INTRODUCTION

With the rapid development of the automotive industry, great emphasis has been put on auto energy consumption. However, owing to the increasing scarcity of resources in response to the urgent need of energy conservation and emission reduction, an important concern is aroused in terms of how to reduce energy consumption and realize energy harvesting.1-4 Automobile energy harvesting, a valuable topic, can facilitate the full use of waste energy and reduce carbon emissions for sustainable development. In particular, as the new-energy vehicles increase, it plays a rather significant role in improving the endurance mileage and extending the working hours of batteries. In this regard, it is indispensable to probe the auto energy harvesting systems.5,6

Existing researches mostly focus on the energy-regenerative suspension mechanism which aims to harvest vibration energy and optimize damping characteristics of suspension.7,8 So far, no mature product has been put into actual use due to the complex structure and its impact on the mechanical property of vehicles.9-11 In addition, together with other methods, slow progress of vibration energy harvesting is achieved as a result the difficulty in collecting vibration energy as well as the low conversion efficiency.

A large number of studies have been conducted on vehicle vibration energy harvesting, which covers simulation, modeling, suspension design, etc. Ataei M et al., based on a two-degree of freedom 1/4 vehicle model, built a hybrid electromagnetic suspension system model and further evaluated its performance.12 Zuo et al. analyzed the recovery
power of vehicle suspension in view of the same evaluation standards, through which the simulation and experimental results were obtained. A detailed analysis was carried out by Xiao et al. of a two-degree of freedom piezoelectric vibration energy harvesting model, thereby theoretically improving the efficiency of piezoelectric vibration energy harvesting. Nevertheless, there was a lack of researches on external motivators. In terms of road identification, Reference 15-19 has provided a ground for relevant researches.

In an attempt to study the unit which harvests vibration energy by piezoelectric conversion, Li Y et al. used the energy-regenerative mechanism of gears and piezoelectric cantilevers to harvest vibration energy and discussed the filtering issue involved in vibration energy harvesting, considering that reasonable gear pitch may help to improve the recovery power of the energy-regenerative mechanism. However, the effect of other external factors like road and vehicle speed on vibration energy harvesting efficiency has not been expounded. With regard to the researches initiated by Mouapi A et al., a piezoelectric cantilever beam was placed under the front passenger seat to generate electrical energy, but available vibration displacement was rather limited. A method was put forward by Viet NV et al. who used rotating machinery structure and piezoelectric materials for energy recovery but without fully considering the difference caused by gear pitch. An energy harvesting system using piezoelectric cantilever was designed, in which the speed bump could be triggered twice in the process of shock compression. Accordingly, the size of energy harvested could be affected by the length of vibration time. Zheng Q et al. also attempted to recover vibration energy with the use of piezoelectric cantilever, but since the object was limited to microelectromechanical system, major attention was not paid to the approaches to improve the harvesting efficiency. Xie and Wang took into full account the effect of road excitation on the acquisition of piezoelectric cantilever beam energy based on a simulation, although the effect of vehicle speed was not mainly elaborated.

Our previous research in Ref. 20 was focused on the virtual displacements filtering and power calculation method. Though indications were made in a value-taking method for gear pitch, the research was still carried out under a fixed vehicle speed and road condition. Following the route of the previous study, the present paper explored the relationship between vibration displacement and road tolerance grade and analyzed the effect of road tolerance grade and vehicle speed on the average power. It was found that a small gear pitch was suitable for good road conditions but limited to harvest power for bad road. Relatively, a big gear pitch harvested power hardly for good road but effectively for bad road. Thus, a piezoelectric vibration energy-regenerative device was designed with the double-gear excitation structure, which allows automatic-switch under comprehensive driving conditions, and the reasonable gear pitch and switching principle were determined. In this way, the problem concerning the maximization of vibration energy harvesting under comprehensive driving conditions could be settled.

