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

(11) Publication number: 0 183 258
A2

(12) EUROPEAN PATENT APPLICATION

(21) Application number: 85115091.2

(22) Date of filing: 28.11.85

(51) Int. Cl.⁴: F 01 B 11/08
F 04 F 11/02

(30) Priority: 28.11.84 JP 251414/84

(43) Date of publication of application:
04.06.86 Bulletin 86/23

(84) Designated Contracting States:
DE GB

(71) Applicant: Daikichiro, Isogai
305 Yanagisawa Haitsu 3-252-1, Nakakiyoto
Kiyose-shi Tokyo(JP)

(72) Inventor: Daikichiro, Isogai
305 Yanagisawa Haitsu 3-252-1, Nakakiyoto
Kiyose-shi Tokyo(JP)

(74) Representative: Lehn, Werner, Dipl.-Ing. et al,
Hoffmann, Eitle & Partner Patentanwälte Arabellastrasse
4 (Sternhaus)
D-8000 München 81(DE)

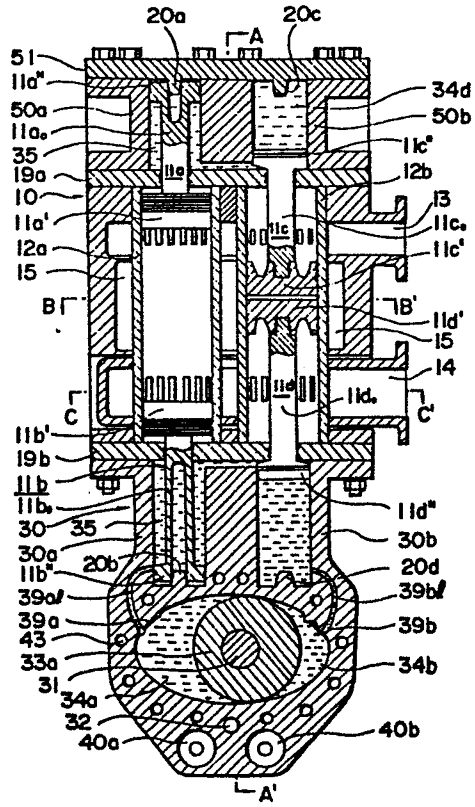
(54) Free piston engine having a hydraulic power transmission mechanism.

(57) A free piston engine having a hydraulic power transmission mechanism. The engine includes at least one free piston (11a, 11b, 11c, 11d) adapted to reciprocate linearly, a rotary piston (33a) rotatably mounted on the crank pin of the crankshaft (31) to rotate about the crank pin and a hydraulic power transmission system (34a, 34b, 34c, 34d, 35). One end face of the or each free piston is exposed to pressure of combustion gas or compressed gas, whereas the other end face of the same is exposed to hydraulic pressure. The rotary piston (33a) has the contour of epitrochoidal curve as seen in the cross-sectional plane. Working hydraulic medium is filled in a space (35) of the hydraulic system as defined between the other end face of the free piston and the rotary piston so as to convert linear reciprocable movement of the free piston (11a, 11b, 11c, 11d) to rotational movement of the crankshaft (31).

EP 0 183 258 A2

./...

FIG. 4



20a
20c
A
A'
51
11a"
50a
11a0
35
19a
10
11a'
12a
15
B
B'
C
C'
11b'
19b
11b
11b0
30
30a
11b"
20b
39a/
39a
43
33a
31
34a
32
40a
34d
50b
11c"
12b
13
11c0
11c'
11d'
14
11d0
11d"
30b
20d
39b/
39b
34b
40b
11a
11c
11d

0183258

FREE PISTON ENGINE HAVING A HYDRAULIC POWER TRANSMISSION MECHANISM

The present invention relates to a free piston engine having a hydraulic power transmission mechanism.

An object of the invention to provide a free piston engine having a hydraulic power transmission mechanism which is simple in structure and requires little space.

It is another object of the invention to provide a free piston engine having a hydraulic power transmission mechanism which is operated at a high efficiency with low vibration.

It is another object of the invention to provide a free piston engine having a hydraulic power transmission mechanism which can be manufactured at low cost.

According to the invention, there is provided a free piston engine having a hydraulic power transmission mechanism characterised by: at least one free piston one end face of which is exposed to pressure of combustion gas or compressed gas and the other end face of which is exposed to hydraulic pressure, said free pistons being adapted to reciprocably move in a cylinder; a rotary piston rotatably mounted on a crank pin of a crankshaft to rotate about the crank pin, said rotary piston having the contour of epitrochoidal curve as seen in the cross-sectional plane; and a hydraulic power transmission system having working hydraulic medium filled in a space defined between the other end face of the free piston and the rotary piston so as to convert linear reciprocable movement of the free piston to rotational movement of the crankshaft.

Other objects, features and advantages of the present invention will become readily apparent from reading of the following description of preferred embodiments of the invention in conjunction with the accompanying drawings, in which:

Figs. 1 (a) to (e) are a schematic side view of a single knot epitrochoidal rotary body, particularly illustrating how it operates in the housing;

Figs. 2 (a) to (e) are a schematic side view of a combination of ring gear and pinion which are brought in meshing engagement along the inner periphery of the ring gear respectively, particularly illustrating how the pinion operates in the ring gear, wherein a ratio of diameter of pitch circle of the ring gear to diameter of pitch circle of the pinion is 2 : 1;

Figs. 3 (a) to (c) are a schematic side view similar to Figs. 2 (a) to (e) respectively, particularly illustrating the pinion operates in the ring gear, wherein a ratio of diameter of pitch circle of the ring gear to diameter of pitch circle of the pinion is 5 : 4;

Fig. 4 is a vertical sectional view of a free piston engine in the form of oppositely located piston type two cycle linearly reciprocable engine in accordance with an embodiment of the invention;

Fig. 5 is a vertical sectional view of the free piston engine taken on line A - A' in Fig. 4;

Fig. 6 is a cross-sectional view of the free piston engine taken on line B - B' in Fig. 4; and

Fig. 7 is a cross-sectional view of the free piston engine taken on line C - C' in Fig. 4.

