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Dynamics simulation research on load vehicle of deep submergence rescue vehicle (LV-DSRV)

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SUMMARY

Submarine accidents can cause loss of human life and economy, as well as environment damage. Also submarine rescue is difficult for the complexity of rescue process and the uncertainty of the sea state. Success of rescue process is determined by reliability and safety of rescue device. Therefore, research on related device design is rather important. The existing knowledge always supplies some empirical formulas and generates a design scheme by general design rules. The scheme plan obtained may not be a good one due to simplicity of mechanical calculation analysis and particularity of design requirements. To improve design safety and reliability of shipborne device on submarine rescue system, dynamics simulation model of LV-DSRV based on sea state excitation was created in this paper. Wave excitation input at sea state 5 and 8 was considered as the extreme marine working conditions. Kinetic property of LV-DSRV was calculated making use of virtual prototype technology through ADAMS software, and mechanical characteristics of key parts were also analyzed. Optimization strategy was proposed and verified by increasing the number of horizontal wheels and adding gap between horizontal wheels and the track, providing with a case study for similar marine special mechanism design.

Key words: *shipborne device, dynamic analysis, sea state, submarine rescue.*

1. INTRODUCTION

According to the statistics, in the non-war period since 1900, there have been 170 major submarine accidents resulting from collision, material failure, disoperation, fire, explosion or other unknown reasons, which do not include common submarine accidents [1]. In *75%* of these accidents, submarines sank in shallow water region less than *200 m*, while in *25%* of these accidents submarines sank in deep water area. With the increasing importance of submarine in the modern sea battle, the world's major countries have paid great attention to the development of advanced submarine rescue system. Therefore, it is of great significance to put much effort on the research on mechanism design relating to submarine rescue systems.

Submarine rescue ship, an important surface ship for deep submergence rescue vehicles (DSRV's) during submarine rescue operations, carries saturation diving system, deep-sea remotely operated vehicle, rescue bell, DSRV and other types of life-saving equipments [2]. As it plays an important role in submarine rescue mission, the navies of many countries such as France, UK and Norway are all equipped with submarine rescue vessels with different functions [3].

The object in this research is the load vehicle of DSRV(LV-DSRV), which carries DSRV on a new type of submarine rescue ship. The vehicle operates on the special fixed track along the deck, with DSRV moving at a low speed between midship parking location and stern lifting position. Because of the complexity of the submarine rescue mission and the uncertainty of the sea state, design of LV-DSRV is required to guarantee its stability so as to avoid damage to the expensive DSRV.

The higher sea state is at which submarine rescue ship works, the more violently LV-DSRV will move due to ship motion as the wave and wind is intensive. The limit sea state at which LV-DSRV can work is 5, and the limit sea state in which DSRV can be loaded on the vehicle is 8. Thus, it is necessary to investigate dynamic characteristics of LV-DSRV for its stability and reliability. However, related research references at home are hardly found because it is a novel mechanism designed in China. Besides, it costs much money and time with high risks to implement physical prototype tank experiment. All this causes difficulties to improve and optimize structure of LV-DSRV.

In this paper, dynamic characteristics of LV-DSRV are calculated by using virtual prototype technology. Wave excitation at sea state 5 and 8 are input in accordance with design requirements of LV-DSRV; kinetic property of LV-DSRV is simulated for the extreme condition of marine environment; contact force between wheels and track is calculated; mass center displacement/speed of LV-DSRV, motors driving torque curves and tensile force of lashing bars are also achieved. Consequently, device optimization strategy is proposed and verified by increasing the number of horizontal wheels and adding gap between horizontal wheels and the track, providing with a method for the optimization design of LV-DSRV.

The rest of this paper is organized as follows. In Section 2, research on related marine special mechanism is surveyed, and design characteristics of LV-DSRV are introduced. In Section 3, dynamics simulation analysis of LV-DSRV is illustrated. In Section 4, device optimization strategy is discussed. Finally in Section 5, the main contribution of this paper is summarized and future work is discussed.

2. PROBLEM STATEMENT

According to the statistics, there are different sizes of waves at sea in about *70%* of the time. As wave load has significant effects on ship offshore operations, design and experiment of marine special mechanism based on ship motion is the issue of the research [4].

Although numerous researches on ship seakeeping theory have been achieved [5-6], study of dynamic analysis on shipborne device is limited and focuses on offshore crane and naval artillery system. In the study of offshore crane, Idres et al. [7] developed a nonlinear 8-DOF crane-ship dynamic model incorporating hull motions with nonlinear large-angle load swings. The ship and crane are treated as one rigid body including arbitrary, bi-angular swings of the suspended load coupled with the surge, sway, heave, roll, pitch, and yaw motions of the ship [7]. Then, presented a mathematical model of the crane's mechanical systems to simulate offshore cranes performing sealift operations [8]. A Cave Automated Virtual Environment (CAVE) was built by Daqaq as a platform to study the dynamics of ship-mounted cranes under dynamic sea environments [9]. Meanwhile, in the study of naval artillery system, Ma created a dynamic model of a self-mechanized gun by adding rotating gun turret, cradle assembly and recoil gun barrel to the tank model in ATV [10]. Motion state and forces of the warship gun breechblock cam was analyzed by Luo with ADAMS software [11].

