General Modeling Method of Power Losses in Transmission with Parameter Identification

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To reduce the CO_2 emission of vehicles, improvement is always considered for an energy-efficient ICE (internal combustion engine) or a substitutional EM (electric motor). But the optimization of efficiency is also necessary for all other components in the powertrain, no matter which motor concept is used (Ref. 1). One of the most relevant loss sources of the powertrain is the transmission. For example, if one energy unit in a manual or automatic transmission is saved, the engine input energy can be decrease by 2.5–4 energy units to achieve the same power output (Ref. 2). Thus, it is also an effective way to reduce the energy consumption through increasing the efficiency of the transmission, which means the power losses inside the transmission shall be reduced.

Although research on power losses of vehicle transmissions has been carried out for decades, the research focus is always the modelling of component power losses. On one hand, the procedure to estimate the overall power losses of the transmission is based on the sum of all the components power losses (Refs. 3–5). On the other hand, only the overall transmission efficiency can be obtained through most experiments. Therefore, this paper aims to develop a method to unify the component power loss models and the experimental results, in order to achieve a more accurate power losses prediction for the transmission. The method can be applied to different types of transmissions in a flexible way and provide a platform to compare several different products from different OEMs or suppliers in parallel.

Power Loss Mechanisms inside a Transmission

According to Niemann/Winter (Ref. 6), power losses in a transmission consist of load-dependent power losses and load-independent power losses, which originate from gears, bearings, seals and auxiliary power losses:

$$P_V = P_{VZP} + P_{VZ0} + P_{VLP} + P_{VL0} + P_{VD} + P_{VX} \tag{1}$$

In Equation 1, gear losses are divided into load dependent power losses P_{VZP} and load-independent power losses P_{VZO}. The P_{VZP} is evoked mainly by the friction-related mechanical power losses in gear meshing (Refs. 3, 7–9). And the cause of P_{VZO} is the gear pair spin that is bathed in the lubricant oil or surrounded by an oil-air-mixture, like the oil churning losses (Refs. 10–12). Same as the gear losses, the bearing losses P_{VL} are composed also of load-dependent power losses P_{VLP} and load-independent ones P_{VLO}, which is well-described (Refs. 13–14). Seal losses P_{VD} and other losses P_{VX} are not load-dependent (Refs. 15–16). For different transmission types, other losses P_{VX} can be losses of synchronizers or clutches. For the power loss calculation of the example transmission, gear meshing losses P_{VZP}, churning losses P_{VZO} and bearing losses PVL will be considered and therefore discussed in the next section.

Gear meshing losses. Power losses due to gear meshing result from the sliding and rolling of the two tooth flanks of the wheel and the pinion against each other on the path of contact (Fig. 1).

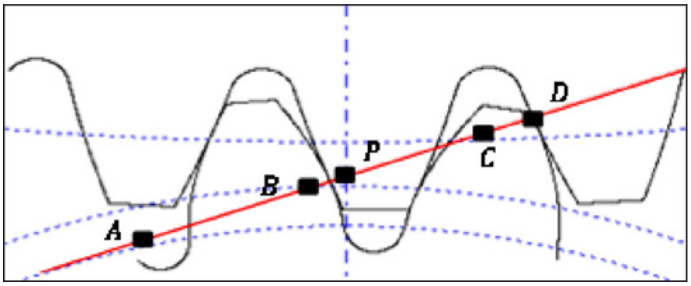

Figure 1 The path of contact between two tooth flanks of the wheel and the pinion.
A: the start point of the contact; B, C: point of the load switching; P: pitch point; D: the end point of the contact.

It is obvious that for each point x along the path of contact, the instantaneous power loss due to sliding is the product of the sliding velocity V_s, the coefficient of friction f and the tooth normal force F_n, shown in Equation 2.

$$P_S(x) = |V_S(x)| \cdot f(x) \cdot F_n(x) \tag{2}$$

Through tooth geometry and input rotational speed, the sliding velocity can be easily calculated. The normal force distribution is assumed to be ideal, which means the normal force between the tooth flanks from point B to C is constant and double the value of the one from point A to B and point C to D (Fig. 1).

