

**DETC2007-35045**

## EXPERIMENTAL INVESTIGATION OF SPUR GEAR EFFICIENCY

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### ABSTRACT

In this study, a test methodology was developed for measurement of spur gear efficiency under high-speed and variable torque conditions. A power-circulating test machine was designed to operate at speeds to 10,000 rpm and transmitted power levels to 700 kW. A precision torque measurement system was implemented and its accuracy and repeatability in measuring torque loss in the power loop was demonstrated. Tests were conducted on gears with two values of module, and two surface roughness levels, operating in a dry sump jet-lubrication environment with three different gear lubricants. These tests were used to quantify the influence of these parameters on load-dependent (mechanical), load-independent (spin), and total power loss. Trends in mechanical gear mesh efficiency and total gearbox efficiency were discussed in terms of rotational speed and transmitted torque. Finally, recommendations were made for the design of spur gear pairs, surface roughness, and lubricant selection for improved efficiency.

### 1 INTRODUCTION

Efficiency of spur and helical gear systems has become an increasingly important research topic as the fuel economy requirements for today's passenger vehicles and rotorcraft are more stringent, not only due to fuel cost, but also environmental concerns associated with energy utilization and air pollution. Improved gear system efficiency also results in less frictional heat generation within the gearbox, resulting in improvements in gear failure modes such as scoring and pitting, and lower-capacity lubrication systems.

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. Since a mechanical efficiency model requires a friction coefficient model 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 constant friction coefficient  $P$  along the entire contact surface gears. Another group of models [5-10] adapted the same approach with the exception that their formulation for  $P$  was not constant, but was based on various empirical formulae obtained from twin-disk tests. Most recently Diab *et al* [10] used such friction formulae to compute the dynamic friction forces and power loss in a spur gear mesh. Physics based elastohydrodynamic lubrication (EHL) models of varying complexity [11-16] were developed in order to compute  $P$  without the need of empirical formulae. 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 coefficients were calculated. Rough-surface EHL models were used 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. While these models were sophisticated in handling EHL aspects of the problem in the gear mesh, they did not consider load-independent power loss, and are thus insufficient to predict total gearbox efficiency. Furthermore, most of these models, perhaps with the exceptions of Diab *et al* [10] and Xu *et al* [16], were not validated.

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 type [17-24]. Of these studies, Naruse *et al* [17-18] 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. Naruse *et al* [17-18] also reported that gear mesh frictional power loss increases with transmitted torque at low torque values, but becomes nearly constant at higher loads. Lastly, Naruse *et al* [17-18] showed that reducing module can have a significant influence on efficiency losses. These studies also showed that efficiency losses are insensitive to changes in surface roughness between 0.5  $\mu\text{m}$  and 3  $\mu\text{m}$  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* [19] demonstrated that below 10 cSt, gear tooth friction coefficient decreases as lubricating oil viscosity increases, while friction coefficient changes little as viscosity increases beyond that point. Ikejo and Nagamura [20] also claimed based on measurements with a similar test setup that gear mesh frictional loss is weakly dependent on oil viscosity, and decreases slightly with increasing viscosity. Conversely in terms of total power loss, Coy *et al* [21] reported that the overall efficiency of a helicopter transmission generally decreased with increasing viscosity.

Martins *et al* [22] measured average friction coefficient of dip-lubricated FZG gears and reported that friction coefficient decreases with increasing 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* [23] 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* [24] reported that reduction of average surface roughness height from 0.4 to 0.05  $\mu\text{m}$  by an isotropic chemical polishing technique reduced gear 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 high-performance applications that are more likely to take advantage of a chemical polishing process.

Given this limited volume of experimental data, there is a need for an extensive database for spur gear efficiency, not only to validate the mathematical efficiency models but also to experimentally quantify the influence of gear module, lubricant and surface related parameters on spur gear efficiency using the same test method and test gears. 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 precision experiments is the first objective 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. Numerous experimental studies [5-6,20,25-38] have shown that spin losses can be significant, especially under high-speed operating conditions. Modifications to a gearbox to minimize mechanical power loss might also increase spin losses, resulting in larger overall power loss. Therefore, it is critical that both mechanical and spin components of gear power loss be considered simultaneously in an effort to maximize efficiency.

Sources of load-independent spin power loss include air windage, oil churning for the case of dip lubrication, inertial power loss from an impinging oil jet for the case of jet lubrication, fluid trapping and squeezing between meshing gear teeth, rotating seals, and bearings. Various models have been developed to quantify these sources of power loss.

The windage models of Dawson [25] and Diab *et al* [26] 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* [26] 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 [5-6] developed a model for meshed spur gears based on pitch radius, face width, rotational speed, and viscosity of the ambient fluid. All these models resulted in a nearly cubic relationship between rotational speed and power loss.

Ariura *et al* [27] 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* [27] using a jet lubricated spur gear system led to the observation that inertial power losses are linearly related to rotational speed. Ariura *et al* [27] 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 [28] 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 [29]. This model elucidated the phenomenon of eddy current flow between gear teeth and introduced a vectorial model for penetration depth of impinging fluid on the rotating gear teeth. Later, Pechersky and Wittbrodt [30] 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* [31] 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.

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 [32-33]. Like Diab *et al* [26] for the case of windage, the oil churning studies were mostly devoted to developing empirical equations for the dimensionless moment coefficient. Daily and Nece [32] 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 [33] 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.

Boness [34] 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. Terekhov [35] 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. More recent efforts using similar methods include Höhn *et al* [36], Luke and Olver [37], and Changenet and Vexel [38]. The model of Changenet and Vexel [38] investigated the influence of a meshing gear of both unity and non-unity ratio on oil churning power loss. Load independent power losses due to bearings and shaft seals can easily be approximated using equations found in [39] and [40], respectively.

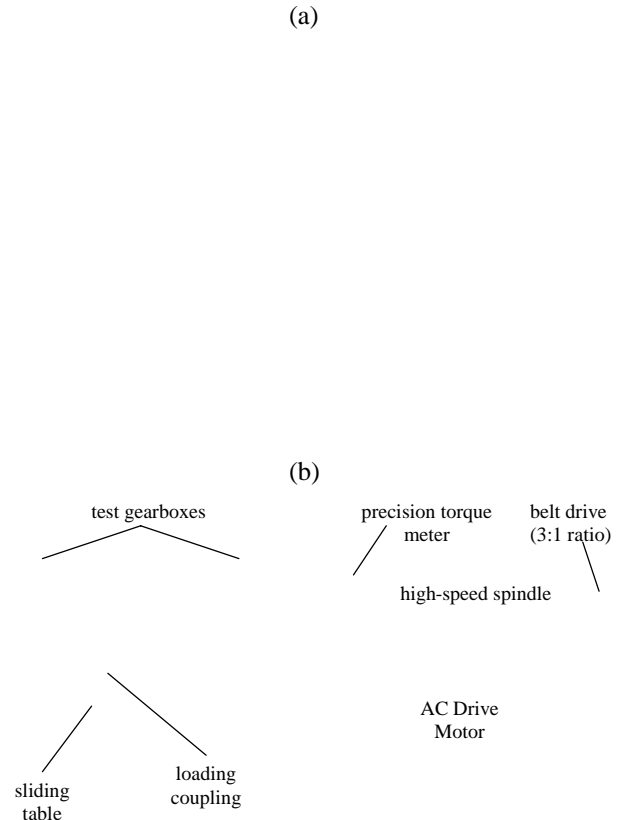
Most of these spin power loss models were developed from specialized test rigs intended to study only a specific component or components of spin power loss. Therefore, there is very little experimental data available to quantify both spin and mechanical losses of the same gear system simultaneously. The second objective 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. At the end, conclusions will be made in regards to the influence of surface roughness, lubricant type and gear parameters on components of spur gear efficiency.

