# Research on application of artificial intelligence model in automobile machinery control system

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**Abstract:** To improve the kinematics and ride comfort of the vehicle suspension system, this paper combines the ADAMS software with the artificial intelligence optimisation algorithm to simulate and optimise the suspension system of a certain type of off-road vehicle. The front suspension of a 1/2 virtual prototype model is established in ADAMS/view, and the kinematics analysis of the prototype model is made. In order to improve the kinematics characteristics of the front suspension, the genetic algorithm and the immune algorithm are used to optimise the positioning parameters of the front suspension based on the kinematics analysis of the suspension. The dynamic model of the whole vehicle is established in ADAMS/view, and the vehicle ride comfort is simulated and calculated. The spring stiffness and shock absorber damping of the rear suspension are optimised by using the immune algorithm, and the purpose of improving the ride comfort of the vehicle is realised.

**Keywords:** artificial intelligence; automobile; mechanical control system.

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**Biographical notes:** Xiuyu Chen is currently an Associate Professor in Jiujiang Vocational and Technical College. He received his degree from the Nanchang University. His research concerns automotive machinery and vehicle engineering.

## 1 Introduction

Because of the powerful function of ADAMS software, it is widely used in automobile industry. Using ADAMS software to develop the vehicle virtual prototype model, the research is very convenient. ADAMS software effectively combines multi-body system dynamics, kinematics analysis and optimisation to make mechanical optimisation design more convenient and flexible (Jalalkamali et al., 2015).

However, in the aspect of optimisation design, the traditional optimisation algorithms are mainly used in ADAMS software at present, such as generalised decreasing gradient algorithm and quadratic programming algorithm. These traditional optimisation algorithms are difficult to achieve ideal results in practical applications. In recent years, biomimetic optimisation algorithms based on simulated biological systems, such as genetic algorithm, immune algorithm and other artificial intelligence optimisation algorithms, have been developed rapidly and successfully applied in many fields. Most of these algorithms are global optimisation search algorithms, which are much better than the traditional optimisation methods in terms of large scale population search ability, parallelism and so on. Genetic algorithms and immune algorithms are also widely used in mechanical optimisation. For example, the basic principles and methods of genetic algorithms are introduced in detail in Chen Lunjun’s book on genetic algorithm for mechanical optimisation design. Genetic algorithm is used to solve the problem of mechanical optimisation design, which avoids the local optimal appearance of traditional methods and achieves the global optimal solution. Xie Kaigui, from Chongqing university, elaborated the comparative analysis of immune algorithm and other stochastic optimisation algorithms systematically (Azizi, 2017). These studies fully prove the advantages of artificial intelligence optimisation algorithm and have great potential in automotive system optimisation.

However, there are few examples of the combination of artificial intelligence optimisation algorithm and ADAMS software in the above research, so how to combine artificial intelligence optimisation algorithm with mechanical system simulation analysis software ADAMS is very necessary. Chen Xiaokai and others combined genetic algorithm with mechanical system simulation and analysis software ADAMS, with genetic algorithm to construct optimiser, and optimised design of automobile five-bar suspension, which achieved good results. But the optimisation process is more complex, and only on the suspension system positioning parameter optimisation design (Padhy et al., 2017). What’s more, for ADAMS, there is no related literature on the performance optimisation analysis of vehicle with artificial intelligence optimisation algorithm.

## 2 State-of-the-art

In 1970s, China began to study the theory, method and application of mechanical optimisation design. Compared with the traditional algorithms, such as trial algorithm, table method and graph algorithm, the optimal design technique can greatly shorten the design cycle and improve the design quality. The design requirements from different angles can be considered to the maximum extent with the optimisation design technique, and the target values that meet predetermined requirements under various constraints can be found (Bello et al., 2015).

In recent years, with the expansion of the function of multi-body dynamics software, the optimisation of the performance parameters of each subsystem will begin to be used gradually under the environment of multi-body complex dynamics model. Thus, the precision of the model and the result of optimisation calculation are greatly improved. On the basis of the equivalent model of three-dimensional vehicle vibration, more influential factors can be taken into account by applying the vehicle multi-body system dynamics model, such as the rubber sleeve used in the suspension, and the nonlinear vehicle model including suspension spring, shock absorber and tyre is optimised. Based on ASAMS software, the optimisation design of vehicle suspension system and vehicle matching is given in this paper. Based on the multi-body dynamics and its analysis software ASAMS/Car, the simulation study of vehicle suspension and vehicle handling stability and ride comfort is carried out in the literature (Agarwal et al., 2015). In this paper, the vehicle ride comfort is simulated and analysed, and the suspension parameters are optimised with the aim of improving the ride comfort. Dr. Su Xiaoping, from Nanjing University of Technology, took a certain model of Iveco vehicle as the research object, and did a lot of simulation and optimisation work on the suspension system and the whole vehicle combined with ADAMS software.

