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Johnson

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(54) **ONE-WAY CLUTCH**

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CPC *F16D 41/07* (2013.01); *F16D 41/063* (2013.01)

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CPC F16D 41/06; F16D 41/063; F16D 41/07; F16D 41/24; F16D 41/28; F16D 2041/0601; F16D 2041/0603
See application file for complete search history.

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Primary Examiner — Jacob S Scott

(57) **ABSTRACT**

A one-way clutch, including: a load rotor, a driving member, and at least one self-locking gripper. The load rotor is spool-shaped and includes two circular contact races, one race at each end of the rotor. The driving member includes a plurality of inward facing cams on an inner circumference arranged to rotate co-axially with the load rotor. The cams rotate within the space separating the two load rotor races. When in a locking position each gripper presses against both load rotor races and a cam over area-to-area contacting surfaces. Rotation of the load rotor and driving member are locked in the first direction when one or more grippers are in the locking position. The driving member is rotatable independent of the load rotor in the second, free-wheeling direction. During free-wheeling rotation centrifugal force acts to disengage the grippers from the load rotor, thus reducing or eliminating friction.
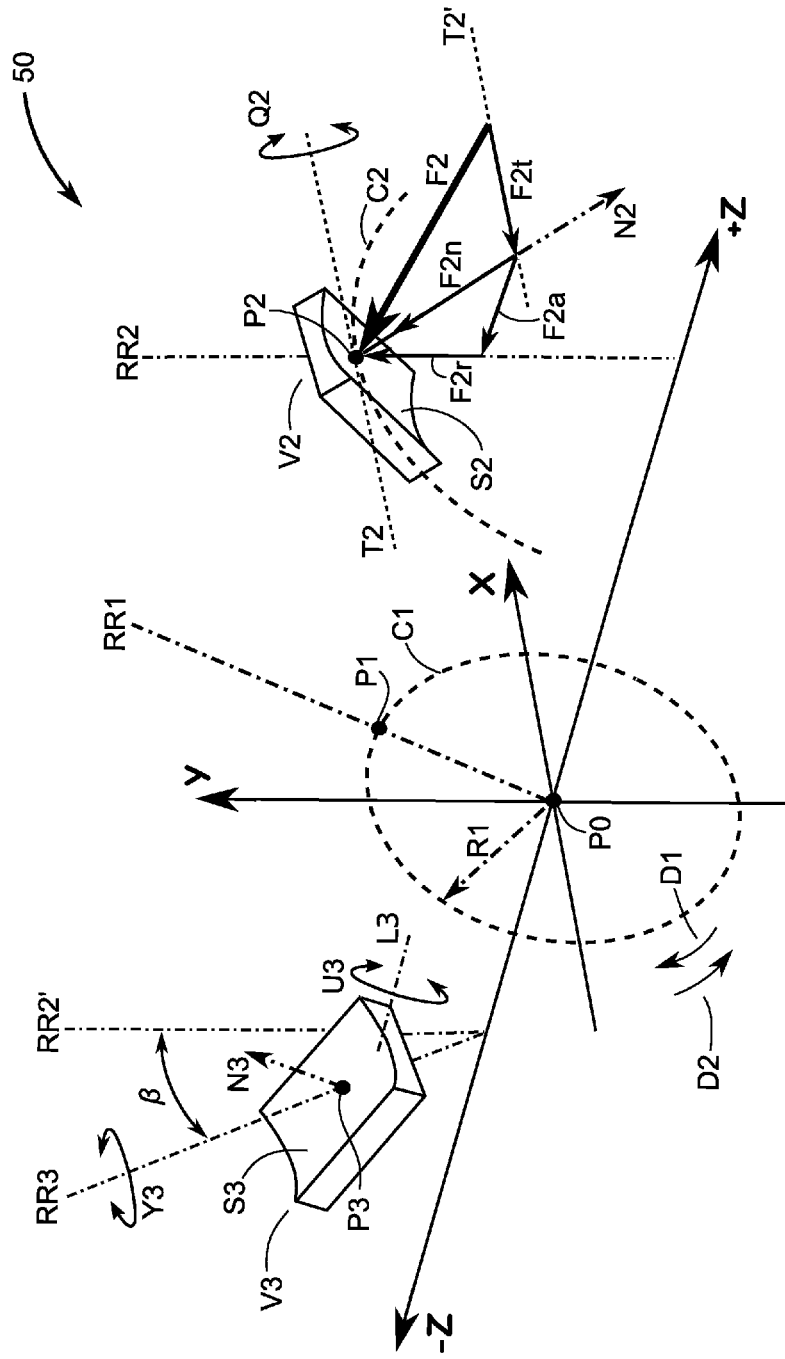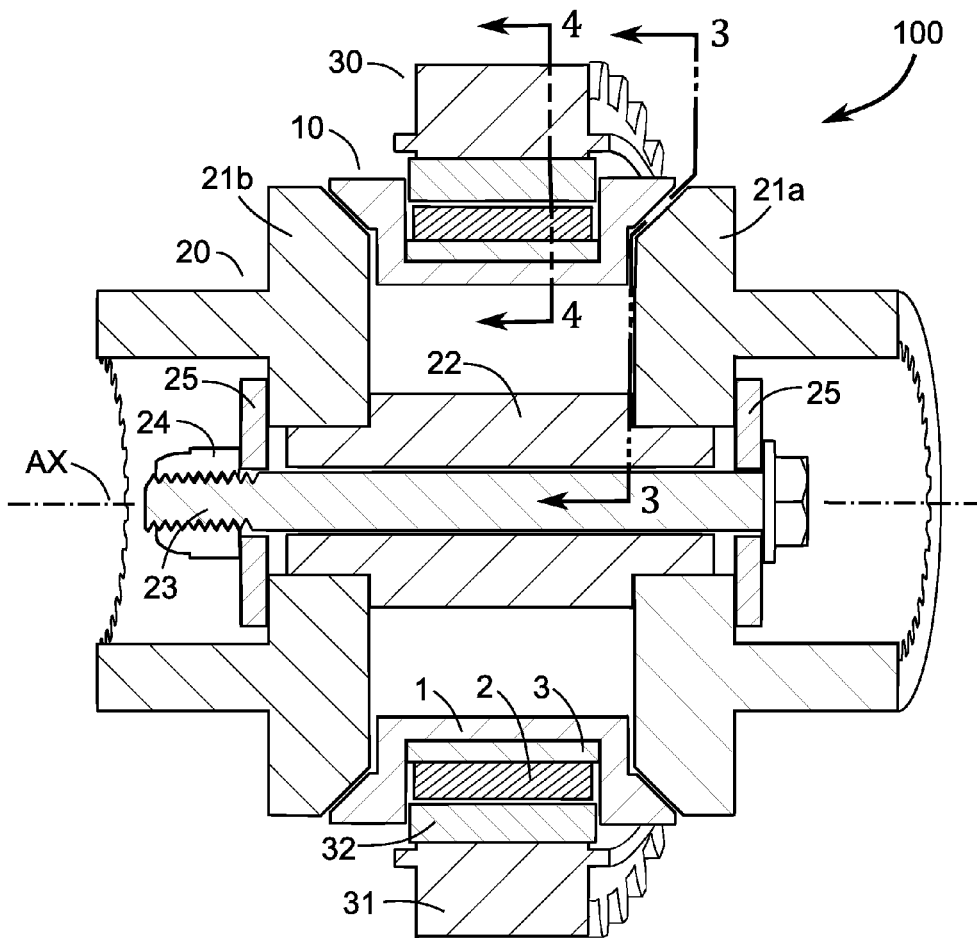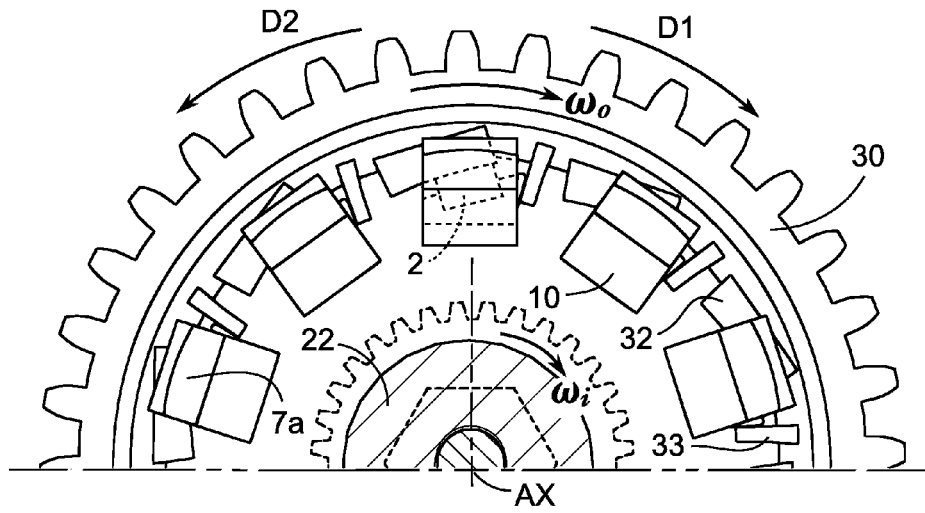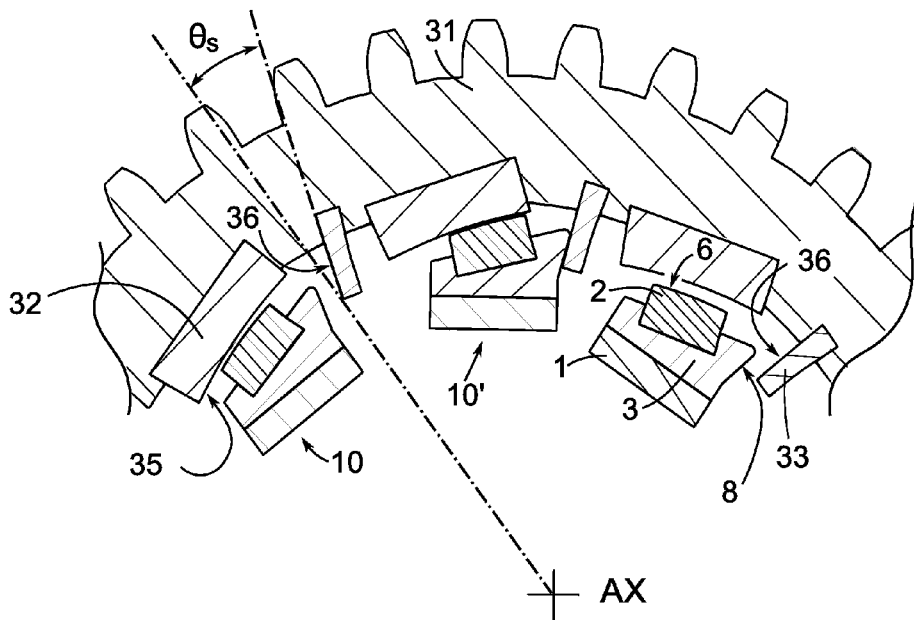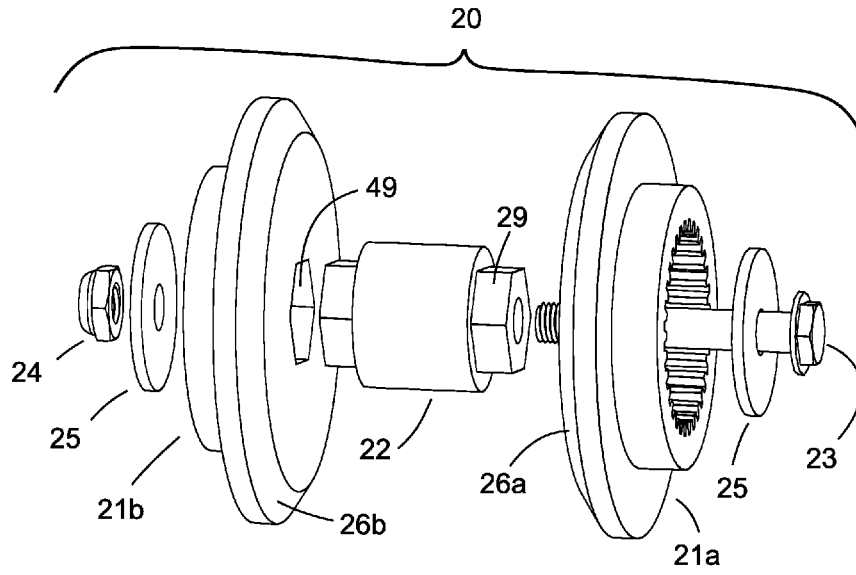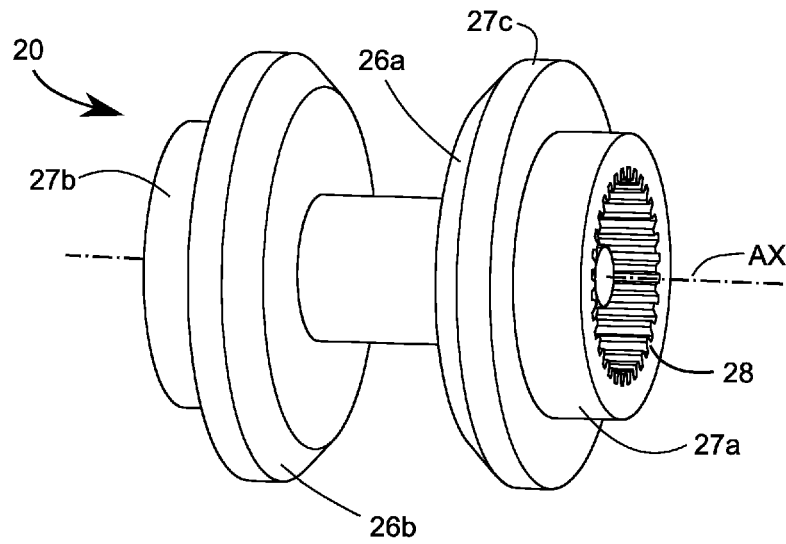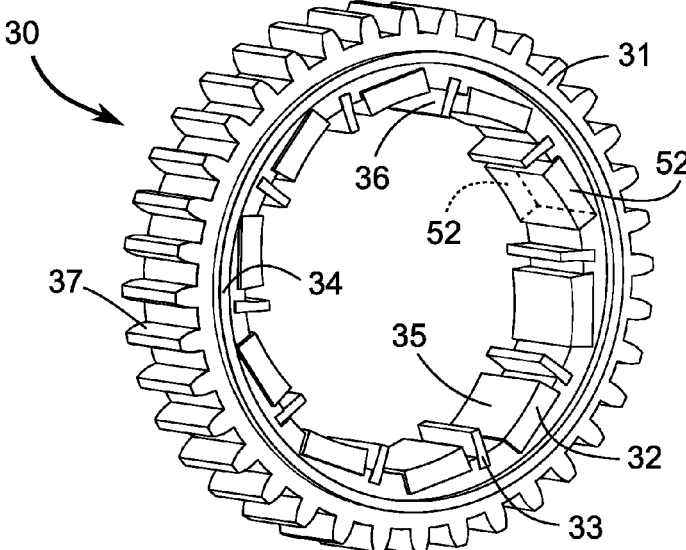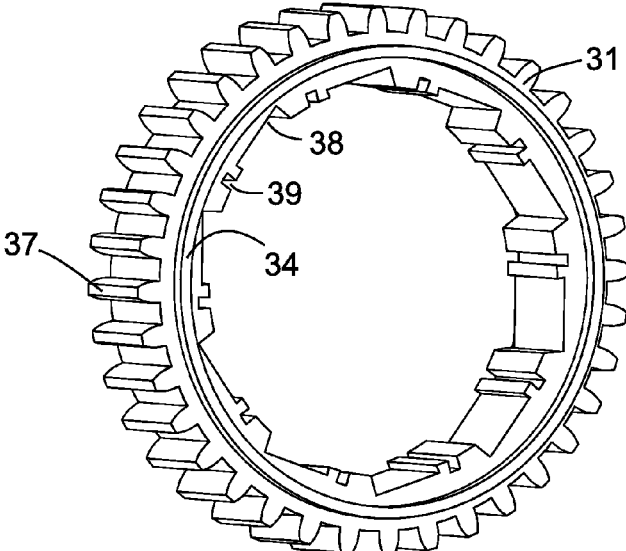
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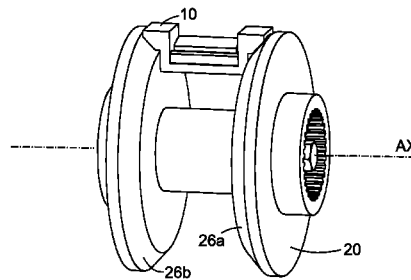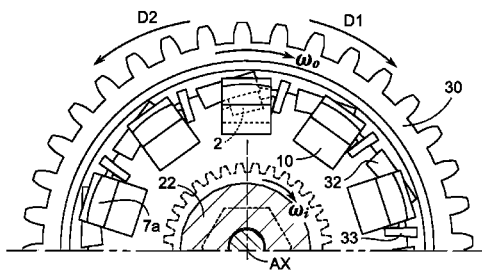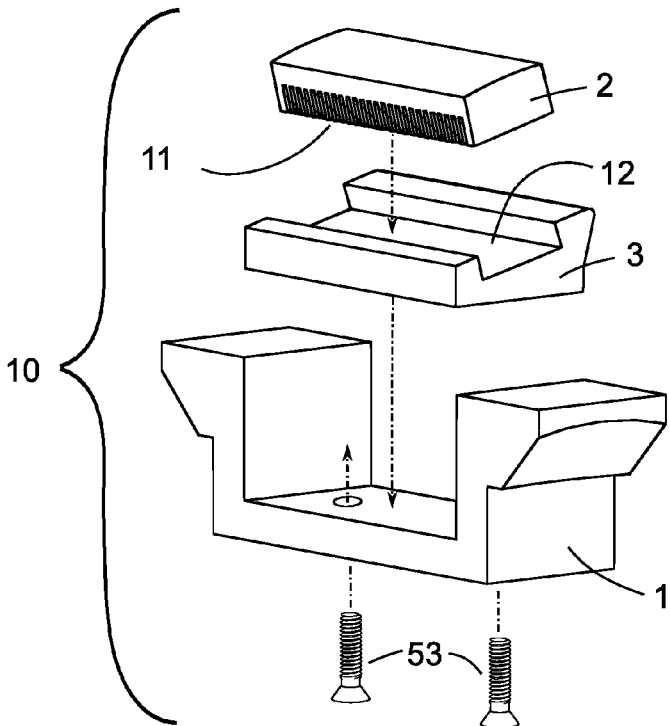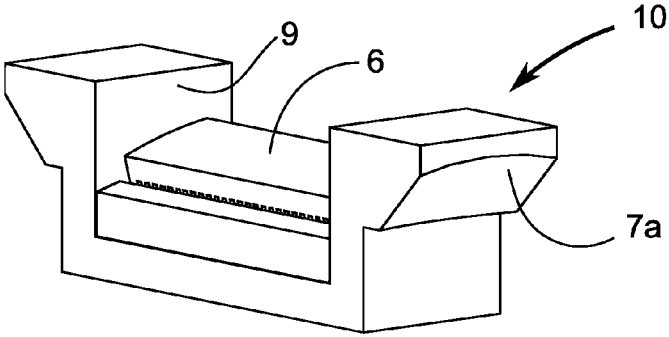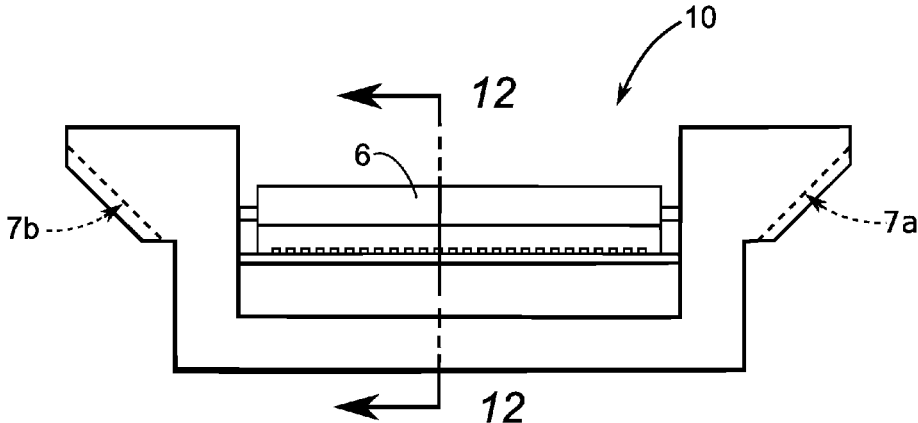
FIG. 1

