EXPERIMENTAL INVESTIGATION
OF SPUR GEAR EFFICIENCY

A Thesis

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By

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ABSTRACT

An experimental investigation of high-speed spur gear efficiency was conducted for both jet-lubricated, dry sump conditions and dip-lubricated conditions. Inspection methodologies were developed for the documentation of gear surface roughness and wear after each test. An experimental test matrix including gears of two different modules and surface roughness levels operating under jet-lubrication conditions with four different gear lubricants was developed to quantify the influence of these parameters on load-dependent (mechanical) and load-independent (spin) power losses.

The spur gear efficiency test machine was modified for dip-lubricated load-independent power loss measurements, allowing direct comparison to jet-lubricated conditions using the same test fixtures. An experimental test matrix including unity ratio gears of different module and face width operating in an oil bath of four different levels for a range of rotational speed and oil viscosity was developed. The influence of rotational speed, oil viscosity, oil bath level, and rotational direction on load independent power loss was quantified.

Dedicated to my parents

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NOMENCLATURE

<u>Symbol</u>	<u>Definition</u>
b	gear face width, mm
d	diameter, mm
h	oil level height, mm
m	module, mm
M	moment, N-m
n	number of samples
Ω	rotational speed, rpm
P	power Loss, kW
r	radius, mm
R	surface Roughness, μm
s	sample standard deviation
T	torque, N-m
W	load, N
μ	coefficient of friction
ρ	density, kg/m^3
σ	standard error
ω	angular velocity, rad/s

<u>Subscript</u>	<u>Definition</u>
a	average
b	bearing
bore	bore
c	closed power circulation loop
churn	churning
drag	fluid drag
inertia	acceleration of impinging oil jet
L	loaded
m	mear mesh
mech	mechanical
p	pitch
r	rolling
s	sliding
seal	shaft Seal
T	total
Trap	oil and air squeezed from between gear teeth as they mesh
$\bar{X}$	sample mean

CHAPTER 1

INTRODUCTION

1.1 Background and Motivation

Efficiency of spur and helical gear systems has become an increasingly important research topic as the demand for increased power density and fuel economy for today's passenger vehicles and rotorcrafts are more stringent. Increased requirements on efficiency are not only due to fuel cost, but also due to environmental concerns associated with energy sustainability and air pollution. Improved gear system efficiency also results in less frictional heat generation within the gearbox, improving on several gear failure modes such as scoring and pitting, and requiring a lower-capacity lubrication system, which can lead to improved power density of the geared system. Primary sources of efficiency losses in transmissions include tooth friction in the gear mesh, bearing power loss, and various load-independent power losses stemming from the interactions of rotating components with lubricating oil and air, and frictional losses of various oil seals.

Previous theoretical and experimental studies on gear efficiency focused primarily on friction loss in the gear mesh or overall power loss of geared systems, but little

research has been dedicated to investigation of individual components of power loss in the same system. Theoretical studies focused on prediction of friction coefficient in the gear mesh, which was then applied to predict gear mesh mechanical efficiency. Only a few of these models were experimentally validated. Experimental studies were generally dedicated to obtaining an average friction coefficient for the theoretical models or defining influences of operating conditions or a specific parameter on efficiency. Design parameters such as module and face width, surface roughness parameters and manufacturing processes, and lubricating oil were also considered in these studies under low to medium speed and load conditions. Recent advancements in gear manufacturing processes have made it possible to apply low friction coatings to gear surfaces or reduce gear tooth average surface roughness amplitudes significantly. However, very little experimental data is available on the influence of these processes on gear efficiency.

Given the limited volume of experimental data, there is a great need for an extensive database for spur gear efficiency, not only to validate the mathematical models but also to quantify experimentally the influence of gear design parameters, manufacturing process, and lubricant and surface related parameters on spur gear efficiency. Also missing in the literature is experimental data on efficiency of spur gears under high-speed (pitch line velocities to 50 m/s) and high-torque conditions. Development of such a database through tightly controlled experiments is one of the main goals of this study.

Most of the previous studies on spur gear efficiency did not refer to other sources of power loss, especially load independent (spin) losses. Modifications to the gearbox to minimize mechanical power loss might indeed increase spin loss, resulting in larger

overall power loss. Therefore, it is critical that both mechanical and spin components of gear power loss are considered simultaneously so that overall gear system efficiency can be maximized. Previous efforts in defining these load independent losses generally employed empirical mathematical models for air drag or lubricant churning losses on rotating components based on results from simple bench tests. Most of these models fail to consider the effects of the meshing gear and other sources of power loss present in a gearbox, and thus cannot be validated using geared transmissions. The second goal of this study is to generate experiment data that compares spin and mechanical losses of the same gear sets under a range of operating conditions for various lubricants.

Existing models for spin losses can be sorted in two categories, windage and churning. Windage power loss refers to drag of a disc or gear rotating in air, or an oily atmosphere, and is assumed to apply to dry sump applications in which the gears are not in contact with an oil bath. Churning models refer to viscous drag of a rotating disc or gear that is partially submerged in oil. These models are intended to apply to dip-lubricated conditions. Existing windage power loss models assume that the gears are fully enveloped in air, while the churning models assume that the only source of drag is from the oil sump. Currently, no published model is available that considers both sources of power loss together. Since any dip-lubricated gear system will simultaneously contain both windage and oil churning power losses, especially when only a small percentage of the gear surface area is submerged, there is a need for a comprehensive spin power loss study to handle both windage and oil churning losses. The third goal of this study is to develop an experimental test methodology for both jet-lubricated and dip-lubricated spur

gear spin power losses, and develop an experimental database that can be used in the future to validate spin power loss models.

1.2 Literature Review

A large number of theoretical studies have been published on the efficiency of gear trains [1-16]. These models typically focused only on the friction related mechanical efficiency losses in the gear mesh for spur and in some cases, helical gears. As a mechanical efficiency model requires a friction coefficient at gear contact interfaces, these models differed mainly in terms of the formulations used to determine the friction coefficient. A group of spur gear efficiency models [1-4] used a uniform and constant friction coefficient μ along the entire contact surface gears. The normal force acting at a given contact point and this constant μ was used to calculate a tangential friction force in the direction of sliding, and the reduction in torque transmitted was used to calculate the mechanical efficiency η of the gear pair. Another group of models [5-10] adapted the same approach with the exception that their formulation for μ was not constant, but was based on various empirical formulae obtained from twin-disk tests.

With the concern that these models are as good as their empirical μ formula, another group of studies aimed at eliminating the need for μ formulae by using elastohydrodynamic lubrication (EHL) models of varying complexity. Several studies [11-13] considered smooth surface EHL formulae to determine the surface shear stress distribution caused by the fluid film, from which instantaneous friction coefficient can be calculated. Rough surface EHL models were employed by Wu and Cheng [14],

Mihalidis *et al* [15], and more recently by Xu *et al* [16]. The model of Xu *et al* [16] incorporated a gear load distribution model, a friction coefficient model, and a mechanical efficiency formulation, which resulted in agreement with experimental measurements to within 0.1% in terms of gear mesh mechanical efficiency. These models had the advantage of eliminating the need for prior knowledge of μ , at the expense of significantly more computational demand. While these models were sophisticated in handling EHL aspects of the problem, the application was limited to simple spur gears with ideal load distributions and no tooth bending deformations. More importantly, they did not consider other sources of power loss in a gearbox, and were therefore insufficient to predict overall gearbox efficiency.

While the body of theoretical work on spur gear mechanical efficiency is extensive, the same cannot be said for the experimental treatment of the problem. Most experimental studies have been limited to estimating average gear mesh friction coefficient under low to medium speeds and correlating this to gear operating characteristics such as sliding velocity, torque transmitted, and lubricating oil [17-26]. Early studies by Naruse *et al* [17-19] sought to define influences of certain gear design parameters, manufacturing process, lubricating oil, surface roughness and operating conditions on spur gear efficiency. Naruse and Haizuka [17] demonstrated on a disk-machine that friction coefficient for lubricated rolling-sliding contacts decreases with sliding velocity, and is not clearly dependent on oil viscosity. In the case of geared systems, Naruse *et al* [18-19] showed from measurements with a power re-circulating loop bench test that spur gear mesh efficiency increases with rotational speed for pitch-line velocities below approximately 24 m/s, and also reported that gear mesh frictional

power loss increases with transmitted torque at low torque values, but becomes nearly constant at higher loads. Naruse *et al* [19] showed that reducing module can significantly influence efficiency losses at the gear mesh. In these tests, reducing module from 6 mm to 2 mm reduced gear mesh efficiency losses from nearly 1.5% to 0.5%. In this same study, it was shown that ground gears of 4.5 mm module operated at approximately 0.25% lower efficiency losses than shaved gears with the same module and similar surface roughness. These studies also showed that efficiency losses are insensitive to changes in surface roughness between 0.5 μm and 3 μm for high load, and low to medium speed conditions. However, the influence of gear surface roughness over a wider range of roughness values, operating conditions, and lubricating oil viscosity has not been studied in detail. Yoshizaki *et al* [20] demonstrated that below 10 cSt, gear tooth friction coefficient decreases as lubricating oil viscosity increases, but friction coefficient changes little as viscosity increases beyond that point. Ikejo and Nagamura [21] also concluded from measurements with a similar test setup that gear mesh frictional loss is weakly dependent on oil viscosity, but decreases with increasing viscosity.

Conversely in terms of total power loss, Coy *et al* [22] reported that the overall efficiency of a helicopter transmission generally decreased with increasing lubricating oil viscosity, indicating that load-independent power losses can be significant in practical applications. Handschuh and Kilmain [23] observed from power loss measurements of a jet-lubricated high-speed helical geartrain that overall efficiency increases with transmitted torque, and decreases with rotational speed. It was shown that at rotational speeds of 5,000 to 15,000 rpm, the influence of transmitted torque on total power loss

was less significant than the influence of rotational speed, but efforts were not made to characterize loaded and unloaded power losses separately.

Martins *et al* [24] measured average friction coefficient of dip-lubricated FZG gears and reported that friction coefficient decreases with rotational speed, and increases with transmitted torque for pitch-line velocities less than 19 m/s. Traction measurements of combined rolling-sliding contacts of Xiao *et al* [25] suggested that gear mesh power loss is lower for smooth surfaces than rough surfaces, but few experimental studies have been pursued to quantify this. One such study by Britton *et al* [26] reported that reduction of average surface roughness height from 0.4 to 0.05 μm through an isotropic chemical polishing technique reduced mesh power loss by between approximately 20 and 30%. The range of operating conditions considered for this study were Hertzian contact pressures ranging from 0.4 to 1.3 GPa, pitch-line velocities up to 22 m/s, and operating temperatures of 50 to 100°C. While these test conditions represent a wide range of applications, it is desirable to expand this investigation to higher Hertzian pressures and pitch-line velocities in order to encompass the high-performance applications that are more likely to take advantage of a chemical polishing process.

Most of the previous studies on spur gear efficiency did not refer to secondary sources of power loss, especially load independent (spin) losses. Several experimental studies [5, 21, 30-31] have shown that spin losses, primarily due to windage and/or oil churning, can be rather significant, especially under high-speed operating conditions. Therefore, it is critical that both mechanical and spin components of gear efficiency losses to be considered simultaneously in an effort to maximize efficiency.

Existing empirical spin power loss models can be categorized into windage or oil churning models. The windage models of Dawson [27-28] and Diab *et al* [29] were obtained by measuring the deceleration of a single gear or disc in air on a spindle, then applying the kinetic energy theorem to obtain power loss from air drag. Diab *et al* [29] employed dimensional analysis to define a windage moment coefficient in terms of angular velocity, fluid density and viscosity, Reynolds Number, face width, pitch radius, number of gear teeth, and nearby fluid flow obstructions such as close-fitting gear case walls. While applicable to a single disc or gear rotating in air, these models did not consider the effects of a meshing gear or impinging oil jet, and so cannot be validated using geared transmissions. As an alternative, Anderson and Loewenthal [5] developed a model for meshed spur gears based on pitch radius, face width, rotational speed, and viscosity of the ambient fluid. These models all resulted in a nearly cubic relationship between rotational speed and power loss.

Ariura *et al* [30] identified sources of load-independent power loss in a spur gear system as (1) the inertial power loss to accelerate the impinging oil jet by the rotating gear teeth, (2) the power to trap and squeeze oil from between gear teeth as they meshed (trapping and squeezing), (3) the power to induce an air draft around the gear (windage), and (4) the churning power loss of the lubricant in an oil bath. Torque measurements of Ariura *et al* [30] using a jet lubricated spur gear system led to the observation that inertial power losses are linearly related to rotational speed. Ariura *et al* [30] also proposed a simple model to estimate the power loss due to oil trapping between the gear teeth. Comparison to experimental results indicated that the model qualitatively captured trends

in the data, but the magnitude of calculated power loss was consistently lower than measured values.

Mizutani [31] reported from measurements of high-speed, long addendum spur gears that windage power loss was proportional to a power of rotational speed of 2.28 and the inertial losses resulting from the impinging oil jet were linearly proportional to rotational speed. Here, the inertial losses from the impinging oil jet were shown to increase with oil jet pressure, and composed a significant portion of load-independent power loss.

An early analytical theory on the fluid flow of air and lubricant around jet-lubricated spur gear teeth was proposed by Akin and Mross [32]. This model elucidated the phenomenon of eddy current flow between gear teeth and introduced a vectoral model for penetration depth of impinging fluid on the rotating gear teeth. Later, Pechersky and Wittbrodt [33] introduced a computational method for the trapping and squeezing of fluid between meshing gear teeth. The model suggested that the volume and velocity of the fluid squeezed between gear teeth is higher for coarse-pitch gears than fine-pitch gears. Most recently, Diab *et al* [34] developed a numerical model to analyze fluid trapping and squeezing in the case of windage by defining a matrix of intertooth volumes and flow restrictions, and numerically solving the equation of continuity for each discretized volume. Analytical results compared favorably to experimental data from pressure transducers placed at the roots of spur and helical gear teeth in a test rig operated without lubricating oil. In the case of helical gears, power loss values calculated by the model indicate that trapping and squeezing power loss decreases as helix angle increases. This

result can have significant implications in the realm of improving efficiency if it can be validated with in-situ spin power loss measurements.

