July 20, 1937. F. A. HOWARD 2,087,619

AUTOMOTIVE INTERNAL COMBUSTION ENGINE

Filed Aug. 4, 1933 6 Sheets-Sheet 1

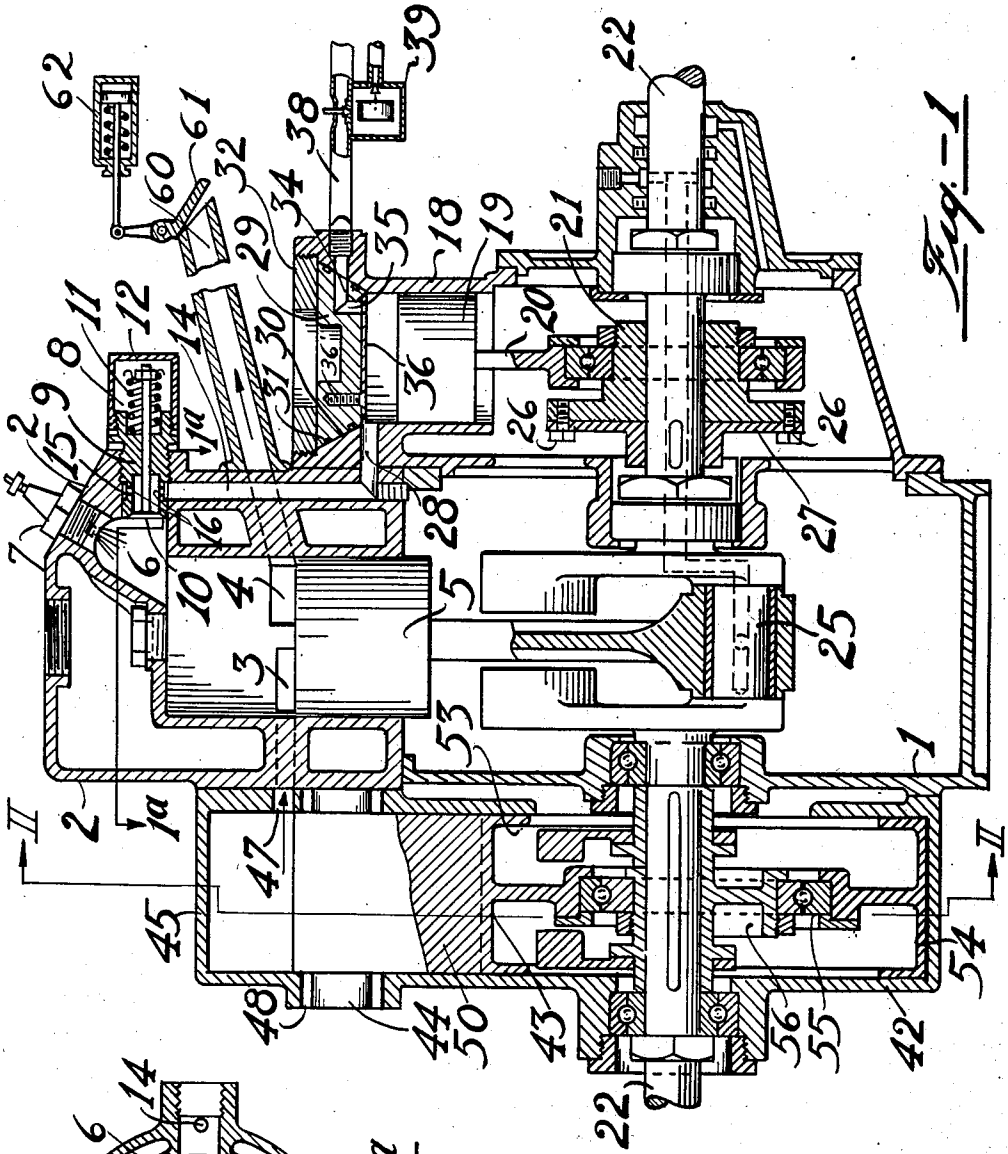


Fig. 1

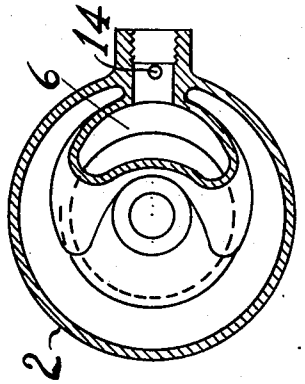


Fig. 1a

Frank A. Howard Inventor

P. L. Young Attorney

July 20, 1937. F. A. HOWARD 2,087,619

AUTOMOTIVE INTERNAL COMBUSTION ENGINE

Filed Aug. 4, 1933 6 Sheets-Sheet 2

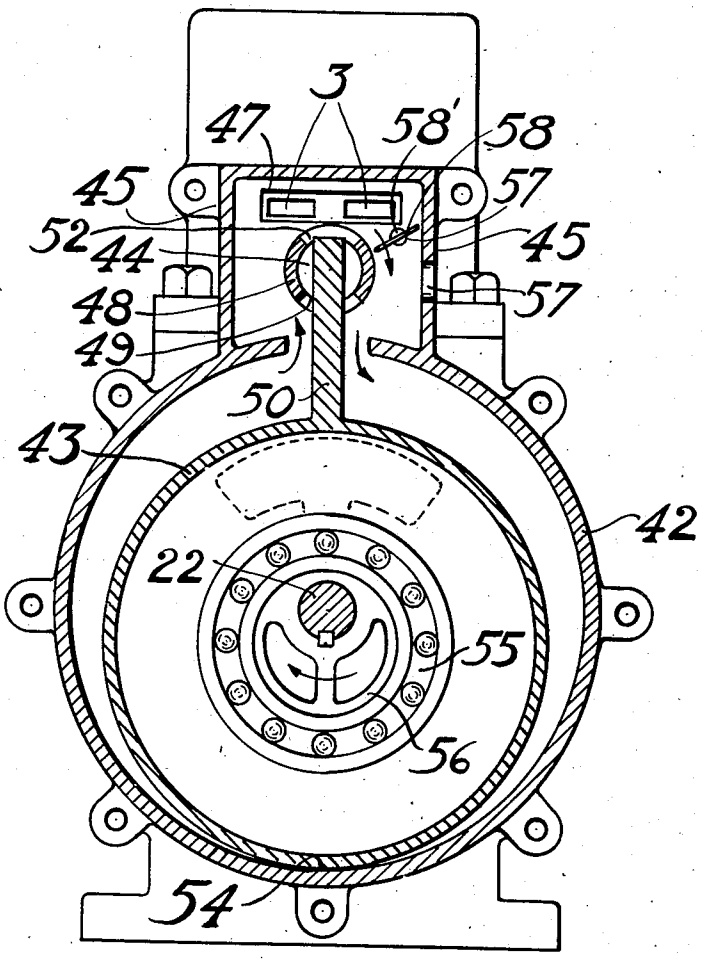


Fig. - 2

Frank A. Howard Inventor

P. L. Young Attorney

July 20, 1937. F. A. HOWARD 2,087,619

AUTOMOTIVE INTERNAL COMBUSTION ENGINE

Filed Aug. 4, 1933 6 Sheets-Sheet 3

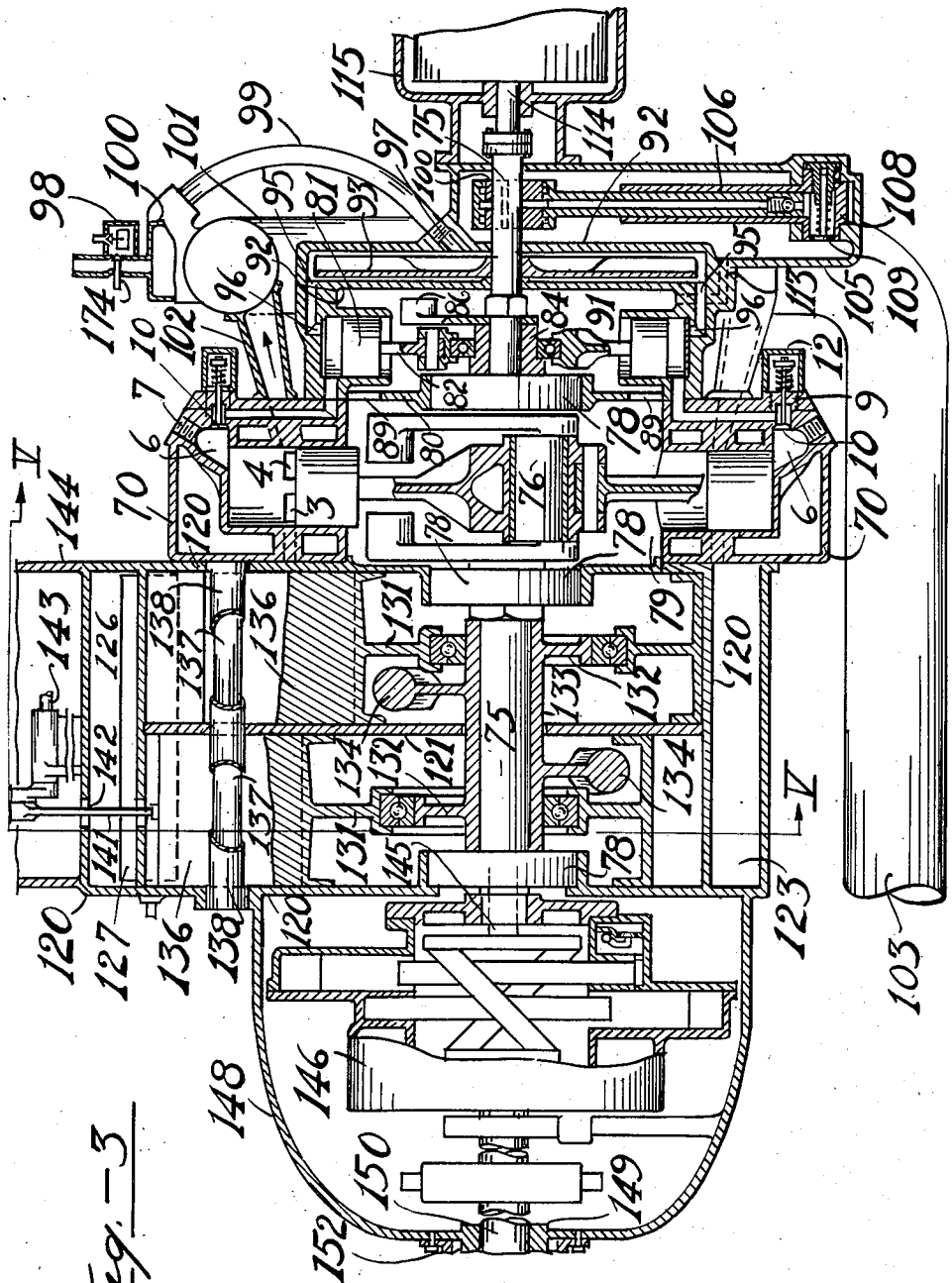


Fig. -3

Frank A. Howard Inventor

P. L. Young Attorney

July 20, 1937. F. A. HOWARD 2,087,619

AUTOMOTIVE INTERNAL COMBUSTION ENGINE

Filed Aug. 4, 1933 6 Sheets-Sheet 4

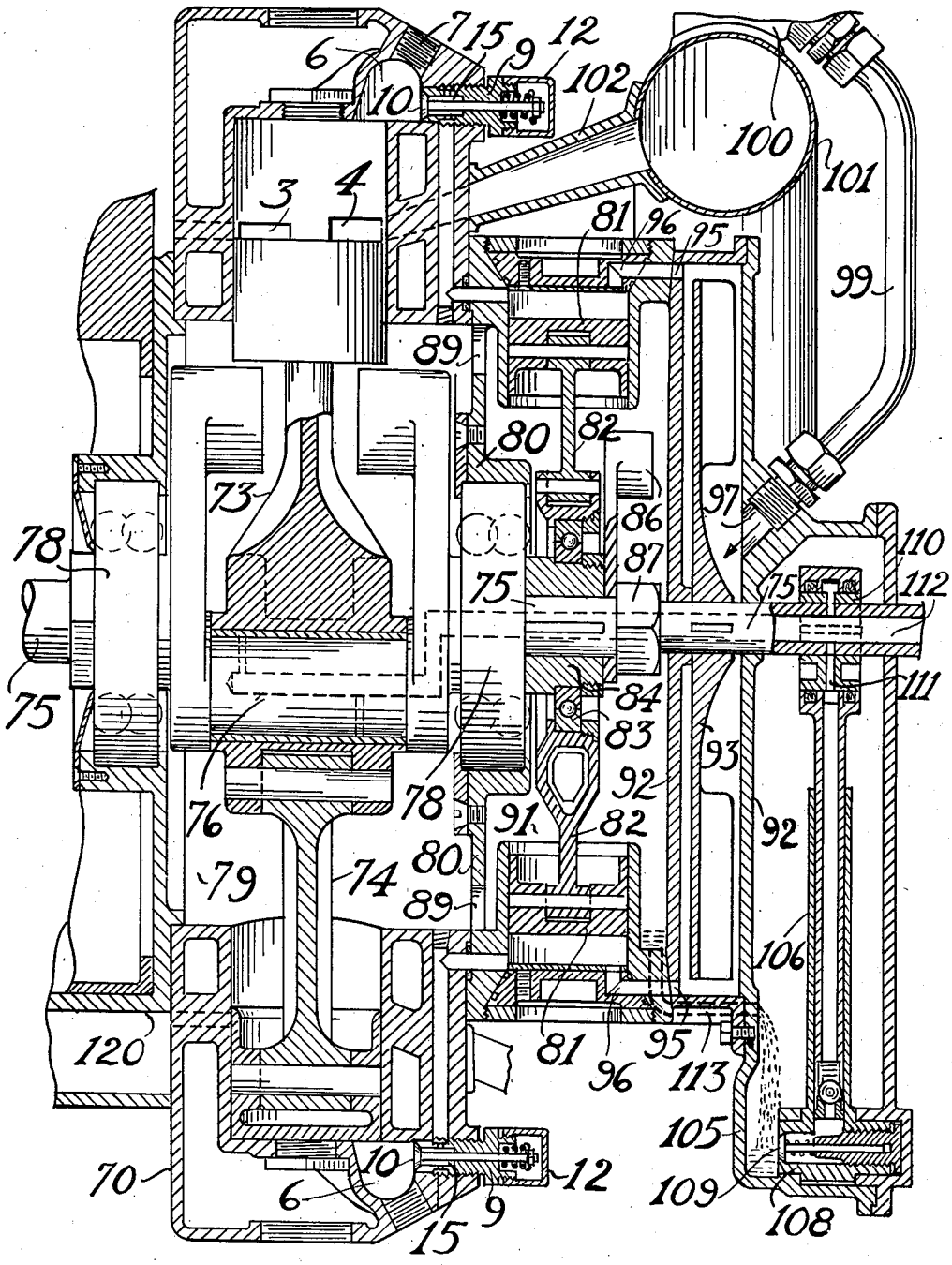


Fig. — 4

Frank A. Howard Inventor

P. L. Young Attorney

July 20, 1937. F. A. HOWARD 2,087,619

AUTOMOTIVE INTERNAL COMBUSTION ENGINE

Filed Aug. 4, 1933 6 Sheets-Sheet 5

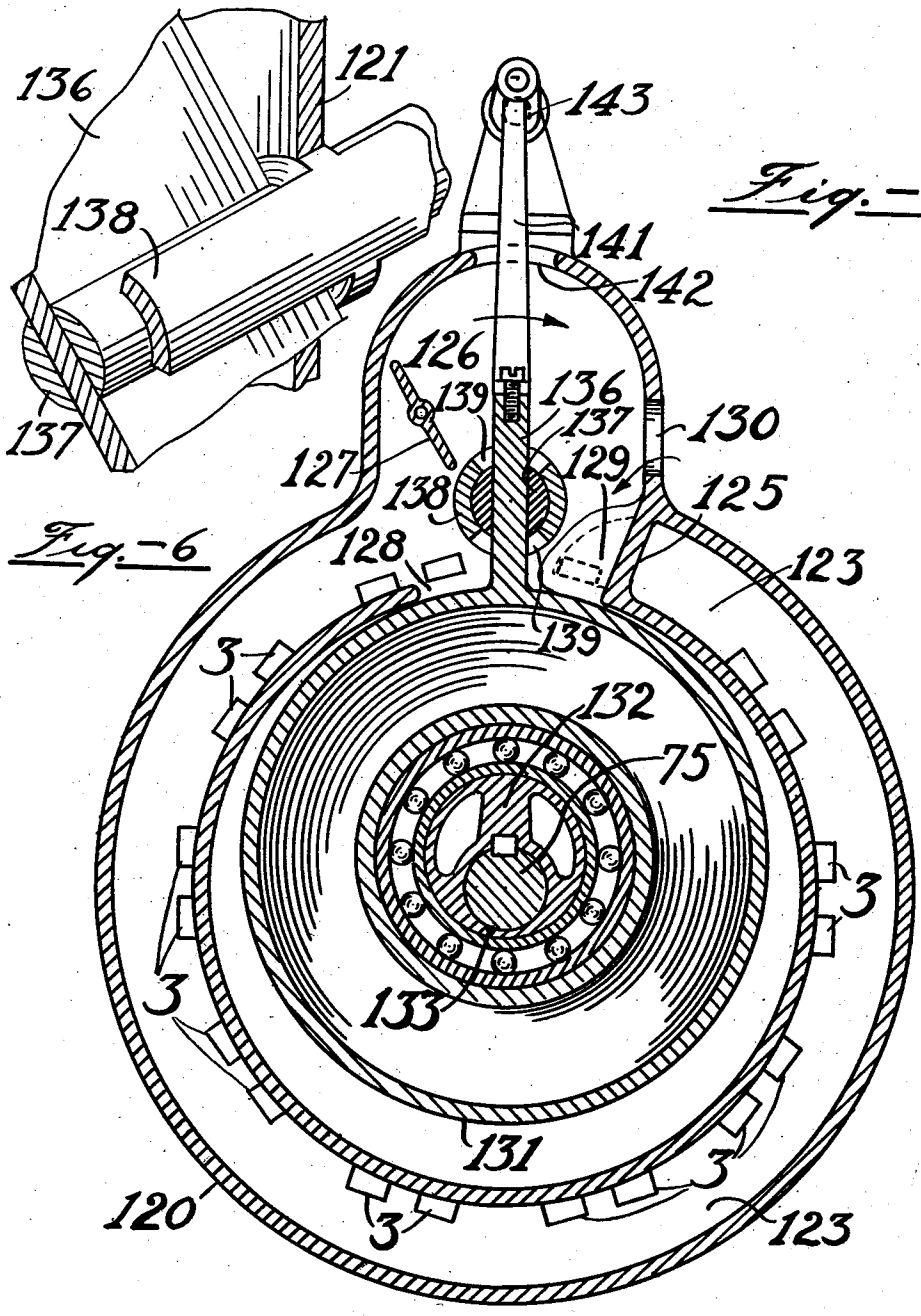


Fig.-5

Fig.-6

Frank A. Howard Inventor

P. L. Young Attorney

July 20, 1937. F. A. HOWARD 2,087,619

AUTOMOTIVE INTERNAL COMBUSTION ENGINE

Filed Aug. 4, 1933 6 Sheets-Sheet 6

Fig. 7a

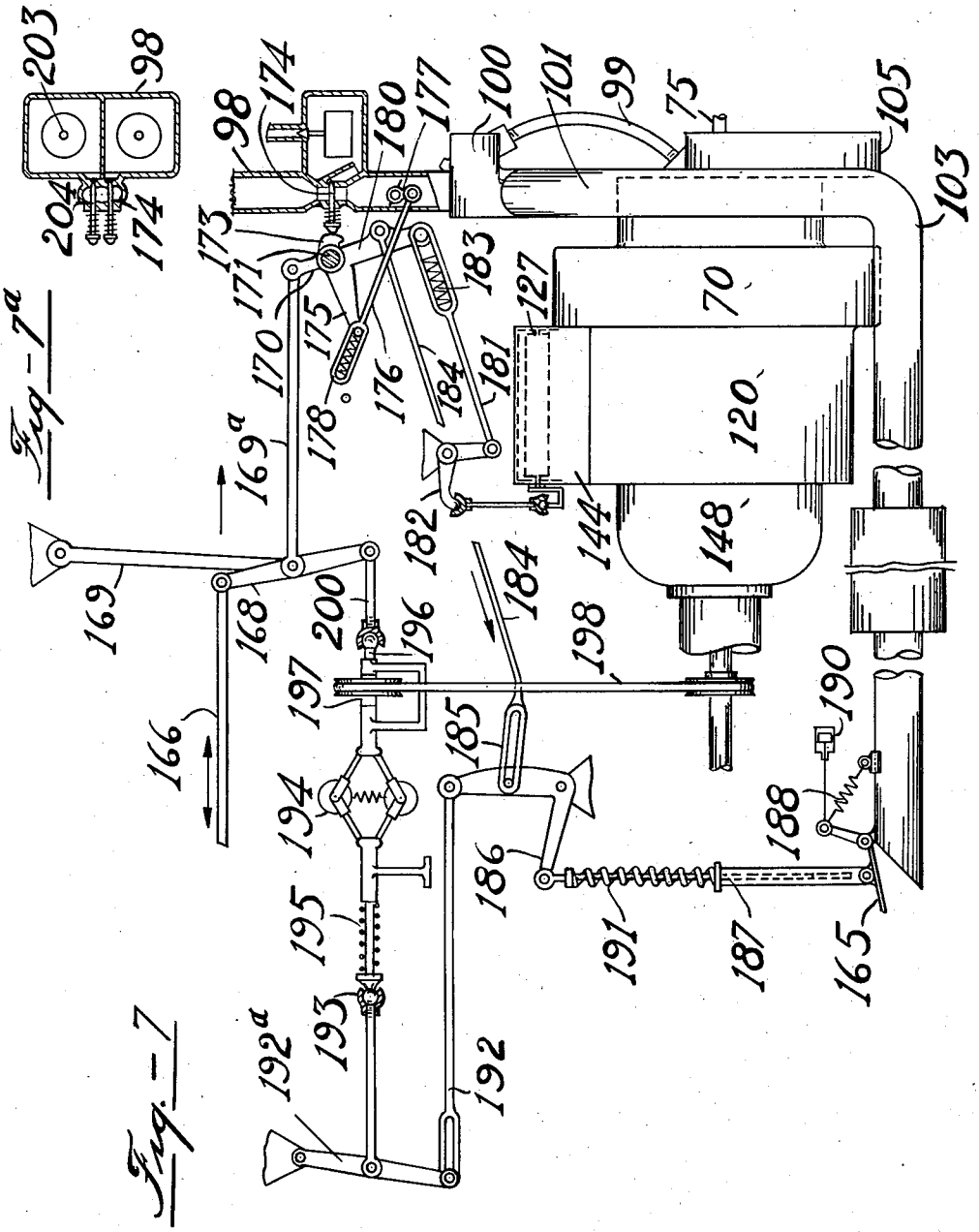


Fig. 7

Frank A. Howard Inventor

