

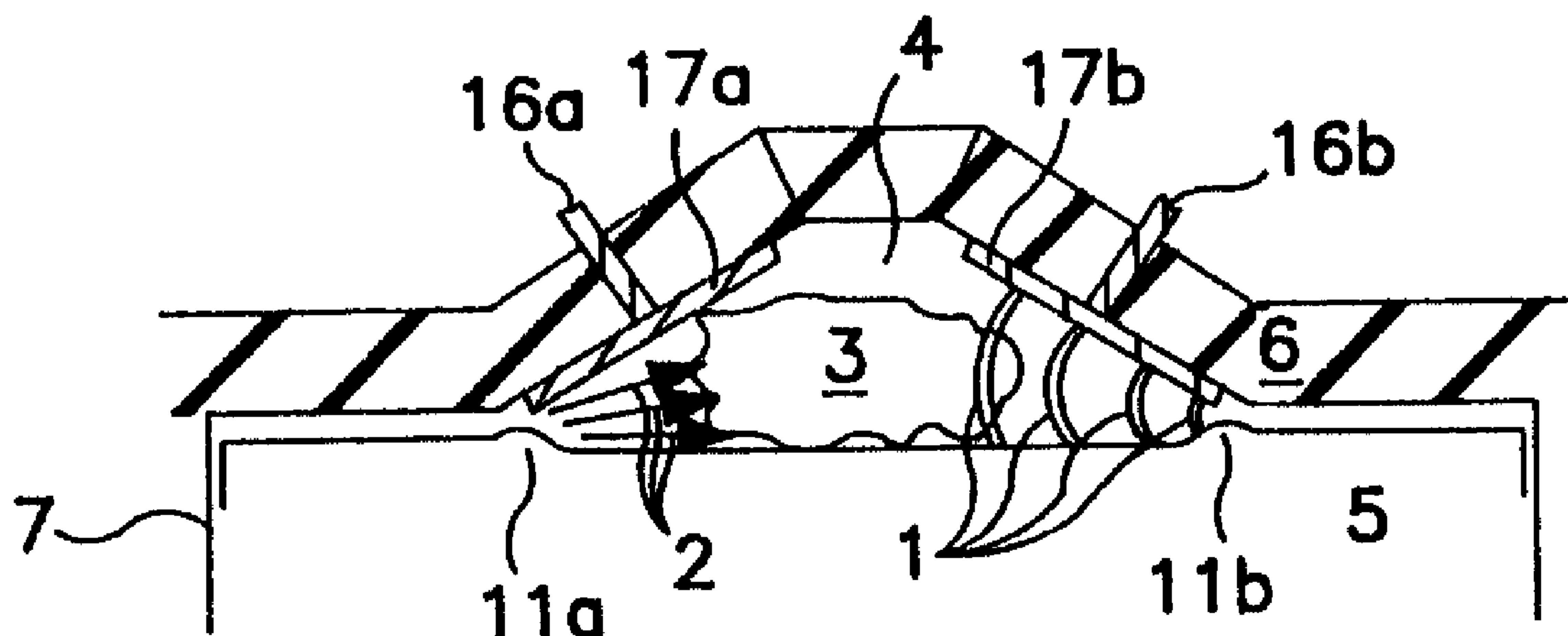


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(54) Title: FLOW COUPLED ARC DISCHARGE IGNITION IN AN IC ENGINE



(57) Abrégé/Abstract:

An improved ignition-combustion system for internal combustion engines comprising a compact combustion chamber zone (4) mainly in the engine cylinder head (6) mainly under the exhaust valve (8) and with large air-squish zones (12a, 124b) formed at the edge of the combustion zone which produce colliding squish flows (2, 2a, 3a, 3b) with high turbulence (3c) at the center of the combustion zone, with one or two spark plugs (12a/118, 12b/18a) located at the edge of the combustion zone within the high squish zones, resulting in a combined ignition and combustion system of colliding-flow-coupled-spark discharge (CFCSD), with the ignition employing high energy flow-resistant ignition sparks which move under the influence of the squish flow towards the central turbulence region as the piston nears engine top center, to produce rapid and complete burning of lean and high EGR mixtures for best engine efficiency and lowest emissions.



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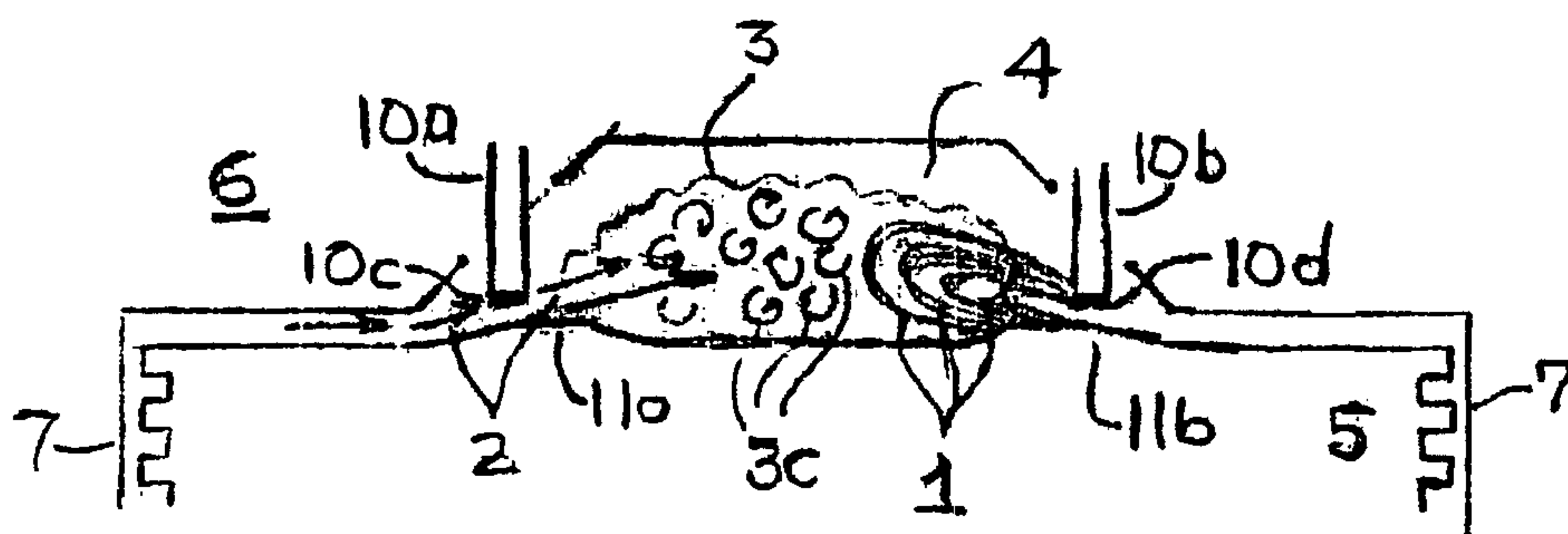
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(54) Title: FLOW COUPLED ARC DISCHARGE IGNITION IN AN IC ENGINE



(57) Abstract: An improved ignition-combustion system for internal combustion engines comprising a compact combustion chamber zone (4) mainly in the engine cylinder head (6) mainly under the exhaust valve (8) and with large air-squish zones (124a, 124b) formed at the edge of the combustion zone which produce colliding squish flows (2, 2a, 3a, 3b) with high turbulence (3c) at the center of the combustion zone, with one or two spark plugs (12a/118, 12b/18a) located at the edge of the combustion zone within the high squish zones, resulting in a combined ignition and combustion system of colliding-flow-coupled-spark discharge (CFCSD), with the ignition employing high energy flow-resistant ignition sparks which move under the influence of the squish flow towards the central turbulence region as the piston nears engine top center, to produce rapid and complete burning of lean and high EGR mixtures for best engine efficiency and lowest emissions.

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FLOW COUPLED ARC DISCHARGE IGNITION IN AN IC ENGINE

FIELD OF THE INVENTION

This invention relates to spark ignition internal combustion (IC) engines and to the improved ignition and combustion of air-fuel mixtures in IC engines through the faster and more vigorous combustion of the air-fuel mixture brought about by high air-flows of both the directed bulk flow type and of the more random turbulent flow type produced in a highly compact combustion zone of the combustion chamber. In particular, the invention relates to combustion systems which can simultaneously produce high directed bulk flow at the spark plug site at the time of ignition and more random and intense turbulent flow inside a compact combustion chamber during the time of ignition and combustion to speed-up the burn. In particular, the invention relates to a system which involves the design and control of high flows, high energy ignition spark discharges, and resulting intense flame kernels, all three of which interact among themselves in ways that produce a very rapid and controlled burn in a compact combustion zone for low NO_x emissions and high engine efficiency through lean and high EGR (exhaust gas recirculation) combustion at a high engine expansion ratio.

BACKGROUND OF THE INVENTION AND PRIOR ART

High flows are used in IC engines to improve engine efficiency and emissions through rapid combustion of lean and high EGR air-fuel mixtures. These flows are either swirl, tumble (vertical vortex), or squish. The predominant type of flow used is swirl, especially for the purpose of generating mixture stratification at the spark plug site to allow for ignition of lean mixtures. Numerous examples of these exist, especially with Japanese manufacturers such as Toyota, Mazda, and others. To a lesser extent tumble flow is used, as in the Mitsubishi Vertical Vortex (MVV) engine now in production. Squish is rarely used for efficiency and emissions improvement. It is used principally in racing and performance for speeding up the combustion of high speed engines. Moreover, such squish flow is not used in conjunction with the spark except in my U.S. patent No. 5,517,961, where it is used in conjunction with a high energy, flow-resistant spark to help spread the spark and speed up the burn.

SUMMARY OF THE INVENTION

In this patent application is disclosed an ignition and combustion system based on the interaction of two generic types of flows which can be generated in an engine cylinder. One is bulk flow which can be produced at the spark plug site at the time of ignition to interact /couple to the spark discharge to direct and spread the spark, along arc runners if practical; the other is turbulent flow which can be produced by impinging squish flows directed towards each other as colliding flows reinforced by the piston compression of swirl or tumble flow, into which the spark discharges and resulting initial flame kernels can move in the form of flow-coupled-spark-discharges of the type disclosed in my U.S. patent No. 5,517,961. Preferably the more random turbulent flows (colliding flows or piston converted swirl or tumble) occur in a highly confined region or zone of the engine combustion chamber defined by the piston or other movable element nearing top center (TC) and the walls of the combustion volume, made up, for example, of the engine cylinder wall and cylinder head in a conventional piston engine. Preferably, the combustion zone is located partly or mostly under the exhaust valve, or in some other high temperature zone of the combustion chamber, with the bulk flow occurring at one or more edges of the turbulent zone to move and direct one or more high energy flow-resistant spark discharges into the mixture turbulence to produce a rapid and complete combustion of an air-fuel mixture without engine knock or other deleterious effect. In this way, very lean and high EGR mixtures can be burnt rapidly to produce high engine efficiencies and low emissions, especially low NOx emissions whose formation depends on both temperature and time, both of which are reduced.

With the use of one spark plug and two valves, the spark plug is preferably located between the valves disposed preferably along or near a center diameter line of the engine cylinder if bulk flow is present at the spark plug site, e.g. if the combustion chamber is under the exhaust valve; or the spark plug may be placed offset of the valve center line at a high bulk flow region if the combustion chamber includes part or most of the region under both valves, either in the cylinder head or in the piston as has been disclosed in my U.S. Patent No. 5,517,961. If two spark plugs are used, with one or two intake and one exhaust valve, more options are available for achieving spark coupling to the bulk flow. The spark plugs can be disposed on either side of the exhaust valve under which is defined the main combustion chamber and region of turbulent flow, with two inward moving bulk flows (BFs) occurring at the two spark plug sites for two flow-coupled-spark discharges (FCSD), producing a more elongated combustion zone and even more

rapid burning of the air-fuel mixture. Placing the combustion chamber under the exhaust valve improves the engine's lean burn capability and reduces its knock susceptibility, especially if the end-gas region of the combustion chamber is narrow and subject to cooling. These are preferred embodiments of the invention. Furthermore, in these preferred embodiments, the piston is flat, or slightly cupped under the combustion zone, with thermal barrier coatings used to limit heat transfer to the piston.

Near top center (TC), i.e. within 30° of TC, the bulk flow (BF) in the spark gap is produced by the piston which induces squish flows which move towards each other and collide to, in turn, produce an intense turbulence in the combustion zone (a colliding turbulence generating flow, or CTF). Advanced of TC, the bulk flows in the spark gap are produced by swirl or tumble, or by a combination of these which are converted by the piston motion near TC to turbulent flows to produce a very rapid and vigorous burn. When squish flow dominates this process, it can be viewed as a colliding-flow-coupled-spark-discharge (CFCSD), versus simply an FCSD for swirl or tumble. Both are made up of high energy, flow-resistant, flow-coupled-spark-discharges which move into, and/or produce initial flame kernels which move into the high turbulence region (of small eddies), produced by colliding flows or piston compression converted bulk flow which help homogenize the mixture and help spread the flame.

In another preferred embodiment, the inlet air flow is used to reinforce the combustion process. Such an inlet flow can be a vertical vortex flow which reinforces one or more of the squish induced flows, especially at advanced ignition timings when squish is low. Alternatively, the combustion chamber and inlet valve, one or two inlet valves, and intake runners may be designed to produce swirl flow to reinforce the squish flows.

For the ignition, preferably triangular distribution spark with a peak amplitude above 200 milliamps (ma) is used, say 300 to 800 ma, operating in part in the flow-resistant arc discharge mode (versus glow discharge) which can sustain flow velocities of 20 meters/sec (m/s) or greater without spark segmentation, as disclosed in my U.S. Patent No. 5,517,961, to produce a very fast ignition and combustion of the mixture.

In another preferred embodiment, the spark may be fired to the piston given the ability to operate the engine with ignition near top center, between 5° and 30° before top center (BTC), with special electrode tips acting as arc runners which improve coupling of the spark to the flowing mixture. Special design halo-disc spark plugs of my U.S. Patent No. 5,577,471 can be

used, or similar plugs with large thin discs at their ends to define a runner along which the spark can move under the influence of the flow. Preferably, erosion resistant material such as tungsten-nickel-iron is used for the spark plug electrode material.

5 In another preferred embodiment, the valve timing is selected to make best use of CFCSD. This includes setting the intake valve opening preferably near TC (a few degrees before TC) for the cases with no combustion volume under the intake valve. The exhaust valve closing is preferably set at or before TC (BTC), e.g. 0° to 30° BTC to trap the last part of the exhaust for both high internal exhaust gas residual fraction to eliminate or minimize the need for EGR as well as minimize hydrocarbon (HC) emissions since the last part of the burnt mixture to be exhausted
10 contains proportionally more unburnt HC. Having the combustion chamber under the exhaust valve can help the process of trapping the (last) part of the exhaust coming from the ring clearances and other chamber crevices which have the most unburnt HC emissions, to be burnt in the subsequent ignition and combustion firing cycle.

In another preferred embodiment using preferably a high energy, high efficiency, flow
15 resistant ignition spark, an improved combustion chamber of the squish type is used which has the combustion chamber in the cylinder head under both the intake and exhaust valves which are centrally located and made as small as practical to allow for large squish flows, especially in a direction transverse to the line joining the two valves, with one or two spark plugs located in the squish zone. Preferably, the intake valve is recessed so that in its fully open condition it clears the
20 top of the piston when positioned at top center (TC), and likewise for the exhaust valve which is preferably even further recessed. This represents a free-wheeling system which will not damage the engine should the timing belt break (in the preferred overhead cam configuration). In such a design several advantages result, including thorough mixing of the air-fuel mixture due to the squish induced colliding flows, rapid burning of the air-fuel mixture, especially of high dilution,
25 high exhaust gas recirculation (EGR) and high air-fuel ratio (AFR) mixtures which produce high efficiency and low emissions, and high internally retained exhaust gas, especially that located at the further regions of the combustion chamber, e.g. in the ring crevices where most of the unburnt hydrocarbons are formed which can be re-burned in the subsequent ignition firing cycle. Preferably, a squish land exists around the entire periphery of the piston so that the end gas can
30 be cooled and therefore allow for higher compression ratio operation.

More ideally, the disclosed ignition/combustion/flow systems are used in a Miller Cycle engine or variable-valve-timing engine with high expansion ratios of 10 to 12 to one or higher and delayed intake valve closing (for effective lower compression ratio). If practical, the systems disclosed are used in the virtual 3-stroke engine disclosed in my U.S. patent SN 5,454,352 where in FIG. 7 of that patent is shown a preferred embodiment of the virtual 3-stroke engine in the form of an opposed piston with two valves and two spark plugs. Preferably, the valves are smaller which is more practical with the use of a turbocharger to create a more compact combustion chamber in the cylinder head and higher squish at the spark plug sites. The low compressive friction makes this engine easy to start and ideal for the forthcoming dual rail 14/42 volt battery systems with integrated starter/generator (ISG). It also makes it very efficient and of very low emissions. NOx emissions are low due to the low adiabatic heating of the intake air on compression and low overall temperatures due to the long expansion stroke (preferably of 12 to one or higher expansion ratio). Because of the inherently high efficiency (BSFC, or brake specific fuel consumption), especially at part load, combustion can be somewhat delayed, as in the diesel engine, to limit peak combustion temperatures below the level where significant NOx emissions are formed. This engine can be operated either premixed, i.e. indirect injection or carbureted, or direct in-cylinder injection, either moderate pressure injection as in the GDI engine, or higher pressure injection as in the diesel engine.

Other improvements of the ignition-engine system include improved forms of dual ignition, especially of the low inductance type, high energy, high efficiency inductive ignition, which use capacitance type plugs, if practical, with erosion resistance electrode materials. Also, such low inductance type coils can be improved by having magnetically uncoupled secondary high voltage windings to aid in reducing an over-voltage coil end effect associated with low resistance high efficiency coil windings.

Other means for producing CFCSD are possible, including in other types of engines, such as two-stroke, rotary, gas direct injection, and others, the main inventive principle disclosed herein being the combination and interaction of flow-coupled-spark-discharge, especially of high-energy high-efficiency flow-resistant sparks with turbulence generating flows, especially in a compact, hot region of the combustion chamber of a spark ignition engine.

BRIEF DESCRIPTION OF THE DRAWINGS

FIGS. 1a and 1b are drawings of air-flow vectors and flow interactions in a compact engine combustion chamber (combustion zone) that characterize a key feature of the invention.

5 FIGS. 2a and 2b are approximately to-scale drawings of the side and top view of a combustion chamber with a preferred embodiment of the invention with central combustion zone under a central exhaust valve, and two intake valves and spark plugs located in the squish zones.

FIG. 3 is a top view similar to that of FIG. 2b but with one intake valve.

FIGS. 4a to 4c show various arc runners for the spark to move along under the action of bulk flow, such as squish flow.

10 FIG. 5 is an approximately to-scale side view drawing through a line intersecting the two spark plugs of FIGS. 2b and 3 with arc runner construction.

FIGS. 6a and 6b are two types of spark plug firing ends for ignition firing to the piston.

15 FIG. 7 is a more idealized embodiment of FIG. 5 with special spark plugs located on an electrically insulating cylinder head, achievable in a two stroke engine or other engine where valves are absent from the cylinder head or highly confined in the head.

FIG. 8 is a preferred embodiment of a direct injection version of the invention with the fuel injector located at the edge of a combustion zone under the exhaust valve across from a single spark plug in the squish zone.

FIG. 8a is a possible side view of the embodiment of FIG. 8 during the intake process.

20 FIG. 8b is a view of FIG. 8 without the fuel injector, with the piston near top center.

FIGS. 9a and 9b is a top and side view drawing of a combustion chamber with two intake valves and one exhaust valve with the combustion zone under the exhaust valve and a single spark plug located at a more central point at the edge of the combustion zone.

25 FIGS. 10a and 10b is a more conventional pent-roof chamber with combustion zone both in the head and piston under the single intake and exhaust valves and two spark plugs at the edge of the combustion zone in the squish zones,

30 FIGS. 11, 11a, 11b, 11c and 11d are partial side views and a top view of a combustion chamber with one small intake and exhaust valve centrally located and with the compact combustion zone under the two valves, and with two spark plugs located at two opposite sides of the combustion zone in regions of high squish flow, and a central fuel injector shown as an option in FIGS. 11c and 11d.

FIGS. 12a, 12b, 12c, and 12d are approximately to-scale partial top and side views of a combustion chamber of FIGS. 11, 11a and 11b detailing and emphasizing certain aspects of the combustion chamber for improved operation.

5 FIG. 13 is a cam timing diagram showing a preferred valve timing for the present invention.

FIG. 14 is a design of the ignition coil output structure for providing dual ignition firing to the piston of the present invention in which the return path for the electrical current is entirely through the spark plugs and not through the engine block.

10 FIG. 15a is a partial, not to scale, top view layout of two side-by-side engine cylinders in a multi-cylinder engine, with FIG. 15b depicting a preferred approximately to-scale combustion chamber with dual ignition of the present invention in which the valves are lined up approximately in the length axis of the engine.

15 FIG. 16a is a partial, not to scale, top view layout of two side-by-side engine cylinders in a multi-cylinder engine, with FIG. 16b depicting a preferred approximately to-scale combustion chamber with dual ignition of the present invention in which the valves are lined up transverse to the length axis of the engine.

FIGS. 17a, 17b and 17c are partial top views of low inductance, high energy ignition coil structures with open magnetic paths suitable for application where greater than 12 volts power is available and designed for having two high voltage towers suitable for dual ignition.

20 FIG. 18 is a top partial view of an open E-type low inductance, high energy ignition coil with open magnetic path with some lower turns loosely coupled secondary windings constituting a series of secondary circuit external leakage inductors, usable also in the designs of FIGS. 17a and 17b, for reducing the end effect voltage overshoot. FIG. 18a is an equivalent circuit of the coil of FIG. 18 used in an inductive ignition circuit.

DESCRIPTION OF PREFERRED EMBODIMENTS

FIGS. 1a and 1b are drawings depicting, in schematic form, flow vectors and flow interactions that lead to the principles of this invention. Region 1 denotes the region of the ignition source which can be the gap between spark plug electrodes and the region adjacent to the electrodes into which the spark discharge moves by the action of the air-flow indicated by flow vectors 2, representing bulk flow, BF, to produce what has been described as a flow-coupled-spark-discharge, FCSD. Region 3, shown with small partially circular curves with arrows 3c, indicating tiny turbulent eddies, is the region of high turbulence generated by the approximately equal and oppositely directed colliding flows 3a and 3b, producing what has been designated as CT. The spark discharge region 1 preferably penetrates the turbulence region 3 at approximately right angles to the direction of the colliding flow vectors 3a and 3b, as shown. A second ignition source region can be equally well placed at the right side of the turbulence region 3, as is indicated with reference to FIGS. 2a, 2b, and other figures.

