A Comparative Study of the Stiller-Smith and Slider-Crank Mechanisms for Eight-Cylinder Internal Combustion Engine Use

J. E. Smith

J. C. Smith

A. D. McKisic
Mechanical and Aerospace Engineering,
West Virginia University,
Morgantown, WV 26506

Of the possible alternatives to the slider-crank for internal combustion engine use, the Scotch yoke in its various forms and inversions has received considerable attention. Among these, the Stiller-Smith mechanism has shown promise as being a viable option. Kinetostatic models were formulated to determine loading within similar eight-cylinder, four-stroke, compression-ignition engines with emphasis placed on comparing the number and similarity of mechanism components, implications of component and linkage motions, the loading experienced by similar bearing surfaces, and the friction losses of specific components.

Introduction

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. Multicylinder 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 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 is the subject of 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 direction of the translation. 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 previous attempts to employ the cross-slider, the trammel 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.

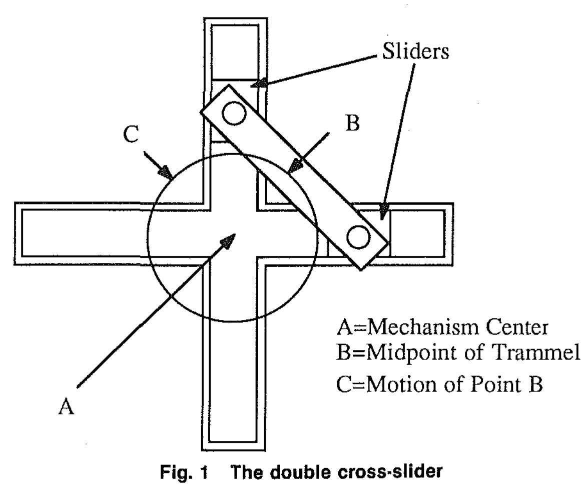

Fig. 1 The double cross-slider

Contributed by the Internal Combustion Engine Division and presented at the Twelfth Annual Fall Technical Conference, Rockford, Illinois, October 7-10, 1990. Manuscript received by the Internal Combustion Engine Division July 1990.

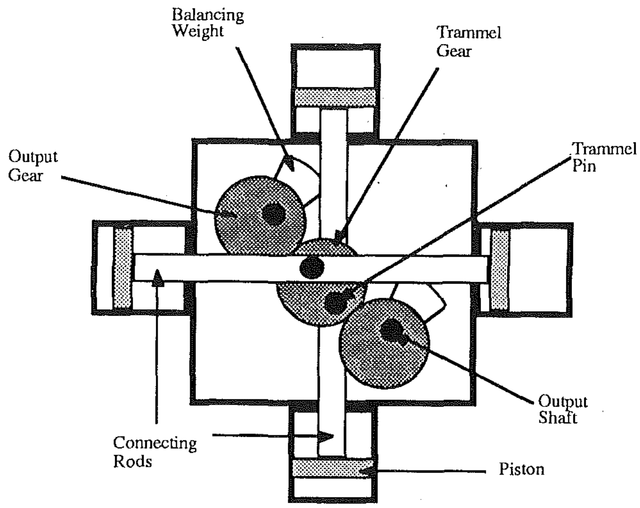

Fig. 2 The Stiller-Smith mechanism

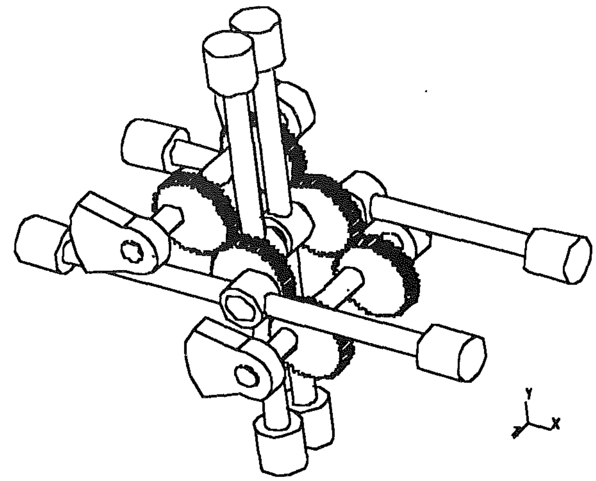

Fig. 3 The ABBA eight-cylinder configuration

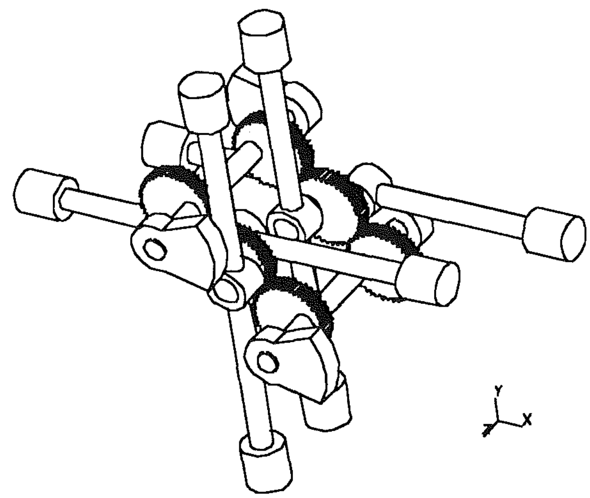

Fig. 4 The ABAB eight-cylinder configuration

2. 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. 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. In the case of the slider-crank, the piston motion is often approximated as a two-term harmonic. This results in the need for complicated balancing schemes. Since these high-order terms are absent from the equations of motion in the Stiller–Smith mechanism, the mechanism is easily balanced [10–12].

