

- 1 -

SCROLL-TYPE MACHINE

The present invention relates to fluid displacement apparatus and more particularly to an improved scroll-type machine especially adapted for compressing gaseous fluids and a method of manufacture thereof.

A class of machines exists in the art generally known as "scroll" apparatus for the displacement of various types of fluids. Such apparatus may be configured as an expander, a displacement engine, a pump, a compressor, etc., and many features of the present invention are applicable to any one of these machines. For purposes of illustration, however, the disclosed embodiments are in the form of a gaseous fluid compressor.

Generally speaking, a scroll apparatus comprises two spiral scroll

wraps of similar configuration each mounted on a separate end plate to define a scroll member. The two scroll members are interfitted together with one of the scroll wraps being rotationally displaced 180° from the other. The apparatus operates by orbiting one scroll member (the "orbiting scroll") with respect to the other scroll member (the "fixed scroll") to make moving line contacts between the flanks of the respective wraps defining moving isolated crescent-shaped pockets of fluid. The spirals are commonly formed as involutes of a circle, and ideally there is no relative rotation between the scroll members during operation, i.e., the motion is purely curvilinear translation (i.e. no rotation of any line in the body). The fluid pockets carry the fluid to be handled from a first zone in the scroll apparatus where a fluid inlet is provided, to a second zone in the apparatus where a fluid outlet is provided. The volume of a sealed pocket changes as it moves from the first zone to the second zone. At any one instant in time there will be at least one pair of sealed pockets, and when there are several pairs of sealed pockets at one time, each pair will have different volumes. In a compressor the second zone is at a higher pressure than the first zone and is physically located centrally in the apparatus, the first zone being located at the outer periphery of the apparatus.

Two types of contacts define the fluid pockets formed between the scroll members: axially extending tangential line contacts between the spiral faces of the wraps caused by radial forces ("flank sealing"), and area contacts caused by axial forces between the plane edge surfaces (the "tips") of each wrap and the opposite end plate ("tip sealing"). For high efficiency, good sealing must be achieved for both types of contacts, however, the present invention is primarily concerned with flank sealing. In a conventional scroll compressor (i.e. one in which

the wraps are involutes of a circle) good flank sealing requires that there be no relative rotation between the scrolls.

Scroll devices are generally described in an early patent to Creux, U.S. Patent No. 801,182. Among subsequent representative patents which 5 disclose scroll compressors and pumps are U.S. Patent Nos. 1,376,291, 2,475,247, 2,494,100, 2,809,779, 2,841,089, 3,560,119, 3,600,114, 3,802,809, 3,817,644, 3,884,599, 4,141,677, 4,300,875, 4,304,535 and 4,357,132.

The concept of a scroll-type apparatus has thus been known for some 10 time and has been recognized as having distinct advantages. For example, scroll machines have high isentropic and volumetric efficiency, and hence are relatively small and lightweight for a given capacity. They are quieter and more vibration free than many compressors because they do not use large reciprocating parts (e.g. pistons, connecting 15 rods, etc.), and because all fluid flow is in one direction with simultaneous compression in plural opposed pockets there are less pressure-created vibrations. Such machines also tend to have high reliability and durability because of the relative few moving parts utilized, the relative low velocity of movement between the scrolls, 20 and an inherent forgiveness to fluid contamination. In spite of this, however, it is believed that the reason use of scroll machines of the prior art has not hitherto become wide spread is because of the difficulty of manufacturing such machines, as well as inherent sealing and unusual wearing problems.

25 One of the more difficult areas of design in a scroll machine concerns the technique used for preventing relative angular movement between the scrolls as they orbit with respect to one another. One of the more popular approaches resides in the use of an Oldham coupling operative between the orbiting scroll and a fixed portion of the

apparatus. An Oldham coupling typically comprises an Oldham ring and two sets of key members or slider blocks. The Oldham ring is formed on one side thereof with grooves which are at right angles to a similar grooves formed on the other side thereof. One set of key members is
5 connected to a surface of the orbiting scroll and is disposed in the grooves on one side of the Oldham ring, while the other set of key members is fixed to either the fixed scroll or the machine housing and is disposed in the grooves on the other side of the Oldham ring. The Oldham ring reciprocates in a motion parallel to the grooves containing
10 the set of key members fixed to the fixed scroll or housing. The Oldham coupling thus acts as a means for controlling (i.e. preventing) angular rotation of the orbiting scroll relative to the fixed scroll, a concept that is believed to be essential to the successful functioning of a conventional scroll apparatus. U.S. Patent No. 4,121,438 shows a
15 machine of this construction.

Unfortunately, a scroll apparatus utilizing an Oldham coupling possesses several assembly, operational, and maintenance disadvantages, primarily due to the large number of parts that make up the coupling. This large number of parts increases material, manufacturing, and
20 assembly cost. The slider blocks or key members in an Oldham coupling are also all sliding within the grooves on the Oldham ring, thus presenting lubrication and wearing problems. The reciprocating ring is also inherently impossible to balance.

There are other known devices for controlling relative rotation of
25 the scrolls, such as the use of multiple drives rotating both scrolls about different centers, and like concepts; however, because of their complexity these devices also have many of the undesirable features of the Oldham coupling.

The present invention approaches the problem from a different

direction. The basic concept of this invention resides in the use of a very simple orbiting scroll rotation controlling device which does not eliminate relative rotation, but simply controls it at a relatively small magnitude, in combination with techniques for easily modifying the contour of the scroll wraps to accommodate the limited relative rotation of the scrolls which occurs, in order to maintain flank sealing contact between the wraps. Three different versions of the rotation controlling device are disclosed, each being a simple linkage operatively connected between the orbiting scroll and a fixed portion of the compressor. Two
10 embodiments are four-bar linkages and two additional embodiments are crank and slider arrangements. All versions couple the orbiting and fixed scrolls in a predetermined angular relationship in all relative positions as the orbiting scroll orbits with respect to the fixed scroll.

15 The orbiting scroll rotation controlling linkage of the present invention overcomes the aforesaid disadvantages of known techniques for accomplishing this function. Because it has fewer parts, it is less expensive to manufacture and is much easier to assemble. Lubrication problems are also reduced to a minimum because there are fewer
20 connections and in one embodiment because the connections are entirely pivotal, rather than sliding. The anti-rotation member also contains no reciprocating ring.

In order to accommodate for the slight degree of rotation of the orbiting scroll, the present invention utilizes either one of two
25 unique techniques for modifying the scroll wrap contours. Both of them are very easy to implement and provide highly efficient flank sealing (one of them provides theoretically perfect flank sealings) when used in combination with the aforesaid linkage arrangements. A novel wrap machining method is also provided.

23

The compressor of the present invention also embodies improved drive means for orbiting the scroll which is compact, as well as being simple to manufacture and assemble. This drive means, two embodiments of which are disclosed, provides the necessary contact between the scroll wraps to have efficient wrap sealing while at the same time provides an automatic unloading function in the event slugging or the like occurs. Should a non-compressible fluid be sucked into the compressor the orbiting radius of the orbiting scroll will automatically decrease to permit this fluid to escape between the scroll wraps to an area of less pressure (i.e. the scrolls will simply ride over any liquid). This function is accomplished without extra parts, such as valves or the like, and is very smooth in operation. The drive means is also nested within the upper crankshaft bearing, thereby reducing the axial height of the machine and lessening vibration.

The compressor of this invention also incorporates an improved thrust bearing arrangement to accommodate the axial loading of the orbiting scroll. The thrust bearing takes advantage of the inherent rotating movement of the orbiting scroll to create a squeeze film which provides the oil support for the bearing. The resulting bearing provides significantly less friction than conventional needle bearings and is much less subject to harmful wear than conventionally used balls. Ball-type thrust bearings, especially those with relatively small balls, have such high point loading that they tend to wear excessively. The present bearing is also relatively inexpensive, very simple in design and easy to manufacture and assemble.

Additional advantages and features of the present invention will become apparent from the subsequent description and the appended claims taken in conjunction with the accompanying drawings.

26

BRIEF DESCRIPTION OF THE DRAWINGS

Figure 1 is a vertical sectional view through a refrigerant compressor embodying the principles of the present invention;

Figure 2 is a partial plan view of the top of a portion of the compressor illustrated in Figure 1;

5 Figure 3 is a sectional view taken along line 3-3 in Figure 1;

Figure 4 is a sectional view taken along line 4-4 in Figure 1;

Figure 5 is a sectional view taken along line 5-5 in Figure 4;

Figure 6 is a sectional view taken along line 6-6 in Figure 1;

10 Figure 7 is a partial sectional view taken along line 7-7 in Figure 6;

Figure 8 is an enlarged exaggerated illustration of the orbiting drive means of Figure 6 showing it in its normal driving position;

Figure 9 is a view similar to Figure 8 but showing the drive means in its unloaded position;

15 Figure 10 is a sectional view taken along line 10-10 in Figure 1;

Figure 11 is a sectional view taken along line 11-11 in Figure 1;

Figure 12 is a bottom plan view of the fixed upper scroll member;

Figure 13 is a top plan view of the lower orbiting scroll member;

20 Figure 14 is a view similar to Figure 4 showing a second embodiment of the rotation controlling means of the present invention;

Figure 15 is a fragmentary horizontal sectional view showing a third embodiment of the rotation controlling means of the present invention;

25 Figure 16 is a view similar to Figure 15 illustrating a fourth embodiment of the rotation controlling means of the present invention;

Figure 17 is a fragmentary horizontal sectional view illustrating a second embodiment of a drive means of the present invention in its normal driving condition;

Figure 18 is a view similar to Figure 15 but illustrating the drive means in an unloaded condition.