FIG. 2

FIG. 3
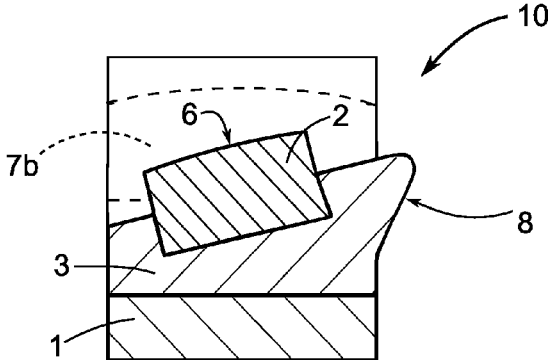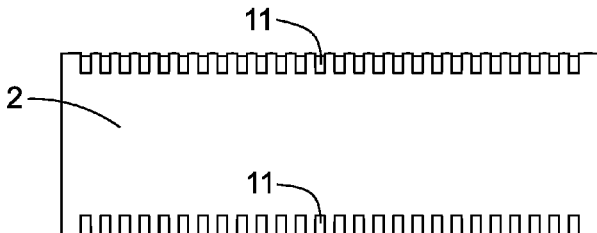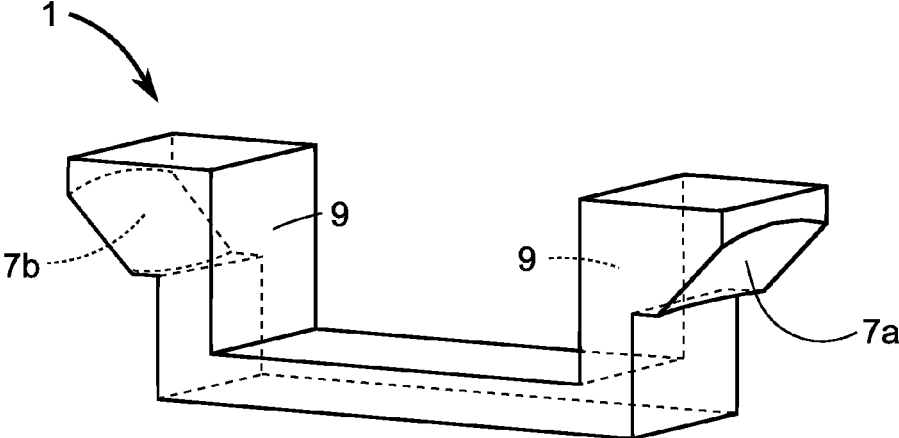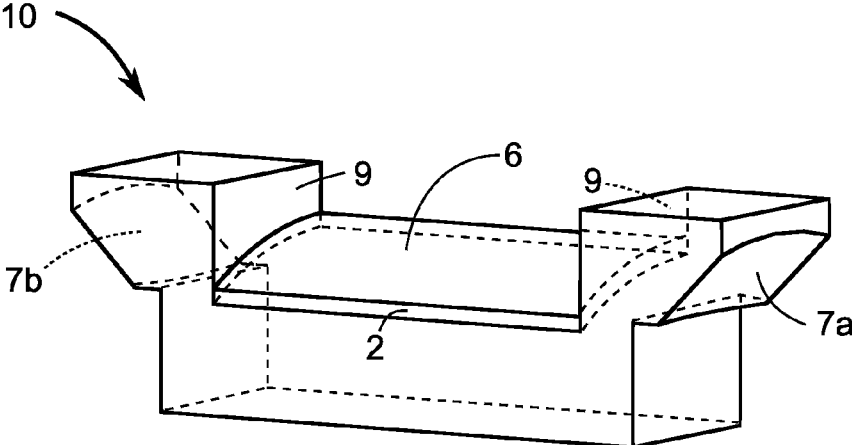

