United States Patent [19]

Young et al.

[11] **3,884,599**

[45] **May 20, 1975**

[54] **SCROLL-TYPE POSITIVE FLUID DISPLACEMENT APPARATUS**

[75] Inventors: **Niels O. Young**, Mason, N.H.; **John E. McCullough**, Carlisle, Mass.

[73] Assignee: **Arthur D. Little, Inc.**, Cambridge, Mass.

[22] Filed: **June 11, 1973**

[21] Appl. No.: **368,907**

[52] **U.S. Cl.** **418/55**

[51] **Int. Cl.** **F01c 1/02**

[58] **Field of Search** 418/55, 56, 57, 131

[56] **References Cited**

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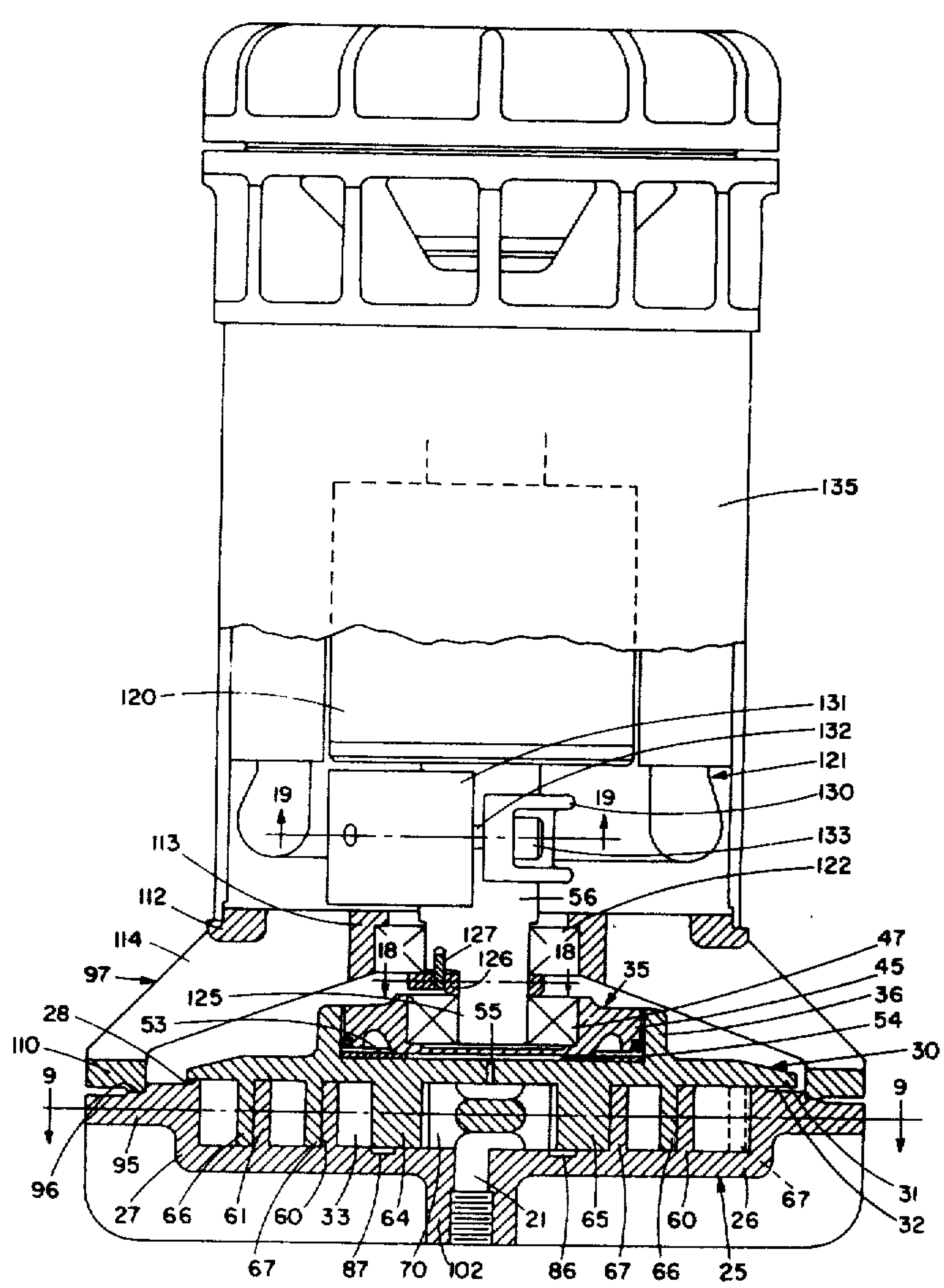
Primary Examiner—C. J. Husar
Attorney, Agent, or Firm—Bessie A. Lepper

[57] **ABSTRACT**

A positive fluid displacement apparatus employing scroll members having interfitting spiroidal wraps angularly and radially offset such that as the spiral centers experience an orbiting motion, they define one or more moving fluid pockets of variable volume. The zones of lowest and highest pressures are connected to fluid ports. Radial sealing is accomplished with minimum wear by using a driving mechanism which provides a centripetal radial force adapted to oppose a fraction of the centrifugal force acting on the orbiting scroll member. The line contact sealing force between the wraps of the scroll members constitutes the sole radial constraining force. Coupling means which are separate from the driving means, and hence from the radial constraining means, are provided to maintain the desired angular relationship between scroll members. Axial sealing is attained by withdrawing a portion of fluid from the zone of highest pressure and using this high-pressure fluid to generate axial sealing forces. The apparatus may serve as a compressor, expander or pump.

58 Claims, 41 Drawing Figures



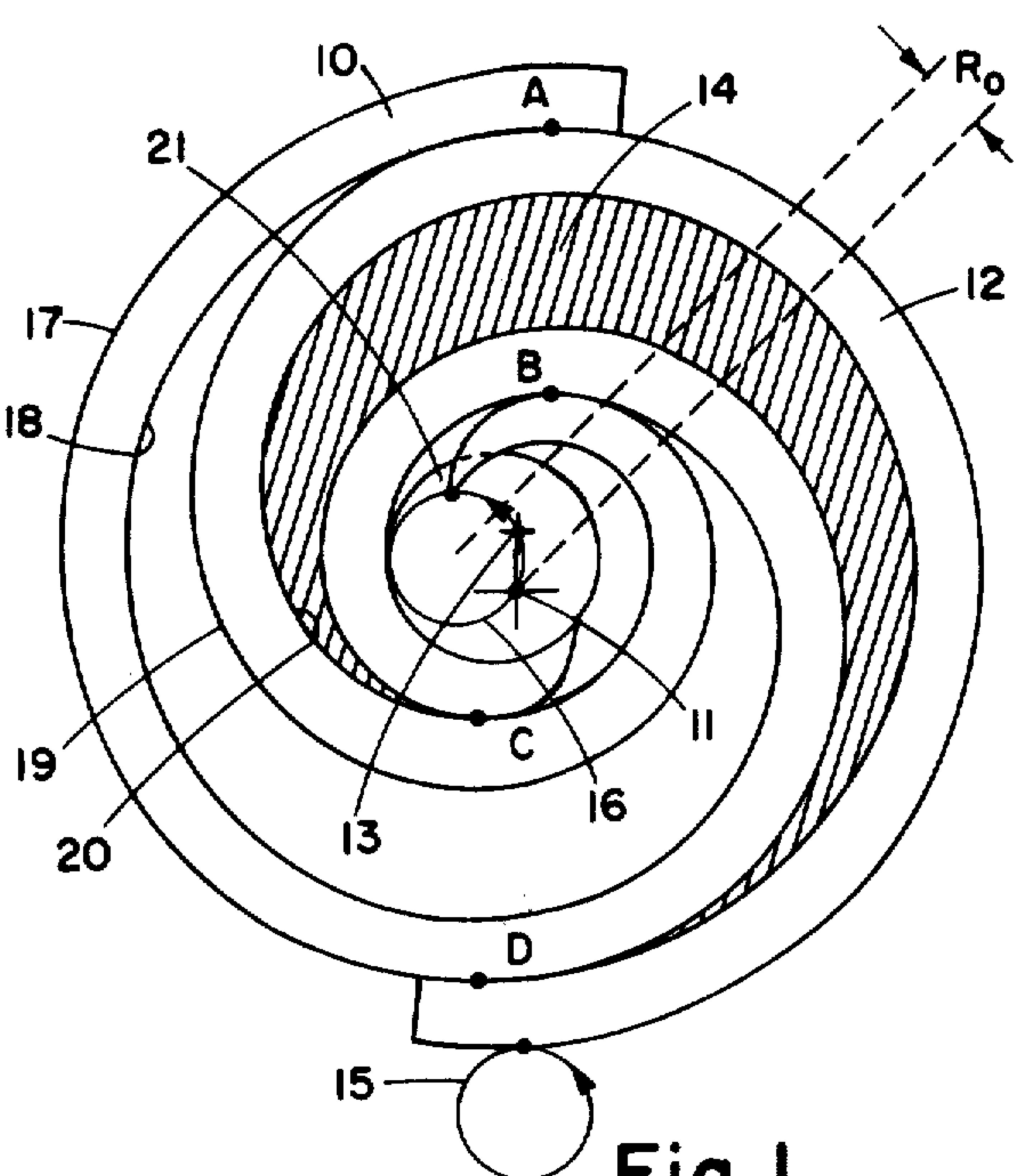


Fig. 1

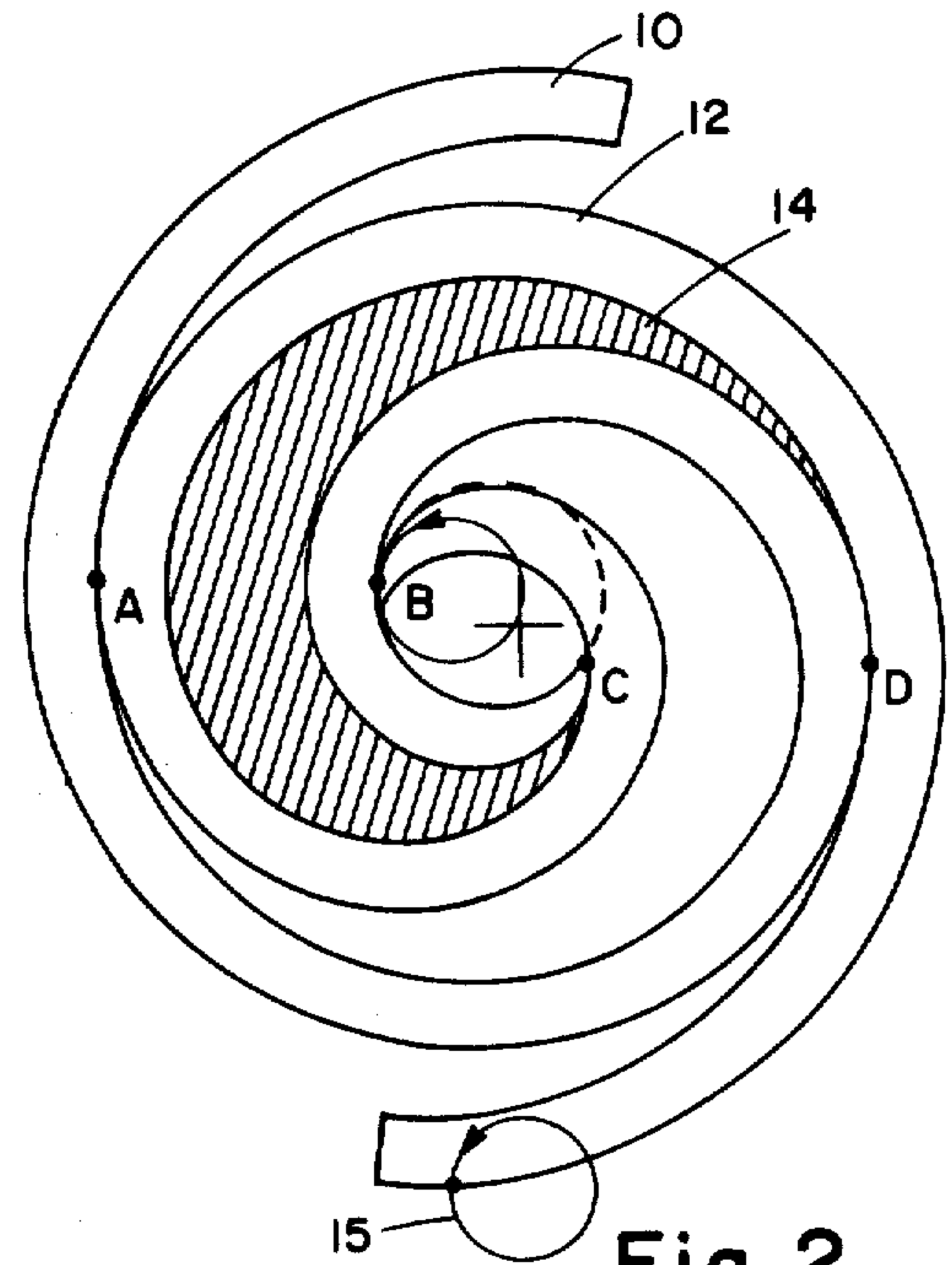


Fig. 2

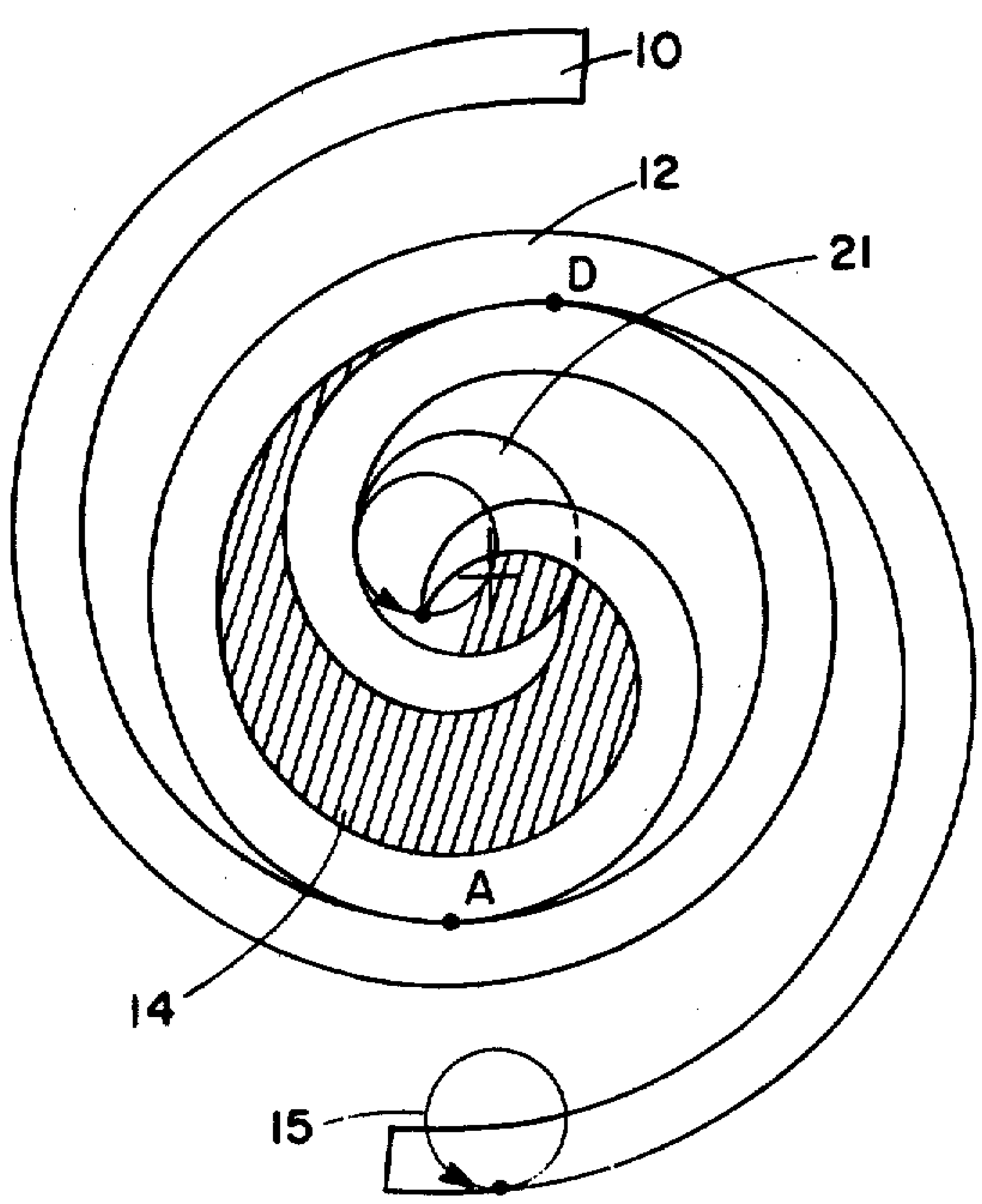
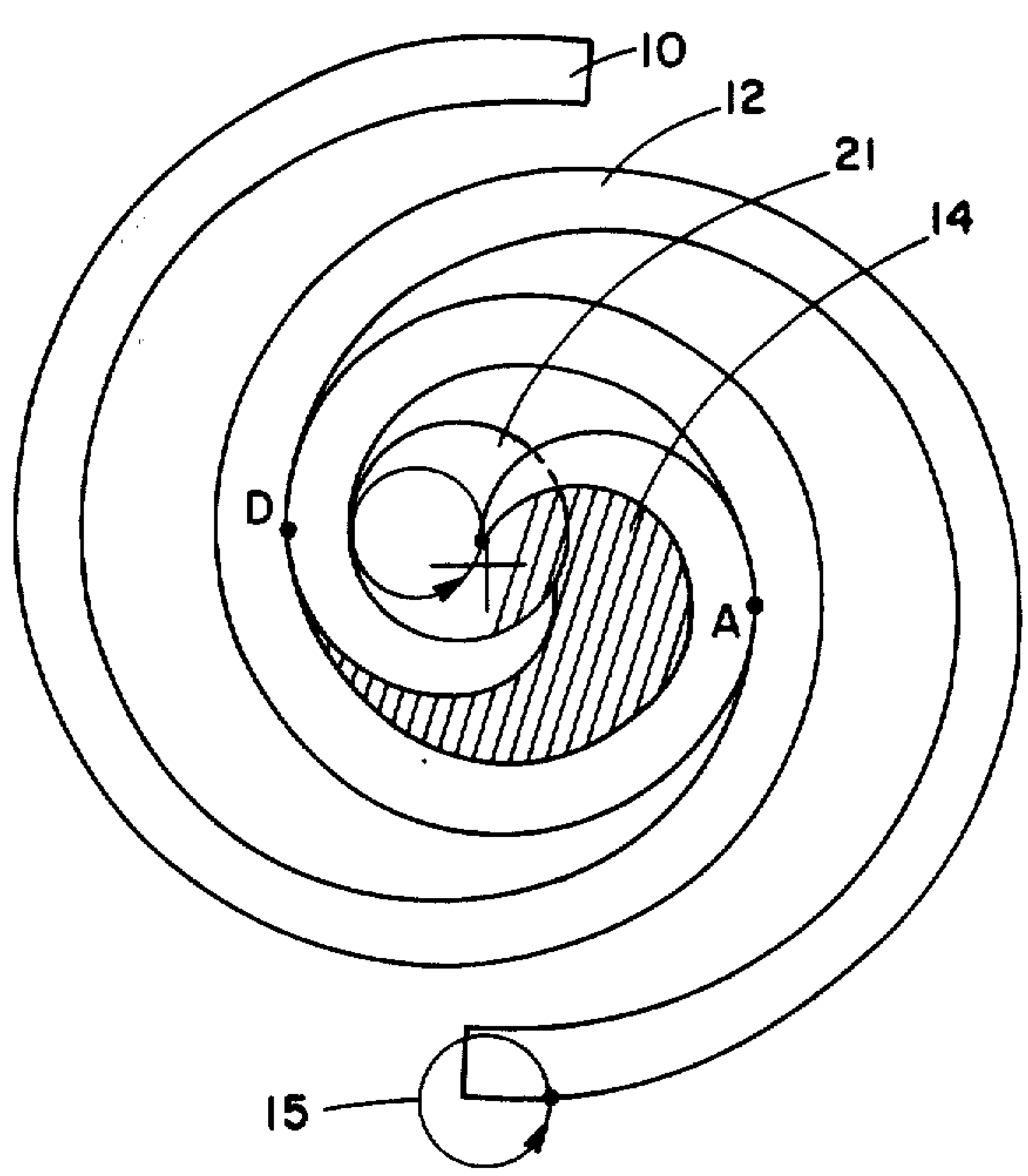


Fig. 3



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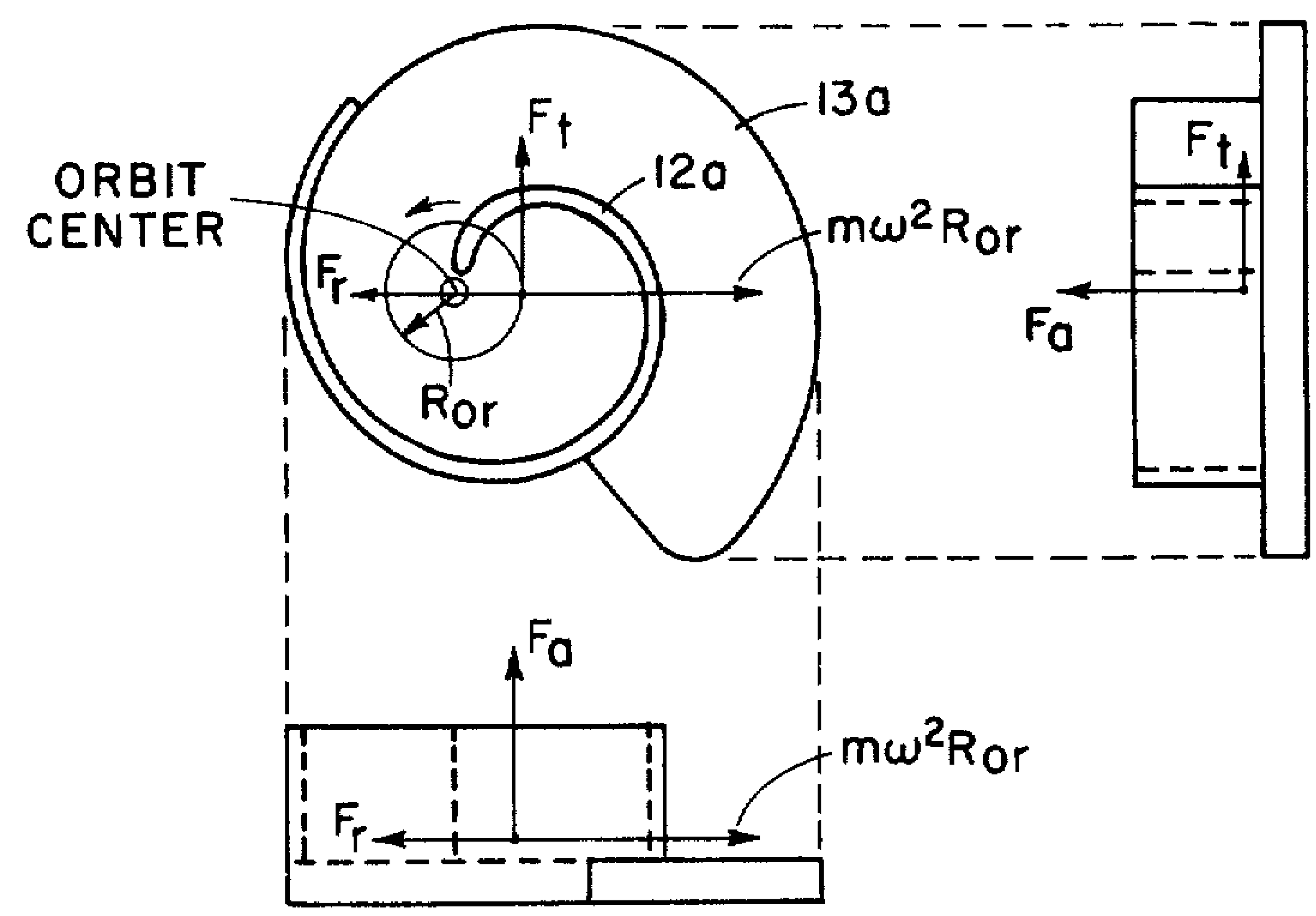

