Advances in Engineering Research, volume 123
2nd International Conference on Materials Science, Machinery and Energy Engineering (MSMEE 2017)

# Dynamic Modeling and Test Verification of Tank Multi - body System

Bing-fei Hao[a,*] Hong-yan Wang[b] ,Qiang Rui[c] and Qin-long Wang[d]

Academy of Armored Force Engineering, Beijing 100072, china

[a]haobing952@163.com, [b]wanghongyan@126.com, [c]rui_qiang@126.com, [d]qinlong_w@126.com

**Abstract:** In order to study the driving vibration characteristics and off-road maneuverability of tanks, the topology of tank system is analyzed by virtual prototyping technology and multi-body dynamics modeling method. The dynamic model of tank system is built. The dynamic simulation is compared with the actual vehicle test. The results show that the dynamic model of the tank can reflect the vibration characteristics of the real vehicle. The model provides a reference model for the study of tank vibration characteristics.

**Key words:** Tank, Virtual Prototyping, Topology, Real Vehicle Test

## 1. Introduction

Tank, as the main combat equipment in the modern army, most works and combats in the complex road. In recent years, with the rapid development of virtual prototyping technology, multi-body dynamics modeling and simulation is a very important method to study tank ride comfort, handling stability and off-road maneuverability [1, 2].

Nai-jun Ju, according to the analyzing of vertical vibration characteristics of the tank, the influence of the track on the vibration of the hull is neglected. Assuming that the road-wheels are in direct contact with the ground, the mathematical model of the vibration system is simulated and analyzed. Shi Li-Chen [4-6] Based on the theory of multi-body dynamics, the dynamic model of high-speed tracked vehicle is established, and the cannon is built on the suspension system. CHEN Bing[7], through the analysis of the relationship between the contact force between the components of the tank suspension system, the multi-body dynamics software Recurdyn was used to built the tank multi-body system model The researches mentioned above lack the verification of the model credibility.

Based on the analysis of topological structure of tank system, this paper builds up the model of tank multi-body dynamics based on multi-body dynamics software, and verifies the credibility of the model by real vehicle test method.

## 2. To establish a tank multi-body dynamics model

### 2.1 Model topology analysis

The main battle tanks are mainly composed of the weapon system, the power transmission system, the hull system and the mobile system. The mobile system is symmetrical about the left and right sides of the hull, including the sprocket, the support roll, the road-wheel, the idler and the song

Arm type track tensioning mechanism, track shoe, track pin and suspension system spring element and damping element [8].

According to the analyzing of constraints among the simplified components of relationship, and the topology of the model is analyzed, as shown in Figure 1. See table 1 and table 2 for details of the parts and constraints.

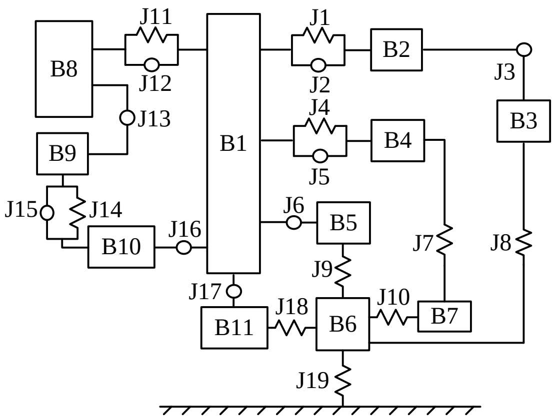



Fig 1 Topological structure of model

Tab 1 Part list

| Number | Name | Quality | Number | Name | Quality |
|---|---|---|---|---|---|
| B1 | Hull | 1 | B7 | Track pin | 196 |
| B2 | Equilibrium elbow | 12 | B8 | Cannon | 1 |
| B3 | Road-wheel | 12 | B9 | Driving piston | 1 |
| B4 | Sprocket | 1 | B10 | Driving hydraulic cylinder | 1 |
| B5 | Support roll | 6 | B11 | Idler | 2 |
| B6 | Belt | 196 | | | |

Tab 2 constraint list

| Number | Name | Quality | Number | Name | Quality |
|---|---|---|---|---|---|
| J1 | Spring damping torque | 12 | J11 | Friction damping torque | 1 |
| J2 | Rotation joint | 12 | J12 | Rotation joint | 1 |
| J3 | Rotation joint | 12 | J13 | Rotation joint | 1 |
| J4 | Driving mechanism | 2 | J14 | Hydraulic driving force | 1 |
| J5 | Rotation joint | 2 | J15 | Translate joint | 1 |
| J6 | Rotation joint | 6 | J16 | Rotation joint | 1 |
| J7 | Contact force | 2 | J17 | Rotation joint | 2 |
| J8 | Contact force | 12 | J18 | Contact force | 2 |
| J9 | Contact force | 6 | J19 | Contact friction force | 196 |
| J10 | Shaft force | 392 | | | |