FIG. 4

FIG. 5

FIG. 6
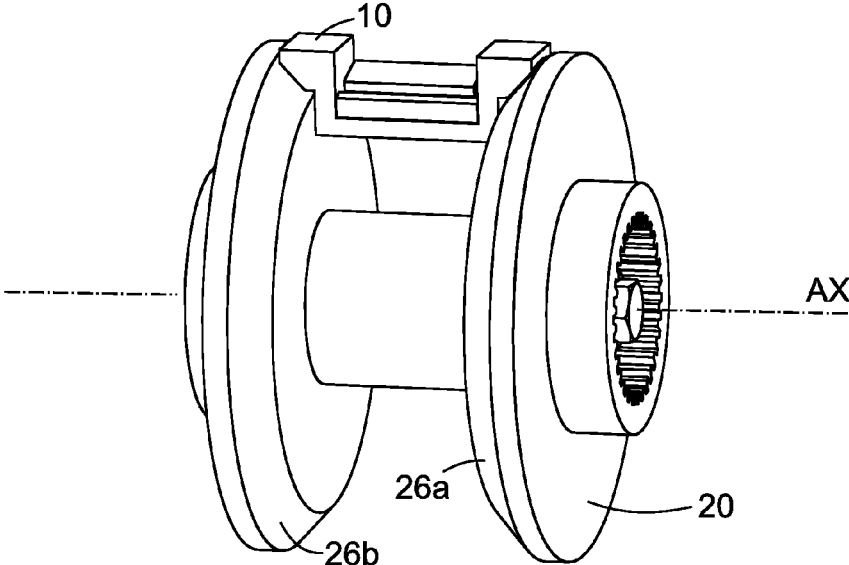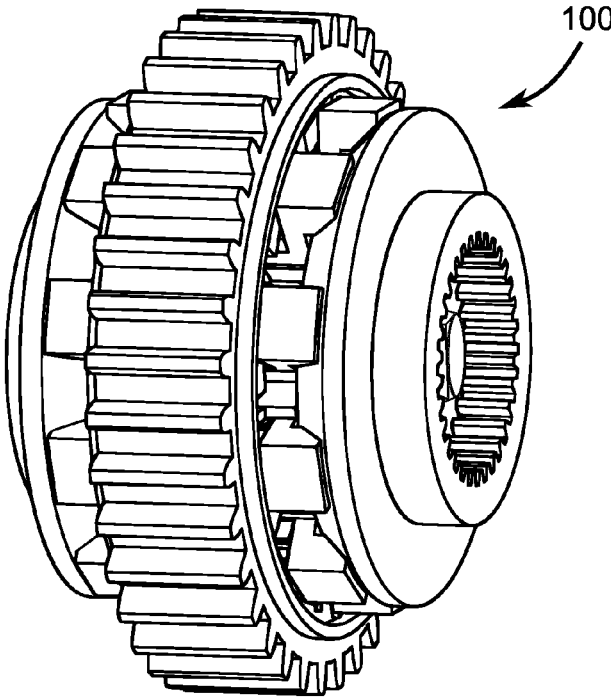
FIG. 7

FIG. 8

FIG. 9

FIG. 10

FIG. 11

FIG. 12

FIG. 13

FIG. 14

FIG. 15

FIG. 16

FIG. 17
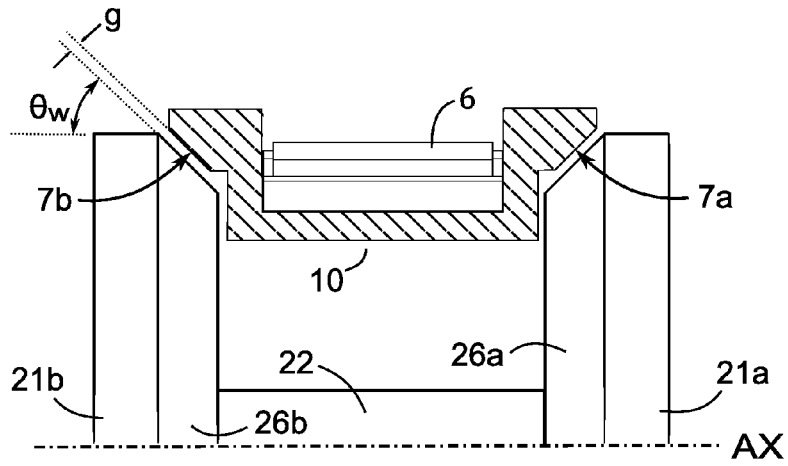
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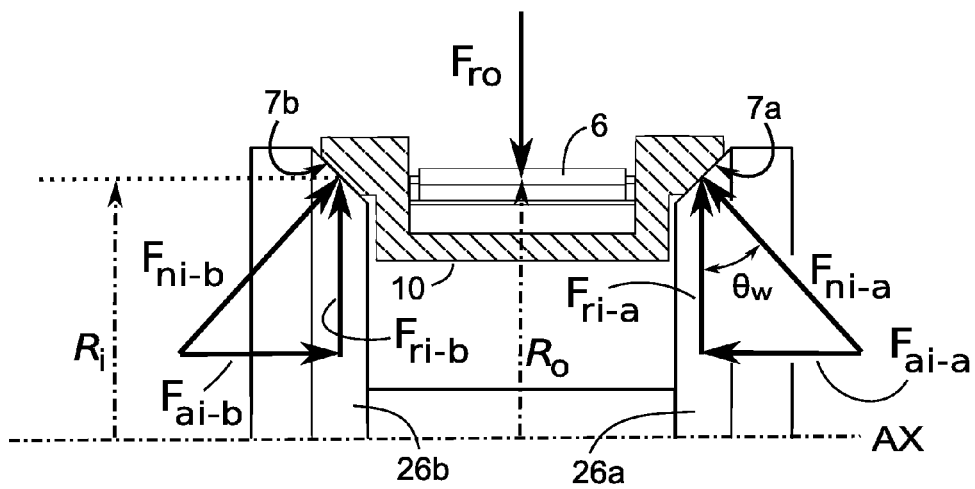
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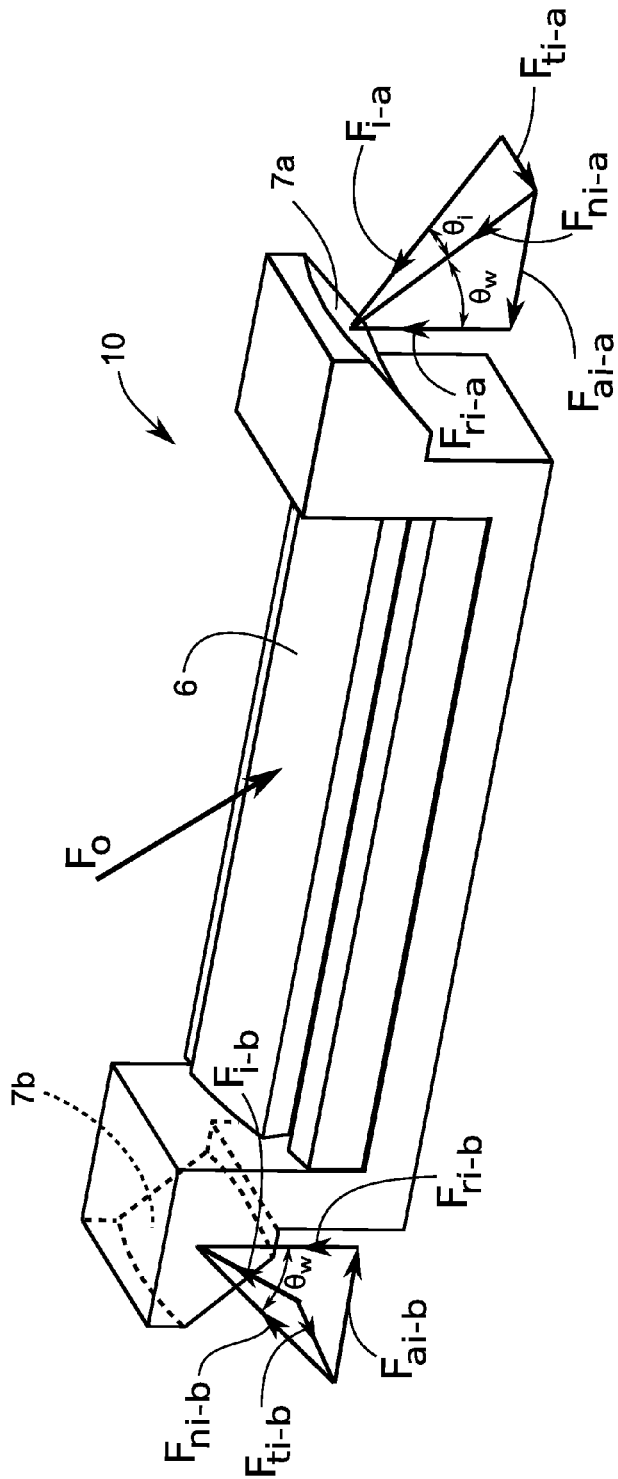
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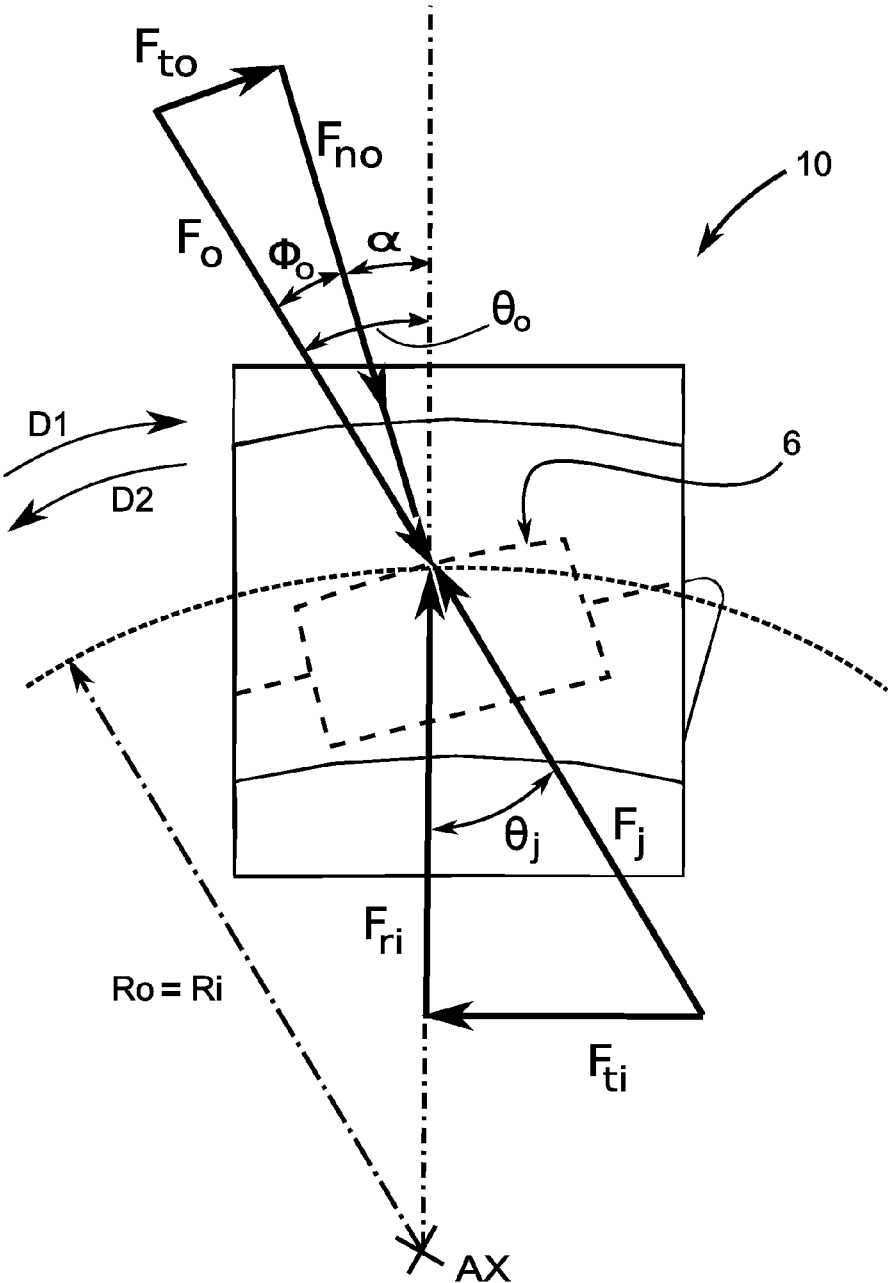
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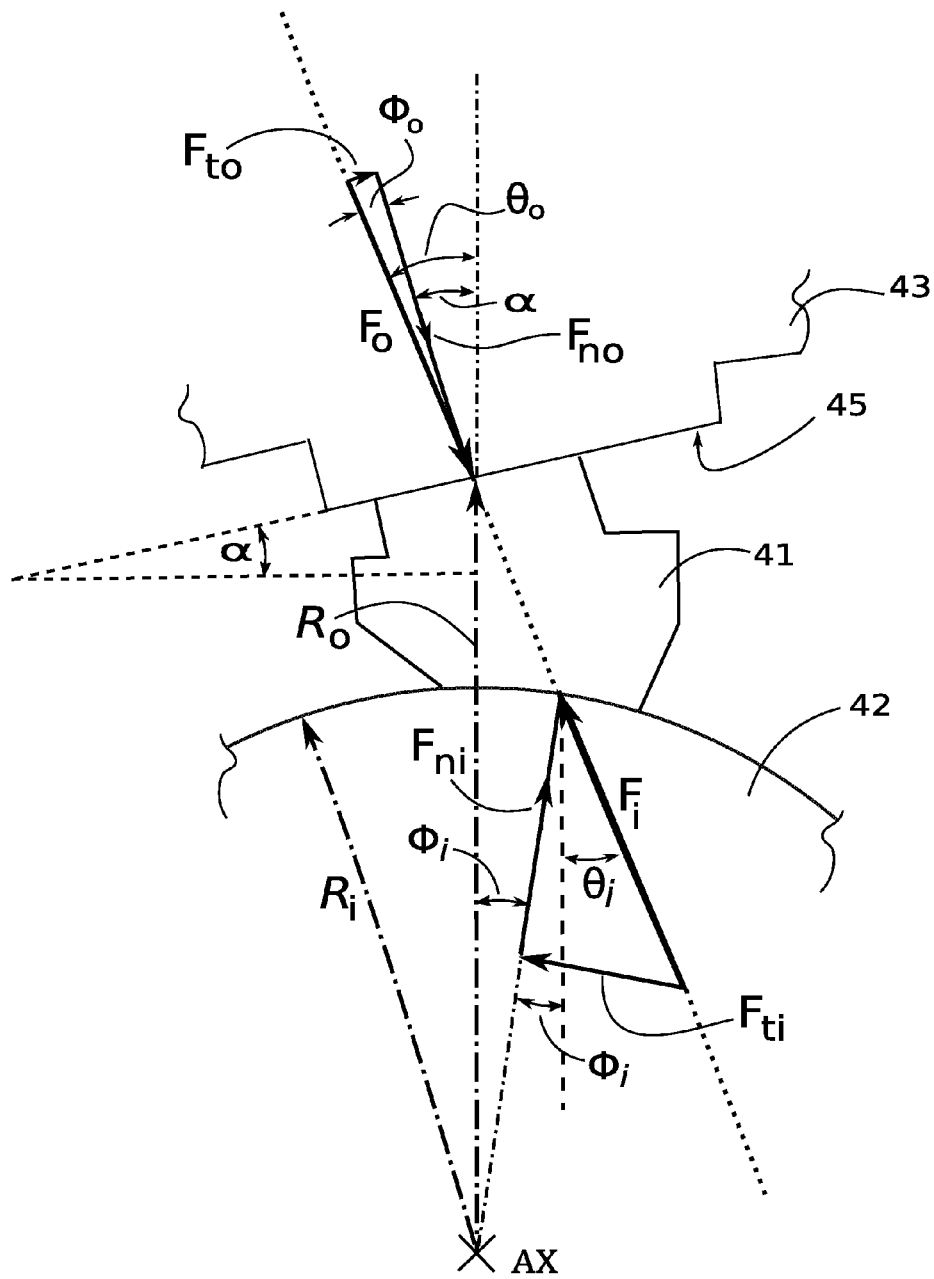
(Prior Art)

