WEAR AND TEAR-COMPENSATING HIGH-PRESSURE GEAR PUMP

Filed Aug. 29, 1968 7 Sheets-Sheet 1

Fig.1

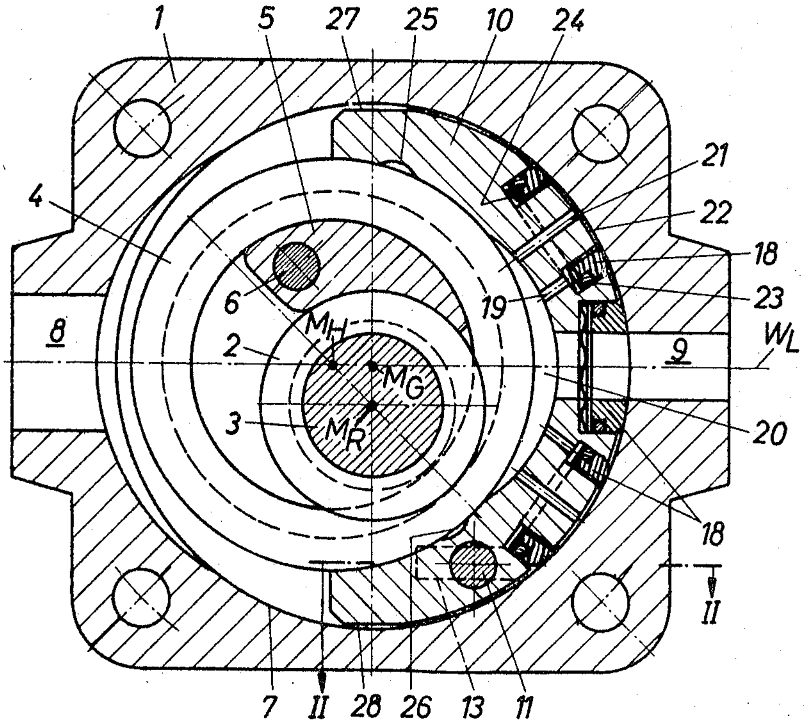

Fig.2

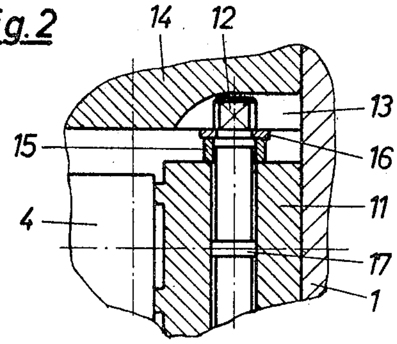

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Fig. 3

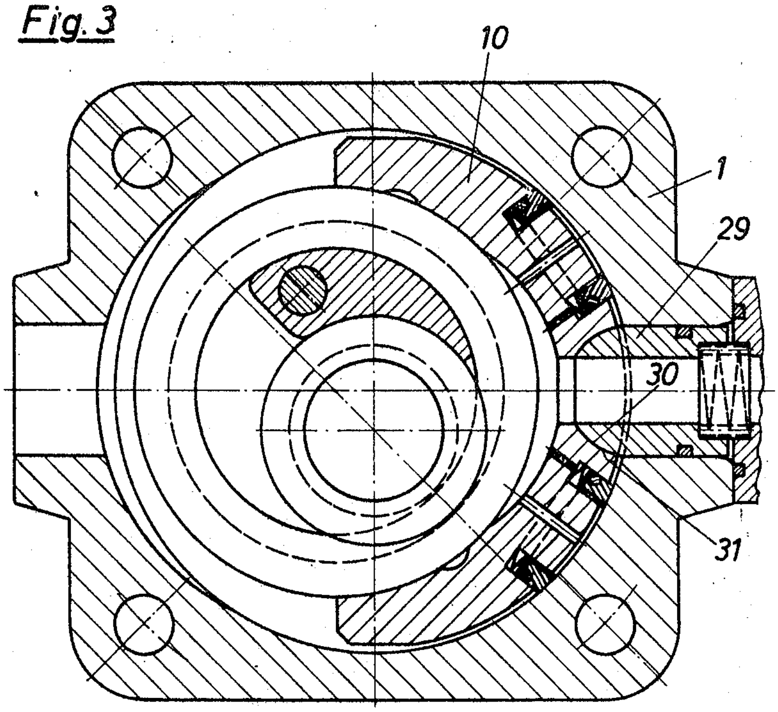


Fig. 4

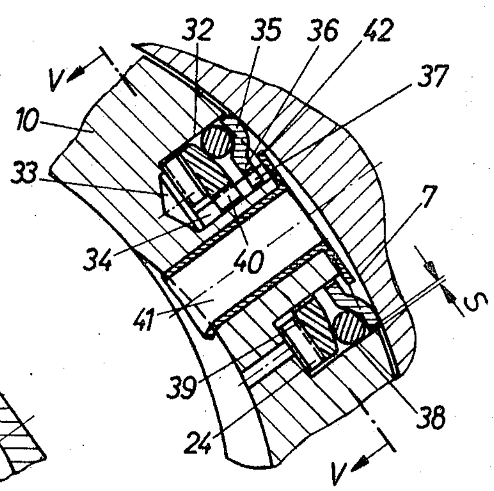
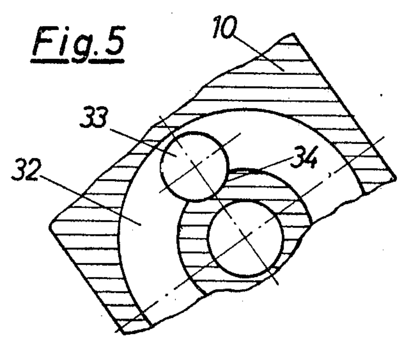


