

The simulation analysis of gear rattle based on Adams

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Abstract. The rattle noise of automobile transmission is always a difficult problem to solve, it seriously affects the comfort and quality of vehicles. A lot of technical researches have been carried out on this academic field, mainly using dynamic software to analyze and solve the problem by changing parameters to reduce rattle force then reduce the rattle noise, which is often time-consuming. Due to the theoretical research is not perfect at present stage, there is no clear solution. In this paper, mathematical model based on dynamic system combined with Adams software is used to obtain inherent law between the rattle force and the related parameters.

1. Introduction

The rattle noise of automobile transmission is always a difficult problem to solve, it seriously affects the comfort and quality of vehicles. the theoretical research is not perfect at present stage, the premier theory^[1] only says the rattle force is associated with driving wheel speed but the inner function relation is not clear, so the analysis of rattle force and wheel speed is particularly important to reduce the rattle noise and improve the quality of products. Transmission rattle noise mainly occur when driving wheel speed is between 900-2000 r/min, In this paper, mathematical model based on dynamic system combined with Adams^[2] software and Data processing method^[3] is used to obtain inherent law between the rattle force and related parameters. It has carried on a beneficial attempt to improve the premier theory.

2. The mathematical model of the gear vibration.

neutral gear couple system in transmission is equivalent to a double mass- spring- damping vibration system^[4], the gear rotational inertia and torsional rigidity is equivalent to M and K on the gear's base circle in linear coordinates: the mass of driving gear and neutral gear is equivalent to mass (M_1 and M_2), the stiffness of input axle and output axle (spring) torsion, bending and bearing is equivalent to stiffness k_1 and k_3 , two gears meshing stiffness is k_2 , the vibration model and the related parameters of the meshing process as shown in figure 1 and table 1. Since M_2 is related to the neutral gear, the model is simplified to make $k_3=0$. The system's vibration differential equations are as follows:

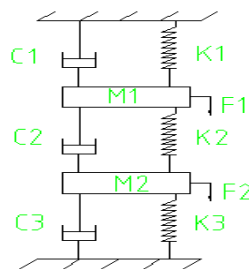


Figure 1. Vibration mode



Table 1. The gear parameters

Parameter	Value	Unit	Parameter	Value	Unit
m_n	2.5	mm	α	20	°
Z_1	23		Z_2	36	
β	30	°	B	20	mm
m_1	0.301	kg	m_2	1.056	kg
M_1	0.248	kg	M_2	0.95	kg
k_1	43303	N/mm	r_{b1}	30.6	mm
K_2	280000	N/mm	K_3	0	N/mm
$\omega_{n1,2}$			6382.2Rad/s; 40600.6Rad/s		

$$\begin{cases} k_1(x_0 - x_1) - k_2(x_1 - x_2) - c_1\dot{x}_1 - c_2(\dot{x}_1 - \dot{x}_2) = m_1\ddot{x}_1 \\ F_2(t) - k_2(x_2 - x_1) - c_2(\dot{x}_1 - \dot{x}_2) - c_3(\dot{x}_2 - \dot{x}_3) - k_3(x_2 - x_3) = m_2\ddot{x}_2 \end{cases}$$

there is no driving force on neturle gear, so $F_2(t) = 0$, x_0 is the driving wheel torsional vibration equivalented to the displacement on the gear base circle,

$$x_0 = A\sin(\omega t) = \eta\omega_0\sin(\omega t):$$

$$\begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix} \begin{bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \end{bmatrix} + \begin{bmatrix} c_1 + c_2 & -c_2 \\ -c_2 & c_2 + c_3 \end{bmatrix} \begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \end{bmatrix} + \begin{bmatrix} k_1 + k_2 & -k_2 \\ -k_2 & k_2 + k_3 \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = \begin{bmatrix} F_1 \\ F_2 \end{bmatrix} e^{i\omega t}$$

$$\Rightarrow \begin{bmatrix} k_{11} - \omega^2 m_{11} + ic_{11}\omega & k_{12} - \omega^2 m_{12} + ic_{12}\omega \\ k_{21} - \omega^2 m_{21} + ic_{21}\omega & k_{11} - \omega^2 m_{22} + ic_{22}\omega \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = \begin{bmatrix} F_1 \\ F_2 \end{bmatrix} e^{i\omega t}$$

The natural frequency of the system is as follows:

$$\omega_n^2 = \frac{1}{2}(\omega_{11}^2 + \omega_{22}^2) \mp \frac{1}{2}\sqrt{(\omega_{11}^2 - \omega_{22}^2)^2 + 4\omega_{12}^4}$$

$$\omega_{11}^2 = \frac{K_1 + K_2}{m_1}; \omega_{22}^2 = \frac{K_2 + K_3}{m_2}; \omega_{12}^2 = \frac{K_2}{\sqrt{m_1 m_2}}$$

After importing data, we can see for 4 cylinder and 6 cylinder engine :

The natural frequency of the system $f = \frac{\omega_n}{2\pi} \gg$ the driving wheel torsional vibration frequency is (13.33–100Hz),

