(54) **Internal combustion engine with 'V' configuration of the cylinders**

(57) An internal combustion engine (1) having two cylinders (2, 3), which are arranged in "V" configuration and accommodate respective pistons (4, 5), and a first balancing body (12), which is fixed to a crankshaft (6) at a first piston (4) so that the first order reciprocating inertial forces of the first piston (4) lie on a first straight line (L1) forming a first inclination angle (β1) which is not zero with the first cylinder (2); the combination of an angle (α) between the cylinders with the first inclination angle (β1) is such that the first straight line (L1) on which the first order reciprocating inertial forces generated by the first piston (4) lies is parallel to a second straight line (L2) on which the first order reciprocating inertial forces generated by the reciprocating motion of a second piston (5) lie.
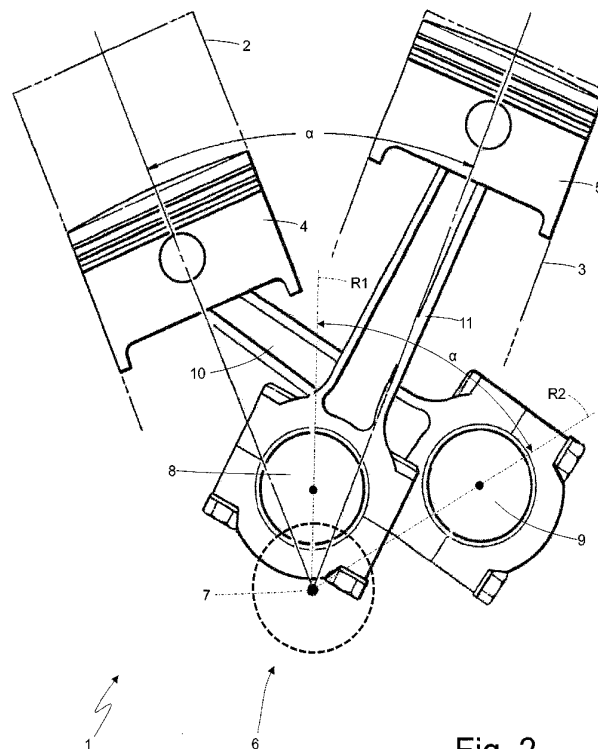
Fig. 2

EP 2 679 782 A1

Description

TECHNICAL FIELD

[0001] The present invention relates to an internal combustion engine with "V" configuration of the cylinders.

PRIOR ART

[0002] The main vibrations of an internal combustion engine are not caused by the combustion of the mixture in the cylinders (even if the crankcase is particularly rigid) and are not even caused by the rotation of the crankshaft (which is balanced by means of specific counterweights or special shapes), but are determined by the reciprocating motion of the pistons which accelerate under the bias of the gases, slow down suddenly and pick up speed again reversing their direction of motion, i.e. are determined by the so-called reciprocating inertial forces. The piston causes the most violent shuddering (i.e. the maximum intensity of the reciprocating inertial forces) when the direction of motion is reversed (i.e. when its acceleration/deceleration is maximum), but reciprocating inertial forces displaying variable intensity over time with a sinusoidal law are present in all instants of motion.

[0003] Reciprocating inertial forces generate vibrations which are detrimental for the engine (because they can cause failure by fatigue over time) and are also annoying for vehicle occupants. In order to reduce such vibrations, the internal combustion engine must be balanced, i.e. by using balancing countershafts (also known as balancing shafts), which allow to reduce (compensate) the reciprocating inertial forces. However, the use of balancing countershafts gives rise to various drawbacks due to the weight and dimensions of the balancing countershafts, the weight and dimensions of the drive needed to turn the balancing countershafts and the greater weight of the engine crankcase which must also support the stress induced by the balancing countershafts (the balancing countershafts generate further reciprocating inertial forces which are transmitted to the engine crankcase and cancel out the reciprocating inertial forces of the pistons in the engine crankcase itself).

[0004] Patent US4517933A1 describes an internal combustion engine with a "V" configuration of the cylinders, wherein the crankshaft is provided with balancing masses 31a, 31b and 31c to reduce the vibrations generated by the reciprocating motion of the pistons.

DESCRIPTION OF THE INVENTION

[0005] It is the object of the present invention to provide an internal combustion engine with "V" configuration of the cylinders, which is easy and cost-effective to make and free from the aforesaid drawbacks.

[0006] According to the present invention, an internal combustion engine with "V" configuration of the cylinders is provided as claimed in the accompanying claims.

BRIEF DESCRIPTION OF THE DRAWINGS

[0007] The present invention will now be described with reference to the accompanying drawings, which show a non-limitative embodiment thereof, in which:

- figure 1 is a diagrammatic view of a vehicle provided with an internal combustion engine made according to the present invention;
- figure 2 is a diagrammatic view of the internal combustion engine in figure 1;
- figure 3 is a side view of a crankshaft of the internal combustion engine in figure 2; and
- figure 4 is a cross section view of the crankshaft in figure 3;
- figures 5-9 are a sequence of diagrammatic views which illustrate how the reciprocating inertial forces generated by the reciprocating motion of a piston are modified as the position and mass of a balancing body applied to the crankshaft varies.

PREFERRED EMBODIMENTS OF THE INVENTION

[0008] In figures 1 and 2, reference numeral 1 indicates as a whole an internal combustion engine, which is provided with two cylinders 2 and 3 arranged in "V" configuration to form an angle α between cylinders which is not equal to zero (by way of example, in the embodiment shown in figure 1, the angle α between the cylinders is equal to 52°). A piston 4 which runs along the axis of the cylinder 2 is mounted so as to slide inside cylinder 2, while a piston 5 which runs along the axis of the cylinder 3 is mounted so as to slide in cylinder 3. The internal combustion engine 1 comprises a crankshaft 6, which is mounted so as to rotate around a rotation axis 7 and is provided with a crank shape, so as to have a corresponding crank 8, 9 for each piston 4, 5. The piston 4 is mechanically connected to the crank 8 of the crankshaft 6 by means of a connecting rod 10 which has one end hinged to the piston 4 and an opposite end hinged to the crank 8; similarly, the piston 5 is mechanically connected to the crank 9 of the crankshaft 6 by means of a connecting rod 11 which has one end hinged to the piston 5 and an opposite end hinged to the crank 9.

[0009] A balancing body 12 (shown in figure 3) is fixed to the crankshaft 6 at the piston 4, and in particular at the crank 8 of the piston 4. The balancing body 12 (as explained in greater detail below) is such that the first order reciprocating inertial forces generated by the reciprocating motion of the piston 4 lie along a straight line L1 (shown in figure 1) which forms an inclination angle $\beta 1$ which is not equal to zero with the cylinder 2 (i.e. with the axis of the cylinder 2 along which the piston 4 moves with reciprocating motion). In the embodiment shown in figure 1, the inclination angle $\beta 1$ is equal to +26° (i.e. 26° clockwise).

