(19) Europäisches Patentamt
European Patent Office
Office européen des brevets

(11) EP 4 006 323 B1

(12) EUROPEAN PATENT SPECIFICATION

(45) Date of publication and mention of the grant of the patent:
07.06.2023 Bulletin 2023/23

(21) Application number: **20210071.5**

(22) Date of filing: **26.11.2020**

(51) International Patent Classification (IPC):
F02B 75/04 *(2006.01)*

(52) Cooperative Patent Classification (CPC):
F02B 75/048

(54) AN INTERNAL COMBUSTION ENGINE INCLUDING VARIABLE COMPRESSION RATIO

BRENNKRAFTMASCHINE MIT VARIABLEM VERDICHTUNGSVERHÄLTNIS

MOTEUR À COMBUSTION INTERNE À TAUX DE COMPRESSION VARIABLE

(84) Designated Contracting States:
AL AT BE BG CH CY CZ DE DK EE ES FI FR GB GR HR HU IE IS IT LI LT LU LV MC MK MT NL NO PL PT RO RS SE SI SK SM TR

(43) Date of publication of application:
01.06.2022 Bulletin 2022/22

(73) Proprietor: **Gomecsys B.V.**
1411 AR Naarden (NL)

(72) Inventors:
- **De Gooijer, Lambertus Hendrik**
1401 EP BUSSUM (NL)
- **Wagenaar, Sander**
1274 GR HUIZEN (NL)

(74) Representative: **De Vries & Metman**
Overschiestraat 180
1062 XK Amsterdam (NL)

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Description

[0001] The present invention relates to an internal combustion engine including variable compression ratio comprising a crankshaft being rotatable about a crankshaft axis and having a crankpin, wherein the crankshaft axis and a centreline of the crankpin span a crank plane, a connecting rod, a piston being rotatably connected to a distal end portion of the connecting rod, an eccentric member being rotatably mounted on the crankpin and comprising a bearing portion having an outer circumferential wall which bears a proximal end portion of the connecting rod such that the connecting rod is rotatably mounted on the bearing portion, wherein the bearing portion is eccentrically disposed with respect to the crankpin and the eccentric member is provided with an eccentric member gear which is drivably coupled to a gear train for rotating the eccentric member with respect to the crankpin.

[0002] Such an internal combustion engine is known from EP 3 365 544. In the known internal combustion engine the gear train comprises a first intermediate gear which meshes with the eccentric member gear and which is fixed to a second intermediate gear that has a smaller diameter than the first intermediate gear. The first and second intermediate gears are rotatably mounted to the crankshaft and have a common axis of rotation which extends parallel to the crankshaft axis and a centreline of the crankpin. The gear train further comprises an actuating gear which meshes with the second intermediate gear and which is fixed to an actuating shaft that extends through the crankshaft. The actuating shaft is rotatable with respect to the crankshaft about the crankshaft axis. It can be turned by a control mechanism in order to vary the compression ratio of the internal combustion engine. In case of running at fixed compression ratio the actuating shaft has a fixed angular position with respect to a crankcase of the engine. The gear ratios of the gears of the gear train and the eccentric member gear are such that under operating conditions, in case of a standstill of the actuating gear, the eccentric member is rotated with respect to the crankpin at half speed of the speed of the crankshaft with respect to the crankcase and in opposite direction thereof. Hence, if the crankshaft rotates twice about the crankshaft axis, the eccentric member rotates once about the crankpin in opposite direction; as seen from the crankcase the eccentric member rotates once about the centreline of the crankpin in the same rotational direction as the crankshaft.

[0003] Due to the eccentrical arrangement of the bearing portion an inertial force on the eccentric member may lead to a torque about the centreline of the crankpin under operating conditions when the direction of such inertial force is angled with respect to a plane that is spanned by the centreline of the crankpin and a centreline of the bearing portion of the eccentric member. The location of maximum eccentricity of the eccentric member lies in this plane. The torque on the eccentric member is transferred to the gear train and may lead to a relatively high force peak on gear teeth, particularly on gear teeth of the relatively small actuating gear. This drawback typically increases in case of small high-speed engines, which may run at a speed of more than 6000 rpm, for example.

[0004] The present invention aims to improve the internal combustion as described above.

[0005] For this purpose the internal combustion engine according to the invention is characterized in that the gear train comprises a balancing gear which is rotatably mounted to the crankshaft and rotates with respect to the crankshaft about a balancing gear axis at the same speed as the eccentric member gear under operating conditions, wherein the crankshaft axis and the balancing gear axis span a balancing gear plane, wherein the balancing gear has an eccentrical centre of gravity which is located such that under operating conditions it causes a counter torque against a torque that is exerted by the eccentric member gear onto the balancing gear due to an inertial force on the eccentric member.

[0006] The counter torque is generated by a centrifugal force on the balancing gear which has a direction from the crankshaft axis to the centre of gravity of the balancing gear. An advantage of providing a counter torque by the balancing gear is that the torque which is exerted by the eccentric member onto the balancing gear is not or not fully transferred to one or more gears of the gear train which are drivably coupled to the eccentric member gear via the balancing gear. The inertial force on the eccentric member may be generated by inertia of at least one of the piston, the connecting rod and the eccentric member itself.

[0007] For example, when the piston is in bottom dead centre or top dead centre the inertial force of the piston and the connecting rod onto the eccentric member is high; when in this situation a plane that is spanned by the centreline of the crankpin and a centreline of the bearing portion is angled with respect to the direction of the inertial force, the inertial force generates a torque on the eccentric member about the centreline of the crankpin. This torque is transferred to the balancing gear, but due to the invention it is not or not fully transferred further to one or more gears of the gear train which are drivably coupled to the eccentric member gear via the balancing gear. It is noted that the gear train has more than one gear and is adapted such that the balancing gear rotates about the balancing gear axis at the same speed as the eccentric member gear rotates about the crankpin.

[0008] In a preferred embodiment the gear train is adapted such that the eccentric member rotates with respect to the crankpin at half speed of the speed of the crankshaft and in opposite direction thereof, since the relative motion of the eccentric member and the crankpin, on the one hand, and of the eccentric member and the connecting rod, on the other hand, is relatively small, which minimizes friction losses. In fact, bearings between the bearing portion and the connecting rod and between the eccentric member and the crankpin experience half the crankshaft speed.

[0009] The internal combustion engine may also have a control mechanism which is drivably coupled to the gear train in order to vary the compression ratio of the internal combustion engine.

[0010] In a particular embodiment the eccentric member has an eccentrical centre of gravity and the inertial force is a centrifugal force caused by the eccentrical centre of gravity of the eccentric member.

[0011] The centrifugal forces of the eccentric member and the balancing gear in the centres of gravity of the eccentric member and the balancing gear, respectively, are directed radially from the crankshaft axis; centrifugal forces in the centres of gravity due to rotation of the eccentric member about the crankpin and the balancing gear about the balancing gear axis, respectively, are neglected since their rotational speeds are half of the rotational speed of the crankshaft.

[0012] In an embodiment the centre of gravity of the eccentric member lies in a plane that is spanned by the centreline of the crankpin and a centreline of the bearing portion, wherein the eccentric member and the balancing gear are arranged such that when under operating conditions the centre of gravity of the eccentric member lies in the crank plane, the centre of gravity of the balancing gear lies in the balancing gear plane. This synchronizes the torque and counter torque under operating conditions, such that the torque and counter torque increase at the same time and decrease at the same time. When the centre of gravity of the eccentric member lies in the crank plane and the centre of gravity of the balancing gear lies in the balancing gear plane the torque and counter torque are zero. It is noted that in this case there may still be another inertial force than the centrifugal force onto the eccentric member, which generates a torque on the eccentric member.

[0013] In a more specific embodiment, when under operating conditions the centre of gravity of the eccentric member lies in the crank plane and the centreline of the crankpin lies between the crankshaft axis and the centre of gravity of the eccentric member, the centre of gravity of the balancing gear lies between the crankshaft axis and the balancing gear axis.

[0014] The eccentric member gear and the balancing gear may rotate in opposite direction with respect to each other under operating conditions.

[0015] In the event that the eccentric member gear meshes with the balancing gear, they rotate in opposite direction with respect to each other under operating conditions.

