

Study on the dynamic characteristics of planetary gear transmission mechanism of metal cold rolling mill

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Abstract. Using the parametric modelling feature in Solidworks software, a three-dimensional solid model of a planetary gearbox with linear reciprocating motion of the output shaft was constructed. By conducting theoretical calculations related to the kinematics of the transmission system and combining Adams virtual prototyping simulation technology, a dynamic simulation model of the planetary gear transmission mechanism was established to analyse the motion laws and dynamic characteristics. The simulation results were compared and analysed against the theoretical calculation results, and they showed good agreement and consistency. The study also investigated the impact of counterweights on the inertia forces and moments of the output shaft in the transmission mechanism. The results indicated that adding counterweights effectively reduced the inertia impact caused by inertia forces at the start and end positions of the stroke, as well as the inertia torque caused by changes in angular acceleration during the startup and stopping phases, thereby enhancing the smooth operation of the mechanism. Additionally, different thicknesses of counterweights had varying effects on balancing the inertia forces and moments of the output shaft. The study aimed to find the optimal thickness of counterweights to achieve the best balance effect. Furthermore, the study examined the influence of different speeds of the driving gear on the inertia forces of the output shaft. The results showed that at a driving gear speed of 1500 rpm, the horizontal thrust generated by the output shaft was 63566 N, which could provide the required thrust for the rolling mill operation, and the inertia forces of the output shaft tended to stabilize.

Keywords: Planetary gear transmission mechanism / 3D model construction / dynamics
virtual prototyping technology / counterweight

1 Introduction

The planetary gear transmission mechanism has a novel and unique structure, superior performance, and wide application prospects in various fields such as wind power generation, agricultural machinery equipment, and aerospace precision equipment [1]. Stainless steel seamless pipes, as an indispensable material in modern society, are produced using various forming methods, including cold rolling, hot rolling, cold drawing, and extrusion. With the development of society, the application scenarios of seamless pipes are continuously expanding, and the performance requirements for them have become more stringent [2]. After continuous exploration and improvement by researchers in the manufacturing process and equipment of seamless pipes, it has been found that seamless steel pipes formed through cold rolling process can better meet the required performance standards [3].

The Pilger cold rolling mill is an important equipment for the cold rolling process, and its transmission system is a key component that ensures the high-speed and smooth operation of the rolling mill [4,5]. Traditional Pilger cold rolling mills mostly use a crankshaft transmission system as their driving mechanism, as shown in Figure 1 [6]. Due to the force exerted on the connecting rod during the operation of driving the Pilger cold rolling mill, there is a component of force in the vertical direction, causing the Pilger cold rolling mill to undergo irregular small oscillating movements in the vertical direction. The displacement of the Pilger cold rolling mill along the vertical direction is shown in Figure 2. The unstable operation of the mechanism exacerbates system wear and increases noise. Therefore, the development of a transmission system capable of driving the Pilger cold rolling mill to operate at high speed and stability has become a research hotspot for scholars both domestically and internationally.

To address the above issues, this paper presents the design of a planetary gearbox with an output shaft capable of reciprocating linear motion. The structure of the gearbox

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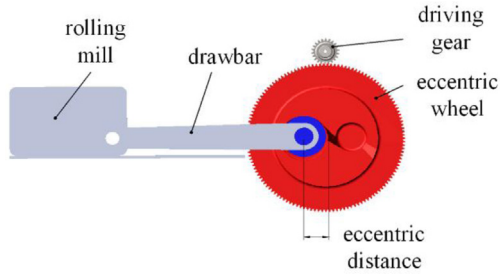


Fig. 1. Using crankshaft transmission for Pilger cold rolling mills.

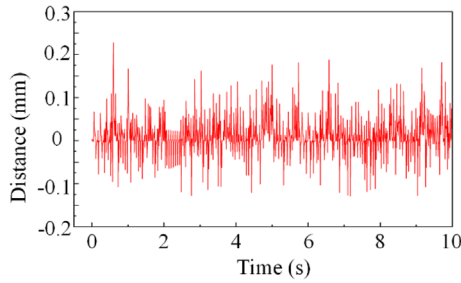


Fig. 2. Vertical displacement of the Pilger cold rolling mill.