Fig. 5

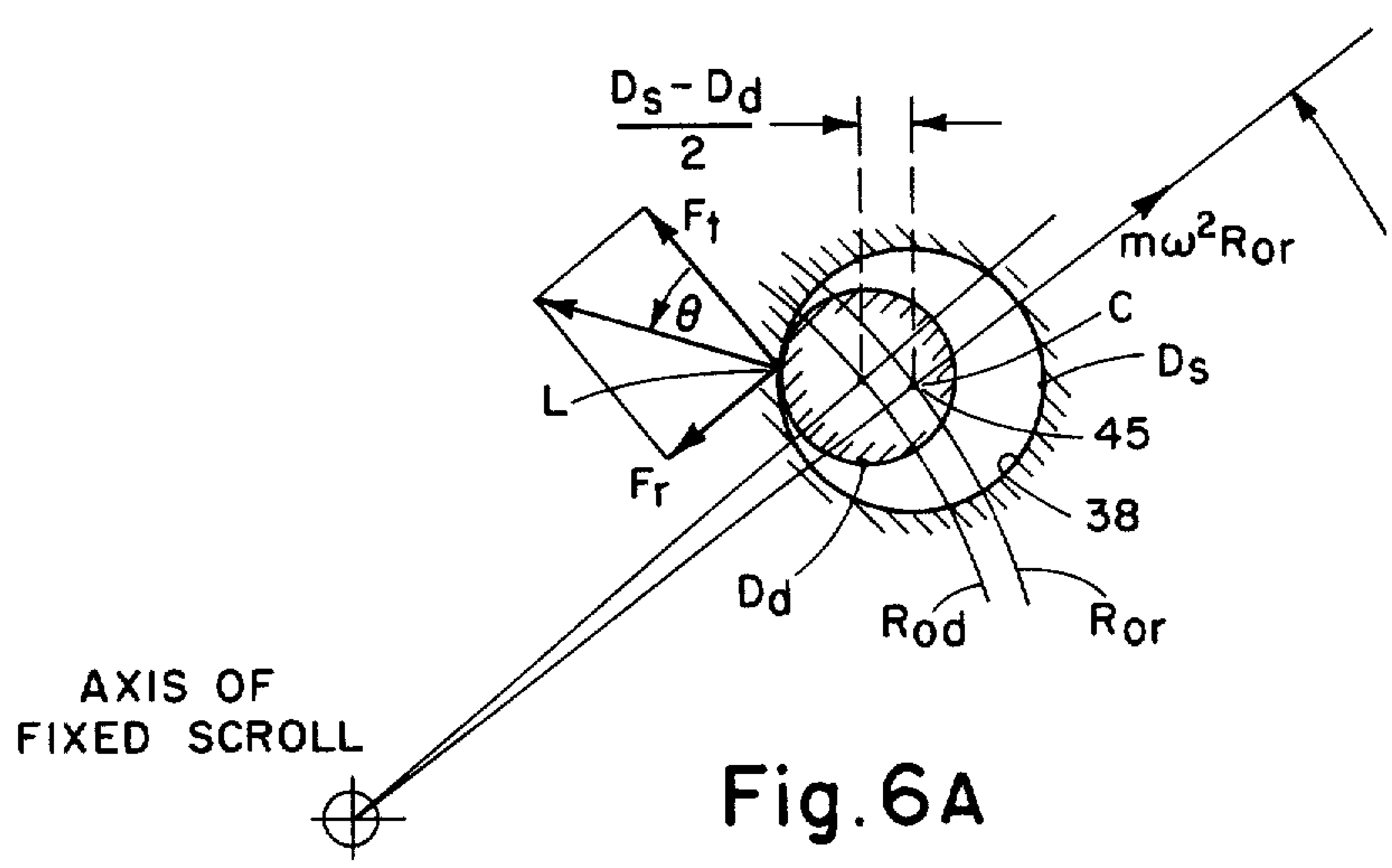

Fig. 6A

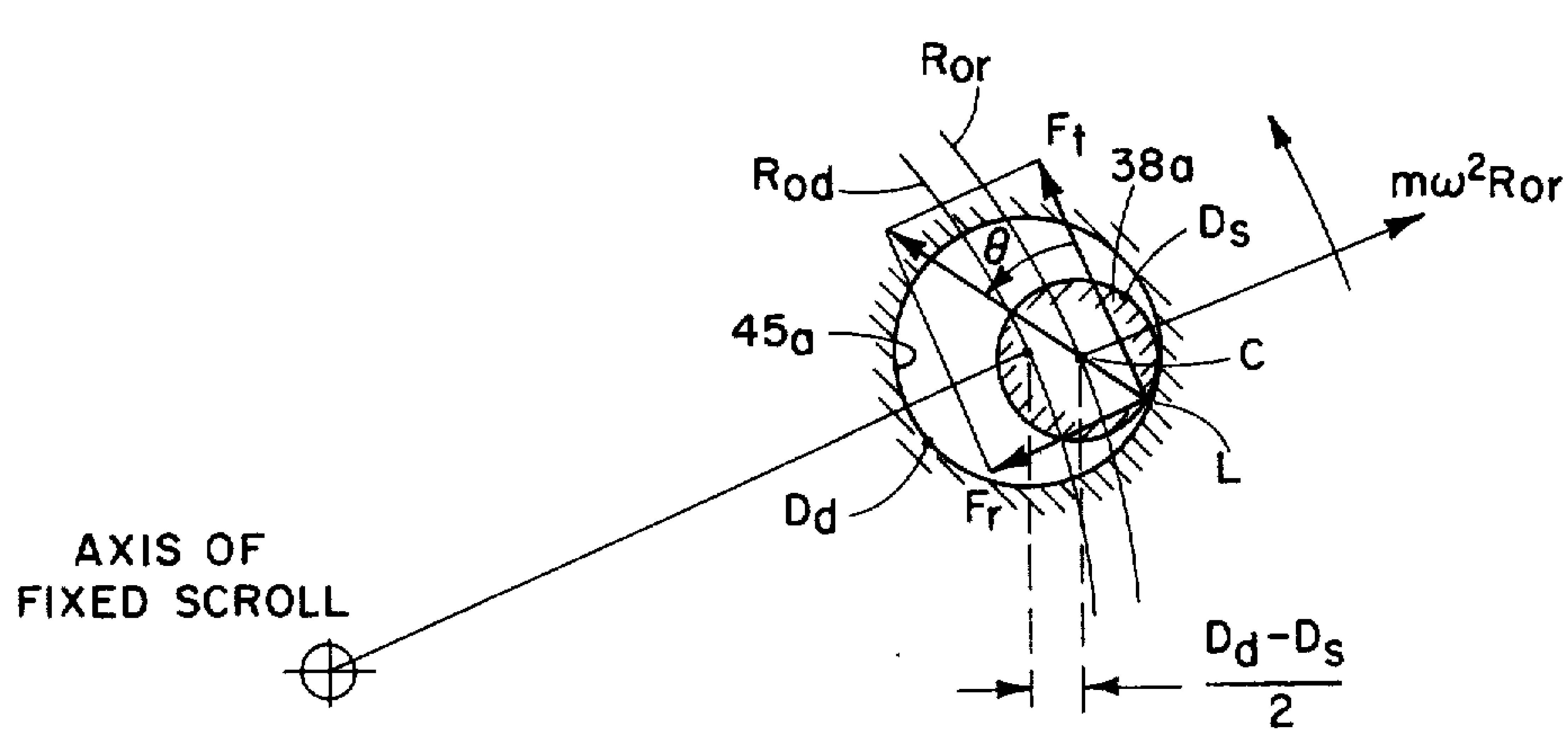

Fig. 6B

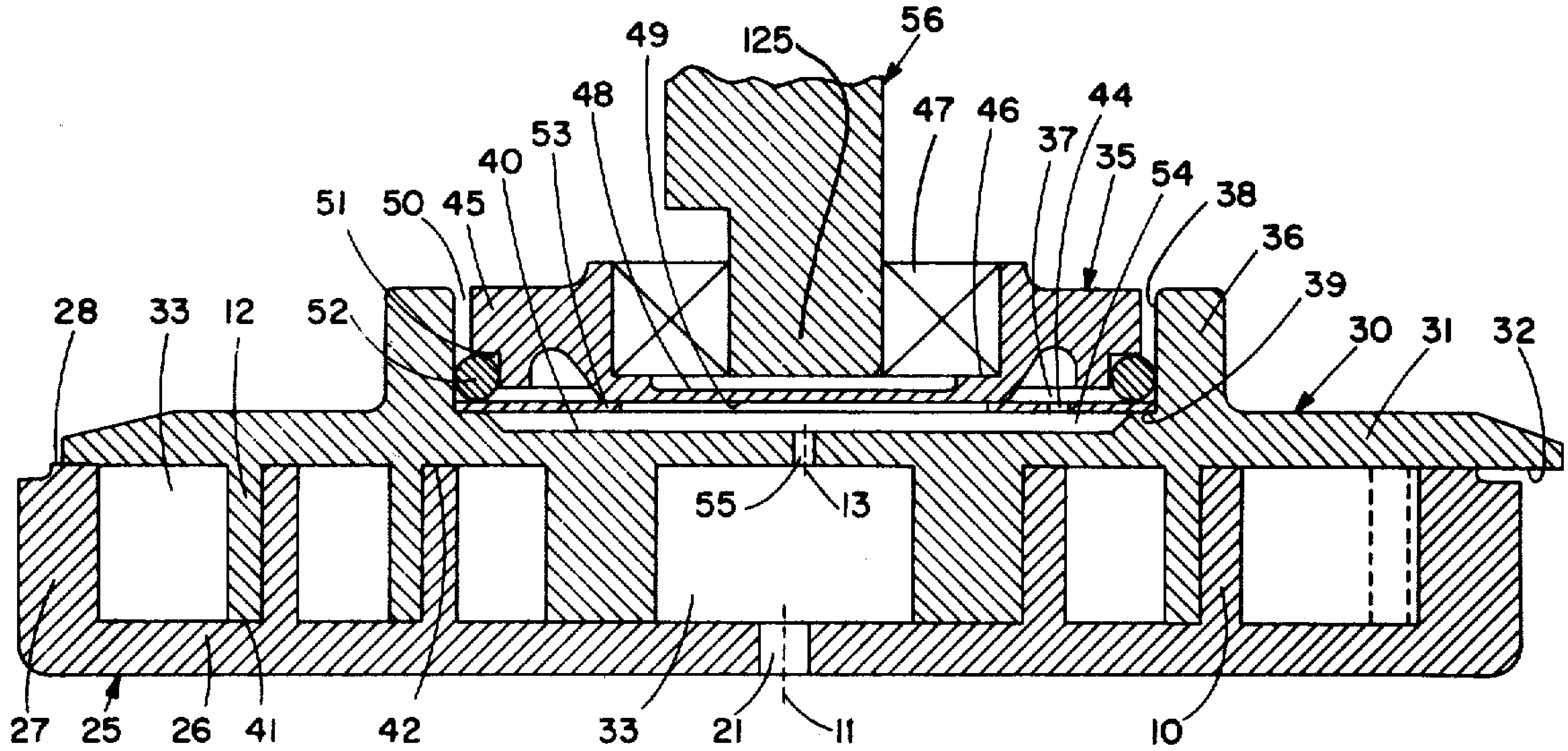
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Fig. 7

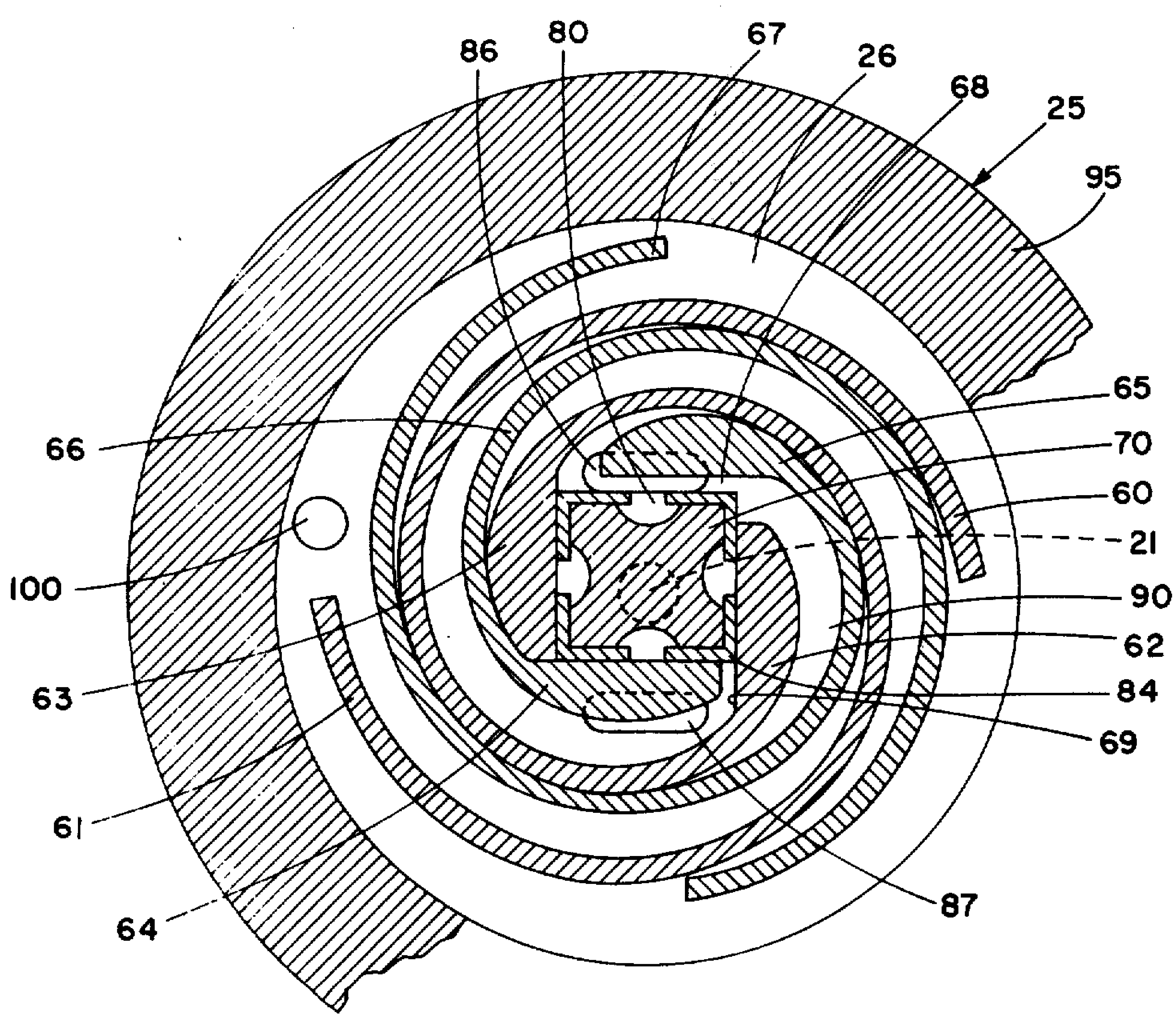
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Fig. 9

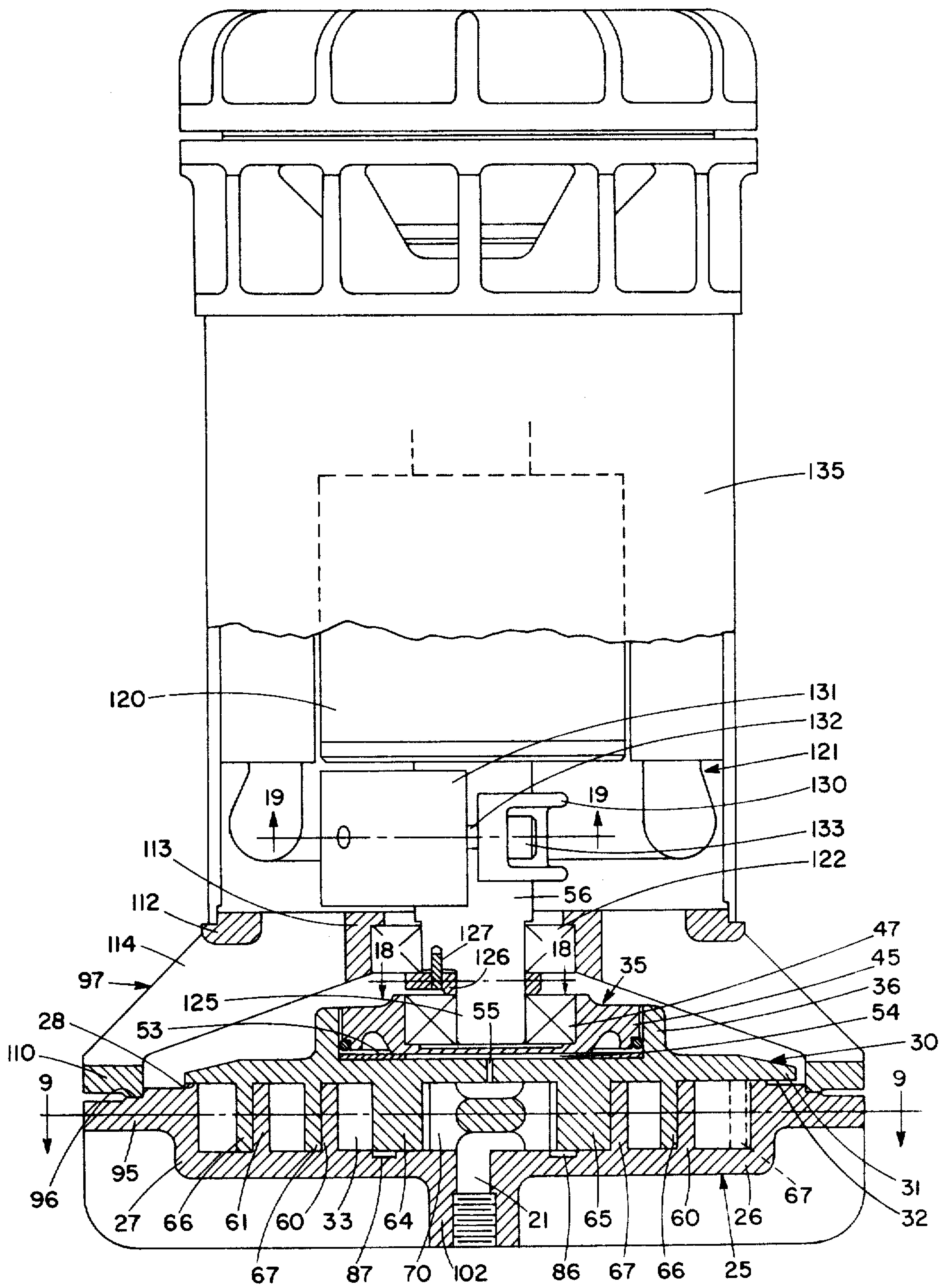
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Fig. 8

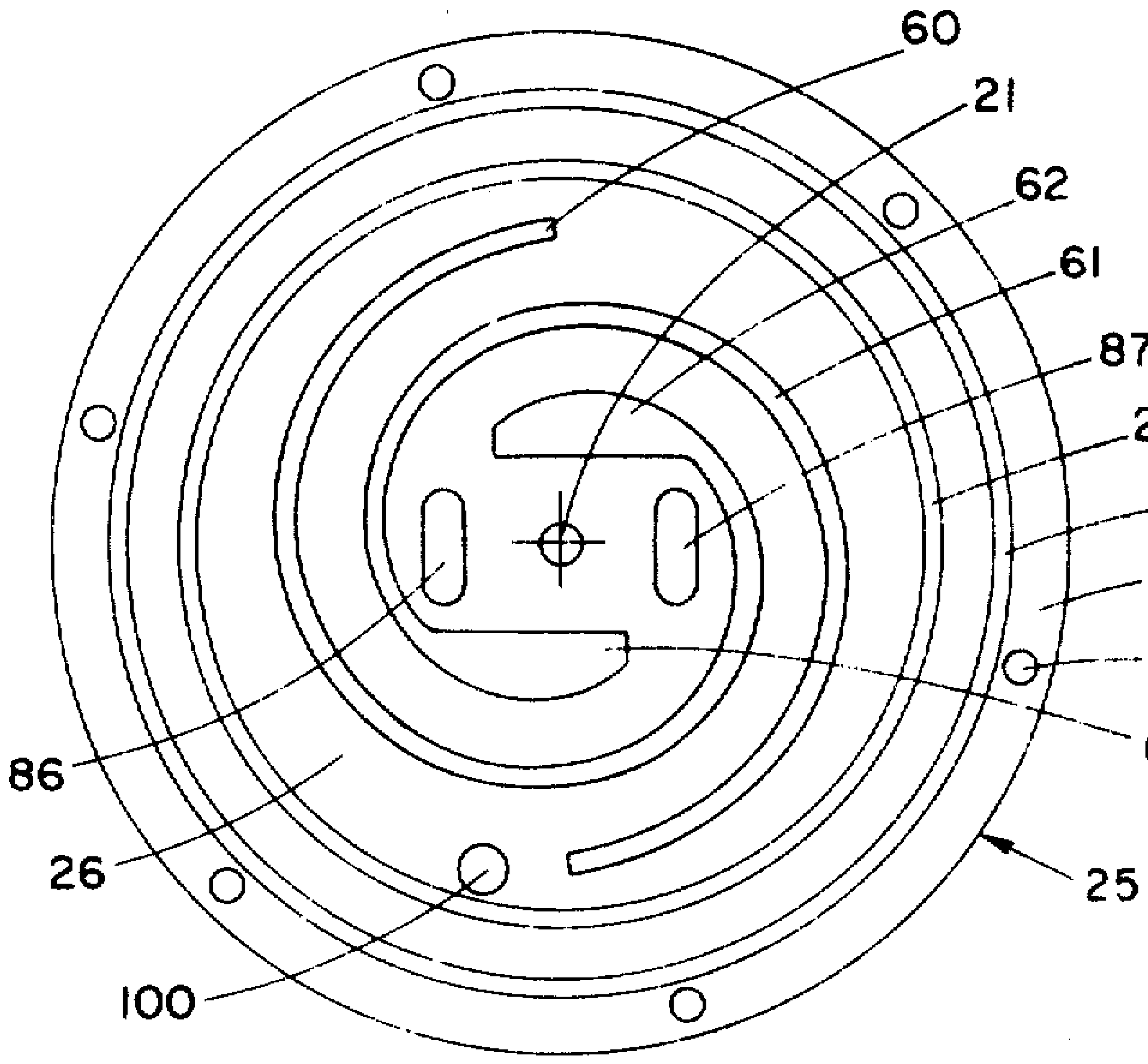


Fig. 10

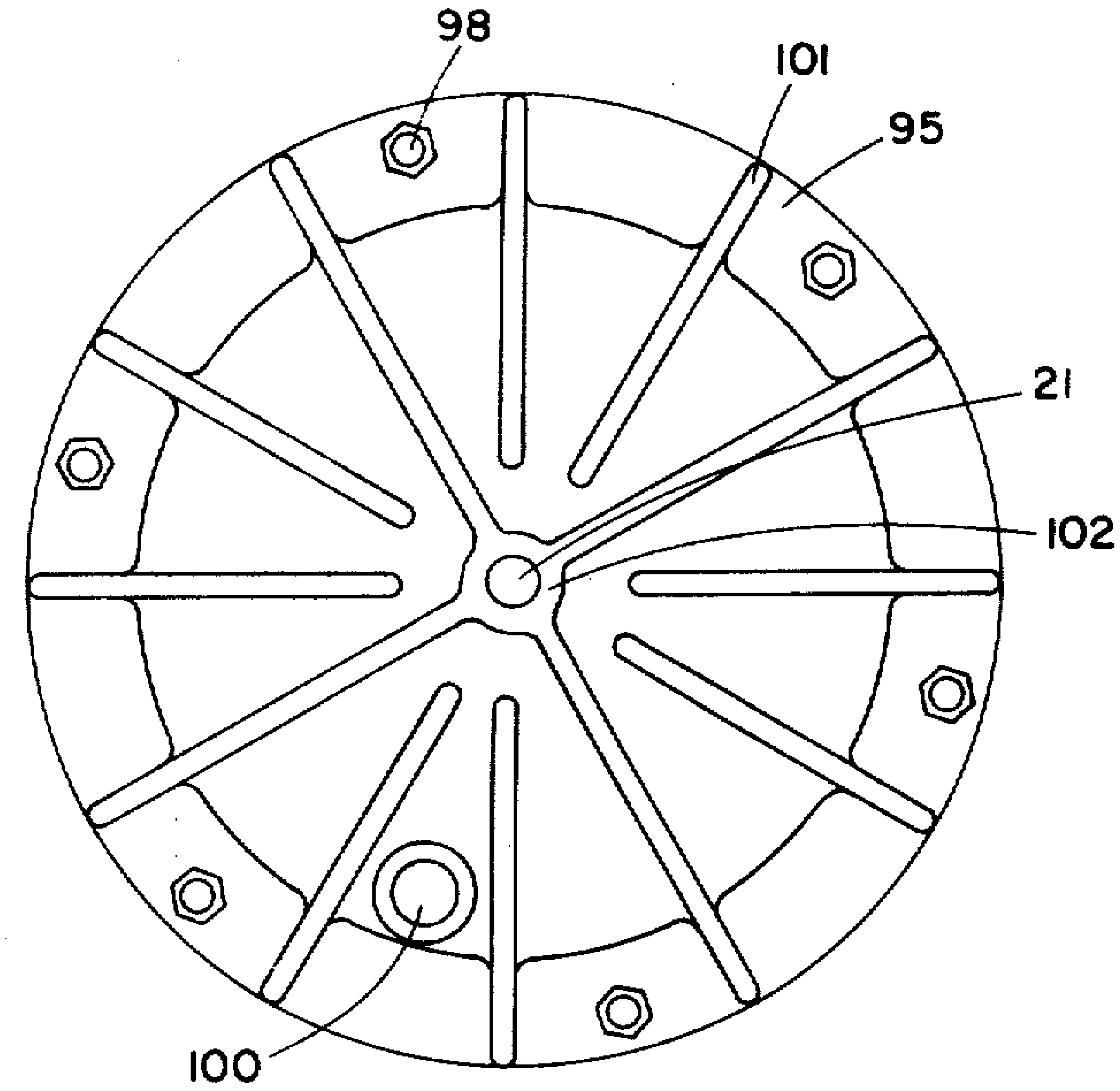
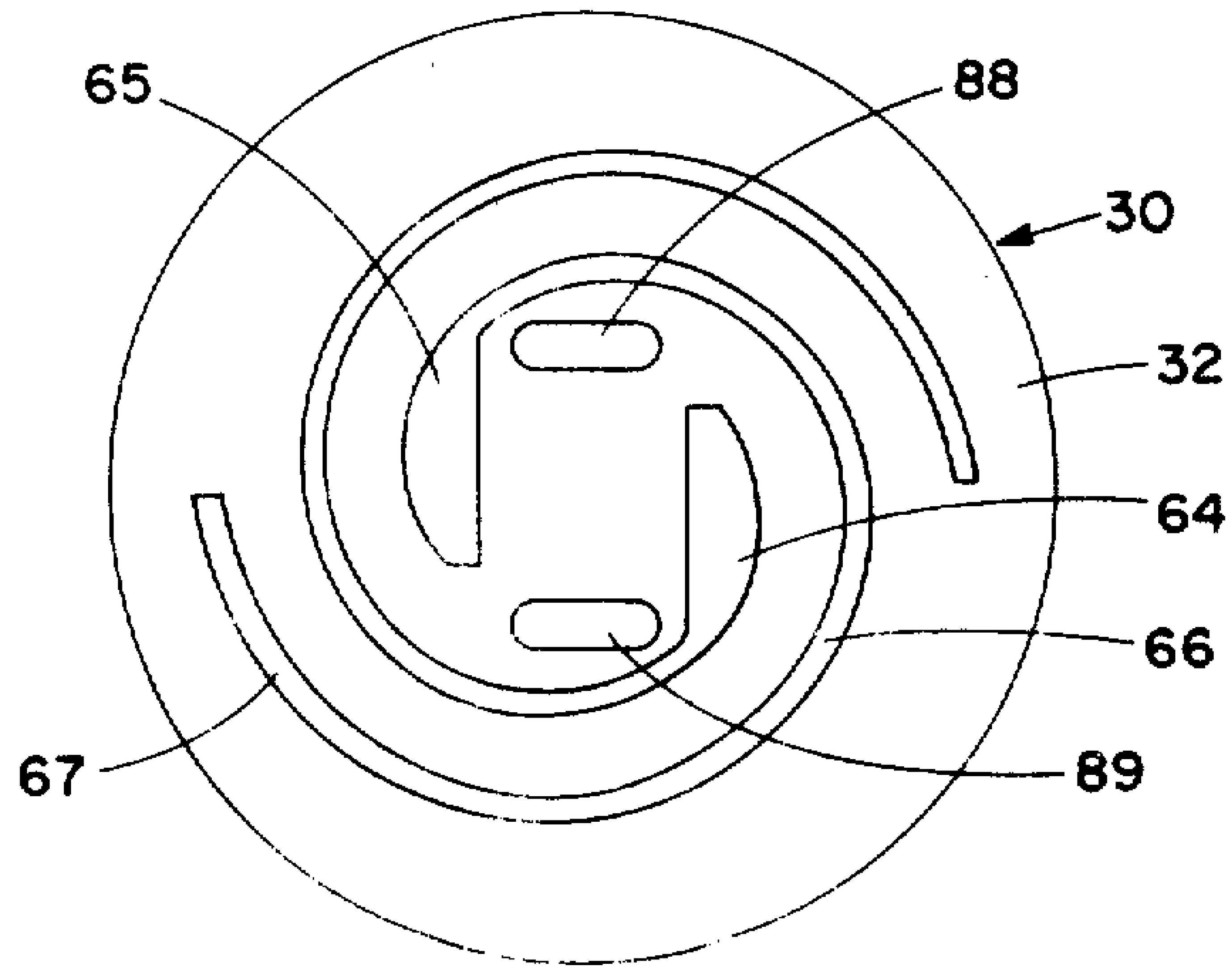


