FRAME STRUCTURE DESIGN AND FINITE ELEMENT ANALYSIS OF CORN COMBINE HARVESTER FOR HILLS AND MOUNTAINS
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丘陵山地玉米联合收获机车架结构设计与有限元分析

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DOI: https://doi.org/10.35633/inmateh-68-13

Keywords: *Corn harvester, frame, strength analysis, modal analysis, optimization design, stability analysis*

ABSTRACT
In view of high center of gravity and poor stability of traditional corn harvesters, a corn combine harvester frame is designed for hill and mountain operations based on TRIZ theory. The frame supports engine mode of middle engine rear drive, consisting of a front frame and a rear frame. The tail of the front frame is welded under the head of the rear frame. The front frame has reduced height and increased width to allow lower center of gravity and better stability of the whole machine. The left and right longitudinal beams of the front frame have different heights to allow better trafficability of the whole machine. A 3D model is established using SolidWorks software and incorporated with ANSYS software to perform finite element analysis and modal analysis on the frame. It turns out that under full-load bending and full-load torsion conditions, the frame strength and stiffness meet the mechanical performance requirements, and the frame displays fine dynamic characteristics. According to the analysis results, the frame is optimized under the goal of light weight. While the frame strength and stiffness requirements are met, the frame mass is lowered by changing the frame component thickness. After optimization, the entire frame volume is reduced by 14.27%, with mass reduced by 14.3%, and the strength and stiffness conform to the requirements, thus achieving lightweight optimization of the frame. Moreover, the stability analysis of the corn combine harvester shows the overturning angle of uphill is 45.3°, the overturning angle of downhill is 45.7°, and the overturning angle of slope is 40.2°.

摘要
为解决传统玉米收获机重心高、稳定性差等问题，基于 TRIZ 理论设计了一种适合丘陵山地作业的玉米联合收获机车架。该车架适合于发动机中置后驱形式，车架由前框架和后框架组成，前框架尾部焊接在后框架头部之下，前框架高度降低、宽度增加，有利于降低整机重心，提高整机稳定性，前框架左右纵梁高度不同，提高整机的通过性。利用 Solidworks 软件建立三维模型，导入 ANSYS 软件对车架进行有限元分析和模态分析，结果表明在满载弯曲和满载扭转工况下，车架强度和刚度均满足力学性能要求，车架动态特性良好。根据分析结果，以轻量化为目标对车架进行优化，在满足车架强度和刚度要求的前提下，通过改变车架构件板厚的方式实现车架的轻量化，优化后整个车架体积减少 14.27%，质量减少 14.3 %，强度和刚度均符合要求，达到了车架轻量化优化目的，最后整车稳定性分析表明收获机上坡极限倾翻角度为 45.3°，下坡极限倾翻角度为 45.7°，横向极限倾翻角度为 40.2°。

INTRODUCTION
Corn, one of the three major food crops in China, had a planting area of about 4×10^7 hm^2 in 2022. The corn planting area on hills and mountains exceeds 1.2×10^7 hm^2, accounting for a non-negligible proportion *(Wei et al., 2022). D*ue to the limitation of terrain conditions, hills and mountains have far lower level of mechanized corn harvesting than plain areas. Hence, it is of great significance to study and develop corn harvesters for hills and mountains. Hilly land is uneven, with multiple potholes and great slope, which imposes high requirements on the trafficability and stability of corn harvester *(Xu et al., 2021*).

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The frame bears the load impact from various working parts of the corn harvester and receives excitation from the hilly road, playing an important role in determining trafficability and stability of the whole machine.

There are many important theoretical achievements in terms of frame design, mainly focusing on the frame structure optimization. For example, Wu et al. conducted finite element modal analysis on the frame of the hillside orchard transporter, changed the middle beam position based on topology optimization to achieve model improvement (*Wu et al., 2016)*. Zhang et al. modified the rice transplanter frame through goal-driven optimization, with mass decreased by 16.77% after optimization (*Zhang et al., 2014*). Using finite element software, Mohd et al. analyzed the frame stress, deformation and fatigue characteristics under dynamic and static loads to optimize and perfect the local frame structure. There are also some achievements in frame design pattern of corn harvester (*Mohd et al., 2012; Masanori H., 2022; S. Krishna et al., 2014*). For instance, Ma Lina et al. performed function analysis of corn harvester based on axiomatic design, extracted the main structural design parameters of the side beam frame, and then designed the frame using mathematical language (*Ma et al., 2019*). Gong Jingfeng et al. designed a trapezoid frame parametric platform in MATLAB/GUI software environment to perform frame structure design. Nevertheless, these research methods are to optimize the traditional frame structure, without innovative changes in the frame structure, and the frame design process fails to take into account particularity of hill and mountain operations *(Gong et al., 2021)*. Accordingly, it is particularly urgent to design a corn harvester frame for hills and mountains.

