Design of cab seat suspension system for construction machinery based on negative stiffness structure

Xin Liao¹², Xiaofei Du³ and Shaohua Li²

Abstract
In order to improve the vibration isolation performance of cab seat and ride comfort of the driver, a seat suspension structure of construction machinery cab is proposed based on negative stiffness structure (NSS) in this paper. The influences of different parameters of suspension system on dynamic stiffness are analyzed. The configuration parameter range of suspension system is obtained. Then, the nonlinear dynamic equation of the seat suspension system is established and the NSS optimization model is proposed. The vibration transmissibility characteristics of suspension structure are analyzed by different methods. The results show that the displacement and acceleration amplitude of optimized seat suspension system are obviously reduced, and the VDV and RMS in the vertical vibration direction for the seat are respectively decreased by 87% and 86%. The vibration transmissibility rate SEAT and the Ttrans are both decreased. Moreover, the peak frequencies of the vibration transmitted to the driver are not near the key frequency values which are easy to cause human discomfort. It indicates that the design of seat suspension system has no effect on the health condition of the driver after being vibrated. The advantages of vibration isolation performance of the designed NSS suspension system are demonstrated, improving the driver’s ride comfort and the working environment.

Keywords
Negative stiffness structure, seat suspension system, cubic stiffness, dynamic properties, ride comfort

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Introduction
In nearly 20 years, more and more attention has been paid to the ride comfort and humanized design of vehicle seats with the continuous improvement of road traffic. Especially, construction machinery as modern construction transport equipment, the cab seats will be vibrated to various degrees during vehicle operation.¹ Previous researches mainly focus on the vibration isolation of engines and vehicle mounted equipment of construction machinery.²⁻⁴ However, the drivers have been in the working environment of forced vibration for a long time, which will cause fatigue and lead to frequent accidents. The driver’s mental and physical health will also be seriously damaged. Researches have shown that cervical, spinal, and pelvis injury of human body will occur frequently.⁵,⁶ Thus, the research and development design of seat suspension system have a great influence on people’s health life. Advocating people-

¹School of Mechanical Engineering, Shijiazhuang Tiedao University, Shijiazhuang, China
²State Key Laboratory of Mechanical Behavior and System Safety of Traffic Engineering Structures, Shijiazhuang Tiedao University, Shijiazhuang, China
³School of Mechanical Engineering, Nanjing Institute of Technology, Nanjing, China

Corresponding author:
Xin Liao, School of Mechanical Engineering, Shijiazhuang Tiedao University, No. 17 North Second Ring East Road, Shijiazhuang 050043, China.
Email: sinna_liaoxin@stdu.edu.cn

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oriented, fully considering the driver’s physical feelings and strengthening the protection of the driver, and improving the working environment are an important development orientation of vehicle seat suspension system.

The vibration isolation mechanism containing parallel spring with negative stiffness structure (NSS) is a new type of nonlinear vibration isolation equipment, which can have a obvious effect on improving the vibration isolation effect, in particular, the low frequency vibration isolation effect. van Eijk and Dijksman⁷ first used negative stiffness in the mechanical design of leaf spring to reduce the total stiffness of the system. South Korean scholars Lee et al.⁸ developed the seat vibration isolation mechanism with negative stiffness based on thin shell theory, and gave the theoretical derivation and experimental verification, which achieved good results. In the theoretical research area, Liu et al.⁹ used a seismic isolation system including quasi-zero stiffness and vertical damper to control near-fault vertical earthquakes. The formula for the maximum bearing capacity of quasi-zero stiffness isolation considering the stiffness of vertical spring components is obtained by theoretical derivation. Junshuet al.¹⁰ used a kind of bending mounted spring roller mechanism as a negative stiffness calibrator in parallel with a vertical linear spring, and developed and designed a passive nonlinear isolator, and analyzed the dynamic characteristics of the isolator. Sun and Jing¹¹ developed a novel vibration isolator with 3D quasi-zero-stiffness property. The remarkable feature of the proposed system is to apply symmetrically scissor-like structures in the horizontal directions, together with a traditional spring-mass-damper system assembled vertically with positive stiffness. Liu and Yu¹² added an auxiliary system to the high-static–low-dynamic-stiffness isolator to overcome disadvantages, with the static displacement of the isolation object remaining unchanged, and discussed the dynamic response and most importantly analyzed the stability of the steady-state response. Zhang et al.¹³ superposed the quasi-zero stiffness system and the inertial nonlinear energy sink (NES) together, and proposed a combined vibration control technique. The results show that the combined control system provides a smaller resonance amplitude and a wider vibration isolation band. The combined control scheme has both effects of the nonlinear isolation and the nonlinear absorption. In the recent years, many scholars have carried out a series of studies on negative stiffness structure.¹⁴⁻²⁰ However, for cab seats of construction machinery, there is little literature about the in-depth research of vibration isolation equipment with negative stiffness structure. Therefore, it is necessary to establish an accurate seat suspension system with NSS characteristics, and design the most desired parameters of system according to the space size of cab seat, and analyze its dynamic characteristics.

For the above reasons, in this paper, the model description of designed cab seat suspension system for construction machinery is firstly given. The theoretical model based on negative stiffness structure (NSS) is established, and different structural parameters and their influences on dynamic stiffness are respectively discussed. Secondly, the nonlinear dynamic model of seat suspension system of the construction machinery cab based on negative stiffness structure is proposed. Based on this, the optimization model of seat suspension structure of construction machinery cab is put forward, and nonlinear dynamic analysis is carried out to study the vibration transfer characteristics of suspension system. Finally, according to the ISO-2631 standard, different methods are used to evaluate the ride comfort of the seat in order to improve the driver’s working environment and reduce vibration.

