# The Relationship Between High-Temperature Oil Rheology and Engine Operation

*A Status Report*

ASTM DS 62

# THE RELATIONSHIP BETWEEN HIGH-TEMPERATURE OIL RHEOLOGY AND ENGINE OPERATION— A STATUS REPORT

Prepared by

ASTM Task Force D02.07.0B TF/EC on the Correlation of High-Temperature Oil Rheology with Engine Performance. ASTM Sub-committee D02.07.0B on High-Temperature Rheology of Non-Newtonian Fluids, Committee D-2 on Petroleum Products and Lubricants.

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# FOREWORD

PURPOSE. The ASTM High-Temperature Rheology/Engine Correlation Task Force (ASTM D02.07.0B TF/EC) has written this status report in partial response to a Society of Automotive Engineers (SAE) request to "develop a test method which incorporates high-temperature, high-shear rate viscometrics or other rheological characteristics to predict the performance of both single and multigrade (i.e., both Newtonian and VI-improved) engine oils in engine bearings and/or the ring and cylinder area (1). "Although this report does not deal with the development of particular viscosity measurement techniques, it does have two specific objectives which relate to this SAE request: (1) to summarize, and interpret, as necessary, all pertinent published studies relating high-temperature oil rheology to selected measures of engine performance and durability, and (2) to suggest areas for possible future research needed to resolve any uncertainties which remain regarding the effects of oil rheological properties on these same engine operating factors.

SCOPE. This review summarizes those available data which describe the effects of oil rheological properties on (1) the operating parameters (oil film thickness and load capacity of journal bearings), (2) the wear of engine components, and (3) the frictional characteristics and fuel economy of engines. These three subjects were selected for review because each is related to the specific areas of interest indicated in the SAE request. Furthermore, only high-temperature (that characteristic of warmed-up engine operation) engine performance is considered. Those engine performance parameters (cranking, pumping) which are associated with low-temperature operation are not within the scope of this review.

# Related ASTM Publications

The Relationship Between Engine Oil Viscosity and Engine Performance (Part V and VI), STP 621-S4 (1980), 04-621040-12

The Relationship Between Engine Oil Viscosity and Engine Performance (Part IV), STP 621-S3 (1978), 04-621030-12

The Relationship Between Engine Oil Viscosity and Engine Performance (Part III), STP 621-S2 (1977), 04-621020-12

The Relationship Between Engine Oil Viscosity and Engine Performance (Part II), STP 621-S1 (1977), 04-621010-12

The Relationship Between Engine Oil Viscosity and Engine Performance (Part I), STP 621 (1977), 04-621000-12

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# OVERVIEW

High-temperature oil rheology affects each one of the measures of engine performance reviewed in this report, but the magnitude of such effects is greatly influenced by engine design, engine operating parameters, and oil chemistry. In addition, engine oils are subjected to different conditions of temperature and shear rate at different locations in the engine. As a result, it is not surprising that this report contains references showing various aspects of engine performance correlating with oil viscosities measured at a variety of different temperatures and shear rates, ranging from 100 to 150°C and from $10^5$ to $10^6 s^{-1}$. The references also show that kinematic viscosity at 100°C, although serving as an adequate measure of oil rheology for single-grade, Newtonian oils, generally does not correlate with these engine performance measurements when polymer-containing, multigrade oils are considered.

In this report, when the expression, "high-temperature, high-shear rate viscosity," is used without any other qualification, it means a viscosity measured at generally greater values of temperature and shear rate than the current kinematic method specified in SAE J300. It does not always imply a viscosity measured at a specific set of temperature and shear rate values (e.g. 150°C and $10^6 s^{-1}$). Also, although some references in the report cite correlations between engine performance and used-oil viscosities, it should be emphasized that SAE J300 has always been, and is anticipated to remain, based on new-oil viscosities.

Journal Bearing Performance. In laboratory rig, bearing studies, conducted under steady-load conditions, the available data are in good agreement that load capacity, defined as the load at which the bearing contacts the journal, and film thickness correlate with some measure of high-temperature, high-shear rate viscosity. In engine bearing systems, limited data are available which also show a correlation between load capacity and high-temperature, high-shear rate viscosity, but at least one reference points out that the detergent/inhibitor (DI) package as well as the friction modifier used in an oil can affect load capacity measurements. A correlation between film thickness in engine bearing systems and high-temperature, high-shear rate viscosity can be shown for many, but not all, oils evaluated at this time.

Engine Wear. In some studies, bearing wear correlates with both low- and high-shear rate viscosity for oils with the same DI additive package, base stock components, and various concentrations of a specific viscosity index (VI) improver. Furthermore, bearing wear may correlate with high-shear rate viscosity when the DI package and base stock components are held constant and only the type of VI improver is varied. The degree of correlation appears to depend on the DI package in the oil and the service to which the oil is subjected. When oils with different DI packages and different VI improvers are considered, it is impossible to predict the amount of bearing wear in a test solely from the viscometric properties of the oil. For engine components which operate primarily in the boundary lubrication regime (e.g. valve train), additive composition affects durability and wear to a much greater degree than oil viscosity.

Engine Friction and Fuel Economy. For comparisons made in the same engine under the same operating conditions, friction measurements correlate with high-temperature, high-shear rate viscosities. However, even under well-controlled conditions, the friction-modifying properties of the DI package in each oil can influence the degree of correlation. Under realistic, cyclic engine operating conditions, available fuel economy measurements correlate better with high-temperature, high-shear rate viscosity than with kinematic viscosity.

The data in this report demonstrate that it will be difficult to select single values of temperature and shear rate at which to measure viscosity which will correlate with all aspects of engine performance within the scope of this report.

## RECOMMENDATIONS

It is recommended that the Society of Automotive Engineers (SAE), through the activities of its Engine Oil Viscosity Classification Task Force, review the findings of this report to:

1. determine if there are sufficient data to revise the high-temperature portion of SAE J300 on the basis of a high-temperature, high-shear rate viscosity, and
2. decide whether it wishes the ASTM D02.07.0B Engine Correlation Task Force to continue in its efforts to relate oil rheological properties to various measures of engine operation and durability.

If additional work is justified and requested by the SAE, it is recommended that ASTM conduct a cooperative research effort using a set of industry reference oils and similar analytical techniques. This should provide a more precise correlation between engine operation and oil rheological properties. As an example of the type of effort which could be conducted within the task force, a program is outlined in the section entitled "Possible Future ASTM D02.07.0B TC/EC Efforts" at the end of this report. It should be emphasized, however, that the size and the goals of any future program should be determined after it is decided by the SAE that further work is required.

## INTRODUCTION

In December, 1977, ASTM established within what was then referred to as RDD VIIB a Task Force on High-Temperature Oil Rheology (TF-HTR) in response to a request by the SAE Fuels and Lubricants Technical Committee. The request asked that ASTM "develop a test method which incorporates high-temperature, high-shear rate viscometrics or other rheological characteristics to predict the performance of both single and multigrade (i.e., both Newtonian and VI-Improved) engine oils in engine bearings and/or the ring and cylinder area (1)."[1] To define the problem and its scope, the TF-HTR developed a special questionnaire which was sent world-wide to engine manufacturers, lubricant marketers, and additive suppliers for lubricants. Specifically, the questionnaire asked for responses on:

1. the suitability of kinematic viscosity as a predictor of high-temperature engine lubrication,
2. evidence of field failures related to high-temperature oil rheology,
3. ranking of engine performance areas with respect to high-temperature rheological properties, and
4. estimation of engine temperatures, pressures, and shear rates for specific engine components.

No reports of widespread field failures related to oil rheology were reported by the questionnaire responses from U.S. manufacturers, but several instances of bearing failures in specific engines were reported in Europe. Although the problems cited were eventually eliminated, in part through mechanical design changes, the incidence of such failures contributed to the demand by European manufacturers for limits on the high-temperature, high-shear rate viscosity of commercial engine oils (2).

The questionnaire responses also indicated that few technologists believe that kinematic viscosity adequately predicts oil performance in engines. Bearing wear was deemed the measure of performance most affected by oil rheology, with bearing operating conditions estimated to range from 100-160°C in temperature and from $10^4$-$10^7$ $s^{-1}$ in shear rate.

The TF-HTR next considered its database for developing a correlation between engine performance and high-temperature oil rheology and concluded that the database in 1978 was too narrow and fragmented to support a major change in the SAE Viscosity Classification. This would require an extensive engine or field test program to systematically develop a broad data base on a number of reference oils. Unfortunately, such a program was beyond the resources of ASTM. This data base limitation was brought to the attention of SAE in October 1980 (3). Since that time, ASTM has pursued an alternate approach of utilizing all published work and ongoing industry research in this area. Companies that

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[1]Numbers in parentheses indicate references tabulated at the end of this report.

published engine data have supplied ASTM with ten reference oils, designated HTR's 30-39.

At the same time in Europe, a Coordinating European Council (CEC) committee, CL-23 (formerly IGL-9), was actively engaged in a research program whose objectives were similar to those of TF-HTR. From its inception, the TF-HTR decided to cooperate closely and fully with CL-23 so that both groups would achieve their goals more efficiently and effectively than working independently. To foster communication, minutes of both committees were exchanged. In 1979 and 1980, samples of European reference lubricants were made available to interested TF-HTR members. In turn, samples of HTR reference oils were sent to CL-23 for various testing programs.

It became clear from early reviews of the technical literature that, of the possible measures of oil rheology that might correlate with engine performance, high-temperature, high-shear rate viscosity was a likely candidate to be included in a revised viscosity classification. Since there were no standardized test methods for measuring this viscometric parameter, the TF-HTR narrowed its objective to the following: "To develop a test method which measures high-temperature, high-shear viscosity." To implement this objective, three new panels were appointed:

- Rotational Viscometer Panel
- Capillary Viscometer Panel
- Panel on the Correlation of High-Temperature Oil Rheology with Engine Performance

The two viscometer panels were to address the new objective directly, while the third panel was to consider the needs of the SAE Viscosity Classification. In subsequent years, ASTM Subcommittee RDD VII was reorganized. In the reorganization, TF-HTR became Section B of Subcommittee 7, on High-Temperature Oil Rheology, and each panel became a separate task force.

Currently each task force is still operating. A new, low-cost rotational viscometer called the Tapered Bearing Simulator (TBS) was developed in 1981 and made available to members of the Rotational Viscometer Task Force. Plans were developed for a cooperative Round Robin for both ASTM and CEC purchasers of the TBS. Later, when a second low-cost rotational viscometer, the Ravenfield viscometer, became commercially available, it too was added to the round robin. Other rotational viscometers had been used to measure high-shear rate viscosity at high temperatures. However, most of these were many times more expensive than either the TBS or Ravenfield viscometers.

High-shear capillary viscometers are also under investigation. While two viscometers of this type are commercially available, most have been individually designed and built by specific companies. The Capillary Viscometer Task Force is attempting to write a method in ASTM format that will cover the many one-of-a-kind capillary viscometers currently in use. Results to date show excellent agreement at high shear rates between rotational and capillary viscometers.

The Task Force on the Correlation of High-Temperature Oil Rheology with Engine Performance has been actively gathering data from both the technical literature and individual task force members. As a first step in determining the direction in which future research in this area should go, the task force decided that a comprehensive status report on what is currently known regarding the effects of high-temperature oil rheology on engine performance should be written. Specifically a report of this nature would: (1) provide a summary of the publicly available data which describe the effects of high-temperature oil rheology on different facets of engine performance, and (2) suggest directions for future research which might be needed to resolve the questions holding up revision of the SAE Viscosity Classification. This report is the result of that task force effort.

# Oil Rheology

## OIL RHEOLOGY

The traditional definition of rheology is that science concerned with the deformation and flow of matter. Using this definition, the rheology of lubricants in general, and of engine oils in particular, represents an exceedingly complex subject which is affected by numerous independent variables and parameters. For example, the viscosity of an engine oil (that property of an oil which relates shear stress to shear rate) can be affected at different times and under different engine operating conditions by temperature, pressure, shear rate, and oil composition. Even time and material phase changes, such as wax crystallization, can influence the viscosity of an oil, although these are more commonly associated with low-temperature phenomena and, thus, are not within the scope of this report. To complicate matters further, because commercial engine oils are composed of different base stocks and chemical additives, it is even difficult to generalize regarding the effects of chemical composition on oil rheological properties. Whereas, under certain conditions, the viscosity of an oil might be affected one way by a particular additive, it could be affected in an entirely different way by another.

In addition to those factors listed in the previous paragraph, oil and additive degradation can also significantly affect oil rheology. With use in an engine, either mechanical degradation due to shearing of the high-molecular-weight polymeric additives, physical degradation due to fuel or water dilution, or chemical degradation due to oil or additive oxidation can alter the rheological properties of an oil. Although used oil rheological properties are employed by the authors of some references described later in this report to develop correlations with various measures of engine performance, the Society of Automotive Engineers (SAE) Viscosity Classification, J300, has traditionally used only new oil properties to describe the rheological characteristics of oils.

## Oil Rheological Models and Definitions

To aid in the review of the effects of oil rheological properties on engine operation and performance presented in later sections of this report, it is worth defining the rheological terms and concepts to be used. Engine oils can be grossly divided into two categories: those referred to as purely viscous, and those referred to as viscoelastic. Purely viscous oils, in turn, can be categorized as being either Newtonian, or non-Newtonian. Viscoelastic oils are non-Newtonian.

Newtonian oils are defined as those oils for which the shear stress, $\tau$ is directly proportional to the shear rate, S,

$$\tau = \eta S \tag{1}$$

and for which the viscosity, $\eta$, is only a function of temperature, pressure, and oil composition. Many single-grade oils fall into this category at temperatures above their cloud point, and it was with such Newtonian oils that the original SAE Viscosity Classification was developed over 70 years ago (4).

The effect of temperature on the viscosity of Newtonian oils has been described over small ranges of temperature, T, by the Arrhenius relationship,

$$\eta = Ae^{-B/T} \tag{2}$$

and over wider temperature ranges by the Walther Equation,

$$\log \log (\eta + 0.7) = A_1 - B_1 \log T \tag{3}$$

where A, B, $A_1$, and $B_1$ are constants determined from experimental measurements. In both Equations 2 and 3, temperature is expressed in Kelvins. In addition, Equation 3 was developed for viscosities expressed in cSt. This equation has been used for many years in the construction of the ASTM Viscosity - Temperature Charts, ASTM D 341.

The effect of pressure on Newtonian viscosities has been described by the Barus Equation,

$$\eta^P = \eta^o e^{\alpha(P/P_o)} \tag{4}$$

where $\eta^o$ is the oil viscosity at atmospheric pressure, $P_o$, $\alpha$ is the pressure-viscosity coefficient, and P is the pressure at which the viscosity $\eta^P$ is measured. This relationship is only applicable over limited ranges in pressure. For wider ranges more complicated models have been proposed (5,6,7,8).

The specific additives and base stocks used in blending an engine oil will determine how Newtonian oil viscosities are affected by composition. In general, the effect which most base stocks and low-molecular-weight-additives have on the viscosity of a fully-formulated oil can be calculated according to simple blending rules (9). However, certain "high-molecular-weight" dispersants and pour point depressants (which are actually "low-molecular-weight" polymers) can not only increase an oil's viscosity, but also impart non-Newtonian behavior to the oil.

Non-Newtonian oils are defined as those oils for which shear stress is not directly proportional to shear rate. Although a viscosity, $\eta$, for a non-Newtonian oil can also be defined by Equation 1, the viscosity is, in addition to being a function of temperature, pressure, and oil composition, also a function of shear rate.

$$\eta = \eta(S) \qquad (5)$$

Several different forms of non-Newtonian behavior exist. Pseudoplasticity (viscosity decreases with increasing shear rate), dilatancy (viscosity increases with increasing shear rate), and thixotropy (viscosity decreases with increasing time of shearing) as well as others have been measured for different fluid systems. For oils at operating engine temperatures, however, the only non-Newtonian behavior of importance is that of pseudoplasticity (also known as shear thinning). Virtually all high-molecular-weight viscosity index (VI) improvers used in multigrade oils, as well as some high-molecular-weight dispersants and pour point depressants used in single-grade oils, will produce the type of shear thinning behavior shown in Figure 1. At low shear rates the oil exhibits a constant (low-shear "Newtonian") viscosity. As shear rate increases, this viscosity begins to decrease. The difference between the low-shear "Newtonian" viscosity and the viscosity at higher shear rates is referred to as the temporary viscosity loss, which obviously also varies with shear rate. In much of the literature this temporary viscosity loss is expressed as a percentage of the "low-shear" Newtonian viscosity value. At very high shear rates, theory suggests that a second, constant (high-shear "Newtonian") viscosity is reached, but experimental data in this region are sparse. In contrast, the low-shear "Newtonian" and the non-Newtonian regions of the viscosity-shear rate curve have been well documented for many oils by several authors (10,11,12).

Many different analytical models have been used to describe the shear thinning behavior shown in Figure 1. For limited ranges of shear rate within the non-Newtonian region, the Power Law Model has been useful,

$$\eta = m|S|^{n-1} \qquad (6)$$

In Equation 6, m and n are experimentally determined constants.

Temperature influences both the thickening ability and the degree of non-Newtonian behavior provided by all VI improvers. The resulting viscosity of VI improver-containing oils at any temperature depends on not only the base oil, but also the DI package and VI improver type used. Recently a Coordinating European Council (CEC) task force, CL-23, completed an evaluation of several non-Newtonian viscometric models (13). The model which did the best job of describing the entire viscosity-shear rate function of a series of reference oils, while at the same time taking into account the effect of temperature (measured in Celçius), and being relatively simple to use, was referred to as the Extended Cross Equation,

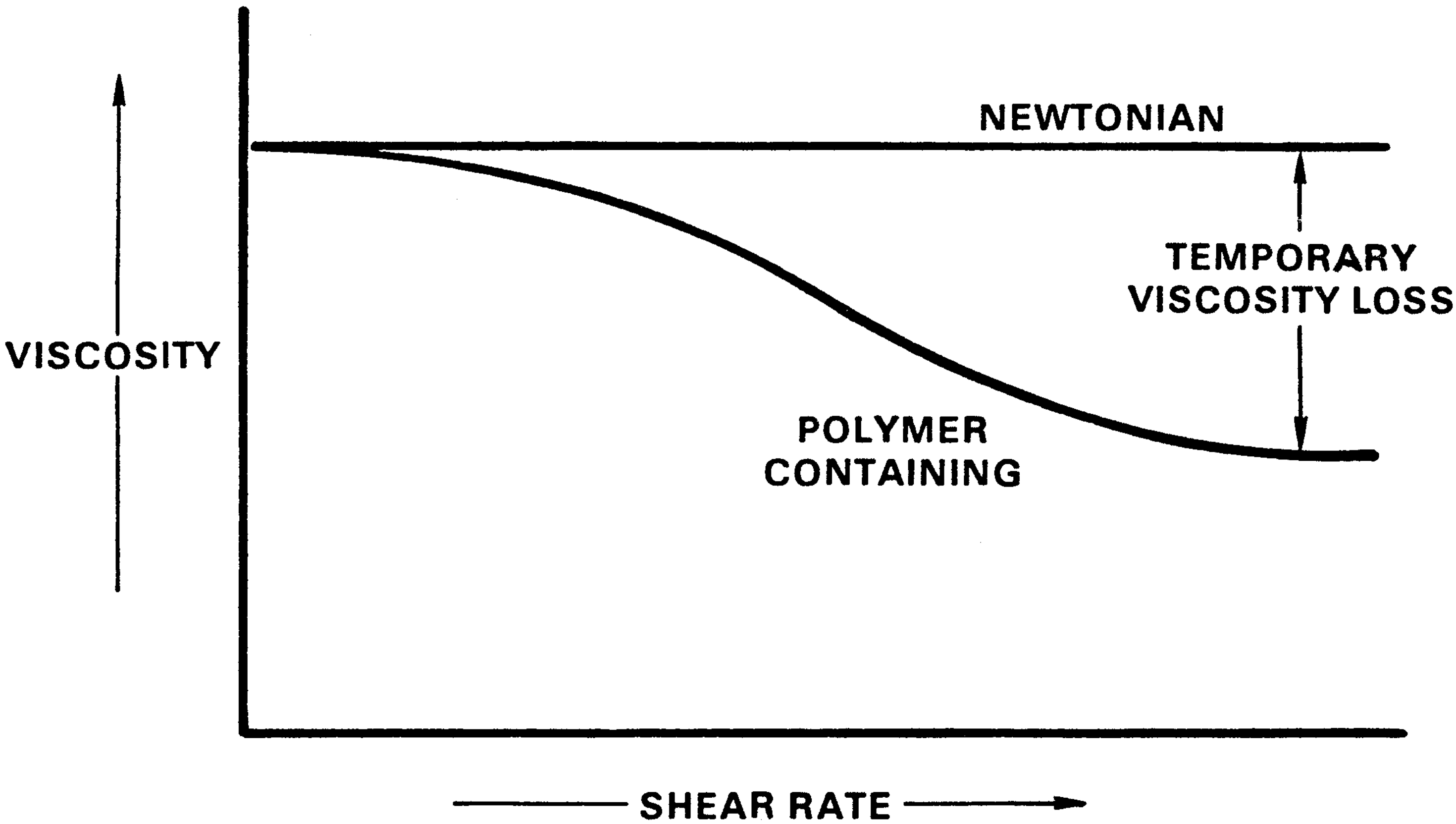


Figure 1. The effect of shear rate on polymer-containing and Newtonian oils.

$$\log \frac{\eta_o - \eta}{\eta - \eta_\infty} = M + R \log S + U/(T + 273) \qquad (7)$$

where $\eta_o$ and $\eta_\infty$ are the low- and high-shear rate limiting viscosities, and M, R, and U are adjustable, experimentally determined constants.

In addition to temporary viscosity losses which are completely reversible with respect to shear rate, polymer-containing oils are also subject to permanent viscosity losses in service due to shear degradation of the polymer molecule. The amount of permanent viscosity loss experienced by an oil is related to the molecular structure of the VI improver, as well as to engine clearances, and the severity and duration of service. While permanent shear loss is an important consideration and may be useful for some of the correlations described in this report, the SAE J300 Viscosity Classification has always been, and is expected to remain, based on new oil viscosities.

The effect of pressure on the behavior of polymer-containing, non-Newtonian oils has also been investigated. Although much of this work was done at shear rates low enough that the viscosities of the oils were in the low-shear "Newtonian" region, Novak and Winer have studied oil viscosities at elevated pressures and a range of shear stresses (14). It was concluded that the Walther Equation, Equation 3, is also valid at elevated pressures at any constant shear stress. In addition, as shear stress is increased at a constant temperature and pressure, the behavior illustrated in Figure 1 is observed. It was clear from this work, however, that different chemical species of polymers produce different temporary viscosity losses in an oil, both at elevated and atmospheric pressures. McMillan and Murphy (12) came to the same conclusion after measuring the temporary viscosity losses for a series of oils containing different polymer species at high shear rates and atmospheric pressures. It is not possible to generalize on the degree of non-Newtonian behavior provided by one polymeric additive relative to another.

For some polymers, the oil will exhibit, in addition to non-Newtonian viscous behavior, an elastic response to applied deformations. Such oils are referred to as viscoelastic and their most striking characteristic is the appearance of "extra" or "normal" stresses. These stresses act in a direction normal to the plane of shear in an oil and are responsible for such dramatic experimental artifacts as the Weissenberg effect (oil is forced to climb up a rotating stirring rod) and die swell (oil jet expands in diameter after exiting from a small orifice). Of a more practical nature, it has been theorized, as described in later sections of this report, that normal stresses in a viscoelastic oil can increase load capacities and film thicknesses in operating journal bearings.

The analytical models that have been developed to describe the rheological behavior of viscoelastic fluids are necessarily much more complex than those for purely viscous oils. Although some analyses have been formulated from molecular theory, many more viscoelastic models have been developed phenomenologically to explain observed trends in experimental data. In general, such models fall into two categories: derivative and integral. The derivative models (15,16,17,18) describe the shear behavior of a fluid in terms of the shear stress, shear rate, and selected time derivatives of either one or both of these variables. The integral models (18,19,21,23,24) describe the shear stress of the fluid in terms of a weighted integral of the shear rate over the entire history of the fluid sample. The weighting function is usually an expression which minimizes the effect of the fluid's most distant history relative to that of its most recent past. Both analytical models can contain numerous fluid properties, particularly elastic moduli and relaxation times, as parameters, and substantial amounts of data are required to evaluate the utility of any of these models in a given flow situation. The details of the development of such models are beyond the scope of this review. It should be emphasized, however, that no single model is currently accepted as being the best for describing the rheological behavior of viscoelastic fluids under all flow conditions.

