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McCullough

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[45] **Dec. 9, 1975**

[54] **POSITIVE FLUID DISPLACEMENT APPARATUS**

[75] Inventor: **John E. McCullough**, Carlisle, Mass.

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

[22] Filed: **Oct. 23, 1973**

[21] Appl. No.: **408,912**

Related U.S. Application Data

[63] Continuation-in-part of Ser. No. 368,907, June 11, 1973.

[52] **U.S. Cl.** **418/55**; 418/57
[51] **Int. Cl.**^{2} **F04C 1/02**
[58] **Field of Search** 418/57, 55

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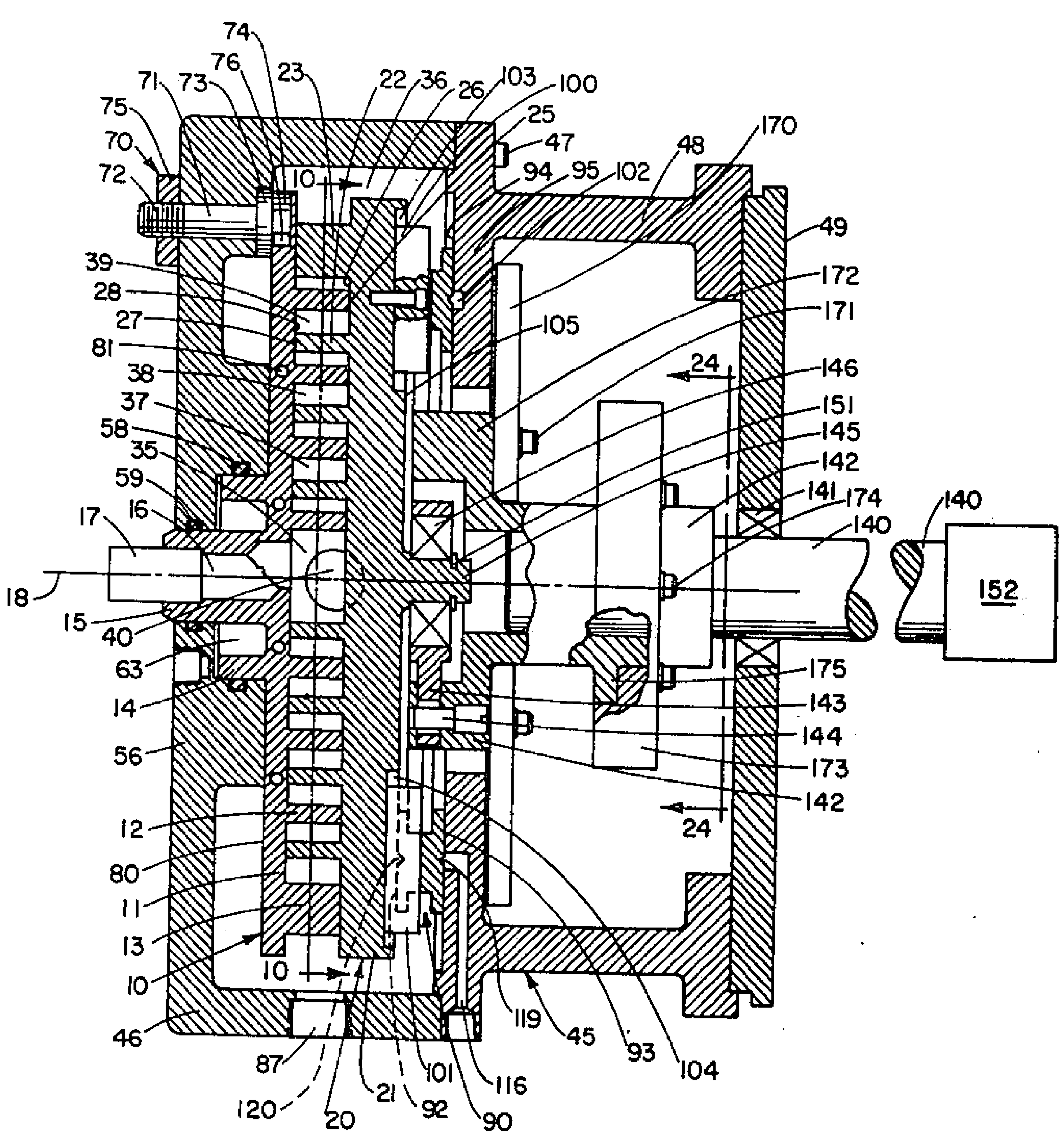
Assistant Examiner—Leonard Smith
Attorney, Agent, or Firm—Bessie A. Lepper

[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 pressure are connected to fluid ports. Radial sealing is accomplished with minimum wear by linking the orbiting scroll member to the driving mechanism through a radially compliant mechanical linking means which also incorporates means to counteract at least a fraction of the centrifugal force exerted by the orbiting of the orbiting scroll member. If essentially all of the centrifugal force is counteracted, then the compliant mechanical linking means is designed to supply the necessary radial sealing force. Coupling means, which are separate from the driving means, and hence from the radial constraining force, are provided to maintain the desired angular relationship between scroll members and to provide one opposing force for axial sealing. The other opposing axial sealing force is a combination of biasing means and fluid pressure. The apparatus may serve as a compressor, expander or pump.

