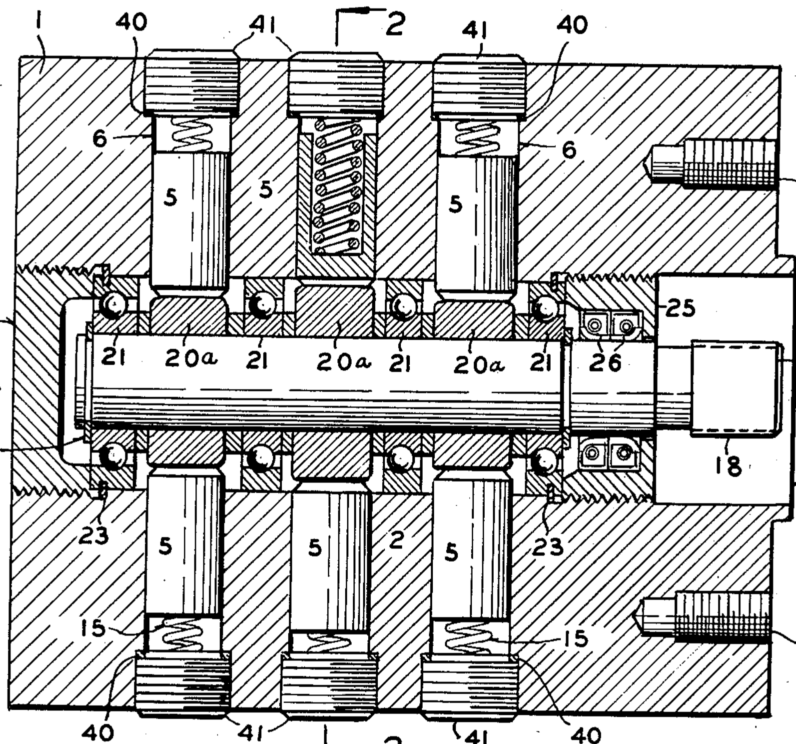

FIG. 1.

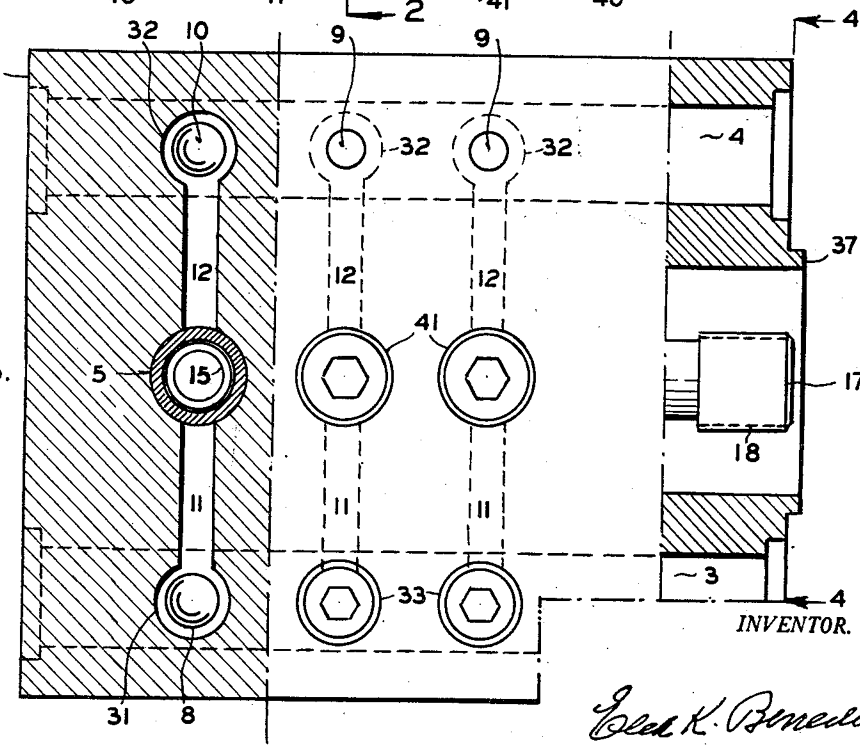

FIG. 3.

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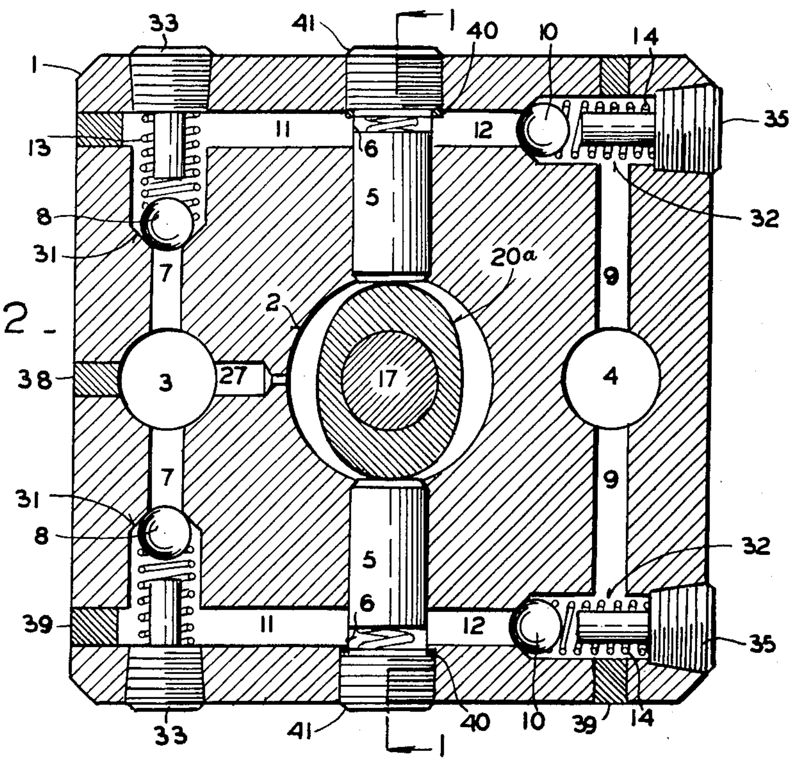

FIG. 2

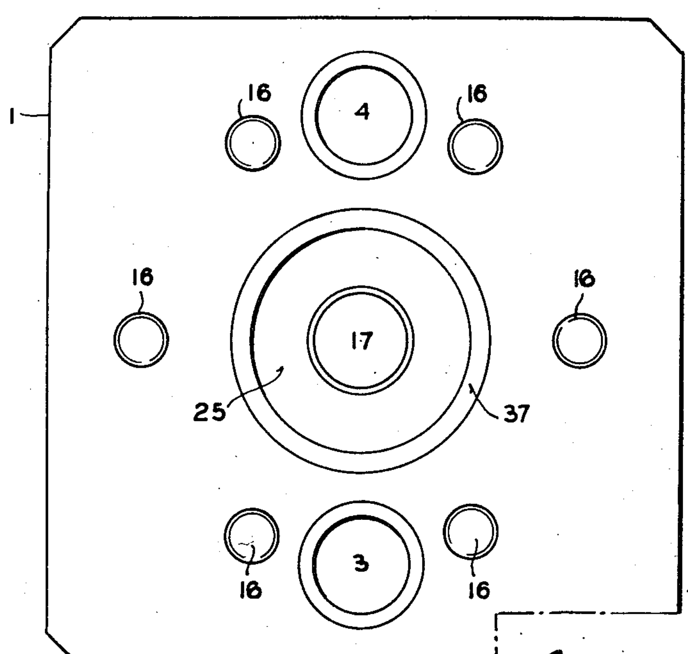

FIG. 4.