Fig. 5



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WEAR AND TEAR-COMPENSATING HIGH-PRESSURE GEAR PUMP

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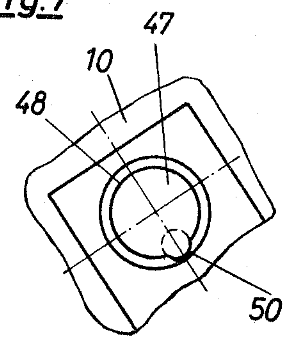
Fig.7



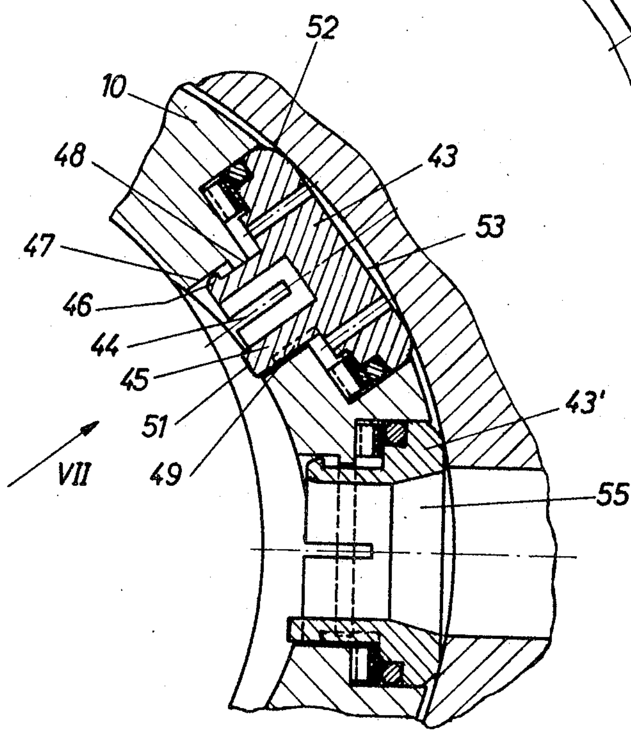


Fig.6

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Fig.8

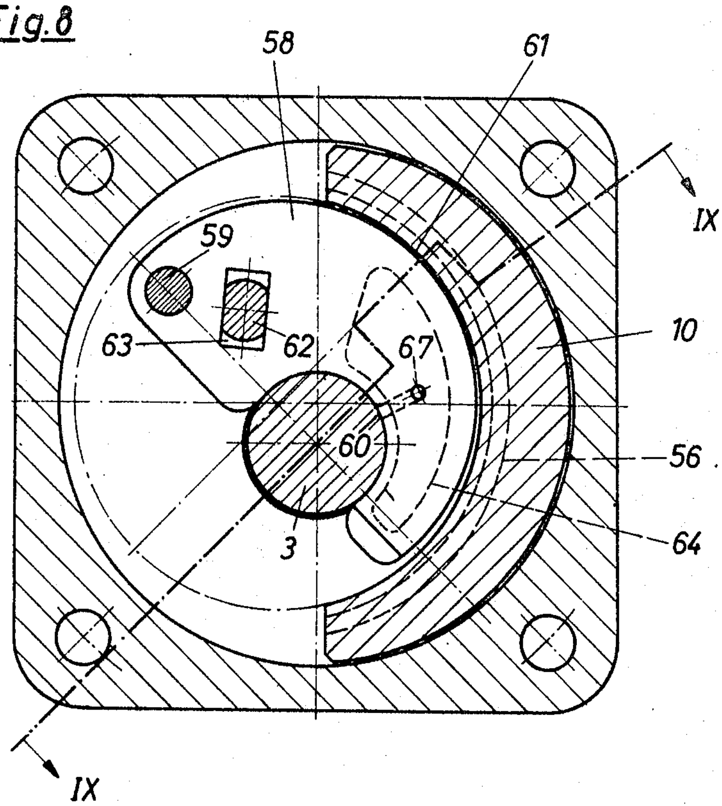
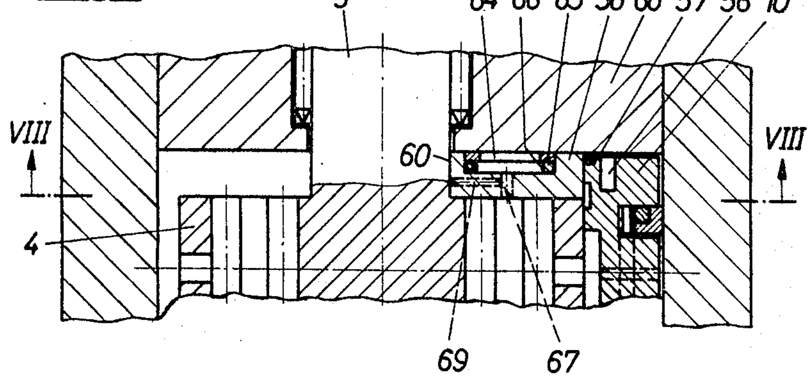