## 2 MEASUREMENT OF SPUR GEAR EFFICIENCY

### 2.1 Test Machine and Test Procedure

The spur gear efficiency test machine used in this study, shown in Fig. 1(a), was designed and procured as part of an earlier spur gear efficiency investigation [41]. All of the essential details of this test machine will be provided here. Additional information about it can also be found in Ref. [16]. The test machine consists of two identical gearboxes, each containing one gear mesh, four identical cylindrical roller bearings, and five 30 mm shaft seals. This machine utilizes a *four-square* type, power circulation concept such that one gear from each gearbox is connected to the corresponding gear of the other gearbox, as illustrated in the schematic of the machine layout shown in Fig. 1(b). 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 closed loop through a torque arm and calibrated dead weights. This torque is transmitted by both test gear meshes.

A flexible coupling connects the input shaft of the reaction gearbox to a high-speed spindle through a precision non-contact type torque-meter. The high-speed spindle is driven by a belt with a 3:1



**Fig. 1 (a) The high-speed spur gear efficiency test machine, and (b) a schematic layout of the machine.**

ratio speed increase from a variable speed AC motor. A separate temperature-controlled oil circulation system supplies oil through directed jets to the gear meshes at 2.25 Lpm and bearings at 1.25 Lpm. Oil temperature is measured with type-J thermocouples at both the supply and return points to the gearboxes. The test machine is capable of operating at a rotational speed of : 10,000 rpm, which corresponds to a gear pitch-line velocity of 48 m/s for unity ratio gears given the fixed center distance of 91.5 mm. Maximum circulation loop torque  $T_c$  is 680 N-m (500 ft-lb). This results in a circulation loop power  $P_c = T_c \omega$  ( $\omega = \frac{2\pi S}{60}$ ) of: over 700 kW (950 hp), representative of commercial automotive, racing and some aerospace applications.

A run-in cycle formed by one hour of operation at  $T_c = 413$  N-m and : 6,000 rpm was run when new gears and/or bearings were used for the first time. An additional two-hour cycle at  $T_c = 0$  and : 6,000 rpm was used to run-in new shaft seals. Previous experimental studies in gear efficiency using the same machine indicated that most transient temperature behavior was eliminated in the first 5 minutes of operation [41] due to preheating the oil and test fixtures before beginning a test. Accordingly, the test duration was selected as 10 minutes, with the last 5 minutes of data used to calculate average values for oil supply temperature, : , and torque  $T_T$ . Here,  $T_T$  is the total torque provided to the closed loop by the external AC drive through the torque-meter to maintain an operation at the set : value. Therefore,  $T_T$  represents the total torque loss of the closed loop. Lastly, after the test was completed, the test

apparatus was disengaged from the flexible coupling, and any electronic drift of the torque-meter was recorded. Electronic drift in the torque signal occurs partly due to thermal expansion of the high-speed spindle during the test, and was subtracted from the measured  $T_T$  in order to obtain the actual input torque. This drift torque was typically less than 0.3-0.4% of the measured  $T_T$  value.

## 2.2 Gear Specimens, Test Matrix and Repeatability

A total of four sets of spur gears, shown in Fig. 2, were used in the test matrix shown in Table 1. Each test gear set is formed by two identical gear pairs, or four gears total. The scope of the test program was limited to the investigation of the impact of gear module  $m$ , surface finish amplitude, and lubricant type on spur gear efficiency.

For the study on gear module, 23-tooth (23T) ground gears with  $m = 3.95$  mm were compared to 40-tooth (40T) ground gears with  $m = 2.32$  mm. Table 2 lists basic design parameters of these gears. In order to match the bending strength of the gear teeth, the gear face width for 40T gears was slightly larger than for the 23T gears as shown in Table 2. However, since other experimental and theoretical

**Table 2 Gear design parameters for the test gears used in this study. Units are in millimeters unless otherwise specified.**

Parameter	23T	40T
Module	3.95	2.32
Pressure Angle, deg	25.0	28.0
Pitch Diameter	90.86	92.74
Base Diameter	82.34	81.89
Outside Diameter	100.34	95.95
Root Diameter	81.30	85.80
Start of Active Profile	85.38	87.73
Circular Tooth Thickness	6.435	2.925
Root Fillet	1.63	0.83
Face Width	19.5	26.7
Center Distance	91.5	91.5

studies [4,16,41] have shown that face width has negligible effect on gear mesh power loss, the comparison of the 23T gears to 40T gears used in this study constitutes the influence of  $m$  on gear mesh power loss. The influence of surface roughness was investigated with sets of 23T ground and 40T ground gears that were finished using a commercial isotropic chemical polishing process that reduced average surface roughness ( $R_a$ ) values to below 0.1 Pm. In order to study the influence of lubricating oil on efficiency, power loss measurements were performed with 23T and 40T ground and chemically polished gears in three different oils. The first two oils, lubricants A and B, are variations of synthetic 75W90 type gear oils. The third oil, lubricant C, is lower viscosity synthetic gear oil marketed as a more efficient blend. The viscosity-temperature characteristics of these three lubricants are compared in Fig. 3, with other mechanical properties listed in Table 3.

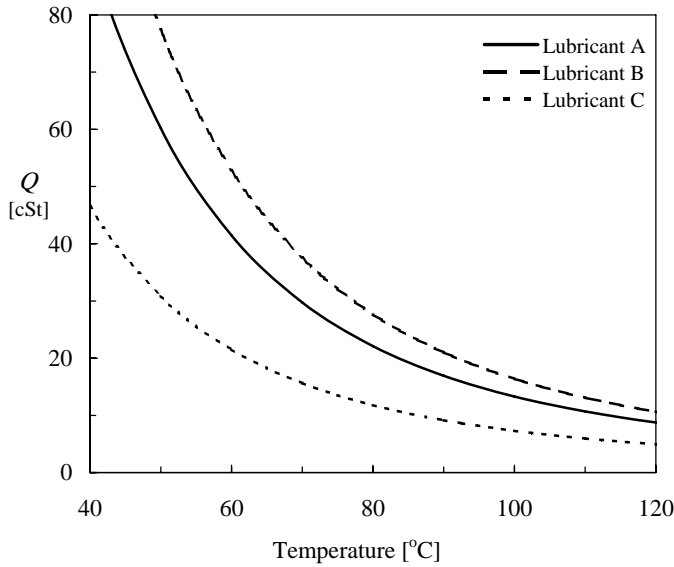
The 23T and 40T ground gear pairs were used to simulate high performance gearing applications within the transmitted power range of  $P_c = 259$  to 716 kW. These gears were operated at rotational speeds of 2,000 to 10,000 rpm for unloaded tests and at 6,000 and 10,000 rpm under  $T_c = 413$  to 684 N-m for loaded efficiency tests. Loaded efficiency tests were also conducted at rotational speeds of 2,000, 4000, and 8,000 rpm in Lubricant A in order to obtain data over a wider range of operating conditions. Lubricant temperature for all tests was 110°C. Table 4 summarizes operating test conditions for 23T and 40T gears. The test condition numbers listed in this table will be used in the remainder of this paper to identify  $T_c$  and  $\dot{\gamma}$  values associated with each test.