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FIG.5

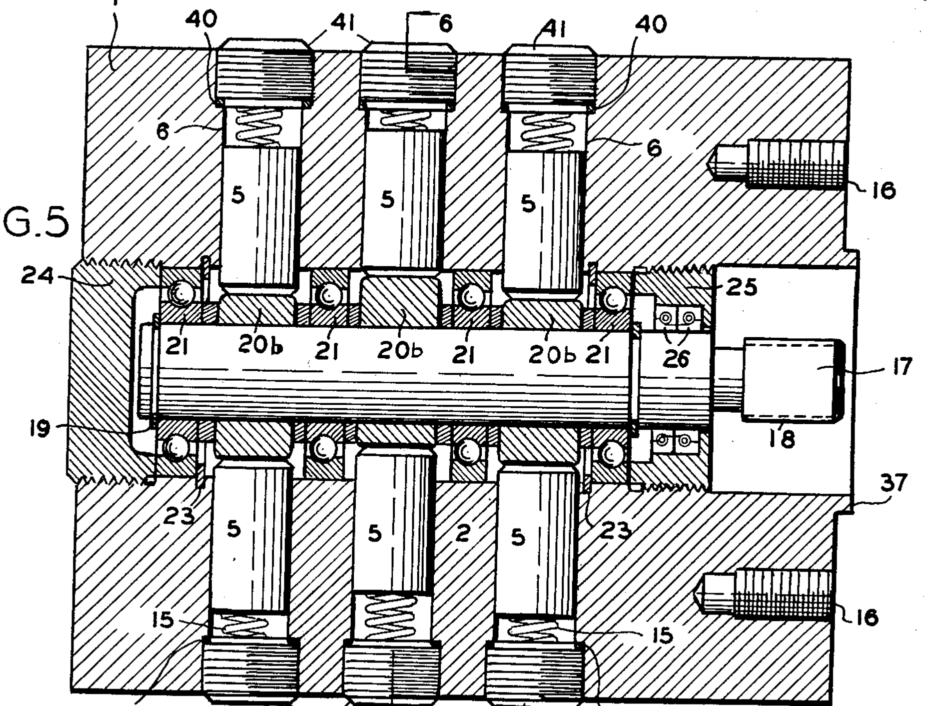
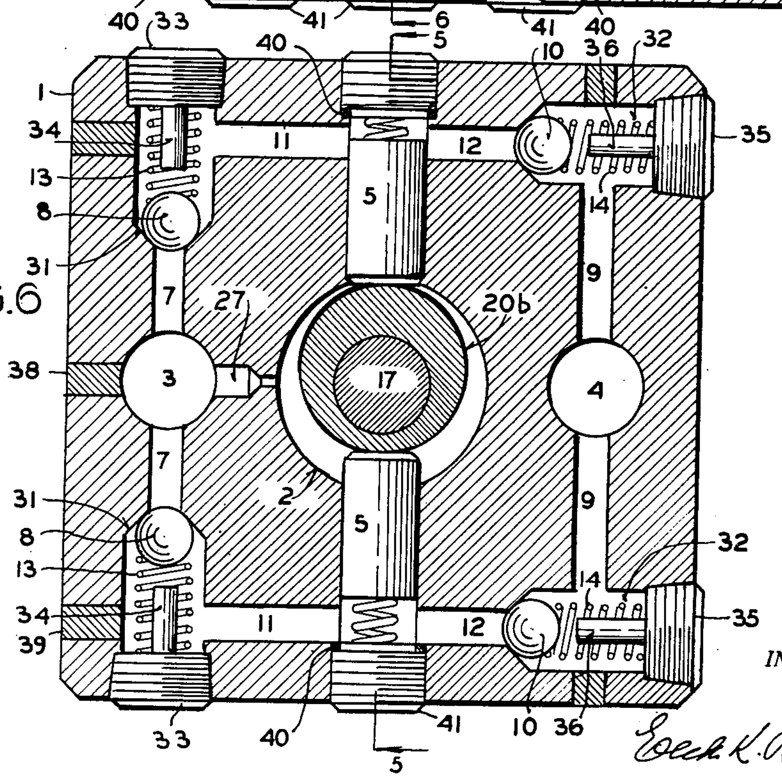


FIG.6



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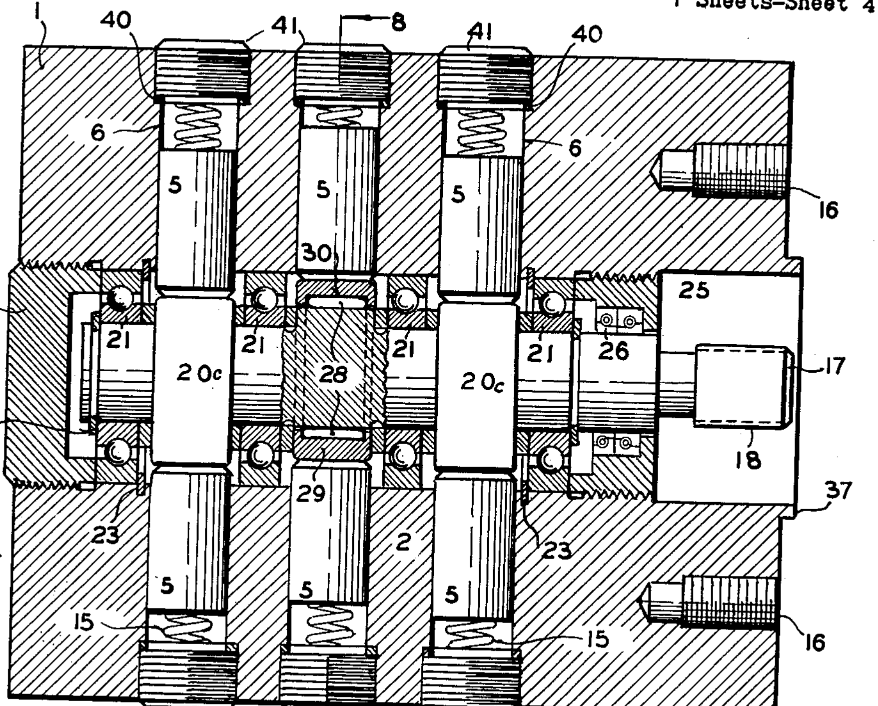

FIG. 7.

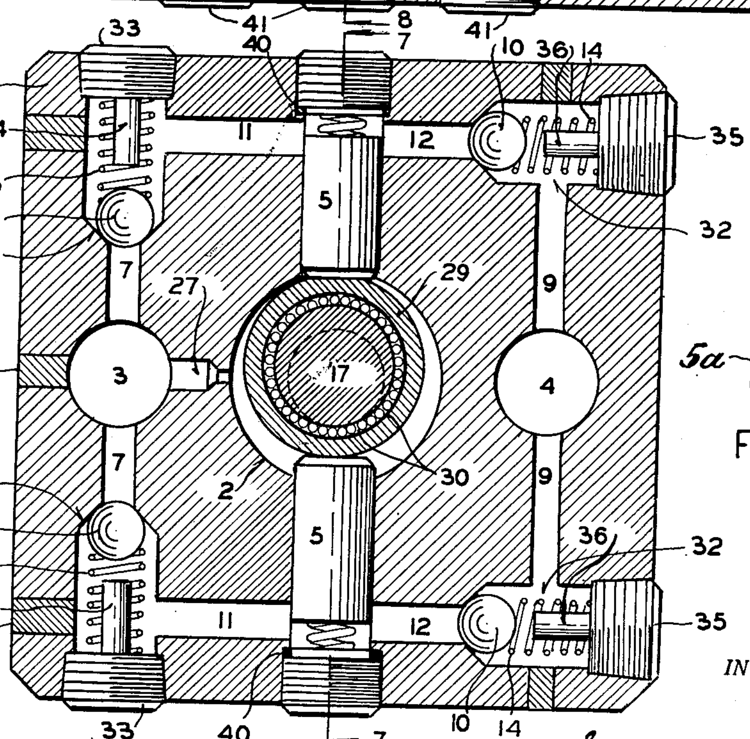

FIG. 8.

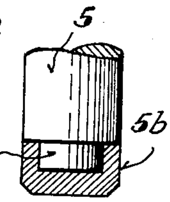

FIG. 18.