[0010] Similarly, a balancing body 13 (shown in figure 3) is fixed to the crankshaft 6 at the piston 5, and in par-

ticular at the crank 9 of the piston 5. The balancing body 13 (as explained in greater detail below) is such that the first order reciprocating inertial forces generated by the reciprocating motion of the piston 5 lie along a straight line L2 (shown in figure 1) which forms an inclination angle β2 which is not equal to zero with the cylinder 3 (i.e. with the axis of the cylinder 3 along which the piston 5 moves with reciprocating motion). In the embodiment shown in figure 1, the inclination angle β2 is equal to -26° (i.e. 26° counterclockwise).

[0011]   The combination of the angle α between the cylinders and the two inclination angles β1 and β2 is such that the first straight line L1, on which the first order reciprocating inertial forces generated by the reciprocating motion of the first piston 4 lie, is parallel to a second straight line L2, on which the first order reciprocating inertial forces generated by the reciprocating motion of a second piston 5 lie. Indeed, the two cylinders 2 and 3 are distanced 52° apart (i.e. the angle α between the cylinders is equal to 52°), the straight line L1 is shifted clockwise (i.e. towards the cylinder 3) by 26° (i.e. the inclination angle β1 is equal to +26°) and the straight line L1 is moved counterclockwise (i.e. towards the cylinder 2) by 26° (i.e. the inclination angle β2 is equal to -26°); consequently, the two straight lines L1 and L2 are parallel to each other.

[0012]   The crank angles of the two cylinders 2 and 3 are such that the first order reciprocating inertial forces generated by the reciprocating motion of the piston 4 and the first order reciprocating inertial forces generated by the reciprocating motion of the piston 5 are in phase opposition; in this respect, it is worth noting that after having defined the angle α between the cylinders it is simple (i.e. by applying a simple geometric calculation) to establish the crank angles so that the first order reciprocating inertial forces generated by the reciprocating motion of the piston 4 and the first order reciprocating inertial forces generated by the reciprocating motion of the piston 5 are in phase opposition.

[0013]   In this manner, the first order reciprocating inertial forces generated by the reciprocating motion of the two pistons 4 and 5 are parallel to each other (i.e. lie on two mutually parallel straight lines L1 and L2) and are also in phase opposition; consequently, the first order reciprocating inertial forces generating the reciprocating motion of the two pistons 4 and 5 always cancel each other out.

[0014]   In order to make that the first order reciprocating inertial forces generated by the reciprocating motion of a piston 4 or 5 lie on a straight line L1 or L2 forming an inclination angle β1 or β2 which is not zero with the cylinder 2 or 3, it is possible to shift the balancing body 12 or 13 out of phase with respect to the corresponding crank 8 or 9 (as shown in figure 4), i.e. it is possible to arrange the balancing body 12 or 13 so that the radius R1 (shown in figure 4) connecting the barycentre B of the balancing body 12 or 13 to the rotation axis 7 of the crankshaft 6 forms a phase angle γ1 or γ2 which is not zero with the prolongation of the radius R2 connecting the crank 8 or

9 of the piston 4 or 5 to the rotation axis 7. In other words, in a conventional internal combustion engine, the balancing body and the corresponding crank (i.e. the barycentre of the balancing body are in perfect phase opposition along the prolongation of the radius connecting the crank of the piston to the rotation axis of the crankshaft); instead, in the internal combustion engine 1 according to the present invention, the barycentre B of the balancing body 12 or 13 is not along the prolongation of the radius R2 connecting the crank 8 or 9 of the piston 4 or 5 to the rotation axis 7, but is "shifted out of phase" by a phase angle γ1 or γ2 which is not zero with respect to the prolongation of the radius R2 connecting the crank 8 or 9 of the piston 4 or 5 to the rotation axis 7.

[0015]   It is worth noting that, as clearly shown in figure 4, the two radii R1 and R2 mutually form an angle α identical to the angle α between cylinders 2 and 3 (i.e. an angle α equal to 52° in the embodiment shown in the accompanying figures).

[0016]   Furthermore, in order to make the first order reciprocating inertial forces generated by the reciprocating motion of a piston 4 or 5 lie along a straight line L1 or L2 forming an inclination angle β1 or β2 which is not zero with the cylinder 2 or 3, it is necessary to combine the phase angle γ1 or γ2 which is not zero with a particular mass value of the balancing body 12 or 13. According to a preferred embodiment, the phase angles γ1 and γ2 are equal to ±64° (i.e. to 64° clockwise or counterclockwise), while the balancing body 12 or 13 has a mass equal to 43.8% of the equivalent mass which generates the reciprocating inertial forces of the piston 4 or 5 (i. e. of the entire mass of the piston 4 or 5, and of the part of the mass of the connecting rod 10 or 11). At this time, we are not aware of other phase angle γ1 or γ2 and balancing body 12 or 13 mass combinations (i.e. other than ±64° and 43.8% of the equivalent mass, respectively) which allow to make the first order reciprocating inertial forces lie along a straight line which is inclined by an inclination angle β1 or β2 which is not zero and not straight with respect to the corresponding cylinder 2 or 3 (i.e. with respect to the axis of the corresponding cylinder 2 or 3); however, we cannot exclude beforehand the existence of other phase angle γ1 or γ2 and balancing body mass 12 or 13 combinations which allow to make the first order reciprocating inertial forces lie along an inclination angle β1 or β2 which is not zero with respect to the corresponding cylinder 2 or 3 (i.e. with respect to the axis of the corresponding cylinder 2 or 3).

[0017]   In order to better understand the present invention, the method used to modify the reciprocating inertial forces F generated by the reciprocating motion of a piston 4 or 5 as the position and mass of a balancing body 12 or 13 applied to the crankshaft 6 at the respective crank 10 or 11 varies is explained below with reference to figures 3-7.

[0018]   As shown in figure 5, when the shaft 6 is free from the balancing bodies, the reciprocating inertial forces F generated by the reciprocating motion of the piston

4 or 5 lie on a straight line parallel to axis z (i.e. to the axis of the cylinder 2 or 3).

[0019] As shown in figure 6, when the shaft 6 comprises a balancing body 12 or 13 having a balancing mass equal to 100% of the balancing mass which generates the reciprocating inertial forces F in perfect phase opposition with the corresponding crank 8 or 9 (i.e. with a phase angle γ1 or γ2 which is not zero), the reciprocating inertial forces F generated by the reciprocating motion of the piston 4 or 5 lie on a line parallel to axis y (i.e. are turned by 90° with respect to the case shown in figure 4) which is perpendicular to axis z (i.e. to the axis of cylinder 2 or 3).

[0020] As shown in figure 7, when the shaft 6 comprises a balancing body 12 or 13 having a mass equal to 50% of the equivalent mass which generates the reciprocating inertial forces F arranged in perfect phase opposition with the corresponding crank 8 or 9 (i.e. with a phase angle γ1 or γ2 which is zero), the reciprocating inertial forces F generated by the reciprocating motion of the piston 4 or 5 lie on a circumference.