48 Claims, 24 Drawing Figures



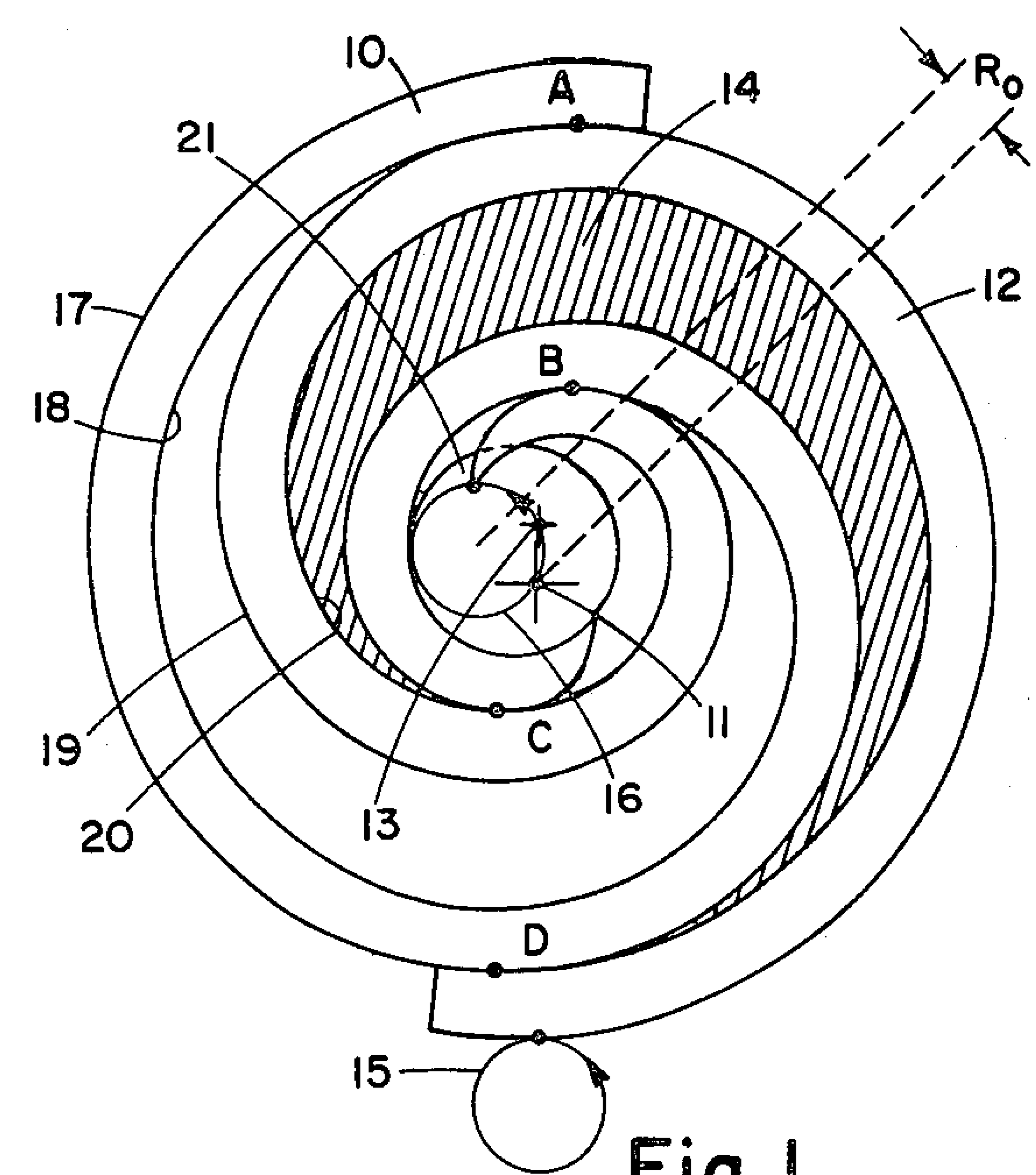


Fig. 1

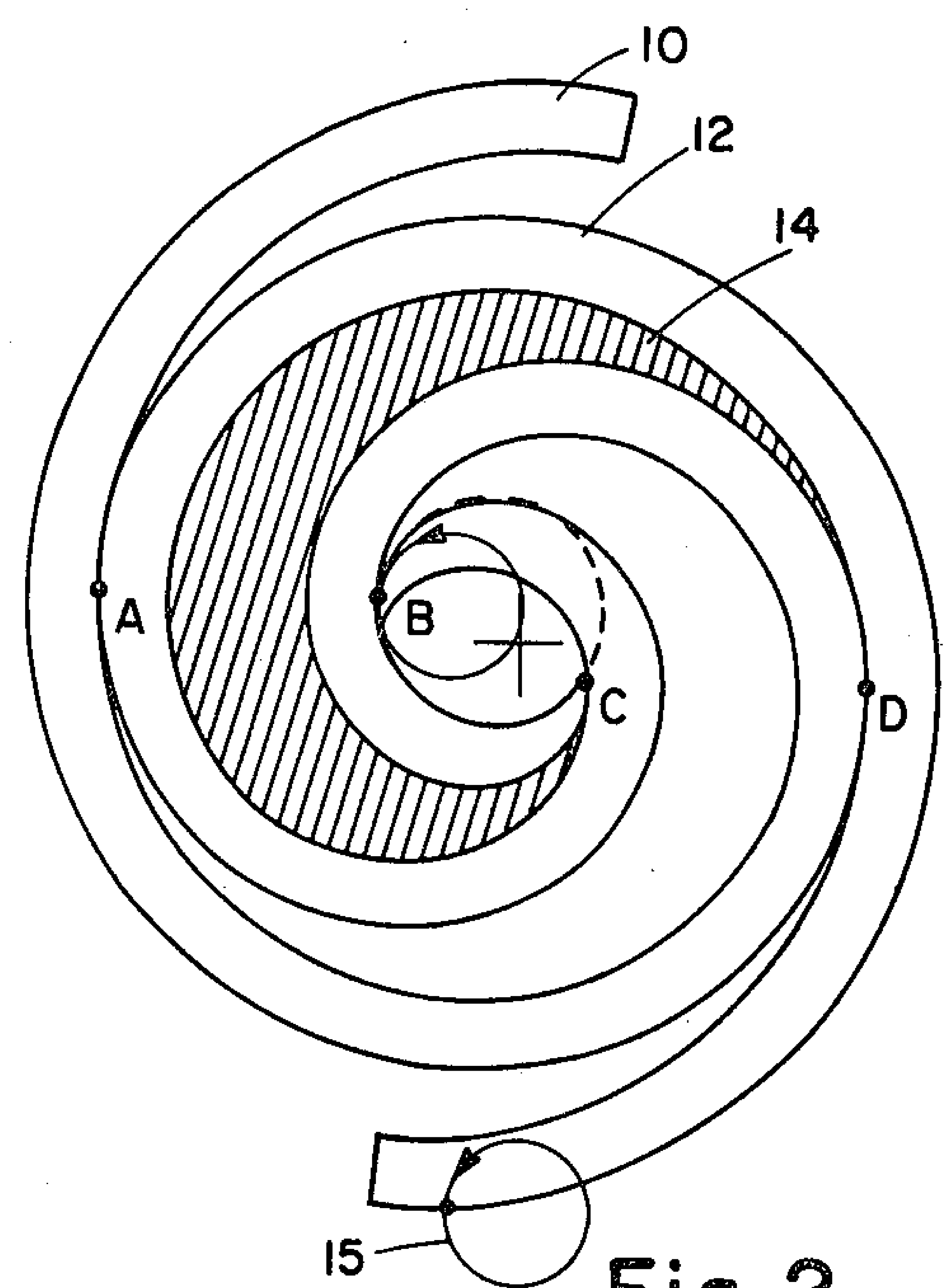


Fig. 2

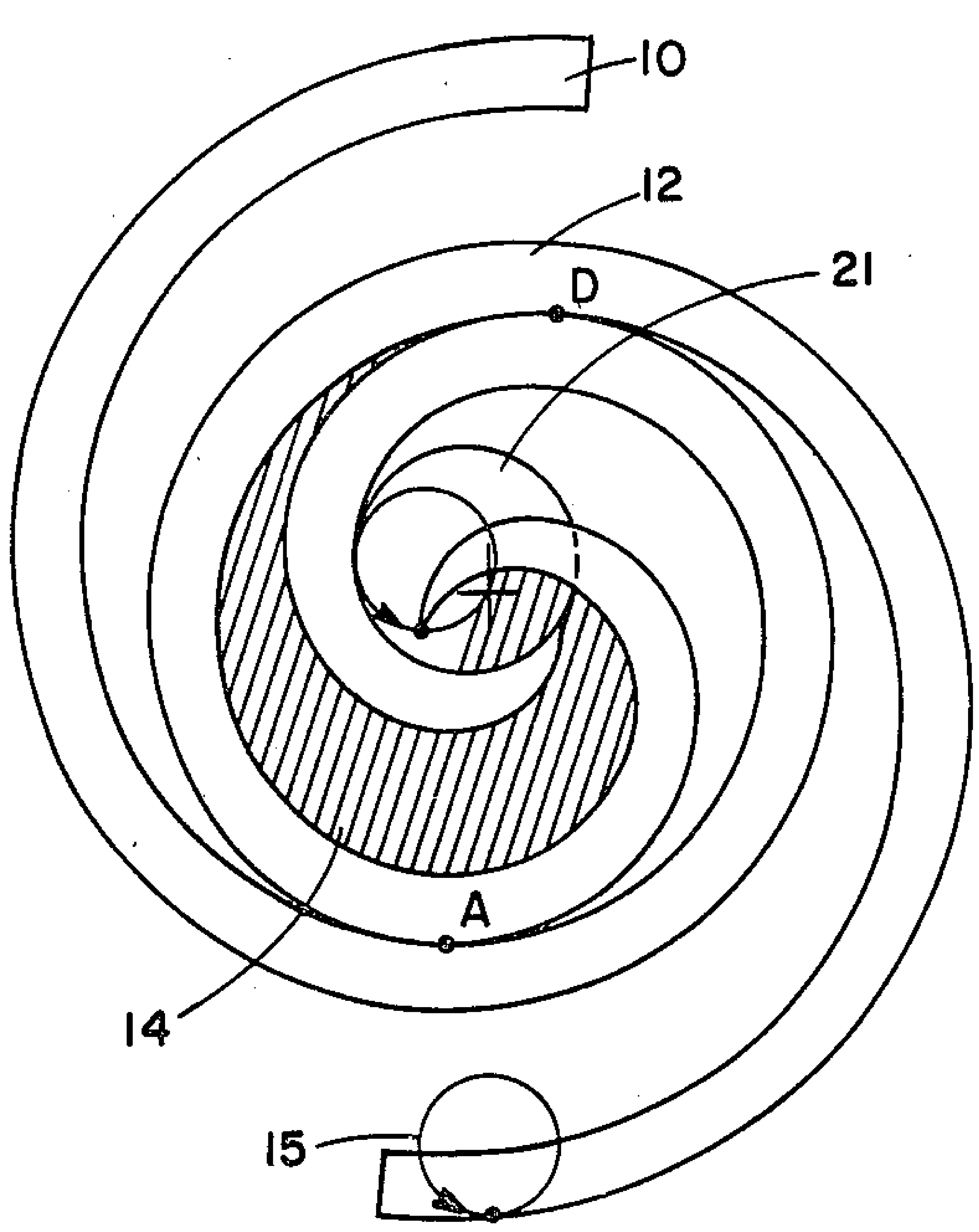
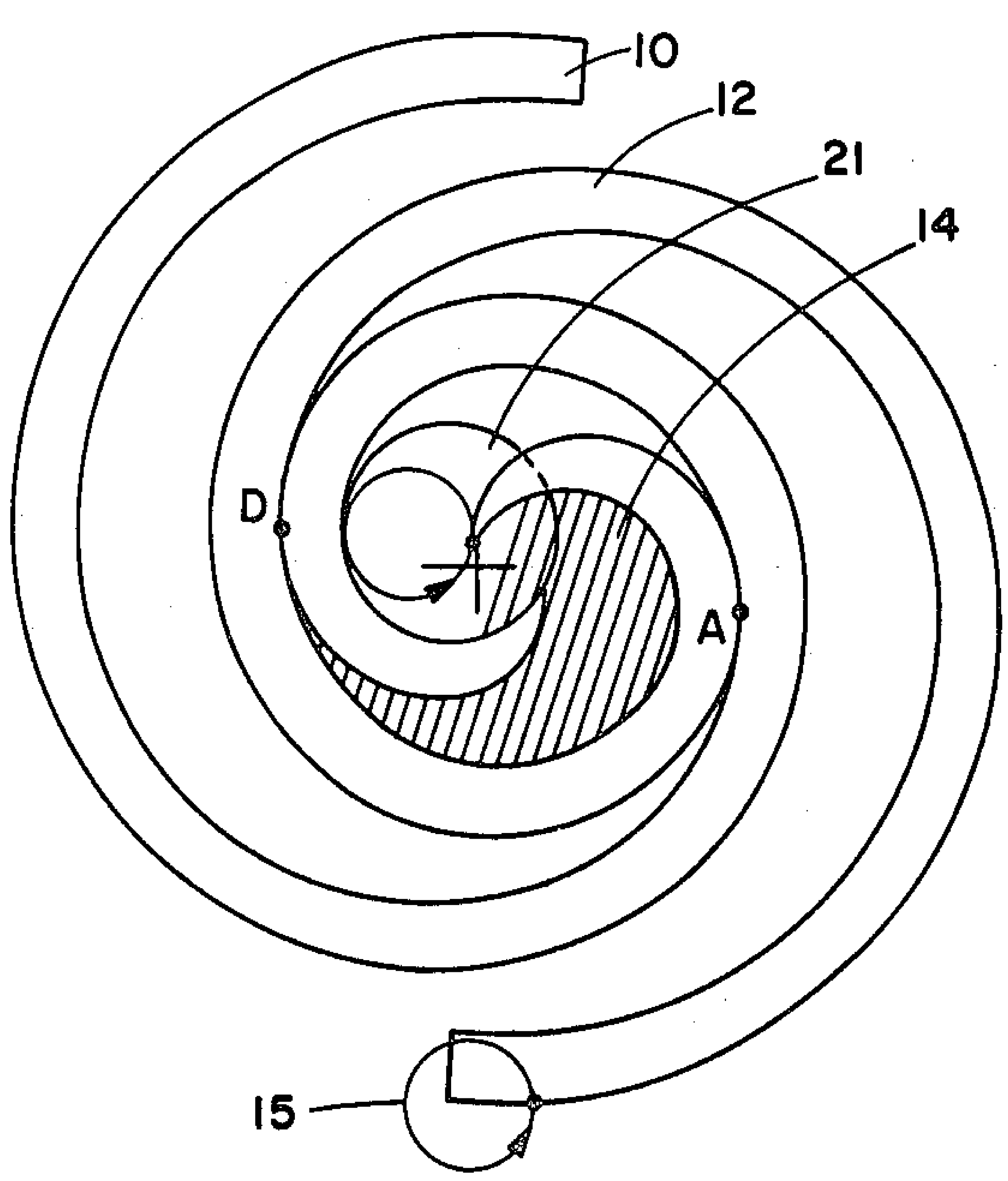


Fig. 3



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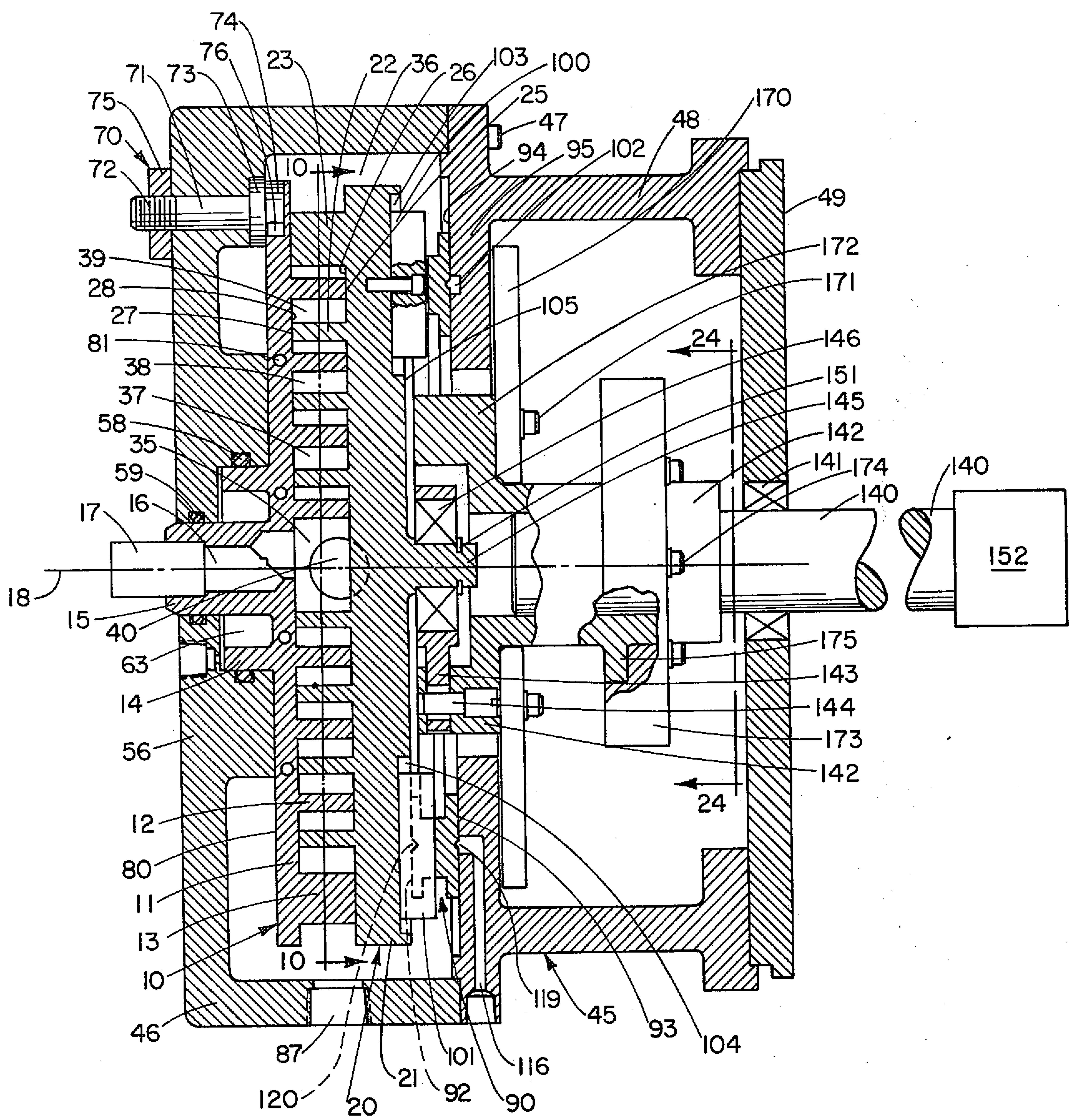
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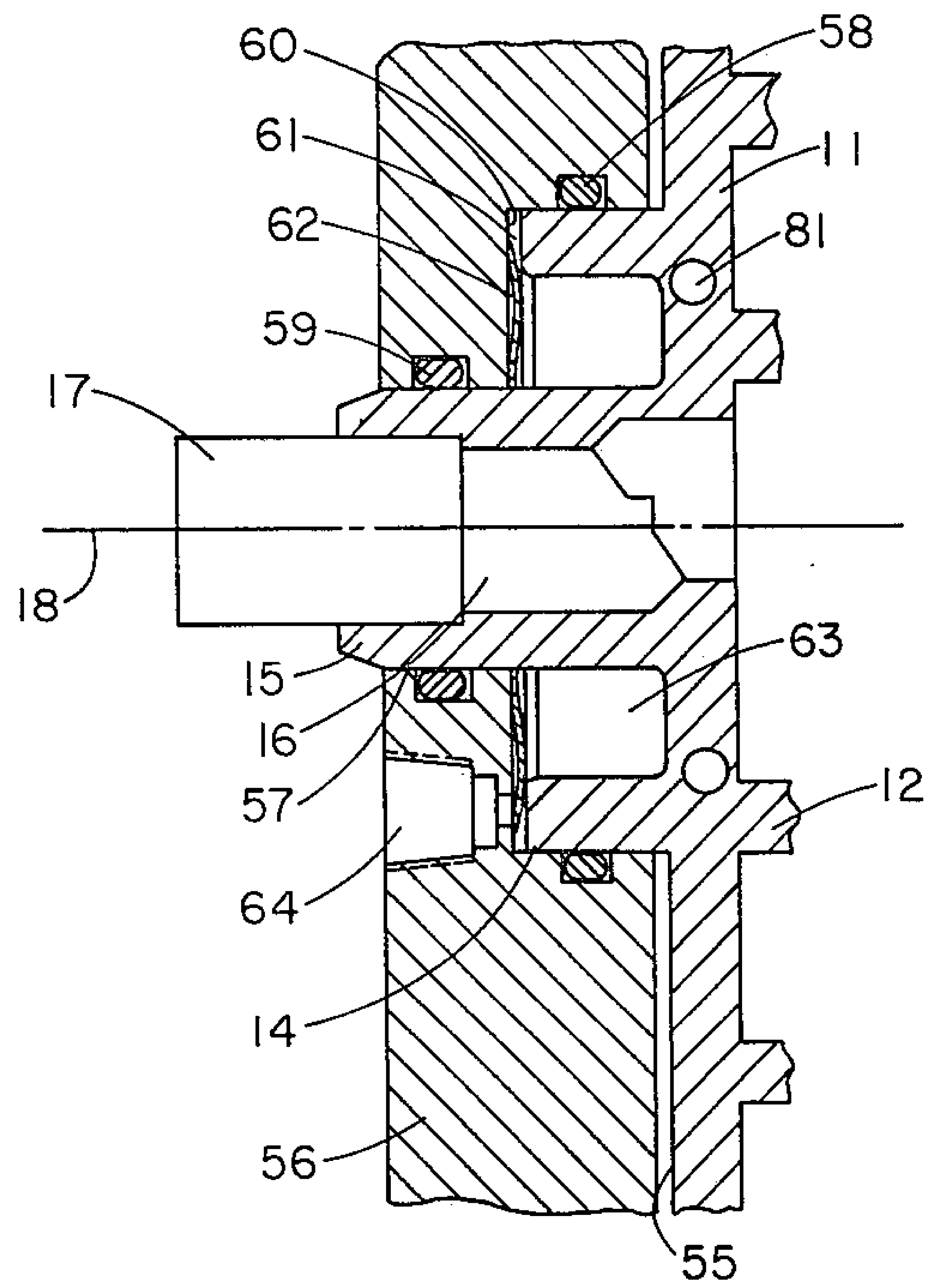
Fig. 5



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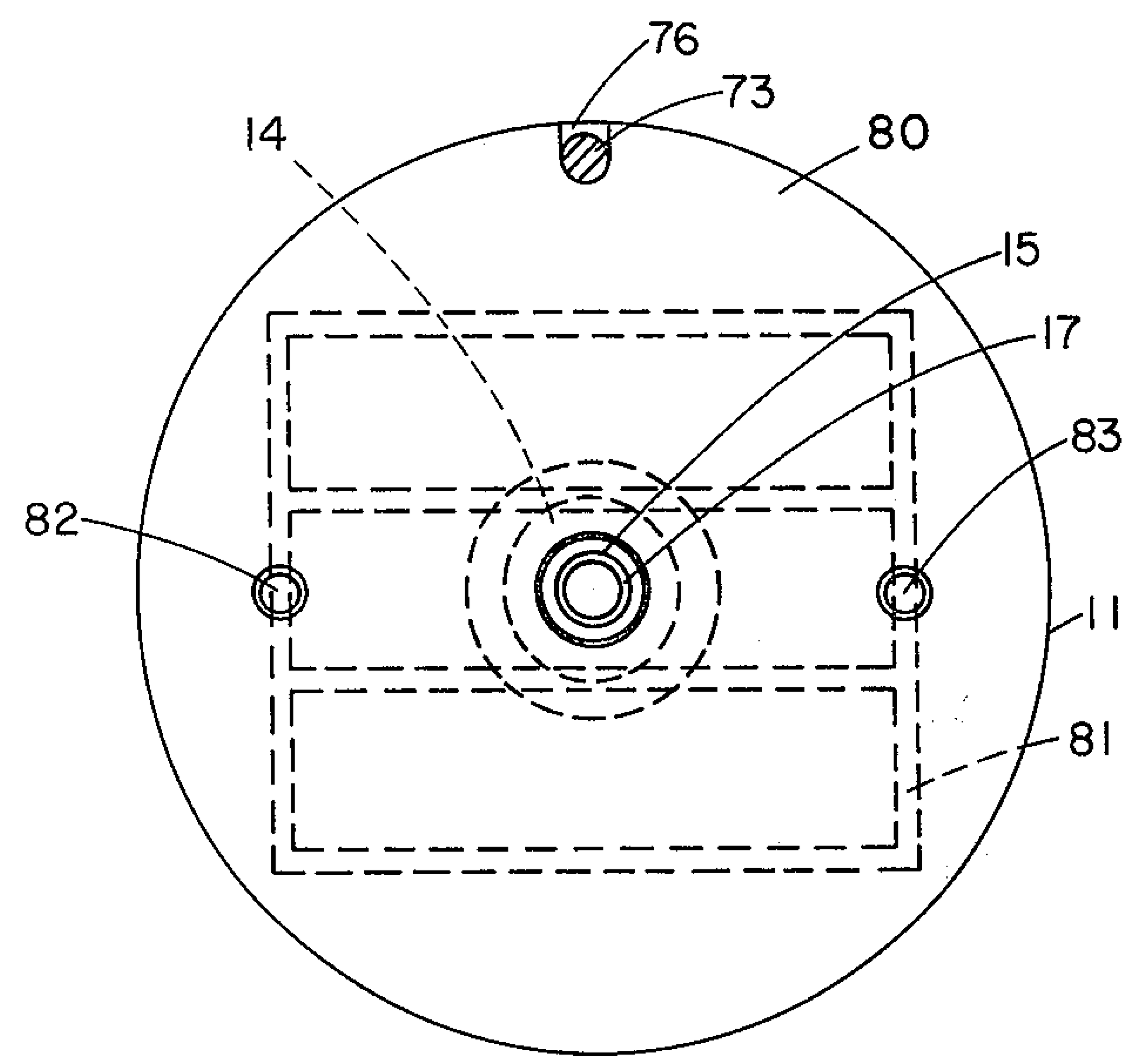