In Section 2, the design of the energy-regenerative device and expected system volume were presented. Section 3 deduced and processed the identification of the road tolerance grade. In Section 4, a value-taking method for gear pitch and a switching principle for the double gear were proposed by analyzing the relationship between gear pitch, road tolerances, vehicle speeds and the recovery energy. Regarding different road grades, the size of energies fed back through the UDDS (Urban Dynamometer Driving Schedule) driving condition test was calculated in Section 5, with the verification of the effectiveness of the double-gear excitation mechanism. Conclusions were drawn in Section 6 based on the analyses throughout the full text.

2 | DEVICE DESIGN AND SYSTEM VOLUME

In our previous research in Ref. 20 great emphasis was put on the design of excitation form from the angle of unit, without the presentation of system volume design in practical application. In this section, this paper gave a solution in volume design and installation from the perspective of system.

For the energy-regenerative mechanism, piezoelectric bending elements were designed into cantilever beams, and the change in the relative displacement between suspension and vehicle body in the process of driving served as the excitation source which enabled the piezoelectric device to generate electric power. Figure 1 shows a model of the energy-regenerative mechanism of piezoelectric bending elements. Due to the material limitations for piezoelectric bending elements in the form of cantilever beam, only a few millimeters of the maximum displacement were allowable at the free end, while the change in the relative displacement between suspension and vehicle body in the process of vibration was far beyond the
value. Therefore, a linear displacement was converted into the reciprocating circular motion in this design, and a gear mechanism was added into the circular motion which divided the whole large displacement into small sections, so as to achieve multiple excitations within the scope of allowed maximum displacement for piezoelectric bending elements. In consideration of the output power and the space utilization efficiency of mechanical structure, the energy-regenerative mechanism of piezoelectric bending elements in the form of cantilever beam was designed into a cascading group which was split into several subgroups on the gear circumference. Moreover, light thin rods were used to make a connection between pieces of the cascading group in order to eliminate the phase difference and avoid the offset of voltage between the pieces.

Under the effect of multiple factors such as comprehensive conditions in the process of vehicle running, it was difficult to ensure high recovery efficiency with the single gear. Therefore, in the design of the excitement gear mechanism of piezoelectric bending elements, the switchable double-gear structure was applied, and the distinguished gear pitch was available for the two gears, so as to apply to different harvesting situations along with the adjustment of the excitement gear under the actual driving conditions and the optimization of the harvesting efficiency.

With regard to the piezoelectric energy-regenerative device structure, several groups of multilayer piezoelectric cantilever structure were installed at the vehicle bottom and arranged into a type surrounding the gear circumference. A double-gear excitation mechanism was fixed on an axle which could move axially, and the gear and the rack were used to transmit the displacement between suspension and vehicle body into the axle, so as to be converted into circular motion. At one end of the axle, the excitement gear could be driven by the driving mechanism to produce a small axial displacement and achieve the switch between excitement gears. The structure design of energy harvesting mechanism was shown in Figure 2.

According to the mentality of designing the actual size, 12 pieces of piezoelectric bending elements could be concatenated as a group. 4 groups could be set on a fixed bracket. And about 18 brackets could be distributed on the transmission shaft. A total of 864 pieces of piezoelectric bending elements could be arranged in a quarter vehicle. Considering the research core of this paper, the double crystal elements were chosen from a existing product, which has been proved suitable with its proper first and third eigenfrequency for the application scene in our previous research. The element character and system volume were shown in Table 1.

### 3 | ROAD TOLERANCE IDENTIFICATION

The vibration energy harvesting power of vehicle could be affected by the road tolerance grade. A most direct sense was that the worse motion the vehicle was in, the more violent vibration occurred. According to ISO 8606:1995(E) Standard, the road tolerance grades were described in Table 2 and Figure 3. When a vehicle ran on the road with different tolerance grades, a large difference in the change of the relative displacement between suspension and vehicle body would be produced. The virtual displacement filtering (VDF), which we have expounded in Ref. 20 would lead the vibration displacement into different results. Accordingly, the harvesting power would vary significantly when the piezoelectric blending element was excited to feed back electric energy. For example, Class A road led to continuous small linear displacements which would be filtered with the big gear pitch. A small gear pitch was recommended to harvest power. Relatively, a big gear pitch was recommended for Class D road to harvest power effectively. To improve the harvesting efficiency and

