1-1996

On the Effectiveness of Dry Film Lubricant Coatings in Reducing Automotive Valve Train Wear

Evan Benstead

Martin L. Smith

Joseph Foyos

Omar S. Es-Said

On the Effectiveness of Dry Film Lubricant Coatings in Reducing Automotive Valve Train Wear

Evan Benstead
Martin L. Smith
Joseph Foyos
Omar S. Es-Said
Loyola Marymount University

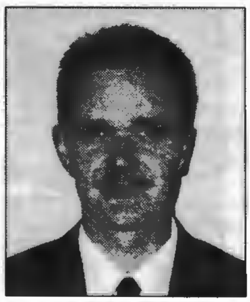
ABSTRACT. The effect of three dry lubricants on automotive valve train wear resistance was studied experimentally. Scuffing wear occurs as the cam slides across the lifter face where the rotating motion of the camshaft is converted into the linear motion necessary to drive the cylinder head valves. This scuffing is caused by localized microscopic bonding between the skidding surfaces. It can be minimized by using dry film lubricant coatings to increase the boundary lubrication depth adjacent to the contact area. To compare valve train wear resistance in the laboratory, rotating cam lobes coated with dry lubricants—parkerization, spray-applied graphite coating, and brush-applied molybdenum disulfide coating—were pressed against valve lifters that were constrained in a fixture. The brush-applied molybdenum disulfide coating was the most effective of the three tested lubricants in reducing scuffing wear.

INTRODUCTION

The operating conditions at the contact line between the rotating camshaft lobe and the reciprocating valve lifter in an automotive engine are extremely severe. High loading forces, high sliding velocity, and high friction between the cam and lifter make this interface one of the most wear-prone areas in an internal combustion engine. Scuffing wear is the common wear mechanism. Fortunately, judicious material selection and processing, excellent cam surface alignment, and proper lubrication can minimize scuffing wear and provide long engine life.

Since proper lubrication is essential for minimum wear, virtually all production four-stroke internal combustion engine designs provide for the splashing of copious amounts of oil in and around the camshaft during continuous engine operation. However, during engine start-up, the situation is different since splashing cannot occur until oil is first pumped upward from the oil pan located at the bottom of the engine. In the seconds before the oil pump reaches full pressure, thereby filling oil galleys and running clearances, camshaft surface lubrication is scant. During this time, dry lubricant coatings can best protect the engine.

This article provides an overview of the scuffing wear mechanism and a laboratory evaluation of the three coatings in a test fixture designed to simulate cam/lifter conditions during engine start-up. The experimental procedure is performed on camshafts and lifters from a 1992 Chevrolet 350 V-8 (5.7-L) engine.



Evan Benstead

BACKGROUND

The term *wear* may be defined as "damage to a solid surface, usually involving progressive loss of material, due to relative motion between that surface and a contacting substance or substances" [1]. Mechanical systems that include components in direct surface-to-surface contact, with one surface in motion relative to the other, experience a type of wear known as *scuffing wear* or *adhesive wear*. This type of wear, by definition, is due to localized bonding between contacting surfaces, leading to material transfer between them or material loss from either surface. Surface adhesion between contacting solids may best be visualized microscopically as they occur at mating surface asperities. On a microscopic level, the normal force from the loaded surface is supported by contacting surface asperities. Scuffing wear occurs when these contact points plastically deform, transfer, or fracture [2].

In the camshaft-to-lifter pair under consideration, the magnitude of the follower axial loading force is of primary importance to wear rate [3,4,5]. This force tends to be large, ranging between 86 and 206 lbf. Hertzian stress exists at the cam-to-follower contact line due to the loading force, and a small area of contact actually exists [6]. This pressure area brings a large number of surface asperities into intimate contact, causing rapid plastic deformation and fracture.

Other factors inherent to the design of any automotive camshaft-to-follower interface have a detrimental effect on the rate of scuffing wear of the system. These factors are related to the dynamic and geometric properties of the interface. The first of these factors is the relative velocity between the cam and follower. This property is defined as

$$\mathbf{V}^{\star} = \mathbf{V}_c + \mathbf{V}_f \qquad (1)$$

where

$\mathbf{V}^{\star}$ = relative velocity between the cam and follower

$\mathbf{V}_f$ = follower velocity component along the follower axis

$\mathbf{V}_c$ = cam velocity component normal to the follower axis

Figure 1 shows how these vectors resolve [7]. It has been observed that, near the top of the follower lift point, a condition exists such that the velocity of the follower is nearly equal to that of the cam lobe. At this point, lubricant entrainment speed approaches zero, and thus lubricant film thickness becomes negligible. This leads to a condition of unlubricated metal-to-metal contact. This point of contact is thought to be one of the sites of accelerated scuffing wear [8].

