May 19, 1964 P. MERCIER 3,133,447

RECIPROCATING ENGINE OR PUMP

Filed Feb. 17, 1958 4 Sheets-Sheet 1

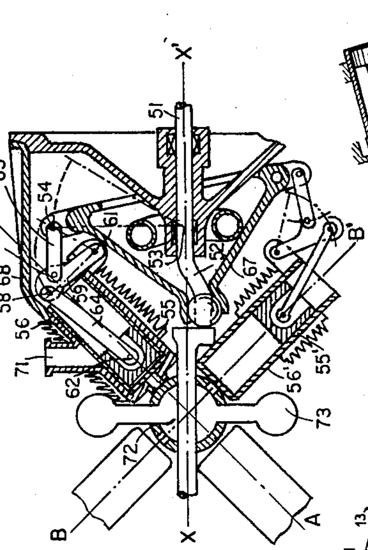


FIG. 3

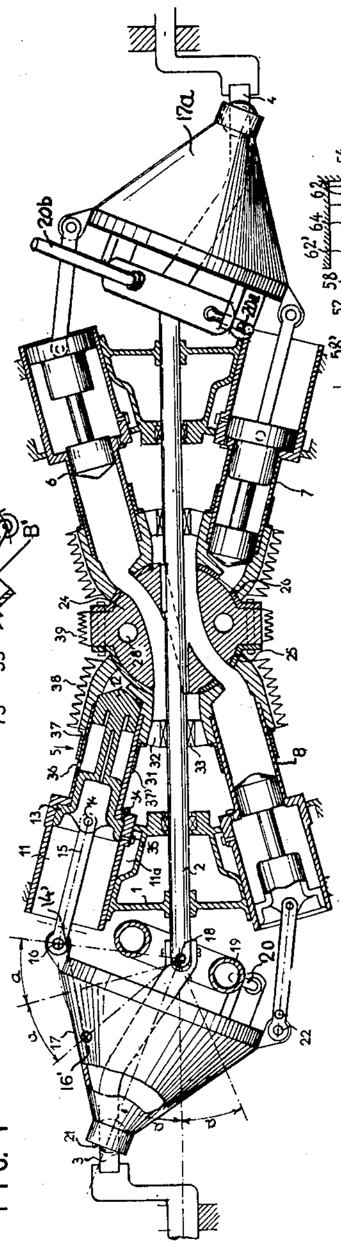


FIG. 1

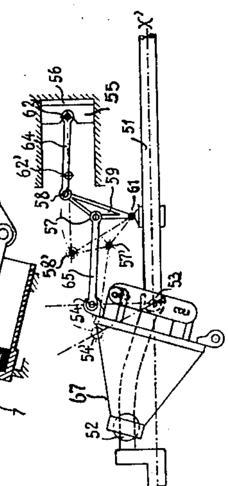


