Parameter influence law analysis and optimal design of a dual mass flywheel

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Abstract
The influence of the dynamic parameters of a dual mass flywheel (DMF) on its vibration reduction performance is analyzed, and several optimization algorithms are used to carry out multiobjective DMF optimization design. First, the vehicle powertrain system is modeled according to the parameter configuration of the test vehicle. The accuracy of the model is verified by comparing the simulation data with the test results. Then, the model is used to analyze the influence of the moment of inertia ratio, torsional stiffness, and damping in reducing DMF vibration. The speed fluctuation amplitude at the transmission input shaft and the natural frequency of the vehicle are taken as the optimization objectives. The passive selection method, multiobjective particle swarm optimization, and the nondominated sorting genetic algorithm based on an elite strategy are used to carry out DMF multiobjective optimization design. The advantages and disadvantages of these algorithms are evaluated, and the best optimization algorithm is selected.

KEYWORDS
dual mass flywheel, vehicle powertrain system, multiobjective optimization, multiobjective particle swarm optimization, nondominated sorting genetic algorithm based on an elite strategy

1 | INTRODUCTION

The powertrain vibration takes different forms, including torsional vibration, bending vibration, and longitudinal vibration, as well as coupled vibration caused by mutual coupling. The most important one is torsional vibration, which has a significant impact on the ride comfort. In recent years, to improve vehicle performance, automobile manufacturers have steadily increased the number of vehicles equipped with high-power engines, and the application of diesel engines has become increasingly more common, which undoubtedly increases the irregularity of engine operation and torque fluctuations. At the same time, it is necessary to improve the vibration-isolation performance of the torsional vibration damper of the powertrain. Due to the limited installation space of the clutch torsional damper (CTD), the relative angle of the damping spring cannot be set too large; therefore, it can only be equipped with a damping spring with higher stiffness, leading to a reduction of the damping effect. Furthermore, for vehicles equipped with CTD, the resonance speed of the powertrain system is often near the idle speed, causing the vehicle to experience resonance in idle conditions.

A dual mass flywheel (DMF) addresses the deficiencies of CTD and can lead to better vibration isolation. Compared with CTD, DMF mainly has the following two advantages. First, there is a large installation space between the primary and secondary flywheels, and a torsional damper with a larger relative angle can be arranged. This means that damping springs with smaller torsional stiffness can be used. Second, by properly distributing the moment of inertia of the flywheels on both sides, the
natural frequency corresponding to the frequency of fundamental mode of the powertrain system can be reduced, and the resonance of the vehicle at idle speed can be eliminated.\(^4\)

Hartmut et al.\(^5\) developed the dynamic model of the DMF drivetrain system, analyzed the rotational speed fluctuation and angular acceleration of the engine, transmission input, and output shaft, and evaluated the vibration reduction effectiveness of the DMF using the angular acceleration amplitude of the secondary flywheel. Kang et al.\(^6\) used several test methods and theories to establish a performance test platform for DMF, which provided experimental conditions for verifying the theoretical research, performance improvement, and parameter optimization of DMF. Chen et al.\(^7\) analyzed how to distribute the moment of inertia on both sides of DMF and design the multistage torsional stiffness, and designed a DMF based on test vehicle information matching, which was verified using ADAMS software to have good working performance, but no design optimization was carried out. Cheng\(^8\) used the optimization analysis tool in ADAMS to optimize the three parameters of the DMF’s moment of inertia ratio, torsional stiffness, and damping coefficient with the angular acceleration fluctuation amplitude of the transmission input shaft as the optimization target. However, Cheng only used the optimization tools of ADAMS without comparing the advantages and disadvantages of the optimization algorithm.

The structure of this paper is as follows: In Section 2, a certain MPV vehicle is taken as the study model. First, the vehicle powertrain model is developed according to its structural parameters, and the torsional vibration of the vehicle is simulated. The vehicle test is carried out, and the simulation data are compared with the experimental data to verify the accuracy of the simulation model. In Section 3, the vehicle powertrain-system model is used to study the influence of the DMF’s primary and secondary flywheel moment of inertia ratio, torsional stiffness, and damping on the torsional vibration of the drivetrain. In Section 4, the rotational speed fluctuation amplitude at the transmission input shaft and the natural frequency of the powertrain system are selected as optimization objectives. The moment of inertia ratio, torsional stiffness, and damping are taken as optimization variables. The passive selection method (PSM), multiobjective particle swarm optimization (MOPSO), and the nondominated sorting genetic algorithm (NSGA) based on an elite strategy are used to optimize DMF. The workflow of this article is shown in Figure 1.

