| AEROSPACE INFORMATION REPORT | AIR5433 | REV. B |
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# Lubricating Characteristics and Typical Properties of Lubricants Used in Aviation Propulsion and Drive Systems

## RATIONALE

Since issuance of AIR5433A, it was discovered that the pressure viscosity coefficient values for 3 and 4 cSt oils listed in Table 3 were in error. In addition, the specific heat units in Table 6 were incorrect. After review in the E-34C Lubricating Characteristics Subcommittee, these issues have been resolved in this revision.



## INTRODUCTION

This SAE Aerospace Information Report (AIR) is intended as a guide toward standard practice, but may be subject to change to keep pace with experience and technical advances. Hence, its use, where flexibility of revision is impractical, is not recommended.

The information contained herein is an attempt to establish guidelines for the selection of properties governing the lubricating characteristics of lubricants to be used in current and future aviation propulsion and drive systems. It is the intent of the SAE Committee E-34 on Propulsion Lubricants, that this document reflects currently accepted thinking in the industry and government agencies concerned with the lubrication of components in aviation primary propulsion systems, and associated power transfer gearboxes and transmissions.

The content of this AIR is the result of communication among lubricant manufacturers, hardware manufacturers, lubricant specialists, tribologists, lubricant specifiers, and lubricant users. Continued communication will be encouraged to maintain the information contained herein as current as possible. Users are advised that they can contribute to subsequent changes and additions to this document by their comments, suggestions and criticism.

In present standards, some properties of importance in determining lubricating ability and the methods to quantify these properties tend to be ignored and in some cases there are no standardized test methods. Because these properties are important to designers, they are included in this document even if methods are not specified or standardized. In these cases, typical values obtained by particular methods are presented. It is intended that by including these data, focus will be brought on the deficiencies in these areas.

It must also be recognized that the selection of properties relevant to lubricating characteristics must be made in the light of the other characteristics necessary or desirable in the lubricant. This document is not intended to specify the properties of fluids optimized for lubricating ability; instead it is intended to provide the guidance to be considered, along with the other requirements for the application, in determining the values to be specified for each property. Trade-off in the selection of limits for various properties must be made based on the relative impact of these properties on each other and on the system.

TABLE OF CONTENTS

## 1. SCOPE

This SAE Aerospace Information Report (AIR) establishes guidance for the specification of formulated lubricant properties which contribute to the lubricating function in bearings, gears, clutches and seals of aviation propulsion and drive systems.

## 2. REFERENCES

### 2.1 Applicable Documents

The following publications form a part of this document to the extent specified herein. The latest issue of SAE publications shall apply. The applicable issue of the other publications shall be the issue in effect on the date of purchase. In the event of conflict between the text of this document and references cited herein, the text of this document takes precedence. Nothing in this document, however, supersedes applicable laws and regulations unless a specific exemption has been obtained.

[1] Anonymous. Glossary of Terms and Definitions in the Field of Friction, Wear and Lubrication - Tribology, Research Group on Wear Engineering Materials, Organization for Economic Cooperation Development, Paris, 1969.

[2] Cameron, A., “Principles of Lubrication”, Longmans, London, 1966, P.31.

[3] Annual Book of ASTM Standards 2000, Section 5, Petroleum Products, Lubricants, and Fossil Fuels, Volume 05.01 Petroleum Products and Lubricants (I): D 56 – D 2596

[4] Cheng, H.S., Micro-Elastohydrodynamic Lubrication, U.S. National Congress of Applied Mechanics, 21-25 June 1982, pp.161-170.

[5] Hamrock, Bernard J., Fundamentals of Fluid Film Lubrication, 1994. McGraw-Hill. ISBN 0070259569.

[6] Gupta, P.K., Cheng, H.S., Forster, N.H., Schrand, J.B., “Viscoelastic Effects in MIL-L-7808 Type Lubricant, Part I: Analytical Formulation, Trib. Trans. Vol. 35, 2, pp 269-274 (1992).

### 2.2 Definitions and Terminology

The following terms are defined according to the definitions given in Reference [1] and according to the customary engineering descriptions used in the aerospace community.

FATIGUE - Removal of particles detached by fatigue arising from cyclic stress variation. (1) Spalling is removal of material by fatigue resulting from the global contact stress. It can be surface or subsurface initiated; (2) Pitting is removal of material by fatigue resulting from local stress within the contact. It is surface or near surface initiated. Fatigue pits are of smaller scale than fatigue spalls; (3) Micro-pitting (Peeling, Frosting) – This type of fatigue is characterized by surface initiated spalling in the order of 5 to 13 micron (0.0002 to 0.0005 inch) in depth which occurs where the surface finishes have many asperities greater than the lubricant film thickness. Normally micro-pitting is only a slight distress to the contact surface and appears to be nothing more than a frosted appearance.

FRETTING - A wear phenomena occurring between two surfaces having oscillatory relative motion of small, <1 mm (<0.040 inch) amplitude.

SCRATCHING - The formation of fine scratches in the direction of sliding. Scratching may be due to asperities on the harder surface or hard particles between the surfaces or embedded in one of the surfaces. Scratching is considered to be less damaging than scoring (scuffing).

SCORING (SCUFFING) - The formation of severe scratches in the direction of sliding. (1) Scoring may be due to local solid phase welding or to abrasion. (2) In the USA the term scuffing is sometimes used as a synonym for scoring. (3) In the U.K. scuffing implies local solid-phase welding only. (4) Minor damage should be called scratching rather than scoring (scuffing).

GLAZING - The formation of a glazed or polished surface due to localized plastic flow.

SMEARING - Plastic flow and removal of material from a surface and re-deposition as a thin layer on one or both surfaces.

CORROSION - Reaction at surfaces due to electrochemical or direct chemical attack resulting in discoloration and/or pitting of the surface. Can occur on functional and nonfunctional surfaces.

CORROSIVE WEAR (TRIBOCHEMICAL WEAR) - A wear process in which chemical or electrochemical reaction with the environment predominates. (1) Usually, corrosive wear is a mild form of wear, but it may become very serious, especially at high temperatures or in moist environments. (2) In some instances, chemical reaction takes place first, followed by the removal of corrosion products by mechanical action; in other instances, mechanical action takes place first, characterized by the formation of very small debris which subsequently is chemically transformed; the phenomena may be mutually enhancing. Oxidative wear is one form of corrosive wear.

FRETTING CORROSION - A form of fretting in which chemical reaction predominates. (1) Fretting corrosion is often characterized by the removal of particles and subsequent formation of oxides, which are often abrasive, and so increases wear. (2) Fretting corrosion can involve other chemical reaction products, which may not be abrasive.

WEAR - The progressive loss of substance from the operating surface of a body occurring as a result of relative motion at the surface. (1) Wear is usually detrimental, but in mild form it may be beneficial, e.g., during running in. (2) A body may become unserviceable as a result of major plastic deformation. Small-scale plastic deformation occurs in almost all wear processes.



2.3 Symbols and Abbreviations

| | |
|---|---|
| a | Major axis of the Hertzian contact |
| AIR | Aerospace Information Report |
| ASTM | American Society for Testing and Materials |
| b | Minor axis of the Hertzian contact |
| c, C | Constants |
| d | Diameter |
| E | Elastic modulus |
| EHD | Elastohydrodynamic |
| EP | Extreme Pressure |
| f | Regression coefficient |
| $F_N$ | Normal load |
| $F_S$ | Shear force |
| g | Regression coefficient |
| h | Lubricant film thickness |
| k | Ellipticity of contact |
| K | Thermal conductivity |
| m | Slope of viscosity-temperature graph, ASTM D341-43 |

| | |
|---|---|
| p | Lubricant pressure |
| $P_{max}$ | Maximum Hertzian stress |
| R | Equivalent radius |
| S | Silde-to-roll ratio |
| Subscripts 1, 2 | Refer to bodies 1 and 2 |
| T | Temperature |
| $T_b$ | Bulk temperature |
| TCP | Tricresyl Phosphate |
| $T_f$ | "Flash" temperature |
| $T_K$ | Absolute temperature |
| U | Velocity |
| $U_e$ | Entraining velocity |
| $U_s$ | Sliding velocity |
| VI | Viscosity Index |
| w | Load per unit width |
| $\alpha$ | Pressure viscosity coefficient |
| $\beta$ | Temperature viscosity coefficient |
| $\lambda = h/\sigma$ | Lambda ratio |
| $\eta$ | Kinematic viscosity |
| $\mu$ | Dynamic viscosity |
| $\nu$ | Poisson's ratio |
| $\rho$ | Specific gravity |
| $\sigma$ | Surface roughness (Ra) |
| $\theta$ | Contact angle |

# 3. FUNDAMENTALS OF LUBRICATION AND LUBRICANT PROPERTIES

## 3.1 Lubricant Function

The design and formulation of aeropropulsion lubricating oils is a compromise among attributes associated with lubrication, low temperature pumping, high temperature evaporation, thermal stability and coking. Specification requirements seek to find a balance of properties.

In any lubricated contact, the lubricant generally must perform two primary functions. These functions are: (1) lubricant film generation for the reduction of wear and (2) heat removal to stabilize component temperatures. While the primary purpose of a lubricant is to extend component life by reduction of wear and surface distress, only a small portion of the lubricant supplied to a component is required for this purpose. In components like rolling element bearings and gears with high load Hertzian contacts, less than 10% of the lubricant supplied is required for lubrication. The remaining 90% is supplied to provide adequate cooling to control and stabilize component temperatures at an acceptable level.

Both physical and chemical properties of a lubricant affect these two functions. The primary physical properties that determine the lubricating and wear reduction performance of a lubricant include its viscosity and pressure-viscosity coefficient. These are the properties that generate hydrodynamic films, the thickness of which are determined by contact geometry, lubricant viscosity, pressure viscosity coefficient, load, surface speed and temperature. Another inherent physical property of the lubricant is the traction (friction) coefficient. The traction of a lubricant film is a function of temperature and pressure. The traction of lubricant films under shear is directly linked to heat generation.

In addition to the above physical properties, the type and quantity of surface active boundary lubricating additives that are present play an important role in a lubricant’s surface film formation to minimize adhesion and wear. Boundary lubricating additives are particularly important at low speeds, high loads and high temperatures where the films generated by hydrodynamic or elastohydrodynamic (EHD) mechanisms may be insufficient. In this “mixed-film” lubrication regime the oil film thickness is less than the combined surface roughness of the surfaces in contact.

The lubricant also serves to help capture and filter debris. The lubricant furthermore transports fresh chemistry to the lubricated contacts.

The primary properties, which determine a lubricant’s ability to remove heat from a component, are density, heat capacity, thermal conductivity and thermal stability. These properties determine how well the lubricant can absorb heat from the mechanical component and how well it can stand up to the thermal environment it is subjected to. Antioxidant additives are used to increase the thermal stability of a lubricant base stock to permit it to be used at elevated temperatures.

While not primarily a lubricating function in the strict sense of the word, corrosion protection is an additional performance requirement for many lubricants. Corrosion preventive additives are frequently used to improve the corrosion protection of a given lubricant. Corrosion protection can be provided by a combination of effects including chemically adherent surface film formation, acid neutralization and moisture elimination or control. The effectiveness of anticorrosion additives is controlled by their detailed chemical properties.

Another critical property is the electrical conductivity of the lubricant. This relates to the design and function of electrical components such as rectifying diodes and estimates the susceptibility of the oil to spontaneously discharge a static electrical charge. Electrical conductivity is known to be a function of temperature. It is desirable to have the electrical conductivity measured at both 20 and 80 °C to determine the effect of temperature and allow for proper calibration. Electrical conductivity is used to assess the risk of generating static charges due to oil flowing past insulating materials. Electrical conductivity of an oil may increase rapidly in use.

3.2 Lubrication and Failure Mechanisms

3.2.1 Introduction

The performance of the lubricant, with its physical and chemical properties, in most cases cannot be formulated into simple and reliable equations to predict the role of lubricant properties in the failure process. Failure in the form of wear, for example, is not an intrinsic property of a bearing or gear material. Also, the performance of an oil is not an intrinsic property of the lubricant alone, but it is the result of the physical and chemical interactions within an entire lubricated contact system. Consequently, the design or selection of a lubricant is based on both science and experience. The rationale is derived from both lubrication and failure mechanisms and their interactions within a lubricated contact system.

The lubricant prevents failure through the formation of lubricating films by hydrodynamic lubrication, elastohydrodynamic (EHD) lubrication and boundary lubrication. The criteria for failure are judged by the user. If the deterioration of the surfaces or level of friction has progressed to the degree that it threatens the essential function of the component, it can be considered to have failed.

Through experience the engineer recognizes distinct classes of failure (see 2.2). These failures are defined without presupposing the exact mechanism by which they occur. They are defined in engineering terms based on a description of observations. The observations and their classifications reflect the remaining evidence of a complicated sequence of events involving many physical and chemical processes that preceded it. Associated with the physical and chemical interactions on the surfaces are several mechanistic wear processes, which generally fall under the basic mechanisms of:

Adhesion
Chemical reaction
Plastic flow
Fatigue

During the service life of a component, the lubricant must provide the lubrication mechanisms required to prevent or control these basic wear processes.