## 3 Methodology

### *3.1 Analysis of ADAMS dynamics simulation calculation principle*

In auto control process, mechanical automation, one of the key technologies, can be sufficient to reduce the accidents in the process of running cars, reduce the security hidden danger, and also can timely solve many unexpected problems, and reduce the costs in control, input of manpower and material resources. At the same time, the quality of monitoring can be improved, and mechanical automation technology can be better applied. The methods for the specific application to automobile in control are analysed.

Mechanical automation, a technique with ever-increasing scientific and technological level of a new kind of mechanical manufacturing technology, improves the work efficiency and also conforms to the sustainable development strategy. Therefore, it becomes a new force for manufacturing industry in China, and has gained great attention and application in manufacturing industry. However, compared with developed countries, in China mechanical automation technology is relatively backward, and there is a big gap between them. Promotion of mechanical automation technology, to strengthen its application in the motor control, to improve the understanding of this technology is very good, and, for collection of wisdom of mechanical automation technology for the rectification and improve also has a great contribution.

In Table 1, the degree of freedom of the mechanical system represents the number of independent motions of each component in the mechanical system relative to the ground frame.

The degree of freedom of the mechanical system is related to the number of components that make up the machinery, the type and quantity of the motion pairs, the type and quantity of the original motive, and the other constraints. (Mazinan and Khalaji, 2016). For example, a rigid body floating in three-dimensional space has six degrees of freedom (DOF), and a cylindrical pair has two movements and two rotations, which provides four constraints.

The degree of freedom of mechanical system (DOF) can be calculated by the following formula:

$$\mathrm{DOF} = 6n - \sum_{i=1}^{m} p_i - \sum_{j=1}^{x} q_j - \sum R_k \tag{1}$$

In the formula, $n$ is the total number of active components, $p_i$ is the number of constraint conditions in the $\boldsymbol{i}$ motion pair and the total number of pairs of motion pairs is $m$; $q_j$ is the driving constraint condition number of the $\boldsymbol{j}$ prime mover, $x$ is the total number of prime movers, and $R_k$ is the number of other constraints.

**Table 1** The number of sportswear and degree of freedom constraints used in ADAMS

| *Motion pair*(*joint*) | *Degree of freedom constraint* (*constraints*) | | *Total degree of freedom constraint number* |
|---|---|---|---|
| | *Turn* | *Translation* | |
| Articulated pair | 2 | 3 | 5 |
| Prismatic pair | 3 | 2 | 5 |
| Cylindrical pair | 2 | 2 | 4 |
| Spherical pair | 0 | 3 | 3 |
| Plane pair | 2 | 1 | 3 |
| Constant speed pair | 1 | 3 | 4 |
| Fixed pair | 3 | 3 | 6 |
| Universal vice | 1 | 3 | 4 |

The number of degrees of freedom (DOF) and prime mover of a mechanical system are closely related to the kinematic characteristics of the system. In ADAMS software, the degree of freedom of the mechanism determines the analytical characteristics of the mechanism: kinematic analysis or dynamic analysis (Jalalkamali et al., 2015).

When DOF = 0, the kinematic analysis of the mechanism is carried out, that is to say, only the law of motion of the system is considered, without taking into account the external forces that produce motion. In kinematics analysis, when the motion state of a member is determined, the variation of displacement, velocity and acceleration with time is determined not by Newton's law but by the constraint relation between the members of the mechanism. It is solved by the iterative operation of the nonlinear algebraic equation of displacement and the linear equation of velocity and acceleration (Wu et al., 2018).

When DOF > 0, the dynamic analysis of the mechanism is carried out, that is to say, the motion of the mechanism is caused by the action of conservative force and non-conservative force, and the component motion is required not only to meet the constraint requirements, but also to satisfy the given motion law. It also includes static analysis, quasi-statics analysis and transient dynamics analysis. The dynamic equation of motion is the Lagrangian multiplier differential equation and the constraint equation of motion in the mechanism.