Figure 19 is a diagrammatic illustration of one embodiment of a pair of mating scroll wraps embodying the principles of the present
5 invention;

Figures 20A, 20B, 21, 22 and 23 are diagrams of the geometry of the present invention showing the nomenclature used; and

Figure 24 is a schematic view of an apparatus which can be used to practice the machining method of the present invention.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

General Layout

Referring to Figure 1, there is illustrated an exemplary refrigerant compressor assembly of the hermetic type embodying the principles of the present invention. The compressor is disposed within a hermetic enclosure consisting of an upper shell 10 and a lower shell 12 welded together in the usual manner as at 14. Affixed to the bottom of lower shell 12 are a plurality of feet 16, one of which is shown. The assembly comprises a motor 18 drivingly affixed to a crankshaft 20 which in turn drives a scroll-type compressor 22 connected to the upper end thereof, an oil pump 24 being driven by the lower end of crankshaft 20 to provide lubrication to the machine. Lower shell 12 has a sump of lubricating oil therein, the upper level of which is indicated at 26.

Motor-Compressor Assembly

The motor-compressor assembly comprises a lower bearing housing 28 having a bearing 30 therein in which the lower end of crankshaft 20 is rotationally supported and journaled, and an upper bearing housing 32 defining a bore 33 having spaced sleeve bearings 34 and 36 disposed therein which rotationally support and journal the upper end of crankshaft 20. The bearing housings are affixed to one another by means of a plurality of threaded fasteners 38 which operate to clamp the stator 40 of motor 18 therebetween. Affixed to crankshaft 20 and disposed within the central bore of stator 40 is a rotor 42. The motor is in all respects conventional in design and operation. The motor compressor assembly is mounted within lower shell 12 by means of a plurality of mounting studs 44 (one shown) which are threadably connected to lower bearing housing 28 and are bolted to the bottom of lower shell 12 in the manner best shown at 46. Leakage from the shell is prevented by means of an O-ring 48 sealingly engaging the outer surface of each mounting stud 44.

The compressor, per se, comprises an upper fixed scroll member 50 and a lower orbiting scroll member 52 having a flat lower face 53 and a downwardly extending drive hub 54 disposed within an eccentrically positioned bore 56 in the upper end of crankshaft 20. The end
5 of hub 54 has a recess 55. An upper counterweight 57 is provided on crankshaft 20 adjacent bore 56 to help balance the eccentricity of the drive. The crankshaft has a lower counterweight 59 affixed adjacent the lower end thereof to complete the balancing system, in accordance with known principles. Fixed scroll member 50 is located an accurately
10 predetermined distance above planar upper surface 58 of upper bearing housing 32 by means of a spacer ring 60, and is bolted to housing 32 by means of a plurality of studs 62 which also serve to clamp spacer ring 60 therebetween. The upper end of each stud 62 projects into a nylon spacer 64 disposed within a cup-like element 66 affixed to a mounting
15 ring 68 welded to the top of upper shell 10. In this manner the top of the motor-compressor assembly is mounted to the top of the shell. Although only one is shown, a number of such connections are provided.

Fixed scroll member 50 comprises a planar tip seal surface 70 from which projects a spiral scroll wrap 72. Scroll member 50 is provided
20 with a centrally disposed kidney-shaped discharge opening 74 communicating with a discharge chamber 76 disposed at the upper surface thereof. Chamber 76 is covered by a suitably gasketed valve plate 78 having ports 79 therethrough and being retained in place by means of a cylinder head 80 bolted to the upper surface of scroll member 50 by
25 means of a plurality of threaded fasteners 82. The direction of flow through ports 79 is controlled by a conventional leaf-type check valve 81 which is affixed to valve plate 78, along with the usual backer 83, by means of a rivet 85.

Discharge fluid passes from within cylinder head 80 via an opening

84 in which is disposed one end of a discharge tube 85. Tube 85 extends over the top of the assembly downwardly between the latter and the shell to a point where it exits from the shell in the normal manner (not shown). Suction gas is introduced into the shell in the normal manner
5 via a fitting 88 extending through the wall of upper shell 10. The motor-compressor disclosed herein is of the "low side" type with suction gas present in the hermetic shell for purposes of motor cooling and the like. The gas enters compressor 22 at the periphery of the scroll wraps.

10 Orbiting scroll 52 comprises a planar tip sealing surface 90 from which projects a spiral scroll wrap 92 adapted to sealingly engage wraps 72 upon relative orbital movement of the scroll members in the usual manner to form fluid pockets of decreasing volume. The tips of wrap 92 are intended to sealingly engage surface 70 of the fixed scroll and the
15 tips of wrap 72 are intended to sealingly engage surface 90 on the orbiting scroll. The central area of orbiting scroll 52 is provided with a kidney-shaped recess 94 to facilitate the flow of discharge fluid. Both wraps 72 and 92 are equally and uniformly tapered in cross-section in order to maximize wrap strength at the root thereof.
20 Discharge opening 74 and recess 94 are configured to provide maximum flow area without weakening the root of the adjacent wraps. Tip seal surfaces 70 and 90 should lie in parallel planes which are parallel to the planes in which the tips of both wraps lie, and which are perpendicular to the rotational axis of crankshaft.20.

25 Orbiting Drive Means

The drive means of the present invention does not utilize a rigid coupling between the crankshaft and the orbiting scroll, but instead is one which will yield radially to permit automatic unloading of the compressor in the event of ingestion of an incompressible fluid or

misalignment of the parts. The drive means relies on centrifugal and driving forces to maintain the flanks of the orbiting scroll in sealing contact with those of the fixed scroll. This is accomplished in a very axially compact manner using an unloader hub 100, as best illustrated in
5 Figures 1 and 6-9.

Hub 100 comprises an outer circular cylindrical retainer 102 and axially spaced inner sleeve bearings 104 and 106 in which hub 54 of the orbiting scroll is journaled. Retainer 102 has a flat driven surface 108 on the periphery thereof, surface 108 lying in a plane
10 parallel to the axis of rotation of the crankshaft. Driven surface 108 drivingly engages a complementary flat driving surface 110 in the wall of bore 56, the latter surface being of greater width than the former. Bore 56 is otherwise of sufficient size and shape to permit limited relative tangential sliding between hub 100 and crankshaft 20. The
15 center of rotation of the crankshaft is indicated at cc and the center of the hub and orbiting scroll is indicated at cs. Rotation of crankshaft 20 causes hub 100 to revolve about the axis cc. This in turn causes hub 54, and hence orbiting scroll member 52, to orbit about the axis cc. The radius of the orbit of the orbiting scroll member is
20 determined by the geometry of the wrap profiles (i.e. the engagement of the flanks of the wraps of the orbital scroll member with the flanks of the wraps of the fixed scroll member).

In the event the wraps encounter a slug of incompressible fluid, the forces thereby created tending to separate the wraps will be
25 accommodated by unloader hub 100 sliding from the normal driving position shown in Figures 6, 8 and 9 to the exaggerated position shown in Figure 9. This will permit the wraps to separate and pass over the fluid slug, whereupon centrifugal force will cause the orbiting wrap to move back into sealing engagement with the fixed wrap.

5 The axially nested relationship of the parts minimizes the axial height of the assembly. Furthermore, it permits locating the upper counterweight very close to the center of mass of the orbiting scroll, which reduces bearing loads and bending forces on the crankshaft, as well as the size of the counterweight required. It also eliminates a cantilevered drive bearing at the top of the crankshaft, which further reduces bearing loads.

10 The present invention embodies another unique version of an orbiting drive means which will also give the desired orbital motion and yet still permit automatic unloading of the compressor in the event an incompressible fluid is encountered. This embodiment, illustrated in horizontal section in Figures 17 and 18, comprises a crankshaft 112 (having a center of rotation cc) in the upper end of which is disposed an axial bore 114 of the shape illustrated, the significant part of which is a circular cylindrical surface 115 (having a center of radius 15 cp) in which is nested a hub 116 having a projection 117 of radius equal to that of surface 115. Hub 116 has a circular bore 118 therethrough (having a center axis cs) in which is journaled hub 54 of the orbiting scroll. Suitable bearings may be provided at the interface of bore 118 and hub 54 if desired. As can be seen, the geometry of the arrangement 20 is such that upon rotation of crankshaft 112 about axis cc centrifugal force will normally maintain the respective parts in the positions shown in Figure 17, with the orbiting radius being dictated by the geometry of the scroll flanks, as aforesaid. When an unloading situation occurs, however, hub 116 can rotate about center cp to the position shown in 25 Figure 18 to permit the wrap flanks to separate to the extent necessary to pass the incompressible matter disposed therebetween.

