June 17, 1941. J. J. WYDLER 2,245,890

AIR COMPRESSOR

Filed Feb. 15, 1940

Fig. I.

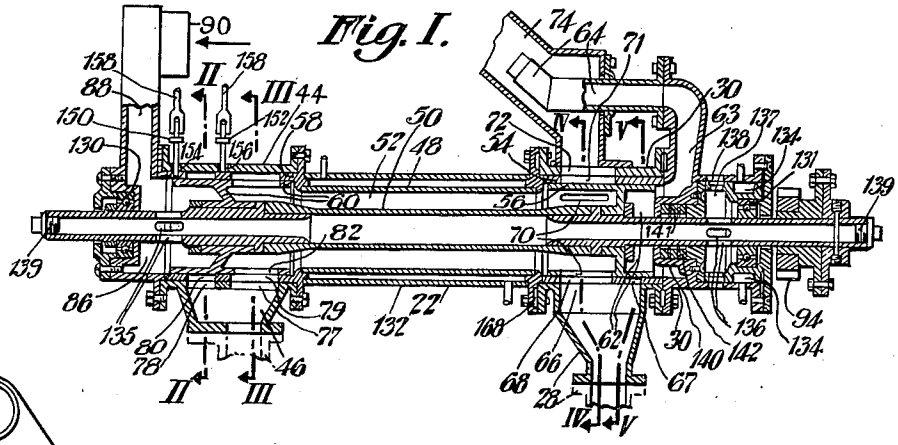


Fig. II.

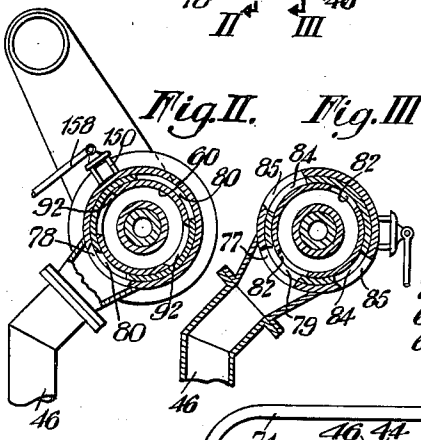


Fig. III

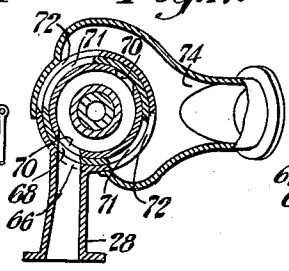


Fig. IV.

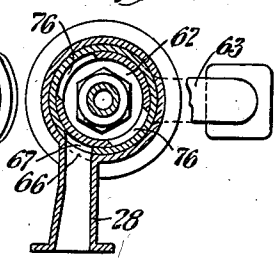


Fig. V.

Fig. VI.

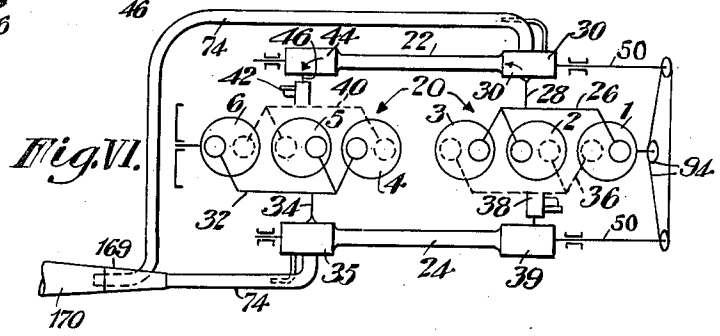


Fig. VII.

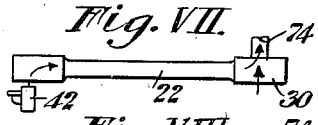


Fig. VIII.

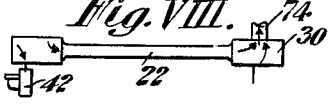
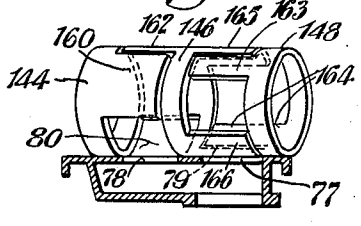


Fig. IX.



