A Multi-Point Iterative Analysis Method for Vibration Control of a Steering Wheel at Idle Speed

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ABSTRACT Steering wheel vibration control is of great significance to improve the ride comfort of vehicles. However, the coupling effect of the multi-source vibration transfer path makes it difficult to screen out the abnormal vibration transfer intervals. Therefore, a multi-point iterative analysis method (MIAM) is presented in this paper. In addition, this paper researches the characteristics of the main vibration transmission path of a truck and proposes a new vibration isolation structure. Considering the structural characteristics of the modal deformation of the car body, the installation orientation of the measuring point is further adjusted according to the area formed by modal deformation nodes, and a simplified vibration transfer path is presented. After weighing the vibration data of the sensor and considering the vibration transfer rate as the analysis index, the abnormal interval of vibration transmission on the main path was detected. The effectiveness of the improved structure is verified with the modal verification analysis of an extracted steering system model. In addition, the experimental results show that the improved measures reduce the total vibration of the steering wheel by 34% and offsets the peak corresponding frequency by 4.8%, showing the effectiveness of the proposed method and proving its advantages for other vibration control problems.

INDEX TERMS MIAM, extracted steering system model, new vibration isolation structure, simplified vibration transfer path, modal verification analysis.

I. INTRODUCTION

Recently, with the development of the logistics and transportation industry, commercial models have gradually become popular for market transportation. The idle stage is a common driving condition for commercial vehicles because the comfort evaluation mainly comes from the intuitive feelings of drivers, and its comfort index has attracted more attention from researchers. Therefore, more effectively controlling the jitter of an idle steering wheel is of great significance to improve the ride comfort of commercial vehicles. In recent decades, many researchers have proposed different research methods on the vibration control of steering wheels. For example, several studies have developed different innovations and verified the effectiveness of the control algorithm [1]–[3] and structural optimization [4]. At the same time, various design models [5] also have been used in vehicle body development and vibration reduction design. In addition, some researchers obtained the source and fault cause of abnormal vibration through vibration signal extraction and analysis [6] and sensory comfort study [7], [8]. Overall, many vibration control methods have been developed and analyzed based on the vibration transfer path, for example, by focusing on the excitation source in the link of vibration transmission, Zheng et al. [9] adopted the genetic algorithm method for the powertrain mounting system optimization. Zagorski et al. [10] researches the interaction of the chassis and steering system. In other aspects, based on the vibration transmission intermediate link and the response source, Lu et al. [11] analyzed the effect of multi-clearance joints.
on the steering mechanism, and Yin et al. [12] investigated the coupled ride and directional performance characteristics between the hydro-mechanical frame steering and hydro-pneumatic suspension systems. In addition, transfer path analysis (TPA) and operational transfer path analysis (OPTA) [13]–[15] are two main existed research techniques for vibration transmission path analysis, and these are widely used in the field of engineering vibration control. However, TPA is only suitable for studying the static transfer function and its analysis steps are tedious. OPTA cannot describe the modal frequency and modal mode of the mechanical system. In addition, although TPA and OPTA analysis technology can achieve accurate vibration fault analysis, because of the existing equipment, the limited test cost and analysis cycle make engineers more strongly demand a cheaper and more convenient analysis method. In this paper, to reduce the period and workload of measuring point arrangement and analysis, the experimental measuring points are arranged evenly according to the path of the system associated with the vibration transmission of the steering wheel.

By focusing on the identification of the main path of vibration transmission and the extraction of vibration data, this work proposes a new method (MIAM) concentrated on the analysis of vibration acceleration transfer, and the basic ideas of the proposed method are as follows: i) extract possible vibration fault areas by calculating the vibration acceleration transfer rates, showing the effects of vibration attenuation or amplification, along the transfer path, and ii) find the extracted possible vibration fault areas through continuous iteration, and the most possible fault area is further extracted by referring to the modal deformation node position. In addition, other main vibration sources of the truck, i.e., the engine and gearbox, are also considered. Then, by focusing on the overall vibration of the steering system and avoiding the vibration interference between the other related systems, the steering system model is extracted according to the constraint and connection relationship of the real vehicle and its accuracy is further verified. By applying the finite element analysis (FEA) method, modal analysis of the simulation model, and entity model of the extracted steering system model, the modal frequency of a certain order frequency of the whole vehicle is greatly reduced and the effectiveness of the method results show that the vibration level of the steering wheel as a feature of the final extracted vibration region, the experimental vibration transmission and the extraction of vibration data, this section is described in Section 3. In Section 4, an extracted steering system model is introduced, and after its accuracy is verified, the possible fault causes of the fault intervals are found by the hammering method. In addition, a structural optimization scheme is proposed, and with the verification of the real vehicle test, the results show the effectiveness of the implementation method and improvement measures. Conclusions are given in Section 5.