Shipborne device does not only move with ship under wave actions, but also performs specific work. Therefore, shipborne device design is different from ground device design, as efforts such as wind action, inertia force resulting from ship motion and transmission failure must be considered [12-13]. Based on seakeeping theory, ship motion is usually analyzed by coupling of all degrees of freedom. Thus, kinetic property of the vehicle is complicated.

The vehicle has the following characteristics: (1) <u>High load capacity</u>: the self weight of the vehicle is *10 tons*, and the weight of DSRV is *30 tons*; (2) <u>High position of the device's gravity center</u>: Because DSRV is required to connect with access hatch of pressurized oxygen chamber after submarine rescue mission is completed, structure height of the vehicle reaches *2.3 m* from the deck; (3) <u>High stability requirement</u>: The length of the track is *24 m*, engaged transmission of gear and rack should maintain normal operation so as to avoid motion-block problem.

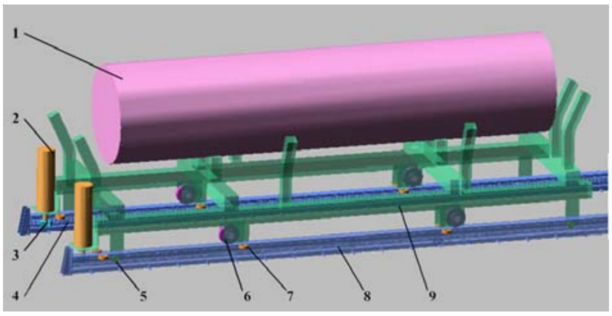
This paper mainly aims at proposing mechanism optimization strategies for LV-DSRV through ADAMS analysis. Study of ship motion is not the focus in this paper. According to theoretical analysis, sea state can be represented by a sum of sine waves, its frequency and amplitude value at different sea state can be obtained through looking up sea state table. In ship oscillation, heave, pitch and roll severely affect kinetic property of the vehicle and its key parts. LV-DSRV dynamics simulation based on these 3 degrees of freedom is calculated and analyzed in this paper, but deep research needs to be further studied.

3. DYNAMICS SIMULATION ANALYSIS

3.1 Dynamics modelling

3.1.1 Solid model

Solid modelling software Pro/E is used to generate three-dimensional model of LV-DSRV, and its assembly model is shown in Figure 1. As the meshing calculation and contact force calculation take up large computer resources in dynamics simulation, parts such as bolts and washers are omitted, and motor and DSRV are simplified as cylinders to enhance simulation efficiency and reduce computing time.



Legend: 1. DSRV; 2. Motor; 3. Drive gear; 4. Rack; 5. Reverse-wheel; 6. Load-bearing wheel; 7. Horizontal wheel; 8. Track; 9. Vehicle body

Fig. 1 Solid model of LV-DSRV

According to Figure 1, LV-DSRV is arranged on the I-shaped tracks. The load-bearing wheels, horizontal wheels and reverse-wheels contact with the upper surface, the lateral surface and the bottom surface of the tracks respectively. Drive gears are mounted on the one end of the vehicle, realizing engaged transmission with the racks fixed on the internal surface of the track. I-shaped tracks are riveted on the submarine rescue ship's deck with oscillating motion due to wave actions.

Contact force between wheels and track is the focus of dynamics analysis. To indicate calculation results, each wheel is marked respectively. In Figure 2, numbers 1-4 represents load-bearing wheels; 5-16 represents horizontal wheels, 17-20 represents reverse-wheels, and 21-22 are driving gears.

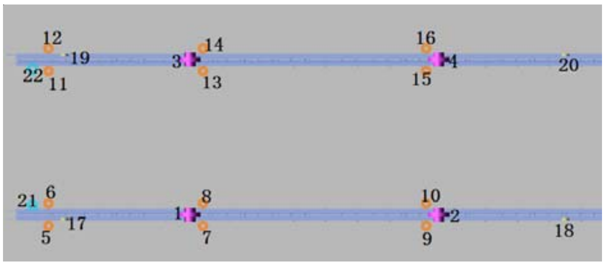


Fig. 2 Position and numbering of wheels

It is obvious that the load-bearing wheels support the weight of the vehicle and DSRV, keeping contact with the track. Ship motion is subjected to severe sea environments, and transient contact force is generated between horizontal wheels and the track under wave actions. Horizontal wheels play the role of guidance and support, preventing the vehicle departure from the track, and shares inertial force resulting from pitch-roll coupling as well. Reverse-wheels also guide the vehicle move along the tracks, and restrict the LV-DSRV freedom by contacting with the bottom surface of the tracks.