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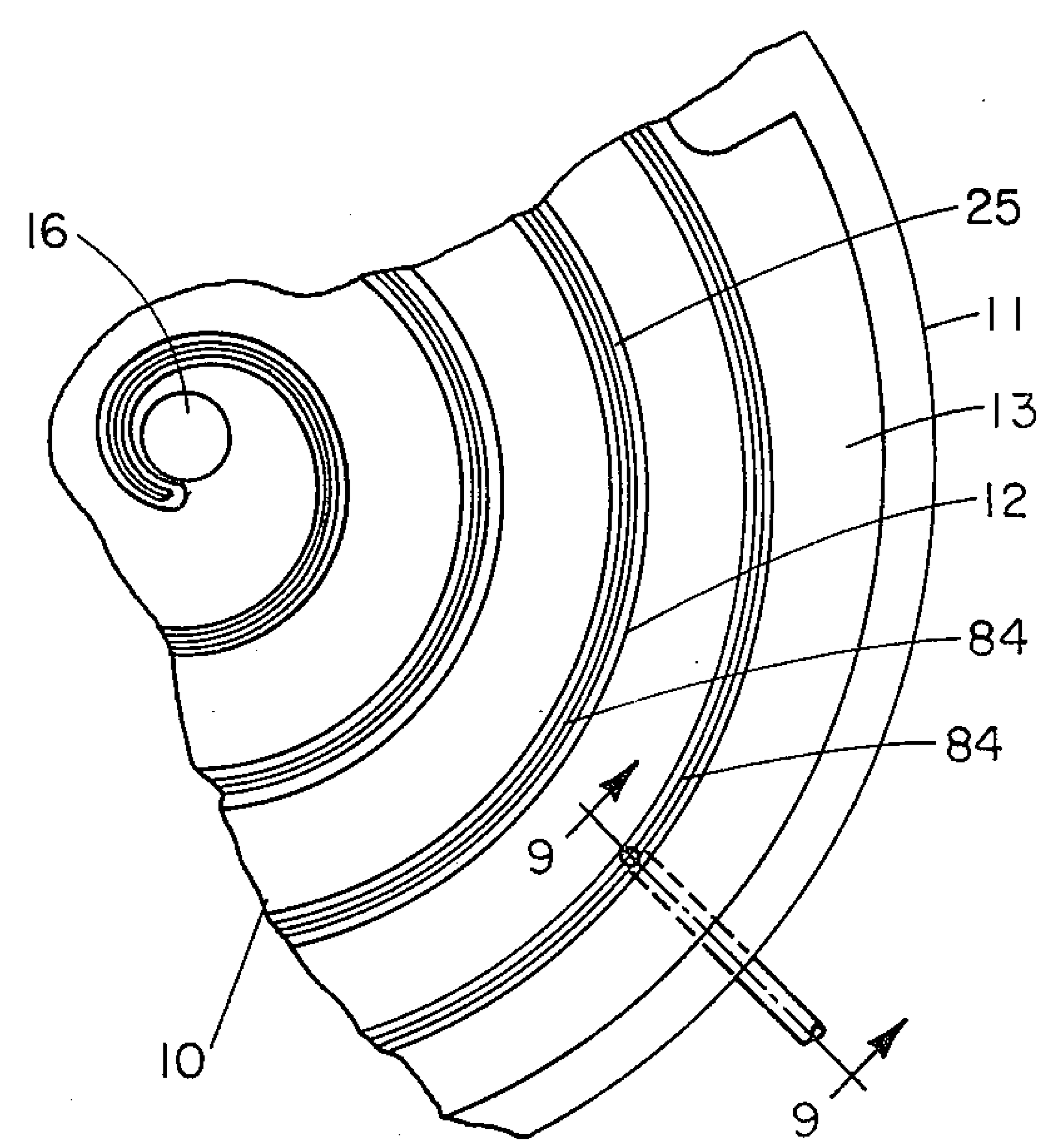
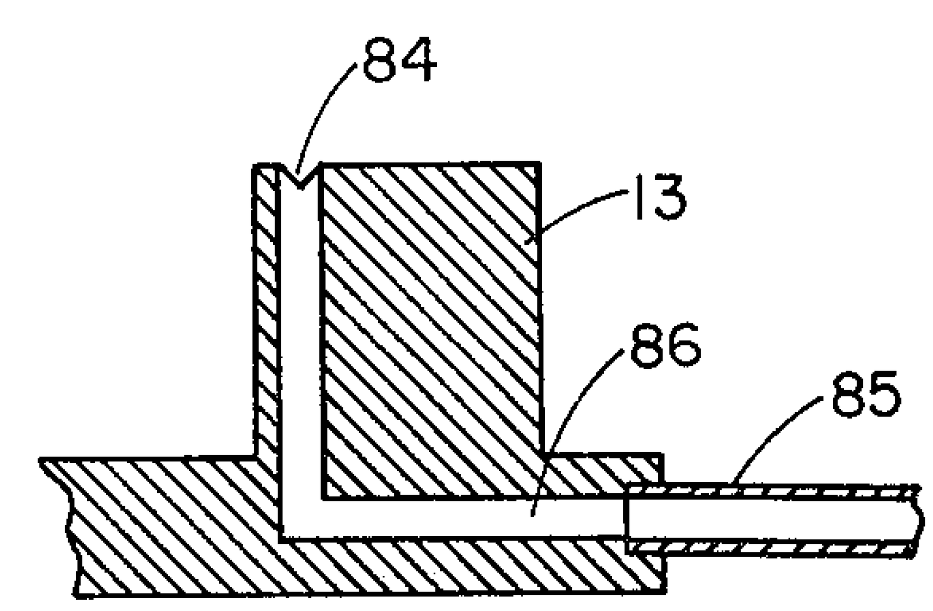


Fig. 8



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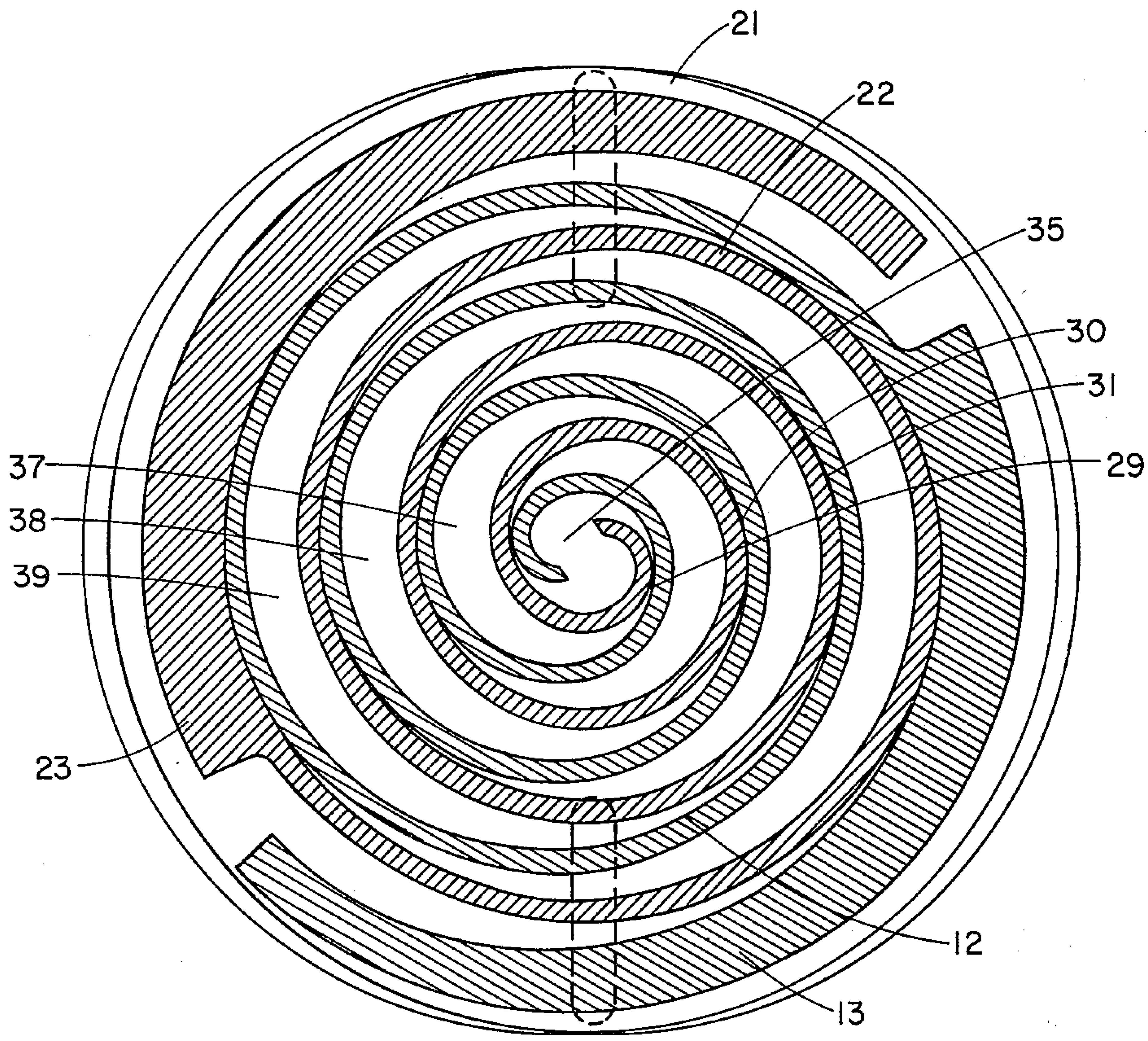

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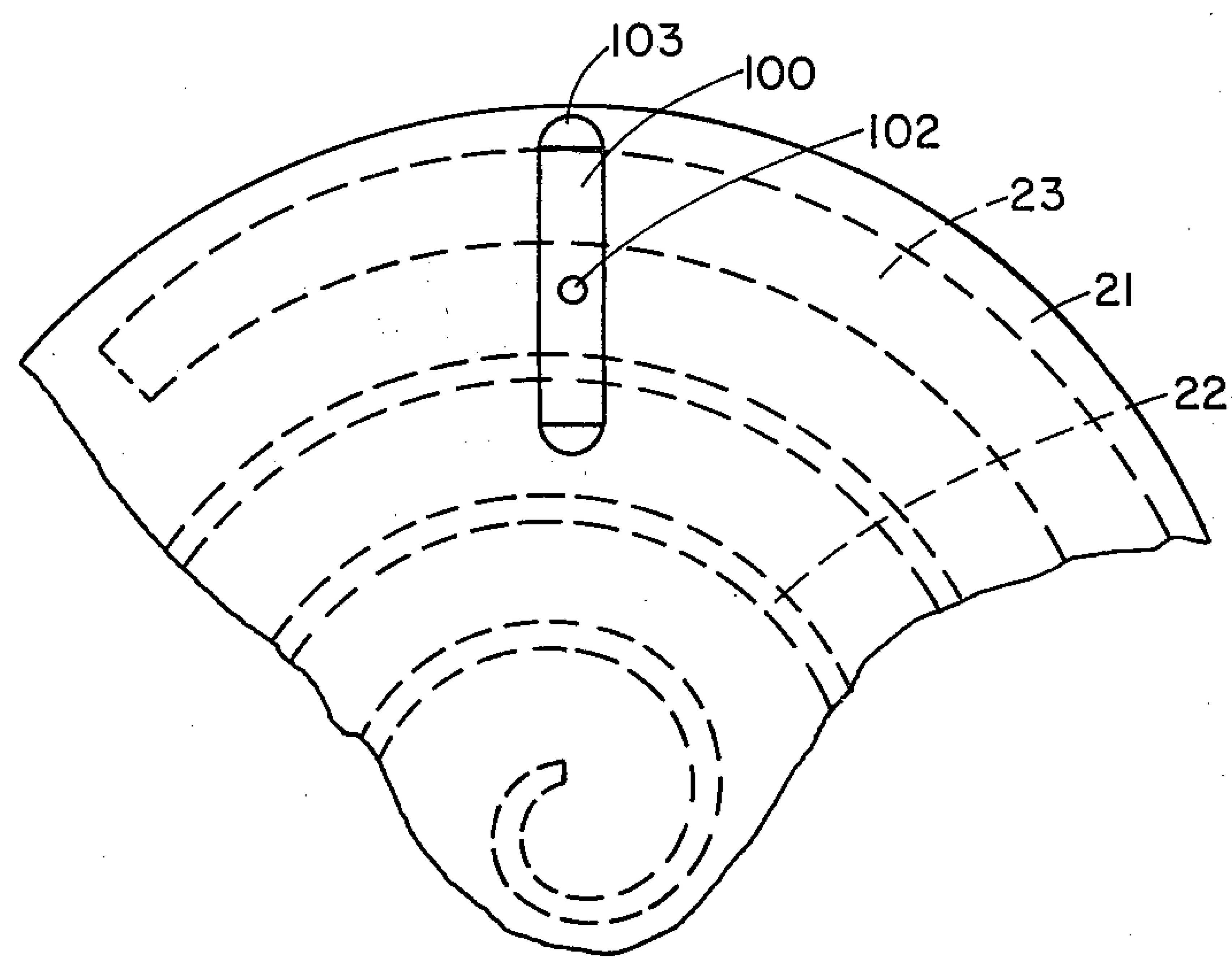

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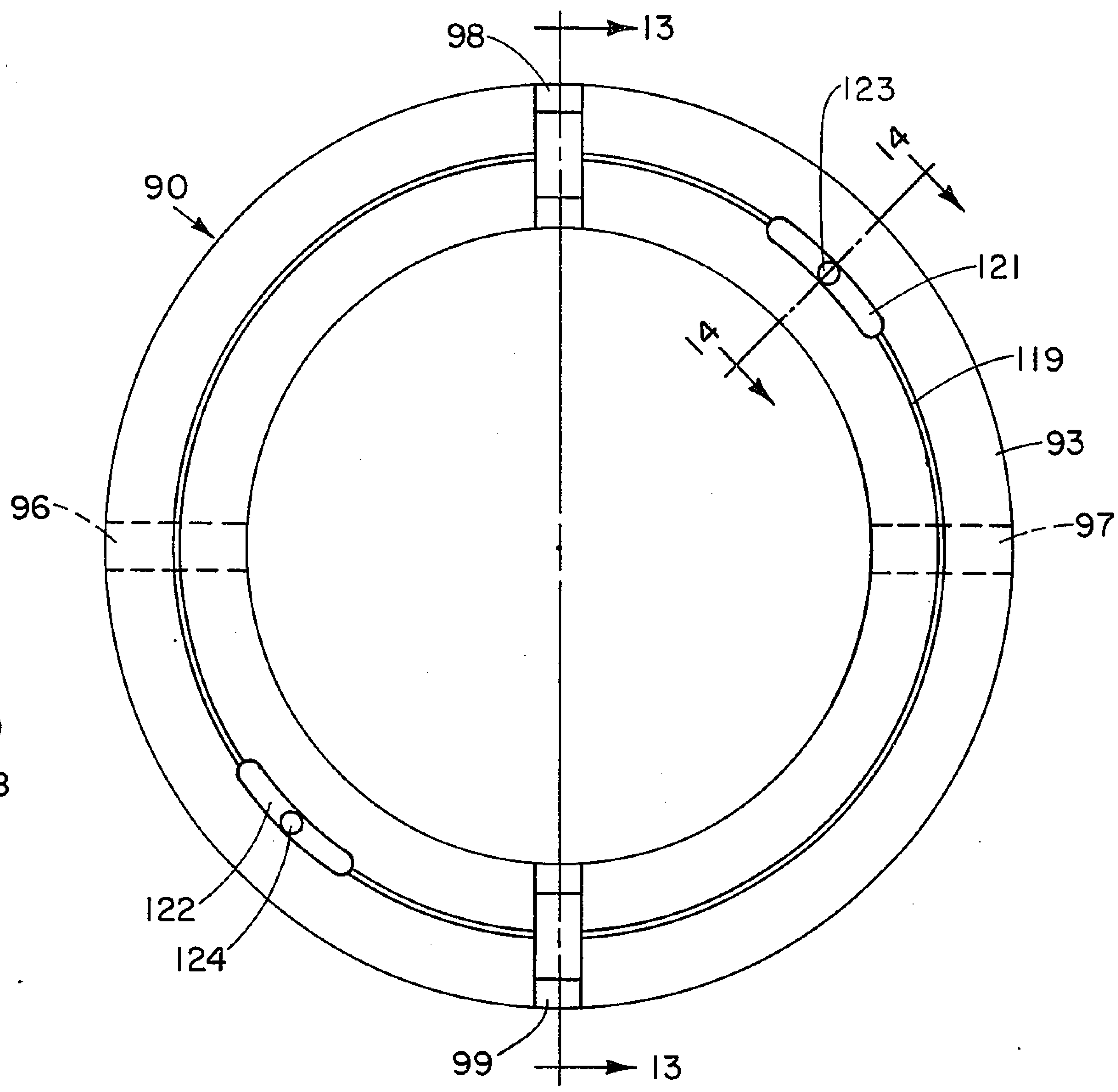