The second factor observed to accelerate scuffing wear at the camshaft-to-follower interface is related to the follower slipping ratio. This property is a function of the velocities of both the cam lobe and the follower and is a measure of the degree to which the follower surface moves tangent to the cam profile (Figure 1). It is defined as

$$S_f = \frac{|\mathbf{V}_c + \mathbf{V}_f|}{|\mathbf{V}_f|} \qquad (2)$$

where S_f = follower slip ratio.

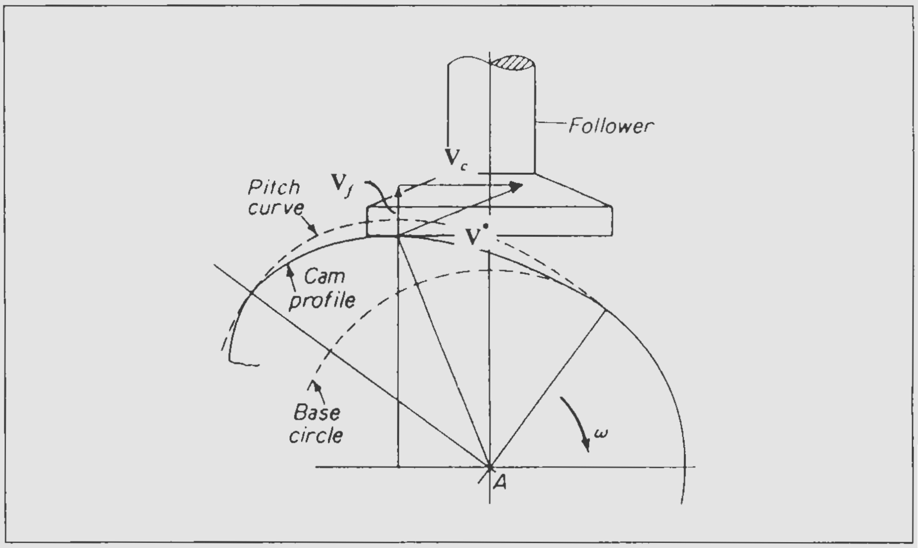
It has been reported that, near the reversal point of follower travel, complete slipping occurs. Again, this contact point is thought to be one of the sites of accelerated scuffing wear [8].

The rate of scuffing wear for the system under consideration may be reduced by providing adequate lubrication. However, the formation of adequate boundary lubrication using standard engine oils is difficult when the camshaft rotates at high speed. A solution to this problem is to create a fixed layer of boundary lubrication at the interface, using dry lubricant coatings applied to the surface of the camshaft. In choosing a dry lubricant coating for use in this application, the predominant consideration is the ability of the coating to continue to adhere to the camshaft surface under conditions of repeated loading. The service life of various coatings may be determined experimentally, and the results can then be compared to ascertain the best dry lubricant coating to be used in the application.

INVESTIGATION METHOD

Testing was performed to determine the resistance of various dry lubricant coatings to scuffing wear at the camshaft-to-follower interface. Mating hydraulic lifters and valve springs from the V-8 engine were employed. Camshafts used in the tests were made of alloyed gray iron (ASTM A-159 Gr. G4000D, SAE J431C Gr. G4000D), with a lobe surface hardness of Rc 50. The lifters were made from hardenable iron, having a minimum hardness of Rc 55. An unlubricated camshaft was tested to serve as a control. Various camshaft surfaces were individually treated in one of three ways: by parkerization (a patented type of manganese phosphate coating); by spray-applied graphite coating; or by brush-applied mo-

FIGURE 1 Velocity Vector Diagram of Cam-Lobe-to-Lifter Pair



lybdenum disulfide coating. Dry lubricant coatings were applied to the camshaft surfaces in accordance with the manufacturers' specifications.

To measure the cam-lobe-to-follower friction coefficient, the camshaft was supported in the lathe on centers coincident with the camshaft rotation axis (Figure 2). Oil was applied to the center contact areas to minimize bearing friction. The lathe headstock was adjusted to press the follower (in its fixture) against the cam lobe, which was positioned at top dead center.

