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Rogers, Sr.

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(54) **METHOD AND APPARATUS FOR OPERATING AN ENGINE ON COMPRESSED GAS**

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F02B 65/00	(2006.01)
F01L 25/08	(2006.01)
F02B 69/04	(2006.01)
F02B 63/06	(2006.01)
F01B 17/02	(2006.01)

(52) **U.S. Cl.**

CPC ***F01B 29/04*** (2013.01); ***F01L 25/08*** (2013.01); ***F02B 65/00*** (2013.01); *F01B 17/02* (2013.01); *F02B 63/06* (2013.01); *F02B 69/04* (2013.01)

(58) **Field of Classification Search**

CPC F01B 29/04; F01B 17/02; F01L 25/08; F02B 65/00; F02B 63/06; F02B 69/04

USPC 60/370, 371, 407; 91/187, 275; 415/143, 415/122.1, 140, 182.1, 199.6

See application file for complete search history.

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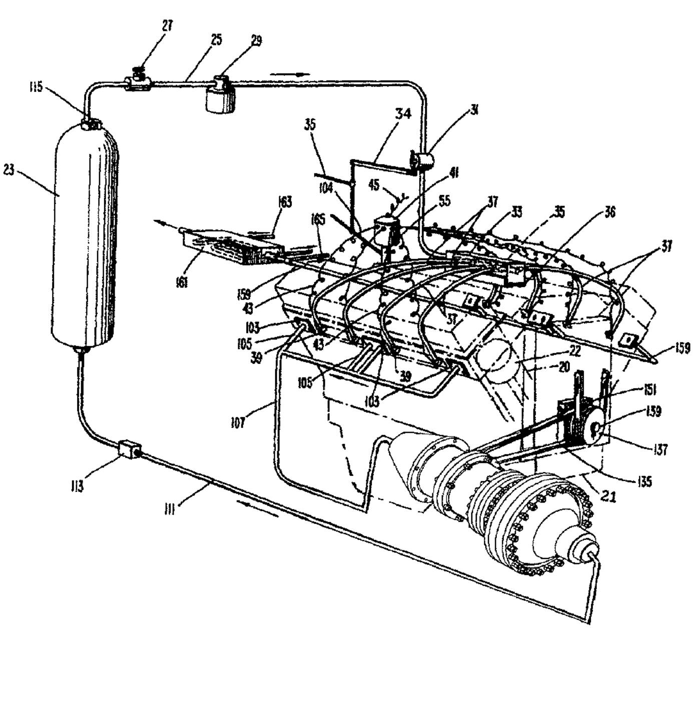
(57) **ABSTRACT**

The present invention relates to a method and apparatus for operating an engine having a cylinder and a piston reciprocable therein on compressed gas. The apparatus comprises a source of compressed gas connected to a distributor which distributes the compressed gas to the cylinder. A valve is provided to selectively admit compressed gas to the cylinder when the piston is in an approximately top dead center position. Compressed gas is provided by a compressor comprising a axial compressor, a deflector blade which is located downstream of the axial compressor, a radial compressor which is located downstream of the deflector blade and a housing with a which encloses the axial compressor, deflector blade, and radial compressor.

7 Claims, 12 Drawing Sheets



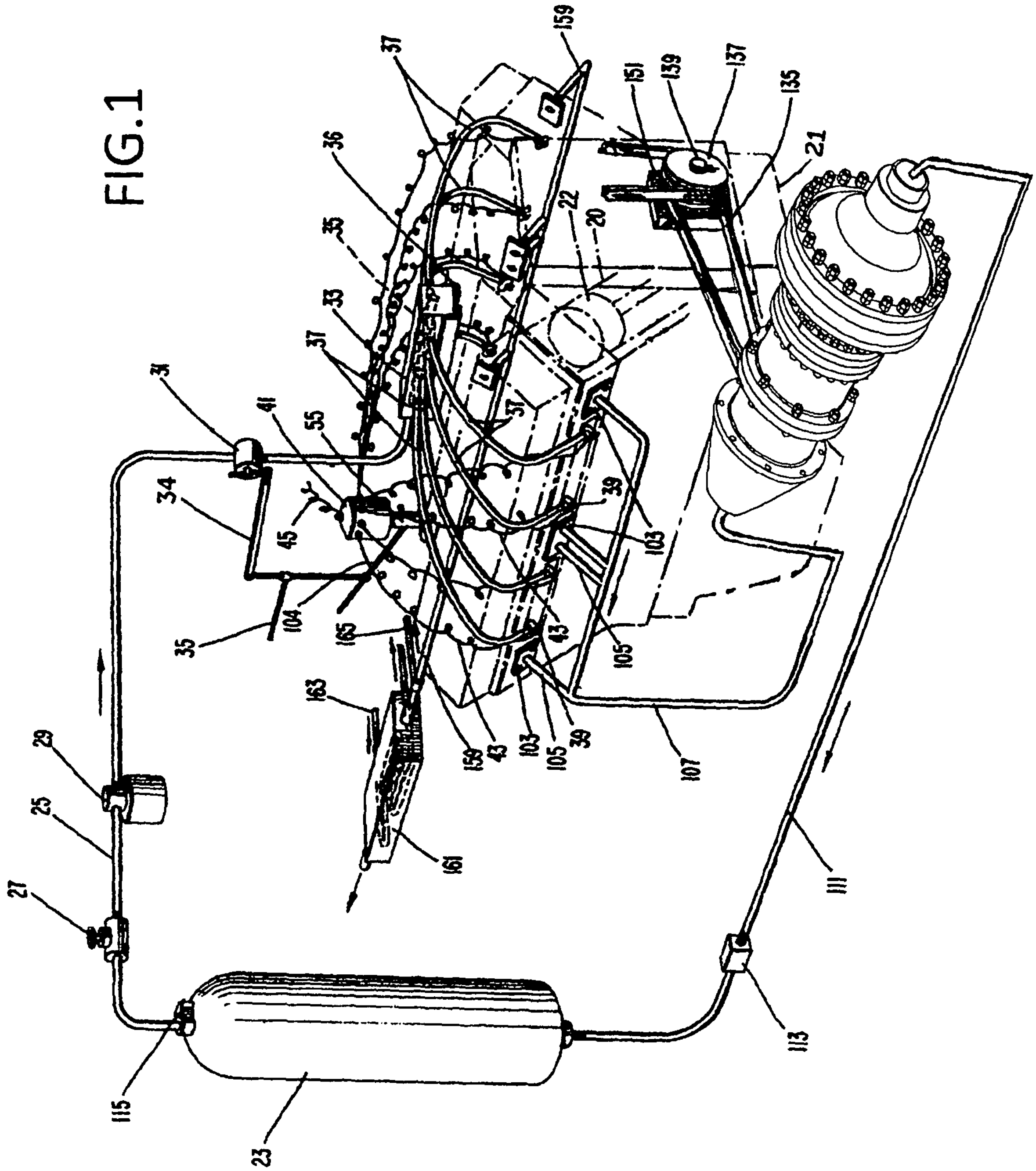

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FIG.1

FIG.2

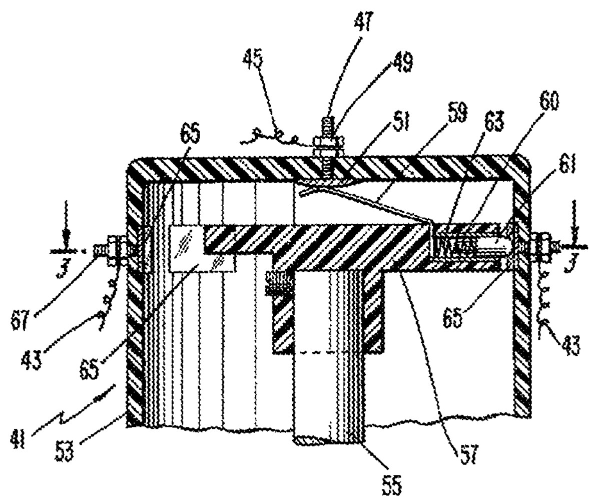
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FIG.3

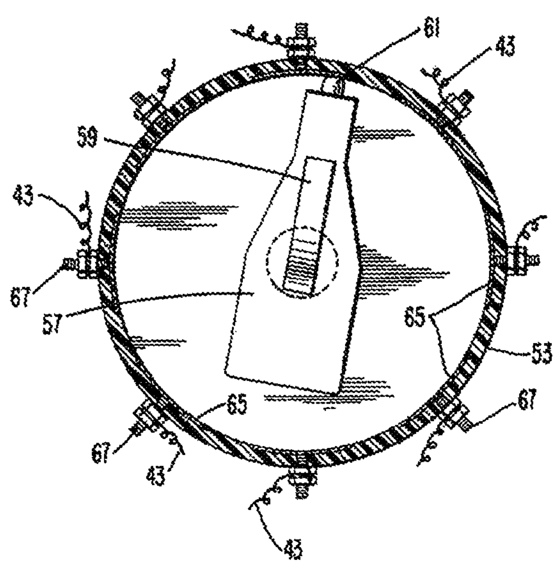
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FIG.4

