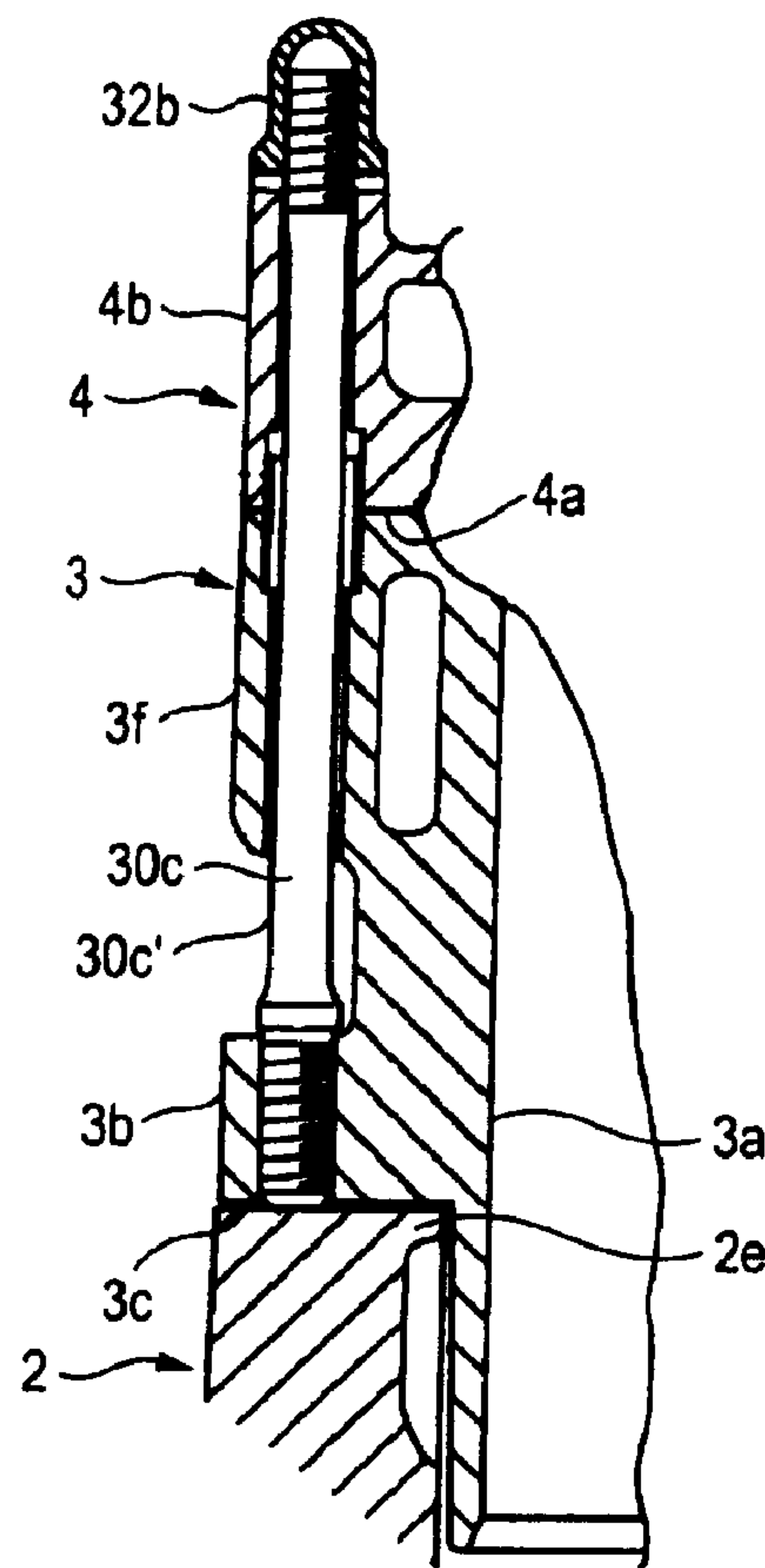




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(72) Inventeurs/Inventors:
UTSUMI, YOJI, JP;
ITO, MASAHIRO, JP
(73) Propriétaire/Owner:
YAMAHA HATSUDOKI KABUSHIKI KAISHA, JP
(74) Agent: G. RONALD BELL & ASSOCIATES

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(57) Abrégé/Abstract:

An engine fastening structure in which a cylinder body and a cylinder head are stacked on and fastened to a crankcase, wherein a case side flange portion 3b formed at a crankcase side end portion of the cylinder body 3 is fastened to the crankcase 2 with case



(57) **Abrégé(suite)/Abstract(continued):**

bolts 30a, and at least part of head bolts 30c for fastening the cylinder head 4 and the cylinder body 3 together are screwed into the case side flange portion 3b.

ABSTRACT

An engine fastening structure in which a cylinder body and a cylinder head are stacked on and fastened to a crankcase, wherein a case side flange portion 3b formed at a crankcase side end portion of the cylinder body 3 is fastened to the crankcase 2 with case bolts 30a, and at least part of head bolts 30c for fastening the cylinder head 4 and the cylinder body 3 together are screwed into the case side flange portion 3b.

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ENGINE FASTENING STRUCTURE

TECHNICAL FIELD

The present invention relates to an engine fastening structure in which a cylinder body and a cylinder head are stacked on and fastened to a crankcase, and more particular, to an engine fastening structure which can improve the durability of the cylinder body by reducing the combustion pressure load which is applied to the cylinder body.

BACKGROUND ART

10 A fastening structure for a motorcycle engine exists in which a crankcase side flange portion of a cylinder body is fastened to a crankcase with bolts, and a cylinder head side flange portion of the cylinder body is fastened to a cylinder head with bolts.

15 With the conventional construction, however, when a single cylinder and large displacement engine are subjected to a large load due to combustion pressure, the large load eventually generates a large tensile stress at an axially intermediate portion of the cylinder body.

Conventionally, it is a generally accepted practice to achieve a required durability by increasing the thickness of the axially intermediate portion of the cylinder body. However, increasing the thickness of the cylinder body increases the weight of the engine.

A conventional engine fastening structure exists which can avoid the increase in the engine weight. See for example, an engine fastening structure disclosed in the Japanese Patent Application No. 06-184100 published on January 30, 1996, as JP-A-8-21210, to Honda Motor Co. Ltd. In this engine fastening structure, a crankcase side flange portion of a cylinder body 2 is fastened and fixed to a crankcase 3, 4 with case bolts 11, and a cylinder head side flange portion of the cylinder body 2 is fastened and fixed to a cylinder head with bolts 15. Furthermore, the cylinder head 1 is fastened and fixed to the crankcase 3, 4 with bolts 17 which screw through the cylinder body 2.

In the structure disclosed in the above publication, the cylinder head 1 is fastened and fixed to the crankcase 3, 4 with the bolts 17 which screw through

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the cylinder body 2, part of a combustion pressure applied to the cylinder body is borne by the bolts 17, and stress generated in the cylinder body can be reduced accordingly, thereby making it possible to improve the durability of the cylinder body.

With the engine fastening structure disclosed in the publication,
5 however, while the head bolts are screwed into the crankcase at positions which align with fixing positions of the cylinder head, since cooling water jackets are located in the cylinder head, the head bolts must be disposed outwardly to avoid the cooling water jackets. Due to this, as seen from the top, the crankcase is fastened
10 at positions apart from the axis of a cylinder, and must be enlarged accordingly, which would otherwise be unnecessary. In addition, since the head bolts have to be disposed at positions where they do not interfere with a web of a crankshaft, and the fixing positions of the cylinder head and fixing positions of the crankcase have to be aligned with each other, the degree of freedom in design is reduced.

The present invention was made in view of the problems inherent in
15 the conventional engine fastening structure, and an object of the invention is to provide an engine fastening structure which can achieve engine durability without needing to enlarge a crankcase unnecessarily or reduce the degree of freedom in arrangement of head bolts.

20 SUMMARY OF THE INVENTION

According to the present invention, there is provided an engine fastening structure in which a cylinder body and a cylinder head are stacked on and fastened to a crankcase, wherein case bolts pass through a case side flange portion formed at a crankcase side end portion of the cylinder body and are screwed into
25 a cylinder body side end portion of the crankcase to fasten the cylinder body to the crankcase, at least part of head bolts which fasten the cylinder head and the cylinder body together is made to be a flange screw-through head bolt, and the flange screw-through head bolt is screwed into a screw portion formed in the case side flange portion.

30 According to an embodiment of the invention, there is provided an engine fastening structure, wherein the flange screw-through head bolt and the case

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bolt overlap each other by a distance which is substantially the same as the thickness of the case side flange portion in the axial direction of a cylinder bore.

According to another embodiment of the invention, there is provided an engine fastening structure, wherein the flange screw-through bolt and the case bolt are disposed close to each other, when viewed in the axial direction of the cylinder bore.

According to a further embodiment of the invention, there is provided an engine fastening structure, wherein the case bolt is disposed such that a distance from the case bolt to a first straight line which passes through the axis of the cylinder bore and which is normal to a crankshaft becomes shorter than a distance from the flange screw-through head bolt to the first straight line, when viewed in the axial direction of the cylinder bore.

According to another embodiment of the invention, there is provided an engine fastening structure, wherein the flange screw-through head bolt is disposed such that a distance from the head bolt to a second straight line which passes through the axis of a cylinder bore and which is parallel to the crankshaft becomes shorter than a distance from the case bolt to the second straight line, when viewed in the axial direction of the cylinder bore.

According to a further embodiment of the invention, there is provided an engine fastening structure, wherein an upper flange portion is formed at a cylinder head side end portion of the cylinder body, the flange screw-through head bolt passes the upper flange portion and is screwed into the case side flange portion, and a part of the flange screw-through head bolt which is disposed between the case side flange portion and the upper flange portion is exposed to the outside.

According to a yet another embodiment of the invention, there is provided an engine fastening structure, wherein at least three head bolts are disposed on either side of the cylinder bore across the second straight line, when viewed in the axial direction of the cylinder bore, and the central head bolt along the second straight line is set to have a length which does not reach the case side flange portion.

According to another embodiment of the invention, there is provided an engine fastening structure, wherein the flange screw-through head bolt is

disposed between a chain compartment formed on a side to the cylinder bore in which a camshaft driving chain which connects the crankshaft to a camshaft is disposed, and the cylinder bore.

5 According to yet another embodiment of the invention, there is provided an engine fastening structure, wherein the flange screw-through head bolt is screwed into the case side flange portion at one end and is fastened and fixed to the cylinder head with a cap nut at the other end thereof.

10 According to a further embodiment of the invention, there is provided an engine fastening structure, wherein a tip of the flange screw-through head bolt is positioned closer to a cylinder body side than a cylinder body side end surface of the crankcase.

BRIEF DESCRIPTION OF THE DRAWINGS

15 Fig. 1 is a right-hand side view of an engine according to an embodiment of the invention.

Fig. 2 is a sectional plan view showing a development of the engine according to the embodiment of the invention illustrated in Figure 1.

Fig. 3 is a left-hand side view showing a valve train device of the engine according to the embodiment of the invention illustrated in Figure 1.

20 Fig. 4 is a sectional rear elevation of the valve train device of the engine according to the embodiment of the invention illustrated in Figure 1.

Fig. 5 is a sectional plan view showing a development of a balance shaft of the engine according to the embodiment of the invention illustrated in Figure 1.

25 Fig. 6 is a bottom view of a cylinder head of the engine according to the embodiment of the invention illustrated in Figure 1.

Fig. 7 is a bottom view of a cylinder body of the engine according to the embodiment of the invention illustrated in Figure 1.

30 Fig. 8 is a sectional side view showing a portion where the cylinder head of the engine is connected to the cylinder body according to the embodiment of the invention illustrated in Figure 1.

Fig. 9 is a sectional side view showing a portion where the cylinder body of the engine is connected to the crankcase of the engine according to the embodiment of the invention illustrated in Figure 1.

5 Fig. 10 is another sectional side view showing a portion where the cylinder body of the engine is connected to the crankcase of the engine according to the embodiment of the invention illustrated in Figure 1.

Fig. 11 is a left-hand side view showing a balancer unit of the engine according to the embodiment of the invention illustrated in Figure 1.

10 Fig. 12 is an enlarged cross-sectional view of a portion where a holding lever of the balancer unit is attached in the engine according to the embodiment of the invention illustrated in Figure 1.

Fig. 13 is a side view of constituent components of a rotational lever of the balancer unit in the engine according to the embodiment of the invention illustrated in Figure 1.

15 Fig. 14 is a side view showing a damping construction of a balancer drive gear of the balancer unit in the engine according to the embodiment of the invention illustrated in Figure 1.

Fig. 15 is a right-hand side view of the balancer unit in the engine according to the embodiment of the invention illustrated in Figure 1.

20 Fig. 16 is a sectional right-hand side view of a bearing bracket of the engine according to the embodiment of the invention illustrated in Figure 1.

Fig. 17 is a sectional left-hand side view of a bearing bracket of the engine according to the embodiment of the invention illustrated in Figure 1.

25 Fig. 18 is an explanatory drawing showing the construction of a lubrication system of the engine according to the embodiment of the invention illustrated in Figure 1.

Fig. 19 is a drawing showing the construction of the lubrication system according to the embodiment of the invention illustrated in Figure 1.

30 Fig. 20 is a sectional side view of an area surrounding a lubricating oil pump of the lubrication system in the engine according to the embodiment of the invention illustrated in Figure 1.

Fig. 21 is a sectional left-hand side view of the lubrication system in the engine according to the embodiment of the invention illustrated in Figure 1.

BEST MODE FOR CARRYING OUT THE INVENTION

5 Hereinafter, one embodiment of the present invention will be described with reference to the accompanying drawings.

Figs. 1 to 21 are drawings for one embodiment of the invention. In the drawings, reference numeral 1 denotes a water-cooled, 4-cycle, single cylinder, 5-valve engine. The engine has a construction in which a cylinder body 3, a cylinder head 4 and a cylinder head cover 5 are stacked on and fastened to a crankcase 2, and a piston 6 slidably disposed in a cylinder bore 3a in the cylinder body 3 is connected to a crankshaft 8 via a connecting rod 7 as shown in Figure 2.

The cylinder body 3 and the crankcase 2 are securely connected together by screwing four case bolts 30a which pass through a lower flange portion 3b of the cylinder body 3 into a cylinder side mating surface 2e of the crankcase 2. To be more specific, the case bolts 30a are screwed into bolt connecting portions 12c of left-side and right-side iron alloy bearing brackets 12, 12' (which will be described later) embedded in left and right wall portions of the aluminum alloy crankcase 2, respectively, through insert casting. Note that reference numeral 31a in Fig. 10 denotes a positioning dowel pin for positioning the crankcase 2 and the cylinder body 3.

In addition, the cylinder body 3 and the cylinder head 4 are connected together with two short head bolts 30b and four long head bolts 30c, being flange screw-through head bolts. The short head bolt 30b is screwed into a threaded portion below an induction port 4c and an exhaust port in the cylinder head 4, and extends downwardly to pass through an upper flange portion 3f of the cylinder block 3 and protrudes downwardly therefrom. Then, a cap nut 32a is screwed on the downwardly protruding portion of the short head bolt 30b. The upper flange portion 3f and hence the cylinder body 3 are thereby fastened to a cylinder side mating surface 4a of the cylinder head 4.

In addition, the long head bolt 30c is screwed into the lower flange portion 3b of the cylinder body 3, and extends upwardly to pass through the upper

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flange portion 3f of the cylinder block 3, then through a flange portion 4b of the cylinder head 4 and protrudes upwardly therefrom. Then, a cap nut 32b is screwed on the upwardly protruding portion of the long head bolt 30c. The lower flange portion 3b and hence the cylinder body are thereby fastened to the cylinder side mating surface 4a of the cylinder head 4. Note that a portion 30c' of the long head bolt 30c which is situated between the lower flange portion 3b and the upper flange portion 3f of the cylinder body 3 is exposed to the outside.

Here, when viewed in a direction normal to the axis A of the cylinder bore (refer to Fig. 10), the long head bolt 30c and the case bolt 30a overlap each other by a distance which is substantially the same as the thickness of the lower flange portion 3b being the case side flange portion, along the axis A of the cylinder bore.

In addition, when viewed in a direction along the axis A of the cylinder bore (refer to Figs. 6, 7), the long head bolt 30c and the case bolt 30a are disposed to establish a relationship proximate to each other. Namely, the case bolt 30a is either disposed such that a distance a1 (from the case bolt 30a to a first straight line C1) which passes through the axis A of the cylinder bore normal to the crankshaft 8 becomes shorter than a distance a2 (from the head bolt 30c to the first straight line C1) or is, in the alternative, disposed such that the case bolt 30a is situated closer to the center of the cylinder bore as viewed in the direction of the crankshaft.

In addition, the head bolt 30c is either disposed such that a distance b2 (from the head bolt 30c to a second straight line C2) which passes through the axis A of the cylinder bore parallel to the crankshaft 8 is shorter than a distance b1 (from the case bolt 30a to the second straight line C2) or is, in the alternative, disposed such that the head bolt is situated closer to the crankshaft side.

Furthermore, three head bolts 30c, 30b, 30c' are disposed on either side of the cylinder bore across the second straight line C2. The head bolt situated centrally along the direction of the second straight line C2 is constructed to be the short head bolt 30b. This short head bolt 30b has a length which corresponds to the upper flange portion 3f, and which does not reach the lower flange portion 3b.

Then, the long head bolts 30c, 30c' are disposed on either side of the cylinder bore across the second straight line C2. Here, on one side of the cylinder

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bore 3a along the direction of the crankshaft (on the left-hand side of Fig. 7), a chain compartment 3d is formed in which a camshaft driving chain 40 for transmitting the rotation of the crankshaft to the camshaft is disposed. The long head bolts 30c situated on the one side of the cylinder bore along the direction of the second
5 straight line C2 are disposed between the chain compartment 3d and the cylinder bore 3a.

