

Description

This invention relates to a scroll type fluid displacement apparatus according to the preamble of Claim 1, and known from GB—A—1165435.

Scroll type fluid displacement apparatus are well known in the prior art. For example, U.S. Patent No. 801,182 discloses a scroll type fluid displacement apparatus including two scroll members, each having a circular end plate and a spiroidal or involute spiral element. These scroll members are maintained angularly and radially offset so that both spiral elements interfit to make a plurality of line contacts between spiral curved surfaces to thereby seal off and define at least one pair of fluid pockets. The relative orbital motion of the two scroll members shifts the line contacts along the spiral curved surfaces and, therefore, the fluid pockets change in volume. The volume of the fluid pockets increases or decreases depending on the direction of the orbiting motion. Therefore, the scroll type fluid displacement apparatus is applicable to compress, expand or pump fluids. The discussion which follows deals only with scroll type devices used as compressors for the sake of convenience.

In comparison with a conventional compressor of the piston-type, the scroll type compressor has certain advantages, such as fewer parts and continuous compression of fluid. However, there have been several problems, primarily in the sealing of the fluid pockets. Sealing of the fluid pockets must be sufficiently maintained at axial and radial interfaces in the scroll type fluid apparatus, because the fluid pockets are defined by the line contacts between the interfitting spiral elements and axial contact between the axial end surfaces of the spiral elements and the inner end surfaces of the plates.

The principles of operation of a typical scroll type compressor will be described with reference to Figs. 1a—1d, Fig. 2 and Fig. 3. Figs. 1a—1d schematically illustrate the relative movement of interfitting spiral elements to compress the fluid. Fig. 2 diagrammatically illustrates the compression cycle in each of the fluid pockets. Fig. 3 schematically illustrates the typical interfitting relationship of prior art spiral elements. Figs. 1a—1d may be considered to be an end view of a compressor wherein the end plates are removed and only the spiral elements are shown.

Two spiral elements 1 and 2 are angularly offset and interfit with one another. As shown in Fig. 1a, the orbiting spiral element 1 and fixed spiral element 2 make four line contacts as shown at four points A—D. A pair of fluid pockets 3a and 3b are defined between line contacts D—C and line contacts A—B, as shown by the dotted regions. The fluid pockets 3a and 3b are defined not only by the wall of spiral elements 1 and 2 but also by the end plates from which these spiral elements extend. When orbiting spiral element 1 is moved in relation to fixed spiral element 2 so that the center O' of orbiting spiral element 1 revolves around the center O of fixed spiral element 2 with

a radius of $O—O'$, while the rotation of orbiting spiral element 1 is prevented, the pair of fluid pockets 3a and 3b shift angularly towards the center of the interfitted spiral elements with the volume of each fluid pocket 3a and 3b being gradually reduced, as shown in Figs. 1a—1d. Therefore, the fluid in each pocket is compressed.

Now, the pair of fluid pockets 3a and 3b are connected to one another while passing the stage from Fig. 1d to Fig. 1a and as shown in Fig. 1a, both pockets 3a and 3b merge at the center portion 5 and are completely connected to one another to form a single pocket. The volume of the connected single pocket is further reduced by each further revolution of 90° as shown in Figs. 1b, 1c and 1d. During the course of rotation, outer spaces which open in the state shown in Fig. 1b change as shown in Figs. 1c, 1d and 1a, to form new sealed off pockets in which fluid is newly enclosed.

Accordingly, if circular end plates are disposed on, and sealed to, the axial facing ends of spiral elements 1 and 2, respectively, and if one of the end plates is provided with a discharge port 4 at the center thereof as shown in Fig. 1a, fluid is taken into the fluid pockets at the radial outer portion and is discharged from the discharge port 4 after compression.

Referring to Fig. 2 and Fig. 1, the compression cycle of fluid in one fluid pocket will be described. Fig. 2 shows the relationship of fluid pressure in a fluid pocket to crank angle, and shows that the one compression cycle is completed at a crank angle of 4π in this case.

The compression cycle begins (Fig. 1a) with the outer end of each spiral element in contact with the opposite spiral element, the suction stroke having finished. The state of fluid pressure in the fluid pocket is shown at point K in Fig. 2. The volume of the fluid pocket is reduced and compressed by the revolution of the orbiting scroll member until the crank angle reaches 2π , which state is shown by the point L in Fig. 2. Immediately after passing this state, and hence, passing point L, the pair of fluid pockets are connected to one another and simultaneously are connected to the space filled with high pressure, which is connected to the discharge chamber and is formed at the center of both spiral elements. At this time, if the compressor is not provided with a discharge valve, the fluid pressure in the connected fluid pockets suddenly rises to equal the pressure in the discharge chamber. If, however, the compressor is provided with a discharge valve, the fluid pressure in the connected fluid pockets rises slightly due to the mixing of the high pressure fluid and the fluid in the connecting fluid pockets. This state is shown at point M in Fig. 2. The fluid in the high pressure space is further compressed by revolution of the orbiting scroll member until it reaches the discharge pressure. This state is shown at point N in Fig. 2. When the fluid pressure in the high pressure space reaches the discharge pressure, the fluid is discharged to the discharge chamber through the discharge

hole by the operation of the discharge valve. Therefore, fluid pressure in the high pressure space is maintained at the discharge pressure until a crank angle of 4π (point O).

