

April 21, 1936.

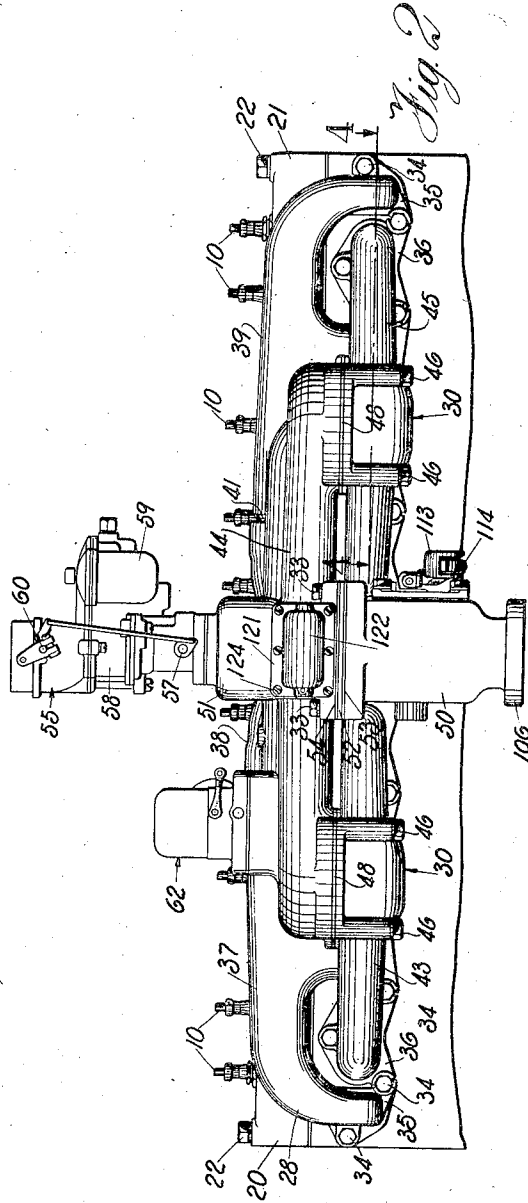
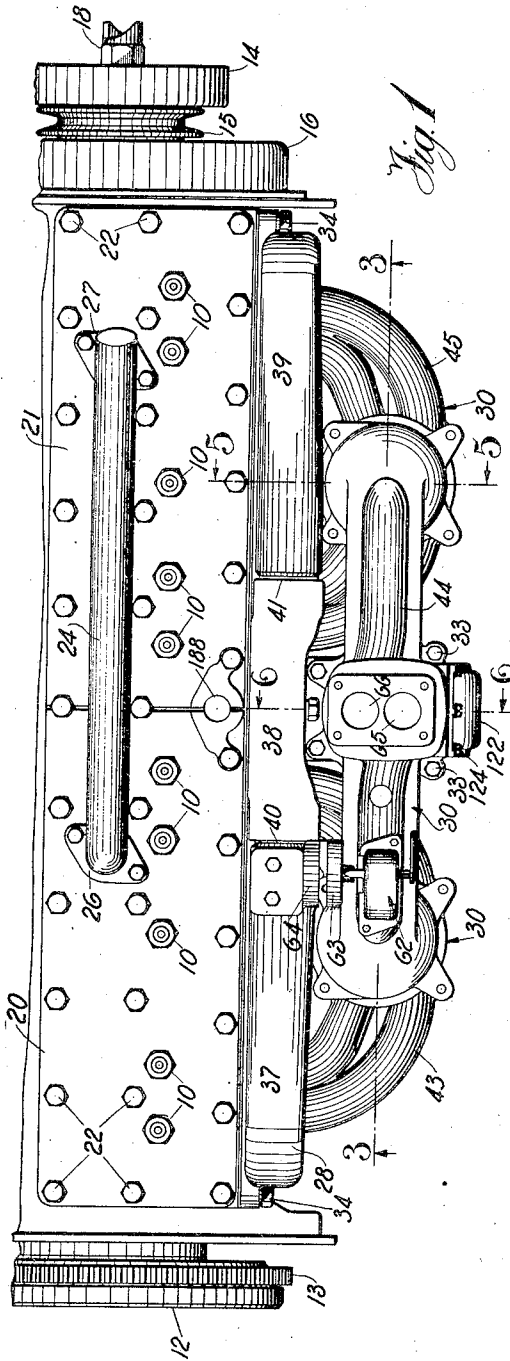
D. G. ROOS

2,038,085

INTERNAL COMBUSTION ENGINE

Filed Dec. 10, 1932

5 Sheets-Sheet 1



INVENTOR.
BY *Delmar G. Roos*
R. M. Pomeroy
ATTORNEY

April 21, 1936.