3.1.2 Dynamics modelling of LV-DSRV under working condition

Three-dimensional model of LV-DSRV is imported into ADAMS software with Parasolid interface, considering model transformation time, integrity of geometric features and physical information [14]. Topology constraint relations among parts are determined in establishing dynamic model of LV-DSRV in ADAMS.

The whole topology relations of the vehicle are shown in Figure 3. Wherein, fixed constraints (FIX) are applied between DSRV and vehicle body, and between motor reducer and vehicle body; rotating constraints (ROL) are applied between wheels, driving gears and vehicle body; contact constraints (CONT) are applied between wheels and tracks, and between driving gears and rack. Moreover, motion of tracks is added with multi-degree of freedoms (DOF), and motion of driving gears is added with rotating freedom.

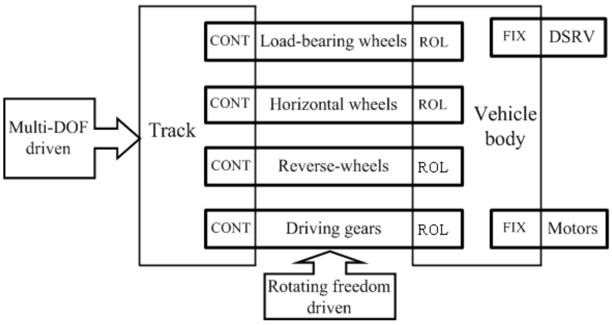


Fig. 3 Topology relations of LV-DSRV

Degrees of freedom of LV-DSRV can be calculated from the formula below [15]:

$$\begin{aligned} F &= 6n - \sum_{i=1}^{m} p_i - \sum_{j=1}^{l} q_j - \sum R_k = \\ &= 6n - p_{ROL} - p_{FIX} - q_{MOTION} = \\ &= 6 \times 28 - 5 \times 22 - 6 \times 3 - 14 = 26 \end{aligned} \tag{1}$$

The result shows that the motion of LV-DSRV is uncertain due to the sea state excitation, and each part of the vehicle may generate motions in multi-directions.

3.1.3 Dynamics modelling of LV-DSRV with lashing bars

At the sea state 8, LV-DSRV is arranged on the midship location. Contact force between reverse-wheels and track is up to *150 kN* based on pitch calculation, which may cause yield deformation for track. Thus, lashing bar is required to fix LV-DSRV.

There are altogether 8 lashing bars on the vehicle. Their force directions are along themselves and do not change due to their high rigidity and low damping property as 2D-bars. Stiffness of a bar with length L can be obtained according to mechanics of materials:

$$K = E \cdot A / L \tag{2}$$

wherein, E is the Young's modulus of elasticity, and A is the cross-sectional area.

The lashing bar material is *40 Cr*, *E=211 GPa*, stiffness of 8 lashing bars with different lengths can be calculated by Eq. (2) as shown in Table 1.

Table 1. Stiffness of 8 lashing bars (N/m)

No.	1	2	3	4	5	6	7	8
Value	22.85	22.85	34.262	34.262	33.27	33.27	28.59	28.59

The dynamics model of LV-DSRV with lashing bars is shown in Figure 4. In this figure, lashing bars are distributed symmetrically, with positions and numbers marked in both sides.

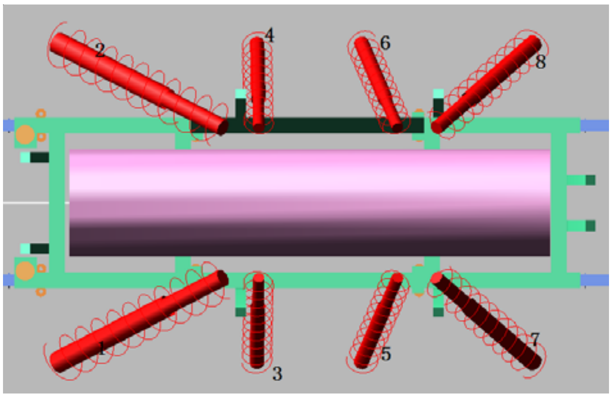


Fig. 4 Dynamics model of LV-DSRV with lashing bars

3.2 Excitation input at different sea states

As an important index for analyzing marine environment and its role in the fields of meteorological oceanography and marine engineering, sea state is a comprehensive description of the sea surface fluctuations under different marine environments, which relates to various factors such as wind speed and significant wave height. The vehicle studied in this paper is required to perform offshore operations under severe sea state environment; wherein, the extreme working condition is sea state 5, and the extreme loading condition is sea state 8. Therefore, ship motion excitations under the two limit sea state codes need to be considered.