Thus, in connecting the cylinder body 3 and the cylinder head 4 together, the upper flange portion 3f of the cylinder body 3 is fastened and fixed to the cylinder head 4 with the short head bolts 30b and cap nuts 32a and the long
10 head bolts 30c are screwed into the lower flange portion 3b which is bolted and connected to the mating surface 2e of the crankcase 2. The cylinder body 3 is thus fastened and fixed to the flange portion 4b of the cylinder head 4 with the screwed long head bolts 30c and cap nuts 32b, and the tensile load due to the combustion pressure is borne by the cylinder body 3 and the four long head bolts 30c.
15 Therefore, the load applied to the cylinder body 3 can be reduced accordingly. As a result, the stress generated at, in particular, the axially intermediate portion of the cylinder body 3 can be reduced, and even when the thickness of the cylinder body 3 is reduced, durability can be achieved.

When only the upper flange portion 3f of the cylinder body 3 is
20 connected to the cylinder head 4, an excessive tensile stress is generated at the axially intermediate portion of the cylinder body 3, and in an extreme case, a crack may be generated at the portion in question. According to this embodiment of the present invention, however, the generation of such an excessive stress at the intermediate portion of the cylinder body can be avoided due to the existence of the
25 long head bolts 30c, thereby making it possible to prevent the generation of such a crack.

In addition, by screwing the four long head bolts 30c into the lower flange portion close to the case bolt 30a for fastening the crankcase 8, the long head bolt 30c transmits part of the load generated by the combustion pressure to
30 the case side flange portion 3b, which further transmits the load to the crankcase 2 via the case bolt 30a, whereby the load applied to the cylinder body 3 can be

reduced in a reliable fashion. From this point of view, the durability of the cylinder body 3 against the load can be improved.

5 In addition, since the long head bolt 30c and the case bolt 30a are disposed to overlap each other by a distance substantially the same as the thickness of the case side flange portion 3b in the axial direction of the cylinder bore, the long head bolt 30c ensures that part of the load generated by the combustion pressure is transmitted to the case side flange portion 3b, thereby reducing the load applied to the intermediate portion of the cylinder body 3.

10 Additionally, since the case bolt 30a is either disposed such that the distance a1 to the first straight line C1 which passes through the axis of the cylinder bore normal to the crankshaft becomes shorter than the distance from the long cylinder bolt 30c to the first straight line C1, or is, in the alternative, disposed such that the case bolt 30a is situated closer to the center of the cylinder bore in the direction of the crankshaft, when viewed in the direction of the axis A of the cylinder
15 bore, (as shown in double-dashed lines in Fig. 7) the dimensions of the mating surface 2e of the crankcase 2 that is attached to the cylinder body in the crankshaft direction can be reduced to the vicinity of positions in which the long head bolts 30c are disposed. As a result, the dimension of the crankcase 2 in the crankshaft direction can be reduced.

20 Furthermore, since either the long head bolts 30c are screwed into the case side flange portion 3b of the cylinder body 3, or in the alternative, the long head bolts 30c are not screwed into the mating surface 2e of the crankcase 2 which is attached to the cylinder body, there is no risk that the long head bolts 30c will interfere with the web 8b of the crankshaft 8, and the long head bolt 30c can be
25 disposed either such that the distance b2 to the second straight line C2 which passes through the center of the cylinder parallel to the crankshaft becomes shorter than the distance b1 from the case bolt 30a to the second straight line C2, or in the alternative, disposed such that the long head bolt 30c is situated closer to the crankshaft side, thereby reducing the dimension of the cylinder head 4 and the
30 cylinder body 3 in the direction normal to the crankshaft.

In addition, since the axial part 30c' of the long head bolt 30 c is exposed to the outside from the side wall of the cylinder body 3, the wall which

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surrounds the long head bolt 30c can be reduced, and the weight of the cylinder body can be reduced accordingly.

Additionally, since the three head bolts 30c are disposed on either side of the cylinder bore across the second straight line C2, while the head bolt 30b situated centrally along the direction of the second straight line C2 is situated apart from the axis A of the cylinder bore in the direction normal to the crankshaft, the case side flange portion 3b corresponding to the central head bolt 30b can be minimized because the length of the head bolt 30b does not reach the case side flange portion 3b, thereby making it possible to avoid enlargement of the crankcase.

In addition, since the long head bolts 30c are disposed between the cylinder bore 3a and the chain compartment 3d formed on the side to the cylinder bore 3a, the long head bolts 30c can be disposed making effective use of a dead space formed therebetween.

Furthermore, since the long head bolt 30c is screwed into the case side flange portion 3b at the one end and is fastened and fixed to the cylinder head with the cap nut 32b at the other end thereof, the cylinder head can be removed without requiring a large space above the cylinder head, thereby making it possible to improve the maintenance properties of the engine.

Here, as shown in Figs. 5 and 16, the right-side bearing bracket 12' has a boss portion 12b in which a right-side bearing 11a' of the crankshaft 8 is inserted to be fitted in a bearing hole 12a through press fit. Then, the front and rear bolt connecting portions 12c, 12c' extend upwardly from front and rear portions of the boss portion 12b which are situated opposite to each other across the crankshaft 8, as viewed in the direction in which the crankshaft 8 extends to the vicinity of the cylinder-sidemating surface 2e of the crankcase 2.

In addition, in the left-side bearing bracket 12, as shown in Figs. 5 and 17, the front and rear bolt connecting portions 12c, 12c' extend upwardly from front and rear portions which are situated opposite to each other across the crankshaft 8, as viewed in the direction in which the crankshaft 8 extends to the vicinity of the cylinder-sidemating surface 2e of the crankcase 2. In addition, a collar hole 12e is formed in the boss portion 12b into which an iron bearing collar 12d having an outside diameter larger than that of a balancer driving gear 25a (which will be

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described later) is press fitted. A left-side crankshaft bearing 11a is inserted to be fitted in the bearing hole 12a of the bearing collar 12d.

Here, the bearing collar 12d is provided to facilitate the assembly of the crankshaft 8 in the crankcase 2 with a gear unit 25 having the balancer driving gear 25a being press fitted on the crankshaft 8.

In addition, as shown in Fig. 5, a seal plate 25d is interposed between the gear unit 25 on a left shaft portion 8c of the crankshaft 8 and the left-side bearing 11a. An inside diameter side portion of the seal plate 25d is held by the gear unit 25 and an inner race of the left-side bearing 11a, and a slight gap is provided between an outside diameter side portion thereof and an outer race of the left-side bearing 11a for avoiding the interference therebetween. In addition, an outer circumferential surface of the seal plate 25d is brought into sliding contact with an inner circumferential surface of a flange portion 12h of the bearing collar 12d.

Furthermore, a seal tube 17i is interposed between the right-side bearing 11a' of a right shaft portion 8c' of the crankshaft 8 and a cover plate 17g. An inner circumferential surface of the seal tube 17i is fixedly fitted on the right shaft portion 8c'. In addition, a seal groove having a labyrinth construction is formed in an outer circumferential surface of the seal tube 17i, and the outer circumferential surface of the seal tube 17i is brought into sliding contact with an inner circumferential surface of a seal hole 2p formed in the right case portion 2b.

Thus, the leakage of pressure within a crank compartment 2c is prevented by interposing the seal plate 25d and the seal tube 17i on the outside of the left-side and right-side bearings 11a, 11a' at the left and right shaft portions 8c, 8c' of the crankshaft 8.

Thus, according to this embodiment, since the front and rear bolt connecting portions 12c, 12c' which extend toward the cylinder body 3 side are integrally formed on the sides situated opposite to each other across the axis A of the cylinder bore of each of the iron alloy crankshaft supporting left-side and right-side bearing brackets 12, 12' which are insert cast in the aluminum alloy crankcase 2 and the case bolts 30a for connecting the cylinder body 3 to the crankcase 2 are screwed into the front and rear bolt connecting positions 12c, 12c' respectively, the load generated by the combustion pressure can be borne uniformly by the two front

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and rear bolt connecting portions 12c, 12c' which are situated opposite to each other across the axis A of the cylinder bore, whereby the connecting rigidity between the cylinder body 3 and the crankcase 2 can be improved.

5 In addition, since the front and rear balance shafts 22, 22' which are disposed in parallel with the crankshaft 8 in the vicinity thereof are supported by the iron alloy left-side and right-side bearing brackets 12, 12' at least one ends thereof, the supporting rigidity of the front and rear balance shafts 22, 22' can be enhanced.

10 Furthermore, in embedding the iron alloy left-side and right-side bearing brackets 12, 12' in the aluminum alloy crankcase 2, since the upper end face 12f of the front and rear bolt connecting portions 12c, 12c' are positioned inwardly without being exposed to the cylinder side mating surface 2e of the crankcase 2. As a result, metallic members at a joint between the crankcase 2 and the cylinder block 3 are of the same material and hardness, and reduction in sealing properties can be avoided. For example, in the event that the upper end faces 12f
15 of the front and rear bolt connecting portions 12c, 12c' are made to abut with a case side mating surface 3c formed on the lower flange 3b of the aluminum alloy cylinder body 3, the sealing properties are reduced due to a difference in thermal expansion coefficients.

20 In the left-side bearing bracket 12, the bearing collar 12d having the outside diameter larger than that of the balancer driving gear 25a is attached to the outer circumference of the bearing 11a. When assembling the crankshaft 8 in the crankcase 2 with the balancer driving gear 25a being fixed onto the crankshaft 8 through press fit or the like, there is no risk that the balancer driving gear 25a will interfere with a minimum inside diameter portion of the boss portion 12b of the left-
25 side bearing bracket 12, and hence, the assembling of the crankshaft 8 can be implemented without any problem. Even in the event that the balancer driving gear 25a is instead formed as an integral part of the crankshaft 8, there is no problem.

30 The crankcase 2 is a two-piece type in which the crankcase 2 is divided into the left and right case portions 2a, 2b. A left case cover 9 is detachably attached to the left case portion 2a, and a space surrounded by the left case portion 2a and the left case cover 9 constitutes a flywheel magnet compartment 9a. A flywheel magnetic generator 35 attached to the left portion of the crankshaft 8 is

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accommodated in this flywheel magnet compartment 9a. Note that the flywheel magnet compartment 9a communicates with a camshaft arranging compartment via the chain compartments 3d, 4d (which will be described later) and most of the lubricating oil which has been used to lubricate the camshafts falls into the flywheel magnet compartment 9a via the chain compartments 3d, 4d.

In addition, a right case cover 10 is detachably attached to the right case portion 2b, and a space surrounded by the right case portion 2b and the right case cover 10 constitutes a clutch compartment 10a.

The crank compartment 2c and a transmission compartment 2d are formed at front and rear portions of the crankcase 2, respectively. The crank compartment 2c is made to open to the cylinder bore 3a but is defined substantially to be separated from the other compartments, including the transmission compartment 2d. As a result, the pressure within the transmission compartment 2d is caused to fluctuate as the piston reciprocates vertically, thereby allowing the transmission compartment 2d to function as a pump.

A transmission 13 is accommodated and arranged in the transmission compartment 2d. The transmission 13 has a constant mesh construction in which a main shaft 14 and a drive shaft 15 are provided and arranged in parallel with the crankshaft 8, and first-speed to fifth-speed gears 1p to 5p attached to the main shaft 14 constantly mesh with first-speed to fifth-speed gears 1w to 5w attached to the drive shaft 15.

The main shaft 14 is rotationally supported by the left and right case portions 2a, 2b via left and right main shaft bearings 11b, 11b', whereas the drive shaft 15 is rotationally supported by the left and right case portions 2a, 2b via left and right drive shaft bearings 11c, 11c'.

A right end portion of the main shaft 14 passes through the right case portion 2b and protrudes to the right side, and a clutch mechanism 16 is attached to the protruding portion. This clutch mechanism 16 is located within the clutch compartment 10a. Then, an input gear, being a large reduction gear 16a of the clutch mechanism 16 meshes with a small reduction gear 17 fixedly attached to the right end portion of the crankshaft 8.

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A left end portion of the drive shaft 15 protrudes outwardly from the left case portion 2a and a driving sprocket 18 is attached to the protruding portion. This driving sprocket 18 is connected to a driven sprocket on a rear wheel.

A balancer unit 19 according to this embodiment of the present invention includes front and rear balancers 20, 20' disposed opposite across the crankshaft 8 and having substantially the same construction. The front and rear balancers 20, 20' include the front and rear balance shafts 22, 22' which do not rotate, and front and rear weights 24, 24' which are rotationally supported on the front and rear balance shaft via front and rear bearings 23, 23'.

Here the front and rear balance shafts 22, 22' also function as the case bolts for fastening and connecting the left and right case portions 2a, 2b together in the direction of the crankshaft 8. The front and rear balance shafts 22, 22' function to connect the left and right case portions 2a, 2b together by causing flange portions 22a formed on insides of the rotationally supported weights 24, 24' in a transverse direction of the engine to abut with boss portions 12g integrally formed on the front and rear bearing brackets 12, 12' which are insert cast into the left and right case portions 2a, 2b and screwing fixing nuts 21b, 21a on opposite end portions of the front and rear balance shafts 22, 22'.

The weight 24 includes a semi-circular weight main body 24a, 24a' and a circular gear supporting portion 24b which is integrally formed on the weight main body 24a, and a ring-shaped balancer driven gear 24c, 24c' is fixedly attached to the gear supporting portion 24b, 24b'. Note that reference numeral 24b denotes a hole made in a part of the weight 24 which is situated opposite to the weight main body 24a so as to reduce the weight of the part to as low a level as possible.

The rear balancer driven gear 24c attached to the rear balancer 20' meshes with the rear balancer driving gear 25a, which is rotationally attached relative to the gear unit 25, which is securely attached to the left shaft portion 8c of the crankcase 8 through press fit.

Note that reference numeral 25b denotes a timing chain driving sprocket integrally formed on the gear unit 15 and has, as shown in Fig. 11, an aligning or timing mark 25c for alignment of timing marks for valve timing. The gear unit 25 is press fitted on the crankshaft 8 such that the timing mark 25c aligns with

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the cylinder bore axis A, as viewed in the direction in which the crankshaft 8 extends when the crankshaft 8 is situated at a top dead center of a compression stroke.

In addition, the balancer driven gear 24c attached to the front balancer 20 meshes with a front balancer driving gear 17a which is supported rotationally
5 relative to the small reduction gear 17, which is fixedly attached to the right shaft portion 8c' of the crankshaft 8.

Here, the rear balancer driving gear 25a is supported rotationally relative to the gear unit 25, and the front balancer driving gear 17a is supported rotationally relative to the small reduction gear 17. Then, U-shaped damper springs
10 33 each made up of a plate spring are interposed between the rear and front balancer driving gears 25a, 17a and the gear unit 25 and the small reduction gear 17, respectively, to thereby restrain the transmission of impact generated due to a torque fluctuation occurring in the engine to the front and rear balancers 20, 20' is restrained from being transmitted.

Here, while the front balancer driving gear 17a for driving the front
15 balancer 20 will be described in detail by reference to Fig. 14, the same description would be given if the balancer driving gear 25a for driving the rear balancer 20' were described. The front balancer driving gear 17a is formed into a ring shape and is supported by a sliding surface 17b formed so as to have a smaller diameter than the
20 small reduction gear 17 rotationally relative to a side of the small reduction gear 17. Then, a number of U-shaped spring retaining grooves 17c are formed in the sliding surface 17b by setting them back into the surface thereof in a radial fashion about the center of the crankshaft 8, and the U-shaped damper springs 33 are arranged to be inserted in place within the spring retaining grooves 17c. Opening side end
25 portions 33a, 33a of the damper spring 33 are locked at front and rear stepped portions formed in a locking recessed portion 17d formed in an inner circumferential surface of the front balancer driving gear 17a.

When a relative rotation is generated between the small reduction gear
30 17 and the front balancer driving gear 17a due to a torque fluctuation, the damper springs 33 resiliently deform in a direction in which the space between the end portions 33a, 33a narrows so as to absorb the torque fluctuation so generated. Note that reference numeral 17g denotes a cover plate for retaining the damper

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springs 33 within the retaining grooves 17c, reference numeral 17h denotes a key for connecting the small reduction gear 1 with the crankshaft 8, and reference numerals 17e, 17f denote, respectively, alignment marks for use in assembling the small reduction gear 17 and the front balancer driving gear 17a.

5 A mechanism for adjusting a backlash between the front and rear balancer driven gears 24c, 24c' and the front and rear balancer driving gears 17a, 25a is provided on the front and rear balancers 20, 20'. This adjusting mechanism is constructed such that the balancer axis of the front and rear balance shafts 22, 22' slightly deviates from the rotational center of the front and rear balancer driven
10 gears 24c, 24c'. Namely, when the front and rear balance shafts 22, 22' are made to rotate about the balancer axis, the space between the rotational center line of the front and rear balancer driven gears 24c, 24c' and the rotational center line of the front and rear balancer driving gears 17a, 25a changes slightly, whereby the backlash is changed.

15 Here, a mechanism for rotating the front and rear balance shafts 22, 22' differs between the front balancer 20 and the rear balancer 20'. In the rear balancer 20', a hexagonal locking protruding portion 22b is formed on a left end portion of the rear balance shaft 22', and a spline-line (a polygonal star-like) locking
20 hole 26a formed in one end of a rotational lever 26 is locked on the locking protruding portion 22b. In addition, an arc-like bolt hole 26b is formed in the other end portion of the rotational lever 26 in such a manner as to extend about the balancer axis.

25 A fixing bolt 27a passed through the bolt hole 26b is screwed into a guide plate 28. The guide plate 28 is generally formed into an arc-like shape and is fixedly bolted to the crankcase 2. Note that the guide plate 28 also functions to control the flow of lubricating oil.

30 The adjustment of the backlash of the rear balancer 20' is implemented by rotating the rotational lever 26 to bring the backlash to an appropriate state, with the fixing nut 21a being loosened and thereafter fixing the rotational lever 26 with the fixing bolt 27a and a fixing nut 27b, and thereafter, the fixing nut 21a is refastened.