Model description of seat suspension

The vehicle cab seat model consists of three parts: seat suspension system, seat structure frame, and driver, as shown in Figure 1. In order to investigate the dynamic characteristics of the seat suspension system specifically, the weight of the seat and human body is simplified into a rigid mass block M, and the elasticity of the cushion, the damping inside the human body, the weight of the connecting rod and joint, etc. are neglected. The simplified model contains two symmetrical negative stiffness structures, dampers, mass block and the supporting spring in vertical direction, as shown in Figure 2. Moreover, two additional tunable inerter elements (TIE) are attached on the two rods symmetrically, and \( m_a \) is the mass of the inerter element, and the location of the attached inerter element can be adjusted to achieve a better vibration isolation performance of seat. The rotation freedom of the seat structure is limited, and the seat can only move in the vertical direction. The vibration reduction amount of

![Figure 1. Vehicle seat simplified model.](image)
suspension system will be affected by different parameters, such as spring stiffness, damping coefficient, length of connecting rod, and so on.

### Static analysis of system

#### Theoretical model

In Figure 2, $K_v$ and $K_h$ respectively represent vertical spring stiffness and horizontal spring stiffness, $K_c$ represents cubic spring stiffness, $L_0$ represents the original length of horizontal spring, $L_h$ represents the length of horizontal spring at any position, $F$ represents the elastic restoring force of the vibration-isolated object, $b$ represents the length of connecting rod between horizontal spring and the isolated equipment M, $a$ represents the distance from the isolated equipment M to the fixed end of horizontal spring. The initial state of the isolated equipment M is the position of the dashed frame, and each spring of the suspension system is in a compression state during the vehicle operation. The dynamic stiffness of the suspension system can be adjusted by changing the above parameters.

Assuming that the spring in Figure 2 is in equilibrium, the deformation length of the horizontal spring is

$$\Delta_l = \Delta_v = \sqrt{b^2 - (H - x)^2} - (a - L_0)$$

where $H$ is the initial deformation of the vertical spring.

$$H = \sqrt{b^2 - (a - L_0)^2}$$

Then the horizontal force produced by the horizontal spring is

$$F_{hv} = 2K_h(\sqrt{b^2 - (H - x)^2} - (a - L_0))\tan \alpha$$

where $\alpha$ is the angle between the rod and the horizontal plane

$$\tan \alpha = \frac{H - x}{\sqrt{b^2 - (H - x)^2}}$$

The force produced by the vertical spring in the vertical direction can be expressed as

$$F_v = K_v \cdot x + K_c \cdot x^3$$

Thus, according to the principle of virtual work, the restoring force $F$ of the system in the vertical direction is obtained

$$F = F_v + F_{hv} = K_v \cdot x + K_c \cdot x^3 + 2K_h(H - x)(1 - \frac{a - L_0}{\sqrt{b^2 - (H - x)^2}})$$

For dimensionless treatment, introducing the structural parameters of the system are respectively as follow

$$F' = \frac{F}{K_vL_0}, \alpha = \frac{K_h}{K_v}, \eta = \frac{K_vL_0^2}{K_h}, \chi' = \frac{x}{L_0}$$

$$\gamma_1 = \frac{b}{L_0}, \gamma_2 = \frac{a}{L_0}, H' = \sqrt{\gamma_1^2 - (\gamma_2 - 1)^2}$$

where $F'$ is dimensionless restoring force, $\chi'$ is dimensionless displacement of polynomial, Thus, the dimensionless force-displacement relation of system is obtained, as follow

$$F' = \chi' + \eta \chi'^3 + 2\alpha(1 - \frac{\gamma_2 - 1}{\sqrt{\gamma_1^2 - (H' - \chi')^2}})$$

Then, by differentiating equation (8) the relationship between the dimensionless dynamic stiffness $K'$ and displacement $\chi'$ of system is obtained.

$$K' = 1 + 3\eta \chi'^2 - 2\alpha(1 - \frac{H' + (\gamma_2 - 1)(\gamma_2 - 1)\chi'^2}{(\gamma_1^2 - (H' - \chi')^2)^{3/2}}) - \frac{\gamma_2 - 1}{\sqrt{\gamma_1^2 - (H' - \chi')^2}}$$

According to equation (9), it can be seen that there are four design parameters $\alpha$, $\gamma_1$, $\gamma_2$, $\eta$, and their
variation can have the influence on the dynamic stiffness of system.

**Quasi-zero-stiffness (QZS) conditions**

Letting $H' - x' = 0$, the dimensionless equivalent stiffness of the suspension system at the static equilibrium position is given by

$$K_{eq}' = 1 + 3\eta H'^2 - 2\alpha \left( \frac{\gamma_1 - \gamma_2 + 1}{\gamma_1} \right)$$

(10)

$K_{eq}'$ may be larger or smaller than zero. However, QZS conditions of the suspension system could be obtained by letting the dynamic stiffness $K_{eq}'$ and the second derivation $\frac{d^2 K_{eq}'}{dx'^2}$ be equal to zero.

Under these two conditions, $\alpha = 1/4$ is obtained which can make the QZS range wider around the equilibrium position, approximately like a horizontal straight line. In addition, some existing parameter combinations will lead to the inflexion points of $K_{eq}'$ at the static equilibrium position so that the lower QZS in a wide region could be generated. And the second derivative of dynamic stiffness of the suspension system is always larger than zero. It is worth noting that design of the parameters $\alpha$, $\gamma_1$, $\gamma_2$, $\eta$ could be achieved according to practical applications. Among the four independent parameters, $\alpha$ represents the stiffness ratios and $\eta$ is the combination of stiffness ratio and structural dimension. Both $\gamma_1$ and $\gamma_2$ are the structural dimension ratio. Thus, by considering space dimensions of construction machinery cab and different working conditions, the values of parameters $\alpha$, $\gamma_1$, $\gamma_2$, $\eta$ can be readily realized and optimized.

