Estimation of Sliding Loss in a Parallel-Axis Gear Pair*

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Abstract
The meshing model and the method used for estimation of sliding loss in a parallel-axis gear pair are presented. First the sliding loss model for single tooth meshing spur gear is constructed. The sliding loss is attributed to the deviation in direction of force from the line of action due to friction. Afterwards the sliding loss of a helical gear pair is considered. The helical gear is modeled as the multi-section spur gears aligned slantwise along with base helix angle. Thus the sliding loss of each section can be considered as same as the loss of the spur gear pair. Then the total sliding loss is estimated by combining the losses of all sections together. To verify the method, the estimated results were compared with the experimental results and estimated results done by the other researchers. The estimated results agree with the experimental results. The sliding loss increases with loads and speeds. The gear with larger module has higher sliding loss. Increasing the helix angle, the sliding loss will be increased, but increasing the transverse pressure angle the sliding loss will be decreased. Various empirical formulae of friction coefficient are used in this paper, and the accuracies of various formulae are shown.

Key words: Spur Gear, Helical Gear, Sliding Loss

1. Introduction
Power losses of transmission systems become more important research topic due to more rigorous environmental regulations and the rising of fuel prices. High efficiency transmission gearbox is usually required in the automotive industry. As the large number of automobiles nowadays, increasing the efficiency of a gearbox just only a few percents can reduce amount of the fuel used each year significantly. One of the dominant sources of the power loss in transmission comes from the large number of transmission gears used in the gearbox. Although power losses of each gear pair are quite small and may have a mechanical efficiency well over 99 percent, power loss increases significantly when the multi-stage gear reduction is used. Therefore reducing the power loss of each gear pair is an important approach to increase the efficiency of the transmission systems.

A paralleled-axis gear pair including spur and helical gear pairs are normally used in many transmission systems such as, automotive transmissions, agricultural machines and industrial gearboxes. Generally tooth profiles of these gears are designed to be involute curve for power can be transmitted at accurate ratio consistently. However involute profile leads to sliding between meshing surface and consequentially brings about power loss unavoidably. Besides the sliding loss, power losses also come from the rolling resistance between meshing surfaces, air resistance during gear spin at high speed or windage loss and
also come from oil churning. The sliding and rolling loss can be categorized as the load-dependent loss, whereas the windage loss and churning loss are categorized as load-independent losses. For gears that are small and are operated in low and moderate speed conditions such as gears used in passenger vehicles or in agricultural tractors, power loss is mainly caused by friction from the sliding while power losses from the rolling friction and windage are very small. In those cases considering just only the sliding loss is sufficient to approximate the overall power loss. With this reason, a method for estimation of sliding loss in a paralleled-axis gear pair is focused in this paper.

There are many researches on power loss and efficiency of the gear systems. Y. Michlin and V. Myunster (1) proposed a model to predict the sliding and rolling losses in a spur gear pair by using geometric and kinematic analysis. The friction coefficient between gear tooth surfaces used in this research is considered to be a constant to simplify the calculation. However, it is known from the former research (2) that the friction coefficient is not constant throughout meshing period, but it depends on the sliding and rolling velocity and load that vary along the meshing process. Because the friction coefficient is very important parameter affecting the amount of the power loss, there are several researches which study on friction coefficient, and proposed the empirical formulae to determine the friction coefficient (2)-(6). These friction formulae are obtained by curve fitting of the results obtained from such twin-disk experiments, hence the uses of these formulae are restricted to match with some conditions depended on their base experiments. To solve these restrictions, there are some researches tried to determine the friction coefficient base on EHL model (7), (8). Estimation of the power loss by using the friction coefficient based on EHL method seems to give the satisfy results, however due to the complicated calculation that requires long computational time, the method base on EHL model is not suitable for practical implementation. With this reason, a simplified model with acceptable accuracy is still required for estimation of the power loss of gear transmission systems. The use of a suitable empirical formula of friction coefficient to determine the power loss is probable the good compromise between the practical implementation and the accuracy of the results and will be used here.