[0016] In an embodiment the gear train comprises an actuating gear which is drivably coupled to the balancing gear and fixed to an actuating shaft, wherein the actuating shaft is rotatably mounted to the crankshaft and rotatable about an axis which coincides with the crankshaft axis, wherein under operating conditions the actuating shaft stands still at fixed compression ratio. In practice the actuating gear may be located at the same side of a crank arm of the crankshaft as the eccentric member gear and the balancing gear, whereas the actuating shaft extends through the crankshaft. It is noted that a stand still of the actuating shaft means that the actuating shaft has a fixed position with respect to a crankcase of the internal combustion engine.

[0017] In a compact design of the internal combustion engine the balancing gear is a first stage gear that is fixed to a second stage gear and which has a larger diameter than the second stage gear, wherein the second stage gear meshes with the actuating gear. The second stage gear also forms part of the gear train.

[0018] The crank plane and the balancing gear plane may coincide.

[0019] In an alternative embodiment the balancing gear is an intermediate gear which meshes with the actuating gear. In this case the number of teeth of the intermediate gear equals the number of teeth of the eccentric member gear and equals twice the number of teeth of the actuating gear. The intermediate meshes with the actuating gear and the eccentric member gear.

[0020] The intermediate gear may extend beyond the eccentric member gear, whereas the actuating gear at least partly overlaps the eccentric member gear. This provides the opportunity to design an internal combustion engine including a relatively short stroke.

[0021] The crank plane and the balancing gear plane may be angled with respect to each other.

[0022] The counter torque may be smaller than the torque that is exerted by the eccentric member onto the balancing gear. The level of the counter torque can be adapted by selecting the location of the centre of gravity of the balancing gear.

[0023] It is possible that the counter torque is higher than the torque which is exerted by the eccentric member onto the balancing gear due to the centrifugal force caused by the eccentrical centre of gravity of the eccentric member in order to at least partly balance an additional imbalance, which is caused by another inertial force on the eccentric member. For example, under operating conditions there is also a centrifugal force of the proximal end portion or big end of the connecting rod, which has a direction from the crankshaft axis to its centre of gravity and at least partly acts onto the eccentric member through the centreline of the bearing portion which is located eccentrically with respect to the centreline of the crankpin. This leads to a torque onto the eccentric member when the centrifugal force is angled with respect to the plane that is spanned by the centrelines of the crankpin and the bearing portion. The torque is transferred to the balancing gear; this torque may also at least partly be reduced by selecting an appropriate location of the centre of gravity of the balancing gear. Similarly, it is also possible to balance inertial forces of the piston and/or connecting rod, particularly peak inertial forces at or near top dead centre and bottom dead centre of the piston, in the event that these forces occur at a certain rotational position of the eccentric member, which leads to relatively high torque. Because of several different inertial forces the location of the centre of gravity of the balancing gear may be compromised in practice.

[0024] The eccentrical centre of gravity of the balancing gear may be created by a cavity in the balancing gear, but alternative manners are conceivable. For example, the balancing gear may be thicker at a side of the balancing gear axis where its centre of gravity is intended such that its teeth are also longer at that side.

[0025] The invention will hereafter be elucidated with reference to the schematic drawings showing embodiments of the invention by way of example.

Fig. 1 is a side view of a part of an embodiment of an internal combustion engine according to the invention.
Fig. 2 is a perspective view of the embodiment as shown in Fig. 1.
Fig. 3 is an enlarged view of a part of the embodiment as shown in Figs. 1 and 2.
Fig. 4 is a similar view as Fig. 3, showing the part of the embodiment from a different side.
Figs. 5-9 are side views of the embodiment as shown in Fig. 1, illustrating different situations under operating conditions.
Figs. 10 and 11 are similar views as Figs. 5 and 7, respectively, showing parts thereof.
Fig. 12 is a similar view as Figs. 5-11, showing another embodiment on a larger scale.
Fig. 13 is a perspective view of a part of the embodiment of Fig. 12.

[0026] Figs. 1-4 show parts of an embodiment of an internal combustion engine 1 including variable compression ratio according to the invention. Figs. 5-11 show different situations of this embodiment under operating conditions. The engine 1 comprises a crankshaft 2 which is rotatably mounted to a crankcase (not shown) and rotatable about a crankshaft axis 3. The crankshaft 2 has a crankpin 4 on which an eccentric member 5 is rotatably mounted. The eccentric member 5 is rotatable about a centreline of the crankpin 4. The eccentric member 5 comprises a bearing portion 6 which has an outer circumferential wall that bears a proximal end portion or a big end of a connecting rod 7. Hence, the connecting rod 7 is rotatably mounted on the bearing portion 6 and rotatable about a centreline of the bearing portion 6, which is parallel to the centreline of the crankpin 4. In other words, the bearing portion 6 is eccentrically disposed with respect to the crankpin 4. Fig. 4 shows a small circle on the eccentric member 5 which indicates the angular location where the eccentricity of the bearing portion 6 has its maximum. This angular location lies in a plane that is spanned by the centreline of the crankpin 4 and the centreline of the bearing portion 6. Due to the eccentricity the eccentric member 5 has an eccentrical centre of gravity, see Figs. 10 and 11 in which the centre of gravity is indicated by reference sign 5a. In this case the angular location of the centre of gravity 5a is the same as the angular location where the eccentricity of the bearing portion 6 has its maximum.

[0027] The internal combustion engine 1 is further provided with a piston 8 which is rotatably connected to a distal end portion or a small end of the connecting rod 7. In this case the engine 1 has a single cylinder but a multicylinder is also conceivable.

[0028] The eccentric member 5 is provided with an eccentric member gear 9 which has an axis of rotation that coincides with the centreline of the crankpin 4. The eccentric member 5 is drivably coupled to a gear train 10 for rotating the eccentric member 5 with respect to the crankpin 4 at half speed of the speed of the crankshaft 2 and in opposite direction thereof. This means that if the crankshaft 2 rotates twice about the crankshaft axis 3 in clockwise direction, the eccentric member 5 rotates once about the crankpin 4 in anti-clockwise direction.

[0029] In the embodiment as shown in Figs. 1-11 the gear train comprises a first stage gear 11 that meshes with the eccentric member gear 9, a second stage gear 12 which is fixed to the first stage gear 11 and an actuating gear 13 which meshes with the second stage gear 12. The first stage gear 11 and the second stage gear 12 are rotatably mounted to the crankshaft 2 and have a common axis of rotation. The actuating gear 13 is fixed to an actuating shaft 14 that extends through the crankshaft 2. The actuating shaft 14 is rotatable with respect to the crankshaft 2 about the crankshaft axis 3.

[0030] A worm wheel 15 is fixed on the actuating shaft 14 and the worm wheel 15 meshes with a worm 16 that is drivable by an electric motor 17. The electric motor 17, worm 16 and worm wheel 15 form part of a control mechanism which is drivably coupled to the gear train 10. The actuating shaft 14 can be turned by the electric motor 17 in order to vary the compression ratio of the internal combustion engine 1. This provides the opportunity to operate the internal combustion engine 1 at a high compression ratio under low load conditions in order to improve its efficiency. Under high load conditions, the compression ratio can be decreased to avoid detonations. In case of running at fixed compression ratio the actuating shaft 14 has a fixed angular position with respect to the crankcase.

[0031] When the internal combustion engine 1 is running the eccentrical centre of gravity 5a of the eccentric member 5 generates a centrifugal force on the eccentric member 5, which is directed radially from the crankshaft axis 3. During periods in which the centre of gravity 5a of the eccentric member 5 lies outside a crank plane in which the crankshaft axis 3 and the centreline of the crankpin 4 lie, the centrifugal force leads to a fluctuating torque about the centreline of the crankpin 4. The fluctuating torque is exerted onto the first stage gear 11 by the eccentric member 5 via the eccentric member gear 9. In order to at least partly reduce transferal of the fluctuating torque via the second stage gear 12 to the relatively small actuating gear 13 so as to avoid overload of its gear teeth, the first stage gear 11 is a balancing gear. This means that the first stage gear 11 has the same number of teeth as the eccentric member gear 9, such that it rotates at the same speed as the eccentric member gear 9 under operating conditions and that the first stage gear 11 has an eccentrical centre of gravity 11a, see Figs. 10 and 11 in which the centre of gravity is indicated by reference sign 11a.