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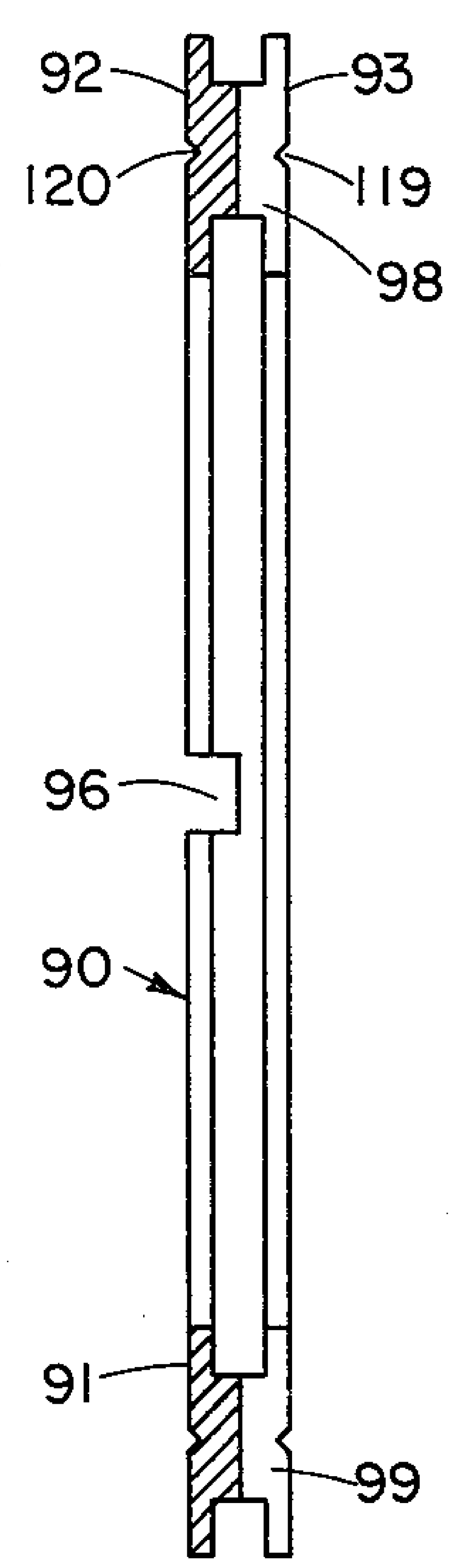

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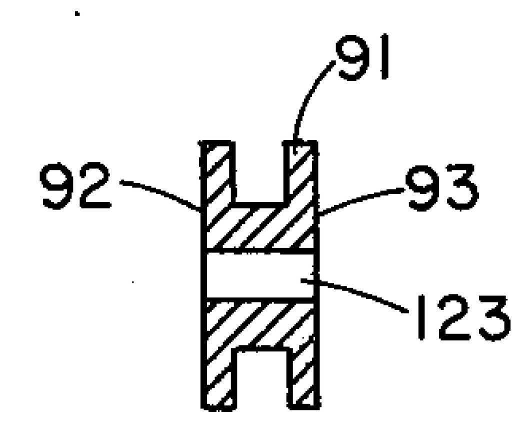
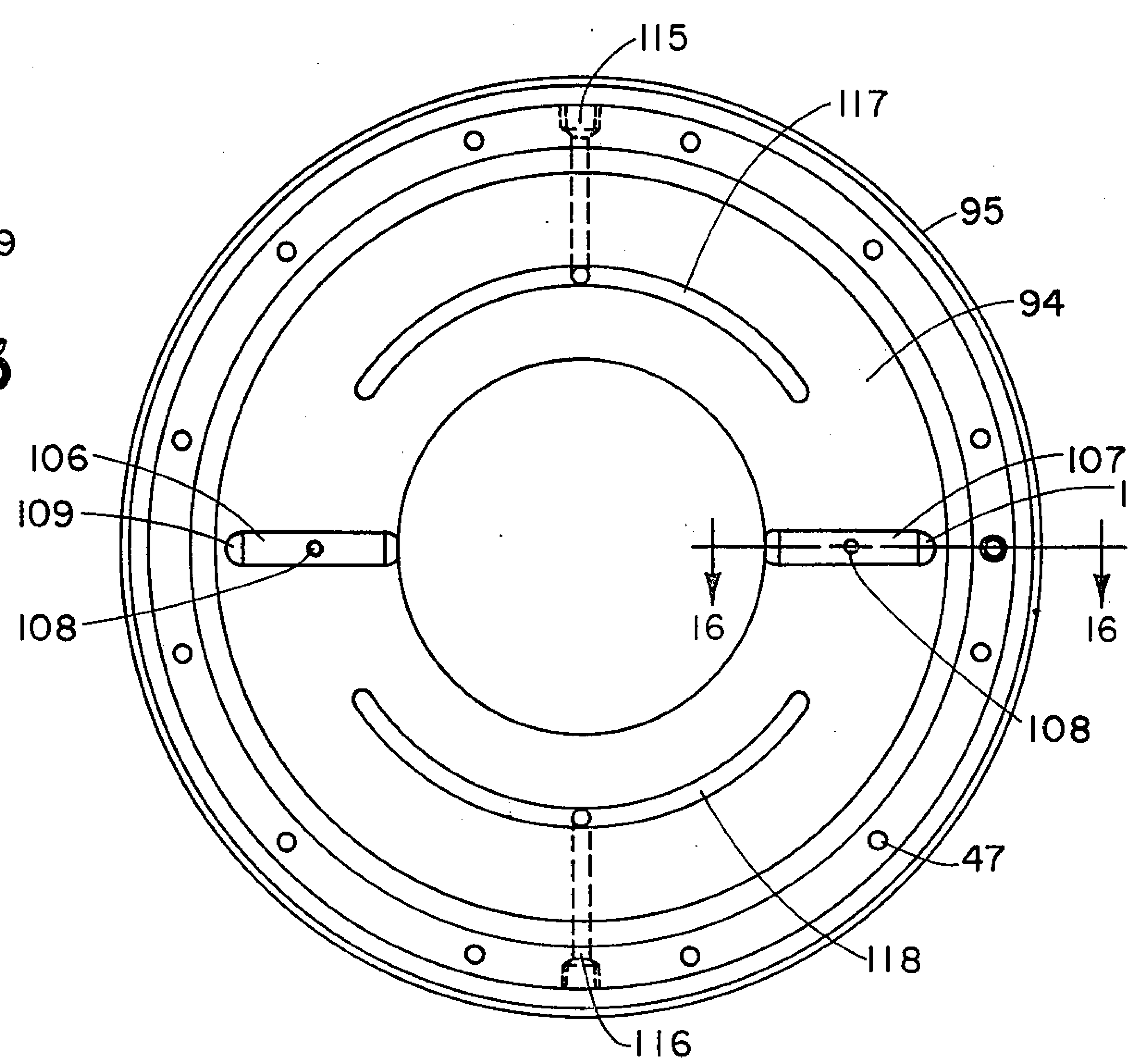


Fig. 14



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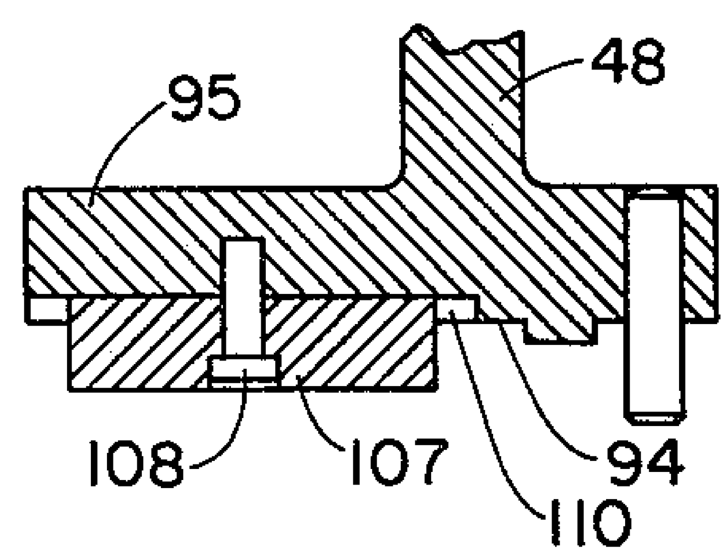

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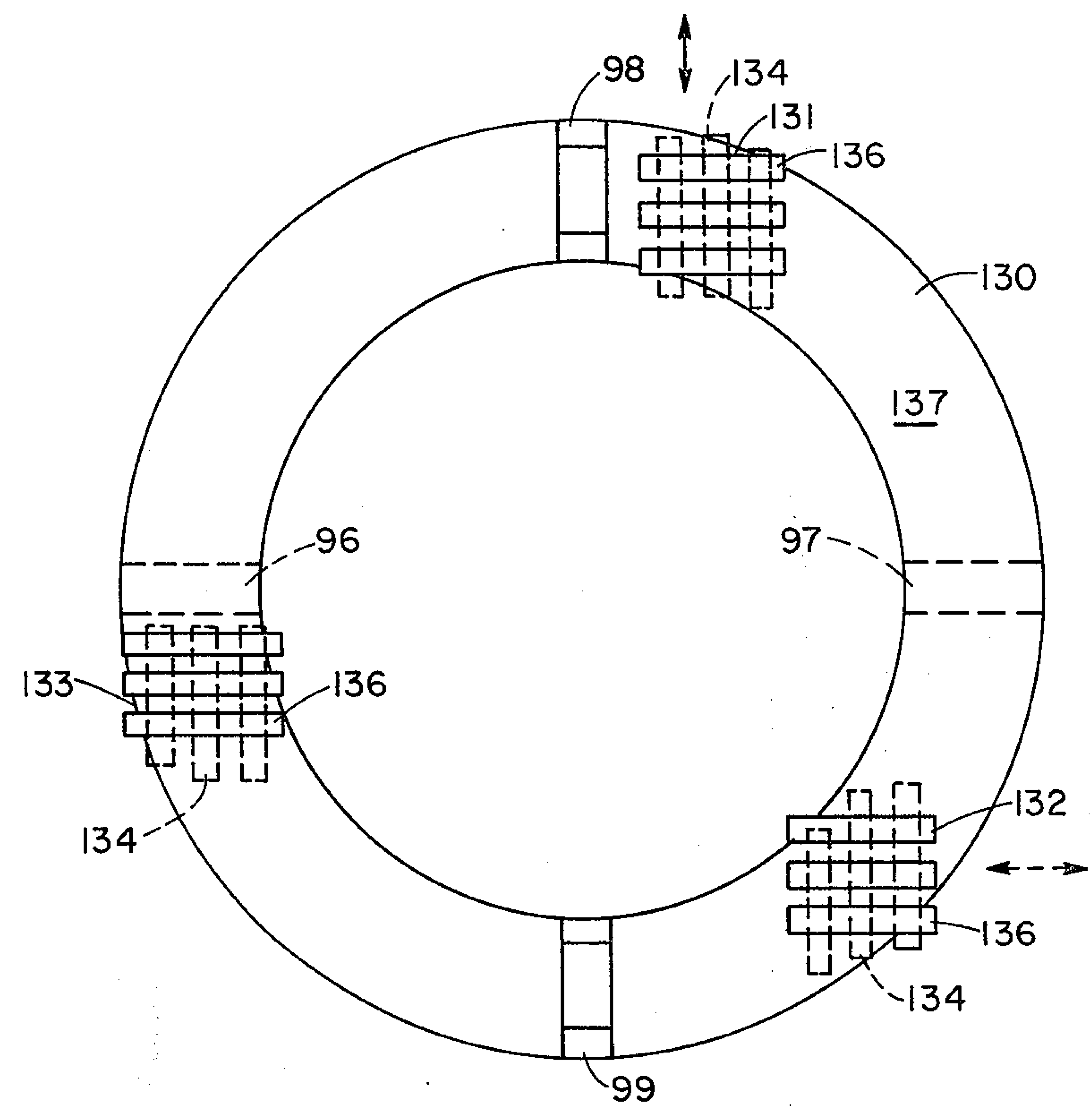

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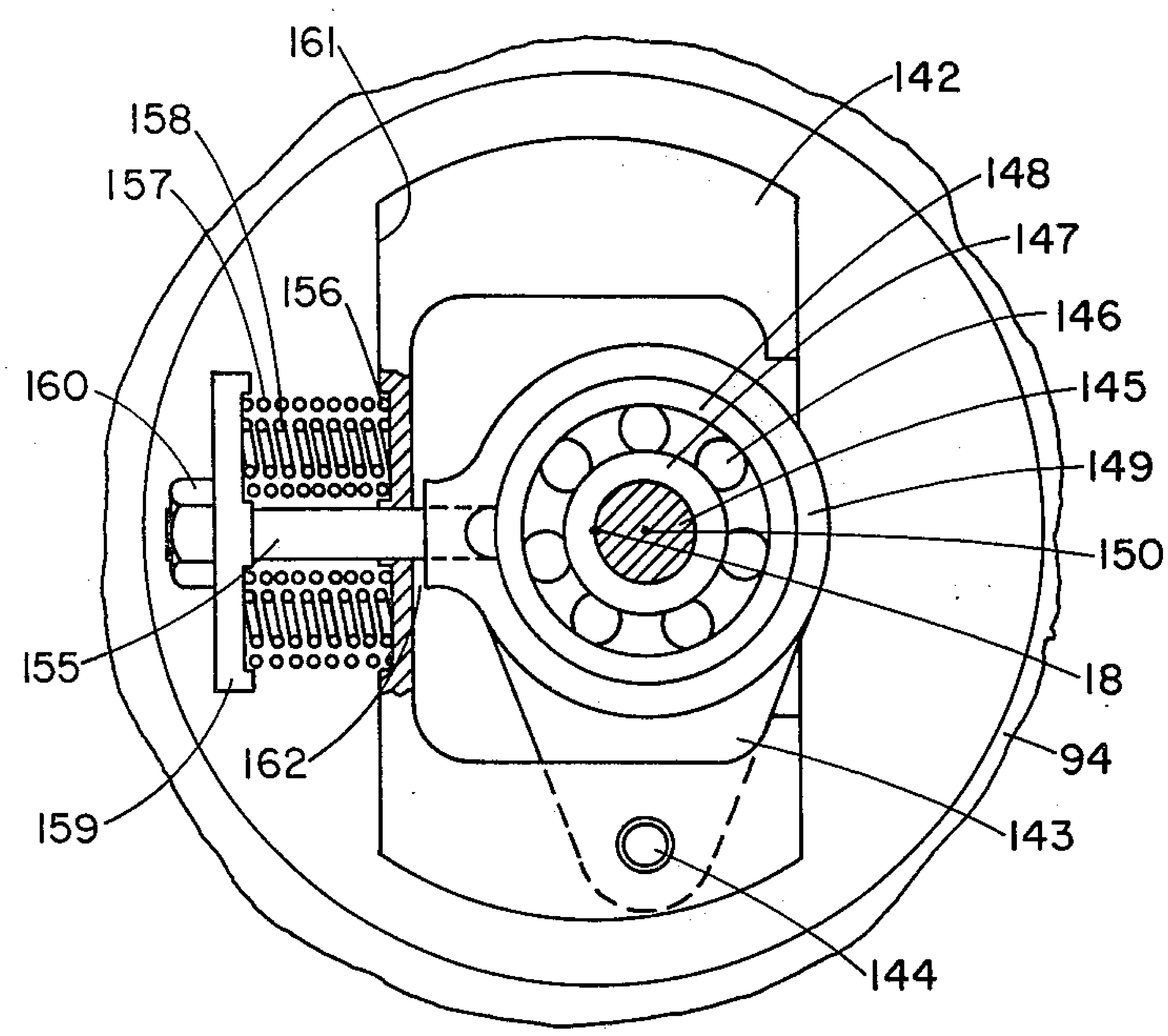
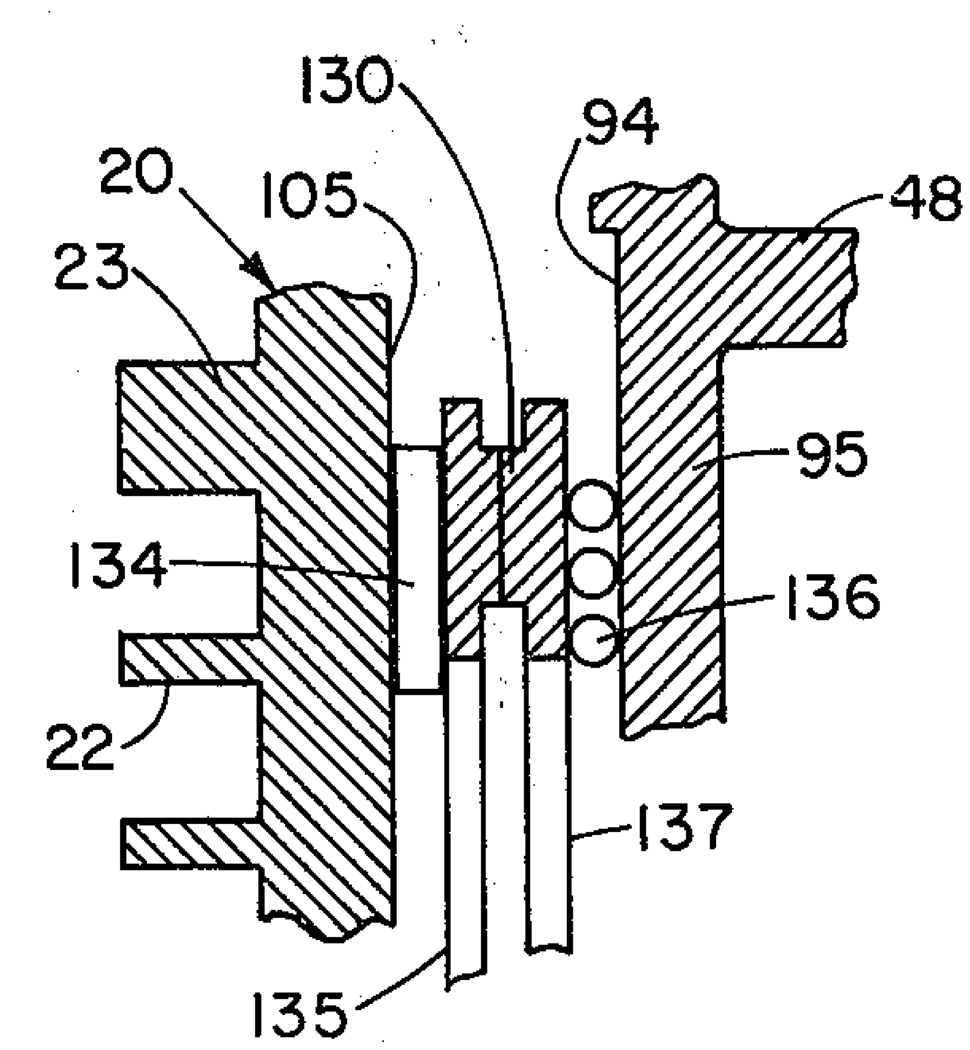


