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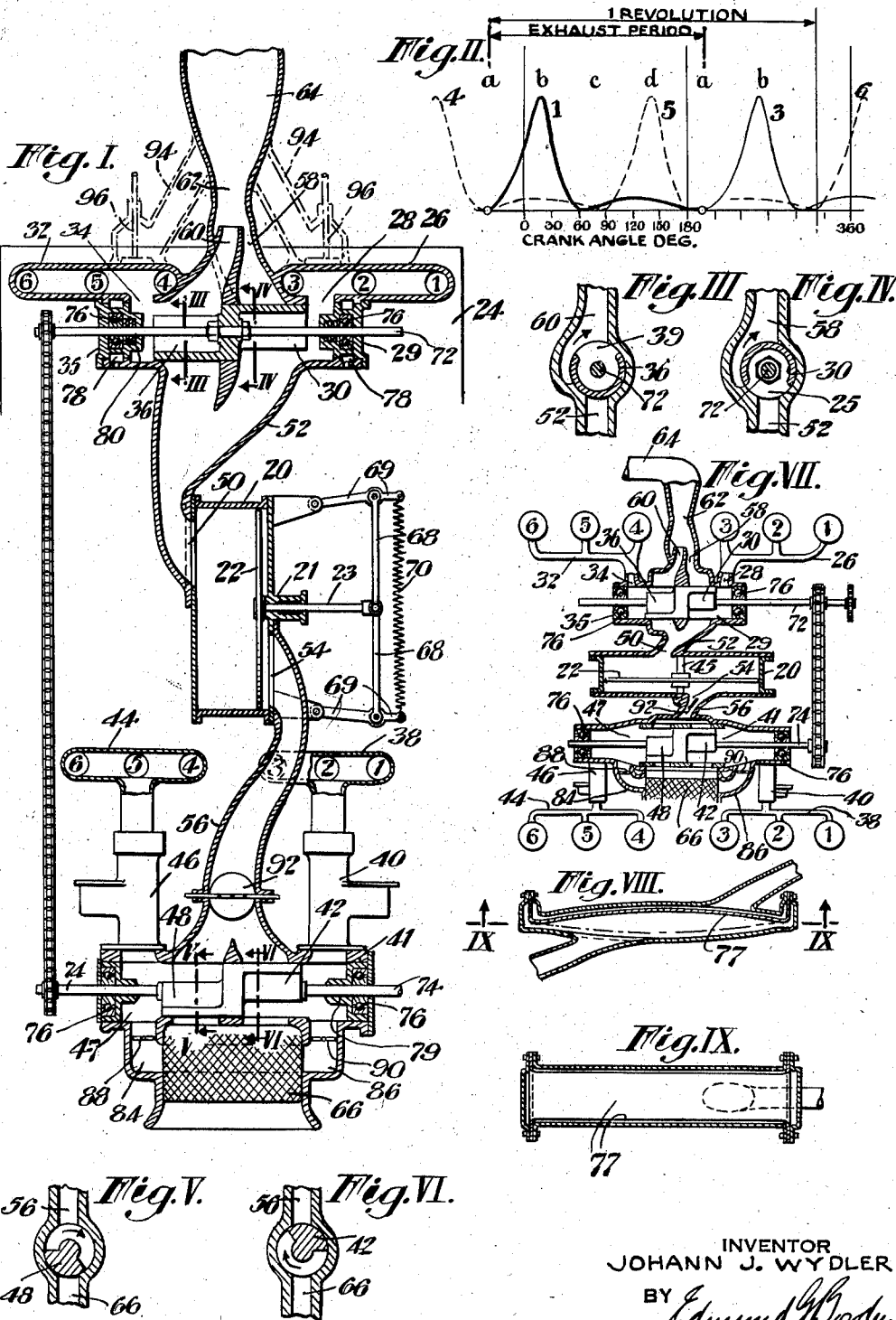
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GAS PUMPING

Filed April 11, 1940

2 Sheets-Sheet 1



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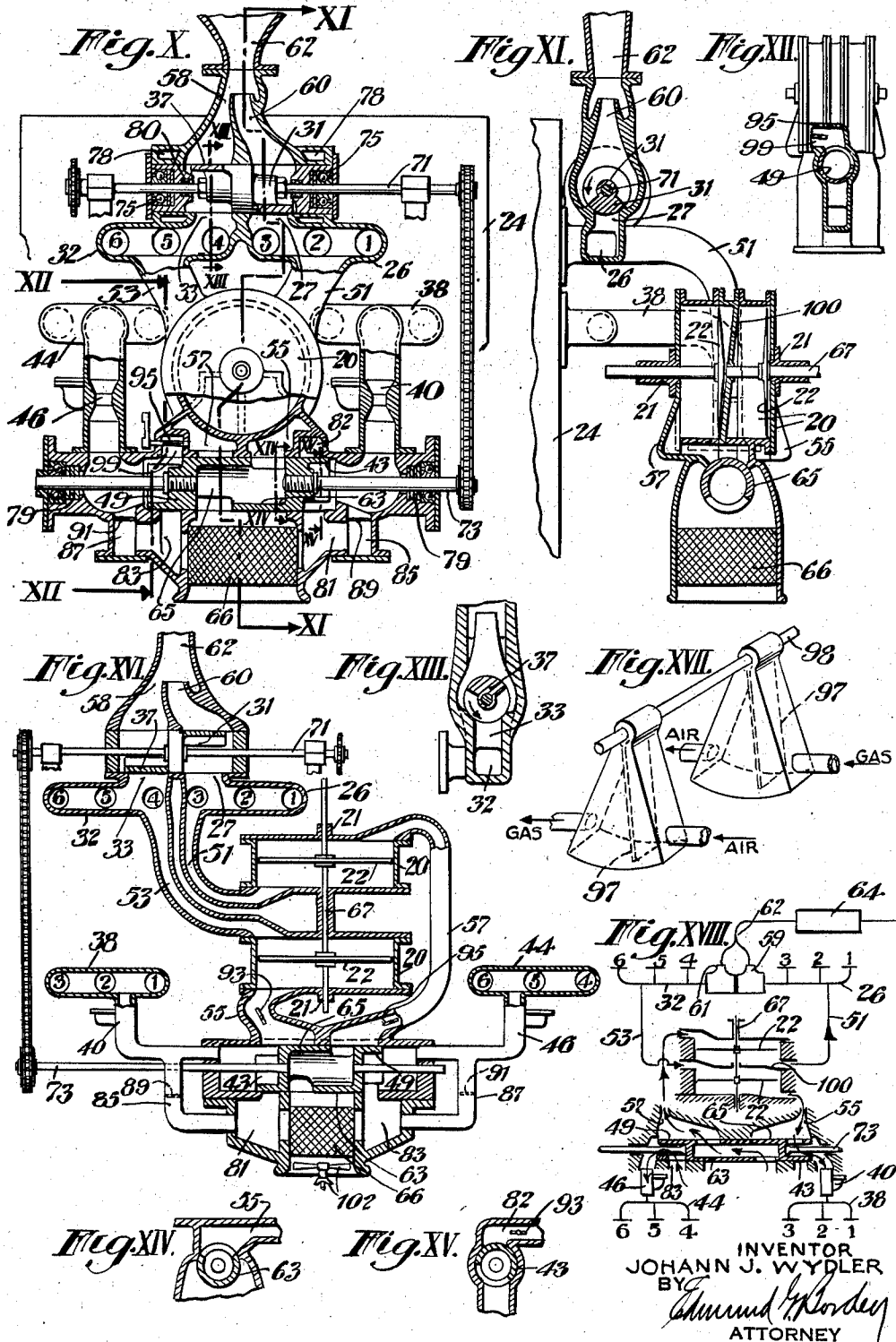
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UNITED STATES PATENT OFFICE