3.2.2 Types of Contacts

The geometry of contacting bodies creates the character of the contact with respect to stress and the mechanisms of lubrication. Conformal contacts, where contacting body geometries closely match one another and distribute the load over relatively large areas. Conformal contacts, like journal bearings and the cage/land interface in rolling element bearings, have clearance spaces that must be preserved during operation. Concentrated contacts are non-conformal. They concentrate the load over small areas like that found in rolling element bearings and gear mesh contacts. The lubrication of conformal contacts is primarily governed by hydrodynamic mechanisms where oil viscosity is the controlling property. The lubrication of concentrated contacts is primarily governed by elastohydrodynamic mechanisms where velocity, viscosity, pressure-viscosity coefficient and traction coefficient are the controlling properties. Boundary lubrication from oil chemical attributes is associated with both conformal and concentrated contacts, when lubricant film thickness becomes less than the average combined surface roughness.



It is helpful to grasp the perspective of tribological contacts for their scale. The lubricated length of a conformal contact is generally less than a centimeter ($10^{-2}$ m). The lubricated length of a concentrated contact is generally less than a millimeter ($10^{-3}$ m). These load-bearing areas are on a “macro-scale” where they can be viewed with the unaided eye.

Within the world of macro-scale contacts are “micro-scale” features which control tribological behavior. Surface roughness features (asperities) and the grain sizes of metals are on the order of microns ($10^{-6}$ m). These microscopic features are critically linked to yet a lower scale of things that control lubrication and failure phenomena. Adsorbed films of molecular size that prevent adhesion between surfaces are on the order of nanometers ($10^{-9}$ m). Here lies a fundamental difficulty within the tribology of component hardware. Lubrication and failure mechanisms are controlled by micro-scale phenomena, but the parameters that control loads and motions of lubricated contacts are on the macro-scale engineering level. To connect lubrication and failure mechanisms, which take place on a micro-scale, surface stresses and kinematics on the engineering level should be considered as input to a tribological contact system. The role of the lubricant and its design and selection rationale can be understood by considering the structural elements of a tribological contact system.

3.2.3 Structural Elements of a Lubricated Contact

Engine and transmission systems have a very wide range of lubricated contacts within their basic components of bearings, gears, seals, clutches and splines. The lubricated contacts cover a wide range of contact stresses and rolling/sliding motion.

For a concentrated contact, the performance is derived from the integrity of four general regions as shown in Figure 1. Each region performs certain functions in the lubrication and failure process. The success of the lubricant depends upon how well it handles the normal stress and accommodates the tangential shear within these regions.

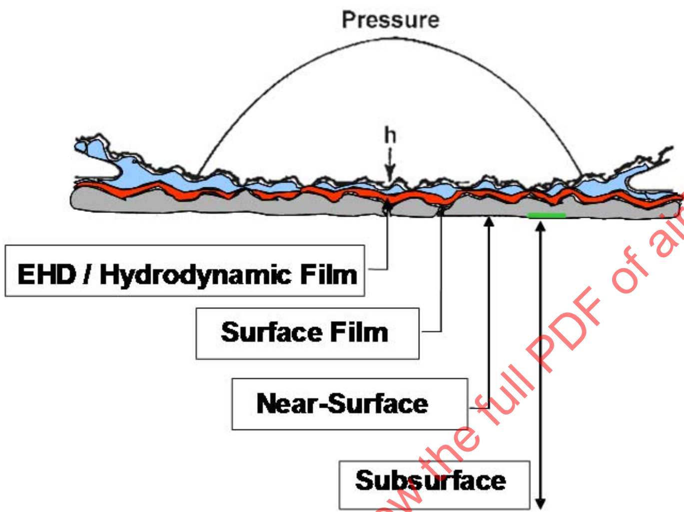


FIGURE 1 - STRUCTURAL ELEMENTS OF A LUBRICATED CONTACT



The hydrodynamic/EHD lubricated region is created by the generation of an elastohydrodynamic (EHD) lubricant film. On a global scale, the EHD film is derived from the hydrodynamic pressure generated in the inlet region of the contact. On a local scale, it is derived from the micro-EHD lubrication action associated with the local topography of the surfaces Reference [4]. Micro-EHD is an EHD effect on an asperity scale, mostly associated with surface scratches transverse to the direction of motion. The EHD/micro-EHD region is typically less than 1 μm thick.

The surface film region contains the thin outer layers of the surface. These consist of surface oxides, adsorbed films and chemical reaction films derived from the lubricant and its additives. The surface films are almost always less than 1 μm thick.

The near-surface region contains the inner layers of the surface. This region may include a finely structured and highly worked or mechanically mixed layer. The deformed layers, which are of a different microstructure than the material below them, may arise from surface preparation techniques such as grinding and honing. These may also be induced during operation; for example, during running-in. Hardness and residual stress may vary significantly in this region, and may also be substantially different from the bulk material below. The near-surface region may be on the order of 50 μm below the surface.

For concentrated contacts, a subsurface region can be defined, which may be 50 to 1000 μm below the surface. This region is not significantly affected by the mechanical processes that produce the surface or the asperity-induced changes that occurring during operation. Its microstructure and hardness may still be different from the bulk material below it, and significant residual stresses may be present. These stresses and microstructures, however, are the result of macro processes such as heat treatment, surface hardening and forging. For typical Hertzian contact pressures, the maximum shear stress is located within the subsurface region. After millions of stress cycles, alterations in residual stress and microstructure have been noted in the subsurface region. These alterations are attributed to the rolling contact stress field and may be the precursors for rolling contact fatigue. In other words, the detrimental global contact stresses are communicated to the subsurface region where subsurface-initiated fatigue commences.

One can also define a "quiescent zone" which is located between the near-surface region and the subsurface region. The "quiescent zone" resides at a depth below the surface in which the local asperity and surface defect stresses are not significant and the stress field from the macroscopic Hertzian contact stress is not yet appreciable. This zone is quiescent from the point of view of stress and the accumulation of plastic flow and fatigue damage. The existence of the quiescent zone is important with regard to rolling contact fatigue. It inhibits the propagation of cracks between the stress field in the near-surface region and the stress field in the subsurface region.

3.2.4 Stress Fields of a Lubricated Contact

The performance of a lubricated contact is associated with a normal stress field and a tangential stress field. The normal stress of conformal contacts is typically in the range of mega-Pascals (MPa), 1 000 000 N/m$^2$. The normal stress of concentrated contacts is typically in the range of giga-Pascals (GPa), 1 000 000 000 N/m$^2$. Concentrated contacts encounter significant elastic deformation. The elastic contact area and stress distribution can be calculated by using Hertzian theory, Reference [2].

The tangential stress is directly linked to the coefficient of friction. The tangential stress of a contact under full elastohydrodynamic lubrication is the shear stress in the lubricant film. Under mixed mode and boundary lubrication, friction events associated with interacting asperities between the surfaces also contribute to the tangential stress, so the resulting tangential stress is a combination of lubricant shear stress and shear stress at the asperity contact. The friction coefficient (also called traction coefficient) is determined from the ratio of the resulting tangential force to the applied normal load. The friction coefficient can also be determined from the ratio of the tangential stress to the mean Hertzian stress. The traction coefficient generally ranges from 0.002 to 0.100% for lubricated rolling contacts, while values as high as 0.500 have been recorded under adverse conditions.

Because lubrication and failure mechanisms are frequently controlled by asperity-scale phenomena, it is important to remember that normal and tangential stresses at asperities may not be that different for conformal contacts as they are for highly stressed concentrated contacts. Asperity stresses depend on how the load is momentarily distributed among the surface features. Conformal contacts, with rough surfaces and a few contacting asperities, can have stresses similar to asperities found in concentrated contacts with smoother surfaces. Because concentrated contacts tend to have finer surface finishes than conformal contacts, asperity stresses for both conformal and concentrated contacts may be on the same order of magnitude, even if their “global” or overall pressure across the contact area is substantially different. The major difference between conformal and concentrated contacts is that the stress field of the latter penetrates deeper into the material. In addition, high global pressure for concentrated contacts can dramatically change fluid viscosity

Whether normal and tangential stresses are on a global scale or asperity scale, the objective of lubrication is to create or preserve interposing material that bears the normal stress and accommodates the tangential shear. In lubricated contact systems lubricating mechanisms are applied to suppress failure mechanisms. The most robust contact systems are those that have boundary lubricating mechanisms preserve surface integrity to allow hydrodynamic and EHD mechanisms to maintain operative after incidental asperity contact.

3.2.5 Lubrication Mechanisms

The physical and chemical properties of the lubricant work within the structural elements of the contact through the mechanisms of hydrodynamic (or elastohydrodynamic) lubrication and boundary lubrication.

Elastohydrodynamic Lubrication: The formation of a hydrodynamic or EHD lubricant film contributes to lubrication by reducing the local stresses between the surfaces and by creating a lubricant film that is easy to shear. For a concentrated contact illustrated in Figure 2, the pressure and elastic deformation are similar to the Hertzian contact condition of dry contact. This Hertzian elastic deformation is critical as it creates the geometry of the surfaces to be lubricated. The Hertzian geometry creates three functional regions that the lubricant must traverse.

The formation of an EHD lubricant film is created by the hydrodynamic pressure generated in the converging space upstream of the Hertzian region. The function of the inlet region is to generate sufficient pressure to squeeze a small quantity of fluid into the high pressure Hertzian region. The success of this action is attributed to a significant rise in viscosity with pressure. The lubricant film thickness created from the inlet pressure generation can be accurately calculated from EHD theory. The required lubricant properties are its viscosity and pressure-viscosity coefficient at the temperature of the fluid in the inlet region. The inlet fluid temperature is generally close to the surface temperature of the contacting bodies. High speeds, typical of many aeropropulsion components, increase the temperature within the inlet region due to viscous shear action. In addition to the lubricant properties, the thickness of the EHD oil film is a function of the radius of curvature and the entraining (rolling) velocity of the surfaces in contact. Remarkably, the film thickness is less dependent upon the contact stress and elastic modulus of the contacting materials. EHD theory has proven to be a very useful design tool for predicting the lubrication regime for various applications. However, it is not sufficient to accurately predict wear, scuffing, and fatigue failure. This is partly because EHD lubrication is primarily an inlet phenomenon; that is, its major role occurs in a region removed from the Hertzian region where the more local events involved in failure initiation take place. The EHD lubrication process can significantly influence the severity of these local events. The nominal thickness of an EHD film determines the degree of asperity interaction between the surfaces. Failure mechanisms are associated with several less quantifiable processes that occur in the Hertzian region of the contact.

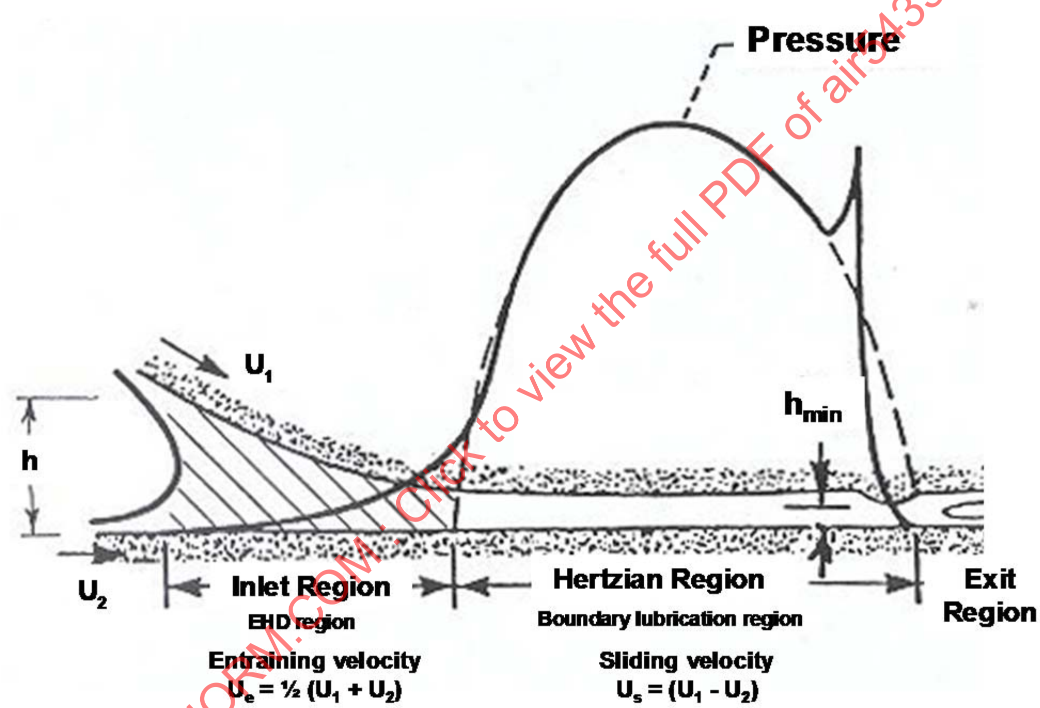


FIGURE 2 - EHD LUBRICATION



Surface temperature is a key link between lubrication and failure. Temperature significantly influences the viscous properties of the lubricant that control the thickness of an EHD generated film. Temperature is also a major driving force in the formation of chemical reaction films. It also influences the rate of lubricant degradation. Surface temperature influences the strength of surface films, as well as the flow properties of the material in the near-surface region. Consequently, it is not surprising that the total temperature is a frequently used criterion for failure, such as scuffing (or scoring).