Accordingly, one cycle of compression is completed at a crank angle of 4π , but the next cycle begins at the mid-point of compression of the first cycle as shown by point K', L', and M', and the dot-dash line in Fig. 2. Therefore, fluid compression proceeds continuously by the operation of these cycles.

Line contact between spiral elements is defined by several pairs of points, as shown in Fig. 3. However, it is very difficult to attain complete contact at all points. If the line contact between spiral elements is imperfect at one or more points to form a gap, fluid leakage through the gap will occur during operation to allow the outer pockets to contain gas with higher pressure than the ideal case. The volumetric efficiency of the compressor and, hence, its refrigeration capacity will thereby be reduced. Especially, fluid leakage across the line contact separating pair of fluid pockets from the high pressure space is very serious problem. If such leakage occurs, the pressure in the fluid pocket rises, as shown by the dotted lines and letters l, m, n in Fig. 2; therefore, the torque or the power required in the compressing operation is increased as a result, the energy efficiency ratio (refrigeration capacity performed by a unit horse power) is greatly reduced. Thus, sealing of the high pressure space must be tightly secured.

The curve of the spiral elements is usually an involute curve of a circle, each spiral having the same pitch (the pitch shown as distance a_1-a_2 , a_2-a_n , or b_1-b_2 , b_2-b_n in Fig. 3), and these two spiral elements interfit at an angular and radial offset, so that the spiral elements make a plurality of line contacts which are represented by points a_1-a_n and b_1-b_n in Fig. 3. Therefore, if the pitch of the spiral element is slightly different or if the inner and outer wall curve deviates from a true involute curve due to manufacturing inaccuracies, the line contacts will be imperfect, and the apparatus which uses these spiral elements will suffer fluid leakage. In order to avoid this problem, high accuracy is required in manufacturing of the spiral elements, resulting in high cost.

Even when two perfect spiral elements (having no dimensional errors) are interfitted and used in a compressor, heat developed during operation creates a thermal expansion of the elements. If the temperature is uniform throughout the spiral elements, the line contacts between both spiral elements change uniformly, and sealing of the fluid pockets is maintained. However, under actual operating conditions, thermal expansion of the spiral elements is non-uniform due to the temperature gradient, material non-uniformity or other imperfection resulting in a non-uniform pitch variation or deviation of wall curves from a true involute. This causes a gap at the line contacts between the spiral elements, resulting in fluid leakage from the high pressure space.

A scroll type fluid displacement apparatus

according to the preamble of the main claim is known from the GB-A-1,165,435. This apparatus has a thickened portion at the inner end of both spiral wraps and a thinned portion extending around 180° at the entry side at one side of both spiral elements. The thickened portions are each formed by the innermost half circle of the spiral wraps, thereby increasing their strength in this region. The thinner portion extends over a region of each spiral elements which provides a suction passage for at least one fluid pocket thereby to provide a common suction channel for the outermost fluid pockets. The thinnest portion of one of the wraps does not contribute to the compression operation. The line contacts of both spiral wraps defining the fluid pockets are shown as being formed perfectly. Therefore, this apparatus will also suffer fluid leakage in case of manufacturing errors and thermal expansion.

It is the object of the present invention to provide a scroll type fluid displacement apparatus with a good compression efficiency without the need to produce spiral elements without dimensional error.

This object is achieved by a scroll type fluid displacement apparatus according to the preamble of the main claim, which is characterized in that those portions of said wrap means extending substantially from the inner ends of said wrap means outwardly at least throughout the portions thereof which come into line contact with one another to define a single fluid pocket into which two innermost fluid pockets are merged, are slightly thicker than the remaining portions thereof.

Hence, according to the invention these two line contacts which define a central single fluid pocket are perfectly ensured for obtaining an efficient compression and thus the sealing of the high pressure space which is formed at the center of the wrap means is maintained without being affected by dimensional errors of the wrap means or by thermal expansion with help of the compliant drive mechanism.

