Design of the Corn Field Straw Chopper Transmission System Based on Virtual Prototype Technology

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Abstract: The planting area of corn is large in our country, and the straw yield is high in our country. But the number of the straw returned to field is low at present, and the cost of returning straw to field is high. Thus, a set of corn straw returning machine whose productivity is high, cost is low and can be used together with self-propelled corn harvest machine is designed. This paper introduced the corn straw returning machine’s general parameters and drive system’s working process and design process. Then the virtual prototype model of the drive system is built, and virtual prototype technology is used to build the model of the drive system and to conduct kinematics simulation and dynamics simulation. The results show that the angular velocity and the stress of all sample points are all different in different working conditions. The response of the sample point is obvious when the pound load is big, and the response decayed to zero after three cycles of concussion. The relationships between angular velocity and stress of each sample point prove that the virtual prototype model’s validity. This study provides a new method and a way for the design and optimization of corn straw returning machine.

1. Introduction
The planting area of corn is large in our country, which number is 36.3184 million hm² in 2013, and the straw yield is high in our country. The corn straw is high enough to achieve 27.39% of total straw [1], so how to handle corn stalks becomes a major topic in the field of agricultural production and environmental protection. Straw is one of the most important organic fertilizers in agricultural production which contains large amounts of organic matter, nitrogen, phosphorus, potassium and trace elements[2,3]. Straw returned to field can increase soil organic matter, soil fertility, and reduce waste of resources and environmental pollution brought by straw piled up and burning. It can make agricultural ecological environment friendly, improve farmland productivity [4-6].

Technology of straw returned to field in China began in 1990s. After years of improvement and optimization, many machines related to straw returned to field had appeared[7]. But the design and analysis methods of the existing corn straw returning machine are relatively backward. It has many problems, such as high failure rate of working part, large noise and vibration, low energy efficiency[8]. There is much research on straw returning machine[9-12], but the transmission system less. Lu Xinchun, Zhang Yongqin and others designed and analyzed the transmission system of Straw Returning and Fertilization Seeder[13]. They designed a compact Straw Returning and Fertilization Seeder whose transmission is smooth. Xu Lichao, Cai Yang and others created a virtual prototype modeling of reverse straw returning machine transmission and analyzed it[14], verified the accuracy of virtual prototype technology. These two machines are always used together with tractors.
This paper showed a transmission system design method of 1JHY-200C Corn Field Straw Chopper, created a transmission system virtual prototype modeling of corn field straw chopper and analyzed it based on ADAMS/Machinery of ADAMS. It provided development & optimization methods and technical guidance for the transmission system of counters-field machine.

2. Transmission System Design of Corn Field Straw Chopper

2.1 Transmission plan design

Figure 1 is the structure of 1JHY-200C corn Field Straw Chopper. It includes frame, suspension system, transmission system, cutting device and ground wheel. 1JHY-200C Corn Field Straw Chopper must work together with self-propelled corn harvester, lifted by hydraulic equipment, hanged under the chassis through three-point.

![Fig.1 The structure of corn field straw chopper](image)

| Parameters’ name                  | Design value |
|-----------------------------------|--------------|
| Rated power/ kW                   | 28-38        |
| Working width/ mm                 | 2000         |
| Stubble height/mm                 | 80           |
| Rotation speed of knife shaft/ r/min | 1850         |
| Dimensions/mm                     | 2200×1324×890|
| Type of blade                     | Y-type       |
| Number of blade                   | 48           |
| Weight/kg                         | 250-350      |

Y-blades knife relative to hammer claw grinding knife has a smaller size, lighter weight, better adaptability, higher crushing efficiency, et al. When operating conditions such as speed, spindle speed is constant, there is an optimum value of the shredding blades number. Blade is too few, it won’t reach the request of cut quality, on the contrary, will cost more, and block easily. The density of blade
arrangement is 0.23-0.4/cm$^{15}$, so the knife-shaft was designed and installed 48 Y-blades. Theoretical studies and tests had shown that, linear velocity of spinning knife top must be greater than 30m/s when cut without support. Knives shaft rotational speed is determined by top velocity and stem-cutting length, and current blade shaft speed is 1800-2000r/min$^{16}$. Power consumption of mincing knife can be gained according to the calculated$^{10,16}$.

