Development of an Efficiency Model for Manual Transmissions

A computational model was proposed by GM Powertrain Europe and Ohio State University, Columbus (USA), to predict friction-related mechanical efficiency losses of manual transmissions. Individual power losses from gear meshes are combined in an efficiency model based on elasto-hydrodynamic lubrication theory with the bearing losses to determine the total mechanical power losses of the transmission. The model was used to predict mechanical power losses of the F40 six-speed manual transmission of GM Powertrain. These predictions were shown to agree well with the power loss measurements from the same transmission.

1 Introduction

Power loss experienced by drivetrains of passenger vehicles has been one of the major concerns in automotive powertrain engineering. Such losses directly impact fuel consumption of the vehicle, helping define how good the vehicle is in terms of its fuel economy and gas/particulate emission levels. While fuel consumption alone is a sufficient reason for seeking reduced drivetrain power losses, there are other auxiliary reasons as well. Excessive power losses within the transmission amounts to additional heat generation and higher temperatures, thus, adversely impacting gear contact fatigue and scuffing failure modes [1]. In addition, the design of the lubrication system as well as quantity of the lubricant within the transmission is also related to the amount of heat generated.

In a manual transmission [12] sources of power losses can be classified into two groups: (i) load-dependent (friction induced) mechanical power losses P_m and (ii) load-independent (viscous) spin losses P_s, meaning of abbreviations see the **Table**. Overall power loss P_T of the transmission is the summation of the two. Mechanical losses are largely defined by sliding and rolling friction losses of the loaded gear meshes and rolling friction losses of the bearings while spin losses are caused by a host of factors including viscous dissipation of bearings, oil churning and windage.

Loaded gear pairs experience combined sliding and rolling, which result in frictional losses. These losses are related to the coefficient of friction, normal load on the tooth and sliding velocity on the gear surface [1, 13]. Rolling frictional losses occur due to the formation of an elasto-hydrodynamic (EHL) film [1]. Windage and churning losses are present as a result of oil/air drag on the face and sides of the gears as well as in the meshing zone. Similarly, rolling element bearings of the transmission also account for mechanical load-dependent and spin losses of their own. Accordingly, the bulk of the load-dependent losses in a manual transmission is accounted for by considering gear mesh (P_{mesh}) and bearing (P_b) losses such that $P_m \approx P_{mesh} + P_b$. This paper focuses on prediction of mechanical power loss P_m of a six-speed manual transmission while the predictions of spin losses will be addressed in another work.

There has been a significant body of research on gear train efficiency as reviewed in papers by Martin [2]. These studies can be grouped into three categories based upon the methodology used to model friction. The first set of studies (for example ref. [3]) investigated gear pair efficiency by using a uniform user-defined coefficient of friction μ along the contact surface. The assumption of a constant μ in these models contradicts another group of published work on traction of contacts

Table: Abbreviations

Name	Unit	Explanation
P_{mesh}	W	gear mesh losses
P_{sg}	W	speed gear pair mechanical power loss
P_{fd}	W	final drive gear mesh power loss
P_b	W	bearing losses
$P_{b, in}$	W	bearing mechanical power losses of the input shaft
$P_{b, t1}$	W	bearing mechanical power losses of the first output shaft
$P_{b, t2}$	W	bearing mechanical power losses of the second output shaft
$P_{b, out}$	W	bearing mechanical power losses of the differential
P_m	W	load-dependent, mechanical power losses
P_S	W	load-independent, viscous spin losses
P_T	W	overall power losses
P_{in}	W	input power
S	µm	or RMS (Root mean square), in Europe: R_q

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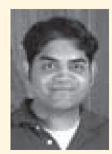
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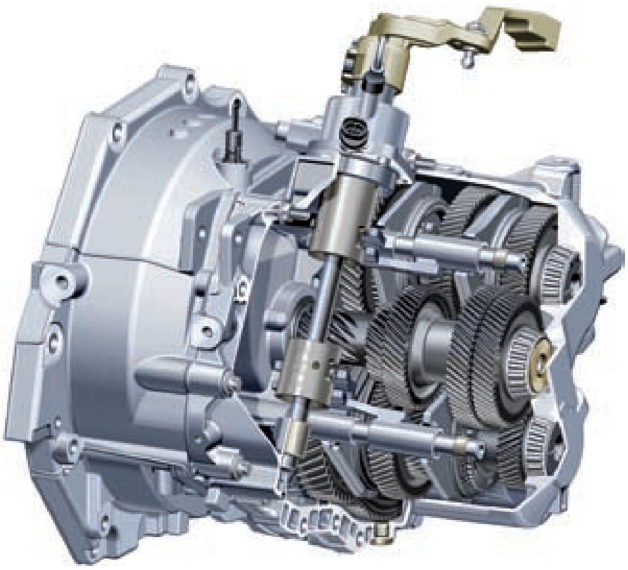