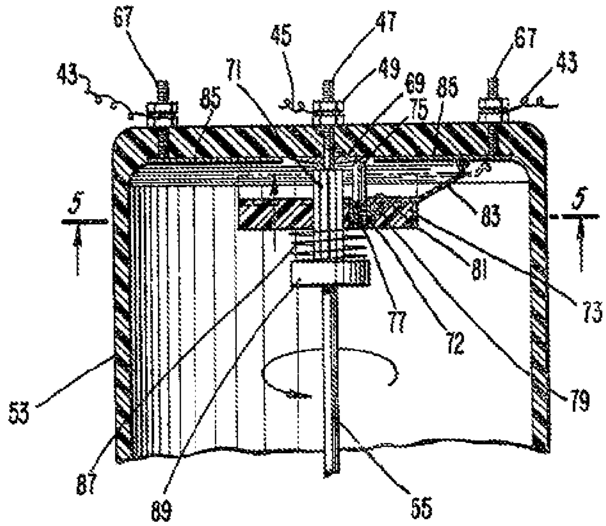
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FIG.5

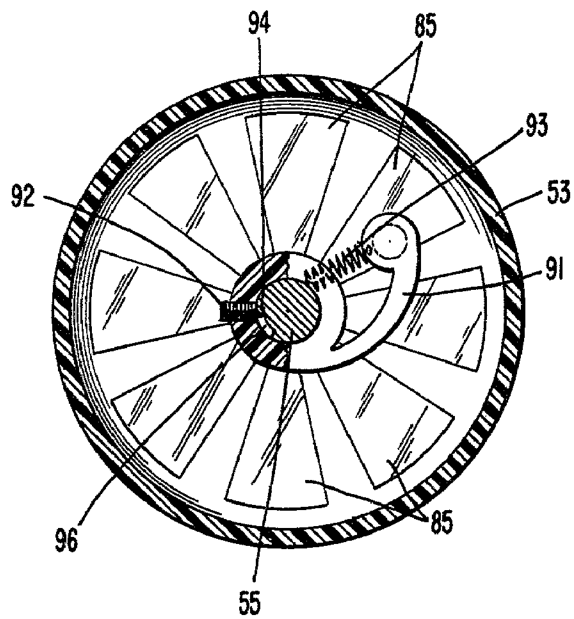
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FIG.6

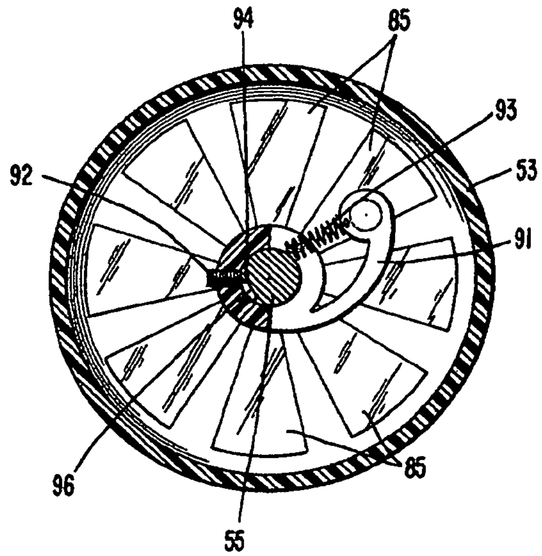
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FIG.7

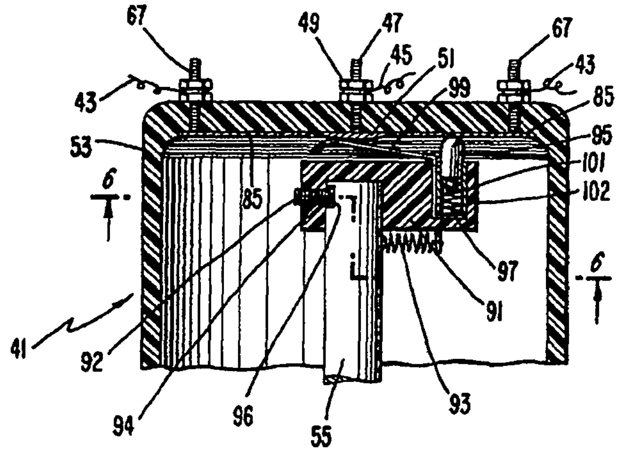
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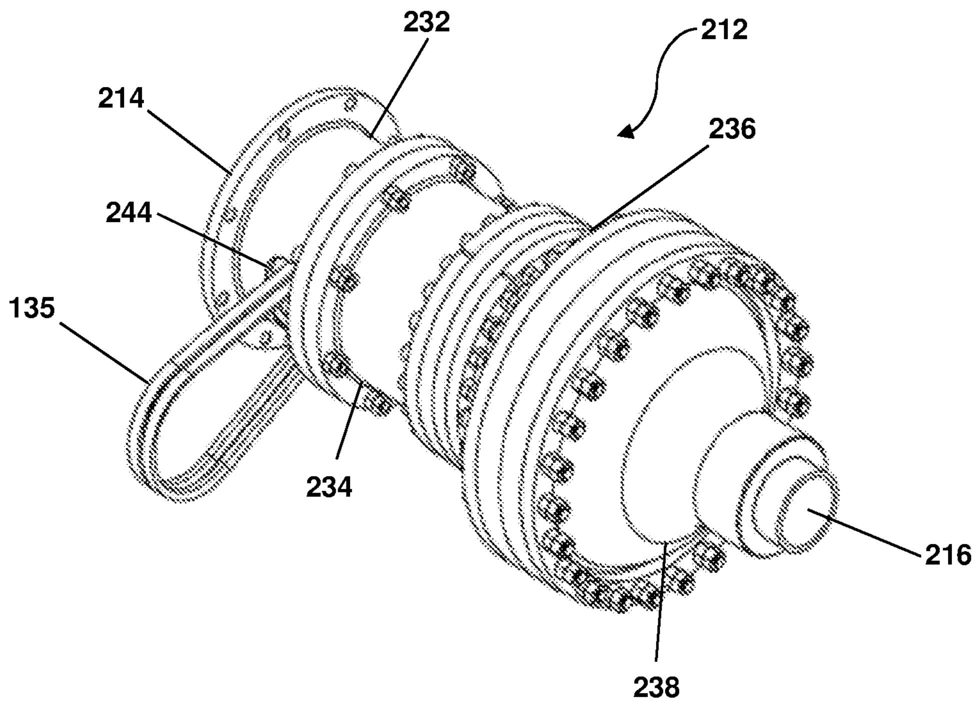
FIG.8



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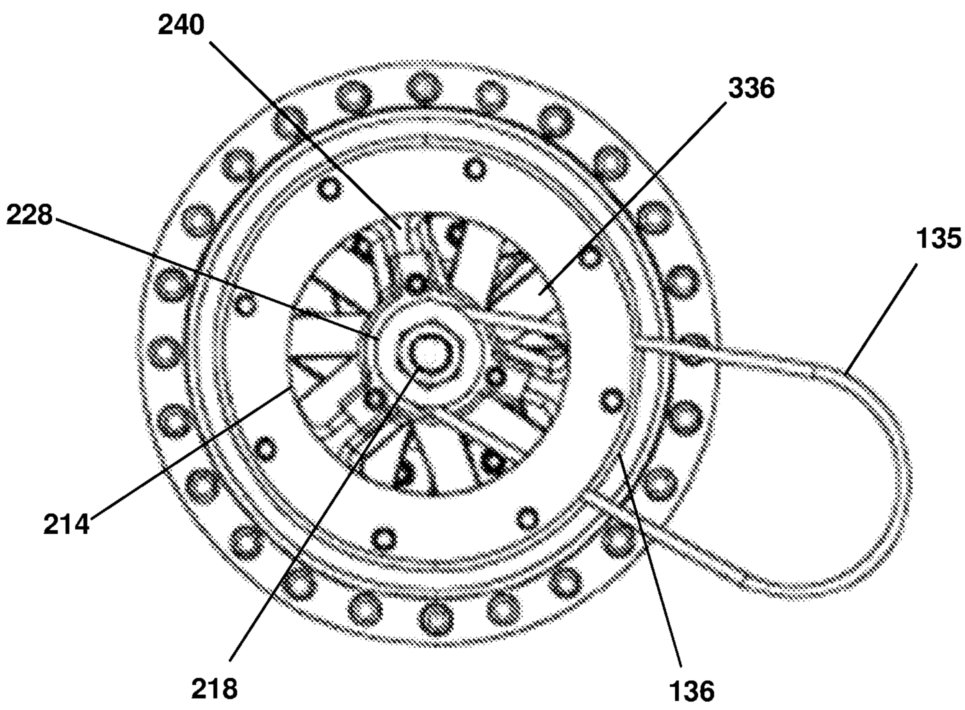

