Development of the Minimum-Friction Adiabatic Engine

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The Minimum-Friction Adiabatic Engine Program was initiated by Cummins Engine Company and the U.S. Army Tank-Automotive Command to research the feasibility of a significant reduction in an engine's mechanical friction. Basically, three areas are unique to this engine: gas-supported piston assembly; solid-lubricated ceramic ball and roller bearings for the crankshaft, crankpin, wristpin, and turbocharger; and solid-lubricated timing gears, camshaft, and valve- and injector-train bearings. Oil lubrication and water cooling of the entire engine, including the turbocharger, are eliminated. Based on computer calculations, a reduction in the engine mechanical friction of approximately 75% is estimated. Rig tests of each component have been completed and a single-cylinder engine has been successfully operated for a limited time.

Introduction

THE concept of the "adiabatic engine" has stirred great interest worldwide among engineers engaged in the development of diesel engines. Reducing the engine's heat rejection and thereby increasing its "adiabaticity" maximizes the thermodynamic cycle efficiency of the engine. This leads to increased energy in the engine exhaust, which can then be utilized by some additional energy-recovery system such as turbocompounding. At present, many projects to develop the adiabatic engine are in progress in the United States, Europe, and Japan.

One of the critical problems facing the adiabatic engine pertains to high-temperature tribology. When water cooling is eliminated, the liner temperature at the top ring reversal area increases beyond the capability of conventional mineral oil to lubricate it. As the in-cylinder engine heat rejection is decreased further, the liner temperature becomes so high that even the most advanced high-temperature synthetic oil may not be adequate.[1]

As a way of solving this problem, a gas-supported piston was proposed. In addition to providing a solution to the high-temperature tribology problem, the concept also offers a significant reduction in the engine mechanical friction loss. Thus, the Minimum-Friction Adiabatic Engine Program was initiated by Cummins Engine Company and the U.S. Army Tank-Automative Command (TACOM).

A schematic depicting the basic concept of the engine is shown in Fig. 1. Oil lubrication and water cooling of the entire engine, including the turbocharger, are eliminated. For low friction and ease of dry lubrication, ceramic roller bearing are employed at the wristpin, crankpin, and crankshaft main bearing locations. Dry lubrication of the balance of the engine is accomplished by utilizing solid-lubricant bushings and dry-lubricated metallic needle bearings.

Figure 2 shows a schematic of the turbocharger bearing system. A ceramic roller bearing is located at the turbine end of the shaft and a ceramic ball bearing, which takes the thrust load as well as the radial load, is used at the compressor end. Other components unique to the minimum friction engine are discussed more thoroughly in the section on design considerations.

Table 1 lists the specifications of the baseline engine used for this program. The reduction in the mechanical friction loss due to the minimum friction engine concept was estimated against the baseline engine and the results are tabulated in Table 2.[2] It is estimated that approximately 75% of the base engine friction loss is eliminated. The gas-supported piston has a small friction loss when compared to a baseline piston, although the blowby will be about 2.5 times higher than the baseline engine. The 72% reduction in the piston friction was estimated by adding the loss due to the blowby to the friction of the gas-supported piston. The total friction loss of the ceramic roller bearings at the connecting rods and crankshafts was determined using computer analysis.

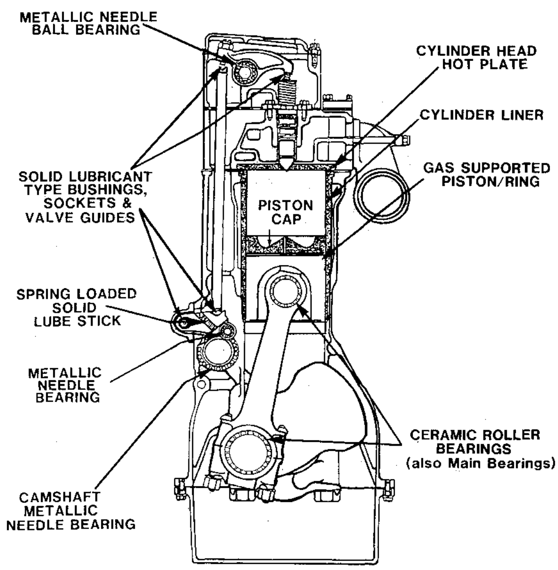

Fig. 1 Minimum-friction engine schematic.

