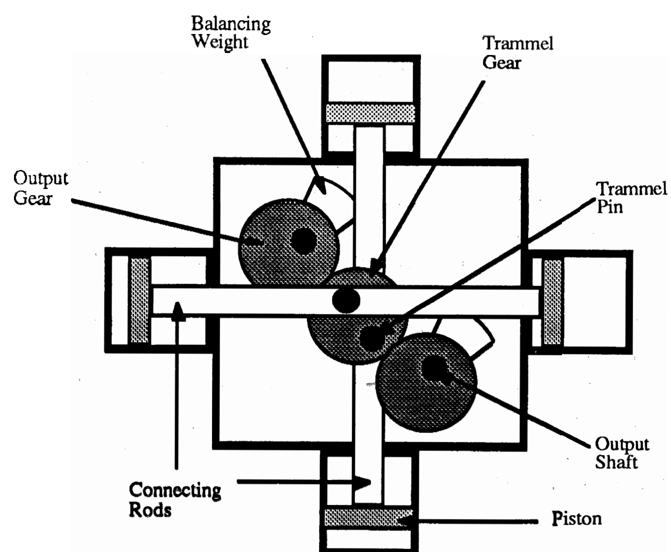

Fig. 3 The Stiller-Smith Mechanism

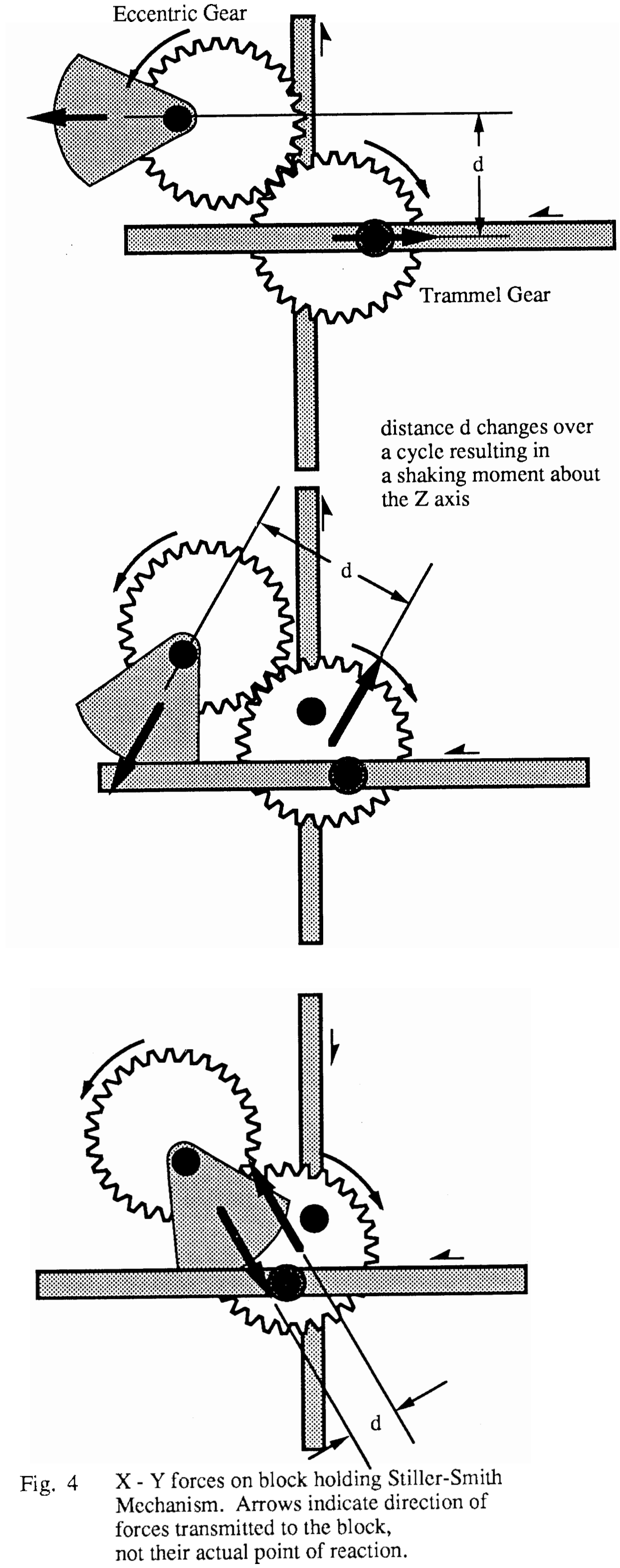

Fig. 4 X - Y forces on block holding Stiller-Smith Mechanism. Arrows indicate direction of forces transmitted to the block, not their actual point of reaction.

motion in the Stiller-Smith Mechanism, as can be illustrated, the mechanism is easily balanced [10-12].

By first treating the device as a planar mechanism, we can quickly balance the major forces. These X-Y forces occur sinusoidally at the linear bearings of the piston-rods. Their vector sum is equivalent to that of a mass being rotated on a shaft and, therefore, can be balanced with a weight placed counter to the rotating mass on a similarly rotating shaft, ie. the output shaft. (Fig. 4) The distance from the center of the engine to the center of the trammel gear multiplied by the mass of the mechanism must be equal to the mass of the counter weights multiplied by the radius from the output shaft center to the center of gravity of the counterweight[10].