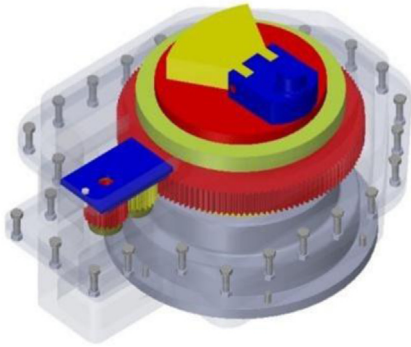


Fig. 3. Linear reciprocating planetary gearbox.

is shown in [Figure 3](#). Since there is no component of force in the vertical direction, there is no oscillating displacement in the vertical direction, which enhances the stability of the mechanism during operation. Taking this as the research objective, theoretical calculations and analysis of system kinematics were conducted. Additionally, the dynamic characteristics were studied using Adams virtual prototyping simulation technology. The influence of counterweight blocks on the inertia force of the system output shaft was also investigated, providing guidance for system dynamic balancing optimization [7–10]. In recent years, scholars both domestically and internationally have devoted significant research efforts to the dynamic characteristics of planetary gear mechanisms [11–14]. Zhang et al. [15] constructed a physical model of a two-stage gearboxes with rotational characteristics and conducted dynamic simulation analysis using Adams software, obtaining regular characteristics such as gear meshing forces. Tao [16]

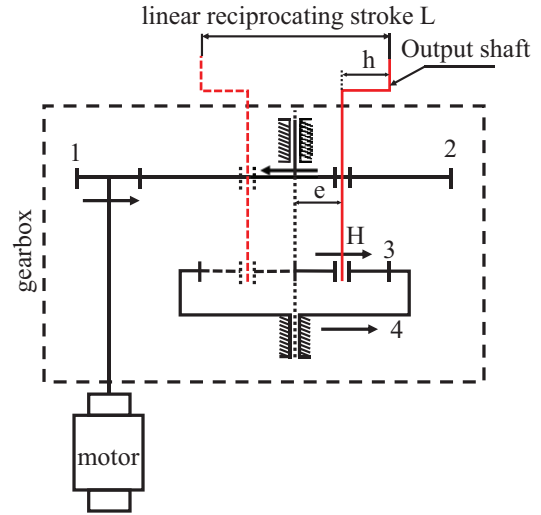


Fig. 4. Diagram of the structural principle of a planetary gearbox.

conducted in-depth research on the dynamic characteristics of offshore wind turbine gearboxes. By constructing a faulty gear model and studying its dynamic characteristics, the mechanism behind typical gear failures was revealed. Furthermore, optimization design was performed on offshore wind turbine gearboxes.

2 Parameterized modelling of planetary gearbox

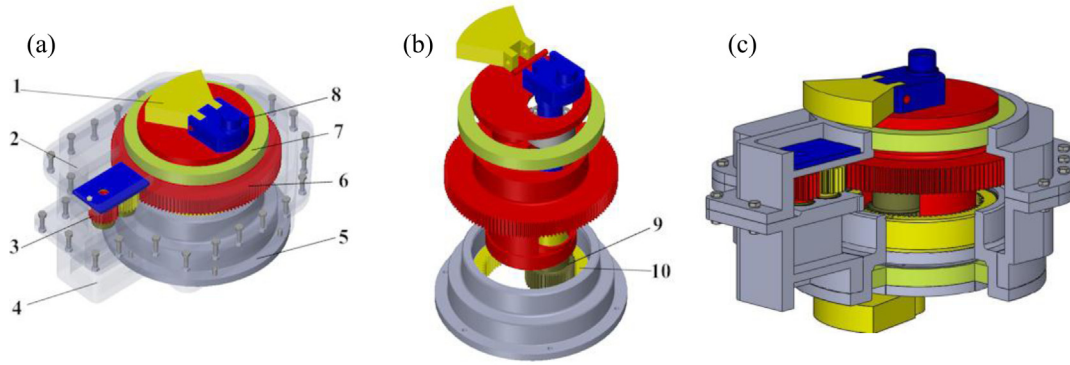
Based on the principles of mechanical design, the working principle of the reciprocating linear motion of the output shaft in a planetary gearbox was studied, and the structure of the gearbox was designed. [Figure 4](#) shows the schematic diagram of the working principle of the planetary gearbox, visually illustrating the meshing conditions of the gears and the reciprocating linear motion of the output shaft.

In Solidworks parametric modelling software, the internal components of the planetary gear mechanism were modelled. By adding constraint relationships and contact relationships to the components, the assembly of the gearbox structure was performed. [Figure 5](#) shows the schematic diagram and exploded view of the structure of the planetary gearbox. [Table 1](#) presents the relevant important parameters of the internal gears in the planetary gearbox.

3 Virtual prototyping of the planetary gearbox

3.1 Material property settings

Taking the designed planetary gearbox as the research object, its model was imported into Adams simulation software to establish a virtual prototype of the planetary gearbox. The dynamic characteristics of the gearbox were studied, as shown in [Figure 6](#). Prior to the study, material properties were first assigned to each component. In this



1- clump weight; 2- upper case; 3- driving gear; 4- lower case; 5- base; 6- large gear; 7- bearing; 8- output shaft; 9- planetary gear; 10- ring gear

Fig. 5. Schematic diagram of planetary gearbox. (a) structural schematic diagram; (b) exploded view; (c) sectional view.

Table 1. Planetary gearbox parameters.

