

Characterizing the Onset of Manual Transmission Gear Rattle

Part II: Analytical Results

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ABSTRACT

Lumped parameter analysis is a simple and commonly used technique for performing torsional analysis or design parameter sensitivity studies on automotive powertrains and drivelines. The purpose of this paper is to demonstrate the application of lumped parameter analysis to manual transmission gear rattle. A representative model is developed for a FWD manual transmission, as operated in a dynamometer test cell. Once validated by experimental data, the model is used to investigate the influence on gear rattle of parameters not readily modified or controlled during hardware evaluations. A sinusoidal torque is used to excite the system, and a signal processing technique similar to that derived in Part I of this two part paper is used to identify the inception of gear rattle. Functional relations for torque losses associated with shafts, gears, seals, lubricating oil flow and bearing clearances as a function of basic design parameters are included within the model. Correlation between the lumped parameter model and experimental results contained in Part I are presented. The effects of modifying basic design and operating parameters associated with various subsystems of the manual transmission are reported.

INTRODUCTION

Meeting the ever-increasing demand for fuel efficiency drives demand for continued reduction in parasitic losses in manual transmissions, as well as operating engines at lower speeds with higher throttle openings. Therefore, inherent conflicts result between fuel efficiency and gear rattle prevention. Drag torque (a parasitic loss) on the loose gears is typically relied on to keep them in contact with the oscillating gears in the main powerflow. Low speed, open throttle engine operation subjects the driveline to low-frequency, high amplitude torque pulses, resulting in large torsional oscillations of the gears in the powerflow. Both of these factors make it more important to understand the rattle sensitivity of the transmission.

Accurate prediction of gear rattle can help ensure the transmission contains the necessary hardware to minimize the risk of rattle. From a modeling perspective, a number of factors must be considered in developing a representative model of the transmission system and governing boundary conditions.

The first consideration is the modeling technique to be used to represent the system. A manual transmission can be well-represented as a lumped parameter system: discrete elements of stiffness and inertia. Kinematics of the system are maintained by including the appropriate gear ratios and connecting the discrete elements

consistent with the powerflow of the transmission. This simplified modeling philosophy has previously been utilized in [1-6] in modeling geared transmission systems for assessing loose component rattle and parameter sensitivity. Some researchers prefer to use multi-body simulation (MBS), see [7-11], to examine loose gear rattle. In MBS, the motion of each body is described by its kinematic behavior. Conservation of force and momentum are used to compute the resulting dynamics of each body. Typically, this approach is computationally demanding and requires a high degree of detail in model assembly. Both modeling techniques have demonstrated to be equally valid through correlation with experimental findings, see [3 and 5].

The dynamics occurring at the point of contact between the two gears is a second consideration when modeling gear rattle. The MBS modeling philosophy usually requires more rigorous attention to the equations governing motion, compared to the lumped parameter approach. The nonlinearity of motion across the lash, formulation of accurate contact stiffness and loss parameters, and their influence on gear rattle has been the topic of much research; see [6-9 and 11].

Drag torque resulting from the motion of unloaded components is a critical boundary condition to take into account when predicting the occurrence of gear rattle. Analytical work in [3] and [8] included only the viscous drag occurring due to the relative speed between loose gear and shaft, ignoring the interaction of loose gears with the oil sump. Delprete and Rosso [10] included three sources of drag torque within an MBS model: 1) viscous drag at the loose gear – shaft hydrodynamic bearing, 2) windage loss, and 3) oil sump churning loss. Investigations to predict manual gearbox efficiency [12-14] have included or developed detailed formulations for loaded and unloaded component power loss. The unloaded component drag torque formulations developed are directly applicable to estimating drag torque acting on a loose gear for the prediction of rattle.