FIGS. 2a and 2b depict in partially schematic, approximately to-scale drawings, side and top views of a combustion zone 4 of a conventional piston engine contained within a larger combustion chamber volume defined by a piston 5, cylinder head 6, and cylinder sleeve 7. The term "combustion zone" 4 shall be used to indicate the region where the major and first part of the combustion, or burning, occurs as distinguished from "combustion chamber" which indicates the entire volume as already defined. In this preferred embodiment, the combustion zone 4 is shown located in a central part of the combustion chamber under an approximately centrally located exhaust valve 8 with exhaust tube 8a, with two intake valves 9a, 9b shown at one side of the combustion chamber with intake tubes 9c and 9d (allowing for the more central exhaust valve 8 and combustion zone 4 under the exhaust valve 8). In this preferred embodiment is shown two high voltage spark plug electrodes 10a and 10b disposed symmetrically at opposite ends of the elongated combustion zone 4 with ground electrodes 11a and 11b defined by small rises on the piston for producing spark discharges between the tips 10c and 10d of electrodes 10a and 10b and the piston surface regions 11a and 11b, i.e. ignition firing to the piston versus to conventional "J" ground electrodes. Spark firing tips 10c, 10d, 11a, and 11b are made of erosion resistant material.

For ease of visualization in FIG. 2a are shown, around the gap of the left-most electrodes 10c/11a, bulk flow vectors 2, and around the right-most electrodes 10d/11b spark discharges 1 being directed by the bulk flow vectors into the turbulence region 3, understanding that both the bulk flow (2 and 2a of FIG. 2b) and spark discharges (1 and 1a of FIG. 2b) are present around

both spark gaps. In this preferred embodiment, a central combustion zone is provided with high bulk flow at the plug sites and high turbulence inside the combustion zone. When used with a high energy ignition system with flow resistant sparks (direct current arc discharge mode) of long duration, i.e. of order of magnitude one millisecond, one can achieve a very rapid burn of a dilute
5 air-fuel mixture, diluted with excess air (lean mixture) and/or with retained exhaust gas (high exhaust residual or EGR), for very high efficiency and low emissions. Preferably, a high and optimum compression ratio (CR) is used, e.g. between 10:1 CR and 15:1 CR, for best efficiency, which depends on several factors including fuel type used and heat transfer to the combustion chamber surfaces which increases with compression ratio (and can be reduced with thermal barrier
10 coatings). More ideally, a Miller type cycle is used with late intake valve closing of 60° after bottom center (ABC) or greater, and a high expansion ratio (ER) of 12 to one or higher.

FIG. 3 is an approximately to-scale, partially schematic top view of a spark ignition piston engine representing a preferred embodiment of CFCSD of the type of FIGS. 2a and 2b except that it has one intake valve 9 instead of two. Like numerals represent like parts with respect to the
15 earlier figures. In this preferred embodiment, the spark plugs 12a and 12b are shown (instead of the electrodes), understanding that the spark plugs can be located at an angle to the vertical as is depicted in FIG. 5, versus vertical as shown in FIGS. 2a, 2b. This design is especially useful for use with a centrally located single overhead cam 13, as depicted, with the two spark plugs located with an angle to the vertical to help clear the cam as well as to better position their tips in the
20 combustion chamber. Another embodiment is to have two smaller intake valves, one at the present location and the other at the opposite side, 180 degrees away.

For better coupling of the spark discharge to the flow, an electrode geometry that acts in part as an arc runner may be practical. FIGS. 4a to 4c show a pair of electrodes 10 and 11 with firing tips 13 and 14, the electrodes subtending progressively smaller included angles from FIG.
25 4a to 4c. Flow vectors 2 are shown emanating from the spark gaps 15 as are spark discharges 1 which move along the runners 10/11. It can be appreciated that the smaller the included angle between the electrodes, say from 30° to 90° , the more directed is the flow and the less stretching there is of the spark discharge as it moves along the runners 10/11 per unit motion, making FIG.
30 4c the more preferred embodiment for electrode geometry which can produce greater motion of the spark discharges 1 and hence greater penetration of the spark into the turbulence region 3. FIG. 5 shows an embodiment of the spark plug electrodes of FIGS. 2b and 3 to produce better spark penetration and size.

FIG. 5 is a partially schematic, approximately to-scale side view drawing through a line intersecting the two spark plugs of FIGS. 2b and 3, representing a preferred embodiment of CFCSD. Like numerals represent like parts with respect to the earlier figures. In this embodiment are shown spark plugs 12a and 12b, preferably 18 mm, with large diameter, thin, erosion resistant electrodes 10c, 10d disposed at right angles to the plug axis forming a small included angle (approximately 45°) with the piston top (shown close to TC) to produce a preferred arc runner of FIG. 4c (versus FIG. 2a which represents arc runners of FIG. 4b). As in FIG. 2a, the flow vectors 2 are shown on the left and the spark discharges 1 on the right for ease of visualization, both moving into the turbulence region 3. The spark gaps 15 are made with small rises 11a and 11b on the piston surface, which represents the second arc runner combination 11 of FIGS. 4a to 4c. Systems involving firing to the piston have been disclosed in several of my prior patents, including U.S. patents Nos. 4,774,914 and 4,841,925 among others. Also, many electrode types have been disclosed, two of several possible types being depicted in FIGS. 6a and 6b.

FIGS. 6a and 6b are approximately to-scale, partial side view drawings of two spark plug ends firing to the piston 5. Like numerals represent like parts with respect to the earlier figures. In FIG. 6a the center conductor 16 is preferably copper extending axially into the combustion chamber near the piston surface at the TC point with a coating 10 (representing the arc runner) of erosion resistant material, such as tungsten-nickel-iron. The runner geometry conforms to that of FIG. 4b (90° included angle). In FIG. 6b the arc runner is erosion resistant disc 17 at right angles to the plug axis shown located slightly away from the end surface of insulator 18 (versus on the surface of the insulator, as in FIG. 5). These piston firing embodiments produce better spark penetration towards the center of the combustion zone.

FIG. 7 is a more idealized embodiment of FIG. 5 which can be attained in a two-stroke engine where valves are absent from the cylinder head or rotary type valves are used (with a four stroke engine also). The engine cylinder is a partially schematic, approximately to-scale side-view drawing representing another preferred embodiment of CFCSD. Like numerals represent like parts with respect to the earlier figures. In this preferred embodiment, the cylinder head is shown made of insulating thermal barrier material (as in an adiabatic engine), with a central triangular cross-section combustion zone 4, on the two angled surfaces of which are placed large, flat, thin electrodes 17a, 17b connected to high voltage terminal 16a and 16b respectively. In this embodiment, extremely effective and large arc runners between 17a and 17b and the piston surface are achievable for extensive penetration of the arc discharge into the turbulence region 3 of the

combustion zone 4. However, to accomplish this a high energy ignition with long duration flow-resistant sparks of the types I have disclosed elsewhere is preferred. Also, special treatment of the piston surface is required, e.g. such as coating with erosion resistant material which can also be a relatively poor thermal conductor.

5 In many applications arc runners are not practical, so more conventional spark gaps with proper oriented electrodes may be used (that do not interfere with the flow), or circularly symmetric spark gaps may be used. In such a case, the high energy-flow coupled-spark-discharge (FCSD) acts as a stationary extended spark (a plume) through which the mixture flows under the action of bulk flow to produce flame kernels which move and grow as they enter the turbulence region 3 of the
10 combustion zone 4. FIG.8 depicts such an embodiment.

In FIG. 8 is shown a partially schematic, approximately to-scale top view of a piston IC engine representing a preferred embodiment of CFCSD with a single spark plug 12 and single intake valve 9, with the valves and spark plug placed collinear in an approximately central cylinder
15 7 diameter line with the spark plug in the center between the valves. Like numerals represent like parts with respect to the earlier figures. In this embodiment, the combustion zone 4 is located mainly under the exhaust valve with small piston clearance for the intake valve to allow for squish bulk flow 2 at the spark plug site. The use of a rotary intake valve would obviate any concern associated with the preferred small clearances between the valve and piston.

Shown also in the figure is a fuel injector nozzle 18 preferentially located across from the
20 spark plug, representing a gas-direct-injection (GDI) version of the present invention allowing for the use of various fuels. The injector injects the fuel spray 19 into the turbulence region where it becomes rapidly mixed and homogenized for clean burning and prevention of spark plug fouling (a problem with spark ignition diesels).

A partially schematic, side view, approximately to-scale drawing of a cylinder of the
25 engine of FIG. 8 with the piston near the end of the intake stroke is shown in FIG. 8a. The intake air 20 is shown entering in a vertical circular form representative of tumble flow which can reinforce the flow 2 at the spark gap 21 of FIG. 8b, which represents the engine of FIG. 8a with the piston at the end of the compression stroke near TC. Like numerals for both FIGS. 8a and 8b represent like parts with respect to the earlier figures. In FIG. 8a is shown the alternative possible
30 fuel nozzle 18 (not shown in FIG. 8b representing the more conventional engine).

FIG. 9a shows a partially schematic, approximately to-scale top view of a piston IC engine representing a preferred embodiment of CFCSD with a single spark plug 12 and two intake valves

9a and 9b, with the exhaust valve 8 and spark plug 12 placed collinear in an approximately central cylinder 7 diameter line. Like numerals represent like parts with respect to the earlier figures. The shaded portion shown indicates the sloping region of the combustion zone from its peak just under the exhaust valve 8.

5 FIG. 9b is a partially schematic side-view drawing of the engine cylinder of FIG. 9a, with like numerals representing like parts with respect to the earlier figures. As in most of the drawings, the piston is shown near TC in the compression ignition firing stage. In this embodiment, the combustion chamber zone 4 is under the exhaust valve 8 and the spark gap is in a high squish zone as required.

10 FIG. 10a and 10b indicates the combustion chamber of a more conventional type pent-roof engine with the combustion zone shared between the cylinder head 6 and the piston 5, but mostly in the piston. Two spark plugs 12a and 12b are shown as one of several options (one spark plug being the more conventional option). The spark plug tips may be more conventional or similar to those of FIGS. 5 and 6b which can act as arc runners as already discussed. Like numerals represent
15 like parts with respect to the earlier figures.

 An interesting and relevant variation of the combustion chamber of FIGS. 10a and 10b is that of Jim Feuling most recently reported in the February 1999 issue of Hot Rod magazine in an article entitled "Lean, Mean, and Clean, Part One". The combustion chamber is a Figure 8 combustion chamber in the cylinder head under the intake and exhaust valves with the spark plug
20 in the middle. This design creates high turbulence in the combustion chamber except that the volume is larger (combustion zone is under both valves). Moreover, since the spark plug is in the middle and not at the edges of the combustion zone, as in FIGS. 10a and 10b, and does not use a high energy, flow resistant spark, it is not subject to bulk flow, as is disclosed in the present invention, to spread the spark discharge and create a rapid initial burn.

25 In the preferred embodiments of this invention, the ignition and combustion stage is so fast that ignition can occur near TC, even for dilute mixtures (which permits piston firing as one alternative for generating and locating the spark, although it is not required). Preferably, ignition occurs as close to 10° BTC as practical since the inward directed squish flow (bulk flow) is at its maximum near that ignition timing. Under such conditions, the very powerful and rapid ignition
30 and burn means that highly dilute mixtures can be burnt effectively for low exhaust emissions and high fuel economy. Preferably mixture dilution is produced internally (versus EGR) by valve timing, i.e. by advancing the exhaust valve closure, although it may be supplemented with EGR

and with excess air (lean mixtures). The very fast burn of dilute mixtures results in very low NO_x emissions due to both the faster burn (less time for NO_x to form) and lower peak combustion temperatures, while the higher trapped exhaust results in lower HC emissions. Also the bulk flow and colliding flow turbulence generation produces very good air-fuel mixing for a more homogeneous mixture for lower HC emissions and higher engine efficiency. Rapid burn also allows for higher compression ratio and higher engine efficiency.

FIGS. 11, 11a, 11b, 11c and 11d depict various partial views of an engine combustion chamber in which the compact combustion zone 110 is contained in the cylinder head under both the intake and exhaust valve of preferably small size and with two spark plugs in the combustion chamber located in the region of high flow. FIG. 11 depicts a top part of a spark plug 118 with a preferred capacitive spark plug boot 117, which may also be built into the spark plug to provide a typical capacitance of 30 to 60 picofarads.

The combustion chamber zone 110 disclosed has preferably intake valve 121a and exhaust valve 121b disposed parallel to the flat top of the piston 119b and are made as small as practical to provide squish zones 122a and 122b at the far edges as shown. The valves are located in line as close as is practical to each other, as shown in the top view of FIG. 11a, where like numerals represent like parts with respect to FIG. 11. The combustion chamber zone 110 is in the head in an elongated region defined by a volume under each valve and open in the central region 123 between the two valves. Preferably, the larger volume is under the exhaust valve 121b, which is more recessed than the intake valve 121a, which is preferably recessed to just to miss the piston at its TC position when the intake valve is fully open (which can occur if the timing belt breaks for an overhead cam engine design).

With further reference to FIG. 11a, the design shown provides large squish areas 124a and 124b, as is also shown in FIGS. 11b, 11c, 11d which are side-views disposed 90° from that of FIG. 11. In FIGS. 11b, 11c, 11d like numerals represent like parts with respect to FIG. 11. In both FIGS. 11a, 11b, 11c, 11d are shown the possibility for inclusion of two, instead of just one, symmetrically placed spark plugs 118 and 118a, making preferably about 30° with the vertical and placed at the inner edge of the large squish zones 124a and 124b so that the spark gaps 125a and 125b are subjected to high squish flow near TC during ignition. In a preferred design, the squish lands 124a and 124b have a transverse length "l" approximately equal to or greater than the transverse radial dimension "r" of the compact combustion chamber, as is shown to produce

high squish at the spark plug site and high colliding flows in the compact combustion chamber zone 110. This helps vaporize, by the radial intense flow prior to TC, any fuel that may have collected on the piston lands, and produces rapid mixing of the air-fuel mixture in the combustion chamber zone 110 due to the turbulent colliding flows, even at low speeds where combustion is normally poor. The piston to head clearance in the land areas 122a, 122b, 124a, 124b is typically 0.04" to 0.1" to create adequate squish without undue heat losses for a typical engine with a bore and stroke in the 3" to 4" range. The clearance is scaled appropriately for larger engines. The engine cylinder views of FIGS. 11, 11a, 11b, 11c are approximately 2/3 of full scale of a typical IC engine cylinder. In FIGS. 11c and 11d, where FIG. 11d is an exploded view of the central part of FIG. 11c, are shown a centrally located fuel injector 130, operable at moderate pressure as in a GDI engine, or at high pressure as in a diesel engine, located between the two valves and spark plugs with central fuel spray cone 131 covering the central combustion region 123. By spraying the fuel in the central region 123, it does not directly impinge on the spark plug electrodes but locates close to them for a better mixed state for ignition. In FIG. 11d, the squish flow vectors 2 are shown on the left side at the tip of spark plug 118 (representing the piston near TC) which help prevent fuel from settling on the spark plug end and clean the end; the fuel spray is shown on the right side (representing the piston position prior to ignition), being directed by the slight cup 133 in the piston up and to the right as a curved spray 132 moving towards the spark gap 125b to be ignited. This combination of flow, fuel spray, and ignition location and intensity (high energy), make for a practical system.

An advantage of the high energy ignition as used in this engine design is that it can sustain flows as high as 20 meters/second (m/s) without spark break-up and can deliver about 100 mJ of spark energy, which allows for very rapid ignition and rapid early flame propagation (which occurs in large part due to the high speed squish and turbulent colliding flows). This allows for ignition timing near TC even for lean and high EGR (or high internal exhaust residual) mixtures, which has several advantages including lower burn temperature and less time available for NO_x formation, to minimize NO_x emissions. Also, the more recessed exhaust valve 121b allows for more of the partially burnt gas formed at the piston ring regions 126 to be trapped in the combustion volume under the valves than to be exhausted because of the long travel path, allowing this partially burnt mixture to be reburned which minimizes hydrocarbon (HC) emissions. Also, the combination of rapid burn, the larger combustion volume under the exhaust

valve 121b, and the large squish zones where the end gas (last part of the mixture to burn) is located and subjected to cooling by the reverse squish flow after TC, means that higher compression ratios can be used without engine knock for higher efficiency. Moreover, the flat or slightly cupped piston can be more readily coated with thermal barrier coatings (TBC) to minimize heat transfer, reduce HC, and increase engine efficiency.

FIGS. 12a, 12b, 12c, and 12d are approximately to-scale partial top and side views of a combustion chamber of FIGS. 11, 11a and 11b detailing and emphasizing certain aspects of the combustion chamber for improved operation. Like numerals represent like parts with respect to the earlier figures.

FIG. 12a is a partial top view of the combustion chamber similar to FIG. 11a. However, in this case the combustion zone 110 wraps tightly around the outer edges of the valves 121a and 121b to have a tight fit with the edge of the combustion zone, defining the tight fit regions 141a and 141b, also shown in partial side view FIGS. 12c and 12d. This allows for larger squish zones 122a and 122b on the smaller squish lands which are important in this case not for flow-coupling (produced by the large squish lands 124a and 124b) but for cooling of the end gas to permit higher compression ratio operation of 10 to 12 to one compression ratio without knock. The spark plugs 118 and 118a are shown located more towards the larger combustion volume 110a under the exhaust valve versus in-line with the central region 123 in between the two valves, for shorter flame travel and faster burn.

FIGS. 12b and 12c are partial side views of the combustion chamber 12a, with FIG. 12b showing conventional type spark plugs 118 and 118a with squish bulk flow (vectors 2) induced by the piston 119b near TC. The side view of FIG. 12c shows views of the two combustion volumes 110a and 110b, approximately scaled to each other, under the exhaust and intake valves respectively. The piston shows an optional combustion zone region 142 conforming to the combustion zone in the cylinder head which would result in smaller volumes in the cylinder head.

FIG. 12d is a partial view of FIG. 12c showing the intake valve 121a at its approximately fully open position with intake flow vectors 143a emerging from the intake 143 with larger flow (vectors) directed towards the center and right-most region of the combustion chamber which can generate tumble flow, useful for flow coupling to the spark gaps at more advanced ignition firing times (shown also with reference to FIGS. 8a and 8b). Likewise, exhaust flow vectors 144a are shown leaving the combustion chamber into the exhaust runner 144, with partially open exhaust

valve 121b, with minimum flow (small flow vectors 144a) along the far edge 141b nearest to the piston ring crevice where unburnt hydrocarbons are formed, to help trap them as part of the exhaust residual, as already discussed, with the major exhaust coming from the central region of the combustion zone 11 (large flow vectors 144a). The net result is that more unburnt hydrocarbons are trapped and higher compression ratios can be used because of the squish land at the entire far edge of the combustion chamber (where preferably the clearance at TC is approximately 0.04" as is known to those versed in the art).

FIG. 13 is a valve timing diagram depicting a preferred exhaust valve closing (EC) at TC for trapping more exhaust residual, and late intake closing (70° ABC) for higher part-load efficiency and higher, high engine speed output power (higher high speed volumetric efficiency). Intake opening (IO) and exhaust opening (EO) are shown 30° BTC and 30° BBC respectively.

FIG. 14 is a design of the ignition coil output structure for providing dual ignition firing to the piston of the present invention in which the return path for the electrical current is entirely through the spark plugs and not through the engine block. Like numerals represent like parts with respect to the earlier figures.

In this preferred embodiment, the coil secondary winding 146 (primary 147) has two floating high voltage outputs connected to spark plug electrodes 10a and 10b which fire to the surface of the piston 5 at tips 11a and 11b respectively, as already disclosed. The advantage in this design is that current flows from one electrode (say 10a shown as positive) to the piston surface 11a and back again to the other electrode 10b at piston surface point 11b, so that the spark current, which produces two sparks to the piston surface, does not have to find a return path along the piston/cylinder interface 148 or through other parts connected to the piston. Preferably, one spark gap 149a is smaller than the other (149b) to reduce the required spark firing voltage, or the two plugs have differing capacitances to divide unevenly the output voltage, as is known to those versed in the art.

FIG. 15a is a partial, not to-scale, top view layout of two side-by-side engine cylinders in a multi-cylinder engine with a preferred combustion chamber FIG. 15b of the present invention in which the valves are lined up approximately in the length axis of the engine and a single overhead cam 150 (broken lines) is shown for actuating the valves (which can also be actuated by other means, i.e. push-rods and rockers, electronically, etc). Like numerals represent like parts with respect to the earlier figures. This embodiment is more suitable for swirl generation as

dictated by the intake runner 143 which is at the side of a combustion chamber diameter line which produces flow vectors 151 and 152 (FIG 15b) at the spark plug sites (two plugs indicated although plug 118 is preferred if only one plug is used since it will move the spark towards a more central part of the combustion zone 110a.). The squish flow vectors are not shown (but will be approximately orthogonal to the swirl vectors). The exhaust runner may be on the opposite side or same side as the intake runner, as indicated in FIG. 15a, it being understood that in an actual engine one or the other would be used (the drawing showing two options, as is also done in FIG. 16a, to conserve drawing space). The combustion zone is indicated as pear shaped with larger diameter zone 110a under the exhaust valve.

FIG. 16a is a partial, not to-scale top view layout of two side-by-side engine cylinders in a multi-cylinder engine with a preferred combustion chamber FIG. 16b of the present invention in which the valves are lined up transverse to the length axis of the engine. Like numerals represent like parts with respect to the earlier figures. This embodiment is more suitable for tumble flow generation as dictated by the intake runner which is along a combustion chamber diameter line. One or two intake valves (121aa and 121ab) are indicated. Two (double) overhead cams 153 and 154 (broken lines) are indicated for separate operation of the intake and exhaust valves, which makes it easy to provide independent variable (intake and exhaust) valve timing. In FIG. 16b the version with two intake valves 121aa and 121ab and two intake runners 143a and 143b are shown, which can allow for swirl generation by deactivating one of the intake runners, resulting in flow vectors 155 and 156 if valve IN2 (121ab) is deactivated. Squish vectors 2 are also shown in this figure.

FIGS. 17a, 17b and 17c are partial top views of low inductance, high energy ignition coil structures with open magnetic paths suitable for application where greater than 12 volts power is available and designed for having two high voltage towers 160a and 160b suitable for dual ignition. As disclosed elsewhere, this low inductance structure normally has a concentric winding structure with preferably a two layer primary winding 161 and secondary winding 162 made up of typically segmented bobbin winding. Each of the designs have two or more open ends 163a, 163b (and 163c and 163d for FIG. 17b) to provide a typical low primary winding inductance L_p of 0.3 milliHenry (mH) to 1.0 mH.

The coil of FIG. 17a features two side-by-side U-cores 164 and 165 of approximately equal area of each leg placed 180° to each other so that the secondary winding 162 can have its two

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ends brought out at the two open ends at the high voltage towers for compactness. The high voltage towers 160a and 160b act to also place a clearance between the magnetic fields 166a and 166b and any external metallic surface where the fields might otherwise couple. Likewise for FIG. 17b where two of the four open end magnetic field lines are shown (166a and 166b), FIG. 17c where one of the two open end magnetic field lines are shown. In FIG. 17a preferably the core is made of stacked U-laminations.

FIG. 17b features a center leg magnetic core 167 (preferably stacked lamination) and outer magnetic core 168 which can be a tube of wound insulated magnetic tape, e.g. 4 mil silicon iron tape, or two stacked laminations, or other material of different type or shape. Three of four mounting holes 169a, 169b, and 169c are shown.

FIG. 17c is a two piece 170a, 170b bobbin type magnetic core, preferably made of stacked laminations, disclosed elsewhere but here designed to have the two high voltage outputs 160a, 160b at the two ends.

