Oct. 14, 1952 — A. G. HERRESHOFF — 2,613,651

ENGINE

Filed March 24, 1948 — 3 Sheets-Sheet 1

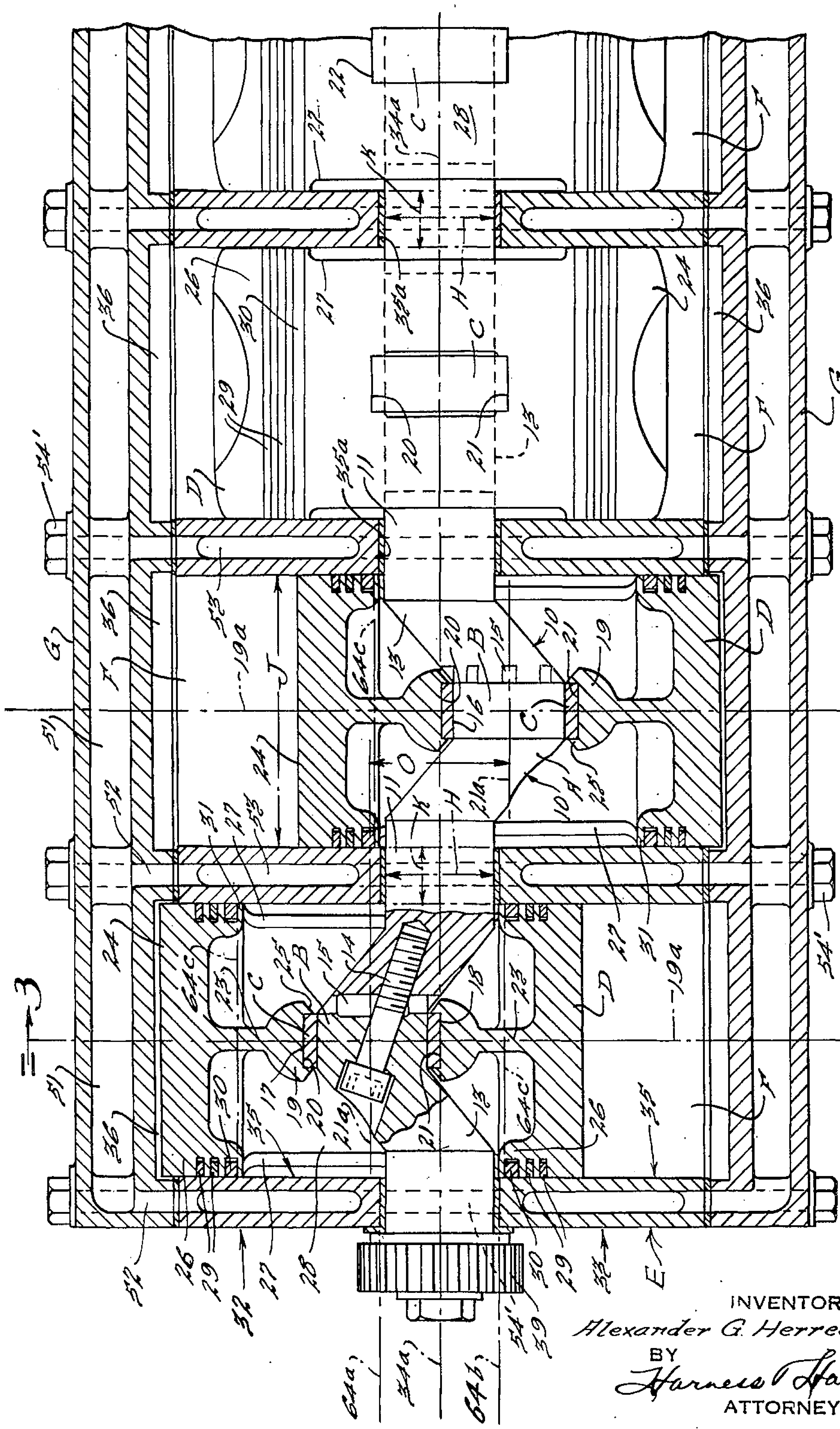

FIG. 1.

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Oct. 14, 1952 A. G. HERRESHOFF 2,613,651