Fig.9



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WEAR AND TEAR-COMPENSATING HIGH-PRESSURE GEAR PUMP

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Fig.10

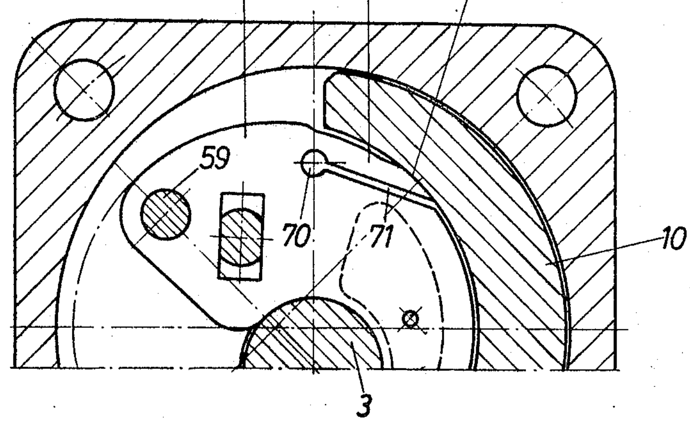
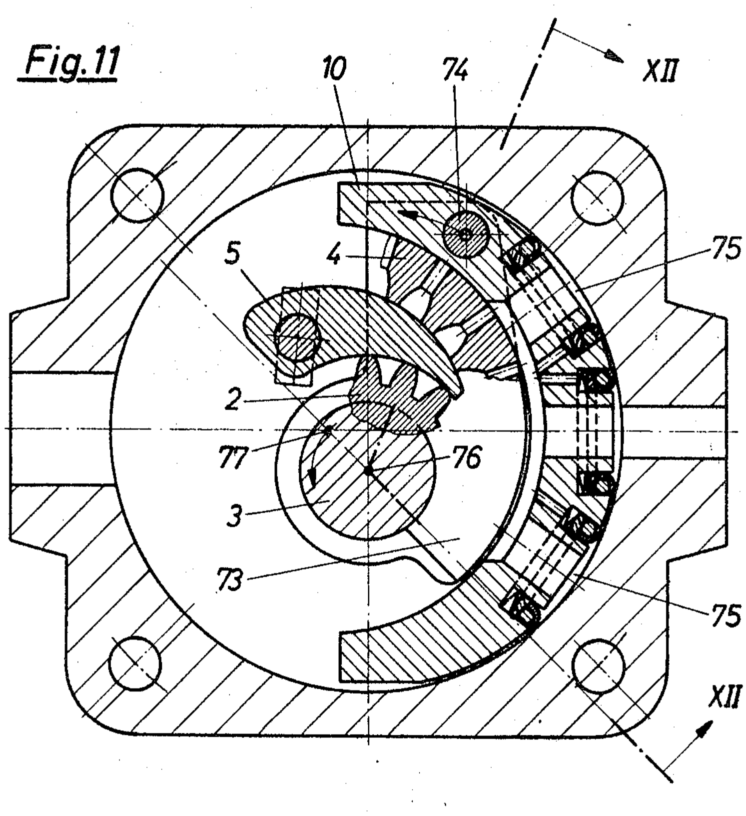


Fig.11



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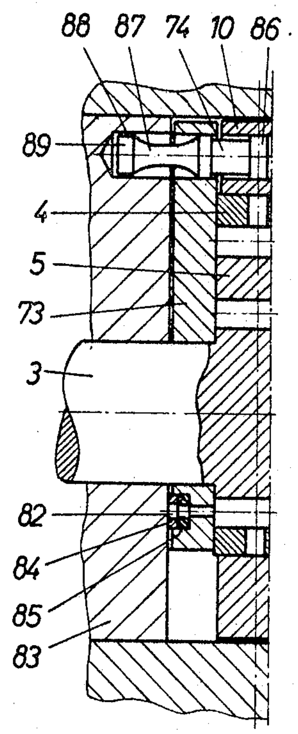