2,293,186

GAS PUMPING

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17 Claims. (Cl. 230-69)

This invention relates to gas pumping, and is particularly concerned with improvements in method and apparatus for utilizing energy of gases under superatmospheric pressure for producing flow of and compressing other stationary bodies of gas under lower pressure.

A particular object of the invention is to provide improved method and means for utilizing the potential energy which is available in the hot waste exhaust gases discharged from the cylinders of an internal combustion engine for compressing and pumping air.

The present invention is directed to modifications of and improvements on the invention described in my copending application Serial No. 240,015, filed November 12, 1938, for Gas pumping, U. S. Patent No. 2,234,100, issued March 4, 1941.

The gas exhaust period of the cycle of any four stroke 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 relatively high initial pressure and at high velocity. During the latter part of the exhaust period the remaining portion of the exhaust gases leaves the cylinder as a relatively low pressure wave moving in front of the advancing piston, this period of the cylinder being referred to as the "stroke" period of the exhaust. During the stroke period of the exhaust, back pressure in the exhaust manifold may interfere markedly with the movement of the piston in the exhausting cylinder.

The operating cycle of the pump of the present invention generally follows that of my aforesaid copending application, in that it includes first a displacement period during which a stationary body of air or other gas is trapped at atmospheric pressure within the pump while being compressed and then pushed out of the pump by pressure balancing displacement action of a flowing stream of gas under pressure, such as hot engine exhaust gases introduced into the pump during the puff discharge period of a single engine cylinder exhaust cycle. This displacement period is followed by a scavenging period during which the puff exhaust gases which have been trapped in the pump during the displacement operation are discharged from the pump and the pump is scavenged with air, preferably by means of energy derived at least in part from the exhaust gases

which are discharged from the same engine cylinder during the stroke period of the cylinder exhaust cycle. Air thus compressed and discharged from the pump by an operation deriving energy from the hot gas pressure wave discharged from one cylinder of a multi-cylinder engine, may be delivered as supercharge air to another engine cylinder having a coinciding air intake period.

The present invention provides that any assembly of gas pumping units and multi-cylinder four cycle engine should include a sufficient number of engine exhaust manifolds to insure that the exhaust puff waves in any manifold shall not follow each other at intervals shorter than 180° engine crank angle travel. Some of the pumps of the present invention, however, are so designed that they can be operated on a cycle which is completed within a period encompassed by 120° crank angle travel of the engine supplying the exhaust gas for energizing the pump. Consequently a single pump may be operated by the pressure waves occurring alternately in two exhaust manifolds of a six cylinder engine. Where the pump is also connected at its air discharge end with one or more air intake manifolds of the engine for the purpose of supplying compressed air to each engine cylinder during the last part of its air intake period, provision is made for supplying air at atmospheric pressure to each engine cylinder during the first part of its air intake period. Consequently the pump need only be of small capacity, and can be assembled in closely spaced relation to the engine, with short air and gas transfer connections.

Another feature which differentiates the pump of the present invention from that of my aforesaid copending application is in the use of a light, and in some designs, flexible sheet metal diaphragm or floating piston mounted for reciprocal movement within the pump in response to small pressure differentials applied to opposite faces thereof, and having low mechanical resistance or inertia to movement in either direction. The diaphragm or piston serves to substantially inhibit contamination of the air or other gas undergoing compression by the engine exhaust gas; insures more nearly perfect adiabatic compression by reducing transfer of heat; and affords more positive and efficient scavenging. The diaphragm or piston, therefore, is an important contributing factor in reducing the size of the pump or compressor to a volumetric capacity not substantially exceeding that of an engine cylinder, and in per-

mitting efficient operation of the pump over a wide speed range.

With the foregoing and other objects and features in view, the invention consists in the improved method of and apparatus for pumping and compressing air or other gas as hereinafter described and more particularly defined by the accompanying claims.

The invention will be described more particularly by reference to the accompanying drawings, in which:

Fig. I is an assembly view, showing a single floating piston displacement pump operatively connected to exhaust and intake manifolds of a six-cylinder four cycle internal combustion engine; the pump, engine intake and exhaust manifolds, and gas transfer connections and valve chambers being shown in longitudinal section.

Fig. II is a pressure-time chart showing in full and dotted lines, respectively, the gas pressure waves which can be built up in two engine exhaust manifolds of a six-cylinder four cycle engine over a period of $\frac{1}{2}$ engine cycle or one revolution.

Figs. III, IV, V and VI are cross-sectional views of the gas and air transfer control valves taken respectively along the lines III—III, IV—IV, V—V and VI—VI of Fig. I.

Fig. VII is another diagrammatic assembly view showing a single floating diaphragm displacement pump and the intake and exhaust manifolds of a six-cylinder, four cycle engine operatively connected by gas and air transfer connections and transfer valves (parts being shown in longitudinal section).

Fig. VIII is a view in vertical section of a modified form of displacement pump equipped with a flexible diaphragm having its ends anchored to the pump housing.

Fig. IX is a plan view of the pump which is illustrated in Fig. VIII, taken on the line IX—IX of Fig. VIII.