## Oil Rheological Measurement Techniques

The only high-temperature measurement currently part of the SAE Viscosity Classification is kinematic viscosity at 100°C (ASTM Test Method for Kinematic Viscosity of Transparent and Opaque Liquids (and the Calculation of Dynamic Viscosity) [D 445]). The kinematic viscosity of an oil is determined in a glass capillary tube. Oil flow is produced by the pressure head resulting from several inches of oil. The shear rates to which the oil is subjected as it flows through the capillary are of the order of 100 $s^{-1}$. Shear rates of this magnitude generally lie within the low-shear "Newtonian" region for polymer-containing oils, and, thus, a single kinematic viscosity measurement does not differentiate between Newtonian single-grade and non-Newtonian multigrade oils. To determine non-Newtonian behavior, a viscometer which allows the control of either shear stress or shear rate is required. Many such viscometers have been described and used. These range from instruments which use a couette geometry (rotating concentric cylinders) (25,26,27,28), to those that use capillary tube designs (10,11,12,32,33), to those that use a cone rotating above a flat plate (29,30,31).

Recent viscometer developments have led to the measurement of oil viscosity at very high shear rates ($\sim 10^6$ $s^{-1}$) in both couette and capillary tube instruments (34,35,36,37,38,39,40). This represents an important advance, since this level of shear approaches that which occurs in engine journal bearings and between piston rings and cylinder walls. As described in the Introduction, ASTM D02.07.0B currently has two task forces working on the standardization of high-shear rate, high-temperature methods for determining oil viscosity: one using couette devices, and the other using capillary techniques.

Measuring the viscoelastic properties of an oil is more difficult than measuring oil viscosity. The primary artifact of viscoelasticity, normal stresses, can be determined in both cone and plate and parallel plate viscometers if the plates are instrumented to measure forces or deflections normal to the direction of their rotation (41). Measurements of this sort are limited to relatively low shear rates, however, since the rotational speeds needed to produce high shear rates also generate flow instabilities due to centrifugal forces. Such instability problems at high shear rates can be avoided by using a technique developed by Metzner et al. (42), where normal stress values are calculated from measurements of the amount of "die swell" exhibited by a fluid jet flowing from a capillary. Schowalter (43) has written a recent review of this technique and pointed out several possible limitations.

A new technique for measuring normal stresses has recently been developed (44). It is based on the experimentally verified hypothesis that normal stresses are directly related to the "hole" pressure produced by a fluid flowing past a solid boundary in which a small liquid-filled hole has been drilled. The "hole" pressure is specifically defined as the difference between the pressure measured in the static liquid at the bottom of the hole, and the pressure which the flowing fluid would exert against the solid boundary if the hole were not there. Since this concept lends itself to use in capillary-type viscometers, normal stress measurements at the very high shear rates representative of those in an operating engine should be possible in a laboratory instrument.

# The Effect of Oil Rheology on Bearing Film Thickness and Load Capacity

## THE EFFECT OF OIL RHEOLOGY ON BEARING FILM THICKNESS AND LOAD CAPACITY

### Analytical Models and Predictions

The fact that a lubricant film has, under certain conditions, sufficient strength to separate two loaded, sliding surfaces was intuitively known and used long before it could be analytically described. A fundamental understanding of hydrodynamic lubrication, defined as the lubrication occurring when sliding surfaces are separated by a continuous oil film, dates back only 100 years to the experiments of Tower (45,46) and the analyses of Reynolds (47) and Petrov (48). In particular, the assumptions and arguments used by Reynolds to solve the hydrodynamic flow equations in a bearing have served as a foundation for virtually all subsequent theoretical analyses of bearing performance. For a very simple, one dimensional, plain bearing, the equation which incorporates all of Reynolds' assumptions is:

$$\frac{d}{dx}\left(\frac{h^3}{\eta}\frac{dp}{dx}\right) = 6U\frac{dh}{dx} \tag{8}$$

where x is the sliding distance along the bearing, U is the relative sliding speed between the surfaces, h is the film thickness, p is the film pressure at x, and $\eta$ is the lubricant viscosity.

Integration for a bearing subjected to a constant load, W, defined as the force acting per unit bearing length, gives:

$$h = k\left(\frac{\eta U}{W}\right)^{0.5} \tag{9}$$

where k is a geometric constant for the bearing.

Because of its fundamental importance in theoretical bearing analysis, it is worth reviewing the specific assumptions used in deriving Equation 8. Probably the most important from the view point of this report is:

1. The lubricant is isoviscous. This assumption neglects not only the effect of temperature on viscosity, but also the effects of shear rate and pressure.
2. Constant lubricant density.
3. Laminar flow.
4. No side flow (leakage out the ends of the bearing).
5. Negligible external and inertial forces.
6. Motion only in the x-direction.
7. Rigid bearing and journal surfaces.
8. Zero lubricant slip at the bearing or journal surface. Finally, although it is an assumption that is so commonly used in hydrodynamic analyses that it is often accepted as fact, it should also be stresssed that Reynolds' equation assumes:

9. The lubricant represents a homogeneous continuum. This implies that the film thickness being calculated is large with respect to molecular or surface dimensions, and that any additives or base stock components in the lubricant remain in homogeneous solution.

Although it is clearly a generalization, it is nonetheless accurate to state that virtually all of the theoretical journal bearing analyses that have been made since Reynolds have attempted to relax the constraints imposed by one or more of the above assumptions. Such analyses have had as their objective to improve on the predictions of film thickness and load-carrying ability provided by Equation 9 for practical bearing systems. Reviews of published work on analytical descriptions of journal bearing performance have been written by Campbell et al. (49), and more recently by Martin (50). Although extensions of Reynolds analysis were attempted early, with the most notable success by Sommerfeld (51,52), sophisticated bearing analyses became a reality only in the 1950's with the development of improved computational facilities and techniques. The first constraints to be loosened were those which allowed the calculation of performance parameters for dynamically loaded bearings as described by Burwell (53), Milne (54), and Lloyd, et al. (55). The mobility method, developed by Booker (56), increased the computational efficiency of numeric solutions to the hydrodynamic equations, and was used to predict the performance of bearings of any finite length with oil film rupture (57). Finite element analyses have also been developed to predict the performance of many types of nonideal bearings (bearings with taper, misalignment, oil-supply hole, grooves, etc.) (58,59). Considerable effort has also been directed toward describing journal bearings that are subject to elastic deformations (58,60-67).

Although computers and numerical techniques have produced tremendous improvements in the ability to predict bearing performance for a wide variety of engine operating conditions and geometric designs, the predictions have fallen short in describing the effects of one major component in any engine journal bearing - the lubricant. All of the analyses previously referenced assume the lubricant to be isoviscous. Although in some analyses this constant viscosity is evaluated at the average values of temperature, pressure, and shear rate for the entire bearing at any instant (68,69), it is assumed that the viscosity does not vary from point to point within the oil film. The variation in viscosity due to variations in temperature and pressure at each point is ignored for those oils that can be considered Newtonian. Furthermore, of particular interest to this review, the variation in viscosity due to variations in shear rate at each point is, in many cases, ignored for those oils which contain high-molecular-weight polymeric additives and thus are, to some degree, non-Newtonian. Several recent papers have considered the case of non-Newtonian oils (70,71), but ignore the possibility of any viscoelastic behavior. The reasons for ignoring these effects are not trivial. First, the numerical techniques which account for those complexities already listed can require significant amounts of computer time. Adding additional relationships into the analysis to account for variations in viscosity with temperature, pressure, or shear rate would compound the computational time required, even for simple bearing systems. Second, although the rheological properties of Newtonian oils (in general, those single-grade oils containing minor amounts of polymeric additives) can be described accurately as a function of temperature and pressure using accepted analytical relationships, there is no universally accepted equation for describing the rheological behavior of polymer-containing, non-Newtonian oils (most multigrades). In addition to the dependence of viscosity on shear rate for these polymer-containing oils, there is also the question of how to accurately describe their elastic properties. Even if a sufficiently complex constitutive equation were available to accurately describe the rheological properties of polymer-containing oils under the high rates of shear and within the "squeeze films" that occur in an operating journal bearing, not even an analytical technique as powerful as finite element analysis could solve the hydrodynamic equations that would result from its use without consuming prohibitive amounts of computer time.

The search for a universal constitutive equation to describe the rheological properties of polymer-containing oils has also been under way for many years. In early work, Tanner (15), arguing that elastic normal stresses for most lubricating oils are negligible, used a Maxwell-type constitutive equation in combination with a power-law relationship for viscosity as a function of shear rate in an analysis of "squeeze film" lubrication. He concluded that shear thinning behavior reduces the load-carrying capacity of a bearing. Metzner (16,17), arguing that normal stresses cannot be neglected, also used a Maxwell-Oldroyd constitutive equation in combination with a set of kinematics which assumed extension rates were more important than shear rates to predict that polymer-containing, elastic fluids can significantly increase bearing load capacity compared to Newtonian lubricants.

In a later paper Williams and Tanner (18) argued that the Maxwell-Oldroyd constitutive equation was, by itself, too simplistic. Instead they used a form of the Network Rupture Theory (19) in combination with kinematics containing both a shear and an extensional flow component to derive equations which suggest, contrary to Metzner's conclusions, that extensional flow fields are not important in journal bearing geometries. Trying to resolve the conflict between the analyses of Tanner and Metzner, Davies and Walters (20) attempted to numerically solve the steady bearing lubrication equations for an Oldroyd model fluid. Because of the complexity of the resulting equations, solutions could only be obtained for small eccentricity values. For this limited case, they concluded that if a polymer-containing lubricant has the correct elastic and viscous properties, it could provide improved performance in journal bearings relative to a purely viscous oil.

Leider and Bird (21) questioned the validity of the steady or quasi-steady flow assumptions in all of the preceding analyses. They argued that any constitutive equation used to describe the flow of squeeze films must be able to accurately portray transient stress buildup and, similarly, "stress overshoot" - an experimentally verified phenomenon. Using an equation which predicts "stress overshoot" in transient shearing flows, they solved the squeeze film equations to demonstrate that for certain values of fluid elasticity, it is predicted that polymer-containing oils would provide thicker films for a given applied load than would Newtonian oils.

Although the equation used by Leider and Bird is not, strictly speaking, a valid constitutive equation (it contains a specific reference time), it does demonstrate the importance of being able to portray transient stress behavior. More impressively, in a related paper, Leider (22) provides experimental verification for his squeeze film predictions between flat parallel plates subjected to a fixed load.

Recently, Shirodkar and Middleman (23) and Shirodkar, Bravo, and Middleman (24) resolved the parallel plate squeeze film equations for a legitimate constitutive model suitable for describing transient stress experiments. The particular equation used portrays the fluid stress tensor as an integral function of the fluid strain multiplied by an exponential memory function. The memory function comprises a spectrum of fluid relaxation times and elastic moduli, and the integral is evaluated over the fluid's entire past history. Despite the complexity of this constitutive assumption, the predictions from the squeeze film analysis also suggest that, if the squeezing rate is high enough, fluid elasticity can increase the load capacity of polymer-containing oils relative to purely viscous fluids.

Although recent constitutive models for describing viscoelastic behavior suggest real benefits for polymer-containing oils, such models have only been applied to very simple flow patterns (i.e., squeeze films between parallel circular plates). There is currently little hope, because of the mathematical complexity involved, for using these models in more practical flow geometries such as between an unsteady eccentric bearing and journal. Even if a numerical scheme could be devised to use one of these constitutive relationships in solving the Reynolds equation, the fluid parameters in such an equation would have to be assumed constant with temperature and pressure, since little work has been done to determine the effect of these variables on fluid viscoelastic properties. The theoretical work done to date, although impressive in its mathematical complexity and analysis, falls short of what is needed to accurately describe the film thickness and load capacity of a bearing operating on a polymer-containing oil. Thus, to relate actual bearing performance to the viscometric or elastic properties of such oils, the use of correlations between engine and laboratory bearing experiments, and laboratory fluid properties measurements must be accepted.

## Oil Film Thickness Measurements

To experimentally determine the effects of oil rheology on bearing performance, effort has been directed toward measuring primarily three different quantities: bearing wear during a prescribed test, bearing oil film thickness, and bearing load capacity (the load required to cause contact between a bearing and journal). The first of these quantities will be discussed in a later section of this report dealing with engine wear. The latter two quantities are reviewed in what follows.

The desire to measure the dimensions of thin oil films in operating engines or mechanical laboratory rigs has been the goal of experimentalists for many years. Since direct physical measurement is unattainable, various indirect techniques and methods have been devised including measuring the electrical capacitance (72,76,82,84), electrical inductance (74,75,79,80), electrical resistance (78,83,85), magnetic field (73,81) and oil temperature rise (77) occurring between the bearing and the journal. In early work, Stone and Underwood (72) used a single-point capacitance measurement to determine the oil film thickness in a laboratory journal bearing rotating at constant speed. The bearing was subjected to a load of constant magnitude whose direction rotated at the same speed as the journal. Only results for one oil are presented in this work, but the data collected clearly demonstrate how the point of minimum film thickness in a bearing, in general, lags behind the load vector at any instant.

One of the earliest investigations of oil viscosity effects on journal bearing film thickness was conducted by Sims (73). By modifying one of the bearing caps on a V-8 engine so that it accepted a single transducer capable of indicating relative displacement by measuring the magnetic field produced by the rotating shaft, the film thickness at the bottom of the bearing could be monitored during engine operation. Sims found film thickness at this point to be only a slight function of speed, but to decrease with engine load substantially. Of the three oils tested, two Newtonian oils and one containing 7.5 volume percent of a polymethylmethacrylate VI improver additive, no effect of viscosity was measured. This anomaly was attributed to viscous heating effects since during the testing no attempt was made to control oil film temperature. Presumably the higher viscosity oils produced more viscous heating which, in turn, lowered their viscosity and associated film thickness to that of the lower viscosity oil. However, it should be recognized that a film thickness measurement at a single point in a test where the load vector is varying in both magnitude and direction will not, in general, identify the minimum film thickness at any instant. Thus, it is not known if the most critical measure of film thickness was determined in this work.

Rosenberg (74,75) developed an inductance technique which he used to measure the relative film thickness at a single point in a laboratory test bearing which was subjected to a rotating eccentric load of constant magnitude. Relative film thickness was measured because of difficulties in calibrating the zero film thickness point for the transducer, and because of temperature effects on the transducer calibration. In one paper (74), Rosenberg demonstrated that although relative minimum film thickness correlated with kinematic viscosity at 98°C (210°F) for five nonpolymer-containing oils, the film thickness values for four oils containing various VI improver additives did not. As shown in Figure 2, an oil containing a dispersant polymethacrylate additive and a commercial oil of unknown composition produced film thickness values lower than those for nonpolymer containing oils. An oil blend containing a polyisobutylene VI improver additive provided the same film thickness value, and another dispersant polymethacrylate-containing oil produced a film thickness greater than that of the nonpolymer-containing oils based on kinematic viscosity measurements.

In a later paper (75), Rosenberg examined the performance of as many as five base oils, sixteen laboratory polymer-containing blends, and six commercial multigrade oils in the same laboratory bearing test apparatus. In all cases, the polymer-containing oils provided lower friction, lower oil-film thickness (Figure 3), and higher oil flow to the bearing than did the base oils when compared on the basis of a Sommerfeld Number, S, defined as,

$$S = \frac{\eta N}{W} \left(\frac{R}{C}\right)^2 \tag{10}$$

where R is the bearing radius, C is the radial clearance, N is the rotational speed, W is the specific load on the bearing, and $\eta$ is the viscosity of the oil evaluated at the temperature of the bearing back. The differences among oils were attributed to different amounts of temporary viscosity loss with the polymer-containing oils at the high shear rates in the bearing rig. To justify this conclusion, it was demonstrated that oil friction data correlated in approximately a 1:1 fashion with oil film thickness. It was thus concluded that polymer-containing oils did not provide any significantly different friction or film thickness values than the base oils examined after accounting for temporary viscosity losses. McMillan and Murphy (12) came to this same conclusion in comparing data on eleven polymer-containing blends tested in Rosenberg's bearing rig with high-shear rate ($5 \times 10^5\ s^{-1}$) viscosities measured in a capillary viscometer at 98.9°C. High-shear viscosity correlated well with friction and oil flow, and moderately with oil film thickness, Figure 4, as determined in the bearing rig. Unfortunately, no comparison was made with a series of Newtonian oils on a similar basis to determine if the same degree of correlation could be achieved.

Van Os, Bos, van Namen, and de Rooij (76) did compare both Newtonian and polymer-containing oils in a steadily loaded laboratory bearing rig in which film thickness was measured with three capacitance probes. Under hydrodynamic lubrication conditions it was demonstrated that oil film thickness was the same function of high-temperature (150°C), high-shear rate ($2.5 \times 10^5\ s^{-1}$) viscosity for polymer-containing oils as for Newtonian oils, as shown in Figure 5. In this figure, the oil film thickness is plotted against a Sommerfeld number which is defined essentially as the inverse of that used by Rosenberg (75). In addition, the viscosities used by Van Os et al. in calculating their Sommerfeld numbers were evaluated at the high-temperature, high-shear rate conditions described, whereas Rosenberg used kinematic viscosities measured at the temperature of the bearing back. The result of these differences is that film

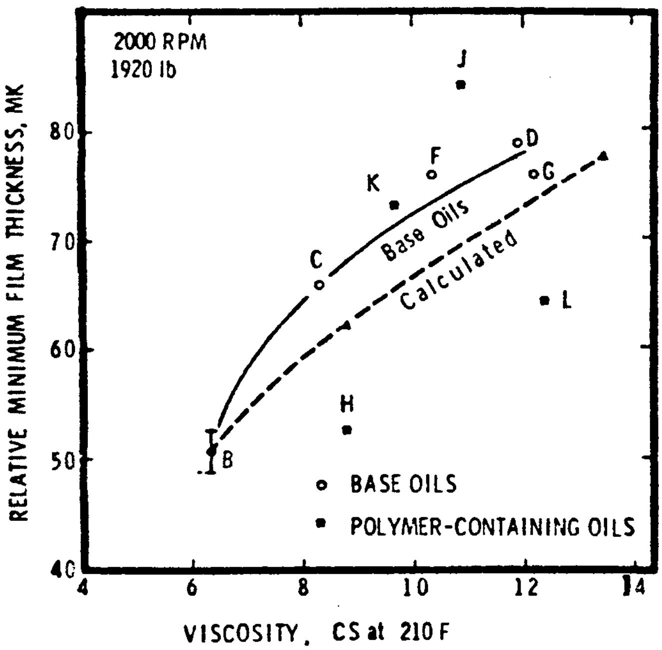


Figure 2. Relative minimum oil film thickness versus final test oil viscosity (Reference 74). Reprinted with permission ©1973 Society of Automotive Engineers, Inc.

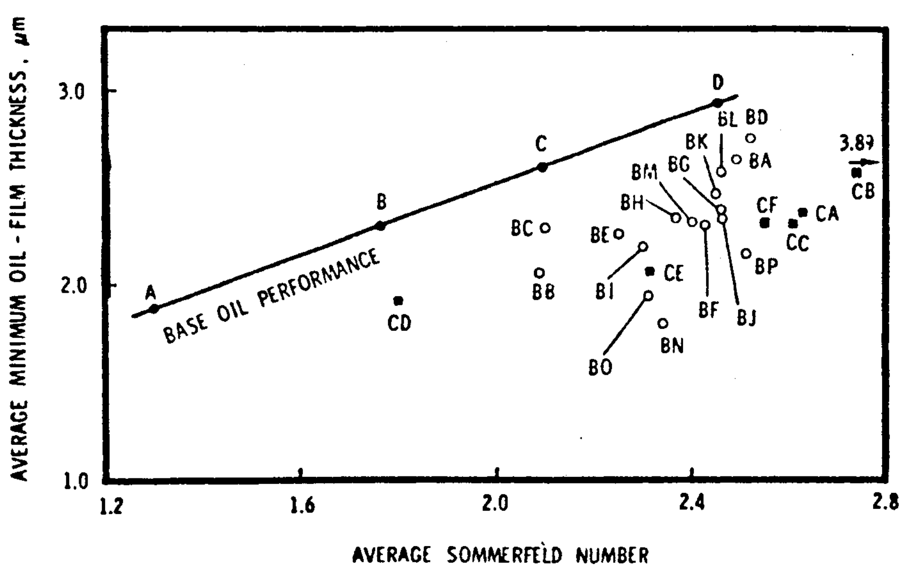


Figure 3. Film thickness of VI improved oils compared to base oil performance (Reference 75). Reprinted with permission ©1975 Society of Automotive Engineers, Inc.

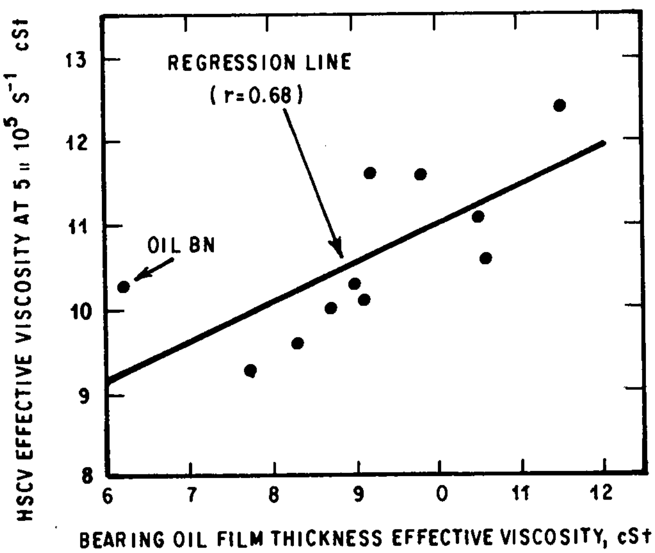


Figure 4. Correlation of HSCV and bearing oil-film thickness effective viscosities (Reference 12). Reprinted with permission ©1978 Society of Automotive Engineers, Inc.

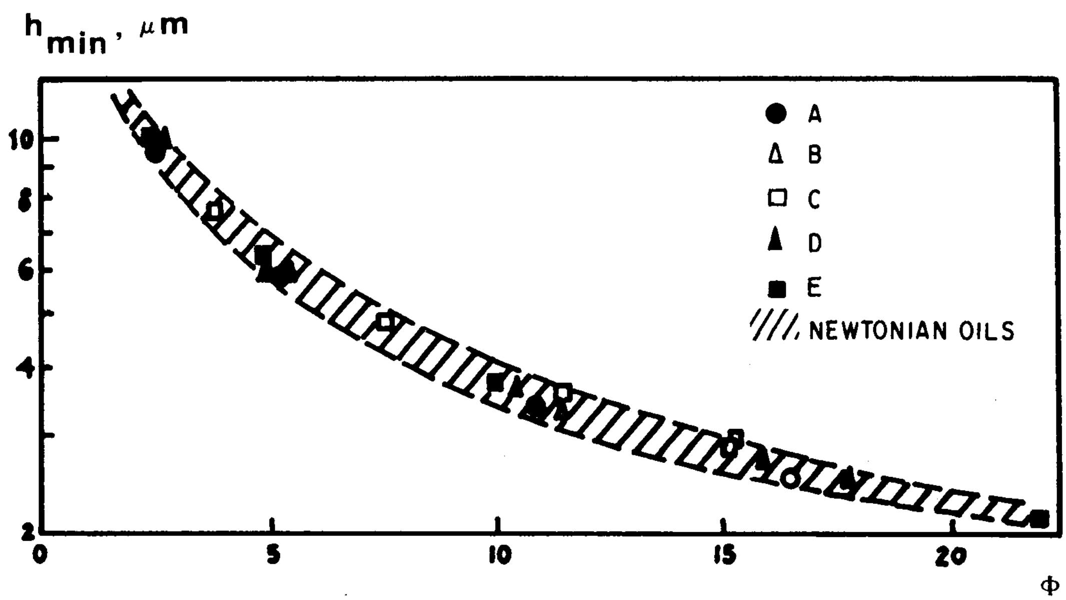


FIGURE 5
$h_{min}$ AS A FUNCTION OF THE SOMMERFELD NUMBER Φ

Figure 5. $h_{min}$ as a function of the Summerfeld number $\phi$ (Reference 76). Reprinted with permission ©1978 Society of Automotive Engineers, Inc.

thickness increases with increasing Sommerfeld number in Rosenberg's work, with the data for polymer-containing oils falling below those for Newtonian oils. In the work of Van Os et al., oil film thickness decreases with increasing Sommerfeld number and the polymer-containing and Newtonian oils provide similar oil film thickness values. Under more severe boundary lubrication conditions, Van Os et al. show that bearing wear is a function of the total additive package in the oil as will be discussed in other sections of this report.