Presented as Paper 84-1396 at the AIAA/SAE/ASME 20th Joint Propulsion Conference, Cincinnati, OH, June 11-13, 1984; received Aug. 4, 1984; revision received March 25, 1985.

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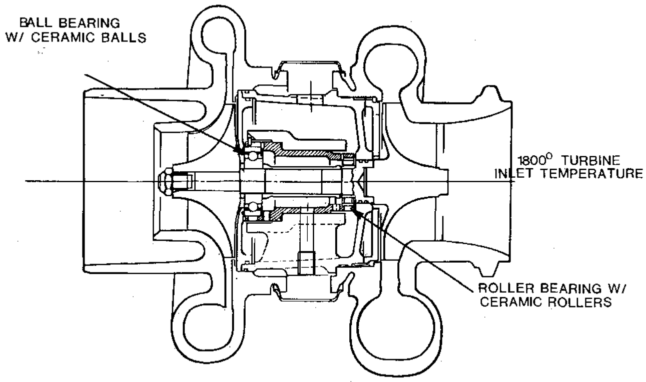

Fig. 2 Schematic of dry-lubricated turbocharger.

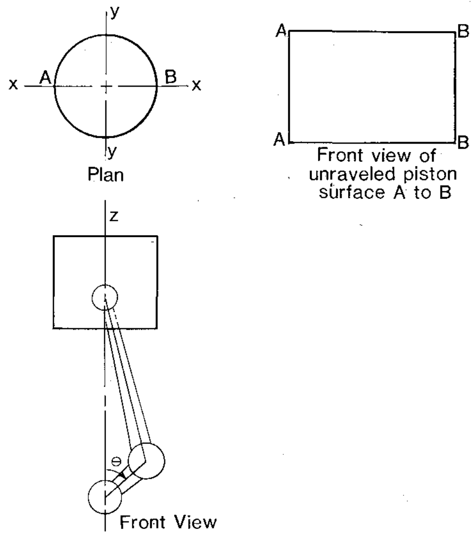

Fig. 3 Schematic of the gas-supported piston computer model.

Table 1 Engine specifications

Engine model	Cummins NTC-400, turbocharged, after-cooled
Displacement	14 liter (855 in.^{3})
Bore × stroke	140×152 mm (5.5×6.0 in.)
Engine configuration	In-line six cylinder
Engine cycle	4 cycle
Combustion chamber	Direct injection Quiescent chamber
Rating	298 kW (400 hp) at 1900 rpm engine speed

Table 2 Estimated reduction of mechanical friction losses of a firing engine at 1900 rpm

Engine component	Base engine Friction, % of total	Friction reduction, %
Piston and rings	53	72
Connecting rod and crankshaft bearings	32	81
Oil pump	11	100
Camshaft and gears	4	0
Total	100	75

Design Considerations

Gas-Supported Piston

A finite element computer code was developed to simulate the piston movement during the entire diesel cycle to study the feasibility of the concept.[3] The program solves the governing two-dimensional, transient, compressible Reynolds equation that describes the gas film thickness between the piston and liner, as well as the pressure distribution of the gas film.

The computation field is shown in Fig. 3. The pressure distribution in the gas film was assumed to be symmetric about plane x-x in the figure, hence only half of the piston surface was analyzed. The dynamics of the piston, including tilting, piston thrust loading, and variations of the in-cylinder pressure during a diesel cycle, were taken into consideration for the calculation. Figure 4 shows the radial movement of the piston as a function of crankangle. Each point on the curve represents the piston position in relation to the cylinder liner. The piston outside diameter is assumed to be 0.025 mm (0.001 in.) from the cylinder liner at 0 deg crankangle and moves from side to side during the engine cycle as shown.

In this case, the piston touches the liner near the end of the compression stroke, at about 335 deg crankangle. However, when the piston thrust load is at its maximum value at about 387 deg crankangle, the calculation indicates no piston-to-liner contact. This is attributed to the high-squeeze film effect due to the high gas film pressure at the time of the maximum thrust load, in addition to the time delay for the piston to respond.

Piston design was optimized using this computer model with the ideal piston design that eliminated the piston-to-liner contacts during the entire diesel cycle.

Thermal and stress analyses were conducted to calculate the thermal deformation of the piston and liner during engine operation.[4] These analyses enabled the proper piston-to-liner clearance to be maintained at the engine operating conditions to ensure sufficient gas support of the piston.