A grip portion 22f having an oval cross section, which is formed by forming a flat portion 22e on both sides of a cross-sectionally circular shape, is formed on a left end portion of the front balance shaft 22 (refer to Fig. 12). A collar 29a having an inner circumferential shape which matches an outer circumferential shape of the grip portion 22f is attached to the grip portion 22f, and furthermore, a holding portion 29b of a holding lever 29 is attached to an outside of the collar 29a in such a manner as to move axially but as not to rotate relatively. A distal end portion 29e of the holding lever 29 is fixed to a boss portion 2f of the left case portion 2a with a bolt 29f. In addition, a tightening slit 29c is formed in the holding portion 29b of the holding lever 29, so that the rotation of the collar 29a, and hence of the front balance shaft 22, is prevented by tightening up the fixing bolt 29d. Furthermore, the fixing nut 21b is screwed on the front balance shaft 22 to an outer side of the collar 29a and secured thereto via a washer.

The adjustment of the backlash of the front balancer 20 is implemented by loosening or removing the fixing nut 21b, gripping the grip portion 22f of the front balance shaft 22 with a tool to rotate the shaft to bring the backlash to an appropriate state, and thereafter tightening up the fixing bolt 29d, and thereafter, the fixing nut 21b is fastened.

In addition, a lubricating oil introducing portion 22c is formed in an upper portion of the locking protruding portion 22b by cutting out the upper in an arc. A guide bore 22d is made to open to the introducing portion 22c, and the guide bore 22d extends into the balance shaft 22 and passes therethrough to below an outer circumferential surface of the front balance shaft 22, whereby the lubricating oil introducing portion 22c is made to communicate with an inner circumferential surface of the balancer bearing 23. Thus, lubricating oil that has fallen in the lubricating oil introducing portion 22c is supplied to the balancer bearing 23.

Here, while the weight 24 and the balancer driven gear 24c are disposed at the right end portion along the direction in which the crankshaft extends in the front balancer 20, in the rear balancer 20', they are disposed at the left end portion. In addition, the balancer driven gear 24c is located rightward relative to the weight 24 in both the front and rear balancers 20, 20', and therefore, the weight 24

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and the balancer driven gear 24c are set into the same configuration in both the front and rear balancers 20, 20'.

Thus, according to this embodiment, since the weight main body 24a and the front balancer driven gear 24c of the front balancer 20 are disposed on the right-hand side (one side) of the front balance shaft 22 along the direction in which the crankshaft 8 extends, and the weight main body 24a' and the rear balancer driven gear 24c' are disposed on the left-hand side (the other side) of the rear balance shaft 22' along the direction in which the crankshaft extends, the reduction in balance in weight in the crankshaft 8 direction that would otherwise result when providing a two-shaft balancer unit can be avoided.

In addition, since the front and rear balance shafts 22, 22' also function as the case bolts for connecting the left and right case portions 2a, 2b together, when adopting a two-shaft balancer unit, the connecting rigidity of the crankcase 2 can be enhanced without causing undue construction complexity and an increase in the number of components.

Additionally, since the balancer weight main body 24a and the balancer driven gear 24c are made integral and are supported rotationally by the front and rear balance shafts 22, 22', only the weight of the balancer weight main body 24a and the balancer driven gear 24c may be driven to rotate, and therefore, the engine output can be used effectively to such an extent that the front and rear balance shafts 20, 20' themselves do not need to be driven to rotate.

In addition, the degree of freedom in assembling can be improved, as compared to engine construction where a balancer weight and a balance shaft are made integral.

Additionally, since the rotational center lines of the balancer driven gears 24c are caused to deviate relative to the axes of the front and rear balance shafts 22, 22', the backlash between the balancer driven gears 24c and the front and rear balancer driving gears 17a, 25a on the crankshaft 8 side can be adjusted by the simple construction or only by a simple operation of rotating the front and rear balance shafts 20, 20', thereby preventing undue generation of noise.

On the front balance shaft 22, the backlash adjustment is implemented by gripping the grip portion 22f formed on the left-hand side of the balance shaft 22

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with a tool so as to rotate the front balance shaft 22. On the rear balance shaft 22', the backlash adjustment is implemented by rotating the rotational lever 26 provided on the left-hand side of the rear balance shaft 22'. Thus, on either one of the front and rear balance shafts 22, 22', the backlash can be adjusted from the left-hand side of the engine, and the backlash adjusting work can be implemented efficiently.

Additionally, since the front balancer driving gear 17a on the crankshaft 8 side which meshes with the balancer driven gear 24c rotates relatively to the sliding surface 17b of the small reduction gear 17 fixed to the crankshaft 8, and the U-shaped damper springs 33 are disposed in the spring retaining grooves 17c formed by setting them back from the sliding surface 17b, the impact generated due to the torque fluctuation in the engine can be absorbed by the compact construction and the balancer unit can be operated smoothly. Note that the same description applies with respect to the rear balancer drive gear 25a.

Furthermore, a coolant pump 48 is disposed at the right end portion of the front balance shaft 22 and is coaxially therewith. A rotating shaft of the coolant pump 48 is connected to the front balance shaft 22 by an Oldham's coupling which has a similar construction to that of a lubricating oil pump 52 (which will be described later) so that a slight deviation between the centers of the rotating shaft and the front balance shaft 22 can be absorbed.

In a valve train device of this embodiment, an intake camshaft 36 and an exhaust camshaft 37 which are disposed within the cylinder head cover 5 are constructed to be driven to rotate by the crankshaft 8. To be specific, a crankshaft sprocket 25b of the gear unit 25 press fitted on the left shaft portion 8c of the crankshaft 8 is connected by a timing chain 40 to an intermediate sprocket 38a rotationally supported by a support shaft 39 in the cylinder head 4. An intermediate gear 38 formed integrally on the intermediate sprocket 38a has a diameter smaller than that of the intermediate sprocket 38a, and meshes with intake and exhaust gears 41, 42 secured to end portions of the intake and the exhaust camshafts 36, 37. Note that the timing chain 40 passes through the chain compartments 3d, 4d formed on the left walls of the cylinder block 3 and the cylinder head 4.

The intermediate sprocket 38a and the intermediate gear 38b are rotationally supported by the support shaft 39 which passes through the chain

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compartment 4d on the cylinder head 4 in the direction in which the crankshaft extends along the cylinder bore axis A via two sets of needle bearings 44. The support shaft 39 is fixed at a flange portion 39a thereof to the cylinder head 4 with two bolts 39b. Note that reference numerals 39c, 39d denote a sealing gasket, respectively, at Fig. 4.

Here, commercially available standard bearings are adopted for the two sets of needle bearings 44, 44. A space adjusting collar 44a is disposed between the respective bearings 44, 44, and thrust washers 44b, 44b for receiving thrust load are provided at ends of the bearings. The thrust washer 44b is formed into a stepped shape having a large diameter portion which is brought into sliding contact with an end face of the intermediate sprocket and a stepped portion which protrudes axially toward the needle bearing 44.

Thus, since the space adjusting collar 44a is interposed between the two sets of bearings 44, 44, commercially available standard bearings can be adopted for the needle bearings by adjusting the length of the collar 44a, thereby reducing construction costs.

In addition, since the washer having the stepped configuration is adopted as the thrust washer 44b, the assembly of the intermediate sprocket 38a can be improved. Namely, the support shaft 39 is inserted from the outside such that the intermediate sprocket 38a and the intermediate gear 38b are disposed within the chain compartment 4d with the thrust washers 44b positioned at the ends so as not to fall therefrom. The thrust washer 44b can then be prevented from falling by locking the stepped portion thereof in a shaft hole in the intermediate sprocket 38a, and hence the assembling properties can be improved.

In addition, an oil hole 39e is formed in the support shaft 39 for supplying lubricating oil introduced from the cam compartment via an oil introducing bore 4e formed in the cylinder head 4 to the needle bearing 44.

Additionally, four weight reduction holes 38c and two inspection holes 38c (adapted to be used at the time of assembling and also functioning as weight reduction holes) are formed at intervals of 60 degrees. Then, an alignment or timing mark 38d is stamped on a tooth situated substantially at the center of the inspection hole 38c' for the intermediate gear 38b, and timing marks 41a, 42a are also

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stamped on two teeth of intake and exhaust camshaft gears 41, 42 which correspond to the timing marks 38d. Here, when aligning the left and right timing marks 38d, 38d with the timing marks 41a, 42a, the intake and exhaust camshaft gears 41, 42 are located at respective positions corresponding to a top dead center of a compression stroke.

Furthermore, timing marks 38e, 38e are also formed at portions of the intermediate sprocket 38a situated on a cover side mating surface 4f of the cylinder head 4 when the timing marks 38d align with 41a, 42a.

To align valve timings, first, the crankshaft 8 is held at a top dead center of a compression stroke by aligning the timing mark 25c (refer to Fig. 11) with the cylinder bore axis A. In addition, the intermediate sprocket 38a and the intermediate gear 38b attached to the cylinder head 4 via the support shaft 39 are positioned so that the timing mark 38e of the intermediate sprocket 38a aligns with the cover side mating surface 4f. The crankshaft sprocket 25b and the intermediate sprocket 38a are then connected by the timing chain 40. Then, the intake and exhaust camshaft gears 41, 42 on the intake and exhaust camshafts 36, 37 are brought into mesh engagement with the intermediate gear 38b. The inspection hole 38c' is used to confirm that the timing marks 41a, 42a align with the timing mark 38d on the intermediate gear 38b, and the intake and exhaust camshafts 36, 37 are fixed to an upper surface of the cylinder head 4 via cam carriers.

Thus, the inspection holes 38c' that also function as weight reduction holes to reduce the weight of the large diameter intermediate sprocket 38a are provided therein. The alignment of the timing marks 38d on the small diameter intermediate gear 38b with the timing marks 41a, 42a on the camshaft gears 41, 42 can be confirmed through the inspection holes 38c', and the meshing positions of the intermediate gear 38b with the camshaft gears 41, 42 can be visually confirmed in an easy and reliable fashion while the small diameter intermediate gear 38b is placed on the back of the large diameter intermediate sprocket 38a, thereby permitting the alignment of the valve timings without any problem.

In addition, since the intermediate gear 38b can be disposed on the back side of the intermediate sprocket 38a, the dimension from the camshaft gears 41, 42 which meshes with the intermediate gear 38b to a cam nose 36a can be

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made shorter, whereby the torsional angle of the camshaft can be decreased to the extent that the dimension is made so shorter, thereby reducing an area surrounding the camshafts.

For example, in a case where the intermediate gear 38b is disposed on a front side of the intermediate sprocket 38a, while the valve timings can easily be aligned, the dimension from the camshaft gears 41, 42 to the cam nose is increased, and the torsional angle of the camshafts is increased to such an extent that the dimension is increased, thereby reducing the control accuracy of valve opening and closing timings.

In addition, in a case where the intermediate gear 38b is disposed in front of the intermediate sprocket 38a, a space between the intermediate sprocket support shaft 39 and the camshafts 36, 37 needs to be expanded in order to avoid any interference between the intermediate sprocket 38a and the camshaft 36, 37, thereby causing a concern that the area surrounding the camshafts is enlarged.

Here, a backlash adjusting mechanism is provided between the intermediate gear 38b and the camshaft gears 41, 42. This adjusting mechanism has a construction in which the intake camshaft gear 41 and the exhaust camshaft gear 42 are made up of two gears, such as a driving gear 46, being a power transmission gear, and a shift gear 45 being on an adjusting gear. The angular positions of the driving gear 46 and the shift gear 45 can be adjusted.

In particular, the shift gear 45 and the driving gear 46 are fixed to flange portions 36b, 37b formed at the respective end portions of the camshafts 36, 37, in such a manner that the angular positions thereof can be adjusted by four circumferentially long elongated holes 45a, 46a and four long bolts 68a. A clearance portion 46b is cut and formed in the driving gear 46 and disposed outwardly, and only the shift gear 45 is fixed in such a manner that the angular position thereof can be adjusted with two elongated holes 45b and two short bolts 68b by making use of the clearance portion 46.

A backlash adjustment is implemented according to the following procedure. Note that in the engine according to this embodiment, the intermediate gear 38b rotates counterclockwise as shown in Fig. 3 when viewed from the left-hand side of the engine. Consequently, both the intake camshaft gear 41 and the

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exhaust camshaft gear 42 rotate clockwise. In addition, here, while the backlash adjustment will be described with respect to the intake camshaft gear 41, the same description applies with respect to the exhaust camshaft gear 42.

5 First, all the fixing bolts 68a, 68b of the intake camshaft gear 41 are loosened, and the shift gear 45 is rotated clockwise so that the front side surfaces of teeth of the shift gear 45 in the clockwise direction slightly abut with the rear side surfaces of teeth of the intermediate gear 38b in the counterclockwise direction. In this state, the shift gear 45 is fixed to the flange portion 36b of the camshaft 36 with two short bolts 68b. Then, the driving gear 46 is rotated counterclockwise so that
10 the front side surfaces of teeth of the driving gear 46 in the counterclockwise direction, being the driven surfaces, abut with the front side surfaces of the intermediate gear 38b in the counterclockwise direction, being the driving surfaces, thereby obtaining a required backlash. In this state, four long bolts 68a are tightened up, and the driving gear 46 and the shift gear 45 are fixed to the intake
15 camshaft 36.

Thus, since the intake and exhaust camshaft gears 41, 42 are made up of the driving gear 46, being the power transmission gear, and the shift gear 45, being the adjusting gear, adapted to rotate relatively to the driving gear, the backlash can be adjusted by rotating the shift gear 45 relative to the driving gear 46,
20 either forward or backward in the rotating directions.

Note that while, in this embodiment, both the driving gear 46 and the shift gear 45 which constitute the camshaft gears 41, 42 are described as being able to rotate relatively to the camshafts, in other alternative embodiments either one of the driving gear 46 and the shift gear 45 may be adapted to rotate relatively,
25 and the other gear may then be integrated into the camshaft. In this case, it is desirable that the gear integrated into the camshaft constitutes the power transmission gear. Even when constructed in this way, similar functions and advantages to those obtained by the present embodiment can be obtained.

In addition, while in this embodiment, the invention is described for the
30 chain drive method construction, the invention can of course also be applied to a drive method construction using a toothed belt.

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Next, the lubricating systems will be described. An engine lubrication system 50 according to this embodiment is constructed such that lubricating oil stored within a separate lubricating oil tank 51 is picked up and pressurized by a lubricating oil pump 52 via a down tube 56c on a vehicle body frame, lubricating oil discharged from the pump 52 is divided into three systems (such as a cam lubricating system 53, a transmission lubricating system 54 and a crank lubricating system 55) so as to be supplied to parts to be lubricated at the respective systems, and lubricating oil used for lubricating the respective parts is returned to the lubricating oil tank 51 by making use of pressure fluctuation occurring within the crank compartment 2c as the piston 6 reciprocates vertically.

The lubricating oil tank 51 is formed integrally within a space surrounded by a head pipe 56a, a main tube 56b, the down tube 56c and a reinforcement bracket 56d of the vehicle body frame 56. This lubricating oil tank 51 communicates with a cross pipe 56e which connects lower portions of the down tube 56c via the down tube 56c.

The cross pipe 56e communicates with a pick-up port of the lubricating oil pump 52 via an outlet tube 56f connected thereto, an oil hose 57a, a joint pipe 57b and a pick-up passageway 58a formed in a crankcase cover 10. A discharge port of the lubricating oil pump 52 is connected to an oil filter 59 via an oil discharge passageway 58b, an external portion connecting chamber 58c and an oil passageway 58d and is divided into the three lubrication systems 53, 54, 55 on a secondary side of the oil filter 59.

The oil filter 59 is constructed such that an oil element 59e is disposed in a filter compartment 59d defined by detachably attaching a portion of a cover 47 to a filter recessed portion 10b provided in the right case cover 10, by setting part thereof further back from the rest.

The cam lubricating system 53 is generally such that a lower end of a vertical member 53a of a T-shaped lubricating oil pipe is connected to a cam side outlet 59a of an oil passageway formed on the outside of the filter recessed portion 10b. Left and right ends of a horizontal member 53b of the lubricating oil pipe are connected to a camshaft oil supply passageway 53c. Lubricating oil is thereby

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supplied to parts, such as bearings of the camshafts 36, 37, which are lubricated via the passageway 53c.

The transmission lubrication system 54 has the following construction.

A right transmission oil supply passageway 54a formed within the right case portion 2b is connected to a transmission side outlet 59b of the oil filter 59, and the oil supply passageway 54a communicates with the interior of a main shaft bore 14a formed in the main shaft 14 along the axial center thereof via a left transmission oil passageway 54b formed in the left case portion 2a. The main shaft bore 14a communicates with sliding portions between the main shaft 14 and change-speed gears via a plurality of branch bores 14b, whereby lubricating oil supplied to the main shaft bore 14a passes through the branch bores 14b to be supplied to the sliding portions.

In addition, an intermediate portion of the left transmission oil passageway 54b communicates with a bolt bore 60a through which a case bolt 60 is inserted for connecting the left and right case portions 2a, 2b together. This bolt bore 60a is such as to be formed by forming a bore having an inside diameter slightly larger than the outside diameter of the case bolt 60 in tubular boss portions 60c, 60c. These tubular boss portions 60c, 60c are formed to face and abut with each other on the mating surface between the left and right case portions 2a, 2b. The boss portion 60c is situated in the vicinity of a portion where a gear train on the main shaft 14 meshes with a gear train on the drive shaft 15, and a plurality of branch bores 60b are formed from which lubricating oil within the bolt bore 60a is spouted out toward the gear trains meshing portion. Note that the bolts 60 shown in Fig. 19 as being developed into the left and right case portions are the same bolt.