When DOF < 0, a statically indeterminate problem cannot be solved by ADAMS.

The following special problems should be noted in calculating the DOF of mechanical systems:

1 *Compound hinge*. Two or more members are connected to each other by rotating pairs, forming a so-called compound hinge. When there are $m$ members (including fixed members) are connected by composite hinges, the number of rotating pairs is $m$–1.

2 *Partial degree of freedom*. The degree of freedom has nothing to do with the motion that needs to be analysed in a mechanical system which is called a local degree of freedom. In calculating the degree of freedom of a mechanical system, the local degree of freedom can be removed.

3 *Virtual constraint*. Constraints that act as repetitive constraints are called virtual constraints, so virtual constraints are also called superfluous constraints. Virtual constraints often occur when:

1 The trajectory overlaps. If two components are connected by rotating pair on the mechanism and the locus of the connection points on the two members coincide, then the connection will be brought into virtual constraint. In the process of mechanism motion, when the distance between two points of different components remains constant, the two points are associated with one member and two rotating sidings, which will also bring virtual constraints.

2 The axis of the rotating pair is coincident. When the two components are composed of multiple rotational pairs and their axes overlap each other, only one rotating pair plays the role of constraint, and the rest of the rotation pairs are virtual constraints.

3 The moving pair moves parallel paths. Two components form multiple moving pairs and their moving paths are parallel to each other. In this case, only one moving pair acts as a constraint and the rest of the rotating pairs are virtual constraints.

4 The symmetrical part of the mechanism with the action of repetitive constraint on motion. In a mechanical system, the constraints brought in by some repetitive parts that do not affect the movement of the mechanism are also virtual constraints. Although the existence of virtual constraint has no effect on the movement of mechanical system, the introduction of virtual constraint can not only improve the mechanical force, but also increase the rigidity of the system, so it is widely used in the mechanism of mechanical system.

However, to solve the equations of motion, there should be no virtual constraint (that is, the correlation equation). Therefore, when the computer analyses the motion of mechanical system, the program will find the virtual constraint automatically. If the virtual constraint exists in the mechanical model, the computer will delete the redundant virtual constraint at any time. This method makes the calculation results different from the actual situation, and there may be multiple solutions. For example, if a door is connected with two rotating pairs (hinges), one of which has a virtual constraint, the computer program randomly removes the constraint of one of the rotating pairs, the result of which is that one of the rotating pairs bears all the connecting forces. The connection force of the other pair is zero. Since one of the rotating pairs is deleted randomly, the result may be in two cases.

## *3.2 Establishment of 1/2 model of front suspension and determination of optimisation conditions*

### *3.2.1 Establishment of 1/2 model of front suspension*

According to the structure of actual suspension system, a simplified front suspension 1/2 model is established in ADAMS/View. The whole front suspension 1/2 model includes: upper swing arm, lower pendulum arm, main pin, spring and damping, steering bar, pull arm, steering joint, tyre and test platform. The key coordinates of the front suspension 1/2 model are shown in Table 2. Each unit of value is millimetre (mm).

**Table 2** Key point coordinates of 1/2 model

| *Key point name* | *Coordinate name* | *X coordinates* (mm) | *Y coordinates* (mm) | *Z coordinates* (mm) |
|---|---|---|---|---|
| Centre point of wheel | P1 | 0 | –210 | –733 |
| Inside point of steering knuckle | P2 | 0.3 | –210.5 | –655 |
| The front point of the upper arm and the frame | P3 | –120.23 | 2.13 | –368.5 |
| Connection of the upper arm and the frame | P4 | 95 | 2.13 | –368.5 |
| Connection point of upper arm and steering knuckle | P5 | 12 | –24.3 | –568.8 |
| The front point of the hem arm and the frame | P6 | –266.9 | –250.87 | –297 |
| The back point of the hem arm and the frame | P7 | 142.87 | –250.87 | –297 |
| The connection point of the hem arm and the steering knuckle | P8 | –6.13 | –299.3 | –641.6 |
| Connection point of shock absorber and frame | P9 | –34.3 | 72.6 | –425.4 |
| Connection point of shock absorber and swinging arm | P10 | –58.5 | –279.6 | –467.2 |
| Steering knuckle arm and cross bar intersection | P11 | –137 | –138 | –406 |
| Intersection of cross bar and steering arm | P12 | –155 | –138 | –356 |