FIG. 18 is a top partial view of an open E-type low inductance, high energy ignition coil with open magnetic path with some lower turns loosely coupled secondary windings constituting a series of secondary circuit external leakage inductors, usable also in the designs of FIGS. 17a and 17b, for reducing the end effect voltage overshoot. FIG. 18a is an equivalent circuit of the coil of FIG. 18 used in an inductive ignition circuit. Like numerals represent like parts with respect to the earlier figures.

In FIG. 18, the secondary winding 162 is a segmented winding separated into bays (10 shown as shaded rectangles), with the last three 162a being thinner and having fewer turns as well as extending beyond the inner concentric primary winding 161 and having looser magnetic coupling as indicated by the magnetic field lines 170 shown in the left half of the drawing. This is to reduce the bay-to-bay voltage immediately following breakdown of the spark gap (gap 15 of FIG. 18a with gap capacitance 117a across it). The three windings are chosen by way of example only. In FIG. 18a the last three windings are shown as three external inductors 171 of total inductance L_{end} to which can be added an external inductor L_{ext} , as disclosed elsewhere, to reduce the voltage overshoot end-effect immediately following spark breakdown. That is, upon spark breakdown, the high voltage terminal across the spark gap immediately drops to zero volts, and the voltages at the end of the various segments cannot drop as quickly, so that an anomalously high voltage can appear across the secondary bays.

In the case where two spark plugs are shown per combustion chamber, a range of options exist for firing them. Under some conditions, e.g. wide-open-throttle (WOT), only one plug may be fired. In some conditions, the plug firing may not be simultaneous. In some conditions, e.g. cold start, the plugs may be multi-fired.

- 5 Since certain changes may be made in the above combustion chamber designs, in the location and use of the ignition, and in the air-fuel mixture flows without departing from the scope of the invention herein disclosed, it is intended that all matter contained in the above description, or shown in the accompanying drawings, shall be interpreted in an illustrative and not limiting sense.

What is claimed is:

1. An internal combustion engine ignition system for ignition, combustion and expansion of an air-fuel mixture and producing work by means of a movable piston within a cylinder that has a cylinder head powered by the combustion-expansion through cyclic compression and expansion strokes, the engine employing a colliding-flow-coupled-spark-discharge process (CFCSD), comprising, in combination:

a) a combustion chamber placed and shaped to form a combustion zone with intake valve means and exhaust valve means mainly within the cylinder head constructed and arranged to induce high air-flow with intense turbulence regions as the piston moves near the cylinder head,

b) means defining a high energy ignition system with at least one spark plug located at the edge of said combustion zone in the high air-flow regions generated therein by the piston motion as it nears the cylinder head on a compression stroke,

c) the system constructed and arranged such that as the piston approaches the cylinder head, at least one spark plug is fired to produce sparks which are forced at least radially inwards by said piston induced air-flows towards the center of the combustion zone where there exists said intense turbulence by colliding flows, whereby a rapid combustion of the air-fuel mixture is established resulting in high engine efficiency and low emissions.

2. An engine ignition system as defined in claim 1 wherein the high air-flows are of a squish type and are produced by a surface of the movable piston, when near top center (TC) of the cylinder, generated by closely spaced cylinder head regions around said combustion zone.

3. An engine ignition system as defined in any one of claims 1 to 2 wherein said combustion zone is located mainly under the exhaust valve means.

4. An engine ignition system as defined in any one of claims 1 to 2 wherein said chamber, comprises at the two centrally located adjacent valves, the intake and exhaust

valve means and the combustion zone is located mainly under said valves.

5. An engine ignition system as defined in any one of claims 1 to 4 wherein said ignition system has flow resistant spark of peak current between 300 and 800 ma.

6. An engine ignition system as defined in any one of claims 1 to 5 wherein two spark plugs are used at two edges of the combustion zone where high air-flows are generated at the time of ignition firing.

7. An engine ignition system as defined in any one of claims 1 to 6 further comprising fuel injection means for introducing fuel directly into the combustion zone.

8. An engine ignition system as defined in any one of claims 1 to 7 wherein at least one of said one or more spark plugs has a center high voltage electrode and wherein the spark formed from said one or more spark plugs is formed between a center high voltage electrode thereof and the piston surface when the piston is near TC.

9. An engine ignition system as defined in any one of claims 1 to 8 wherein said piston has an interface, and the said valve means are above it comprising distinct intake and exhaust valved combustion zone as projected onto a piston surface is defined by arcs around and substantially paralleling an outer edge of each of two valve openings and joined by essentially straight line barriers defining large squish lands on the two sides of said lines at whose inner edges one or two spark plugs are located to enable high inward radial squish flow to impinge on their sparks formed during ignition firing.

10. An engine ignition system as defined in any one of claims 1 to 9 wherein a fuel injector is placed in the center of the combustion zone between two openings of the valve means.

11. An engine ignition system as defined in any one of claims 1 to 10 and further comprising at least two spark plugs in the head which form an angle of 15 to 45

degrees with the vertical with their spark gaps in a central portion of a squish flow to direct the sparks radially in and upwards.

12. An engine ignition system as defined in any one of claims 1 to 11 wherein the engine has a compression ratio that is high and ranges between 10:1 and 12:1.

13. An engine ignition system as defined in any one of claims 1 to 12 and further comprising a squish land located substantially at the entire periphery of the combustion chamber to cool the end gas to allow for a higher engine compression ratio.

14. An engine ignition system as defined in any one of claims 1 to 13 wherein the combustion chamber is part of a cam type virtual three stroke engine defined by intake and compression stroke approximately $\frac{1}{2}$ of the expansion stroke and an expansion ratio approximately 12 to 1 or greater.

15. An engine ignition system as defined in any one of claims 1 to 14 and further comprising two spark plugs, each having a high voltage end, per combustion chamber with a dual output ignition coil with at least two open magnetic paths wherein two high voltage towers are located for connecting to high voltage ends of said spark plugs.

16. An engine ignition system as defined in claim 15 wherein the two spark plugs are fired to the piston and use a dual output coil with primary and secondary windings and a magnetic core in an electrical circuit which allows for the spark current to complete an electrical circuit entirely within the coil dual output secondary coil winding.

17. An engine ignition system as defined in any one of claims 1 to 16 wherein the ignition system used is an inductive ignition utilizing a coil with primary and secondary windings and a magnetic core in an electrical circuit with low primary inductance below 1 milliHenry and with at least one open magnetic path end where a secondary winding, comprised of segmented bays, extends to locate one or two high

voltage towers.

18. An ignition system as defined in claim 17 wherein said secondary winding is a segmented winding with at least two of the last winding bays at the high voltage end having fewer turns than the previous bays and with at least one of said last winding bays extending beyond the primary winding at a magnetic core open end.

19. An engine ignition system as defined in any one of claims 7 to 8 wherein one spark plug is located at said edge of the combustion zone and a fuel injector is located at the opposite side of the combustion zone.

20. An engine ignition system as defined in any one of claims 1 to 18 wherein two spark plugs are used and wherein only one spark plug is fired under some operating conditions of the engine including high engine load conditions with stoichiometric or rich of stoichiometric air-fuel ratios.

21. An engine ignition system as defined in any one of claims 1 to 20 wherein the exhaust valve means is located essentially at the center of the combustion chamber with the combustion zone located mainly underneath it.

22. An engine ignition system as defined in claim 15 wherein a magnetic core of said coil is made up of two U-cores located at 180° of each other with open ends at opposites ends wherein are located the high voltage towers.

23. An engine ignition system as defined in claim 15 wherein a magnetic core of said coil is made up of a central I-core over which primary and secondary windings are wound and an outer magnetic core section which together define a core with two open ends where the high voltage towers are located.

24. An engine ignition system as defined in claim 15 wherein magnetic core of said coil is made up of a central I-core over which primary and secondary windings are

wound and two orthogonal magnetic core legs located at each end of the I-core which all together define a bobbin type core with two open sides where high voltage towers of the secondary winding are located.

25. An ignition-combustion system for internal combustion with intake and exhaust port means and fixed and movable elements within each combustion chamber of an engine, comprising:

(a) means defining a compact combustion chamber zone mainly in a fixed portion of the combustion chamber,

(b) means for forming large air-squish zones at an edge of a combustion zone which produce colliding squish flows with high turbulence at about centrally of the combustion zone,

(c) one or more igniting means located at or near the edge of the combustion zone with at least one ignition means located within the high squish zone,

(d) ignition means comprising means for generation of one or more high energy flow-resistant ignition spark means which move, for at least some conditions of operation of the engine, under the influence of the squish flow towards the central turbulence region as the movable element nears engine top center (TC) and ignition firing occurs near TC,

e) means for reinforcing the squish flow further by other bulk air-flows at a spark plug site when ignition occurs substantially earlier of TC when squish flow is relatively low and such other bulk air-flows are relatively high,

to produce rapid and complete combustion of lean and high EGR mixtures whereby high engine efficiency and lowest emissions are achieved.

26. An ignition-combustion system as defined in claim 25 wherein said other bulk air-flow is of a swirl type circulating around said combustion chamber.

27. An ignition-combustion system as defined in claim 25 wherein said other bulk air-flow is of a tumble flow type circulating up and down said combustion chamber.

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28. An ignition-combustion system as defined in any one of claims 25 to 27 wherein at least approximately 50% of said compact combustion zone is under the exhaust port means.