The cruciform shape allows for multiple mechanisms to be linked via output shafts to form multicylinder (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. 3 and 4.

The first configuration, Fig. 3, 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. Figure 4 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

Nomenclature

B = distance between linear bearing supports
$D_{()}$ = gear diameter
$F_{()()}$ = force
$I_{()}$ = moment of inertia
$P_{()}$ = cylinder pressure
$R_{()}$ = bearing reaction
$T_{()}$ = shaft torque
$a_{()}$ = acceleration of center of gravity
$d_{()}$ = bearing diameter
$f_{()}$ = friction force
$m_{()}$ = mass
q = axial distance between bearings
$r_{()}$ = location vector magnitude
s = distance between parallel connecting rod
x = Cartesian coordinate
y = Cartesian coordinate
z = Cartesian coordinate
t = time
$\mathbf{F}_{()}$ = force vector
$\mathbf{i}$ = unit direction vector
$\mathbf{j}$ = unit direction vector
$\mathbf{k}$ = unit direction vector
$\mathbf{M}$ = moment vector
$\mathbf{R}_{()}$ = bearing reaction vector
β = mass center location angle
$\theta_{()}$ = location angle
$\mu_{()}$ = coefficient of friction
ρ = torque ratio
ϕ = pitch angle
$\chi_{()}$ = coordinate rotation angle

Subscripts

A = crank throw #1
B = crank throw #2
C = crank throw #3
D = crank throw #4
L = left bank
R = right bank
a = journal
b = bearing
f = friction circle
g = center of gravity
i = component number
j = component number
n = cylinder number
net = net
$/o$ = with respect to margin

Superscripts

x = component
y = component
$'$ = local cordinate
$\dot{}$ = first derivative with respect to time
$\ddot{}$ = second derivative with respect to time

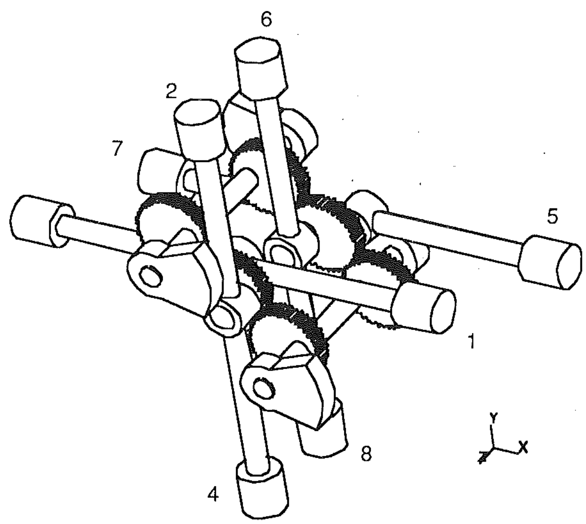

Fig. 5 Eight-cylinder ABAB cylinder designation

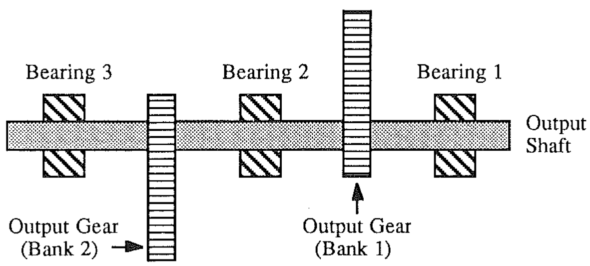

Fig. 6 Single output shaft, gears, and bearings

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-takeoff shaft is also readily available.

Analysis of an Eight-Cylinder Stiller–Smith Engine

Modeling. The kinematic through dynamic analysis of the Stiller–Smith engine has been well described in the literature with the corresponding development for the slider-crank left to the open literature [13–18]. Those readers interested in a detailed analysis or information on unique features of the Stiller–Smith mechanism are referred to the works of Smith et al. [9, 19], Sivaneri et al. [20], and Mucino et al. [21].

Stiller–Smith Firing Order. Since the Stiller–Smith eight-cylinder engine is still in the research stage, no firing order exists. Of the two possible arrangements of the engine that were discussed, this investigation will include only the ABAB configuration. This configuration and the cylinder numbering convention are shown in Fig. 5. Any firing order must be consistent with the balancing requirements. As determined in [12], cylinders 5–8 must be 180 degrees out of phase with cylinders 1–4 for three-dimensional balancing to be possible. This investigation will use a firing order of 1-2-5-4-7-8-3-6. This firing order is one of eight combinations and was chosen for ease of illustration.

Output Shaft Bearing Distributions. The open literature discusses several procedures for determining the load distribution on the five main bearings for a V-8 engine. For the load analysis for the six output shaft bearings for the eight-cylinder Stiller–Smith engine, each output shaft contains three journal bearings. Figure 6 shows one output shaft not including the balancing weights. In the slider-crank analysis, the loads on any main journal are assumed to be affected by only those arrows adjacent to the bearing. When two throws share a common

