Concept of a new overrunning clutch with the characteristic of reverse force transfer at high speed

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Abstract
To extend the functionality of the overrunning clutch. A new overrunning clutch with the characteristic of reverse torque transfer direction at high speed is proposed in this paper. This clutch consists of a traditional one-way overrunning clutch and a reverse torque transfer direction subsystem. It has three work scenarios according to the angular velocity relationship between the master shaft and the slave part, which are positive torque transfer direction, overrunning, and reverse torque transfer direction. The principle and schematic structure of this clutch are detailed, relationships among the rollers, springs, and hammers are discussed, and threshold angular velocity of reverse torque transfer direction is achieved. The dynamic model of the reverse torque transfer direction part of this concept is established with Simulink software. A series of simulation scenarios that the slave shaft's rotational speed changed at different spring stiffness is conducted. The results show that the threshold value of the reverse torque transfer direction is increasing with the decrease of the spring stiffness. At last, a prototype system is made to roughly validate the feasibility of this concept.

KEYWORDS
mechanisms, overrunning clutch, reverse torque transfer direction

1 | INTRODUCTION

In the machine systems, actuating mechanisms will work when receiving the torque from prime movers or other additional power sources, such as gravity. For example, the vehicle could be accelerated without the torque from the engine in the downhill scenario. Although the driver could reduce fuel consumption and emissions by shifting the transmission ratio to the neutral and shutting down the engine in this scenario, excessive speed of the vehicle will lead to the accident safety problem, and the speed controlling with brake system raises the wear of brake pads. The problem will be easily
solved if developing an overrunning clutch which has two functions: overrunning, reconnecting to realize reverse force transfer when the angular velocity of the slave shaft is bigger than the threshold speed. During the downhill scenario, the vehicle equipped with this overrunning clutch could adaptively realize neutral coasting and engine anti drag braking at low speed and high speed, respectively. It benefits to improve the economy and reliability of mechanisms.

Overrunning clutch is widely used in the machine system due to its advantages in adaptive one-way torque transmission. Yang analyzed the influence of the combination of a controllable overrunning clutch on the shift characteristic of transmission. Liu discussed the sliding friction characteristics of an overrunning clutch which is used in pulse-type continuously variable transmission. Zhang introduced the static and dynamic behaviors of belt-drive dynamic systems with a one-way clutch. Ding presented the speed coupling control strategy of the Overrunning-clutch in a parallel plug-in hybrid electric vehicle. A continuous rotary mini-motor is designed based on the shape-memory wires and overrunning clutches. A kind of load-sensitive dual-speed transmission is developed with overrunning clutches and gears. A novel structure of the two-speed I-AMT is discussed in Reference 8, which utilizes the overrunning clutch to realize the gear shifting automatically. Ma used the finite element method to analyze the characteristic of the dynamic load of the overrunning clutch for a dual turbine torque converter. However, the research on the aforementioned scenario has not been seen due to the lack of the new kind of overrunning clutch proposed in this paper.

There are four types of overrunning clutches widely utilized in the machine system based on its structure, centrifugal clutch, roller overrunning clutch, spring overrunning clutch, and sprag overrunning clutch. Many researchers devote themselves to the performance of these clutches. Yan analyzed the work characteristic roller overrunning clutch, and revealed the relationships among clutch structure state, load, bearing performance, and strength. Xue built the mathematical model of the combined energy consumption of roller overrunning clutch. Some researchers analyzed the dynamic wedging characteristic, working principle, and design method of the spring clutches. The dynamics characteristics of the variable cross-section overrunning spring clutch during the engagement process are discussed, and the relationship between the minimum friction coefficient and roughness under contact conditions is established. In the aspect of the sprag overrunning clutch, some researchers analyzed its mechanical characteristics, discussed its self-locking condition and the influence of inner and outer ring wear on working wedge angle, analyzed the shape of the wedge, and put out methods to reduce the wear of wedge.

With these structures, researchers extended the functions of the overrunning clutch. Bidirectional overrunning clutch and bidirectional controllable overrunning clutch are developed and utilized in the mechanical engineering area. Some researchers analyzed the dynamic characteristic of the bidirectional controllable overrunning clutch. However, these overrunning clutch transfers torque in one way, from the master shaft to the slave shaft, which limits the application of the overrunning clutch.