Fig. X is an assembly view of two floating piston displacement pumps arranged in tandem and communicably connected respectively to two engine exhaust manifolds, together with gas and air transfer valves and connections adapting the pumps for engine supercharging, parts being shown in longitudinal section.

Fig. XI is a cross-sectional view of the apparatus of Fig. X, taken on the line XI—XI of Fig. X.

Fig. XII is a cross-sectional view through one of the air transfer valves of Fig. X, taken along the line XII—XII of Fig. X.

Figs. XIII, XIV and XV are cross-sectional views of the gas and air transfer valves, taken respectively along the lines XIII—XIII, XIV—XIV, and XV—XV of Fig. X.

Fig. XVI is a diagrammatic assembly view, chiefly in longitudinal section, showing a pair of floating piston pumps mounted in tandem and each communicably connected to individual engine exhaust and intake manifolds, together with gas and air transfer connections and control valves.

Fig. XVII is a perspective view of a modified arrangement of two gas displacement pumps having rectangular pistons mounted on a common oscillating shaft.

Fig. XVIII illustrates schematically an arrangement of two displacement pumps in tandem with connections to separate engine exhaust manifolds, no hot gas transfer valves being provided.

In the apparatus assemblies which are illustrated in Figs. I, VII, X, XI, XVI and XVIII of the drawings, one or more displacement pumps 20 are arranged for the compression and pumping of air by means of energy supplied thereto from the hot exhaust gases discharged under low superatmospheric pressure from a six cylinder four cycle internal combustion engine 24. Within the pumps piston-like floating diaphragms 22 are mounted to reciprocate with small clearance for the purpose of preventing substantial contact or intermixing between the compressing gas (exhaust gas) and the air or other gas being compressed, thereby insuring efficient adiabatic compression. The air which is compressed in the pump by a pressure balancing operation, is illustrated as being utilized for supercharging the engine cylinders. However, as previously indicated, the invention is not limited to the use of engine exhaust gases as the energizing medium for the pump, nor to the compression of air, nor to the use of such air for engine supercharging.

In Figs. I and VII, the cylinders of engine 24 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 passage 28 to housing 29 of a hot gas transfer control valve 30; while the exhaust ports of cylinders 4, 5 and 6 have been shown as connected through exhaust manifold 32 and passage 34 to housing 35 of a hot gas transfer control valve 36. Likewise, the intake ports of cylinders 1, 2 and 3 have been shown as connected through an intake manifold 38 and carburetor 40 to housing 41 of an air transfer control valve 42; while the intake ports of cylinders 4, 5 and 6 have been shown as connected through an intake manifold 44 and a carburetor 45 to housing 47 of an air transfer control valve 48. Hot valve housings 29 and 35 are in turn connected to a hot gas intake and exhaust port 50 of pump 20 by a forked transfer conduit 52; and cold valve housings 41 and 47 are in turn connected to an air intake and exhaust port 54 of pump 20 by a forked air transfer conduit 56. Concentric gas ejector nozzles 58 and 60 are connected, respectively, to the housings 29 and 35 of the hot gas transfer control valves, and afford the means by which gas may be discharged from either of the exhaust manifolds or from the pump to atmosphere by way of a Venturi throat 62 and muffler 64. An atmospheric air intake filter 66 is connected to housings 41 and 47 of the air transfer control valves in position to deliver air at atmospheric pressure to the pump and to either of the air intake manifolds and carburetors.

The full line pressure time curve of Fig. II shows the successive steep pressure waves built up in an exhaust manifold (such as manifold 26) by the exhaust gas discharges from two cylinders (for example cylinders 1 and 3) over one engine revolution. The exhaust of cylinder 1 begins about 45° crank angle before bottom dead center of crank 1, producing a strong puff wave which builds up a peak and then subsides within a period of about 100° crank angle, and is followed by a smooth weak stroke exhaust extending over about 140° crank angle. The dotted line pressure time curve of Fig. II shows the successive pressure waves built up in another exhaust manifold (such as manifold 32) by the exhaust gas discharged from other engine cylinders (for example cylinders 4 and 5) at pe-

riods shifted in phase against the waves produced by gas discharges from cylinders 1 and 3 by half of an exhaust period or by 120° crank angle firing intervals. The puff discharge wave of cylinder 1 occurs simultaneously with the stroke exhaust period of cylinder 4, and the puff discharge wave developed in manifold 32 by cylinder 5 occurs simultaneously with the stroke exhaust wave of cylinder 1 in manifold 26. The pressure waves as portrayed in Fig. II occur in an exhaust piping system which is continuously open to atmospheric discharge. When, however, discharge of the exhaust gases to atmosphere is temporarily blocked over the length of a cylinder puff discharge period, the pressure peak of the puff wave may be forced up higher and may be maintained over a longer period. The subsiding side of the puff wave has a slope and shape which depends on the rapidity with which the exhaust piping system is reopened to free atmospheric discharge.

In the single floating piston type displacement pump-engine assemblies which are illustrated in Figs. I and VII, the puff discharge waves which are produced successively by all six cylinders of engine 24, operating with crank angle spacings of 120°, are all put to work within the same pump space in rapid succession, which means that the operating cycle of the pump must be completed within a time period corresponding to a 120° crank angle movement of the engine.

The rotary gas and air transfer valves, arranged respectively between the pump and the engine exhaust manifolds and between the pump and the engine intake system, are activated from the engine crank shaft in the manner illustrated by Fig. VII, the drive being taken for example by chain from the engine shaft to the shaft 72 on which valves 30 and 36 are mounted and from shaft 72 to shaft 74 to which the air pressure valves 42 and 48 are keyed. The valve actuating and timing mechanism is arranged for operating the respective valves at rates proportional to the intervals between pressure peaks at the pulsating gas source of energy for operating the pump, namely in the engine exhaust manifolds.

In the operation of all six cylinder four cycle internal combustion engines, the cylinders each fire once during every two engine revolutions, and the cylinders operate on cycles with a crank angle spacing of 120°. Thus, while cylinder 1 is starting its gas exhaust, cylinder 6 is finishing its air intake; and while cylinder 4 is starting its gas exhaust, cylinder 3 is finishing its air intake; and while cylinder 5 is starting its gas exhaust, cylinder 2 is finishing its air intake. In other words, the assemblies of displacement pumps and engine exhaust manifolds and intake manifolds as illustrated in Figs. I, VII, X and XVI are designed to pair the cylinders of the multi-cylinder internal combustion engine when utilizing energy supplied to the pump by the engine exhaust waves for supercharging the engine. With the engine cylinders thus paired, energy carried by the exhaust gas discharge from one cylinder of a pair can be utilized for compressing air and transferring such air as supercharge air into the other paired cylinder during the last portion of its air intake period. During the first part of the air intake period of each cylinder, air can be supplied to the cylinder 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°.