This paper presents a lumped parameter model capable of predicting the onset of loose gear rattle. The manual transmission model consists of an arrangement of discrete inertias and stiffnesses, each coupled through the appropriate gear ratio and backlash, and with various drag torque components applied to the loose gears. The transmission, which was discussed in Part I, contains three shafts (input, output and reverse idler) and a differential/final drive unit. Input and output shafts are mounted on tapered roller element bearings. The reverse idler shaft is located by two journal bearings on each end of its shaft. The following sources of torque dissipation are included in the model: oil sump churning, roller element bearing drag, hydrodynamic bearing drag and synchronizer drag acting on the free pinions. The contribution of load dependent friction at mating gear teeth is neglected. The resulting transition to rattle for loose gears is shown to be in good agreement with

experimental data and observations discussed in Part I, highlighting the importance of accounting for all drag torque components on individual loose gears.

LUMPED PARAMETER MODEL

The system under consideration is shown by the kinematic diagram in Figure 1, a transverse manual transmission with five forward gear ratios, of two axis design, with a maximum input torque rating of 353 N-m. First and second gears utilize triple cone synchronizers, which are located on the output shaft, and supported by hydrodynamic journal bearings. To minimize noise during no-load operation, third through fifth gears feature single cone synchronizers with leaf springs between the baulk ring and hub. To minimize drag, these higher gears are located on the input shaft and use needle bearings and splash lubrication. Reverse is achieved with an idler shaft and a gear reduction of two stages. Stage one is driven by first gear and is splined to the reverse idler shaft. Stage two is a synchronized spur gear, piloted by a journal bearing on the idler shaft driving the first/second gear synchromesh sleeve on the output shaft.

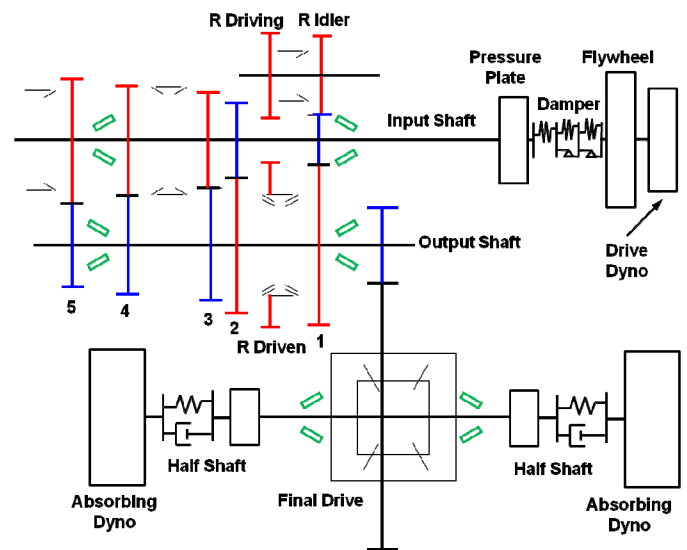


Figure 1. Kinematic diagram for transverse, two axis transmission, starting device and test cell, driving gears are blue, loose gears are red and roller bearings are green

Included in the lumped parameter model are torsional properties of the dynamometer system used to test the transmission as described in Part I. Dynamometer and shafting inertias are lumped together, since the shafting is assumed rigid in comparison to the transmission shafts and half shafts. The torsional characteristics of the flywheel assembly (flywheel, friction disk, damper assembly and pressure plate) are also represented in the model. The nominal hysteresis friction torque for the first and second stages of the damper assembly are 1.25 and 19.6 N-m, respectively.

dynamometer test cell inertia and stiffness properties are detailed in Table 1.

Table 1. Lumped model torsional properties

Model Element	Inertia kg-m ²	Stiffness Nm/deg
Drive dyno	0.12	-
Flywheel	0.1162	-
Friction disk & damper assembly	0.0049	$K_1=0.29$, $K_2=16.7$, $K_3=38.1$
Pressure plate	0.0357	-
Output intermediate shaft	0.0007	639
Output half shafts	0.0072	461
Absorbing dyno	0.57	-

Each gear pair of the transmission is modeled as an input gear inertia driving a loose gear inertia through a backlash and contact stiffness. This is shown in Figure 2, where the driving gear inertia is firmly coupled to the dynamics of the rest of the gears in the power flow. Thus, it is connected to a set of parallel axis gears, and modeled with the number of teeth to preserve the kinematics. A torque reaction to the transmission case from the parallel axis gear pair is also provided. The driven gear connects to a rotary lash element, containing the backlash of the gear set and conditions for the contact dynamics. For this investigation the contact dynamics were assigned constant values to simplify the analysis and decrease simulation time. The best estimate of contact stiffness and damping properties were determined by comparison with experimental data. The values for contact stiffness and damping properties used are 1000 Nm/deg and 0.5 Nm/deg-s, respectively. The loose gear inertia is acted on through the contact model (in the forward direction), and drag torque element (rearward). When active in torque transfer, the drag torque element is replaced with a rigid spring element that couples the loose gear to the shaft.