Figure 1: GMPT F40 6-speed manual transmission

having combined rolling and sliding. In studies mentioned in ref. [4], empirical μ formulae were presented based on actual roller contact experiments to show that μ varies with a number of parameters including lubricant viscosity, sliding and rolling velocities, radii of curvature, surface characteristics and operating load. Every contact point in a gear mesh interface is subjected to different values of these parameters, suggesting that μ is not constant. In an attempt to overcome this shortcoming, a second group of studies (for example ref [5]) employed these empirical μ formulae. This approach also had limited success since the empirical μ formulae adapted often did not represent the operating parameter ranges of gear meshes. The third group of studies (for example ref. [6]) took on a physics-based approach by using elasto-hydrodynamic lubrication (EHL) formulations to compute the variation of μ across the gear contacts. This EHL-based approach was shown to work well for spur gears with idealized load distribution while the computational time required was very significant. A low-loss gearing according to [14, 15] with different gearing parameters is not being discussed in this paper, the aim is to gain optimization parameters by looking at micro geometrical details.

In order to reduce the computational demand of EHL-based predictions, Xu et al [1] proposed a new friction coefficient relationship for gear contacts, which is found by applying multiple linear regression analysis to a large number of EHL simulations covering a wide range of all key contact parameters. This μ formula was first validated by comparisons to direct simulations from a mixed-EHL model as well as actual traction measurements over a wide range of speed, load, roughness and sliding conditions. Then this μ formula was incorporated with a gear load distribution model (LDP) to predict instantaneous mechanical gear mesh power losses and efficiency of a spur or helical gear pair. It was also shown in the same study [1] that the predicted gear mesh mechanical losses agreed well with measurements from spur gear pairs having different tooth sizes.

The literature on the power losses of geared transmissions is very sparse. As one significant body of work, Velex et al [7] proposed a power loss model for a 6-speed manual transmission using thermal networks. The model predicts temperature distribution and efficiency by considering several sources of dissipation such as rolling and sliding friction, rolling bearing elements and oil shearing taking place in the synchronizers.

In this study, a generalized mechanical efficiency model for manual transmission will be developed and applied to the F40 transmission shown in **Figure 1** by employing the mechanical gear mesh power loss model of Xu et al [1] and bearing power loss model of Harris [8]. The specific objectives of this study are as follows:

- Develop a mechanical power loss model for multi-axis manual transmissions and apply it to GM Powertrain's F40 transmission shown in Figure 1.
- Validate the transmission mechanical power loss model by comparing the predictions to measured values collected from the same transmission under tightly controlled conditions.
- Investigate the influence of some of the key parameters affecting mechanical power losses including surface roughness of gear teeth, lubricant temperature, as well as complex transmission duty cycles.

2 Transmission Power Loss Model

Figure 2 shows the flowchart for the transmission power flow model. Input torque T_j, input speed ω_j, gear state g_j ($g_j \in [1, G]$), an integer, where G is the number of gear ranges) and the average transmission temperature θ_j from a duty cycle at a discrete time instant j ($j \in [1, J]$) are input to a transmission power flow model. This model identifies the loaded gear meshes according to the value of g_j and determines the forces transmitted by each loaded gear mesh as well as the rotational speed of the loaded gears. For the F40 transmission shown in Figure 1, there are six forward gear ratios ($G = 6$) allowing power to be transmitted by different paths as shown in **Figure 3**. Here the differential gear pair is always loaded regardless of the value of $g(t)$ while a different speed gear pair is loaded at each gear range. The speed and gear mesh forces at each loaded gear mesh for given values of T_j and ω_j are input to the gear pair mechanical power loss model to predict the power losses of the speed gear pair P_{sg} and the final drive gear pair P_{fd} at a given g_j. Accordingly, the total transmission gear mesh power loss is defined as:

$$P_{mesh} = P_{sg} + P_{fd} \qquad \text{Eq. (1)}$$

For the F40 transmission in a forward gear ratio, gear mesh power loss computations are done twice, once for the load carrying speed gear pairs and then for the final drive gear pair. This is captured in Figure 2 by an integer index $k \in [1, K]$ where $K = 2$ for most manual transmission forward gear ranges including those of F40.

Similarly, given the gear mesh force vectors $F_{sg}(t)$ and $F_{fd}(t)$, and position vectors at the gear meshes defined relative to the same reference frame are used to compute the forces and moments carried by the bearings of each shaft. For the input and output shafts there is only one loaded gear pair, making this calculation rather simple, while the calculation of the forces on the bearings of the transfer shafts require more effort and additional parameters such as the angle of the shaft centers of the loaded meshes. Knowing the bearing loads and speeds, the total bearing power losses can be computed. For the F40 transmission, shown in Figure 1, the following equation is established:

$$P_b = P_{b,in} + P_{b,t1} + P_{b,t2} + P_{b,out} \qquad \text{Eq. (2)}$$

where $P_{b,in}$, $P_{b,t1}$, $P_{b,t2}$ and $P_{b,out}$ are individual bearing mechanical power losses of the input shaft, the first and second output shafts, and the differential, respectively. Here, mechanical power losses for each shaft must be computed, even when one of the shafts does not carry any load. This is due to the fact that each bearing operates under preloaded conditions.