2 | POWERTRAIN SYSTEM MODELING AND EXPERIMENTAL VERIFICATION

2.1 | Vehicle powertrain branched model

2.1.1 | Engine transient torque model based on GT Power

The engine torque model can be divided into a steady-state torque model and a transient torque model according to whether it can...
reflect the fluctuation of engine torque in a working cycle.\textsuperscript{9} Since it is difficult for the engine steady-state torque model to simulate the torsional vibration component, the Fourier series is usually superimposed on this basis to simulate the torsional vibration of the engine,\textsuperscript{10} and different orders are selected according to different accuracy requirements.\textsuperscript{11} However, in general, this method cannot simulate the torque fluctuation of the engine accurately.

According to the working principle of the engine, the key to obtain the transient output torque of the engine is to obtain the gas pressure data in the cylinder at different speeds and different throttle valve openings.\textsuperscript{12} The gas pressure in the cylinder is obtained through a series of processes such as fuel injection, gas distribution, timing, combustion, and so forth. It is difficult to establish a mathematical model for the speed and throttle opening. This paper uses the cosimulation between GT Power and MATLAB/Simulink to simulate the transient torque of the engine. The engine model implemented in GT Power is shown in Figure 2. The engine speed and throttle opening signals are implemented in MATLAB/Simulink as shown in Figure 3. The GT-SUITE Model module in Simulink is responsible for the data interaction between MATLAB/Simulink and GT Power and transmits the engine speed and throttle opening signals to GT Power, so that the engine model works at the corresponding speed and throttle opening. The Simulink-Harness module in GT Power imports the engine transient torque into Simulink as the power source for the vehicle powertrain model.

\subsection*{2.1.2 Force analysis of the vehicle}

The branch model of the drivetrain system needs to consider not only the torsional degrees of freedom of the drivetrain system but also the translational degrees of freedom of the vehicle. Through the force analysis of the driving wheel and the driven wheel, the interaction between the translational mass of the vehicle and the drivetrain system is studied. The force analysis of the whole vehicle is shown in Figure 4.

By simplifying the two driven wheels of the rear axle of the vehicle into one, the rolling resistance of the driven wheels of the rear axle can be obtained from

\begin{equation}
F_{tr} = F_{0}r f, 
\end{equation}

where $f$ is the rolling resistance coefficient.
Due to the consideration of the left and right half-axle branch structures of the front-wheel drive vehicle, the front axle needs to be analyzed separately according to the left and right drive wheels. Assuming that the vertical loads on the left and right drive wheels are equal, the rolling resistance of the left and right drive wheels of the front axle can be obtained from

\[ F_{fl} = F_{fr} = \frac{F_{ff}}{2}. \]  

(2)

The resistance torques acting on the left and right front wheels are, respectively,

\[ T_l = F_{fl}r_L + F_{fL}r_L, \]  

(3)

\[ T_r = F_{fr}r_L + F_{fR}r_L. \]  

(4)

The vertical load distributions of the front and rear axles of the vehicle are given, respectively, as

\[ F_{l} = \frac{L_m g - m_u \ddot{x}_u H_u - F_w H_w}{L_a + L_b}, \]  

(5)

\[ F_{r} = \frac{L_m g + m_u \ddot{x}_u H_u + F_w H_w}{L_a + L_b}. \]  

(6)

The air drag of the vehicle is

\[ F_w = \frac{1}{2} C_D A \rho_a \ddot{x}_u^2, \]  

(7)

where \( C_D \) is the air drag coefficient, \( A \) is the frontal area of the vehicle, and \( \rho_a \) is the air density.