**Parameter analysis**

Different parameters will have different influences on suspension system. In order to evaluate the mechanical properties of the system, the value ranges for parameters need to be further determined. The dimensionless force-deflection characteristic curves under different values of $\alpha$ are shown in Figure 3. With the increase of the value of $\alpha$, the curvature of the curve has been changed. Among these curves, the flatness degree of the curve is even higher at $0.6 < \alpha < 1$.

As shown in Figure 4, by MATLAB numerical calculation, it can be found that with the increase of displacement, the restoring force curve of the system has been on a downward trend, and the proposed structure has obvious negative stiffness characteristics. When $\gamma_1 < 1$, the occurrence time of the first peak of the curve can be faster and the change of the corresponding displacement is smaller. When $\gamma_1 \geq 1$, the force change trend of the system in the vertical direction can be gentle, which indicates that the seat has better vibration isolation performance, and the selection of the NSS parameters is desired. It can be found that the dimensionless force $F''$ of the system will peak in a short time when the value of $\gamma_1$ is especially small. Therefore, this situation should be avoided to occur when designing parameters so as not to affect the dynamic characteristics of the suspension system. In addition, by contrasting and analysis, the results show that the change of $\alpha$ will affect the trend of the overall force change of the system. With the increase of $\alpha$, the trend in the change of force will accelerate, indicating that the vibration amplitude of the seat in the vertical direction will increase, which will reduce the driver’s ride comfort. As a result, it is not recommended that $\alpha$ take a larger value and try to make the stiffness of the horizontal spring smaller than that of the vertical spring.

Figure 5 shows the dimensionless force-deflection characteristic curves with different values of $\gamma_2$. The results show that when the value of $\gamma_2$ is greater than 1, curve changes very regularly, which is similar to an elongated sinusoidal curve, existing the maximum and minimum peak value. It indicates that the dimensionless force $F''$ between the two peaks decreases with the increase of displacement, and inversely in the other positions the value of $F''$ increases with the increase of displacement. However, when the value of $\gamma_2$ is less than 1, the initial value of the dimensionless force $F''$ begins to change irregularly, and the value of $F''$ decreases with the increase of displacement. In this case, although the stiffness of the suspension system is negative stiffness, it is not suitable for the design of the above negative stiffness model structure. And its reason can be found in Figure 7. When $\gamma_2 < 1$, the shape of stiffness curve is convex parabola, and the stiffness reaches the maximum peak value in the static...
equilibrium position. But when it is distant enough from the static equilibrium position, the stiffness of the system is always negative, and the vibration isolation system will be in an unstable state.

Considering the nonlinear characteristics of the suspension system and multiple design parameters, the dimensionless dynamic stiffness of the system should be used to evaluate the vibration isolation system so that the appropriate suspension configuration parameters can be selected. As can be seen in Figure 6, as the value of $\gamma_1$ becomes larger, the basic outline of the stiffness characteristic curves of system does not change obviously, while the curvature of the stiffness characteristic curve changes. For dimensionless displacement $x' < 0$, when the value of $\alpha$ increases, the absolute values of stiffness of the suspension system also increases. For dimensionless displacement $x' > 0$, when the value of $\alpha$ increases, the curve moves to the right overall, and the suspension system deviates from the quasi-zero stiffness more and more. It also illustrates that the value of the parameter $\alpha$ should not be choose to be too large.

From Figure 7, the corresponding stiffness curves under different values are nearly symmetric at $z = 0$. When $\gamma_2 \geq 1.2$, the dynamic stiffness curve is concave parabola shape, and the stiffness value at static equilibrium position is the smallest, and when $0.9 \leq \gamma_2 < 1.2$, the stiffness value is unstable with varying between positive and negative. Therefore, the configuration of structural parameters needs to be reasonably selected to make the dynamic stiffness of the static equilibrium position zero or greater than zero. Additionally, when $\gamma_2 = 1$ and $\alpha = 1.2$, its dynamic stiffness curve is an oblique line with a tiny angle, and the stiffness value can be considered as $K' = -1$. This situation shows that the distance from the isolated equipment to the static equilibrium position can be infinite, and dynamic stiffness of the region above the line $K' = -1$ is always less than static stiffness. However, in case of $\gamma_2 > 1$, the

**Figure 4.** Dimensionless force-deflection curves with the various values of $\gamma_1$. (a) $\alpha = 0.8$, $\gamma_2 = 1.2$, (b) $\alpha = 1.0$, $\gamma_2 = 1.2$, and (c) $\alpha = 1.2$, $\gamma_2 = 1.2$. 

Liao et al. 5
infinite range from the isolated equipment to static equilibrium position decreases with the increase of values of $g_2$.

Figure 7 also shows several sets of stiffness curves corresponding to different values of $\alpha$. Through comparison, it can be concluded that when $\alpha = 0.5$ and $\gamma_2 \geq 1.2$, the dynamic stiffness of the system around the static equilibrium position is very close to zero, that is, near quasi-zero stiffness. This illustrates that the selection of parameters $\alpha$ and $\gamma_2$ has a certain impact on the stiffness characteristics of the system.

Furthermore, the dynamic stiffness curves of the system under different variations of $\alpha$ are shown in Figure 8. It can be seen that as the value of $\alpha$ increases, the minimum value of the dimensionless stiffness curve will decreases. When $\alpha = 1.2$ and $\alpha = 1.4$, the system is close to quasi-zero stiffness, but the concave parabola is closer to the center axis. The correctness of the above $\gamma_2 \geq 1.2$ analysis results is verified again. If the larger the range of displacement changes for the suspension system, the better the vibration isolation effect. To sum up, the design parameters of suspension system $1.2 \leq \gamma_2 \leq 1.4$ and $\alpha = 0.25$ are the most desired configuration parameters of the system. But in the actual design process, it is difficult to obtain these accurate values because of the errors in manufacturing and assembly, and $\gamma_2 < 1.2$ maybe also appear, so the quasi-zero stiffness should be satisfied as far as possible during the design.