ENGINE

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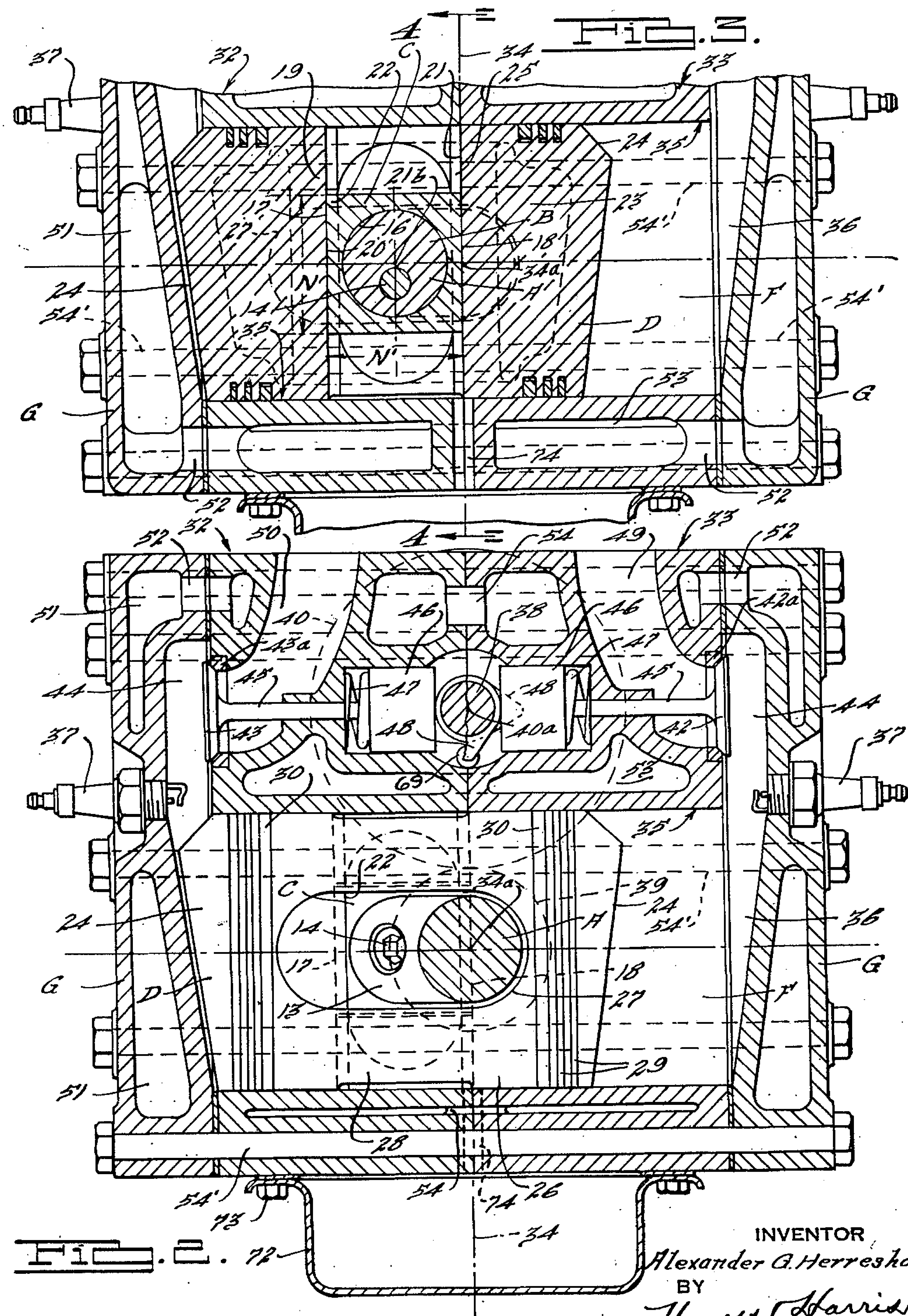

FIG. 3.

FIG. 2.

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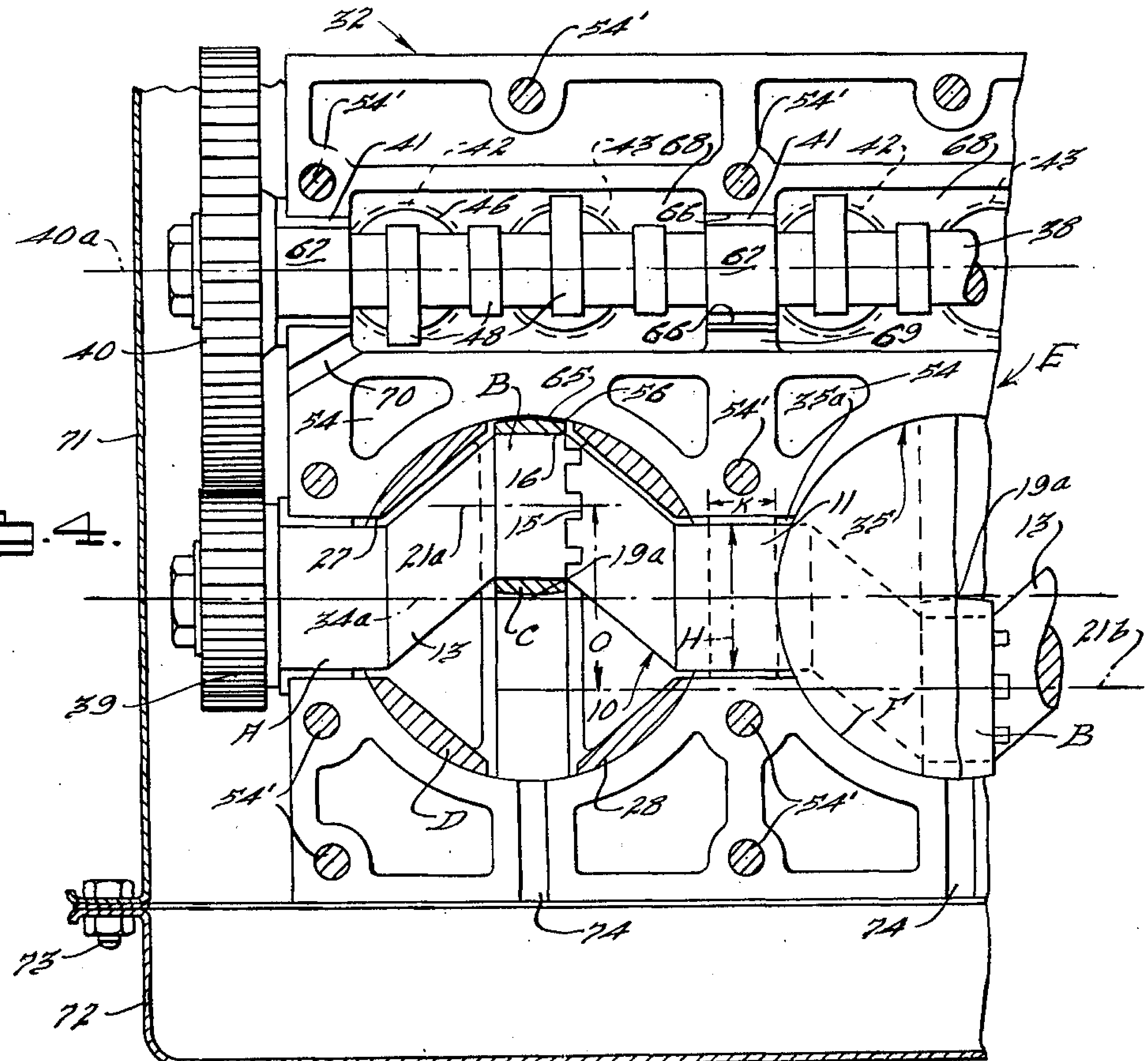

FIG. 4.

