WORM ROTARY COMPRESSORS WITH LIQUID JOINTS

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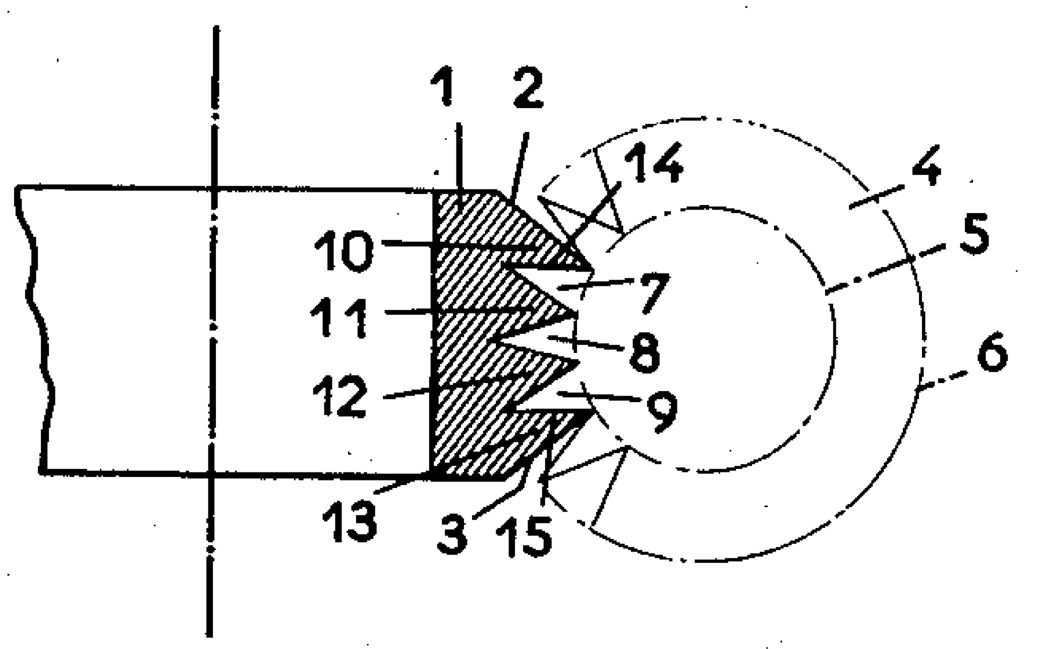