Rotation Controlling Means

The present invention solves the problem of coupling the scroll

33

members to control relative rotation therebetween in a very simple and unique way. In the embodiment of Figures 1 through 13 (see particularly Figures 1, 4 and 5), this is accomplished by providing a projection 120 on the outer periphery of orbiting scroll member 52 having an axially extending bore 122 therethrough in which is pivotally disposed a
5 projection 124 at one end of a simple link 126, the opposite end of which is pivotally mounted to any fixed part of the apparatus. In the embodiment illustrated, link 126 is journaled on one of the axially extending locating pins 128 which project into and accurately locate with respect to one another fixed scroll member 50 and upper bearing
10 housing 32. The lower surface of link 126 is flat and slidably rests upon the flat upper surface of spacer ring 60.

Link 126 is preferably as long as possible (to minimize the effect of the fact that projection 124 will move in an arcuate rather than a straight line path) and should operate as close as reasonably possible
15 to perpendicular to an imaginary line extending between the center of the orbiting scroll member and bore 122. This very simple linkage arrangement replaces the complex Oldham coupling used in known devices. The small degree of relative rotation between the scroll members which this linkage arrangement does permit is accurately controlled and is of
20 a defined magnitude and nature so that it can be readily accommodated without sacrificing performance or efficiency, as will be described hereinbelow.

The present invention also embodies other simple linkage arrangements, such as those shown in Figures 14-16. The embodiment of
25 Figure 14 is almost identical to that of Figures 1 through 13 except that the simple link is relatively straight, as indicated at 130, and is journaled at its fixed end to a projection 132 mounted upon a bracket 134 rigidly affixed to upper shell 10. The movable end of link 130 is



in all respects the same as that described with respect to the preceding embodiments and the function of the two links is the same. Both of these linkage arrangements are known in the art as four-bar linkages. The first "bar" of the linkage is link 126 or 130, which extends from a
5 fixed portion of the structure to the periphery of the orbiting scroll. The second "bar" of the linkage is the imaginary link between the pivot point at the periphery of the orbiting scroll and the geometric center of the scroll. The third "bar" is the imaginary link which extends between the geometric center of the scroll and the center about
10 which the scroll orbits, and the fourth "bar" is the fixed structure which extends from the fixed pivot of the first "bar" to the fixed pivot of the third "bar".

The embodiments of the rotation controlling means illustrated in Figures 15 and 16 are both of the crank and slider type. In the
15 embodiment of Figure 15 projection 120 on orbiting scroll member 52 is provided with a radially extending elongated slot 136 in which is slidably disposed a pin 138 which is mounted upon a bracket 140 which is rigidly affixed to upper shell 10. In the embodiment of Figure 16, projection 120 of orbital scroll member 52 has affixed thereto a pin 142
20 which is slidably disposed within a radially extending elongated slot 144 in a bracket 146 which is affixed to upper shell 10. In both of these last two embodiments, rotation of the orbital scroll member is prevented as a result of the interengagement of pin 138 in slot 136 or pin 142 in slot 144.

25 Thus, in all four embodiments the rotation of the orbital scroll members is prevented by pivoting it about a single relatively fixed pivot point. The only significant difference between the arrangements is that in the four-bar linkage arrangements the single pivot moves in a slightly arcuate path whereas in the two crank and slider arrangements

relative movement of the pivot is in a straight-line path. In addition, in the embodiment of Figure 15 the imaginary link between the single pivot point and the geometric center of the orbiting scroll varies as the scroll orbits, whereas it is fixed in the other
5 embodiments.

Thrust Bearing

The handling of high thrust loads, which are an inherent characteristic of scroll machinery, is an area which has presented many problems to designers in this field. Applicants have discovered that these thrust loads can be efficiently handled using a bearing which
10 relies on the inherent imbalance of the orbiting scroll member to support the thrust load through a squeeze film mechanism. This bearing has very low friction, is very simple in design, requires only flat finishing, has high stiffness, and because of the low relative speeds encountered has small shear losses. It also transmits the thrust loads
15 directly to the frame of the apparatus (i.e. upper bearing housing 32).

As best shown in Figures 1, 10 and 11, the thrust bearing assembly comprises a relatively thin bearing 150 formed of Teflon^(Registered Trade Mark) and lead impregnated bronze, or other suitable bearing material, having opposed parallel flat faces slidably engaging on its lower side flat upper
20 surface 58 of upper bearing housing 32 and on its upper side flat surface 53 on orbiting scroll member 52. Relative movement between bearing 150 and scroll member 52 is prevented by means of a pin 152 which projects into both parts. There are no oil grooves on bearing 150, but instead upper surface 58 of housing 32 is provided with two
25 concentric grooves 154 and 156. Groove 156 is an oil supply groove and is supplied oil via a plurality of passages 158 which communicates through housing 32 to bore 33 in the area of the space between sleeve bearings 34 and 36. Groove 154 is an oil drain groove which permits oil

30
to drain back to the open crankcase defined by housing 32 via a plurality of passages 160, in order to reduce oil carryover into the suction gas entering the compressor. Surfaces 53 and 58 should be parallel to one another and perpendicular to the axis of rotation of 5 crankshaft 20.

It is believed that this bearing functions because the inherent miniscule tilting or wobbling of the orbiting scroll as it orbits creates a rotary oil wave which traverses the entire face of the bearing at a speed determined by the rotational speed of the crankshaft. Thus 10 if the motor is operating at 3600 rpm the relative axial velocity of the parts will cause this wave to traverse the entire face of bearing 150 and surface 58 in a circular path at 3600 rpm. A standard hydrodynamic bearing would not, it is believed, function properly because there is not sufficient relative movement (i.e. high enough tangential velocity) 15 between the scroll and housing because it is only orbital movement of relatively small displacement. Because the bearing of the present invention utilizes a relatively large flat surface, unit loading is relatively small and long bearing life therefore possible.

The thrust loads encountered at the top of the orbiting scroll 20 member are taken by the axial interengagement of the two scroll members; specifically the engagement of the wrap tips on one scroll member with the tip sealing surface on the other scroll member. Fasteners 62 tend to urge the scroll members axially together and the fluid pressures generated by operation of the compressor tends to urge them apart. It 25 is desirable to have sufficient axial force urging the scrolls together, or maintaining them in a sufficiently close relationship, that good tip sealing is achieved. On the other hand, it is desirable to minimize friction and wear as much as possible.

In the present compressor this tip wear is controlled by

substantially increasing the thickness of the end of the outer wrap of each scroll member, as best shown in Figures 12 and 13. It is well known that the outer flank of the outer 180° of each scroll wrap is not utilized in the compression cycle. This fact is taken advantage of in the compressor of the present invention to control tip wear. As can be seen in Figure 12, approximately the last 180° of wrap 72 is of much greater thickness than the rest of the wrap, starting at 170 and ending at 172. This surface slidably rides on surface 90 of orbiting scroll member 52. Lubricating oil is supplied to the interface via a passage 174 in orbiting scroll member 52 which in turn receives oil from a vertical passage 178 communicating with the space between sleeve bearings 104 and 106 via a radial passage 180 (Figures 7 and 13). The outer end of passage 176 is plugged in the usual manner. As shown in Figure 13, approximately the last 180° of wrap 92 is also of much greater thickness than the remainder of the wrap, starting at 182 and ending at 184. In addition, this widened wrap portion is provided with an oil supply groove 185 which receives oil from a passage 186 communicating with passage 176. The use of these relatively wide surfaces reduces unit loading in the axial direction and thereby reduces tip wear.

Lubrication

Lubrication of the respective parts of the machine is accomplished by using oil pump 24 which is mechanically driven by the crankshaft and operates to pump oil under pressure from the oil sump at the bottom of the shell, indicated generally at 200, to all of the moving parts of the machine which require lubrication. Oil pump 24 may be of any suitable type, in accordance with known criteria, however, for exemplary purposes it is of the type shown in U.S. Letters Patent Nos. 4,331,420 or 4,331,421, the disclosures of which are incorporated herein by



reference. The inlet to the pump is indicated at 202 and the outlet of the pump is via a vertically extending oil passage 204 extending up the center of crankshaft 20.

As best seen in Figures 1, 6 and 7, the upper end of passage 204 communicates with a transverse passage 206 in the crankshaft, the opposite ends of which communicate with vertically extending passages 208 and 210. The radially outer end of passage 206 and the upper ends of passages 208 and 210 are plugged in the usual manner. Passage 208 communicates with a radially extending passage 212 which places the space between sleeve bearings 34 and 36 in communication with a supply of lubricating oil under pressure. Passage 210 communicates with a generally radially extending passage 214 which communicates with the interface of surfaces 108 and 110 and a radially extending passage 216 in hub 100 for placing the space between sleeve bearing 104 and 106 in communication with a supply of lubricating oil under pressure. Lubricating oil is communicated from the space between the outer pair of sleeve bearings to the thrust bearing via passage 158, as aforesaid, and from the space between the inner pair of sleeve bearings to the mating scroll tip surfaces via passages 180, 178, 176, 174 and 186, as aforesaid.

Because the lower counterweight 59 is disposed below the liquid level of the oil sump (due to the size of the counterweight required to balance the rather large eccentric mass of the orbiting scroll), provision is made to prevent the rotating counterweight from pumping oil and thereby needlessly consume energy. This is accomplished by forming a recess 220 in lower bearing housing 28 which closely approximates the outside surface of counterweight 59 as it rotates. The outer periphery of chamber 220 communicates with an oil outlet conduit 222 which extends radially outwardly and then upwardly between the housing and

shell. Upon starting of the compressor, the rotating of counterweight 59 will cause oil to be pumped out of chamber 220 via conduit 222 into the sump area of the shell. In order to permit this evacuation of oil from chamber 220, housing 28 is provided with a passage 224 which
5 communicates at one end with the central portion of chamber 220 and at the other end with a conduit 226 which extends to above the level of oil in the bottom of the sump. This conduit permits gas or vapor to enter chamber 220 when the compressor is started, thereby permitting the evacuation of all oil therefrom. Excess oil at the top and bottom of
10 the inner sleeve bearings, the top of the outer sleeve bearings and the inside of the thrust bearings is drained back to the crankcase via passageways 217 and 218.