**Dynamic model**

**Dynamic modeling of the proposed seat suspension**

With respect to the seat suspension system, $u_m(t)$ is the absolute displacement of the isolated equipment in space. When the isolated equipment $M$ experiences upward a vertical displacement $z$ from its initial position, the kinetic energy $T$ of the system is
The vertical upward external excitation from the base is \( u_e(t) \). The relative vertical motion is \( z(t) = u_m(t) - u_e(t) \). \( z \) is the displacement of the isolated equipment relative to the base of cab.

The potential energy of the system contains the elastic potential energy of the vertical springs and the two springs in horizontal direction, which is as

\[ T = \frac{1}{2} M u_m^2 + m_a v_a^2 \quad (11) \]

\[ z = H - x \quad (13) \]

where

\[ v_a = \sqrt{\left\{ \frac{d(\lambda x)}{dt} \right\}^2 + \left\{ \frac{d \left\{ (1 - \lambda) \sqrt{b^2 - (H - x)^2} - (1 - \lambda)(a - L_0) \right\}}{dt} \right\}^2} \]

\[ = \dot{x} \sqrt{\lambda^2 + (1 - \lambda)^2 \frac{(x - H)^2}{b^2 - (H - x)^2}} \quad (12) \]

The relative velocity and acceleration of the isolated equipment are respectively defined as below

\[ \dot{z} = \dot{u}_m - \dot{u}_e \quad (14) \]

\[ \ddot{z} = \ddot{u}_m - \ddot{u}_e \quad (15) \]
The dissipation function $D$ is given by

$$D = \frac{1}{2} C_l \Delta^2_l + \frac{1}{2} C_r \Delta^2_r + \frac{1}{2} C_v \Delta^2_v + \frac{1}{4} k_v \Delta^4_v$$

$$= \frac{1}{2} C_v (\dot{u}_m - \dot{u}_c)^2$$

(17)

$C$ is the damping coefficient ($N/m/s$), $k$ is the spring stiffness ($N/m$), $\Delta$ is the variation of the spring length, $\dot{\Delta}$ is the relative velocity at both ends of the spring ($m/s$), Subscripts $l$ and $r$, respectively represent the left, right direction of the suspension system. $\lambda$ ($0 \leq \lambda \leq 1$) is the distance ratio. Since the system is symmetrical structure, it is considered that $k_l = k_r = K_i$, $C_l = C_r$, $\Delta_l = \Delta_r$. For the model of this suspension system, the damping in the horizontal direction is designed to be $C_l = C_r = 0$. Note that the weight of the isolated equipment $M$ and the two attached inerter elements can be offset by the pre-compression of the vertical spring. So, we can have

$$K_i \Delta_i + K_v \Delta^3_v = Mg + 2\lambda m_v g$$

(18)

$$\Delta_x = x + \Delta_s$$

(19)

where $\Delta_x$ is the pre-compression of the vertical spring.

Applying the Lagrangian equation of the second kind, it can be expressed as

$$\frac{d}{dt} \left( \frac{\partial T}{\partial \dot{u}_m} \right) - \frac{\partial T}{\partial u_m} (T - V) = Q_x$$

(20)

where $Q_x$ comprises of the excitation force $F_e$ and the damping force introduced to account for energy dissipation, and it can be expressed as

$$Q_x = C_v (\dot{u}_m - \dot{u}_c)^2$$

(21)
By substituting equation (12) into equation (11), substituting equations (18) and (19) into equation (16), and then substituting the expressions of potential energy, kinetic energy into the Lagrange equation (20), the dynamic equation of the system can be derived as follows

\[ Q_x = F_e - \frac{\partial D}{\partial u_m} \]  

By substituting equation (12) into equation (11), substituting equations (18) and (19) into equation (16), and then substituting the expressions of potential energy, kinetic energy into the Lagrange equation (20), the dynamic equation of the system can be derived as follows

\[
\begin{align*}
\left\{ 1 + 2 \frac{m_a}{M} \left[ \alpha^2 + \left( 1 - \frac{a}{b} \right) \frac{z^2}{b^2 - z^2} \right] \right\} \ddot{z} \\
- \frac{2 m_a}{M} \left( 1 - \frac{a}{b} \right) \frac{z}{b^2 - z^2} \ddot{z}^2 + \frac{C}{M} \dot{z} + \frac{K_e}{M} z + \frac{K_e}{M} \dot{z}^3 \\
+ 2 \frac{K_h}{M} \left( 1 - \frac{a}{b} \right) \frac{z}{\sqrt{b^2 - z^2}} \dot{z} = -\ddot{u}_c
\end{align*}
\]  

where \( \Delta_z \) and gravity terms no longer exist due to equations (18) and (19).

The left-hand side of the equation of motion includes damping term and restoring force term respectively, which are the same as those of the equivalent quasi-zero-stiffness (QZS) system without the attached inerter element. From equation (22), it also can be seen that the nonlinear stiffness term and the inertia term have nonlinear characteristics. The structural parameters of the system can be adjusted to satisfy different engineering requirements. Especially when the construction machinery is excited by larger amplitude, the vibration peak transmissibility rate can be further reduced by adjusting the parameter of nonlinear inerter \( \lambda \).