This paper bases itself on the TRIZ theory, analyzes the impact of frame installation height of the corn harvester on the stability and trafficability of the whole machine. Adhering to the physical contradiction solving principle and idea, a staggered frame is designed for corn harvester for hills and mountains to increase the whole machine trafficability and stability in hill and mountain operations. To determine the frame structure reliability, stress and deformation of the corn harvester frame are analyzed using ANSYS software under two typical working conditions: full-load bending and full-load torsion, so that frame modal analysis can be made. Lightweight improvement of the frame is performed by optimizing sectional dimension. This research carries practical significance for the development and design of harvester frame for hill and mountain operations.

MATERIALS AND METHODS

Overall frame design based on TRIZ theory

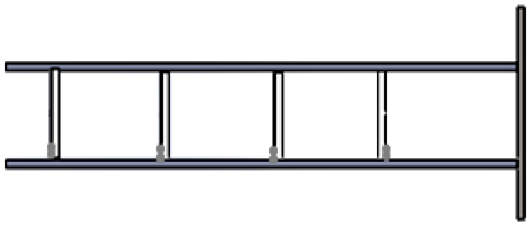
TRIZ represents a new theory that comprehensively and systematically discusses and solves invention-creation problems to achieve technological innovation. It provides scientific principles for humans to obey in the process of invention, creation and technical problem solving *(Zhang et al., 2014; Li et al., 2022)*. According to the TRIZ conflict resolution principle, demand analysis is made on corn harvester frame to innovatively design the corn harvester frame structure for hill and mountain operations, so that whole machine stability is higher.

An effective approach to increase roll stability of the corn harvester is to reduce the height-width ratio of the whole machine. The specific measures are to reduce the height of the center of gravity and increase the wheel base. As an important component of the corn harvester, the frame bears and connects the working parts such as the engine, gearbox, cab, header, peeler and granary. Reduced frame installation height of the whole machine helps lower its center of gravity, but will also reduce the ground clearance and then impair the field trafficability of the whole machine. The frame should be installed both low and high, resulting in conflicts between geometric physical parameters. By solving physical contradictions in TRIZ theory, innovative design of a higher level is possible. The core idea for solving physical contradictions is to separate the two contradictory sides by spatial separation *(Li et al., 2022)*. The recommended invention principles for space separation include 1 division, 2 extraction, 3 local masses, 4 asymmetry, 7 nesting and 17 shift from one-dimension to multi-dimension. Comparison between the above 6 invention principles reveals that, for the innovative frame structure, 1 division, 4 asymmetry and 17 dimension change are more valuable invention principles.

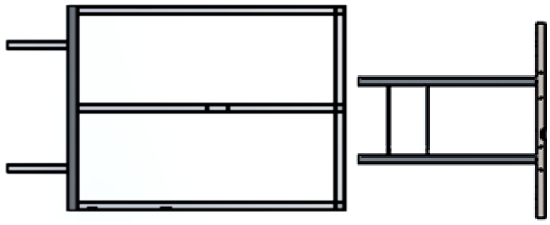
Normally, the frame of corn combine harvester adopts a side-beam trapezoidal frame structure, which is welded by longitudinal beam, cross beam, reinforcing rib, etc. *(Ma et al., 2019)*, as shown in Figure 1(a). According to the segmentation principle, solution 1 is proposed: front and rear frame structure. The conventional side-beam integrated trapezoid frame consists of two parts: the front frame and the rear frame.

The front frame bears the header, engine, transmission, elevator and peeler. The front frame has reduced height and increased width to lower the center of gravity of the whole machine, with roll stability of the whole machine increased; the rear skeleton bears the ear box at a height to maintain grain unloading efficiency of the ear box, as shown in Figure 1(b).

According to the dimension change principle, solution 2 is proposed: the front and rear layered frame. The front and rear frames are arranged in layers, the rear frame head is welded on the front frame tail, and the front frame is installed with engine and gearbox. The engine has greater geometric dimension than the gearbox. By lowering the engine installation position, it is possible to greatly reduce the center of gravity of the whole machine, as shown in Figure 1(c). According to the asymmetry and dimension change principle, solution 3 is proposed: staggered frame structure. The left and right longitudinal beams of the front frame are changed in height so that one longitudinal beam is placed under the front frame beam, while the other longitudinal beam has the same height as the front frame beam, making the cross beam and the form a staggered frame structure, as shown in the Figure 1(d).

