

Nonlinear Characteristics Study and Parameter Optimization of DMF-RS

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ABSTRACT

Dual Mass Flywheel (DMF) has better damping capacity than the conventional Clutch Torsional Damper (CTD), and is more suitable for diesel engine, Dual Clutch Transmission (DCT) and hybrid vehicles. Dual Mass Flywheel-Radial Spring (DMF-RS) is a DMF that has a specific structure. In the light of working principle and static analysis, the hard nonlinear torsional stiffness of DMF-RS is derived in this paper, which is very important to a driveline damper. On this basis, a simulation model is developed to analyze the dynamic response of DMF and CTD excited by idle engine; the comparison of the two dampers reveals that the DMF has better damping capacity, high-frequency filter ability and can reduce crankshaft load. Further, the nonlinear vibration transfer behavior of DMF-RS and its influencing factors are investigated using the method of harmonic balance; it not only indicates that appropriate damping ratio will avoid unstable jump and bifurcation, but also illustrates that resonance frequency of DMF-RS is lower than that of CTD. Besides, the virtual prototype is optimized by Generalized Reduced Gradient (GRG) method, and the results show the influence of variables on damping capacity. The use of the model can be applied as theoretical foundation and can reduce the time to design a DMF-RS since the optimal approach is more efficient, which will be utilized to guide the future experimental studies.

INTRODUCTION

Though the concept of electric vehicle is attractive recently, its poor dynamic performance and driving range limit the popularization. The internal-combustion powered vehicles like diesel or hybrid electric vehicles are still in great demand. However, these economic and powerful vehicles still meet vibration problems in the driveline. For example, the diesel engine always knocks seriously, and driveline of

hybrid electric vehicle vibrates fiercely when there is an inertia change caused by driving power source switching.

The traditional Clutch Torsional Damper (CTD) with limited damping capacity cannot solve these problems. Dual Mass Flywheel (DMF), developed in 1980s, is a more reasonable vibration damper for automotive driveline because of more effective vibration attenuation capacity and greater development potential. What's more, the DMF can match the Dual Clutch Transmission (DCT, a promising type of automatic transmission) well, improve the working condition of the transmission greatly, and be mounted on the original vehicle prototype with few modifications in the driveline.

Actually, a DMF is to move the damper from the clutch driven plate to the flywheel that cut into two parts. This kind of reformation will enlarge the space for designing a better damper and distribute rotary inertia of driveline rationally. Recently, the DMF arouses wide concern in virtue of the advantages of simple structure, low cost, and the ability to avoid idling resonance, as well as the better performance of ride comfort, vehicle dynamics, economics and emission [1].

Typically, there are several types of DMF: the DMF-Circumferential Spring (DMF-CS) that has advantages of low stiffness and simple structure, but shortcomings of tightness and easy affecting by centrifugal effects; the DMF-Circumferential Short Spring (DMF-CSS) which diminishes centrifugal effects but is complex in structure; the DMF-Radial Spring (DMF-RS) that possesses favorable nonlinear torsional stiffness but mainly matches small engine; as well as the different forms of damping and bearing configurations such as DMF-Hydraulic Springs, DMF-Quadrangle Spring, DMF-Friction Bearing, DMF-Thrust Bearing, etc.

A lot of research on spring circumferentially mounted DMF (DMF-CS and DMF-CSS) has been done, e.g. the influence

