July 22, 1958 C. BANCROFT 2,844,040

FLUID DISPLACEMENT MACHINE

Filed May 25, 1953 8 Sheets-Sheet 1

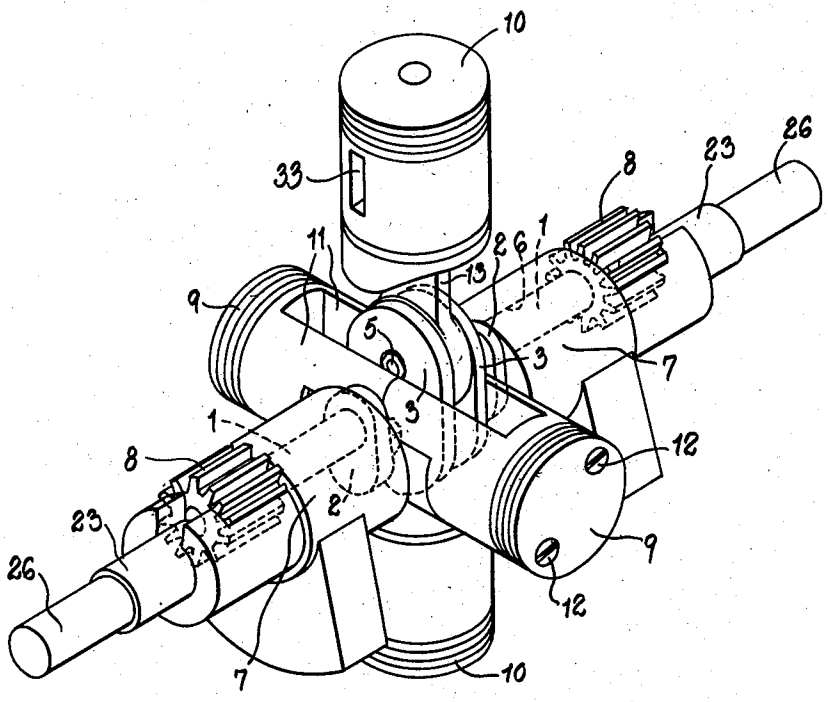


Fig. 1.

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FLUID DISPLACEMENT MACHINE

Filed May 25, 1953 8 Sheets-Sheet 2

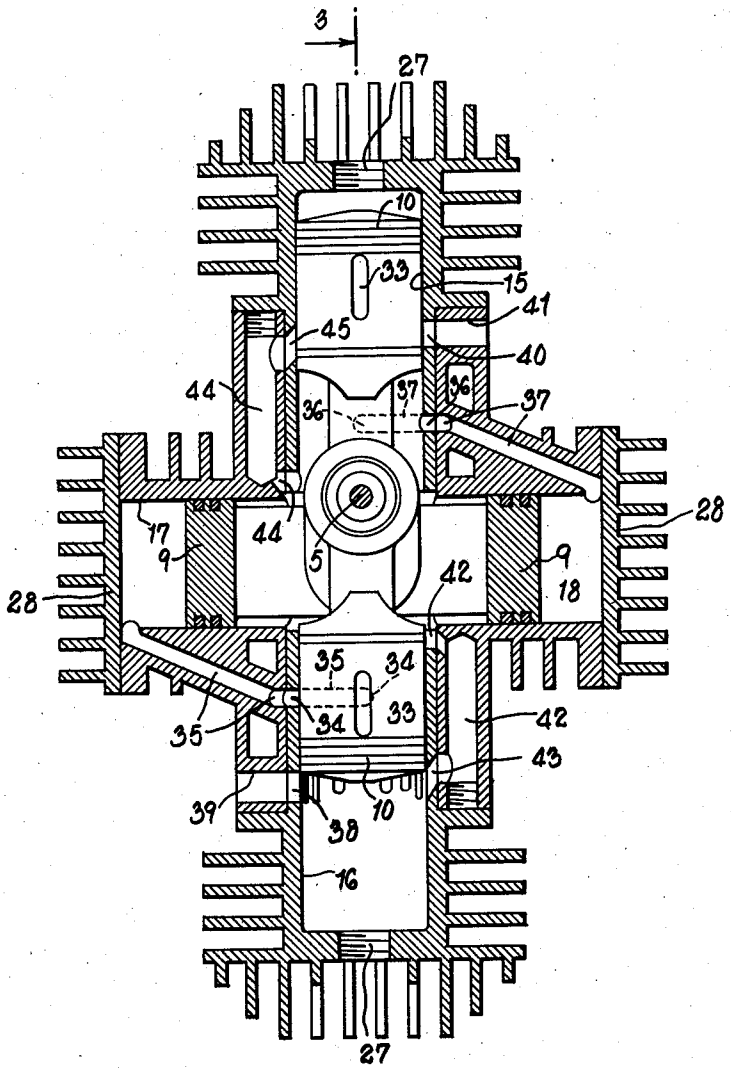


Fig. 2.