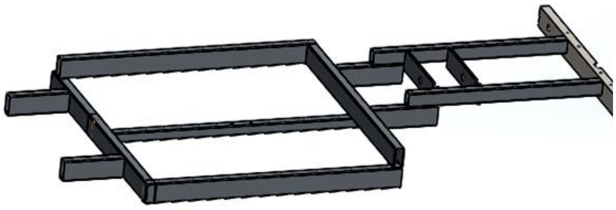


(a) Side-beam trapezoid frame



(b) Front and rear separation frame

(c) Front and rear separation layered frame



（d）Staggered frame

Fig. 1 - The frame design evolution process of corn harvester

Attach the engine, gearbox, cab, header, peeler, granary and other parts to the frame. The structure of the corn combine harvester is shown in Figure 2.

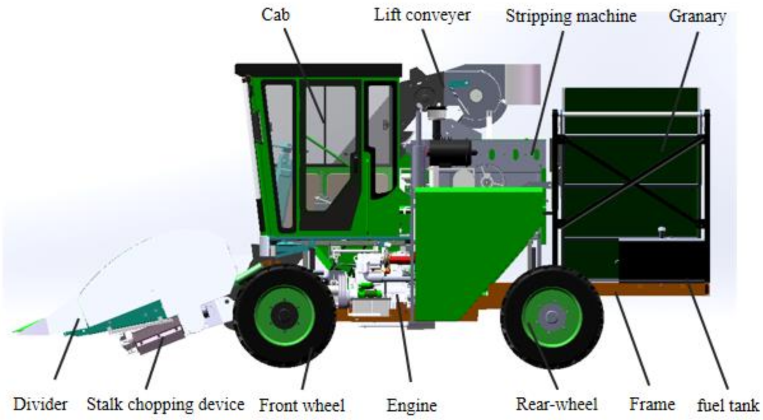


Fig. 2 - The structure diagram of corn harvester for hilly area

Design of main structural parameters of the frame

The frame length has something to do with the front and rear wheelbases. The wheelbase should be decided based on factors such as vehicle performance, loading area and axle load distribution. The shorter, the better. According to the preset whole machine dimension, the wheelbase is initially determined to be 2400 mm, and the frame length is generally1.3~1.7 times that of wheelbase. The frame length (that is, the length of the longitudinal beam) is thus selected as 3900 mm to basically meet the above requirements.

Frame width is the distance between the outer edges after welding of the two longitudinal beams and the cross beam, which is subject to limitation of the wheelbase and the suspension elastic element. Judging from the need for better handling stability, the wider the frame, the better, but too wide frame will make whole machine mass greatly increased, which harms economic and power performance. Accordingly, both aspects need to be taken into consideration.

Based on comprehensive consideration of the body layout, the front frame width is 1140 mm, and the rear frame width is 800 mm. The frame height is the vertical distance between the frame bottom and the frame top, which is 360 mm herein. The frame is made of 45Mn square tube with a rectangular section and a thickness of 6 mm.

Finite element static and modal analysis of the frame

The established Solidworks frame geometry model is imported into ANSYS software, and the frame structure is simplified with the main mechanical properties of the frame structure maintained. The frame is welded by various components, and the component joints are bound by contact to form a rigid connection. The frame material properties are described in Table 1. In this calculation, SOLID 95 element is taken, the element length is 10 mm, and 447,366 elements and 108,222 nodes are derived from mesh division of the frame.

Table 1

Material parameter

Project	parameter	unit
Material	45 Mn	-
Poisson's Ratio	0.28	-
Elastic Modulus	210	GPa
Density	8	$g \cdot cm^{-3}$
Yield Strength	375	MPa
Tensile Strength	620	MPa

Each part assembly on the frame directly acts on the corresponding frame parts with a uniform load *(Wang et al., 2014; Badretdinov et al., 2020)*, and the frame dead-weight can be automatically entered by setting the material density software. The loads required for the calculation and its definitions are illustrated in Table 2.

Table 2

Frame load values and loading modes in the finite element analysis

Name	Load value（N）	Loading mode
Header	8250	uniform load
Engine	3310	uniform load
Transmission	2520	uniform load
Cab	2480	uniform load
Elevator	2575	uniform load
Peeler	5935	uniform load
Granary (full load)	16120	uniform load
Frame dead-weight	5160	inertial load
Total	46350	

Considering working environment and stress conditions of corn harvester for hills and mountains, the constraints of the analysis model are determined: 1) Full constraints are applied at the welding point between the front axle and the left and right longitudinal beams; 2) 5 degrees of freedom are constrained at the two ends of the shaft extending from the rear axle shell, with rotational degrees of freedom released around the axis (*Ali Mohammod et al., 2021)*.