FIG. 1

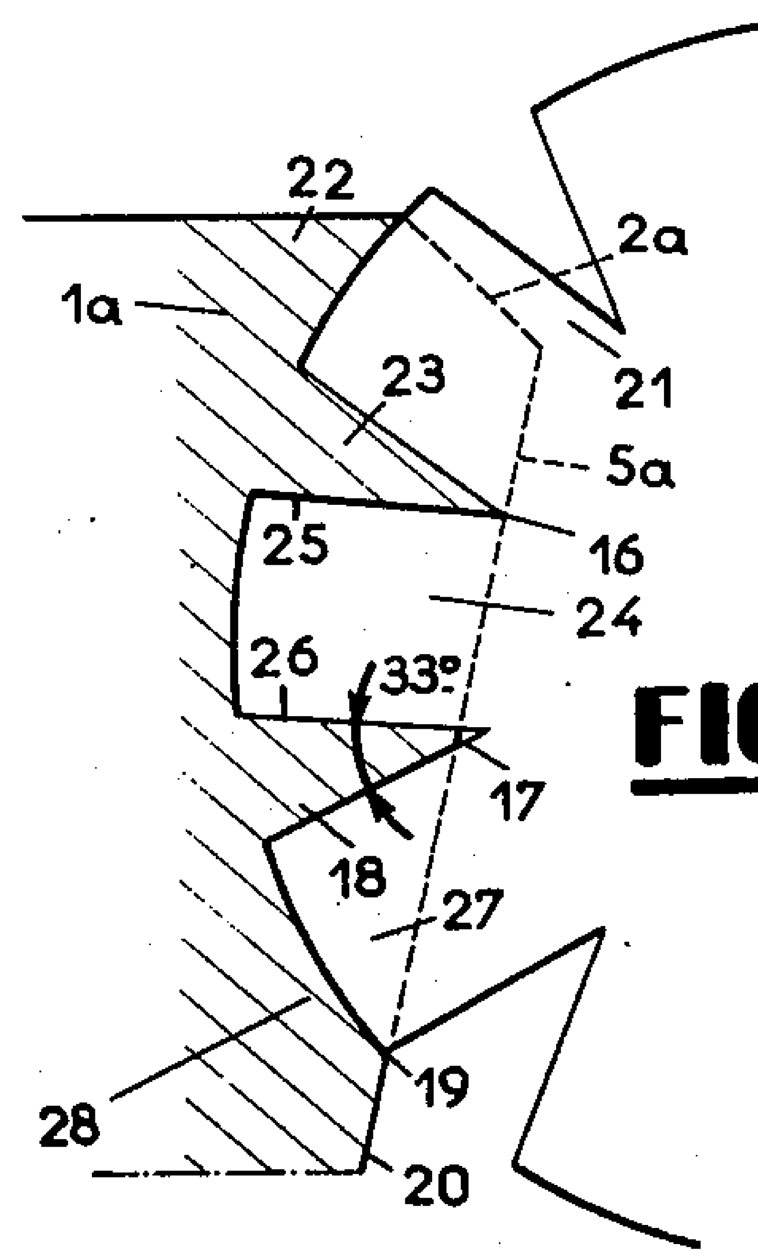

FIG. 2

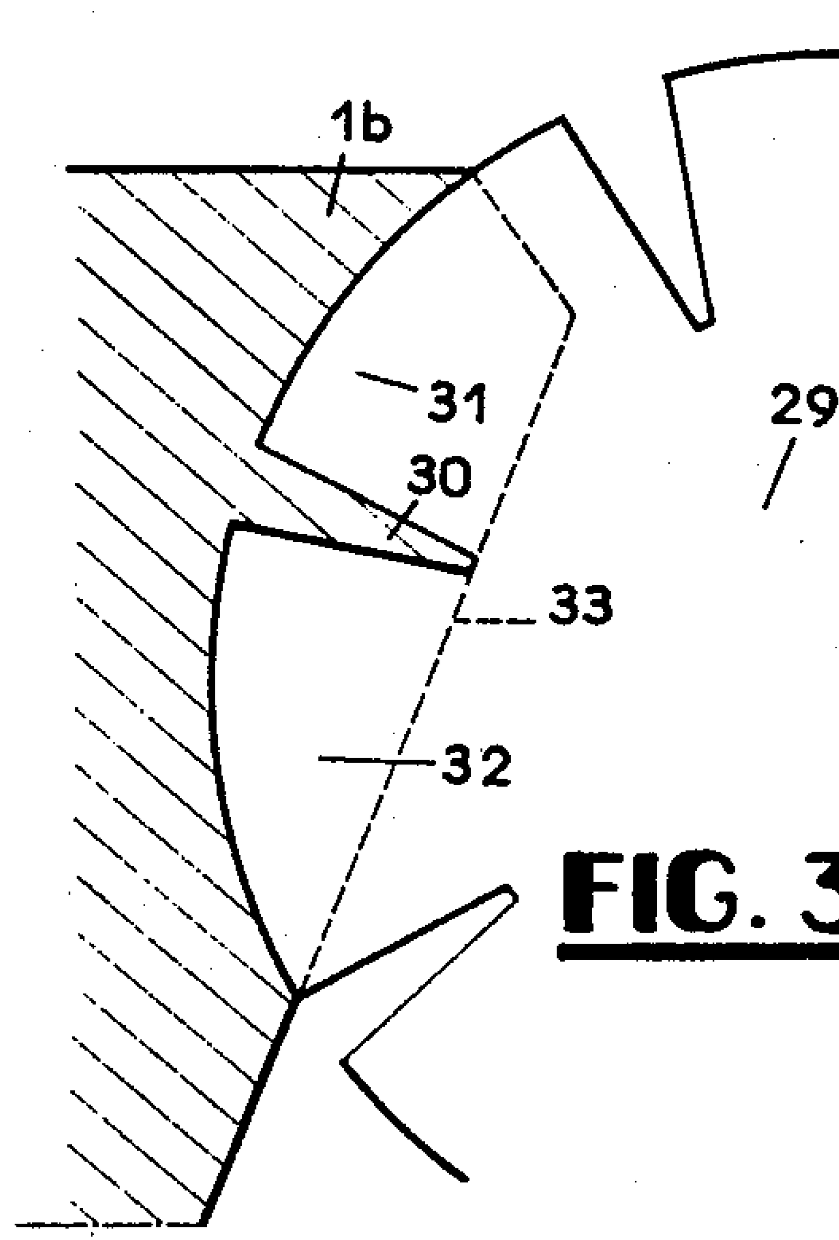

FIG. 3

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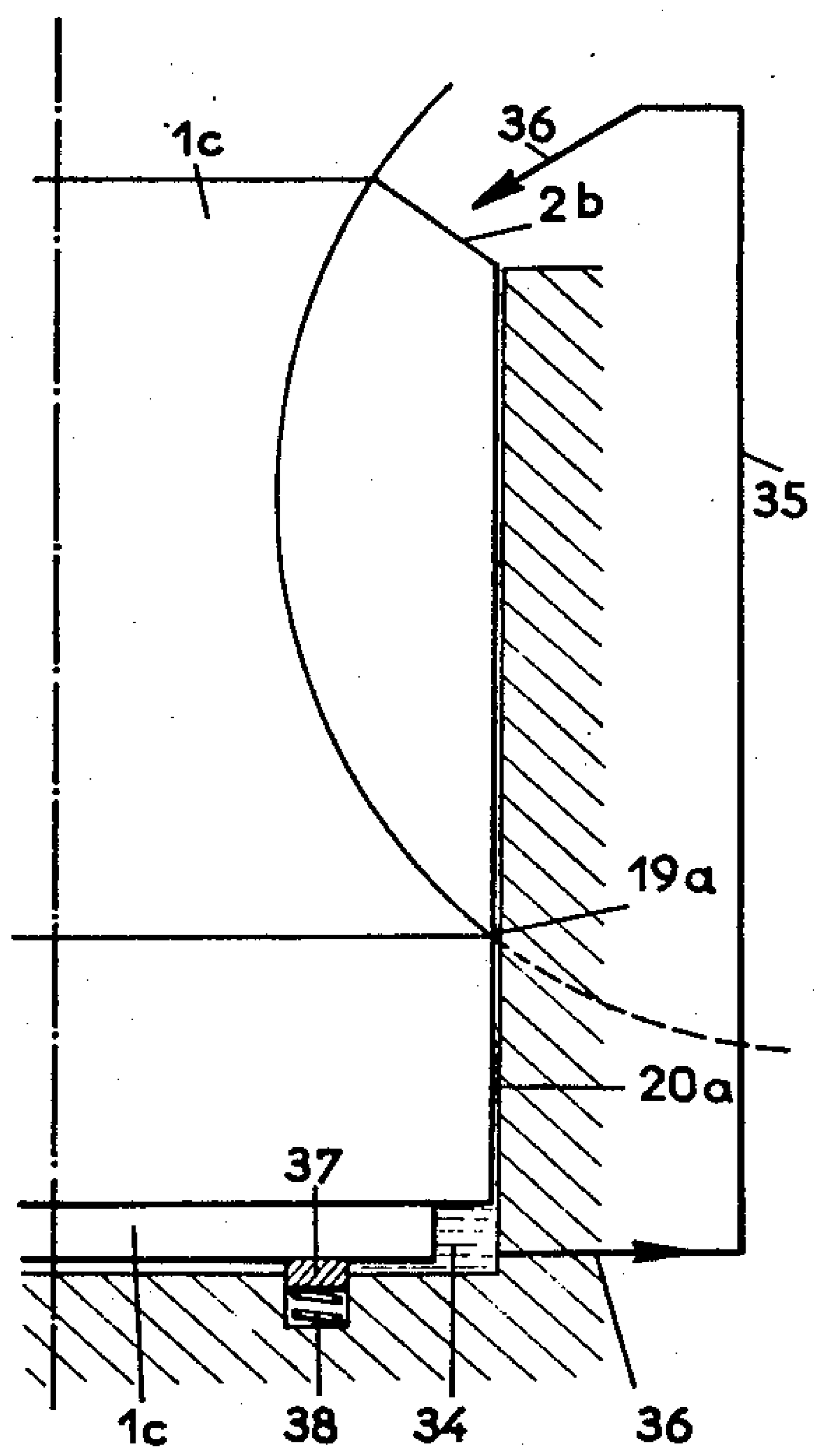

