Structure optimization and mechanical study on lifting mechanism of the multifunctional car transporter

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ABSTRACT

In order to realize the multifunctional purpose, the rear column of the car transporter is replaced by a lifting mechanism which can be regarded as a four-bar mechanism. However, for the multifunctional car transporter, lifting difficulty and lifting lock occur shortly after the beginning of lifting. Dynamic analysis of the lifting mechanism is carried out, and the computational results reveal that the unreasonable structure of the lifting mechanism is the main reason for the above problems. In order to overcome lifting difficulty and lifting lock, and to improve lifting performance, a back push bar combined lifting mechanism is proposed to replace the original lifting mechanism, and dynamic analysis and finite element analysis (FEA) are carried out in both lifting mechanisms. After using the back push bar combined lifting mechanism, the numerical results show that the required hydraulic cylinder thrust is reduced to a certain degree, and the horizontal pulling force imposed on the front column by the upper platform is declined sharply; the lifting test shows that the lifting process becomes fluent and labor-saving.

1. Introduction

In Chinese market, the lifting mechanism of the car transporter usually includes the front and rear columns, in which the hydraulic cylinder, the sliders, the sprocket wheels and the chain work together, as shown in Fig. 1. The rise and fall of the upper platform is fulfilled by the cooperative movement of components inside of the front and rear columns so as to load or unload the upper cars. However, for this kind of car transporter, existence of the rear column limits the load space, so that it can only load the cars. After arriving at the destination, the car transporter will return emptyly, resulting in a waste of resources and an increase in transportation cost.

In order to solve the above problems, a four-bar mechanism is adopted to replace the rear column, so that space occupied by the rear column is released, and the upper platform can load cargo wider than a car. The upper platform is located at a high position for cars' transportation (Fig. 2a), at a middle position for upper cars' loading or unloading (Fig. 2b), and at a low position for loading container or bus (Fig. 2c), thus to realize the multifunctional purpose.

As a large-scale car transporter, the multifunctional car transporter usually loads the large cars, the mass of a single car usually exceeds 2000 kg. During the lifting process, some problems such as lifting difficulty, lifting lock, large pulling forces imposed on the front column by the upper platform exist, as shown in Fig. 3.

Contraposibly the arisen problems, dynamic analysis of the lifting mechanism has been carried out to discover the causes of the problems in this paper. Structure optimization of the lifting mechanism and corresponding dynamic analysis have also been conducted. In the end, FEA

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1 Under the provisions of the “Regulation on the Administration of Over-limit Transport Vehicles Driving on Highway (2021 revised edition, China)”, total length, width and height of the vehicle and cargo should not exceed 18.1 × 2.55 × 4 m, respectively. The width of the rear column is about 0.13 m, therefore, the rest of space used to load car is about 2.0 m wide.

2 The length, width and height of standard 40 GP container are 12.192 × 2.438 × 2.591 m, respectively. The length, width and height of large-scale bus (12 m length) are 12 × 2.5 × 3.7 m, respectively; of middle-scale bus (8 m length) are 7.999 × 2.45 × 3.365 m, respectively; of small-scale bus (6 m length) are 5.995 × 2.27 × 2.75 m, respectively. The length and width of large-scale bus (55 seats) are 10.549 × 2.5 m, respectively. Upper platform's length of the multifunctional car transporter is 12.53 m, so that it can load the standard 40 GP container, the middle- and small-scale bus, even the large-scale bus.
of the overall vehicle (without consideration of the lower vehicle frame) under five working conditions has been done, and the principal performance parameters calculated in FEA are the required hydraulic cylinder thrust and the horizontal pulling force of the upper platform acting on the front column. There have been some publications addressing the finite element method (FEM) for the vehicle frame applications in recent years (Wei et al., 2021; Qin et al., 2018; Gawande et al., 2018; Zhang et al., 2018; Zagarin et al., 2020; Cai et al., 2017; Li and Feng, 2020; Honda et al., 2021; Lin et al., 2021; Liu et al., 2019; Wang et al., 2019, 2018; Wu et al., 2016). Zhang et al. (2015) discovered a characteristic triangle method to calculate input vectors of scissor lift mechanism and applied it to modeling and analysis. Nguyen et al. (2021) presented the study of the dynamics of a hydraulic static-pile-pressing machine during the process of lifting and slewing a pile using a mounted crane. Jiang et al. (2022) proposed a 2-DOF parallel lifting mechanism of the stereoscopic parking robot, and the forward and inverse kinematics, workspace, and singular configuration analysis were carried out to achieve further analysis of the mechanism. Gu et al. (2014) established a model of a rigid-flexible coupling system to study the fatigue life of the lift mechanism of mining wrecker by integrating multi-body dynamic simulation and FEM. However, reports on applications of FEM to the double-deck car transporter are few and far between.