INVENTOR
JOHANN J. WYDLER
BY Edmund G. Morley
ATTORNEY

Patented June 17, 1941

2,245,890

UNITED STATES PATENT OFFICE

2,245,890

AIR COMPRESSOR

Johann J. Wydler, Westfield, N. J., assignor, by mesne assignments, to Cities Service Oil Company, New York, N. Y., a corporation of Pennsylvania

Application February 15, 1940, Serial No. 318,985

11 Claims. (Cl. 230—95)

This invention relates to an improved apparatus for deriving energy from engine exhaust gases and utilizing such energy for compressing air for supercharging the engine.

The gas exhaust period of the cycle of any four cycle internal combustion engine cylinder consists of two parts. During the first part of the exhaust period, just after the exhaust valve has been opened, a substantial proportion (roughly 50%) of the total weight of gas in the cylinder is rapidly discharged as a high pressure "puff" wave moving outwardly from the cylinder into the exhaust manifold at high initial pressure and at high velocity. During the latter part of the exhaust period the remaining portion of the exhaust gases leave the cylinder under a relatively low pressure head and moderate velocity in front of the advancing piston; this period of the cycle being referred to as the "stroke" period of the exhaust.

The present invention is directed to an improvement on that described in the copending joint application of Robert G. Griswold and Johann J. Wydler, Serial No. 240,014, filed November 12, 1938, for "Engine supercharging." According to the invention of the aforesaid joint application, a displacement compressor is employed as the medium for compressing air for supercharging one cylinder of a multicylinder, four cycle internal combustion engine by means of the pressure energy of the coinciding "puff" discharge wave of exhaust gases leaving another cylinder of the engine which is operating 360° apart in crank angle phase with respect to the first cylinder. This displacement compressor operates as a gas piston pump or compressor in which a body of air first introduced into the pump chamber is compressed and then discharged by a nonturbulent stratified layer or wave of hot exhaust gases under higher pressure moving forward in direct contact with and displacing the air in the pump chamber without substantial mixing with and contamination of the air by the gas.

The principal object of the present invention is to provide an improved apparatus for deriving energy from the waste exhaust gases of an internal combustion engine and using such energy for compressing air.

A feature of the present invention is that an improved design of displacement compressor or pump is provided having relatively rotatable valves and housing therefor, together with mechanism for adjusting the area and timing of the valve ports controlling transfer of atmospheric and supercharge air and exhaust gases. A preferred compressor design embodies a pair of rotary cylinder valves and adjustable sleeves therefor, located respectively at the hot and cold gas ends of each displacement compressor. The single valve at the cold end is designed to control and properly time the periods of engine atmospheric air intake, pump discharge of compressed air, and air scavenging of the pump. The valve at the hot end operates to control and time the periods of pump intake of engine puff exhaust gases, discharge of exhaust gases from the pump, and transfer of engine stroke exhaust gases to suction aspirators.

The air compression and displacement period in the operating cycle of each displacement compressor is followed by a scavenging period during which the exhaust gases are discharged from the compressor and the compressor is scavenged with atmospheric air. A major part of the energy for such scavenging operation is preferably derived from the exhaust gases which are discharged from the exhausting engine cylinder during the stroke part of the cylinder exhaust cycle. To provide an additional scavenging effect, the gas exhaust ducts of a pair of dispacement compressors are preferably interconnected by concentric twin nozzles, each of which operates to develop repeated suction impulses at the respective compressor exhaust ports during one and the same scavenging period.

With the above and other objects and features in view, the invention consists in the improved apparatus for compressing air and for supercharging a four cycle internal combustion engine, which is hereinafter described and more particularly defined by the accompanying claims.