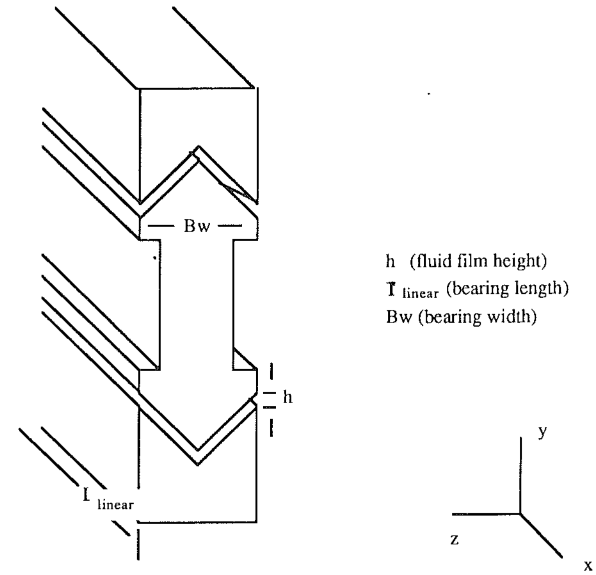

Fig. 7 Cross-sectional view of linear slider bearing

bearing, the loads on the bearing are determined by first assuming that each throw acts separately and then vectorially superimposing the loads resulting from the individual throws. In the Stiller–Smith analysis it is assumed that mechanism bank number 1 has no effect on bearing number 3. If the output gear is located midway between the two bearings, each bearing will support half the net reaction. The same procedure is followed for the second mechanism bank. In this case bearings 2 and 3 are involved. The net reaction due to the two mechanisms on the middle bearing is determined by superimposing the loads resulting from the two individual mechanism banks.

Balancing Weights. As can be shown in [17 and 22] the primary and secondary inertial forces in a 90 degree V-8 engine are inherently balanced. The same can be said for the secondary out-of-plane couple. The primary couple that exists can be easily eliminated by the addition of simple rotating counterweights. Higher order forces cannot be eliminated by the use of simple rotating counterweights. For the eight-cylinder Stiller–Smith mechanism, [12] shows that the primary shaking forces are also inherently balanced for the ABAB configuration. Due to its simple harmonic motion, the Stiller–Smith mechanism, unlike the slider-crank, contains no higher order harmonic forces. The same is true for higher order harmonic couples. However, a first-order couple does exist in the absence of rotating counterweights. This couple is easily eliminated by placing counterweights on the output shafts so that they are 180 degrees out of phase with the closest output gear [12]. This means that they are also out of phase with each other and contribute no net shaking force. They also induce no moment about the z axis [19].

Linear Bearing Analysis. The most significant difference resulting from a comparison of Stiller–Smith and slider crank mechanisms comes from the linear bearing analysis. A cutaway view of the linear bearing in the Stiller–Smith engine can be seen in Fig. 7. The reader may note that if these bearings are well designed and if the rod is sufficiently rigid, sidewall forces between the pistons and cylinder walls of the engine will be negligible. This will reduce wear and gas blowby past the rings and will negate the need for a piston skirt, thus reducing some moving mass.

In the first working prototype of the Stiller–Smith engine, the rods were round in cross section and ran in plain metal bushings in the engine casing so that they had a long unsupported span. For an engine with acceptable longevity this would

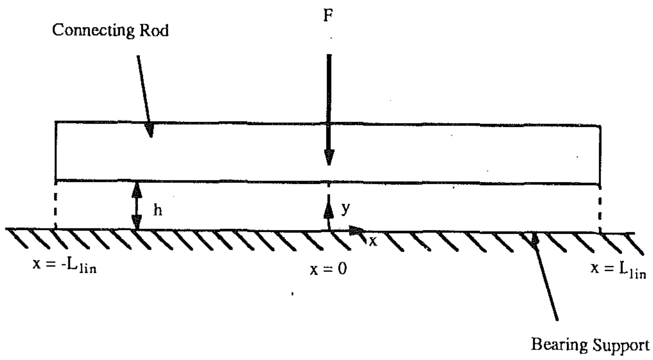

Fig. 8 Simplified linear slider bearing

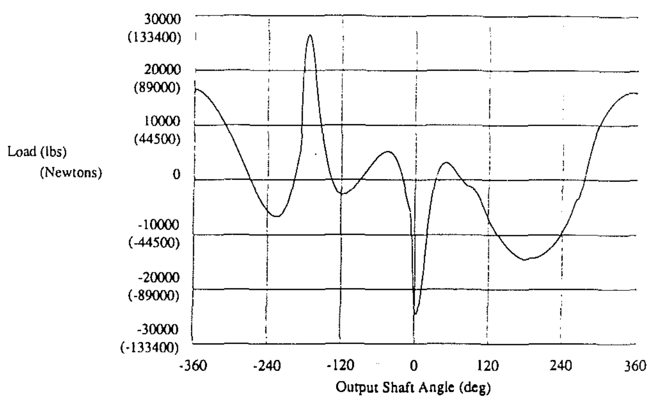

Fig. 9 Load history for linear bearings as a function of output shaft rotation (engine speed 2400 rpm)

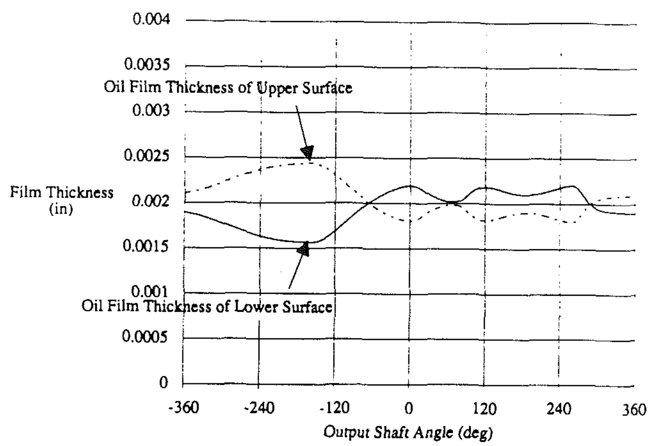

Fig. 10 Oil film thickness of upper and lower bearing surfaces for the linear bearing as a function of output shaft rotation (engine speed 2400 rpm)

prove unsatisfactory, so the new design in Fig. 7 has been proposed. A cross section of the new connecting rod is supported over much of its length by plain bearing surfaces on webs attached to the engine case. Oil may be fed to the bearing surfaces via holes in the webs.

