(19) Europäisches Patentamt
European Patent Office
Office européen des brevets

(11) Publication number: 0 065 261
A2

(12) EUROPEAN PATENT APPLICATION

(21) Application number: 82104075.5

(22) Date of filing: 11.05.82

(51) Int. Cl.³: F 04 C 27/00

(30) Priority: 11.05.81 JP 68235/81
11.07.81 JP 103282/81

(43) Date of publication of application: 24.11.82
Bulletin 82/47

(84) Designated Contracting States: DE FR GB IT SE

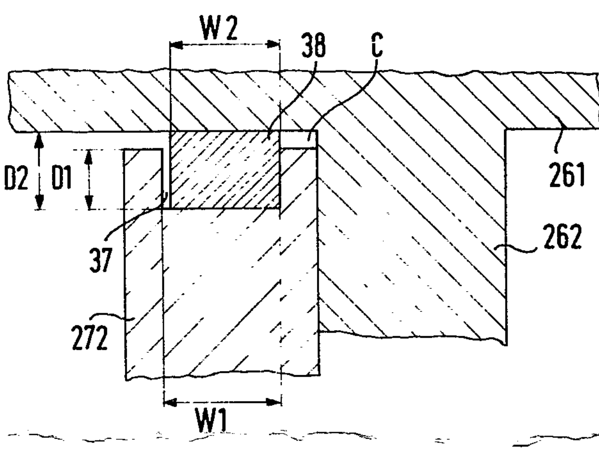
(71) Applicant: SANKYO ELECTRIC COMPANY LIMITED,
20 Kotobuki-cho, Isesaki-shi, Gunma-ken (JP)

(72) Inventor: Hiraga, Masaharu, 8-34, Honjo 4-chome,
Honjo-shi Saitama-ken (JP)
Inventor: Terauchi, Kiyoshi, 8-14, Heiwa-cho, Isesaki-shi
Gunma-ken (JP)

(74) Representative: Prüfer, Lutz H., Dipl.-Phys.,
Willroiderstrasse 8, D-8000 München 90 (DE)

(54) **Axial sealing mechanism for scroll type fluid displacement apparatus.**

(57) In a scroll type fluid displacement apparatus having an orbiting scroll member (27) and a fixed scroll member (26) which form at least one pair of outer fluid pockets and a central pocket therebetween for fluid compression, the axial end surface of each spiral element (262, 272) of the scroll member has a groove (37) along the spiral curve. A seal element (38) is fitted in the groove (37). The axial dimension of the seal element (38) is formed greater than the depth of groove (37). This seal element (37) is usually in contact with the facing end plate of the scroll member without any axial force urging means to secure the sealing of the fluid pockets. Also, the gap (C) between the seal element (38) and the groove (37) which defines the channel of the fluid leakage, is reduced.



EP 0 065 261 A2

AXIAL SEALING MECHANISM FOR SCROLL TYPE FLUID DISPLACEMENT APPARATUS

This invention relates to a fluid displacement apparatus, and more particularly, to an axial sealing mechanism for a scroll type fluid displacement apparatus.

Scroll type fluid displacement apparatus are well known in the prior art. For example, U.S.Patent No. 801,182 (Creux) discloses a device 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 their 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. Since the volume of the fluid pockets increases or decreases dependent on the direction of the orbiting motion, the scroll type fluid displacement apparatus is applicable to compress, expand or pump fluids.

In comparison with conventional compressors of the piston type, the scroll type compressor has certain advantages, such as reduced parts and continuous compression of fluid. However, one of the problems encountered in prior art scroll type compressors has been ineffective sealing of the fluid pockets. Axial and radial sealing of the fluid pockets must be maintained in a scroll type compressor in order to achieve efficient operation. The fluid pockets in a scroll type compressor are defined by both line contacts between the interfitting spiral elements and axial contacts between the axial end surfaces of the spiral elements and the inner end surface of the end plates.

One prior art solution to the radial sealing problem is described in co-pending European application No. 81301155.8, filed March 18, 1981. The co-pending application discloses an orbiting scroll member supported on a crank pin of which axis is placed on the radially offset or eccentric of axis of a drive shaft through a bushing. The orbiting scroll is rotatably supported on the crank pin, and therefore, during operation of the apparatus, the orbiting scroll is pushed against the fixed scroll due to the moment created by the differential of the driving point and reaction force acting point to thereby effect the radial sealing.

Furthermore, various techniques have been used in the prior art to resolve the sealing problem, particularly, the axial sealing problem. For example, U.S.Patent No. 3,874,827 (Young) discloses a non-rotatable fixed scroll member supported within the housing of scroll apparatus in an axially floating condition. A high pressure fluid is introduced behind the fixed scroll member to establish sufficient axial sealing. However, because the fixed scroll member in Young patent is supported in the axial floating condition, the fixed scroll member may wobble due to the eccentric orbital motion of the orbiting scroll member. Therefore, axial and radial sealing of the fluid pockets and resultant fluid compression, tend to be imperfectly performed.

In order to avoid these disadvantages, the pressure of the high pressure fluid must be increased and the clearance between radial supporting parts must be made as small as possible. However, minimizing the clearance is expensive due to the close tolerance requirements, while an increase in pressure increases contact pressure between both scroll members, which increases mechanical loss or causes damage to the scroll members.

Another technique for improving the axial seal of the fluid pockets is to be use sealing element mounted in the axial end surface of each spiral elements, as described in U.S.Patent No. 3,994,635. In this technique the end surface of each spiral element facing the end plate of the other scroll member is provided with a groove formed along the spiral. A seal element is placed within the groove and an axial force urging device, such as a spring, is placed behind the seal element to urge the seal element toward the facing end surface of the end plate to thereby effect axial sealing. In this technique, the construction of the axial force urging device is complex and it is difficult to obtain the desired uniform sealing force along the entire length of the seal element.