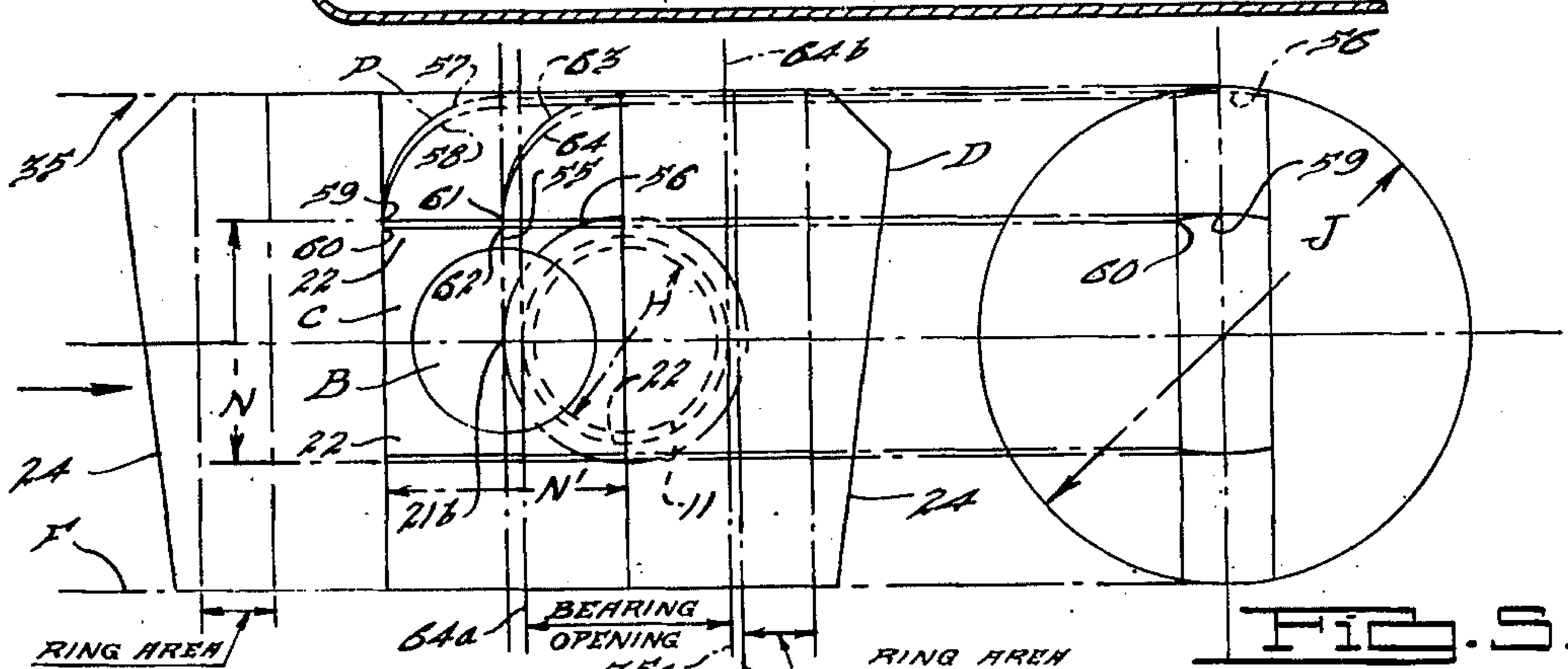

FIG. 5.

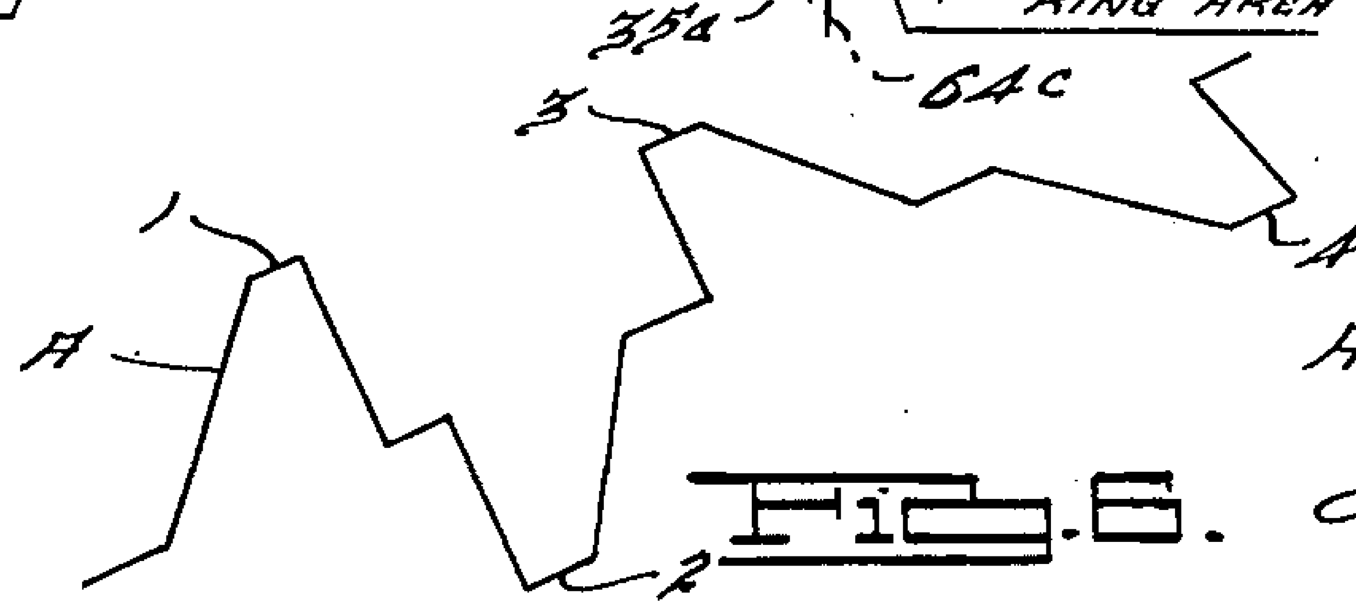

FIG. 6.