[0032] Furthermore, the centre of gravity 11a of the first stage gear 11 is located such that under operating conditions it causes a counter torque against the torque that is exerted by the eccentric member 5 onto the first stage gear 11 due to the centrifugal force. In this case the centre of gravity 11a of the first stage gear 11 is located eccentrically because of the presence of a cavity 18 in the first stage gear 11, but this can be created in an alternative manner. The centre of gravity 11a of the first stage gear 11 and the cavity 18 are located at opposite sides of the common axis of rotation of the first stage gear 11 and the second stage gear 12. The common axis of rotation of the first stage gear 11 and the second stage gear 12 may be called a balancing gear axis. It is noted that since the first stage gear 11 and the second stage gear 12 are fixed to each other the eccentrical centre of gravity 11a may also be created at the second stage gear 12, for example by creating a cavity in the second stage gear 12.

[0033] Figs. 5-9 show successive situations of the engine 1 under operating conditions when the piston 8 moves from top dead centre, where firing starts, to bottom dead centre, where gas exchange starts, and back to top dead centre where gas exchange stops, hence during one revolution of the crankshaft 2. The figures show that during one revolution of the crankshaft 2 in clockwise direction the crank member 5 rotates by a half revolution about the crankpin 4 in anti-clockwise direction. Figs. 10 and 11 illustrate the locations of the centres of gravity 5a, 11a of the eccentric member 5 and the first stage gear 11 in the situations as shown in Figs. 5 and 7, respectively.

[0034] In the situation as shown in Figs. 5 and 10 the crankshaft axis 3, the centreline of the crankpin 4 and the centre of gravity 5a of the eccentric member 5 lie in the crank plane, whereas the centreline of the crankpin 4 lies between the crankshaft axis 3 and the centre of gravity 5a of the eccentric member 5. In this situation the crankshaft axis 3 and the axis of rotation of the first stage gear 11 lie in a balancing gear plane, which coincides with the crank plane because of the arrangement and dimensions of the actuating gear 13, the first stage gear 11, the second stage gear 12 and the eccentric member gear 9. Furthermore, the centre of gravity 11a of the first stage gear 11 is located in the balancing gear plane and lies between the crankshaft axis 3 and the axis of rotation of the first stage gear 11.

[0035] Fig. 10 illustrates the centrifugal forces in the respective centres of gravity 11a and 5a, which centrifugal forces are directed radially from the crankshaft axis 3. Because of the locations of the centre of gravity 11a of the first stage gear 11 and the centre of gravity 5a of the eccentric member 5 the respective centrifugal forces are directed in opposite direction. The centrifugal force on the eccentric member 5 is directed through the centreline of the crankpin 4 and the centrifugal force on the first stage gear 11 is directed through its axis of rotation, which means that in this situation they do not generate a torque about the centreline of the crankpin 4 and the axis of rotation of the first stage gear 11, respectively. It is noted that the centrifugal forces in the centres of gravity 5a, 11a due to rotation of the eccentric member 5 about the crankpin 4 and the first stage gear 11 about its axis of rotation are neglected since their rotational speeds are half of the rotational speed of the crankshaft 2. Besides, they do not result in a torque on the eccentric member 5 and the first stage gear 11.

[0036] Fig. 11 shows a situation in which the piston 8 is in bottom dead centre. Since the first stage gear 11 and the eccentric member gear 9 mesh with each other the centres of gravity 11a, 5a of the first stage gear 11 and the eccentric member 5 have rotated in opposite directions with respect to each other compared to the situation as shown in Fig. 10. In the situation as shown in Fig. 11 the centre of gravity 5a of the eccentric member 5 lies outside the crank plane that is spanned by the crankshaft axis 3 and the centreline of the crankpin 4, and the centre of gravity 11a of the first stage gear 11 lies outside the balancing gear plane that is spanned by the crankshaft axis 3 and the axis of rotation of the first stage gear 11.

[0037] The centrifugal force in the centre of gravity 5a of the eccentric member 5 generates a torque in clockwise direction which torque is exerted via the eccentric member gear 9 onto the first stage gear 11. Due to the opposite rotational motions of the meshing eccentric member gear 9 and the first stage gear 11 the centrifugal force in the centre of gravity 11a of the first stage gear 11 also generates a torque in clockwise direction which forms a counter torque against the torque that is exerted by the eccentric member gear 9. The counter torque avoids that the torque which is exerted by the eccentric member 5 is entirely transferred to the relatively small actuating gear 13 such that its teeth are prevented from overload. In practice the first stage gear 11 may be adapted such that the counter torque is smaller than the torque that is generated by the eccentric member.

[0038] Figs. 12 and 13 show a part of an alternative embodiment of the internal combustion engine 1 of the invention. the gear train 10 of this embodiment is different from that of the embodiment as shown in Figs. 1-11. The eccentric member gear 9 meshes with an intermediate gear 19, which forms the balancing gear. The intermediate gear 19 extends beyond the eccentric member gear 9 such that it also meshes with the actuating gear 13. The actuating gear 13 is located in front of the eccentric member gear 9 in axial direction thereof and partly overlaps the eccentric member gear 9. In this case the crank plane is also spanned by the crankshaft axis 3 and a centreline of the crankpin 4 and the balancing gear plane is also spanned by the crankshaft axis 3 and the axis of rotation of the intermediate gear 19, but the crank plane and the balancing gear plane extend perpendicularly with respect to each other.

[0039] In the situation as shown in Fig. 12 the centre of gravity 5a of the eccentric member 5 lies at a side of the crank plane which is directed clockwise and the centre of gravity of the intermediate gear 19 lies at a side of the balancing gear plane which is directed clockwise such that the centrifugal force on the eccentric member 5 generates a torque in anti-clockwise direction about the centreline of the crankpin 4 and the centrifugal force on the intermediate gear 19 generates a counter torque in anti-clockwise direction about the axis of rotation of the intermediate gear 19.

[0040] The invention also provides the opportunity to generate a counter torque against a different inertial force onto the eccentric member 5, for example an inertial force that is generated by the piston 8 and/or the connecting rod 7. For example, under operating conditions the big end of the connecting rod 7 generates a centrifugal force which is directed from the crankshaft axis 3 to its centre of gravity, which at least partly acts onto the eccentric member 5 through the centreline of the bearing portion 6. This leads to a torque on the eccentric member 5 when the direction of this centrifugal force is angled with the plane that is spanned by the centrelines of the crankpin 4 and the bearing portion 6. This condition would happen in the situation as shown in Fig. 11, for example. Hence, even if the eccentric member 5 was balanced itself by having its centre of gravity 5a at the centreline of the crankpin 4, the first stage gear 11 or balancing gear could at least partly compensate the centrifugal force that is generated by the big end.

[0041] Another example of other inertial forces onto the eccentric member 5 is inertia of the piston 8 and the connecting rod 7 due to their reciprocating motion, which inertial forces are highest in top dead centre and bottom dead centre of the piston 8. For example, in the situation as shown in Fig. 11 the inertial forces of the piston 8 and the connecting rod 7 are exerted in downward direction onto the eccentric member 5 and are directed through the centreline of the bearing portion 6. This leads to a torque on the eccentric member 5 since the direction of the inertial forces are angled with respect to the plane that is spanned by the centrelines of the crankpin 4 and the bearing portion 6. The resulting torque can at least partly be compensated by the first stage gear 11 or balancing gear.

[0042] From the foregoing, it will be clear that the invention provides an effective solution to prevent overload of teeth of the actuating gear due to an inertial force onto the eccentric member, in particular at high engine speed.

[0043] The invention is not limited to the embodiments shown in the drawings and described hereinbefore, which may be varied in different manners within the scope of the appended claims, defining the invention.

[0044] For example, the arrangement and dimensions of the eccentric member gear and the gears of the gear train may be different.