Fig. 11



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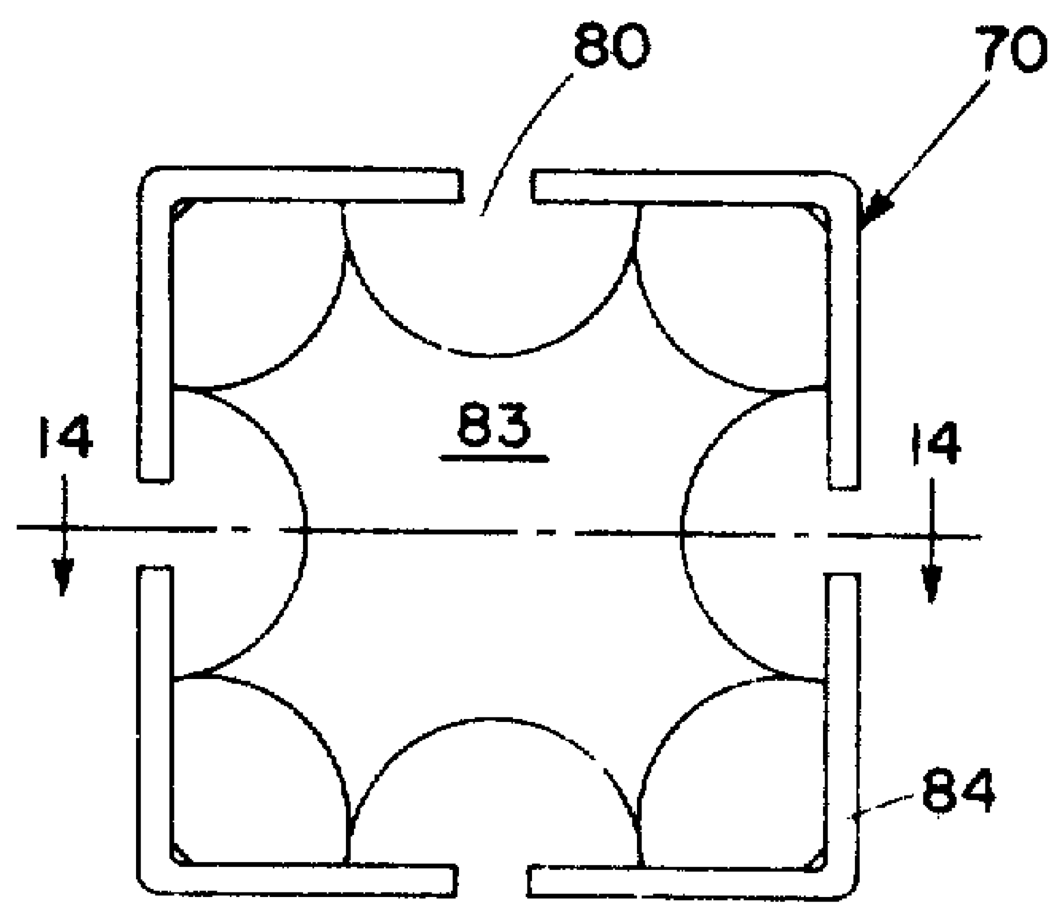

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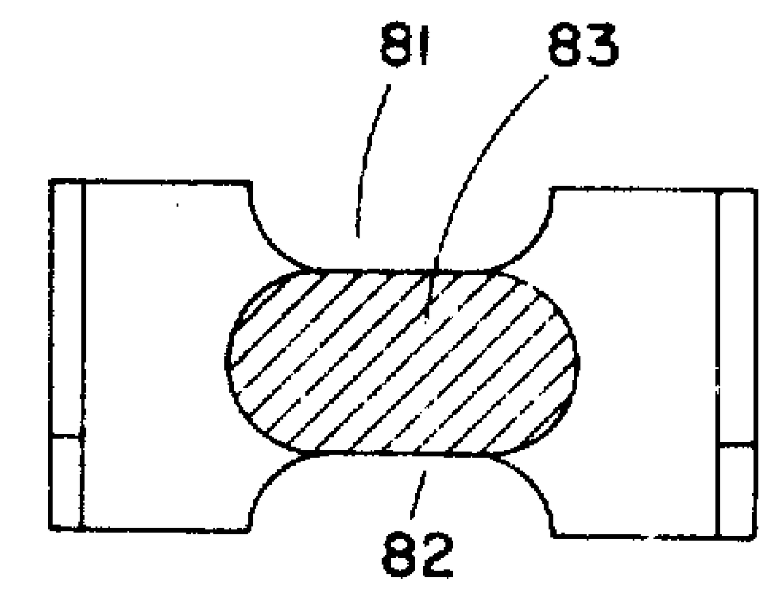

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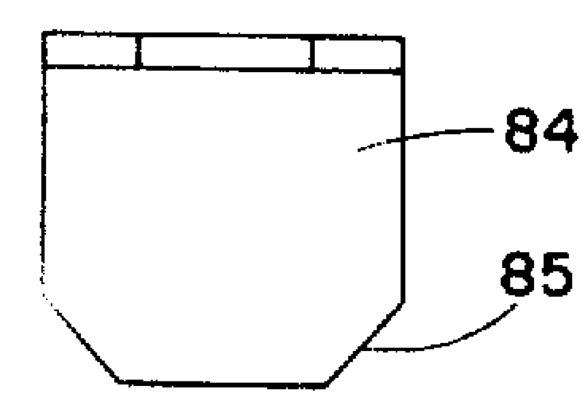

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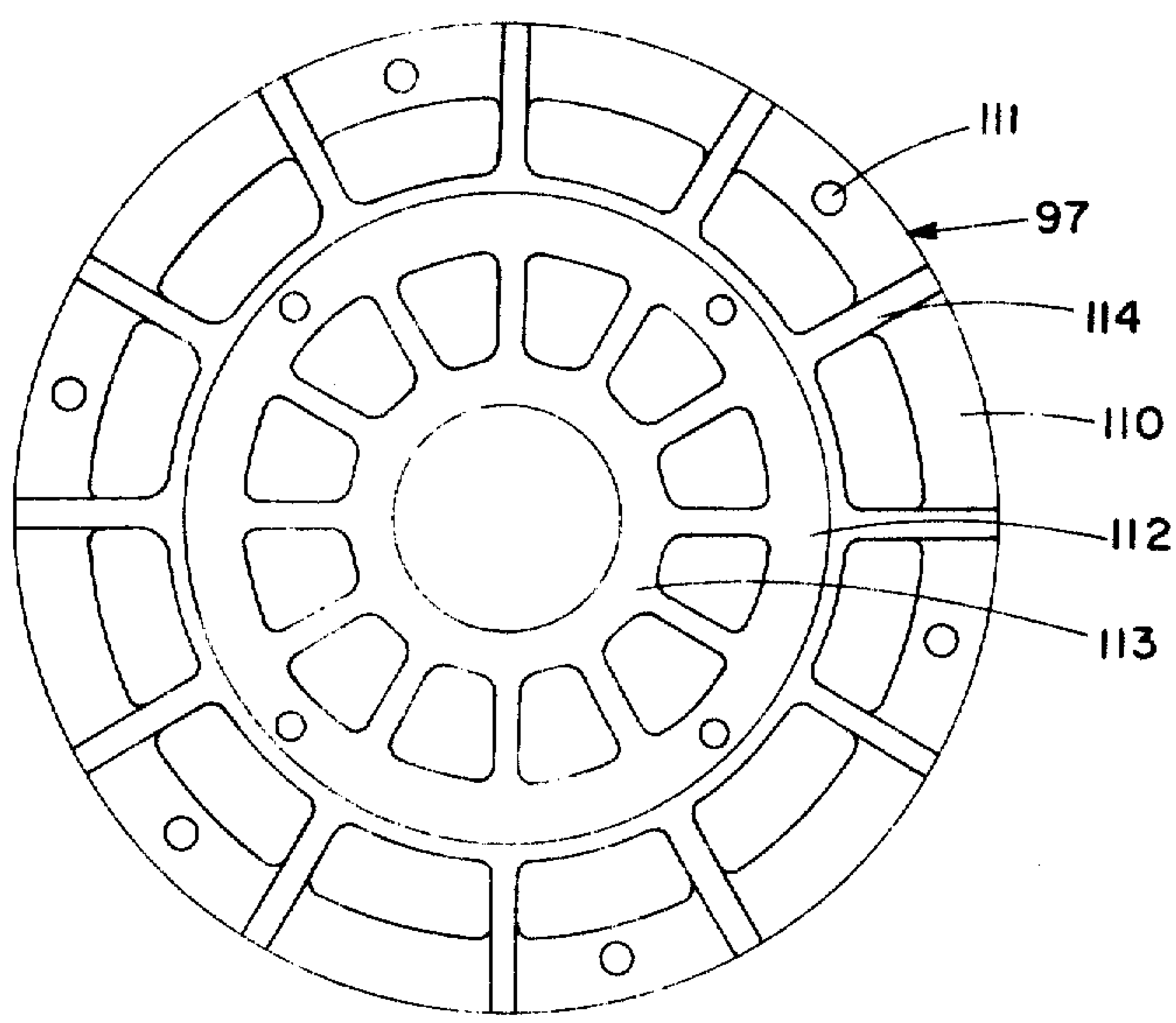
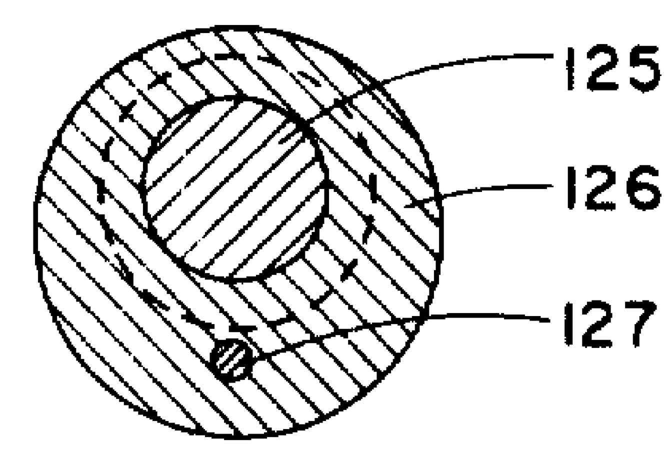


Fig. 16



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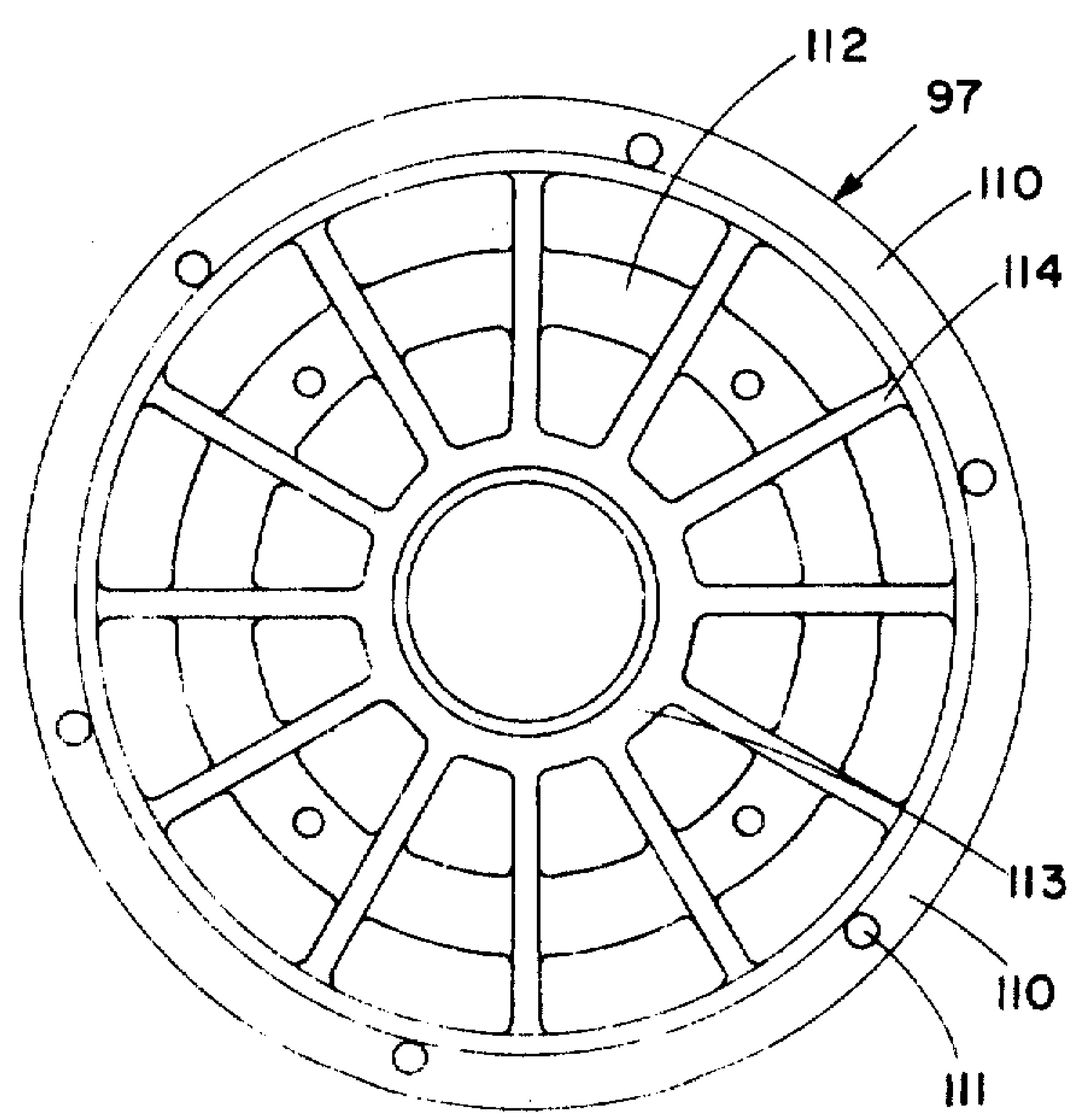

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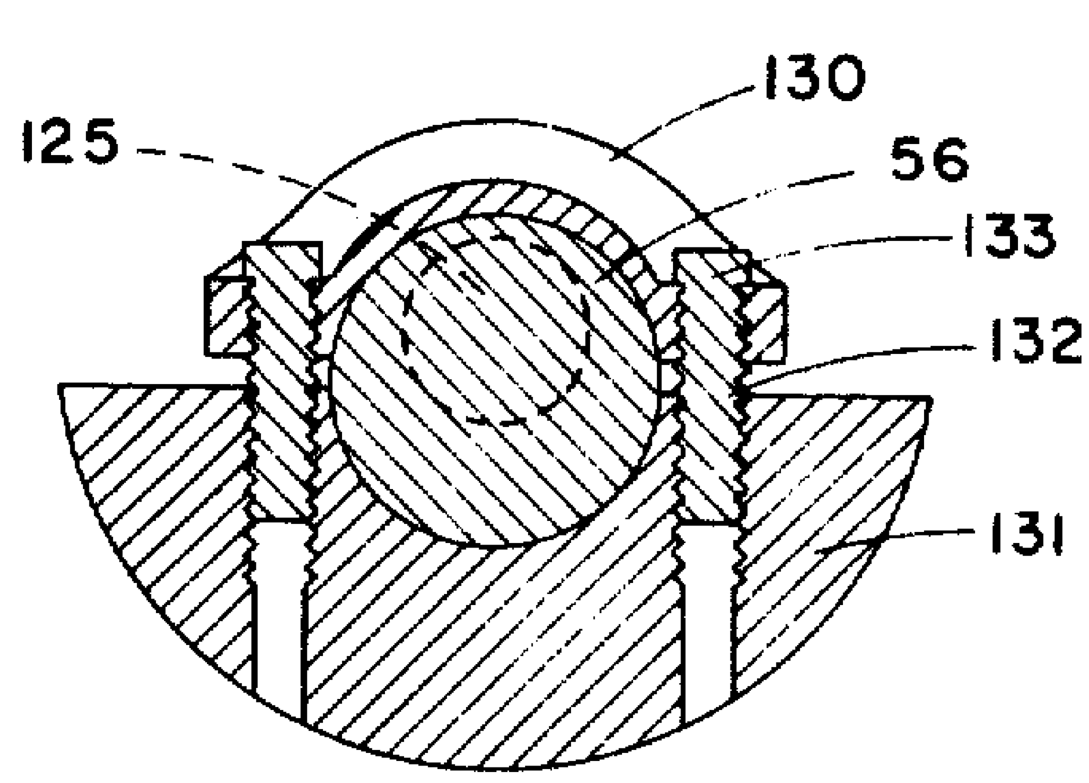

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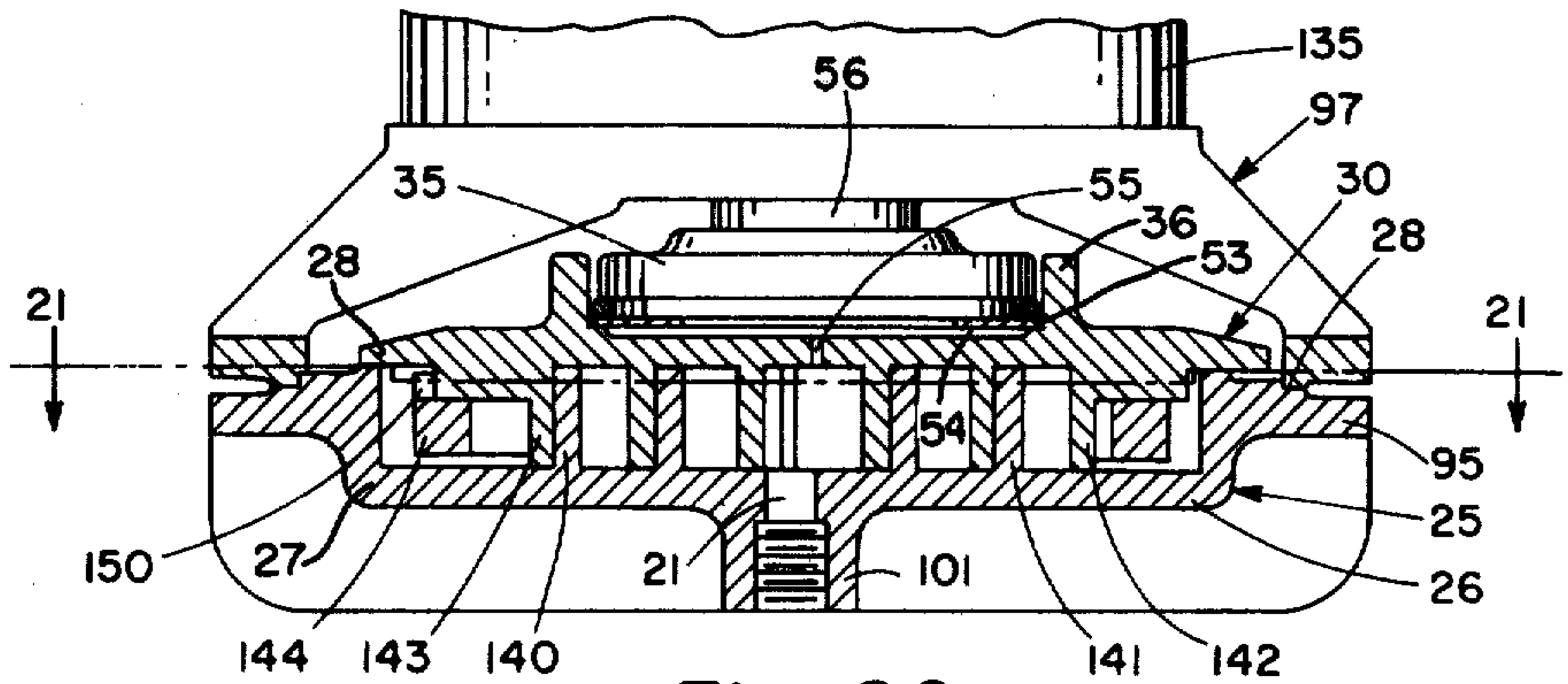

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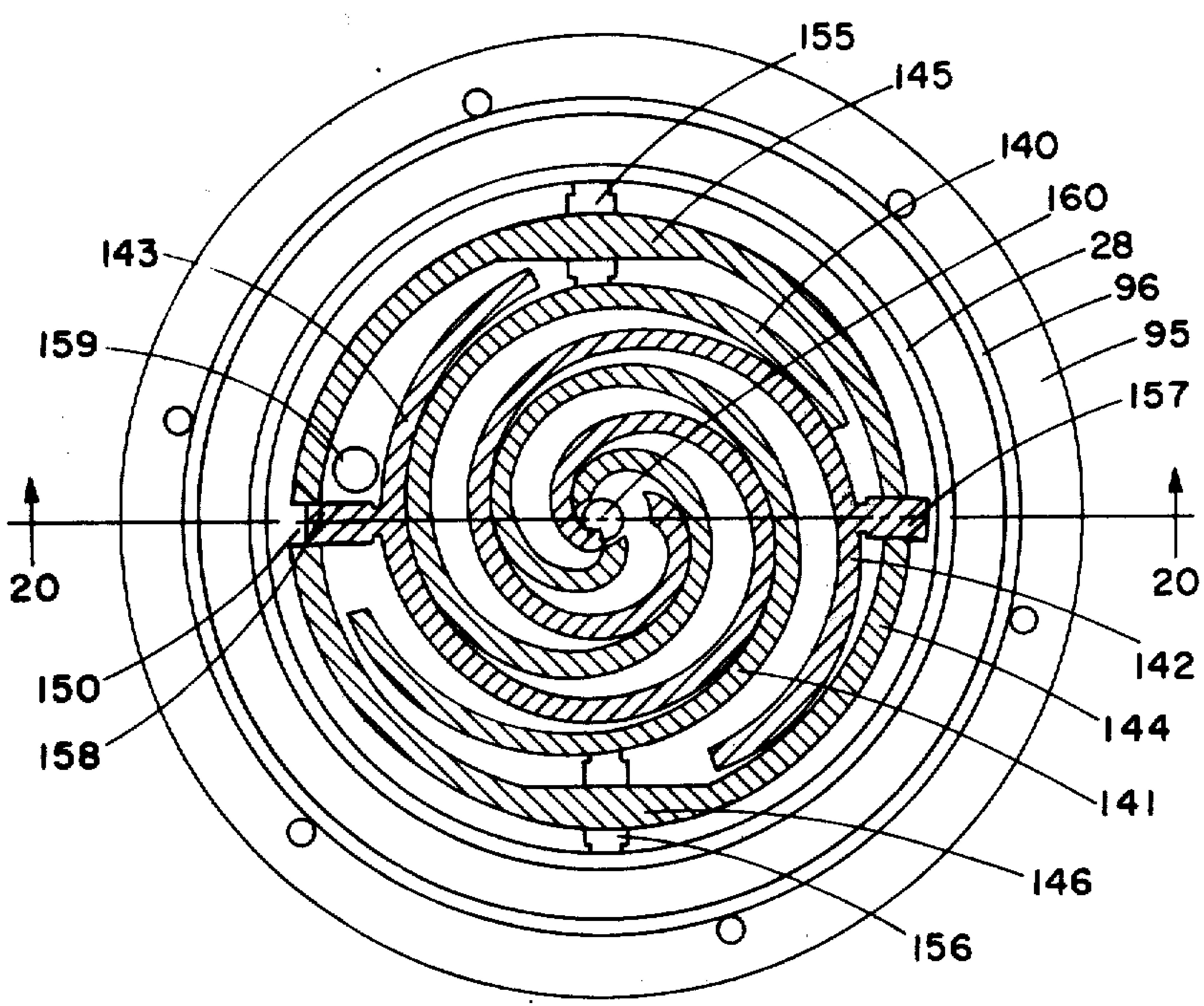

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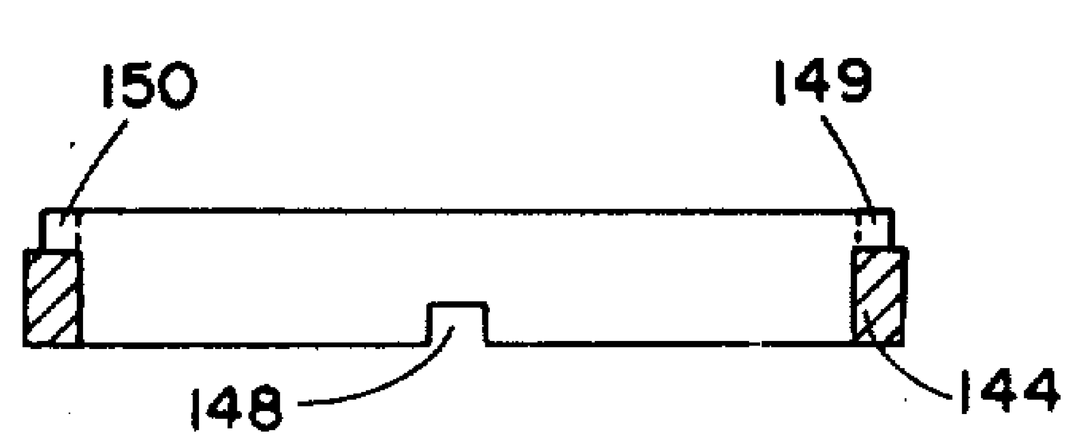