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FLUID DISPLACEMENT MACHINE

Filed May 25, 1953 8 Sheets-Sheet 3

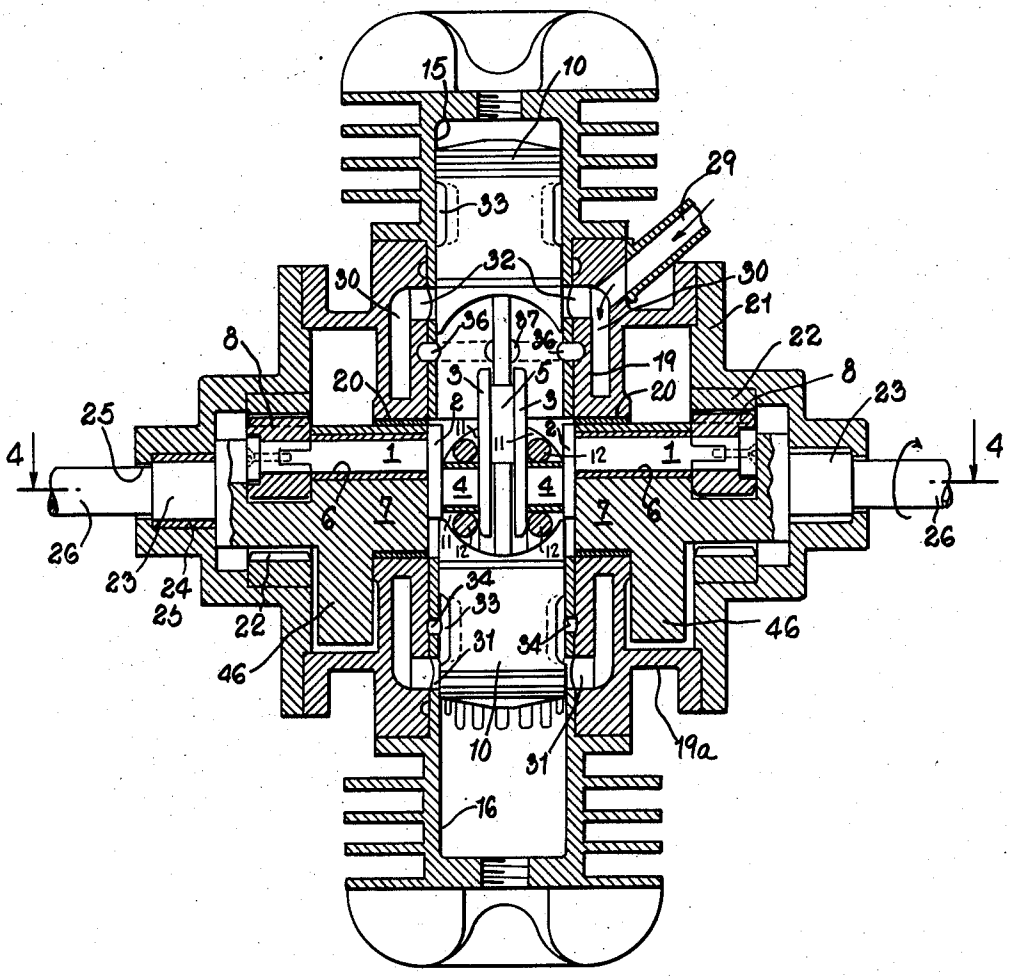


Fig. 3.

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FLUID DISPLACEMENT MACHINE

Filed May 25, 1953 8 Sheets-Sheet 4

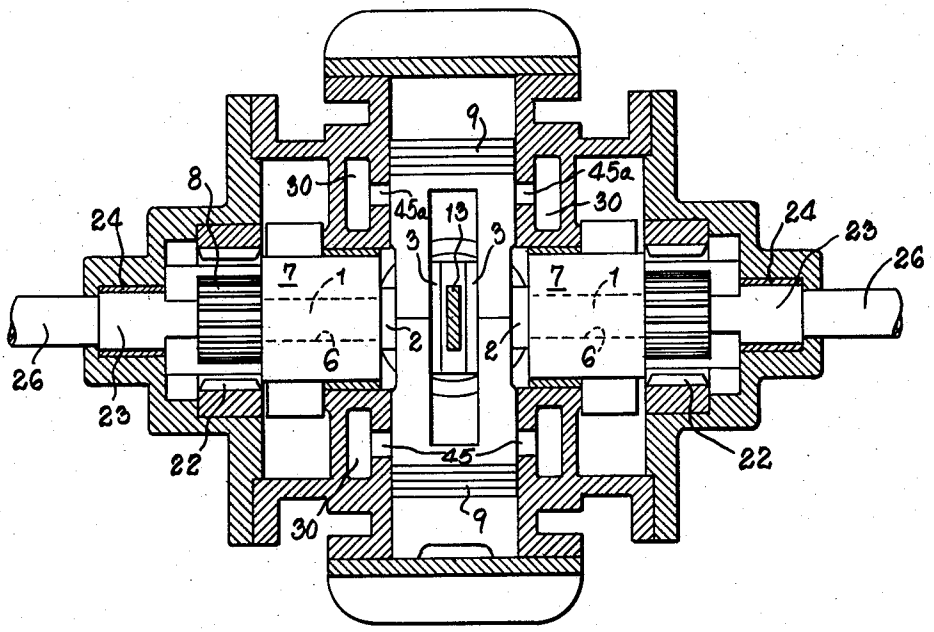


Fig. 4.