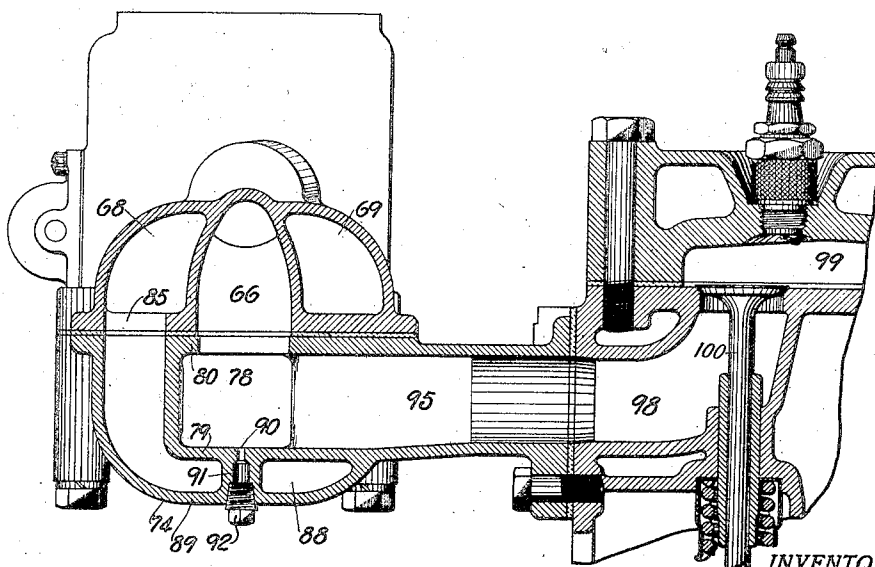
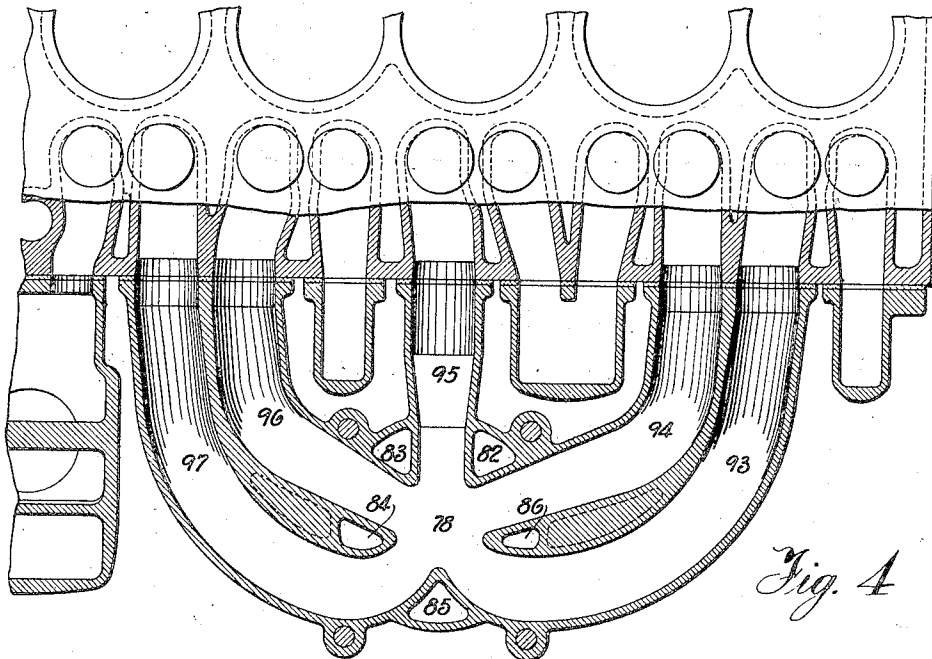
D. G. ROOS

2,038,085

INTERNAL COMBUSTION ENGINE

Filed Dec. 10, 1932

5 Sheets-Sheet 3



INVENTOR.
BY *Delmar G. Roos*
R. P. Ramsey
ATTORNEY

April 21, 1936.

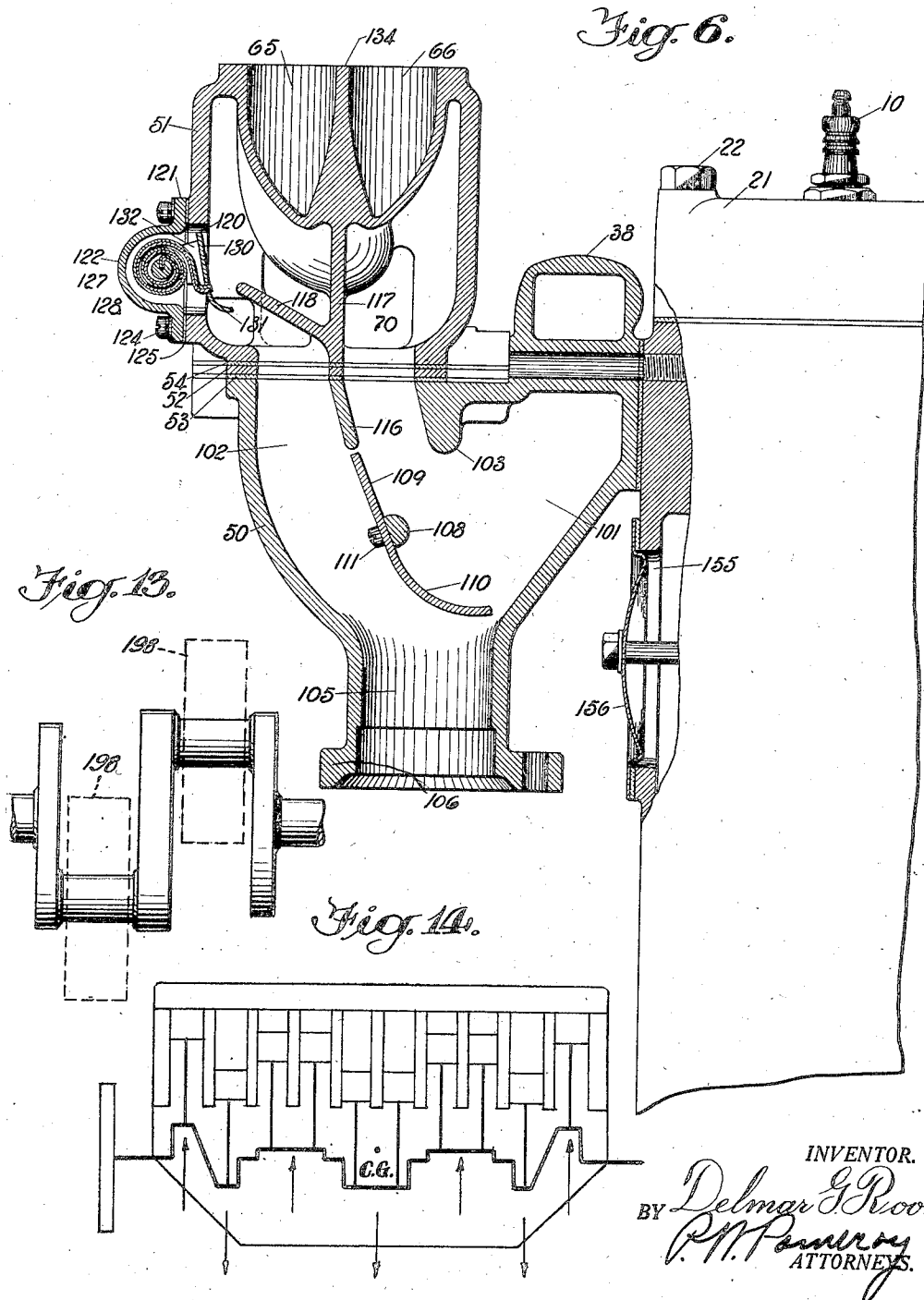
D. G. ROOS

2,038,085

INTERNAL COMBUSTION ENGINE

Filed Dec. 10, 1932

5 Sheets-Sheet 4



April 21, 1936.