### Table 1 Parameters of a piezoelectric bending element and system volume

| Parameter                                           | Value          |
|-----------------------------------------------------|----------------|
| Size of piezoelectric ceramic (mm)                  | 60 × 31 × 0.2  |
| Size of base plate (mm)                             | 80 × 33 × 0.2  |
| Quality factor Qm                                   | 70             |
| Electromechanical coupling factor Kp                | 0.65           |
| Piezoelectric constant D_{31} 10^{-12} (C/N)        | -186           |
| Piezoelectric constant D_{33} 10^{-12} (C/N)        | 670            |
| Piezoelectric constant D_{15} 10^{-12} (C/N)        | 660            |
| Dielectric constant ε_{11}/ε₀                        | 3130           |
| Dielectric constant ε_{33}/ε₀                        | 3400           |
| System volume (piece)                               | 864            |
| Load (Ω)                                            | 330            |
rationally determine the excitement gear, it is necessary to identify the tolerance grade of the current road in the process of vehicle running and determine the change trend first.

For the purpose of road identification, 1/4 car body was taken as the object of study. Sensor KTC-75 mm was installed between the vehicle bottom and suspension. In the process of vehicle running, the measured voltage signal was converted into the relative displacement $y(t)$ between suspension and vehicle body. Given the road change $q(t)$, the model equation would be as follows:

\[
\begin{align*}
    m_2\ddot{y}_{2(0)} + C(y_{2(0)} - \dot{y}_{1(0)}) + k_1(y_{2(0)} - y_{1(0)}) &= 0 \\
    m_1\ddot{y}_{1(0)} + C(y_{1(0)} - \dot{y}_{2(0)}) + k_2(y_{1(0)} - y_{2(0)}) + k_1(y_{1(0)} - q_{(0)}) &= 0 \\
    y(0) &= y_{2(0)} - y_{1(0)}
\end{align*}
\]

(1)

For Laplace transformation of the model equation, the effect of the initial state should be considered in practice, from which we may obtain the following:

\[
\begin{align*}
    m_2s^2Y_{2(0)} - m_2sY_{2(0)} + Cs(Y_{2(0)} - Y_{1(0)}) - C(y_{2(0)} - y_{1(0)}) + k_2(Y_{2(0)} - Y_{1(0)}) &= 0 \\
    m_1s^2Y_{1(0)} - m_1sY_{1(0)} + Cs(Y_{1(0)} - Y_{2(0)}) - C(y_{1(0)} - y_{2(0)}) + k_2(Y_{1(0)} - Y_{2(0)}) + k_1(Y_{1(0)} - Q_{(s)}) &= 0 \\
    Y_{(s)} &= Y_{2(0)} - Y_{1(0)}
\end{align*}
\]

(2)

(3)

(4)

Using Equation 2 minus Equation 3, and substituting Equation 4, then it can be simplified into:

\[Q_{(s)} = R_{(s)}Y_{(s)} + W_{(s)}\]

(5)

Equation showed the relationship between the displacement of the vibration and the road tolerance. Where

\[R_{(s)} = \frac{m_1m_2s^4 + C(m_1 + m_2)s^3 + (k_1m_2 + k_2m_1 + k_2m_2)s^2}{-k_1m_2s^2} + \frac{Ck_1s + k_1k_2}{-k_1m_2s^2}\]

(6)

\[W_{(s)} = \frac{m_1m_2y_{(0)s} + C(m_1 + m_2)y_{(0)s}^2 + k_1m_2y_{2(0)s} + Ck_1y_{(0)}}{k_1m_2s^2}\]

(7)

where $y_{1(0)}$, $y_{2(0)}$, and $y_{(0)}$ stands for the initial states (Table 3).

Based on the observation of $R_{(s)}$ in Equation 6 and $W_{(s)}$ in Equation 7, it could be found that under the effect of the initial state, a constant term was added in the relationship between $q_{(0)}$ and $y_{(s)}$ during the transformation of time domain integral, which would result in a baseline drift and a disturbance magnitude far higher than $q_{(0)}$. However, the effect of the constant term, through which the tolerance grade of the current road could be reflected by the filtered $q_{(s)}$ signal clearly.

