Research Article

Study on the Influence of Suspension Parameters on Longitudinal Impact Comfort

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Ride comfort criteria are a key challenge for vehicle dynamic design and optimization. Currently optional parameter is the vertical impact, and longitudinal impact is neglected. With further requirements for future comfortability, effects of longitudinal impact should be investigated in detail. A longitudinal impact model is firstly proposed to evaluate the ride comfort factors based on the dynamic theory and commercial ADAMS® software. Predictions revealed that the hardpoints of the suspension and the stiffness of rubber bushing (SORB) are the primary factors. A novelty finding is that travel of rubber bushing (TORB) in the linear region is the most important parameter for ride comfort optimization and suspension factor is the weakest, and experimental validation is performed with better agreements.

1. Introduction

When a driving vehicle passes over a speed bump, the impact force caused by road is being transferred to the car body by means of the suspension causing vibration [1, 2]. As we know that the better attenuation performance of suspension contributes to the generation of smaller vibrations to car body, if it is in worse condition, the stronger vibrations will be transferred [3, 4]. Large vibrations give rise to the discomfortability for passengers and damage for cargo [5, 6]. Therefore, they are of great importance for suspension design and optimized strategy.

In recent years, several studies on impact comfort have been done. Abdulgazi [7] investigated the impact of vehicle load, speed, speed bump height, and other factors on impulse comfort using the maximum vertical acceleration as the evaluation index and determined that vehicle speed and speed bump height were the key indices determining it. Donggyun Kim [8] used a neural artistic style extraction to construct a human evaluator model for the ride comfort of a car over a speed bump, and the model was shown to be considerably more accurate than any other associated models. Lai [9] investigated the effects of roads with distinctive impacts (e.g., potholes, bumps, bulges, and slopes) on ride comfort and discovered that there was an ideal speed to reduce the vertical vibration of automobiles travelling over them for each individual road. Song [10] used other pulse roads not mentioned in Chinese national standards (e.g., rectangular bumps, bevel bumps, potholes, and speed bumps), ran impulse road simulations, compared the results, and concluded that rectangular bumps or speed bumps could be used as pulse input in vehicle ride comfort analysis. Gao [11] used impact road simulations to investigate flexible subframe deformation and found that it may increase impact comfort. It is vital to take into account both a vertical and a longitudinal impact force along the driving direction [12]. The research described above mostly look at the effect of vertical impact on ride comfort rather than longitudinal impact. However, it is very sensitive for passengers and should not be overlooked; more research into the long-term influence is both promising and necessary.

Longitudinal impact model to evaluate ride comfort is proposed based on dynamics theory. Vehicle dynamics are analyzed while passing through a speed bump; that is, hard points of suspension, SORB, and TORB, as well as their effects on longitudinal impact comfort (LIC), are predicted in detail.
2. Influence of the Suspension Parameter on the LIC

2.1. Attenuation Mechanism and Influencing Factors of the LIC. When wheels hit a speed bump during driving, the impact force will cause the wheels to move upward and rearward simultaneously, resulting in vertical and longitudinal deformation of the suspension [13, 14]. The larger the suspension deformation is, the more the impact force is attenuated. The maximum upward travel of the suspension is relatively large, between 70 and 120 mm, and can greatly attenuate the vertical impact force compared with the vertical upward travel. The maximum rearward travel of the suspension is affected by the thickness of the bushing and kinematic characteristics of the suspension. It is generally small, between 2 and 10 mm. When the longitudinal deformation of the suspension reaches its limit and the longitudinal impact force is not significantly attenuated, the impact energy is directly transferred to the car body, causing discomfort to the vehicle passengers. Therefore, to improve a vehicle’s LIC, it is necessary to increase the rearward travel of the suspension after impact. This consists of two parts: automatic rearward travel of the wheels in the process of the suspension bumping upward, which is determined by the hard points, and longitudinal deformation of the suspension after impact, which is determined by the SORB. The mechanism of these two kinds of withdrawal is studied here.

2.2. Influence of Hard Points on the LIC. We did not evaluate the impact of rubber bushing deformation during suspension movement in this research; instead we focused on the suspension’s vertical and longitudinal movement characteristics when the wheels encounter a speed bump from a kinematic standpoint.