The temperature distribution of the piston is shown in Fig. 5. The three-dimensional nature of the temperature is clearly seen near the wristpin and in the skirt area. The radial deflection of the piston due to these temperatures is shown in Fig. 6. The deflection is much higher at the top of the piston than at the bottom and the pin axis direction expands more than the thrust direction.

The temperature distribution of the ceramic liner is shown in Fig. 7. The top of the liner reaches 625°C (1154°F), while the liner bottom temperature remains around 300°C (575°F). Therefore, the radial deflection of the liner is also different from top to bottom.

Based on the analysis, the piston and liner were designed so that the clearance would remain approximately 0.025 mm (0.001 in.) during most of the engine operating conditions.

Dry Lubrication of Engine Bearings

Rolling Element Bearings

Dry-lubricated rolling element bearings were used where the engine speeds and loads were too severe for dry-lubricated journal bearings. Such places include the crankshaft bearings, wristpin bearings, camshaft bearings, rocker lever bearings, and cam-follower roller bearings.

Carbon graphite is the lubricant used in such bearings. The carbon is assembled into the bearings as rings upon which the rollers rub. Figure 8 shows a cross section of a typical bearing. Specific bearing configurations are determined by the loading direction and magnitude and the allowable bearing size.

In addition to being dry lubricated, the rollers used in the crankshaft main, crankpin, and wristpin bearings are of

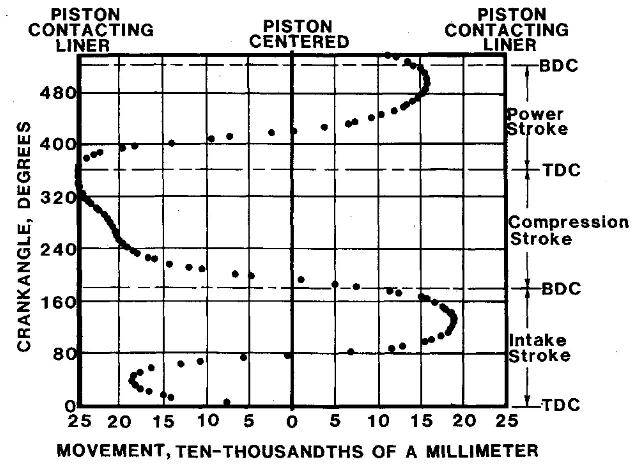

Fig. 4 Radial movement of piston in bore.

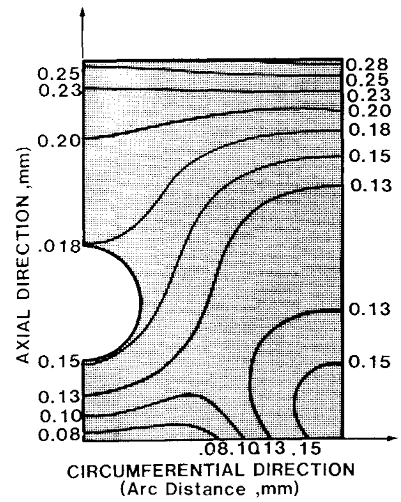

Fig. 6 Radial deflection contours of surface of the ceramic piston.

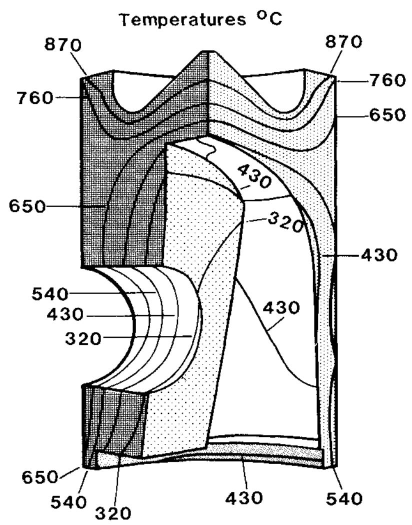

Fig. 5 Temperature contour lines of the ceramic piston.