With this configuration the reciprocating rod is permitted to move in the x direction. Vertical movement (y direction) is restricted to a few thousandths of an inch because of the bearing supports. This clearance is necessary for incorporation of an oil film, which will separate the connecting rod and bearing support preventing metal to metal contact. Movement in the z direction is likewise restricted. Once again some small movement in this direction is permitted; however, large displacements are restricted by the presence of the **V**-shape design of the bearing.

The load applied to this bearing is the result of the load imposed by the trammel pin in the connecting rod. As the engine runs, the linear reciprocating rods perform a linear sinusoidal translation at the same time the oil film thickness on each side of the bearing is changing as a result of the loading on the rod. This motion along with a load fluctuating in magnitude and direction pose a very difficult yet interesting lubrication problem. Due to the limited research in this type of bearing and the complexity of the problem, the analysis has been simplified. Consider the bearing system shown in Fig. 5.2. This differs from the true Stiller-Smith bearing in that the base is flat, and not a **V**-trough, and in that a film on only one side of the rod is considered. Power and pressure requirements have prescribed the use of a hydrostatic oil film, and analysis has shown that hydrodynamic (wedge) effects will be small. It is therefore proposed that the force on the center of the rod will be opposed solely by "squeeze-action" of the oil film. It will also be assumed for this analysis that no breakup of the film occurs during the linear motion.

Due to the complexity of the lubrication of this type of linear bearing system, several simplifying assumptions have been made in the solution of the Reynolds equation:

1 The linear connecting rod will not be permitted to move in the z direction.

2 The **V**-shape will be flattened and the bearing will be modeled as flat plates.

3 The oil viscosity is independent of direction.

4 A pressurized oil system will feed oil to the bearing surface via holes in the bearing support. For this reason the bearing surface will be assumed to possess a complete oil film.

With the simplifying assumptions imposed, the Reynolds equation can be reduced to a more workable form [23]. This work can be found in Smith [23] where the bearing load history was described by Fig. 9. The resulting oil film thickness for this analysis is represented in Fig. 10.

Results

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. 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 for the engine used in the investigation may be found in Tables 1 and 2.

Table 3 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.

This table shows that the V-8 mechanism contains over twice as many moving parts as does the eight-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. While both engines have eight cylinders, the Stiller-Smith has only four reciprocating parts. Its pistons are rigidly attached to the connecting rods, one at each end.

Joint Identification. Figure 11 shows the joint (bearing surface) types of the two mechanisms. A breakdown of the joints into bearing type is shown in Table 4 for both mechanisms. The slider-crank engine contains more bearing surfaces, by 42 percent, than the Stiller-Smith engine. The table also identifies bearing surfaces that serve similar purposes in the two mechanisms. Both mechanisms contain eight 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-

Table 1 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 2 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 3 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

Table 4 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

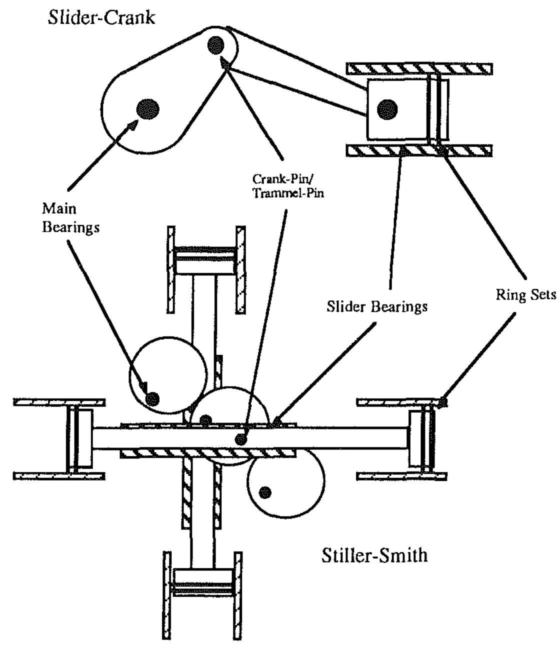

Fig. 11 Mechanism joint identification

Smith. The slider-crank engine therefore has 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.

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-takeoff shaft to the V-8 will require the addition of a minimum of two journal bearings and one higher kinematic pair.

Comparison of Main Bearing and Output Shaft Bearing Load. 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. 12 and 13. 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, bearings 4, 5, and 6 have significantly higher peak loads than do bearings 1, 2, and 3. This is a direct result of 90 percent 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.