One end of the upper swing arm and the lower pendulum arm is connected to the frame (ground) and can rotate around two connecting points to the frame (ground), the other end is connected to the steering joint through the ball hinge, and the upper end of the shock absorber is connected with the frame (ground). The lower end is connected to the lower swing arm, because the spring in ADAMS has two options: stiffness and damping, so the stiffness of spring and damping of shock absorber can be replaced by a simple spring. The knuckle and the wheel pin are connected to the wheel through a fixed hinge (there is no need for the wheel to rotate at this time); there are two connections at the left and right ends of the bar, the left end and the steering arm are connected by the ball hinge, the right end is connected to the steering arm (ground) through the universal joint hinge.

Create a double-arm front independent suspension model in ADAMS/View, as shown in Figure 1.

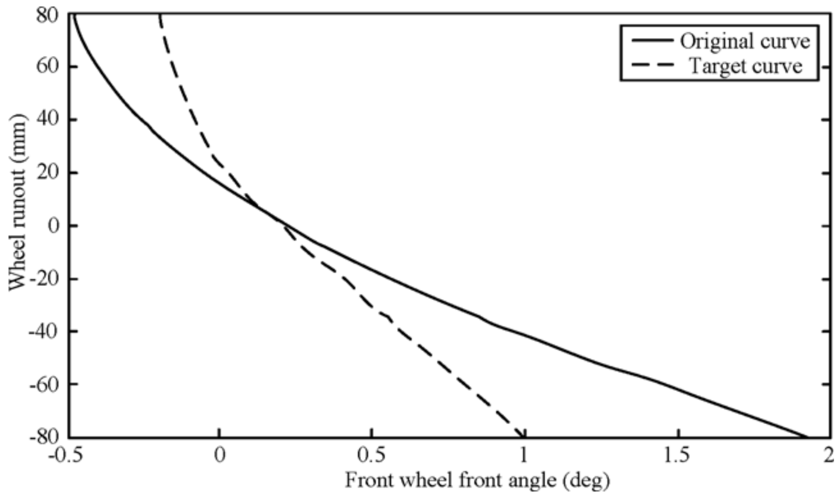
The whole suspension model is placed on the test stand. The test bench is connected to the wheel through a point side constraint pair, and the test bench to the earth through a prism pair. In simulation, the variation of parameters is calculated by applying vertical motion on the test stand.

There are eight objects, three fixed pairs, three spherical pairs, three hinged pairs, one universal joint pair, one prism pair, one point side constraint pair and one vertical constraint pair.

Degree of freedom:
DOF = 8 × 6 – 3 × 6 – 3 × 3 – 2 × 5 – 1 × 4 – 1 × 5 – 1 × 1 – 1 × 1 = 0.

The model has 0 DOF and can be analysed by kinematics.

**Figure 1** The change of the jump momentum of the front wheel front of the front wheel of the front wheel and the target curve




### *3.2.2 Establishment of optimal conditions*

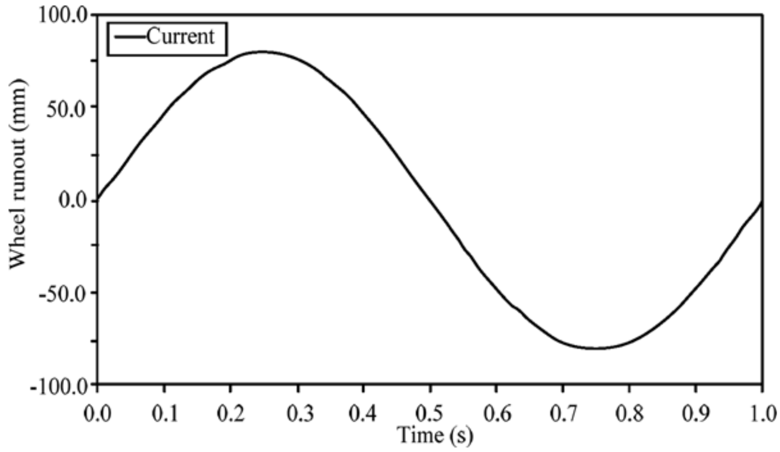
*(1) Object function*

The front beam angle of front wheel is the main guarantee condition to reduce the running deviation and the wear of the tyre, which has a direct effect on the safety and economy of the vehicle. In order to reduce the running deviation, reducing the tyre wear and improving the kinematic characteristics of the front suspension, the minimum deviation between the curve of the front angle of the wheel and the provided curve is taken as the optimisation objective during the run-up and down of the wheel. The objective function can be expressed as:

$$\min \text{ OBJ} = \sum_{i=1}^{p} \left[ \text{ABS}\left(C_i - T_i\right) \right] \tag{2}$$

