STUDY ON THE TRESHOHLD OF GEARBOX RATTLE NOISE IN AUTOMOTIVE

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Introduction

Automobile noise has been considerably reduced as interest in environmental concerns as well as a need for more acoustic comfort has increased.
One source of noise in a car, among many, is associated with powertrain. With improvement in engine design, the perception of noise that went formerly unnoticed has become a problem, such as gearbox noise, a well-known source of irritation detrimental to vehicle comfort.

This study is dedicated to one of these noises, known as "rattle noise". It is caused by fluctuation of the engine torque which, under certain conditions, can cause multiple impacts inside the gearbox.

Problematic

Numerous authors got interested in the definition of the rattle noise threshold, i.e. the conditions of appearance of that phenomenon [1] , [2], [3], [4], but the excitation is always considered as sinusoidal. For the first time, we take into account realistic engine torque fluctuations composed of several harmonics of the engine speed, which are function of the engine and clutch design and of the working conditions.

Design of test benches

Our needs in terms of excitation led us to design and build new test benches. Indeed, we have to be able to control perfectly the engine torque fluctuations on the primary shaft, not only made of a 2nd harmonic of the engine speed, but also 4th and 6th harmonics whose relative amplitudes and phases must be adjustable at will so as to explore all the possible excitation configuration.

The rattle phenomenon is due to teeth impacts after backlash crossing, it is the relative motion between driving gear and unloaded gear that matters. So we work with a gearbox on neutral, which allow us to eliminate the continuous part of the excitation torque and to impose only an oscillating torque. For that purpose, we use a electrodynamic translation exciter, tied to the gearbox with a crank and commanded by a signal generator.

The Figure 1 shows one of our test benches ([5]).

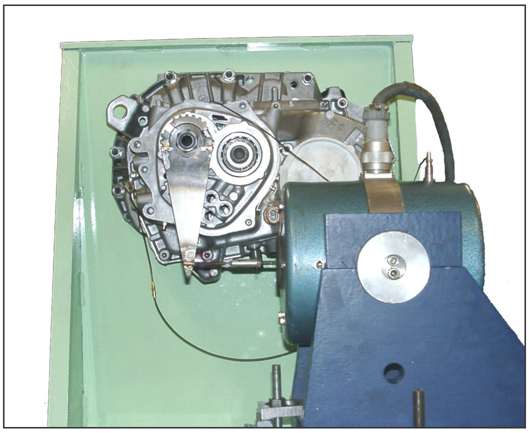

Figure 1: Photography of one of our test benches

So as to isolate the multiple teeth impacts phenomenon, our tests have been carried out on a simplified gearbox.

Definition of the rattle threshold

The first experiments showed that the threshold mainly evolves with the frequency of the imposed H_2 harmonic. In order to get more precise results about the influence of H_4 and H_6 harmonics, we first observe the evolution of the threshold with the frequency for a sinusoidal excitation, and then we bring in the other parameters for the same frequency.

"Stable" threshold and "free" threshold

We consider as threshold the acceleration amplitude of torque fluctuation from which rattle is stable. The Figure 2 gives, for a sinusoidal excitation at several frequencies, the threshold called "stable" and the threshold called "free", got without imposing teeth backlash crossing to the unloaded gear.

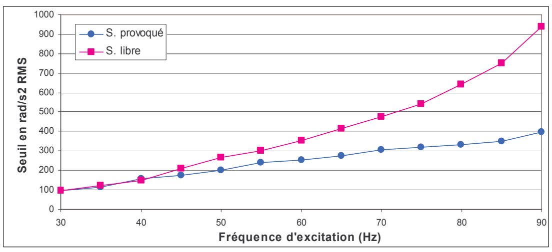

Figure 2: Comparison of the evolution of "free" and "stable" rattle thresholds with the excitation frequency.

The Figure 3 gives the acceleration (in rad/s² RMS), the speed (in rad/s, ×500) and the angular displacement (in rad, ×10000) measured at the stable threshold.

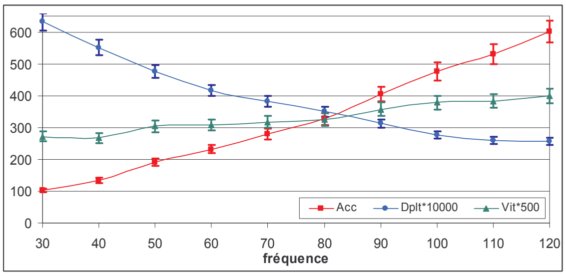

Figure 3: Evolution, in function of the frequency, of the acceleration, the speed and the angular displacement imposed to the primary shaft at the rattle threshold.