FIG. 2

May 19, 1964 P. MERCIER **3,133,447**

RECIPROCATING ENGINE OR PUMP

Filed Feb. 17, 1958 4 Sheets-Sheet 2

FIG. 5

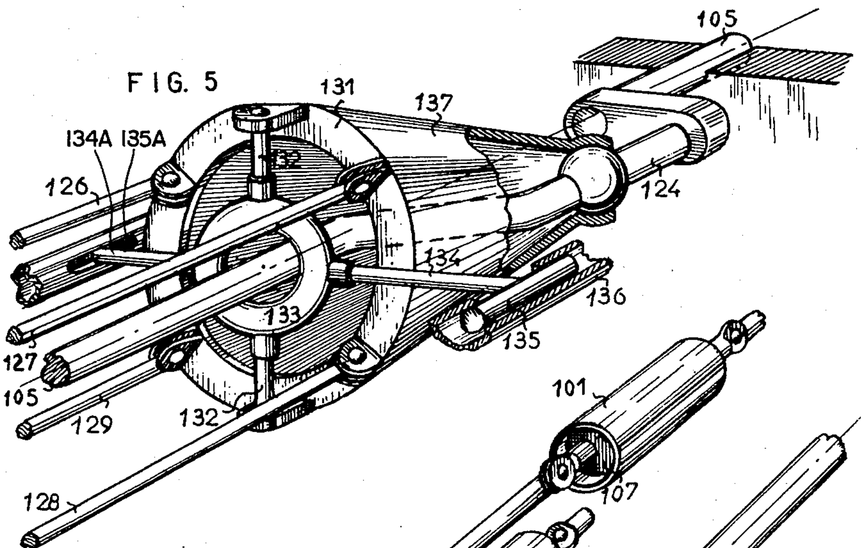
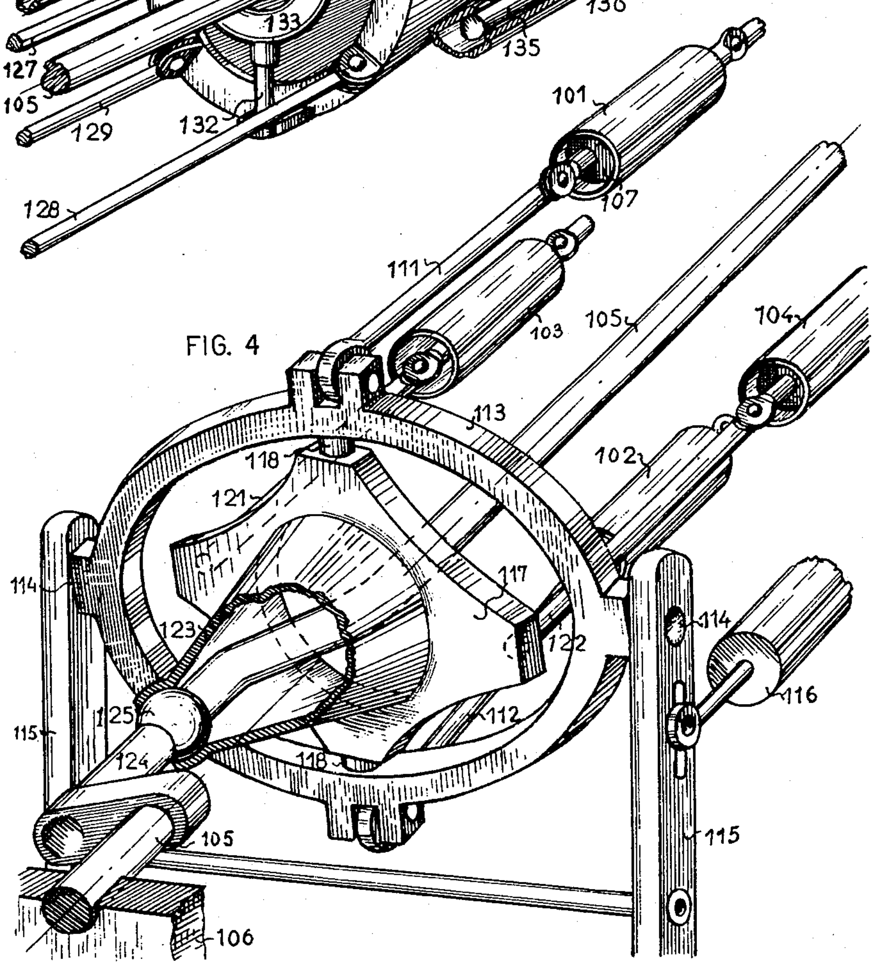