From a simplistic point of view the contact temperature (T) is the sum of a bulk temperature ($T_b$) of the component and the "flash" temperature ($T_f$) associated with the instantaneous temperature rise derived from the friction within the lubricated contact. Flash temperature may arise from the traction of the lubricant film as well as from the energy dissipated from the adhesion, plastic flow of surface films, and deformation of the material within the near-surface region. The magnitude of $T_f$ can be predicted if simplifying assumptions about the coefficient of friction and conductive heat transfer are made.

Boundary Lubrication: It is well known that surface films are important to boundary lubrication because they prevent adhesion and provide a film that is easy to shear. These films may be in the form of oxides, adsorbed films from surfactants and chemical reaction films from other additives or the bulk fluid itself. Typical aviation oils have anti-wear additives or EP (extreme pressure) additive chemistry to create surface films to prevent adhesion. Other additives, such as anti-oxidants, can also create surface films for boundary lubrication. Boundary lubricating surface films are schematically shown in Figure 3. Boundary lubricating surface films can be formed from energy generated in localized asperity interaction. The plastic flow and removal of oxide films create sufficient surface energy for chemical reaction films to form.

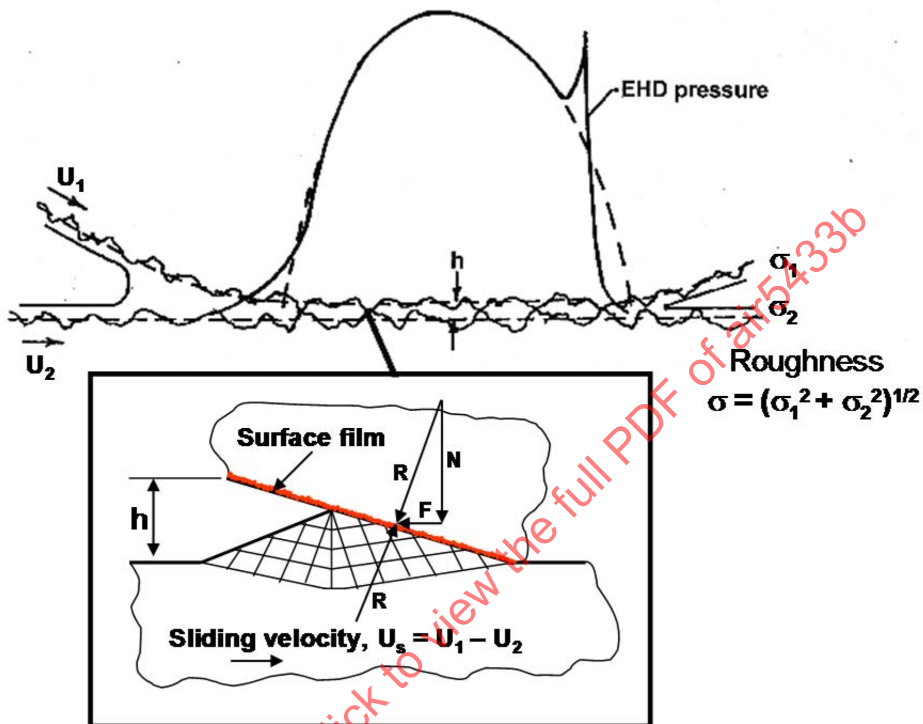


FIGURE 3 - SURFACE FILMS FROM BOUNDARY LUBRICATION MECHANISMS



The interactions of surface films are very complex. Most studies on the subject have focused on the chemical identification or phenomenological effect of surface films, but little is known about the mechanism of protection, the means of removal, or the rate of reformation. At high temperatures the oxidation of the base fluid can contribute to surface film formation. There have been many studies of the catalytic effect of metals on the bulk oxidation of lubricants. Similar oxidative processes can occur under the thermal stress environment in the contact region where intermediate oxidation species can react with the surface or organo-metallic material that may be present. These reactions can influence boundary lubrication in several ways; such as by corrosive wear, by competition with other additives, or by forming polymeric material (friction polymer).

The contribution of surface films in preventing wear and controlling scuffing and fatigue is significant. The time and spatial distribution of the various surface films within the contact seem to be important. It is essential that active and effective boundary lubricating chemistry form surface films, and allow some degree of polishing, for lubricated contacts to operate with extremely thin EHD films.

Mixed-Film Lubrication: Mixed-film lubrication is a term used to describe operation with a mixture of hydrodynamic/EHD lubrication mechanisms and boundary lubrication mechanisms. Mixed film lubrication, which can range from almost a full hydrodynamic film to almost complete boundary lubrication, is common among many aeropropulsion lubricated contacts. A transition through this range occurs during startup and shutdown. The lambda ratio ( $\lambda = h/\sigma$), is a useful engineering quantity to describe the ratio of the hydrodynamic or EHD lubricant film thickness (h) to the average combined roughness height ($\sigma$) of the interacting surfaces (see Figure 3). The combined roughness has been defined as the average of the Ra roughnesses of the surfaces or the square root of the sum of the squared RMS roughnesses of those surfaces, Reference [5]. When lambda is $\lambda$>3, wear, scuffing, and micro-pitting (or micro-spalling) are mostly eliminated. Surface initiated fatigue and wear are controlled by $h/\sigma$ as a result of affecting normal and tangential stress on an asperity scale. Its connection with surface-initiated fatigue seems to be more obvious than failure modes associated with wear or scuffing. The latter failure modes generally appear at low lambda, less than 1, where the concept of lambda loses some of its meaning.

When the combined roughness ($\sigma$) is the same order of magnitude as the film thickness (h), the surface topography becomes intimately involved in the lubrication process itself in the form of micro-EHD lubrication. Local hydrodynamic or EHD pressures can be generated at asperity sites or topographical features to actually depress these features and boost local film formation. Micro-EHD lubrication should not be confused with boundary lubrication. Micro-EHD lubrication can also affect the friction, or traction within the contact, see Reference [4].

3.2.6 Failure Processes

3.2.6.1 Five Key Parameters

The challenges of lubricant formulation, along with bearing and gear design, are to invoke lubrication mechanisms of full-film EHD, partial-film EHD and boundary film lubrication and to avoid failure mechanisms associated with wear, scuffing, and fatigue. With the exception of hydrodynamic and EHD film thickness prediction, lubricant performance is determined by testing and service experience. Tribological testing must capture the micro-scale lubrication and failure mechanisms controlling performance and somehow link the testing results to the macro-scale design space that the design engineer can utilize. Five key tribology parameters have been found to be effective in linking tribological mechanisms in service with testing and design. The five parameters, which are derived from EHD theory, are shown in Figure 4.

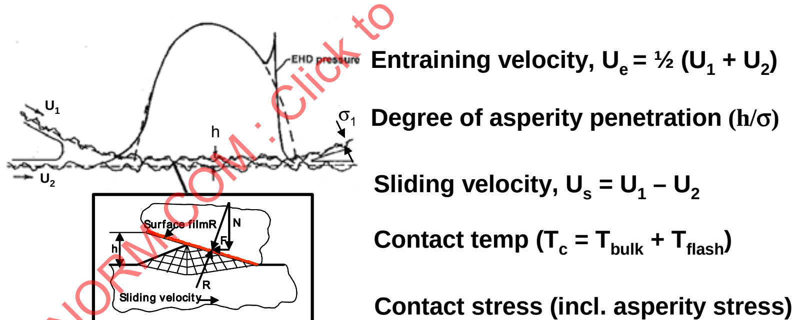


FIGURE 4 – FIVE KEY TRIBOLOGY PARAMETERS

The entraining velocity ($U_e$) controls the formation of EHD film thickness through the generation of pressure in the inlet region. The sliding velocity ($U_s$) controls the shear and heat generation within the Hertzian region. The film thickness-to-surface roughness ratio ($h/\sigma$), or Lambda ratio ($\lambda$), controls the degree of asperity interaction. The contact stress is considered on a global Hertzian contact scale and a local asperity scale. Wear, micro-pitting and micro-scuffing are generally initiated on a local scale and subsequently propagated to a global scale. Contact temperature ($T_c$) is considered the sum of the bulk temperature ($T_b$) and the friction-generated flash temperature ($T_f$). The contact temperature controls reaction rates between the lubricant and surface material. It also controls the shear properties at the interface.

3.2.6.2 General Failure Scenarios

The five key tribological parameters provide an engineering linkage to lubrication and failure scenarios. With little or no asperity penetration ($\lambda >> 1$), major failures of wear, scuffing, and surface initiated fatigue are avoided. It is generally recognized that the loss of an EHD film is a necessary, but not sufficient, condition for failure mechanisms. Perhaps the most important quantity in connection with failure is the deformation attributes of the near-surface region. It is unfortunate that there is little understanding of near-surface mechanical properties or the attributes needed to complement the various lubricating mechanisms to prevent failure. To maintain surface integrity, the near-surface region must prevent micro-fracture and maintain a viable surface finish even in the presence of some plastic flow.

The severity of asperity interaction is reflected in the normal load $F_N$ (see Figure 3). The normal load on the asperity is influenced by the thickness (h) of lubricant film that is generated. The shear force, $F_S$, is influenced by the various surface films and micro-EHD lubricant films, along with the flow properties in the near-surface region. The exact mechanism whereby shear stress is applied to the near-surface region is not well understood. This could come about through metal-to-metal adhesion, or possibly through shear stresses applied locally through a thin lubricant film.

The severity of interaction is important to the initiation and propagation of the events toward failure. The severity of interaction will determine whether the result is (1) a benign elastic encounter, (2) a further accumulation of plastic fracture sites that can lead to the generation of wear particles (e.g., micro-pitting, or polishing wear debris), (3) oxidative or corrosive wear, or (4) the advancing of adhesive material transfer, which can lead to adhesive wear or scuffing.

Failure can occur through four basic processes of adhesion, plastic deformation, fatigue and chemical reactions. The recognized failure descriptions in bearings and gears, such as smearing, micro-pitting, spalling and scuffing, are not singly connected to these basic processes but are associated with the interaction of the processes both simultaneously and sequentially. The importance of connecting these basic processes with the commonly accepted failure modes is associated with engineering decisions required to extend the onset of failure through improved lubricant properties, material selection, design or allowable operating conditions.



3.2.6.3 Local Surface Film Removal

Reasonable high asperity penetration ($\lambda$ <1), along with high asperity stress and sliding speed ($U_s$), can cause boundary films trapped between contacting asperities to become stretched until they rupture. This creates the formation of metal-to-metal contact on an atomic scale and the result is strong adhesive or welded junctions.

With relative motion between the contacting asperities, “junction growth” occurs by plastic deformation. Fracture ultimately takes place, and it can occur at a location different from the original interface resulting in material transfer from one surface to the other surface. The formation and rupture of adhesive junctions is accompanied by high local temperatures that can form reaction films on the newly formed surface and change the mechanical properties of the underlying material. "Adhesive wear" occurs when the adhesive transfer of material is the most important controlling mechanism.

3.2.6.4 Gross Surface Failure

Very high asperity penetration ($\lambda$ <<1), along with high asperity stress and high sliding speed ($U_s$) can result in catastrophic failure. Adhesive wear on a large scale is called scuffing (or scoring). This is a gross failure of the near-surface region. Scuffing is accompanied by a sudden rise in traction (friction) coefficient. Scuffing, which is only confined to roughness features, is called micro-scuffing. It results in sudden smearing of roughness features. If the smearing makes the surfaces smoother, the traction (friction) coefficient will subsequently decrease. Micro-scuffing can sometimes precede scuffing. Scuffing occurs on gear teeth at locations where the contact stress and sliding velocity are high. A similar mechanism in rolling element bearings is called "smearing." The precise mechanism of scuffing is not well understood, but it does involve the gross failure of the surface and is accompanied by a rapid increase in friction and contact temperature. A current view of scuffing is that under conditions yet to be defined it is a gradual breakdown in the lubrication of interacting asperities, the nature of which may be boundary, micro-EHD, or a mixture of the two lubricating mechanisms. Although the final scuffing mode may represent the gross breakdown of various lubrication films and the near-surface region, it may be triggered by the deterioration in surface topography as a result of adhesive wear or local plastic flow. While lubrication mechanisms affect scuffing, the scuffing event is a shear stability failure of the material in the near-surface region. Anti-wear and EP attributes of the oil serve to limit the tangential shear stress at the interface.

3.2.6.5 Chemical Reaction

The mechanistic processes associated with fatigue and plastic flow are the result of material deformation caused by stress. A significant part of wear and its control involves chemical reaction processes with the environment. The environment is defined as that portion of the contact system that is not an intrinsic part of the surfaces. The environment includes the surrounding atmosphere as well as the lubricating film.

Pure chemical reactions should be distinguished from "tribochemical" reactions, which are a consequence of these tribological interactions between the contacting surfaces. "Corrosion" results from reaction of the surfaces with the ambient environment under the prevailing ambient conditions. Tribochemical behavior is activated by mechanical interaction of the contacting surfaces. Corrosion often occurs on surfaces because of improper handling or storage resulting from the absence or removal of a protective film.

The prevention of adhesive wear is accomplished by forming tribochemical films. These films may be formed from oxygen in the atmosphere or from anti-wear or extreme pressure additives. "Tribochemical wear" generally involves a continuous process of surface film formation and removal. The formation process involves chemical reaction or adsorption of chemical species on the surface. The removal process results from mechanically induced crack formation and abrasion of the reaction products in the contact. The process introduces "clean," that is, activated, local areas where new tribochemical films can be formed and subsequently removed. The tribochemical process introduces thermal and mechanical activation of the near-surface region, which can cause (1) greater chemical reactivity as a result of increased asperity temperature and (2) changes in the microstructure and mechanical properties of the near-surface layer due to high local temperatures and mechanical working.

