United States Patent [19]

Young

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[54] **POSITIVE DISPLACEMENT SCROLL APPARATUS WITH AXIALLY RADIALLY COMPLIANT SCROLL MEMBER**

[76] Inventor: **Niels O. Young,** Hurricane Hill Rd., Mason, N.H. 03048

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[52] **U.S. Cl.**...................... **418/55,** 418/57, 418/151
[51] **Int. Cl.**..... **F01c 1/02,** F01c 21/00, F04c 17/02
[58] **Field of Search**......................... 418/55, 57, 151

[56] **References Cited**

UNITED STATES PATENTS

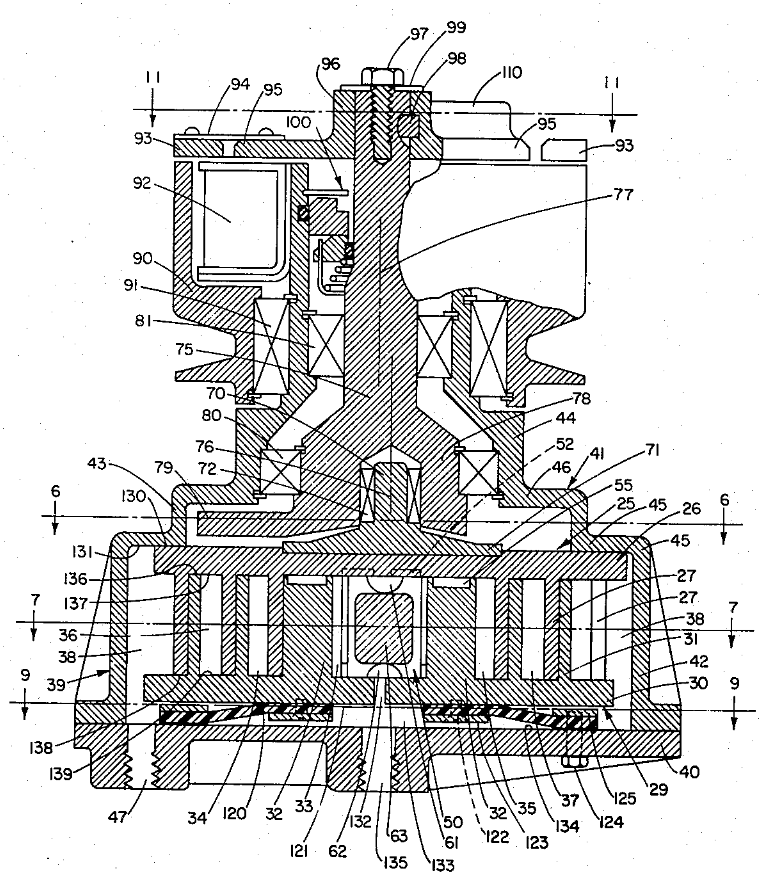
3,817,664 6/1974 Bennett et al. 418/55

Primary Examiner—John J. Vrablik
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 prots. One of the scroll members is caused to orbit along a path which is rigidly defined with respect to the machine frame by bearings and mechanical elements. The other of the scroll members does not orbit except in yielding response to forces applied to it by the fluid being processed and by forces applied by the orbiting scroll member. This other scroll member therefore makes small radial and axial accommodating motions which effect axial and radial sealing through externally applied axial forces and internally generated radial forces. The small radial motions of the accommodating scroll member result in its performing a small orbit in response to the totality of forces acting upon it. Hence this scroll member may be said to perform a minor orbit of small dimensions and is referred to as the "accommodating scroll member." The centrifugal force component produced by the orbiting of the one scroll and the motion of the other scroll is essentially all counterbalanced to leave the internally applied radial force as the primary radial sealing force. Fluid withdrawn from the zone of highest pressure is used as an axial sealing force. Coupling means are provided to maintain the desired angular relationship between the scroll members. The apparatus may serve as a compressor, expander or pump.