Modeling of the friction coefficient is complex because it is affected by tribology factors like tooth surface structure, roughness, type and viscosity of the lubricant. Besides, it is also influenced by macro-geometric properties of the gear set. Various authors (Refs. 8, 17–20) developed models to predict the friction coefficient on the tooth flank. Among them, the model by Xu (Ref. 20) is obtained by using multiple linear regression analyses includes all the key features of a gear contact, based on the Newtonian thermal elastohydrodynamic

This paper was first presented at the International VDI Conference on Gears 2017, Garching/Munich [VDI-Berichte 2294, 2017, VDI Verlag GmbH] and is reprinted here with VDI approval.

lubrication (EHL) model.

The instantaneous sliding power loss is then available through multiplying the sliding velocity V_s, the coefficient of friction f and the tooth normal force F_n. The average sliding power loss of the gear meshing $\overline{P_S}$ is obtained by integrating P_s over the path of contact and then dividing by the length of the path of contact:

$$\overline{P_S} = \frac{1}{x_D - x_A}\left[2\int_{x_A}^{x_B} P_S(x)dx + \int_{x_B}^{x_C} P_S(x)dx + 2\int_{x_C}^{x_D} P_S(x)dx\right] \tag{3}$$

Equation 4 was initially proposed by Ohlendorf (Ref. 21) and is widely applied (Refs. 5, 22); it is also used in ISO 14179-2 (Ref. 3). The average friction coefficient is a function of load, roughness, lubricant viscosity, velocity and geometrical properties (Refs. 3, 6). The normal force distribution is assumed as described above. Together with Equations 2 and 3, the sliding power loss is deduced:

$$\overline{P_S} = H_v f_m P_{in} \tag{4}$$

Where P_{in} is the input power of the gear pairs and

$$H_v = \frac{\pi(i+1)}{z_1\, i \cos(\beta)}(1 - \varepsilon_a + \varepsilon_1^2 + \varepsilon_2^2), \tag{5}$$

With i, ratio of the wheel and the pinion; z_1, tooth number of the pinion; β, helical angle (0 for the spur gears); ε_a, profile contact ratio; ε_1, ε_2, tip contact ratio.

The instantaneous rolling frictional power loss is expressed as the product of the rolling velocity V_R and the rolling frictional force F_R, as in Equation 3.

$$P_R(x) = V_R(x) \cdot F_R(x) \tag{6}$$

However, power loss due to rolling is often neglected because the sliding power loss dominates the gear mesh losses.

Power losses due to oil churning. Modeling of power losses due to spinning gears is complicated, so most models in the literature are empirical, based on dimensional analysis. Mauz (Ref.23) investigated various influencing factors like spinning direction, oil viscosity, housing effect and oil volume experimentally, and developed an equation to predict oil churning power losses:

$$T_{pl} = 1.86 \cdot 10^{-3} \cdot \left(\frac{\eta_{oil}}{\eta_0}\right)^{-1.255} \left(\frac{R_a}{R_0}\right) \cdot C_{WZ} \cdot C_{wa} \cdot C_M \cdot C_V \cdot \eta_{oil} \cdot v_t \cdot A_B, \tag{7}$$

With R_a, gear tip radius; C_{WZ}, C_{WA}, factor of distance from wall; C_M, factor of module; C_v, factor of oil volume; v_t, peripheral speed; A_B, gear immersion area.

Although a range of influence factors are considered in the model, many of the parameters are not available for the vehicle transmission test and the validity range of the model is also limited. Another model is employed in the ISO 14179-2 (Ref. 3):

$$T_H = C_{SP} C_1 e^{C2\left(\frac{V_T}{V_{T0}}\right)} \tag{8}$$

where C_{SP}, C_1 and C_2 depend on the parameters of the immersion depth of pinion and wheel.