FIG.10

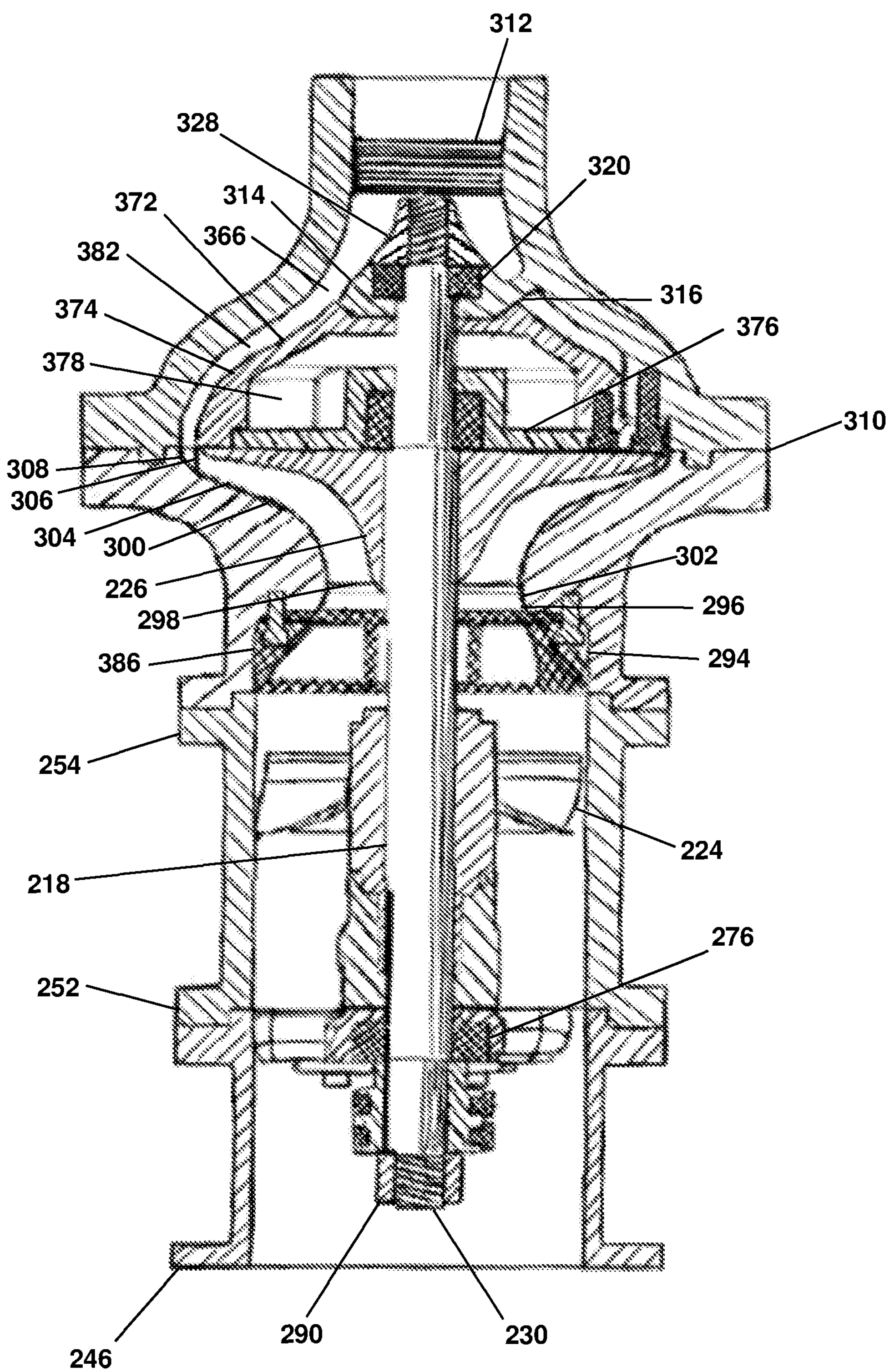
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FIG. 11

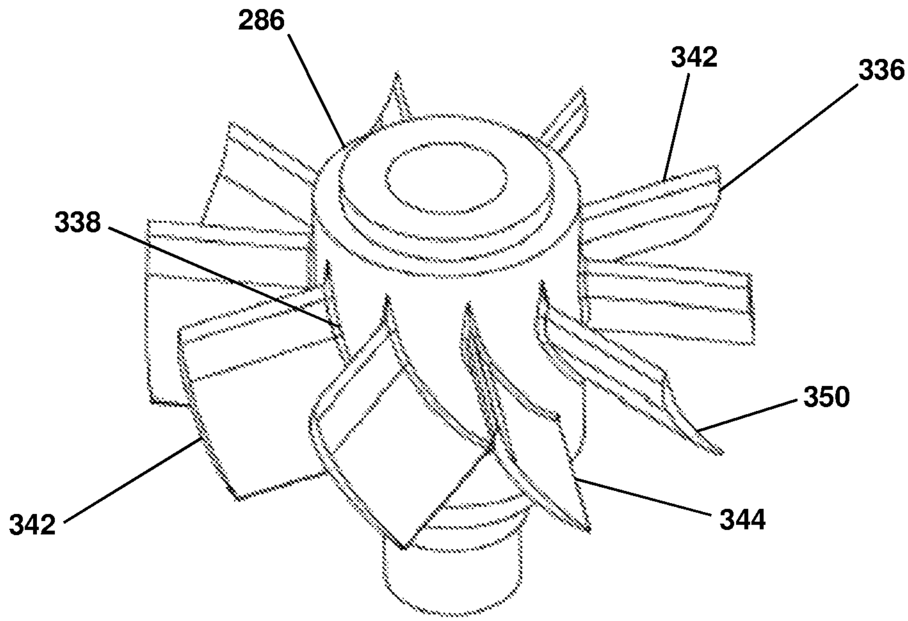
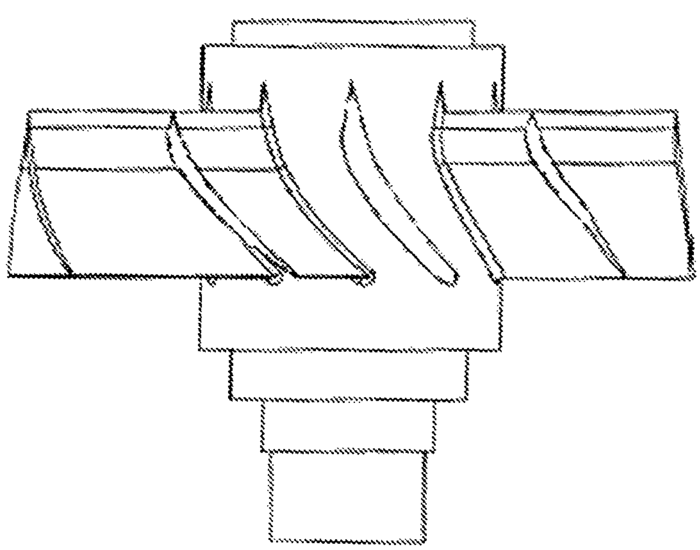


FIG. 12



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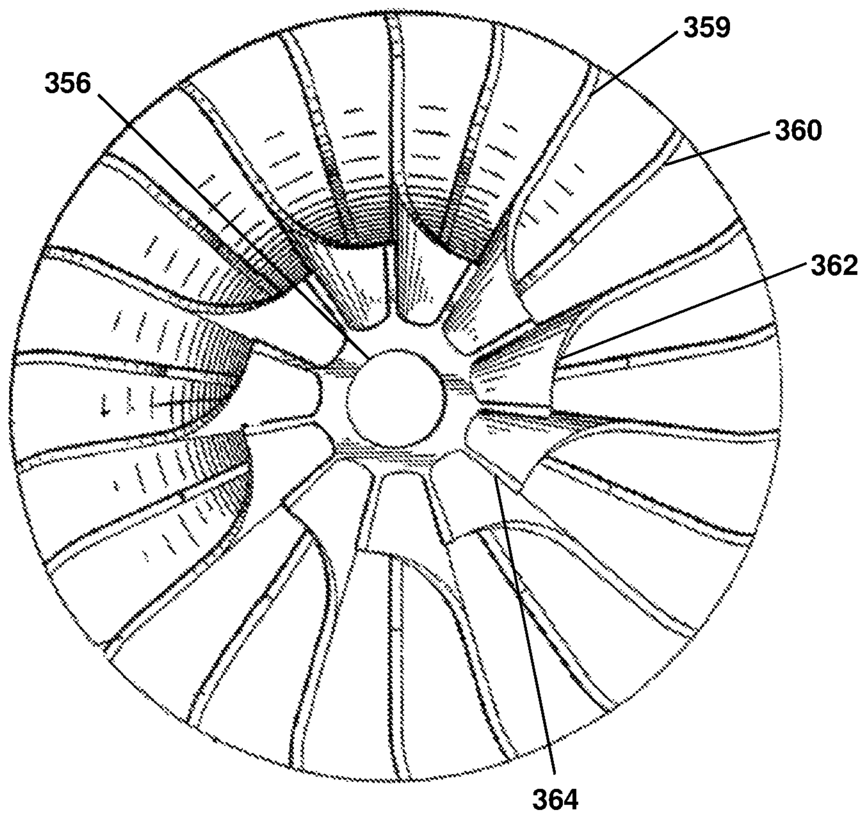
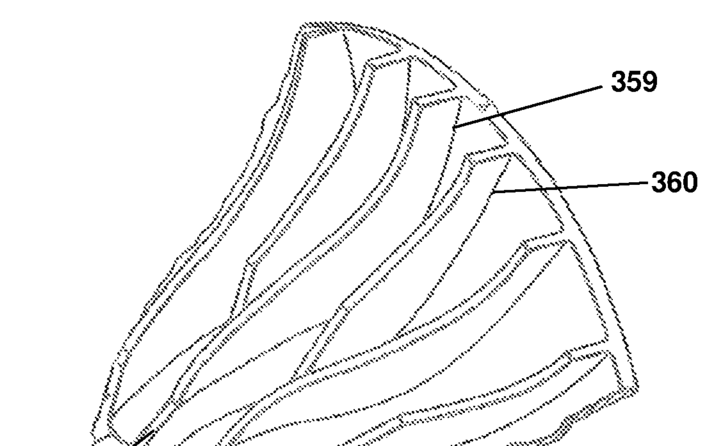


