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CRITERIA FOR GEAR RATTLE NOISE SOURCE

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INTRODUCTION

Vibro-impacts or rattle, induced by backlash between meshing gears, lead to excessive vibration and noise in many geared rotating systems, and the problem is more pronounced in essentially unloaded meshes. In this paper the neutral rattle problem in a five-speed manual transmission of a front wheel drive automobile is examined. The main focus is on establishing an appropriate rattle criterion and the rattle noise index. Figure 1 shows schematically a generic reduced physical model; similar models have been used by other investigators [1-3]. The sign convention for the torsional displacements $\theta_i(t) = \theta_{mi}(t) + \theta_{pi}(t)$, i = 1-4, is shown, where $\theta_{mi}(t)$ and $\theta_{pi}(t)$ are mean and vibratory parts respectively. The input gear moment of inertia I_3 also includes the reflected moments of inertia from other gears. Nonlinear characteristics of the clutch torque, similar to the curve chosen by Ohnuma et al. [2], are chosen. The elastic compression force F_b between the conjugate gear pair of radii R_3 and R_4 is given in Figure 2(d). Here gear meshing stiffness k_g and backlash $2x_b$ are assumed to be constants; and impact damping is ignored. The relative displacement between gears is x_r. The torque dissipation terms due to drag and viscous dampers are given as: $T_{di} = -c_i\dot{\theta}_i(t) = -c_i(\dot{\theta}_{mi}+\dot{\theta}_{pi})$, i = 3,4. The torque excitation is $T_e = T_m + T_p(t)$.

RATTLE CRITERIA AND RESULTS

Criterion I: Vibro-impacts or rattle should not occur when the gears remain in contact at the driving side as shown in Figure 2(a); this implies that $x_r(t) > x_b$. Rattle between conjugate gears will obviously take place under the following conditions: (i) gears are separated and free to move within the backlash as shown in Figure 2(b), and/or (ii) the gears are in contact on the driven side as shown in Figure 2(c). The second case is not compatible with the mean torque requirements and hence separation will occur very quickly; consequently, case (ii) is an unstable case and must be avoided. Mathematically, a rattle criterion can be given in terms of $x_r(t)$ as shown in Table 1.

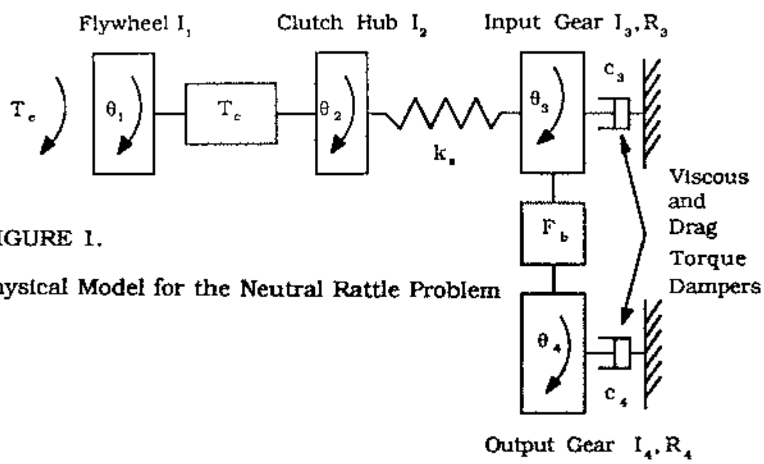

FIGURE 1.

Physical Model for the Neutral Rattle Problem

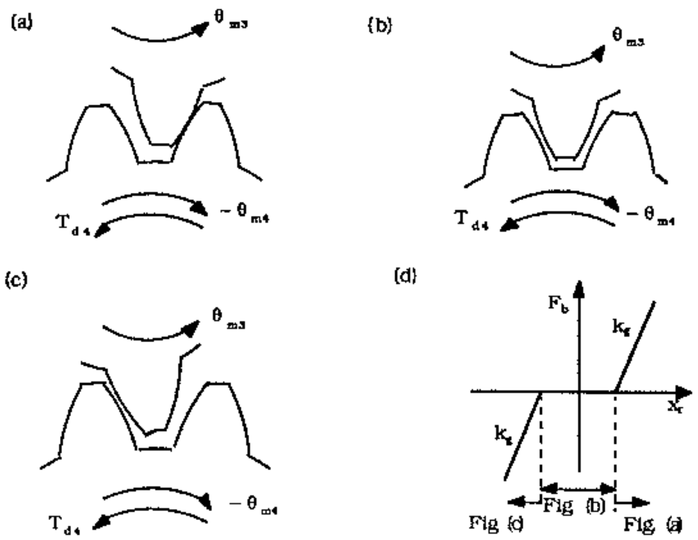

FIGURE 2. Meshing Gears and Elastic Compression Force F_b.
(a) Gears in Contact on Driving Side.
(b) Gears Separated.
(c) Gears in Contact on the Driven Side.
(d) Elastic Compression Force (F_b) with Mesh Stiffness (k_g) and Backlash $(2x_b)$. Here $x_r = R_3\theta_3 + R_4\theta_4$.