This simple solution does counter the primary imbalance

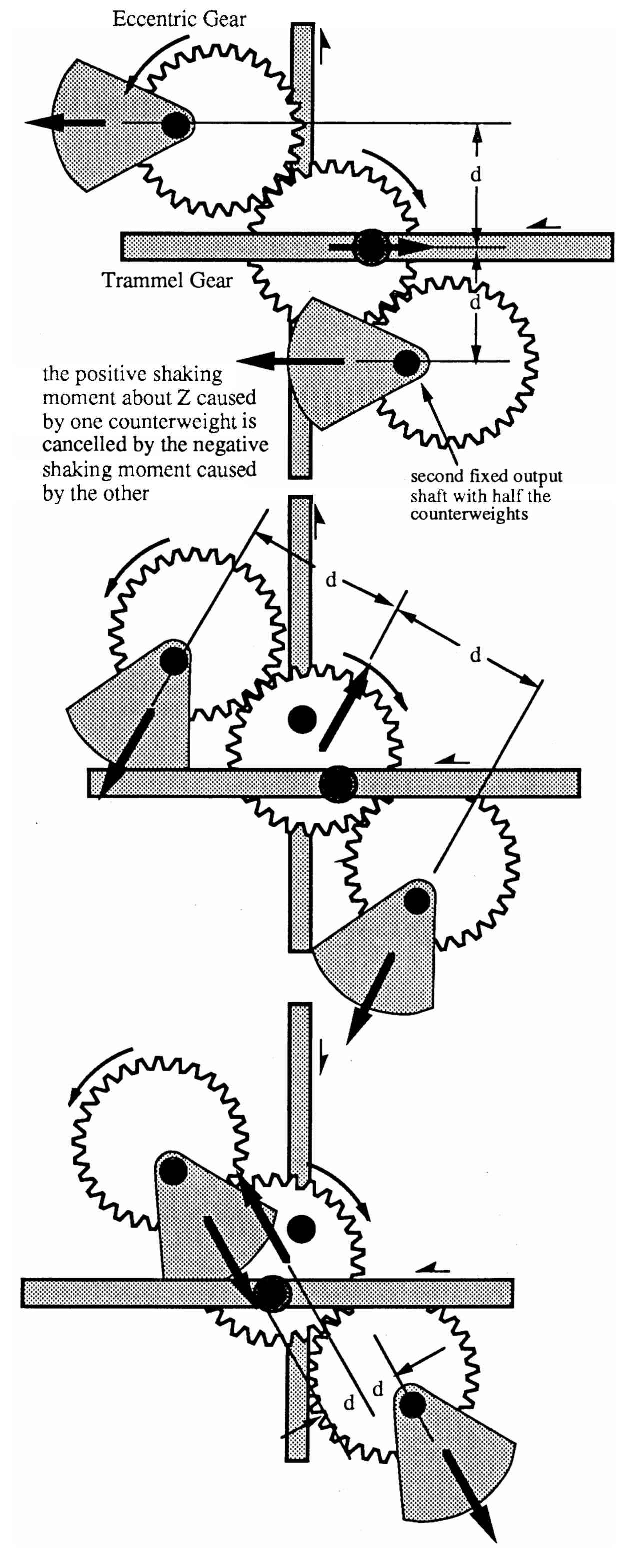

Fig. 5 A five moving part Stiller-Smith Mechanism with X - Y shaking forces and Z axis shaking moments balanced

however not all six degrees of freedom are accounted for. In some applications this may be sufficient - depending on the amount of imbalance which is acceptable.

By inspection of the arrows in Fig. 4 it can be seen that the perpendicular distance measured between the two arrows is varying. This cyclic change in distance will result in a shaking moment about the Z axis. To eliminate this imbalance the output shaft must either coincide with the center of the engine or another shaft must be added in the opposite quadrant of the mechanism which must have half of the counterweights on it to produce an equal and opposite shaking moment about the Z axis, Fig. 5. The complete vector analysis of this shaking moment may be found in Nesbit et. al.[10,11].

The basic four cylinder Stiller-Smith Engine is completely balanced for all six degrees of freedom. This is accomplished with counterweights distributed on three shafts - two using the eccentric gears mated to the trammel gear and one counter rotating shaft to counter out-of-plane moments. It is important to note that most 4-cylinder engines are not completely balanced but instead tolerate high order imbalances from the irregular motion of the connecting rods. In the Stiller-Smith Engine, all motions are either linear or in a circle. Further, the orthogonal linearly moving parts add to circular motion, thus making the planer balancing of the mechanism trivial. Out-of-plane moments arise due to the three dimensional aspects of the mechanism, but counterweights on a counter rotating shaft eliminate this imbalance leaving a completely balanced engine. If as with many of today's four cylinder applications, some imbalance can be tolerated, these counterweights and their counter rotating shaft may be eliminated.

The cruci-form shape allows for multiple mechanisms to be linked via output shafts to form multi-cylinder (greater than four) arrangements. By the addition of balancing weights to the output shafts, two different eight-cylinder configurations can be balanced in three-dimensions [12]. These configurations are illustrated in Figs. 6 and 7.

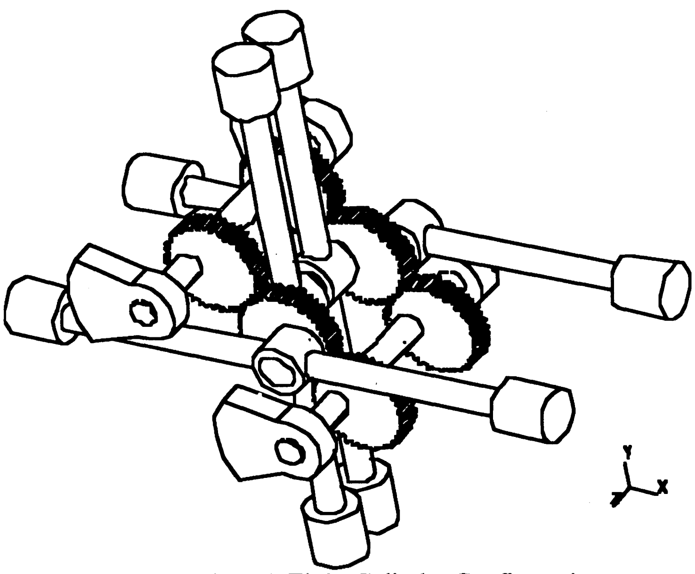

Fig 6 The ABBA Eight Cylinder Configuration

The first configuration, Fig. 6, is designated ABBA. While proceeding along an axis parallel to the output shafts the connecting rods are encountered in the following order; horizontal, vertical, vertical, and then horizontal. Fig. 7 shows the second balanceable configuration, the ABAB. The connecting rods are in the order of horizontal, vertical, horizontal, and vertical.

It is possible for a mechanism to contain up to five output shafts. One shaft can be used as the main drive shaft. It can also be externally coupled with another to change the internal load distribution of the engine. Separate shafts can be used to drive accessories such as cooling fan, distributor, cam shaft, and generator. An auxiliary power-take-off shaft is also readily available.