Gear	Modulus	Pressure angle (°)	Number of teeth	Dividing circle diameter (mm)
Driving gear 1	8	20	22	176
Large gear 2			133	1064
Planetary gear 3			37	296
Ring gear 4			74	592

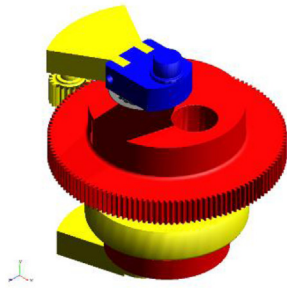


Fig. 6. Virtual prototype of planetary gearbox.

paper, the geometric shape and density were chosen to define the mass of each component, and the parameters such as mass and rotational inertia of the components were automatically calculated.

3.2 Constraint and load addition

To ensure the proper operation of the planetary gearbox in simulation software, appropriate constraints such as motion pairs and contact pairs are added between the components of the mechanism. The motion pairs added to the planetary gear reducer include: (1) Setting a rotational pair between the driving gear and the ground, allowing the driving gear to rotate relative to the ground. (2) Setting a rotational pair between the large gear and the ground, enabling the large gear to rotate relative to the ground. (3) Setting a rotational pair between the output shaft and the bearing, allowing the output shaft to rotate relative to the

bearing. (4) Setting a rotational pair between the eccentric hole of the large gear and the bearing, enabling the eccentric hole of the large gear to rotate relative to the bearing. (5) Setting a rotational pair between the planet gear and the output shaft, allowing the planet gear to rotate relative to the output shaft. (6) Fixing the ring gear to the ground. (7) Setting a rotational pair between the planet gear and the eccentric hole of the large gear, enabling the planet gear to rotate relative to the eccentric hole of the large gear. (8) Setting a rotational pair between the large gear and the ring gear, allowing the large gear to rotate relative to the ring gear. (9) Fixing the counterweight block 1 to the upper end of the output shaft. (10) Fixing the counterweight block 2 to the lower end of the output shaft.

3.3 Setting of operating conditions

To reduce the vibration impact generated during the instantaneous start of the planetary gearbox, a STEP function is used to drive the driving gear, allowing the speed of the driving gear to increase from 0 to 1500 r/min within 0.5 s. The total simulation duration is set to 2.5 s with a calculation step size of 0.005 s. The angular velocity variations of each component obtained from the simulation analysis are shown in Figure 7.

By observing (a)–(c) in Figure 7, we can draw the conclusion: Within the range of 0–0.5 s, as the speed of the input gear gradually increases from 0, the speeds of the large gear and the planet gear also increase accordingly. After 0.5 s, the speed of the input gear stabilizes at 157 rad/s. Due to the presence of vibration and impact during the

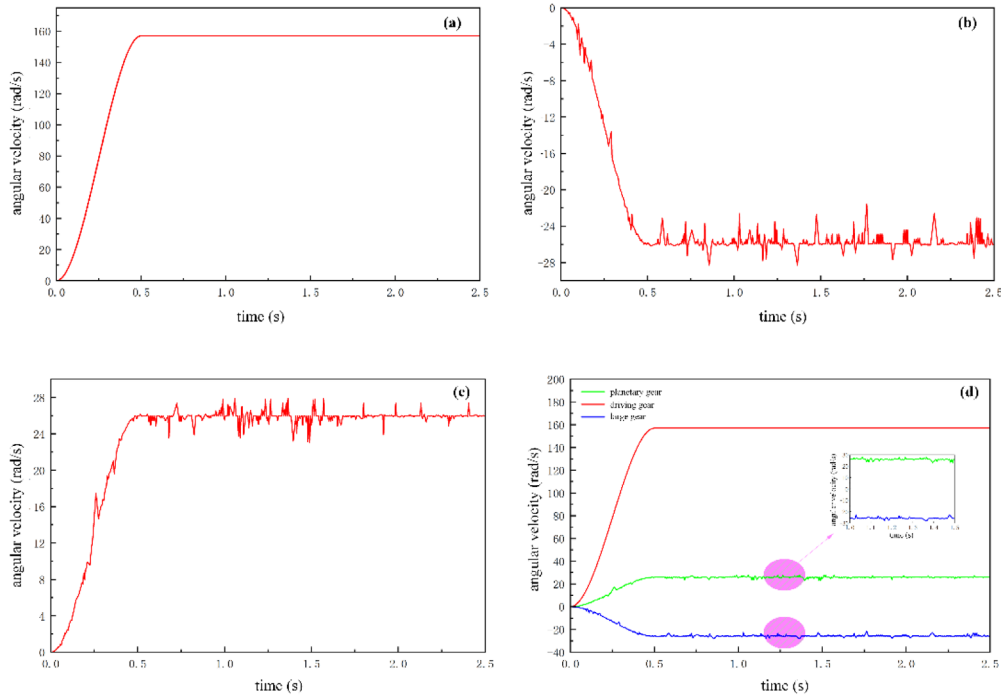


Fig. 7. Angular velocity change curve (a) driving gear; (b) large gear; (c) planetary gear; (d) overall view.

operation of the mechanism, there are fluctuations in the speeds of the large gear and the planet gear, but they both stabilize at around 26 rad/s, with relative errors of 0.12% and 0.5% respectively, compared to the theoretical value of 25.97 rad/s.