Further objects, features and aspects of this invention will be understood from the following detailed description of the preferred embodiments of this invention referring to the annexed drawings. In the drawings:

Figs. 1a-1d are schematic views illustrating the relative movement of interfitting spiral elements to compress the fluid;

Fig. 2 is a pressure-crank angle diagram illustrating the compression cycle in each of the fluid pockets;

Fig. 3 is a schematic view illustrating the interfitting relationship of prior art spiral elements;

Fig. 4 is vertical sectional view of a compressor of the scroll type according to the invention;

Fig. 5 is an exploded perspective view of the driving mechanism used in the compressor of Fig. 4;

Fig. 6 is an explanatory diagram of the motion of the eccentric bushing illustrated in Fig. 4;

Fig. 7 is an exploded perspective view of the

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rotation preventing/thrust bearing mechanism used in the compressor of Fig. 4;

Fig. 8 is a schematic view illustrating the interfitting relationship of spiral elements according to one embodiment of this invention;

Fig. 9 is a schematic view similar to Fig. 8 illustrating a modified interfitting relationship of spiral elements according to this invention;

Fig. 10 is a schematic view illustrating the configuration of the transition portion of a spiral element; and

Figs. 11a—11c are schematic views illustrating the configuration of spiral elements according to a third embodiment of this invention.

Referring to Fig. 4, a refrigerant compressor unit according to the invention is shown which includes a compressor housing 10 comprising a front end plate 11 and a cup-shaped casing 12 disposed on the end surface of front end plate 11.

A fixed scroll member 13, an orbiting scroll member 14, a driving mechanism and a rotation preventing/thrust bearing mechanism of orbiting scroll member 14 are disposed within an inner suction chamber of cup-shaped casing 12. These mechanisms are described in detail below. The inner chamber is defined by the side wall of cup-shaped casing 12, the inner end surface of front end plate 11, and fixed scroll member 13.

Fixed scroll member 13 includes a circular end plate 131 and an involute wrap or spiral element 132 affixed to and extending from one major end surface of end plate 131. End plate 131 of fixed scroll member 13 is formed with a plurality of internally threaded bosses 133 axially projecting from a major end surface of plate 131 opposite the side thereof from which spiral element 132 extends. The end of each boss 133 abuts the inner surface 121 of cup-shaped casing 12, and is fixed to casing 12 by screws 15 which screw into bosses 133 from the outside of casing 12. Hence, fixed scroll member 13 is fixedly disposed within cup shaped casing 12. End plate 131 of fixed scroll member 13 partitions the interior of cup-shaped casing 12 into two chambers, a discharge chamber 16 and a suction chamber 17, and a sealing member 135 is disposed between the outer periphery of end plate 131 and the inner wall of cup-shaped casing 12 to isolate these two chambers.

Orbiting scroll member 14 is disposed in suction chamber 17 and also comprises a circular end plate 141 and an involute wrap or spiral element 142 affixed to and extending from one end surface of end plate 141. Spiral element 142 and spiral element 132 of fixed scroll member 13 are interfitted at an angular offset of 180° and a predetermined radial offset. A pair of fluid pockets are thereby defined between spiral elements 132, 142. Orbiting scroll member 14 is connected to the driving mechanism and the rotation preventing/thrust bearing mechanism. These mechanisms effect the orbital motion of orbiting scroll member 14 at a circular radius R_0 by the rotation of a drive shaft 18, to thereby compress

the fluid in the fluid pockets, as described in connection with Figs. 1a—1d.

Thus, when orbiting scroll member 14 is allowed to undergo the orbital motion with the radius R_0 by rotation of drive shaft 18, fluid or refrigerant gas, introduced into suction chamber 17 from an external fluid circuit through an inlet port 19 on casing 12, is taken into the fluid pockets formed between spiral elements 132, 142. As orbiting scroll member 14 orbits, fluid in the fluid pockets is moved to the center of the spiral elements with a consequent reduction of volume thereof. Compressed fluid is discharged into discharge chamber 16 from the fluid pocket at the center of the spiral elements through a hole 134 which is formed through circular plate 131 at a position near the center of spiral element 132, and a reed-type valve 136, and therefrom is discharged through an outlet port 20 to an external fluid circuit.

Referring to Figs. 4 and 5, the driving mechanism of orbiting scroll member 14 will be described. Drive shaft 18 is rotatably supported by a sleeve portion 111 of front end plate 11 through a bearing 21 and is formed with a disk portion 181 at its inner end portion. Disk portion 181 is also rotatably supported by front end plate 11 through a bearing 22 which is disposed within an opening of front end plate 11.

A crank pin or drive pin 182 projects axially from an end surface of disk portion 181 and, hence, from an end of drive shaft 18, and is radially offset from the center of drive shaft 18. End plate 141 of orbiting scroll member 14 is provided with a tubular boss 143 axially projecting from the end surface opposite to the surface thereof from which spiral element 142 extends. A discoid or short axial bushing 23 is fitted into boss 143, and is rotatably supported therein by bearing, such as a needle bearing 24. Bushing 23 has a balance weight 231 which is shaped as a portion of a disc or ring and extends radially from bushing 23 along a front surface thereof. An eccentric hole 232 is formed in bushing 23 radially offset from the center of bushing 23. Drive pin 182 is fitted into the eccentrically disposed hole 232 within which bearing 25 may be applied. Bushing 23 is therefore driven by the revolution of drive pin 182 and permitted to rotate by the needle bearing 24.