The invention will hereinafter be described with particular reference to the accompanying drawing, in which:

Fig. I is a horizontal sectional view of a preferred design of displacement air compressor;

Fig. II is a vertical sectional view of the air transfer ports and control valve therefor, taken on the line II—II of Fig. I;

Fig. III is another vertical sectional view through the air transfer port end of the compressor, taken on the line III—III of Fig. I;

Fig. IV is another vertical cross sectional view through the hot gas transfer port end of the compressor, taken on the line IV—IV of Fig. I;

Fig. V is still another vertical cross sectional view through the hot gas transfer port end of the compressor, taken on the line V—V of Fig. I;

Fig. VI is a diagrammatic view illustrating the

hook-up of all six cylinders of a six cylinder engine, through a pair of exhaust manifolds and a pair of intake manifolds and carbureters, with a pair of displacement compressors;

Fig. VII is a diagrammatic view illustrating by means of arrows the flow of gases during the re-expansion period in the operating cycle of the upper displacement compressor shown in Fig. VI. In Fig. VI the direction of flow of air and gas during the puff supercharging period is indicated;

Fig. VIII is a diagrammatic view of the atmospheric air intake period of the scavenging cycle of the upper compressor illustrated in Fig. VI, and of the simultaneous atmospheric air intake of the engine cylinder next in cycle with the pump; and

Fig. IX is a perspective view of the three-sectioned adjustable sleeve which journals the rotary valve at the air transfer end of the displacement compressor.

The apparatus illustrated diagrammatically in Fig. VI includes a six cylinder four stroke cycle internal combustion engine 20 and two displacement air compressors 22 and 24 which are operatively connected for compressing air at the expense of energy derived from the engine exhaust gases, and utilizing the compressed air for supercharging the engine. In its broadest scope, however, the invention is not limited to the use of the compressed air for engine supercharging; nor is the invention limited in application to an engine having six cylinders.

In Fig. VI the cylinders of a six cylinder, four cycle engine have been numbered respectively 1, 2, 3, 4, 5 and 6; and cylinders 1, 2 and 3 have been shown with their exhaust ports connected through an exhaust manifold 26 and conduit 28 to a hot gas inlet valve housing 30 at one end of a compressor 22; while the exhaust ports of cylinders 4, 5 and 6 have been shown as connected through an exhaust manifold 32 and conduit 34 to the hot gas inlet valve housing 35 at one end of compressor 24. Likewise the intake ports of cylinders 1, 2 and 3 have been shown as connected through an intake manifold 36 and carburetor 38 to an air valve housing 39 at the other end of compressor 24; while the intake ports of cylinders 4, 5 and 6 have been shown as connected through an intake manifold 40, carburetor 42 and pipe 46 to an air valve housing 44 at the other end of compressor 22 (Fig. I).

The cylinders of a multicylinder engine must be paired in practicing the preferred supercharging operation of this invention, so that the energy carried by the exhaust gas discharged from one cylinder of a pair can be utilized for compressing the air introduced into the other paired cylinder at the end of its intake period. During the first part of the air intake period for each cylinder air is supplied at atmospheric pressure. The pistons in each cylinder of a pair, such as 2 and 5, pass simultaneously through their top and bottom dead center positions. However, the power strokes of the pistons are 360° crank angle apart in phase. In the case of engines having an uneven number of cylinders, as for example nine cylinders, the dead center positions of the pistons in paired cylinders are not exactly together (for example 40° apart) and therefore the power strokes are apart in phase less than 360° (for example 320°).