So there's no resonance

The steady - state solution is as follows.:

$$\begin{pmatrix} x_1 \\ x_2 \end{pmatrix} = \begin{bmatrix} \frac{z_{22}F_1 - z_{12}F_2}{z_{11}z_{22} - z_{12}^2} \\ \frac{-z_{22}F_1 + z_{11}F_2}{z_{11}z_{22} - z_{12}^2} \end{bmatrix} e^{i\omega t}; \quad z_{ij} = k_{ij} - \omega^2 m_{ij} + ic_{ij}\omega$$

Due to the damping value is very small. We can make $C_1=C_2=C_3=0$. When the two gears contact, the striking force (F) is as follows:

$$F = k_2(x_1 - x_2)e^{i\omega t} = \frac{\omega^2 F_1}{\frac{k_1 - m_1\omega^2}{m_2} - \frac{k_1}{k_2}\omega^2 - \omega^2 + m_1\omega^4} \approx \frac{m_2\omega^2 k_1 A \sin(\omega t)}{k_1} = m_2\omega^2 \omega_0 \eta \sin(\omega t)$$

$\omega = n\omega_0$; parameter(n) is the frequency of torsional vibration of the transmission input shaft (Unit: (n/r)n per revolution),

$$n = \begin{cases} 1(\text{The engine is Horizontal 4-cylinder engine(H4)}) \\ 2(\text{The engine is straight-line 4-cylinder engine(L4)}) \\ 3(\text{The engine is 6-cylinder engine}) \end{cases}$$

$$\Rightarrow F \propto n^2 \omega_0^3 \eta$$

It indicates that the speed volatility of transmission input axle(η), Driving wheel torsional vibration (n/r) and the basic angular speed of input axle(ω_0) are the 3 main parameters affecting the rattle force. Now we consider the Primary and secondary relations of the 3 main parameters, using Adams and the linear least squares interpolation to simulate and analyse the relationship between the rattle force F and $\eta\omega_0^3n$, the orthogonal analysis was also performed.

3. Establishment of virtual prototype.

Using Solidworks^[5] and geartrax to generate a 3-dimensional transmission gear model, import the model to Adams software then define two gear's meshing Force by CONTACT Force, using the sin function:

$$\omega = \omega_0 + \eta\omega_0 \sin(n\omega_0 t)$$

To create driving wheel speed curve, CONTACT Force reflects the the rattle force, we should mainly to determine the three coefficients: stiffness, damping and Force Exponent, the stiffness is the material stiffness, it means the elastic modulus E of the gear material (unit: Mpa), for steel gear couple, stiffness = 210000Mpa; Damping = (0.1%- 1%) stiffness value, Force Exponent =1, the CONTACT Force in Adams is defined as figure 2, and the virtual prototype model is shown in figure 2 and 3:

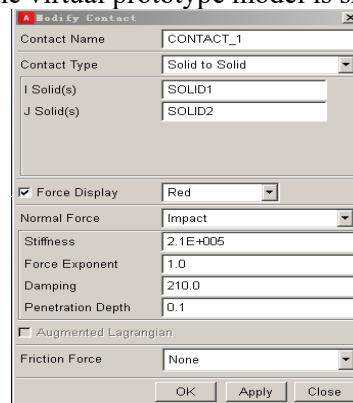


Figure 2. CONTACT force definition.

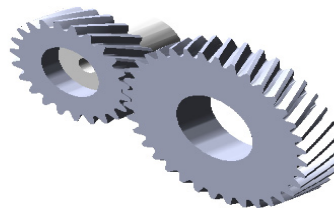


Figure 3. Virtual prototype model

4. Simulation analysis.

When the driving wheel rotational speed is between 900-2100r/min, $1\% \leq \eta \leq 5\%$ ^[6], and n respectively 1, 2, and 3 is the least square fitting of F and $n^2\omega_0^3\eta$ and its function is shown as follows:

$$F = 2.80518 \times 10^{-5} n^2 \omega_0^3 \eta + 98.87502729$$

The linear correlation coefficient ($r=0.951 > r_{\min}=0.632$) is linearly dependent. In figure 4, series 1 is the computer simulation value curve, and series 2 is the fitting curve. From figure 4, it can be seen that When the rotational speed of the driving wheel is between 900-2100r/min, and $1\% \leq \eta \leq 5\%$, the striking force F and $n^2\omega_0^3\eta$ are linearly proportional, there is no mutation, the result is consistent with the theoretical derivation.

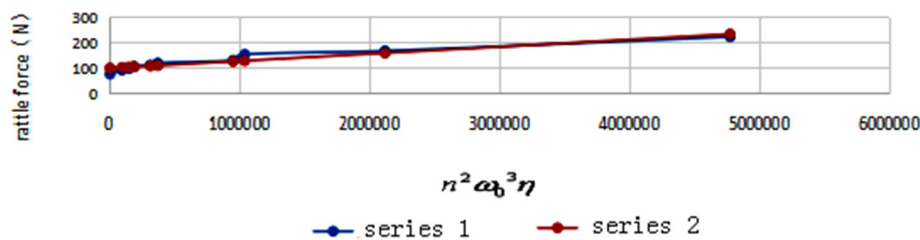


Figure 4. the relationship between the rattle force F and $n^2\omega_0^3\eta$.