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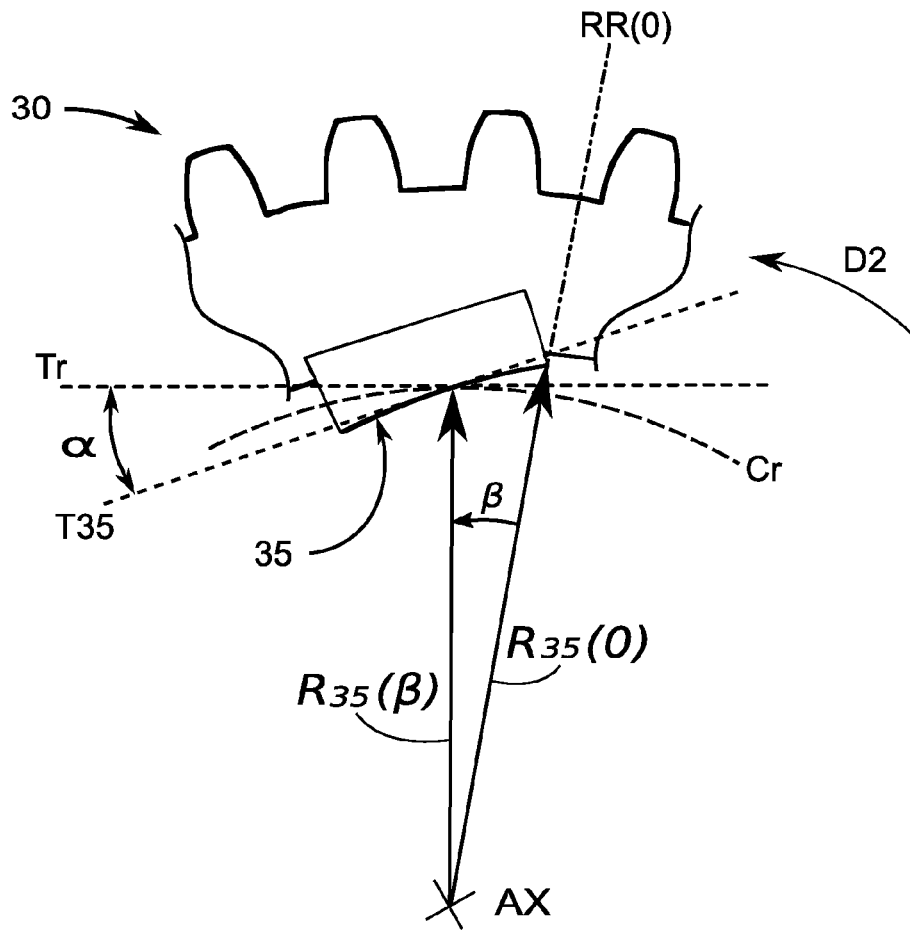
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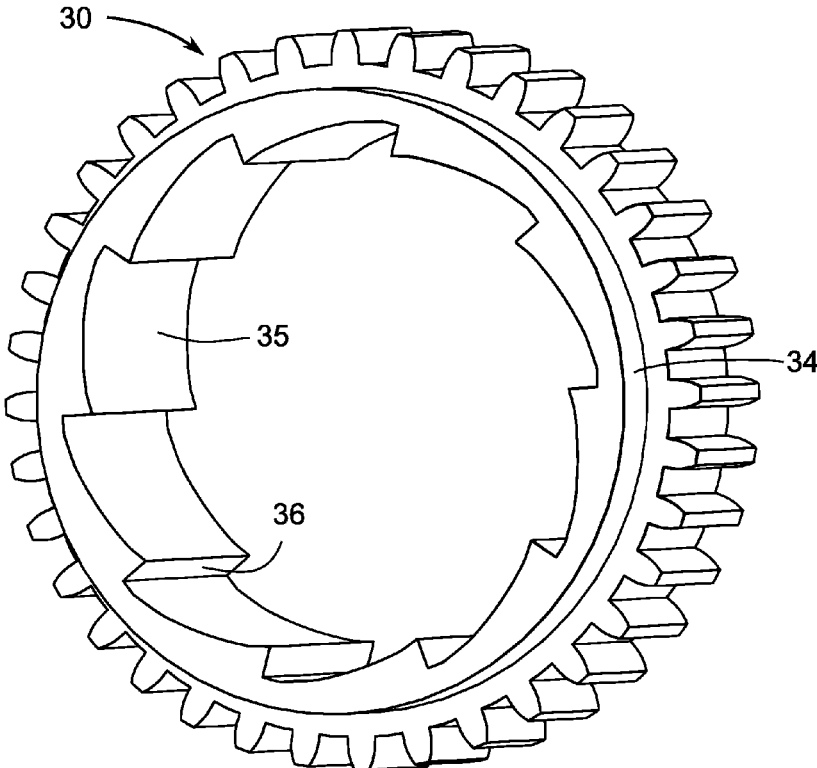
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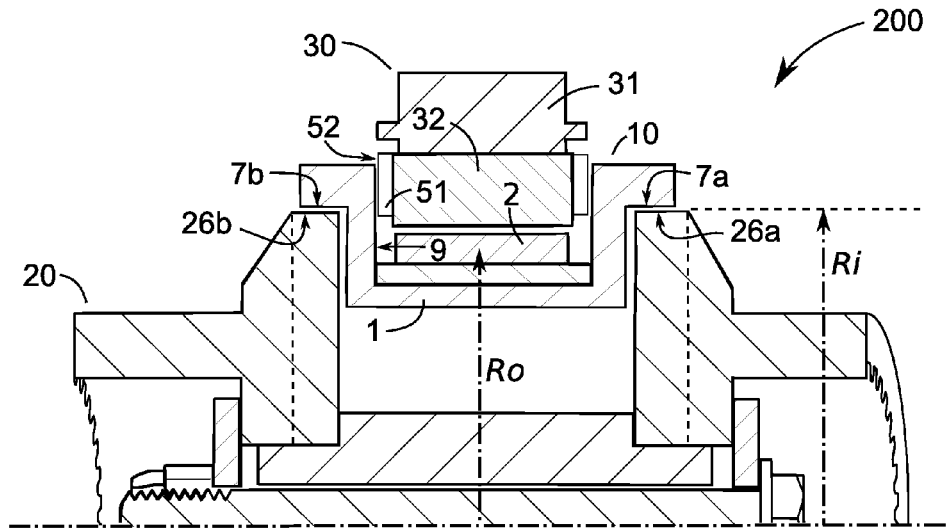
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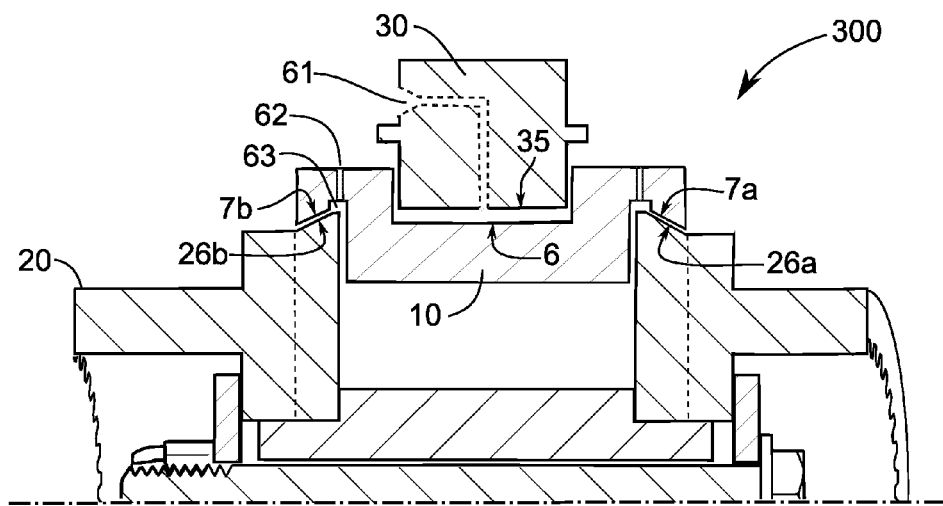
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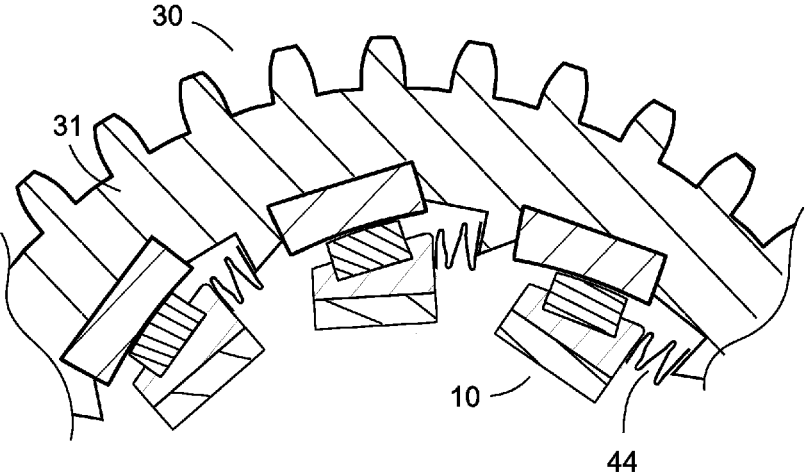
FIG. 27
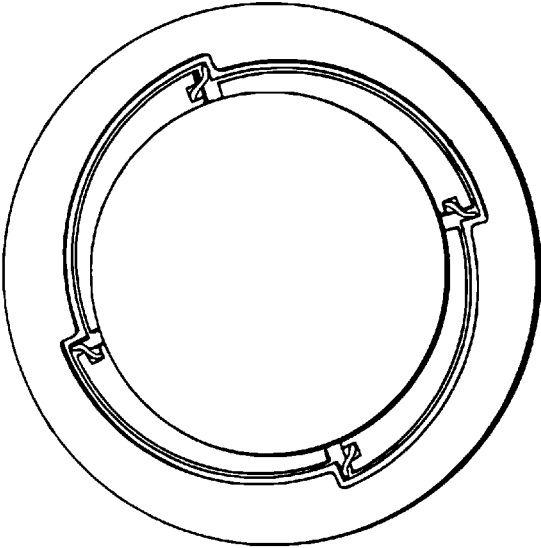
(Prior Art)