Fig. 19



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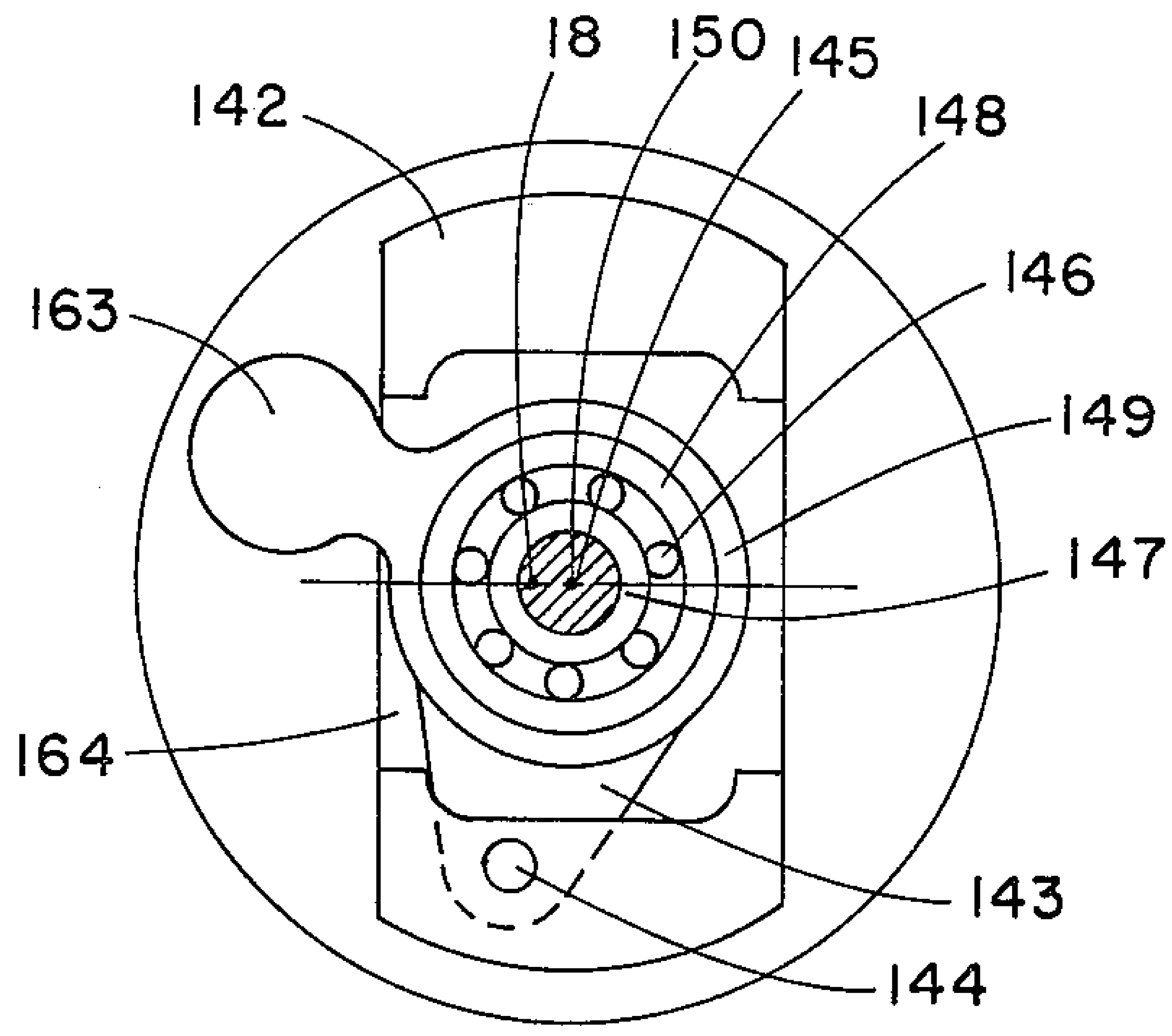