The total gear mesh and bearing mechanical power losses are then added to find total (gear and bearing related) mechanical power losses of the transmission as $P_m = P_{mesh} + P_b$ and the mechanical efficiency of the transmission is given as:

$$\eta_m = 1 - \frac{P_m}{P_{in}} \qquad \text{Eq. (3)}$$

where the input power is given as $P_{in} = T_j\,\omega_j$. These computations are repeated at every discrete point of the user defined duty cycle given by T_j, ω_j, g_j and θ_j to predict P_m and η_m as a function of the duty cycle.

2.1 Helical Gear Pair Power Loss Model

The gear mesh mechanical power loss model used in this study combines a gear load distribution model, a gear contact friction model and a gear mechanical efficiency formulation [1]. The gear load distribution model predicts the load and contact pressure distribution at every contact point over the tooth surface of the gear, as the mesh position is incremented. The corresponding load distribution along with geometric parameters is input to the gear contact friction model, which determines distribution of μ along the contacting surfaces. This friction coefficient value is then input to the mechanical efficiency formulation to determine the instantaneous power loss at an incremental rotational position n ($n \in [1, N]$). The same procedure is repeated at other discrete rotational positions until $n = N$, when a complete gear mesh cycle of rotation is achieved. Following this, the instantaneous mechanical efficiency values over the entire mesh cycle are then averaged to obtain the average mechanical power loss of the gear pair at the given instantaneous duty cycle condition.

The gear load distribution model (LDP) [9] computes the elastic deformation at any point on the gear surface given the tooth compliance, input torque and initial unloaded tooth separations. Compatibility and equilibrium conditions are considered for the solution to the contact problem. The load distribution problem is solved for each angular position of the gears by using a Simplex algorithm. The model then computes the load distribution along the lines of contact as well as maximum Hertzian pressure based on an equivalent cylindrical contact. Radii of curvature and surface velocities at each contact point on the gear pair, as needed in the friction and efficiency computation, are also calculated by LDP.

Xu et al [1] used the thermal EHL model of Cioc et al. [10] as the basis to predict μ distribution along the gear contacts. Consider any contact point having certain values of sliding and entrainment velocities, $V_s = u_1 - u_2$ und $V_e = (u_1 + u_2)/2$, combined radius of curvature, $R = r_1 r_2/(r_1 + r_2)$, where u_1 and u_2 are velocities of the contacting surfaces in the direction of sliding and r_1 and r_2 are the radii of curvatures of the contacting surfaces on the transverse plane of the gears. By solving the transient Reynolds equation along with a film thickness equation, a viscosity-pressure-temperature relationship, a density-pressure-temperature relationship and the energy balance equa-