3.2.1 Excitation input at the sea state 5

LV-DSRV is designed to work safely at the sea state 5, as it runs along the tracks at a low speed about *0.133 m per second*.

As the vehicle's tracks are riveted with the deck, motion characteristic of the tracks are determined by ship motion, and the tracks motion can be simulated by ship oscillation. Ship motion with heave-pitch coupling may severely affect stable movement of LV-DSRV. Excitation of the sea state 5 is simulated with the consideration of heave-pitch coupling. The motion cycle and amplitude can be calculated according to the ship seakeeping theory [16]. Figure 5(a) represents motion curve of heave, and Figure 5(b) represents motion curve of pitch.

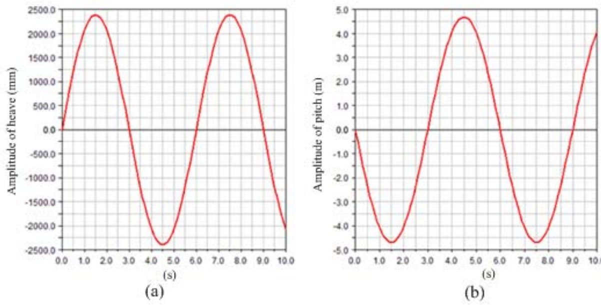


Fig. 5 Ship motion in heave (a) and pitch (b) at the sea state 5

Motion characteristic of LV-DSRV with heave-pitch coupling can be simulated by applying the following setups in ADAMS software: the tracks are arranged to swing around the ship core, its driving type is restricted with 6-DOF point driving, and its degree of freedom along which there is no motion is removed.

3.2.2 Excitation input at the sea state 8

The submarine rescue ship can move in seaway at the limit sea state 8 when LV-DSRV is parked in the storing position close to the ship core (midship parking position). Though the vehicle is fixed by the lashing bars and does not work, *30 tons* of DSRV is positioned on the top of the vehicle. Ship roll severely affects the stability of LV-DSRV, which may cause vehicle tip-over or lashing bars fracture. Motion cycle and amplitude of ship roll at the sea state 8 is calculated as the excitation input of LV-DSRV, and the motion curve of roll is shown in Figure 6.

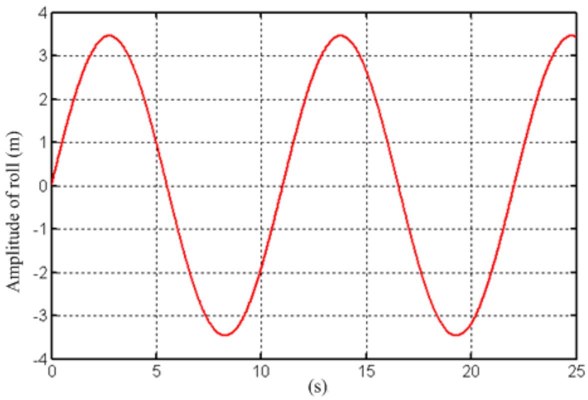


Fig. 6 Ship motion in roll at the sea state 8

3.3 Dynamics simulation analysis of LV-DSRV

3.3.1 Dynamics analysis at the sea state 5

Contact forces between the load-bearing wheels and the tracks are shown in Figure 7 wherein, Figures 7(a) and 7(c) show the force of 1 and 3 load-bearing wheels, which are far from the ship core; Figures 7(b) and 7(d) show the force of 2 and 4 load-bearing wheels, which are close to the ship core. The results show that sea state excitation has different effects on load-bearing wheels, maximum value of the load-bearing wheels which are far from the ship core is obviously larger than that of the wheels close to the ship core.

As the contact forces of the horizontal wheels on both sides are basically equal, forces of the horizontal wheels on one side are given. In Figure 8, forces of the horizontal wheels from 5 to 10 are shown in Figures 8(a) to 8(f) respectively. The results show that the action between horizontal wheels and track at the sea state 5 is unilateral contact, the wheels 6, 8 and 10 contact with one track as the wheels 5, 7, 9 do not. Moreover, value of the contact force on horizontal wheels is related to their locations along the ship body vertical direction.