II. THE PRINCIPLE AND APPLICATION OF MIAM

In this section, MIAM is proposed to analyze the vibration transmission characteristics of vibration data, and a vibration transfer rate index is used as the analysis criterion to explain the relationship of the vibration transmission principle. Based on the transfer index and the modal deformation characteristics of the whole vehicle, the vibration path point is monitored in real time and the optimal arrangement of the sensor is adjusted. In addition, the correlation vibration transfer relationship and the characteristics of the steering system from measured accelerations are briefly described.

A. IDENTIFICATION OF VIBRATION PATH AND LAYOUT ADJUSTMENT OF MEASURING POINTS

The vibration of the steering wheel is driven by multiple sources of vibration at idle speed, such as ignition excitation of the engine and unbalanced torque of the gearbox, and these sources can be generally summarized as the vibration source, the vibration target, and the vibration transfer path [16]. Because of the non-stationary working conditions and background noise, collected vibration signals are usually complex, nonlinear, and expressed in a series of pulses [17]–[19]. The existing analysis techniques often focus on the analysis of the characteristics of the excitation signal of the vibration excitation source at a specific speed and need to be denoised, filtered, and checked at the process of processing, resulting in interference to the excitation analysis of the vibration source and an extension of the analysis cycle. However, for generalized vibration transfer paths with multiple vibration subpaths, fuzzy point data processing analysis is more helpful to avoid the signal processing of vibration source data. Based on the vibration measurement standard ISO 2613-1:1997(E), the vibration acceleration time domain signal, $a_i$, and the recorded acceleration time, $T$, was weighted using a certain frequency weighting function, $w(f)$, and the corresponding root-mean-square value of the weighted acceleration was calculated as $a_i$, based on the vibration signal in a particular $i$ direction, which is expressed as follows [20], [21]:

$$a_i = \left[ \frac{1}{T} \int_0^T a_i^2(t) dt \right]^{\frac{1}{2}}$$

(1)

Furthermore, for a selected measuring point on the transfer path, the root-mean-square value of the total weighted acceleration can be expressed as:

$$S : a_w = \left( a_x^2 + a_y^2 + a_z^2 \right)^{\frac{1}{2}}$$

(2)

where, $a_x$, $a_y$, and $a_z$ represent the time domain data of vibration acceleration data in the $x$, $y$, and $z$ directions, respectively. When a vehicle runs at idle speed, following the vibration transfer direction of the excitation source-transfer
path-response object, the steering wheel can be taken as the vibration responds object, and the structure such as engine and gearbox can be used as the excitation source; thus, the vibration transmission links to the steering wheel can be shown in Figure 1.

The analysis of steering transfer path in Figure 1 shows that the frame is a common transmission link in the process of vibration transmission, and the accuracy of the vibration data on which should be considered. Although the traditional circumferential symmetric measuring point arrangement is convenient for the layout, there may still be unreasonable arrangements that lead to errors in the collected vibration data.

For the i-order modal mode ($i \leq 5$), there are symmetric and antisymmetric modal shapes in symmetric structures, but symmetric measuring points placed on area formed by deformation nodes, as shown in Figure 2, make it difficult or even impossible to identify a certain order of modal mode data. Therefore, the selected data acquisition measuring points need to avoid the symmetry points of symmetric structures, so that more comprehensive and accurate information such as vibration modes can be obtained based on these vibration data.

In this study, to obtain significant vibration measurements, the region S formed by the deformation nodes of different main order modal modes represents the main area of the multi-measuring point placement on the frame, as shown in Figure 3. By using the measuring point data associated with the steering system on the frame as the equivalent starting point ($a_3$) and the steering wheel measuring point ($a_8$) as the response object, the data of each link in the vibration transmission are converted and extracted, as shown in Figure 4.

Under the vibration excitation of the engine and gearbox, the vibration acceleration of each link on the transmission path is further converted into the root-mean-square value of weighted acceleration and finally flows to the vibration of the research response target (steering wheel), which is convenient to facilitate further iterative positioning analysis. To express the effect of vibration transmission, the vibration isolation parameters (such as stiffness, mass, and damping value) between the vibration measuring point have different vibration suppression effects, and the degree of acceleration transfer between different nodes can be defined by the transfer rate $a_{p,a}$, which is expressed as follows [22]:

$$a_{p,a} = 1 - \frac{|a_p|}{|a_a|} \times 100\%$$  \hspace{1cm} (3)

where $a_p$ is the root-mean-square value of weighted acceleration at passive measuring points and $a_a$ is the root-mean-square value of weighted acceleration at active measuring points.