Furthermore, a right end portion of the bolt bore 60a communicates with a drive shaft bore 15a formed in the drive shaft 15 along the axial center thereof via a communication bore 54c. The drive shaft bore 15a is closed by a partition wall 15c at a left-hand side portion and communicates with sliding portions between the drive shaft 15 and driving gears via a plurality of branch bores 15b. Thus, lubricating oil supplied into the drive shaft bore 15a passes through the branch bores 15b to be supplied to the sliding portions.

The crank lubricating system 55 has the following construction. A crank oil supply passageway 55a is formed in the filter cover 47 in such a manner as to extend from a crank side outlet 59c toward the lubricating oil pump 52. The passageway 55 is made to communicate with a communication bore 62a which is
5 formed in a rotating shaft 62 of the lubricating oil pump 52 to pass therethrough along the axial center thereof. Furthermore, the communication bore 62a communicates with a crank oil supply bore 8e formed in the crankshaft 8 to pass therethrough along the axial center thereof via a connecting pie 64. Then, this crank oil supply bore 8e communicates with the interior of a pin bore 65a in a crank
10 pin 65 via a branch bore 8f, and the pin bore 65a is made to open to the rotating surface of a needle bearing 7b at a large end portion 7a of a connecting rod 7 via a branch bore 65b. Thus, lubricating oil filtered in the oil filter 59 is supplied to the rotating surface of the needle bearing 7b.

The lubricating oil pump 52 has the following construction. A pump
15 compartment 61c is provided in a right case 61b of a two-piece casing made up of left and right cases 61a, 61b by setting a relevant portion of the case further back from the rest, and a rotor 63 is disposed rotationally within the pump compartment 61. The rotating shaft 62 is inserted into the rotor 63 along the axial center thereof in such a manner as to pass therethrough to be disposed in place therein, and the
20 rotating shaft 62 and the rotor 63 are fixed together with a pin 63a. Note that the oil pick-up passageway 58a and oil discharge passageway 58b are connected to a pump compartment on the upstream side of the left case 61a and a pump compartment on the downstream side of the left case 61a. In addition, reference numeral 66 denotes a relief valve for maintaining the discharge pressure of the
25 lubricating oil pump 52 lower than a predetermined value. The relief valve 66 is adapted to relieve the pressure in the lubricating oil pump 52 toward the oil pick-up passageway 58a side when the pressure on the discharge side reaches or exceeds the predetermined value.

The rotating shaft 62 is a tubular shaft which passes through the pump
30 case 61 in the axial direction and opens to the crank oil supply passageway 55a at a right end portion thereof as shown in the Fig. 20. In addition, a power transmitting flange portion 62b is formed integrally at a left end portion of the rotating shaft 62

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as shown in the drawing. The flange portion 62b faces a right end face of the crankshaft 8, and the flange portion 62b and the crankshaft 8 are connected together by an Oldham's coupling 67 to absorb a slight deviation of the centers of the shafts.

5 The Oldham's coupling 67 is constructed such that a coupling plate 67a is disposed between the crankshaft 8 and the flange portion 62b, a pin 67b set in the end face of the crankshaft 8 and a pin 67c set in the flange portion 62b are inserted into a connecting bore 67d in the coupling plate 67a.

10 In addition, the connecting pipe 64 connects a right end opening in the crankshaft 8 to a left end opening in the rotating shaft 62, and sealing is provided by an oil seal 64a between the inner circumference of the crankshaft opening and the inner circumference of the rotating shaft opening and the outer circumference of the connecting pipe 64.

15 As has been described above, the crank compartment 2c is defined separately from the other transmission compartment 2d, the flywheel magnet compartment 9a and the clutch compartment 10a. An oil return mechanism is constructed in which the pressure within the crank compartment 2c is fluctuated between positive and negative values as the piston 6 strokes, so that lubricating oil in the respective compartments is returned to the lubricating oil tank 51 by virtue of
20 the pressure fluctuation.

 In detail, a discharge port 2g and a suction or pick-up port 2h are formed in the crank compartment 2c. A discharge port reed valve 69 (adapted to open when the pressure within the crank compartment 2c is positive) is disposed in the discharge port 2g, and a pick-up port reed valve 70 (adapted to open when
25 the pressure within the crank compartment 2c is negative) is disposed in the pick-up port 2h. See Fig. 18.

 The discharge port 2g communicates with the clutch compartment 10a from the crank compartment 2c via a communication bore 2i, then communicates with the transmission compartment 2d from the clutch compartment 10a via a
30 communication bore 2j. Furthermore, the transmission compartment 2d communicates with the flywheel magnet compartment 9a via a communication bore 2k. A return port 2m formed to communicate with the flywheel magnet compartment

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9a communicates with the lubricating oil tank 51 via a return hose 57c, an oil strainer 57d and a return hose 57e.

5 Here, a guide plate 2n is provided at the return port 2m. This guide plate 2n functions to ensure the discharge of lubricating oil by modifying the return port 2m to provide a narrow gap (described in Fig. 18 as "a") between a bottom plate 2p and itself and to secure a wide width b.

10 Additionally, an oil separating mechanism is connected to the lubricating oil tank 51 for separating oil mists contained in the air within the tank. By virtue of centrifugal force, oil mists are separated and returned to the crank compartment 2c. This oil separating mechanism has an introduction hose 72a which is connected to an upper portion of the lubricating oil tank 51 at one end, and which is tangentially connected to an upper portion of a cone-shaped separating compartment 71 at the other end. A return hose 72b is connected to a bottom portion of the separating compartment 71 at one end and is connected to the pick-up port 2h of the crank compartment 2c at the other end. Note that the air from which the oil mists are separated is discharged to the atmosphere via an exhaust hole 72c.

20 Thus, according to this embodiment, since the crank compartment 2c comprises a substantially closed space wherein the pressure fluctuates as the piston 6 reciprocates vertically, lubricating oil in the crank compartment 2c is sent back to the lubricating oil tank 51 through the use of this pressure fluctuation and the need for an exclusive oil sending pump, or a scavenging pump, is obviated, the construction of the engine is simplified, and costs are likely reduced.

25 In addition, the discharge port reed valve 69, being an outlet side check valve which is adapted to open when the pressure in the crank compartment 2c increases and to close when the pressure lowers, is disposed in the vicinity of the location where the oil sending passageway is connected to the crank compartment 2c. The selection of this location ensures that the lubricating oil within the crank compartment 2c can be sent back to the lubricating oil storage tank 51 in a more reliable fashion.

30 In addition, a portion above the oil level within the lubricating oil storage tank 51 is connected to the crank compartment 2c via the return hoses 72a,

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72b and the discharge port reed valve 70, being a pick-up side check valve which is adapted to open when the pressure in the crank compartment 2c lowers and to close when the pressure increases, is provided in the vicinity of the location where the return hoses are connected to the crank compartment 2c. As a result, the required air is pumped into the crank compartment 2c when the piston 6 moves upwardly, and the inside pressure of the crank compartment 2c increases when the piston 6 lowers. Lubricating oil within the crank compartment 2c can be discharged in a more reliable fashion.

In a case where no air is supplied from the outside to the interior of the crank compartment 2c, a negative pressure, or a lower positive pressure, is formed inside the crank compartment 2c, and oil cannot be sent out properly.

Furthermore, since the centrifugal lubricating oil mist separating mechanism 71 used for separating lubricating oil mist is interposed at the intermediate position along the length of the return passageways 72a, 72b, so that lubricating oil mist so separated is returned to the crank compartment 2c via the return hose 72b and the air from which the mist content is removed is discharged to the atmosphere. Since only lubricating oil mist can be returned to the crank compartment 2c, the reduction in efficiency that would otherwise occur if an excessive amount of air was allowed to flow into the crank compartment 2c is avoided. Lubricating oil can thereby be returned to the crank compartment 2c in a more reliable fashion while preventing atmospheric pollution.

In addition, the lubricating oil pump 52 is disposed so as to be connected to the one end of the crankshaft 8, and the discharge port of the lubricating oil pump 52 communicates with the crank oil supply bore (an in-crankshaft oil supply passageway) 8e formed within the crankshaft 8 via the communication bore (an in-pump oil supply passageway) 62a formed within the lubricating oil pump 52 and the connecting pipe 64. Lubricating oil can thereby be supplied to the parts of the crankshaft 8 that need lubrication using a simple and compact construction.

In addition, the crankshaft 8 and the lubricating oil pump 52 are connected together by the Oldham's coupling 67 which can absorb the displacement of the shafts in the direction normal thereto, and the communication

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bore 62a and the crank oil supply bore 8e are made to communicate with each other via the connecting pipe 64. The resilient O ring 64a is interposed between the connecting pipe 64, the communicating bore 62a, and the crank oil supply bore 8e. Even in the event that the centers of the crankshaft 8 and the pump shaft 62 are
5 caused to deviate slightly from each other, lubricating oil can be supplied to the parts needing to be lubricated without any problem, thereby making it possible to secure the required lubricating properties.

Furthermore, the tubular boss portion 60c is formed in the vicinity of the main shaft 14 and the drive shaft 15 which constitute the transmission. The
10 crankcase connecting case bolt 60 is inserted into the bolt bore 60a in the boss portion 60c so that the space between the inner circumferential surface of the bolt bore 60a and the outer circumferential surface of the case bolt 60 forms the lubricating oil passageway. The branch bore 60b, being the lubricating oil supply bore, is directed to the change-speed gears at the boss portion 60c. As a result,
15 lubricating oil can be supplied to the meshing surfaces of the change-speed gears while avoiding the need to provide an exclusive lubricating oil supply passageway.

In addition, the other end of the lubricating oil passageway defined by the inner circumferential surface of the bolt bore 60c and the outer circumferential surface of the case bolt 60 communicates with an opening of the drive shaft bore
20 15a formed within the drive shaft 15 and situated opposite to an outlet side of the bore. As a result, lubricating oil can be supplied to the portions on the drive shaft 15 brought into sliding contact with the change-speed gears while avoiding the need for providing an exclusive lubricating oil supply passageway.

25 INDUSTRIAL APPLICABILITY

According to one embodiment of the invention, since head bolts which fasten the cylinder head and the cylinder body together are screwed into the case side flange portion, the load applied to the cylinder body is reduced and the load generated by the combustion pressure is partially borne by the head bolts. The
30 stress generated in the cylinder body can be reduced accordingly, thereby making it possible to improve the durability of the cylinder body.

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In the case of a construction in which a head side flange portion of a cylinder body and a cylinder head are simply fastened together with bolts, and a case side flange portion and a crankcase are simply fastened together with bolts, the load generated by the combustion pressure is completely transferred to the cylinder body, which may then be of insufficient strength depending upon its thickness, and in the worst case, a crack may be generated in the cylinder body. This embodiment of the present invention avoids this problem.

According to another embodiment of the invention, since the flange screw-through head bolt and the case bolt overlap each other by the distance which is substantially the same as the thickness of the case side flange portion, the flange screw-through head bolts ensure that part of the load generated by the combustion pressure is transferred to the case side flange portion, thereby reducing the load applied to the intermediate portion of the cylinder body.

According to another embodiment of the invention, since the flange screw-through bolt and the case bolt are disposed close to each other, when viewed in the axial direction of the cylinder bore, the flange screw-through head bolts ensure that part of the load generated by the combustion pressure is transferred to the case side flange portion. Furthermore, the case side flange portion transfers this load to the crankcase via the case bolts, thereby reducing the load applied to the cylinder body in a secure fashion.

According to another embodiment of the invention, the case bolt is either disposed such that the distance from the case bolt to the first straight line which passes through the axis of the cylinder bore and normal to the crankshaft becomes shorter than the distance from the flange screw-through head bolt to the first straight line when viewed in the axial direction of the cylinder bore, or in the alternative, the case bolt is situated closer to the center of the cylinder bore in the crankshaft direction. As a result, the dimension in the crankshaft direction of the mating surface of the crankcase which is attached to the cylinder body can be reduced to the vicinity of the positions where the flange screw-through head bolts are disposed, and the dimension in the crankshaft direction of the crankcase can be reduced.

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According to another embodiment of the invention, either the flange screw-through head bolts are screwed into the case side flange portion of the cylinder body or, in the alternative, the flange screw-through head bolts are not screwed into the crankcase and do not interfere with the crankshaft web incorporated in the crankcase. The flange screw-through head bolts can either be disposed such that the distance to the second straight line which passes through the axis of the cylinder bore and which is parallel to the crankshaft becomes shorter than the distance from the case bolt to the second straight line or, in the alternative, the flange screw-through head bolts can be situated closer to the crankshaft side. As a result, the dimension of the cylinder body in the direction normal to the crankshaft can be reduced.

According to another embodiment of the invention, since the axial part of the flange screw-through head bolt is exposed to the outside, the weight of the cylinder body can be reduced.

According to yet another embodiment of the invention, the three head bolts are disposed on either side of the cylinder bore across the second straight line, and the central head bolt along the second straight line, and the central head bolt along the second straight line is situated apart from the axis of the cylinder. However, since the head bolt is set at a length which does not reach the case side flange portion, the portion of the case side flange portion which corresponds to the center can be minimized, thereby avoiding any requirement to enlarge the cylinder body and the crankcase.

According to still another embodiment of the invention, since the flange screw-through head bolt is disposed between the cylinder bore and the chain compartment formed on the side to the cylinder bore, the flange screw-through head bolt can be disposed by making the effective use of the dead space formed therebetween.

According to a further embodiment of the invention, since the flange screw-through head bolt is screwed into the case side flange portion at one end, and is fastened and fixed to the cylinder head with the cap nut at the other end thereof, the cylinder head can be removed without requiring a large space above the cylinder head, thereby improving the maintenance properties of the engine.

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THE EMBODIMENTS OF THE PRESENT INVENTION IN WHICH AN EXCLUSIVE PROPERTY OR PRIVILEGE IS CLAIMED ARE DEFINED AS FOLLOWS:

1. An engine fastening structure in which a cylinder body and a cylinder head are stacked on and fastened to a crankcase, wherein a case bolt pass through a case side flange portion formed at a crankcase side end portion of the cylinder body and are screwed into a cylinder body side end portion of the crankcase to fasten the cylinder body to the crankcase, wherein at least part of a head bolt which fasten the cylinder head and the cylinder body together is made to be a flange screw-through head bolt, and wherein the flange screw-through head bolt is screwed into a screw portion formed in the case side flange portion.

2. An engine fastening structure according to claim 1, wherein the flange screw-through head bolt and the case bolt overlap each other by a distance which is substantially the same as the thickness of the case side flange portion in the axial direction of a cylinder bore.

3. An engine fastening structure according to claim 1 or 2, wherein the flange screw-through bolt and the case bolt are disposed close to each other, when viewed in the axial direction of the cylinder bore.

4. An engine fastening structure according to any one of claims 1 to 3, wherein the case bolt is disposed such that a distance from the case bolt to a first straight line which passes through the axis of the cylinder bore and which is normal to a crankshaft becomes shorter than a distance from the flange screw-through head bolt to the first straight line, when viewed in the axial direction of the cylinder bore.

5. An engine fastening structure according to any one of claims 1 to 4, wherein the flange screw-through head bolt is disposed such that a distance from the head bolt to a second straight line which passes through the axis of the cylinder bore and which is parallel to the crankshaft becomes shorter than a distance from the case bolt to the second straight line, when viewed in the axial direction of the cylinder bore.

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6. An engine fastening structure according to any one of claims 1 to 5, wherein an upper flange portion is formed at a cylinder head side end portion of the cylinder body, wherein the flange screw-through head bolt passes the upper flange portion and is screwed into the case side flange portion, and wherein a part of the flange screw-through head bolt which is between the case side flange portion and the upper flange portion is exposed to the outside.

7. An engine fastening structure according to claim 5, wherein at least three head bolts are disposed on either side of the cylinder bore across the second straight line, when viewed in the axial direction of the cylinder bore, and wherein the central head bolt along the second straight line is set to have a length which does not reach the case side flange portion.

8. An engine fastening structure according to claim 1, 2, 3, 4, or 6, wherein at least three head bolts are disposed on either side of the cylinder bore across a second straight line, when viewed in the axial direction of the cylinder bore, and wherein the central head bolt along the second straight line is set to have a length which does not reach the case side flange portion.