In the formula, $C_i$ is the angle of the front beam of the wheel is calculated. ***T*** is the numerical value of the point on the target curve of the front beam of the wheel, ***i*** is the iterative ordinal number for solving multi-body kinematics. *P* is the total step number of the solution. ABS () is ADAMS's own absolute function. The front beam angle target and original values are shown in Figure 2. The change of the front beam angle of the front wheel is as small as possible when the wheel jumps up and down, and the target value of the front beam angle of the front wheel is set according to the actual needs of the vehicle.

**Figure 2** Wheel run out curve




When doing kinematics analysis in ADAMS, the objective function is usually a function with time as an independent variable, so the target value and the original value of the front beam angle of the wheel are transformed into a time function. At the end of the optimisation, the time function is transformed into the change of the front beam angle of the front wheel with the runout of the wheel. At the same time, the expression of the linear driving function of the test platform in Section 4.3 is changed to be: (80 × time-80), that is to say, the test platform moves from the highest point to the lowest point. When the wheel beats, the objective function is fun. Its expression is as follows:

ABS (ATAN (Dx (MARKER_86, MARKER_88) /Dz (MARKER_86, MARKER_88))-
+step (time, 0, –0.2d, 0.125, –0.15d)
+step (time, 0.125, 0d, 0.75, 0.15d)
+step (time, 0.75, 0d, 1, 0.2d)
+step (time, 1, 0d, 1.5, 0.4d)
+step (time, 1.5, 0d, 2, 0.4d)

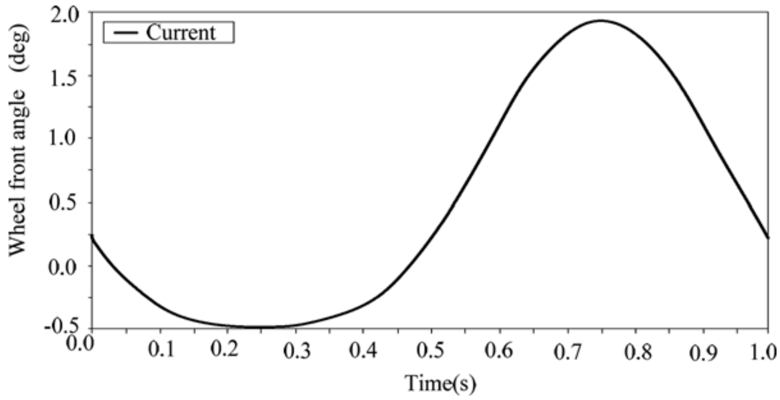
ATAN (Dx (MARKER_86, MARKER_88)/Dz (MARKER_86, MARKER_88)) is the toe angle which changes with time in the original function, the original curve corresponding to Figure 3:

step (time, 0, –0.2d, 0.125, –0.15d)
+step (time, 0.125, 0d, 0.75, 0.15d)
+step (time, 0.75, 0d, 1, 0.2d)
+step (time, 1, 0d, 1.5, 0.4d)
+step (time, 1.5, 0d, 2, 0.4d)

The objective curve function represents the change of the front beam angle with time corresponds to the target curve in Figure 3.

ABS is an absolute function in ADAMS.

**Figure 3** Change curve of the inclination angle of the main pin




*(2) Design variables and constraints*

The design variables are as follows: swinging arm and steering joint connection point(DV_1, DV_2, DV_3, upper pendulum arm and frame connection front point:(DV_4, DV_5, DV_6, upper pendulum arm and frame connection point(DV_7, DV_8, DV_9, lower pendulum arm and steering joint connection point (DV_10, DV_11, DV_12, upper pendulum and frame connecting front point(DV_13, DV_14, DV_15, lower pendulum arm and frame connection point(DV_16, DV_17, DV_18 and parameterise the coordinates of the points above. The specific value and range of variation are shown in Table 2, in which the numeric units of the coordinates of the coordinates are all millimetre (mm). At the same time, the length of the corresponding rod is parameterised to optimise these design variables to achieve the purpose of optimising the front suspension.