### B. BRIEF INTRODUCTION OF METHOD PRINCIPLE

The abnormal root-mean-square value of weighted acceleration and vibration transfer rate on the vibration path are the vital factors considered in the experiment. Based on these values, the area of the fault structure is shortened and determined by the criteria shown in Figure 5.
In the experiment, because the influence of external interference in a single path is small, measure points placed on the engine, gearbox, middle frame, front frame, direction engine, steering column, steering wheel, and steering wheel side are represented symbolically by \( a_1, a_2, a_3, a_4, a_5, a_6, a_7, \) and \( a_8 \), respectively. Because the vibration amplitude of reference measure points on the seat slide, engine, and gearbox meet the expected value, the characteristics of the method can be summarized as follows.

The principle and flows of the theorem shown in Figure 5 are as follows:

1) Referring to the direction of the vibration transfer path, the equivalent vibration excitation point \( a_3 \) is found as the first term and the vibration point \( a_8 \) of the steering wheel is seen as a target that needs to be studied.

2) Several main reasons contribute to the vibration variation of the main vibration path, as shown in Figure 6.

As shown in Figure 6, \( \Delta E_1 \) represents the resonance excitation energy by the cab, \( \Delta E_2 \) represents the comprehensive resonance excitation energy with steering rod substructure resonance, and \( \Delta E_3 \) represents the energy transmitted by the damping layer after vibration attenuation. In addition, the mass and the stiffness of the front panel structure of cab connected to the steering system are represented by \( m_c \) and \( k_c \), respectively; the mass and stiffness of the steering system substructure are represented by \( M_2 \) \( (M_2 = \sum_{i=1}^{N} m_i) \) and \( K_2 \) \( (K_2 = \sum_{i=1}^{N} k_i) \), respectively; and the value of damping and stiffness of the cab suspension damping system are expressed by \( c_1 \) and \( k_1 \), respectively. Based on the analysis of the vibration response characteristics of the damped system, the vibration dynamic equilibrium equation can be established as follows:

\[
[M] \{\ddot{u}(t)\} + [C] \{\dot{u}(t)\} + [K] \{u(t)\} = \{p(t)\}
\]

\[
[M] = \begin{bmatrix}
m_c & 0 \\
0 & M_2
\end{bmatrix}
\]

\[
[K] = \begin{bmatrix}
k_c & 0 & 0 \\
0 & K_2 & 0 \\
0 & 0 & k_1
\end{bmatrix}
\]

\[
[C] = [c_1]
\]
Then, by using the transformation matrix, \( \phi_i \), the modal analysis method can be used to transform the dynamic equation, and the suspension damping of the cab is represented by a linear combination of the stiffness and mass matrices. After establishing the converted mass matrix \( m_\& \), damping matrix \( c_\& \), stiffness matrix \( k_\& \), and dynamic load matrix \( P_\&(t) \) of the \( i \)-order structural mode, the dynamic equations can be further extracted and expressed as [23]–[26]:

\[
m_\& \ddot{y}_i(t) + c_\& \dot{y}_i(t) + k_\& y_i(t) = \{ P_\&(t) \}
\]

\[
m_\& = (\phi_i)^T \{ M \}(\phi_i), \quad c_\& = (\phi_i)^T \{ C \}(\phi_i) = 2\lambda_i m_\&, \quad k_\& = (\phi_i)^T \{ K \}(\phi_i) = w_i^2 m_\&, \quad \{ P_\&(t) \} = (\phi_i)^T \{ P(t) \}
\]

(5)

where \( \lambda_i \) is the modal damping ratio and \( \omega_i \) is the modal natural frequency of \( i \)-order. Furthermore, it is assumed that the vibration excitation energies defined by the engine and the gearbox are \( Q_2 \) and \( Q_1 \), respectively, and transmit the three points and vary as \( q_1 \), \( q_2 \), and \( q_3 \) in this transfer process. In addition, because the frame damping \( c_\text{frame} \) value is limited, the transfer energy consumption \( \Delta E_\text{frame} = \int_0^1 c_\text{frame} x_\text{frame}^2 \) (\( t \) is sampling time) can be ignored in the vibration reduction design process of the steering wheel. Therefore, the vibration effects of the steering wheel are mainly reflected as described in the following section.

C. VIBRATION TRANSFER INTERFERENCE OF CAB VIBRATION SELF-EXCITED RESONANCE TO STEERING WHEEL

Under the influence of the excitation frequency of the engine and gearbox, the modal steady-state energy of \( i \)-order produced by the cab structure is the following:

\[
E_i(t) = \frac{(\phi_i)^T \{ M \}\{ G \} \{ \ddot{v}_w \}^2}{m_\& w_i^2 (1 - \xi_i^2) + 2\lambda_i w_i \xi_i} \cdot (w_i^2 \cos^2(wt - \alpha_i) + w^2 \sin^2(wt - \alpha_i)),
\]

\[
\{ \ddot{v}_w \} = (\ddot{a}_x, \ddot{a}_y, \ddot{a}_z)^T, \quad w \in [w_E, w_G]
\]

\[
\xi_i = \frac{w}{w_i}, \quad \alpha_i = \arctg \left( \frac{2\lambda_i \xi_i}{1 - \xi_i^2} \right),
\]