15 Claims, 20 Drawing Figures



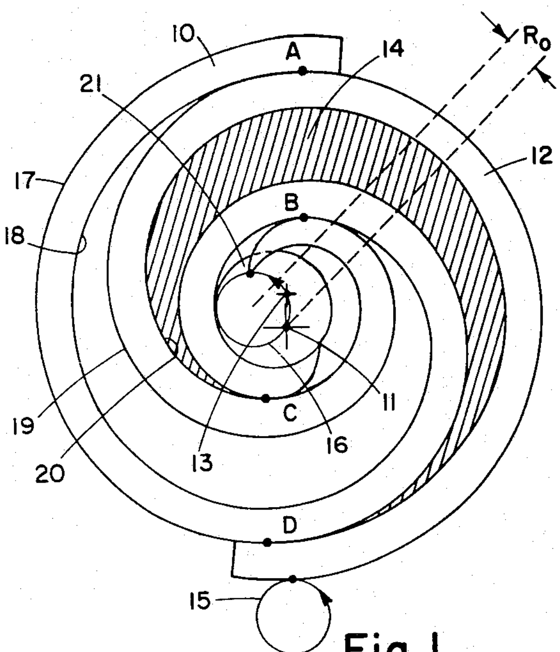


Fig. 1

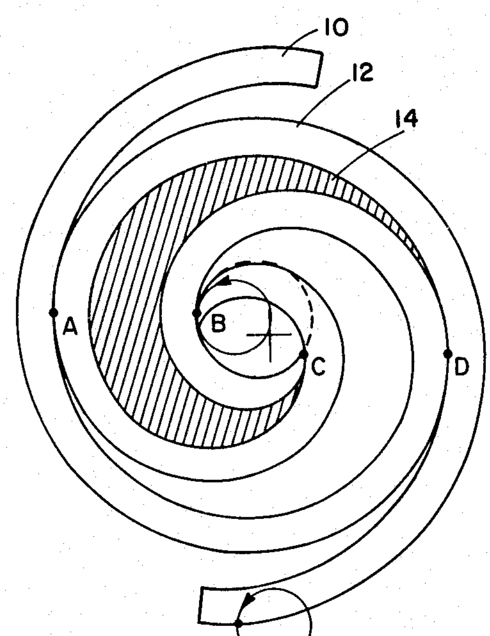


Fig. 2

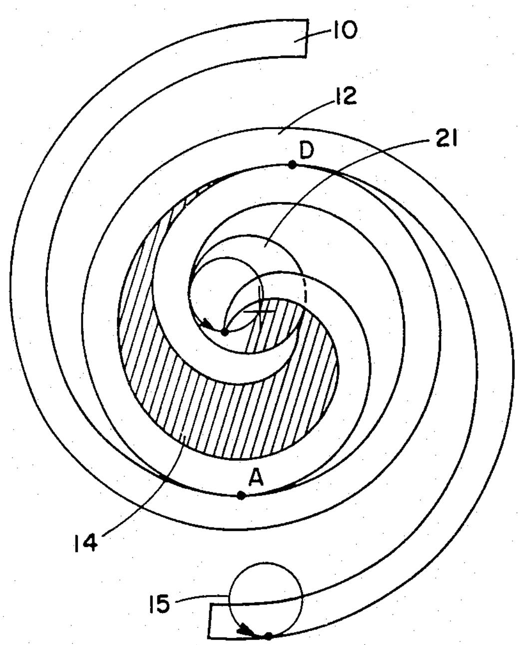


Fig. 3

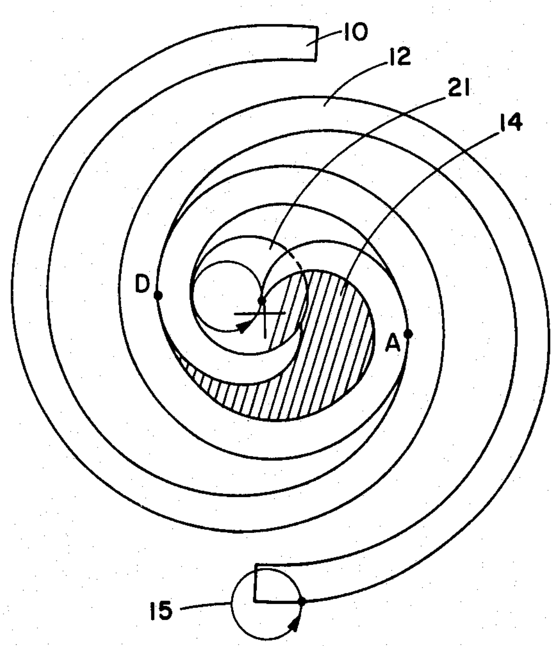


Fig. 4

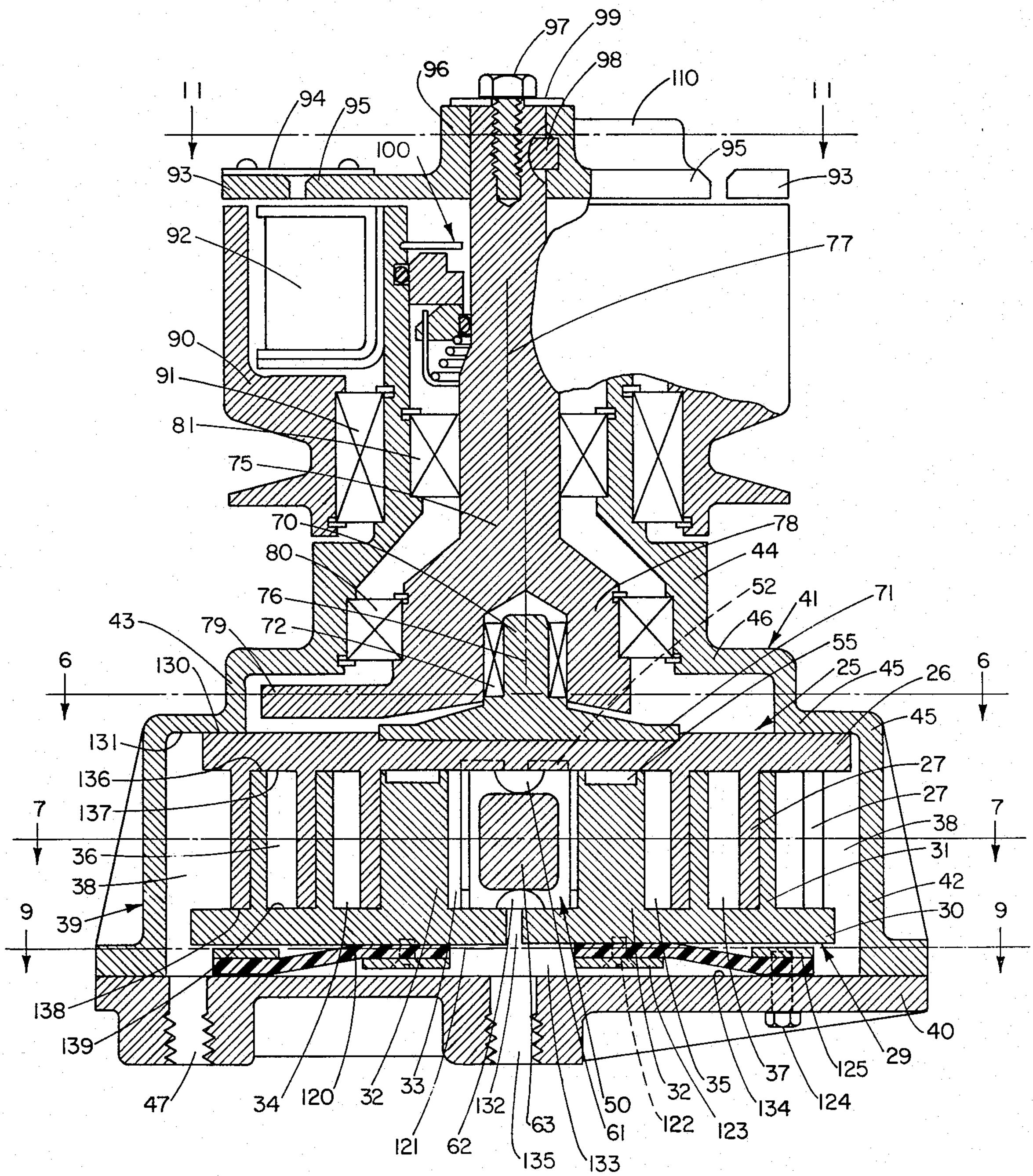

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Fig. 5

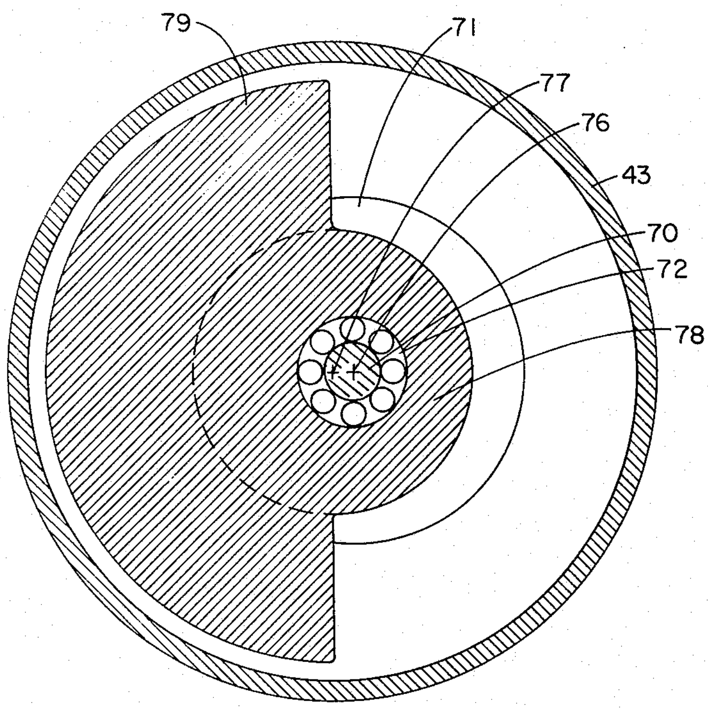

Fig. 6

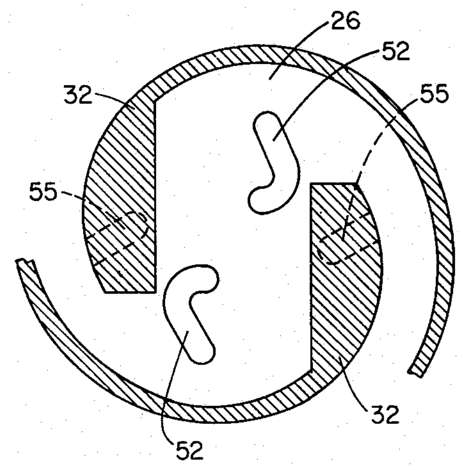

Fig. 8