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FIG. 9

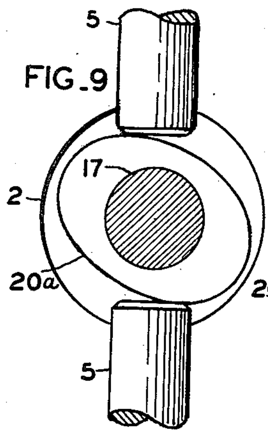

FIG. 10

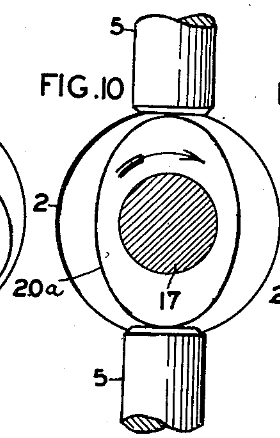

FIG. 11

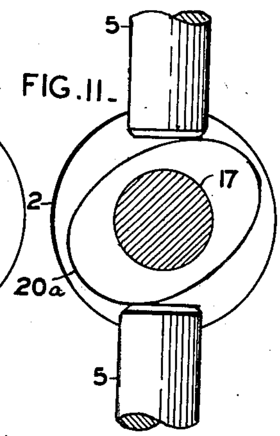

FIG. 12

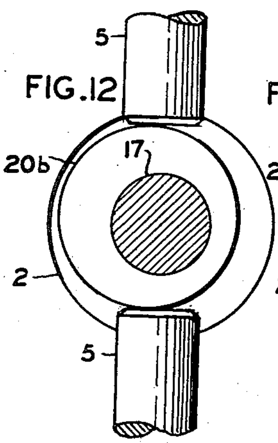

FIG. 13

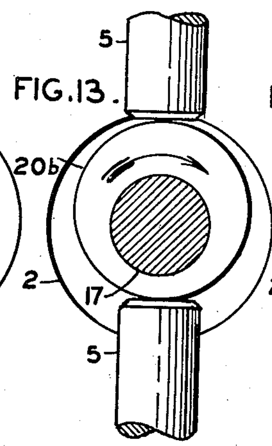

FIG. 14

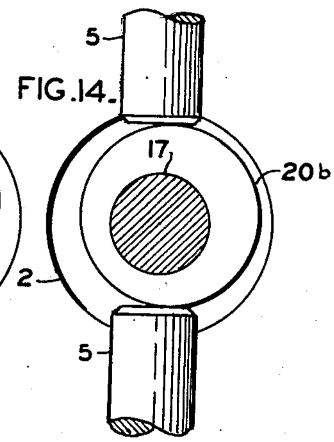

FIG. 15

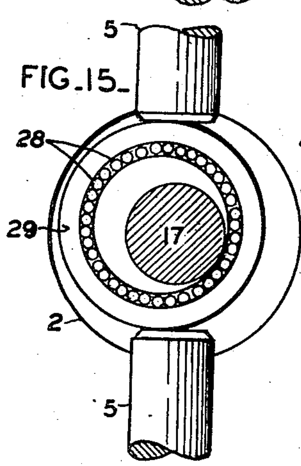

FIG. 16

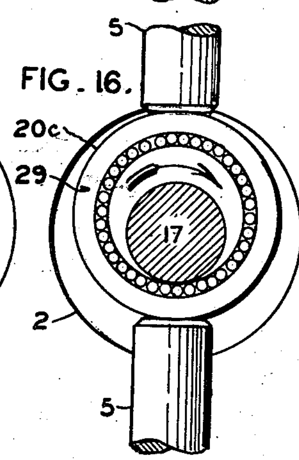

FIG. 17

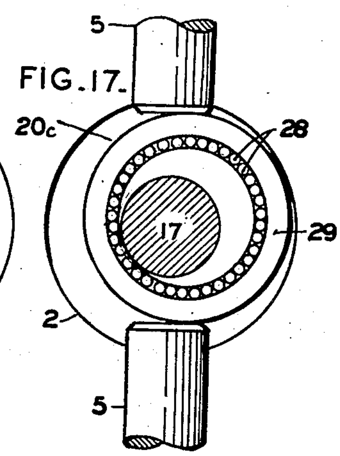

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FIG_19_

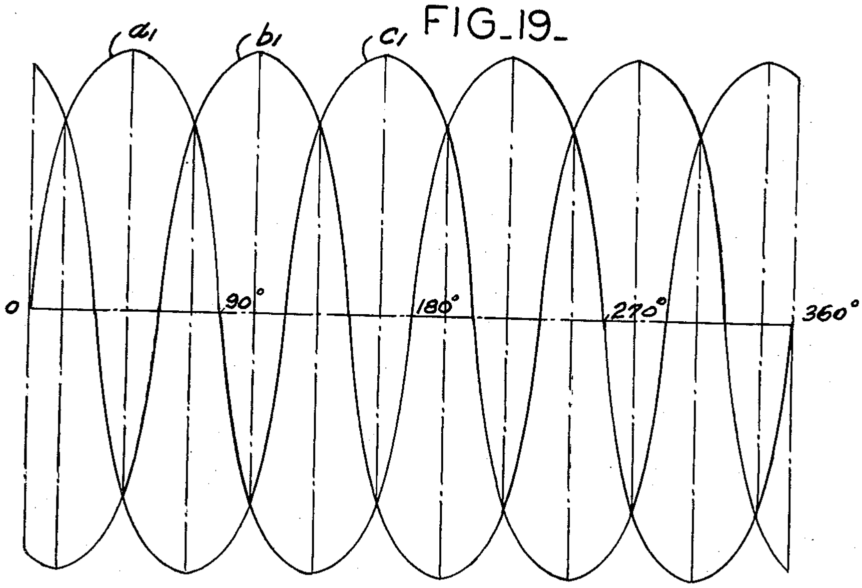


FIG_20_

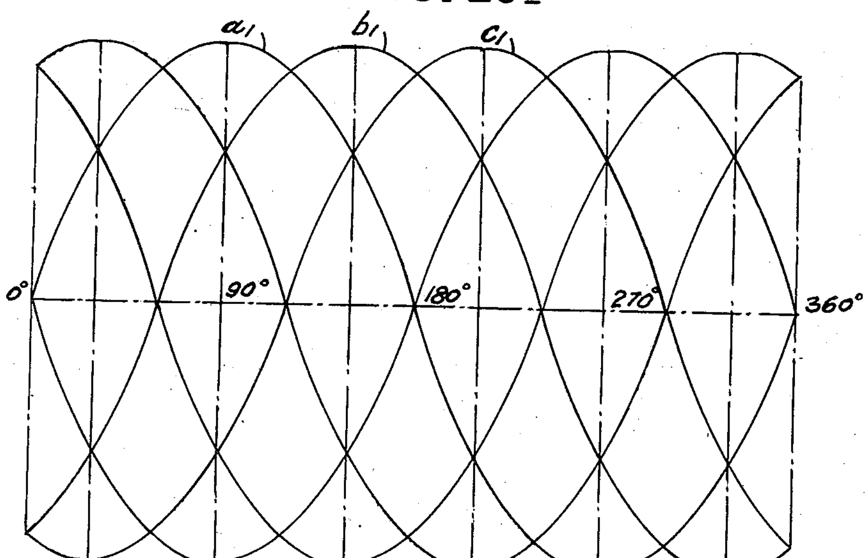


FIG_21_

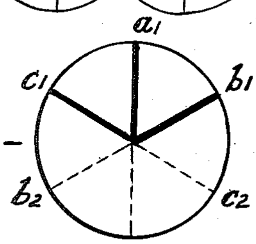
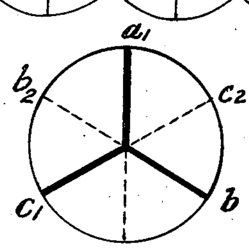


FIG_22_



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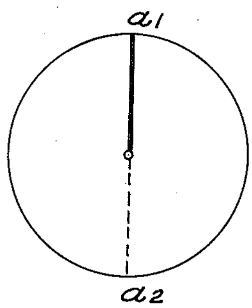

FIG_23_

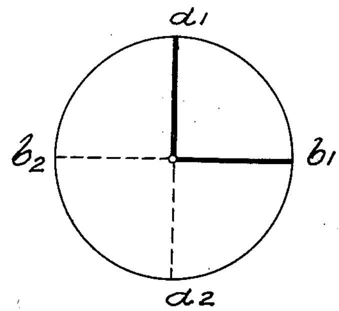

FIG_24_

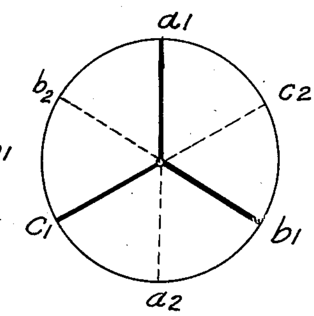

FIG_25_

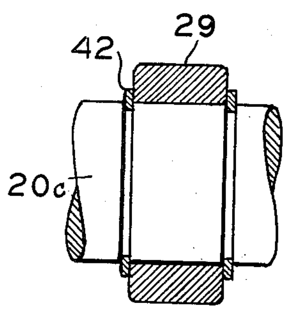

FIG. 26.

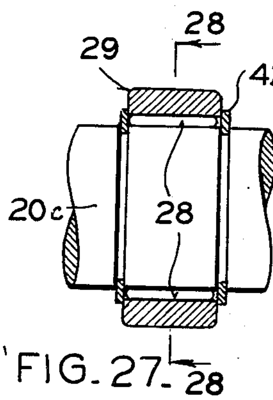

FIG. 27.

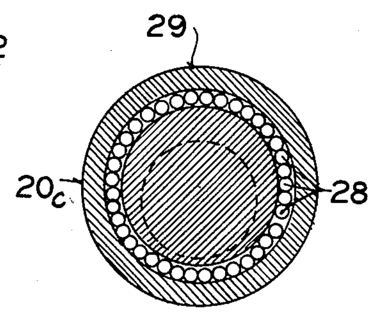

FIG. 28

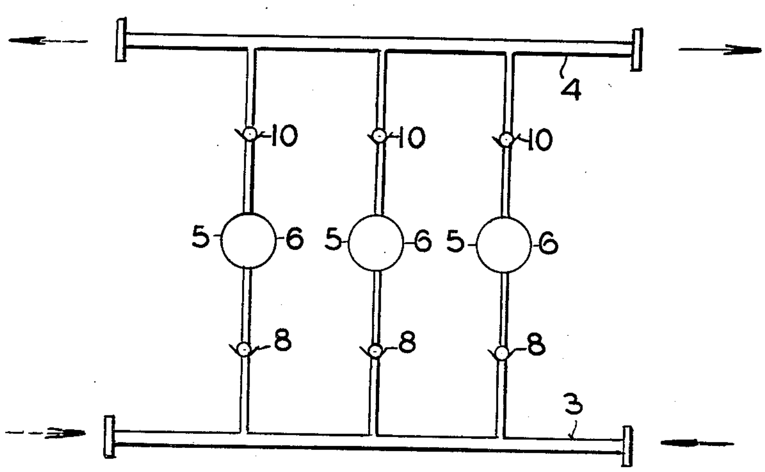

FIG. 29.

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United States Patent Office

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Patented Dec. 21, 1954

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HYDRAULIC PUMP OR MOTOR

Elek K. Benedek, Chicago, Ill.; Melba L. Benedek, administratrix of said Elek K. Benedek, deceased, assignor to Melba L. Benedek

Application June 6, 1949, Serial No. 97,392

10 Claims. (Cl. 103—174)

This invention relates to opposed piston and cylinder type hydraulic pump or motor, and more particularly to pumps and motors wherein one or a plurality of sets of opposed piston and cylinder assemblies are arranged in parallel and axially spaced operating positions with one another in an integral piece of solid block housing.

One main object of the present invention is to provide a solid steel or alloy steel block housing for a set of opposed piston and cylinder assemblies.

Another main object is to provide a solid steel or alloy steel forged block housing for a plurality of sets of opposed piston and cylinder assemblies.