P. L. Young Attorney

Patented July 20, 1937 2,087,619

UNITED STATES PATENT OFFICE

2,087,619

AUTOMOTIVE INTERNAL COMBUSTION ENGINE

Frank A. Howard, Elizabeth, N. J.

Application August 4, 1933, Serial No. 683,593

37 Claims. (Cl. 123—65)

My invention relates to improvements in internal combustion engines especially suitable for motor vehicle propulsion. This application is a division in part and continuation in part of my previously filed application Serial No. 650,018, filed January 3, 1933, entitled "Gyral engine". The invention will be fully understood from the following description taken in connection with the accompanying drawings in which I have illustrated the engine itself and also the preferred arrangement of the engine in combination with a vehicle chassis.

In these drawings—

Fig. 1 is a longitudinal vertical sectional view through a single cylinder engine, and Fig. 1*a* is a cross section through the combustion chamber of the engine taken along the line Ia—Ia of Fig. 1;

Fig. 2 is a transverse sectional view taken along the line II—II of Fig. 1;

Fig. 3 is a longitudinal sectional view of a complete automotive internal combustion engine showing a multiple cylinder engine;

Fig. 4 is an enlarged longitudinal sectional view of the engine of Fig. 3 drawn on the same scale as Figs. 1 and 2 and taken through the radial working and charge compression cylinders of the automotive internal combustion engine of Fig. 3;

Fig. 5 is an enlarged sectional view taken along the line V—V of Fig. 3, showing the blower construction;

Fig. 6 is an enlarged fragmentary perspective view of the blower arm and its rock-shaft;

Fig. 7 is a diagrammatic layout of the control system for the engine;

Fig. 7*a* is a transverse sectional view through the fuel jet of the carbureter showing the arrangement for use of a supplementary water jet.

Referring particularly to the single cylinder construction shown in Figs. 1 and 2, reference numeral 1 designates the crankcase which is of the usual form. The cylinder 2 associated with the crankcase is of water cooled two-cycle design. The cylinder 2 is provided with a pair of inlet ports 3 on one side of the cylinder and with a pair of exhaust ports 4 on the opposite side of the cylinder. This construction provides for large port areas with four lands between the ports. The cylinder 2 is provided with a piston 5 which is rotatably connected to a crankpin 25 to be later described. A combustion chamber 6 is provided at the top of cylinder 2 and is of crescent form and partly offset as shown in Figs. 1 and 1*a*. The combustion chamber 6 receives a spark plug 7 and charge admission valve 8.

