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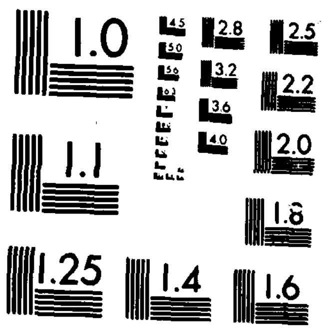

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HIGH-TEMPERATURE LUBRICATION SYSTEMS FOR RING/LINER APPLICATIONS IN ADVANCED HEAT ENGINES

INTERIM REPORT
BFLRF No. 189

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By

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to the 8th power to the 10th power

An assessment of tribological requirements in upper-cylinder area of advanced high-temperature engines was made in terms of critical components and their operating conditions. The ring-liner applications are the most severe since they have both high temperature (>800°F) and large number of sliding cycles ($10^8 - 10^{10}$). This report explores and evaluates various high-temperature tribological systems lubricants, materials and lubricant delivery or supply systems for high-temperature ring-liner applications. Keywords: solid lubricants; low heat rejection engines; friction; wear; composite materials; displacement engines; gas turbine engines.

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FOREWORD

The work reported herein was conducted at the Belvoir Fuels and Lubricants Research Facility (SwRI), Southwest Research Institute, San Antonio, TX, under Contract Nos. DAAK70-82-C-0001 and DAAK70-85-C-0007 and covers the period October 1984 to July 1985. The work was funded by the U.S. Army Belvoir Research, Development and Engineering Center, Fort Belvoir, VA. Contracting Officer's representative was Mr. F.W. Schaekel, Materials, Fuels, and Lubricants Laboratory/STRBE-VF, and the technical monitor was Mr. M.E. LePera, Fuels and Lubricants (STRBE-VF).

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I. INTRODUCTION

A variety of advanced heat engines are under development (Table 1); these engines are either displacement engines or gas turbine engines. The characteristics of the advanced displacement engines are shown in Table 2. Also shown in Table 2 is a list of the critical mechanical components and their operating conditions. A similar table for gas turbine engines is shown in Table 3. A review of the component operating conditions (Table 4) shows that temperatures far above those normally encountered in engine applications are anticipated. In addition, other constraints must be recognized. These are also listed in Table 4. The most difficult constraint is the large number of sliding cycles (10^8 to 10^{10}) encountered. Under such conditions, some method of lubricant resupply will undoubtedly be required since the limit without such resupply is 10^4 to 10^7 sliding cycles. If fluids are used above 600°F (316°C), some system must be devised which prevents lubricant deterioration. If solids are used as lubricants, some lubricant resupply technique must be developed.

The ring-liner applications are the most severe since they have both high temperatures and a large number of sliding cycles. In addition, the geometries are such that effective lubrication is difficult to achieve. The objective of this report is to explore and evaluate various high-temperature lubrication systems (lubricants, materials, and lubricant supply systems) for high-temperature ring-liner applications.

In this study, a review of technical literature was conducted to identify potential lubricant delivery systems. A number of different systems were identified; these are described in Table 5. Each of these systems was evaluated for ring-liner applications. The best materials and lubricants are chosen for each system. The advantages and disadvantages of each system are discussed and estimates made for friction and wear. These are then compared with the requirements for the engine. A selection is made of those systems which hold the greatest promise for future development and recommendations made as to what form that development should take.

TABLE 1. ADVANCED HEAT ENGINES

DISPLACEMENT ENGINES
Adiabatic Diesel, Heavy-Duty
Adiabatic Diesel, Light-Duty
Minimum Friction Adiabatic Diesel (Military)
Rotary Advanced Automotive
Rotary Military
Rotary General Aviation
Rotary Adiabatic
Stirling Automotive
Stirling Free Piston
Spark Ignition LD Automotive
Spark Ignition LD Commercial
Stirling Space Power

GAS TURBINE
Aircraft Turbojets and Fan-Jets
Propjets, Turboshafts (HELO)
Limited Life (RPV's, Cruise Missile)
Military Ground Vehicles
Heavy-Duty Vehicles (Off Road)
Light-Duty AIG (Automotive)
Power Boost Energy Recovery

TABLE 2. ADVANCED DISPLACEMENT ENGINES

		ADIABATIC DIESEL - HEAVY DUTY ON-, OFF-ROAD	ADIABATIC DIESEL - LIGHT DUTY AND AUTOMOBILE	ADV. ADIABATIC DIESEL (MFE) - MILITARY VEHICLE	ROTARY - ADV. AUTOMOTIVE & TRACTOR	ROTARY - MILITARY VEHICLE	ROTARY - GENERAL AVIATION	ROTARY - ADIABATIC
CHARACTERISTIC PARAMETERS AND FACTORS	RATED SHAFT SPEED	2300	3000	2800	7000	3000	7000	–
	POWER RANGE HP (RATED)	400-600 HP	70 HP	460-1500 HP	350-1500	350-1500	120-1900	–
	EXHAUST GAS TEMP. (RATED CONDITIONS) °F	1500	1400	1000	1472°F	1000°F	1000-2000°F	–
	COMPRESSION RATIO	15:1	14:1	16:1	6:1	8.5-11:1		–
	PEAK CYLINDER PRESSURE, PSI	2000 PSI	3000 PSI	2000 PSI				
	TYPICAL LIFE, TBO, HOURS	500,000 MI	B_{10} - 60000 MI	1000 - 2000 HR	5000-1000 HR	1000-2000 HR	2000 HRS	
	REMARKS BMEP, PSI	195 PSI	PRELIM ANALYSIS 240 PSI, 79 MPG (COMBINED FDC)	300 PSI	180 PSI	180 PSI	ARM TIP SPEED 5400 FPM 211 PSI BMEP	PRELIMINARY STUDIES UNDERWAY
CRITICAL MECHANICAL COMPONENTS AND INTERFACES REQUIRING SPECIAL LUBRICATION CONSIDERATION	PISTON, TOP RING LAND TEMP, °F	644	940	1000	NA	NA	NA	NA
	PISTON SKIRT TEMP, °F	–	640	–	NA	NA	NA	NA
	LINER (TOP RING REVERSAL TEMP, °F)	707	520	1343	NA	NA	NA	NA
	RING/RING GROOVE TEMP, °F	–	–	900	NA	NA	NA	NA
	WRIST PIN, BEARING TEMP	300-600 °F	300-600°F	300-600°F	NA	NA	NA	NA
	WRIST PIN, BEARING LOAD, PSI							
	MAIN BEARING TEMP, °F	180-210°F	180-210°F	180-210°F	NA	NA	NA	NA
	MAIN BEARING LOAD, °F							
	LARGE END CONN. ROD BEARING, TEMP	180-210°F	180-210°F	180-210°F	NA	NA	NA	NA
	LARGE END CONN. ROD BEARING, LOAD							
	CROSS HEAD BEARINGS	–	–	180-210°F	NA	NA	NA	NA
	SCOTCH YOKE BEARINGS			180-210°F				
	VALVES, VALVE GUIDES RECIP/SLIDE	800-1200°F	800-1200°F	800-1200°F	PORTED	PORTED	PORTED	PORTED
	ROCKER ARMS, TAPPETS	CONTACT ≅ 3.5 IN 950 CPM 700-800°F		CONTACT 3.5 IN. 950 CPM 700-800°F	NA	NA	NA	NA
	CAMS/FOLLOWERS GEAR/ACCESSORIES	1300 LB LOAD 300000 PSI, HZ STRESS 300-400°F, ROLL & SLIDE < 300°F HIGH HZ LOAD	– SLIDING	1300 LB LOAD 350000 PSI HZ STRESS 300-400°F	NA	NA	NA	NA
	TRANSMISSIONS, CVT TRANSMISSION/ENGINE CONTROL ELEMENTS	C.T.	C.T.	C.T.	C.T.	C.T.	C.T.	C.T.
	APEX SEALS	NA	NA	NA	*1200 EST	*1000 EST	*1000-2000 EST	*2000
	TRACHOID FLANK ROD SEALS	NA	NA	NA	500-600	500-600	500-600	NA

NOTES:
C.T. – CURRENT TECHNOLOGY
NA – NOT APPLICABLE

TABLE 2. ADVANCED DISPLACEMENT ENGINES (CONT'D)

		STIRLING AUTOMOTIVE	STIRLING FREE PISTON GEN. SET, COMMERCIAL	SPARK IGNITION · LIGHT DUTY AUTOMOTIVE	SPARK IGNITION LIGHT DUTY COMMERICAL NATURAL GAS FUEL	STIRLING SPACE POWER F-P
CHARACTERISTIC PARAMETERS AND FACTORS	RATED SHAFT SPEED	4000	3000 CPM	4000-6000	1200-3000	100 CPS
	POWER RANGE HP (RATED)	80 HP (60 KW)	.5-2.5 KW	80-200 HP	50-1000 KW	25-30 KW
	EXHAUST GAS TEMP. (RATED CONDITIONS) °F	EXTERNAL BURNER	EXTERNAL BURNER	1400-1900°F	1500°F	
	COMPRESSION RATIO			8.0-9.5:1	10.5-12:1	
	PEAK CYLINDER PRESSURE, PSI	2700 PSI				2500 PSI
	TYPICAL LIFE, TBO, HOURS	3500 HRS	10000 HRS	3000	20000 HRS VALVES 30000 HRS RING & LINER	100000
	REMARKS BMEP, PSI	1500-1800°F HEATER HEAD TEMP 1320°F WORKING GAS TEMP	HERMETIC POWER SYSTEM, GAS BEARINGS 1400°F MEAN HEATER WALL TEMP	9.25 KG/CM KNOCK DETECTION WITH T-C	TARGET: TBO 5000 HRS	WEIGHT KEY FACTOR HERMETIC SEALED; GAS LUBE; HIGH-CYCLE PRESSURE
CRITICAL MECHANICAL COMPONENTS AND INTERFACES REQUIRING SPECIAL LUBRICATION CONSIDERATION	PISTON, TOP RING LAND TEMP, °F	1000°F	GAS LUBE	C.T.	CONVENTIONAL PISTON ENGINE TECH-NOLOGY	1300°F
	PISTON SKIRT TEMP, °F	100°F	GAS LUBE	C.T.		GAS LUBE
	LINER (TOP RING REVERSAL TEMP, °F)	< 1000°F DOWN 120°F TRR	GAS LUBE	C.T.		GAS LUBE
	RING/RING GROOVE TEMP, °F	210°F	GAS LUBE	C.T.		GAS LUBE
	WRIST PIN, BEARING TEMP	100-210°F	NA	C.T.		NA
	WRIST PIN, BEARING LOAD, PSI					
	MAIN BEARING TEMP, °F	100-210°F	NA	C.T.		NA
	MAIN BEARING LOAD, °F	100-210°F	C.T.			
	LARGE END CONN. ROD BEARING, TEMP	NA				NA
	LARGE END CONN. ROD BEARING, LOAD					
	CROSS HEAD BEARINGS	100-210°F	NA	NA		NA
	SCOTCH YOKE BEARINGS					
	VALVES, VALVE GUIDES RECIP/SLIDE	CGR 2000°F NO VALVES	NO VALVES	C.T.	CRITICAL	NA
	ROCKER ARMS, TAPPETS	NA	NA	C.T.	C.T.	NA
	CAMS/FOLLOWERS GEAR/ACCESSORIES	NA	NA	C.T.	C.T.	NA
	TRANSMISSIONS, CVT TRANSMISSION/ENGINE CONTROL ELEMENTS	C.T.	NA	C.T.	C.T.	NA
	APEX SEALS	NA	NA	NA	NA	NA
	TRACHOID FLANK ROD SEALS	100-210°F HIGH ΔP (2000 PSI) RECIPROCATING	NA	NA	NA	NO ROD OR SEAL