FIG. 4

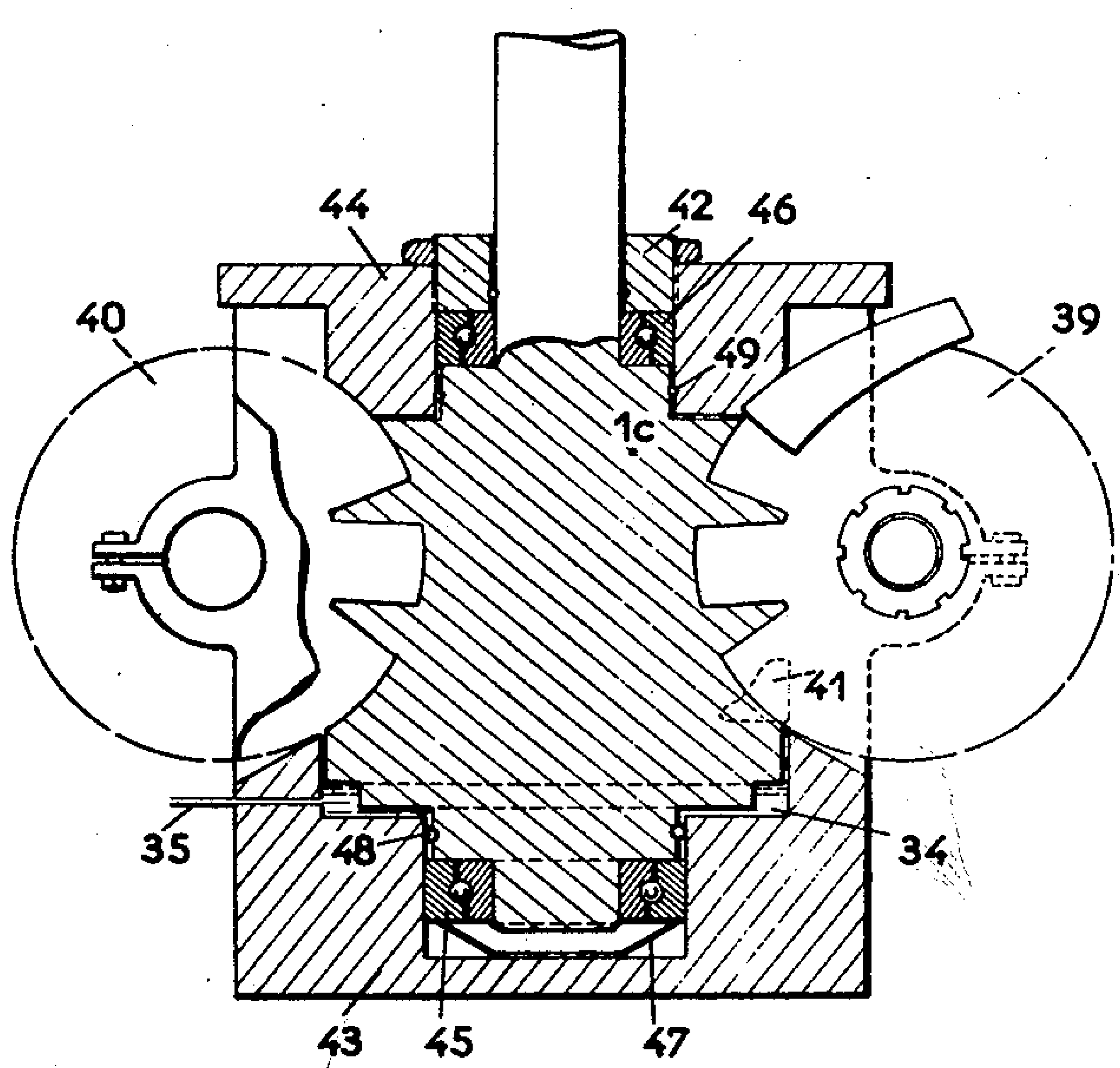

FIG. 5

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WORM ROTARY COMPRESSORS WITH LIQUID JOINTS

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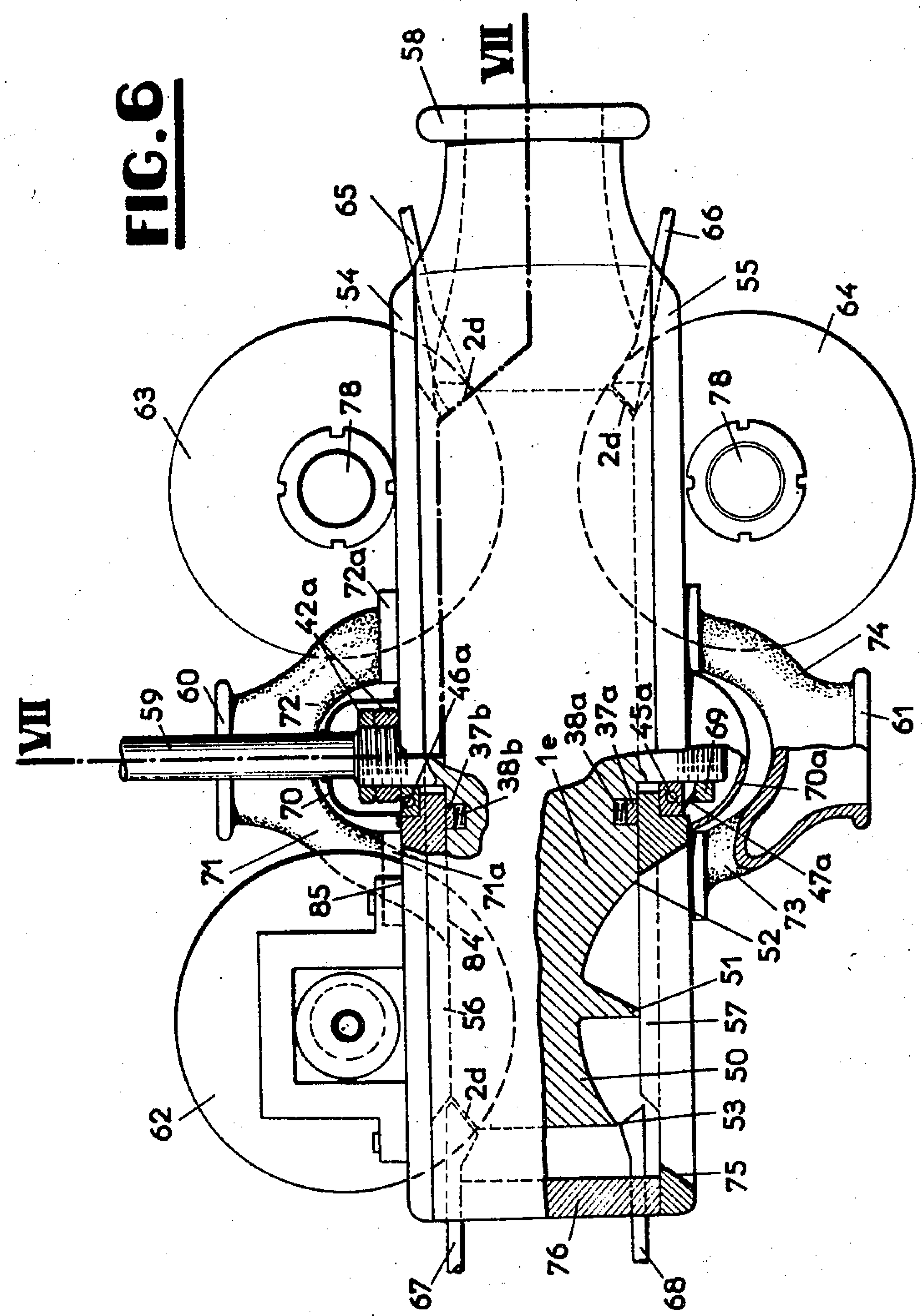