The nonlinear model of the system shown in Figure 1 is solved numerically using a fixed integration time step (Δt), fourth order Runge-Kutta technique. Since the elastic deformation between the two meashing gears $y = x_r - x_b$ is very small, double precision calculation must be performed in order to ensure accurate simulation. A step size of $\Delta t = 10^{-4}$ seconds is adequate to simulate the case without any impacts. However, extremely small integration steps, of the order 10^{-6} to 10^{-7} seconds, are required when the rattle impacts between gears take place. Still several problems were encountered in this numerical technique; this issue has not been discussed by other investigators [1-4]. Typical results given in Table 2 show a large difference in $\ddot{\theta}_{4rms}$ values for two time steps. However, $\ddot{\theta}_{3rms}$ and $\ddot{\theta}_{2rms}$ are not sensitive to the choice of Δt as long as Δt is of the order of 10^{-6}s. This illustrates that the computations of $\theta_1(t)$, $\theta_2(t)$, and θ_{3rms} are fairly accurate; conversely accurate results for $\theta_3(t)$, $\theta_4(t)$, θ_{4rms}, and $x_r(t)$ are difficult to obtain.

Criterion II: Using the equations of motion of the system shown in Figure 1 and Criterion I, we can derive the criterion based on the comparison of drag torque $T_{d4}(t)$ and inertia torque $I_4\ddot{\theta}_4(t)$ as given in Table 1. This implies that T_{d4} be sufficient to overcome the $I_4\ddot{\theta}_4$ torque to ensure rattle free transmission. This is in fact Seaman, et al's rattle threshold theory [4] which is based on a single gear pair analysis with the assumption of a constant drag torque on the driven gear. Earlier, Sakai, et al [1] had measured the increase in rattle noise ΔL_N in dB for a single input and counter gear pair. These measurements are qualitatively consistent with Criterion II.

Criterion III: Since $\ddot{\theta}_4(t)$ and $T_{d4}(t)$ are difficult to calculate when the rattle occurs due to the numerical difficulties, we propose a new rattle criterion based on $\ddot{\theta}_3(t)$. When the gears are in contact, the elastic deformation is very small and therefore we can approximate $\ddot{\theta}_4(t)$ as $-R_3\ddot{\theta}_3(t)/R_4$. This along with Criterion II and restricting the absolute motion for the rattle free case will give us Criterion III. Included in here is the fact that for a practical gear box, x_b and T_{d4} may not be constants, and, $\ddot{\theta}_3(t)$ signature may be fairly complicated. Hence, it is more convenient to calculate the root-mean-square (rms) or decibel value which is related to the energy contents of the motion. Table 2 shows that the rattle level L_β is substantially high when the rattle takes place.

Criterion IV: This is based on the clutch hub motion which can be considered as the input to the geared mesh system. The approximate calculation of $L_\beta*$, as seen from Table 2, can not only be used as a rattle index but also is useful to examine the transmissibility across the clutch.

CONCLUDING REMARKS

Based on the rattle criteria proposed here and the mathematical model of the system, one can examine the effect of the following parameters on rattle noise level: engine torque pulses, flywheel inertia, clutch characteristics, and drag torque. One can also predict how the modern automotive design changes such as the use of synthetic lubricants, lower flywheel inertia, turbocharging, etc., have affected the rattle noise problem.

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Table 1. Gear Rattle Criteria

	Criterion Basis	Rattle Condition	No Rattle Condition
I	Relative displacement between meshing gear $x_r = R_3\theta_3 + R_4\theta_4$	$x_r(t) < x_b$	$x_r(t) > x_b$
II	Output gear motion	$T_{d4}(t)-I_4\ddot{\theta}_4(t) < 0$	$T_{d4}(t)-I_4\ddot{\theta}_4(t) > 0$
III	Input gear motion	$\lvert\beta(t)\rvert > 1$ or $L_\beta > 0$, dB	$\lvert\beta(t)\rvert < 1$ or $L_\beta < 0$, dB
IV	Clutch hub motion	$L_{\beta^*} > 0$, dB	$L_{\beta^*} < 0$, dB

$$\beta(t) = I_4R_3\ddot{\theta}_3(t)/T_{d4}(t)R_4,\ \beta^* = I_4R_3\ddot{\theta}_2(t)/T_{d4}(t)R_4$$
$$L_\alpha = 20\ \mathrm{Log}_{10}(\alpha_{rms}/0.707),\ \mathrm{dB};\ \alpha = \beta, \beta^*$$

Table 2. Computer Simulation Results

Case	Δt s	$\overline{x_r} - x_b$ (mm) $\overline{x_r}$ = mean x_r	$R_4\ddot{\theta}_{4rms}/R_3$ (rad/s^2)	$\ddot{\theta}_{3rms}$ (rad/s^2)	$\ddot{\theta}_{2rms}$ (rad/s^2)	L_β (dB)	L_{β^*} (dB)
Rattle	5×10^{-7}	-0.05	573.0	172.0	180.0	16.1	16.5
	2.5×10^{-7}	-0.039	511.0	170.0	179.0	16.0	16.45
No Rattle	$< 10^{-4}$	0	8.41	8.41	8.41	-10.1	-10.1