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# Development of a high-speed and low-torque loss two-way clutch

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**Abstract**

In this study, a two-way clutch mechanism using planetary gear meshing was proposed. The proposed mechanism allows low torque loss and high-speed rotation instead of a larger backlash compared to a roller type two-way clutch. The unique structure of this mechanism that makes such functionality possible comprises two independent planetary gears with variable distances between the axes. When the input shaft rotated, the planetary gears rotated at a distance. Torque is transmitted by the input shaft pushing directly on the output shaft, so losses on the tooth surface are small. As with gear reducers, high-speed rotation is possible. When the output shaft rotated, the two planetary gears engaged, and the shaft locked. A flat cam was used to switch between the above two states, so the overall structure was simple. The diameter, thickness, and weight of the developed prototype were 44 mm, 24 mm, and 78.8 g, respectively. We experimentally confirmed that the mechanism worked as expected. The theoretical locking limit torque was 1.16 Nm. The torque loss was 5 mNm at the static condition and less than 10 mNm at a high speed of 4000 rpm.

**Keywords:** Two-way clutch, Torque diode, Non-back-drivability, High speed, Low torque loss

## Background

A two-way clutch is a mechanical element used to switch between transmission and disconnection of power. Although there was a one-way clutch with a similar name, the one-way clutch switches between transmission and disconnection depending on the direction of rotation, whereas the two-way clutch is independent of the direction of rotation (Fig. 1a). As shown in Fig. 1b, two-way clutches are divided into active and passive types based on whether or not an external actuator is required to switch the action [1]. The active type is a type of electromagnetic clutch and is incorporated in transmissions in wind power generators [2] and four-wheel-drive systems of automobiles [3] for operation switching. The passive type is a type of non-back drive device. It transmits the power from the input shaft to the output shaft but interrupts the power from the output shaft. When a passive two-way clutch is placed between the actuator and the load, it protects the actuator from the load. The passive type is further divided into the locked type and the free type according to the form of power interruption. In the lock type, the output shaft is fixed to make rotation impossible [1, 4], while in the free type, the output shaft is disconnected from the input shaft, and only the output shaft idles to interrupt the power [5, 6]. The free type is used, for instance , to switch the zoom of a camera lens manually and automatically [6]. Since the locking type maintain the state without continuous energy supply, its installation in robots leads to energy saving and miniaturization of actuators. Therefore, it is actively used in the field of artificial limbs [7–9]. Based on these characteristics, a locking two-way clutch into the transmission mechanism with a two-motor coupled drive which was developed in the past was investigated [10]. This transmission has a high-speed drive system and a high-torque drive system, and these two types of drive systems is switched by coupled drive. When the load is small, the high-speed drive system is used, and when the load is large, the high-speed drive system is locked to operate so that the load is supported by the high-torque drive

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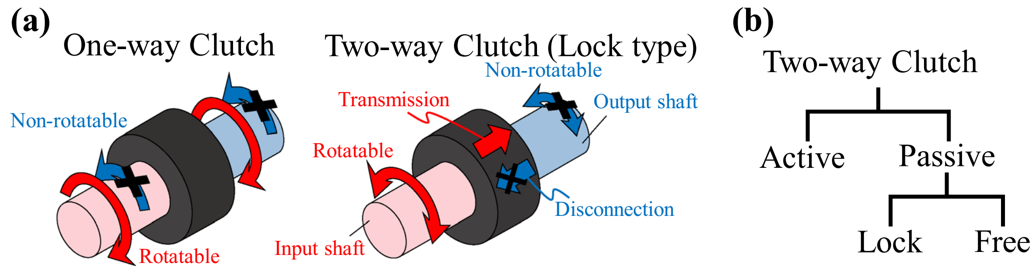


**Fig. 1** **a** Difference in function between one-way clutch and two-way clutch. **b** Classification of two-way clutches

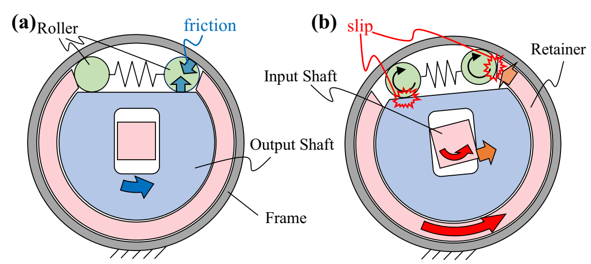


**Fig. 2** Existing two-way clutch operation. **a** Power from the output shaft is not transmitted. **b** Power from the input shaft is transmitted

system. The two-way clutch is used to switch the operation of the high-speed drive system.