To verify the effectiveness of the road identification method, a four-section road tolerance was set on a roughly 12 km road. The vibration displacement measured in the process of continuous driving was shown in Figure 4, under the driving condition of UDDS.

In the light of the road identification in Group 5 and high-pass filtering of the waveform, the value $q_{(s)}$ was obtained, as shown in Figure 5.

According to the model identification process, no direct relationship was detected between the road change $q_{(t)}$ and the vehicle speed based on the relative displacement $y_{(t)}$ between vehicle body and suspension. However, when the vehicle speed reaches zero, the relative displacement would also be zero, causing an error in the process of identification. In this case, the road tolerance identified within the period of stop would also approach zero. Nonetheless, it failed to truly reflect the road condition, which made it necessary to change the result of road identification based on the abscissa axis of time into the one relying on the path length. Additionally, as for the existing road construction standard, since continuous change would not be witnessed by the tolerance grade of a section of road within a short distance, the whole road could be divided into several sections at a certain length unit by taking into account the error in data collection and calculation, with the tolerance mean of each section representing this section of road. As a result, the identification error caused by accidental factors could be effectively lowered. After the abscissa axis was converted from time to path length and the tolerance value was taken per 100 m, the identification result of the road tolerance could be obtained, as shown in Figure 6.

According to the road identification result in Figure 6, a 12 km road could be basically divided into four sections, including 0-3 km section, 3-6 km section, 6-9 km section, and 9-12 km section where the tolerance was around 10 mm, 3 mm, 20 mm, and 5 mm respectively, each corresponding to Class C, Class A, Class D, and Class B roads. The method

### Table 2: Surface roughness conditions based on ISO 8606:1995(E)

| Road class | A | B | C | D | E |
|------------|---|---|---|---|---|
| Roughness coefficient $S_d(f_0)$ (m$^2$/cycle) $\times 10^{-6}$ | 6 | 16 | 64 | 256 | 1024 |
has been proved to be effective in the identification of road tolerance.

When a vehicle runs on the road with different tolerance grades, a large difference could be observed in the change of the relative displacement between suspension and vehicle body. In relation to the energy-regenerative mechanism herein, the relative displacement presented a reciprocating change, and the gear was driven to rotate forward and reversely in the case of a shift to circular motion. Due to the existence of virtual displacement, judgment should be made on identifying which gear was more suitable for the energy-regenerative mechanism to excite piezoelectric blending elements and obtain more energy for the road with the current tolerance grade. Therefore, it was considered necessary to identify the road tolerance grade in the energy-regenerative mechanism.

On the basis of identifying the road tolerance grade, the average power of vibration energy harvesting could be calculated for the sections of different road tolerances, so as to obtain the effect of the road tolerance grade on the average power and to calculate the maximum average power with different gear pitches.

**TABLE 3** Parameters of a quarter vehicle in road identification

| m_1 (kg) | m_2 (kg) | k_1 (N/mm) | k_2 (N/mm) | C (N/mm/s) |
|----------|----------|------------|------------|------------|
| 35.5     | 317.5    | 240        | 30         | 1.91       |

**FIGURE 3** The examples of the road surface roughness conditions for five classes calculated according to ISO 8606:1995(E)

**FIGURE 4** Vehicle vibration displacement y(t) in UDDS cycle

**FIGURE 5** Identification of road q(t) in UDDS cycle
4 | MAXIMIZATION OF THE HARVESTED POWER WITH DOUBLE GEAR

In our previous research, the determination of the gear pitch was carried out under a certain road tolerance grade and at a constant speed. On account of the momentarily changing actual driving conditions, this section employed a value-taking method and switching principle for the gear pitch of the double gear, which contributed to obtaining the maximum harvested power.

4.1 | Power calculation

To obtain vibration energy harvesting power, consideration should be given to the effect of vehicle speed and gear pitch, in addition to the tolerance grade of the road where the vehicle ran. Under the combined effect of the vehicle speed and the road tolerance grade, an adequate gear pitch could be determined through the comparison of the change trend in the harvesting power obtained based on different gear pitches.