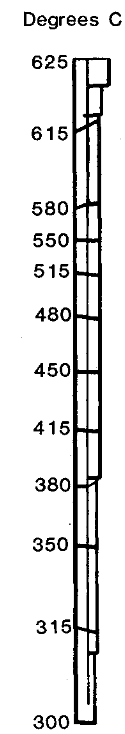

Fig. 7 Temperature variation along the length of the ceramic cylinder liner (shown in cross section).

ceramic material. The high-hardness, high-temperature strength and the lightweight properties of ceramic materials make them an attractive material for these applications. This weight characteristic of ceramic materials is especially important for crankpin bearings where roller skidding limits the maximum engine speed. (Skidding is the tendency of the rollers to slide instead of roll.)

Skidding occurs when the load forcing the roller to maintain contact with both races is smaller than the load forcing the roller away from one of the races. Therefore, the critical speed at which skidding will occur in a crankpin bearing is a function of the rolling element inertia (or the size of the rollers) and the throw of the crankshaft. Figure 9 shows the relationship of these variables.[5]

An additional design consideration is the assembly of the crankshaft bearings to the crankshaft. For this engine, the crankshaft main and crankpin bearings were split to allow them to be placed around the one-piece forged crankshaft.

Sliding Bearings

Sliding bearings were used in various parts throughout the engine. For example, large spherical push tube ends, as shown in Fig. 10, were coated with a commercially available dry lubricant. This same coating was used on the inside of the valve guides.

The cam follower lever pivots in a solid-lubricant bushing. This bushing consists of a steel backing and a bronze bead matrix filled with a polytetrafluoroethylene (PTFE). The cam follower levers are separated by similarly constructed thrust washers. Both the bushing and the thrust washer are commercially available.

Dry-Lubricated Turbocharger

The turbocharger chosen for this program is the Cummins T-46 used on many Cummins NH engines. The target specifications of the turbocharger are shown in Table 3.

The production turbocharger runs in a single oil-lubricated sleeve bearing. Dry lubrication of this bearing was accomplished by using a graphite-lubricated ceramic roller bearing at the turbine end of the shaft and a graphite-lubricated ceramic ball bearing at the compressor end of the shaft. The ball bearing was designed to take thrust loads from either direction. The bearings are shown in Fig. 11.

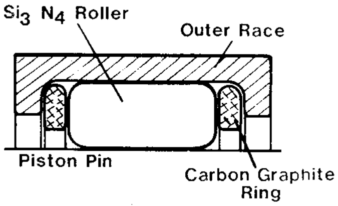

Fig. 8 Cross section of a bearing with ceramic rollers and carbon graphite lubrication.

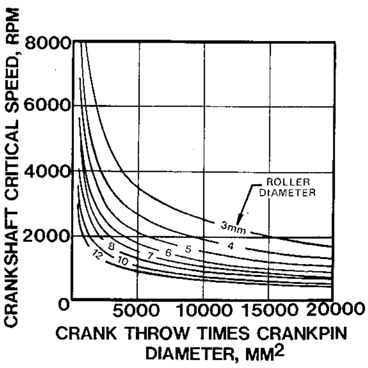

Fig. 9 Engine crankpin roller diameter limitation.

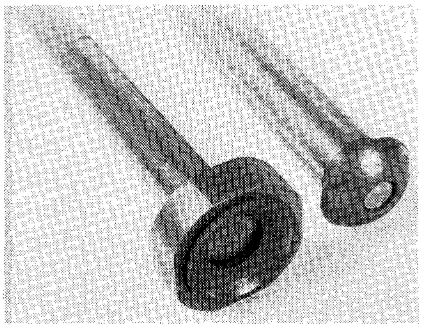

Fig. 10 Dry-lubricated spherical pushtube ends.

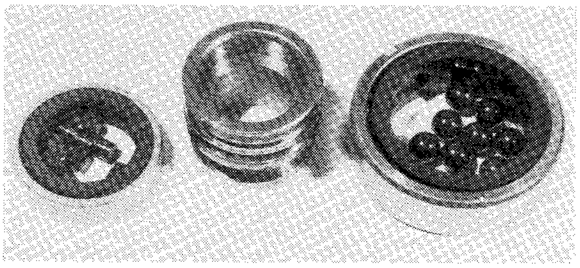

Fig. 11 Turbocharger ceramic roller bearing (left), M-50 steel inner race (center), and ceramic ball bearing (right).

Several design iterations were made on the graphite cage before the bearing system was actually tested in a turbocharger. A special bearing test rig was utilized for this purpose.

All the races were made of M-50 tool steel, except the roller bearing inner race. Two types of materials were prepared for this race: M-50 steel and silicon nitride. The ceramic was used to alleviate the overexpansion of the inner race at the elevated temperatures predicted by thermal analysis.