RESULTS AND ANALYSIS

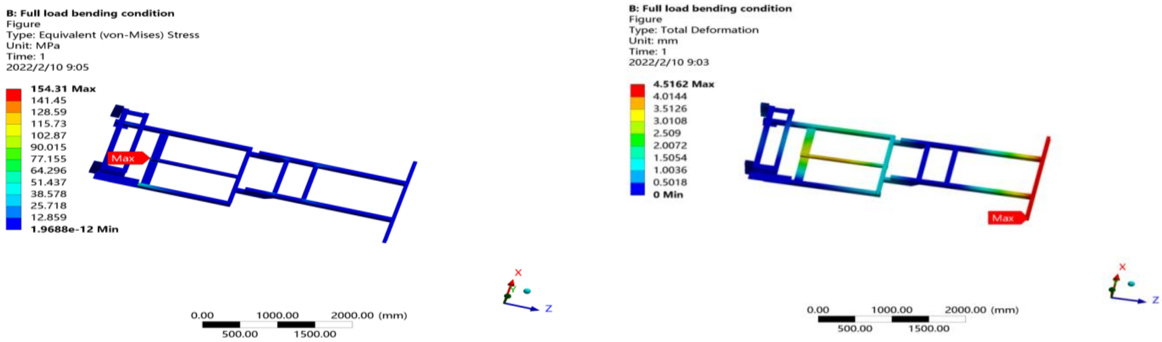
Corn harvester for hills and mountains face relatively complicated operating environment. In view of its operating conditions, two typical working conditions are selected for analysis: the full-load bending condition, which is a working condition of the corn harvester during normal operation; the full-load torsion condition, which is an abominable working condition for corn harvester.

Analysis of the frame under full load bending condition

Under full-load bending condition, when the corn harvester is running at full load, the wheels are in full contact with the ground, the wheels are on the same working plane, and the whole machine is in a balanced state.

Under full-load bending condition, corn harvester for hills and mountains has relatively low operating speed, and the main load on the frame is the static load at full load. To check the structural strength and stiffness of the frame, the full-load bending safety factor is selected as 1.7 *(Wang et al., 2011)*, and the allowable stress of the frame is 220 MPa (375/1.7) now. Under the full-load bending condition, the stress of the corn harvester frame is distributed as shown in Figure 3(a). The overall stress on the frame is small, and the maximum stress on the frame appears at the juncture between the front and middle beams of the front frame, which is 154 MPa and smaller than the allowable material stress of 220 MPa, so frame strength meets the mechanical performance requirements under full-load bending condition.

Under full-load bending condition, the deformation and displacement of the corn harvester frame are distributed as shown in Figure 3(b). The maximum frame displacement appears at the tail beam of the frame, with a value of 4.5 mm. The reason is that it bears the granary weight, with a location far from the frame supporting point. Seen from the overall displacement cloud map, the front frame has greater deformation in the middle, with a value of about 2.9 mm. This relates to the actual load distribution, and conforms to the actual situation. The deformed frame has a safety factor of 2.4 (375/154), indicating good deformation resistance of the frame. In addition, the deformation of the longitudinal beams is symmetrically distributed on the left and right sides of the frame, resulting in a good load ratio of the frame.