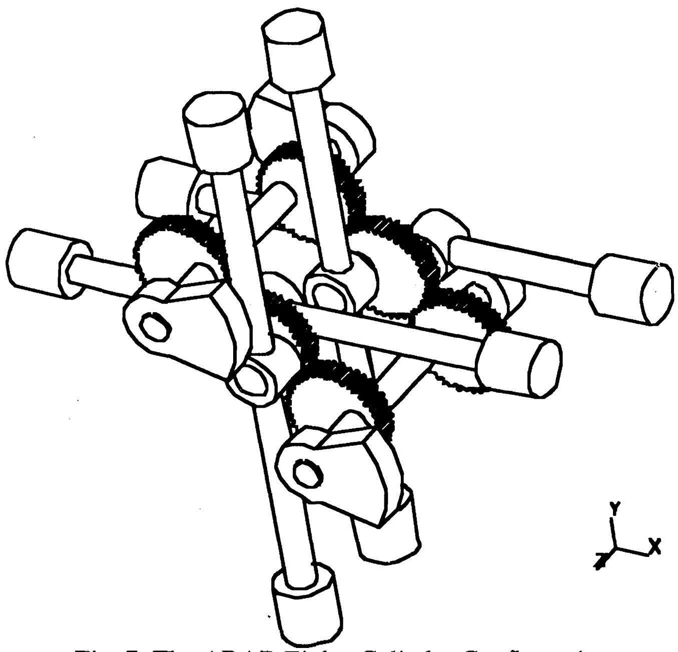

Fig. 7 The ABAB Eight -Cylinder Configuration

The Stiller-Smith Engine is ideally suited for aeronautic or marine applications which are more sensitive to imbalances. The balanced engine should provide a reduced vibration environment for sensitive instrumentation applications such as reconnaissance drones.

Multiple bank engines, i.e., those with more than one set of four cylinders, have the added benefits of reduced or zero counterweight requirements to archive a complete balance. Many configurations exist where various components of imbalance in one bank of cylinder counter a component of imbalance in another bank. Certain eight cylinder configurations have a vastly reduced counterweight requirement. This phenomenon is manifest most fully in two 16 cylinder configurations which do not require any counterweights. This self balancing capability allows engines with very low inertia-to-displacement ratios to be constructed. This is beneficial if fast response is desired and it also allows the total weight of the engine to be reduced.

RESULTS

The Mechanisms

The objective of the study was to compare the slider-crank and Stiller-Smith mechanisms in eight cylinder, four-stroke, compression-ignition engine environments. The slider-crank engine investigation is based on the Cummins VT-903 turbocharged diesel engine. The derivations and analyses used for this comparison are available in the open literature [13-22]. All calculations for the Stiller-Smith Mechanism were based upon an engine whose stroke, bore, and displacement are equivalent to those found in the Cummins VT-903. Specifications used in the investigation may be found in Tables I and II for both engines.

Table I: V-8 Engine Specifications

General Specifications			
Base Engine		Cummins VT-903	
Stroke		12.1 cm	(4.75 in)
Bore		14 cm	(5.5 in)
Displacement		14.8 l	(903 cu. in.)
Operating Cycle		4-Stroke, CI	
Bank Angle		90o	
Compression Ratio		15.5:1	
Governed Speed		2400 rpm	
Crank Specifications			
Throw Length	r1	6.033 cm	(2.375 in)
Mass Rotation Radius	r1g	6.033 cm	(2.375 in)
Rotating Weight/Throw	W1	17.87 N	(4.017 lbf)
Main Bearing Diameter	d1	9.53 cm	(3.75 in)
Crank-Pin Diameter	d2	7.938 cm	(3.125 in)
Bearing Separation	q	17 cm	(6.6 in)
Con-Rod Separation	s	3.8 cm	(1.5 in)
Balancing Weight Separation	h	59.06 cm	(23.25 in)
Con-Rod Specifications			
Length	r2	20.81 cm	(8.193 in)
Mass Center Location	r2g	6.716 cm	(2.644 in)
Weight	W2	30.9 N	(6.94 lbf)
Moment of Inertia	I2	0.02977 N-m-s2	(.2635 in-lbf-s2)
Piston Specifications			
Weight	W3	33.26 N	(7.477 lbf)
Skirt Area		84.677 cm2	(13.125 in2)
Friction Coefficients			
Main Bearings	m1	0.01	
Crank Pins	m2	0.01	
Piston	m3	0.05	

Table II: Stiller-Smith Engine Specifications

General Specifications			
Base Configuration		ABAB	
Stroke		12.1 cm	(4.75 in)
Bore		14 cm	(5.5 in)
Displacement		14.8 l	(903 cu. in.)
Operating Cycle		4-Stroke, CI	
Compression Ratio		15.5:1	
Governed Speed		2400 rpm	
Output Shaft/Gear Specifications			
Gear Diameter	D	15 cm	(6.0 in)
Mass Rotation Radius	r5g	3.0163 cm	(1.1875 in)
Rotating Weight/Gear	W5	54.94 N	(12.35 lbf)
Bearing Separation	q	17.1 cm	(6.75 in)
Con-Rod Separation	s	22 cm	(8.5 in)
Balancing Weight Separation	h	52.1 cm	(20.5 in)
Pitch Angle	f	20o	
Torque Ratio	r	0.111	
Output Shaft/Gear Specifications			
Gear Diameter	D	15 cm	(6.0 in)
Mass Rotation Radius	r2g	3.0163 cm	(1.1875 in)
Rotating Weight/Gear	W2	54.94 N	(12.35 lbf)
Pitch Angle	f	20o	
Pin Separation		2.375	
Con-Rod/Piston Specifications			
Bearing Mount Separation	B	44.133 cm	(17.375 in)
Weight	W2	143.8 N	(32.33 lbf)
Linear Bearing Area		280.3 cm2	(43.44 in^2)
Friction Coefficients			
Main Bearings	m1	0.01	
Trammel Pins	m2	0.01	
Linear Bearings	m3	0.05	

Table III shows a breakdown of the moving parts of the engines. The members listed include only those involved in the motion conversion mechanisms themselves and are listed by functionally similar motions.

Table III: Moving Part Breakdown

	VT - 903 (8-Cylinder)		Stiller-Smith (8-Cylinder)	
Motion	Component	Number	Component	Number
Reciprocating	Pistons	8	Piston-Rods	4
Rotating	Crankshaft	1	Output Shafts	2
Complex Motion	Con-Rod	8	Trammel Gears	2
	Total	17		8

This table shows that the V-8 mechanism contains over twice as many moving parts as does the 8-cylinder Stiller-Smith. For members experiencing complex motion, that is motion other than just simple translation or rotation, the V-8 has four times as many members as does the Stiller-Smith. The complex motion experienced by the V-8's connecting rod also requires multiple harmonic terms for an accurate description.