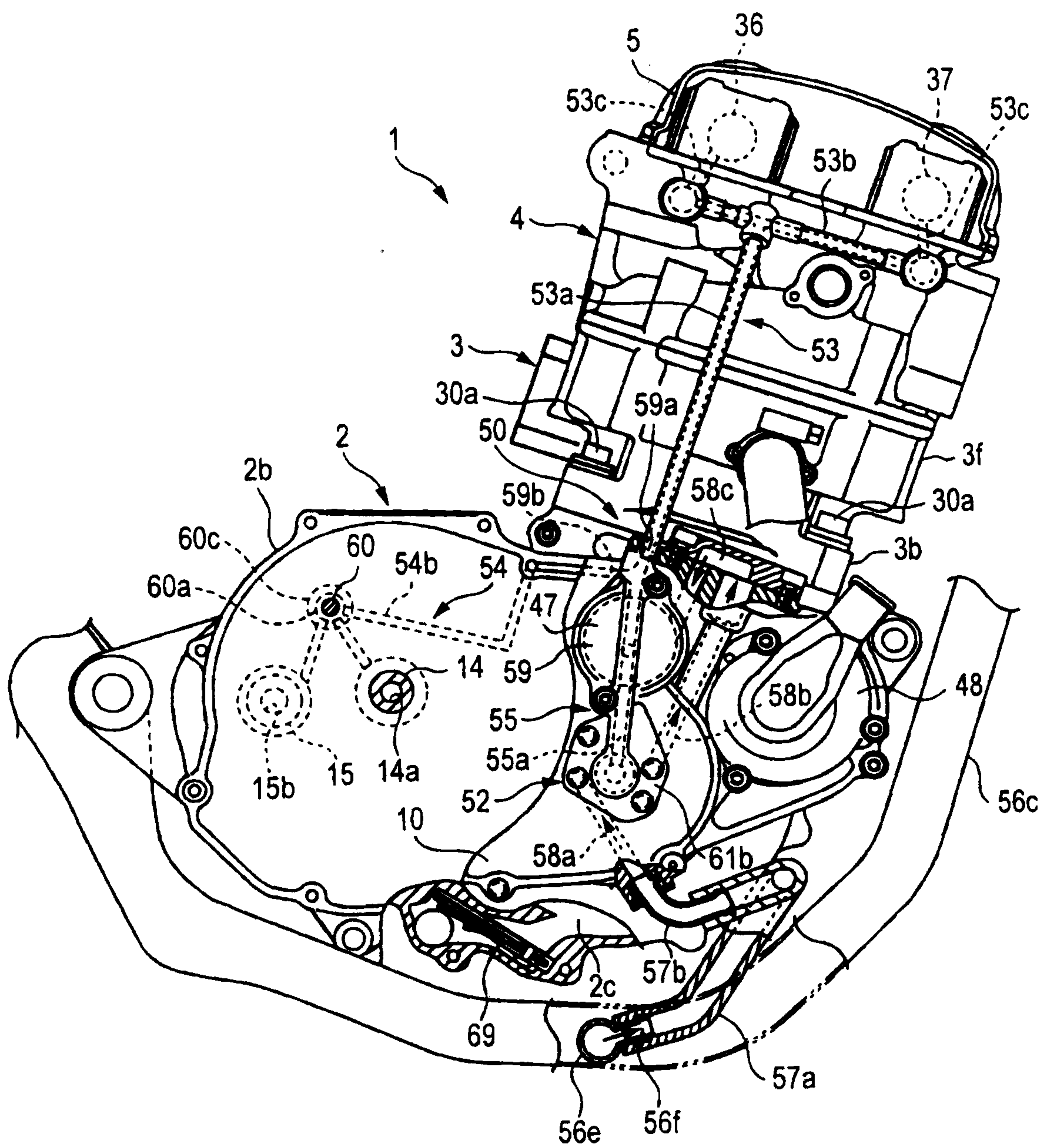
9. An engine fastening structure according to any one of claims 1 to 8, wherein the flange screw-through head bolt is disposed between a chain compartment formed on a side to the cylinder bore in which a camshaft driving chain which connects the crankshaft to a camshaft is disposed and the cylinder bore.

10. An engine fastening structure according to any one of claims 1 to 9, wherein the flange screw-through head bolt is screwed into the case side flange portion at one end and is fastened and fixed to the cylinder head with a cap nut at the other end thereof.

11. An engine fastening structure according to claim 1, wherein a tip of the flange screw-through head bolt is positioned closer to a cylinder body side than a cylinder body side end surface of the crankcase.

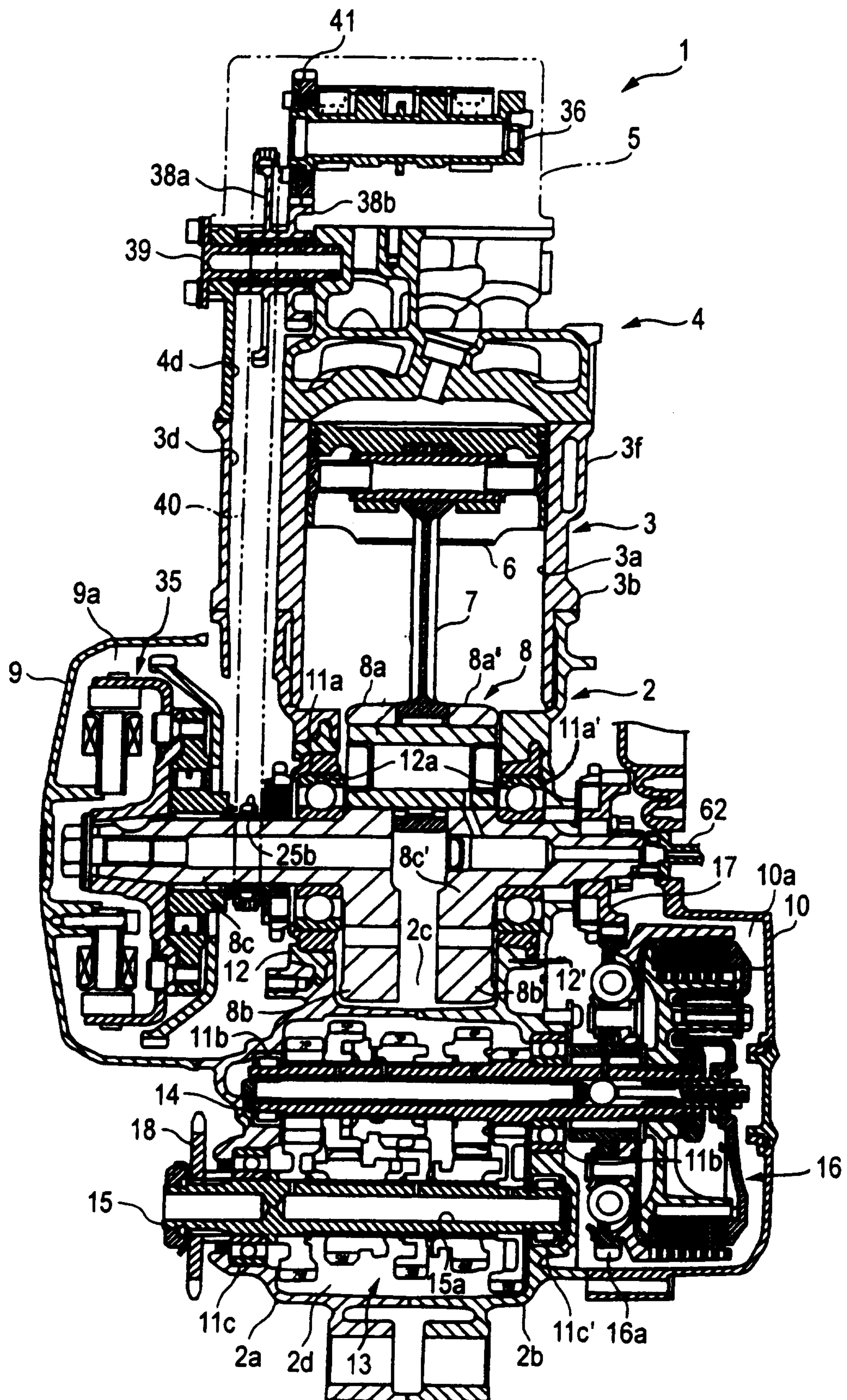
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FIG. 1



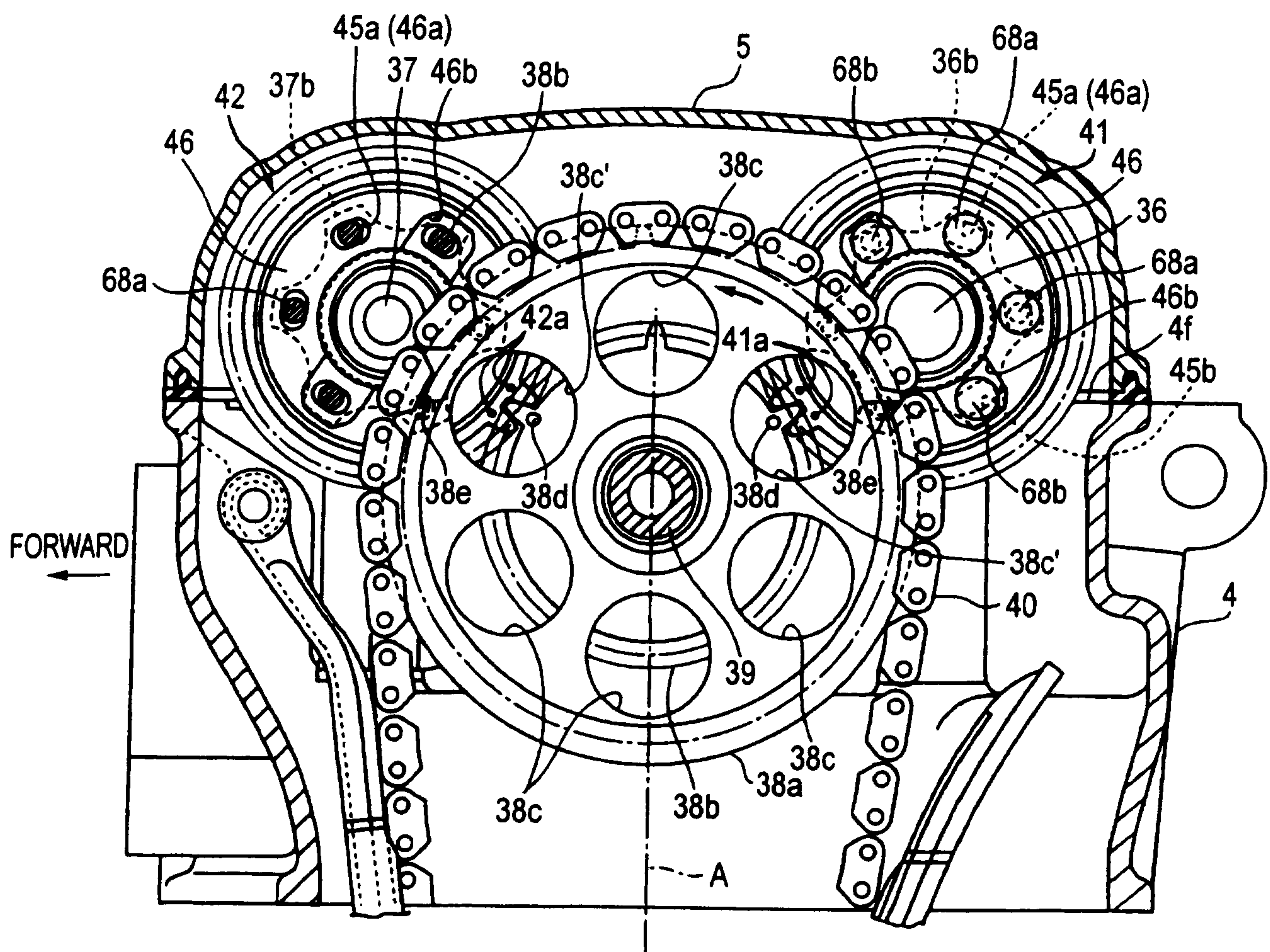
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FIG. 2



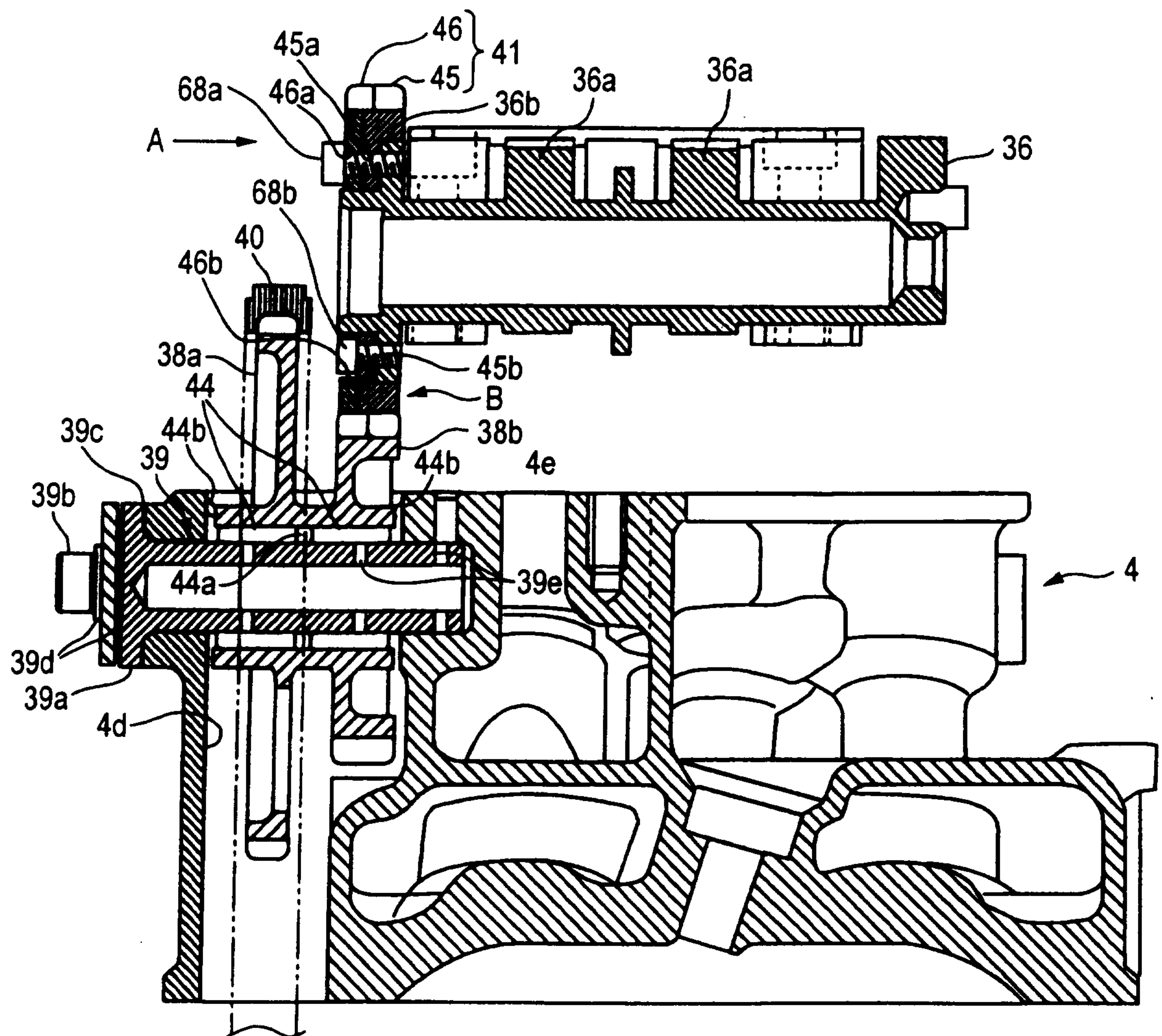
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FIG. 3



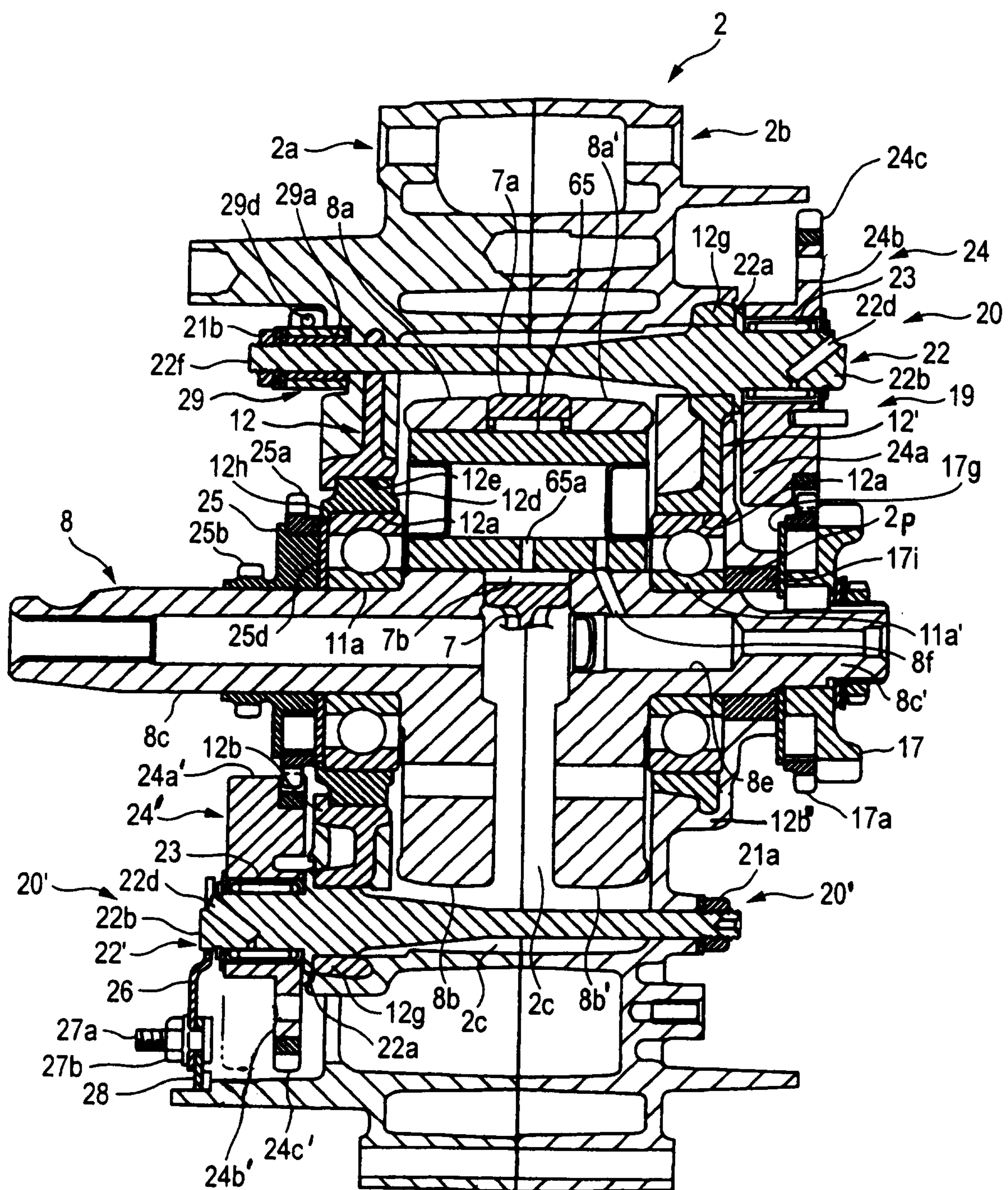
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FIG. 4