A further main object is to provide a compact and centrally located driving means for the opposed piston and cylinder assemblies in a pump of the above character.

Another main object lies in the provision of compact and centralized thrust transmitting means for the opposed piston and cylinder assemblies, in a pump of the opposed piston and cylinder type.

A still further main object of the invention lies in the provision of a fluid tight chamber for the housing of the driving and thrust transmitting means of the pump of the above character.

A still further main object lies in the provision of a centrally located fluid tight chamber in a pump of the above character for the housing of the bearing means and the reception of the driving and thrust transmitting means of the pump of this invention.

A further main object lies in the provision of a simplified intake and exhaust fluid passage in the block type forged housing of this invention.

A further main object lies in the provision of simplified intake and exhaust individual valve means for the opposed piston and cylinder assemblies.

A further main object lies in the provision of novel arrangements for the intake and exhaust valve chambers and for their corresponding valve means respectively.

A further main object lies in the provision of simplified individual fluid passages for the opposed piston and cylinder assemblies.

A further main object lies in the provision of novel individual fluid passages for the individual sets of opposed piston and cylinder assemblies.

A further object lies in the provision of a solid non-yielding metallic block as the housing means for a pump of this character.

Another object lies in the novel combination of thrust transmitting and driving means for the relative reciprocation of the opposed pistons and cylinders.

A still further object lies in the provision of a novel bearing supporting means for the said thrust transmitting and driving means.

A further object lies in the novel relative disposition of the said thrust transmitting and driving means.

A further object lies in the novel disposition of the main fluid passages with respect to each other and with respect to the axis of the block housing of the pump.

A further object lies in the novel disposition of the individual intake and exhaust fluid passages of the piston and cylinder assemblies with respect to each other and the main fluid passages respectively.

A further object lies in the novel disposition of the intake and discharge valves of an opposed piston and cylinder assembly relative to the individual fluid passages respectively.

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A further object lies in the provision of novel thrust transmitting and driving means for the reciprocating opposed pistons and cylinders.

A further object lies in the provision of means for reciprocating the pistons during the pressure cycle operation of the pump.

A further object is in the provision of novel means for reciprocating the pistons during the suction cycles of the pump.

A still further object lies in the provision of a block type housing in which the cylinders can be bored and lapped or burnished by improved, simplified processes and operations.

A still further object lies in the provision of a novel finishing, boring and lapping process to finish the central longitudinal main bore of the block bearing housing in proper concentric relation with the block.

A further object is to provide an improved pump housing and an improved process for finishing the outside wall surfaces of the housing in proper relation to the central fluid tight chamber thereof.

Further objects and advantages will be apparent from the following description of the three illustrative embodiments of the present invention which form part of the specifications.

In the accompanying drawings:

Fig. 1 is a longitudinal main section of the pump constructed in accordance with the spirit of this invention, taken on line **1—1** of Fig. 2.

Fig. 2 is a transverse main section taken on line **2—2** of Fig. 1.

Fig. 3 is a combined section and elevation of Fig. 1, showing parts of Fig. 1 in top view, parts in section through the axes of the main fluid passages, and a further part through the axes of the horizontal intake and exhaust passages of the top piston of Fig. 2.

Fig. 4 is an end view of Fig. 3 taken on line **4—4** of Fig. 3.

Fig. 5 is a longitudinal main section taken on line **5—5** of Fig. 6. It shows the second embodiment of this invention. It is similar to Fig. 1 except that the thrust transmitting means are single acting cam means instead of the double acting ones of Fig. 1.

Fig. 6 is a transverse main section taken through line **6—6** of Fig. 5.

Fig. 7 is a longitudinal main section similar to Fig. 1 taken through line **7—7** of Fig. 8, and shows the third embodiment of the invention; this embodiment shows a stationary thrust ring as a direct thrust transmitting means, instead of a double acting rotary cam as in Fig. 1.

Fig. 8 is a transverse main section taken on line **8—8** of Fig. 7.

Figs. 9 to 11 inclusive illustrate the relative keyed-up positions of the thrust transmitting cam means relative to the drive shaft and to each other respectively, of the first embodiment as shown in Fig. 1 of this invention.

Figs. 12 to 14 inclusive illustrate the relative keyed-up positions of the thrust transmitting cam means relative to the drive shaft and to each other respectively, of the second embodiment of this invention, as shown in Fig. 5.

Figs. 15 to 17 inclusive illustrate the relative keyed-up positions of the thrust transmitting cam means relative to the drive shaft and each other respectively, of the third embodiment of this invention, shown in Fig. 7.

Fig. 18 shows a fragment of the two part piston, including the piston proper and a sliding shoe or cap therefor.

Fig. 19 shows the wave pattern of the suction and the delivery flow of the first embodiment of this invention as shown in Fig. 1 and Fig. 2.

Fig. 20 shows the wave pattern of the suction and the delivery flow of the second and third embodiments as shown in Figs. 5 and 6, and Figs. 7 and 8 respectively of this invention.

Fig. 21 and Fig. 22 show the relative position of three sets of opposed piston and cylinder assemblies when the sets are disposed at 60° and 120° phase angles respectively, relative to one another.

Fig. 23 shows the relative disposition of the opposed piston and cylinder assemblies of one set of opposed assemblies.

Fig. 24 shows the relative disposition of two sets of opposed piston and cylinder assemblies at 90° phase angle between the sets.

Fig. 25 shows the relative disposition of three sets of opposed piston and cylinder assemblies at 120° phase angle between the sets.

Figs. 26 to 28 inclusive show details of the cam drive of the third embodiment of the invention, as shown in Fig. 7 and Fig. 8.

Fig. 29 shows the internal oil circuit of the pump or motor constructed in accordance to the present invention.

Construction of the pump or motor

Referring now to the opposed piston and cylinder type pumps or motors, such as shown in Figs. 1, 2, 3 and 4 and Figs. 9, 10 and 11 respectively, and more particularly to pumps and motors wherein one or more sets of opposed piston and cylinder assemblies, **5—6** and **5—6**, are arranged in parallel pumping positions in a solid, one piece pump body **1**, preferably of the solid forged block type. The arrangement is such that two piston and cylinder assemblies **5—6** and **5—6** are arranged in sets of opposed assemblies about a rotary driving and thrust transmitting means as at **17** and **20**, respectively, in one common meridian plane of block **1**. The manner of the assembly being such that the axis of each opposed piston and cylinder assembly **5—6** and **5—6** will lie in the same straight line so that the paired opposed assemblies **5—6** and **5—6** of each set will reciprocate in one common line, the common line of dead centers.

The assemblies **5—6** and **5—6** of each set will lie on diametrally opposite points of bearing housing bore **2** of the driving and thrust transmitting means **17** and **20**.

One or more sets of opposed pistons and cylinders **5—6** and **5—6** are combined in a pump housing to constitute a single or multi-row pumping unit, as shown in Fig. 1. The common axis of each set of opposed assemblies **5—6** and **5—6** is parallel with the common axis of the other sets of opposed piston and cylinder assemblies **5—6** and **5—6**, and falls in a common main meridian plane which is horizontal or vertical to the block housing **1**. In the present instance this plane is vertical, and contains also the axis of rotation of the longitudinally extending driving and thrust transmitting means **17** and **20** respectively. The axis of each set of opposed piston and cylinder assemblies **5—6** and **5—6** being parallel with the axis of the other sets, the assemblies of the multi-row combination also will be parallel with each other and will occupy a central vertical portion of the block housing **1**, as shown more particularly in Fig. 1 and Fig. 2.

Each sub-assembly **5—6** of the opposed assembly **5—6** and **5—6** is provided with a set of intake and exhaust automatic valve means **8** and **10**, and appropriate intake and exhaust fluid passages **7—11** and **9—12** respectively between one sub-assembly **5—6** and each set of valve means **8** and **10**. The complete assembly of the entire set **5—6** and **5—6** thus will include two intake **8—8** and two exhaust **10—10** valves, two complete sets of intake passages **7—11** and **7—11**, and two complete sets of exhaust passages **9—12** and **9—12**, as shown in Fig. 2. The arrangement is such that the axis of the bores of the intake and exhaust passages **7—11** and **9—12** and the axis of the opposed piston and cylinder assemblies **5—6** and **5—6** fall in the same main transverse meridian plane of the block housing. Such a transverse meridian plane is normal to the axis of rotation of the associated driving and thrust transmitting means **17** and **20**, and coincident with the central axis of the block housing **1**.

A centrally located housing bore as at **2** will receive and house the driving and thrust transmitting means **17** and **20**, and will pass in the direction and about the central longitudinal axis of the housing block **1**.

Parallel with the center bore **2** and in the same central horizontal meridian plane of the housing block **1** are located the main intake and exhaust fluid passages **3** and **4** of the pump. Thus the three parallel axes, namely of the center bore **2** and of the main passages **3** and **4**, are in one and the same central meridian of the pump housing **1**. See also Fig. 2. All three bores, namely the central bore **2** and the main fluid passages **3** and **4**, are running in the same longitudinal direction of the pump housing **1** and are disposed about parallel axes in the said main horizontal meridian plane.