FIG. 14



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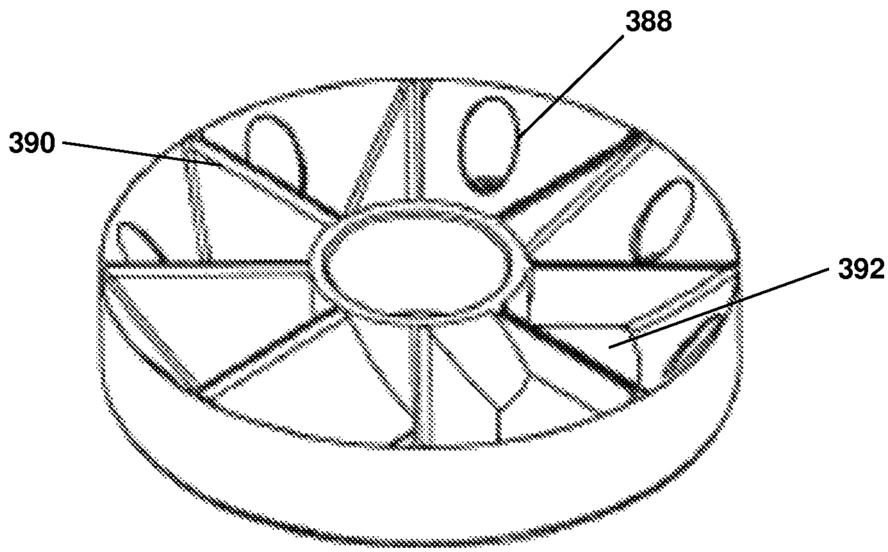

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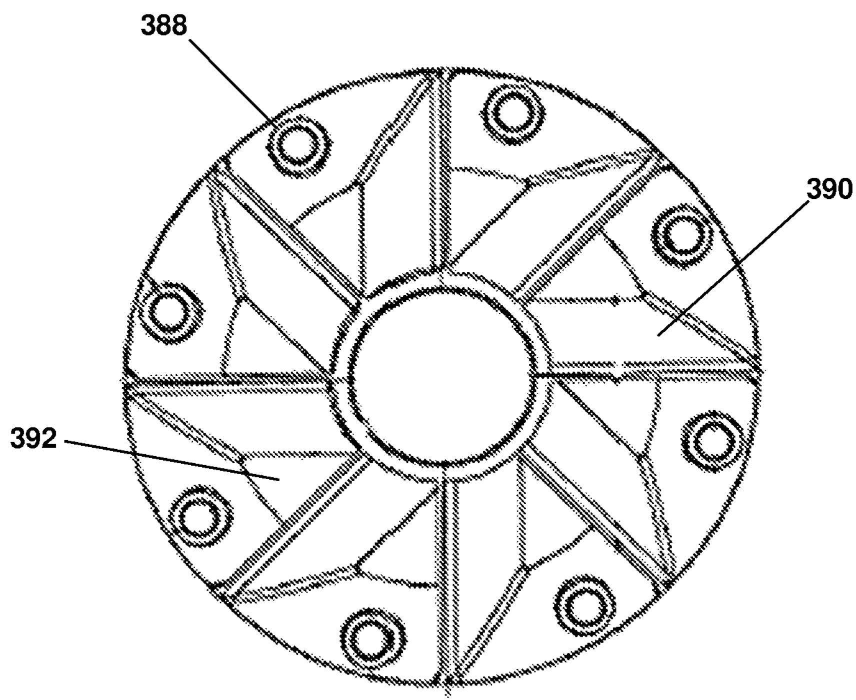