Several methods have been proposed to realize the locking type two-way clutch, but one that is currently widely used is the roller friction method [1, 4]. Figure 2 shows the structure and operating principle of a locking two-way clutch using roller friction. Fig. 2a shows the neutral state. When rotation is applied to the output shaft, the roller bites into the space between the output shaft and the frame like a wedge, and the shaft is locked by the frictional force generated by it. When rotation is applied to the input shaft, the retaner pushes out the roller as shown in Fig. 2b. Then, the input shaft inside pushes the output shaft to transmit power. The most important feature of the two-way clutch based on this principle is its simple structure. A simpler and smaller device with an integrated input shaft and retainer has been developed by Controzzi et al. and Kobayashi et al. [11, 12]. In addition, the locking performance is changed by adjusting the parameters, and the design method has been proposed by Controzzi et al. [1].

However, the roller-type two-way clutch has several issues, which prevent its introduction into the transmission mechanism with a two-motor coupled drive [10]. The first is that the torque loss during shaft rotation is large. This has been pointed out in the research by Yamamoto et al. [13]. The TDL28 of NTN Corporation is shown that has an idling torque loss of more than 200 mNm [14]. Second, the clutch is not suitable for high-speed rotation. The allowable speed of the OSCM3-5TW of Origin Electric Co., Ltd. is 560 rpm [15], a small value compared to the rotational speeds (above thousand rpm)of electric motors usually used in the robotics field. Therefore, it is necessary to go through a reduction gear to use it, as it cannot meet the need for a faster rotation. In addition, a gear reducer with a large reduction ratio does not require a two-way clutch because it functions to prevent back drive. The mechanism by which these problems occur is explained below. When the shaft rotates, as shown in Fig. 2b, the rollers are pushed by a spring and come into contact with two or more surfaces. The rollers rotate along the frame, but slipping occurs at other contact points. We consider the frictional loss owing to this slippage to be the main cause of torque loss in two-way clutches. In addition, prolonged slippage causes wear. In a roller-type two-way clutch, back-driving is prevented by the static frictional force; therefore, if the condition of the contact surface changes owing to wear, it cannot lock sufficiently. In other words, the wear rate is directly related to life. Because the wear rate increases with rotational speed, we consider that the allowable rotational speed is limited to ensure life. As a supplement, grease cannot be used to solve problems because it also reduces the static friction force during locking.

Since slippage led to losses, was wondered whether it was possible to realize a two-way clutch using gears that did not slip during rotation. A typical back drive prevention mechanism using gears is a worm gear [16]. The worm gear prevents back-drive by directing the force generated on the tooth flanks in the direction of the rotation axis. Other methods have been proposed to direct the force applied to the tooth flanks toward the center of rotation [17], but all these mechanisms involve a speed reduction. In this study, we focused on the phenomenon that an odd number of gears cannot rotate when they mesh in an annular pattern and proposed a new two-way clutch mechanism that used this phenomenon. A lock control system [18] has been developed using the same phenomenon. Our device is different from the conventional ones in that it did not use an actuator for control. This two-way clutch had a planetary gear mechanism using internal and planetary gears. As the gears of a general gear reducer can rotate at high speeds, this mechanism can also rotate at high speeds. In this planetary gear mechanism, the input and output shafts are directly connected to the planetary carrier without using a sun gear. This means that the rotational speed is transmitted at a high speed on a one-to-one basis, and no load is applied to the gears. Therefore, losses on the meshing tooth surfaces can be reduced. However, gears have several disadvantages. These disadvantages are discussed in "Disadvantages of Using Gears" in the "Analysis and Discussion" section.

## Two-way clutch using gear meshing

### Overall structure

Figure 3 shows the structure of the proposed two-way clutch. The planetary gear mechanism was sandwiched between an input shaft and an output shaft with a cam shape. The planetary gear mechanism had an internal gear, four planetary gears and two planetary carriers. The two planetary carriers were independent of each other. The point-symmetrically arranged planetary gears were held together by the same planetary carrier on the central axis. If there were only two planetary gears, there would have been a possibility that the gears move unintentionally due to the effect of gravity acting on the gears as shown in Fig. 4a, or that the center of gravity, which was far from the center of rotation, generated vibration.

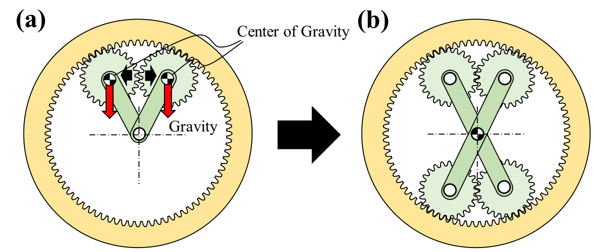


**Fig. 4** Four symmetrically arranged planetary gears balance the center of gravity and counteract the effects of gravity

Therefore, another pair of planetary gears was prepared as a counterweight to prevent these effects, as shown in Fig. 4b. Since the central axis of the gear was fixed to the planetary carrier, a bearing was built into the gear. The central axis of the gear protrudes from both sides of the gear, and this protruding part is called the pin.

Within this structure, the planetary gear mechanism had the role of locking function, and the cam structure had the role of state switching. The detailed principle of the locking function is shown in "Planetary gear mechanism and locking principle" section, the detailed principle of the state switching is shown in "Cam structure and state switching principle" section, and the design conditions are shown in "Design condition" section.