Wrap Flank Contours

15 Conventional scroll compressors having wrap contours which are involutes of a circle require curvilinear translation between the fixed and moving scrolls in order for line-to-line flank contact to be maintained. If the curvilinear anti-rotation device is replaced with the linkage of the present invention, which allows slight pivoting of the
20 orbiting scroll, some means of maintaining flank line-to-line contact must be employed to maximize efficiency. Two such techniques are disclosed. The first is an involute shifting technique, and the second is an involute modification which results in scroll flank profiles which are not true involutes or are not involutes of a circle. Flank leakage
25 distances (i.e. the distance between the mating scroll wraps) are extremely small when the first technique is used. The second technique gives zero theoretical flank leakage distances.

Generally speaking, scroll compressors utilize two or more spiral wraps generated from the involute of a plane geometry and mounted on

base plates. If a generating circle is used, its radius, the scroll wrap thickness and the total number of wraps are chosen to give the desired volumetric displacement and pressure ratio. When the scrolls are rotated 180° and brought together so that the flanks and tips touch, 5 pockets are formed. If one scroll is made to move in curvilinear translation (no rotation of any line in the body) with respect to the other scroll, then these pockets are sealed and move inwardly toward the center or outwardly toward the scroll ends, depending on the direction of rotation. When there are more than 1-1/2 working wraps, the fluid in 10 the pockets will be compressed or expanded. If the motion is true curvilinear translation, perfect flank sealing will be obtained (discounting involute surface irregularities). Because of the relatively long sealing line of the flanks and tips, small clearances can result in substantially reduced efficiencies. Tip clearance management techniques 15 are not specifically the subject matter of this invention; flank clearance management techniques are.

By way of background, and with reference to Figure 20A, the equations that describe the scroll involute profiles when the relative motion is curvilinear translation and the relative rotational 20 displacement between the generating (x-y) axis of the scroll members is 180°, are:

$$R = C(1 + (Ar - \pi T/C)^2)^{1/2}$$

$$Ar = Ac + \pi/2 = Ap - \arctan(Ap)$$

25 Where: R is the distance from the center of the base circle to the point on the scroll profile

C is the base generating circle radius

Ar is the roll angle

T is the thickness of the scroll wrap

41

- 22 -

M is a logic modifier; $M = 1$ for surface S1 and $M=0$ for surface S2

5 A_c is the Numerical Control Angle that is incremented or it may be thought of as the angle of the crank when contact occurs between the moving and fixed scroll at the point under consideration.

A_p is the polar coordinate angle

Curvilinear translation motion requires the use of an anti-rotation device such as the complex and relatively expensive Oldham coupling. A
10 much simpler and less costly device is the rotation controlling linkage arrangement described above. This arrangement, however, allows a slight pivoting of the orbiting scroll, which normally would cause the scroll flanks to separate or interfere and reduce machine efficiency. By changing the geometry of the wrap this clearance or interference can be
15 reduced or eliminated. The present invention incorporates two novel profile modifications, as follows:

Shifting Technique

It has been discovered that conventional involutes of a circle can be used with the aforesaid linkage arrangement and excellent flank sealing characteristics obtained by merely forming the inside involute
20 profile of a wrap flank on a different base circle center than the outside involute profile of the same wrap flank. The involute profiles are identical, but the resulting wrap varies in thickness throughout its length. The base circles of either or both inner and outer flanks on either or both scrolls may be shifted. Thus there are
25 three possible cases:

- (1) shifting the inner and outer profiles on only the fixed scroll;
- (2) shifting the inner and outer profiles on only the orbiting scroll
- (3) shifting the inner and outer profiles on both scrolls.

30 Case No. 1, surprisingly, has been discovered to provide the best



sealing and Case No. 2 the least acceptable. This is based on theoretical analysis; it is believed that in actual practice the leakage which could occur between mating flanks even in Case No. 2 will probably be acceptable. A pair of such wraps 250 and 252 mated together are 5 illustrated in an exaggerated form in Figure 19. This is a Case No. 1 machine in which wrap 250 is the orbiting wrap and wrap 252 is the fixed wrap and has all of the correction.

The geometry of the linkage mechanism of the present invention is illustrated in Figure 21, where:

10 P = location of rotation controlling pivot point on orbiting scroll (e.g. the axis of projection 124)

cs = location of the geometric center of the orbiting scroll

cc = rotational axis of crankshaft

15 L = distance from the geometric center of the orbiting scroll to point P (i.e. the length of the imaginary link formed by the orbiting scroll).

Am = Relative angle between scrolls

B = angle from start of involute on orbiting base circle to the pivot pin center (polar)

20 E = Eccentricity (radius of orbit of the orbiting scroll)

V = Crankshaft angular velocity

Tm = Time

S1 = Inner flank profile

S2 = Outer flank profile

25 In order to design the scrolls of the present invention, one first must design conventionally shaped scrolls (such as would be used with an Oldham coupling) to provide the proper displacement, pressure ratio, and exit port geometry according to conventional procedures. This results in known values for the involute base circle radius C, the scroll

(43)

thickness T and the number of wraps of outer surface. Once this is done, one needs to choose the location of point P , the pivot point of the orbiting scroll. The distance between the geometric center cs and point P is L . For purposes of manufacture, the distance L is chosen to be convenient. It could be less than the distance from the geometric scroll center to the outside of the scroll if pivot point P is on the underside of the scroll member, but preferably it will be outside of the outer wrap as shown in Figures 1, 4, and 13. Since the end of the outer wrap makes a convenient reference point, for purposes of demonstration pivot point P will be located there, although it could be at any location around the scroll periphery. It should be appreciated that length L represents the link formed by the orbiting scroll between the center cc and pivot point P . The line between cc and cs is referred to as the "line of centers".

As described above, it has been discovered that there are a number of basic linkage mechanisms which will accomplish the objective of permitting only a small amount of controlled rotation between scrolls. Because of ease of fabrication, low cost, and easy lubrication, the four-bar linkage is the preferred approach. If length L is sufficiently long and the link is substantially perpendicular to the line extending between P and cc in Figure 21, the four-bar linkage motion becomes similar to the crank and slider embodiment of Figure 16 since point P moves in essentially a straight line and length L does not change. This embodiment is therefore the easiest to analyze. The other mechanisms disclosed will follow a similar analysis with similar results.

The basic problem, looking at Figure 21, is that the orbiting scroll will rotate through the angle Δm as the eccentric moves the center of the orbiting scroll around the circle of the orbit with eccentricity E . These angular excursions result in a mismatch of

scroll surfaces causing a simultaneous gap and interference condition to occur at the seal points between the scroll flanks, resulting in low capacity and efficiency. The amount of gap and interference depends upon the amount of mismatch between the scroll surfaces, which in turn depends upon the link geometry and the scroll configuration.

- 5 Assuming P moves radially on a straight line through the center of the generating circle of the fixed scroll, the value of A_m is calculated as follows:

$$\sin(A_m) = \frac{E \sin(V_{Tm})}{L}$$

- 10 For small values of A_m (which will be the case here) it will be assumed:

$$A_m = \sin(A_m) = \frac{E \sin(V_{Tm})}{L} \text{ (radians)}$$

Therefore, because the maximum angle the orbiting scroll rotates with respect to the fixed scroll is at $V_{Tm} = 90^\circ$:

$$A_{m \max} = E/L \text{ (radians)}$$

- 15 Looking at Figure 21, when the line of centers intersects position a, the value of A_m is 0° . Thus, there is zero relative angle between the scrolls at that point, which means there is no scroll surface mismatch at that position. However, as the line of centers moves to point b, the value of A_m increases to the value of $A_{m \max}$, resulting in the maximum
20 amount of surface mismatch. If one requires that the sealing points with the present linkage arrangement be at the same points as with an Oldham coupling (i.e. with no relative rotation permitted), then the profiles must be modified to permit this. What has been observed is that at positions a and c, there is 0° relative angle between the
25 scrolls and thus no radial correction required. At positions b and d, where the relative angle between the scrolls is maximum, the radial correction is maximum and equal to:

(45)

$$R_{\max} = [(E/L)/2(\pi)]^2(\pi)C = EC/L$$

In between the points of zero correction and maximum correction, the radial correction will have some intermediate correction approximately equal to:

5 $(EC/L)\sin(V\theta_m)$

Thus, when the uncorrected scrolls are at the position requiring maximum correction, one set of seal points has a gap requiring a correction to close up, whereas the other set of seal points actually has interference. It has been discovered that these gaps and
10 interferences can be corrected by appropriately shifting the involute profiles in a direction normal to a line through the scroll centers and pivot point P when the scrolls are in their unrotated position.

If the correction or shifting is to be applied to only one scroll wrap (Case Nos. 1 and 2) then the inner wrap profile is shifted in one
15 direction along the normal line an amount equal to $R_{\max}$ in order to close the gap and eliminate interference, and the outer wrap profile is shifted the same amount in the opposite direction for the same reason.