Besides, many models from literature take the form of the model by Boness (Ref. 24).

$$T_{pl} = \frac{1}{2}\rho(1.1047n)^2 R_p^3 S_m C_m, \tag{9}$$

With ρ, density of the lubricant; n, gear rotational speed; R_p, gear pitch radius; S_m, surface area of contact between the gear and the lubricant; C_m, dimensionless drag torque based on dimensional analysis.

Deduced from dimensional analysis, the corresponding expression of C_m depends on the flow regimes characterized by Reynolds and Froude numbers. Changenet (Ref. 25) characterizes additional regimes and broadens the range of application of dimensional analysis to helical gears. The dimensionless churning torque C_m is expressed in five groups — depending on speed factor γ and Reynolds number R_e.

Bearing losses. It is difficult to directly model the load-dependent power losses and load-independent losses of bearings because there are different bearing types that vary in terms of geometry and design parameters. Therefore, the manufacturers (INA/FAG (Ref. 13) and SKF (Ref. 26)) provide empirical models to estimate the drag torques on the bearings that are widely applied in the literature.

According to INA/FAG, the drag torque of the bearing can be calculated through the following equation. It is also included in the ISO 14179-2 (Ref. 3):

$$T_{pl} = T_{B0} + T_{B1} \tag{10}$$

where T_{B0} is the load-independent bearing power losses and T_{B1} the load-dependent bearing power losses.

The SKF model is composed of four loss contributors to calculate the drag torque of a bearing, i.e. — rolling frictional torque $T_{rr;}$ sliding frictional torque T_{sl}; drag torque of oil bath lubrication T_{drag} and frictional torque of the sealing T_{seal}.

$$T_{pl} = T_{rr} + T_{sl} + T_{drag} + T_{seat} \tag{11}$$

Simulation on a 2-Speed Transmission in an Electric Vehicle

As discussed earlier in this paper, there are different models to predict the component power losses inside a transmission, corresponding to the different loss sources in Equation 1; the combinations are therefore varied to calculate the overall power losses of the transmission. To compare the models, two sets of combinations are chosen here. One of them is to apply all of the models from ISO 14179-2. The other one is to combine the integration model for gear meshing losses, the model of Changenet for the oil churning losses, and the SKF model for bearing losses — referred to as the joint model.

A 2-speed, 2-stage transmission for an electric vehicle (Fig. 2) with a high-speed electric motor is selected for the case study. The ratio of the 1st gear is 21.83 and the ratio of the 2nd gear is 16.04. The center distance from the input shaft to the intermediate shaft is 69 mm, and the one from intermediate shaft to output shaft is 115 mm. Some gears are immersed in the oil, and there are only deep-groove ball bearings in the system; the experiment is conducted by an industry partner. The efficiency map of the transmission for the lower ratio was acquired at a stationary temperature of 100ºC and with a constant oil level on the test bench. There was no speed difference between the two out-put shafts, therefore no differential losses needed to be considered. The input torque varied from 3.6 Nm to 20 Nm, and the input speed varied from 1,000 rpm to 22,000 rpm.

The results of the two calculation methods are illustrated in Figure 3. In general, the joint model predicts a higher efficiency than the ISO 14179-2 methods. The largest difference is found in the low-input torque area. The efficiency calculated by the joint model (Fig. 3b) is 4.47% higher than predicted by ISO 14179-2 (Fig. 3a). Figure 4 shows a side-by-side comparison of the delta efficiency maps (measurement vs. calculation) for each method, and the results of the two methods vary considerably. A larger difference between the prediction by ISO 14179-2 and the measurement (Fig. 4a) can be clearly observed, which ranges from 0.2% to 12.2%. In contrast, the difference between the prediction by the joint model and the measurement ranges from –2.1% to 4.0%. From the comparison, it can be concluded that the joint model provides a better overall efficiency prediction for the example transmission.