Since the same gears were operated with all three lubricants, it was critical to ensure that the tooth surfaces did not undergo changes in geometry and roughness throughout testing. For this purpose, tooth profiles were inspected using a gear coordinate measurement machine before and after every test for any signs of wear. These inspections showed that the maximum wear amounts were negligibly small. Similarly, a contact type surface profiler was used to quantify the surface roughness in the direction of sliding before and after every block of tests with different lubricants. Table 5 shows  $R_a$  and  $R_q$  surface roughness values measured after blocks of tests with different lubricants were completed. Table 5 indicates that surface roughness values did not change significantly throughout the test program, ensuring that measured power loss comparisons for all three lubricants represent the sole influence of the lubricant.

**Fig. 2 Experimental test gears, (a) 23T chemically polished gear, (b) 23T ground gear, (c) 40T chemically polished gear, and (d) 40T ground gear.**

**Table 1 Spur gear efficiency test matrix.**

Test	Gear	Process	Lubricant
1	23T	Ground	A
2		Ground + Chemically Polished	
3	40T	Ground	
4		Ground + Chemically Polished	
5	23T	Ground	B
6		Ground + Chemically Polished	
7	40T	Ground	
8		Ground + Chemically Polished	
9	23T	Ground	C
10		Ground + Chemically Polished	
11	40T	Ground	
12		Ground + Chemically Polished	



**Fig. 3 Viscosity-temperature characteristic for the lubricants used in this study. Other mechanical properties are displayed in Table 3.**

**Table 3 Mechanical properties of the lubricants.**

Property (at 60 °C)	Lubricant A	Lubricant B	Lubricant C
Absolute Viscosity (cP)	34.7	43.6	18.5
Pressure-Viscosity Coefficient (cP <sup>-1</sup> )	16.8	15.1	15.0
Specific Gravity	0.82	0.81	0.83

**Table 4 Operating test conditions used in this study.**

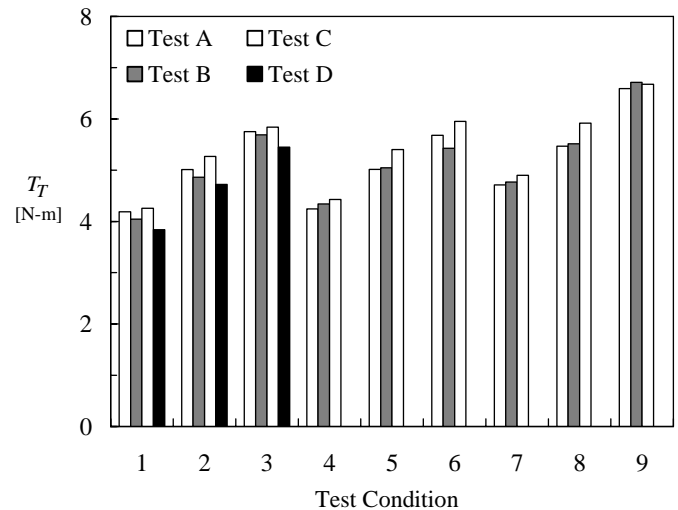
Test Condition	Rotational Speed [rpm]	Torque [N-m]
1	6,000	413
2	6,000	546
3	6,000	684
4	8,000	413
5	8,000	546
6	8,000	684
7	10,000	413
8	10,000	546
9	10,000	684
10	10,000	0
11	8,000	0
12	6,000	0
13	4,000	0
14	2,000	0

**Table 5 Measured  $R_a$  and  $R_q$  values in  $\mu\text{m}$  for the test gears, each block of tests with a different lubricant.**

Gear	Surface	Lubricant A		Lubricant B		Lubricant C	
		$R_a$	$R_q$	$R_a$	$R_q$	$R_a$	$R_q$
23T	Ground	0.24	0.32	0.21	0.28	0.23	0.31
23T	Chem. Polished	0.06	0.08	0.03	0.05	0.03	0.04
40T	Ground	0.16	0.20	0.14	0.18	0.13	0.17
40T	Chem. Polished	0.05	0.07	0.05	0.06	0.04	0.06

The test parameters monitored during gear efficiency tests were input torque  $T_T$ , rotational speed, oil stand reservoir temperature, oil supply temperatures for both test gearboxes, and oil temperatures on return from the gearboxes. Torque was measured with a non-contact digital torque-meter. It has a full-scale range of 50 N-m, and overall accuracy of 0.03% of full scale for the installed system. The machine rotational speed is typically maintained within 0.2-0.3% of the set (intended) speed, and exhibits a negligibly small ( $\pm 2$  rpm) fluctuation about this value which is averaged out during data collection. The lubrication system maintains the oil supply temperatures effectively within  $\pm 2^\circ\text{C}$  of the set value.

In order to evaluate the repeatability of power loss measurements for the test machine, a 40T ground and chemically polished gear set was identified for checking repeatability and tested periodically with Lubricant A. These gears were used to demonstrate the repeatability of the test machine over a 2-year period, through several tear-downs, shaft, bearing and seal replacements and with different operators. Figure 4 shows the results of this repeatability study, displayed in terms of total measured torque loss  $T_T$ . Four sets of measurements were performed with these gears, except for Test D, in which only test conditions 1, 2, and 3 were run. The numbers along the x-axis of this figure represent the test conditions described in Table 4. Figure 4 indicates that the test fixtures maintained repeatability to within 0.5 N-m in terms of  $T_T$  when  $T_c = 684$  N-m, which corresponds to 0.04% in total efficiency per gearbox, and can be considered satisfactory.



**Fig. 4 Comparison of repeatability tests using 40T chemically polished gears in lubricant A, 19.5 mm face width.**

## 2.3 Calculation of Components of Power Losses from $T_T$ Measurements

There are a large number of sources of power loss in a gearbox due to complex mechanical and fluid-mechanical interactions. These losses can be grouped based on their dependence on load such that

$$P_T = P_{mech} + P_{spin} \quad (1)$$

where  $P_T$  is total measured power loss,  $P_{mech}$  is the total mechanical (load-dependent) power loss, and  $P_{spin}$  is the total load-independent spin power loss, which includes gear windage, inertial losses from the impinging oil lubrication jet, shaft seals, and bearing viscous power losses. In order to experimentally separate  $P_{mech}$  and  $P_{spin}$ , a two-step measurement procedure is required. First the total power loss  $P_T$  is measured at a certain test condition under a given torque transmitted  $T_c$ . Then the same test is repeated under no load  $T_c = 0$ . In this second test,  $P_{mech} = 0$  and hence,  $P_T = P_{spin}$ . These two companion tests allow measurement of  $P_{mech}$  according to Eq. (1). The experimental set-up shown in Fig. 1 contains two identical gear pairs and eight identical cylindrical roller bearings, so that the load-dependent power loss can be further decomposed as  $P_{mech} = 2P_m + 8P_{b,L}$ . Here  $P_{b,L}$  is the load-dependent bearing power loss, where the load-dependent bearing friction torque is  $M_L = 0.5P_bW_b d_{bore}$  [39]. Here,  $P_b$  is an experimentally determined average bearing friction coefficient obtained under full-film lubrication conditions,  $W_b$  is bearing radial load, and  $d_{bore}$  is the bearing bore diameter. For the NJ 406 J cylindrical roller bearings used in this study,  $P_b = 0.0011$  and  $d_{bore} = 30$  mm. This bearing power loss formula was used to separate bearing losses from  $P_{mech}$  so that the gear mesh mechanical power loss  $P_m$  was obtained from the measured values of  $P_T$  for loaded and unloaded cases as  $P_m = \frac{1}{2}(P_T - P_{spin}) - 4P_{b,L}$ . Finally, the average total efficiency of each gearbox (gear pair, bearings and seals) and the average gear mesh mechanical efficiency were calculated, respectively, by

$$\bar{\eta}_T = \frac{P_c - 0.5P_T}{P_c}, \quad \bar{\eta}_m = \frac{P_c - P_m}{P_c} \quad (2,3)$$