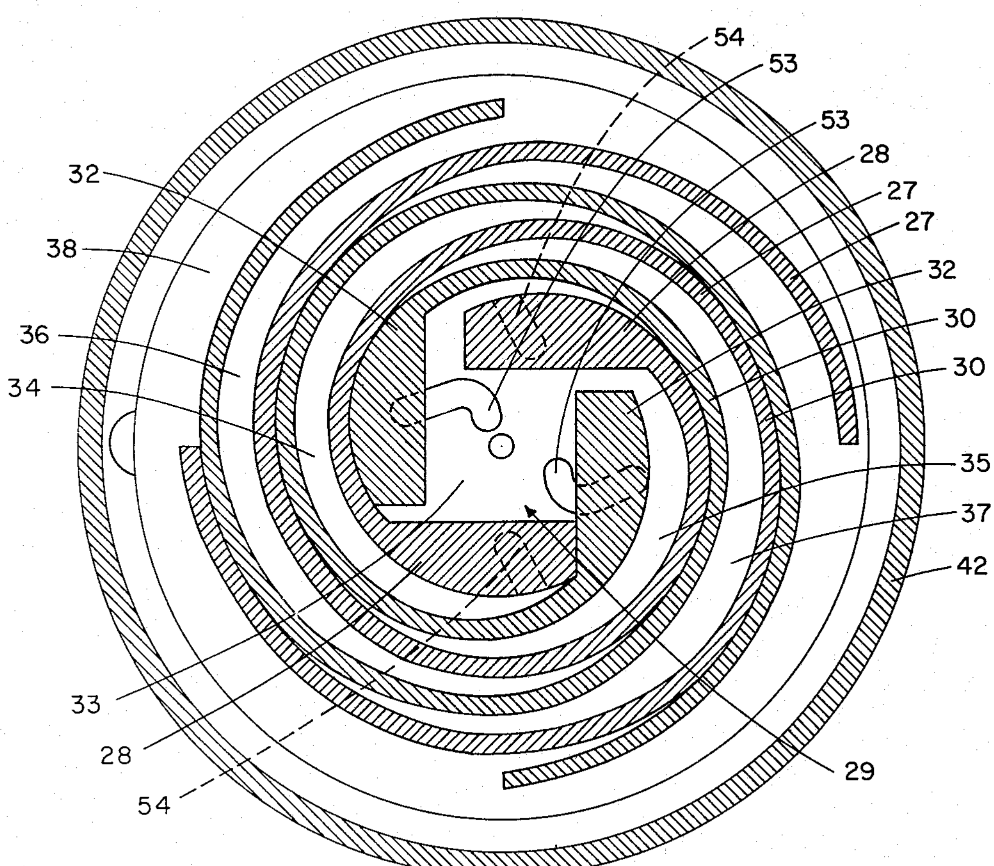

Fig. 7

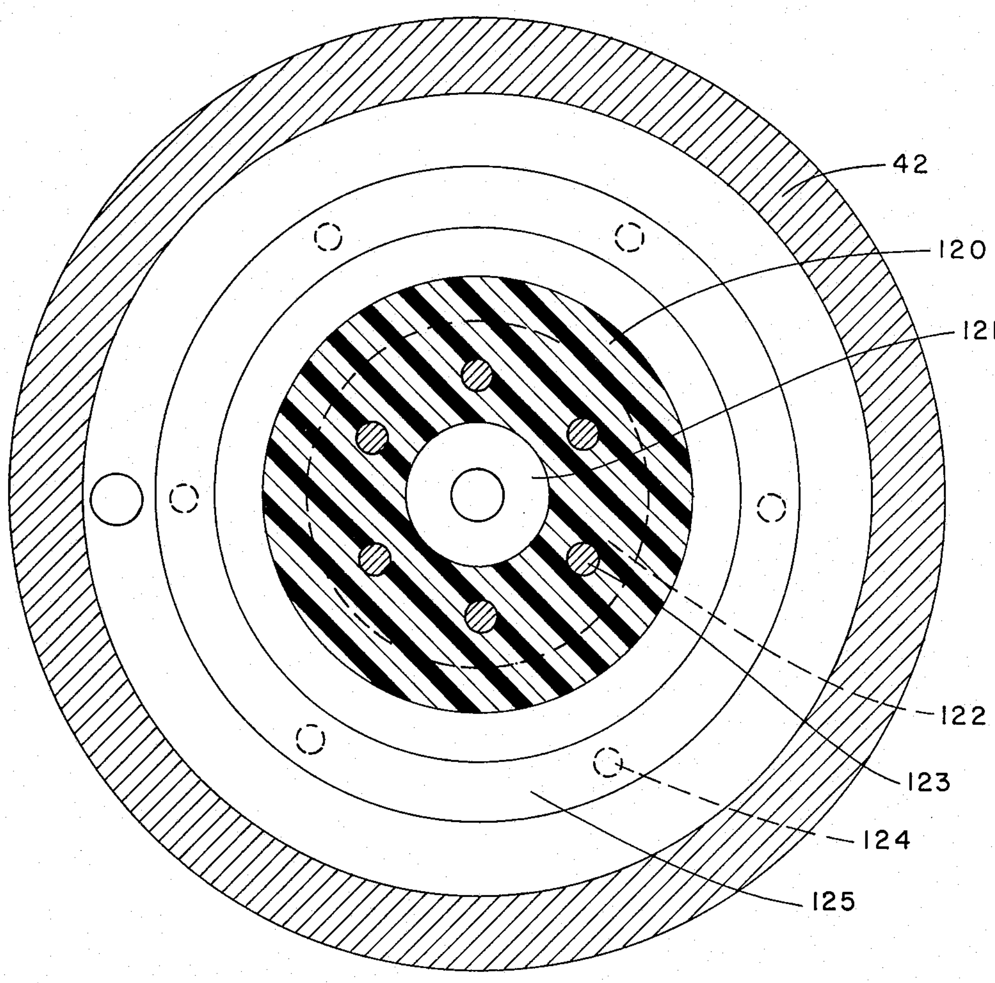

Fig. 9

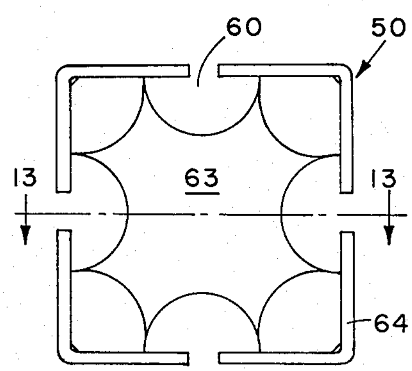

Fig.12

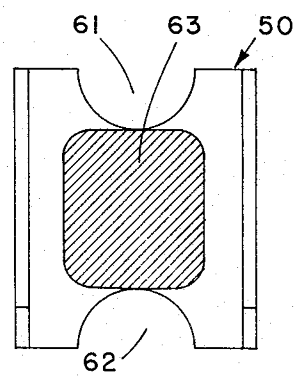

Fig.13

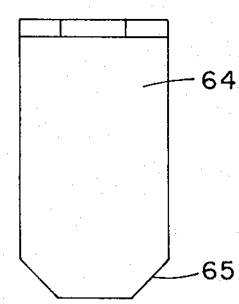

Fig.14

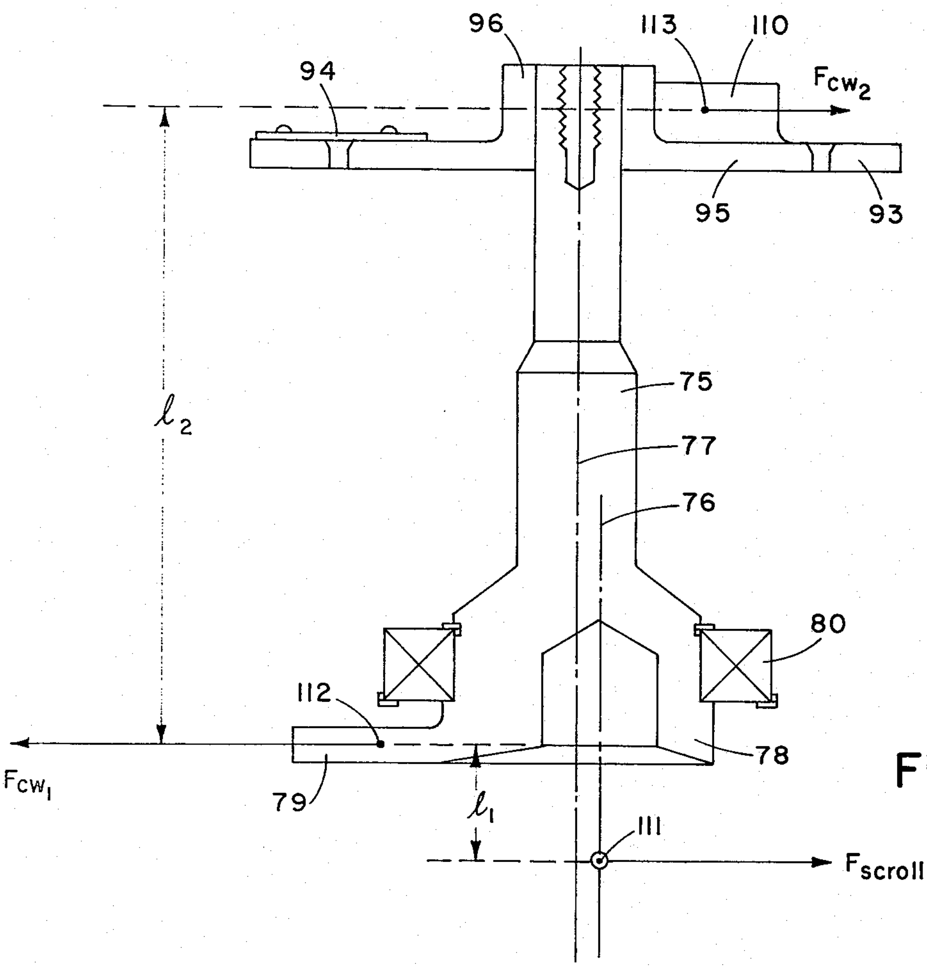
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Fig. 10

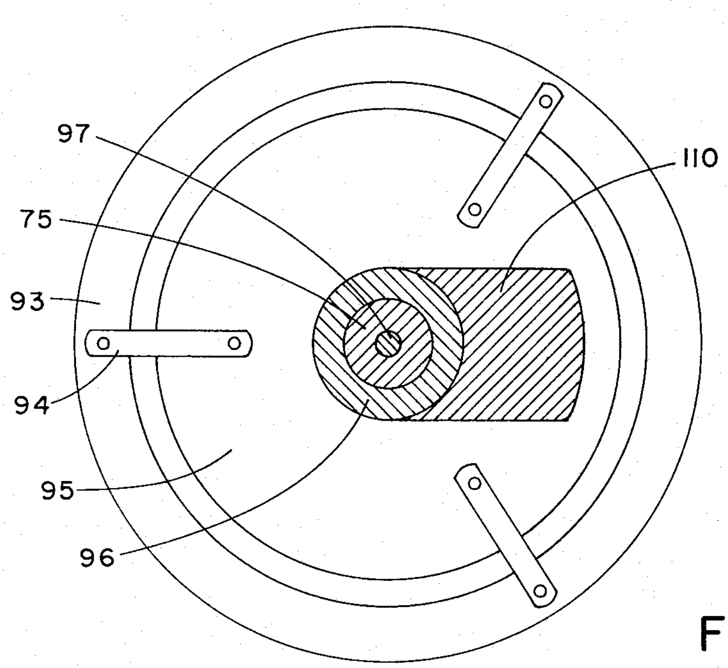
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Fig. 11

