Original Article

Radial mechanics and simulation analysis of main-auxiliary structure metal spring wheel

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A B S T R A C T
Aiming at the various special circumstances of automobile driving process, a main-auxiliary structure metal spring wheel is proposed, which prevents effectively the abrupt accident caused by blown or punctured tire. Its working principle is described in detail. Besides, the simplified mechanical models of the wheel, the main and auxiliary spring are established. The radial stiffness FEM models of main and auxiliary spring are established. The radial deformation and contact pressure of the wheel are analyzed when the maximum deformation of main spring is 6.22 mm. The radial stiffness curve of a single auxiliary spring is obtained. The relationship among the radial force, the radial deformation, the grounded angle and the grounded circles of auxiliary spring is deduced combining the deformation and stress characteristics of the spring mesh. Finally, the reliability of the mechanical and simulation models is verified by the radial stiffness test.

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1. Introduction

Tire is the only contact part between the vehicle and the ground. It is used to support the vehicle weight and transmit driving force, braking force and steering force, meanwhile buffer the ground impact, ensuring the handling stability and ride comfort [1]. Non-pneumatic tires do not rely on the inflatable structure, which avoids tire accidents from the root [2]. According to the structure form, non-pneumatic tire can be divided into solid, spoke radial, spoke non-radial and metal spring tire [3]. Among them, the spring tire is produced by the hooking, weaving and assembling of the springs with different structure, size and number. The spring tire makes use of the spring's lateral or radial vibration damping characteristics. This kind of tire structure is much more flexible and diversified [4]. In 2009, the US Goodyear Rubber Company and NASA have jointly developed a new spring tire-Spring Tire [5]. And Zhao et al. from Nanjing University of Aeronautics and Astronauts, proposed a new type of spring elastic tire [6].

In this paper, a main-auxiliary structure metal spring wheel is designed composed of main spring, spring mesh, fixed ring and rim. The radial stiffness characteristics of the main and auxiliary spring are analyzed theoretically and simulated. Finally, the radial stiffness is obtained through the test and compared with the theoretical and simulation results.

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2. Composition of the main-auxiliary structure wheel

This wheel is made by the mutual winding, hooking and weaving of the main spring and auxiliary springs. It generally consists of five parts including main spiral spring, spring mesh, rim, fixed belt and fixed ring. The structure is shown in Fig. 1. The main spring (the green section) assembles itself head-to-tail into a ring by welding. It is embedded in the arc groove of the rim, and the wire of main spring is correspondingly clamped into the groove of the fixed belt (the yellow section). The fixed belt is composed of 6 metal rings with the same structure and size. The fixed belt and the rim are fixed by the screw threads, which limit the circumferential and radial movement of the main spring. The circles of the auxiliary springs (the gray section) are the double of the main spring circles, and one end is tight, the other is free. The spring mesh is formed by the winding and weaving of the auxiliary springs around each ring and the clearance between two rings of the main spring. The fixed rings (the blue section) are used to fix the auxiliary spring mesh from both sides of the rim through the threaded hole. Fig. 2 shows the details of the weaving process. Fig. 3 presents the exploded figure and the diagram of assembly process according to the installation procedure step by step [7–9].

3. Mechanical properties analysis of the wheel

3.1. Radial force analysis of the main spring

3.1.1. The simplified mechanics model
The metal wheel adopts the rigid rim. When loaded, the wheel relies on the rim to compress the main spring and the auxiliary spring mesh, which contacts with the ground. In the force process, only the contact part is under stress, the rest is not, which belongs to bottom-bearing way. The mechanical interaction of the structure is complex due to the winding and hooking of the
springs. So the direct analysis is difficult. The fixed belt separates the main spring into independent metal rings with the same number of turns as the main spring. Compared with the main spring, the structural parameter of the auxiliary springs is smaller. Therefore, the wheel force is mainly borne by the main spring in static radial load. According to the above analysis, the radial mechanics model is simplified as follows:

1. The effect of the main spring pitch is ignored. The main spring is reduced to metal rings with the same number of turns, pitch, pitch diameter as the main spring.
2. The effect of the spring mesh on wheel radial stiffness is ignored. The simplified mechanical model of the main spring is shown in Fig. 4.