(6)

where \( \{ G \} \) is the transformation matrix of spatial degree of freedom, \( \{ \ddot{v}_w \} \) is the excitation amplitude matrix of acceleration, \( w \) is the excitation frequency from engine or gearbox, and \( w_i \) is the \( i \)-order modal frequency of cab. When \( w \) is equal to \( w_i \), \( E_i(t) \) can produce a large increase of vibration energy (\( \Delta E_3 \)), greatly affecting the vibration degree of the steering wheel and increasing the vibration energy.

D. RESONANCE EFFECT OF STEERING ROD SUBSTRUCTURE

The steering rod system is a discrete system of multi-mass elements composed of multi-link structures. Therefore, the excitation frequency emitted by the engine can easily cause resonance excitation of discrete mass elements. For any mass unit, there is the following resonance possibility [27], [28]:

\[
x(t) = \frac{F}{m_i(w_i^2 - w_0^2)} \sin w_0 t = \frac{F}{(1 - \frac{w_0^2}{w_i^2})} \cdot \frac{1}{m_i w_i^2} \sin w_0 t
\]

\[
= X\delta \sin w_0 t
\]

\[
\delta = \frac{x_{\text{max}}}{x_{\text{min}}} = \frac{1}{1 - (\frac{w_0}{w_i})^2} = \frac{1}{1 - \beta^2}
\]

(7)

(8)

where the \( X \) is static displacement under dynamic load, \( \delta \) is the dynamic coefficient, and \( \beta \) is the external excitation frequency and natural frequency ratio. While the external frequency is coupled with the natural frequency of part, the value of \( \beta \) is close to 1, resulting in a vibration large amplification in the transmission characteristics and generating increased resonance energy (\( \Delta E_2 \)).

E. DAMPING CHARACTERISTICS OF DAMPING SYSTEM

The cab mounting system is the main buffer damping layer structure existing between the cab and the frame. Having a good front damping matching characteristic helps minimize the vibration energy of the frame; however, because the damping characteristics of the suspension spring are non-linear, large stress shock makes this damping characteristic tend to be unstable so the additional vibration transfer increment (\( \Delta E_1 \)) of the cab can be obtained. In the vibration fault analysis of the steering system, the vibration data analysis of the upper and lower fulcrums of the cab mount is often used as the generalized criterion to judge whether the suspension is qualified. Overall, when there is a resonance problem, the transfer rate is negative and the transmission effect can be qualitatively determined by the vibration transfer rate of the weighted accelerations \( a_i \) and \( a_j \) between the two adjacent measuring points. To obtain a more vibration reduction effect, the first case of the best acceleration vibration transfer rate is found and can be represented by \( p_1 \), as follows:

\[
P_1 : a_{i,j} = \begin{cases} a_{y,j} & a_{y,j} < a_{y,j} < 0 \\ a_{x,j} & a_{x,j} < a_{x,j} < 0 \\ a_{z,j} & a_{z,j} < a_{z,j} < 0 \\ \end{cases}
\]

and as its value decreases, the response vibration becomes more violent.

Second, when the excitation frequency is larger than the natural frequency, the value \( \beta \) is far larger than 1. By this time, the function of items 1 to 3 in equation (2) reduce the vibration energy under the damping effects on the corresponding structure between some adjacent parts and cause shifts in the amplitude and phase of the vibration characteristic curve shift, the second \( P_2 \) of the cases is expressed as follows.

\[
P_2 : a_{i,j} = \begin{cases} a_{y,j} & 0 < a_{y,j} < a_{y,j} \\ a_{x,j} & 0 < a_{x,j} < a_{x,j} \\ a_{z,j} & 0 < a_{z,j} < a_{z,j} \end{cases}
\]

(9)

(10)

In addition, when resonance and attenuation occur simultaneously in the front and rear intervals corresponding to a measuring point of the main vibration transfer path, we may synthesize the above characteristics and it is not difficult to find that the main select target can be set as the resonance.
FIGURE 7. Method implementation process.

intervals, and third $P_3$ of the cases can be represented as follows.

$$P_3 : a_{i,j} = \begin{cases} a_{3j}, & a_{3j} < 0, \ a_{j7} > 0 \\ a_{j7}, & a_{3j} > 0, \ a_{j7} < 0 \end{cases}$$

After locating the final fault area, a related optimization scheme focusing on the structure of the fault area is further proposed according to the procedure shown in Figure 7. Through multiple local optimization iterations on the main transfer path, the optimal fault area was found.