FIG. 17

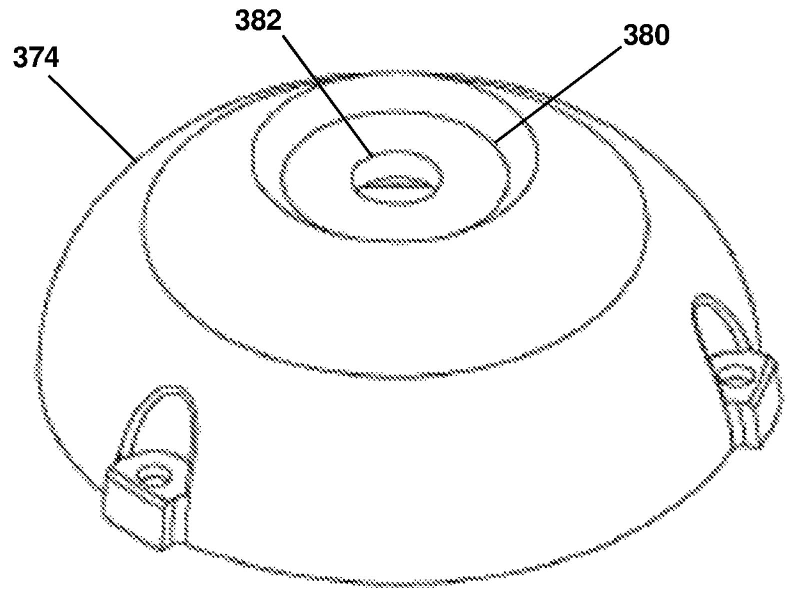


FIG. 18

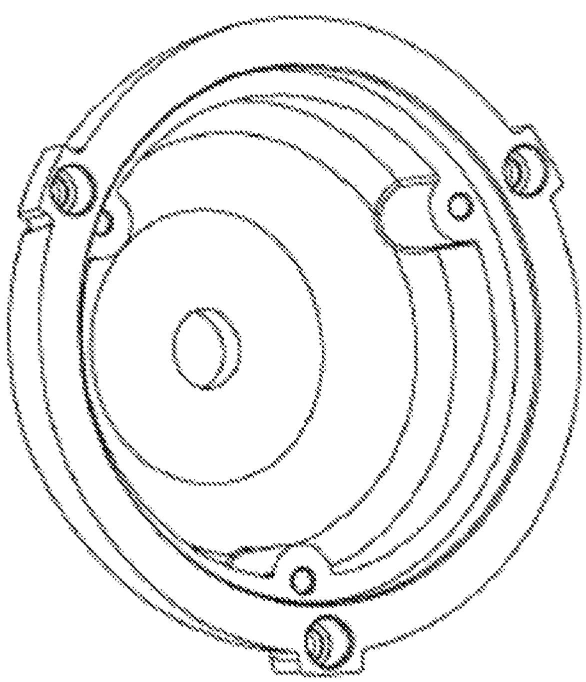


FIG. 19

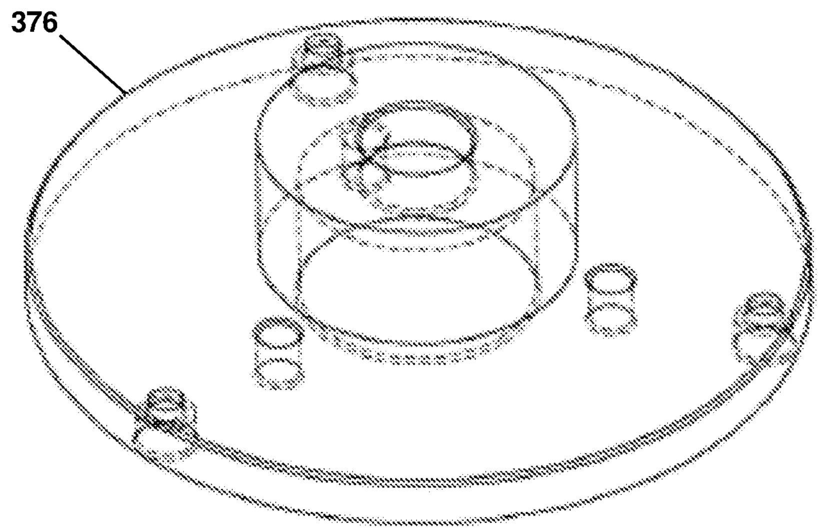


FIG. 20

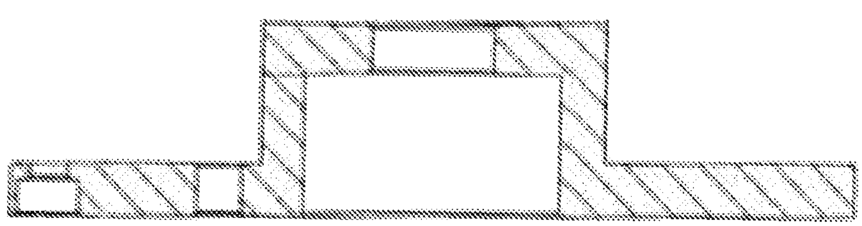


FIG. 21

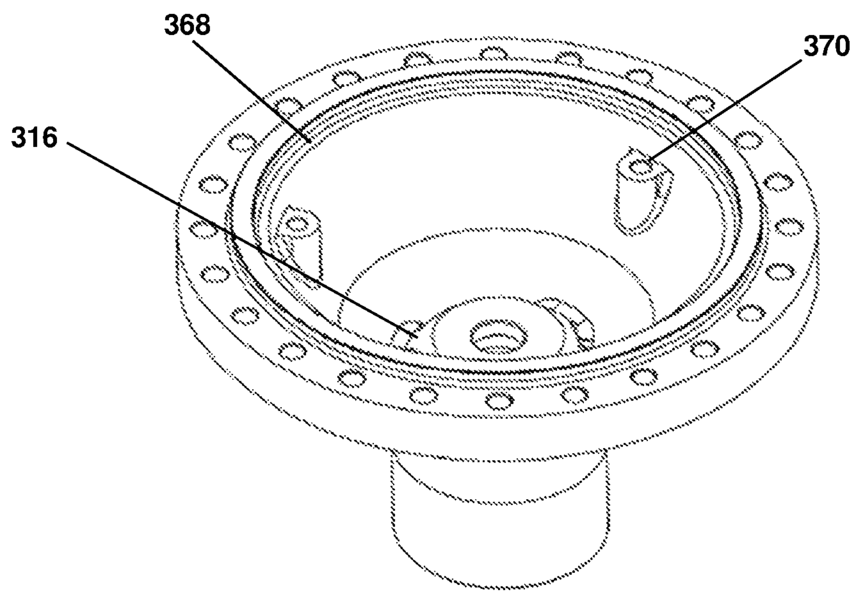
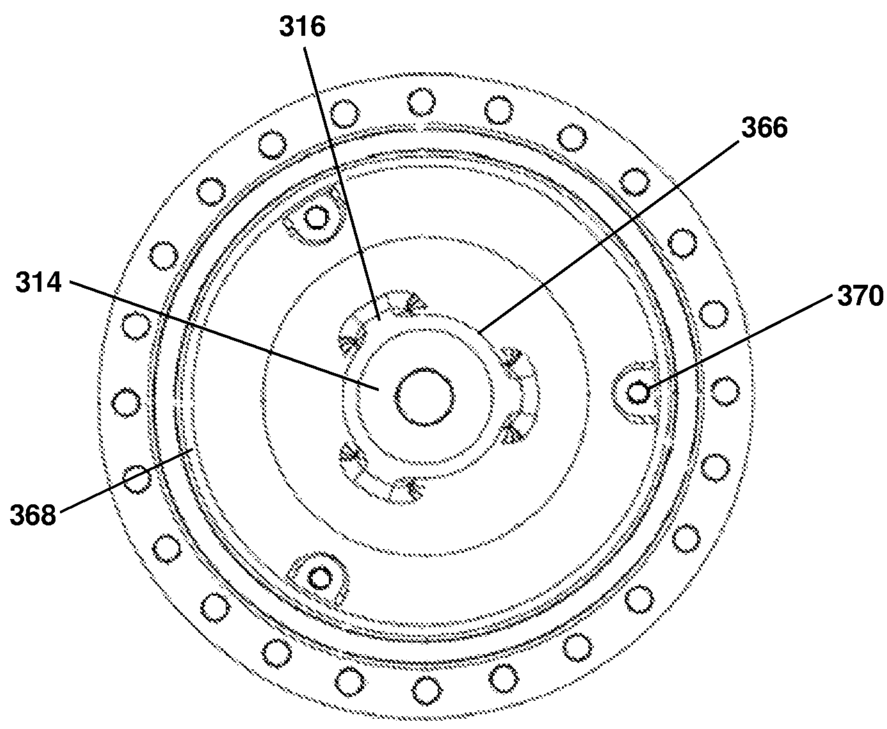


FIG. 22



METHOD AND APPARATUS FOR OPERATING AN ENGINE ON COMPRESSED GAS

CROSS-REFERENCE TO RELATED APPLICATIONS

Not Applicable

FIELD OF THE INVENTION

The present invention relates to a method and apparatus for operating an engine using a compressed gas as the motive fluid. More particularly, the present invention relates to an apparatus for adapting a pre-existing internal combustion engine for operation on a compressed gas.

BACKGROUND

Air pollution is one of the most serious problems facing the world today. One of the major contributors to air pollution is ordinary internal combustion engine which are used in most motor vehicles today. Various devices, including many items mandated by legislation, have been proposed in an attempt to limit the pollutants which an internal combustion engine exhausts to the air. However, most of these devices have met with limited success and are often both prohibitively expensive and complex. A clean alternative to the internal combustion engine is needed to power vehicles and other machinery.

A compressed gas, preferably air, would provide an ideal motive fluid for a engine since it would eliminate the usual pollutants exhausted from an internal combustion engine. An apparatus for converting an internal combustion engine for operation on compressed air is disclosed in U.S. Pat. No. 3,885,387 issued May 27, 1975 to Simington. The Simington patent discloses an apparatus including a source of compressed air and a rotating valve actuator which opens and closes a plurality of mechanical poppet valves. The valves deliver compressed air in timed sequence to the cylinders of an engine through adapters located in the spark plug holes. However, the output speed of an engine of this type is limited by the speed of the mechanical valves and the fact that the length of time over which each of the valves remains open cannot be varied as the speed of the engine increases.

Another apparatus for converting an internal combustion engine for operation on steam or compressed air is disclosed in U.S. Pat. No. 4,102,130 issued Jul. 25, 1978 to Stricklin. The Stricklin patent discloses a device which changes the valve timing of a conventional four stroke engine such that the intake and exhaust valves open once for every revolution of the engine instead of once every other revolution of the engine. A reversing valve is provided which delivers live steam or compressed air to the intake valves and is subsequently reversed to allow the exhaust valves to deliver the expanded steam or air to the atmosphere. A reversing valve of this type however does not provide a reliable apparatus for varying the amount of motive fluid injected into the cylinders when it is desired to increase the speed of the engine. Further, a device of the type disclosed in the Stricklin patent requires the use of multiple reversing valves if the cylinders in a multi-cylinder engine were to be fired sequentially.

The present inventor, Rogers, disclosed in U.S. Pat. No. 4,292,804 issued Oct. 10, 1981 a method and apparatus for operating an engine having a cylinder and a piston reciprocable therein on compressed gas. The apparatus comprises a source of compressed gas connected to a distributor which distributes the compressed gas to the cylinder. A valve is provided to selectively admit compressed gas to the cylinder when the piston is in an approximately top dead center position. In one embodiment of the present invention the timing of the opening of the valve is advanced such that the compressed gas is admitted to the cylinder progressively further before the top dead center position of the piston as the speed of the engine increases. The engine however, operated utilizing a centrifugal compressor, that was unable to supply sufficiently high volumes of compressed air for the engine to run reliably.

The present inventor, Rogers, disclosed in U.S. Pat. No. 4,693,669 issued Sep. 15, 1987 a supercharger for delivering compressed air to the engine disclosed in U.S. Pat. No. 4,102,130, comprising a shrouded axial compressor, a radial compressor which is located downstream of the axial compressor and a housing. The housing comprising four sections, including a section defining a highly convergent, frustoconical transition duct which favorably directs the discharge of the axial compressor to the inlet of the radial compressor and a hollow, highly convergent, exhaust cone section immediately downstream of the radial compressor which converges into the exhaust port of the supercharger. An annular flow deflector is provided for directing the discharge of the radial compressor into the exhaust cone. The supercharger was able to supply a greater volume of air to the to the engine however, the disclosed supercharger was unable to supply sufficiently high pressure air for the engine to run reliably.