## 3 MEASURED EFFICIENCY RESULTS

### 3.1 Gear Mesh Mechanical Power Losses

The test matrix defined by Table 1 covers a wide range of operating conditions, gear module, surface roughness, and lubricants. 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 these influences on  $P_m$ .

The influence of  $\omega$  on measured  $P_m$  is shown in Fig. 5(a) for 23T and 40T ground gears at  $T_c = 413$  N-m. Here,  $P_m$  increases nearly linearly with  $\omega$  over the range of rotational speeds studied. The influence of  $T_c$  on  $P_m$  is shown in Fig. 5(b) for 23T and 40T ground gears at  $\omega = 6,000$  rpm. Similarly,  $P_m$  also increases nearly linearly with  $T_c$ . These two figures suggest that the gear mesh mechanical efficiency  $\bar{\eta}_m$  should remain relatively unchanged within these ranges of  $T_c$  and  $\omega$  due to the nearly linear influence of  $T_c$

and  $\omega$  on  $P_m$ . In order to demonstrate this, Fig. 6 shows measured gear mesh mechanical efficiency  $\bar{\eta}_m$  for 23T and 40T ground gears in lubricant A. Here, test conditions 1-3 correspond to  $\omega = 6,000$  rpm and a range of  $T_c$  from 413 N-m to 684 N-m. Test conditions 4 to 6 correspond to the same values of  $T_c$ , but at  $\omega = 8,000$  rpm, and test conditions 7-9 correspond to  $\omega = 10,000$  rpm. The  $\bar{\eta}_m$  values calculated by Eq. (3) for 40T gears range between 99.76% at test condition 2 ( $T_c = 546$  N-m and  $\omega = 6,000$  rpm), and 99.81% at test condition 8 ( $T_c = 546$  N-m and  $\omega = 10,000$  rpm). For 23T gears,

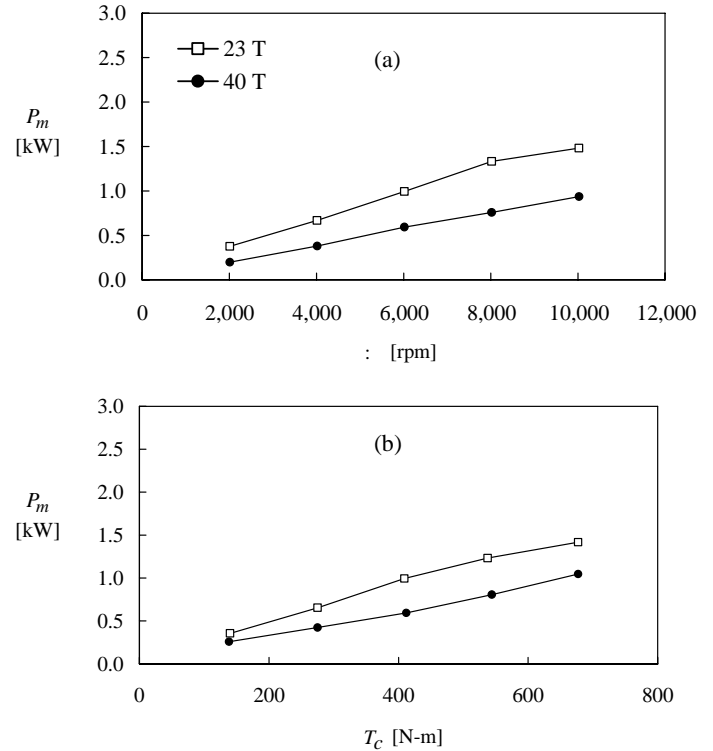


Fig. 5 Comparison of measured  $P_m$  for 23T and 40T ground gear pairs as a function of (a)  $\omega$  at  $T_c = 413$  N-m and (b)  $T_c$  at  $\omega = 6,000$  rpm.

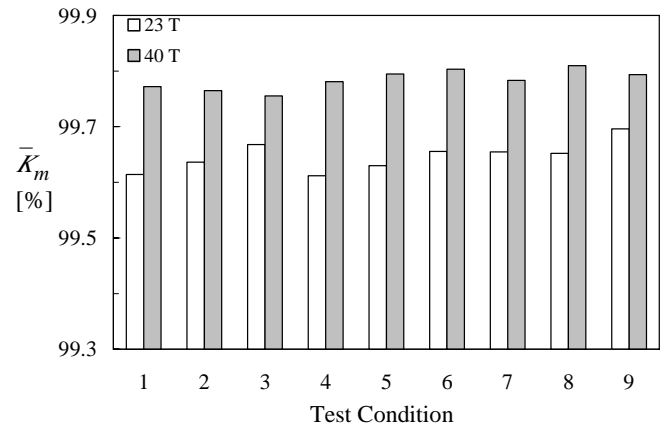
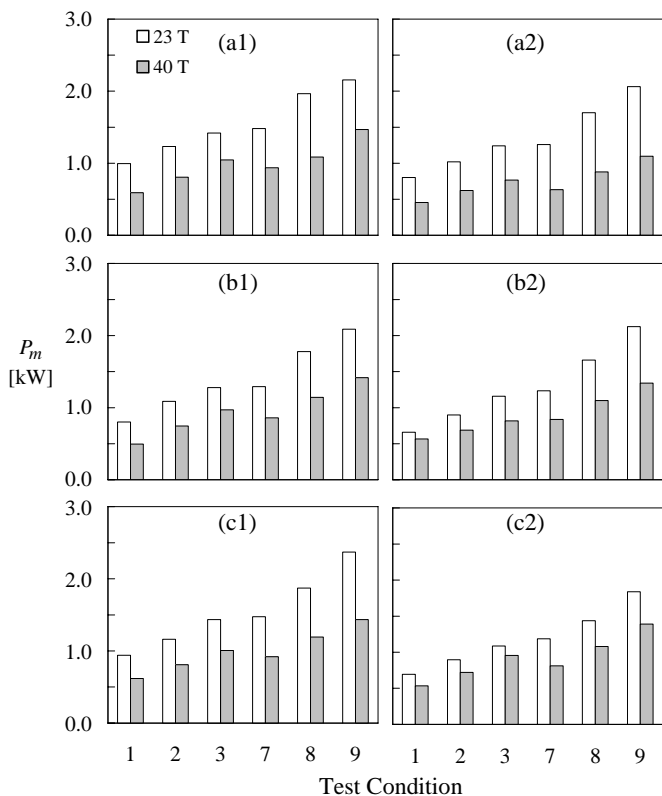


Fig. 6 Measured  $\bar{\eta}_m$  values of 23T and 40T ground gear pairs in Lubricant A at test conditions listed in Table 3.

measured  $\bar{X}_m$  values range between 99.61% at test condition 1 ( $T_c$  413 N-m and  $\omega$  : 6,000 rpm), and 99.70% at test condition 9 ( $T_c$  684 N-m and  $\omega$  : 10,000 rpm). For the wider range of  $\omega$  included in Fig. 5(a), measured  $\bar{X}_m$  values for 40T ground gears stays nearly constant with  $\bar{X}_m$  99.77% at 2,000 rpm and 99.78% at 10,000 rpm. For the range of  $T_c$  in Fig. 5(b), measured  $\bar{X}_m$  ranges between 99.70% at  $T_c$  140 N-m and 99.76% at 684 N-m. Therefore, the influences of  $T_c$  and  $\omega$  on  $\bar{X}_m$  are not dominant in the range of test conditions studied.