Furthermore, testing of the real car was performed, the root-mean-square value of vibration acceleration between each measuring points on the single-path model was obtained, and the iterative optimization between the measuring points was performed according to the criterion in Equation 11 to continuously obtain the optimal vibration transfer rate solution sets.

III. EXPERIMENTS IMPLEMENTATION AND ANALYSIS

By taking the problem of steering wheel jitter of a truck at idle speed as an example, the following experiments can be used to study and demonstrate the solutions for abnormal vibration of the steering wheel when the idle speed of the vehicle reaches approximately 666 rpm. Experimental requirements mainly include:

1. PCB(PCB Piezotronics Inc.) vibration sensors were installed on the measuring points separately, the experimental sampling frequency was set to 512 Hz, and the active accelerations of 10 sampling points in the X, Y, Z direction were measured.

2. The truck started when the engine rotate speed reached 600 rpm with a growth step of 50 rpm. Each test under a given speed lasted for 20 seconds. After the final test under the speed of 1000 rpm was finished, the truck was stably braked...
FIGURE 8. The experimental setup (a) as the 24 bit compact peripheral component interconnect signal data acquisition instrument and measuring point were separately placed on (b) steering fixed support, (c) steering gear, (d) the upper end of the rear suspension of the cab, (e) the upper end of the front suspension of the cab, (f) middle frame, (g) front frame, (h) gearbox, and (i) engine.

until it stops, and all the acceleration signals of the measuring points during the deceleration process were collected.

The vibration data of the experiment on the steering wheel was collected by using a PCB (PCB Piezotronics Inc.) acceleration vibration sensor and 24-bit compact peripheral component interconnect signal data collector. The actual vehicle layout measuring points are shown in Figure 8. When driving the vehicle at idle speed, test data from sensors placed on these measuring point were collected, as shown in Figure 9, and the installation orientation of them are shown in Table 1.

The test results shown in Figure 9 show that the root-mean-square values of acceleration in all directions and the total weighted acceleration of the engine, the frame, and the cab itself do not exceed the limit value (the limit values of the engine, frame, cab are 7.00 m/s², 7.00 m/s², and 4.00 m/s², respectively). Therefore, the vibration directly interference of cab and engine to the steering wheel can be ignored, but in the analysis, it is still impossible to avoid the interference and influence of random factors such as collision caused by gap interference and insufficient constraint of some sheet metal parts.

In addition, because of the change of the exciting frequency of the engine, the steering system part is easy to vibrate. Based on the results of the installed sensor shown in Table 1, through the experiment, the vibration contribution of the vibration transfer path and the exciting frequency value of each part are judged by using the steering wheel, i.e., $a_8$, as the vibration response point and the point $a_3$ as the equivalent vibration standing point. This study focuses on the identification of vibration main path and the extraction and analysis of vibration data and provides the basis and facility for fast tracing the source of the fault by finding the resonance interval, frequency range, and vibration transfer rate caused by the influence.

As shown in Figure 10, after considering the characteristic analysis of three-direction acceleration in the region with more concentrated modes and making adjustments for the measuring point, the installation positions of these mea-
suring points were readjusted to an offset area surrounded by deformed nodes to overcome the defects of the lack of modal shape. Through the experimental test, the vibration test data of each measuring point were obtained and are shown in Table 2.

Here, $a_w$ represents the weighted acceleration of the measured points on the vibration path of the steering wheel as the truck passes the flat horizontal ground.

As shown in Table 2, when the truck is moving at idle speed, the accelerations of engine and gearbox satisfy the
Based on each test point of the extracted single-path transfer chain, for ease of observation, the measured value (black font annotation) and the desired standard value (red font annotation) of each link are indicated in Figure 11. Furthermore, by taking the root-mean-square value of the vibration acceleration of the adjacent measuring points as the calculation basis, the transmission rate between the adjacent measuring points is obtained by using the existing equation for calculating the index of the vibration transfer rate.

By referring to the data in Table 3, the vibration data of the reference measuring points at the seat slide are lower than the standard value, indicating the effect is minor and is not a major factor in steering wheel jitter. Then, by extracting the abnormal vibration transfer rate and referring to the actual structure arrangement, the vibration of engine interferes with the vibration of the front frame or the middle frame. However, because the acceleration values of the direct transfer measurement points relate to the engine and gearbox sources meet the requirements of the standard values, the judgment of the final results has little effect on the determination of the final results. Referring to the search algorithm flow of vibration method, the large analysis range is gradually reduced from \( x \) to \( y \) and the final intervals are narrowed down from the steering fix support to the steering wheel.

### IV. STEERING SYSTEM DESIGN AND VERIFICATION

#### A. EXTRACTION OF STEERING SYSTEM MODEL

The existing experimental analysis shows that there is an abnormal vibration transfer interval in the steering system structure. However, because most of the steering system...
structures are rigid parts, the limited vibration isolation capacity makes the vibration control research mainly focus on the influence of the main order modal frequency (close to the natural frequency) distribution of the rigid body structure of the steering system. To avoid the interference of the external additional system, the steering system is further extracted and isolated according to the original constraint relationship, as shown in Figure 12.