In this paper, we propose a new concept of the overrunning clutch whose master shaft could receive the torque from the slave shaft when the rotational speed of the slave shaft beyond the threshold value. The working principle of this new overrunning clutch is detailed based on the structure of the roller-type overrunning clutch. The remainder of the paper is shown as follows: Section 2 introduces the design principle of power reverse on an ordinary overrunning clutch. Section 3 presents the dynamic geometric relationship and force analysis of roller and fly-hammer which will work at ultra-high speed. Section 4 establishes the Simulink model of this device and discusses the relationship between the threshold value of ultra-rotation speed and the stiffness of the spring with the model simulation method. Section 5 presents the roughly testable of this new clutch. The last section is the conclusion.

## 2 | THE PRINCIPLE OF POWER REVERSE OF SLAVE SHAFT

### 2.1 | Working principle

As is known, the wedge bond of friction overrunning clutch is achieved by the self-lock principle based on the friction wedge angle. As shown in Figure 1, when the rotation speed of the inner star wheel is bigger than that of the outer shell during anticlockwise rotating, the friction wedge angle \( \alpha \) is produced due to contacting among the roller, the lane of the inner star wheel, and the inner surface of the outer shell. Define the friction coefficient between the roller and the surface of the inner star wheel or out shell as \( \mu \), and we can know that self-lock will work and the power of the inner star wheel will be transmitted to the outer shell when \( \tan(\alpha) < \mu \). Similarly, a self-locking state could also occur when the rotation...
speed of the inner star wheel is lower than that of the outer shell during clockwise rotating. In other conditions, this overrunning clutch will be in a state of disconnection.

2.2 | Design principle of power reverse at high threshold speed

The function of torque transfer direction reverse could be achieved if there is a device installed on the slave shaft of the original overrunning clutch, which could be connected to the master shaft when the rotation speed of the slave shaft beyond the threshold value. According to the principle of centrifugal force, this device could be developed with components such as rollers and springs. The roller is connected to the slave shaft of the original clutch through a spring. With the increase of the rotating speed of the slave shaft, the roller can overcome the spring resistance and move outward. At a certain speed point, the roller touches both the inner star wheel and the outer shell, resulting in self-locking and consequently reversing transmission of the power of the slave shaft. This speed point is the threshold speed of reverse torque transfer direction.

Thus, a kind of reverse devices could be designed, as shown in Figure 2, which is composed of a fly-hammer, a guide rail, and a rod. In this structure, the friction damping generated by the sliding of the fly-hammer along the guide rail can attenuate the vibration of the fly-hammer. The rod pushes the roller moving along the inner surface of the outer shell under the centrifugal force of the fly-hammer and makes it easier to realize self-lock and separation among roller, inner star wheel, and outer shell.

2.3 | Design principle of the new overrunning clutch

According to the former discussion, this new overrunning clutch could be designed as a combination of two overrunning clutches, as shown in Figure 3. Here, the two clutches have the contrary wedge notch orientation.

The working flow is shown as follows. When the rotational speed of the slave shaft 8 is below the threshold value of the torque transfer direction, traditional overrunning clutch works while the torque transfer direction reverse device keeping
the separation state. In this condition, the torque will be transmitted from the master shaft 1 to slave shaft 8 through the self-lock of the traditional clutch and bolt 2 if the rotational speed of the master shaft is bigger than that of the slave shaft. And the torque transfer direction will be cut off by this traditional clutch once the speed of the slave shaft bigger than that of the master shaft. During the speed increase of the slave shaft, fly-hammer 7 overcomes the resistance of spring 6 and pushes the roller 9 moving to the inner surface of the outer shell 4, as shown in Figure 2(B). When the speed of the slave shaft 8 reaches the threshold value of the torque transfer direction, the roller 5 will move to a position touching both the part 4 and part 8. Thus, the torque transfer direction reverse device realizes self-lock, and the torque of the slave shaft will be reversely transmitted to the master shaft.

According to the schematic diagram of Figure 3, this overrunning clutch is modeled with CAD software, as shown in Figure 4. Here, the outer shell is connected to the master shaft, the inner star wheel is connected to the slave shaft.