According to analysis of Figure 1, table 1 and table 2, they shows that the number of components required to establish a dynamic model is 426 and the number of displacement constraints is 36, where the degree of freedom of the rotation joint and the mobile secondary limit is 5 and the total degree of freedom of the model is 1217.

### 2.2 Force Constraint Description

In Fig 2, the constraint J1 is the spring damping moment acting between the hull and the equilibrium elbow, and it is obtained by equating the torsion bar spring-blade friction damper of the suspension system [9,.10]. The equivalent formula is:

$$T_k = \frac{\pi d^4 G}{32L}\alpha \qquad T_c = i^2 c' \dot{\alpha} \tag{1}$$

Where $T_k$ and $T_c$ are the equivalent spring torque and the equivalent damping torque, $d$ is the diameter of the torsion bar, $G$ is the shear modulus of the torsion bar, $L$ is the length of the torsion bar, $i$ is the transmission ratio from shock absorbers to the installation point of equilibrium elbow; $c'$ is the damping coefficient of the shock absorber; $\alpha$ is the rotation angular of equilibrium elbow with its rate of changing $\dot{\alpha}$.

In the mobile system, the interaction between the sprockets, the idlers, the support rolls and the road-wheels and the belts through the contact force, named J7, J8, J9, J17, the formula is:

$$F = \begin{cases} \tau(\delta_1 - \delta)^e - \varepsilon\dot{\delta}, & \delta \le \delta_1; \\ 0, & \delta \ge \delta_1; \end{cases} \tag{2}$$

Where $F$ is the contact force; $\tau$ and $\varepsilon$ is the stiffness coefficient and the damping coefficient of the contact surface; $e$ is the nonlinear exponent of the contact force and is the generalized contact displacement. $\delta$ is generalized contact displacement with its rate of change $\dot{\delta}$, $\delta_1$ is the base distance to produce contact force.

The interaction force between the track shoe and the ground is treated according to the contact-friction force, the driving torque J4 of the sprocket and the hydraulic driving torque J14 are defined according to the specific working conditions. The restraining force is treated according to the linear spring damping force.

### 2.3 Description of displacement constraints

Any part of the dynamical model is expressed as a generalized coordinate.

$$q^T = [x \ y \ z \ \psi \ \theta \ \varphi] \tag{3}$$

Where x, y, z are the coordinates (center of mass) of the component in the Cartesian coordinate system, $\psi$, $\theta$ and $\varphi$ are the coordinates of the Euler angle. Taking the rotation joint as an example, the displacement constraint is described by the rotation joint, and the rotation joint restricts the translation of the three directions and the rotation in both directions. Therefore, the displacement constraint equation can be expressed as:

$$\begin{cases} x_{ij} = y_{ij} = z_{ij} = 0 \\ \sin\theta_{ij} \sin\varphi_{ij} = 0 \\ \sin\theta_{ij} \cos\varphi_{ij} = 0 \end{cases} \tag{4}$$

Where $x_{ij}$, $y_{ij}$and $z_{ij}$ are the displacement of the component $i$ with respect to the component $j$ in three directions; and the Euler angle of the component $i$ with respect to the component $j$.

### 2.4 Multi-body dynamics modeling of vehicle

The first-order Lagrange equation based on generalized Cartesian coordinates is used to establish the tank multi-body dynamics model. The generalized coordinate matrix is:

$$q=[q_1^T,q_2^T,\cdots,q_n^T]^T \tag{5}$$

Where $n$ is the number of component elements in the model and the constraint equation of the system is expressed as:

$$\Phi(q,t)=[\Phi_1(q,t),\cdots,\Phi_m(q,t)]=0 \tag{6}$$

Where: $m$ is the number of system displacement constraints and force constraint equations. The dynamics of the Euler-Lagrange Formation is:

$$\begin{bmatrix} M & \Phi_q^T \\ \Phi_q & 0 \end{bmatrix}\begin{bmatrix} \ddot{Q} \\ \lambda \end{bmatrix}=\begin{bmatrix} P^A \\ \eta \end{bmatrix} \tag{7}$$

Where: $M$ is the generalized mass matrix of the model component, $\Phi_q^T$ is the Jacobian matrix, obtained by the differential equation (6); $\lambda$ is the Lagrange's multiplier; $P^A$ is the generalized external force matrix; $\eta$ is the right term of the acceleration constraint equation. Figure 2 is the establishment of the tank multi-body dynamics model with Recurdyn.

