The Vibration and Noise Study of Dry Double Clutch with Clutch Rigidity at Low Speed in Starting

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Abstract: By establishing engine-clutch-vehicle body nonlinear dynamic, the nonlinear dynamic behavior of DDCT is studied with Clutch rigidity. The nonlinear dynamic behavior of the chaotic and bifurcation of the vibration of the clutch system is analyzed. The stability of the system is determined by the maximum Lyapunov index method; And combined with experimental study, the clutch noise and the vibration acceleration of the gearbox during the movement of the car are tested. It shows as follows, Reducing the Clutch stiffness can effectively reduce the noise and vibration acceleration. It is also provide a theoretical basis for solving the noise problem in the starting process.

1. Introduction
Recently, the vibration and noise problem of the car during the engine starting process is one of the hot issues. This problem may be caused by the existence of nonlinear dry friction in the drive train, which causes the sliding state-slimy state-viscous state of the clutch friction plate to engage with the engine flywheel. Especially in the slow acceleration process, it will cause the car to appear jitter or abnormal noise [1-2]. In the research [3], a detailed excitation analysis is performed for the chattering phenomenon during clutch motion. The study [4] optimizes the internal structure of the transmission and the transmission path structure in two directions to alleviate the transmission gear knocking problem caused by engine torque fluctuation. Researches [5-6] study the clutch damping characteristics based on solving the abnormal acceleration characteristics of the vehicle, which mainly analyze the torsional rigidity and friction damping of the clutch against the torsional vibration attenuation rate of the clutch. In another research [7], deriving from solving the problem of gear slap of automobile transmission, a four-freedom-degree torsional vibration model is established, and the role of system parameters in reducing gear slamming vibration is discussed.

Many domestic and foreign studies have suggested that, the system will have complex nonlinear dynamics character when the relative velocity is low in systems with nonlinear dry friction. The studies [8-9] point out that when the car starts at low gear, the motion of the dry double-clutch system exhibits complex nonlinear dynamics character as the engine excitation increases, especially when the engine excitation increases at a low speed. This performance is more obvious. This paper uses a dry double-clutch car as the prototype, the three-freedom-degree dynamic model of the drive train is established. Based on the HOPF bifurcation theory, the influence of the change of clutch rigidity on the stability of the system is studied in detail. The stability of the system is determined by the largest Lyapunov index method.
2. Establish engine-clutch-drive system dynamics model

When the car starts at low block, one set of clutches of the dry double-clutch automatic transmission is engaged with the engine flywheel, and the other set is in the separated state [7-8]. To better study the influence of clutch rigidity changes in this process of the whole system, this paper ignores the interaction between the two sets of clutches, and considers the engine flywheel, clutch friction plate and pressure plate as a disc with a certain quality for axial rotation, and there is nonlinear friction force $T_f(t)$ during the start of engagement between the engine and the flywheel. The remaining part of the transmission system such as the transmission-drive shaft is simplified into a disc with a certain mass at the same time, and the disc has a torque $T_r$, as shown in Fig. 1.

In this Figure: $I_1, I_2, I_3$ are the corresponding rotational inertia of engine flywheel, the clutch friction plate, the drive system equivalent plate; $\theta_1, \theta_2, \theta_3$ are the absolute angular displacement of $I_1, I_2, I_3$ respectively; $K_1$ and $K_2$ are the clutch torsional vibration damping rigidity and the transmission connection rigidity, the concentrated viscous damping of $C_2$ connected to the clutch.

Based on the three-degree-of-freedom dynamics model of the engine-clutch-drive train shown in Fig. 1, and according to Newton's mechanics laws, the differential equations of motion of the system can be obtained as follows:

$$
\begin{align*}
I_1\ddot{\theta}_1 + K_1(\theta_1 - \theta_2) + T_f(\dot{\theta}_1 - \dot{\theta}_2, t) &= T_r(t) \\
I_2\ddot{\theta}_2 + K_2(\theta_2 - \theta_3) - K_1(\theta_1 - \theta_2) + c_2(\dot{\theta}_2 - \dot{\theta}_3) &= T_f(\theta_1 - \theta_2, t) \\
I_3\ddot{\theta}_3 + K_2(\theta_2 - \theta_3) + c_2(\dot{\theta}_2 - \dot{\theta}_3) &= -T_r
\end{align*}
$$