In this paper the simplified gear meshing model for estimation of sliding loss in a paralleled-axis gear pair is proposed. First the power loss in single tooth meshing spur gear pair is considered, and then the power loss of a helical gear pair will be described. The estimated power losses were compared with the experimental results and the estimated results done by the other researchers (9), (10). The various empirical formulae of friction coefficient are adopted in the estimation process. The suitable formula for estimation of the sliding loss is also proposed in this paper.

2. The estimation process

The process to estimate the sliding loss of a parallel-axis gear pair is shown by the diagram in Fig.1. First, gear parameters, tooth surface condition, viscosity of the lubricant oil and operating conditions are input into the estimation program. The gear parameters are used to calculate the lengths of contact lines at any arbitrary moment. Then the sum of lengths of contact lines is used along with the gear parameters and loading conditions to calculate the load per length further. Next, the friction coefficient throughout the meshing period is calculated by the several empirical formulae proposed by the former researchers. The gear parameters, the tooth surface roughness, viscosity of the lubricant oil, operating conditions, and the load per length are the necessary parameters used for calculation of the friction coefficient.

The coefficient of friction is used to determine the sliding loss ratio of single tooth meshing of a spur gear pair. Then the sliding loss ratio in the case of double teeth meshing of spur gear or in the case of helical gear pair can be estimated by combining the sliding
loss ratio of single tooth meshing together according to the helix angle and the meshing order. The details of the methods to determine the sliding loss of spur and helical gear will be described in sections 3 and 4 respectively, and the empirical formulae used to determine the friction coefficient will be described in section 5.

3. Sliding loss in a spur gear pair

3.1 Sliding loss

Sliding loss of a spur gear pair is caused by the difference in velocity vectors at the contact point of a gear and a pinion during meshing. The sliding of the tooth surface during meshing is represented in Fig 2. Gear $O_1$ is a driving gear, and gear $O_2$ is a driven gear. At this moment the gear pair is meshing at point $K$. The velocity vectors at point $K$ of the driving and driven gear are $v_1$ and $v_2$, respectively. From Fig 2, the directions of both velocity vectors at the meshing point are different. The velocity components in direction $(n-n)$ must be equal to make the driving gear contact with the driven gear throughout the meshing period. This makes the velocity component in $(t-t)$ direction are different, and sliding between the meshing tooth are occurred.
Because sliding velocity vectors change when meshing position is changed, sliding velocity in each period will be unequal. At the meshing point located before the pitch point the velocity of the contact point on the driven gear is faster than that on the driving gear. In contrast the velocity of the contact point on the driving gear is faster after the pitch point. These make the direction of friction force at the contact point before and after the pitch point are opposite. In addition, the sliding velocity is directly proportional to the distance from the pitch point to the contact point. Therefore, the maximum sliding velocity will occur at the tooth tips, and the minimum sliding velocity will be found at the pitch point where the sliding velocity is zero.

3.2 Power loss model for the spur gear pair

The sliding loss of a spur gear pair can be determined by the method proposed by Y. Michlin and V. Myunster (1). Figure 3 show the force acting at the driven gear in a spur gear pair during meshing. The meshing starts at point \(A\) and moves along the line of action, and finishes meshing at point \(B\). Point \(k\) is the meshing point at that moment. The transmitted force acting on the driven gear is shown by vector \(R_{12}\). The angle \(\alpha\) is the pressure angle. From the figure, the direction of force \(R_{12}\) is deviated from the line of action by the angle \(\theta\) due to the effect of friction. The sliding loss ratio \(\varphi\) can be determined in term of input power \(H_1\), output power \(H_2\) and power loss \(H_3\) by

\[
\varphi = \frac{H_1}{H_1} = \frac{H_1 - H_2}{H_1} \quad (1).
\]

The power \(H\) can be determined from the product of torque \(T\) and the rotational speed \(\omega\) as

\[
H_1 = T_1\omega_1 = R_{12} \cdot \overrightarrow{OF_1} \cdot \cos(\alpha + \theta_\mu) \cdot \omega_1 \quad (2),
\]

\[
H_2 = T_2\omega_2 = R_{12} \cdot \overrightarrow{OF_2} \cdot \cos(\alpha - \theta_\mu) \cdot \omega_2 \quad (3).
\]

