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CHAPTER - 1

INTRODUCTION

1.1 Background and Motivation

Gears are widely used in many applications including automotive, rotorcraft and off-highway vehicle drive trains, machine tools, and industrial gearboxes. Gears transmit power and rotational motion from one shaft to another. In this process, some of the power is inevitably lost due to friction in the system and the drag caused by the atmosphere around the gears.

Nowadays, reducing frictional loss of gears to save energy is a big concern. There are many sources of printed information on this problem for gears but most of them are practical specialized information and are not developed to give a generalized basic consideration. With the present interest in energy saving the improvement of gear efficiency is being looked at again. Future energy shortages have not only to be fought with exploitation of new renewable energy resources but also with reduction of energy losses in all technical fields.

Looking at wind turbines as a growing market for alternative energy production, a modern equipment of the 5 MW class consist of 8 or more gear meshes and more than 12 bearing supports. A reduction of the overall losses by 50 % would save some 200 kW power losses per wind turbine unit. Hohn, et al [1] suggested the challenges of substantial power loss reduction with only minor impact on load carrying capacity, component size and weight and noise generation.

Automotive applications optimization attempts are made in all operating areas and for all components of vehicles to achieve minimum fuel consumption. Weight reduction and thermal management are possible approaches as well as hybrid systems and mechanical and software features for high efficient engines. Power loss reduction at the end of the power train has a large impact on overall optimization although
absolute efficiency in gearboxes and rear axles are already high. However, 1 kW savings in the gearbox mean 4 kW savings in fuel energy.

James and John [2] pointed that the efficiency of power transmission systems is an important design factor due to the following reasons:

● Efficient power transmission systems ensure fuel economy of automobiles, marine vessels and aircrafts.

● With less fuel consumption less pollutant gases and particulates are emitted to the environment.

● Since power losses amount to heat generation within the gearbox several gear failure modes such as scoring and fatigue can be directly influenced by the efficiency of the gearing system.

● Improved efficiency of a gearing system can reduce the requirements on the capacity of the lubrication system and the gearbox lubricant and thereby reducing the operation costs of the system.

● Efficiency prediction can assist in estimating the power requirements during the design stage of a machine and thus ensuring that the system operates reliably. It can also assist in estimating the power output for a given power input.

Internal combustion engines and other forms of prime movers were the main focus of efficiency improvement efforts in the past year. In view of the high fuel prices, fuel economy of a vehicle became important and these factors are influencing the decision of the consumer. Independent of this environmental pressures and government regulations are becoming more stringent in terms of allowable emission in the form of hazardous gases and particulates released to the environment. One percent improvement in drive train efficiency would reduce the fuel consumption and air pollutants by the same percentage. These form the motivation for friction and gear efficiency studies.
There are other benefits to improving gear efficiency as well. The efficiency losses amount to additional heat generation within the gearbox, several gear failure modes including scoring and contact fatigue failures are directly impacted by the efficiency of the gear pair. A more efficient gear pair generates less heat and therefore it is likely to perform better in terms of such failures. In the process, demands on the capacity and size of the lubrication system and the amount and quality of the gearbox lubricant are also constraint with improved efficiency. This also reduces the overall weight of the unit contributing to further efficiency improvements.