**Driver body vibration**

WBV (Whole Body Vibration) exposure that occurs during transportation or as part of the work can cause physical and mental problems for humans. It becomes more and more important for drivers of construction machinery due to permanent and extreme exposure against vibration. Various studies show that the vibration at a specific frequency range or under larger excitation amplitude can lead to physical responses in different parts of the human body which may be deleterious to human health. Table 1 shows the effects of the vibration at specific frequencies on components of the human body. However, considering effects of WBV on human health, the resonant frequencies for the entire human torso are considered between 4 and 6 Hz for the lower spine and pelvis, and between 10 and 14 Hz for the upper torso with forward flexion movements of the upper vertebral column respectively. The dominant frequency of vibrations transmitted through the seat to the driver is often below 20 Hz. However, the frequencies above 20 Hz also occur occasionally when construction machinery is working due to the poor construction environment, such as land mixed with stones, a bumpy slope, etc. Discomfort could be increase with acceleration from 2 to 20 Hz or higher in the vertical vibration. In addition to the frequency, magnitude, and duration of exposure are also other primary factors that affect human health. Vibrations below 0.01 m/s² are rarely felt and vibrations above 10 m/s² are assumed to be hazardous. When the vibration root mean square (RMS) magnitude exceeds 2.0 m/s², driver feel extremely uncomfortable. The duration of WBV can be reflected in the health guidance caution zone (HGCZ). The magnitude of the upper limit of the HGCZ decreases as the duration of the exposure increases due to the energy of vibration exposure being equivalent. Therefore, the longer duration of acceleration with lower magnitude can be equal to the shorter duration of acceleration with larger magnitude or the shorter duration of excitation with larger amplitude.

In general, the ISO-2631 standard is the most widely used to assess the effect of the WBV exposure, which describes the measurements and analysis for evaluating the vibrations transmitted to the human body through the supporting objects. ISO 2631-1 defines the methods to measure random, periodic, and transient WBV, which is relevant to human health, comfort, perception,
and motion sickness. Meanwhile, the related level of comfort is given in the ISO-2631 standard. According to the ISO 2631 standard, the range of vibration frequencies affecting health, activities, and comfort is between 0.5 and 80 Hz. Frequencies out of that range (below 0.5 Hz and above 80 Hz) are considered as not important regions for health evaluation.

### Vibration of cab floor

Cab Floor vibration is a common phenomenon in construction machinery working. The vibration source is one of the prerequisites for the simulation of the seat suspension system. The vibration of most construction machinery vehicles is produced by the engine, the main/tail rotor, and the transmission, which are mechanically connected together. The variable speed motion is carried out by the engine governor, and the vibration is finally passed through the seat to the driver, which makes the whole body of driver in vibration (Whole Body Vibration). In this paper, taking the same type of loader as an example, the frequency of the vehicle generating vibration source is extracted, as shown in Table 2. The data in the right side of the table can be used to identify the frequency component in the frequency domain diagram, that is, the vibration signal of the cab floor can be reproduced for simulation. In addition, the amplitude of the seat model also depends on the external natural conditions during the operation of the construction machinery (such as, the soft degree of the road surface, the slope turning road, and other different road conditions). And the excitation source is changeable and the vibration source propagation path is long, resulting in the external excitation at the connection between the cab floor and seat connection very unstable. For this reason, it is difficult to obtain the vibration amplitude data, thus the data in other reference is referred. Fast Fourier transform (FFT) is used and the vibration signal is converted from time domain to frequency domain. The amplitude of frequency component of the vibration signal for the cab floor is reproduced, as shown in Table 3. Since the effects of the vibration frequencies above 50 Hz on the human body gradually decreases according to the ISO-2631, the frequencies above 50 Hz are ignored when the vibration signal of the cab floor is reproduced. Figure 9 shows the time domain and frequency domain diagram of the vertical vibration signal of the cab floor and the reproduced signal can be simulated as the excitation signal of the floor.

### Dynamic analysis of system

Taking a medium loader as an example, the weight of the cab seat is set to 12 kg, the mass range of the human body is 50–85 kg, and the weight range of the mass block in the model is 62–97 kg. Therefore, the fluctuation difference of mass accepted for the suspension system is 35 kg. According to the design requirements of NSS, it is necessary to force the horizontal spring to be

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### Table 2. Vibration sources and frequencies of loader.

| Vibration source    | Speed(r/min) | Frequency(Hz) |
|--------------------|--------------|---------------|
| Main rotor         | 336          | 5.2           |
| Tail rotor         | 591          | 12.7          |
| Hydraulic pump     | 3200         | 19.5          |
| Rotor drive shaft  | 3700         | 25.3          |
| Transmission input shaft | 5500 | 32.6          |
| Oil cooler         | 5800         | 41.7          |
| Rotor brake        | 6300         | 53.4          |

### Table 3. The amplitude of frequency component of the vibration signal for the cab floor.

| Frequency | Amplitude (g) | Amplitude (m/s²) |
|-----------|---------------|------------------|
| 5.2       | 0.052         | 0.59             |
| 12.7      | 0.065         | 0.73             |
| 19.5      | 0.136         | 1.58             |
| 25.3      | 0.017         | 0.14             |
| 32.6      | 0.041         | 0.36             |
| 41.7      | 0.043         | 0.37             |

### Figure 9. The reproduced vibration signal of cab floor in vertical direction: (a) time domain map and (b) frequency domain map.
kept to be placed horizontally in the static balance. In addition, under the condition of keeping the stiffness of the supporting spring unchanged, the maximum floating range is 80 mm in the vertical direction of the designed seat. According to the calculation of the stiffness of the vertical spring, additional displacement should be required for no load as follow

\[
\frac{(50\sim85)K_g \cdot 9.8N}{40N/mm} = 12.25\sim20.83mm \tag{23}
\]

The equation (23) describes the distance to be required when the system reaches the equilibrium position at the moment the driver first takes his seat. In this paper, the value is selected to be a little larger, and in order to reduce the displacement deviation caused by the input of different mass of the system, the stiffness of the support spring is increased moderately.

### Evaluation criteria for suspension systems

There are many different methods to evaluate the vibration of the system. At present, weighted root mean square RMS is widely used to evaluate the suspension vibration. However, many studies have proved that using RMS method will underestimate the effects of vibration characteristics with many peaks. Therefore, the four-time power vibration dose value VDV method is used to evaluate the vibration characteristics of suspension system, due to the VDV method more sensitive to the peak value of vibration. Its expression is as follow

\[
VDV = \left(\frac{1}{T} \int_0^T [a_w(t)]^4 dt\right)^{1/4} \tag{24}
\]

where the unit of VDV is \(m/s^1.75\) or \(rad/s^1.75\), \(a_w(t)\) is the instantaneous frequency weighted acceleration, \(T\) is the duration of the measurement.