3 | DYNAMICS OF THE TORQUE TRANSFER DIRECTION REVERSE DEVICE

The work state of this new overrunning clutch is divided into three states according to the speed of the slave shaft, positive torque transfer direction, overrunning, reverse transfer direction. No matter positive torque transfer direction or reverse transfer direction, the fundamental principle is to use the self-lock of friction angle, which is discussed in detail in References 10,19. In this paper, we mainly focus on the kinematics and dynamics analysis among the fly-hammer, spring, and the roller.

3.1 | Geometrical relationships

As is known from Figure 2, the roller of the torque direction transfer reverse device always touches the inner surface of the outer shell due to the centrifugal force during the rotation of the clutch. According to Figure 2 the geometrical relationship among the center point of the roller, the center point of the inner star wheel, and the fixed point of the spring could be drawn, as shown in Figure 5. Where, $O_1$ is denoted as the center point of the inner star wheel, $O_2$ is denoted as the center point of roller, A is denoted as the fixed point of spring, B is denoted as the center of the fly-hammer, $L_1$ is denoted as the length of rod, $R_0$ is the distance between the Point A to the point $O_1$, $R_r$ is the distance from the point B to the point $O_1$, $L_s$ is the length of spring, $R_1$ is the distance from point $O_2$ to $O_1$, $h$ is the distance from $O_2$ to the horizon line that through the point $O_1$, $R$ is denoted as the radius of the inner surface of the outer shell, $r$ is denoted as the radius of the roller, $\theta_1$ is denoted as the angle between the line $BO_1$ and the vertical line, $\theta_2$ is denoted as the angle between the guide rail and vertical direction, $\theta_3$ is denoted as the angle between the line $O_1O_2$ and vertical direction, $\theta_4$ is denoted as the angle between the rod and horizontal direction, $\theta_5$ is denoted as the angle between the rod and the perpendicular of the line $O_1O_2$, $\theta_6$ is denoted as the angle between the guide rail and the rod, $\theta_7$ is denoted as the angle between the vertical line and the perpendicular of the line AB.

The roller of the torque direction transfer reverse device will rotate and move along the inner surface of the outer shell if the rotational velocity changing. So, parameters, $R$, $r$, $\theta_2$, $L_1$, $R_0$, are constants that are determined during the structure
F I G U R E  5  Geometrical relationship of power reverse part. (A) is the position relationship of the guide rail, the fly-hammer, the roller and the inner star wheel. (B) presents the parameters of $h_0$ and $h$ design of the system, and parameters $R_r, L_s, R_1, h$, are the variables. Define $h_0$ as the distance from the plane of the inner star wheel to the point $O_1$, there is,

$$R_1 = R - r.$$  \hfill (1)

$$h \geq h_0 + r.$$ \hfill (2)

According to the geometric relationship of triangles, Equations (3)–(10) could be deduced.

$$h = R_1 \cos(\theta_5).$$ \hfill (3)

$$R_0 = R_r \cos(\theta_1) + L_s \cos(\theta_2).$$ \hfill (4)

$$R_r \sin(\theta_1) = L_s \sin(\theta_2).$$ \hfill (5)

$$\theta_3 = \arccos \left( \frac{R_r^2 + R_1^2 - L_s^2}{2R_rR_1} \right) - \theta_1.$$ \hfill (6)

$$\theta_5 = \frac{\pi}{2} - \arccos \left( \frac{R_r^2 + L_s^2 - R_1^2}{2L_sR_1} \right).$$ \hfill (7)

$$\theta_6 = \arccos \left( \frac{R_r^2 + L_s^2 - R_0^2}{2L_sR_r} \right) - \arccos \left( \frac{R_r^2 + L_s^2 - R_1^2}{2L_1R_r} \right).$$ \hfill (8)

$$\theta_4 = \frac{\pi}{2} \theta_2 - \theta_6.$$ \hfill (9)

$$\theta_7 = \arccos \left( \frac{R_r^2 + L_s^2 - R_0^2}{2L_sR_r} \right) - \frac{\pi}{2}.$$ \hfill (10)

Define $R_r$ as an independent variable, Equations (11) and(12) could be deduced from Equations (3) to (5).