Figs. 1–5 illustrate a more simple construction of the axial sealing mechanism to avoid these disadvantages. Fig.1 shows a diagrammatic sectional view illustrating the spiral element of the fixed and orbiting scroll members and Fig. 2 shows a sectional view taken along a line II–II in Fig. 1. In the construction, a seal element 1 is loosely fitted into a groove 2 formed in the axial end surface of each spiral element 3, i.e., the width W_2 of seal element 1 is formed smaller than the width W_1 of groove 2 and also the depth D_2 of seal element 1 is formed smaller than the depth D_1 of groove 2, as shown in Fig. 2. As a substitute for a mechanical axial force urging device, the pressurized fluid then is introduced into groove 2 from adjacent pockets through the gap C, which is defined between the side surface of seal element 1 and the inner end surface of groove 2, and seal element 1 is urged toward the axial and radial direction to contact with the facing end plate 4 and the one side surface of groove 2. The urging force towards radial direction is shown by arrow in Fig. 1.

As shown in Fig. 1, the fluid within a high pressure chamber A, which is defined by the center portions of the spiral elements, leaks into the adjacent lower pressure chambers B_1 and B_2, and also fluid within lower pressure chambers B_1 and B_2 leaks into the suction chamber defined by the outer peripheral portion of the scroll members. Since groove 2 in which seal element 1 is disposed extends from the center of spiral element to near the terminal end thereof. Furthermore, as shown in Fig. 4 which is a sectional view taken along a line IV-IV in Fig. 1, the fluid within the high pressure chamber A or lower pressure chambers B_1 and B_2 leaks into the adjacent lower chambers B_1 and B_2 or the suction chamber through the contact surface between the axial end surface of seal element 1 and the inner end surface of facing end plate 4. Therefore, the volume efficiency of the compressor is reduced.

In order to avoid the fluid leakage from the high pressure chamber or lower pressure chambers to lower pressure chamber or suction chamber, the clearance between the axial end surface of spiral element and the inner end surface of facing end plate will be made as small as possible to reduce the gap C. However, during the operation of the compressor, the axial end surface of the spiral element comes into contact with the inner end surface of the facing end plate to thereby cause the abnormal wearing. Because, these disadvantages arise the following conditions. Intrinsically, both scroll members are expanded each other by the pressure within the fluid pockets and the orbiting scroll member is pushed against an axial thrust bearing device. However, the driving point of the orbiting scroll member, which point is placed on the side surface of the end plate opposite the spiral element extending, is offset from the acting point of the reaction force caused by the compression of gas, which point is placed on the intermediate along the height of spiral element of the orbiting scroll member. Therefore, the resultant moment causes the axial slant of the orbiting scroll member. Also, in a transit condition of such as a moment to turn on or off the magnetic clutch or sudden change of the operating condition, the orbiting scroll member is not uniformly pushed against the axial thrust bearing which is placed between the end plate of the orbiting scroll member and the fixed plate portion. Therefore, the axial slant of the orbiting scroll member is caused.

Furthermore, the seal element is disposed within the groove in an axially and radially floating condition, and pushed against toward facing end plate by pressurized fluid. However, as shown in Fig. 1 and Fig. 5 which is a sectional view taken along a line V-V in Fig. 1, a part of seal element 1 extends the no pressure differential area. The area is shown by a part of D in Fig. 1.. In this condition, the seal element 1 cannot receive the urging force toward the axial and radial direction, and seal element 1 can therefore freely move within the groove 2 and comes into contact with the inner end surface of facing end plate 4, as shown in Fig. 5, to thereby arise the abnormal wear.

It is a primary object of this invention to provide an improvement in a scroll type fluid displacement apparatus with high volumetric efficiency and thus with high energy efficiency.

It is another object of this invention to provide a scroll type fluid displacement apparatus wherein the abnormal wear of the axial seal element is prevented and the axial sealing of the fluid pockets is enhanced to attain a long life.

It is still another object of this invention to provide a scroll type fluid displacement apparatus wherein the wear of the scroll member is prevented and the endurance life of the apparatus is improved.

It is a further object of this invention to provide a scroll type fluid displacement apparatus which is simple in construction and simple in manufacturing to achieve the above objects.

A scroll type fluid displacement apparatus according to this invention includes a pair of scroll members each comprising a circular end plate and a spiral wrap extending from one side of the circular end plate. A groove is formed in the axial end surface of each spiral wrap and extends along the spiral curve of the wrap. A seal element is fitted in the groove. The axial dimension of the seal element is greater than the depth of the groove. The axial end surface of seal element is, therefore, usually contact with the inner end surface of the facing end plate without any axial force urging device, after assembling the compressor. Accordingly, the gap between the seal element and the groove which defines the channel of leakage fluid is reduced.

Further objects, features and other aspects of this invention will be understood from the detailed description of the preferred embodiment of this invention referring to the annex d drawings.

Fig. 1 ia a diagrammatic sectional view illustrating the spiral element of the fixed and orbiting scroll members;

Fig. 2 is a sectional view taken along a line II-II in Fig. 1;

Fig. 3 is a perspective view of an axial end portion of the spiral element illustrating the leakage channel of the high pressure fluid;

Fig. 4 is a sectional view taken along a line IV-IV in Fig. 1;

Fig. 5 is a sectional view taken along a line V-V in Fig. 1;

Fig. 6 is a vertical sectional view of the compressor unit type of fluid displacement apparatus according to one embodiment of this invention;

Fig. 7 is an exploded perspective view of the driving mechanism of the embodiment of Fig. 6;

Fig. 8 is a perspective view of the scroll member in the embodiment of Fig. 6;

Fig. 9 is a sectional view taken along a line IX - IX in Fig.8;and

Fig. 10 is a sectional view of an axial end portion of the spiral element according to another embodiment of this invention.