### *3.3 Evaluation method of vehicle ride comfort*

The main index to evaluate the vehicle ride comfort is the dynamic response result of the vehicle body under the random road surface input. According to the requirement of GB/T4970-1996 “Standard Test Method for Random Input Exercise of Vehicle Ride Comfort”, the random input ride comfort of the whole vehicle is simulated and analysed.

The total weighted acceleration root-square root (RMS) value of the driver’s human body is used to evaluate the index, which is the weighted RMS value of the longitudinal, transverse and vertical acceleration of the vehicle. In the test, the vehicle is in a full load state, and the test point is the driver seat. The acceleration curve of the driver is measured in the direction of *X*, *Y*, *Z*, and then the power spectral density curve is obtained by Fourier transform FFT. The coordinate system shows that a car is: *X* axis is vertical, a car with a forward backward, *Y* axis is vertical, a car with a upward and a car on the *Z* axis is transverse, forward to the left.

The root-mean-square value of the weighted acceleration of the human body, in the direction of *X*, *Y* and *Z*, can be directly integrated from the self-power spectral density function of the axial vibration acceleration:

$$\sigma_{pi} = \left[ \int_{0.9}^{90} w_i^2(f) G_{pi}(f) \mathrm{d}f \right]^{\frac{1}{2}} \quad (i = x, y, z) \tag{3}$$

$\sigma_{pi}$ is the root mean square value of the weighted acceleration of the human body in the direction of *X*, *Y*, *Z*, which is in the units of m/s$^2$. $G_{pi}(f)$ is the self-power spectral density function of the acceleration of the human body vibrating in the direction of *X*, *Y*, *Z*, which is in the units of m$^2$/s$^2$. $w_i(f)$ is the frequency weighting function of the human body in the direction of *X*, *Y*, *Z*.

In the *Y* axis:

$$w_y(f) = \begin{cases} 0.5 f^{\frac{1}{2}} & 0.9 < f \le 4 \\ 1 & 4 < f \le 8 \\ 8/f & 8 < f \end{cases} \tag{4}$$

In the direction of *X*, *Z*:

$$w_{x,s}(f) = \begin{cases} 1.0 & 0.9 < f \le 2 \\ 2/f & 2 < f \end{cases} \tag{5}$$

Therefore, the root means square values of the three-axial total weighted accelerations are:

$$\sigma_{pw} = \left[ (1.4\sigma_{px})^2 + (1.4\sigma_{ps})^2 + (\sigma_{py})^2 \right]^{\frac{1}{2}} \tag{6}$$

In the formula, $\sigma_{px}, \sigma_{ps}, \sigma_{py}$ are represented of the weighted acceleration root-square values of the three directions of the *X*, *Y*, *Z* axis respectively.

At present, the automatic technological progress acceleration, due to mechanisation technology is focused on the discovery of the driver at the time of running car, if the operation of mistakes, this technology can be timely to remind, at the same time corresponding treatment measures. This kind of automation technology in a certain extent, can use rational automation technology for processing, using the technology, can effectively reduce the risk of safety accident, and you can also to have found the problem of implementation timely processing.

## 4 Result analysis and discussion

### 4.1 Simulation analysis of front suspension

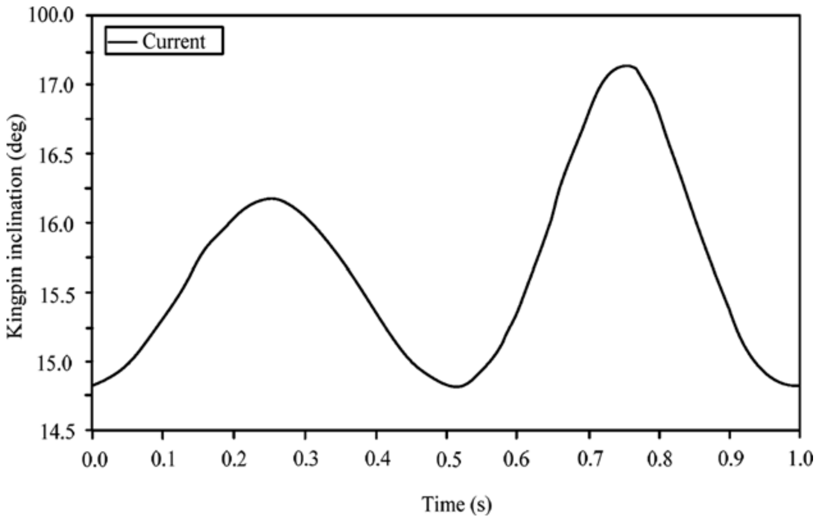
*(1) Add driver*

Create a straight-line drive for the test platform, perpendicular to the ground, and set the driver function expression to:

$$F(\text{time})=80*\sin(360\text{d}*\text{time}) \tag{7}$$

It indicates that the wheel jumps up and down according to the sine law on the test platform, and the stroke is 160 mm. The wheel jumps on the test platform as shown in Figure 4.