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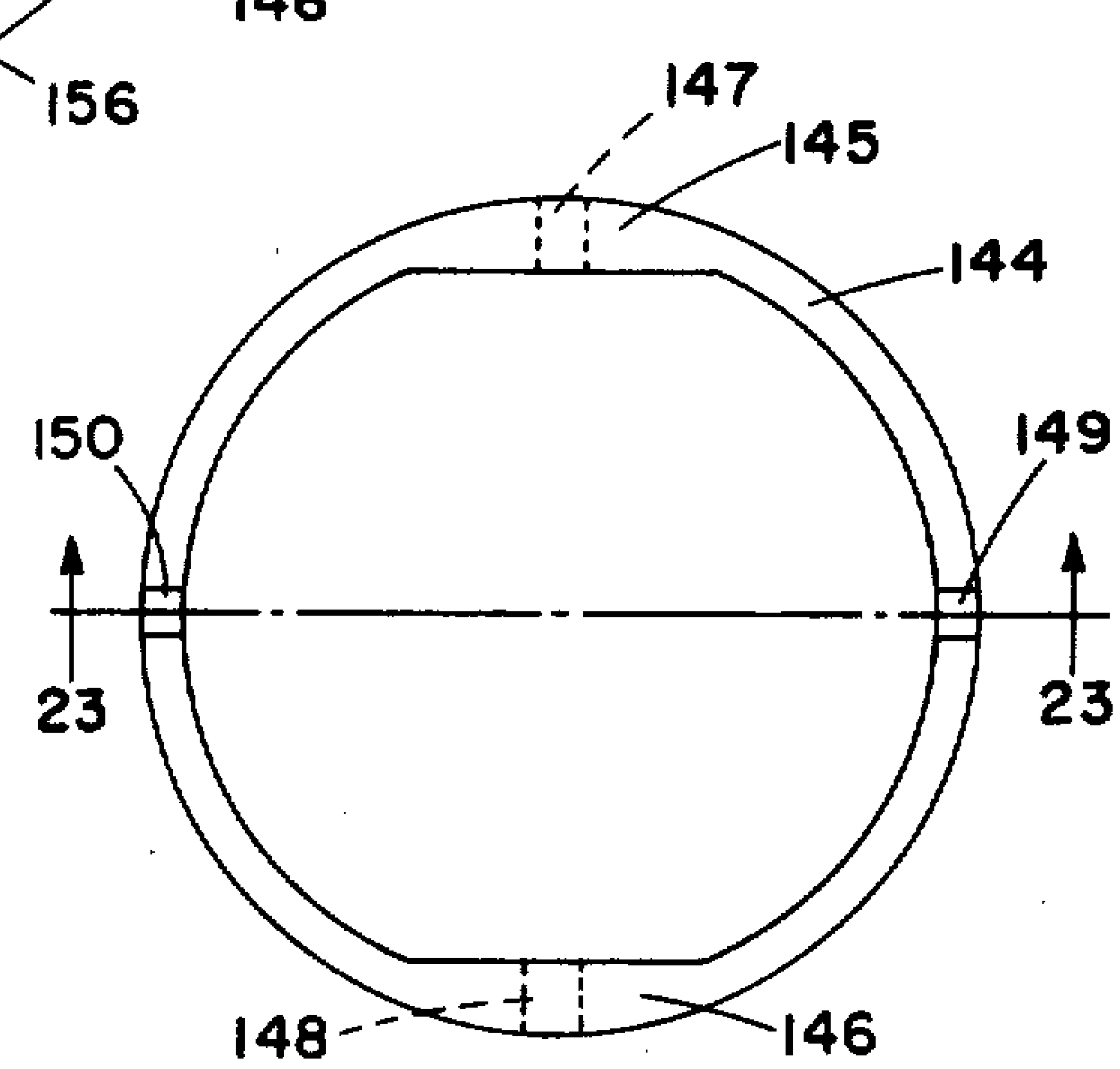

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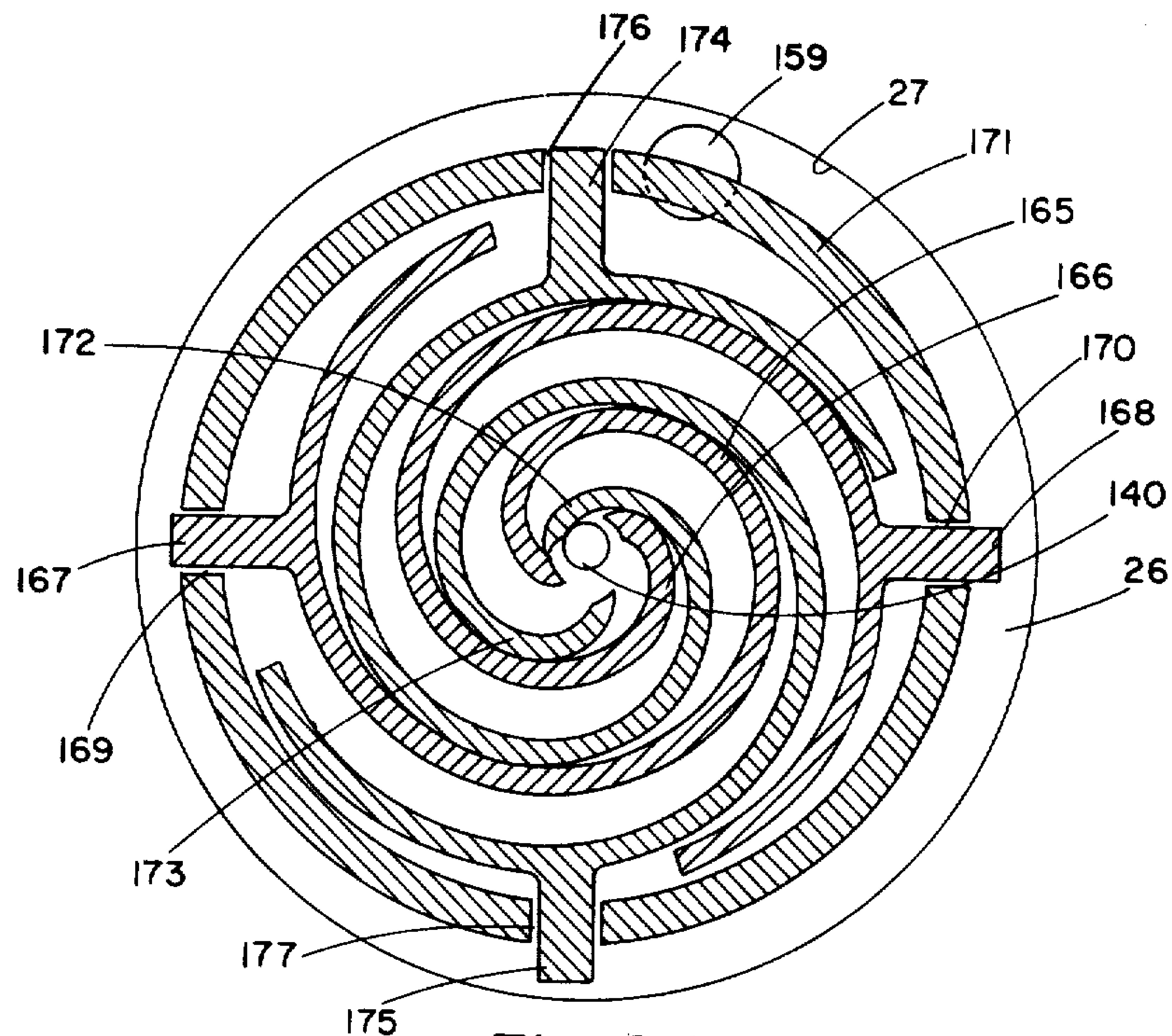
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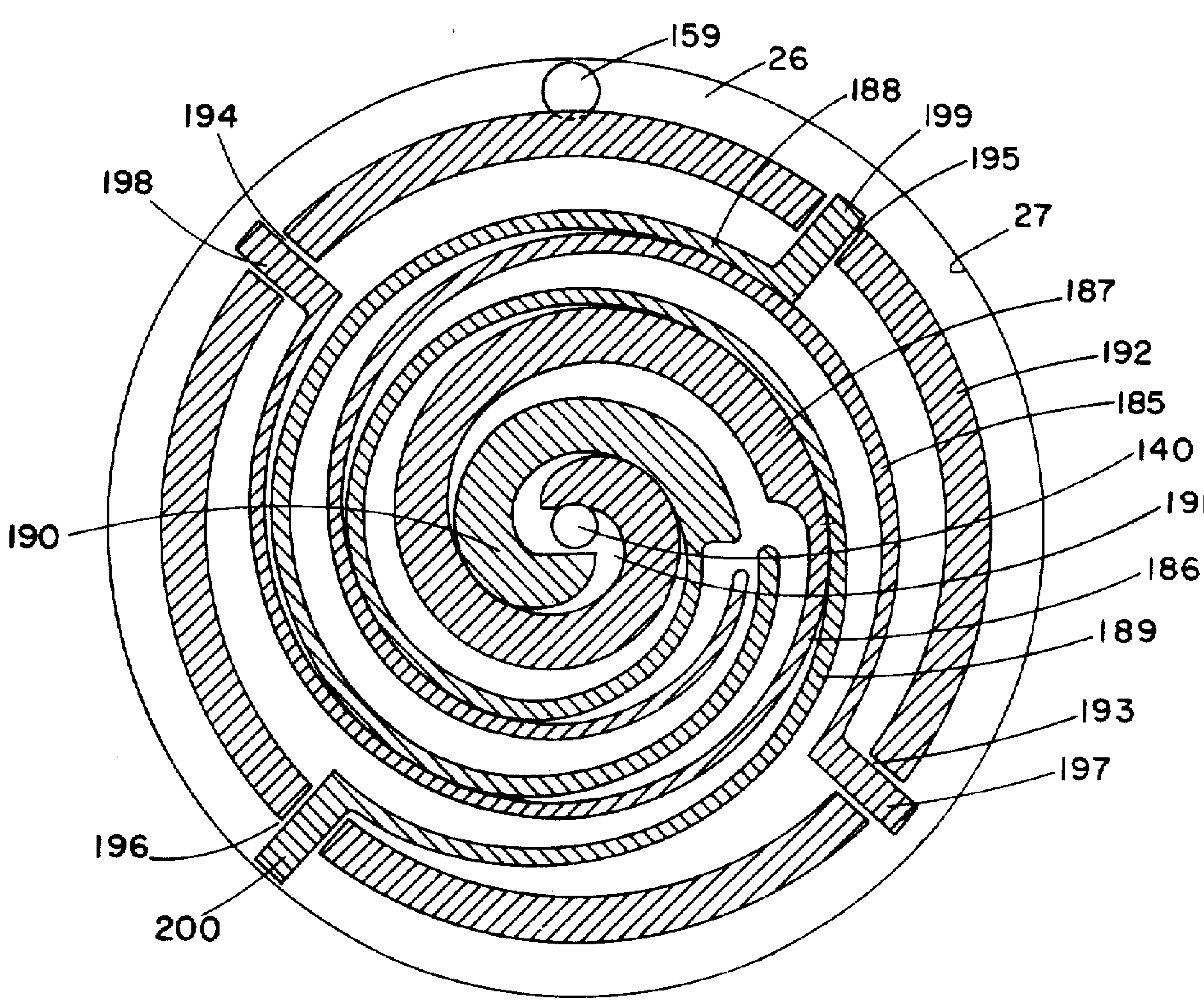
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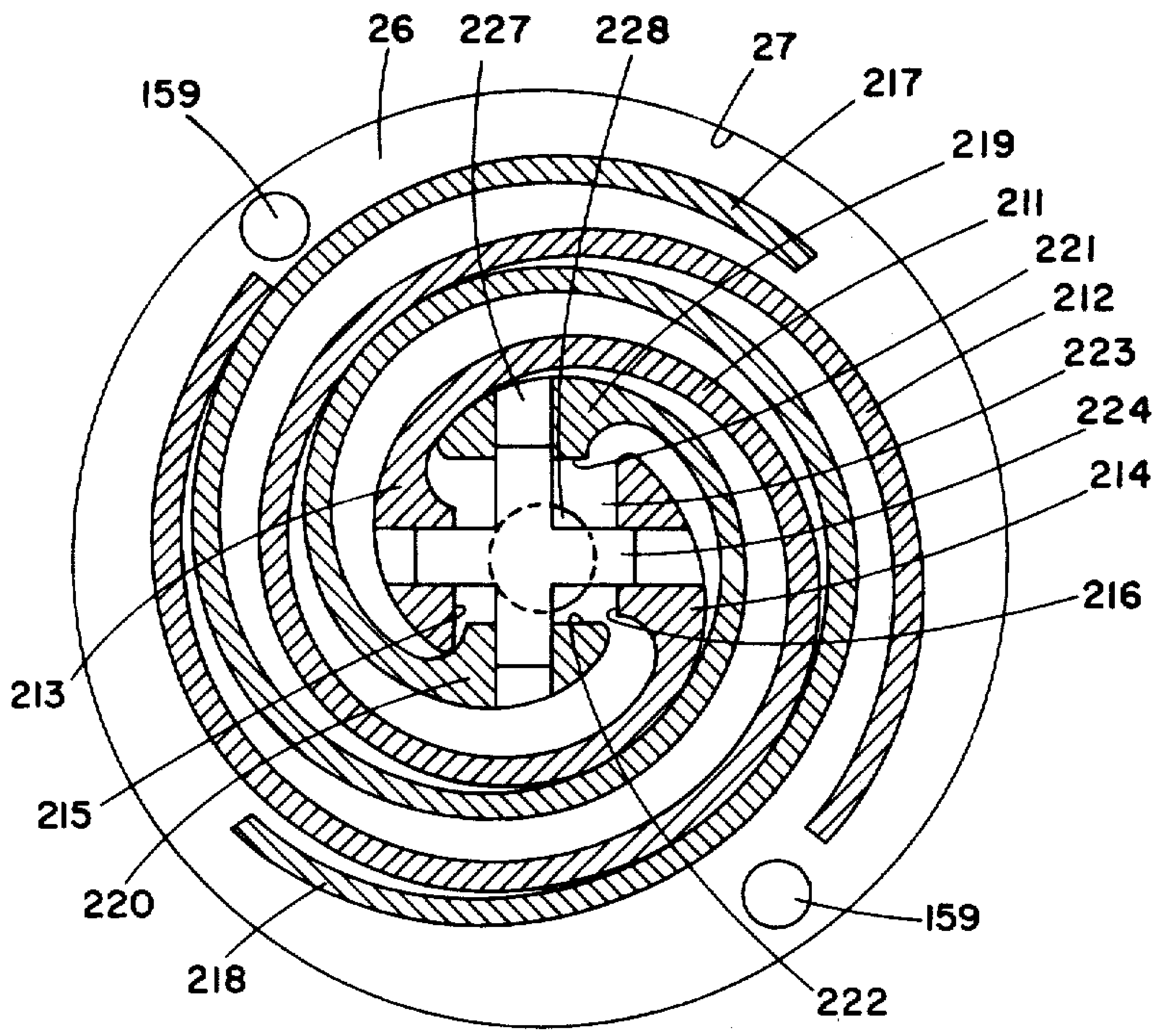
Fig. 25



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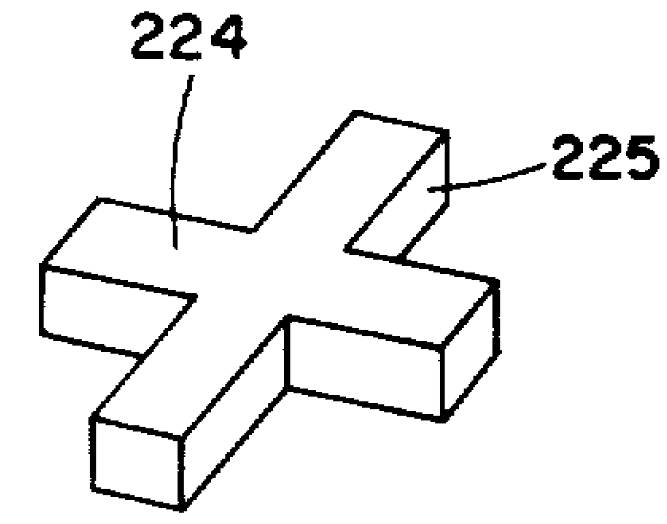

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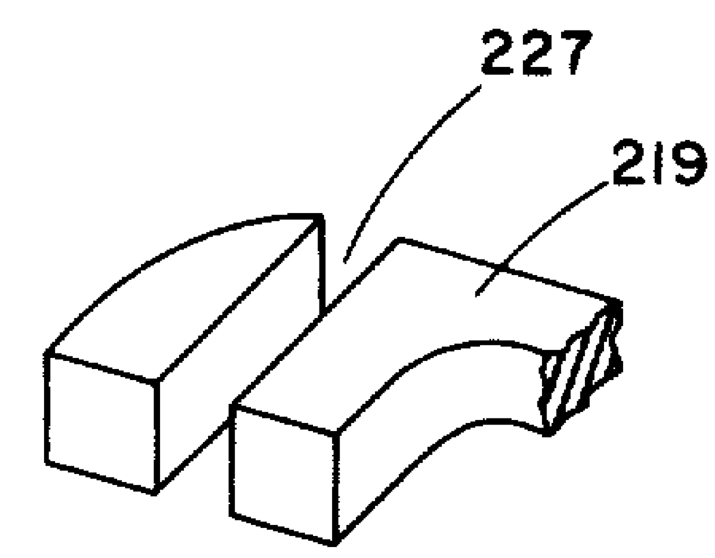

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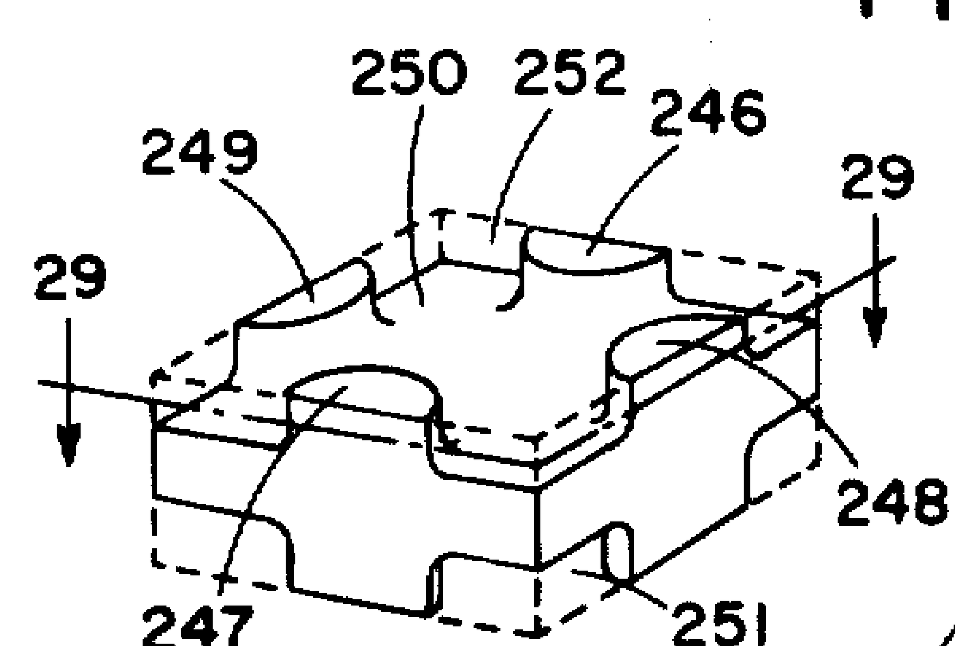

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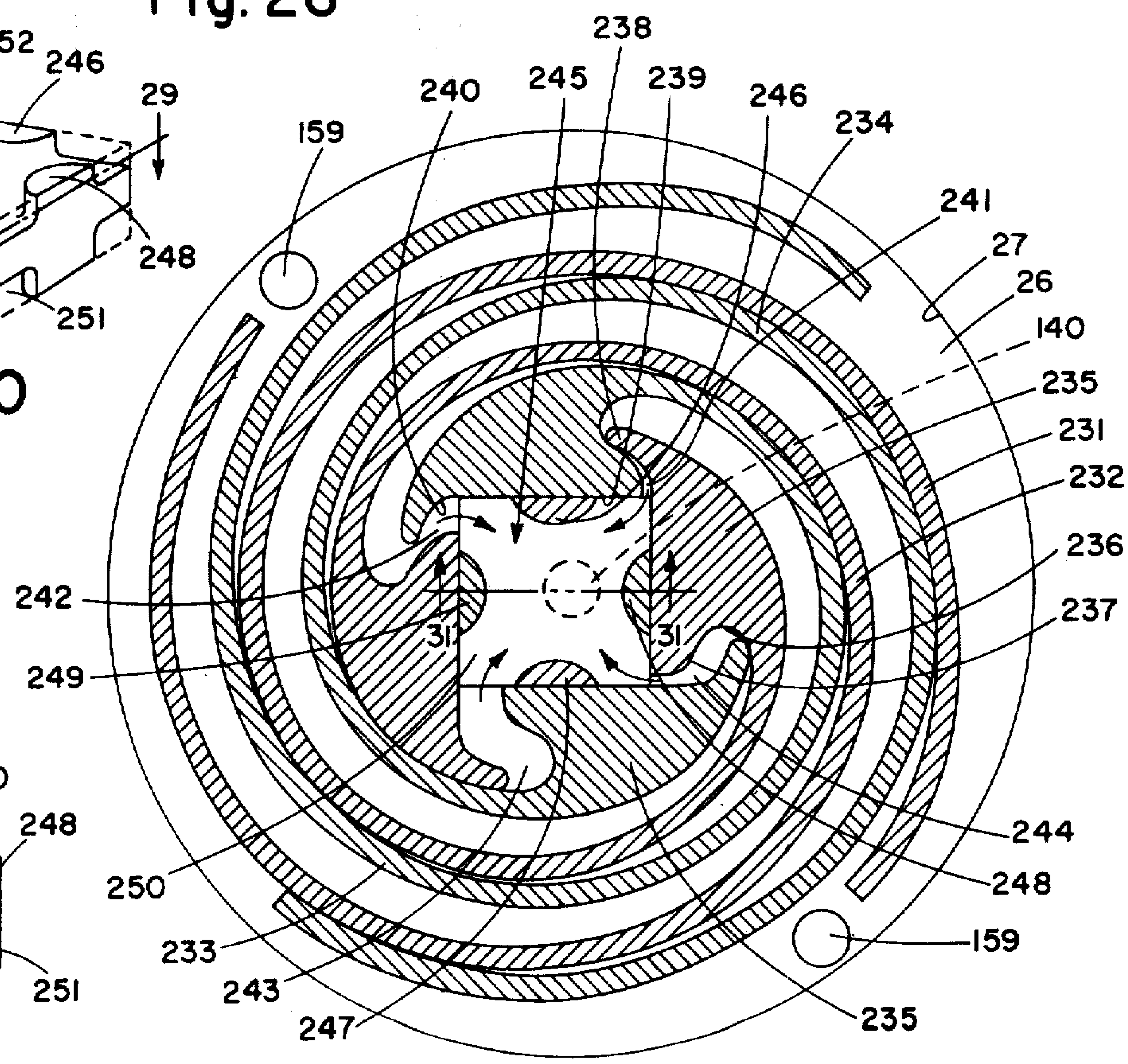

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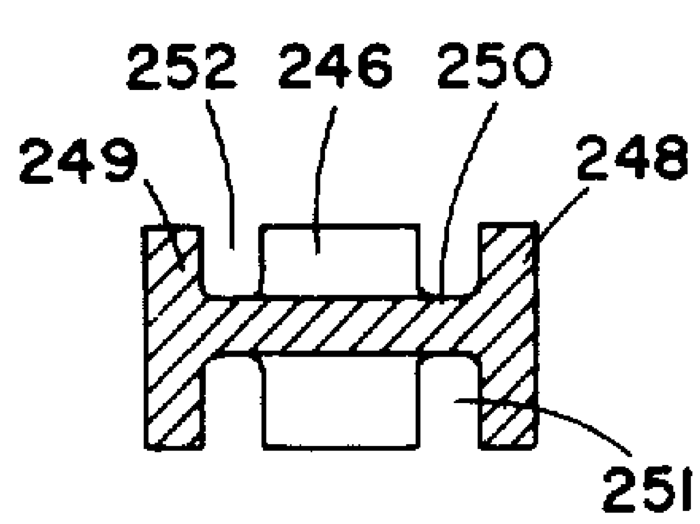

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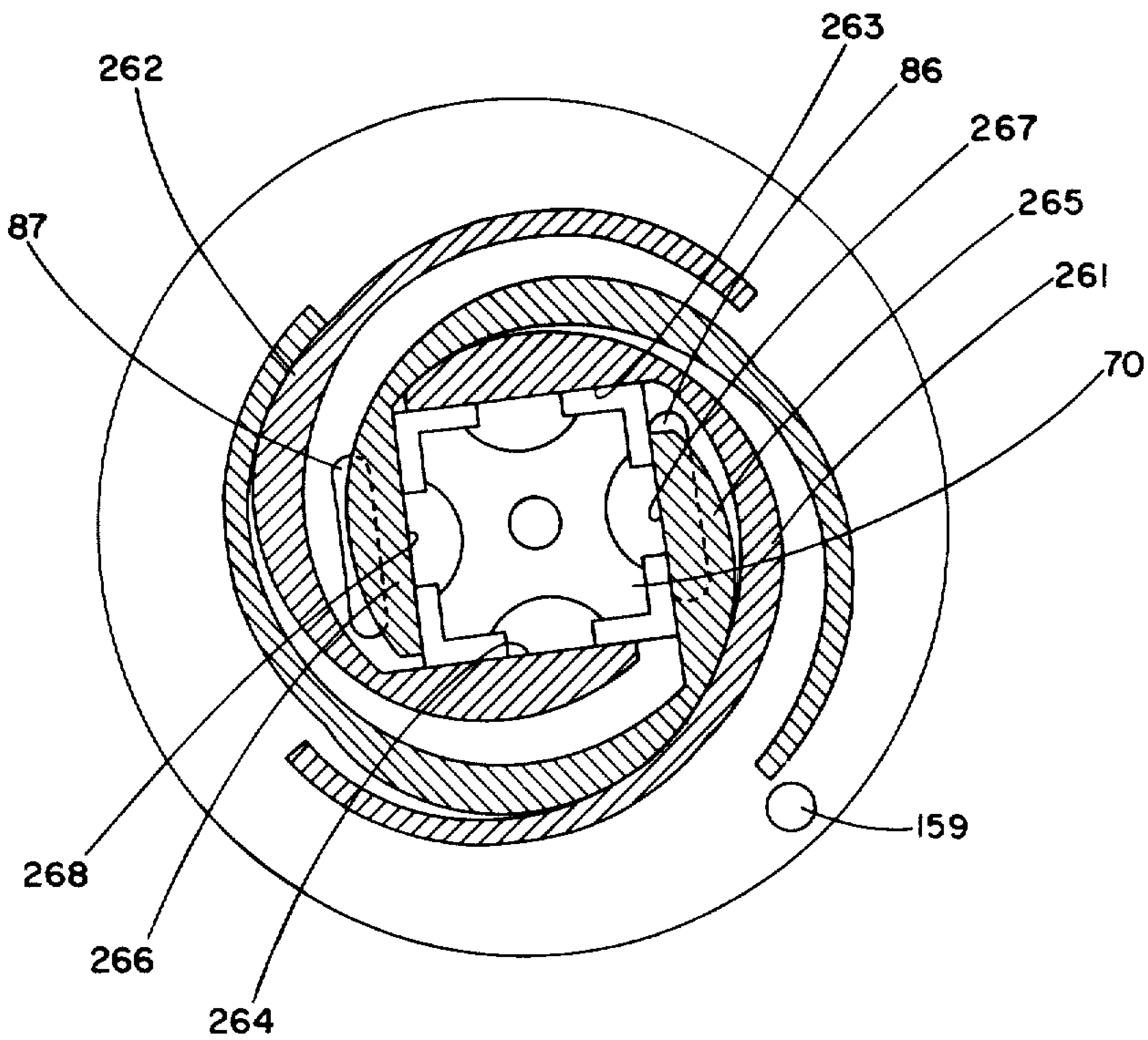