The principle of operation of the engine is based on the conversion of reciprocable movement of the piston, through a hydraulic system, to rotary movement of the crankshaft and vice versa.

As is well known, the method of transmitting movement of the piston to the crankshaft has been established for the conventional reciprocable piston engine.

On the other hand, the pistons in the engine of the invention act as free pistons of which one end face serves as cover for inhibiting leakage of working liquid, for instance, hydraulic oil.

An epitrochoid rotary piston to be described later carries out quasi-reciprocable movement as the crankshaft rotates and its effective volume varies in the sine-curved fashion. When pressure is exerted on the epitrochoid piston, the crankshaft is caused to rotate in such a direction that the aforesaid pressure decreases.

Now, it is assumed that one of both the opened ends of the cylindrical space is closed with the rotary piston and the other one of the same is closed with the free piston so that the thus closed space is filled with working liquid. As long as the free piston operates in such a manner as to normally impart thrust to the working liquid and a volume of working liquid is kept constant at all time, it is possible to convert reciprocable movement of the free piston to rotary movement of the crankshaft. This means that the functioning is similar to that of a conventional connecting rod.

Principles of operation of the engine according to the present invention will be described below with reference to Figs. 1 to 3 of the accompanying drawings.

When it is assumed that a rotary body has the crosssectional configuration comprising an epitrochoidal curve which extends around the crank pin of the crankshaft, a predetermined relation is established between ratio of the number of revolutions of the crankshaft to the number of revolutions of the epitrochoidal body and direction of rotation by employing a train of phase gears. Further, when the extent of eccentricity of the epitrochoidal rotary body is equalized to an extent of eccentricity of the crank pin, it results that the epitrochoidal rotary body acts as a rotary piston. Fig. 1 schematically illustrates the case where a single epitrochoidal rotary body is rotatably mounted on the crank pin of the crankshaft, wherein a mark ● designates a center of rotation of the crankshaft and a mark ○ does a center of rotation of the epitrochoidal rotary body. Further, reference letter R designates a radius of generation of epitrochoid and reference letter e does an extent of eccentricity. When the crankshaft rotates by one revolution in the clockwise direction as shown in Figs. 1 (a) to (d), the rotary body is caused to rotate around the crank pin by two revolutions in the anticlockwise direction. In other word, the crankshaft rotates in the clockwise direction and the epitrochoidal rotary body does in the anticlockwise direction by the same extent. It should be noted that the epitrochoidal rotary body carries out a kind of reciprocable movement relative to the crankshaft during their rotation as mentioned above. To facilitate understanding of the invention the epitrochoidal rotary body is referred to below as epitrochoidal rotary piston or simply as rotary piston.

As is well known by any expert in the art, sealing members adapted to come in slidable contact with the rotary piston are secured not to the rotary piston but to the housing in which the rotary piston is accomodated.

As mentioned above, the engine is provided with phase gears in order to associate the number of revolutions of the crankshaft with the number of revolutions of the epitrochoidal rotary piston.

Description will be made below as to the phase gears with reference to Figs. 2 and 3.

As shown in Fig. 2 which schemically illustrates the case where a pinion (smaller diameter gear) having a pitch circle of which diameter is determined to 1/2 $d\ell$ comes in rolling contact with a stationary gear wheel (ring gear) having a pitch circle of which diameter is determined to $d\ell$, the pinion is caused to rotate as shown in Figs. 2 (a) to (d) when it is operatively connected to the epitrochoidal rotary piston. Thus, it can be concluded that movement of the pinion satisfactorily meets the requirement as mentioned with respect to the movement of the rotary piston as shown in Fig. 1. (In other word, the phase relation between the crankshaft and the pinion in Fig. 2 corresponds exactly to the phase relation between the crankshaft and the epitrochoidal rotary piston in Fig. 1.) In the illustrated case difference between diameter $d\ell$ of the pitch circle of the ring gear and diameter ds of the pitch circle of the pinion is determined to be two times as wide as the extent of eccentricity e of the epitrochoidal rotary piston and this dimensional difference constitutes an essential condition for the present invention.

Namely, the following equations are established.

$$d\ell - ds = 2e \qquad \text{-----} \qquad (1)$$

$$d\ell = 2ds \qquad \text{-----} \qquad (2)$$

Since the dimension of ds should be determined larger than the diameter of the pin of the crankshaft, there is a limitation with respect to the dimensional lower end side. Once the dimension of ds is determined, the extent of eccentricity e is automatically determined from both the equations (1) and (2).

However, it is found that dimensions of an ideal rotary piston can not always be determined, because determination of dimensions $d\ell$, ds and e made in that way leads to restriction of designing of the epitrochoidal rotary piston (particularly, determination of thickness of the rotor in the axial direction), as mentioned below. A countermeasure for resolving the problem will be described below.