3.1.2. Force analysis of the ring

The rings are the force components in the simplified model. Combining with the force symmetry, half of the structure is taken for force analysis. The radial force of single ring is set for P, as shown in Fig. 5.

Where the bending moment of any ring section is:

\[ M = PR_1 \sin \phi \]  

(1)

Since the section dimension of the ring is far smaller than its radius of curvature, the deformation energy of axial force and shear force can be ignored. It only considers the effect of bending moment [10], its deformation energy is:

\[ U = \int_0^\pi M^2(\phi) \frac{R_1}{2EI} d\phi = \frac{\pi R_1^3 p^2}{4EI} \]  

(2)

where E is the elastic modulus of the main spring, which value is \(1.96 \times 10^{11}\) Pa. I is the inertia moment of the ring section to neutral axis, which value is \(\pi d_1^4/64\), d1 is the wire diameter of the main spring.

\( \delta \) is the displacement which P moves along its point of action. The work done by P is W.

\[ W = \frac{1}{2}P\delta \]  

(3)

According to the energy conservation theory:

\[ U = W \]  

(4)

By formula (2)-(4):

\[ \delta = \frac{\pi R_1^3 p}{2EI} \]  

(5)

In the case of small deformation, the vertical deformation of the ring is linear with the radial load. K = \(\frac{2EI}{\pi R_1^4}\) then by Hooke’s law:

\[ P = K\delta \]  

(6)

Suppose the deformation amount of the ‘i’ ring is \(L_i\) and the force is \(N_i\) in the process of grounding. It can be obtained that:

\[ N_i = K \Delta L_i \]  

(7)

According to the force symmetry of the wheel, the circles of grounded rings are odd or even, and the two kinds of force cases are analyzed respectively.

3.1.3. Force analysis of odd grounded circles

When the grounded circles are odd, the force diagram is shown in Fig. 6. \(R_3\) is the radius of the rim. \(R\) is the radius of the wheel that does not take into account the influence of the auxiliary spring, \(\phi\) is the angle between the adjacent rings of main spring, which value is 6.54°. \(\alpha\) is the grounded angle.

Assuming that the wheel is in a static radial load, and the load is F, the maximum deformation of the wheel in
the vertical direction is L. The radio of F and L is the radial stiffness of the wheel.

Assuming that there are i circles of spring rings contacting with the ground, each bears the force of \( N_i \). Then by the static equilibrium relationship:

\[
\begin{align*}
\sum X &= \begin{bmatrix} 0 & \sin \theta & \cdots & \sin \frac{n-1}{2} \theta & \sin \frac{n-1}{2} \theta \\ 1 & \cos \theta & \cdots & \cos \frac{n-1}{2} \theta & \cos \frac{n-1}{2} \theta \end{bmatrix} \begin{bmatrix} N_1 \\ N_2 \\ \vdots \\ N_i \\ \vdots \\ N_n \end{bmatrix} \\
&= \begin{bmatrix} 0 \\ F \end{bmatrix}
\end{align*}
\]

By the force symmetry of the main spring:

\[
\begin{align*}
N_2 &= N_3 = \ldots = N_{i-1} = N_i \\
\Delta L_2 &= \Delta L_3 = \ldots = \Delta L_{i-1} = \Delta L_i
\end{align*}
\]

With the increase of deformation amount, the circles of grounded rings increased by 2. The grounded angle, the maximum deformation, the grounded circles also gradually increases, and the relationship between the three is:

\[
\begin{align*}
0 &\leq \alpha < \theta & 0 &\leq \Delta L < R(1 - \cos \theta) & n = 1 \\
0 &\leq \theta < 2\theta & R(1 - \cos \theta) &\leq \Delta L < R(1 - \cos 2\theta) & n = 3 \\
2\theta &\leq \alpha < 3\theta & R(1 - \cos 2\theta) &\leq \Delta L < R(1 - \cos 3\theta) & n = 5
\end{align*}
\]

By the deformation coordination:

\[
\begin{align*}
(R - \Delta L_2) \cos \theta &= \ldots = (R - \Delta L_{i-1}) \cos \frac{i-1}{2} \theta = R \cos \theta \\
R \cos \alpha &= R - \Delta L_i
\end{align*}
\]