Figures 5 and 6 show that  $P_m$  is consistently higher for 23T gears than 40T gears, which is in agreement with previous experimental and theoretical studies [4-7,16-17,41]. This is primarily due to the higher relative sliding velocities experienced by the 23T gears (sliding ratio (SR) range of -1.0 to 1.0), in comparison to 40T gears (-0.3<SR<0.3). Figure 7 provides further comparisons of measured  $P_m$  values but for gear pairs using lubricants B and C in addition to lubricant A under test conditions 1-3 and 7-9 in Table 4 ( $\omega$  : 6,000 and 10,000 rpm, and  $T_c$  413, 546 and 684 Nm). Here for Lubricant A,  $P_m$  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. 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. Similarly for lubricant C,  $P_m$  values for 40T ground gear pairs are 35% lower than those for 23T ground gears



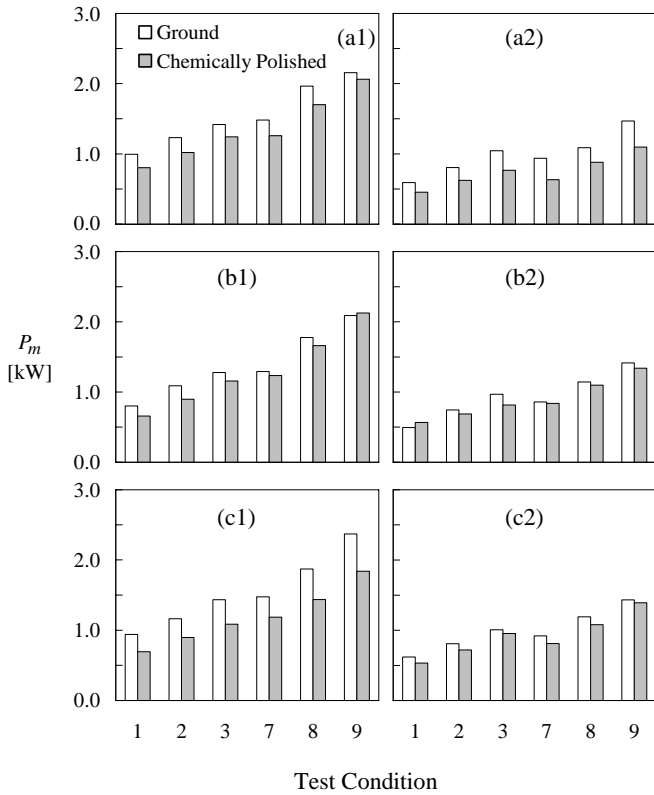
**Fig. 7 Influence of  $m$  on measured  $P_m$  for 23T and 40T gear pairs; (a1) ground gears in lubricant A, (a2) chemically polished gears in lubricant A, (b1) ground gears in lubricant B, (b2) chemically polished gears in lubricant B, (c1) ground gears in lubricant C, (c2) chemically polished gears in lubricant C.**

and 23% lower for chemically polished gears. Percent reduction in  $P_m$  for 40T gears versus 23T gears generally increases slightly with  $\omega$ , but there appears to be no definite trend with  $T_c$ .

$P_m$  is composed of sliding and rolling power losses, i.e.  $P_m = P_s + P_r$ .  $P_s$  is the frictional sliding power loss due to asperity contact and fluid shearing, and  $P_r$  is the power loss resulting from fluid viscous resistance to rolling in the gear mesh contact zone. Sliding and rolling friction in a gear mesh cannot be separated experimentally. In terms of efficiency, the goal of super-finishing gear surfaces is to reduce asperity contact, and thus reduce sliding friction. Commercially available chemical polishing processes have been suggested to improve gear efficiency as they reduce the surface roughness amplitude and the resultant sliding friction coefficient of the contacting surfaces [4,16]. However little published experimental proof of this is available, especially when lubricants of different viscosities are considered.

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 the three different lubricants. Figure 8 shows a comparison of the measured  $P_m$  values for ground and chemically polished surfaces with lubricants A, B and C over a range of operating conditions. For gears operating in lubricant A, experimental data was obtained at test conditions 4, 5, and 6, in addition to the test conditions shown in Fig. 8. Data for these operating conditions can be found in Figures 6, 10, 11, and 12. For 23T gears operating in Lubricant A, shown in Fig. 8(a1), chemical polishing gears to  $R_a$  0.06  $\mu\text{m}$  compared to ground gears having  $R_a$  0.24  $\mu\text{m}$  reduces  $P_m$  by as little as 4% at test condition 9 ( $T_c$  684 N-m and  $\omega$  : 10,000 rpm), and as much as 26% at test condition 4 ( $T_c$  413 N-m and  $\omega$  : 8,000 rpm). The average reduction of measured  $P_m$  for these gears is 16% for all of the test conditions considered. For 40T gears operated in lubricant A, shown in Fig. 8(a2), chemical polishing gears to  $R_a$  0.05  $\mu\text{m}$  compared to ground gears having  $R_a$  0.16  $\mu\text{m}$  results in 19% and 31% reductions of  $P_m$  at test conditions 6 ( $T_c$  684 N-m and  $\omega$  : 8,000 rpm) and 7 ( $T_c$  413 N-m and  $\omega$  : 10,000 rpm), respectively. The average reduction of measured  $P_m$  for these gears is 23% for all of the test conditions considered. 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. Finally in Fig. 8(c1) and 8(c2) for lubricant C chemical polishing reduced  $P_m$  of 23T and 40 T gears by 23 and 9%, respectively. The larger difference in measured  $P_m$  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  $P_s$ , which is directly reduced by chemical polishing. However, this result is not the same with all lubricants tested because of differing oil viscosities as shown in Fig. 3. The general trend in Fig. 8 is that measured  $P_m$  values are reduced by varying amounts when gear surface roughness is reduced significantly through chemical polishing. The amount of improvement in  $P_m$  due to chemical polishing is dependent on operating conditions, gear module, surface roughness amplitudes before and after the chemical polishing process, and by oil viscosity because both  $P_s$  and  $P_r$  are influenced by viscosity.