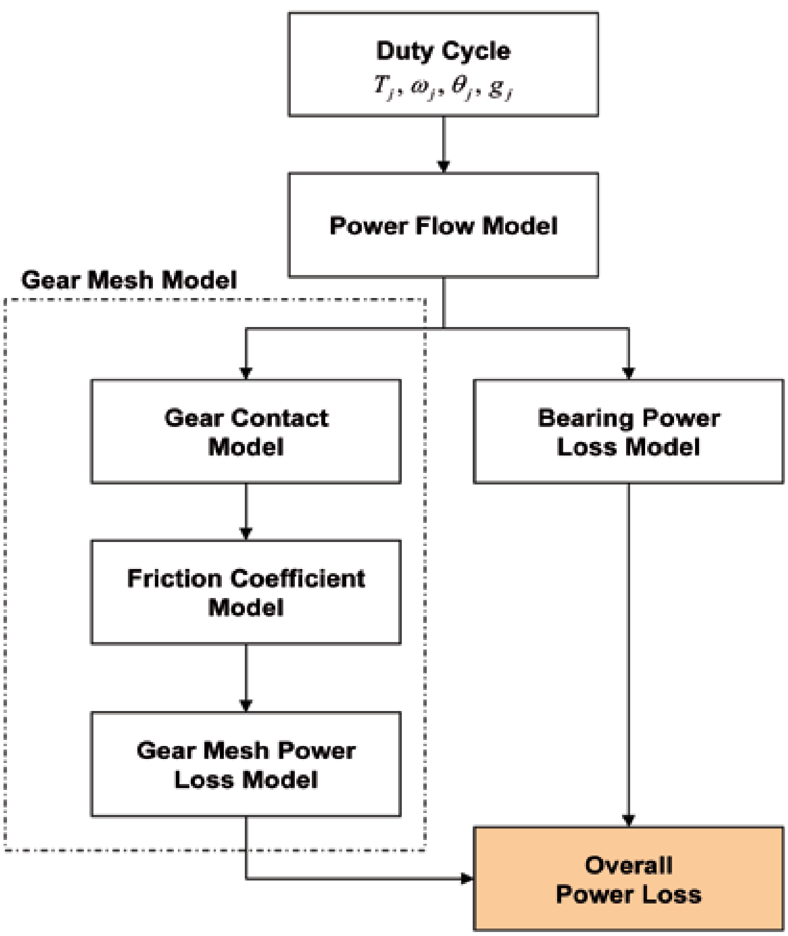

Figure 2: Flowchart of the transmission power loss model

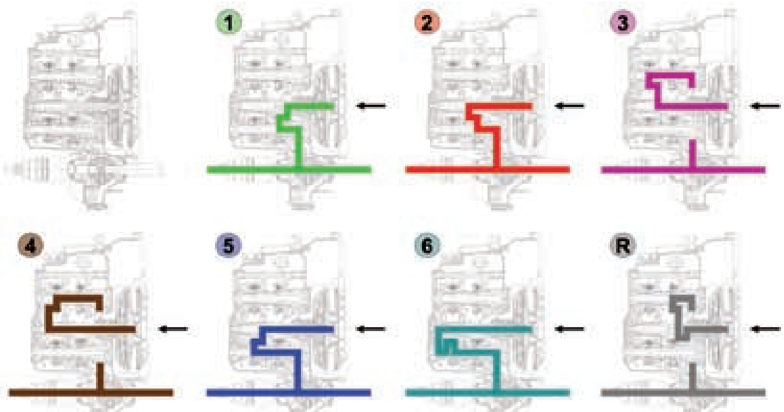

Figure 3: Power flows of the F40 transmission

tion, the thermal EHL model predicts the film pressure distribution $p(x)$, viscosity distribution $v(x)$ and film thickness distribution $h(x)$ across an elastic lubricated contact zone [1]. The friction force per unit width of contact segment is then given as the sum of the rolling and sliding components of the friction force as:

$$F_t' = \int \tau(x) dx = F'_r + F'_f \qquad \text{Eq. (4a)}$$

where rolling and sliding friction components are defined as:

$$F_r' = -\int \frac{h(x)}{2} \frac{\partial p}{\partial x} dx \qquad \text{Eq. (4b)}$$

$$F_f' = \int v(x) \frac{u_1 - u_2}{h(x)} dx + \int \mu_s p(x) dx \qquad \text{Eq. (4c)}$$

Here μ_s is the coefficient of friction between the asperities under mixed EHL conditions. The friction coefficient value is calculated as $\mu = F'_f / W'$ where W' is the total load per width of the contact segment.

The main disadvantage of using the EHL model to compute µ within a gear efficiency model is its computational demand. To overcome this difficulty, Xu et al [1] first performed a large number of EHL analyses for the lubricant considered within typical ranges of all key contact parameters representative of common automotive gear contact conditions. These several thousand cases covered combinations of surface characteristics, operating conditions and contact pressure distribution. Then a multiple linear regression analysis of the analyzed EHL results was performed to reduce these simulations into a single friction coefficient formula:

$$\mu = e^f P_h^{b_2} |SR|^{b_3} V_e^{b_6} v_0^{b_7} R^{b_8} \qquad \text{Eq. (5a)}$$

where

$$f = b_1 + b_4 |SR| P_h \log_{10}(v_0) + b_{5^{e^{-|SR| P_h \log_{10}(v_0)}}} + b_{9^{e^S}} \qquad \text{Eq. (5b)}$$

This formula contains most of the key parameters including the absolute viscosity of the lubricant v_0 (in cPs), the RMS composite surface roughness S (in µm), the effective radius of curvature R (in meters), the contact pressure P_h (in Pa) and the sliding ratio $SR = 2(u_1 - u_2)/(u_1 + u_2)$. In Eq. 5a, b_i (i = 1, 2, ..., 9) are constant coefficients that are different for every lubricant. This formula was shown to agree well with both actual EHL analyses and transition measurements.

After finding of friction coefficient at each contact segment centered by the coordinates (z, θ) at each incremental rotational angle ϕ_n (n = 1, 2, ..., N), the sliding friction forces is calculated as:

$$F_s(z, \theta, \phi_n) = \mu(z, \theta, \phi_n) W(z, \theta, \phi_n) \qquad \text{Eq. (6a)}$$

where $W(z, \theta, \phi_n)$ is the normal force at the center of the contact segment. In addition, using a smooth surface assumption, the rolling friction force is given in its approximate form as [1]:

$$F_r(z, \theta, \phi_n) = \frac{4.318}{\alpha} \varphi_T (\tilde{G}\tilde{U})^{0.658} \tilde{Q}^{0.0126} R \qquad \text{Eq. (6b)}$$

with φ_T being the thermal reduction factor to account for temperature rise at higher speeds, $\tilde{G}$ is the dimensionless material parameter, $\tilde{U}$ is the dimensionless speed parameter, $\tilde{Q}$ is the dimensionless load parameter, and α is the pressure-viscosity coefficient. When friction forces at each contact point are computed, the instantaneous gear mesh mechanical power loss is given as:

$$\tilde{P}_{mesh}(\phi_n) = P_{in} - \sum_{q=1}^{Q} [|F_s(u_1 - u_2)| + |F_r(u_1 - u_2)|] \qquad \text{Eq. (7)}$$

where $q \in [1, Q]$ represents a discretized load distribution segment and Q is the total number of segments. The average value of the power loss over a mesh cycle is given as $\tilde{P}_{mesh} = \frac{1}{n} \sum_{n=1}^{N} \tilde{P}_{mesh}(\phi_n)$. If this particular gear mesh is a speed gear pair, then $P_{sg} = \tilde{P}_{mesh}$ in Eq. 1; if it is the final drive mesh, then $P_{fd} = \tilde{P}_{mesh}$.