The displacement compressors or pumps 22 and 24 are identical in design. Each has a cylindrical main housing 48 which preferably has a length several times its maximum internal diameter (Fig. I). Valve housings 30 and 44 of compressor 22 are shown as attached respectively to the opposite ends of housing 48 as extensions thereof. A hollow valve drive shaft 50 is mounted within the housing 48, extending from end to end thereof along the longitudinal axis of the compressor. Displacement and compression chamber 52 of the compressor is of annular cross section and lies between the inner wall of the housing 48 and the outer circumference of the shaft 50. The chamber 52 preferably has a cubic capacity only sufficient to handle the volume of hot gas which is discharged from a single engine cylinder during the first or puff portion of its exhaust, and to compress only the air with which a cylinder is supercharged at the end of its air intake period. Within that extension 30 of the compressor 22 into which hot exhaust gases are admitted from exhaust manifold 26, there is mounted an apertured cylindrical sleeve or bushing 54 within which is journaled a ported cylinder valve 56. The hub of valve 56 is keyed to shaft 50 for rotation therewith. Within the valve housing 44 at the opposite end of the compressor, there is assembled a multi-sectioned apertured cylindrical sleeve or bushing 58 within which is journaled a ported cylinder valve 60. Valve 60 has a hub which is preferably fastened to shaft 50 by a set screw for rotation therewith. The imperforate web extensions of the hubs of valves 56 and 60 which attach the cylindrical valve walls to the hubs, form end wall closures for the compressor chamber 52.

Referring to Figs. I, IV and V, it will be seen that that end of the compressor 22 into which hot exhaust gases are introduced from the exhaust manifold 26 houses an annular gas by-pass chamber 62 which is in permanent communication with the atmosphere through a funnel 63 and an aspirator pressure nozzle 64. The hot gases from manifold 26 are conducted by conduit 28 to a permanent aperture 66 in the wall of valve housing 30. That portion of sleeve 54 which overlies housing aperture 66 is also provided with two apertures 67, 68, which register with the aperture 66. Valve 56 is provided with a pair of identical ports 70 in opposite sides of that portion of its cylindrical wall which is rotatably aligned with sleeve aperture 68. Sleeve 54 and valve housing 30 are also provided respectively with registering apertures 71, 72, which are adapted to communicably connect the interior of housing 30 and sleeve 54 with a gas discharge funnel 74. That portion of the cylindrical wall of valve 56 which lies to the right of the hub web, as viewed in Fig. I, is also provided with a pair of identical oppositely disposed wall ports 76 (Fig. V), which on rotation of the valve about 30°–40° clockwise from the position illustrated in Figs. I and V, will come into register with the apertures 66 and 67.

Valve housing 44 at the opposite end of the compressor is permanently apertured at 77 and 78 (Figs. I, II and III). Sleeve 58 is also provided with wall apertures 79 and 80 which normally register with apertures 77 and 78, respectively. Valve 60 is provided with a pair of identical ports 82 in opposite sides of that portion of its cylindrical wall which is rotatably aligned with apertures 77 and 79. When valve 60 is rotated about 90° clockwise from the position illustrated in Fig. III, its ports 82 are moved into full register with other apertures 84 and 85 located respectively in the walls of sleeve 58 and housing 44. An air by-pass chamber 86 is formed within valve hous-

ing 44 to the left of the hub web of the valve as viewed in Fig. I. Chamber 86 is permanently connected to atmosphere through an air supply conduit 88 and filter 90. As shown in Fig. II, that portion of the cylindrical wall of valve 60 which lies to the left of the hub web, as viewed in Fig. I, is provided with a pair of oppositely disposed identical ports 92 which, on rotation of the valve about 90° clockwise from the position illustrated in Figs. I and II, will come into full register with apertures 78 and 80.

Thus with the shaft 50 and valves 56 and 60 in place, the displacement compressor comprises a pair of short annular chambers 62 and 86 located at opposite ends thereof, each chamber being permanently connected to the outside atmosphere, together with a long displacement air compression chamber 52 of annular cross section lying between shaft 50, housing 48 and the hubs of valves 56 and 60.

The engine 20 is of course equipped with the usual air and fuel valves and exhaust valves for each cylinder intake and exhaust port. Combustion within any cylinders of the engine is initiated by means of a spark plug, the timing of which is controlled by a distributor. The engine intake and exhaust valves and the distributor are all actuated from the engine through a cam shaft which operates at half the speed of the engine crank shaft. The drive shafts 50 for the valves 56 and 60 in each of the compressors 22 and 24 are driven at three-fourths the speed of the engine through suitable drive mechanism such as the chain and gear drives 94, which are illustrated in Figs. I and VI.