A light thin rod was used to connect piezoelectric bending elements in the piezoelectric energy-regenerative mechanism, between which there was no phase difference. Therefore, analyses could focus on a single piezoelectric bending element before extending to the whole energy-regenerative mechanism.

In view of the above conditions, an analysis was made on the state of piezoelectric blending elements in the process of single excitation: under the action of gear pitch \( d \), the excitation end experienced a shape change of displacement \( d \), and after the gear was separated from the excitation end, the damped vibration should be performed on the element, with the damping factor of \( \beta_1 \). However, at the moment of separation, the next gear had moved to the equilibrium position of piezoelectric blending elements. When the element returned to the equilibrium position, a complete \( \beta_1 \) damped vibration would not be achieved, but \( \beta_2 \) damped vibration would be performed around the equilibrium position. After such a vibration lasted for \( T_1 \), a second peak of \( \beta_1 \) damped vibration would appear to excite the piezoelectric blending elements to re-perform \( \beta_2 \) damped vibration, and so on. The waveform of the voltage generated by the element was shown in Figure 7.

The average excitement time interval between two adjacent teeth could be calculated by the VDF and defined as the excitation time \( t_d \). At the vehicle speed of 20 km/h, \( t_d \) was calculated with the VDF at different gear pitches, as listed in Table 4.

The method of time integration was adopted to calculate the power based on voltage signal. According to the optimized algorithm for the average power of piezoelectric harvesting (OAAP) which we have proved in Ref. 20 we may obtain the following:

\[
\begin{align*}
\text{TABLE 4} & \quad \text{The excitation time } t_d \text{ of different gear pitches using the VDF} \\
\hline
\text{Filtering distance, } d \text{ (mm)} & \quad \text{Excitation speed, } v \text{ (mm/s)} & \quad \text{Excitation time interval } t_d \text{ (ms)} \\
2 & \quad 62.633 & \quad 31.932 \\
3 & \quad 57.361 & \quad 52.300 \\
4 & \quad 51.792 & \quad 77.232 \\
5 & \quad 49.420 & \quad 101.175 \\
6 & \quad 46.539 & \quad 128.923 \\
7 & \quad 43.971 & \quad 159.195 \\
8 & \quad 41.845 & \quad 191.182 \\
9 & \quad 37.919 & \quad 237.347 \\
10 & \quad 33.809 & \quad 295.776 \\
\end{align*}
\]

FIGURE 6 Identification of the road tolerance in UDDS cycle

FIGURE 7 Voltage signal of piezoelectric bending element excited by gear

\[
\begin{align*}
\text{If } t_d < T_1 & : \\
\quad p = \int_0^{t_d} \frac{U_0 e^{-\beta_1 t} \cos \omega_2 t}{R} dt \\
& \quad + \int_{T_1}^{T_2} \frac{U_0 e^{-\beta_1 t} \cos \omega_2 t}{R} dt \\
& \quad + \int_{T_2}^{T_1} \frac{U_0 e^{-\beta_1 t} \cos \omega_2 t}{R} dt \\
\text{If } t_d > T_1 & :\\
\quad p = \int_0^{t_d} \frac{U_0 e^{-\beta_1 t} \cos \omega_2 t}{R} dt \\
& \quad + \int_{T_1}^{T_2} \frac{U_0 e^{-\beta_1 t} \cos \omega_2 t}{R} dt \\
\end{align*}
\]
where $\beta_1$ and $\beta_2$ are damping factors; $T_1$ and $T_2$ mean the period of $\beta_1$ damped vibration and the period of $\beta_2$ damped vibration, respectively; $\omega_2 = 2\pi/T_2$ is the angular frequency; $U_n$ is the starting voltage of each $\beta_2$ damped vibration; $U_0$ means the initial voltage.

According to the voltage signal generated from piezoelectric blending elements, $U_0$ would appear when the element returned to the equilibrium position by neglecting the gear in the stationary state. At this time, the movement speed reached its maximum at the free end, and the corresponding value was only associated with the gear pitch. $t_d$ refers to the time of excitation between two gears. Due to the addition of VDF, it became associated with the gear pitch. The faster the speed of a vehicle running on a road was, the more bumps there would be within a unit time, and the faster the rotation speed of the excitement gear would become. That is why $t_d$ is also related to vehicle speed. $\beta_1$, $\beta_2$, $T_1$, $T_2$ and $f_1, f_2$ are dependent on the characteristics and size of piezoelectric blending element material. The parameters of piezoelectric power adopted herein are listed in Table 5.