Figure 2 is transmission program of 1JHY-200C Corn Field Straw Chopper. Transmission system includes pulleys, side gears set and flail shaft.

The axis of drive wheel is coaxially connected with the power output axis of clutch device. Power of engine is passed to drive wheel by clutch device, then passed to gear1 of side box that coaxially connected with driven pulley. Next, the power is passed to knife shaft by two groups gear pair in side box to drive knife shaft rotate, achieve broken and return of corn straw. Through the interaction between pulleys and gear set, it achieves the purpose speed-down and torque-up, and provides suitable speed and torque for cutter shaft.

Figure 3 is the lift mechanism transmission program of 1JHY-200C Corn Field Straw Chopper. It includes hydraulic cylinders, rocker and linkage.

Part 11 and 12 showed in figure.3 is piston rod and cylinder, which formed a hydraulic cylinder, so hydraulic energy can be transformed into mechanical energy, to provide power, drive rocker rotating. Linkage is located at the leg of the rocker. When the rocker rotates, it can drive linkage upward. Linkage is hinged with frame, which can make the frame rotary around the other hinge point, to achieve the purpose of raising corn straw returning machine.

2.2 Transmission design

The clutch output shaft of the straw returning machine is at speed of 2400r/min when normal operate, and the total transmission ratio of drive train is 1.297. By belt drive design theory, the greater angle the small pulley has, the less prone to slipping in the same conditions and the more powerful it can deliver.
So belt transmission ratio is 1, gear transmission ratio is 1.297.

The belt transmission has many advantages like that simple structure, stable transmission, low price, absorbing shock, suitable for long-distance transmission, and opened belt transmission efficiency is high enough 0.97. The transmission distance of output power from the clutch to the corn straw returning machine is far and in order to costing less, I chose belt drive.

According to the power consumption of straw returning machine, figure out calculation-power, as follows:

\[ P_{ca} = K_A P \]  
(1)

In this formula: \( P_{ca} \) is calculation-power, kW; \( K_A \) is coefficient of performance; \( P \) is the rated power required to pass, kW. Select C-type V-belt based on the calculation-power \( P_{ca} \) and the driven pulley speed.

To enable the force uniform, the number of belts is generally smaller than 10. The number \( z \) is:

\[ z = \frac{P_{ca}}{P_r} = \frac{K_A P}{(P_0 + \Delta P_0) K_a K_l} \leq 10 \]  
(2)

In this formula: \( z \) is the number of belts; \( P_r \) is rated power of single V-belt, kW; \( P_0 \) is the basic rated power of single V-belt, kW; \( \Delta P_0 \) is the single V-belt increment of rated power, kW; \( K_a \) is correction factors when angle is not equal to 180°; \( K_l \) for the belt length is not equal to pilot belt length correction factor.

According to formula (1) ~ (3), the required number of V-belt is 4.

When calculate the initial tension, we need to take the effects of centrifugal force and angle into account. So, minimum initial tension is:

\[ F_{0\text{min}} = 500\left( \frac{2.5 - K_a}{K_a^{2v}} \right) \frac{P_{ca}}{K_a^{2v}} + qv^2 \]  
(3)

In the formula: \( F_{0\text{min}} \) is the minimum initial tension, N; \( q \) is mass per unit length, kg/m; \( v \) is the tangential velocity, m/s.

According to equation (4) we can achieve that belt drives initial minimum force of 1JHY-200C Corn Field Straw Chopper is 477.71N, for new installations of V-belts, force should be 1.5\( F_{0\text{min}} \); for operation of V-belts, force should be 1.3\( F_{0\text{min}} \).

Tangential velocity of belt should be 5~25m/s. According to belt type, belt speed and V-belt pulleys diameter series we select the base diameter of the pulley which is 200mm.