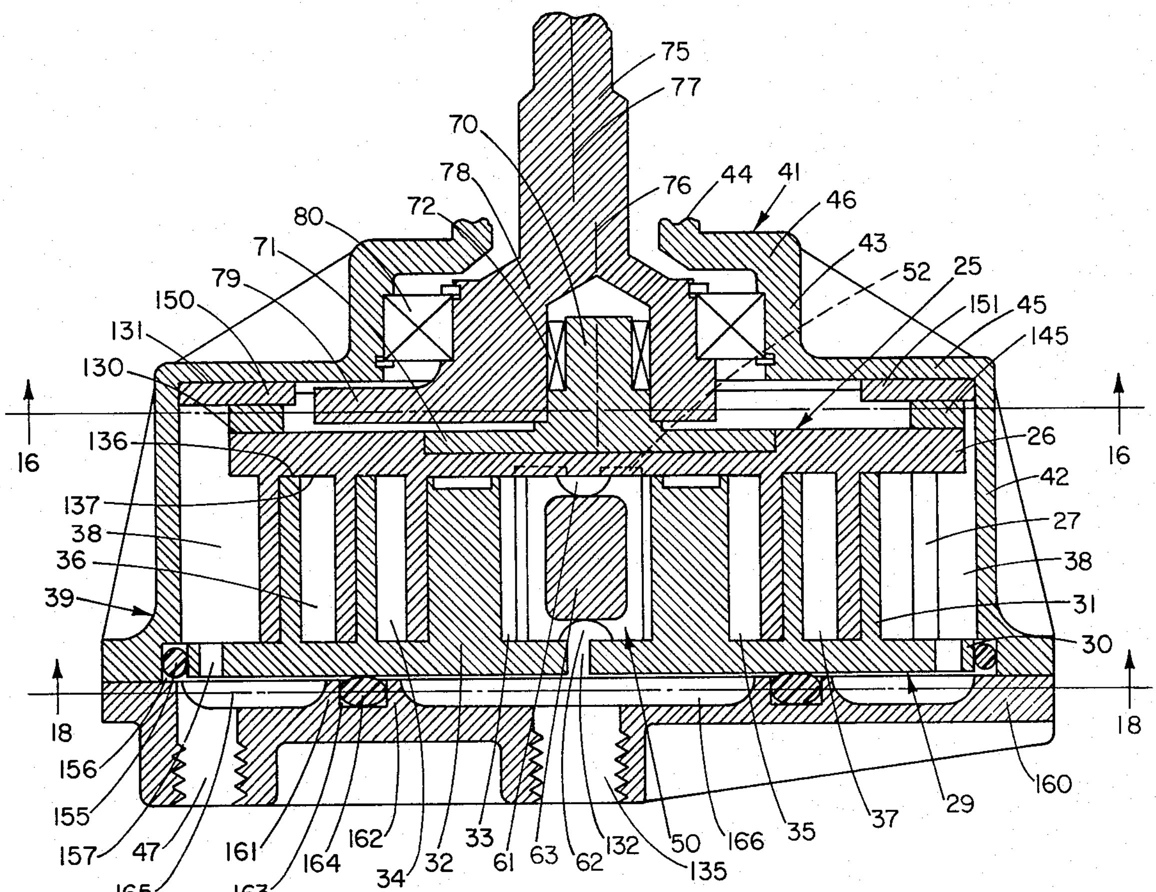
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Fig. 15

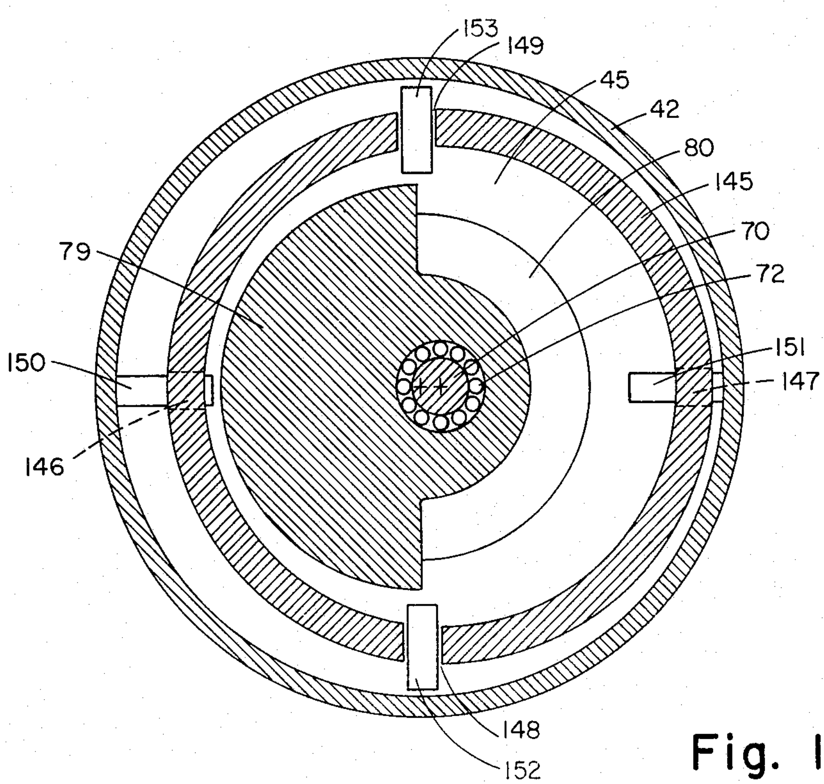
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Fig. 16