The paper is outlined as follows. In the following section, dynamic study on lifting mechanism of the multifunctional car transporter is carried out. In Section 3, a back push bar combined lifting mechanism is proposed to replace the original rear lifting mechanism, and dynamic analysis is executed for the new lifting mechanism. FEA of the overall vehicle is performed in Section 4, in which some important physical parameters and stresses and displacements of some principal components are studied. In Section 5, lifting test of the sample vehicle is described. Some concluding remarks are mentioned in Section 6.
2. Dynamic analysis of lifting mechanism of the multifunctional car transporter

2.1. Determination of motion parameters of a four-bar mechanism

Lifting mechanism of the car transporter consists of a big arm, a small arm, a hydraulic cylinder, and two hinged supports welded on the lower vehicle frame, as shown in Fig. 4. The upper platform, the big arm, the small arm and the lower vehicle frame constitute a four-bar mechanism, as shown in Fig. 5.

The upper platform is supposed as component A, its length is $l = 12.53 \text{ m}$, and the distance between two hinged points $O_1O_2$ is $l_A = 11.024 \text{ m}$; the small arm is supposed as component B, its length is $l_B = 0.797 \text{ m}$; the big arm is supposed as component C, its length is $l_C = 2.147 \text{ m}$; the distance between two hinged points $O_1O_2$ is $l_{O_1O_2} = 11.632 \text{ m}$. The included angles between components A, B, C and x direction are $\theta_A$, $\theta_B$ and $\theta_C$, respectively; the acute angle between the hydraulic cylinder and the x direction is $\varphi$. According to the lengths of $l_A$, $l_B$, $l_C$, $l_{O_1O_2}$, and the determination method for the type of a four-bar mechanism (Sun et al., 2006), it can be determined that the lifting mechanism of the multifunctional car transporter is a double-rocker mechanism with a definite motion trajectory.

The hinged support of the big arm $O_2$ is appointed as the origin of coordinates, the coordinates of the pin shaft $O_1$ used to link the upper platform and the front column are set as $(l_1, l_y)$, in which $l_1 = -12.339 \text{ m}$, $l_y = 1.983 \text{ m}$; and the length of each component is projected to the x and y directions, then the following geometric relationships can be obtained as

\begin{align}
l_{x} &= l_A \cos \theta_A + l_B \cos \theta_B + l_C \cos \theta_C \tag{1} \\
l_{y} &= l_A \sin \theta_A + l_B \sin \theta_B + l_C \sin \theta_C \tag{2}
\end{align}

Let $P = l_1 - l_C \cos \theta_C$, $Q = l_y - l_C \sin \theta_C$, then Eqs. (1) and (2) can be expressed as

\begin{align}
l_A \cos \theta_A + l_B \cos \theta_B &= P \tag{3} \\
l_A \sin \theta_A + l_B \sin \theta_B &= Q \tag{4}
\end{align}

Squaring both sides of Eqs. (3) and (4), then adding them, we can obtain

\begin{align}
l_A^2 + 2l_A l_B \cos (\theta_A - \theta_B) + l_B^2 &= P^2 + Q^2 \tag{5}
\end{align}

Eq. (5) can also be expressed as

\begin{align}
\theta_A - \theta_B &= \arccos \frac{P^2 + Q^2 - l_A^2 - l_B^2}{2l_A l_B} \tag{6}
\end{align}

Let $\psi = \arccos \frac{P^2 + Q^2 - l_A^2 - l_B^2}{2l_A l_B}$, Eq. (6) can be written as

\begin{align}
\theta_A &= \psi + \theta_B \tag{7}
\end{align}

Substituting Eq. (7) into Eq. (3), we can get

\begin{align}
(l_A \cos \psi + l_B) \cos \theta_B - l_A \sin \psi \sin \theta_B = P \tag{8}
\end{align}