Temperature increase in a big end bearing of a connecting rod in a four cylinder engine relative to sump oil temperature was used by du Parquet and Godet (77) to evaluate the lubrication provided by a series of four Newtonian and ten polymer-containing oils. Bearing temperature increased as engine load, speed and sump oil temperature increased. After using the four Newtonian oils to calibrate the bearing as a relative viscometer, the temperature rise for each polymer-containing oil provided an effective viscosity at engine operating conditions. As shown in Figure 6, the effective engine viscosities correlated well with a quadratic function of high-shear rate viscosity at 6 x $10^5$ $s^{-1}$. These high-shear rate viscosities were obtained by extrapolating data collected at 100 and 125°C to the bearing temperature for each oil using ASTM D 341 graphical procedures. How much of the observed quadratic behavior is real and how much is an artifact of the extrapolation procedure is unknown. However, a single correlation was obtained for both sets of oils.

At this point in the development of bearing analyses, several researchers published papers which described a variety of techniques for measuring oil film thickness in the bearings of an operating engine. Bassoli, Cornetti, and Bilei (78) developed the equations for calculating the oil film thickness in a bearing assuming the entire oil film is a simple electrical resistor, and that the bearing and journal are perfectly circular. Goodwin and Holmes (79) and Goodwin (80) described a single point induction transducer which compensates for temperature variations during engine operation. The transducer was used to measure oil film thickness in a big end bearing of a diesel engine. Ishihama, et al. (81), described a single point eddy current sensor used to measure oil film thickness in a main bearing. Dobson (82) provided a limited description of a capacitance bridge circuit which apparently assumes the entire oil film is a simple capacitor in calculating bearing oil film thickness. In none of these five papers, however, are the effects of Newtonian and polymer-containing oils on bearing film thickness compared, although several authors suggest such comparisons are possible.

A series of Newtonian and polymer-containing oils were compared in a main journal bearing of an operating engine using an electrical discharge technique described by Filowitz, King, and Appeldoorn (83). The degree or duration of electrical discharge during a given combustion cycle was equated to a relative oil film thickness. At a fixed set of engine operating conditions, the relative oil film thickness values for a series of Newtonian oils were smaller than for those of a series of polymer-containing oils when plotted versus a high-temperature (150°C), high-shear rate ($10^6$ $s^{-1}$) viscosity. Although, as suggested in the paper, this result could be due to viscoelastic effects in the polymer-containing oils, it could also have been simply caused by measuring viscosity at the wrong shear rate or temperature for correlating with the

| Curve | nu = | r = | |
|---|---|---|---|
| 1 | $1.21 nu_{HSR} - 2.06$ | 0.92 | r coefficient of correlation |
| 2 | $0.57 nu_{HSR} - 0.045 nu_{HSR}$ | 0.98 | nu effective viscosity |
| 3 | $0.76 nu_{LSR} - 0.07$ | 0.93 | $nu_{HSR}$ high shear rate viscosity |
| 4 | $1.75 nu_{BO} - 0.75$ | 0.92 | $nu_{LSR}$ low shear rate viscosity |
| | | | $nu_{BO}$ base oil viscosity |

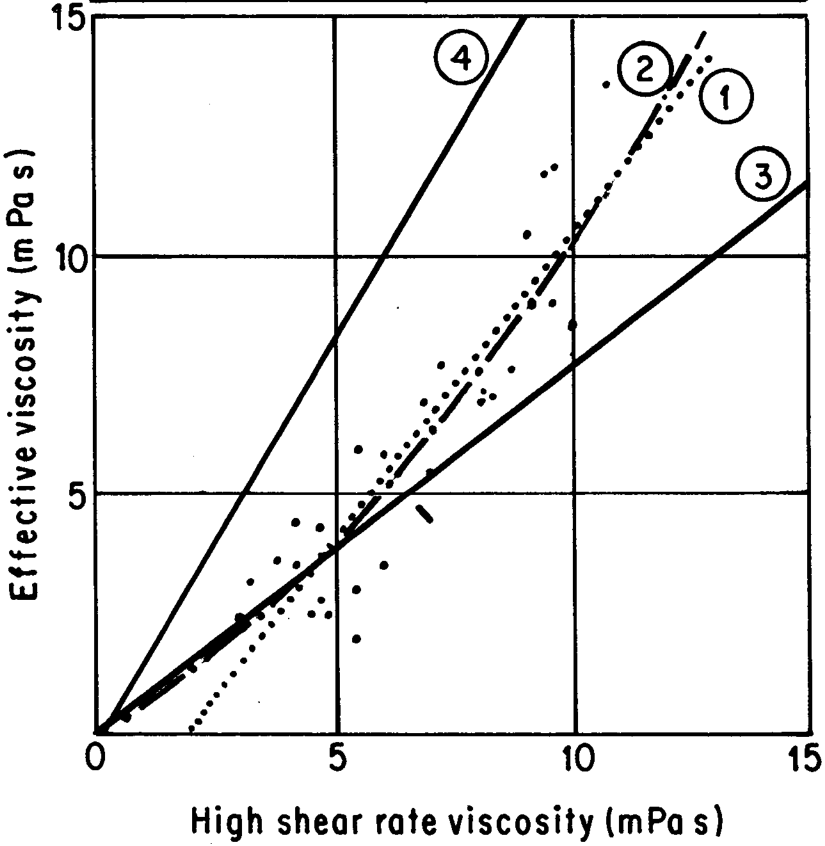


Figure 6. Correlation between effective viscosity and viscometric data (Reference 77). Reprinted with permission ©1978 Society of Automotive Engineers, Inc.

engine. Specifically, measuring viscosity at a lower shear rate could have caused the relative oil film thickness values for the polymer-containing oils to more closely align with those of the Newtonian oils. Since only relative film thicknesses were measured it was not possible to calculate the actual shear rates existing in the bearing. Craig, King, and Appeldoorn (84) described a technique for measuring the absolute minimum film thickness in the main bearing of an operating single-cylinder engine by assuming the entire oil film is an electrical capacitor. However, only Newtonian oil data are described. The technique is sufficiently precise to demonstrate that the minor fluctuations in the value of the minimum film thickness from cycle to cycle are related to the variation in peak cycle pressure in the engine used in the study.

In a subsequent paper, Girshick and Craig (94) used this same engine and capacitance technique to measure minimum oil film thickness values for six Newtonian and fifteen polymer-containing oils. Film thickness increased uniformly with both low- and high-shear viscosity for all oils. However, as shown in Figure 7a, the specific film thickness values measured for the polymer-containing oils were all lower than those for Newtonian oils when plotted versus kinematic viscosity at 100°C. When plotted versus high-shear rate ($10^6$ $s^{-1}$) viscosity at 100°C as shown in Figure 7b, the film thicknesses for the polymer-containing oils were still, in general, lower than those of the Newtonian oils, but the differences were smaller. No physical explanation was offered for the lower film thicknesses with the polymer-containing oils.

Spearot, Murphy, and Rosenberg (76) using a similar technique which measured oil electrical resistance rather than capacitance found that the minimum film thickness values for the majority of twenty-two commercial and experimental oil blends, both Newtonian and polymer-containing, correlated with the viscosity of the used oils measured at 150°C and 5 x $10^5$ $s^{-1}$, conditions which were believed to represent those within the oil film. As shown in Figure 8a, when all of the oils are considered, a good correlation is achieved for single-grade oils ($R^2$ = 0.80), while a poor correlation exists for multigrade oils ($R^2$ = 0.04). In evaluating each of the data independently, however, it is reported that four multigrade oils, ranging from SAE 10W-40 to SAE 20W-50, provided film thickness values which were different from a single correlation line through the remainder of the data by an amount that could not be attributed to random experimental error. Three of the anomalous polymer-containing oils provided film thickness values greater than the correlation line, while one provided a lower film thickness. As shown in Figure 8b, the correlation between the film thickness values provided by the remaining eighteen oils and used oil viscosity at 150°C and 5 x $10^5$ $s^{-1}$ is good. No specific cause for the deviation of the four anomalous oils was identified, although it was concluded that other physical properties of the oil could affect bearing oil film thickness values.

Another recent paper by Van Os, Garcia-Franco, Gottenberg, and Trip (86) presented data which were obtained from a steadily loaded laboratory bearing rig in which oil film thickness was measured using eight eddy current probes, positioned in pairs, at four locations around the circumference of the bearing. Data collected showed no difference in oil film thickness between two Newtonian and seven polymer-containing oils provided the viscosity of each oil was

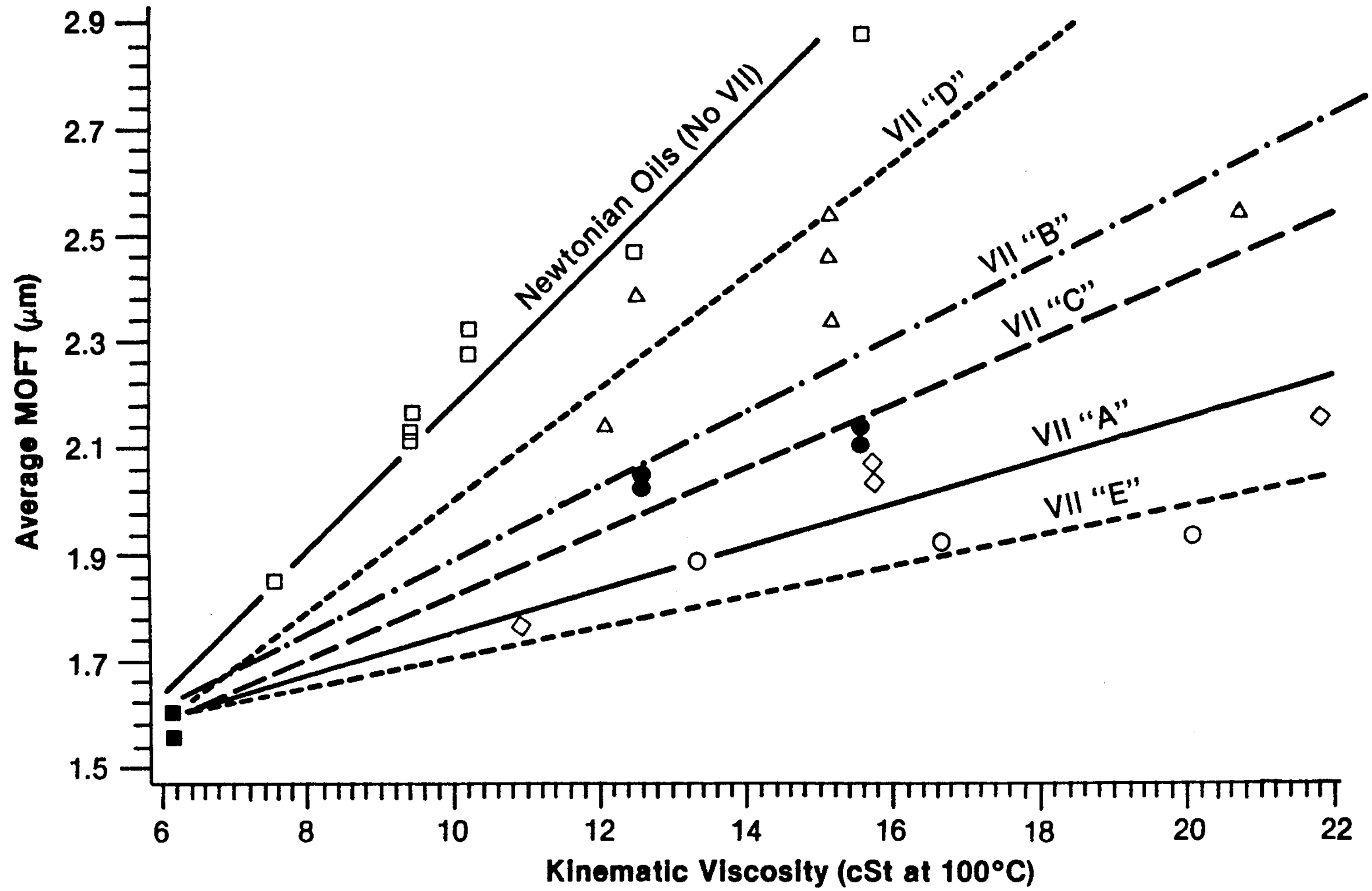


Figure 7a. The effect of different VI improvers on minimum oil film thickness (kinematic viscosity at 100°C) (Reference 94). Reprinted with permission ©1983 Society of Automotive Engineers, Inc.

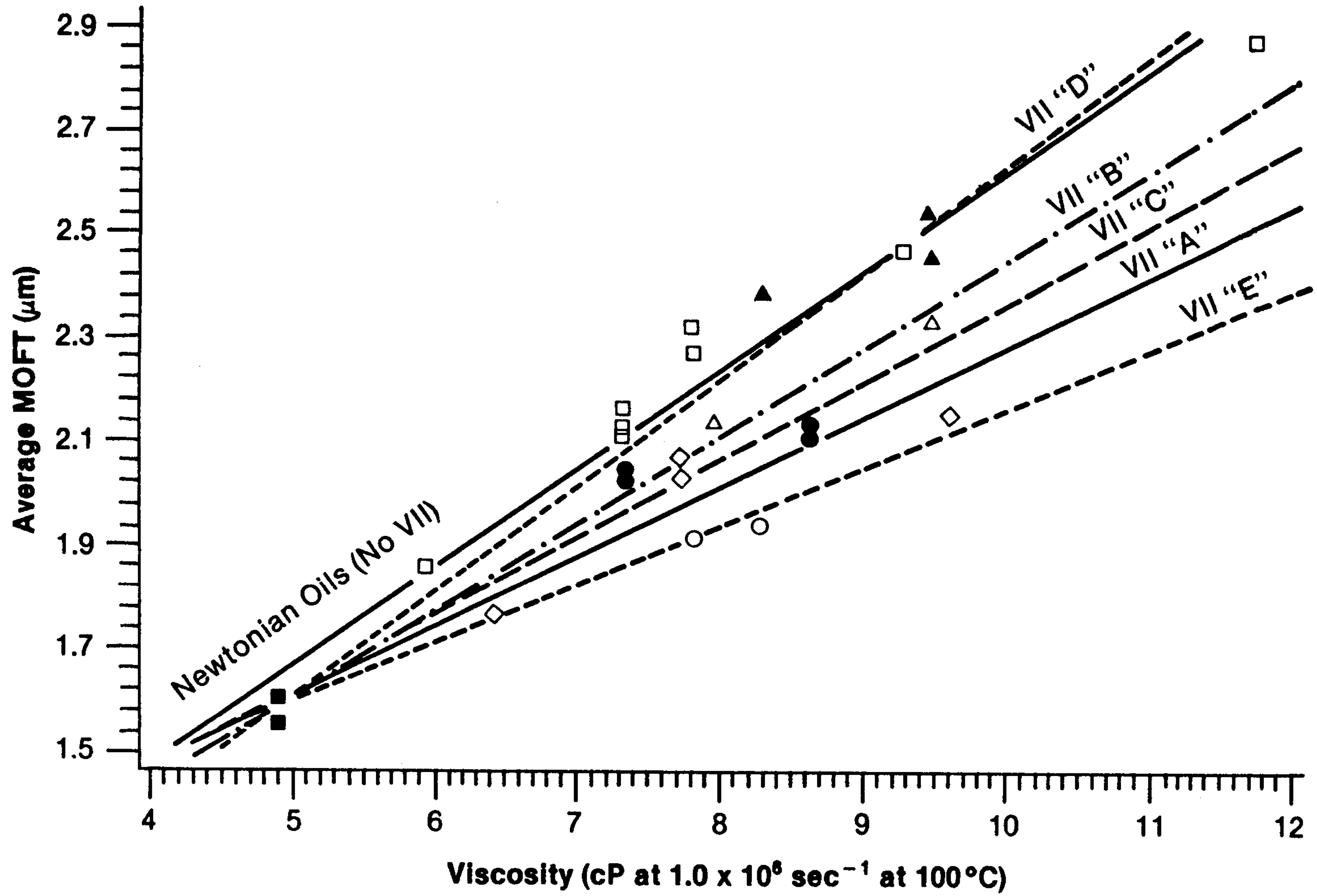


Figure 7b. The effect of different VI improvers on minimum oil film thickness (high-temperature, high-shear-rate viscosity at 100°C) (Reference 94). Reprinted with permission ©1983 Society of Automotive Engineers, Inc.

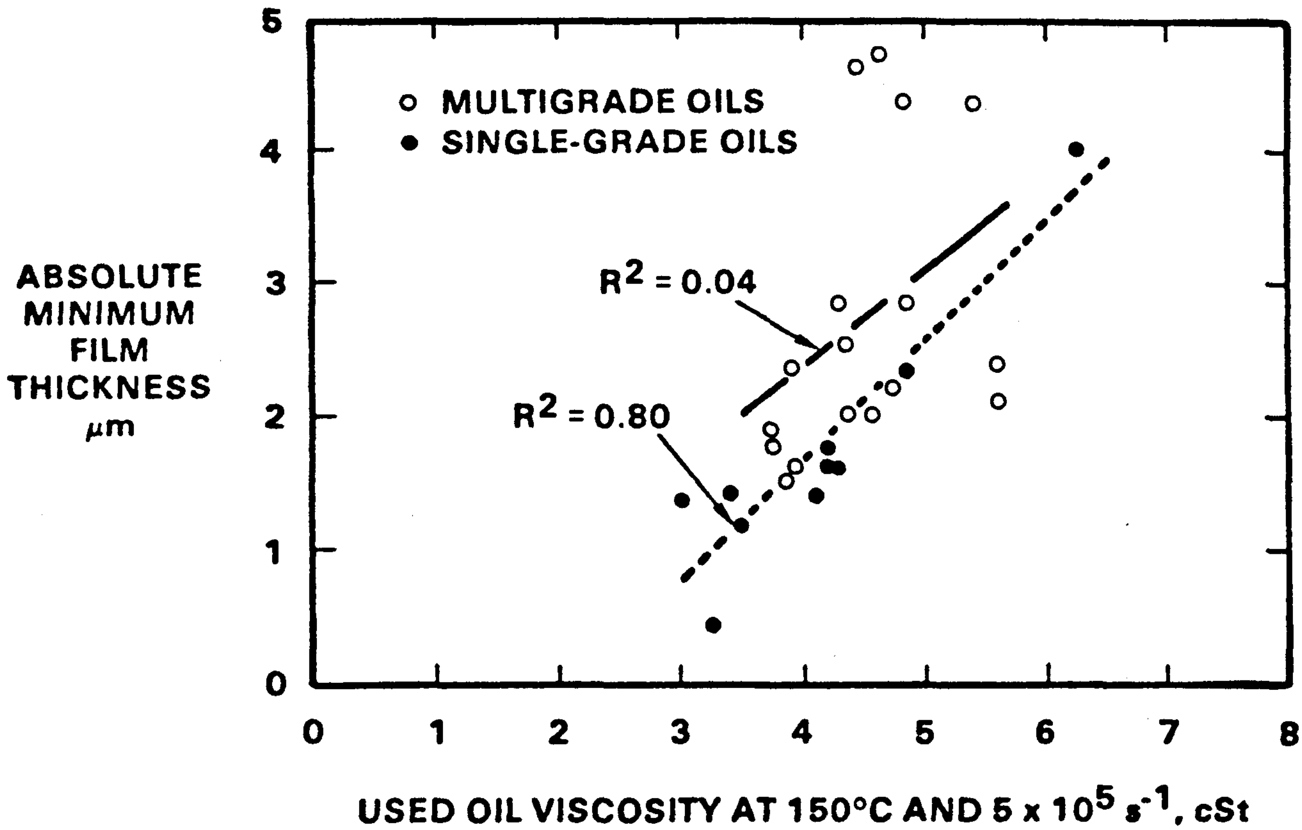


Figure 8a. Individual correlations between absolute minimum film thickness and viscosity for single- and multigrade oils (Reference 85). Reprinted with permission ©1983 Society of Automotive Engineers, Inc.

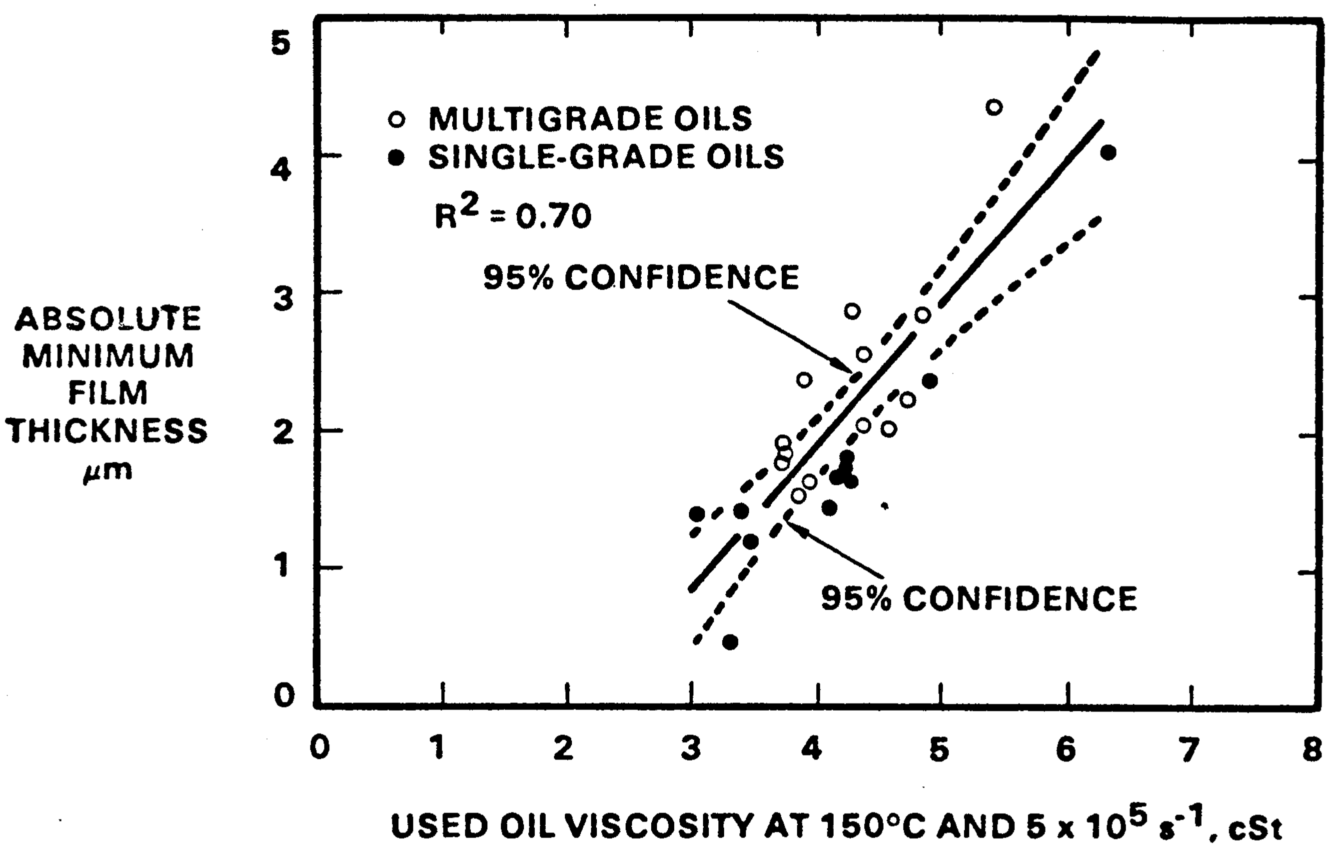


Figure 8b. The correlation between absolute minimum film thickness and viscosity for oils which lie within acceptable limits of a linear regresssion (Reference 85). Reprinted with permission ©1983 Society of Automotive Engineers, Inc.

evaluated at the temperature and shear rate of the bearing. The attitude angle of the bearing (the angle between the load line and the line of minimum film thickness) was significantly less for the polymer-containing oils than for the Newtonian oils (a fact which the authors attribute to viscoelasticity), but it has not been firmly established whether this has any practical effect on bearing performance.

For this review to be complete, it should be mentioned that attempts to measure oil film thickness between the compression rings and the cylinder wall of an operating engine have also been made. Recent work using a capacitance technique by Furuhama, Asaki, and Hiruma (87), and Shin, Tateishi and Furuhama (88) have demonstrated the feasibility of such a technique, but they have not investigated the effects of polymer content on oil film thickness.

## Load Capacity Measurements

Load capacity in this report is defined as the load applied to a bearing system at which contact between the journal and bearing occurs. As in the measurement of bearing oil film thickness, several different methods have been used to measure load capacity. These include radiometric (89,90), temperature rise (91), and electrical contact (92,93) techniques. These references exclude those that claim to have measured bearing load capacity by measuring wear, as they will be described in a later section of this report.