To evaluate the effectiveness of the seat suspension system and reduce the vibration transfer amplitude, the frequency weighting function is considered based on the ISO 2631 standard, and seat effective amplitude transmissibility (SEAT) is adopt, which is defined as

\[
SEAT = \frac{(\ddot{x}_c)_{RMS}}{(\ddot{x}_f)_{RMS}} = \frac{\sqrt{\frac{1}{T} \int_0^T (w_k(t) \cdot z_w)^2 dt}}{\sqrt{\frac{1}{T} \int_0^T (w_k(t) \cdot z_{sw})^2 dt}} \tag{25}
\]

where \((\ddot{x}_c)_{RMS}\) is seat weighted RMS acceleration, \((\ddot{x}_f)_{RMS}\) is weighted RMS acceleration for cab floor.

To evaluate the vibration transfer characteristics of NSS suspension system in frequency domain, the equation (26) is used to calculate the vibration transmissibility.

\[
T_{trans} = G_{RX}/G_{BB} \tag{26}
\]

where \(G_{RX}\) is the cross power of vibration signals of cab floor and seat, \(G_{BB}\) is the auto power of vibration signals of cab floor in the vertical direction. Calculation equation is as follow

\[
\begin{align*}
G_{RX} &= \int B(t)X^*(t - \tau)dt \\
G_{BB} &= \int B(t)B^*(t - \tau)dt
\end{align*} \tag{27}
\]

where \(B\) and \(X\), respectively represent floor vibration and seat bottom vibration, \(\tau\) is the time delay between signals, * is conjugate signal. The frequency with transmissibility ratio greater than 1 represents the amplified frequency through the vibration transfer path.

### Optimization Model of Cab Seat Suspension System

The design variable is the stiffness of the seat suspension system and the optimization model of low frequency response for cab suspension system is as below

\[
\begin{align*}
\text{Min} & \quad V \left( \frac{T_{trans}}{T} \right) \\
\text{Min} & \quad SEAT \\
1.2 \leq & \quad \gamma \leq 1.4, \quad \alpha = 0.25 \\
K' & = K_{eq}/(K_v + K_c) \\
r & = \frac{\omega_{exc(Min)}}{\omega_n} \geq 4 \\
\omega_n & = \sqrt{\frac{K_{eq}}{M}}
\end{align*}
\]

where \(\omega_{exc}\) is excitation frequency, \(\omega_n\) is natural frequency of the system, \(K_{eq}\) is the total vertical stiffness of suspension structure. As shown in Figure 6, the value of the dimensionless equivalent stiffness is \(K' = 0.3\). The established dynamic model is simulated and calculated by using the MATLAB, and the constraint optimization method is used to solve the problem. The design domain is discrete points, and then the optimal solution is searched. Considering the actual engineering design, the parameter values are generally rounded. Table 4 shows the structural parameters of the designed seat suspension system.

### Results

Based on the above NSS suspension system dynamics model, the displacement response curve and acceleration response curve of the seat are obtained by optimization design, as shown in Figure 10. The vibration signal of the floor is compared with the vibration signal transmitted to the seat. The results show that the
The response amplitude of the optimized system is obviously reduced, which indicates that the suspension model based on NSS effectively reduces the vibration transmitted to the seat through the cab floor. At the same time, Table 5 shows the comparison between the vibration characteristics of cab floor after optimization and the vibration characteristics transmitted to the seat. It can be seen that the VDV and RMS values of the seat in the vertical vibration direction are respectively 0.1936 and 0.1537. Compared with the vibration of cab floor, its vibration amplitude and peak amplitude both decreased by 87% and 86%. It also illustrates that the vibration isolation performance of the seat suspension system is better. Besides, according to the ISO-2631 standard, the results in Table 5 also show that the driver’s ride comfort and vibration environment have been obviously improved.

Since VDV and RMS only reflect the main characteristics of vibration signals, so the performance of suspension system can be evaluated intuitively by calculating $SEAT$ and $T_{trans}$. The evaluation results after optimal design are shown in Table 6. The values of $SEAT$ and $T_{trans}$ are respectively 0.0526 and 0.0492. Compared with other types of seat suspension systems, the vibration transmissibility rate of the seat decreases, which illustrates that the NSS suspension system with TIE has the enhanced vibration isolation performance.

To analyze the difference between the vibration frequency transmitted to the driver and the resonance frequency of human body, the vibration characteristics of the seat are analyzed in the frequency domain in this paper. The vibration FFT spectrum transmitted to the driver through the suspension structure is shown in Figure 11. The results show that for the optimization model the vibration frequency transmitted to the driver changes within 1–50 Hz, in which the minimum frequency value is 4.7 Hz. Because the human body is sensitive to certain special frequencies, it will cause diseases in the human body or in a certain part when the driver has been in these vibrational frequencies for a long time. Some studies have shown that 4 Hz is a critical frequency of human body vibration, and 4.7 Hz is very close to 4 Hz. Since the internal damping of the human body and the low amplitude transmitted to the seat, the frequency 4.7 Hz will not cause the driver’s physical discomfort. Moreover, none of the other frequency values in the spectrum are near the other critical frequency values of the human body. To sum up, the design of the seat suspension system has no adverse effect on the health of the driver after being vibrated.