The valve drive shaft 50 for each compressor is journaled at opposite ends of the compressor on ball bearings 130 and 131 (Fig. I). An important feature in the design of the compressor is the provision of means for limiting transfer of heat from the hot gas end to the cold air transfer end of the compressor and to the valve shaft bearings on the hot side. This protection may include a water jacket 132 for the main section 48 of the compressor, together with an annular water cooled compartment 134 around or adjacent bearing 131. Means are also shown for cooling the bearing 131 at the hot gas intake side of the compressor by air ventilation. To effect this, apertures 135 are provided in that portion of the wall of shaft 50 which rotates in air intake chamber 86. Additional wall apertures 136 are provided in that portion of the shaft which is centered in a chamber 137 within which the bearing 131 is mounted. Apertures 138 are also ported out from chamber 137 through the compressor housing, and the ends of the hollow shaft 50 are closed by plugs 139. By this arrangement, a cooling draft of air can enter chamber 137 from the outside atmosphere and flow thence through the hollow bore of shaft 50 into air intake chamber 86 during that portion of the cycle of each engine cylinder during which air at atmospheric pressure is being taken into the cylinder. To seal the bearing 131 against contact with the hot gases entering the chamber 62 of the compressor, a partition stuffing box 140 is mounted between chamber 62 and air chamber 137. Essentially this stuffing box comprises a housing for a plurality of shim stock seal rings 141 encircling the shaft 50 and separated from each other by spacing rings 142. All the rings are prevented from rotating with the shaft by a key; however, they are fitted loosely with respect to each other and with a minute clearance on the shaft to permit true self-centering adjustment on the shaft.

When operating engine 20 with supercharging, the air supply end of the compressor with its rotary valve 60, must function to: (*a*) transfer air at atmospheric pressure through chamber 86 to the intake port of a cylinder operatively connected to the compressor during the major part of the cylinder intake period (Fig. VIII); (*b*) cut off the atmospheric air transfer and transfer compressed air from the compressor chamber 52 as a puff supercharging wave through the carbureter into the intaking cylinder at the end of the intaking period (Figs. I, III and VI); (*c*) simultaneously with at least part of the atmospheric air transfer period (*a*), admit scavenging air from atmosphere through ports 85, 84, 82, into the compressor chamber 52 (Fig. VIII). Likewise, the hot exhaust gas receiving end of the compressor with its rotary valve 56 must function to: (*d*) introduce the first or puff exhaust gas discharge from a cylinder operatively connected to the compressor into the chamber 52 during the supercharging period (*b*) (Figs. I, IV and VI); (*e*) cut-off transfer of exhaust gas to chamber 52 and switch the exhaust gas discharged during the second or stroke portion of the engine exhaust into the aspirator pressure nozzle 64 (Figs. VII, VIII); (*f*) simultaneously with actions (*c*) and (*e*) open the connection between 52 and exhaust discharge funnel 74 for the purpose of scavenging the compressor before beginning a new cycle. When operating the engine normally without supercharging, the air supply end of the compressor and valve 60 should also function to (*g*) transfer atmospheric air to the engine intaking cylinder connected therewith throughout the entire intake period.

In order to enable the air supply end of the compressor and valve 60 to function as specified under *a, b, c,* and *g,* sleeve 58 has been designed in three ring sections, 144, 146 and 148 (Fig. IX), and two of these sections 144 and 146 have been made rotatably adjustable within housing 44 for the purpose of varying the length of time during which sleeve ports 80 are in position to allow air passed by valve ports 92 to be transferred to conduit 46. Sleeve sections 144 and 146 are provided respectively with handles 150 and 152 (Fig. I) extending to the outside of housing 44 through arcuate slots 154 and 156 in the housing. Actuating mechanism 158 is provided attached to handles 150 and 152 of the sleeve sections 144 and 146 at the air intake ends of both compressors 22 and 24 whereby to simultaneously move sections 144 and 146 of the sleeve 58 in each compressor in opposite directions to vary the size of openings 80. Timing adjustments of the atmospheric air intake periods can thus be made by means of sleeve 58, as well as by shifting the point of attachment of valve 60 to shaft 50 by the set screw. The hot valve 56 is preferably keyed to shaft 50, but the timing of the gas exhaust periods of the compressor may be varied by shifting the angular position of the shaft 50 and valve 56 relative to the engine crank shaft at the chain drive sprocket 94.