Charge admission valve

The charge admission valve 8 is made up of four main parts as follows: First, a body 9 of the size of a spark plug shell is screwed into a threaded hole in the cylinder 2, in the same manner as a spark plug; second, a poppet type valve 10 is seated on the inner end of the body 9; third, a loading spring 11 is associated with the valve 10 holding the valve closed against pressure within the body 9 up to a predetermined limit; fourth, a cap 12 is threaded over the exposed end of the body 9 and encloses the valve spring 11 to exclude dirt and prevent leakage along the stem of the valve 10. A compressed charge which is admitted to the cylinder by the valve 10 reaches the cavity in the valve body 9 through a drilled passage 14 in the cylinder casting 2. The passage 14 extends the length of the cylinder 2 and passes through the wall dividing the two exhaust ports 4. The valve body 9 is grooved externally at 15 opposite the end of the passage 14. A plurality of holes 16 through the body 9 deliver the charge from the groove 15 into the cavity of the body 9 back of the head of the valve 10. By virtue of its small size the valve 10 is capable of operation at very high speeds. The valve 10 is constructed with a seating area large in proportion to its free area as shown in Fig. 1 and therefore remains open under pressure less than the pressure required to open the valve initially. For example, the valve 10 may be loaded to open under a pressure difference of two atmospheres and with a seat area greater than its free area the valve will remain open until the pressure difference drops below one atmosphere.

Charge compressor

On the open side of the crankcase 1 there is mounted a charge compression cylinder 18 of a displacement very much smaller than that of the working cylinder 2. The compressor 18 has a piston 19 which is driven through a connecting rod 20 by means of an eccentric 21 mounted on a crankshaft 22. The eccentric 21 is set ahead of the crankpin 25, for example 45° more or less, the exact setting being adjustable by loosening cap screws 26 which latter pass through arcuate slots in a driving flange 27 keyed to the crankshaft 22. The compressor 18 is provided with a delivery port 28 which opens directly into the passage 14 in the cylinder casting 2. By means of this arrangement the volume of the delivery passages back of the valve 8 may be kept to a small fraction of the displacement of

the compressor, and the charge admission valve 8 becomes the outlet valve of the compressor, still permitting the compressor 18 to operate at fair volumetric efficiency. The compressor 18 has a head 29 which is seated in the compressor cylinder on a conical seat 30 having spaced packing grooves 31. A hard packing or ring gasket is used in the inner groove and a soft packing in the outer groove so that upon forcing the head 29 to its seat by a ring nut 32 there will be formed a pressure-tight seal on the inner packing. The outer packing is sufficiently tight to prevent infiltration of air during the suction stroke. This latter requirement arises from the use of the head as a suction connection by drilling the head horizontally at 34 to communicate with an inlet port 35. An inlet valve is provided for closing the port 35 and consists of a flat spring or reed 36 which is disposed in a recess in the inner face of the head 29 and is secured at its extremity by a screw as illustrated in Fig. 1. The design is such as to reduce to a minimum the clearance volume within the compressor cylinder 18. It is possible for the piston 19 to practically contact the flat inner face of the head 29 at its upper position. The head 29 may be notched to register with the outlet port 28 and slightly increase the delivery port area at the top of the stroke of the piston 19. These details of construction are described as indicative of the necessity of avoiding unnecessary clearance volume in the compressor, which, although always good practice, becomes of especial importance and difficulty in the present design in which the outlet valve 8 of the compressor is located at some distance from the compressor head 29 as required for its double functioning as a charge admission valve and a compressor outlet valve.

The inlet passage 34 of the compressor head 29 registers with a suction passage 38 connected to a carbureter 39. The carbureter 39 is of usual design constructed to maintain a constant ratio of fuel to air with variable delivery, but in addition is provided with means for opening the fuel jet to increase very greatly this ratio. The details of suitable arrangements for this purpose will be later described.

Air blower

On the side of the crankcase 1 opposite the compressor 18 there is located a blower of the type described in my previously filed application Serial No. 650,018, filed January 3, 1933, entitled "Gyral engine", which supplies air for scavenging, internal cooling and for combustion. The blower comprises a housing 42 and two moving parts in addition to the crankshaft 22, namely a gyror 43 and a rockshaft 44. The housing 42 comprises a cylindrical casing having a connected extension 45 at its upper side. The extension 45 is provided with a port 47 in its face adjacent the cylinder 2 which registers with the inlet ports 3 of the cylinder. A fixed sleeve 48 extends through the housing 42 from one face of the housing to the other and serves as a bearing for the rockshaft 44. The rockshaft 44 is slotted in its center at 49 to form a slide bearing and seal for an arm 50 of the gyror 43, and the sleeve 48 is likewise slotted with a wider slot 52 of the same length to permit of the passage and oscillation of the gyror arm 50. The gyror arm 50 is carried by a drum shaped gyror 43 which is the displacing member of the blower. The side faces 52 and 53 of the gyror 43 have a sliding fit against the inner faces of the casing 42 while the cylindrical surface 54 of the gyror 43 has a minimum working clearance with the inner cylindrical surface of the casing 42. The gyror is driven through a bearing 55 from an eccentric 56 keyed to the crankshaft 22 in the same angular position as the crankpin 25. The admission port of the blower is constituted by a hole 57 in the side of the housing extension 45. A tight fitting balanced valve of the butterfly type designated 58 is carried on a shaft parallel with the rockshaft 44 and is adapted to control by-passing of air over the top of the arm 50 and thence back to the suction side of the blower.

The engine construction shown in Figs. 1 and 2 is completed by an exhaust pipe 60 which registers with the exhaust ports 4 and carries at its outer end a loaded valve 61 by means of which a regulated back pressure may be put upon the exhaust for purposes of supercharging. The valve 61 is preferably dampened to prevent chattering by means of a dash pot 62. Details of means for regulating the loading of such a valve will be later described.

Engine operation

The engine as described operates as follows: Upon rotation of the crankshaft 22 a charge of combustible mixture is drawn into the compressor 18 and its compression is begun by return movement of the compressor piston 19. At or about the time the working piston 5 has overrun the exhaust ports 4 on its compression stroke, the compressor piston 19, which is for example now at 105° past bottom dead center, will have built up a pressure sufficient to overcome the loading of the valve 9 and the compressed charge will be delivered into the combustion chamber 6 progressively as the rotation continues. This delivery will end only when the compressor piston 19 has reached top dead center. The compression of the charge in the compressor aids in its vaporization, and the delivery of this charge under substantial pressure difference through the valve 8 atomizes unvaporized fuel. The expansion through the valve 8 cools this valve, the combustion chamber 6 and more especially the spark plug 7 which may be located as shown in the path of the expanding charge and close to the valve 8. The compressor piston 19 may be timed to reach its outer dead center before or shortly after ignition occurs.

Following ignition, the charge burns and expands in the usual way to deliver power through the working piston 5. When the working piston 5 approaches bottom dead center it uncovers the exhaust ports 4 and the residual pressure is relieved through the ports 4 at or before the time the inlet ports 3 are uncovered. Assuming that the blower by-pass valve is not too far open, the opening of the inlet ports will permit the compressed air in the blower housing to deliver through the inlet ports 3. In part this air will be deflected upwardly to scavenge out the exhaust gas and in part it will pass directly across the piston 5 cooling the latter and emerging with the scavenged exhaust gases. In the engine illustrated the displacement of the blower is three times that of the working cylinder 2. There is therefore a large excess of air available and while it has been considered that scavenging air which by-passed directly through the cylinder without pushing out its quota of exhaust gas was wasted, the present design definitely contemplates a substantial percentage of such by-passing for the express purpose of cooling the piston head 5 over

which the air flow in question occurs with the present simplified design.

The design further contemplates the retention of a substantial proportion of unscavenged exhaust gas in the combustion chamber 6, for the purpose of aiding vaporization, improving ignition, and smoothing pressure rise during combustion. This retention is affected by the partly pocketed form of the offset combustion chamber 6. The loss of power from this retention of exhaust gas is capable of being offset by keeping a small back pressure on the exhaust gases by the valve 61.

By further increasing this back pressure it is possible to obtain a considerable super-charging effect. For example if the valve 61 be loaded to maintain a pressure of one atmosphere (gauge) on the exhaust, then the blower will deliver at that pressure one and one-half displacement volumes of air, sufficient to give substantially complete scavenging with an excess available for cooling purposes by direct flow across the top of the piston, leaving the cylinder substantially filled with air at a pressure of two atmospheres absolute at the time of closing of the exhaust ports 4. If this be assumed to be the maximum blower super-charging effect desired, there is then the additional super-charging effect of the charge compressor 18. I prefer to proportion the compressor 18 for a delivery (allowing for volumetric loss from suction and clearance volume) between 10% and 25% of the engine displacement. This amount of additional super-charging will therefore take place during the compression stroke and constitutes the second stage of super-charging. The final effect may therefore be super-charging up to an equivalent volumetric efficiency of the working cylinder of over 200% corresponding to a brake mean effective pressure of over 200 pounds per square inch as compared with about 50 pounds per square inch for a normal two cycle engine with crankcase compression. Such high brake mean effective pressures put low limits on the permissible compression ratio unless there be used fuels of most exceptional anti-detonation value or unless the cooling be exceptional. In the engine of this invention the blower supercharging is arranged to effect intensive cooling of the piston head 5 which without this cooling is the principal contributing cause for detonation, and the use of a pre-compressor 18 for the charge with expansion injection through the valve 9 directly into the combustion chamber 6 and in proximity to the spark plug 7 avoids true pre-ignition. To the extent that exhaust gas is retained in the combustion chamber 6 it is a further factor in preventing detonation, the retained heat thereof being of less importance than its heat adsorptive capacity during combustion of the fuel.