Fig. 20

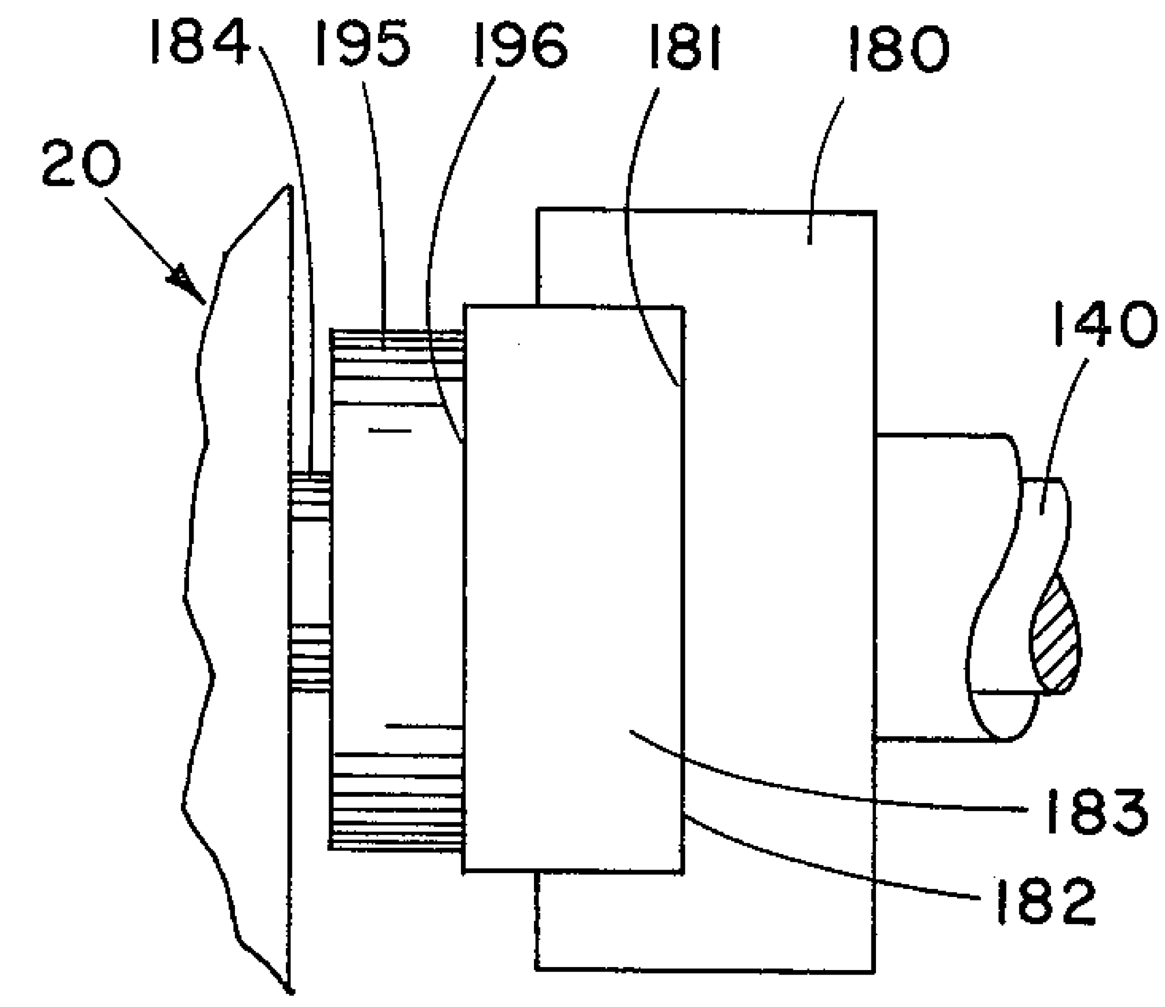

Fig. 23

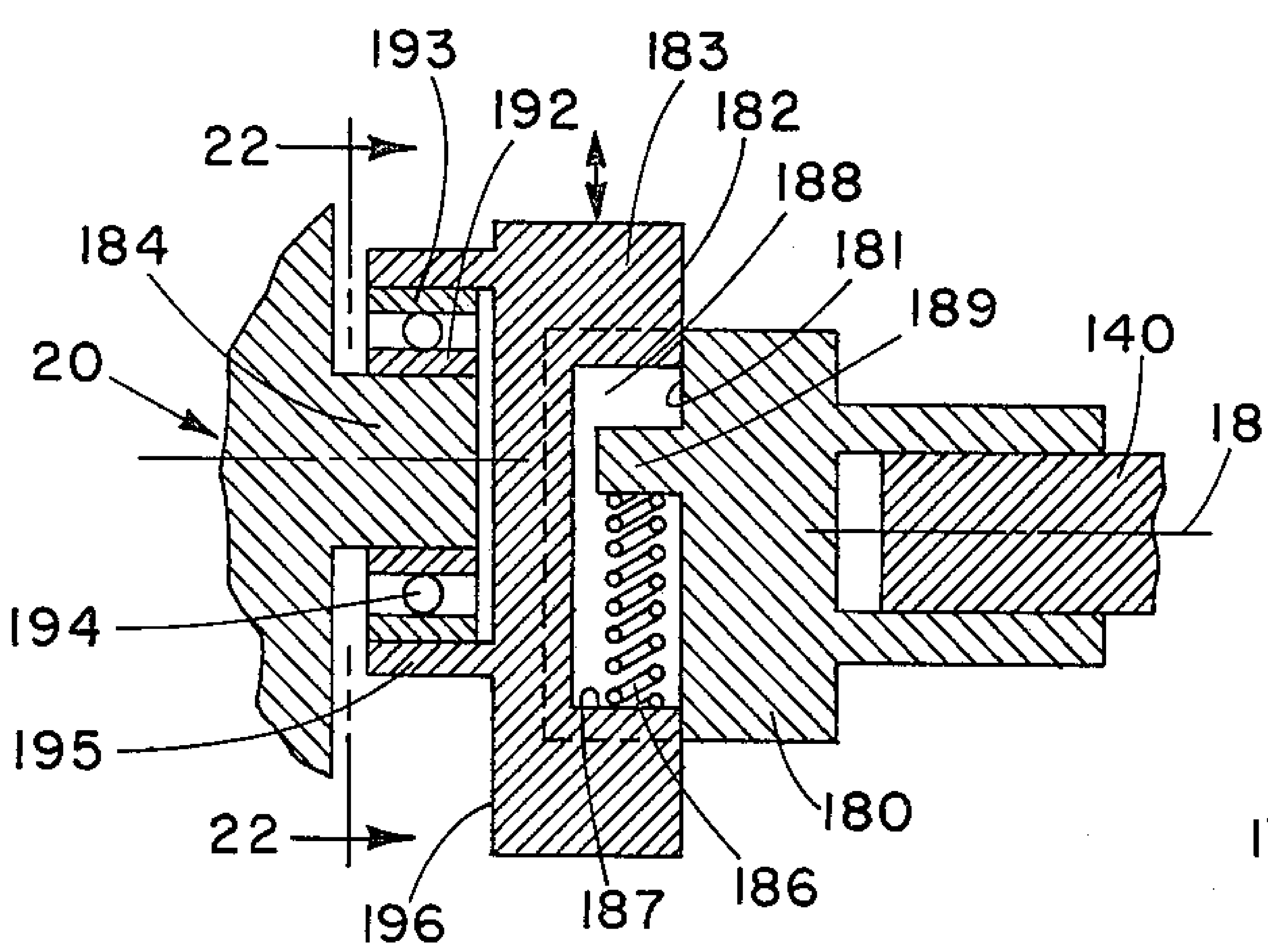

Fig. 21

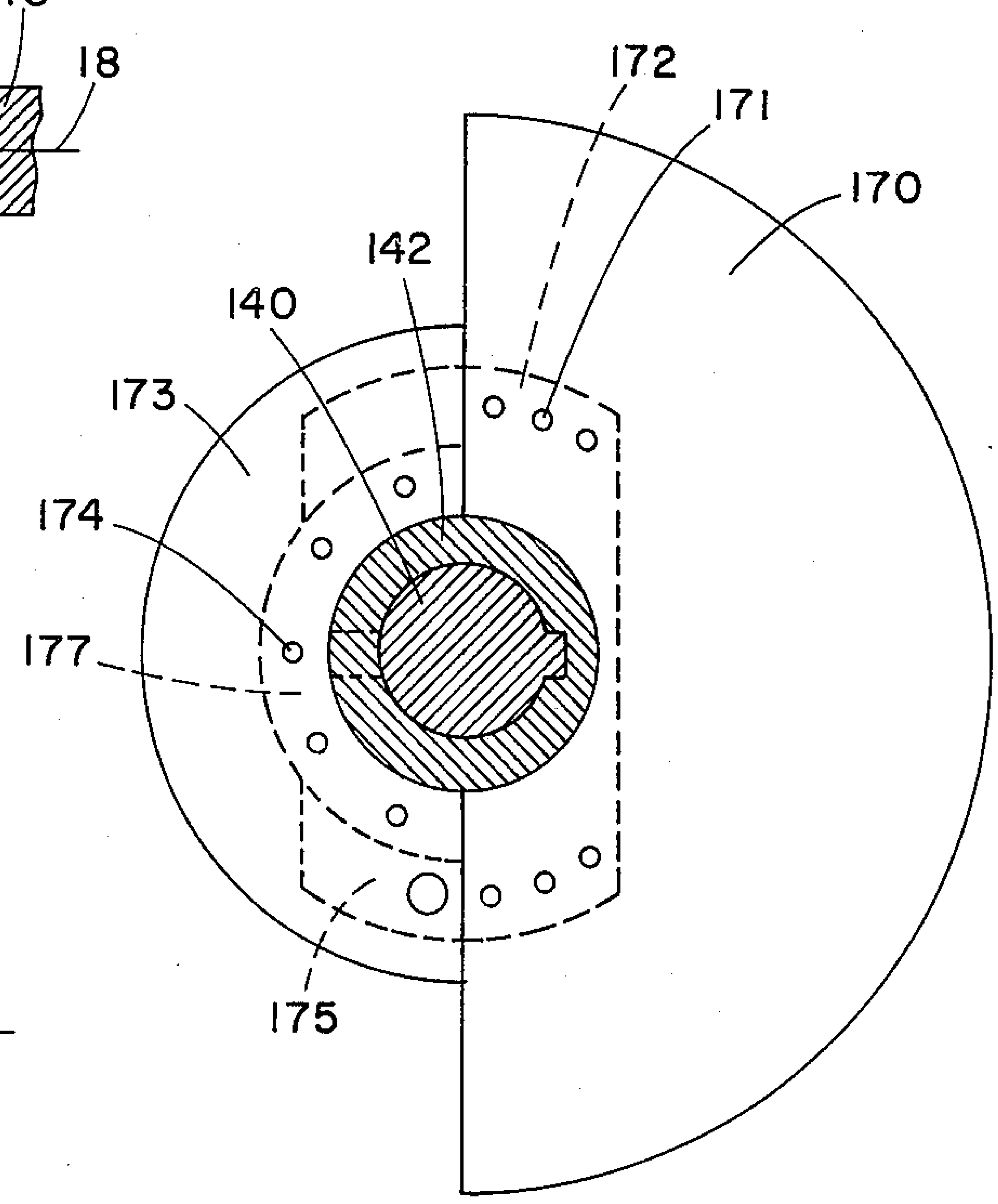

Fig. 24

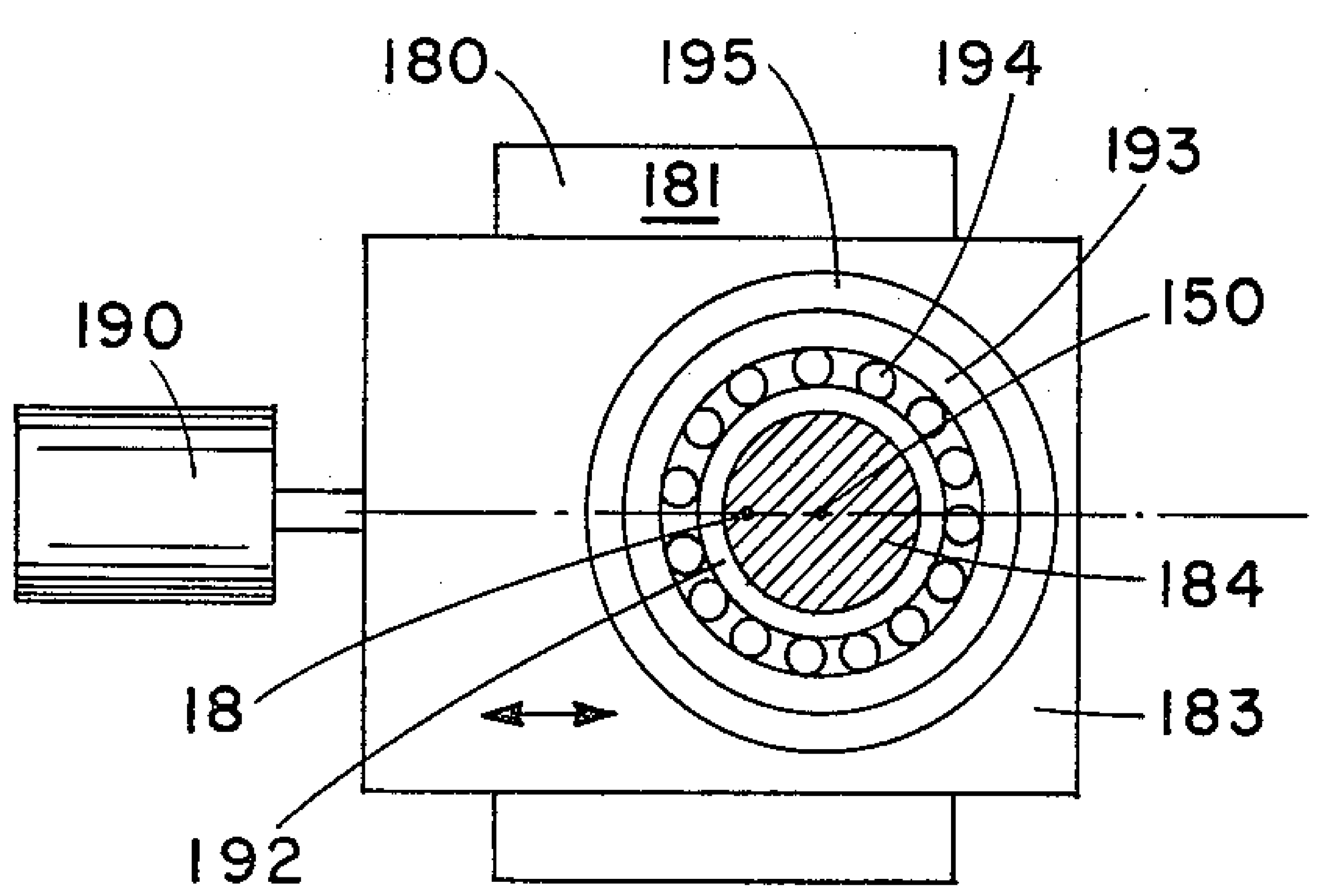

Fig. 22

POSITIVE FLUID DISPLACEMENT APPARATUS

This application is a continuation-in-part of U.S. Pat. application Ser. No. 368,907 filed June 11, 1973, in the names of Niels O. Young and John E. McCullough.

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 percent 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 may not, however, 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 wrap 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.

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 curved surfaces. A 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 the angular position of which varies with relative orbiting of the spiral centers; and all pockets maintain the same relative angular position. 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 Patent 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 constraints, 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 obtained.

The resulting solutions to the sealing, wearing and porting problems through these and other prior art 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 outweighted by sealing, wearing and porting problems. 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 to sealing, wearing and porting problems heretofore encountered.

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 an 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 may be configured to control delivery of fluid into and discharge of fluid from the apparatus.

In copending application Ser. No. 368,907 filed in the names of Niels O. Young and John E. McCullough there is disclosed a novel scroll apparatus in which 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. This requires a flexible linking of the orbiting scroll with its driving or orbiting means while the fixed scroll remains rigidly fixed with respect to the housing as well as to the orbiting scroll. In specific embodiments of the improved scroll apparatus described in U.S. Pat. application Ser. No. 368,907, the required flexible linking of the orbiting scroll with its driving means comprises means defining a cylindrical drive surface associated with the orbiting scroll member having an orbit radius R_{or} and a scroll driving means defining a cylindrical driving surface with an orbit radius R_{od}. The driving surface is designed to drive the orbiting scroll member through line contact with the cylindrical drive surface by virtue of the fact that R_{od} is less than R_{or}. Thus, when the scroll driver is orbited to develop a drive force acting upon the orbiting scroll, a centripetal radial force is provided to oppose a fraction of the centrifugal force acting on the orbiting scroll member, and the difference between this centirpetal radial force and the normal centrifugal force acting upon the orbiting scroll member appears as the only contact force, e.g., radial sealing force, between the scroll members. This embodiment of a flexible linking of orbiting scroll member to the driving mechanism is particularly suitable for smaller-size, relative low-load machines where simplicity of construction and minimization of the number of parts is desirable.

There are, however, a number of applications for compressors, expanders and pumps where large horsepower sizes are required that can carry large drive loads with good wear life, operate over extended periods of time and be easily started up and shut down. For such applications, a positive mechanical linkage between scroll members to achieve radial sealing through balancing of centrifugal forces with centripetal forces has advantages over the use of cylindrical driving surfaces relying on line contacts to provide force control. The apparatus of this invention comprises a unique combination of elements which achieve such a positive mechanical linkage in a scroll type apparatus.

It is therefore a primary object of this invention to provide scroll-type apparatus particularly suited for large housepower sizes and where drive forces on the scroll members are significant. It is another object of this invention to provide apparatus of the character described which has radial compliance permitting the orbiting scroll member to move inwardly and outwardly relative to the machine axis in response to initial component misalignment or gradual wear or to compensate for the temporary presence of noncompressible material such as a slug of liquid, accumulated wear debris or ingested dirt.

Another object of this invention is to provide scroll-type apparatus of the character described which places no load on the drive motor when used as a compressor or pump during start-up and shut-down and which permits the use of conventional journal or ball bearing elements to carry the drive loads with acceptable wear life.

This invention has as yet another object the providing of a scroll-type compressor which is capable of achieving near isothermal compression by virtue of relatively large internal heat transfer capability, which has excellent volumetric efficiency, which is capable of operating at relatively low noise levels, which has a sealed inlet volume, and which is suitable for high- and low-speed operation and for mounting directly to a motor to utilize the drive motor's bearings.

Other objects of the invention will in part be obvious and will in part be apparent hereinafter.

The scroll apparatus of this invention provides means to control the radial contacting forces such that radial sealing is continuously and effectively attained even with wear or when noncompressibles are temporarily present. This means to control radial contacting comprises means to counterbalance at least a fraction of the centrifugal force acting upon the orbiting scroll member and radially compliant mechanical linking means between the orbiting scroll and its drive means. In one embodiment, the radially compliant mechanical linking means is capable of providing a centripetal force to counterbalance a fraction of the centrifugal force thereby having a portion of the centrifugal force available for achieving controlled radial sealing. In this embodiment the compliant mechanical linking means incorporates mechanical springs to counteract a portion of the centrifugal force.

In another embodiment, means separate from the radially compliant mechanical linking means, e.g., counterweights, are provided to counterbalance all or nearly all of the centrifugal forces acting upon the orbiting scroll member and the radially compliant linking means, i.e., mechanical springs, are incorporated to provide the desired radial sealing forces. In this embodiment, the radial sealing force may be made independent of changes in pressure at the inlet and outlet of the machine and of variations in operational speed.

The scroll members are angularly positioned by a coupling of the sliding friction type or rolling element type; the radially compliant linking means may be a slide-link or swing-link; either one or both of the scroll members may be cooled and the contacting surfaces may be lubricated if desired.

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 longitudinal cross section of a compressor constructed in accordance with this invention incorporating a sliding friction type coupling, a swing-link and springs to counterbalance a fraction of the centrifugal force acting on the orbiting scroll member;

FIG. **6** is a fragmentary cross section of the fixed scroll member illustrating in detail one embodiment of an axial sealing force generating means;

FIG. **7** is an end view of the fixed scroll member showing the back or external side of the end plate and the location of coolant passages drilled in the end plate;

FIG. **8** is an end view of a portion of the front or internal side of the end plate of the fixed scroll member illustrating the lubricant groove in the wrap end and the introduction of a lubricant into the groove;

FIG. **9** is a cross section through plane **9—9** of FIG. **7** showing the lubricant inlet line;

FIG. **10** is a cross section through the scroll wraps taken through plane **10—10** of FIG. **5;**

FIG. **11** is a fragmentary end view of the front or external side of the end plate of the orbiting scroll member showing the coupling key;

FIG. **12** is a top plan view of one embodiment of a scroll coupling member;

FIG. **13** is a cross section of the coupling member of FIG. **12** taken through plane **13—13** of that figure;

FIG. **14** is a cross section of the coupling member of FIG. **12** taken through plane **14—14** of that figure;

FIG. **15** is a plan view of the inside surface of the housing frame support member showing the coupling member keys and lubricant channels;

FIG. **16** is a cross section through plane **16—16** of the housing back plate of FIG. **15;**

FIG. **17** is a top plan view of another embodiment of a coupling member designed to make rolling contact with the housing frame and orbiting scroll member;

FIG. **18** is a fragmentary cross section showing the coupling member in place between the housing frame and the orbiting scroll member;

FIG. **19** is a plan view of a swing-link assembly serving as the compliant mechanical linking means between the orbiting scroll member and the driving means;

FIg. **20** illustrates a modification of the swing-link of FIG. **19;**

FIGS. **21–23** are cross section, end and side views of a sliding block assembly serving as the complicant mechanical linking means between the orbiting scroll member and the driving means; and

FIG. **24** is a cross section through the main drive shaft through plane **24—24** of FIG. **5** showing the configuration of the centrifugal force counterbalancing weights used to eliminate vibration.

Before describing specific embodiments of the apparatus of this invention, the principles of the operation of scroll apparatus in general 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 compressed while being 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 volume remain essentially constant independent of pressure 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 cylindrical surfaces defined by the involute of a circle or other suitably curved configuration. The scroll members have parallel axes since in only this way can the continuous sealing contact between the plane surface of the scroll members be maintained. A sealed pocket moves 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 is accomplished, for example, by maintaining one scroll fixed and oribiting the other scroll. 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.

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 follows, 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 arc 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.** In this illustrative example of scroll wraps, congruency results from the fact that one scroll pattern can be brought into coincidence with the other by a simple rotation of one-half turn or less about its axis, followed by a small translation to bring their centers together. The thicknesses, t, of the spiral walls are shown to be identical, although this is not necessary. As will be discussed in the following description 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, delivering mechanical energy in the form of rotary motion as the fluid pockets expand to increasing volume. In such an arrangement the device is an expansion engine and may be used, if desired, to develop refrigeration.

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 encourage 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 mechanism other than the line contacts of the wraps themselves while using such mechanisms also to control angular phase relationships between the scroll members. Failure to provide efficient continued axial and radial sealing permits blow-by and it 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 aggravate 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 compound the problem of wear so that extended operation becomes impractical.

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. The total external axial force on a scroll pair is the sum of a contact sealing force acting between plane surfaces 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.

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 orbiting scroll 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 inherently determine radial sealing of the scroll members comprise tangential forces due to the reaction of the fluid within the scroll volume which is resolved by mechanical radial constraints and centrifugal forces due to the orbiting of the orbiting scroll member. In addition to these inherent forces, other external forces may be supplied.

In the apparatus of this invention the disadvantages associated with scroll apparatus of the prior art are eliminated or minimized by counterbalancing all or a portion of the centrifugal forces acting upon the orbiting scroll and by linking the orbiting scroll to the drive means through a radially compliant mechanical linkage. If all of the centrifugal forces are counterbalanced, then the compliant mechanical linkage is designed to provide a radial force component of a desired magnitude to achieve radial sealing. If less than all of the centrifugal forces are counterbalanced, that fraction which is not counterbalanced is used to provide radial sealing. Axial sealing is accomplished by two opposing forces acting on the scroll members. The first of these forces is a biasing means, including fluid pressure acting on the fixed scroll member and the second force acts upon the orbiting scroll through the coupling means located externally of the scroll pockets and arranged to oppose the force of the biasing means. The coupling means, which maintains the desired angular relationship of the scroll members functions independently of the radial sealing forces which are independently controlled to minimize wear of the line contacts between the wraps of the scroll members and to be able to compensate for noncompressible contaminants which may accidentally enter the fluid pockets.