According to the Lugre tire model, the longitudinal force between the left and right front wheels and the ground can be expressed as

\[ F_{fl} = \left[ \sigma_0^l \ddot{x}_u^l + \sigma_1^l \dot{x}_u^l + \sigma_2^l v_{1u} \right] F_{fr} \]  

\[ \ddot{x}_u \]  

(8)

where \( \sigma_0^l \) is the equivalent stiffness of the bristles, \( \sigma_1^l \) is the equivalent damping of the bristles, \( \sigma_2^l \) is the relative damping of the friction surface, \( \ddot{x}_u \) is the average deformation of the bristles, and \( v_{1u} \) is the relative velocity of the friction surface.

According to Newton’s Second Law, Equation (9) yields

\[ F_{fl} + F_{fr} - F_w = m_u \ddot{x}_u. \]  

(9)

2.1.3 Vehicle powertrain model

This paper considers a certain MPV vehicle running on a level road and working in the first gear creeping condition as the study example, and develops the vehicle powertrain model as shown in Figure 5.

According to Newton’s Second Law, the equation of motion of the powertrain is obtained as

\[ J \ddot{\theta} + C \dot{\theta} + K \theta = T(t). \]  

(10)

where \( J \) is the moment of inertia matrix, \( C \) is the torsional damping matrix, \( K \) is the torsional stiffness matrix, \( \theta(t) \) is the angular displacement vector, and \( T(t) \) is the external moment vector.

The meanings and values of the parameter and symbols in the vehicle powertrain model are listed in Table 1. Some of these parameters are directly provided by the cooperative enterprises, and some are obtained by CAD software according to the materials and dimensions of the parts provided.

2.2 Comparison of test and simulation results

2.2.1 Results of the test

The signals that need to be collected in the vehicle test are the speed signals at the DMF primary flywheel and the transmission input shaft.
The test equipment used in the test mainly includes Hall-type speed sensors, a ROTEC torsional vibration tester, and a laptop equipped with a RASnbk control and analysis system, as shown in Figure 6.

Figure 7 shows the speed signals of the engine and the transmission input shaft collected under the first gear creeping condition. The speed of both fluctuates around 840 rpm, but the engine speed fluctuation amplitude reaches 140 rpm, but only 20 rpm at the transmission input shaft. After the DMF vibration reduction, the amplitude of the speed fluctuation is reduced by 120 rpm, and the attenuation range of the speed fluctuation is 85.7%, indicating that the DMF can considerably attenuate the torsional vibration of the transmission system.

Figure 8 shows the spectrum analysis results of engine and transmission input shaft speed signals obtained by Fourier Transform. It can be seen from the figure that the speed fluctuation of the engine has peaks of 57.8 and 16.6 rpm at 28 and 56 Hz, respectively. These two frequencies correspond to the second-order and fourth-order excitation frequencies of the four-cylinder four-stroke engine. Similarly, the speed fluctuation of the input shaft of the transmission also has peaks at 28 and 56 Hz, and the amplitudes are 4.2 and 0.4 rpm, respectively. The amplitudes of the second-order and fourth-order speed fluctuations are reduced by about 53.5 and 12.5 rpm, respectively, and the attenuation amplitudes reach 92.6% and 97.8%, which also indicates the excellent DMF vibration damping performance.
2.2.2 | Results of simulation

The developed vehicle powertrain model is used to simulate and process the obtained simulation data to obtain the results presented in this section. Figure 9 shows the rotational speed of the engine and the input shaft of the transmission. The rotational speed of both fluctuates around 840 rpm, but the fluctuation amplitude of the rotational speed of the engine reaches 150 rpm, while that of the transmission input shaft is only 20 rpm. After the DMF vibration reduction, the speed fluctuation is attenuated by 86.7%. It can be concluded that the engine speed and the speed fluctuation of the transmission input shaft are in good agreement with the test results, indicating that the established vehicle powertrain system model can reproduce the torsional vibration accurately.

Figure 10 shows the spectral analysis results of engine and transmission input shaft speed signals obtained by Fourier Transform. It can be seen from the figure that the speed fluctuations at both the engine and the transmission input shaft have peaks at 28 and 56 Hz. The amplitudes of the second-order and fourth-order speed fluctuations are reduced by about 67.1 and 14.8 rpm, respectively, and the attenuation amplitudes reach 90.6% and 97.1%. This is also consistent with the experimental results, indicating that the frequency components of the simulation data and experimental data are basically the same.