The finite element model of the extracted steering system contained 278973 FE entity elements and 995735 element nodes, and its constraints mainly concluded as three types. First, the solder joint fixed connection was the main connection between the steering fixed support and the front enclosure of the cab, which was usually set in the circumferential bolt hole of the front enclosure and constrains the degree of freedom in all directions. Second, the constraints connection was performed at the bottom of the steering fixed support through circumferential constraint points, and its freedom in all directions was also constrained. Third, the hinge connection was the main connection mode between the opposite directions of the universal joint, and the circumferential direction of the hole was constrained by a rigid node in each hole.
of the universal joint, and the rigid nodes were connected by a rigid element. The rotational degree of freedom and axial degree of freedom parallel to the steering column are also released to ensure the relative rotation between the steering knuckles.

### B. MODAL TEST OF STEERING SYSTEM

The steering system is evenly arranged with a series of equivalent measuring points according to the transfer path to obtain accurate experimental modal shapes, and the positions of the measuring points are shown in Figure 13.

![Figure 12: Model of the steering system extraction (a) solid Model extracted from the steering system (b) simulation model of the extraction system.](image)

![Figure 13: Layout of the measuring points of the extracted steering system model.](image)

After the experimental measuring points of the PCB vibration acceleration sensor shown in Figure 13 were evenly arranged on the steering system, three points of the steering system were hammered by using the force hammer, and the intensity of hammering was controlled by the same experiment. The direction of the hammer is represented by local coordinates, as shown in Figure 14.

To avoid the influence of human factors, the hammering experiment adopted multiple grouping hammering. After a series of hammering experiments, more credible percussion data were obtained from the results, and the collected vibration data were filtered and processed. Then, the modal...
the structure. In fact, because the modal mass of the lower vibration mode is often related to the modal effective mass of vibration and can be expressed as follows:

\[
\alpha_j = \frac{\{\phi_{ij}\}^T [m] \{l\}^2}{\{\phi_{ij}\}^T [m] \{\phi_{ij}\} \sum_j m_j}
\]  (12)

where \(\alpha_j\) is the modal mass coefficient corresponding to the \(j\)th and \(l\)th, while \([m]\) is the mass matrix and \([\phi_{ij}]\) is the influence matrix, respectively. Correspondingly, \(R_j\) is set to the relative mode contribution factor for each mode, which is defined as the following:

\[
R_j = \frac{\alpha_j}{\alpha_{\text{max}}} \quad (13)
\]

\[
\alpha_{\text{max}} = \max(\alpha_1, \alpha_2, \ldots, \alpha_j) \quad (14)
\]

The equations show that the participation of the vibration mode is often related to the modal effective mass of the structure. In fact, because the modal mass of the lower order is larger, the total effective mass of the low order vibration mode accounts for a large proportion of the whole vibration mode. In general, the effective mass of the first six vibration modes is close to 90% of the actual mass of the structure. Therefore, in the analysis, the vibration modal analysis of the first six orders is enough to obtain more complete vibration modal information. However, because the multi-rigid structure of the steering wheel makes the natural frequency of the steering wheel lower (generally less than 30 Hz), the modal frequency of the sixth order structure has deviated far from the resonance frequency in the steering wheel vibration experiment, which can be ignored in the vibration modal test.

After preprocessing and solving the test data, based on the obtained modal geometric model, the vibration mode and frequency of the geometric model in the main order modes (order 1~5) of the experiment were further obtained. By comparing with the modal data solved by the simulation model, the results verify the fitting and accuracy between the solid model and the simulation model.

The comparison results of Figure 16 show that the modal shapes of the solid model and the simulation model are roughly consistent, and the corresponding modal frequency interval difference between them is less than 1 Hz; consequently, there is a good fit between the simulation model and the solid model. In addition, the test results also show that the second-order modal frequency of the steering system is close to the ignition frequency of the engine, easily causing resonance problems.

C. OPTIMIZATION MEASURES

Considering the influence of the stiffness and mass of the steering system on the frequency of the steering structure, under the constraint of the installation limit condition of the steering structure, the discrete reinforcement method is used to enhance the lateral stiffness on the basis of the original steering fixed support, and also convenient to realize the lightweight design of the structure, just as displayed in Figure 17.