As indicated in Figure 1, the rear suspension was used as an analytical example. Due to the impact of suspension kinematics, when the vehicle hits a speed bump while travelling, the wheels will move upward and backward about the instantaneous center. The suspension’s upward displacement was recorded as Dz, while the suspension’s backward displacement was recorded as Dx. Dz had the biggest influence on vertical impact comfort, while Dx had the biggest impact on the LIC. We focused on Dx’s impact mechanism on the LIC in this paper.

The larger Dx of the suspension moving backward when the wheels hit the speed bump, the more impact energy attenuated and the better the LIC. Dx was determined by the kinematic characteristics of the suspension, that is, the change rate of the longitudinal displacement of the wheel center with the wheel jump. The larger the value was, the more the wheels moved backward when meeting a speed bump and the better the attenuation of the impact comfort. Therefore, to improve the LIC, it was necessary to adjust the hard points to make the change rate of the longitudinal displacement of the wheel center with the wheel jump higher.

2.3. Influence of Rubber Bushing Stiffness on the LIC. When wheels hit a speed bump during driving, they were affected by the impact force, F, from the ground, as shown in Figure 2. Decomposing the impact force along the vertical and longitudinal directions, the vertical component force, Fz, mainly affects the vertical comfort of the vehicle, and the longitudinal component force, Fx, mainly affects the longitudinal comfort of the vehicle. In this section, we studied the influence mechanism of longitudinal component force Fx on ride comfort.

Due to the action of the rubber bushing, when the wheels were exposed to an impact force, the suspension was elastically deformed in the direction of the force and attenuated the force. It may be roughly viewed as a bushing put at the wheel center to clearly demonstrate the deformation characteristics of the suspension when exposed to longitudinal force. To reduce the impact force, the rubber bushing bent elastically in the direction of the force, and the driving wheels moved rearward. The greater the attenuation of the impact comfort was, the more the wheels went backward. As a result, in order to enhance the LIC, the rubber bushing stiffness had to be reduced, resulting in a bigger longitudinal displacement of the wheels.

3. Building the Vehicle Model

3.1. Building the Tire Model. The tire plays a decisive role in the accuracy of pulse road simulation. Considering the high input frequency of pulse road [15–18], only F-tire model and PAC2002 with Belt Dynamics tire model with high upper limit frequency can be selected for simulation [13, 14]. Considering the relatively slow operation speed of F-tire model, PAC2002 with Belt Dynamics tire model was selected in this paper for simulation.

The Belt Dynamics tire model in PAC2002 was an enhanced version of the “basic” PAC2002 model, with an upper frequency limit of 80 Hz. The PAC2002 tyre type with Belt Dynamics comprises a rim and a belt; between the rim and the belt, a six-degree-of-freedom bushing with stiffness and damping [19, 20] was used, as illustrated in Figure 3.

The vertical force $F_z$, longitudinal force $F_x$, and lateral force $F_y$ of the wheel are calculated as follows [21–23]:

\[ F_z = F \cos \theta \]
\[ F_x = F \sin \theta \]
\[ F_y = 0 \]

Where $F$ is the impact force, and $\theta$ is the angle between the force and the vertical axis.

\[ \theta = \arctan \left( \frac{y}{x} \right) \]

Where $y$ is the vertical displacement and $x$ is the longitudinal displacement.

\[ F_z = F_k y + F_d \frac{dy}{dt} \]
\[ F_x = F_k x + F_d \frac{dx}{dt} \]
\[ F_y = F_k z + F_d \frac{dz}{dt} \]

Where $F_k$ and $F_d$ are the stiffness and damping coefficients, respectively.

Figure 1: Schematic diagram of wheel motion based on suspension kinematic characteristics.
\[
F_z = \left( q_{RE0} + q_{V2} \right) \left( F_{c0} \right) - \left( \frac{F_x}{F_{c0}} \right) - \left( \frac{F_y}{F_{c0}} \right)
\]