Table 3 Target T-46 turbocharger specifications

Turbine wheel diameter	11.7 cm (4.6 in.)	
Design shaft speed	65,000 rpm	
Overspeed shaft speed	81,000 rpm	
	Altitude	
	152 m (500 ft)	3,658 m (12,000 ft)
Mass flow rate, kg/s (lb/min)	0.575 (76)	0.454 (60)
Compressor pressure ratio	2.6	3.24
Turbine inlet temperature, °C (°F)	593 (1100)	704 (1300)

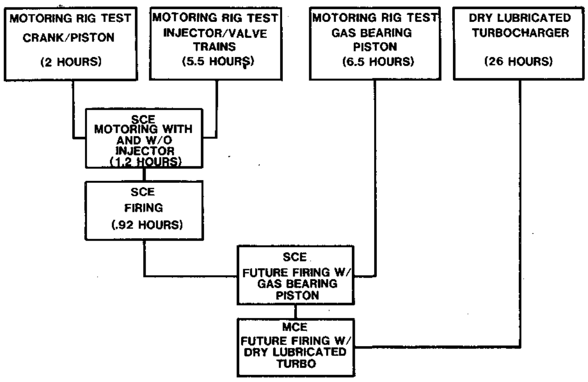

Fig. 12 Minimum-friction engine test flow chart.

Testing

Rig Tests

The flow chosen for developmental testing is shown in Fig. 12. The crankshaft bearings were tested separately from the valve and injector train. When developed, they were placed together into a single-cylinder engine rig for additional testing. Meanwhile, the gas-supported piston and dry-lubricated turbocharger were being tested and developed independently. The tests are summarized below.

Crankshaft Test

A single-cylinder engine test rig was built containing the crankshaft, camshaft, piston, and liner. The piston was ductile iron with a PS101-coated skirt and compression ring. PS101 is a high-temperature solid lubricant developed by NASA Lewis Research Center.[6]

Two motoring tests were run with this rig. Each ran for 1 h up to 900 rpm. The motoring torque at 900 rpm averaged 26.2 N·m (232 lbf·in.). Design modifications were made after these tests that improved the bearing performance in subsequent tests.

Valve/Injector Train Test

A similar single-cylinder engine rig was built for testing the valve and injector trains. This rig closely simulated the engine loading on the injector train, since fuel was supplied to the head and injected into the cylinder as is done in an operating engine.

Motoring tests lasting 5.5 h were completed on this rig with an equivalent maximum crankshaft speed of 1250 rpm. Problem areas were identified and successful design alterations were made.

Engine Tests

Motoring Test

The crankshaft and valve/injector train tests described above resulted in improved dry-lubricated bearing designs in both of these areas. With confidence in the design improvements, these two test rigs were built together into a single-cylinder engine to evaluate the concepts of dry lubrication more stringently.

This engine was successfully motored for 70 min up to 900 rpm with and without the injector installed. The motoring torque averaged 36.4 and 61.4 N·m (322 and 543 lbf·in.), respectively.

The maximum liner temperature was approximately 93°C (200°F) in the area of maximum piston velocity during these tests. This heat generation is due to the sliding of the solid-lubricated piston skirt against the liner—solid lubricants have a high coefficient of friction compared to hydrodynamic oil films.

Firing Test

After this motoring test, the engine was fired and run without load for 55 min up to 1500 rpm. Figure 13 shows the cylinder liner temperatures throughout this engine test. Although fairly warm, the temperatures stabilized after about 28 min at 900 rpm. As the speed was increased to 1500 rpm, the liner temperature climbed rapidly. It was at this time that the ductile iron piston overexpanded in the bore and terminated the test. This overexpansion of the piston was again, in part, due to the heat generated by the sliding friction of the piston skirt.

It is interesting to note that the engine friction (in terms of required fuel input) decreased as the firing test continued. This was probably due to the running-in of the bearing lubricating carbons during the test.

The post-test disassembly showed that sufficient lubrication existed at all bearing surfaces. In addition, the ceramic rollers in the crankshaft main, crankpin, and wristpin bearings operated without a problem.

This test showed the feasibility of a dry-lubricated engine, as well as pointing out the need for a different approach to piston lubrication.