NOTES:
C.T. – CURRENT TECHNOLOGY
NA – NOT APPLICABLE

TABLE 3. ADVANCED GAS TURBINES

		FOR GROUND VEHICLES			
		MILITARY VEHICLES	HEAVY DUTY (ON- AND OFF-ROAD)	LIGHT DUTY AUTOMOTIVE (AGT)	POWER BOOST ENERGY RECOVERY (TURBOCHARGERS, COMPOUND AND BOTTOMING SYSTEMS)
CHARACTERISTICS	PEAK TURBINE INLET TEMPERATURE, °F	2500	2300-2800	2300-2500	1400-1900
	OPERATING SPEED, RPM	22500	34000	8000-100000	> 100,000
	POWER RANGE, LB THRUST OR HP	1500-2000 HP	300-800 HP	100 HP	NA
	PRESSURE RATIO	15:1	5-7:1	4.0-5.0:1	1.5-3.5:1
	TYPICAL LIFE OR TBO, HR	1000-2000	20000	3500	3500
	REMARKS	GT1500; 601	GT 404; GT 601		
CRITICAL MECHANICAL COMPONENTS AND INTERFACES REQUIRING SPECIAL LUBRICATION CONSIDERATION	HIGH SPEED ROTOR BEARING AND DAMPERS	POSSIBLE GAS BEARING		800-1200°F POSSIBLE GAS BEARING	464°F OIL OUT POSSIBLE GAS BEARING
	ROTARY REGENERATOR SEALS & GEARS	2000°F	1950-2000°F 500-1000	1950-2000°F 544-1634°F	NA
	VARIABLE GEOMETRY AND FLOW CONTROL BEARINGS & LINKAGES	YES	YES	YES	400-1000°F
	COUPLINGS & SHAFT SPLINES	C.T.	C.T.	C.T.	C.T.
	SPLINES (CURVEC COUPLINGS)	C.T.	C.T.	C.T.	C.T.
	GEARS	C.T.	C.T.	C.T.	C.T.
	BLADE FASTENINGS	INTEGRAL	INTEGRAL	INTEGRAL	INTEGRAL
	OIL SEALS	C.T.	C.T.	–	300°F
	ABRADABLE SEALS	1000-2000	C.T.	1000°F	–
	REMARKS BEARING THERMAL SOAK BACK TEMPERATURE, °F			800-1200	800

NOTES:
C.T. – CURRENT TECHNOLOGY
NA – NOT APPLICABLE

TABLE 3. ADVANCED GAS TURBINE (CONT'D)

		FOR AIRCRAFT		
		TURBOJETS FANJETS	PROPJETS, TURBO-SHAFT (HELICOPTERS, GENERAL AVIATION)	LIMITED LIFE (RPV'S, CRUISE MISSILE)
CHARACTERISTICS	PEAK TURBINE INLET TEMPERATURE, °F	2800-2700	2400-2700	3000 PLUS
	OPERATING SPEED, RPM	10000-20000	10000-30000	30000-60000
	POWER RANGE, LB THRUST OR HP	15000-50000 LB-T	1000-10000 HP	500-1000 LB-T
	PRESSURE RATIO	20-30:1	25-40:1	6.0-10.0:1
	TYPICAL LIFE OR TBO, HR	10000-15000	5000-10000	< 50
	REMARKS	LIMITED BY USE OF CURRENT MATERIALS AND LUBRICANTS; COOLING AVAILABLE IN LARGER ENGINES		
CRITICAL MECHANICAL COMPONENTS AND INTERFACES REQUIRING SPECIAL LUBRICATION CONSIDERATION	HIGH SPEED ROTOR BEARING AND DAMPERS	500-800°F 3 × 10⁶ DN	500-800°F 3 × 10⁶ DN	1200-2000°F
	ROTARY REGENERATOR SEALS & GEARS	NA	NA	NA
	VARIABLE GEOMETRY AND FLOW CONTROL BEARINGS & LINKAGES	*500-1200°F	*500-1200°F	NA
	COUPLINGS & SHAFT SPLINES	300-400	300-800	–
	SPLINES (CURVEC COUPLINGS)	1000-1200	1000-1200	–
	GEARS	300-400	300-800	–
	BLADE FASTENINGS	1000-1200	1000-1200	INTEGRAL
	OIL SEALS	300-400°F	300-400°F	–
	ABRADABLE SEALS	4000°F GOAL	3000°F GOAL	–
	REMARKS			
	BEARING THERMAL SOAK BACK TEMPERATURE, °F	800		

NOTES:
NA – NOT APPLICABLE

TABLE 4. COMPONENTS FOR POTENTIAL SOLID LUBRICATION IN HIGH-TEMPERATURE DIESEL ENGINES

Components	Temp, °F	Other Constraints
Ring/Liners	600-1100	10^8 to 10^9 Sliding Cycles
Valve/Guides	800-1200	—
Tappets	700-800	1300 lb Load
Cams	350-400	350,000 psi
Rolling Contact Bearings	500-1200	3×10^4 DN
Sliding Bearings	500-1200	—
Splines	1000-1200	Lubrication Methodology
Gears	300-600	Lubrication Methodology

TABLE 5. HIGH-TEMPERATURE LUBRICATION SYSTEMS

System	Description
1. Unlubricated	Piston ring and liner materials are chosen which will meet friction and life requirements without being lubricated.
2. Solid-Film Lubricant	Lubricating films are originally applied to ring and liner materials but do not require resupply.
3. Composite Materials	Materials are used which contain lubricants to reduce friction, wear, and surface damage.
4. Synthetic Lubricants	Conventional hydrocarbons are replaced by more stable high-temperature synthetic fluids along with high-temperature materials. Current lubricant delivery techniques are used.
5. Decomposed Fluid	A once through lubrication system provides a fluid lubricant to a hot surface. The decomposing fluid, both liquid and solid, is an adequate lubricant.
6. Liquid-Carried Suspension	A liquid is used in either a "once through" or recirculating lubricant which contains a solid lubricant or the ingredients of a solid lubricant in suspension. Evaporation or decomposition of the liquid at operating temperatures deposits the solid lubricant.
7. Gas-Carried Suspension	Similar to "6" except that a gas is used as the carrier.
8. Reactive Gases	A gas is supplied to the component which reacts with the surface to form a lubricating film.
9. Stick Transfer Films	A solid stick of lubricant is rubbed against a surface to supply a lubricant film.
10. Sacrificial Stick System	A stick of solid lubricant is inserted into or placed adjacent to a ring or bearing surface. As the bearing surface wears, lubricant is supplied.

II. RING-LINER LUBRICATION REQUIREMENTS

Current diesel engines usually consist of water-cooled cast iron liners with steel piston rings which can be coated with chrome or molybdenum for high-temperature application. The rings are either compression rings which seal the hot gases or oil control rings which seal the lubricating oils and distribute it over the circumference of the liner. Although the number of rings is a function of the combustion pressure, in most cases the rings consist of two or three compression rings and one oil control ring. A wide variety of designs are possible, depending upon the engine in question.

Current lubricant delivery systems consist of oil from the crankcase being supplied to the liner wall via throw-off from the crankshaft and piston pin or by spray jets on the connecting rods. This oil supply is provided per cycle which both cools and lubricates the contacting surfaces.

Studies have been made to define the type of lubrication which exists at each ring-liner position.(1-4)* Although a complete description is still not available, it seems that fluid film or mixed lubrication exist at all positions except top dead center and bottom dead center; here boundary lubrication prevails with one exception. At top dead center and bottom dead center, squeeze film lubrication exists for a brief interval while the lubricant flows from the ring/liner interfaces. The highest wear and friction occurs immediately after top dead center since the pressures are the highest and the piston forces pulling the rings down tend to cock the rings to a position unfavorable to fluid film lubrication. Conditions are also most severe at top dead center where temperatures and ring pressures are the highest.

Ting (5) has studied the wear and lubrication of piston rings and provided the following information which can be used in a general way for ring-liner design studies. Average ring contact pressures are approximately 300 psi, being the sum of the gas forces behind the ring, the ring elastic tension, hydrodynamic forces acting at the ring-liner interface, and the piston side

* Underscored numbers in parentheses refer to the list of references at the end of this report.

thrust. At top dead center, the top ring will reach a pressure of approximately 1000 psi which exists over 10 percent of the cycle. Friction coefficient varies based upon the type of lubrication. Where boundary lubrication exists at TDC, the friction coefficient, f, is approximately 0.08; in other areas, an average value of f = 0.04 can be assumed. Ting also determined typical wear coefficients for truck diesel engines:

$$K^* = 8 \times 10^{-10} \qquad (1)$$

This is an extremely low value and confirms that mixed film lubrication exists during most of the cycle. It is this mixed film lubrication which provides the long wear life in current engines. Liner wear is the most critical since the rings are designed to expand. When the liner wear reaches a value of 0.002 in., performance begins to degrade so this value may be taken as a design limit. As a minimum design limit, 2 mils wear in a 400-hour duty cycle may be selected. This 400-hour life is a minimum value based on the requirement to complete the NATO mission-profile cycle during acceptance testing.

Typical ring-liner operating conditions can be based upon a typical military diesel, the Cummins VT-903TA. A 5.5-in. bore and 4.75-in. stroke piston operating at a speed of 2600 rpm was selected as typical for the comparison of the various lubrication schemes. This gives an average value of velocity of 2058 ft/min.

A typical design of a lubricated and unlubricated (6) piston is shown in Figure 1. Some critical dimensions are also shown. It should be noted that unlubricated compressors use wider rings (3X) so pressures are reduced by at least one third. In addition, a large wear ring is included to take the piston side loads and the weight of the piston. Ring pressures on the wear ring are normally in the 5- to 10-psi range.

---

* This is the dimensionless wear coefficient based on a material hardness of 400 Brinell. K is determined from the equation $K = \frac{hH}{PVt}$ where h is depth of wear; H is the hardness of the wearing material; P, pressure; V, velocity; and t is time.

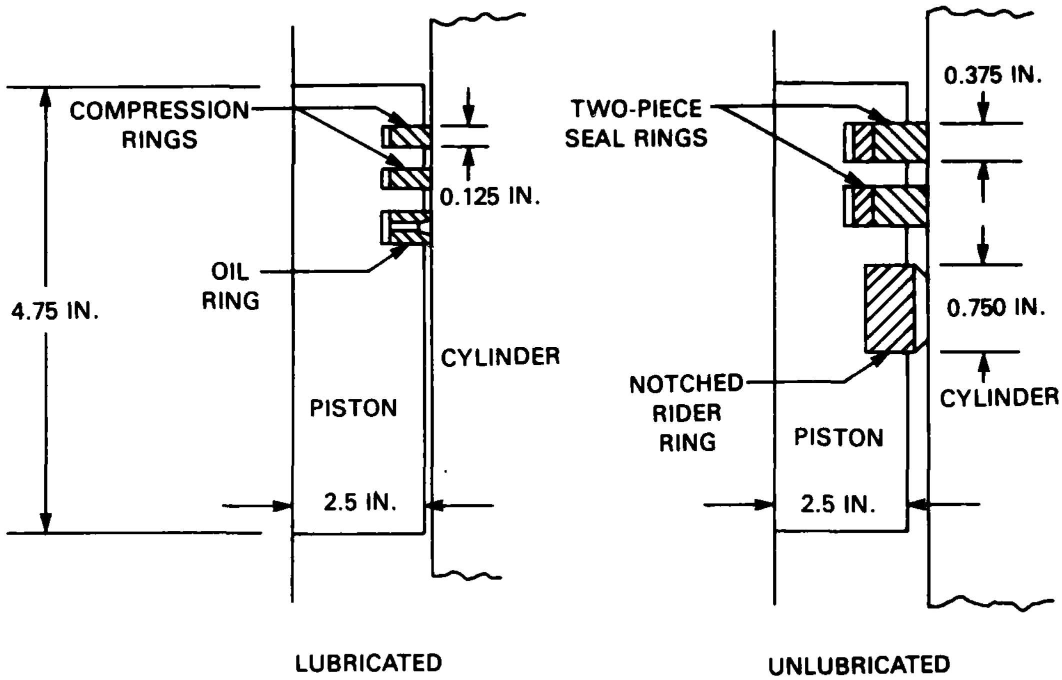

FIGURE 1. COMPARISON OF PISTON CONFIGURATIONS BASED ON PRESENCE OF LUBRICATION (REF 6)

The primary question considered in this study was which of the 13 lubrication schemes most nearly meet the engine requirements using the typical design parameters previously suggested. Four factors must be considered:

1. Ring-liner friction
2. Friction-generated interface temperatures
3. Wear
4. Ring-liner surface damage from sliding

These are discussed in more detail in the following subsections:

1. Friction

Average friction coefficients reported for internal combustion engines generally fall in the range of 0.02 to 0.04. Any increase in this value will result in a loss of engine efficiency and reduce any fuel consumption benefits gained by reducing the heat losses from the engine. The importance of this friction coefficient should not be underestimated. Such low values of friction are characteristic of partial fluid film lubrication. For boundary lubrication where there is no fluid film effects, values of 0.09 to 0.12 may be expected. For lubrication with solids, values usually range from 0.03 to 0.15; however, values of 0.09 to 0.12 are usually found. Values below these are usually obtained from experiments conducted at very high pressures (100,000 psi). Thus, it can be concluded that partial fluid film friction values of 0.04 will be difficult to achieve under any form of dry, boundary, or solid-film lubrication. These higher friction values will not only reduce the engine efficiency, but also increase ring temperatures and surface stresses which will increase surface damage possibilities.