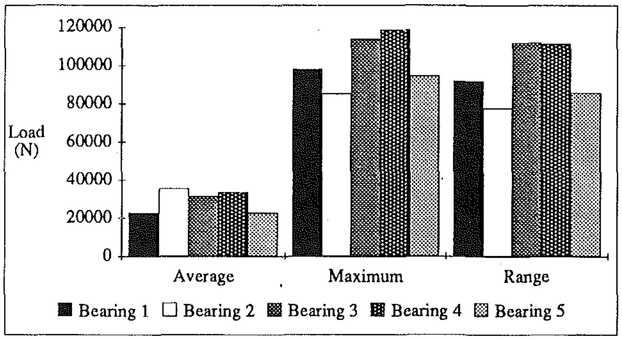

Fig. 12 V-8 main bearing load distribution

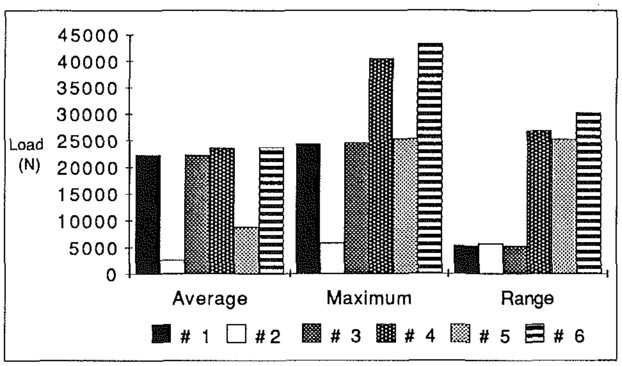

Fig. 13 Stiller-Smith output shaft load distributions

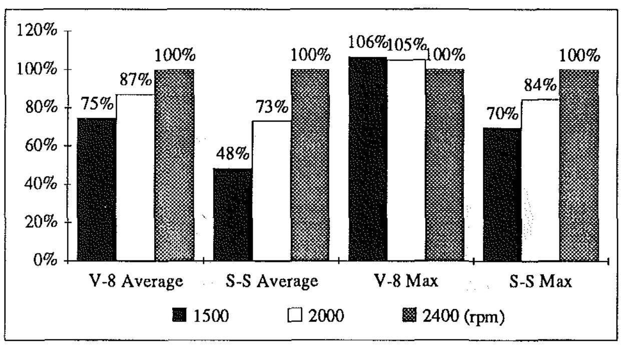

Fig. 14 Effect of engine speed on main bearing loads: normalized to 2400 rpm

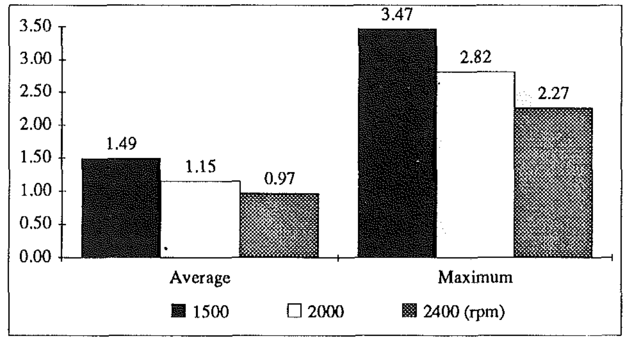

Fig. 15 Ratio of V-8 main bearing to Stiller-Smith output shaft bearing load

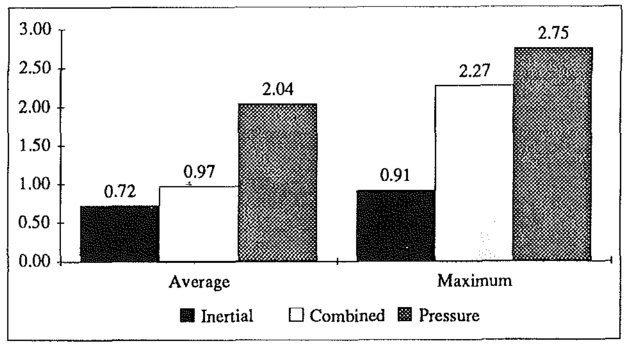

Fig. 16 Relative magnitude of contributing loads: V-8 main bearing load/ Stiller-Smith output shaft bearing load

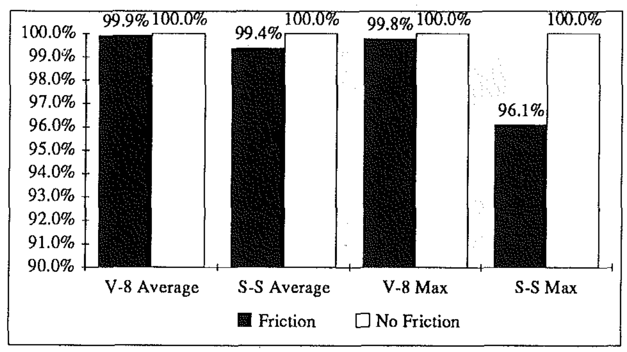

Fig. 17 Effect of friction on main bearing loads

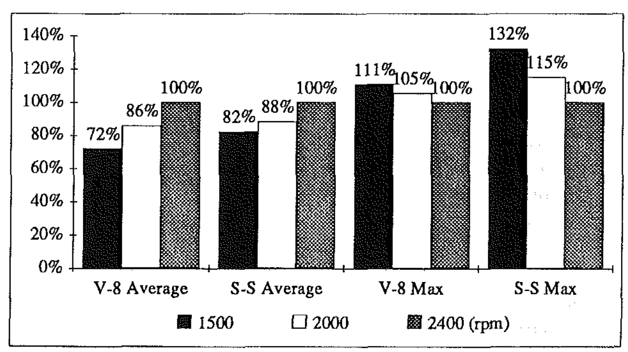

Fig. 18 Effect of engine speed on pin loads: normalized to 2400 rpm

It is assumed that the engine load is held constant over the range of engine speeds investigated where the maximum pressure was 12 MPa (1750 psi) with 30 percent supercharging. The effect of engine speed on the bearing load is similar in both engines. As shown in Fig. 14, 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. 15, 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. 16, 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.