[0021] As shown in figure 9, when the shaft 6 comprises a balancing body 12 or 13 having a mass equal to 50% of the equivalent mass which generates the reciprocating inertial forces F arranged at a phase angle γ1 or γ2 of +45°, the reciprocating inertial forces F generated by the reciprocating motion of the piston 4 or 5 lie on an ellipse the main axis of which is inclined by -34° with respect to axis z (i.e. with respect to the axis of cylinder 2 or 3).

[0022] As shown in figure 9, when the shaft 6 comprises a balancing body 12 or 13 having a mass equal to 43.8% of the equivalent mass which generates the reciprocating inertial forces F arranged at a phase angle γ1 or γ2 of +64°, the reciprocating inertial forces F generated by the reciprocating motion of the piston 4 or 5 lie on a straight line L1 or L2 which is inclined by -26° with respect to axis z. As previously mentioned, at this time, this is the only known balancing body 12 or 13 mass and phase angle γ1 or γ2 combination which allows to make the reciprocating inertial forces F generated by the reciprocating motion of the piston 4 or 5 lie on a straight line L1 or L2 which is inclined by -26° with respect to the axis z (i.e. with respect to the axis of the cylinder 2 or 3).

[0023] In the embodiment shown in the accompanying figures, the angle α between the cylinders is equal to 52° and both pistons 4 and 5 are coupled to balancing masses 12 and 13 which are shifted out of phase with respect to the corresponding cranks 8 and 9 (i.e. are arranged with phase angles γ1 and γ2 which are not zero with respect to the corresponding cranks 8 and 9).

[0024] According to alterative embodiments (not shown and perfectly equivalent), the angle α between the cylinders can be modified by reversing either only one of the two phase angles γ1 and γ2 or both phase angles γ1 and γ2 which are not zero; indeed, as mentioned above, the angle α between the cylinders must always be such to allow the parallelism between the two straight lines L1 and L2, and by varying the phase angles γ1 and γ2 (i.e. by changing the sign of the phase angles γ1 and γ2) the inclination angles β1 and β2 which determine the position of the straight lines L1 and L2 vary as a consequence.

[0025] According to further embodiments (not shown and perfectly equivalent), only the piston 4 is coupled to the balancing body 12, which is shifted out of phase with respect to the corresponding crank 8 (i.e. which is arranged by a phase angle γ1 which is not zero with respect to the corresponding crank 8), while the balancing body 13 coupled to the piston 5 is not present.

[0026] By varying the phase angles γ1 and γ2 by using the balancing body 13 or not, the angle α between the cylinders may assume the following values: 26°, 52° (as shown in the accompanying figures), 64°, 116°, 128°, 154°; such a limitation of the angle α between the cylinders is, as mentioned above, determined by the fact that the parallelism must always be guaranteed between the two straight lines L1 and L2.

[0027] In the embodiment described above, the internal combustion engine 1 comprises only two cylinders 2 and 3 in "V" configuration; according to other embodiments (not shown and perfectly equivalent), the internal combustion engine 1 comprises a higher number of cylinders 2 and 3 in "V" configuration, and in particular a higher number of pairs of cylinders 2 and 3 in "V" configuration (e.g. four, six, eight or ten cylinders 2 and 3).

[0028] The internal combustion engine 1 described above has many advantages, because it allows to perfectly balance (i.e. to perfectly cancel out) the first order reciprocating inertial forces (i.e. which have the same frequency as the rotation of the crankshaft 6) generated by the reciprocating motion of the pistons 4 and 5 without using balancing countershafts but only by working on the balancing masses 12 and 13 added to the crankshaft 6.

[0029] In the internal combustion engine 1 described above, the second order reciprocating inertial forces (i.e. those which have a frequency double that of the rotation frequency of the crankshaft 6) are not balanced (but only partially reduced); however, the second order reciprocating inertial forces have a modulus which is approximately one third of the first order reciprocating inertial forces, and thus the negative effect is much lower than that of the first order reciprocating inertial forces.

Claims

1. An internal combustion engine (1) comprising:

   at least two cylinders (2, 3), which are arranged in a "V", so as to form an angle (α) that is not equal to zero between the cylinders (2, 3);
   two pistons (4, 5), each of which is mounted so as to slide inside a corresponding cylinder (2, 3);
   a crankshaft (6), which is mounted so as to rotate around a rotation axis (7) and is provided with a

crank shape, so as to present, for each piston (4, 5), a corresponding crank (8, 9);

two connecting rods (10, 11), each of which presents an end that is hinged to a corresponding piston (4, 5) and an opposite end that is hinged to a corresponding crank (8, 9); and

a first balancing body (12), which is fixed to the crankshaft (6) in correspondence to a first piston (4);

the internal combustion engine is **characterised in that** the first balancing body (12) is such that the first order reciprocating inertial forces generated by the reciprocating motion of the first piston (4) lie along a first straight line (L1) forming a first inclination angle ($\beta1$) that is not equal to zero with the first cylinder (2);

the combination of the angle (α) between the cylinders with the first inclination angle ($\beta1$) is such that the first straight line (L1), on which the first order reciprocating inertial forces generated by the reciprocating motion of the first piston (4) lie, is parallel to a second straight line (L2), on which the first order reciprocating inertial forces generated by the reciprocating motion of a second piston (5) lie; and

the crank angles of the two cylinders (2, 3) are such that the first order reciprocating inertial forces generated by the reciprocating motion of the first piston (4) are in phase opposition with respect to the first order reciprocating inertial forces generated by the reciprocating motion of the second piston (5).

2. An internal combustion engine (1) according to claim 1, wherein the radius (R1) connecting the barycentre (B) of the first balancing body (12) to the rotation axis (7) of the crankshaft (6) forms a first phase angle ($\gamma1$), which is not equal to zero, with the prolongation of the radius (R2) connecting the crank (8) of the first piston (4) to the rotation axis (7).

3. An internal combustion engine (1) according to claim 2, wherein the two radii (R1, R2) mutually form and angle (α) identical to the angle (α) between the two cylinders (2, 3).

4. An internal combustion engine (1) according to claim 2 or 3, wherein the first phase angle ($\gamma1$) is equal to $\pm 64°$.

5. An internal combustion engine (1) according to claim 2, 3 or 4, wherein the first balancing body (12) presents a mass that is equal to 43.8% of the equivalent mass generating the reciprocating inertial forces of the first piston (4).

6. An internal combustion engine (1) according to any of the claims from 1 to 5 and comprising a second balancing body (13), which is fixed to the crankshaft (6) in correspondence to the second piston (5) and is such that the second straight line (L2), on which the first order reciprocating inertial forces generated by the reciprocating motion of a second piston (5) lie, forms a second inclination angle ($\beta2$) that is not equal to zero with the second cylinder (3).