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FLUID DISPLACEMENT MACHINE

Filed May 25, 1953 8 Sheets-Sheet 5

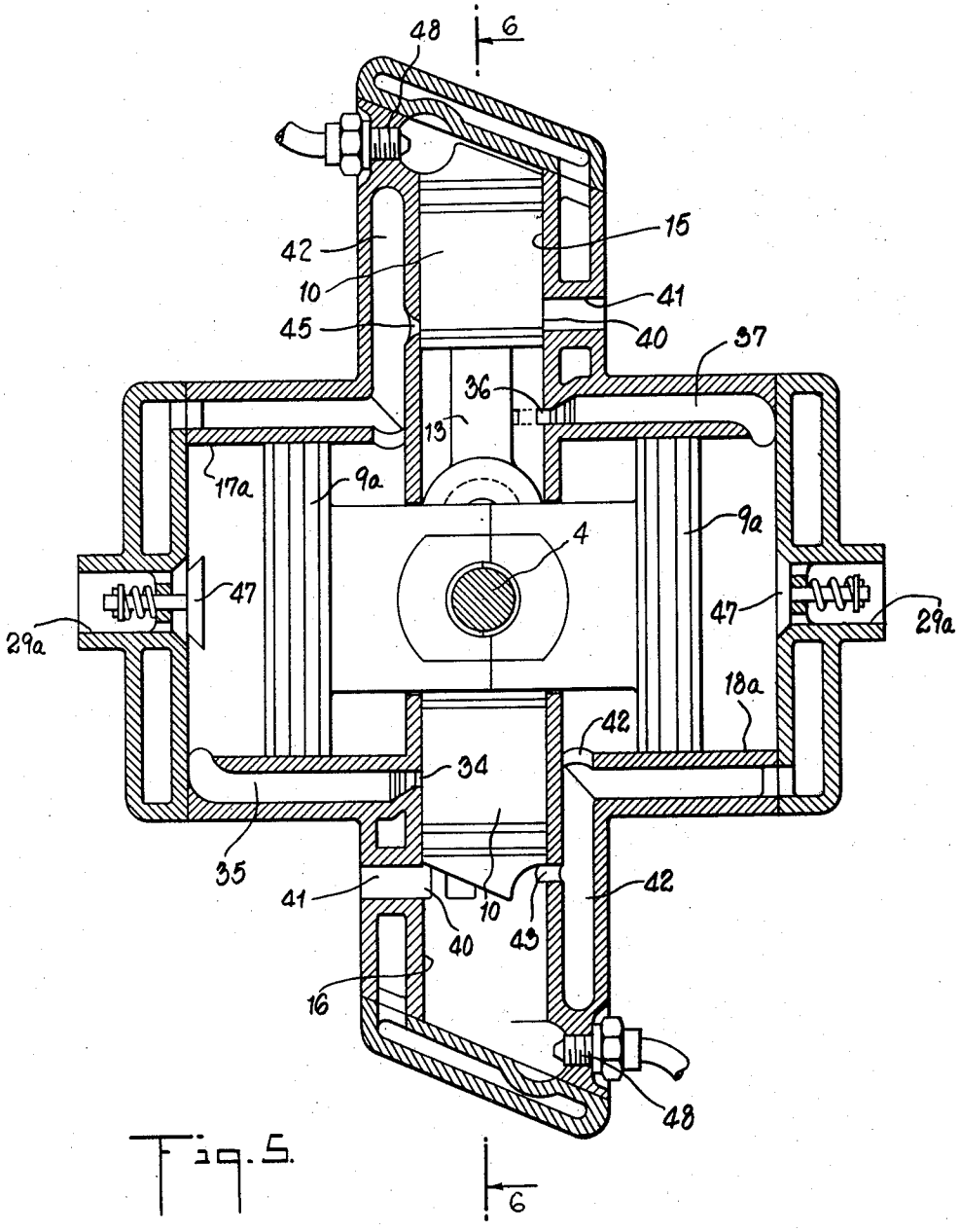


Fig. 5.

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FLUID DISPLACEMENT MACHINE

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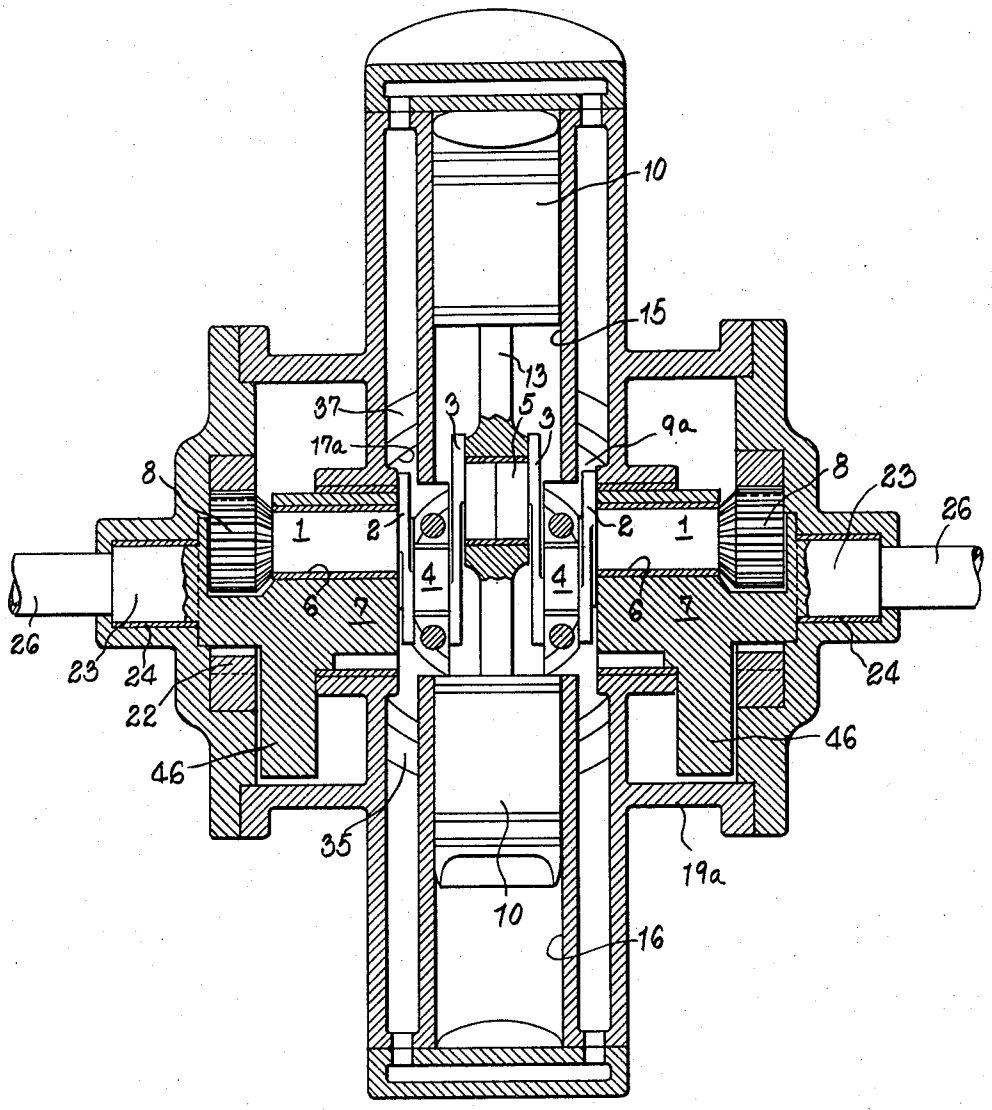


Fig. 6.