The study of oil churning power loss has received much attention in the past, but primarily for the rotation of a single disc or bladed rotor in a fluid [35-36]. Like Diab *et al* [34] for the case of windage, the oil churning studies were mostly devoted to developing empirical equations for the dimensionless moment coefficient. Daily and Nece [35] proposed the existence of four different flow regimes around a rotating disc fully submerged in fluid, and correlated these flow regimes to Reynolds Number and enclosure effects based on experimental results. Mann and Marston [36] studied friction drag of bladed and unbladed discs, and related experimental results to moment coefficient based on Reynolds Number and axial clearance with the chamber. In the case of dip-lubricated gear applications Terekhov [37] developed empirical relations for dimensionless moment coefficient from numerous experiments on gears rotating partially submerged in a fluid, and identified differing power loss equations for meshed gears rotating upward or downward in an oil bath. Boness [38] conducted friction torque tests with a simple bench setup on smooth discs of various diameter and width partially submerged in high-viscosity oil, and compared these results to results with a gear. Boness' results [38] varied widely from Terekhov [37] in predicted friction torque for similar operating conditions.

More recently, Höhn *et al* [39] proposed a simpler calculation method for churning friction torque. Luke and Olver [40] investigated the discrepancy between the results of Terekhov and Boness, conducting numerous bench tests of their own with unmeshed and mating nylon and steel gears of varying module, face width, and number

of teeth. Luke and Olver [40] concluded that the models of Terekhov [37], Boness [38], and Höhn *et al* [39] were all inadequate in predicting their own measured churning power loss values. Luke and Olver [40] challenged the previous assumption that dimensionless moment coefficient can be related to Reynolds Number for dip lubricated gear systems. Most recently, Changenet and Velez [41] completed an extensive experimental churning power loss study, and proposed a new model that includes the effect of the mating gear depending on the sense of rotation. As in previous studies, empirical formula for dimensionless moment coefficient were developed using dimensional analysis.

The dimensionless moment coefficients calculated by the models of Terekhov [37], Boness [38], Höhn *et al* [39], and Changenet and Velez [41] have all been shown to agree well with data obtained on the test rigs from which the models were developed, but poor agreement is achieved when comparing moment coefficients to those calculated by other models under similar test conditions [40]. A universally applicable, physics-based churning power loss model still eludes researchers.

1.3 Scope and Objective

The existing body of experimental gear efficiency research illuminates useful trends in spur gear efficiency for a wide range of gear designs and operating conditions. However, the influences of recent refinements in manufacturing techniques such as isotropic chemical polishing and low-friction coatings have yet to be studied in detail. Additionally, experimental data for high-power gearing applications, is not readily available in the literature. Even less data is available on load-independent (spin) power

loss of jet lubricated and dip lubricated gearboxes. Lastly, existing semi-empirical models for both lubrication conditions fail to agree well with experimental data.

The main purpose of this study is to establish a spur gear efficiency database through tightly-controlled gear efficiency experiments. Experimental data will be generated to compare spin and mechanical power losses of the same gear sets under a range of operating conditions for various lubricant types and lubrication methods. Specific objectives of this study are as follows:

- Develop a test methodology that will be capable of measuring power losses of a high-speed spur gear pair operating under a wide range of torque.
- Apply this test methodology to describe trends in spur gear mesh and total efficiency under various operating conditions.
- Investigate the influences of gear module, manufacturing finishing processes such as grinding and shaving, surface finish technologies such as chemical polishing and thin-film coatings, and lubricant type on gear mesh and total power loss.
- Quantify load-independent spin power loss for the same test conditions as loaded power loss measurements.
- Develop a test methodology for dip lubricated churning power loss of spur gear pairs, and quantify the influence of lubricant viscosity, oil level and rotational direction on churning power loss.
- Provide a detailed comparison between the power losses associated with jet-lubricated and dip-lubricated gear sets.

1.4 Thesis Outline

Chapter 2 provides a description of the high speed gear efficiency test machine and test plan utilized in this study. Inspection procedures for the test gears are outlined and inspection results are presented. The test specimens and test matrix are described, and the purpose of each test is noted. The accuracy and repeatability of the experimental test machine over the duration of the project are presented.

Chapter 3 presents results of the efficiency experiments under jet-lubricated conditions. Sources of power loss in a gearbox are discussed, as well as the methods utilized to separate certain power losses using the experimental test machine. Measured load-dependent and load-independent power loss results are presented. Heuristic arguments are made for the influence of certain design parameters and operating conditions on components of gear power loss. Influences of the experimentally investigated parameters are discussed in terms of average total and gear mesh mechanical efficiency.

Chapter 4 describes modifications made to the experimental test machine to study dip-lubricated spin power loss. The accuracy and repeatability of the experimental test machine for this mode of operation is presented. The sources of load-independent power loss in a jet-lubrication and dip-lubrication environment are outlined. Experimental results for the influence of several gear design parameters, and operating conditions are presented. Lastly, chapter 5 summarizes the accomplishments of this study, and presents the main conclusions drawn from the experimental results. Finally, recommendations for future work are offered.

CHAPTER 2

EXPERIMENTAL TEST METHODOLOGY

2.1 Description of Gear Efficiency Test Machine

The spur gear efficiency test machine used in this study, shown in Fig. 2.1, was designed and procured as part of an earlier spur gear efficiency investigation [42]. A brief description of the test machine will be provided here with only the most relevant features included. Readers are referred to Chase [42] for details of the test machine and its operation.

The test machine consists of two opposing identical gearboxes, each containing one gear mesh, four identical cylindrical roller bearings, and shaft seals. This machine utilizes a four-square, power circulation concept such that one gear from each gearbox is connected to the corresponding gear of the other gearbox. Mounted on one of the flexible shafts connecting the gearboxes is a split coupling that is used to apply a constant torque T_c to the system. This torque is transmitted by both test gear meshes. Figure 2.2 shows a schematic of the test machine layout. The two gearboxes are specified “test” and

Figure 2.1 The spur gear efficiency test machine.

Figure 2.2 A schematic representation of the spur gear efficiency test machine.

“reaction” to differentiate them, but there is no physical difference except for the input shaft on the reaction gearbox. A flexible coupling connects the input shaft of the reaction gearbox to a high-speed spindle through a high-precision, non-contact type torque-meter. The high-speed spindle is driven by a belt with a 3:1 ratio speed increase from a variable speed AC motor. A separate temperature-controlled lubricant circulation system supplies directed oil jets to the gear meshes and bearings. Oil temperature is measured with type-J thermocouples at the supply points as well as return from the gearboxes. The test apparatus shown in Fig. 2.2 sits on a slide mechanism, allowing the splined input shaft to be easily disengaged from the coupling, so that the power circulation loop torque T_c can be safely changed and any electronic drift of the torque-meter can be checked and documented between tests.

The test machine is capable of operating at rotational speed up to $\Omega = 10,000$ rpm, which corresponds to 48 m/s gear pitch-line velocity for unity ratio gears with a fixed center distance of 91.5 mm. With a circulation loop torque T_c up to 680 N-m (500 ft-lb), circulation loop power $P_c = T_c \omega$ ($\omega = \frac{2\pi}{60} \Omega$) of over 700 kW (950 hp) is possible, making the machine capable of running at representative conditions for commercial automotive, racing and some aerospace applications.

The test machine is almost entirely computer controlled and monitored through a LabView interface. Rotational speed, lubricant temperature, and test duration are all controlled through the LabView interface. Oil flow rates are controlled manually with a needle valve on a machine-mounted oil supply manifold. The computer interface permits a number of diagnostics and shutdown limits to be set according to test parameters.

Limits on oil temperature, oil pressure, rotational speed, and driving torque can be specified by the user. Additionally, the data logging period and test parameter tolerances can be adjusted.

2.2 Test Procedure

Most of the high-speed gear efficiency tests are conducted at an oil supply temperature of 110°C or 90°C depending on the gear application represented. Therefore, the lubricant must be heated to this set operating temperature prior to beginning an efficiency test. Lubricant is circulated throughout both gearboxes to bring their temperatures to the desired temperature level. Test duration and rotational speed are set through the LabView interface.

The gear efficiency test machine is mechanically loaded through the split coupling with a torque arm and deadweights as shown in Fig. 2.3. Once the desired T_c is achieved, friction discs in the split coupling are tightened, locking the torque in to the closed loop formed by the two gear pairs and the shafts connecting them. Torque arm angle is measured with a line level attached to the arm and with a compass. This measurement is used to correct for small changes in torque arm length due to its angle, so that applied T_c can be determined precisely. The deadweights and torque arm are then removed, and the test machine is readied for testing.

New gears, shaft seals, and bearings typically have a run-in period during which they exhibit elevated levels of frictional power loss. During this period, gear and bearing surface roughness decrease slightly, and seals wear-in with the shaft. At the beginning of

Figure 2.3 Static load application of the power circulation loop.

this test plan, it was thought that gear run-in time should drive the run-in cycle. For this reason, a run-in cycle formed by one hour of operation at $T_c = 413$ N-m and $\Omega = 6,000$ rpm is used. This run-in cycle was applied to new gears and bearings. An additional run-in cycle of two hours of operation at $T_c = 0$ and $\Omega = 6,000$ rpm was used for run-in of the new shaft seals.

Previous experimental studies on gear efficiency using the same machine indicated that most transient temperature behavior is eliminated in the first five to six minutes of a test [42]. Accordingly, ten minutes was selected as the test duration, with the last five minutes averaged together to obtain the values of oil supply temperature, Ω , and torque measured by the torque-meter $\hat{T}_T$. Here, $\hat{T}_T$ is the total torque provided to the closed loop by the external AC motor to maintain an operation at the set Ω value, representing the total torque loss of the closed loop. Lastly, after a test is completed, the test apparatus is disengaged from the flexible coupling, and any electronic drift of the torque meter is recorded, regardless how small it is. Electronic drift occurs partly due to thermal expansion of the high speed spindle during the test, so it must be subtracted from $\hat{T}_T$ in order to obtain the actual input torque. This drift torque is typically less than 0.3-0.4% of the measured $\hat{T}_T$ value.

2.3 Inspection Procedure for Gear Specimens

A total of twelve sets of spur gears were used throughout the experimental test regimen, six of which were used to study the influence of lubricating oil on power loss. Each set is formed by 2 identical gear pairs (4 gears). Since the gears were used for more

than one test, it was critical to qualify that the tooth surfaces did not undergo changes in geometry and roughness throughout testing. For this purpose, two types of gear inspections were performed before and after each test. The first inspection was performed by using a Gleason M&M 255 gear coordinate measurement machine (CMM), pictured in Fig. 2.4, to measure gear lead and profile variations from a perfect involute. This inspection documented the surfaces of new gears and quantified any wear accumulated after a given test.

Full CMM gear inspections of lead, index, pitch, and spacing were performed on the left flanks of all new gears. Right flanks were not measured, except in a few cases, because the left flanks were the loaded flanks during power loss testing. After each test, gears were removed from the gear efficiency test machine, cleaned with an alcohol solution, and re-inspected. After the first inspection, lead and profile measurements were performed in order to document the amount of wear, if any.

Figure 2.5 shows typical inspection results for a new ground test gear having 40 teeth. For lead measurements, deviation from a horizontal line represents deviation from a perfect involute profile across the face width. Index measurements indicate the deviation of gear tooth position from a perfect gear with no errors. Pitch inspections measure variations in circular pitch from a theoretically perfect tooth. Lastly, gear spacing inspections measure spacing errors between adjacent gear teeth. Pitch and tooth spacing are derived from index measurements. Thus, if the gear teeth are perfectly spaced with a uniform circular pitch, then each bar would be the same height of zero microns. These results show exceptional gear accuracy. The waveform observed in the index measurement is primarily due to the eccentricity introduced by the mounting arbor

Figure 2.4 Gear CCM machine with a test gear mounted on the expanding arbor.

(a)

(b)

Figure 2.5 Gear CMM inspection results for a 40-tooth gear prior to run-in; (a) gear tooth lead measurements at three roll angles for four gear teeth, (b) gear index measurements, (c) gear pitch measurements, and (d) gear tooth spacing measurements.

(Continued)

Figure 2.5 Continued.

(c)

(d)

during inspection, rather than being an actual manufacturing error. The waveform is not observed in pitch or spacing figures because pitch and spacing depend on the difference between adjacent index measurements.

The second type of inspection was intended to measure gear tooth surface roughness profiles by using a Taylor-Hobson Form Talysurf-120 surface profiler. This inspection ensured that the changes in surface roughness amplitudes remained minimal throughout the test program. This surface profiler is capable of measuring variations in surface roughness to a resolution of 12 nm. Measured roughness profiles are analyzed to compute typical roughness parameters, including centerline average roughness R_a , root-mean-square roughness R_q , and several other relevant surface parameters in characterizing topography. In order to ensure repeatable measurements, a gear-locating fixture was constructed and is pictured in Fig. 2.6 along with the Talysurf surface profiler. This fixture securely holds the gear in a position such that the tooth profile to be measured is mostly horizontal and perpendicular to the inspection probe. This was accomplished by mounting the gear on a pin designed for a close running fit with the gear bore. Above the gear, a bracket holds a brass locating screw against a gear tooth. A rubber band is wrapped between a gear tooth and the locating screw such that the gear is held snugly against the locating screw. The gear fixture has a rectangular locating recess on the bottom that fits the vee-block on the base plate of the surface profiler. This ensures that the desired position of the gear tooth surface is maintained throughout the test and the same exact position can be achieved in subsequent inspections.

Like the CMM gear profile accuracy inspections, gear surface roughness measurements were performed prior to gear run-in and after each subsequent test.

Figure 2.6 The gear locating fixture and the probe of the Talysurf surface profiler during a surface roughness measurement.

Surface roughness measurements were performed for each test gear on both right and left flanks of two gear teeth spaced 90 degrees apart. Since the right flanks were not used for efficiency measurements, and thus remained unchanged throughout the test regimen, their measurements were used to establish statistical limits on the reliability of left flank measurements. Typical tooth roughness traces in the profile direction of the teeth of the 40-tooth ground and chemically polished run-in gears are illustrated in Fig 2.7.

Since surface topography measurements are highly dependent on the parameters defining the inspection routine, a study was performed in order to determine which inspection parameters resulted in the most accurate surface roughness measurement. Surface roughness measurements with the Talysurf machine were performed with sets of 23-tooth and 40-tooth gears. These measurements were compared to inspection results performed by the sponsor on a similar machine. The study resulted in selection of 16 cutoffs of 0.08 mm length for 40-tooth gears, and 16 cutoffs of 0.25 mm length for 23-tooth and 17/26-tooth gears.