FIG. 4



May 19, 1964 P. MERCIER 3,133,447

RECIPROCATING ENGINE OR PUMP

Filed Feb. 17, 1958 4 Sheets-Sheet 3

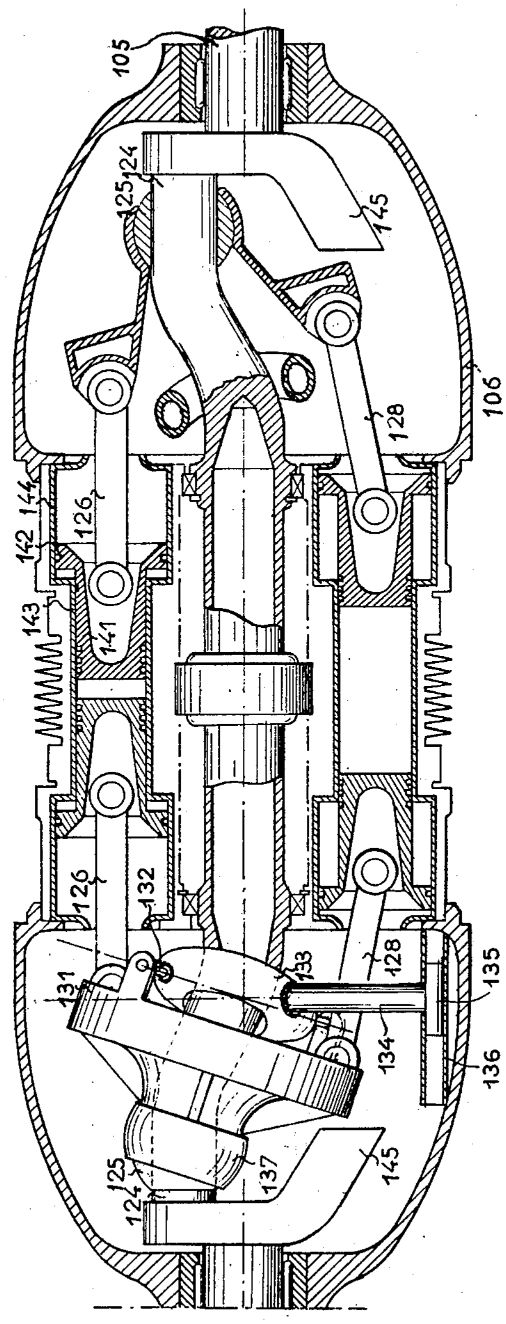


FIG. 6

May 19, 1964 P. MERCIER 3,133,447

RECIPROCATING ENGINE OR PUMP

Filed Feb. 17, 1958 4 Sheets-Sheet 4

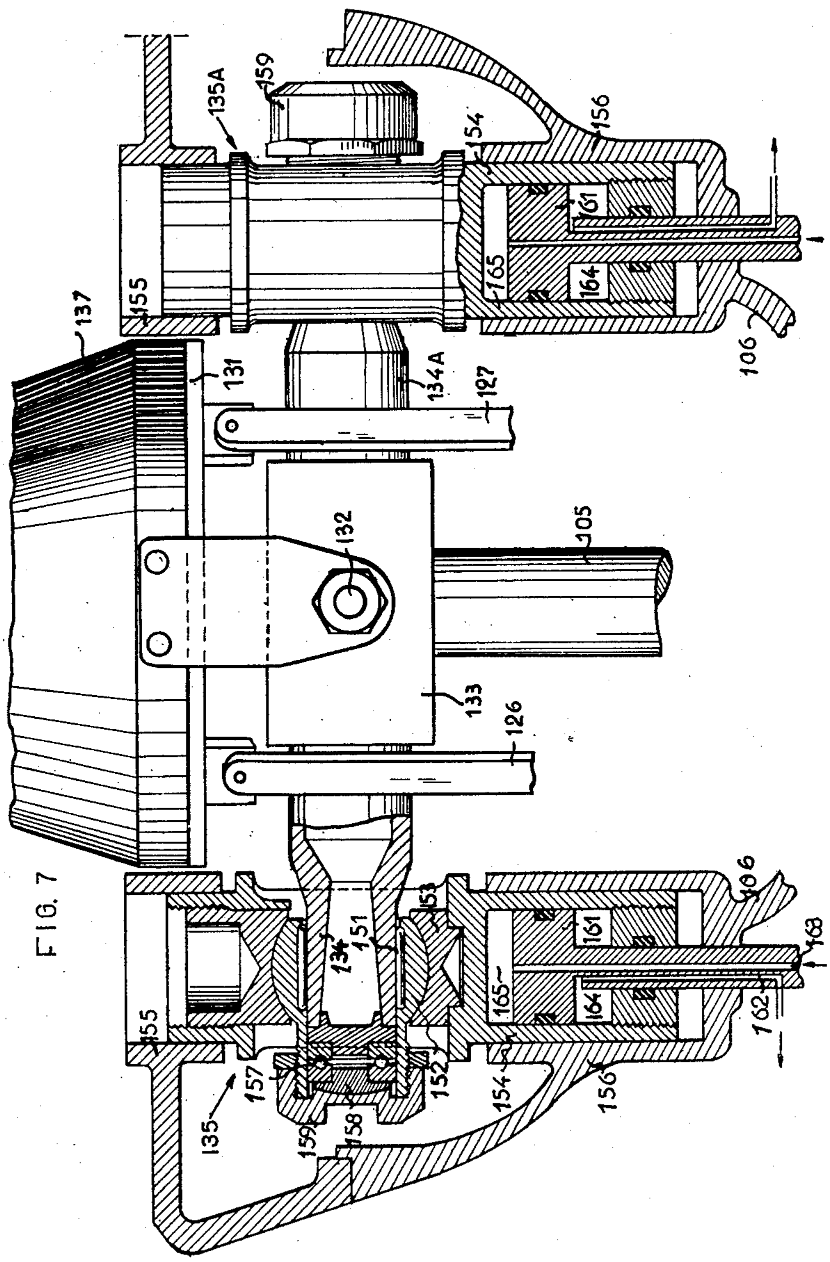


FIG. 7

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RECIPROCATING ENGINE OR PUMP

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4 Claims. (Cl. 74—60)

The invention relates to a power transmission mechanism of the wobbler or swash-plate type for internal combustion engines, pumps or compressors having a plurality of cylinders and pistons reciprocable therein. The swash plate is mounted for oscillatory movement without rotating relative to the pistons, and the pistons are connected to the swash plate whereby reciprocatory motion of the pistons is transformed into rotary motion of the crankshaft, or vice-versa.