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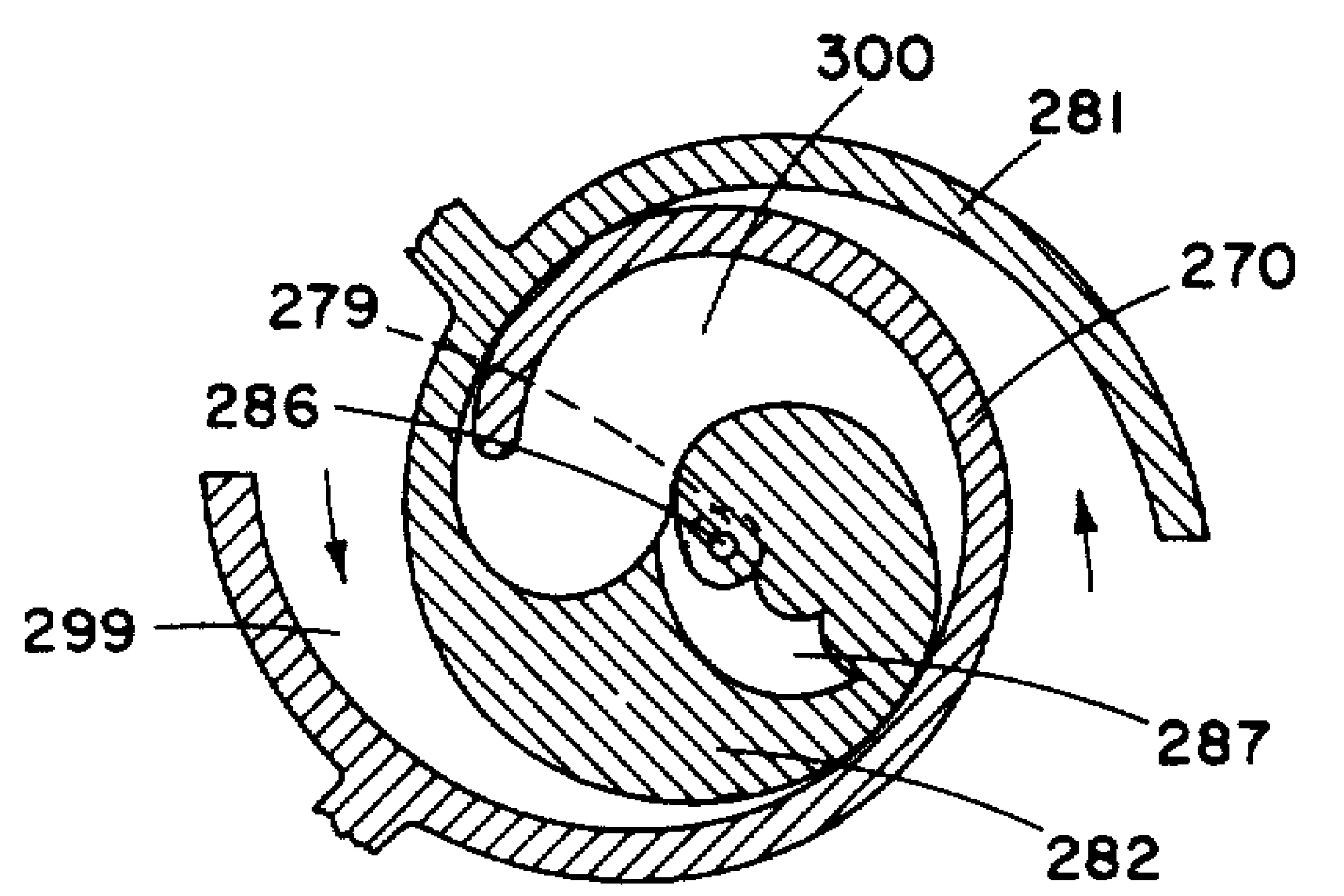

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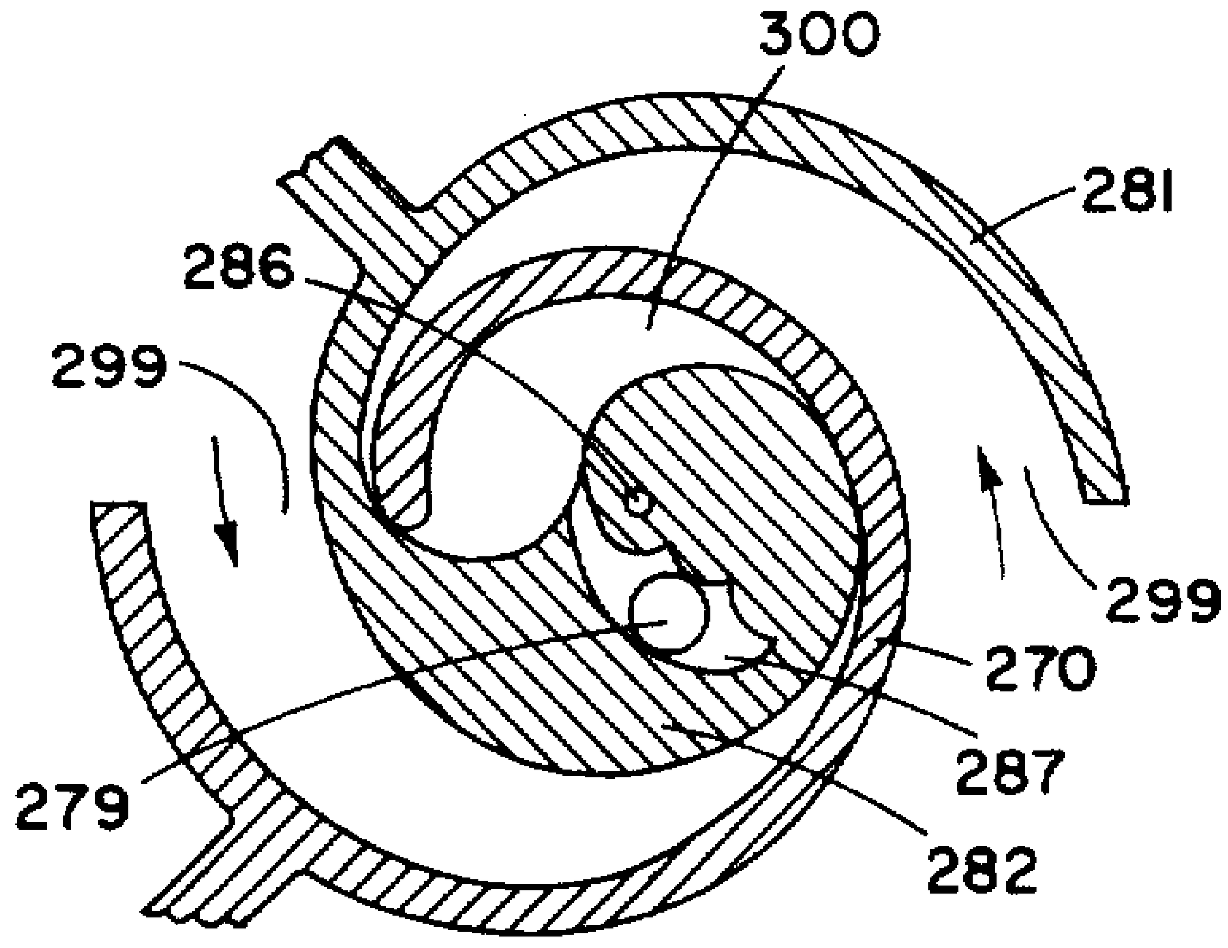

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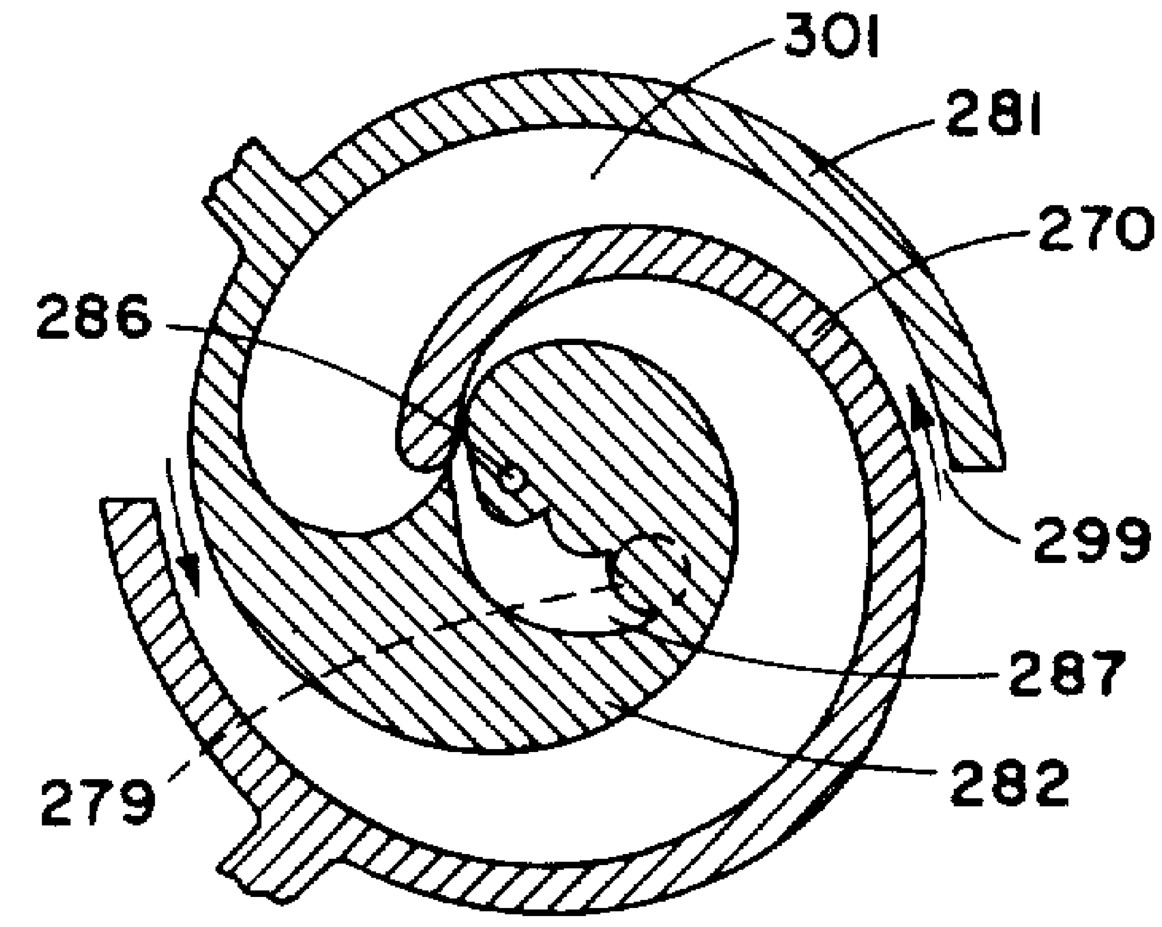

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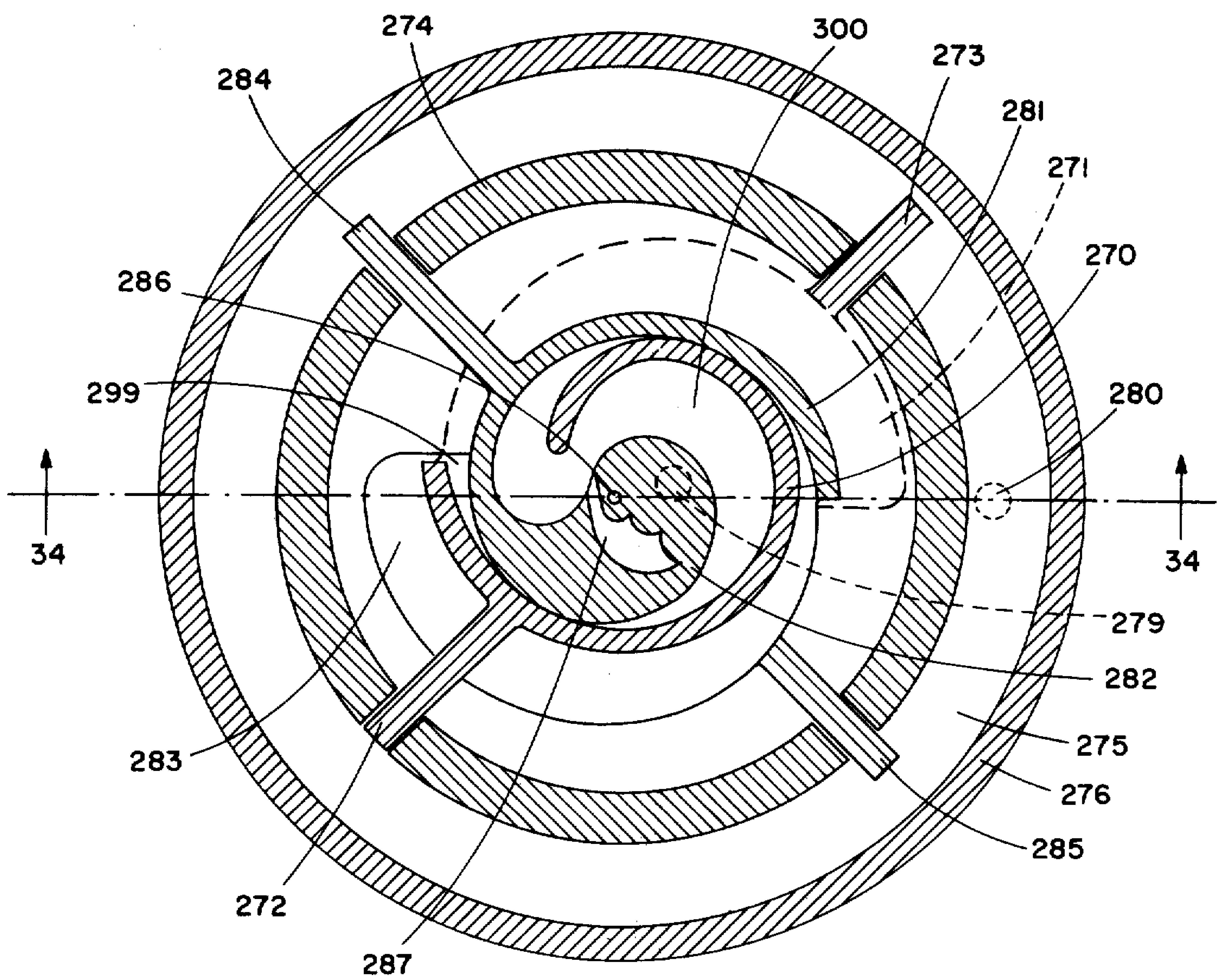
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Fig. 33

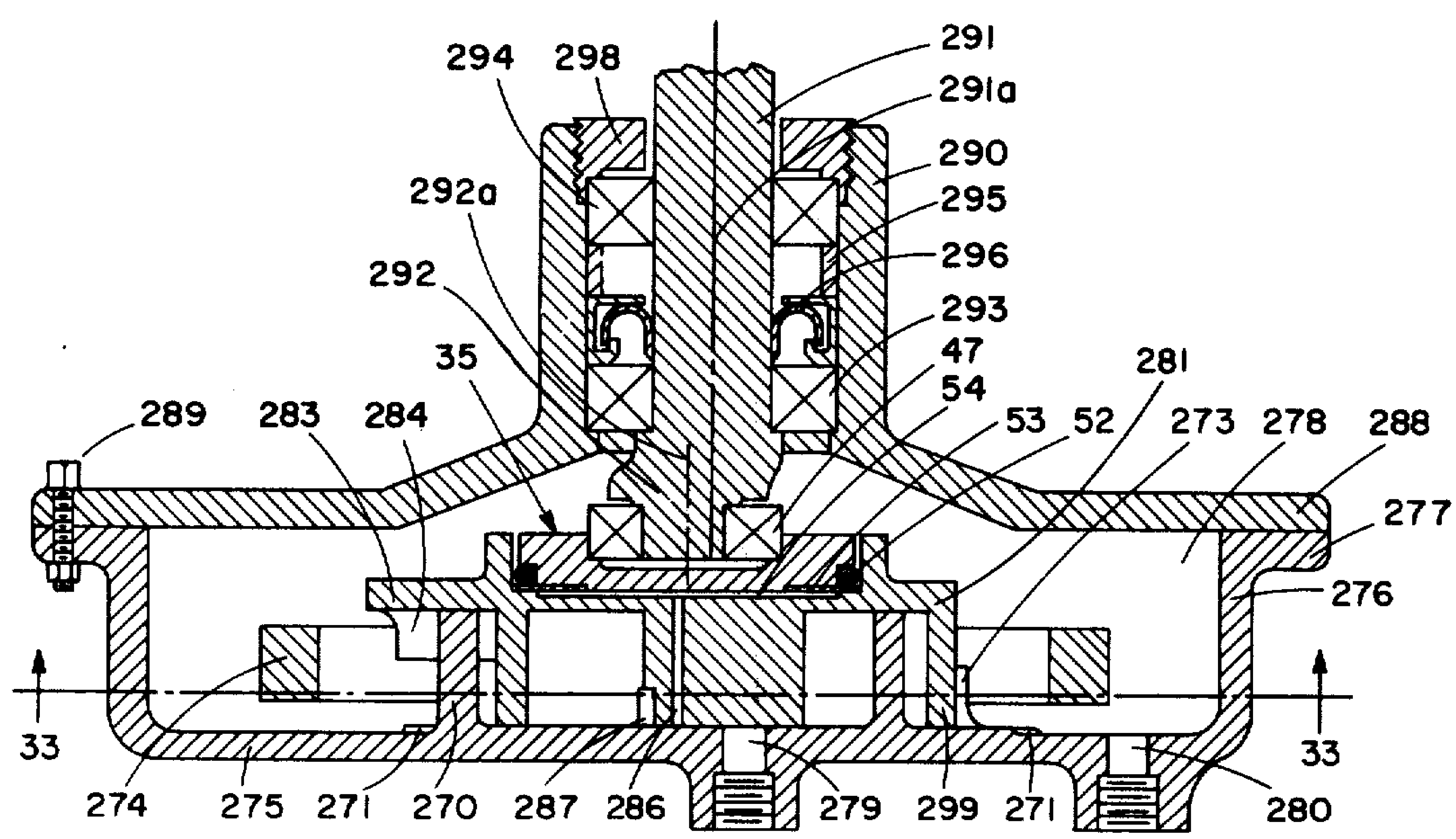
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Fig. 34

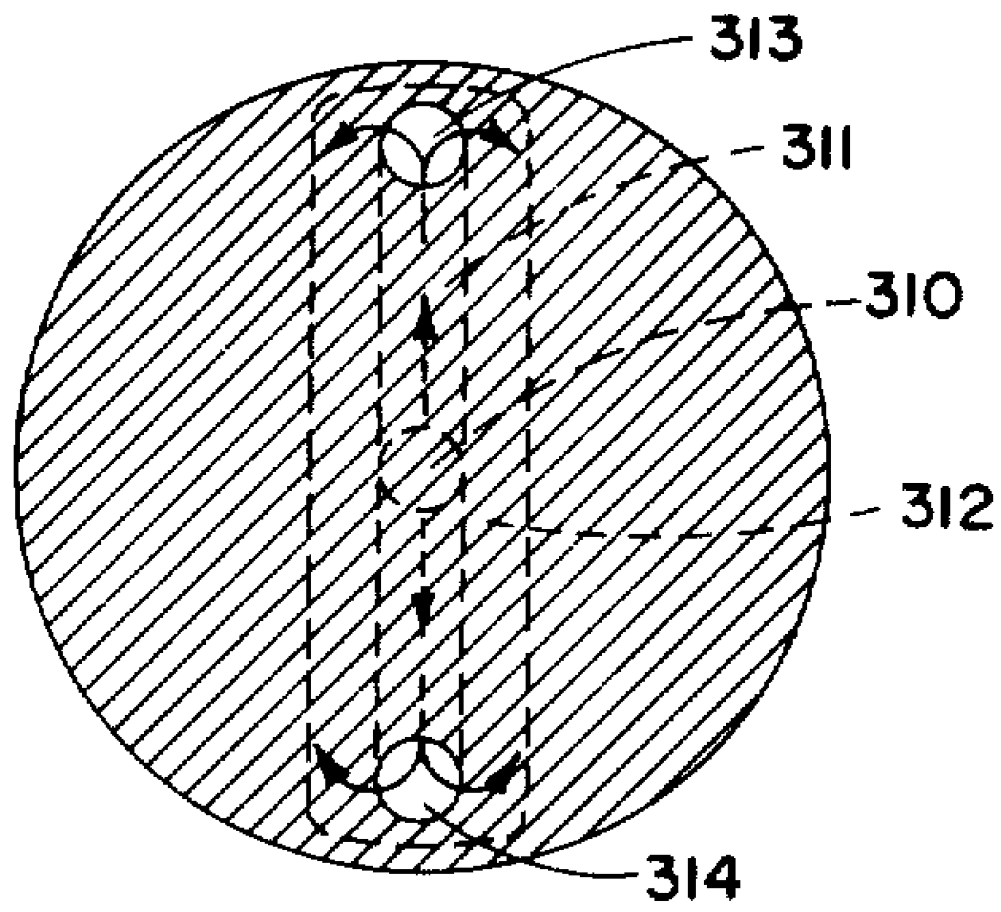
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Fig. 39

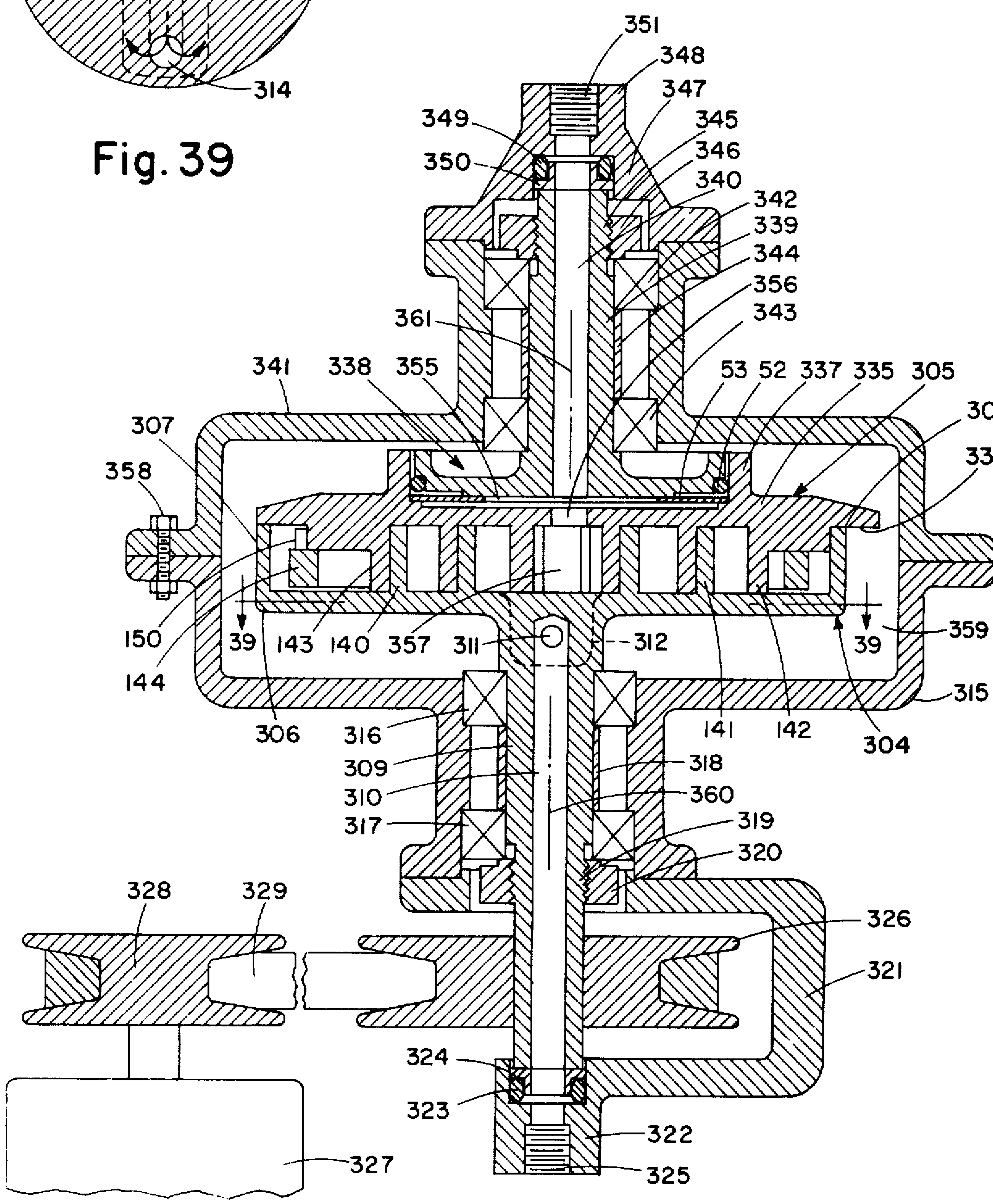
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Fig. 38

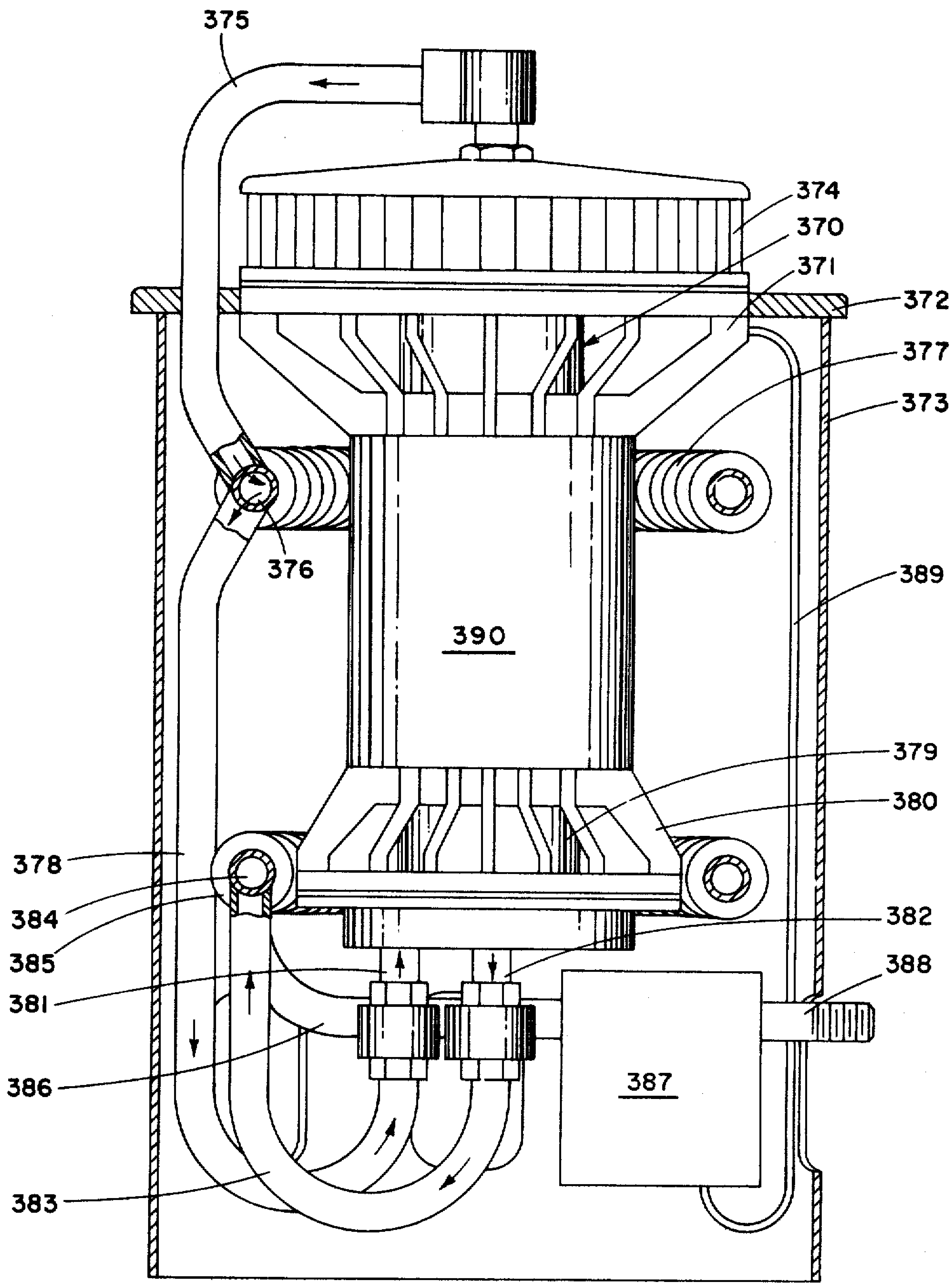
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Fig. 40

SCROLL-TYPE POSITIVE FLUID DISPLACEMENT APPARATUS

This invention relates to fluid displacement apparatus and more particularly to apparatus for handling fluids to compress, expand or pump them.