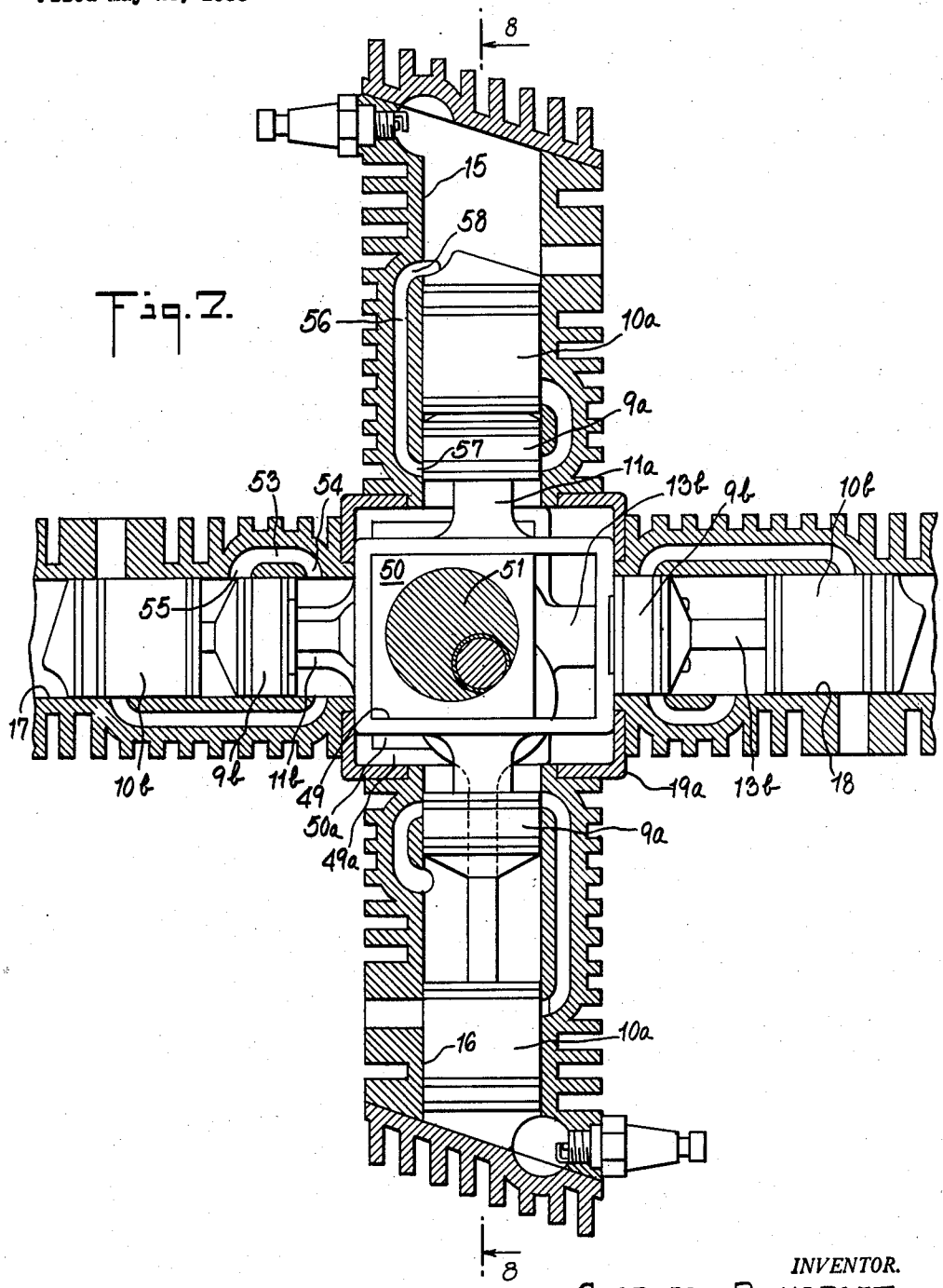
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FLUID DISPLACEMENT MACHINE

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Fig. 7.



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FLUID DISPLACEMENT MACHINE

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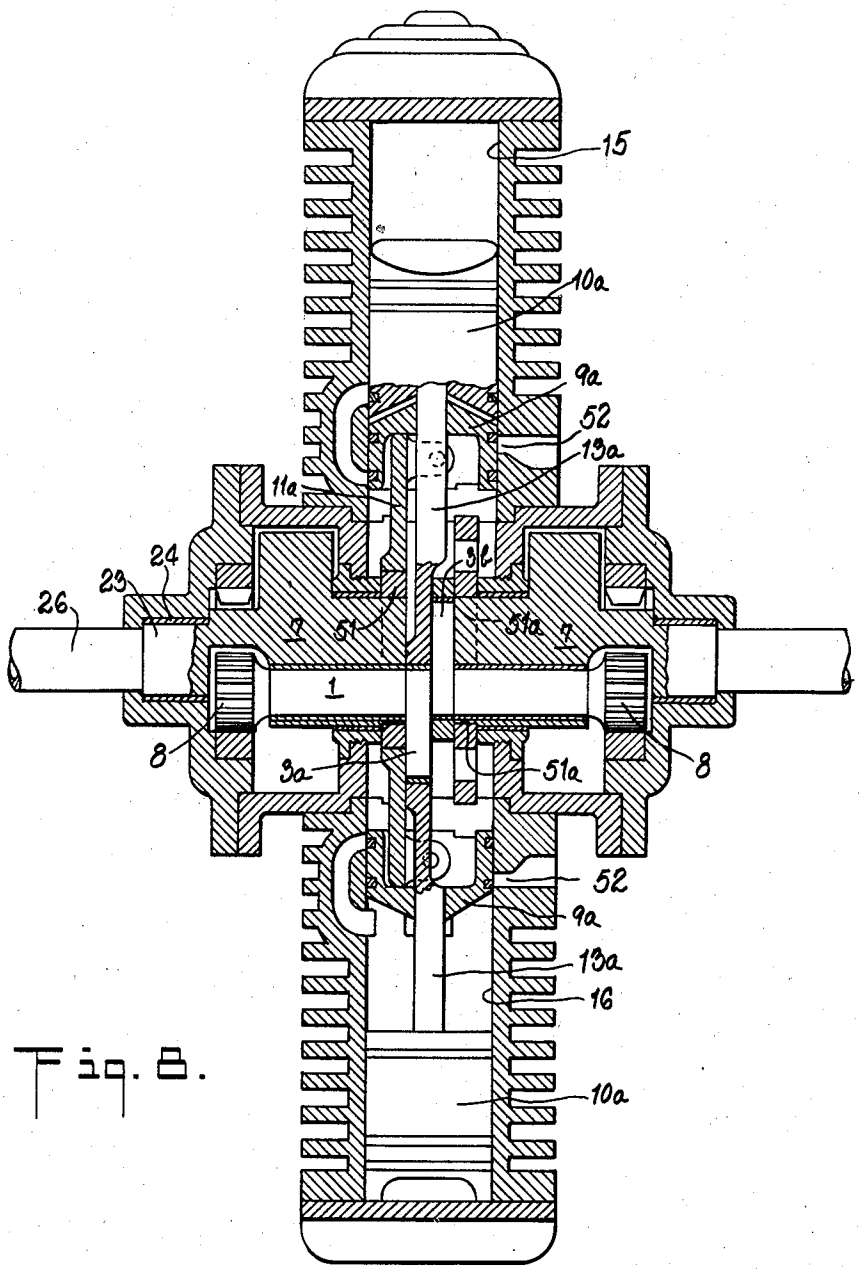


Fig. 8.