An object of the invention is to provide a power transmission mechanism of the above-noted type of which the cylinders and pistons, connecting rods, swash-plate and crank-shaft are so designed as to ensure an optimum performance by causing the pistons to move at predetermined accelerations in the vicinity of their dead centers by providing means for adjusting the piston inner dead centers.

In known devices of this type, and in particular in two-stroke engines, each connecting rod is subjected to stresses of different values at the ends of its stroke respectively corresponding to the inner and outer dead centers of the associated piston. This is due to the fact that, in the vicinity of the inner dead center, the inertia of the moving parts and the motive forces, are in opposite directions, while in the vicinity of the outer dead center they have the same direction.

Under these conditions and, in particular, in a high speed engine, it is desirable to establish a discrepancy between the acceleration corresponding to the inner dead center position of the piston and that corresponding to the outer dead center. This can be effected by increasing the acceleration at the compression stroke, i.e., at the inner dead center and/or decreasing the acceleration at the outer dead center or exhaust stroke. Such a discrepancy also improves the engine performance, since it will reduce the time during which the hot gases will be trapped within the combustion chamber while lengthening the time of scavenging and gas exhausting.

In swash-plate mechanisms hitherto constructed, the swash-plate is mounted for oscillating movement either concentrically with the crankshaft or concentrically with an inclined crankpin, and the connecting rods of the pistons of the several cylinders are pivotally connected to the swash-plate at points spaced around its periphery. In operation, if the pistons are reciprocated in properly timed relation, their movement is transformed into a conical oscillation of the swash-plate around its center of oscillation, the points of articulation of the connecting rods to the swash-plate being constrained to describe an arc of a circle around said center whereby each piston arrives at the end of its compression stroke or expansion stroke when said point of articulation is situated at one end or the other of said arc respectively corresponding to the maximum inclined positions of the swash-plate. The arc is intersected by the axis of the corresponding connecting rod when the piston is at the end of its compression stroke and exhaust stroke so that the swash-plate will impart to the piston strokes of successive acceleration and deceleration as the articulation point moves along said arc of circle.

According to the present invention there is provided a swash-plate mechanism for engines, pumps or compressors having a plurality of cylinders and pistons reciprocable therein comprising a swash-plate mounted concentrically with the crankshaft so as to be capable of oscillating about a center and connected to a crank pin of the crankshaft by means of a universal joint of the spherical type, the points of articulation of the connecting rods to the swash-plate all lying in a transverse plane which is spaced from the center of oscillation. The distance between said plane and the center of oscillation is such that when a piston arrives at the end of its compression stroke, the axis of its connecting rod will extend substantially tangential to the arc of circle described by the corresponding articulation point during oscillation of the swash-plate, whereby the motion of the piston will have a substantially greater acceleration towards the end of its compression stroke than towards the end of its expansion stroke.

Thus, the invention provides a novel geometrical configuration permitting a differentiation between the acceleration of the piston at the end of its compression stroke and the acceleration of the piston at the end of its expansion stroke. Thus, the axial spacing between the plane containing the points of articulation and the center of oscillation results in a consequent shift of the arc of circle described by each point of articulation during oscillatory movement of the swash-plate. Moreover, the axes of the connecting rods, or at least a portion thereof, are arranged to extend substantially tangential to the corresponding arc of circle when the piston is at the end of its compression stroke.

A more particular object of the invention, with this purpose in view, is to arrange the engine cylinders along generatrix lines of a revolution conical surface.

Other objects and advantages of the invention will be hereinafter described with reference to the accompanying drawings, given merely by way of example.

In these drawings:

FIG. 1 is a diagrammatic partly axial sectional view of a two-stroke engine according to the invention with four cylinders.

FIG. 2 shows another embodiment comprising cranked connecting rods articulated on intermediate oscillating levers.

FIG. 3 is a diagrammatical partly axial sectional view of another two-stroke engine with four cylinders and cranked connecting rods according to FIG. 2.

FIG. 4 is a diagrammatic perspective view of an engine according to the invention including means for adjusting the position of the dead centers.

FIG. 5 shows a modification of FIG. 4.

FIG. 6 is a diagrammatical longitudinal sectional view of an engine having opposed pistons and provided with the arrangement shown in FIG. 5, and

FIG. 7 shows a practical construction of a portion of the engine of FIG. 6.

In the embodiment shown in FIG. 1 a two-stroke engine according to the invention comprises four cylinders having their axes in the plane of the drawing; it is clear that any number of similar groups of four cylinders may be arranged in a similar manner in other planes containing the longitudinal axis of the crank-shaft. In the example shown, the end-of-stroke accelerations are given different values for the inner and outer dead centers respectively by a suitable inclination of the cylinder axes with respect to the crank-shaft axis and by a shifting of the plane containing the articulation points of the connecting rods on the swash-plate with respect to the geometrical center of oscillation of said plate.