In principle, it is known that a volume of discharged oil or displacement volume V of the epitrochoidal rotary piston is represented by the following equation, when a plurality of working chambers whose number is represented by Z are formed in the space as defined between the epitrochoidal rotary piston and the cylinder (see, for instance, "The Wankel R C Engine" authored by R. F. Ansdale).

$$V = e \cdot R \cdot B \cdot \frac{4\,Z}{Z-1} \sin\frac{\pi}{Z} \quad \ldots\ldots \quad (3)$$

where e: extent eccentricity

R: generation radius of epitrochoid

B: length (width) as measured in the axial direction

In the case where the epitrochoidal rotary piston is a single knot epitrochoidal rotary piston as shown in Fig. 1, the number of Z amounts to 2. Thus, the equation (3) can be represented in the modified manner as noted below.

$$V = 8 \cdot e \cdot R \cdot B \qquad \text{-----} \qquad (4)$$

Accordingly, when the extent of eccentricity e in the equation (4) is determined, the value of R is determined correspondingly because there is a necessity for practically determining a trochoid constant K (= R/e) more than a certain value. Thus, the value of B could become excessively small from the viewpoint of manufacturing, when a certain value is given to the volume of discharged oil V.

In order to obviate the foregoing problem it is not necessarily required that the diameter $d\ell$ of pitch circle of the gear wheel is determined two times as wide as the diameter ds of pitch circle of the pinion but dimensional determination should be made in such a manner that the epitrochoidal rotary piston carries out movement as shown in Fig. 1. This can be achieved by allowing the gear wheel to rotate at a certain rate relative to the crankshaft without any necessity for immovably holding the gear wheel. In this case it should be noted that the dimensional relation as represented by the equation (1), that is, $d\ell - ds = 2e$ is maintained unchanged at all time.

Once the diameter ds of pitch circle of the pinion is determined on the basis of dimension of the crankshaft and the dimension e is determined on the basis of volume of discharged oil, the diameter $d\ell$ of pitch circle of the gear wheel is determined automatically because there is a necessity for meeting the requirement associated with the relation as represented by the equation (1). At this moment the equation (2) fails to be established and therefore decision should be made as to at what rate the gear wheel should rotate relative to the crankshaft in order to maintain the operative relation between the pinion and the crankshaft.

Next, description will be made below as to an example of the above-mentioned relation with reference to Fig. 3.

For instance, it is assumed that a ratio of the diameter $d\ell$ of pitch circle of the gear wheel (ring gear) to the diameter ds of pitch circle of the pinion (smaller diameter gear) is determined as noted below.

$$d\ell : ds = 5 : 4$$

In this case the same results are obtainable as the case where the above-mentioned ratio is determined as noted below, that is,

$$d\ell : ds = 2 : 1$$

while the gear wheel is caused to rotate in the reverse direction relative to the crankshaft, that is, in the anticlockwise direction at the number of revolutions equal to 3/4 times as high as the number of revolutions of the crankshaft as well as the case where the gear wheel is held immovably. When it is assumed that points at which the periphery of pitch circle of the pinion is equally divided into four segments are identified by P_1, P_2, P_3 and P_4 and points at which the periphery of pitch circle of the gear wheel is equally divided into five segments are identified by P_1', P_2', P_3', P_4' and P_5', the following equations are established.

$$\widehat{P_1P_2} = \widehat{P_2P_3} = \widehat{P_3P_4} = \widehat{P_4P_1}$$

$$= \widehat{P_1'P_2'} = \widehat{P_2'P_3'} = \widehat{P_3'P_4'} = \widehat{P_4'P_5'} = \widehat{P_5'P_1'}$$

As the crankshaft rotates in the clockwise direction, the pinion is caused to rotate in the anticlockwise direction. When movement is effected from the position as shown in Fig. 3 (a) to the position as shown in Fig. 3 (b), the crankshaft rotates about the center C' by angle of $360° \div 5 = 72°$ and the pinion rotates about the crank pin C by an angle of $72° - 90° = -18°$, that is, rotates by an angle of 18° in the anticlockwise direction. When the crankshaft continues to rotate further by one revolution, the pinion is caused to rotate by an angle $18° \times 5 = 90°$ in the anticlockwise direction. Due to the fact that as the crankshaft rotates by one revolution, the pinion rotates by one revolution (360°) in the case as shown in Fig. 1, it is required that the gear wheel rotates by an angle of 270°, that is, by 3/4 revolution in the anticlockwise direction during rotation of the crankshaft by one revolution in the clockwise direction in the case as shown in Fig. 3 in order that the operative relation between both the gears in Fig. 1 corresponds to that in Fig. 3. When a ratio of diameter of pitch circle of the gear wheel to diameter of pitch circle of the pinion is selected to another value different from the above-mentioned one, it is naturally required that the gear wheel rotates at a different rate of rotation.

Thus, the same results as those in Fig. 1 are obtainable by rotating the gear wheel at a properly selected rate corresponding to a certain ratio of diameter of pitch circle of the gear wheel to diameter of pitch circle of the pinion.

Next, description will be made below as to a hydraulic circuit.

The engine of the invention includes a space filled with hydraulic medium in the area located between reciprocable free pistons and epitrochoidal rotary pistons in order to transmit generated power from the free pistons to the rotary pistons. The space of which both opened ends are closed by the free piston and the rotary piston is called below as "working hydraulic chamber". This working hydraulic chamber should meet the following two requirements.

(a) Each of the free pistons functions to impart thrust to working hydraulic oil at all time so that hydraulic pressure is always kept at a level higher than a predetermined pressure to prevent from occurring air bubbles in the oil of the working hydraulic chamber.

(b) Working hydraulic oil is replaced with new one by a predetermined volume.

These requirements can be satisfactorily met by means of the following structure.