Figure 8 shows the joint (bearing surface) types of the two mechanisms. A breakdown of the joints into bearing type is shown in Table IV for both mechanisms.
The slider-crank engine contains more bearing surfaces, by 42%, than the Stiller-Smith engine. The table also identifies bearing surfaces that serve similar purposes in the two mechanisms. Both mechanisms contain 8 sets of piston rings, one per cylinder. These serve the function of containment of combustion gases and isolation of lubricant from the combustion chamber. While the rings will provide some support for lateral load, this is primarily accomplished by the piston skirt in the slider-crank and the linear bearings in the Stiller-Smith. The slider-crank engine therefore has

Joint Identification

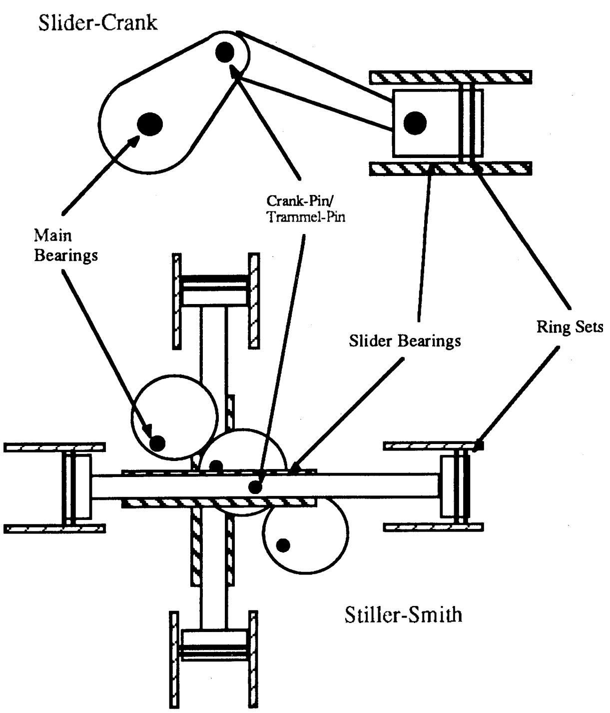

Fig. 8 Mechanism Joint Identification

twice as many linear reciprocating bearing surfaces. The slider-crank also has twice as many crank-pin, or big-end, bearings as the Stiller-Smith engine has trammel pin bearings. Conversely, the Stiller-Smith engine has six journal bearings for its two output shafts compared to the five main journal bearings supporting the V-8 crankshaft.

Table IV: Bearing Surface Breakdown

	VT - 903		Stiller-Smith	
Bearing Type	Component	Number	Component	Number
Linear Reciprocating	Ring Sets	8	Ring Sets	8
	Piston Skirts	8	Linear Bearings	4
Rotating Journals	Main	5	Output Shaft	6
	Crank Pins	8	Trammel Pin	4
Oscillating	Wrist Pin	8	Wrist Pin	0
Gear Contacts	Gears	0	Gear Teeth	4
	Total	37		26

It should be noted that the Stiller-Smith Engine can operate with one to five output shafts. A Stiller-Smith Engine employing a single output shaft requires only three bearings. Overall the V-8 contains 13 rotating journals to the 10 for the Stiller-Smith. The Stiller-Smith Engine contains no functionally similar bearing surface to the slider-crank oscillatory wrist-pin. Likewise there are no gear contacts or any higher order kinematic pairs in the slider-crank. Adding an auxiliary power-take-off shaft to the V-8 will require the addition of a minimum of two journal bearings and one higher kinematic pair.

Journal Bearing Load Comparison

The main bearings in the V-8 and the output shaft bearings in the Stiller-Smith Engine are functionally similar. The load distributions for the main bearings are shown in Figs. 9 and 10. For the Stiller-Smith Engine the outer bearings are the most heavily loaded whereas the V-8 is most heavily loaded in the center bearings. In the Stiller-Smith Engine, bearing 4,5, and 6 have significantly higher peak loads than do bearings 1,2, and 3. This is a direct result of 90% of the torque being carried by the output shaft containing these bearings. The following results presented will be for the Stiller-Smith bearing #6, which is the most heavily loaded Stiller-Smith bearing, and the first V-8 main bearing.

It is assumed that the engine load is held constant over the range of engine speeds investigated. The effect of engine speed on the bearing load is similar in both engines. As shown in Fig. 11, bearing load increases with increasing engine speed except for the maximum V-8 load. This indicates that inertial forces and gas forces work together instead of in opposition. The effects are approximately the same with inertial forces having a greater influence for the Stiller-Smith Engine. In a direct comparison of the loads, as shown in Fig. 12, the maximum V-8 load is nearly four times that of the Stiller-Smith at 2400 rpm. The maximum bearing load due to gas pressure, shown in Fig. 13, in the V-8 is 2.75 times that experienced in the Stiller-Smith. After comparing the maximum loads it is concluded that the V-8 bearings are more likely to fail due to fatigue.