Substituting Eqs.(2) and (3) into Eq.(1), and rearranging. Equation (1) becomes

\[
\varphi = -\frac{n \cdot \tan \alpha \cdot \mu \cdot (1 + m)}{1 - (n + 1) \cdot \tan \alpha \cdot \mu} \quad (4),
\]

where \(n\) is the position ratio relating to the meshing position and \(n = \frac{KP}{N_1P}\),

\(m\) is the speed reduction ratio and \(m = \frac{\omega_2}{\omega_1}\),

\(\mu\) is friction coefficient and \(\mu = \tan \theta_\mu\).
The sample result of the sliding loss ratio of single tooth meshing calculated from Eq.(4) is shown by a V-line in Fig.4. From the figure, the maximum sliding loss ratio occurs at the tooth tips, and the minimum is found at the pitch point.

### 3.3 The total sliding loss of a spur gear pair

The sliding loss ratio calculated from Eq.(4) based on the method proposed by Y. Michlin and V. Myunster are sliding loss ratio due to single tooth meshing which power is transmitted by only single tooth pair. In actual operation, there are some moments that the spur gear pair is meshing with more than one tooth pair. Therefore to estimate the total sliding loss ratio, the sliding loss ratio of each meshing pair must be combined together according to the meshing order.

Figure 5(a) shows the arrangement of sliding loss ratios of the meshing teeth according to the meshing order. Each V-line in the figure corresponds to the sliding loss ratio of one meshing pairs. The overlap parts show that double-teeth meshing occur at that moment. During double-teeth meshing, load is assumed to distribute equally among both teeth. Hence the power transmitted by each tooth pair can be determined by

\[ H_{Z1} = H_{Z2} = \frac{H_i}{2} \]  

where \( Z_1 \) and \( Z_2 \) are the first and the second tooth pair meshing at that moment. With this assumption, the total sliding loss ratio can be determined by

\[ \varphi_{total} = \frac{(Loss_{Z1} + Loss_{Z2})}{H_i} \]  

\[ \varphi \]

Meshing position

**Figure 4** The sliding loss ratio at each meshing position of single tooth meshing spur gear

**Figure 5** The power loss ratio of a spur gear pair
Sliding loss of the \( i \)th tooth pair can be calculated from

\[
\text{Loss}_{Zi} = \varphi_{Zi} \cdot H_{Zi}
\]  

(7).

Substituting Eqs.(5) and (7) into Eq.(6), and rearranging, Equation (6) becomes

\[
\varphi_{\text{total}} \cdot H_1 = \varphi_{Z1} \cdot \frac{H_1}{2} + \varphi_{Z2} \cdot \frac{H_1}{2} = \frac{H_1}{2} \left( \varphi_{Z1} + \varphi_{Z2} \right),
\]

\[
\varphi_{\text{total}} = \frac{\varphi_{Z1} + \varphi_{Z2}}{2}
\]

(8),

where \( \varphi_{Z1} \) and \( \varphi_{Z2} \) are sliding loss ratio of the first tooth pair and the second tooth pair. These sliding loss ratios can be calculated from Eq.(4).

Equation (8) shows that the sliding loss ratio of the double-teeth meshing is equal to the average value between the sliding loss ratios of both tooth pairs meshing at that time. The plot of the total sliding loss along with meshing position is shown in Fig.5(b). From the figure the maximum sliding loss ratios are occurs at the interval that double-teeth are meshing, and the minimum losses are at the pitch point \( P \). The dash line in Fig.5(b) shows the average value of the total sliding loss ratio \( \varphi \) which can be determined by

\[
\bar{\varphi} = \frac{1}{Pb} \int_{0}^{Pb} \varphi \, dn
\]

(9),

where \( Pb \) is base pitch.

4. Sliding loss in a helical gear pair

4.1 The forces acting on a helical gear

Because the teeth of helical gears are not parallel with the axis of rotation, but they are slantwise with the helix angle, the force acting on the tooth surface also lies slantwise according to the helix angle and the friction coefficient between contact surfaces. This force can be divided into three rectangular components that are the force along the axial direction, \( F_a \), the transverse force along the plane of action, \( F_p \), and the another transverse force \( F_f \) along the tooth surface perpendicular with force \( F_a \) and \( F_p \) as shown in Fig.6. The force \( F_p \) is the transmitted force, on the other hand the force \( F_f \) is the sliding friction between meshing surface that leads to the power loss. For the axial force \( F_a \), since there is no displacement along the axis, the power loss attributed to the axial force does not occur.