Next, a string was wrapped around the axis of the camshaft and attached to a weight. By varying either the weight or the follower spring compression, the assembly could be adjusted to equalize the weight. Knowing the spring constant, the spring compression, the weight, and the camshaft geometry, the follower contact force and the torque could then be calculated. The test was repeated several times to ensure consistent readings.

The coefficient of friction was calculated by using the following equations:

$$\mu = \frac{F_s}{N} \qquad (3)$$

where
F_s = force of static friction
μ = coefficient of static friction
N = normal force applied by the lifter

$$N = kx \qquad (4)$$

where
N = normal force applied by the lifter
k = spring constant of lifter spring
x = linear compression of lifter spring

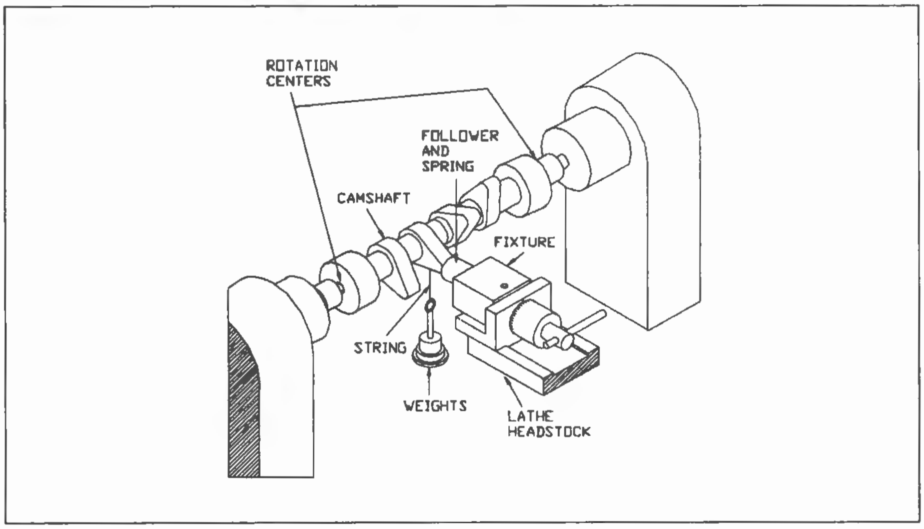
$$T = F_w(r_1) \qquad (5)$$

where
T = torque on the camshaft
F_w = force applied by the weights
r_1 = radius of camshaft at the point where weights were applied

$$F_s = \frac{T}{r_2} \qquad (6)$$

where
F_s = force of static friction
T = torque on the camshaft
r_2 = radius of the cam lobe at top dead center (where normal force was applied)

FIGURE 2 Experimental Set-up Used to Determine the Cam-Lobe-to-Follower Coefficient of Friction



Engine conditions were simulated [9] by rotating each camshaft in a lathe at 500 rpm. In order to prevent deflection in the camshafts, each was cut at its bearing points prior to mounting. Individual cam lobes were placed in contact with a follower and loaded to 270 lbf at 0° cam rotation by use of a valve spring. At the beginning of each test, engine oil was applied to each cam lobe assist break-in at the interface. Each camshaft was then rotated for 12 h under load before removal from the test apparatus for measurement of surface wear. The conditions and duration of the test were chosen to simulate the harsh break-in period that exists in most four-stroke engines. The rotational speed chosen is representative of the camspeed of an engine running at 1000 rpm, or at a speed appreciably above idle. The time duration was arbitrarily chosen. The test apparatus is shown in Figure 3.