7. An internal combustion engine (1) according to claim 6, wherein the radius (R1) connecting the barycentre (B) of the second balancing body (13) to the rotation axis (7) of the crankshaft (6) forms a second phase angle ($\gamma2$), which is not equal to zero, with the prolongation of the radius (R2) connecting the crank (9) of the second piston (5) to the rotation axis (7).

8. An internal combustion engine (1) according to claim 7, wherein the second phase angle ($\gamma2$) is equal to +64°.

9. An internal combustion engine (1) according to claim 7 or 8, wherein the second balancing body (13) presents a mass that is equal to 43.8% of the equivalent mass generating the reciprocating inertial forces of the second piston (5).

10. An internal combustion engine (1) according to any of the claims from 1 to 5, wherein the second piston (5) is not coupled to any balancing body (13).

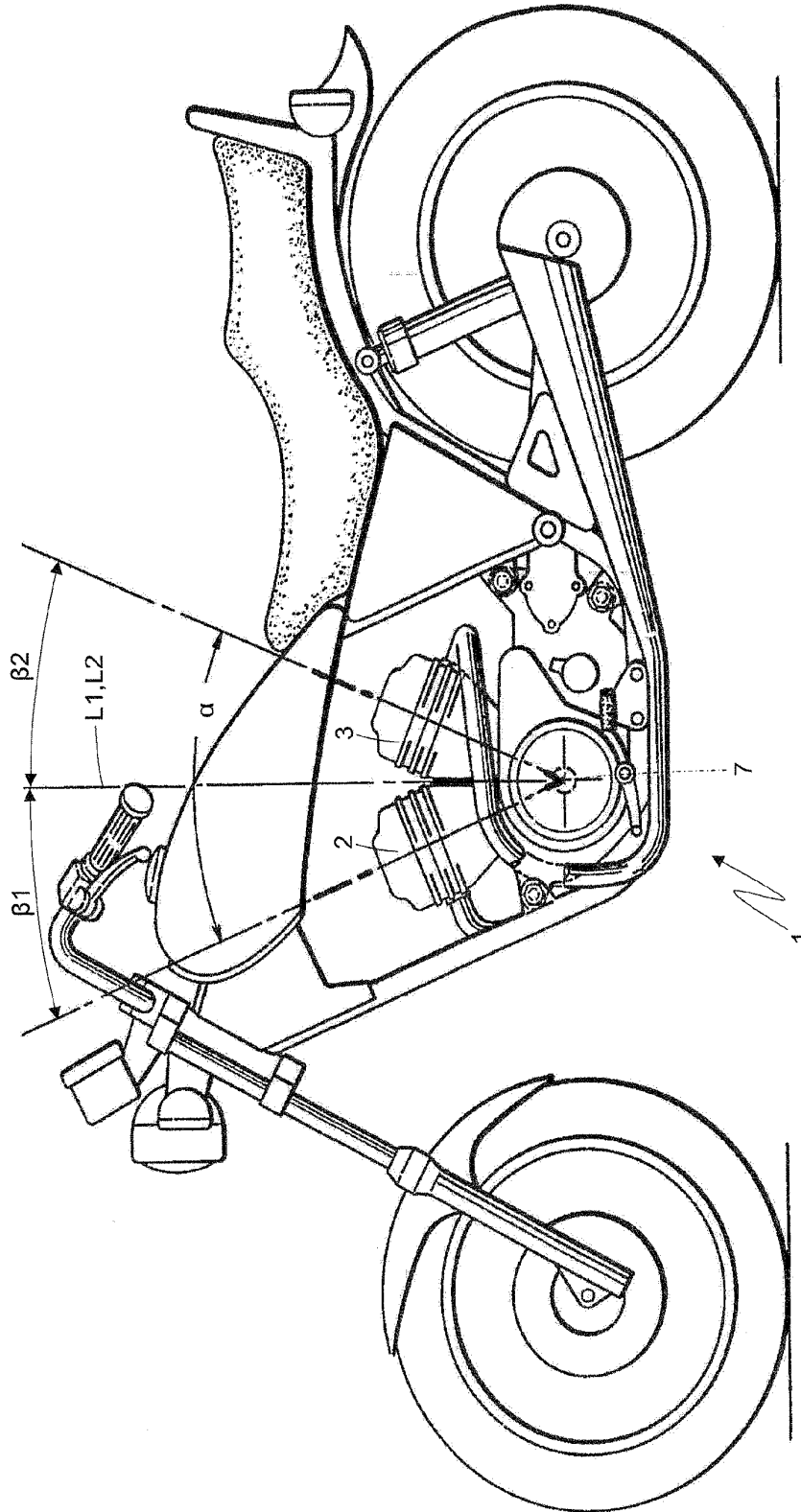11. An internal combustion engine (1) according to any of the claims from 1 to 10, wherein the angle (α) between the cylinders is selected within the following set: 26°, 52°, 64°, 116°, 128°, 154°.
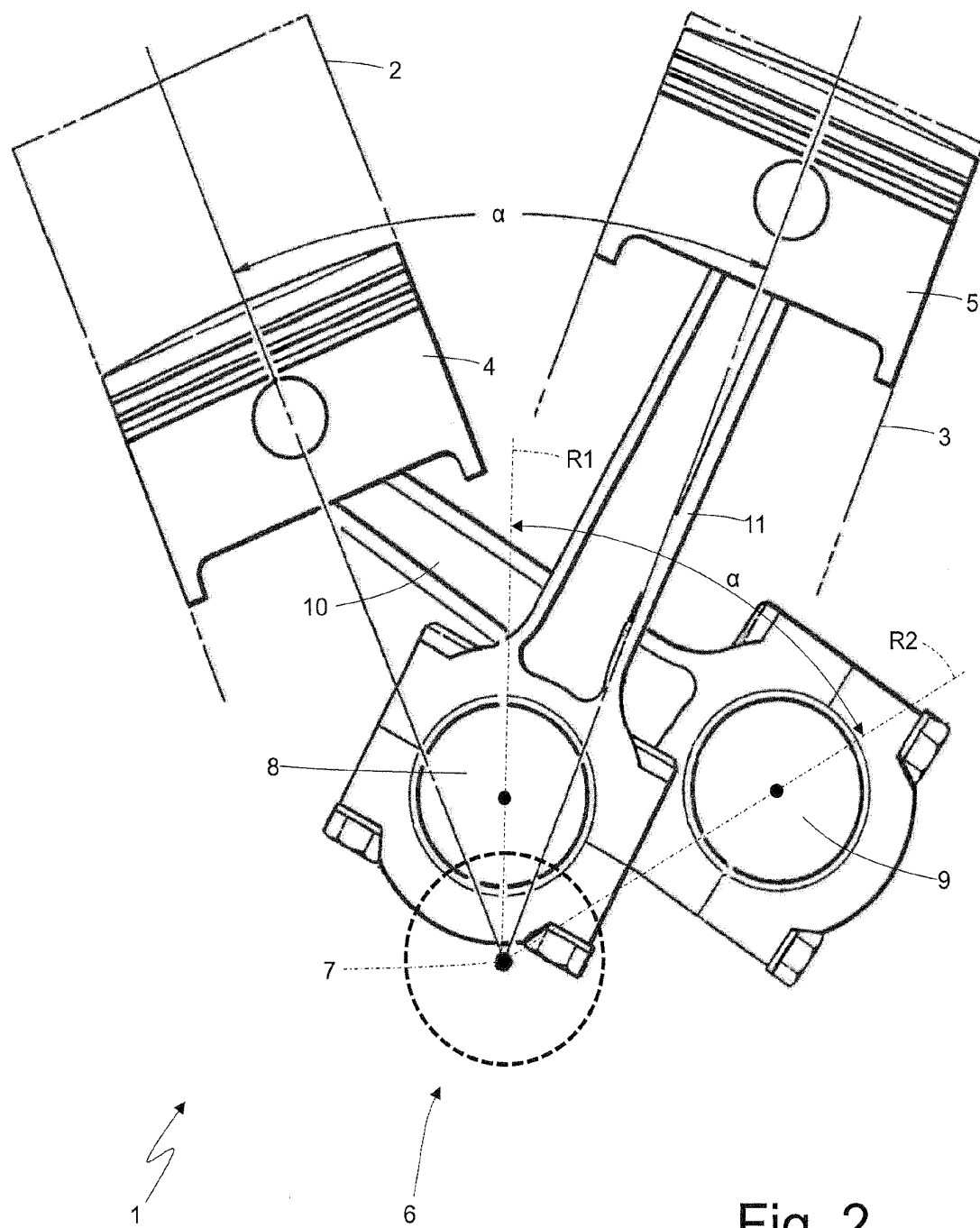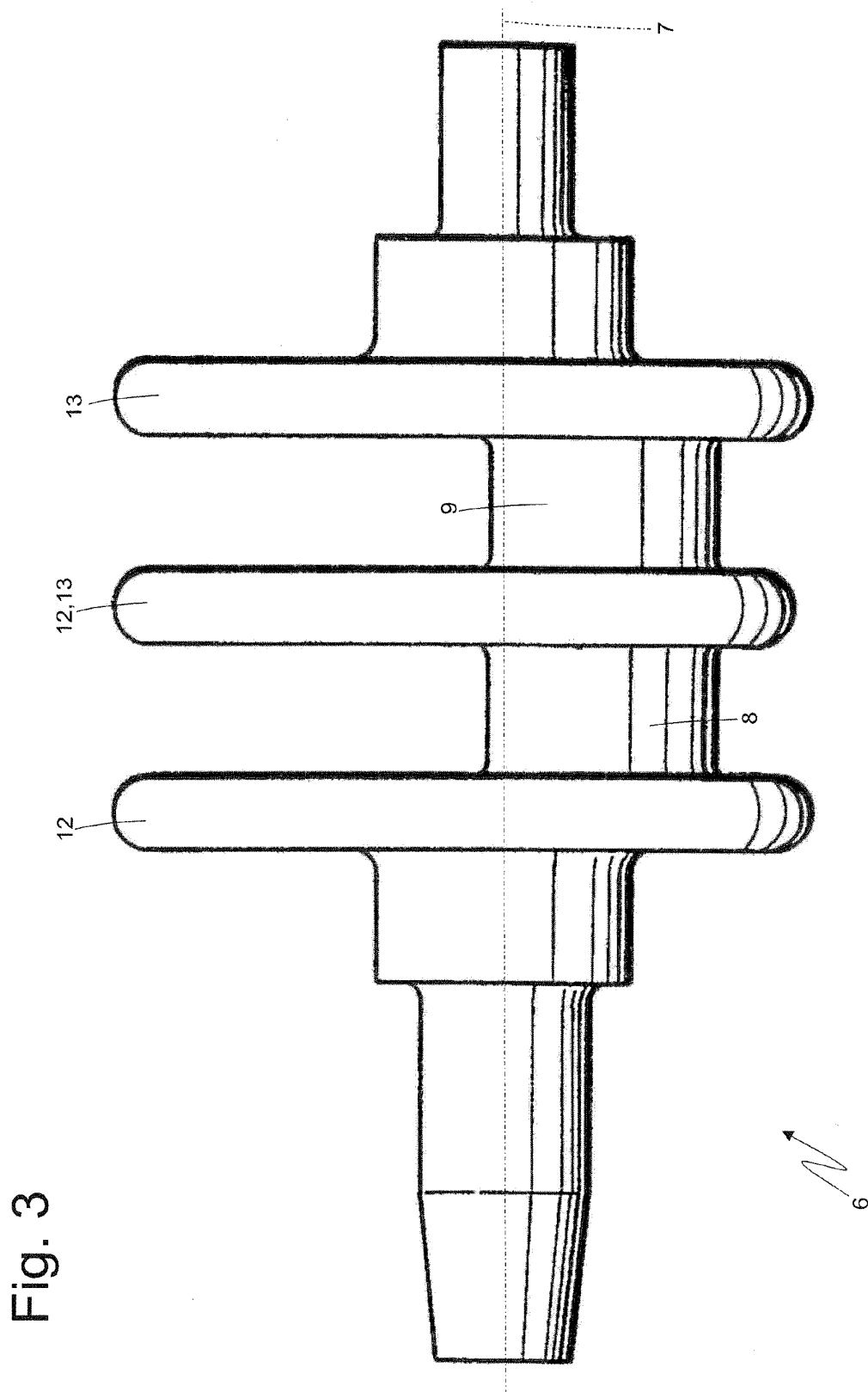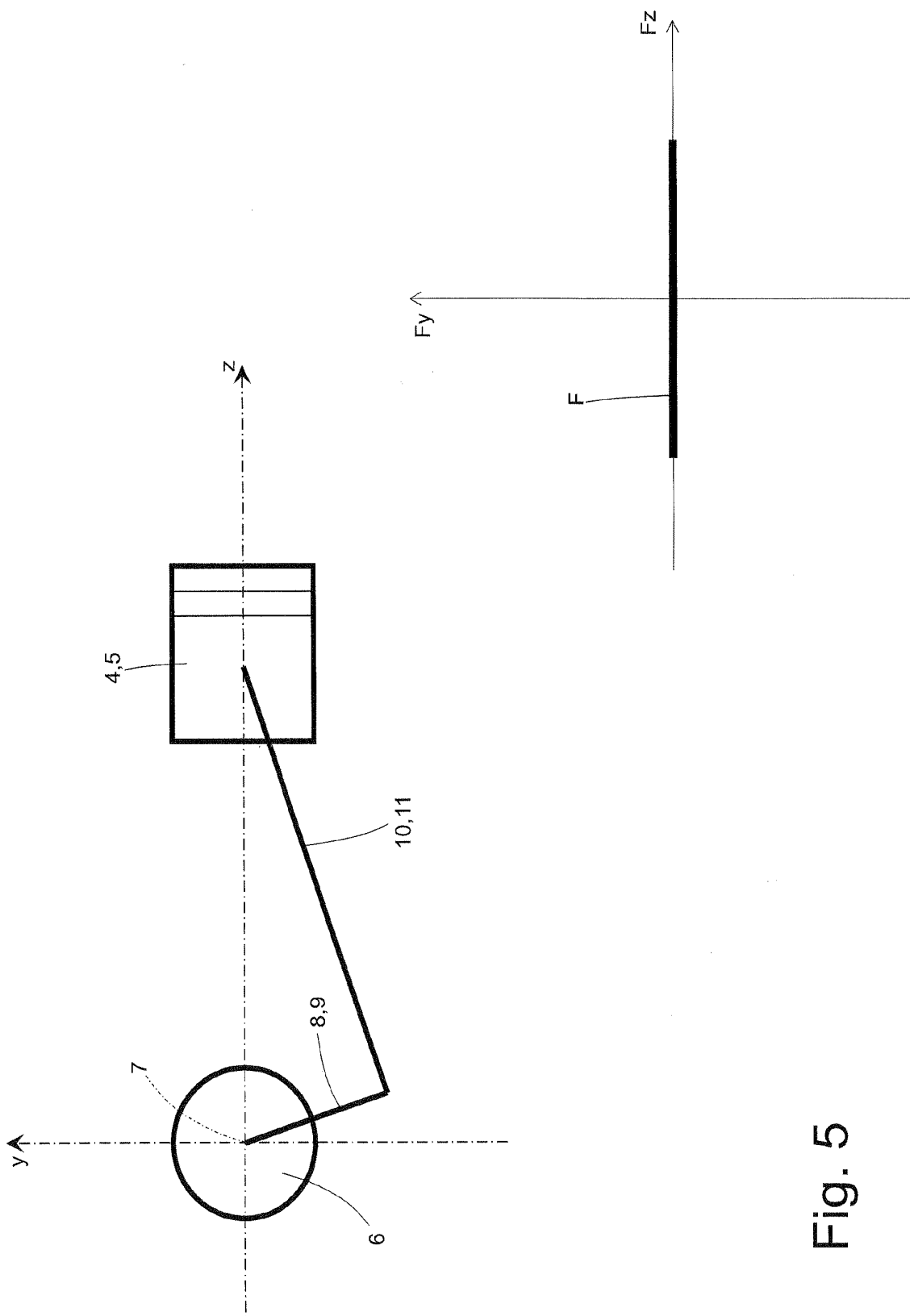
Fig. 1