Let $m = l_A \cos \psi + l_B$, $n = l_A \sin \psi$, Eq. (8) can be expressed as

\begin{align}
\frac{m}{\sqrt{m^2 + n^2}} \cos \theta_B - \frac{n}{\sqrt{m^2 + n^2}} \sin \theta_B = \frac{P}{\sqrt{m^2 + n^2}} \tag{9}
\end{align}

Let $\cos \delta = \frac{m}{\sqrt{m^2 + n^2}}$, $\sin \delta = \frac{n}{\sqrt{m^2 + n^2}}$, Eq. (9) can be expressed as Eq. (10)

\begin{align}
\theta_B &= \arccos \frac{P}{\sqrt{m^2 + n^2}} - \delta \tag{10}
\end{align}

The connection relationship between the big arm and the hydraulic cylinder is shown in Fig. 6, and the coordinates of the hydraulic cylinder support $O_3$ are $(x_{cil}, y_{cil})$, in which $x_{cil} = -0.925 \text{ m}$, $y_{cil} = -0.27 \text{ m}$; the coordinates of the hinged point $O$ between the big arm and the hydraulic cylinder are computed as $(l_C \cos \theta_C$, $l_C \sin \theta_C)$, the distance between two hinged supports $O_2$ and $O_3$ is $a = \sqrt{x_{cil}^2 + y_{cil}^2}$. 

Fig. 4. Lifting mechanism of the multifunctional car transporter.

Fig. 5. A four-bar mechanism.

Fig. 6. Schematic diagram of the big arm and the hydraulic cylinder.
The length of hydraulic cylinder at current position is

\[ l_{act} = \sqrt{\left(x_{act} - l_C \cos \theta_C \right)^2 + \left(y_{act} - l_C \sin \theta_C \right)^2} \]  \hspace{1cm} (11)

Define \( \tan e = \frac{x_{act}}{y_{act}} \), \( \sin e = \frac{y_{act}}{\sqrt{x_{act}^2 + y_{act}^2}} \), the relationship between \( \theta_C \) and \( l_{act} \) from Eq. (11) can be expressed using Eq. (12)

\[ \theta_C = e - \arccos \frac{x_{act}^2 + y_{act}^2 - l_{act}^2}{2l_C} \]  \hspace{1cm} (12)

### 2.2. Calculation of applied forces for each component of the four-bar mechanism

Except for the upper platform, without consideration of the masses of the big arm, the small arm and the hydraulic cylinder, they can be regarded as the two-force bars, in which the direction of force is along its axial direction. In accordance with the requirements of the plus assumption method, all bars are assumed to be subjected to tension. During establishment of the equilibrium equations, right hand Cartesian coordinate system is used.

The upper platform is taken as the research object, and the forces acting on it are shown in Fig. 7, in which, \( \gamma = \pi - \theta_A + \theta_B \) is the mechanism transmission angle; \( G_A \) is the sum of the weight of the upper platform and the loaded cars, the weight of the upper platform is 40 KN, and the weight of the loaded cars is 80 KN, therefore \( G_A = 120 \) KN.

The equilibrium equations along \( x \) and \( y \) directions are established as

\[ \sum F_x = F_{O_{iy}} - F_B \cos \theta_B = 0 \]  \hspace{1cm} (13)

\[ \sum F_y = F_{O_{ix}} - G_A - F_B \sin \theta_B = 0 \]  \hspace{1cm} (14)

where \( F_{O_{ix}} \) and \( F_{O_{iy}} \) are the horizontal and the vertical support reactions applied to the upper platform by the front column through pin shaft, \( F_B \) is the axial force imposed on the upper platform by the small arm.

According to Eqs. (13) and (14), support reactions of the upper platform can be obtained by Eqs. (15) and (16)

\[ F_{O_{ix}} = F_B \cos \theta_B \]  \hspace{1cm} (15)

\[ F_{O_{iy}} = G_A + F_B \sin \theta_B \]  \hspace{1cm} (16)

Hinged point \( O_i \) is set as the origin of the moment, and the moment equation of the upper platform is presented as

\[ \sum M_{O_i} (F) = -F_B \sin \gamma l_A - G_A \cos (\pi - \theta_A) = 0 \]  \hspace{1cm} (17)

The force acting on the small arm \( F_B' \) in Eq. (18) can be obtained from Eq. (17)

\[ F_B' = F_B = \frac{-G_A \cos (\pi - \theta_A)}{\sin \gamma l_A} = \frac{G_A \cos \theta_A}{\sin \gamma l_A} \]  \hspace{1cm} (18)

where \( F_B' \) and \( F_B \) are a pair of action and reaction.