2.4 Test Specimens and Test Matrix

The objective of the test program was to investigate the impact of gear module, surface finish, manufacturing processes, low-friction coatings, and lubricant type on spur gear efficiency. Seven different sets of spur gears were procured for this purpose. For the study on gear module, 23 tooth (23T) ground gears with $m = 3.95$ mm are compared to 40 tooth (40T) ground gears, having $m = 2.32$ mm. The center distance was identical for both gear pairs. In order to study the influence manufacturing processes used to finish tooth surfaces, 40T hobbed-shaved gears were procured to compare to ground gears. The

Figure 2.7 Typical surface roughness profiles for run-in 23-tooth test gears; (a) ground gears, and (b) chemically polished gears.

hob-shaving process produces tooth surfaces that are not as accurate as ground surfaces. In addition, hobbed-shaved gears must be case-hardened after shaving, adding additional profile deviations due to heat treat distortions. Therefore, a comparison of hobbed-shaved and hard-ground gears will also provide information about the influence of gear profile accuracy on gear efficiency.

The influence of the surface roughness will be documented by comparing sets of 23T ground, 40T ground, and 40T hobbed-shaved gears that were finished using a chemical polishing process that reduced R_a to below 0.1 micron. Similarly, four sets of 40T ground and chemically polished gears were coated with three different diamond-like carbon (DLC) coatings to investigate their influence on gear efficiency. One DLC coating, thought to be most effective, was applied to two different 40T gear sets. Finally, to study the influence of lubricating oil on efficiency, power loss measurements were performed with 23T and 40T ground and chemically polished gears in four different oils. The first two oils, designated Lubricants A and B, respectively, are heavy synthetic 75W90 gear oils, the third oil, Lubricant C, is a lower viscosity synthetic gear oil marketed as a more efficient blend, and the fourth oil, Lubricant D, is a common automatic transmission fluid.

Additionally, two sets of non-unity ratio gear pairs (17 tooth pinion and 26 tooth gear, 17/26T) were procured to represent a certain final drive application. Hobbed-shaved, and hobbed-shaved plus chemically polished gear pairs were tested in both Lubricants A and D. The purpose of these tests was to approximate the influences of surface roughness and oil type in the actual application without procuring a special test

machine specifically for the actual production gears. Table 2.1 summarizes the various tests and their purposes for the gear efficiency study. Table 2.2 lists design parameters of the 23T, 40T and 17/26T test gears. Figure 2.8 shows samples of all the experimental test gears, while Fig. 2.9 shows a closer view of the gear tooth surfaces of the 23T and 40T gears shown in Fig. 2.8. Figures 2.9(a) and 2.9(b) show 23T ground and chemically polished gear surfaces, respectively, while Fig. 2.9(c) and (d) show 40T ground and chemically polished gear surfaces. Figure 2.9(e) shows the 40T hobbed-shaved gear surfaces, while the 40T hobbed-shaved plus chemically polished surfaces look identical to the ground plus chemically polished surfaces shown in Fig. 2.9(d). Figure 2.9(f) shows a 40T DLC coated gear surface.

Operating test conditions were varied for the different gear types and lubricants. The 23T and 40T gear pairs were used to simulate high performance gearing applications within a transmitted power range of $P_c = 259$ to 716 kW (348-961 hp). These gears were operated at rotational speeds of 2,000 rpm to 10,000 rpm for unloaded tests and at 6,000 rpm to 10,000 rpm under $T_c = 413$ to 684 N-m for loaded efficiency tests. Lubricant temperature for these tests was $110^\circ C$. In order to simulate commercial automotive applications, 23T and 40T gears were operated at lower speed, torque and temperature levels with Lubricant D as the lubricating oil. In Lubricant D, the gears were tested at $\Omega = 2,000$ to 6,000 rpm, and $T_c = 140$ to 413 N-m, with lubricant temperature of $90^\circ C$. These test conditions correspond to $P_c = 23$ to 259 kW (39-348 hp), and represent typical production automotive operating conditions. 40T hobbed-shaved gears were operated

Test	Gear	Process	Lube Oil	Purpose
1	23T	Ground	Lubricant A	Effect of Lubricant A for Ground and Chemically Polished Gears
2		Ground + Chem. Polished		
3	40T	Ground		
4		Ground + Chem. Polished		
5	23T	Ground	Lubricant B	Effect of Lubricant B for Ground and Chemically Polished Gears
6		Ground + Chem. Polished		
7	40T	Ground		
8		Ground + Chem. Polished		
9	23T	Ground	Lubricant C	Effect of Lubricant C for Ground and Chemically Polished Gears
10		Ground + Chem. Polished		
11	40T	Ground		
12		Ground + Chem. Polished		
13	40T	DLC-A	Lubricant A	Effect of DLC Coatings
14		DLC-B		
15	40T	DLC-C		
16		DLC-D		
17	23T	Ground	Lubricant D	Effect of Hob/Shave and chemical polishing - commercial automotive applications
18		Ground + Chem. Polished		
19	40T	Ground		
20		Ground + Chem. Polished		
21		Hob / shave		
22		Hob / shave + Chem. Polished		
23	17/26T	Hob / shave	Lubricant A	Hypoid Simulation - commercial automotive applications
24		Hob / shave + Chem. Polished	Lubricant D	
25		Hob / shave		
26		Hob / shave + Chem. Polished		

Table 2.1 Spur gear efficiency experimental test plan.

Parameter	23T	40T	17T	26T
Module	3.95	2.32		4.23
Pressure Angle, deg.	25.0	28.0		22.5
Face Width	19.50	26.7	14.00	20.29
Center Distance	91.50	91.50	91.50	91.50
Pitch Diameter	90.86	92.74	71.97	110.07
Base Diameter	82.34	81.89	66.49	101.69
Outside Diameter	100.34	95.95	80.02	117.11
Root Diameter	81.30	85.80	62.87	99.82
Start of Active Profile	85.38	87.73	67.92	105.33
Circular Tooth Thickness	6.435	2.925	7.81	5.65
Root Fillet	1.63	0.83	1.68	2.74

Table 2.2 Gear design parameters for 23T, 40T, 17T, and 26T gears used in the spur gear efficiency study. Units are in mm unless otherwise specified.

Figure 2.8 Samples of the experimental test gears used in the gear efficiency study

under these test conditions as well. In accordance with their intended application, tests with 17/26T gears were performed at pinion (17T) speeds of $\Omega = 500$ to 2,500 rpm (gear speeds of $\Omega = 327$ to 1,635 rpm), and pinion torques of $T_c = 214$ N-m to 621 N-m (gear torques of $T_c = 140$ N-m to 413 N-m, representing $P_c = 7$ to 106 kW (10-142 hp). The 17/26T gears were operated in Lubricant D at 90°C , and in Lubricant A at 48°C . Tables 2.3 and 2.4 summarize operating test conditions for the 23T and 40T gears, and 17/26T gears, respectively.

Test sequence was governed by time constraints, torque acquisition system characteristics, and availability of rotating components. Tests generally progressed from low torque to high torque, and high speed to low speed. Testing began with low torques because the gear run-in procedure was at the lowest torque level. That way, T_c would be changed a minimum number of times. A typical test sequence started at 10,000 rpm and decreased in subsequent tests due to the most desirable torque-meter operating characteristics. The manufacturer recommends exercising the torque meter to full scale three times before calibration. Since drive torque is higher for higher rotational speeds, operating at the highest speed first had the approximate effect of exercising the torque meter to the highest range first since subsequent measurements at lower rotational speeds would fall below the initial drive torque. Table 2.5 displays the test sequence of operating conditions for all gears.

Figure 2.9 Various test gear surfaces; (a) 23T ground gear, (b) 23T chemically polished gear, (c) 40T ground gear, (d) 40T chemically polished gear, (e) 40T hobbed-shaved gear, and (f) 40T DLC coated gear.

Test Condition	Rotational Speed, Ω [rpm]	Pitch-Line Velocity [m/s]	Loop Torque, T_c [Nm]
1	2,000	10	239
2	2,000	10	413
3	2,000	10	546
4	4,000	19	239
5	4,000	19	413
6	4,000	19	546
7	6,000	29	239
8	6,000	29	413
9	6,000	29	546
10	6,000	29	684
11	8,000	38	413
12	8,000	38	546
13	8,000	38	684
14	10,000	48	413
15	10,000	48	546
16	10,000	48	684
17	10,000	48	0
18	8,000	38	0
19	6,000	29	0
20	4,000	19	0
21	2,000	10	0

Table 2.3 Operating test conditions for 23T and 40T ground, hobbled and shaved, and chemically polished gear pairs operated with lubricants A, B, and C at 110°C , and Lubricant D at 90°C .

Test Condition	Pinion Speed, Ω [rpm]	Pitch-Line Velocity [m/s]	Pinion Torque [Nm]
22	500	2	140
23	500	2	239
24	500	2	406
25	1,000	4	140
26	1,000	4	239
27	1,000	4	406
28	1,500	6	140
29	1,500	6	239
30	1,500	6	406
31	2,530	10	140
32	2,530	10	239
33	2,530	10	406
34	500	2	0
35	1,000	4	0
36	1,500	6	0
37	2,530	10	0

Table 2.4 Operating test conditions for 17/26T hobbed-shaved and hobbed-shaved plus chemically polished gear pairs in lubricant D at 90°C, and lubricant A at 48°C.

23T and 40T gears in Lubricants A, B, and C					
Loop Torque [N-m]	Rotational Speed [rpm]				
	2,000	4,000	6,000	8,000	10,000
0	20	19	18	17	16
413	5	4	3	2	1
546	10	9	8	7	6
684	15	14	13	12	11

23T and 40T gears in Lubricant D			
Loop Torque [N-m]	Rotational Speed [rpm]		
	2,000	4,000	6,000
0	12	11	10
239	9	8	7
413	3	2	1
546	6	5	4

17/26T gears in Lubricants A and D				
Pinion Torque [N-m]	Pinion Rotational Speed [rpm]			
	500	1,000	1,500	2,530
0	16	15	14	13
239	4	3	2	1
413	8	7	6	5
546	12	11	10	9

Table 2.5 Experimental test sequence for all gears and operational parameters

2.5 Accuracy and Repeatability

In order to have confidence in the efficiency data collected, all test specimens must initially be qualified in terms of their specified dimensions, tooth profile accuracy, and the required initial surface roughness amplitudes. Once qualified, these initial baseline profiles must be regularly compared to profiles measured after each test to ensure that profiles remain relatively unchanged to alter the data and the efficiency comparison. Likewise, the changes in tooth profile surface roughness after each test must also be monitored. If significant changes in surface roughness amplitudes of a test gear are observed, then the interpretation of the collected data must account for such changes. The following sections outline gear CMM and surface roughness measurements performed regularly to ensure the stability of the tooth surfaces in terms of their micro-geometry (wear) and surface roughness. Once the consistency of the test specimens throughout the tests are verified, the next item to check is the repeatability and the accuracy of the test machine and its sensors. The last two sub-sections are intended to show the repeatability and accuracy of the test machine and its auxiliary components.

Gear Accuracy and Check for Surface Wear

In order to document gear tooth surface wear, three CMM lead measurements were performed on four gear teeth for each test gear after every test. 23T and 40T chemically polished gears were operated for four tests in the four lubricating oils of interest, while 23T and 40T ground gears were operated in the same tests as well as an additional repeatability test. 40T hobbed-shaved gears were only operated in Lubricant D, while the 17/26T hobbed-shaved gears were operated in both Lubricants A and D.

DLC coated gear pairs were operated in both Lubricant A and Lubricant B. Figure 2.10 compares new CMM inspection results of one of the new gears in each set to the same gear after all tests were completed. Figures 2.10(a) and 2.10(b) show CMM inspection results for a 23T ground and a 23T chemically polished gear, respectively. Figures 2.10(c) and 2.10(d) show CMM inspection results for a 40T ground, and a 40T chemically polished gear, respectively. Figures 2.10(e) and 2.10(f) show lead traces for both left and right flanks for 40T hobbed-shaved gears, and 40T hobbed-shaved plus chemically polished gears, respectively. Figures 2.10(g) and 2.10(h) show lead traces for a 17/26T hobbed-shaved gear, and a 17/26T hobbed-shaved plus chemically polished gear, respectively. Results from Figure 2.10 can be summarized as follows:

- All gear surfaces showed little to no discernable wear throughout the test regimen (less than 5 μm).
- Hobbed-shaved gears were characterized by manufacturing profile deviations in surface height of about 5 μm , each usually less than 1 mm in length across the face width.
- Ground gears were characterized by flat, uniform lead traces with no local variations in surface height that is characteristic of hobbed-shaved gears.
- 40T hobbed-shaved plus chemically polished gears exhibited significantly better lead traces compared to the baseline hobbed-shaved gears.
- 23T and 40T gears exhibited a lead slope of nearly 5 μm across the face width.

(a)

New

Run-In

Figure 2.10 CMM gear accuracy inspection results of new gear surface and gear surface after completion of all testing for one gear in each design category; (a) 23T ground gear, (b) 23T chemically polished gear, (c) 40T ground gear, (d) 40T chemically polished gear, (e) 40T hobbled and shaved, (f) 40T hobbled and shaved plus chemically polished, (g) 17/26T hobbled and shaved, (h) 17/26T hobbled and shaved plus chemically polished.

(continued)

Figure 2.10 Continued.

(b)

(Continued)

Figure 2.10 Continued.

(c)

(Continued)

Figure 2.10 Continued.

(d)

(Continued)

Figure 2.10 Continued.

(e)

New

Run-In

(Continued)

Figure 2.10 Continued.

(f)

(Continued)

Figure 2.10 Continued.

(g)

(Continued)

Figure 2.10 Continued.

(h)

- All 23T and 40T gears had lead crown of between 5 and 10 μm , consistent with design specifications.
- 17/26T gears had a lead crown of approximately 5 μm , consistent with design specifications, but comparable in magnitude to local variations in surface height.

Variation of Gear Surface Roughness through Efficiency Tests

Since gear surface roughness has been shown to be a key parameter influencing efficiency [4,16,26,42], it was important to document surface roughness of the test gears before and after each test. Table 2.6 shows centerline average surface roughness R_a , and root-mean-square roughness R_q for each of the test gear sets, measured after operation in the lubricating oils indicated. Two roughness traces were performed on the left and right flanks of each test gear, resulting in eight measurements per flank per gear set. The surface roughness numbers in Table 2.6 are the average R_a and R_q of those eight measurements. Table 2.6 indicates that surface roughness of the test gears remained nearly constant throughout testing, stating that surface conditions were quite stable throughout the test program. An initial slight decrease in surface roughness generally due to wear down of asperity peaks is followed by a saddle increase.