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United States Patent Office 2,844,040 Patented July 22, 1958

2,844,040

FLUID DISPLACEMENT MACHINE

Charles Bancroft, New Canaan, Conn.

Application May 25, 1953, Serial No. 357,160

2 Claims. (Cl. 74—52)

This invention relates to a fluid displacement machine adapted for many uses.

One of the objects is to provide a machine design featuring improved mechanical balance, fewer parts, simplified construction, and greater compactness and lower cost, than are conventional. Another object is to provide a construction making possible a multi-cylinder, two cycle, internal combustion engine of improved volumetric efficiency, which requires no valves, other than the pistons themselves, for control of porting, and which does not require the partitioned crank-case ordinarily characteristic of prior art multi-cylinder two cycle designs. Still another object is to make possible a compact, vibrationless gas generator of relatively simple construction, or, alternately, an exceptionally compact, eight cylinder, four cycle engine, using relatively few moving parts, or an unusually compact, multi-cylinder, pump. Other objects will become apparent from the following.

The invention involves the use of reciprocatory pistons requiring a crank-shaft arrangement. In developing this arrangement advantage has been taken of the fact that when a second crank-shaft is rotatably mounted on the crank-pin of a crank-shaft of equal crank-throw radius, and is controlled in its rotation on that crank-pin so that it rotates at a speed, relative to the stationary main body of the unit, equal and opposite to that of the crank-shaft on which it is mounted, its crank-pins reciprocate in straight lines intersecting the axis of rotation of the latter crank-shaft. When the pistons are rotatably mounted at their gravitational centers of these crank-pins, they can be confined within cylinders in which they will reciprocate through a stroke equal to twice the sum of the crank-throw radii of the two crank-shafts although all of the mechanical movements involved are purely rotary and can be balanced as such.

Since, when two crank-shafts are employed together as described above, the crank-pins of the second crank-shaft will move in straight lines intersecting the axis of the first or supporting crank-shaft; conversely, it follows that if two or more crank-pins of the second crank-shaft are forcibly restrained to such linear motion, as for example by pistons mounted on the crank-pins being confined within suitable cylinders, that crank-shaft will both be supported in its movement through the circle which it follows on the crank-pin of the first crank-shaft and will also be restrained to the "opposite and equal" rotation, while moving through that circle, which the circumstances require. Thus it follows that the pistons mounted on the crank-pins of the second crank-shaft act as supporting bearings for that crank-shaft and also control, or assist in controlling, that crank-shaft's rotation on the crank-pin of the first crank-shaft.

Specific examples of mechanisms and machines embodying the principles of the invention are illustrated by the accompanying drawings in which:

Fig. 1 is a perspective view showing an example of two sets of pistons provided with the type of crank organization described but constructed and arranged in accordance with the present invention;

Fig. 2 is a cross section showing the assembly of Fig. 1 in use in a machine which may be used as a two cylinder, two cycle gasoline engine;

Fig. 3 is a section taken substantially on the line 3—3 in Fig. 2;

Fig. 4 is a section taken substantially on the line 4—4 in Fig. 3;

Fig. 5 is a view similar to Fig. 2 excepting that it shows the principles of the invention applied to a machine adapted to function as a gas generator useful for operating turbines and the like;

Fig. 6 is a section taken substantially on the line 6—6 in Fig. 5;

Fig. 7 is a view similar to Fig. 2 but showing a machine which may be used as a four cylinder, two cycle gasoline engine, this incorporating a modification of the arrangement shown by Fig. 1;

Fig. 8 is a section taken substantially on the line 8—8 in Fig. 7.

In the assembly shown by Fig. 1, the described second crank-shaft is in the form of an intermediate crank-shaft having oppositely extending shafts 1, single-throw crank-arms 2 and double-throw crank-arms 3, with crank-pins 4 extending between the arms 2 and 3, in each instance, and a crank-pin 5 extending between the arms 3, the latter being formed as a pair of spaced arms. The crank-arms are aligned with each other, the crank-pins 4 are eccentric to the shafts 1, and the crank-pin 5 is eccentric to the shafts 1 opposite to the crank-pins 4. The displacement of the crank-pins 4 and 5 relative to the shafts 1 are equal and opposite.

The shafts 1 oppositely extend through bearing holes 6 formed eccentrically in two shafts 7 of large diameters with the axial displacement of the shafts 1 relative to the axes of the shafts 7 being equal to the axial displacement of the crank-pins 4 or 5 relative to the crank-shaft 1. The shafts 7 are completely separate from each other and function as cranks mounting the crank-shaft previously described. This or an equivalent arrangement provides adequate room for proper bearings for the shafts 1 of the crank-shaft. The outer ends of the shafts 7 are reduced and cut-out to provide room for pinions 8 secured to the outer ends of each of the shafts 1. When these pinions 8 are engaged by stationary ring gears of proper diameter and axially concentric with the shafts 7, rotation of the shafts 7 in one direction results in opposite rotation of the intermediate crank-shaft under the control of the pinions 8, and therefore the crank-pins 4 and 5 move linearly in intersecting paths.

The pistons are made as two sets of oppositely facing pistons 9—9 and 10—10. The inner or back ends of the pistons 9 are rigidly interconnected by laterally spaced struts 11 which may be externally contoured as continuations of the cylindrical piston sides, whereby to in effect form a double-ended piston having a central slot. The struts may be centrally divided transversely and the parts clamped together by long screws 12 so as to form bearings journalling the struts 11 to the two crank-pins 4. The spacing between the struts 11 is sufficient for rotation of the arms 3. The inner ends of the pistons 10 are interconnected by a single strut 13 in a rigid manner, and this strut 13 may also be centrally split transversely with the parts held together by screws so as to form a bearing journalling the strut 13 to the crank-pin 5, or, as shown by the drawings, the strut may be solid and the crank-pin 5 made as separate parts projecting from the crank-arms 3 and bolted together in the manner common to radial engine crank-pins.

When the pistons are arranged in right angularly related cylinders the pistons and their interconnecting struts

are firmly guided to follow the reciprocating paths of the respective crank-pins, whereby the intermediate crank-shaft is supported rotatively. By making the two halves of each piston set so that the crank-pin journal bearings are at the center of gravity of the piston set, in each instance, it becomes possible to balance the moving parts as purely rotary parts even though the pistons reciprocate. The forces involved are purely rotary in effect. This effect is obtained when single pistons are journaled to each crank-pin, instead of the sets, when there is enough overhanging weight to make the center of gravity of each piston coincide with its crank-pin, and the piston is adequately guided linearly.