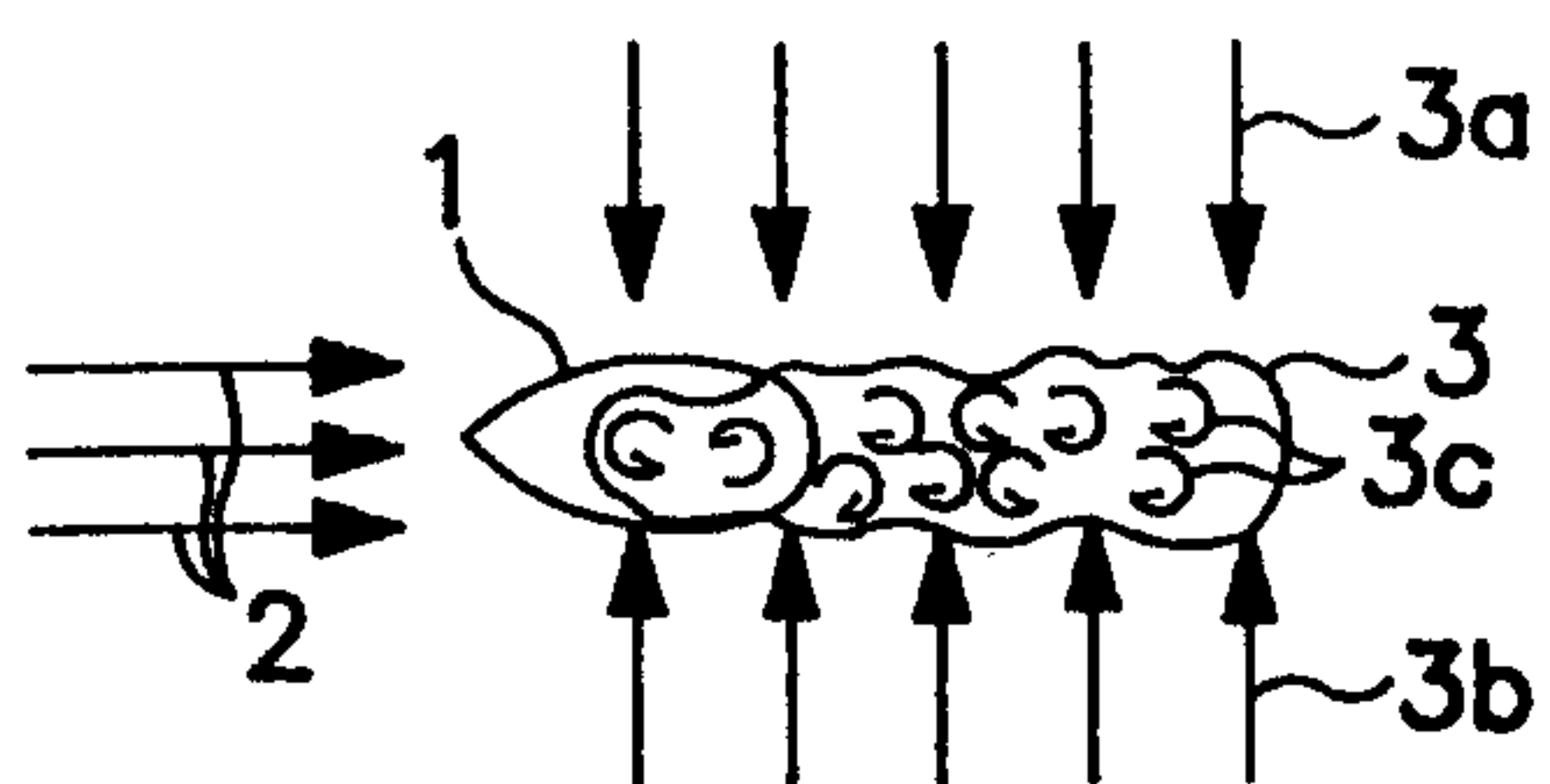


FIG. 1A

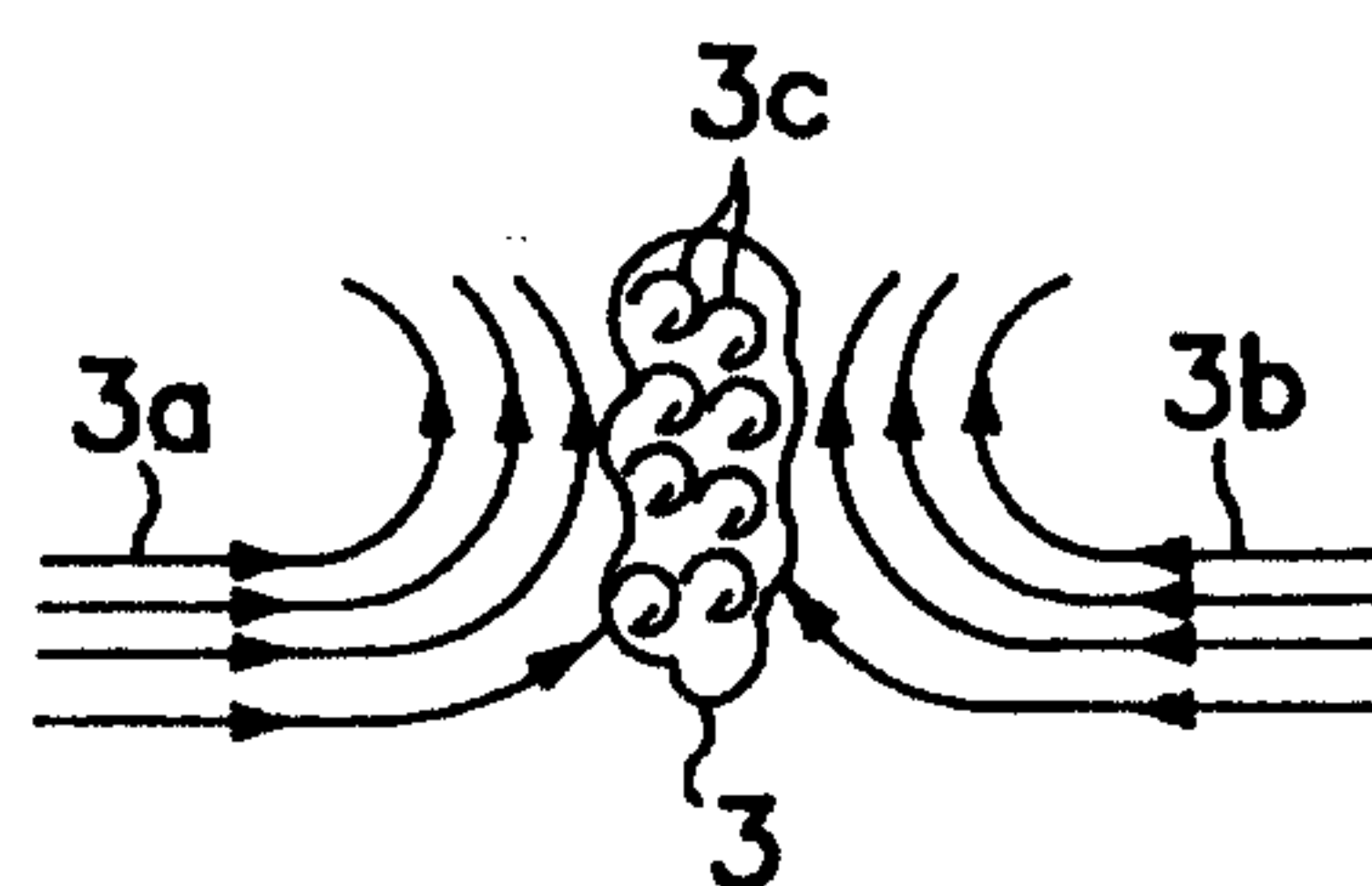


FIG. 1B

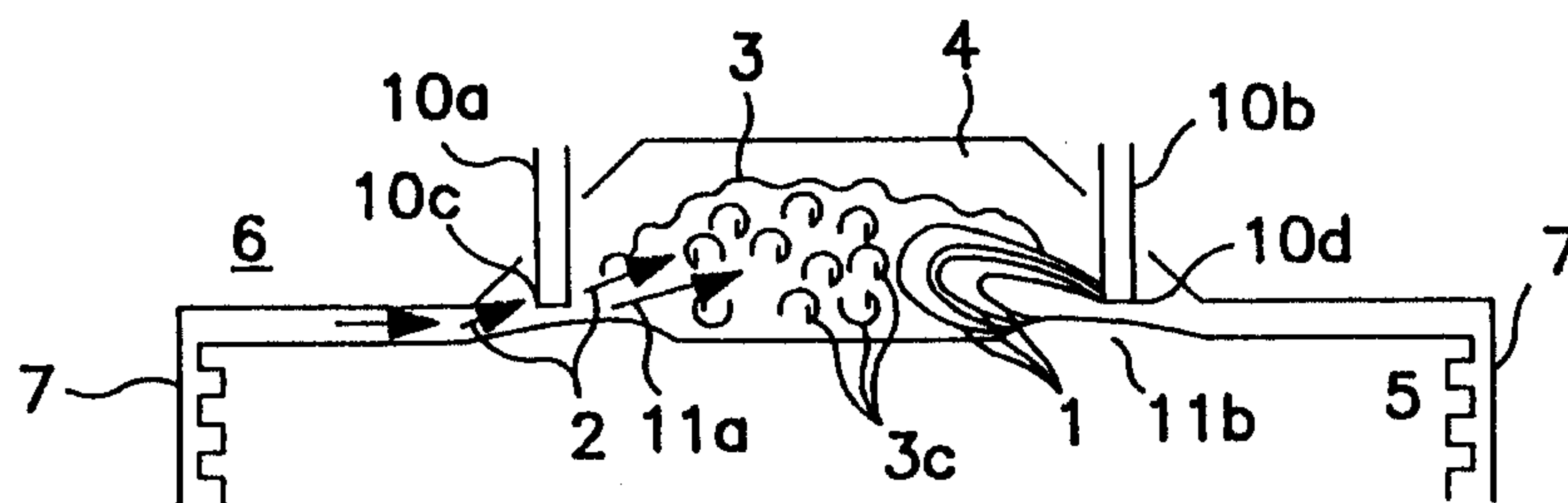


FIG. 2A

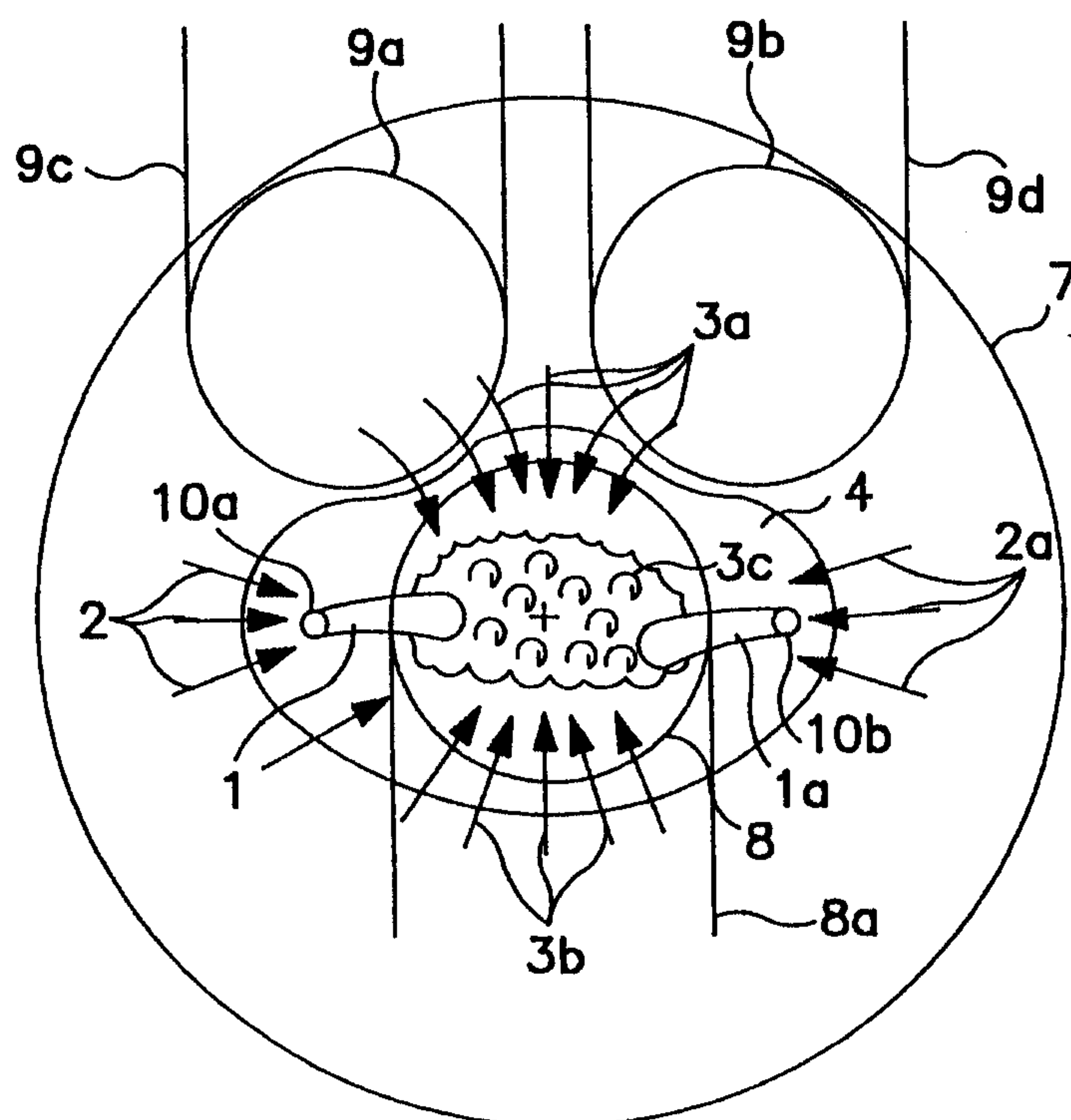


FIG. 2B

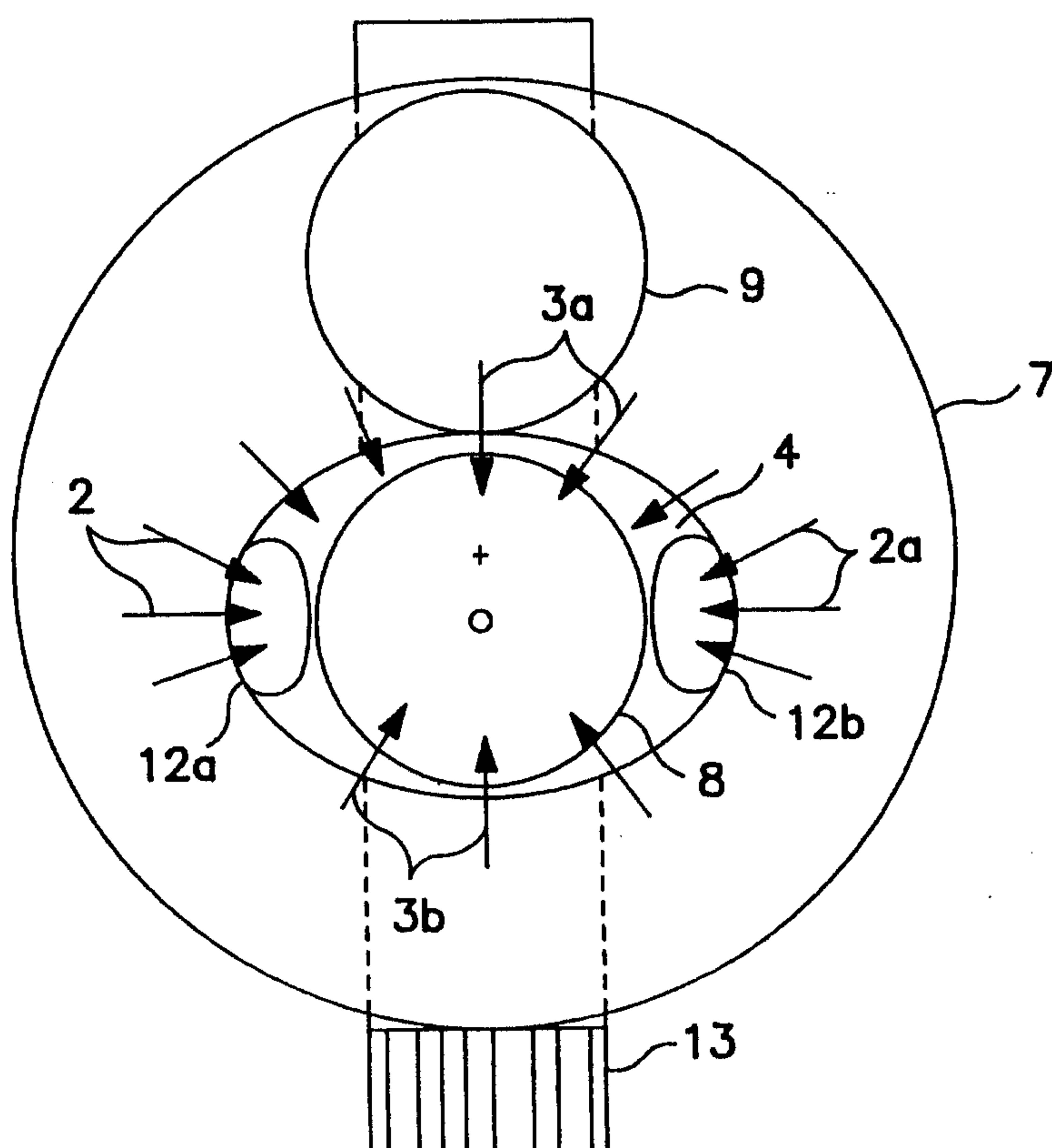


FIG. 3

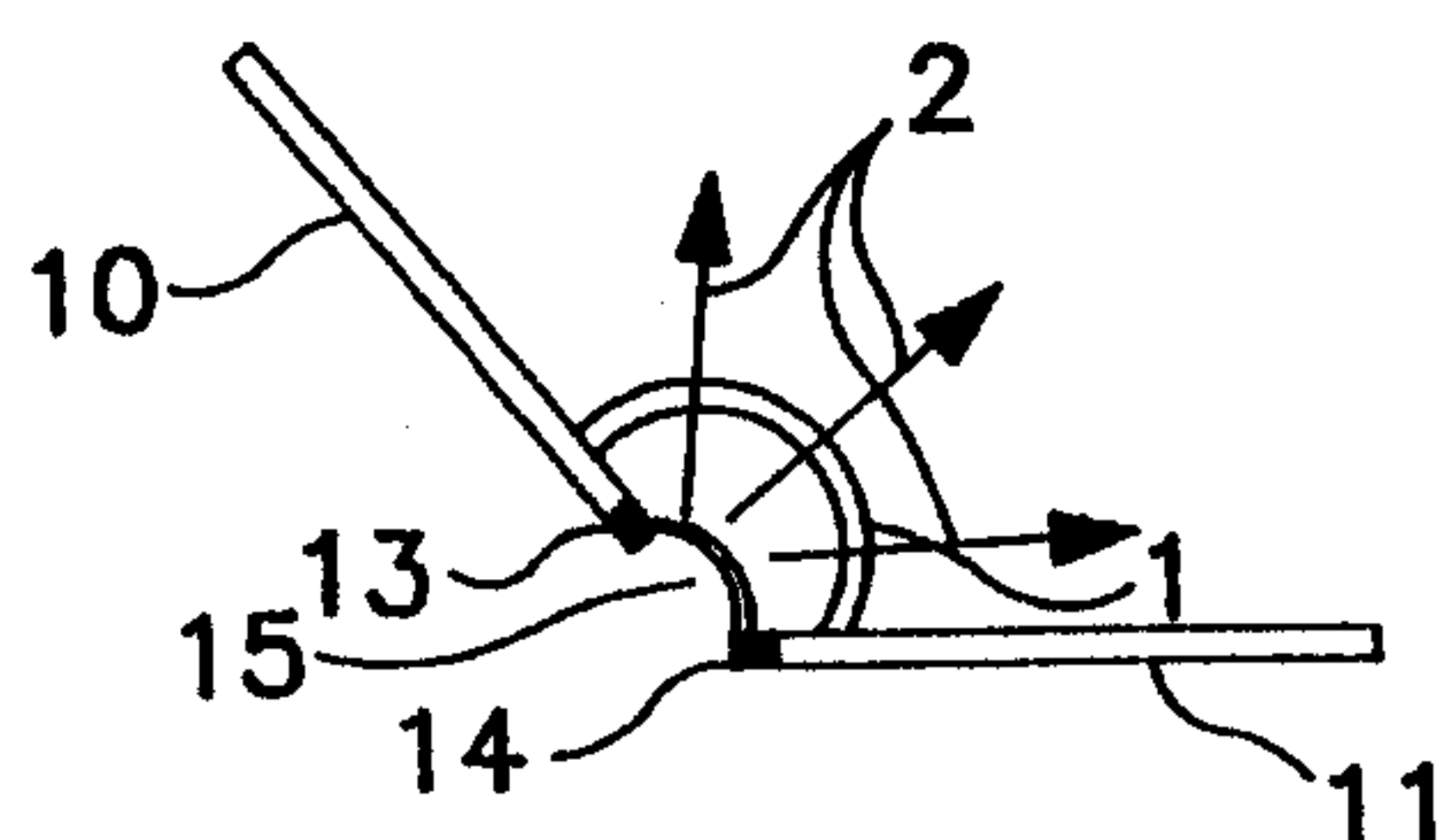


FIG. 4A

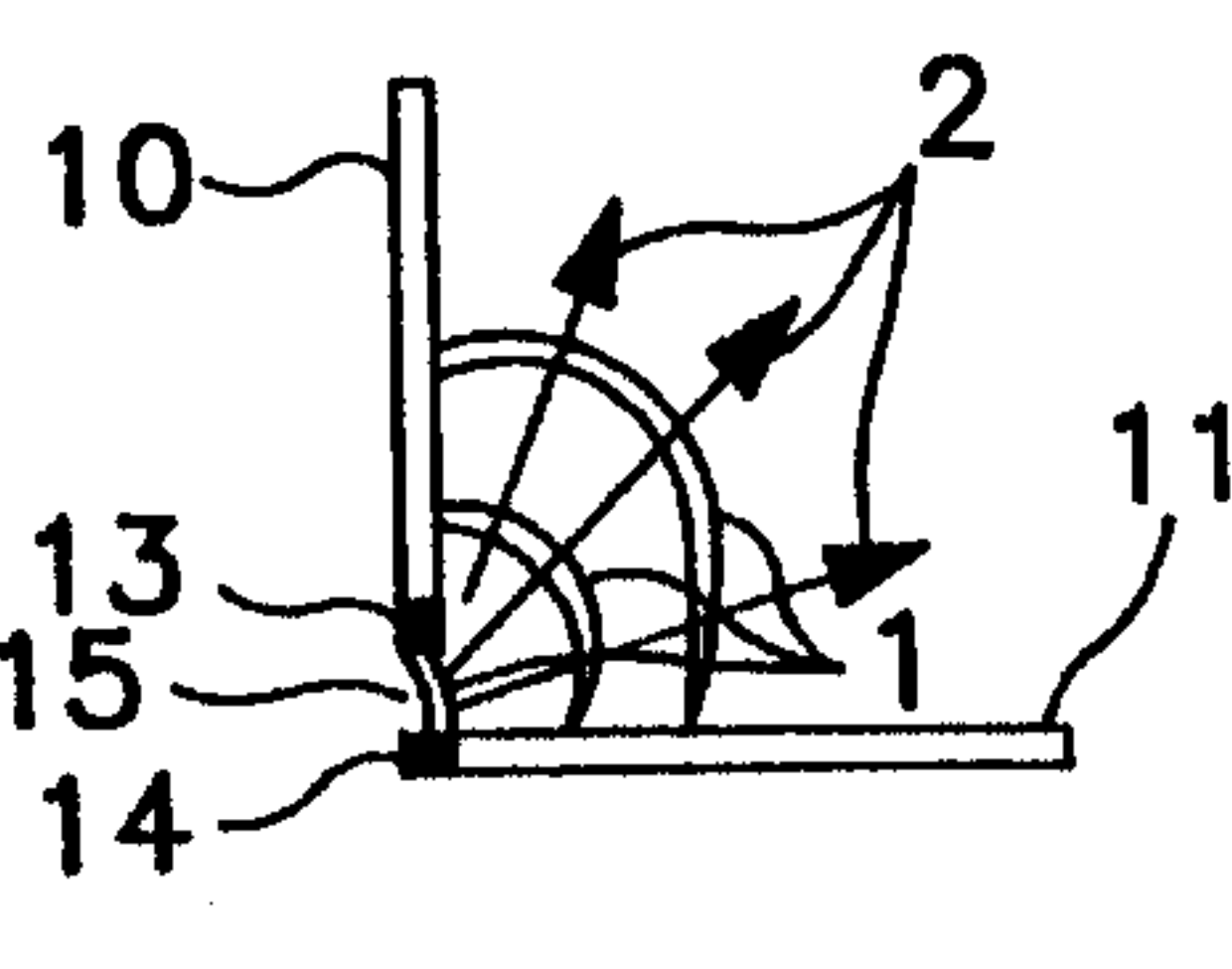


FIG. 4B

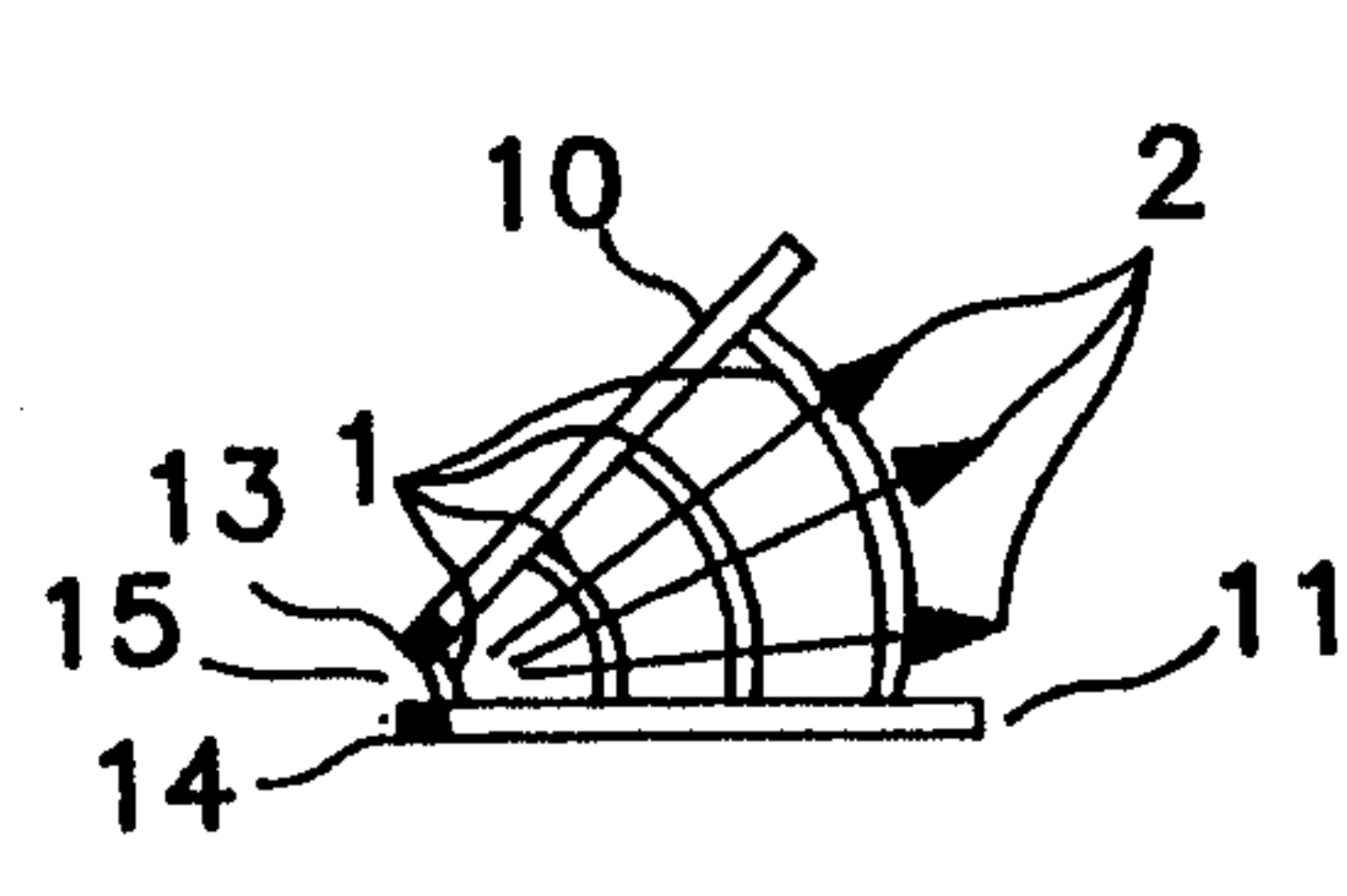