Figure 2.11 shows the R_a measurements of Table 2.6 for 23T and 40T ground and chemically polished gears with 95% confidence limits imposed. These confidence limits were obtained for each gear set using the right flank surface roughness measurements. Since the right flanks were not used in this study, measurements of these

(a)

		Average Ra (micrometer)			
# of Teeth	Surface Treatment	Lubricant A	Lubricant B	Lubricant C	Lubricant D
23	Ground	0.24	0.21	0.23	0.24
	Chem. Polished	0.06	0.03	0.03	0.04
40	Ground	0.16	0.14	0.13	0.17
	Chem. Polished	0.05	0.05	0.04	0.06
40	DLC-B	N/A	0.10		
40	DLC-A	0.13			
40	DLC-C	0.11			
40	DLC-D	0.11			

		Lubricant A	Lubricant D
40	Hob + Shave	0.10	N/A
	Chem. Polished	0.06	N/A
17/26	Hob + Shave	0.25	0.17
	Chem. Polished	0.03	0.03

Table 2.6 Taylor Hobson Talysurf surface roughness measurements for test gears, measured after testing in the lubricating oil indicated; (a) R_a , and (b) R_q . Left flank measurements averaged from two teeth on four gears.

(continued)

Table 2.6 Continued.

(b)

		Average Rq (micrometer)			
# of Teeth	Surface Treatment	Lubricant A	Lubricant B	Lubricant C	Lubricant D
23	Ground	0.32	0.28	0.31	0.32
	Chem. Polished	0.08	0.05	0.04	0.06
40	Ground	0.20	0.18	0.17	0.21
	Chem. Polished	0.07	0.06	0.06	0.09
40	DLC-B	N/A	0.18		
40	DLC-A	0.17			
40	DLC-C	0.19			
40	DLC-D	0.22			

		Lubricant A	Lubricant D
40	Hob + Shave	0.15	N/A
	Chem. Polished	0.08	N/A
17/26	Hob + Shave	0.34	0.26
	Chem. Polished	0.04	0.04

Figure 2.11 Variation of left flank surface roughness measurements for all 23T and 40T gears, and 95% confidence bands established from right flank measurements; (a) 23T gears, and (b) 40T gears.

flanks represent repeatability of the test machine and fixtures. The 95% confidence limits are calculated from standard error of the sample mean, given by

$$\sigma_{\overline{X}} = \frac{s}{\sqrt{n}} \quad (2.1)$$

where s is the sample standard deviation, and n is the number of samples. Hence, 95 percent confidence limits are given by $\pm 1.96\sigma_{\overline{X}}$. Due to repeatability tests performed by using ground gears, n was not the same for every gear set. However, every gear set was measured at least four times, resulting in a minimum of 32 individual surface roughness measurements for the right flanks of each gear set. Calculated 95% confidence limits for R_a are $\pm 0.008 \mu\text{m}$ for 23T ground gears, $\pm 0.006 \mu\text{m}$ for 23T chemically polished gears, $\pm 0.013 \mu\text{m}$ for 40T ground gears, and $\pm 0.003 \mu\text{m}$ for 40T chemically polished gears.

Figures 2.10, 2.11 and Table 2.6 show that the test gear surfaces remained consistent throughout testing, both in terms of profile accuracy and surface roughness. CMM surface inspections revealed key differences in gear accuracy between ground and hobbed-shaved gears, as well as documenting no appreciable wear from the first to last test. The surface roughness measurements indicate with a significant degree of confidence that measured R_a values changed very little after run-in. Having documented excellent consistency in the test gears throughout the test sequence, experimental power loss comparisons can be made on basis of the influence of surface roughness, module, gear accuracy, low-friction coatings, and lubricating oil.

Measurement Accuracy of High-Speed Gear Efficiency Test Machine

The experimental test parameters that are monitored during gear efficiency testing are the input torque $\hat{T}_T$, the rotational speed Ω , the oil reservoir temperature, the oil supply temperatures at the test and reaction side gearboxes, and the oil return temperatures from the test and reaction side gearboxes. Torque is measured with a Lebow 9000 series non-contact digital telemetry torque meter. It has a full-scale range of 50 N-m, and installed linearity to 0.03% of full scale. Figure 2.12 shows a typical torque signal for 40T ground gears operating under 546 N-m at 6,000 rpm. The rapid rise in torque at the beginning of the test is due to a 100-point running average in the LabView data acquisition system. Figure 2.13 shows the rotational speed Ω for the same test where the set speed was 6,000 rpm. In most tests, a 0.2-0.3% deviation of the actual speed from the set value was observed. While this is quite negligible, still the gear efficiency is calculated using the actual speed instead of the set-speed to eliminate any secondary errors coming from the speed condition. Except for the small constant bias between the set and actual Ω values, rotational speed remained nearly constant within ± 2 rpm.

Figure 2.14 shows temperature measurements for the same test. Lubricant reservoir and supply temperatures were very close, with supply temperatures usually several degrees higher. This suggests that the oil is further heated as it is forced through the supply system, but it could also be due to location of the thermocouple in the oil stand. Oil return temperatures are higher because the return oil absorbs waste heat from the gear system. For most of the tests, the lubrication system of the test machine was

Figure 2.12 Typical variation of input driving torque versus time. 40T ground gears, $T_c = 546$ N-m, $\Omega = 6,000$ rpm.

Figure 2.13 Typical variation of rotational speed versus time. 40T ground gears, $T_c = 546$ N-m, $\Omega = 6,000$ rpm.

Figure 2.14 Typical variation of temperature signals versus time. 40T ground gears, $T_c = 546$ N-m, $\Omega = 6,000$ rpm.

able to maintain the oil supply temperatures effectively within $110 \pm 2^\circ C$ or $90 \pm 2^\circ C$, based on the set test conditions.

Repeatability of Power Loss and Efficiency Measurements

In order to quantify the repeatability of loaded power loss measurements for the test machine, three sets of gears were identified and tested periodically. The first set of gears was 40T type (ground and chemically polished with 19.5 mm face width), operated in Lubricant A. These gears were used to demonstrate the repeatability of the test machine in Fig. 2.15 over a 2-year period, through several teardowns, shaft, bearing, and seal replacements. Four sets of $\hat{T}_T$ measurements were performed with these gears, except for Test D, for which only test conditions 8, 9, and 10 were tested due to time constraints. The numbers along the x-axis of this figure represent certain test conditions listed in Table 2.3. Figure 2.15 indicates that the test fixtures maintained repeatability to within 0.5 N-m in terms of $\hat{T}_T$ when $T_c = 684$ N-m, which corresponds to 0.04% in total efficiency per gearbox, and this can be considered satisfactory.

The second and third sets of gears were 40T ground gears with 26.7 mm face width, and 23T ground gears with 19.5 mm face width, respectively. Both of these gear sets were operated in Lubricant A. These gears were used to characterize repeatability of the test machine on a shorter timeline through one replacement of shafts, bearings, and seals. Figures 2.16 and 2.17 show repeatability in terms of $\hat{T}_T$ for the 40T gears and 23T gears, respectively. The 40T gears show excellent repeatability in measured power loss, and the 23T gears show satisfactory agreement.

Figure 2.15 Comparison of repeatability tests using 40T chemically polished gears in lubricant A, 19.5 mm face width.

Figure 2.16 Comparison of repeatability tests using 40T ground gears, in Lubricant A, 26.7 mm face width.

Figure 2.17 Comparison of repeatability tests using 23T ground gears, in Lubricant A, 26.7 mm face width.

2.6 Summary

Inspection methodologies were developed for the documentation of gear surface roughness and wear after each test. Surface roughness measurements of the unloaded gear flanks were used to establish 95% confidence limits of about $\pm 0.01 \mu\text{m}$ for all surface roughness measurements. Results show that after the initial run-in phase, gear surface roughness amplitudes remained constant throughout the test regimen. CMM inspections show negligible wear for all gear surfaces. With a high degree of confidence in the integrity of the gears throughout testing and with the demonstrated repeatability of the test machine and instrumentation, experimental results presented in the next chapters for power loss and efficiency can be accepted confidently.

CHAPTER 3

EFFICIENCY EXPERIMENTS UNDER JET LUBRICATED CONDITIONS

3.1. Components of Power and Efficiency Losses

There are a large number of sources of power loss in a gearbox due to complex mechanical and fluid-mechanical interactions. Retaining only the primary sources of power loss, the total power loss of a gearbox can written as

$$P_T = P_m + P_{churn} + P_b + P_{seal} \quad (3.1)$$

where P_T is total gearbox power loss, P_m is the mechanical (load-dependent) power loss in the gear mesh, P_{churn} is the gear fluid churning power loss that includes both oil churning and air windage, P_b is the bearing power loss, and P_{seal} is the shaft seal power loss. P_m can be resolved into

$$P_m = P_s + P_r \quad (3.2)$$

where P_s is the frictional sliding power loss due to asperity contacts and fluid shearing, and P_r is the power loss resulting from fluid viscous resistance to rolling in the gear mesh contact zones. Sliding and rolling friction in a gear mesh cannot be separated experimentally, but predictive methods have been developed to estimate their contributions in combined rolling-sliding contacts [14, 16, 45]. P_{churn} applies to both jet-lubricated and dip-lubricated applications and is described by

$$P_{churn} = P_{inertia} + P_{trap} + P_{drag} \quad (3.3)$$

where $P_{inertia}$ applies only to jet-lubricated applications, and is the power loss resulting from acceleration of the impinging oil jet by the rotating gear teeth [30-31]. P_{trap} arises from squeezing oil and air out of the spaces between gear teeth as they mesh [30, 32-34], and P_{drag} is the combined fluid drag power loss on the sides of the gears, faces of the gear teeth, and from fluid trapped between gear teeth as the gears rotate.

Widely used formulas for load-depended and load-independent power losses of rolling element bearings were originally developed empirically by Palmgren [46], then later popularized by Harris [47], and further refined by bearing manufacturers [48]. Load-dependent bearing friction torque M_L in N-mm is given by [43] as

$$M_L = 0.5\mu_b W_b d_{bore} \quad (3.4)$$

where μ_b is an experimentally determined average bearing friction coefficient obtained under full film lubrication conditions, W_b is radial bearing load in N, and d_{bore} is the bearing bore diameter in mm. For the NJ 406 J cylindrical roller bearings used in this study, $\mu_b = 0.0011$ and $d_{bore} = 30$ mm. Load-independent friction torque is caused by viscous shear of the lubricant, and is given in N-mm by

$$M_v = \begin{cases} 1.6 \times 10^{-5} f_o d_m^3, & \nu\Omega < 2,000 \\ 10^{-7} f_o (\nu\Omega)^{2/3} d_m^3, & \nu\Omega > 2,000 \end{cases} \quad (3.5)$$

where f_o is a constant depending on bearing type and lubrication method, ν is kinematic viscosity in cSt, Ω is rotational speed in rpm, and d_m is bearing mean diameter in mm. For the NJ 406 J cylindrical roller bearings used in this study under jet-lubricated conditions, $f_o = 2.2$, and $d_m = 60$ mm. Lastly, P_b is defined in mW as

$$P_b = (M_L + M_v)\omega \quad (3.6)$$

where $\omega = \frac{2\pi}{60}\Omega$ is the angular velocity of the bearing in radians/sec. Finally, shaft seal power loss in kW is estimated approximately by [49]

$$P_{seal} = 2.544 \times 10^{-7} d\Omega \quad (3.7)$$

for a Buna type rubber radial lip seal where d is shaft diameter in mm. For the shaft seals used in this study, $d = 30$ mm.

3.2. Calculation of Power Loss and Efficiency from Experimental Results

Total torque loss for both gearboxes $\hat{T}_T$ and rotational speed Ω were measured while operating the experimental test setup at constant speed and under constant torque T_c in the power circulation loop. Since the experimental test setup has two gearboxes, the total torque loss for each gearbox is found by $T_T = 0.5\hat{T}_T$. Measured total gearbox power loss can be decomposed into its load-dependent and load-independent components by writing

$$P_T = P_{mech} + P_{spin}. \quad (3.8)$$

This requires a two-step measurement procedure. First, the total power loss $P_T = \omega T_T$ is measured at a certain rotational speed Ω under a given torque transmitted T_c . Then, the same test is repeated under no load $T_c = 0$. In this second test, $P_{mech} \approx 0$ and the measured power loss is nearly equal to P_{spin} . These two companion tests allow measurement of P_{mech} , according to Eq. (3.8). For the experimental set-up shown in Fig. 2.2, which contains 2 identical gear pairs, 8 identical cylindrical roller bearings and 4 identical shaft seals, load-dependent and load-independent power losses can be further decomposed as

$$P_{mech} = P_m + 4P_{b,L} , \quad (3.9a)$$

$$P_{spin} = P_{churn} + 4P_{b,v} + 2P_{seal} . \quad (3.9b)$$

Here $P_{b,L} = \omega M_L$ and $P_{b,v} = \omega M_v$ where M_L and M_v are defined earlier in Eq. (3.4) and Eq. (3.5), respectively. Accordingly, the mechanical power loss of an individual gear mesh while rotating at speed Ω and transmitting torque T_c is given in terms of measured total power losses, P_T and P_{spin} , and the predicted load-dependent bearing losses $P_{b,L}$ as

$$P_m = P_T - P_{spin} - 4P_{b,L} . \quad (3.10)$$

Finally, the average total efficiency of each gearbox (gear pair, bearings and seal) and the average gear mesh mechanical efficiency are calculated, respectively, by

$$\bar{\eta}_T = \frac{P_c - P_T}{P_c}, \quad \bar{\eta}_m = \frac{P_c - P_m}{P_c} . \quad (3.11a,b)$$

3.3. Gear Mesh Mechanical Power Loss

The test matrix defined by Table 2.1 covers a wide range of operating conditions, gear module, surface roughness, manufacturing processes, lubricants and thin-film coatings. This allows quantification of the direct influence of these parameters and

features on gear mesh mechanical power loss P_m . This section presents experimental results to demonstrate influences of the following specific parameters, lubricants, engineered surface treatments and manufacturing processes on P_m :

- (i) *Rotational speed, Ω* : Ω will be varied within a range from 2,000 to 10,000 rpm with increments of 2,000 rpm at various T_c values, and the resultant P_m values will be measured for both fine (40T) and coarse (23T) pitch gear pairs.
- (ii) *Torque transmitted, T_c* : This parameter will be varied between 0 (unloaded spin tests) and 684 N-m (504 ft-lbs) at increments of nearly 140 N-m (103 ft-lbs) at various Ω values to quantify its impact on P_m .
- (iii) *Gear module, m* : Two different gear pair designs, one 23-tooth pair having $m = 3.95$ mm and a 40-tooth gear pair having $m = 2.32$ mm are included here. In order to closely match bending strength of the gear teeth, gear face width for 40T gears was slightly larger than for the 23T gears as shown in Table 2.3. However, since other experimental and theoretical studies [4, 16, 42] have shown that face width has negligible effect on P_m , the comparison of P_m values for these gear pairs at different operating conditions will quantify the influence of m on P_m .
- (iv) *Gear manufacturing processes*: Two of the most common gear processes, grinding and shaving, will be considered here. The hard grinding process typically achieves more accurate gears since shaved gears must be hardened after shaving, while ground gears are finished after hardening. 40T gear

specimens produced using these two methods will provide a comparison of their efficiency performance in terms of P_m .