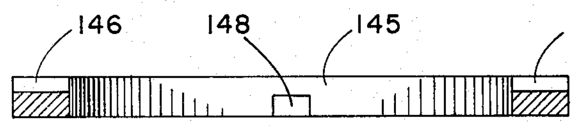

Fig. 17

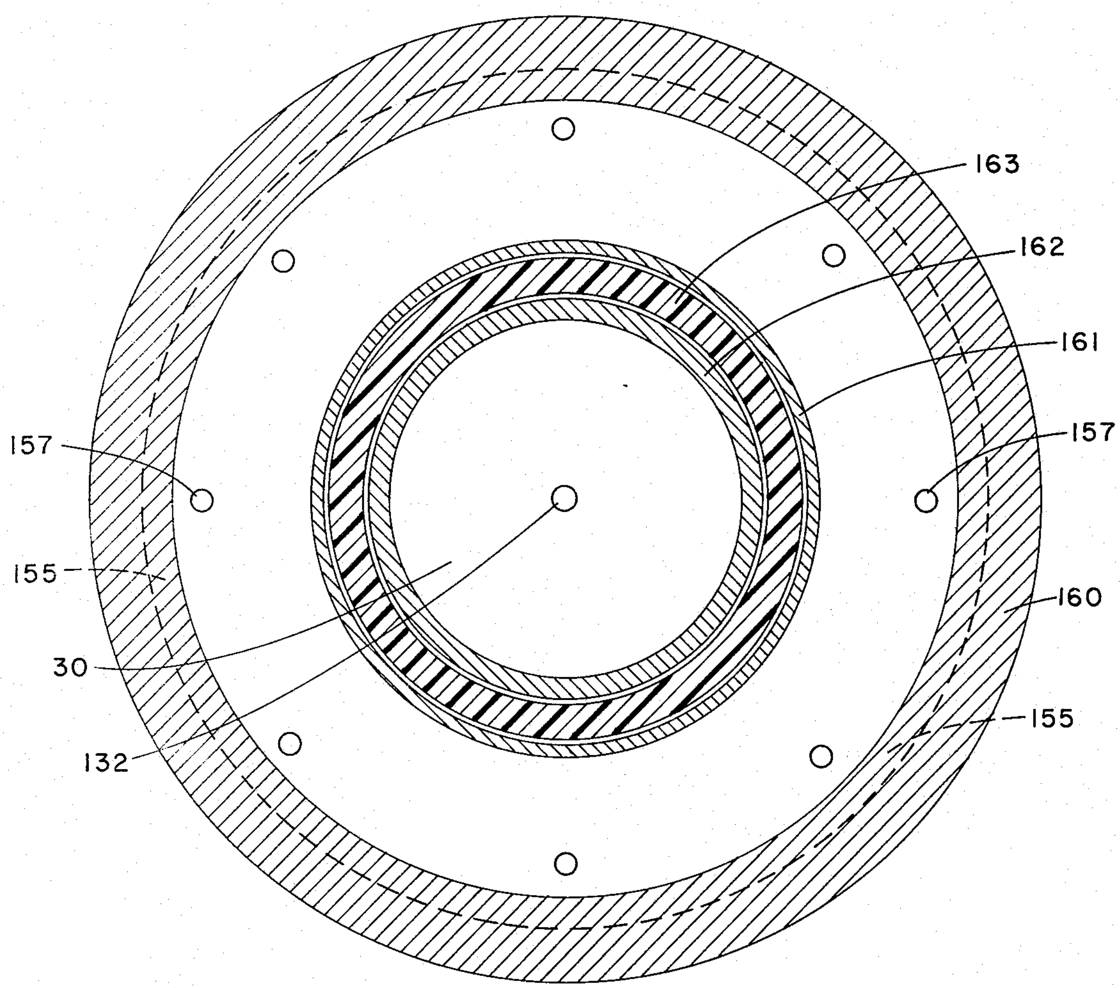

Fig. 18

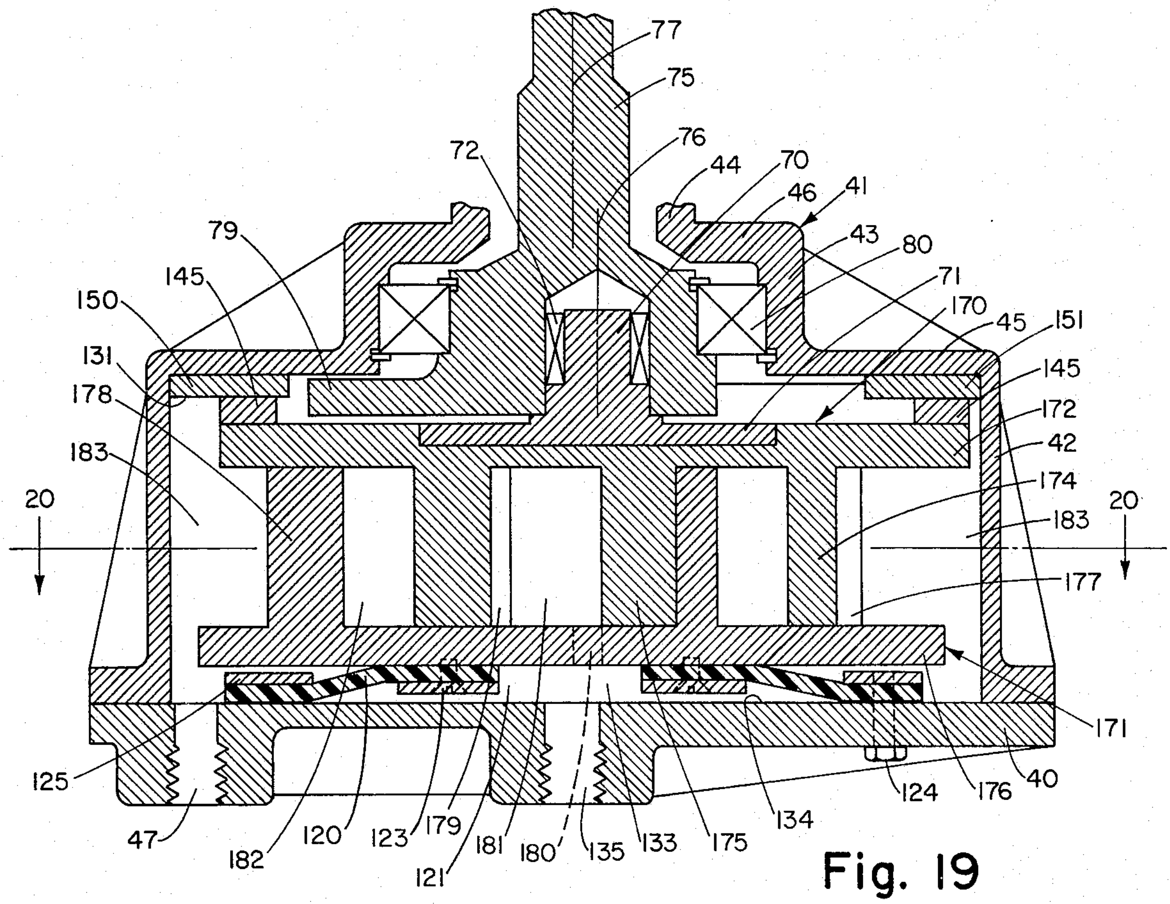
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Fig. 19

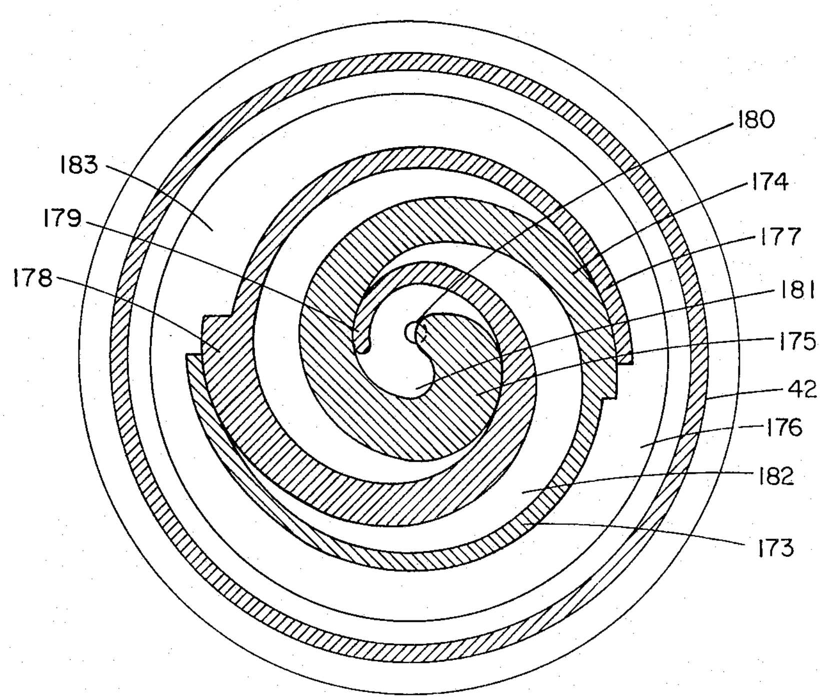
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Fig. 20

POSITIVE DISPLACEMENT SCROLL APPARATUS WITH AXIALLY RADIALLY COMPLIANT SCROLL MEMBER

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 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 cylinders. 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 which vary with the orbit angle of one of the spiral centers with respect to the other. As the angle between the centers of the two spirals changes, the angle of the radius which can be drawn between the line of contacts changes. 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 parts.

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 operations 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 long obtained.

The resulting solution 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 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 of sealing, wearing and porting problems heretobefore 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 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.

In copending application Ser. No. 368,907 filed on June 11, 1970, 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.

For some applications, particularly those such as in automobile air conditioners and the like, it would be desirable to have a scroll-type compressor which possessed all the advantages associated with this type of apparatus, and which at the same time is relatively simple to construct from a minimum number or parts and easy to maintain. Such a scroll-type apparatus should, of course, retain the advantages of maintaining efficient radial and axial sealing over extended periods of time as realized by the apparatus described in application Ser. No. 368,907.

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 from a minimum number of parts, 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 and wherein the axial and radial sealing forces are proportional to the total fluid pressure interval acting upon the apparatus.

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 equipment, automobile air conditioners etc. Other objects of the invention will in part be obvious and will in part be apparent hereinafter.

The scroll apparatus of this invention incorporates an orbiting scroll member rigidly affixed to an eccentric drive shaft and a unique "accommodating" scroll member which is free to move slightly in response to axial and radial forces applied thereto. The term "accommodating" is used to identify that scroll member which is fixed in the sense that it does not orbit except in response to force applied to it by the orbiting scroll member. Thus the accommodating scroll member may be said to perform a minor orbit of small dimensions. Therefore, the accommodating scroll member is constructed to experience small radial and axial excursions under externally applied axial forces and internally applied radial forces. The internally applied radial force is proportional to the fluid pressure within the pockets and is designed to maintain line contacts between the wrap members of the scroll members to achieve radial sealing; and the applied axial force is designed to counteract the internal fluid-generated axial forces to attain axial sealing between the scroll members. Since the radial force used to achieve radial sealing is other than the centrifugal force components produced by the orbiting of the orbiting scroll member and by the motion of the accommodating scroll member, means are provided to counteract or compensate for the centrifugal force components and to essentially eliminate all those centrifugal forces which would vibrate the frame of the apparatus as well as those centrifugal forces acting to cause unwanted forces at the line contacts between scroll members where radial sealing is effected. As will be seen in the detailed description of different apparatus embodiments, the means for applying radial and axial forces may take several different forms and their functions may overlap to some extent. There is thus provided a contacting, i.e., radial sealing, force which minimizes wear and which may be, if desired, 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 at least in part through pressurized fluid withdrawn from the zone of highest pressure.

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 longitudinal cross section of a compressor constructed in accordance with this invention and incorporated in a typical automotive type air conditioner;

FIG. **6** is a cross section taken through plane **6—6** of FIG. **5** showing the eccentric drive mechanism;

FIG. **7** is a cross section taken through plane **7—7** of FIG. **5** showing the wraps of the scroll members;

FIG. **8** is a planar view of a portion of the end plate of the orbiting scroll member and a cross section of a portion of the wraps of the accommodating scroll member;

FIG. **9** is a cross section taken through plane **9—9** of FIG. **5** showing one embodiment of a compliant attachment means joining the accommodating scroll member to the housing;

FIG. **10** is a side view of the drive shafts illustrating diagrammatically the attainment of static and dynamic balancing of the system;

FIG. **11** is a top cross section of the drive mechanism of FIG. **10** through plane **11—11** of FIG. **10.**

FIGS. **12, 13** and **14** are top planar, cross sectional and end views of one embodiment of a coupling member designed to prevent the relative angular motion of the two scroll members in an apparatus such as illustrated in FIG. **5**;

FIG. **15** is a longitudinal cross section of another embodiment of a compressor constructed in accordance with this invention;

FIG. **16** is a cross section taken through plane **16—16** of FIG. **12** showing the external coupling means;

FIG. **17** is a side elevational view of the external coupling means of FIG. **15**;

FIG. **18** is a cross section taken through plane **18—18** of FIG. **15** showing another embodiment of a complaint attachment means joining the accommodating scroll member to the housing;

FIG. **19** is a longitudinal cross section of yet another embodiment of a compressor constructed in accordance with this invention; and

FIG. **20** is a cross section taken through plane **20—20** of FIG. **19** illustrating another form of scroll member wraps.