In Fig. II, the two apertures 80 in the wall of the left hand side of sleeve 58 as viewed in Fig. I, are illustrated as of unequal area. Fig. IX illustrates the construction whereby the relative areas of these two apertures can be varied by rotating sleeve segments 144 and 146 in opposite directions. Sleeve segment 144 consists of a ring to which is attached a cylinder segment 160

which in the preferred design subtends an arc of 115°. Segment 160 extends to the right of ring 144 as viewed in Figs. I and IX, with its outer edge forming a close sliding fit with ring 146. Ring 146 carries a corresponding cylinder segment 162 which also subtends an arc of 115° and which extends to the left, as viewed in Fig. IX, with its outer edge in close sliding relation to ring 144. Thus the apertures 80 are defined as to length and width by the rings 144 and 146 and their segments 160 and 162. Similarly, apertures 84 and 79 at the right hand side of sleeve 58 may be varied as to relative areas by relative movement of ring 146 with respect to stationary ring 148. Referring to Fig. IX, it will be seen that ring 148 is provided with two oppositely disposed cylinder segments 163, 164, one of which in this case subtends an arc of 48°, and the other an arc of 42°. The ends of segments 163 and 164 extend to close sliding relation with ring 146. Ring 146 carries at its right hand side as viewed in Fig. IX, a pair of oppositely disposed cylinder segments 165 and 166, one of which subtends an arc of 48°, and the other an arc of 42°. Thus the rings 146 and 148, together with their segments 163, 164, 165 and 166 afford the means for varying the lengths of the air scavenging period and of the compressed air discharge period for the compressor.

Valves 56 and 60 are designed to rotate with very small clearance within the sleeve bushings 54 and 58. To reduce conduction of heat by the metal parts from the hot ends of the compressor, a heat dam 168 is disposed between housing sections 30 and 48. Other heat dams are provided between chamber 62 and the housing of bearing 131, and circumferentially between the retainer of bearing 131 and the housing.

It is possible to obtain wide adjustments in the timing and intensity of the aspirator effect when operating with a multi-cylinder engine. The aspirator jet 64 has its greatest force during the re-expansion period in the cycle of the compressor, and reaches a second and smaller peak at about the mid-portion of the exhaust stroke of the piston in the exhausting engine cylinder. Considerable benefit has been obtained by operation of the aspirator 64 prior to the instant of cut-off of transfer of exhaust gases into chamber 52 by valve 56.

Referring to Fig. VI, there has been illustrated diagrammatically an arrangement of multiple aspirator jets and interconnected compressor discharge conduits 74, by means of which it is possible to utilize more efficiently suction impacts developed by one compressor discharge for promoting efficient scavenging of a second compressor interconnected therewith. Thus, in Fig. VI, the exhaust lines 74 of compressors 22 and 24 are interconnected at the throat 169 into a common discharge funnel 170. Each conduit 74 terminates at the throat in the equivalent of a secondary aspirator nozzle adapted to develop suction in the other exhaust line. By this arrangement, the tail end of the scavenging period in one compressor 22 is most efficiently supplied with an additional suction impact produced by the high velocity discharge of gases during the re-expansion portion of the operating cycle of compressor 24. Similarly, the tail end of the scavenging period in compressor 24 is boosted by the suction impact developed by high velocity discharge of gases from compressor 22. The secondary aspirator thus provided at the throat 169 is made up of two concentric orifices, each having an area equal to or slightly smaller than the area of its respective exhaust pipe 74. The cone 170 of the secondary aspirator is of gradual widening cross-sectional area ultimately leading to atmosphere, preferably by way of a muffling device.