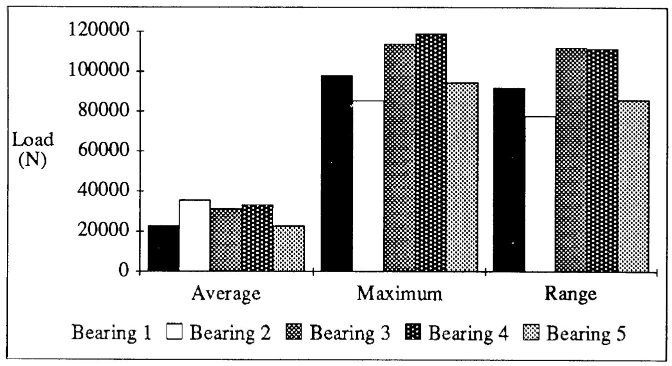

Fig. 9 V-8 Main Bearing Load Distribution

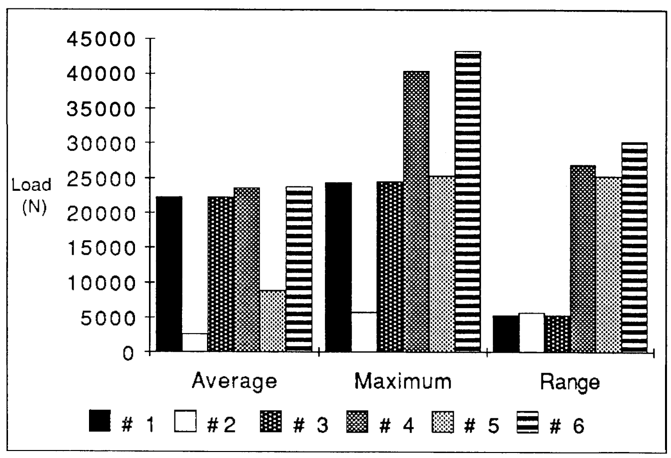

Fig. 10 Stiller - Smith Output Shaft Load Distributions

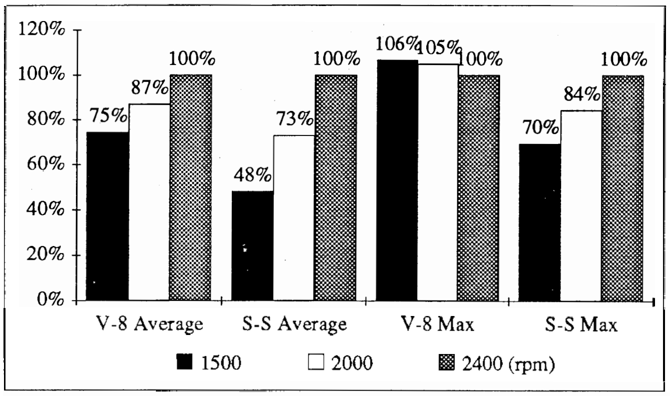

Fig. 11 Effect of Engine Speed on Main Bearing Loads: Normalized to 2400 RPM

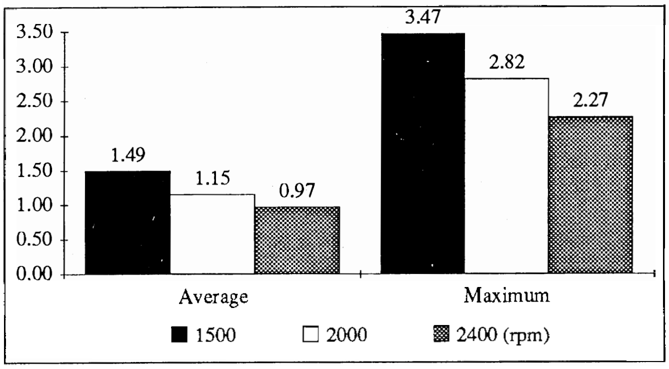

Fig. 12 Ratio of V-8 Main Bearing to Stiller-Smith Output Shaft Bearing Load

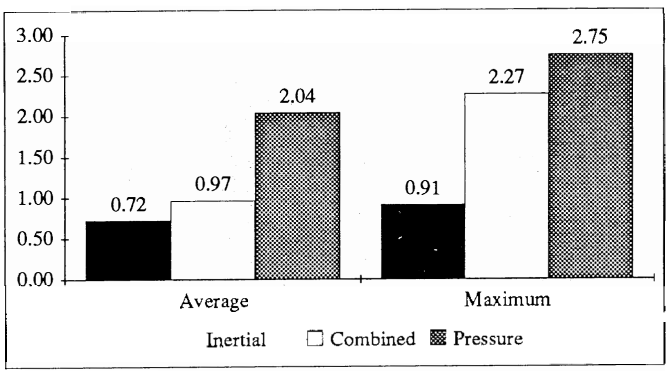

Fig. 13 Relative Magnitude of Contributing Loads: V-8 Main Bearing Load / Stiller - Smith Output Shaft Bearing Load

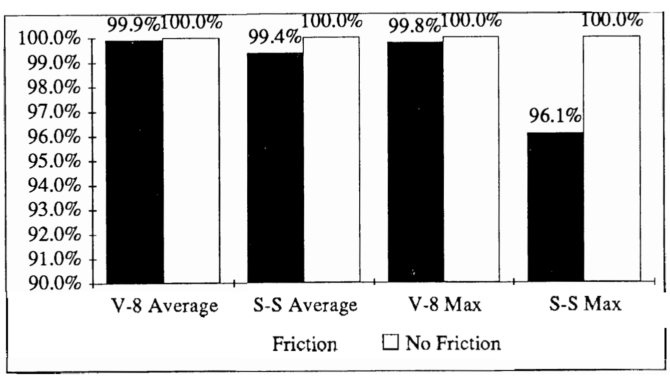

Fig. 14 Effect of Friction on Main Bearing Loads

The dual output shafts of the Stiller-Smith Engine introduce possibilities for design variation not available to the V-8. As previously discussed the difference in loading between bearings on the two output shafts is a result of the torque distributions. It was assumed in the analysis that the balancing weights were identical on both output shafts.

The difference between the output shaft and crankshaft bearing loading increases even more with the introduction of friction into the system. As seen in Fig. 14 the introduction of friction decreases the loading on all bearings. The effect is greatest in the Stiller-Smith Engine. The percentages shown are based upon a Coulomb friction model.

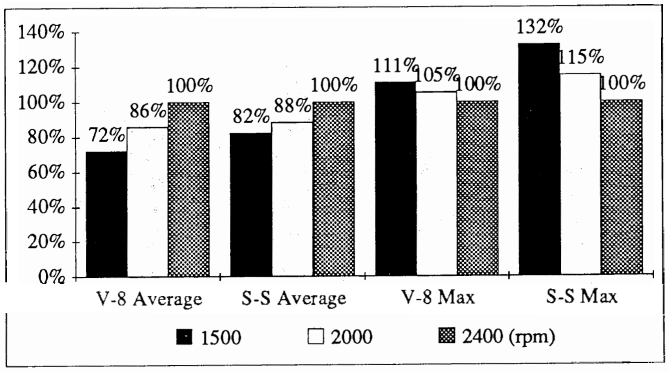

Fig. 15 Effect of Engine Speed on Pin Loads: Normalized to 2400 RPM

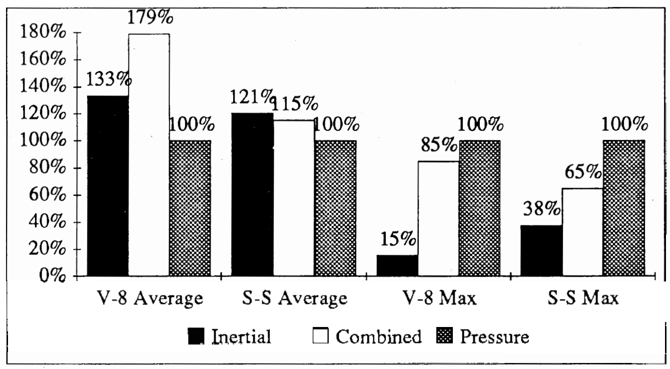

Fig. 16 Relative Magnitude of Contributing Pin Loads: Normalized by Pressure Loads