### Planetary gear mechanism and locking principle

In the planetary gear mechanism shown in Fig. 3, the distance between the two planetary gears was variable because the two planetary carriers were independent.

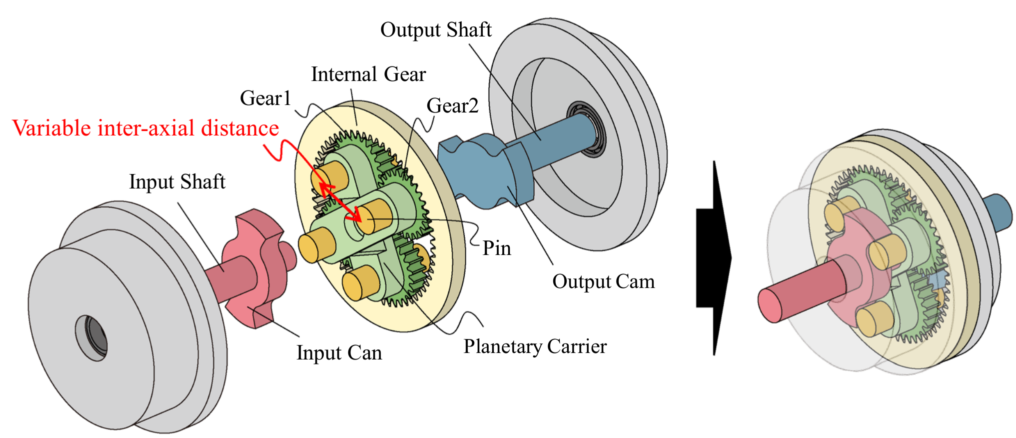


**Fig. 3** Planetary two-way clutch mechanism (3D CAD drawing of a prototype machine)

Therefore, it took two states, as shown in Figs. 5 and 6. If the internal gears and gears 1 and 2 were annularly engaged as shown in Fig. 5, the rotation and revolution of gears 1 and 2 would have been restricted. In other words, the mechanism would have been in a locked state. On the other hand, if gears 1 and 2 had not meshed as shown in Fig. 6, the two gears would have revolved and rotated around along the internal gear. In this two-way clutch, when the output shaft rotated, the rotational force was applied to the two gears from the outside to engage them, and the shaft was locked in the state shown in Fig. 5. When the input shaft rotated, the force was applied from the inside to disengage the gears, and the shaft was unlocked in the state shown in Fig. 6.

The distance between the planetary gears became the minimum and maximum value in the locked state and unlocked state, as shown in Figs. 5 and 6, respectively. In addition, this difference became the backlash of the two-way clutch. The minimum value was the standard meshing distance of the gears and depended on the specifications of the gears, but the maximum value was set arbitrarily. In addition to the minimum distance for disengagement, a certain amount of gap was necessary to prevent interference of the tooth tips during rotation. However, if the gap is too large, the backlash will also be large, so it is necessary to determine the limit based on machining errors.

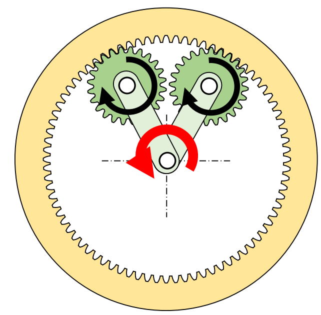

**Fig. 6** When the internal gear and the two spur gears mesh in a chain, they can rotate and revolve

### Cam structure and state switching principle

If the two planetary gears were taken out of Fig. 3, the cam of the input shaft would have been located inside the pins of the two gears and the cam of the output shaft would have been located outside the pins of the two gears. Fig. 7 shows the internal state when clockwise rotation was applied to both, the input and output shafts. When the input shaft rotated, the cam of the input shaft contacted pin 2 and pushed gear 2 outward, as shown in Fig. 7a,i. As a result, the meshing of the gears was disengaged and the gears became unlocked, and the gears started to revolve while spinning, as shown in Fig. 7a,ii. Pin 2 then pushed the cam of the output shaft, and the rotation was transmitted. Initially, pin 1 was not in contact with the cam of the input shaft, so gear 1 rotated in contrast to gear 2 due to the meshing, as shown in Fig. 7a,i. When the gears were disengaged and the distance between the shafts reached its maximum value, pin 1 contacted the other cam on the output shaft, and gear 1 began to rotate in the same direction as gear 2 while maintaining a constant distance, as shown in Fig. 7a,ii. On the other hand, when the output shaft rotated, the cam of the output shaft pushed pin 1 inward, as shown in Fig. 7b,i. This caused the gears to engage and lock, and the force to be transmitted to the frame, as shown in Fig. 7b,ii. Therefore, the shaft could not rotate any further. To prevent the gears from meshing more than necessary, both pins contacted the cams of the input shaft when they were engaged at the standard pitch (minimum distance between the gear shafts). In the counterclockwise