Therefore, it is an object of the present invention to provide a reliable method and apparatus for operating an engine or converting an engine for operation with a compressed gas.

A further object of the present invention is to provide a method and apparatus which is effective to deliver a constantly increasing amount of compressed gas to an engine as the speed of the engine increases.

A still further object of the present invention is to provide a method and apparatus which will operate an engine using compressed gas at a speed sufficient to drive a conventional automobile at highway speeds.

It is still a further object of the present invention to provide a method and apparatus which is readily adaptable to a standard internal combustion engine to convert the internal combustion engine for operation with a compressed gas.

Another object of the invention is to provide a method and apparatus which utilizes cool expanded gas, exhausted from a compressed gas engine, to operate an air conditioning unit and/or an oil cooler.

These and other objects are realized by a method and apparatus according to the present invention for operating an engine having at least one cylinder and a reciprocating piston therein using compressed gas as a motive fluid. The apparatus includes a source of compressed gas and a distributor connected with the source of the compressed gas for distributing the compressed gas to the at least one cylinder. A valve is provided for admitting the compressed gas to the cylinder when the piston is in approximately a top dead center position within the cylinder. An exhaust is provided for exhausting the expanded gas from the cylinder as the piston returns to approximately the top dead center position.

SUMMARY

In a preferred embodiment of the present invention a device is provided for varying the duration of each engine cycle over which the valve remains open to admit compressed gas to the cylinder dependent upon the speed of the engine. In a further preferred embodiment of the present invention, an apparatus for advancing the timing of the opening of the valve is arranged to admit the compressed gas to the cylinder progressively further before the top dead center position of the piston as the speed of the engine increases.

Further features of the present invention include a valve for controlling the amount of compressed gas admitted to the distributor. Also, a portion of the gas which has been expanded in the cylinder and exhausted through the exhaust valve is delivered to a compressor to be recompressed and returned to the source of compressed gas. A gear train is selectively engagable to drive the compressor at different operating speed depending upon the pressure maintained at the source of compressed air and/or the speed of the engine. Still further, a second portion of the exhaust gas is used to cool a lubricating fluid for the engine or to operate an air conditioning unit.

In a preferred embodiment of the present invention, the valve for admitting compressed gas to the cylinder is electrically actuated. The device for varying the duration of each engine cycle over which the intake valve remains open as the speed of the engine increase comprises a rotating element whose effective length increases as the speed of the engine increases such that a first contact on the rotating element is electrically connected to a second contact for a longer period of each engine cycle. The second contact actuates the valve whereby the valve remains in an open position for a longer period of each engine cycle as the speed of the engine increases.

Features of the present invention include an adaptor plate for supporting the distributor above an intake manifold of a conventional internal combustion engine after a carburetor has been removed to allow air to enter the cylinders of the engine through the intake manifold and conventional intake valves. Another adaptor plate is arranged over an exhaust passageway of the internal combustion engine to reduce the cross-sectional area of the exhaust passageway.

Compressed air for the is supplied by a compress for delivering high volume-high pressure compressed air to the engine. The compressor is comprised of a shrouded axial compressor, a radial compressor which is located downstream of the axial compressor, an insert downstream of the axial compressor, and a housing. The housing comprising four sections, including a section defining a frustoconical transition duct which favorably directs the discharge of the axial compressor to the inlet of the radial compressor and a convergent exhaust cone section immediately downstream of the radial compressor which converges into the exhaust port of the compressor with connection points for the insert to attach to, and at least one opening at the exit of the convergent exhaust cone section with a smaller cross-sectional area than the convergent exhaust cone exit port. An annular flow deflector is provided for directing the discharge of the radial compressor into the exhaust cone.

The insert in the compressor prevents the air compressed by the axial and radial compressors from expanding and de-pressurizing in the exhaust port of the compressor, and the at least one opening at the exit of the convergent exhaust cone section further keeps the compressed air from expanding. The result of the compressor design allows the compressor to produce high pressure air, in excess of 400 psi, at high volumes, in excess of 20 cfm (@400 psi) to supply air to the engine.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. **1** is a schematic representation of an apparatus according to the present invention arranged on an engine;

FIG. **2** is a side view of one embodiment of a valve actuator according to the present invention;

FIG. **3** is a cross-sectional view taken along the line **3-3** in FIG. **2**;

FIG. **4** is a cross-sectional view of a second embodiment of a valve actuator according to the present invention;

FIG. **5** is a view taken along the line **5-5** in FIG. **4**;

FIG. **6** is a cross-sectional view of a third embodiment of a valve actuator according to the present invention;

FIG. **7** is a view taken along the line **7-7** in FIG. **6**;

FIG. **8** is a perspective view drawing of the compressor of the present invention;

FIG. **9** is an air intake end view drawing of the compressor of the present invention;

FIG. **10** is a cross-sectional view of the compressor of the present invention;

FIG. **11** is a perspective view drawing of the axial compressor component of the compressor of the present invention;

FIG. **12** is a side view drawing of the axial compressor component of the compressor of the present invention;

FIG. **13** is a top view drawing of the radial compressor component of the compressor of the present invention;

FIG. **14** is a partial sectional view drawing of the radial compressor component of the compressor of the present invention;

FIG. **15** is a perspective view drawing of the diverter plate component of the compressor of the present invention;

FIG. **16** is a top view drawing of the diverter plate component of the compressor of the present invention;

FIG. **17** is a top perspective view drawing of the upper bell housing component of the compressor of the present invention;

FIG. **18** is a bottom perspective view drawing of the upper bell housing component of the compressor of the present invention;

FIG. **19** is a perspective view drawing of the lower bell housing component of the compressor of the present invention;

FIG. **20** is a cross sectional drawing of the lower bell housing component of the compressor of the present invention;

FIG. **21** is a perspective view drawing of the exhaust cone component of the compressor of the present invention;

FIG. **22** is a top view drawing of the exhaust cone component of the compressor of the present invention;

DETAILED DESCRIPTION OF THE INVENTION AND PREFERRED EMBODIMENT

With reference to FIG. **1**, an engine block **21** (shown in phantom) having two banks of cylinders with each bank including cylinders **20** having pistons **22** reciprocable therein (only one of which is shown in phantom) in a conventional manner. While the illustrated engine is a V-8 engine, it will be apparent that the present invention is applicable to an engine having any number of pistons and cylinders with the V-8 engine being utilized for illustration purposes only. A compressed gas tank **23** is provided to store a compressed gas at high pressure. It may also be desirable to include a small electric or gas compressor to provide compressed gas to supplement the compressed gas held in the tank **23**. In a preferred embodiment, the compressed gas is air which can be obtained from any suitable source.

A line **25** transports the gas withdrawn from the tank **23** when a conventional shut off valve **27** is open. In addition, a solenoid valve **29** preferably operated by a suitable key operated switch (not shown) for the engine is also arranged in the line **25**. In normal operation, the valve **27** is maintained open at all times with the solenoid valve **29** operating as a selective shut off valve to start and stop the engine **21** of the present invention.

A suitable regulating valve **31** is arranged downstream from the solenoid valve **29** and is connected by a linkage **34** to a throttle linkage **35** which is operator actuated by any suitable apparatus such as a foot pedal (not shown). The line **25** enters an end of a distributor **33** and is connected to an end of a pipe **35** which is closed at the other end. A plurality of holes, which are equal to the number of cylinders in the engine **21**, are provided on either side of the pipe **35** along the length of the pipe **35**.

When the present invention is used to adapt a conventional internal combustion engine for operation on compressed gas, an adaptor plate **36** is provided to support the distributor **33** in spaced relation from the usual intake opening in the intake manifold of the engine after a conventional carburetor has been removed. In this way, air is permitted to enter the internal combustion engine through the usual passageways and to be admitted to the cylinders through suitable intake valves (not shown). The adaptor plate **36** is secured to the engine block **21** and the distributor **33** by any suitable apparatus, e.g., bolts.

Each of the holes in the pipe **35** is connected in fluid-tight manner to a single line **37**. Each line **37** carries the compressed gas to a single cylinder **20**. In a preferred embodiment, each of the lines **37** is ½ inch high pressure plastic tubing attached through suitable connectors to the distributor **33** and the pipe **35**. Each of the lines **37** is connected to a valve **39** which is secured in an opening provided near the top of each of the cylinders **20**. In the case of a conversion of a standard internal combustion engine, the valves **39** can be conveniently screwed into a tapped hole in the cylinder **20** typically provided for a spark plug of the internal combustion engine. In a preferred embodiment, the valves **39** are solenoid actuated valves in order to provide a fast and reliable opening and closing of the valves **39**.