Using orthogonal analysis to establish the primary and secondary relations of the 3 parameters: (η , n , ω_0), choosing $L_9(3)^4$ orthogonal table, the orthogonal table and orthogonal analysis are as follows: column 1 to 3 listed in the table respectively represent the 3 parameters (η , ω_0 , n)

Table 2. $L_9(3)^4$ orthogonal table.

Test no.	Column number				Test combination	Rattle force $F(N)$
	1 (η)	2 (ω_0)	3 (n/r)	4 (none)		
1	1 (1%)	1 (900r/min)	1 (1/r)	1	$\eta_1\omega_0(1) n_1$	75.3647
2	1 (1%)	2 (1500r/min)	2 (2/r)	2	$\eta_1\omega_0(2) n_2$	91.0564
3	1 (1%)	3 (2100r/min)	3 (3/r)	3	$\eta_1\omega_0(3) n_3$	129.7278
4	2 (3%)	1 (900r/min)	2 (2/r)	3	$\eta_2\omega_0(1) n_2$	97.1735
5	2 (3%)	2 (1500r/min)	3 (3/r)	1	$\eta_2\omega_0(2) n_3$	153.016
6	2 (3%)	3 (2100r/min)	1 (1/r)	2	$\eta_2\omega_0(3) n_1$	111.0269
7	3 (5%)	1 (900r/min)	3 (3/r)	2	$\eta_3\omega_0(1) n_3$	119.019
8	3 (5%)	2 (1500r/min)	1 (1/r)	3	$\eta_3\omega_0(2) n_1$	105.105
9	3 (5%)	3 (2100r/min)	2 (2/r)	1	$\eta_3\omega_0(3) n_2$	166.9421
K1	296.1489	291.5572	291.4966			
K2	361.2164	349.1774	355.172			
K3	391.0661	407.6968	401.7628			
range	94.9172	116.1396	110.2622			

The ranges of columns 1 to column 3 from big to small in turn are: $2 > 3 > 1$, thus driving wheel speed main influence parameters on the the rattle force from big to small in turn is ω_0 , n , η . From a practical point of view, reducing engine per revolution torsional vibration (n/r) and the transmission driving wheel speed volatility (η) can effectively reduce the rattle noise, the method is: on the premise of guarantee the dynamic performance, choose fewer cylinder engines as far as possible, as to 4-cylinder engine, choices Horizontal 4-cylinder engine (H4) is superior to straight-line 4-cylinder engine (L4), at the same time install a double quality flywheel between the transmission input axle and the engine output shaft can effectively reduce the speed volatility of transmission input axle (η) and then reduce the rattle force. Now take the Driving wheel basic speed of 1500r/min as an example, the driving wheel speed curve is

as follows: the parameters n , η and their respective effects on the rattle force F are showing in fig3 and fig4.

Table 3. the influence of volatility on the rattle force F when the active wheel speed is 1500r/min and the torsional vibration frequency is 1/r.

Volatility(η)	1%	2%	3%
Rattle force F (N)	55.2081	66.3015	83.0707

Table 4. the influence of the torsional vibration frequency on the rattle force F when the active wheel speed is 1500r/min and the fluctuation rate is 1%.

Torsional vibration frequency (n/r)	1	2	3
Rattle force F (N)	55.2081	78.9829	98.0605

From table3 and 4, we can see: when the basic driving wheel speed and the driving wheel torsional vibration frequency is constant, speed volatility(η)increases to 3 times, rattle force F increases by 57.7%, when the basic driving wheel and its volatility is constant, the driving wheel torsional vibration frequency increase to 3 times, rattle force increases by 77.6%, the simulation result show that reducing torsional vibration frequency of driving wheel(n) changes to reduce rattle force (F) is stronger than the effect of driving wheel speed volatility η , the first column of table 4 represents the Horizontal 4-cylinder engine(H4), the second column represents the straight-line 4-cylinder engine(L4) and the third column represents 6-cylinder engine, the above simulation results are in conformity with the orthogonal analysis results.

5. Conclusions

1) in this paper, theoretical derivation and simulation verification proves the specific internal relations between the transmission input axle speed and the transmission rattle force F : $F \propto \eta \omega_0^3 n^2$, resonance phenomenon don't occur within the speed range(900-2100r/min, $1\% \leq \eta \leq 5\%$), rattle force F don't mutate, transmission designers also don't need to consider resonance problem.

2) From orthogonal analysis we obtain the primary and secondary relationship between the 3 main parameters η , ω_0 , n , from the perspective of the power supply, on the premise of guarantee the dynamic performance, we should first try to reduce the number of engine cylinder, then install a double quality flywheel between the transmission input axle and the engine output shaft can effectively reduce the rattle force. When the 4-cylinder engine is used, the rattle force of Horizontal 4-cylinder engine(H4) is obviously less than that of straight-line 4-cylinder engine(L4).

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