Fig. 2

Fig. 3

Fig. 4

Fig. 5
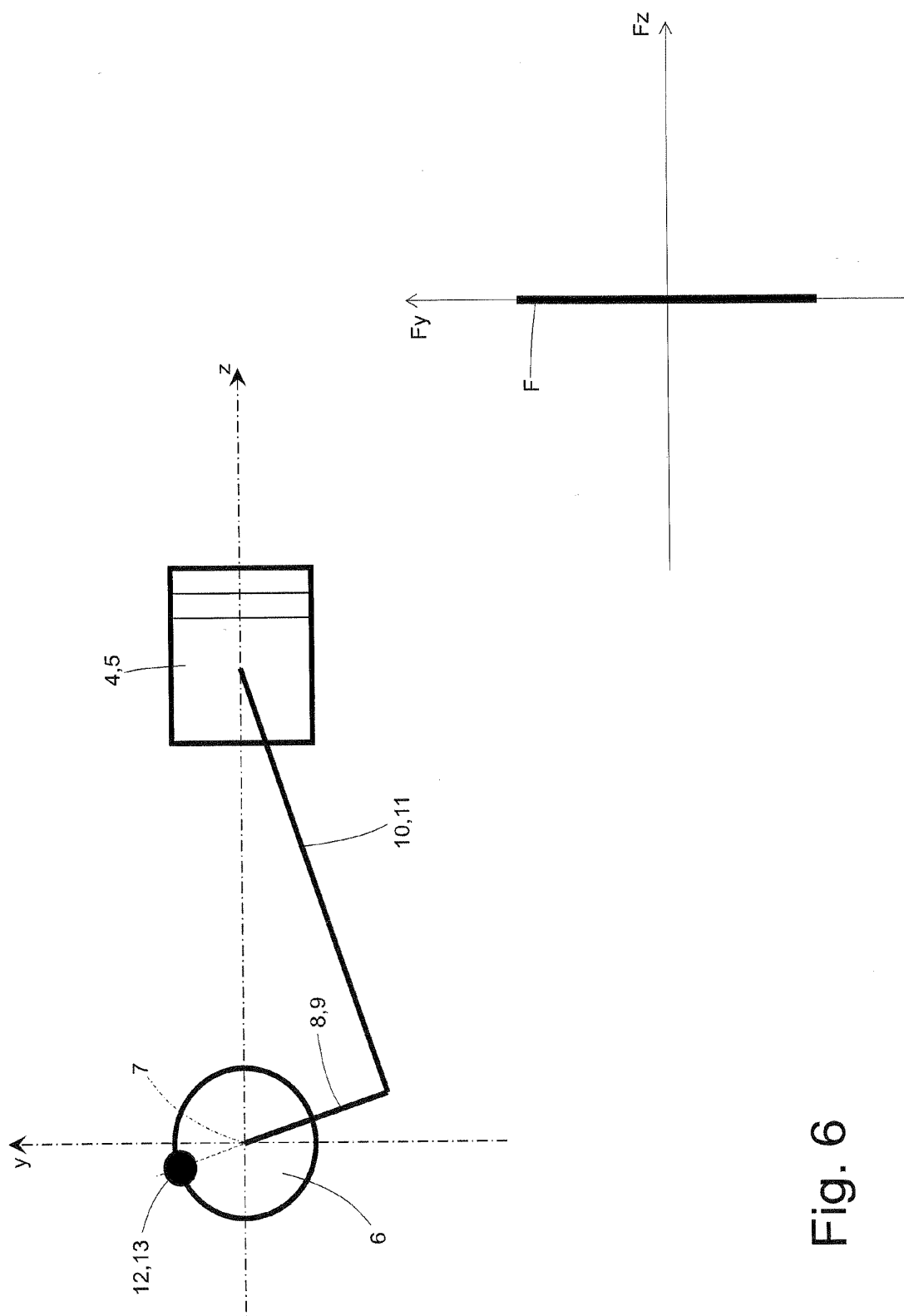
Fig. 6
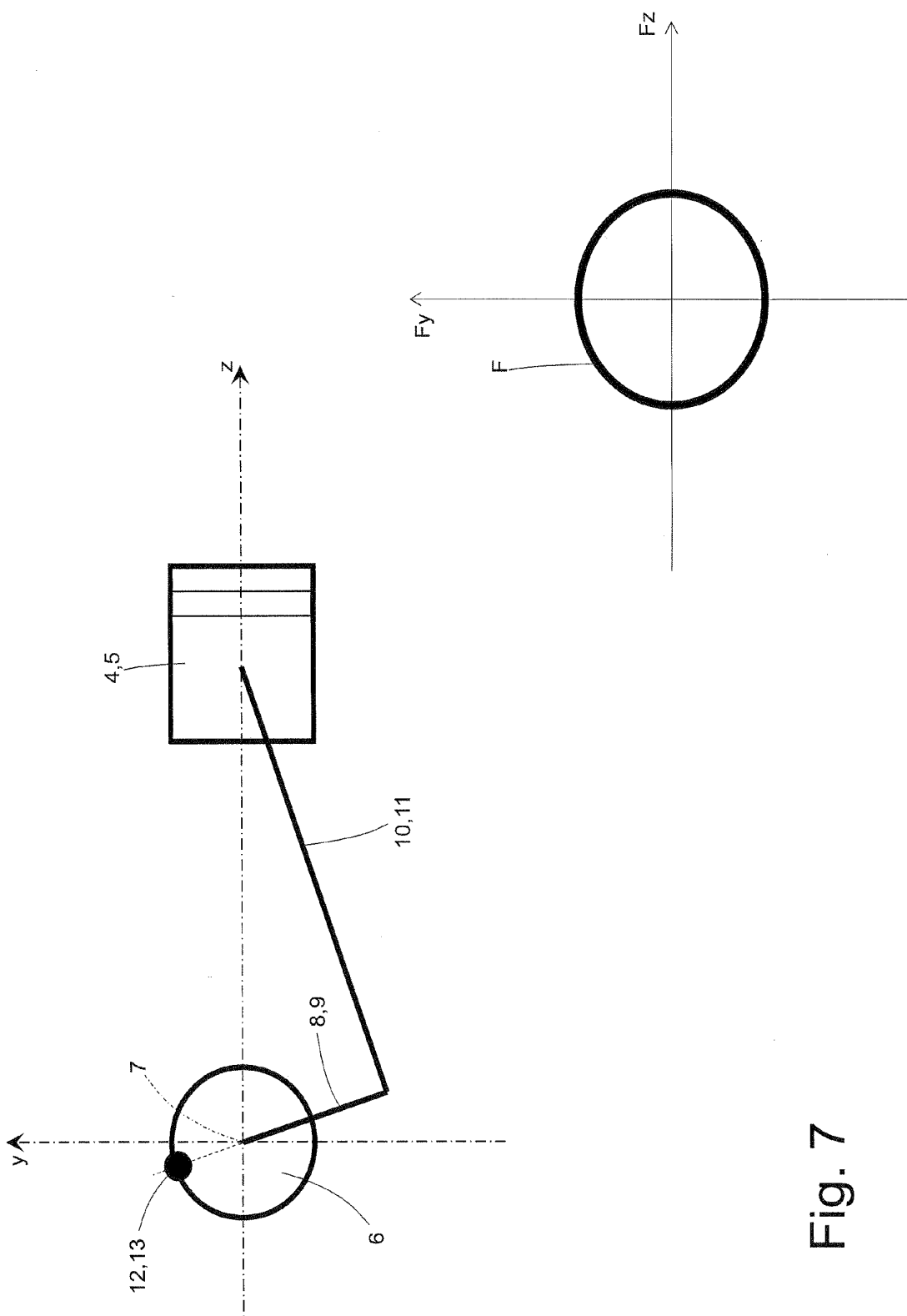
Fig. 7