**Figure 4** Change curve of the inclination angle of the main pin




*(2) Create a measurement function*

In ADAMS/View, the inner inclination of the main pin and the angle of caster angle after the main pin are created, the camber angle of the wheel is camber angle, and the angle of the front beam of the wheel is used to measure the function. The simulation results, as shown in Figure 3, show the curves with time of the internal inclination of the main pin (Kingpin_Clination), the angle of caster angle after the main pin(Caster_Angle), the camber angle of camber angle(Camber_Angle), and the angle of the front bundle of the wheel(Toe_Angle).

From the simulation results in Figures 3 and 4, it can be seen that when the suspension changes with sinusoidal law on the test platform, the internal inclination of the main pin changes greatly. The change of the inclination angle after the main pin is small. The amplitude of the outside inclination angle of the wheel and the front beam angle of the wheel are basically the same.

*(3) Front suspension characteristic curve*

By importing Figure 3 into the post process module of ADAMS/View, the curves of the internal inclination of the main pin with the run out of the wheel, the curve of the change of the back angle of the main pin with the run out of the wheel, and the curve of the outside angle of the front wheel with the run out of the wheel are obtained. The front beam angle of the front wheel varies with the wheel run out, as shown in the diagram.

From the simulation results in Figures 5 and 6, it can be seen that the simulation curves of the internal inclination of the main pin, the rear inclination of the main pin, the outside inclination of the front wheel and the front beam angle of the front wheel with the wheel run out basically meet the design requirements. The change of the internal inclination angle of the main pin with the wheel run out is larger than the change of the rear inclination angle of the main pin with the wheel run out, and the change of the outer inclination angle of the front wheel with the wheel run out and the front beam angle of the front wheel with the wheel run out are not significant.

**Figure 5** The change curve of the obliquity of the front wheel with the wheel

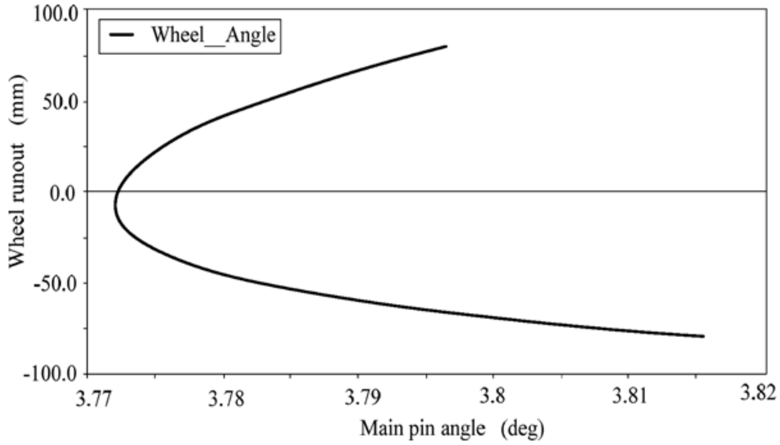
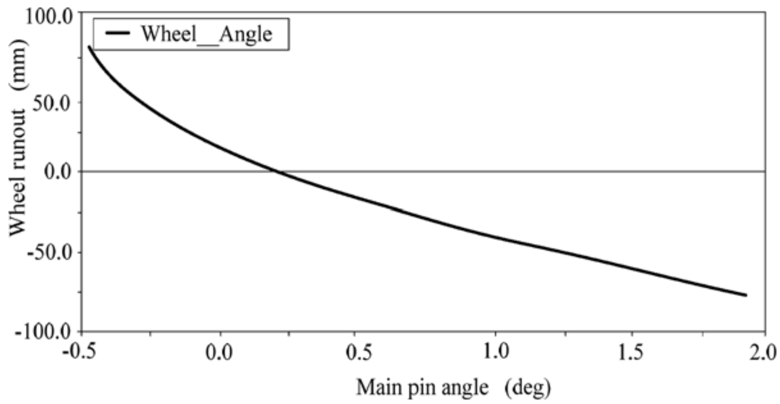



**Figure 6** The curve of the front wheel front angle of the front wheel with the change of the wheel




### *4.2 Optimisation results and analysis*

The prototype model of suspension system is optimised by using ADAMS's own algorithm OPTDES-GRG (generalised decreasing gradient algorithm), genetic algorithm and immune algorithm respectively.