Two conclusions can be drawn from this discussion:

1. Every effort should be made to find materials and operating conditions which yield solid friction values below 0.10.

2. Every effort should be made to provide some degree of fluid or gas film lubrication to further reduce friction and wear.

2. Interface Temperatures

High friction is not only important from an efficiency standpoint, but also from a temperature standpoint. Sliding interface temperatures are directly proportional to friction coefficients. If a dry system is used, friction coefficient will increase; in addition, the cooling effects of the oil will be lost. Fortunately, with appropriate materials, higher temperatures can be tolerated. However, limits would have to be imposed to prevent destruction of the lubricant or damaging the surfaces. Temperature predictions for non-uniform geometries are exceedingly complex; however, some estimates for comparative purposes can be made using a simple linear heat flow model of Figure 2. If the temperature at the point of maximum velocity is determined, all the heat flow is considered to enter the liner (not a bad assumption since the piston is heated and the rings are in poor thermal contact with the liner) and T_o is maintained at room temperature then,

$$T_i - T_o = \Delta T = \frac{fPVl}{K} \qquad (2)$$

where T_i = Liner surface temperature
T_o = Base temperature (cooled wall)
f = Friction coefficient
P = Pressure at ring-liner interface
V = Velocity
l = Thickness of liner wall
K = Conductivity

however, since the ring is only supplying heat to the maximum velocity point, approximately one-fortieth of the stroke time and three rings are being considered

$$\Delta T = \frac{3fPVl}{40K} \qquad (3)$$

The following listing shows increases in interface temperatures above the room temperature as a function of coefficient of friction and thermal conductivity.

f	ΔT Metal, °F (°C)	ΔT Ceramic, °F (°C)
1.0	2460 (1349)	24600 (13649)
0.50	1230 (666)	12300 (6816)
0.25	615 (324)	6150 (3399)
0.10	246 (119)	2460 (1349)
0.04	98 (37)	980 (527)

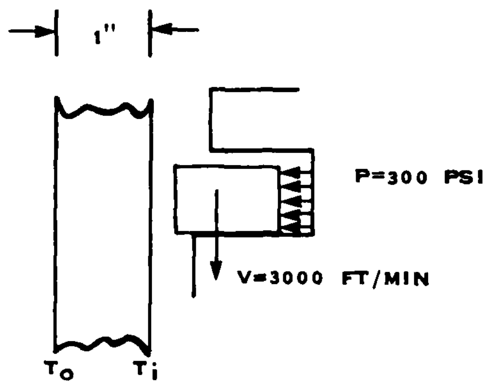

FIGURE 2. RING/LINER HEAT TRANSFER MODEL

Although a much better analysis of ring-liner temperatures is required, this analysis illustrates again the importance of friction control or choosing materials which will withstand the indicated temperature rise. Particularly suspect will be completely unlubricated systems where values of friction of 0.50 to 1.00 are expected.

3. Wear

If adequate friction can be maintained over a life of 400 hours, then due consideration must be given to the wear which takes place. It has been shown (7) that system comparisons can be made if one calculates a desired wear coefficient, K*, for the application and compares this with the values derived in bench test or similar applications. The wear equation is used:

$$h = \frac{KPVt}{H} = \frac{Kpd}{H} \qquad (4)$$

where K = Dimensionless wear coefficient

h = Depth of wear

p = Pressure

V = Velocity

t = Time

d = Sliding distance

H = Hardness

As Ting (5) has shown, this equation can be applied effectively to ring-liner applications by summing the pressure on all rings and calling wear zero when the calculated film thickness is greater than the composite surface roughness (50 μin. was chosen). Without fluid film lubrication, Equation 4 can be used directly as follows:

$$K^* = \frac{hH}{pVt} \tag{5}$$

where h liner = 0.002 in. (wear limit)

h ring = 0.020 in. (wear limit)

H metal = 600 brinell = 0.8×10^6 psi

H ceramic = 3000 Vickers = 4.3×10^6 psi

V average = 2058 ft/min.

t = 400 hours

Number of wearing rings = 3

Using these values, the following K* values can be determined

	K* Liner Wear	K* Ring Wear
Metal/metal combinations	0.30×10^{-8}	8.9×10^{-8}
Ceramic/ceramic combinations	1.6×10^{-8}	48×10^{-8}

To be acceptable, a lubricant/material combination must be found whose K > K*. For oil lubrication under boundary lubrication conditions, Rowe reports K values for metals from 0.1 to 20×10^{-8}. Since 0.1×10^{-8} is lower than any needed K*, it may be concluded that complete boundary lubrication would be acceptable; however, K values must be located for the dry and solid lubricated systems to evaluate their potential for this application.

If a smooth conforming piston and liner were used instead of rings, then the pressure would be reduced and wear depth would be considerably reduced. Assuming a piston thrust load of 400 lb, then the following wear coefficients would be required based on a given wear limit in 400 hours:

Wear Limits (inches)	Required Wear Coefficient (K*)
0.0020	88×10^{-8}
0.0010	41×10^{-8}
0.0005	21×10^{-8}
0.0002	8.8×10^{-8}
0.0001	4.4×10^{-8}

The amount of wear which could be tolerated would be determined by the amount of leakage. Since the leakage increases with the cube of the clearance, it must be held to an absolute minimum; thus

$$K < K^* \sim 5 \times 10^{-8} \text{ for the ringless system}$$

much higher wear coefficients could be tolerated if the piston radial thrust loads could be reduced.

4. Surface Damage

In current engines, final finishes of 20 to 30 μin. are used. During effective boundary lubrication, polishing occurs and finishes are reduced to values of 4 to 8 μin. This polishing promotes fluid film lubrication and if it occurs, the friction coefficient is reduced from values of 0.10 to values between 0.10 and 0.01. This polishing is also necessary to provide effective sealing. In unlubricated, boundary lubricated, and solid lubricated, polishing may not occur. In fact, data show that roughening can occur for a variety of reasons. Unfortunately, a definite limit cannot be placed on allowable surface roughness nor a direct correlation established between roughness and leakage. Rather it is a factor to consider when selecting material combinations, and such data should be reported where possible combinations are being evaluated.

5. Summary

From the preceding discussion, it can be concluded that the following criteria can be used to evaluate the potential of various lubricating systems for ring-liner applications.

1. Does it yield friction coefficient values less than 0.10, preferably 0.04 to 0.06?

2. Is it capable of sustaining a temperature rise of 25°F for each 0.01 value of its friction coefficient for metals and 250°F for ceramics?

3. Will the lubrication system provide wear coefficients of approximately 10^{-8} or less?

4. Will the sliding process damage the surface so that the resulting finish allows excessive leakage?

Using these criteria, the various systems (Table 5) are evaluated in the following section.

III. EVALUATION OF LUBRICATION SYSTEMS

To evaluate the various lubricating systems, data have been accumulated from the literature on the friction, wear, and surface damage of metal and ceramic couples with various types of lubrication. These data are applied to the ring-liner applications in the following paragraphs.

1. Unlubricated Systems

In the past 40 years, extensive data have been accumulated on the frictional behavior of various types of material combinations under a variety of conditions. Some selected data are shown in Table 6, which gives the friction coefficient and wear rates of materials sliding against themselves at room and elevated temperatures.

From these data, some general trends can be noted. Pure ceramic materials give high friction and high wear at both high and low temperatures. The lowest values of friction are about 0.25 for TiB_2. Metal-bonded ceramics give lower friction, wear, and surface damage than ceramics because of several factors: the metal can act like a lubricant for the carbide grains; they are softer and flow to mitigate stresses rather than fracture; and oxides can form which prevent adhesive wear. Certain oxides are soft and can act as solid lubricants.

Tool steels have the best sliding characteristics for operation to 1000°F (538°C); however, they also yield high friction and wear when unlubricated. Generally speaking, the friction and wear properties of metals are controlled by the nature of the friction-generated oxide films. If such films are not formed, high friction, wear, and surface damage will result [see, for example, I650 at 80°F (27°C)]. The best results are found with hard materials that form soft lubricating oxides. Those materials that form soft oxides contain alloying elements of molybdenum, tungsten, copper, cobalt, boron, or vanadium.

Obviously, the best results will be found for metal-bonded carbides containing metals which form lubricating oxides and low friction carbides. K161B is such a material since it is TiC with a nickel molybdenum binder. For unlubricated sliding, primary consideration should be given to such materials. Unfortunately, they have higher values of thermal conductivity than ceramics. If they are used in adiabatic engines, alternative insulating schemes must be devised.

TABLE 6. FRICTION AND WEAR BEHAVIOR OF UNLUBRICATED MATERIALS

Material Combination	T, °F(°C)	H, kg/mm^2	K	f	Ref. No.
AL_2O_3	800(427)		3.07×10^{-2}	0.82	8
	400(204)		1.5×10^{-2}	0.90	8
PSZ	800(427)		4.4×10^{-3}	0.65	8
	400(204)		5.7×10^{-4}	0.96	8
SiC	800(427)		12×10^{-4}	0.78	8
	400(204)		3.7×10^{-3}	0.78	8
AL_2O_3	1600(871)		--	0.70	9
	800(427)		--	0.80	9
ZrO_2	1600(871)		--	0.60	9
	800(427)		--	0.60	9
TiC	1600(871)		--	0.40	10
	800(427)		--	0.50	10
TiB_2	1600(871)		--	0.24	11
	800(427)		--	0.31	11
Cemented Oxide	1600(871)		--	0.25	12
	800(427)	3000	6.7×10^{-4}	0.32	12
K161B	1600(871)		3×10^{-5}	0.28	12
	80(27)	1400	3×10^{-5}	0.34	12
C608	1600(871)		5.6×10^{-4}	0.27	12
	80(27)	1200	7.8×10^{-5}	0.39	12
M2	1000(538)	300	0.40×10^{-4}	0.46	12
	80(27)	580	0.45×10^{-4}	0.67	12
S6	1200(649)	--	3.4×10^{-4}	0.21	12
	400(204)	500	0.9×10^{-4}	0.26	12
I650	1600(871)	--	0.67 x 10-4	0.35	12
	80(27)	300	44×10^{-4}	0.68	12

The values reported in Table 6 are for cleaned surfaces. Much lower values of friction for ceramic combinations are often reported in the literature. These low friction values are due to contamination which is strongly bonded to the surface or contained in the surface pores. Murray (13), for example, found a friction coefficient of 0.09 for Al_2O_3 sliding against itself at room temperature (see Figure 3). As the temperature was increased, a point was reached (250°F)(121°C) at which friction increased to values of 0.80 due to removal or destruction of surface contaminating films. At this temperature, severe surface damage could result.