FIG. 6

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WORM ROTARY COMPRESSORS WITH LIQUID JOINTS

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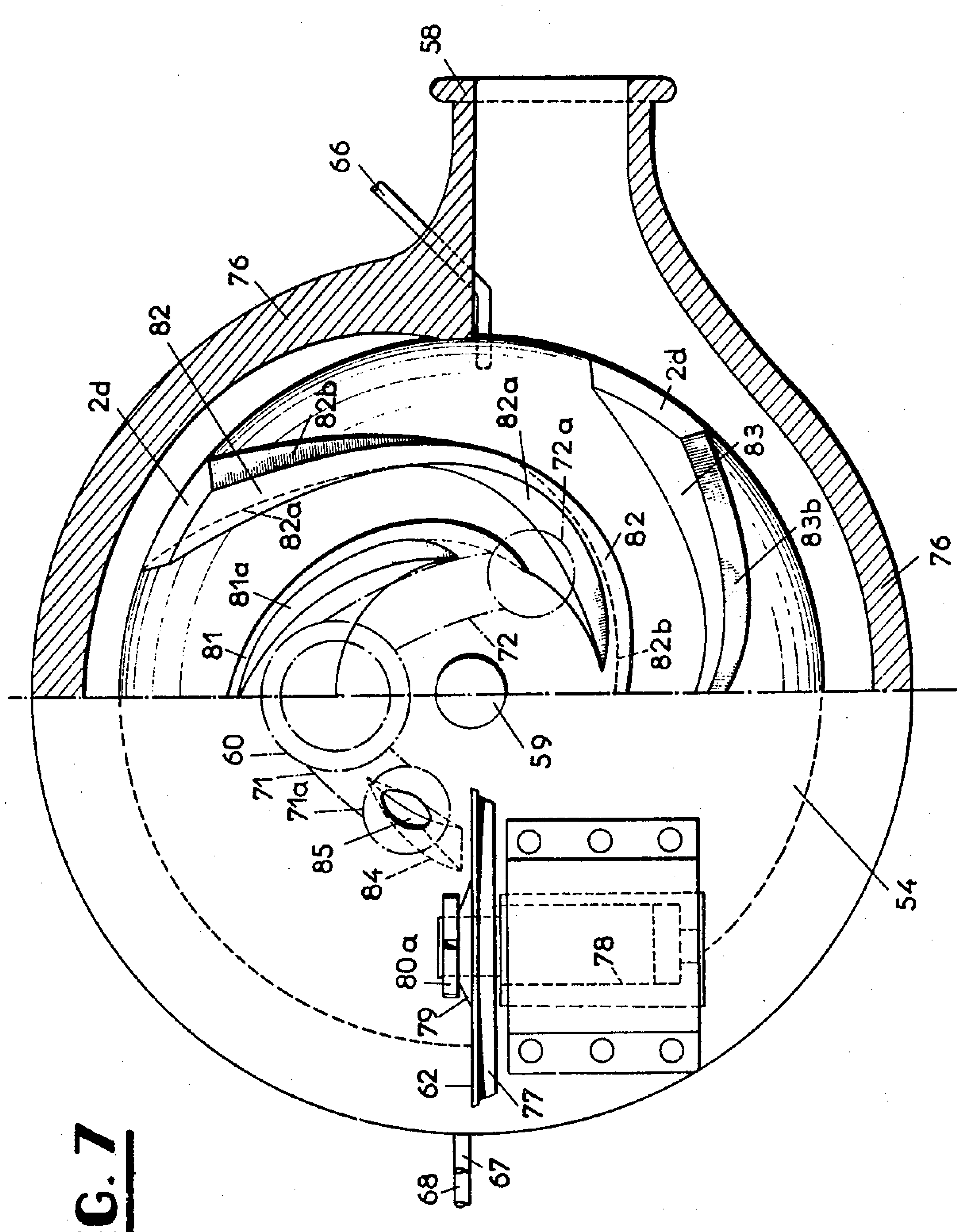