As shown in Figure 17, the axial dimensions \(L_i\) (\(i = 1, 2, 3, 4\)) and \(R_i(i = 1, 2, 3, 4)\), mass \(M_j(j = 1, 2)\), and thickness \(S_j(j = 1, 2)\) of the steering fixed bearing were defined under the spatial limitation of the local coordinate system (\(o-xyz\)), respectively. In the modeling of the improved and original steering system, the steel material with elastic modulus (210 GPa), Poisson’s ratio (0.3), and density (7.9 mg/mm\(^3\)) was used. The division element adopts the mixed element type divided into nearly equal element sizes. In addition, by comparing the dynamic characteristics of the vibration response between the improved structure and the original steering bearing structure, the dynamic characteristics of the vibration response can be further analyzed, as shown in Figure 18.

The damping attenuation ability of the structure determines the loss of vibration energy. As shown in Figure 18 (a), the vibration damping force of the original structure was
mainly because of the bending tension of the two bending shapes of the steering fixed support. It is assumed that the critical points \(D_1\) and \(D_2\), along with \(D_3\) and \(D_4\) at both ends of the deflection structure, form the mutual extrusion pressure \((f_1, f_2, f_3, f_4)\) under the action of a certain vibratory extrusion pressure \((N_1\) and \(N_2)\). By using the effect of porosity distribution on the buckling resistance of the structure [34], the use of a porous fence structure, as shown in Figure 18(b), helps make the vibration stress \((f_{12}, f_{13}, f_{24}, f_{34})\) closer to the radial direction of the structural element compared to the original structure. By comparing the two different steering fixed support structures, the vibration characteristics of the two structures under vibration impact can be presented, as shown in Figure 18 (c). Under the assumption that the vibration compression forces on both ends of the steering bearing are \(N_1\) and \(N_2\), the vibration displacement produced by the joints \((D_1, D_2, D_3, D_4)\) of the original structure is \(M_1, M_2, M_3, M_4\), and the stiffness and damping between \(M_1\) and \(M_2\), along with the values between \(M_3\) and \(M_4\), can be expressed by \(K_{12}, C_{12}\) and \(K_{34}, C_{34}\), respectively. Similarly, the vibration displacement on fence shape structure nodes \(P_1, P_2, P_3, P_4, \ldots, P_{i-1}, P_{i-2}, P_i\) can be set as \((X_1, X_2, X_3, \ldots, X_{i-2}, X_{i-1}, X_i)\) and the stiffness and damping between these nodes can be marked as \((k_{12}, c_{12}), (k_{13}, c_{13}), \ldots, (k_{i-1,i}, c_{i-1,i}), (k_{i-2,i}, c_{i-2,i})\), respectively.

The analysis of the motion characteristics of the structure shows that the improved structure adopts the fence structure to increase the number of internal movable nodes; thus, the degree of freedom constraint of these joints also released.

As shown in Figure 19, compared with the steering support structure before and after the improvement, the angle \(\theta (\max \theta = 90^\circ)\) between the surface stress \(f_{i,j}\) and the transverse component \(f_{x}(i,j)\) of the structural elements of the original steering fixed support changes with the geometric distribution. In addition, the maximum deflection of
FIGURE 18. Comparison analysis included vibration impact stress changes between (a) original steering fixed support and (b) improved steering fixed support, and (c) dynamic structural characteristics between the original and improved steering fixed support.

FIGURE 19. Stress analysis of steering support structure elements before and after improvement.

the structural element (max $\theta = 90^\circ$) is the working condition of the maximum ultimate stress angle. Referring to the Bernoulli–Euler beam model [35], [36], the equilibrium equation of the beam can be set as:

$$M_x + Y_{xy} = -\left(EI + \mu Al^2\right) \frac{d^2w(x)}{dx^2}$$

$$\theta = \frac{dw(x)}{dx}, \quad \theta \in (0^\circ, 90^\circ)$$

Here, material constant, $\mu$, the resultant moment, $M_x$, and couple moment, $Y_{xy}$, Young’s modulus, $E$, and second moment of cross-sectional area, $I$, determine the vibration bending characteristics of the structure. In particular, $EI$ mainly affects the bending stiffness of the structure element, while $l$ is length parameter and represents the structural element’s attributes to affect the couple stress [37]. Because the material of the plates of steering fixed support is consistent, the total length $l$ and the area $A$ of the side faces of improved steering fixed support increase, also increasing $M_x$ and $Y_{xy}$. Therefore, it is will more possible to make the improved steering fixed support to produce a slight deformation ($X_1, X_2, X_3, \ldots, X_{i-2}, X_{i-1}, X_i$) when addressing lower amplitude vibration excitation ( $N_1$ and $N_2$), thus showing certain damping characteristics and avoiding the direct transmission of vibration impact so parts of the vibration energy transferred can be consumed internally. Furthermore, based on the kind of discrete damping and stiffness characteristics of the structure formed by the vibration of discrete nodes (i.e. $P_1$, $P_2$, $P_3$, $P_4$), we may constraint its relationship as follows:

$$C_{12} \approx C_{34} \geq \sum_{i=1}^{n} \sum_{j=1}^{n} c_{ij}$$

$$K_{12} \approx K_{34} \geq \sum_{i=1}^{n} \sum_{j=1}^{n} k_{ij}$$
FIGURE 20. The structure (a) is the steering system structure with original steering fixed support, (b) the steering system structure with improved steering fixed support.