In the operation of the engine-displacement pump assemblies of Figs. I and VII three cylinder discharge puff waves are supplied to the pump from the engine discharge manifold system during each engine crank shaft revolution. These gas discharge waves are designed to produce by means of the pump three similar air compression waves in the engine intake manifold system. The rising side of each exhaust puff wave measures the period during which the puff exhaust gas surges into the pump against the air, though separated from it by the diaphragm or floating piston, and during this period air is compressed in the pump and discharged from the pump to a transfer conduit. The receding side of the puff wave represents the period during which exhaust gases are released from the pump to atmosphere and the period during which air rebounds from the transfer conduit into the pump to fill it preliminary to a new operating cycle. Thus one pump cycle may be said to be completed during the period spanned between the two points *a*—*c* in the diagram of Fig. II.

The exhaust gas distributing valves 30 and 36 must operate on cycles which correspond with those of the pump, but also on cycles which include the additional phase of passing the stroke exhaust from each cylinder directly to the atmospheric discharge system 62 and 64 during the second half of each engine cylinder exhaust period. Likewise, each of the air valves 42 and 48 must complete its cycle during the period of the pump cycle, but has to accomplish the additional duty of supplying atmospheric air to the intake manifolds during the first part of the air intake period of each engine cylinder. The hot gas and cold gas transfer valves may be rotated at a speed $1\frac{1}{2}$ times the speed of the engine crank shaft, or at a straight fraction of such speed, for example with a speed $\frac{3}{4}$ crank shaft speed in the case where each of the valves is provided with two opposite sets of ports, in place of the single ports possessed by the valves illustrated in Figs. I and VII. By providing the valves with three sets of ports the speed of the valves may be reduced to $\frac{1}{2}$ the speed of the crank shaft. With a single pump assembly the number of strokes of the pump piston is always three times the number of revolutions of the crank shaft.

As shown in Figs. I, III and IV, each of the valves 30 and 36 is a rotary tubular valve having a bore of annular cross section which opens at one end into the valve chamber and engine manifold connected therewith, and which is closed at the other end by a common cylindrical hub joining both valves to shaft 72. Each of the valves 30, 36 has a single lateral port 25, 39 (Figs. III, IV) extending the full length of the valve wall and having a width subtending a cylinder arc of approximately 120°. Each of the air pressure valves 42 and 48 is a rotary cylinder segment subtending an arc of approximately 120° (Figs. V, VI).

Each of the hot gas transfer valves 30 and 36 performs three functions during one revolution.

During the first part of a cycle of pump 20, one of the valves rotates to a position permitting passage of engine puff exhaust gases under superatmospheric pressure into the pump from one exhaust manifold. During the second part of the pump cycle, the valve rotates further to open the passage whereby puff exhaust gases trapped in the pump, exhaust manifold, and exhausting cylinder, are released to atmosphere. Also during this last part of the pump cycle and for some time after the pump cycle is completed, the valve must cut off any further transfer of exhaust gases to the pump and pass stroke exhaust gases from the same engine manifold directly through the engine muffler system to the outside atmosphere. In doing so the pump is disconnected from this same exhaust manifold and enabled to perform another pumping cycle in connection with another branch exhaust manifold.

Similarly, one of the air transfer valves 42 and 43 must operate during the first air displacement and compression period of each pump cycle to transfer compressed air from the pump space into the proper engine intake manifold. This period of communication between the pump and the intake manifold extends over all of the displacement and compression period of the pump cycle and over a part of the air rebound period. After completion of the air rebound period, the air transfer valve must operate to admit scavenging air into the pump from atmosphere through the air filter 56. Simultaneously, with this scavenging air transfer period, the air transfer valve must also operate to pass atmospheric air directly to one of the engine intake manifolds. During the period when the one air transfer valve is in position to pass compressed air from the pump to one engine intake manifold, the other air transfer valve must be closed to cut off additional parasitic spaces or escapes open to the outside and to prevent transfer of compressed air into the other engine intake manifold. It will be noted that the period in which intake of scavenging air to the pump takes place coincides for a short time with the period in which intake of atmospheric air takes place to another of the cylinders beginning its intake period. Both of these atmospheric air intakes may be served by the same air transfer valve, or in part by

To some extent the valve timings of a particular pump design may differ from those illustrated. However, the valve timings must always be such as to avoid upsetting interference between the different pressure waves by which the pump operates. Special attention must also be given to designing the engine discharge system so as to prevent the building up of a back pressure in one of the two engine exhaust manifolds particularly during periods when puff waves are being transferred from the other manifold into the pump space. The concentric discharge nozzles 58 and 60 and the expanding Venturi throat 62 have been provided for the specific purpose of promoting rapid and powerful scavenging of the pump system while avoiding development of back pressure opposing the stroke exhausts.

The shafts 72 and 74 which, respectively, actuate the hot and cold gas transfer valves, are supported within the valve chambers by ball bearings 76. The bearings supporting the shaft 72 for the hot gas transfer valves may be protected by water jackets 78 against excessive heat. Also, the hot valve shaft bearings may be pro-

tected against gas leakage by labyrinth gaskets 80 (Fig. I). The bearings for supporting the cold gas transfer valve shaft 74 may be protected against gas leakage by the usual type of bushings 79 with oil sealing, which has been found to provide sufficient tightness for cold gas pressures never fluctuating between positive and negative pressure maxima of more than a few pounds per square inch.

Each of the hot gas and cold air transfer valves is shown in Figs. I, III, IV, V and VI, in the position which it assumes just prior to termination of the displacement compression operation period *a-b* of Fig. II. During this period the hot puff exhaust gases from cylinder 1 and manifold 26 are being transferred past valve 30 into pump 20. Simultaneously the valve 36 is in a position to transfer stroke exhaust gases from cylinder 4 and manifold 32 directly to atmosphere through discharge nozzle 60 and funnel 64. At this same time compressed air is being transferred from the pump directly through valve 48, carburetor 46, and air intake manifold 44, into cylinder 6. Also during this period, air transfer valve 42 is in position for passing atmospheric air through carburetor 40 and manifold 38 into cylinder 2.