Fig.12

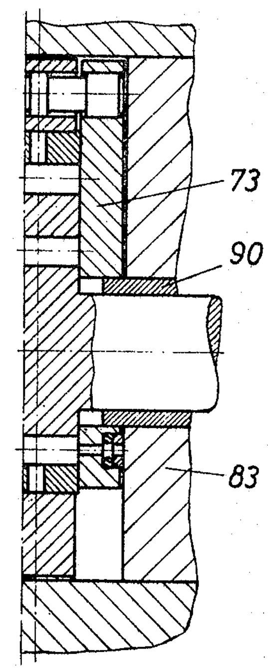

Fig.13

Fig.14

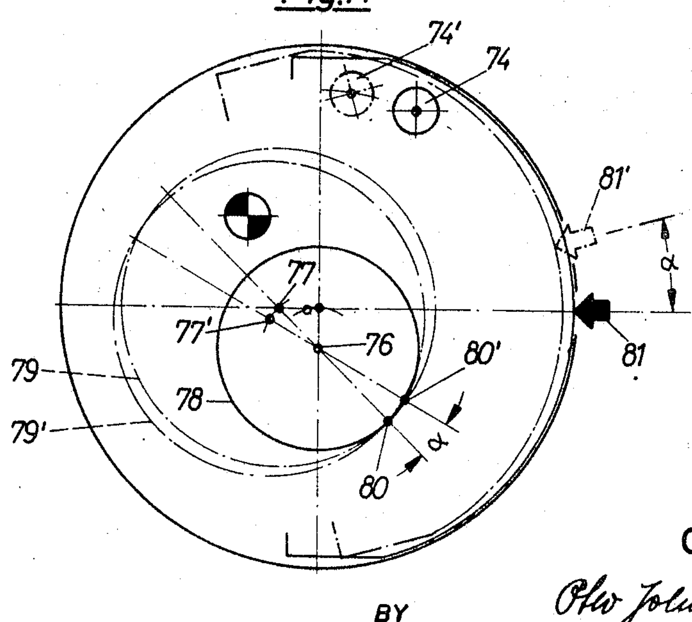

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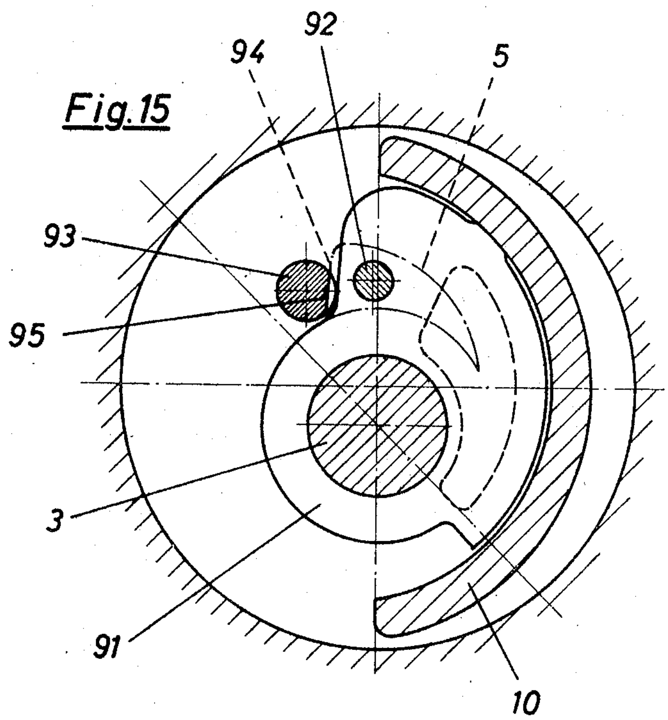
Fig. 15



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United States Patent Office 3,525,580
Patented Aug. 25, 1970

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3,525,580
WEAR AND TEAR-COMPENSATING HIGH-PRESSURE GEAR PUMP

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Claims priority, application Germany, Sept. 1, 1967, 1,653,826
Int. Cl. F04c *1/06*
U.S. Cl. 418—126 17 Claims

ABSTRACT OF THE DISCLOSURE

Wear and tear-compensating high-pressure gear pump comprising a driven externally-geared pinion, a concomitantly rotating internally-geared wheel, a movable sickle-shaped filler member between the pinion and the internal gear and either one or two axial disks covering the gears laterally, as well as an insert member, called "control piston," being disposed on the pressure side of the gears and being radially displaceably arranged with respect thereto, enclosing the outer circumferential surface of the internal gear in a specific angular range and touching the latter as the sole stator part, said control piston containing the first portion of the pressure outlet duct, being relieved for the major part with regard to radial forces and hence pressing against the circumference of the internal gear with only a limited amount of excess force so that as a result the internal gear is essentially supported only on the pinion, on the one hand, and on the filler member, on the other hand, characterized in that the pump housing (**1**) is eccentrically bored with respect to the pinion shaft (**3**) in such a manner that the center (M_G) of the housing bore (**7**) is positioned on the hydraulic line of application or effective curve (W_L) which extends of necessity also through the internal gear center (M_H), and in that the control piston (**10**) is accommodated in the sickle-shaped space being formed due to the eccentricity between the internal gear (**4**) and the housing bore (**7**), being essentially adapted to the configuration thereof.

BACKGROUND OF THE INVENTION

Field of the invention

The present invention relates to a wear and tear-compensating high-pressure gear pump comprising a driven externally-geared pinion, a concomitantly rotating internally-geared wheel and a movable sickle-shaped filler member between the pinion and the internal gear, as well as an insert member, called "control piston," which is disposed on the pressure side of the gears and radially displaceably arranged with respect thereto, and which encloses or surrounds the outer circumferential area of the internal gear in a specific angular range, touching the latter as the sole stator part. This insert member contains the first portion of the pressure outlet duct, is relieved for the major part with regard to radial forces and hence presses against the circumference of the internal gear with a limited excess force so that the internal gear is supported essentially only on the pinion, on the one hand, and on the filler member, on the other.

Description of the prior art

In the known pumps of this type, the control piston is positioned in one recess each of the pump housing and of a cover being specifically provided for the control piston. By virtue of this arrangement, the angle at which the control piston encloses the outer circumferential surface of the internal gear is only small. Furthermore, the control piston has a complicated construction since it must not only be radially movable with respect to the internal gear, but also has to compensate for inclined or slanted positions of the internal gear which are due to wear and tear.