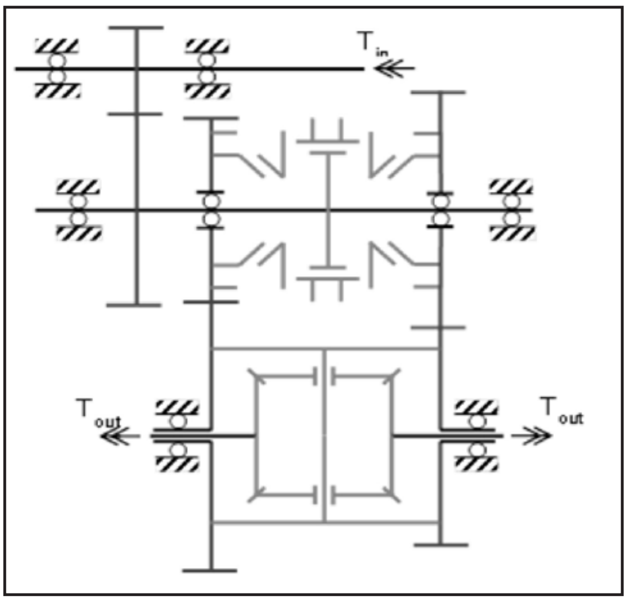

Figure 2 The structure of a 2-speed transmission in an electric vehicle.

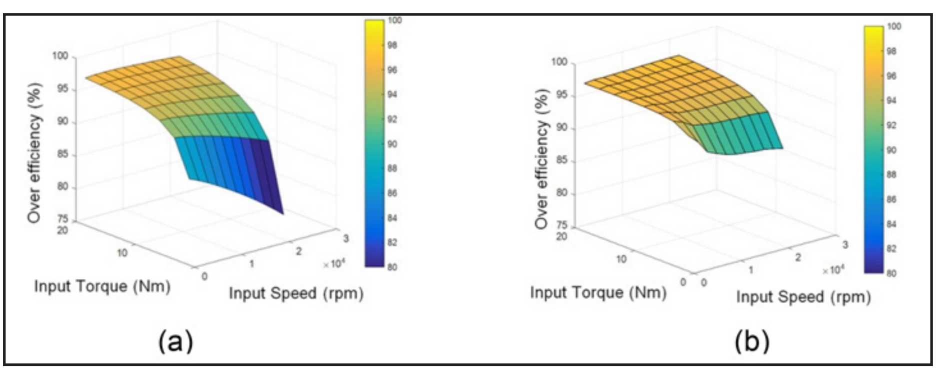

Figure 3 Comparison of the calculation results of (a) ISO 14179-2 and (b) the joint model.

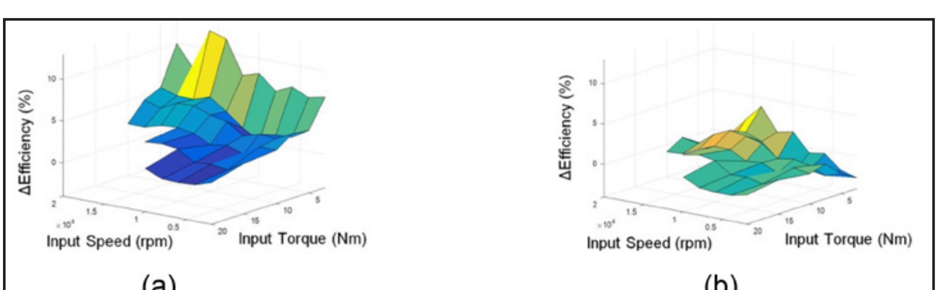

Figure 4 Comparison to the measurement with the calculation by (a) ISO 14179-2 and (b) the joint model.