When cylinder 1 is operatively connected to the compressor 22 during its exhaust period, the puff exhaust gas discharge wave which enters the compressor chamber 52, compresses the air charge within the compressor and interconnected spaces until the total mass of air and gas has reached a balancing pressure. If at the same time cylinder 6 is operatively connected to the compressor, valve 60 of the compressor operates to cut off the supply of atmospheric air to cylinder 6 and to connect the compressor space 52 with the intake port of the cylinder to permit transfer of compressed air from the compressor through the carbureter and intake manifold into this cylinder during the latter part of its intake period and during the period when the piston therein is just starting its compression stroke. When the piston in the air intaking cylinder 6 reaches a point in its travel about 40°–60° crank angle beyond bottom dead center position on the compression stroke, its air intake valve closes. During this period of transfer of air from the compressor to the cylinder 6 substantially the entire original air charge for the compressor is discharged under substantially the peak pressure impressed thereon by the puff exhaust wave from exhausting cylinder 1 of the engine. At the instant that the intake valve of cylinder 6 closes, the interior of the compressor 22 is still communicably connected with the intake manifold 40. However, at this instant the valve 56 of the compressor operates to cut off any further supply of hot exhaust gases from cylinder 1 to the compressor, and to open the compressor for discharge of gas therefrom to atmosphere through the exhaust funnel 74.

As soon as compressor valve 56 opens to discharge gas into funnel 74, a rapid discharge takes place as a result of suction developed by the final transfer of exhaust gases from cylinder 1 to the aspirator nozzle 64 during the stroke portion of the cylinder exhaust period. The suction thereby developed quickly reduces the pressure within the compressor to a point below atmospheric pressure. This puff exhaust period extends in this particular example over 88° crank angle, and the puff intake ports stay open over 120° crank angle, thus allowing for re-expansion within the compressor over a period of 120° minus 88° equals 32° crank angle.

At the end of the re-expansion period valve 60 disconnects space 52 from the intake manifold and carbureter and brings ports 82 into register with atmospheric air intake ports 84 and 85, thus permitting influx of fresh scavenging air to the compressor preparatory to operation of the compressor on a second cycle, during which the exhausting cylinder is cylinder 3, and the cylinder taking in supercharge air is cylinder 4.

In Fig. I the compressor ports are shown at the position which they occupy during the period of puff supercharging. The positions of the ports which are illustrated in Figs. II to V inclusive, represent the instant about 45° crank angle after the beginning of the puff period, i. e. when the exhaust gas inlet port 70 of the compressor is fully open. At this time the puff air discharge port 82 has still about 11°

crank angle to move to reach full open position.

It will be appreciated that the air scavenging of the compressor can be effected without developing suction by means of an aspirator, and without supplying scavenging air at atmospheric pressure. Thus scavenging air may be supplied to the air intake end of the compressor, under the positive pressure furnished by a fan or blower. Likewise the invention is not limited in other respects to the specific apparatus which has been described. For example, the invention contemplates a compressor in which the casing or shell may be rotatable and the cylindrical valves stationary, and in which other means may be substituted for those specifically described whereby to vary the timing and/or area of the gas and air transfer port openings.

The invention having been thus described, what is claimed as new is:

1. In a gas displacement air compressor, an elongated tubular housing, individual pressure gas inlet and waste gas discharge ports at spaced points in said housing adjacent one end thereof, individual atmospheric air inlet and compressed air discharge ports at spaced points in said housing adjacent its other end, a valve unit mounted coaxially with said housing for relative rotation with respect thereto and having tubular side walls positioned and dimensioned to form a close journal fit with the ported sections of the housing, and ports in the valve unit rotatably aligned respectively with the housing gas inlet and discharge ports and with the housing air inlet and discharge ports, said valve ports being so disposed as to communicably connect the housing gas inlet and air discharge ports at one period of the compressor cycle, and to communicably connect the housing gas discharge and air intake ports at a later period of the cycle.

2. A displacement compressor as defined in claim 1, including a pair of cylindrical apertured sleeve bushings within which the ported sections of the valve unit at each end of the housing are respectively journaled, the apertures in the bushings being adapted to register with the ports in the housing.