Through the above analyses, the VDF and the corresponding OAAP were adopted to comprehensively investigate the relationship between the average power of vibration energy recovery and gear pitch and vehicle speed. Thus, the average recovery power was figured out under each road tolerance grade and at different vehicle speeds as well as gear pitches, as shown in Figure 8, which demonstrated the power of 15 pieces of piezoelectric bending element.

According to Figure 8A, when a vehicle ran on the road of Class A, the vibration energy recovery power would experience an increase before decreasing with the enlargement of the gear pitch, and it almost reached the peak with the corresponding gear pitch of 3 mm. With the increase of vehicle speed, a general uptrend with a slight fluctuation was displayed for the power, which was evidenced by a fast increase at the position of $v = 60$ and $d = 2$.
close to the pitch of 3 mm and a slow increase at the position far away from 3 mm. But the peaks at some speed were gained by different pitches as shown by the arrows. According to Figure 8B, when a vehicle ran on the road of Class B, the vibration energy harvesting power first increased and then decreased with the enlargement of the gear pitch, and it almost reached the peak with the corresponding pitch of 5 mm. With the increase of vehicle speed, the power would witness a general up-trend with a slight fluctuation, which was evidenced by a fast increase at the position close to the pitch of 5 mm and a slow increase at the position far away from 5 mm. Similarly, different pitches could be resulted by different speed. Synthesizing the analysis of Class A and B road, it could be merged to deal by considering the contiguous power and the complexity of the device.

As shown in Figure 8C,D, when a vehicle ran on the road of Class C, D or, below, the maximum deformation amount would be limited to 10 mm at the free end of piezoelectric blending elements, and the maximum gear pitch would be 10 mm, in the meanwhile, which was no higher than the road tolerance. In such case, the harvesting power would increase with gear pitch and vehicle speed, and the reasonable gear pitch would be 10 mm.

Accordingly, a general uptrend of the average power could be observed with the increase of vehicle speed. At the same time, when the road tolerance was within the maximum gear pitch of 10 mm, the average power of different road tolerance grades would equal the larger value when the value of gear pitch was taken at a position close to the road tolerance. When the road tolerance exceeded 10 mm, the average power would reach the peak at the maximum gear pitch limit of 10 mm. Although the more gears used the higher power recovered, it was uneconomical and complicated for the device. In this scene, double gear was chosen to equilibrate. Gear A was used for Class A and B road according to the similar situations. Gear B was used for Class C, D, or worse road.

### 4.2 Value-taking method and switching principle

On the basis of the average harvesting power for each road tolerance grade and different vehicle speeds and gear pitch in Figure 8, an investigation was conducted into the method of double-gear pitch selection, so as to maximize the harvesting efficiency under comprehensive driving conditions.

1. Definitions were given to gear A and gear B, with the gear pitch corresponding to $d_A$ and $d_B$, respectively, and $d_A$ and $d_B$ would gradually increase in turn;
2. The average harvesting power was obtained according to the vehicle speed and road tolerance identified;
3. The time integral operation was performed with the average harvesting power, with an aim to figure out the harvested energy on a cumulative basis;
4. A diagram of the relationship between the gear pitch and the energy was drawn to obtain reasonable values of $d_A$ and $d_B$.

Through the above analyses, 3 mm and 10 mm were taken as the gear pitch of the two gears in the switchable double-gear structure of the piezoelectric harvesting system of vehicle vibration energy designed herein, which helped to optimize the power harvesting efficiency and could be proved in the following result section. The switching principle between the two gears is presented as follows:

1. Identify the road according to the signal gathered by the sensor;
2. Average the tolerance grades identified for the road on which a vehicle runs for 500 m and take the average as the current road tolerance;
3. When the gear pitch is 3 mm, the road tolerance is greater than 7 mm, with a switch to the gear pitch of 10 mm; otherwise,
no switch is made; in the case of the gear pitch with 10 mm, the road tolerance is smaller than 6 mm, switching to the gear pitch of 3 mm; otherwise, no switch is made.