Parameter Identification Applied to the Joint Model

Even though the joint model shows a better agreement for the overall efficiency of the 2-speed transmission, there is still potential to improve the quality of the prediction. Parameter sensitivity studies were carried out for each loss type in the joint model. Based on the results of the parameter studies, several parameters from the models are selected, such as immersion depth, oil viscosity, gear surface roughness, correction factor for axial/radial forces on bearings and so on, and formed as the parameter vector. According to the models, the reasons that these parameters shall be identified are apparent. For example, the oil viscosity parameter plays a certain role in every model. The axial/radial forces on the bearing are deduced from the static analysis on the shaft at each input torque, which shall be corrected due to the deformation of the shaft at different torque input.

The parameter identification is applied to the joint model, with the help of the experimental data. The efficiency differences $\Delta\eta$ between measurement and calculation result by the joint model at corresponding transmission input torque and input speed are set as the criteria for optimization:

$$\Delta\eta = \left|1-\hat{\eta}(T_{in}, n_{in}) - \frac{P_V(T_{in}, n_{in})}{P_{in}(T_{in}, n_{in})}\right| \qquad (12)$$

In Equation 12, $\hat{\eta}$ denotes the experiment data for the overall efficiency of the transmission at specific input torque T_{in} and input speed n_{in}. The trust-region method (Ref. 27) is employed to search the best-fitting parameter vector. The boundary of the parameter vector is given according to different parameters' physical meaning and limitation. At end, the best fit parameter vector is identified (Fig. 5).

The identified parameters are put back in the joint model. In Figure 6, contour maps are used to illustrate how the parameter identification assists to minimize $\Delta\eta$. In the high-input torque area, as well as the high-input speed area, the difference between measurement and calculation is clearly decreased with the best-fit parameter set. The comparison in Figure 7 shows a power loss stack up comparison for a transmission input torque of 10 Nm. The overall power loss of the best-fit parameter set shows good agreement with the measurement. The change in the gear churning power loss appears to be the dominant factor in this case.

The best-fit parameter set provided a better agreement between the measurement and the joint model than the initial parameter set. In this way, it was also possible to analyze which components inside the transmission contribute to which extent. Thus the target to compare and assess different transmission products in detail is realized. In addition, it is also possible to search for the optimization potential inside the transmission assisted by the identified parameters.

Conclusion and Perspective

In this paper, the models for different power loss sources in transmissions from literature are applied and compared, which form the two methods — ISO 14179-2 and the joint model to estimate the power losses in a transmission. A 2-speed transmission in an electric vehicle is used as a case study to compare the two methods and validate them with experimental data. Parameter identification is carried out on the selected model, and the overall efficiency of the transmission calculated by the joint model agrees better with the measurement. Using this process, a better understanding of the breakdown of losses inside a transmission is gained; it is then possible to further improve the overall transmission efficiency. In the future, more models to predict other component power losses in the transmission will be investigated and added in the joint model. More transmission types are expected to be studied and compared. **PTE**

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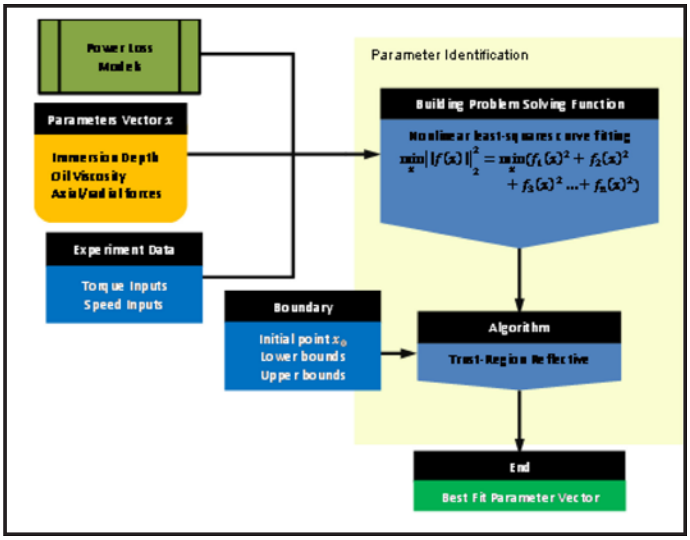

Figure 5 The process of parameter identification for the joint model.