Lubricant viscosity  $Q$  is a key parameter influencing  $P_m$ . This can be demonstrated by performing efficiency tests using a particular lubricant at various oil supply temperatures, or by using oils of



**Fig. 8 Influence of chemical polishing on measured  $P_m$  for (a1) 23T gears in lubricant A, (a2) 40T gears in lubricant A, (b1) 23T gears in lubricant B, (b2) 40T gears in lubricant B, (c1) 23T gears in lubricant C, (c2) 40T gears in lubricant C.**

increasing  $Q$  increases fluid shear forces arising from relative sliding of the two surfaces, which can potentially increase  $P_s$  if the sliding friction component is small compared to fluid shear forces. Increasing  $Q$  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  $P_s$ .

Film thickness can also be used to help demonstrate the influence of chemical polishing on  $P_m$ . The classic minimum film thickness formula of Dowson and Higginson [11] is used here. Table 6 lists minimum pitch point lambda ratio ( $O = \text{film thickness} / R_a$ ) for 23T and 40T ground and chemically polished gears for the test conditions and lubricants shown in Fig. 8. Table 6 demonstrates that lambda ratio for all chemically polished gears was greater than unity, indicating minimal metal-to-metal contact for all test conditions. Conversely, lambda ratio for all ground gears was below unity for all test conditions and lubricants. Table 6 combined with Fig. 8 show that the reducing asperity contact through chemical polishing reduces  $P_m$ , but the extent depends on the viscosity of the lubricant. Furthermore, Table 6 and Fig. 8 indicate that the sliding friction component of  $P_s$  dominates  $P_m$  for the gears and lubricants used in this study.

Figure 9 provides an overall comparison of average  $\bar{K}_m$  over the range of operating conditions considered for 23T and 40T ground and chemically polished gear pairs operated with lubricants A, B, and C. Since measured  $\bar{K}_m$  was found to be approximately constant, a

single, average  $\bar{K}_m$  value is used to compare gear designs and lubricant performance. From this figure, the following observations can be made: (i) 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 since lubricant B is the thickest. For the 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  $P_s$ . (ii) Chemically polished 23T gears performed best with the least viscous lubricant C. Since chemically polished surfaces exhibit limited (or no) asperity contact regardless of oil type, both the fluid shear component of  $P_s$  and  $P_r$  were minimal with the lowest viscosity oil. (iii) Chemically polished 40T gears performed best with the medium-viscosity lubricant A. This result is not consistent with the 23T gears probably due to the lower range of sliding velocities found in the 40T gears. Due to the lower sliding velocities in the 40T gear mesh,  $P_s$  may represent a smaller portion of  $P_m$  than for 23T gears.

Figure 9 also summarizes the influences of gear module, surface roughness, and lubricant type on  $P_m$ . It illustrates that chemically polished 40T gears with lubricant A outperformed all other gears in terms of  $P_m$  with average mechanical efficiency over all test conditions tested of  $\bar{K}_m$  99.83%. Meanwhile, ground 23T gears

**Table 6 Calculated minimum lambda ratio at pitch point for the test conditions represented in Fig. 8.**

Test Condition	23T		40T	
	Chem. Polished	Ground	Chem. Polished	Ground
<b>Lubricant A</b>				
1	2.2	0.5	1.8	0.4
2	2.2	0.5	1.8	0.4
3	2.2	0.5	1.8	0.4
7	3.2	0.8	2.6	0.6
8	3.1	0.8	2.5	0.6
9	3.1	0.7	2.5	0.6
<b>Lubricant B</b>				
1	2.5	0.6	2.0	0.5
2	2.4	0.6	2.0	0.5
3	2.4	0.6	2.0	0.5
7	3.5	0.8	2.9	0.7
8	3.4	0.8	2.8	0.7
9	3.4	0.8	2.8	0.7
<b>Lubricant C</b>				
1	1.4	0.3	1.1	0.3
2	1.4	0.3	1.1	0.3
3	1.3	0.3	1.1	0.3
7	2.0	0.5	1.6	0.4
8	1.9	0.5	1.6	0.4
9	1.9	0.5	1.5	0.4

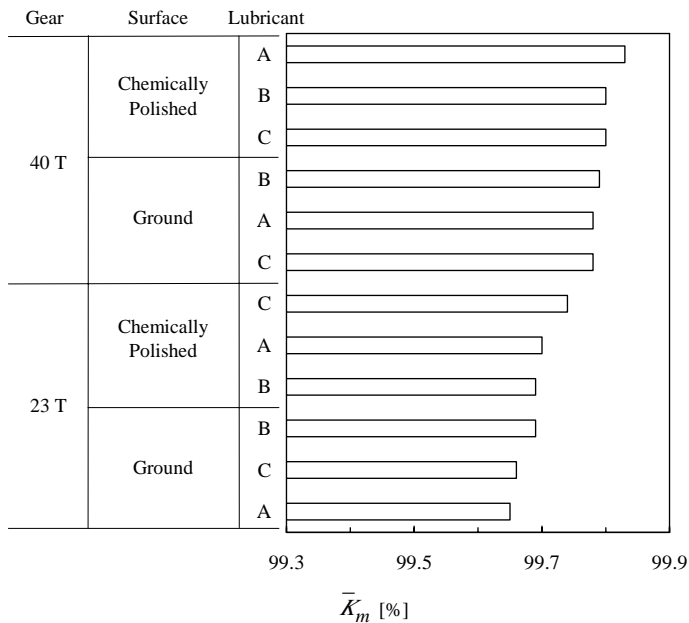


Fig. 9 Influence of  $m$ , chemical polishing, and lubricant type on measured  $\bar{K}_m$  for 23T and 40T gears in lubricants A, B, and C.

lubricated with the same oil performed the worst with average  $\bar{K}_m$  99.65%. These results show that module is the most influential parameter on  $P_m$ , followed by surface roughness amplitude, then by lubricant type.

### 3.2 Measured Gearbox Spin Power Losses

Total gearbox power losses were defined earlier in Eq. (1) as a sum of  $P_{mech}$  and  $P_{spin}$ . The previous section focused on how to minimize  $P_m$ , and hence  $P_{mech}$ . Although it is important to study  $P_m$  independently of other sources of power loss, doing so without consideration of gearbox spin power losses can lead to erroneous conclusions regarding lubricant selection on the basis of efficiency. This section will focus on the unloaded gearbox power losses  $P_{spin}$ .

In order to study the influence of module and lubricating oil on unloaded power loss, total gearbox spin loss  $P_{spin}$  was measured for each test outlined in Table 1 at rotational speeds of 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 represent the combined influence of  $m$  and  $b$ . However, a previous study using this test machine [41] showed that the influence of face width on  $P_{spin}$  between 14.2 mm and 26.7 mm is negligible, so  $P_{spin}$  will be discussed in terms of the effect of  $m$  with the understanding that the true influence of  $m$  is probably slightly higher.