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FIG. 5



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FIG. 6

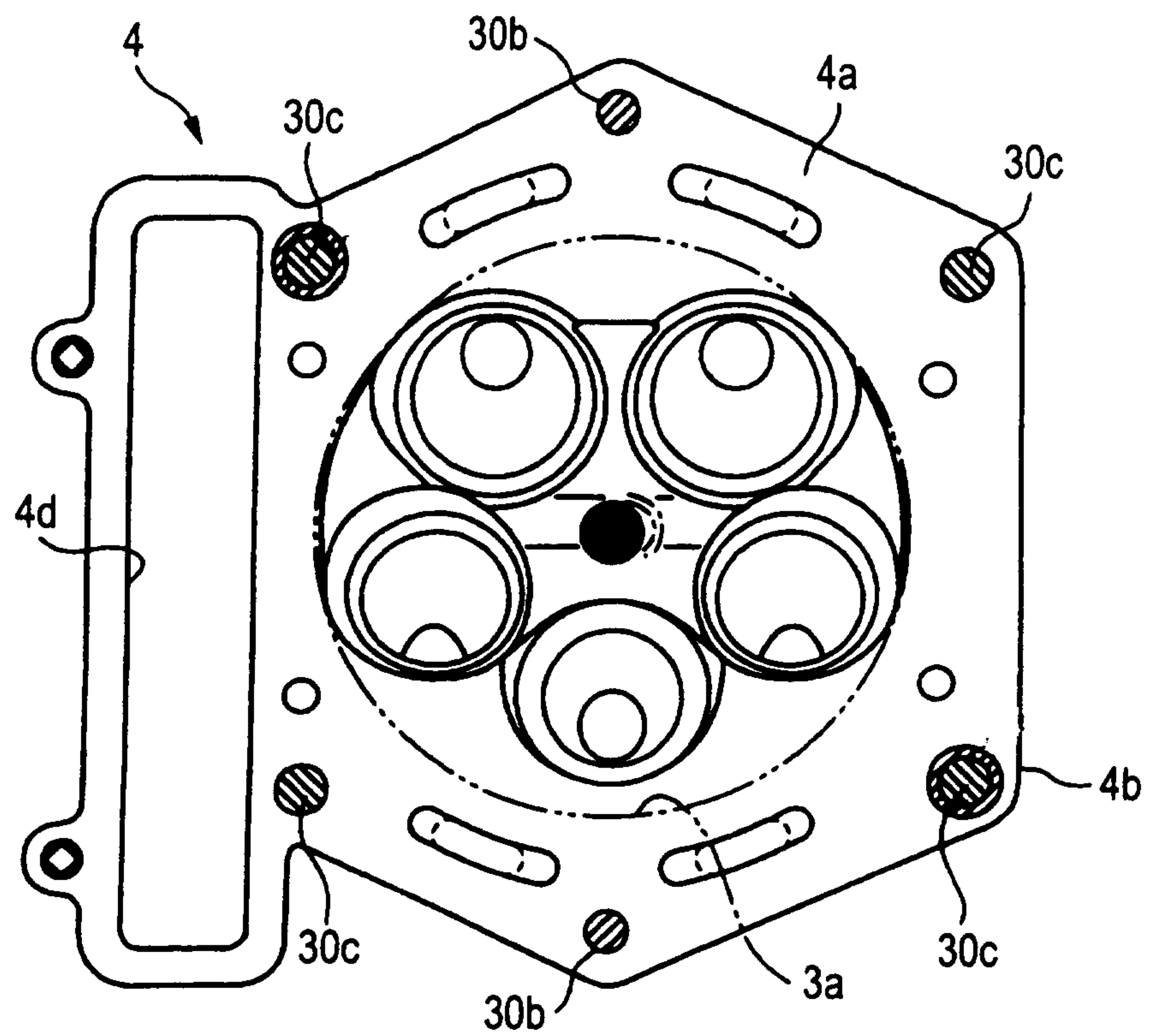
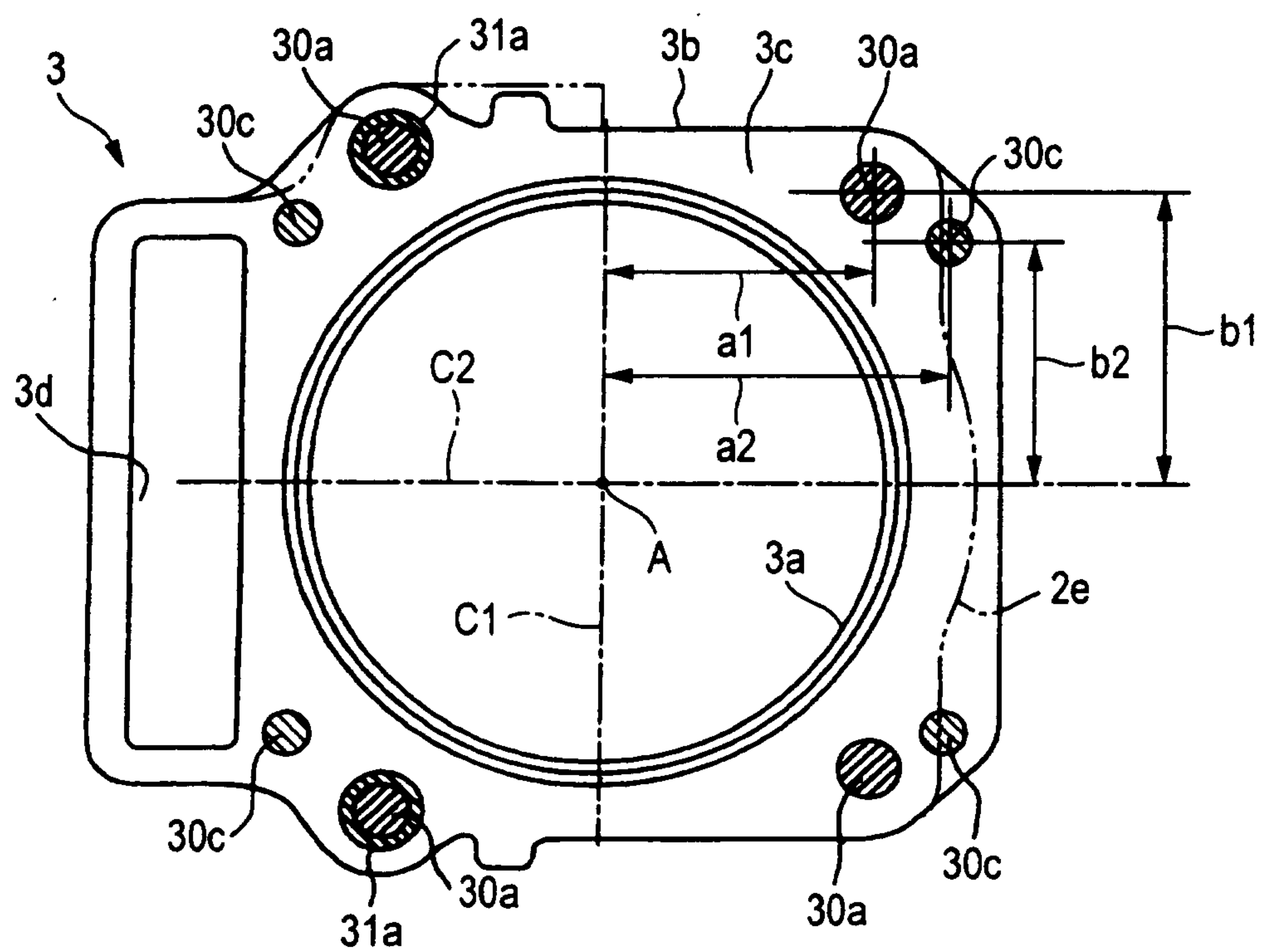
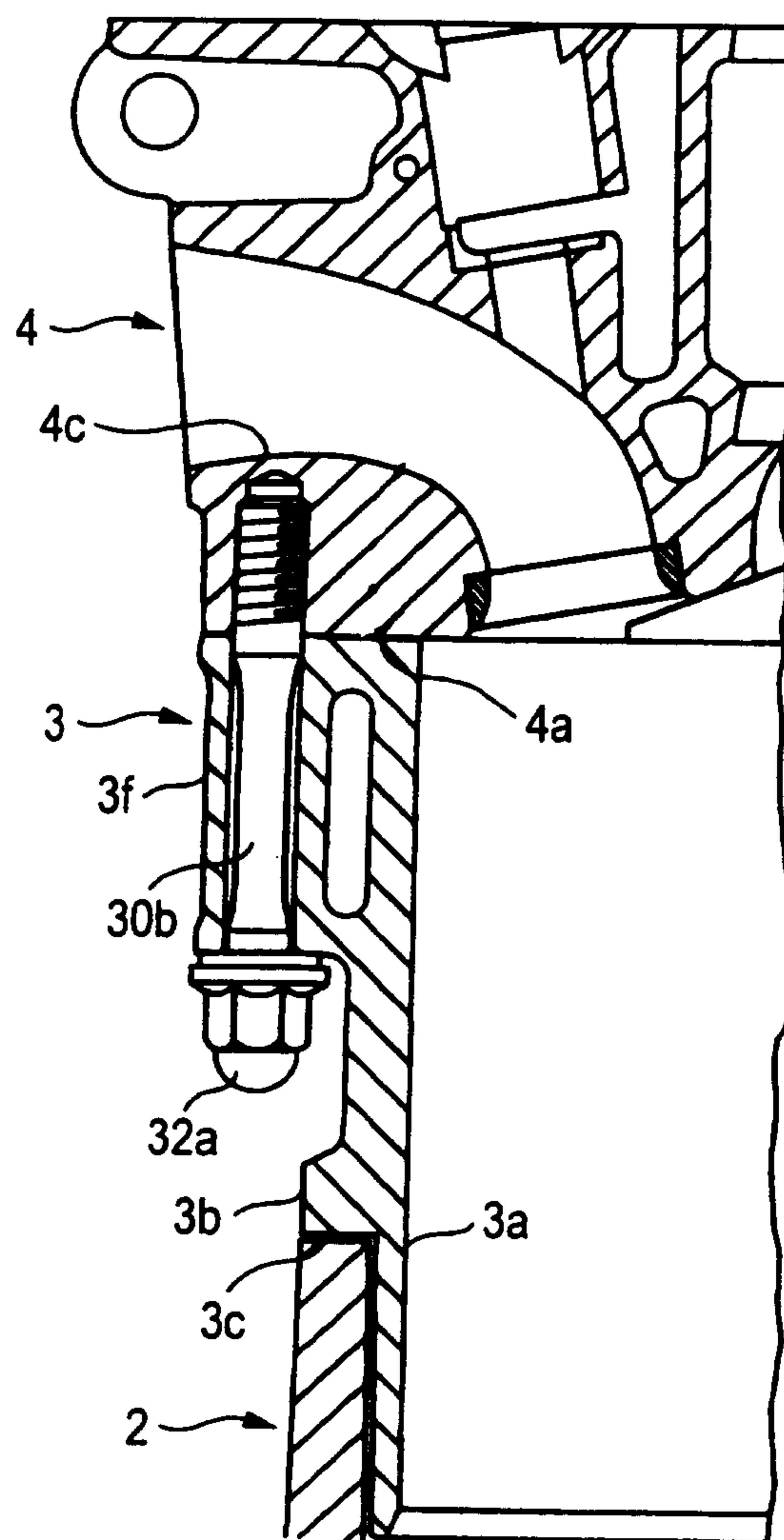