4.2 Power loss of a helical gear pair

Because only transverse forces \( F_p \) and \( F_f \) are relevant to the sliding loss of the helical gear pair, to estimate the sliding loss, a helical gear is divided along the transverse direction or face width direction into multi-sections. In other words, the helical gear is modeled as the multi-section spur gears as shown in Fig.7. Each section is aligned slantwise along with base helix angle, and is subjected to the transverse forces \( F_p \) and \( F_f \). Thus the sliding loss of each section can be considered as same as the loss of the spur gear pair that has been described in sections 3.2 and 3.3, and the whole sliding losses of all sections can be determined by combining the losses of each section together according to the helix angle and the meshing order.

Figure 8 shows the sample plot of the sliding loss ratios along with meshing positions of a helical gear pair. Figure 8(a) to (c) show the power loss ratio of the 1st, the 2nd, and the 3rd tooth pair respectively. In these figures, the helical gear is modeled as 5-section spur gears. Each V-line corresponds with the sliding loss ratio of each spur gear section. The positions of V-lines in each figure are aligned to match with the meshing position relevant
to the base helix angle and the base pitch. The interval that multi-teeth are in mesh is shown by the interval that the graphs in Fig.8(a) to (c) are overlapped. For example at the meshing moment “\( t \)”, there are 3 teeth in mesh. The 1st tooth is meshing with 2 spur gear sections (section #1 and #2), the 2nd tooth is meshing with 5 spur gear sections (section #3-#7), and the 3rd tooth is meshing with 3 spur gear sections (section #8-#10). Figure 9 shows the positions of section #1 to #10 of Fig.8 on the plane of action.

Here, the load distribution on the line of contact is assumed to be uniform, therefore the tooth pair having longer contact line must be subjected to larger load. Because the load distribution is uniform, the power transmitted by each spur gear section must be equal, and can be determined by

\[
H_{S1} = H_{S2} = \ldots = H_{Sn} = \frac{H_1}{nS}
\]

(10).

\( S1, S2 \) and \( Sn \) are the 1st, 2nd and \( nth \) tooth of spur gears and \( nS \) is the number of sections meshing at that time.

As same as the spur gear case, the power loss ratio of the helical gear pair can be determined from

\[
\varphi_{total} = \frac{(Loss_{S1} + Loss_{S2} + \ldots + Loss_{Sn})}{H_1}
\]

(11).

Sliding loss of the \( ith \) tooth pair can be calculated from

\[
Loss_{Si} = \varphi_{Si} \cdot H_{Si}
\]

(12).
Substituting Eqs. (10) and (12) into Eq. (11), and rearranging, Equation (11) becomes

\[ \phi_{total} \cdot H_1 = \phi_{S1} \cdot \frac{H_1}{nS} + \phi_{S2} \cdot \frac{H_1}{nS} + \ldots + \phi_{Sn} \cdot \frac{H_1}{nS}, \]

\[ \phi_{total} = \frac{\phi_{S1} + \phi_{S2} + \ldots + \phi_{Sn}}{nS} \]  \hspace{1cm} (13),

where \( \phi_{S1}, \phi_{S2} \) and \( \phi_{Sn} \) are sliding loss ratio of the 1st, the 2nd and the \( n \)th tooth pair, respectively. Each sliding loss ratio can be calculated from Eq. (4).

The total sliding loss ratio of the helical gear pair is shown in Fig. 8(d). The average value of the total sliding loss ratio can be calculated from Eq. (9) as same as the total sliding loss of the spur gear pair, and is shown by the dash line in Fig. 8(d).