Friction can be stated as the resistance to motion between two surfaces in relative sliding and rolling under dry or lubricated contact conditions. Sliding friction is a direct result of the relative sliding between the two contacting surfaces while rolling friction exists due to the resistance to the rolling motion that takes place when one surface is rolled over another. Coefficient of friction usually refers to the coefficient of sliding friction, as a coefficient of rolling friction has no physical meaning. The total efficiency loss of the gearbox is attributable to sliding and rolling frictional losses between the gear teeth, windage losses due to complex interactions with the air surrounding the gears, and oil splashing and churning losses inside the gearbox, as well as the losses associated with the bearing and seals. While churning and windage losses are mostly geometry and speed related, friction losses are mainly associated with sliding velocities and load. Friction losses of hypoid gears are of primary interest here since they are a major source of losses in a drive train. As a common automotive application, the drive train of a rear-wheel-drive (or all-wheel-drive) car or truck contains one (or two, one rear and one front) axle gearbox that is formed by a hypoid final drive reduction unit and a differential. Unlike parallel axis gears that may have a mechanical efficiency well over 99 percent, the efficiency of hypoid gears usually falls into the range of 86 to 97 percent depending on the amount of relative sliding induced by the gear geometry. This is mainly because sliding velocities in a hypoid gear contact are significantly larger than parallel-axis gears. As a result of this, an axle unit that is formed by a single hypoid gear set has the same levels of (and in many applications more) power loss as a manual or automatic transmission that contains many spur or helical gears.
While friction power losses of an individual spur or helical gear pair are quite small, efficiency of gear trains that use multiple gear pairs as in transmissions might be significant. The mechanical efficiency of a gear train formed by in-series gear pairs is the product of efficiencies of individual gear meshes. Therefore, any incremental efficiency improvement that can be achieved from each gear pair often amounts to sizable improvements when the entire gear train or transmission considered.

Gears are usually operated under mixed elastohydrodynamic lubrication conditions where the lubricant film thickness is comparable to the surface asperity heights such that actual metal to metal contacts are possible. Therefore the friction between the gear teeth can be considered as a hybrid of dry and fluid friction. This effective friction at each contact point of a gear plays an important role in defining friction dependent efficiency losses as well as influencing scoring limits, contact fatigue lives and the dynamic behavior including vibration and noise.

1.2 Literature Review

Friction and efficiency of gear system has been of acute interest to gear researchers in last century. The study of power losses in geared transmissions dates back to the pioneering works of Reuleaux [3] and Weisbach [4]. A large number of studies have been proposed in the literature, especially on prediction of friction and efficiency of gear trains by Martin [5], Yada [6], Li and Seireg [7] and Buckingham [8]. In terms of the methodologies used for the study of gear efficiency it can be broadly classified into three approaches as:

(i) the friction and gear efficiency measured directly by using actual gears or representative hardware,

(ii) semi analytical prediction of efficiency using a pre-known constant or empirical coefficient of friction information, and

(iii) physics-based analytical prediction of gear efficiency by computing coefficient of friction using an elastohydrodynamic lubrication model.
An experimental studies [9-17] were focused on measurement of power losses of gear pairs. Naruse, et al [9,10 and 11] carried out an experiment to study the power loss of gear pairs, and suggested that, the mechanical power loss decreases with a decrease in oil temperature and reduction in gear surface roughness. Mizutani and Isikawa [12] were conducted an experimental analysis to determine power loss of spur gears with long addendum under various conditions of load and speed. They found that the gear power loss was proportional to tooth load and rate of oil flow at all gear speeds. Hori, et al [13] proposed a new gravity pendulum method for precisely measurement of the tooth surface friction coefficient of a mating gears, excluding the bearing loss. As the result, it is found that it is required to oscillate the gravity pendulum from the initial amplitude which is slightly larger than that corresponding to the width of the Hertzian contact, and measuring the value which becomes almost a constant in the region equal to or larger than the width of the Hertzian contact.

Yoshino and Ohshima [14] were studied the effect of tooth profile and addendum modification on efficiency of cylindrical worm gears. They proposed a new method for finding the geometrical dimensions of the cylindrical worm gears based on the conditions required for that between the flanks, load transmission and hydrodynamical lubrication. Yada [15] measured gear tooth normal and frictional forces using strain gauges mounted in the fillets of the gear teeth. These measured forces were further used to compute the dynamic coefficient of friction existing between contacting teeth. Friction coefficients increase at low sliding speeds. The results show that the reversal of sliding which occurs at the pitch-point does not cause a discontinuity in the friction coefficient, which shows a smooth transition as the friction force reverses direction.