The pump is disposed in the solid block **1** of the steel or other suitable metallic material such as nitralloy, alloy-steel, nickel iron, aluminum forging or the like. In this matter all pistons and cylinders **5—6**, main and individual fluid passages **3—4** and **7—9** and **11—12** are cast or finish bored in the solid block of steel, and therefore fluid losses, slip, or air infiltration through loose joints are confined to a practical minimum. Also the driving and thrust transmitting means **17**, **20** and **21** are housed in the rigid and permanent central bore **2**, and therefore misalignment, deformation, warpage and distortion of these vital working parts of the pump are reduced also to a practical minimum if not fully eliminated.

The novel driving and thrust transmitting means **17**, **20**, and **21** respectively, are all disposed in a lubricated central bore **2** of the block type housing **1**. In this manner the axis of rotation and the axis of the bore will be coincident. The thrust transmitting means include oval shaped, double acting cam means **20*a*—20*a*—20*a*** in the first embodiment of the invention. Cam means **20*a*—20*a*—20*a*** is provided for each set of opposed piston and cylinder assemblies **5—6** and **5—6**, and may be made integral or separable from the driving or camshaft **17** as in Fig. 1 and Fig. 2. When the cams **20*a*—20*a*—20*a*** are made removable from their respective camshaft **17** a separate anti-friction roller or ball bearing **21** may be interposed between the camshaft **17** and the housing bore **2** as at **21*a*—21*a*—21*a***. The camshaft **17** then can be assembled with its three cams **20*a*—20*a*—20*a*** as in Figs. 9, 10 and 11 including the press-fitted bearings **21*a*—21*a*—21*a*** and spacers **22** as a self-contained subassembly unit, and as such it may be manufactured and assembled as a separate unit, slip-fitted in the housing bore **2**, and locked in it against axial displacement by bearing snap rings **23—23**. The bearing **21** and cam assembly **20*a*—20*a*—20*a*** then can be secured against axial displacement to the drive shaft **17** by means of shaft snap rings **19—19**.

As shown in Fig. 9, Fig. 10 and Fig. 11, individual cams **20*a*—20*a*—20*a*** of Fig. 1 and Fig. 2 are rotated about the camshaft **17** relative to another at 60° or 120° respectively. Comparative figure diagrams of Fig. 21 and Fig. 22 explain this situation as follows: since double acting cams **20*a*—20*a*—20*a*** have diametrally opposed identical cam portions as ***a*1—*a*2**, ***b*1—*b*2** and ***c*1—*c*2**, it does not matter whether ***a*1—*b*1—*c*1** are disposed at 60° or 120° relative to one another. When ***a*1—*b*1—*c*1** are disposed at 60° phase angle as in Fig. 21, the opposed and equivalent cam portions ***a*2—*b*2—*c*2** will complete the division of the circle of one revolution into six equal angles, since $6\times60°=360°$. When, however, the cams ***a*1—*b*1—*c*1** are disposed at 120° phase angle relative to one another as in figure diagram Fig. 22, the opposed cam portions ***a*2—*b*2—*c*2** will complete the division of the circle of one revolution into six equal angles, since again $6\times60°=360°$. Thus the opposed nature of the cams **20*a*—20*a*—20*a***, corresponding to the opposed nature and disposition of the sets of opposed piston and cylinder assemblies **5—6**, **5—6** and **5—6**, calls for evenly distributed phase angles between the cams of the respective sets of opposed assemblies.

This question of the phase angles of the respective sets of multi-row opposed assemblies has great significance, and permits a deeper insight and understanding into the operation of the pump or motor of this invention as hereinafter will be more specifically pointed out in connection with the operation of the pump or motor. The above described rule for the relative disposition and rotation of cams **20*a*—20*a*—20*a*** with respect to each other and with respect to the cam shaft **17** will be readily understood from Figs. 9, 10 and 11 if one superimposes the three cams upon cam **20*a*** of Fig. 10. In this manner one obtains the diagram of Fig. 21 wherein the cam of Fig. 9 will correspond to ***c*1—*c*2**, the cam of Fig. 10 will correspond to ***a*1—*a*2** and the cam of Fig. 11 will coincide with ***b*1—*b*2**. That this disposition of Fig. 21 is equivalent to the disposition of Fig. 22 was already pointed out above more specifically.

For the sake of the principle of the invention, the cam phase angles for a one set, for a two set, and for a three set opposed piston and cylinder type pump are shown in Fig. 23, Fig. 24 and Fig. 25 respectively. It will be seen also that these latter figures enable one to compare the opposed piston type pump of the present invention with the radial piston type pump of conventional design, as far as the equivalence of the two different type of pumps with regard to the effect of the number of their pistons. Accordingly, a one set opposed piston pump with either single or double acting cams is equivalent with a two piston radial pump. A three set opposed piston pump is equivalent with a six plunger radial pump. See Fig. 22 and Fig. 23.

This combination offers the free and unrestricted side spaces for the main intake and exhaust passages 3 and 4. Fig. 3 and also Fig. 29 show that these convenient and straight passages qualify the pump for a flange suspension at either end of the pump around suspension lip 37 by means of suspension tapped holes 16, and at the same time intake and delivery of fluid is also possible from either end of the pump. This means that the pump may be suspended as in Fig. 3 about lip 37 but the piping will be reversed and connected with the left ends of fluid passages 3 and 4. Fig. 3 shows the possibility for such a reversal of the piping, and the flow of fluid in the main passages. By the same finish of the ends of the said passages, pipe connection and fittings will be interchangeable and will fit equally both ends of the passages. In this manner the plugs of the closed ends will be transferred into the open ends and the piping of the pump may thus be reversed.

An important object of the invention comprises the use of a high quality metallic block 1 as pump housing, without any covers, bolts or nuts, attachments or externally removable parts. As shown in Fig. 1 and Fig. 2 this is possible by the virtue of the specific arrangement of the pistons 5 and cylinders 6 with regard to the internal oil circuit of the pump. See Fig. 29. The arrangement is such that all hydraulic means and functions are provided in and within the solid block 1, and all mechanical and driving means and functions are provided in and within central housing bore 2 and in direct load transmitting relation with the bore wall 2. Such suitable material as nitralloy or manganese bronze may be used for making the housing block 1.

It will be understood that while I choose the form and shape of the block housing 1 as of a prismatic or rectangular shape in the preferred embodiment, a cylindrical shape of circular cross-section also may be used for great advantage for certain applications. In a cylindrical solid block, made out of cold rolled finished bar stock material instead of cold rolled rectangular bar stock, the stock may be sewed up for the required length, and then finished from there on. Central bore 2, cylinder bores 6 and fluid main passages 3 and 4 will be bored with equal ease as those of the rectangular body stock 1.

The opposed piston and cylinder assembly 5—6 and 5—6 as shown in Fig. 1 and Fig. 2 has an unique disposition in the housing block 1 with regard to the central bore 2, main passages 3 and 4, and individual intake and exhaust passages 7—11 and 9—12. As shown in Fig. 1 and Fig. 2, the individual intake and exhaust passages 7—11 and 9—12 are connecting the main passages 3 and 4 with the intake ports of the opposed assemblies 5—6 and 5—6. These individual passages 7—11 and 9—12 are separated by their coacting valves and valve chambers, such as 8—31 and 10—32 respectively. Each suction valve 8 and its coacting valve chamber 31, and each pressure valve 10 and its coacting chamber 32 is thus in proper communication with its respective piston and cylinder assembly 5—6, and main supply line 3 and 4.

For efficient flow and efficient production reasons, the first portion 7—7 of the individual suction passages, and the first portion 9—9 of the discharge passages are in one line and opposite to each other with respect to the respective main fluid passages 3 and 4, and both sets of passages 7—7 and 9—9 are parallel with each other and the axis of the coacting opposed piston and cylinder assembly 5—6 and 5—6. In a similar manner the other portions of the individual intake and discharge passages 11—11 and 12—12 which connect valve chambers 31 and 32 with the ports of their coacting assembly 5—6 are in one line and parallel with each other, and in communication with the outer ends of their coacting assembly 5—6. The center lines of the combined passages 7—11 and 9—12 of an upper assembly 5—6 are parallel with the corresponding center lines of the combined passages 7—11 and 9—12 of a lower assembly 5—6 as in Fig. 2. Thus the total center lines of the individual passages of a set of opposed piston and cylinder assemblies comprises a rectangular shaped passage such as 7—11 and 9—12, and 7—11 and 9—12 respectively. The parallel sides of the rectangular constitute the different passages which communicate with the valves 8—10 and 8—10. Valve chambers 31—32 are located at the corners of the right angled parallelogram. At diagonally opposite mid-points of the right angled parallelogram we find the ports of the opposed piston and cylinder assemblies 5—6 and 5—6 in communication with the passages 11—12 and 11—12, and the main passages 3 and 4 of the pump. The axis of all above four passages lie in a transverse plane of the block 1. The said transverse plane contains also the axis of the respective opposed assemblies 5—6 and 5—6. This plane of one set of total individual passages for a set of opposed pistons and cylinders is also normal to the longitudinal center meridian plane of the block 1 which meridian contains all the axes of all the opposed assemblies 5—6 and 5—6. The plane of the axes of the main intake and exhaust passages 3—4 and the plane of all the axes of all the opposed pistons and cylinders determine a conjugated set of main central meridian planes by their respective parallel axes, which meridians are both longitudinal and at right angles to one another.