Figure 8 The sliding loss ratio of a helical gear pair

Figure 9 Meshing position on the plane of action
4.3 The effect of the number of sections in modeling of the helical gear

From the former topic, the helical gear is modeled as the multi-section spur gears. The larger number of sections used, the model becomes closer to the helical gear. With this reason, the number of sections used for calculation affects directly to the accuracy of the estimated sliding loss. Figure 10 shows the relation between the number of sections and the average value of sliding loss ratio of some gear pairs. It is found that the values of sliding loss ratio slightly change along with the number of sections used, and it will converge when the number of sections is large enough. Figure 11 shows the minimum number of sections that makes the sliding loss ratio converges for the 800 helical gear pairs that their geometries come from the combination of various parameters shown in Table 1. The horizontal axis is the number of sections, and the vertical axis is gear set number (No.1, 2, 3…800). Each data point shows the number of sections must be used for that gear set. It is found from the figure that for the gear pairs having the parameters in the range shown in Table 1, the number of sections should be more than 36 sections to assure the convergence.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Range</th>
<th>Increment</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of teeth, ( N )</td>
<td>20-50</td>
<td>10</td>
</tr>
<tr>
<td>Transverse module, ( m_t ) (mm)</td>
<td>2-6</td>
<td>1</td>
</tr>
<tr>
<td>Transverse Pressure angle, ( \Phi_t ) (deg)</td>
<td>14.5,20</td>
<td></td>
</tr>
<tr>
<td>Helix angle, ( \Psi ) (deg)</td>
<td>15-30</td>
<td>5</td>
</tr>
<tr>
<td>Face width, ( FW ) (mm)</td>
<td>20,30,40,50,100</td>
<td></td>
</tr>
</tbody>
</table>
5. Determination of the friction coefficient

In former topic, the methods to estimate the sliding loss in a parallel-axis gear pair have been proposed. In the calculation process, the accurate value of friction coefficient is necessary to know for the sliding loss can be estimated accurately. From many experimental results, it is obviously known that many parameters such as sliding and rolling velocity, lubricant viscosity, load parameter and surface roughness affect the value of friction coefficient, and there are many researches that propose the empirical formulae for estimation of the friction coefficient. The several empirical formulae used in this paper are listed in Table 2. In these formulae, $v_k$ and $v$ are the kinematic and dynamic viscosities of the lubricant, $v_s$ is the relative surface sliding velocity, $v_r$ is sum of the rolling velocities, $R$ is the combined radius of curvature, $W$ is the unit normal load, $P_{\text{max}}$ is the maximum contact pressure, and $S$ is the surface finish parameter.

6. Verification with experimental results

To verify abilities of the method, the estimated results are compared with the experimental results done by the other researchers (9), (10). These experiments were done with the back-to-back gear test rig. The system power loss was measured by a precision torque

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Table 2 Empirical formulae used to determine the friction coefficient

<table>
<thead>
<tr>
<th>Empirical formulae</th>
<th>Published author</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\mu = 0.0127 \left[ \frac{50}{50 - S} \right] \log_{10} \left[ \frac{3.17(10)^8 W}{v_k v_r^2} \right]$</td>
<td>Benedict and Kelley $^{(2)}$</td>
</tr>
<tr>
<td>$\mu = \left[ 0.8 \sqrt{v_k v_s} + v_r \phi + 13.4 \right]^{-1}$</td>
<td>Drozdov and Gavrikov $^{(3)}$</td>
</tr>
<tr>
<td>$\phi = 0.47 - 0.13(10)^{-4} P_{\text{max}} - 0.4(10)^{-3} v_k$</td>
<td>ISO TC60 $^{(4)}$</td>
</tr>
<tr>
<td>$\mu = 0.12 W S / (R V_r v)^{0.25}$</td>
<td>Misharin $^{(5)}$</td>
</tr>
<tr>
<td>$\mu = 0.325 [v_r v_k]^{-0.25}$</td>
<td>O’donoghue and Cameron $^{(6)}$</td>
</tr>
<tr>
<td>$\mu = 0.6(S + 22) / 35[v^{1/8}v_s^{1/3}v_r^{1/6}R^{1/2}]^{-1}$</td>
<td></td>
</tr>
</tbody>
</table>