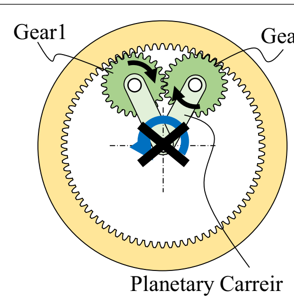


**Fig. 5** When the internal gear and the two spur gears mesh in an annular shape, rotation and revolution become impossible

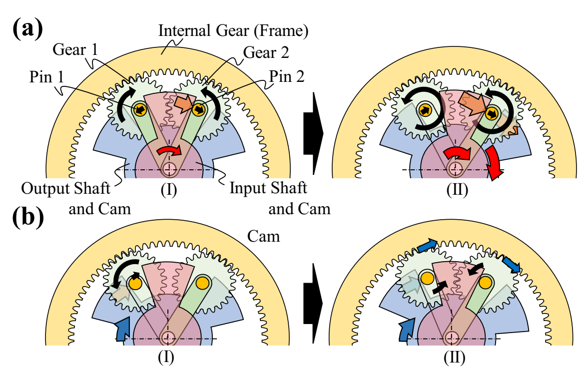


**Fig. 7** Gear movement inside the two-way clutch when clockwise rotation is applied. **a** When rotation is applied to the input shaft. **b** When rotation is applied to the output shaft

rotation, the cam of the input shaft worked to separate the gears, while the cam of the output shaft worked to bring them closer together, resulting in the same behavior. That was how the cam and pin changed the distance between the shafts to switch the state.

### Design condition

Based on the above operation, the design conditions for the cams of the input and output shafts were determined as follows

- When the distance between the gear shafts was at its minimum values (meshing distances at the standard pitch circle), the pins were in contact with the cams of both input shafts.
- When the distance between the gear shafts were at its maximum value, both pins were in contact with the cam of the output shaft.

Next, the tooth number condition of the planetary gear was determined. Since the planetary gear meshed with the internal gear, the lower limit was the minimum value that did not cause involute interference. The upper limit was the value of the limit at which two pairs of four gears were placed in the internal gear at the maximum distance between shafts. Figure 8 shows the geometric state when the planetary gears were placed at the maximum distance between shafts. In the figure, $a_{01}$ is the distance between the internal gear; the planetary gear, $a_{12}$, is the distance between the planetary gears; $d_{1a}$, is the tip diameter of the planetary gear, and $e$ is a small gap to ensure that the tips do not interfere in the unlocked state. $\alpha$ is the angle between the y-axis and OA., and $\beta$ is the angle between OA and OB. The maximum condition for the number of teeth is $\alpha + \beta < 90$ [°] because the planetary gear should be above the x-axis. Since $\alpha > \beta$ for $e$, the maximum condition is $\alpha < 45$ [°]. Using the module $m$ and the tooth numbers $z_0$ of the internal gear and $z_0$ of the planetary gear, we get,

$$\sin^{-1}\frac{a_{12}}{2a_{01}} = \sin^{-1}\frac{m(z_1+2)+e}{m(z_0-z_1)} < \frac{\pi}{4} \tag{1}$$

The gap is defined as $e = m\epsilon$, assuming that it is proportional to the module. Thus, the maximum number of teeth can be obtained as follows, noting that the gears are assumed to be untransformed.

$$\begin{aligned} \sin^{-1}\frac{z_1+2+\epsilon}{z_0-z_1} &< \frac{\pi}{4} \\ z_1 &< \frac{z_0\sin\frac{\pi}{4}-2-\epsilon}{\sin\frac{\pi}{4}+1} \end{aligned} \tag{2}$$

Figure 9 shows the tooth number range of the gears that were used based on the minimum and maximum tooth number conditions. The gap was assumed to be $\epsilon = 0.6$. The range became wider as the number of teeth in the internal gear increased. The minimum number of internal gears was 55.

## Results

### Design and fabrication of prototypes

Figure 3 shows the 3DCAD diagram of the designed prototype. Internal gears (module 0.5, tooth number 60, tooth width 3 mm, C3604) and spur gears (module 0.5, tooth number 23, tooth width 3 mm, C3604) were used. These tooth numbers were within the region shown in Fig. 9. The backlash of this machine was 10.69