Respective location of center O_s of drive shaft 18, center O_b of bushing 23, and center O_d of hole 232 and thus drive pin 182 is shown in Fig. 6. In the position shown in Fig. 6, the distance between O_s and O_b is the representative radius R_0 of orbital motion of the orbiting scroll member 14; and when drive pin 182 is placed in eccentric hole 232, center O_d of drive pin 182 is placed, with respect to O_s , on the opposite side of a line L_1 , which is through O_b and perpendicular to a line L_2 through O_s and O_b , and also beyond the line through O_b and O_s in direction of rotation A of drive shaft 18.

In this construction of the driving mechanism, center O_b of bushing 23 is permitted to swing about the center O_d of drive pin 182 at a radius E_2 .

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As shown in Fig. 6, such swing motion of center O_c is illustrated as arc $O_c'-O_c''$ in Fig. 6. This permitted swing motion allows the orbiting scroll member 14 to compensate its motion for changes in radius R_0 due to wear on the spiral elements or due to dimensional inaccuracies of the spiral elements. When drive shaft 18 rotates, a drive force F_d is applied to the left at center O_d of drive pin 182 and a reaction force F_r of gas compression appears to the right at center O_c of bushing 33, both forces being parallel to line L_1 . Therefore, the arm O_c-O_d can swing outwardly by creation of the movement generated by the two forces. Spiral element 142 of orbiting scroll member 14 is thereby forced toward spiral element 132 of fixed scroll member 13 to make at least one pair of contact among several pair of sealing points which will be explained later, and the center of orbiting scroll member 14 orbits with the representative radius R_0 around center O_d of drive shaft 18. The rotation of orbiting scroll member 14 is prevented by the rotation preventing/thrust bearing mechanism 26 (Fig. 7), whereby orbiting scroll member 14 orbits while maintaining its angular orientation relative to fixed scroll member 13.

Referring to Figs. 7 and 4, a rotation preventing/thrust bearing mechanism 26 surrounds boss 143 and comprises a fixed ring 261 and Oldham ring 262. Fixed ring 261 is secured to an inner surface of housing 10. Fixed ring 261 is provided with a pair of keyways 261a, 261b in an axial end surface facing orbiting scroll member 14. Oldham ring 262 is disposed in a hollow portion between fixed ring 261 and end plate 141 of orbiting scroll member 14. Oldham ring 262 is provided with a pair of keys 262a, 262b on the surface facing fixed ring 261, which are received in keyways 261a, 261b. Therefore, Oldham ring 262 is linearly slidable relative to fixed ring 261 by the guide of keys 262a, 262b within keyways 261a, 261b. Oldham ring 262 is also provided with a pair of keys 262c, 262d on its opposite surface. Keys 262c, 262d are arranged along a diameter perpendicular to the diameter along which keys 262a, 262b are arranged. Circular end plate 141 of orbiting scroll member 14 is provided with a pair of keyways (in Fig. 7 only one keyway 141a is shown; the other keyway is disposed diametrically opposite keyway 141a) on the surface facing Oldham ring 262 in which are received keys 262c, 262d. Therefore, orbiting scroll member 14 is linearly slidable relative to Oldham ring 262 by the guide of keys 262c, 262d within the keyways of end plate 141.

Accordingly, orbiting scroll member 14 is slidable in one radial direction with Oldham ring 262, and is independently slidable in another radial direction perpendicular to the first radial direction. Therefore, rotation of orbiting scroll member 14 is prevented, while its movement in two radial directions perpendicular to one another is permitted. Now, Oldham ring 262 is provided with a plurality of holes or pockets 27, and a bearing element, such as balls 28, having a diameter

which is greater than the thickness of Oldham ring 262, is retained in each pocket 27. Balls 28 contact and roll on the surface of fixed ring 261 and circular end plate 141 of orbiting scroll member 14. Therefore, the thrust load from orbiting scroll member 14 is supported on fixed ring 261 through balls 28.

As explained below, the radius R_0 of orbital motion is determined by one contact point between the spiral elements having the minimum of the angle $\angle O_c O_d O_c$. Bushing 23 is supported to permit swing motion about drive pin 182, and this swing motion allows the orbiting scroll member 14 to compensate its motion for variation of radius R_0 . On the other hand, spiral element 142 of orbiting scroll member 14 is forced toward spiral element 132 of fixed scroll member 13 by the driving moment. The radius R_0 is determined by the combination of the errors of the spiral elements, for example by, either a combination of the maximum inward deviation of the inner wall of the fixed spiral element 132 and the maximum outward deviation of the outer wall of the orbiting spiral element 142 or a combination of the maximum outward deviation of the outer wall of the fixed spiral element 132 and the maximum inward deviation of the inner wall of the orbiting spiral element 142, from the theoretical involute curve for each wall. There are various manners in which the actual orbiting radius varies as the crank angle proceeds, but the first portion to contact the opposite wall of the other spiral element, determines the radius R_0 as a function of the crank angle, in other words the orbiting scroll member 14 orbits with radius R_0 which is determined by the first contact point between spiral elements 132, 142, and the actual contact point to determine the radius can be near outer end of the spiral to form gaps between the two spiral walls in the area of the high pressure space.