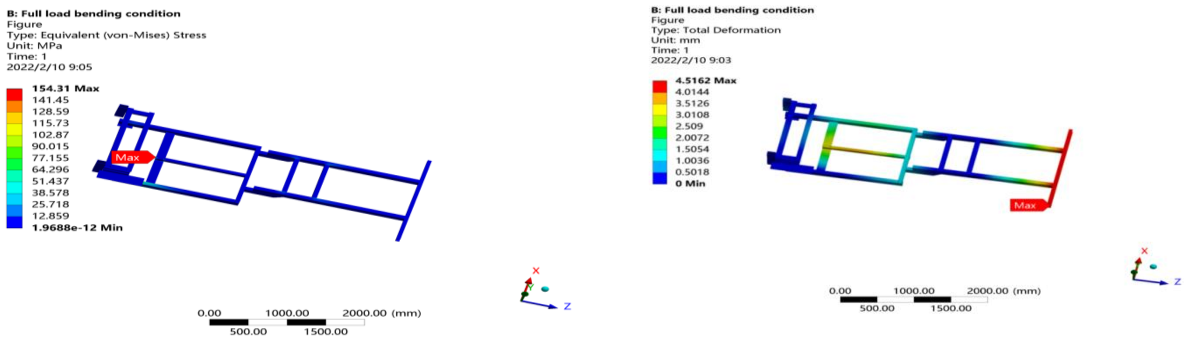


（a）Stress distribution diagram （b）Deformation distribution diagram

Fig. 3 - Full load bending condition

Analysis of the frame under full-load torsion condition

Hilly areas feature complicated terrain, uneven land and widespread potholes. When the corn harvester drives on potholes, the contact point between the wheel and the ground is not on the same plane, with the frame subject to torsional deformation under torsional load. The torsional condition studied herein is when the left front wheel is in the suspended state, and the full load torsional safety factor is selected as 1.3 *(Ma et al., 2019*). The allowable stress of the frame structure is 288 MPa at this time. Under full-load torsion condition, the stress of the corn harvester frame is distributed as shown in Figure 4(a). The maximum frame stress is 246 MPa at the juncture between the front and middle beams of the front frame, and there is greatly increased stress at the left column juncture of the front frame, though smaller than the allowable frame stress. Hence, frame strength meets the mechanical performance requirements under full-load torsion condition.



(a) Stress distribution diagram (b)Deformation distribution diagram

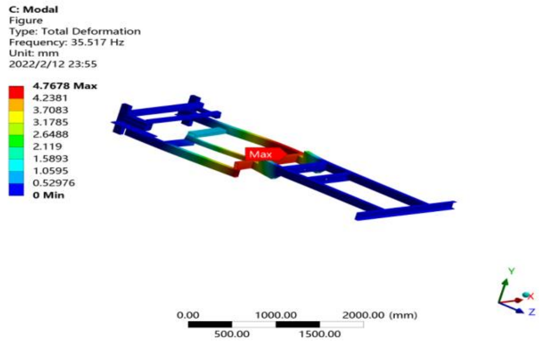
Fig. 4 - Full-load torsion condition

Under full-load torsion condition, the deformation and displacement of the corn harvester frame is distributed as shown in Figure 4(b). The maximum frame displacement occurs at the middle beam of the front frame, with a value of 5.1 mm, which is within the allowable range and conforms to the actual stress situation.

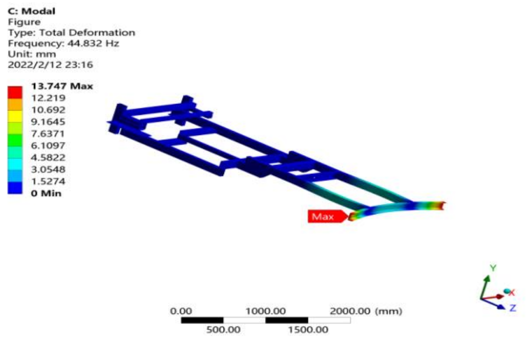
Finite element analysis of the frame under the full-load static state indicates that, under full-load bending and torsion conditions, the frame structure strength and stiffness meet the usage requirements. Mastery of the load distribution and weak frame parts under the full-load condition provides a theoretical basis for frame improvement in trial manufacturing.

Modal analysis of the frame

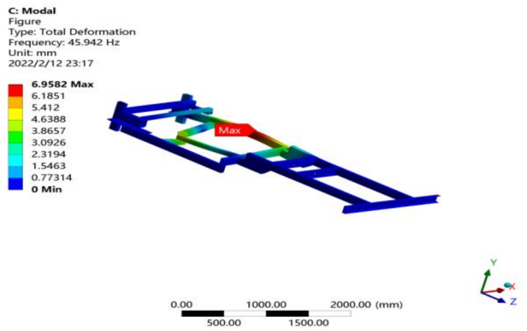
Modal analysis is performed on the corn harvester frame, and the calculated natural frequency is compared with the external frequency to analyze the reliability of the designed frame. Figure 5 displays the first sixth-order modes of the Modal module solved by ANSYS. Solution of the modes takes full-load bending condition as an example. Table 3 lists the sixth-order mode of the frame and its corresponding vibration modes.



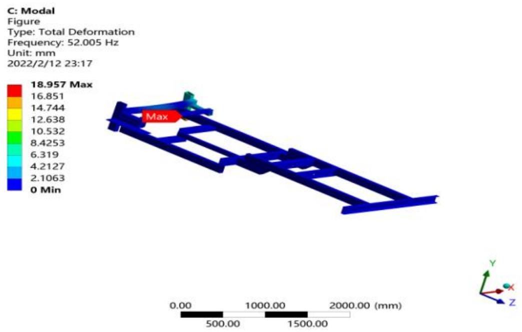
(a) The first-order modal shape diagram



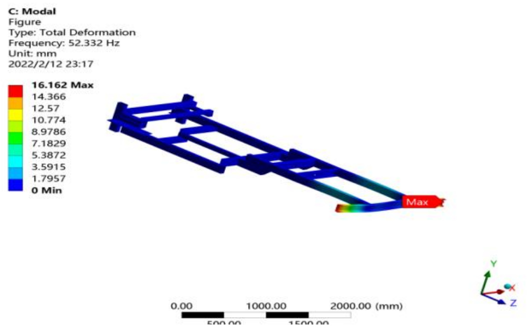
(b) The second-order modal shape diagram