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. 17, 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.

Comparison of Crank-Pin and Trammel Pin Bearing Loads. The trammel pins on the Stiller-Smith and crank-pin in the slider-crank are functionally similar. As shown in Fig. 18,

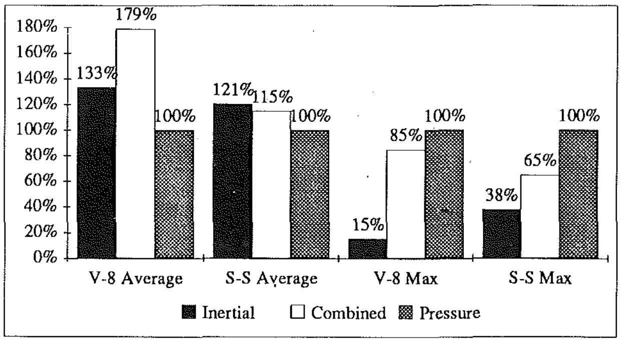

Fig. 19 Relative magnitude of contributing pin loads: normalized by pressure loads

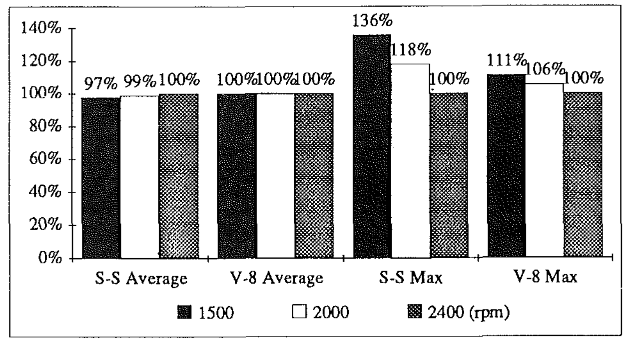

Fig. 22 Effect of engine speed on reciprocating bearing loads: normalized to 2400 rpm

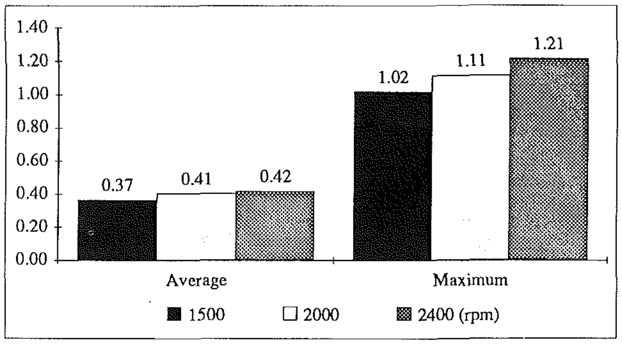

Fig. 20 V-8 pin loads/Stiller-Smith trammel pin loads

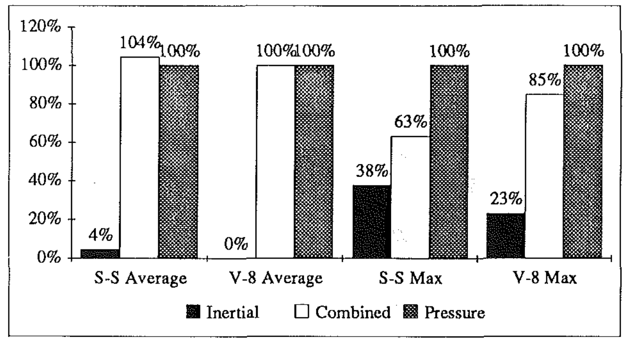

Fig. 23 Relative magnitudes of contributing reciprocating bearing loads: normalized by gas pressure loads

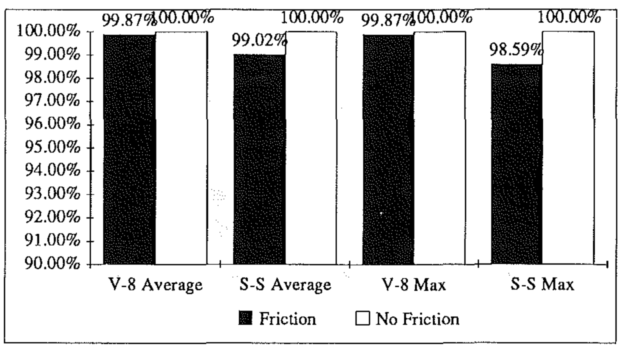

Fig. 21 Effect of friction on pin loads

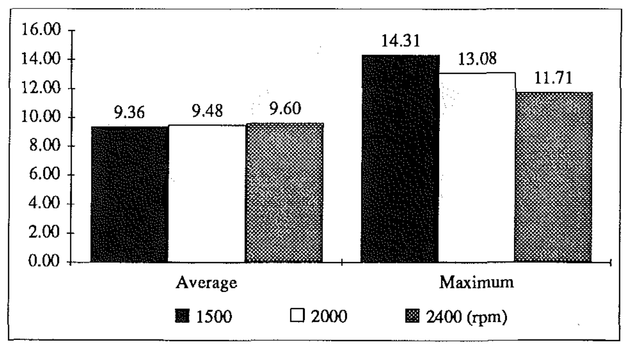

Fig. 24 Stiller-Smith linear bearing loads/V-8 piston sidewall load

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. Figure 19 helps to explain why engine speed has a greater effect on maximum load in the Stiller-Smith engine. Maximum inertial forces are 38 percent of the maximum gas forces in the Stiller-Smith engine. They account for only 15 percent of the maximum gas forces in the V-8. It is also noteworthy that the combined forces are 85 percent 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. 20. Figure 21 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. Given minimum capacity ratio, the Stiller-Smith journal bearings are less likely to have fluid film breakdown and suffer seizure.