FIG. 7

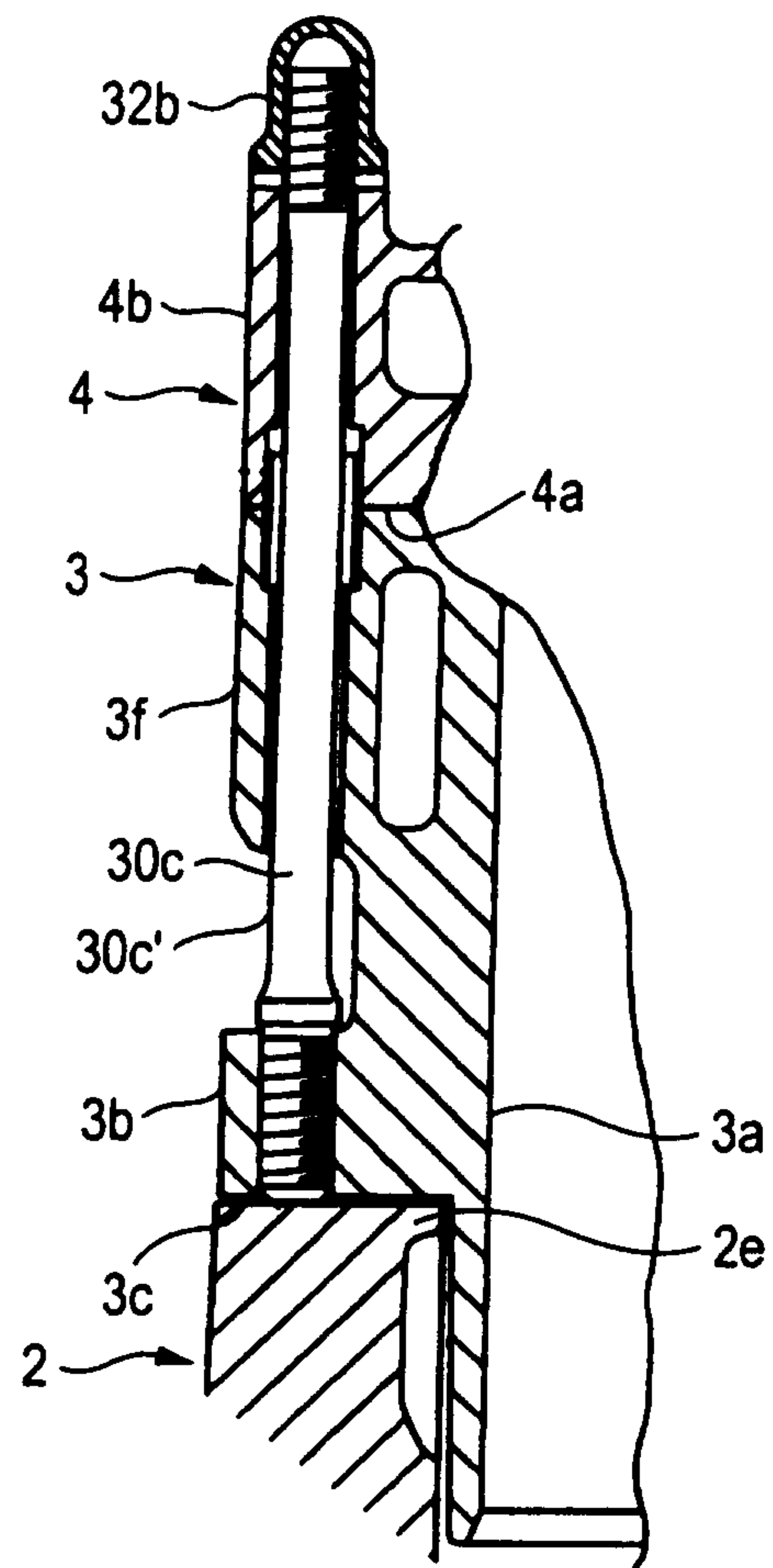


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FIG. 8

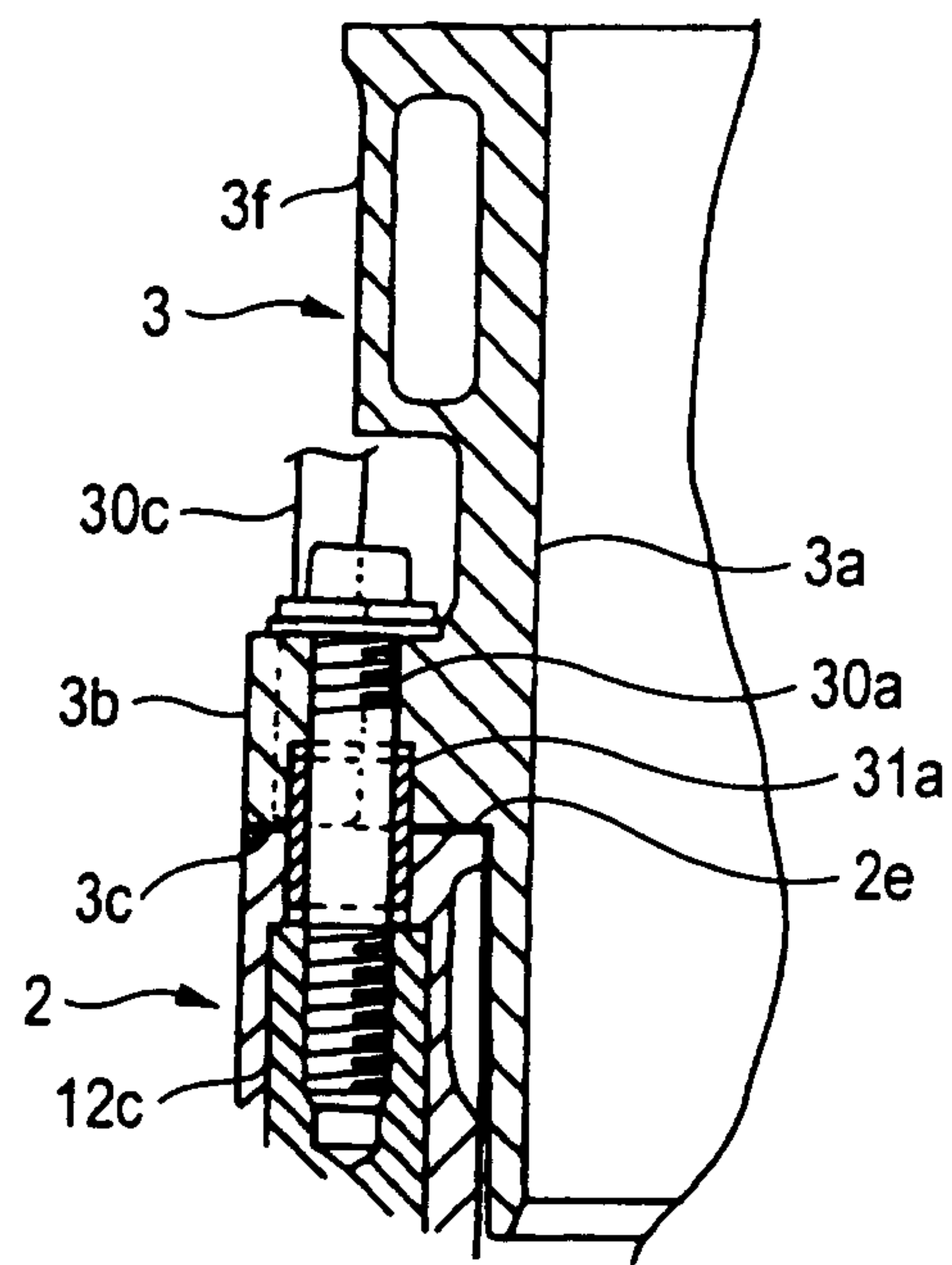


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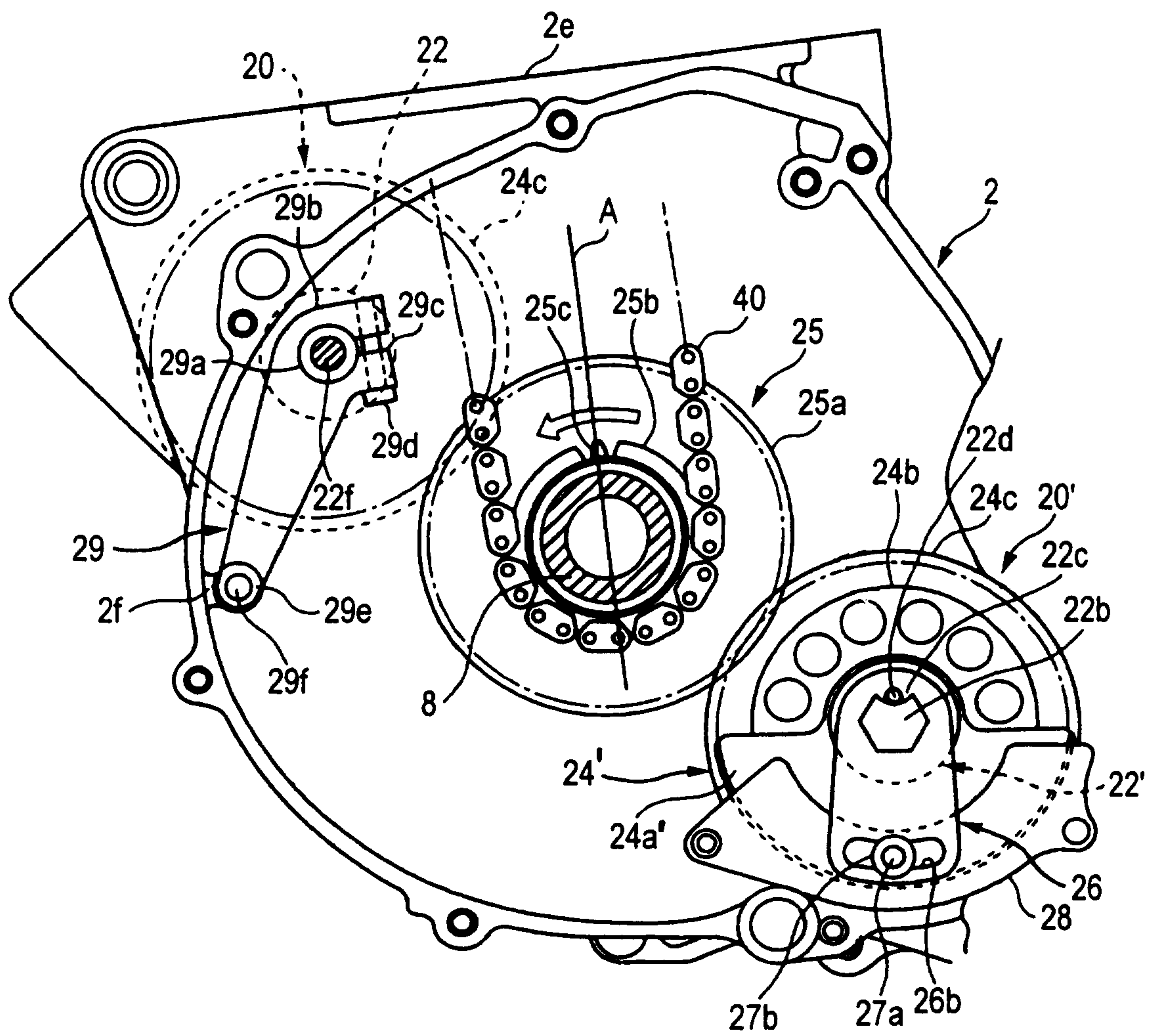
FIG. 9

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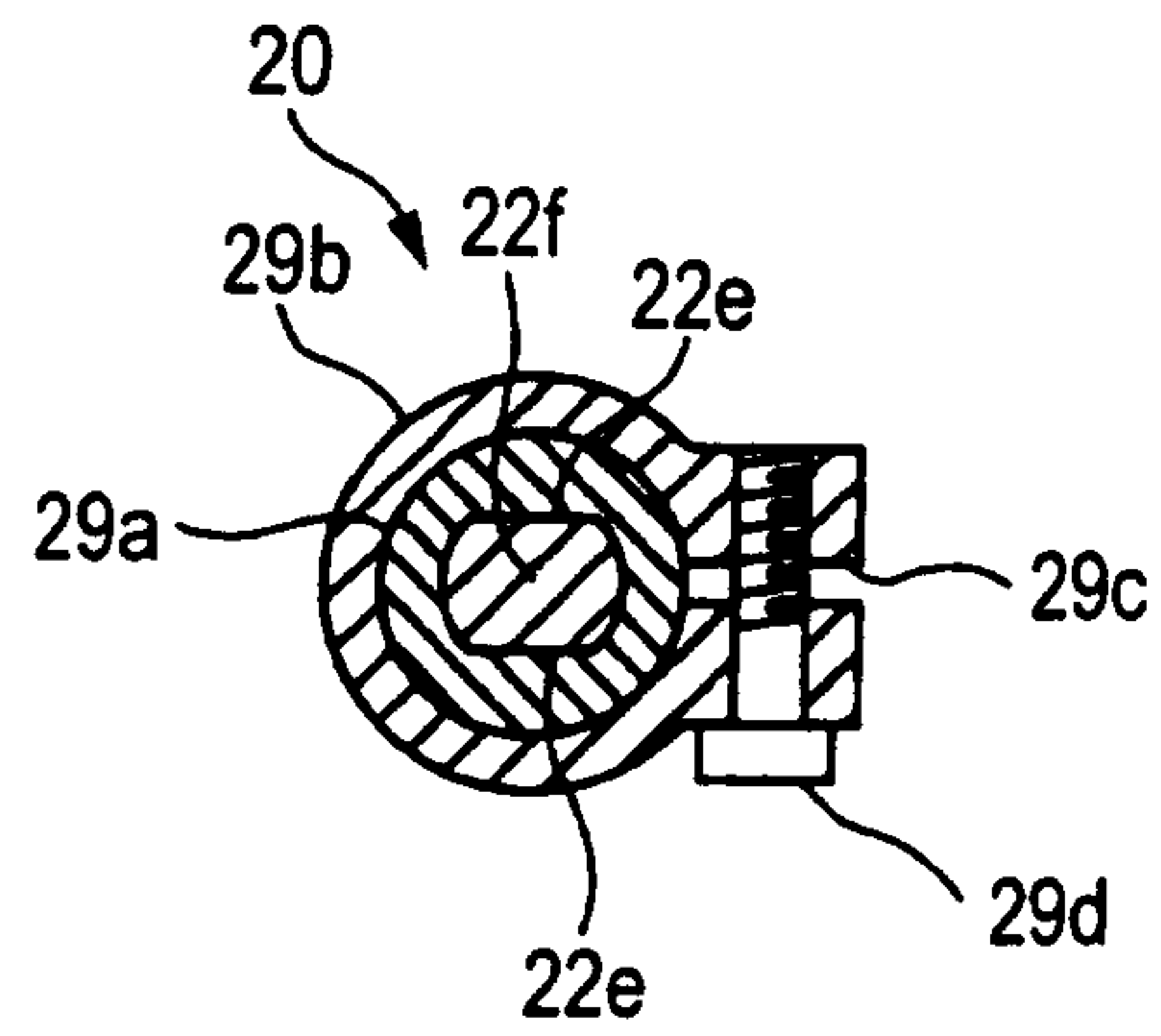
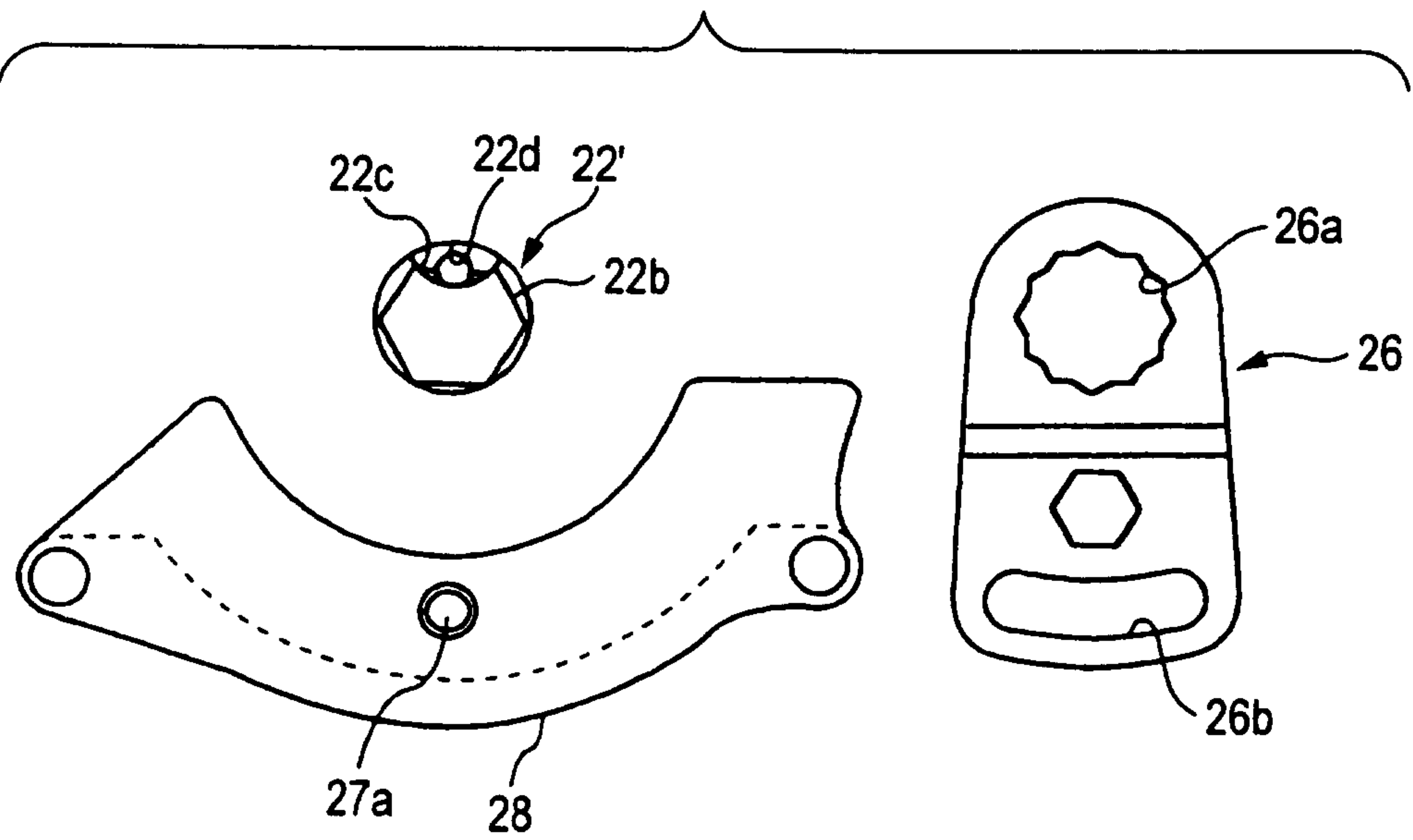
FIG. 10



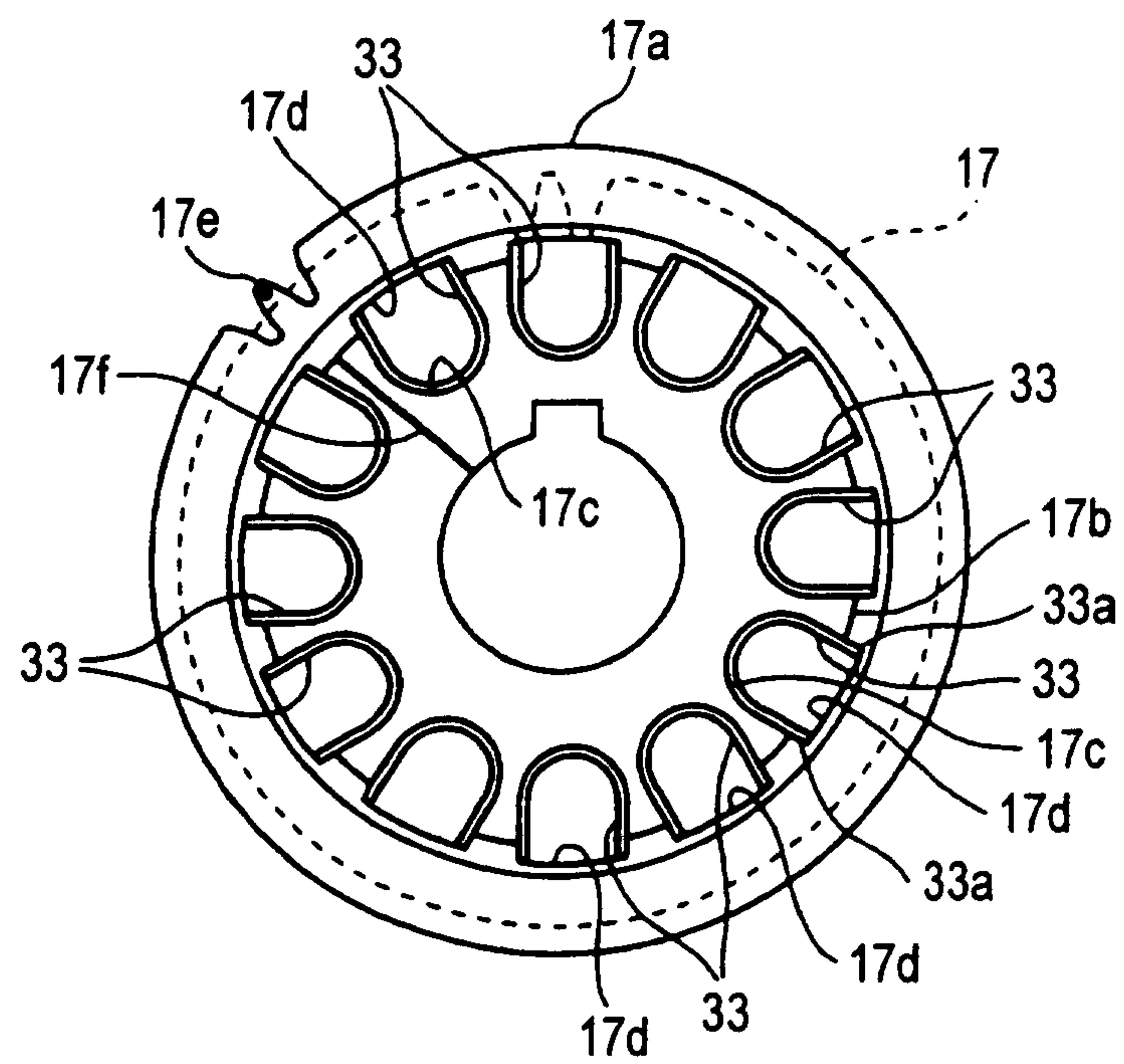
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FIG. 11LEFT-HAND SIDE VIEW