The air compression chamber of pump 20 as viewed in Fig. I, always lies to the right of piston 22, and is of annular cross section surrounding the stem of the piston. The volumetric displacement of the pump 20, exclusive of the cubic displacement of the piston, which is relatively small, is never appreciably more than sufficient to handle the volume of hot gas which is discharged from a single engine cylinder during the first puff discharge period, and to compress only the air with which a cylinder is supercharged at the end of its air intake period.

The atmospheric air intake ports under the control of air transfer valves 42 and 48 have been illustrated as by-passed, respectively, with a pair of air by-pass conduits 84 and 86. Valves 88 and 90 are respectively mounted in conduits 84 and 86; and another valve 92 is mounted in the conduit 56 connecting the cold gas transfer valve housing with the pump. Valves 88, 90 and 92 afford the means whereby engine 24 can be switched from normal operation to supercharging operation, or back to normal operation, at will. During supercharging, valves 88 and 90 are closed and valve 92 is opened, as shown in Fig. I. During normal operation of the engine without supercharging, valve 92 is closed and valves 88 and 90 are opened for passing atmospheric air directly and continuously from the air cleaner to the corresponding carburetors and engine intake manifolds.

The design of the pump-engine assembly illustrated in Fig. I is such that the displacement face of the pump piston is always exposed to the impacts of the puff discharge waves from the engine cylinders. The whole system responds more rapidly to switching from atmospheric air intake to supercharging when the hot gas side of the pump is continuously subjected to pulsating pressure. If, however, the operator desires to shield the pump during normal operation (without supercharging) against the hot puff exhaust waves, a by-pass 94 may be provided leading from each of the engine exhaust manifolds directly into the muffler line (indicated in dotted lines in Fig. I), and special valves 96 may be provided which on opening by-pass the exhaust puff waves directly into the engine muffler.

The diaphragm of the pump illustrated in Fig.

VII has been shown as slidably journaled on a post 45 which is mounted on the main axis of the pump with its end supported by the end walls of the pump. A preferred design of the single diaphragm piston pump, however, has been shown in Fig. I, in which the piston 22 is attached rigidly at its center to one end of a stem 23. The other end of stem 23 carries a pin on which adjacent ends of two links 68 are pivotally hinged. The opposite ends of links 68 in turn carry pins on which are respectively hinged two oscillating rods 69. Rods 69 are in turn pivotally mounted on brackets attached to the casing of the pump. The oscillating ends of rods 69, to which links 68 are respectively connected, are connected together by a retractile spring 70. An oil-sealed stuffing box 21 is mounted in an aperture in the end plate of the pump within which stem 23 is reciprocally journaled.

During the compression period of each pump cycle all of the air which is trapped between the pump diaphragm and the air intake port of the intaking engine cylinder is subjected to compression by the full discharge gas wave. During the second half of the pump cycle, when the full discharge wave is subsiding by reexpansion from the pump, the forces acting on the diaphragm to move it in the opposite direction include the suction developed in the Venturi exhaust orifice 62 by the stroke exhaust from one engine cylinder, and also the expansion force of the compressed air still trapped between the engine cylinder intake valve which has just closed and the pump diaphragm. The pressure of the air thus trapped is rapidly reduced to atmospheric pressure by the complete expansion of the engine exhaust gases on the other side of the diaphragm, so that there is a tendency for the diaphragm movement to terminate somewhere in mid-stroke, without the aid of a device such as the spring 70. The air pressure on the air side of the diaphragm is rapidly reduced for another additional reason, and that is that during the air rebound period of the diaphragm of the pumps shown in Figs. I and VII, a second cylinder of the engine is beginning its air intake period.

The spring 70 of the piston return mechanism illustrated in Fig. I has been designed as an energy absorbing element which converts to mechanical energy a small part of the energy imparted to the piston during the displacement period of the pump cycle, by building up tension on the spring. The spring need only be strong enough to absorb a very small proportion of the energy imparted to the piston. The construction of the spring mechanism is such that the farther the piston moves toward the right (as viewed in Fig. I) the smaller the amount of opposition to movement of the piston. In other words, the spring has no effect whatsoever on movement of the piston at the time that the piston has reached the position shown in Fig. I, that is, when the back pressure of the compressed air is greatest. However, the spring exerts its full force against the piston when the piston is near the end point of its travel toward the extreme left hand position within the pump. The only forces bearing on the piston in the position shown in Fig. I are the balancing gas pressures on opposite sides thereof. On release of the trapped exhaust gases lying to the left of the piston at the end of the displacement compression period, the piston will start to move backward toward the left hand side of its path of travel, and the tension on the spring then comes into action to draw the oscillating

ends of the links 69 toward each other, forcing the piston toward its extreme left hand position and thereby producing air scavenging of the pump. During full speed operation of the piston, in a pump-engine assembly such as illustrated in Fig. I, the piston of the pump will not travel the full length of its stroke, the length of the path of travel which it does traverse depending on the exact dynamics of the particular pump design and on the speed of the engine and pump.

The floating diaphragms of the pumps shown in Figs. I, VII, X and XVI are circular metal discs, while the diaphragms of the pumps illustrated in Figs. VIII, IX and XVII have a rectangular shape. In all cases the pump diaphragms are dimensioned to reciprocate within the pump housings with a definite small clearance between the walls of the housing and the edges of the diaphragms. Very little leakage of gas occurs past the diaphragm through such small clearance space during the operation of the pump, since the gas pressure differential between opposite faces of the diaphragm is always very small. In fact such pressure differential is only sufficient to overcome any inertia resistance of the floating diaphragm, which is kept as small as possible. The displacement pumps are preferably designed with a large cross sectional area in comparison with the diaphragm stroke, for the purpose of reducing the diaphragm speed and the inertia forces operating on the diaphragm to a minimum. This construction also has the effect of magnifying any motive force impressed on the diaphragm and giving instantaneous response of the diaphragm to any gas pressure differential impressed thereon.

In the pump modification which is illustrated in Figs. VIII and IX, the sheet metal diaphragm 77 has its ends rigidly attached to the shell of the pump and has sufficient elasticity to provide for a self-flexing operation between the full line position and the dotted line position. The method of suspending the ends of the diaphragm with respect to the pump housing of Figs. VIII and IX must be such as to allow for free play of the elastic self-retroactive properties of the diaphragm at any instant of the pump operating cycle.