FIG. 28
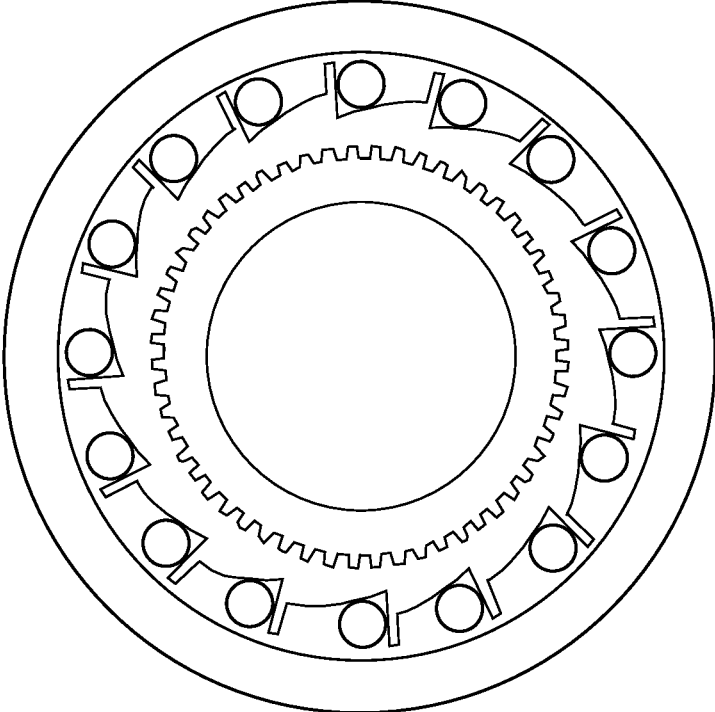
(Prior Art)

FIG. 29

ONE-WAY CLUTCH

TECHNICAL FIELD

The present invention relates generally to a one-way 5
clutch, and more specifically to a one-way clutch capable of
sustaining a high number of high torque lock-and-unlock
cycles while operating with reduced freewheeling friction.

BACKGROUND 10

Roller-ramp and sprag clutches are one-way clutches that
operate automatically and are capable of supporting high
torque loads. However, metal fatigue limits the cycle life of
roller-ramp and sprag clutches in high torque uses. The ramp 15
angle in roller-ramp clutches and the strut angle in sprags
must be shallow to achieve locking action. These shallow
angles produce very high compressive force on the sprags or
rollers, also called "grippers." Because the force on these
grippers is focused along narrow line contacts, high torque 20
loading causes extreme pressure at the contacts. As a result,
even the hardest steel alloys suffer metal fatigue after some
number of high torque lock and release cycles. Cycle
lifetimes of current technologies range from a few hundred
thousand to a few million high torque cycles. 25
An advantage of roller-ramp and sprag clutch technolo-
gies is that the gripper elements are "self-locking." That is,
once friction contact is established the grippers lock auto-
matically without need of any external forces or mecha-
nisms. Biasing springs are commonly present in these 30
clutches but only function to initiate friction contact between
the grippers and the surfaces of the clutch races. When used
in an application in which inertial forces reliably initiate
friction contact no springs are required for the function of
self-locking grippers. The Langen Overrunning Clutch (U.S. 35
Pat. No. 67,659 August 1867), shown in FIG. 29, is an
example of a roller-ramp clutch using no biasing springs.
Numerous attempts have been made to replace the rollers
and sprags of conventional one-way clutches with wedge
shaped gripper elements so as to increase the contact area 40
and thereby decrease pressure and stress, thus minimizing
metal fatigue. For example U.S. Pat. No. 8,020,681 includes
an embodiment with large contact surfaces on wedge shaped
grippers. But wedge grippers are not generally self-locking
and require springs to press the wedges against the ramp 45
surfaces to hold the torque load. The necessary spring force
for maintaining high torque grip produces exceedingly high
freewheeling friction. In addition, if the torque load momen-
tarily exceeds the spring force the wedges lose grip, pro-
ducing a dangerous runaway clutch failure. Runaway clutch 50
failures do not occur in a one-way clutch with self-locking
grippers.
U.S. Pat. No. 3,202,250 by Bertram Fulton discloses a
wedge clutch with self-locking grippers. The patent reveals
that application of a low-friction coating at the ramp surface 55
can make a wedge-shaped gripper self-locking if the ramp
angle is sufficiently small. FIG. 28 shows an embodiment of
Fulton's disclosure in which the ramp surfaces are placed on
the outer race for the expressed purpose of reducing free-
wheeling friction. With radially inward facing ramps cen- 60
trifugal force urges the wedges away from the inner race
during freewheeling rotation, reducing friction or even lift-
ing the wedge off the counter-rotating race. The following
friction analysis of Fulton's model is necessary for disclo-
sure of the present invention's innovations. 65
FIG. 22 is a diagram of the forces acting on wedge 41 in
a clutch of the same configuration as the model in FIG. 28.

Fo is the sum of the pressure on the wedge from contacting
ramp surface 45 of outer race 43. Fi is the sum of the
pressure on the wedge from contacting circular inner race
42. During the locked state Fi and Fo must be equal in
magnitude, opposite in direction and co-linear, and therefore
angle θo is equal to θi. To produce self-locking action the
wedge must slip at the ramp surface when the wedge slips
against the inner race. For the wedge to slip at the ramp
surface the ratio Fto/Fno must be greater than the coefficient
of friction μo at the ramp, where Fto is the frictional
component and Fno is the normal component of force Fo.
Therefore angle ϕo must be greater than the friction angle
$\arctan(\mu o)$ for locking action to occur. During slip at the
inner race the ratio Fti/Fni is the kinetic coefficient of
friction μi, where Fti is the frictional component and Fni is
the normal component of Fi. Therefore the sum of angles
$(\theta i + \phi i)$ is equal to friction angle $\arctan(\mu i)$. It can be shown
that combining these conditions leads to the following
requirement for self-locking action:

$$\arctan(\mu o) < \arctan(\mu i) - \alpha - \phi i \qquad (1)$$

where α is the slope angle of the ramp. The larger the
magnitude of ϕi the more difficult it is to comply with
condition (1). But ϕi increases as the ratio Ro/Ri increases,
where Ro is the rotational radius of Fo and Ri is the
rotational radius of Fi. That is, the magnitude of ϕi increases
with the thickness of the wedge. In practice ϕi often reaches
a value that makes satisfying requirement (1) unattainable,
or attainable for only small values of α. Therefore wedge-
clutch designs with outer race ramps use thin spiral wedges
and very shallow ramp angles. However a shallow ramp
angle is known to lead to "lock up," a state in which a wedge
permanently jams between the races. Substantial force may
be necessary to free up a locked up wedge. Additionally,
wedge-clutch designs typically specify a circular ramp cur-
vature or leave the ramp curvature unspecified. But a cir-
cular or undefined ramp curve does not distribute pressure
evenly along the contacting surfaces, especially in spiral-
type ramps. Instead these curvatures focus most of the
compressive force on a small section of the wedge, leading
to lock ups and fatigue processes.
Placement of the ramps on the inside race in a wedge-
clutch design reverses the effect of the ϕi term in equation
(1) and makes self-locking action more easily attainable.
Most all wedge-clutch designs therefore use inner race
ramps. But inner race ramps put the wedges in constant
friction contact with the outer race during freewheeling
rotation. This effect generates undesirable wear and friction,
especially during long periods of high speed counter-rota-
tion. High speed freewheeling is also known to cause
freewheeling lockup in operation of inner ramp wedge-
clutch designs. U.S. Pat. No. 9,016,451 discloses an inner
ramp wedge-clutch with reduced freewheeling friction by
use of a spring mechanism built into a wedge ring. Though
reduced the design still requires significant freewheeling
friction to initiate locking action and is still subject to
freewheeling lockup. Other designs, for example U.S. Pat.
No. 9,353,802, use an external actuator to disengage the
wedges from the outer race during freewheeling operation.
These designs do not operate automatically, however, and
require additional and complex external mechanisms for
operation of the clutch.