The acceleration which causes the free rattle corresponds to a constant angular displacement, equal to the teeth backlash. The acceleration needed for the stable threshold is doubled (×2,2) when the excitation frequency is double.

Threshold for a multi-harmonics excitation

We consider an angular acceleration imposed at gearbox input shaft with the following expression (1), the frequency being fixed.

$$\ddot{\theta} = \sin(2\pi f t) + H_4 \sin(4\pi f t + \varphi_4) + H_6 \sin(6\pi f t + \varphi_6) \quad \textbf{(1)}$$

The number of different excitation configurations being very important, we decided to use the design experiment method to carry out this study. Each excitation factor (H_4, H_6, φ_4 and φ_6) can have two levels. This leads us to choose a L_{16} Tagushi table, where the second order interactions can be integrated.

The chosen domain for the amplitudes of the harmonics H_4 and H_6 is between 20 and 80% of the amplitude of H_2, φ_4 changes from 0 to 45° and φ_6 from 0 to 30°. For each of the 16 excitation configurations, three measures are made : the peak-to-peak amplitude of the signal, the global RMS value and the RMS value of the second harmonic.

The Tagushi method allows us to calculate the effects and interaction of the factors. Results show that the threshold can be pushed away by choosing high values for H_4 and H_6 harmonics and a nil value for φ_4. A clear interaction exists between the parameters H_6 and φ_4 and between φ_4 and φ_6.

A variance analysis allows us to highlight the really significant parameters. Checking the mean point and the optimal point allow us to validate this model. In the end, the threshold can be given with the following expression (not centred and not reduced variables):

$$\begin{aligned} Seuil = {} & 342{,}40 + 7{,}76 H_4 + 9{,}57 H_6 - 0{,}0379 H_4 H_6 + 3{,}89 \varphi_4 \\ & - 0{,}0389 H_4 \varphi_4 + 0{,}0913 H_6 \varphi_4 + 3{,}3473 \varphi_6 + 0{,}1488 \varphi_4 \varphi_6 \end{aligned} \quad \textbf{(2)}$$

When the considered response is the amplitude of the H_2 harmonic, the variance caused by a modification of the excitation configuration is slow, regarding the natural variance of the measure. The Figure 4 gives an example of the evolution of RMS amplitudes of the harmonics H_2 and H_4 measured at the rattle threshold, in function of the proportion of H_4 added to the H_2, for a frequency of H_2 at 60Hz.

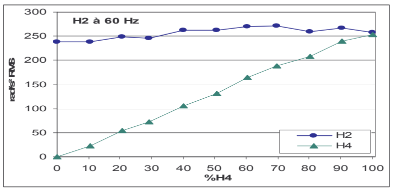

Figure 4: RMS amplitudes of the harmonics H_2 and H_4 measured at the rattle threshold in function of the proportion of H_4 introduced at 60Hz.

At a given frequency, the rattle threshold increases linearly with the introduction of H_4 harmonic.

These results confirm the important role played by the H_2 harmonic of the acyclism, since, for a given frequency, the rattle phenomenon appears for a slightly constant value of H_2, whatever the part of added H_4. The peak-to-peak value of the excitation at the threshold is then even higher than the percentage of H_4 is high.

Conclusions

In the first part, we highlighted the really important role of the excitation frequency parameter. For rattle phenomenon to appear, kinetic energy imposed to the primary shaft must be sufficient, or the angular displacement of the input shaft must be at least equal to the teeth backlash.

The introduction of upper-order harmonics in the excitation signal has but little influence on its kinetic energy. Rattle threshold is determined for an approximately RMS constant 2nd order harmonic level, but the peak-to-peak amplitude of the angular acceleration at threshold is very sensitive to the spectral composition (relative amplitude and phase of the harmonics).

Therefore, it is important to underline the major role of the amplitudes and phases of the 4th and 6th order harmonics in the sound perception of rattle noise. Indeed, the temporal shape of the acyclism imposed at gearbox input, directly linked to these parameters, has a major influence on the rattle rhythmic and so on its perception.

References

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