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 orbiting 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. Although in the apparatus of this invention the accommodating scroll member does undergo small excursions axially and radially, it may still be considered to be the equivalent of a fixed scroll in the following discussion of FIGS. **1–4.**

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 or less about its axis, followed by a small translation to bring their centers together. The thickness, 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 begine the discharge of high-pressure fluid and this discharge of the high-pressure fluid is continued as shown in FIG. **4** unti such time as the moving wrap has completed its 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 delivery 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 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 mechanisms 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.

In the apparatus of this invention the disadvantages associated with scroll apparatus of the prior art are eliminated or minimized by counteracting all of the centrifugal forces acting between the scroll members, and by introducing a degree of compliance into the nonorbiting scroll which is constructed to be able to accommodate to axial and radial forces applied by the orbiting scroll, and in so doing to undergo small excursions in the axial and radial directions while maintaining the axial and radial sealing of the scroll members. The coupling means, which maintains the desired angular relationship of the scroll members, is independent of the radial sealing means; but the coupling means may form a part of that portion of the apparatus used to attain the axial positioning of the scroll members. The operation of the scroll apparatus is designed so that fluid pressure events between scroll members act to maintain radial and axial sealing forces between the scroll members. Hence, wear of the line contacts between the wraps of the scroll members is essentially self-compensating and thus efficient continued radial sealing is attained over extended periods of operation. Axial sealing is accomplished at least in part by using gas from the highest pressure zone of the apparatus and such gaseous sealing forces are augmented by suitable elastic biasing means to continuously force the scroll members to make axial contact during start-up and shut-down.

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. 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, to act upon the accommodating scroll member and to augment this gas force with a suitable biasing spring force, particularly for start-up and shut-down.

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 determine radial sealing of the scroll members comprise a fraction of the tangential force due to the reaction of the fluid within the scroll volume which is resolved by mechanical radial constraints, a fraction of the centrifugal force due to the orbiting of the orbiting scroll member, and a force due to springs or other compliant elements which may act in the radial direction. Any combination of these radially acting forces can be used to effect radial sealing by which one scroll is pressed into line contact with the other scroll member to seal off the pockets. The centrifugal force is in excess of that which is required to attain efficient radial sealing and does in fact bring about excessive wear of the cylindrical surfaces of the wrap members. In the apparatus of this invention, therefore, these centrifugal forces are essentially eliminated and replaced by a fraction of the tangential forces due to the reaction of the fluid being compressed within the scroll pockets. This is in contrast to the apparatus of Ser. No. 368,907 wherein a fraction, but not all, of the centrifugal force is counteracted by a centripetal force developed by the driving means. In contrast to the practice of this invention as well as that of Ser. No. 368,907, is the prior art teaching which discloses the use of an augmented centrifugal force to attain radial sealing. (See for example British Specification 486,192).

Embodiments illustrating the scroll apparatus of this invention are shown in FIGS. **5–20.** FIG. **5** is a longitudinal cross section of one embodiment wherein the coupling means is located within the zone of highest pressure, the compliant attachment means is a spring-loaded compliant disk member joining the accommodating scroll member to the housing and the means for applying an axial force is the compliant member in combination with fluid pressure, the fluid being derived from the zone of highest pressure defined by the scroll members.

To illustrate one possible use for the compressor of this invention, FIG. **5** shows it incorporated into the drive mechanism of a typical FREON compressor as used in an automobile air conditioner. It will, of course, be appreciated that the embodiment of the scroll apparatus illustrated in FIG. **5** may be used as an expander or a pump and may be used in applications other than in automobile air conditions.

In the apparatus of FIG. **5**, the orbiting scroll member, generally indicated at **25**, is formed of an end plate **26** and wraps **27** which terminate in enlarged inner sections **28** (FIG. 7) and the accommodating scroll member, generally indicated at **29**, is formed of an end plate **30** and wraps **31** which terminate in enlarged inner sections **32.** In the following description of the embodiment of FIG. **5,** reference should also be made to FIGS. **6–14** which include transverse cross sectional drawings and an end view of a portion of the orbiting scroll member. A series of fluid pockets such as **33–37** are defined between the wraps **27** and **30** and their enlarged inner sections **28** and **32**, and a peripheral or low-pressure plenum **38** is defined between the wraps and the interior wall of a housing. The zone of lowest pressure represented by peripheral plenum **38** is connected through low-pressure port **47** to a low-pressure line not shown. The housing, generally indicated at **39**, comprises a bottom plate member **40** and a main housing **41** made up of scroll housing section **42,** counterweight housing section **43** and a bearing and shaft housing section **44** all formed, if desired, as one integral piece. The housing **39** is formed in a generally stepped configuration, and the scroll housing section **42** is joined to the counterweight housing section **43** by a shoulder **45**; while the counterweight housing section is in turn joined to the bearing and shaft housing section through shoulder **46.**

Fluid pocket **33** is the zone of highest pressure and in this embodiment the coupling means **50** is positioned therein. (The coupling means is omitted in FIG. 7 to better illustrate the porting system.) Inasmuch as the coupling means occupies a large portion of the rectangularly-configured zone of highest pressure, passages must be provided to connect the highest-pressure zone **33** with the moving fluid pockets adjacent it. This is accomplished through shallow passages **52** cut in end plate **26** (FIG. **8**) and comparable passages **53** cut in end plate **29** (FIG. 7) of the orbiting and accommodating scroll members, respectively, and shallow passages **54** and **55** cut in the contacting edges of the enlarged central sections **28** and **32** of the wrap members of the orbiting and accommodating scroll members, respectively. It will be seen that as scroll member **25** orbits, the shallow passages in the enlarged wrap sections are periodically opened to the shallow passages in the end plates to complete a fluid path between the central pocket **33** and an adjacent inwardly moving pocket.

Coupling member **50** is detailed in FIGS. **12–14** 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 pocket, it must be configured to provide fluid passages within the pocket to communicate with the flow paths defined by passages **52–55** and through them to the adjacent fluid pockets as well as to the porting system described below. The coupling member therefore has relatively large cutouts **60** on each side and cutouts **61** and **62** on the top and bottom leaving, in effect, a solid central piece **63.** Right-angled sealing plates **64** are located at each corner and these sealing plates are beveled at **65** to permit the opening of fluid communication between the inner fluid pocket and outer adjacent pockets through the fluid paths described. It will be appreciated that with the orbiting of scroll member **25,** this fluid communication path shifts to use first one flow path and then another.

The orbiting scroll member **25** must be driven to cause it to orbit in the manner explained in the description of FIGS. **1-4.** This is most conveniently done by mounting orbiting scroll endplate **26** rigidly to a scroll drive shaft **70** through a shaft plate **71** configured to serve as retaining means for a bearing **72** through which scroll drive shaft **70** is eccentrically connected to main drive shaft **75.** It will be seen that the axis **76** of the scroll drive shaft is spaced from and parallel to axis **77** of main shaft **75.** The distance between axes **76** and **77** is the orbit radius of scroll member **25.** Main drive shaft **75** has an enlarged diameter section **78** in which scroll drive shaft **70** is mounted. Affixed to or integral with this enlarged main shaft section is the primary counterweight **79** used to form a part of the means to counterbalance the centrifugal force component of the orbiting scroll member. Main shaft **75** is mounted for rotation within the bearing and shaft section **44** of the main housing through bearings **80** and **81.**

FIG. **5** illustrates the attachment of a magnetic clutch and sheave of a typical Freon compressor to the main shaft. This portion of the apparatus is, of course, not part of the invention herein described, but it is included to illustrate one way in which the rotation of shaft **75,** and hence the operation of the compressor, may be effected. Briefly the magnetic clutch and sheave can be seen to comprise a pulley **90** mounted for rotation about housing section **44** through a bearing **91**, a solenoid coil **92**, an armature **93** joined through springs **94** (FIG. **11**) to a clutch boss **95** which is rigidly connected to shaft **75** through a collar **96**, screw **97**, key **98** and washer **99**. Between the upper end of main drive shaft **75** and shaft and bearing housing section **44** is a shaft seal of standard design and generally indicated at **100**.