1 is the casing of the engine, and **2** is the crankshaft which is provided with two opposed crank-pins **3** and **4**. Four cylinders are shown at **5**, **6**, **7** and **8**. The four cylinders are obviously alike, as well as the parts cooperating with them, so that it will be sufficient to describe one cylinder and its associated parts.

The driving cylinder **5** is associated with a scavenging air cylinder **11**, coaxial therewith but having a greater diameter. There is provided a double piston comprising a piston member **12**, or head, slidably mounted in the driving cylinder **5** and a piston member **13**, or body, slidably mounted in the scavenging cylinder **11**. On the piston body **13** is pivoted at **14** the small end of a connecting rod **15**, the large end of which is pivoted at **16** on the swash-plate **17**. The swash-plate **17** is supported in the casing **1** so as to be freely tiltable around an oscillation point **18** on its axis which is also located on the axis of the crank-shaft **2**. The support of the swash-plate by the casing may be made for example, through a universal joint comprising an annular member **19** having two right-angled axles, one of which is journalled by diametrically opposite pivot pins **20** in the swash-plate **17** while the other one is journalled in the casing **1** by means of diametrically opposite pivot pins **20*a***, another point of the axis of the swash-plate being secured to the crank-pin **3** of the crank-shaft through a ball-and-socket joint **21**. The point **16** of the swash-plate at which the connecting rod **15** is pivoted, as well as the diametrically opposed point **22** at which the connecting rod of the cylinder **8** is pivoted, thus move in the vicinity of the plane containing the axes of both cylinders **5** and **8**, as well as the axis of the crank-shaft, as the crank-shaft rotates while the swash-plate describes its wobbling motion. According to a feature of the invention the plane containing the points **16** and **22** is shifted (towards the left in FIG. 1) with respect to the point of oscillation **18** of the swash-plate, which contributes to give to the end-of-stroke accelerations of the piston different values in the respective vicinity of the inner and outer dead centers, as explained hereafter.

In the position shown in the drawing, the piston head **12** lies at its inner dead center or compression stroke and the leverage between the points **16** and **18** of the swash-plate **17** is substantially right-angled to the direction of the cylinder axis while the connecting rod is substantially aligned with the cylinder axis.

For a position of the crank-shaft angularly spaced by 180° with respect to that shown in the drawing, i.e., when the piston will lie at its outer dead center or exhaust stroke, the axis of the swash-plate **17** will make an angle **2*a*** with respect to its position shown in the drawing, so that the above-mentioned leverage **18—16** will then also make the same angle **2*a*** with respect to its position shown in the drawing; then the point **16** will be brought to **16′** while the connecting rod will be shifted from the position **14—16** to the position **14′—16′**. It may be seen that the above-mentioned leverage, which will then lie at **18—16′**, will make the cylinder axis an angle having a value different from that of the angle the said leverage makes, as shown in the drawing, when the piston lies at its inner dead center. This angle difference results, on the one hand, from the fact that the cylinder axis is inclined with respect to the crank-shaft axis and, on the other hand, from the fact that the plane of the anchoring points of the connecting rods on the swash-plate **17** is shifted with respect to the point of oscillation of the latter. It is the angular relation of the aforenoted leverage that results in the desired accelerations.

FIG. 2 diagrammatically shows an alternative embodiment in which the axis of the crank-shaft **51** is shown at XX′, the crank-pin at **52**, the point of oscillation of the swash-plate at **53**, the connecting rod anchoring point at **54**, the associated piston at **55** and at **56** the cylinder in which said piston is slidably mounted, which has been assumed to be parallel with the axis XX′. The connecting rod interposed between the swash-plate and the piston is cranked and its two bending points are respectively pivoted at **57** and **58** on an intermediate lever **59** which is, in turn, pivoted at **61** on the engine casing.

At the inner dead center, the piston axis lies at **62** while at the outer dead center it lies at **62′**; then the intermediate lever lies at **57′**, **58′**, **61** and the swash-plate at **53**, **54′**. The end-of-stroke accelerations of the piston are a function of the inclination of the connecting rod section **54**, **57** with respect to the radius **53**, **54** of the swash-plate as well as with respect to the radius **61**, **57** of the intermediate lever and the inclination of the section rod **58**, **62** with respect to the radius **61**, **58** of the intermediate lever, as well as with respect to the axis of the cylinder **56**.

By varying the location of the points of articulation of these parts and the length of the various leverages, many combinations of piston acceleration in the vicinity of its inner dead center and its acceleration in the vicinity of its outer dead center are obtained. It is also possible to so dispose the cylinders that they be steeply inclined to the crank-shaft axis which offers important advantages for the structural design of the engine; finally, the oblique thrust of the piston on the cylinder walls may be reduced, which permits omitting guiding members.