In constant load, radiometric experiments conducted in laboratory bearing rigs, DeHart, Rosenberg, Hill, and Schneider (89) demonstrated that load capacity for four oils, two Newtonian, and two containing polymethacrylate VI improvers, correlated with oil viscosity measured at 110°C and $6 \times 10^5\ s^{-1}$. In a later paper, however, Schneider and Rosenberg (90) found at least one commercial multigrade oil which contained an insoluble friction modifier that provided significantly greater bearing load capacity than did other polymer-containing oils.

In a series of papers Dancy, Marshall, and Oliver (92) and Oliver and Dancy (93) described a steady load experiment in a laboratory rig in which bearing load capacity was measured by an electric contact technique. For all oils investigated, including multigrades formulated with both olefin copolymer and styrene-ester VI improvers, the load at which contact between the journal and bearing occurred correlated with the viscosity of the oil measured at 100°C and $10^6\ s^{-1}$.

Finally, in a recent engine test in which bearing load capacity was measured by temperature rise in the bearing, Kotani, Yamada, Okumura, and Kobayashi (91) demonstrated for a series of thirty-three oil blends that both temporary and permanent shear stability of the polymer-containing oils must be considered in relating bearing load capacity to oil viscosity.

# The Effect of Oil Rheology on Engine Wear

## THE EFFECT OF OIL RHEOLOGY ON ENGINE WEAR

Engine wear is dependent on several properties of both the engine and the engine lubricant. For the engine, these properties include operating parameters, such as oil flow rates, oil temperature and pressure, and the physical and metallurgical properties of the surfaces that are subjected to conditions that might initiate wear. For the lubricant, the reduction of service life is an important consideration. Combustion products cause additive depletion. Sludge and acids build-up as the oil becomes contaminated with materials that can interfere with the normal mechanisms by which the oil impedes engine wear. Viscosity changes will occur that are the result of oil oxidation, contaminant entrainment, and polymer degradation.

The subject of this portion of the status report, however, is not oil degradation, but rather the effect of the rheology of new oil on engine wear. It is well-established that the viscosity of single-grade, Newtonian fluids can affect engine wear, but the rheological properties of non-Newtonian multigrade oils are more complex and more difficult to measure at conditions representative of the primary engine wear regions. With these lubricants, it is possible that engine wear is a function of more than one rheological parameter.

Before the effect of lubricant rheology on wear is reviewed, it is advisable to consider the types of lubrication that occur in the internal combustion engine. The lubrication regimes of interest can best be considered by observing the frictional characteristics of a journal bearing that is lubricated by a Newtonian fluid. As shown in Figure 9, when the coefficient of

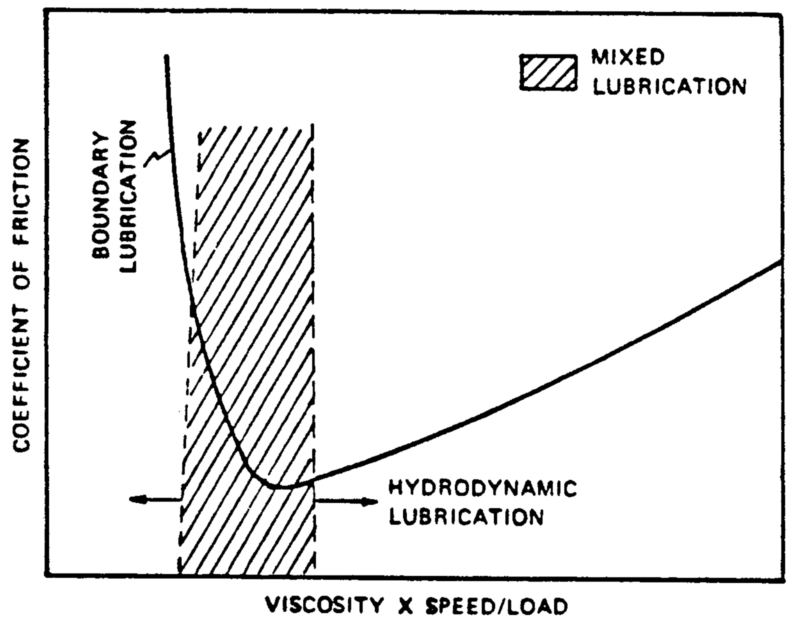


Figure 9. Coefficient of friction in a journal bearing.

friction is plotted versus $\eta N/W$, where $\eta$ is viscosity, N is rotational speed, and W is specific load, the hydrodynamic, mixed, and boundary lubrication regimes are described by the so-called Stribeck curve. For any N/W ratio, the viscosity of the lubricant will determine the lubrication regime in which the bearing operates.

If the viscosity of the Newtonian lubricant is insufficient to maintain full-film lubrication, then mixed or boundary lubrication will occur in which metal-to-metal asperity contact takes place. In boundary lubrication this asperity contact occurs throughout the high-pressure region between loaded surfaces, whereas in mixed lubrication there are regions of asperity contact mixed with regions of continuous fluid films. Once these regions begin to prevail over hydrodynamic lubrication, where a continuous fluid film exists, then wear may no longer correlate simply to viscosity. DeHart and Harwick (95) recognized this in a 1969 paper which reviewed bearing design principles. They indicated that bearing load capacity can be derived from either boundary or hydrodynamic lubrication conditions. For boundary lubrication conditions, they discussed the importance of a renewable surface film that was dependent on the chemistry of the additives in the lubricant.

When contact pressures become sufficiently high, the metal surfaces can deform and elastohydrodynamic (EHD) lubrication will occur. In this lubrication regime the chemical composition of the surface films, the high-pressure viscosity of the lubricant, and the elasticity and roughness of the surfaces become important parameters. The valve train is usually the engine area that is most exposed to EHD/boundary lubrication and it is therefore expected that wear here should be most affected by the chemistry of the lubricant. Successful lubrication of this area is usually achieved by carefully matching the extreme pressure/antiwear additives with other lubricant additives so that good valve train protection is achieved.

The piston ring-belt area is exposed to considerable boundary lubrication at the top-dead-center and the botton-dead-center of the piston stroke, but, unlike the valve train, there is also extensive hydrodynamic lubrication. This area is more likely to be affected by oil rheology considerations than the valve train, although oil chemical effects are also likely to affect cylinder/piston ring wear.

The engine journal bearings, however, are primarily designed to operate in the hydrodynamic lubrication regime. Except for transient boundary lubrication conditions during engine start-up, shutdown, or unusually heavy lugging under load, the engine bearings should operate with full lubricant films between the journal and the bearing. Bearing-to-shaft contact should thus be controlled by the rheological properties of the bulk lubricant.

Based on these considerations, it is not surprising to find that the number of studies that observe a relationship between oil rheology and engine wear decline in the order: bearing wear, cylinder wear, valve train wear.

## Journal Bearings

Studies of the lubrication of journal bearings can be divided into experiments which are designed to control the lubrication regime so that either hydrodynamic or boundary lubrication predominates. As reviewed in the other sections of this report, hydrodynamic lubrication studies may be characterized by the investigation of lubricant rheology effects on bearing minimum film thickness, oil flow rates, friction, and determination of the threshold of entry from hydrodynamic lubrication to mixed or boundary lubrication. Boundary lubrication studies are concerned with how oil properties affect bearing wear or seizure. These experiments are designed to cause significant shaft-to-journal contact so that actual bearing wear can be measured. It is this type of experiment which is reviewed in the following material.

Journal Bearing Rig Studies. Early journal bearing rig experiments by Rudinger (96), using a radiotracer technique to detect wear, indicated that multigrade oils gave poorer load-carrying capacity than single-grade oils with the same low-shear rate viscosity. This same type of experiment was later used by DeHart et al. (89), where load-carrying capacity was defined as the point where the applied load results in metal-to-metal contact as detected by the measure of radioactive wear debris. From test results on two multigrade oils and several single-grade oils, these investigators concluded that there was a correlation between the load-carrying capacity and the high-shear rate viscosities of the oils, as measured at $6 \times 10^5$ $s^{-1}$ and 110°C.

Van Os et al. (97) obtained actual bearing wear data using a statically loaded journal bearing rig. Fifteen polymer-containing oils were evaluated and found to give reduced wear relative to single-grade oils. Wear was found not to be influenced by high-shear rate viscosity, at least for fluids having viscosities between 2.1 cP and 4.9 cP at 150°C and $1 \times 10^6$ $s^{-1}$. Single-grade oils, however, gave increased wear in this viscosity range. Van Os also reported in other work (76) that the wear due to two SAE 10W-50 oils was lower than single-grade oils having the same viscosity in the journal bearing rig and that a change in the composition of the additive performance package could reduce wear by 60%.

Engine Bearing Studies. Okrent (98) reported over 20 years ago that polymer-containing oils could actually reduce friction, including friction in the mixed lubrication regime of a motored engine. Although this would appear to indicate that more than one lubricant physical property, i.e., more than low-shear rate viscosity, is required to characterize the frictional behavior of an engine journal bearing operating with a non-Newtonian fluid, it is important to note that Okrent did not have the facilities to measure high-shear rate viscosities.

In addition to the motored engine experiments, Okrent (98) studied the effect of lubricant composition on connecting-rod bearing wear in a fired engine. As shown in Figure 10, polymer-containing oils gave lower wear rates than their mineral oil counterparts that had the same low-shear rate viscosities at 98°C (210°F). When the concentration of any polymer was increased to give higher kinematic viscosities, using the same base oil and additive package, wear rate was reduced. The wear rate tended to level off as test oil viscosity

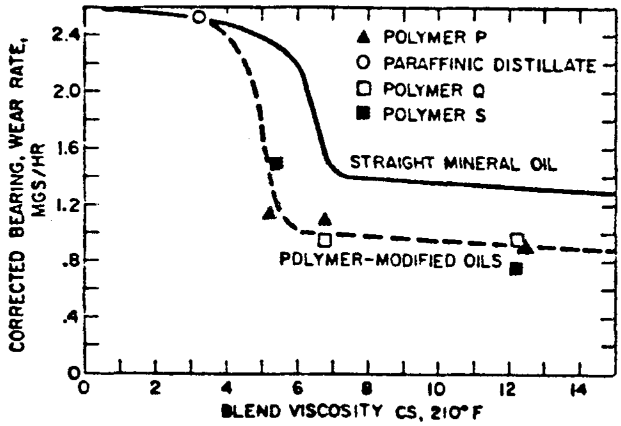


Figure 10. Effect of blend viscosity on bearing wear rate (3000 rpm, 90 ihp -- paraffinic distillate A base oil) (Reference 98). Reprinted with permission of the American Society of Lubrication Engineers. 

approached 6.5 cSt at 98°C (210°F). The wear reduction of polymer-containing oils was found to depend on polymer type and concentration, Figure 11. Further work (99) showed that wear rates were also sensitive to detergent additives. Savage and Bowman (100) also reported that polymer-containing oils were better than Newtonian oils and Savage (101) provided performance data from taxicab service and over-the-road gasoline engine truck service that indicated that multigrade oils gave markedly lower bearing wear than single-grade analogs. Okrent (98) suggested that the reason for the improvement with polymer-containing oils was due to viscoelasticity and perhaps surface adsorption of the polymer molecules. Later, experiments were carried out to measure the recoverable shear of polymer-containing oils using a vibrating crystal; a correlation was observed between this measure of viscoelasticity and bearing wear (102).

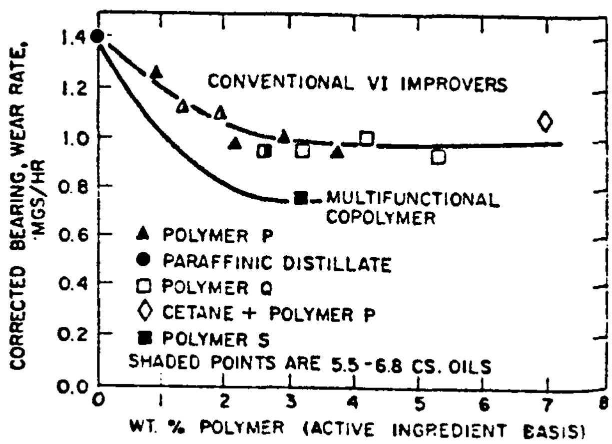


Figure 11. Effect of polymer concentration on bearing wear reduction (12.4 cSt blended oils) (Reference 98). Reprinted with permission of the American Society of Lubrication Engineers. 

Further bearing wear data were obtained by Neudörfl (103) in 1976. Fired engine tests were used to determine actual bearing weight loss, after subjecting test engines to polymethacrylate-containing oils. Bearing wear decreased as polymer concentration in the oil increased. Wear of the non-Newtonian fluids was found to be either higher or lower than single-grade oils with the same kinematic viscosities, but wear could be reduced by maintaining the low-shear rate viscosity and increasing the polymer content. It was also observed that the substitution of a low-molecular-weight polymer for a high-molecular-weight polymer at the same concentration in oil gave reduced wear.

There have been other reports that in field service multigrade oils perform better than single-grade oils which have the same low-shear rate viscosities. If high-shear rate viscosity dominates the wear mechanism, one would expect that multigrade oils would provide less wear protection. Waddey and Pearce (104) found, however, that an SAE 10W-40 oil gave lower gasoline engine connecting-rod bearing wear than an SAE 40 grade oil while Overton et al. (105), found reduced bearing wear in diesel engine service for multigrade oils relative to single-grade oils.

All of the authors thus far referred to in this section reported evidence for enhanced protection of bearings with some, if not all, multigrade oils. This evidence was called into question in two papers reported in 1977. Bell and Voisey (106) studied the behavior of a series of multigrade oils versus single-grade oils in laboratory engine bearing wear tests. Two different VI improvers and one performance package were used to prepare a series of test oils. Points P and Q in Figure 12 represent total bearing wear obtained with SAE 10W-30 and 10W-50 oils which contain different concentrations of the same VI improver. Wear for these two oils, as well as multigrade oil N, were found to correlate with kinematic viscosity; however, when results for single grade oils S and R were also evaluated, high-shear viscosity gave the best correlation with bearing wear.

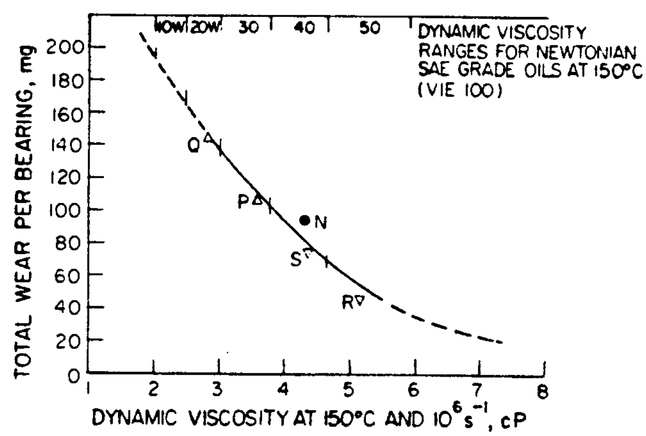


Figure 12. Relationship between high-temperature, high-shear rate viscosity and bearing wear (Reference 106). Reprinted with permission ©1977 Society of Automotive Engineers, Inc.

Stambaugh, Kopko, and Bennett (107), following up leads from earlier work (108), developed an on-the-road vehicle test designed to severely distress engine bearings. A relatively large sampling of Newtonian and non-Newtonian oils was evaluated in the vehicle tests and also in a pumping test designed to estimate the temporary shear stability of the test oils. The authors concluded that SAE 5W-20 fluids were not satisfactory and that a high-shear rate viscosity of at least 2.8 cP (3.5 cSt) at $5 \times 10^5\ s^{-1}$ was necessary to provide adequate protection. They also observed that base oil composition appeared to influence wear inhibition. Pike and coworkers (37) later evaluated the test oils in a high-shear rate viscometer and observed that the pumping test viscosities had provided Stambaugh with realistic estimates of high-shear rate viscosity.

One year later, McMillan, Rosenberg, and Murphy (109) reported on the results of a high-speed track test, using four test cars equipped with V-6 engines. Four SAE 10W-40 oils, identical except for VI improvers, were evaluated. The test oils were subjected to exceptionally severe service conditions so that their kinematic viscosities rose markedly during the test; filter plugging was reported during the operation of each test so particle entrainment may have been largely responsible for the increase in used oil viscosity. From the wear measurements, Figure 13, obtained on both main and connecting-rod bearings the authors concluded that bearing wear correlated with high-shear rate viscosity of the new test oils. Furthermore, they concluded that a viscosity of at least 3.6 cP (4.5 cSt) at $6 \times 10^5\ s^{-1}$ was required to provide protection under these driving conditions with this particular (V-6) engine.

Several years later, Lonstrup and Smith (110) demonstrated that components of the oil's detergent/inhibitor package can affect engine bearing wear to a greater degree than does oil viscosity. They evaluated additive package components using the ALI Bearing Distress Test (a laboratory engine bearing wear test) after this test was shown to rank the four oils used by McMillan et al. (109), in the same order as in the vehicle tests. A commercial detergent/ inhibitor package was shown to produce 25% less bearing wear than the experimental package used by McMillan et al., when blended without VI improvers in the same 200 N base stock. Both oils were essentially "Newtonian", and had the same SAE 20 grade viscosity. Further research demonstrated significant changes in bearing wear with component substitution and additions to the experimental additive package. Lonstrup and Smith (110) also analyzed taxicab data and found a relationship between oil pentane insolubles and average connecting rod bearing wear as shown in Figure 14. This figure also shows that the data from the track test conducted by McMillan et al. fit this correlation.

Further work by Rhodes and Henderson (111) confirmed the importance of the chemical nature of the oil to bearing protection in the ALI bearing wear test. The ALI test results of the four track-test oils were compared with a series of multigrade and single-grade oils which contained a different additive system. While the critical high-shear rate viscosity of the track-test oils was approximately 3.6 cP at $1 \times 10^6\ s^{-1}$ and 150°C, as shown in Figure 15, the oils formulated by Rhodes gave a satisfactory level of inhibition of bearing wear at a viscosity as low as 2.3 cP at the same conditions. Further evidence was presented that illustrated the effect of additive chemistry on bearing wear in this

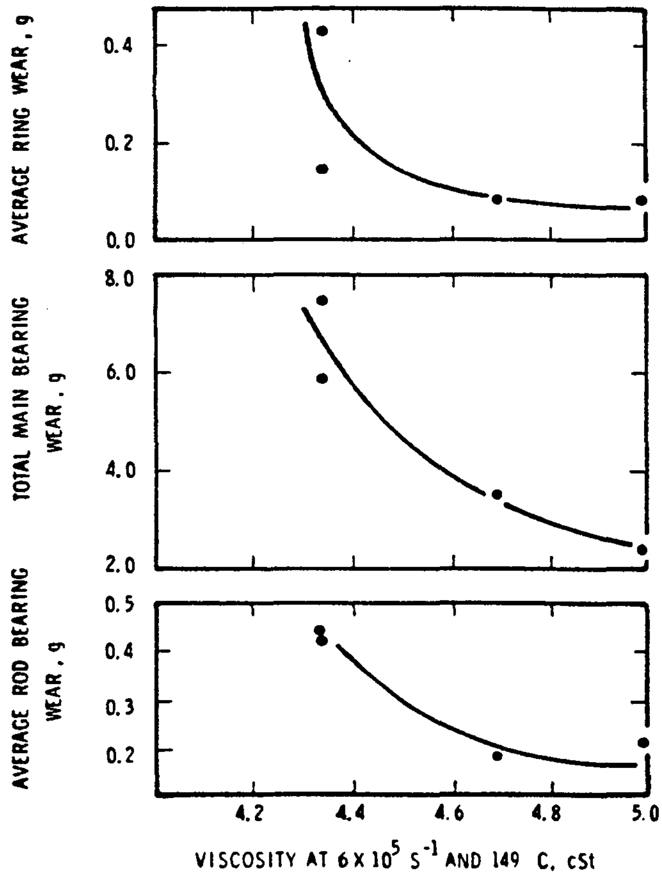


Figure 13. Relationship between wear and high-temperature, high-shear rate viscosity (Reference 109). Reprinted with permission ©1978 Society of Automotive Engineers, Inc.

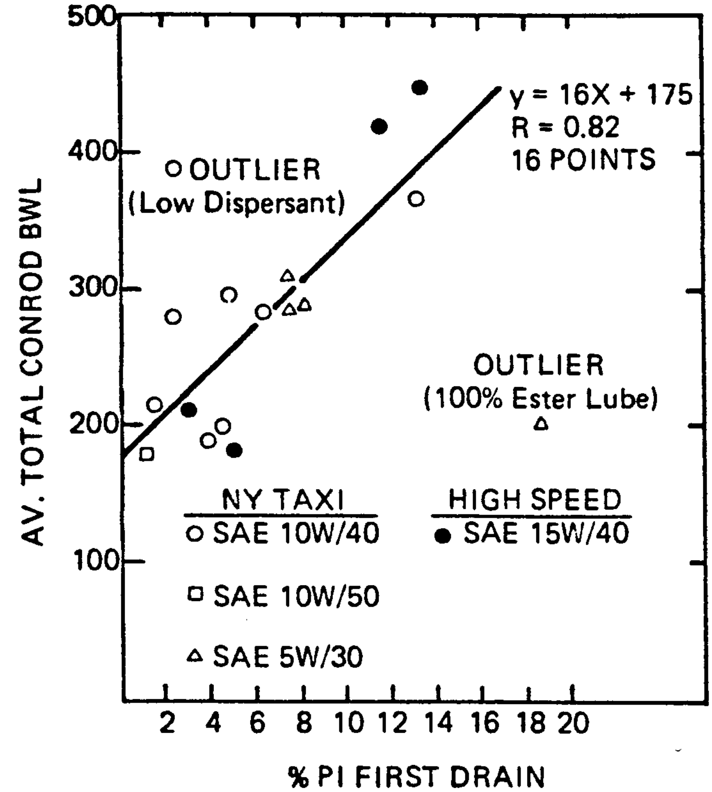


Figure 14. Con rod bearing wear-pentane insolubles first drain - 3 cabs or more (Reference 110). Reprinted with permission ©1981 Society of Automotive Engineers, Inc.

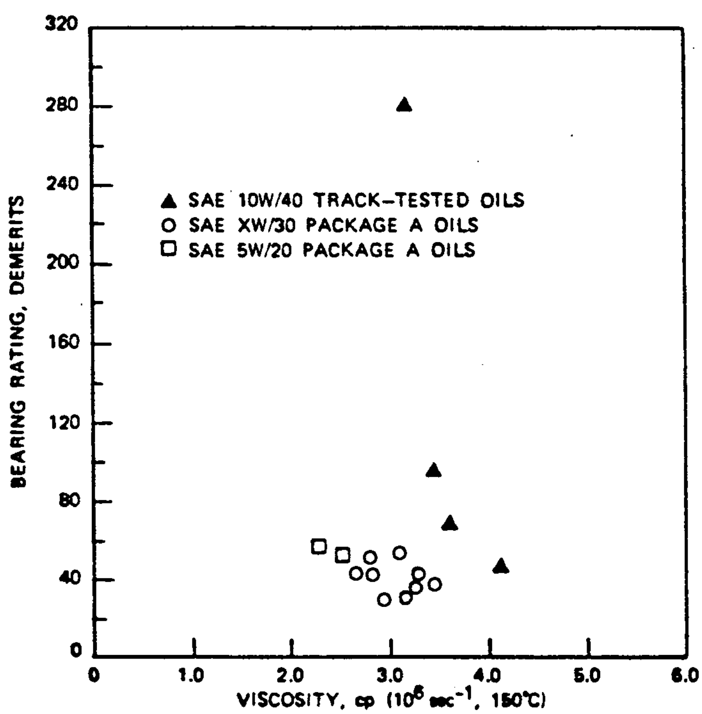


Figure 15. HTHSR viscosity and bearing demerits. All multigrade oils (Reference 111). Reprinted with permission ©1981 Society of Automotive Engineers, Inc.

test. Also, for a series of polymer-containing multigrades which contained the same performance package, the correlation between high-shear rate viscosity and wear was poor. Only when the same VI improver was evaluated at two different concentrations in the same package and base oil, could wear be related to either high-shear rate viscosity or kinematic viscosity.

In a different set of experiments using the ALI bearing test, Hutton, Jones, and Bates (112) also reported poor agreement between high-shear rate viscosity and bearing wear; however, when high-shear rate, high-pressure viscosity was compared with the wear results, the correlation was much improved, Figures 16 and 17.