RESULTS AND DISCUSSION

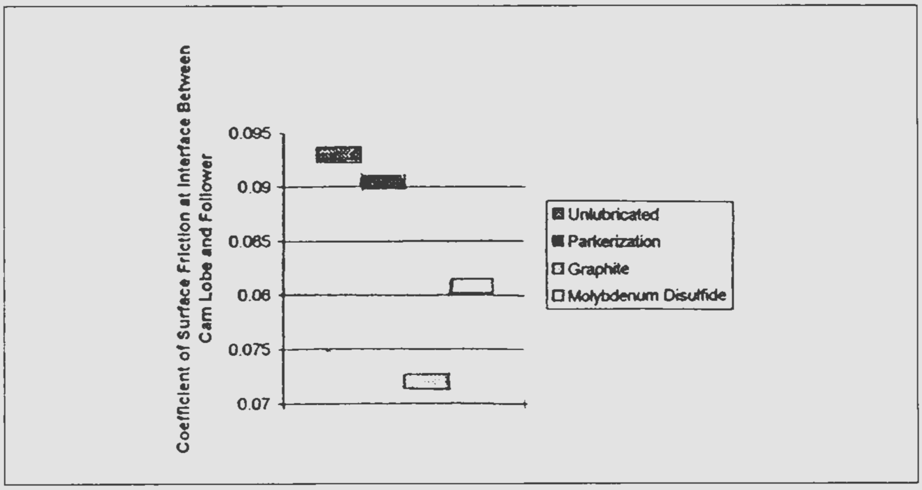
Coefficients of static friction at the camshaft-to-follower interface, correlating to each of the various dry film lubricant coatings, are displayed in Table 1 and Figure 4. Figures 5a–5c

FIGURE 3 Test Apparatus: Test Equipment Mounted in Lathe (Front View)

TABLE 1 Coefficients of Static Friction for Each Camshaft-to-Lifter Interface, by Lubricant Type

Lubricant Type	Coefficient of Static Friction at Interface between Cam Lobe and Follower
Unlubricated	0.0920–0.0937
Parkerization	0.0899–0.0911
Graphite	0.0704–0.0727
Molybdenum Disulfide	0.0802–0.0815

FIGURE 4 Histogram of Coefficients of Static Friction for Each Camshaft-to-Lifter Interface, by Lubricant Type



display some of the geometric and dynamic properties of the camshaft-to-lifter interface.

Regardless of the type of lubricant coating tested, the follower surface experienced the most scuffing wear. All followers exhibited visible wear at the lifter face, and the layout of the wear patterns generally was identical, differing only in depth (see Figure 6 on page 36). Wear at the camshaft surface appeared minimal, with only a mild polishing effect (and in some cases, a mild abrasive effect) in evidence. Most of the following discussion focuses on the wear pattern at the follower face; individual results of the tests of each lubricant coating are then presented.

In the case of all followers, the site of maximum wear depth was found to correspond to the interface point where the magnitude of **V*** (the relative velocity between the cam and follower) was at a minimum. As shown in Figure 5b, this condition existed at a cam angle of approximately 100°, and again at a cam angle of approximately 105°. On the face of each follower, this site was offset from the center. It is believed that the thickness of the layer of engine oil lubricant here reached zero, and dry sliding contact occurred. The experiment confirmed that the conditions at this point are the most detrimental to the acceleration of scuffing wear in the system. However, it was found that the parkerization and molybdenum disulfide coatings proved successful in significantly reducing the wear rate at this site.

Significant wear of all followers also occurred at the interface point corresponding to maximum Hertzian contact stress and maximum follower slip ratio. As displayed in Figure 5a, the site of maximum Hertzian contact stress occurred at a cam angle of 90°, or at the point of maximum follower lift. As shown in Figure 5c, the follower slip ratio simultaneously became infinite at this point (i.e., complete slipping occurred). This point corresponds to the center of each follower face. The experiment confirmed the detrimental effects of these combined conditions on the acceleration of scuffing wear. Again, the parkerization and molybdenum disulfide lubricants were successful in inhibiting accelerated scuffing wear at this site.

Following is a description of the results of the experiment for each type of lubricant coating tested.

Unlubricated Interface

The magnitude of the coefficient of static friction was the highest for this interface (Table 1). The profile of each follower face was visibly affected after only 4 h of testing, and each cam lobe surface began to display a polishing effect. After 6 h of testing, wear depth at maximum wear sites was visible on each follower face. From this time forward, scuffing wear at the follower surfaces accelerated rapidly. The final wear conditions at each follower face included deep scuffing wear, with minimal abrasive wear evident. Observed abrasive wear was believed to be caused by the presence of microscopic particles on the follower surface, freed by microscopic fracturing of the follower face and present at the camshaft-to-follower interface. It was noted that cam lobe surfaces continued to display polishing-type wear after 12 h of testing, with minimal abrasive wear also in evidence.

Parkerized Interface

As recorded in Table 1, the coefficient of static friction at the parkerized interface* was relatively high. Follower faces, tested in the presence of this dry lubricant coating, experienced minimal wear. The profile of each follower

*This coating is manufactured by Consolidated Manufacturing, 1600 N. Halstead, Hutchinson, Kansas 67504-1800; also by Turco Products, Box 6200, Carson, California 90749.