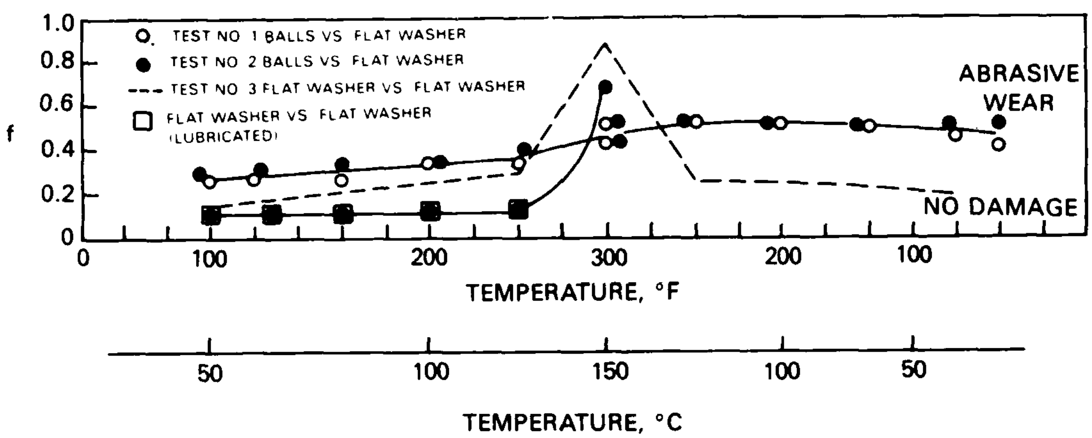

FIGURE 3. EFFECT OF TEMPERATURE ON THE COEFFICIENT OF FRICTION FOR COLD-PRESSED Al_2O_3 USING DIFFERENT SURFACE GEOMETRIES

This point has also been demonstrated in other studies.(14-16) These contaminate films may be either fluid or solid and arise from either handling or processing. Upon cooling, friction may return to the original low value if the source of contamination remains and if the surfaces are not damaged during the high friction sliding. Softer materials such as metals are not as sensitive to contamination since any contaminants are more readily destroyed during the sliding process. The point being made is that unlubricated sliding of high-temperature materials almost always gives high friction and wear; the low values often reported in the literature are for contaminated surfaces which will not be sustained at high temperatures or for long periods of time. There are two exceptions:

1. If the ceramic contains a soft phase which can act as a lubricant, lower friction will result. Note the lower friction for the metal-bonded carbides in Table 2.

2. If soft surface reaction or interaction films are formed on high-temperature materials, lower friction can result. Note the lower friction with oxidizing materials at high temperatures (Table 2).

Without these exceptions, the following friction coefficients apply

Unlubricated ceramics	0.60 - 1.00
Unlubricated cermets	0.20 - 0.50
Unlubricated metals	0.40 - 1.80

In comparing the Table 6 data with the friction ($f < 0.10$) and wear ($K = 10^{-8}$) requirements, it is clear that unlubricated systems are far from satisfactory and some lubricant scheme must be developed.

Fortunately a very simple lubrication scheme may suffice. For example, for alumina sliding against itself in water, Church (17) measured wear rates of 2.8×10^{-8} while Wallbridge (18) reported 1×10^{-8} under much lower loads. Ceramic materials are effectively lubricated with what would normally be considered ineffective lubricants. Diesel fuel (17) gave a wear coefficient of 2.4×10^{-8}. Ceramics are high-energy surfaces and will readily adsorb lubricating films. However, such lubrication system must be planned so that a specified lubricant is delivered at the needed rate to all the required locations in the piston liner area. These lubricants are discussed in later sections.

2. Solid-Film Lubrication

Thin solid films are often applied to lubricate surfaces especially at high temperatures. These films currently consist of molybdenum disulfide, Teflon®, graphite, or other lubricants in either a metal or polymer binder; they also contain other additives for specific purposes. Current commercial materials are useful up to approximately 500°C, but higher temperature films have been developed and used.(19) Friction coefficients are usually in the range of 0.08 to 0.12 so they approach the desired values. Wear lives of some commercially available films are shown in Figure 4 as a function of temperature. It should be noted that the wear lives of the films are short (several million cycles at best).

Finkin (20) has reviewed a wide variety of such test results and has concluded that the following wear rates apply to bonded solid lubricant films:

	K
Normal values	$10^{-6} - 10^{-7}$
Best films	3×10^{-8}

These are mostly MoS_2 films; however, they have received a great deal of research attention and probably represent the best that can be achieved with the film approach.

Although the best films have adequate wear coefficients, they apply to films only 0.0001 inch in thickness. If a 3×10^{-8} wear coefficient is applied to this film thickness, a life of only 0.12 hours will result. The important point being made is that the life of such films are about 10 to 20 million cycles, which are achieved in a short time at 3200 rpm. Obviously, some method of resupply for the solid film is necessary.

3. Composite Materials

The simplest solid lubricant supply system is a self-lubricating composite material. Here a polymer, metal, or ceramic matrix contains a solid lubri-

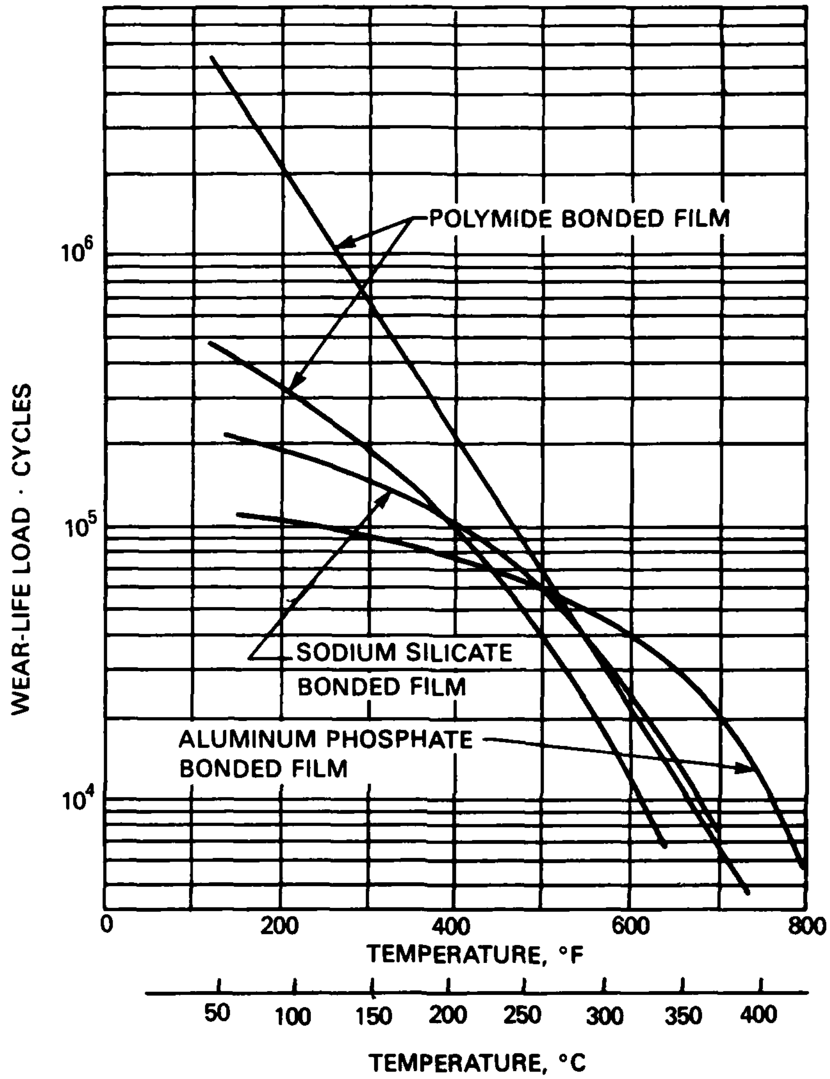

FIGURE 4. EFFECT OF TEMPERATURE ON WEAR-LIFE OF TYPICAL BONDED LUBRICANT FILMS FROM REFERENCE 21

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cant which is supplied as the polymer wears. This wearing of the material to provide sufficient lubricant is a serious limitation since it is wear that one is trying to prevent. However, if adequate rates are achieved to meet ring or liner requirements, then such materials can be used. Wear rates reported in the literature on some typical materials measured under different operating conditions are listed in Table 7.

TABLE 7. TYPICAL WEAR RATES OF SELF-LUBRICATING COMPOSITES

Material	$K \times 10^{-8}$	Reference
Filled Teflons		26
Acetals	70-600	26
Polyesters	9-90	26
Polyimides		
	40	22
	270	23
	20-150	24
	18	25
	45	27
	290	28
PPS	550-240	26
Polyamide-imide	90-900	
Metal Matrix		
Ag/Cu/MoS_2	20	29
Cb/Mo/MoS_2	100	30
Ag/Graphite Fibers	12	31
Cu/WS_2	--	32
Porous Stainless/Glass	--	32
Carbon Graphites	$2\text{-}1500 \times 10^{-8}$	33-36

Polymer materials developed to date are probably limited to temperatures in the 700° to 800°F (371° to 427°C) range so they would be inadequate for high-temperature piston ring application. However, their wear rates illustrate the ultimate of what can be achieved using this approach. At high temperatures, most of the development work has been carried out with the polyimides.

Lewis (22) first reported wear coefficients of 40 x 10^{-8} for temperatures up to 750°F (399°C). While using pressures to 1000 psi, Jepson (23) reported somewhat higher values. Coefficients of friction of approxmately 0.07 were reported. Martin and Murphy (24) reported values of 20 to 150 x 10^{-8} for several polyimide composites. Lewis' results were with graphite, while Martin and Murphy used a variety of lubricants and fillers. Gilthrow and Lancaster (25) added carbon fibers to polyimides and measured wear coefficients of 18 x 10^{-8} which are near the desired range. More recently, other fiber-filled composites have been developed by NASA and Hughes.

NASA in a series of investigations (26-29) studied the friction and wear characteristics of 50/50 mixtures of chopped graphite fibers and a highly crosslinked addition-type polyimide. Wear rates in sliding and bearing tests at room temperature and 570°F (299°C) gave similar wear coefficients of 45 x 10^{-8}. Friction coefficients were 0.10 at room temperature and .05 at 570°F. A similar material has been developed by Hughes and carefully investigated by Mecklenberg.(28) Over a wide range of conditions, a wear coefficient of approximately 300 x 10^{-8} was reported. These materials represent about the limit of current technology for high-temperature applications. Limited work has also been reported using two other high-temperature polymers--polyphenycline sulfide and polyamide imide. Their wear rates are also listed in Table 4. Friction coefficients of these materials are somewhat higher (0.20).

Metal matrix materials offer greater promise for high temperature applications but have received far less attention. In the 1950's, NASA (29) developed silver/copper/MoS_2 composites which had wear coefficients of 20 x 10^{-8}. Somewhat later, Boeing, under Air Force sponsorship (30), developed a series of metal matrix-MoS_2 compounds (Fe/Pt, Fe/Pd, CbMo) and measured wear rates of 100 x 10^{-8} at 350°F (177°C). Variations of these materials are now commercially available. More recently the wear and friction behavior of continuous graphite fiber reinforced copper and silver base alloys were investigated.(31,32) With the correct fiber orientation, low wear rates (K = 12 x 10^{-8}) were found. Other reported work without wear measurements include Cu/WS_2 and Ag/WS_2 and porous stainless steel filled with glass lubricants.

Carbon-graphites have been widely used in high-temperature bearing and seal applications. They can be considered as composites consisting of graphite in a carbon matrix. In the Wear Handbook, McNab and Johnson (33) list wear coefficients from 2 to 1500 x 10^{-8}. For seals, values of 10^{-8} to 100 x 10^{-8} are reported (34) depending upon the counterface material. Against high-temperature alloys, values of 200 x 10^{-8} were found.(35) Lancaster (36) found values of 100 x 10^{-8}, while Paxton (37) gave K's of 14 x 10^{-8} at high velocity and low pressures. Some of these values are near the range required to achieve 400 hours of life; they do, however, have some undesirable properties. They are difficult to fabricate into thin sections and cannot be applied as coatings. Friction coefficients below 0.10 are quite common, and surface damage is not a problem since they polish under sliding conditions.

Work on the development of ceramic self-lubricating composites is almost nonexistent. The fabrication process for ceramic composites requires high temperatures which destroys any low-temperature lubricants currently available. This applies to either sintered or plasma spray processes. Ceramics with sufficient porosity along with adequate strength are not available for infiltration. Ho and Peterson (38) attempted to develop ceramic brake materials using low temperature cements containing solid lubricants without much success. However, the approach is promising and further work is justified in order to achieve the wear rates reported for polymer and metallic materials.

This brief review of the tribological properties of self-lubricating composites shows that the approach may have some potential for development. Friction coefficients are in the correct range and, with appropriate materials, wear coefficients in the range of 10^{-8} to 10 x 10^{-8} range could be achieved. This approach should be explored in detail, particularly as to how such rings could be designed.