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UNITED STATES PATENT OFFICE

2,613,651

ENGINE

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Application March 24, 1948, Serial No. 16,801

2 Claims. (Cl. 121—120)

This invention relates primarily to internal combustion engines of the double acting type.

In the double acting type of engine a double ended piston is employed for reciprocatory movement in a working cylinder, such engines being arranged to provide a combustion chamber at each end of the cylinder for cooperation with each end of the double ended piston. Engines of this general type are known in the art and employ a crankshaft which is directly operably connected to the double piston without the use of a connecting rod of the more conventional type of engine.

It is an object of my invention to provide an engine having improved compactness for a given output whereby in comparison with known engines of equivalent output, an engine constructed in accordance with my invention occupies considerably less space. Such an engine is desirable for many uses as, for example, for driving motor vehicles wherein space saving, along with weight and cost reduction, are constantly striven for.

Another object is to provide an engine of improved simplicity of parts especially in connection with a double acting type of engine.

A further object of my invention is to provide an improved double acting engine having its parts so arranged as to provide improved mechanical relationships of the parts and at the same time minimize the overall dimensions of the engine for an engine of a given displacement.

Further objects of my invention in its more limited aspects are to provide a double acting engine of improved compactness in which, notwithstanding the necessity for the crankshaft to extend through the double ended piston, the piston rings do not overtravel the slots or openings in the cylinder; the double ended piston is of uniform diameter as is the cylinder in which the piston slides; the piston intermediate body structure, providing the ties between the piston ends, lies entirely within the cylinder bore; and the crank as well as at least those portions of the bearing block which are disposed immediately adjacent the piston slideway are so arranged as to at all times lie within the cylinder bore. In addition, in order that the optimum of compactness and engine output be obtained, I preferably locate the innermost piston ring of each piston head of the double piston as closely as practicable to the crankshaft and to the cylinder block openings for the crankshaft bearings when such piston head is in the extreme limit of its stroking movement inwardly.

In carrying out my invention, I have provided,

as a further object of my invention, a double acting engine of relatively large bore stroke ratio which is most desirable for compactness while at the same time realizing all the other objects of my invention. Heretofore, in engines of this type, it has been deemed necessary to provide a relatively small bore stroke ratio at a sacrifice of compactness or to incorporate structural arrangements at a sacrifice of other desirable characteristics. This is especially so in connection with engines dealing with conventional high gas explosion pressures, and while my invention is primarily directed to such engines it may also be of advantage in other machines utilizing unusually high pressure operation in the order of engine pressures rather than ordinary pressures usually dealt with in compressors, for example. However, generally speaking, compressor problems are quite different from engine problems and many arrangements acceptable for compressors would result in failure if employed in modern engine structures or in other high pressure machines.

Another object is to provide a double acting engine arranged for improved installation in a motor vehicle by reason of a novel construction and disposition of the cylinder block, valving, cylinder heads, and oil reservoir.

An additional object is to provide an improved and simplified assembly of engine block, crankshaft, and camshaft components.

Yet another object is to provide an improved lubricant drainage from the valve chamber, above the pistons, to the oil reservoir below the pistons.

Further objects and advantages of my invention reside in the novel combination and arrangement of parts, more particularly hereinafter set forth, reference being made to the accompanying drawings in which:

Fig. 1 is a longitudinal sectional plan view through a portion of the engine;

Fig. 2 is a transverse sectional elevational view through a typical cylinder of the Fig. 1 engine;

Fig. 3 is a sectional view of the left hand piston and slidehead or bearing block in their Fig. 1 positions, the view being taken as indicated by section line 3—3 of Fig. 1;

Fig. 4 is a transverse sectional view taken along line 4—4 of Fig. 3 but showing the bearing block of the piston at the left end of Fig. 1 in one of its extreme positions of movement laterally 90° from the position of Figs. 1 and 3, portions of the camshaft being also shown;

Fig. 5 is a diagrammatic view illustrating the motion of the bearing block relative to the piston;

Fig. 6 is a diagrammatic perspective view of the crankshaft.

In the drawings, reference character A represents the engine crankshaft structure which, for reasons of assembly of the engine, is shown built up in a generally conventional manner. This crankshaft structure has a plurality of sections 10 each composed of a central axial bearing portion 11, and offset crank-forming end portion B, and the diagonal portions 13 obliquely connecting each of the bearing portions with a pair of the terminal crank portions B. The sections 10 are appropriately connected in any well-known manner, as by fasteners 14 which join a crank portion B with an oblique portion 13 of an adjacent section 10, the engaged portions having matching serrations 15 to facilitate the assembly and increase the efficiency of the joint.