FIG. 7

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WORM ROTARY COMPRESSORS WITH LIQUID JOINTS

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18 Claims. (Cl. 230—150)

The present invention concerns various improvements relating to rotary compressors utilizing hour-glass worms as drive members and in which the tightness between the worm threads and the pinion teeth is ensured by liquid joints, the said improvements making it possible, in particular, to produce compressors having an appreciable effective volume corresponding to a reduced number of worm threads and affording nevertheless a high compression rate per stage, whilst using hour-glass worms and pinions made in one piece.

The inventor has noticed in fact that in order to obtain an important effective volume between the threads of such of a screw, without having to split either the screw or the pinions into several pieces and assemble those two types of components by successive segments, it becomes necessary to resort to worms having but four to six threads, and capable of cooperating with unitary pinions having preferably seven or eleven teeth respectively.

The present description will show the means whereby the inventor can successfully obtain in a single stage, a pressure of 7 kg./cm.^{2} for current industrial uses, in spite of the above-mentioned reduced number of threads.

Another improvement within the scope of the present invention concerns the relative forms to be given to the worm extension on the high pressure side, as well as to the surrounding casing, in order to eliminate all axial thrusts on the worm and make it possible for the compressor bearings to rotate at a high speed, so as to obtain the highest possible power to weight ratio.

The inventor has noticed that when it is desired to obtain with only four threads the high compression rate required to produce the above-mentioned pressure of 7 kg./cm.^{2}, it becomes necessary to combine three means which the known art has used either separately or in pairs and to provide simultaneously, between the compressed gas inlet and outlet, a reduction in the length of the compression chambers defined by adjacent threads, a reduction in the depth of worm threads, and a reduction in the diameter of the casing surrounding said worm.

It is possible, theoretically at least, to combine these three means, by giving to the casing surrounding the worm a curvature having the same direction as that of the worm, together with a suitable offset, but the rectilinear path of liquid particles injected in a direction tangential to the casing at one end of the latter in order to form the said liquid joints, produces normally a break in continuity within these joints.

For that reason, and quite independently from the known cylindrical shape, the inventor advocates, as a means of avoiding said breaks in continuity and simplifying the construction of compressors, the use of casings having a linear profile in cross-section, i.e. conical casings producing a compression rate which increases in proportion to the angle of opening of the cone, the said casings becoming at the extreme limit, for an angle of opening of 180°, plane-shaped casings cooperating with discs carrying spirally shaped threads, and placed at right angles to the casing axis.

It is possible to accomplish in practice the required aim and to produce a 7 kg./cm.^{2} pressure in a single stage and with one-piece drive members, by using a cylindrical casing, provided that the worm contains six

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threads, whereas with four threads only it becomes necessary to combine all three above mentioned means, and to use preferably plane-shaped casings.

Other objects of the present invention will appear from the following description, to be read in conjunction with the annexed drawings, in connection with various embodiments of compressors according to the invention, it being understood that said embodiments are given as examples only.

On the drawings:

FIGURE 1 shows diagrammatically a diametral cross-section of an ordinary hour-glass worm mounting, indicating the shape ordinarily given to the teeth of a unitary pinion in order to enable them to be inserted between the threads of a unitary globoid worm, the delivery rate of compressed gas being very low with this type of mounting.

FIGURE 2 shows one half of a diametral cross section of a compressor having a higher rate of delivery, with a unitary six thread hour-glass worm cooperating with a truncated cone shaped casing together with two unitary pinions having eleven teeth.

FIGURE 3 shows one half of a cross section similar to FIGURE 2, but relates to a compressor with a unitary four thread hour-glass worm cooperating with two unitary pinions of seven teeth.

FIGURE 4 shows a half cross-section of a compressor having a unitary hour-glass worm, together with a cylindrical casing and an exhaust groove communicating with the open air.

FIGURE 5 shows diagrammatically a compressor comprising a unitary six thread hour-glass worm having an outer cylindrical profile, together with two unitary pinions of eleven teeth, said compressor including an exhaust groove placed on the high pressure side, and being provided in addition with means for adjusting the relative positions of the worm and pinions.

FIGURE 6 represents an elevational view, with parts taken away, of a disc having four spiral threads on each face cooperating with four unitary pinions of seven teeth and with two plane-shaped casings, said compressor comprising offset plastic rings for adjusting the position in height of the axial lines of said pinions.

FIGURE 7 represents a plan view, seen from above, of the compressor of FIGURE 6, with the casing cut away along line VII—VII of said FIGURE 6.