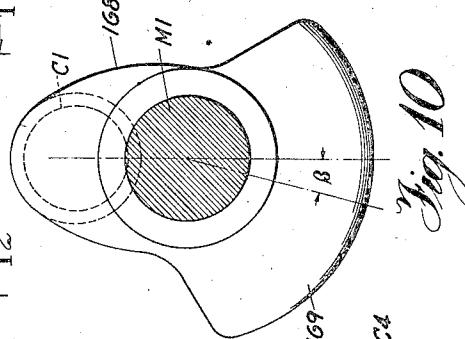
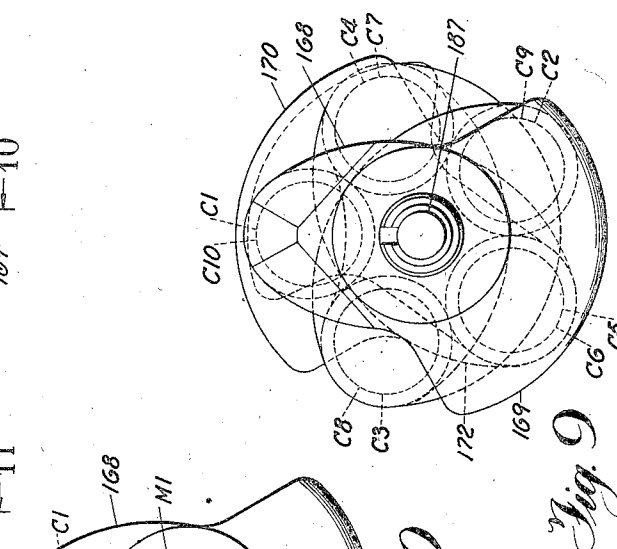
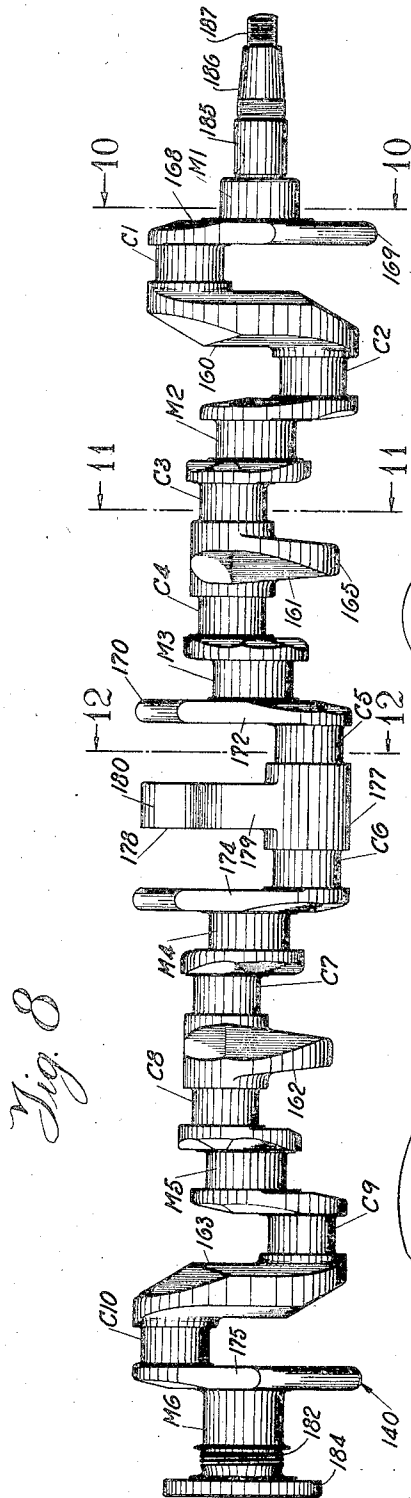
D. G. ROOS

2,038,085

INTERNAL COMBUSTION ENGINE

Filed Dec. 10, 1932

5 Sheets-Sheet 5



UNITED STATES PATENT OFFICE

2,038,085

INTERNAL COMBUSTION ENGINE

Delmar G. Roos, South Bend, Ind., assignor to
The Studebaker Corporation, South Bend, Ind.,
a corporation of Delaware

Application December 10, 1932, Serial No. 646,649

10 Claims. (Cl. 123—52)

This invention is a continuation in part of my application, Serial No. 568,968 filed October 15, 1931, and relates to improvements in internal combustion engines and particularly to an internal combustion engine having ten cylinders in line, and has for its principal object the provision of an in-line ten cylinder internal combustion engine which will develop the maximum horse power, and at the same time be smooth and quiet in operation.

A more specific object lies in the provision in an engine of the character described, of a system of manifolding a crank shaft and a timing arrangement which will be conducive to improved engine performance in matters of power output, fuel economy, smoothness of operation and durability of the engine.

A further object resides in the provision of an intake manifold by means of which each cylinder will be fed directly from a central distributing chamber and there will be no tendency for one cylinder to obtain a greater or less charge of explosive mixture than any other cylinder.

A still further object lies in the provision of a crank shaft so arranged that the inertia forces of the moving parts of the engine are balanced and one force will cancel out another force to provide smooth and quiet engine operation and reduce the stresses on the bearings and moving parts of the engine.

An additional object resides in the provision of a firing order which will so distribute the power loads upon the crank shaft as to minimize the torsional vibration in the crank shaft and at the same time will distribute the operating forces of the engine about the center of gravity thereof in such a manner that there will be no tendency for the engine to rock in either direction about its center of gravity.

Another object lies in the provision of an internal combustion engine of the character described which will be economical to manufacture, easy to operate and durable in use.

Other objects and advantages of the invention will appear as the description proceeds.

The accompanying drawings illustrate an acceptable mechanical embodiment of the idea of the invention. The drawings, however, are to be taken as illustrative only and not as limiting the invention, the scope of which is to be measured entirely by the scope of the sub-joined claims.

In the drawings:

Figure 1 is a top plan view of an internal combustion engine constructed according to the idea

of this invention, showing my improved intake manifold applied thereto.

Figure 2 is a vertical side elevation of the upper portion of an internal combustion engine showing intake and exhaust manifolds constructed according to the idea of this invention applied thereto.

Figure 3 is a vertical elevational view similar to Figure 2 showing a sectional view of the intake manifold taken on the line 3—3 of Figure 1.

Figure 4 is a sectional view through one element of the intake manifold and a portion of the cylinder block of the engine taken on the line 4—4 of Figure 2.

Figure 5 is a sectional view of the intake manifold taken on the line 5—5 of Figure 1 looking in the direction of the arrows.

Figure 6 is a vertical sectional view of the intake and exhaust manifolds in assembled relation taken on the line 6—6 of Figure 1.

Figure 7 is a vertical side elevational view of an internal combustion engine constructed according to the idea of this invention, a portion thereof being broken away to better illustrate the construction thereof.