A secondary counterweight **110** is mounted on, and preferably integral with, main drive shaft **75** and is a part of collar **96**. Counterweights **79** and **110** make up the means for counterbalancing the centrifugal force developed by the orbiting scroll as well as the centrifugal force component which is generated by the motion of the accommodating scroll member in moving circularly in its minor orbit. The balancing of the major centrifugal components due to the orbiting scroll member may be described with reference to FIG. **10** in which only the main drive shaft and counterweights are shown. The center of gravity of the orbiting scroll is indicated at **111**, of the primary counterweight (cw_1) at **112** and of the secondary counterweight (cw_2) at **113**; and the distances between these centers of gravity are designated l_1 and l_2. The direction and relative magnitude of the centrifugal forces are indicated by the arrows. From elementary force considerations it will be seen that dynamic balance of these centrifugal forces is attained when

$$F_{scroll} \times l_1 = F_{cw} \times l_2$$

and that static balance is attained when

$$F_{cw} = F_{scroll} + F_{cw}\ .$$

The balancing of the centrifugal component due to the accommodating scroll member's performing its minor orbit is accomplished in a similar manner to and concurrently with the balancing of the centrifugal force of the orbiting scroll member. This combined balancing of the centrifugal components is accomplished by a small angular displacement of counterweights **79** and **110** in a direction around the main drive shaft. By having this small angular displacement the centrifugal components which are due to the minor orbit of the accommodating scroll member and which are directed vertical to the plane of FIG. **10** can be balanced out both dynamically and statically.

In determining this angular displacement it is first necessary to note that counterweights **79** and **110** lie in a plane which includes the axis of the main shaft and that this plane would contain the center of gravity of the orbiting scroll member if only the orbiting scroll member were to be balanced. However, in order to balance the centrifugal force components of both the orbiting scroll member and accommodating scroll member, it is necessary to cause the plane of counterweights **79** and **110** to occupy a position between the plane formed by the center of gravity of the orbiting scroll member and the shaft axis and the plane formed by the center of gravity of the accommodating scroll member and the shaft axis. The angle between these two planes, i.e., the displacement angle, will be determined for each apparatus design and it will depend primarily upon the following factors: the orbit radius of the orbiting scroll member, the minor orbit radius of the accommodating scroll member, the radius at which one scroll member moves with respect to the other, the centrifugal force which has been engineered to act between the two scroll members, the degree of compliance of the accommodating scroll member with respect to the frame of the machine and the size of the tangential driving force relative to the radial sealing force which is required.

By proper choice of counterweights and their positioning on the main drive shaft **75**, taking into account the abovelisted factors, it is possible to counterbalance all of the centrifugal forces acting upon the scroll members. With the elimination of all centrifugal force components which would, in the absence of counterweights **79** and **110**, act between the scroll members as well as upon the frame to vibrate it, it is possible to apply controllable radial forces to achieve controlled radial sealing.

In any practical embodiment of the scroll apparatus of this invention there will be random inertial force effects (due primarily to machining errors) which may tend to detract from the attainment of effective radial sealing. Any such forces will be proportional to the square of the operational speed of the machine. It may, therefore, be desirable to have a small fraction of the centrifugal force component available to counterbalance such inertial force effects. Since any centrifugal force component is likewise proportional to the square of operational speed, such balancing of inertial force with a small fraction of centrifugal force is desirable and is readily accomplished. The required amount of such centrifugal force, if employed, may easily be realized by adjusting the size and location of the counterweights as well as the displacement angle described above. Thus, it is possible to allot a small fraction of the centrifugal components to counterbalance other unwanted force effects while still achieving the desired radial sealing forces. The important feature of this invention lies in the fact that the centrifugal forces of both of the scroll members can be brought completely under control in the manner described. No unwanted portions of the centrifugal forces need be tolerated and the control of these centrifugal components is an essential feature in regulating the necessary radial sealing forces between the two scroll members.

An important part of this control of centrifugal component lies in permitting the accommodating scroll member to undergo small excursions in the radial direction by affixing the accommodating scroll member to the housing (i.e., bottom plate member **40**) through compliant attaching means. In the embodiment of FIG. **5** this compliant attaching means comprises a strong but somewhat elastic disk **120** having a central opening **121** and being spring loaded to apply a small force to urge the wraps of scroll member **29** to contact the end plate of scroll member **25**. This disk is preferably formed of a material which possesses some degree of elasticity, such as for example a fabric-reinforced elastomer. It is attached to the bottom (outside) surface of the end plate **30** of the accommodating scroll member in any suitable manner such as by screws **122** and washer **123** and to the housing by means of bolts **124** and washer **125**. This compliant attachment of the accommodating scroll member to the housing permits the accommodating scroll to move radially in response to the pressure events within the fluid pocket and thus to be continuously forced against the orbiting scroll to make the line contacts between the wraps in order to achieve radial sealing. Thus radial sealing is attained with minimum wear and in a manner to be at least partially self-compensating. In addition to providing radial accommodating for the accommodating scroll member, compliant disk **120** also prevents the accommodating scroll from rotating and hence serves as an additional secondary coupling means, and further it spring loads the nonorbiting scroll member in the radial sealing direction at low speeds and/or low pressure conditions especially during start-up and shut-down.

Axial sealing is accomplished in part by fluid forces and in part by the compliant attaching means. As will be seen in FIG. **5** the outer surface **130** of the orbiting scroll member **25** (i.e., the surface opposite to that to which wraps **27** are affixed or integral therewith) bears against the inner surface **131** of shoulder **45** of the main housing and the sliding between these two surfaces **130** and **131** takes up axial loads acting upon the orbiting scroll member and maintains the motion of the orbiting scroll member within a plane. The forces applied to effect this are through the accommodating scroll member **29** and are derived from high-pressure fluid withdrawn from the zone of highest pressure **33** as well as from the configuration of compliant disk **120.** The high-pressure fluid porting system comprises a central port **132** in end plate **30** of the accommodating scroll member, the fluid sealing chamber **33** defined between the compliant disk **120** and the internal surface **134** of bottom member **40,** and a high-pressure fluid passage **135** in bottom member **40** which is preferably threaded for easy connection to a threaded fluid line not shown. This high-pressure fluid porting system thereby provides for axial sealing through the action of the high-pressure fluid in chamber **133** forcing the accommodating scroll member upwardly to engage in a small axial excursion, and ensure that the contacting surfaces **136** of the accommodating scroll wraps **31** and **32** make sealing contact with the inner surface **137** of the end plate of the orbiting scroll and likewise the contacting surfaces **138** of the orbiting scroll wraps **27** and **28** make sealing contact with the inner surface **139** of the end plate of the accommodating scroll member to achieve axial sealing.

Since during start-up and shut-down, the fluid pressure in fluid sealing chamber **133** may not be sufficient to effect complete axial sealing, the compliant disk is configured to axially spring load the accommodating scroll member so that it effects axial sealing during these transient periods of operation. The compliant disk also serves to augment the axial sealing force of the high-pressure fluid in sealing chamber **133** during normal operation.

FIG. **15** illustrates an embodiment of this invention in which the coupling means comprises two coupling members, one positioned within the volume defined between the scroll members and the other positioned between the orbiting scroll member and the housing; the compliant connection between the accommodating scroll member and the housing comprises elastomeric sealing rings; radial sealing is effected by one of the sealing rings; and axial sealing is attained by means of the other sealing rings and fluid pressure in a fluid sealing chamber. In FIG. **15** like reference numerals are used to identify like apparatus elements of FIG. **5.**

Since the constructions of the orbiting scroll member **25,** of the accommodating scroll member **29** (with the below-noted exceptions), of the housing **39** and of the driving means, along with the means to counterbalance centrifugal forces, are essentially the same as described for the embodiment of FIG. **5,** these elements will not be described again. The cross sectional and elevational views of FIGS. **7, 8** and **10–14** are applicable to the embodiment illustrated in FIG. **15.**

In the following description of the embodiment of FIG. **15,** reference should also be had to FIG. **16-18** which are cross sections of the "external" coupling means and of the compliant attachment means. In the embodiment of FIG. **15,** an additional "external" coupling member **145** is used to prevent the orbiting scroll member **25** from rotating within the housing. The term "external" is used to differentiate coupling **145,** which is external of the volume defined between the scroll members, from coupling member **50** which, as previously noted, is located within the zone of highest-pressure **33.** External coupling **145** comprises a ring having oppositely disposed channels **146** and **147** cut into that surface facing the internal surface **131** of shoulder **45** of the housing and two oppositely disposed channels, **148** and **149** cut into the ring surface which makes contact with outer surface **130** of orbiting scroll member **25.** The center line of channels **146** and **147** form a right angle with the center line of channels **148** and **149.** Affixed to, or integral with, the inner wall of housing shoulder **145** are four equally spaced keys **150,** 151, 152 and **153** arranged in oppositely disposed pairs for engagement is coupling ring channels **146–149,** as best shown in FIG. **16.** This external coupling means (channeled ring and engaging keys) prevents orbiting scroll member **25** from rotating within the housing; while the internal coupling member **50** prevents the relative rotation of the orbiting and accommodating scroll members.