(v) *Surface roughness*: Grinding and shaving processes result in gear tooth surfaces that are relatively rough. Knowing that surface roughness is a key parameter in defining the friction coefficient [16], chemically polished gear specimens will be evaluated for their efficiency and compared to ground and shaved specimens to quantify the influence of the surface roughness amplitude on P_m .

(vi) *Low-friction coatings*: It has long been claimed that certain PVD coatings designed to reduce friction should improve gear efficiency, but little experimental proof is available. This test matrix includes a number of diamond-like-carbon coatings applied to baseline 40T ground and chemically polished gear specimens to assess their effect on P_m .

(vii) *Lubricant type*: Four different lubricants, three gear oils of varying mechanical properties, and an automatic transmission fluid, will be used in the efficiency tests and their influence on P_m will be quantified.

Influence of Speed and Load

The influence of Ω on P_m is shown in Fig. 3.1 for ground and chemically polished 23T and 40T gears at $T_c = 413$ N-m. Here, P_m increases nearly linearly with Ω over the range of rotational speeds studied. The influence of T_c on P_m is shown in Figure 3.2 for 23T and 40T ground and chemically polished gears at $\Omega = 6,000$ rpm. Similarly, P_m also increases nearly linearly with T_c . These two figures suggest that the

Figure 3.1 Comparison of measured P_m for 23T and 40T gear pairs as a function of Ω for (a) ground gears and (b) chemically polished gears in Lubricant A. $T_c = 413$ N-m.

Figure 3.2 Comparison of measured P_m for 23T and 40T gear pairs as a function of T_c for (a) ground gears and (b) chemically polished gears in Lubricant A. $\Omega = 6,000$ rpm.

gear mesh mechanical efficiency $\bar{\eta}_m$ should remain relatively unchanged within these ranges of T_c and Ω due to the nearly linear influence of T_c and Ω on P_m . In order to demonstrate this, Fig. 3.3 shows measured gear mesh mechanical efficiency $\bar{\eta}_m$ for 23T and 40T ground gears in Lubricant A at certain test conditions listed in Table 2.3. Here, the $\bar{\eta}_m$ values calculated by Eq. (3.11b) for 40T gears range between 99.76% at test condition 9 ($T_c = 546$ N-m and $\Omega = 6,000$ rpm), and 99.81% at test condition 15 ($T_c = 546$ N-m and $\Omega = 10,000$ rpm). For 23T gears, measured $\bar{\eta}_m$ values range between 99.61% at test condition 8 ($T_c = 413$ N-m and $\Omega = 6,000$ rpm), and 99.70% at test condition 16 ($T_c = 684$ N-m and $\Omega = 10,000$ rpm). For the wider range of Ω included in Fig. 3.1, measured $\bar{\eta}_m$ values for 40T ground gears stays nearly constant with $\bar{\eta}_m = 99.77\%$ at 2,000 rpm and 99.78% at 10,000 rpm. For the range of T_c in Fig. 3.2, measured $\bar{\eta}_m$ ranges between 99.70% at $T_c = 140$ N-m and 99.76% at 684 N-m. Therefore, the influences of T_c and Ω on $\bar{\eta}_m$ are not dominant in the range of test conditions studied.

Influence of Gear Module

In order to study the influence of gear module on gear mesh power loss, power loss measurements were conducted with both ground and chemically polished gear pairs with the various lubricant types indicated in Table 2.1 over a range of T_c and Ω . Figure 3.4 compares measured P_m values for ground and chemically polished 23T and 40T gears in Lubricant A for T_c from 413 N-m to 684 N-m, and Ω from 6,000 rpm to 10,000

Figure 3.3 Measured $\bar{\eta}_m$ values of 23T and 40T ground gear pairs in Lubricant A at various test conditions listed in Table 2.3.

Figure 3.4 Influence of m on measured P_m for 23T and 40T gear pairs in Lubricant A; (a) ground gears and (b) chemically polished gears.

rpm. P_m values for the 40T gear pairs are on average 38% and 43% lower than those for the 23T gear pairs formed by ground and chemically polished gears, respectively. Figure 3.5 shows the same comparison, but for gear pairs lubricated with Lubricants B and C. When Lubricant B is used, P_m values for ground 40T gear pairs are measured to be about 33% lower than for ground 23T gears and about 35% lower for chemically polished gears. In case of Lubricant C, P_m values for 40T ground gear pairs are 35% lower than those for 23T ground gears and 23% lower for chemically polished gears. Likewise, P_m for 40T gear pairs in Lubricant D was on average 32% and 31% lower than for 23T gear pairs for ground and chemically polished gears, respectively, as shown in Fig. 3.6. Percent reduction in P_m for 40T gears versus 23T gears generally increases slightly with Ω , but there appears to be no definite trend with T_c . This is in agreement with previous studies [4-5, 16, 18, 42], and is primarily due to the higher sliding velocities in the mesh of gear pairs with larger module m . Lastly, since the 40T ground gear pairs had a slightly lower surface roughness (R_a) value than the 23T gear pairs as indicated in Table 2.6, the influence of m in this case is somewhat exaggerated.

Influence of Gear Manufacturing Processes

Since the shaving process is applied prior to case hardening, heat treat distortions make shaved gears typically less accurate than their hard ground counterparts. In chapter 2, this was demonstrated for the test gears used in this study through gear tooth lead, measurements using a gear CMM. Figure 3.7 compares the efficiency performance of

Figure 3.5 Influence of m on measured P_m for 23T and 40T gear pairs (a) ground gears in Lubricant B, (b) chemically polished gears in Lubricant B, (c) ground gears in Lubricant C, and (d) chemically polished gears in Lubricant C.

Figure 3.6 Influence of m on measured P_m for 23T and 40T gear pairs in Lubricant D; (a) ground gears and (b) chemically polished gears.

Figure 3.7 Comparison of measured P_m for 40T, hobbed-shaved, and 40T ground gears in Lubricant D.

40T ground gear pairs to 40T hobbed-shaved gear pairs over a range of T_c and Ω using Lubricant D.

Despite having smoother surfaces as shown in Table 2.6, the hobbed-shaved gears performed consistently worse than ground gears with an average of 35% higher P_m , or 0.17% lower $\bar{\eta}_m$. This finding is comparable to the study of Naruse and Haizuka [18], which found that hobbed shaved gear pairs with $m = 4.5$ mm loaded between 35 N-m and 452 N-m at pitch-line velocity of 7.2 m/sec exhibited roughly 0.25% lower $\bar{\eta}_m$ than their ground counterparts, despite having a smoother surface finish. While this indicates that the quality of the gears has a sizable effect on the mechanical efficiency of the gear pair, the exact reason for this is not readily available. However, visual inspection of the run-in hobbed-shaved gear surfaces in Fig. 3.8 suggests that the load distribution on a hobbed-shaved surface is not as uniform as on a ground surface. Figure 3.8 shows the loaded flank used for efficiency tests. Here, the original pre-heat treat surface finish is still visible as a pattern of fine striations from root to tip of the gear tooth. This pattern indicates that little to no tooth load was carried along these striations, which would concentrate contact pressures over a smaller area than on ground gears. Higher contact pressures would cause the sliding friction coefficient μ_s at such contact locations to be higher, resulting in higher P_m values.

Influence of Surface Roughness

In terms of efficiency, the goal of super-finishing gear surfaces is to reduce asperity contact, and thus reduce sliding friction. Commercially available chemical

Figure 3.8 The run-in flank of a 40T hobbed-shaved gear after efficiency testing.

polishing processes have been suggested to improve gear efficiency as they reduce surface roughness amplitude and the resultant sliding friction coefficient of the contacting surfaces [4, 16, 26, 42]. However little published experimental proof of this is available, especially when lubricants of different viscosities are considered. In order to provide definitive experimental data on the influence of surface roughness, the measured P_m values of 23T and 40T ground gears are compared to their chemically polished counterparts for all of the four different lubricants listed in Table 2.1.

Figure 3.9 shows the viscosity-temperature characteristics of these lubricants. Lubricant B has the highest kinematic viscosity at 110°C with $\nu = 13.1$ cSt, followed by Lubricant A with $\nu = 10.7$ cSt at the same temperature. Although Lubricant C is more viscous than Lubricant D at a given common temperature, tests with Lubricant D were conducted at 90°C , whereas tests with Lubricant C were conducted at 110°C . The kinematic viscosity of Lubricant D at 90°C ($\nu = 8.0$ cSt) is higher than the viscosity of Lubricant C at 110°C ($\nu = 5.9$ cSt).

Figure 3.10 shows a comparison of the measured P_m values for ground and chemically polished surfaces with Lubricant A over a range of operating conditions. For 23T gears in Fig. 3.10(a), chemical polishing gears to $R_a = 0.06\text{ }\mu\text{m}$ compared to ground gears having $R_a = 0.24\text{ }\mu\text{m}$ reduced P_m by as little as 4% at test condition 16 ($T_c = 684\text{ N}\cdot\text{m}$ and $\Omega = 10,000\text{ rpm}$), and as much as 26% at test condition 11 ($T_c = 413\text{ N}\cdot\text{m}$ and $\Omega = 8,000\text{ rpm}$), with an average reduction of 16% for all of the test conditions presented in Fig. 3.10. For 40T gears as shown in Fig. 3.10(b), chemical

Figure 3.9 Viscosity-temperature characteristic for the lubricants used in this study.

Figure 3.10 Comparison of measured P_m for ground and chemically polished gear pairs in Lubricant A; (a) 23T gears and (b) 40T gears.

polishing gears to $R_a = 0.05 \text{ } \mu\text{m}$ compared to ground gears having $R_a = 0.16$ results in 19% and 31% reductions of P_m at test conditions 13 ($T_c = 684 \text{ N-m}$ and $\Omega = 8,000 \text{ rpm}$) and 14 ($T_c = 413 \text{ N-m}$ and $\Omega = 10,000 \text{ rpm}$), respectively, with an average reduction of 23% for all of the test conditions considered.

The general trend in Fig 3.10 is that the measured P_m values are reduced by varying amounts when gear surface roughness is reduced significantly through chemical polishing. However, the amount of improvement is dependent on the operating conditions as well as the gear module and the surface roughness amplitudes before and after the chemical polishing process. This trend generally holds with other lubricants as well. Figure 3.11 shows the same experimental comparison for Lubricants B and C. For Lubricant B, chemically polishing the tooth surfaces reduced P_m by an average of 9% for 23T gears, and 3% for 40T gear pairs. Comparing this to 23% when lubricated with Lubricant A, the reduced improvement in Lubricant B can be explained by lubricant viscosity. Since Lubricant B is more viscous than Lubricant A, the former provides a thicker oil film. Hence, reducing R_a is expected to have less of an effect on P_m when gears are operated with a thicker oil film, as asperity contacts are more limited.

With Lubricant C, chemical polishing reduced P_m for 23T and 40T gears by 23% and 9%, respectively. The larger difference observed for 23T gears is to be expected because the 23T gears have larger sliding velocities than the 40T gears, and hence larger sliding frictional loss P_s . However, this result is not the same with all lubricants tested because of differing oil viscosities, which is expected to influence both P_s and P_r at different levels.

Figure 3.11 Comparison of measured P_m for ground and chemically polished gear pairs; (a) 23T gears in Lubricant B, (b) 40T gears in Lubricant B, (c) 23T gears in Lubricant C, and (d) 40T gears in Lubricant C.

Figure 3.12 compares the P_m values of chemically polished 23T and 40T gears to their ground counterparts with Lubricant D. Here, chemical polishing resulted in 19% lower P_m values for both 23T and 40T gear pairs. Similarly, Fig. 3.13 shows the measured influence of chemical polishing on 40T hobbed-shaved gear surfaces, again lubricated with Lubricant D. Here, the chemically polished gear pairs operated at measured P_m values nearly 53% lower than their hobbed-shaved counterparts. This result suggests that chemical polishing has a larger influence on gears with inaccurate profiles. However, the gear lead measurements of Fig. 2.10 indicate that the hobbed-shaved gears that are chemically polished are quite comparable to those of chemically polished ground gears, while the baseline hobbed-shaved gears appear to be less accurate. Therefore, the improvements in P_m observed in Fig. 3.13 represent the combined influence of chemical polishing and a more accurate involute profile.

The gear inspections of Fig. 2.10 might lead one to conclude that the chemical polishing process can improve gear accuracy, but one should be cautioned against this conclusion. Although chemical polishing processes are known to reduce surface roughness amplitudes to less than $0.1\text{ }\mu\text{m } R_a$, they can provide only limited improvements to the larger profile deviations observed with the hobbed-shaved gears. Since gear inspections were not performed prior to the chemical polishing treatment, no conclusions can be drawn as to the effect of this process on gear accuracy.

Lastly, the influence of chemical polishing on a 17/26T hobbed-shaved gear pair lubricated with Lubricant D and Lubricant A is shown in Fig. 3.14. When Lubricant D is used, the chemically polished gear pair operates at 38% to 47% lower P_m , with an

Figure 3.12 Comparison of measured P_m for ground and chemically polished gear pairs in Lubricant D; (a) 23T gears and (b) 40T gears.

Figure 3.13 Comparison of measured P_m for 40T hobbed-shaved, and hobbed-shaved plus chemically polished gear pairs in Lubricant D.