Such principle may effectively solve the problem concerning the harvesting efficiency of vibration energy during the vehicle running on the road with each grade.

5 | RESULTS AND DISCUSSIONS

5.1 | Simulation results

By means of simulation, a segment of road of Class A and Class B was used to test and calculate the harvested energy of vehicle vibration. The vehicle ran in the UDDS cycle, and a diagram was drawn concerning the relationship between the gear pitch and the harvested energy therefore according to the double-gear pitch selection method. The energy in Figure 9 stood for the amount which recovered by a single piezoelectric bending element and dissipated on the load of 2 kΩ during the whole process. In the simulation, the energy was obtained by integrating instantaneous power during the UDDS cycle. As Figure 9A indicated, any value concerning gear B has no effect on the size of energy harvested, showing that only gear A is involved in excitation, while gear B is in an idle state during the vehicle running. In such case, with $d_A$ of 3 mm, the vibration energy harvested reaches its maximum when the vehicle runs on such a section of road. The test and calculation was repeated on another section of the road of Class C and Class D. It was found that the vehicle ran in the UDDS cycle, so the relationship between the gear pitch and the harvested energy is shown in Figure 9B. On this basis, it can be concluded that any value concerning gear A has no effect on the size of energy harvested. Therefore, it has been indicated that only gear B is involved in excitation, while gear A is in an idle state in the process of vehicle running. In such case, with $d_B$ of 10 mm, the vibration energy harvested reaches its maximum when the vehicle runs on such a section of road.

According to these simulations, determination was made that the value-taking method of the double gear was reliable, and it was satisfied with the usage of various tolerance road when $d_A = 3$ mm and $d_B = 10$ mm.
5.2 Bench testing results

A bench testing platform was built according to the model design of double-gear excitation vibration energy harvesting device, as shown in Figure 10. The system volume for experiment was set as 1 bracket including three groups of piezoelectric bending elements which were composed of five pieces per group. The connection of the elements was chosen as the series within groups, and parallel between groups. According to the connection characteristics of piezoelectric elements, the experimental load was selected as 3.3kΩ. By loading the vibration displacement between vehicle body and suspension on the gear-rack, the stacking piezoelectric bending element was excited by the double gear to recover vibration energy. In accordance with the road surface tolerance levels which were identified and analyzed by software platform, the switching action of double gear was carried out by the actuating device, in order to gain the maximum recovery power. The transmission shaft was fixed by bearings at both ends. At one end, the actuating device was composed of electromagnet and acted by pulsing signals. At the other end, the switching status was detected and kept by a self-locking mechanism with a Hall sensor. Figure 10C is a sectional drawing of the self-locking mechanism. The slideways and guides were processed with slopes and fracture-surface, which made sure the slider moving in the order of A→B→C→D→A. According to the switching principle and the status signal of the Hall sensor, when gear 10 mm is needed, the slider will move from position A→B→C by a pulsing signal which is loaded on the electromagnet. Relatively, when gear 3 mm is needed, the Hall sensor and electromagnet will work together to move the slider from position C→D→A.

TABLE 6 Recovery energy and comparison of different gear pitch of road section 1

| Gear pitch | Recovery energy (J) | Double-gear recovery energy (J) | Double-gear improving efficiency |
|------------|--------------------|-------------------------------|---------------------------------|
| Single-gear pitch | 3 mm | 10 mm | 83.631 | 177.132 | 131.66% | 9.37% |
simulation result calculated with one piece of element indicated that reasonable values of the gear pitch are $d_A = 3$ mm and $d_B = 10$ mm. In such case, the vibration energy harvested would reach the maximum value when the vehicle ran on such a section of road. Then, an experiment was carried out on the testing platform. Figure 11C showed the digital signal of automatically switching action which was sent to the actuating device. The moments of high levels were consistent with the average tolerance per 500 m which exceeded the threshold value. In Figure 11D, the red line showed the instantaneous harvested power of testing platform with the automatically switching of double gear enable. The black line and blue line were gained with the switching disable. It was worth noting that the harvested power of double gear was equal or greater than the power of single gear during the whole test. The reason for the low power of single gear 10 mm between 270 s and 530 s as the red star shown was that the vibration displacements on Class A road excited the gear hardly so as to be filtered almost all.