BRIEF SUMMARY OF THE INVENTION

The present invention broadly comprises a one-way
clutch including a driving member, a load rotor, and at least

one gripper element. The load rotor includes two radially outward facing circular contact races and an inner shaft. The load rotor inner shaft and contact races are arranged for rotation about the axis of rotation of the clutch, with the two contact races separated at a fixed longitudinal distance and rigidly connected to the inner shaft. The rotational radius of the load rotor inner shaft is smaller than the radius of the contact races, forming a spool-shaped rotor. The load rotor is arranged for rotational connection to a torque load in most embodiments, or to a power source in others.

The clutch driving member includes an inner circumference with a plurality of radially inward facing cam surfaces on the inner circumference. The driving member is arranged co-axially to the load rotor with the cam surfaces rotatable around the load rotor inner shaft and within the longitudinal distance separating the two load rotor races. A space is provided between the driving member inner circumference and the load rotor inner shaft and races. The driving member is arranged for rotational connection to a source of torque in most embodiments, or to a rotating load in others.

The clutch gripper elements are disposed in the space provided between the load rotor and the driving member. Each gripper includes three contact surfaces: two gripping surfaces and one slipping surface, with the slipping surface disposed between the two gripping surfaces. The gripper element contact surfaces are arranged to provide a locking position in which one gripping surface presses against one load rotor race, the second gripping surface presses against the other load rotor race, and the slipping surface presses against a driving member cam surface. The driving member includes a stop for each gripper element. Movement of the grippers relative the driving member in the first rotational direction is limited by the stops. In some embodiments the stops are made of an impact absorbing material; in other embodiments the stop includes an elastic element, such as a spring.

When in a locked mode of operation at least one gripper element is disposed in its locking position. In locked mode torque is transferred from the driving member to the load rotor through the gripper element(s) and rotation of the driving member and load rotor are locked in the first rotational direction. In a freewheeling mode of operation the driving member is free to rotate independently relative the load rotor in the second rotational direction, opposite the first direction of rotation. During freewheeling rotations of the driving member centrifugal forces urge the grippers to move radially outward, acting to disengage the grippers from the load rotor races.

Some embodiments of the present invention include aligning surfaces on the driving member and gripper elements arranged to limit motion of the grippers in the longitudinal, pitching and yawing directions during clutch operation. In some embodiments the curvature of the driving member cams follows a logarithmic spiral.

In the present invention the rotational radius of the forces at the clutch driving member cams, Ro, is adaptable in relation to the rotational radius of forces at the load rotor races, Ri. Specifically the ratio Ri/Ro can be made to approach or exceed unity. Friction physics discloses that increasing the ratio Ri/Ro also increases the cam surface ramp angle at which self-locking action of the clutch grippers can be achieved. Increasing the cam surface ramp angle decreases pressure and stress at the clutch contact surfaces, thereby increasing fatigue life and preventing clutch lock-ups. Some embodiments of the present invention include a means for decreasing friction at the interface between the gripper slipping surfaces and driving member cams, thereby

further increasing the attainable cam surface ramp angle. Low-friction materials such as polyfluorotetraethylene are used in some embodiments and lubricating grease or oil is used in others. In some embodiments of the present invention the load rotor races are beveled at an angle to the axis of rotation so as to form a conical surface, a method which further acts to increase the attainable cam surface ramp angle.

BRIEF DESCRIPTION OF THE DRAWINGS

Various embodiments are disclosed, by way of example only, with reference to the accompanying schematic drawings in which corresponding reference symbols indicate corresponding parts, in which:

FIG. 1 is a perspective view of coordinate system 50, demonstrating the spatial terminology used in the disclosure;

FIG. 2 is a sectional view of an exemplary embodiment of the invention, clutch 100, with the section taken along the rotational axis of the invention;

FIG. 3 is a sectional view of clutch 100 taken along the line 3-3 of FIG. 2;

FIG. 4 is a partial sectional view of clutch 100 taken generally along the line 4-4 of FIG. 2;

FIG. 5 is an exploded view of load rotor 20 of clutch 100;

FIG. 6 is a perspective view of load rotor 20 of clutch 100;

FIG. 7 is a perspective view of driving member 30 of clutch 100;

FIG. 8 is a perspective view of main gear 31 of clutch 100;

FIG. 9 is an exploded view of gripper element 10 of clutch 100;

FIG. 10 is a perspective view of gripper element 10 of clutch 100;

FIG. 11 is a longitudinal view gripper element 10 of clutch 100;

FIG. 12 is a sectional view of a gripper element 10 of clutch 100 taken along line 12-12 of FIG. 11.

FIG. 13 is a bottom view of pad 2 of gripper element 10 in which the extent of expansion slots 11 is visible;

FIG. 14 is a perspective view of cradle 1 of gripper element 10;

FIG. 15 is a perspective view of an alternative embodiment of gripper element 10;

FIG. 16 is a perspective view of load rotor 20 and a single gripper element 10, with the gripper element located in its locking position;

FIG. 17 is a perspective view of clutch 100;

FIG. 18 is schematic longitudinal view of load rotor races 26a, 26b, and gripper element 10 with the gripper located in a recessed position;

FIG. 19 is a schematic longitudinal view of load rotor races 26a, 26b, and gripper element 10 in its locking position, with a diagram of the forces acting upon the gripper projected onto the illustration;

FIG. 20 is a perspective view of gripper 10 with a diagram of the forces acting upon the gripper projected onto the illustration;

FIG. 21 is a front view of gripper 10 with a diagram of the forces acting upon the gripper projected onto the illustration;

FIG. 22 is a schematic front view of a prior art one-way wedge clutch with a diagram of the forces acting upon the wedge projected onto the illustration;

FIG. 23 is a schematic front view cam surface 35 depicting the variables used to calculate the curvature of the surface;

FIG. 24 is a perspective view of an alternative embodiment of driving member 30;

FIG. **25** is longitudinal sectional view of an alternative embodiment of the present invention, clutch **200**;

FIG. **26** is longitudinal sectional view of an alternative embodiment of the present invention, clutch **300**;

FIG. **27** is a partial front sectional view of an alternative embodiment of the present invention illustrating the inclusion of elastic elements **44**;

FIG. **28** is a front view of a prior art one-way wedge clutch;

FIG. **29** is a front view of a prior art one-way roller-ramp clutch;

DETAILED DESCRIPTION

At the outset, it should be understood that the drawings of the disclosure are schematic in nature and may not be to scale. Features such as proportions, sizes, lengths, spacing, widths, and gaps may be exaggerated or otherwise altered for the sake of clarity. For example the size of gap g in FIG. **18** will likely be much smaller in practice than depicted, and the length of gripper **10** as depicted in FIG. **11** and other figures may be longer or shorter relative to other features in testing and practice. Additionally, the number of grippers, stops, and cam surfaces of the disclosure is not limited to the number depicted in these drawings.

FIG. **1** is a perspective view of coordinate system **50** used for demonstrating the spatial terminology used in the present invention. System **50** has orthogonal axes X, Y and Z and also includes elements of a cylindrical coordinate system. Axis Z is the invention's axis of rotation and the term "longitudinal" refers to the direction of the Z axis. A plane containing the Z axis will be termed an "axial plane." The origin Po is located approximately equidistant between the invention's load rotor races (described later). Any plane that is parallel to the X-Y plane is designated a "radial plane." A radial is a line extending from the Z axis and lying on a radial plane. The term radial motion and radial distance will refer to motion and distance respectively along a radial. P1 is a point on the X-Y plane and RR1 is the radial of P1. Circle C1 is the path of P1 as it rotates about axis Z and is termed the "rotational circle" of P1. R1 is the radius of C1 and is called the "rotational radius" of P1. D1 is the first direction of rotation and D2 is the second direction of rotation.

To demonstrate the terminology used for describing the orientation of various forces, lines and surfaces FIG. **1** includes objects V**2** and V**3**. Point P2 lies on surface S2 of V**2**. Line N2 is the normal to surface S2 at point P2. Because the normals of S2 point away from the X-Y plane S2 is termed "longitudinally outward" facing. C2 is the rotational circle of P2 and N2. Because the normals of S2 are directed inside their rotational circles S2 is termed "radially inward" facing. Referring to object V**3**, N3 is the surface normal at point P3 of surface S3. Surface S3 is "longitudinally inward" facing and is "radially outward" facing. The terms "front" and "rear" will refer to the positive and negative directions of axis Z. For example, object V**2** is in front of object V**3**. F2 is a force impinging on surface S2 at point P2. Force F2n is the normal component of F2. That is, F2n is the component of F2 that is perpendicular to the surface at P2. Line T2 is the tangent of C2 at P2. Line T2' is tangent T2 translated to the tail of F2 for diagrammatic purposes. Force F2t is the component of force F2 parallel to T2 and will be called the rotational component of F2. Line RR2 is the radial at point P2. Force F2r is the radial component of F2 and lies on RR2. Force F2a is the component of F2 that is parallel to axis Z and will be called the longitudinal component of F2. Force

F2 is the sum of F2t, F2a and F2r. If F2n lies on an axial plane then F2 is also the sum of F2t and F2n. Referring to object V**3**, RR2' is the radial of P2 translated along the Z axis to the radial plane of P3. The angle between radials RR3 and RR2' will be called the rotational angle between point P3 and P2 and is marked β in FIG. **1**. Rotation of an object about a radial passing through the object will be termed a rotation in the "yaw" direction. Rotation Y3 of object V**3** is a yawing rotation. Rotation of an object about the tangent to a rotational circle passing through the object will be termed rotation in the "pitch" direction. Rotation Q2 of object V**2** is a pitching rotation. Rotation of an object about an axis parallel to the Z axis, but not the Z axis, will be termed a rotation in the "roll" direction. In FIG. **1** line L3 is a line parallel to axis Z that passes through V**3**. Rotation U3 of V**3** is a rolling rotation. An object that remains motionless in a frame of reference that is rotating about axis Z has no pitching, yawing or rolling motion in that frame of reference.

FIG. **2** is a longitudinal sectional view of an exemplary embodiment of the present invention, one-way clutch **100**, with the section taken along the axis of rotation AX. Following standard practice, like drawing numbers on different drawing views of clutch **100** and the drawing views of other embodiments of the invention will identify identical, or functionally similar, structural elements. Load rotor **20** of clutch **100** includes front disc **21***a*, rear disc **21***b* and inner shaft **22**. Driving member **30** includes main gear **31** and cam blocks **32**. Grippers **10** include cradles **1**, pad holders **3** and pads **2**. The load rotor is arranged for rotation about axis AX and is arranged for rotational connection to a torque load (not shown). The driving member is arranged to be rotatable co-axially to the load rotor and is arranged in a manner that limits longitudinal displacement in relation to the load rotor. The driving member is arranged for rotational connection to a source of power, that is, a source of rotating torque (not shown).