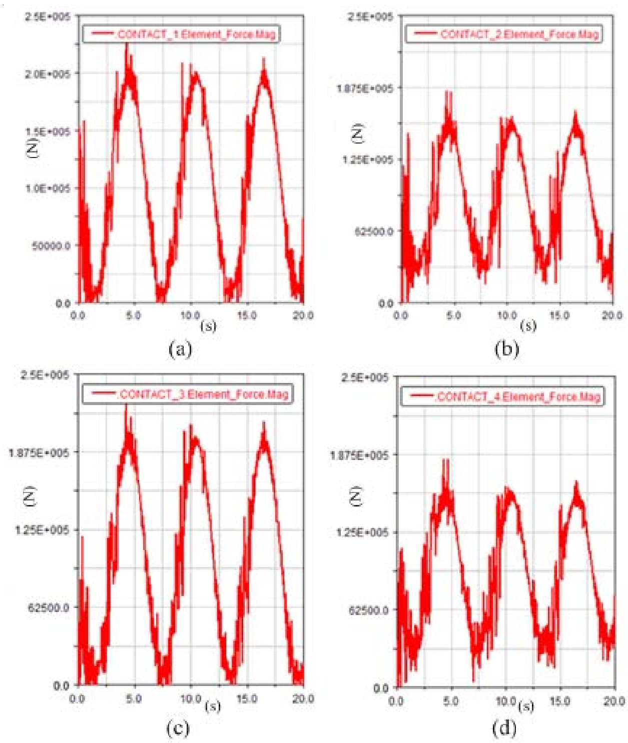


Fig. 7 Contact forces of the load-bearing wheels

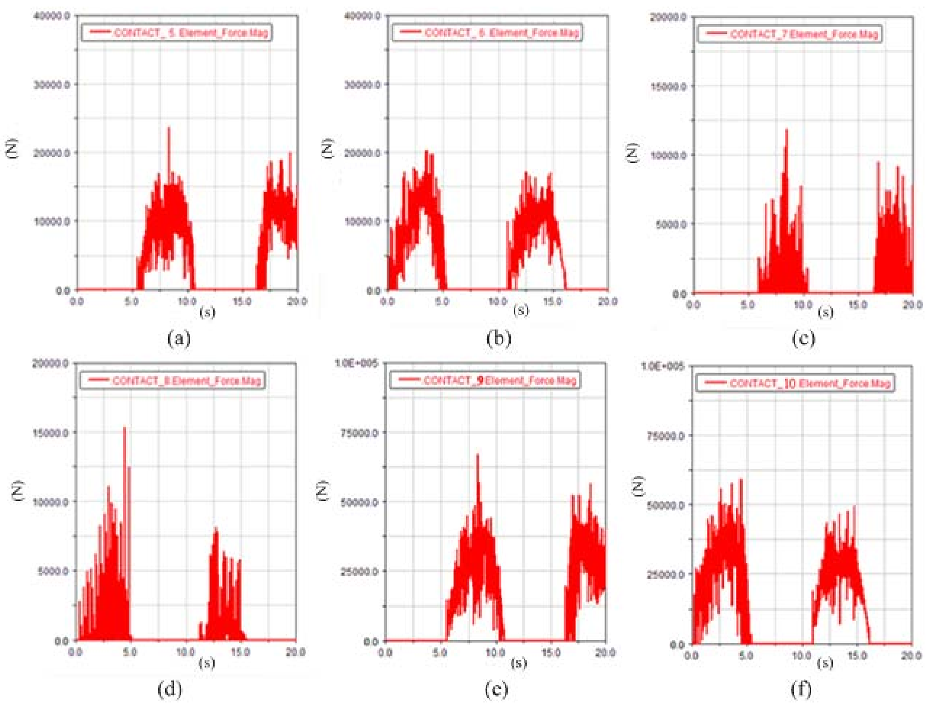


Fig. 8 Contact forces of the horizontal wheels

At the sea state 5, motor is required to drive the vehicle to move along the track. Driving torque of motors should be large enough to overcome not only the friction caused by self weight, but the inertia force resulting from pitch as well. If motor's driving torque is not sufficiant, the vehicle may fail to complete operation mission. Therefore, it is necessary to calculate driving torque of motors to ensure that the output power of selected motor meets design requirement.

Motors driving torque is shown in Figure 9. In Figure 9, variation of motors driving torque is nonlinear because of the pitch effect on the vehicle. Though the driving torque of two motors is basically equal, they are not entirely identical. This is because, the transmission of gear and rack is not consistent due to the gap between rack and gear. The results also show that the output driving torque (*16600 Nm*) of selected motor is much larger than calculated driving torque on gears (*8031 Nm*). Thus, the vehicle can operate smoothly with sufficient driving supply.

3.3.2 Dynamics analysis at the sea state 8

The vehicle is fixed on the deck at the sea state 8, and its inertial force resulting from the ship oscillation is shared by 8 lashing bars. To prevent the vehicle departs from the track, it is important to check the allowable tensile stress of the lashing bars.