Referring now to Figs. 2 through 4, the pistons **10** are shown reciprocatively positioned in a set of opposed cylinders **15** and **16**, and the pistons **9** reciprocatively positioned in a set of opposed cylinders **17** and **18**. Each set of cylinders is radially arranged at right angles to each other. The cylinders are mounted by a crank-case **19** which completely encloses the crank-shaft assembly and is provided with axial holes in which the respective shafts **7** are journaled in bearings **20**. The crank-case **19** outwardly extends axially from the radial walls in which the shafts **7** are journaled. End plates **21** are secured to the ends of the crank-case extensions **19***a* and these end plates immovably mount the previously described ring gears, shown at **22**, with which the pinions **8** are constantly meshed as the pinions revolve due to rotation of the shafts **7**. The shafts **7** are continued outwardly with reduced portions **23** forming journals mounted by bearings **24** in the end plates **21**, the latter having holes **25** through which the outermost ends of the shafts **7** extend as at **26**.

Due to the fact that the pinions **8** force the crank-shaft **1** to rotate on its axis in the same fashion at both ends, the individually separate shafts **7** are rotatively locked together so that they function as a single shaft in which the crank-shaft **1** is eccentrically journalled. Neither of the shafts **7** can rotate independently of the other, the effect being that of a single crank-shaft formed as two rotatively interconnected halves. It is to be noted that the shaft **7** function as cranks having their outer ends journalling the crank-shaft **1** and revolving about axes common to the axes of the shafts **7**.

The cylinders **15** and **16** have their outer ends closed and provided with threaded holes **27** in which spark plugs may be screwed to fully enclose these ends. The outer ends of the cylinder **17** and **18** are completely closed by cylinder heads **28** in each instance.

The crank-case **19** is made with its previously described radial walls, which mount the bearings **20**, in the form of hollow walls. A fluid intake passage **29** leads to the space **30** inside of these hollow walls, the spaces **30** being interconnected as to both crank-case bearing-mounting end walls. The crank-shaft arrangement is such that the pistons **9** differ in angular phase from the pistons **10** during the operation of the machine. In the illustrated form the phase difference is 90° and, therefore, when one piston set is stationary at either of its reciprocating extremes, the other piston as at its mid-position and moving at its maximum velocity.

In Figs. 2 through 4 the cylinders **15** and **16** and the pistons **10** funtion as a two-cycle internal combustion engine, and the cylinders **17** and **18** and pistons **9** function as a scavenging blower for this engine. Passages from the cylinders **17** and **18** feed into the inner ends of the cylinders **15** and **16** where they are alternately either connected by slots in the pistons **10** with the intake manifold space **30** or, when the pistons are out of the way, directly with the crank-case. Thus the pistons **10** control the intake and output of what is in effect a scavenging blower formed by the cylinders **17** and **18** and the pistons **9**. Since the pistons **10** are moving at their maximum velocities when the blower is shifted from input to output, the valve action is rapid, which is advantageous. The blower output is thus fed into the crank-case from which it discharges into the cylinders **15** or **16** through passages opening into these cylinders and uncovered by the pistons when at their inner limits, whereby to both scavenge and charge these cylinders. Suitable exhaust ports are uncovered by the pistons at this time, as is usual. Additional intake passages for the scavenging blower cylinders **17** and **18** may be arranged to open from the intake manifold so that they are uncovered when the pistons **9** are at their inner limits. In such instances as when the valving action is at the end of a passage connecting with the cylinder, the volume of the passage is added to that above the piston during the compression phase, and the features of the present invention permit such passage to be made very short with a consequent ability to effect adequate volumetric efficiency.

With the foregoing in mind, the intake manifold space **30** connects with the portion of the cylinder **16** that is inner respecting the center of the engine, through ports **31**, and with the inner portion of the cylinder **15**, respecting the engine center, through ports **32**. Both of the pistons **10** are provided in their side walls with longitudinally extending grooves **33** which register with the ports **31** and **32** as either piston approaches and is adjacent to its innermost limit, or in other words, throughout the inner half of its piston stroke.

The cylinder **16** has ports **34** spaced inwardly from the port **31** and with which the latter is connected by the grooves **33** when the latter register with the ports **31**. These ports **34** connect with a passage **35** leading into the outer end of the cylinder **17**, as shown by Fig. 2. The cylinder **15** has comparable ports **36** connecting with a passage **37** leading into the outer end of the cylinder **18**. An exhaust port **38** is located in the cylinder **16** so as to be uncovered by the piston when the piston approaches and is adjacent to its innermost position, this port **38** connecting with an exhaust passage **39**. The cylinder **15** has a comparable exhaust port **40** connecting with an exhaust passage **41**.

The ports **34** and **36** and their respective passages **35** and **37** are in part shown in broken lines in Fig. 2 only so as to indicate their locations. As shown in Fig. 3 there are three of these ports in each instance and the outer two are diametrically located for registration by the piston grooves. The passages **35** and **37**, as the case may be, extend as a semi-circle portion from the outer two ports, around behind the respective cylinder walls, and thus to the straight portions leading to the cylinders **17** or **18** in each instance. Referring to Fig. 3, the semi-circle portion of the passage **35** is in a plane above that of the drawing, as is indicated in Fig. 2 which also shows that the semi-circle portion of the passage **37** is below the plane of the drawing so it can be indicated by the broken lines in Fig. 3. In each instance, the center one of the ports **34** or **36** are in effect auxiliary ports which function when the charge is blown into the crank-case as subsequently described.

The crank-case interior connects with the cylinder **16** through a passage **42** and a port **43** located through the cylinder wall diametrically opposite to the exhaust port **39**. The crank-case, in a similar manner, connects with the cylinder **15** through a passage **44** and a port **45**. Preferably, the cylinders **17** and **18** have auxiliary ports **45***a* connecting with the intake manifold space **30** and located in each instance so as to be uncovered by the piston as the latter approaches and is at its innermost limit.