Both ends of the central housing bore 2 are tapped or otherwise provided for receiving fluid tight plugs 24 and 25 respectively, thus providing a fluid tight lubricated chamber 2 for the mechanical means 17—20—21 of the pump. The inner faces of plugs 24 and 25 are finish-machined in such a manner that a tight push-fit against the coacting side faces of bearing means 21a and 21a will seal the fluid inside the bore 2. The shaft seal 26 is provided in plug 25. For providing outlet flow for the excessive slip from the inside of the bore into suction passage 3 of the pump, a small orifice or equivalent low pressure flow restrictive means, such as a relief valve is provided as at 27. In this manner, if the orifice is designed for 100 p. s .i., chamber 2 will contain a bath of 100 p. s. i. pressurized lubricant. The rotating driving and thrust transmitting rolling devices, such as 17—20a and 21 will be force-feed lubricated. Impeller shaft 17 ends in a splined piece 18 so that in suspension drive a female spline may engage it and with the impeller 17 for driving relation.

The pump may be suspended in any desired position as to the directions of the above named conjugated main meridians of the block 1. It is, however, preferred that the meridian of all axes of the assemblies 5—6 and 5—6 be in vertical position as in Fig. 2 to thereby eliminate side wear caused by the forces of the weight of the pistons 5 against their cylinder bores 6.

Intake and exhaust valve means 8 and 10 are selected for simplicity and durability so that dirt and foreign matter in the operating fluid medium will not damage the working surfaces of the hardened and lapped balls. A light intake and discharge spring is provided as at 13 and 14 for prime-seating their valves 8 and 10 before pressure is raised by the operation of the pump. After the pump starts to operate, pressure will be raised to seat both of the valves 8 and 10. In a similar manner, pistons 5 are provided with a prime-seating spring 15 to seat the pistons upon their coacting cam means 20 before the raising of sufficient pressure by the pump. Chamber plugs 33 and 35 are provided with extension means 34 and 36 so that the valve lift may be set and limited for efficient operation of the valves. I do not want, however, to limit my invention to this one form of valve means, as the application of the pump will more specifically determine the best form or type of the valve which automatically may function under the given application. Such other valve, for instance, is a poppet valve, diaphragm valve, etc.

Referring now to the second embodiment of this invention, as shown in Fig. 5 and Fig. 6 and Figs. 12, 13 and 14 respectively.

The arrangement of the parts and the hydraulic circuit of the pump is the same as that of embodiment one, except the cam means 20a—20a—20a of the first embodiment are changed to cam means 20b—20b—20b. The keyed-up relative positions of the cam means 20b—20b—20b are shown more specifically in Figs. 12, 13 and 14 respectively, and in Figs. 21, 22 and 25. As more clearly shown in Fig. 13, the opposed pistons and cylinders **5—6** and **5—6** are now not in the same pumping cycle, but are in exactly the opposed pumping cycle, in 180° phase angle difference. Thus when upper piston **5** is in its pressure cycle, lower piston **5** is in its suction cycle, and vice versa. A 60° or 120° phase angle rotation of the three different sets is equivalent and the same as the 120° rotation of the sets, because of the 180° phase angle difference between the assemblies **5—6** and **5—6** of one opposed set as shown in Fig. 21 and Fig. 22. In final result, the upper piston **5** of Fig. 13 will be in 120° phase angle difference with the lower pistons **5—5** of Fig. 12 and Fig. 14, while the lower piston **5** of Fig. 13 will be in 120° phase difference with upper pistons **5—5** of Fig. 12 and Fig. 14. From the above it can be seen that the differences between the first and second embodiments of the invention lie in the nature of the respective cams **20***a* and **20***b* as well as in the hydraulic nature of the opposed combinations of both embodiments. In the first embodiment the double acting cam means effects simultaneous suction and pressure cycles in the respective opposed assemblies of a set, while in the second embodiment the single acting cam means effects different cycles in the opposed assemblies of a set.

The arrangement of suction and pressure valves **8** and **10** and intake and discharge chambers **31** and **32**, intake and discharge individual passages **7—11** and **9—12** is the same as in Fig. 2 of the first embodiment. Since, however, single acting cams **20***b*—**20***b*—**20***b* are different in their mechanical characteristics against speed, load and frictional engagement, it is useful for best results to closely consider these characteristics in connection with a specific application of the pump. There is, however, further noteworthy differences between a pump of the first embodiment of Fig. 1 and Fig. 2, and of the second embodiment of Figs. 5 and 6. In the first embodiment the double acting valve arrangement **20***a*—**20***a*—**20***a* provides hydrostatic balance between the opposed pistons and cylinders **5—6** and **5—6** and thus any transverse load incident to the building up of pressure in these cylinders will be in balance and cancelled out by the opposed assemblies. Thus impeller shaft **17** and shaft bearings **21** will be fully relieved from any and all hydraulic thrust loads. This means longer bearing life and longer pump life at all operating pressures.

The pump of Figs. 5 and 6 is subjected to the full amount of the hydraulic piston load, and therefore the bearings **21** of Fig. 5 will wear out much faster than the same bearings of Fig. 1 under the same load and speed conditions. It will be noted also that the camshaft **17** and the double acting cams **20***a*—**20***a*—**20***a* in the first embodiment constitute perfect static and dynamic balance for the rotating masses of the sub-assembly **17—20***a*, while in the second embodiment the same sub-assembly **17—20***b*—**20***b*—**20***b* presents a new problem as to balance and torsional vibrations. The cams **20***b*—**20***b*—**20***b* are press fitted or keyed under the proper phase angle 60° or 120° respectively to the camshaft **17** with the camshaft bearings **21** and spacers **22** therebetween. Also there will be a marked difference between the operation of the valve means as hereinafter will be pointed out, when the valves **8** and **10** operate in a single or double acting pump. It will be seen that three cams **20***b*—**20***b*—**20***b* are in static and dynamic balance in any position of camshaft **17**, but one cam alone would not be.

It can be seen then, that in spite of the fact that the cam means **20***b*—**20***b*—**20***b* of the second, and **20***c*—**20***c*—**20***c* of the third embodiment of this invention are in static and dynamic unbalance individually with respect to their camshaft **17**, the cam and camshaft assembly as a whole will be in static and dynamic balance. Namely, the cams of both latter embodiments are rotated against one another with 120° phase angle, that is in evenly distributed radial positions around the camshaft **17**, and therefore they balance each other about the axis of said camshaft for both static and dynamic balance. Since the cams are always evenly distributed around the camshaft, irrespective of the number of cams provided there is more than one and the number of sets of opposed assemblies **5—6** and **5—6** respectively, it is clear that the present design of the driving and thrust transmitting means **17** and **20** will provide always a balanced combination of cams and camshafts. This is a great advantage of my opposed piston and cylinder type of pumps, single or multi-row, over the radial piston and cylinder type of pumps in which there is but one cam for all the pistons and therefore it is in static and dynamic unbalance. The balancing of a single acting cam means additional space and weight. The present invention eliminated the necessity of such balance, and therefore the additional space and weight for such a balance.

Figs. 7 and 8 and Figs. 15, 16 and 17 show the third embodiment of the invention. This embodiment is similar also to embodiment two of Figs. 5 and 6. The difference being in the utilization of a planetating thrust ring **29** which serves as an outer race member of an anti-friction bearing assembly **28** around the periphery of a single acting cam **20***c*. The thrust ring **29** or cam **20***c* includes, as the case may be, a recess **30** in its periphery so that a full complement elongated roller bearing assembly **28** may be inserted in said recess **30** for free individual and rotating rolling motion betwen the cam **20***c* and recess **30** as shown by the arrow. The cam assembly of this embodiment is designated by **20***c* in distinction from the previously described cams **20***a* and **20***b* of the first and second embodiments respectively.

The relative rotation of cams **20***c*—**20***c*—**20***c* with respect to each other upon the camshaft **17** is 60° or 120° as that of the cams of the previous embodiments. In this manner there will be six suction and six pressure strokes of the pistons **5** during every revolution of the pump. The advantages of the third embodiment over the second is obvious in larger size pumps where the eccentricity and stroke of the pump will be too great for the sliding friction and the accompanying development of heat between cams **20***c* and pistons **5**. This friction may become a menace for the maintenance of an effective wedge shaped oil film, and therefore in such larger pumps and motors it becomes necessary to employ a relatively stationary thrust ring such as **29** instead of the directly rotating cams **20***c*—**20***c*—**20***c*. The construction and operation of the full complement roller bearing **28** will be more specifically described hereinafter in connection with the operation of this invention. It will be seen that the three sets of cams **20***c*—**20***c*—**20***c* will be in static and in dynamic balance in any position of the camshaft **17**.

Operation of the pump or motor

From Figs. 9 to 17 inclusive it can be seen that the units of the three opposed sets of pistons and cylinders operate not only as a multirow unit, but each opposed set will constitute and consequently operate as a different pumping unit because there is a rotation between the cams **20** of the respective rows relative to each other, preferably with an angle of

$$\alpha=\frac{360}{n} \text{ and } \alpha=\frac{360}{2n}$$

where n is the total number of the opposed piston and cylinder assemblies utilizing single acting or double acting cams respectively. For $n=3$, $\alpha=120°$ phase angle for single acting cams, while for $n=3$, $\alpha=60°$ phase angle for double acting cams. It will be seen that due to the effect of the relative rotation of the dead center positions of the cams there will be a corresponding and similar rotation of the pumping cycles and the wave patterns of the assemblies which will result in a uniform flow of fluid of pump supply and delivery.