Referring to Fig. 6, a fluid displacement apparatus in accordance with the present invention is shown which is a scroll type refrigerant compressor. The compressor includes a compressor housing 10 having a front end plate 11 and a cup shaped casing 12 fastened to an end surface of front end plate 11. An opening 111 is formed in the center of front end plate 11 for supporting a drive shaft 13. An annular projection 112, concentric with opening 111, is formed on the rear end surface of front end plate 11 facing cup shaped casing 12. An outer peripheral surface of annular projection 112 fits into an inner wall of the opening of cup shaped casing 12. Cup shaped casing 12 is fixed on the rear end surface of front end plate 11 by a fastening device, such as bolts and nuts, so that the opening of cup shaped casing 12 is covered by front end plate 11. An O-ring 14 is placed between the outer peripheral surface of annular projection 112 and the inner wall of cup shaped casing 12. Front end plate 11 has an annular sleeve 15 projecting from the front end surface thereof; this sleeve 15 surrounds drive shaft 13 to define a shaft seal cavity. As shown in Fig. 6, sleeve 15 is attached to the front end surface of front end plate 11 by screws 16, one of which is shown in Fig. 6. An O-ring 17 is placed between the front end surface of front end plate 11 and an end surface of sleeve 15 to seal the mating surface of front end plate 11 and sleeve 15. Alternatively, sleeve 15 may be formed integral with front end plate 11.

Drive shaft 13 is rotatably supported by sleeve 15 through a bearing 18 disposed within the front end of sleeve 15. Drive shaft 13 has a disk shaped rotor 19 at its inner end; disk shaped rotor 19 is rotatably supported by front end plate 11 through a bearing 20 disposed within opening 111 of front end plate 11. A shaft seal assembly 21 is assembled on drive shaft 13 within the shaft seal cavity of sleeve 15.

A pulley 22 is rotatably supported by a bearing 23 on the outer surface of sleeve 15. An electromagnetic coil 24, which is received in an annular cavity of pulley 22, is mounted on the outer surface of sleeve 15 by a supported plate 241. An armature plate 25 is elastically supported on the outer end of drive shaft 13 which extends from sleeve 15. A magnetic clutch is formed by pulley 22, magnetic coil 24 and armature plate 25. Thus, drive shaft 13 is driven by an external power source,for example, an engine of vehicle, through a rotation transmitting device, such as the above described magnetic clutch.

A number of elements are located within the inner chamber of cup shaped casing 12 including a fixed scroll 26 ,an orbiting scroll 27, a driving mechanism for orbiting scroll 27 and a rotation preventing/thrust bearing device 28 for orbiting scroll 27. The inner chamber of cup shaped casing 12 is formed between the inner wall of cup shaped casing 12 and front end plate 11.

Fixed scroll 26 includes a circular end plate 261, a wrap or spiral element 262 affixed to or extending from one end surface of circular end plate 261, and a plurality of internal bosses 263 axially projecting from the end surface of circular end plate 261 on the side opposite spiral element 262 extending. The end surface of each boss 263 is seated on the inner surface of end plate portion 121 of cup shaped casing 12 and is fixed to end plate portion 121 by a plurality of bolts 29, one of which is shown in Fig. 6. Hence, fixed scroll 26 is fixedly disposed within cup shaped casing 12. Circular end plate 261 of fixed scroll 26 partitions the inner chamber of cup shaped casing 12 into a rear chamber 30 having bosses 263, and a front chamber 31, in which spiral element 262 of fixed scroll 26 is located. A sealing member 32 is disposed within circumferential groove 264 of circular end plate 261 for sealing the outer peripheral surface of circular end plate 261 and the inner wall of cup shaped casing 12. A hole or disharge port 265 is formed through circular end plate 261 at a position near the center of spiral element 262; discharge port 265 connects the fluid pockets at the center of spiral element 262 and rear chamber 30.

Orbiting scroll 27, which is disposed in front chamber 31, includes a circular end plate 271 and a wrap or spiral element 272 affixed to or extending from one end surface of circular end plate 271. The spiral elements 262 and 272 interfit at an angular offset of 180° and a predetermined radial offset. The spiral elements define at least a pair of fluid pockets between their interfitting surfaces. Orbiting scroll 27 is connected to the driving mechanism and the rotation preventing/thrust bearing device 28. The driving mechanism and the rotation preventing/ thrust bearing device 28 effect orbital motion of orbiting scroll 27 by the rotation of drive shaft 13 to thereby compress fluid passing through the compressor.

Referring to Figs. 6 and 7, the driving mechanism of orbiting scroll 27 will now be described. As described above, drive shaft 13, which is rotatably supported by sleeve 15 through bearing 18, has disk shaped rotor 19 at its inner end. Disk shaped rotor 19 is also rotatably supported by front end plate 11 through bearing 20. A crank pin or drive pin 191 projects axially from an axial end surface of disk shaped rotor 19 and is radially offset from the center of drive shaft 13. Circular end plate 271 of orbiting scroll 27 is provided with a tubular boss 273 axially projecting from the surface opposite to the end surface from which spiral element 272 extends. A discoid or short axial bushing 33 fits into boss 273, and is rotatably supported therein by a bearing, such as needle bearing 34. An eccentric hole is formed on bushing 33; the eccentric hole is radially offset from the center of bushing 33. Drive pin 191, which is surrounded by bearing 39, fits into the eccentric hole. Therefore, bushing 33 is driven by the revolution of drive pin 191 to thereby rotate within bearing 34. The spiral element of orbiting scroll 27 is thus pushed against the spiral element of fixed scroll 26 due to the moment created by the differential of the driving point and the reaction point of pressure gas for secured the line contacts.