Figure 3.14 Comparison of measured P_m for hobbed and shaved, and hobbed and shaved plus chemically polished 17/26T gear pairs in (a) Lubricant D, and (b) Lubricant A.

(continued)

Figure 3.14 continued.

average of 43% for all operating conditions tested. When Lubricant A is used instead of Lubricant D, P_m reductions were between 28% and 42%, with an average of 36% for all the operating conditions tested. This sizeable reduction in P_m compared to the 23T and 40T gears can be attributed primarily to the larger reduction in surface roughness from $R_a = 0.25 \mu\text{m}$ to $0.03 \mu\text{m}$ due to chemical polishing process, as seen in Table 2.6. Secondly, the pitch-line velocities, and hence rolling velocities, achieved in the 17/26T gear tests were lower as listed in Table 2.6. Since oil film thickness becomes thinner at lower rolling velocities, reductions in surface roughness, and thus asperity contacts, are likely to have a more dominant influence on P_m . Additionally, the influence of chemical polishing on P_m is greater when lubricated with Lubricant D than Lubricant A. Again, this is likely due to the increased asperity contacts with Lubricant D, the less viscous oil, and as a result, measured P_m is more sensitive to changes in R_a . This is consistent with the reduced influence of R_a on measured P_m for 40T gears when lubricated with Lubricant B compared to Lubricant A.

Influence of Low-Friction Coatings

Diamond-like carbon (DLC) surface coatings are known to reduce dry friction compared to uncoated steel contacts. Traction tests with a ball-on-disk tribometers as well as preliminary studies with gears in a transfer gearbox [16, 50] have shown that low-friction coatings do reduce friction coefficient, and by extension P_m . However, the extent to which power loss improvements are possible depends on the amount of asperity contact present in the gear mesh.

In order to evaluate the influence of low-friction coatings, three different DLC coatings were applied to four sets of chemically polished 40T gear pairs and tested with Lubricant A as the lubricating oil. Figure 3.15 compares measured P_m for the four coated gears pairs to a baseline chemically polished gear pair. DLC-A and DLC-B are proprietary coatings supplied by two separate sources, while DLC-C and DLC-D are the same coating supplied by a third source.

All four gear sets exhibited wear of the coating layer to varying extents. The numbers above each bar group in Fig. 3.15 indicate the order in which tests were conducted. Testing began with the lowest load level of $T_c = 413$ N-m, and highest speed $\Omega = 10,000$ rpm, and progressed downward in speed before increasing T_c . At each torque level, tests were conducted at 10,000 and 6,000 rpm. Observing P_m at each condition in the test sequence, it appears that DLC-B and DLC-C exhibited initially lower power loss than the baseline chemically polished gear pair, but as the tests progress, these gains deteriorated. P_m for the highest T_c at test conditions 10 ($T_c = 684$ N-m and $\Omega = 6,000$ rpm) and 16 ($T_c = 684$ N-m and $\Omega = 10,000$ rpm) show the worst performance of these coatings. Hence, it appears that DLC-B and DLC-C initially reduced friction at asperity contacts, but as the coatings wore through, power loss increased to a level consistent with surface roughness as shown in Table 2.6. Figure 3.16(a) shows wear of the coating layer for the DLC-B coated gears and Fig. 3.16(b) shows wear of the coating layer for the DLC-C coated gears.

In summary, before a particular coating can be proposed as a method to improve mechanical gear mesh efficiency, it must be durable enough to sustain a long period of

Figure 3.15 Influence of low-friction coatings on measured P_m for 40T gear pairs in Lubricant A.

(a) DLC-B

(b) DLC-C

Figure 3.16 Failed DLC coatings after efficiency tests; (a) DLC-B, and (b) DLC-C

operation without wearing off so that its potential to improve efficiency can be assessed. All four coatings tested in this study failed to endure even several hours of operation needed to complete efficiency tests. Therefore, the data provided in Fig. 3.15 cannot be relied upon to assess the influence of coatings on P_m .

Influence of Lubricant Type

Lubricant viscosity ν is a key parameter influencing P_m . Increasing ν increases oil film thickness, which helps reduce asperity contacts, and thus reduces the sliding friction component P_s of mechanical power loss. Meanwhile, increasing ν increases fluid shear forces arising from relative sliding of the two surfaces, potentially increasing P_s . It also increases rolling power losses P_r in the form of fluid squeezing, and pumping in the formation of the film. The goal of lubricant selection for increased efficiency in terms of P_m is to reduce both P_r and P_s . Figure 3.17 compares $P_m = P_s + P_r$ for 23T and 40T ground and chemically polished gear pairs lubricated with Lubricants A, B, and C. From this figure, the following observations can be made:

- For all ground gears, P_m is minimized when Lubricant B is used. This is likely due to reduced asperity contact that follows a larger film thickness, due to higher viscosity. For 23T ground gears, Lubricant B appears to have a larger impact on P_m compared to the other oils. This can be attributed to the higher sliding velocities observed with these gears, which should result in higher P_s .

- 23T chemically polished gears performed best with the least viscous lubricant, Lubricant C. Here, since the surfaces were chemically polished, there was little asperity contact with all oils, and both the fluid shear component of P_s and P_r were minimal with the lowest viscosity oil.
- 40T chemically polished gears performed best with the medium-viscosity lubricant, Shell 75W90. This result is not consistent with the 23T gears probably due to the lower range of sliding velocities encountered with the 40T gears. Since the sliding friction component of P_s represents a smaller portion of P_m for the 40T gears, the efficiency performance seems to be more dependent on optimizing fluid shear forces versus rolling losses.

Finally, Fig. 3.18 compares P_m for 17/26T hobbed-shaved, and hobbed-shaved plus chemically polished gears with Lubricant D at 90°C , and Lubricant A at 48°C . At these operating temperatures, Lubricant D has $\nu = 8.0$ cSt, while Lubricant A has $\nu = 65.2$ cSt. This comparison serves to illustrate the effect of lubricant viscosity on P_m for large differences in ν . In this comparison, the hobbed-shaved gears (without chemical polishing) benefit marginally from the thicker film afforded by Lubricant A at $\nu = 65.2$ cSt, indicating that the reduction in asperity friction forces simply offsets increases in fluid shear and rolling losses. Accordingly, it can be expected that hobbed-shaved gears that are chemically polished should perform better with Lubricant D at $\nu = 8.0$ cSt under the same operating conditions because these gears do not benefit from

Figure 3.17 Comparison of measured P_m for 23T and 40T, and ground and chemically polished gear pairs in Lubricants A, B, and C; (a) 23T ground gears, (b) 23T chemically polished gears and, (c) 40T ground gears, and (d) 40 T chemically polished gears.

Figure 3.18 Comparison of measured P_m for 17/26T hobbled and shaved, and hobbled and shaved plus chemically polished gear pairs in Lubricant D at 90°C and Lubricant A at 48°C ; (a) hobbled-shaved gears, (b) hobbled-shaved plus chemically polished gears.

a thicker film as much as their hobbed-shaved counterparts. Figure 3.18(b) bears this out, showing that the chemically polished 17/26T gears operated at 11% lower P_m in Lubricant D compared to Lubricant A. This observation can be explained by the fact that fluid shear forces and rolling losses dominate P_m when lubricant viscosity is high.

Figure 3.19 serves as a summary of the influences of gear module m , surface roughness, manufacturing process, and lubricant type on P_m . For Fig. 3.19, $\bar{\eta}_m$ was averaged over all of the test conditions. 40T chemically polished gears lubricated with Lubricant A outperformed all other gears in terms of P_m with average mechanical efficiency of $\bar{\eta}_m = 99.83\%$, while 23T ground gears lubricated with Lubricant A performed the worst with average $\bar{\eta}_m = 99.65\%$. These results show that module is the most influential parameter in determining P_m , followed by surface roughness amplitude, then lubricant type. Furthermore, these results indicate that lubricant selection should be considered at the design stage in conjunction with gear parameters in order to maximize efficiency.

3.4 Gearbox Spin Power Loss

Total gearbox power loss is defined earlier in Eq. (3.8) as a sum of P_{mech} and P_{spin} . The previous section focused on minimizing P_m , and hence P_{mech} , according to Eq. (3.9a). Although it is important to study P_m independently of other sources of power loss, doing so without consideration of gearbox spin power loss can lead to

Figure 3.19 Comparison of average measured $\bar{\eta}_m$ for 23T and 40T, ground and chemically polished gear pairs lubricated with Lubricants A, B, and C.

erroneous conclusions regarding lubricant selection on the basis of efficiency. This section will focus on the unloaded gearbox power loss P_{spin} .

In order study the influence of m on the spin power losses, P_{spin} was measured for each test outlined in Table 2.1 at rotational speeds of $\Omega = 2,000$ to $10,000$ rpm in $2,000$ rpm increments. In actuality, since the 23T and 40T gears had different face width b , the reported spin losses are the combined influence of m and b . However, previous work reported that m is more influential on P_{spin} than b [27-29]. Therefore, P_{spin} will be discussed in terms of effect of m with the understanding that the true influence of m is probably slightly higher.

Figure 3.20 shows the influence of lubricant type on P_{spin} for ground 23T and 40T gears. Since the test machine was unloaded during these tests, i.e. $T_c = 0$, the load-dependent losses were nearly zero, so that the influence of m , ν , and Ω on P_{spin} can be quantified. Since 23T and 40T gears are displayed separately, Fig. 3.20 illustrates the influence of ν on P_{spin} as a function of Ω . Referring to the lubricant temperature-viscosity relationship in Fig. 3.9, it is observed that P_{spin} follows ν . Lubricant B, being the most viscous lubricant studied, exhibited the highest P_{spin} for both 23T and 40T gears over the entire range of Ω considered. Likewise Lubricant C, having lowest ν , exhibited lowest P_{spin} . P_{spin} data was unavailable for Lubricant D at $\Omega = 8,000$ and $10,000$ rpm.

In Fig. 3.20, P_{spin} varies to a power of Ω ranging from 1.4 to 1.7, depending on the gear pair and lubricant. These results do not agree with previous studies on windage

Figure 3.20 Comparison of measured P_{spin} of 23T and 40T gear pairs lubricated with Lubricants A, B, C, and D as a function of Ω for (a) 23T ground gear pairs and (b) 40T ground gear pairs.

[27-29, 31], for which power loss was found to be proportional to $\Omega^{2.8}$ as a result of applying the kinetic energy theorem to a disc rotating in a viscous fluid. However, the present results do agree with the unloaded power loss measurements of the jet-lubricated gear friction test rig of Britton *et al* [26]. For example, measured unloaded power losses in this study for the 23T gears in Lubricant A at 110°C were found to follow the relation $P_{spin} = 6 \times 10^{-7} \Omega^{1.6}$ kW, whereas $P_{spin} = 1 \times 10^{-7} \Omega^{1.5}$ kW in ref. [26] for a gear pair lubricated with Mobiljet II synthetic oil at 50°C . These results indicate that the mechanisms of unloaded power loss in gearboxes are not as well understood as loaded power loss, and deserve a more detailed treatment than that in this study.

Although P_m was only modestly influenced by oil type, P_{spin} is shown to be strongly dependent on oil type due to their differences in ν . P_{spin} values for both 23T and 40T gears were similarly influenced by ν , with an average of 42% reduction in P_{spin} for Lubricant C compared to Lubricant B. This indicates that the selection of a lubricant is a critical step in optimizing power loss for high-speed gearing.

Prior studies on high-speed gear windage indicated that gear module is an important parameter in gearbox spin losses [27-28]. To show this, Fig. 3.21 compares measured P_{spin} at $\Omega = 2,000$ to $10,000$ rpm for 23T and 40T ground gear pairs lubricated with four gear oils. This figure shows that module plays an important role in P_{spin} regardless of lubricating oil. At 2,000 rpm, differences in P_{spin} for 23T and 40T gears are too small to be reliably captured by the test machine, but at 4,000 rpm, the 40T

Figure 3.21 Influence of m on measured P_{spin} for 23T and 40T ground gear pairs lubricated with all oils; (a) Lubricant A, (b) Lubricant B, (c) Lubricant C, and (d) Lubricant D.

gear pairs operated at an average of 7% lower P_{spin} over all oils tested. This difference is increased to an average of 16% at $\Omega = 10,000$ rpm.

3.5 Gearbox Total Power Loss

Although P_m was only slightly influenced by lubricant type, it was shown that P_{spin} is heavily dependent on ν . While the most viscous oil (Lubricant B) yielded lowest P_m for ground gears, it resulted in highest P_{spin} for the same gears. It is therefore necessary to investigate total gearbox power loss P_T in order to determine which combination of gear geometry, surface finish and oil selection resulted in the highest efficiency gear pair. To this end, Fig. 3.22 compares measured P_T values for 23T and 40T ground and chemically polished gears lubricated with Lubricants A, B, and C. Although there was no clear choice of most efficient gear oil in terms of P_m , Lubricant C can be identified as the overall best-performing lubricant. Lubricant C resulted in the lowest P_T for all gears at all operating conditions. For ground gears and 40T chemically polished gears, reduction of P_{spin} with Lubricant C offsets improvements in P_m with Lubricant B. For 23T chemically polished gears lubricated with Lubricant C, both P_{spin} and P_m are minimum, resulting in the greatest difference in P_T between Lubricant C and Lubricant A.

Finally, influence of lubricant on P_T for the 17/26T hobbled-shaved, and hobbled-shaved plus chemically polished gears is shown in Fig. 3.23. Due to the large difference

Figure 3.22 Comparison of measured P_T for 23T and 40T, and ground and chemically polished gear pairs in Lubricants A, B, and C; (a) 23T ground gears, (b) 23T chemically polished gears and, (c) 40T ground gears, and (d) 40T chemically polished gears.

Figure 3.23 Comparison of measured P_T for 17/26T hobbed and shaved, and hobbed and shaved plus chemically polished gear pairs in Lubricant D at 90C and Lubricant A at 48C; (a) hobbed and shaved gears, (b) hobbed and shaved plus chemically polished gears.

in P_{spin} , both hobbed-shaved and hobbed-shaved plus chemically polished gears performed best in Lubricant D. The baseline hobbed-shaved gears had lower P_T with Lubricant A at $T_c = 413$ N-m for all Ω tested. In this case, reduction of P_m with Lubricant A due to higher film thickness offset increases in P_{spin} due to higher ν .