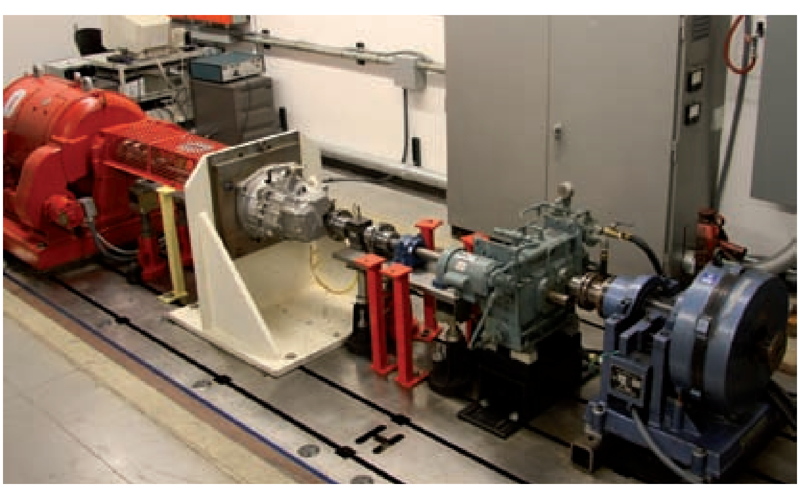

Figure 4: Experimental set-up for measuring the power losses of the F40 transmission

2.2 Rolling Bearing Element Power Losses

The axial and radial components of forces acting on bearings of the transmission shown in Figure (1) are computed at each duty cycle increment. Given the force vector of a bearing i on shaft s as $\mathrm{F}_{b,s}^{(i)} = [F_{b,s}^{(i,x)}, F_{b,s}^{(i,y)}, F_{b,s}^{(i,z)}]$, the radial bearing force is defined as $F_{b,s}^{(i,r)} = [(F_{b,s}^{(i,x)})^2 + (F_{b,s}^{(i,y)})^2]^{1/2}$, while the axial force $F_{b,s}^{(i,z)}$ is adjusted to incorporate the preload on shaft s. With this, the moment caused by the friction in the bearing i is calculated as [8]:

$$M_s^{(i)} = \hat{F} \lambda d \qquad \text{Eq. (8a)}$$

where λ is a material factor, d is the mean bearing diameter, and $\hat{F}$ is the effective bearing force obtained from force and momentum balance. The value of $\hat{F}$ is taken to be the greater of $F_{b,s}^{(i,r)}$ and $\frac{4}{5} F_{b,s}^{(i,z)} \cot \beta$, where β is the bearing contact angle. Once $M_s^{(i)}$ is calculated for each bearing i on each shaft s, the power loss due to friction on the all I number of bearings on shaft s ($s = in, t1, t2, out$) is calculated as:

$$P_{b,s} = \sum_{i=1}^{I} M_s^{(i)} \omega_s \qquad \text{Eq. (8b)}$$

where ω_s is the rotational speed of shaft s.

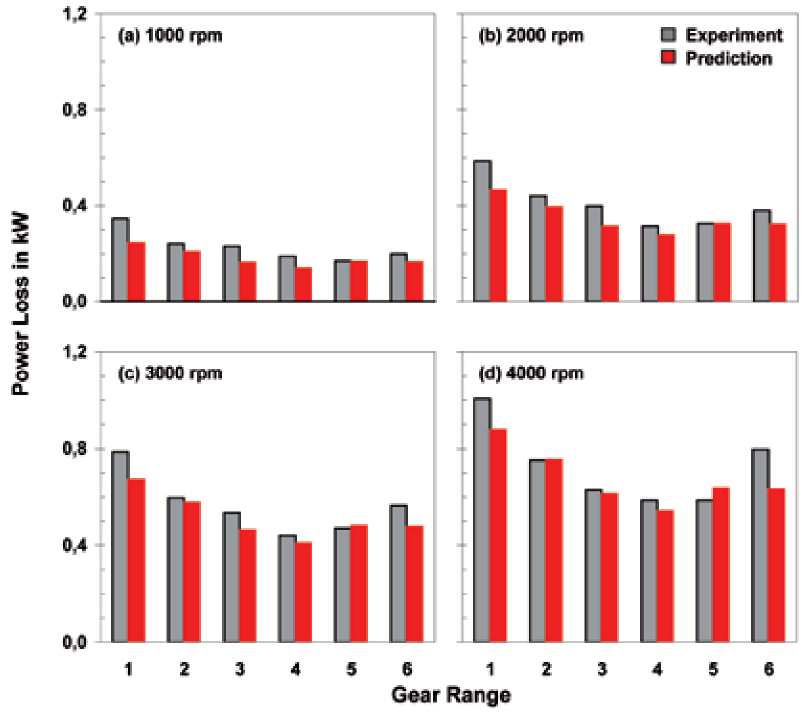

Figure 5: Comparison of measured and predicted P_m values of the F40 transmission at T = 100 Nm, θ = 80° C und S = 0.56 µm; (a) Ω = 1000 pm, (b) Ω = 2000 pm, (c) Ω = 3000 rpm, and (d) Ω = 4000 pm

3 Results

In the following the results of validation of the model predictions and parametric studies are described.