Fig. 8 shows the configuration of spiral elements according to one embodiment of the present invention. As shown in Fig. 8, the wall of the center portion of each spiral element is made slightly thicker (by α in Fig. 8) by making a slight step along the inner wall thereof. The thicker portion of each spiral element extends from the inner end portion or tip of the spiral element (shown at point A in Fig. 8) to a location along the spiral which is spaced from the tip by an involute angle of at least 2π (shown at point B in Fig. 8). The outer portion of each spiral element extends from point B to the outermost end of the spiral element (shown at point D in Fig. 8) with a reduced thickness. When the two spiral elements are interfitted at an angular and radial offset for the involute portions from A to B of both spiral elements to make line contact, a small gap may arise at the point where the line contacts should be between the spiral elements in the involute range from B to D. However, the more important seal of the high pressure space which is defined in the center of both spiral elements is ensured by the thicker portion from A to B (hereinafter designation A—B will be used) of the inner wall of the

spiral elements. The same effect may alternatively be achieved by a step on the outer (rather than the inner) wall thereof at corresponding points to B for each spiral.

In accordance with the above construction of spiral elements, when the thickness of portion B—D as a dimensional error (ΔE) of less than the step (a) between portion A—B and portion B—D, such error will not disturb the sealing of the high pressure space. The fluid leakage across the gap at the line contacts between the outer portion (B—D) of the spirals is considered to be minimal because the pressure difference between outer fluid pockets is small. Deterioration of resultant volumetric efficiency of the compressor is thereby permissible.

Fig. 9 shows a modification of the embodiment shown in Fig. 8, wherein the center portion of each spiral element is made thicker by a slight step (a) on the inner *and* outer walls thereof. These thicker portion extend from the inner end portion or tip of each spiral element 132, 142 (shown at point A in Fig. 9) at least throughout the portion of the spiral elements which contact one another when the pair of fluid pockets are connected to the high pressure space (shown at points B and C in Fig. 9). The slightly thinner outer portion extends from the points B or C to the terminal ends of both spiral elements 132, 142 (shown at point D and E in Fig. 9). Therefore, when these two spiral elements are interfitted with one another, a gap (shown as 2a in Fig. 9) between the portion B—D and the portion C—E of both spiral elements results. However, the important seal of the high pressure space which is defined at the center of the spiral elements is insured.

The transition between the thicker portion and the thinner portion of each spiral is shown in Figs. 8 and 9 to be steplike. However, the transition can be arcuate, rather than stepped, as shown in Fig. 10. The radius of curvature of the arcuate transition portion is determined by the radius of the milling tool M used to form the spiral element. The arcuate transition portion is formed when the milling tool reaches the end of its travel after forming an adjacent portion of spiral.

Fig. 11a shows another embodiment of the present invention, which is characterized in that the inner wall of the outer portion of the spiral element starts deviating from a true involute curve at point B to form a portion of gradually reduced thickness. The wall thickness of the inner portion, which is between the inner end portion or tip of each spiral element (point A) and point B, is uniform. Since the wall thickness between point B and the outer terminal end (point D) gradually reduces, the gap (a) between the spiral elements will be a function of the involute angle.

Figs. 11b and 11c show modifications of the embodiment shown in Fig. 11a, wherein the center portion of each spiral element is formed to a true involute curve and the outer wall of the outer portion of the spiral element starts deviating from a true involute curve at point C to form a

portion of gradually reduced thickness (shown in Fig. 11b) or the inner *and* outer wall of the outer portion of the spiral elements start deviating from a true involute curve at points B and C to form a portion of gradually reduced thickness (shown in Fig. 11c).

Claims

10 1. A scroll type fluid displacement apparatus including a housing (10), a pair of scroll members (13, 14), one of said scroll members (13) being fixedly disposed relative to said housing (10) and having an end plate (131) from which a first wrap means (132) extends into the interior of said housing and the other scroll member (14) being movably disposed for non-rotative orbital movement within the interior of said housing (10) and having an end plate (141) from which a second spiral wrap means (142) extends, said first and second wrap means (132, 142) substantially inter-fitting at an angular and radial offset to define at least one pair of substantially sealed off fluid pockets (3), and a driving mechanism (18, 181, 182) operatively connected to said other scroll member (14) to effect the orbital motion of said other scroll member (14), whereby said fluid pockets (3) move inwardly and change in volume, the two innermost fluid pockets eventually merging into a single pocket (5) near the center of said wrap means (132, 142), a center part of each of said wrap means (132, 142) being thicker than the remaining portions thereof, and both the center parts and the remaining portions being arranged to define said fluid pockets, characterized in that those portions of said wrap means (132, 142) extending substantially from the inner ends of said wrap means outwardly at least throughout the portions thereof which come into line contact with one another to define a single fluid pocket into which two innermost fluid pockets are merged, are slightly thicker than the remaining portion thereof.