A compressor constructed in accordance with this invention is shown in longitudinal cross section in FIG. **5.** Reference should also be had to FIGS. **6–16, 19** and **24** where indicated. In all of the drawings like reference numerals are used to refer to like elements. As seen in FIG. **5** the fixed scroll member, generally indicated at **10,** is comprised of an end plate **11;** a wrap **12,** which makes more than three revolutions (see FIG. **10**) and terminates in an enlarged peripheral section **13**; an annular sealing ring member **14**; and a central ported extension **15**. Within extension **15** is a high-pressure fluid passage **16** and extending into it is an connector tube **17**, aligned on the machine axis **18**, for connecting a high-pressure line, not shown.

The orbiting scroll member, generally indicated at **20**, comprises an end plate **21** and a wrap **22**, which makes more than three revolutions and terminates in an enlarged peripheral section **23** (FIG. **10**). In order to achieve axial sealing the end surface **25** of the fixed scroll member wraps **12** and **13** must make sealing contact with the internal surface **26** of the end plate **21** of the orbiting scroll; and in like manner, the end surface **27** of the orbiting scroll member wraps **22** and **23** must make sealing contact with the internal surface **28** of the fixed scroll member. In order to achieve radial sealing the wraps of the two scroll members must make rolling line contacts, e.g., **29, 30** and **31** of FIG. **10**. Innermost fluid pocket **35** defines the zone of highest pressure while plenum chamber **36**, defined around the wraps, comprises the zone of lowest pressure. The fluid pockets, e.g., **37, 38** and **39** are at intermediate pressures increasing toward the center pocket **35**. This zone of highest pressure communicates with a high-pressure line through fluid passage **16** and connector tube **17**. The low-pressure plenum chamber **36** communicates through one or more low-pressure ports **40** with a low-pressure fluid source or reservoir. If the apparatus is a compressor, low-pressure fluid is brought in through low-pressure port **40** and compressed fluid is delivered through a suitable conduit connected to connector tube **17**. If, on the other hand, the apparatus is used as an expander, high-pressure fluid is brought in through passage **16** and expanded low-pressure fluid is discharged through one or more low-pressure ports **40**.

FIG. **10** illustrates but one design of the scroll wraps which may be used in the apparatus. It is also within the scope of this invention to use, for example, wraps of more or less than three revolutions, wraps which are configured as other than true spiroids (e.g., arcsoof circle) and wraps which have their innermost ends configured to control the volume of the highest-pressure pocket. A number of such suitable wrap design variations are included in copending application Ser. No. 368,907.

The scroll apparatus is contained within a housing, generally indicated at **45**, which in the embodiment of FIG. **5** comprises a front housing cover plate **46** affixed through screws **47** to housing back plate **48** which in turn is covered by a back cover **49**. Low-pressure port **40** is cut through the front cover plate **46**.

The fixed scroll member **10** is mounted on and aligned in front housing cover plate **46** through annular sealing ring member **14** and central ported extension **15** which are affixed to, or preferably integral with, the external surface **55** of end plate **11**. Front housing cover plate **46** has an internally disposed mounting ring **56** and a central opening **57**, each of which are grooved to hold elastomeric sealing rings **58** and **59**, these sealing rings providing a seal with ring member **14** and extension **15**. As seen in the detail drawing of FIG. **6**, the annular sealing ring member **14** terminates short of the inner surface **60** of front cover plate **46** to define a shallow annular spacing **61** into which is placed a wave spring washer **62** biased to apply an axial force on the fixed scroll member **10**. The force thus applied constitutes a portion of the axial sealing force required. An additional axial sealing force is provided by introducing a high-pressure fluid into spacing **61** and into annular fluid sealing chamber **63** (defined between ring **14** and extension **15**) through fluid port **64** which is, in turn, connected to a source of high-pressure fluid, not shown. This source of high-pressure fluid may be an external one, e.g., nitrogen from a storage tank, or it may be the zone of highest pressure of the apparatus, e.g., high-pressure pocket **35**. In the latter case, suitable fluid communication means (not shown) must be provided between pocket **35** and fluid sealing chamber **63**. If the high-pressure fluid is to be supplied from the zone of highest pressure within the machine (i.e., from pocket **35**) then the axial force provided by the wave spring washer **62** must be sufficient to provide sealing during start-up since little, if any, high-pressure fluid will be available from pocket **35** during this period of operation. If, however, as in the case of the embodiment of FIGS. **5** and **6**, the high-pressure axial sealing fluid is applied from an external source, the sealing fluid may be introduced into sealing chamber **63** prior to start-up and continued subsequent to shut-down.

The fixed scroll member is angularly positioned relative to the front housing cover plate **46** and is thereby maintained fixed with respect to the housing frame. This is accomplished through the use of locking means generally indicated at **70**. The locking means of FIG. **5** permits minor adjustments to be made during assembly or operating in angularly locating the fixed scroll so that it may be placed at the optimum angle. This locking means comprises a bolt **71** with threads **72** and a head **73** to which a pin **74** is fixed eccentric to the bolt axis. A nut **75** engaging threads **72** serves to hold the bolt in the front plate. Pin **74** engages a notch **76** cut into the back of end plate **11** (See FIG. **7**). Because of the eccentricity of pin **74** relative to the axis of the bolt, turning of bolt **71** causes pin **74**, and hence fixed scroll end plate **11**, to undergo small angular excursions to find the optimum position for the fixed scroll member. Alternatively, the locking means may be a simple constant-diameter pin extending from within front cover **46** into notch **76**, in which case the angular position of the fixed scroll member is not adjustable.

In the embodiment of FIG. **5**, the scroll members are cooled by circulating a coolant through end plate **11** of the fixed scroll member as can best be seen in FIG. **7** which is an end view of the external surface **80** of end plate **11**. A series of interconnecting passages **81** forming a conduit network are drilled in end plate **11** and they are connected through suitable connectors **82** and **83** to fluid conduits (not shown) capable of introducing a coolant into and withdrawing it from the coolant passage network.

In the embodiment illustrated, a lubricant is provided to lubricate the contacting ends **25** of the fixed scroll member wraps. As seen in the fragmentary detail drawings of FIGS. **8** and **9**, the contacting spiraling end surface **25** of the continuous wraps **12** and **13** have an oil channel **84** cut into it. A suitable lubricant is delivered to the outermost part of the channel through a line **85** connected to a passage **86** drilled into enlarged wrap **13** and extending to channel **84**. The lubricant is forced through channel **84** which follows the entire wrap and any excess eventually drains by gravity to the bottom of the housing to be withdrawn through oil drain port **87**. Lubrication of the contacting wrap ends may not be required, particularly in such cases where the wrap and end plate surfaces are of a self-lubricating nature or are not subject to intolerable wear, or where the size and mode of operation either does not require or even prohibits the introduction of lubricants into the system.

The orbiting scroll member must be prevented from moving angularly with respect to the fixed scroll member and with respect to the frame of the housing; and it must be driven in an orbit in a way to counteract all or a portion of the centrifugal force developed in its orbiting while providing the required radial sealing forces.

The maintaining of the desired angular relationship between the oribiting scroll and the fixed scroll member and housing is accomplished through the use of a coupling member generally indicated by the numeral **90.** As will be seen in FIGS. **12–14,** this coupling member comprises an annular ring **91** with an "H"-shaped cross section. What may, for convenience, be designated the front surface **92** faces the orbiting scroll; and in keeping with this terminology, the opposite surface, called the back surface **93,** faces the inside wall **94** of an internal supporting frame member **95** of the housing. Front surface **92** has two oppositely disposed keyways **96** and **97** cut into it (FIG. **12**); and back surface **93** also has two oppositely disposed keyways **98** and **99** cut into it. The axes of the keyways on the front and back surfaces are at right angles. The keys which slidingly engage these keyways are attached to the orbiting scroll and the housing frame member **95.** As will be seen in FIGS. **5** and **11,** keys **100** and **101** are fastened, such as by countersunk screws **102** into shallow recesses **103** and **104** cut into external surface **105** of the orbiting scroll member end plate. These keys slide in keyways **96** and **97** of the coupling member, respectively. FIG. **15,** which is a plan view of internal surface **94** of housing frame member **95** shows the position of keys **106** and **107** mounted through countersunk screws **108** (FIG. **16**) into recesses **109** and **110,** respectively, cut into surface **94.** These keys **106** and **107** slidingly engage keyways **98** and **99.**

Inasmuch as there is sliding friction contact between the front and back surfaces of the coupling and the surfaces of the orbiting scroll member end plate and the support frame, it may be desirable to lubricate these surfaces. In the embodiment of FIG. **5,** this is accomplished by injecting oil through oppositely disposed passages **115** and **116** (FIG. **15**) drilled into frame member **95** to communicate with arcuate channels **117** and **118** cut into surface **94** of the frame member. Oil flowing through these arcuate channels is introduced into oil groove **119,** cut into back face **93** of the coupling, and then transferred to oil groove **120,** cut into front face **92** of the coupling, through short arcuate channels **121** and **122** by way of passages **123** and **124** (FIGS. **12** and **14**) cut through the coupling. Excess oil discharged from the coupling eventually flows out through oil drain port **87.**

In some apparatus it may not be desirable to use a lubricant or to tolerate the losses developed by the friction between the coupling member and the surfaces it contacts as in the embodiment of FIG. **5.** It is possible to eliminate the use of a lubricant and at the same time materially reduce friction losses by the use of a rolling coupling member as illustrated in FIGS. **17** and **18.** The coupling member **130** is an annular ring constructed with essentially the same cross section as the coupling member **90** of FIGS. **12** and **13,** and it has keyways **96–99** as described above. It is, however, supported on three sets of rolls **131, 132** and **133,** each set comprising a plurality of rolls **134** contacting the front face **135** of the coupling and surface **105** of the orbiting scroll member, and a plurality of rolls **136** contacting the back face **137** of the coupler and surface **94** of the support frame member **95.** Rolls **134** and **136** are oriented at right angles to each other and their axes are oriented to be normal to the axis of the keyway cut in the surface which they contact. Thus each roll is positioned to travel with the coupling member as it moves in the directions indicated by the arrows.

Although the embodiments of the coupling means illustrated in FIGS. **12–18** show keyways in the coupling means and slidingly engageable keys in the orbiting scroll member and housing frame, it is also within the scope of this invention to reverse this arrangement and affix the keys to the coupling means and locate the keyways in the orbiting scroll member and the housing frame.

Several embodiments of suitable mechanisms for driving the orbiting scroll member with the desired radial compliance to attain a predetermined sealing force are illustrated in FIGS. **5** and **19–23.** As will be seen in the detailed description of these drawings, the radial compliant means may take one of several forms. Moreover, it is possible to choose between several operating modes, i.e., counterbalancing a fraction of the centrifugal force and using that fraction which is not counterbalanced as a radial sealing force, or counterbalancing essentially all of the centrifugal force and incorporating into the mechanical compliant linkage, means to provide a radially outward force which can alone serve as the radial sealing force.

In the embodiment of FIGS. **5, 19** and **20** the radially compliant mechanical linkage is a swing-link, while in FIGS. **21–23** it is a sliding-block linkage. Both of these embodiments may use springs in compression to counteract all or a part of the centrifugal force, or they may use counterweights in place of or in addition to the springs.

In order to attain radial compliance, the orbiting scroll member must have the ability to move inwardly or outwardly relative to the machine axis in response to gradual wear of the scroll wraps or to encounter noncompressible objects such as a slug of liquid, accumulated wear debris or ingested dirt particles. This radial compliance feature also allows the use of less perfect geometry scrolls in that it allows the orbiting scroll member to ride inside of the fixed scroll member and adjust its trajectory, as required, to suit the geometries of the wraps of the two scrolls. In the embodiments of FIGS. **5, 19** and **20,** a ball bearing is mounted on the axial drive shaft of the orbiting scroll member and the outer periphery of this ball bearing is connected to a crank mechanism with a swing-link. The axis of the swing-link in FIG. **19** is shown to the nominally perpendicular to the eccentricity radius of the orbiting scroll member. During rotation of the drive crank, the orbiting scroll member swings radially outward under the action of centrifugal force acting on its center of mass. The orbiting scroll member is confined to a given locus of motion by virtue of contact with the wrap of the fixed scroll member. The radial contact force between the orbiting and fixed scroll members is adjusted by the use of mechanical springs, or equivalent devices, to counteract some predetermined fraction of the centrifugal force exerted on the orbiting scroll member.

Turning now to FIGS. **5** and **19,** the orbiting scroll is driven by the main drive shaft **140** which is mounted in the back cover plate through a bearing **141.** Affixed to main drive shaft **140** is a crank **142** to which a connecting rod **143** is pivotally mounted through connecting rod pin **144.** This connecting rod is affixed to the orbiting scroll member through a stub shaft **145** by means of snap ring **151** and ball bearing **146** retained by an inner race **147** and outer race **148** mounted on ring **149** of the connecting rod. The axis of shaft **145** is designated in FIG. **19** by the numeral **150.** The axis of the main shaft and of the machine is the same as that of the fixed scroll member and is therefore designated by the numeral **18.** The distance between the orbiting scroll axis **150** and the machine axis **18** is R_{or}, the orbit radius.

Since the description of the apparatus is, for convenience, presented in terms of its serving as a compressor, main shaft **140** is shown attached to a motor **152** of a type suitable to rotate shaft **140.** However, it is also, as previously noted, within the scope of this invention to use this apparatus as an expansion engine, in which case the element **152** may be considered to be any suitable work absorbing means, e.g., another compressor, or a brake in the case of an expansion engine used to develop refrigeration.

The swing-link, which is comprised of the connecting rod, ball bearing assembly, and pin is connected to the crank through one or more springs in compression as shown in FIG. **19.** A T-bolt **155** is attached to connecting rod **143** and extends through the wall of crank **142** which has a shallow well **156** on its external surface to seat concentric springs **157** and **158** held in compression by means of a spring retainer **159** adjustably affixed to T-bolt **155** by means of nut **160.** Springs **157** and **158** are preloaded to a desired force through turning nut **160.** The number of springs and the degree of preloading may be so chosen as to overcome a predetermined fraction of the centrifugal force exerted on the orbiting scroll member while it is achieving full running eccentricity. Thus the springs in effect pull back on the swing-link and thereby exert a centripetal force on the orbiting scroll member, the difference between the centrifugal and centripetal forces being in essence equal to the radial sealing force. This in turn means that the radial sealing force may be adjusted by adjusting the degree of preloading of the springs.

During start-up and shut-down (as well as periods of nonorbiting) the springs cause the swing-link to be pulled inwardly, that is toward the inner surface **161** of the crank so that the small gap **162** shown in FIG. **19** is not present. This means that the eccentricity radius of the orbiting scroll member is slightly less than the normal operating orbit radius. As the speed of the machine increases, the centrifugal force of the orbiting scroll increases, reaching a point where it balances the restraining spring force and eventually achieves a value greater than the restraining spring force. In this fashion, the initial start-up and the final portion of the shut-down operations of the machine occur without wrap-to-wrap contact of the scroll members. This in turn means that the motor **152** which turns main shaft **140** does not have to start under a load but picks up the load as the speed of the machine increases. The same desirable condition also, of course, occurs during shut-down.

It will be apparent that in the embodiment described, the axis of the swing-link is oriented perpendicular to the eccentricity radius of the orbiting scroll member. This has the advantage that the radial sealing force does not vary with changes in the inlet and outlet conditions of the machine.

Modifications of the swing-link embodiment of FIG. **19** are possible and within the scope of this invention. Exemplary of one such modification is the use of a counterweight on the swing-link to exert a force on the link equal and opposite to the centrifugal force exerted by the orbiting scroll. Such a counterweight may be affixed to the spring retainer **159,** or alternatively, the spring retainer itself may be configured to serve as such a counterweight. In the case where a counterweight is used to counterbalance all of the centrifugal force, the springs (e.g., **157** and **158** or a single spring) will be used to generate the radial sealing force. Again, this modification permits accurate control of the radial sealing force. Moreover, the combination of springs and counterweight allows the radial sealing force to be independent of changes in pressure at the inert and outlet of the machine as well as of variations in operating speed.