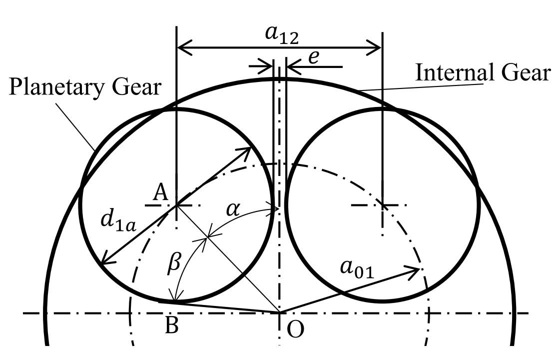


**Fig. 8** Model when the planetary gears are placed at the maximum inter-axis distance

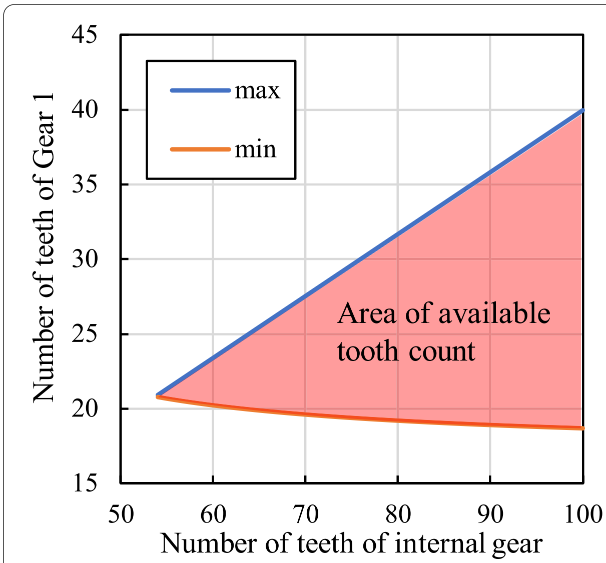


**Fig. 9** The range of usable teeth determined from the tooth number condition

°. The difference from Fig. 7 was the cam shape of the output shaft. It had a cross shape to evenly transmit the force to the four gears in the locked state. The new cam part was designed so that the pins and cams all contact in the same direction when locked.

The allowable torque in the locked state was calculated based on the allowable circumferential force of the gear. The strength of the spur gear is referred to as the dangerous side between the internal gear and the spur gear. The allowable circumferential force was calculated to be 19.3 N based on the bending strength of the teeth using the gear technical data [19] of Kohara Gear Industry Co. However, the number of repetitions should be $10^7$ or more. Assuming that the torque applied from the output shaft was equally distributed to the four gears and transmitted to the internal gear, the theoretical value of the allowable locking torque was obtained to be 1.16 Nm.

Figure 10 shows an image of the fabricated prototype. The dimensions were $\phi 44 \times 24$ mm and the weight was 78.8 g. It has been confirmed that the proposed mechanism can switch the state as expected using this prototype.

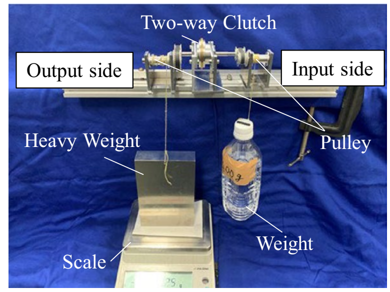


**Fig. 11** Experimental device for measuring torque loss

### Measurement of torque loss

Figure 11 shows the experimental device for measuring static torque loss. When a weight of a certain mass was hung from the input shaft through a pulley, the torque applied to the output shaft was measured, and the static torque loss was calculated from the difference. The output torque was calculated from the tension of a fixed thread attached to the output shaft through a pulley. The radius of the pulley was 5.9 mm for both, the input and output shafts. The results of the experiment are shown in Fig. 12. This graph shows only the average of the two measurements because the errors were small. The horizontal axis was the input torque, and the vertical axis was the output torque and torque loss. The torque loss in the figure was the value after subtracting the device loss other than the two-way clutch (6.1 mNm). Figure 12 shows that the torque loss was almost constant, although it increased slightly as the input torque increased. The average value was 4.4 mNm.

Next, the dynamic torque loss was measured. The experimental device is a modification of Fig. 11, where the input side was replaced by a motor (maxon, 249245) and the output side by a certain weight. The motor connected to the input shaft lifted a weight hanging on the output shaft in a stable state. The dynamic torque loss was obtained from the difference between the input torque calculated from the current value and the output shaft torque due to the weight. The result is shown in Fig. 13. This graph shows the average of two measurements. Error bars show the maximum and minimum values. The horizontal axis was the load on the output shaft, and the vertical axis was the input torque and torque loss. As in the static case, the torque loss in the figure was the value after subtracting the device loss from the two-way clutch (8.6 mNm). The error in Fig. 13 shows the range of variation in a stable state. Figure 13 shows that the torque loss was almost constant regardless of the load. The average value of torque loss was 2.7 mNm.