In the formula above: $T_f(t)$ are the torque excitation outputted by the engine; $T_f(\theta_1 - \theta_2, t)$ is the dry friction torque caused by the clutch friction plate; $T_r$ is the partial remained torque of the drive system. Replace the relative angular displacement with the absolute angular displacement in the equation (1), for example: $\delta_1 = \theta_1 - \theta_2$; $\delta_2 = \theta_2 - \theta_3$; $\delta_3 = \theta_3 - \theta_3$; $\delta_1 = \theta_1 - \theta_2$; $\delta_2 = \theta_2 - \theta_3$; $\delta_3 = \theta_3 - \theta_3$. Substituting the above formula into equation (1) and arranging it:

$$
\begin{align*}
\delta_1 - \frac{C_2}{I_2} \delta_1 + \frac{I_1 + I_2}{I_1 I_2} K_2 \delta_2 - \frac{K_1}{I_1} \delta_1 - \frac{I_1 + I_2}{I_1 I_2} T_f(\dot{\delta}_1, t) &= \frac{T_r}{I_1} \\
\delta_2 + \frac{I_2}{I_1 I_2} C_2 \delta_1 + \frac{I_2 + I_3}{I_1 I_2} K_2 \delta_2 - \frac{K_1}{I_1} \delta_1 &= \frac{T_f(\dot{\delta}_2, t)}{I_2} + \frac{I_2}{I_3} T_r
\end{align*}
$$

Equation (2) is the system motion differential equation based on relative angular displacement. Among them, $T_f(\dot{\delta}_1, t)$ is the friction force of the system calculated by the BOUC-WEN friction model. The characteristic curve of the BOUC-WEN friction model is shown in the following formula:
\[ N(t) = N_m + N_p \sin(\omega t + \phi) \]
\[ T_f(\delta, t) = \mu(\delta) N(t) R + \frac{\mu(\delta)}{\delta} \left( 2 + \frac{\varepsilon \delta}{\pi} \right) \]

In the above formula, \( \varepsilon \) and \( \gamma \) are the adjustment coefficients, which are taken as \( \varepsilon = 1000 \), \( \gamma = 110 \) in this paper; the relative torsion speed of the friction model \( \dot{\delta} \) is the difference between the torsion speed of the engine flywheel and the clutch friction plate. When the car starts to increase, the engine rotational speed increases gradually. When the friction disc and the flywheel start to engage, the excitation of the flywheel speed caused by the change of engine speed is \( \omega_1 \). The expression of \( \dot{\delta} \) is:

\[ \dot{\delta} = \omega_1 + \dot{\theta}_1 - \dot{\theta}_2 = \omega_1 + \dot{\delta} \]

The BOUC-WEN friction model characteristic curve is shown in Fig. 2.

Equations (2), (3), and (4) are the system equations of the engine clutch power train model shown in Fig. 1.

3. Analysis of the influence of clutch torsional vibration-damping rigidity on system vibration

In order to study the behavior of flutter and noise during the starting process of the car, based on the three-degree-of-freedom dynamic system equation established above, the system equation is transformed into the state equation in the MATLAB platform (set \( x_1 = \delta_1 \); \( x_2 = \dot{\delta}_1 \); \( x_3 = \delta_2 \); \( x_4 = \dot{\delta}_2 \)), the bifurcation diagram, phase diagram and other methods in nonlinear dynamics are used to analyze the influence of clutch rigidity variation on the system. The parameters required for calculation are shown in Table 1.

| Parameter          | Value                          |
|--------------------|--------------------------------|
| Rotational Inertia | \( I_1 = 0.1 \text{kg.m}^2; I_2 = 0.1 \text{kg.m}^2; I_3 = 6.6 \text{kg.m}^2 \) |
| Rigidity           | \( K_2 = 950 \text{ N.m/}^\circ \) |
| Resistance         | \( C_2 = 0.6 \) |
| Other Parameters   | \( T_e = 200 \text{N.m}; N_m = 300 \text{N.m}; N_p = 100 \text{N.m}; T_r = 200 \text{N.m}; \sigma = 50; \alpha = 2; \omega_1 = 2 \text{rad/s} \) |

Because the relative speed of the friction plate and the flywheel represented by \( \dot{\delta}_1 \) (x2), if \( \dot{\delta}_1 = 0 \), the relative speed is 0, which indicates that the friction plate and the flywheel are purely viscous, and
the larger $\delta$ indicates the larger the relative speed. According to the parameter value is shown in the table 1, a bifurcation diagram of the clutch friction plate and the flywheel relative speed $x_2$ with the change of the clutch torsional rigidity $K_1$ when the dry double-clutch system is in the slowly acceleration of start process, as shown in the Fig. 3:

The horizontal axis in Fig. 3 is $K_1$(N.m/°); the vertical axis is $x_2$ (rad/s). It can be seen from the figure, in the range of $K_1=0$~$60$ N.m/°, with the increase of $K_1$, the bifurcation diagram of $x_2$ appears complex dynamic behavior. According to the bifurcation diagram, when $K_1=0$~$24.5$, $x_2$ has multiple points and the value of the maximum point decreases with the increase of $K_1$, and when $K_1=0$~$60$, there are three intervals, $x_2$ is a single point, the three intervals is $K_1=(24.5$~$26.5)$, $(34.9$~$38.4)$, $(50.5$~$55.5)$, respectively. In other words, when the value of $K_1$ is in these ranges, the vibration of the system may be a stable periodic vibration.