(a') The free piston is so designed that the diameter of the intermediate part thereof constituting an intermediate rod is determined smaller than that of the piston's end face which is exposed to hydraulic pressure. The intermediate rod extends through an opening of the intermediate wall whereby a cylindrical space is formed between the inner wall of the hydraulic cylinder and the intermediate rod. The cylindrical space is supplied with pressurized hydraulic oil having a predetermined pressure which is transported from an accumulator or the like. Thus, the free piston is affected by pressure which is effective in such a direction as to impart thrust to working hydraulic oil. As a result the requirement (a) can be met satisfactorily. This cylindrical space which is affected by hydraulic oil having a predetermined hydraulic pressure is called below as "predetermined pressure hydraulic chamber".

(b') As oil in the working hydraulic chamber is extracted therefrom by means of a metering pump in order to meet the requirement (b), the lower end of the free piston is caused to gradually descend toward the epitrochoidal rotary piston. When it reaches a predetermined lower positional limit, communication is established between the predetermined pressure hydraulic chamber and the working hydraulic chamber by way of a hydraulic oil passage in the cylinder whereby hydraulic pressure in both the hydraulic chambers is equalized. As a result the free piston stops its movement under the influence of pressure which acts against gas pressure. At this moment a certain volume of hydraulic oil flows from the predetermined pressure hydraulic chamber to the working hydraulic chamber while the free piston is held at the position located in the vicinity of its lower positional limit. As a result the requirement (b) can be met satisfactorily.

Principle of operation and structure of the engine of the invention can be applied to a variety of existent reciprocable piston engines. To facilitate understanding of the invention, an oppositely located piston type two cycle diesel engine will be employed below as typical example. As is well known, two cycle diesel engine of this type has many advantageous features such as no necessity for valve mechanism, excellent dynamic balance, excellent scavenging performance, possibility of carrying out supercharging by adjusting relative phase of the oppositely located pistons or the like. On the other hand, it has drawbacks such as necessity for two crankshafts, a train of gears for operatively connecting the crankshafts to one another or the like.

However, the foregoing drawbacks can be obviated by employing the hydraulic mechanism of the invention which will be described next.

Now, the present invention will be described in a greater detail hereunder with reference to Figs. 4 to 7 which schematically illustrate a preferred embodiment thereof.

The engine of the invention includes a cylinder housing 10 in which cylinders 12a and 12b are fitted. Intake holes on both the cylinders 12a and 12b are communicated with a scavenging pipe 13, whereas exhaust holes on the same are communicated with an exhaust pipe 14. The middle part of each of the cylinders 12a and 12b is surrounded by a cooling water passage 15. As will be best seen from Fig. 6, the middle parts of the cylinders 12a and 12b are communicated with pre-combustion chambers 17a and 17b.

An intermediate wall 19a, a top housing 50 and a top cover 51 are arranged one above another on the cylinder housing 10, whereas an intermediate wall 19b and an output section housing 30 are arranged below the lower end of the cylinder housing 10. The top cover 51, the top housing 50, the intermediate wall 19a, the cylinder housing 10, the intermediate wall 19b and the output section housing are firmly fastened by means of a plurality of tightening bolts 53 to build an integral structure. The top housing 50 is provided with hydraulic cylinders 50a and 50b which are located in vertical alignment with the cylinders 12a and 12b, whereas the output section housing 30 is provided with hydraulic cylinders 30a and 30b which are located in vertical alignment with the cylinders 12a and 12b.

Piston portions 11a', 11b' 11c' and 11d' adapted to receive gas pressure are accomodated in the cylinders 12a and 12b as well as the cylinders 12a and 12b and piston portions 11a", 11b", 11c" and 11d" adapted to be exposed to hydraulic pressure are accomodated in the hydraulic cylinders 50a, 50b, 30a and 30b of both the top housing 50 and the output section housing 30. The piston portions as mentioned above are connected to one another via intermediate rods 11ao, 11bo, 11co and 11do whereby free pistons 11a, 11b, 11c and 11d are built.

The intermediate walls 19a and 19b have holes formed thereon through which the intermediate rods 11ao, 11bo, 11co and 11do are slidably displaced and each of the holes is fitted with an O-ring for the purpose of inhibiting an occurrence of oil leakage.

A predetermined pressure hydraulic chamber 35 is formed in the space as defined between the hydraulic cylinders 30a, 30b, 30c and 30d and the intermediate rods 11ao, 11bo, 11co and 11do. As is apparent from Fig. 5, each of the hydraulic chambers

35 is communicated with hydraulic pressure source such as accumulator or the like (not shown) via an inlet port 18 and a hydraulic passage 35 extending through the cylinder housing 10 in the vertical direction.

An output shaft 31 extends through the output section cylinder 30 in the horizontal direction and epitrochoid rotary pistons 33a and 33b are rotatably mounted on the crank portion of the output shaft 31. The one rotary piston 33a defines working hydraulic chambers 34a and 34b between both the free pistons 11b and 11d.

The other rotary piston 33b defines working hydraulic chambers 34c and 34d between both the free pistons 11a and 11c.

Hydraulic oil in the working hydraulic chambers 34a and 34c is extracted therefrom by a predetermined volume by operating a metering pump 40a for the purpose of inhibiting an occurrence of deterioration of hydraulic oil and the situation is the same with hydraulic oil in the working hydraulic chambers 34b and 34d which is extracted therefrom in the same manner.

As a predetermined volume of oil in the working hydraulic chambers 34a and 34b is extracted by operating the metering pumps 40a and 40b, the free pistons 11b and 11d are caused to descend toward the rotary pistons. When they reach the predetermined position, oil flows from the predetermined pressure hydraulic chamber 35 into the working hydraulic chambers 34a and 34b via passage holes 39aℓ and 39bℓ.