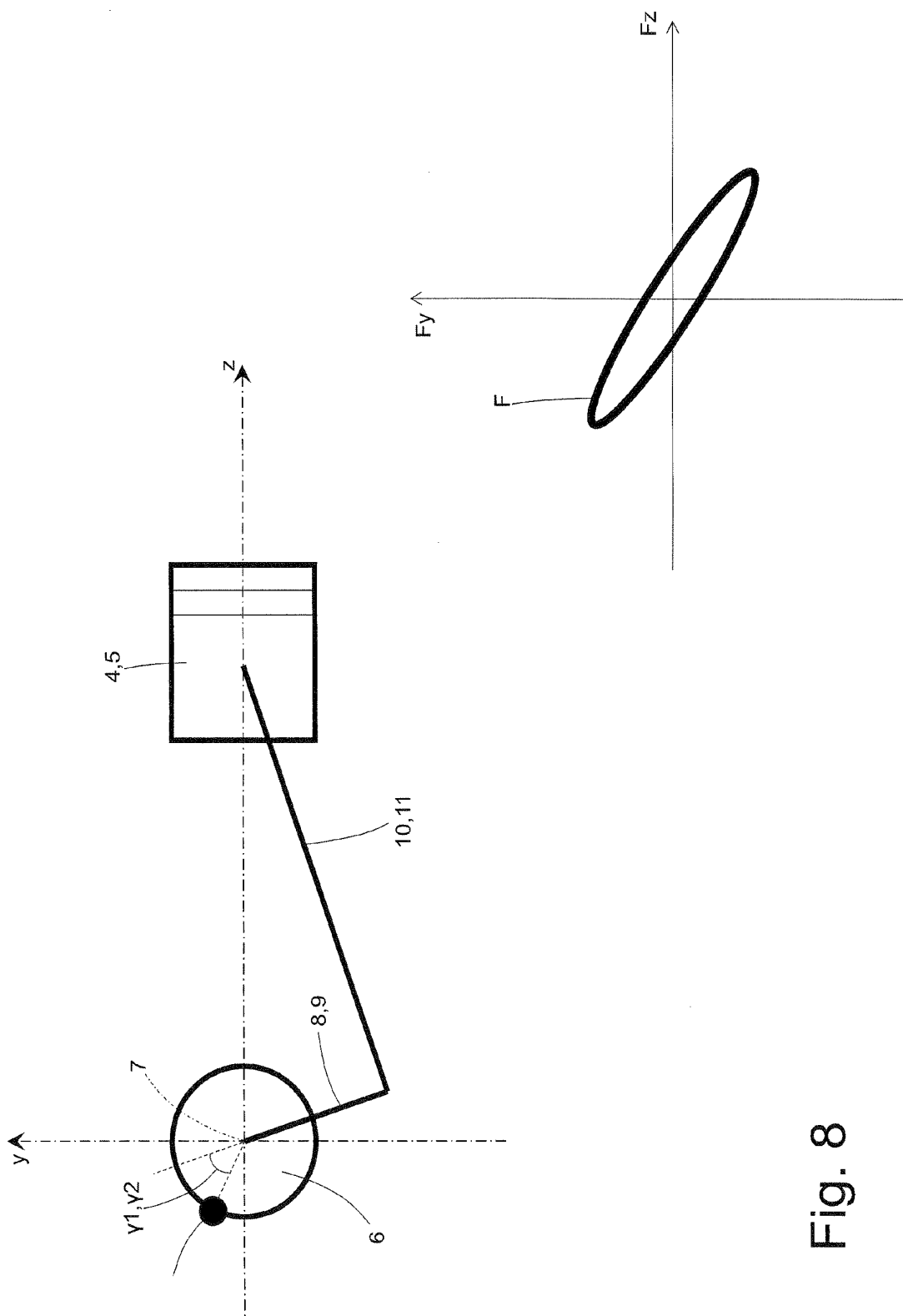
Fig. 8
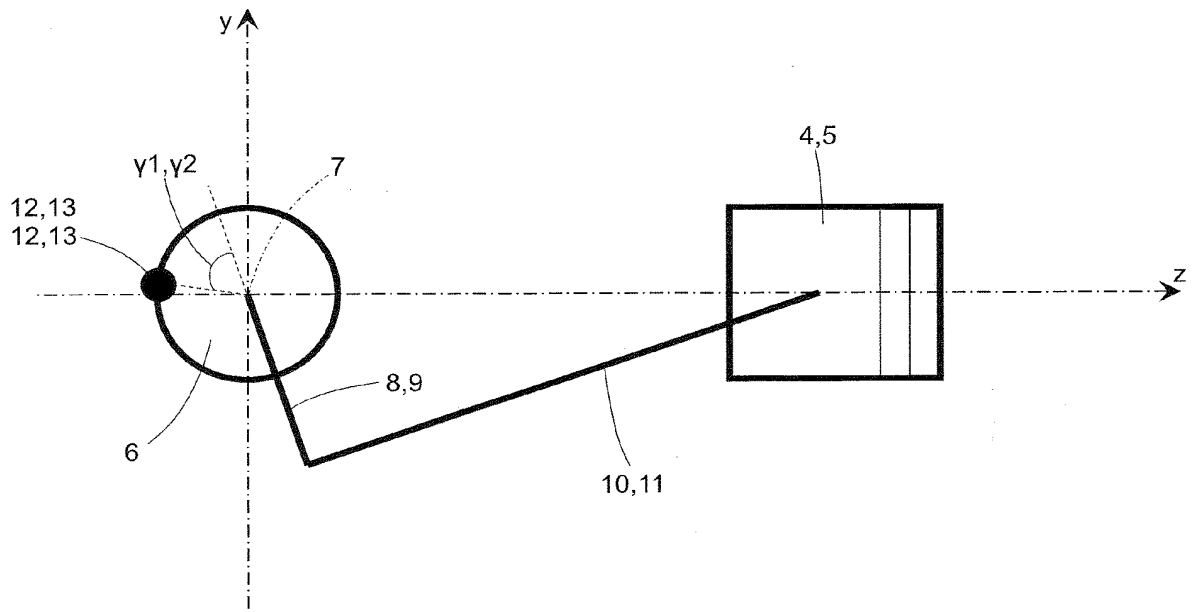
Fig. 9
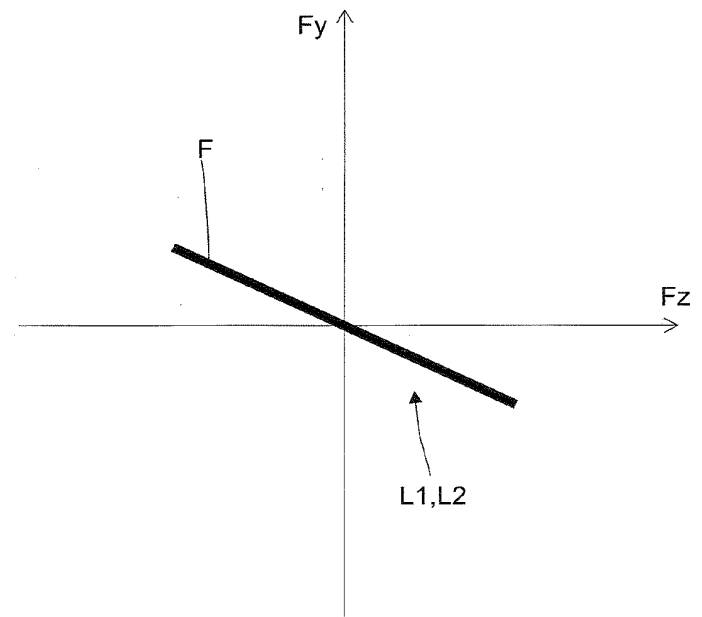
EUROPEAN SEARCH REPORT