In road section 1, when a single gear is used, the gear pitch is 3 mm and 10 mm, respectively, and the energy harvested from the experiment platform will be 83.631 J and 177.132 J, respectively. When the double-gear recovery mechanism is used with $d_A = 3$ mm and $d_B = 10$ mm which are both automatically switched, the energy harvested will be 193.737 J. Compared with a single gear, the efficiency has been improved by 76.70% and 22.38%, respectively. The comparison was shown in Table 6.

A similar processing was carried out on road Section 2 with the tolerance identified in Figure 12A. The same value-taking result of simulation could prove the effectivity of the method this paper proposed. The platform experiment also proved the switching principle could work validly and reliably under different comprehensive driving conditions. Noting the moment of 1400 s in Figure 12C, this experiment also tested the restoration function in order to confirm the durability of the mechanism when the vehicle shut down. The red star in Figure 12D also meant that the vibration displacements excited the single gear 10 mm hardly.

**TABLE 7** Recovery energy and comparison of different gear pitch of road section 2

| Gear pitch | 3 mm  | 10 mm |
|------------|-------|-------|
| Recovery energy (J) | 66,859 | 96,529 |
| Double-gear recovery energy (J) | 118,137 |
| Double-gear improving efficiency | 76.70% | 22.38% |
With regard to Section 2 of the road, when a single gear is used, the gear pitch is 3 mm and 10 mm, respectively, and the energy harvested from the experiment platform will be 66.859 J and 96.529 J, respectively. When the double gear recovery mechanism is used with \( d_A = 3 \text{ mm} \) and \( d_B = 10 \text{ mm} \) which are both automatically switched, the energy harvested will be 118.137 J. Compared with a single gear, the efficiency has improved by 76.70% and 22.38%, respectively. The comparison was shown in Table 7.

Comparing the results of road section 1 with those of road section 2, it can be found that the difference of efficiency improvement between double gear and single gear is due to the road surface and driving condition. In the proportion of road surface tolerance grades, the more A and B road surface are taken up, the more \( d_A \) is used in double gear. The improvement efficiency will be reduced compared with single 3 mm, but will increase compared with single 10 mm. Relatively, the more C and D road surface is took up, the more \( d_B \) is used in double gear. The improvement efficiency will improve compared with the single 3 mm, but will decrease compared with the single 10 mm. Totally, comparing with single-gear pitch 3 mm, the efficiency improvement of double gear will increase from 0(Class A and B) to 141.34% (Class C) ~ 183.3% (Class D). Comparing with single-gear pitch 10 mm, the efficiency improvement of double gear will increase from 0(Class C and D) to 96.27(Class B) ~ ∞(Class A).

6 | CONCLUSIONS

In relation to the vibration energy harvesting of vehicle, a simulation model and an experimental platform are designed in this paper. By shifting the relative displacement between suspension and vehicle body to the circular motion, an automatically switchable double-gear excitation mechanism is used to achieve the continuous electric energy feedback within the allowed scope of piezoelectric bending elements, thus effectively improving the output power of the elements. Additionally, the road tolerance grade is identified based on the vibration displacement data gathered by the sensor, and an investigation is carried out to explore the effect of the excitation gear pitch and the vehicle speed on harvesting efficiency of the energy-regenerative device, thereby further proposing a value-taking method for gear pitch and a switching principle for the automatically switchable double-gear excitation mechanism to maximize the vibration energy harvesting efficiency under comprehensive driving conditions. The results show that the switchable double-gear excitation mechanism is superior to the single-gear excitation mechanism in terms of the application in energy-regenerative devices, and it is more applicable to meet the need of vibration energy harvesting under comprehensive driving conditions. It provides referential theoretical basis for predictive control on the vibration energy harvesting and experimental basis for a real car on-field test with a full system volume in the future.

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