FIGURE 21. Main order modes and frequencies of steering system structures with improved steering fixed supports: (a) First-order vibration mode (b) second-order vibration mode (c) third-order vibration mode (d) fourth-order vibration mode (e) fifth-order vibration mode.

where the \( n \) value is constrained by the spatial size of the steering bearing. Compared with the original structure, the degrees of freedom of the internal structure of the steering support increased, resulting in greater losses of vibration energy. In addition, the original fixed support opposite hole pressure is discretized, the flexible force embodied by the flexible deformation of the discrete stiffener plate improves the vibration damping of the stiffener plate, and its mass also has been further reduced, meeting the requirements of vibration reduction and lightweight design. Furthermore, the improved discrete stiffener design drives the overall frequency of the existing steering bearing to deviate from the original frequency, thus reducing the influence of the resonance effect. To save the cost of the experiment, because the fitting of the simulation model is verified, the improved fixed support is applied to the extracted steering model and compared with the original extracted steering system structure, as shown in Figure 20.

Comparing the simulation results of the improved model shown in Figure 21 with the mode of the original structure shows that the vibration mode and frequency of the improved steering system structure have changed, and the frequency changes before and after optimization are shown in Table 3.

The results in Table 3 show that the improved steering system structure has the greatest influence on the first-order frequency of the mode. To detect the effectiveness of the improved structure in the real vehicle test, the original steering structure and the improved structure, as shown in Figure 22, are installed into the real vehicle test. In the verification test, the test conditions were set as before, and the PCB vibration sensor measuring points were evenly arranged on the steering wheel, as Figure 23. The responding vibration data from them were collected to objectively and conveniently to evaluate the practicability and feasibility of the improvement.
The vibration test results show that the large side vibration amplitude of the steering wheel is more convenient and intuitive to analyze the influence of rectification effect, and the vibration sensor signal at measuring point 1 of the steering wheel is taken as an example. The frequency domain data of vibration in three directions were preprocessed and analyzed, and the results are shown in Figure 24.

Figure 24a & 24b show that the effect of improving the steering system drives the frequency value corresponding to the peak value of the original spectrum signal to change from 22.99 Hz to 21.88 Hz. In the resonance frequency range, the vibration acceleration in the three directions of the sensor also produces a non-degree of attenuation, of which the attenuation degree in the y-direction is the most obvious. To evaluate the overall effect of the improvement measures on the vibration of the steering wheel, the vibration signal of the sensor at measuring point 1 of the steering wheel in three directions is the weighted average, and the relationship...
between the root-mean-square value of the total weighted acceleration and the speed of the engine (excitation source) was obtained (refer to Equation 3), as shown in Figure 25.

The analysis results show that the improved steering system structure reduces the peak value of the total weighted acceleration from 13.5 m/s² to 7.89 m/s², the resonance effect is obviously weakened, and the speed of the engine corresponding to the resonance is also offset from 666.36 rpm to 677.60 rpm, which deviated from the original initial speed. Because of the subjective evaluation of the driver, the improvement measures make the vibration degree of the steering wheel address a satisfactory state, in line with the expected vibration reduction design requirements, and indirectly verified the effectiveness of the methods.

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**V. CONCLUSIONS**

This paper demonstrated a new MIAM, combined with the FEA, modal analysis, and the vibration transfer characteristics analysis, effectively solving the problem of steering wheel jitter at idle speed. Based on the established model of simplified vibration transfer path, the abnormal area of vibration transfer is found based on the analysis of the proposed allocated algorithm and a new structure proves its effectiveness by improving the ability of vibration isolation of steering wheel.

One of the advantages of the proposed method lies in avoiding the defects that the existing transfer path technology cannot describe the modal frequency and modal mode of the mechanical system. In this method, the global vibration transfer function analysis of the whole vehicle model shows that the whole transfer path is simplified to a main vibration transfer path according to the vibration transfer target and vibration source. The modal deformation nodes of the frame determined the distribution area of experimental measuring points, and the experiment at idle speed is also performed by continuously adjusting the speed. Through iterative searching for global optimal fault area by applying the proposed algorithm based on the collected data of each sensor, the most appropriate fault area on the vibration transfer path is found. Then, based on an extracted steering model, the validity of the proposed method, along with the proposed structure improvement, is verified by the modal comparative analysis between the original steering structure and the improved structure. The results show that the vibration amplitude of the steering wheel is greatly reduced, meeting the design requirements of vibration reduction, and provided a reference for other similar engineering problems.

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