It was observed in Section 3.4 that P_{spin} increases with $\Omega^{1.4-1.7}$, while P_m increases linearly with Ω . Therefore, P_{spin} comprises a larger percentage of P_T at high speed than at low speed, and it becomes instructive to compare the magnitudes of the components of power loss for the range of operating conditions studied. The percentage of P_{spin} attributed to P_T is largest (51%) at low torque and high speed ($T_c = 413$ N-m, $\Omega = 10,000$ rpm), and lowest (28%) at high torque and low speed ($T_c = 684$ N-m, $\Omega = 6,000$ rpm). To demonstrate the effect of P_{spin} on total efficiency, $\bar{\eta}_T$ as calculated by Eq. (3.11a) is shown in Fig. 3.24 for the same test conditions considered in Fig. 3.3. Here, the first three groups correspond to $\Omega = 6,000$ rpm and a range of T_c from 413 N-m to 684 N-m. The next three groups correspond to the same values of T_c , but at $\Omega = 8,000$ rpm, and the last three groups correspond to $\Omega = 10,000$ rpm. It is observed here that $\bar{\eta}_T$ increases as T_c increases for any given speed because the load-dependent (mechanical) power loss increases, but the load-independent (spin) power loss remains the same. Conversely, as Ω increases for any given T_c , $\bar{\eta}_T$ decreases due to the non-linear relationship between Ω and P_{spin} .

Figure 3.24 Measured $\bar{\eta}_T$ values of 23T and 40T ground gear pairs in Lubricant A at various test conditions listed in Table 2.3.

Unlike $\bar{\eta}_m$, which remained relatively constant, $\bar{\eta}_T$ varies more widely due to the inclusion of load-independent power loss. Since P_{spin} is lower at low Ω , $\bar{\eta}_T$ is found to be highest under high torque, low speed conditions, and lowest under low torque, high speed conditions. For instance, the 23T ground gears in lubricant A operated with $\bar{\eta}_T = 99.47\%$ at $T_c = 684$ N-m and $\Omega = 6,000$ rpm, but at $\bar{\eta}_T = 99.21\%$ at $T_c = 413$ N-m and $\Omega = 10,000$ rpm. Similarly, the 40T ground gears operated at $\bar{\eta}_T = 99.56\%$ and 99.41% under the same respective operating conditions.

Figure 3.25 is intended to serve as a summary of the influences of m , surface roughness amplitude, and lubricant type on $\bar{\eta}_T$. Although $\bar{\eta}_T$ is not constant with Ω , the influences of Ω and ν on P_{spin} are well defined and consistent for 23T and 40T gear designs. Therefore, an average $\bar{\eta}_T$ value over all test conditions considered represents the overall performance of any gear design, surface finish and oil relative to the others. Here, 40T chemically polished gears lubricated with Lubricant C outperformed all other gears in terms of P_T with $\bar{\eta}_T = 99.60\%$, while 23T ground gears lubricated with Lubricant A performed the worst with $\bar{\eta}_T = 99.35\%$, resulting in a possible 0.25% overall efficiency improvement.

It is important to note here the relative importance of the parameters studied when considering $\bar{\eta}_T$ versus $\bar{\eta}_m$. Comparing Fig. 3.19 to Fig. 3.25, it is immediately obvious that m has the largest impact on both $\bar{\eta}_T$ and $\bar{\eta}_m$ under the operating conditions studied. The reason for this is that m is an important parameter in P_m due to its influence on

Figure 3.25 Comparison of average measured $\bar{\eta}_T$ for 23T and 40T, ground and chemically polished gear pairs lubricated with Lubricants A, B, and C.

sliding velocities and P_{spin} due to its influence on drag losses from the gear teeth. The influence of surface roughness amplitude is the second largest indicator of $\bar{\eta}_m$ due to sliding friction arising from asperity contacts, but it is of tertiary importance in $\bar{\eta}_T$. The reason for this is that P_{spin} is heavily influenced by lubricant ν .

3.6 Summary

An experimental test matrix including gears of different modules and surface roughness levels operating under jet lubrication conditions with four different gear lubricants was developed to quantify the influence of these parameters on load-dependent (mechanical) and load-independent (spin) power losses. Experimental data indicate that gear mesh mechanical power losses increase nearly linearly with both rotational speed and applied torque, indicating that gear mesh mechanical efficiency is approximately constant over the ranges of speed and load considered. Experimental results show that gear module is the most influential parameter on gear mesh mechanical efficiency for the operating conditions considered, followed by gear tooth surface roughness. It was shown that gear mesh mechanical power loss for all ground gears was minimized with the most viscous lubricant due to reduction in sliding power losses through decreased asperity contacts. For chemically polished gears, the 40T gears performed best in terms of mesh power losses with the medium-viscosity lubricant, while the 23T gears performed best with the lowest viscosity lubricant due to minimization of both rolling and sliding power losses at the gear mesh.

Experimental results for four different lubricants show that load-independent gearbox (spin) power losses are influenced heavily by lubricant viscosity and to a lesser extent gear module, and can comprise a significant portion of overall gearbox power loss. Due to the inclusion of spin losses, total gearbox power loss was minimized for both ground and chemically polished gear pairs with the lowest-viscosity lubricant studied, despite differing trends in mesh mechanical power loss.

Gear mesh mechanical power losses have been discussed in terms of rolling and sliding power losses, but has been on basis of postulation since these power losses can not be separated experimentally in a geared system. It is therefore desirable to define these sources of power loss on a physical basis and perform efficiency predictions in order to quantify their individual contributions to power loss in a gear mesh.

CHAPTER 4

EFFICIENCY EXPERIMENTS UNDER DIP-LUBRICATED CONDITIONS

4.1 Introduction

Up until this point, the method of gear lubrication discussed has been dry sump jet-lubrication, with the oil jet directed at the exit of the gear mesh zone opposite the sense of rotation. While this lubrication method is commonly applied in high-speed gearing applications such as helicopter gearboxes as well as automotive automatic transmissions, it is not used for many low to medium-speed applications where an external oil pump cannot be afforded. Many industrial gearboxes, automotive manual transmissions, rear and front-axle differentials, and transfer cases all employ dip lubrication. In this lubrication method, the gear housing is partially filled with oil. All or some of the gears rotate partially submerged in oil, causing significant amount of splashing and churning of oil. This results in load-independent spin power loss that is typically larger than those for jet-lubricated gear sets. This chapter aims at experimentally investigating the spin losses of dip-lubricated gear pairs.

A set of simple modifications to the gear efficiency test machine used in the previous chapters allows dip lubricated spin power loss to be quantified under the same conditions as jet-lubricated efficiency experiments using the same gear specimens, test fixtures, and instrumentation. This provides a direct quantitative comparison between spin power loss associated with both lubrication methods. The objective of this chapter is to experimentally quantify the influence of several operating conditions on dip-lubricated spin power loss, and discuss the compatibility of previous churning power loss studies with the present experimental results. The experimental results presented here represent the initial phase of an extensive spur gear efficiency study. The primary objective here is to identify overall trends in measured dip lubricated spin power loss, while detailed analysis is left to future work.

4.2 Description of Dip-Lubricated Test Setup

Figure 4.1 shows a picture of the modified test-side gearbox. Same modifications were applied to the reaction-side gearbox as well. For the dip-lubrication power loss study, the oil supply jets to the bearings were removed and the ports were plugged. Oil supply jets directed into the gear mesh were retained for filling the gearboxes with oil between tests and represent a minimal flow obstruction within the gearbox. Ball valves of the oil return lines were closed in order to maintain an oil bath in the gearboxes, and were positioned such that the static oil levels in the gearboxes were equal via a flow path between the gearboxes. Sheet metal baffle plates were added inside the gearboxes in front of the oil drainage ports to minimize oil circulation in the drainage line. Large diameter graduated site glasses were added to the gearboxes to document static oil level

Figure 4.1 Modifications to the jet-lubricated gear efficiency test machine to allow dip-lubricated spin power loss studies.

within the gearbox. Ball valves were placed between the gearbox and site glasses. These ball valves were closed during machine operation in order to ensure that the site glasses did not act as fluid capacitors.

4.3 Test Specimens and Test Matrix

Previous studies on oil churning power loss [37-41] reported conflicting results regarding the influence of operational parameters such as the influence of rotational speed, lubricant viscosity, and static oil bath level. The objective of this study is to quantify these influences using the test machine in hand. Any possible influences of gear surface roughness on oil churning will be excluded here by simply using only the chemically polished test gears. Additionally, all gears used in this study have a gear ratio of unity that eliminates to certain extent the oil swell effect observed in some of the earlier studies [41]. This has the advantage of allowing the influences of the parameters under study to be quantified directly, as well as simplifying the validation of future analytical modes.

In order to investigate the influence of immersion depth oil level on dip-lubricated spin power loss, tests were performed at four different oil levels identified by a h/r ratio, where h is the oil level measured from the pitch point at the bottom of the test gears, and r is the pitch radius, shown schematically in Fig. 4.2. Here $h/r = 0$ indicates that the static oil level is at tangent to the pitch circles of both gears in the bottom, i.e. only the addendums of both gears are submerged in the oil. Meanwhile, $h/r = 1.0$ means the static oil level is as high as gear centers such that exactly half of both gears are within the oil initially.

Figure 4.2 A test gearbox in the dip-lubrication arrangement.

Rotational speeds of $\Omega = 1000$ to 7,000 rpm were considered for $h/r = 1.0$ in order to quantify the dependence of spin loss on rotational speed. For other h/r values of 0, 0.5 and 1.5, rotational speed values of 1,000, 2,000, 4,000, and 6,000 were selected due to time constraints. At $h/r = 1.5$, the 6,000 rpm, test point was replaced with a 3,000 rpm test point due to concerns of oil foaming. Table 4.1 shows the test matrix for the dip lubricated spin power loss study. A typical manual transmission fluid (identified in this study as Lubricant E) was used in all of these tests. The viscosity-temperature relationship of this lubricant is shown in Fig. 4.3. Tests were performed within an operating temperature of 30 to 100°C so that the influence of oil viscosity on dip lubricated spin power loss can be quantified. The torque loss $\hat{T}_L$ was measured continuously as the temperature of the oil changes in time so that real-time torque loss-temperature data can be captured.

Since an active control of the oil temperature was not sought in the tests, frictional heating of the oil bath was relied upon to vary operating oil temperature, and hence lubricant viscosity. Since gear box heat balance results in different steady-state temperatures at different h/r and Ω values, it was not possible to obtain power loss data at particularly high or low temperatures for some of the test conditions. For instance, rapid frictional heating of the oil bath at 6,000 rpm prevented the acquisition of torque data below 50°C. Similarly for low speed tests, say $\Omega = 1,000$ rpm, frictional heating of the gearboxes was not sufficient to reach the temperatures beyond 90°C. Figure 4.4 shows the oil temperature-time characteristic of the gearboxes at various Ω values with

Test Number	# of Teeth	Face Width	h/r	Rotation Direction	Purpose
1	40	19.5 mm	1.5	Up in Mesh	Viscosity-Speed Characteristic
2			1.0		
3			0.5		
4			0.0		
5	40	19.5 mm	1.0	Up in Mesh	Repeatability
6	40	19.5 mm	1.0	Down in Mesh	Influence of Rotational Direction

Table 4.1 Spur gear dip-lubricated spin power loss experimental test plan.

Figure 4.3 Viscosity-temperature characteristic for Lubricant E.

Figure 4.4 Variation of oil temperature as a function of time for several values of Ω . 40T chemically polished, 19.5 mm face width gears rotating upward in the oil bath.

40T, 19.5 mm face width gears operating at $h/r=1.0$. Here it is evident that for most rotational speeds, it was not possible to obtain torque data over the full temperature range by relying on frictional heating alone. However, since a real-time torque-temperature signal was acquired by the data acquisition system, torque data was also acquired while the system was cooling from high temperatures achieved by operating at a high speed. For the low-speed tests, the gearboxes were preheated to nearly 100°C by preceding these tests with high-speed tests. In this fashion, by alternating high-speed and low-speed tests, data for the complete range of temperature was collected by simply relying on frictional heating alone.

4.4 Accuracy and Repeatability

For data processing purposes, the continuous torque-temperature data was simplified into discrete data points by averaging torque loss and oil sump temperature in 5°C increments. The results of this discretization is shown in Fig. 4.5 for 40T, 19.5 mm face width gears rotating upward in the oil bath. Figure 4.5 shows that such discretization is quite accurate. For the remainder of this chapter, all dip lubricated spin loss data will be displayed in the discretized format.

Test 5 in Table 4.1 is a repeat of test 2 to demonstrate the repeatability of the test machine and fixtures using 40T, 19.5 mm face width chemically polished gears with $h/r=1.0$ and upward gear rotation in the oil bath. Figure 4.6 demonstrates the repeatability of the test machine over a wide range of rotational speed and oil kinematic viscosity due to changes in temperature. These torque measurements represent over 23 hours of data collection, or nearly 84,000 individual torque measurements. Figure 4.6

Figure 4.5 Discretization of continuous temperature-torque characteristic. 40T chemically polished, 19.5 mm face width gears rotating upward in the oil bath.

Figure 4.6 Repeatability of measured dip-lubricated spin power loss for 40T chemically polished gears as a function of oil bath kinematic viscosity, 19.5 mm face width, rotating upward in the oil bath.

shows repeatability of the data for Ω in the range of 1,000 to 5,000 rpm is very good, while it is not as good at 6,000 and 7,000 rpm, probably due to a high degree of oil/air turbulence inside the gearbox. Since dip lubricated systems are rarely operated at such high rotational speeds, the repeatability of the test fixtures can be considered satisfactory for this study.

4.5 Experimental Dip Lubricated Spin Power Loss Results

Influence of Rotational Speed and Oil Level

Figures 4.7 to 4.10 show measured P_{spin} values as a function of oil temperature at various Ω for $h/r = 0, 0.5, 1.0$ and 1.5 , respectively. These figures represent 40T, 19.5 mm face width gear pairs rotating upward in the oil bath. The P_{spin} values at discrete oil bath temperatures of 53, 73, and 88°C are extracted from these figures to illustrate the influence of Ω in Fig. 4.11. Here, P_{spin} increases with Ω to the power 1.35 to 2.05, depending on h/r and oil temperature. The exponent on Ω appears to generally increase with oil bath level. For instance, at 73°C, $P_{spin} \sim \Omega^{1.39}$ for $h/r = 0$, $P_{spin} \sim \Omega^{1.72}$ for $h/r = 0.5$, $P_{spin} \sim \Omega^{1.93}$ for $h/r = 1.0$, and $P_{spin} \sim \Omega^{1.97}$ for $h/r = 1.5$. This general trend is consistent over the full temperature range for which P_{spin} data was available. Additionally, the exponent on Ω appears to increase slightly with oil bath temperature, or rather increase as oil bath viscosity decreases. As an example, for $h/r = 0$, the exponent on Ω increased from 1.24 at 38°C to 1.47 at 88°C.