FIG. 3 shows an engine designed according to the diagram of FIG. 2, but wherein, however, the cylinder axes, instead of extending along generatrix lines of a cylindrical surface coaxial with the crank-shaft, are disposed along generatrix lines of two vertically opposite conical surfaces generated by the two straight lines A, A′ and B, B′ symmetrical with respect to the axis XX′ of the crank-shaft and both intersecting the latter at the same point. In this figure there are shown, as previously, at **56** a cylinder, at **55** the associated piston, at **64** a section of the related connecting rod, at **59** an intermediate lever and its pivot **61**, at **65** another section of the connecting rod, at **53** the point of oscillation of the swash-plate **67** and, at **52**, the crank-pin of the crank-shaft **51**. The engine casing is indicated at **68**. On the axis B—B′ is diagrammatically shown a second cylinder **56′** and, on the other side of the general center O of the engine, are located two other cylinders (not shown) respectively symmetrical with the cylinders **56** and **56′**, with respect to the said center O. Thus, the engine shown in FIG. 3 includes at least four cylinders. It would be possible, furthermore, to dispose other pairs of cylinders on each of the two conical surfaces vertically opposed at O.

In the drawing, there is further shown at **71** the fluid outlet, at **72** a central distributor for the scavenging air, and at **73** a tore chamber communicating with the said sliding valve.

The piston **55** lies at its inner dead center while the piston **55′** is located at its outer dead center; the various parts respectively lie in positions similar to those of the parts shown in FIG. 2. It may be seen that the inclination of the connecting rod is negligible at explosion time when the piston lies at its inner dead center in spite of the very strong inclination of the axis of the cylinders on the crank-shaft axis. This arrangement permits of designing a high compact and, hence, very light engine.

In the embodiment shown in FIG. 4, a set of four cylinders **101**, **102**, **103**, **104** are arranged with their axes on four equidistant generatrices of a cylinder on the axis of which is mounted a crank-shaft **105** rotatably mounted in the engine casing only a small portion of which is indicated at **106** in the drawing. In each cylinder are arranged two opposed pistons, such as the piston **107** which is visible in cylinder **101**. Alternately, each cylinder could have one end closed and contain one piston only. The connecting rods **111** and **112** of two diametrically opposed pistons are pivoted at the ends of a diameter of a gimbal member **113** which is supported in the casing by two stub shafts **114** located on an axis at right angles to the straight line joining the two points of the gimbal member where the connecting rods **111**, **112** are pivoted.

The stub-shafts **114** are mounted on a supporting member which, in this example, consists in two arms **115** pivotally mounted in the casing of the engine about an axis parallel with the common axis of the stub-shafts, in such a manner that the latter may be shifted along over a short distance in a substantially rectilinear path parallel with the crank-shaft axis, for instance, under the action of a hydraulic jack indicated as at 116.

A swash-plate 117 with a central aperture for accommodating the crank-shaft, is arranged within the gimbal member 113 and is provided with two stub-shafts 118 which enable said swash-plate 117 to pivot within said gimbal member about an axis at right angles to the axis of the stub-shafts 114 of the gimbal member 113. The other two connecting rods are pivoted to the plate 117 respectively at the ends of a diameter at right angles to the axis of both stub-shafts 118 of the plate.

The swash-plate 117 is provided with a hub-shaped member 123 the end of which is connected to a crank-pin 124 of the crank-shaft 105 through a sleeve 125 the outer surface of which is spherical and fitted in a conjugated spherical housing in the hub-shaped member 123, and which has a cylindrical bore which enables it to slide along the rectilinear crank-pin 124 which is parallel with the general axis of the crank-shaft.

The assembly so far described could operate as a compressor but it is more particularly designed to be used as an internal combustion engine.

Under the influence of the pressures being successively exerted by the pistons, the outer end of the hub 123 drives the crank-pin in a circular path about the crank-shaft axis. Every axial reaction is absorbed by the casing, through the Cardan-like connection formed together by the plate 117 and the gimbal member 113. The connection between the hub and the crank-pin through the spherical sleeve with a cylindrical bore, is particularly desirable because the bearing surfaces, namely the spherical bearing surface between the hub and the sleeve, as well as the cylindrical bearing surface between the sleeve and the crank-pin are permanently moving one with respect to the other and, therefore, are subjected to a permanent grinding operation. No local wear of these surfaces, therefore, is to be expected, and the joint is able to operate for long periods of time without any material clearance taking place. Furthermore, the lubrication is correct, the unitary pressures are uniform and no excessive heating is to be feared. Finally, the slight possible distortions of the crank-pin, principally flexure distrotions, are absorbed by the spherical joint all the more easily that the surfaces in contact are already moving one with respect to the other, as already stated hereinabove.