By Eqs. (8)–(11):

\[
F = \left\{ \begin{array}{ll}
KR(1 - \cos \alpha) & i = 1 \\
KR \left[ 1 - \cos \alpha \right] + \cdots + 2 \left( \cos \frac{i-1}{2} \theta - \cos \theta \right) & i = 3, 5, \ldots
\end{array} \right.
\]

To ensure that the wheel can withstand the rated load in the case of small deformation, it is assumed that the maximum grounded angle is 18°. In this range, \( \alpha \) is taken as different values, and the grounded angle, radial force, maximum deformation, the grounded circles, radial stiffness of the wheel can be obtained, as shown in Table 1.

### 3.1.4. Force analysis of even grounded circles

When the grounded circles are even, the force diagram is shown in Fig. 7.

| Table 1 – Deformation relation of odd grounded circles. |
|----------------|--------|---------|--------|-----------|
| \( \alpha \) | F/N   | \( \Delta L/mm \) | n/circle | K/N mm\(^{-1} \) |
| 3.27          | 7.62  | 0.21    | 1      | 36.29     |
| 6.55          | 30.13 | 0.83    | 1      | 36.29     |
| 9.82          | 142.3 | 1.86    | 3      | 108.90    |
| 13.09         | 299.31| 3.30    | 3      | 108.90    |
| 16.36         | 644.69| 5.14    | 5      | 184.66    |
| 18.00         | 829.09| 6.22    | 5      | 184.66    |

The same procedure may be easily adapted to obtain the following result:

\[
F = 2KR \left[ \cos \frac{\theta}{2} - \cos \alpha \right] + \ldots + \left( \cos \frac{i-1}{2} \theta - \cos \alpha \right) i = 2, 4, \ldots
\]

Similarly, the grounded angle, radial force, maximum deformation, the grounded circles, radial stiffness of the wheel are shown in Table 2.

### 3.2. Radial force analysis of the auxiliary spring

#### 3.2.1. Radial force analysis of single auxiliary spring

The sections and axes of the spring are shown in Fig. 8. The cross section that plan V goes through the axis of the spring is the oblique section. The angle between the plane V and plane V is \( \alpha \). Each plane is perpendicular to the center line of the spring. In the center of the spring section, the tangent of the center line is taken for t-axis, the normal line for n-axis, the subnormal line for b-axis. The t-axis is located in the tangent plane of spring center line. The n-axis is on the intersection of plane V and plane V. The b-axis is in the V plane.

In order to facilitate the force analysis, the radial mechanical model of the auxiliary spring is simplified as follows:

![Fig. 7 – Forced diagram of even grounded circles.](image)

| Table 2 – Deformation relation of even grounded circles. |
|----------------|--------|---------|--------|-----------|
| \( \alpha \) | F/N   | \( \Delta L/mm \) | n/circle | K/N mm\(^{-1} \) |
| 3.27          | 0     | 0.21    | 0      | 0         |
| 6.55          | 45.01 | 0.83    | 2      | 72.60     |
| 9.82          | 119.79| 1.86    | 2      | 72.60     |
| 13.09         | 328.87| 3.30    | 4      | 145.30    |
| 16.36         | 596.77| 5.14    | 4      | 145.30    |
| 18.00         | 829.08| 6.22    | 6      | 215.10    |

![Fig. 8 – Spring cross-section coordinate.](image)
As the helix angle of spiral spring is very small, take 
\[ \cos \alpha = 1, \sin \alpha = 0 \]
By the formula (14)–(19):

\[ T_t = \frac{FD_2}{2} \sin \delta + \frac{FD_1}{2} \sin \delta \cos \phi \]  \hspace{1cm} (20)

\[ M_b = -\frac{FD_2}{2} \cos \delta \sin \phi \]  \hspace{1cm} (21)

\[ M_n = \frac{FD_1}{2} \sin \delta \sin \phi \]  \hspace{1cm} (22)

According to the energy law, the axial force and strain energy corresponding to the shear force are ignored. The deformation calculation formula can be obtained.