Figure 7d illustrates the variation of component speeds over time in the planetary gearbox. It can be observed that the angular velocity variation pattern of the large gear is essentially similar to that of the input gear. The meshing arrangement between the two gears is external, resulting in opposite rotation speeds. Due to the high gear ratio, it results in a significant difference in rotational speeds; The eccentricity of the planetary carrier is set on one side of the large gear, and its revolution speed variation trend is consistent with that of the large gear; The planet gear and the driving gear are connected through a spline connection, with both rotating in the same direction and having equal angular velocities.

4 Study on meshing force between gears

Inside the planetary gearbox, there are primarily two pairs of meshing gear sets. During the smooth operation of the mechanism, motion and power are transmitted through the meshing of gears. Both pairs of gear sets exhibit regular variations in meshing force. Figures 8 and 9 respectively show the meshing force curves between the two pairs of gear sets.

From Figure 8, it can be observed that within 0–0.5 s, as the speed of the driving gear gradually increases, the meshing force between it and the large gear exhibits significant fluctuations, showing irregular variations. Once the speed of the driving gear tends to stabilize, the meshing

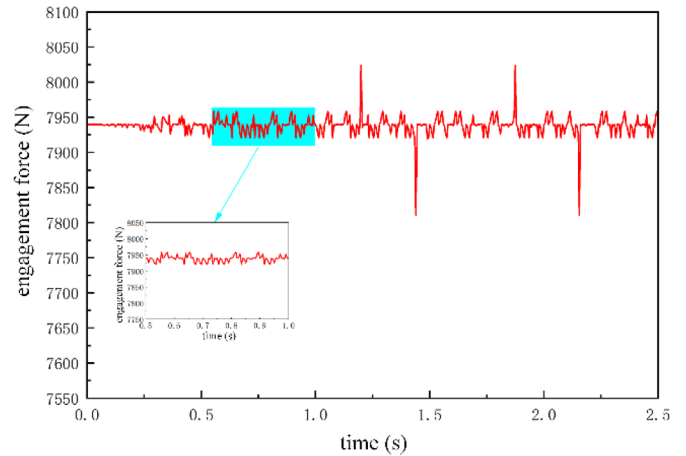


Fig. 8. The meshing force between the driving gear and the large gear.

force demonstrates periodic changes. Although it fluctuates up and down, it remains relatively stable at around 7939.63 N. In the design process of the planetary gear transmission, the calculated meshing force between the driving gear and the large gear is 7957.5 N. Compared with the theoretical calculation, the relative error of the simulation result is 0.2%, confirming the accuracy of the simulation result.

From Figure 9, within 0–0.5 s, the gearbox is in the startup phase, and the meshing force between the planet gear and the ring gear exhibits significant fluctuations, showing irregular variations. Once it reaches 0.5 s, the mechanism operates in a stable manner, and the meshing

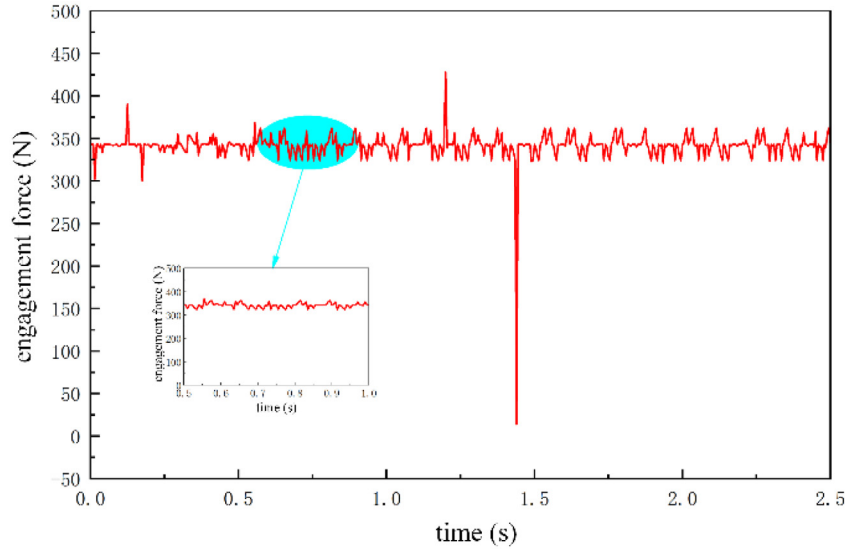


Fig. 9. The meshing force between the planet gear and the ring gear.