Operation of the jack 116 makes it possible to move the gimbal member 113 along the crank-shaft axis. During this movement, the spherical sleeve 125 slides along the crank-pin, so that the only result of this adjustment is a shifting movement as a whole of all the pistons either in one direction or in the other. It is thus possible to increase or to reduce the "inner clearance" in the cylinders, according to the operating conditions of the engine.

FIG. 5 shows a modification of the embodiment of FIG. 4, which is different from the latter by the fact that the connecting rods 126, 127, 128, 129, on one side are pivoted to the plate 131, at the ends of two perpendicular diameters, said plate being supported by two diametrical stub shafts 132 of a cross-like gimbal member 133 the two further diametrical stub shafts 134, 134A of which are carried on the casing of the engine, through a supporting member which, in this example, consists in pistons 135, 135A adapted to slide in cylinders 136 extending in directions parallel with the crank-shaft axis. The axis of the stub shafts 132 of the cross member extends along the bisectrix of the angle formed by the diameters which join the points on the plate where the connecting rods are pivoted. The cross member 133 is centrally apertured for accommodating the crank-shaft. The plate 131 also is provided with a hub-shaped member 137 the outer end of which is connected to the crank-pin 124 of the crank-shaft 105 by a spherical sleeve with a cylindrical bore, as in the embodiment described hereinabove.

This device operates in the same manner as that of FIG. 4 with only the slight difference that, even practically, the center of the Cardan-like joint moves in a direction parallel with the crank-shaft axis instead of describing a short arc of a circle.

FIG. 6 shows a motor-compressor set designed according to the kinematic principle illustrated in FIG. 5. The crankshaft 105 is rotatably mounted in the casing 106 and is provided with two crank-pins 124 on which the two spherical sleeves 125 are slidably mounted. Each piston comprises two portions of different diameters, namely a portion 141 which serves as a power piston and a portion 142 of a larger diameter which serves as a compressor piston; these two pistons operate, respectively, in cylinders 143 and 144 of corresponding diameters. For the sake of clarity, some details have been omitted, namely the timing systems of the motor and of the compressor. Balancing weights rigid with the crank-shaft have been indicated as at 145.

FIG. 7, wherein the same reference numerals are used for corresponding elements of FIG. 5, shows a device for shifting the gimbal member 133 of FIG. 5 axially with respect to the crank-pin 124 to thereby adjust the compression ratio of the engine cylinders (not shown) corresponding to the connecting rods 126, 127, 128, 129. As shown in FIG. 7, the stub shafts 134 and 134A are mounted through self-aligning needle bearings 151, the spherical outer race 152 of which is mounted in a bearing 153 rigid with a double piston 154 adapted to have its end portions sliding in a bore 155 of a support cast in one piece together with the casing 106 of the engine. The weak axial component forces possibly exerted upon the stub shafts 134, 134A of the cross member are absorbed by a thrust ball bearing 157, through a spherical thrust-piece 158 which cooperates with a cup-member 159 screwed on a lateral extension of the sleeve 152.

Stub shaft 132 provides the same pivotal mounting between gimbal member 133 and plate 131 as shown in FIGURE 5. The axial shifting movement of each piston 154 is insured by a hydraulic system which comprises a stationary piston 161, the piston 154 being itself designed as a cylinder reciprocably mounted on the piston 161. Channels 162, 163 within the rod of the piston 161 ensure a communication between both chambers 164, 165 respectively, of the cylinder, either with a source of pressure fluid, or with a zone of low pressure, so as to cause said cylinder to move, either in one direction or in the other, as it is desired either to increase or to reduce the inner clearance in the cylinders 143 of the engine. The positioning of the inner dead centers of the pistons of the engine may be achieved in a positive manner or, else, with a given resiliency, in order automatically to limit the maximum effort exerted upon the pistons at the moment of the maximum power impulse. For this purpose, in order to drive the pistons 154, use may be made for instance, of an installation which comprises, in combination, a pressure liquid and a pressure gas, the latter being intended to serve as a resilient cushion under a pressure which corresponds to the pressure prevailing in the cylinders of the engine at the moment where the maximum permissible forces are being exerted.

In a general manner, in the above description, there have been disclosed practical and efficient embodiments of the invention, and it should be well understood that changes may be made in the arrangement, disposition and form of the parts without departing from the principle of the present invention as defined in the following claims.