Under favorable operating conditions, tribochemical reactions may be associated with polishing wear or "mild wear." Mild wear is characterized by low wear rates and smooth surfaces frequently caused by oxidation of the surfaces and subsequent removal of the oxide. Unfavorable operating conditions can result in "severe wear," where the surfaces are extensively disturbed by gross adhesion and plastic flow rather than oxidative wear. Severe wear can be prevented by increasing the rate of chemical reactions to form protective surface films at the same rate as clean activated local areas are generated. In this way, a balance can be obtained between adhesive wear and "chemical wear." "Corrosive wear" is a term used when chemical wear dominates the adhesive wear mode by a wide margin. Additive composition, concentration, and temperature affect the rate of chemical wear. An optimum additive formulation is achieved when there is a balance between adhesive and chemical wear for a given degree of asperity penetration.



3.2.6.6 Plastic Deformation

Depending on the contact geometry, relative hardness and load, the shape of a contacting surface can be permanently deformed on both a macroscopic and microscopic scale as a result of plastic deformation. On a macroscopic scale the overload of rolling elements of a bearing can cause "Brinell marks" or distortion of the rolling track.

Almost all wear processes involve plastic flow on a microscopic scale. The plastic deformation that occurs from overrunning of hard particles, such as contaminants and wear debris, is "denting." "Plowing" occurs when there is displacement of material by a hard particle under the presence of sliding or combined rolling/sliding conditions. "Abrasive wear" occurs when the plastic deformation leads to material removal and wear debris.

General plastic deformation of asperities and ridges on rolling contact surfaces, such as bearings, is referred to as "surface distress" or at least the initial stages of surface distress. The final stages of surface distress involve the loss of material through micro-fracture (micro-pitting).

3.2.6.7 Surface Fatigue Processes

Reasonably high asperity penetration ($\lambda$ <1), giving high asperity stress with low sliding speed ($U_s$), are favorable conditions for surface initiated fatigue (micro-pitting). Fatigue is caused by cyclically repeated stresses on the contact surface, which eventually introduce permanent damage within the material. Damage begins as a crack. After repeated stress cycles, cracks can propagate and eventually lead to loss of surface material. Fatigue may initiate and propagate from the macro-stresses induced in the subsurface region. This results in "spalling" characterized by relatively large craters. Fatigue can also be initiated in the near-surface region as a result of micro-stresses from asperities or surface defects, such as dents, grooves, nicks and scratches. If the combined micro- and macro-stress fields propagate cracks through the quiescent zone and into the subsurface region, surface-initiated fatigue spalling can occur. Micro-pitting occurs when crack propagation is confined to the near-surface region. These processes are associated with the final stages of surface distress discussed above. The role of the lubricant is to reduce the local stresses at, and near, the surface to prevent the initiation of fatigue cracks. The effect of lubricant chemistry on surface initiated fatigue is associated with how the chemistry affects asperity stress. Surface films with good anti-wear attributes will preserve roughness features and asperity stress. Surface films which allow polishing wear, decrease asperity stress. Wear mechanisms almost always accompany surface initiated fatigue.

3.2.6.8 Comments on Lubrication and Failure Processes

The lubricant is an integral part of the contact system. The physical properties of viscosity, pressure-viscosity and traction coefficient affect oil film formation, its internal friction and heat generation. The lubricant film serves to reduce local stress and provides a separating film of low shear strength.

Lubricant chemistry forms surface films for boundary lubrication. They prevent local adhesion and disruption of the surface. The formation, removal and performance of these films are complex and very application dependent. The additives that produce these films are frequently classified as anti-wear or EP depending upon how they function toward reducing wear or preventing scuffing. Together the lubricant physical properties and chemical attributes determine the micro-stresses at the surface which must be accommodated by the contacting material surface properties to avoid failure mechanisms. The hydrodynamic processes are predictable and basic lubricant properties are used to predict performance. There is a rational basis for the selection of current values and desired values for future aircraft systems. While chemical attributes are complex, it is clear that chemical boundary film mechanisms can condition the surface to allow hydrodynamic and EHD mechanisms to be operative for long life with performance margin. The state-of-art tribology does not provide sufficient predictive capability for wear, scuffing or fatigue. Consequently, lubricant and bearing/gear material performance must rely on relevant testing. Aeropropulsion oil qualification relies on wear and scuffing testing. While testing advances have been made to provide a degree of performance assurance, there is still room for improvement. Tribological qualification tests for wear and scuffing provide a ranking or minimum performance level. There is no direct link to engineering design for wear, scuffing or fatigue. Progress toward linking oil formulation and tribological performance testing with engineering design may be accomplished through the utilization of the five identified key tribological parameters.

## 4. PROPERTIES AND ATTRIBUTES

Customary lubricant properties are listed below under two headings of physical and chemical:

PHYSICAL PROPERTIES

Kinematic Viscosity
Temperature-Viscosity Coefficient
Pressure-Viscosity Coefficient
Traction Coefficient
Surface Tension
Specific Heat or Heat Capacity
Thermal Conductivity
Density
Volatility
Pour point
Foaming
Bulk Modulus
Vapor Pressure
Electrical Conductivity
Miscibility

CHEMICAL PROPERTIES/ATTRIBUTES

Oxidative and Thermal Stability
Anti-Wear
Adhesive Wear
Scuffing (EP)
Surface Initiated Fatigue
Oil-off and Recovery
Corrosion

Some of the above properties are directly related to lubrication functions involving viscosity, anti-wear and EP performance. Others are indirectly related. Specific heat is a design consideration since the lubricant is used to remove heat as well as provide lubrication. Surface tension and density influence the supply and distribution of the lubricant within the mechanical components. Decomposition properties influence the ability of the lubricant to operate within a thermal environment as well as lubricate the components within that environment. Pour point is connected to low temperature starting ability. The lubricant serves multiple functions, including the removal of wear debris.

## 4.1 Properties Significance

The significance of lubricant properties can be revealed by observing their influence in a lubricated contact over a wide range of conditions such as load, speed and temperature. Additional insight can be obtained by observing the influence of lubricant properties as a function of film thickness. This approach helps to separate the role of the physical and chemical properties.

### 4.1.1 Properties in Hydrodynamic Lubrication

Hydrodynamic lubrication is the generation of a load supporting pressure by the motion of two surfaces, which pumps the lubricant into a convergent space. The theory for pressure generation, film thickness and lubricant flow is derived from Reynolds Equation where viscosity is the most important fluid property. The role of viscosity is seen in Equation 1 for film thickness assuming rigid cylinders and a viscosity independent of temperature or pressure.

$$\frac{h_o}{R} = \frac{U_e \mu_T}{F_N} \qquad \text{(Eq. 1)}$$



where:

$h_o$ = film thickness on the line of centers

$R$ = equivalent radius of curvature

$U_e$ = entraining velocity

$\mu_T$ = dynamic viscosity

$F_N$ = load

Since viscosity is directly related to film thickness, the level of viscosity can be selected for a given application to provide sufficient film thickness for complete surface separation. Since most applications are not constant speed, or constant load devices, lubricant composition decisions are made to select an appropriate level of viscosity over a range of conditions. In many cases design decisions are made to obtain the geometry, surface finish, material and load to allow a customer selected lubricant to operate successfully.

The process of generating a hydrodynamic film, results in power consumption and a temperature rise of the lubricant film. Both internal and external thermal effects can decrease the viscosity substantially. The rate of change of viscosity with temperature can be significant. The thermal effect on viscosity also varies with lubricant type.

The approach taken to model viscosity-temperature correlation has been mostly empirical. Most viscosity measurements are performed with capillary tubes in temperature baths. This measurement provides kinematic viscosity. The standard viscosity-temperature relation is the Walther equation [3] from which the ASTM D341-43 viscosity-temperature chart was formed:

$$\log\log(\eta_T + 0.7) = m \log T_K + C \qquad \text{(Eq. 2)}$$

where:

$\eta_T$ = kinematic viscosity as a function of temperature

$T_K$ = absolute temperature

$m$ = slope of ASTM viscosity-temperature chart

$C$ = constant

While many lubricants give reasonably straight lines on the ASTM D341-34 chart, the extrapolation capability at the extreme ends of temperature axis are not very accurate.

The slope m is an important lubricant property. The performance of a lubricant will improve, or at least be more predictable, if the sensitivity between viscosity and temperature is diminished. A common approach to the viscosity-temperature characterization of lubricants is the viscosity index (VI). VI is based on a comparison with two families of lubricants having widely different viscosity-temperature behavior. One is designated zero and the other is designated 100.

For most engineering calculations the dynamic, also called the absolute, viscosity is required. To convert the kinematic viscosity to the dynamic viscosity, and for various heat transfer calculations, the density of the oil is required. The density of the oil is nearly linear with temperature. A quadratic equation captures the slight curvature with temperature:

$$\rho = (C_1T^2 + C_2T + C_3) \quad \text{(Eq. 3)}$$

The dynamic viscosity can now be calculated as a function of temperature:

$$\mu_T = \rho\eta_T \quad \text{(Eq. 4)}$$

To ensure that accurate data is available for determination of the cooling and heat transfer qualities of a lubricant it is desirable to have density measurements at 15, 40, and 70 °C, heat capacity at 15, 40, 100, 150, and 200 °C and thermal conductivity covering 40, 100, 150, and 200 °C.

#### 4.1.2 Properties in Elastohydrodynamic Lubrication

High loads or concentrated contact geometries introduce elastic deformation into the hydrodynamic problem. A more important feature is the tremendous change in viscosity that results from the high pressures that are developed. The result is the rather remarkable load carrying mechanism of EHD lubrication. The lubricating film thickness formula derived from the Reynolds Equation for EHD conditions includes both elastic and viscosity-pressure parameters. The EHD lubricant film thickness has been solved by Crook, Dowson, and others. Today the most common version in use is the equation developed by Hamrock for both point and line contacts. A detailed description of this equation is given in Reference [5]:

$$h_{cen} = 2.69U^{0.67}G^{0.53}W^{-0.067}\left(1 - 0.61e^{-0.73k}\right)R_x \quad \text{(Eq. 5)}$$

$$h_{min} = 3.63U^{0.68}G^{0.49}W^{-0.073}\left(1 - e^{-0.68k}\right)R_x \quad \text{(Eq. 6)}$$

where:

$h_{cen}$ = central film thickness

$h_{min}$ = minimum film thickness

$$U = \text{dimensionless speed parameter} = \frac{\mu_T U_e}{E'R_x}$$

$$W = \text{dimensionless load parameter} = \frac{F_N}{E'R_x^2}$$

$$G = \text{Elasticity parameter} = E'\alpha$$

$$R_x = \text{equivalent radius} = \frac{r_1 r_2}{r_1 + r_2}$$

$$R_x = \text{equivalent radius} = \frac{r_1 r_2}{r_1 + r_2}$$

$$k = \text{ellipticity ratio} = \frac{a}{b}$$

$$E' = \text{materials parameter} = \frac{2}{\frac{1-\upsilon_1^2}{E_1} + \frac{1-\upsilon_2^2}{E_2}}$$

In contrast to the hydrodynamic problem where film thickness is directly proportional to viscosity, under EHD conditions the central lubricant film thickness is proportional to the lubricant properties of viscosity and pressure-viscosity raised to the power of 0.67 and 0.53, respectively. With a coefficient of -0.067, the above central film thickness equation indicates the lubricant film thickness is not very sensitive to load.

The interaction between pressure and lubricant viscosity is important to the success of an EHD lubricating film. The variation of viscosity with pressure is usually described by an exponential relation proposed by Barus, Equation 7.

$$\mu_{T,p} = \mu_T e^{\alpha p} \quad \text{(Eq. 7)}$$

where:

$\mu_{T,P}$ = viscosity as function of temperature and pressure

$\mu_T$ = viscosity as a function of temperature

$\alpha$ = pressure-viscosity coefficient

p = pressure

This equation works reasonably well for predicting film thickness provided that the variation in the pressure-viscosity coefficient with temperature is taken into account.

Direct pressure-viscosity measurements can be made in specialized viscometers that apply pressure to the lubricant and measure the time of a sinker to fall in the pressurized oil. In practice these measurements are difficult to make. Another method is to indirectly obtain the pressure-viscosity coefficient from EHD film thickness measurements using optical interferometry. The pressure-viscosity coefficient $\alpha$ is then derived from EHD film thickness theory. This resulting pressure-viscosity coefficient is an “effective” $\alpha$.

Measured pressure-viscosity data from optical EHD measurements for a number of 5 cSt qualified products are shown in Figure 5. The data in Figure 5 is interpolated and extrapolated to nominal temperatures. Measurements are made at discrete temperatures close to the nominal temperatures of 40, 70, 100, 130 and 150 °C. The pressure-viscosity coefficient varies due to blends with different molecular structures.