Each of the cranks B is embraced by a slide head structure or bearing block structure C which has a cylindrical bearing surface 16 having a bearing fit with a crank B and a pair of diametrically disposed bearing surfaces 17, 18 having a sliding fit in the slideway 19 of a double ended piston D. This slideway extends transversely of the common piston and cylinder axis 19^a and the elements of the slideway bearing surfaces 20, 21 engaged by surfaces 17, 18 extend parallel with a plane 21^a which contains the axis 21^b of the crank B and which is perpendicular with the piston axis 19^a, the same being also true of the elements of surfaces 17, 18. The slideway 19 is thus disposed perpendicular to a plane containing the piston axis 19^a and the axis 34^a of the bearing portions 11 of the crankshaft structure A. The slideway 19 and bearing block C are important in that they provide sufficient area to take the high thrusts under sliding action without developing undue wear and rattle. The bearing block, as will be presently more apparent, presents critical regions at its corners or regions 22 which lie immediately adjacent the opposite extremities of the surfaces 17 and 18 in a direction transversely of the piston axis, as viewed in Fig. 3.

The slideway 19 is carried by the struts 23 of the hollow double ended piston D, each strut extending centrally inwardly from an associated pressure-receiving head 24 of the piston, as shown in Figs. 1 and 3. The struts have marginal portions 25 extending inwardly beyond the slideway bearing surfaces 20, 21 to retain the block C in proper position axially of the crank B. The piston D is of generally cylindrical formation and is provided with an inwardly extending annular skirt portion 26 at each end. The heads 24 of the piston are tied or connected together structurally as, for example, by an intermediate body structure 28. This intermediate body structure is provided with a pair of diametrically disposed piston slots or openings 27 extending axially of the piston. The slots 27 are so arranged as to provide unobstructed reciprocation of the piston and are of minimum length such that when the piston is at its extreme limits of reciprocatory travel the respective ends of the slots 27 are disposed with minimum practicable clearance with the crankshaft structure in order to attain the maximum compactness for the engine.

Each of the skirt portions 26 is formed with a plurality of ring-receiving grooves opening outwardly toward the cylinder bore for accommodating the desired number of outwardly expanding compression rings, oil rings, and the like indicated at 29, the innermost ring being designated at 30, the location of the latter being of material significance in the more limited aspects of my invention, as will be presently more apparent. Each ring is provided with a support land, that for the innermost ring being indicated at 31.

The engine cylinder block structure E, according to my invention, is shown disposed such that the cylinders extend horizontally as shown in Fig. 2, but in the broader aspects of my invention the axes of the pistons may extend vertically or the engine turned at any other desired angle in which event the oil reservoir and certain accessories may require re-orientation as will be readily understood. It will be understood that terms of reference to the horizontal and vertical are employed only for convenience except as may be expressly otherwise set forth in the appended claims. Furthermore, my description will for the most part, be directed to a cylinder and piston assembly which is typical of the remaining cylinders and pistons of the engine.

The block E is formed in mating component half-portions 32, 33 having planar face contact vertically at plane 34 which passes through the axis 34^a of crankshaft structure A. The component portions 32, 33 are formed with companion cylinder bore portions 35 so that when the block portions are brought together each piston D will be slidably contained in what in effect is a single continuous uniform diameter cylinder F open at each side of the block and made up of the two coaxial cylinder bore portions 35, as shown in Fig. 2. The cylinder block portions are formed with pairs of mating or companion openings or recesses for receiving the bearing portions 11 of the crankshaft structure as will presently be more apparent. Thus the cylinder block portions together provide the openings 35^a for receiving the crankshaft structure. The cylinders F are closed at each end thereof by a cylinder head G seated at the side of the portions 32, 33 of block E, each cylinder head having a plurality of combustion chambers 36 for the multi-cylinder engine illustrated. Each combustion chamber closes one end of a cylinder F and is provided with the usual spark plug 37 where the engine is of the conventional spark ignition type.

The engine illustrated is of the four-stroke cycle type having a camshaft structure 38 extending longitudinally of the engine and driven from one end of the crankshaft by gears 39, 40 in Fig. 4. The camshaft has its axis 40^a in the plane 34 and is journalled in bearings 41 located at intervals above bearings 11 and in suitable mating or companion openings or recesses in the portions 32, 33 of block E. The camshaft operates intake and exhaust valves 42 and 43 respectively, it being understood that each combustion chamber 36 has an upwardly extending valve-receiving portion 44 accommodating the lift of a set of valves 42, 43 for controlling the fuel mixture intake to the chamber 36 and the discharge of exhaust gases therefrom as is well known for four stroke cycle engines in general. Each of the component cylinder block portions is provided with intake and exhaust valve seats 42^a and 43^a engaged by the respective valves 42 and 43.

The valves 42, 43 have their stems 45 extending toward camshaft 38 for operation by cam-followers or tappets 46, a spring 47 yieldingly seating each valve and maintaining the tappets in contact with the various cams 48 of the camshaft 38. The block portions 32, 33 are provided with intake and exhaust passageways 49, 50 extending from the seats 42^a and 43^a of the respective intake and exhaust valves **42**, **43** upwardly so as to open at the upper faces of block portions **32**, **33** for communication with conventional intake and exhaust manifolds (not shown).

For an engine of the liquid-cooled type, the cylinder heads G have cored chambers **51** for the circulation of a coolant, these chambers opening at **52** to the cored chambers **53** of the block portions **32**, **33** for surrounding the cylinders F with the coolant. The chambers **53** are open to each other for circulation of the coolant at **54**.

In order to prevent leakage of the coolant at the plane **34** where the block portions **32**, **33** are brought together in face contact, these face portions are preferably lapped and coated prior to assembly with a commercial rubber cement or other suitable sealing medium which will permit the block portions to be accurately fitted together to preserve the desired relationship in the component portions of the engine elements as, for example, the cylinders and the bearings for the crankshaft and camshaft.