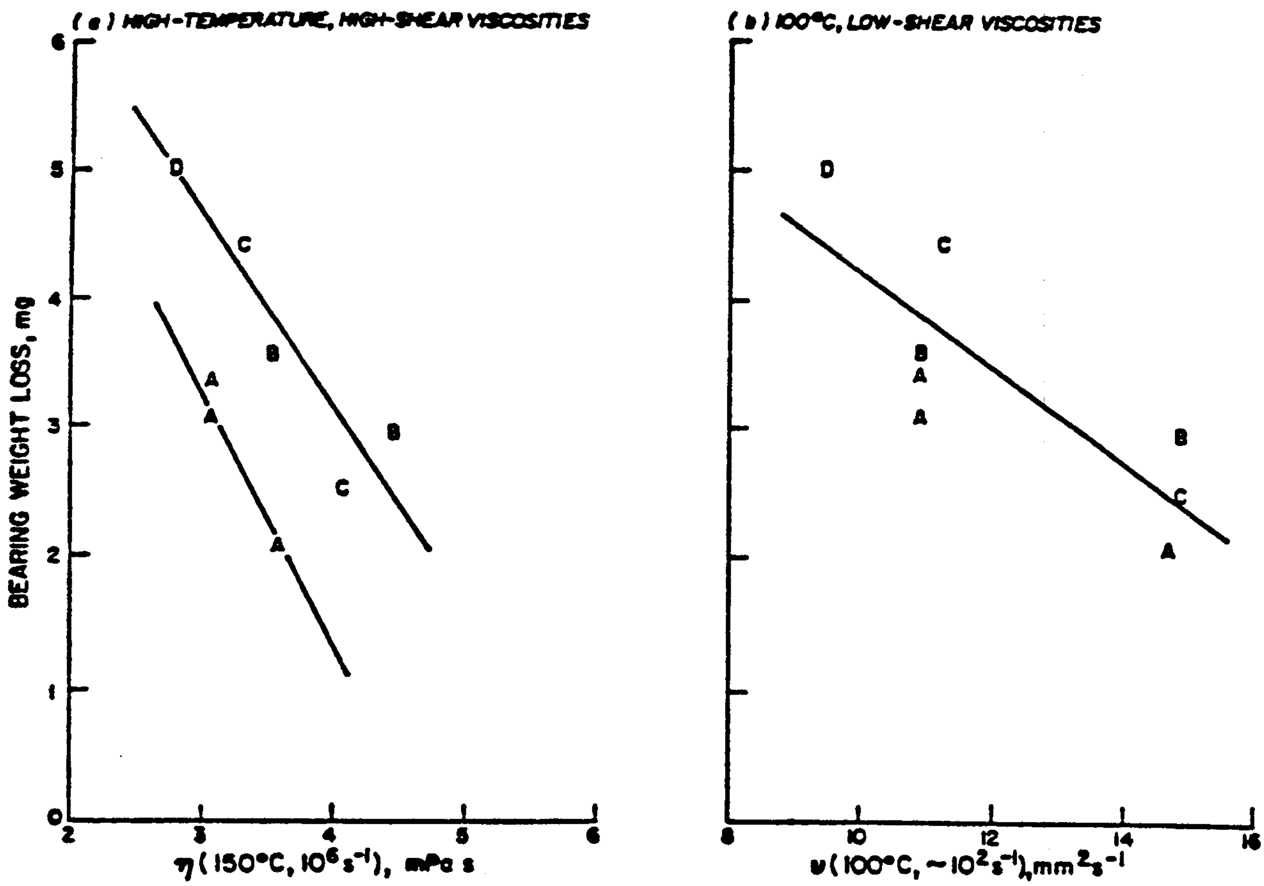


Figure 16. Dependence of bearing weight loss (ALI bearing distress test) on atmospheric pressure viscosities of 10W-30 and 10W-40 motor oils based on different VI improvers (Reference 112). Reprinted with permission ©1983 Society of Automotive Engineers, Inc.

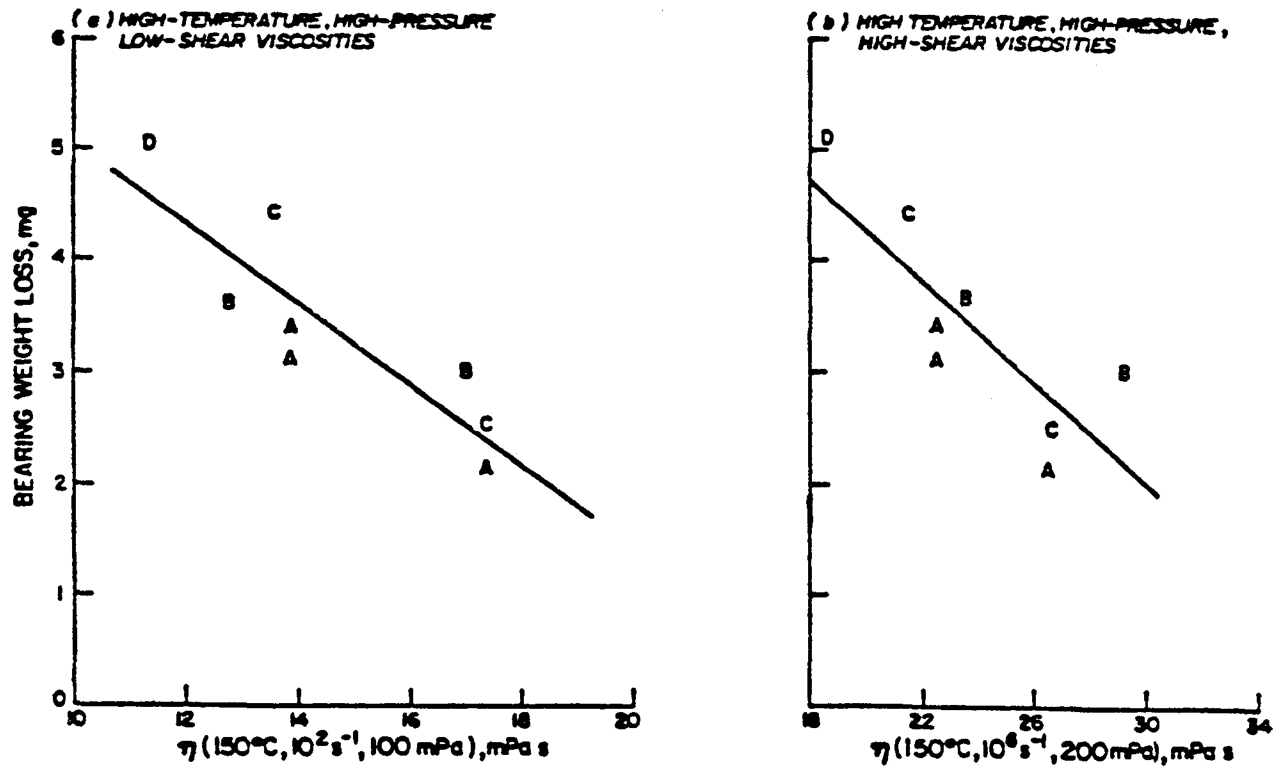


Figure 17. Dependence of bearing weight loss on high-temperature high-pressure viscosities of 10W-30 and 10W-40 motor oils based on different VI improvers (Reference 112). Reprinted with permission ©1983 Society of Automotive Engineers, Inc.

Concern over bearing wear protection led Flathmann, Barker, and Wylie (113) to compare the performance of an SAE 0W-30 and an SAE 10W-30 oil in a three-car, 80,000 kilometer, highway-speed test program. The test cars selected contained engines that were thought to be most sensitive to low-viscosity lubricants. It was found, however, that both oils provided satisfactory bearing wear performance, even though one of the oils had a viscosity at 150°C and $1 \times 10^6$ $s^{-1}$ of less than 2.64 cP. The authors proposed that additive effects might explain, to some extent, the fact that these oils were able to perform well at a much lower viscosity than the track-test oils of McMillan. However, it should be noted that the temperature of the sump oil in the vehicles described by Flathmann, Barker, and Wylie averaged less than 110°C during their test program. Based on this fact, viscosity at 150°C may not be the appropriate property to use in correlating with the bearing wear results.

The relative antiseizure performance of a large number of experimental and commercial single- and multigrade oils was recently reported by Kotani et al. (91), by monitoring the temperature rise in a main bearing of a 1.8-L gasoline engine. It was found that the seizure tendency with single-grade oils could be predicted from the viscosity of the test oil. For multigrade oils, seizure tendency was dependent on the type of VI improver used and, to a lesser extent, on the detergent/inhibitor package and friction modifiers in the oil.

Antiseizure properties of SAE 5W-30 oils were found to fall between the SAE 20 and SAE 30 results. Permanent viscosity losses of the multigrade oils tested were also found to increase the seizure tendency, presumably because the used oil viscosities were too low.

In summary, several of the authors referenced in this section have proposed minimum critical viscosity limits, below which lubricant engine bearing protection is expected to rapidly diminish. These critical limits are reviewed in Table I, together with estimates that can be derived from test data reported by Rhodes and Henderson (111) and Flathmann, Barker, and Wylie (113). The critical viscosity estimates range from less than 2.3 cP (2.9 cSt) to 3.6 cP (4.5 cSt) at shear rates of $6 \times 10^5\ s^{-1}$ to $1 \times 10^6\ s^{-1}$. It is possible that different engine and vehicle test severities are responsible for the differentiation in some cases, but it is also possible that chemical effects and rheological properties other than high-shear rate viscosity also are responsible for the lack of agreement. For example, in the ALI Bearing Distress Test one series of lubricants gave results which indicated that a 3.6 cP minimum viscosity was required at 150°C and $1 \times 10^6\ s^{-1}$. A different series of multigrade oils, with a different detergent/inhibitor package, gave results which indicated that the minimum was less than 2.3 cP, a value which approachs the viscosity of the base oil/detergent/inhibitor combination from which the multigrade oils were derived (1.8 cP at 150°C) (111).

**TABLE I**

Critical High-Shear-Rate Viscosities

| Minimum High-Shear-Rate Viscosity, cP | Shear Rate | Temperature, °C | Test, Engine Type | Ref. |
|---|---|---|---|---|
| 2.6[a] (3.2 cSt) | $1 \times 10^6$ | 160 | Police Fleet, na | 108,115 |
| 2.8[a] (3.5 cSt) | $5 \times 10^5$ | 149 | High-Speed Road, na | 107 |
| 3.6[a] (4.5 cSt) | $6 \times 10^5$ | 149 | Track, GM 3.8 L V-6 | 109 |
| 3.6[b] (4.5 cSt) | $1 \times 10^6$ | 150 | ALI Bearing Dist., CLR | 111 |
| <2.3[b] (<2.9 cSt) | $1 \times 10^6$ | 150 | ALI Bearing Dist., CLR | 111 |
| <2.6 (<3.3 cSt) | $1 \times 10^6$ | 150 | High-Speed Driving, VW 2.5 L, Ford 2.3 L, Buick V-6 3.8 L | 113 |

[a] Estimate of cP from cSt made using density term of 0.8 $g/cm^3$.
[b] Different performance packages used with each oil series.

Piston Ring and Cylinder

It is reasonable to expect that both rheological and chemical effects should also influence piston ring and cylinder wear. As recently determined by Uras and Patterson (114), piston ring frictional force measurements are consistent with boundary or mixed lubrication at top-dead-center and at bottom-dead-center of the piston stroke, while hydrodynamic lubrication takes precedence as the mid-stroke position is approached. Thus, as with journal bearing wear, piston ring face and cylinder wear might be expected to be influenced by both the rheology and the chemical composition of the engine oil.

In 1977, Stewart and Selby (115) made a comprehensive review of the effect of oil viscosity on engine performance. They cited at least six studies that found a relationship between kinematic viscosity and wear in this region. One of these studies was carried out by Neudörfl (103), who investigated the performance of SAE 10W-50 multigrade oils as well as single-grade lubricants. Differences in Daimler Benz OM 616 cylinder wear were found for the multigrades which could not be explained by kinematic viscosity, and it was concluded that polymer concentration, molecular weight, and solubility may have affected wear in this test. Stewart and Selby also discussed the work of Ambrose and Taylor (116), who discovered that an SAE 5W-20 oil could be made equivalent in top-ring antiwear performance to an SAE 20W-20 oil by the inclusion of an antiwear agent in the blend. Before the addition of the antiwear agent, the SAE 5W-20 oil had given 16 times the wear of the SAE 20W-20 oil.

McMillan and coauthors (109) used a single performance package for the preparation of four SAE 10W-40 multigrades that were used in the high-speed track test discussed in the previous section. It was found in this test program that high-shear rate viscosity was related not only to bearing wear but also to average ring wear. Two SAE 10W-40 oils that had high-shear rate viscosities of at least 4.69 cSt at 149°C and $6 \times 10^5\ s^{-1}$ gave satisfactory protection, but oils having high-shear rate viscosities of ca. 4.3 cSt allowed excessive wear. Unusual cylinder bore wear patterns were also found in one engine which had been operated on one of the oils having relatively low viscosity at high shear rates.

Bell and Voisey (106) also used a single performance package for the preparation of a variety of single- and multigrade test oils that were evaluated in a Ford Cortina 1200 $cm^3$ engine. The engine was equipped with radioactive, chrome-plated, top-compression rings so that ring-face wear could be monitored. The authors found that the viscosity of the oils at 100°C and a shear rate of $2 \times 10^5\ s^{-1}$ correlated well with ring-face wear.

In a recent review of European viscosity/engine performance programs, there was a small section devoted to the disclosure of piston ring-wear test data on reference oils RL-59 through RL-67 of the Coordinating European Council (CEC). Seven of these oils were SAE 15W-50 blends, containing the same additive package but different VI improvers in each oil. Radioactive wear tests in a fired engine were used to obtain ring-face, ring-side and ring-back, and cam follower pad wear. No relationship was found between high-shear rate viscosity and any wear parameter, although there were significant differences found at the 95% confidence level between ring face and cam follower pad wear for several of the multigrade oils. The authors conclude, "The lack of wear to viscosity correlation is consequent on the high apparent viscosity of the SAE 15W-50 oils which have been used in these experiments." However, it should be pointed out that these reference oils have high-shear rate viscosities in the range of about 10 to 13 cP at 100°C and 2 x $10^5$ $s^{-1}$ (117), which was within the high-shear rate viscosity range that provided Bell and Voisey with the best correlation of viscosity with ring-face wear. These reference oils would also be expected to provide a correlation between ring face wear and high-shear rate viscosity, unless changes in the chemical composition of the oils, due to differences in polymer composition and base stocks, are affecting wear inhibition.

It has been observed that base oil composition can affect piston-cylinder assembly scuffing. Cassiani-Ingoni and Miorali (118) found that scuffing in a two-stroke engine depended on the viscosity of non-additive-containing lubricants as well as their composition. Mineral oils, poly-glycols, esters, and other fluids were studied but no multigrade oils were included in the testing.

Several other workers since Neudörfl have investigated the effect of viscosity on OM 616 cylinder wear. Pike, Banks, and Kulik (37) reported test results on five SAE 10W-50 oils which only differed in polymer type, and five SAE 10W-40 oils which contained the same polymer but had different antiwear compositions. There was found to be some correlation between high-shear rate viscosity and cylinder wear for the former oil series, but the oils with different antiwear compositions gave large differences in cylinder bore wear. Klüever (119) designed a short-term OM 616 engine test, since he was interested in the effect of new oil, or "quasi-fresh" oil properties on cylinder wear. He observed that the chemical composition of the oil, rather than the oil physical properties, was important in controlling cylinder wear.

A recent paper by Nautigal and coworkers (120), which includes a review of publications that are concerned with the mechanisms of piston ring lubrication and wear, provides evidence that a correlation may exist between lubricant heat of adsorption and wear caused by top piston-ring/cylinder wall contact. In this report, a theoretical model was first proposed for piston ring wear in terms of the energy of adsorption, and the material properties and operating parameters of the system of interest. The theoretical model was then tested via friction studies in an experimental test rig and by the determination of piston ring wear in a single cylinder diesel engine. Good agreement was obtained between the theoretical and the experimental results using one fluid system and it was concluded that a correlation exists between heats of adsorption and wear. Since heats of adsorption depend on the surface active agents in the lubricant, this work provides a link between wear theory and actual engine test experiments which demonstrate that the additive performance package can affect cylinder wear.

For oils that have seen extensive service, there are contaminants that may become primary sources of corrosive or abrasive wear. For example, McGeehan (121) found that there was a correlation between ring weight loss and cylinder bore polishing in diesel engines and attributed the wear to the buildup of "hard carbon" particles which caused abrasive wear. For the new or quasi-new oil properties that we are interested in, however, it would appear that chemical effects and, perhaps to a lesser extent, rheological effects have the greatest influence on wear in this area of the engine.

## Valve Train

In an analysis of camshaft-lifter lubrication, Baldwin (122) stressed that wear prevention appeared to be due to localized elastohydrodynamic lubrication and surface film formation. A laboratory Falex test was developed to simulate engine cam-lobe/lifter contact. Test results from experiments using this apparatus indicated that viscosity effects could be significant if lubricant viscosity was below a critical minimum value.

A test rig was designed by Coy and Dyson (123) to reproduce the kinematics and geometry of a typical finger-follower assembly. Lubrication of the contact area was believed to be mixed, with contributions from both boundary and elastohydrodynamic lubrication. However, based on their experimental findings, these workers concluded that hydrodynamic lubrication was also a factor. Wear was found to be affected by both lubricant viscosity and zinc dithiophosphate. Neither Coy and Dyson nor Baldwin reported on the evaluation of non-Newtonian fluids in their equipment.

K. Ninomira and coworkers (124) developed a procedure to measure the conductivity of lubricant films that were formed between the cam and lifter of an OHV engine. Comparable conductivity was found for an SAE 30 and SAE 10W-30 oil, but there was a higher conductive duration observed for an SAE 5W-30, which suggests that the SAE 5W-30 oil would provide less wear inhibition. At higher valve spring loads, the SAE 10W-30 oil was found to exhibit higher conductivity duration than the single-grade oil, and this conductivity increase was manifest in pitting of the lifter surfaces.

Neudörfl (103) attempted to neutralize the variance of chemical effects on wear by evaluating a series of multigrade oils that contained the same detergent/inhibitor package. In a Daimler Benz OM 616 Kombi test, an SAE 10W-50 oil gave less cam wear than a SAE 10W-30 oil that contained the same VI improver. Subsequent studies indicated that oils having zinc dithiophosphate concentrations from 0 to 1.5% varied little in cam wear protection, but polymer concentration produced noticeable differences with low wear favored for oils having high polymer content.

In high-speed track test studies, McMillan et al. (109) found no differences in cam and lifter wear for oils that had significantly different high-shear rate viscosities, but valve guide wear was increased with oils which contained shear unstable polymers (or which had relatively low viscosities at high shear rates).

In their 1977 review, Stewart and Selby (115) concluded that viscosity could influence wear in the valve train after reviewing the work of Ambrose and Taylor (116), Slater (125), and Roux (126). Each of these authors had also indicated, however, that the additive performance package was a primary factor. Slater showed that cam wear varied linearly with the zinc content of oils that were tested in a 2-hour, 2-liter Volvo engine test while Roux stated in reference to valve train wear that "the benefit of viscosity should be mentioned, although it is often hidden by the effect of the detergent/dispersant additives."

A recent review (127) reported on attempts of certain European laboratories to detect differences in wear protection afforded by CEC reference oils that contained the same performance package but were formulated with different VI improvers. It was found that oil viscosity at high rates of shear could not be used to predict the performance of the multigrade oils, which were all SAE 15W-50 grades.

Several very comprehensive studies over the last ten years have dealt with the effects of lubricant additives and engine operational parameters, including oil service life and deposit buildup, on valve train wear. Pless and Rodgers (128) found that zinc dithiophosphate content could greatly affect valve train scuffing and cam/lifter wear; wear was also influenced by the dispersant level. Rounds (129) found that diesel soot contamination was a primary cause of valve train wear. Torii and coworkers (130) studied the effect of additive chemistry, but not viscosity, on valve train wear.

In a 1981 paper, Roberts and Tourier (131) reviewed the influence of engine oil formulation practice on European engine valve train wear. They concluded that wear could be reduced by increasing oil viscosity, but that viscosity effects were small compared to lubricant additive effects. Of particular interest are results they reported on a Peugeot 504 follower pitting test. By optimization of additive components it was possible to obtain excellent pitting resistance for SAE 15W-50, 10W-40, and 5W-30 multigrades.

Sugiura (132) reported on four test procedures that are used to assess valve train wear in Japan. Test results indicated that the type and quantity of zinc additive, together with the choice of detergent and dispersant, can affect wear. It was observed that even an SAE 5W-20 oil could give good wear protection if it was properly formulated.

# The Effect of Oil Rheology on Engine Friction and Fuel Economy

## THE EFFECT OF OIL RHEOLOGY ON ENGINE FRICTION AND FUEL ECONOMY

Oil-related frictional losses in a gasoline powered automotive engine are of two general types: rubbing and viscous-resistance losses (88,123). Both types of frictional loss directly affect engine fuel economy. Rubbing losses occur during starting of the engine before the oil reaches each of the engine's critically lubricated areas (134,135). These losses also occur while the engine is running, in areas of boundary and/or squeeze film lubrication where a well-defined oil film is not always present and metal-to-metal or asperity contact is probable. Top and bottom dead center (TDC, BDC) (98,134-142) of the piston stroke, the valve train area (143), and bearing/shaft interaction directly after engine firing (due to extreme squeezing down of the oil film) have been identified as the most important sources of rubbing losses in an automotive engine. Viscous-resistance or hydrodynamic losses occur where there is a definite oil film between moving engine parts. The oil's resistance to the movement of one metal surface relative to another, or viscous drag, is directly proportional to oil viscosity; the higher the viscosity, the larger the viscous drag. Viscous-resistance losses are primarily the result of the oil's viscosity working against the shaft as it rotates in the bearing area (98,140,142), although some hydrodynamic friction is also experienced at mid-piston stroke (maximum piston speed) (134,135,138).

From the foregoing analysis it is evident that both low- and high-shear rate oil viscosities are important to overall engine friction. The time it takes for an oil to get to the critically lubricated parts of an engine (oil flow) during ambient start has been identified as a low-shear rate phenomenon (144,145), and as has been stated previously, a delay in lubrication to these critical engine parts can lead to substantial rubbing losses. On the other hand, many studies have shown that the engine lubricant experiences a maximum shear rate of $\sim 10^6\ s^{-1}$ as the shaft rotates in the bearings and it is the high-shear rate viscosity which affects the level of viscous-resistance losses in the bearings (36,146,147). In addition, the temperature of the oil can also affect the degree of oil flow to critical parts during ambient start and also the relative high-shear rate viscosity of the oil in the bearings; the higher the oil temperature, the thinner the oil film or viscosity and the smaller the viscous resistance.

Treating the engine as a whole, certain engine parts always operate under rubbing conditions while others operate in a hydrodynamic regime, depending on the engine, driving cycle and environmental conditions. A graphical representation of the complex effect of oil rheology on total engine friction was previously shown in Figure 9. A replot of this "Stribeck Curve" in a slightly different fashion, Figure 18, depicts the relationship between overall engine friction and oil composition, including oil viscosity (film thickness) and additive makeup.

Lowering the viscosity (film thickness) of an oil (from right to left) leads to reduced viscous drag friction. Friction is continuously reduced until a critical low viscosity, point F, is reached. At this point, metal-to-metal (asperity) contacts through the oil film become increasingly important to overall friction, and engine friction increases exponentially. The engine friction produced with commercial SAE 10W multigraded oils of the early to mid 1970s can best be described by point A. Overall engine friction had a strong rubbing component with these oils, and friction reducing additive technology developed during this period helped reduce this metallic contact friction (point B) without having to reduce the viscosity of these oils. During this time period, the major effort in automotive engine friction reduction was directed towards reducing rubbing losses via new oil chemistry.

Engine friction associated with higher viscosity multigrades (e.g. SAE 15W and SAE 20W multigrades) and SAE 30 and higher single grades marketed during the early to mid 1970's is best described by points to the right of point F on the Streibeck curve, Figure 18. With these higher viscosity oils, reducing the oils viscosity until point F is reached will also reduce engine friction. However, the critical level of oil viscosity can be extended to the left beyond point F, to even lower viscosities. This can be accomplished by incorporating even more effective friction-reducing additives into the oil. Mechanism studies (148,149) have shown that these additives operate by forming a strong bond with the metal surface, leading, longer term, to a smoothing of surfaces (a reduction in the number and height of asperity peaks). Engine parts can therefore come closer to one another (a thinner or less viscous oil can be used) without increasing rubbing friction. In other words, simultaneous advantage can be taken of reduced viscous drag and reduced metallic contact friction to achieve maximum reduction of oil related frictional losses (maximum increase in fuel economy) in the automotive engine.