\[ f = \int_0^1 \frac{T_t T_{1n}}{G l_p} ds + \int_0^1 \frac{M_b M_{1b}}{E l_p} ds + \int_0^1 \frac{M_n M_{1n}}{E l_n} ds \]  \hspace{1cm} (23)

\[ ds = \frac{D_2}{2} \frac{d \delta}{\cos \alpha} = \frac{D_2}{2} d \phi \]  \hspace{1cm} (24)

\[ l = \frac{\pi D_2 n_2}{\cos \alpha} \]  \hspace{1cm} (25)

where \( l_p \) is the polar moment of inertia, \( l_p = \frac{p D_2^4}{32} \). \( l_b \) and \( l_n \) are the moments of inertia around \( t \)-axis and \( n \)-axis. For the material with circular cross section, \( l_b = l_n = \frac{p D_2^4}{32} \). \( ds \) is a minor segment of the auxiliary spring. \( l \) is the length. \( T_{1n}, T_{1b} \) and \( T_{1n} \) are the corresponding torque and bending moments under the unit force.

The deformation amount of single circle ring for auxiliary spring is obtained.

\[ \Delta f = \int_0^{2\pi} T_{1r} T_{1n} \frac{D_2}{2} d \phi + \int_0^{2\pi} M_{1b} M_{1n} \frac{D_2}{2} d \phi + \int_0^{2\pi} M_{1n} M_{1n} \frac{D_2}{2} d \phi \]  \hspace{1cm} (26)

\[ = \frac{F}{2G l_p} \frac{\pi (D_2^2 + D_2^2)}{4} \sin^2 \delta + \frac{F}{2E l_p} \frac{\pi D_2^2}{4} \sin^2 \delta \]  \hspace{1cm} (27)

Since \( D_2^2 2D_2^2, 2G l_p 2E l_p, l_b = l_n \) and the compression stiffness of the auxiliary spring in straightened state is \( K = G D_2^4 / 8D_2^4 n_2 \).

\[ E = 2(1 + \mu) G \]

where \( H_2 \) is the length of the auxiliary spring, \( n_2 \) is the effective number of turns.

Since the spring pitch \( t_2 = H_2 / n_2 \), it can be known that \( \Delta f \) is the deformation amount which the auxiliary spring generates in the range of height \( t_2 \) from the formula (27). So the deformation amount along the arc of the auxiliary spring is as follows:

\[ f = \int_0^{t_2} \frac{1}{2(1 + \mu) H_2} \frac{1}{4} \cos^2 \delta dx + \frac{1}{2(1 + \mu) H_2} \frac{1}{4} \sin^2 \delta dx \]

\[ = \frac{F}{2(1 + \mu) H_2} \frac{1}{4} n_{max} \cos^2 \delta dx + \frac{F}{2(1 + \mu) H_2} \frac{1}{4} n_{max} \sin^2 \delta dx \]  \hspace{1cm} (28)

\[ = \frac{F}{2(1 + \mu)} \left[ \frac{D_2^2}{4} \frac{t_2}{D_2^2} - \frac{D_2^2}{2} \frac{t_2}{D_2^2} \sin^2 2D_2^2 \right] \]  \hspace{1cm} (29)

The influence of the axial force, helix angle and the minimum term is neglected in the above analysis. And in the actual deformation process, the deformation of auxiliary spring is
limited by the deformation of the main spring. Therefore, this expression can only be used as the reference for the relationship between the deformation and force of the auxiliary spring.

3.2.2. Force analysis of the auxiliary spring mesh

If the mutual hooking of main and auxiliary spring wires is neglected, the radial force of the auxiliary spring is similar to that of the main spring. According to the derivation process of the formula (12), the radial force expression of spring mesh can be obtained as follows.

\[
F = \begin{cases} 
K_a R_i (1 - \cos \omega) & \text{i} = 1 \\
K_a R_i \left[ (1 - \cos \omega) + \cdots + 2 \left( \cos \frac{1}{2} \theta_i - \cos \omega \right) \right] & \text{i} = 3, 5, \ldots 
\end{cases}
\]  

(29)

where \( \theta_i \) is the central angle between the auxiliary springs, which value is 3.27°. \( \alpha \) is the grounded angle. \( R_i \) is the distance from the outer end of the auxiliary spring to the center of the wheel.