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SUMMARY OF THE INVENTION

In order to eliminate the disadvantages and drawbacks of the known pumps as outlined above, the present invention proposes that the pump housing be eccentrically bored with respect to the pinion shaft in such a manner that the center of the housing bore is positioned on the hydraulic line of application, or effective curve, which extends of necessity also through the internal gear center, and so that the control piston is accommodated within the sickle-shaped space formed due to the eccentricity between the internal gear and the housing bore and is adapted to the configuration of that space.

Since the control piston is thus positioned within the housing bore, the recesses in the housing and in the cover, which serve for guiding it, as well as the otherwise required seals or gaskets are rendered unnecessary. The control piston can enclose the internal gear within an angular range of up to 180° and above, which results in a stabler positioning, that is less sensitive to vibrations, of the internal gear on the control piston. In addition thereto, the control piston may be manufactured in a simple manner inasmuch as its surface is composed of plane and circular surfaces.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a cross-sectional view of an embodiment of the invention.

FIG. 2 is a cross-sectional view taken along lines II—II of FIG. 1.

FIG. 3 is a cross-sectional view of another embodiment of the suspension means of the control piston **10** of FIG. 1.

FIG. 4 shows a detail of the radial pressure field **22** of FIG. 1.

FIG. 5 is a cross sectional view taken along V—V of FIG. 4.

FIG. 6 is a cross sectional view, with portions cut off, of another embodiment of the control piston **10** of FIG. 1.

FIG. 7 is a view along the arrow **7** of FIG. 6.

FIG. 8 is a cross sectional view of the pump of the invention dealing primarily with means to eliminate the differences in pressure and excessive wear and tear upon the filler member of the control piston **10** and the internal pressure and is shown in cross section VIII—VIII of FIG. 9.

FIG. 9 is a cross sectional view with portions eliminated, taken on line IX—IX of FIG. 8.

FIG. 10 is a cross sectional view of a portion of the pump of the invention showing an additional improvements of the support of the control piston.

FIG. 11 is a cross sectional view of the pump of the invention showing still additional means of suspending the control piston **10**.

FIG. 12 is a cross sectional view taken along line XII—XII of FIG. 11.

FIG. 13 is a cross sectional view also taken along line XII—XII of FIG. 11, showing a different manner of positioning the axial disk **73**.

FIG. 14 is a schematic diagram showing another means of suspension of the control piston **10**.

FIG. 15 is a modification of the embodiment of FIG. 11.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

Various embodiments of the present invention are illustrated in the accompanying drawings wherein like elements are identified with the same reference numerals.

The pump which is shown in FIG. 1 in a cross sectional view and in FIG. 2 in a view taken along line II—II of FIG. 1 consists of the housing **1**, the pinion **2** with the driven pinion shaft **3**, the concomitantly rotating internally-geared wheel **4** and the sickle-shaped filler member **5** which is movably positioned on the bolt **6**. The housing bore **7** is eccentrically hollowed out with regard to the center M_R of the pinion shaft **3** in such a manner that the hydraulic line of application or effective curve W_L which extends axially through the suction connection **8** and the pressure connection **9** also extends through the center M_G of the housing bore **7** and the center M_H of the internal gear **4**.

Mounted on the pressure side of the pump between the internal gear **4** and the housing bore **7** is a control piston **10** which is secured in its position against rotation by means of a pin **11**. As illustrated in FIG. 2, the pin **11** engages on both sides with one square head **12** each in a flat milled-out portion **13** of the housing parts **14**. Accordingly, the control piston **10** is supported on the housing part **14** respectively by way of a spacer tube **15** and a retainer ring **16**. The pin **11** has in its center a small collar or flange **17** which, due to the short guide thereof in the bore of the control piston **10**, allows for a limited alignment of the control piston **10** toward the gear elements **2** and **4**.

In order to achieve a compensation of the radial forces, three pressure pistons or rams, for example ring pistons **18** made from plastic material, are provided within the control piston **10** and their end faces are rounded off or chamfered in such a manner that they have only a limited area of contact with the housing bore **7**. The space below each ring piston is in operative connection with the pressure space of the pump by way of a bore **19** which terminates in a recess **20** of the control piston **10**. A further bore **21** extends from the recess **20** to the space **22** between the control piston **10** and the housing bore **7** which is delimited by the ring piston **18**. The ring pistons are further provided with sealing rings **23** and are subjected to the action of cup or undulated springs **24**. By virtue of this construction, the ring pistons **18** are pressed against the housing bore **7**—by means of the undulated springs when the pump begins to run, and by means of the pressure medium during the operation of the pump—with only a small amount of force, and only low frictional forces will be produced when the control piston **10** becomes displaced due to wear and tear of the internal gear **4**. Pressure fields are formed or built up in the spaces **22** between the ring pistons **18** and the housing bore **7** which press the control piston **10** against the internal gear **4** and the latter against the filler member **5** and the pinion **2**, so that the pinion **2** will comb or mate with the internal gear **4** without backlash of the toothed wheels. The relieving pressure field which is being built up within the recess **20** and which is delimited by means of transverse grooves **25** and **26** in the control piston **10**, that are under the effect of suction pressure, counteracts the pressure fields mentioned above. The control piston **10** encloses the internal gear **4** within an angular range of slightly more than 180°. In order not to impair the possibility of a radial displacement of the control piston **10**, the latter is provided at the ends of the outer circumference with milled-out surfaces **27** and/or **28**. In order to assure that the conveying medium can flow from the suction side to the pressure side, the internal gear **4** has radial bores (not shown) extending between each tooth gap and the circumferential surface of the internal gear **4**.