FIGURE 1 shows hour-glass worm **1**, truncated at its upper and lower portions according to conical profiles **2** and **3**, and cooperating with a pinion **4** having fifteen teeth adapted to engage with the threads of said worm.

Worm **1** has a circular outer profile, corresponding to chain dotted circle **5**, the latter being concentric with circle **6** which corresponds to the bottom of the threads in worm **1**.

It will be understood, by inspecting the position shown on FIGURE 1, that the three pinion teeth **7**, **8** and **9**, can easily be inserted between threads **10**, **11**, **12** and **13** in worm **1**, which makes it possible to use a unitary worm and a unitary pinion, and to avoid having to split either the worm or the pinions into several pieces which would require to be assembled in successive stages, in step with the insertion of said pinion teeth between the threads of said worm.

In this embodiment, the required result is obtained by reason of the fact that the faces **14** and **15** in threads **10** and **13** are both horizontal and parallel to each other in the position shown on FIGURE 1, whilst median tooth **8**, which is symmetrical relatively to the median horizontal plane, also penetrates easily between threads **11** and **12**.

The arrangement in FIGURE 1 affords therefore the possibility of producing a worm compressor in which worm and pinions can be either made in one piece or formed by assembling several segments into one piece prior to inserting the pinion teeth within the worm threads.

However, one disadvantage in this arrangement stems from the fact that the volume between adjacent threads is very reduced, which results in a rather low value of delivery rate and power to weight ratio.

It will be readily understood that the effective volume in the compression chambers increases appreciably when providing a much reduced taper on the pinion teeth, in which case the use of unitary components, becomes possible only by reducing the number of threads, the compression rate thereby obtained being entirely insufficient in known types of compressors.

FIGURE 2 shows the hour-glass worm **1**a, whose top portion is truncated according to conical profile indicated in **2**a, and whose effective outer profile is also conical as may be seen in **5**a, on said figure. The roots of the threads define a concave portion of the worm surface which is part circular when seen in axial section.

It will be seen on FIGURE 2 that the compression chambers contained between adjacent threads decrease progressively in cross section, towards the end of compression, particularly owing to the fact that, starting from point **16** level, the depth of threads decreases gradually, in such a way that the truncation, indicated by **17** on thread **18**, progressively increases down to the base of the effective part in the worm, i.e. down to point **19**.

The worm extends in **20**, beyond said effective part, said extension corresponding in shape to the profile of the conical casing which surrounds it.

It will further be seen on FIGURE 2 that the conical shape **5**a tends to make up for the increase in diameter at the bottom of the worm, below the median plane of said worm, and that the centre of gravity of the compression chambers tends, at the end of compression, to take up a position, such as that of point **19**, which is nearer to the worm axis than would be the case if the worm cooperated with cylindrical casings.

Therefore, as mentioned above, a combination of the three means enumerated in the introduction to the present description, results in the production of a compression rate equal to or even higher than that of the compressor shown on FIGURE 1, but with a much higher volume of delivery, whilst affording the possibility of using unitary pinions and worm having relative profiles enabling them to be assembled for given suitable angular positions of worm **1**a.

Indeed, since the said worm is truncated on line **2**a, there are no threads above pinion tooth **21**; therefore tooth **21** can easily penetrate between thread **23** and worm top portion **22**.

Median tooth **24** in the effective sector of said pinion can therefore penetrate easily between threads **18** and **23**, owing to the fact that sides **25** and **26** respectively associated to threads **23** and **18**, adapted to cooperate on either side of tooth **24**, are parallel to each other.

Finally, the lower pinion tooth **27** penetrates easily under thread **18**, between the latter and worm lower portion **28**.

It will be seen that, with the alternative arrangement on FIGURE 2, the mounting of the pinions on the worm can be carried out simply by bringing two of the six worm threads opposite one pinion, in a position approximately symmetrical to the median plane of constriction of said worm, and by giving parallel profiles, seen particularly in **25** and **26**, to the sides of adjacent worm threads, so as to enable them to cooperate with pinion teeth having also parallel sides.

It will be seen that the number of teeth which have to be inserted simultaneously within the worm, is lower in the case of FIGURE 2 than in that of FIGURE 1.

This advantage become even more marked when there exists the possibility of using a four thread worm cooperating with pinions having seven teeth, as shown on FIGURE 3.

In this case, the compressor delivery comes to a maximum owing to the larger volume of the compression chambers contained by the casings and the worm, as a result of the reduced number of threads.

Besides, in the arrangement of FIGURE 3, the sides of adjacent worm threads may be made to converge outwardly, since it becomes possible to place pinion **29** in a position wherein, as in the case shown in cross section on FIGURE 3, there is on worm **1**b but one thread **30** requiring to be engaged with two adjacent teeth **31** and **32** on pinion **29**, said thread being evenable to have in cross section any convex shape capable of being easily inserted between said teeth **31** and **32** on pinion **29**.

It will be readily understood that the placing of pinion **29** on the worm could become impossible if there were, opposite the said worm, two adjacent threads defined by sides converging towards the centre of the said piston.

However, owing to the reduction from six to four in the number of threads, the production of a compression rate comparable to that of the compressor on FIGURE 2 makes it necessary to appreciably increase the reduction in diameter, i.e. to increase appreciably the slope of profile **33** relatively to the vertical, and to go even as far as using plane-shaped casings placed at right angles to the worm axis.

In practice, it will be preferable to use either a six thread cylindrical worm, or a four thread worm, cooperating with plane-shaped casings, in order to produce compression rate values of about 5.

When using the said plane-shaped casings, it is advantageous to provide two compressors using respectively two symmetrical threads associated to one double worm, in order to balance the axial thrusts exerted by the gases in the said compressors on the spindle of said double worm.

The arrangement on FIGURE 4 concerns a compressor in which worm **1**c, having a cylindrical profile on the major part of its height, is truncated in **2**b, at its top portion, for suction.