Claims

1. An internal combustion engine (1) including variable compression ratio comprising

> a crankshaft (2) being rotatable about a crankshaft axis (3) and having a crankpin (4), wherein the crankshaft axis (3) and a centreline of the crankpin (4) span a crank plane,
> a connecting rod (4),
> a piston (8) being rotatably connected to a distal end portion of the connecting rod (4);
> an eccentric member (5) being rotatably mounted on the crankpin (4) and comprising a bearing portion (6) having an outer circumferential wall which bears a proximal end portion of the connecting rod (4) such that the connecting rod (4) is rotatably mounted on the bearing portion (6), wherein the bearing portion (9) is eccentrically disposed with respect to the crankpin (4) and the eccentric member (5) is provided with an eccentric member gear (9) which is drivably coupled to a gear train (10) for rotating the eccentric member (5) with respect to the crankpin (4), **characterized in that** the gear train (10) comprises a balancing gear (11, 19) which is rotatably mounted to the crankshaft (2) and rotates with respect to the crankshaft (2) about a balancing gear axis at the same speed as the eccentric member gear (9) under operating conditions, wherein the crankshaft axis (3) and the balancing gear axis span a balancing gear plane, wherein the balancing gear (11) has an eccentrical centre of gravity (11a) which is located such that under operating conditions it causes a counter torque against a torque that is exerted by the eccentric member gear (9) onto the balancing gear (11, 19) due to an inertial force on the eccentric member (5).

2. An internal combustion engine (1) according to claim 1, wherein the gear train (10) is adapted such that the eccentric member (5) rotates with respect to the crankpin (4) at half speed of the speed of the crankshaft (2) and in opposite direction thereof.

3. An internal combustion engine (1) according to claim 1 or 2, wherein the eccentric member (5) has an eccentrical centre of gravity (5a) and the inertial force is a centrifugal force caused by the eccentrical centre of gravity (5a) of the eccentric member (5).

4. An internal combustion engine (1) according to claim

3, wherein the centre of gravity (5a) of the eccentric member (5) lies in a plane that is spanned by the centreline of the crankpin (4) and a centreline of the bearing portion (6), wherein the eccentric member (5) and the balancing gear (11, 19) are arranged such that when under operating conditions the centre of gravity (5a) of the eccentric member (5) lies in the crank plane, the centre of gravity (11a) of the balancing gear (11, 19) lies in the balancing gear plane.

5. An internal combustion engine (1) according to claim 4, wherein when under operating conditions the centre of gravity (5a) of the eccentric member (5) lies in the crank plane and the centreline of the crankpin (4) lies between the crankshaft axis (3) and the centre of gravity (5a) of the eccentric member (5), the centre of gravity (11a) of the balancing gear (11, 19) lies between the crankshaft axis (3) and the balancing gear axis.

6. An internal combustion engine (1) according to any one of the preceding claims, wherein the eccentric member gear (9) and the balancing gear (11, 19) rotate in opposite direction with respect to each other under operating conditions.

7. An internal combustion engine (1) according to claim 6, wherein the eccentric member gear (5) meshes with the balancing gear (11, 19).

8. An internal combustion engine (1) according to claim 7, wherein the gear train (10) comprises an actuating gear (13) which is drivably coupled to the balancing gear (11, 19) and fixed to an actuating shaft (14), wherein the actuating shaft (14) is rotatably mounted to the crankshaft (3) and rotatable about an axis which coincides with the crankshaft axis (3), wherein under operating conditions the actuating shaft (14) stands still at fixed compression ratio.

9. An internal combustion engine (1) according to claim 8, wherein the balancing gear is a first stage gear (11) that is fixed to a second stage gear (12) and which has a larger diameter than the second stage gear (12), wherein the second stage gear (12) meshes with the actuating gear (13).

10. An internal combustion engine (1) according to any one of the preceding claims, wherein the crank plane and the balancing gear plane coincide.

11. An internal combustion engine (1) according to any one of the claims 1-5 and claim 8, wherein the balancing gear is an intermediate gear (19) which meshes with the actuating gear (13).

12. An internal combustion engine (1) according to claim 11, wherein the intermediate gear (19) extends beyond the eccentric member gear (9) and the actuating gear at least partly overlaps the eccentric member gear (9).

13. An internal combustion engine (1) according to claim 11 or 12, wherein the crank plane and the balancing gear plane are angled with respect to each other.

14. An internal combustion engine according to any one of the preceding claims, wherein the counter torque is smaller than the torque that is exerted by the eccentric member (5) onto the balancing gear (11, 19).

15. An internal combustion engine according to any one of the preceding claims, wherein the centre of gravity (11a) of the balancing gear (11) is created by a cavity (18) in the balancing gear (11).

Patentansprüche

1. Brennkraftmaschine (1) mit variablem Verdichtungsverhältnis, die aufweist:

   eine Kurbelwelle (2), die um eine Kurbelwellenachse (3) drehbar ist und einen Kurbelwellenzapfen (4) hat, wobei die Kurbelwellenachse (3) und eine Mittellinie des Kurbelwellenzapfens (4) eine Kurbelwellenebene aufspannen,
   eine Verbindungsstange (7),
   einen Kolben (8), der drehbar mit einem distalen Endabschnitt der Verbindungsstange (7) verbunden ist,
   ein exzentrisches Element (5), das drehbar auf dem Kurbelwellenstift (4) montiert ist und einen Lagerabschnitt (6) mit einer Außenumfangswand, die einen proximalen Endabschnitt der Verbindungsstange (7) trägt, aufweist, so dass die Verbindungsstange (7) drehbar auf dem Lagerabschnitt (6) montiert ist, wobei der Lagerabschnitt (6) in Bezug auf den Kurbelwellenstift (4) exzentrisch angeordnet ist und das exzentrische Element (5) mit einem exzentrischen Elementzahnrad (9) versehen ist, das antreibbar mit einem Rädertrieb (10) zum Drehen des exzentrischen Elements (5) in Bezug auf den Kurbelwellenstift (4) gekoppelt ist, **dadurch gekennzeichnet, dass** der Rädertrieb (10) ein Ausgleichszahnrad (11, 19) aufweist, das drehbar an der Kurbelwelle (2) montiert ist und sich unter Betriebsbedingungen in Bezug auf die Kurbelwelle (2) um eine Ausgleichszahnradachse mit der gleichen Geschwindigkeit wie das exzentrische Elementzahnrad (9) dreht, wobei die Kurbelwellenachse (3) und die Ausgleichszahnradachse eine Ausgleichszahnradebene aufspannen, wobei das Ausgleichszahnrad (11) einen exzentrischen Schwerpunkt (11a) hat, der

derart angeordnet ist, dass er unter Betriebsbedingungen aufgrund einer Trägheitskraft des exzentrischen Elements (5) ein Gegendrehmoment gegen ein Drehmoment, das von dem exzentrischen Elementzahnrad (9) auf das Ausgleichszahnrad (11, 19) ausgeübt wird, bewirkt.

2. Brennkraftmaschine (1) nach Anspruch 1, wobei der Rädertrieb (10) derart angepasst ist, dass sich das exzentrische Element (5) in Bezug auf den Kurbelwellenstift (4) mit der halben Geschwindigkeit der Geschwindigkeit der Kurbelwelle (2) und in die dazu entgegengesetzte Richtung dreht.

3. Brennkraftmaschine (1) nach Anspruch 1 oder 2, wobei das exzentrische Element (5) einen exzentrischen Schwerpunkt (5a) hat und die Trägheitskraft eine Zentrifugalkraft ist, die durch den exzentrischen Schwerpunkt (5a) des exzentrischen Elements (5) bewirkt wird.

4. Brennkraftmaschine (1) nach Anspruch 3, wobei der Schwerpunkt (5a) des exzentrischen Elements (5) in einer Ebene liegt, die von der Mittellinie des Kurbelstifts (4) und einer Mittellinie des Lagerabschnitts (6) aufgespannt wird, wobei das exzentrische Element (5) und das Ausgleichszahnrad (11, 19) derart angeordnet sind, dass der Schwerpunkt (11a) des Ausgleichszahnrads (11, 19) in der Ausgleichszahnradebene liegt, wenn der Schwerpunkt (5a) des exzentrischen Elements (5) unter Betriebsbedingungen in der Kurbelwellenebene liegt.

5. Brennkraftmaschine (1) nach Anspruch 4, wobei der Schwerpunkt (11a) des Ausgleichszahnrads (11, 19) zwischen der Kurbelwellenachse (3) und der Ausgleichszahnradachse liegt, wenn der Schwerpunkt (5a) des exzentrischen Elements (5) unter Betriebsbedingungen in der Kurbelwellenebene liegt und die Mittellinie des Kurbelwellenstifts (4) zwischen der Kurbelwellenachse (3) und dem Schwerpunkt (5a) des exzentrischen Elements (5) liegt.