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. Figure 22 shows the effect of engine speed on reciprocating bearing load for the two mechanisms.

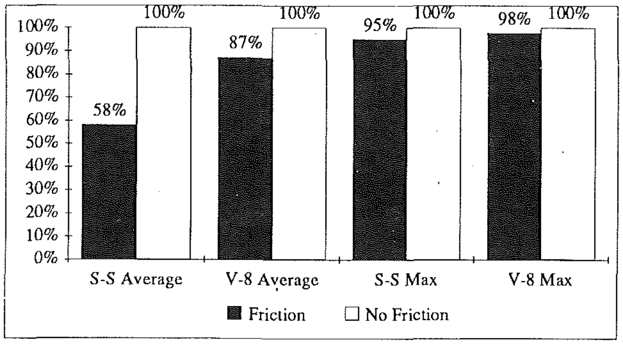

Fig. 25 Effect of friction on reciprocating bearing loads

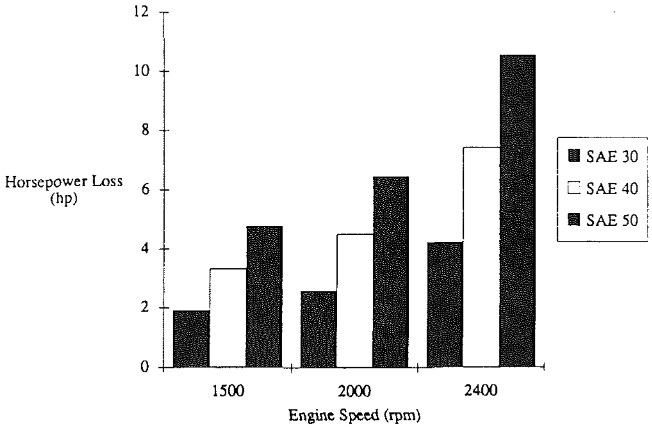

Fig. 26 Horsepower loss as a function of engine speed for linear bearings

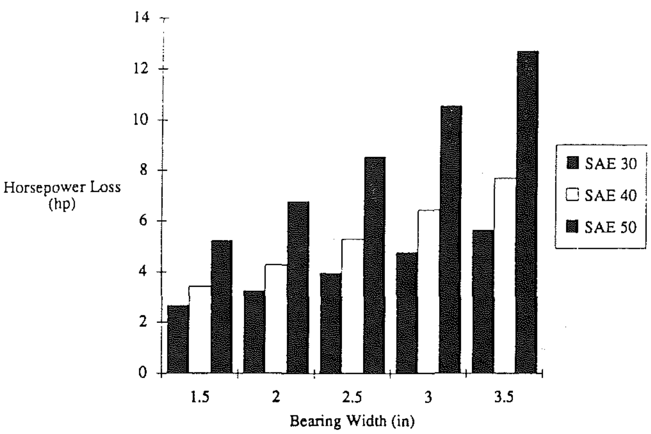

Fig. 27 Horsepower loss as a function of bearing width for linear bearings (engine speed 2400 rpm)

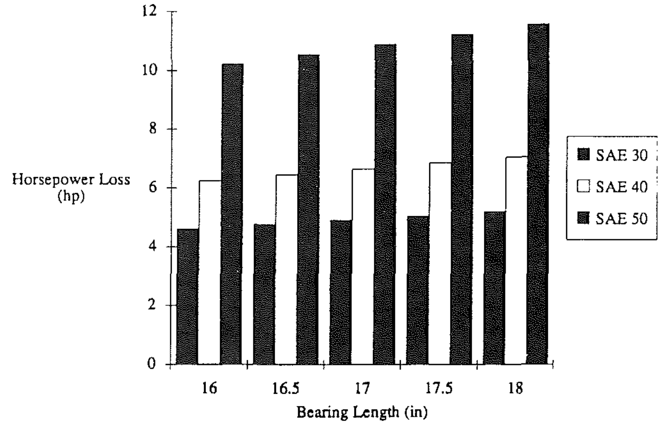

Fig. 28 Horsepower loss as a function of bearing length for linear bearings (engine speed 2400 rpm)

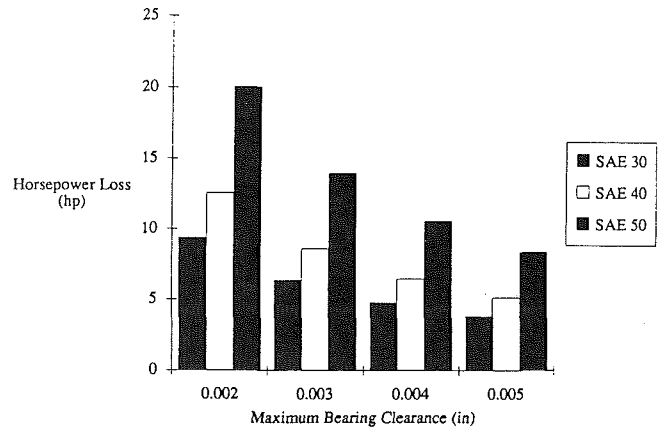

Fig. 29 Horsepower loss as a function of maximum bearing clearance for linear bearings (engine speed 2400 rpm)

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. 23 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. 24, 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. 25.