3. In a gas displacement air compressor, an elongated tubular housing, a displacing gas inlet and two gas discharge ports at peripherally spaced points in the wall of said housing adjacent one end thereof, a pair of air inlet ports and a compressed air discharge port at peripherally spaced points in the wall of said housing adjacent its other end, a valve unit mounted coaxially within said housing for relative rotation with respect thereto and having tubular side walls positioned and dimensioned to form a close journal fit with the ported sections of the housing, and two pairs of wall ports in the valve unit rotatably aligned respectively with the housing gas inlet and discharge ports and with the housing air inlet and discharge ports, said valve ports being so disposed that individual ports of each pair respectively register with the housing gas inlet and air discharge ports at one operating position of the valve unit, and at a later operating position the paired ports at each end of the valve unit register respectively with the housing gas discharge ports and the air inlet ports.

4. A displacement air compressor as defined in claim 1 including a cylindrical apertured sleeve bushing at the air inlet and discharge port end of the housing within which the corresponding end of the valve is journaled, said sleeve comprising at least two relatively rotatable ring sections arranged for adjusting the areas and position of the apertures.

5. A displacement compressor as defined in claim 1 including wall closures for each end of the housing and end walls at each end of the valve unit, gas and air bypass chambers located respectively between the ends of the valve and the adjacent housing end closures, two additional pairs of housing wall ports opening respectively into the air and gas bypass chambers at spaced points in the walls thereof, and wall extensions of the valve unit at each end thereof beyond the adjacent end wall arranged respectively for periodically making and interrupting communication through the air and gas bypass chambers between the housing wall ports opening therein.

6. A displacement compressor as defined in claim 1 together with an aspirator having a suction throat communicably connected with the housing gas discharge port and having a compressor nozzle mounted in said throat, and a communicable bypass connection under the control of the compressor valve unit between the housing gas inlet port and the aspirator pressure nozzle.

7. A displacement compressor as defined in claim 1 in which the ends of the housing in which the gas and air inlet and discharge ports are located are constructed as detachable tubular segment extensions of the main housing, and in which the main housing is water jacketed.

8. A displacement compressor as defined in claim 1 together with a hollow rotatable shaft mounted longitudinally within said housing on which said valve unit is rotatably mounted, bearings for the shaft at each end of the housing, and a stuffing box arranged as a partition between the gas inlet port and the bearings at the gas inlet end of the housing.

9. A displacement compressor as defined in claim 1 in which that end of the housing which contains the gas inlet and discharge ports is constructed as a detachable tube segment extension of the housing, and in which heat dams in the form of knife edge joints are provided at the points of contact of the housing and said housing extension.

10. In a gas displacement air compressor, an elongated tubular housing, individual gas inlet and discharge ports at spaced points in the wall of said housing adjacent one end thereof, individual air inlet and discharge ports at spaced points in the wall of said housing adjacent its other end, and a valve unit mounted coaxially within said housing for rotation with respect thereto, said valve unit having a rotatable shaft mounted longitudinally within the housing, tubular side walls positioned and dimensioned to form a close journal fit with the ported sections of the housing, imperforate webs at each end of the valve unit connecting the side walls to the shaft, and wall ports in the valve unit rotatably aligned respectively with the housing gas inlet and discharge ports and with the housing air inlet and discharge ports, said valve ports being so disposed relative to each other and to the housing ports that one valve port registers with the housing air discharge port while another registers with the housing gas inlet port, and so that the housing air inlet and gas discharge ports are communicably connected by co-registering valve ports only when the gas inlet and

air discharge ports of the housing are blanked off by the valve.

11. A gas displacement compressor as defined in claim 1 in which the several housing ports and valve ports are so positioned and dimensioned with respect to each other that the housing air discharge port is communicably connected to the compressor for a short re-expansion period after the gas inlet port has been closed and after the gas discharge port has been opened and that the housing air inlet port is open only when the air discharge port is closed and when the gas inlet port is also closed.

JOHANN J. WYDLER.