The need for gas compressors and expanders and for fluid pumps is well known and there are many different types of such apparatus. In these apparatus a working fluid is drawn into an inlet port and discharged through an outlet port at a higher pressure; and when the fluid is a gas its volume may be reduced before delivery through the outlet port, in which case the apparatus serves as a compressor. If the working fluid is a pressurized gas when it is introduced and its volume is increased, then the apparatus is an expansion engine capable of delivering mechanical energy and also if desired of developing refrigeration. Finally, a fluid may be introduced and withdrawn at different pressures but without any appreciable change in volume, in which case the apparatus serves as a fluid pump.

In the following description of the fluid displacement apparatus of this invention it will be convenient to refer to it, and to the prior art, as a compressor. However, it is to be understood that the apparatus of this invention may also be used as an expansion engine and as a pump and its use as such will be described for the various apparatus embodiments.

It is not necessary to discuss the prior art in detail as it pertains to such dynamic apparatus as centrifugal compressors and pumps, or as it pertains to the more commonly used positive-displacement devices of the vane, gear or other rotary types. However it is of interest to note some of the features which characterize these general types of prior art apparatus as a basis for comparison with the fluid displacement apparatus of this invention.

Those pumps, compressors and blowers which may be termed "dynamic" apparatus must operate at high speeds to achieve large pressure ratios and they typically have efficiencies of less than 90% in terms of mechanical energy converted to flow and compressional energy. Apparatus of the dynamic type find their widest application in large sizes in such applications as gas turbine compressors, stationary power plant steam expanders, and the like.

The positive displacement pumps or compressors of the vane type have rubbing speeds proportional to the radius of the vanes and the vanes rub at varying angles. Furthermore, the vanes operate within a housing of fixed axial length so that any wear upon their flat surface ends will always act to increase the clearance, and hence, the blow-by or leakage of the apparatus. The positive-displacement pumps and compressors of the rotary type are typically constructed to have the rotating components movable between end plates, an arrangement which demands close tolerances to reduce blow-by while permitting free rotation. Wear between the rotating components and end plates increases blow-by, a fact which requires the adjustment of the spacings of the end plates through the use of screws and very precisely constructed gaskets in the form of shims. The gaskets, in turn, may not be able to withstand corrosive fluids or fluids at extreme temperatures, e.g., cryogenic liquids or hot gases. Furthermore, these gaskets require precisely located edges to prevent injury by the moving vanes, a fact which adds to the delicacy of assembling the apparatus.

In most industrial applications, particularly those of large scale, the fluid pumps and compressors now being used are adequate for the uses for which they are employed. However, there remains a need for a simple, highly efficient apparatus, essentially unaffected by wear which can handle a wide range of fluids and operate over a wide range of conditions to serve as a pump, compressor or expansion engine. The apparatus of this invention which meets these requirements is based on the use of scroll members, having wraps which make moving contacts to define moving isolated volumes, called "pockets", which carry the fluid to be handled. The contacts which define these pockets formed between scroll members are of two types: line contacts between spiral cylindrical surfaces, and area contacts between plane surfaces. The volume of a sealed pocket changes as it moves. At any one instant of time, there will be at least one sealed pocket. When there are several sealed pockets at one instant of time, they will have different volumes, and in the case of a compressor, they will also have different pressures.

There is known in the art a class of devices generally referred to as "scroll" pumps, compressors and engines wherein two interfitting spiroidal or involute spiral elements of like pitch are mounted on separate end plates. These spirals are angularly and radially offset to contact one another along at least one pair of line contacts such as between spiral cylinders. The pair of line contacts will lie approximately upon one radius drawn outwardly from the central region of the scrolls. The fluid volume so formed therefore extends all the way around the central region of the scrolls. In certain special cases the pocket or fluid volume will not extend the full 360° but because of special porting arrangements will subtend a smaller angle about the central region of the scrolls. The pockets define fluid volumes which vary with relative orbiting of the spiral centers while maintaining the same relative spiral angular orientation. As the contact lines shift along the scroll surfaces, the pockets thus formed experience a change in volume. The resulting zones of lowest and highest pressures are connected to fluid ports.

An early patent to Creux (U.S. Pat. No. 801,182) describes this general type of device. Among subsequent patents which have disclosed scroll compressors, and pumps are U.S. Pat. Nos. 1,376,291, 2,809,779, 2,841,089, 3,560,119, and 3,600,114 and British Pat. No. 486,192.

Although the concept of a scroll-type apparatus has been known for some time and has been recognized as having some distinct advantages, the scroll-type apparatus of the prior art has not been commercially successful, primarily because of sealing, wearing and to some extent porting problems which in turn have placed severe limitations on the efficiencies, operating life, and pressure ratios attainable. Thus in some of the prior art devices the apparatus components have had to be machined to accurate shapes and to be fitted with very small tolerances to maintain axial sealing gaps sufficiently low to achieve any useful pressure ratios. This is difficult to do and resembles the problem of constructing apparatus with a reciprocating piston without the use of sealing rings. In other prior art devices, radial sealing has been achieved through the use of more than one form of radial constraint, each being imposed by separate apparatus components requiring precise interbalancing to attain efficient radial sealing. If during extended operation of such devices this interbalancing is disarranged by one component experiencing more wear, or by any other mechanism, the problem of wear of other components may grow progressively worse until satisfactory radial sealing is no longer possible.

In place of ports, delivery of the compressed fluid in a number of the prior art scroll apparatus has been made through the scroll passages, and compression ratios have previously been limited to approximately the ratio of the radius to the outermost pocket to the radius to the innermost pocket at the moment fluid delivery begins, i.e., the moment the inner pocket opens. Therefore, in the design of prior art scroll-type apparatus another important approach to the obtaining of compression ratios greater than about two has been to construct the scrolls and their end plates to resemble very large flat pancakes. In contrast, the scroll apparatus of this invention possesses features making it possible to reduce the outside diameter of the scroll members while attaining desired compression ratios. Among such features are wraps which are configured at their inner ends to delay delivery of fluid into a receiver, wraps having a transition between a double scroll to a single scroll pattern, and special types of porting.

The resulting solutions to the sealing, wearing and porting problems through these and other approaches have not been satisfactory. Thus in the prior art devices, the inherent advantages of scroll-type apparatus (simplicity, high efficiency, flexibility, reversibility, and the like) have not been attained and have, in fact, been usually outweighed by sealing, wearing and porting proglems. It would therefore be desirable to be able to construct scroll-type fluid displacement devices which could realize the inherent advantages of this type of apparatus and which could be essentially free of sealing, wearing and porting problems heretofore encountered.

It is therefore a primary object of this invention to provide improved practical and useful fluid-displacement apparatus which may serve as compressors, expanders or pumps. It is another object of this invention to provide apparatus of the character described which are of the so-called scroll type and which achieve efficient axial and radial sealing over extended operating periods. It is a further object to provide a fluid displacement apparatus which is simple and relatively inexpensive to construct, which has relatively few moving parts and a limited number of rubbing surfaces, and which experiences less friction and wear than other types of apparatus designed for the same purpose. Still another object is to provide such apparatus wherein wear is essentially self compensating.

Another primary object of this invention is to provide fluid displacement apparatus which, as a compressor or an expansion engine, is capable of handling a wide variety of fluids over a large temperature range. Yet another object of this invention is to provide a small, versatile, compact and quiet compressor to achieve compression ratios up to about ten to supply compressed air for such uses as dentist's drills, garage requirements, automobile air conditioner etc. Still another object is to provide an efficient, simple, liquid pump usable in hydraulic systems, and the like. Other objects of the invention will in part be obvious and will in part be apparent hereinafter.

The scroll apparatus of this invention incorporates a unique driving means which permits reducing the radial constraints within the apparatus to only those imposed by the moving line contacts between the surfaces of the wraps forming the fluid pockets. The unique driving means is characterized in part by including means to counteract a fraction of the centrifugal force on the moving scroll member with an inwardly directed radial (centripetal) force. There is thus provided a contacting, i.e., radial sealing, force which minimizes wear and which is independent of the functioning of such other apparatus components as the means to maintain the desired angular relationship between the scroll members and the axial sealing means. Axial sealing is preferably attained through pressurized fluid withdrawn from the zone of highest pressure and a spring biased to exert a force on one of the scroll members to urge it against the other scroll member.

An exemplary embodiment of the apparatus of this invention incorporates a unique scroll driver as the driving means. The scroll driver is fixed through bearings to the main drive shaft while the moving scroll member is free to float axially to respond to fluid pressure acting upon its outer surface to attain axial sealing. The scroll driver effects the orbiting of the movable scroll member by making a rolling line contact between its cylindrical surface and a drive surface assoclaated with the movable scroll. By maintaining the orbit radius of the scroll driver less than the orbit radius of the movable scroll, the required opposing centripital force is provided in the driving means.

Valved porting where required is provided in the fluid displacement apparatus of this invention to better control the flow of fluid in and out. Valved porting generally need not be required for liquid pumps or for gas compressors and expanders wherein the pressure ratios are small or when a "pancake" geometry is acceptable. A wide range of scroll designs may be used to achieve a variety of desired results such as different compression ratios, control of fluid volume at the time of discharge, overall size of the apparatus and the like. The apparatus of this invention is readily reversible from a compressor to an expansion engine and it is capable of handling a wide variety of fluids over a wide temperature range. Many of the embodiments illustrated may also be used as pumps for liquids.

The invention accordingly comprises the features of construction, combinations of elements, and arrangement of parts which will be exemplified in the constructions hereinafter set forth, and the scope of the invention will be indicated in the claims.

For a fuller understanding of the nature and objects of the invention, reference should be had to the following detailed description taken in connection with the accompanying drawings in which

FIGS. 1–4 are diagrams of exemplary spiral wraps, one moving in a circular orbit with respect to the other, illustrating the manner in which a device incorporating such spiral members can achieve compression of a gas;

FIG. 5 is a combination of a top plan view and two side elevational views of spiral member illustrating the principal forces to which the scroll is subjected;

FIGS. 6A and 6B are diagrams showing forces in the radial-tangential plane which act upon a spiral member and illustrating the mechanism by which radial sealing forces are developed;

FIG. 7 is a cross section through the scroll driver and the moving and stationary scroll members illustrating the manner in which axial sealing is attained;

FIG. 8 is a side elevation view, partly in cross section, of one embodiment of a compressor constructed in accordance with this invention;

FIG. 9 is a cross section of the compressor of FIG. 8 through the plane 9—9 of FIG. 8;

FIGS. 10 and 11 are top planar and bottom planar views of the fixed spiral member of the compressor of FIG. 8;

FIG. 12 is a top planar view of the moving spiral member of the compressor of FIG. 8;

FIGS. 13, 14 and 15 are top planar, cross sectional and end views of one embodiment of a coupling member designed to prevent the relative angular motion of the scroll members during the orbiting of the movable scroll member;

FIGS. 16 and 17 are top planar and bottom planar views of the frame of the compressor of FIG. 8;

FIG. 18 is a cross section of the main shaft and bearing spacer taken through plane 18—18 of FIG. 8;

FIG. 19 is a cross section of the main shaft and balance weights taken through plane 19—19 of FIG. 8;

FIG. 20 is a partial side elevation view, partly in cross section, of another embodiment of a compressor constructed in accordance with this invention;

FIG. 21 is a cross section of the compressor of FIG. 20 through the plane 21—21 of FIG. 20;

FIGS. 22 and 23 are top planar and cross sectional views of another embodiment of a coupling member such as used in the compressor of FIGS. 20 and 21;

FIG. 24 is a cross section through the scroll members of an embodiment of a compressor constructed in accordance with this invention in which each scroll member has two wraps;

FIG. 25 is a cross section through the scroll members of an embodiment of a compressor constructed in accordance with this invention in which each scroll member has two outer wraps which are transformed into single inner wraps;

FIg. 26 is a cross section through the scroll members of an embodiment of a compressor constructed in accordance with this invention in which the innermost ends of the wraps are configured to use a cross-shaped coupling;

FIG. 27 is a perspective view of a cross-shaped coupling.

FIG. 28 is a perspective view of a portion of the innermost end of a wrap of FIG. 26 showing the channel which engages one arm of the cross-shaped coupling;

FIG. 29 is a cross section through the scroll members of an embodiment of a compressor constructed in accordance with this invention in which the wraps are configured to provide a modified porting arrangement;

FIGS. 30 and 31 are perspective and cross section views of the coupling used in the apparatus embodiment of FIG. 29;

FIG. 32 is a cross section through the scroll members of an embodiment of a compressor constructed in accordance with this invention in which the wraps are circular arcs rather than involute spirals;

FIG. 33 is a cross section through the scroll members of an embodiment of a compressor constructed in accordance with this invention in which the wraps are configured so that the clearance volume at the start of delivery of compressed gas is under control of a central port to attain intermittent porting;

FIG. 34 is a cross section across plane 34—34 of the apparatus of FIG. 33;

FIGS. 35–37 are cross sections through the wraps of the scroll members of FIG. 33 illustrating the operation of the apparatus;

FIG. 38 is a cross sectional view of a compressor constructed in accordance with this invention in which the scroll members rotate on parallel axes;

FIG. 39 is a cross section through the driving scroll of FIG. 38 taken through plane 39—39 of FIG. 38; and

FIG. 40 is a side elevational view of a compressor constructed in accordance with this invention including housing, heat transfer means, etc.

Before describing specific embodiments of the apparatus of this invention, the principles of its operation may be discussed briefly in order to understand the way in which positive fluid displacement is achieved. The scroll-type apparatus operates by moving a sealed pocket of fluid taken from one region into another region which may be at a different pressure. If the fluid is moved from a lower to higher pressure region, the apparatus serves as a compressor; if from a higher to lower pressure region it serves as an expander; and if the fluid volumes remain essentially constant, then the apparatus serves as a pump.

The sealed pocket of fluid is bounded by two parallel planes defined by end plates, and by two cylindrial surfaces defined by the involute of a circle or other suitably curved configuration. The scroll members are aligned on parallel axes. A sealed pocket moves along between these parallel planes as the two lines of contact between the cylindrical surfaces move. The lines of contact move because one cylindrical element, e.g., a scroll member, moves over the other. This may be accomplished by maintaining one scroll fixed and orbiting the other scroll or by rotating both of the two scrolls on their parallel axes. In the detailed discussion which follows, it will be assumed for the sake of convenience that the positive fluid displacement apparatus is a compressor and that one scroll member is fixed while the other scroll member orbits in a circular path. The embodiment in which both of the scroll members rotate on parallel axes is shown in FIG. 38.

FIGS. 1–4 may be considered to be end views of a compressor wherein the end plates are removed and only the wraps of the scroll members are shown. In the descriptions which follow, the term "scroll member" will be used to designate the component which is comprised of both the end plate and the elements which define the contacting surfaces making movable line contacts. The term "wrap" will be used to designate the elements making movable line contacts. These wraps have a configuration, e.g., an involute of a circle (involute spiral), arc of a circle, etc., and they have both height and thickness. The thickness may vary over the length of the wrap.

In the diagrams of FIGS. 1–4, a stationary scroll member wrap 10 in the form of an involute spiral having axis 11 and a movable scroll member wrap 12 in the form of another involute spiral of the same pitch as spiral 10 and having axis 13 constitute the components which define the moving sealed fluid pocket 14 which is crosshatched for ease of identification. The involute spirals 10 and 12 may be generated, for example, by wrapping a string around a reference circle having radius R_g. The distance between corresponding points of adjacent wraps of each spiral is equal to the circumference of the generating circle. This distance between corresponding points of adjacent wraps of any scroll member is also the pitch, P. As will be seen in FIG. 1, the two scroll members can be made to touch at a number of points, for example in FIG. 1, the points A, B, C and D. These points are, of course, the line contacts between the cylindrical surfaces previously described. It will be seen that line contacts C and D of FIG. 1 define the cross-hatched pocket **14** being considered. These line contacts lie approximately on a single radius which is drawn through point **11**, thus forming pocket **14** which extends for approximately a single turn about the central region of the scrolls. Since the spiral wraps have height (normal to the plane of the drawings) the pocket becomes a fluid volume which is decreased from FIG. 1 to FIG. 4 as the movable scroll member is orbited around a circle **15** of radius $(P/2)-t$, where t is the thickness of the wrap. Since wrap **12** does not rotate as it orbits, the path traced out by the walls of wrap **12** may be, in addition, represented as a circle **16**. As illustrated in FIGS. 1–4, wrap **10** has a shape characterized by two congruent involute spirals **17** and **18** and wrap **12** has a shape characterized by two congruent involute spirals **19** and **20**. The thicknesses, t, of the spiral walls are shown to be identical, although this is not necessary. As will be shown in the description below of FIGS. 21, 24–26, 29, 32 and 33, the wraps may take a number of different configurations and may vary in the number of turns used.

The end plate (not shown in FIGS. 1–4) to which stationary wrap **10** is fixed has a high-pressure fluid port **21** and as the moving wrap **12** is orbited the fluid pocket **14** shifts counterclockwise and decreases in volume to increase the fluid pressure. In FIG. 3, the fluid volume is opened into port **21** to begin the discharge of high-pressure fluid and this discharge of the high-pressure fluid is continued as shown in FIG. 4 until such time as the moving wrap has completed it orbit about circle **15** and is ready to seal off a new volume for compression and delivery as shown in FIG. 1.

If high-pressure fluid is introduced into the fluid port **21**, the movable scroll **12** will be driven to orbit in a clockwise direction under the force of the fluid pressure and will deliver mechanical energy in the form of rotary motion as it expands into fluid pockets of increasing volume. In such an arrangement the device is an expansion engine.

Although this principle of the operation of scroll apparatus has long been known as evidenced by the prior art, the attainment of practical scroll equipment in a form which would permit the use of such apparatus on a commercial scale has so far not been realized. The failure of prior art scroll equipment to attain its potential has, at least in part, been due to problems of sealing and wearing. More particularly, the scroll devices of the prior art, as far as is known, have not provided an efficient combination of continuous axial and radial sealing; and they have in many cases sought to impose radial constraints on the scroll members by mechanisms other than the line contacts of the wraps themselves while using such mechanisms also to control angular phase relationships between the scroll members. Failing to provide efficient continued axial and radial sealing can materially decrease the efficiency of the apparatus to the point where it is no longer economical to operate. Imposing radial constraints through means other than through the line contacts of the wraps of the scroll members eventually leads to the wearing of the contacting surfaces and then to leakage. Generally such wear will vary from surface to surface and will not be self-compensating, a fact which only serves to aggrevate the problem of wear with continued operation. Combining mechanisms to achieve a desired angular phase relationship between the scroll members with such means to impose radial constraints can add to the problem of wear so that extended operation becomes impractical.

In the apparatus of this invention the disadvantages associated with scroll apparatus of the prior art are eliminated or minimized by limiting the radial displacement constraints to only those imposed by the line contacts of the wraps themselves, by providing a drive force on the movable scroll which has an inwardly directed radial component which opposes at least a fraction of the centrifugal force acting on the movable scroll, and by providing coupling means to control the angular phase relationship of the two scroll members which function independently of any constraint imposing means. The operation of the scroll apparatus remains independent of any pressure events within the scroll; wear of the line contacts between the wraps of the scroll members is essentially self-compensating and hence efficient continued radial sealing is attained over extended periods of operation. Axial sealing is preferably accomplished by using gas from the highest pressure zone of the apparatus in combination with a suitably biased spring to continuously force the scroll members to make axial contact.

To understand the problem of sealing a scroll-type apparatus and to describe the mechanism by which axial and radial sealing is achieved in the apparatus of this invention, it is helpful to examine the principal axial and radial forces acting upon a scroll member. Referring to FIG. 5, it can be shown that in the axial direction, the total external axial force F_a on a scroll pair is the sum of an involute contact sealing force and an internal gas load. Therefore, if an external force is provided which is always greater than the internal axial gas force, axial sealing is accomplished. In the apparatus of this invention, this desired condition is achieved by withdrawing fluid from the highest pressure zone and using it to generate an axial sealing force substantially proportional to the highest pressure within the fluid pockets. Referring to FIG. 5, there must then be a surplus axial force acting upon the scroll which is directed along the arrow F_a. This surplus force performs sealing of the edges of one scroll member against the other, and is opposed by a as force directed against the force F_a shown in FIG. 5.

FIG. 7 illustrates one exemplary embodiment of an apparatus which attains continued axial sealing through the use of gas withdrawn from the highest pressure zone of the apparatus supplemented by the use of a spring biased to force the scroll members toward axial sealing contact. In the embodiment of FIG. 7, which is a cross section through the scroll members and scroll driver, there is illustrated the use of a scroll driver along with a "floating" movable scroll member to achieve continuous self-adjusting axial sealing. The wrap **10** forming the line-contacting surfaces of the stationary scroll member, generally indicated at **25**, is integral with or affixed to a stationary scroll end plate **26** terminating around its peripheral edge in an annular housing **27** which has an annular sealing surface **28.** The wrap **12** forming the line-contacting surfaces of the movable scroll member, generally indicated at **30,** is integral with or affixed to a movable scroll end plate **31,** the peripheral sealing surface **32** of which is adapted to contact annular sealing surface **28** of stationary scroll member **25** to achieve axial sealing of the internal volume **33** defined between end plates **26** and **31.** It will be appreciated that rubbing contact must be completely and continually achieved between sealing surfaces **28** and **32** even as these surfaces experience wear during operation. Furthermore, and more importantly, the axial sealing force is used to force the end plates to make sealing contact with the end surfaces of the wraps of the opposing scroll member such as at **41** and **42** to seal the pockets at these areas of contact. This desired axial sealing is attained through the use of a scroll driver, generally indicated at **35,** in conjunction with movable scroll member **30** which is allowed to "float" under the influence of forces upon it. That is, movable scroll member **30** moves under the influence of the forces upon it until there is sufficient contact to seal the pockets.

Movable scroll member **30** includes, in the embodiment of FIG. **7,** a scroll driver annular housing ring **36** defining a cylindrical volume **37** in which scroll driver **35** is located. Internally, annular ring **36** has a wall **38** normal to the plane of end plate **31,** an annular shoulder **39,** and a pressure sealing surface **40** which is, in effect, the central external surface of the movable scroll member **30.** The scroll driver **35** is generally configured as a piston, and in this embodiment, comprises a ring **45** having an internal annular shoulder **46** for seating a bearing **47** and a central closure plate **48,** the external wall **49** of which faces the driving surface **40.** The ring **45** of the scroll driver has a diameter, D_d, slightly less than the diameter, D_s, of internal wall **38,** thereby defining with it a clearance **50.** The difference in diameters may be expressed as $(D_s-D_d)/D_s$ and it may range from about 0.001 to about 0.2. Ring **45** is contoured to define a peripheral groove **51** suitable for positioning an elastomeric sealing ring **52** between the scroll driver and internal wall **38** of scroll driver housing ring **36.** A preloading spring **53** is designed to exert an axial force on the movable scroll member at those times when the delivery pressure and hence the gas loading force is zero. Preloading spring **53** is positioned to contact annular shoulder **39** and the periphery of scroll driver end plate **48** thereby defining a shallow axial sealing fluid volume **54** between the scroll driver end plate **48** and driving surface **40** of the movable scroll member. Some means, such as hole **44** in spring **53,** must be provided so that the spring does not adventitiously seal off volume **37** from **54,** for these volumes must be in fluid communication at all times. A fluid port **55** provides fluid communication between the zone of highest fluid pressure in volume **33** and sealing volume **54.** Scroll driver **35** is fixed to driver shaft **56** through bearing **47** and the mechanism by which it drives the movable scroll member **30** will be described below.

Inasmuch as the movable scroll member **30** is not rigidly connected to the scroll driver it will be seen that it is free to move axially, i.e., to float. By bleeding high pressure fluid through port **55** into sealing volume **54,** a force F_a which is essentially equal to the internal gas force is provided as the axial sealing force so long as the area of the force applying surface of the scroll driver is sufficiently large. In effect, the fluid pressure within sealing volume **54** forces the movable scroll member away from the scroll driver and against the fixed scroll member to achieve sealing between surfaces **28** and **32,** as well as to effect sealing contact between the wrap edges and scroll member end plates. As these sealing surfaces wear, sealing contact is maintained because of the ability of the movable scroll member to float under the force of the pressure of the fluid in sealing volume **54.** In practice, it is desirable to bias the total end force F_a by means of spring **53** so that F_a does not go to zero even should the differential pressure in the system go to zero. Thus spring **53** provides an axial sealing force at start up and some additional axial sealing force during operation.