3 | PARAMETER INFLUENCE LAW ANALYSIS

Based on the vehicle powertrain-system model developed in Section 2, the effects of the rotational inertia ratio of primary and secondary flywheels \( \mu = J_{PF}/J_{SF} \), the first-level torsional stiffness \( k_1 \), and the damping \( c \) on DMF damping performance are analyzed.

DMF not only has the function of a torsional vibration damper but also has the function of energy storage as an ordinary flywheel. Therefore, it is necessary to make sure that the total inertia of both flywheels remains unchanged. Figure 11 shows the variation of the rotational speed fluctuation amplitude of the transmission input shaft when the DMF moment of inertia ratio is changed. When the moment of inertia ratio increases, the amplitude of speed fluctuation shows a trend of increasing first and then decreasing. When \( \mu = 0.9 \), the fluctuation amplitude of the speed of the transmission input shaft is the lowest, which is 17.78 rpm. It can be seen from the figure that a reasonable selection interval of the moment of inertia ratio should be 0.5–2.

Figure 12 shows the variation of the speed fluctuation amplitude of the transmission input shaft when the DMF first-stage torsional stiffness is changed. It can be seen from the figure that when the torsional stiffness increases, the rotational speed fluctuation amplitude also increases, and the graph trend presents a quadratic curve...
| Parameter | Meaning                                                                 | Value       | Unit        |
|-----------|--------------------------------------------------------------------------|-------------|-------------|
| $J_1$     | Moment of inertia of the primary flywheel                                 | 0.133       | kg m$^2$    |
| $J_2$     | Total moment of inertia of the secondary flywheel and clutch cover assembly | 0.062       | kg m$^2$    |
| $J_3$     | Moment of inertia of the clutch disc                                      | 0.01        | kg m$^2$    |
| $J_4$     | Total moment of inertia of the transmission input shaft                   | $1.086 \times 10^{-3}$ | kg m$^2$    |
| $J_5$     | Total moment of inertia of the transmission output shaft                  | $5.62 \times 10^{-3}$ | kg m$^2$    |
| $J_6$     | Moment of inertia of the driving gear of the final drive                  | $4.6 \times 10^{-4}$ | kg m$^2$    |
| $J_7$     | Total moment of inertia of the driven gear ring of the final drive, the differential housing, planetary gear and the pin shaft around the axis of rotation | 0.025 | kg m$^2$    |
| $J_{10}$, $J_{11}$ | Moment of inertia of the axle shaft gear around its axis | $7.96 \times 10^{-5}$ | kg m$^2$    |
| $J_{12}$, $J_{13}$ | Total moment of inertia of the left/right half shaft and the hub, rim, and brake disc on the same side | 0.7423 | kg m$^2$    |
| $c_1$     | Equivalent DMF viscous torsional damping                                 | 0.2         | Nm s/rad    |
| $c_2$     | Equivalent viscous torsional damping of the transmission input shaft between the clutch driven disc and the first/reverse driving gear | 0.1 | Nm s/rad    |
| $c_3$     | Equivalent viscous torsional damping of the transmission output shaft l between the first/second speed synchronizer and the driving gear of the final drive | 0.1 | Nm s/rad    |
| $c_4$, $c_5$ | Equivalent viscous torsional damping of the left/right half shaft | 0.2 | Nm s/rad    |
| $c_6$, $c_7$ | Equivalent viscous torsional damping of the tire of left/right half shaft | 1.0 | Nm s/rad    |
| $k_1$     | First-stage DMF torsional stiffness                                       | 138         | Nm/rad      |
| $k_2$     | Torsional stiffness of the transmission input shaft                       | 8780        | Nm/rad      |
| $k_3$     | Torsional stiffness of the transmission output shaft                      | $1.18 \times 10^5$ | Nm/rad      |
| $k_4$, $k_5$ | Torsional stiffness of the left/right half shaft                           | 8100, 5900  | Nm/rad      |
| $k_6$, $k_7$ | Torsional stiffness of the tire of the left/right half shaft              | $1.96 \times 10^4$ | Nm/rad      |
| $r_{bp1}$ | Radius of base circle of the first gear driving gear                      | 17.08       | mm          |
| $r_{rg1}$ | Base circle radius of the first gear driven gear                          | 63.08       | mm          |
| $r_{rgfr}$ | Base circle radius of the main reducer drive gear                         | 19.71       | mm          |
| $r_{rgfr}$ | Base circle radius of the main reducer driven gear                        | 93.30       | mm          |
| $H_g$     | Height of vehicle center of mass                                          | 0.643       | m           |
| $H_w$     | Height of the wind pressure center                                        | 0.643       | m           |
| $L_a$     | Distance from vehicle center of mass to the front axle                   | 1.209       | m           |
| $L_b$     | Distance from vehicle center of mass to the rear axle                     | 1.891       | m           |
| $m_v$     | Vehicle mass (curb weight + 4 occupants)                                 | 2500        | kg          |
| $f$       | rolling resistance coefficient                                            | 0.014       | –           |
| $i_i$     | Transmission ratio of the planetary gear to side gear                     | 1.44        | –           |
| $\sigma'_0$ | Equivalent stiffness coefficient of bristles                             | 315         | m$^{-1}$    |
| $\sigma'_1$ | Equivalent viscous damping coefficient of bristles                       | 1.1         | s/m$^{-1}$  |
| $\sigma'_2$ | Equivalent viscous damping coefficient between the tire and the ground  | 0.0005      | s/m$^{-1}$  |
relationship, indicating that when the torsional stiffness is too large, the rotational speed fluctuation increases sharply. As can be seen from the figure, a reasonable selection interval for torsional stiffness should be between 1 and 5 Nm/(°).