45 2. A scroll type fluid displacement apparatus as claimed in Claim 1, characterized in that said slightly thicker portions of said wrap means (132, 142) are formed with a step on the inner wall of said center parts.

50 3. A scroll type fluid displacement apparatus as claimed in Claim 1, characterized in that said slightly thicker portions of said wrap means (132, 142) are formed with a step on the outer wall of said center parts.

55 4. A scroll type fluid displacement apparatus as claimed in Claim 1, characterized in that said slightly thicker portions of said wrap means (132, 142) are formed with a step on both the inner and outer walls of said center parts.

60 5. A scroll type fluid displacement apparatus as claimed in Claim 1, 2, 3 or 4, characterized in that a transition portion is formed on each of said wrap means (132, 142) between said slightly thicker portion and the thinner remaining portion thereof.

65 6. A scroll type fluid displacement apparatus as

spiral elements. The same effect may alternatively be achieved by a step on the outer (rather than the inner) wall thereof at corresponding points to B for each spiral.

In accordance with the above construction of spiral elements, when the thickness of portion B—D as a dimensional error (ΔE) of less than the step (α) between portion A—B and portion B—D, such error will not disturb the sealing of the high pressure space. The fluid leakage across the gap at the line contacts between the outer portion (B—D) of the spirals is considered to be minimal because the pressure difference between outer fluid pockets is small. Deterioration of resultant volumetric efficiency of the compressor is thereby permissible.

Fig. 9 shows a modification of the embodiment shown in Fig. 8, wherein the center portion of each spiral element is made thicker by a slight step (α) on the inner and outer walls thereof. These thicker portion extend from the inner end portion or tip of each spiral element 132, 142 (shown at point A in Fig. 9) at least throughout the portion of the spiral elements which contact one another when the pair of fluid pockets are connected to the high pressure space (shown at points B and C in Fig. 9). The slightly thinner outer portion extends from the points B or C to the terminal ends of both spiral elements 132, 142 (shown at point D and E in Fig. 9). Therefore, when these two spiral elements are interfitted with one another, a gap (shown as 2α in Fig. 9) between the portion B—D and the portion C—E of both spiral elements results. However, the important seal of the high pressure space which is defined at the center of the spiral elements is insured.

The transition between the thicker portion and the thinner portion of each spiral is shown in Figs. 8 and 9 to be steplike. However, the transition can be arcuate, rather than stepped, as shown in Fig. 10. The radius of curvature of the arcuate transition portion is determined by the radius of the milling tool M used to form the spiral element. The arcuate transition portion is formed when the milling tool reaches the end of its travel after forming an adjacent portion of spiral.

Fig. 11a shows another embodiment of the present invention, which is characterized in that the inner wall of the outer portion of the spiral element starts deviating from a true involute curve at point B to form a portion of gradually reduced thickness. The wall thickness of the inner portion, which is between the inner end portion or tip of each spiral element (point A) and point B, is uniform. Since the wall thickness between point B and the outer terminal end (point D) gradually reduces, the gap (α) between the spiral elements will be a function of the involute angle.

Figs. 11b and 11c show modifications of the embodiment shown in Fig. 11a, wherein the center portion of each spiral element is formed to a true involute curve and the outer wall of the outer portion of the spiral element starts deviating from a true involute curve at point C to form a

portion of gradually reduced thickness (shown in Fig. 11b) or the inner and outer wall of the outer portion of the spiral elements start deviating from a true involute curve at points B and C to form a portion of gradually reduced thickness (shown in Fig. 11c).

Claims

1. A scroll type fluid displacement apparatus including a housing (10), a pair of scroll members (13, 14), one of said scroll members (13) being fixedly disposed relative to said housing (10) and having an end plate (131) from which a first wrap means (132) extends into the interior of said housing and the other scroll member (14) being movably disposed for non-rotative orbital movement within the interior of said housing (10) and having an end plate (141) from which a second spiral wrap means (142) extends, said first and second wrap means (132, 142) substantially inter-fitting at an angular and radial offset to define at least one pair of substantially sealed off fluid pockets (3), and a driving mechanism (18, 181, 182) operatively connected to said other scroll member (14) to effect the orbital motion of said other scroll member (14), whereby said fluid pockets (3) move inwardly and change in volume, the two innermost fluid pockets eventually merging into a single pocket (5) near the center of said wrap means (132, 142), a center part of each of said wrap means (132, 142) being thicker than the remaining portions thereof, and both the center parts and the remaining portions being arranged to define said fluid pockets, characterized in that those portions of said wrap means (132, 142) extending substantially from the inner ends of said wrap means outwardly at least throughout the portions thereof which come into line contact with one another to define a single fluid pocket into which two innermost fluid pockets are merged, are slightly thicker than the remaining portion thereof.