$$L_s = R_0 \cos(\theta_2) - \sqrt{R_r^2 - R_0^2 \sin^2(\theta_2)}.$$ \hfill (11)
\[ \theta_1 = \arcsin \left( \frac{(R_0 \cos(\theta_2) - \sqrt{R_r^2 - R_0^2 \sin^2(\theta_2)}) \sin(\theta_2)}{R_r} \right). \] (12)

### 3.2 Dynamic analysis of the fly-hammer

The fly-hammer suffers the centrifugal force $F_{r1}$, friction force $F_{f1}$, normal force $N_1$, gravity, the push force $F_d$ from the rod, and spring force $F_s$, during the rotation of the overrunning clutch, which makes it and moves along the guide rail. To simplify the analysis, define the direction along the guide rail as the $x$-axis, the plane $x$-$B$-$y$ is perpendicular to the rotation axis of the clutch, and ignore the influence of gravity and spring weight, the forces acting on the hammer could be drawn as shown in Figure 6.

Thus, the following equations could be deduced according to the mechanical principle.

\[ m_1 \ddot{x} + F_s + F_{f1} + F_d \cos(\theta_6) - F_{r1} \sin(\theta_7) = 0. \] (13)

\[ N_1 - F_d \sin(\theta_6) - F_{r1} \cos(\theta_7) = 0. \] (14)

\[ F_{f1} = \text{sign}(\dot{x})|N_1|\mu. \] (15)

\[ x = L_0 - L_s. \] (16)

\[ F_{r1} = m_1 R_r \omega^2. \] (17)

\[ F_s = k_s x. \] (18)

where $x$ is denoted as the compression value of spring, $L_0$ is denoted as the free length of spring, $k_s$ is denoted as the stiffness of spring, $\omega$ is denoted as the rotating speed of the slave shaft, $m_1$ is denoted as the weight of the fly-hammer, $\mu$ is denoted as the friction coefficient between the fly-hammer and the guide rail. $\mu$ is usually set to 0.15.\(^{20}\)

### 3.3 Dynamic analysis of roller

For the roller of the torque transfer direction reverse device, there are two typical states during the working of the clutch, before wedging and wedging. The first is that the roller moves along the inner surface of the outer shell under the push of the rod and the centrifugal force, but it does not touch the inner star wheel. The other is that the surface of the roller touches both the plane of the inner star wheel and the inner surface of the outer shell. The function of self-locking or unlocking will occur in this condition, which is similar to that of the traditional one-way overrunning clutch.

![Force diagram of fly-hammer](FIGURE 6)
3.3.1  |  Before wedging

The roller rolls along the inner surface of the outer shell under the push of the rod at this state. It bears the normal pressure force, adhesive force, and the rolling resistance force from the inner surface of the shell. The rolling resistance force is the result of the contact deformation, which is usually the product of the rolling friction coefficient and the normal pressure force. Figure 7 is the state of the forces acting on the roller. Here, $F_{g2}$ is the adhesive force of the roller on the inner surface of the outer shell, $F_{r2}$ is the centrifugal force of the roller, $\mu_g$ is the adhesion coefficient; $d$ is rolling friction coefficient, the $x$-axis and the $y$-axis are the centripetal direction and circumferential direction of a roller in circular motion, respectively, $m_2$ is denoted as the weight of the fly-hammer, $N_2$ is the normal pressure the inner surface of the outer shell acting on the roller which also is the centripetal force. $\mu_g$ and $d$ are valued as 0.15 and 0.05, respectively.#20

Thus, force relationships are deduced and shown in the following equations.

$$|F_{g2}| \leq \mu_g N_2.$$  \hspace{1cm} (19)

$$F_{g2} + F_d \cos(\theta_3) = m_2 \ddot{y}.$$  \hspace{1cm} (20)

$$N_2 = F_{r2} + F_d \sin(\theta_3).$$  \hspace{1cm} (21)

$$J_2 \dot{\omega}_g = -\text{sign}(\omega_3) N_2 d - F_{g2} r$$  \hspace{1cm} (22)

$$F_{r2} = m_2 (R - r) \omega^2.$$  \hspace{1cm} (23)

$$J_2 = \frac{1}{2} m_2 r^2.$$  \hspace{1cm} (24)