If the net delivery of the charge compressor 18 be as small as 10% of the displacement of the working cylinder 5, it is possible to operate it at constant volumetric efficiency, that is without a throttle valve since this is the approximate minimum charge volume required for idling conditions. During idling the blower by-pass valve 58 will therefore remain wide open, the entire air supply as well as the fuel being delivered by the compressor 18. This results in exceptionally smooth and reliable low speed idling. The exhaust back pressure valve 61 will, of course, also be open and unloaded under such conditions.

To increase the power output of the engine, the by-pass valve 58 is partly closed. This results in building up a pressure in the blower housing with consequent flow of air through the ports 3 into the working cylinder. The carbureter 39 must now supply fuel in proportion not alone to the air flowing through it, but in proportion also to that delivered by the blower. This result is obtained by increasing the fuel-to-air ratio of the carbureter proportionately to the closing of the by-pass valve 58. Changes in total air supply per minute resulting from changes in engine speed without change of setting of the by-pass valve 58 will be taken care of by the metering characteristics of the carbureter as in normal practice. Preferably the by-pass valve 58 will be fully closed before the exhaust back pressure valve 61 is loaded. The engine will therefore operate up to an equivalent volumetric efficiency in excess of 100% (neglecting the capacity lost through late closing of the exhaust) without exhaust back pressure, and under these conditions will deliver up to two displacement volumes of cooling air through the cylinder and especially over the piston head 5 at each cycle. Where the ratio of surface to volume of the combustion chamber is high, as for example upwards of 1.5 to 1, corresponding to the small cylinders especially useful in automotive work this internal cooling is efficient in the sense that the air used picks up heat in important amounts from the combustion chamber walls. As the surface to volume ratio falls the efficiency of utilization of the excess air for cooling of the walls also drops, and in large engines of the slow speed type is practically negligible. The operation described is therefore especially the desirable condition for high speed engines in which the heat liberation in unit time and consequent required rate of heat flow through the walls of the working chamber is high. Without adequate internal cooling the rate of heat flow in a two cycle engine operating at high speeds and high brake mean effective pressure becomes excessive with consequent uncontrollable pre-ignition or detonation and failure of cylinder lubrication.

On the other hand relative average rate of heat liberation and heat flow through the walls at low speeds even at high brake mean effective pressure is low, and it is possible to reduce the required internal cooling. The excess blower capacity employed for internal cooling of small engines at high speeds is therefore available and may be used for super-charging at low speeds to level off the speed-power curve of the engine to cause it to approach the variable cut-off reciprocating steam engine or the series wound electric motor.

These unusual performance characteristics render the engine of the present invention especially suitable for vehicle propulsion. For such service, however, there is required in addition the maximum attainable degree of continuity of torque, which comes only with multiplicity of working chambers or cylinders even where the two stroke instead of the four stroke cycle is employed. There is further required compactness of design, light weight, absence of vibration and easy accessibility of parts requiring frequent service attention. In adapting the engine of the present invention to this service in the manner here shown and described it has been possible to meet these conditions.

Radial engine

Referring to the construction shown in Figs. 3 to 7 inclusive, the ten radial cylinders which

may be for example 2½" bore by 2" stroke are preferably cast integrally within a smooth annular water jacket 70. The common annular water jacket 70 surrounding all of the cylinders is so large that thermo-siphon or evaporation cooling is adequate for the engine, thus dispensing with a water circulating pump. Each of the ten radial cylinders is of the construction previously described in connection with Figs. 1, 1a, and 2. I prefer to use an articulated connecting rod for the ten pistons, the master rod being designated 73 and one of the nine link rods being designated 74. The crankshaft 75 is divided, the crankpin being formed integrally with the loaded end of the crankshaft and secured in the crankthrow of the unloaded end of the crankshaft. The big-end bearing may thus be a one-piece bushed bearing. The main bearings 78 of the crankshaft are shown as ball bearings. The engine crankcase is designated 79.

Radial compressor

The compressor, like the engine itself, is preferably formed as a single casting, designated 80, and carrying ten radial compressor cylinders preferably aligned with the respective engine cylinders which they serve. The compressor casting 80 fastens directly against the face of the engine casting 70 and forms the support for the adjoining main bearing 78 as in the construction illustrated in Fig. 1. The compressor pistons 81 are driven by articulated connecting rods 82, the big-end bearing of the master rod being a ball bearing 83. The stroke of the compressor pistons is so short that they may be conveniently operated by an eccentric 84 which is keyed to the crankshaft 75. Because of the compact arrangement of the parts as shown in Fig. 4, it is practical to consider the eccentric 84 as an over-hung load on the adjoining main bearing 78. A compressor counterweight 86 may be locked against the face of the eccentric 84 by a retaining nut 87 which secures the eccentric 84 in position. The compressor eccentric 84 is set ahead of the engine crankpin 76 by 45° more or less as previously described. In addition to the counterweight 86 the adjacent crankthrow of the crankshaft 75 is formed to complete the balancing of the compressor thus avoiding the interposition of a counterweight between the compressor eccentric 84 and the main bearing 78. By virtue of perforations 89 in the compressor casting 80, the crankcase 91 of the compressor opens freely into the engine crankcase 79. The engine crankcase 79 is flared outwardly at a small angle to form a cone frustrum with its larger end opening to the compressor. Both centrifugal force and gravity thus operate to divert oil from the engine crankcase 79 into the compressor, where it may collect in the dead space around the lowermost cylinder and be drained off to a sump tank to be later described.

Fuel distributor

The outer face of the compressor crankcase 91 is closed by a charge distributor housing 92 which encases a distributor impeller 93 keyed to the crankshaft 75. The housing 92 is drilled axially at its inner periphery to form openings 95 which register with compressor inlet ports 96. The central portion of the cover plate of the distributor housing 92 has a port 97 for the admission of carburetted air which is delivered from a carbureter 98 through an intake pipe 99 which extends downwardly from a heater chamber 100 carried by an exhaust collector ring 101. The exhaust collector ring 101 communicates with the exhaust ports 4 through exhaust passages 102. An exhaust pipe 103 leads from the exhaust collector ring 101.

Lubricating oil feed

The cover plate of the distributor housing 92 is formed as a unit with a pump housing and oil sump designated 105. Within the housing 105 there is mounted an oscillating cylinder lubricating oil pump 106. The bearing for the oscillating cylinder is a hollow trunnion 108 which carries the pump foot valve 109. The pump 106 is driven by an eccentric 110 mounted on the crankshaft 75 and delivers the oil pumped through holes 111 in the bearing surface of the eccentric 110 to an oil duct 112 in the crankshaft 75 from whence the oil is fed in the usual way to the crankpin bearing as shown and to any other crankshaft bearing if desired. Oil collecting around the lowermost compressor cylinder drains by gravity through drainage ducts 113 to the sump 105 in which the oil suction valve 109 is located.

Preferably the front end of the crankshaft 75 is coupled directly to the shaft 114 of a generator 115, as shown in Fig. 3. The generator 115 is secured to the cover plate of the oil pump housing 105. The generator shaft 114 and its bearings therefore serve to support the free end of the crankshaft 75. The front end of the generator shaft 114 carries a cooling fan 116, (see Figs. 8 and 9), which is thus located opposite the central portion of a cooling radiator 117. The cooling radiator 117 is preferably inclined as shown in Figs. 8 and 9.

Twin blower

A blower casing 120 is attached to the back face of the engine casting 70 as clearly seen in Fig. 3, and forms a closure for the rear end of the crankcase 79 and support for the rear main bearing 78. The blower casing 120 is divided by a central web 121 to form two blower chambers both of which open at the top into an annular passage 123 which forms a manifold delivering air to the ring of inlet ports 3 of the engine. The passage 123 is blanked off by a web 125 on the intake side of the blower casing 120 and opens to a by-pass 126 on the delivery side of the casing 120. By-pass control valve 127 controls flow through the by-pass 126 from blower outlet ports 128 to inlet ports 129. An air inlet opening 130 is provided in the side of the by-pass 126, as can be seen in Fig. 5. Gyrors 131 of the blowers are driven by eccentrics 132 set at 180° on a sleeve 133 keyed to the crankshaft 75 and are counter-weighted as indicated at 134. Each gyror 131 is provided with an arm 136 which passes through a slot in a rock-shaft 137, as clearly shown in Figs. 5 and 6. The two rock-shafts 137 are assembled in end to end relation in a common sleeve 138 which has wide slots 139 to permit of the oscillation of the arms 136. The sleeve 138 is supported at its ends by the casing 120 and in the center by the partition web 121. One of the two blower arms 136 carries a driving rod 141 affixed to its end and projecting through a slot 142 in the housing 120. At its outer end the rod 141 has a bearing on a crank which is affixed to an accessory shaft 143 mounted atop the blower housing 120 in a casing 144. The accessory drive-shaft 143 is thus coupled directly and positively to the engine

crankshaft 75 and may therefore be employed to drive timed accessories such as an electrical ignition apparatus. The parts being of the proportion shown in the drawings, the end of the driving rod 141 describes a path so nearly a true circle that the variations may be ignored for practical purposes, the elasticity of the parts and the ordinary bearing clearances being sufficient to take up the microscopic deviations. The angular velocity ratio of crankshaft 75 and the accessory shaft 143 also deviates slightly from unity but this deviation is small, constant and predeterminable for each angular position and may therefore be compensated for in the design of the timed accessory where necessary. As will be apparent from Figs. 8 and 9, the location of the accessory drive-shaft is especially convenient, giving maximum accessibility for service adjustments of accessories driven thereby.