Rotation preventing/thrust bearing device 28 is placed between the inner end surface of front end plate 11 and the end surface of circular end plate 271 which faces the inner end surface of front end plate 11, as shown in Fig. 6, and includes a fixed ring 281, which is fastened against the axial end surface of annular projection 112, an orbiting ring 282, which is fastened against the end surface of circular end plate 272 by a fastening device, and a bearing element, such as a plurality of spherical balls 283. Rings 281 and 282 have a plurality of indentations 284 and 285 and one of spherical ball 283 is retained between each of these indentations 284 and 285. Therefore, the rotation of orbiting scroll 27 is prevented by balls 283, which interact with the edges of indentations 284 and 285 to prevent rotation. Also, these balls 283 carry the axial thrust load from orbiting scroll 27. Therefore, orbiting scroll 27 orbits while maintaining its angular orientation to fixed scroll 26.

As orbiting scroll 27 orbits, the line contacts between spiral element 262 and 272 shift toward the center of the spiral elements along the surfaces of the spiral elements. The fluid pockets defined by the line contacts between spiral elements 262 and 272 move toward the center with a consequent reduction of volume, to thereby compress the fluid in the fluid pockets. Therefore, fluid or refrigerant gas introduced into front chamber 31 from an external fluid circuit through an inlet port 35 mounted on the outside of cup shaped casing 12 is taken into the fluid pockets formed at the outer portion of spiral elements 262 and 272. As orbiting scroll 27 orbits, the fluid in the fluid pockets is compressed as the pockets move toward the center of the spiral element. Finally, the compressed fluid is discharged into rear chamber 30 through hole 265, and thereafter, the fluid is discharged to the external fluid circuit through an outlet port 36 formed on cup shaped casing 12.

Referring to Fig. 8, each spiral element 262 and 272 is provided with a groove 37 formed in its axial end surface along the spiral curve of the spiral element. Groove 37 extends from the inner end portion of the spiral element to a position close to the terminal end of the spiral element. A seal element 38 is fitted within each groove 37. In this construction, as shown in Fig. 9, at least the axial dimension D_2 of seal element 38 is formed greater than the depth D_1 of groove 37. Seal element 38 thus usually contacts with the inner end surface of facing circular end plate without any axial force urging device. Therefore, axial sealing of the fluid pockets

0065261

will be secured.

In this construction, the gap C defined by the end plate 261, side wall of the spiral element 262, the axial end surface of facing spiral element 272 and seal element 38 fitted in the groove 37 of facing spiral element 272 will be reduced, and the amount of the fluid leakage from the high pressure chamber to lower pressure chamber is reduced. Therefore, the volume efficiency of the compressor is improved.

Furthermore, the width of the groove will be made as large as possible and the width of the seal element is made larger dependent on the width of the groove. Therefore, the width of contact surface between the seal element and the inner end surface of facing end plate will be made larger, so that the fluid leakage through the contact surface will be reduced.

In this construction, the width W_2 of seal element 38 is formed smaller than the width W_1 of groove 37. Therefore, the problem in that the assembling of the seal element into the groove would be in difficulty which is caused by the limit of the tolerance to surface roughness or width of the groove to be worked, or the width of seal element, is resolved. Also, if seal element is formed of hard plastic and/or self-lubricating material, the deformation of the seal element would occur in the long operation of the compressor. That is, the axial dimension of the seal element is reduced by the axial slant of the scroll, while the width of the seal element is increased. The increase of the width can be permitted by the differential of the width, and thereby secures the long life sealing of the fluid pockets.

Fig. 10 shows an alternative embodiment of the present invention which is characterized in that the width W_2 of seal element 38 is formed substantially same as the width W_1 of groove 37. The construction will more reduce the gap C, and therefore, the amount of fluid leakage is reduced and volume efficiency of the compressor is improved.

CLAIMS

1. A scroll type fluid displacement apparatus including a pair of scroll members (26, 27) each comprising an end plate (261, 271) and a spiral wrap (262, 272) extending from one side of said end plate (261, 271), said spiral wrap (262, 272) having a groove (37) formed in the axial end surface thereof along the spiral curve, said spiral wraps (262, 272) interfitting at an angular and radial offset to make a plurality of line contacts which define at least one pair of fluid pockets, driving means (13, 19, 28) operatively connected to one of said scroll member (27) for orbiting said one scroll member (27) relative to the other scroll member (26) and for preventing rotation of said one scroll member (27) to change the volume of the fluid pockets, characterized by a seal element (38) fitted within said groove (37) and at least the axial dimension of said seal element (38) formed greater than the depth of said groove (37).

2. The scroll type fluid displacement apparatus of claim 1, characterized in that the width of said seal element (38) is formed substantially same as the width of said groove (37).