Hinged point \( O \) is taken as the research object, as shown in Fig. 8. The equilibrium equations along \( x \) and \( y \) directions are established as

\[ \sum F_x = F_B' \cos \theta_B + F_C \cos (\pi - \theta_C) + F_{act} \cos \varphi = 0 \]  \hspace{1cm} (19)

Fig. 7. Forces diagram of the upper platform.

Fig. 8. Schematic diagram of forces acting on joint O.

Fig. 9. Required hydraulic cylinder thrust versus elongation of the hydraulic cylinder.

\[ \sum F_y = F_B' \sin \theta_B - F_C \sin (\pi - \theta_C) - F_{act} \cos \varphi = 0 \]  \hspace{1cm} (20)

Using Eqs. (19) and (20), we can obtain the required hydraulic cylinder thrust expressed in Eq. (21)

\[ F_{act} = \frac{\sin \theta_B - \cos \theta_B \tan \theta_C \cdot F_B'}{\sin \varphi + \cos \varphi \tan \theta_C} \]  \hspace{1cm} (21)

### 2.3. Analysis of computational results of the lifting mechanism

It is assumed that lifting process of the upper platform is a uniform lifting movement. The change rules of the required hydraulic cylinder thrust and horizontal pulling force imposed on the front column with elongation of the hydraulic cylinder can be obtained by using Fortran program, as shown in Fig. 9 and Fig. 10.

For the phenomenon of lifting difficulty and lifting lock, the first factor should be considered is whether the hydraulic cylinder thrust can meet the lifting requirement or not. However, it is obvious from Fig. 9 that the maximum value of the required hydraulic cylinder thrust is about 126 KN at the initial moment of lifting, the vehicle-mounted hydraulic system is able to meet the requirement. In fact, lifting difficulty and lifting lock do not occur at the initial moment, but occur shortly after the beginning of lifting, whose angle between the upper platform and the horizontal plane is in the range from 5° to 6°. As we know, transmission angle \( \gamma \) between the upper platform and the small arm is an important physical parameter affecting lifting performance, the larger the better. Transmission angle \( \gamma \) and its variation curve to elongation of the hydraulic cylinder is plotted in Fig. 10. Transmission angle \( \gamma \) is 37.5° at the beginning of lifting, it reaches the minimum value of 33.5°, and then increases steadily with the increase of elongation. The time when the transmission angle \( \gamma \) reaches the minimum value coincides with the moment that lifting difficulty, lifting lock, and maximum horizontal pulling force occur, as shown in Fig. 11. Therefore, we have
the conclusion that the unreasonable structure of the existing rear lifting mechanism is the main reason for the problems.

3. Structure optimization of lifting mechanism and its dynamic analysis

3.1. Structure optimization of the original lifting mechanism

Some limiting conditions should be considered in the structure optimization of the lifting mechanism, for example, the vertical distance between the upper platform and the lower vehicle frame should be no less than 2.1 m, and the existing vehicle-mounted hydraulic system should continue to be used. Considering the limiting conditions, and referring to Mallely lifting mechanism of the dump truck (Li and Liang, 1992), we propose a back push rod combined lifting mechanism to replace the existing rear lifting mechanism in this paper, as shown in Fig. 12. For the optimized lifting mechanism, transmission angle increases gradually with the lifting process, so that it has good lifting performance. For the car transporter adopted the optimized lifting mechanism, the required hydraulic cylinder thrust should be no more than 120 KN, and the horizontal force imposed on the front column should be no more than 20 KN.

3.2. Dynamic analysis of the optimized lifting mechanism

Dynamic analysis of the optimized lifting mechanism is carried out using the same procedure in Section 2. Fig. 13 demonstrates that the required hydraulic cylinder thrust decreases monotonously with elongation of hydraulic cylinder, the maximum value is 114.5 KN, which is reduced by 9.1% compared with that before optimization. Similarly, Fig. 14 demonstrates that the horizontal pulling force imposed on the front column decreases monotonously with elongation of hydraulic cylinder, the maximum value is 15.0 KN, which is reduced by 72.6% compared with that before optimization.
Dynamic analysis of the lifting mechanism is only a sketchy computation, it is necessary to conduct a detailed FEA for the overall vehicle frame.