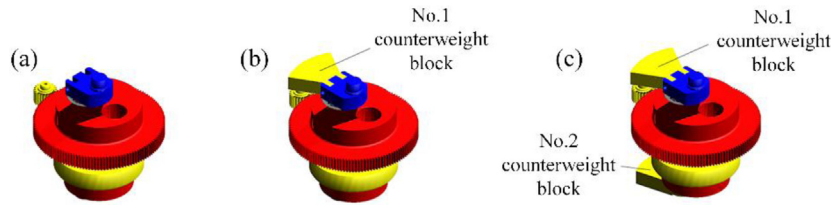


Fig. 10. Simulation model of the planetary gearbox (a) without counterweight; (b) adding counterweight block 1; (c) adding counterweight blocks 1 and 2.

force between the planet gear and the ring gear follows a periodic pattern. It fluctuates within a certain range, and the meshing force remains relatively stable at around 343.4 N. In the design process of the planetary gear transmission, the calculated meshing force between the planet gear and the ring gear is 343.24 N. Compared with the theoretical calculation, the relative error of the simulation result is 0.05%, validating the accuracy of the simulation result. Additionally, at 1.45 s, the force value becomes 0, which may be a numerical error generated during the simulation calculation.

5 Impact of counterweight blocks on inertial force of the system's output shaft

5.1 Impact of the presence or absence of counterweight blocks on the inertial force of the system's output shaft

The output end of the planetary gearbox exhibits reciprocating linear motion, which generates inertial forces due to the periodic variation of its centre of mass acceleration. In this design, counterweight blocks are installed at the output shaft end in a position opposite to

the direction of motion of the output shaft to reduce the generated inertial forces, thereby buffering the inertial impact generated at the extreme positions of the output end. From this perspective, it is necessary to study the impact of the presence or absence of counterweight blocks on the inertial forces at the output end. Therefore, in this section, we will conduct an in-depth study on the impact of inertial forces on the output shaft under three conditions: without counterweight blocks, adding counterweight block 1, and simultaneously adding counterweight blocks 1 and 2. Figure 10 shows the schematic diagram of the simulation model of the planetary gearbox under the three conditions.

Figure 11 respectively show the inertial forces along the direction of motion of the output shaft and the corresponding RMS (root mean square) values of the inertial forces under the three conditions. It can be seen that without counterweight blocks, the variation trend of the inertial force on the output shaft is quite noticeable, with an RMS value of 35607.23 N, which is relatively large; When adding counterweight block 1, there is a certain mitigation effect on the inertial force along the direction of motion compared to the case without any counterweight block. The maximum value of the inertial force also decreases. The RMS value of the inertial force reduces to

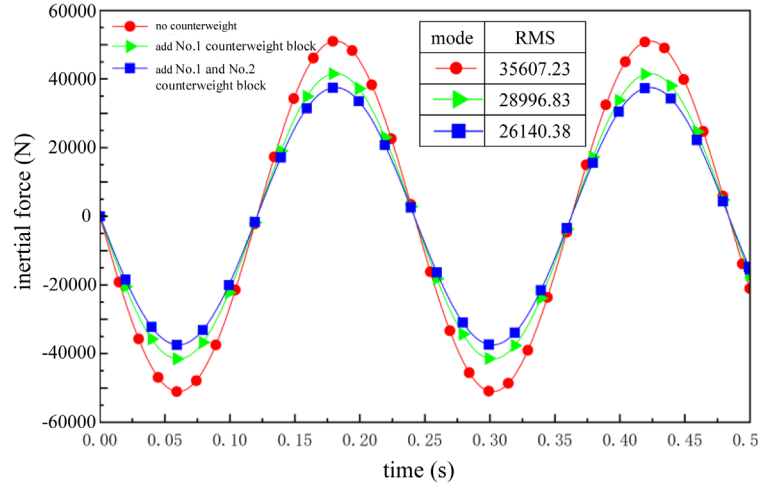


Fig. 11. Inertial force along the direction of motion of the output shaft.

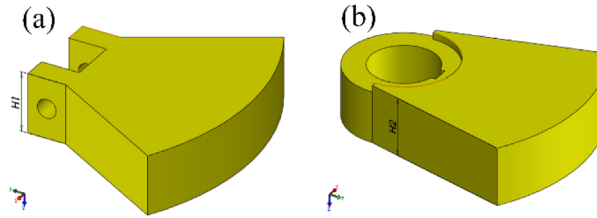


Fig. 12. Schematic diagram of counterweight thickness (a) Counterweight block 1; (b) Counterweight block 2.

Table 2. Counterweight block thickness table.