2. A scroll type fluid displacement apparatus as claimed in Claim 1, characterized in that said slightly thicker portions of said wrap means (132, 142) are formed with a step on the inner wall of said center parts.

3. A scroll type fluid displacement apparatus as claimed in Claim 1, characterized in that said slightly thicker portions of said wrap means (132, 142) are formed with a step on the outer wall of said center parts.

4. A scroll type fluid displacement apparatus as claimed in Claim 1, characterized in that said slightly thicker portions of said wrap means (132, 142) are formed with a step on both the inner and outer walls of said center parts.

5. A scroll type fluid displacement apparatus as claimed in Claim 1, 2, 3 or 4, characterized in that a transition portion is formed on each of said wrap means (132, 142) between said slightly thicker portion and the thinner remaining portion thereof.

6. A scroll type fluid displacement apparatus as



claimed in Claim 5, characterized in that said transition portion is stepped.

7. A scroll type fluid displacement apparatus as claimed in Claim 5, characterized in that said transition portion is arcuate.

8. A scroll type fluid displacement apparatus as claimed in Claim 1 characterized in that the thickness of the thinner outer portion of each of said wrap means (132, 142) gradually diminishes towards the outer terminal end thereof.

9. A scroll type fluid displacement apparatus according to one of Claims 1 to 8, characterized in that said slightly thicker portions have a uniform thickness.

Patentansprüche

1. Spiralf fluidverdrängungsvorrichtung mit einem Gehäuse (10), einem Paar von Spiralelementen (13, 14), von denen eines relativ zum Gehäuse (10) feststehend angeordnet ist und eine Endplatte (131) aufweist, von der sich ein erstes Spiralteil (132) in das Innere des Gehäuses erstreckt, und das andere Spiralelement (14) im Inneren des Gehäuses (10) für eine nicht rotierende Umlaufbewegung beweglich angeordnet ist und eine Endplatte (141) aufweist, von der sich ein zweites Spiralteil (142) erstreckt, wobei das erste und zweite Spiralteil (132, 142) im wesentlichen mit einer winkelmäßigen und radialen Versetzung ineinandergreifen, um zumindest ein Paar von im wesentlichen abgedichteten Fluidtaschen (3) zu umgrenzen, und einer Antriebsvorrichtung (18, 181, 182), die mit dem anderen Spiralelement (14) zum Bewirken der Umlaufbewegung des anderen Spiralelementes (14) wirkungsmäßig verbunden ist, wodurch die Fluidtaschen (3) nach innen wandern und ihr Volumen verändern, wobei die beiden innersten Fluidtaschen sich gegebenenfalls zu einer einzigen Tasche (5) nahe der Mitte der Spiralteile (132, 142) vereinen und ein Mittelbereich jedes der Spiralteile (132, 142) dicker ist als dessen übrige Bereiche und sowohl die Mittelbereiche als auch die übrigen Bereiche so angeordnet sind, daß sie die Fluidtaschen umgrenzen, dadurch gekennzeichnet, daß diejenigen Bereiche der Spiralteile (132, 142), die sich im wesentlichen von den Innenenden der Spiralteile nach außen über zumindest diejenigen Bereiche der Spiralteile erstrecken, die miteinander in Linienberührung zum Umgrenzen einer einzigen Fluidtasche, zu der sich die beiden innersten Fluidtaschen vereinen, gelangen, geringfügig dicker als deren übrige Bereiche ausgebildet sind.

2. Spiralf fluidverdrängungsvorrichtung nach Anspruch 1, dadurch gekennzeichnet, daß die geringfügig dickeren Bereiche der Spiralteile (132, 142) mit einer Stufe an der Innenwand der Mittelbereiche gebildet sind.

3. Spiralf fluidverdrängungsvorrichtung nach Anspruch 1, dadurch gekennzeichnet, daß die geringfügig dickeren Bereiche jedes Spiralteiles (132, 142) mit einer Stufe auf der Außenwand der Mittelbereiche gebildet sind.

4. Spiralf fluidverdrängungsvorrichtung nach Anspruch 1, dadurch gekennzeichnet, daß die geringfügig dickeren Bereiche der Spiralteile (132, 142) mit einer Stufe an sowohl der Innenwand als auch der Außenwand der Mittelbereiche gebildet sind.