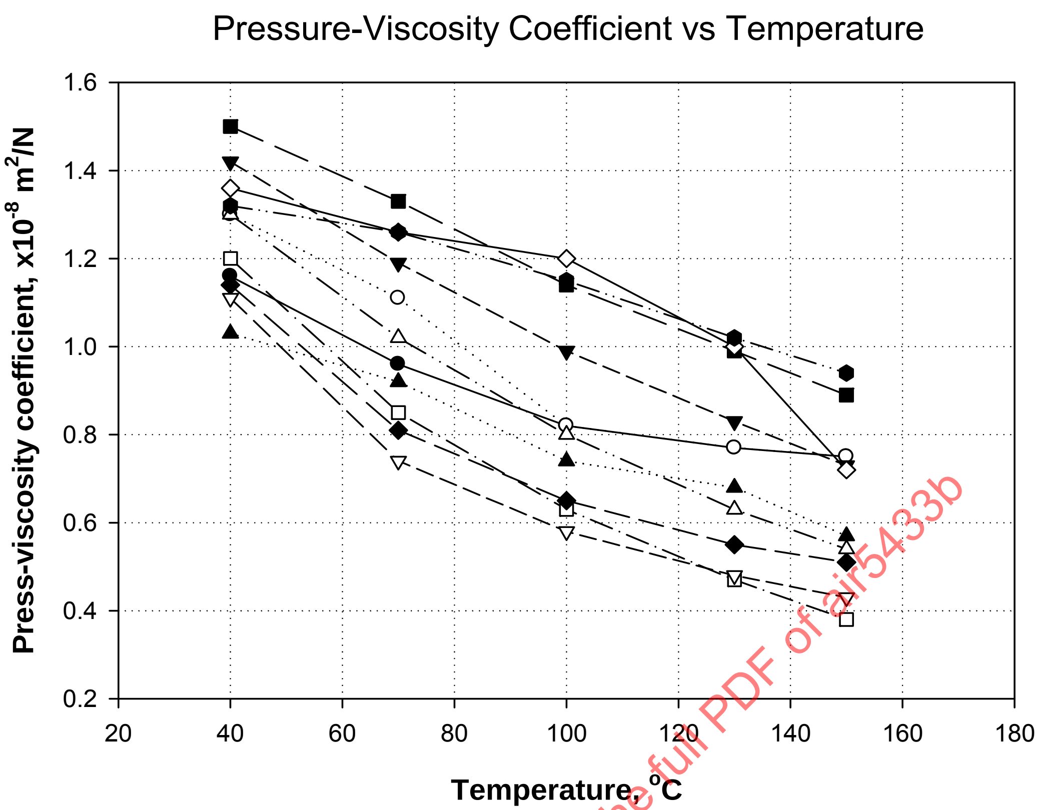


FIGURE 5 – PRESSURE-VISCOSITY COEFFICIENTS FOR 5 CST QUALIFIED PRODUCTS

For engineering purposes it is convenient to average the pressure-viscosity coefficient to obtain a nominal value for 5 cSt qualified products. It is also useful to be able to predict the pressure-viscosity coefficient as a function of temperature. This can be done using Equation 8. A comparison of the averaged data and the fit obtained with Equation 8 is shown in Figure 6. For the averaged data shown here, the coefficient f = 2.795 x $10^{-3}$ and g=-2.142 when $T_K$ is expressed in degrees Kelvin.

$$\alpha = f\,T_K^g \qquad \text{(Eq. 8)}$$

where:

$\alpha$ = pressure-viscosity coefficient

f = regression coefficient obtained from experimental data

g = regression coefficient obtained from the experimental data

$T_K$ = Absolute temperature

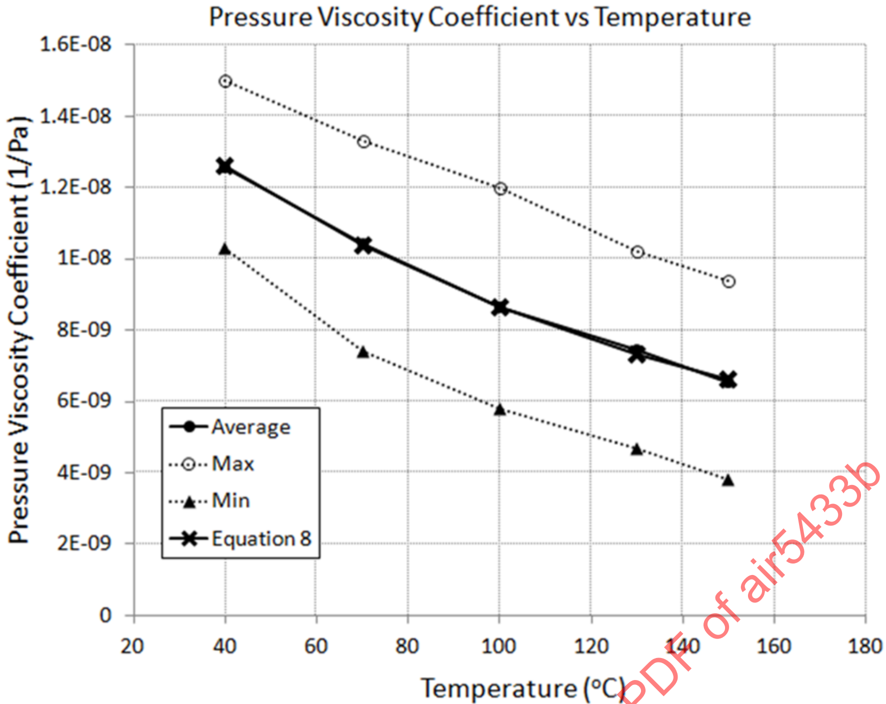


FIGURE 6 – AVERAGE PRESSURE-VISCOSITY COEFFICIENT REPRESENTING 5 CST QUALIFIED PRODUCTS

There are several lubricant related factors that can cause the film thickness to be less than the EHD film thickness equation predicts. The inlet region to the contact (see Figure 2) does not require very much lubricant, but the inlet boundary must be sufficiently upstream of the contact to avoid starvation. The location of this inlet boundary is influenced by the surface tension and the density of the lubricant. Starvation results in a thinner film thickness.

The fluid molecules also see a great deal of shear in the inlet region due to rapidly moving surfaces and a converging geometry. The maximum oil shear stress occurs at a location within the inlet region where the film thickness is approximately $2h_o$. This is close to the "heart" of the pressure generating inlet region. Any non-Newtonian fluid behavior may influence film thickness. Finally, the viscous flow, and backflow, in the inlet region can cause significant heat generation within the lubricant film. While the heat is quickly dissipated, this thermal effect is sufficient to reduce the film thickness, particularly at high speeds. Film thickness calculations in high speed components must be adjusted with an inlet heating thermal correction. An equation for the thermal correction developed by Herb Cheng is shown in Equation 9 and described in more detail in Reference [6].

$$\Phi_t = \frac{1+4.15PL^{0.42}}{1+0.241(1+2.42S^{0.83})L^{0.64}} \qquad \text{(Eq. 9)}$$

$$L = \frac{\beta \mu_T U_e^2}{K_f}$$

$$P = \frac{P_{max}}{E'}$$

$$\beta = \frac{-\ln\left({}^{\mu_T}/_{\mu_o}\right)}{T - T_o}$$

where:

$\Phi_t$ = Thermal reduction factor

S = Slide to roll ratio = $U_s/U_e$

$\beta$ = Temperature viscosity coefficient

E' = Combined elastic modulus $1/E' = [\,(1-\nu_1^2\,)/\,E_1 + (1-\nu_2^2)/\,E_2]/\,2$

$P_{max}$ = Maximum Hertzian stress

$K_f$ = Thermal conductivity of the oil

$\mu_T$ = Dynamic viscosity at temperature T

$\mu_o$ = Reference viscosity at room pressure and at the reference temperature $T_o$

The thermally corrected film thickness equation can now be rewritten as:

$$h_{cen,t} = h_{cen}\Phi_t \qquad \text{(Eq. 10)}$$

Where $h_{cen,t}$ is the thermally corrected central film thickness and $h_{cen}$ is the isothermal central film thickness calculated in Equation 5. A comparison of the film thickness obtained from both solutions is shown in Figure 7. The geometry and operating conditions used to calculate the dimensilonless parameters in Equations 5, 9, and 10 were dervied from an engine bearing at typical engine operating conditions. The relative speed of an engine bearing has a noticeable thermal effect in the calculated film thickness value. The effect of viscosity can also be seen. Additional details of the calculation are provided in the Appendix.



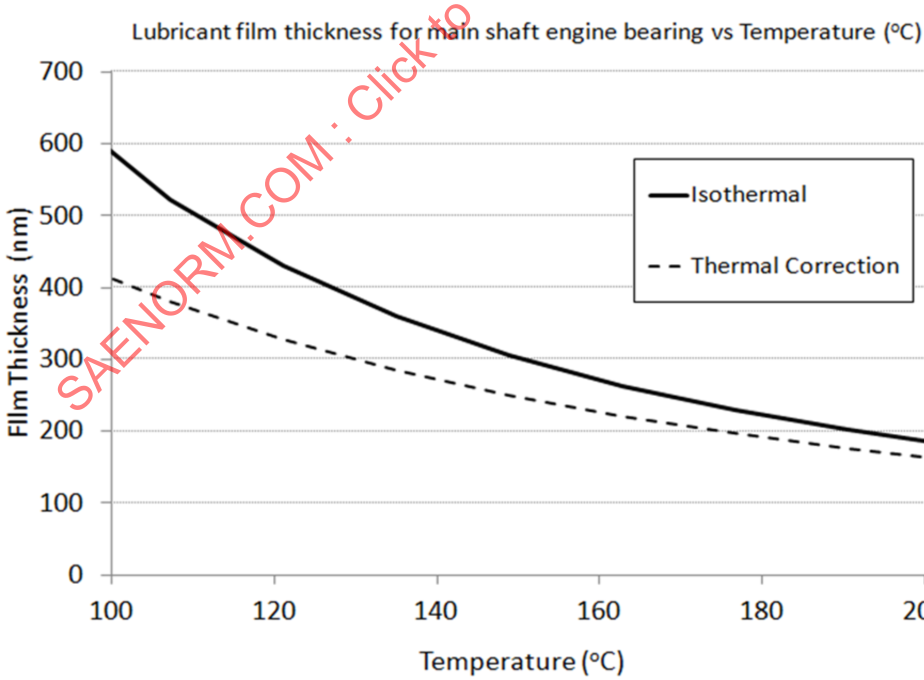


FIGURE 7 - COMPARISON OF THE ISOTHERMAL AND THERMAL SOLUTION FOR THE LUBRICANT FILM THICKNESS IN A GAS TURBINE ENGINE BEARING AS A FUNCTION OF TEMPERATURE

4.1.3 Properties for EHD Traction

Under EHD conditions, the inlet region is primarily responsible for the generation of lubricant film thickness. The Hertzian region (see Figure 2) is primarily responsible for the friction, or traction, of the lubricated contact. The high pressure in the Hertzian contact causes the lubricant to become pseudo-solid. If there is relative motion between the surfaces, the solid-like lubricating material must accommodate shear between the surfaces. Under incipient sliding conditions, where the sliding velocity between the surfaces is very small compared to the rolling velocity, the lubricant behaves like an elastic solid. As the sliding velocity is increased, the lubricant will shear like a highly viscous fluid or like a plastic solid. This causes the traction between the surfaces to reach a maximum at some level of sliding. Depending on contact temperature, the maximum traction, or traction coefficient (load/traction force), is usually reached within a few percent slip. Slip is defined as the sliding velocity $U_s$ divided by the entraining velocity, $U_e$. Under high pressure, the traction coefficient reflects the limiting shear strength of the lubricant. The traction within the contact is a major cause of heat generation in bearings and gears.

Traction coefficients vary with molecular structure. Typical polyol esters used in aeropropulsion systems have similar, but not exactly the same traction coefficients. Formulated oils may differ by approximately 10%. The effect of pressure and slip on traction coefficient is shown in Figure 8.

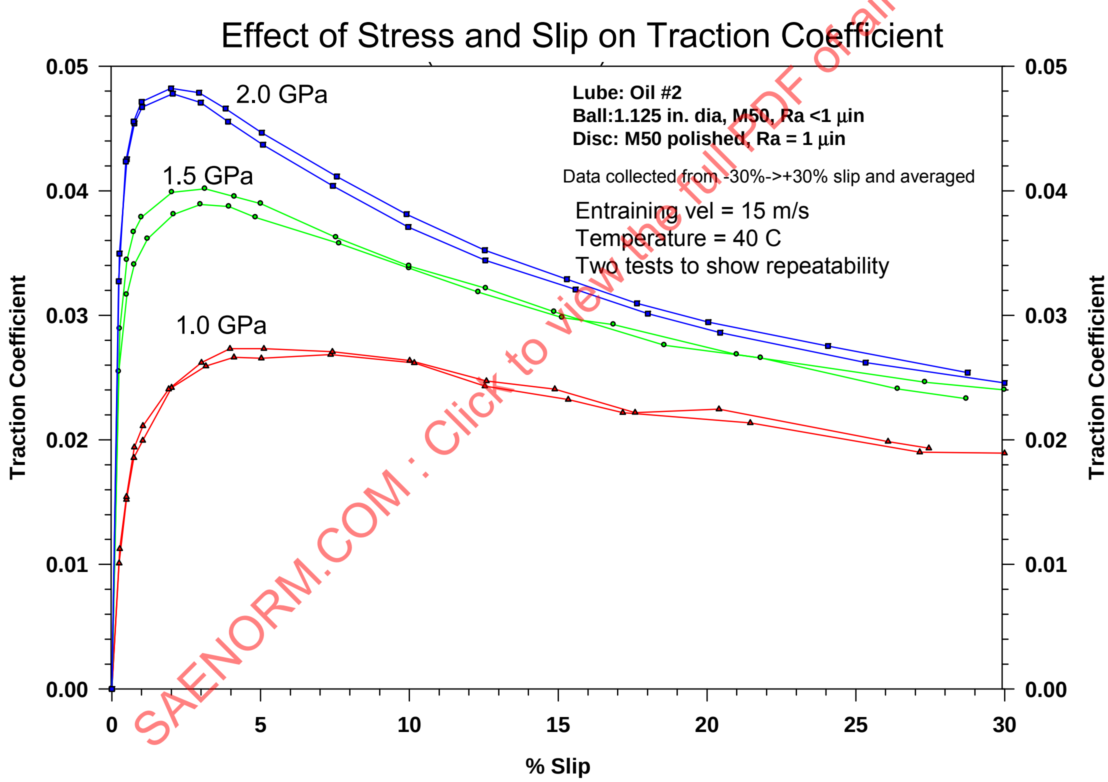


FIGURE 8 - EFFECT OF STRESS AND SLIP ON TRACTION COEFFICIENT

At 40 °C the traction coefficient reaches a maximum at less than 5% slip. The drop off in traction is due to thermal effects where frictional heating reduces the shear strength of the interposed pressurized fluid. Rolling element bearings generally operate in the low slip region. Gear contacts can cover a large range of slip as the contact moves along the active profile of the gear tooth.