	1	2	3	4	5
H1 (mm)	85	105	125	145	165
H2 (mm)	90	110	130	150	170

28996.83 N, which is a decrease of 18.56% compared to the case without any counterweight blocks; When further adding counterweight block 2, the mitigation effect on the inertial force becomes more pronounced. The maximum value of the inertial force decreases even more. The RMS value of the inertial force reduces to 26140.38 N, which is a decrease of 26.59% compared to the case without any counterweight block. This indicates that adding counterweight blocks can have a significant effect in balancing the inertial impact on the output shaft.

5.2 Impact of different thickness counterweight blocks on the inertial force of the system's output shaft

Based on the findings from the previous section, it was discovered that by adding counterweight blocks, it is possible to effectively balance the inertial force generated on the output shaft of the gearbox due to acceleration variations, thus improving the stability of the mechanism

during operation. However, different thicknesses of counterweight blocks correspond to different masses, resulting in varying balancing effects on the inertial force of the gearbox's output shaft. Therefore, in this section, we will investigate the impact of several sets of different thicknesses of counterweight blocks on the inertial force of the gearbox's output shaft. By varying the thickness of two counterweight blocks, we aim to find the optimal thickness configuration of the counterweight blocks.

Figure 12 shows the schematic diagram of the two counterweight blocks' thicknesses, and Table 2 lists the specific thickness dimensions of five sets of different thickness counterweight blocks.

Figure 13 shows the different inertial force curves of the gearbox's output shaft corresponding to the different thicknesses of the counterweight blocks. It can be observed that as the thickness of the two counterweight blocks increases, their corresponding masses also increase. Consequently, the inertial force curve of the gearbox's

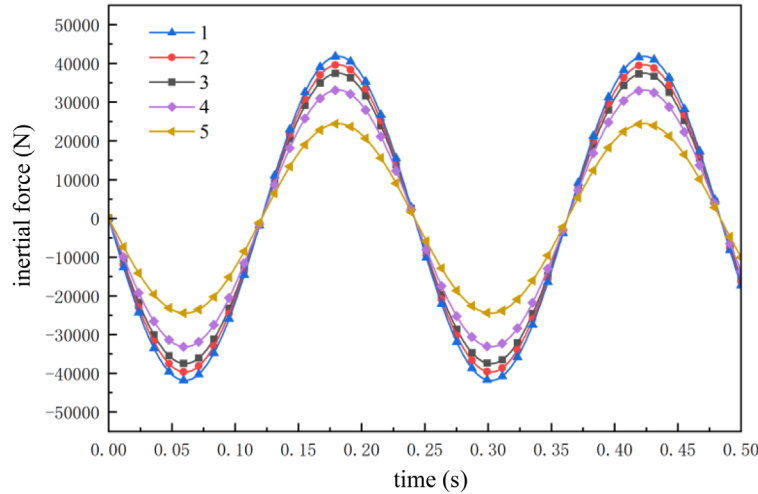


Fig. 13. Inertial forces of the output shaft with different thickness counterweight blocks.

output shaft along the direction of motion gradually becomes smoother. This indicates that the thicker the counterweight blocks are, the greater their mass, and the more pronounced the balancing effect on the inertial force of the gearbox's output shaft. However, the thickness of the counterweight blocks cannot be arbitrarily increased. Taking into account the constraints of installation space and bearing life, as well as the balance effect of inertia forces, the optimal balance is achieved when the thickness of counterweights 1 and 2 are 125 mm and 130 mm, respectively. Continuing to increase the thickness of the counterweight blocks not only enhances the balancing effect on the inertial force of the output shaft but also leads to a significant increase in system weight. However, excessively thick counterweight blocks can affect the overall coordination of the transmission mechanism and occupy more space. Therefore, it is important to choose the thickness of the counterweight blocks reasonably. In this case, the third option is selected.

5.3 Impact of the presence or absence of counterweight blocks on the inertial torque of the system's output shaft

In Section 5.1, we investigated the impact of adding counterweight blocks on the inertial force of the system's output shaft. In this section, we will follow the same approach to study the impact of the presence or absence of counterweight blocks on the inertial torque of the system's output shaft.

During the startup phase, the rotating axis, eccentrically positioned on one side of the large gear, undergoes a continuous increase in angular velocity. This results in the rotation of the axis around the centre of the large gear, creating an inertial torque. On the upper and lower sides of the extended end of the rotating axis, counterweight blocks 1 and 2 are respectively installed in opposite directions. As the rotating axis rotates clockwise around the centre of the large gear, it generates a torque. At the same time, the two

counterweight blocks rotate counterclockwise, generating an inertial torque that partially counteracts the impact of the inertial torque caused by the startup on the rotating axis.