Figure 10 shows the influence of lubricant type on total gearbox spin loss  $P_{spin}$  for ground 23T and 40T gears. Since the test machine was unloaded during these tests ( $T_c = 0$ ), the load-dependent losses were nearly zero, so that the influence of  $m$ ,  $Q$ , and  $\omega$  on  $P_{spin}$  can be quantified. Referring to the lubricant temperature-viscosity relationships in Fig. 3, it is observed that total gearbox spin loss  $P_{spin}$  follows  $Q$ . Lubricant B, being the most viscous lubricant studied, exhibited the highest  $P_{spin}$  for both 23T and 40T gears over the

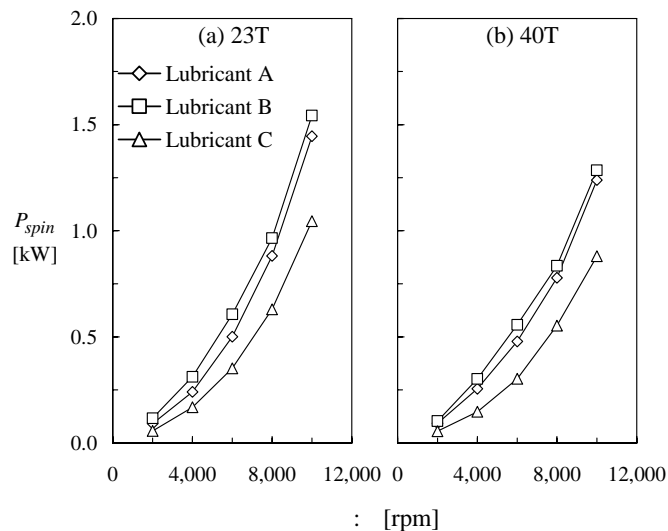


Fig. 10 Comparison of measured  $P_{spin}$  of 23T and 40T gear pairs in lubricants A, B, and C as a function of  $\omega$ ; (a) 23T ground gear pairs and, (b) 40T ground gear pairs.

entire range of  $\omega$  considered. Likewise, lubricant C with the lowest  $Q$  exhibited the lowest  $P_{spin}$  values. Since the specific gravity is nearly equal for all these lubricants, density can be taken to be constant. In Fig. 10,  $P_{spin}$  varies to a power of  $\omega$  ranging from 1.4 to 1.7, depending on the gear pair and lubricant. This result does not agree with most previous studies on windage [25-26], 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. This is likely due to the fact that this study uses a meshed gear pair with an oil jet, so  $P_{spin}$  presented here includes other sources of power loss, namely oil pumping/squeezing and inertial losses of the oil jet. However, the present results do agree with the unloaded power loss measurements of the jet-lubricated gear friction test rig of Britton *et al* [24], which also included power losses of shaft seals, bearings, and oil pumping/squeezing. For example, measured unloaded power losses in this study for the 23T gears in Lubricant A at  $110^\circ\text{C}$  were found to follow the relation  $P_{spin} = 6 \cdot 10^{-7} \omega^{1.6}$  kW; whereas  $P_{spin} = 1 \cdot 10^{-7} \omega^{1.5}$  kW; in Ref. [24] for a gear pair lubricated with Mobiljet II synthetic oil at  $50^\circ\text{C}$ . 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 this study.

Although  $P_m$  was only modestly influenced by oil type,  $P_{spin}$  is shown to be very dependent on oil type due to differences in  $Q$ .  $P_{spin}$  for both 23T and 40T gears was similarly influenced by  $Q$ , with an average of 42% reduction in  $P_{spin}$  for Lubricant C (the least viscous lubricant) compared to Lubricant B (the most viscous lubricant). This indicates that lubricant selection can be very important in optimizing total power loss for high-speed gearing.

Previous studies on gear windage and fluid pumping/squeezing between gear teeth indicated that gear module is an important parameter in gearbox spin losses [25, 30]. Figure 10 also shows that module  $m$  plays an important role on measured  $P_{spin}$  regardless of type of the lubricant. At 4,000 rpm, the 40T gear pairs operated at an

average of 7% lower  $P_{spin}$  over all oils tested. This benefit increased with  $\phi$  to an average of 16% at 10,000 rpm, showing that reducing gear module has the advantage of reducing both  $P_m$  and  $P_{spin}$ .

### 3.3 Total Gearbox Power Losses

It was observed in section 3.2 that total gearbox spin loss  $P_{spin}$  increases with  $\phi$ :  $^{1.4}$   $^{1.7}$ , while  $P_m$  increases linearly with  $\phi$ . 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. Figure 11 shows the variation of  $P_m$ ,  $P_{b,L}$  and  $P_{spin}$  as a function of  $\phi$  at  $T_c$  413 N-m, 546 N-m, and 684 N-m for 40T ground gears in lubricant A. As previously observed,  $P_m$  and  $P_{b,L}$  increase nearly linearly with  $\phi$  and  $T_c$ , but  $P_{spin}$  increases as a function of  $\phi$ :  $^{1.6}$ . The percentage of  $P_T$  attributed to  $P_{spin}$  is the largest (51%) at low torque and high speed ( $T_c$  413 N-m,  $\phi$ : 10,000 rpm), and the lowest (28%) at high torque and low speed ( $T_c$  684 N-m,  $\phi$ : 6,000 rpm). The effect of  $P_{spin}$  on total efficiency,  $\bar{K}_T$  as calculated by Eq. (2) is shown in Fig. 12 for various test conditions considered in Table 4. Here, test conditions 1-3 correspond to  $\phi$ : 6,000 rpm and a range of  $T_c$  from 413 N-m to 684 N-m. Test conditions 4-6 correspond to the same values of  $T_c$ , but at  $\phi$ : 8,000 rpm, and test conditions 7-9 correspond to  $\phi$ : 10,000 rpm at the same values of  $T_c$ . It is observed here that  $\bar{K}_T$  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  $\phi$  increases for any given  $T_c$ ,  $\bar{K}_T$  decreases due to the non-linear relationship between  $\phi$  and  $P_{spin}$ .

Unlike  $\bar{K}_m$ , which remained relatively constant,  $\bar{K}_T$  varies more widely due to the inclusion of load-independent power loss. Since

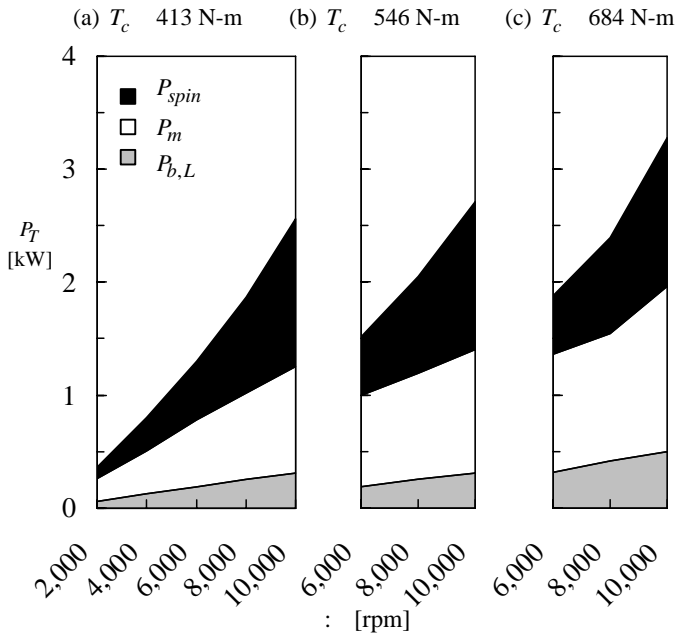


Fig. 11 Components of measured  $P_T$  for 40T ground gear pairs in lubricant A as a function of  $\phi$ ; (a)  $T_c = 413$  N-m, (b)  $T_c = 546$  N-m, and (c)  $T_c = 684$  N-m.