FIG. 4C

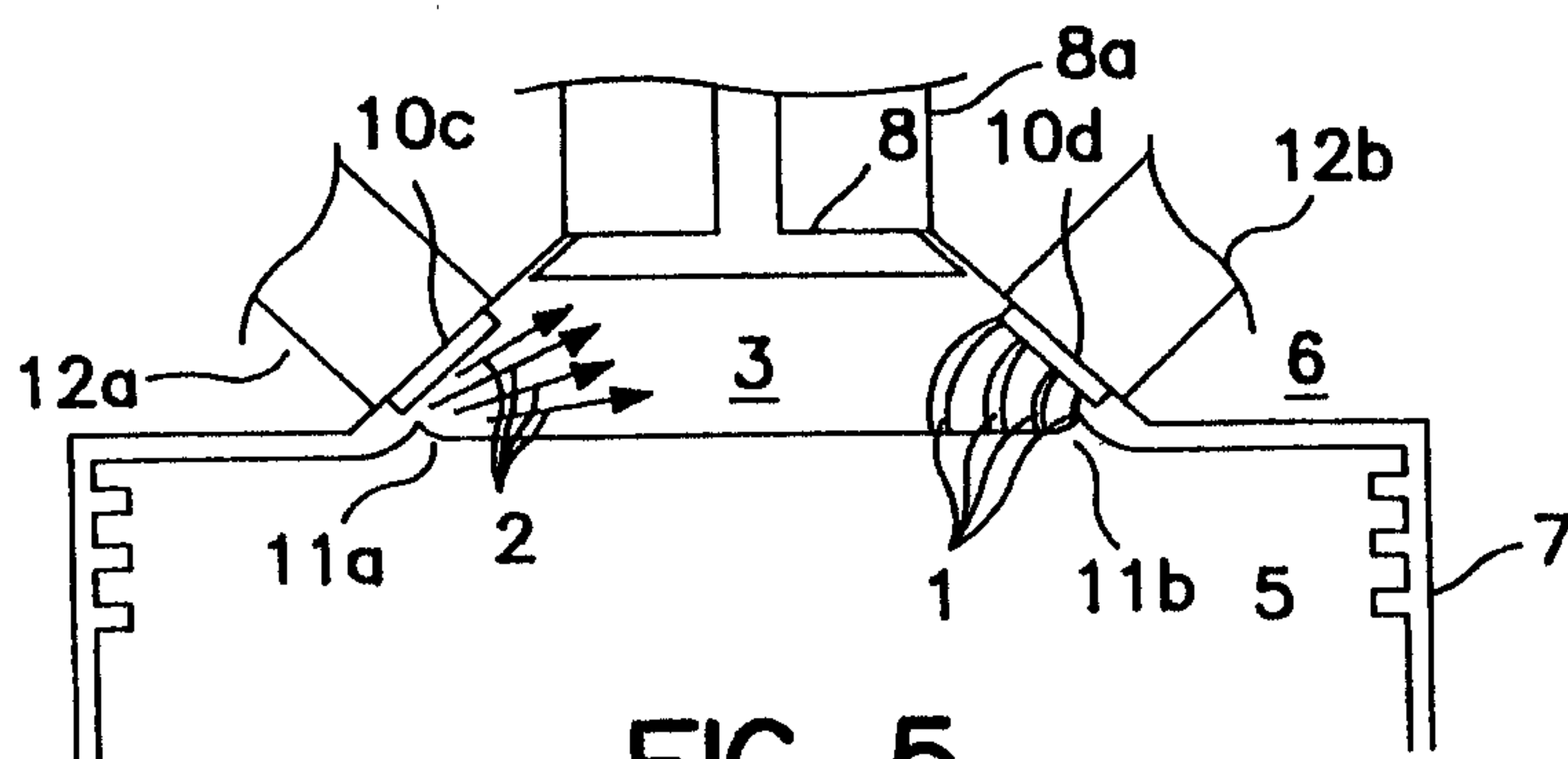


FIG. 5

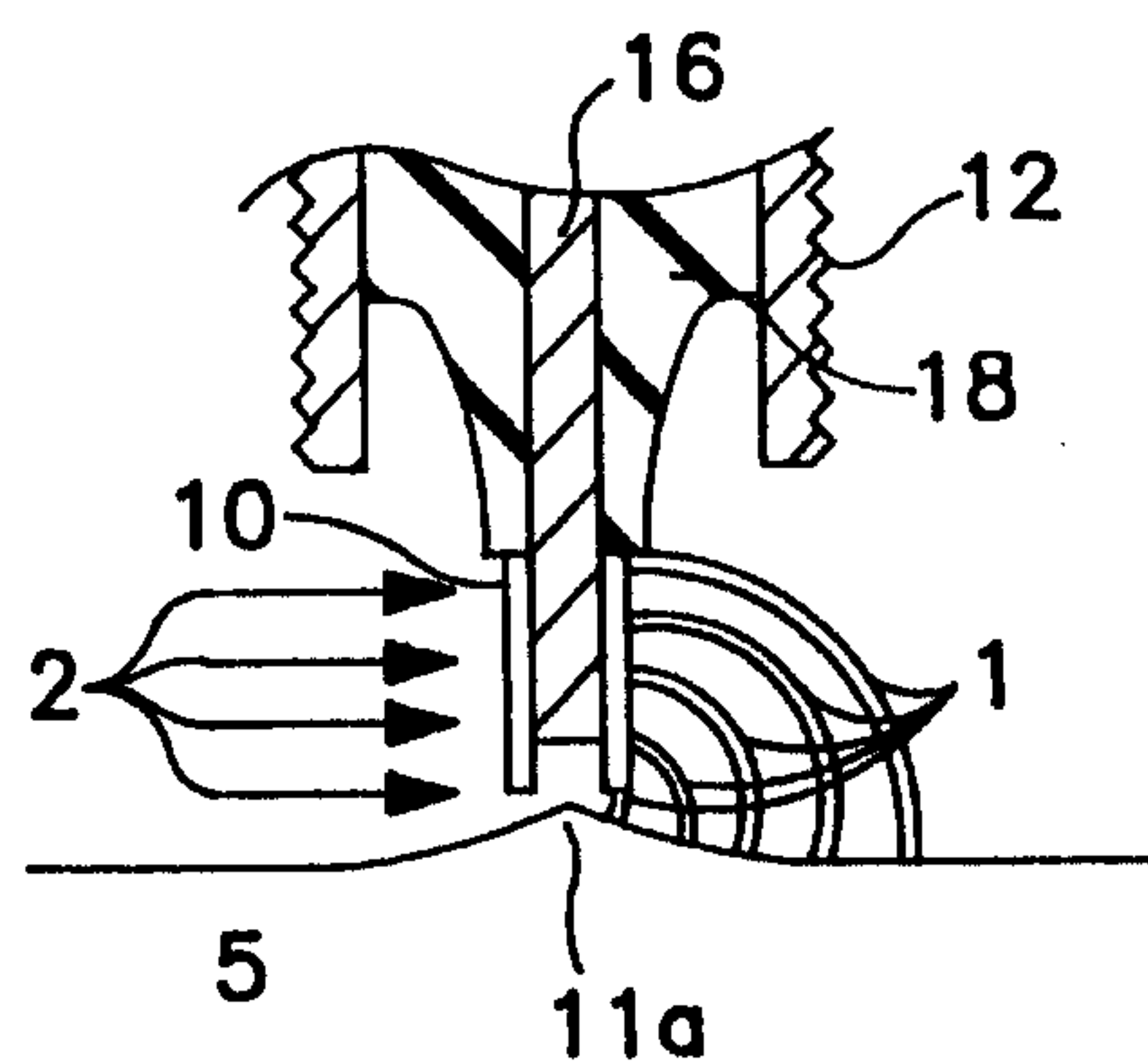


FIG. 6A

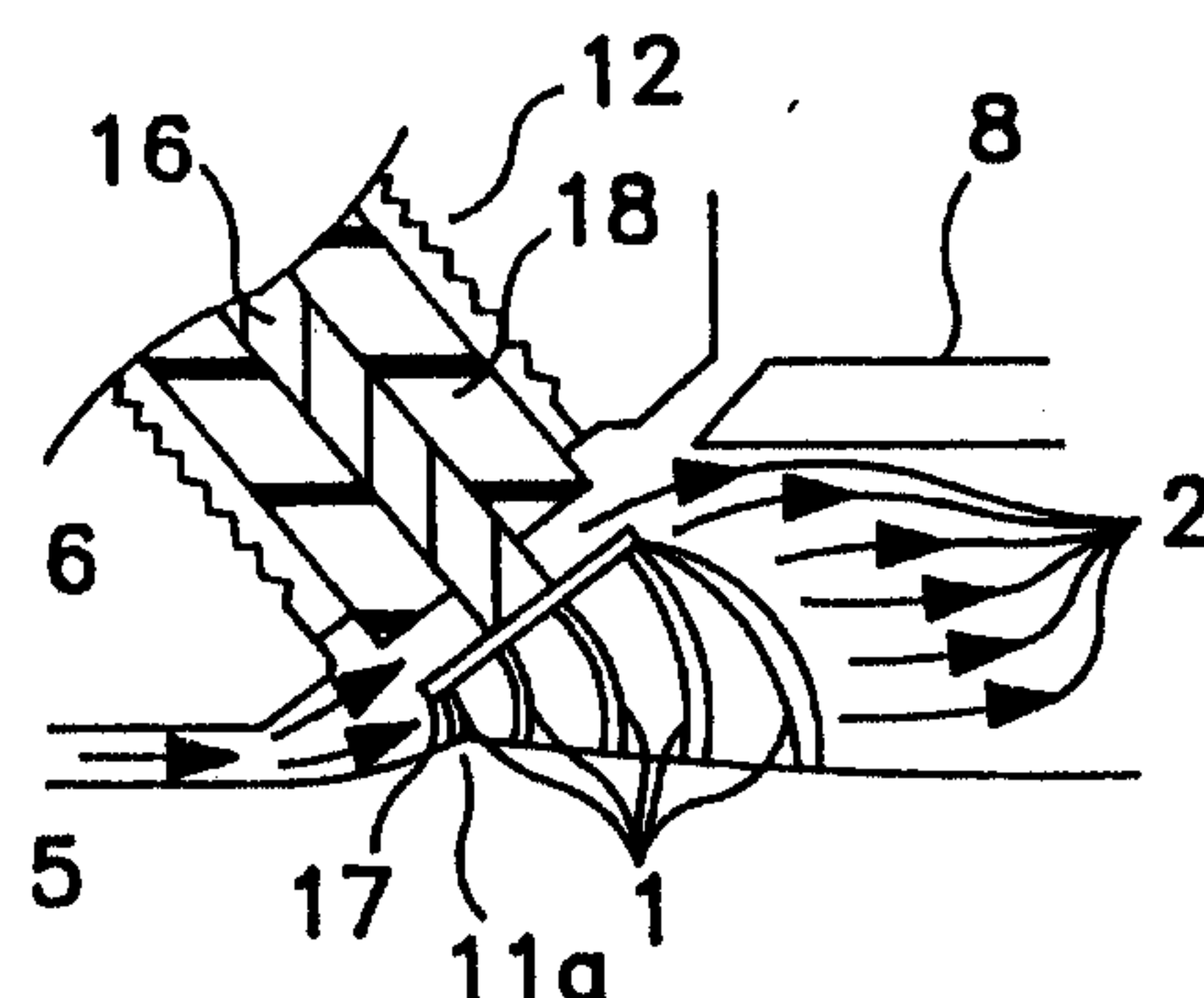


FIG. 6B

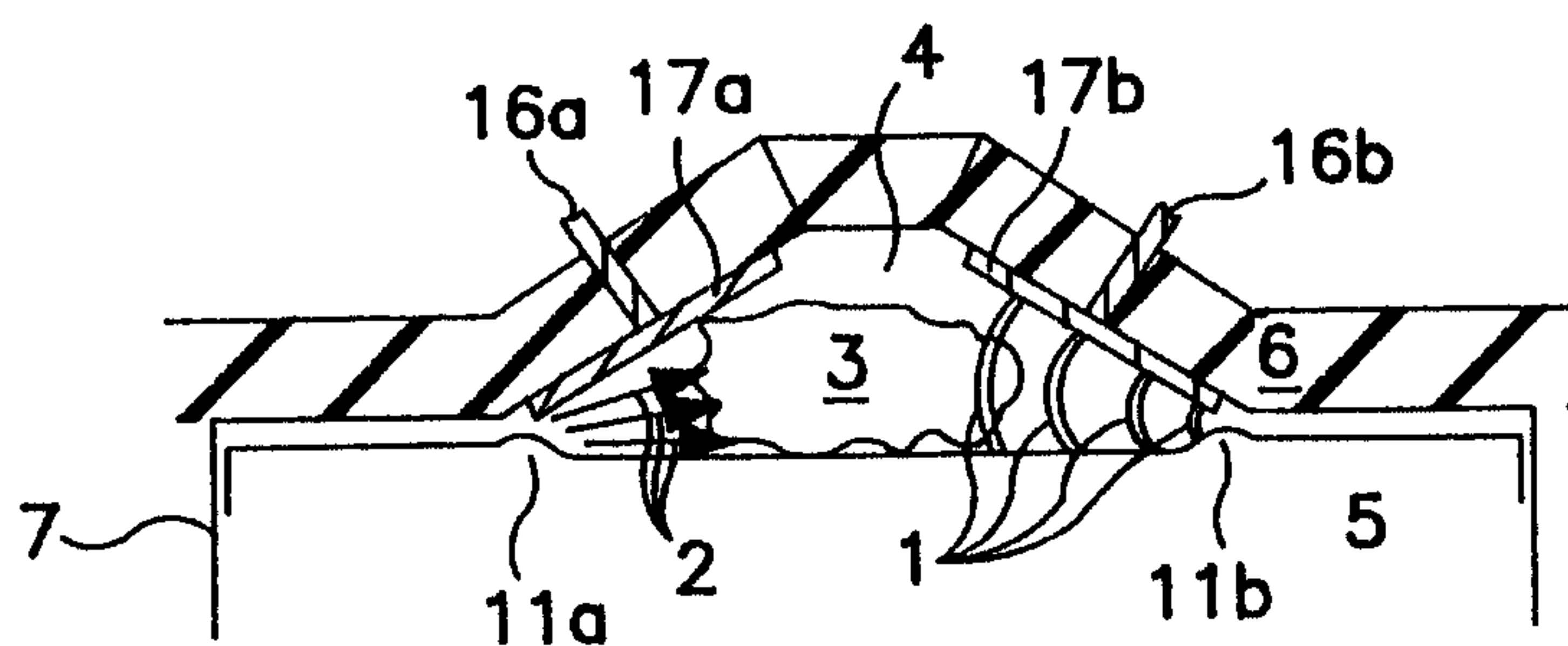


FIG. 7

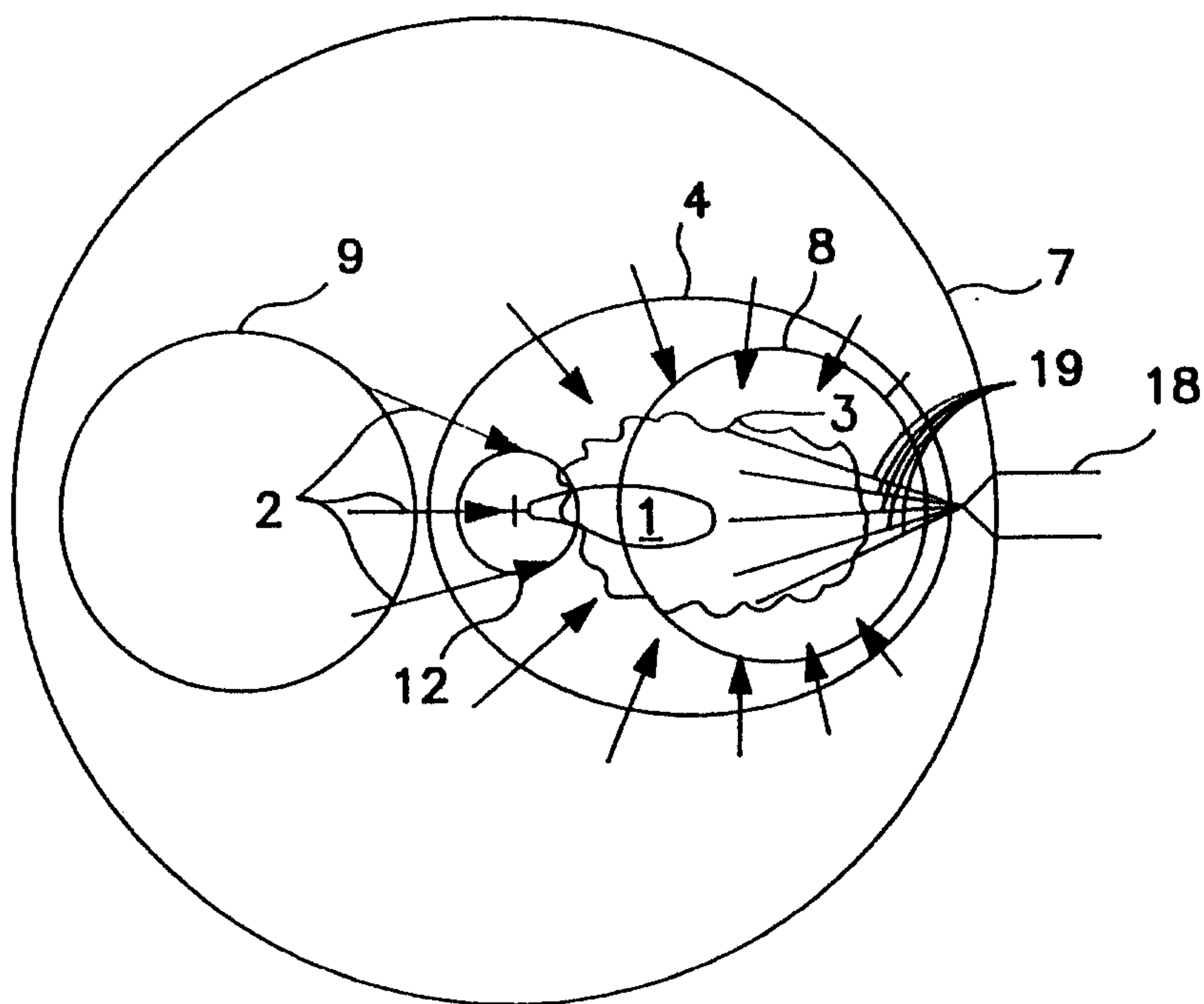


FIG. 8

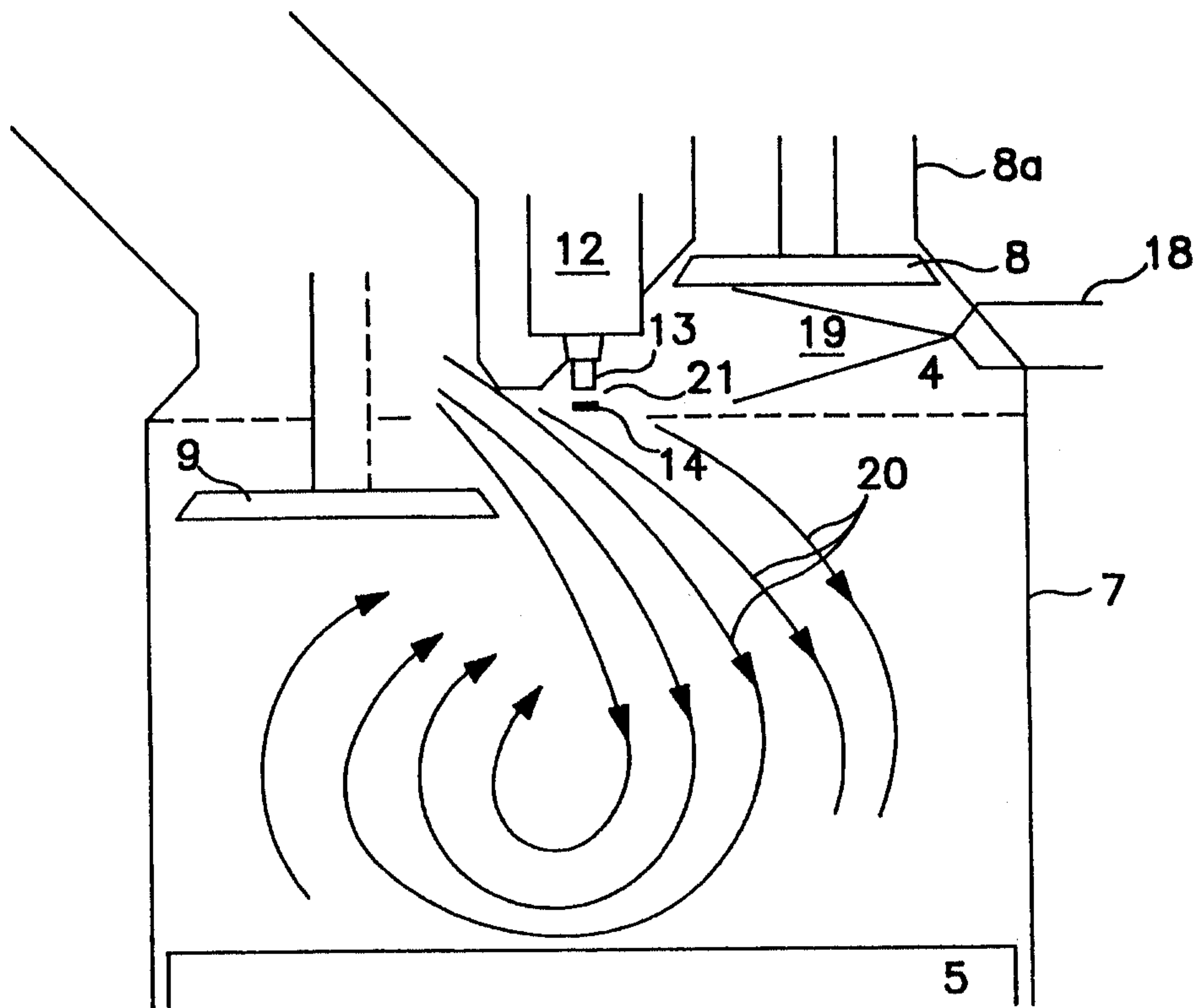


FIG. 8A

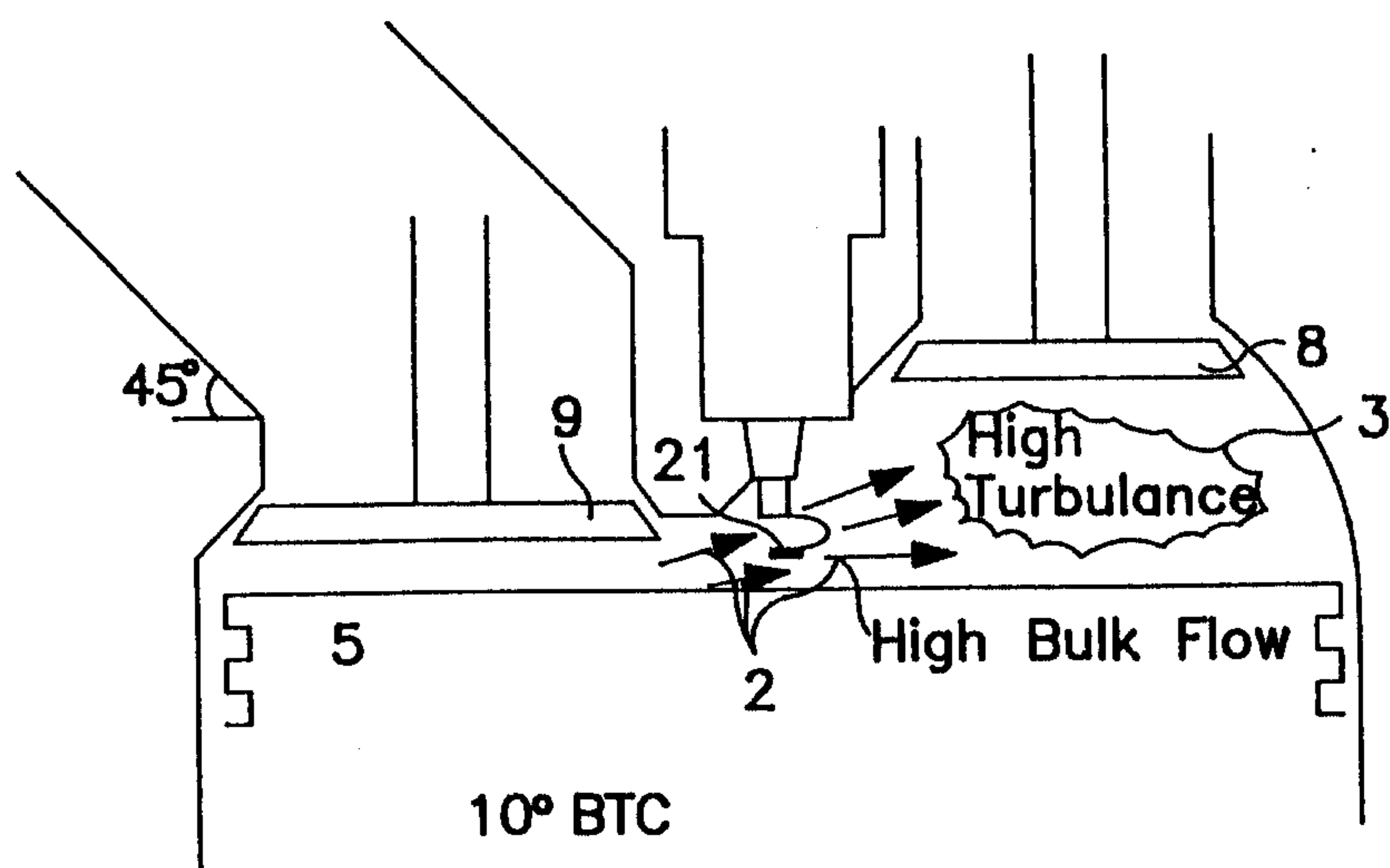


FIG. 8B