Each of the valves **39** is energized by a valve actuator **41** through one of a plurality of wires **43**. The valve actuator **41** is driven by a shaft of the engine similar to the drive for a conventional distributor of an internal combustion engine. That is, a shaft **55** of the valve actuator **41** is driven in synchronism with the engine **21** at one half the speed of the engine **21**.

A first embodiment of the valve actuator **41** (FIGS. **2** and **3**) receives electrical power through a wire **45** which is energized in a suitable manner by a battery, and a coil if necessary (not shown) as is conventional in an internal combustion engine. The wire **45** is attached to a central post **47** by a nut **49**. The post **47** is connected to a conducting plate **51** arranged within a housing **53** for the valve actuator **41**. Within the housing **53**, the shaft **55** has an insulating element **57** secured to an end of the shaft **55** for co-rotation therewith when the shaft **55** is driven by the engine **21**. A first end of a flexible contact **59** is continuously biased against the conducting plate **51** to receive electricity from the battery or another suitable source. A second end of the contact **59** is connected to a conducting sleeve **60** which is in constant contact with a spring biased contact **61** which is arranged within the sleeve **60**. The contact **61** is biased by a spring **63** which urges the contact **61** towards a side wall of the housing **53**.

With reference to FIG. **3**, a plurality of contacts **65** are spaced from one another and are arranged around the periphery of the housing **53** at the same level as the spring biased contact **61**. Each contact **65** is electrically connected to a post **67** which extends outside of the housing **53**. The number of contacts **65** is equal to the number of cylinders in the engine **21**. One of the wires **43**, which actuate the valves **39**, is secured to each of the posts **67**.

In operation, as the shaft **55** rotates in synchronism with the engine **21**, the insulating element **57** rotates and electricity is ultimately delivered to successive ones of the contacts **65** and wires **43** through the spring biased contact **61** and the flexible contact **59**. In this way, each of the electrical valves **39** is actuated and opened in the proper timed sequence to admit compressed gas to each of the cylinders **20** to drive the pistons **22** therein on a downward stroke.

The embodiment illustrated in FIGS. **2** and **3** is effective to actuate each of the valves **39** to remain open for a long enough period of time to admit sufficient compressed gas to each of the cylinders **20** of the engine **21** to drive the engine **21**. The length of each of the contacts **65** around the periphery of the housing **53** is sufficient to permit the speed of the engine to be increased when desired by the operator by moving the throttle linkage **35** which actuates the linkage **34** to further open the regulating valve **31** to admit more compressed gas from the tank **23** to the distributor **33**. However, it has been found that the amount of air admitted by the valves **39** when using the first embodiment of the valve actuator **41** (FIGS. **2** and **3**) is substantially more than required to operate the engine **21** at an idling speed. Therefore, it may be desirable to provide a valve actuator **41** which is capable of varying the duration of each engine cycle over which the solenoid valves **39** are actuated, i.e., remain open to admit compressed gas, as the speed of the engine **21** is varied.

A second embodiment of a valve actuator **41** which is capable of varying the duration of each engine cycle over which each of the valves **39** remains open to admit compressed gas to the cylinders **20** dependent upon the speed of the engine **21** will be described with reference to FIGS. **4** and **5** wherein members corresponding to those of FIGS. **2** and **3** bear like reference numerals. The wire **45** from the electrical source is secured to the post **47** by the nut **49**. The post **47** has an annular contact ring **69** electrically connected to an end of the post **47** and arranged within the housing **53**. The shaft **55** rotates at one half the speed of the engine as in the embodiment of FIGS. **2** and **3**.

At an upper end of the shaft **55**, a splined section **71** slidably receives an insulating member **73**. The splined section **71** of the shaft **55** positively holds the insulating member **73** for co-rotation therewith but permits the insulating member **73** to slide axially along the length of the splined section **71**. Near the shaft **55**, a conductive sleeve **72** is arranged in a bore **81** in an upper surface of the insulating element **73** generally parallel to the splined section **71**. A contact **75**, biased towards the annular contact ring **69** by a spring **77**, is arranged within the conductive sleeve **72** in contact therewith. The conductive sleeve **72** also contacts a conductor **79** at a base of the bore **81**.

The conductor **79** extends to the upper surface of the insulating element **73** near an outer periphery of the insulating element **73** where the conductor **79** is electrically connected to a flexible contact **83**. The flexible contact **83** selectively engages a plurality of radial contacts **85** arranged on an upper inside surface of the housing **53**. A weak spring **87** arranged around the splined section **71** engages a stop member **89** secured on the shaft **55** and the insulating element **73** to slightly bias the insulating element **73** towards the upper inside surface of the housing **53** to ensure contact between the flexible contact **83** and the upper inside surface of the housing **53**. As best seen in FIG. **5**, the radial contacts **85** on the upper inside surface of the housing **53** are arranged generally in the form of radial spokes extending from the center of the housing **53** with the number of contacts being equal to the number of cylinders **20** in the engine **21**. The number of degrees covered by each of the radial contacts **85** gradually increases as the distance from the center of the upper inside surface of the housing **53** increases.

In operation of the device of FIGS. **4** and **5**, as the shaft **55** rotates, electricity flows along a path through the wire **45** down through post **47** to the annular contact member **69** which is in constant contact with the spring biased contact **75**. The electrical current passes through the conductive sleeve **72** to the conductor **79** and then to the flexible contact **83**. As the flexible contact **83** rotates along with the insulating member **73** and the shaft **55**, the tip of the flexible contact **83** successively engages each of the radial contacts **85** on the upper inside of the housing **53**. As the speed of the shaft **55** increases, the insulating member **73** and the flexible contact **83** attached thereto move upwardly along the splined section **71** of the shaft **55** due to the radial component of the splines in the direction of rotation under the influence of centrifugal force. As the insulating member **73** moves upwardly, the flexible contact **83** is bent such that the tip of the contact **83** extends further radially outwardly from the center of the housing **53** (as seen in phantom lines in FIG. **4**). In other words, the effective length of the flexible contact **83** increases as the speed of the engine **21** increases.

As the flexible contact **83** is bent and the tip of the contact **83** moves outwardly, the tip remains in contact with each of the radial contacts **85** for a longer period of each engine cycle due to the increased angular width of the radial contacts with increasing distance from the center of the housing **53**. In this way, the length of time over which each of the valves **39** remains open is increased as the speed of the engine is increased. Thus, a larger quantity of compressed gas or air is injected into the cylinders as the speed increases. Conversely, as the speed decreases and the insulating member **73** moves downwardly along the splined section **71**, a minimum quantity of air is injected into the cylinder due to the shorter length of the individual radial contact **85** which is in contact with the flexible contact **83**. In this way, the amount of compressed gas that is used during idling of the engine **21** is at a minimum whereas the amount of compressed gas which is required to increase the speed of the engine **21** to a level suitable to drive a vehicle on a highway is readily available.

With reference to FIGS. **6** and **7**, a third embodiment of a valve actuator **41** according to the present invention includes an arcuate insulating element **91** having a first end pivotally secured by any suitable device such as screw **92** to the shaft **55** for co-rotation with the shaft **55**. The screw **92** is screwed into a tapped hole in the insulating element **91** such that a tab **94** at an end of the screw **92** engages a groove **96** provided in the shaft **55**. In this way, the insulating element **91** positively rotates with the shaft **55**. However, as the shaft **55** rotates faster, a second end **98** of the insulating element **91** is permitted to pivot outwardly under the influence of centrifugal force because of the groove **96** provided in the shaft **55**. A spring **93** connected between the second

end **98** of the element **91** and the shaft **55** urges the second end of the element **91** towards the center of the housing **53**.

A contact **99** similar to the contact **59** (FIG. **2**) is arranged such that one end of the contact **99** is in constant contact with the conducting plate **51** located centrally within the housing **53**. The other end of the contact **99** engages a conductive sleeve **101** arranged in bore **102**. A contact element **95** is arranged in the conductive sleeve **101** in constant contact with the sleeve **101**. The bore **102** is arranged generally parallel to the shaft **55** near the second end of the arcuate insulating element **91**. The contact **95** is biased by a spring **97** towards the upper inside surface of the housing **53** for selective contact with each of the plurality of radial contacts **85** which increase in arc length towards the outer peripheral surface of the housing **53** (FIG. **6**).

In operation of the device of FIGS. **6** and **7**, as the shaft **55** rotates the arcuate insulating element **91** rotates with the shaft **55** and the second end **98** of the insulating element **91** tends to pivot about the shaft **55** due to centrifugal force. Thus, as the effective length of the contact **95** increases, i.e., as the arcuate insulating element **91** pivots further outwardly, the number of degrees of rotation over which the contact **95** is in contact with each of the radial contacts **85** on the upper inside surface of the housing **53** increases thereby permitting each of the valves **39** to remain open for a longer period of each engine cycle to admit more compressed gas to the respective cylinder **20** to further increase the speed of the engine **21**.