Changenet and Pasquier [16] stated that the power dissipation mechanism inside a gearbox is due to the churning losses and frictional losses they are commonly called the no-load dependent losses and load dependent losses, respectively. These forms of heat evacuation depend mainly on the case temperature, the environment temperature, the case geometry and the coefficients that govern each one of these heat exchange mechanisms, such as gearbox surface emissivity, etc. The thermal equilibrium of a gearbox is reached when the operating temperature stabilizes i.e. when the power dissipated inside the gearbox is equal to the heat evacuated from gearbox to the
surrounding environment. The higher the power losses the higher will be the equilibrium temperature of the gearbox. Ikejo and Nagamura [17] described the effect of traction oil on the power loss of spur gear drive using several traction oils. They investigated the relationships between the power loss of the gear drive lubricated with the traction oil, the surface temperature on the gear tooth, and the lubrication condition.

A simpler contact interface condition were adapted [18-28], most commonly a pair of cylindrical disks (twin-disks), to measure coefficient of friction under conditions simulating a gear pair, so that this measured friction coefficient can be used to predict the efficiency of a gear pair. The empirical formula proposed by Misharin [18] do not include any surface roughness parameter, and hence influence of varying surface roughness cannot be studied using these formulae. In the formula of Benedict and Kelley [19] considers the load per unit length as the load parameter and consider a surface parameter S that is either the root mean square (RMS) roughness profiles. Similarly, the formulae of O’Donoghue and Cameron [20] and Mishrin [18] are not a function of the normal load and hence they cannot account for any load effects on coefficient of friction. They consider a surface parameter S that is the centerline average (CLA) of the roughness profiles. Also, Drozdov and Gavrikov [21] formula uses the maximum Hertzian pressure as the load parameter and do not included any surface roughness parameter, and hence, effect of surface roughness cannot be studied well.

The other empirical formulae proposed [22-25], that coefficient of friction is a function of a number of parameters such as sliding and rolling velocities, radii of curvature of the surfaces in contact, load related parameters (unit load or contact pressure), amount of surface roughness and the lubricant viscosity. The formula stated by Plint [22] included sliding velocities that are normal to the line of contact and are uniform along the contact line due to the geometry of the roller specimens and the twin-disk test configuration. There is no sliding in the direction of the contact line. Consequently, the formulae proposed by Kelley and Lemanski [23] and Ku, et al [24] from the twin-disk experiments represent a contact line of a spur gear pair and cannot be used directly for other types of gears including helical, worm, and hypoid gears. For these types of gears, the radii of curvature vary along the contact
line. In addition, there is also a component of sliding velocity in the direction of the contact line. Naruse and Haizuka [25] proposed an empirical friction coefficient formula from disk machine to study the effect of lubricating oil and sliding velocity in the direction of contact line. Influence of the sliding in the direction of a contact line on the value of coefficient of friction or the lubrication conditions is not well understood.

An experimental investigations [29-32], attempted to bridge this gap between the contact conditions defined by a pair of cylindrical disks and the contact conditions of an actual gear pair. Hirano, et al [29] studied the effect of angle between directions of sliding and line of contact on friction and wear of a roller and concluded that the sliding in the direction of the contact line results in the largest coefficient of friction while the opposite is true for the direction normal to the contact line. Tan [30] and Hohn [31] stated in two independent studies for worm gears that increasing sliding in the direction of the contact line, while keeping the sliding and rolling in the direction of normal to the contact line constant, should increase the value of coefficient of friction.

Semi-analytical efficiency models can be reviewed in two groups. The first group of studies [33-35] investigated the efficiency of a spur gear pair by assuming a uniform coefficient of friction along the entire contact surface and at every rotational position. Denny [33] stated that, the tangential friction force along the sliding direction was computed by using this user defined constant coefficient of friction and basic geometric and kinematic parameters of a spur gear pair. The amount of reduction of torque transmitted to the driven gear was used to calculate the mechanical efficiency of the gear pair. These models, while being very useful in bringing a qualitative understanding to the role of spur gear geometry on efficiency, had some major shortcomings. The first shortcoming stems from the definition of coefficient of friction. It must be constant for every contact point of a gear pair, and it must be known beforehand. Data collected by Pedrero [34] from the published twin-disk tests indicate that coefficient of friction of a combined rolling/sliding contact is not constant. It is influenced by many contact parameters [18-25]. The second shortcoming is that these studies were limited to spur gears and many complicating effects of the tooth bending and contact deformations, tooth profile
modifications and manufacturing errors were not included in the efficiency models. Michlin and Myunster [35] proposed a methodology for analysis of gear transmissions with allowance for the power losses due to both the rolling and sliding friction. The approach makes for improved accuracy in predicting the losses and the wear already at the design stage.