Whereas axial sealing is required to seal the end surfaces of the wrap edges to the end plate of the opposing scroll member, radial sealing is required to maintain a seal along the line contacts made by the cylindrical surfaces of the wraps of the scroll members as the movable scroll member is orbited (See for example points A, B, C and D of FIGS. **1–4** which illustrate the shifting positions of such line contacts). The principal forces which determine radial sealing of the scroll members are sketched out in simplified manner in FIG. **5** which deals with the forces on the moving scroll having a single wrap **12a** affixed to an end plate **31a.** As the scroll member is orbited about a path with radius R_{or} it experiences a tangential force F_t and a normal contacting force which is, of course, the centrifugal force, $m\omega^2R_{or}$, where m is the scroll member mass and ω is its angular velocity. This centrifugal force is in excess of that which is required to attain efficient radial sealing, and the magnitude of such excess centrifugal force determines the extent of wear experienced by the contacting cylindrical surfaces of the wraps. In the apparatus of this invention, the driving means associated with the orbiting of the movable scroll member incorporates means to counteract, or oppose, a fraction of the centrifugal force to provide a contacting force which is of a magnitude sufficient to attain effective radial sealing and at the same time is not conducive to excessive wear. Thus the driving means of the apparatus of this invention includes means to provide a centripetal radial force F_r to oppose a fraction of the centrifugal force acting upon the orbiting scroll member. This is in direct contrast to prior art teaching which discloses the use of an augmented centrifugal force to attain radial sealing (See for example British Specification No. 486,192.)

In the practice of this invention the actual fraction of centrifugal force which is counteracted by the centripetal radial force applying means will depend upon several factors which may be interrelated. The optimum balance between centrifugal force, which is never reduced to zero, and centripetal force can be determined for any scroll apparatus by consideration of such factors as the specific application for which the scroll apparatus is used, the use or nonuse of lubricants, the material from which the wraps are made, the speed of operation, the desired life of the apparatus, and the like. For example, a compressor running dry will generally require that a greater fraction of the centrifugal force be opposed than one operating with a suitable lubricant; and a compressor having wraps formed of materials conducive to wear will require that a larger fraction of the centrifugal force be opposed than one having wraps formed of materials which are not as subject to wear. In general, higher operational speeds and longer operational lives dictate that a greater fraction of the centrifugal force be opposed by a centripetal force.

In conjunction with the providing of means to oppose a fraction of the centrifugal force on the orbiting scroll member, the apparatus of this invention is characterized by additional features which make it possible so to regulate the contacting force as to continuously maintain the radial sealing force between the scroll members at a level consistent with minimum wear and minimum fluid leakage. One of these features is the limiting of the radial constraints within the scroll apparatus to the moving line contacts between the wraps. Thus these radial constraints, controlled solely through the centripetal force providing means, not only minimize wear but impart a flexibility to the operation of the apparatus such that a great part of any wear that does occur is self-compensating. The limiting of the radial constraints to only the moving line contacts between the wraps is contrary to teaching in the prior art as exemplified by U.S. Pat. No. 3,600,114.

Another important feature of the apparatus of this invention is the use of a coupling means, adapted to maintain a fixed angular relationship between the scroll members, which is separate and distinct from the driving means. By using such a coupling means in combination with the unique driving means of this invention, and by limiting the radial contraints to the moving line contacts between the cylindrical surfaces of the wraps, the apparatus of this invention overcomes, at least to a very large extent, the radial sealing and wear problems of the prior art apparatus.

In the embodiments of the apparatus of this invention illustrated in FIGS. **7–40** the unique driving means, providing a centripetal force to oppose a fraction of the centrifugal force to give the desired contact force and radial sealing, is exemplified by a combination of means to define a cylindrical drive surface associated with the movable scroll member and a scroll driver which defines a cylindrical driving surface adapted to orbit the movable scroll member through line contact with the drive surface. By choosing the orbit radius, R_{or}, of the movable scroll member to be greater than the orbit radius, R_{od}, of the scroll driver, the desired centripetal, inwardly directed radial force opposing a fraction of the centrifugal force is attained.

In the embodiment of this invention using a movable scroll member which provides a cylindrical drive surface in combination with a scroll driver which defines a cylindrical driving surface to provide the desired contacting force, an important feature is that the diameter, D_d, of the scroll drive must be different from the diameter, D_s, of the cylindrical drive surface. In FIG. **6A**, which is, in essence, a cross section taken through plane **6A—6A** of FIG. **7**, the diameter, D_s of the internal wall **38** which serves as the cylindrical drive surface is larger than the external diameter, D_d, of the ring **45** of scroll driver **35** which serves as the cylindrical driving surface. The difference between D_s and D_d is greatly exaggerated in FIG. **6A** better to illustrate the forces involved. Due to the difference in these diameters, the scroll driver, represented in FIG. **6A** by the ring **45**, makes an essentially rolling line contact at L with the movable scroll member, represented in FIG. **6A** by internal wall **38**.

The moving scroll is contained in the radial-tangential plane by forces applied to the internal wall **38** as shown in FIG. **6A**. (For continuity of presentation, the center, C, of the movable scroll driving ring is assumed to contain the origin of the involute of the scroll member as well as the center of gravity of the movable scroll member.) These forces applied to the movable scroll member through wall **38** are seen to be the centrifugal force, $m\omega^2R_{or}$, and the tangential force F_t. By making the orbit radius, R_{od}, of the scroll driver less than the orbit radius, R_{or}, of the movable scroll member there is developed a centripetal force which is an inwardly directed radial force F_r. The magnitude of F_r is a function of the contact angle which in turn is a function of the difference in orbit radii R_{or} and R_{od} as well as of the difference in diameters D_s and D_d. Thus this contact angle θ between the scroll driver and the movable scroll member can be expressed in terms of diameters and orbit radii as

$$\sin\theta = 2(R_{or} - R_{od})/(D_s - D_d). \tag{1a}$$

For a given operational speed and fluid pressure, θ is designed into the apparatus to provide an adequate, but not excessive, radial sealing force which is always less than the centrifugal force $m\omega^2R_{or}$ on the orbiting scroll as discussed previously.

It is also within the scope of this invention to construct the movable scroll member, as illustrated somewhat diagrammatically in FIG. **6B**, to have a cylindrical drive surface **38***a*, such as a shaft in place of the internal wall **38** of the annular ring housing and to use a scroll driver which defines an internal driving wall **45***a*, having a diameter greater than the drive surface **38***a*, in place of the external wall of the scroll driver piston. Thus, in this embodiment D_d is greater than D_s. However, R_{or} remains greater than R_{od} and the various forces which make up the contacting or radial sealing forces are comparable to the reverse situation as a comparison of FIG. **6A** and FIG. **6B** makes evident. In the arrangement of FIG. **6B** the contact angle θ is defined as

$$\sin\theta = 2(R_{or} - R_{od})/(D_d - D_s). \tag{1b}$$

The geometry which regulates the radial sealing force also tends to reduce the effects of manufacturing errors and the wasting away of material through wear. If the full centrifugal force were exerted on the radial sealing lines, excessive scroll wear would result. For example in manufacturing the scroll driver, the orbit center might not precisely coincide with the scroll orbit center; in such case θ would have a component oscillating at the fundamental frequency ω. By making the actual values of the numerator and denominator of equation (1*a*) or (1*b*) some 10 times larger, for example, than the expected manufacturing error, there results less than 10% peak-to-peak a.c. component of radial sealing force compared to its steady value.

FIGS. **8–19** illustrate in detail a compressor constructed in accordance with this invention directly coupled to an electric motor as a driving means. The driving means illustrated is that discussed in conjunction with FIGS. **5–7**. FIGS. **8** and **9** clearly illustrate the absence of any radial constraining means other than the contacting forces developed along the line contacts of the wraps. A number of different embodiments of coupling means which are connected only to the scroll members are illustrated. It will be seen that the scroll members and scroll driver are similar to the components shown in FIG. 7 and like reference numerals are used to identify like components throughout FIGS. 7–19.

As will be seen from FIG. 9, the fixed scroll member **25** has two spiral wraps **60** and **61** defining the line-contacting surfaces, and these wraps terminate at their inner ends in enlarged sections **62** and **63** which are configured to define with similar enlarged inner sections **64** and **65** of spiral wraps **66** and **67** of the movable scroll member an essentially rectangular central fluid pocket **68** which is, of course, the zone of highest fluid pressure. The flat inner faces (e.g., face **69** of enlarged inner section **62** of wrap **60**) of the four inner sections of the spiral wraps make it possible to use a small square-cornered coupling member **70**, the purpose of which is to permit the movable scroll member **30** to orbit without rotating with respect to the stationary scroll member.

Coupling member **70** is detailed in FIGS. **13–15** which are top plan and cross sectional views of the coupling member and an end view of one of the sealing plates, respectively. Since the coupling member is located within the highest-pressure fluid pocket, it must be configured to provide fluid passages within the pocket to communicate with both fluid ports **21** and **55**. It therefore has relatively large cutouts **80** on each side and cutouts **81** and **82** on the top and bottom leaving in effect a solid central piece **83**. Right-angled sealing plates **84** are affixed to each corner and these sealing plates are beveled at **85** to permit the opening of fluid communication between an outer fluid pocket and the inner fluid pocket through the two recesses **86** and **87** cut into the end plate **26** of the fixed scroll member. Thus as shown in FIG. **9**, there is established fluid communication from fluid pocket **90** by way of recess **86** and cutout **80** into the central pocket. It will be appreciated that with the orbiting of the movable scroll, this fluid communication path shifts to use each of the recesses **86** and **87** in turn.

The stationary scroll member **25** of, FIG. **8** (shown in top and bottom plan views in FIGS. **10** and **11**) is formed to have a lip **95**, which provides the sealing surface **28**, and an outer grooved seating ring **96** for engagement with a frame **97** (FIGS. **8**, **16** and **17**) to which it is fixed by means of a series of screws **98** (FIG. **11**) through holes **99**. An inlet fluid port **100** in end plate **26** provides for the intake of fluid, e.g., air, into the apparatus. It will be seen from FIG. **11** that the bottom of the stationary scroll member has a series of radially extending fins **101** to dissipate heat to the atmosphere and to contribute strength to this member. Finally, as will be seen in FIG. **8**, the high-pressure fluid port **21** has an adapter ring **102** suitable for internal threading or other modification to permit ready attachment of a conduit, e.g., flexible hose, thereto.

The end-on view in FIG. **12** of the movable scroll member shows that it has recesses **88** and **89** similar to recesses **86** and **87** of the stationary scroll member and designed for the same purpose.

As will be seen in FIGS. **8**, **16** and **17**, the frame **97** which serves as a support for the drive motor, shaft bearings and stationary scroll member comprises an outer ring **110** engageable with grooved seating ring **96** through screws in threaded holes **111**, a grooved ring **112** adapted to seat the housing of an electric motor and an center inwardly-lipped bearing retaining ring **113**. These rings are joined through a plurality of radial ribs **114**.

The main drive shaft **56** which is the shaft of the rotor **120** of electric motor **121** is mounted in a bearing **122** seated in bearing ring **113**. Main shaft **56** is in axial alignment with the central axis of the stationary scroll member. Shaft **56** terminates in an eccentric shaft section **125** which is axially aligned with the movable scroll member. The two axes are parallel and the distance between them is, of course, R_{od}, the radius of the orbiting circle of the scroll driver. Shaft bearing **47** associated with ececctric shaft **125** is held in spaced relationship with main shaft bearing **122** by means of an eccentric bearing spacer **126** which is fixed to the main shaft by means of pin **127**. (See also FIG. **18**)

In those embodiments wherein one scroll member remains stationary and the other orbits, machine vibrations are minimized by the use of counterweights spinning synchonously in proper phase with the scroll drive motor. In the embodiment of FIG. **8**, as well as in FIG. **19** which is a cross section through the drive shafts, these counterweights are shown to comprise a smaller mass **130** and a larger mass **131** clamped to main shaft **56** by means of bolts **132** and nuts **133**. FIG. **19** shows the general counterweight configurations and means for attachment to the shafts.

The counterweight assembly comprising masses **130** and **131**, and the bolts which tie them together onto shaft **56**, make up an eccentric weight. This weight has a mass m_b and an effective radius R_b in the sense described earlier. The purpose of this counterweight assembly is to provide static balancing of the scroll machine due to the orbiting of the moving scroll member. Of course, there is an unbalanced couple, that is, the system of scroll member and counterweight assembly is statically but not dynamically balanced. Dynamic balance of the whole apparatus is achieved nonetheless by means of the high pressure stage in the compressor unit of FIG. **8**. The dynamic unbalance of the high pressure stage is designed to cancel the dynamic unbalance of the low-pressure stage. By this means the entire scroll compressor is balanced both dynamically and statically. Of course, it is also possible to achieve full balancing by means of a second counterweight attached at an angle of 180° to the first. By such means both the radial inertial force and the rotating moments due to the orbiting of one moving scroll member may be balanced out individually.

The motor **121** and its housing **135** may be any suitable sized commercially available rotary drive mechanism. Although the embodiment of FIG. **8** illustrates the compressor shaft being connected directly to the motor shaft, it is of course within the scope of this invention to connect the apparatus through any suitable mechanism including belts, gears and the like. If the apparatus of this invention is to be used as an expander, than shaft **56** will be connected to any suitable energy absorbing means such as a drive shaft, the shaft of a dynamo, and the like.

FIGS. **20–23** illustrate embodiments of compressors constructed in accordance with this invention in which the scroll members have two complete spiral wraps forming the line-contacting surfaces and a circular coupling member. In FIG. **20**, like reference numerals are used to refer to like components shown in FIG. **8**.

In FIG. 21 the stationary scroll member is formed of two involute spiral wraps **140** and **141** and the movable scroll member of two involute spiral wraps **142** and **143.** The coupling means joining the stationary and movable scroll members comprises a ring **144** which has a diameter which is sufficiently great to permit it to surround the surfaces defining the scroll wraps. As shown in detail in FIGS. **22** and **23,** this coupling ring **144** has oppositely disposed internal flat surfaces which define slightly thickened wall sections **145** and **146** in the ring. Channels **147** and **148** are cut into one side (e.g., the bottom side as shown in FIGS. **22** and **23**) and oppositely disposed channels **149** and **150,** the axis of which is normal to the axis of channels **147** and **148,** are cut into the opposite side of ring **144.** As will be seen more clearly from FIG. **21,** wrap **140** of the stationary scroll member has a radial extension **155** affixed thereto or integral therewith which is positioned and sized to fit into channel **147** of the ring. In like manner, radial extension **156** affixed to wrap **141** fits into channel **148,** radial extension **157** affixed to wrap **142** of the movable scroll is adapted to slidably engage channel **149** and radial extention **158** affixed to wrap **143** is adapted to slidably engage channel **150.** In orbiting, the radial extensions are free to slide within the channels while the scroll members are prevented from experiencing relative angular motion. One or more low-pressure fluid ports **159** are located between ring **144** and the outermost wraps and a central high-pressure port **160** communicates with the high-pressure zone.

FIGS. **24–32** illustrate a number of scroll member embodiments and coupling members suitable for these scroll members. In the drawings showing cross sections of the scroll members and couplings the annular housing **28** (see FIG. **8**) which is an integral part of the stationary scroll member, along with the lip flange, (e.g., **95** of FIG. **8**) is omitted since these components will be identical in cross section to those shown in FIG. **9.** Thus FIGS. **24, 25, 26, 29** and **32** may be considered to be cross sections through the plane **9—9** of FIG. **8** terminating with the internal wall of housing **27** which is lined in and identified by that reference numeral.

The modification of FIG. **24** illustrates one preferred approach to the attainment of a compression ratio of the order of three. Each scroll member has two wraps, each of which makes about one and one-half turns. A greater number of turns is of course possible, but may not be desirable because of the large diameter of housings required. The stationary scroll member wraps **165** and **166** have radial extensions **167** and **168** which engage channels 169 and **170** for slidably engaging the stationary scroll member with the coupling ring **171**; and the movable scroll member wraps **172** and **173** have radial extensions **174** and **175** which slidably engage channels **176** and **177** in ring **171.** As in the case of the embodiment of FIG. **21,** the porting is simple, comprising a high-pressure fluid port **140** centrally located in the end plate of the stationary scroll member and one or more low-pressure fluid ports **150** located under and outside of the coupling ring. Coupling ring **171** in FIG. **24** is configured as a peripheral ring with the channels cut as rectangular slots entirely through it.

The embodiment of FIG. **25** illustrates one way in which acceptable compression ratios may be attained without having to unduly increase the diameters of the scroll members. In this embodiment it will be seen that in each scroll member, two outer wraps are transformed into a single, thick-walled inner wrap. Thus thin-walled stationary scroll wrap **185** makes almost a full turn, but terminates before it approaches the center of the end plate; while stationary thin-walled scroll wrap **186** makes something more than one full turn and then is transformed into a thick-walled wrap section **187** which terminates near the center of the end plate and is cut out to free high-pressure fluid port **140.** In like manner, the movable scroll member has a thin-walled scroll wrap **188** making almost one full turn and a thin-walled wrap **189** which is transformed into a thick-walled wrap section **190.** The effect of the thick-walled wrap sections is to materially decrease the volume of the high-pressure chamber **191** and hence to increase the overall compression ratio.

The coupling in FIG. **25** is similar to that of FIG. **24** in that it comprises a ring **192** formed to have four cutouts to define channels **193–196** through which radial extensions **197** and **198** at the outer ends of wraps **185** and **186** and radial extensions **199** and **200** at the outer ends of wraps **188** and **189** can slide so that the movable scroll member may orbit but not experience angular motion relative to the stationary scroll member.

By delaying the opening of the high-pressure fluid chamber into the high-pressure fluid port, the volume at the beginning of fluid discharge from a compressor may be reduced. This in turn results in the use of a smaller outside diameter device for a given volumetric compression ratio. Such an arrangement is illustrated in FIG. **26.** The stationary scroll member wraps **211** and **212** terminate at their inner ends in enlarged sections **213** and **214** which define oppositely disposed inwardly facing flat surfaces **215** and **216.** Likewise, movable scroll member wraps **217** and **218** terminate at their inner end in similarly configured enlarged sections **219** and **220** defining oppositely disposed inwardly facing flat surfaces **221** and **222** which with surfaces **215** and **216** define a rectangular-like chamber **223** in which the coupling member **224** is located. This coupling member takes the form of a cross, (shown in FIG. **27**) each arm **225** of which serves as a key adapted to slide in a keyway **227** cut through each of the enlarged wrap sections **213, 214, 219** and **220** as shown in perspective detail in FIG. **28** which shows the keyway cut into section **219.** The high-pressure fluid port **228** lies under a portion of the central solid section of coupling member **224.** The diameter of port **228** is such that a minor area of it is always open to high-pressure chamber **223.** Alternatively, the cross-shaped coupling member **224** need not extend the full depth of the wraps. In this case, the keyway such as channel **227** would not extend all the way through the wraps. Furthermore, the high-pressure fliud port **228** could be of a smaller diameter and there would still remain free passage of fluid from the fluid pockets discharging to the delivery port. Low-pressure fluid port **159** is located around the outer edge of the stationary scroll member end plate **26.**

The configuration of the enlarged inner sections of the wraps, the use of a coupling which is located within the high-pressure fluid chamber and the continual partial covering of the high-pressure fluid port achieve the desired results of delaying the opening of the high-pressure fluid chamber and the reduction in volume of this chamber at the beginning of fluid discharge.

The scroll members of FIG. 29 illustrate what may be termed an extreme form of porting achieved by a unique configuration of the inner ends of the wraps and by the use of a square, centrally-positioned coupling means. Each of the stationary scroll member wraps 231 and 232 and of the movable scroll member wraps 233 and 234 terminates at its inner or central end in an enlarged section. Inasmuch as each of these enlarged sections is identical, section 235, terminating in wrap 231, may be described as illustrative of them. It will be seen that the end of the enlarged section which joins the wrap is configured to define what may be described as reversed curves 236 and 237. At its other end, the enlarged section is formed as an extension 238 of the wrap, this extension being faired into flat surface 239 through curve 240. By adjustment of the contours of curve 236 and 237 of one enlarged section and curve 240 of the adjacent enlarged section it is possible to define the range of the size of the fluid passages 241–244 between the enlarged wrap sections and to obtain the degree of porting desired.

Since the coupling 245 is square and makes contact with all four flat surfaces 239, it must be constructed to define fluid flow paths to provide fluid communications to the hgih-pressure fluid port 140 and to the port (e.g., port 55 of FIGS. 7 and 8) leading to the sealing volume associated with the scroll driver. This is done through the design of the coupling shown in perspective and cross sectional views in FIGS. 30 and 31. In essence, this coupling may be described as comprising two pair of oppositely disposed legs 246 and 247 and 248 and 249 formed integral with a square plate 250 positioned midway between the ends of the legs to define a lower fluid passage 251 opening into port 140 and an upper fluid passage 252 opening into port 55.

As will be seen in the modification of FIG. 32, the scroll member wraps may be configured as arcs rather than as involute spirals. In this modification the stationary scroll member wraps comprise arcs 261 and 262 which vary in thickness throughout their length and which define flat inner surfaces 263 and 264 similar to FIG. 9. Likewise, the movable scroll member wraps comprise arcs 265 and 266 terminating in flat inner surfaces 267 and 268. The coupling member 70 is identical to that of FIGS. 8 and 9, and detailed in FIGS. 13–15. Fluid port recesses 86 and 87 are provided in the end plate of the stationary scroll member and fluid port recesses corresponding to 88 and 89 (as in FIG. 12) in the end plate of the movable scroll member are also used to complete the high-pressure fluid flow path.

The modification of FIGS. 33–37 is designed to achieve high compression ratios with relatively small outside diameters. The porting of this modification is designed to open the high-pressure fluid port intermittently and the scroll drive is loaded by gas pressure intermittently when the high-pressure port is opened. Through this arrangment, the axial load can be a function of the angular position of the scroll driver if desired as well as a function of the delivery pressure.

As will be seen in FIG. 33, the stationary scroll member wrap 270 is an involute spiral of one turn with a semicircular skirt 271 having an edge which is an involute curve (shown by dotted line in FIG. 33) at its base and two radial extensions 272 and 273 to engage channels in coupling ring 274 which is similar to the coupling ring 192 of FIG. 25. The end plate 275 of the stationary scroll member along with annular member 276 and flange 277 form part of a housing which defines a chamber 278 in which the movable scroll member and scroll driver are located. An off-center high-pressure fluid port 279 provides fluid communiation between the highest pressure zone of the apparatus and any attached fluid conduit (not shown); and a peripherally located low-pressure fluid port 280 provides fluid communiation with the surrounding atmosphere on some low-pressure fluid reservoir (not shown).

The movable scroll member has a wrap 281 which terminates at its inner end in an enlarged section 282, the cross sectional configuration of which is defined by a circle and an involute curve. Wrap 281 has a skirt 283 of involute outline at its base and two radial extensions 284 and 285 for engagement with coupling 274. The enlarged central section 282 of the movable scroll member wrap has a narrow fluid passage 286 extending throughout its height and a high-pressure fluid passage 287 which, as is seen in FIG. 34 extends for a short distance upwardly into enlarged section 282 form its contacting surface. This fluid passage 287 has a scalloped edge and a curved edge, the purpose of which will become clear from the description of the operation of this apparatus.

The scroll driver 35 and its associaed bearing 47, preloading spring 53, ring seal 52, and sealing volume 54 are similar to those shown and described in FIGS. 7 and 8.

The main housing is completed by housing member 288 which is affixed to flange 277 by suitable means such as bolts 289 and which has a central shaft housing extension 290. The scroll driver shaft 291 with axis 291a terminates in an eccentric shaft 292 with axis 292a, the distance between these parallel axes being R_{od}, the radius of the scroll driver. Shaft 291 is mounted for rotation in shaft housing 290 through bearings 293 and 294 which are held in spaced relation by bearing spacer ring 295 and rotary shaft seal 296. A bearing retaining ring 298 holds bearing 293 and 294 in place.

In contrast to the arrangement of FIG. 8, the movable scroll member does not make an orbiting rubbing seal with the housing portion of the stationary scroll member. Rather, the seals are formed between the end surfaces of the wraps and the skirts 271 and 281 as shown in FIG. 34. The low-pressure fluid port 280 opens into chamber 278 and enters the low-pressure pockets defined by the scroll wraps through passages between the wraps and the skirts, such as passage 299.

FIGS. 35–37 along with FIG. 33 illustrate four different positions occupied by the orbiting movable scroll member relative to the stationary member. These figures, in which like reference numerals are used to identify like components, illustrate the manner in which intermittent porting in attained. It will be assumed for the purpose of this description of the operation of the device that it is working as a compressor. Beginning with FIG. 33, which represents the relative wrap positions shortly after high-pressure chamber 300 has sealed off a freshly-loaded pocket of inlet fluid, it will be noted that the enlarged section 282 of the wrap of the movable scroll member covers high-pressure fluid port 279. In FIG. 35, which represents wrap positions immediately prior to the uncovering of port 279, the volume 300 has decreased, thus increasing the fluid pressure. Volume 300 continues to decrease as fluid passage 286 begins to uncover port 279, then opens it completely (FIG. 36) until volume 300 reaches zero volume (FIG.

37) at which time fluid port **279** is again closed. After the complete closing of port **279**, the orbiting of wrap **281** closes off chamber **301** which is open to the low-pressure side so as to form high-pressure chamber **300** to begin the cycle anew.

The intermittent loading of the high-pressure sealing volume **54** through passage **286** is achieved during that period of the cycle when passage **286** is in fluid communication with high-pressure port **279.** Such a condition is shown in FIG. **35.** During the remaining portion of the cycle, high-pressure fluid is stored in sealing volume **54** until such time as it receives a new charge of pressurized gas. This intermittent porting into the sealing volume has the advantage of making it possible to directly vary the axial sealing forces with the axial gas forces exerted by the movable scroll and hence to maintain the difference between these two opposing forces approximately constant. Moreover, it will be seen that the phase of the drive shaft with respect to the open condition of port **286** is subject to choice, thereby making it possible to achieve a balance between the axial gas force and the loading force as previously discussed.

The apparatus of FIGS. **33–37**, although possessing distinct advantages as a gas compressor or expander, is not suitable as a liquid pump because of the intermittent porting feature.

As pointed out in the general description of this invention, the desired relative motion of the cylindrical surfaces defining the variable volume fluid chambers of the positive fluid displacement device may be attained by maintaining one scroll member stationary and orbiting the other without effecting any relative angular motion, or by spinning both of the scroll members about their respective parallel axes without effecting relative angular motion of the scroll members. The apparatus illustrated in the preceding figures incorporate the first of these operational principles.

The apparatus of FIGS. **38** and **39** operates using the second of these principles. Inasmuch as the scroll wraps and couplings may take any of the forms previously illustrated, these components need not be further described. Thus the scroll members wrap and coupling of FIG. **20** are used to exemplify these components in FIG. **38** and the same reference numerals are used to identify like components as illustrated in FIG. **20.**

The stationary scroll member of the previous embodiments becomes a driving scroll member, generally indicated at **304**; and the movable scroll member becomes a driven scroll member, generally indicated at **305.** Driving scroll member **304** has an end plate **306** and an annular outer wall **307** which terminates in an annular sealing surface **308** (identical in function to sealing surface **28** of FIG. **8.** End plate **306** of driving scroll member **304** is affixed to or formed integral with a scroll shaft **309** which has a central low-pressure fluid passage **310** extending throughout its length to within the point where the shaft joins end plate **306.** Passage **310** is the low-pressure fluid inlet (or outlet) conduit and therefore it is necessary to provide means for passage **310** to communicate with the outer low-pressure chambers. This is done through transverse fluid passage **311** located in web **312** on the underside of end plate **306.** Transverse passage **311** terminates in one or more ports, e.g., ports **313** and **314** drilled in end plate **306** as seen in FIG. **39.**

Driving scroll shaft **309** is mounted for rotation in main frame section **315** and is supported and aligned through bearings **316** and **317** which are separated by bearing spacer **318.** An externally threaded section **319** of the shaft permits a threaded ring **320** to be used as a bearing retainer ring. A pulley housing and shaft frame **321** is affixed to frame section **315** and driving scroll shaft **309** terminates within an extension **322** of this housing. An elastomeric ring **323** and a ring retaining member **324** are provided to seat the shaft. Housing extension **322** has a threaded fluid passage **325** aligned with shaft passage **310.** This threaded passage is adapted for connection with any suitable conduit if such a conduit is required. A pulley **326** is mounted on shaft **309** to drive the shaft and the driving scroll **304.** This may be accomplished by a motor **327** through another pulley **328** and V-belt **329.** It is, of course, within the scope of this invention to use any other suitable means for rotating shaft **309,** such as gearing.