In the first embodiment of the invention shown in Fig. 1 and Fig. 2 more particularly, it can be seen that in the middle set of opposed assemblies as in Fig. 10, both pistons **5—5** are in their outer dead center positions, that is at the end of an outwardly going piston compression stroke. After this both pistons will start to descend on the cam and move inwardly, which movement corresponds to a suction stroke in each **5—6** assembly. The upper **5—6** assembly of Fig. 9 is 60° behind its outer dead center position following a clockwise rotation of the camshaft **17**, and in both sub-assemblies **5—6**, upper and lower, the pistons **5** are moving outwardly to reach the peak points of the double acting cam **20***a*. The opposed assemblies of Fig. 11, **5—6** and **5—6** upper and lower, are 60° ahead of the assemblies of Fig. 10 and they are in descending piston positions which means suction in both assemblies. Since the pressure is the same at both sides of the camshaft **17**, because the upper and lower assembly **5—6** and **5—6** are in the same pumping cycle, the resulting load of the operation of the pump as far as hydraulic load is concerned upon the camshaft **17** is naught. And this is an outstanding feature of pumps of the first embodiment of the invention. The hydrostatic balance of the pistons reflects also mechanical and dynamic balance upon the bearings **21** of the camshaft **17** and this mechanical and dynamic balance means long, useful life, silent operation, and the applicability of high unit pressures in the pump. Also the double acting nature of the cam **20**_a_ means a shorter sliding pressure contact between cam profile and piston since there are two pressure and two suction cycles during each revolution of the pump. The intermediate suction cycles between pressure cycles will give a chance for the cams and pistons to cool and be ready for the next pressure cycle. Also from the consideration of Fig. 10 it will be seen that in a simplified pump wherein only one single set of opposed piston assemblies is used, such as **5—6** and **5—6**, there will be four suction and four delivery strokes during each revolution of a single stage pump. Although two pistons actually act as one piston, yet in an high speed pump the delivery will be so smooth that it supersedes any commercial expectation. And the reason for it is that in spite of the duplication of the cycles of the opposed pairs **5—6** and **5—6**, the wave pattern of this first embodiment is such that the pressure waves of each piston **5** as well as the suction waves follow each other in 90° intervals due to the double acting of the cams **20**_a_. Therefore, the resultant flow of a single set of opposed piston and cylinder assemblies alone such as shown in Fig. 10 is equivalent in smoothness to a four plunger unit of the second embodiment. A four plunger unit of this first embodiment is equivalent to an eight plunger unit of the second embodiment. According to one concept of this invention, in pumping units utilizing double acting cams, a two row pump will have 90° phase angle between the two cams **20**_a_—**20**_a_ of the two sets of assemblies **5—6**, **5—6** and **5—6**, **5—6** respectively.

Referring to Figs. 1 and 2 of the first embodiment, it will be seen that due to the limitation of vibration and deformation in the support **2** of the camshaft **17** and camshaft bearings **21**, the driving and thrust transmitting means will be able to keep their alignment and will operate smoothly and without excessive vibrations at substantial load transmission. When the slip fills up the inter-spaces of bore **2** and the chamber formed by it and the end plugs **24** and **25**, the excess fluid will leave the chamber through slip hole **27** at a predetermined set pressure. Thus the vital moving parts of the pump are well under pressure lubrication at all times. Slip hole **27** may be provided with an orifice or with a small relief valve as the case may be.

With reference to the operation of valves **8** and **10** and their associated spring elements **13** and **14**, it is important to harmonize the valve lift with the actual stroke of the coacting plunger **5** in order to obtain smooth flow and valve operation. The maximum lift of the valve is a function of the size of the valve seat, the flow as well as of the speed of the pump. Chamber plugs **33** and **35** can be so designed that plug legs **34** and **36** will limit the maximum lift of the valves to a predetermined amount and thus give sufficient time for the valves to return to their seats by the working pressure of the pump. Since the suction valve **8** will be seated simultaneously with the opening of the pressure valve **10**, it will be understood that the closing of valve **8** and the opening of valve **10** is effected by the discharge pressure of the piston and cylinder assembly **5—6** and not by the valve springs **13** and **14**. The valve springs **13** and **14** are only for the purpose of pre-seating the valves **8** and **10** during the idle periods of the pump. The pressure valve **10** always will be seated and kept closed by the working pressure of the main pressure line **4** as soon as pressure is raised in the line, while the suction valve also will be kept closed by the working pressure.

The profiles of the cams **20**_a_ may be modified without departing from or sacrificing the spirit of the invention. The valve lift should be calculated for uniform passage flow in **7—11** and **9—12** during the peak velocity of the piston **5**. The first rule that ¼ of the diameter of the valve seat should be used as lift gives satisfactory results. The performance of the second embodiment shown in Fig. 5 and Fig. 6, and the third embodiment shown in Figs. 7 and 8 can be observed most conveniently from the developed illustration of the cams and camshafts as in Figs. 12 to 14 and Figs. 15 to 17 respectively.

The single acting cams **20**_b_ and **20**_c_ will perform substantially in the same manner, and effect one pressure stroke and one suction stroke during each revolution of the pump. The phase angle of the cams is also the same, and is given above by the invention as

$$\alpha=\frac{360}{n}=\frac{360}{3}=120°$$

The points of the dead center lines of the cams, however, mark different suction cycles. Fig. 13 and Fig. 16 show the cams and the pistons in their dead center positions. The upper assembly **5—6** is in an outer position which marks the end of a pressure stroke in each figure. Fig. 12 and Fig. 15 show a difference of 60° between the pistons and the cylinders of Figs. 12 and 13, and in a similar manner the pistons and cylinders of Figs. 14 and 17 show a difference of 60° in phase angle with regard to the pistons and cylinders of Figs. 13 and 16. Since, however, the pistons of an opposed set of embodiments two and three are not in the same phase, but in 180° difference, it is obvious that in rotating the cams this phase difference must be taken into consideration. In the calculation of the absolute and relative phase angles of the assemblies 180°—60°=120° is the relative phase angle. A clear visualization can be obtained by keeping the center cam and its assemblies in their vertical fixed positions as shown in Fig. 13 and Fig. 16 respectively, and then rotating the other two assemblies, evenly distributing all three sets of opposed piston and cylinder assemblies around the camshaft. Then it will be seen that since there are three rows of pistons, there is a direct distance of 120° between consecutive rows of pistons, but there is 60° distance between subsequent pistons of the rows. The effect of the opposed pistons brings about the existence of these two main figures, 60° and 120° in phase angles, whereas actually there is only 120° between the various opposed sets, and there is only 60° between the consecutive single assemblies of all the various opposed sets.

Needle roller assemblies **28** and thrust ring **29** around the single acting cam **20**_c_, as eccentric drive, reduces sliding friction between cam **20**_c_ and piston **5** to the amount of the tangential component of the eccentricity of the cam **20**_c_, which is very small as compared to the one-half of the periphery of the cam profile without the bearings **28** and thrust ring **29**. In cam drives where the single acting cam has to act bodily with sliding frictional thrust engagement with the ends of the pistons, the path of the friction extends to the entire periphery of the cam instead of the above small portion just described. The elongated roller bearings when spaced to the total of one needle roller diameter per bearing show a remarkable efficiency of rolling and carrying load with capillary fluid cage as lubricating film around the entire periphery of the needles. Additionally, recess **30** also contains fluid by capillary attraction, and this energized fluid in the recess acts as a shock absorbing medium during the pressure cycles of the pistons against shock load.

With reference to the technological processes for finishing the pump, it will be seen from Fig. 1 and Fig. 2 that the opposed cylinder bores **6—6** and the in line bores **7—7**, **9—9**, **11—12** and **11—12** of the individual intake and exhaust passages respectively are all in opposed aligned position with the horizontal main meridian plane of the pump housing **1**, and therefore in one boring operation always an opposed set of cylinder bores or fluid passages may be bored. Also in production set-ups, all six holes for the cylinders **6** may be bored in one multi-spindle operation. While the passage bores are smooth enough for the passages **7—11** and **9—12**, the cylinder bores **6** must be finish bored or lapped and ground in nitrited nitralloy steel blocks. Lapping and grinding operations need the free passage of the lapping and grinding tools through the two opposed cylinders **6—6**, and therefore it can be seen that the block set-up with open holes is suitable for rapid boring and mass production lapping processes. The blind ends of the passage holes **7—11** and **9—12** are plugged with tight permanent plugs as at **38** and **39** respectively. Copper gaskets **40** and piston plugs **41** will assure pressure tight sealing means at the outer ends of the pistons and cylinders **5** and **6**.

Hollow pistons **5** may be chromium plated for extreme hardness and maximum durability, and for the reduction of sliding friction between pistons **5** and cylinders 6. When the block 1 is made out of hard nitralloy, I prefer to make the pistons 5 also out of nitralloy and operate the assembly 5—6 on all hard steel. For smaller pump units a very efficient combination of material is that of hard manganese bronze block 1 with chromium plated pistons 5. With the availability of dry ice and deep freeze chambers, it is very advantageous in cam and shaft assembly to cool one piece and heat the other, and thus obtain a real squeeze fit between cams 20 and camshaft 17 after the equalization of the temperatures. In this manner keys and key-ways may be avoided, as they always mean undue weakening of one member or the other, and the distortion thereof by the cuting of the key-way.