Figure 4.7 Measured dip-lubricated spin power loss for 40T chemically polished gears as a function of oil bath temperature, 19.5 mm face width, rotating upward in the oil bath. $h/r = 0.0$.

Figure 4.8 Measured dip-lubricated spin power loss for 40T chemically polished gears as a function of oil bath temperature, 19.5 mm face width, rotating upward in the oil bath. $h/r = 0.5$.

Figure 4.9 Measured dip-lubricated spin power loss for 40T chemically polished gears as a function of oil bath temperature, 19.5 mm face width, rotating upward in the oil bath. $h/r = 1.0$.

Figure 4.10 Measured dip-lubricated spin power loss for 40T chemically polished gears as a function of oil bath temperature, 19.5 mm face width, rotating upward in the oil bath. $h/r = 1.5$.

Figure 4.11 Measured P_{spin} as a function of Ω for four oil levels; (a) 53°C oil bath temperature, (b) 73°C oil bath temperature, and (c) 88°C oil bath temperature. 40T chemically polished gears, 19.5 mm face width, rotating upward in the oil bath.

(continued)

Figure 4.11 continued

(continued)

Figure 4.11 continued

Similarly, for oil levels at the other extreme, $h/r = 1.5$, the exponent on Ω increased from 1.89 to 2.05 for the same temperature range.

Figure 4.11 also shows the influence of oil bath level on P_{spin} for three different oil bath temperatures. The general trend here is that P_{spin} decreases as the oil bath level decreases. Taking $h/r = 1.0$ as the baseline oil level and averaging over the entire range of Ω and oil bath temperatures considered, P_{spin} is reduced by 26% by reducing the oil level to $h/r = 0.5$, and by 49% by reducing the oil level to $h/r = 0.0$. Likewise, by increasing the oil level to $h/r = 1.5$, P_{spin} increased by 23%. Comparing P_{spin} for the highest oil level, $h/r = 1.5$, to the lowest oil level, $h/r = 0$, it is seen that P_{spin} is approximately four times as large for $h/r = 1.5$ at the highest comparable speed of $\Omega = 4,000$ rpm, and 1.7 times as large at $\Omega = 1,000$ rpm.

Influence of Lubricant Viscosity and Rotational Direction

Figure 4.12 shows directly the influence of oil bath viscosity ν on measured P_{spin} for a range of Ω using 40T, 19.5 mm face width gears at $h/r = 1.0$ for both upward and downward rotation in the oil bath. Here, P_{spin} is observed to be weakly dependent on ν , and is somewhat linear. This observation agrees with the results of references [40, 41], who reported that P_{churn} appears to be weakly dependent on ν , especially at high Ω . For all oil levels at low Ω , P_{spin} decreased by as much as 50% by reducing ν from 140 cSt to 18 cSt, representing a typical savings of 0.04-0.05 kW. At

higher Ω , 4,000 rpm for instance, the dependence of P_{spin} on ν was less pronounced, with reductions in P_{spin} of approximately 5 to 20% over the same viscosity range, depending on oil bath level. This represents a power savings of 0.01-0.16 kW, depending on oil bath level.

Figure 4.12 shows the influence of rotation direction on P_{spin} for a range of Ω and oil bath temperatures using 40T, 19.5 mm face width gears at $h/r = 1.0$. Here, it is shown that the direction of rotation has a quantifiable influence on P_{spin} , with upward direction in the oil bath resulting in higher P_{spin} . The effect of the rotational direction of gears on P_{spin} is rather negligible for low Ω , while the difference becomes more significant as Ω increases. Averaging over all test conditions considered, upward rotation in the oil bath resulted in 15% higher P_{spin} than for downward rotation in the oil bath for $h/r = 1.0$. This phenomenon was also observed in references [37, 41], and can be attributed to a larger magnitude of P_{trap} for the case of upward rotation in the oil bath. This can be explained by the higher density and viscosity of the lubricating oil compared to air. For the case of upward rotation gears in the oil bath, especially for higher h/r values, the fluid pumping and trapping of the meshing gear teeth consists primarily of the lubricating oil. For the case of downward rotation in the oil bath, the fluid in the meshing zone probably consists of a higher percentage of air than for the former case because oil is flung off of the gear teeth as they exit the bath [52-53]. By the time the gear teeth come back into mesh, there is only a thin film of oil adhering to the surface, and so P_{trap} consists mainly of power loss due to pumping of air.

Figure 4.12 Influence of rotation direction on measured P_{spin} for 40T chemically polished gears, 19.5 mm face width.

Comparison to Jet Lubricated Spin Power Loss

The method of jet-lubrication with a dry sump is generally accepted as a more efficient method of lubrication than dip lubrication due to lower magnitudes of P_{trap} and P_{churn} , but little experimental data is available to support this claim. As the oil bath level in a dip-lubricated system is lowered, the magnitudes of P_{trap} and P_{churn} should approach that of a jet-lubricated system. That is, the effects of gear windage become more dominant as oil level lowers. While this argument make sense, no experimental proof of it has been provided. This section attempts to make direct comparisons between the measured P_{spin} values the same test gears operated under both jet and dip-lubricated conditions.

To demonstrate the difference in P_{spin} under jet and dip-lubrication conditions using the same test fixtures, a jet-lubricated spin power loss was conducted using the manual transmission fluid at $110^{\circ}C$ with 40T, 26.7 mm face width chemically polished gears. Figure 4.13 compares P_{spin} under these jet-lubricated conditions to dip-lubricated P_{spin} values at $h/r=0$ and $88^{\circ}C$ $88^{\circ}C$. Although the oil temperatures for these two tests were not the same, the difference in ν was small (about 6 cSt). Figure 4.13 shows no practical difference in measured P_{spin} for dip-lubricated and jet-lubricated gear pairs with the oil bath level at the gear pitch line, $h/r=0$. The dip-lubricated P_{spin} values in Fig. 4.13 correspond to the $h/r=0$ values in Fig. 4.7(c). This result suggests that the

Figure 4.13 Comparison of jet-lubricated spin power loss for 26.7 mm face width gears at 110°C to dip-lubricated spin power loss for $h/r = 0.0$, 88°C , 19.5 mm face width, upward rotation in the oil bath.

mechanisms of power loss are very similar for low oil level dip-lubrication and-jet lubrication.

4.6 Summary

The gear efficiency test machine presented in Chapter 2 was modified for dip-lubricated spin power loss measurements, allowing direct comparison of load independent power loss for dip-lubricated or jet-lubricated conditions using the same test fixtures. An experimental test matrix including unity ratio gears operating in an oil bath of four different levels for a range of rotational speed and oil viscosity was developed. Satisfactory repeatability of the test fixtures was established for rotational speeds less than 5,000 rpm. The influence of rotational speed, oil viscosity, oil bath level, and rotational direction on load independent power loss was quantified.

Experimental data indicate that dip-lubricated spin power loss can be several times higher than jet-lubricated spin power loss under similar operating conditions. It was shown that dip-lubricated spin power loss increases with rotational speed to the power 1.35-2.05, depending on oil bath level, with the higher exponents corresponding to higher oil bath level. This dependence on speed is consistent with the jet-lubricated spin power loss measurements presented in Chapter 3, as well as the measurements of previous researchers [26, 41], but does not agree with the widely accepted windage models of Dawson [27-28] and Anderson and Loewenthal [5], or the more recent windage model of Diab *et al* [29]. Experimental results also show that dip-lubricated spin power loss decreases with oil bath level, and is weakly dependent on oil viscosity. Lastly, dip-lubricated spin power loss for the case of the oil bath level at the lowest point

of the gear pitch point is found to be nearly identical to the jet-lubricated spin power loss under similar conditions.

CHAPTER 5

SUMMARY AND CONCLUSIONS

5.1 Summary

An experimental investigation of high-speed spur gear efficiency was conducted for both jet-lubricated dry sump conditions and dip-lubricated conditions. Inspection methodologies were developed for the documentation of gear surface roughness and wear after each test. Results show that after the initial run-in phase, gear surface roughness amplitudes remained constant throughout the test regimen, and CMM inspections show negligible wear for all gear surfaces. An experimental test matrix including gears of different modules and surface roughness levels operating under jet-lubrication conditions with four different gear lubricants was developed to quantify the influence of these parameters on load-dependent (mechanical) and load-independent (spin) power losses.

The spur gear efficiency test machine was modified for dip-lubricated spin power loss measurements, allowing direct comparison of load-independent power loss for dip-lubricated or jet-lubricated conditions using the same test fixtures. An experimental test matrix including unity ratio gears of different module and face width operating in an oil

bath of four different levels for a range of rotational speed and oil viscosity was developed. The influence of rotational speed, oil viscosity, oil bath level, and rotational direction on load-independent power loss was quantified.

5.2 Conclusions

The extensive experimental study of spur gear efficiency under highly loaded conditions and unloaded conditions for both jet-lubrication and dip-lubrication has resulted in the following conclusions:

- Gear mesh mechanical power losses increase nearly linearly with both rotational speed and applied torque, indicating that gear mesh mechanical efficiency is approximately constant over the ranges of speed and load considered. Gear mesh mechanical efficiency calculations from measured data support this conclusion.
- Gear module is the most influential parameter on gear mesh mechanical efficiency for the operating conditions considered, followed by gear tooth surface roughness, then lubricant selection.
- Gear mesh mechanical power loss for all ground gears was minimized with the most viscous lubricant due to reduction in sliding power losses through decreased asperity contacts. For chemically polished gears, the fine pitch gears performed best in terms of mesh power losses with the medium-viscosity lubricant, while

the coarse pitch gears performed best with the lowest viscosity lubricant due to minimization of both rolling and sliding power losses at the gear mesh.

- Experimental results for four different lubricants show that load-independent gearbox (spin) power losses are influenced heavily by lubricant viscosity and to a lesser extent gear module, and can comprise a significant portion of overall gearbox power loss.
- Due to the inclusion of spin losses, total gearbox power loss was minimized for both ground and chemically polished gear pairs with the lowest-viscosity lubricant studied, despite differing trends in mesh mechanical power loss.
- Dip-lubricated spin power loss can be several times higher than jet-lubricated spin power loss under similar operating conditions.
- Dip-lubricated spin power loss increases with rotational speed to the power 1.35-2.05, depending on oil bath level, with the higher exponents corresponding to higher oil bath level. Jet-lubricated spin power loss increases with speed to the power 1.4-1.7 for both fine and coarse pitch gears.
- Dip-lubricated spin power loss decreases with oil bath level, and is weakly dependent on oil viscosity.

- Dip-lubricated spin power loss for the case of the oil bath level at the lowest point of the gear pitch point is found to be nearly identical to the jet-lubricated spin power loss under similar conditions.

5.3 Recommendations for future work

Although this experimental investigation of spur gear efficiency was extensive, it was by no means adequate to cover all ranges of gearing applications available. As the topic of gear efficiency becomes more crucial, especially in high-power applications, it will become necessary to expand on some of the objectives in this study. The following are some recommendations for future work:

- *Renew attempts to apply DLC coatings:* The low friction DLC coatings applied in this study failed to endure even several hours of operation. Since such coatings have been shown to reduce dry friction, efforts should be made to improve coating durability enough that they maintain integrity throughout the life of the gear.
- *Expand efficiency testing to include helical and hypoid gears:* Helical and hypoid gears are widely used in automotive applications, as well as in industry. Up until this point, only the efficiency of spur gears has been considered. The test methodology and test machine have been proven to be highly accurate and well suited for efficiency testing of spur gears. Simple modifications to the machine

can expand testing to other parallel axis gears. Additional modifications to the machine are possible to allow testing of cross-axis gears such as hypoid gears.

- *Expand dip-lubricated spin power loss test plan:* Load independent power losses have been shown to represent a significant portion of overall gearbox power loss, and deserve more detailed study. In particular, the influence of module and face width on dip-lubricated spin power loss still need to be quantified, as well as the oil swell effect caused by non-unity ratio gears. Also of particular interest is the influence of enclosure effects on spin power loss. Enclosure effects have been shown to play an important role in windage, but have not been quantified for dip-lubricated gearing.
- *Quantify gear mesh mechanical power loss under dip-lubricated conditions:* Frictional power loss in the gear mesh has been quantified for the case of jet-lubrication. However, no experimental studies have investigated whether gear mesh mechanical power loss is the same under dip-lubrication conditions. This is an important step in populating a comprehensive gear efficiency database, and requires no further modification of the existing test fixtures.
- *Develop a physics-based spin power loss model:* Theoretical models for load-dependent power loss have become increasingly sophisticated and have been shown to agree quite well with experimental data, but no such model exists for load-independent spin power loss. Existing empirical formulae for both windage

and oil churning do not agree with experimental data, so calculation of total gearbox power loss cannot be accomplished with confidence. Therefore, there is a need for a physics-based model that applies to both jet and dip-lubrication conditions.

In addition to these suggestions for future experimental and theoretical studies, several modifications to the existing spur gear efficiency test machine can improve experimental accuracy, as well as expand on the capability of the test machine. Suggestions for improving the experimental test machine are as follows:

- *Develop a methodology to precisely measure loop torque:* For this study, loop torque was calculated based on dead weight and moment arm of the torque arm, but was not measured directly. Since load-dependent gear efficiency is not particularly sensitive to load, this is not a major concern. However, better documentation of experimental parameters will lead to more accurate results.
- *Experimentally quantify friction torque of bearings and shaft seals:* The dip-lubricated spin power loss measurements include power losses resulting from viscous friction in the bearings and shaft seals. Formulae exist for calculating these values, but their accuracy is unknown. In order to validate a physics-based spin power loss model, the magnitudes of these power losses must be accurately known.

- *Employ temperature control for dip-lubricated efficiency studies:* The jet-lubricated loaded efficiency studies employed temperature control in the form of a temperature controlled oil jet. This allowed the test machine to operate at a user-defined steady-state temperature for an extended period of time. In order to obtain similar data under dip-lubrication, it will be necessary to regulate the temperature of the oil bath. This will also permit acquisition of dip lubricated spin power loss data for temperature ranges that were not possible in this study.

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