New material developement should begin with the selection of the best lubricants in the 500° to 1000°F (260° to 538°C) range (PbO, Graphite + Oxides, Rhenates). Metal and ceramic matrixes should be selected which are compatible with both liner ceramics and the chosen lubricants. Probably the best approach would be co-spraying so that thick films (10 to 20 mils) could be built up on metal rings. One approach which has been used is to first spray ceramic coatings (e.g., Cr_2O_3), which are by nature somewhat porous on the metal rings, and then to coat this ceramic with solid lubricant films by spraying or bonding. Murray (39) used this technique to develop high-temperature gas-bearing surfaces. Another technique is surface modification by ion plating and implantation. Controlled thickness ceramic coatings can be built up on machine surfaces and implanted with lubricating substances.

4. Synthetic Lubricants

The decomposed fluid system approach to high-temperature lubrication has been called a "throwaway system" in which the lubricant is used once and discarded. It is used in high-temperature metalworking (40) and has been studied for sliding and rolling contacts. The essence of the approach is that a lubricant film applied to a hot surface requires a finite time to decompose. During this time, it provides effective lubrication. Lubrication is provided by the fluid; a decomposed oxidized film containing fluid, and a decomposition film.

The primary questions to be answered concern the maximum temperature at which the fluid will lubricate and how good a lubricant is the decomposition film.

Murray (41) ran pin on disk tests in a pot of heated oils at low speeds and loads and recorded the following failure temperatures:

Petroleum	145°C
Di-2-ethylhexyl sebacate	263°C
Phosponate	300°C
Silicate	271°C

Cowley (42) and Hopkins (43) ran similar tests and found that the failure temperature depended upon the operating conditions and the materials. Higher loads and speeds produced frictional heating which reduced the observed failure temperature. Hopkins found a failure temperature of 336°C for a naphthenic mineral oil and up to 385°C for an ester. From these data, it can be concluded that bulk fluid lubrication will be limited to temperatures less than 400°C. Under these circumstances, decomposition films are probably not formed because of limited oxygen availability. Fein (44) found that friction polymer is formed only at certain oxygen (dissolved) to hydrocarbon ratios. If decomposition films are formed and retained at the interface, then higher temperatures can be achieved. Buckley (45) was able to lubricate up to 538°C under the proper conditions using a pin on disk apparatus and a nitrogen atmosphere.

Rolling contact bearings have been effectively lubricated to temperatures of 538°C.(46-48) In these experiments, cold oil droplets (15 per min) were used to lubricate a 20-mm tool steel bearing. With a petroleum fluid, a temperature of 288°C was reached in air and 371°C in nitrogen. With a synthetic fluid, the corresponding temperatures are 454°C and 538°C. By adjusting the oil flow conditions, a temperature of 454°C could be reached with a 1010 turbine oil. Reichenbach (48) reported lower failure temperatures in higher speed rolling contacts. He also found that an air oil mist fails at a much lower temperature which he attributed to insufficient oil for hydrodynamic lubrication or excessive vaporization. Thus, it is clear that a "throwaway" system can be used to high temperatures (538°C); however, the conditions must be carefully controlled to deliver the right amount of fluid without excessive decomposition. Excessive fluid is also undesirable since it prohibits decomposition film formation required for lubrication. The essential parameter is controlled oxygen availability.

Very little data on friction and wear of decomposition films are available in the literature. In an early NASA program (49), the effects on friction of films formed on steel surfaces by decomposition of common lubricants of several types were studied. The films were formed by heating surfaces to which a thin film of lubricant had been applied. The data show that the films were beneficial and reduced friction and wear. Measured friction values were as follows:

Decomposed Lubricant	f
White Oil	0.11
Grade 1010	0.09
Grade 1005	0.10
Diester	0.10
Polyglycol Ether	0.12
Silicone	0.25

The best results were obtained when the lubricant was applied to the surface of the decomposition film. Wear rates or film life was not measured.

From these data, it can be seen that decomposition films are effective boundary lubricants although friction coefficients are higher than desired. This effective lubrication might be expected since many lubrication scientists believe that such films are always formed due to frictional heating and, in fact, provide the friction and wear reduction attributed to boundary lubrication.

Such a system would seem to hold considerable potential for the 700° to 900°F (371° to 482°C) range or even higher. It is only a minor adaptation of an existing technology which has been shown to be theoretically possible. The primary questions to be answered deal with film mechanics and control. What film thickness is required, and how long will it take to build up? Will the film continue to build up and eventually jam the ring? What will happen in the nonrubbing areas--will the piston eventually fill up with soot or decomposition products? Only experimental investigations conducted under conditions which simulate ring applications will answer these questions. The most important question will be the maximum temperatures achievable. With the simplest conventional lubricant system, this would be the fire point which is in the 400° to 700°F (204° to 371°C) temperature range. With synthetic fluids in nitrogen atmospheres, this probably could be extended to 1000°F (538°C).

It would be desirable to have a lubricant which is effective either as a fluid or a decomposition film and to use that lubricant for all engine components. However, if this approach does not extend the operating temperature range, then a special lubricant and lubricant delivery system could be designed for the cylinder area. An approach evaluated by Devine (50) may have merit. He used volatile organic compounds carried in nitrogen to lubricate rolling contact bearings.

5. Liquid-Carried Suspension

In this system, a liquid in either a circulating or "throwaway" system is used to bring another lubricant to the piston ring area. This is one part of a larger matrix "Lubricant/Carrier Suspension Systems" illustrated in Table 8 and discussed in this section as well as the following on gas suspension systems, lube additive in the fuel, and reactive gases.

TABLE 8. LUBRICANT/CARRIER SUSPENSION SYSTEMS

Lubricant Form	Carrier	Delivery System
Solid Particle	Conventional Lubricant	Conventional crankcase re-circulation
Soluble Additive	Nonlubricating Fluids	Jet directed to cylinder walls
Reacting Ingredient	Gas Carrier	Jet directed through cylinder walls
	Fuel	Fuel additive
		Intake air

In general, the lubricant can be either a solid particle such as graphite or any other solid lubricant. It could also be a soluble additive dissolved in water or an organic compound dissolved in a conventional lubricant. The lubricant could also be a specific atomic species which is contained within the carrier structure and reacts with or is deposited on the surfaces to be lubricated.

Carriers include conventional lubricants, a nonlubricating carrier such as water or fuel, or a gas carrier.

A variety of delivery systems could be used either singly or in combination. The first is a conventional crankcase system which provides passages through the connecting rod and wrist pin to supply lubricant to the ring/piston interface. Splash systems are also frequently used. In such systems, fluid control is provided by the oil control rings which distribute the oil and determine its thickness. Advanced systems include jets directed from the crankcase to the cylinder walls or jets directed through the cylinder walls. These jet systems would be especially appropriate if an additional high-temperature lubrication system was needed to supplement conventional crankcase lubrication. These jet systems could be either "throwaway" or recirculating, depending on the nature of the lubricant and the carrier. Other systems include fuel additives or adding the lubricant to the intake air.

Although all these variations seem to present endless permutations or combinations for evaluation, the actual process is much simpler. It begins with the specific solid lubricant. Once a lubricant/material combination is selected which meets the friction, wear, surface damage, and temperature rise criteria and its effective temperature range is defined, then rational decisions can be made as to its form, potential carriers, and supply systems.

Liquid/solid lubricant suspensions are quite conventional, and most common solid lubricants are available in either conventional lubricants or non-lubricating fluids. When solids are added to conventional lubricants, it has been shown (51) that from 5 to 10 percent is necessary to affect friction and wear. Solids can reduce boundary friction from values of 0.10 to values of 0.06 to 0.08. Actually the fluid lubricant adsorption on the surface interferes with the adhesion of the solid lubricant. As a result, only extremely thin films of solid lubricant are built up on the surface which are quickly worn away once the fluid lubricant becomes ineffective. Generally speaking, solid lubricants are added to fluid lubricants as an adjunct to give increased load or temperature capability. However, the question remains as to the effectiveness of a solid film when continuously supplied from a lubricant which is no longer providing any lubricating function. If the lubricant is quickly destroyed, then the solid film should be effective; however, in the transition region, damage and wear may be high. Such a question can only be determined by experiments using the specific materials involved.

It is known that nonlubricating fluids make better use of solids than lubricating fluids. However, such an approach would require an extremely effective solid lubricant and a complete redesign of the lubricant supply system in the cylinder area. Actually, the selection of the system is so dependent upon the lubricant that system choices can only be discussed in a very general way.

One of the main advantages of the liquid-carried suspension is that it flushes away any excess solid lubricant and prevents piston build up. It also distributes the solid better over the surfaces to be lubricated.

A second major advantage is that a viscous fluid is available to provide hydrodynamic or squeeze film lubrication (and low friction) at less than maximum operating conditions. Such a system could be seen as merely an extension of the previously discussed decomposing system.

The difficulty, of course, is that stable suspensions are hard to produce. Once produced, the suspension limits the degree of filtration that can be applied to the system. However, it seems logical to conclude that liquid-carried suspensions should receive major emphasis for ring-liner applications. Two systems can be considered:

1. Use the conventional crankcase lubrication system combined with a lubricant development program to find an appropriate lubricant and additives which will lubricate to maximum temperatures as a synthetic fluid, decomposing film, and residual solid lubricant film.

2. If necessary, tailor the lubricant to the ring-liner application and provide an independent cylinder lubrication recirculation system. This will prevent contamination of the crankcase system and avoid the environmental problems of the throw away system.

6. <u>Gas-Carried Suspensions</u>

If gas reaction systems are not included, then this approach consists primarily of gases carrying solid lubricant particles which are applied through the air intake of engine or by a separate cylinder lubrication system. Such a system has the advantage that once an effective lubricant has been found, it may be used without all the complicating factors of fluid decomposition, desorption, and preferential adsorption. Secondly, extensive lubricant development is not necessary, just grind the lubricant to minimum particles sizes.

Unfortunately, there are so many disadvantages that the use of such a system seems remote. First, it is very difficult to obtain and sustain particles in suspension. The design of such a system must have the capability of adding solid to a gas stream in a very uniform manner. In Reference 52, a variety of techniques were investigated.

1. A lubricant plug system in which the gas flows through a tube packed with a solid lubricant
2. A bed pick-up in which gas flows over a bed of solid lubricant
3. A diffuser in which gas is passed through a diffuser stone into a bed of solid lubricant
4. Gravity feed
5. Mechanical feed by rotating a container filled with solid lubricant which has a fine screen at its periphery
6. Mechanical agitation feed
7. Mechanial wear feed
8. Subliming solid

All of these systems had one thing in common; initial high particle flow which ceased in time due to agglomeration of the powder. Mechanical agitation of the system improved the suspension process but did not solve the problem of nonuniform flow. Although some positive mechanical system certainly can be devised, it would not be simple, for example, Reference 53.

After the particles are in suspension, it is difficult to keep them there. They tend to collect at corners of the flow path and on moving parts where they are held by centrifugal forces. Changes in these forces produce clots of material moving through the system.

In Reference 53, an air suspension of solid lubricant was used to lubricate high-speed, high-temperature rolling contact bearings. Although successful results were obtained, much more lubricant than necessary to lubricate the bearing had to be supplied. This excess material (0.11 grams/min of solid in an airflow of 0.67 cfm) collected in the bearing and caused jamming. The amount needed to lubricate is very small. If, as discussed in a later section, a 0.40 μin. film will last 1.67 min, then for a piston stroke area of 78.5 inches, 0.0018 in.^{3} or 0.143 grams/min is all that is actually needed. This is a very small concentration.

Special attention would also be necessary to get the lubricant to stick to the surface. Unlike fluids which wet surfaces, particles must be trapped. This is more complicated on a vertical surface where gravity does not assist in the process. Furthermore, the air currents in front of the moving piston ring would also displace the solid particles and possibly prevent them from forming films.

Considering all these problems, this approach should be given a low priority for ring cylinder lubrication.

