

Design and Analysis of a Gear Rattle Test Rig

A. R. Crowther and M. K. Rozyn
Romax Technology Limited

Copyright © 2009 SAE International

ABSTRACT

Gear rattle in manual transmission systems is a complex phenomenon and problematic to transmission and vehicle manufacturers. In the last three decades some inroads have been made in applying analytical and numerical analysis to understand rattle, however solutions and design methodologies for low rattle or no rattle are elusive. Problems with rattling transmissions continue to plague the industry. Controlled experiments for transmission rattle are required for research, to aid in the development of design tools and for validation of these tools. This paper presents a test rig which utilises a Hooke's joint to induce the torsional oscillations that lead to rattle. The design process relies on torsional lumped mass modeling and linear and transient nonlinear analyses and this paper details how the correct specification of the required driving motor isolation system is achieved. The rig complements the ongoing development of advanced rattle simulation tools and design methodologies and is intended for validation and research work.

INTRODUCTION

Gear rattle is the audible noise caused by repeated loss of contact of gear teeth. It is often associated with automotive transmissions, but may also occur in other geared systems such as pumps or power-take-off units. Gear rattle is typically excited by a time-varying input, although it may also, in rare cases, be caused by excitation from gear teeth themselves [1]. Neutral gear rattle occurs when an externally applied pulsating torque, with a small or zero mean value, is applied to the gears causing the gear mesh vibration amplitude to exceed their mean twist due to drag torque [2]. This vibration level is often referred to as the rattle threshold. The idling power plant provides the torsional fluctuations

from engine firing and dynamic characteristics of the crankshaft. Then the flywheel and clutch springs provide a transmission path which modifies the vibration before the geartrain. The gear impacts result in impulsive responses which are broadband in the frequency spectrum.

This paper details the design and analysis for a test rig intended for the study of neutral gear rattle under controlled laboratory conditions. The test rig utilises a telescopic Hooke's joint with gimbals phased so as to deliberately induce torsional oscillations at the second order of input speed. At this order lies the major component of (crankshaft) torsional vibration when a vehicle is fitted with a four cylinder engine. The vibration amplitude for a particular test may be controlled by adjusting the angle across the Hooke's joint. An essential design requirement is that the induced torsional vibration is pronounced at the transmission and suppressed at the electric drive and the large inertia of the transmission flywheel significantly complicates this task. The analysis relies on a torsional lumped mass model with electric drive, isolation system, Hooke's joint, flywheel/clutch and transmission. Linear analysis tools are first used to understand the natural frequencies and modes of the system and for isolation design. Then transient non-linear time domain analyses are performed for drive speed run-ups and used to check the maximum loads when passing through critical speeds, to predict the vibratory torque seen at the transmission and to confirm low accelerations at the drive motor. By this analysis the key design decisions were able to be made, e.g. the sizing of shafts, bearings and motor, and importantly, the correct design of an isolation system was ascertained. To highlight the latter point, the paper provides analysis results with no isolation system, a simple isolation system and finally a dual mass system, which is implemented in the final design.

The testing project aims to validate a numerical code used to predict gear rattle [3]. The code is used by transmission designers for rattle analysis prior to construction of prototypes, which allows for reduced lead times and a superior end product. The rig will also be used for research and validation of models for drag torque, clutch characteristics and impact damping. The gear rattle code is part of an advanced gearbox design software (RomaxDesigner), where specific models for gear rattle are exported from the main package for analysis. Similarly, the software has gear whine and other noise, vibration, harshness (NVH) analysis capabilities. This allows rattle and whine studies and optimisation to be performed in parallel using one model. The software includes calculation of the drag torque from bearings, gear meshes, churning and windage, for which various industry accepted models, including those from ISO standards [4] may be applied.

GEAR RATTLE TEST RIG

The purpose of the rattle test rig is to provide controlled conditions where the amplitude of input torsional vibrations may be increased so as to excite neutral gear rattle in a manual transmission (the test article). The rig is designed to provide sufficient excitation across typical engine idle speeds with some extra range (500-1500rpm). The design maximum for transmission input shaft peak to peak vibration is 2Nm. Figure 1 provides a schematic of the rattle rig concept. The electric motor drives the transmission at a steady mean speed via a double telescoping Hooke's joint (universal joint), which is installed with the gimbals deliberately phased so as to induce the torsional vibration. By changing the angle of the joints (shown as α in Figure 1) the amplitude of the vibration may be adjusted. This is practically achieved by sliding the motor in the lateral direction on the base plate. The telescopic function considerably simplifies the mounting as the longitudinal dimension remains fixed.