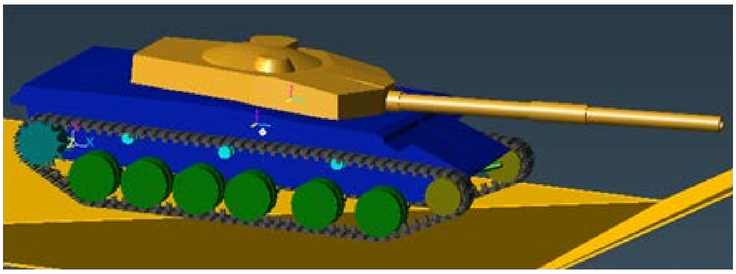


Fig.2 Tank multi body dynamic model

### 3 Simulation and Analysis of Real Vehicle test

The test device shown in Figure 3, where the NI data acquisition system is used to obtain the three acceleration signals when the tank is running on the road, included the first point above road-wheel's axis on the hull, the first road-wheel axle, and driving Speed signal acquisition.

a) NI Data acquisition system

b) Vehicle platform

c) Rotation rate of sprocket photoelectric sensor

Fig.3 Real vehicle testing device

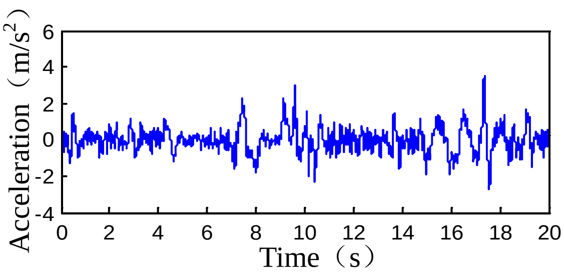
During the test, the tank was tested at a low speed through the vehicle platform, NI data acquisition system is to record the acceleration signals in three directions. During the test, the speed of the sprocket is measured, and then the speed is loaded into the sprocket of the simulation model to drive the simulation tank.

Comparing the vertical vibration acceleration signal of the hull with the first road-wheel calculated by the real test and the simulation, the Welch algorithm based on the periodic graph method is used to calculate the power spectral density of the acceleration signal data and measured by the attitude test sensor The variation of the pitch angle of the hull through the vehicle platform is shown in Fig 4, Fig 5 and Table 3.

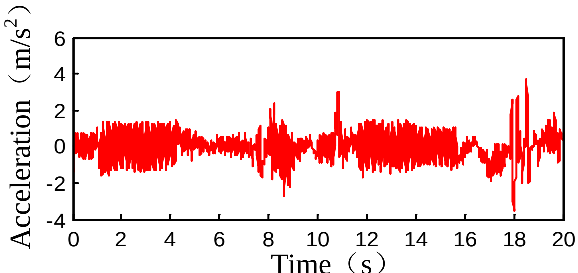
Tab 3 Comparison of vertical acceleration of hull
above the first road-wheel

| Project | Test result | Simulation result | Error |
|---|---|---|---|
| Peak | 3.7042 $m/s^2$ | 3.892 $m/s^2$ | 5% |
| RMS | 0.6395 $m/s^2$ | 0.6959 $m/s^2$ | 1% |
| Vertical natural frequency | 1.51 Hz | 1.54Hz | 1.3% |

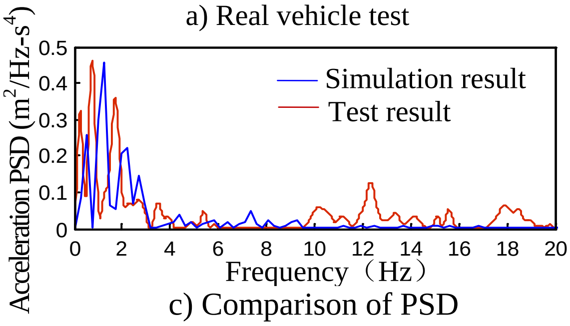
Through the comparison of the statistical data in Fig. 4 and Table 3, it can be found that the simulation results are in good agreement with the experimental results. The vertical natural frequencies of the simulation data and the experimental data are 1.51Hz and 1.54Hz respectively, which are close to the natural vibration of the theoretical analysis result, Through the test and simulation results in Figure 5 comparison found that the test and simulation is in good agreement, the vehicle in the 5s~12s climbing vehicle platform, 12s~16s stable on the platform, 16s~22s running down the platform. Test results verify the rationality of the low-order vibration characteristics of the tank model was established.