3.1 Validation of the Model Predictions

As mentioned earlier, the friction coefficient model Eq. (5) of the gear mesh efficiency model was validated in ref. [1] through comparison to sliding contact friction measurements. The gear pair power loss model was also validated using tightly controlled spur gear power loss experiments [11]. While these provide a certain level of confidence on the transmission power loss predictions, it is still necessary to validate the transmission mechanical efficiency model by using the actual transmission power loss measurements, since (a) the gears of the transmission are helical type and (b) multiple gear meshes and bearings contribute to the transmission power losses. For this purpose, an experimental setup was devised to measure the power losses of the F40 transmission under varying torque, speed and temperature conditions at gear each gear range. **Figure 4** shows the dynamometer setup used for this purpose. Here, a support bracket between two precision torque-meters holds an F40 transmission. A DC motor drives the transmission and an eddy-current load brake dynamometer provides the reaction torque through a speed increaser. The temperature of the transmission is measured by a thermocouple inserted from the bottom drain hole.

The test setup uses a 110 kW electric motor and a 400 kW brake to load the transmission at input speeds up to 5000 rpm. A pair of precision torque-meters are mounted next to the transmission to measure the input and output torque. A thermocouple is inserted into the transmission in the vicinity of the differential ring gear to measure the instantaneous oil temperature.

The tests were repeated at various input speeds of $\omega_j = \omega_{in}$ and gear state g values with and without a given level of input torque T_j. For the unloaded case of $T_j = 0$, the torque measured at the output side is zero and the torque loss measured at the input side τ_{in} is used to compute the spin loss $P_s = \omega_{in} \tau_{in}$. When the same test is repeated with a given nonzero torque value $\mathrm{T}_j \neq 0$, measured torque levels τ_{in} and τ_{out} can be used to calculate the total power loss as $P_T = \omega_{in} \tau_{in} - \omega_{out} \tau_{out}$. Here $\omega_{out} = \omega_{in} / \Gamma_j$ where Γ_j is the overall speed reduction ratio of the transmission at this particular gear range. From these two measurements, the mechanical loss of the transmission is calculated as $P_m = P_T - P_s$.

Figure 5 compares the measured and predicted P_m values of the F40 transmission at $\theta = 80°$ C and $T = 100$ Nm for $\Omega = \frac{60\omega_{in}}{2\pi} = 1000$ bis 4000 rpm. Here, every gear pair has a composite root-mean-square surface roughness of $S = 0.56$ µm. As is evident from this figure, the predicted and measured mechanical power loss values agree well over the entire range of operating conditions and at all gear stages of the transmission, with maximum deviations between the measured and predicted power loss values well within 0.2 kW. This suggests that the transmission mechanical power loss model proposed above is sufficiently accurate to be used for engineering studies to reduce power losses.

3.2 Parametric Studies

In part a) of **Figure 6**, the predicted contributions of individual gear meshes and bearings to the total mechanical power loss are shown for the same case of θ = 80° und T = 100 Nm, S = 0.56 µm, Ω = 4000 rpm. Here, the final drive gear mesh power loss P_{fd} is seen to experience a slight linear decay as the g increases. Considering that the final drive gear pair remains the same for all the gear ranges, this slight reduction in P_{fd} with increased g can be attributed solely to the increase in input speed and decrease in torque transmitted by this gear pair. The speed gear pair mechanical power loss P_{sg} is also seen to decrease as the transmission is shifted upwards from the 1st gear range to the forth gear range, increasing beyond that from the fifth gear stage onwards. This is mainly due to the fact that each gear range involves a completely different speed gear pair having its own speed ratio and geometry. In Figure 6a, at the first gear range $P_{sg} \approx 0.45\ P_m$, $P_{sg} \approx 0.35\ P_m$ and $P_b \approx 0.2\ P_m$ while $P_{sg} \approx P_{fd} \approx 0.35\ P_m$ and $P_b \approx 0.3\ P_m$ at the 6th gear range. This suggests that any effort to minimize mechanical power loses for this case should focus initially on the losses of gear meshes.