4. Simulation of the wheel radial stiffness

In order to further verify the feasibility of the wheel design, this section uses the nonlinear finite element analysis software ABAQUS to simulate the radial stiffness of the simplified main spring model. And the correctness of the theoretical calculation is also verified [13,14].

4.1. Radial stiffness simulation of the main spring

Figs. 10, 11 and Figs. 12, 13 respectively show the radial deformation and the contact pressure nephogram of the model which circles of grounded rings are odd and even, when the wheel’s grounded angle is 18° (i.e., a vertical displacement of 6mm is applied to the ground). It can be seen from the figures that there are respectively five and six circles of spring rings that contact with the ground in the two conditions, which is consistent with the theoretical results. And with the middle plane as the symmetry plane, the deformation decreases in turn. Any grounded ring has the largest deformation in the area that contacts with the ground, and gradually decreases along the vertical direction. The area that the main spring contacts with the ground shows a certain angle with the vertical direction, which is called the helix angle. From the simulation results in Fig. 14, it can be seen that the radial stiffness of the wheel increases with the increase of the deformation amount. Under the assumption of permissible deformation, the wheel can withstand a radial maximum load of 700N, proving the carrying capacity of the wheel. There is a same changing trend for the theoretical calculation and simulation results, but the simulation results are smaller. With the increase of the deformation amount, the error between two is gradually increasing. When the deformation reaches 6.22mm, the error is 13.97% and 13.33%.

![Fig. 10 – Radial deformation diagram of five grounded circles.](image1)

![Fig. 11 – Radial deformation diagram of six grounded circles.](image2)
It can be known that the grounded ring of the auxiliary spring has the largest deformation. With the grounded area as the symmetry center, the deformation of both sides decreases in turn. There is no deformation at the fixed points of the auxiliary spring. When the radial displacement reaches the maximum of 11.22 mm, there are 4 circles of spring rings in contact with the ground.

The contact pressure between the auxiliary spring and the ground under different radial displacements can be obtained by applying different displacement loads to the pavement control points. Fig. 16 shows the radial stiffness curve of the single auxiliary spring.

It can be seen from the figure that the radial force is increasing with the increase of the radial deformation. But the slope of the curve becomes smaller, that is, the radial stiffness of the single auxiliary spring is getting smaller. The radial stiffness curve is stable and the changing trend is slow. The radial stiffness tends to be linear before the deformation amount is less than 5 mm, and the stiffness value is approximately 0.4 N/mm. When the deformation is larger than 5 mm, the nonlinear factors increase and the curve becomes nonlinear. When the deformation amount reaches a maximum of 11.2 mm, the radial stiffness is approximately 0.35 N/mm. Therefore, in order to facilitate the next analysis and calculation, the overall stiffness of the auxiliary spring is set linear. The stiffness from the beginning to the deformation of 5 mm is taken as the approximate radial stiffness, which value is $K_1 = 0.4$ N/mm.

Substituting $K_1$ into the formula (29), the radial deformation amount is taken as different values in the range of the maximum radial deformation of 11.22 mm. Table 3 shows the values of the grounded angle, radial force, radial deformation, grounded circles and radial stiffness.

### 5. Stiffness characteristics test of the main-auxiliary structure wheel

With the assembly method determined above, the overall size of the wheel specimen is 9270 x 75 mm, as shown in Fig. 17. And the springs used in the wheel are purchased directly from the spring factory with the material of spring steel (65Mn), which is consistent with the setup of the material property in the simulation analysis.

![Fig. 12 – Contact pressure nephogram of five grounded circles.](image)

![Fig. 13 – Contact pressure nephogram of six grounded circles.](image)

![Fig. 14 – Result comparison between calculation and simulation.](image)

The main reason is that the axial force, shear force and the torque generated by the spring pitch are neglected in the process of theoretical calculation. And the main spring is approximated as unrelated metal rings. However, the whole force analysis is carried on in the simulation.