7. Gas-Reaction Systems

One way to avoid many of the problems associated with solid or liquid suspension systems is to lubricate with a reactive gas. In this approach, the gas itself or a gaseous ingredient is used which reacts with the surface to form a lubricating film. Examples of such systems are:

1. Air used at high temperatures to form lubricating oxides at high temperatures.(54-56)
2. H_2S used to lubricate molybdenum.(57)
3. Chlorine- and sulfur-containing gases used to lubricate high-temperature metals.(58-59)
4. Gas lubrication of rolling contacts.(60)

Such systems have always been applied to metals, although there is no reason that the same technique could not be applied to ceramics or ceramic composite materials which contain the necessary ingredients. There are many advantages to such systems. First, there are no carrier problems and no difficulties in attaching the solid lubricant to the surface. Unlike particle systems, no unnecessary material is being carried. Very small amounts of reactants can be added to the gas, and only thin films build up on the wearing parts. With proper material choices, reactions can be limited to the sliding contacts.

Unlike particle or liquid supply systems, gas supply systems are relatively simple. A simple pressurized reservoir could be used to inject the reactive ingredient into the engine air intake or directly into the cylinder.

Fortunately, experience with oxide lubrication at high temperature has shown that the system is selective. If there is a small amount of molybdenum in an alloy, a lubricating molybdate will be formed on the surface in spite of the presence of other oxides from other alloy constituents.

The disadvantage of the system is in film control. If too thin a film is formed at low temperatures, failure will take place; at high temperatures, excessive wear can occur because of film build up. The low temperature range is the lesser problem since frictional heating promotes film formation; nevertheless, there is a limited range of utiliy for each material/reactant combination. The second disadvantage is that only certain lubricants can be applied this way. However, in the temperature range above 1200°F (649°C), oxides are the primary lubricants and they can easily be formed with a gas-reaction system.

The real question which must be answered concerns the choice of lubricants. Those identified to date are not adequate for the whole temperature range 80° to 1500°F (27° to 816°C). To develop this system, it would be necessary to explore improved lubricants which could be applied by reaction techniques. This would probably concentrate on double or tertiary oxides which have been tailored to give the desired properties. Some effective oxide lubricants are shown in Figure 5. Because of the simplicity of this system and its ease of adaptability to the engine, further development is warranted. The development should concern itself with what oxides are good lubricants and what materials will produce them.

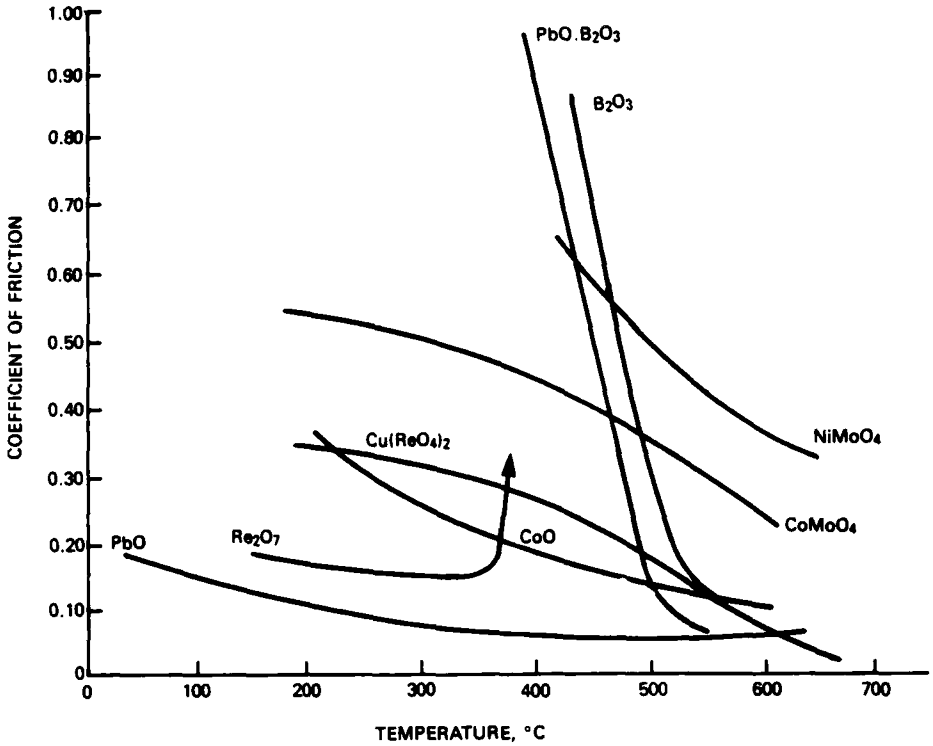

FIGURE 5. METAL OXIDE LUBRICANTS

8. Stick-Transfer Systems

An illustration (61-66) of a stick lubrication system is given in Figure 6. Basically, it consists of a solid lubricant in the form of stick, L_S, which is transferred to a moving surface to lubricate a loaded bearing material, L_B. In this system, one is concerned with:

1. Controlled transfer from the stick to the moving surface
2. Wear rate of the stick
3. Wear rate of the film
4. Wear rate of the bearing material.

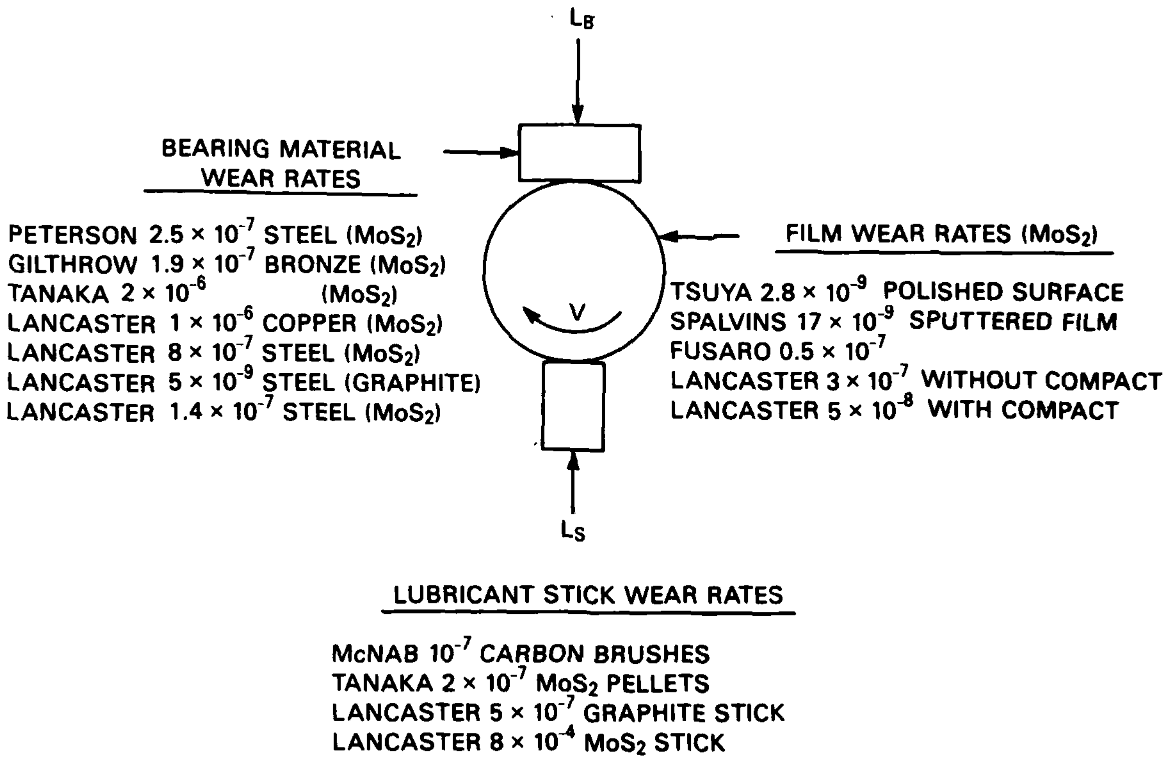

FIGURE 6. WEAR RATES FOR MoS_2 STICK TRANSFER FILMS

In operation, a given film thickness must be maintained to provide effective lubrication. Based upon the work of Fusaro (67), this film thickness should be approximately 1 μm thicker than that necessary to fill around the asperities. This 1 μm is the working lubricant film which is built up and worn away; the solid lubricant in the valleys acts as a reservoir to supply the 1-μm film. If the film wears down to expose the asperity tips, friction will rise to 0.10. If the film builds up, it will quickly be worn away. The critical element in this system is the life of this 1-μm film. If it has a short life, a more rapid rate of supply is needed. This supply process causes bearing material wear by abrasion by the unoriented solid lubricant particles. Some wear rates under different operating conditions, taken from the literature for MoS_2, are listed in Figure 6.

Using these wear rates, estimates can be made of the amount of lubricant required. Using a wear rate of $K = 5 \times 10^{-8}$, a 1-μm (40 μin.) film will last 1.67 minutes as predicted by the previously described wear equation. For a 5 x 5 piston, a 40-μin. film requires a volume of 0.0031 $in.^3$ to be delivered in 1.67 minutes or 0.143 grams/min for each ring. A 400-hour engine would require 3432 grams/ring. This is a volume of 432 $in.^3$ (the piston volume is 78.5 in^3). This suggests that such a system is impractical unless longer life films can be developed. The same simple analysis is probably valid for conventional boundary lubrication; however, the large lubricant supply is provided by constant recirculation.

The bearing wear rates are also seen to be higher than desired. Although these apply to MoS_2 only, it can be seen that the wear rates are no better than those reported for composite materials. A composite material is certainly preferred over a complex lubrication system.

A second difficulty with this system was pointed out by Lancaster.(68) Initially, transfer from the lubricant stick to the moving surface is high; but with continuous operation, this transfer rate is reduced to zero. The reason for this is that the surfaces become polished and the MoS_2 stick is an effective bearing material. Although additives can be used to increase the transfer, this is an additional development effort required before the system could be utilized.

Other lubricants may give better performance and should be evaluated; however, MoS_2 is one of the best solid lubricants resulting from years of research. Finding an improved material may be difficult.

This is not to say that stick transfer films are always impractical. For many applications which do not accumulate a large number of cycles (160 million), this delivery system may be adequate.

9. Sacrificial-Stick System

In the previous system, the lubricant stick and the bearing material were independently loaded against the moving shaft. In this system, the stick and lubricant are loaded together as shown in Figure 7, an example taken from the literature. Such a system is very much like the composite material except that the lubricant occupies a larger volume. The difficulty with the system is the same as that with the composite material, namely that the bearing material must wear to provide the lubricant. However, it does provide an advantage; sacrificial-stick systems do not require the extra processing step required in the manufacture of composite materials. This will be a major advantage when working with ceramic materials. A ceramic piston ring shown in Figure 6 using an oxide lubricant could be easily fabricated by plasma spraying the lubricant, while a composite material combining the two could be impossible to fabricate.

The main factor to be determined is how effective such a system would be. Murray (69) has evaluated this effect in developing piston rings for unlubricated cryogenic compressors. He found that this approach reduced wear by a factor of approximately 10 with ceramic materials. Ho (38) investigated the wear rates of potassium silicate buttons filled with carbon inserts of various volume percentages. His data are shown in Figure 8. In a comparable load range, wear was reduced by a factor of 6 to 10 over the unfilled (0 percent) specimen with the best combinations (25 to 29 percent). Abe (70) measured the wear of bronze bearings with and without lubricant plug inserts. Wear reductions by a factor of 16 were reported with the best lubricants.

These results, like the others, do not predict what can be accomplished at the present time since they deal with low-temperature lubricants; however, they do give an indication of what might be accomplished with ceramic sacrificial systems. If this past work is duplicated, wear reductions by a factor of 10 may be expected. This is significant when applied to the wear rates of ceramic or metal-bonded ceramic materials.