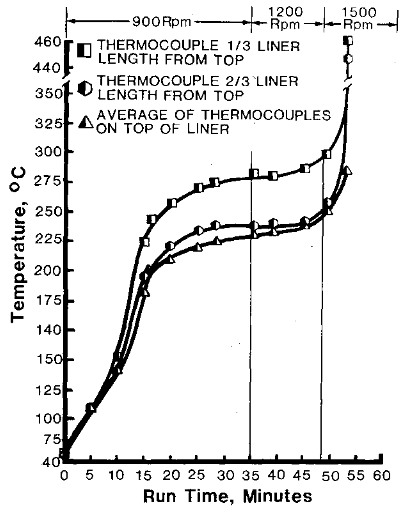

Fig. 13 Firing test cylinder liner temperatures.

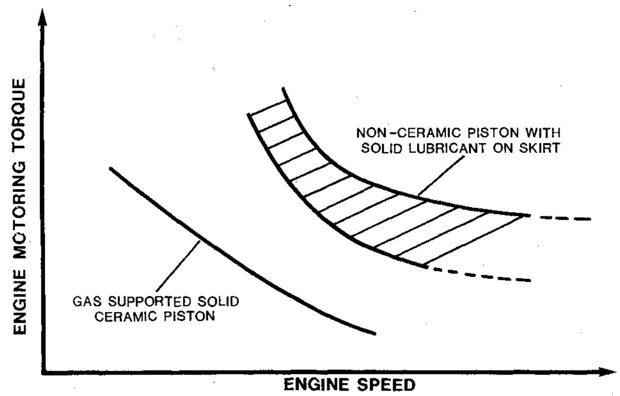

Fig. 14 2.98 kW engine motoring torque.

Gas Bearing Piston Tests

Preliminary Test

To screen the gas-supported ceramic piston design concept, preliminary tests were run using a 2.98 kW (4 hp) gasoline test engine. Figure 14 shows the motoring friction results of this testing. The gas-supported pistons had a motoring friction much lower than the pistons that relied on a solid-lubricant coating on the piston skirt.

After these results indicated that a gas film was supporting it, this piston was fired in the engine for 50 min up to 2000 rpm without failure. The disassembly of the engine was encouraging, since no contact between the ceramic piston and liner was noted.

Since both the motoring and firing test results of this small engine showed that a gas-supported piston was working, further development of this concept was initiated.

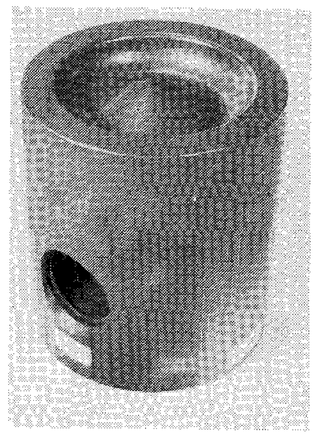

Fig. 15 Gas-supported ceramic piston after motoring test.

Initial Motoring Test

A ceramic piston and liner for a Cummins NTC-400 engine were procured and assembled into a single-cylinder engine rig. This piston was motored for a total of 2 h. Half of this time was with the cylinder top entirely open to atmospheric pressure; the rest was run with only a small hole in the cylinder head open to the atmosphere.

Although some evidence existed that the piston made contact with the liner, the temperature of the liner increased less than 10°C (20°F) during the 2 h test, indicating that a gas film was supporting the piston during most of the piston stroke.

Second Motoring Test

A new ceramic piston and liner were assembled into the single-cylinder engine rig for a second test. This piston was coated with a dry lubricant to lessen the effects of the piston contacting the liner. Also, the piston contour was modified slightly.

The rig was motored for a total of 4.5 h up to 900 rpm. About one-third of this motoring was done with the top of the cylinder open to the atmosphere; the remainder of the motoring was done with only a small hole in the head open

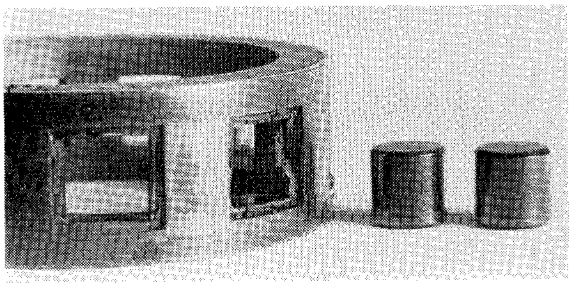

Fig. 16 Roller bearing cage after 26 h of engine testing.