This figure shows how it is possible to avoid the production by the compressed gases of an upward axial thrust on the lower face of a vertically placed worm having a cylindrical outer profile, and consequently to relieve the bearings, so as to make it possible to increase the rotational speed and the power to weight ratio of the compressor, taking into account the fact that the worm is not subjected to radial thrusts, since the compressor is provided with pinions mounted in pairs.

The worm **1**c in FIGURE 4 comprises beyond its effective portion, i.e. beyond point **19**a, a cylindrical extension **20**a.

The compressor in FIGURE 4 is of a known type, but includes however, in its lower portion, an exhaust groove which communicates with the open air by means of a duct, diagrammatically shown in **35**, the said groove affording the possibility to eliminate all pressures on the lower face of the worm, and to reinject the leaking liquid recovered within said groove **35** in the direction of arrows **36**.

It will be understood that the liquid in the joints can penetrate between the worm and the casings below point **19**a, and accumulate within exhaust groove **34**, which makes it necessary to provide, below worm **1**c, a circular segment **37**, forming a plane joint, said segment resting on at least one spring **38** in order to protect the roller bearings carrying the spindle **8** of worm **1**c.

Since this exhaust groove communicates with the open air there is no possibility of water infiltrating between the worm base and the casing and accumulating under pressure beneath the worm in a manner liable to produce an upward axial thrust.

The compressor shown on FIGURE 5 includes worm **1**c having a cylindrical outer profile, as well as an exhaust groove **34** communicating with the suction side by means of a duct **35**, as in the case of FIGURE 4. The roots of the threads define a concave portion of the worm surface which is part circular when seen in axial section.

The worm **1c** has six threads, and it will be seen that both pinions **39** and **40** are provided with eleven teeth having two parallel sides and similar in type to those shown on FIGURE 2.

Taking into account the injection of liquid, the effective compression rate in a compressor of this kind is in fact slightly higher than the theoretical rate. It may range from 5.5 to 6, whereas its theoretical value is between 4 and 5, which makes it possible to produce compressed air at a 7 kg./cm.^{2} pressure, with a good efficiency factor.

The figure shows in **41** an exhaust port, outlined in dotted lines and formed within the casing thickness, said port being placed in fact in front of the plan of FIGURE 5.

The position in height of worm **1c** can be adjusted by means of threaded plug **42**.

It will be seen further that worm **1c** is mounted between casing **43** and cap **44**, by means of lower bearing **45** and upper bearing **46**, lower bearing **45** resting on a spring washer **47** and both bearings being protected from all contact with the liquid in the joints by means of circumferential gaskets **48** and **49** respectively.

Pinion **39**, made in a plastic substance, is mounted on a steel support.

An inspection of the embodiment of compressors shown on FIGURES 6 and 7 will reveal that the disc shaped drive member **1e** is provided with threads on surfaces perpendicular to the axis of the compressor. The roots of these threads define concave surface portions in the form of annular depressions which are coaxial with the axis of said compressor and part circular in a plane containing said axis.

The portion of FIGURE 6 in which the casing is shown broken away shows in cross section one of the effective portions of the compressor, indicated by **51**, wherein the threads are spiral shaped, and not helix shaped as usual, the depth of said threads increasing gradually from point **52** up to the median portion corresponding substantially to the position of thread **51**, and decreasing beyond that position towards the periphery of the disc, i.e. in the direction of point **53**.

The disc includes two spiral threads symmetrically placed relatively to the median plane of the compressor and situated respectively one on the top of the disc, the other on the bottom.

The crests of the threads on the disc bear upon two plane shaped caps **54** and **55** provided respectively above and below the disc and including internal bosses **56** and **57**.

Gas entering the compressor periphery by way of piping **58** becomes gradually compressed towards the interior of the compressor, i.e. in the vicinity of spindle **59** of disc **1e**. This air escapes at bottom and at top by way of duct **60** and **61** which may thereafter converge within a compressed air manifold.

As mentioned above, the decrease in diameter of the compressor from the inlet of gases to their outlet in the vicinity of the spindle, acting in combination with the gradual reduction in the depth of threads provided immediately beyond the position shown in FIGURE 6, and with the gradual reduction in the length of compression chambers, is operative in producing a high compression rate of about 5.

By combining these three means, it is possible to obtain a pressure of at least 7 kg./cm.^{2}, as in the case of the cylindrical compressor of FIGURE 5, by using drive members with only four threads, instead of the six threads used in the case of FIGURE 5.

The compressor in FIGURE 6 has four pinions, three of which are indicated on the drawing by the numerals **62**, **63** and **64**.

The liquid which is intended to ensure the tightness between the various pinions and the threads of drive member **1e**, is supplied by ducts **65**, **66**, **67** and **68**, towards the compressor periphery.

The spindle carrying the disc **1e** is mounted on covers **54** and **55** by means of bearings **45a** and **46a** which are protected against water infiltration by means of plane shaped gaskets **37a** and **37b**, loaded by springs **38a** and **38b**.

The assembly of said drive member includes in the lower part a washer **47a** and a tightening screw **69**.

This assembly includes in the upper part two screws **42a**. A lubricator cap **70** is provided for lubricating bearing **46a**.

It will be seen in FIGURE 7 that the compressed air outlet duct **60** is made up in fact by bringing together two ducts **71** and **72** which receive respectively the compressed air in the vicinity of upper pinions **62** and **63**.

Similarly, the lower duct **61** is made up of two ducts **73** and **74** brought together. A portion of duct **73** has been taken away in order to show that the said duct receives compressed air at the level of the lower left pinion, not shown on the drawing, whereas duct **74** receives compressed air at the level of pinion **64**.

A grease box **70a** is also provided in the lower portion of spindle **59** for lubricating bearing **45a**.