The second group of semi-analytical models [36-43] can be considered as an improvement over the constant coefficient of friction type models. These models relied on published experimental coefficient of friction formulae such as those of references [18-21] to predict the efficiency of spur [37-40, 42] and helical [36] gear pairs. Anderson, et al [37, 38, 39] were developed a power loss methodology to include involute spur gear of nonstandard properties. They analyzed the effects of modified addendum, tooth thickness, and gear center distance in addition to the parameters which included gear diameter, pitch, pressure angle, face width, oil viscosity, speed and torque. They concluded that, addendum elongation and pressure angle reduction are the most effective means of increasing contact ratio with a corresponding increase in power loss. In general, peak efficiencies were found to be greater for larger diameter and fine pitched gears and tare (no-load) losses were found to be significant. Barnes [40] were introduced a relationships for involute spur gear geometry with methods of correlating lubricant traction and windage test data for finding gearbox efficiency. Compact math models for lubricant density and viscosity under contact pressure are presented. Heingartner and Mba [36] were proposed a mathematical model to predict the power losses on helical gears highlighting the major contributor to losses in the gear mesh. They concluded that the sliding friction losses are heavily load dependent, increasing with load. However, the rolling friction losses decreased slightly with an increased load, and this is due to a decrease in oil film thickness. While they are potentially more accurate than the models of the first group, these models are still limited in the sense that their accuracy is dependent largely on the accuracy of the empirical coefficient of friction formula used. These empirical coefficient of friction formulae are not general and often represent certain types lubricants, operating temperatures, speed and load ranges, and surface roughness conditions of roller specimens that might differ from those of the actual gear pair of interest.
There are a number of gear efficiency models that use different forms of elastohydrodynamic lubrication models for predicting coefficient of friction [44-56]. Gear teeth operate different forms of lubricant, boundary lubrication, mixed lubrication and full EHD oil film lubrication. Dudley [44] shows the general characteristic of three regimes of lubrication. The tendency of gears to score (scuff) or to pit is strongly influenced by the regime of lubrication. Snidle and Evans [45] were conducted an experimental and theoretical work on the elastohydrodynamics lubrication of gears. Gear tooth contacts tends to operate under conditions where the film is thin compared to surface roughness. This feature is shown to have a significant effect on scuffing capacity and friction and also thought to be a factor in micropitting. Crook [46] stated that measurements of the thickness of the oil film have shown that the lubrication of loaded rollers is hydrodynamic in character. The dissipation of the heat by conduction through the oil to the surfaces of the disks and by transport with the oil (i.e. convection) are both considered. Viscosity distributions is used to develop the expressions for the effective viscosity within the pressure zone (i.e. that constant viscosity which would give the same frictional traction) and for the frictional traction.

Dyson [47] were proposed a model which refers to behavior in oscillatory shear, and the implications for the behavior in continuous shear have been examined. The experimental curves of frictional traction against sliding speed have been analyzed into three regions; the linear region, the nonlinear (ascending) region, and the thermal (descending) region. Experimental features are presented in terms of the new visco-elastic model. Trachman [48] presented a simplified calculation of the coefficient of traction for an elastohydrodynamic line contact, based upon a complete numerical solution to the coupled thermal and non-Newtonian hydrodynamic equations.