Figure 8 is a longitudinal elevational view of a crank shaft forming an integral part of the internal combustion engine of this invention.

Figure 9 is a front end view of the crank shaft illustrated in Figure 8.

Figure 10 is a sectional view of the crank shaft illustrated in Figure 8 taken on the line 10—10 of Figure 8 looking in the direction of the arrows.

Figure 11 is a sectional view of the crank shaft illustrated in Figure 8 taken on the line 11—11 of Figure 8 looking in the direction of the arrows.

Figure 12 is an additional sectional view of the crank shaft illustrated in Figure 8 taken on the line 12—12 of Figure 8 looking in the direction of the arrows.

Figure 13 is a schematic illustration of the crank shaft showing the method of counterbalancing the same.

Figure 14 is a schematic view of the engine showing the manner in which the inertia forces are opposed to each other and distributed about the center of gravity of the engine.

Referring to the drawings in detail and particularly to Figure 1, it will be observed that I have provided an internal combustion engine of ten cylinders in line as indicated by the spark plugs 10. At the rear end thereof the engine is provided with a flywheel 12 which may be of conventional construction and carry gear teeth 13 which comprise a part of the self-starting mecha-

nism. At the forward end thereof the engine is provided with a vibration dampener 14, a fan belt pulley 15, and a timing gear casing 16, the vibration dampener and fan belt pulley being secured to the engine by means of a starting crank nut 18. The top of the engine as illustrated in Figure 1 is covered by a cylinder head formed in two parts, 20 and 21, secured to the engine cylinder block by means of the hold-down bolts 22. A cooling water conduit 24 extends along the top of the engine above the cylinder heads 20 and 21 and is operatively connected to the cylinder heads 20 and 21 by means of the flanged fixtures 26 and 27 respectively. An exhaust manifold 28 extends along one side of the engine and an intake manifold generally indicated at 30 extends along the same side of the engine in assembled relation with the exhaust manifold 28.

From an inspection of Figure 2, it will be observed that the exhaust manifold 28 and intake manifold 30 are secured together by means of stud bolts 33 and that the assembled manifolds are secured to the engine by means of stud bolts 34 and cooperating flanged portions 35 and 36 on the exhaust and intake manifolds respectively.

Referring again to Figure 1 it will be observed that the exhaust manifold 28 is formed in three parts as indicated at 37, 38 and 39, the end portions 37 and 39 being telescopically united with the center portion 38 as indicated at 40 and 41. The intake manifold 30 is also formed in three separate parts as indicated at 43, 44 and 45, the end portions 43 and 45 being secured to the center portion 44 by means of the stud bolts 46 illustrated in Figure 2, gaskets 48 being interposed between the adjacent surfaces of the members 43 and 45 and the member 44 to provide a gas-tight seal therebetween. A chamber 50 is formed on the center portion 38 of the exhaust manifold and underlies a chamber 51 formed in the center portion 44 of the intake manifold. An apertured plate 52 is inserted between the adjacent surfaces of the chambers 50 and 51 and gaskets 53 and 54 are disposed, one upon either side of the plate 52 between the plate and the adjacent surfaces of the chambers 50 and 51 respectively to form a gas-tight seal between the chambers 50 and 51, and, as explained above, the chambers 50 and 51 are rigidly secured together by means of the stud bolts 33. In the form of the invention illustrated, a down-draft carburetor generally indicated at 55 is mounted upon the top surface of the chamber 51. The invention, however, is not specifically limited to the carburetor as shown nor to a down-draft carburetor, as an up-draft carburetor may be used upon slight modification of the construction of the chambers 50 and 51. The carburetor illustrated is provided with a throttle valve 57, a Venturi chamber 58, a float bowl 59 and a choke valve 60. An automatic choke control device generally indicated at 62 and including a thermostat is mounted upon the intake manifold and may be operatively connected with the choke valve 60 by means of linkage, not illustrated, for the purpose of automatically controlling the choke valve in accordance with the operating temperature of the engine. A heating pad 64 is mounted upon the exhaust manifold and positioned in heat exchanging relation with the thermostat 63.

Attention is now directed to Figures 3, 4 and 5 for a more detailed description of the intake manifold. As illustrated in section in Figure 3, the center portion, or bar, 44 of the intake manifold, comprises a hollow member having two oppositely extending curved channels 65 and 66

leading from the top of the chamber 51 to the centers of the members 43 and 45 respectively. These are the channels through which the explosive mixture is drawn from the carburetor 55 to the manifold portions 43 and 45 for distribution therethrough to the individual cylinders. In addition to the channels 65 and 66, the bar 44 is provided upon each side thereof with additional channels as indicated at 68 and 69 in Figure 5, and at 69 and 70 in Figure 3. These additional channels extend along the channels 65 and 66 upon each side thereof and lead from the interior of the chamber 51 to the interiors of the heating chambers 73 and 74 formed in the manifold portions 43 and 45 respectively at the centers thereof. In all, there are four such additional passages extending along each side of each of the intake passages 65 and 66 from the interior of the chamber 51 to the interiors of the heating chambers 73 and 74 separated at their outer ends by partitions 75 and 76 respectively and these passages provide outlet and return channels for exhaust gases led from the chamber 51 to the heating chambers 73 and 74 to heat the intake charge before it is drawn into the cylinders. Each of the heating chambers 73 and 74 is provided interiorly thereof with a distributing chamber as indicated at 77 and 78 respectively in open communication with the intake channels 65 and 66, and with the interiors of the manifold portions 43 and 45 respectively. Each distributing chamber lies at the junction of the branches of the manifold portion in which it is located and is provided with a bottom wall 79 and a top wall 80 provided with suitable apertures for the passage of intake and exhaust gases there-through. Annularly spaced ports indicated at 82, 83, 84, 85 and 86 located between the adjoining branches of the manifold portion extend from below the bottom wall to the apertures provided in the top wall. A space 88 is provided between the bottom wall of each distributing chamber and the bottom of the outer wall 89 of the containing heating chamber.