Table 4 Engine rig friction torque comparison,[a] N·m (in.-lb)

Rig descriptions	600 rpm	900 rpm
Oil-lubricated standard piston and crank[b]	13.2 (117)	15.8 (140)
Solid-lubricated iron piston, roller bearing crank and piston pin	15.5 (137)	26.2 (232)
Gas-supported piston, grease lubed roller bearing crank, solid-lubed piston pin	9.0 (80)	12.4 (110)

[a]All rigs are of NH engine configuration without a cylinder head. All friction data include the friction loss of a balancer box that runs on oil-lubricated roller bearings. Each engine used a different balancer box. [b]Extrapolated data.

Table 5 Minimum friction engine oil-less test summary

Components tested	Cycles completed
Camshaft, crankshaft, ductile iron piston, liner	80,000 motoring
Camshaft, valve/injector trains	168,000 motoring
Camshaft, valve trains, crankshaft, ductile iron piston, liner	14,250 motoring
Entire engine	77,400 motoring
	61,500 firing
Gas-supported piston	225,650 motoring

to the atmosphere. The torque for the open-head motoring was 12.4 N·m (110 lbf·in.) at 900 rpm, while that for the small hole was 57.1 N·m (505 lbf·in.).

Disassembly showed very little rubbing of the piston coating. Only near the bottom of the piston was there evidence of the piston rubbing against the liner. This area corresponds to the part of the piston that is below the liner at bottom dead center. In addition, this rubbing occurred only on the pin sides and the minor thrust side of the piston. The major thrust side exhibited no such rubbed areas. Figure 15 shows the piston after the motoring test.

The coating used on this piston is quite soft and easily marked. Since few signs of piston rubbing were evident, a gas film was supporting the piston as it moved in the liner.

Turbocharger

The dry-lubricated turbocharger was first tested in a turbocharger test cell. Operation up to 60,000 rpm shaft speed for approximately 1 h was achieved. Then the turbocharger was mounted on a Cummins NTC-400 engine and tested up to 137 kW (183 hp) at 2030 rpm. During the test, the turbocharger speed and turbine inlet temperature reached 45,600 rpm and 416°C (780°F), respectively. After 26 h of testing, the engine was shut down due to a problem with the heat shield of the turbocharger. Both dry-lubricated bearings looked excellent except for some minor damage to the roller bearing cage. The graphite cage was beginning to chip out at the o.d. edge adjacent to the outer steel shroud as shown in Fig. 16.

Conclusions from Testing to Date

The 55 min firing test shows the basic feasibility of the minimum-friction engine. The gas-supported ceramic piston seems to be a viable solution to the problem of piston skirt friction that causes the piston to overheat. In addition to expanding less for each degree of temperature increase, the gas-supported piston generates less heat than the ductile iron piston during testing due to the gas support. Table 4 is a comparison of the engine and rig friction torque for various motoring test runs. As shown, the gas-lubricated piston has lower friction than the standard piston at both 600 and 900 rpm. The percentage improvements shown here, 32 and 22%, respectively, are actually higher since the friction torque of each test includes the unknown frictional loss of the balancer machine. Although these tests are inconclusive, since they were not fired but motored, results show that the gas-supported piston is a low-friction choice.

The feasibility of an oil-less engine is further shown by the bearing surfaces that were sufficiently lubricated. The wear ratio of the lubricating carbons was 2.4×10^{-9}m worn per 1 m rubbed. Although this value seems a bit high, solid lubricants generally exhibit high wear during the break-in process; the wear rate drops considerably as running continues. At this time, data are not available to accurately gage the expected wear life of these lubricating carbons.

The ceramic rollers in the crankshaft and wristpin bearings operated without failure. These rollers were not damaged when problems developed in other bearing components. Table 5 is a summary of the test cycles each engine component has undergone.

The dry-lubricated turbocharger also operated successfully, running for 26 h without failure.

Acknowledgments

The authors thank Roy Kamo for initiating the minimum-friction adiabatic engine concept and Robert Graham and Dr. Robert Walson for their contributions to the dry-lubricated turbocharger work. In addition, the experimental contributions of Mike Cooper, Jim Dunagan, and Don Spencer are appreciated as are the efforts of Cathy Rhoades and Mary Dougherty in preparing this paper for publication.

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