In addition, the change in torque loss with increasing rotation speed was measured. The experimental device

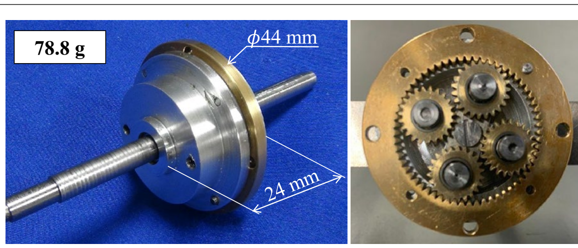


**Fig. 10** External and internal views of the prototype

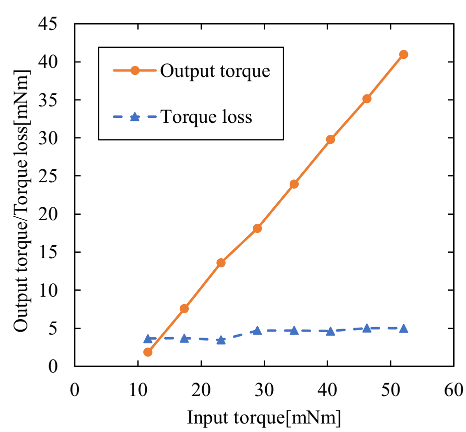


**Fig. 12** Measurement result of static torque loss

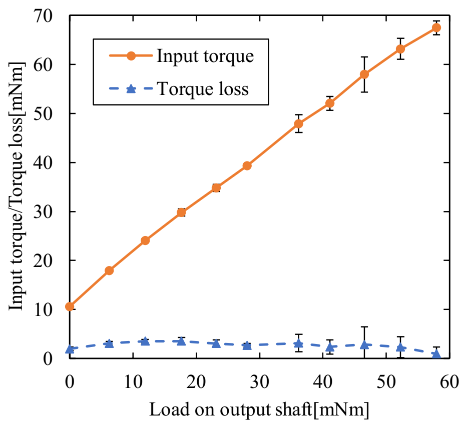


**Fig. 13** Measurement result of dynamic torque loss

is a modification of Fig. 11, where both the input and output sides were replaced with motors. The motor on the output side was not connected to the power supply, and the rotation speed was determined from the back EMF voltage of that motor. The measured value was the median value because the value kept changing approximately 0.01V during the measurement. Figure 14 shows the measurement results of torque loss against rotation speed. Measurements were taken three times, and the third result is shown because the results gradually improved as the gears became more familiar. The horizontal axis was the rotation speed, and the vertical axis was the torque loss. The load generated by the back EMF of the motor on the output shaft was measured in advance, and the value was subtracted in the figure. Figure 14 shows that the torque loss increased depending on the rotation speed. However, the torque loss was about 10 mNm at 3660 rpm, which was the highest among all the speeds measured in this experiment.

### Input torque required for unlocking

The input torque required to unlock the output shaft with load was one of the parameters that describe the performance of a two-way clutch. When rotating in the opposite direction of the load, the input torque should be greater than the load, when rotating in the same direction as the load, the input torque may not be greater than the load. In the case of roller-type two-way clutches, it has been shown that this value depends on the parameters of the design according to Controzzi et al. [1]. Therefore, the torque required for unlocking was also measured for this two-way clutch. First, a weight was hung from the output shaft through a pulley to lock the device. Then, the torque was gradually applied to the input shaft, and the input torque at the moment the unlocked device was measured. Figure 15 shows the average values of the five measurements. However, results that were out of the range of the measuring instrument were excluded. Error bars show the maximum and minimum values. The horizontal axis was the load torque applied to the output shaft, and the vertical axis was the torque required for unlocking. The dashed line in the figure connected the points where the unlocking torque was the same as the load torque. The figure shows that the unlocking torque became larger than the load torque.

## Analysis and discussion

### Validity of the gap

In Eq. (2), the gap was assumed to be proportional to the module. This makes Eq. (2) a simple conditional equation that dose not include the module in the coefficients. However, this assumption is not valid. This gap is provided such that when the planetary gears are separated, they do not interfere with each other owing to the machining or assembly accuracy. The larger the module,

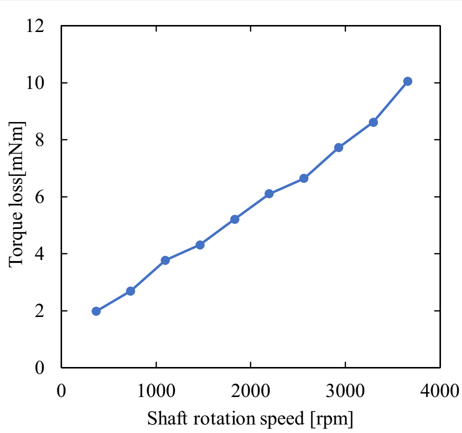


**Fig. 14** Increase in torque loss due to increase in shaft speed

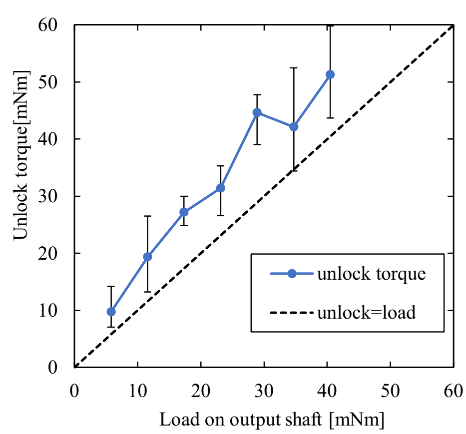


**Fig. 15** Torque required to unlock when trying to rotate the input shaft in the same direction as the load

the larger the error. Therefore, the gap must be increased accordingly. However, gear accuracy and tolerance are not proportional to the module, and their sizes vary depending on the number of teeth.