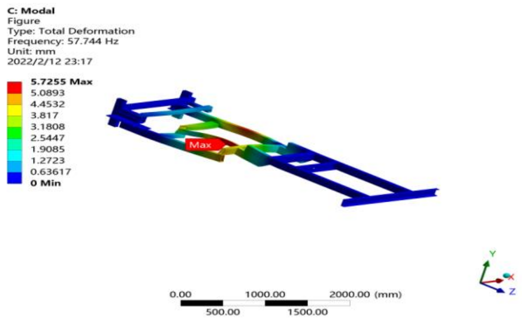
(c) The third-order modal shape diagram



(d) The fourth-order modal shape diagram



(e) The fifth-order modal shape diagram



(f) The sixth-order modal shape diagram

Fig. 5 - Sixth-order Modal Shape Diagram of the frame

Seen from vibration mode of each order in Table 3 and Figure 5, the natural frequency of the corn harvester frame is uniformly distributed, displaying reasonable vibration mode. Vibration deformation mostly occurs in the middle of the front frame and the tail of the rear frame. Regarding the great deformation in the middle of the front frame, it is because the front frame bears components like engine, transmission, with the frame heavily loaded. For the great deformation of the frame tail, it is because the outer beam of the compartment at the frame tail is an overhanging beam structure prone to vibration deformation. In addition, other frame parts have relatively small vibration deformation, featuring smooth vibration mode and relatively stable structure.

Table 3

Modal frequency of frame structure

Mode Order	Frequency (Hz)	Vibration mode
1	35.517	Bending vibration
2	44.832	Bending vibration
3	45.942	Bending-torsion combination vibration
4	52.005	Bending-torsion combination vibration
5	52.332	Bending-torsion combination vibration
6	57.744	Bending-torsion combination vibration

The modal analysis result is compared with some important excitation forces of the frame under actual working conditions to analyze rationality of the designed vehicle frame. During normal driving, corn harvester is exposed to road excitation, engine excitation, wheel imbalance excitation and drive shaft excitation. The road excitation frequency is generally within 3 Hz, the wheel unbalance excitation is generally within 5 Hz. The engine speed frequency varies depending on the engine speed, and the engine vibration frequency can be calculated via the following formula *(Huang et al., 2020)*:

$$f = \frac{2nm}{60\,p} \tag{1}$$

Where:

f stands for the engine speed frequency.

n denotes engine speed;

m represents number of engine cylinders;

p stands for the number of engine strokes;

The transporter herein has a four-cylinder four-stroke engine with a maximum speed of 2500 r/min. According to the formula calculation, the engine has a vibration frequency of 83.3Hz under the maximum speed. The engine excitation frequency during normal driving is much lower than the vibration frequency under maximum speed. Modal calculation results reveal that the first sixth-order natural frequencies of the corn harvester frame are all lower than the engine excitation frequency, thus meeting the frame design requirements.

Frame optimization and result analysis

Using ANSYS optimization calculation module, an optimization model with minimal mass is established while strength and stiffness requirements are met. In comprehensive analysis with the maximum deformation under full-load bending condition as the state variable, the optimization goal is to minimize the frame mass, and the optimal design variable is the section geometry parameter of the frame. After 20 iterations, the optimization results are shown in Table 4.

Table 4

Comparison of main parameters before and after price optimization model

Optimization variables	Initial value	Optimization results	Change ratio/%
Volume of car frame U/mm	0.39231×10^8	0.33632×10^8	-14.27
Maximum deformation of bending condition M/mm	4.5162	5.828	22.51
External beam thickness	6	5.6074	-6.54
Thickness of other beam of frame T/mm	6	4.5459	-24.23

Seen from the optimization results, although the maximum deformation is increased from the original 4.5 mm to 5.8 mm with an increase ratio of 22.51%, it is still within a reasonable range, and the section thickness of each frame beam is greatly reduced. To verify whether the stiffness and strength of the optimized new frame meet the requirements, the material is assigned with new thickness in the optimization scheme and compared with the original frame strength, so that the main performance parameters can be compared before and after the frame model optimization, as shown in Table 5.