When all four flank profiles are corrected, (Case No. 3), then a similar correction is made to both scroll wraps. Specifically, the
20 inner profile of one scroll wrap is shifted in one direction along the normal line to close up the gap and eliminate interference, and the outer profile of the same wrap is shifted in the other direction for the same reason. The inner profile of the second scroll wrap is also shifted, but in the opposite direction as the inner profile on the first
25 scroll wrap, and the outer profile is shifted in the opposite direction as the outer profile on the first scroll wrap. The amount of correction to be applied to each of the wrap flanks under Case No. 3 is as follows:

Fixed Scroll Wrap

Orbiting Scroll Wrap

<u>Outer Flank</u>	<u>Inner Flank</u>	<u>Outer Flank</u>	<u>Inner Flank</u>
$(X)(R_{max})$	$(Y)(R_{max})$	$(1-Y)(R_{max})$	$(1-X)(R_{max})$

where X is the proportion (%/100) of correction desired on the outer flank of the fixed scroll wrap and Y is the proportion of correction desired on the inner flank of the fixed scroll wrap. Thus, the correction to the outer flank of the fixed scroll plus the correction to the inner flank of the orbiting scroll will total R max, as will the combined correction to the inner flank of the fixed scroll and the outer flank of the orbiting scroll.

The result is one or a pair of scrolls having a wrap which is no longer of uniform thickness. Observe in Figure 19 that fixed scroll wrap 252, to which all correction has been applied, is the same thickness wherever it intersects the x axis, is of reduced thickness wherever it intersects the y axis below the x axis, and is of increased thickness wherever it intersects the y axis above the x axis. Thus, as one continues inwardly along a corrected scroll, the thickness changes continually, having a maximum, minimum, or "nominal" value approximately every 90°. The result is that the profiles have a shape and position during operation such that they come extremely close to sealing at all the required points at the required times.

As noted, the involute shifting method does produce a small flank clearance or interference. The magnitude of this clearance or interference depends not only on crank angle and basic scroll geometry, but also on the ratio K, which is the ratio of modification added to the fixed scroll divided by the total modification. Therefore, assuming both scrolls are modified the same, if the total modification is added to the fixed scroll, K = 1; whereas, if the total modification is added to the moving scroll, K = 0. Satisfactorily operating compressors have

been made and tested having $K = 0.5$, which was originally believed to be optimum. Subsequent analysis, however, surprisingly revealed that the smallest error is encountered when $K = 1$, whereas the largest error occurs when $K = 0$. There is a nearly linear relationship between the two so the best value for K is 1. K may also have values greater than unity or smaller than zero, but these values have no known advantages.

Furthermore, as noted above, the same amount of correction need not be applied to each scroll. In the case of unequal correction there will be different values of K for different flank interfaces in the same machine. As used herein the ratio K_1 represents the K ratio for the fixed outer and orbiting inner flanks, and K_2 the K ratio for the fixed inner and orbiting outer flanks. Thus, the optimum situation (all correction on the fixed scroll) will occur when $K_1 = 1$ and $K_2 = 1$, which is the same as having $K = 1$.

Overall, the potential theoretical errors in the shifting method come from three different causes:

- (1) The sine error; due to the approximation $A = \sin(A)$.
- (2) The link arc error. When the link length is sufficiently long this error becomes very small; practical packaging, however, usually restricts this length. This error can be completely eliminated by using a crank and slider.
- (3) The choice of the surface to correct. By placing the shift entirely on the fixed scroll, this error is zero. By placing the shift entirely on the moving scroll, this error has its maximum value. Since there is no known advantage to placing it on the moving scroll, it should be placed on the fixed scroll.

It is believed, however, that in practice all these errors are sufficiently negligible that they will not adversely effect compressor

performance.

Involute Modification Technique

Figure 22 shows a flank contact between the outer fixed and inner orbiting flanks, and Figure 23 shows a flank contact between the outer orbiting and inner fixed flanks. These points are defined for any given geometric parameters (C,T,L) and compressor crank position (Ac). The additional material that must be added to or subtracted from the involute profile for continuous contact between scrolls has been determined to be in accordance with the following:

$$R = C(1 + \text{Doc}^2)^{1/2}$$

10	$\text{Doc} = \text{Ar} + \text{K1}(\text{Am})$	Fixed Outer Scroll Surface
	$\text{Doc} = \text{Ar} + (\text{K2} - 1)\text{Am}$	Moving Outer Scroll Surface
	$\text{Doc} = \text{Ar} + \text{K2}(\text{Am}) - \text{T/C}$	Fixed Inner Scroll Surface
	$\text{Doc} = \text{Ar} + (\text{K1} - 1)\text{Am} - \text{T/C}$	Moving Inner Scroll Surface
	$\text{Ar} = \text{Ac} + \text{B} + 5 (\text{Pi})/2$	Fixed Outer Scroll Surface
15	$\text{Ar} = \text{Ac} + \text{B} + 5 (\text{Pi})/2 + \text{Am}$	Moving Outer Scroll Surface
	$\text{Ar} = \text{Ac} + \text{B} + 3(\text{Pi})/2$	Fixed Inner Scroll Surface
	$\text{Ar} = \text{Ac} + \text{B} + 3(\text{Pi})/2 + \text{Am}$	Moving Inner Scroll Surface
	$\text{Ap} = \text{Ar} - \arctan(\text{Doc})$	

For restraining link (4-bar):

$$\text{Am} = \text{B} - \text{Bb}$$

$$\text{Bb} = \text{[see calculations in steps 3-5 of the Algorithm set forth below]}$$

For restraining slider (crank and slider with pin on scroll):

$$\text{Ap} = \arcsin([E/L]\sin(\text{Ac}))$$

$$\text{K1} = \text{Modification amount added to fixed scroll outer flank divided by total modification between fixed outer and moving inner flanks.}$$

K2 = Modification amount added to fixed scroll inner flank divided by total modification between fixed inner and moving outer flanks.

Doc = Dummy variable

5 Ac = "crank angle", although it is better thought of as a mapping increment angle.

B = angle from start of involute on orbiting base circle to the pivot pin center (polar).

C = radius of base generating circle

T = thickness of scroll wrap

10 Increment Ac beginning with Ac such that Ar = 0.

Nw = number of wraps = (Ar to end of scroll - Pi/2) / 2(Pi)

Stop when Ac = 2 (Pi)(Nw) + Pi/2

In cartesian coordinates:

$$X = C(1 + Doc^2)^{1/2} \cos(Ar - \arctan(Doc))$$

15
$$Y = C(1 + Doc^2)^{1/2} \sin(Ar - \arctan(Doc))$$

In numerically controlled milling machines, the cartesian equations must be programmed to give position coordinates. One possible Algorithm is as follows:

1. If involute profile = inner surface then $Ac = - B - 3(Pi)/2 + Au$
If involute profile = outer surface then $Ac = - B - 5(Pi)/2 + Au$

- 20 2. If restraining slider (crank and slider with pin on scroll) is used then:

$$Am = \arcsin[(E/L)\sin(Ac)]$$

Go to step 7

- 25 3. If restraining link (4-bar linkage) is used then

$$Xa = (E)\cos(Ac + B + Pi)$$

$$Ya = (E)\sin(Ac + B + Pi)$$

$$Xgb = Xa + (L)\cos(B + Pi)$$

X_c = x coordinate of the center of projection 124 (Figure 4)

Y_c = y coordinate of the center of projection 124 (Figure 4)

$$A_z = ((X_c - X_a)^2 + (Y_c - Y_a)^2)^{.5}$$

$$Y_u = (X_a - X_c)/A_z$$

5 $C_c = \arctan(Y_u)$

$$F_{11} = F_{12} = 1$$

If $(Y_a - Y_c) > 0$ then go to step 4

$$C_c = 2(\pi) - C_c$$

$$F_{11} = -1$$

104. $X_{um} = X_c + L_{ik}(\cos(C_c))$

If $X_{gb} - X_{um} < 0$ then go to step 5

$$F_{12} = -1$$

5. $Y_u = (A_z^2 + L_{ik}^2 - L^2) / 2(A_z)(L_{ik})$

$$C_b = (F_{11})(F_{12})\arctan(Y_u)$$

15 $X_b = X_c + L_{ik}(\cos(C_c + C_b))$

$$Y_b = Y_c + L_{ik}(\sin(C_c + C_b))$$

$$X_{gb} = X_b$$

$$B_b = \arctan((X_a - X_b)/L)$$

If $Y_a - Y_b > 0$ then go to step 6

20 $B_b = 2(\pi) - B_b$

6. $A_m = B - B_b$

7. $A_r = A_c + B + 5(\pi)/2 - \pi(L_1) + A_m(L_2)$

$$D_{oc} = A_r + K(A_m)(L_3) + (K-1)(A_m)(L_2) - (T/C)(L_1)$$

$$R = C(1 + D_{oc}^2)^{1/2}$$

25 $p = A_r - \arctan(D_{oc})$

$$X = ((R)\cos(p) + [(T_1)\sin(A_r)]TT)L_4$$

$$Y = (R)\sin(p) - [(T_1)\cos(A_r)]TT$$

8. $A_c = A_c + A_i$

9. If $Ar = 2(p_1)(Nw) + P_1/2$ then go to step 2

10. The procedure is complete

NOMENCLATURE

E = Eccentricity = $C(P_1) - T$

5 C = Generating circle radius

L = Distance between geometric center of orbiting scroll and link pin center P

Ac = Mapping angle

10 K1 = Modification amount added to fixed scroll outer flank divided by total modification between fixed outer and moving inner flanks

K2 = Modification amount added to fixed scroll inner flank divided by total modification between fixed inner and moving outer flanks

R = Polar vector magnitude

Ar = Roll angle

15 Aro = Roll angle (Ar) on moving (orbiting) scroll

Arf = Roll angle (Ar) on fixed scroll

B = Angle from start of involute on orbiting base circle to the pivot pin center (polar).