Referring to Figures 1 and 5, it will be observed that the heating chambers 73 and 74 are substantially circular bowl-shaped castings, and that these bowl-shaped heating chambers are covered by substantially bowl-shaped portions formed on the ends of the bar 44 to provide a substantially spherical distributing and heating chamber at each end of the bar 44 and at the center of each of the manifold members 43 and 45. In the end of the intake manifold illustrated in Figures 4 and 5, the passages are so arranged that the exhaust gases flow to the heating chamber 74 through the channel 69 and return to the chamber 51 through the channel 68. Referring to Figure 4, it will be observed that two of the ports, namely 82 and 83, lie upon one side of the intake channel 66 under the heating channel 69 and as the other three, namely 84, 85 and 86, lie upon the opposite side of the intake channel 66 and under the heating channel 68, the heating gases will flow into the heating chamber through the ports 82 and 83, circulate beneath the distributing chambers 78 through the space indicated at 88 in Figure 5, and out through the ports 84, 85 and 86 and the return channel 68. As illustrated in Figure 5, a bore 90 has been provided leading from the bottom of the distributing chamber through a circular well 91 to the exterior of the heating chamber at the bottom thereof, and this bore is normally closed by a screw plug 92 which may be removed to drain condensed gasoline from the intake manifold if necessary. As 75

illustrated in Figure 4, each of the branches 93, 94, 95, 96 and 97 lead directly from the distributing chamber 78 to one of the engine intake ports. As illustrated in Figure 7, each of the intake ports as indicated by "I" is separate, that is, there are no Siamesed intake ports in this engine. This construction gives a direct and short intake passage from each intake port to the distributing chamber, thereby obviating the possibility of one cylinder starving an adjacent cylinder due to overlapping intake periods.

A further advantage of this construction resides in the fact that with the firing sequence which forms an integral part of this invention, namely a firing sequence in the order 1, 7, 2, 6, 3, 10, 4, 9, 5, 8, or the reverse, 1, 8, 5, 9, 4, 10, 3, 6, 2, 7, the suction impulses through the branches 93, 94, 95, 96 and 97 occur in a sequence in the nature of a regular and recurring rotation about the distribution chamber 78. This rotational effect tends to give a whirling motion to the mixture in the distributing chamber and obviates the possibility of pocketing or stagnation with consequent condensation of gasoline, and insures an adequate supply of properly evaporated fuel to all of the cylinders at all times. As indicated in Figure 5 it will be observed that the intake ports, one of which is indicated at 98, are short and direct and lead directly into the combustion chamber of the engine as indicated at 99. The particular engine illustrated is of the L-head type and the section illustrated in Figure 5 shows an intake valve 100 having a stem extending through the intake port 99 and a head at the junction of the intake port with the combustion chamber. It is to be understood, however, that the invention is not specifically limited to an L-head engine as illustrated and that the idea may with equal facility be applied to a valve-in-head or sleeve-valve engine or other equivalent forms of internal combustion engines.

The section illustrated in Figure 6 shows the construction of the interiors of the chambers 50 and 51 and the manner in which the exhaust gas is diverted from the exhaust manifold into the heating chambers when it is desirable to heat the intake charge. As illustrated the chamber 50 formed on the central portion 38 of the exhaust manifold comprises a substantially Y-shaped member having a branch 101 in open communication with the exhaust manifold, a second branch 102 opening into the bottom of the chamber 51 formed in the intake manifold, a partition 103 between the two branches and an outlet portion 105 extending downwardly from the junction of the branches 101 and 102 and provided at its lower end with an apertured flange 106 for the attachment thereto of an exhaust conduit, not illustrated. A shaft 108 extends transversely through the chamber 50 at approximately the center thereof and is rotatably mounted in the chamber walls. A valve member 109 having a curved lower end 110 is secured at approximately its center to the shaft 108 as by screws, one of which is indicated at 111. Referring now to Figure 2 it will be observed that one end of the shaft 108 extends outwardly of the chamber and has attached thereto a weighted arm 113 controlled by a thermostat 114. The construction of this portion of the device is fully illustrated and described in application, Serial No. 632,118 by Stanwood W. Sparrow for Heat control valve, filed September 8, 1932, now Patent No. 1,986,542, and it is believed that a further description is not essential to the disclosure of

the present invention. A partition 116 is provided in the upper portion of the branch 102 and forms a continuation of the valve 109 when the valve is in the operative position illustrated in Figure 6. This partition 116 is continued in the chamber 51 as illustrated at 117 and is also provided in the chamber 51 with a branch indicated at 118 extending at an angle from the partition 117 toward the outer wall of the chamber 51. From the structure so far described it will be observed that when the valve 109 is in the position indicated in Figure 6 exhaust gases entering the branch 101 from the exhaust manifold 38 will be diverted upwardly through the portion of the branch 102 between the partition 116 and the partition 103 into the chamber 51. By reason of the partition 117, the exhaust gases will then be diverted into the channels 69 and 79 from which they will flow through the heating chambers 73 and 74 illustrated in Figure 3, and return from the heating chambers through the channel 68 and the fourth channel, not illustrated, to the opposite half of the chamber 51. From this half of the chamber 51, the exhaust gases will flow around the outer end of the partition 118 and into that portion of the branch 102 opposite the partition 116 from the partition 103, and from this portion of the branch 102 through the outlet 105 into the exhaust conduit. An aperture 120 is provided in the outer wall of the chamber 51 opposite the outer end of the partition 118 and this aperture is covered by a plate 121 provided with a semi-cylindrical extrusion as indicated at 122. The plate is secured to the wall of the chamber 51 by means of the screws 124 or other equivalent securing means and a gasket 125 is interposed between the adjacent surfaces of the plate and the chamber 51 to provide a gas-tight seal between the plate and the chamber. A shaft 127 is rigidly mounted in the plate 121 along the axis of the semi-cylindrical extrusion 122 and carries a coiled thermostat 128, the free end of which is operatively connected with a valve member 130 having a curved lower end 131. The valve 130 is positioned between the outer end of the partition 118 and the adjacent inner surface of the plate 121 and is rotatably mounted upon the shaft 127 by means of apertured ears, one of which is indicated at 132. As the temperature of the thermostat 128 and its environment rises, the thermostat will exert a force to urge the lower end of valve 130 toward the outer edge of partition 118 and it will be observed that when the thermostat 128 is in a position to bring the curved lower end of the valve 130 against the outer edge of the angular partition 118, the valve 130 will provide a restriction to the flow of exhaust gases past the outer end of the partition 118, thereby causing a rise in the pressure of the exhaust gases in the heating channels upon the lower portion of valve 109 to overcome the effect of weight 113 whereupon the valve 109 will be moved to a position to cut off the flow of exhaust gas to the chamber 51 and heating of the intake charge will cease.

Figure 6 also illustrates in detail the manner in which the channels 65 and 66 are curved up to the top of the chamber 51 to open into the fuel outlet passages of the dual carburetor 55. The upper portions of these channels are separated by a central partition 134 and from their upper ends, as illustrated in Figure 6, each of the channels 65 and 66 is curved downwardly and has a

downward inclination throughout its length so that any gasoline condensed in the channels will flow by gravity to the bottom of the distributing chambers 77 and 78. The channels 65 and 66 are also curved inwardly toward each other so that outside of the chamber 51 the axes of the two channels are substantially co-incident.