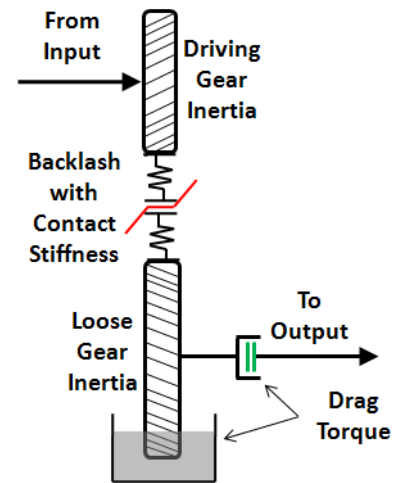


Figure 2. Lumped parameter model of a single gear pair with backlash and torque loss element

Angular backlash data relative to loose gear pitch radii are provided in Table 2 along with basic gear geometry.

Table 2. Loose gear backlash and basic geometry

Gear	Backlash (°)	Teeth Driven (loose)	Pressure Angle (°)	Radius (m)
1st	0.074	44	20.0	0.059
2nd	0.060	44	17.0	0.049
3rd	0.105	46	17.0	0.035
4 th	0.121	42	16.5	0.040
5 th	0.046	38	17.0	0.045
Rev 1	0.161	21	20.0	0.031
Rev 2	0.143	49	20.0	0.061

The lumped parameter model is torsionally excited using a simple harmonic torque input to the dynamometer drive inertia. The excitation source was chosen to be identical to that used in the experimental investigation, described in Part I, to correlate with those results. Figure 3 shows the torque based torsional sweep used for loaded gear conditions. The torque amplitude was stepped in 10 N-m increments from 0 to 200 N-m (0-pk) at 0.5 second intervals in order for the response at each loose gear to stabilize. For the neutral gear state, the excitation source is prescribed by a mean input speed and a dynamic speed component with amplitude swept from 0 rpm to 40 rpm (0-pk) at 2 rpm increments every 0.5 seconds (see Figure 4).

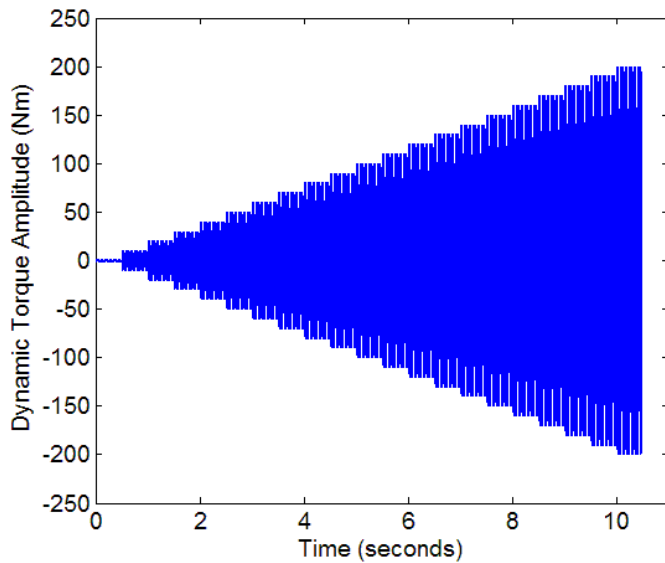


Figure 3. Dynamics component of torque based torsional excitation signal used for loaded gear simulations