Figure 13 shows the change in the amplitude of the speed fluctuation of the transmission input shaft when the damping of the DMF is changed. When the damping increases, the speed fluctuation amplitude also increases gradually. The selection of the damping coefficient generally depends on the test and the empirical values, usually 0.03–0.15 Nms/(°).

4 | MULTIOBJECTIVE OPTIMIZATION

In this paper, the rotational speed fluctuation amplitude of the transmission input shaft \( n_a \) and the natural frequency of the vehicle \( f \) are selected as the optimization goals. The smaller the rotational speed fluctuation amplitude of the transmission input shaft, the more significant the damping effect of the DMF. The lower the natural frequency of the vehicle, the greater the avoidance of vehicle resonance. Therefore, the general mathematical expression of the DMF multiobjective optimization problem is

**FIGURE 6** Vehicle test equipment layout: (A) test vehicle, (B) hall rotational speed sensor at the primary flywheel ring gear, (C) hall rotational speed sensor at the first driving gear, (D) ROTEC torsional vibration tester, and (E) laptop.

**FIGURE 7** Speed fluctuations of the engine and the transmission input shaft: (A) 0–2 s and (B) 0–0.2 s.

**FIGURE 8** Spectral analysis of the engine and the transmission input shaft speed: (A) engine and (B) transmission input shaft.
where $X_i$ is the variable input of the $i$th design parameter.

The parameter configuration and performance of the DMF equipped with the test vehicle are shown in Table 2.

### 4.1 Optimal design based on the PSM

The PSM refers to the simulation calculation of the DMF finite set of parameters using the vehicle dynamics model, and a set of parameters with relatively good performance is selected from the calculation results as the optimal solution of the optimal design. The method should first determine the design parameter variables to be optimized and limit its variation range, and complete the parameter combination design based on design of experiment (DOE). The optimization result is considerably affected by the DOE samples and has a randomness, so it is called the PSM. Its processing steps are shown in Figure 14.

This paper uses the Latin hypercube to generate 100 sets of optimization schemes. The principle of the Latin hypercube is to divide the coordinate interval of each dimension into $m$ small intervals evenly in the $n$ dimensional space, and randomly select $m$ points from them to ensure that each dimension is sampled only once in each small interval. In this way, a Latin hypercube
design with \( n \) dimensional space and \( m \) samples is formed. The simulation results and the corresponding Pareto frontier points are shown in Figure 15. There are only four Pareto optimal solutions in the sample points, and the corresponding detailed data are shown in Table 3.