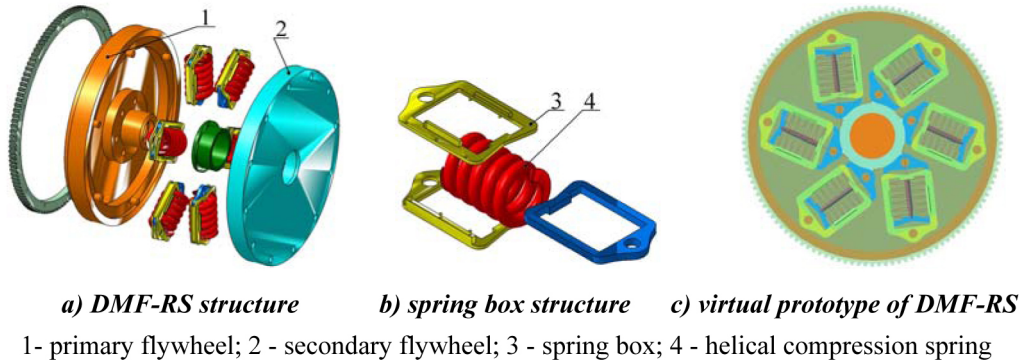
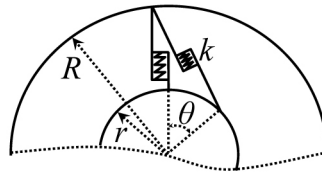


Figure 1. Structure and virtual prototype of DMF-RS



R - outer install radius of spring box; r - inter install radius of spring box
 θ - relative angle between two parts of flywheel; k - stiffness coefficient of spring

Figure 2. Working principal of DMF-RS

on the vehicle NVH behavior when the driveline is equipped with DMF [1, 2], the analysis methods for DMF [3] and engine control based on the applications of DMF [4, 5, 6, 7, 8]. And these DMFs usually mounted in high class vehicles. While the spring radially arranged DMF-RS which matches small displacement vehicles well has been rarely studied [9, 10], especially the dynamic performance. As a matter of fact, with the special advantages and characteristics like favorable nonlinear torsional stiffness realized by simple structure, high-frequency filter ability and particular transfer behavior, DMF-RS is quit suitable to be a driveline damper.

In this paper, a thoroughly investigation of DMF-RS emphasized on nonlinear dynamic behavior is discussed, and the work proceeds with working principle and static characteristics study of DMR-RS, then followed by dynamic performance and transfer behavior research, and finally the optimization of virtual prototype.

1. WORKING PRINCIPAL OF DMF-RS

Actually, a DMF is to move the driveline damper from the clutch driven plate to the flywheel which is cut into two parts, the primary and secondary flywheels. This structure will increase the space for damper design and improve the vibration condition of driveline since the rotary inertia is more rationally distributed. Figure 1 shows the structure of DMF-RS, the springs are helical compression springs

mounted in spring boxes. Each spring box is connected to the two flywheels via two drive pins. Figure 2 is the mechanical model that shows the working principal. When there is a relative angle between the two flywheels, the distance between the two drive pins would be lengthened and the spring box would be stretched, which makes the spring compressed so as to transfer torque.

The torque characteristic of DMF-RS is different to normal damper, assuming that $\delta = \sqrt{R^2 + r^2 - 2Rr \cos(\theta)}$, $\Delta l = \delta - R + r + \Delta l_0$, according to moment balance, the relation between restoring torque T_K (transferred by DMF-RS), and the relative angle θ , can be derived as

$$T_K = Nk\Delta l \frac{Rr \sin(\theta)}{\delta} \quad (1)$$

where, N and Δl_0 stand for the number and precompaction of helical compression springs, respectively.

The torsional stiffness K is obtained by deriving T_K to θ

$$K = \frac{NkRr}{\delta} \left[\Delta l \cos(\theta) + \frac{(\delta - \Delta l)Rr \sin^2(\theta)}{\delta^2} \right] \quad (2)$$