Referring now to Figure 7, it will be observed that in the broken away portion of the drawings, I have illustrated a crank shaft generally indicated at 140 connected with pistons, one of which is indicated at 142, by connecting rods as indicated at 143, in the conventional manner. The pistons are reciprocally mounted in cylinders as indicated at 145 which are cast in a cylinder block 147 and substantially completely surrounded by water jackets as indicated at 149. A cam shaft 150 is rotatably mounted in the cylinder block and operatively connected with the crank shaft by means of timing gears contained in the timing gear casing 16. The cam shaft 150 operates intake and exhaust valves as indicated at 152 and 153 respectively, and in combination with the crank shaft establishes the firing sequence of the engine cylinders. The exhaust and intake valves are provided with springs, guides, tappets and adjusting means in the conventional manner. The cylinder block 147 is further provided with apertures as indicated at 155 in Figure 6 to afford access to the valve adjusting means and these apertures are covered by plates as indicated at 156.

In an internal combustion engine constructed according to the idea of this invention, the crank shaft 140 and the cam shaft 150 are so constructed and so related as to provide for a firing sequence in the order 1, 7, 2, 6, 3, 10, 4, 9, 5, 8, or the reverse order, 1, 8, 5, 9, 4, 10, 3, 6, 2, 7, assuming that the cylinders of the engine are numbered consecutively from one end to the other and that the numbers indicating the above firing sequence are the numbers applied in consecutive order to the engine cylinders. It will be observed that this particular firing sequence gives a succession of power impulses which alternate between the forward and rearward halves of the engine and proceed in consecutive steps from the center toward each end thereof. As the firing order is definitely dependent on the form of the crank shaft, a further discussion thereof will be deferred until the construction of the crank shaft as illustrated in Figures 8, 9, 10, 11 and 12 has been described.

A crank shaft involving to a considerable degree the principles and construction of the crank shaft forming an integral part of this invention has been previously described in the application of Delmar G. Roos, Serial No. 568,968, filed October 15, 1931. The present crank shaft, however, is regarded as an improvement upon the crank shaft illustrated and described in that prior application.

Referring to Figure 8, it will be observed that the crank shaft forming an integral part of this invention comprises ten connecting rod bearings, and six main bearings arranged in two similar but reversely positioned groups of five connecting rod bearings and three main bearings in each group. Each group of five crank bearings is integrally balanced insofar as it is possible to balance a group of an odd number of crank bearings, the unbalanced forces remaining in each group are then counter-acted by the manner in which the two groups are related in the com-

plete crank shaft. For the sake of convenience in description, the connecting rod bearings or crank bearings are numbered beginning at the front end of the crank shaft in consecutive order as C—1 to C—10 inclusive, and the main bearings are numbered beginning at the same end of the crank shaft as M—1 to M—6 inclusive. As illustrated in Figure 9, the consecutive crank bearings in each half of the crank shaft are spaced at a rotational angle of two-fifths of a circle from each other, thus C—2 is two-fifths of a circle from C—1, and C—3 is two-fifths of a circle or 144° from C—2. C—5 and C—6, the adjacent end crank bearings of the two halves of the crank shaft, are arranged in co-axial position relative to each other and this arrangement causes the bearings C—1 and C—10, C—4 and C—7, C—2 and C—9, and C—3 and C—8 to be positioned in co-axial relation.

The first four bearings C—1, C—2, C—3 and C—4 and the last four C—7, C—8, C—9 and C—10 are arranged in pairs with a flying crank as indicated at 160, 161, 162 and 163 between the two crank bearings of each pair. As the two crank bearings in each pair are spaced apart a rotational angle of 144° , they are not directly opposite each other and hence each crank bearing does not simply balance out the rotational force of the opposite bearing but there is a rotational force on one side of the crank shaft contributed to by both bearing pins. In order to balance out this rotational force in each pair, a counter-weight portion is formed on each flying crank as illustrated at 165 in Figure 11. This counter-weight portion is so proportioned that the combined centers of gravity of the two crank pins and the flying crank in each pair will lie on the axis of the crank shaft. It will now be observed from an inspection of Figure 9, that when the two pairs of crank bearings, C—1, C—2, C—3 and C—4, are placed together in assembled relation as illustrated in Figures 8 and 9 that this portion of the crank shaft is then out of balance in a manner which would give an unbalanced rocking or torsional vibration to the crank shaft. This force is compensated by extending the crank throws between the main bearing M—1 and the crank bearing C—1, and the main bearing M—3 and the crank bearing C—5 in the front half of the crank shaft, and the crank throws between the crank bearing C—6 and the main bearing M—4 and the crank bearing C—10 and the main bearing M—6 in the rear half of the crank shaft to form counter-weights as illustrated in Figure 10. The crank throw 163 between the main bearing M—1 and the crank bearing C—1 is illustrated in elevation in Figure 10 and this figure clearly illustrates that the counterweight portion 169 is rotated at an angle β from a position directly opposite the crank bearing C—1 so that the combined centers of gravity of the counterweight portion and the crank bearing C—1 will lie outside of the axis of the crank shaft. Referring to Figure 9, it will be observed that the counterweight portion 170 of the crank throw 172 between the main bearing M—3 and the crank bearing C—5 is rotated at an angle equal to β from a position directly opposite the crank bearing C—5 and that the counterweight portions 169 and 170 bear such angular relation to each other that a line joining the combined centers of gravity of the crank pin C—1 and the counterweight 169 and the crank pin C—5 and the counterweight 170 will pass at its center through

the axis of the crank shaft. The same system of counterweighting is applied to the crank throws 174 and 175 in the rear half of the crank shaft. It will be apparent from the above description that the four front crank bearings and the four rear crank bearings are placed in perfect static and dynamic balance in each group. This construction then leaves the two center crank bearings C—5 and C—6 and that portion 177 of the crank shaft which joins these two bearings entirely unbalanced. However, as the two crank bearings C—5 and C—6 and the portion 177 of the crank shaft which lies between these two bearings are all co-axial, the unbalanced effect of these crank bearings and the intermediate portion of the crank shaft can readily be counteracted by a simple counterweight 178 illustrated in elevation in Figure 12. This counterweight comprises a stem portion 179 and a weight portion 180, the weight portion being positioned directly opposite the crank bearings C—5 and C—6 and the end of the stem opposite the weight portion being formed integrally with the portion 177 of the crank shaft. This counterweight 178 is of such weight and shape as to bring the combined centers of gravity of the counterweight and the crank bearings C—5 and C—6 and the intermediate portion 177 of the crank shaft to the axis of the crank shaft.