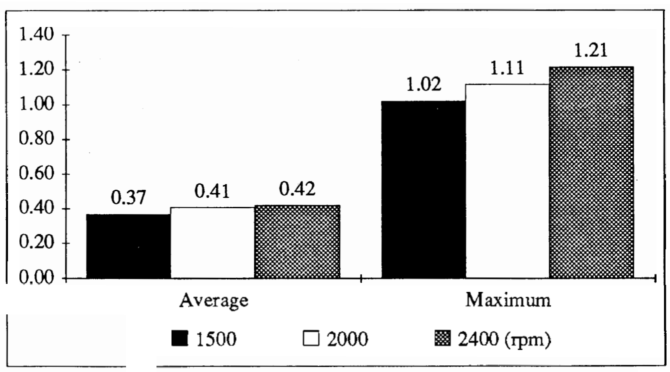

Fig. 17 V-8 Pin Loads / Stiller Smith Trammel Pin Loads

<u>Comparison of Crank-Pin and Trammel Pin Bearing Loads</u>

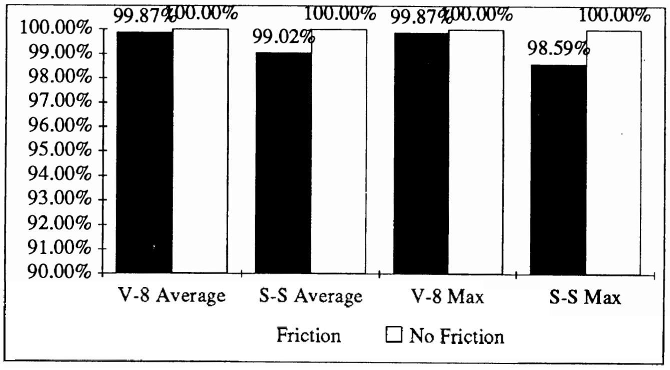

Fig. 18 Effect of Friction on Pin Loads

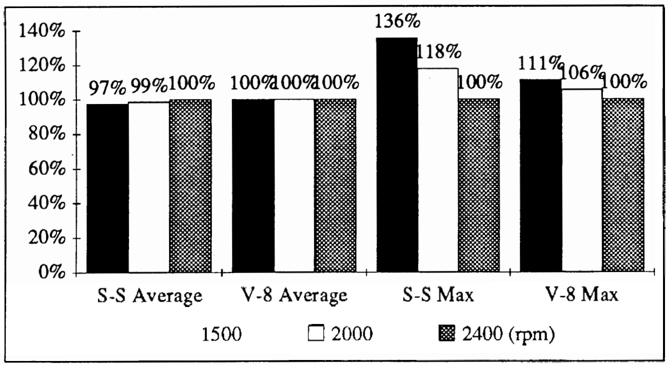

Fig. 19 Effect of Engine Speed on Reciprocating Bearing Loads: Normalized to 2400 RPM

The trammel pins on the Stiller-Smith and crank pin in the slider-crank are functionally similar. As shown in Fig. 15, average pin loads again increase with engine speed. The maximum pin loads decrease with increasing engine speed. This trend is again more pronounced in the Stiller-Smith Engine. The trend in both shows that inertial and gas forces oppose each other in their contribution to the maximum pin load. Fig. 16 helps to explain why engine speed has a greater effect on maximum load in the Stiller-Smith Engine. Maximum inertial forces are 38% of the maximum gas forces in the Stiller-Smith Engine. They account for only 15% of the maximum gas forces in the V-8. It is also noteworthy that the combined forces are 85% of the pressure forces in the V-8. Therefore the maximum inertial and gas forces occur nearly simultaneously and directly oppose each other. This is not the case on the Stiller-Smith Engine. Because the maximum inertial forces in the Stiller-Smith are closer in magnitude to the maximum gas forces, the combined load is actually less than that in the slider-crank. The effect is increased with engine speed as seen in Fig. 17. Figure 18 shows that for all cases the introduction of friction reduces pin loads. Like the main bearings, the pin bearings in the Stiller-Smith Engine are less susceptible to fatigue failure based on maximum loading.

Some important conclusions can be made from summarizing the preceding discussions on the journal bearing loadings for the two mechanisms. In all cases the maximum loads experienced by the V-8 journal bearings are greater than those in the corresponding Stiller-Smith journal bearings. If all other factors are considered equal the journal bearings in the Stiller-Smith engine are less susceptible to fatigue failure. The drastic difference between the maximum loading on the main bearings can be attributed to the corresponding gas force reactions. In the V-8 engine the large loads due to cylinder pressure are transmitted directly through the crank and must be supported by the main bearings. These same loads in the Stiller-Smith engine are carried by the linear bearings instead of the output shaft bearings. The introduction of friction in the system decreases the loads on all journal bearings. Based on minimum capacity ratio, the Stiller-Smith journal bearings are less likely to have fluid film breakdown and suffer seizure.

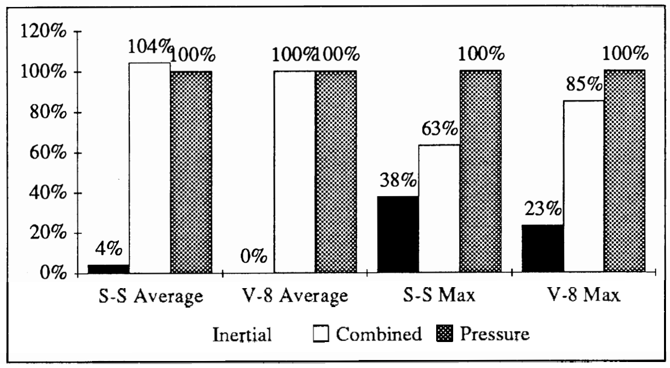

Fig. 20 Relative Magnitudes of Contributing Reciprocating Bearing Loads: Normalized by Gas Pressure Loads