When hydraulic pressure becomes equalized, the free pistons 11b and 11d stop their descending movement. In order to prevent oil from flowing in the reverse direction through the passage holes 39aℓ and 39bℓ, nonreturn valves 39a and 39b are fitted in the passage holes 39aℓ and 39 bℓ. The same arrangement is made for the free pistons 11a and 11c.

In the drawings reference numerals 20a, 20b, 20c and 20d each designate a respective mechanical stop which serves to stop further upward and downward movement of the free pistons. These stops provide oil by-pass holes not to disturb the flow.

Since a volume of oil extracted by means of the metering pump per one cycle of the free piston is very little compaired with cubic volume of each of the working hydraulic chamber, it can be considered that a volume of oil required for one cycle is kept substantially constant. Accordingly, movement of the free pistons is exactly converted into displacement of the rotary pistons, that is, rotation of the crankshaft. Although the rotary pistons 33a and 33b produce phase difference to some extent for the purpose of supercharging, the free pistons 11a and

11b located opposite to one another are caused to simultaneously come close to one another or move away from one another.

The rotary pistons 33a and 33b are provided with phase gears 38a and 38b in the form of pinion at the outer end part of their axial extension. The pinions 38a and 38b mesh with phase gears 37a and 37b (having a larger diameter) in the form of ring gear along the inner periphery of the latter. The gears 37a and 37b are rotatably mounted in such a manner as to rotate relative to the output shaft 31. Further, the gears 37a and 37b mesh with gears 36d and 36e which are fixedly mounted on an auxiliary shaft 32 extending in parallel with the output shaft 31. Another gear 36c fixedly mounted on the auxiliary shaft 32 is brought in meshing engagement with a gear 36a fixedly mounted on the output shaft 31 via an intermediate gear 36b. A series of speed reduction gears as mentioned above are accomodated in the space as defined between the side covers 41a and 41b for the working hydraulic chambers and the end covers 42a and 42b.

A speed reduction ratio is properly determined for each of the gears in accordance with the principle as described above.

Finally, advantageous features of the invention will be described below.

(1) The engine of the invention has the following advantages compaired with the conventional oppositely located piston type two cycle diesel engine.

1) The number of crankshafts can be reduced from 2 to 1. Further, since a distance of displacement of the crank section is short in the same way as in the case of Wankel engine, the crankshaft can be constructed in the structure having an excellently high mechanical strength.

2) The engine of the invention does not require a train of connection gears arranged between the crankshafts which are used for the conventional engine.

3) Any connecting rod for operatively connecting piston to crankshaft is not required.

4) Since volumeric variation of the rotary piston is achieved in the sine-curved pattern, no harmonic component is produced during reciprocable movement of the free piston, resulting in vibration being reduced remarkably.

(2) In the case where the power transmission mechanism is employed for the conventional serially arranged four cylinder engine it results that the cylinders are arranged not in the four series type but in the two parallel-two series type. This leads

to an advantage that the length of the engine can be reduced remarkably in the direction of the output shaft with generation of vibration being minimized.

While the present invention has been described above merely with respect to a single preferred embodiment thereof, it should of course be understood that it should not be limited only to this but various changes or modifications may be made in any acceptable manner without departure from the scope of the invention.

Claims:

1. A free piston engine having a hydraulic power transmission mechanism characterised by:

at least one free piston (11a,11b,11c,11d) one end face of which is exposed to pressure of combustion gas or compressed gas and the other end face of which is exposed to hydraulic pressure, said free pistons (11a, 11b,11c,11d) being adapted to reciprocably move in a cylinder (12a,12b);

a rotary piston (33a,33b) rotatably mounted on a crank pin of a crankshaft (31) to rotate about the crank pin, said rotary piston (33a,33b) having the contour of epitrochoidal curve as seen in the cross-sectional plane; and

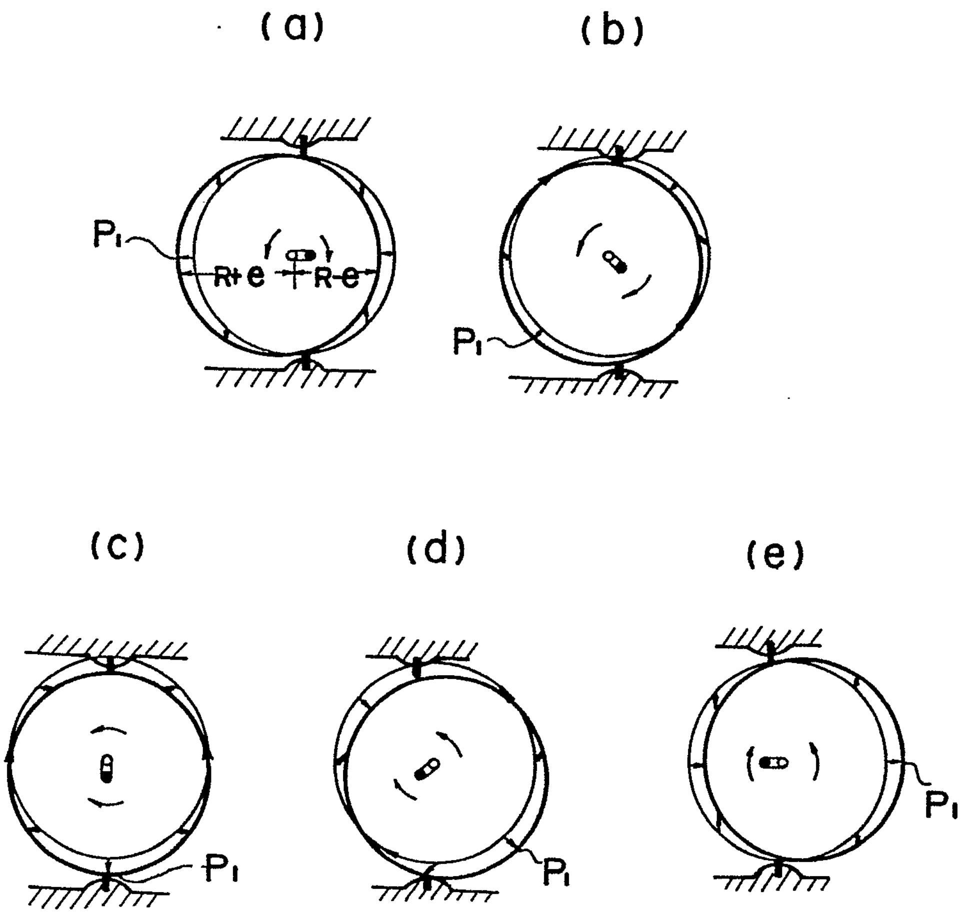
a hydraulic power transmission system (34a,34b,34c, 34d,35) having working hydraulic medium filled in a space defined between the other end face of the free piston (11a,11b,11c,11d) and the rotary piston (33a,33b) so as to convert linear reciprocable movement of the free piston (11a,11b,11c,11d) to rotational movement of the crankshaft (31).