What is claimed is:

1. In a reciprocating device comprising a crank-pin and a crankshaft, said crankshaft including an inclined crank-arm for supporting said crank-pin parallel to the axis of said crankshaft, a plurality of reciprocating pistons, each of said pistons disposed around said crankshaft so that at one end of a stroke, the piston lies at a greater axial distance from said crank-pin than at the other end

of the stroke, each of said pistons including a connecting rod pivotally connected thereto, a ball on said crank-pin, a wobble member including a bearing for forming a socket for said ball and a conical skirt portion surrounding said inclined crank-arm, said conical skirt portion including a base remote from said bearing, said ball and said bearing forming a spherical type universal joint between said crank-pin and said wobble member, means for supporting said wobble member for oscillating movement concentrically with said crank-arm about a point of oscillation, and means for pivotally connecting said connecting rods to said base of said conical skirt portion of said wobble member in a common transverse plane axially spaced from said point of oscillation in a direction towards said crank-pin, the distance between said transverse plane and said point of oscillation being such that when a piston arrives at the end of its stroke remote from said crank-pin, the respective connecting rod will extend along an axis substantially tangential to the arc of a circle described by the pivotal connection of the last mentioned connecting rod to said base during oscillation of the wobble member for causing the motion of said piston to have a substantially greater acceleration towards one end of its stroke than towards the other end.

2. In a reciprocating device comprising a crank-pin and a crankshaft in displaced parallel relationship, a crank-arm extending from said crank-pin to said crankshaft, said crank-arm being connected to said crankshaft to define a point of oscillation, a plurality of reciprocating pistons disposed around said crankshaft, each of said pistons having a stroke in a plane including the axis of said crankshaft and wherein each of said pistons at one end of a stroke is closer to said crank-pin than the other end of the stroke, each of said pistons including a connecting rod including first and second ends and means for pivotally connecting said first end of each connecting rod to a respective of said pistons, a ball on said crank-pin, a wobble member including a bearing for forming a socket for said ball and a conical skirt portion surrounding said inclined crank-arm, said conical skirt portion including a base, said ball and said bearing forming a spherical type universal joint between said crank-pin and said wobble member, and means for pivotally connecting said second ends of said connecting rods to said base of said conical skirt portion of said wobble member at locations lying in a common transverse plane axially spaced from said point of oscillation in a direction toward said crank-pin whereby said wobble member oscillates about said point of oscillation in response to the reciprocating movement of said pistons.

3. In a reciprocating device comprising a crank-pin and a crankshaft in displaced parallel relationship, a crank-arm including a first end rigidly connected to said crank-pin and a second end connected to said crankshaft and defining a point of oscillation, a plurality of reciprocating pistons, each of said pistons having a reciprocating stroke lying in a plane which includes the axis of said crankshaft and wherein one end of a stroke of the piston lies at a greater axial distance from said crank-pin and closer to said crankshaft than the other end of the stroke, each of said pistons including a connecting rod including a first end pivotally connected thereto and a second end, a ball on said crank-pin, a wobble member including a bearing for forming a socket for said ball and a conical skirt portion surrounding said inclined crank-arm, said conical skirt portion including a base, said ball and said bearing forming a spherical type universal joint between said crank-pin and said wobble member, and means for pivotally connecting the second ends of said connecting rods to said base of said conical skirt portion of said wobble member, whereby said wobble member oscillates about said point of oscillation in response to the strokes of said reciprocating pistons, said connecting rods being connected to said base at locations lying in a common transverse plane axially spaced from said point of oscillation in a direction towards said crank-pin for causing the motion of said pistons to have a substantially greater acceleration towards one end of its stroke than towards the other end.

4. In a reciprocating device comprising a crank pin and a crankshaft in displaced parallel relationship, a crank-arm including a first end rigidly connected to said crank-pin and a second end connected to said crankshaft, to define a point of oscillation, a plurality of reciprocating pistons, each of said pistons having a reciprocating stroke lying in a plane which includes the axis of said crankshaft and wherein one end of a stroke of the piston lies at a greater axial distance from said crank-pin and closer to said crankshaft than the other end of the stroke, each of said pistons includes a connecting rod including a first end pivotally connected thereto and a second end, a ball on said crank-pin, a wobble member including a bearing for forming a socket for said ball and a conical skirt portion surrounding said inclined crank-arm, said conical skirt portion including a base, said ball and said bearing forming a spherical type universal joint between said crank-pin and said wobble member, and means for pivotally connecting the second ends of said connecting rods to said base of said conical skirt portion of said wobble member whereby said wobble member oscillates about said point of oscillation, said connecting rods being connected to said base at locations lying in a common transverse plane axially spaced from said universal coupling means in a direction towards said crank-pin, the distance between said transverse plane and said point of oscillation being such that when a piston arrives at the end of its stroke remote from said crank-pin, the respective connecting rod will extend substantially tangential to the arc of a circle described by the pivotal connection of the last mentioned connecting rod to said base during oscillation of the wobble member for causing the motion of said piston to have a substantially greater acceleration towards one end of its stroke than towards the other end.

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