Figure 14 respectively depict the inertia torque curves of the rotating axis during the startup phase and the corresponding RMS values of the inertia torque. It can be observed that without counterweight blocks, the inertia torque generated by the rotation of the axis around the centre of the large gear is relatively high, with a RMS value of 42827.37 N·m; After adding counterweight block 1, there is a certain reduction in the rate of change of the inertia torque, although not particularly significant. The RMS value decreases to 36322.62 N·m, which is a reduction of 15.19% compared to the case without counterweight blocks; With the further addition of counterweight block 2, there is a noticeable reduction in the rate of change of the inertia torque. The RMS value decreases to 25885.70 N·m, which is a reduction of 39.56% compared to the case without counterweight blocks. Adding counterweight blocks can effectively reduce the significant impact of inertia torque caused by angular acceleration changes during the startup phase, thereby further improving the smooth operation of the mechanism.

5.4 The impact of counterweight blocks with different thicknesses on the inertia torque of the system output shaft

In the previous section, it was found that adding counterweight blocks can effectively reduce the impact of inertia torque caused by the angular acceleration changes of the eccentric rotating shaft. However, different thicknesses of counterweight blocks have varying masses, which in turn result in different rotational inertias around the axis of rotation. As a result, their effectiveness in balancing the inertia torque of the rotating shaft also differs. Therefore, in this section, we will investigate the

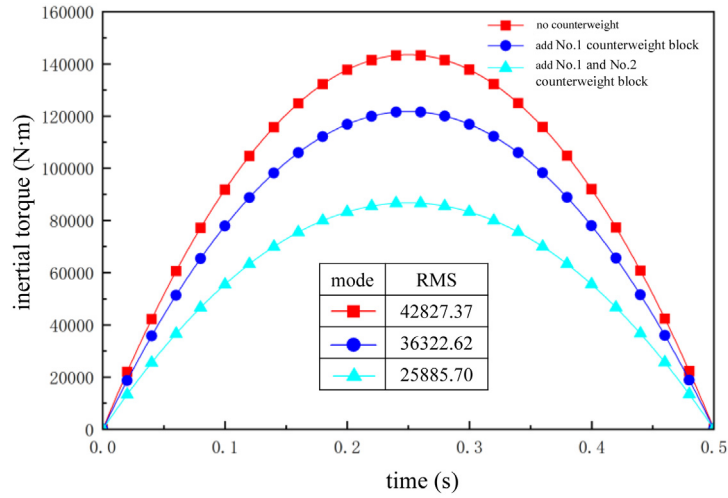


Fig. 14. Initial stage rotational axis inertial torque.

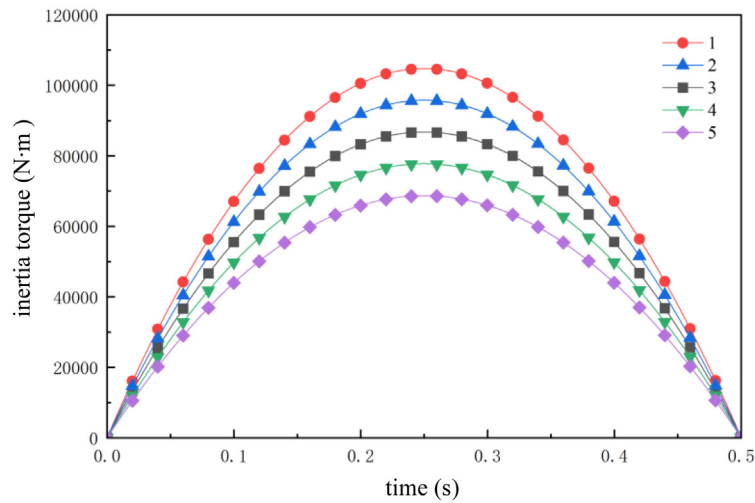


Fig. 15. Different thickness of counterweight blocks corresponding to the inertial torque of the rotating shaft.

impact of different thicknesses of counterweight blocks on the inertia torque of the eccentric rotating shaft, based on the five sets of configurations listed in Table 2.

Figure 15 shows the inertia torque curves of the rotating shaft for different thicknesses of counterweight blocks. It can be observed that as the thickness of the two counterweight blocks increases and their mass increases, the moment of inertia of their rotation around the axis also increases. As a result, the balancing effect on the inertia torque of the eccentric rotating shaft becomes more pronounced. When the thickness of the counterweight block 1 is 125 mm and the thickness of the counterweight block 2 is 130 mm, a good balance has been achieved for the inertia torque of the rotating shaft. As the thickness of the counterweight blocks continues to increase, although the balance effect on the inertia torque becomes more pronounced, it leads to a significant increase in system weight and occupies a larger space. Therefore, it is

necessary to select the thickness of the counterweight blocks reasonably, considering not only achieving the optimal balance effect but also considering the actual total weight of the system and the coordination of the mechanism.

6 The influence of the rotational speed of the driving gear on the inertial force of the output shaft

During the operation of the gearbox, the different input speeds result in varying levels of inertial force along the direction of motion of the output shaft. The presence of inertial force can cause inertial impact on the output shaft at extreme positions. Therefore, it is necessary to study the influence of different rotational speeds of the driving gear on the inertial force of the output shaft.