T = Scroll wrap thickness

20 Ai = Increment angle (i.e., the angle by which AC is incremented)

25 Au = Truncation angle in Figure 20B (the angle from the x axis to a line perpendicular to a line tangent to the base generating circle and passing through the points of tangency of the physical inner end of a wrap and the involute curves from which its flanks were generated)

Tl = Tool radius

D = Distance between base circle center and anchor pin center

Lik = Length of rotation controlling link (4-bar)

LOGIC COEFFICIENTS:

30 Surface:	L1	L2	L3	L4	TI
Fixed inner	1	0	1	-1	-1

Fixed outer	0	0	1	-1	1
Moving inner	1	1	0	1	-1
Moving outer	0	1	0	1	1

Method of Manufacture

5 The present invention includes a very simple and unique method of manufacturing the fixed and orbiting scrolls on a N.C. or similar machine utilizing the aforesaid shifting technique. This method is schematically illustrated in Figure 24, where 300 is a workpiece to
10 be formed into a scroll wrap, 302 is a fixture supporting workpiece 300 while it is being machined by a cutter 304, which could be an end mill. The machine which positions cutter 304 with respect to workpiece 300 is programmed in the conventional manner to form the inner and outer flank profiles (identical curves) from the common
15 base generating circle of the involute scroll. Thus, if the workpiece is fixed in one position on the fixture for the entire flank machining operation, a conventional scroll wrap of uniform thickness will be generated, with both flanks being involutes of the same base circle.

20 In the practice of the present invention, however, the outer flank of the wrap is machined when the workpiece is in the position shown in which it is disposed against stop 308, and the inner flank of the wrap is machined when the workpiece is disposed against stop 306, no other changes being made to the orientation of the workpiece.
25 The distance between stops 306 and 308 is equal to the total amount of shifting of the involute base generating circle which is desired, calculated in accordance with the criteria set forth with respect to the aforesaid shifting method, plus the overall dimension of the workpiece between the stops; i.e. the distance the workpiece moves
30 from stop 308 to 306 is equal to the desired displacement of the base

53

generating circles. The distribution of the amount of correction applied to each flank is in turn determined by the base position (reference point) of cutting tool 304 with respect to stops 306 and 308. If the base position is centered, then an equal amount of
5 correction will be applied to each flank.

C L A I M S

1. A scroll-type machine including: a first scroll member having a first spiral wrap; a second scroll member having a second spiral wrap and being mounted for movement with respect to said first scroll member; said
5 second wrap being intermeshed with said first wrap with flank sealing between the wraps so that when said second wrap is moved with respect to said first wrap along a predetermined path fluid pockets of progressively changing volume are formed; and means for causing said second wrap
10 to move along said predetermined path, including drive means for causing a first point on said second scroll member to move in a generally circular orbital path with respect to said first scroll member, and rotation controlling means for restricting but not preventing
15 rotational movement of said second scroll member relative to the first scroll member by limiting movement of a second point thereon.



55

2. A scroll-type machine as claimed in claim 1, wherein movement of said second point is limited to a substantially straight line path with respect to said first scroll member.

3. A scroll-type machine as claimed in claim 2, wherein said
5 straight line path extends generally radially with regard to said second spiral wrap.

4. A scroll-type machine as claimed in claim 2, wherein said straight line path is slightly arcuate.

5. A scroll-type machine as claimed in claim 2, wherein said
10 straight line path is a true straight line.

6. A scroll-type machine as claimed in any one of claims 1-5, wherein said first point is disposed at the center of said second spiral wrap.

7. A scroll-type machine as claimed in any one of claims 1-6,
15 wherein said second point is disposed radially outwardly of said second spiral wrap.

8. A scroll-type machine as claimed in any one of claims 1-7, wherein each of said spiral wraps has an inner and outer flank having a profile which is the involute of a circle.



* C L M S *

9. A scroll-type machine as claimed in any one of claims 1-7, wherein each of said spiral wraps has an inner and outer flank having a profile which is the involute of a plane geometric shape.

10. A scroll-type machine as claimed in claim 9, wherein the center of the generating shape of the involute profile of the outer flank of both of said wraps is displaced from the center of the generating shape of the involute profile of the inner flank of the same wrap.

11. A scroll-type machine as claimed in claim 9, wherein the center of the generating shape of the involute profile of the outer flank of one of said wraps is displaced from the center of the generating shape of the involute profile of the inner flank of the same wrap.



12. A scroll-type machine as claimed in claim 11, wherein said rotation controlling means is connected to said second scroll member at substantially a single point and said centers are displaced from one another substantially along a line normal to a line extending between said single point and the center of said second wrap when said scroll members are not rotated relative to one another.

13. A scroll-type machine as claimed in claim 12, wherein said geometric shapes are circles of equal radius, said centers being displaced from one another on each said wrap by an amount equal to EC/L , where:

E = the radius of orbit;

C = the radius of said base generating circle

L = the distance from the geometric center of said second wrap to said second point.

14. A scroll-type machine as claimed in claim 4 or any of claims 6 to 13 when appendant thereto wherein said rotation controlling means comprises a link having one portion pivotally connected to said second scroll member and another portion mounted for rotation about an axis which is fixed with respect to said first scroll member.





15. A scroll-type machine as claimed in claim 14, wherein the axis of said link extending between said link portions is generally perpendicular to a radius of said second wrap passing through said pivotal connection of said link to said second scroll member.

5 16. A scroll-type machine as claimed in claims 14 or 15, wherein pivotal connection of said link to said second scroll member is disposed radially outwardly of said second wrap.

17. A scroll-type machine as claimed in claims 14, 15 or 16, wherein said other portion of said link is mounted to said first scroll
10 member.

18. A scroll-type machine as claimed in claims 14, 15 or 16, wherein said machine is disposed within a hermetic shell, and said other portion of said link is mounted to said shell.





19. A scroll-type machine as claimed in any one of the preceding claims wherein said rotation controlling means comprises a four-bar linkage interconnecting said moving scroll member and a fixed support structure.

5 20. A scroll-type machine as claimed in claim 1

wherein said rotation controlling means comprises a crank and slider mechanism interconnecting said moving scroll member and a fixed support structure.

21. A scroll-type machine as claimed in any one of the preceding
10 claim, further comprising a crankshaft driven for rotation about a first axis, said crankshaft having a generally flat driving surface lying in a plane perpendicular to a radius of and spaced from said first axis, a drive element having a generally flat driven surface in driving engagement with said driving surface, and driven means on said
15 second scroll member pivotally connected to said drive element for rotation with respect thereto about a second axis spaced from said first axis, whereby rotation of said crankshaft causes said second axis on said second scroll member to orbit relative to said first scroll member about said first axis.



* C L H S *



22. A scroll-type machine as claimed in claim 21, wherein the plane of said driving surface is substantially parallel to a plane passing through said first and second axes.

23. A scroll-type machine as claimed in claims 21 or 22, wherein
5 said drive element is free to slide on said driving surface in the event an obstruction temporarily prevents said second scroll member from following its normal course of movement.

24. A scroll-type machine as claimed in claims 21, 22 or 23, wherein said drive element is generally annular in cross-section.

10 25. A scroll-type machine as claimed in claims 21, 22, 23 or 24, wherein said drive element is generally cylindrical and said driven surface is disposed on the outer periphery thereof.



(51)

26. A scroll-type machine as claimed in any preceding claim wherein:

said first scroll member has a first flat sealing surface and a said spiral wrap has a first flat tip surface,

5 said second scroll member has a second flat sealing surface and said second spiral wrap has a second flat tip surface,

said second scroll member being mounted for movement with respect to said first scroll member,

10 said second wrap being intermeshed with said first wrap so that when said second wrap is moved with respect to said first wrap along a predetermined path, the inner and outer flanks of a substantial portion of each said wraps engage one another to form fluid pockets of progressively changing volume,

15 said first tip surface sealingly engaging said second sealing surface and said second tip surface sealingly engaging said first sealing surface to seal said pockets,

20 each of said wraps being arranged so that the outside flank on the outer terminal end portion thereof does not engage the flank of the other wrap, said terminal end portion of each said wrap being substantially thicker in the radial direction than the remainder thereof in



order to bear a disproportionately large share of the axial loads on said scroll members.



63

27. A scroll-type machine as claimed in claim 26, further comprising an oil supply groove in one of said tip surfaces, and means for supplying oil thereto in order to reduce friction.

28. A scroll-type machine as claimed in claims 26 or 27, further
5 comprising means for supplying lubricating oil to at least one of said sealing surfaces in an area where it is engaged by a tip surface.