According to the design requirements, the gear1, driven pulley and the three-point suspension shaft must be in the same axis. Gear3 and flail shaft must be in the same axis too, and gear2 is the idler. Center distance sum of the two gear sets should be equal to the distance from the driven pulley axis to flail shaft axis, so the gear pitch circle diameter is relatively large, the number of teeth is big. In order to make the gear axial load balance, increase gears’ life, gear 1, 2, 3 are used bearing symmetrical brace. Width of gear3 is 6mm bigger compared to gear 1 and 2, in case of increasing work load leded by tooth axial meshing decrease because of assembly errors. Chose the 40Cr steel as the material of gear 2. And chose 45 steel as the material of gear 1 and gear 3. So that it can improve gear tooth surface fatigue limit of gear 3 and gear 1, and make contact fatigue stress increase 20%. Considering there is a sliding rate in belt drives, gear transmission ratio should be slightly smaller than 1.297, so set the gear ratio as 1.28.

The main work of gear design is contact fatigue strength design and tooth root bending fatigue strength design. Fatigue design of tooth contact is:

\[ d_t \geq 2.32 \frac{K_T}{\phi_d} \times \left( \frac{u}{u^2 \times (Z_E \times \sigma_t)} \right)^{0.5} \]  
(4)

In the formula: \( d_t \) is gear pitch circle diameter, mm; \( K \) is the load factor; \( T_1 \) is the torque on gear, N•mm; \( \phi_d \) is width factor; \( u \) is tooth profile radius of curvature ratio; \( Z_E \) is elasticity coefficient, MPa \( 0.5 \); \( [\sigma_t] \) is contact fatigue allowable stress, MPa.

Bending fatigue strength design equation:
\[ m \geq \frac{2KT_{T1}(\frac{Y_F}{Y_{Sa}})}{\phi_1 z_1^2 \left[ \frac{\sigma_f}{\sigma_f} \right]} \]  

In the formula: \( m \) is gear module, \( \text{mm} \); tooth number \( Z_1 \) is for gear 1; \( Y_F \) is tooth form factor; \( Y_{Sa} \) is stress correction coefficient; \( [\sigma_f] \) is bending fatigue stress, \( \text{MPa} \). Gear module depends primarily on the size of the bearing capacity of bending strength determined by the tooth surface contact fatigue strength of the decision bearing capacity only link to gear diameters.

The number in table 2 is 1JHY-200C Corn Field Straw Chopper agencies main gear parameters.

| Table. 2 Main parameters of the gears |
|--------------------------------------|
| parameter                           | gear1 | gear2 | gear3 |
| modulus/mm                          | 4     | 4     | 4     |
| Number of teeth                     | 50    | 61    | 64    |
| Width of teeth/mm                   | 40    | 46    | 40    |
| Pressure angle/°                     | 20    | 20    | 20    |
| Modification coefficient            | 0     | 0     | 0     |

3. Virtual Prototype Development

This paper built a virtual prototype of 1JHY-200C Corn Field Straw Chopper based on ADAMS & Solidworks.

In the process of dynamic analysis, the deformation of rack effects is negligible, so the rack can be used as a rigid body, therefore we need to ensure the characteristics such as gear, hob shaft, the point of lifting and turning the relative position size when establish visual modeling of rack, hob axis and lifting mechanism based on Solidworks. Flail and flail shafts wear connected with a hinge in the physical prototype. Because of the effects of centrifugal force, flail radial distributed along the radial of the cutter axis under the condition of high speed. At the same time, in order to reduce the amount of computation, reduce simulation time, flail shaft can be related with the cutter shaft into a part model. Finally, assemble the parts, and save it in Parasolid(*x_t) format, then import it to ADAMS/View through IMPORT.

Figure 4 is 1JHY-200C Corn Field Straw Chopper transmission system which had been imposed constraints. Establish belt drive models and gear models in the right rack position with the use of ADAMS/Machinery. "Fixed Joints" can be used as spline exist in physical prototype. So, the driven pulley and shaft "axisA", gear 1 and shaft "axisA", gear 2 and shaft "axisB", gear 3 and flail shafts are all can be restrained by "Fixed Joints". "Revolution Joint" can be used as bearings in the mechanical connection. So the driving pulley and the earth, the shaft "axisA" and the frame, shaft "axisB" and the frame, flail shafts and racks and rack wear all connected with "Revolution Joint". Constrain the lifting mechanism with the same method.