3. A scroll type fluid compressor comprising:

a compressor housing (10) having a fluid inlet port (35) and fluid outlet port (36);

a fixed scroll member (26) fixedly disposed relative to said housing (10) and having an end plate (261) and a first spiral wrap (262) extending from said end plate (261) into the interior of said housing (10);

an orbiting scroll member (27) having an end plate (271) and a second spiral wrap (272) extending therefrom, said first and second spiral wraps (262, 272) interfitting at an angular and radial offset to make a plurality of line contacts defining at least two fluid pockets which merge to form a single pocket;

driving means (13, 19, 28) supported by said housing and connected to said orbiting scroll member (27) for orbiting said orbiting scroll member (27) and for preventing the rotation of said orbiting scroll member (27) to change the volume of the fluid pockets;

a groove (37) formed in the axial end surface of each of said first and second spiral wraps (262, 272) along the spiral curve, characterized by

a seal element (38) fitted within said groove (37) and at least the axial dimension of said seal element (38) formed greater than the depth of said groove (37).

4. The scroll type fluid compressor of claim 3, characterized in that the width of said seal element (38) is formed substantially same as the width of said groove (37).

5. Apparatus as claimed in one of the claims 1 to 4, characterized in that the said seal element (37) is formed of hard plastic and/or self-lubricating material.