There are only four optimal solutions in this optimization design, and the selection is relatively simple. However, the number of Pareto optimal solutions is uncertain and has a large randomness, and a large number of Pareto optimal solutions may appear, so a quantitative index for the selection of the final optimization result is introduced. In this paper, the expectation value method (EVM) is selected to determine the final optimization scheme. Different weights are set for each optimization goal to obtain the final expected value. The formula is as follows:

\[
D_j = (d_{ij} \times d_{ij} \times \ldots \times d_{ij})^{\frac{1}{i}},
\]

(12)

where \( i \) is the number of optimization objectives, \( j \) is the number of optimization scheme, and \( d_{ij} \) is the normalized value.

\[
d_{ij} = \left( \frac{\text{high}_j - Y_{ij}}{\text{high}_j - \text{low}_j} \right)^m, \quad 0.1 \leq w_i \leq 10,
\]

(13)

where \( w_i \) is the weight of the \( i \)th optimization objective, \( Y_{ij} \) is the \( i \)th optimization result of the \( j \)th optimization objective, and \( \text{high}_j \) and \( \text{low}_j \) are the maximum and minimum values of the \( i \)th objective's optimization results.

For this optimization design, EVM is used. The optimization schemes considered are as follows: (1) only the rotational speed fluctuation amplitude is considered, and scheme 1 is selected, (2) only the natural frequency is considered and scheme 4 is selected, and (3) the two responses are given the same weight value and the optimization scheme is selected as 2. Since the natural frequencies of the four optimization schemes are all lower than the excitation frequency of idle speed, scheme 1 is selected as the optimization result only considering the amplitude of speed fluctuation. Compared with the DMF equipped with the test vehicle, the amplitude of speed fluctuation is reduced by 40.8%.
To summarize, it can be seen that the advantage of the PSM is that the optimization process is relatively simple, and the entire multiobjective optimization design process can be completed only by performing simulation calculations on a limited number of samples, but the disadvantages are also obvious. On the one hand, there is no guarantee that the optimal solution in the sample is the DMF optimal design point. On the other hand, there are relatively few non-dominated solutions to choose from, which may not fully satisfy the corresponding needs.

4.2 Optimal design based on MOPSO

PSO was proposed by Kennedy and Eberhart in 1995 based on the foraging behavior of birds. Bird flock foraging is essentially an optimal decision-making process, and PSM treats each bird as a particle with specific position and velocity information. In addition to recording the best position and current position that each particle has passed through, it also knows the best position the group has passed through at present. By adjusting the flight direction of the particles using the above information, the group will gradually be directed to the location of the food.

MOPSO is the application of PSO in multiobjective optimization, and its process steps are shown in Figure 16.

MOPSO is used for DMF multiobjective optimization. After 50, 100, 500, and 1000 iterations, the Pareto solution set is shown in Figure 17. It can be seen from the figure that the optimization results of MOPSO basically converge under different iteration times. With the increase of optimization times, the number of Pareto frontier points increases, but the uniformity is still not ideal.

Three special solutions are selected from the Pareto solution set of 1000 iterations by EVM, as shown in Table 4. Scheme 1 only considers the rotational speed fluctuation amplitude, scheme 2 only considers the natural frequency, and scheme 3 assigns the same weight to the two responses. Scheme 1 is selected as the optimization result only considering the amplitude of speed fluctuation. Compared with the DMF equipped with the test vehicle, the amplitude of speed fluctuation is reduced by 41%.

4.3 Optimal design based on the NSGA

The NSGA was proposed by Srinivas and Deb in 1994. In 2000, Deb et al. improved NSGA and obtained a nondominated sorting genetic algorithm based on an elite strategy (NSGA-II). By introducing the crowding distance, the complex calculation is optimized, the optimization

![Figure 16](image)

Start

Initialize particle swarm

Calculate the fitness value of each particle

External storage

Determine the individual best position

Determine the group best position

Update particle swarm

Calculate the fitness value of each particle

External storage

Update the individual and group best position

Satisfy the termination condition?

Y

End

FIGURE 16 Optimization process of MOPSO. MOPSO, multiobjective particle swarm optimization.

![Figure 17](image)

FIGURE 17 Optimization results under different iterations of MOPSO. MOPSO, multiobjective particle swarm optimization.

| Optimization scheme | $\mu$ | $k_i$ (Nm/rad) | $c$ (Nms/°) | $n_r$ (rpm) | $f$ (Hz) |
|---------------------|------|---------------|-------------|-------------|---------|
| 1                   | 0.84 | 1.31          | 0.03        | 11.80       | 6.41    |
| 2                   | 1.01 | 1.00          | 0.03        | 12.15       | 5.67    |
| 3                   | 0.84 | 1.00          | 0.03        | 11.92       | 5.71    |

TABLE 4 Optimization scheme of MOPSO

Abbreviation: MOPSO, multiobjective particle swarm optimization.
accuracy is improved, and the exploration performance is enhanced. The evolutionary calculation process is shown in Figure 18.