Fig.4 System constraints for the transmission of working device
Figure 5 is a transmission system constraint of lift mechanism. Frame and connecting rod, connecting rod and rocker arm, rocker arm and piston rod, cylinder and the earth are all restrained with "Revolution Joint". Piston rod and cylinder is restrained with "Prismatic Pair".

Take the method of transforming rigid body to flexible body to transform rigid flail shafts into flexible components, then analysis kinematics and dynamic.

Figure 6 is 1JHY-200C Corn Field Straw Chopper drive system virtual prototyping.

4. Transmission Analysis

The documents give the definition, kinematic and dynamic equations, solving algorithms about the equations of the collision force in ADAMS. ADAMS/Postprocessor module is used to handle the simulation result data, display animation, statistics, analysis, comparison of the data results. It can run in ADAMS/View, and also run independently.

| working conditions | marker1 | marker2 | marker3 | marker4 | marker5 |
|--------------------|---------|---------|---------|---------|---------|
| 1                  | ●       | ○       | ○       | ○       | ○       |
| 2                  | ○       | ●       | ○       | ○       | ○       |
| 3                  | ○       | ○       | ●       | ○       | ○       |
| 4                  | ○       | ○       | ○       | ●       | ○       |
| 5                  | ○       | ○       | ○       | ○       | ●       |
| 6                  | ●       | ○       | ○       | ○       | ●       |
| 7                  | ○       | ●       | ○       | ●       | ○       |
| 8                  | ●       | ○       | ●       | ○       | ●       |

Annotation: ● replace loading, ○ replace no loading

The simulation process did not take the influence of environmental factors and lubrication into
consideration. To avoid mutation of the speed and force because of the beginning impact, speed drive is added by step function. Steady speed is working speed. Speed drive function is step (time, 0, 0, 0.2, 40*360d), which is imposed on the driving pulley. According to the equivalence principle we can calculate that the load of the section of a single flail is about 66N•m. Under the condition of stable operating, the load is striking, so we could use impact function to load. In flail shafts, took five uniform points to load, and named the load points as: Marker1, Marker2, Marker3, Marker4 and Marker5. Select eight conditions to analysis. Different loads under different operating modes are shown in table 3. Be careful that when step size is large it can’t reflect the high frequency response of the prototype. On the contrary, it will greatly increase simulation time, and the output files are large too. So the time is set 0.3s, steps is 600, then simulate. Simulation analysis sample points are shown in figure 6. They are the driving pulley centre point (O), the driven pulley center point (P), gear 3 center point (Q), the related Center shaft input end (M) and flail shaft output center point (N).

4.1 Kinesiology analysis

Figure 7 shows the angular velocity curve and angular acceleration curve of simulation point O under the condition 1.

![Angular velocity & angular acceleration of point O](image)

There was a same trend between angular velocity curve and speed drive. Angular velocity changed same as step function. Angular velocity rose up in 0-0.2s, and stabilized at 2400r/min during 0.2-0.3s. Angular acceleration curve showed that the acceleration is max at 0.1s, 300r/s², 0.2s later, angular accelerated is 0r/s², which is consistent with the angular velocity curve of point O.