The compliant attachment between accommodating scroll member **29** and the housing comprises an elastomeric sealing ring **155** between the periphery of end plate **30** of the accommodating scroll member and the wall of groove **156** cut into the internal wall of scroll housing section **42** of the main housing. Inasmuch as sealing ring **155** isolates peripheral pocket **38** from the low-pressure port **47,** plurality of fluid ports **157** are cut through end plate **30** of the accommodating scroll. The internal surface of the bottom plate **160** of the main housing is configured to have two concentric rings **161** and **162** which define a groove **163** adapted to seat a second elastomeric sealing ring **164.** This sealing ring is sized such that it provides an axial force against the accommodating scroll member to force it in axial sealing contact with the orbiting scroll member, particularly during start-up or shut-down. An annular low-pressure passage **165** is thus defined between end plate **30** and bottom member **160** and it provides fluid communication between low-pressure port **47** and low-pressure pocket **38** via ports **157.** This low-pressure annular passage is sealed from the central high-pressure fluid sealing chamber **166** which is in fluid communication with pocket **33** (the zone of highest pressure) through port **132.** Sealing chamber **166** serves the same role as chamber **133** of FIG. **5,** that is, it provides an axial sealing force acting against the accommodating scroll member proportional to the highest pressure obtaining within the scroll apparatus. During operation, this axial sealing force is augmented by the axial force of elastomeric ring **164.**

It will be seen that sealing rings **155** and **164** in combination serve the same function as the compliant disk **120** of the embodiment of FIG. **5,** that is they permit small axial and radial excursions of the accommodating scroll member to attain controlled axial and radial sealing along with self-regulating wear of the contacting surfaces. The external coupling means of FIG. **15** assumes that portion of the compliant disk's function of preventing the rotation of the orbiting scroll member relative to the housing.

In the embodiment shown in FIG. **19,** it is possible, by using the external coupling member of FIG. **15** with the compliant disk of FIG. **5** to eliminate the internal coupling member. This in turn permits the use of scroll wraps and a central porting arrangement which, in combination, make it possible to achieve relatively large pressure ratios without having to resort to grossly enlarged wrap sections and/or narrow fluid passages. Thus the configurations of the wraps may be optimized and the wraps of the orbiting and accommodating scroll members may be different and need not be involute spirals of identical pitch.

In the embodiment of FIG. **19,** like reference numerals are used to identify like components shown in FIGS. **5** and **15.** Thus the grooved external coupling member **145,** along with the keys **150-153** on the internal wall of shoulder **45,** serves to prevent angular motion of orbiting scroll member **170**; and compliant disk **120,** serving as an attachment between accommodating scroll **171** and bottom housing member **40,** serves to permit small axial and radial excursions of the accommodating scroll member to effect axial and radial sealing.

The orbiting scroll member **170** is comprised of an end plate **172** and a wrap which has an outer, thin-walled section **173** and an inner thicker-walled section **174** terminating at the center of the scroll member in an even thicker-walled section **175** (FIG. **20**). The accommodating scroll member **171** is comprised of an end plate **176** and a wrap which has, from outside to inside, a thin-walled section **177** and a thick-walled section **178** which tapers to an inner thin-walled section **179.** End plate **176** of the accommodating scroll has an off-center port **180** which communicates with fluid sealing chamber **133** and through it to high-pressure port **135.** It will be evident from FIG. **20** that as orbiting motion is imparted to the wrap of the orbiting scroll member, the thickest-walled section **175** of that wrap closes and opens port **180.** By proper choice of the configuration of orbiting wrap section **175** and of the location and size of port **180,** the timing of high-pressure fluid discharge or introduction can be set. By proper choice of the configurations of both wraps, the relative sizes of the highest-pressure pocket **181,** the intermediate-pressure pocket **182** and the low-pressure pocket **183** may be set, thus fixing the pressure ratio.

It will, of course, be apparent that many other wrap configurations may be used including, in addition to involute circles, arcs of circles, multiple wraps for each scroll member, dual wraps which fair into single wraps, etc. It is also within the scope of this invention to use any suitable porting systems and to use other eccentric drive arrangements, e.g., a main drive shaft having an eccentrically placed stud shaft connected to the orbiting scroll member.

It will be seen from the above description of the scroll type apparatus of this invention that by counterbalancing essentially all of the centrifugal forces, and by giving the accommodating scroll member axial and radial compliance with relation to the housing and hence with relation to the orbiting scroll, it is possible to employ the pressure events within the volume defined by the wraps of the scroll member as primary axial and radial sealing forces. The number of apparatus components is minimized, wear is minimized and largely self-compensating, and flexibility in porting, pressure ratios and wrap designs is possible.

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, and in which first and second 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 said first scroll member is driven to perform an orbit with respect to said second scroll member and to a stationary frame to which said apparatus is mounted while maintaining a fixed angular relationship between said scroll members and said stationary frame, the improvement which comprises compliant attachment means joining said scroll member to said stationary frame in a manner to permit said second scroll member to undergo excursions in the axial direction in response to externally applied axial forces and to undergo excursions in the radial direction solely in response to internally applied radial forces; and means to counterbalance all of that part of the centrifugal force component produced by the orbiting of said scroll members which would otherwise increase the radial sealing force.

2. A positive fluid displacement apparatus, comprising in combination

a. a first orbiting scroll member having first wrap means affixed to a first end plate;

b. a second accommodating scroll member having second wrap means affixed to a second end plate and providing for moving line contacts with said first wrap means when said first scroll member is positively driven to orbit said second member, thereby to seal off and define at least one moving fluid pocket of variable volume and zones of different fluid pressures between said scroll members;

c. coupling means maintaining said first and second scroll members in a fixed angular relationship;

d. stationary housing means defining a chamber in which said scroll members are located;

e. axial force applying means for applying an axial force external of said fluid pocket to counteract the internal fluid-generated axial forces thereby to attain axial sealing of said first and second scroll members;

f. compliant attachment means joining said second scroll member to said stationary housing means in a manner to permit said second scroll member to undergo excursions in the axial direction in response to said axial force applied by said axial force applying means and to undergo excursions in the radial direction and thereby perform a minor orbit solely in response to internal radial forces, at least a portion of which is applied by fluid pressure within said fluid pocket;

g. drive means including a main drive shaft rigidly connected to said first scroll member for orbiting said first scroll member in a full orbit radius rigidly defined with respect to said housing and to said second scroll member; and

h. means associated with said drive means for counterbalancing essentially all of that part of the centrifugal force component produced by the orbiting of said first and second scroll members which would otherwise increase the radial sealing force.

3. A fluid displacement apparatus in accordance with claim **2** wherein said coupling means is positioned within the zone of highest pressure.

4. A fluid displacement apparatus in accordance with claim **2** wherein said coupling means is external of said fluid pocket and couples said orbiting scroll member to said housing means.

5. A fluid displacement apparatus in accordance with claim **2** wherein said coupling means is external of said fluid pocket and couples said first and said second scroll members.

6. A fluid displacement apparatus in accordance with claim **2** wherein said coupling means comprises a first coupling member positioned within the zone of highest pressure and a second coupling member external of said fluid pocket and adapted to couple said orbiting scroll member to said housing means.

7. A fluid displacement apparatus in accordance with claim **2** wherein said compliant attachment means comprises a spring-loaded stiff but flexible disk joined to said nonorbiting scroll member and said housing means.

8. A fluid displacement apparatus in accordance with claim **2** wherein said compliant attachment means comprise first sealing ring means positioned to apply an axial force against said second end plate and second sealing ring means positioned between the periphery of said second end plate and the wall of said housing means.

9. A fluid displacement apparatus in accordance with claim **2** wherein said drive means comprises a main drive shaft and a scroll drive shaft connected thereto, the axes of said main drive shaft and said scroll drive shaft being parallel and displaced from each other by a distance equal to the radius of orbit of said orbit scroll member.

10. A fluid displacement apparatus in accordance with claim **2** wherein said means for counterbalancing said centrifugal force component and said motion of said second scroll member comprise counterweight means attached to said main drive shaft.

11. A fluid displacement apparatus in accordance with claim **2** including means external of said fluid pocket to apply a radial force to said second scroll member.

12. A fluid displacement apparatus in accordance with claim **2** wherein said axial force applying means comprises means to define a fluid-tight high-pressure sealing chamber in force-applying relationship with said second scroll member and fluid passage means between said sealing chamber and the zone of highest pressure.

13. A fluid displacement apparatus in accordance with claim **12** wherein said axial force applying means provides a force greater than that required to counteract said internal fluid-generated axial force and includes said compliant attachment means.

14. A fluid displacement apparatus in accordance with claim **2** including orbiting scroll member coupling means to couple said first scroll member to said housing means thereby to maintain a fixed angular relationship between said first scroll member and said housing means.

15. A fluid displacement apparatus in accordance with claim **14** wherein said orbiting scroll member coupling means comprises a ring with grooves and key members affixed to said housing engagable with said grooves.

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