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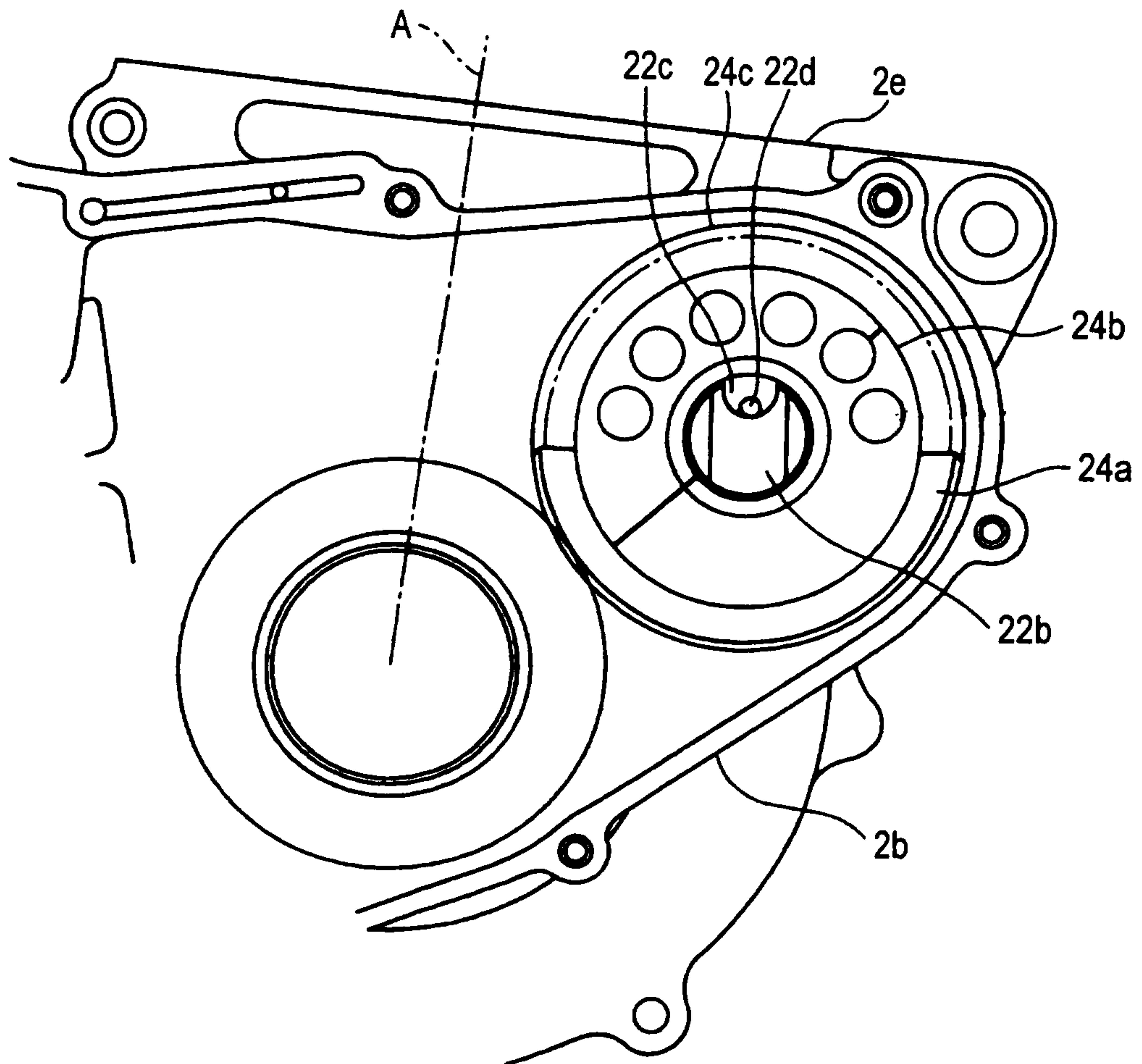
FIG. 12**FIG. 13**

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FIG. 14LEFT-HAND SIDE VIEW

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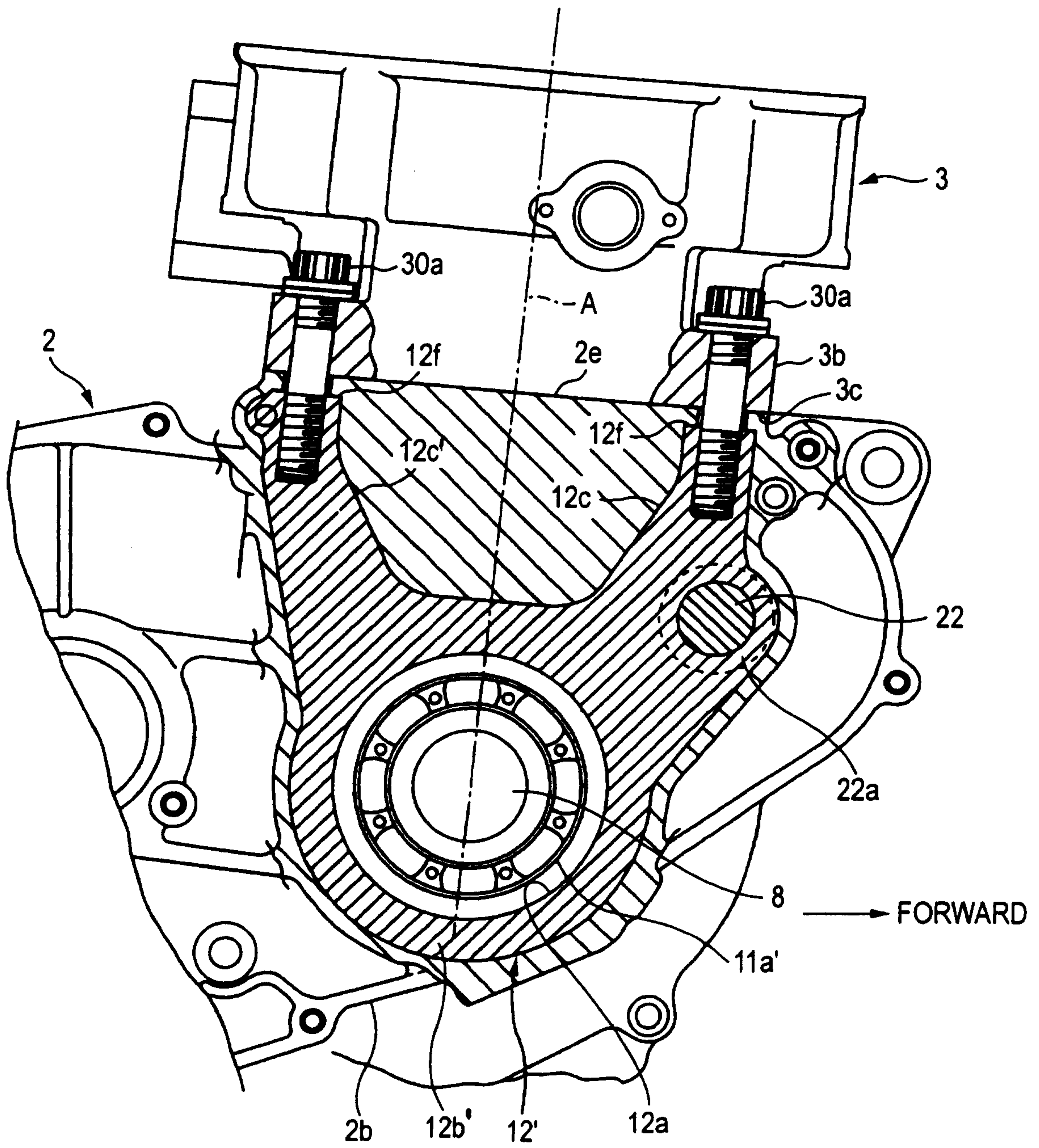
FIG. 15



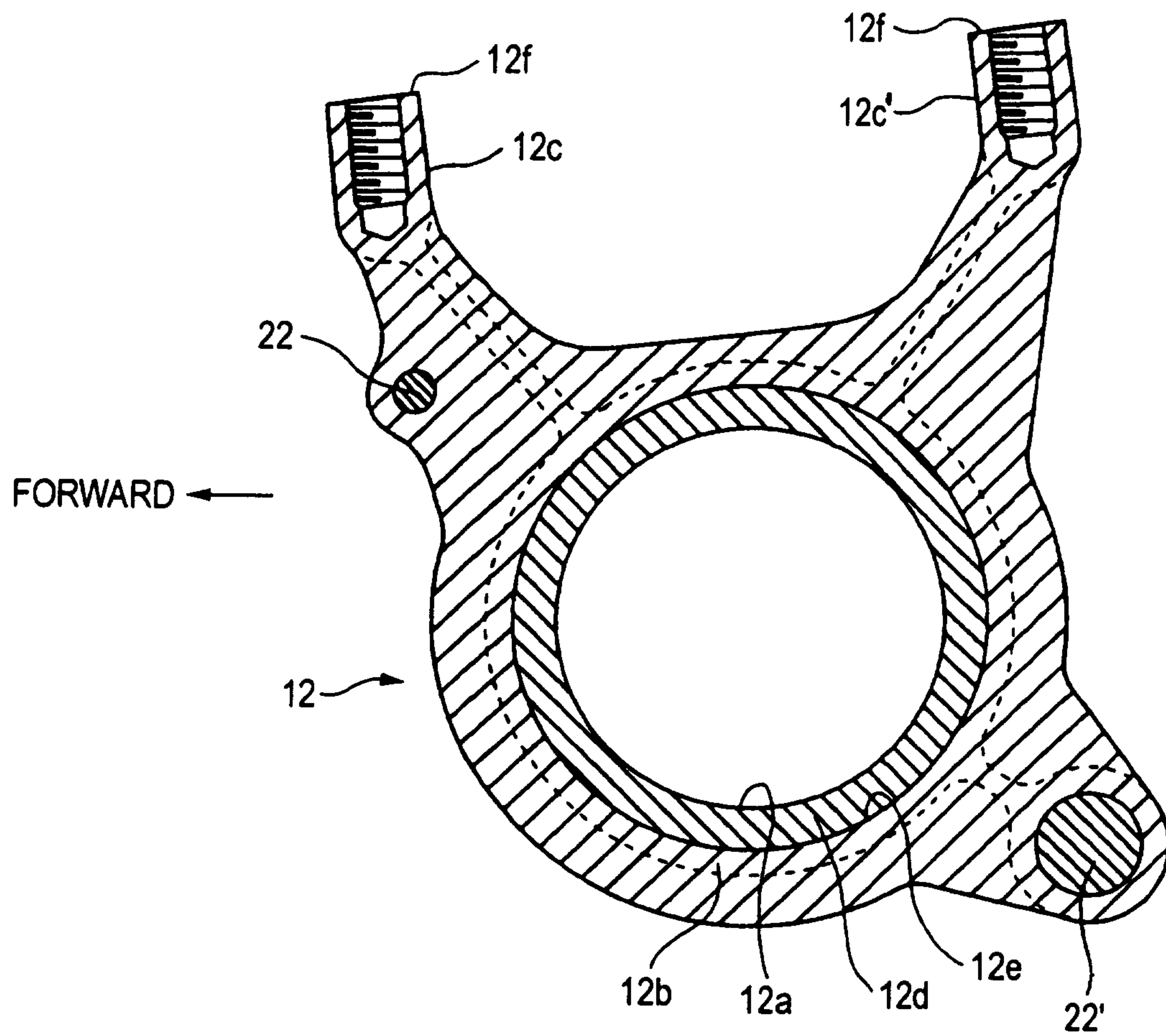
RIGHT-HAND SIDE VIEW

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FIG. 16

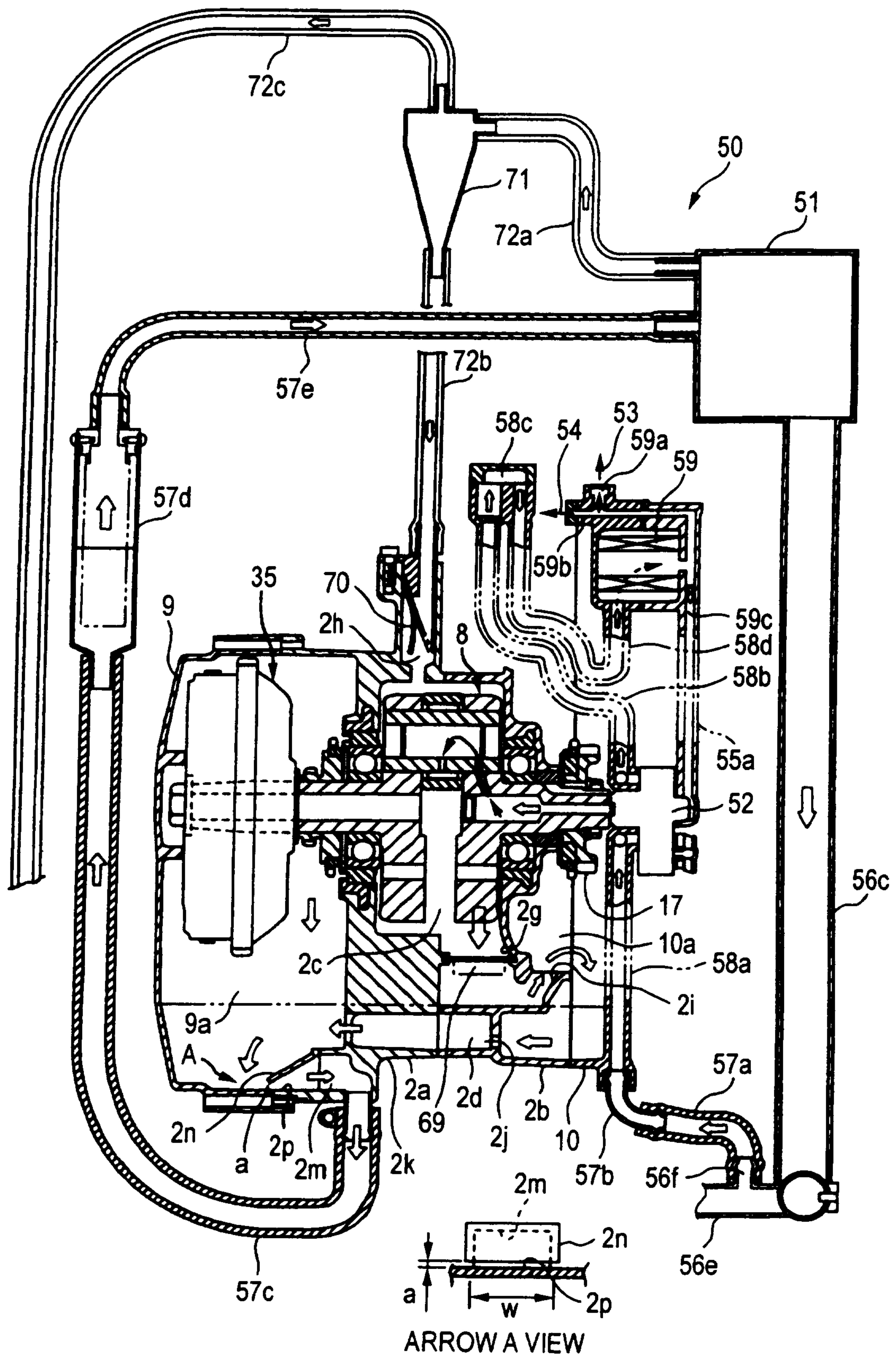
RIGHT-HAND SIDE VIEW

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FIG. 17LEFT-HAND SIDE VIEW

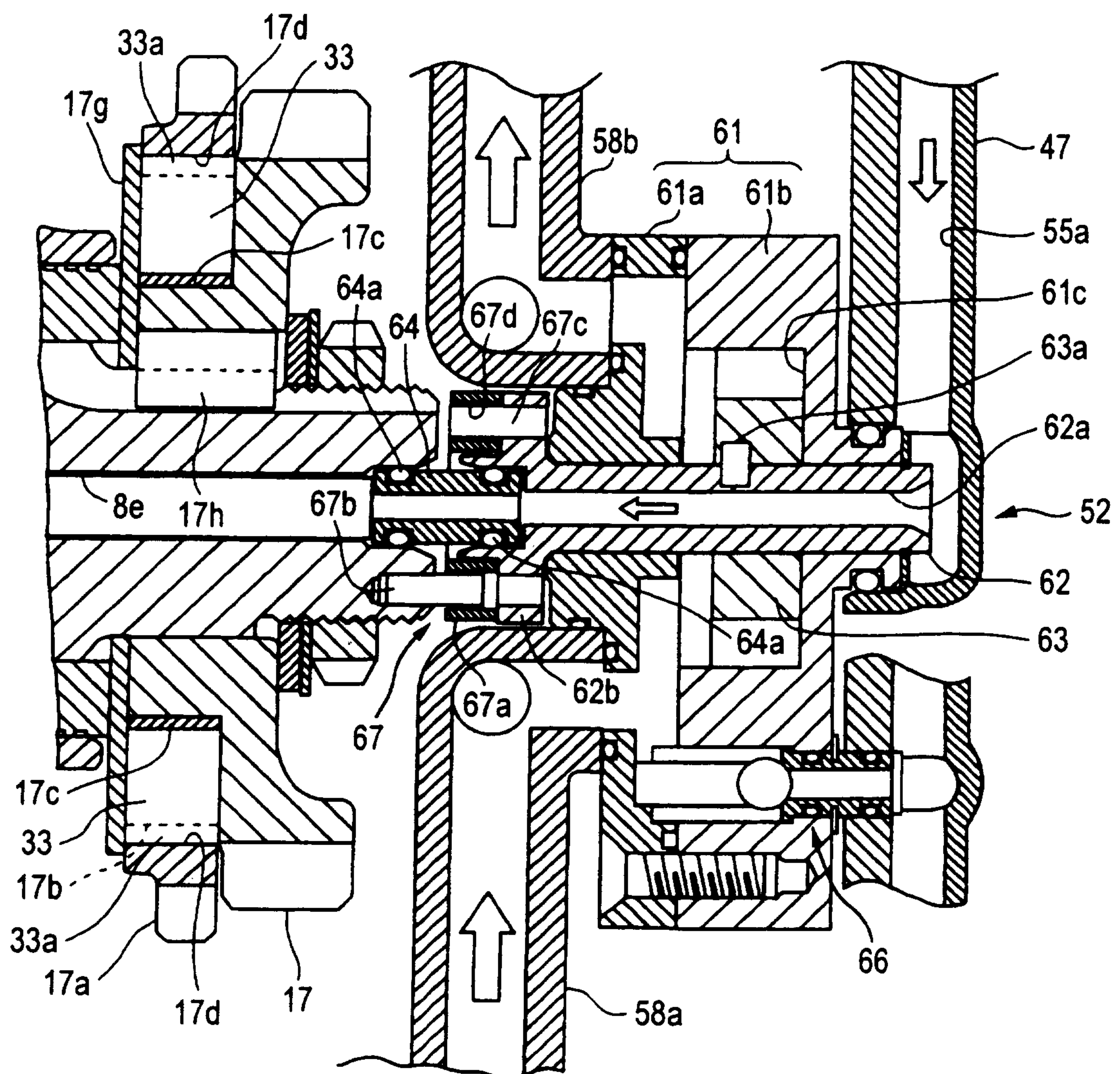
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FIG. 18



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FIG. 20



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FIG. 21