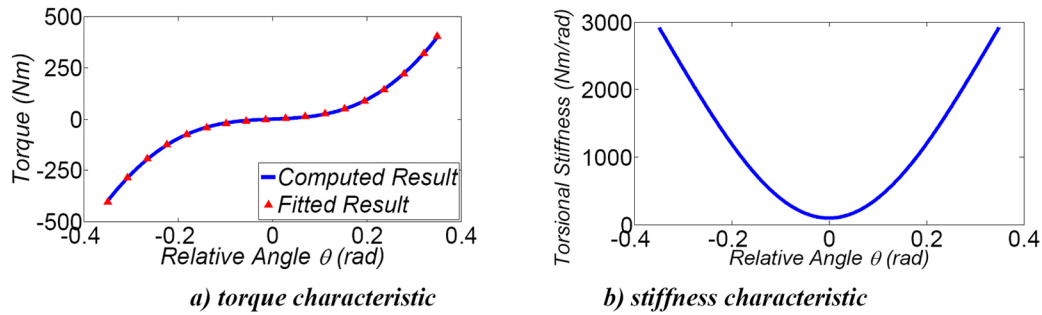


Figure 3. Torque and stiffness characteristics of DMF-RS

Based on Equations (1) and (2), torque and stiffness characteristics are acquired as shown in Figure 3. DMF-RS turns linear stiffness of helical compression spring to fine hard nonlinear torsional stiffness, which meets the requirement of vehicle driveline torsional damper. Other than normal damper, of which torsional stiffness is piecewise linear; the stiffness and torque of DMF-RS vary smoothly, without impacting parts and decreasing life expectancy of damper.

2. SIMULATION COMPARISONS BETWEEN DMF-RS AND CTD

Idle noise is an important evaluating indicator of ride comfort. In idling condition, the excitation of engine is fierce in fluctuation and low in frequency, which results in resonance vibration of the driveline system obviously. This paper takes idling condition as a case study to analyze damping capacity of DMF-RS. Noticing the low stiffness of torsional damper, the other parts of the driveline can be considered as rigid connection, and the rotational inertia of these parts is equivalently attached to the two sides of DMF-RS, thus, the driveline can be simplified as a two-degree of freedom torsional vibration system subjected to the output torque of engine, as shown in Figure 4, and the differential equation is

$$\begin{cases} I_2 \ddot{\theta}_2 = K(\theta_1 - \theta_2) + C_d(\dot{\theta}_1 - \dot{\theta}_2) \\ I_1 \ddot{\theta}_1 = T - K(\theta_1 - \theta_2) - C_d(\dot{\theta}_1 - \dot{\theta}_2) \end{cases} \quad (3)$$

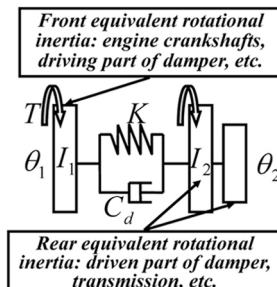


Figure 4. Equivalent torsional vibration system

where, I_1 is the front equivalent rotational inertia (for DMF-RS, it contains crankshaft and primary flywheel; and for CTD, it contains crankshaft, flywheel and clutch cover), I_2 is the rear equivalent rotational inertia (for DMF-RS, it includes secondary flywheel, clutch assembly and transmission shaft; for CTD, it includes driven plate, transmission shaft, etc); θ_1 and θ_2 are the angular displacement of the front and rear equivalent rotational inertia, respectively; T is the excitation torque, K is torsional stiffness (for DMF-RS, it's nonlinear stiffness, and liner stiffness for CTD), C_d is torsional damping.

Besides the values of rotational inertia and damping coefficient, the major difference between CTD and DMF-RS in Equation (3) is the torsional stiffness. The torsional stiffness of CTD can be simplified as linear, but for DMF-RS, it's a nonlinear function of relative angle between the front and rear rotational inertia as defined in Equation (2). A model is built according to Equation (3) in MATLAB/Simulink to contrast dynamic response subjected to idle engine between DMF-RS and CTD. In the light of practical application of the damper, the more stable the output angular acceleration is, the more stable the output torque is, and the better vibration attenuation effect is, so the output angular acceleration is chosen to evaluate the damping capacity.

Figure 5 shows the simulation results, and the amplitude attenuation exhibited in Figure 5a) indicates that DMF-RS is more effective in driveline torsional vibration attenuation than CTD; Figure 5b) shows that DMF-RS can obviously inhibit high frequency components of the excitation, and it can be used as low-pass filter to eliminate high order resonance; Figure 5c) shows that the load on crankshaft can be lowered because DMF employs smaller primary flywheel rotational inertia. Furthermore, DMF-RS can use smaller damping to achieve equal vibration damping capacity, so it can realize higher power transfer efficiency.