6. Brennkraftmaschine (1) nach einem der vorhergehenden Ansprüche, wobei das exzentrische Elementzahnrad (9) und das Ausgleichszahnrad (11, 19) sich in Bezug aufeinander unter Betriebsbedingungen in entgegengesetzte Richtungen drehen.

7. Brennkraftmaschine (1) nach Anspruch 6, wobei das exzentrische Elementzahnrad (5) mit dem Ausgleichszahnrad (11, 19) in Eingriff steht.

8. Brennkraftmaschine (1) nach Anspruch 7, wobei der Rädertrieb (10) ein Betätigungszahnrad (13) aufweist, das antreibbar mit dem Ausgleichszahnrad (11, 19) gekoppelt und an einer Betätigungswelle (14) fixiert ist, wobei die Betätigungswelle (14) drehbar an der Kurbellwelle (3) montiert ist und um eine Achse, die mit der Kurbelwellenachse (3) zusammenfällt, drehbar ist, wobei die Betätigungswelle (14) unter Betriebsbedingungen bei festen Verdichtungsverhältnis stellsteht.

9. Brennkraftmaschine (1) nach Anspruch 8, wobei das Ausgleichszahnrad ein Zahnrad (11) der ersten Stufe ist, das an einem Zahnrad (12) der zweiten Stufe fixiert ist und das einen größeren Durchmesser als das Zahnrad (12) der zweiten Stufe hat, wobei das Zahnrad (12) der zweiten Stufe mit dem Betätigungszahnrad (13) in Eingriff steht.

10. Brennkraftmaschine (1) nach einem der vorhergehenden Ansprüche, wobei die Kurbelwellenebene und die Ausgleichszahnradebene zusammenfallen.

11. Brennkraftmaschine (1) nach einem der Ansprüche 1 - 5 und Anspruch 8, wobei das Ausgleichszahnrad ein Zwischenzahnrad (19) ist, das mit dem Betätigungszahnrad (13) in Eingriff steht.

12. Brennkraftmaschine (1) nach Anspruch 11, wobei das Zwischenzahnrad (19) sich über das exzentrische Elementzahnrad (9) hinaus erstreckt und das Betätigungszahnrad das exzentrische Elementzahnrad (9) wenigstens teilweise überlappt.

13. Brennkraftmaschine (1) nach Anspruch 11 oder 12, wobei die Kurbelwellenebene und die Ausgleichszahnradebene in einem Winkel aufeinander stehen.

14. Brennkraftmaschine nach einem der vorhergehenden Ansprüche, wobei das Gegendrehmoment kleiner als das Drehmoment ist, das von dem exzentrischen Element (5) auf das Ausgleichzahnrad (11, 19) ausgeübt wird.

15. Brennkraftmaschine nach einem der vorhergehenden Ansprüche, wobei der Schwerpunkt (11a) des Ausgleichszahnrads (11) durch einen Hohlraum (18) in dem Ausgleichszahnrad (11) erzeugt wird.

Revendications

1. Moteur à combustion interne (1) comportant un taux de compression variable comprenant

un vilebrequin (2) qui est rotatif autour d'un axe de vilebrequin (3) et ayant un maneton (4), dans lequel l'axe de vilebrequin (3) et une ligne centrale du maneton (4) délimitent un plan de vilebrequin,
une bielle (4),
un piston (8) qui est relié de manière rotative à une partie d'extrémité distale de la bielle (4) ;

un organe excentrique (5) qui est monté de manière rotative sur le maneton (4) et comprenant une partie d'appui (6) ayant une paroi circonférentielle externe sur laquelle s'appuie une partie d'extrémité proximale de la bielle (4) de sorte que la bielle (4) soit montée de manière rotative sur la partie d'appui (6), dans lequel la partie d'appui (9) est disposée de manière excentrique par rapport au maneton (4) et l'organe excentrique (5) est muni d'un pignon d'organe excentrique (9) qui est couplé en entraînement à un train d'engrenages (10) pour la mise en rotation de l'organe excentrique (5) par rapport au maneton (4), **caractérisé en ce que** le train d'engrenages (10) comprend un pignon d'équilibrage (11, 19) qui est monté de manière rotative sur le vilebrequin (2) et est en rotation par rapport au vilebrequin (2) autour d'un axe de pignon d'équilibrage à la même vitesse que le pignon d'organe excentrique (9) dans des conditions de fonctionnement, dans lequel l'axe de vilebrequin (3) et l'axe de pignon d'équilibrage délimitent un plan de pignon d'équilibrage, dans lequel le pignon d'équilibrage (11) possède un centre de gravité (11a) excentrique qui est situé de sorte que, dans des conditions de fonctionnement, il provoque un contre-couple à l'encontre d'un couple qui est exercé par le pignon d'organe excentrique (9) sur le pignon d'équilibrage (11, 19) en raison d'une force d'inertie sur l'organe excentrique (5).

2. Moteur à combustion interne (1) selon la revendication 1, dans lequel le train d'engrenages (10) est adapté de sorte que l'organe excentrique (5) soit en rotation par rapport au maneton (4) à la mi-vitesse de la vitesse du vilebrequin (2) et dans la direction opposée de celui-ci.

3. Moteur à combustion interne (1) selon la revendication 1 ou 2, dans lequel l'organe excentrique (5) possède un centre de gravité (5a) excentrique et la force d'inertie est une force centrifuge provoquée par le centre de gravité (5a) excentrique de l'organe excentrique (5).

4. Moteur à combustion interne (1) selon la revendication 3, dans lequel le centre de gravité (5a) de l'organe excentrique (5) se trouve dans un plan qui est délimité par la ligne centrale du maneton (4) et une ligne centrale de la partie d'appui (6), dans lequel l'organe excentrique (5) et le pignon d'équilibrage (11, 19) sont agencés de sorte que lorsque, dans des conditions de fonctionnement, le centre de gravité (5a) de l'organe excentrique (5) se trouve dans le plan de vilebrequin, le centre de gravité (11a) du pignon d'équilibrage (11, 19) se trouve dans le plan de pignon d'équilibrage.

5. Moteur à combustion interne (1) selon la revendication 4, dans lequel lorsque, dans des conditions de fonctionnement, le centre de gravité (5a) de l'organe excentrique (5) se trouve dans le plan de vilebrequin et la ligne centrale du maneton (4) se trouve entre l'axe de vilebrequin (3) et le centre de gravité (5a) de l'organe excentrique (5), le centre de gravité (11a) du pignon d'équilibrage (11, 19) se trouve entre l'axe de vilebrequin (3) et l'axe de pignon d'équilibrage.

6. Moteur à combustion interne (1) selon l'une quelconque des revendications précédentes, dans lequel le pignon d'organe excentrique (9) et le pignon d'équilibrage (11, 19) sont en rotation dans une direction opposée l'un par rapport à l'autre dans des conditions de fonctionnement.

7. Moteur à combustion interne (1) selon la revendication 6, dans lequel le pignon d'organe excentrique (5) s'engrène avec le pignon d'équilibrage (11, 19).

8. Moteur à combustion interne (1) selon la revendication 7, dans lequel le train d'engrenages (10) comprend un pignon d'actionnement (13) qui est couplé en entraînement au pignon d'équilibrage (11, 19) et fixé à un arbre d'actionnement (14), dans lequel l'arbre d'actionnement (14) est monté de manière rotative sur le vilebrequin (3) et est rotatif autour d'un axe qui coïncide avec l'axe de vilebrequin (3), dans lequel, dans des conditions de fonctionnement, l'arbre d'actionnement (14) reste immobile à un taux de compression fixe.

9. Moteur à combustion interne (1) selon la revendication 8, dans lequel le pignon d'équilibrage est un pignon de premier étage (11) qui est fixé à un pignon de second étage (12) et qui possède un plus grand diamètre que le pignon de second étage (12), dans lequel le pignon de second étage (12) s'engrène avec le pignon d'actionnement (13).

10. Moteur à combustion interne (1) selon l'une quelconque des revendications précédentes, dans lequel le plan de vilebrequin et le plan de pignon d'équilibrage coïncident.

11. Moteur à combustion interne (1) selon l'une quelconque parmi les revendications 1 à 5 et la revendication 8, dans lequel le pignon d'équilibrage est un pignon intermédiaire (19) qui s'engrène avec le pignon d'actionnement (13).