Table 3 Parameters of gears used in T.T. Petry-Johnson et al.’s experiments

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Gear set</th>
<th>23T</th>
<th>40T</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of teeth</td>
<td>23</td>
<td>40</td>
<td></td>
</tr>
<tr>
<td>Module (mm)</td>
<td>3.95</td>
<td>2.32</td>
<td></td>
</tr>
<tr>
<td>Pressure angle (deg)</td>
<td>25</td>
<td>28</td>
<td></td>
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<tr>
<td>Face width (mm)</td>
<td>19.5</td>
<td>26.67</td>
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<tr>
<td>Absolute Viscosity (cP)</td>
<td>10.7</td>
<td>10.7</td>
<td></td>
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<tr>
<td>Surface roughness ($\mu$m)</td>
<td>0.32</td>
<td>0.2</td>
<td></td>
</tr>
<tr>
<td>Load operation condition (Nm)</td>
<td>140, 275, 413, 546, 684</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Speed operation condition (rpm)</td>
<td>2000, 4000, 6000, 8000, 10000</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
meter. In the experiments, the mechanical power loss that is the summation of sliding loss and rolling loss was determined by subtract the spin loss and bearing losses from the measured system power loss. Although the mechanical power losses presented in these researches are not exactly the same as the sliding losses estimated in this paper because they are also comprised of the rolling losses, it is known from the former researches (1), (11) that the magnitude of the rolling losses are very small comparing with the sliding losses, hence the sliding losses can be approximated the same and are comparable to the mechanical losses.

T.T. Petry-Johnson et al. (9) measured the power loss of spur gears operated at various speeds and loads. The gear parameters and relevant information are shown in Table 3. The gear ratio is 1:1. Figure 12 shows the comparison of the results estimated from various friction formulae including the constant friction coefficient with the experimental results.

![Figure 12 Comparison of the estimated results and the experimental results done by T.T. Petry-Johnson et al.](image-url)
The figures in the left hand side show the effect of rotational speeds on the sliding loss. The loads in these results are fixed at 413 Nm. On the other hand, the figures in the right hand side are the results at various load conditions. The speeds in these cases are fixed at 6,000 rpm. The black lines with solid marks are experimental results and the dash lines with blank marks are estimated results.

It is obvious from all results that the sliding losses increase with increasing loads and speeds, and the gear pair having 23 teeth (having larger module) has higher losses than

Table 4 Parameters of gears used in A. Vaidyanathan’s experiments

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Gear set</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>A</td>
</tr>
<tr>
<td>Number of teeth</td>
<td>23</td>
</tr>
<tr>
<td>Transverse module, m&lt;sub&gt;t&lt;/sub&gt; (mm)</td>
<td>3.95</td>
</tr>
<tr>
<td>Transverse pressure angle, Φ&lt;sub&gt;t&lt;/sub&gt; (deg)</td>
<td>25</td>
</tr>
<tr>
<td>Helix angle, Ψ (deg)</td>
<td>0</td>
</tr>
<tr>
<td>Face width (mm)</td>
<td>26.67</td>
</tr>
<tr>
<td>Absolute Viscosity (cP)</td>
<td>6.732</td>
</tr>
<tr>
<td>Surface roughness (µm)</td>
<td>0.37</td>
</tr>
<tr>
<td>Load operation condition (Nm)</td>
<td>140, 239, 413, 546</td>
</tr>
<tr>
<td>Speed operation condition (rpm)</td>
<td>2000, 4000, 6000</td>
</tr>
</tbody>
</table>

Figure 13 Comparison of the estimated results and the experimental results done by A. Vaidyanathan (effect of speeds)
those of the 40 teeth-gear. Although in these points all estimated results agree with the experimental results, they are different in details. All estimated sliding losses are higher than experimental results. From the relation between power losses and rotational speeds, the estimated results those use the coefficient of frictions estimated by the formulae proposed by Benedict and Kelly and by O’donoghue and Cameron seem to give the most similar shape of trend line that the increment of the power loss slightly reduces at high speed. From the relation between the power losses and applied torque, the graphs of all estimated results have higher slopes than those from experiments. This means the power loss increasing rate is overestimated when applied load is increased. It is also found from the results that the used of the constant coefficient of friction in the calculation may give the satisfy results, if the suitable value of the friction coefficient is used.