Because the direction of the $y$-axis is the tangent direction of O when it is rotating around the point $O_2$ with the radius of $r$. Equation (25) could be deduced from Figure 5.

$$\dot{y} = R_1 \dot{\theta}_3.$$  \hspace{1cm} (25)

Assuming that the roller purely rolls on the surface of the inner surface of the outer shell, Equation (26) could be deduced.

$$\ddot{y} = r \dot{\omega}_g.$$  \hspace{1cm} (26)

3.3.2  |  Wedging state

A wedging state occurs when the roller moves to a position whose surface touches both the inner surface of the outer shell and the plan of the inner star wheel. The plane of the inner star wheel provides an additional normal force $N_1$ and an additional adhesion force $F_{g1}$ due to the trend of relative motion between it and the roller. Forces acting on the roller at this state is shown in Figure 8.

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**FIGURE 7**  Force diagram of the roller before wedging
Thus, the following equations could be deduced.

\[ F_{g2} + F_d \cos(\theta_5) - F_{g1} \cos(\theta_3) - N_1 \sin(\theta_3) = m_2 \ddot{y}. \]  
(27)

\[ N_2 = F_{i2} + F_d \sin(\theta_5) + N_1 \cos(\theta_3) - F_{g1} \sin(\theta_3). \]  
(28)

\[ J_2 \dot{\omega}_g = (N_2 + N_1)d - F_{g2}r + F_{g1}r. \]  
(29)

\[ |F_{g1}| \leq \mu g N_1. \]  
(30)

When the speed of the inner star wheel exceeds the threshold speed and continues to increase, the pushing force acting on the roller by rod will increase continually, self-locking works at this time and the torque is transmitted from the slave shaft to the input shaft. When the speed of the inner star wheel dropped to the threshold speed and continue to decrease, the fly-hammer pulling the roller with the rod to depart from the inner star wheel surface, torque-transmitting is cut off. The force of the rod acting on the roller is like a compression spring of a traditional one-way overrunning clutch, but its pressure is changing according to the rotation speed of the inner star wheel. As the dynamic analysis of the traditional one-way overrunning clutch is discussed in detail in Reference 19, it is not repeated in the paper. According to this reference, it can be easily concluded that, along with the increasing of the slave shaft’s rotational speed, the wedging angle of the torque transfer direction reverse device increases, and the roller is more easily separated from the plane of the inner star wheel due to the change of the centrifugal force of the fly-hammer.

4  |  SIMULATION AND ANALYSIS BASED ON THE SIMULINK MODEL

4.1  |  Modeling based on Simulink

In this paper, only the model of the fly-hammer and roller system of the torque transfer direction reverse device is established due to the process of separating and wedging of the roller had been well validated by man researchers. The model is mainly the dynamics of the roller and the fly-hammer established with Matlab/Simulink software. The Simulink model is composed of three parts, geometric relationship model, free body diagram models of the fly-hammer and the roller, as shown in Figure 9. Geometric relationship model describes the dynamic geometric relationship of the roller and the fly-hammer. Free body models of the fly-hammer and the roller are divided into two subsystems in the simulation model, force relationship of the roller and the fly-hammer before wedging and force relationships of the fly-hammer after wedging. The simulation scenario is from the starting of rotation to the state that the roller touches both the inner star wheel and outer shell.

In the model, the input variable is the rotation speed of the inner star wheel, and the middle variable is the length of the compression spring. The dynamic process of wedging is neglected during the simulation because it is well studied in Reference 19. The paper just utilizes the length of the compression spring to judge whether the roller is in the state of wedging or not, \( X_0 \) is the maximum compression of spring. Initial values of parameters are set as shown in Table 1.
TABLE 1 Parameters used in the simulation

| Parameter name | Parameter value | Parameter name | Parameter value |
|----------------|----------------|----------------|-----------------|
| R              | 42 mm          | L₀             | 12 mm           |
| R₀             | 40 mm          | X₀             | 4 mm            |
| r              | 5 mm           | μ              | 0.15            |
| a              | 6°             | kₜ             | 100 N/m         |
| L₁             | 10.21 mm       | m₁             | 0.005 kg        |
| h₀             | 36.4 mm        | m₂             | 0.002 kg        |
| d              | 0.05 mm        |                |                 |

4.2 Analysis of simulation result

In the simulation scenario, the angular speed of the slave shaft increases from 10 to 100 rad/s. The results indicate that the push force of the rod and the length of spring compression are in coincide with the design idea, which is shown in Figures 10 and 11, respectively.