12. Moteur à combustion interne (1) selon la revendication 11, dans lequel le pignon intermédiaire (19) s'étend au-delà du pignon d'organe excentrique (9) et le pignon d'actionnement chevauche au moins partiellement le pignon d'organe excentrique (9).

13. Moteur à combustion interne (1) selon la revendication 11 ou 12, dans lequel le plan de vilebrequin et le plan de pignon d'équilibrage sont inclinés l'un par rapport à l'autre.

14. Moteur à combustion interne selon l'une quelconque des revendications précédentes, dans lequel le contre-couple est inférieur au couple qui est exercé par l'organe excentrique (5) sur le pignon d'équilibrage (11, 19).

15. Moteur à combustion interne selon l'une quelconque des revendications précédentes, dans lequel le centre de gravité (11a) du pignon d'équilibrage (11) est créé par une cavité (18) dans le pignon d'équilibrage (11).

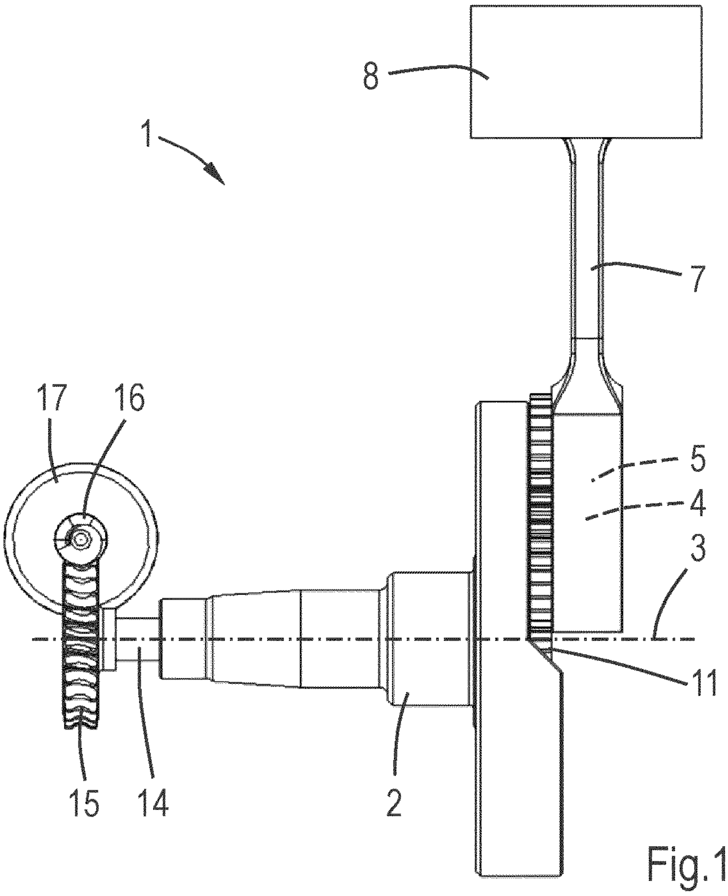

Fig.1

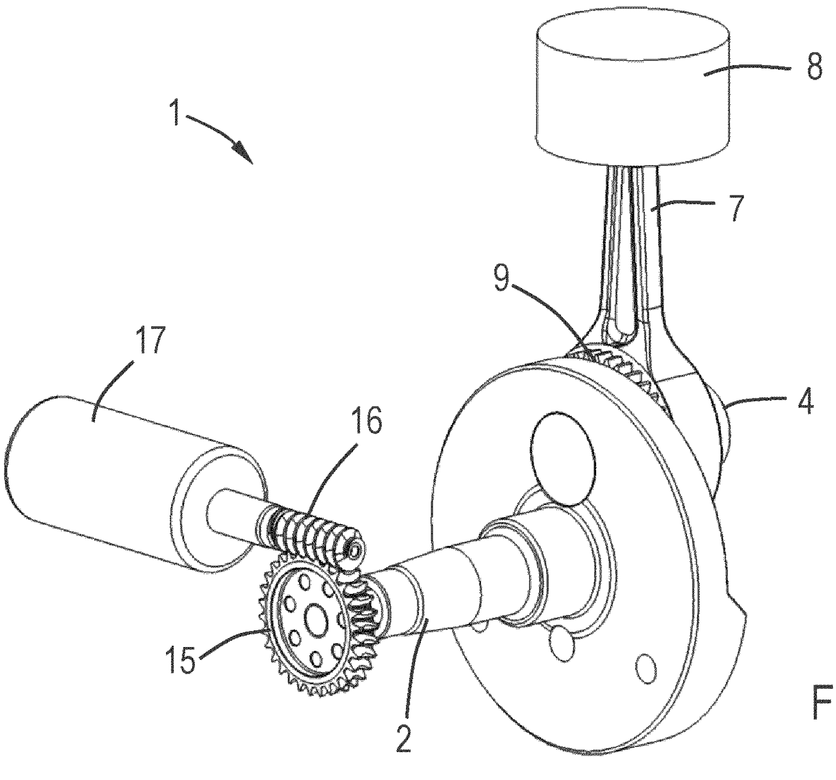

Fig.2

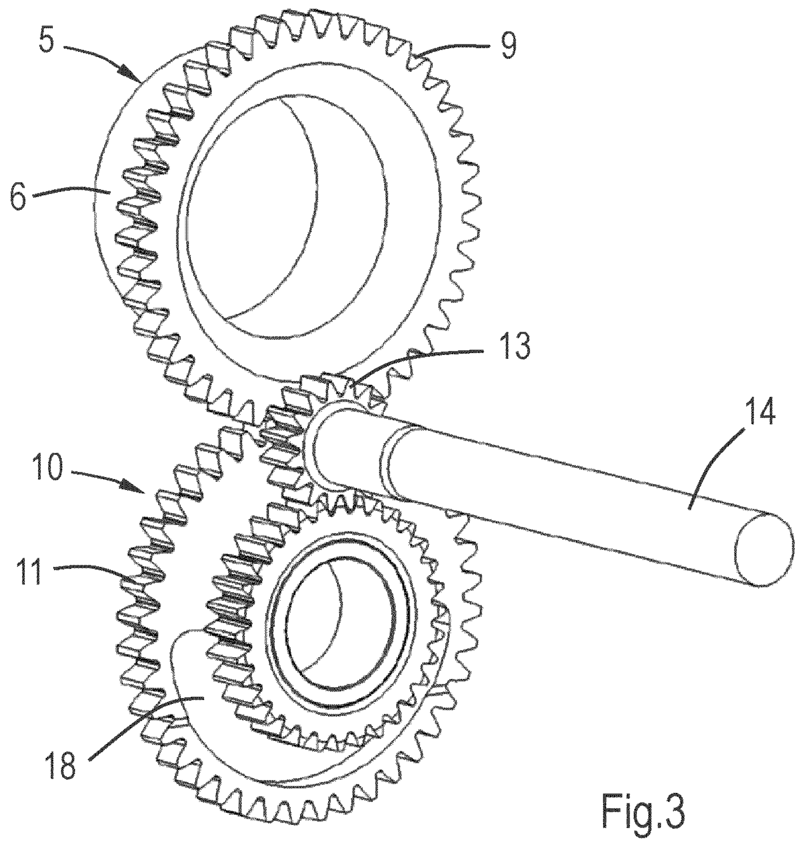

Fig.3

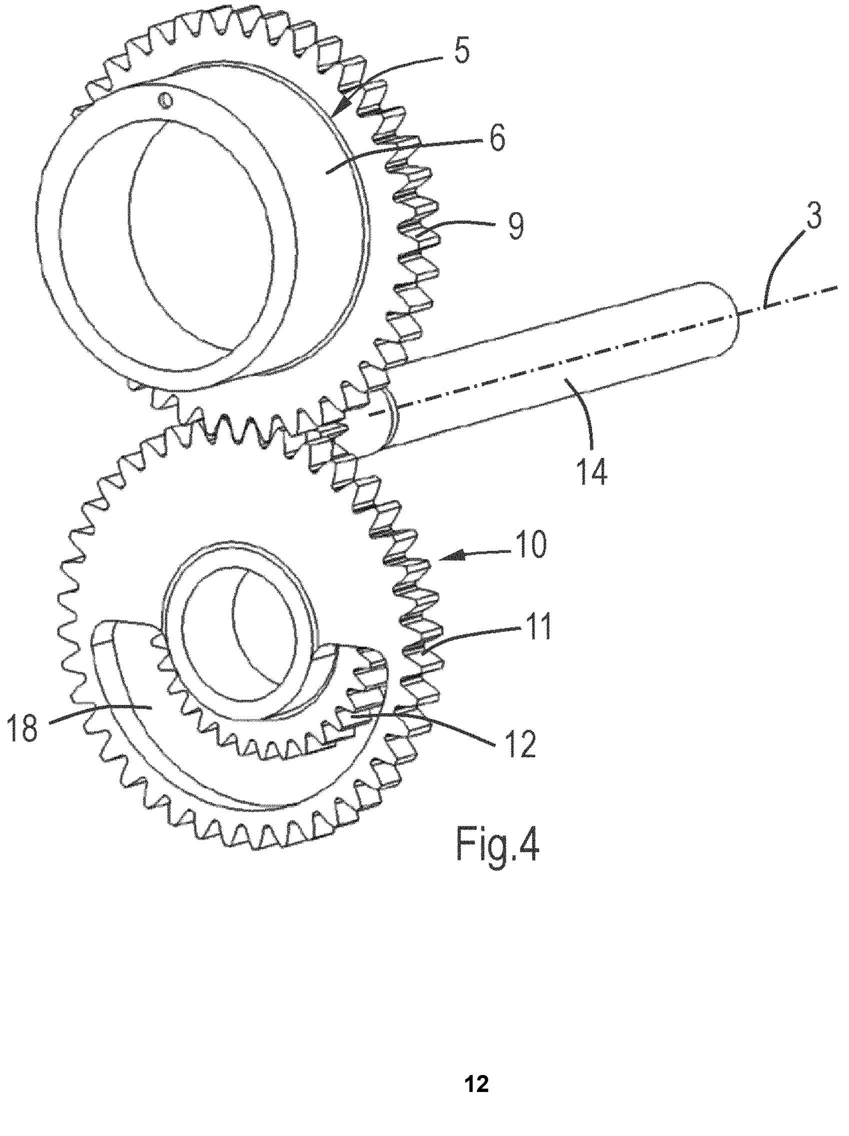

Fig.4