Control system

Referring to Fig. 7 in which the control system is shown in an enlarged diagram, the manual control rod by which the power output is regulated is designated 166. The control rod 166 is movable axially to the right, as shown in Fig. 7, to increase power output, and to the left to decrease the power output of the engine. At the front end the control rod 166 is pinned to one end of a floating link 168 supported by a hanging link 169 from a fixed point. The center of the link 168 is coupled by a link 169a to a crank-arm 170 on a cam shaft 171. A cam 173 on the shaft 171 moves a fuel valve 174 of the carbureter 98. A second arm 175 on the cam shaft 171 is coupled by a link 176 with the carbureter throttle valve 177. The link 176 has a slotted end 178 within which the pin of the arm 175 may slide against the resistance of a spring extending the length of the slot.

A third arm 180 on the cam shaft 171 has coupled thereto at its end a link 181 which through a bell crank 182 as shown operates the blower by-pass valve 127 previously described. The link 181 likewise has a slotted head 183 with an axial spring, which spring however extends only through half the length of the slot, as shown. The arm 180 carries an additional link 184, the slotted outer head 185 of which is pinned to a bell crank 186, the other arm of which is pinned to a telescopic loading rod 187 fixed to the exhaust back pressure valve 165 which has previously been described. The exhaust back pressure valve 165 has a spring 188 fastened to it and of sufficient strength to hold the valve 165 normally open and take up lost motion in the linkage connected with the valve. The valve 165 is coupled also to a dash pot 190 which prevents chattering of the valve. The two ends of the telescopic loading rod 187 are held in an extended position by a valve loading spring 191. The upper end of the bell crank 186 is coupled through a link 192 having a slotted head and through a lever 192a to the end of a governor shaft 193. A ball governor 194 is affixed to the governor shaft 193 so that upon expanding by centrifugal force it is capable of drawing in the shaft 193 axially against the resistance of a spring 195. Such resistance, however, causes the governor 194 to first retract, up to limit of its movement, a shaft 196 on its other end. This shaft 196 is splined in a driving-sleeve 197 of the governor 194. The driving sleeve 197 is indicated diagrammatically as being driven by a belt 198 from the driving member 150. The shaft 196 is coupled through a link 200 to the floating link 168.

Operation of the control linkage

The sequence of operation of the control linkage just described is as follows: Initial opening movement of the control rod 166 rocks the cam shaft 171 to open the throttle 177 progressively up to the full open position without effecting any other motions, except that as the speed of the engine increases the governor retracts the shaft 196 and thus offsets the opening movement of the rod 166 by moving the lower end of the floating link 168 to the left, as viewed in Fig. 7. This control action of the governor 194 is optional, and is intended only to prevent the engine from tending to race at low throttle openings when unloaded. The throttle 177 is also optional. If the compressor displacement be limited to the charge volume required for idling the engine, the throttle 177 may be omitted, permitting the compressor to operate at full volumetric efficiency at all times. If it be desired to employ the compressor for super-charging beyond this low limit (approximately 10%) the compressor must be of larger displacement and in such case requires the throttle 177 for control of the engine at low loads, as described. In the event the throttle 177 is omitted there will be no lost motion provided between the arm 180 and the link 181. Initial movement of the control rod 166 will therefore begin to close the by-pass valve 127, thus creating a pressure in the blower delivery duct 123 which will result in the delivery of air through the inlet ports 3 of the engine when the latter are exposed by the pistons. The air entering the ports 3 will displace exhaust gas through the opposite ports 4. Each cylinder will therefore contain a mixture of exhaust gas and air when the piston over-runs the exhaust port 4 on its outward stroke. The mixture supplied by the carbureter 98 must now contain sufficient fuel to burn not only the air passing through the carbureter 98, thence through the compressor to the engine, but in addition fuel for the air supplied by the blower. This result is obtained by enriching the mixture delivered by the carbureter 98 to the compressor proportionately to the delivery of blower air to the engine. I employ the cam 173 for this purpose, forming the surface of the cam to cooperate with the head of the fuel valve 174 to open the valve as the shaft 171 is turned.

Progressive opening movement of the rod 166 therefore either first opens the throttle 177 to full open position without affecting any other controls, and subsequently progressively closes the by-pass valve 127, simultaneously opening the fuel valve 174, or if the throttle 177 be omitted, causes the immediate progressive closing of the by-pass valve 127 with simultaneous opening of the fuel valve 174. In either case the lost motion connection formed by the slotted head of the link 184 prevents the exhaust back pressure valve 165 from being affected until the by-pass valve 127 has been fully closed. After the closing of the by-pass valve 127 further movement of the control rod 166 causes the bell crank 186 to rock in counter clockwise direction to compress the spring 191 and thus force the valve 165 to its seat against the resistance of the spring 188 which has been holding the valve 165 in the full open position. The slotted head of the link 192 permits this motion without interference from the governor if the lathe is in a low speed position. There is thus created a pres-

sure within the exhaust pipe 103, and as a result the final pressure in the cylinder itself rises so that the mass of gas retained in the cylinder on the closing of the exhaust port 4 is increased. As heretofore stated, the displacement of the blower is in excess of that of the engine, for example 150 to 300% of the engine displacement. So long as the exhaust system is free to discharge directly to the atmosphere, it results that on complete closing of the by-pass valve 127 the blower delivers a quantity of air in excess of that required to displace the combustion gases, the excess blowing directly out, mainly across the top of the pistons, for cooling the latter. Until the exhaust back pressure valve 165 is loaded therefore the excess blower capacity is used for internal cooling of the engine. When the valve 165 is loaded by the spring 191, however, the volume (but not the mass) of the blower delivery decreases proportionately, and the engine cylinder is charged with air at superatmospheric pressure, that is super-charged. If for example the net delivery of the blower per revolution, measured at atmospheric pressure, is two displacement volumes of the engine, the valve 165 may be loaded by the spring 191 to hold a pressure of one atmosphere gauge on the exhaust system. The volume of the blower delivery under these conditions is approximately equal to the displacement volume of the engine, so that the combustion gases are displaced against a pressure of one atmosphere (gauge) by an equal volume of air at the same pressure, leaving the cylinder filled with twice the normal mass of air at the beginning of the compression stroke. This increased mass of air requires additional fuel for combustion, which is provided as before by further enrichment of the mixture in the carbureter 98 resulting from the progressive opening of the fuel valve by the same control motion which loads the exhaust control valve 165. Under conditions of maximum supercharging the carbureter 98 may deliver a mixture so rich that relatively little vaporization occurs in the induction or intake pipe 99. For that reason it is especially desirable to employ a down draft carbureter with heater and induction pipe arranged for continuous downward flow, as shown, and to deliver from the induction pipe 99 against a rotary charge distributor 93 through which the unvaporized fuel is delivered in equal proportions to each of the compressor cylinders.

Assuming the control rod 166 to be in the fully advanced position, operation under supercharging conditions continues until the engine attains a speed sufficient to overcome the tension of the governor springs, for example 1,000 to 1,500 R. P. M. Beyond this speed the governor shaft 193 begins to be retracted, the lost motion is taken up in the end of link 192 and through the linkage shown the governor forcibly moves the entire control linkage backward, thus progressively unloading the spring 191 and relieving the back pressure on the exhaust system, until at a speed of, for example, 2,500 R. P. M. the governor reaches the limit of its action. At this point the valve 165 is completely unloaded and is returned to its open position by the spring 188. Further increase of engine speed has no effect on the control system, and the engine continues to operate at full volumetric efficiency, but without blower super-charging and without exhaust back pressure, up to its maximum speed of, for example, 5,000 R. P. M. During such operation the excess blower capacity is usefully employed in internal cooling of the engine, serving to prevent excessive temperature rise of the combustion chamber surfaces, especially the piston head. Super-charging to the extent of the net delivery of the compressor continues unchanged throughout the speed range of the engine, the increase of power from this supercharging being sufficient to offset the weight of the compressor. To a considerable extent also the compressor super-charging exerts a cooling action offsetting the tendency toward detonation. The primary cooling effect is from the loss of heat during compression, with resultant temperature of the charge as injected through the valve 8 below that of the compressed gas in the engine. A secondary cooling effect is obtained by the absorption of heat by vaporization of the fuel after the closing of the compressor intake valve, that is during compression and injection.