### 4.2. Radial stiffness simulation of the auxiliary spring

After the wheel assembly, the spring mesh is far away from the wheel axle due to the whole outward expansion of the spring. And in the radial force process, the auxiliary spring mesh bears the force and deformation firstly because of its small overall stiffness, which cannot withstand the load alone. Until the auxiliary spring mesh contacts with the main spring, it begins to bear the radial load along with the main spring. Fig. 15 shows the radial deformation results of auxiliary spring when the radial displacement is applied to the pavement control point.
At this time, the wheel is in the vertical load and ‘zero’ position. A data measurement is performed for each turn of the handle from ‘zero’ position. The radial displacement is determined by recording the rotating revolutions of the handle. The measurement is stopped until the radial deformation amount reaches 11.22 mm. Reset the wheel and measure again in the same way. Measure twice at each test points and three different points have to be selected for a single wheel for testing [15]. The radial stiffness curve under the two grounded states (odd and even grounded circles of rings) is shown in Fig. 19. From the curve, we can see that with the increase of the radial deformation, the radial stiffness of the wheel is increasing. For the two grounded states, the radial stiffness trend is basically the same. Among them, the various range of the wheel stiffness is 0–4.48 N/mm in segment a–b, which has a small changeable scale. The main reason is that the spring mesh is subjected to force deformation firstly in the early stage, and the main spring is not bearing deformation. So, the curve mainly reflects the stiffness change trend of the spring mesh.

In segment b–e and b–e’, the slope of the curve increases rapidly. The main reason is that the main spring starts to bear the load, and with the increase of the deformation amount, the ground circles of the rings increase by 2. The wheel stiffness increases rapidly and presents certain nonlinearity. Among them, single ring of the main spring contacts with the ground in segment b–c. There is an accelerating rise of the stiffness and the overall deformation is small in this stage. The stiffness increases to 44.19 N/mm. Three circles of rings contacts with the ground in segment c–d, and the overall stiffness increases significantly with the maximum of 152.76 N/mm. Five circles of rings contacts with the ground in segment d–e, the radial stiffness is further increased to 190.58 N/mm. Similarly, even circles of rings contacts with the ground, the stiffness
also showed a rapid increase in stage. In segment b–c’, two circles of rings contacts with the ground, and the stiffness increases to 79.67 N/mm. Four circles of rings contacts with the ground in segment c’–d’, the stiffness is further increased to 186.55 N/mm. Six circles of rings contacts with the ground in segment d’–e, the stiffness is 227.98 N/mm. Ignoring the deformation amount of the spring mesh from the wheel starts to contacts with the ground to the main spring starts to bear the load, that is, the segment a–b of the curve in Fig. 19. The contrast curve of the calculation, simulation and test results is shown in Fig. 20.

It can be seen from Fig. 20 that there is the same various trend of the stiffness for the three results. The grounded circles of rings increase by 2 with the increase of the deformation. And the change of the stiffness appears nonlinear. But there is a certain deviation for the three results, which is the smallest in the simulation, followed by the theoretical calculation, the largest in the test. As the axial force, the shear force and the torque caused by the pitch are neglected in the theoretical calculation, the results are larger than that in simulation. Compared with the test results, the theoretical results are smaller. In the actual test, the deformation is very small when the wheel starts to contact with the ground. So the spring mesh has little effect on the radial stiffness, and the two curves are basically consistent. When the deformation increases to 2 mm, the spring mesh becomes tighten, which limits the tangential slip of the main spring. And the impact on the stiffness increases significantly. Besides, there is a surface-to-surface contact between the main spring and the rim in actual operation, which is equivalent to reducing the circumferential length of the main spring and increasing the ring stiffness. Thus the stiffness in the test results is significantly larger than that in the theoretical calculation. When the deformation reaches to the maximum, the deviation in the two grounded states are 17.37% and 16.77%, respectively. However, the theoretical and simulation results can reflect the trend of wheel stiffness and predict the overall stiffness, which has some reference significance in the wheel design.

6. Conclusions

Through the final test of the wheel radial stiffness test, the following conclusions can be obtained.

(1) As for the radial stiffness calculation, the mechanical analysis model is established by simplifying the main and
auxiliary spring structure in the paper. Its numerical calculation results are similar to that by the simulation and test. So the simplified model can be used to guide the structure and size design of this wheel, to meet the wheel performance requirements.

(2) Comparing the simulation and test results, there are some errors. But their trend is the same, and the error is within the allowable range. The next work is to improve the simulation accuracy by modifying the boundary conditions and perfect the model to establish the effective method for simulation analysis of the metal wheel.

Conflicts of interest

The authors declare no conflicts of interest.

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