FIG. 3 illustrates another embodiment relative to the suspension of the control piston **10**. The central pressure piston **29** is guided within a bore of the housing **1** and has at its inner end a spherical surface **30** which engages in a spherical indentation **31** of the control piston **10**, thus securing the latter against rotation. This measure affords the advantage that the control piston **10** can become adjusted or aligned with regard to the gearing almost free from any forces, while nevertheless being secured against rotation.

A particular embodiment of the circular radial pressure field (**22** in FIG. 1) is illustrated in FIG. 4 as well as in FIG. **5**, the latter being a cross-sectional view taken along line V—V of FIG. 4. A pocket bore **33** is so disposed in the annullar bore or recess **32** of the control piston **10** that a groove **34** is produced. A sheet metal ring **36** having a beaded edge **35** is guided within the bore or recess **32** with a small gap S whose beaded edge abuts or rests against the housing bore **7**. The sheet metal ring **36** which engages with a nose or projection **37** in the groove **34**, thus being secured against rotation, has the advantage that it cannot become canted by reason of its small thickness. The gap S is sealed by means of an O-shaped ring **38** which is pressed against the sheet metal ring **36** by a cup and, respectively, an undulated spring **24** by way of a plastic ring **39**. The plastic ring **39** is likewise fixed in position within the groove **34** by way of a nose or projection **40**. As an aid during assembly serves a tubular rivet **41** whose flange **42** prevents the sheet metal ring **36** from falling out.

Another embodiment of the control piston **10** is illustrated in a cross section in FIG. 6, and in FIG. 7 viewed in the direction of the arrow VII. The pressure piston **43** which is made according to the injection molding process for plastics comprises a stud **45** having slots **44** and a collar portion **46** at the lower end thereof. During the introduction of the stud **45** into the bore **47** of the control piston **10**, the collar **46** locks behind a projection **48** in the bore **47**. The pressure piston **43** is secured in position by means of a cam **49** being provided at the stud **45** and engaging in a groove **50** of the control piston **10**. A nose **51** serves as an aid for the introduction of the stud **45**. The outer edge of the pressure piston **43** is provided with a rounded-off portion **52** so as to prevent canting. The surface **53** of the pressure piston **43** is supplied with pressure medium by way of the bores **54**, thus being hydrostatically relieved. This relief is necessary for the following reason: During the deflection of the pinion shaft **3** and as a result of abrasion, the control piston **10** executes a slight rotary movement about its point of suspension near the tooth engagement. During this rotary movement, the pressure fields at the outside diameter of the control piston **10** must execute a slight displacing movement at the inside diameter of the housing. In order not to obstruct this displacing movement, a hydrostatic relief of the pressure fields toward the inside diameter of the housing is necessary. The central pressure piston **43'** is constructed in an analogous manner, but it has an axial opening **55** which serves for the discharge of the medium that has been conveyed.

The pump shown in FIG. 8 in a cross section taken along the line IX—IX of FIG. 9, and in FIG. 9 in a cross section taken along line VIII—VIII comprises means for partially relieving the filler member. It has been found that the filler member is subjected to a greater degree of wear and tear since the pressure of the control piston **10** which is exerted on the filler member by way of the internal gear **4** is greater than the internal pressure. This disadvantage is obviated by the measures which will be described hereinafter.

The control piston **10** is wider than the internal gear **4** and is supported on the axial disk **58** by means of a collar **57** formed by a perforation or aperture **56**. The axial disk **58** is disposed rotatably about the pin **59** and comes to abut or bear against the pinion shaft **3** with a cylindrical bore or recess **60**. The control piston **10**, on the other hand, is supported only against the surface element **61** of the axial disk **58**. Within the area of the tooth engagement, the control piston **10** is not supported on the axial disk **58** so that at that point it will push the internal gear **4** into the flanks of the piston **2**. It is possible to keep the filler member slightly larger in its outside diameter so that the control piston **10** will initially not come to abut against 61, but will do so only after the filler member has been introduced. Due to the spring action of the collar 57, existing tolerances may be bridged. The bolt 62 serving for the suspension of the filler member is chamfered at the ends on both sides and guided displaceably within a slot 63 of the axial disk 58.

The axial disk 58 comprises a recess 64, building up a pressure field, and a piston 56 which is sealed with the aid of a loop ring 66 is guided in the recess 64. The supply of the pressure field is effected by way of the bore 67. The piston 65 is provided annular so that only a limited friction arises between the piston and the axial housing part 68. During the deflection of the pinion shaft 3, the axial disk 58, by virtue of its almost floating position, can be additionally slid on to the pinion shaft 3 by the control piston 10. The contact surface of the axial disk 58 with the pinion shaft 3 may be lubricated with pressure oil through the bore 69.

In the above-described construction of the pump, the surface pressure between the internal gear and the filler member is reduced since the control piston 10 is supported not only upon the internal gear but, by way of the axial disks, also upon the pin thereof and the pinion shaft 3 so that the contact pressure is distributed to a greater area and the total wear and tear is reduced. Also taken into account is the problem that, in a newly installed pump, the internal gear does not simultaneously abut against the filler member with the control piston at the axial disks. If the filler member, when new, is provided with a larger outside diameter, the control piston 10 does not press upon the axial disk while the sealing of the pressure space is nevertheless assured. When the pump is placed in operation, an increased abrasion at the filler member (running-in abrasion) will therefore take place until the control piston with the collar thereof will abut against or bear on the axial disk.