FIGURE 5a Cam-to-Follower Pressure Diagram

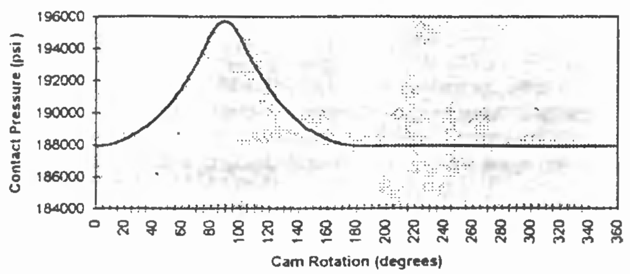


FIGURE 5b Relative Speed versus Cam Rotation

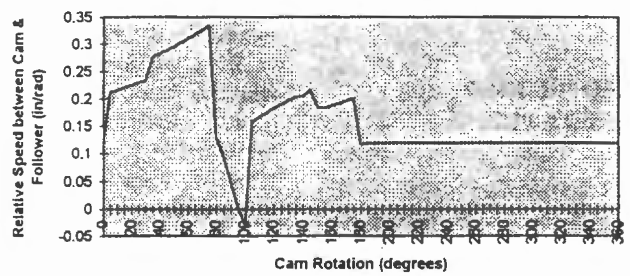
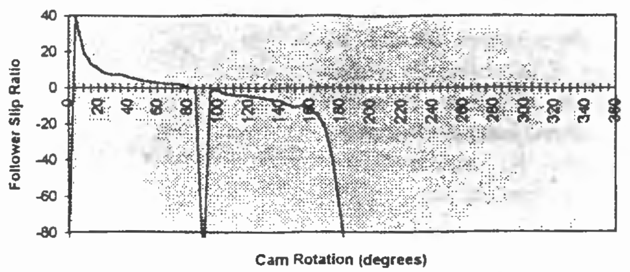


FIGURE 5c Follower Slip Ratio versus Cam Rotation



face became visibly affected only after 8 h of testing, and no visible effect was noted at the cam lobe surfaces for the duration of the test. Final depth of wear at each follower face became visible after 12 h of testing. However, it was noted that the final wear depth at the sites corresponding to maximum wear was several orders of magnitude less than that observed for the followers tested under unlubricated conditions.

The presence of mild abrasive wear across each follower face was also evident after 12 h of testing. As discoloration of the follower faces also occurred, it was believed that this abrasion was caused by the dislodged microscopic particles of the manganese phosphate from the cam lobe surfaces.