In the double pump assembly which is illustrated in Fig. XVII, the floating pistons of both pumps are flaps 97 suspended on a common oscillatory shaft 98. In this design, as in all double pump designs, the air and energizing gas intake and exhaust ports of the two pumps are arranged at opposite sides of the respective pump pistons in order that each pump, when operating on the compression period of its cycle, will effect automatic scavenging of the other pump during the scavenging period of its cycle. In the tandem pumps which are illustrated in Figs. X to XVI and XVIII, the energizing gas intake ports may be located either at adjacent ends of the two pumps or at opposite side covers of the two pumps.

In the apparatus assemblies which are illustrated in Figs. X, XI, XVI and XVIII, two displacement pumps are mounted in tandem with their floating pistons connected by a common stem and interlocked for reciprocation in unison. The two pumps are designed for operation on alternate cycles, so that the displacement period in the operating cycle of one pump coincides with the air scavenging period in the cycle of the other pump, the interlocked piston functioning

to make both the displacement and the scavenging entirely positive. Each pump is operatively connected to only one of the two exhaust manifolds of the six cylinder engine, so that each pump performs only half the number of cycles that are required of the single pump in the assemblies illustrated by Figs. I and VII. In other words, each pump of the double pump assemblies illustrated performs a number of cycles corresponding to $1\frac{1}{2}$ engine crank shaft speed (as compared to a pump cycle speed three times crank shaft speed for the single pump assembly of Figs. I and VII), and the gas and air transfer valves, when equipped with single-phased ports, also revolve at $1\frac{1}{2}$ engine crank shaft speed.

In the assemblies illustrated by Figs. X, XI and XVI and XVIII, exhaust manifold 26 (for cylinders 1, 2 and 3) is in open and constant communication with one of the pumps 20 by means of a hot gas transfer conduit 51; while the other exhaust manifold 32 (for cylinders 4, 5 and 6) is in open and constant communication with the other pump 20, by means of a separate hot gas transfer conduit 53. The floating pistons 22 for the two pumps are preferably light alloy metal discs rigidly mounted on a common stem 67 which is, in turn, reciprocally supported by lubricated bushings and stuffing boxes 21 which are centrally mounted in the end plates of each pump housing and are always cooled by air being pumped. The working chambers of the two pumps shown in Figs. X, XI and XVIII are disposed in tandem within a single housing, on opposite sides of an inclined partition 100. In Fig. XVI, the two pumps are arranged in tandem, each pump within its individual housing.

Manifold 26 is ported out (Figs. X, XI and XVI) at 27 into the housing of a cylindrical gas discharge valve 31; and manifold 32 is similarly ported out at 33 into the housing of a cylindrical gas discharge valve 37. Transfer conduits 51 and 53 are ported out into the respective pumps with which they communicate at adjacent ends of the two pumps (Fig. XVI); or, in the case of Figs. X, XI and XVIII, at symmetrical points located at opposite sides of partition 100.

One engine intake manifold 38 (for cylinders 1, 2 and 3) is shown as connected through a carbureter 40 to the housing of a single-ported tubular air transfer control valve 43; while the other intake manifold 44 (for cylinders 4, 5 and 6) is connected through a carbureter 46 to the housing of an air transfer control valve 49. The housings of valves 43 and 49 are respectively connected to the respective displacement pumps 20 by air transfer conduits 55 and 57. It will be noted that hot gas transfer conduits 51 and 53 are ported out into the respective pumps 20, with which they communicate, at adjacent sides of the pump pistons 22; and that the air or cold gas transfer conduits 55 and 57 are ported out into the respective pumps at the remote sides of the corresponding pump pistons.

Concentric gas ejector nozzles 60 and 58 are connected respectively to the housing of hot gas discharge control valves 31 and 37 and afford the means by which gas may be discharged from the exhaust manifolds, and the pumps connected therewith, to atmosphere by way of the Venturi throat 62 and muffler 64.

Each of the air transfer conduits 55 and 57 is ported out into the common housing for a pair of air transfer control valves 63 and 65, which are single-ported tubular valves. Valves 63 and 65

are mounted to respectively control transfer of atmospheric air for scavenging the pumps from an air intake filter 66 to one of the conduits 57 and 55, while blocking transfer of atmospheric air to the other transfer conduit.

In the apparatus of Figs. X, XI, XII, XIV and XV, conduits 55 and 57 are forked. The main forks of the respective conduits lead directly from the pumps to the corresponding atmospheric air control valves 63, 65. The other forks 82 (of conduit 55) and 99 (of conduit 57) branch out of the main fork at points near the pump and lead up to the ports of transfer valve 43, 49.

A pair of air by-pass chambers 81 and 83 has been illustrated in Figs. X and XVI. These by-pass chambers are ported out at each side of air filter 66. Communication between chamber 81 and intake manifold 38 is under the control of valve 43, while communication between chamber 83 and the intake manifold 44 is under the control of valve 49. For supplying atmospheric air to the intake manifolds of the engine throughout the entire air intake period, in case pumps 20 are not operatively connected to deliver supercharge air during part of the intake period, by-pass pipes 85 and 87 are provided, respectively connecting chambers 81 and 83 to the intake manifolds 38 and 44, bypassing valves 43 and 49. A butterfly valve 89 is mounted in pipe 85, and a similar valve 91 is mounted in pipe 87. A valve 93 is mounted in fork 82 of the air transfer conduit 55 (Fig. XV), and a similar valve 95 is mounted in fork 99 of air transfer conduit 57 (Figs. X, XII). Valves 93 and 95 when closed block transfer of compressed air from the pump to the engine intake manifolds. Valves 89, 91, 93 and 95 afford the means whereby the engine can be switched from normal operation to supercharging operation, or back to normal operation, at will. During supercharging, valves 89 and 91 are closed, and valves 93 and 95 are open. During normal operation of the engine without supercharging, valves 93 and 95 are closed and valves 89 and 91 are open.

Hot gas transfer valves 31 and 37 (Figs. X, XI and XVI) are mounted on a single drive shaft 71. Likewise, air or cold gas control valves 43, 49, 63 and 65 are all mounted on a single drive shaft 73. Shafts 71 and 73 are operatively connected for actuating all of the hot gas and air transfer control valves directly from the crank shaft of the internal combustion engine. The shaft 71 for the hot gas valves is supported within the valve chambers by ball bearings 75, and these ball bearings are protected by water jackets 78 against excessive heat (Fig. X). Also the hot valve shaft bearings are protected against gas leakage by labyrinth gaskets 80. The bearings for supporting the air or cold gas transfer valve shaft 73 are protected against gas leakage by the usual type of bushings 79 with oil sealing.