Among them, Dowson and Higginson [49] and Martin [50] assumed smooth contact surfaces for a spur gear pair and computed the instantaneous friction coefficient caused by surface shear stress distribution caused by the fluid film at the contact from a smooth surface elastohydrodynamic lubrication model. Adkins and Radzimovsky [51] developed an efficiency model for lightly loaded spur gears under hydrodynamic lubrication conditions, assuming that the gear teeth are rigid. Simon [52] provided an enhancement by using a point contact elastohydrodynamic
lubrication model for heavily crowned spur gears with smooth surfaces. The elastic deformations of the surfaces due to the fluid pressure distribution is included in the study. Larsson [53] conducted a transient elastohydrodynamic lubrication analysis for an involute spur gear with smooth surfaces. An isothermal full film thickness was assumed with a non-Newtonian model and also the gear teeth were assumed to be rigid. Wang, et al [54] analyzed involute spur gear lubrication by using a transient thermal elastohydrodynamic lubrication model for a Newtonian fluid. In this model, gear teeth were again assumed rigid and smooth. Also, it was assumed that the load is carried by either one tooth pair or two tooth pairs with the transition from one to other modeled as a trapezoidal variation of the load.

Wu and Cheng [55] developed a spur gear friction model based on mixed elastohydrodynamic lubrication contact analysis. The surface roughness was modeled such that all the asperities have the same radius of curvature whose heights have a Gaussian distribution. In order to account for the effect of the temperature rise at high speeds, a thermal reduction factor was used to modify this formula. Michalidis, et al [56] included the influence of the asperity contacts as well in calculating coefficient of friction. This study used a numerically generated roughness profile as an input. Gear materials were assumed to behave according to the elastic-perfectly plastic model. Thermal effects and non-Newtonian lubricant behaviors were also be considered.

The elastohydrodynamic lubrication based models [49-56] were successful in eliminating the need for prior knowledge of coefficient of friction to a certain extent. However, these models were not practical as they required significant CPU time to run. In addition, they might be more accurate in elastohydrodynamic lubrication aspects of the problem, their modeling of gears was limited to simple spur gears with ideal load distributions and no tooth deformations.

The literature on efficiency of helical gears is quite limited. The work by Akin [57] and Wellauer and Holloway [58] presented formulations to calculate elastohydrodynamic lubrication film thickness along the pitch line of a helical gear contact which uses the film thickness formula of Dowson and Higginson [49]. Chittenden, et al [59] presented a generalized analysis of a smooth surface
elastohydrodynamic lubrication problem under isothermal condition having a fluid entrainment at some intermediate angle from the minor axis of the contact ellipse, as is the case for helical gears. Later on Simon [60] extended his elastohydrodynamic lubrication analysis to helical gears using a smooth surface, point contact thermal elastohydrodynamic lubrication model. Haizuka, et al [61] studied the influence of helix angle on frictional power loss of helical gears experimentally and found the power loss increased with the increasing of the helix angle.

In terms of efficiency of hypoid gears, Buckingham [8] proposed an approximate formula for the power loss of hypoid gears, which is the sum of the losses of corresponding spiral bevel and worm gears. Naruse, et al [10] conducted several tests on scoring and frictional losses of hypoid gears of the Klingelnberg type. Coleman [62] used a simple coefficient of friction formula with a very limited number of parameters to calculate bevel and hypoid gear efficiency. In a later study [63], he proposed a scoring formula for bevel and hypoid gears with a coefficient of friction formula that is a function of only the surface velocity and surface roughness. Smooth surface thermal elastohydrodynamic lubrication formulations were also applied to a modified hypoid gear pair using a point contact model by Simon [64]. Jia, et al [65] analyzed the elastohydrodynamically lubricated hypoid gears by the multi-level techniques to study the film thickness and pressure distributions under isothermal condition.

The models reviewed in this literature survey were quite limited in terms of their representation of gears. Almost all of them assumed gear teeth (no bending deflection, base rotation or gear blank deformations) and relied on theoretical idealized load distributions along the contact line. These models were not capable of including modified profiles and any type of geometric deviations resulting from the manufacturing processes, heat treatment distortions, assembly errors and deflections of support structures. The proposed study is aims at eliminating these shortcomings by using gear contact models that are capable of including such effects. It also aims to develop and validate general physics-based efficiency models of gear pairs which in turn requires less computational time.
1.3 Scope and Objectives

The comparison of main approaches used by researchers to developed different models for prediction of friction and efficiency of gear are shown in Table 1.