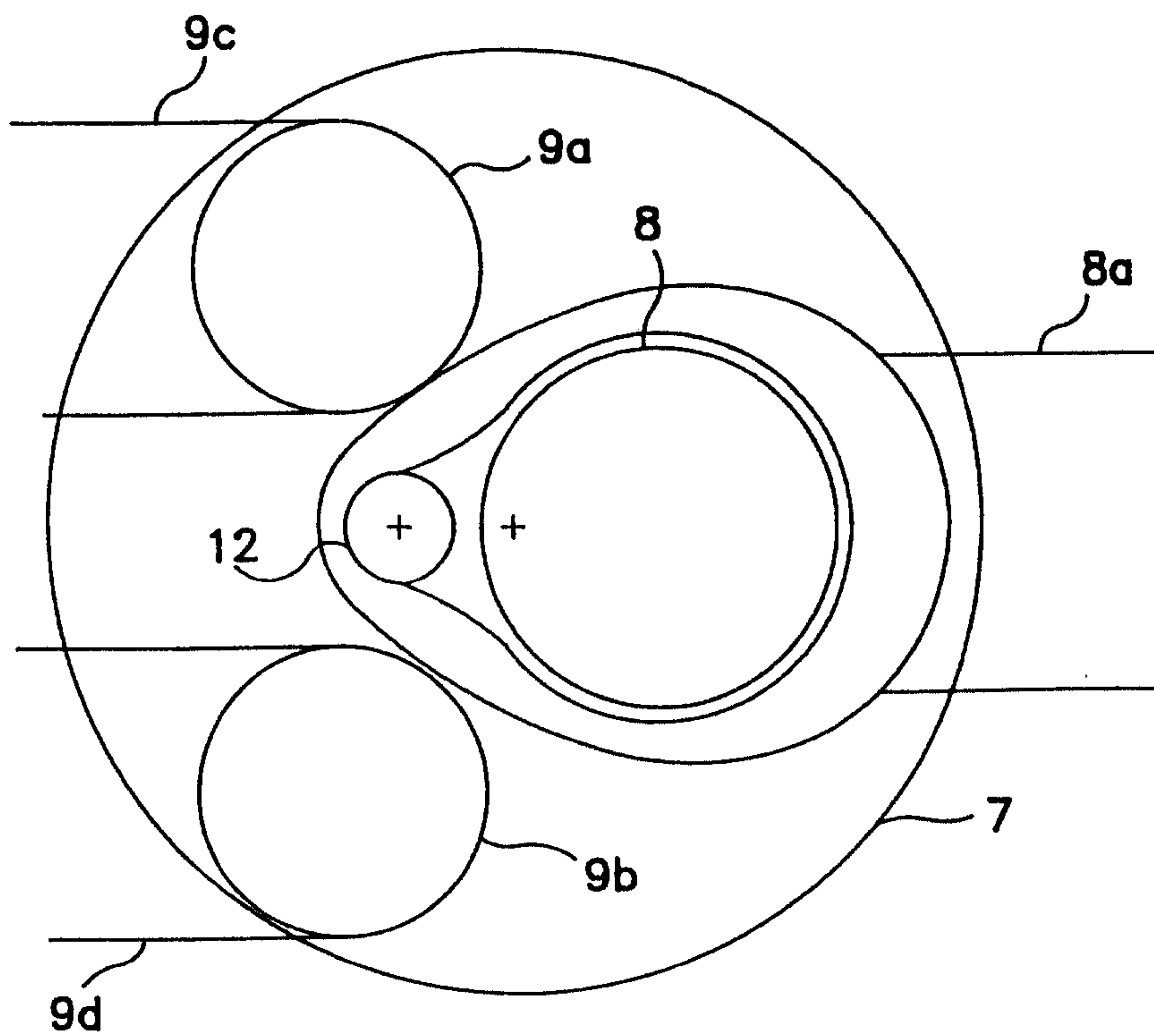


FIG. 9A

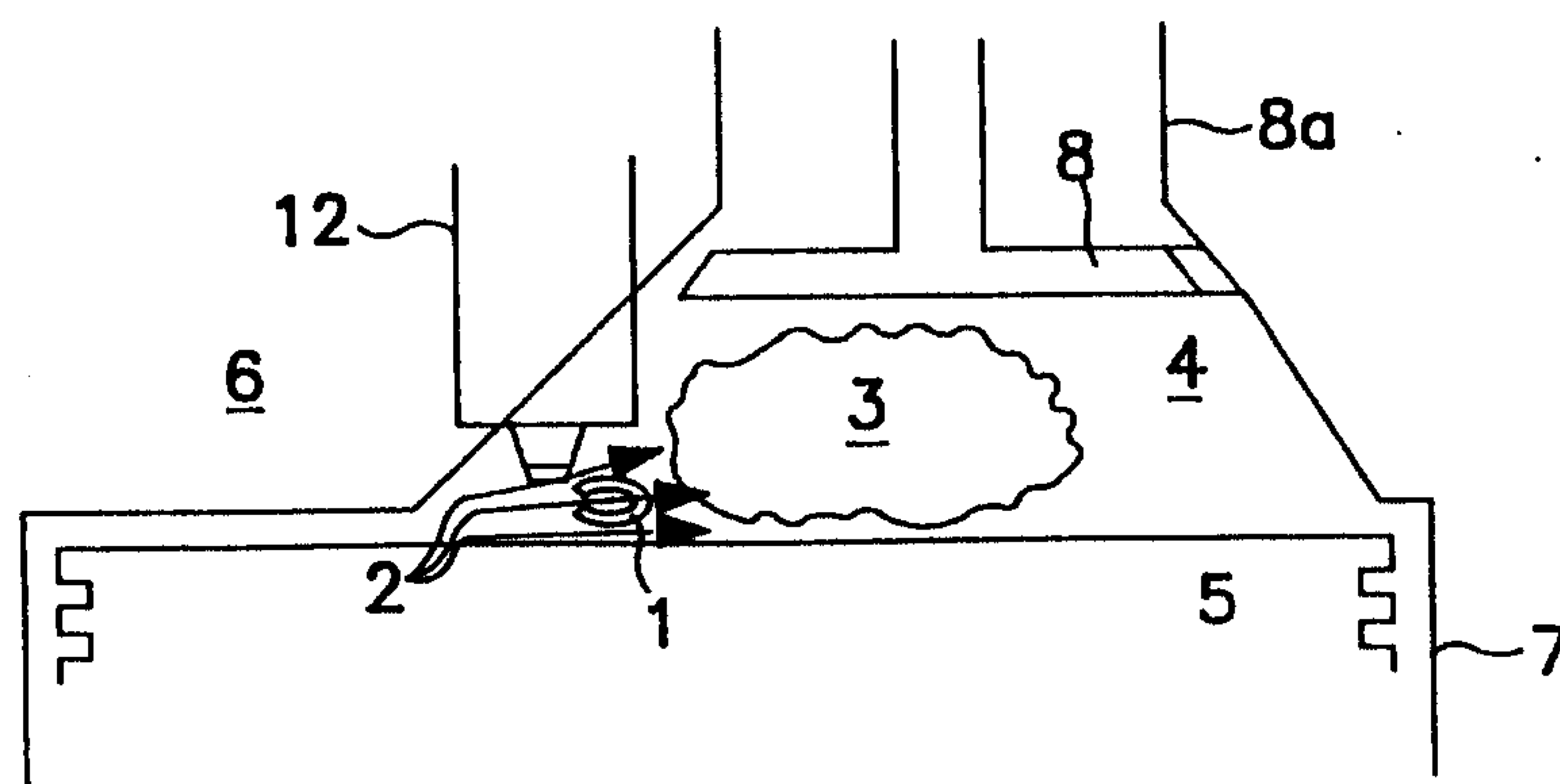


FIG. 9B

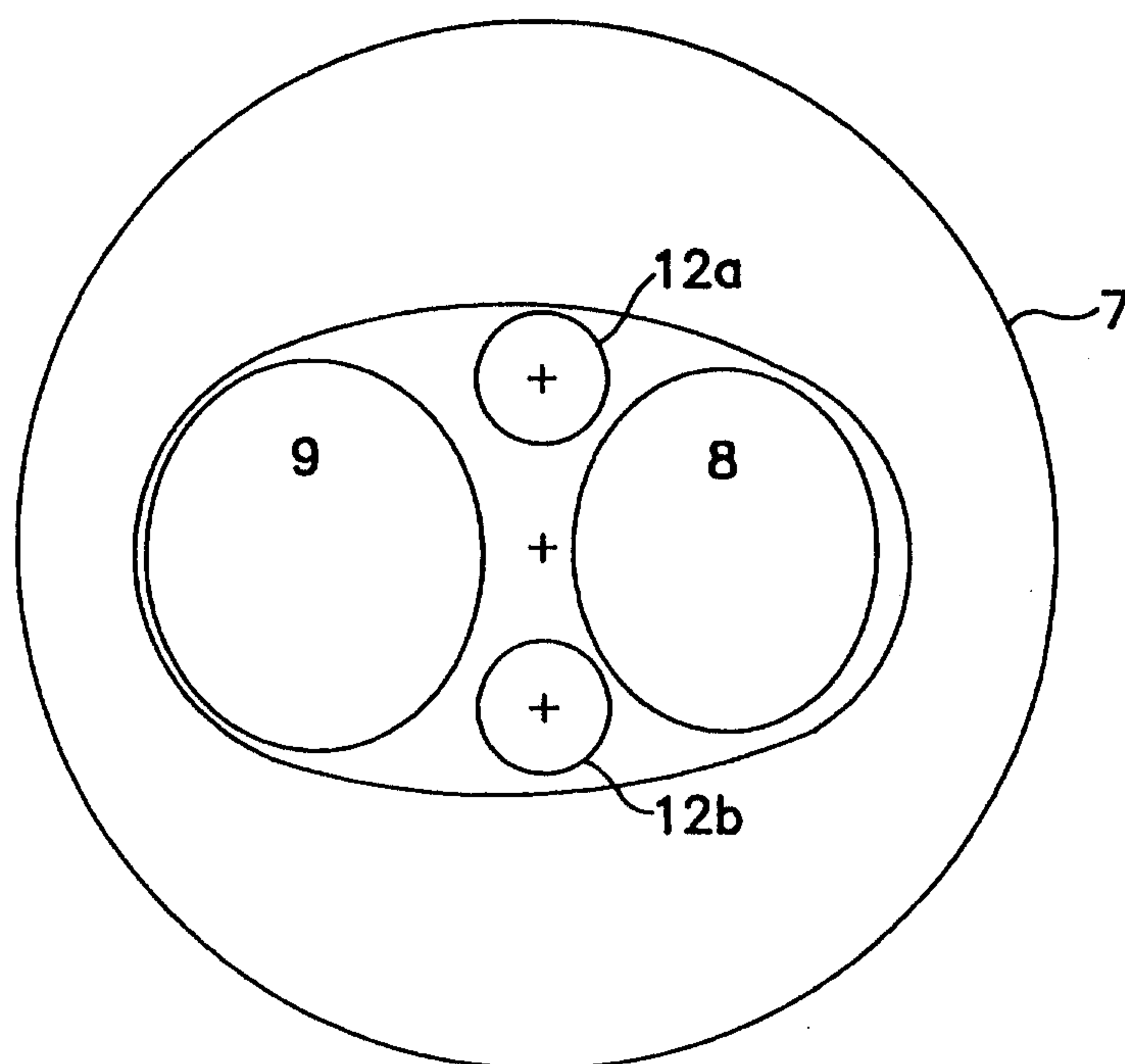


FIG. 10A

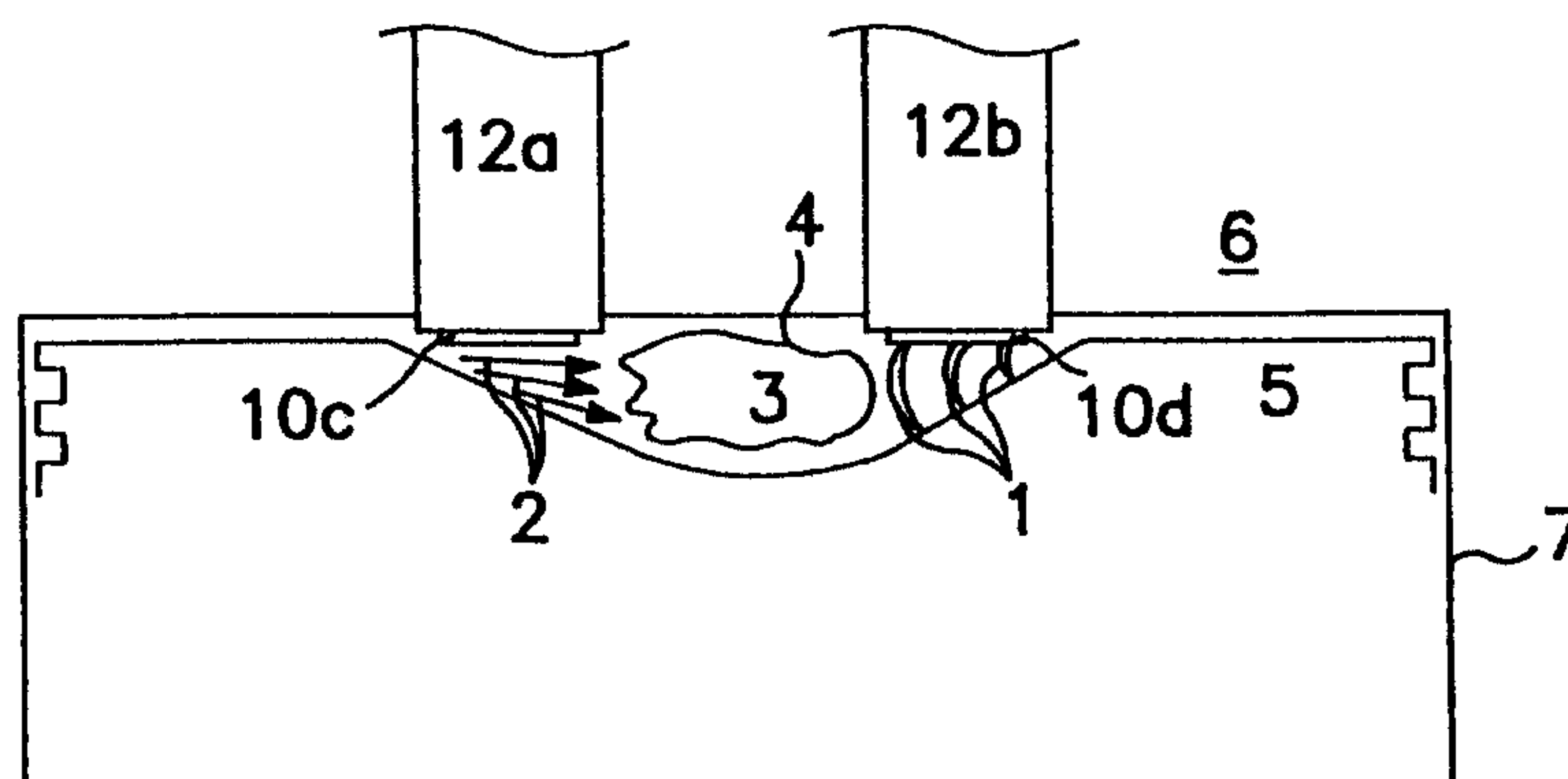


FIG. 10B

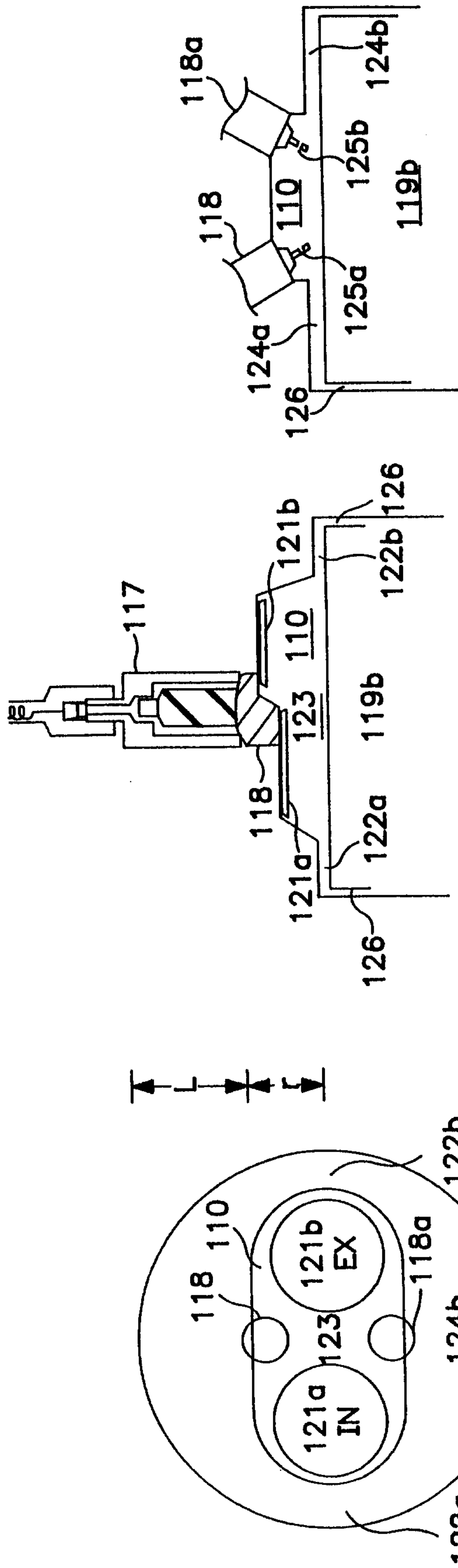


FIG. IIB

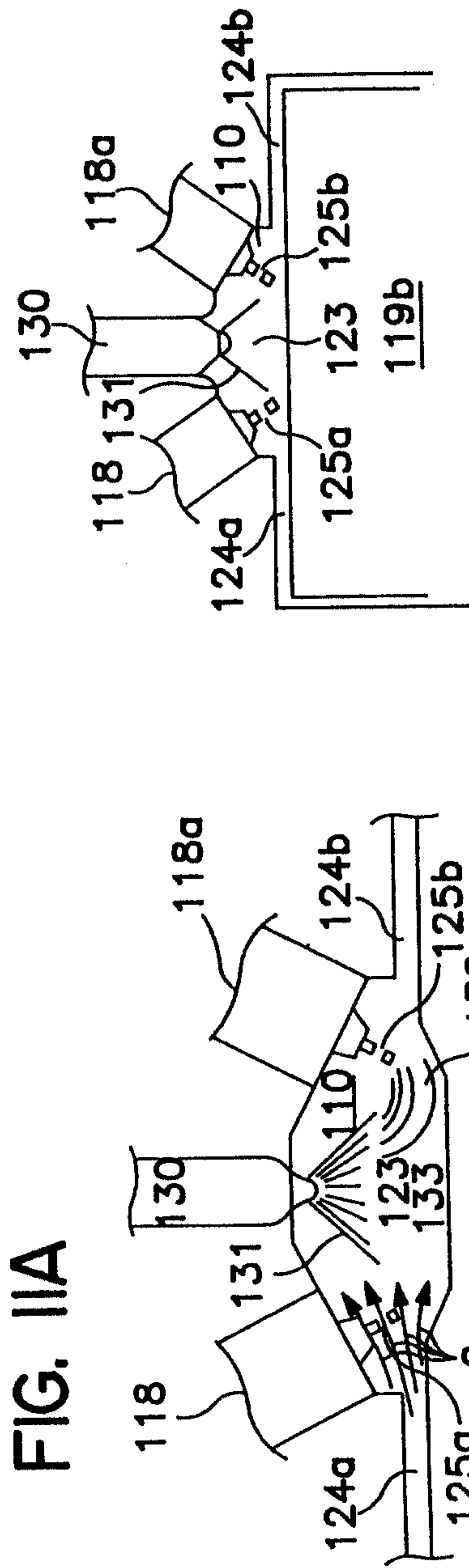


FIG. IIA

FIG. IIC

FIG. IID

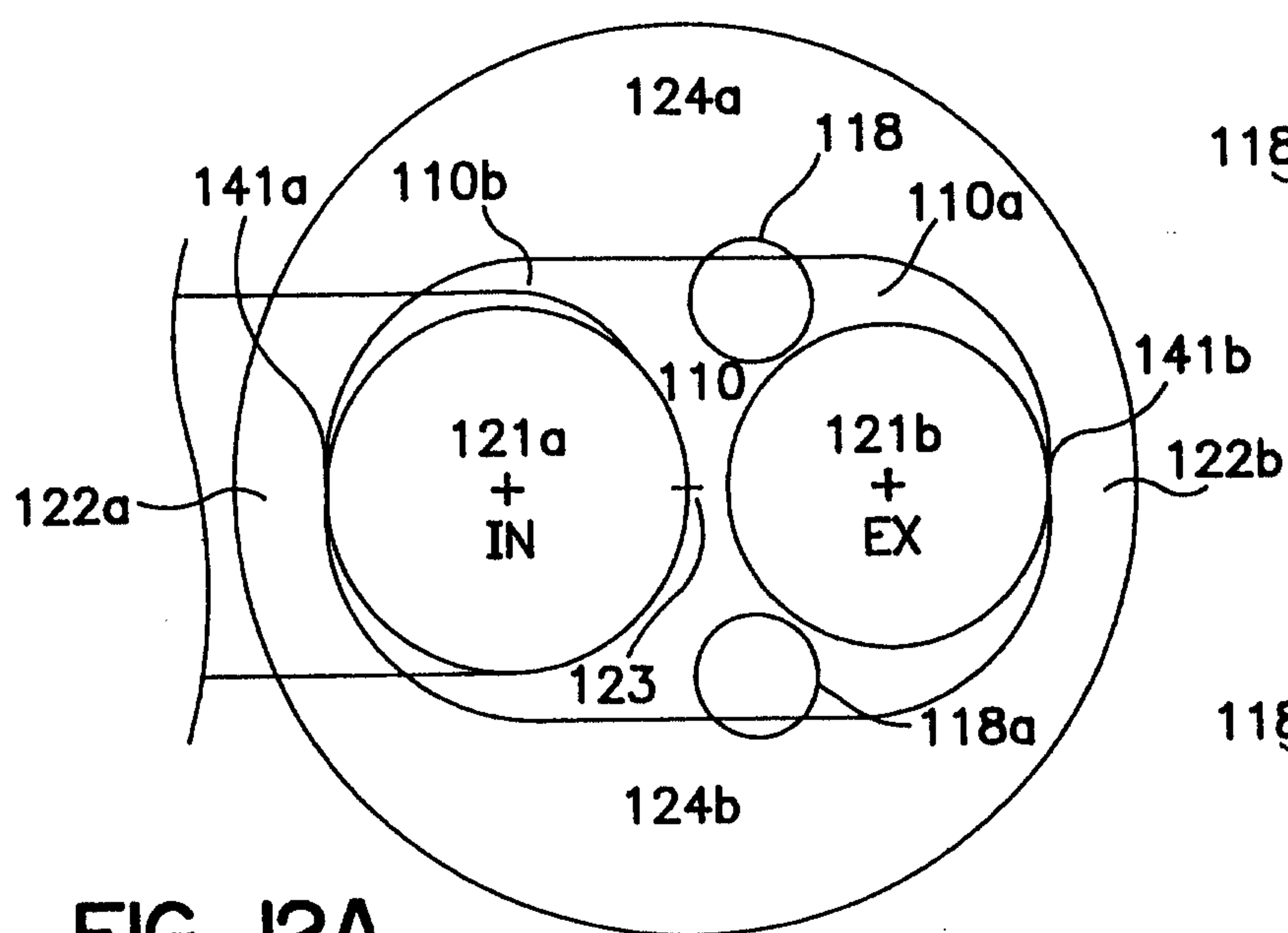


FIG. 12A

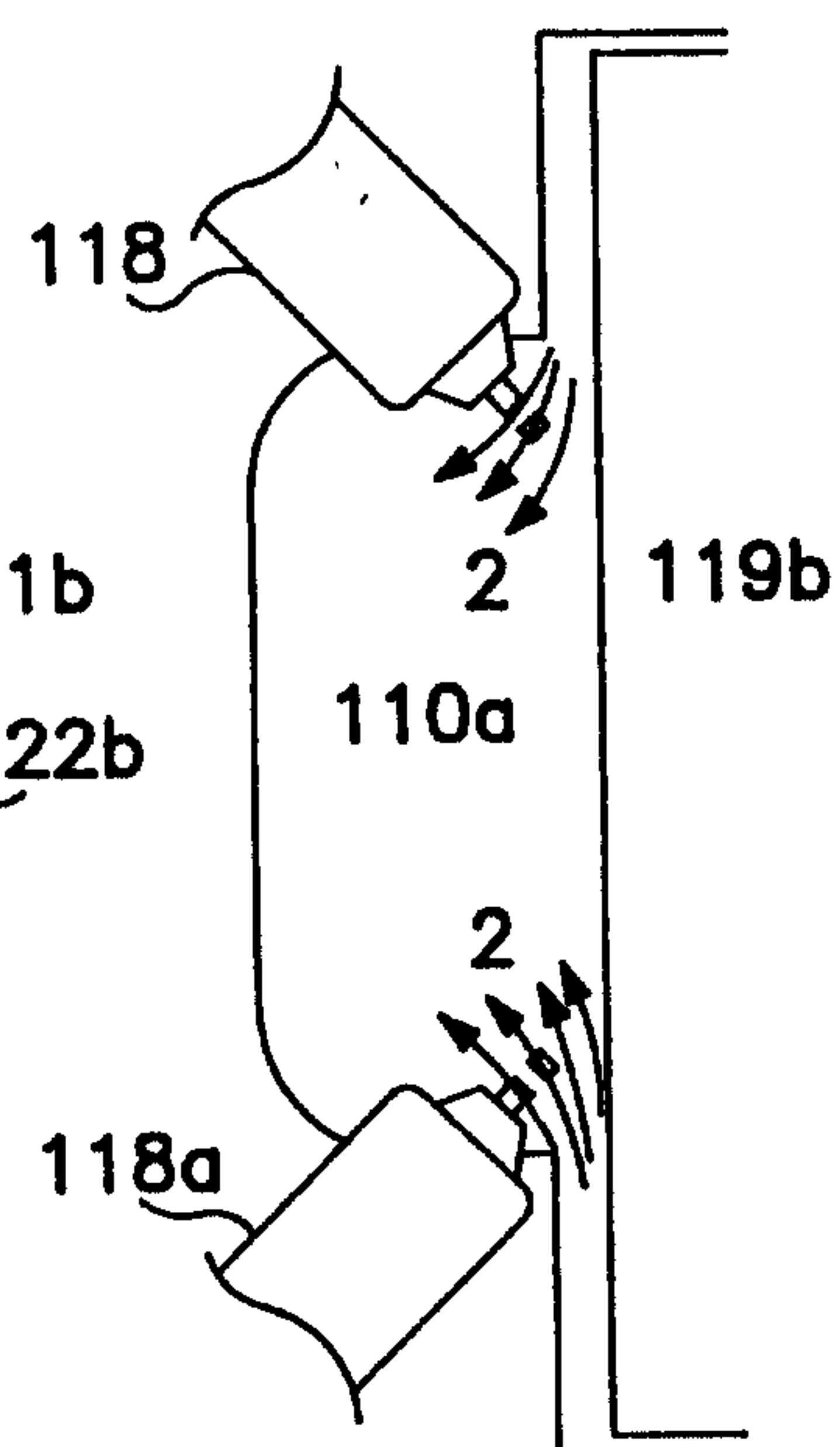