Contact temperature has a significant effect on traction coefficient. This is shown in Figure 9.

Elevated temperatures substantially reduce the traction coefficient. Elevated temperatures also cause the maximum traction coefficient to occur at high values of slip. This has a significant impact on heat generation in high speed rolling element bearings, particularly thrust load angular-contact ball bearings. The lubricated contact becomes much more efficient at elevated temperatures.

While attempts have been made to predict traction coefficients from high pressure shear experiments, the most reliable methods for traction measurements are rolling/sliding experiments covering operating conditions in service. The traction data is then used for bearing thermal analysis.

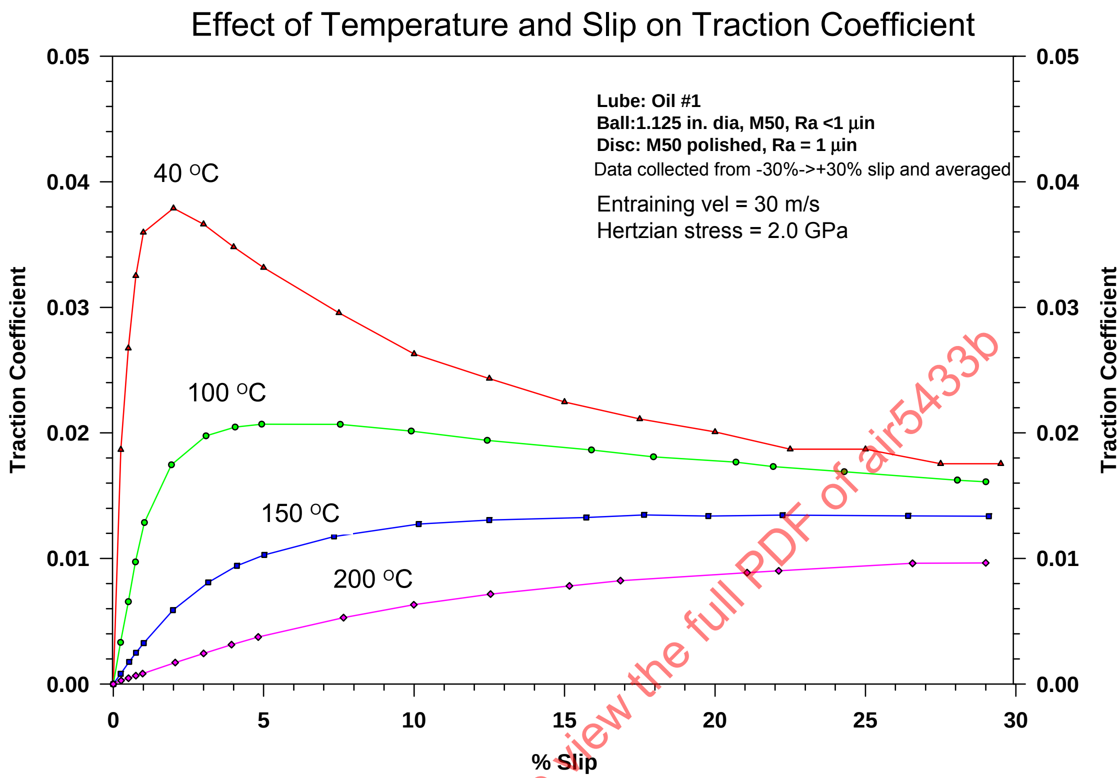


FIGURE 9 - THE INFLUENCE OF TEMPERATURE AND SLIP ON TRACTION COEFFICIENT

4.1.4 Properties for Boundary Lubrication

Lubricating films generated by hydrodynamic mechanisms can totally separate surfaces to prevent wear and control friction. Boundary lubrication occurs under conditions where there is significant surface interaction. It is characterized by the following:

a. The lubrication mechanisms are highly complex; involving metallurgical effects, surface topography, physical and chemical adsorption, corrosion, catalysis and reaction kinetics.

b. The primary function of boundary lubrication is the formation of surface films to minimize wear and surface damage.

c. The formation of surface films is controlled not only by lubricant chemistry, but also its interaction with the surfaces, wear debris and the environment.

d. While the chemical makeup of boundary lubricating surface films can be obtained, very little is published about the critical physical properties of these films. These properties include shear strength, thickness, surface adhesion, film cohesion, melting point or decomposition temperature and solubility.

Some boundary lubricating films are thick enough to be visible as shown in Figure 10.

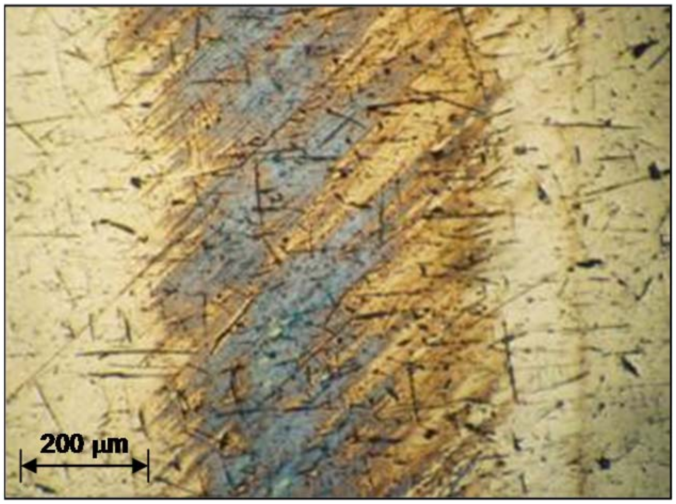


FIGURE 10 - BOUNDARY FILM FROM ANTI-WEAR ADDITIVE TRICRESYL PHOSPHATE (TCP)

Thin, transparent, boundary films may appear colored as a result of light interference. The color changes from light-brown to blue as the thickness increases.

The fundamental physical and chemical properties of boundary lubricating films are mostly too complex to be of practical predictive value for performance. Thin film properties within the transient contact are difficult to measure. The practical approach is to measure the effects of boundary films and how these control friction, wear and catastrophic failure. These measurements characterize the anti-wear and EP performance of the lubricated contact system. The challenge is to make measurements under meaningful conditions that have linkage to engineering parameters and service operation. Careful simulation of user hardware is necessary to reliably asses the performance of boundary lubricating films.

4.1.5 Mixed Film Lubrication

There are not many applications that continuously operate on full hydrodynamic films or purely on boundary lubrication films. In most cases, hydrodynamic and EHD mechanisms operate along with boundary lubrication mechanisms. The former carries the chemistry for the latter to work; and the latter protects the surfaces so that the former can continue to operate. The joint actions of the physical and chemical mechanisms seem to be a significant factor in performance. Lubricant property significance therefore can only be evaluated when both types of mechanisms are allowed to operate as they would in service.

An example of the interaction of both boundary and EHD mechanisms is shown in Figure 11 for a MIL-PRF-23699 aviation oil with and without the anti-wear additive Tricresyl Phosphate (TCP). A bearing steel lubricated concentrated contact is forced to transition from a full-film EHD lubrication mode into mixed-film lubrication by increasing the temperature. The oil without TCP encounters polishing wear immediately upon entering the mixed-film lubrication regime. The polishing of roughness features, resulting in smoother surfaces, allows the low shear strength EHD film to control the traction (friction). However, at 200 °C the surfaces fail by scuffing due to the absence of a protective boundary film to maintain the integrity of the surfaces.

The fully-formulated MIL-PRF-23699 oil with anti-wear additive displays a higher traction coefficient due to surface film protection of roughness features. It does not encounter polishing wear until it is well into the mixed-film lubrication regime. The boundary films preserve the surface integrity and no scuffing failure is observed. Note that while entering the mixed-film lubrication regime with increasing temperature the traction coefficient continues to decrease. This is a result of lower shear strength of the EHD film with temperature (see Figure 9). Boundary films also decrease in shear strength with temperature. The traction coefficient does not increase until scuffing failure.

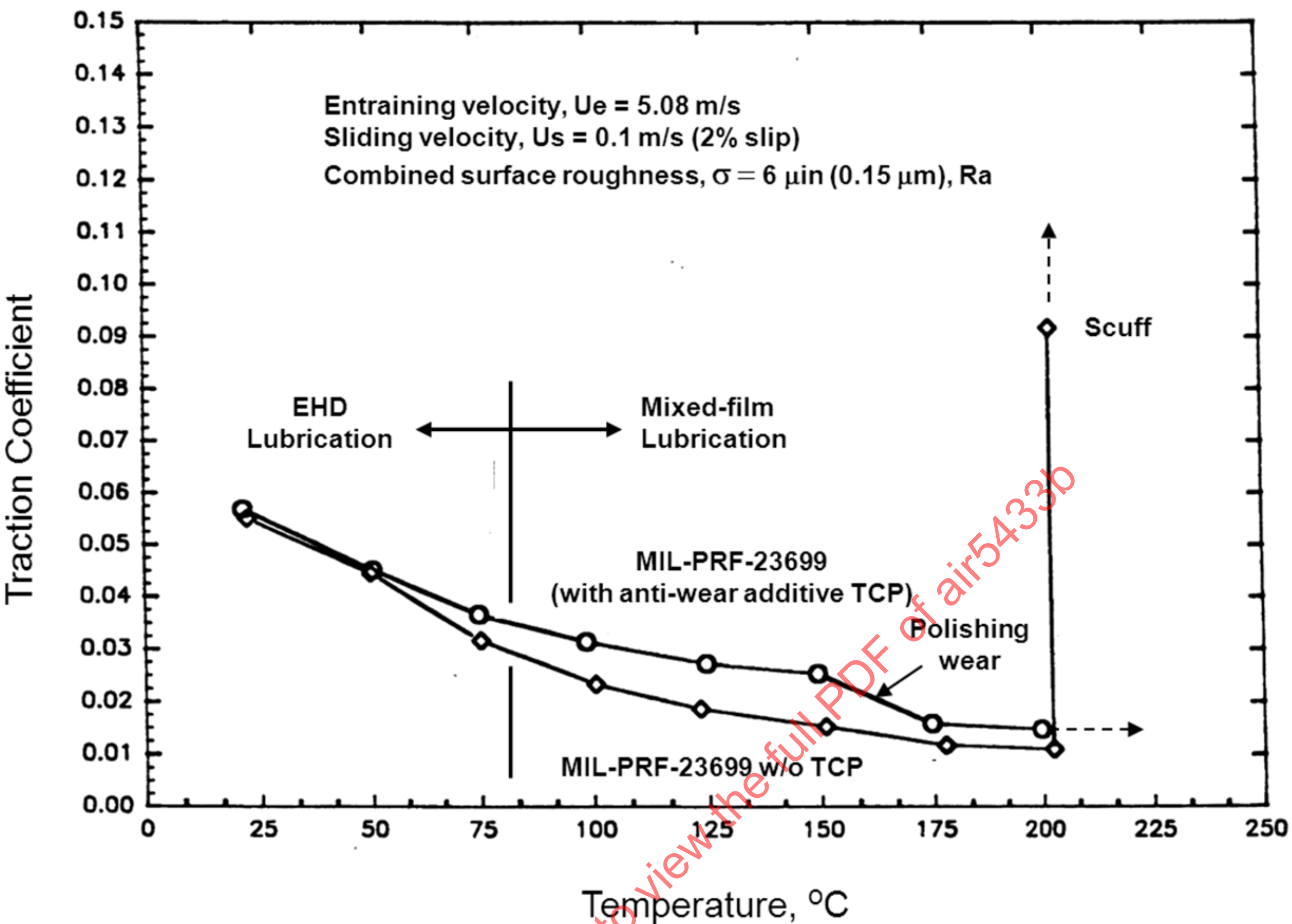


FIGURE 11 - TRANSITION FROM EHD TO MIXED FILM LUBRICATION



The role and significance of both hydrodynamic/EHD and surface film mechanisms can be shown by driving a lubricated contact into the mixed-film lubrication regime from a number of directions. Figure 12 is a performance map obtained with a MIL-PRF-23699 oil. The performance map is plotted in terms of two of the five key tribological parameters, the entraining velocity ($U_e$) and the sliding velocity ($U_s$). The entraining velocity generates an EHD film and the sliding velocity generates a thermal environment that leads to reduction of film thickness and eventually toward a scuffing failure.