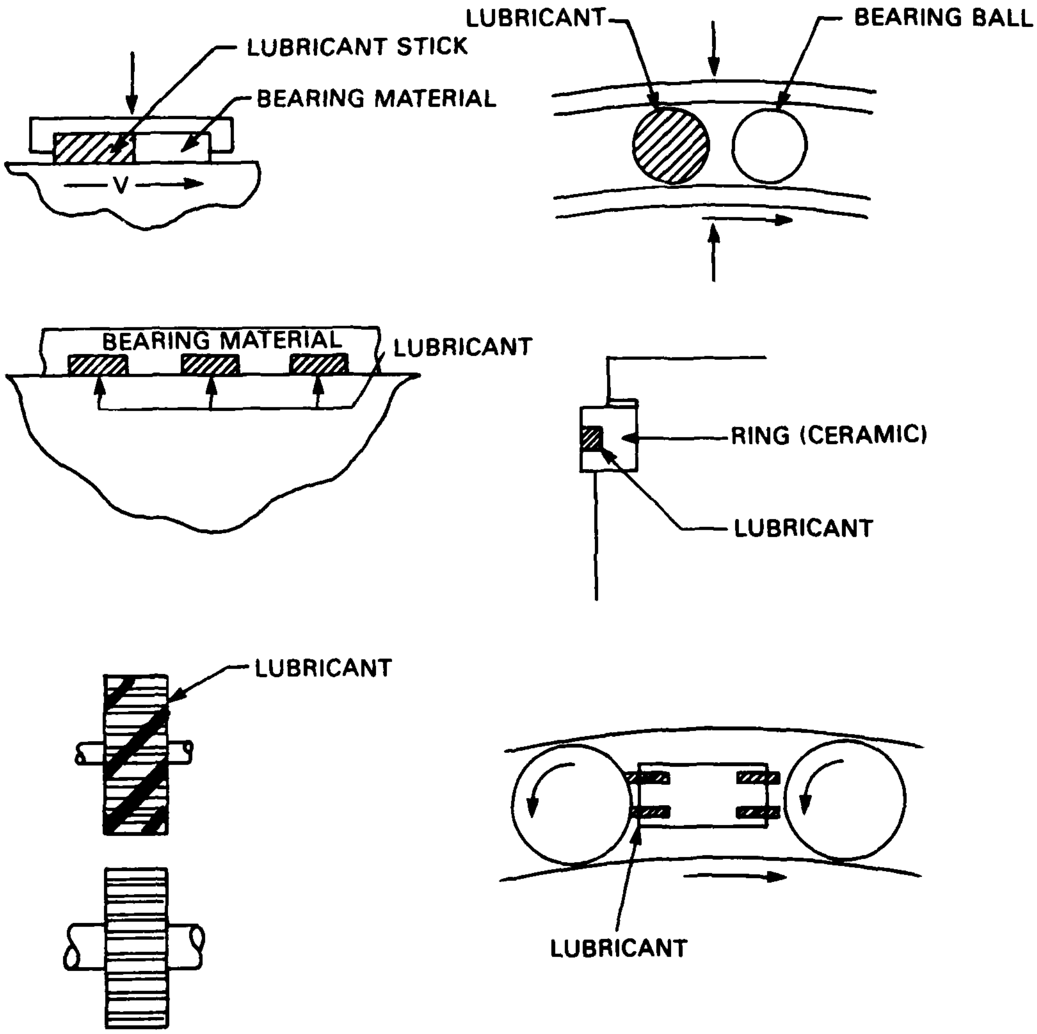

FIGURE 7. SACRIFICIAL-STICK SYSTEMS FOR SOLID LUBRICANT DELIVERY

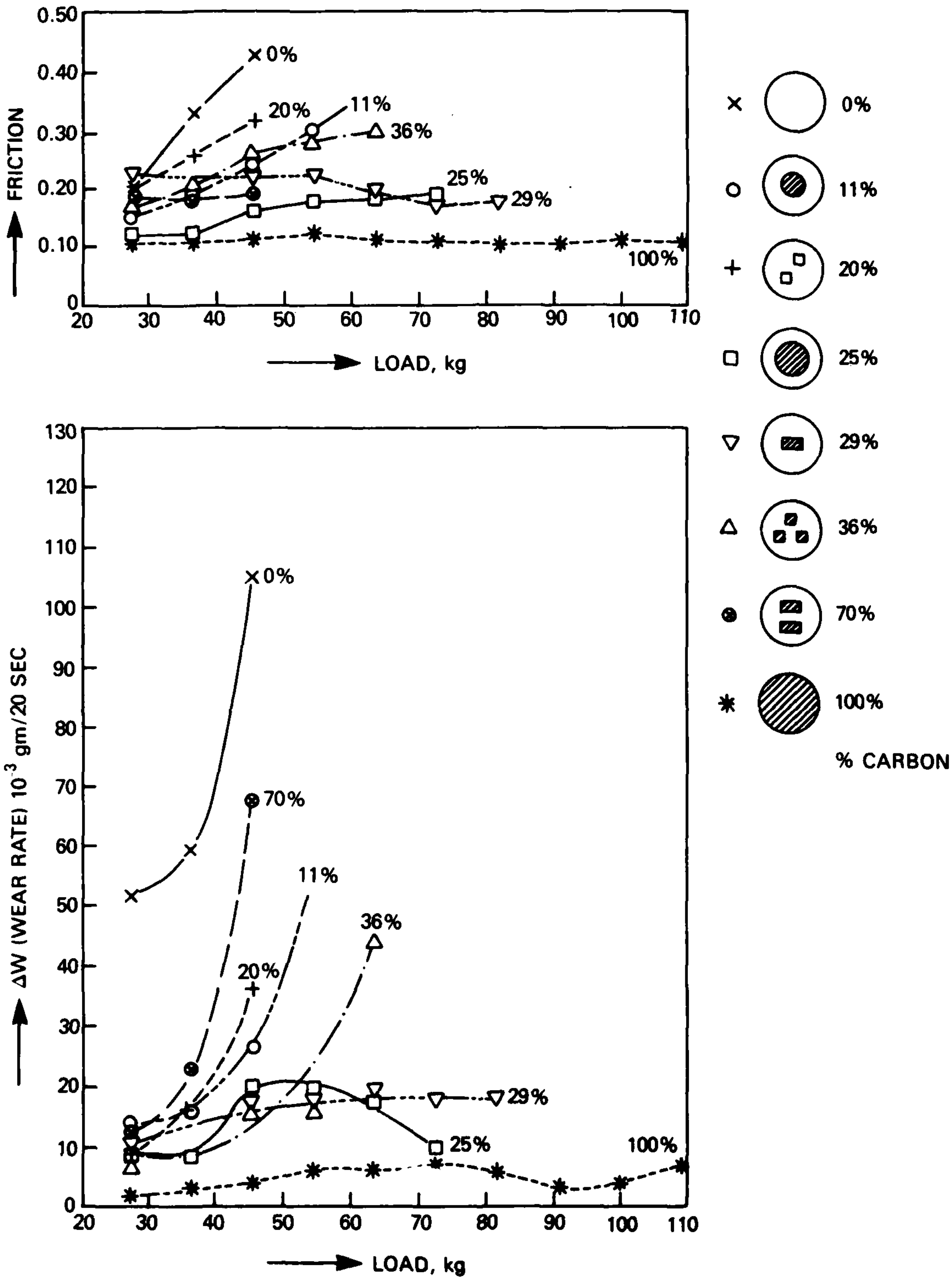

FIGURE 8. THE EFFECT OF LOAD ON THE FRICTION AND WEAR BEHAVIOR OF A POTASSIUM SILICATE CEMENT FILLED WITH CARBON INSERTS

It is an important concept that solid lubricants be applied to ceramic materials. The higher interface pressures and damage-resistant capabilities of ceramic materials allow the use of poorer solid lubricants or solid lubricants with limited temperature range capability. Sacrificial systems could be effectively used in combination with a liquid system, in which the liquid systems provide the low-temperature fluid film and the sacrificial system, the low-temperature boundary lubrication, and the high-temperature lubrication.

10. Synthetic Liquid Lubricants

Liquid lubricants for high-temperature applications may be petroleum based or synthetic. Table 9 presents relative performance areas of synthetic and petroleum (mineral) oils which have outstanding high-temperature stability.

Table 9 can be summarized as follows:

Main Strengths and Limitations of Principal Synthetics

Synthetic Type	Temperature Range, °F	Main Strength	Limiting Characteristics
DAB	-40 to 350	Low Temperature	Volatility - VI
PAO	-85 to 450	Wide Temperature	Additive Solubility
PAG	-30 to 350	Water Versatility	Lubricity
Diesters	-65 to 400	Solvency/ Detergency	Hydrolysis-Antirust
Polyol Esters	-75 to 500	Wide Temp	Hydrolysis-Antirust
Phosphate Esters	-15 to 500	Fire Resistance	Low VI Solvency
Silicones	-100 to 500	Very High VI	Lubricity

In addition to these synthetics, polyphenyl ethers are available; however, they do not have adequate low-temperature properties and are extremely expensive.

Organic esters of the sterically-hindered polyol type have excellent high-temperature oxidation resistance while retaining good low-temperature fluidity and have received much recent attention as potential high-temperature diesel engine (HTDE) oils. Since even the best synthetic lubricants decom-

TABLE 9. RELATIVE PERFORMANCE OF SYNTHETIC OILS VS. MINERAL OIL

	SYNTHETICS							
	Synthesized Hydrocarbons		Organic Esters			Polyalkylene		
Properties	Mineral Oil	Polyalpha-Olephin(PAO)	Dialkylated (C_{12}) Benzene (DAB)	Dibasic	Hindered Polyol	Glycol (PAG)	Phosphate Ester	Silicone Fluid
Viscosity Temperature Properties (VI)	Fair	Good	Fair	Very Good	Good	Good	Poor	Excellent
Low-Temperature Fluidity Low Pour Point	Poor	Good	Good	Good	Good	Good	Fair	Good
High Temperature Oxidation Resistance, with inhibitors	Fair	Very Good	Good	Good	Excellent	Fair	Fair	Good
Compatibility with Mineral Oils	Excellent	Excellent	Excellent	Good	Fair	Poor	Fair	Poor
Low Volatility	Fair	Excellent	Good	Excellent	Excellent	Good	Good	Good
Stability in Presence of water (Hydrolytic Stability	Excellent	Excellent	Excellent	Fair	Fair	Very Good	Fair	Good
Antirust Properties with inhibitor	Excellent	Excellent	Excellent	Fair	Fair	Good	Fair	Good
Additive Solubility	Excellent	Good	Excellent	Good	Good	Fair	Good	Poor
Elastomer Swelling Tendency Buna Rubber	Light	Nil	Light	Moderate	High	Light	High	Light

pose in the 800° to 1000°F range, they will probably be inadequate for the AIPS engine where top piston ring reversal temperatures in excess of 1100°F are expected.

Several organizations have been recently active in HTDE lubrication research. (71-75, 82) In the following section, the HTDE liquid lubrication work of the following organizations is reviewed: U.S. Army Fuels and Lubricants Research Laboratory, Cummins/TACOM, Komatsu, Ltd., and Garrett Turbine Engine Co. (Compound Cycle Turbine Engine program).

The U.S. Army Fuels and Lubricants Research Laboratory (AFLRL) under contract to U.S. Army Belvoir Research and Development Center, Fuels and Lubricants Division has conducted an investigation of "High-Temperature Lubricants for Minimum-Cooled Diesel Engines.(71) This work was jointly funded by U.S. Army Tank Automotive Command, DRSTA-RG, Warren MI, 48090, and U.S. Army Belvoir Research and Development Center, STRBE-VF, Ft. Belvoir, VA 22060.

In this project, lubricant performance at high cylinder wall temperature (630°F CWT) was defined using an uncooled single-cylinder diesel engine operated at conditions which simulate a minimum-cooled/adiabatic diesel engine. The following lubricant-related problems were identified:

- Lubricant oxidation--oil too thick to pump; corrosive products formed and bearings attacked.
- Lubricant volatility--high oil consumption, oil thickening.
- Engine deposits--ring sticking.

Many high-temperature candidate lubricants of varying chemical composition were evaluated in the uncooled single-cylinder diesel engine. Two promising synthetic lubricants were identified; however, both were still deficient in oxidation stability/oil thickening properties, as severe oil degradation occurred by 50 hours when operating at 630°F CWT. This work is continuing at AFLRL with current effort directed at extending the upper temperature use limit of liquid lubricants and developing an understanding of the oil degradation mechanisms in the ring zone area of an HTDE.