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.

Friction Loss Implications. One of the strong points of the diesel engine is its good fuel economy. Shigley and Mitchell [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, Furahama and Takiguchi [25] hypothesize 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-crank's piston skirt the loss in horsepower in the Stiller-Smith linear bearings can be predicted as a function of engine speed, bearing width, bearing length, and clearance (Figs. 26-29). While these results are not definitive without experimental backup they do correlate well with friction loss expectations for piston skirts.

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.

For comparable journal bearing surfaces, the performance of those in the Stiller-Smith engine equaled 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 blowby.

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.

References

1 Hunter, W. J., "Internal Combustion Engine," United States Patent No. 1181892, 1916.

2 Bourke, R. L., "Internal Combustion Engine," United States Patent No. 2122677, 1938.

3 Reitz, D. M., "Bourke Type Engine," United States Patent No. 4013048, 1977.

4 Flinn, H. L., Jr., "Linear to Rotary Motion Converter Utilizing Reciprocating Pistons," Great Britain Patent No. 2038984, 1979.

5 Hope, J., "The Geisel: A New Concept for Reduced Fuel Consumption in Internal Combustion Engines," Haeco, Inc., Report Cincinnati, OH, 1983.

6 Smith, J. E., "The Dynamic Analysis of an Elliptic Trammel Mechanism for Possible Application to an Internal Combustion Engine With a Floating Crank," Ph.D. Dissertation, West Virginia University, 1984.

7 Stiller, A., and Smith, J., "Oscillatory Motion Apparatus," United States Patent No. 4641661, 1987.

8 Stiller, A., and Smith, J., "Oscillatory Motion Apparatus," United States Patent No. 4682569, 1987.

9 Smith, J., Craven, R., and Cutlip, R., "The Stiller-Smith Mechanism: A Kinematic Analysis," SAE Paper No. 860535, 1986.

10 Smith, J., Nesbit, S., and Churchill, R., "The Stiller-Smith Cross-Slider Engine: A Balanced Engine Concept," SAE Paper No. 870614, 1987.

11 Nesbit, S., "A Two-Dimensional Analytical Model for Balancing the Stiller-Smith Engine," Thesis, West Virginia University, 1985.

12 McKisic, A. D., Smith, J., Craven, R., and Prucz, J., "Three-Dimensional Balancing of the Stiller-Smith Mechanism for Application to an Eight Cylinder IC Engine," SAE Paper No. 871917, 1987.

13 Taylor, C., *The Internal Combustion Engine in Theory and Practice*, Vol. 1, MIT Press, Cambridge, MA, 1985.

14 Norling, R., "Continuous Time Simulation of Forces and Motion Within an Automotive Engine," SAE Paper No. 780665, 1978.

15 Doughty, S., Smalley, A. J., and Evans, B. F., "Internal Dynamic Force Analysis for V-Type Engine/Compressor With Articulated Power Cylinder Connecting Rod Mechanism," ASME Paper No. 88-ICE-12, 1988.

16 Taylor, C. F., *The Internal Combustion Engine in Theory and Practice*, Vol. 2, MIT Press, Cambridge, MA, 1985.

17 Wilson, C. E., Sadler, J. P., and Michels, W. J., *Kinematics and Dynamics of Machinery*, Harper & Row, New York, 1983.

18 Nahvi, H., "Analytical Model of Friction in a Slider-Crank Mechanism With Hydrodynamic Bearings," Master's Thesis, West Virginia University, 1986.

19 Smith, J. E., and McKisic, A. D., "Stiller-Smith Versus Conventional V-8 Bearing Load and Friction Comparisons," *The Proceedings of the Institute of Mechanical Engineers*, Issue D4, Vol. 203, 1989.

20 Sivaneri, N. T., et al., "Unique Kinematic Features of the Stiller-Smith Mechanism," *Proceedings of the OSU 10th Applied Mechanisms Conference*, Dec. 6-9, 1987.

21 Mucino, V., Sivaneri, N. T., Smith, J. E., Wang, W. G., and Gokhale, M. R., "Dynamics of the 'Stiller-Smith' Mechanism in an Internal-Combustion Engine Environment," *Proceedings of the OSU 10th Applied Mechanisms Conference*, Dec. 6-9, 1987.

22 Patterson, D., "Engine Torque and Balance Characteristics," SAE Paper No. 821575, 1982.

23 Smith, J. C., "Analysis of Plain Bearings in and Eight Cylinder Internal Combustion Engine Utilizing the Stiller-Smith Mechanism," M.S. Thesis, West Virginia University, 1989.

24 Shigley, J., and Mitchell, L., *Mechanical Engineering Design*, 4th ed., McGraw-Hill, New York, 1983.

25 Pohlmann, J. D., and Kuck, H.-A., "The Influence of Design Parameters on Engine Friction," *Combustion Engines—Reduction of Friction and Wear*, I. Mech. E. Publication No. 1985-3, C73/85, Mechanical Engineering Ltd., London, 1985, pp. 67-74.

26 Rosenberg, R. C., "General Friction Considerations for Engine Design," SAE Paper No. 821576, 1982.

27 Furuhama, S., and Takiguchi, M., "Measurement of Piston Friction Force in Actual Operating Diesel Engine," SAE Paper No. 790855, 1979.