The performance map identifies three regions of operation: (1) EHD region where physical properties of the lubricant completely separate the surfaces; (2) a mixed-film lubrication region where both the physical and chemical properties are sufficient to prevent scuffing; and (3) a scuffing region where the lubricant properties, both physical and chemical, are no longer sufficient of prevent scuffing or severe wear.

The size of the mixed-film region characterizes the lubricating capacity of the oil formulation to operate under marginal film thickness conditions. Anti-wear, and particularly EP, additives affect the size of the mixed-film region. It is important to recognize that the contacting materials, heat treatment and surface processing also affect the lubricating capacity. The oil and the load-bearing materials are integral parts of the lubricated contact system.

For lubricant/material development and qualification, it is convenient to utilize simplified tests that transition through the lubrication and failure regions shown in Figure 12. One such test is a load capacity test shown in Figure 13 where traction coefficient is plotted against increasing load. Each increment in load results in a new load stage. The resultant Hertzian stress from increasing applied load covers a range from 0.7 to 2.4 GPa.

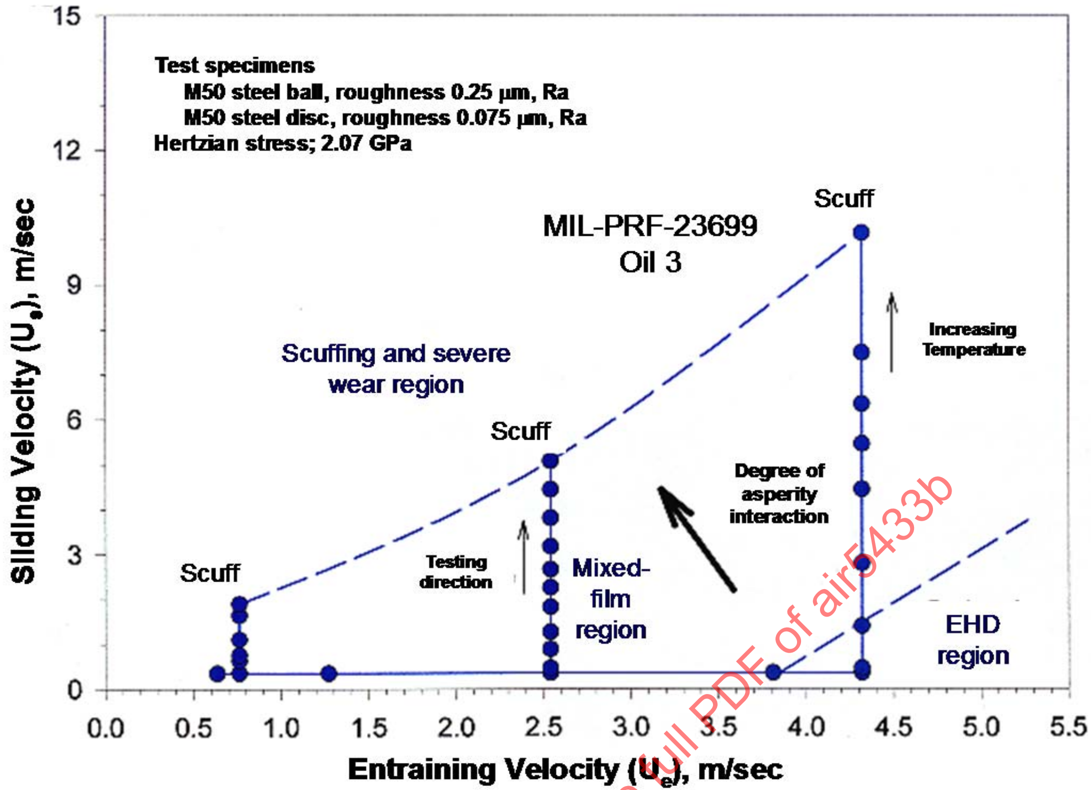


FIGURE 12 - PERFORMANCE MAP IN TERMS OF ENTRAINING AND SLIDING VELOCITY



A load capacity test operates at constant entraining velocity ($U_e$) and sliding velocity ($U_s$). The load, which is changed in discrete stages, increases the temperature and reduces the EHD film thickness until severe wear or scuffing occurs. The test conditions in Figure 13 simulate the Ryder Gear Test at the location where scuffing occurs on the gear mesh. The test conditions give nearly full-film EHD lubrication at the first load stage. Higher load stages with accompanying heat generation drive the lubricated contact deeper into the mixed-film region.

The traction coefficient and scuffing load stage for four oils plotted in Figure 13 show the effect of oil formulation on anti-wear and scuffing performance. The base stock oil is the same for each formulation. Only the additive package is changed. A high traction coefficient reflects good anti-wear performance. The anti-wear surface films preserve the roughness features resulting in higher friction within the contact. Low traction coefficient reflects polishing wear. The vertical arrows identify the average load stage where scuffing occurs. The results in Figure 13 show the dramatic effect additive packages have on anti-wear and scuffing performance. The oil formulations plotted show a trade-off between anti-wear and scuffing performance. High scuffing resistance may come at the expense of some anti-wear performance, but this is not always the case. Formulations for both scuffing and anti-wear performance can be achieved.

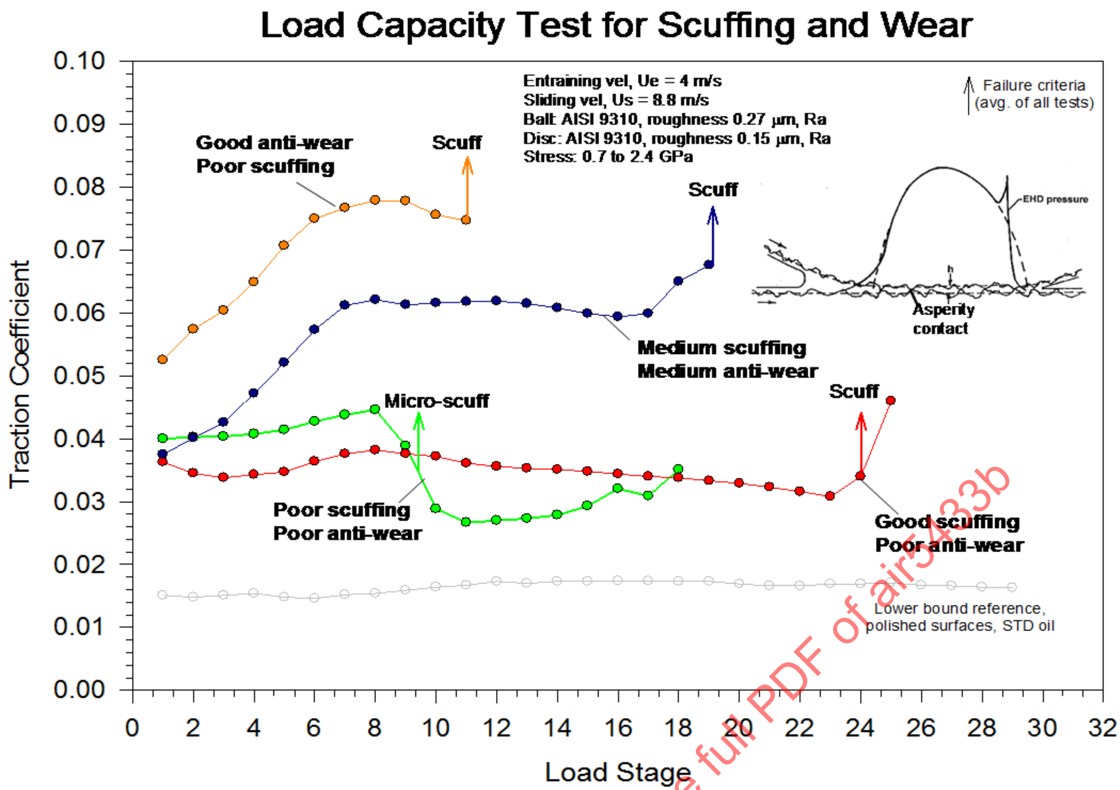


FIGURE 13 – LOAD CAPACITY TEST SHOWING SCUFFING AND WEAR PERFORMANCE

## 5. COMMONLY USED LUBRICANT PROPERTIES FOR AIRCRAFT ENGINE AND POWER SYSTEMS

### 5.1 Background

The properties presented in this section are those related to the lubrication function, i.e., those which directly or indirectly play a role in the performance of the fluid in the contact area of lubricated components.

Several physical properties of the base stock contribute to the formation of the film that separates surfaces in relative motion. In this respect, viscosity is the major physical property related to the lubrication function. For example, viscosity-temperature index, despite having an influence on fluid film thickness, is governed by the low temperature flow requirements of the overall system. Other physical properties which are used in determining a lubricant's film strength are not specified only for their contribution to lubrication but are a consequence of the need for other characteristics (e.g., thermal stability) of the base stock and the economics of producing the fluid. Although, usually not specified, these properties are not expected to vary very much for products qualified to a particular specification because the overall requirements restrict the base stock to similar chemistry. Because these properties are important to designers, they are included in the following tables. Typical values obtained by particular methods are presented.

The "chemical properties" of the lubricant are those which play a major role in establishing load carrying capacity or the performance of the lubricant in the absence of a separating film. This characteristic is very difficult to assess universally because it is not an inherent property of the lubricant itself. It is a manifestation of the interaction between the lubricant additives and the metallurgy of the contacting surfaces.

5.2 Properties and Test Methods for Lubricants Used in Aircraft Propulsion Systems

The tables provided in this section deal with synthetic lubricants intended primarily as engine and/or gearbox lubricants for aircraft propulsion or power generating systems.

Based upon a past survey of the SAE E-34 Aerospace Lubricants Committee members, the tables list commonly used lubricant properties and test methods that are used to determine these properties. The data presented are divided into properties that have direct influence on the lubricant characteristics that affect the contact zone and other properties that influence the overall lubrication system design. Included in these tables is a list of lubricant specifications that were defined as being used in current aircraft propulsion systems along with typical property values. These lists are not intended to be complete but do specify the data provided during the survey. The intent of this section is to provide a list of lubricants, properties, and typical values of these properties that could be used for either the selection of a lubricant or the selection of the property value that can be used for various lubrication calculations.

TABLE 1 - LUBRICANT PROPERTY SUMMARY FOR AIRCRAFT PROPULSION SYSTEMS
KINEMATIC VISCOSITY

ASTM D445 for measurements at -40 °C, 40 °C, and 100 °C is preferred and estimate by ASTM D341 at 200 °C required.

USES: EHL calculations, lubricant flow to lubricated components, heat generation, oil pressure drop and cage drag calculations.

TEST METHODS: ASTM D 445/IP71
ASTM D 341

SYNTHETIC LUBRICANTS:

| | Temperature °C/(°F) | Specification cSt | Typical Range cSt |
|---|---|---|---|
| MIL-PRF-7808L Grade 3 | -51/(-60) | 17 x $10^3$ max | 7106 - 7741 |
| | 40/(104) | 11.5 min | 11.6 - 12.9 |
| | 100/(212) | 3.0 min | 3.04 - 3.23 |
| Grade 4 | -51/(-60) | 20 x $10^3$ max | 17,280 - 19,704 |
| | 40/(104) | 17.0 min | 17.61 - 17.80 |
| | 100/(212) | 4.0 min | 3.95 - 4.07 |
| | 204/(400) | 1.1 min | 1.13 - 1.23 |
| MIL-PRF-23699F | -40/(-40) | 13,000 max | 7,800 - 11,500 |
| | 40/(104) | 23 min | 24.0 - 27.5 |
| | 100/(212) | 4.9 - 5.4 | 5.0 - 5.3 |
| DOD-L-85734 | -40/(-40) | 13,000 max | 8,649 - 9,970 |
| | 37.8/(100) | 25 min | 27.35 - 29.50 |
| | 98.9/(210) | 5.0 - 5.5 | 5.13 - 5.40 |
| DEF STAN 91-100/101 | -40/(-40) | 13,000 max | 7,532 - 11,965 |
| | 40/(104) | 30 max | 24.0 - 27.53 |
| | 100/(212) | 4.9 - 5.4 | 4.97 - 5.378 |

REMARKS:

For calculation of hydrodynamic and elasto-hydrodynamic film thickness, the kinematic viscosity of the lubricant must be converted to dynamic viscosity by multiplying it by the specific gravity of the lubricant. Therefore, when viewing lubricants of different density for film forming ability, direct comparison of kinematic viscosity could lead to considerable error.