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (IPC)
X	US 4 517 933 A (YASUTAKE KOUSUKE [JP]) 21 May 1985 (1985-05-21) * column 3, line 16 - column 5, line 24; figures 1-9 *	1-11	INV. F02B61/02 F02B75/22 F16C3/20 F16F15/28
A	DE 17 76 154 A1 (KLOECKNER HUMBOLDT DEUTZ AG) 27 January 1972 (1972-01-27) * page 1, paragraph 3 - page 2, paragraph 3; figure 1 *	1	
A	US 4 617 885 A (OSHIRO NOBUAKI [JP] ET AL) 21 October 1986 (1986-10-21) * column 2, line 33 - column 8, line 2; figures 4-13 *	1	
A	JP 9 166182 A (YAMAHA MOTOR CO LTD) 24 June 1997 (1997-06-24) * abstract; figures 1-10 *	1	
A	JP 53 067053 A (TOYOTA MOTOR CO LTD) 15 June 1978 (1978-06-15) * abstract; figures 1-4 *	1	TECHNICAL FIELDS SEARCHED (IPC) F02B F16C F16F

The present search report has been drawn up for all claims

Place of search	Date of completion of the search	Examiner
Munich	31 July 2013	Tietje, Kai

1

ANNEX TO THE EUROPEAN SEARCH REPORT
ON EUROPEAN PATENT APPLICATION NO.

EP 13 17 4440

This annex lists the patent family members relating to the patent documents cited in the above-mentioned European search report.
The members are as contained in the European Patent Office EDP file on
The European Patent Office is in no way liable for these particulars which are merely given for the purpose of information.

31-07-2013

Patent document cited in search report		Publication date	Patent family member(s)		Publication date
US 4517933	A	21-05-1985	DE	3336444 A1	12-04-1984
			JP	H0260886 B2	18-12-1990
			JP	S5969514 A	19-04-1984
			US	4517933 A	21-05-1985
DE 1776154	A1	27-01-1972	DE	1776154 A1	27-01-1972
			FR	2019097 A1	26-06-1970
			GB	1285915 A	16-08-1972
US 4617885	A	21-10-1986	JP	H0249422 B2	30-10-1990
			JP	S57195944 A	01-12-1982
			US	4617885 A	21-10-1986
JP 9166182	A	24-06-1997			
JP 53067053	A	15-06-1978			

For more details about this annex : see Official Journal of the European Patent Office, No. 12/82

REFERENCES CITED IN THE DESCRIPTION

Patent documents cited in the description

• US 4517933 A1 **[0004]**