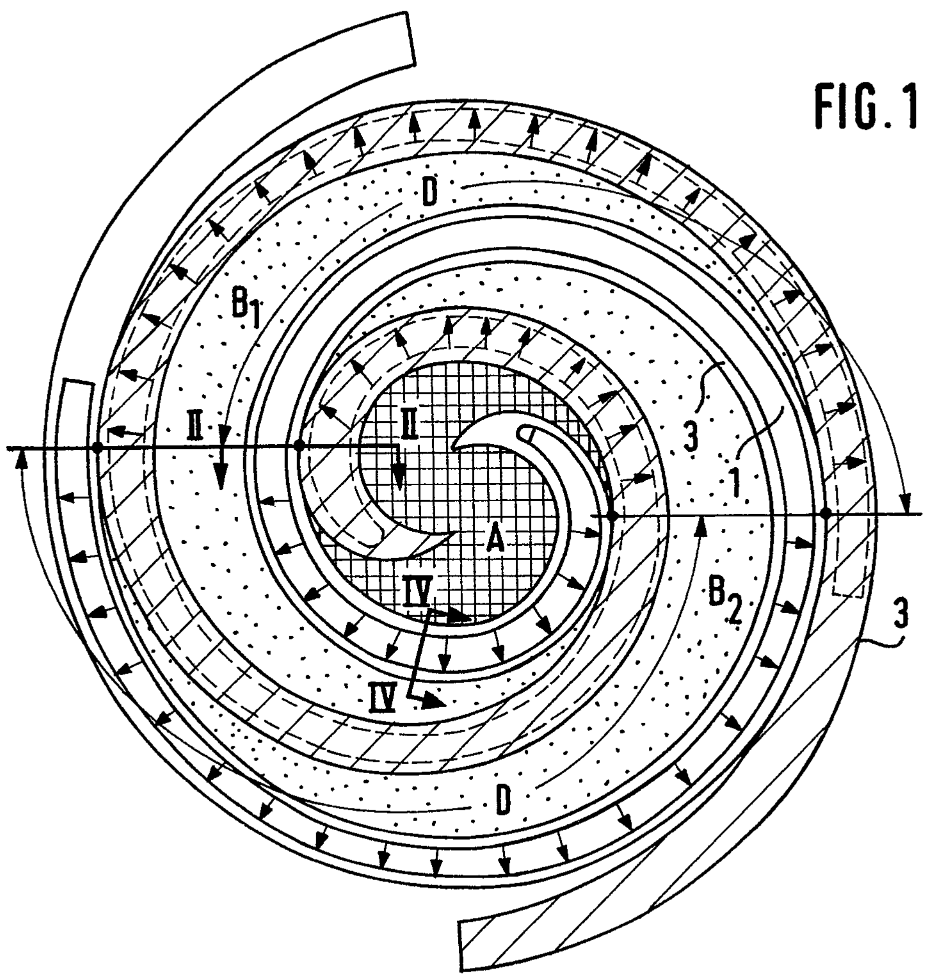


FIG. 1

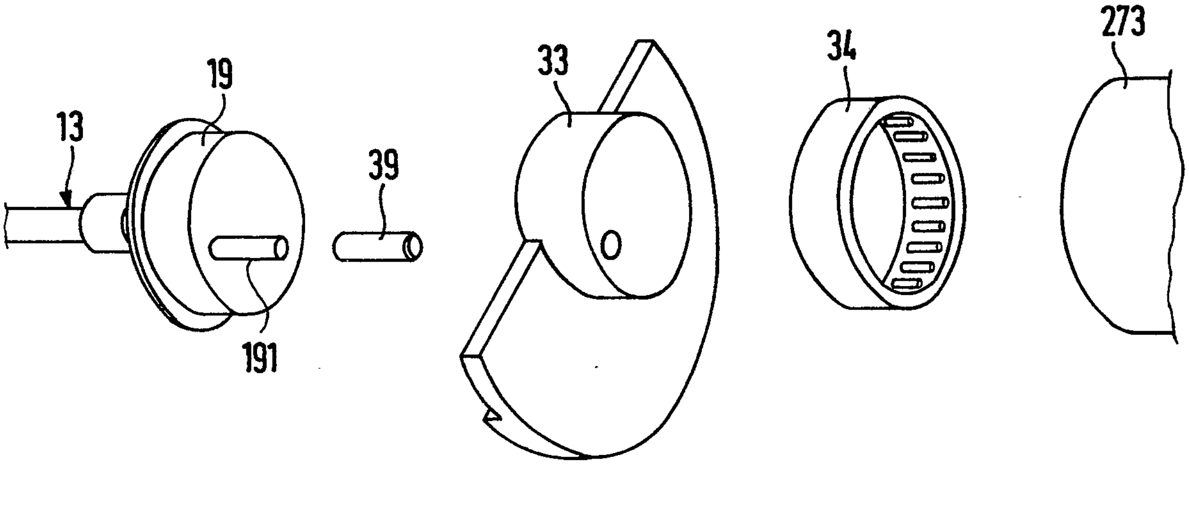


FIG. 7

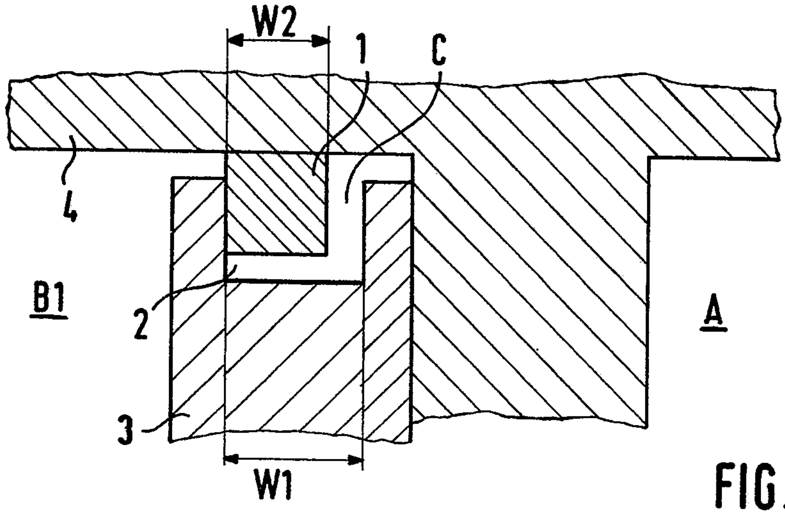


FIG. 2

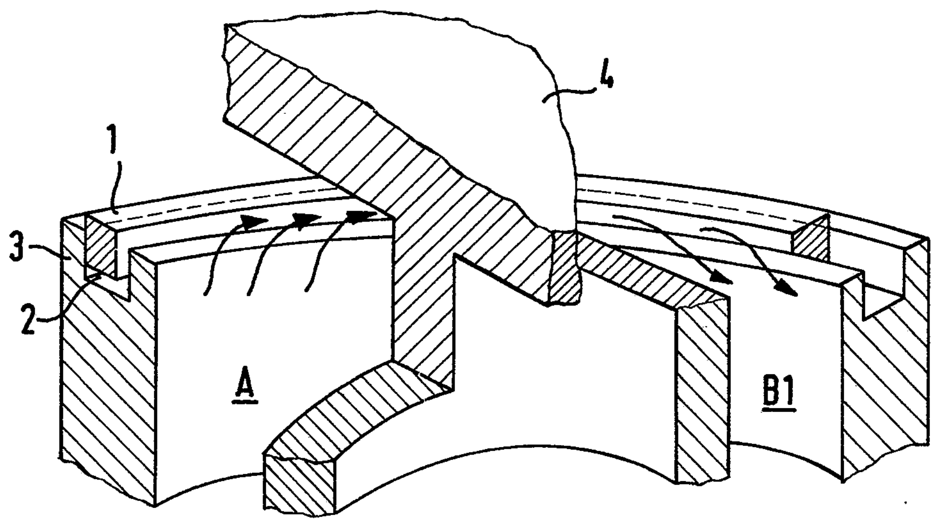


FIG. 3