2. A free piston engine as defined in claim 1, wherein the or each free piston (11a,11b,11c,11d) is so designed that the diameter of the intermediate part thereof constituting an intermediate rod ($11a_0$,$11b_0$,$11c_0$,$11d_0$) is determined smaller than that of the piston's end face which is exposed to hydraulic pressure, said intermediate rod extending through an opening of an intermediate wall (19a,19b) whereby a cylindrical space (35) is formed between the inner wall of the hydraulic cylinder and the intermediate rod, said cylindrical space (35) being supplied with pressurized hydraulic medium having a pre-

determined pressure which is transported from an accumulator or the like, so that the free piston is affected by pressure which is effective in such a direction as to impart thrust to the working hydraulic medium.

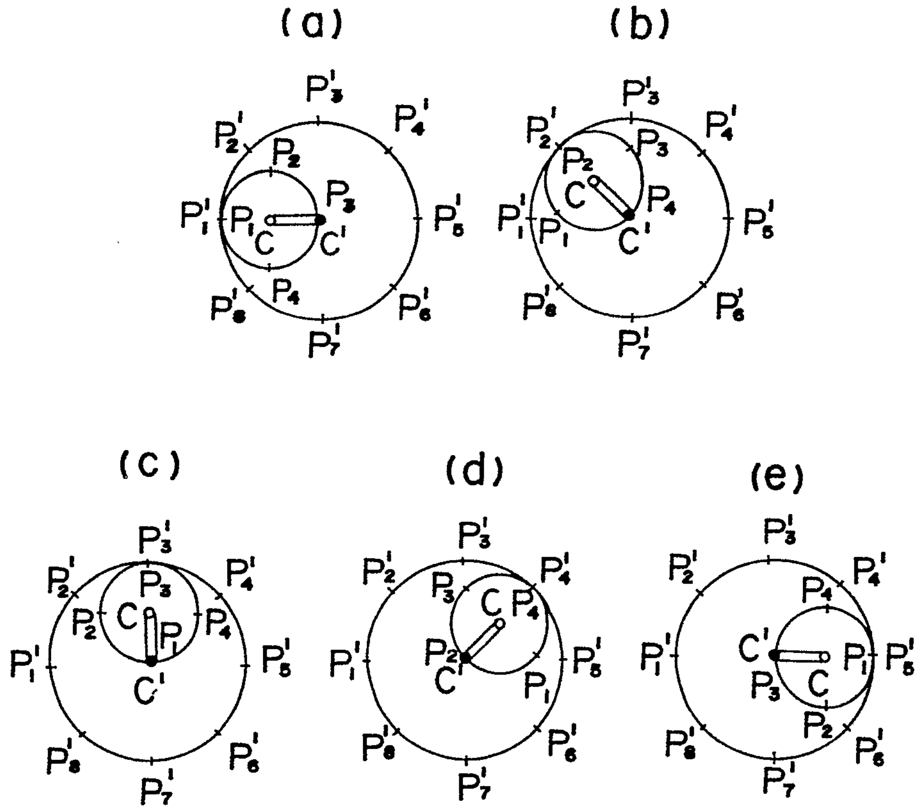
0183258

FIG. 1



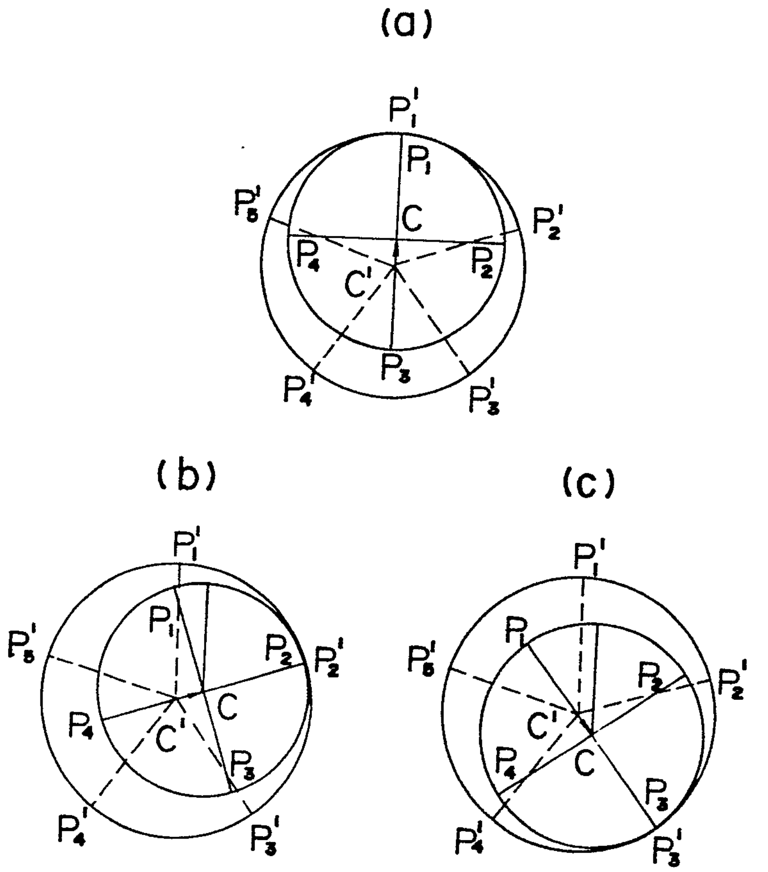
rotation angle	a	b	c	d	e
crankshaft	0°	+45°	+90°	+135°	+180°
epitrochoidal rotary piston	0°	−45°	−90°	−135°	−180°