## TABLE 2 - LUBRICANT PROPERTY SUMMARY FOR AIRCRAFT PROPULSION SYSTEMS TEMPERATURE-VISCOSITY DATA

USES: EHL Calculations

TEST METHODS: ASTM D2270 (Determine Slope within Nominal Temperature Range)

SYNTHETIC LUBRICANTS:

Grade 5 oils

| | | HPC A | SPC A | HPC B | SPC B | SPC C | HPC C | HPC D | SPC D |
|---|---|---|---|---|---|---|---|---|---|
| Kinematic Viscosity, cSt | Temperature | | | | | | | | |
| | -40°C | 11421, 11450 | 11260, 9351 | 12125, 12303 | 7850, 7362 | 10200, 10400 | 11016, 12190 | 9584, 9540 | 9399, 9067, 8162 |
| | 40°C | 24.8, 25.0 | 26.82, 26.87 | 26.81, 26.59 | 26.27, 25.79 | 25.3, 25.1 | 29.7, 27.5 | 25.9, 25.8, 25.9 | 25.6, 25.86, 27.03 |
| | 100°C | 4.91, 4.93 | 5.24, 5.26 | 5.22, 5.20 | 5.06, 5.00 | 5.0, 4.9 | 5.3, 5.3 | 5.2, 5.2, 5.1 | 5.13, 5.17, 5.16 |
| | 200°C | 1.35, 1.4 | 1.45 | 1.38 | 1.36 | | 1.4 | 1.38 | 1.49 |



REMARKS:

ASTM Slope is the slope of the kinematic viscosity vs. temperature curve as plotted on an ASTM graph The ASTM graph is based on an equation relating kinematic viscosity (η) with temperature according

$$\text{Log log } (\eta_T + 0.7) = m \log T_K + C$$

where $T_K$ is the absolute temperature and m is the ASTM slope

Since the rate of decrease in viscosity with temperature is inversely proportional to the ASTM slope, the smaller the slope, m, the better the lubricant is able to retain a level of viscosity as the temperature rises. For a given class of lubricant, the ASTM slope tends to increase with viscosity, i.e., high viscosity lubricants tend to be more sensitive to temperature than low viscosity lubricants. For this reason, selection of lubricants for viscosity should always be based on the viscosity predicted at the expected service temperature of the lubricant. For calculating hydrodynamic or EHD film thickness, the kinematic viscosity must be converted to dynamic viscosity. This is accomplished by multiplying the kinematic viscosity of the lubricant by the specific gravity of the lubricant. For hydrodynamic lubrication, lubricant film thickness is directly proportional to viscosity. For contacts under high stress, with elastohydrodynamic (EHD) lubrication, lubricant film thickness is proportional to the viscosity raised to the power 0.67

## TABLE 3 - LUBRICANT PROPERTY SUMMARY FOR AIRCRAFT PROPULSION SYSTEMS PRESSURE-VISCOSITY COEFFICIENT

USES: Elastohydrodynamic film thickness calculations

TEST METHODS: Data calculated from optical film thickness measurements
Pressure-Viscosity Coefficient Method ARP6157 (draft)

SYNTHETIC LUBRICANTS:

| | Temperature °C | Range $x10^{-8}m^2/N$ | Avg. Value $x10^{-8}m^2/N$ |
|---|---|---|---|
| MIL-PRF-7808 | 40 | | 0.61 |
| 3 cSt | 70 | | 0.44 |
| | 100 | | 0.30 |
| | 130 | | (0.21) extrapolated |
| MIL-PRF-7808 | 40 | | 0.86 |
| 4 cSt | 70 | | 0.54 |
| | 100 | | 0.39 |
| | 130 | | (0.28) extrapolated |
| AS5780 (MIL-PRF-23699) | 40 | 1.03 - 1.50 | 1.26 |
| 5 cSt (representing 11 oils) | 70 | 0.74 - 1.33 | 1.04 |
| | 100 | 0.58 - 1.20 | 0.87 |
| | 130 | 0.47 - 1.02 | 0.74 |
| | 150 | 0.38 - 0.94 | 0.66 |
| DO-PRF-85734 | No data, similar to 5 cSt oils | | |
| DEF STAN 91-100/101 | No data | | |



REMARKS

Measured data is interpolated to nominal temperatures shown in Table 3.
Concentrated contacts with high Hertzian stress generate sufficient pressure within the lubricated inlet region to cause an exponential rise in viscosity with pressure. The relationship between pressure and viscosity is frequently described by the Barus equation:

$$\mu_{T,p} = \mu_T e^{\alpha p}$$

where:

- $\mu$ = viscosity at pressure p
- $\mu_T$ = viscosity at atmospheric pressure
- $\alpha$ = pressure-viscosity coefficient
- P = pressure

Pressure viscosity coefficients can be measured using high-pressure capillary viscometers, which are limited to low shear conditions. Because elastohydrodynamic oil film generation occurs under high-shear conditions the preferred method is to measure the actual oil film thickness in a lubricated contact using high precision optical interferometry. The “effective” pressure-viscosity coefficient is then derived from a commonly used EHD film thickness equation (Hamrock – Dowson). The values obtained can then be used for engineering calculations using the same Hamrock – Dowson equation.

EHD film thickness is proportional to the pressure-viscosity raised to the power of 0.53. Errors in pressure-viscosity coefficient have a lesser effect on the calculated film thickness. Aviation Polyol ester lubricants have median values of pressure viscosity coefficients compared to other molecular structures which give higher or lower values. Pressure-viscosity coefficients tend to increase with molecular weight (viscosity) and decrease with temperature. Within the class of qualified 5 cSt oils, pressure-viscosity coefficients are found to vary slightly depending on base stock blend or perhaps additive formulation.

## TABLE 4 - LUBRICANT PROPERTY SUMMARY FOR AIRCRAFT PROPULSION SYSTEMS LOAD RATING (WEAR AND SCUFFING)

Uses: To provide a minimum performance level and a database for wear and scuffing attributes of qualified products; and to support component design

Test Methods: Ryder Gear (FED-STD-791 Method 6508), (ASTM D1947), ARP 6255 AIR4978, (ARP6156 draft)

SYNTHETIC LUBRICANTS:

| Oil Type Specification | Ryder Gear Specification | Ryder Gear – Typical |
|---|---|---|
| MIL-PRF-7808 | | |
| Grade 3 | 2200 ppi (ASTM D1947) | 2570 – 2749 ppi |
| Grade 4 | 2200 ppi (ASTM D1947) | 2550 ppi |
| AS5780<br>MIL-PRF-23699 | 102% of reference (min.) relative to Herco-A at 74 °C (165 °F) | 109 – 132% of ref. relative to Herco-A at 74 °C (165 °F)<br>2715 – 3245 ppi |
| DOD-PRF-85734 | 145% of reference (min.) relative to Herco-A at 74 °C (165 °F) | 145% of reference relative to Herco-A at 74 °C (165 °F) |

| Oil Type Specification | WAM Specification* | WAM – Typical |
|---|---|---|
| AS5780 | Load Stage 15 (min.) (ARP6156 draft) | Scuffing load stage 16 – 28 |

| Oil Type Specification | ALTE Specification | ALTE – Typical |
|---|---|---|
| AS5780 | ARP 6255 | tbd |

REMARKS:
Historically, lubricating performance has been evaluated with gear tests, such as the Ryder Gear Test Method (FED-STD-791 Method 6508). Gear tests provide a scuffing load rating reflecting the viscous elastohydrodynamic film generating and chemical boundary film-forming capability of the oil formulation. Because of cost and consistent gear availability, alternate test methods were developed (AIR4978). Two alternate meth ods are currently recognized, the WAM High Speed Load Capacity Test Method (ARP6156 draft) and the Aviation Lubricant Tribology Evaluator (ALTE) (ARP6255). These test methods are designed to rank the wear/scuffing performance of reference oils according to the Ryder Gear method. While these methods have proven to screen out oils with limited performance in service, they do not provide a comprehensive evaluation of lubricating performance covering wear, scuffing and surface initiated fatigue. Additional test methods are required for linking tribological tests with design.

* The minimum scuffing load stage of 15 is derived from a body of qualified 5 cSt oil products with satisfactory historical performance in service. Load stage 15 is equivalent to a maximum Hertzian contact stress of 1.54 GPa. Historical testing with a 5 cSt STD Ref Oil results in an average scuffing load stage of 26.7 (2.1 GPa). Due to difficulties in purchasing and processing AISI 9310 gear steel to give scuffing performance equivalent to vintage material this test method, at least temporarily, includes reference oil testing to normalize results. The minimum load stage of 15 is equivalent to 56.2% of the 5 cSt STD Ref Oil.

## TABLE 5 - LUBRICANT PROPERTY SUMMARY FOR AIRCRAFT PROPULSION SYSTEMS TRACTION COEFFICIENT

USES: Bearing and gear thermal analysis

TEST METHOD: WAM ball and disc rolling/sliding traction method

SYNTHETIC LUBRICANT: 5 cSt candidate (typical for 5 cSt qualified oils)

| Hertz Stress | Bulk Temp. | Contact Slip % | Traction Coef. | Max Trac Coef | Initial Slope |
|---|---|---|---|---|---|
| 1.0 | 40° C | 1 | 0.0100 | 0.0194 | 0.0101 |
| | | 2 | 0.0166 | | |
| | | 5 | 0.0188 | | |
| | | 10 | 0.0175 | | |
| | | 30 | 0.0132 | | |
| | 100° C | 1 | 0.0016 | 0.0110 | 0.0015 |
| | | 2 | 0.0029 | | |
| | | 5 | 0.0056 | | |
| | | 10 | 0.0083 | | |
| | | 30 | 0.0110 | | |
| | 150° C | 1 | 0.0011 | 0.0063 | 0.0038 |
| | | 2 | 0.0011 | | |
| | | 5 | 0.0013 | | |
| | | 10 | 0.0030 | | |
| | | 30 | 0.0063 | | |
| | 200 ° C | 1 | 0.0017 | 0.0051 | 0.0066 |
| | | 2 | 0.0015 | | |
| | | 5 | 0.0013 | | |
| | | 10 | 0.0015 | | |
| | | 30 | 0.0051 | | |

| Hertz Stress | Bulk Temp. | Contact Slip % | Traction Coef. | Max Trac Coef | Initial Slope |
|---|---|---|---|---|---|
| 1.5 GPa | 40° C | 1 | 0.0260 | 0.0305 | 0.0101 |
| | | 2 | 0.0304 | | |
| | | 5 | 0.0283 | | |
| | | 10 | 0.0233 | | |
| | | 30 | 0.0161 | | |
| | 100° C | 1 | 0.0051 | 0.0151 | 0.0051 |
| | | 2 | 0.0086 | | |
| | | 5 | 0.0129 | | |
| | | 10 | 0.0150 | | |
| | | 30 | 0.0145 | | |
| | 150° C | 1 | 0.0011 | 0.0111 | 0.0011 |
| | | 2 | 0.0024 | | |
| | | 5 | 0.0050 | | |
| | | 10 | 0.0078 | | |
| | | 30 | 0.0111 | | |
| | 200 ° C | 1 | 0.0005 | 0.0075 | 0.0006 |
| | | 2 | 0.0009 | | |
| | | 5 | 0.0020 | | |
| | | 10 | 0.0038 | | |
| | | 30 | 0.0075 | | |

| Hertz Stress | Bulk Temp. | Contact Slip % | Traction Coef. | Max Trac Coef | Initial Slope |
|---|---|---|---|---|---|
| 2.0 GPa | 40° C | 1 | 0.0360 | 0.0379 | 0.0748 |
| | | 2 | 0.0379 | | |
| | | 5 | 0.0332 | | |
| | | 10 | 0.0263 | | |
| | | 30 | 0.0175 | | |
| | 100° C | 1 | 0.0129 | 0.0207 | 0.0126 |
| | | 2 | 0.0175 | | |
| | | 5 | 0.0207 | | |
| | | 10 | 0.0202 | | |
| | | 30 | 0.0161 | | |
| | 150° C | 1 | 0.0033 | 0.0135 | 0.0032 |
| | | 2 | 0.0059 | | |
| | | 5 | 0.0103 | | |
| | | 10 | 0.0127 | | |
| | | 30 | 0.0134 | | |
| | 200 ° C | 1 | 0.0008 | 0.0097 | 0.0008 |
| | | 2 | 0.0017 | | |
| | | 5 | 0.0040 | | |
| | | 10 | 0.0063 | | |
| | | 30 | 0.0097 | | |

The traction test data in the above table was conducted with a 28.6 mm diameter ball contacting a flat disc and operating with an entraining velocity of 30 m/sec. The traction coefficient is an inherent and fundamental property of the oil under Hertzian contact conditions. It is primarily a function of stress and temperature and it is almost independent of film thickness under typical thin film conditions. Traction coefficient increases from zero at pure rolling to a maximum as slip within the contact increases. As the sliding velocity increases, the traction coefficient becomes affected by heat generation within the contact. The data in the above table represent s effects due to sliding velocities based on a percentage of the rolling velocity of 30 m/sec. Traction coefficients at higher or lower entraining velocities may be slightly different, particularly at the higher values of percent slip. Additional traction data may be required for accurate values at higher or lower entraining velocities.

While the above data was collected with a “point contact” configuration, tests conducted with an elliptical contact give essentially the same results.

See Figures 8 and 9 for typical traction coefficient test plots.

TABLE 6 - LUBRICANT PROPERTY SUMMARY FOR AIRCRAFT PROPULSION SYSTEMS
SPECIFIC HEAT

ASTM E1269 is the preferred method.

USES: Determine heat exchanger cooling requirements and conduct heat transfer analysis.

SYNTHETIC LUBRICANTS:

| | Temperature °C | Typical Value J/g °C | Std Deviation |
|---|---|---|---|
| AS5780 SPC | 15 | 1.825 | |
| | 40 | 1.896 | 0.098 |
| | 100 | 2.037 | 0.044 |
| | 150 | 2.167 | 0.044 |
| | 200 | 2.307 | 0.045 |
| AS5780 HPC | 15 | 1.770 | |
| | 40 | 1.838 | 0.047 |
| | 100 | 2.026 | 0.062 |
| | 150 | 2.178 | 0.054 |
| | 200 | 2.308 | 0.055 |