The portions **32**, **33** of the block E are held together in assembled relationship, along with the cylinder heads G, crankshaft A, and camshaft **38**, by a plurality of tie-bolts **54'** which extend laterally all the way through the component cylinder block portions and through the cylinder heads G, the opposite headed ends of the tie-bolts being seated on suitable faces provided by the outer sides of the longitudinally extending cylinder heads G.

In carrying out my invention, I have provided certain physical relationships of the various parts in order to obtain maximum compactness of the engine which is of the greatest practicable importance. In automotive, aircraft, or even stationary power plant installations, engineers have striven to provide an engine which will produce more power for a given physical size, the latter also necessarily including weight and cost. In automotive installations such compactness in engine constructions is of the greatest importance in providing a minimum of space for the power plant and a maximum of space for the passenger or load compartment as well as a maximum of space for wheel suspensions, steering mechanism, and other vehicle equipment. This, of course, means that if such vehicle is powered commensurate with standard practice my engine would occupy much less space and would be much lighter and less costly to manufacture.

I have discovered that there are certain critical relationships of certain of the parts of a double acting engine, these being interdependent on each other so that if they are brought together according to my teachings then the desired compactness will result. On the other hand, if certain of the component relationships are not observed, then this in turn requires such distortion of other of the inherent relationships that the advantages of my invention cannot be realized, at least not to the extent possible by such observance.

In an engine having conventional carburetor type of aspiration, pressures in the combustion chamber are developed, with fuels of the most common octane rating, in the order of 600 p. s. i. (pounds per square inch) although with supercharging and for high octane fuels or for diesel engine practice pressures of 1200 p. s. i. and even more are utilized. In constructing a double acting engine, it is important that the

diameter H of the crankshaft bearings **11** should be a minimum consistent with allowable stress and bearing loads. For engines of the conventional carburetor aspirator type, I have determined that by utilizing available commercial forged material for the crankshaft structure, the ratio of the cylinder bore J to H should be in excess of approximately 2.4, say 2.4 to 2.6. If the ratio is materialy higher, then excessive deflection and loads are placed on the crankshaft leading to failures at bearings **11** and at the bearing block C, for example. If this ratio is materially lower, then H is too large in relation to J from a standpoint of necessary strength and, of more importance, such relationship exacts a magnified penalty on the ultimate size, weight, and cost of the engine for a given power output. The general relationship of J in terms of H is as follows:

$$\left(\frac{J}{H}\right)^3+\left(\frac{J}{H}\right)^2\left(\frac{K}{H}\right)-\frac{L}{2M}=0$$

where K is the length of the bearing **11**, L is the total allowable stress, and M is the applied pressure on the piston. For good practice, I have illustrated the bearings **11** as being of a length K equal to half their diameter. As the allowable stress of the material is about 20,000 p. s. i. it will be found that the ratio of J to H is 2.4. In time other materials may be discovered which will permit this ratio of approximately 2.5 to be increased which will be of advantage in following the teachings of my invention and in my reference herein to specific values of such ratio, I naturally contemplate such deviations therefrom as may be made possible by the utilizing of materials for the crankshaft which may be suitable as well as deviations occasioned by the use of combustion pressures higher than the assumed conventional pressure of around 600 p. s. i. However, for modern practice as a substitute for present popular automotive engines the ratio of approximately 2.5 is chosen as being determinative of achieving the improvements set forth herein for purposes of illustrating my invention.

With the values of J and H having been determined, the proper stroke of the engine is also determined thereby as will be presently apparent and is not to be taken as any arbitrary value according to my inventon. I determine the maximum proper engine stroke for the maximum of compactness as that value which will cause the bearing block C to travel in a path such that the continuity of the cylinder F will not be interrupted outwardly of the crankshaft axis 34^a at a distance greater than the interruption necessarily required for accommodating the crankshaft A whereby the innermost piston ring **30** will not travel over any such interruption in the cylinder wall. As it is generally desirable to provide the crankpin B with a diameter equal to H, and as the bearing block C surrounds the crankpin, it will be apparent that the sweep of the corner regions **22** of block C present critical limitations to the maximum proper engine stroke. However, by providing an engine stroke of approximately 50% of the value of bore diameter J, it will be found that my requirements can be readily fulfilled.

The minimum length N as well as the width N' of the bearing block C as seen in Fig. 3 should be, from the standpoint of practical considerations, approximately 1.25 times that of the crankpin diameter H. By proportioning the ratio of the bore J to the stroke O substantially

2 to 1, while providing adequate ties for the opposite piston heads **24**, then the travel of the advancing corner region **22** will describe an arcuate path P which, as seen in Fig. 5, will at all times lie within the cylinder bore.

In Fig. 5, the regions **22** are preferably curved at **56** in the direction of the thickness of block C concentrically with the cylinder so that they may swing closely to the cylinder with substantially minimum clearance and without breaking into the cylinder and thereby not penalizing the maximum available stroke. Such clearance can, as a practical matter, be maintained as little as .025 times the diameter of the crank portion **12**.

The travel of the extreme element of an advancing region **22** is illustrated by circular paths **57** and **58** for the respective points **59** and **60** on such element. It will be noted that these arcs do not cut the cylinder. In similar manner the travel of two points **61**, **62** lying in an element at the intermediate region **55** is shown by circular paths **63**, **64** respectively which likewise do not cut the cylinder and the end faces of the block C between the corner regions **22** are provided with the aforesaid curvature **56** concentric with the cylinder bore. Thus, in the engine illustrated, the crank portion B and block C lie within the cylinder bore during operation of the engine, this being especially desirable in connection with the crank and the corner regions **22**.