### Table 4. Parameter Setting of Seat Isolation System.

| Parameter | Values |
|-----------|--------|
| $L_0$/mm  | 200, 220, 230, 240, 250, 260 |
| $a$/mm    | 260, 275, 285, 300, 325, 351 |
| $b$/mm    | 180 |
| $K_c$/(N/mm$^3$) | $1.4 \times 10^{-4}$ |
| $K_v$/(N/mm) | 28, 34, 40, 46, 52, 58 |
| $K_h$/(N/mm) | 7.0, 8.5, 10, 11.5, 13, 14.5 |
| $M$/kg     | 62–97 kg |

### Table 5. Vibration characteristics of cab floor and vibration characteristics transmitted to seat after optimization.

| Vibration source position | Cab floor | Seat (transmitted to driver) |
|---------------------------|-----------|-----------------------------|
| VDV($m/s^{1.75}$)         | 1.5268    | 0.1936                       |
| RMS($m/s^2$)              | 1.0975    | 0.1537                       |
| Driver’s ride comfort      | Poor comfort and feel apparent discomfort | Good comfort, and not cause adverse disease |
Besides, Figure 12 shows the vibration spectrum contrast of the cab floor and the driver. It can be seen that the acceleration amplitude is obviously reduced. Also, the spectrum diagrams of the seat and the cab floor are similar, which illustrates that in fact there is no frequency regulation occurring in the vibration transfer path.

In order to study the influence of different loads and distance ratio $\lambda$ on the performance of suspension system, based on the optimization model, the simulations are carried out according to different driver weights and the uncertainty of distance ratio. The comparative analysis results are shown in Table 7. $SEAT$ values of the seat suspension system are given in the table. When the load $\geq 80$ kg, the value of $SEAT$ becomes larger with increase of the uncertainty of distance ratio, and the vibration transmissibility rate also increases. When the load $< 80$ kg, an decrease in the uncertainty of the spring stiffness factor first leads to the smaller $SEAT$ value. It indicates that the performance of the suspension system has been improved. And then the $SEAT$ value increases and the vibration isolation performance of the system decreases gradually. Table 8 shows the $T_{trans}$ values of the seat suspension system under different percentages of uncertainty. Results show that as the increases of uncertainty of distance ratio, the value of $T_{trans}$ increases gradually and the vibration isolation performance of seat decreases. Since the evaluation criteria between $SEAT$ and $T_{trans}$ values are different, their difference is that the $SEAT$ calculation results express the reduction of the vibration amplitude itself, while the $T_{trans}$ values emphasize the relative dependence between the excitation signal and the seat vibration signal. Furthermore, the results from Tables 7 and 8 show that the performance of the suspension system is not significantly changed by the variations of load mass, which indicates that the change of the driver’s weight in the range of 65–90 kg has no effect on the performance of the seat suspension system. To sum up, even if there is uncertainty of distance ratio and change

### Table 6. Vibration sources and frequencies of loader after optimization.

| Vibration transmissibility                              | SEAT  | $T_{trans}$ |
|---------------------------------------------------------|-------|-------------|
| Seat suspension system with NSS                         | 0.0681| 0.0522      |
| Seat suspension system with NSS (without TIE)           | 0.0872| 0.0753      |
| Seat suspension system with only vertical spring stiffness| 2.1937| 0.1463      |
| Seat without vibration isolation system                 | I     | I           |

### Table 7. $SEAT$ values of NSS suspension systems under different uncertainties.

| Percentage | 70%  | 50%  | 20%  | 10%  | 5%   |
|------------|------|------|------|------|------|
| 65 kg      | 0.3218 | 0.2076 | 0.1252 | 0.2241 | 0.2493 |
| 70 kg      | 0.3387 | 0.2205 | 0.0966 | 0.1338 | 0.1546 |
| 75 kg      | 0.4691 | 0.3662 | 0.0847 | 0.1302 | 0.1363 |
| 80 kg      | 0.5075 | 0.4526 | 0.1291 | 0.0814 | 0.0695 |
| 85 kg      | 0.5872 | 0.5417 | 0.2005 | 0.1464 | 0.1126 |
| 90 kg      | 0.6288 | 0.6054 | 0.2348 | 0.2159 | 0.2016 |

About 65–90 kg in the weight of seat and body, Percentage represents percentage of uncertainty in distance ratio.

### Table 8. $T_{trans}$ values of NSS suspension systems under different uncertainties.

| Percentage | 70%  | 50%  | 20%  | 10%  | 5%   |
|------------|------|------|------|------|------|
| 65 kg      | 0.0813 | 0.0782 | 0.0757 | 0.0734 | 0.0716 |
| 70 kg      | 0.0785 | 0.0762 | 0.0745 | 0.0633 | 0.0609 |
| 75 kg      | 0.0776 | 0.0694 | 0.0671 | 0.0598 | 0.0582 |
| 80 kg      | 0.0692 | 0.0583 | 0.0538 | 0.0494 | 0.0471 |
| 85 kg      | 0.0682 | 0.0557 | 0.0511 | 0.0483 | 0.0465 |
| 90 kg      | 0.0543 | 0.0529 | 0.0472 | 0.0457 | 0.0448 |

About 65–90 kg in the weight of seat and body, Percentage represents percentage of uncertainty in distance ratio.
of load mass, the results shows that the designed NSS suspension system for construction machine has better vibration isolation performance and the change in the vibration transmissibility also can be achieved by the adjust of distance ratio of additional inerter elements especially when construction machine is in poor working conditions.

Conclusion
A cab seat suspension system based on NSS is designed in this paper. The different structural parameters of the system and its influence on the dynamic stiffness characteristics are respectively analyzed. The desired configuration parameter range of the suspension system is obtained. Meanwhile, the nonlinear dynamic model of NSS suspension system is established. For seat suspension structure of construction machinery cab, the optimization model is also put forward, and then the reasonable design parameters are finally adopted. Through simulation and comparative analysis, the vibration transmissibility is studied. The results show that the acceleration and displacement amplitude of the optimized seat suspension system are obviously reduced. According to the ISO-2631 standard, compared with the vibration source of the cab floor, the RMS and VDV values of the seat in the vertical vibration direction are reduced by 87% and 86%, respectively, and the vibration transmissibility $SEAT$ and $T_{trans}$ of seat are both reduced.