In operation the intake passage **29** is connected with a carburetor or the like for supplying a suitable combustible mixture, and, of course, spark plugs will be located in the holes **27** and energized by a suitably timed sparking voltage. Upon proper rotation of the shafts **7** the charge is sucked through the intake manifold **30**, which would be by the left-hand piston **9** as the operation is shown by Fig. 2, through the ports **31** and **34**, by way of the registering piston grooves **33** of the piston

in the cylinder 16, and thus into the cylinder 17 from the head of which the pistons 9 are retreating. As the operation continues the pistons 9 reverse so as to compress the charge. In the drawings this compression is underway in the cylinder 18, which was previously sucked full of the charge. The compressed charge is driven through the passage 37 and into the crank-case, the piston in the cylinder being now clear of the ports 36. The compressed charge in the crank-case drives in through the passage 42 and port 43 into the cylinder 16.

Thus in succession the charge is appropriately introduced to the cylinders so as to scavenge them free from burnt gases and provide them with fresh charges. As the appropriate one of the pistons 10 advances it cuts off all of the ports ahead of its working face and goes into its compression phase ready for firing. After firing, the exhaust is, of course, through the ports 38 and 40 or the exhaust passages 39 and 41, as the case may be.

It can now be seen that in a single compact unit the set of pistons 9 provide the previously described scavenging blower with its valve action controlled by the working pistons so as to eliminate valves as such. The crank-case is used as part of the fluid transfer system but it is not required to operate alternately under sub-atmospheric and super-atmospheric pressure, so there is no need for separating webs or diaphragms in the crank-case for any reason. The spaces between the radial crank-case walls and the end plates 21 provide gear cases for the pinions 8 and ring gears 22 and may be flooded with lubricant as desired. The crank-shaft 1 and pinions 8 are balanced by counterbalances 46, and the gear box space also provides room in which these counterbalances may move.

Referring now to the gas generator, shown by Figs. 5 and 6, the same fundamental parts and principles are used as have been described in connection with the example shown by Figs. 2 through 4. Therefore, wherever possible the same numerals are used to identify corresponding parts and where these parts are modified the same numerals are used in conjunction with a letter.

In this gas generator the piston 10 in the cylinders 15 and 16 function to power the pistons 9*a* in the cylinders 17*a* and 18*a*, the pistons 9*a* and the last-named cylinders being of much larger diameter than are shown in connection with the first example for the corresponding scavenging blower parts. Another difference is that the intake system is modified by the provision of intake passages 29*a* directly in the heads of each of the cylinders 17*a* and 18*a*, respectively, these intake passages being provided with poppet check valves 47.

As to the operation of this second example, the left-hand pliston 9*a* is shown moving to the right and sucking air in through the passage 29*a*, the poppet check valve 47 being sucked open to permit this. The right-hand one of the pistons 9*a* is on its compression stroke and is forcing a large volume of air, previously sucked in as described, through the passage 37 and port 36 into the crank-case. The upper one of the pistons 10 has cleared the port 36 at this time, the piston having formerly blocked this port in the manner the lower one of the pistons 10 is shown doing in the case of the port 34, as required to close the passage 35 during the suction stroke. From the crank-case the air drives up through the passage 42 and out of the port 43 so as to scavenge and cool the cylinder 16 and from therethrough the exhaust port 41 mixed with the hot products of previous combustion, a residue of air being left in the cylinder to charge it for the next compression. In this case the passage 42 surrounds all of the cylinders so as to be heated thereby and to cool the cylinders by the air moving through the passage 42.

The upper one of the pistons 10 is at the limit of its compression stroke. Because there is an excess of air involved this example shows the use of solid fuel injection nozzles 48 so that the machine operates in the manner of a diesel. A suitable fuel injection control must, of course, be provided.

This second example is particularly adapted to produce a large volume of highly compressed gas which may be used, by connections to the exhaust ports 41, for powering a turbine or the like. The pistons 9*a* work as pumps compressing large volumes of air into the passage 42 surrounding the cylinders and thus into the power cylinders above the pistons 10 where the air is heated by the combustion, as to the portion of the air retained by the cylinders, so as to obtain a relatively high temperature before being mixed with fresh scavenging air and discharged through the exhaust ports. The principles of the present invention provide for a compact and smooth operating unit capable of being controlled easily and which produces a large volume of hot compresed gas.

The four cylinder, two cycle, gasoline engine illustrated by Figs. 7 and 8 involves a modification of the crank-shaft and piston assemblies shown by Fig. 1.

In this modification the intermediate crank-shaft involves the use of two eccentrics 3*a* and 3*b*, instead of the crank-arms 3 of the first example. All of the power pistons 10*a*, of which there are four, are organized in the form of sets of two pistons each with the vertical ones interconnected by a rigid piston rod 13*a* journaled on the eccentric 3*a* and the horizontal ones interconnected by a rigid piston rod 13*b* journaled on the eccentric 3*b*. This arrangement provides a four cylinder radial engine requiring some means for scavenging and recharging the cylinders above the pistons. This is provided for as described below.

A vertical scavenging blower piston set 9*a* reciprocates with its piston heads respectively in the cylinders 15 and 16, the pistons being apertured so as to permit the sliding passage of the rod 13. This piston set 9*a* has the inner ends of its respective pistons interjoined by a strut 11*a* having a central cross head 49 in which a block 50 transversely reciprocates. This block 50 is journaled on an eccentric 51 fixed to the inner end of the left-hand one of the shafts 7 as viewed in Fig. 8. The horizontal set of the scavenging blower pistons 9*b* are interconnected by a strut 11*b* provided with a similar cross head 49*a* in which a block 50*a* reciprocates and is journaled on an eccentric 51*a* fixed on the inner end of the right-hand one of the shafts 7. Horizontal pistons 9*b* are also apertured to slidingly pass the piston rod 13*b*. The crank-case 19*a* is made in rectangular fashion so that it may be internally provided with suitable guiding surfaces for supporting the two cross heads in their relatively right angular motions and in line with their respective piston sets.

The described assembly is timed so that the power piston sets 10*a* are 90° out of phase with each other and so that the scavenging blower pistons 9*a* each lags in phase behind its associated power piston by 90°. Furthermore, the eccentrics 51 and 51*a* are contoured so that in each instance the stroke of the blower piston is shorter than the power piston stroke.

Each cylinder and its two pistons functions the same as the other so the following description is applicable to all. In each instance each cylinder has an intake port 52 opening into its inner end so as to be cleared by the porting piston 9*a* or 9*b* when the latter approaches and is at the limit of its innermost stroke limit. As shown in the right-hand one of the horizontal cylinders in Fig. 7, this results in sucking a charge, as from a carburetor, into the space formed between the two pistons 9*b* and 10*b*. As the action continues the piston 9*b* advances, as shown in the case of the cylinder 17, while the piston 10*b* is moving inwardly. Each cylinder has a transfer passage 53 opening into the cylinder through an inner port 54 and an outer port 55 respectively located so that one or the other is closed by the adjacent piston 9*b* when the latter is at either of its stroke limits or adjacent thereto, but so as to span the piston 9*b* when the latter is

at an intermediate position. Therefore, as the pistons **9***b* and **10***b* move, so as to diminish the space between them, as shown on the left-hand side of Fig. 7, the compressed charge is transferred to the crankcase.