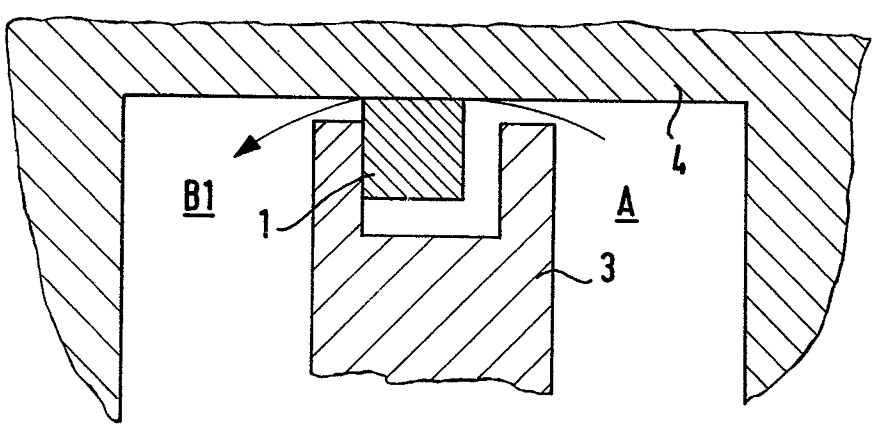


FIG. 4

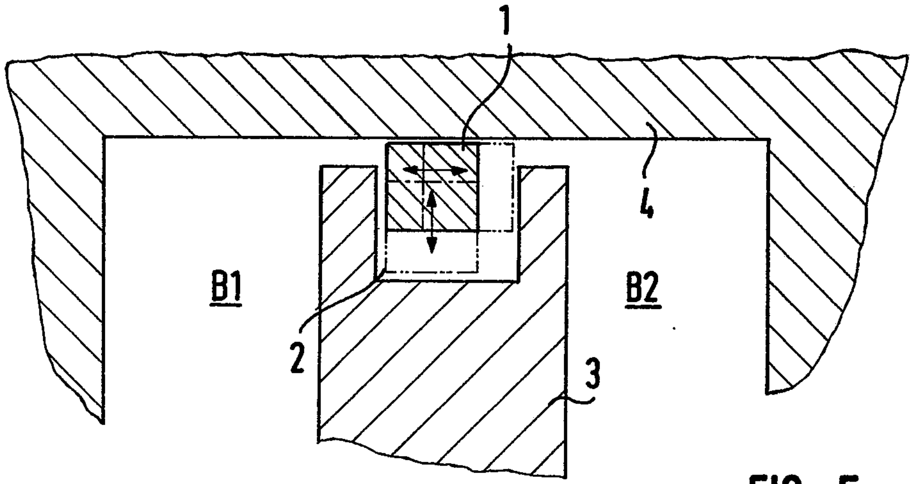


FIG. 5

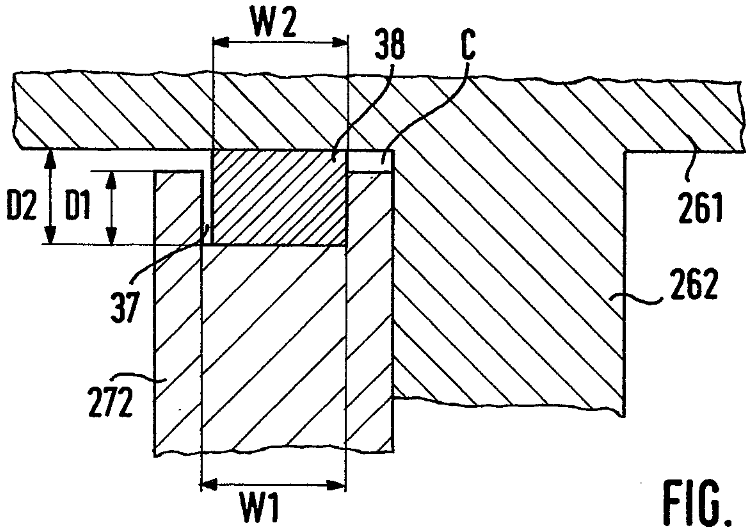


FIG. 9

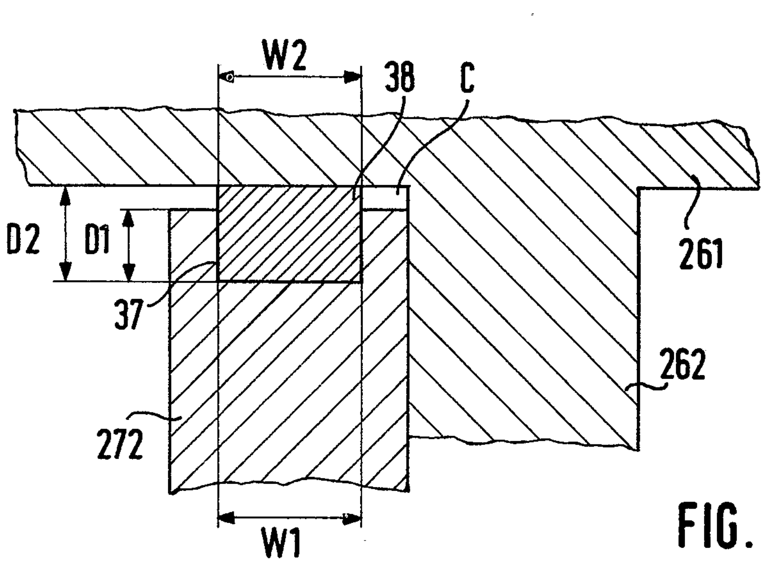
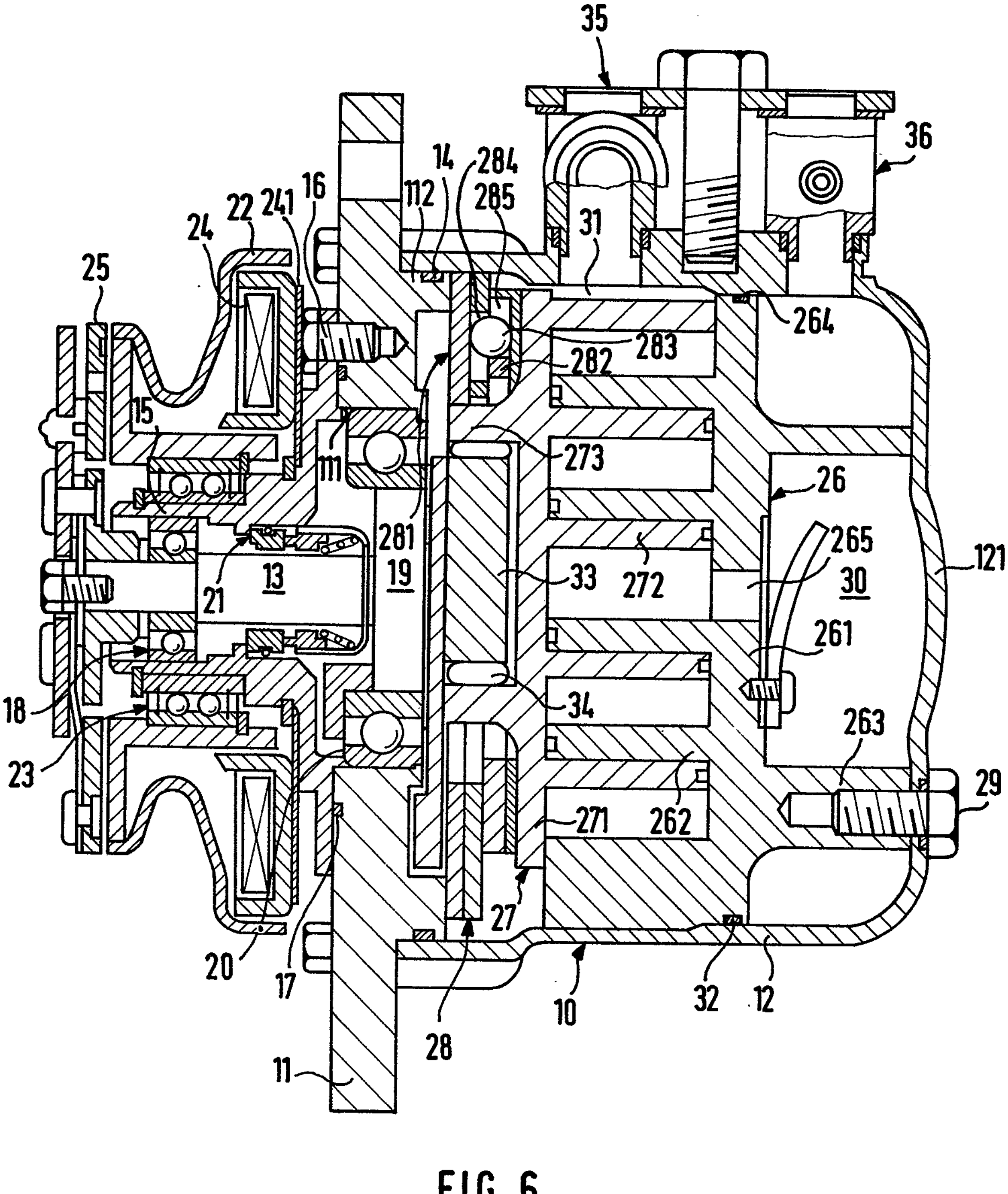