Mass center displacement of LV-DSRV is shown in Figure 10(a). The displacement in *X* direction of global coordinate system (the longitudinal direction of ship) is represented by solid line, the displacement in Y direction of global coordinate system (the vertical direction of ship) is represented by dotted line, and the displacement in Z direction of global coordinate system (the horizontal direction of ship) is represented by point line. Mass center speed of LV-DSRV is shown in Figure 10(b), the speed in *X*, *Y* and *Z* directions are represented by solid line, dotted line and point line respectively. As the vehicle does not move along the track at the sea state 8, there is no change of mass center speed and displacement in *X* direction.

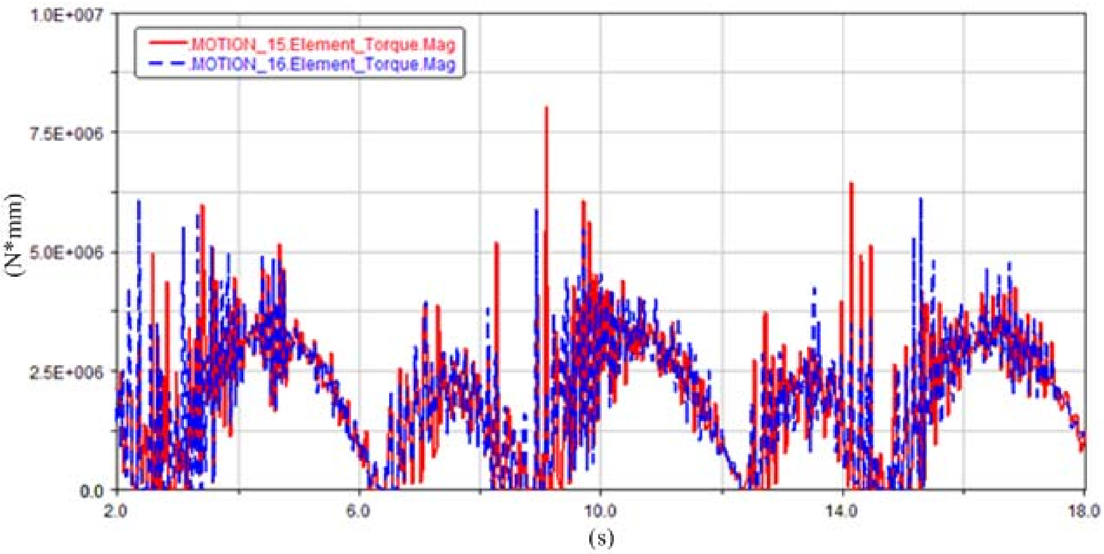
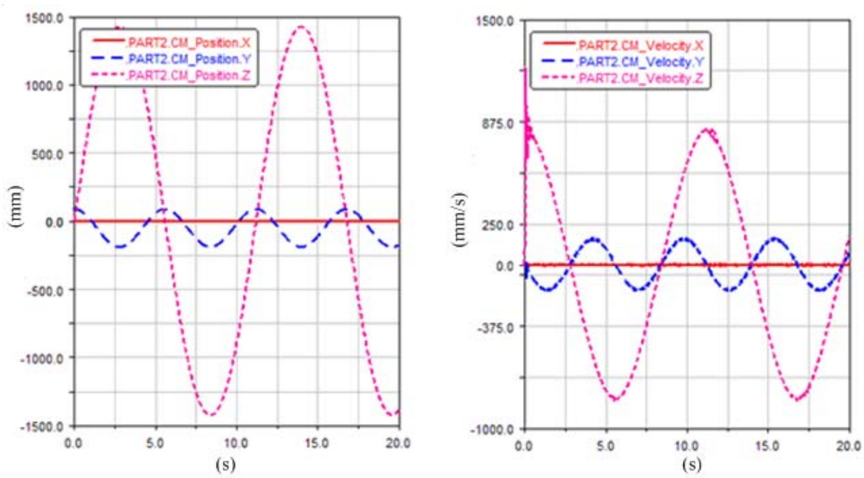


Fig. 9 Driving torque of two motors



(a) Displacement (b) Speed

Fig. 10 Kinetic property of the vehicle's mass center

Tensile forces of the lashing bars from 1 to 8 are shown in Figure 11. The results show that the tensile force varies periodically under wave actions, and it is related to acceleration due to the distance from the mass center of the vehicle to the mass centers of the lashing bars.

Maximum tensile forces of 8 lashing bars are listed in Table 2. The parameters of the lashing bars are: yield limit of *40 Cr* is *340 MPa*, cross-sectional area is *400* mm^2 and the maximum tensile force is *71.38 kN*. Therefore, maximum tensile stress of 8 lashing bars is calculated as *178.45 MPa*, which is less than the yield limit of *40 Cr* and meets allowable tensile stress requirement.