<table>
<thead>
<tr>
<th>By using actual gears or representative hardware model</th>
<th>Semi analytical prediction model</th>
<th>The physics-based analytical prediction model</th>
</tr>
</thead>
<tbody>
<tr>
<td>In this model, the researchers adapted a simpler contact interface, most commonly a pair of cylindrical disks (twin-disks), to measure coefficient of friction under conditions simulating a gear pair.</td>
<td>In this model, the researchers investigated the efficiency of gear pair by assuming a uniform (constant) coefficient of friction along the entire contact surface and at every rotational position</td>
<td>In this model, the researchers used different forms of EHL lubrication models for predicting coefficient of friction</td>
</tr>
</tbody>
</table>

Table 1.1: Comparison of different friction and efficiency models

Most of these efficiency models proposed by researchers are shown in Table-1.1 were not validated. Therefore, their accuracy and effectiveness in representing real-life gear pairs are not known. Consequently, this study will emphasize the validations of not only the friction coefficient models but also the overall gear pair efficiency models.

The developed efficiency methodology using ANSYS™ software will combine a gear contact analysis model, an elastohydrodynamic lubrication based coefficient of friction model, and an efficiency computation formulation. This methodology will be applied to parallel and cross axis gear pairs. Since the focus of this study is the prediction of mechanical efficiency in terms of frictional power losses, the prediction of other losses related to oil churning, gear windage, and bearings will rely on published work of others [34,38, 66-71].
The primary objectives of this study are as follows:

1. To develop a model for the prediction of coefficient of friction and mechanical efficiency of gear pairs including all key gear, lubricant, surface finish, and operating parameters.

2. To assess the accuracy of the published friction coefficient formulae by comparing them to the elastohydrodynamic lubrication based friction coefficient values predicted by an elastohydrodynamic lubrication model and experimental traction data.

3. To validate the friction coefficient predictions of the elastohydrodynamic lubrication model by comparing them to the measured traction data at various load, speed and sliding conditions as well as lubricant prosperities.

4. To derive a new elastohydrodynamic lubrication based friction coefficient formula to minimize computational effort associated with elastohydrodynamic lubrication based model.

5. To apply the developed efficiency model to parallel axis (spur and helical) and cross axis (hypoid and bevel) gear pairs.

6. To validate the efficiency model by comparing efficiency predictions to actual spur gear efficiency measurement [72].

7. To perform parametric studies to quantify the influence of basic gear parameters on mechanical efficiency of gears.

The ultimate goal of this study is to develop an efficiency methodology which were be carried by using ANSYS™ software for prediction of coefficient of friction and mechanical efficiency of a gear pair.
1.4 Thesis Outline

Chapter 2 outlines the overall methodology proposed in this study for the prediction of the mechanical efficiency of a gear pair and describes the main components of this methodology. In Chapter 3, differences between friction coefficient models are identified and explained by comparing them with the measured traction data. Based on the results of this comparison, Multiple Linear Regression Analysis is employed to derive a new friction coefficient formula based on the elastohydrodynamic lubrication model predictions. A set of friction coefficient formulae is also developed with the measured traction data.

Chapter 4 is devoted to the application of the mechanical efficiency model to the parallel-axis gears. The load distribution model and formulations of surface velocities and radii of curvature are presented. Efficiency prediction results are presented for an example of spur and helical gear pair. The application of the developed mechanical efficiency methodology to cross-axis gears is described in Chapter 5. This chapter gives an introduction to the contact mechanics model used, provides the relevant details for the calculation of the hypoid gear tooth surface curvatures and velocities that are inputs to friction coefficient models. A set of efficiency predictions is presented for an example of hypoid and bevel gear pair.

The gear pair mechanical efficiency methodology is validated in Chapter 6 by using an experimental database established for high-speed spur gears [72]. This extensive validation effort includes different speed-load conditions, different gear design variations as well as different lubricant and surface conditions.

Chapter 7 presents the parametric studies of parallel axis (helical) and cross axis (hypoid) gear pairs. Variables in these parametric studies include the gear geometry, tooth modifications, surface finish and treatments, lubricant parameters, manufacturing and assemble errors as well as operating conditions. Finally, Chapter 8 presents the main conclusions and provides a list of recommendations for future work.