The crank shaft is provided at the rear end thereof with oil grooves 182 and a clutch flange 184 and at the front end thereof with a timing gear bearing 185 and fan belt pulley and vibration dampener bearing 186 and a threaded end portion 187 to receive the starting crank nut 18.

It will now be apparent that in an engine constructed as described above, the pistons in cylinders 1 and 10 connected to crank bearings C—1 and C—10 will move together and will reach top dead-center simultaneously. Likewise the pistons in cylinders 3 and 8 will move together and will reach top dead-center simultaneously 72° of crank shaft rotation behind the top dead-center position of pistons 1 and 10. Pistons 5 and 6 will reach top dead-center together 72° behind pistons 3 and 8, pistons 2 and 9 will reach their top dead-center position together 72° behind pistons 5 and 6 and pistons 4 and 7 will reach top dead-center 72° behind pistons 2 and 9. If we consider the position of the crank shaft in relation to the engine at which piston No. 1 is at top dead-center having just completed its compression stroke and being about to begin its power stroke, piston No. 10 at that time will be at the top of its exhaust stroke and about to begin its intake stroke. The next two pistons coming up to top dead-center are pistons 3 and 8. However, piston No. 3 is in the same half of the engine as piston No. 1 and as it is desirable to alternate firing strokes between the two halves of the engine, cylinder No. 8 will be the next cylinder to fire. Following cylinder No. 8, cylinder No. 5 in the opposite half of the engine from No. 8 will fire, then cylinder 9 followed by cylinder 4 and the remaining cylinders will then fire in the sequence 10, 3, 6, 2, 7. This firing order will be readily apparent from an observation of the positions of the consecutively numbered crank pins in Figure 9. This is the preferred firing order and is the firing order which would preferably be applied to an engine in which the crank shaft is rotated in the clockwise direction in the conventional manner. It is quite possible, however, to construct an engine in which the crank shaft would rotate in the anti-clockwise direction in which case the firing

order would simply be the reverse of that stated above, namely, 1, 7, 2, 6, 3, 10, 4, 9, 5, 8. With the preferred firing sequence described above it will now be apparent that as the crank bearings C—1, C—8 and C—5 altogether describe an angle of less than 180°, which is the angle from top dead-center to bottom dead-center, this engine will have the distinct advantage that power from three cylinders will be continuously, overlappingly applied to the crank shaft, thereby giving an extremely smooth power flow from the engine. However, as three of the engine cylinders continuously overlap on the power stroke, it must naturally follow that three cylinders will continuously overlap on the intake stroke and from a careful observation of the firing sequence it will be observed that two adjacent cylinders in each half of the engine will be continuously overlapping on the intake stroke. When this condition occurs, that cylinder which begins its intake stroke first has a tendency to rob fuel mixture from the subsequently intaking cylinder, thereby tending to cause uneven power strokes among the different cylinders of the engine. This condition, however, has been entirely obviated by the intake manifold as illustrated in Figures 1, 2, 3 and 4 and described in detail above. This intake manifold besides assuring an independent and ample supply of explosive mixture to the individual engine cylinders, at the same time provides for adequate heating of the intake mixture by heat transfer from the exhaust gases.

Due to the unusual length of the engine illustrated and described herein, I have found it desirable to form the cylinder head in two sections as indicated at 20 and 21 in Figure 1 and to supply an individual outlet for cooling fluids as indicated at 26 and 27 in each portion of the cylinder head.

An aperture 188 has been provided between the two portions of the cylinder head for the extension therethrough of a commutator shaft adapted to actuate an ignition commutator, not illustrated. I have also found it desirable to reinforce the crank case of the engine by additional flanges and ribs as indicated at 189 and 190 in Figure 7. An oil pan 192 is secured to the bottom of the crank case in the conventional manner as by stud bolts 194 and is provided interiorly thereof with a drip pan 196.

While I have indicated in the above description that the crank shaft is statically and dynamically balanced as an integral unit, I have found it highly desirable in practice to actually over-balance the crank shaft in order to counter-balance the rotating mass of the connecting rods journaled on the crank shaft as well as to counter-balance the actual mass of the crank pins themselves. This construction is accomplished by simply temporarily adding to each crank pin a weight equal to the weight of the rotating mass of the connecting rod as indicated diagrammatically at 198 in Figure 13 and then accurately counter-balancing the crank shaft with the additional weights attached. It will be apparent that this construction necessitates the use of somewhat larger counter-weights than would be necessary if the weight of the crank pins alone were counter-balanced.

An inspection of diagrammatic Figure 14 clearly illustrates the fact that by providing a crank shaft of the form illustrated and described above, the rocking couples in the engine due to the inertia of the unbalanced rotating parts in each half of the engine are at all times equally

spaced upon each side of the center of gravity of the engine and are of equal intensity, whereby the force exerted by the rocking couple upon one side of the center of gravity of the engine cancels out the force exerted by the rocking couple on the opposite side of the center of gravity of the engine and entirely obviates any tendency of the engine to move about its center of gravity.

A consideration of the above description taken with the accompanying drawings will now indicate that I have provided an in-line ten cylinder engine having among others the following advantageous characteristics. In the first place the explosive mixture fed to the cylinders is adequately heated to completely evaporate the fuel charge before the mixture enters the cylinders and the heating thereof is automatically controlled in accordance with the operating temperature and power output of the engine. Secondly, each cylinder has a separate and independent intake passage from the cylinder to one of the two distributing chambers positioned opposite each half of the engine, and the sequence of intake impulses occurs in such order that the fuel mixture flowing through the distributing chamber is given a definite rotary motion about the vertical axis of the chamber to create a more complete mixture of the fuel and air comprising the mixture, and to prevent stagnation of any part of the mixture with a consequent condensation of fuel. Thirdly, the firing sequence occurs in such order that the power strokes alternate between the two halves of the engine and occur at equally spaced intervals, so that the power of three of the ten cylinders is continuously overlappingly applied to the crank shaft of the engine. Fourth, the crank shaft is so arranged and so counter-balanced that the effect of both torsional and rocking forces are entirely eliminated. Fifth, the particular structure of the crank shaft illustrated and described above provides a crank shaft having ample strength and rigidity which at the same time requires only six main bearings to support the ten crank shaft bearings, thereby providing for an engine length with practical and useful limits, and also obviating the necessity of reducing the width of the main bearings to a point where such bearings would not be durable in use, and lastly the engine described above has the important characteristics of a rhythmic and uniform flow of power coupled with an absolute dynamic balance of all moving parts which renders the engine smooth and quiet in operation to a superlative degree.