Realizing early in the adiabatic diesel engine program that HTDE lubrication was a potential technical barrier, Cummins Engine Co. and TACOM have been conducting research in this area.(72-81) In 1983, Cummins reported that a minimum cooled big cam NT engine with a ring coating/liner coating/lubricant combination of chrome carbide/RC-6/Lubricant B (polyol ester base stock) proved to be successful for a 250-hour endurance run with average liner top ring reversal temperature of 371°C at rated conditions of 298 kW and 1900 rpm. Average oil consumption was 1.37 gr/kW-hr.(81) In recent work in the VTA903 low heat rejection engine (LHR-600), they have operated for 250 hours at 450-475°F (230-245°C) liner temperature using a special synthetic lubricant blend of 65 percent polyol ester and 35 percent triaryl phosphate. Very low engine wear was observed in this 250-hr test. This lubricant blend has been operated for 1 hour at 1100°F (593°C) TRR, at which time problems were encountered due to high blowby. Figure 9 is a thermogravimetric analysis of this type of polyol ester based synthetic lubricant. The thermogram shows weight loss and the rate of weight loss as a function of temperature when a small sample of polyol ester based lubricant is heated in air at a programmed temperature rise rate of 5°C/min. It is obvious that this oil starts degrading rapidly after 250°C and the maximum rate of degradation occurs at approxiamtely 275°C. At 290°C, only 20 percent of the original mass of the lubricant remains in the condensed phase. Clearly, additional work on HTDE lubricants is needed for operation at 1100°F TRR.

Komatsu, Ltd. has reported on their high-temperature tribology efforts. (82,83) They investigated the high-temperature performance of a petroleum and an ester synthetic oil at 662°F CWT and found that the synthetic oil produced less piston ring groove carbon filling than the petroleum oil. Komatsu also reported that the high top ring reversal temperature of the heat insulated engine caused severe side wear of the liner at TRR.

Garrett Turbine Engine Co. addressed the need for high-temperature liquid lubricants in their development of the compound cycle turbine engine (CCTE) for potential cruise missile applications.(84) For 800°F CWT, Garrett's approach was to lubricate with a clean-burning ash-free lubricant based on MIL-L-23699 (synthetic turbine engine oil) technology, with 10 percent triaryl phosphate added.

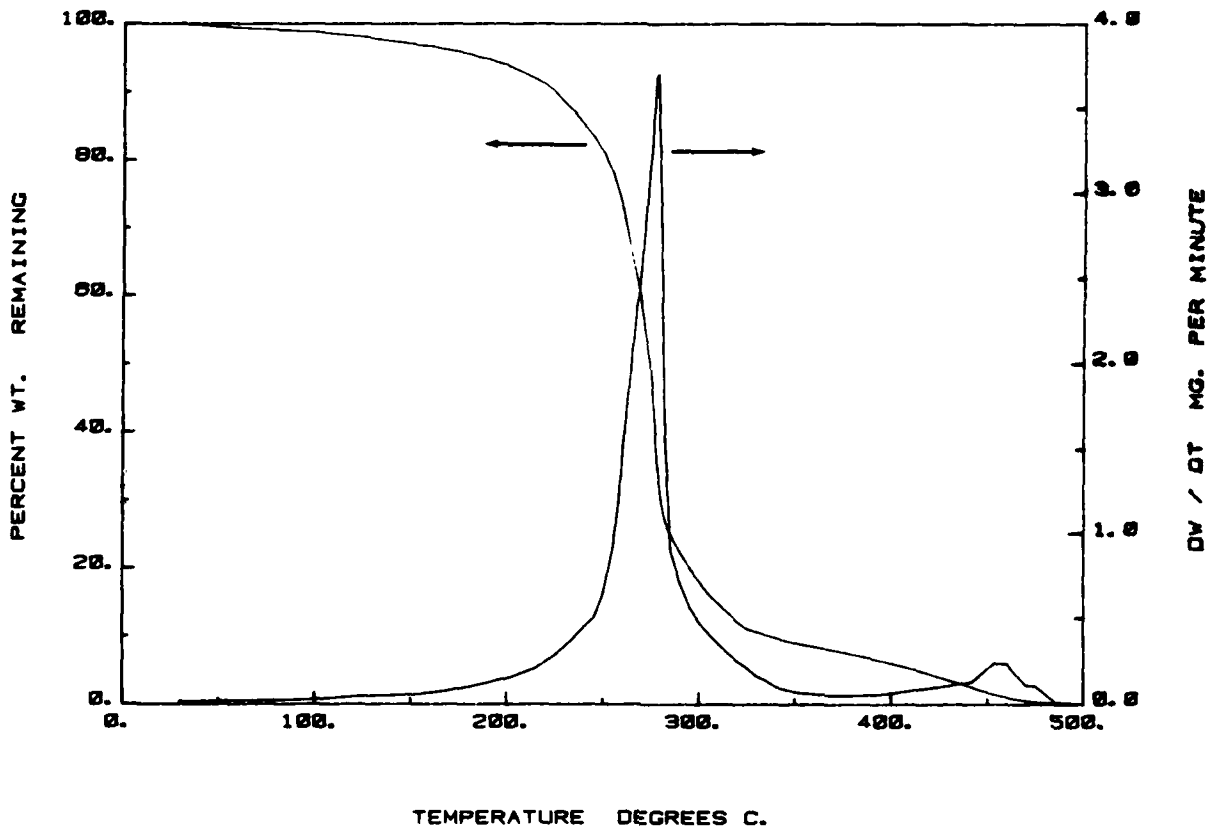

FIGURE 9. THERMOGRAVIMETRIC ANALYSIS OF POLYOL ESTER TYPE SYNTHETIC LUBRICANT

In conclusion, the TRR temperature of 1100+°F expected in the AIPS engine is beyond the capability of current synthetic oil technology. The development of a "system approach" to engine lubrication which includes engine materials, novel high-temperature lubricants, and a novel lubricant delivery scheme will be required to adequately lubricate the AIPS engine.

IV. LUBRICATION SYSTEM SUMMARY

From the previous discussions, it can be seen that a wide variety of approaches offer promise for increasing the temperatures in the piston/cylinder area of diesel engines. The elements of such systems are summarized in the following paragraphs.

1. Lubricants

The major problem to be solved is the selection of adequate lubricants. Although synthetic fluids may extend the current temperature range to 700°F (371°C) with the current crankcase delivery system, operation above that temperature is questionable due to lack of information concerning lubricant stability. Although preformed decomposition products have been shown to give low friction at room temperature, it is not known how effective they will be during formation or how far they can increase fluid operating temperatures. Solid lubricants are available in the 700°F (371°C) to 1000°F (538°C) temperature range, but they will give higher friction than that found for fluid film lubrication. These solid lubricants also give higher wear rates than reported for boundary lubrication. There are effective solid lubricants for temperature above 1000°F (538°C), but they have the additional limitation that they are ineffective at low temperatures. A major emphasis must be placed on identifying improved lubricants and their effective temperature ranges. The major emphasis should be placed on finding effective "residual" lubricant films deposited from synthetic fluids which consist of both decomposition films, surface active additives, and solid lubricants. The goal would be to extend operating temperature as high as possible without jeopardizing fluid film or boundary lubrication at lower temperatures. The residual film should lubricate both as a decomposition film and, when that is no longer effective, leave a solid lubricant film. The most effective solid lubricants based upon current information would be mixtures of graphite and soft metal oxides such as cadmium or copper.

2. Materials

A wide variety of materials are available for operation to temperatures above 1500°F (816°C). However, some form of lubrication will be required. The accumulated information shows that unlubricated friction and wear rates are much too high for ring-liner applications. Ceramics offer the advantage of insulation, high hardness, and maximum utility of ineffective lubricants.

They also have the ability to operate unlubricated for short periods of time. Their main disadvantages are that they are prone to surface damage (microfracture) during sliding, generally have high friction coefficients and high interface temperatures, and are difficult to fabricate and use. If a lubricated system is used, consideration should be given to metallic materials such as tool steels which will withstand temperatures to 1000°F (538°C). For solid or boundary lubrication systems, consideration should be given to metal-bonded ceramic combinations. Metal-bonded ceramic combinations have lower unlubricated friction coefficients and are less prone to microsurface damage than either metals or ceramics. They are also easier to apply and finish. Particular attention should be given to metal-bonded carbides which produce lubricating oxides. Such materials are currently available but also offer promise for further development. For these higher conductivity materials, some form of insulating backing may be necessary.

Self-lubricating composite materials have wear rates which approach the desired levels at lower temperatures but have higher friction coefficients than desired. Each of these problems could be avoided by supplemental fluid or gas lubrication. Unfortunately, materials for temperatures above 700°F (371°C) are limited consisting of carbon graphites and a few metal-bonded composites. Such materials should be evaluated along with metal-bonded carbides and tool steels for the 700° to 1000°F (371° to 538°C) temperature range with both liquid and solid lubrication.

Additional development will be required for temperatures above 1000°F (538°C). Efforts should be directed toward ceramics and metal-bonded carbides which contain solid lubricant inserts or which provide lubricants by oxidation.

The best lubricating composite materials should be used irrespective of the lubricant or the delivery system.

3. Lubricant-Delivery Systems

Changing the lubricant-delivery system in the cylinder requires major redesign, thus every effort should be made to keep the current crankcase delivery system. This can be done by exploiting the residual lubricant concept previously discussed. It is difficult to project how high temperature can be achieved, but that limitation would be either ring scuffing or wear or the generation of excessive amounts of decomposition products which accumulates in the crankcase or are expelled through the exhaust. Above this temperature, it may be necessary to isolate the cylinder area and provide a special lubricant and system. A lubricant might be developed which burns clean or the contaminating elements could be removed by the recirculating system.

One might estimate that the above systems could be used to 1000°F (538°C). Above that temperature, improved materials will be required. Two systems seem to offer the most promise, especially when used in conjunction with self-lubricating composites: the gas bearing or the gas reaction system. It is entirely possible that they can be used together. The gas reaction system provides an auxiliary solid lubricant, while the gas bearing reduces the contact loads and thus reduces friction and wear. The technology for gas bearings is available while that for gas reaction systems would need to be developed.

V. CONCLUSIONS

A study has been made to identify potential lubricant systems for ring-liner applications in advanced heat engines. From this study, the following conclusions were drawn:

1. Current metal and ceramic materials do not possess adequate friction and wear properties to allow unlubricated operation for extended periods of time.

2. Solid lubricant coatings do not have adequate wear lives to be useful except for "run in".

3. Composite materials approach the necessary wear rates ($K = 10^{-8}$) for 400 hour life but do not have the temperature capability. Increased emphasis on the development of metal and ceramic self-lubricating composites is recommended. In this development, priority should be given to composites made by inserting lubricant plugs.

4. Synthetic lubricants are currently available which should allow satisfactory operation to temperatures of 700°F (371°C).

5. Lubricant decomposition films are good boundary lubricants and should allow operation to temperatures above 700°F (371°C) if increased friction coefficients (0.10) can be tolerated. However, adequate film control will be required for effective lubrication.

6. Solid lubricant additives in the decomposition films may lower friction and allow higher temperature operation.

7. Regardless of the materials selected (including self-lubricating composites), some lubricant delivery system will be necessary. Those that appear to have great potential are:

   a. A liquid recirculation system designed only for the ring-liner area which lubricates with the fluid at low temperatures and with decomposition films and solid lubricants at high temperatures.

   b. A reactive gas system in combination with appropriate ring-liner materials. This might be a combination gas bearing solid lubricant system.

8. Gas/particle systems are considered impractical based upon suspension, retention, and transfer properties.

9. Too much lubricant is required to make solid stick transfer processes practical for ring-liner lubrication.

10. Gas bearings, even if only partially effective and in spite of fabrication difficulties, offer promise because of their friction-reducing capabilities.

11. A logical development of lubrication systems for high-temperature piston-liner applications would consist of the following:

    a. The use of a synthetic crankcase lubricant with high temperature wear resistant metal alloys, rings, and liners. The system is designed for minimum wear using larger rings, lubricating coatings, minimum loads, and maximum wear tolerances in the rings. Designs such as foils which compensate for wear should also be explored.

    b. Develop ceramic self-lubricating composites using the sacrificial stick (plug) approach. Substitute these materials along with ceramic liners into the synthetic/crankcase system. The materials should be developed for 1500°F (816°C) capability.

    c. Develop a piston/liner fluid lubrication system which exploits the decomposition/solid lubricant approach. The fluid carrier would be expected to "flash" away without leaving harmful residue. Rings and liners could be the previously developed self-lubricating composite rings and/or liners.

    d. For the maximum temperatures, a pressurized gas-reaction system which maximizes the gas-bearing effect should be designed. Modify the self-lubricating ceramic conept using materials which will be lubricated by reactive pressurized gas.

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