FIG. 10



35
36
31
264
283
282
273
26
265
30
121
261
263
29
272
33
34
271
262
27
28
32
12
10
11
14
284
285
112
16
241
22
24
25
15
111
281
19
13
21
18
23
20
17

FIG. 6

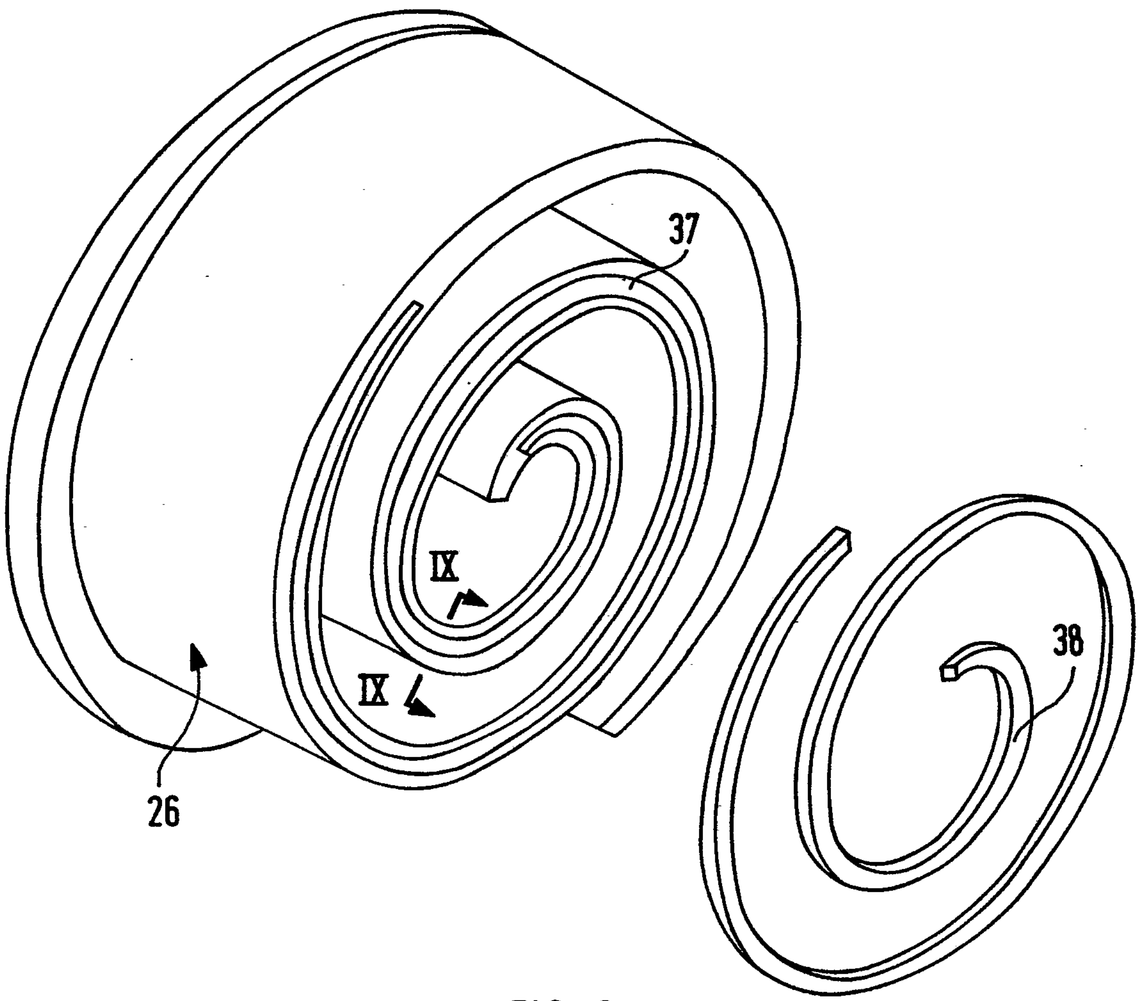


FIG. 8