While I have illustrated and described a particular mechanical embodiment of the idea of the invention, it is to be understood that the invention is not limited to the specific device as illustrated and described but that such changes in the size, shape and arrangement of parts may be resorted to as come within the scope of the subjoined claims.

Having now described my invention so that others skilled in the art may clearly understand the same, what I desire to secure by Letters Patent is as follows:

1. A multi-cylinder four stroke cycle internal combustion engine having cylinders in line, a crank shaft having crank throws arranged in progressive spiral formation from the ends of said shaft to the center thereof, pistons in said cylinders, connecting rods operatively connecting said pistons with said crank throws, intake and exhaust valves, appropriate intake and exhaust ports for said valves, an ignition system having

a firing sequence such that a plurality of said engine cylinders overlap on their power strokes and adjacent engine cylinders overlap on their intake strokes, and an intake manifold having a central distributing chamber and independent conduits from said distributing chamber to each of said engine cylinders, said conduits surrounding said chamber in equally spaced relation to cause the fuel charge to have a rotary motion in said chamber.

2. In a four stroke cycle internal combustion engine having ten cylinders arranged in line and provided with suitable intake and exhaust ports, a piston in each cylinder, an engine crank shaft having ten throws angularly disposed from the ends of said shaft to the center thereof in such order that if the shaft be rotated the throws will pass through a radial plane including the shaft axis in the order 1-10-8-3-6-5-2-9-4-7, or the reverse according to the sense of rotation, reaching firing position in the order 1-7-2-6-3-10-4-9-5-8 or the reverse; connecting rods operatively connecting the several pistons with the several crank bearings; an intake manifold having five delivery ports connected with a central distributing chamber in each half thereof, and communicating each with a single delivery port in the engine; said delivery ports being arranged to cause the fuel charge to have a rotary motion in said chambers and to cause each cylinder to draw its firing charge directly from the associated central distributing chamber so that no two cylinders will draw mixture from the same delivery port of the intake manifold.

3. In an internal combustion engine having cylinders, pistons in said cylinders and suitable intake and exhaust ports connected with said cylinders, a balanced crank shaft, connecting rods operatively connecting said pistons with said crank shaft, said crank shaft having ten crank bearings no two of which in the same half of the crank shaft lie in the same radial plane, said crank bearings being angularly disposed from the ends to the center of said crank shaft in such order that if the shaft be rotated no two longitudinally adjacent crank bearings will pass through a radial plane including the shaft axis in succession, means for conducting combustible mixture to the intake ports of said engine, said means comprising separated distributing chambers and separate branches surrounding said chambers arranged to cause the fuel charge to have a rotary motion in said chambers and so that no two cylinders firing in succession draw mixture from the intake manifold adjacent to the same delivery port.

4. In an internal combustion engine of the four-cycle type having ten cylinders in line consecutively numbered 1 to 10, the combination of a crank shaft having crank pins arranged in independently balanced pairs and in two independently balanced groups of five consecutive crank pins, consecutive crank pins except the two center pins being spaced relative to each other at a rotational angle of two-fifths of a circle to provide for a firing sequence of the order 1-7-2-6-3-10-4-9-5-8, and a manifold having a plurality of distributing chambers and separate branches leading therefrom to each cylinder arranged to cause the fuel charge to have a rotary motion as it enters each of said branches.

5. In an internal combustion engine of the four-cycle type having ten cylinders in line consecutively numbered 1 to 10, the combination of

a crank shaft having crank pins arranged in independently balanced pairs and in two independently balanced groups of five consecutive crank pins, consecutive crank pins except the two center pins being spaced relative to each other at a rotational angle of two fifths of a circle to provide for a firing sequence of the order 1-8-5-9-4-10-3-6-2-7, and a manifold having a plurality of distributing chambers and separate branches leading to each of said cylinders arranged to cause the fuel charge to have a rotary motion as it enters each of said branches.

6. In an internal combustion engine of the four-cycle type having ten cylinders in line and a firing order of a sequence wherein at least two adjoining cylinders overlap on their intake periods, an intake manifold comprising a carburetor chamber, a plurality of distributing chambers, pneumatically connected with said carburetor chamber and a separate branch from said chambers to each cylinder, said branches being positioned in respect to said distributing chambers to cause the fuel charge to have a rotary motion as it passes through said distributing chambers.

7. In an internal combustion engine of the four-cycle type having ten cylinders in line and a firing order of a sequence wherein two or more cylinders in the same half of the engine overlap on their intake periods, an intake manifold therefor comprising, a separate distributing chamber for each half of the engine, and substantially equally spaced separate manifold branches surrounding the respective distributing chamber, and communicating with the engine cylinders where- by the fuel charge has a rotary motion as it passes through the distributing chamber.

8. In an internal combustion engine of the four-cycle type having ten cylinders in line and a firing sequence wherein two or more cylinders in the same half of the engine overlap on their intake periods, an intake manifold therefor com-

prising, a separate distributing chamber for each half of the engine, and a manifold portion interposed between each distributing chamber and the respective half of the engine, each of said manifold portions comprising five branches leading directly from the respective distributing chamber to the five cylinders in the respective half of the engine so arranged in respect to the firing sequence of the engine as to give the fuel charge a rotary motion as it passes through the distributing chamber.

9. In an internal combustion engine of the four-cycle type having ten cylinders in line and a firing sequence wherein two or more cylinders in the same half of the engine overlap on their intake periods, an intake manifold therefor comprising, a carburetor chamber substantially centrally located with respect to said engine, a separate distributing chamber for each half of said engine, and a header leading from said carburetor chamber to each of said distributing chambers, each of said distributing chambers having branches so arranged in respect to the firing sequence of the engine as to give the fuel charge a rotary motion as it passes through the distributing chamber.

10. In a multi-cylinder internal combustion engine of the four-cycle type, an intake manifold comprising, a carburetor chamber, a distributing chamber pneumatically connected with said carburetor chamber, independent manifold branches separately connecting the individual engine cylinders with said distributing chamber, said manifold branches being so arranged with respect to the firing sequence of said engine that said cylinders draw their intake charges in succession substantially around the circumference of said distributing chamber, thereby giving to the fuel charge in said distributing chamber a rotary motion about a vertical axis.

DELMAR G. ROOS.