The estimated results in the case of the helical gear pairs are compared with the experimental results done by A. Vaidyanathan (10). The parameters of gears used in the experiments are shown in Table 4. The gear ratio is 1:1. The comparison between the estimated results and the experimental results in the case of the gear pair B are shown in Figs.13 and 14. From the results, the sliding losses are increase with increasing operating speeds and loads. The results estimated using the formula proposed by Benedict and Kelly are the most conformable to the experimental results. The trends of sliding loss variation with speed and load are almost the same, and the magnitudes of the estimated sliding losses are just slightly higher than the experimental results. On the other hand, the results estimated from the other formulae differ from the experimental results both the magnitudes and the forms of graphs. The results estimated by using the constant coefficient of friction
that seem to make good approximations with the experimental results in Fig.12, do not agree well in these cases. This is because the coefficients of friction change according to the operating conditions. The constant coefficient of friction can be selected to match with a specific driving condition that lead to the good estimated result, but when the operating conditions are changed the coefficient of frictions must also be changed consequently therefore the use of constant friction coefficient will lead to erroneous results.

Figure 15 shows the comparison of the sliding loss of various gear pairs having different parameters. The gear parameters are shown in Table 4. The torques applied to all of the gear pairs in Fig.15 are set to be constant at 546 Nm. In the figure, $\Phi$ is transverse pressure angle and $\Psi$ is helix angle. The effect of helix angles can be known by comparing the gear pair A and C which have almost the same transverse pressure angles but different helix angles. On the other hand to find the effect of pressure angle, the gear pair B and C which have the same helix angle are compared.

From all results, for the gears pairs A and C that have almost the same transverse pressure angle, the helical gear pair C having larger helix angle has higher sliding loss than the spur gear pair A. On the other hand when the helix angle is fixed, the gear pair C that has smaller transverse pressure angle has higher sliding loss than the gear pair B. As same as the results in Figs. 13 and 14, the estimated sliding losses are slightly higher than the
experimental results except in the case of the results estimated by using constant coefficient of friction equal to 0.3 and by using Drozdov and Gavrikov’s formula. The Benedict and Kelly’s formula and O’donoghue and Cameron’s formula seem to give the better approximation of the amount of sliding loss than the others.

Although the method proposed here can be used to estimate the effect of either transverse pressure angle or helix angle correctly, but if there are both variations in transverse pressure angle and helix angle simultaneously, the estimations done by using some friction formulae are not correct. From the experimental results shown in Fig.15, the gear pair A has lower transverse pressure angle and lower helix angle than gear pair B, and sliding loss in this case is slightly lower than that of the gear pair B. The results estimated using constant coefficient of friction, using Drozdov and Gavrikov’s formula and using Misharin’s formula have the same trend with the experiment. In contrast the results estimated using Benedict and Kelly’s formula, ISO TC60 formula, and O’donoghue and Cameron’s formula show the opposite results that the gear pair A has higher sliding loss than the gear pair B.

7. Verification with other estimated results

The estimated results are also compared with the estimations done by A. Vaidyanathan\(^{(10)}\). In his estimations, the EHL model is used to predict the friction coefficient. The comparison results in the case of the gear pair B in table 4 are shown in Fig.16. Most of the estimated results have higher losses than the A. Vaidyanathan’s estimations. By comparing both estimations with the experimental results shown in Fig.14, it is found that the method

![Figure 16 Comparison of the estimated results and the estimated results done by A. Vaidyanathan](image-url)
presented here tend to overestimate the sliding loss, on the other hand the estimated results done by A. Vaidyanathan are underestimate. The results estimated by using Benedict and Kelly’s formula seem to give better prediction comparing with A. Vaidyanathan’s results.

8. Conclusion

The simplified gear meshing model and the method used to estimate sliding loss of a parallel-axis gear pair is presented in this paper. The estimated results show the same trend with the experimental results. The sliding loss increases with increasing loads and speeds. The gear pair with larger module has higher sliding loss. Increasing transverse pressure angle will reduce the sliding loss, on the other hand increasing helix angle will increase the sliding loss. The friction coefficient formula proposed by Benedict and Kelly is the most suitable to use for sliding loss estimation.

Acknowledgement

This work is a part of the research “A Study on the mechanism and the parameters affecting to sliding loss of a parallel-axis gear pair” that was financial supported by The Thailand Research Fund (MRG5480164).

References

(4) ISO TC 60, DTR 13989