FIG. 2



rotation angle	a	b	c	d	e
crankshaft	0°	+45°	+90°	+135°	+180°
pinion	0°	−45°	−90°	−135°	−180°

FIG. 3



rotation angle	a	b	c
crankshaft	0°	+72°	+144°
pinion	0°	−18°	−36°

FIG. 4

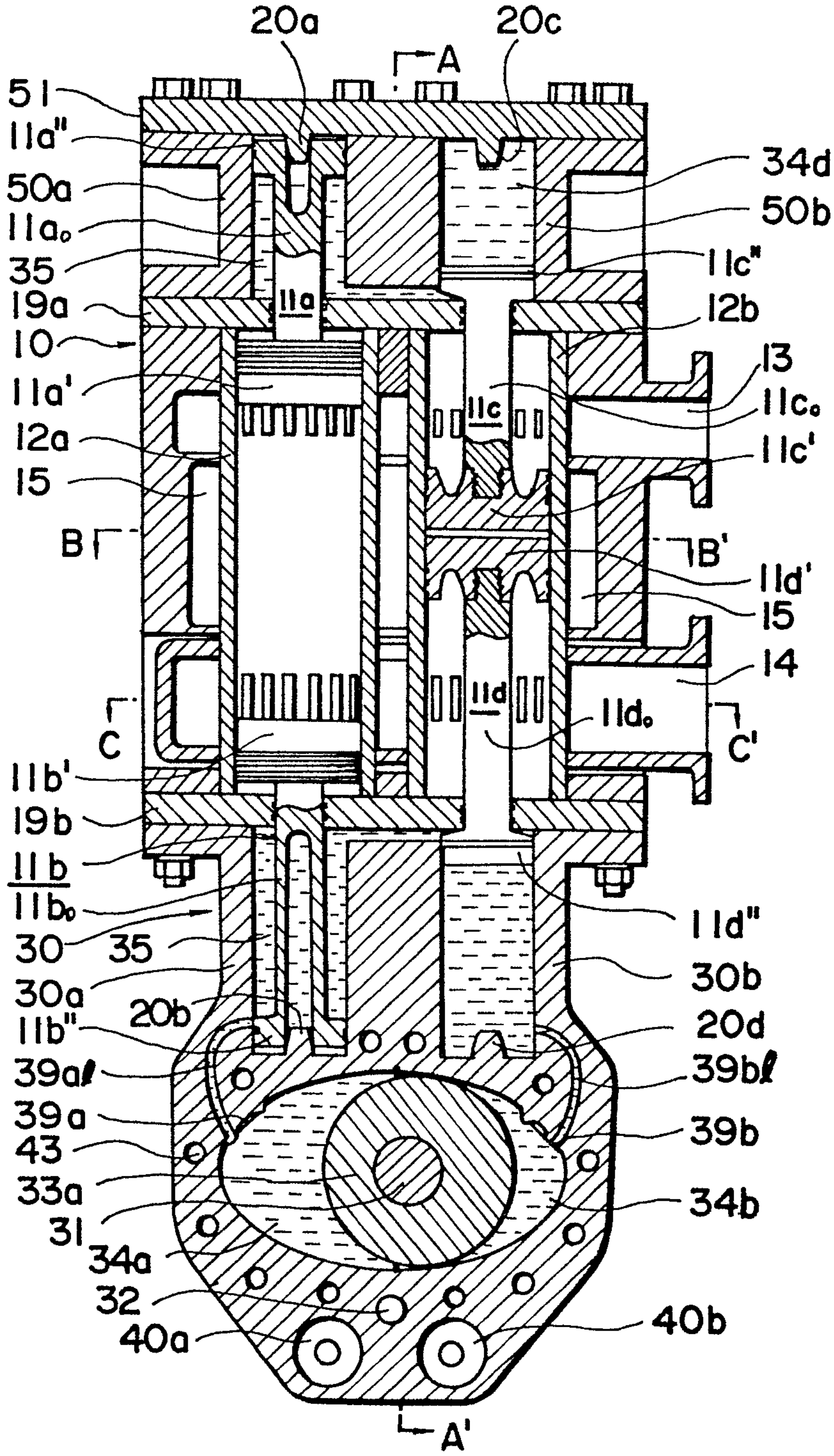

20a
20c
A
51
11a"
50a
11a₀
35
19a
10
11a'
12a
15
B
C
11b'
19b
11b
11b₀
30
35
30a
20b
11b"
39aℓ
39a
43
33a
31
34a
32
40a
A'
34d
50b
11c"
12b
13
11c₀
11c'
B'
11d'
15
14
11d₀
C'
11d"
30b
20d
39bℓ
39b
34b
40b
11a
11c
11d