Two additional modifications of the swing-link are illustrated in FIG. **20.** In the first of these, the springs are replaced by a counterweight **163** which is affixed to or integral with the ring **149** of the connecting rod **143** and extends beyond opening **164** in crank **142.** In this modification, counterweight **163** is designed to counterbalance a fraction of the centrifugal force exerted on the orbiting scroll member, the remaining fraction being used to provide a radial sealing force of a predetermined magnitude. Thus counterweight **163** serves in the same role as springs **157** and **158.**

The second modification illustrated in FIG. **20,** which is equally applicable to the swing-link embodiment of FIG. **19,** is the orientation of the axis of the swing-like to form an angle less than 90° with the eccentricity radius of the orbiting scroll. This is evident from the shift in position of swing-link pivot axis **144** relative to axes **18** and **150.** Reducing this angle has the advantage of reducing the size of the counterweight, or spring, required to counteract the centrifugal force acting upon the orbiting scroll member; but it does mean that the resulting radial sealing force is somewhat dependent upon machine inlet and outlet pressure variations.

It is also possible to form the mechanical, compliant link between the main drive shaft and the orbiting scroll member through a sliding block mechanism in place of the swing-link mechanism illustrated in FIGS. **5, 19** and **20.** An exemplary sliding link mechanism is illustrated in FIGS. **21–23,** wherein the same reference numerals are used to define similar elements identified in FIGS. **5, 19** and **20.** In the cross section, end-on and sides views of FIG. **21–23,** it will be seen that crank **180** provides a flat contacting surface **181** adapted for slidable engagement with surface **182** of sliding block **183** which slides in a groove defined within crank **180,** (FIG. **23**). Sliding block **183** is affixed to flanged stub shaft **184** or orbiting scroll member **20** through a bearing assembly, comprised of an inner race **192,** an outer race **193** and ball bearings **194,** held by retaining member **195** affixed to the back side **196** of sliding block **183.** Crank **180** is preferably linked to sliding block **183** through compression spring **186** (FIG. **21**) which is anchored to the inner wall **187** of a chamber **188** cut into the sliding block and to a spring support **189** which is an integral extension of crank **180** extending into chamber **188.** This spring **186** serves the same role as springs **157** and **158** of the embodiment of FIG. **19,** i.e., the counterbalancing of a fraction of the centrifugal force exerted on the orbiting scroll member. As in the case of the swing-link assembly, it is also possible to use a counterweight in place of or in addition to the springs in the same manner as previously described. Thus counterweight **190** is shown affixed to sliding block **183** in FIG. **22.** The sliding block linkage of FIGS. **21–23** is preferably used with the coupling member of FIGS. **5** and **12,** i.e., where the coupling member makes sliding friction contact with the housing frame and orbiting scroll member. The advantage of the sliding block linkage is that it can carry the axial restraining force to be exerted on the orbiting scroll member so that the coupling is unloaded in the axial direction and therefore consumes less power.

As will be seen in FIGS. **5** and **24,** the driving mechanism also has oppositely disposed counterweights comprising a primary counterweight **170** affixed through screws **171** to the shoulder **172** of crank **142** and a secondary counterweight **173** affixed through screws **174** to a flanged extension **175** of crank **142.** The counterweights are so configured with respect to size and are so positioned on the crank to eliminate vibrations in the running of the machine. It will be noted that the larger, primary counterweight **170** is positioned to exert a centrifugal force in the same direction as the centripetal force of springs **157** and **158** (FIG. **19**).

The operation of the apparatus of this invention has been detailed above with respect to the various elements. It is, therefore, only necessary to briefly review the overall operation of the apparatus, particularly with respect to its serving as either a compressor or an expansion engine. In the case where it is to serve as a compressor, low-pressure fluid, e.g., air at ambient conditions, is taken in through one or more of the low-pressure ports **40,** and delivered as high-pressure air through high-pressure discharge conduit **16.** During its orbiting, orbiting scroll member **20** is forced by the coupling member **90,** which is interposed between it and the housing frame **95,** against the fixed scroll member **10** to oppose the force of the wave spring washer **62** and fluid pressure in fluid sealing chamber **63,** thus attaining axial sealing between the wrap ends and the scroll end plates which they contact. Radial sealing by forcing the wraps to make moving-line contacts is controlled to the desired degree by the amount of centrifugal force which is counteracted by the amount of centripetal force provided by the radially compliant mechanical linkage used.

If the apparatus is to serve as an expansion engine to develop mechanical energy and/or refrigeration, the gas to be expanded is introduced into the high-pressure port **16** and withdrawn into a low-pressure reservoir through one or more low-pressure ports. The attainment of axial and radial sealing is the same as when the apparatus is used as a compressor.

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.

I 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, in which two scroll members being maintained at a desired angular relationship and having wraps which make a plurality of 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 by driving means to orbit within said other of said scroll members while maintaining a fixed angular relationship therewith, and wherein said driving means associated with said one of said scroll members includes means to provide a centripetal radial force adapted to oppose a fraction of the centrifugal force acting upon said one of said scroll members, the improvement comprising radially compliant mechanical linking means between said driving means and said one of said scroll members, said radially compliant mechanical linking means including said means to provide said centripetal radial force of a magnitude to control the radial sealing force between said scroll members at a level to minimize both wear and internal fluid leakage.

2. A positive fluid displacement apparatus in accordance with claim **1** wherein said radially compliant mechanical linking means includes means to provide a centripetal radial force less than said centrifugal force and the difference of said forces comprises said radial sealing force.

3. A positive fluid displacement apparatus in accordance with claim **2** wherein said radially compliant mechanical linking means comprises swing-like means.

4. A positive fluid displacement apparatus in accordance with claim **3** wherein said swing-link means includes compression spring means to provide said centripetal radial force.

5. A positive fluid displacement apparatus in accordance with claim **2** wherein said radially compliant mechanical linking means comprises sliding-block linking means.

6. A positive fluid displacement apparatus in accordance with claim **5** wherein said sliding-block linking means includes compression spring means to provide said centripetal radial force.

7. A positive fluid displacement apparatus in accordance with claim **1** wherein said radially compliant mechanical linking means includes means to provide a centripetal radial force essentially equal to said centrifugal force and means to provide a separate adjustable radial sealing force.

8. A positive fluid displacement apparatus in accordance with claim **7** wherein said radially compliant mechanical linking means comprises swing-link means having compression spring means to provide said centripetal force and counterweight means as said means to provide a separate controllable radial sealing force.

9. A positive fluid displacement apparatus in accordance with claim **7** wherein said radially compliant mechanical linking means comprises sliding-block linkage means having compression spring means to provide said centripetal force and counterweight means as said means to provide a separate controllable radial sealing force.

10. A positive fluid displacement apparatus into which fluid is introduced through an inlet port for circulation therethrough and subsequently withdrawn through a discharge port, comprising in combination

a. an orbiting scroll member and a fixed scroll member each having end plate means to which are affixed wrap means which, when said orbiting scroll member is orbited with respect to said fixed scroll member, make moving line contacts to seal off and define at least one moving fluid pocket of variable volume and zones of different fluid pressure and develop radial constraints;

b. drive means to effect orbital motion of said orbiting scroll member;

c. radially compliant mechanical linking means between said orbiting scroll member and said drive means;

d. centrifugal force counterbalancing means associated with said mechanical linking means and adapted to counterbalance at least a fraction of the centrifugal force acting upon said orbiting scroll member; and

e. coupling means adapted to prevent relative angular motion of said scroll members, said coupling means being separate and distinct from said scroll drive means whereby said radial constraints within said apparatus are limited to said moving line contacts between said wraps and are controlled through said radially compliant mechanical linking means.

11. A positive fluid displacement apparatus in accordance with claim **10** wherein said centrifugal force counterbalancing means counterbalances less than all of said centrifugal force and that fraction of said centrifugal force which is not counterbalanced constitutes the radial sealing force between said wraps.

12. A positive fluid displacement apparatus in accordance with claim **11** wherein said radially compliant mechanical linking means is a swing-link and said centrifugal force counterbalancing means are compression spring means.

13. A positive fluid displacement apparatus in accordance with claim **11** wherein said radial compliant mechanical linking means is a swing-link and said centrifugal force counterbalancing means is a counterweight.

14. A positive fluid displacement apparatus in accordance with claim **11** wherein said radial compliant mechanical linking means is a sliding-block link and said centrifugal force counterbalancing means are compression spring means.

15. A positive fluid displacement apparatus in accordance with claim **11** wherein said radial compliant mechanical linking means is a sliding-block link and said centrifugal force counterbalancing means is a counterweight.

16. A positive fluid displacement apparatus in accordance with claim **10** wherein said centrifugal force counterbalancing means counterbalances essentially all of said centrifugal force and includes radial sealing force means to provide a separate controllable radial sealing force.

17. A positive fluid displacement apparatus in accordance with claim **16** wherein said radially compliant mechanical linking means is a swing-link and said centrifugal force counterbalancing means and said radial sealing force means are compression spring means.

18. A positive fluid displacement apparatus in accordance with claim **16** wherein said radial compliant mechanical linking means is a swing-link, said centrifugal force counterbalancing means is a counterweight and said radial sealing force means are compression spring means.

19. A positive fluid displacement apparatus in accordance with claim **16** wherein said radial compliant mechanical linking means is a sliding-block link and said centrifugal force counterbalancing means and said radial sealing force means are compression spring means.

20. A positive fluid displacement apparatus in accordance with claim **16** wherein said radial compliant mechanical linking means is a sliding-block link, said centrifugal force counterbalancing means is a counterweight and said radial sealing force means are compression spring means.

21. A positive fluid displacement apparatus into which fluid is introduced through an inlet port for circulation therethrough and subsequently withdrawn through a discharge port, comprising in combination

a. an orbiting scroll member and a fixed scroll member each having end plates affixed to wrap means which, when said orbiting scroll member is orbited with respect to said fixed scroll member, make moving line contacts through a radial sealing force to seal off and define at least one moving fluid pocket of variable volume and zones of different fluid pressure and develop radial constraints;

b. axial sealing means adapted to force said wrap means into sealing contact with said end plates;

c. main drive shaft means, the axis of which is the axis of said fixed scroll member and is parallel to and spaced from the axis of said orbiting scroll member, the distance between said axes being the operational orbit radius of said orbiting scroll member;

d. radially compliant mechanical linking means joining said orbiting scroll member and said main drive shaft means;

e. centrifugal force counterbalancing means associated with said mechanical linking means adapted to counterbalance at least a fraction of the centrifugal force acting upon said orbiting scroll member;

f. coupling means adapted to prevent relative angular motion of said scroll members, said coupling means being separate and distinct from said orbiting scroll member driving means whereby said radial constraints comprising said radial sealing force within said apparatus are limited to said moving line contacts between said wraps and are controlled solely through said radially compliant mechanical linking means and its associated centrifugal force counterbalancing means; and

(g) housing means, including frame support means, defining a fluid chamber within which said scroll members are located.

22. A positive fluid displacement apparatus in accordance with claim **21** wherein said radial compliant mechanical linking means is a swing-link and the axis of said swing-link is oriented perpendicular to said eccentricity radius of said orbiting scroll member.

23. A positive fluid displacement apparatus in accordance with claim **21** wherein said radial compliant mechanical linking means is a swing-link and the axis of said swing-link is oriented to make an angle of less than **90°** with the eccentricity radius of said orbiting scroll member.

24. A positive fluid displacement apparatus in accordance with claim **21** including crank means joining said main drive shaft means to said radially compliant mechanical linking means.

25. A positive fluid displacement apparatus in accordance with claim **21** including counterweight means attached to said crank means to minimize or eliminate vibration in said apparatus.

26. A positive fluid displacement apparatus in accordance with claim **21** including means to circulate a coolant within the end plate of said fixed scroll member.

27. A positive fluid displacement apparatus in accordance with claim **21** wherein the contacting surface of the wrap means of said fixed scroll member has shallow lubricant channel means therein, and said apparatus includes means to introduce a lubricant into said channel means.

28. A positive fluid displacement apparatus in accordance with claim **21** including locking means to lock said fixed scroll member to said housing means, said locking means being adjustable whereby the angular orientation of said scroll member may be adjusted within said housing.

29. A positive fluid displacement apparatus in accordance with claim **21** having fluid inlet port means associated with the zone of lowest pressure and fluid discharge port means associated with the zone of highest pressure and means to rotate said main drive shaft means, whereby said apparatus is a compressor.

30. A positive fluid displacement apparatus in accordance with claim **21** having fluid inlet port means associated with the zone of highest pressure and fluid discharge port means associated with the zone of lowest pressure and work absorbing means connected to said main drive shaft means, whereby said apparatus in an expansion engine.

31. A positive fluid displacement apparatus in accordance with claim **21** wherein said coupling member is an annular ring, positioned between the endplate of said orbiting scroll member and said frame support means, having oppositely disposed keyways on each surface thereof and wherein said endplate of said orbiting scroll member and said frame support means have keys slidably engageable with said keyways of said coupling means.

32. A positive fluid displacement apparatus in accordance with claim **31** wherein said coupling means is further characterized by having shallow lubricant channels on each of its surfaces and passage means providing fluid communication between said channels.

33. A positive fluid displacement apparatus in accordance with claim **31** including sets of roller means, each comprising a plurality of rollers, interposed between one surface of said coupling means and said end plate of said orbiting scroll member and between the other surface of said coupling means and said frame support means, said roller means being so oriented that the axes of said rollers contacting any one surface of said coupling means are normal to the axis of said keyways on that surface.

34. A positive fluid displacement apparatus in accordance with claim **21** wherein said centrifugal force counterbalancing means counterbalances less than all of said centrifugal force and that fraction of said centrifugal force which is not counterbalanced constitutes said radial sealing force.

35. A positive fluid displacement apparatus in accordance with claim **34** wherein said radially compliant mechanical linking means is a swing-link and said centrifugal force counterbalancing means are compression spring means.

36. A positive fluid displacement apparatus in accordance with claim **34** wherein said radial compliant mechanical linking means is a swing-link and said centrifugal force counterbalancing means is a counterweight.

37. A positive fluid displacement apparatus in accordance with claim **34** wherein said radial compliant mechanical linking means is a sliding-block link and said centrifugal force counterbalancing means are compression spring means.

38. A positive fluid displacement apparatus in accordance with claim **34** wherein said radial compliant mechanical linking means is a sliding block link and said centrifugal force counterbalancing means is a counterweight.

39. A positive fluid displacement apparatus in accordance with claim **21** wherein said centrifugal force counterbalancing means counterbalances essentially all of said centrifugal force and includes radial sealing force means to provide a separate controllable radial sealing force.

40. A positive fluid displacement apparatus in accordance with claim **39** wherein said radially compliant mechanical linking means is a swing link and said centrifugal force counterbalancing means and said radial sealing force means are compression spring means.

41. A positive fluid displacement apparatus in accordance with claim **39** wherein said radial compliant mechanical linking means is a swing-link, said centrifugal force counterbalancing means is a counterweight and said radial sealing force means are compression spring means.

42. A positive fluid displacement apparatus in accordance with claim **39** wherein said radial compliant mechanical linking means is a sliding-block link and said centrifugal force counterbalancing means and said radial sealing force means are compression spring means.

43. A positive fluid displacement apparatus in accordance with claim **39** wherein said radial compliant mechanical linking means is a sliding-block link, said centrifugal force counterbalancing means is a counterweight and said radial sealing force means are compression spring means.

44. A positive fluid displacement apparatus in accordance with claim **21** wherein said axial sealing means comprise in combination (1) mechanical and fluid force applying means providing a first axial force acting on said fixed scroll member and (2) said coupling means positioned between said orbiting scroll member and said frame support means adapted to oppose said first axial force with a second axial force.

45. A positive fluid displacement apparatus in accordance with claim **44** wherein said mechanical force applying means comprises a wave spring washer.

46. A positive fluid displacement apparatus in accordance with claim **44** wherein said fluid force applying means comprises a fluid chamber positioned to exert fluid pressure upon the end plate of said fluid scroll member and means to deliver high-pressure fluid to said fluid chamber.

47. A positive fluid displacement apparatus in accordance with claim **46** wherein said means to deliver high-pressure fluid to said fluid chamber comprises conduit means providing fluid communication between an external source of a high-pressure fluid and said fluid chamber.

48. A positive fluid displacement apparatus in accordance with claim **46** wherein said means to deliver high-pressure fluid to said fluid chamber comprises conduit means providing fluid communication between the zone of highest pressure and said fluid chamber.

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