4. Finite element analysis of the overall vehicle frame

In order to obtain the more reliable performance parameters, and the more accurate strength and deformation distribution of the principal components, FEA of the overall vehicle frame has been carried out.

4.1. Establishment of finite element model

According to the geometric characteristic and load conditions of each component, appropriate elements are selected for simulation. The side beams on both sides of the upper platform, the front column plates, the big arm and the small arm of the lifting mechanism are thin plate structures, so that SHELL181 elements are used to simulate them. BEAM188 elements are used to simulate the cross beams, the longitudinal beams and the pin shaft. The hydraulic cylinder and the chain are simulated by LINK180 elements. Hinge joints located in the places such as the big arm-hinged support, the small arm-upper platform, the hydraulic cylinder-hinged support, and the big arm-small arm-hydraulic cylinder are modeled in use of MPC184 elements (Ma et al., 2020), where the common nodes are generated and are used to create the hinge joints. On the premise of maintaining the integrity of the overall vehicle frame, some details that have no obvious influence on the overall stress distribution, such as small round corners, chamfers and unloaded small round holes, are removed during the simplification of vehicle frame. Finite element model of the lifting mechanism is plotted in Fig. 15.

The front column is connected to the upper platform through the pin shaft, which includes the built-in hydraulic cylinder, a movable sprocket wheel, two fixed sprocket wheels, a chain, a hydraulic cylinder-slider, and a upper platform-slider, as shown in Fig. 16. According to the equivalent principle of the magnitude and direction of forces, the sprocket and chain system inside the front column is simulated by the “equivalent triangle” structure (Lu et al., 1998; Wang and Zhou, 2005). Fig. 17 illustrates a simple sprocket and chain system, if efficiency loss is not considered, the force \( S_1 \) applied to winding in of chain is equal to the force \( S_2 \) applied to winding out of chain. At the same time, the rigid body displacement of \( S_1 \) is equally transferred to \( S_2 \). Points \( a \) and \( b \) are the two tangential points from which the chain winds in and out of the sprocket. Connect points \( a \) and the central point \( O \) of the sprocket to create the first hinged LINK180 element, connect points \( b \) and \( O \) to create the second hinged LINK180 element, and connect points \( a \) and \( b \) to create the third hinged LINK180 element, thus forming the so called “equivalent triangle” structure. It is obvious that the “equiva-
Serpa, 2015; Korayem et al., 2021; Tian et al., 2021; Sun et al., 2005). In order to reduce the static indetermination of the sprocket and chain system caused by rigid body motion, a rigid region is established at the “equivalent triangle” structure to constrain the sprocket and chain system.

The side beams of the upper platform, the front column plates, the big arm and the small arm are all made of T700L high strength steel, whose elastic modulus is 210 GPa, yield strength is no less than 700 MPa, tensile strength is no less than 800 MPa, and Poisson’s ratio is 0.307 (Duarte et al., 2021; Šmak et al., 2021). The material used by other components is Q345B, whose elastic modulus is 206 GPa, yield strength is no less than 340 MPa, tensile strength is in the range from 490 MPa to 620 MPa, and Poisson’s ratio is 0.28 (Wen, 2010).

4.2. Treatments of the boundary conditions and the applied loads

The nodes at the bottom of the front column are imposed on all constraints. In order to simulate the hinge joints between the hydraulic cylinder and the hinged support as well as between the big arm and the hinged support, we release the rotational degree of freedom around the pin shaft and constrain all translational degrees of freedom and other rotational degrees of freedom. The weight of the three cars loaded on the upper platform is 80 KN, which is applied to the contact places between the tires and the upper platform in the form of concentrated force. After the modeling of the upper platform is finished, its weight should be 40 KN, if the mass of the upper platform is not 4000 kg, it can be corrected by mass compensation method (Botero et al., 2017). Considering the fact that the weight of the loaded cars and the upper platform is fixed, no time-dependent loads are considered in finite element model.