In the schematic assembly of two displacement pumps and six cylinder engine shown by Fig. XVIII, hot gas transfer control valves have been omitted, and in place thereof there have been substituted a pair of gas ejector nozzles 59 and 61 of predetermined restricted cross-section, which may be located in the same relative position as are the gas discharge nozzles 58 and 60 of the assemblies portrayed in Figs. X, XI and XVI. The modified assembly, which is illustrated in Fig. XVIII, has been particularly designed for operation at substantially constant speed. The cross sectional area of each of the discharge nozzles 59

and 61 is so chosen as to just handle the volume of gas discharged from an engine cylinder during the exhaust period without developing substantial back pressure during the stroke period of the exhaust. Consequently, because of the restricted area of these nozzles 59 and 61, they have a considerable blocking effect against the strong puff exhaust wave which exits from an engine cylinder during the first or puff period of the exhaust cycle. As a result of the partial blocking effected by nozzles 59 and 61, a supercharging pressure of moderate intensity is impressed on the piston of whichever one of the pumps is connected to the manifold receiving that particular puff discharge wave. The reduction in compression efficiency or intensity which is obtained by the design of Fig. XVIII, in comparison with the assemblies of Figs. X, XI and XVI, may in some cases be justified by the greater simplification of apparatus which result from the elimination of the hot gas transfer control valves.

Each of the hot gas and cold air transfer valves is shown in Figs. X to XV, inclusive, in the position which it assumes just prior to termination *b* of the displacement compression operation period *a-b* of Fig. II. During this period the hot puff exhaust gases from one of the engine cylinders, for example cylinder 1, are being transferred directly from manifold 25 into the pump 20 which is connected with that manifold. Simultaneously, valve 37 is in position to transfer stroke exhaust gases from cylinder 4 and manifold 32 directly to atmosphere through discharge nozzle 53 and funnel 62. At this same time, compressed air is being transferred from the chamber in the same pump at the opposite side of the piston directly past valve 49, carbureter 46 and air intake manifold 44, into cylinder 6. Also during this period, air transfer valves 63 and 65 are in position for passing atmospheric air from air cleaner 66 through transfer conduit 55 into the air chamber side of the second pump connected with manifold 32 during the scavenging period of the cycle of this second pump.

Positive scavenging of the displacement pump with a fresh charge of air during the last part of each pump cycle is assured by providing the pump piston with a spring fly-wheel construction such as shown in Fig. I, or by connecting the piston with the piston of another pump (Figs. X, XI, XVI, XVII and XVIII) in such a manner that the first piston is moved on a suction stroke by the second piston operating on its displacement stroke. The discharge nozzles 53 and 60 are disposed in concentric relation at the entrance of the Venturi funnel 62 to assist scavenging of the pumps by applying the suction aspiration effect of a jet of engine exhaust gases discharged directly from one engine exhaust manifold to atmosphere through one of said nozzles during the stroke exhaust period of an engine cylinder connected to said manifold for promoting development of suction in the pump connected to the other exhaust nozzle during the scavenging period of the pump cycle. Any interference to pump scavenging which may be offered by air cleaner 66 may be compensated by mounting a fan 102 (Fig. XVI) at the entrance of the air cleaner to supply air thereto under slight pressure.

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

1. In compressing and pumping air, the steps comprising moving combustion gases under super-atmospheric pressure in a confined stream in pressure waves following each other at substan-

tially uniformly spaced rapidly repeated intervals, during an interval between successive wave peaks trapping a body of air in a compression vessel at substantially atmospheric pressure, during the period of the next wave peak introducing gas from said stream into one end of said vessel thereby compressing the air by a pressure balancing displacement operation, mechanically absorbing part of the energy carried into the vessel by said combustion gases, discharging the compressed air from the vessel while trapping the gas against escape therefrom and expanding the gas trapped therein to substantially atmospheric pressure, developing a partial vacuum by discharging gas from said stream directly to atmosphere in a high velocity expanding column during an interval between wave peaks immediately following the displacement compression step, and utilizing a partial vacuum and mechanically absorbed energy to scavenge the vessel with a fresh supply of air at atmospheric pressure before repeating the cycle.

2. In gas pumping apparatus, a pump chamber, a diaphragm mounted within said chamber for reciprocation therein in response to slight gas pressure differentials at opposite sides thereof, a source of fixed gas under pulsating superatmospheric pressure, means permitting discharge of gas directly from said source to atmosphere at a restricted rate, a conduit communicably connecting the pump at one side of said diaphragm with said gas source, a branched conduit communicably connecting the pump at the opposite side of the diaphragm both to atmosphere and to a compressed air chamber, valve mechanism mounted for controlling supply of atmospheric air to and delivery of compressed air from the pump through the branched conduit, and valve actuating and timing means for operating said valve mechanism at a rate proportional to the interval between pressure peaks at the gas source.

3. Apparatus as defined in claim 2, in which the means permitting discharge of gas from said source to atmosphere includes a valve, said valve being arranged for periodically interrupting communication between said source and atmosphere.

4. In gas pumping apparatus, a pumping vessel, a diaphragm mounted within said vessel for reciprocation therein in response to slight gas pressure differentials at opposite sides thereof, a source of fixed gas under pulsating superatmospheric pressure, a valve chamber having a valve mounted therein, a discharge nozzle leading off from said valve chamber to atmosphere, a conduit communicably connecting the pump at one side of said diaphragm with said valve chamber, a connection between the valve chamber and the gas source, said valve being arranged for periodically switching communication between the gas source and pump and between the pump and atmosphere, a branched conduit communicably connecting the pump at the opposite side of the diaphragm to atmosphere and to a compressed air chamber, and valve mechanism mounted to control alternate supply of atmospheric air to and delivery of compressed air from the pump through the branched conduit, together with valve actuating and timing means for operating the respective valves at a rate proportional to the interval between pressure peaks at the fixed gas source.

5. In gas pumping apparatus, a wall-enclosed pump housing of relatively large cross-section and short length, a diaphragm mounted transversely in the housing for reciprocation therein in response to slight gas pressure differentials at

opposite sides thereof, a source of fixed gas under pulsating superatmospheric pressure, a gas transfer connection communicably connecting said gas source and the interior of the pump housing at one end thereof, a waste gas discharge conduit leading off from the gas source to atmosphere, valve mechanism arranged to control discharge of gas from the source and from the pump to atmosphere through the waste gas discharge conduits, a forked air transfer connection ported out of the pump through its other end wall, and a pair of valves mounted to respectively control introduction of atmospheric air into and removal of compressed air from the pump through said air transfer connection.