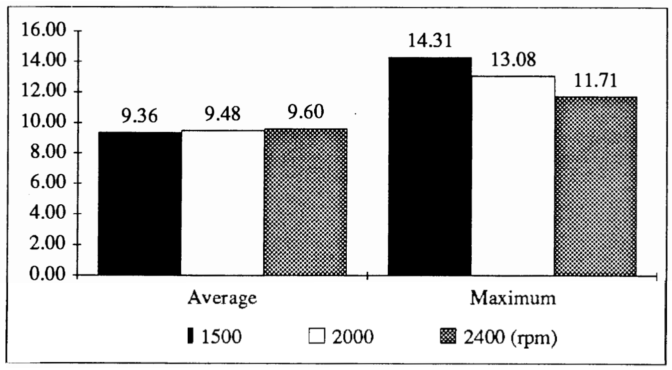

Fig. 21 Stiller - Smith Linear Bearing Loads / V-8 Piston Sidewall Load

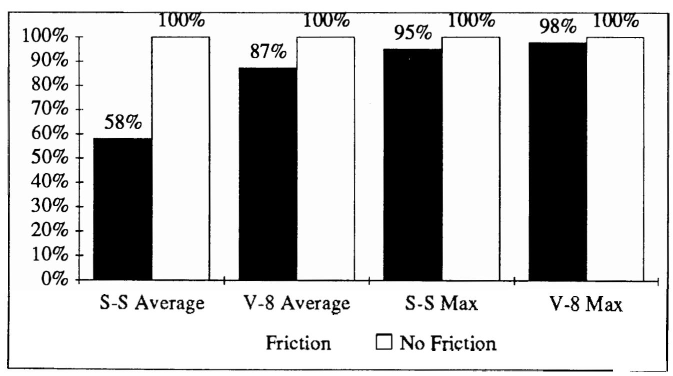

Fig. 22 Effect of Friction on Reciprocating Bearing Loads

Reciprocating Bearing Load Comparisons

In a standard slider-crank engine, the piston skirt serves as a bearing surface that must reciprocate linearly and at the same time support a load. The same purpose is accomplished by the linear bearings in the Stiller-Smith Engine. Fig. 19 shows the effect of engine speed on reciprocating bearing load for the two mechanisms. The maximum normal load decreases with increasing engine speed for both mechanisms indicating that inertial forces oppose gas forces for both. The effect is more drastic in the Stiller-Smith Engine. This is reflected in Fig. 20 in that the maximum inertial and gas pressure forces are closer in magnitude in the Stiller-Smith Engine than in the V-8. A direct comparison of the loads, Fig. 21, shows that the Stiller-Smith slider is much more heavily loaded than the V-8 piston sidewall. The ratio of average load increases with engine speed while that for maximum load decreases. As would be expected in both engines, the loads decrease with the introduction of friction. This is shown in Fig. 22.

The reason that the Stiller-Smith linear bearings are so much more heavily loaded is that they, like the V-8 main bearings, must experience the main force exerted on the piston by the cylinder pressure. One redeeming factor for the linear bearings is they carry their highest load at the time their velocity is the greatest. This is very beneficial for hydrodynamic lubrication. Even though the Stiller-Smith linear bearings are much more heavily loaded than the slider-crank piston sidewall, there is no indication that these bearing surfaces will not provide the required support.

Balancing Implications

It has been shown how in the past [23] the planar forces in the X and Y directions have been balanced through the use of counterweights on the output shaft. Shaking moments about the Z axis were eliminated by the use of a second output shaft and dividing the counterweights between the two shafts. In this way two shaking moments about Z are created which cancel while still balancing the X and Y inertia forces.

The concept of "wobble" is the cyclic nature of both the moments about the X axis and the Y axis and is given a sign convention based on whether the moment about X followed the moment about Y by 90^{o} or vica versa with Y following X being positive. A Stiller-Smith Engine with a positive angular velocity creates a negative wobble while counterweights on a shaft with positive angular velocity can only cause positive wobble. Wobbles with opposite sign will cancel in one direction while becoming additive in the other direction. Simple counterweights on the existing output shaft will not suffice. Through the introduction of counter rotating shafts, via centrally mounted gears with a 1:1 gear ratio, counterweights can be placed giving a negative wobble, 180 degrees out of phase from the mechanisms negative wobble, thus completely balancing the mechanism.

In the interest of lower inertia, it may in some cases be desirable not to completely balance the mechanism. Keep in mind that the common place 4 cylinder slider-crank engines are not usually completely balanced and, therefore, a certain amount of imbalance may be acceptable. Not using counterweights reduces the moment of inertia and results in a more responsive engine.

In the 8, 12, and 16 cylinder engines, there exist many configurations where the counter rotating shaft may be eliminated, or further there may be a reduction in either the force imbalance or the wobble imbalance. The best of these configurations seems to be the ABAB180^{o}BA180^{o}BA0^{o} and the ABBA180^{o}AB180^{o}BA0^{o} 16 cylinder engines which do not need any counterweights at all. It is important to note that though some configurations enjoy an advantage from a balancing perspective over other configurations, they all can be completely balanced through the use of simple counterweights and counter rotating shafts.

Friction Loss Implications

One of the strong points of the diesel engine is its good fuel economy. Millington and Hartles [24] state that the difference between a very good diesel engine and an average diesel engine is almost invariably due to a difference in their frictional losses. All other things being equal, a reduction in the number of bearings reduces the friction losses in the system containing those bearings. The extent of the resulting reduction of friction losses is dependent upon a number of conditions. Primarily, a significant friction loss reduction can be achieved if the number of bearings can be reduced without subjecting the remaining bearings to excessive loading.

It is commonly held that power losses are proportional to bearing width and the cube of the bearing diameter [25, 26]. If the bearing dimensions must be changed to obtain the same specific loading, the difference in losses can change considerably [25]. If a bearing system has high specific loadings after a reduction in the number of contained bearings, then the bearing dimensions must be changed accordingly. If the same diameter-width ratio is kept and dimensions are changed so that the same specific load is achieved, the friction losses are approximately the same as with a greater number of bearings [25]. Following the same logic, for an equal number of bearings with the same specific loading and diameter-width ratios, smaller bearings can result in significant reductions in friction losses.

For the remaining journal bearings in the engines, the trammel and crank pins, the loadings are much more similar in magnitude. It must be assumed that the bearings are similar in size to be conservative. There are still indications that friction losses will be greater in the V-8 crank-pin bearings because there are twice as many bearings.

Predictions as to comparative losses in the reciprocating slider bearings are much less certain. The majority of published literature covering slider bearings friction losses concerns rotating bearings. The projected bearing area for a Stiller-Smith linear bearing is greater than that of a V-8 piston skirt. The smaller area of the V-8 piston skirt would provide more favorable friction losses [24, 26]. However the V-8 contains twice as many pistons as the Stiller-Smith contains linear bearings and the total projected areas are actually closer than expected.

In this investigation it was assumed that the only motion experienced by the piston was linear reciprocation in the cylinder bore. In actual engines clearances exist between the piston skirt and cylinder walls. This and the ability of the piston to rotate about the wrist-pin allows piston slap to occur. In their work on piston friction losses in diesel engines, Furukama and Takiqucki [27] hypothesis that a large frictional force is generated by the piston slap impulse. Due to the component construction in the Stiller-Smith engine, piston slap is not likely to occur. Without the piston slap it is possible that this large initial friction force is not present.

In addition to these friction losses on a slider cranks's piston skirt the loss in horsepower in the Stiller-Smith linear bearing can be predicted as a function of engine speed, bearing width, bearing length and clearance Figs. 23-26. While these results [28] are not definitive without experimental backup, they do correlate well with friction loss expectations for piston skirts in standard slider-crank engines.