Supplementary internal water cooling

If desired, additional cooling may also be effected most conveniently by the compressor by installing an additional nozzle 203 and valve 204 in the carbureter 98 as indicated in Fig. 7*a*, and supplying the nozzle 203 from a separate float chamber, connected to a water supply. This water valve 204 operates in precisely the same way as the fuel valve save that it remains closed until the control linkage reaches an open position requiring supplementary cooling by water after which it opens progressively. When using compression ratios or super-charging pressures requiring supplementary internal water cooling, it is desirable to insulate the heater chamber 100 from the exhaust pipe 101 so that there will be no substantial vaporization of the water in the induction system and therefore no substantial effect on the air volume inspired by the compressor cylinders or loss of heat absorption power of the water. Vaporization of the water during compression in the compressor reduces the temperature of the charge prior to injection, while vaporization occurring as or after the charge is injected into the combustion chamber 6 through the valve 8 directly cools the combustion chamber contents. In both cases the heat absorption attendant upon the vaporization of the water does not reduce volumetric efficiency. Preferably the compression ratio of the engine as used for general motor vehicle service is such that supplementary water cooling is either unnecessary or is limited to the condition of blower super-charging. The occasions for such blower super-charging in vehicle service are so rare and short of duration that the water requirement may be taken from the engine water cooling system without material enlargement of its reservoir capacity.

It will be understood from the foregoing specification that there is provided by the present invention an engine design of unique performance capabilities which fit it especially for service in vehicle propulsion, and that its mechanical embodiment is such that when combined as shown with a conventional vehicle chassis it produces an automotive vehicle of materially simplified construction. I am aware that various elements of this complete construction are old in similar general combinations, but I believe the combination of the present invention to be inventive, new and useful as a combination, and I have therefore claimed the same in the appended claims. I have also claimed various elements and sub-

combinations which are novel and useful independently or in combinations other than those shown. It is my intention to claim all novelty inherent in the subject matter herein shown and described as broadly as is permissible in view of the prior art.

I claim:

1. A positive displacement two cycle internal combustion engine having an air blower driven by the engine and of a capacity substantially in excess of the engine displacement, means for supplying the air delivered by the blower to the intake ports of the engine during the period in which the exhaust ports are likewise open, whereby the excess air passes through the engine for internal cooling of the combustion chamber walls, means for regulating the air supply to reduce it below the engine displacement for reduction of engine torque below full normal torque, means for applying a regulated back pressure to the exhaust ports of the engine to retain an air volume in excess of the engine displacement volume, and independent means for supplying to the engine carbureted air under pressure after compression has been initiated.

2. An engine according to claim 1 in which means is provided for progressively limiting the back pressure which may be applied to the exhaust as the speed of the engine increases.

3. In combination with a positive displacement internal combustion engine, an air compressor of relatively small displacement as compared with the engine, means for synchronously driving the compressor by the engine with at least the latter portion of the compression phase of the compressor coinciding with a part of the compression phase of the engine, means for supplying water to the compressor during its suction phase, and means for delivering compressed air carrying the water so supplied to the engine for reducing the temperature attained on combustion.

4. A combination according to claim 3, means for supplying fuel as well as water to the compressor during its suction phase.

5. In combination with an internal combustion engine, a blower for supplying air thereto comprising a casing of cylindrical form, a drum shaped rotor of less diameter than the casing and of the same axial length rotatively and eccentrically mounted in the casing substantially in contact on the projected radius of the eccentric throw, means for rotating the eccentric mounting means of the rotor on the axis of the casing, a radial arm extending from the rotor, a rocking member journaled in the casing parallel with the axis thereof and having a sliding seal with the arm, means for preventing air from passing from one side of the arm and rocking member to the other side, means for admitting air to the casing in proximity to the side of the arm which lies in the direction of rotation, and means for delivering air to the engine from the casing in proximity to the other side of the arm.

6. A combination according to claim 5 in which the blower is synchronously driven by the engine and the arm is extended through the rocking member and rotatably connected to a crankpin, and an accessory driveshaft is mounted parallel with the casing axis and carries the crank for said crankpin, whereby the blower serves to drive the accessory shaft synchronously with the engine.

7. In combination with an internal combustion engine, an eccentric synchronously driven by the crankshaft thereof, a driving member rotatably mounted on the eccentric and having a sliding fit with a spaced rocking member mounted parallel with the eccentric shaft, an accessory shaft journaled for rotation on a parallel axis in the plane of the eccentric and rocking member axes and equally spaced, and a crank on the accessory shaft operatively connected with the said driving member whereby the crankshaft drives the accessory shaft synchronously.

8. A combination as described in claim 5 in which the engine has a multiplicity of combustion chambers operating at approximately equal angular intervals, and the blower is in duplicate with the two gyrors at 180° with respect to one another.

9. In combination with a multi-chambered internal combustion engine, a radial charge compressor mounted on the engine shaft and having an equal number of compression chambers, a stationary housing surrounding the shaft adjacent the compressor, an impeller secured to the shaft and turning therewith in the housing, means for supplying liquid and gas to the housing in the path of the impeller, means for supplying liquid and gas distributed by the impeller from the periphery of the housing separately to each chamber of the compressor, and means for supplying the compressed mixture from each compression chamber to a connected engine chamber.

10. In combination a multi-cylinder radial internal combustion engine, a radial charge compressor of an equal number of cylinders mounted on one face of the engine and driven by the crankshaft thereof, a charge distributor housing on the other face of the compressor and having ports in its periphery connected respectively with the compressor cylinders, an impeller secured to the crankshaft and operating in the housing, means for supplying liquid and gas to the housing in the path of the impeller for distribution thereby to the ring of peripheral ports, and means for supplying the compressed mixture from each compressor cylinder to a connecting engine cylinder.

11. In combination a multi-cylinder radial internal combustion engine, a circular housing mounted on one face of the engine having an intake and an outlet, a displacement member mounted on the crankshaft and operating in the housing to displace air under pressure, means for by-passing a portion of the air to the intake side of the housing, an outer circular conduit surrounding and forming a part of the housing, means for supplying air under pressure from the housing to the conduit, and means for delivering the air so supplied through the side face of the conduit directly to the cylinders of the engine.

12. In combination, a multi-cylinder radial internal combustion engine, a rotary blower centrally mounted on one face of the engine and driven by the crankshaft thereof, a multi-cylinder radial compressor centrally mounted on the other face of the engine and likewise driven by the crankshaft thereof, means for delivering the blower discharge in parallel to the engine cylinders, and means for delivering the compressor cylinder charge to a single connected cylinder of the engine.

13. A combination as described in claim 12 having in addition a circular housing adjacent the compressor and from the periphery of which lead individual suction ducts for the compressor cylinders, an impeller on the crankshaft within the housing, and means for delivering liquid and gas to the housing in the path of the im-

peller for distribution thereby to the ring of suction ducts.

14. In combination, a reciprocating single acting internal combustion engine, means for supplying its main air requirement in advance of the compression stroke of the piston, means for supplying additional air during the compression stroke comprising a reciprocating single acting compressor mounted beside the engine cylinder and driven from the crankshaft, means for supplying water as well as air to the compressor for delivery thereby to the engine, and a single valve interposed between the compressor cylinder and the engine cylinder and constituting the outlet valve of the compressor and the means for admitting the air and water delivered thereby into the engine cylinder.

15. In combination, a reciprocating single acting internal combustion engine, means for supplying its main air requirement in advance of the compression stroke of the piston, means for supplying additional air during the compression stroke comprising a reciprocating single acting compressor mounted beside the engine cylinder and driven from the crankshaft, means for supplying fuel and water as well as air to the compressor for delivery thereby to the engine, and a single valve interposed between the compressor cylinder and the engine cylinder and constituting the outlet valve of the compressor and the means for admitting the air delivered thereby into the engine cylinder.

16. A variable torque high speed positive displacement internal combustion engine having a combustion chamber of high surface to volume ratio operating on the two-stroke cycle, an air blower driven by the engine of capacity substantially in excess of the engine displacement, inlet and exhaust ports for the engine controlled by the piston, means for supplying the air delivered by the blower to the intake ports of the engine during the period of its cycle in which the exhaust ports are likewise open, whereby the excess delivery of the blower passes through the engine for substantial internal cooling of its combustion chamber walls, means for regulating such air supply whereby the air volume admitted per cycle may be reduced below the engine displacement for reduction of engine torque below full normal torque, means for delivering fuel to the combustion chamber after the closing of the exhaust port, means for applying a regulated back pressure to the exhaust port, and means responsive to the speed of the engine for progressively limiting the back pressure which may be applied to the exhaust as the engine speed increases.