FIG. 10 shows a further possibility for supporting the control piston 10 by way of the axial disk 58 on the pinion shaft 3 and the pin 59. The surface element 61 of the axial disk 58 is elastically provided by means of a bore 70 and a slot 71 so that, due to the resilience of the element 72, a compensation of the differences in the abutment of the internal gear against the filler member, or its contact therewith, and of the control piston 10 at and, respectively, with the axial disk 58 can take place.

A still further possibility of suspending the control piston 10 is illustrated in FIGS. 11 to 14. The axial disk 73 is positioned on the pinion shaft 3 and provided with a bore into which engages a pin 74 which, in turn, is positioned in the control piston 10. When a force is exerted toward the center of the housing by the pressure fields 75, the control piston 10 will press the internal gear 4 in a tooth engagement upon the flanks of the pinion 2 while simultaneously turning or rotating the axial disk 73 about the center 76 of the pinion shaft 3, until the internal gear 4 comes to rest against or bear against the filler member 5. This solution avoids the disadvantage which arose as a result of the heretofore known measures, and specifically that, during the abrasion of the filler member 5 and due to the change in the position of the internal gear center 77, a displacement of the gearing took place while, however, the control edges in the axial disk and in the control piston as well as the position of the pressure fields were maintained.

The schematic showing of FIG. 14 indicates these interrelations. When the filler member is worn, the internal gear 4 with its pitch circle or circle of contact 79 and/or 79′ rolls along on the pitch circle 78 of the pinion. The center 77 of the internal gear 2 executes at that time a rotary movement about the center 76 of the pinion shaft and travels toward point 77′. The pitch point, or instantaneous center of motion, 80 is shifted to 80′. By means of the axial disk 73 being rotatably suspended in point 76, the pin 74 and/or 74′ is moved, pulling behind the control piston, so that also the resultant 81 of the pressure fields is pulled behind about the angle α to 81′; in other words, the wear and tear of the filler member and the automatic re-setting of the internal gear have changed nothing with regard to the existing control and force conditions.

FIG. 12 is a cross-sectional view taken along line XII—XII of FIG. 11. The axial disk 73 is axially relieved with regard to the cover 83 by means of a pressure field 82 so that the re-setting of the axial disk 73, the internal gear 4 and the control piston 10 will not be met with or opposed by a high frictional resistance. The pressure piston 84 consists of an endless wire having a rectangular crosssection which is formed according to the recess 64 in FIG. 8 and sealed off by means of an O-shaped ring 85. The control piston 10 is secured in position with respect to the axial disk 73 by way of the pin 74 which, in the center of the control piston, can be readily adjusted to the gearing. The pin 74 projects with its extension 87 into a bore 88 of the cover 83. The collar 89 has play in the bore 88 so that the control piston 10 can execute together with the axial disk 73 a rotary movement about the pinion shaft 3 (see FIG. 14) until the collar 89 rests against or makes contact with the bore 88. The play of the collar 89 in the bore 88 is so designed that, following the abutment or bearing of the collar 89 in the bore 88, the filler member has run in and corresponds in its contour precisely to the crown lines or circles of the gears. In case of a pressure increase, both the pinion shaft 3 and also the filler member bolt 6 will deflect. In order to also compensate for the gap being formed during this deflection, the bolt is provided elastic. This may be accomplished, for example, in that the pin is weakened in its moment of resistance in the area of the extension 87 by a reduction of the crosssectional area.

FIG. 13 is equally a cross-sectional view taken along line XII—XII of FIG. 11. The difference as compared to FIG. 12 consists in the manner of positioning the axial disk 73: In this embodiment, the positioning is effected on a bearing bushing 90. The positioning of the axial disk 73 could, of course, also be made on a corresponding cylindrical shoulder portion of the cover 83.

FIG. 15 illustrates a modified embodiment of the pump shown in FIG. 11, wherein the control piston 10 has been indicated only schematically. This embodiment is based upon the following deliberation: As a result of the different operating conditions to which these pumps are subjected, for example pressure, rate of revolutions, temperature, viscosity, the hydraulic load on the filler memmber 5 varies, i.e. in the embodiments in which the control piston 10 is supported by way of the axial disks it is possible that the filler member is pressed to an increased extent against the crown circle diameter. This fact causes a wear and tear which can no longer be compensated by the compensating pressure fields of the control piston because of the support of the control piston on the pinion shaft 3 via the axial disks. In order to obviate this disadvantage, it is necessary that the filler member and the axial disk be so suspended that minor moments acting upon the filler member, the housing, the control piston and the pinion shaft are transferred without loading or charging the gliding surfaces of the filler member.

For this purpose, the axial disks 91 are rotatably positioned on the pinion shaft 3 and connected with the filler member 5 by means of a cylindrical pin 92, and the aggregate which is thus formed is supported against a housing pin 93 being secured to the housing. The filler member 5 is provided at the thicker end thereof with a surface 94 and bears therewith against a surface 95 of the housing pin 93 which is rotatably and/or displaceably positioned in bores of the axial housing parts so as to be adapted to execute the gliding movements that are required due to the deflection of the pinion shaft.