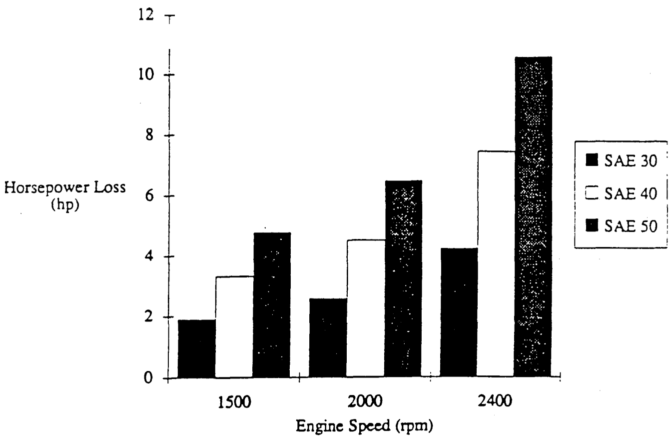

Fig. 23 Horsepower Loss as a Function of Engine Speed for Linear Bearings

CONCLUSIONS

It is impossible to predict the functional success of a machine from theoretical studies alone. Simulations are useful in identifying potential strengths and weaknesses without the expensive and time consuming construction of the actual machines. This analysis attempted to examine specific components in internal combustion engines using two different motion conversion mechanisms and to make comparisons between the component's performance and the effects certain parameters have upon their performance.

On a basis of the number of components it is concluded that an eight cylinder engine using the Stiller-Smith Mechanism is superior to a similar slider-crank design because it has less than half the moving parts. These parts are also less complex in construction. The number of bearing surfaces in the Stiller-Smith Engine is again less than that in a standard V-8.

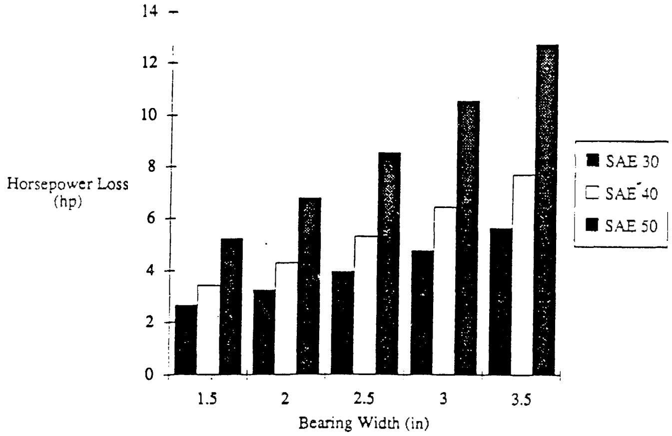

Fig. 24 Horsepower Loss as a Function of Bearing Width for Linear Bearings (Engine Speed 2400 RPM)

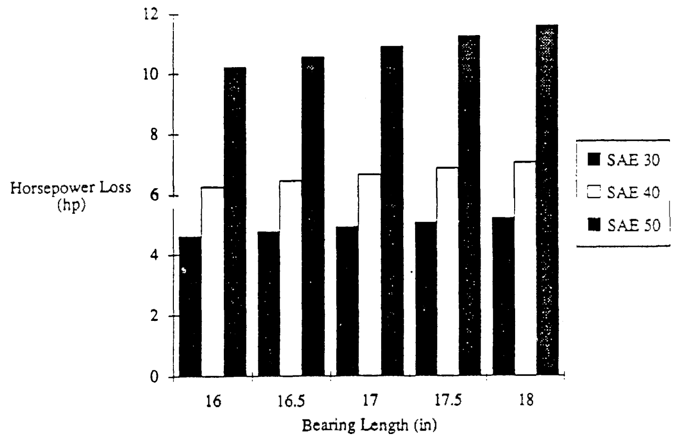

Fig. 25 Horsepower Loss as a Function of Bearing Length for Linear Bearings (Engine Speed 2400 RPM)

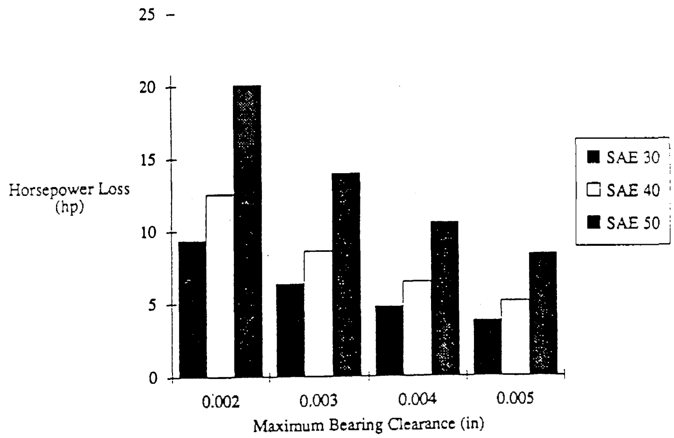

Fig. 26 Horsepower Loss as a Function of Maximum Bearing Clearance for Linear Bearings (Engine Speed 2400 RPM)

Piston motions are different in the two mechanisms with the Stiller-Smith Mechanism providing a simpler motion.

The different piston motions effect both mechanical and thermal behavior of the engines. Both mechanisms are considered balanced but because of the more complex piston motion, the slider-crank is not completely balanced as is the Stiller-Smith Mechanism.

For comparable journal bearing surfaces, the performance of those in the Stiller-Smith Engine equalled or exceeded that of those in the slider-crank engine in the areas of bearing fatigue and minimum capacity ratio.

For the linear reciprocating bearings the Stiller-Smith linear bearings were much more heavily loaded than the V-8 piston skirts. This is a direct result of the Stiller-Smith linear bearings directly receiving gas pressure loads. The Stiller-Smith linear bearings also control piston motion more effectively than the V-8 piston sidewall, minimizing piston slap and blow-by.

In the case of constant load it was determined that engine speed had a greater effect upon bearing loads in the Stiller-Smith Engine. A comparison of relative magnitudes of contributing forces showed that the inertial forces in the Stiller-Smith Engine were closer in magnitude to those due to gas pressure than was the case in the V-8. In direct comparisons of inertial loads, those in the Stiller-Smith Engine were the greatest of the two engines. In both engines the inertial forces tended to reduce bearing loads due to gas pressure.

In general the introduction of friction into the system reduces the magnitudes of bearing loads. It was concluded that the journal bearings in the Stiller-Smith Engine will produce fewer friction losses than those in the V-8 if properly designed.

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