NSGA-II is used for DMF multiobjective optimization. After 50, 100, 500, and 1000 iterations of calculation, the Pareto solution set is shown in Figure 19. It can be seen from the figure that after 50 iterations of NSGA-II, the Pareto calculation results have poor convergence. After 100 iterations, the convergence is improved, but the number of Pareto frontier points is less, and the uniform distribution is poor. At 500 and 1000 iterations, the calculation results basically converge and the Pareto frontier points are well distributed. Therefore, 500 iterations represent the ideal iteration number for using NSGA II to optimize the design.

Three special solutions are selected from the Pareto solution set of 500 iterations, as shown in Table 5. They belong, respectively, to scheme 1, which only considers the amplitude of rotational speed fluctuation, scheme 2, which only considers the natural frequency, and scheme 3, which assigns the same weight to the two responses. Scheme 1 is selected as the optimization result only considering the amplitude of speed fluctuation. Compared with the DMF equipped with the test vehicle, the amplitude of speed fluctuation is reduced by 40.8%.

By using the above three methods to optimize the DMF, the amplitude of the speed fluctuation can reach more than 40% attenuation. However, the number of Pareto frontier points, convergence, and uniform distribution that can be obtained by different algorithms are not the same. The specific comparison results are shown in Table 6. Among them, NSGA-II with the high iteration number performs the best, and PSM is the least ideal.

![Figure 18](image)

**Figure 18** Optimization process of NSGA-II. NSGA-II, nondominated sorting genetic algorithm based on an elite strategy.

**Table 5** Optimization scheme of NSGA-II

| Optimization scheme | $\mu$ | $k_1$ (Nm/rad) | $c$ (Nms/rad) | $n_2$ (rpm) | $f$ (Hz) |
|---------------------|------|----------------|---------------|-------------|---------|
| 1                   | 0.87 | 1.25           | 0.03          | 11.80       | 6.25    |
| 2                   | 1.03 | 1.00           | 0.03          | 12.24       | 5.67    |
| 3                   | 0.88 | 1.03           | 0.03          | 11.92       | 5.76    |

Abbreviation: NSGA-II, nondominated sorting genetic algorithm based on an elite strategy.

**Table 6** Algorithm comparison

| Algorithm              | Pareto frontier points | Convergence | Uniform distribution |
|------------------------|------------------------|-------------|----------------------|
| PSM                    | Poor                   | Poor        | Poor                 |
| MOPSO (low iterations) | Poor                   | Very good   | Poor                 |
| MOPSO (high iterations)| Good                   | Very good   | Good                 |
| NSGA-II (low iterations)| Poor                  | Poor        | Poor                 |
| NSGA-II (high iterations)| Very good             | Very good   | Very good            |

Abbreviations: MOPSO, multiobjective particle swarm optimization; NSGA-II, nondominated sorting genetic algorithm based on an elite strategy; PSM, passive selection method.

**5 | CONCLUSIONS**

In this paper, the influence of the DMF dynamic parameters on its vibration reduction performance is analyzed, and the DMF multiobjective optimization design is studied. The following conclusions can be obtained:

1. The DMF dynamic parameters, including the moment of inertia ratio, torsional stiffness, and damping, have a significant influence on the vibration reduction effect. When the moment of inertia ratio increases, the DMF damping effect first increases and then decreases. When the moment of inertia ratio is 0.9, the vibration absorption capacity is the best. The DMF vibration absorption capacity decreases with the increase of torsional stiffness and damping.
Through the research of this paper, one can obtain the influence law of DMFs dynamic parameters on its damping effect, and can choose a suitable algorithm to complete the DMF parameter optimization design.

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CONFLICT OF INTEREST
The authors declare no conflict of interest.

DATA AVAILABILITY STATEMENT
The data that support the findings of this study are available from the corresponding author upon reasonable request.

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