Table 2 Maximum tensile force of lashing bars (kN)

No.	1	2	3	4	5	6	7	8
Value	29.057	28.96	71.38	71.236	66.046	65.394	53.638	52.914

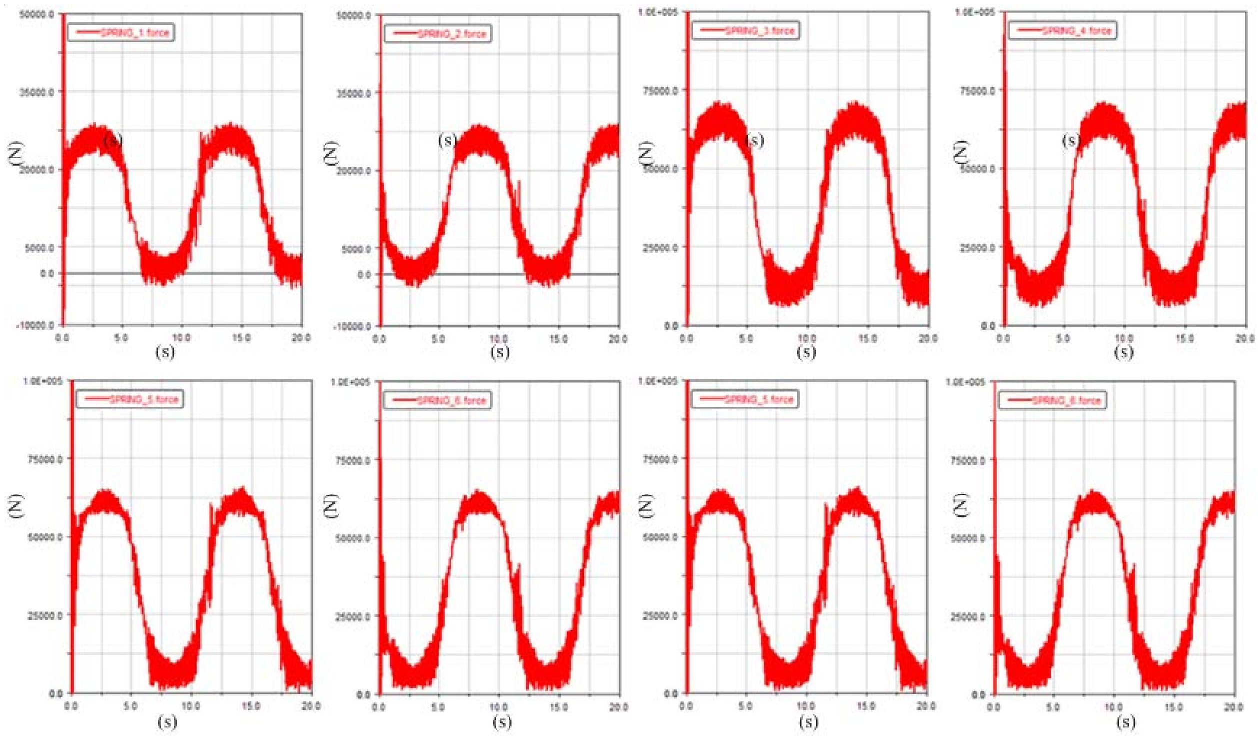


Fig. 11 Tensile forces of 8 lashing bars

4. IMPROVEMENT OF DESIGN

Two improvements of the design as a kind of the optimization strategies are proposed based on simulation analysis mentioned above:

(1) Increase the number of horizontal wheels

According to the simulation analysis, the contact force of horizontal wheel is large in ship roll due to the contact area between wheels and the track is small, which tends to cause track bending deformation. In order to reduce lateral deformation of the track and to prevent the effect on the racks, four horizontal wheels (8, 10, 13 and 15 in Figure 2) can be added in the symmetrical position of the original horizontal wheels.

In Tables 3 and 4, maximum contact forces of load-bearing wheels (horizontal wheels) with different numbers of horizontal wheels are compared. The results show that the maximum contact forces of load-bearing wheels (horizontal wheels) are obviously reduced by increasing the number of horizontal wheels.