In Fig. 7 the upper ones of the pistons **9***a* and **10***a* are shown nearly at their closest positions, respecting each other, and at this time a transfer passage **56** is opened because it has an inlet port **57** positioned to be cleared by the piston **9***a* at this time, and an outlet port **58** cleared by the pistons **10***a*, whereby the charge is transferred from the crank-case to above the power piston so as to scavenge the cylinder thereabove and recharge it ready for compression and firing. The exhaust action after firing is readily apparent.

In all of the foregoing examples it is to be understood that those skilled in the art can effect the necessary timing as to the ignition means in the case of the gasoline type and the fuel injection in the case of the diesel type. The three examples serve to illustrate how the assembly of Fig. 1 permits the designing of unusually compact fluid displacement machines which may be balanced as though comprising only rotary parts. The invention further provides for the scavenging blowers with their necessary valve action provided by the power pistons with the resulting valve action rapidly inaugurated and terminated, due to the oscillatory motion imparted each piston and the timing already explained.

It is to be understood that all of the piston sets are centrally journaled to the crank-arms, whether provided by arms as such or by eccentrics, at the center of gravity of the piston set in each instance. Furthermore, the pistons and their interconnecting members, as to each set, are rigidly guided linearly by the cylinders in which the pistons reciprocate, whereby to rotationally support the intermediate crank-shaft in all instances, due to there being a plurality intersecting lines or paths through which the crank-pins or their equivalent reciprocate and which represent truly rotary motions. With proper rotary balancing smooth operation results. It is to be understood that by interconnecting the two spaced shafts as described, the rigid guiding by the cylinders is not absolutely necessary.

It is to be noted that where the valving of one cylinder is controlled by the piston of another cylinder, offset radially substantially 90°, as shown by the drawings, the mid-point in time of any valving condition in the controlled cylinder occurs when the controlling piston is at the top or bottom of its stroke, and, therefore, if the valve timing is changed at one end of the stroke of the piston in the controlled cylinder, there should be a corresponding change in the valve timing at the other end of the stroke since the mid-point duration of port opening or closing must always correspond to the top or bottom of the stroke of the controlling piston.

If then, as may be true in the case of the piston gas generator shown by the drawings, it is preferable to have the pumping cylinder's connection to the output manifold (in this case the crank-case area) delayed during the output stroke of the pumping pistons until the pressure in these cylinders reach that already existing in the crank-case section, forming the output manifold, and to have this connection broken, as before, at the end of each output piston stroke, it can be seen that the mid-point of duration of such a port opening would not coincide with the top or bottom of the stroke of the controlling piston. However, by changing the radial angular relationship of the controlled cylinder to that of the cylinder in which the controlling piston works, the top or bottom of the stroke can be made to coincide with the mid-point of the port opening duration as required. It is expressly understood that such variations in the radial angular relationship of the various sets of pistons can be introduced for such purposes or other purposes without effecting the basic principles of this invention, as described here, or their advantages such as the balancing advantages, etc.

It is also to be understood that while the invention is illustrated by examples having the crank-case section acting as a connecting manifold between the scavenging pumps and the control cylinders, other examples can be constructed to have the crank-case section acting as a preliminary intake manifold or eliminated entirely from any function in connection with the passage of working fluids through the unit. Also, although all the drawings included herewith relate to four cylinder units, or an equivalent construction, any number of crank-pins and sets of pistons can be employed to provide any number of working cylinders in a single row or multi-row construction or to provide single cylinder, in line, or V-type units.

To simplify the description, reference has been made herein to single ports in instances where the drawings show a multiplicity of openings, to effect loop scavenging for example. It is to be understood, therefore, that herein a reference to a port is to be construed as meaning a single opening or a group of interconnected openings.

I claim:

1. A fluid displacement machine including a plurality of sets of opposed cylinders arranged radially about an axis, a piston in each of the cylinders, struts laterally spaced axially respecting said axis and interjoining the pistons in one of the cylinder sets, a strut positioned between the spaced struts and interjoining the pistons in another of the cylinder sets, cranks journaled to rotate on said axis and positioned on opposite sides of the spaced struts and having arms revolving about said axis, and a crank-shaft journaled by and between the cranks' arms and having crank-arms on the outside of the spaced struts with crank-pins extending therefrom and journaled to and extending through the struts and double crank-arms between the struts and which are joined by a crank-pin journaled to and extending through the strut positioned between the spaced struts.

2. A fluid displacement machine including a plurality of sets of opposed cylinders arranged radially about an axis, a piston in each of the cylinders, struts laterally spaced axially respecting said axis and interjoining the pistons in one of the cylinder sets, a strut positioned between the spaced struts and interjoining the pistons in another of the cylinder sets, cranks journaled to rotate on said axis and positioned on opposite sides of the spaced struts and having arms revolving about said axis, and a crank-shaft journaled by and between the cranks' arms and having crank-arms on the outside of the spaced struts with crank-pins extending therefrom and journaled to and extending through the struts and double crank-arms between the struts and which are joined by a crank-pin journaled to and extending through the strut positioned between the spaced struts, the cranks and crank-arms having equal effective lengths, and means for causing the cranks to rotate synchronously together and for causing opposite rotation of the crank-shaft, whereby the crank-pins move through intersecting linear paths while supported rotatively by the pistons and struts.

References Cited in the file of this patent

UNITED STATES PATENTS

1,056,746	Pitts	Mar. 18, 1913
1,585,453	Williamson	May 18, 1926
1,761,429	Dean	June 3, 1930
1,774,105	Nelder	Aug. 26, 1930
1,787,599	Stickney	Jan. 6, 1931
2,088,863	McClelland	Aug. 3, 1937
2,132,595	Bancroft	Oct. 11, 1938
2,199,625	Fiala-Fernbrugg	May 7, 1940
2,223,100	Foster	Nov. 26, 1940

FOREIGN PATENTS

341,010	Great Britain	July 4, 1929
884,403	France	Apr. 27, 1943