The actions are as follows:

On the ADAMS/View interface, entering Simulate – Design Evaluation Tools, as shown in the diagram to optimise the design 'Design Evaluation Tools' dialog box. In the design evaluation tools dialog box, selecting 'Optimiser' in the lower right corner and entering the 'solver Settings' dialog box for optimal design. The optimisation algorithm is selected from 'algorithm' of 'solver Setup' dialog box, in which genetic algorithm and immune algorithm are defined as 'Use and' and 'User2' interface in programming.

After setting up the algorithm, the prototype model can be optimised. The prototype model of suspension system is optimised by OPTDES-GRG (generalised descending gradient algorithm), genetic algorithm ('Use and' interface) and immune algorithm ('User2' interface), respectively.

After the optimisation, the original curve of the objective function and the curve of the optimisation are shown in the graph as the values of the objective function and the design variable after the end of the optimisation are optimised by using different algorithms.

It can be seen from the diagram that the three algorithms can achieve better optimisation effect, that is, the minimum deviation between the change of the front beam angle of the wheel and the target value is achieved during the wheel run out. From Table 3, the following conclusions can be get: using the algorithm OPTDES-GRG (generalised decreasing gradient algorithm), genetic algorithm and immune algorithm, the objective function value is reduced from 0.290 to 0.060, 0.040 and 0.030 respectively. This shows that the best results are obtained by immune algorithm optimisation.

Before optimisation, the rear suspension parameters are compared as shown in Table 3. It can be seen from Table 3 that the maximum vibration acceleration in the vertical direction of body mass centre decreases, and the root mean square (RMS) of acceleration decreases from 0.52 $m/s^2$ to 0.43 $m/s^2$, which indicates that the ride comfort of the vehicle has been improved. At the same time, the reliability of the immune optimisation algorithm is further explained, but the optimisation speed is slow when the immune algorithm is used to optimise the algorithm.

**Table 3** Comparison of suspension parameters before and after optimisation

| *Parameter* | *Pre optimisation* | *After optimisation* |
|---|---|---|
| Root mean square of acceleration ($m/s^2$) | 0.52 | 0.43 |
| Spring stiffness of front suspension (N/mm) | 26.3 | 25.1 |
| Front suspension damper damping (Ns/mm) | 2.0 | 2.16 |
| Spring stiffness of rear suspension (N/mm) | 33.0 | 29.7 |
| Damping of rear suspension elastic shock absorber (Ns/mm) | 3.0 | 2.81 |

## 5 Conclusion

In this paper, the suspension system of a certain type of off-road vehicle is optimised and simulated by using the dynamic simulation software ADAMS which is a multi-body system, combining a genetic algorithm and immune algorithm of artificial intelligence algorithm.

1 Aiming at the user interface of optimisation algorithm provided by ADAMS software, the code of genetic and immune optimisation algorithm is added to ADAMS/View by using dynamic link library technology, and the link between ADAMS and objective function is established. The artificial intelligence algorithm is successfully implemented in ADAMS software. The simulation results show that the added algorithm can meet the needs of suspension system optimisation design.

2 The vehicle 1/4 vibration model is established, and the kinematics characteristics of the suspension are simulated and analysed. Based on the kinematics analysis of the suspension, the added algorithm is adopted to improve the performance of the suspension. The 1/2 model of front suspension is optimised and the optimum effect is achieved.

3 According to the parameters of the suspension system of a certain type of vehicle, the simulation model of the front and rear suspension is established, and the dynamic simulation model of the whole vehicle system is established on the basis of the suspension model.

4 On the basis of the whole vehicle model, the vehicle ride comfort is simulated and calculated. Considering the various factors of suspension, the immune algorithm is adopted to optimise the suspension characteristic parameters, in order to improve the vehicle ride comfort. The results show that the ride comfort of the whole vehicle has been improved to some extent.

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