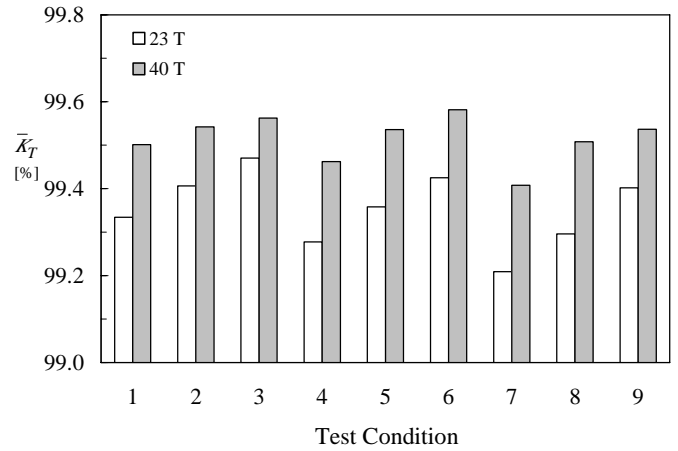
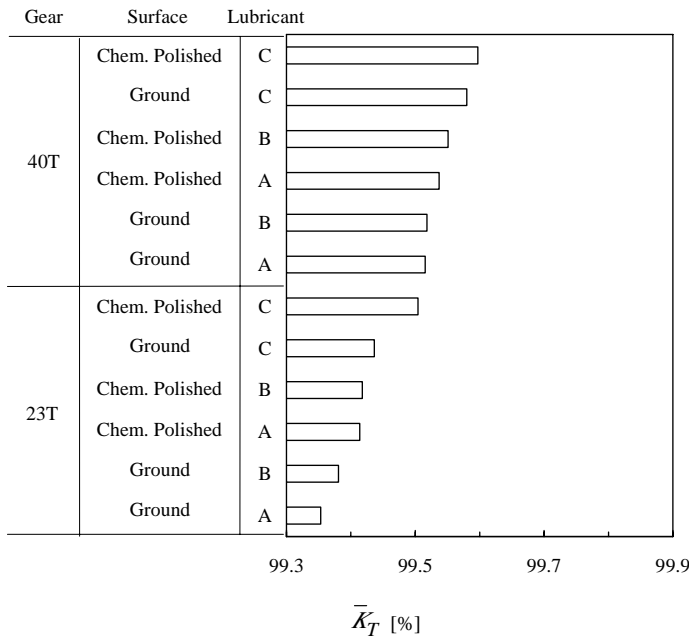


Fig. 12 Measured  $\bar{K}_T$  values of 23T and 40T ground gear pairs in lubricant A at various test conditions listed in Table 3.

$P_{spin}$  is lower at low  $\phi$ ,  $\bar{K}_T$  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  $\bar{K}_T$  99.47% at  $T_c$  684 N-m and  $\phi$ : 6,000 rpm, but at  $\bar{K}_T$  99.21% at  $T_c$  413 N-m and  $\phi$ : 10,000 rpm. Similarly, the 40T ground gears operated at  $\bar{K}_T$  99.56% and 99.41% under the same respective operating conditions.

Although  $P_m$ , and by extension  $\bar{K}_m$ , were only slightly influenced by oil selection, it was shown that  $P_{spin}$  is heavily dependent on  $Q$ . While the most viscous oil yielded lowest  $P_m$  for ground gears, it resulted in highest  $P_{spin}$  for the same gears. It is therefore necessary to investigate the influence of gear geometry, surface finish and lubricant selection on total gearbox efficiency  $\bar{K}_T$ . To this end, Fig. 13 compares the averages of all  $\bar{K}_T$  values over all test conditions considered for 23T and 40T ground and chemically polished gears lubricated with the three types of lubricant tested. Although  $\bar{K}_T$  is not constant with  $\phi$ , the influences of  $\phi$  and  $Q$  on  $P_{spin}$  are well-defined and consistent for 23T and 40T gear designs. Therefore, an average  $\bar{K}_T$  value over all test conditions considered represents the overall performance of gear design, surface finish and oil relative to the others.

Figure 13 demonstrates that module is still the most critical parameter influencing gear efficiency in terms of  $\bar{K}_T$ , but the surface finish is in some cases less influential than oil selection. For instance,  $\bar{K}_m$  for the 23T chemically polished gears was consistently higher than ground gears for all lubricants tested, but  $\bar{K}_T$  is highest with lubricant C, regardless of surface finish. This trend also holds for 40T gears. Hence, although there was no clear choice of most efficient gear oil in terms of  $\bar{K}_m$ , lubricant C is identified as the overall best-performing lubricant among the three lubricants considered, resulting in the highest  $\bar{K}_T$  for all gears at all operating conditions. For all ground gears and chemically polished 40T gears, reductions of  $P_{spin}$  with lubricant C offset improvements in  $P_m$  with lubricants B and C, respectively. For 23T chemically polished gears, both  $P_{spin}$  and  $P_m$  are minimized with lubricant C, which results in the greatest difference in  $P_T$  between lubricants C and A. Figure 13 also serves as a summary of the influences of  $m$ , chemical polishing, and lubricant selection on  $P_T$ . 40T chemically polished gears



**Fig. 13** Comparison of average measured  $\bar{K}_T$  for 23T and 40T ground and chemically polished gear pairs in lubricants A, B, and C.

lubricated with lubricant C outperformed all other gears in terms of  $P_T$  with  $\bar{K}_T$  99.60%, while 23T ground gears lubricated with lubricant A performed the worst with  $\bar{K}_T$  99.35%, resulting in a possible 0.25% overall efficiency improvement.

#### 4 CONCLUSIONS

A test methodology was developed for measurement of spur gear efficiency under high-speed and variable torque conditions. A power-circulating test machine was designed and built to operate at speeds to 10,000 rpm and transmitted power to 700 kW. A precision torque measurement system was implemented and its accuracy and repeatability in measuring torque loss was demonstrated to be within 0.5 N-m or 0.04% of the total efficiency per gearbox at 684 N-m. In order to expand the existing experimental spur gear efficiency knowledge base further into high speed and high torque conditions, an experimental test matrix including gears of different modules and surface roughness levels operating under jet lubrication conditions with three different gear lubricants was developed. Experimental results were analyzed to quantify the influence of these parameters on load-dependent (mechanical) and load-independent (spin) power losses.

Measured trends in gear mesh mechanical power loss agree with previous experimental studies under similar conditions, and show that gear mesh mechanical efficiency remains relatively constant for high torque and speed conditions whereas total efficiency is not constant. Gear surface roughness was conclusively shown to reduce gear mesh mechanical power loss for fine and coarse pitch gears over all test conditions and lubricants considered. Experimental results indicate that gear module is the most influential parameter on gear mesh mechanical efficiency for the high speed and torque 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, fine pitch gears performed better in terms of mesh power losses with the medium-viscosity lubricant, while the coarse pitch gears performed better with the lowest-viscosity lubricant due to minimization of both rolling and sliding power losses at the gear mesh. Experimental results for three different lubricants show that load-independent gearbox (spin) power losses are influenced heavily by lubricant viscosity and to a lesser extent by gear module, and can comprise a significant portion of overall gearbox power loss. With the spin losses included, gearbox total efficiency decreases with rotational speed and increases with transmitted torque. Gearbox total 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. This suggests that selection of the gear lubricant at the design stage is crucial in maximizing overall gearbox efficiency. Since spin losses are shown to significantly influence overall gearbox power loss, especially at high speeds, it is necessary for future theoretical studies to include these power losses in order to obtain a comprehensive gearbox efficiency prediction methodology.

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