In view of this phenomenon, firstly, according to Fig. 3, the clutch torsional vibration rigidity values corresponding to typical bifurcation and chaotic behavior are selected, and the influence of different rigidity values on the vibration characteristics of the system is analyzed. Secondly, the Lyapunov index of the system is further calculated to distinguish the chaotic interval and the periodic window. Finally, combined with theoretical analysis, the test noise and acceleration spectra were compared and verified in the bench test.

According to Fig. 3, we take a clutch torsional vibration damping rigidity in the same cycle interval and calculate the system relative angular velocity vibration characteristics corresponding to the value, taking $K1=5$, 14.6, 20, 25.5, 30, 37, 38.5, 54 (Nm/°), respectively. The value is calculated corresponding with the phase diagram, Poincare diagram and time domain diagrams of the system, as shown in Fig. 4~7.
Fig. 4 Phase diagram of system $x_2$ at different values of $K_1$.

Fig. 4 is a phase diagram of the system friction plate and flywheel relative angular velocity $x_2$ when $K_1$ is different. It can be seen from these figures that the phase diagrams of Fig. 4(a), (b), (c), and (g) are irregular, and the vibrations of Fig. 4(d), (f), and (h) are a closed loop, which indicates that it may be the limit cycle phenomenon. In Fig. 4(e), there are two periodic rings, which indicates that the system vibration may have double cycle bifurcation.

It can be seen from the system $x_2$ Poincare diagram of Fig. 5. Fig. 5(a), (b), (c), (g) in the four figures are multiple irregular points, Fig. 5(d), (f), (h) with only one point in the figure, combined with corresponding three phase diagrams in the Fig. 4, it is ensured that the system performs the single-cycle vibration in this range, which is the limit loop phenomenon of the system vibration; combined with two points in Fig. 5(e) and Fig. 4(e), the system is explained that the vibration appears multiple cycle bifurcation.
The pictures shown in Fig. 6 are the time domain diagram corresponding to clutch rigidity in the system $x_2$. Fig. 6 (a), (b), (c), (g) are irregular vibrations, and steady state in Fig. 6 (d), (f), (h) is equal-amplitude vibration, and the steady state shown in Fig. 6 (e) has two different peaks in one cycle.

As can be seen from Fig. 3, Fig. 4, Fig. 5 and Fig. 6, as the damping rigidity of the clutch increases, the $x_2$ vibration of the system first appears chaotic, and then complex phenomena such as single vibration period and doubling bifurcation period occur within a certain range of values. And as the damping rigidity of the clutch increases, the amplitude of the vibration of the system $x_2$ will become smaller and smaller in different single-cycle vibration intervals.

4. Lyapunov index identification periodic solution and chaos

It can be known from the previous analysis that combined with phase diagram, Poincare diagram and time domain diagram, it can be clearly judged whether the system performs periodic vibration or chaotic vibration under a specific bifurcation parameter value. However, the range of the bifurcation parameters of the bifurcation diagram shown in Fig. 3 is large. It is not preferable to determine the specific motion form of the system by making the Poincare diagram and the spectrogram under each bifurcation parameter point by point. For obtaining the specific bifurcation and chaos interval of the system, the Lyapunov index of the computing system will be used to judge the motion form. This paper uses the 'clone' method to calculate the maximum Lyapunov index of the system. Analyze the motion stability of the system. Set the initial disturbance to $10^{-5}$ in Matlab, the time interval is 0.01, and the number of iterations is $10^4$. The maximum Lyapunov index spectrum of the system when $K_1$ is different, as shown in Fig. 7.

When the maximum Lyapunov index of the system is less than or equal to 0, it corresponds to the periodic motion in the chaotic interval. As can be seen in conjunction with Fig. 3 and Fig. 7, when $K_1 = (24.5 \sim 26.5), (34.9 \sim 38.4), (50.5 \sim 55.5)$ The single cycle is shown in the bifurcation diagram. In Fig. 7, the maximum Lyapunov index is less than or equal to zero. In other regions, the system also has a local maximum Lyapunov index of less than zero, indicating that the system is a single-cycle motion, combined with a phase diagram, a Poincare diagram, and a time-domain diagram to determine...
the limit cycle of the system motion at this time.

5. Conclusion
In this paper, the three-freedom -degree torsional vibration equation of the dry double-clutch vehicle slow-starting time system is established. The numerical simulation method is used to analyze the change process of the system vibration with the increase of the vibration damping rigidity of the clutch.

(1) It is found that with the increase of the vibration damping rigidity of the clutch, the system has complex dynamic behaviors from chaotic cycle multiplication to single-cycle vibration, and multiple single-cycle vibration intervals appear within a certain range.

(2) Further analysis of the bifurcation and chaos behavior of the system reveals that as the damping rigidity of the clutch increases, the amplitude of the vibration of the system will become smaller and smaller in different single-cycle vibration intervals. However, the amplitude will increase with increasing rigidity in the same single cycle interval.

(3) Using the maximum Lyapunov index method to compare with the bifurcation diagram and verify the complex dynamic behavior of this chaotic and bifurcation system.

Provide a theoretical basis for the transmission system in engineering design.

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