17. A variable torque high speed positive displacement internal combustion engine having a combustion chamber of high surface to volume ratio operating on the two-stroke cycle, an air blower driven by the engine and of capacity substantially in excess of the engine displacement, inlet and exhaust ports for the engine controlled by the piston, means for supplying the air delivered by the blower to the intake ports of the engine, means governed by the speed of the engine for recycling through the blower a portion of the air of the blower for regulating such air supply whereby the air volume admitted per cycle may be reduced below the engine displacement for reduction of engine torque below full normal torque.

18. An engine according to claim 17 in which means are provided for applying a regulated back pressure to the exhaust port of the engine to retain therein an air volume in excess of the engine displacement volume and increase the torque above the full normal torque.

19. An engine according to claim 17 in which a loaded valve is provided on the discharge outlet of the exhaust port for applying a regulated back pressure to the exhaust port of the engine to retain an air volume therein in excess of the engine displacement volume and to increase the torque above the full normal torque.

20. A positive displacement two-cycle internal combustion engine having an air blower driven by the engine and of a capacity substantially in excess of the engine displacement, means for supplying the air delivered by the blower to the intake ports of the engine, means for regulating the air supply to reduce it below the engine displacement for reduction of engine torque below full normal torque, and means for applying a regulated back pressure to the exhaust ports of the engine to retain an air volume in excess of the engine displacement volume.

21. An engine according to claim 20 in which means are provided for progressively limiting the back pressure which may be applied to the exhaust as the speed of the engine increases.

22. In an internal combustion engine, a working cylinder and a piston therein, a combustion head lying above the outer limit of the piston stroke and forming a chamber of crescent form the arc of which is centered on a point offset from the cylinder axis, of maximum depth at its point of maximum width and of gradually diminishing depth toward the points of the crescent, means for adding fuel through a valve into the chamber, an ignition plug located in the central portion of the chamber, and inlet and exhaust ports in the lower end of the cylinder whereby the charge is passed through the valve and is conducted through the working cylinder to the exhaust port.

23. A two-cycle reciprocating internal combustion engine combined with a reciprocating compressor of relatively small displacement synchronously driven by the engine, means for supplying water as well as fuel and air to the compressor and for delivering the same from the compressor to the engine, and a rotary blower of displacement not substantially less than that of the engine, driven thereby for supply of the main requirement of the engine.

24. In combination, a reciprocating single acting internal combustion engine, means for supplying its main air requirement, means for supplying additional air comprising a reciprocating single acting compressor mounted beside the engine cylinder and driven from the crank-shaft, means for supplying water as well as air to the compressor for delivery thereby to the engine, and a single valve interposed between the compressor cylinder and the engine cylinder and constituting the outlet valve of the compressor and the means for admitting the air and water delivered thereby into the engine cylinder.

25. In combination, a reciprocating single acting internal combustion engine, means for supplying its main air requirement, means for supplying additional air comprising a reciprocating single acting compressor mounted beside the engine cylinder and driven from the crank-shaft, means for supplying fuel and water as well as air to the compressor for delivery thereby to the engine, and a single valve interposed between the compressor cylinder and the engine cylinder and constituting the outlet valve of the compressor

and the means for admitting the air delivered thereby into the engine cylinder.

26. A variable torque high speed positive displacement internal combustion engine having a combustion chamber of high surface to volume ratio operating on the two-stroke cycle, an air blower driven by the engine of capacity substantially in excess of the engine displacement, inlet and exhaust ports for the engine controlled by the piston, means for supplying the air delivered by the blower to the intake ports of the engine, means for regulating such air supply whereby the air volume admitted per cycle may be reduced below the engine displacement for reduction of engine torque below full normal torque, means for delivering fuel to the combustion chamber after the closing of the exhaust port, means for applying a regulated back pressure to the exhaust port, and means responsive to the speed of the engine for progressively limiting the back pressure which may be applied to the exhaust as the engine speed increases.

27. In a positive displacement two-cycle internal combustion engine having an air blower driven by the engine and of capacity substantially in excess of the engine displacement, means for supplying the air delivered by the blower to the intake ports of the engine during the period in which the exhaust ports are likewise open whereby the excess air passes through the engine for internal cooling of the combustion chamber walls, means for regulating the air supply to reduce it below the engine displacement for reduction of engine torque below full normal torque, and means for applying a regulated back pressure to the exhaust port of the engine to retain an air volume in excess of the engine displacement volume.

28. In combination with an internal combustion engine, an eccentric synchronously driven by the crank-shaft thereof, a driving member rotatably mounted on the eccentric and having a sliding fit with a spaced rocking member, an accessory shaft journaled for rotation, and a crank on the accessory shaft operatively connected with the driving ...ember whereby the crank-shaft drives the accessory shaft synchronously.

29. In combination, a multi cylinder radial internal combustion engine, a housing associated with the engine and having an intake and an outlet, a displacement member mounted on the crank-shaft and operating in the housing to displace air under pressure, means for by-passing a portion of the air to the intake side of the housing, a conduit, means for supplying air under pressure from the housing to the conduit, and means for delivering the air so supplied through the conduit directly to the cylinders of the engine.

30. In combination, a multi cylinder radial internal combustion engine, a blower driven by the crank-shaft of the engine, a compressor driven by the crank-shaft, means for successively delivering the blower discharge to the engine cylinders, and means for delivering the compressor discharge to a single connected cylinder of the engine.

31. In combination, a multi cylinder radial internal combustion engine, a blower driven by the crank-shaft of the engine, a compressor driven by the crank-shaft, means for successively delivering the blower discharge to the engine cylinders, means for applying a regulated back pressure to the exhaust of the engine cylinders, and means for delivering the compressor discharge to a single connected cylinder of the engine.

32. In an internal combustion engine having a drive shaft and having an exhaust pipe, a valve for the exhaust pipe, a governor, means operatively connected with the drive shaft for driving the governor at a speed proportional to the speed of the drive shaft, and means connected to the valve and controlled by the governor for varying the position of the valve to progressively limit the back pressure applied to the exhaust as the speed of the engine increases.

33. In an internal combustion engine having a drive shaft and having an exhaust pipe, a valve for the exhaust pipe, a governor, and means driven by the drive shaft and connected to the valve for altering the position of the valve proportionately to the speed of rotation of the shaft for progressively limiting the back pressure which may be applied to the exhaust as the speed of the engine increases.

34. In an internal combustion engine having a drive shaft and having an exhaust pipe, a weighted valve for the exhaust pipe, a governor, and means driven by the drive shaft and connected to the valve for altering the position of the valve proportionately to the speed of rotation of the shaft for progressively limiting the back pressure which may be applied to the exhaust as the speed of the engine increases.

35. A variable torque high speed positive displacement internal combustion engine having a combustion chamber of high surface to volume ratio operating on the two-stroke cycle, an air blower driven by the engine and of capacity substantially in excess of the engine displacement, means for supplying the air delivered by the blower to the intake ports of the engine, means for recycling through the blower a portion of the air of the blower for regulating such air supply whereby the air volume admitted per cycle may be reduced below the engine displacement for reduction of engine torque below full normal torque and means for applying a regulated back pressure to the exhaust port of the engine to retain therein an air volume in excess of the engine displacement volume and increase the torque above the full normal torque.

36. A variable torque high speed positive displacement internal combustion engine having a combustion chamber of high surface to volume ratio operating on the two-stroke cycle, an air blower driven by the engine and of capacity substantially in excess of the engine displacement, means for supplying the air delivered by the blower to the intake ports of the engine, means for recycling through the blower a portion of the air of the blower for regulating such air supply whereby the air volume admitted per cycle may be reduced below the engine displacement for reduction of engine torque below full normal torque and means for applying a regulated back pressure to the exhaust port of the engine to retain therein an air volume in excess of the engine displacement volume and increase the torque above the full normal torque and a loaded valve on the discharge outlet of the exhaust port for applying a regulated back pressure to the exhaust port of the engine to retain an air volume therein in excess of the engine displacement volume and to increase the torque above the full normal torque.

37. A variable torque high speed positive displacement internal combustion engine having a combustion chamber of high surface to volume ratio operating on the two-stroke cycle, an air blower driven by the engine and of capacity sub-

stantially in excess of the engine displacement, means for supplying the air delivered by the blower to the intake ports of the engine, means for recycling through the blower a portion of the air of the blower for regulating such air supply whereby the air volume admitted per cycle may be reduced below the engine displacement for reduction of engine torque below full normal torque, and means operatively connected to the engine and governed by the speed of the engine for supplying a regulated back pressure to the exhaust port of the engine to retain therein an air volume in excess of the engine displacement volume and increase the torque above full normal torque.

FRANK A. HOWARD.