The three embodiments of the invention shown and described herein for the purpose of illustration are the present preferred forms of the invention, but it will be understood that changes may be made in the specific construction and relative arrangement of the parts without departing from or sacrificing the invention as defined in the appended claims.

I claim:

1. In a fluid pressure pump the combination of a solid block housing, a central bore in said housing, a plurality of sets of opposed piston and cylinder assemblies located in a vertical meridian plane of said block, evenly distributed along the longitudinal axis of said block and parallel with respect to one another, each set including two opposed piston and cylinder assemblies, driving and thrust transmitting means operatively interposed between the opposed assemblies of each set for actuating the same, and plug means operatively engaging the respective ends of said driving and thrust transmitting means for sealing the ends of said bore and positioning axially said first named means in said bore, a liquid supply passage, a liquid discharge passage, said liquid supply passage and said liquid discharge passage being formed entirely inside of and by said block housing and disposed in opposed parallel position to each other longitudinally in said block and in a plane normal to the plane of said opposed piston and cylinder assemblies, valve means having individual fluid passages for connecting each set of the opposed assemblies alternately with said supply passage and said discharge passage respectively, said individual fluid passages for each set of valve means and each set of opposed assemblies respectively being disposed in a transverse plane containing the axis of said assemblies and normal to the axis of said driving and thrust transmitting means, said opposed assemblies including pressure responsive means for actuating the pistons during suction cycles of the pump.

2. In a fluid pressure pump the combination of a solid block housing, a central bore disposed in a longitudinal direction in said housing, one set of opposed piston and cylinder assemblies, located in a vertical meridian plane of said block, said set including two opposed piston and cylinder assemblies, driving and thrust transmitting means operatively interposed between said opposed assemblies for actuating the same, a liquid supply passage, a liquid discharge passage, said liquid supply passage and said liquid discharge passage being formed entirely inside of and by said block housing and disposed in opposed parallel position to each other longitudinally in said block and in a plane normal to the plane of said opposed piston and cylinder assemblies, valve means having individual fluid passages for connecting each opposed assembly alternately with said supply passage and said discharge passage, said individual fluid passages for said valve means and said set of opposed assemblies respectively being disposed in a transverse plane containing the axis of said assemblies and normal to the axis of said driving and thrust transmitting means, said driving and thrust transmitting means disposed in said central bore in concentric relation thereto, and means for sealing the liquid leaking from said cylinders into said bore and forming a fluid tight chamber therewith for pressure lubrication of said driving and thrust transmitting means.

3. In a fluid pressure pump the combination of a solid block housing, a fluid tight chamber disposed in said housing, sets of opposed piston and cylinder assemblies located in a vertical meridian plane of said block, evenly distributed along the longitudinal axis of said block and parallel with respect to one another, each set including two opposed piston and cylinder assemblies, thrust transmitting means operatively interposed between said opposed assemblies for actuating the same, a liquid supply passage, a liquid discharge passage, said liquid supply passage and said liquid discharge passage being formed entirely inside of and by said block housing and disposed in opposed parallel position to each other longitudinally in said block and in a plane normal to the plane of said opposed piston and cylinder assemblies, valve means having individual fluid passages for connecting each opposed assembly alternately with said supply passage and said discharge passage, said individual fluid passages for each set of valve means and each set of opposed assemblies respectively being disposed in a transverse plane containing the axis of said assemblies and normal to the axis of said driving and thrust transmitting means, and flow restrictive means operatively interposed between said chamber and said supply passage for effecting communication between said fluid tight chamber and said liquid supply passage and for relieving said chamber from an excess of fluid supplied by the pressure slip of said assemblies.

4. In a fluid pressure pump the combination of a solid block housing, a fluid tight chamber disposed in a longitudinal direction in the center of said housing, sets of opposed piston and cylinder assemblies located in a vertical meridian plane of said block, evenly distributed along the longitudinal axis of said block and parallel with respect to one another, each set including two opposed piston and cylinder assemblies, driving and thrust transmitting means operatively interposed between said opposed assemblies for actuating the same, a liquid supply passage, a liquid discharge passage, said liquid supply passage and said liquid discharge passage being formed entirely inside of and by said block housing and disposed in opposed parallel position to each other longitudinally in said block and in a plane normal to the plane of said opposed piston and cylinder assemblies, valve means having individual fluid passages for connecting each of said assemblies alternately with said supply passage and said discharge passage, said individual fluid passages for each set of valve means and each set of opposed assemblies respectively being diposed in a transverse plane containing the axis of said assemblies and normal to the axis of said driving and thrust transmitting means, said last named means including pressure responsive means for sustaining a predetermined fluid pressure in said fluid tight chamber for the purpose of force-feed lubricating of said driving and thrust transmitting means.

5. A housing block of polygonal cross section for fluid pressure pumps including a central cylindrical bore, a plurality of sets of opposed cylinders in said housing block and opening in said bore and arranged at diametrally opposite points thereof, each set of opposed cylinders including two opposed cylinders having a common axis, the axis of all sets being parallel with each other and lying in a common meridian plane of said housing, a pair of parallel bores one at each side of said central bore and extending parallel therewith, the axes of said parallel bores and the axis of said central bore lying in a second meridian plane of said housing, said first and second planes being normal to one another and intersecting each other longitudinally in the axis of said center bore.

6. In a housing block for fluid pressure pumps including a central longitudinal bore, a plurality of sets of opposed cylinders in said housing and opening in said bore and arranged at diametrally opposite points thereof, each set including two opposed cylinders, the axes of all sets of cylinders lying in a common meridian plane of the housing block, a pair of parallel bores lying in a meridian plane normal to the meridian plane of the cylinders, a plurality of sets of rectangular shaped fluid passages provided respectively for the sets of opposed cylinders and in communication therewith, said fluid passages lying in the respective transverse planes of the housing block and passing through the respective axes of said cylinders and connecting said parallel bores with said opposed cylinders at their outer ends respectively.

7. A fluid pressure pump including a housing, a central bore in said housing, a plurality of sets of opposed piston and cylinder assemblies in diametrally opposite positions in said housing, a liquid supply passage, a liquid discharge passage, said liquid supply passage and

said liquid discharge passage being formed entirely inside of and by said housing and disposed in opposed parallel position to each other longitudinally in said housing and in a plane normal to the plane of said piston and cylinder assemblies, driving and thrust transmitting means operatively interposed between said opposed assemblies for actuating the same and located in said bore, said means including a camshaft and axially spaced cam means fixedly secured to said camshaft one being provided for each set of opposed assemblies, said cam means including a plurality of double acting cams evenly distributed about said camshaft for effecting static and dynamic balance of said first named means, anti-friction bearing means interposed between said camshaft and said housing bore to transmit the hydraulic thrust load between said camshaft and said assemblies to said housing bore, and means for closing the ends of said housing bore.

8. A fluid pressure pump including a housing, a central bore in said housing, a plurality of sets of opposed piston and cylinder assemblies in diametrally opposite positions in said housing, each set including a pair of assemblies, a liquid supply passage, a liquid discharge passage, said liquid supply passage and said liquid discharge passage being formed entirely inside of and by said housing and disposed in opposed parallel position to each other longitudinally in said housing and in a plane normal to the plane of said piston and cylinder assemblies, driving and thrust transmitting means operatively interposed in said bore between said opposed assemblies for actuating the same, said means including a camshaft and axially spaced cam means fixedly secured to said camshaft, one cam means provided for each set of assemblies, said cam means including a plurality of single acting cams rotated at evenly distributed phase angles about the circumference of said camshaft, and a thrust ring about each cam in relatively rotating position thereto, and anti-friction bearing means interposed between said camshaft and said housing bore to transmit the hydraulic thrust load between said camshaft and said assemblies to said housing bore, and means for closing the ends of said housing bore.

9. A fluid pressure pump including a housing block, a central bore in said housing block, a plurality of sets of opposed piston and cylinder assemblies in diametrally opposite positions in said housing and about said central bore, a liquid supply passage, a liquid discharge passage, said liquid supply passage and said liquid discharge

passage being formed entirely inside of and by said block housing and disposed in opposed parallel position to each other longitudinally in said block and in a plane normal to the plane of said piston and cylinder assemblies, driving and thrust transmitting means operatively interposed between said opposed assemblies for actuating the same and located in said bore, said means including a camshaft and axially spaced cam means fixedly secured to said camshaft, each cam means including a single acting cam and associated anti-friction roller bearing assemblies, said bearing assemblies including an outer ring surrounding said cam in radially spaced relation thereto and a full complement of anti-friction needle roller bearings operatively interposed between said cam and said ring in load transmitting relation thereto, said cam means being keyed in evenly distributed radial positions with respect to said camshaft to evenly distribute the flow of fluid of the individual sets of opposed assemblies respectively, and provide static and dynamic balance for said first named means.

10. In a housing block for pressure pumps, a central longitudinal bore, at least one set of opposed cylinders in said housing and opening in said bore and arranged at diametrically opposite points thereof, said set including two opposed cylinders, the axis of said set of cylinders lying in a plane passing through the axis of said bore, fluid discharge and fluid supply passages integrally contained within said block and extending parallel to said bore on opposite sides of said plane, connecting passages provided respectively for the opposed cylinders and in communication therewith and with said fluid supply and fluid discharge passages, said connecting passages lying transversely of the axis of said bore.

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