Here, \( q_{RE0} \) is correction factor for measured unloaded radius; \( q_{V2} \) is the tire stiffness variation coefficient with speed; \( \Omega \) is the rotational velocity of the wheel; \( R_0 \) is the unloaded rolling radius of the tire; \( F_{c0} \) is the nominal tire load; \( q_{Fcx1}, q_{Fcy1}, \) and \( q_{Fcy1} \) are the tire stiffness interactions with \( F_x, F_y, \) and \( F_z \) and the camber dependency of the tire vertical stiffness; \( \rho \) is the radial tire deflection; \( m_{c} \) is the mass of the tire; \( \psi \) is the sliding velocities of the contact body in longitudinal, lateral, and yaw directions; \( V_{sx}, V_{sy}, \) and \( \psi \) are the sliding velocities of the lower part of the wheel in longitudinal, lateral, and yaw directions; \( V_{c0}, V_{c1}, \) and \( \psi \) are the corresponding velocities of the contact mass; \( F_{z0}, F_{x0}, \) and \( F_{y0} \) are the vertical, lateral, and yaw stiffness factors of belt contact mass; \( q_{mc} \) is the mass parameter of the tire contact mass.

### 3.2. Building the Whole Vehicle Model

After the tire model was built, the front and rear suspension models, wheel models, steering model, body model, and motor model were built in ADAMS software, and assembly model of the whole vehicle was finally built, as shown in Figure 4.

### 4. Simulation and Comparison of the Influence of the Hard Points and Rubber Bushing Stiffness on the LIC

#### 4.1. Building the Evaluation Model of the LIC

The impact road had a quick action time and a high impact energy, both of which caused considerable pain to passengers and were important criteria for determining vehicle ride comfort. During the test, the left and right wheels of the same axle had to pass over the bump at the same time at the stipulated speed, with the bump’s placement perpendicular to the vehicle’s driving direction. The standard approach for vertical impact comfort employs the highest vertical acceleration, \( Z_{max} \) (absolute value) of the passenger’s foot floor, seat cushion, and seatback to evaluate [24]. The larger the value, the worse the impact comfort of the vehicle. On the contrary, it indicates that the impact comfort of the vehicle is better, and the calculation formula is as follows:

\[
\hat{Z}_{max} = \frac{1}{n} \sum_{j=1}^{n} \hat{Z}_{max, j}.
\]

Here, \( n \) represents the times of the pulse test; \( n \geq 5 \); \( \hat{Z}_{max} \) is the maximum value (absolute value) of the vertical acceleration response; and \( \hat{Z}_{max, j} \) is the jth test of the maximum value (absolute value) of the vertical acceleration response.
This paper innovatively presents using the maximum longitudinal acceleration (MLA) response, \( \dot{X}_{\text{max}} \) (absolute value), to evaluate the LIC. The larger the value, the worse the suspension’s attenuation performance. Conversely, the smaller the MLA response, the better the suspension’s attenuation performance. The \( \dot{X}_{\text{max}} \) value was calculated as follows:

\[
\dot{X}_{\text{max}} = \frac{1}{N} \sum_{j=1}^{N} \dot{X}_{\text{max},k}.
\]

Here, \( \dot{X}_{\text{max}} \) is the maximum value (absolute value) of the longitudinal acceleration response and \( \dot{X}_{\text{max},k} \) is the \( k \)th test of the maximum value (absolute value) of the longitudinal acceleration response.

4.2. Building the Triangular Bump Road Model. A triangle bump road was developed in the ADAMS programme to mimic the circumstance of cars travelling over a speed bump in accordance with Chinese national standard GB/T 4970-
Figure 5: The triangular impulse road model.

Figure 6: Variation of longitudinal acceleration with time.

4.3. Simulation and Comparison of the Influence of Hard Points and Rubber Bushing on the LIC. In the vehicle model, the position of the connection point between the twist beam and body was lowered to the same height as the wheel center so that the change rate of the longitudinal displacement of the wheel center with the wheel jump was zero. Thus, when the wheels hit the bump, the wheel recession would mainly be contributed by the bushing. Then, the impact road simulation was carried out to study the influence of the bushing on the LIC.

To study the influence of the hard points on the LIC, the bushing at the joint between the twist beam and body in the model was changed to a ball joint, which was equivalent to infinite SORB. Thus, when the wheels hit the bump, the backward retraction of the wheel would mainly be contributed by the hard points of the suspension. Then, the influence of the hard points on the LIC was studied by pulse road simulation.