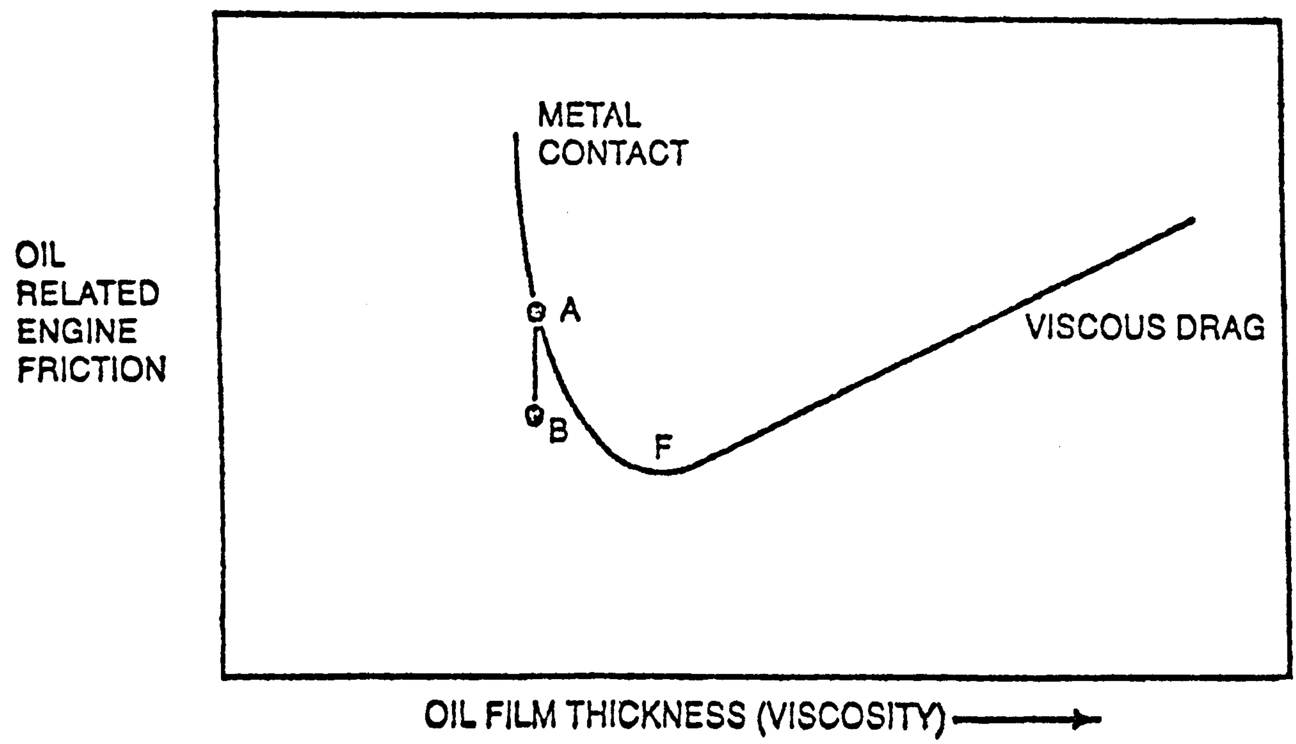


Figure 18. Friction reduction via lower viscosity plus friction lowering additives.

## Friction

Interpretation of tests designed to relate engine oil viscosity to engine friction has provoked a measure of both speculation and controversy. In 1961, Okrent (98), in measuring the torque required to motor a preconditioned engine, observed that some polymer-containing oils gave engine friction values below those of their base stocks. In a carefully designed experiment, he measured engine friction for polymer-containing oils whose base stock viscosities lay below the minimum in the friction/viscosity curve, arguing that, if significant temporary viscosity loss occurred, these oils should lead to increased, rather than decreased friction. In all cases, the polymer-containing oils gave engine friction values below those of their bases stocks. Okrent concluded that temporary viscosity loss could not be invoked to explain his findings, and that some other factor such as viscoelasticity or surface effects must be influencing the test results. In 1976 Selby (150) suggested that Okrent's friction measurements could have been dominated by the cylinder wall-piston ring friction effect rather than by the bearings, and that high-shear viscosities of the oils at 200-260°C would have been more appropriate than the low-shear, 100°C viscosities employed by Okrent.

In 1969 Haviland (151) performed low-temperature cranking tests at -29°C (-20°F), employing oils ranging in viscosity from 4 to 184 poise. He was able to show that for a constant engine speed, instantaneous engine friction torque increased with increasing engine oil viscosity. Conversely, average engine friction horsepower decreased as engine oil viscosity was increased because average cranking speed decreased more than friction torque increased.

In 1975 Rosenberg (75) used a simulated bearing rig to measure friction values for four Newtonian oils ranging in low-shear viscosity from 6.3 to 13.5 cSt at 99°C. The linear plot of friction versus viscosity (at the bearing back temperature) which resulted was used as a calibration from which apparent viscosities of polymer-containing oils could be derived. Friction values for sixteen oils (blended from the same base stock and thirteen different VI improvers to a common low-shear viscosity of 13.5 cSt at 99°C) were always higher than that of the base stock, but never exceeded that of a 13.5 cSt base oil. A linear regression of the apparent viscosities derived from the friction measurements and from oil film thickness determined in the rig yielded a slope of 1.11 and a correlation coefficient of 0.72. This degree of correlation was considered reasonable in view of the experimental errors associated with the film thickness measurements.

In 1978 Badiali et al. (152), measured friction power loss by a modified Morse procedure in a fired engine. Friction power loss values for single-grade reference oils were related linearly to the log of low-shear viscosity (at sump temperature), and from this calibration, the effective viscosity of multigrade oils could be derived. A series of four SAE 20W-50 oils having the same kinematic viscosity at 98.9°C and blended from a single mineral base oil and four different VI improvers yielded effective viscosities very close to that of the base lubricant (fully-formulated oil less VI improver), and in some cases even lower (Table II).

**TABLE II**

Effective Viscosities of Multigrade Oils
(from Reference 152)

| Oil | Composition | Power Loss, kW | Viscosity at 50°C, $mm^2s$ (Sump Temperature) Kinematic | Effective |
|---|---|---|---|---|
| BO | Base Oil | 5.00 | 49.3 | 49.3 |
| M1 | BO + PMA-1 | 5.15 | 89.7 | 56.5 |
| M2 | BO + PMA-2 | 4.92 | 86.7 | 45.5 |
| M3 | BO + PSE[a] | 5.07 | 90.0 | 52.5 |
| M4 | BO + OCP | 4.85 | 101.0 | 42.0 |

[a] Polystyrene Ester

In further tests it was noted that bearing temperatures reached with these multigrade oils were higher than those with the corresponding base oil, even when the friction power loss with the multigrade oil was lower than with the base oil. This was taken as evidence that the oil viscosity in the bearing is anisotropic, comprising a lesser apparent circumferential component which determines friction, and a greater component in the axial direction which determines oil flow through the bearing and hence bearing temperature. Support for this view is given by the rheological model of Tipei and Rohde (153), but as in the case of Okrent's data, a simpler explanation may lie in a consideration of high-shear viscosities at temperatures much higher than those measured in the bearings.

In 1978 McGeehan (154) presented a comprehensive literature review of piston and ring friction and its relationship to oil viscosity. He concluded that (a) piston and ring friction can account for 65% of the mechanical friction in an internal combustion engine, and (b) cylinder liner lubrication is predominantly hydrodynamic, with localized contact between ring and liner at TDC firing. Piston ring friction in the hydrodynamic region is proportional to the square root of the viscosity, and the viscosity is affected by temperature and pressure which can reach peak values of 176°C and 4000 psi, respectively. In a similar analysis, estimates of component contribution to total engine friction by Cleveland and Bishop in 1960 (155) assigned 60% to the piston assembly, 25% to the valve train, and 15% to the crankshaft. The relatively low contribution of the main and connecting rod bearings is indicative of the hydrodynamic lubrication regime in which they operate, and may explain the sparsity of engine bearing friction data in the literature.

In 1980, Dancy et al. (146) described a motored engine test in which the reaction torque of the drive motor could be measured. Engine motoring torque reflects a combination of the frictional effects of both oil viscosity and additive differences. The authors attempted to assess these effects separately by comparing the friction results for a series of single-grade and multigrade oils, all containing the same dispersant/inhibitor package, on an equal viscosity basis. Changing torque versus sump oil temperature plots for each oil to plots of torque versus kinematic viscosity at the corresponding oil temperatures resulted in single-grade and multigrade oil "envelopes" of the same shape and level, but with the multigrade oil envelope displaced to the right of that for the single-grades as shown in Figure 19. Superposition of the two envelopes was achieved by replacing kinematic viscosity with high-shear rate viscosity measured at $10^6$ $s^{-1}$, Figure 20. Oils could thus be placed on an equal viscosity basis by plotting torque versus high-shear rate viscosity. Differences between oils plotted on this basis would be due to chemical differences which affect boundary lubrication friction. By this means, the effects of friction-modifying additives could be clearly distinguished. Conversely, the change in engine friction attributable to temporary viscosity loss of non-Newtonian oils during engine operation could be derived by entering the kinematic and high-shear viscosities at a given temperature into a plot similar to Figure 20, in which the data are reduced to a single average-value curve, and estimating the difference in motoring torque.

In 1980 Cassiani-Ingoni et al. (127), reported power loss measurements in fired and motored engines for a series of seven SAE 15W-50 oils. These oils were formulated with different VI improvers in the same base oil and additive system to the same kinematic viscosity at 100°C; viscosities of the oils at 100°C and $10^6$ $s^{-1}$ ranged from 70 to 90% of the low-shear viscosity. Tests were performed at three independent CEC laboratories (H, L, M) and their data are given in Table III.

**TABLE III**

Power Loss Values in CEC CL23 Reference Oils
(from Reference 127)

| | Fired Engine | Motored Engines | | | |
|---|---|---|---|---|---|
| | Lab H | Lab L | | Lab M | |
| Reference Oil Code | Power Loss, HP 100°C[a] | Indicated Load, kg 100°C[b] | 150°C[b] | Indicated Load, kg[c] 1000 rpm | 4000 rpm |
| RL 59 | 6.20 | 2.87 | 2.74 | 1.06 | 1.85 |
| RL 60 | 6.72 | 2.94 | 2.86 | 1.03 | 1.77 |
| RL 61 | 6.75 | 2.95 | 2.87 | 1.03 | 1.82 |
| RL 62 | 6.90 | 3.04 | 2.96 | 1.08 | 1.85 |
| RL 63 | 6.92 | 2.98 | 2.89 | 1.09 | 1.89 |
| RL 64 | 7.08 | 2.93 | 2.82 | 1.00 | 1.80 |
| RL 65 | 7.10 | 3.02 | 2.95 | 1.12 | 1.91 |

[a] Main bearing oil film temperature. 0.8-L engine.
[b] Sump oil temperature. Coolant outlet controlled at 90°C. 1.3-L engine.
[c] Coolant outlet controlled at 90°C. Oil gallery temperature self-stabilizing. 0.85-L engine.

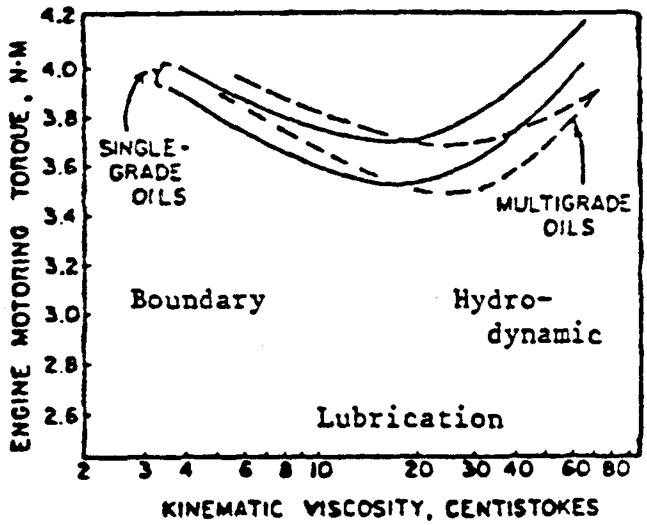


Figure 19. Effect of kinematic viscosity on engine motoring torque (Reference 146). Reprinted with permission ©1980 Society of Automotive Engineers, Inc.

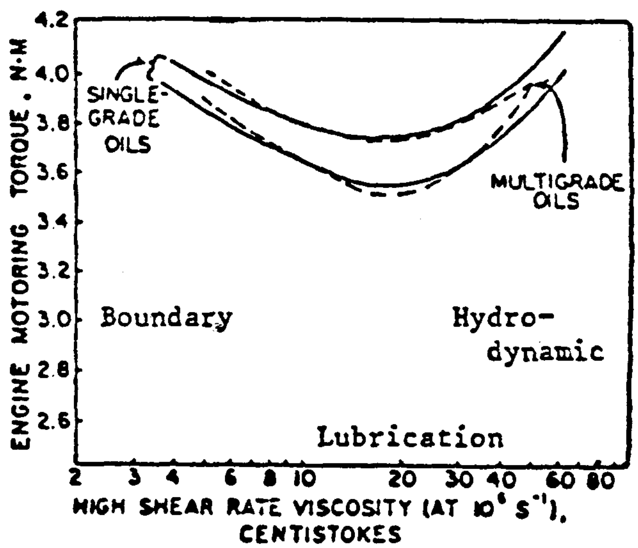


Figure 20. Effect of high-shear rate viscosity on engine motoring torque (Reference 146). Reprinted with permission ©1980 Society of Automotive Engineers, Inc.

Agreement among the laboratories with respect to the ranking of the oils was not good, and it was clear that the engine design and the choice of test conditions were critical in determining the friction response of the lubricants. It should be noted also that differences in ranking of the oils in each laboratory were not always statistically significant when test repeatability factors were applied. Similar conclusions were drawn by the same authors when reporting results for a further series of SAE 10W-40 oils in 1981 (117).

In 1981 Hamaguchi et al. (156), measured the axial torque required to motor a 1.6 L engine lubricated by single and multigrade oils. All oils were formulated with the same additive package, and for the multigrade oils, viscosity was varied by use of different amounts of a polymethacrylate VI improver in the same base oil. Friction torque increased with single-grade oil viscosity over the range 5.2 to 13.1 cSt at 100°C, while for the multigrade oils, as shown in Figure 21, friction torque increased with increasing polymer concentration. However, for 100°C viscosities greater than ~7 cSt, the multigrade oils produced much lower motoring friction torque than did the equivalent single-grade oils. This latter observation was attributed to temporary viscosity loss of the polymer oils under the high-shear conditions of the engine.

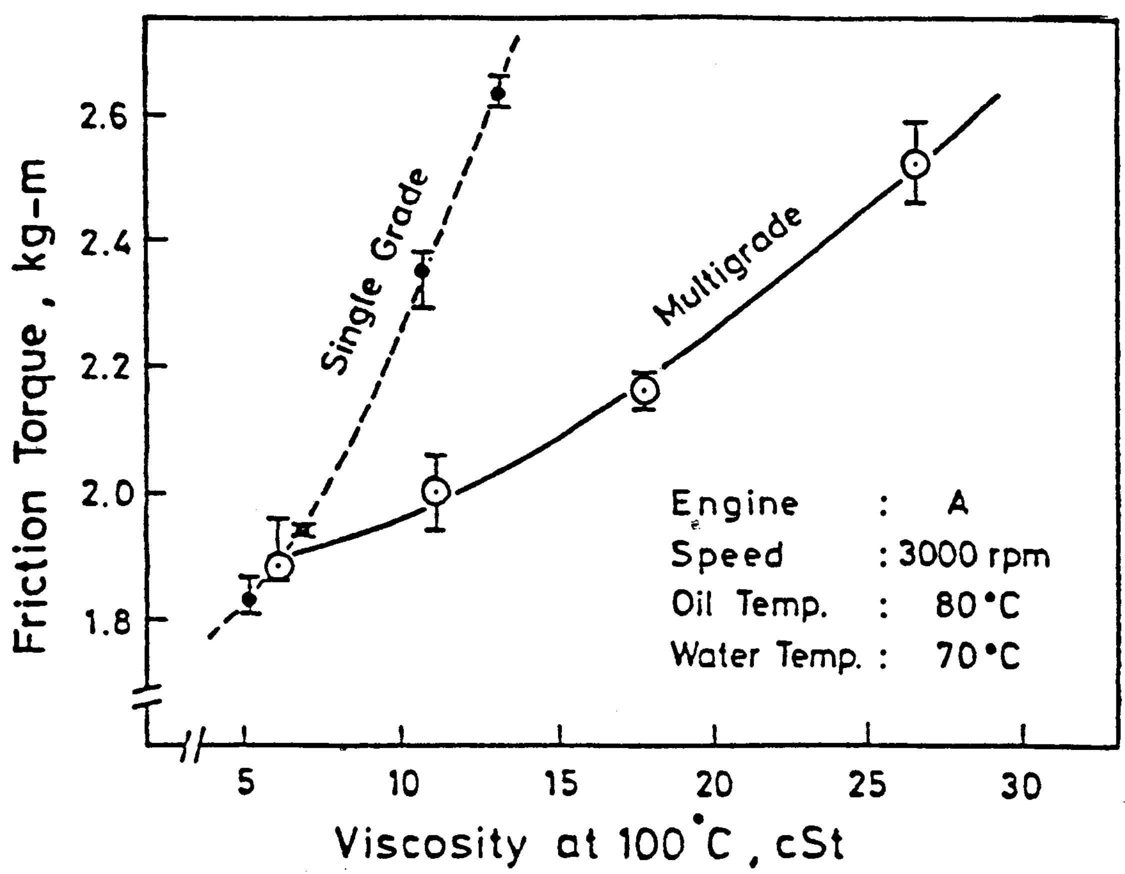


Figure 21. Effect of viscosity index improver on motor friction torque characteristics (Reference 156). Reprinted with permission ©1981 Society of Automotive Engineers, Inc.

Additional tests were made with multigrade oils of different viscosity grades formulated with the same additive package, while varying the base oil combination and polymer concentration to produce the desired viscosities. Again, friction torque values for the multigrade oils were lower than those for the single-grade oils at the same 100°C viscosity; the difference in friction torque between the single and multigrade oils tends to become larger with increasing polymer concentration as shown in Figure 22. By plotting friction torque versus used oil viscosity, it was possible to estimate the relative contributions of permanent and temporary viscosity loss to the reduction in friction torque; in all cases, temporary viscosity loss was the dominant factor. Thus, as in other engine friction studies reviewed in this section, the evidence suggests that engine friction correlates better with oil viscosity measured at the high shear rates typical of piston/cylinder and crankshaft bearing contacts than with conventional, low-shear kinematic viscosity.

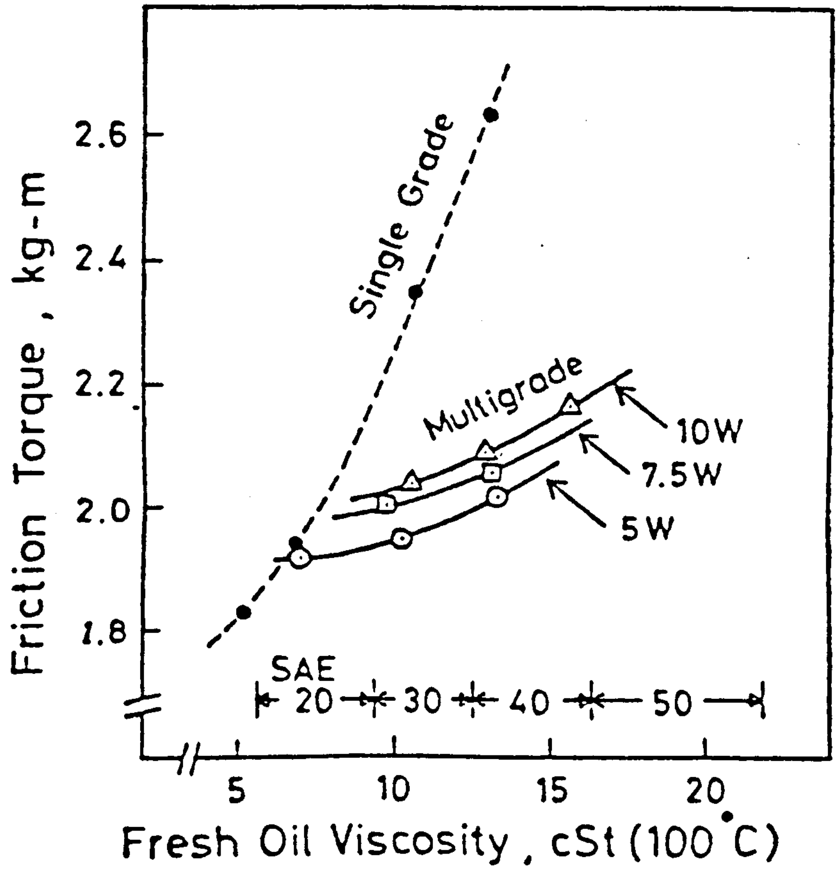


Figure 22. Oil viscosity versus motoring friction torque (Reference 156). Reprinted with permission ©1981 Society of Automotive Engineers, Inc.

In 1982, Rosenberg (157) observed that, while motoring friction tests are normally used to establish the friction contribution of the various engine components, these test procedures do not duplicate the actual combustion forces, nor certain inertia forces that are present during fired engine operation. It may be inferred, therefore, that friction measurements from motored engine tests may be different from those made under fired engine conditions.

In 1983 Staron and Willermet (143) developed a valve train friction model for a 1.6 L OHC engine lubricated with SAE 5W, 10W, 30, and 50 grade oils, from which they concluded

- Boundary friction accounts for most of the frictional loss at the cam and tappet interface. Friction decreases with increased oil viscosity since this leads to increased oil film thickness.
- Viscous friction losses in the cam journal bearings were insensitive to viscosity because of the high loadings. Boundary friction losses become important at low speeds, an effect which is most pronounced with lower viscosity lubricants.
- The rocker arm/fulcrum is a boundary-lubricated contact and, as such, is insensitive to lubricant viscosity.

Experimental results from a motored valve train rig employing (i) a Newtonian SAE 20W-30 oil, (ii) a Newtonian SAE 50 oil, and (iii) a friction-modified version of the SAE 20W-30 oil were in good agreement with the predictions of the valve train friction model. A marked reduction in valve train friction torque was obtained with the friction-modified oil.

## Fuel Economy

Relationship to Kinematic Viscosity at 100°C. Georgi (158) showed in 1954 that the only significant property of an engine oil (at that time) relating to engine friction and fuel economy was kinematic oil viscosity. Moreover, the data presented also indicated that the "engine viscosities" of some motor oils may be somewhat different from their "laboratory viscosities" as measured in conventional Saybolt or Kinematic viscometers.

Chamberlin and Sheahan (159) showed in 1975 that a 2.3% fuel economy benefit could be achieved for SAE 10W-40 oils over an SAE 40 base line oil in steady state (60 mph road load) engine dynamometer fuel consumption tests.

Davison and Haviland (160) reported that by using low viscosity lubricants in the engine, automatic transmission and rear axle, warmed up fuel economy benefits of up to 5% could be realized (depending upon the differences in lubricant viscosity and type of driving regime).

McGeehan, in a literature review paper (154), recounted that fuel economy benefits of 2-3% which were achieved in the early 1950's by reducing an oil's kinematic viscosity one SAE viscosity grade, had been lowered to 0.7-0.9% by the late 1970's because of piston and ring design changes.

Goodwin and Haviland (161) showed that, in general, fuel economy benefits from lowering lubricant viscosity, increased with each of the following: decreasing lubricant temperature, decreasing engine and axle loads, and in road tests, decreasing car size. The authors showed that by reducing both the engine oil and rear axle lubricant kinematic viscosities 60% (engine oil from SAE 10W-40 to 5W-20 and rear axle lube from SAE 90 to SAE 75W), average fuel economy benefits of up to 4.8% could be achieved under warmed-up operation. This is summarized in Table IV.

**TABLE IV**

Fuel Economy Benefits With the Use of Low Viscosity Lubricants for Different Driving Cycles.
(from Reference 161)

| Test | No. Cars | Average Percent Fuel Economy Improvement |
|---|---|---|
| EPA 55/45 | 3 | 1.6 |
| City | 3 | 2.0 |
| Highway (Cyclic) | 3 | 1.0 |
| Constant-Speed, 88 km/h | 2 | 4.8 |
| GM City-Suburban, (Warmed-up) | 2 | 3.8 |
| GM City-Suburban, (Cold-Start, 0°C) | 2 | 7.9 |

The 60% reduction in engine oil viscosity accounted for 60-85% of the benefits (0.65 to 4%) in warmed-up fuel economy tests, the exact quantity depending on the vehicles used for comparison and the driving mode (steady state or cyclic). The cyclic, warmed-up operation (EPA highway cycle, GM city-suburban cycle) gave the smallest viscometric benefits.

As indicated in Figure 23, Roberts, Gaskill and Richard (162) showed that the same range of fuel consumption reductions could be achieved in the cyclic, warmed-up European ECE 15 driving mode when reductions in lubricant kinematic viscosity (at 100°C) similar to those described by Goodwin and Haviland were made.