The offset of the crank portion B is of such amount in relationship with the cylinder bore J, and the bearing block structure C and slideway **19** are so constructed and arranged, that the regions **22** always lie within the bore of cylinder F when, during operation of the engine, the regions **22** are disposed outwardly axially of the bore from parallel planes **64**[a] and **64**[b]. These planes are perpendicular to the cylinder axis **19**[a] and are tangent with the crankshaft-receiving openings **35**[a]. By maintaining such relationship, the regions **22** will move in such a path that they will not require interrupting the cylinder bore axially outwardly from the interruption at **35**[a] necessary to receive the crankshaft structure. Therefore, if the piston rings clear to openings **35**[a] then they may be disposed to move close to these openings and yet always slide on uninterrupted portions of the cylinder bore even though, at times, the critical corner regions **22** of the bearing block structure C lie outside of the parallel planes **64**[a] and **64**[b]. In fact, in my arrangement, no part of the structure C or crank portion B cuts the cylinder bore outside of these parallel planes and this arrangement permits the rings to be positioned for close approach to the openings **35**[a] and the realization of maximum compactness for the engine as a whole.

It will be noted that the offset of the crank portion B, which determines the stroke O, is of such a predetermined amount in relationship with the diameter J of the cylinder bore F that, during operation of the engine, each of the oblique portions **13** of the crankshaft structure A will sweep a path adjacent the intermediate body structure **28** of piston D such that, when the piston is midway of its stroke as shown in Fig. 4, such oblique portion will be disposed in a direction transversely of the piston axis **19**[a] with substantially minimum clearance **65** with respect to a piston tie-portion or intermediate piston body structure **28** which lies between a piston opening **27** and one end of the slideway **19** and which is disposed at such time adjacent a plane perpendicular to the piston axis **19**[a] containing the axis **34**[a] of the crankshaft structure. My reference to minimum clearance will be readily understood as meaning the usual practical nominal minimum clearance for accommodating necessary shop tolerance operations. In production it is usually desirable to assemble crankshafts as forged and without machining except for the bearing and crank journals and where necessary to connect the crankshaft sections. Where such a forged crankshaft structure is employed, without machining the outer surfaces of the oblique portions **13**, and where the piston D is not machined as to these portions adjacent the region of minimum clearance at **65**, this minimum clearance should be about one-eighth of an inch plus or minus usual shop tolerances in order to most fully realize the advantages of my invention. Of course, if the crankshaft or piston is machined at the region **65** then this minimum clearance may be somewhat lessened. The opposite head portions **24** of the piston must, of course, be connected together sufficiently to maintain the desired rigidity of the piston to insure its proper operation and therefore adequate thickness of material must be imparted to the piston intermediate body structure **28**. When this is effected, certain of the important objects of my invention may be realized by providing the oblique crankshaft portions **13** disposed to move with substantially minimum clearance with reference to the piston tie portions thus insuring the maximum piston stroke and the maximum compactness for the engine.

In connection with the piston rings **29**, **30** it will be noted that they are so disposed, in relationship with the crankshaft-receiving openings **35**[a], as to always remain in engagement with portions of the cylinder bore which are disposed outwardly of these openings in a direction axially of the bore during operation of the engine.

An important characteristic of my invention in its somewhat more limited aspects resides in the disposition of the innermost piston ring **30** of each piston D for movement contiguous to the associated crankshaft bearings **11** and contiguous to the cylinder block openings **35**[a] for such bearings. By so locating these inner rings the length of the piston is kept to a minimum and, therefore, the width of the engine is minimized along with the optimum in compactness, low weight, and low cost. Furthermore, such ring location in a sense determines the maximum stroke of the piston as I deem it to be desirable to have the piston rings always engage the continuously cylindrical uninterrupted wall portions of the cylinder bore F to avoid danger of the rings expanding into cylinder openings with consequent damage to the rings, lands, cylinders, and leakage past the rings. Therefore, the rings should not overlap any cylinder openings at least to such an extent as to give rise to tendency of the rings to expand into any cylinder openings or to otherwise cause undesired wear of the cylinder and rings and jamming and breakage of these parts.

The innermost ring **30** associated with each piston head portion **24** has its inner surface boundary, axially of the piston, disposed in a ring-surface plane **64**[c] which is normal to axis **19**[a] and disposed as shown in Fig. 1 when such piston head portion is at the limit of its stroke inwardly of the cylinder bore. The stroke O is such in relationship with the cylinder bore size J and the bearing block structure C and slideway **19** are so constructed and arranged that the regions **22** always lie within the cylinder bore when, during operation of the engine, such region 22 is disposed outwardly axially of said bore from such planes 64^c.

The bearing block C is thus maintained of minimum length in the direction of slideway 19 such that the corner regions 22 remain within the cylinder walls at all times. Inasmuch as openings in the cylinder walls are necessary to journal the crankshaft and inasmuch as I have provided an arrangement which causes the piston rings to avoid overlapping such journal openings, certain of the advantages derived therefrom would be, at least to some extent, lost if the bearing block C was arranged for such movement as to require an opening in the cylinder wall into which the piston rings could expand with destructive results.

Certain of my engines arranged in accordance with my invention provide a bore stroke ratio J/O of 2, as aforesaid, with the desired good results. Obviously some latitude in the ratios set forth herein may be taken, and I have cited specific quantitative examples by way of illustrating the basic principles of my invention and not by way of limitation as a matter of exact figures. Likewise such engines incorporate a ratio of bore to crankshaft bearing diameter J/H of 2.5 and 2.6 and a ratio of stroke to crankshaft bearing diameter O/H of approximately 1.25.

The relationship of O/H is derived from

$$\frac{J}{O}=2 \text{ and } \frac{J}{H}=2.5$$

from which

$$\frac{O}{H}=1.25$$

thus further illustrating the desired relationship of stroke O in terms of crankshaft bearing diameter H dependent on a common relationship which each has with the bore J.