The edge **75** in the sectioned part **58** of cover **55** corresponds to one of the millings provided to secure a clearance for the various pinions. The casing on which bear the two covers **54** and **55** can be seen on the drawing, in particular at **76**. This casing whose outer shape is formed by a surface of revolution has an internal dissymmetry and forms a suction volute before joining up with the suction in the straight portion of the drawing.

The mounting of pinion **62** can also be seen in FIGURE 7, with its steel support indicated under the numeral **77**.

Said pinion and support are mounted on one of the shafts **78** by means of a spring washer **79** and a screw **80a** which fulfils the same function as screw **80** seen in FIGURE 5. The truncated ridges of three out of the four threads in disc **1e** are indicated respectively by numerals **81**, **82** and **83**, the sides of said threads being indicated respectively by numerals **81a**, **82a**, **82b** and **83b**, and the truncated peripheral portions of said threads being indicated by numeral **2d**.

The dotted line indicated by numeral **84** shows the shape given to a compressed air exhaust port leading through cover **54** and opening into duct **71** by means of outlet **85**.

For the sake of simplicity, the casings surroundings the pinions and serving to recover leaking fluids, have been omitted from the drawing.

It is to be clearly understood that the embodiments referred to in the above description have been given by way of an example only, and that they may include various amendments, alterations or improvements without departing from either the scope or the spirit of the present invention.

Having thus described the invention, what is claimed as new and desired to be secured by Letters Patent is:

1. A compressor comprising a casing having a central axis, a disc rotatably and coaxially mounted in said casing, at least one face of said disc being formed with a coaxial annular depression therein, said depression being part-circular in section, a plurality of spirally extending threads projecting upwardly from said depression to the level of the remainder of said disc surface, the inner surface of said casing comprising a portion positioned to be swept by said threads and define therewith a plurality of compression chambers, fluid inlet means in said casing adjacent the periphery of said disc and fluid outlet means in said casing near the center of said disc, and at least one pinion projecting into said casing and having teeth meshing with the threads on said disc, the teeth on each pinion meshing with at least two but not more than three threads on said disc at any one time.

2. A compressor as claimed in claim 1 according to which said disc is spirally threaded on both surfaces and two pinions spaced 180° apart mesh with the threads on surface.

3. A compressor as claimed in claim 2 in which each side carries four threads, with not less than two nor more than three of said threads meshing with each pinion at any one time.

4. A compressor as claimed in claim 3 in which each pinion carries seven teeth.

5. A compressor as claimed in claim 4 in which said threads mesh with said pinions over an arc of about 102°.

6. A compressor comprising a casing having a central axis, a one-piece drive member rotatably and coaxially mounted in said casing, said drive member having at least one concave surface portion which defines a pair of circular arcs in those planes containing said axis, a group of screw threads positioned on said concave surface, the crests of said threads lying in the locus generated by a straight line rotated about a point on said axis, at least part of the inner surface of said casing being positioned to be swept by said crests and define therewith a plurality of pressure chambers extending from one end of said group of screw threads to the other, a plurality of toothed pinions mounted to turn about axes perpendicular to the axis of said drive member and projecting through said casing to mesh with said threads, each thread traversing an arc about the axis of the drive emember no greater than that between adjacent pinions, a low pressure fluid inlet in said casing communicating with one end of said group of threads and a plurality of high pressure fluid outlets in said casing, one immediately adjacent each of said pinions and communicating with the other end of said group, the teeth on said pinion being thicker in said axial planes than said threads, and at least two but not more than three teeth of each pinion meshing simultaneously with said threads, and blocking at least one of said pressure chambers.

7. A compressor as claimed in claim 6 in which said drive member terminates near the inlet end of the thread group in a surface transverse to said locus, and said transverse surface terminates where it meets said locus in a chamfer defining one wall of a passageway communicating with said inlet.

8. A compressor as claimed in claim 7 in which said drive member carries six discrete threads and there are two of said pinions, the two flanks of each tooth on said pinions being parallel to each other.

9. A compressor as claimed in claim 8 in which each pinion has eleven teeth.

10. A compressor as claimed in claim 7 in which said drive member carries four discrete threads and there are two of said pinions, and in which the two flanks of each pinion tooth converge as they approach the center of the pinion on which that tooth is carried.

11. A compressor as claimed in claim 10 in which each pinion has seven teeth.

12. A compressor as claimed in claim 6 in which said straight line is perpendicular to said central axis.

13. A compressor as claimed in claim 7 in which said drive member is a disc, having at least one concave surface portion defining a circular arc in those planes containing said axis and provided with a group of said threads having their crests in a plane which constitutes the locus generated by said straight line, said disc being formed with a peripheral edge transverse to said one face positioned near the inlet end of said thread group, and said edge terminating where it meets said one face in a chamfer defining one wall of a passage which communicates with said inlet.

14. A compressor as claimed in claim 13 comprising a single drive member having two oppositely facing concave surface portions, each of which defines a circular arc in those planes containing said axis, and is provided with a group of said threads with the crests of the threads positioned on said two concave surface portions lying in the loci generated by two parallel lines perpendicular to said axis which are rotated thereabout, and a pair of pinions engaging the threads on each of said concave surface portions.

15. A compressor as claimed in claim 14 in which there are four discrete threads on each concave surface portion and the two flanks of each pinion tooth converge as they approach the center of the pinion on which that tooth is carried.

16. A compressor as claimed in claim 6 in which said straight line is perpendicular to said axis and the inner surface of said casing forms a gas suction volute.

17. A compressor as claimed in claim 6 in which said straight line is parallel to said central axis, said drive member carries six discrete threads, and each pinion carries eleven teeth.

18. A compressor as claimed in claim 17 comprising means for supplying at the inlet end of said casing sealing fluid which flows between said drive member and pinions, said casing terminating at its outlet end in a reservoir positioned to receive said sealing fluid after it has passed between said drive member and pinions, said reservoir being connected to said inlet end of said casing and said drive member comprising a thread-free cylindrical portion extending into said reservoir.

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