The gap $e$ is assumed to be a constant, and its size is defined as 0.3 mm which was used in this prototype. The order of error (1 to 10 $\mu$m) is small compared to the order of the gap (100 $\mu$m); therefore, it can be absorbed sufficiently. In fact, the allowable full engagement of both gear tooth flanks specified in JIS B 1702-2 is 127 $\mu$m maximum for a diameter of 280 mm or less in grade N8 [20]. The value of 0.3 mm is not an optimum value, since experiments have not been conducted for multiple gaps. However, this is a valid value because the prototype performs without problems and can absorb errors even when the dimensions are ten times larger. This leads to eq. (3): eq. (3) is changed to the upper line in Fig. 9, but when the module is made larger than 0.5, it changes so that the area expands, thus Fig. 9 can be used.

$$z_1 < \frac{z_0 \sin\frac{\pi}{4} - 2 - \frac{0.3}{m}}{\sin\frac{\pi}{4} + 1} \tag{3}$$

### About torque loss

Torque loss values were evaluated instead of transmission efficiency. In this two-way clutch, the gear is only used for locking. The torque was not transmitted by the tooth surface, but by the pin that pushed the linkage. This means that losses at the tooth surface are expected to be small and constant, regardless of the input torque. Therefore, we have omitted the discussion on transmission efficiency.

As shown in Figs. 12 and 13, the torque loss of this two-way clutch was nearly constant at less than 5 mNm both statically and dynamically, which was sufficiently low compared to the existing clutches [14] shown in the "background" section. Furthermore, as shown in Fig. 14, the torque loss of this system was about 10 mNm even under high-speed rotation. This indicates that the gears were not used to transmit power in this mechanism and theoretically, the force applied to the tooth surfaces is small. Therefore, the wear of the gears was minimized. On the other hand, the torque loss increased when the rotational speed increased because the centrifugal force increased the force that pushed the teeth. The spur gears used in the prototype were self-made, and low torque loss was achieved even though the accuracy of the gears was not good. This means that this mechanism did not require fine precision in the gears.

In Fig. 12, the slight decrease in torque loss around the load of 60 mNm was due to the decrease in rotation speed. In this experiment, a constant voltage was applied to the motor, so the rotation speed decreased as the load became higher. Fig. 13 shows that the torque loss depended on the rotation speed. Therefore, the torque loss decreased as the rotation speed decreased with a higher load.

### Time response of rotation

The premise is that responsiveness is not an issue with normal two-way clutches. This is because two-way clutches are obviously not responsive owing to backlash and are not used in devices that require feedback control. On the other hand, this two-way clutch will finally be introduced into the transmission shown in the "Background" section. Responsiveness is not a problem when this transmission is used in machines that rotate continuously in one direction, such as the winches and wheel axles of mobile robots [21]. However, when it is used in other applications, such as a robot joint, responsiveness becomes a problem owing to feedback control [22].

The rotational time response was measured by attaching rotary encoders (NEMICON, SBY-30-2MC) to both ends of the two-way clutch and rotating the input shaft with a motor(FAULHABER, 2342S024CR). The motor was driven by a ±2.4 V square wave (period 0.5 s). The experimental results are shown in Fig. 16, where (a) and (b) show the input voltage and rotation speed, and the rotation angle and backlash of the shaft, respectively. Both horizontal axes represent time. In this experiment,

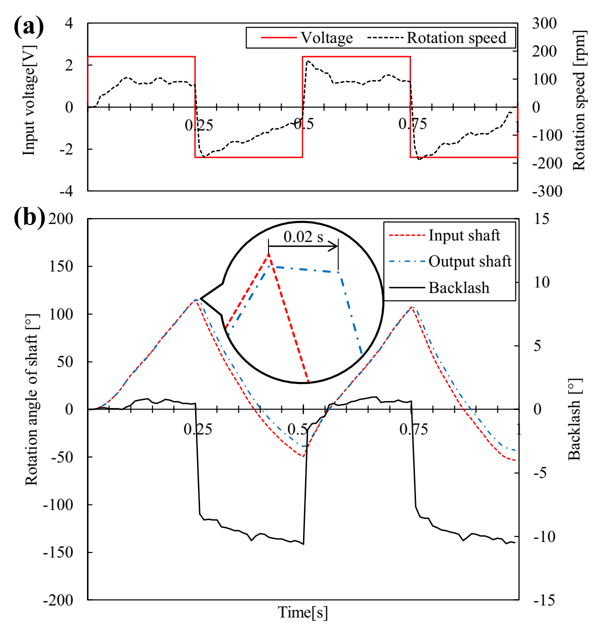


**Fig. 16** Time response between input and output rotation of a two-way clutch.The motor was driven by a ±2.4 V square wave (period 0.5 s)

the backlash at startup was set to zero; therefore, a large backlash occurred when switching the rotational direction. The absolute value of the maximum backlash is 10.62 °, which is consistent with the design value (10.69 °). This backlash causes a 0.02-second delay in the output shaft as shown in Fig. 16b. In the proposing transmission, the reduction gear will be placed ahead of the two-way clutch; therefore, the effect of backlash at the output shaft may be reduced.