Figure 6 depicts the longitudinal acceleration curve at the H-point of the back seat as a function of the time the car travelled over the bump. The front wheel impacted the bump first, creating body vibration, followed by the rear wheel hitting the hump, causing body vibration as well. We focused on the impact of the rear suspension hard points and bushing settings on the LIC in this article, ignoring the impact of the front suspension; therefore we only looked at the change in the body’s longitudinal acceleration when the rear wheels travelled over the bump. In Figure 6, the black solid line illustrates the original car, and as the vehicle travelled over the bump, the bushing and hard points operated together. The bushing operated alone when the car
travelling over the bump, as seen by the red dotted line. The hard points that acted alone as the vehicle passed over the bump are represented by the green-dotted line.

Through comparison, it was concluded that when hard points and bushing worked together, the MLA was the smallest of the three curves, and the attenuation performance of the suspension was the best. When the bushing acted alone, the MLA was slightly larger than that of the original vehicle, and the suspension attenuation performance was slightly worse. When the hard points acted alone, the MLA was significantly larger than that of the original vehicle, and the suspension’s attenuation performance was significantly worse, indicating that the influence of the bushing on the LIC was greater than that of the hard points.

5. Influence of the Rubber Bushing Parameters on the LIC

5.1. Mechanical Model and Calculation Principle of Rubber Bushing. The bushing is used as a mechanical element in ADAMS software to link two components and transmit force and torque. The bushing’s direction is described as follows: the X direction is the hollow direction in the radial direction, the Y direction is the solid direction perpendicular to it, and the Z direction is the axial direction. The bushing mechanics model will calculate the elastic force and damping force in the six directions that need to be applied to the two components based on the relative displacement and velocity of the two components at the connection point using dynamic equations during the ADAMS software solution process and then apply it. In ADAMS software, the dynamic equation for bushing is as follows:

\[
\begin{bmatrix}
F_{bx} \\
F_{by} \\
F_{bz} \\
T_{bx} \\
T_{by} \\
T_{bz}
\end{bmatrix} =
\begin{bmatrix}
K_{11} & 0 & 0 & 0 & 0 & 0 \\
0 & K_{22} & 0 & 0 & 0 & 0 \\
0 & 0 & K_{33} & 0 & 0 & 0 \\
0 & 0 & 0 & K_{44} & 0 & 0 \\
0 & 0 & 0 & 0 & K_{55} & 0 \\
0 & 0 & 0 & 0 & 0 & K_{66}
\end{bmatrix}
\begin{bmatrix}
x \\
y \\
z \\
\theta_x \\
\theta_y \\
\theta_z
\end{bmatrix}
\]

\[
\begin{bmatrix}
C_{11} & 0 & 0 & 0 & 0 & 0 \\
0 & C_{22} & 0 & 0 & 0 & 0 \\
0 & 0 & C_{33} & 0 & 0 & 0 \\
0 & 0 & 0 & C_{44} & 0 & 0 \\
0 & 0 & 0 & 0 & C_{55} & 0 \\
0 & 0 & 0 & 0 & 0 & C_{66}
\end{bmatrix}
\begin{bmatrix}
V_x \\
V_y \\
V_z \\
\omega_x \\
\omega_y \\
\omega_z
\end{bmatrix}
\] +
\[
\begin{bmatrix}
F_1 \\
F_2 \\
F_3 \\
T_1 \\
T_2 \\
T_3
\end{bmatrix}
\]

where \(F_{bx}, F_{by}, F_{bz}, T_{bx}, T_{by}, T_{bz}\), respectively, are the six forces and moments in which the rubber bushing restrains the relative movement of the two components; \(K_{11}, K_{22}, K_{33}\), respectively, are the stiffness of the rubber bushing along the \(x\)-axis, \(y\)-axis, and \(z\)-axis; \(K_{44}, K_{55}, K_{66}\), respectively, are the torsional stiffness of the rubber bushing around the \(x\)-axis, \(y\)-axis, and \(z\)-axis; \(C_{11}, C_{22}, C_{33}\), respectively, are the damping coefficients of the rubber bushing along the \(x\)-axis, \(y\)-axis, and \(z\)-axis; \(C_{44}, C_{55}, C_{66}\), respectively, are the damping coefficients of the rubber bushing around the \(x\)-axis, \(y\)-axis, and \(z\)-axis; \(x, y, z, \theta_x, \theta_y, \theta_z\), respectively, are the linear and angular displacements of the six relative movements at the connection of the two components; \(V_x, V_y, V_z, \omega_x, \omega_y, \omega_z\), respectively, are the linear and angular velocities of the relative movement in the six directions at the connection point of the two components; and \(F_1, F_2, F_3, T_1, T_2, T_3\), respectively, are the initial loads of the rubber bushing in six directions.