FIG. 5

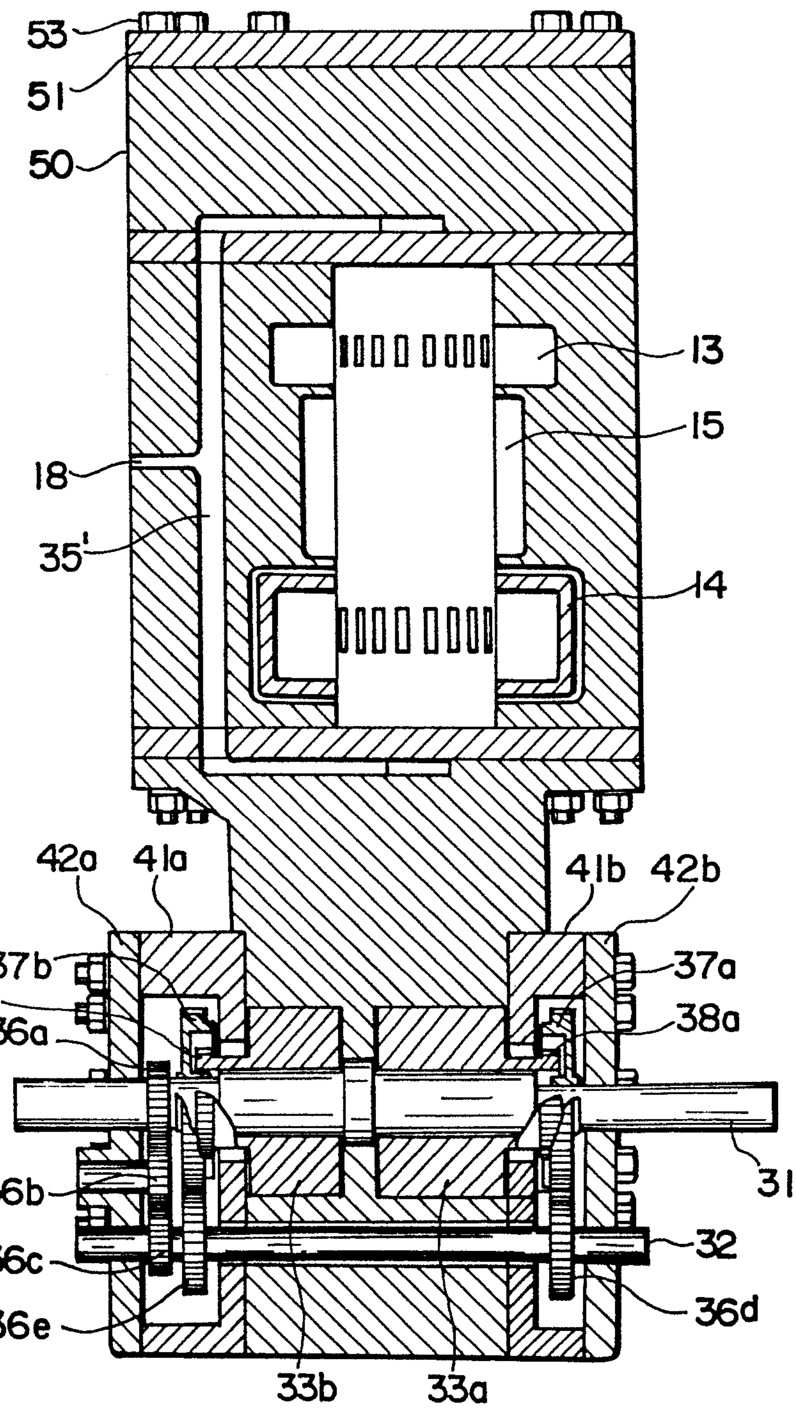

53
51
50
13
15
18
35'
14
42a
41a
41b
42b
37b
37a
38b
38a
36a
31
36b
32
36c
36e
36d
33b
33a

FIG. 6

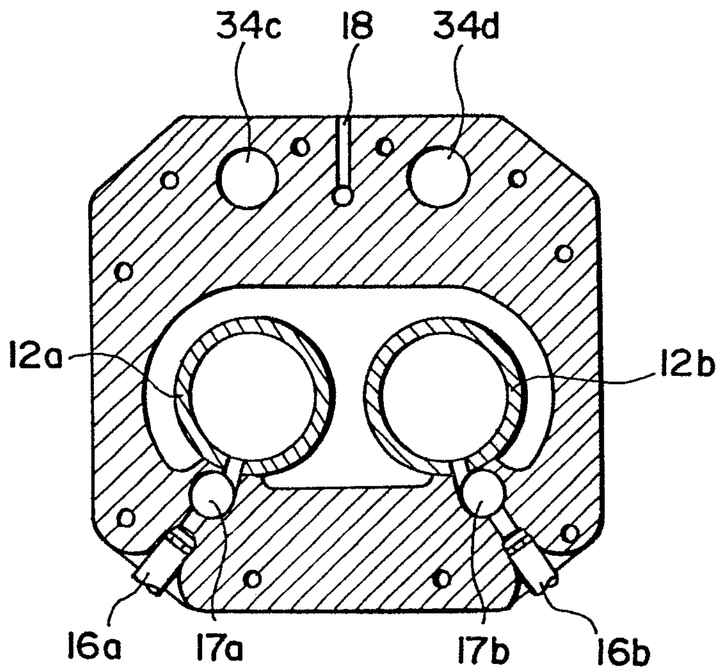


FIG. 7