The driven scroll member **305** is identical in construction with the movable scroll member previously described, having an end plate **335** with a peripheral sealing surface **336** and an annular ring extension **337** which defines a volume in which the scroll driver, generally indicated at **338,** is located. Elastomeric ring **52** and preloaded spring **53** are identical in function as previously described and scroll driver **338** differs from that previously described in that it is affixed to or integral with scroll driver shaft **339** which has a high-pressure fluid passage **340** extending throughout its entire length. Shaft **339** is mounted for rotation in main frame section **341** and is supported and aligned through bearings **342** and **343** maintained in spaced relation by bearing spacer **344.** An externally threaded section **345** of shaft **339** permits threaded ring **346** to serve as a bearing retaining means. A shaft housing **347** is affixed to the end of frame section **341** and scroll driver shaft **339** terminates within an extension **348** of this housing. An elastomeric ring **349** and a ring retaining member **350** are provided to seal shaft **339.** Housing extension **348** has a threaded fluid passage **351** aligned with shaft passage **346.** This threaded passage is adapted for connection with any suitable conduit.

High-pressure fluid passage **340** opens into sealing volume **355** which has a function identical with sealing volume **54** of FIG. **7.** High-pressure fluid port **356** provides fluid communication between the high pressure chamber **357** and sealing passage **355** as well as high-pressure fluid passage **340.**

The main frame sections **315** and **341** may be constructed similarly to frame **97** (FIGS. **8** and **20**) with ribs and heat transfer surfaces and they may be joined in any suitable manner such as with bolts **358.**

In an alternative arrangement, the low-pressure port into the scroll members may be the volume **359** defined by frame sections **315** and **341.** In such a modification, a low-pressure fluid port into volume **359** may be defined as a passage between the frame sections. It, will, of course, be necessary to join the frame sections through a gasket or otherwise make volume **359** fluid-tight. In this alternative arrangement passage **310** and **311,** ports **313** and **314,** shaft sealing ring **323** and annular outer wall **307** of scroll member **304** ar eliminated. The bearings must then operate in the fluid being handled unless appropriate shaft seals are provided to protect them. In many cases such shaft seals would not be required.

In the operation of the apparatus of FIG. **38** driving scroll member **304** is rotated about its axis **360** and in turn drives driven scroll member **305** about its axis **361** which is parallel with axis **360** and spaced therefrom by a distance equal to the radius R_{od} of the scroll driver. Contact between the wraps of the two scroll members is through a number of contact lines, e.g., A–D of FIG. **1** and the operation of the apparatus is the same as that described in conjunction with FIGS. **1–4.** It will be seen that the driven scroll member **335** still floats with respect to scroll driver **338** to achieve axial sealing by the same principle described. Likewise, the diameter D_d of the scroll driver is less than the diameter D_s of the internal wall of annular ring extension **337** of driven scroll **335** to provide the radial sealing load required.

Since each scroll member spins in self-balance, no balance weights (e.g., **130** and **131** of FIG. **8**) are required. This is an inherent advantage of the embodiment of FIG. **38.** Another inherent advantage of this embodiment is the fact that no inertial forces are transmitted through the bearings so that very high rotational speed and a resulting high capacity can be attained.

The roles of the two scroll members may be reversed if desired. For example, the scroll driver may be connected through any suitable means to the motor, thus in effect making scroll member **305** the driving scroll and scroll member **304** the driven scroll.

It is, of course, within the scope of this invention to employ the positive fluid displacement apparatus in a multistaged device. Exemplary of such a multistaged system in the two-staged air compressor illustrated in FIG. **40.** No attempt has been made in FIG. **40** to show the scroll members, scroll drivers, etc., in detail since these components may be any one of the embodiments and modifications previously described. Rather, FIG. **40** shows a complete compressor system with auxiliary components.

In the apparatus of FIG. **40** a first or low-pressure stage **370,** mounted in frame **371** which is in turn supported by cover **372** of housing **373,** has an intake air filter **374** constructed in accordance with any suitable well-known design. Compressed air is withdrawn from first stage **370** through conduit **375** which communicates with the internal passage **376** of a finned heat exchanger **377** serving as an intercooler. The cooled, initially compressed air is then withdrawn from heat exchanger passage **376** via fluid conduit **378** and introduced into the low-pressure side of the second-stage compressor **379,** mounted in frame **380,** by way of intake **381.** The finally compressed air is withdrawn through the high-pressure discharge line **382** and fluid conduit **383** for circulation through internal passage **384** of finned heat exchanger **385** serving as an aftercooler. Finally, the cooled compressed air is withdrawn from heat exchanger **385** by way of conduit **386** into an oil removal sump **387** from where it is directed into any desired conduit (not shown) which may be attached to the compressor discharge port **388.** The oil from sump **387** is recirculated to the first stage by means of line **389.** A single motor **390** drives both stages.

The positive fluid displacement apparatus of this invention is versatile wih respect to its applications (compressor, expansion engine or pump) the types of fluids it can handle and the conditions under which it can operate.

It will thus be seen that the objects set forth above, among those made apparent from the preceding description, are efficiently attained and, since certain changes may be made in the above constructions without departing from the scope of the invention, it is intended that all matter contained in the above description or shown in the accompanying drawings shall be interpreted as illustrative and not in a limiting sense.

We claim:

1. In a positive fluid displacement apparatus into which a fluid is introduced through an inlet port for circulation therethrough and subsequently withdrawn through a discharge port, and in which two scroll members having wraps which make moving line contacts to seal off and define at least one moving pocket of variable volume and zones of different fluid pressure when one of said scroll members is driven to orbit said other scroll member while maintaining a fixed angular relationship therewith, the improvement comprising driving means including means to provide a centripetal radial force adapted to oppose a fraction of the centrifugal force acting upon said other orbiting scroll member, whereby the radial sealing force between said scroll member is maintained at a level to minimize both wear and internal fluid leakage.

2. A positive fluid displacement apparatus in accordance with claim **1** further characterized by having coupling means separate and distinct from said driving means, and wherein the radial constraints within said apparatus are limited to said moving line contacts between said wraps and are controlled through said means to provide said centripetal radial force.

3. A positive fluid displacement apparatus in accordance with claim **2** including axial sealing means comprising means to withdraw a portion of fluid from the zone of highest pressure and scroll member sealing volume means adapted to contain the fluid withdrawn from said zone of highest pressure to generate axial sealing forces which are substantially proportional to said highest pressure.

4. An apparatus in accordance with claim **3** wherein said fluid is a gas and said inlet port is a low-pressure port, whereby said apparatus is a compressor.

5. An apparatus in accordance with claim **3** wherein said fluid is a gas and said inlet port is a high-pressure port, whereby said apparatus is an expansion engine.

6. An apparatus in accordance with claim **3** wherein said fluid is a liquid and said apparatus is a pump.

7. A positive fluid displacement apparatus, comprising in combination

a. two scroll members having wrap means which, when one of said scroll members is orbited with respect to the other, make moving line contacts to seal off and define at least one moving fluid pocket of variable volume and zones of different fluid pressure;

b. scroll orbiting means associated with said one of said scroll members, adapted to effect orbital motion of said one of said scroll members and including means to provide a centripetal radial force adapted to oppose a fraction of the centrifugal force acting upon said one of said scroll members, whereby the radial sealing force between said scroll members is maintained at a level to minimize both wear and internal fluid leakage;

c. coupling means adapted to prevent relative angular motion of said scroll members, said coupling means being separate and distinct from said scroll orbiting means whereby the radial constraints within said apparatus are limited to said moving line contacts between said wraps and are controlled through said means to provide said centripetal radial force;

d. axial sealing means including means to define a sealing volume in pressure applying relationship with said one of said scroll members and means to conduct fluid from the zone of highest pressure into said sealing volume, whereby said one of said scroll members is forced into axial sealing relationship against the other of said scroll members;

e. porting means associated with the zones of lowest and highest pressures and adapted to permit the circulation of a fluid through said pockets; and

f. means to drive said orbiting means.

8. A positive fluid displacement apparatus in accordance with claim **7** including spring means biased to supplement the axial sealing force of said fluid in said sealing volume.

9. A positive fluid displacement apparatus in accordance with claim **7** wherein said scroll orbiting means comprises means defining a cylindrical drive surface associated with said one of said scroll member and having an orbit radius R_{or}, and scroll driver means defining a cylindrical driving surface and adapted to orbit said one of said scroll member through line contact with said cylindrical drive surface associated with said one of scroll members and having an orbit radius R_{od} less than R_{or}, whereby when said scroll driver means is orbited to develop a drive force acting upon said one of said scroll members said centripetal radial force is provided to oppose a fraction of the centrifugal force acting on said one of said scroll members and the difference between said centripetal radial force and said centrifrigal force appears as the only contact force between said scroll members; and wherein said means to drive said orbiting means is connected to said scroll driver means.

10. A positive fluid displacement apparatus in accordance with claim **9** wherein said means defining said cylindrical drive surface comprises housing means having an internal diameter D_s, and said cylindrical driving surface is configured as a piston having an outside diameter D_d which is less than D_s.

11. A positive fluid dispalcement apparatus in accordance with claim **9** wherein means defining said cylindrical drive surface comprises a shaft member having an outside diameter D_s, and said cylindrical driving surface is configured as a cylindrical housing having an internal diameter D_d which is greater than D_s.

12. An apparatus in accordance with claim **9** wherein said other of said scroll members is stationary.

13. An apparatus in accordance with claim **9** including means to rotate said scroll members on separate parallel axes, the distance between said axes being at least as great as the radius of orbit of said one of said scroll members.

14. A positive fluid displacement apparatus, comprising

a. first and second scroll members having wrap means which, when said first scroll member is orbited with respect to said second scroll member, make moving line contacts to define at least one moving fluid pocket of variable volume and zones of different fluid pressure;

b. coupling means adapted to prevent relative angular motion of said first and second scroll members;

c. means defining a cylindrical drive surface having a diameter D_s associated with said first scroll member and having an orbit radius R_{or};

d. scroll driver means defining a cylindrical driving surface adapted to orbit said first scroll member through line contact with said cylindrical drive surface associated with said first scroll member, and having a diameter D_d which is less than D_s and an orbit radius R_{od} less than R_{or}, whereby when said scroll driver means is orbited to develop a drive force acting upon said first scroll member, a component of said drive force opposes a fraction of the centrifugal force acting on said first scroll member, and the difference between said component of said drive force and said centrifugal force appears as the only contact force between said first and second scroll members thereby to effect continuous radial sealing of said fluid pocket;

e. axial sealing means including means to define a sealing volume between said first scroll member and said scroll driver means, conduit means to conduct fluid from the zone of highest pressure into said sealing volume whereby said first scroll member is forced in axial sealing relationship against said second scroll member, and spring means biased to supplement the axial sealing force of said fluid in said sealing volume;

f. means to orbit said scroll driver means; and

g. porting means associated with said zones of highest and lowest pressures.

15. An apparatus in accordance with claim **14** wherein said wrap means are configured as involute spirals.

16. An apparatus in accordance with claim **14** wherein said wrap means are configured as arcs of circles.

17. An apparatus in accordance with claim **14** wherein siad warp means for each of said scroll members comprise two spaced apart involute spirals, one of which terminates short of the center of said scroll member and the other of which terminates at its innermost end in an enlarged section.

18. An apparatus in accordance with claim **14** wherein at least one of said wrap means terminates at its innermost end in an enlarged wrap section thereby to adjust the volume of said zone of highest pressure.

19. An apparatus in accordance with claim **18** wherein each of said wrap means terminates at its innermost end in an enlarged section.

20. An apparatus in accordance with claim **19** wherein said enlarged wrap sections define a central rectangular volume and said coupling means is located within said rectangular volume.

21. An apparatus in accordance with claim **20** wherein said coupling means is of an overall square shape and is configured to provide passages in fluid communication with said conduit means and with said porting means associated with said zone of highest pressure.

22. An apparatus in accordance with claim **20** wherein said coupling means is cross-shaped and said enlarged wrap sections have channels adapted to slidably engage the arms of said cross-shaped coupling means.

23. An apparatus in accordance with claim **14** wherein said wrap means of said first scroll member terminates at its innermost end in an enlarged section and said wrap means of said second scroll member is an involute spiral, said enlarged section having porting means which intermittently vents to said porting means associated with said zone of highest pressure and passage means adapted to serve intermittently as said conduit means.

24. An apparatus in accordance with claim **14** wherein said coupling means comprising an annular ring surrounding said wrap means and defines four equally spaced channels and wherein said wrap means have radial extension means adapted to slidably engage said channels.

25. An apparatus in accordance with claim **14** wherein $(d_s - D_d)/D_s$ ranges between about 0.001 and 0.2.

26. An apparatus in accordance with claim **14** wherein said means to orbit said scroll driver means comprises

a. motor means;

b. main drive shaft means rotatable on a first axis and adapted to be rotated by said motor means; and

c. eccentric shaft means rotatable on a second axis parallel to said first axis and spaced therefrom by a distance no greater than the orbit radius of said one of said scroll members, said eccentric shaft means being adapted to be rotated by said motor means.

27. An apparatus in accordance with claim **14** including weighted mass means associated with said main drive shaft means adapted to substantially balance machine vibrations generated in the orbiting of said scroll driver means.

28. An apparatus in accordance with claim **14** wherein said fluid is a gas and said porting means associated with said zone of highest pressure is the discharge port, whereby said apparatus is a compressor.

29. An apparatus in accordance with claim **14** wherein said fluid is a gas and said porting means associated with said zone of highest pressure is the inlet port, whereby said apparatus is an expansion engine.

30. An apparatus in accordance with claim **14** wherein said fluid is a liquid and said apparatus is a pump.

31. A positive fluid displacement apparatus, comprising in combination

a. two scroll members aligned on parallel axes having end plates with wrap means affixed to facing sides thereof, said wrap means, when one of said scroll members is orbited with respect to the other, being adpated to make moving line contacts to define at least one moving fluid pocket of variable volume and zones of different fluid pressure;

b. coupling means adapted to prevent relative angular motion of said scroll members;

c. cylindrical volume defining means centrally positioned on the outer side of said one of said scroll members and having an internal diameter D_s and an orbit radius R_{or};

d. scroll driver means in the form of a piston having a diameter, D_d, less than D_s and an orbit radius R_{od} less than r_{or}, said scroll driver means being spaced from said outer side of said one of said scroll members thereby to define with sealing ring means a sealing volume disposed between said scroll driver means and said one of said scroll members;

e. conduit means providing fluid communication between said sealing volume and the zone of highest pressure defined by said wrap means thereby to provide a fluid pressure within said sealing volume essentially equal to that in said zone of highest pressure and to generate an axial sealing force between said scroll members substantially proportional to said highest pressure;

f. means to rotate said scroll driver means on an axis parallel to the axis of said other of said scroll members thereby to effect the orbiting of said one of said scroll members through rolling line contact between said scroll driver means and the internal wall of said cylindrically volume defining means; and

g. porting means associated with the zone of lowest pressure and with said zone of highest pressure and adapted to permit the circulation of a fluid through said pockets.

32. An apparatus in accordance with claim **31** including preloading spring means located within said cylindrical volume defining means and biased to force said one of said scroll members against said other of said scroll members.

33. An apparatus in accordance with claim **31** wherein said other of said scroll members is stationary.

34. An apparatus in accordance with claim **31** wherein the end plate of said other of said scroll members includes a peripheral housing wall defining an annular surface adapted to make a fluid seal with the end plate of said one of said scroll members, whereby said scroll members form a housing.

35. An apparatus in accordance with claim **31** including separate housing means defining a fluid volume around said scroll members wherein said porting means associated with said lowest pressure zone comprises spacings between the outer portions of said wrap means.

36. An apparatus in accordance with claim **31** wherein said porting means associated with said zone of lowest pressure is an inlet port and said apparatus is a compressor.

37. An apparatus in accordance with claim **31** wherein said porting means associated with said zone of highest pressure is an inlet port and said apparatus is an expansion engine.

38. An apparatus in accordance with claim **31** including means to rotate said scroll members on said axes, the distance between said axes being no greater than the radius of orbit of said one of said scroll members.

39. An apparatus in accordance with claim **31** wherein said wrap means are configured as involute spirals.

40. An apparatus in accordance with claim **31** wherein said wrap means are configured as arcs of circles.

41. An apparatus in accordance with claim **31** wherein said wrap means for each of said scroll members comprise two spaced apart involute spirals, one of which terminates short of the center of said scroll member and the other of which terminates at its innermost end in an enlarged section.

42. An apparatus in accordance with claim **31** wherein at least one of said wrap means terminates at its innermost end in an enlarged wrap section thereby to adjust the volume of said zone of highest pressure.

43. An apparatus in accordance with claim **42** wherein each of said wrap means terminates at its innermost end in an enlarged section.

44. An apparatus in accordance with claim 43 wherein said enlarged wrap sections define a central rectangular volume and said coupling means is located within said rectangular volume.

45. An apparatus in accordance with claim 44 wherein said coupling means is of an overall square shape and is configured to provide passages in fluid communication with said conduit means and with said porting means associated with said zone of highest pressure.

46. An apparatus in accordance with claim 45 wherein said coupling means is cross-shaped and said enlarged wrap sections have channels adapted to slidably engage the arms of said cross-shaped coupling means.

47. An apparatus in accordance with claim 31 wherein said wrap means of said one of said scroll members terminates at its innermost end in an enlarged section and said wrap means of said other of said scroll members is an involute spiral, said enlarged section having porting means which intermittently vents to said porting means associated with said zone of highest pressure and passage means adapted to serve intermittently as said conduit means.

48. An apparatus in accordance with claim 31 wherein said coupling means comprises an annular ring surrounding said wrap means and defines four equally spaced channels and wherein said wrap means have radial extension means adapted to slidably engage said channels.

49. A positive fluid displacement apparatus, comprising in combination

a. a first scroll member having first wrap means affixed to first end plate means and aligned on a first axis;

b. a second scroll member having an orbit radius R_{or} and second wrap means affixed to one side of second end plate means, said second wrap means being adapted to make at least two moving line contacts with said first wrap means as said second scroll member is orbited relative to said first scroll member thereby to form at least one movable pocket of variable volume and zones of different pressures;

c. cylindrical housing means affixed to the other side of said second end plate means and defining a scroll driving volume, the internal diameter of said cylindrical housing means being D_s;

d. coupling means adapted to prevent relative angular motion of said first and second scroll members;

e. scroll driver means rotatable within said cylindrical housing means and having a diameter D_d which is less than D_s and an orbit radius R_{od} less than R_{or} whereby said scroll driver makes an essentially line rolling contact with said second scroll member to drive it;

f. sealing means providing a fluid seal between said scroll driver means and the internal wall of said annular ring means thereby to define a sealing volume between said scroll driver means and said second scroll member;

g. port means providing fluid communication between the zone of highest pressure and said sealing volume;

h. porting means associated with the zones of highest and lowest pressure and adapted to permit the circulation of a fluid through said pockets;

i. driving means to orbit said scroll driver means about a second axis parallel to said first axis of said first scroll member, the distance between said first and second axes being the orbit radius of said second scroll member, said driving means comprising

1. motor means,
2. main drive shaft means rotatable on said first axis and adapted to be rotated by said motor means, and
3. eccentric shaft means rotatable on said second axis and adapted to be rotated by said motor means; and

j. weighted mass means associated with said main drive shaft means adapted to substantially balance out radial inertial forces generated in the orbiting of said scroll driver means.

50. An apparatus in accordance with claim 49 including frame means adapted to support said first scroll member and said main drive shaft means.

51. An apparatus in accordance with claim 49 wherein said end plate of said first scroll member includes a peripheral housing wall defining an annular surface adapted to make a fluid seal with the end plate of said second scroll member whereby said scroll members form a housing.

52. An apparatus in accordance with claim 49 including separate housing means defining a fluid volume around said scroll members.

53. A positive fluid displacement apparatus, comprising in combination

a. a first scroll member having first wrap means configured as a single-turn involute spiral, the outer half of which is affixed to an involute spiral skirt member, said first wrap member having two oppositely disposed radial extension members;

b. a second scroll member with an orbit radius R_{or} having second wrap means configured as a single-turn involute spiral the outer half of which is affixed to an involute spiral skirt member, said wrap means having two oppositely disposed radial extension members and terminating at its innermost end in an enlarged section, said enlarged section having porting means and passage means extending throughout the thickness of said wrap, said second wrap means being adapted to make at least two moving line contacts with said first wrap means as said second scroll member is orbited relative to said first scroll member thereby to form at least one movable pocket of variable volume and zones of different pressure;

c. cylindrical housing means affixed to said second scroll member defining a scroll driving volume;

d. coupling means comprising an annular ring surrounding said wrap means and having channels adapted for slidable engagement with said radial extensions thereby to prevent any relative angular motion of said scroll members;

e. scroll driver means rotatable within said cylindrical housing means and having a diameter which is less than the internal wall of said cylindrical housing means and an orbit radius R_{od} less than R_{or} whereby said scroll driver makes an essentially line rolling contact with said second scroll member to drive it;

f. sealing means providing a fluid seal between said scroll driver means and said internal wall of said cylindrical housing means thereby to define a sealing volume between said scroll driver means and said second scroll member, said sealing volume being in intermittent fluid communication with the zone of highest pressure through said passage means in said enlarged section of said second wrap means;

g. a high-pressure fluid port in intermittent fluid communication with the zone of highest pressure through said porting means in said enlarged section of said second wrap means;

h. driving means adapted to orbit said scroll driver means about a second axis parallel to said first axis of said first scroll member, the distance between said first and second axes being the orbit radius of said second scroll member, said driving means comprising

1. motor means,
2. main drive shaft means rotatable on said first axis and adapted to be rotated by said motor means, and
3. eccentric shaft means rotatable on said second axis and adapted to be rotated by said motor means;

i. weighted mass means associated with said main drive shaft means adapted to substantially balance out radial inertial forces generated in the orbiting of said scroll driver means; and

j. housing means defining a fluid-tight volume around said scroll members and having low-pressure port means.

54. A positive fluid displacement apparatus, comprising in combination

a. a first scroll member having first wrap means affixed to a first end plate which terminates in a peripheral housing wall defining an annular sealing surface, said first end plate being affixed to and supported by a first shaft rotatable on a first axes;

b. a second scroll member with an orbit radius R_{or} having second wrap means affixed to one side of second end plate means adapted to contact said annular sealing surface, said second wrap means being adapted to make at least two moving line contacts with said first wrap means as said second scroll member is orbited relative to said first scroll member thereby to form at least one movable pocket of variable volume and zones of different pressures;

c. cylindrical housing means affixed to the other side of said second end plate means and defining a scroll driving volume, the internal diameter of said cylindrical housing means being D_s;

d. coupling means adapted to prevent relative angular motion of said first and second scroll members;

e. scroll driver means rotatable within said cylindrical housing means and having a diameter D_d which is less than D_s and an orbit radius R_{od} less than R_{or} whereby said scroll driver makes an essentially line rolling contact with said scroll member to drive it, said scroll driver means being affixed to an supported by a second shaft rotatable on a second axis parallel to and spaced from said first shaft;

f. sealing means providing a fluid seal between said scroll driver means and the internal wall of said annular ring means thereby to define a sealing volume between said scroll driver means and said second scroll member;

g. means to rotate said first shaft and thereby to rotate said first and second scroll members and through said second scroll member to rotate said scroll driver means and orbit said second scroll member;

h. low-pressure porting means adapted to provide fluid communication into the zone of lowest pressure;

i. high-pressure porting means extending from the zone of highest pressure through said second shaft and providing fluid communication between said zone of highest pressure and said sealing volume; and

j. frame means adapted to support said first and second shafts.

55. An apparatus in accordance with claim **54** wherein said low-pressure porting means includes shaft passage means defined within said first shaft and passage means connecting said shaft passage means and the zone of lowest pressure.

56. A postive fluid displacement compressor, comprising in combination

a. at least two compressor stages arranged in series, each of said compressor stages comprising in combination

1. two scroll members having wrap means which, when one of said scroll members, is orbited with respect to the other, make moving line contacts to define at least one moving fluid pocket of variable volume and zones of different fluid pressure,

2. scroll orbiting means associated with said one of said scroll members, adapted to effect orbital motion of said one of said scroll members and including means to provide a centripetal raidal force adapted to oppose a fraction of the centrifugal force acting upon said one of said scroll members, whereby the radial sealing force between said scroll members is maintained at a level to minimize both wear and internal fluid leakage;

3. coupling means adapted to prevent relative angular motion of said scroll members, said coupling means being separate and distinct from said scroll orbiting means whereby the radial constraints within said apparatus are limited to said moving line contacts between said wraps and are controlled through said means to provide said centripetal radial force;

4. axial sealing means including means to define a sealing volume in pressure applying relationship with said one of said scroll members and means to conduct fluid from the zone of highest pressure into said sealing volume, whereby said one of said scroll members is forced into axial sealing relationship against the other of said scroll members;

5. fluid inlet means in fluid communication with the zone of lowest pressure, and

6. fluid discharge means in fluid communication with said zone of highest pressure;

b. means to introduce fluid to be compressed into said fluid inlet means of the first of said compressor stages;

c. means to withdraw compressed fluid from said fluid discharge means of the last of said compressor stages;

d. conduit means connecting said fluid discharge means of each but the last of said stages with said inlet means of that compressor stage which is next in said series;

e. intercooler means associated with said conduit means;

f. aftercooler means associated with said means to withdraw compressed fluid from said fluid discharge means of the last of said stages;

g. means to orbit said scroll driver means of each of said compressor stages.

57. A compressor in accordance with claim **56** wherein at least one of said wrap means in each of said compressor stages terminates at its innermost end in an enlarged wrap section thereby to adjust the volume of said zones of highest pressure.

58. A compressor in accordance with claim **56** wherein said means to orbit said scroll driver means of each of said stages comprises, in combination

1. motor means;

2. main drive shaft means rotatable on a first axis and adapted to be rotated by said motor means; and

3. eccentric shaft means rotatable on a second axis parallel to said first axis and spaced therefrom by a distance no greater than the orbit radius of said one of said scroll members, said eccentric shaft means being adapted to be rotated by said motor means.

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