5. Spiralf fluidverdrängungsvorrichtung nach Anspruch 1, 2, 3 oder 4, dadurch gekennzeichnet, daß auf jedem der Spiralteile (132, 142) zwischen dem geringfügig dickeren Bereich und dem dünneren übrigen Bereich ein Übergangsbereich gebildet ist.

6. Spiralf fluidverdrängungsvorrichtung nach Anspruch 5, dadurch gekennzeichnet, daß der Übergangsbereich gestuft ist.

7. Spiralf fluidverdrängungsvorrichtung nach Anspruch 5, dadurch gekennzeichnet, daß der Übergangsbereich bogenförmig ist.

8. Spiralf fluidverdrängungsvorrichtung nach Anspruch 1, dadurch gekennzeichnet, daß die Dicke des dünneren äußeren Bereiches jedes der Spiralteile (132, 142) zu dessen Außenende hin allmählich abnimmt.

9. Spiralf fluidverdrängungsvorrichtung nach einem der Ansprüche 1 bis 8, dadurch gekennzeichnet, daß die geringfügig dickeren Bereiche eine gleichmäßige Dicke besitzen.

Revendications

1. Appareil de déplacement de fluides, du type à volute, comprenant un corps (10), une paire d'éléments de volute (13, 14), l'un desdits éléments de volute (13) étant disposé d'une manière fixe par rapport audit corps (10) et possédant une première plaque en bout (131) à partir de laquelle un premier moyen d'enroulement (132) s'étend dans l'espace intérieur dudit corps, et l'autre élément de volute (14) étant disposé d'une manière mobile, de façon à subir un mouvement orbital non-rotatif, à l'intérieur dudit corps (10) et possédant une plaque en bout (141) à partir de laquelle s'étend un deuxième moyen d'enroulement spiral (142), lesdits premier et deuxième moyens d'enroulement (132, 142) s'adaptant essentiellement les uns dans les autres selon un décalage angulaire et radial pour définir au moins une paire de poches de fluide (3), essentiellement scellées, et un mécanisme d'entraînement (18, 181, 182), raccordé d'une manière opérationnelle audit autre élément de volute (14) pour provoquer le mouvement orbital dudit autre élément de volute (14), ce par quoi lesdites poches de fluide (3) se déplacent vers l'intérieur et changent de volume, les deux poches de fluides les plus à l'intérieur fusionnant en fin de compte en une poche unique (5) proche du centre desdits moyens d'enroulement (132, 142), une partie centrale de chacun desdits moyens d'enroulement (132, 142) étant plus épaisse que leurs parties restantes, et tant les parties centrales que les parties restantes étant disposées de façon à définir lesdites poches de fluide, caractérisé en ce que les parties desdits moyens d'enroulement (132, 142) s'étendant essentiellement à partir des extrémités inté-



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rieures desdits moyens d'enroulement, vers l'extérieur au moins par leurs parties qui entrent en contact linéaire les unes avec les autres pour définir une poche de fluide unique dans laquelle fusionnent deux poches de fluide les plus à l'intérieur, sont légèrement plus épaisses que leurs parties restantes.

2. Appareil de déplacement de fluides du type à volute selon la revendication 1, caractérisé en ce que lesdites parties légèrement plus épaisses desdits moyens d'enroulement (132, 142) sont formées avec un décrochement sur la paroi intérieure desdites parties centrales.

3. Appareil de déplacement de fluides, du type à volute, selon la revendication 1, caractérisé en ce que lesdites parties légèrement plus épaisses desdits moyens d'enroulement (132, 142) sont formées avec un décrochement sur la paroi extérieure desdites parties centrales.

4. Appareil de déplacement de fluides, du type à volute, selon la revendication 1, caractérisé en ce que lesdites parties plus épaisses desdits moyens d'enroulement (132, 142) sont formées avec un décrochement tant sur la paroi intérieure que sur

la partie extérieure desdites parties centrales.

5. Appareil de déplacement de fluides, du type à volute, selon les revendications 1, 2, 3 ou 4, caractérisé en ce qu'une partie de transition est formée sur chacun desdits moyens d'enroulement (132, 142) entre ladite partie légèrement plus épaisse et sa partie restante plus mince.

6. Appareil de déplacement de fluides, du type à volute, selon la revendication 5, caractérisé en ce que ladite partie de transition possède des décrochements.

7. Appareil de déplacement de fluides, du type à volute, selon la revendication 5, caractérisé en ce que ladite partie de transition est incurvée.

8. Appareil de déplacement de fluides, du type à volute, selon la revendication 1, caractérisé en ce que l'épaisseur de la partie extérieure plus mince de chacun desdits moyens d'enroulement (132, 142) diminue progressivement vers son extrémité terminale extérieure.

9. Appareil de déplacement de fluides, du type à volute, selon l'une des revendications 1 à 8, caractérisé en ce que lesdites parties légèrement plus épaisses ont une épaisseur uniforme.



* C L M S *