Through the analysis of the frequency domain response of the seat suspension system, the comparison results show that according to the ISO-2631 standard, the peak frequency of vibration transmitted to the driver avoids the key frequency which is easier to cause human resonance. Based on the optimization model, the influences of uncertainty of distance ratio $\lambda$ and different driver weights on suspension system are discussed. Study results show that the change of driver’s weight in the range of 65–90 kg has no effect on the performance of seat suspension system. Based on the NSS, the seat suspension system has good vibration isolation performance, improving the working environment of the driver and the ride comfort of the seat. The results of research provide scientific basis for the design and improvement of seat suspension system in the future.

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ORCID iD
Xin Liao https://orcid.org/0000-0001-7336-6052

References
1. Nguyen SD, Nguyen QH and Choi SB. A hybrid clustering based fuzzy structure for vibration control – part 2: an application to semi-active vehicle seat-suspension system. Mech Syst Signal Process 2015; 56-57: 288–301.
2. Swapnil K. Isolation mount assembly. U.S. Patent application No 16/051,785, 2019.
3. Deshmukh SR, Krishnan Balaji NS, Saharabhudhe S, et al. Designing of control strategy for high voltage battery isolation in an electric vehicles. In: 2019 IEEE 5th International Conference for Convergence in Technology (I2CT), Bombay, India, 29–31 March 2019. IEEE.
4. Liu Z and He H. Sensor fault detection and isolation for a lithium-ion battery pack in electric vehicles using adaptive extended kalman filter. Appl Energy 2017; 185: 2033–2044.
5. Griffin MJ. Handbook of human vibration. 1st ed. Cambridge: Academic Press, 2012.
6. Friesenbichler B, Nigg BM and Dunn JF. Local metabolic rate during whole body vibration. J Appl Physiol 2013; 114: 1421–1425.
7. van Eijk J and Dijksman JF. Plate spring mechanism with constant negative stiffness. Mech Mach Theory 1979; 14: 1–9.
8. Lee CM, Goverdovskiy VN and Temnikov AI. Design of springs with “negative” stiffness to improve vehicle driver vibration isolation. J Sound Vib 2007; 302: 865–874.
9. Liu D, Liu Y, Sheng D, et al. Seismic response analysis of an isolated structure with QZS under near-fault vertical earthquakes. Shock Vib 2018; 2018: 1–12.
10. Junshu H, Lingshuai M and Jinggong S. Design and characteristics analysis of a nonlinear isolator using a curved-mount-spring-roller mechanism as negative stiffness element. Math Probl Eng 2018; 2018: 1–15.
11. Sun X and Jing X. Multi-direction vibration isolation with quasi-zero stiffness by employing geometrical nonlinearity. Mech Syst Signal Process 2015; 62-63: 149–163.
12. Liu C and Yu K. A high-static–low-dynamic-stiffness vibration isolator with the auxiliary system. Nonlinear Dyn 2018; 94: 1549–1567.
13. Zhang Z, Zhang YW and Ding H. Vibration control combining nonlinear isolation and nonlinear absorption. Nonlinear Dyn 2020; 100: 2121–2139.
14. Li D, Zhao S, He Y, et al. Dynamic analysis of quasi-zero stiffness isolator with time delay control. Vib Shock 2018; 37: 49–55.
15. Lu Z, Brennan MJ and Chen LQ. On the transmissibilities of nonlinear vibration isolation system. J Sound Vib 2016; 375: 28–37.
16. Sarlis AA, Pasala DTR, Constantinou MC, et al. Negative stiffness device for seismic protection of structures. *J Struct Eng* 2013; 139: 1124–1133.

17. Palomares E, Nieto AJ, Morales AL, et al. Numerical and experimental analysis of a vibration isolator equipped with a negative stiffness system. *J Sound Vib* 2018; 414: 31–42.

18. Davoodi E, Safarpour P, Pourgholi M, et al. Design and evaluation of vibration reducing seat suspension based on negative stiffness structure. *Proc IMechE, Part C: J Mechanical Engineering Science* 2020; 234: 4171–4189.

19. Dong G, Zhang X, Xie S, et al. Simulated and experimental studies on a high-static-low-dynamic stiffness isolator using magnetic negative stiffness spring. *Mech Syst Signal Process* 2017; 86: 188–203.

20. Yao H, Chen Z and Wen B. Dynamic vibration absorber with negative stiffness for rotor system. *Shock Vib* 2016; 2016: 1–13.

21. Hoy J, Mubarak N, Nelson S, et al. Whole body vibration and posture as risk factors for low back pain among forklift truck drivers. *J Sound Vib* 2005; 284: 933–946.

22. Newell GS and Mansfield NJ. Evaluation of reaction time performance and subjective workload during whole-body vibration exposure while seated in upright and twisted postures with and without armrests. *Int J Ind Ergon* 2008; 38: 499–508.

23. ISO. Mechanical vibration and shock: evaluation of human exposure to whole-body vibration. Part 1, general requirements: international standard ISO 2631-1: 1997 (E). Geneva: ISO, 1997.

24. Knothe K and Stichel S. Human perception of vibrations-ride comfort. In: Knothe K and Stichel S (eds) *Rail vehicle dynamics*. New York, NY: Springer, 2017, pp.141–157.

25. Ji X. Evaluation of suspension seats under multi-axis vibration excitations—a neural net model approach to seat selection. PhD Thesis, The University of Western Ontario, Canada, 2015.

26. Carletti E and Pedrielli F. Tri-axial evaluation of the vibration transmitted to the operators of crawler compact loaders. *Int J Ind Ergon* 2018; 68: 46–56.

27. Wickramasinghe VK. Dynamics control approaches to improve vibratory environment of the helicopter aircrew. PhD Thesis, Carleton University, Canada, 2013.

28. Grabau PJ. *The simulation of vibrations experienced by patients during helicopter winching and retrieval*. PhD thesis, James Cook University, Australia, 2016.