Table 3 Comparison of maximum load-bearing wheels contact force

Number of horizontal wheels	Maximum contact forces of load-bearing wheels (kN)			
	No.1	No.2	No.3	No.4
4	239.22	211.27	238.72	223.33
8	228.45	232.01	194.56	209.05
12	194.97	189.95	190.90	174.51

Table 4 Comparison of maximum horizontal wheels contact force

Number of horizontal wheels	Maximum contact forces of horizontal wheels (kN)							
	No.	Value	No.	Value	No.	Value	No.	Value
4	7	94.43	9	129.73	14	91.98	16	115.98
8	5	15.44	6	43.19	7	55.71	9	111.24
	12	14.54	11	39.65	14	51.67	16	100.90
12	5	18.19	6	18.23	7	9.63	8	10.12
	9	47.52	10	46.73	11	18.20	12	18.21
	13	10.02	14	10.14	15	47.57	16	46.71

(2) Add gap between horizontal wheels and the track

As there is no gap designed between the wheels and the track, contact force is generated in simulating ship motion with heave-pitch coupling. In optimization strategies, *1 mm* gap is added to reduce the impact between horizontal wheels and the track, and for the convenience of assembly as well. In Figure 12, contact forces between horizontal wheels and the track with and without *1 mm* gap are compared. Wherein, contact force between horizontal wheels and the track without gap is represented by solid line, and contact force between horizontal wheels and the track with *1 mm* gap is represented by dotted line. The results show that the improved strategy can reduce the amount of the contact force between horizontal wheels and the track to almost zero. Therefore, appropriate gap should be added for the stable movement of the vehicle.

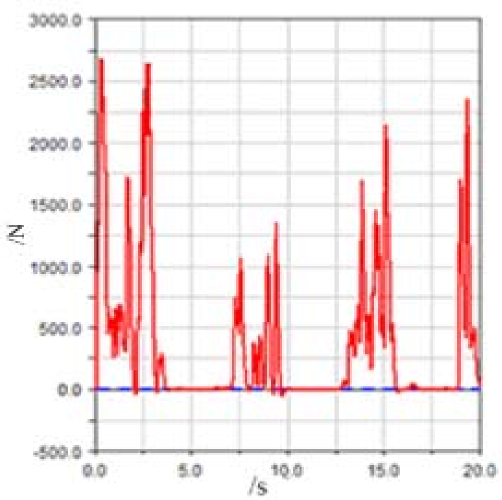


Fig. 12 Comparison of contact forces for cases of 1 mm gap and no gap

5. CONCLUSION

Dynamics simulation is a crucial issue for a successful design of marine special mechanism. However, the current design scheme is mainly obtained by general design specifications and experiences, and may not be a good one because of special design requirements. This paper presents a dynamics simulation model of LV-DSRV based on sea state excitation. Wave excitation input at the sea state 5 and 8 are considered as the limit marine working environments. Kinetic property of LV-DSRV is calculated making use of virtual prototype technology, and mechanical characteristics of key parts are also analyzed. Optimization strategies are proposed and verified by increasing the number of horizontal wheels and adding gap between horizontal wheels and the track. This research can be a case study for design of similar marine special mechanism. Future work is required to optimize design structure of the vehicle in two aspects. The first is to utilize finite element analysis method to achieve stress-strain properties of the track at different sea states. The second is to establish rigid-flexible coupled model for the vehicle.

6. ACKNOWLEDGEMENT

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ISTRAŽIVANJE DINAMIČKE SIMULACIJE TERETNOG VOZILA SMJEŠTENOG UNUTAR VOZILA ZA SPAŠAVANJE PRI DUBOKIM URONIMA (LV-DSRV)

SAŽETAK

Nesreće podmornica mogu uzrokovati ekonomsku i ekološku štetu, te gubitak ljudskih života. Spašavanje podmornica je složen proces kako zbog složenosti samog procesa spašavanja tako i zbog promjenjivosti stanja mora. Uspješnost procesa spašavanja je određena pouzdanošću i sigurnošću sredstva za spašavanje. Stoga je vrlo važno istražiti na koji je način projektirano to sredstvo za spašavanje. Postojeća saznanja pružaju neke empirijske formule kao i shemu projekta pomoću općih pravila. Zbog jednostavnosti analize mehaničkog proračuna, te posebnosti projektnog zahtjeva, može se dogoditi da plan dobivene sheme nije sasvim odgovarajući. Da bi se poboljšala sigurnost i pouzdanost brodskog uređaja u sustavu spašavanja podmornice, u ovom radu predstavljen je model dinamičke simulacije teretnog vozila smještenog unutar vozila za spašavanje pri dubokim uronima (LV-DSRV) na osnovi uzburkanog stanja mora. Ekstremnim radnim uvjetima na moru smatralo se more stanja 5 i 8. Kinetičko svojstvo LV-DSRV-a izračunalo se koristeći virtualno prototipsku tehnologiju pomoću računalnog programa ADAMS, te su se analizirale mehaničke karakteristike ključnih dijelova. Predložena je i potvrđena strategija optimizacije povećavajući broj horizontalnih kotača i veličinu praznog hoda između horizontalnih kotača i tračnice, imajući u vidu i projekte sličnih posebnih pomorskih mehanizama.

Ključne riječi: *brodsko sredstvo, dinamička analiza, stanje mora, spašavanje podmornice.*