a) Real vehicle test




b) Simulation result




c) Comparison of PSD

Fig.4 Vertical acceleration and PSD of vehicle body above the first road-wheel

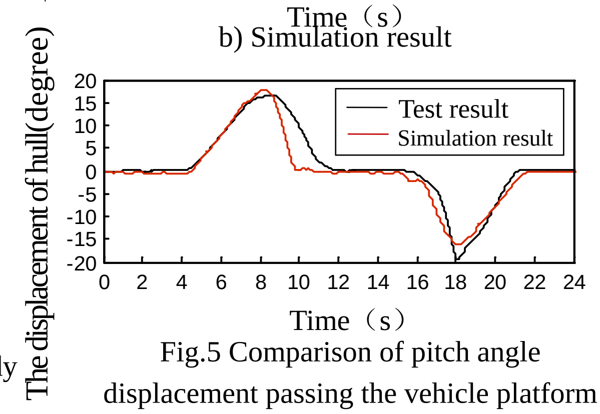



Fig.5 Comparison of pitch angle displacement passing the vehicle platform

It can be seen that the simulation results of the multi-body dynamics model are the same as those of the real vehicle test, and with the increase of the vehicle speed, the simulation results show that the root mean square value of the vibration acceleration signal increases. Compared with the simulation results, it is shown that the dynamic model can reflect the dynamic characteristics of the tank.

## 4. Conclusions

Based on the topology analysis of the tank driving system, the dynamic model of the tank multi-body system is established in Recurdyn based on the virtual prototype technology. Under the typical driving conditions, the simulation analysis is carried out. The conclusions are as follows:

1) The first two-order resonant frequency of the tank is between 1 and 2Hz, mostly the first-order pitch and the second-order vertical vibration of the hull. The third-order vibration instability changes with the change of the vehicle speed.

2) On the contrary, the vibration acceleration, from the first road-wheel above simulated tank model and the real tank, power spectral density curve of the peak point coincidence is better, indicating that the model can be used for tank travel off-road mobility analysis.

3) Based on the virtual prototyping technology modeling, it provides a new method for the study of tank vibration characteristics.

## 5. References

[1] Bao-kun Han, LI Xiao-lei, Feng-chun Sun, Present state and future outlook of the simulation of tracked vehicle, J . Acta Ar-mamentarri, 24 (2003) 246-249.

[2] Ferretti G, Girelli R, Modeling and simulation of an agricultural tracked vehicle, J . Journal of Terramechanics , 36 (1999) 139-158.

[3] Nai-jun Ju, Computer Simulation of Tank Random Dynamic System, J .Acta Ar-mamentarri Tank armored vehicles and engines, 1 (1984) 1-26.

[4] Li-chen Si, Liang-xi Wang, Bing-zhi Zhang, Ride dynamic simulation of armored vehicle and experiment validation, J .Computer Simulation, 21(2004) 21 18-20.

[5] Li-chen Si, Liang-xi Wang, Bing-zhi Zhang, Modeling and simulation of a moving tank-gun system in vibration, J . Acta Armamentarri , 24 (2003) 442:446.

[6] Li-chen Si, Liang-xi Wang, Bing-zhi Zhang, Dynamic Simulation of Tank Gun System. J . Journal of system simulation. 16 (2004) 366-369.

[7] Bing Chen, Liang Gu, Hua Huang. Computer simulation study of tracked vehicle with semi-active suspension, J .Computer Engineering and Design. 27 (2006) 7-11.

[8] Qin-long Wang, Hong-yan Wang, Rui Qiang. Research on parameter updating of high mobility tracked vehicle dynamic model based on Multi-objective Genetic Algorithm, J .Acta Ar-mamentarri , 37 (2016) 969-978.

[9] Nai-jun Ju. Dynamic analysis and simulation for armored vehicle, first ed. National Defense Industry Press , Beijing,2002,pp.80-120.

[10] Fa-qian Ding. Dynamics of tracked armored vehicle suspension system . twice ed. National Defend Industry Press, Beijing, 2004,pp.160-195.