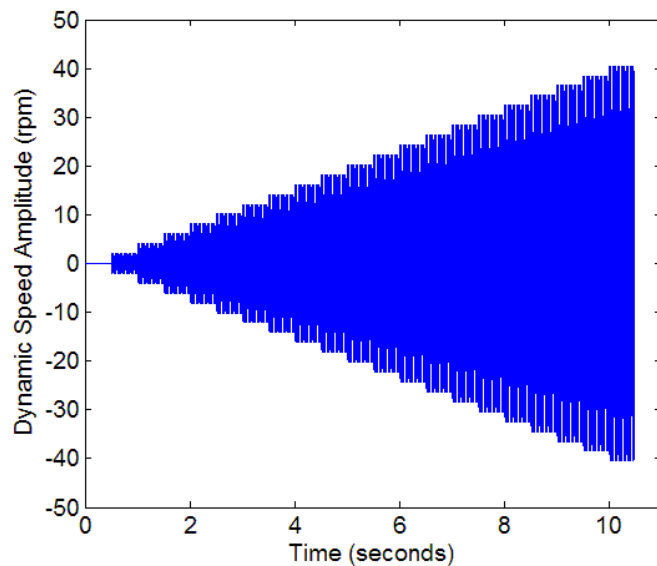


Figure 4. Dynamic component of speed based torsional excitation signal used for neutral gear simulations

The temperature-dependent physical properties of the lubricant used during the experimental testing (see Part I) and for the analysis in Part II are provided in Table 3 at three mean sump temperatures.

Table 3. Physical properties of oil lubricant

Property	Sump Temperature		
	0 °C	60 °C	100 °C
Density (kg/m ³)	876	838	813
Absolute Viscosity (N-s/m ²)	0.1040	0.0172	0.0052
Kinematic Viscosity (m ² /s)	1.19e-4	2.05e-5	6.37e-6

DRAG TORQUE PREDICTION

In geared transmission systems, drag torques (loss torques) act on each rotating element, whether active or inactive in torque transfer. Losses from active elements include sliding and rolling friction at mating gear teeth, bearing drag, oil churning, and oil aeration. For inactive rotating elements, torque loss is mainly generated through oil churning, oil aeration, viscous drag in its bearings (journal or roller element), and synchronizers. Total drag torque, T_{drag} , for a particular loose gear is the sum of roller bearing drag, T_{rb} , oil shearing in journal bearings and synchronizers, T_{sh} , and oil churning, T_{ch} :

$$T_{drag} = T_{rb} + T_{sh} + T_{ch} \quad (1)$$

Gear windage losses are ignored, since gear speeds are relatively low and loose gears on the input shaft are splash lubricated. For the loose gears of the modeled transmission, drag torques were computed with the following formulas and implemented as fixed values for a specified operating condition.

BEARING LOSSES

The friction torque generated by viscous drag in unloaded roller element bearings can be estimated using the following equations developed in [15],

$$T_{rb} = 10^3 f_0 (\nu N)^{2/3} d_m^3 \quad \nu N > 2e^{-3} \text{ m}^2/\text{s min} \quad (2a)$$

$$T_{rb} = 16 f_0 d_m^3 \quad \nu N < 2e^{-3} \text{ m}^2/\text{s min} \quad (2b)$$

where N is shaft speed in rpm, d_m is mean diameter, and f_0 is a lubrication factor. A value of 1.5 was assumed for this investigation as the test transmission considered in Part I contained 90W gear oil. For lighter weight oils a value of 1 can be used. Equations 2a and 2b are used to compute drag torques for 3rd, 4th and 5th loose gears of the modeled transmission.

Oil shearing occurs at the interface between a loose gear, its bounding shaft and synchronizer rings, due to a speed differential, ΔN . Assuming laminar Couette flow, Petroff's bearing equation is applied to estimate drag torque between the loose gear and locating shaft and loose gear and synchronizer:

$$T_{sh} = \frac{4\pi^2 \mu L R^3 \Delta N}{30j} \quad (3)$$

where μ is the absolute viscosity, L is the gear length, R is the pitch radius of the gear and j is the radial clearance of the bearing. Equation 3 is used to estimate oil shearing drag torque on 1st and 2nd loose gear hydrodynamic bearings and for all synchronizers acting on loose gears.