The components of the total bearing power loss for the same case are shown in Figure 6b. For bearings on the input shaft, the rotational speed remains constant at all the gear ranges. For bearings on the output shafts and differential shaft, ω_S is a function of the gear stage and hence the reduction ratio between the speed gear mesh. The bearing power loss on the input shaft $P_{b,\,in}$ shows a decreasing trend. This is because, while input speed remains constant, the loads on the bearings of the input shaft decrease from the first to the sixth gear range. Meanwhile, $P_{b,\,t1}$, $P_{b,\,t2}$ and $P_{b,\,out}$ all experience slight reductions with increased gear range, resulting in a relatively flat bearing power loss P_b regardless of the gear range.

In part a) of **Figure 7**, the influence of surface roughness on mechanical power loss P_m of the F40 transmission is shown at θ = 80° C, T = 100 Nm, and Ω = 4000 rpm. The lambda ratio (the film thickness to surface roughness ratio $\lambda = \frac{h}{S}$) reduces with increase in surface roughness moving the lubrication conditions to a partial-EHL regime. This increases actual asperity contacts to cause increased µ values according to Eq. 5, resulting in sizable increases in gear mesh mechanical power losses. From Figure 7 a, it is observed that P_m is reduced by 40 to 50 % through reducing the composite surface roughness of gear pairs from S = 0.62 µm (a typical value for a ground gear pair) to S = 0.1 µm with actual P_m reductions ranging from 0.45 kW at the firstt gear range to 0.25 kW in the forth gear range.

Similarly, Figure 7b shows the influence of oil temperature on P_m of the F40 transmission at T = 100 Nm, Ω = 4000 rpm and S = 0.56 µm. Significantly lower mechanical power losses are ob-

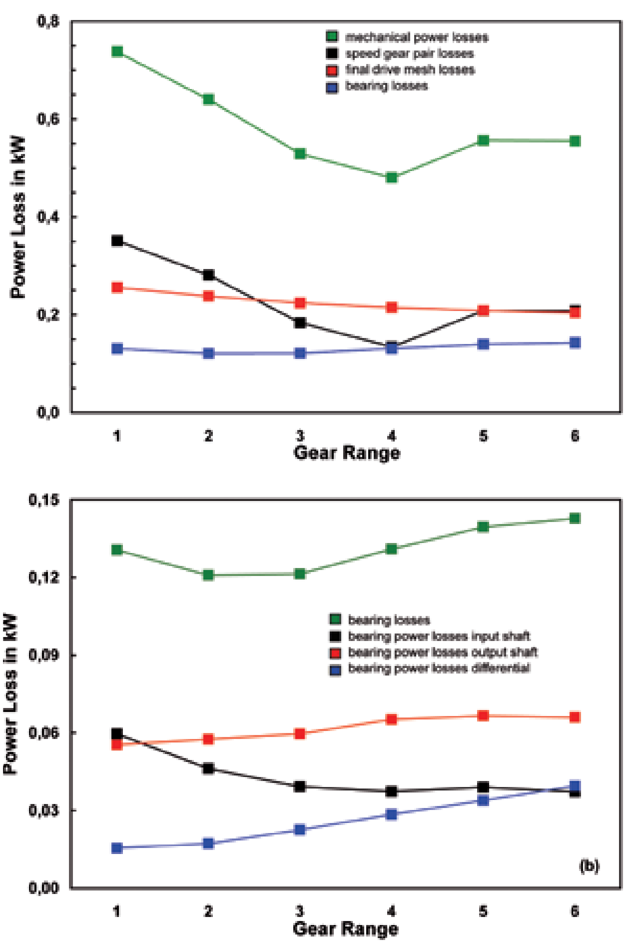

Figure 6: Components of mechanical power losses at T = 100 Nm, θ = 80° C, Ω = 4000 rpm and S = 0.56 µm; (a) components of P_m, and (b) components of P_b

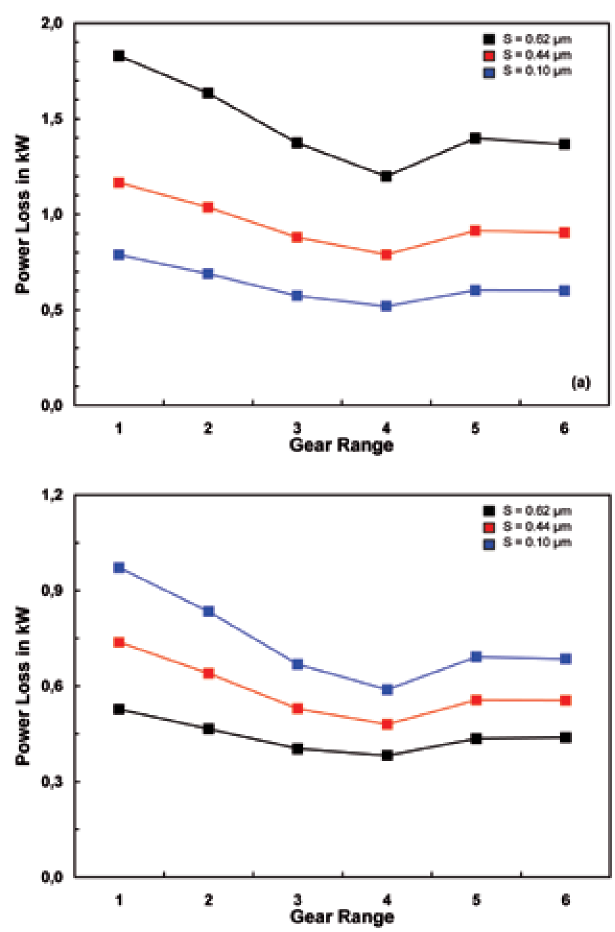

Figure 7: Influence of (a) surface roughness S and (b) temperature θ on P_m at Ω = 4000 rpm