FIG. **3** is a sectional view of clutch **100** taken along line 3-3 of FIG. **2**. Grippers **10** are movable in the radial and rotational directions relative driving member **30**. The grippers are shown in their recessed position and the position of pad **2** is shown in hidden lines. When in the recessed position the grippers rest against both cam blocks **32** and stops **33**. Stops **33** limit the rotation of the grippers in the D1 direction relative the driving member.

FIG. **4** is a partial sectional view of clutch **100** taken along line 4-4 of FIG. **2**. Three grippers **10** are shown and the gripper marked **10'** is shown in its recessed position. Surfaces **36** of stops **33** contact surfaces **8** of grippers **10** when the grippers are in recessed position. Surfaces **36** are raked at angle θs to the radial as shown. During operation of the clutch reversal of rotation of the driving member from direction D1 to D2 may cause the stops to collide with the grippers. Rake angle θs of stop surfaces **36** direct the impact rebound of the grippers radially outward. Surfaces **6** and **8** of grippers **10** are arranged to make area-to-area contact with surfaces **35** and **36** respectively when the grippers are in recessed position.

FIG. **5** is an exploded view of load rotor **20** of clutch **100**. Disc **21***a* and disc **21***b* are attached in a rotationally and longitudinally ridged manner to inner shaft **22** so that a fixed longitudinal distance separates contact races **26***a* and **26***b*. Inner shaft **22** is fashioned to present a smaller rotational radius than races **26***a* and **26***b*. Any means known in the art may be selected for rigid attachment of the discs to the inner shaft. For example hex plugs **29** may be fashioned at the

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ends of inner shaft **22**, and may be fitted to hex sockets **49** in the discs using bolt **23**, nut **24** and washers **25**.

FIG. **6** is a perspective view of load rotor **20** of clutch **100**. Contact races **26a** and **26b** are conical surfaces facing radially outward and longitudinally inward. Load rotor **20** can be rotationally connected to the torque load by any means known in the art. For example splines **28** may be used. Any means known in the art can be used to arrange the load rotor to be rotatable about axis AX. For example, surfaces **27a** and **27b** may be used as bushing surfaces or bearing races for mounting the load rotor into a housing (not shown). In some embodiments connection of the load rotor to the torque load may fix the load rotor to the axis of rotation. In such cases surfaces **27** are not needed for mounting purposes. Alternative configurations for surfaces **27** and **28** and are not precluded. For example, splines or gear teeth may be placed on surfaces **27a** or **27b**, or the radius of surface **27a** may be expanded to merge with surface **27c**.

FIG. **7** is a perspective view of driving member **30** of clutch **100**. Driving member **30** includes main gear **31**, cam blocks **32** and stops **33**. The driving member provides radially inward facing cam surfaces **35**. Stop surfaces **36** protrude from the inner circumference of the driving member. In some embodiments front and back surfaces **52** of cam blocks **32** may be arranged to serve as guide slides for the clutch grippers. Driving member **30** can be rotationally connected to the source of rotating torque by any means known in the art. For example gear teeth **37** may be used. Any means known in the art can be used to arrange the driving member to be rotatable about axis AX in a manner that limits longitudinal movement relative load rotor **20**. For example rim **34** may be used for attachment to a bearing race. Stop blocks **33** are composed of an impact absorbing material such as a visco-elastic polymer, and cam blocks **32** are made of a strong, hard material such as steel.

FIG. **8** is a perspective view of main gear **31** of clutch **100**. Main gear **31** includes slots **38** and **39** for holding cam blocks **32** and stops **33**, respectively. The cam blocks and stop blocks may be press fit into their respective slots or they may be attached to the main gear using other methods known in the art.

FIG. **9** is an exploded view of gripper **10** of clutch **100**. Cradle **1** and pad holder **3** are made of hard, high strength materials such as steel. Pad **2** is made of a polytetrafluoroethylene (PTFE) composite material. Expansion slots **11** of pad **2** provide stress relief from the difference in thermal expansion of the pad and holder materials. Pad **2** and slot **12** may be arranged for a dove-tail fit or other means may be used to attach the pad to the holder, such as using adhesive. Any means known in the art may be used for attaching pad holder **3** to cradle **1**. For example bolts **53** may be used. FIG. **10** is a perspective view of gripper **10** of clutch **100**. In some embodiments surfaces **9** may be arranged to slide against guide surfaces **52** of driving member **30** in the rotational and radial directions.

FIG. **11** is a longitudinal view of gripper **10** of clutch **100**. Surfaces **7a**, **7b** and **6** are shaped and arranged so that when grippers **10** are disposed in a locking position in relation to load rotor **20** and driving member **30**, surfaces **7a** make area-to-area contact with race **26a**, surfaces **7b** make area-to-area contact with race **26b**, and surfaces **6** make area-to-area contact with a driving member cam surface **35**. FIG. **12** is a sectional view of gripper **10** taken along line **12-12** of FIG. **11**. A partial boundary of contact surface **7b** is shown with hidden lines. FIG. **13** is a bottom view of pad **2** showing the extent of expansion slots **11**. FIG. **14** is a perspective

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view of cradle **1** showing contact surface **7a** and showing surface **7b** with hidden lines. Surfaces **9** are also shown.

FIG. **15** is a perspective view of an alternative embodiment of gripper **10**. In this alternative embodiment surfaces **6**, **7a**, **7b** and surfaces **9** are all formed on a single block and low-friction coating **2**, consisting of a material such as PTFE, is deposited on surface **6**. FIG. **16** is a perspective view of load rotor **20** and a single gripper **10**, with the gripper disposed in its locking position in relation to the load rotor.

FIG. **17** is a perspective view of the one-way clutch **100**.

FIG. **18** is a schematic longitudinal view of load rotor races **26a**, **26b** and gripper **10** with gripper **10** shown in a recessed position. Load rotor races **26a** and **26b** are angled at a bevel angle θw to the axis of rotation AX. When the grippers are in recessed position a gap g separates surfaces **7a** and **7b** of the grippers from races **26a** and **26b**. FIG. **19** is a schematic longitudinal view of the load rotor races and a gripper with the gripper **10** shown in locking position.

The operation of the present invention is now described in light of FIGS. **1** through **19**. Referring to FIG. **3**, ω_i identifies the speed of rotation of load rotor **20** and inner shaft **22**. The speed of rotation of driving member **30** is labeled ω_o. For this discussion ω_i and ω_o are referenced to the stationary frame of reference and direction D1 is chosen for the positive rotational direction. When the gripper elements are in a recessed position, with no contact made with the load rotor, the load rotor and driving member are free to rotate independently of one another. When one or more grippers are in locking position the driving member is free to rotate in direction D2 relative the load rotor, and the load rotor is free to rotate in the D1 direction relative the driving member. That is, the driving member and load rotor are allowed to freewheel when ω_o is less than ω_i. In some embodiments of the invention, for example clutch **100**, centrifugal forces during freewheeling rotation of the driving member move the gripper(s) into their recessed position and the grippers cause no frictional resistance to freewheeling. In alternative embodiments using biasing springs, such as illustrated in FIG. **27**, the grippers may make contact with the load rotor during freewheeling but in a manner that causes little frictional resistance, depending on spring length and stiffness.

Referring again to FIG. **3**, when one or more grippers are situated in the locking position the gripper(s) perform a self-locking action that prevents rotational speed ω_o from exceeding ω_i. When torque is applied in direction D1 to the driving member, torque is transmitted through the grippers to the load rotor, counter-torque is transmitted from the load rotor to the driving member, and rotation of the load rotor and driving member is locked. A gripper which is locked and transferring torque will be said to be in locking mode. If the clutch is in an unlocked state, torque applied in direction D1 to the driving member urges the grippers into their locking position. In embodiments such as illustrated in FIG. **27**, elastic elements **44** may be employed to urge the grippers into locking position in addition to the force generated by torque applied to the driving member.

The forces impinging on the contact surfaces of the grippers **10** during locking mode are now described. FIG. **19** includes a longitudinal diagram of the forces acting on gripper **10** during locking mode. Fro is the radial component of the total force on surface **6**. Forces Fni-a and Fni-b are the normal components of the forces on gripper surfaces **7a** and **7b** from contacting surfaces **26a** and **26b**. Force Fri-a is the radial component and Fai-a is the longitudinal component of the force on surface **7a**. Force Fri-b is the radial component

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and Fai-b is the longitudinal components of the force on surface 7b. Fai-a is equal in magnitude and opposite in direction to Fai-b so the total longitudinal force on gripper 10 is null. The angle between Fni-a and Fri-a is angle θw, as is the angle between Fni-b and Fri-b. We observe that the relationship at both surfaces 7a and 7b of the normal to radial force components is:

$$Fni=Fri/\cos(\theta w) \tag{2}$$

Referring again to FIG. 19, the rotational radius of force Fro is Ro, and the rotational radius of forces Fni-a and Fni-b is Ri. The configuration of surfaces 26a, 26b, 6, 7a and 7b is arranged so that Ri approaches the value of Ro. Specifically, because surfaces 6, 7a and 7b are separated longitudinally the ratio Ri/Ro can be made close to or equal to unity.

FIG. 20 is a perspective view of the contact forces impinging on gripper 10 when in locking mode. Fo is the summation of pressure on surface 6 from contact with cam surface 35 of the driving member. Fi-a is the total force on surface 7a from contacting load rotor race 26a. Fni-a is the normal component of Fi-a (also shown in FIG. 19). Fti-a is the frictional force on surface 7a and is also the rotational component of Fi-a. The angle between Fi-a and Fni-a is angle θi. Surface 7b is impinged by corresponding forces Fi-b, Fni-b and Fti-b of near or equal magnitude to forces Fi-a, Fni-a and Fti-a, respectively.

FIG. 21 diagrams the forces acting on gripper 10 in locking mode from a front view. Fo is the total force acting on gripper surface 6. Fto is the frictional component and Fno is the normal component of Fo. Force Fti is the total frictional force from contacting the load rotor races and is the sum of the forces Fti-a and Fti-b described in FIG. 20. Fri is the total of the radial component of the forces from contacting the load rotor races and is the sum of the forces Fri-a and Fri-b shown in FIGS. 19 and 20. Fj is the sum of Fti and Fri. FIG. 21 illustrates the case when rotational radius Ro of force Fo is equal to rotational radius Ri of force Fj.