It is known from the Figure 10 that the push force of the rod is very small before the roller touches the inner star wheel. The maximum value is less than 0.003 N when the angular velocity of the slave shaft is 54 rad/s, which is the moment the roller first touching the inner star wheel. The wedging begins after that, and the push force increases in the mode of a parabola-like curve. Figure 11 indicates that the length of spring compression increases along with the increase of the angular velocity of the slave shaft.

To find the influence of the stiffness of the spring, a scenario that the stiffness of the spring varies from 1 to 1000 N/m is set, and a series of threshold values of the rotational speed of the slave shaft are calculated from simulation, as is plotted in the Figure 12. Two conclusions are deduced from Figure 13. The first one is that the threshold value is increased with the increase of the stiffness of the spring. The other is that this threshold value increases along with the increase of friction ratio μ when the stiffness of spring is constant.

5 EXPERIMENT

To test the feasibility of the design idea, a rough test table is built, as shown in Figure 13. The testable is composed of a table frame, two motors, an overrunning clutch, battery, and signal acquiring device. The signal is acquired from the controller
of the motor. The controller could change the speed of the motor manually or controlled by other controllers. Motor one is used as a motion source of the master shaft, while motor two is used as a motion source of the slave shaft. In Figure 13, the two motors are in visible due to behind the motor controllers and part of the table frame. The overrunning clutch is made of a three-dimensional (3D) printing machine, the printing material is photosensitive resins, the parameters values are same to that utilized in simulation, as shown in Table 1. Considering the strength and stiffness of this material, the experiment just roughly tests the motion characteristic of this device.

The test shows that the traditional part works well when motor one is powered on, which means that the slave shaft rotates along with the master shaft. If powering off the motor one, and powering on the motor two, the force line is cut off and the speed of the slave shaft increases quickly. When the speed of the slave shaft increases to a certain value, the master shaft starts to rotate again, which means the torque from the slave shaft is transferred to the master shaft.

As the testable is too simple to measure the accurate rotation speed, this test just qualitatively analyses the function of this device. Quantitative analysis will be made when test conditions permitting in the future.
In this paper, we present the structure and principle of a novel overrunning clutch to realize the function of reverse torque transfer direction when the rotational speed of the slave shaft is too high. This clutch is the combination of two parts, the traditional overrunning clutch part, and the torque transfer direction reversing part. The traditional part is used to achieve the function of cutting off the torque transmission route from the master shaft when the rotational speed of the slave shaft is bigger than that of the master shaft but below the threshold value. The torque transfer direction reversing part is to realize the function of transfer of the torque of the slave shaft to the master shaft when its rotational speed is beyond the threshold value. The geometric and force relationship of the roller and the fly-hammer are deduced and modeled with Simulink and 3D design software. The results suggest that the fly-hammer could push the roller to a position to achieve wedging and self-locking. It also indicates that the threshold value of the rotational speed of the slave shaft is increased with the increase of the spring stiffness. The simple testable is also made, and the function of this new device is verified qualitatively.

Due to the limited experimental condition, the physical system of this overrunning clutch is manufactured with 3D printing mode, and the testable is too simple to make a quantitative analysis. The friction coefficient used previously is not got from the real test. And the influence of gravity is also not considered in the process of modeling and calculation. In the future, an accurate experimental test should be made to test the real performance of this device and the influence of undisussed factors. Besides, the reliability of this clutch should be tested and improved if the experimental condition is permitted.
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The data that support the findings of this study are available from the corresponding author upon reasonable request.

CONFLICT OF INTEREST
The authors declare that they have no conflict of interest relevant to this article.

AUTHOR CONTRIBUTIONS
huiyong zhao: Conceptualization; methodology; writing-original draft. baohua wang: Conceptualization; funding acquisition; project administration; supervision; writing-review & editing. guangde zhang: Formal analysis; supervision; writing-review & editing. jiao li: Resources; software.

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