4.3. Numerical results and analysis

There are five kinds of typical working conditions, the former three working conditions are for the original lifting mechanism and the later two working conditions are for the optimized lifting mechanism, namely, the upper platform is located in the middle position 8°, which is defined as working condition 1; the upper platform is located in the middle position in the range from 5° to 6°, which is defined as working condition 2; it is also the critical working condition for the original structure; the high position 0° of the upper platform is defined as working condition 3; the upper platform is located in the middle position 8° is defined as working condition 4; it is also the critical working condition for the optimized structure; the high position 0° of the upper platform is defined as working condition 5.

Take the working condition 3 as an example, hydraulic cylinder LINK180 elements 49000 and 49001 are adopted as the research objects, as shown in Fig. 19. Axial stresses of the two elements are extracted as $6.544 \times 10^6$ and $6.528 \times 10^6$ Pa, then multiplied by the cross-sectional area of LINK180 element $2.827 \times 10^{-3}$ m², the required hydraulic cylinder thrusts can be obtained as $18.499 \times 10^3$ and $18.455 \times 10^3$ N, and $18.5 \times 10^3$ N is adopted as the required hydraulic cylinder thrust for element 49000 or 49001. The required hydraulic cylinder thrusts under five working conditions are shown in Table 1.

It can be seen from Table 1 that the required hydraulic cylinder thrust by the back push rod combined lifting mechanism in working condition 4 is 111.3 KN, which is reduced by 9.6% compared with that of working condition 1; furthermore, it is less than 120 KN, the existing vehicle-mounted hydraulic system is able to meet the lifting requirement. For the same position of high position 0°, the required hydraulic cylinder thrust of the optimized lifting mechanism is 60.1 KN, which is larger than 18.5 KN of the original lifting mechanism. This is perhaps the cause of the increased stress in the small arm for the optimized lifting mechanism; however, there is no need to worry about the lifting performance as long as the required hydraulic cylinder thrust is less than 120 KN.

![Fig. 16. Structure diagram of the front column.](image1)

![Fig. 17. The “equivalent triangle” structure.](image2)

![Fig. 18. The equivalent finite element models of the sprocket and chain system, (a) Display of element shapes based on real constant descriptions is Off, (b) Display of element shapes based on real constant descriptions is On.](image3)
Table 1. Required hydraulic cylinder thrusts under five working conditions.

| Working condition | 1   | 2    | 3    | 4    | 5    |
|-------------------|-----|------|------|------|------|
| Required hydraulic cylinder thrust (10^6 N) | 123.1 | 120.7 | 18.5 | 111.3 | 60.1 |

Fig. 19. Computation of hydraulic cylinder thrust via LINK180 elements 49000 or 49001.

Table 2. Horizontal forces acting on the front column under five working conditions.

| Working condition | 1   | 2    | 3    | 4    | 5    |
|-------------------|-----|------|------|------|------|
| Horizontal pulling force (10^6 N) | 56.0 | 15.0 | 4.1   |

As previously mentioned, one end of the pin shaft is connected with the upper platform, and the other end is connected with the upper platform-slider, so that it is the force carrier between the upper platform and the front column. Node 50445 shown in Fig. 20 is the connecting node between the pin shaft and the upper platform-slider, its reaction forces caused by the front column can be obtained through Reaction Solu of the General Postproc of ANSYS17.0. The horizontal components of the forces acting on the front column under five working conditions are shown in Table 2.

As can be seen from Table 2, for the optimized lifting mechanism, the horizontal pulling force imposed on the front column by the upper platform is 15 KN at the beginning of lifting process, which is far less than that of the original lifting mechanism. Compared with 50.9 KN in working condition 1, the horizontal pulling force in working condition 4 is reduced by more than 70%. As we know, the greater the horizontal pulling force imposed on the front column, the greater the deformation of the front column will produce, which is prone to generate cracks at the welding positions located in the bottom of the front column. Therefore, it is of great significance to reduce the horizontal pulling force of the front column in practical application.

Stresses distribution of components in the front column under the working condition 4 is shown in Fig. 21. It is obvious that stress concentration is created in some places, such as the lug used to fix the chain and the upper platform-slider used to connect the pin shaft. Maximum stress value reaches 595 MPa, which appears in the vicinity of pin hole of the upper platform-slider. Similar situations also happen to other working conditions. The maximum values of von Mises stresses exceed the allowable stress of T700L high strength steel more or less, which reveals that these parts are prone to fatigue failure in actual applications.