Figure 8 shows the angular velocity curve of P and point Q from the simulation under condition 1. Combining the angular velocity curves of point O in figure 7 turned that angular velocity of different sample points are all positive, which indicated that sampling points have the same angular velocity, which is consistent with the theoretical analysis. Stable operation phase, the angular velocity of the point O is 2400r/min, the angular velocity of the point P is 2397.5r/min, the angular velocity of the point Q is 1873r/min. Transmission ratio of belt drive system is 0.999 in this simulation. Because of the load is zero, belt transmission slip rate is almost zero. And design transmission ratio was roughly equal to simulation transmission ratio, which was compatible with the theory. Gear transmission system simulation ratio was 1.28, which was equal to design ratio. It verified the accuracy of the virtual prototype.
Figure 9 is the angular velocity curves of point P under the different conditions. By the curve we can see: peak angular velocity of point P under working condition 1 to 5 is 2398.7 r/min, valley angular
velocity is 2395.6r/min; peak angular velocity of point P under working conditions 6 and 7 is 2399.8r/min, valley angular velocity is 2393.9r/min. Peak angular velocity under working condition 8 is 2401.5r/min, valley angular velocity is 2391.9r/min. Macro-analysis, angular velocity amplitude of point P was affected badly by impact loads, and regardless of its position.

From Figure 8 and Figure 9, we could know that instantaneous slip ratio of pulley under the top 5 conditions were all 0.208%, instantaneous sliding rate of pulley under the 6th and 7th condition were 0.254%, instantaneous sliding rate of pulley under the 8th condition was 0.335% through analyzing the speed of point O and point P under different conditions and comparing the speed of point O and point P under different impact loads. The greater the impact load is the more instantaneous slip ratio of the belt is, the lower the belt transmission efficiency is.

4.2 Dynamic analysis
Dynamic analysis existed in two conditions: analysis impact load of transmission system under conditions 1 and analysis loads of transmission system under stable load. Under stable load condition, the loading was loaded by 66000*step (time,0,0,0.2,1), which is loaded in the form of a step function. The load increases during 0s to 0.2s. The load reached a maximum value (66N•m) and then kept constant.

Figure 11 are the tangential meshing force curves of the gears under condition 8. During 0-0.2s, meshing force between two gears changes as parabola, and the maximum of meshing force is located at 0.1s, which corresponds to the angular acceleration of the point O in Figure 7, and also consistent with Newton's second law. During 0.2-0.3s, the system operates at a constant speed. Meshing force appeared between gears at 0.25s because of the impact load. We know that the max tangential contact force between gear 1 and gear 2 is 639N, and the max tangential contact force between gear 2 and gear 3 is 595N through analyzing. Meshing force produced by impact load finally reduced to 0 after several cycles.
Figure 13 shows the torque curve that gear 1 and gears 3 delivered. Average torque that gear 1 delivered is $51.372 \text{N} \cdot \text{m}$, Average torque that gear 3 delivered is $65.672 \text{N} \cdot \text{m}$. 
Figure 14 shows the driving and driven pulley torque curve. The average torque of driving pulley transmits is 53.875N•m, and the average torque of driven pulley transmits is 51.372N•m. From the above data, we achieve that the closer the moving parts to the power source, the bigger torque value it passes.

We can calculate the power of different parts transmit under steady load condition based on the relations between torque and speed. The transmission power of driving pulley is 13359W. The transmission power of driven pulley is 13036W. The transmission power of gear 1 is 13019W. The transmission power of gear 3 is 12881W. The total efficiency of belt transmission is 0.975. The total efficiency of gear transmission is 0.989. The total efficiency of the transmission system is 0.964.

5. Conclusion
1) This essay designed an efficient transmission system for the 1JHY-200C Corn Field Straw Chopper. It provided a set of parameters matching method for corn harvesting machine drive system.

2) It reduced the development time of a corn straw returning machine virtual prototype through using the ADAMS/Machinery module. It analyzed the movement and the stress of the transmission system under condition 1. We achieved that angular velocity of point O, P and Q is respectively
2400r/min, 2397.5r/min and 1873r/min, which verified the validity of virtual prototype model. The transmission system has been designed to meet the requirements.

3) This essay analyzed the dynamics response of the flail shaft and the transmission system, under the condition of impact load and unbalance load. The response time of the fail shaft input end is slightly earlier than the free end under condition 5. On the contrary, the oscillation amplitude of the fail shaft free end is larger, which is consistent with fact.

4) This essay analyzed the movement and the stress of the transmission system under the step function load condition. Achieved the transmission efficiency of belt transmission and gear transmission, which is 0.975 and 0.989, and meet the design requirements.

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