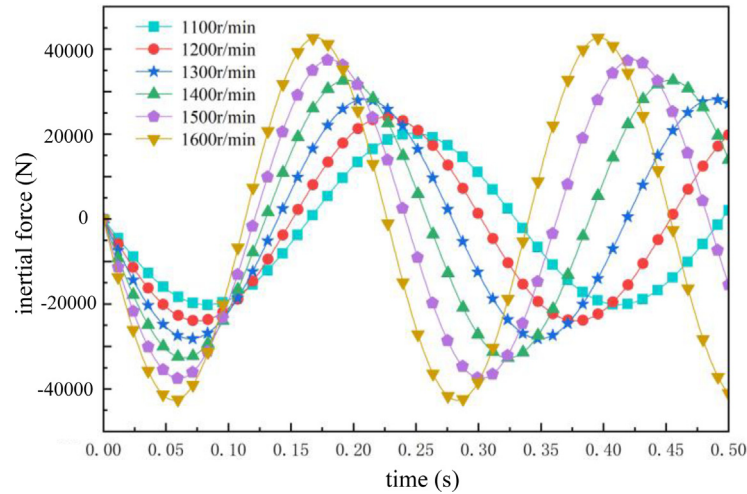


Fig. 16. Inertial force on the output shaft corresponding to different speeds of the driving gear.

The driving gear at the input end was set to rotational speeds of 1100 r/min, 1200 r/min, 1300 r/min, 1400 r/min, 1500 r/min, and 1600 r/min. Figure 16 shows the corresponding curves of the inertial force along the direction of motion of the output shaft. It can be observed that the inertial force of the output shaft exhibits periodic variations over time. Furthermore, as the rotational speed of the driving gear increases, the magnitude of the inertial force on the output shaft also increases, resulting in larger increments. When the rotational speed of the driving gear is set to 1500 r/min, it satisfies the requirement of 63566 N horizontal thrust force for the rolling mill. Although higher input speeds can generate greater horizontal thrust force, they also result in increased inertial forces along the motion direction of the output shaft. Therefore, it is necessary to strictly control the rotational speed of the driving gear in order to achieve both the desired magnitude of horizontal thrust force for the rolling mill and effectively reduce the inertial impact on the output shaft at extreme positions.

7 Conclusion

- Based on virtual prototype simulation technology, a dynamic simulation model of the planetary gear transmission system was constructed to study its dynamic characteristics. The simulation results show that the meshing forces between the high-speed stage and the planetary stage gears fluctuate around 7939.63 N and 343.4 N, respectively. Compared to theoretical values, the relative errors are 0.2% and 0.05%, indicating small relative errors and validating the rationality of the simulation.
- The study investigated the effects on the system's output shaft inertia force and inertia torque under three conditions: without counterweight blocks, with the addition of counterweight block 1, and with the simultaneous addition of counterweight blocks 1 and 2. The results show that after adding counterweight block 1, the RMS values of the system's output shaft inertia

force and inertia torque are 28996.83 N and 36322.62 N·m, respectively, representing a reduction of 18.56% and 15.19% compared to the case without counterweight blocks. The balancing effect is not significant. By further adding counterweight block 2, the RMS values of the system's output shaft inertia force and inertia torque are 26140.38 N and 25885.70 N·m, respectively, showing a reduction of 26.59% and 39.56% compared to the case without counterweight blocks. The balancing effect is more pronounced.

- The study investigated the effects of different thicknesses of counterweight blocks on the system's output shaft inertia force and inertia torque. The results show that the balancing effect on inertia force and inertia torque varies with different thicknesses of counterweight blocks. When the thicknesses of counterweight blocks 1 and 2 are 125 mm and 130 mm, respectively, a good balance effect has been achieved. Further increasing the thickness of the counterweight blocks will significantly increase the total weight of the system and occupy more space. It also affects the overall coordination of the transmission mechanism. Therefore, it is necessary to choose the thickness of the counterweight blocks reasonably.
- Six different speeds were set for the driving gear to investigate their respective effects on the inertia force of the system's output shaft. The results show that when the driving gear speed is set to 1500 r/min, the output force of the output shaft is 63566 N, which can meet the force requirements for normal operation of the Pilger cold rolling mill. At the same time, the inertia impact on the output shaft at the extreme position is not significant, resulting in smoother system operation.

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Conflicts of interest

The authors have no relevant financial or non-financial interests to disclose.

Data availability statement

The data that support the findings of this study are available from the corresponding author upon reasonable request.

Author contribution statement

All authors were involved in the design of the model and discussion of the results. Conceptualization Z.C.; writing original draft preparation Y.H. and Y.Z.; validation, review X.D.; supervision C.M. and L.T.. All authors have read and agreed to the published version of the manuscript.

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