Displacement vector sum nephogram of the upper platform bearing the weight of the loaded cars and its own weight is shown in Fig. 22. The maximum deflection occurs in the middle of the upper platform, reaching 6.6 cm. If the deformation is wished to be reduced, measures such as increasing the number or (and) the wall thickness of the cross beams can be taken to improve the structure stiffness.

The small arm and the big arm are the principal force bearing components of the lifting mechanism. Von Mises stress nephogram and its
local enlarged diagram of the small arm for middle position $8^0$ before optimization (Fig. 23a) and after optimization (Fig. 23b) are plotted. In the same way, Von Mises stress nephogram and its local enlarged diagram of the big arm for middle position $8^0$ before optimization (Fig. 24a) and after optimization (Fig. 24b) are obtained. For the optimized lifting mechanism, the small arm has become the main load.
The bearing component, its von Mises stress increases obviously, and there is obvious stress concentration near the round hole of the bolt. The big arm ceases to be the main load bearing component, and the values of deformation and stress become small, however, it is not appropriate to reduce the thickness of wall excessively. Otherwise, the stiffness loss caused by the oil cylinder passing through the middle of the big arm cannot be compensated for. Of course, the phenomenon of stress concentration also happens near the round hole of the bolt for the big arm.

5. Lifting test of the sample vehicle

In order to verify the correctness of dynamic analysis and FEA, the back push rod combined lifting mechanism is adopted to carry out lifting test, as shown in Fig. 25.

Test parameters of the hydraulic cylinder are shown in Table 3.

Table 3. Test parameters of the hydraulic cylinder.

| Test parameter     | Unit | Value |
|--------------------|------|-------|
| Diameter of the oil cylinder | mm   | 100   |
| Diameter of the push rod | mm   | 60    |
| Trip               | mm   | 910   |
| Maximum working pressure | MPa | 16    |
| Test pressure      | MPa  | 24    |

![Fig. 25. Lifting test of the sample vehicle.](image)

![Fig. 26. Comparison of the numerical results and test solutions.](image)

Comparisons such as the frictional resistance in hinge joints, a small leakage of hydraulic oil, and the accuracy of measuring instruments, test values of the required hydraulic cylinder thrust are larger than those of the numerical results slightly. Moreover, after adopting the optimized lifting mechanism, the lifting process has turned fluent and stable. Lifting test of the sample vehicle verifies the rationality of the optimized lifting mechanism and the validity of the finite element model of the lifting mechanism.

6. Conclusions

(1) Lifting difficulty of the upper platform happens to a type of multifunctional car transporter. Dynamic analysis has revealed that unreasonable structure design of the lifting mechanism is the main reason for the lifting problems.

(2) Consideration the installation sites of the components of the lifting mechanism and the actual requirements of the car transporter, the back push bar combined lifting mechanism is proposed to replace the existing lifting mechanism. After the optimization, the required hydraulic cylinder thrust has been reduced to a certain degree, and the horizontal pulling force imposed on the front column has been reduced obviously; moreover, lifting process becomes smooth and efficient, which can be verified by the lifting test of the sample vehicle.

(3) FEA is carried out for the lifting mechanism of the multifunctional car transporter under five working conditions. As listed in the former data, some important performance parameters such as the required hydraulic cylinder thrust and the horizontal pulling force imposed on the front column are basically similar on the occasions of dynamic analysis, FEA, and lifting test (the pulling force imposed on the front column is not measured in the lifting test). At the same time, the stress and displacement distributions of some structures are also studied, such as the front column, the upper platform, the big arm, and the small arm.

(4) To improve lifting performance, it would be better to keep transmission angle monotonously increasing during lifting process. For the lifting mechanism, the locations and sizes of the hydraulic cylinder, the big arm, and the small arm have important influence on transmission angle.

Declarations

Author contribution statement

Qian-Yue Nie: Performed the experiments; Analyzed and interpreted the data; Wrote the paper.
Zhi-Feng Nie: Conceived and designed the experiments; Contributed reagents, materials, analysis tools or data; Wrote the paper.
Kai Wang: Analyzed and interpreted the data; Contributed reagents, materials, analysis tools or data.
Ting-Rui Liu: Conceived and designed the experiments; Contributed reagents, materials, analysis tools or data.

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Data availability statement

Data will be made available on request.

Declaration of interests statement

The authors declare no conflict of interest.
Additional information

No additional information is available for this paper.

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