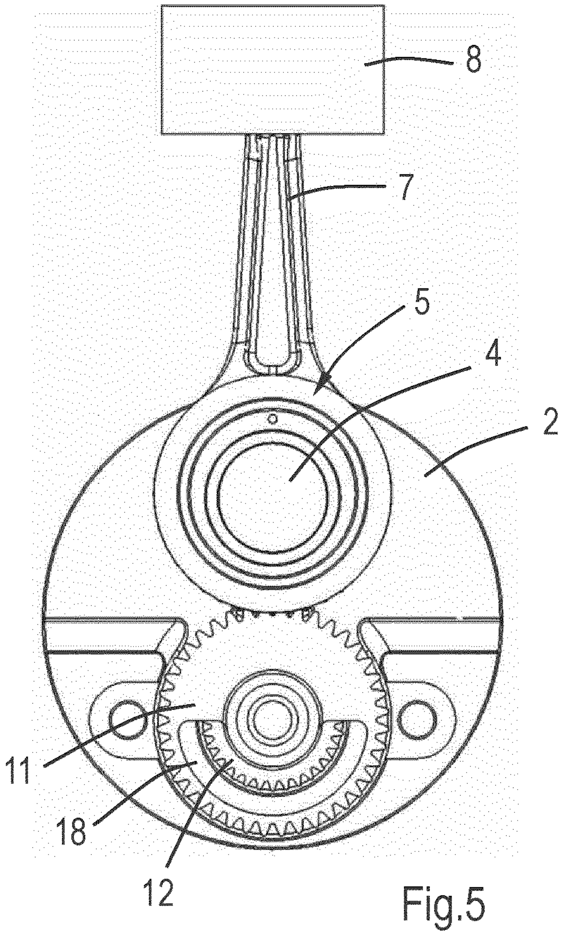

Fig.5

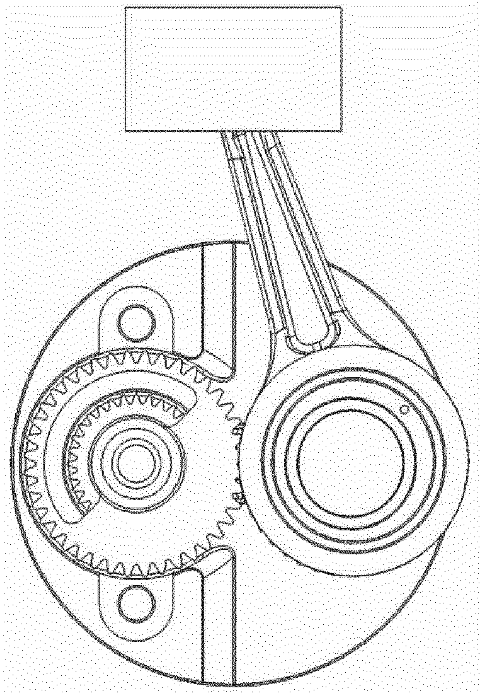

Fig.6

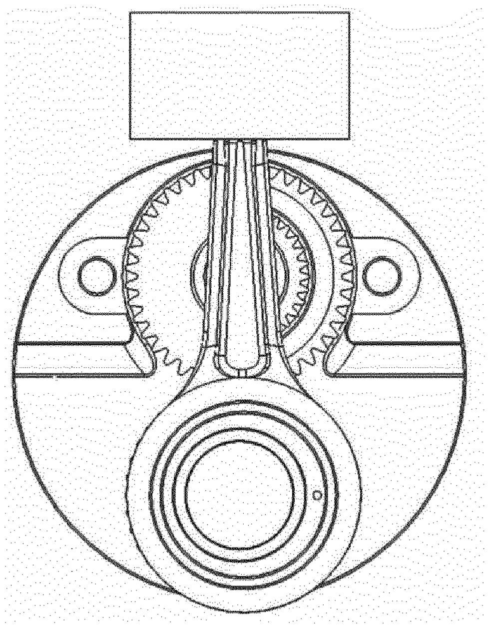

Fig.7

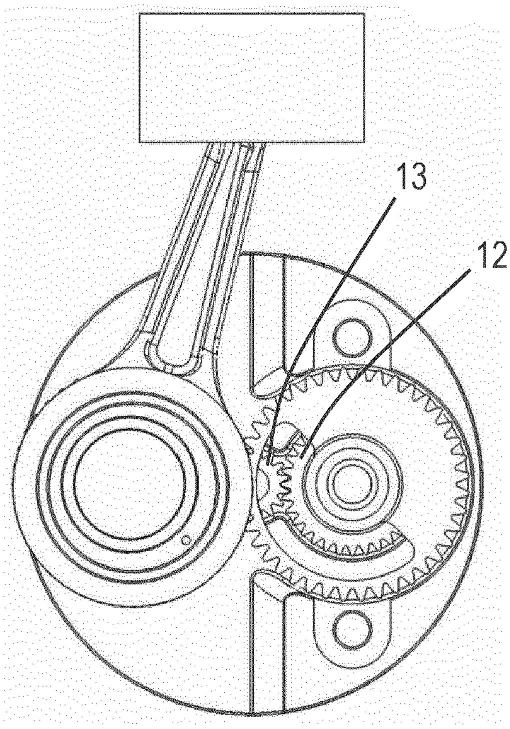

Fig.8

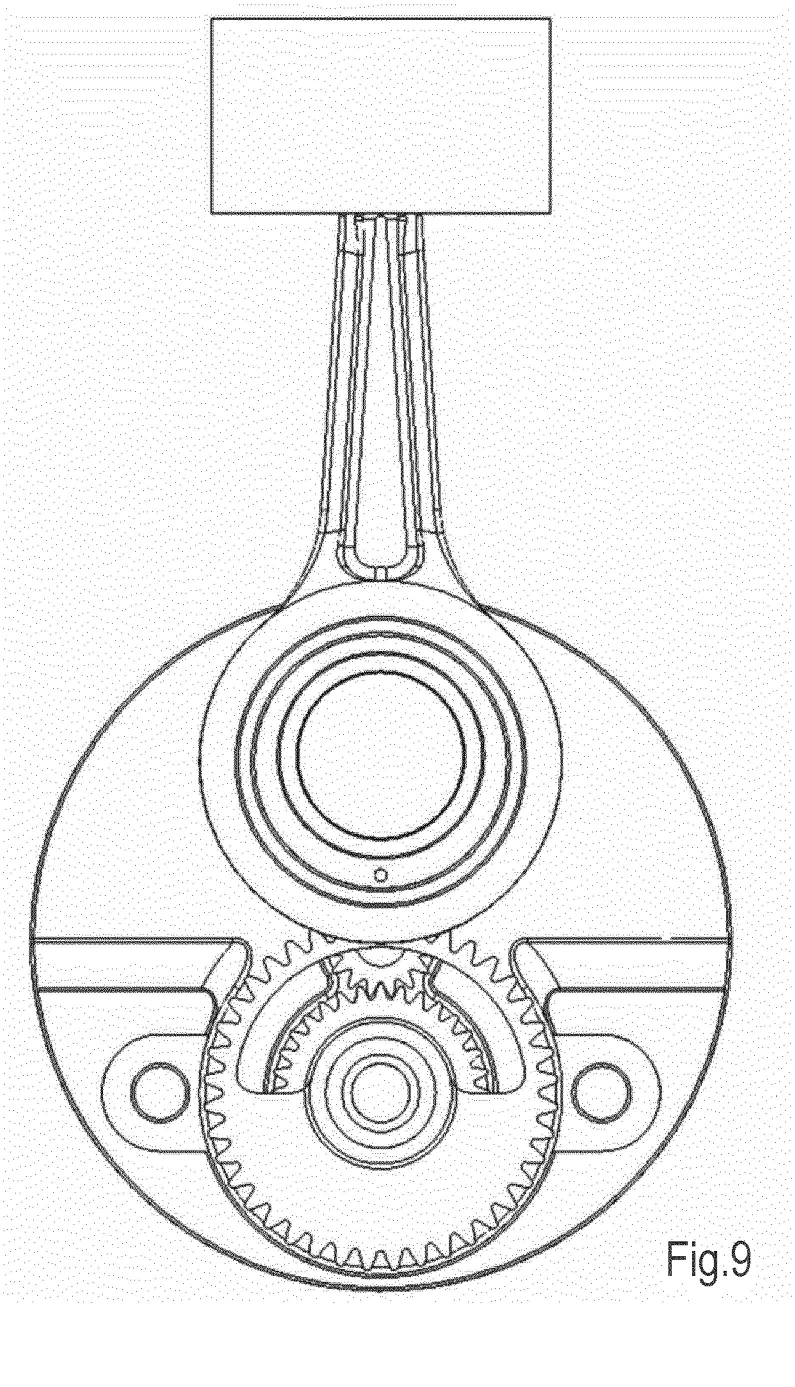
Fig.9

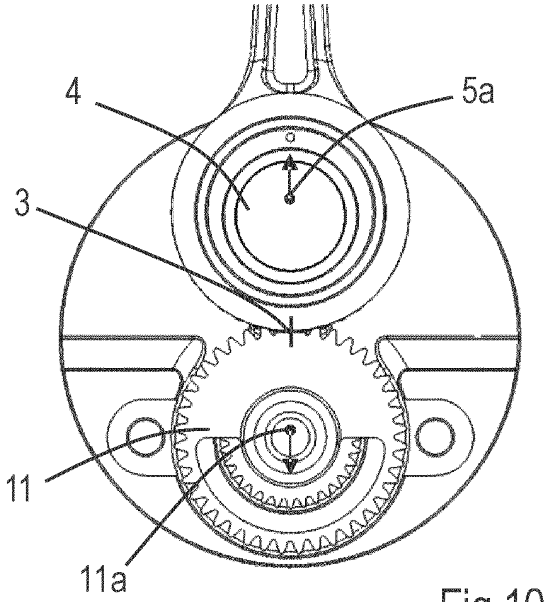

Fig.10

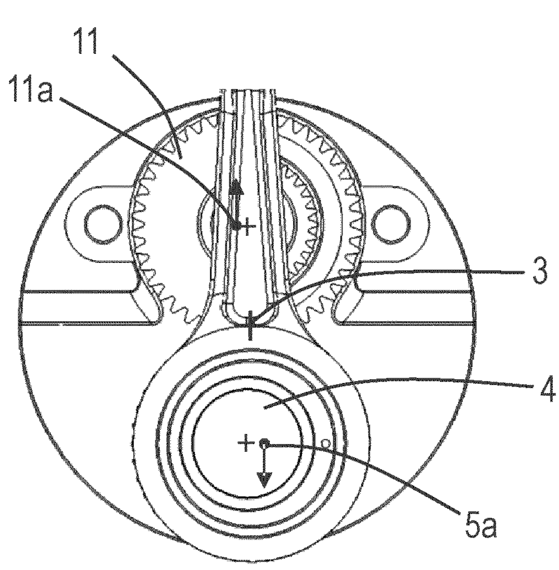

Fig.11

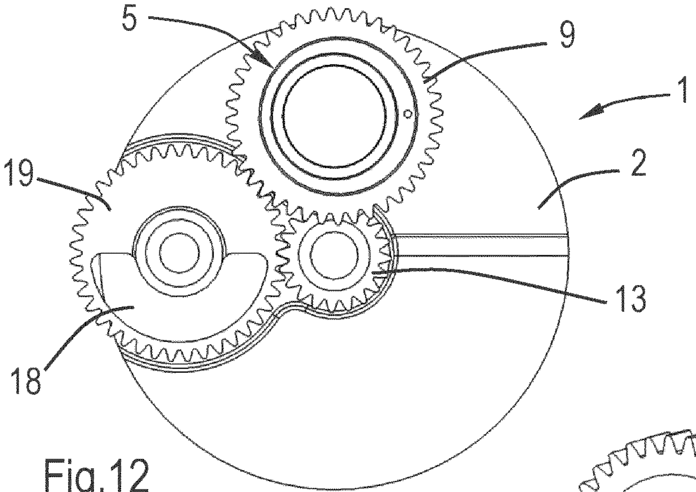

Fig.12

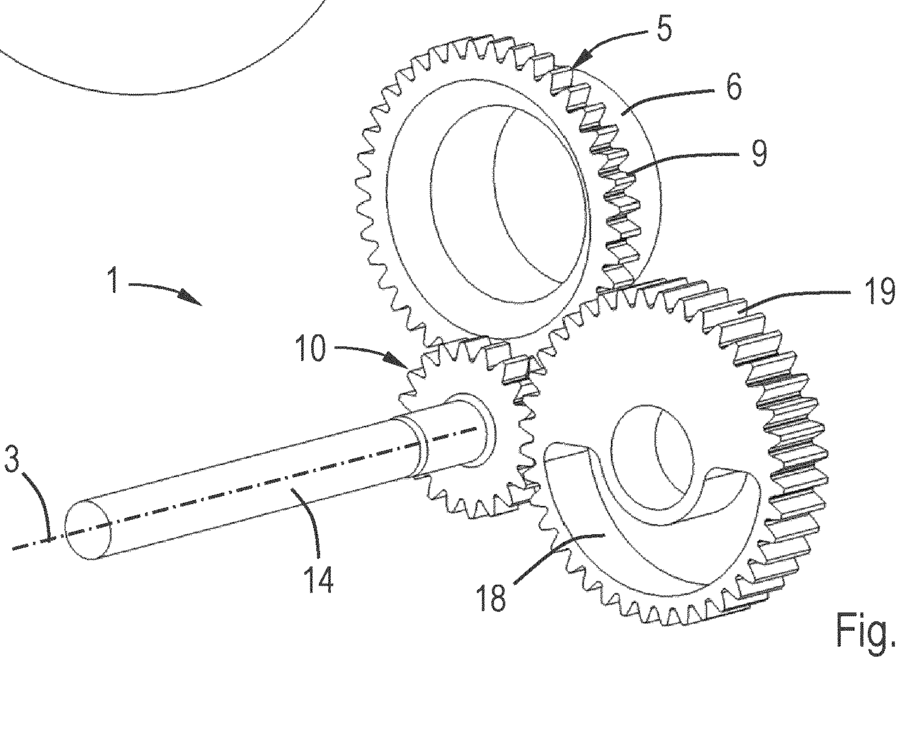

Fig.13

REFERENCES CITED IN THE DESCRIPTION

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