6. A gas pump as defined in claim 5 in which the valve mechanism which is arranged to control discharge of gas from the source and from the pump to atmosphere is so positioned and arranged as to also control transfer of gas between the source and the pump.

7. Apparatus as defined in claim 5 together with a second source of fixed gas under pulsating pressure, branches of said gas transfer conduit and waste gas discharge conduit respectively connecting said second source with the pump and with atmosphere, and a pair of valves mounted and arranged to respectively control transfer of gas between each source and the pump and between the source and the pump and atmosphere.

8. Apparatus as defined in claim 5 in which the pump is cylindrical, and in which the diaphragm is a circular disc piston rigidly mounted on the end of a stem which is reciprocally journaled in an aperture in one end of the pump, together with a spring fly wheel mechanism connected to the piston stem in position tending to move the piston toward the gas transfer end of the pump during the air scavenging period of the pump cycle.

9. In compressing and pumping air the steps comprising, producing flow of air from atmosphere in a confined stream, periodically introducing air at substantially atmospheric pressure into contact with one side of a flexible diaphragm and trapping air thus introduced while increasing its pressure by movement of said diaphragm, thereafter releasing air thus compressed into said stream to increase the rate of flow in the direction of stream flow, causing energy-supplying gas to flow under superatmospheric pressure in a confined path in pulsating pressure waves, applying such gas pressure waves to the other side of said diaphragm to move the same and impart pressure energy therethrough to trapped air, mechanically absorbing and storing some of the energy thus imparted to the diaphragm, and between pressure waves expanding gas from said gas stream to atmosphere and utilizing mechanically stored energy to assist return of the diaphragm to its original position.

10. An air compressing and pumping operation comprising setting up flow of air from atmosphere in a confined stream, introducing air at substantially atmospheric pressure into contact with one side of a movable partition and trapping air thus introduced while increasing its pressure by movement of said partition, thereafter releasing air thus compressed into said stream to increase the rate of setting up flow of a second stream of gaseous products of combustion under superatmospheric pressure in a confined path, contacting pressure gas from said second stream with the other side of said partition thereby displacing the same and imparting

pressure energy therethrough to the trapped air, trapping the pressure gas in contact with the partition during the displacing and air release operations, and after each period of compressed air release expanding gas from said second stream to atmosphere and returning the partition to substantially its original position preliminary to a new cycle.

11. A displacement pump comprising a wall-enclosed housing of relatively large cross-section and short length, a diaphragm partition mounted transversely in the housing between the ends thereof for reciprocation therein in response to slight pressure differentials between opposite sides thereof, a source of energy supplying fluid under pulsating superatmospheric pressure, a continuously open restricted outlet from said source to atmosphere, a fluid transfer connection communicably connecting said fluid source and the interior of the pump housing at one end thereof, connections for introducing a second fluid to be pumped into, and for removing said fluid from, the other end of the housing, and valve mechanism arranged to respectively control introduction of fluid into, and removal from, the pump through said several connections.

12. In compressing and pumping air the steps comprising setting up flow of combustion gases under superatmospheric pressure in a confined stream in pressure waves following each other at substantially uniformly spaced rapidly repeated intervals, during an interval between successive wave peaks trapping a body of air at one side at a diaphragm partition at substantially atmospheric pressure, during the period of the next wave peak impressing said gas stream against the other side of said partition thereby moving the partition ahead of the advancing gas wave and compressing the air by pressure balancing displacement, mechanically absorbing part of the energy imparted to the partition by said combustion gases, discharging the compressed air ahead of the advancing partition, between wave peaks discharging gas from the gas side of the partition to atmosphere to release pressure on the partition, and utilizing mechanically absorbed energy to assist in returning the partition to its original position preparatory to a new cycle.

13. In compressing and pumping air the steps comprising, setting up flow of energy supplying gas in a confined stream from a source thereof under pulsating superatmospheric pressure, building up pressure waves in said stream following each other with a frequency corresponding to the frequency of the source pulsations, between wave peaks trapping air to be compressed at low pressure in contact with one side of a movable diaphragm, impressing the following gas pressure wave against the other side of the diaphragm thereby moving the diaphragm ahead of the advancing gas wave and compressing the air by pressure balancing displacement, discharging air thus compressed ahead of the moving diaphragm and between wave peaks discharging gas from the gas side of the diaphragm to atmosphere to thereby release pressure on the diaphragm, and returning the diaphragm to its original position preparatory to a new cycle.

14. The method which is defined in claim 13 in which the flow of energy supplying gas is directed to atmosphere through a back pressure building flow restriction.

15. The method of operation as defined in claim 13 in which the compressed air is discharged into a confined moving stream thereof

under lower average pressure thereby building up pressure waves in said air stream of the same frequency and substantially the same magnitude as the pressure waves in the gas stream.

16. In gas pumping apparatus, a wall enclosed 5 pump housing of relatively large cross section and short length, a diaphragm partition movably mounted transversely in the housing for reciprocation therein in response to slight gas pressure differentials at opposite sides thereof, a 10 source of fixed gas under pulsating superatmospheric pressure, a gas transfer connection communicably connecting said gas source and the interior of the pump housing at one end thereof, a waste gas discharge conduit communicably 15 connected with the pressure gas side of the hous-

ing, valves arranged to control transfer of gas between the source and the pump and between the pump and the waste gas discharge conduit, valved connections for delivering atmospheric air to, and for removing compressed air from, the pump at the other end thereof, and valve actuating and timing mechanism for operating the respective valves at rates proportional to the interval between pressure peaks at the pulsating pressure gas source.

17. Gas pumping apparatus as defined in claim 16 together with a second source of gas under pulsating pressure and a valve controlled connection between said second gas source and the pressure gas side of the housing.

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CERTIFICATE OF CORRECTION.

Patent No. 2,293,186.

August 18, 1942.

JOHANN J. WYDLER.

It is hereby certified that error appears in the printed specification of the above numbered patent requiring correction as follows: Page 8, first column, line 70, before "setting" insert the word and comma--flow,--; and that the said Letters Patent should be read with this correction therein that the same may conform to the record of the case in the Patent Office.

Signed and sealed this 3rd day of November, A. D. 1942.

(Seal)

Henry Van Arsdale,
Acting Commissioner of Patents.