FIG. 12B

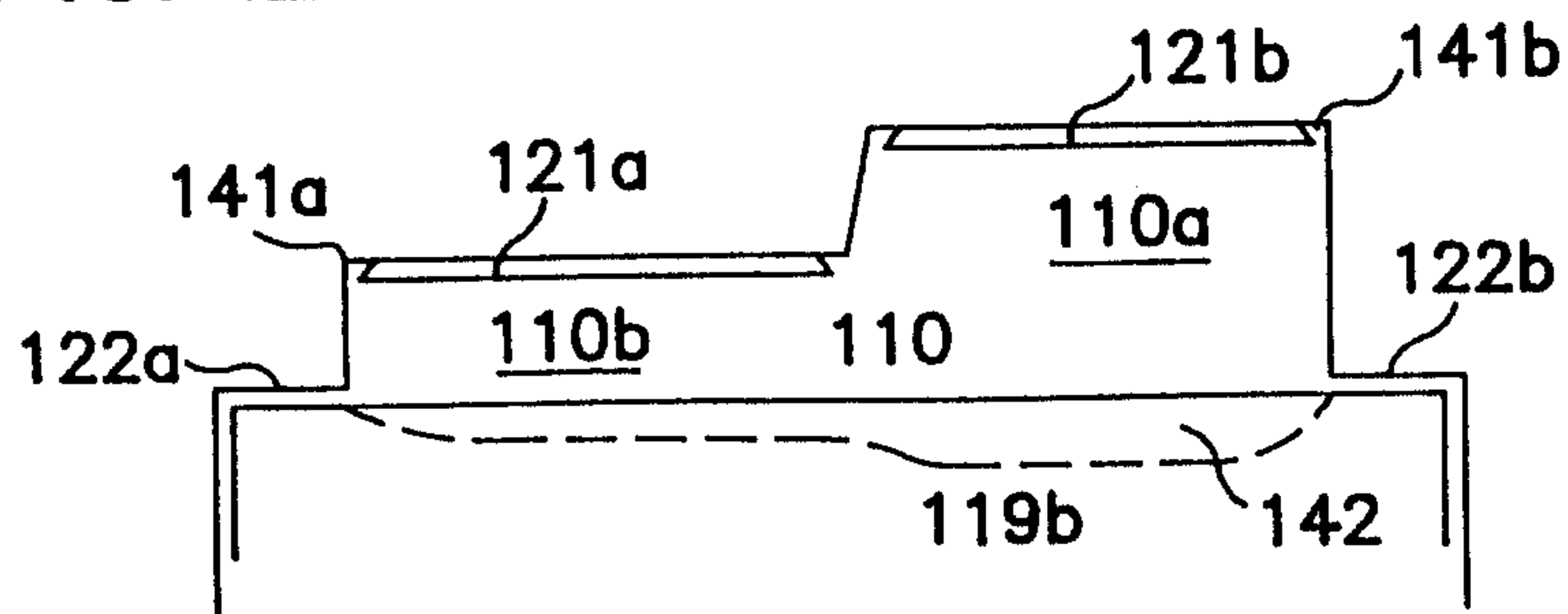


FIG. 12C

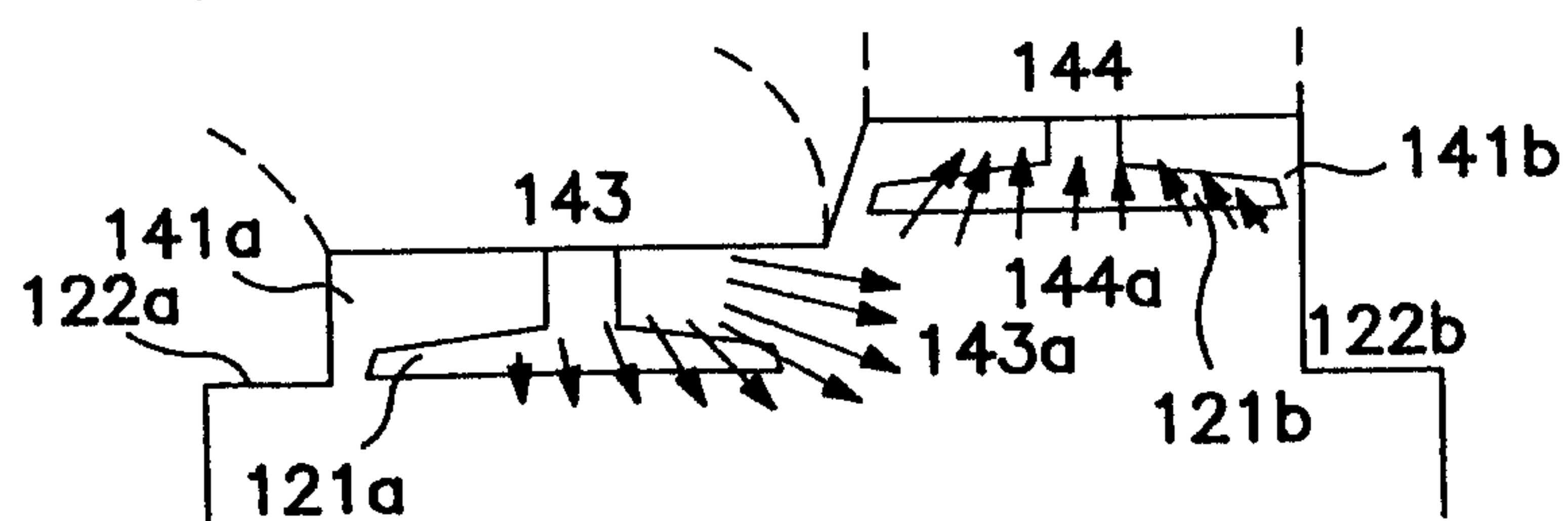


FIG. 12D

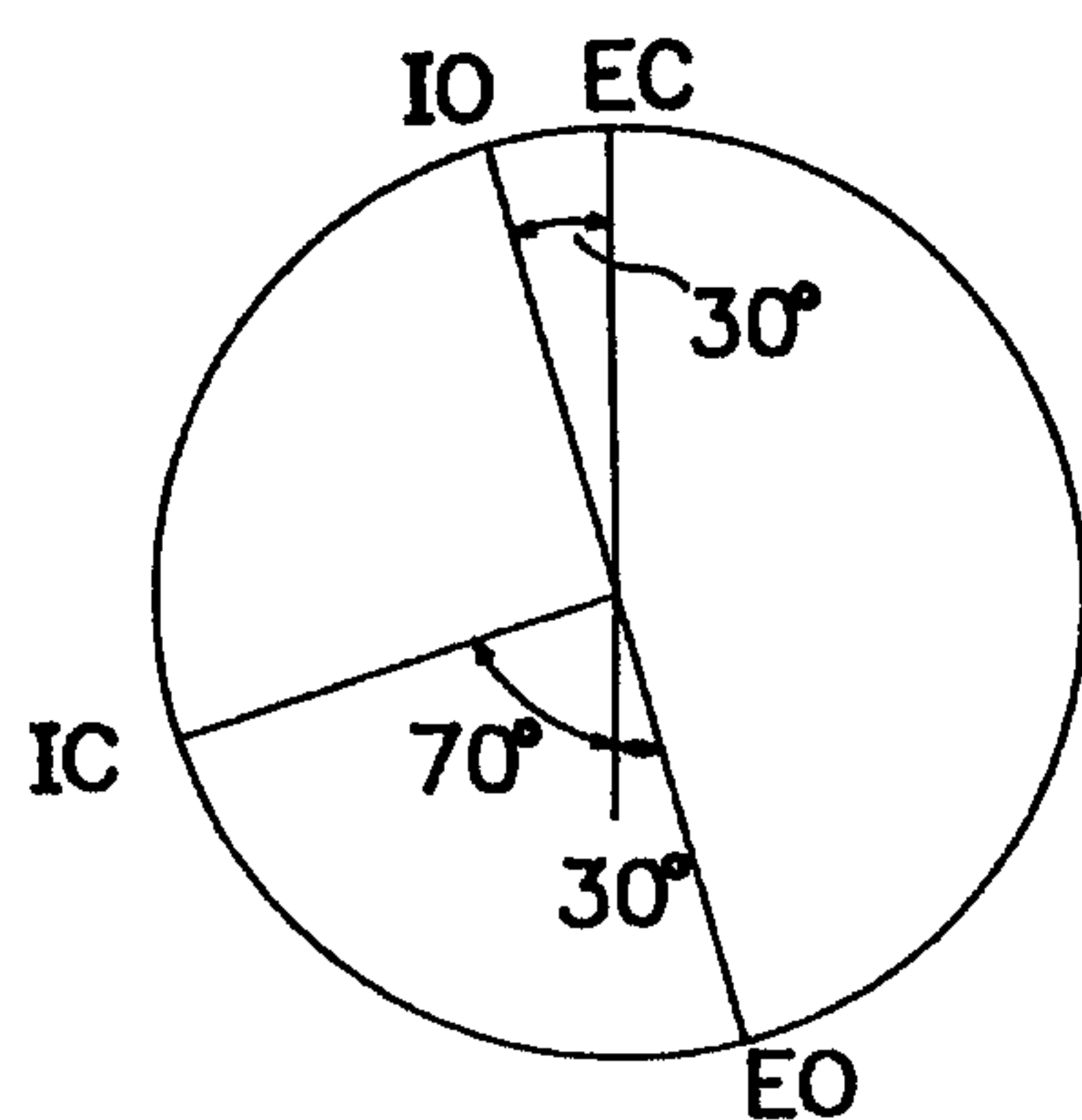


FIG. 13

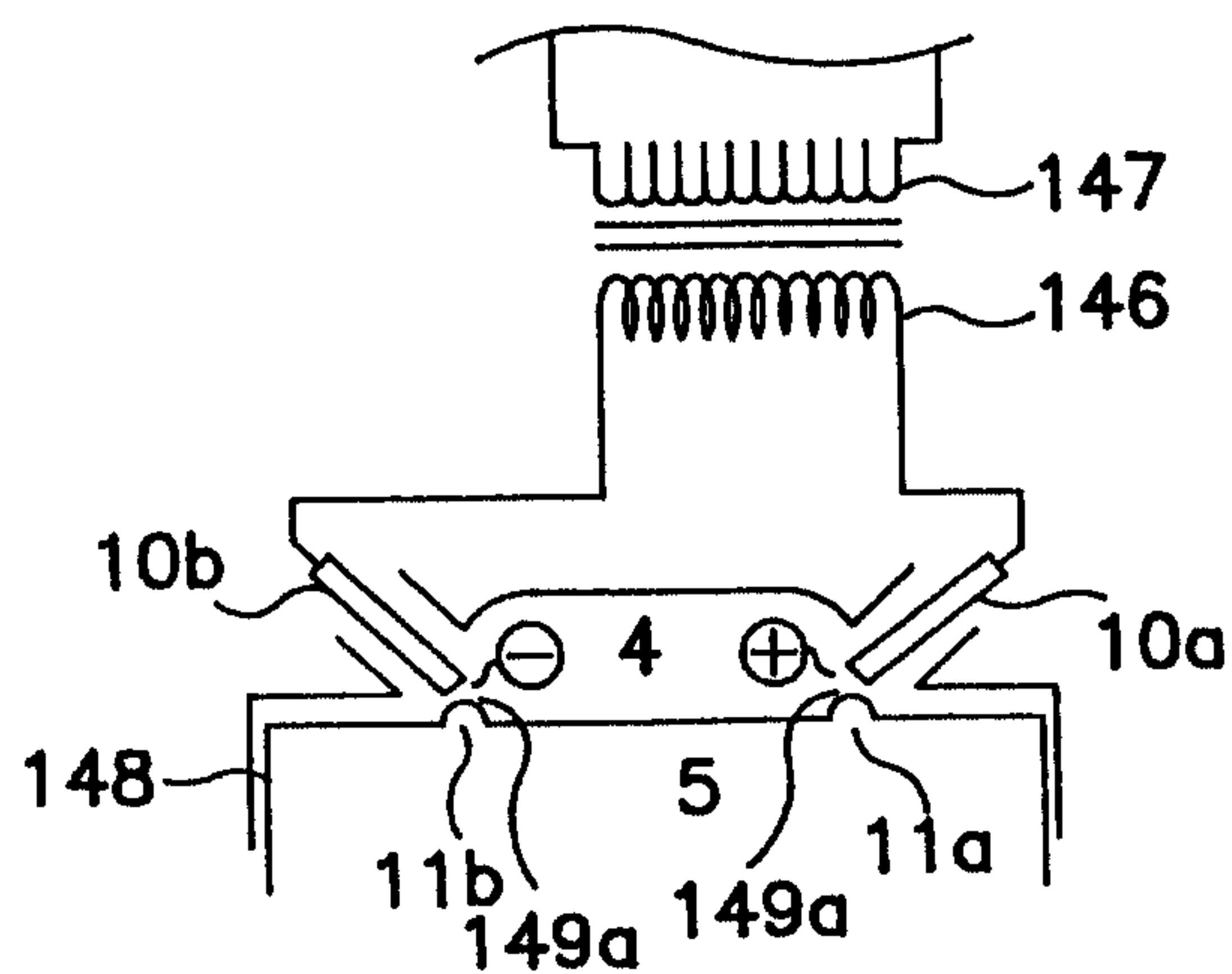
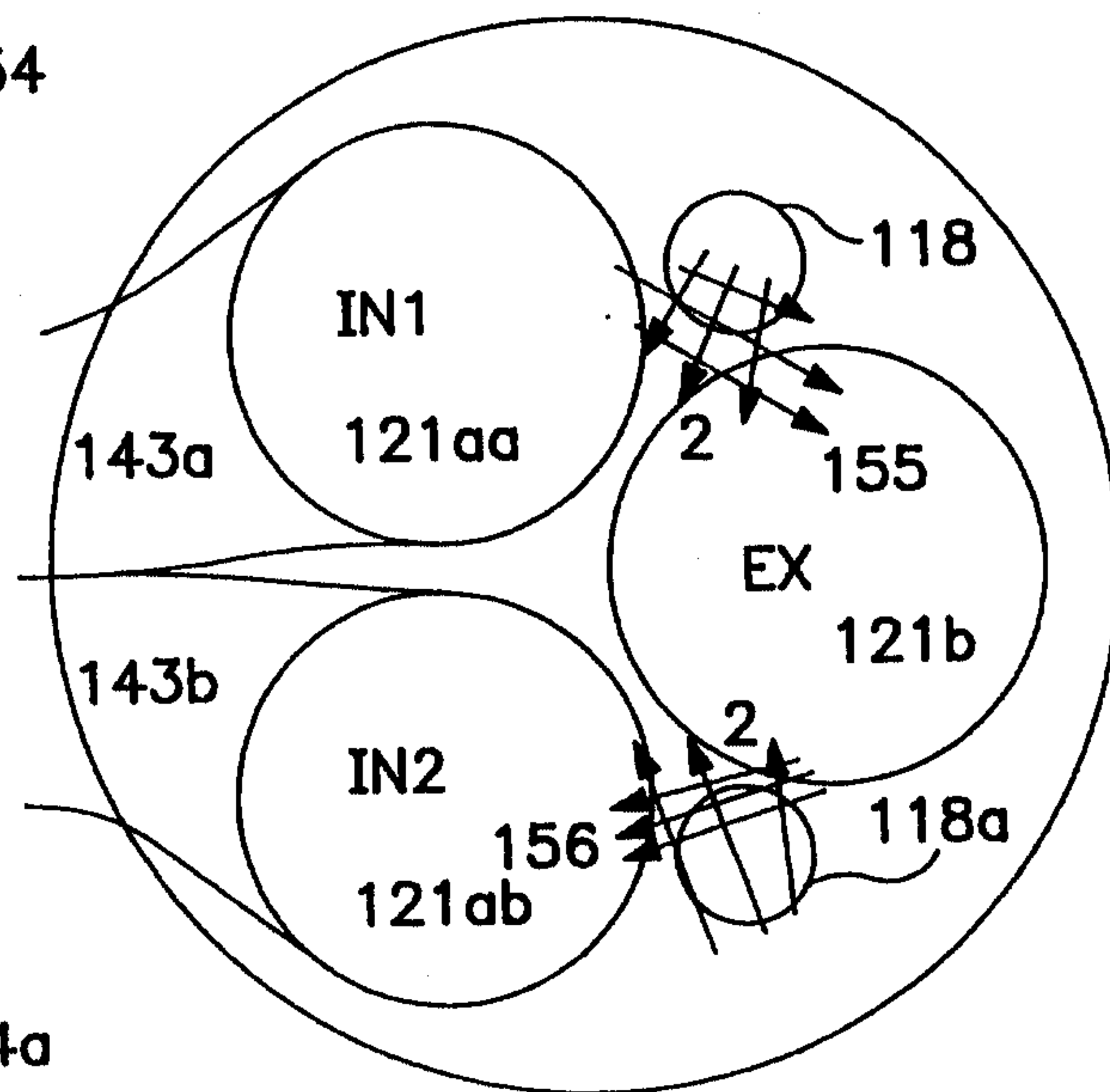
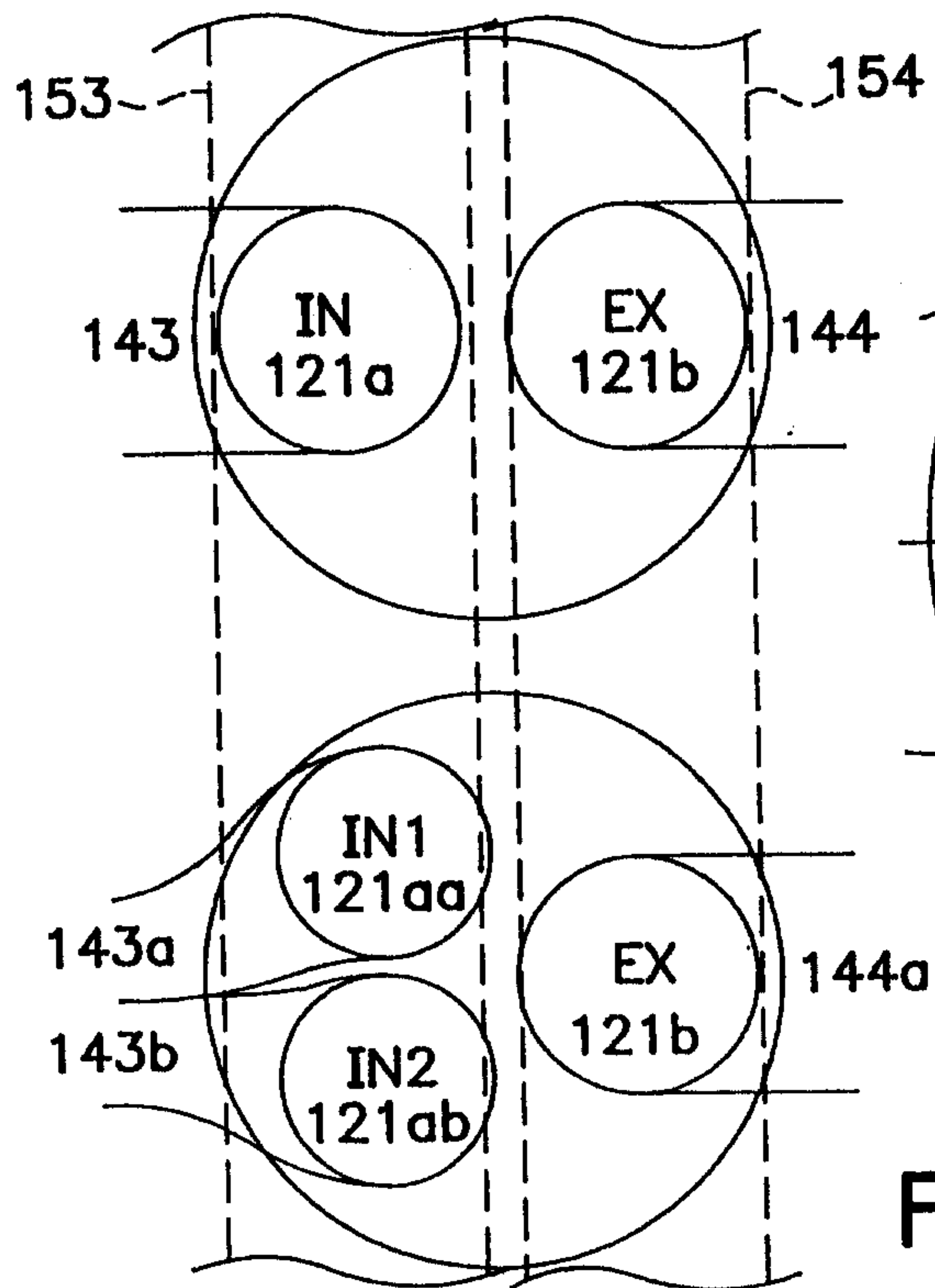
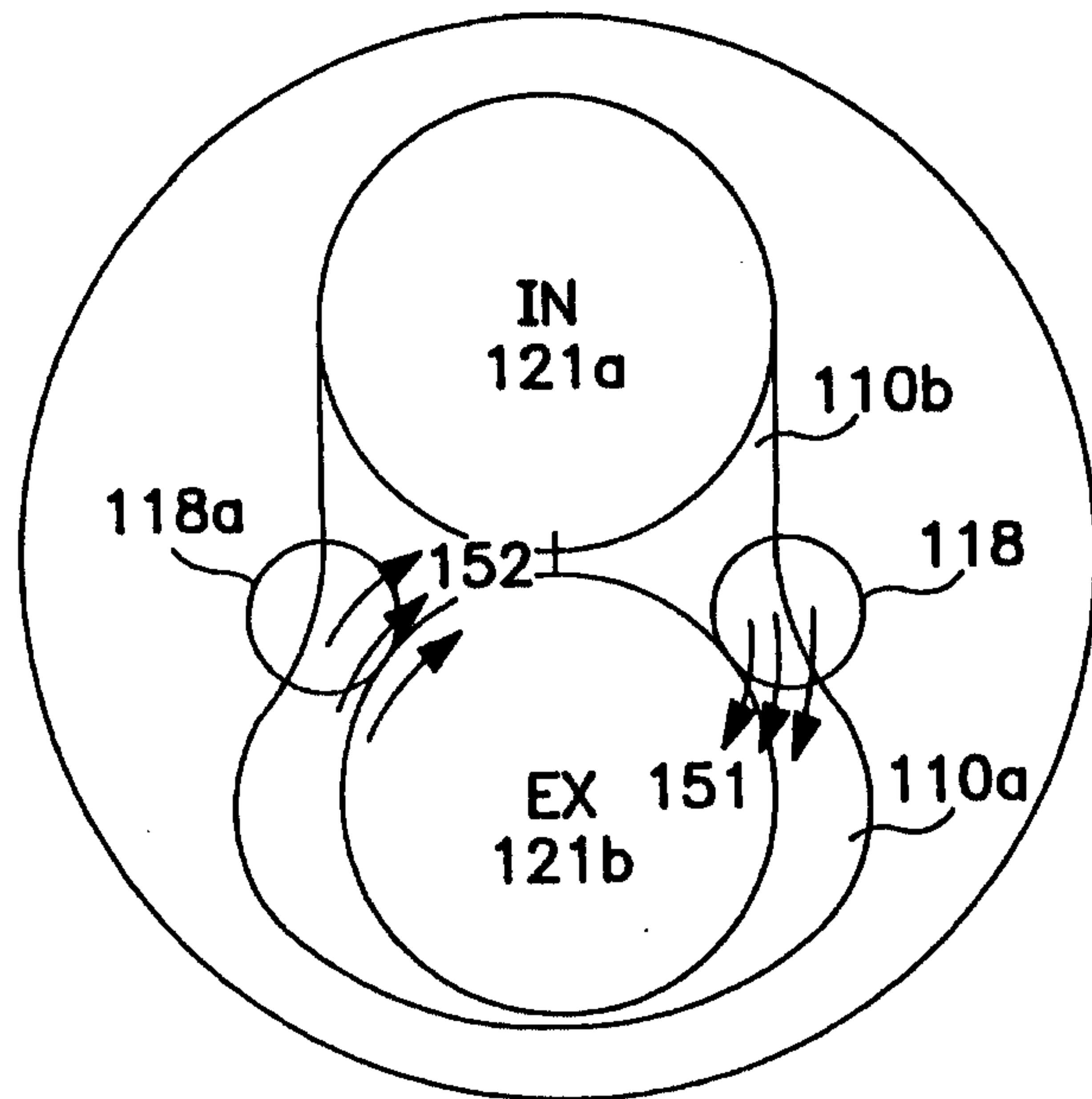
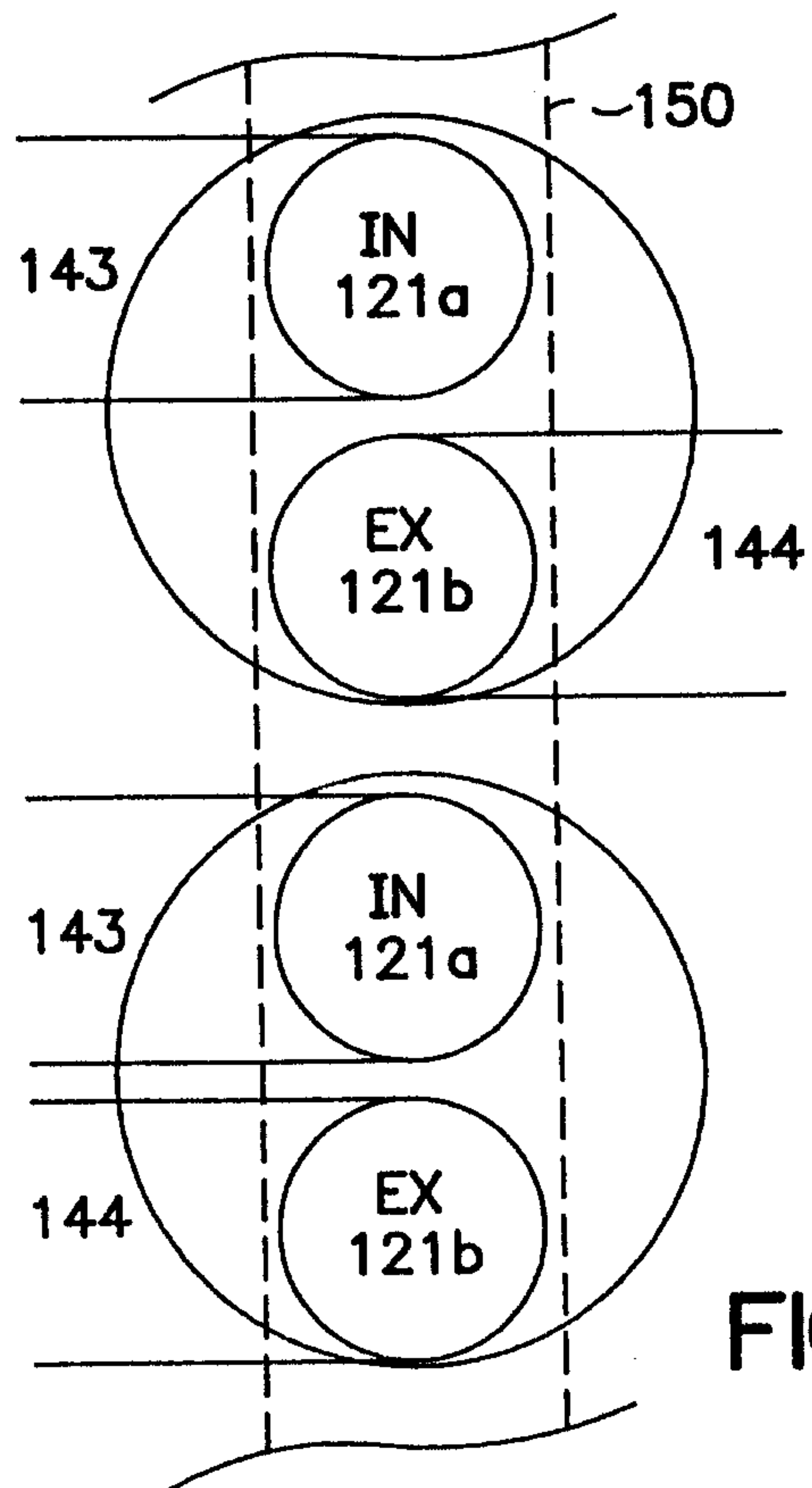