In ADAMS software, the mechanical properties data of rubber bushing are stored in the property file, and the calculation is carried out by calling the property file of rubber bushing during simulation.

5.2. Comparison of the Rubber Bushing Optimization Schemes. The installation diagram of the rubber bushing at the connection point of the twist beam and vehicle body is shown in Figure 7. The hollow direction of the rubber bushing was generally arranged along the longitudinal direction of the vehicle, and the solid direction was arranged along the vertical direction of the vehicle. The hollow direction of the rubber bushing mainly attenuated the longitudinal impact, so as to ensure that the vehicle had a good LIC.
Change the rubber bushing construction to lower the stiffness of the linear area in the hollow direction by 40%, as illustrated in Figure 8, to explore the impact of rubber bushing hollow direction stiffness and limit stroke on the LIC, while ensuring that the outside diameter of the bushing does not change (b). Furthermore, as shown in Figure 8, increasing the clearance in the hollow direction increased the linear region’s trip in the hollow direction by 40% (c). The stiffness curves of the various rubber bushing schemes are shown in Figure 9, where the black solid line represents the original bushing’s stiffness, the red-dashed line represents the linear region’s SORB reduced by 40%, and the green-dotted line represents the linear region’s TORB increased by 40%. Finally, the stiffness data from various rubber bushing modification strategies were saved in the ADAMS software’s characteristic file for use in further simulations.

5.3. Simulation and Comparison of the Influence of the Rubber Bushing Parameters on the LIC. To study the influence of the rubber bushing parameters on the LIC, the simulation of different speeds was carried out based on the triangular bump road built previously.

Figure 10 shows the change curve of longitudinal acceleration at the H-point of rear seat with time when the vehicle passed over the triangular bump road at a typical speed of 30 km/h. Figure 11 shows the change curve of the MLA at the H-point of rear seat when the vehicle drove on the triangular bump road at different speeds. When the

![Figure 8](image1.png)

![Figure 9](image2.png)

**Figure 8**: Different rubber bushing optimization schemes. (a) Original scheme. (b) SORB of the linear region reduced by 40%. (c) TORB of the linear region increased by 40%.

**Figure 9**: Comparison of the SORB of different schemes.
vehicle speed was 10 km/h, the MLA was decreased by the scheme of reducing the SORB of linear region by 40%, but in the scheme of increasing the TORB of linear region by 40%, the MLA was almost unchanged. The reason was that when the vehicle passed over the speed bump at a low speed, the longitudinal impact force was relatively small, and the deformation of the bushing did not enter the nonlinear region. At this time, the bushing mainly worked in the linear region, the stiffness of which was relatively small to better attenuate the impact from the road; thus, increasing the TORB of the linear region had little effect on the MLA. Reducing the SORB of the linear region was beneficial for the impact attenuation, so the MLA became smaller.

When the vehicle speed was between 20 and 60 km/h, using the scheme to reduce the linear region stiffness by 40%, the MLA was slightly reduced compared with that of the original vehicle. In the scheme to increase the linear region travel by 40%, the MLA was significantly reduced compared with that of the original vehicle, and the reduction amount was higher than that of the first scheme. This was because when the vehicle speed increased and the vehicle passed over the bump, the impact force on the suspension became larger,
Figure 12: Impact road test. (a) Impact road. (b) Rear seatback and cushion acceleration sensors. (c) Rear floor acceleration sensors.

Figure 13: Variation of the MLA at the rear passenger’s foot floor with speed.
Figure 14: Variation of the MLA at the rear passenger’s seat cushion with speed.

Figure 15: Variation of the MLA at the rear passenger’s seatback with speed.