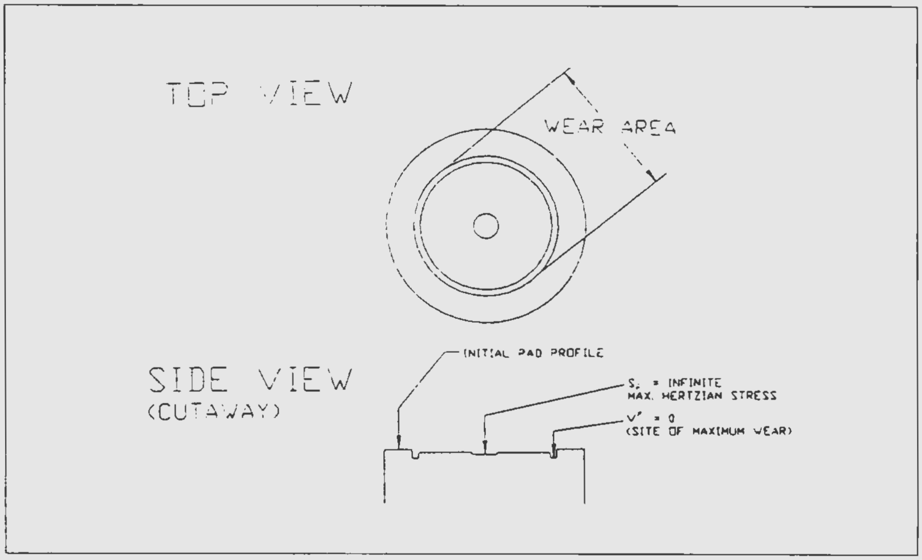
Graphite-Coated Interface

As recorded in Table 1, the coefficient of static friction at the graphite-coated interface† was the lowest recorded for any of the dry lubricant coatings tested. As with the unlubricated interface, the profile of each follower face was visibly affected after only 4 h of testing. Scuffing wear at both the follower face surfaces and at the cam lobe surfaces accelerated in much the same manner as observed for the unlubricated interface. Poor adhesion of the graphite coating to the camshaft surface was responsible for these results. Despite the fact that this coating was applied in the manner recommended by the manufacturer, it was observed to adhere poorly in the presence of engine oil. Approximately 50% of the coating mixed with the engine oil and wore away from the interface within only 30 min of testing; after 2 h of testing, it was observed that approximately 80% of the coating had completely disappeared.

The final wear conditions of both the cam lobe and follower face surfaces exhibited a combination of scuffing and abrasive wear.

†This coating is manufactured by Miracle Power Products, 101 Beltline Street, Cleveland, Ohio 44109.

FIGURE 6 General Layout of Wear Pattern on Follower Faces



Molybdenum-Disulfide-Coated Interface

As recorded in Table 1, the coefficient of static friction at the molybdenum-disulfide-coated interface‡ was considerably lower than that recorded at the unlubricated interface. Experimental results showed that this type of dry lubricant coating provided the most significant resistance to scuffing wear of the tested coatings. The follower face profile was visibly affected only after 12 h of testing, and no visible effect was noted at the cam lobe surfaces. This was believed to result from the high thermal resistance of the coating. Even though approximately 40% of the coating had mixed with the engine oil and worn away from the interface after 4 h of testing, it was observed that the remaining coating did not further disappear. It remained entrained at the interface for the duration of the test.

Final wear depth at each of the follower faces was considerably less than that observed for any of the others tested.

‡This coating is manufactured by National Process Industries, 42250 Baldaray Circle, Temecula, California 92590.

CONCLUSIONS AND RECOMMENDATIONS

In automotive valve trains, where break-in of the camshaft-to-lifter interface is critical to the design life of the engine, it is recommended that camshaft surfaces be coated with dry molybdenum disulfide or another coating that could have equal or better wear-inhibiting performance. Dry molybdenum disulfide provides superior wear resistance at the camshaft-to-follower interface and can be expected to considerably extend the life of valve train components. It is recommended that spray-type application of the coating be employed to ensure uniform coating thickness.

Regardless of the type of coating selected, camshaft surfaces should be coated with dry film lubricants whenever economically feasible. Prior to use, the application process should be tested and refined to ensure an optimum protective film. Though primarily not critical to normal engine operation, dry film lubricants can extend engine life under adverse operating conditions, such as infrequent use or frequent engine starts.

ACKNOWLEDGMENTS

The authors wish to thank Dr. Joseph P. Callinan, chairman of the Mechanical Engineering Department, and Salwa B. Es-Said for reviewing the manuscript. Special thanks to Jonathan Sakomoto for typing the manuscript, and to Thanh Nguyen for his assistance in the initial literature search for this project.

REFERENCES

1. K. G. Budinski, *Surface Engineering for Wear Resistance*, Prentice-Hall, 1988.
2. A. D. Sarkar, *Wear of Metals*, Pergamon Press, 1976.
3. G. H. Martin, *Kinematics and Dynamics of Machines*, Macmillan, 1989.
4. J. M. Prentis, *Dynamics of Mechanical Systems*, 2d edition, Halstead Press, 1980.
5. M. Kano and I. Tanimoto, "Wear Mechanism of High Wear-Resistant Materials for Automotive Valve Trains," *Wear of Materials*, Vol. 1, p. 83, 1991.
6. J. E. Shigley and C. R. Mischke, *Mechanical Engineering Design*, 5th edition, McGraw-Hill, 1989.
7. M. F. Spotts, *Design of Machine Elements*, 4th edition, Prentice-Hall, 1971.
8. J. C. Bell and T. Colgan, "A Predictive Model for Wear in Automotive Valve Train Systems," SAE Paper 892145.
9. D. L. Anglin and W. H. Crouse, *Automotive Engines*, 7th edition, McGraw-Hill, 1986.

Evan Benstead *is a graduate of the Mechanical Engineering Department of Loyola Marymount University.*

Joseph Foyos*, laboratory manager,* ***Martin L. Smith****, graduate student, and* ***Omar S. Es-Said****, associate professor, are with the Mechanical Engineering Department of Loyola Marymount University.*