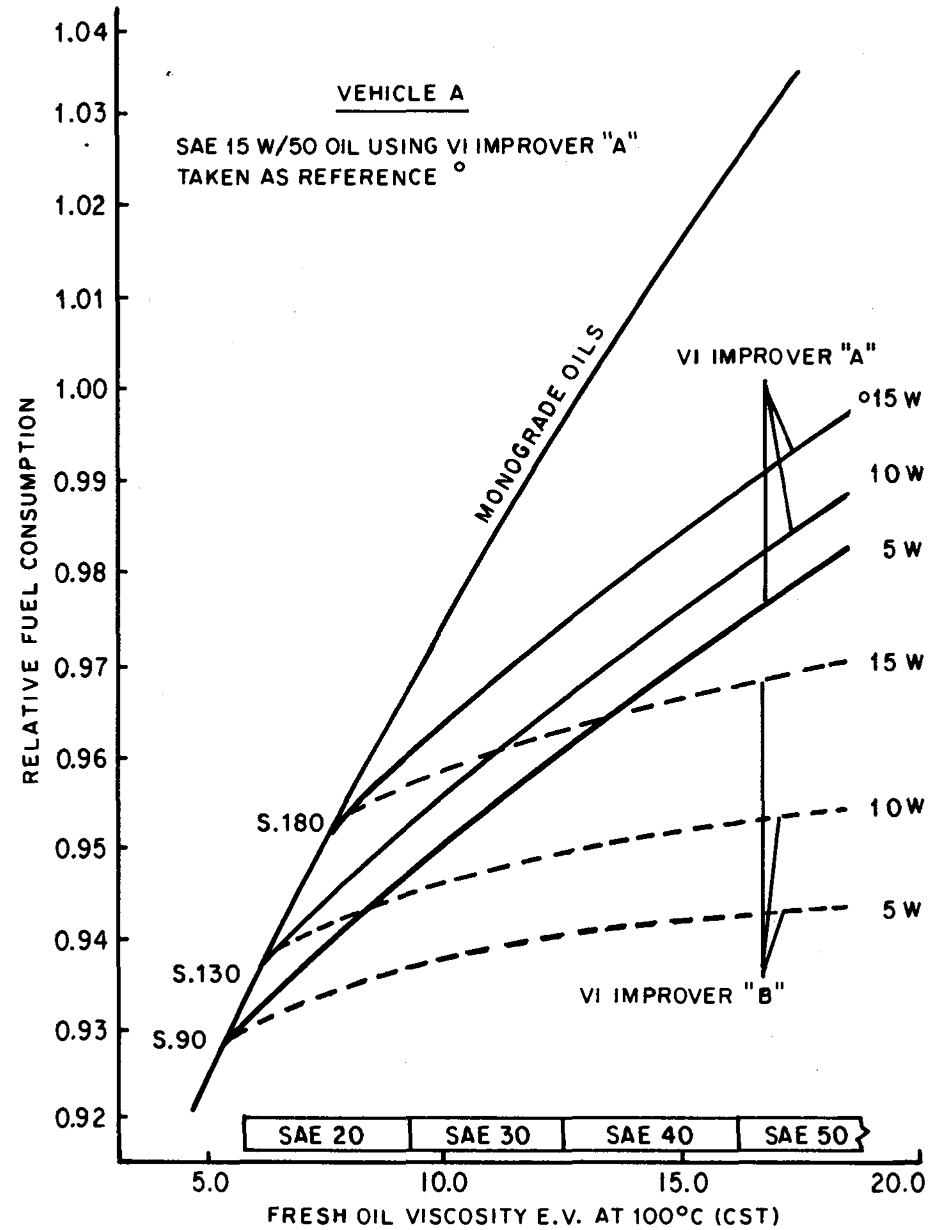


Figure 23. Effects of different VI improvers and viscosity grades on fuel consumption - ECE 15 (Reference 162).Reprinted by permission of the Council of the Institute of Mechanical Engineers.

The effect of kinematic oil viscosity on fuel economy under part (43.3°C oil temperature) and fully warmed-up (93.3°C oil temperature) engine conditions was also studied by Passut and Kollman (163) with five single-grade engine oils ranging from an SAE 5W (kinematic viscosity at 98.9°C = 5.4 cSt) to an SAE 40 (kinematic viscosity at 98.9°C = 16.0 cSt) grade oil. Fuel consumption was determined with each of these oils in a six cylinder (L-6) Ford engine at an engine speed of 1800 rpm (50 mph/80 km/h) and at 18 and 23 bhp (13.4 and 17.2 kW) "level road load" conditions. Oil viscosity was determined from an ASTM viscosity-temperature chart at the temperature measured at the engine oil filter during the test. Figure 24 summarizes the dependence of fuel economy on oil viscosity over the large viscosity range studied.

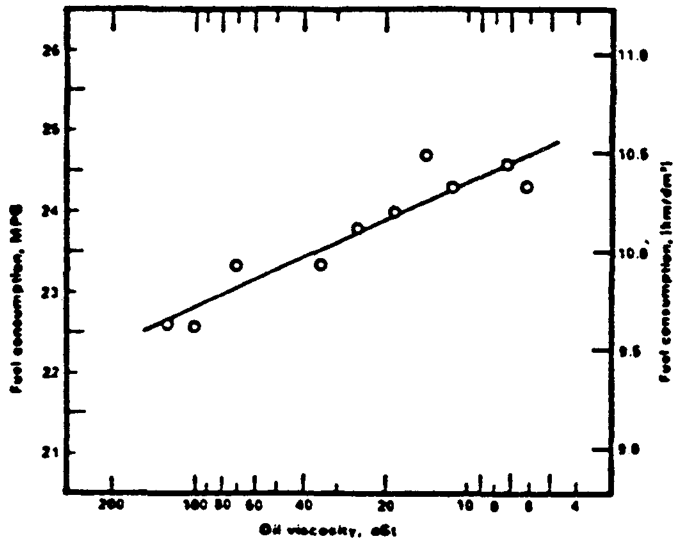

Figure 24. Effect of oil viscosity on fuel consumption, Ford L-6 (Reference 163). Reprinted with permission ©1978 Society of Automotive Engineers, Inc.

Broman, et al. (164), measured the fuel consumption of a series of fully formulated (SAE 10W, 20W-20, 30, 40, and 10W-40) engine oils on a dynamometer test stand using a fully broken-in 2.3 liter engine. The kinematic viscosities of the test oils ranged from 6.32 to 15.31 cSt at 98.9°C. Fuel consumption was measured at constant speed/load conditions corresponding to 30, 40, 50, and 60 mph road load. The results are summarized in Tables V and VI and represent an average, in mpg, over all four speed/load conditions for each of the test oils. With the SAE 40 oil used as the baseline, the most commonly used commercial oils, SAE 20W-20, 30, and 10W-40, showed benefits ranging from 1.1 to 2.8% under these steady state, warmed-up conditions.

**TABLE V**

Viscometric Properties of Test Oils as a Function of SAE Grade
(from Reference 164)

| SAE Grade | 40 | 30 | 10W-40 | 10W-20 | 10W |
|---|---|---|---|---|---|
| KV/100°F | 147.6 | 122.3 | 91.7 | 74.8 | 41.5 |
| KV/210°F | 15.31 | 12.35 | 15.44 | 8.85 | 6.32 |
| Viscosity Index | 98 | 100 | 190 | 100 | 110 |

**TABLE VI**

Effect of Viscosity on Fuel Economy
(from Reference 164)

Engine: 2.3 Liter
Dynamometer Test Conditions: Steady State (30, 40, 50, 60 mph, Road Load)

| SAE Grade Engine Oil | Averaged Fuel Economy, mpg[a] | Improvement Over Baseline mpg | Improvement Over Baseline Percent |
|---|---|---|---|
| 40 | 16.93 | Baseline | Baseline |
| 30 | 17.11 | 0.18 | 1.1 |
| 10W-40 | 17.27 | 0.34 | 2.0 |
| 20W-20 | 17.41 | 0.48 | 2.8 |
| 10W | 17.45 | 0.52 | 3.1 |

[a] Average of 30, 40, 50, and 60 mph steady state results

However, Ghirla and Smith (165) showed that reducing the viscosity of multigrade engine oils by as much as 50% (SAE 10W-40 to SAE 5W-20) did not produce consistent or statistically significant fuel economy benefits on cyclic EPA (FTP/HFET) tests run under the 1975 Federal Test Procedure. Constant speed tests showed larger fuel economy differences in their studies.

Fuel consumption data were obtained on a homologous series of single-graded oils in a 140 CID (2.3 liter) L-4 engine and in a 305 CID (5.0 liter) V-8 engine in a dynamometer study by Hart and Klaus (166). Fuel flow rates were taken at six different engine speed/road load conditions (1500 to 2500 rpm) and at two different water jacket outlet temperatures (160°F/71°C and 190°F/88°C). This allowed them to evaluate the effect of changing the oil's viscosity on fuel consumption without changing the oil. The reference oils were SAE viscosity grades of 5W, 10W, 20W-20, 30, and 40. The fuel consumption data from this series of tests are summarized in Table VII along with the 149°C kinematic viscosities of

**TABLE VII**

Viscosity Calibrations - 140 CID (2.3 Liter) & 305 CID (5.0 Liter) Engines
(from Reference 166)

| | | 140 CID (2.3 Liter) | | 305 CID (5.0 Liter) | |
|---|---|---|---|---|---|
| SAE Grade | Vis @ 300°F (149°C) cs | Average VSI, % | Average Fuel Cons., #/Hr. | Average VSI, % | Average Fuel Cons., #/Hr. |
| 40 | 5.76 | -0.91 | 9.800 | -1.57 | 19.881 |
| 30 | 4.83 | -0.59 | 9.490 | -1.34 | 19.768 |
| 20W-20 | 3.92 | 0.27 | 9.419 | -0.99 | 19.631 |
| 10W | 3.04 | 0.36 | 9.321 | -0.48 | 19.490 |
| 5W | 2.51 | 0.94 | 9.260 | -0.14 | 19.503 |

each oil. A plot of these data against 149°C kinematic viscosity is shown in Figure 25. The warmed-up fuel economy benefits in going from an SAE 30 to an SAE 5W oil ranged from 1.3 to 2.4% while going from a SAE 40 to a SAE 30 grade ranged from 0.6 to 3.1% depending on the engine.

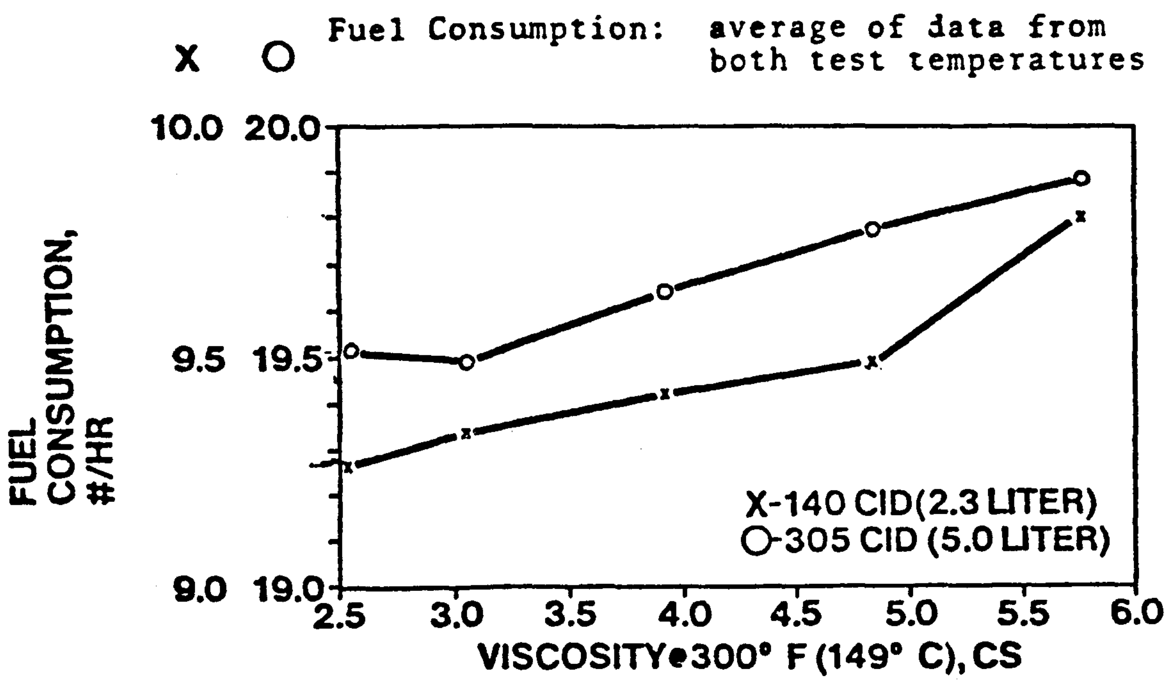


Figure 25. Viscosity calibration fuel consumption (Reference 166). Reprinted with permission ©1979 Society of Automotive Engineers, Inc.

Overall, the studies relating changes in oil kinematic viscosity to fuel economy under warmed-up engine conditions showed that fuel economy effects were greatest under constant (steady state), medium speed (40-60 mph) operation. Under cyclic, warmed-up operation, where boundary (rubbing) friction effects become increasingly more important, fuel economy benefits resulting from a lower engine oil kinematic viscosity, are significantly reduced. In fact, Goodwin and Haviland (161) have shown that there can actually be a reversal in fuel economy effects with lower viscosity engine oils in hot running engines under cyclic operation (HFET EPA highway cycle). Under these conditions, a lower viscosity oil can give poorer mpg than a higher viscosity oil.

Relationship to High-Temperature, High-Shear Viscosity. In 1981, Galluccio and Koller (147) showed that the relative fuel efficiency of multigraded engine oils correlated well with their high-temperature (100°C) high-shear rate ($10^6$ $s^{-1}$) viscosities. Examples of this are summarized in Figures 26 and 27 and in Tables VIII and IX. The authors also demonstrated that as a consequence of the fuel efficiency/high-temperature, high-shear rate viscosity relationship, relative fuel efficiency is a function of the SAE J300 SEP80 W grade for engine oils formulated with a common viscosity index improver.

In May of 1982, the ASTM Fuel Efficient Engine Oils Task Force issued a report (167) summarizing the results of the ASTM Pilot and Demonstration programs to define an industry test procedure for measuring the fuel efficiency of passenger car engine oils. Included in the Task Force report was a plot of percent change in fuel economy (five car average) versus used oil high-temperature (150°C), high-shear rate ($10^6$ $s^{-1}$) viscosity for several of the oils evaluated in the demonstration test program. This plot is included as Figure 28 in this report. The percent change in fuel economy appears to correlate ($R = 0.92$) quite well with high-temperature, high-shear rate viscosity of the used engine oils. However, while oil 508 was run in four different five-car fuel economy programs, the results of only one is plotted in Figure 28 (because used engine oil was only available from this group of five cars at the time of correlation development). Since the other three five-car programs with oil 508 gave fuel economy benefits over the industry reference oil (HR) of ~1.7 to 3%, the correlation suggested by Figure 28 may be somewhat premature. Oil 508's used oil viscosity at 150°C and $10^6$ $s^{-1}$ for the five-car group which showed a 3% benefit over HR would have to be about 3.0 cSt for the correlation in Figure 28 to hold. This is distinctly lower than the ~4.5 cSt viscosity measured for the used 508 oil which gave ~1% benefit.

This suggests another potential problem with attempting to correlate and predict fuel economy of passenger car engine oils from used rather than fresh oil viscometric data. The used oil viscosities from any given oil may vary from car to car (engine to engine) even within the same engine type, but the data the industry has gathered thus far do not assure that fuel economy will vary correspondingly.

Dobson and Pike (168-172) on the other hand, have shown that fresh oil high-temperature, high-shear rate viscosity correlates better with fuel efficiency than does kinematic viscosity in some "gasoline-engined" cars run under certain driving modes. Figures 29a and b show both relationships for a Ford Escort run in an 17.7 km (11 mile) urban cycle on a dynamometer test stand. The

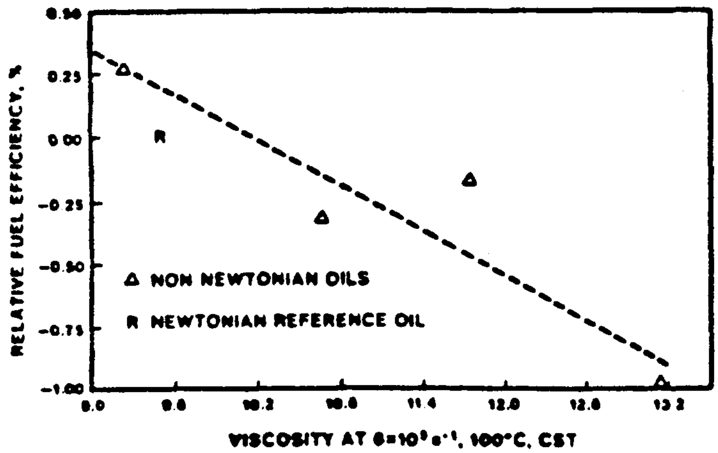


Figure 26. Relative fuel efficiency as a function of high-shear rate viscosity in the 2.5 L engine for oils formulated with VI improvers having different molecular weights (Reference 147). Reprinted with permission ©1981 Society of Automotive Engineers, Inc.

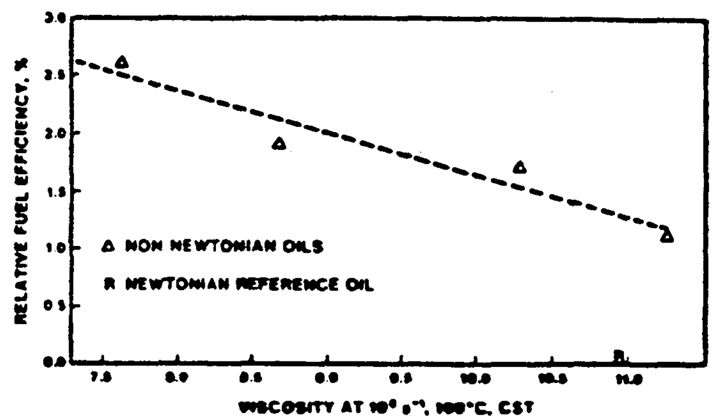


Figure 27. Relative fuel efficiency as a function of high-shear rate viscosity in the 5.7 L engine for oils formulated with a constant VI improver (Reference 147). Reprinted with permission ©1981 Society of Automotive Engineers, Inc.

**TABLE VIII**

Linear Correlation Analysis - Molecular Weight Study
Fuel Efficiency Versus High-Shear-Rate Viscosity

2.3 L Engine (from Reference 147)

| Temperature °C | Shear Rate $s^{-1}$ | $R^2$ [a] Non-Newtonian Oil Only | $R^2$ [a] Newtonian and Non-Newtonian Oils |
|---|---|---|---|
| 100 | 600,000 | 0.84 | 0.83 |
| 125 | 600,000 | 0.81 | 0.75 |
| 150 | 600,000 | 0.80 | 0.70 |
| 175 | 600,000 | 0.75 | 0.61 |
| 100 | 1,000,000 | 0.74 | 0.74 |
| 125 | 1,000,000 | 0.71 | 0.74 |
| 150 | 1,000,000 | 0.70 | 0.62 |
| 175 | 1,000,000 | 0.66 | 0.67 |

[a] $R^2$ is a statistic which estimates the fraction of the variability in the data which has been described by the model, in this case a straight line. R is the correlation coefficient.

**TABLE IX**

Linear Correlation Analysis - SAE Grade Study
Fuel Efficiency Versus High-Shear-Rate Viscosity

5.7 L Engine (from Reference 147)

| Temperature °C | Shear Rate $s^{-1}$ | $R^2$ [a] Non-Newtonian Oil Only | $R^2$ [a] Newtonian and Non-Newtonian Oils |
|---|---|---|---|
| 100 | 600,000 | 0.85 | 0.36 |
| 125 | 600,000 | 0.80 | 0.26 |
| 150 | 600,000 | 0.77 | 0.19 |
| 175 | 600,000 | 0.82 | 0.17 |
| 100 | 1,000,000 | 0.92 | 0.69 |
| 125 | 1,000,000 | 0.89 | 0.62 |
| 150 | 1,000,000 | 0.87 | 0.57 |
| 175 | 1,000,000 | 0.88 | 0.53 |

[a] $R^2$ is a statistic which estimates the fraction of the variability in the data which has been described by the model, in this case a straight line. R is the correlation coefficient.

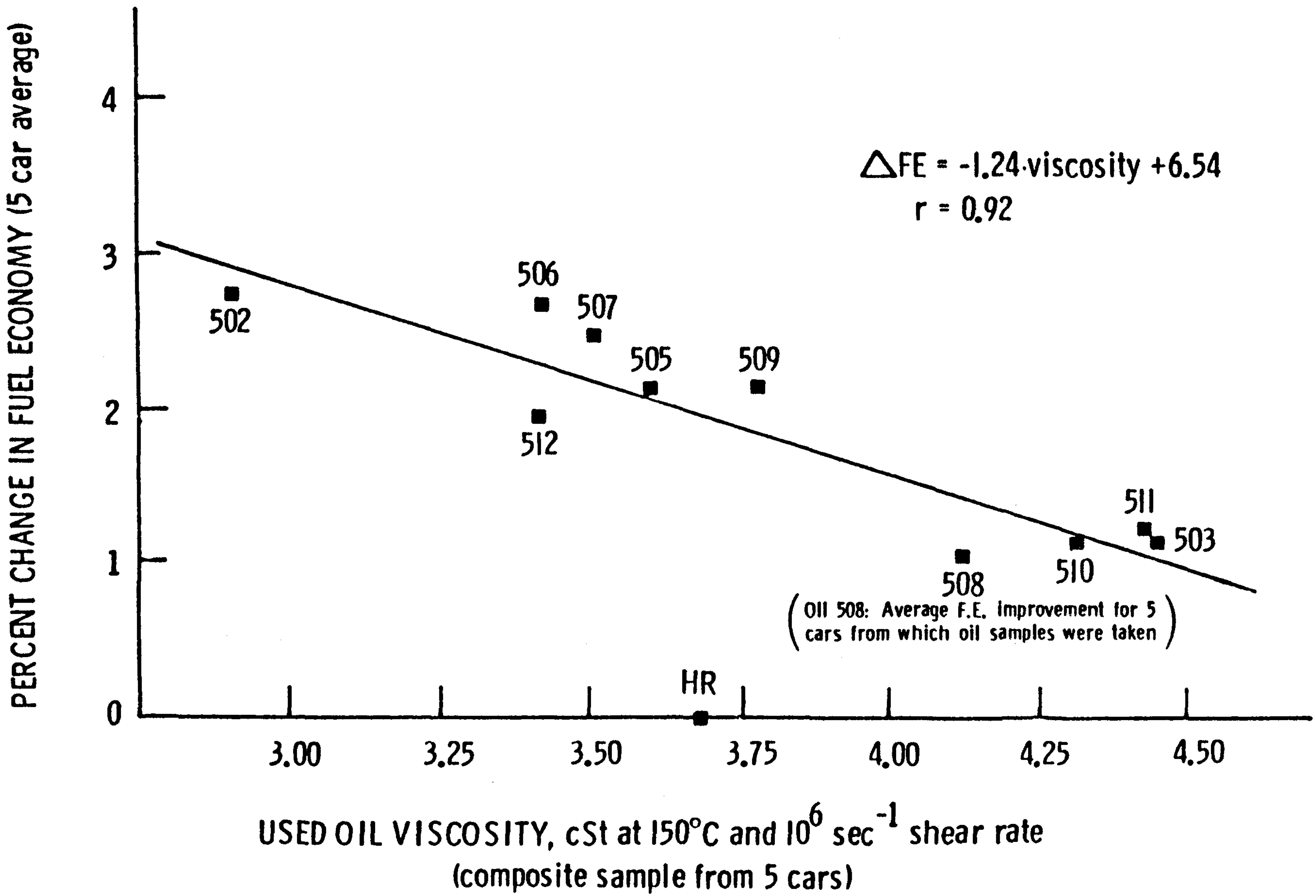


Figure 28. Percent change in fuel economy versus high shear viscosity (Reference 167).