It should be understood of course that the foregoing disclosure relates to only preferred embodiments of the invention and that it is intended to cover all changes and modifications of the examples of the invention herein chosen for the purposes of the disclosure which do not constitute departures from the spirit and scope of the invention set forth in the appended claims.

What is claimed is:

1. Wear and tear-compensating high-pressure gear pump comprising a driven externally-geared pinion, a concomitantly rotating internally-geared wheel, a movable sickle-shaped filler member between the pinion and the internal gear and at least one axial disk covering the gears laterally, as well as an insert member, called "control piston," being disposed on the pressure side of the gears and being radially displaceably arranged with respect thereto, enclosing the outer circumferential surface of the internal gear in a specific angular range and touching the latter as the sole stator part, said control piston containing the first portion of the pressure outlet duct, being relieved for the major part with regard to radial forces and hence pressing against the circumference of the internal gear with only a limited amount of excess force so that as a result the internal gear is essentially supported only on the pinion, on the one hand, and on the filler member, on the other hand,
characterized in that the pump housing **1** is eccentrically bored with respect to the pinion shaft **3** in such a manner that the center M_G of the housing bore **7** is positioned on the hydraulic line of application or effective curve W_L which extends of necessity also through the internal gear center M_H, and in that the control piston **10** is accommodated in the sickle-shaped space being formed due to the eccentricity between the internal gear **4** and the housing bore **7**, being essentially adapted to the configuration thereof.

2. High-pressure gear pump according to claim **1**, characterized in that the control piston **10** encloses the internal gear **4** within an angular range of approximately 180°.

3. High-pressure gear pump according to claim **1** characterized in that hydrostatically relieved circular pressure fields **22** are provided in the outer circumferential surface of the control piston **10**, said pressure fields being formed by pistons positioned in the control piston and being in operative connection with the pressure space of the pump via bores **21**.

4. High-pressure gear pump according to claim **3**, characterized in that the edges of the pistons are chamfered, in that the rear of the pistons is charged by means of a charging spring and are in operative connection with the pressure space of the pump via a bore **19**.

5. High-pressure gear pump according to claim **1**, characterized in that the central pressure and, respectively, ring piston **29** is guided within the housing **1** and is provided at the inner end thereof with a spherical surface **30** engaging in a spherical indentation **31** of the control piston **10**.

6. High-pressure gear pump according to claim **1**, characterized in that the ring pistons **18** are replaced by sheet metal rings **36** having a beaded edge **35** and being supported via an O-shaped ring **38** and a plastic ring **39** each on a charging spring, said sheet metal ring **36** and said plastic ring **39** engaging each with one nose in a groove **34** of the annular recess or bore **32**, said sheet metal ring having a gap S with respect to the recess or bore.

7. High-pressure gear pump according to claim **6**, characterized by a tubular rivet **41** whose flange **42** retains the sheet metal ring **36** during the assembly thereof.

8. High-pressure gear pump according to claim **1**, characterized in that the pressure pistons **43** consist of plastic material and comprise a stud **45** being equipped with slots **44** and a collar **46**, and in that the bore **47** of the control piston **10** receiving the stud **45** is provided with a projection **48** locking behind the collar **46**, as well as with a groove **50** in which engages a cam **49** disposed at the stud **45**.

9. High-pressure gear pump according to claim **1**, characterized in that the control piston **10** is supported not only on the internal gear **4** but also via a collar **57** on the lateral axial disks **58** which rest with a bore on the pinion shaft **3**.

10. High-pressure gear pump according to claim **9**, characterized in that the filler member bolt **62** is displaceably positioned in the axial disks **58** and in that said axial disks **58** are suspended on a pin **59**.

11. High-pressure gear pump according to claim **9**, characterized in that the collar **57** is rendered elastic by means of a perforation or aperture **56** in the control piston.

12. High-pressure gear pump according to claim **9**, characterized in that the part **72** of the axial disks **58** against which the control piston **10** bears or abuts is provided elastic by means of a slot **71** which changes over into a bore **70**.

13. High-pressure gear pump according to claim **9**, characterized in that the filler member **5**, when new, has a slightly greater outside diameter than necessary so that the internal gear **4** being pressed on by the control piston **10** initially runs up on the filler member before the axial disk **59** is supported on the pinion shaft **3**.

14. High-pressure gear pump according to claim **1**, characterized in that the axial disks **73** are positioned on the pinion shaft **3** and comprise a bore into which a pin **74** engages, said pin being positioned in the control piston **10**.

15. High-pressure gear pump according to claim **1**, characterized in that the pin **74** engages with an extension **87** into a bore of the housing cover **83** and is equipped in the bores of the control piston **10** and of the housing cover **83** with a collar **86**, **89**, and in that it has within the area of the extension **87** a reduction of the cross sectional area.

16. High-pressure gear pump according to claim **1**, characterized in that the axial disks **91** are rotatably positioned on the pinion shaft **3** and are connected by means of a cylindrical pin **92** with the filler member **5**, the aggregate **5**, **91**, **92** thus formed being supported against a housing pin **93** being secured in the housing.

17. High-pressure gear pump according to claim **16**, characterized in that the filler member **5** is flattened at its thicker end **94** and rests or bears against the equally flattened but otherwise cylindrical housing pin **93**, the latter being movably positioned in bores of the axial housing parts.

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