FIG. 14



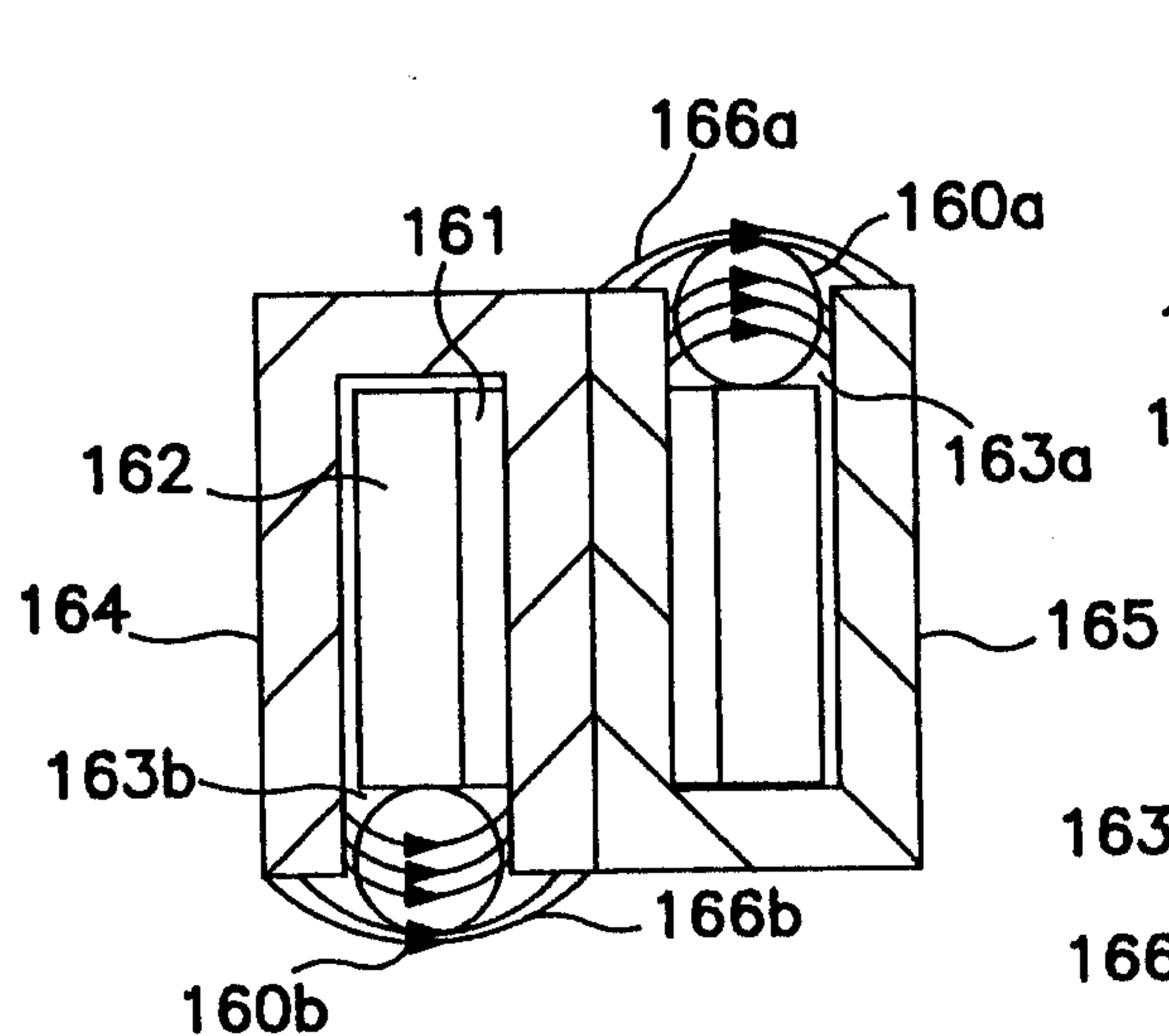


FIG. 17A

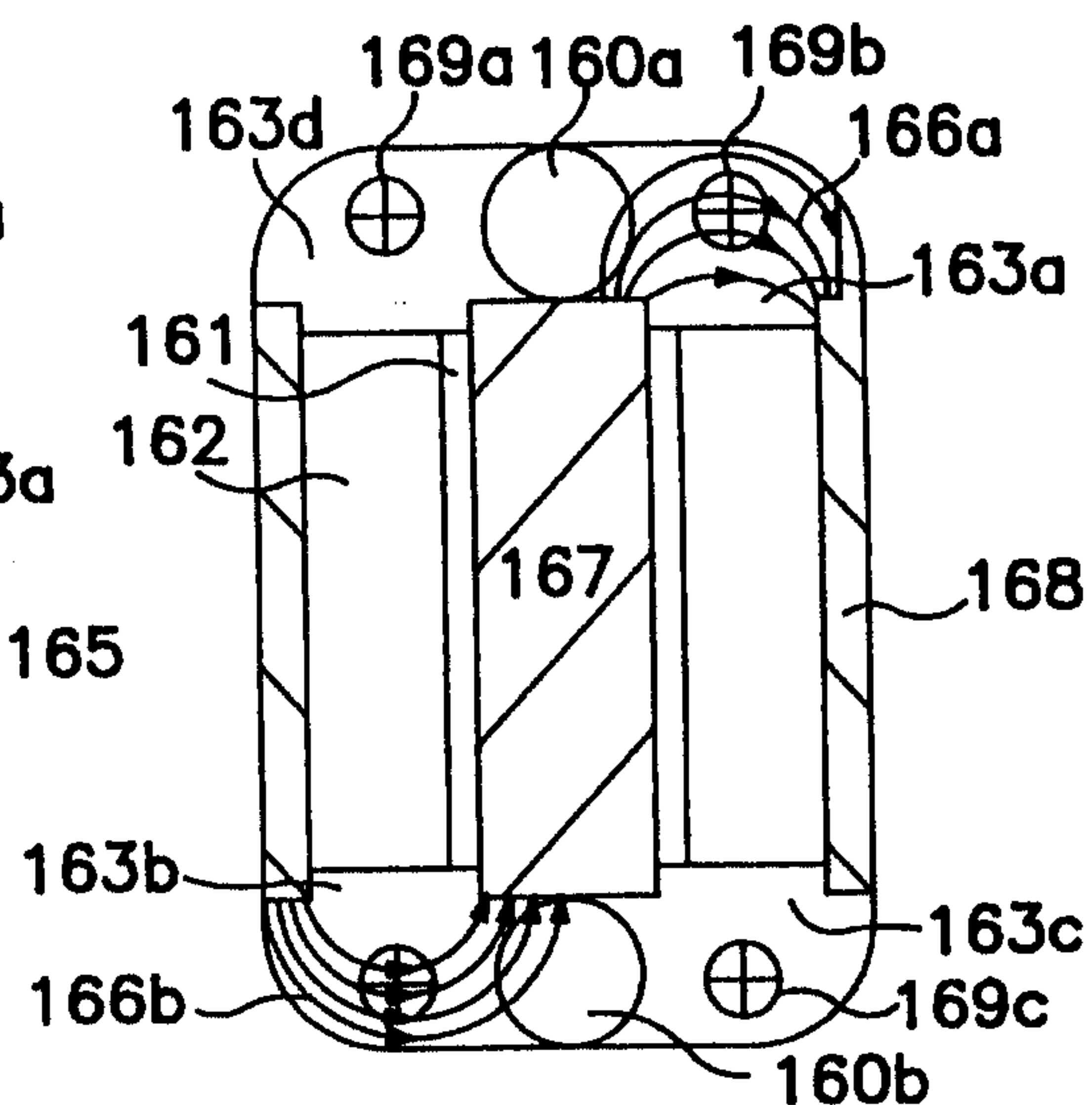


FIG. 17B

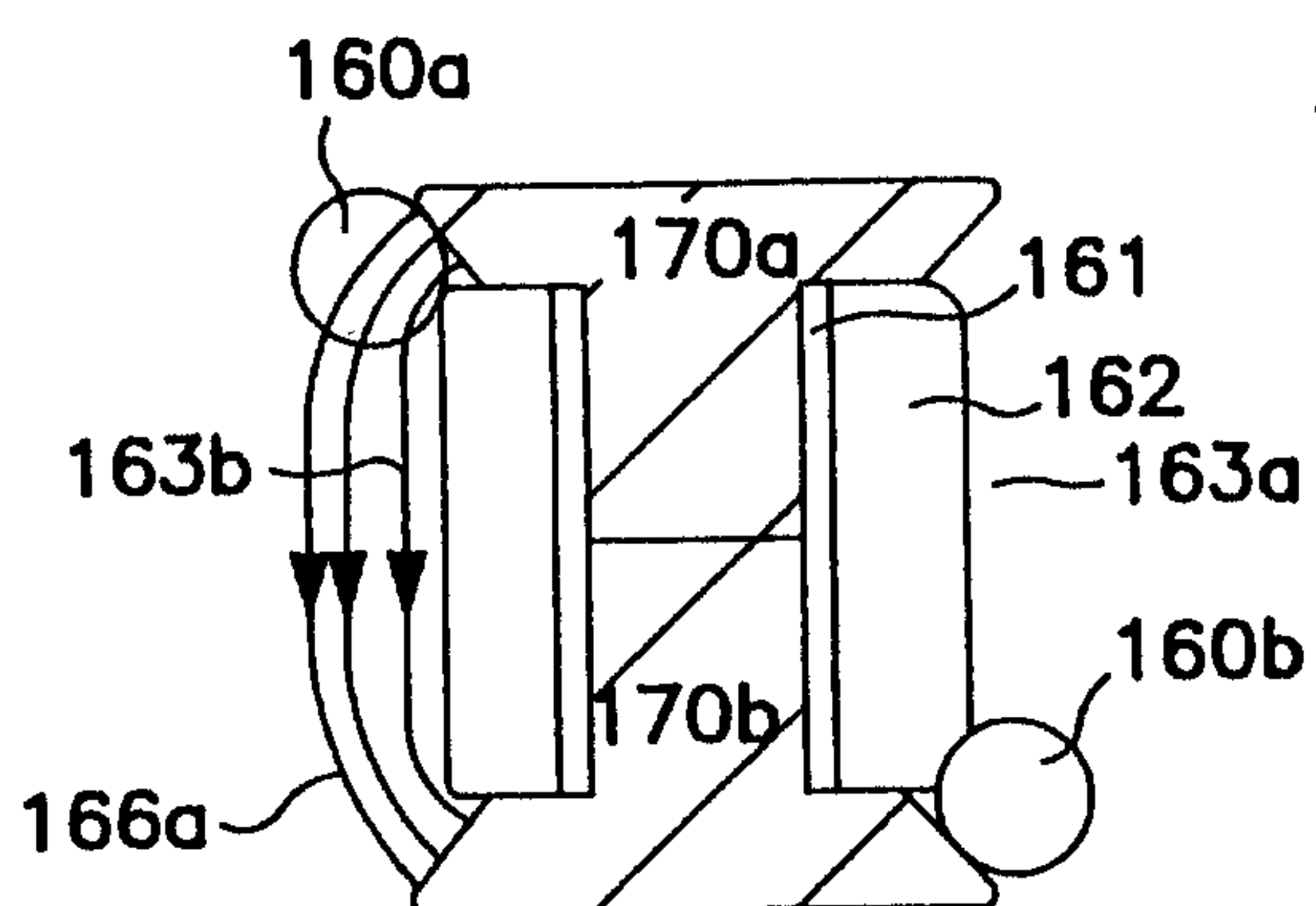


FIG. 17C

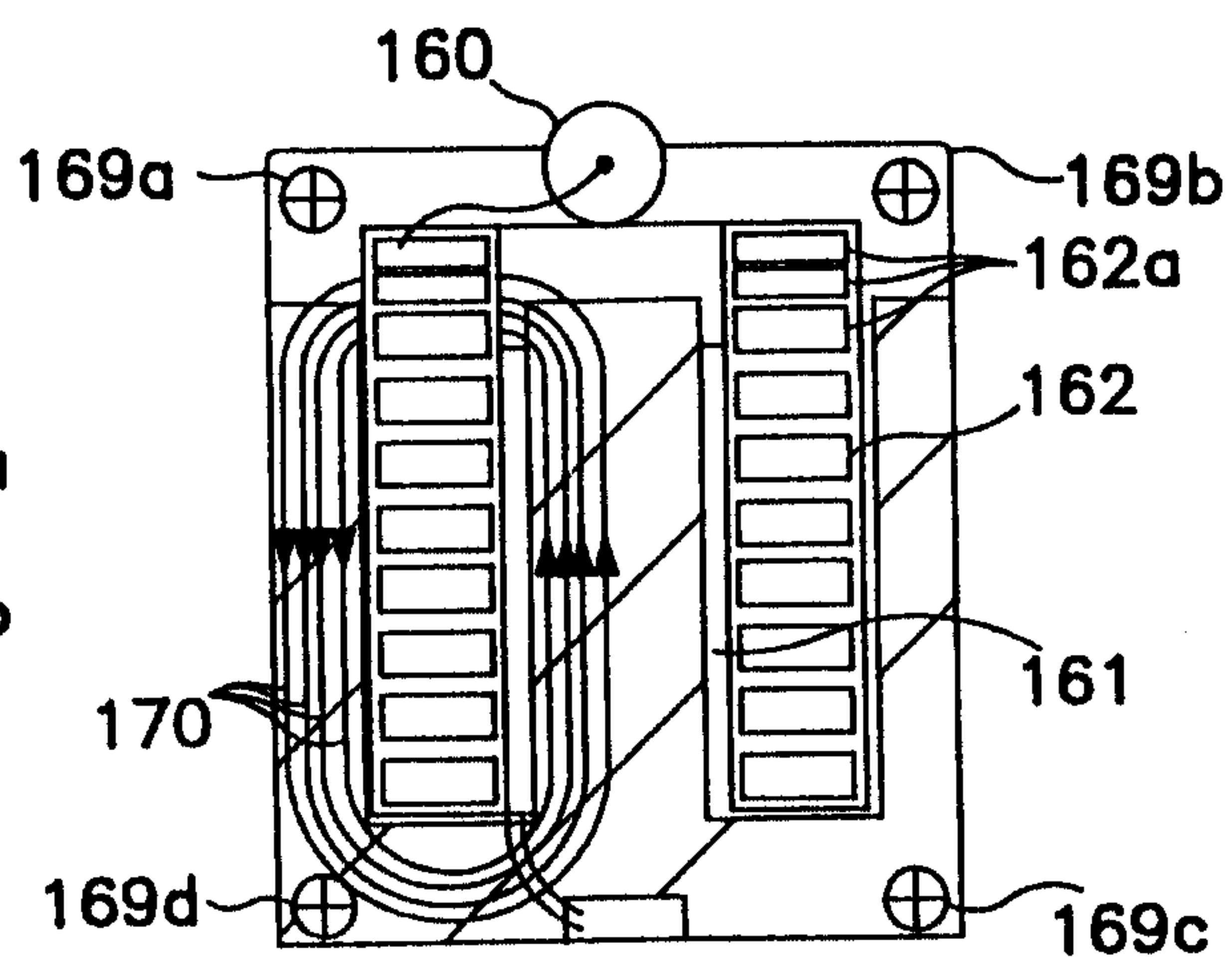


FIG. 18

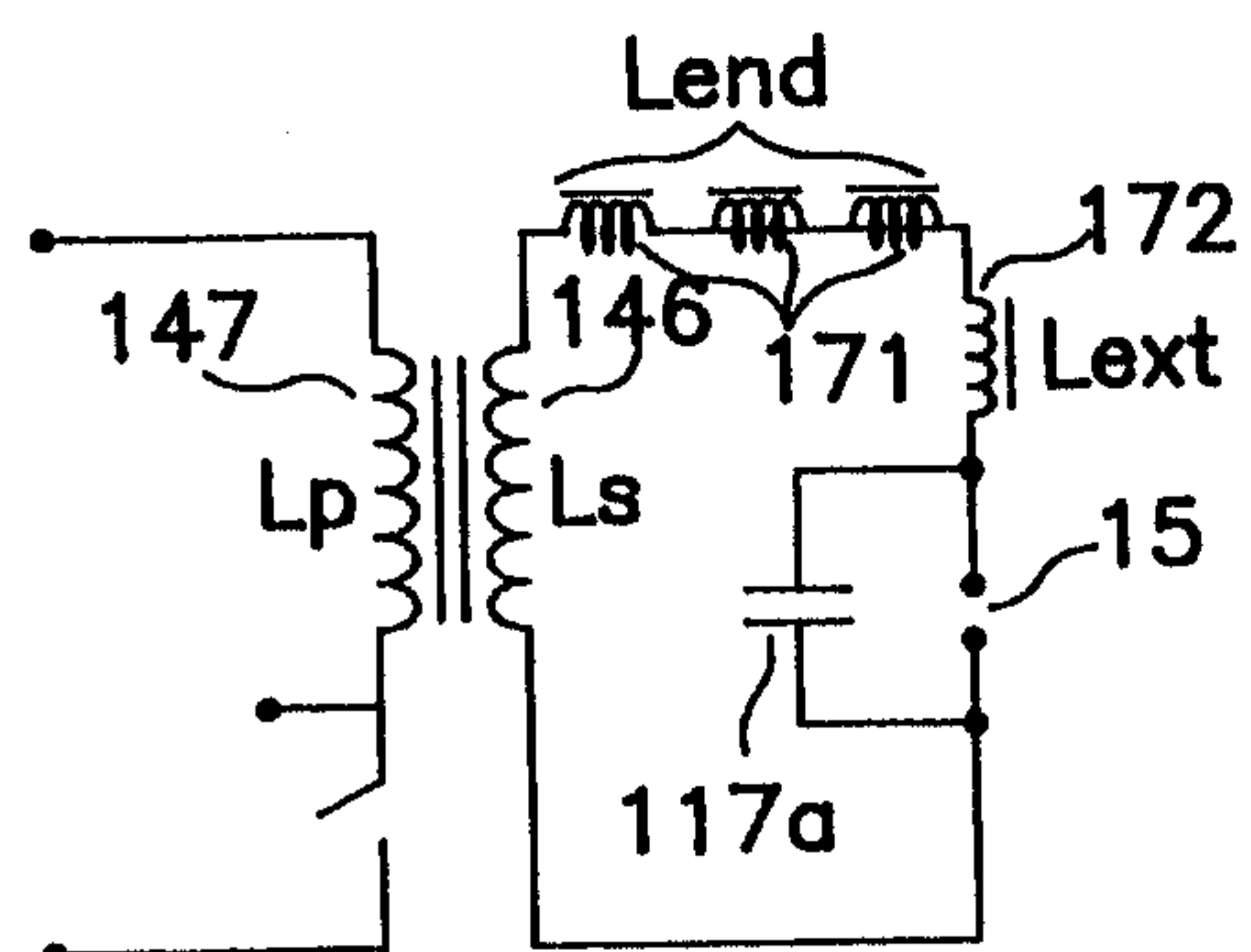


FIG. 18A