29. A scroll-type machine as claimed in claims 26, 27 or 28 wherein at least one of said terminal end portions extends for approximately 180°.



* C L N S *

30. A scroll-type machine as claimed in any of the preceding claims, further comprising:

a fixed support structure for

supporting said first scroll member;

5 a thrust bearing operatively disposed between said fixed support structure and said second scroll member,

said thrust bearing being generally annular in plan and having a parallel flat uninterrupted bearing surface on the face thereof facing said fixed support structures; and

10

lubricating means for supplying oil to the interface between said thrust bearing and said fixed support structure to create an oil film therebetween,

15

the resultant miniscule wobbling of said second scroll member as it orbits with respect to said first scroll member causing a squeeze film phenomenon between said thrust bearing and said fixed support structure.

31. A scroll-type machine as claimed in claim 30, wherein said thrust bearing is fixed to said second scroll member.



* C L M S *

65

32. A scroll-type machine as claimed in any of the preceding claims, further comprising:

5 a crankshaft driven for rotation about a first axis, said crankshaft having a cylindrical driving surface the axis of curvature of which is parallel to said first axis;

a drive element having a cylindrical driven surface in driving engagement with said driving surface; and

10 driven means on said second scroll member pivotally connected to said drive element for rotation with respect thereto about a third axis spaced from and parallel to said first axis, whereby rotation of said crankshaft causes said third axis on said second scroll member to orbit relative to said first scroll member about said first axis.



33. A scroll-type machine as claimed in claim 32, wherein said driving surface axis of curvature is spaced from said first axis.

34. A scroll-type machine as claimed in claims 32 or 33, wherein said driving surface axis of curvature is spaced from said third axis.

5 35. A scroll-type machine as claimed in claims 32, 33 or 34, wherein said driving and driven surfaces have a common axis of curvature.

36. A scroll-type machine as claimed in claims 32, 33, 34 or 35, wherein said first axis is spaced from said axis of curvature a greater
10 distance than from said third axis.

37. A scroll-type machine as claimed in claims 32, 33, 34, 35 or 36, wherein said axis of curvature is spaced from the plane of said first and third axes.

38. A scroll-type machine as claimed in claims 32, 33, 34, 35, 36
15 or 37 wherein said drive element is free to rotate with respect to said crankshaft about said axis of curvature in the event an obstruction temporarily prevents said second scroll member from following its normal course of movement.

39. A scroll-type machine as claimed in claims 32-37 or 38,
20 wherein said drive element is generally annular in cross-section.



67

40. A scroll-type machine as claimed in any of the preceding claims, wherein:

5 the profiles of the inner and outer flanks of said first wrap are each the involute of the same generating plane geometric shape,

10 the center of said generating shape of the profile of said outer flank of said first wrap being displaced from the center of said generating shape of the profile of said inner flank of said first wrap.



* C L M S *

41. A scroll-type machine as claimed in any one of claims 1 to 40 wherein a portion of said second scroll member is adapted to orbit with respect to said first scroll member with an eccentricity E, said second scroll member having rotation controlling means fixed thereto at a point P, each of said scroll members including a scroll wrap having inner and outer surfaces based on a generating circle and defined by the following equations (cartesian coordinates):

$$10 \quad \begin{aligned} X &= C(1 + \text{Doc}^2)^{1/2} \cos(\text{Ar} - \arctan(\text{Doc})) \\ Y &= C(1 + \text{Doc}^2)^{1/2} \sin(\text{Ar} - \arctan(\text{Doc})) \end{aligned}$$

where:

$$R = C(1 + \text{Doc}^2)^{1/2}$$

$$\text{Doc} = \text{Ar} + K1(\text{Am}) \quad \text{First Outer Scroll Surface}$$

$$15 \quad \text{Doc} = \text{Ar} + (K2 - 1)\text{Am} \quad \text{Second Outer Scroll Surface}$$

$$\text{Doc} = \text{Ar} + K2(\text{Am}) - T/C \quad \text{First Inner Scroll Surface}$$

$$\text{Doc} = \text{Ar} + (K1 - 1)\text{Am} - T/C \quad \text{Second Inner Scroll Surface}$$

$$\text{Ar} = \text{Ac} + B + 5(Pi)/2 \quad \text{First Outer Scroll Surface}$$

$$\text{Ar} = \text{Ac} + B + 5(Pi)/2 + \text{Am} \quad \text{Second Outer Scroll Surface}$$

$$20 \quad \text{Ar} = \text{Ac} + B + 3(Pi)/2 \quad \text{First Inner Scroll Surface}$$

$$\text{Ar} = \text{Ac} + B + 3(Pi)/2 + \text{Am} \quad \text{Second Inner Scroll Surface}$$

$$\text{Ap} = \text{Ar} - \arctan(\text{Doc})$$

$$\text{Am} = \arcsin([E/L]\sin(\text{Ac}))$$

25 $K1$ = Modification amount added to first scroll outer flank divided by total modification between first outer and second inner flanks.



* C L M S *

69

- 50 -

K2 = Modification amount added to first scroll inner flank divided by total modification between first inner and second outer flanks.

Doc = Dummy variable

5 Ac = mapping increment angle.

B = angle from start of involute on orbiting base circle to the pivot pin center (polar)

C = radius of base generating circle

T = thickness of scroll wrap

10 E = $C(\text{Pi}) - T$

L = Distance between geometric center of orbiting scroll and link pin center P

Ac being incremented beginning with Ac such that $A_r = 0$.

Nw = number of wraps = $(A_r \text{ to end of scroll} - \text{Pi}/2) / 2(\text{Pi})$

15 Stopping when $A_c = 2 (\text{Pi})(N_w) + \text{Pi}/2$.

42. A machine as claimed in any one of claims 1 to 40 wherein a portion of said second scroll member adapted to orbit with respect to said first scroll member with an eccentricity E, said second scroll member having rotation controlling link
20 fixed thereto at a point P, each of said scroll members including a scroll wrap having inner and outer surfaces based on a generating circle and being of a contour formed by an NC-type machine operating in accordance with the following routine:





- 51 -

1. If involute profile = inner surface then $A_c = -B - 3(\pi)/2 + A_u$
If involute profile = outer surface then $A_c = -B - 5(\pi)/2 + A_u$
 - 5 2. $A_m = \arcsin[(E/L)\sin(A_c)]$
 3. $A_r = A_c + B + 5(\pi)/2 - \pi(L_1) + A_m(L_2)$
 $Doc = A_r + K(A_m)(L_3) + (K-1)(A_m)(L_2) - (T/C)(L_1)$
 $R = C(1 + Doc^2)^{1/2}$
 $p = A_r - \arctan(Doc)$
 - 10 $X = ((R)\cos(p) + [(T_1)\sin(A_r)]TT)L_4$
 $Y = (R)\sin(p) - [(T_1)\cos(A_r)]TT$
 4. $A_c = A_c + A_i$
 5. If $A_r = 2(\pi)(N_w) + \pi/2$ then go to step 2
 6. The procedure is complete
- 15 where:
- $E = \text{Eccentricity} = C(\pi) - T$
- $C = \text{Generating circle radius}$
- $L = \text{Distance between geometric center of orbiting scroll and link pin center P}$
- 20 $A_c = \text{Mapping angle}$
- $K_1 = \text{Modification amount added to first scroll outer flank divided by total modification between first outer and second inner flanks.}$
- 25 $K_2 = \text{Modification amount added to first scroll inner flank divided by total modification between first inner and second outer flanks.}$
- $R = \text{Polar vector magnitude}$
- $A_r = \text{Roll angle}$



Arf = Roll angle (Ar) on first scroll

5 T = Scroll wrap thickness

10 Au = Truncation angle the angle from the X axis to a line perpendicular to a line tangent to the base generating circle and passing through the points of tangency of the physical inner end of a wrap and the involute curves from which its flanks were generated.

15 D = Distance between base circle center and anchor pin center

Surface:	L1	L2	L3	L4	TT
First inner	1	0	1	-1	-1
First outer	0	0	1	-1	1
Second inner	1	1	0	1	-1
Second outer	0	1	0	1	1

43. A machine as claimed in claim 41 or 42, further comprising a crank and slider linkage operatively connected between said first scroll member and said second scroll member to restrict relative rotation of said scroll members to a limited predetermined amount.