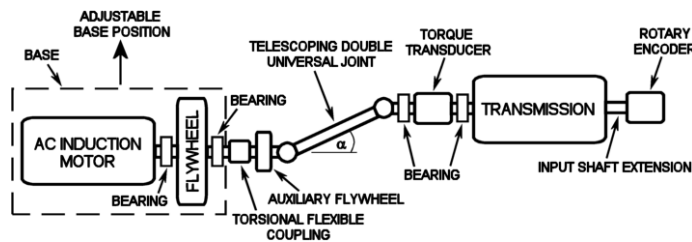


Figure 1: Schematic of Gear Rattle Test Rig.

The transmission flywheel (and clutch housing) rotates at the mean speed set by the motor with superimposed vibrations principally at the second order of rotational speed. The flywheel significantly reduces the vibration amplitude before it reaches the transmission input shaft and thus the induced vibration needs to be large, as it is in the case of the combusting engine. The inertia and stiffness distribution of the system needs to be carefully considered so that the Hooke's joint vibrates the transmission, not the electric drive. The engineering analysis leads to a design utilising a similar concept to

the dual mass flywheel used in vehicle transmissions. A main flywheel, flexible coupling and auxiliary flywheel (see Figure 1) decouple the motor from second order vibrations across the operating speed range.

Instrumentation is fitted for the following measurements: flywheel input torque and velocity, transmission input shaft torque and velocity and housing accelerations. A lubrication circuit is designed for heating to a specified set point and includes a level gauge. This allows the control of oil temperature and level in the transmission; both are critical parameters for rattle testing.

TORSIONAL MODEL

For the analysis the test rig is simplified to a lumped mass torsional model as illustrated in Figure 2 with system parameters of Table 1.

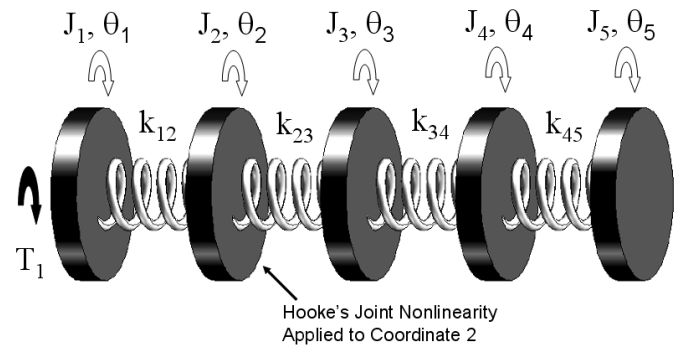


Figure 2: Rattle Rig Torsional Model. Note Damping Elements (not shown) are included in Parallel with the Stiffness Elements.

The five degrees of freedom are sufficient to represent the torsional modes of interest and have the following generalised coordinate vector:

$$\theta = \{\theta_1 \quad \theta_2^* \quad \theta_3 \quad \theta_4 \quad \theta_5\}^T \quad (1)$$

The system of equations of motion is:

$$\mathbf{J}\ddot{\theta} + \mathbf{C}\dot{\theta} + \mathbf{K}\theta = \mathbf{T} \quad (2)$$

Where $\mathbf{J}$, $\mathbf{C}$ and $\mathbf{K}$ are the inertia, damping and stiffness matrices, and $\mathbf{T}$ is the vector of input torque:

$$\mathbf{J} = \text{diag}[\mathbf{J}_1 \quad \mathbf{J}_2 \quad \mathbf{J}_3 \quad \mathbf{J}_4 \quad \mathbf{J}_5] \quad (3)$$

$$\mathbf{C} = \begin{bmatrix} c_{12} & -c_{12} & 0 & 0 & 0 \\ -c_{12} & c_{12} + c_{23} & -c_{23} & 0 & 0 \\ 0 & -c_{23} & c_{23} + c_{34} & -c_{34} & 0 \\ 0 & 0 & -c_{34} & c_{34} + c_{45} & -c_{45} \\ 0 & 0 & 0 & -c_{45} & c_{45} \end{bmatrix} \quad (4)$$

$$\mathbf{K} = \begin{bmatrix} k_{12} & -k_{12} & 0 & 0 & 0 \\ -k_{12} & k_{12} + k_{23} & -k_{23} & 0 & 0 \\ 0 & -k_{23} & k_{23} + k_{34} & -k_{34} & 0 \\ 0 & 0 & -k_{34} & k_{34} + k_{45} & -k_{45} \\ 0 & 0 & 0 & -k_{45} & k_{45} \end{bmatrix} \quad (5)$$