served here at elevated operating transmission temperature values. For instance, at the first gear range, the reduction of the power losses P_m is about 65 % warmth from θ = 25° to 100° C. This trend is preserved at other gear ranges, indicating that the same transmission exhibits three times less power losses P_m at 100° C as at 25° C. This is in direct relation to the change in viscosity of the lubricant as a function of temperature.

The variation of P_m and η_m within the input speed and torque values as defined by the transmission duty cycle conditions are illustrated in **Figure 8** at θ = 80° C and S = 0.56 µm. Figure 8a provides six contour plots (one for each gear range) for the variation of P_m with speed and torque, while Figure 8b presents the η_m values for the same case in the same format. In this format, these simulation results can be used to determine P_m values at each step (defined by T_j, ω_j and g_j in Figure 2) of a user-defined duty cycle. It is also worth mentioning here that all three parameters influence P_m and η_m significantly. This is evident from the predicted ranges of P_m (from 0.01 to 0.77 kW) and η_m (97.46 to 99 %) obtained by changing speed within 1000 to 5000 rpm and torque within 10 to 100 Nm at all of the speed ranges 1 to 6.

Consistently, the degree of efficiency increases slightly towards the 100 % margin when increasing the input torque within the nominal torque range of the transmission.

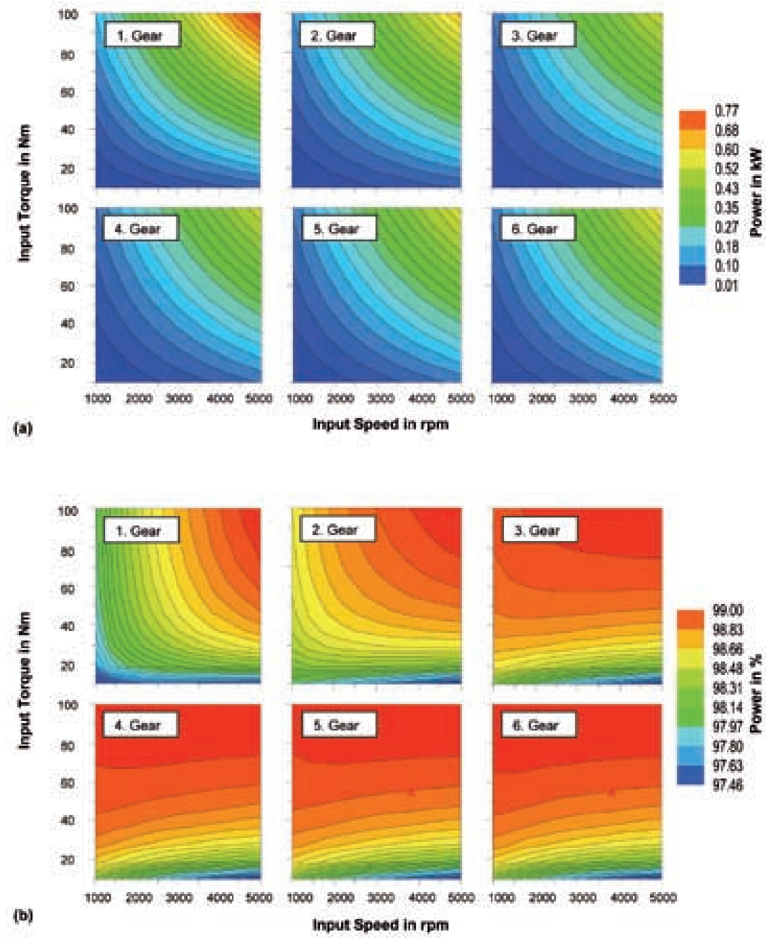

Figure 8: Variation of P_m (a) in kW and η_m (b) in % with duty cycle conditions at θ = 80° C

4 Conclusions

In this study by GM Powertrain Europe and Ohio State University, Columbus (USA), a model to predict the mechanical power loss of all the gear pairs in a manual transmission is proposed. The transmission power loss model integrates a gear mesh loss model and a bearing power loss model to calculate the load-dependent friction losses of all the gear pairs in the transmission. The transmission power loss model is validated through comparisons to measured power loss data from the F40 six-speed transmission. Limited parametric studies are presented to demonstrate the influence of the gear tooth surface roughness values and the lubricant properties on mechanical transmission power loss. A set of contour plots are also presented to show the combined influence of the speed, torque and gear range on transmission power losses. Our current work focuses on expanding this transmission power loss model through novel formulations to predict spin losses.

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