Table 5

Comparison of main parameters before and after price optimization model

Frame performance parameters	Before optimization	After optimization	Change amount	Change rate/%
Frame quality / kg	494.82	424.2	-70.62	-14.3
Bending the maximum equivalent stress / MPa	154.31	203.27	48.96	31.7
Maximum amount of bending deformation / mm	4.5162	5.9536	1.4374	31.8
Maximum torsional equivalent stress / MPa	246.73	278.14	31.41	12.7
Maximum torsional deformation / mm	5.1888	6.4365	1.2477	24.04

According to the above table, by optimizing the key frame parts of the corn harvester for hills and mountains, there is a great room for reducing the frame mass. After optimization, the transporter frame mass is reduced by 70.62 kg. Despite the certain increase in maximum deformation and maximum equivalent stress under torsion and bending conditions, the various performance parameters change little, and the results are ideally within a reasonable range. To conclude, this optimization scheme makes it possible to better distribute the frame mass of the corn harvester, which achieves the purpose of light weight to a certain extent and lowers the manufacturing cost.

Stability analysis of the corn combine harvester

The force analysis of a corn harvester moving uphill on a mountainous terrain with a slope φ is shown in Figure 6(a). When the harvester is tilted longitudinally around the point O_2, the moment balance at point O_2 is $\sum M_{O_2} = 0$, then:

$$F_{Z1} = \frac{A_2 Gcos\varphi - hGsin\varphi}{A} \tag{2}$$

If the corn harvester is to be prevented from tipping longitudinally around the point O_2 when going uphill, then $F_{Z1} \geq 0$, i.e.:

$$A_2 Gcos\varphi - hGsin\varphi \geq 0 \tag{3}$$

$$\varphi \leq \arctan(\frac{A_2}{h}) \tag{4}$$

The limiting tipping angle for longitudinal tipping of a corn harvester while driving uphill in hilly mountainous terrain is:

$$\varphi_{lim} = arctan(\frac{A_2}{h}) \tag{5}$$

Where:

F_{z1} is the support force on the front wheel;

A is the distance between shafts;

A_2 is the distance distance from the center of gravity of the machine to the slope of the rear axle;

G is the weight of harvester;

h is for the vertical height from the center of gravity to the slope;

φ is for the angle of uphill;

φ_{lim} is for the uphill tipping angle limit.

The value of φ_{lim} can be used to measure the ability of the corn harvester to resist longitudinal tipping when going uphill, and it is related to the distance from the center of gravity of the entire machine to the slope of the rear axle A_2 and vertical height h from the center of gravity to the slope.

The force analysis of a corn harvester moving uphill on a slope with a gradient φ' is shown in Figure 6(b). When the harvester tilts longitudinally around the point O_1, the moment balance at the point O_1 is $\sum M_{O_1} = 0$, then:

$$F'_{z2} = \frac{A_1 Gcos\varphi' - hGsin\varphi'}{A} \tag{6}$$

If the corn harvester is to be prevented from tipping longitudinally around the point O_1 when going downhill, then $F'_{z2} \geq 0$, i.e.:

$$A_1 Gcos\varphi' - hGsin\varphi \geq 0 \tag{7}$$

$$\varphi' \leq \arctan(\frac{A_1}{h}) \tag{8}$$

The limiting tipping angle for a corn harvester that tilts longitudinally while driving downhill in hilly mountainous terrain is:

$$\varphi'_{lim} = arctan(\frac{A_1}{h}) \tag{9}$$

Where:

F'_{z2} is the support force on the rear wheel;

A_1 stands the distance from the center of gravity of the entire machine to the front axle slope;

φ' stands for the angle of slope down;

φ'_{lim} stands for the downhill ultimate tipping angle.

The value of φ'_{lim} can be used to measure the resistance of the corn harvester to longitudinal tipping when going downhill, and it is related to the distance from the center of gravity of the entire machine to the front axle slope A_1 and the vertical height *h* from the center of gravity to the slope.

The force analysis of the entire machine when the corn harvester travels laterally on a mountainous terrain with a slope *γ* is shown in Figure 6(c). When the harvester is tilted laterally around the point O_1', the moment balance at point O_1' is $\sum M_{O_1'} = 0$, then:

$$F_{zo} = \frac{B_2 Gcos\gamma - hGsin\gamma}{B_1 + B_2} \tag{10}$$

To keep the corn harvester from tipping laterally around point O_1' when traveling laterally, $F_{zo} \geq 0$, i.e.:

$$B_2 Gcos\gamma - hGsin\gamma \geq 0 \tag{11}$$

$$\gamma \leq \arctan(\frac{B_1}{h}) \tag{12}$$

The limiting tipping angle for lateral tipping of a corn harvester while driving laterally in hilly mountainous terrain is:

$$\gamma_{lim} = arctan(\frac{B_1}{h}) \tag{13}$$

Where:

F_{zo} is the support force on the righr wheel;

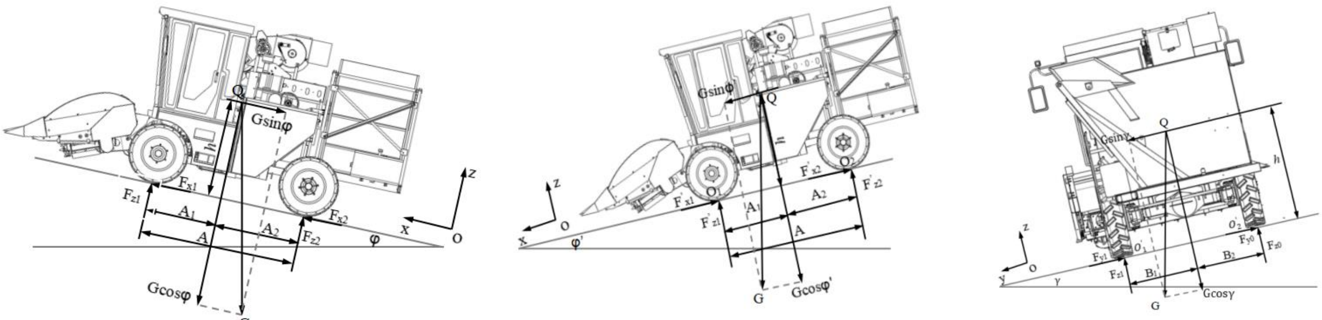
B_1 stands the distance from the center of gravity of machine to the slope of the left walking wheel;

B_2 stands the distance from the center of gravity of machine to the slope of the right walking wheel;

γ stands for the rollover angle;

γ_{lim} stands for the lateral ultimate tipping angle;.

The value of γ_{lim} can be used to measure the resistance of the corn harvester to lateral tipping when traveling laterally, and it is related to the distance from the center of gravity of the entire machine to the slope of the left walking wheel B_1 and vertical height *h* from the center of gravity to the slope.



(a)uphill driving (b) downhill driving (c)slope driving

Fig. 6 - The driving force diagram of harvester

The analysis of the anti-tip ability of the corn harvester in mountainous regions shows that the height of the center of gravity and width of the wheelbase of the harvester affect its anti-vertical tipping ability and anti-transverse tipping ability. The height of center of gravity decreases and the wheelbase width increases, and the anti-tip ability of the harvester increases. The 3D model of the entire machine was created in Solidworks software. Assuming that the mass of the operator is 80 kg and the harvester is in the process of driving with a full load in the grain bin, the vertical height h of the center of gravity of the entire harvester from the ground is 1080 mm, the distance from the front axle A_1 is 1110 mm, the distance from the rear axle A_2 is 1090 mm, and the distance from the left wheel B_1 is 913 mm by evaluating the mass property options in SolidWorks software. By substituting the abovementioned parameters into Eqs. (5), (9), and (13), the limit tilting angle of the mountain corn harvester was calculated to be 45.3°, 45.7°, and 40.2° for travelling uphill, downhill, and laterally, respectively.

CONCLUSIONS

This paper designs a corn combine harvester frame for hill and mountain operations, with contributions mainly reflected in three aspects.

(1) Based on the TRIZ innovation theory, analysis is made on the frame effect on the trafficability and stability of the whole machine, and a staggered frame structure is innovatively designed according to the physical contradiction solving principles. Static analysis and modal analysis of the frame is performed using ANSYS software to examine the stress and deformation under the two typical working conditions of full-load bending and full-load torsion. The maximum value appears in the middle of the front frame and the tail beam of the rear frame. As the first sixth-order natural frequencies of the frame are all below the engine excitation frequency, frame design requirements are met.

(2) An optimization analysis is performed with the maximum deformation under the full-load bending condition as the state variable, the minimum frame mass as the goal, and the geometric parameters of the beam section as the variables. The optimization reduces the frame mass by 14.3%, so that lightweight of the frame is possible. In view of complicated environment in hill and mountain operations, it is necessary to further study how to improve stability of the whole machine in the future researches, such as by optimizing the frame material and the frame welding mode.

(3) Based on the stability analysis of the corn combine harvester, it puts forwards the overturning angle of uphill is 45.3°, the overturning angle of downhill is 45.7°, and the overturning angle of slope is 40.2°.

ACKNOWLEDGEMENT

This research was funded by the Project supported by the National Natural Science Foundation of China (51405311) and the results of the work are obtained using the corn harvester for hilly area provided by Fushun Dayi Agricultural Machinery Equipment Co., Ltd.

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