$$\mathbf{T} = \{T_1 \quad 0 \quad 0 \quad 0 \quad 0\}^T \quad (6)$$

Symbol	Parameter
J_1	Inertia of electric motor and main flywheel
J_2	Inertia of joint and auxiliary flywheel
J_3	Inertia of gearbox transmission flywheel
J_4	Inertia of gearbox clutch hub
J_5	Inertia of transmission gears
c_{12}	Damping between motor and joint
c_{23}	Damping between joint and transmission flywheel
c_{34}	Damping of clutch spring
c_{45}	Damping from clutch hub to 1st gear
k_{12}	Torsional stiffness between motor and joint
k_{23}	Torsional stiffness between joint and transmission
k_{34}	Torsional stiffness of clutch damper spring
k_{45}	Torsional stiffness from clutch hub to 1st gear
θ_1	Coordinate of electric motor and main flywheel
θ_2	Coordinate of auxiliary flywheel and joint input
θ_2^*	Coordinate of joint output
θ_3	Coordinate of transmission flywheel
θ_4	Coordinate of transmission clutch hub
θ_5	Coordinate of transmission gears
T_1	Input torque from electric motor

Table 1: Inertia, Damping, Stiffness and Generalized Coordinate Elements of the Torsional Model Figure 2.

Note that the bearing and gear drag may be neglected for the purpose of these analyses so no drag terms are present in equation (6). To describe the Hooke's joint excitation a non-linear kinematic constraint is applied between coordinates θ_2 and θ_2^* . Reference [5] provides the typical equations and the constraint is applied in the numerical simulations as follows:

$$\theta_2^* = \tan^{-1}(\tan(\theta_2) \cos(2\alpha)) \quad (7)$$

$$\dot{\theta}_2^* = \frac{\dot{\theta}_2 \cos(2\alpha)}{(1 - \sin^2(\dot{\theta}_2) \sin^2(2\alpha))} \quad (8)$$

The angle in the equation (2α) is twice the physical angle due to the joint having two phased gimbals. If the gimbals are not phased, or if their combined angle is zero, then the output motion is identical to the input (less

any deflection due to stiffness of joint). The vibration amplitude increases with joint angle and velocity and the fundamental excitation frequency is twice that of the input (second order). Generally, for proper machine function the natural frequencies of the rig should not coincide with these vibrations. Yet if the test article so happened to have natural modes within the operating range, then the gear rattle would be expected to be severe and coupling the test article with the driving machine should not interfere with these dynamics. This is an important consideration in the design of dynamic tests in general; the test device should not modify the dynamic characteristics of the test article (where possible). For this machine the goal is to design the isolating system so as to decouple the vibration modes of the driving system (upstream of transmission flywheel) from the transmission (transmission flywheel, clutch and gears). Note the rig is designed to run from 500 to 1500 rpm (16.7 to 50 Hz at the second order).

ANALYSIS

This section details some of the analysis undertaken in the design of the rig and the decoupling effects of the dual flywheel and flexible coupling system. This is achieved through an analytical study of the mode shapes for three cases with the inertia and stiffness parameters as per Table 2. The three cases are as follows:

Case 1: No primary or auxiliary flywheel, no flexible coupling.

Case 2: Primary flywheel and flexible coupling.

Case 3: Primary and auxiliary flywheel and flexible coupling.

Thus in these cases the parameters for J_1 , J_2 and k_{12} vary (refer Figure 2 and Table 1). The other system parameters remain fixed. Damping parameters are assigned to give reasonable damping ratios and where time histories are shown the examples have assume an joint angle of 10° ($2\alpha=20^\circ$).

Item	Unit	Case		
		1	2	3
J_1	kg.m ²	7.00e-3	120e-3	120e-3
J_2	kg.m ²	60.0e-6	60.0e-6	40.0e-3
J_3	kg.m ²	180e-3	180e-3	180e-3
J_4	kg.m ²	1.70e-3	1.70e-3	1.70e-3
J_5	kg.m ²	2.40e-3	2.40e-3	2.40e-3
k_{12}	Nm/rad	30000	30000	170
k_{23}	Nm/rad	30000	30000	30000
k_{34}	Nm/rad	490	490	490
k_{45}	Nm/rad	17600	17600	17600

Table 2: Inertia and Stiffness Values for Cases 1-3.