Table 1: The MLA of each measurement point under the different schemes.

| Speed (km/h) | Foot floor | 10 | 20 | 30 | 40 | 50 | 60 |
|--------------|------------|----|----|----|----|----|----|
| MLA of original stiffness scheme (m/s²) | Foot floor | 5.25 | 8.00 | 11.71 | 13.2 | 13.7 | 15.1 |
| | Seat cushion | 3.30 | 4.30 | 6.05 | 8.23 | 8.56 | 10.06 |
| | Seatback | 5.54 | 7.47 | 12.06 | 14.81 | 14.5 | 16.66 |
| MLA of TORB of linear region increased by 40% scheme (m/s²) | Foot floor | 5.15 | 7.03 | 9.92 | 11.4 | 11.7 | 13.0 |
| | Seat cushion | 3.25 | 4.00 | 6.06 | 8.95 | 7.22 | 8.52 |
| | Seatback | 5.45 | 7.05 | 9.97 | 12.31 | 12.0 | 13.8 |

MLA change rate (%)

| MLA change rate (%) | Foot floor | 1.90 | 12.13 | 15.29 | 13.64 | 14.60 | 13.91 |
| | Seat cushion | 1.52 | 6.98 | 16.36 | 15.55 | 15.65 | 15.31 |
| | Seatback | 1.62 | 5.62 | 17.33 | 16.88 | 17.24 | 17.17 |
resulting in the deformation of the bushing into the non-linear region. As the stiffness of the bushing in the nonlinear region was relatively large, the impact force transferred to the H-point of the rear seat could not be further attenuated. Increasing the TORB of the linear region always pushed the deformation of the bushing to the linear region when the wheels hit the bump so that the impact force transferred to the H-point of the rear seat was attenuated better. In the scheme to reduce the SORB of the linear region by 40%, although the reduction was conducive to the attenuation of the longitudinal impact, the deformation of the bushing entered the nonlinear region, and the SORB in the nonlinear region was relatively large, resulting in the impact transferred to the H-point of the rear seat to not be well attenuated.

The impact force from the ground was rather minor when the car travelled over the bump at a relatively low speed of 10 km/h, and the passengers were largely unaffected. When the vehicle speed exceeded 20 km/h, the collision force was significant, resulting in passenger complaints. As a result, the change in longitudinal acceleration over 20 km/h was the major emphasis here. When the vehicle speed exceeded 20 km/h, the strategy of raising the TORB of the linear area could considerably enhance the LIC when compared to the plan of lowering the SORB of the linear area. As a result, it was recommended that the LIC be improved by extending the linear area rubber bushing travel in a real vehicle.

6. Impact Road Ride Comfort Test

To verify the above conclusions, a test was carried out with a real vehicle. In the test, two schemes of bushings were adopted at the installation point of the twist beam and car body: an original stiffness scheme and a scheme to increase the hollow direction of the travel of the linear region by 40%. Then, a pulse road test was conducted after acceleration sensors were installed at the floor of the rear right passenger’s foot as well as at their seat cushion and seatback, as shown in Figure 12, to compare the changes in longitudinal acceleration when the vehicle passed over the speed bump.

The curvature of the MLA at the right rear passenger’s foot floor, seat cushion, and seatback dependent on speed is shown in Figures 13, 14, and 15. It can be seen from the comparison of curves that with the solution of increasing the travel in the linear zone of the bushing by 40%, the MLA at the right rear passenger’s foot floor, seat cushion, and seat back was significantly smaller than that of the original scheme, and the LIC at different speeds was improved. Table 1 compares the MLA of each measurement point to that of the original vehicle after the linear region’s TORB was raised by 40%. It can be shown that when the vehicle speed is 30 km/h, the MLA change rate is the highest, and the LIC improvement is the greatest. At the time, the LIC’s improvement rates for the right rear passenger’s foot floor, seat cushion, and seatback were 15.29 percent, 16.36 percent, and 17.33 percent, respectively.

Via this test, the feasibility of the analysis conclusions in the previous sections was verified, and it was concluded that when the vehicle speed was 30 km/h, the change rates of the MLA at the floor, seat cushion, and seatback were the largest.

7. Conclusions

In this work, longitudinal factors regarding vehicle impact comfort are investigated. The remarks are given as follows:

1. Passenger comfortability is determined by both vertical and longitudinal impacts while passing over bump road

2. Accurate evaluations on impact comfort are obtained by using optimized longitudinal vibration model based on the vertical algorithm

3. Effect of bushing stiffness on LIC is predominantly greater than suspension hard points, and rubber bushing in travel linear region is larger than stiffness as well

Data Availability

The data used to support the findings of this study are included within the article.

Conflicts of Interest

The authors declare no conflicts of interest.

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