44. A scroll-type machine as claimed in any one of claims 1 to 40, wherein a portion of said second scroll member is adapted to orbit with respect to said first scroll member with an eccentricity E, said second scroll member having rotation controlling means fixed thereto at a point P, each of said scroll members including a scroll wrap having inner and outer surfaces based on a generating circle and defined by the following equations (cartesian coordinates):

$$10 \quad \begin{aligned} X &= C(1 + Doc^2)^{1/2} \cos(Ar - \arctan(Doc)) \\ Y &= C(1 + Doc^2)^{1/2} \sin(Ar - \arctan(Doc)) \end{aligned}$$

where:

$$\begin{aligned} R &= C(1 + Doc^2)^{1/2} \\ Doc &= Ar + K1(Am) && \text{First outer scroll surface} \\ 15 \quad Doc &= Ar + (K2 - 1)(Am) && \text{Second outer scroll surface} \\ Doc &= Ar + K2(Am) - T/C && \text{First inner scroll surface} \\ Doc &= Ar + (K1 - 1)Am - T/C && \text{Second Inner scroll surface} \\ Ar &= Ac + B + 5(Pi)/2 && \text{First outer scroll surface} \\ Ar &= Ac + B + 5(Pi)/2 + Am && \text{Second outer scroll surface} \\ 20 \quad Ar &= Ac + B + 3(Pi)/2 && \text{First inner scroll surface} \\ Ar &= Ac + B + 3(Pi)/2 + Am && \text{Second inner scroll surface} \\ Ap &= Ar - \arctan(Doc) \end{aligned}$$

to calculate Am:

$$\begin{aligned} (1) \quad Xa &= (E)\cos(Ac + B + Pi) \\ 25 \quad Ya &= (E)\sin(Ac + B + Pi) \\ Xgb &= Xa + (L)\cos(B + Pi) \end{aligned}$$



73

- 54 -

Xc = x coordinate of point P

Yc = y coordinate of point P

$Az = ((Xc - Xa)^2 + (Yc - Ya)^2)^{.5}$

$Yu = (Xa - Xc)/Az$

5 Cc = arctan(Yu)

F11 = F12 = 1

If (Ya - Yc) > 0 then go to step 2

Cc = 2(pi) - Cc

F11 = -1

10 (2) Xum = Xc + Lik(cos(Cc))

If Xgb - Xum < 0 then go to step 3

F12 0 -1

(3) $Yu = (Az^2 + Lik^2 - L^2) / 2(Az)(Lik)$

Cb = (F11)(F12)arctan(Yu)

15 Xb = Xc + Lik(cos(Cc + Cb))

Yb = Yc + Lik(sin(Cc + Cb))

Xgb = Xb

Bb = arctan((Xa - Xb)/L)

If Ya - Yb > 0 then go to step 4

20 Bb = 2(pi) - Bb

(4) Am = B - Bb

K1 = Modification amount added to first scroll outer flank
divided by total modification between first outer and
second inner flanks.

25 K2 = Modification amount added to first scroll inner flank
divided by total modification between first inner and
second outer flanks.

Doc = Dummy variable



Ac = Mapping increment angle.

B = angle from start of involute on orbiting base circle to the pivot pin center (polar).

C = radius of base generating circle

5 T = thickness of scroll wrap

E = $C(P_i) - T$

L = Distance between geometric center of orbiting scroll and link pin center P

Ac being incremented beginning with Ac such that Ar = 0.

10 Nw = number of wraps = $(Ar \text{ to end of scroll} - P_i/2) / 2(P_i)$

Stopping when $Ac = 2(P_i)(Nw) + P_i/2$.

45. A machine according to any one of claims 1 to 40, wherein a portion of said second scroll member is adapted to orbit with respect to said first scroll member with an
15 eccentricity E, said second scroll member having rotation controlling link affixed thereto at a point P, each of said scroll members including a scroll wrap having inner and outer surfaces based on a generating circle and being of a contour formed by an NC-type machine operating in accordance
20 with the following routine:

1. If involute profile = inner surface then $Ac = -B -$

$3(P_i)/2 + Au$

If involute profile = outer surface then $Ac = -B -$

$5(P_i)/2 + Au$

25 2. $Xa = (E)\cos(Ac + B + P_i)$

$Ya = (E)\sin(Ac + B + P_i)$



78

$X_{gb} = X_a + (L)\cos(B + \pi)$
 $X_c = x \text{ coordinate of point P}$
 $Y_c = y \text{ coordinate of point P}$
 $A_z = ((X_c - X_a)^2 + (Y_c - Y_a)^2)^{.5}$
5 $Y_u = (X_a - X_c)/A_z$
 $C_c = \arctan(Y_u)$
 $F_{l1} = F_{l2} = 1$
 If $(Y_a - Y_c) > 0$ then go to step 3
 $C_c = 2(\pi) - C_c$
10 $F_{l1} = -1$
3. $X_{um} = X_c + L_{ik}(\cos(C_c))$
 If $X_{gb} - X_{um} > 0$ then go to step 4
 $F_{l2} = -1$
4. $Y_u = (A_z^2 + L_{ik}^2 - L^2) / 2(A_z)(L_{ik})$
15 $C_b = (F_{l1})(F_{l2})\arctan(Y_u)$
 $X_b = X_c + L_{ik}(\cos(C_c + C_b))$
 $Y_b = Y_c + L_{ik}(\sin(C_c + C_b))$
 $X_{gb} = X_b$
 $B_b = \arctan((X_a - X_b)/L)$
20 If $Y_a - Y_b > 0$ then go to step 5
 $B_b = 2(\pi) - B_b$
5. $A_m = B - B_b$
6. $A_r = A_c + B + 5(\pi)/2 - \pi(L_1) + A_m(L_2)$
 $Doc = A_r + K(A_m)(L_3) + (K-1)(A_m)(L_2) - (T/C)(L_1)$
25 $R = C(1 + Doc^2)^{1/2}$
 $p = A_r - \arctan(Doc)$



$$X = ((R)\cos(p) + |(T1)\sin(Ar)|TT)L4$$

$$Y = (R)\sin(p) - |(T1)\cos(Ar)|TT$$

7. $Ac = Ac + Ai$

8. If $Ar = 2(pi)(Nw) + Pi/2$ then go to step 2

5 9. The procedure is complete

where:

$E = \text{Eccentricity} = C(Pi) - T$

$C = \text{Generating circle radius}$

10 $L = \text{Distance between geometric center of orbiting scroll and link pin center P}$

$Ac = \text{Mapping angle}$

$K1 = \text{Modification amount added to first scroll outer flank divided by total modification between first outer and second inner flanks.}$

15 $K2 = \text{Modification amount added to first scroll inner flank divided by total modification between first inner and second outer flanks.}$

$R = \text{Polar vector magnitude}$

$Ar = \text{Roll angle}$

20 $Aro = \text{Roll angle (Ar) on moving (orbiting) scroll}$

$Arf = \text{Roll angle (Ar) on fixed scroll}$

$B = \text{Angle from start of involute on orbiting base circle to the pivot pin center (polar).}$

$T = \text{Scroll wrap thickness}$

25 $Ai = \text{Increment angle (i.e., the angle by which AC is incremented)}$

30 $Au = \text{Truncation angle (the angle from the X axis to a line perpendicular to a line tangent to the base generating circle and passing through the points of tangency of the physical inner end of a wrap and the involute curves from which its flanks were generated).}$



77

T1 = Tool radius

D = Distance between base circle center and anchor pin center

Lik = Length of rotation controlling link

5 where logic coefficients are:

Surface:	L1	L2	L3	L4	TT
First inner	1	0	1	-1	-1
First outer	0	0	1	-1	1
Second inner	1	1	0	1	-1
10 Second outer	0	1	0	1	1

46. A machine ~~scroll~~ as claimed in claim 44 or 45, further comprising a four-bar linkage operatively connected between said first scroll member and said second scroll member to restrict relative rotation of said scroll members
15 to a limited predetermined amount.

47. A method of manufacturing a scroll-type machine as claimed in any one of claims 41 to 46, including the steps of machining the inner and outer scroll-shaped flanks of a scroll from a workpiece using a programmable machine, wherein the profile of each
20 flank is generated from the same plane geometric shape, said working steps comprising: (1) programming the machine so that it will machine the inner and outer flanks of said scroll based on said generating shape being at a fixed point with respect to said workpiece; (2) determining the
25 reference point where said workpiece should be located relative to the cutting tool of said machine to machine



said scroll flanks in accordance with said program;

(3) positioning said workpiece relative to said cutting tool in a first position spaced from said reference point and machining said workpiece to form one of said flanks of said scroll; and (4) positioning said workpiece relative to said cutting tool in a second position and machining said workpiece to form the other flank of said scroll; steps (3) and (4) following steps (1) and (2) and being in any order.

48. A method of manufacturing a scroll type machine as claimed in any one of claims 41 to 46, including the steps of machining the inner and outer flanks of a scroll as claimed in claim 47, wherein said second position is spaced from said first position.

49. A method of manufacturing a scroll type machine as claimed in any one of claims 41 to 46, including the steps of machining the inner and outer flanks of a scroll as claimed in claim 47, wherein said first position is spaced in one direction from said reference point and said second position is spaced from said reference point in the opposite direction.

50. A method of manufacturing a scroll type machine as claimed in any one of claims 41 to 46, including the steps of machining the inner and outer flanks of a scroll as claimed in claim 47, 48 or 49, wherein said first and second positions are substantially equidistant from said reference point.

51. A method of manufacturing a scroll type machine as claimed in any one of claims 41 to 46, including the steps of machining the inner and outer flanks of a scroll as claimed in claim 47, wherein said second position is at said reference point.

52. A method of manufacturing a scroll type machine as claimed in any one of claims 41 to 46, including the steps of machining the inner and outer flanks of a scroll as claimed in claim 47, 48, 49, 50 or 51,



79

- 60 -

wherein said geometric shape is a circle.

53. A method of manufacturing a scroll type machine as claimed in any one of claims 41 to 46, including the steps of machining the inner and outer flanks of a scroll as claimed in claim 47, 48, 49, 50, 51 or 52, wherein said profiles are involutes of a circle.

54. A scroll-type machine constructed and arranged to operate substantially as hereinbefore described with reference to and as illustrated in the accompanying drawings.

55. A method of manufacturing a scroll-type machine substantially as hereinbefore described with reference to the accompanying drawings.