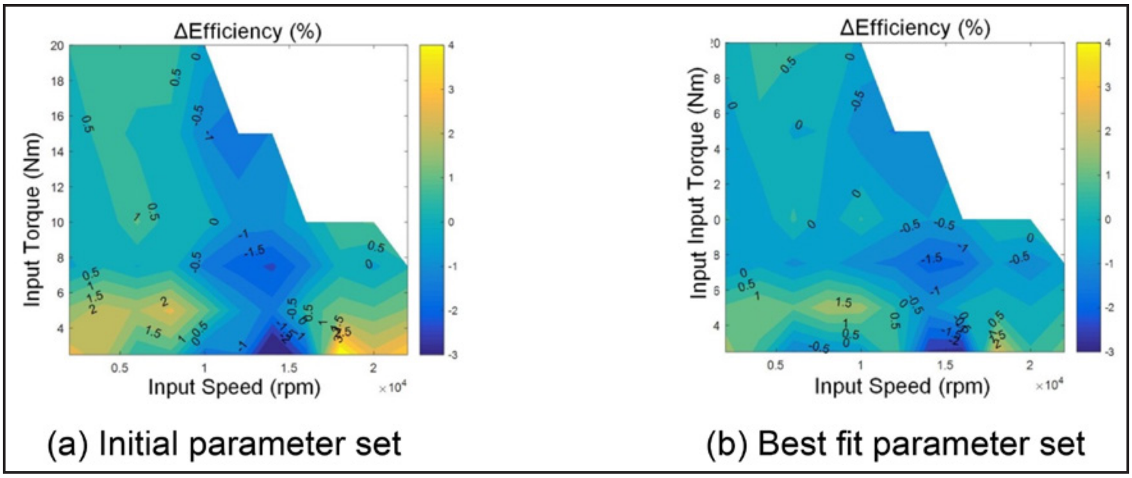

Figure 6 Comparison of $\Delta\eta$ before and after parameter identification.

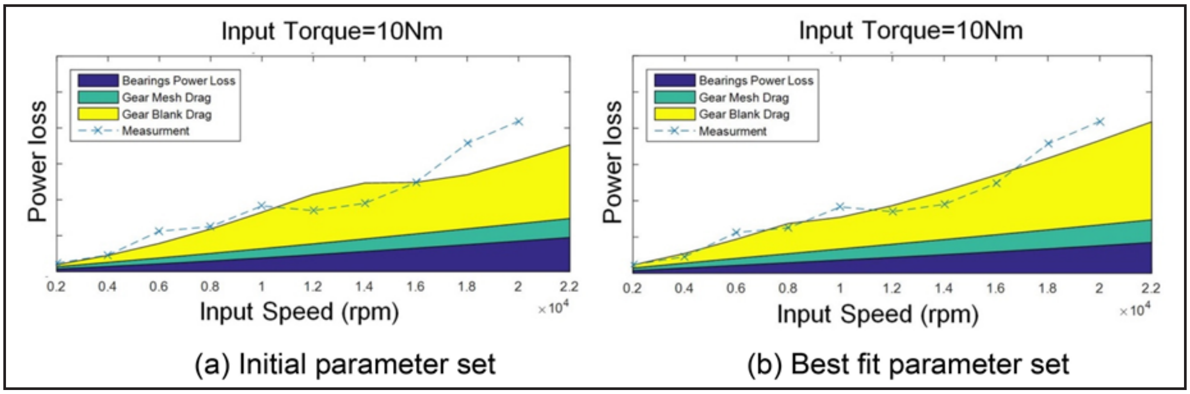

Figure 7 Segment power losses before and after parameter identification.

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