Any suitable crankshaft throw arrangement may be provided such as the Fig. 6 arrangement of cranks for the four cylinder engine illustrated. Cranks 1 and 2 are in the same plane but 180° from each other. Cranks 3 and 4 are likewise in a common plane and 180° apart. These two planes are related 90° from each other. This four cylinder engine is, in effect, an eight cylinder engine as each of the four cylinders hereinbefore discussed in detail provides two combustion chambers each having its piston head for operating the crankshaft. Obviously any number of cylinders may be used to construct an engine within the teachings of my invention.

The split cylinder block E facilitates the assembly of my engine. The bearing block C and pistons are assembled on crank portions which are thereafter connected at 14, 15. Then the block sections 32, 33 are brought together to contain the various parts shown in Fig. 2 and the tie-bolts 54′ secured in place.

The camshaft bearings 41 are formed in halves for assembly in the mating semi-cylindrical openings 66 of the block portions 32, 33 whereby the camshaft structure 38 is rotatably journalled at its journal portions 67. Between the openings 66 the cylinder block structure E is provided with component mating portions 68 providing cam chambers which accommodate rotation of cams 48, these chambers being open to each other by passages formed by component semi-cylindrical openings 69 in the faces of the block portions 32, 33. Each passage 69 extends in the direction of the camshaft axis 40^a and is located adjacent a bearing 41 as best shown in Fig. 4. Lubricant is supplied to the various moving parts in any appropriate manner (not shown) and the lubricant tending to collect in the cam chambers 68 drains through passages 69 thence to the front of the engine and through a similar passage 70 and downwardly in the casing 71 which houses the camshaft drive gears 39, 40 and thence into the oil reservoir or sump provided by the pan 72. This oil pan is appropriately secured to the under face of block structure E, when the engine is mounted as illustrated, by fasteners 73.

In order to relieve the hollow portions of each piston structure D of lubricating oil escaping from the main bearings 11 and the bearing parts at the crank portions B and slideway C, the block portions 32, 33 are provided with passages formed by the component mating semi-cylindrical openings 74. Each of these passages has its lower end open to the pan 72 and its upper end open to a cylinder F in the plane 34 so that each passage provides drainage from one cylinder to the oil pan. It will be apparent that the upper ends of these passages 74 are protected by the heads of each piston from exposure to the combustion chambers 36 or to the portions of the cylinders F which, during operation of the engine, lie outwardly from the piston heads in the direction of the cylinder axes 19^a. The power is taken from the rear end of the crankshaft structure 38 in a manner conventional with engines in general.

Engines of various desired capacities may be readily made following the teachings of my invention. Engines having cylinder bores larger than that illustrated will naturally develop more total pressure on the piston with consequent increase in size of crankshaft end bearing dimension H. This would have a tendency to limit the available stroke except for the necessary increase in crankshaft bearing diameter which, in turn, accommodates a stroke of substantially half the bore diameter. As fuels of higher octane rating become more available or with supercharged fuel mixture or with diesel cycles, higher compression ratios may be used thus increasing the piston load for a given size cylinder bore and requiring increase in size of crankshaft and dimension H. This will have the tendency to somewhat shorten the available stroke in relation to the bore and increase the ratio of J/O hereinbefore set forth. However, such ratio will still in the ultimate not greatly exceed the ratio set forth herein and even in such instances the maximum in compactness may be readily obtained by following the teachings of my invention.

Features of novelty not claimed but disclosed in my subject application are more fully disclosed and claimed in the copending applications of John P. Butterfield, Serial Nos. 16,645, 16646, 16,648, 16,649, 16,650, filed March 24, 1948, and Patent No. 2,571,198.

I claim:

1. A double acting engine comprising a cylinder block with a piston bore therethrough, a crankshaft formed of interconnected sections journaled in said block and extending transversely of said bore intermediate the ends thereof, said crankshaft having journal bearing portions at opposite sides of said bore extending at right angles to the axis of the piston bore and a crank located within said bore between said journal bearing portions of such size and shape as to be continuously positioned within said bore during crankshaft rotation, the diameter of the bore being approxi-

mately 2.5 times the diameter of the crankshaft bearing portions, a double ended piston mountel in said bore and pierced by said crankshaft intermediate the ends thereof, said piston having an interiorly disposed slideway, a unitary bearing block rotatably journaled on the crank and slidably connected to the piston slideway, said piston slideway extending transversely of the piston at right angles to the bore axis, said bearing block being of a length longitudinally of the slideway of not less than 1.25 times the crank diameter, said bearing block being shaped such that the sliding bearing surfaces thereof remain continuously within said bore during crankshaft rotation, and said crank being of such shape that the ratio of the diameter of the piston bore to the length of the piston stroke is approximately 2 to 1.

2. The engine structure set forth in claim 1 wherein the opposite ends of the sliding bearing surfaces of the bearing block are convexly curved to permit the bearing block to have the maximum stroke transversely of the piston while maintaining the sliding bearing surfaces continuously within the cylinder bore during crankshaft rotation.

ALEXANDER G. HERRESHOFF.

REFERENCES CITED

The following references are of record in the file of this patent:

UNITED STATES PATENTS

Number	Name	Date
545,502	Hennegin	Sept. 3, 1895
980,263	Holsman	Jan. 3, 1911
1,018,532	Twombly	Feb. 27, 1912
1,047,860	Twombly	Dec. 17, 1912
1,140,292	Sharpneck et al.	May 18, 1915
1,254,281	Salway	Jan. 22, 1918
1,287,797	Tietzmann	Dec. 17, 1918
1,473,602	Brockway	Nov. 6, 1923
1,578,228	Yearsley	Mar. 23, 1926
1,710,721	Bosmans	Apr. 30, 1929
2,170,099	Stubings	Aug. 22, 1939
2,404,906	Heald	July 30, 1946

FOREIGN PATENTS

Number	Country	Date
772,815	France	Nov. 7, 1934