### Disadvantages of using gears

There are several disadvantages of using gears. For example, they generate noise, require grease, the gear model does not match after heat generation, and backlash occurs. These issues are discussed in comparison with roller types. First, the noise and deformation after heat generation are not gear-specific problems. Rather, we consider that these problems are improved over the roller-type owing to smaller friction. Second, we considered the required grease to be an advantage, because the roller type cannot use grease, which shortens its lifetime. Finally, backlash was the most significant disadvantage. With the roller type, there was a slight backlash to disengage roller contact. The roller-type clutch developed by Controzzi et al. was shown to have a backlash of approximately 6 ° under certain design conditions [11]. With the gears, it increases to 10.69 °. This backlash causes deterioration in the responsiveness of the output shaft, as shown in Fig. 16. However, as mentioned above, this system can be placed in front of the reducer because it can turn at a higher speed than the roller-type, and the effect of backlash at the output shaft can be reduced. In this section, we describe how to reduce the backlash.

As described in Planetary gear mechanism and locking principle section, the backlash in this two-way clutch was generated to disengage the gears. Therefore, its magnitude was proportional to the height of the teeth and inversely proportional to the distance from the center of rotation to the meshing position. There were two possible ways to reduce the backlash: The first method was to minimize the size of the gear module. For instance, if the module was reduced to 0.25 while maintaining the external shape of the prototype, the number of teeth of the internal gear was changed to 120 and the number of teeth of the spur gear to 20. Since the height of the teeth was 1/2 and the distance between the centers of the gears was 1.35 times, the backlash was reduced to less than 0.37 times by simple calculation. However, the disadvantage was that the strength of the teeth was reduced. The other method was to use the profile-shifted gear. The height of the tooth was reduced by forward shifting the gear. Therefore, the backlash was reduced. At the same time, the dedendum was strengthened. The disadvantage was that it was costly.

### Relationship between overrunning and unlocking torque

According to Controzzi et al., in the case of passive two-way clutches, the optimum ratio of locking force to unlocking force is 1 [1]. If the ratio is greater than 1—where the output torque is greater than the input torque—overrunning may occur. If the ratio is less than 1, extra torque is required for control. In this mechanism, Fig. 15 shows that the ratio is less than 1. The error from the optimum value is considered to include the effect of the torque loss obtained in the experiment and the frictional force acting on the tooth surface shown in Fig. 17. In this mechanism, unlocking means disengaging the meshing of gears. When the gears were disengaged, a frictional force was generated on the tooth surface to inhibit the disengagement. Since this friction force varied depending on the condition of the teeth, it was thought that the measurement results of the prototype machine, which were not accurate, varied greatly. However, there was no danger of overrunning because the output shaft load torque would never exceed the unlocking torque under any load condition.

## Concluding remarks

In this study, we proposed a new two-way clutch mechanism using the meshing of an odd number of gears. The features of this mechanism were as follows.

- Low torque loss and high-speed rotation.
- Easy to fabricate because the structure was simple and high accuracy was not required for the parts.
- No overrunning occurs because the unlocking torque was greater than the load torque.
- Enough performance could be obtained regardless of the accuracy.

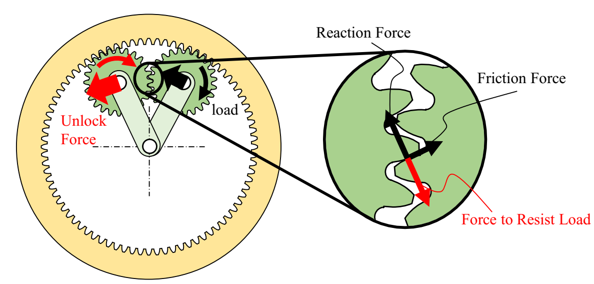


**Fig. 17** Increase in unlocking force by frictional force acting on the tooth surface

When using this system in a transmission under development and control is applied to the reciprocating motion, a backlash problem occurs. In such cases, some control method considering this backlash should be introduce.

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**Author contributions**
MW invented the proposed two-way clutch mechanism and confirm it by the examinations. TT refined the proposed mechanism and improved the quality of principles. Both authors read and approved the final manuscript.

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**Availability of data and materials**
The datasets used and/or analysed during the current study are available from the corresponding author on reasonable request.

## Declarations

**Competing interests**
The authors declare that they have no competing interests.

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