The self-locking action of grippers 10 is now described in terms of the forces diagrammed in FIGS. 19, 20 and 21. First we examine an initial condition in which grippers 10 are in locking position and the load rotor races are slipping in direction D2 against the grippers. This initial condition will be referred to as the lash condition. During the lash condition the force angle θi shown in FIG. 20 is the arctangent of μi, the kinetic coefficient of friction at the interface. The relation of Fti and Fni is therefore:

$$Fti=\mu i * Fni. \tag{3}$$

Substituting equation (2) for Fni in equation (3) reveals:

$$Fti=\mu i * (Fri/\cos(\theta w)) \tag{4}$$

and therefore:

$$Fti/Fri=\mu i/\cos(\theta w). \tag{5}$$

Referring to FIG. 21 we observe that the ratio Fti/Fri is the tangent of angle θj, so the value of θj during the lash condition is known:

$$\theta j = \text{Arctan}(\mu i/\cos(\theta w)). \tag{6}$$

Again referring to FIG. 21, when forces Fo and Fj are equal and opposite in magnitude and co-linear gripper 10 will remain stationary relative the driving member with no slipping at surface 6. In this case angles θo is equal to angle θj and, because the sum of angles φo and angle α is equal to θo, we have:

$$\phi o + \alpha = \theta j, \tag{7}$$

and therefore:

$$\phi o = \theta j - \alpha. \tag{8}$$

However if φo exceeds the friction angle arctan(μo) during the lash condition, where μo is the static coefficient of friction at surface 6, gripper 10 will slip against the driving member in direction D2. This slip at surface 6 increases compression on the gripper until ratio Fti/Fni drops below the coefficient of friction at the load rotor races. Therefore during the lash condition the grippers will grab and lock the load rotor if angle φo exceeds arctan(μo). Inserting this condition into equation (8) gives us the requirement for self-locking action of the grippers:

$$\text{arctan}(\mu o) < \theta j - \alpha \tag{9}$$

Substituting equation (6) for θj in equation (9) the conditions for self-locking action is expressed in terms of angle α, θw, and μi μo:

$$\alpha < \text{arctan}(\mu i/\cos(\theta w)) - \text{arctan}(\mu o). \tag{10}$$

The difference between requirements (10) and (1) should be noted. Requirement (1) describes the self-locking requirement of the conventional wedge-clutch configuration illustrated in FIG. 22 and includes the term φi. But φi goes to zero as the ratio Ri/Ro approaches unity. Therefore the negative effect of φi on the maximum ramp angle is negligible or absent in the self-locking action of the grippers in the present invention.

FIG. 23 is a schematic front view of cam surface 35 demonstrating the curvature of the cam surfaces of clutch 100. RR(0) is the radial at the leading edge of surface 35 of driving member 30. $R_{35}(0)$ is the rotational radius of surface 35 at the leading edge. $R_{35}(0)$ is also the point of maximum rotational radius of the surface and will be designated R_{max}. $R_{35}(\beta)$ is the rotational radius of surface 35 at rotational angle β, measured in radians, from RR(0) in direction D2. In the preferred embodiments of the present invention the curvature of surface 35 follows a logarithmic spiral according to the polar co-ordinate formula:

$$R_{35}(\beta)=R_{max} * \exp(-\tan(\alpha) * \beta) \tag{11}$$

for: $0 \le \beta \le \beta_{max}$

where β_{max} is the value of β at the trailing edge of surface 35 and signifies the angular span of the surface. The value α is the chosen ramp angle of cam surface 35 selected according to requirement (10).

It is now shown that logarithmic curve (11) produces even pressure over surface 6 of grippers 10 when the grippers are in locked position and transferring torque. In the invention's preferred embodiments the curvature of surface 6 matches or closely approximates the curvature of surface 35 when the gripper is in locking position. During the lash condition described above the gripper slides a differential distance dS across surface 35 until lock is achieved. The gripper is compressed against the load rotor races as the gripper is forced radially inward during this lash motion. The pressure at each point on surface 6 is proportional to the differential ratio dR/dS at the point, where dR is the radial displacement caused by motion dS. But the differential $dR_{35}(\beta)/dS$ of curve (11) is a constant value for all points along the surface:

$$dR_{35}(\beta)/dS=-\tan(\alpha). \tag{12}$$

Therefore during torque transfer the pressure is constant or nearly constant over the span of gripper surface 6.

During the operating life of the clutch it is expected that surface wear and temperature variations will change the distance that the grippers slide across cam surfaces 35 as they move from their recessed position to their locking

position. The logarithmic nature of curve (11) maintains alignment of the grippers in the roll direction as this sliding distance changes. Referring again to FIG. **23**, Cr is the rotational circle of point $R_{35}(\beta)$ on surface **35**. Tr is the tangent to Cr and T**35** is the tangent to the surface at $R_{35}(\beta)$. For curve (11) the angle between Tr and T**35** is constant and is equal to ramp angle α at all values β across the span. Therefore the roll alignment of the grippers in relation to load rotor races is constant as the angle β of the locking position changes.

When the curvature of the cam surfaces of the invention is made according to curve (11) the performance of the clutch is generally maximized. In embodiments that include gripper elements that span a large rotational angle implementation of curve (11) is recommended. It should be appreciated, however, that various embodiments of the invention may implement other curvatures as long as the curvature produces self-locking action of the clutch grippers. For example, a circular curvature or even a flat cam surface may be used especially with gripper elements spanning a small rotational angle.

Various alternative embodiments of the current invention are now described. Referring to expression (10), clutch **100** achieves a high ramp angle α and low compression pressures in part by implementing load rotor races with bevel angle θw. A low coefficient of friction μo at the interface between gripper surfaces **6** and cam surfaces **35** also increases the attainable ramp angle. Clutch **100** provides a low μo value by fabricating gripper pad **2** of a PTFE material. Alternative methods of providing a low μo value may be used. For example cam block **32** of the driving member may be composed of a PTFE composite, or cam surface **35** of the driving member may be coated with a low-friction material. In these cases gripper surface **6** may be formed on a material other than PTFE, such as steel.

FIG. **24** is a perspective view of an alternative embodiment of driving member **30**. In this embodiment cam surfaces **35** are formed directly on the inner circumference of the driving member. Although the stops in clutch **100** are made of an impact absorbing material the stop surfaces may be formed on other materials in alternative embodiments. For example, stop surfaces **36** in FIG. **24** are formed directly on the driving member inner circumference. It should also be appreciated that other embodiments of the invention may implement stop surfaces of various curvatures other than the surface implemented in clutch **100**.

FIG. **25** is a longitudinal sectional view of clutch **200**, an additional embodiment of the current invention. Clutch **200** includes an alignment means for limiting longitudinal, pitching and yawing movement of grippers **10** during operation of the clutch. Surfaces **52** of driving member **30** are arranged to form guide slides for the grippers. Contact between surfaces **9** and **52** limit yawing, pitching and longitudinal motion but allow the grippers to slide in the radial and rotational directions relative to the driving member. Low-friction coating **51** may be applied to cam blocks **32**, or to gripper surfaces **9**, to promote easy sliding of the surfaces. Clutch **200** load rotor races **26a** and **26b** are not beveled, therefore the term $\cos(\theta w)$ in expression (10) is unity and does not assist in maximizing α. However the clutch **200** races are arranged so that rotational radius Ri is greater than Ro. This introduces the term ϕi, described above, in a manner that assists in maximizing ramp angle α. In this case the conditions for self-locking action of the grippers is expressed as:

$$\alpha < \arctan(\mu i) - \arctan(\mu o) + \phi i \qquad (13)$$

FIG. **26** is a longitudinal sectional view of clutch **300**, a further embodiment of the current invention. Clutch **300** includes a lubricating substance, such as oil or grease, to lower friction at the cam surfaces, thus providing a low value μo. Channel **61** provides means for ejecting the lubricant to the cam surfaces **35**. Alternatively lubricant may be directed to the cam surfaces by other means, for example with an external spray nozzle or jet (not shown). Sump cavity **63** provides a means for catching the splash of lubricant that may occur during clutch operation. Channel **62** provides a means for evacuating lubricant splash from the sump cavity. Clutch **300** load rotor races **26a** and **26b** are radially outward facing and longitudinally outward facing conical surfaces. The bevel angle of the races serves to spin any lubricant splash, with the influence of centrifugal forces, away from the race surfaces.

FIG. **27** is a partial front sectional view of another embodiment of the present invention. This embodiment includes elastic elements **44** attached to either main gear **31** or grippers **10**, or both. The elastic elements serve to limit rotation of the grippers in direction D1 as do the stops of previous described embodiments. The elastic elements may be employed to serve in the manner of biasing springs used in conventional technologies. Used as biasing springs the elastic elements are placed to urge the grippers into locking position in addition to or in the absence of inertial forces. But the elastic elements may be employed for other purposes. For example, elements **44** can be employed to pull grippers **10** out of contact with the load rotor races and into a slightly recessed position when the driving member is motionless and centrifugal forces are absent.

In yet another embodiment of the present invention the values for ramp angle α of driving member surfaces **35**, bevel angle θw of load rotor contact races, and the ratio Ri/Ro are selected so that self-locking action of the grippers **10** is achieved without use of lubrication or low friction materials to lower the value of μo. This type of embodiment may be useful in some applications, for example when torque loading is relatively light.

It should be noted that for clarity the direction of torque transmission described for clutch **100** corresponds to the expected use of the present invention in most common circumstances. However, the disclosure is not limited to these circumstances and may be used in a manner that reverses the direction of torque transmission described above. That is, the load rotor may be connected to a source of rotating power, or may be held stationary in cases where the clutch is used as a break or backstop. In these uses the driving member may be arranged for rotational connection to a torque load.

What I claim is:

1. A one-way clutch, comprising:
   a load rotor including a first race and a second race and an inner shaft, with each race consisting of a radially outward facing circular contact surface rotatable about an axis of rotation, with each race rigidly connected to the inner shaft so that a fixed longitudinal distance separates the first and second races, with said inner shaft having a smaller rotational radius than the rotational radii of the races;
   a driving member rotatable co-axially with said load rotor, including an inner circumference with at least one radially inward facing cam surface, with most or all of each cam surface rotatable within said longitudinal distance separating said load rotor first and second races, said load rotor and driving member providing a space therebetween;

at least one gripper element for the performance of torque transmission between said driving member and said load rotor, said gripper elements located and movable within the space between said load rotor and driving member, with each gripper element including a first gripping surface and a second gripping surface and a sliding surface, with the sliding surface disposed between the first and second gripping surfaces;

a stop for each said gripper element protruding from said driving member adapted to limit rotation of the gripper element in the first rotational direction relative the driving member;

wherein said gripper surfaces are arranged to provide a locking position in which said first gripping surface makes area-to-area contact with the said load rotor first race, said second gripping surface makes area-to-area contact with said load rotor second race, and said gripper sliding surface makes area-to-area contact with a cam surface of said driving member.

2. The one-way clutch of claim **1** wherein:

the contacting surfaces of said gripper elements, load rotor and driving member are arranged so that during freewheeling rotation of the driving member the gripper elements are urged radially outward from said locking position.

3. The one-way clutch of claim **1** wherein:

the contacting surfaces of said gripper elements, load rotor and driving member are arranged so that when the gripper elements are disposed in said locking position at least one contact point between said gripping surfaces and said load rotor races has a rotational radius equal to or greater than the rotational radius of at least one contact point between said slipping surface and said driving member cam.

4. The one-way clutch of claim **1** wherein:

said gripper element slipping and gripping surfaces, and said load rotor races, and said driving member cam surfaces consist of materials producing a lower coefficient of friction between the gripper elements and driving member cams than the coefficient of friction between the gripping elements and load rotor races.

5. The one-way clutch of claim **1** wherein:

a means is included for providing a lubricating substance to the interface of said gripper element slipping surfaces and driving member cam surfaces.

6. The one-way clutch of claim **1** wherein:

each said load rotor race is slanted to the axis of rotation to form a conical surface.

7. The one-way clutch of claim **1** wherein:

said driving member cam surfaces form a logarithmic spiral.

8. The one-way clutch of claim **1** wherein:

said driving member includes at least one alignment surface adapted to limit motion of said gripper elements, in relation to the driving member, in the yawing or pitching or longitudinal directions, or any combination of yawing or pitching or longitudinal directions.

9. The one-way clutch of claim **1** wherein:

said stops are composed of an impact absorbing material.

10. The one-way clutch of claim **1** wherein:

said stops are arranged to reflect impact rebound of said gripper elements radially outward.

11. The one-way clutch of claim **1** wherein:

said stops include an elastic material.

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