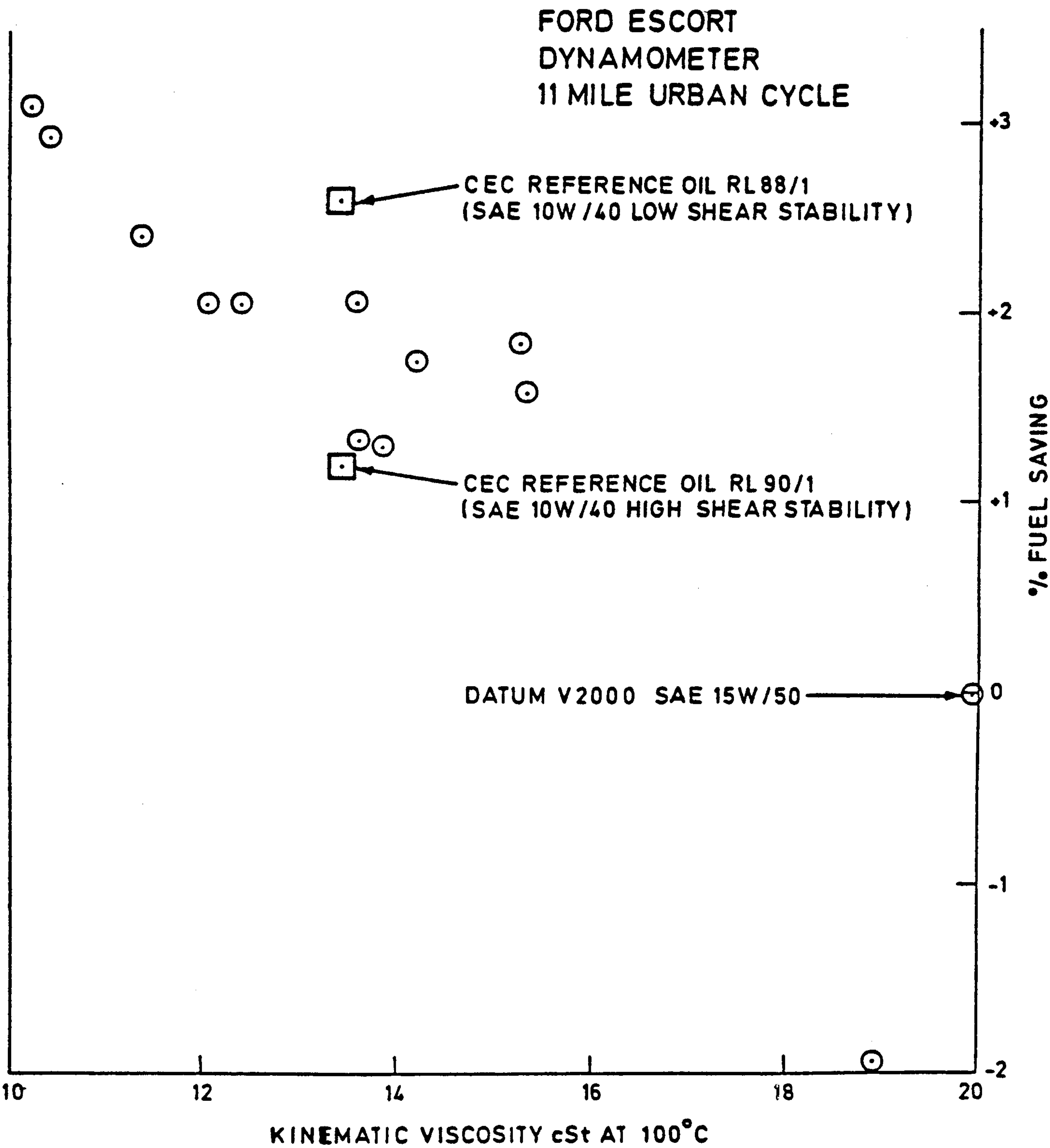


**Figure 29a. Correlation between fuel saving and kinematic viscosity - Ford Escort (Reference 172).** Reprinted with permission of the Coordinating European Council (CEC).

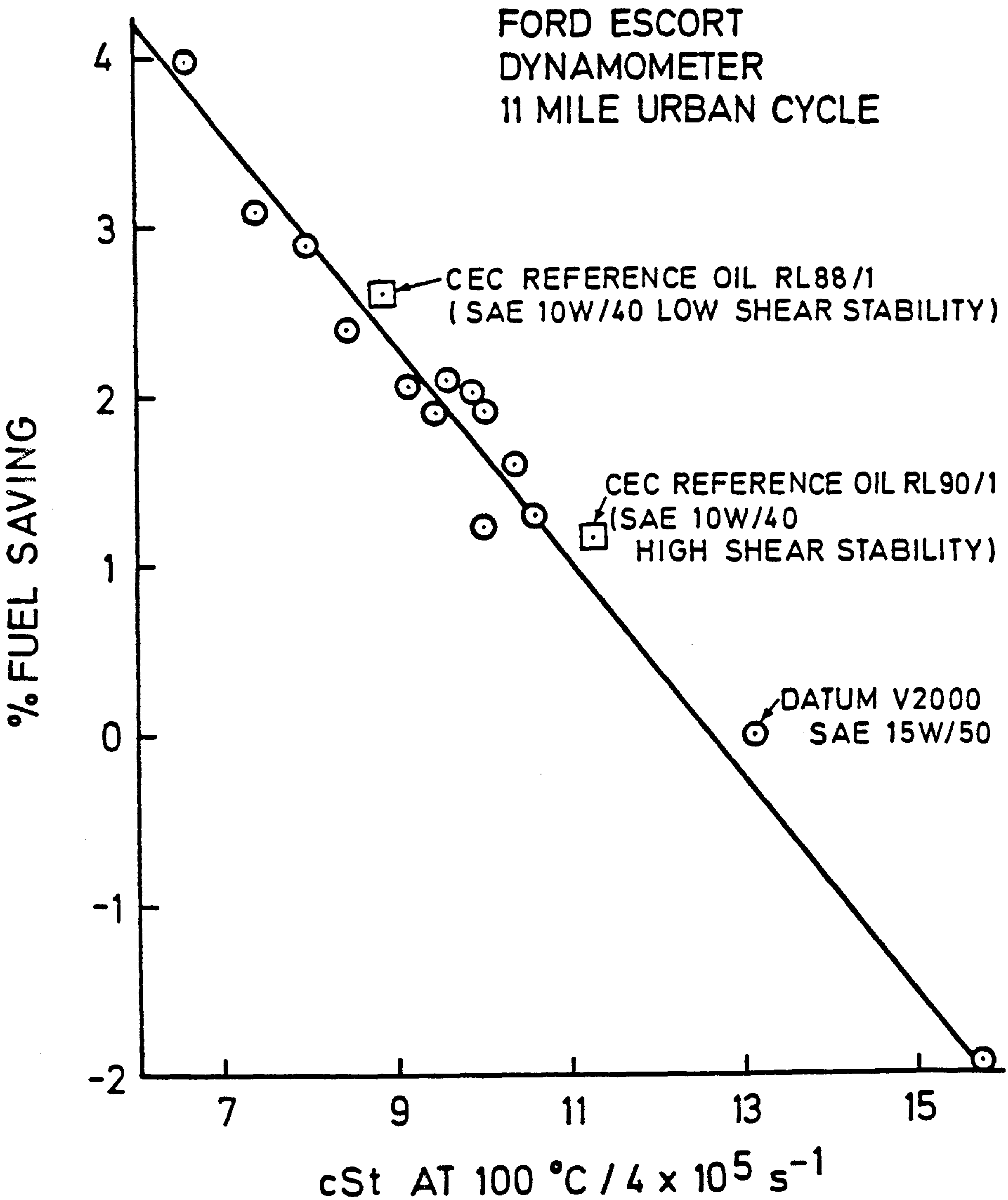


Figure 29b. Correlation between fuel saving and high-shear-rate viscosity - Ford Escort (Reference 172). Reprinted with permission of the Coordinating European Council (CEC).

high-shear correlation was obtained using a viscosity measured at a temperature of 100°C and a shear rate of $4 \times 10^5\ s^{-1}$. These conditions were chosen by the authors as the most relevant to relate to fuel efficiency for the following reasons. First, Whitehouse and Metcalf (173) showed that as far as frictional losses in motored engines are concerned, most of the power loss occurs in the cylinders and not in the bearings. Therefore, the relevant critical conditions are those experienced by the oil film in the ring zone. Furthermore, the argument continues that "since the temperature of the water jacket, even from a cold start, rapidly stabilizes at 80-90°C it is reasonable to define this as the characteristic temperature range for predicting the effect of engine oil viscosity on fuel efficiency." Finally, consideration of shearing conditions experienced by the oil in the ring zone suggests that a shear rate between $10^5$ and $10^6\ s^{-1}$ would be appropriate. Using these fresh oil viscometric criteria, the correlations shown in Figures 29 and 30 were obtained. Although the fleet (on the road) fuel economy benefits (Figure 30) of lower viscosity oils appear to be double those observed from engine-dynamometer measurements (Figure 29b), the type of driving regime and the difference in cars (engines) used in each study could account for a substantial part of the observed differences. This work has given the industry an important lead in relating differences in fuel efficiency to engine oil viscosity. However, much work over the past several years has shown that some engines (cars) and driving conditions are more responsive to engine oil viscosity than others. Therefore, more work needs to be carried out with a variety of engine (vehicle) types using a variety of driving cycles to quantitatively define the overall relationship(s) between engine oil viscosity and gasoline engine fuel economy.

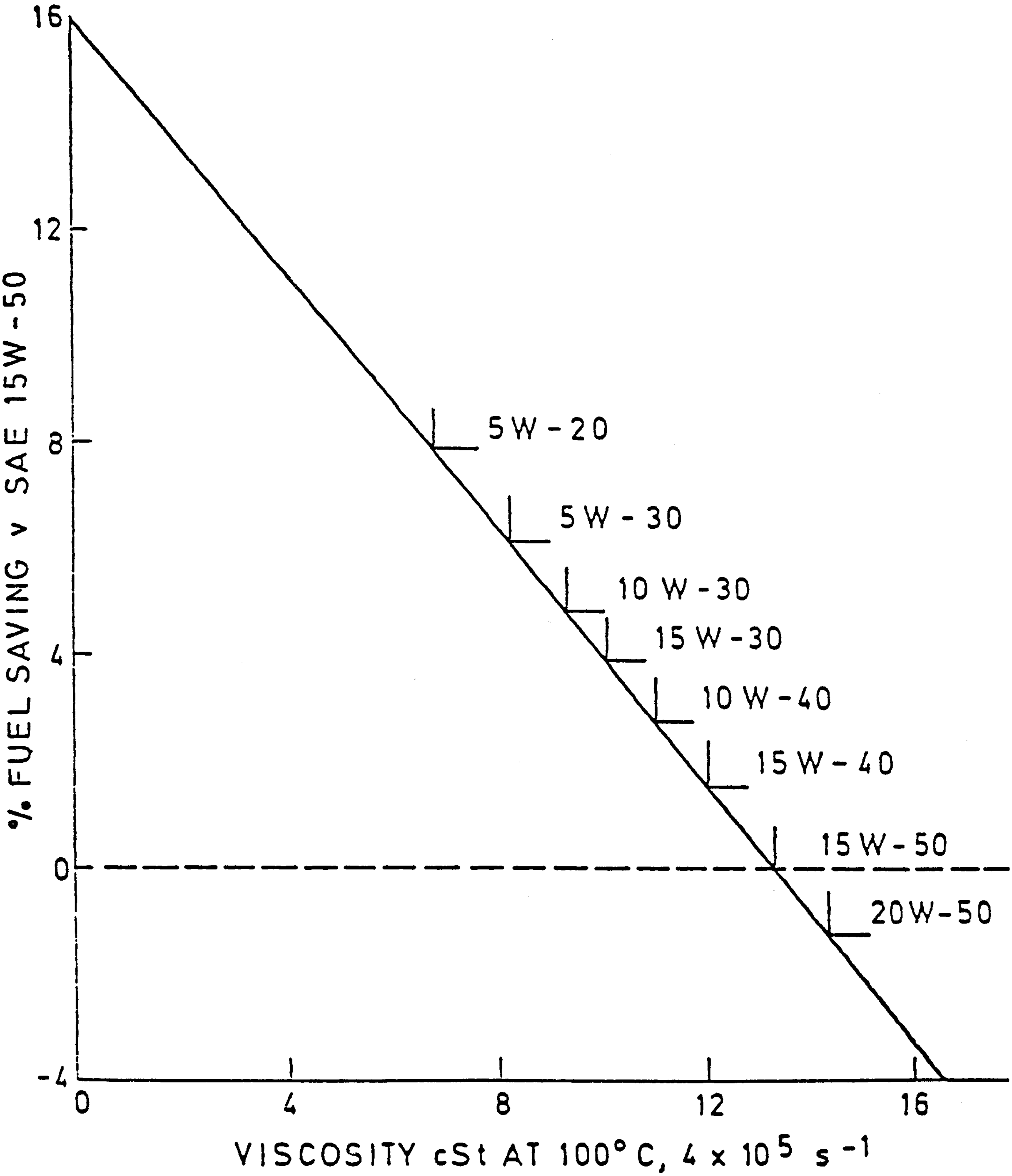


**Figure 30. Predicted fleet average fuel savings - gasoline engined cars (Reference 172).** Reprinted with permission of the Coordinating European Council (CEC).

# Discussion

## DISCUSSION

The importance of understanding the effects of oil rheological properties on engine performance is well demonstrated by the large number of both experimental and analytical investigations that have been described in the literature over the last century. Although engine oils serve other purposes, the major rheological justification for using such lubricants is to provide a fluid film between loaded, moving surfaces, and, by this means, to minimize engine friction and the wear of engine components. As indicated in these literature sources, the oil succeeds in doing this to varying degrees for different engines, engine components, and engine operating conditions. It is improbable that any engine operates with all moving surfaces separated by an oil film, even under ideal operating conditions. It is more likely that, in all engines, some metallic contact will occur between specific components in relative motion with each other, and, in such instances, the surface active additives in the oil's detergent/inhibitor package will influence engine operation, performance, and durability. This is the one fact which makes isolating and quantifying the effects of high-temperature oil rheology on engine performance very difficult. It is also the reason that, although each of the references cited in this report has added to a general knowledge of how high-temperature oil rheology affects engine performance, it is not possible to identify a specific measure of oil rheology which correlates with all aspects of engine operation. All that can be concluded is that certain properties appear to correlate with engine operation better than others.

Summation of Available Data

With regard to efforts at measuring or predicting the oil film thickness or load capacity of a journal bearing, the following conclusions are reached. There is only a marginal chance that bearing performance can be absolutely predicted using analytical models for oils containing high-molecular-weight polymeric additives. Finite element and other numerical analyses have predicted bearing performance for such complicated systems as unsteadily loaded journal bearings of finite length with oil grooves, cavitation, and bearing elasticity taken into account. However, the imposition of a lubricant whose viscosity is a function of temperature, pressure, and shear rate has not been considered, except for the case where average fluid properties have been used. The results from such analyses are accordingly suspect.

Complicating the situation further is the fact that there is no generally accepted equation for describing fluid elasticity. Numerous fluid constituitive equations have been suggested. The fluid elastic parameters in such equations, however, have only been measured for a few model fluids at conditions of temperature, pressure, and shear rate much milder than those expected in engine bearings. Since there are only limited numeric analyses that can handle Newtonian viscosity as a point function, it is doubtful that a comprehensive and detailed analysis of bearing performance for elastic fluids will be formulated in the near future. Analytical solutions are invaluable for investigating the directional effects of bearing design and average lubricant property changes by calculating film thickness and load capacity values on a relative basis. Absolute bearing calculations, however, will require yet more sophisticated calculation schemes and computational facilities.

As there are many assumptions involved in computational analyses for calculating bearing film thickness and load capacity, there are also assumptions involved in the schemes devised to measure these same two parameters. There is no clearly advantageous method for making either of these measurements. A variety of methods have been proposed and described in the literature in recent years. Since one cannot specify which techniques are the most accurate, or which are unsuitable, a summary of experimental results can best be made by listing the generalizations that derive from the experimental data that have been collected.

In general, bearing experiments conducted in laboratory bearing rigs have demonstrated that film thickness (75,76,86) and load capacity (89,90,92,93) are a function of high-temperature, high-shear rate viscosity. This is true, not only for Newtonian fluids, but also for a wide variety of polymer-containing oils. In only one paper (90) was a polymer-containing oil found whose bearing performance (in this case load capacity) did not correlate with that of polymer-free oils on the basis of viscosity measured at some suitable shear rate and temperature. However, the specific polymer-containing oil in question contained an insoluble friction modifier which could have altered the load capacity characteristics of the oil independent of its viscometric properties.

The results of bearing tests conducted in fired engines are less conclusive (73,77,83,85,91,94). In tests that used temperature rise (77,91) as a relative measure of film thickness or load capacity, bearing performance was again found to correlate with high-temperature, high-shear rate viscosity. In tests in which a variety of electrical techniques were used to measure film thickness (83,85,94), however, some oils have been found whose high-shear viscosities do not appear to correlate with bearing oil film thickness. For the majority of oils tested, however, high-temperature, high-shear rate viscosity correlates better with bearing film thickness than does kinematic viscosity measured at 100°C.

In determining the effect of oil rheology on engine wear, the problem has been to separate rheological effects from those due to additive chemistry differences among oils. For those engine components that clearly operate in the boundary lubrication regime with extensive contact between surfaces, such as in the valve train and between the rings and cylinders during specific portions of the combustion cycle, additive chemistry has been shown to overwhelm viscosity effects (37,119,120,125,126,128,130,131,132).

Even in journal bearings, which probably operate within the hydrodynamic regime more than any other engine component, the interpretation of wear measurements must include a consideration of additive chemistry effects. For either single- or multigrade oils that contain the same base stock, the same detergent/inhibitor package, and the same VI improver type at different concentrations, several studies have shown a general relationship exists between bearing wear and either kinematic viscosity (98,106,111), high-shear rate viscosity (89,92, 106,111), or polymer concentration (89,98,99,83,106,111). These relationships can become tenuous for multigrade blends that contain the same base stocks, the same detergent/inhibitor (DI) package, but different VI improver types (compare 12,106,107 with 96,98,99,103,111,112). It has been suggested that either viscoelasticity, pressure effects on viscosity, or polymer adsorption characteristics may be responsible for the lack of correlation between high-shear rate viscosity and bearing wear for certain VI improver-containing oils.

For multigrade oils with different DI packages, there appears to be no universal relationship between high-shear rate viscosity and bearing wear (76,98,99,110,111,112). It can only be concluded that for any given detergent/inhibitor package, as the viscosity is lowered, a point is reached when wear increases rapidly. This suggests a "critical viscosity" below which bearing wear is excessive. However, attempts to measure such a "critical viscosity" (107,109,111,113,115) have shown that the specific value selected is a function of oil chemistry, engine design, and engine operating conditions. Critical values ranging from 2.3 to 3.6 cP have been identified.

The effects of rheology on friction and fuel economy measurements are also difficult to isolate because of the chemical effects of surface active components in the detergent/inhibitor package. For this reason, any comparison of engine friction or fuel economy with oil rheological properties is valid only when the tests are made in the same engine or vehicle operated under identical conditions.

The majority of friction studies reported in the literature (98,146, 151,156) have been made under motored conditions. In the absence of cyclic loading due to gas pressures and its attendant effects upon lubricant squeeze films, these tests may produce different results than those made under fired conditions. The motored engine and rig studies are predominantly affected by friction at the piston ring/cylinder interface, and in the valve train. As with wear, oil viscosity does not appear to have a large influence upon valve train friction (143), presumably because it operates in the boundary lubrication regime. Fired engine friction studies generally yield results which are related to the engine as a whole. Friction measurements made in the same fired engine under a standard set of conditions suggest that high-temperature, high-shear rate viscosities would be more appropriate than kinematic viscosities in defining friction (146).

Although steady state engine tests have provided correlations between kinematic viscosity and engine fuel economy, cyclic tests have demonstrated that fuel economy correlates better with high-shear rate viscosity than with kinematic viscosity (167,168).

## Technical Questions Remaining

The conclusions outlined in the preceding paragraphs point out the need for greater understanding of the effects of oil rheology on engine performance. Based on the data available at this time, there is no single measure of oil rheology which correlates perfectly with all (or in fact any single one) of the engine operating parameters reviewed in this status report. However, a case can be made for stating that a measure of high-temperature, high-shear rate viscosity does a better job of correlating with engine wear, friction, fuel economy, or bearing performance, than other measures of oil rheology (in particular kinematic viscosity at 100°C), although even this generalization has apparent exceptions.

Although numerous questions remain regarding the relationship between oil rheology and wear, friction, or fuel economy, developing a new viscosity classification based on such measures of engine performance will be difficult. Because of complications introduced by oil additive chemistry effects, it is not expected that a unique correlation between any rheological parameters and these measures of engine performance can be developed for all oils in general. The formation of hydrodynamic films within the engine, however, represents a measure of performance which may be less sensitive to oil chemistry differences. Because current data in this area are far from complete, efforts to relate hydrodynamic film thickness, either in journal bearings or some other engine component, to oil rheological properties represents the best opportunity for development of a correlation that could be used to better define the high-temperature portion of a viscosity classification system. Such an effort should not attempt to define a "critical viscosity" for any given engine or conditions, but rather should define a general relationship between hydrodynamic film formation and oil rheological properties.

Before starting an effort to relate film formation to oil rheology, an evaluation of the methods for measuring oil film thickness should be conducted. Although the results reported to date generally tend to support each other, there are differences among specific oils. The question of whether these differences are real or due to the experimental techniques used to measure film thickness needs to be determined.

Finally, any future efforts at determining the effect of oil rheology on hydrodynamic film formation should consider factors in addition to high temperatures and shear rates. The importance of factors such as pressure and fluid elasticity on film formation have not been clearly identified at this time.

## Possible Future ASTM D02.07.0B TF/EC Efforts

Any possible future activities of the Engine Correlation Task Force depend largely on the needs and request of the SAE. Thus, before deciding on a course of action within ASTM, SAE, through the efforts of its Engine Oil Viscosity Classification Task Force, should:

(1) determine if there are sufficient data to revise the high-temperature portion of SAE J300 on the basis of a high-temperature, high-shear rate viscosity, and

(2) decide whether it wishes the ASTM D02.07.0B Engine Correlation Task Force to continue in its efforts to relate oil rheological properties to various measures of engine operation and durability.

If the SAE needs and requests additional information from the Task Force, one alternative is to continue to collect data submitted by individual researchers. This alternative has a major drawback since it involves trying to compare data submitted by different researchers, using different experimental techniques, oils, engines, and engine operating conditions. A more desirable alternative would be to conduct a coordinated industry testing effort to obtain any additional information needed and requested by the SAE. An example of a test program which could be used to answer the remaining questions concerning the effect of oil rheology on engine performance is outlined below.

A prerequisite for any cooperative industry effort would be the development of a set of engine reference oils which spans the current SAE viscosity grades. The reference oils currently being used in the development of test procedures for measuring high-temperature, high-shear rate viscosities are inadequate because they do not cover a wide enough range of viscosity grades. In addition, there is not enough left of many of these oils to run even single engine tests. A new set of reference oils should include single-grades ranging from SAE 10W to SAE 50 oils. Multigrades ranging from SAE 5W-30 to SAE 20W-50 and including all of the major VI improver types should also be included in the reference oil set. Quantities of each reference oil large enough to supply several laboratories with oil sufficient to conduct repeat engine tests should be collected.

To determine whether the different methods for measuring hydrodynamic film thickness in journal bearings that have been developed and reported on in the literature are equivalent, a subset of reference oils should be exchanged between those laboratories currently capable of conducting the different measurement techniques. A comparison of the results should be used to determine if the test methods are equivalent, or, if not, which are the most suitable for use within the industry effort.

After a test method (or methods) has been selected, each of the reference oils should be tested by at least two laboratories at several engine operating conditions in the same type of engine. Other engines would be evaluated by other laboratories. Ideally, five or more engines from several manufacturers should be tested. In each, the oil film thickness in the front main bearing of the engine would be evaluated at several operating conditions.

To compliment this film thickness data, the rheological properties of each of the reference oils should be extensively characterized. The viscosity should be measured over a range of temperatures and shear rates large enough to allow interpolation of viscosity values at any temperature and shear rate characteristic of engine operation. In addition, pressure-viscosity isotherms over a range of temperatures and possibly shear rates should be measured in order to allow calculation of pressure-viscosity coefficients for each oil. Finally, either the normal stress, elastic moduli, or relaxation times generated by each oil at a series of temperatures and shear rates should be measured and used to characterize viscoelastic behavior.

To sort out these data, a multiple linear regression statistical analysis should be conducted. Commercial computer packages designed to perform such regression analyses are readily available. In theory, the hydrodynamic film thickness at a given set of conditions in a specific engine bearing could be compared with an extensive list of rheological properties for each oil. The statistical analysis package should be capable of identifying which variable has the greatest effect on oil film thickness, which variable or variables are required to achieve an acceptable degree of correlation with oil film thickness at some preselected level of confidence, and whether unknown variables not on the list are influencing the film thickness measurements. The information provided by such a statistical analysis should be tabulated for different engines and engine operating conditions and provided to the SAE Viscosity Classification Task Force for use in improving the high-temperature portion of the SAE Viscosity Classification, J-300.

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