With reference to FIG. **1**, a mechanical advance linkage **104** which is connected to the throttle linkage **35**, advances the initiation of the opening of each valve **39** such that compressed gas is injected into the respective cylinder further before the piston **22** in the respective cylinder **20** reaches a top dead center position as the speed of the engine is increased by moving the throttle linkage **35**. The advance linkage **104** is similar to a conventional standard mechanical advance employed on an internal combustion engine. In other words, the linkage **104** varies the relationship between the angular positions of a point on the shaft **55** and a point on the housing **53** containing the contacts. Alternatively, a conventional vacuum advance could also be employed. By advancing the timing of the opening of the valves **39**, the speed of the engine can more easily be increased.

The operation of the engine cycle according to the present invention will now be described. The compressed gas injected into each cylinder of the engine **21** drives the respective piston **22** downward to drive a conventional crankshaft (not shown). The movement of the piston downwardly causes the compressed gas to expand rapidly and cool. As the piston **22** begins to move upwardly in the cylinder **20** a suitable exhaust valve (not shown) arranged to close an exhaust passageway is opened by any suitable apparatus. The expanded gas is then expelled through the exhaust passageway. As the piston **22** again begins to move downwardly a suitable intake valve opens to admit ambient air to the cylinder. The intake valve closes and the ambient air is compressed on the subsequent upward movement of the piston until the piston reaches approximately the top dead center position at which time the compressed gas is again injected into the cylinder **20** to drive the piston **22** downward and the cycle begins anew.

In the case of adapting a conventional internal combustion engine for operation on compressed gas, a plurality of plates **103** are preferably arranged over an end of the exhaust passageways in order to reduce the outlet size of the exhaust passageways of the conventional internal combustion engine. In the illustrated embodiment, a single plate having an opening in the center is bolted to the outside exhaust passageway on each bank of the V-8 engine while another single plate having two openings therein is arranged with one opening over each of the interior exhaust passageways on each bank of the V-8 engine. A line **105** is suitably attached to each of the adaptor places to carry the exhaust to an appropriate location. In a preferred embodiment, the exhaust lines **105** are 1½" plastic tubing.

In a preferred embodiment, the exhaust lines **105** of one bank of the V-8 engine are collected in a line **107** and fed to an inlet of a compressor **109**. The pressure of the exhaust gas emanating from the engine **21** according to the present invention is approximately 25 p.s.i. In this way, the compressor **109** does not have to pull the exhaust into the compressor since the gas exhausted from the engine **21** is at a positive pressure. The positive pressure of the incoming fluid increases the efficiency and reduces wear on the compressor **109**. The exhaust gas is compressed in the compressor **109** and returned through a line **111** and a check valve **113** to the compressed gas storage tank **23**. The check valve **113** prevents the flow of compressed gas stored in the tank **23** back towards the compressor **109**.

The pulley wheel **228** which drives the compressor **109** through the is driven by a belt **135** which is driven by a pulley **137** arranged on a drive shaft **139** of the engine **21**.

Referring to FIGS. **8**, **9** and **10**, a compressor **109** is provided for supplying compressed air to compressed gas storage tank **23**. In accordance with a preferred embodiment of the present invention, the compressor **109** comprises a housing **212** having an axially directed inlet **214** for receiving exhaust and an axially directed outlet **216** for delivering compressed air to the compressed gas storage tank **23**. Rotatably mounted within the housing **212** is a shaft **218** on which are secured an axial compressor **224** and a radial compressor **226**, the radial compressor **226** being positioned downstream of the axial compressor **224**. A pulley wheel **228** is secured to a forward end **230** of the shaft for receiving drive belts **135**, which belts drivingly connect the shaft **218** to a pulley wheel on the crankshaft of the engine (not shown). The drive belt **135** delivers torque to the shaft **218** as required for driving the compressors **224** and **226** of the compressor **109**.

The housing **212** itself is constructed from four sections which are preferably bolted together at flanged connections in end-to-end relationship. These sections include a front housing section **232**, an axial compressor duct section **234**, a rear housing section **236** and an exhaust cone section **238**. The shaft **218** extends along the longitudinal axis of the housing **212**.

The front housing section **232** is a hollow cylinder which extends forward of a front bearing support **240**. The front housing section **232** encloses the forward end **230** of the shaft **218** and the associated pulley wheel **228**. At its forward end, the front housing section **232** defines the inlet **214** for receiving air from an external source (not shown). Referring particularly to FIG. **2**, the front housing section **232** includes a lateral opening **244** on one side in order to accommodate the connection of the drive belts **135** to the pulley wheel **228**. The front housing section **232** also includes a forward flange **246** for accommodating the connection of the exhaust line **107** upstream of the compressor **109** according to the particular engine layout.

Referring again to FIG. **10**, the pulley wheel **228** is interference-fitted upon the forward end **230** of shaft **218**. The pulley wheel **228** is preferably a double-track design which is suitable for the attachment of twin drive belts, although a single-belt type pulley wheel would be adequate. The pulley wheel **228** is preferably sized such that the ratio of its diameter with respect to the diameter of the pulley **137** provides an effective gearing ratio in the range of approximately two. Thusly at idle, when the automobile engine is running approximately 700 rpm, the compressor **109** is running at approximately 1,400 rpm, and at cruise, when the engine is running in the range of 2,500 rpm, the compressor **109** is preferably turning over in the range of 5,000 rpm. It is to be noted that although the diameter of the pulley wheel **228** may be substantially reduced in order to achieve a desired gearing ratio, the double-track wheel **228** presents a sufficient sum total of surface area to avoid slippage of the belts **135**. When the compressor is turning over at approximately 5,000 rpm, the compressor output is approximately 25 cfm at 450 psi.

The next adjacent section of housing **212** is the axial compressor duct **234** comprising a short cylinder which is coaxially disposed about the axial compressor **224**. Preferably, the axial compressor duct **234** is constructed from cast aluminum, with the interior surfaces machined to assure uniform clearance between the duct **234** and the axial compressor **224**. As with other sections of the housing **212**, the axial compressor duct **234** is provided with flanges **252** and **254** for effecting connection to the adjacent housing sections. The axial compressor duct **234** guides air delivered from the front housing section **232** toward the axial compressor **224**.

Referring now to FIG. **10**, a front bearing support **240** is interposed between the front housing section **232** and the axial compressor duct **234**. The front bearing support **240** is rigidly secured to the housing **212** such that loads and shocks to the shaft **218** can be transferred through the front bearing support **240** to the housing **212**.

The outer raceway of the front roller bearing assembly **276** is secured by the front bearing support **240**. In the preferred embodiment, the front bearing assembly **276** is of the sealed, high speed type. The front bearing assembly **276** is preferably secured to the shaft **218** with an interference fit.

The rear housing section **236** is connected by bolts to the downstream end of the axial compressor duct **234**. Preferably, the rear housing section **236** is constructed from a single section of cast aluminum. The walls of the rear housing section **236** define four elements of the compressor **109**: a conical transition duct **296** which favorably directs the output of the axial compressor to an inlet **298** of the radial compressor **226**; a recess **294** for the deflector blade **386**; the inlet **298** of the radial compressor **226**, itself; and a casing **300** for the radial compressor **226**.

The transition duct **296** is a hollow, frustoconical portion having a half-apex angle (from the generatrix to the axis of symmetry) of approximately 35". The angle is selected such that the inlet to the radial compressor **226** is as close as possible to the outlet of the axial compressor **224** without causing undue back-pressure. In the preferred embodiment, the transition duct **296** begins a short distance downstream of the deflector blade **386** and ends at the beginning of the inlet **298** of the radial compressor **226**.

Referring now to FIGS. **10**, **15** & **16**, incorporated into the recess **294** of the transition duct **296** is a deflector blade **386**. In the preferred embodiment, the deflector blade **386** is round, and the outside edge of deflector blade has a plurality of bolt holes **388**, which allow the deflector blade to be bolted to the transition duct **296**. The center of the deflector blade **386** comprises a plurality of angled diffuser blades **390** and veins **392**. The plurality of angled diffuser blades **390** and veins **392** are angled at the opposite direction of the axial compressor blades **336**. Angling the diffuser blades **390** and veins **392** at the opposite direction of the axial compressor **224** blades **336** reverses the rotational direction of the airflow as it travels through the deflector blade **386**, thus, air entering the deflector blade **386** with a clock-wise rotational direction, will exit with a counter clock-wise rotational direction.

At the inlet **298** of the radial compressor **226**, the walls of the rear housing **236** are cylindrical and coaxially disposed about the shaft **218**. It is to be noted that in the preferred embodiment, the surface transition **302** from the transition duct **296** to the inlet **298** is rounded-off.

The casing portion **300** of the rear housing section **236** closely follows the contour defined by blade edges **304** of the radial compressor **226** in a close, substantially sealing manner as is well known in the art of radial compressors. The casing portion **300** of the rear housing section **236** channels air between the rotating blades of the radial compressor **226** so that the blades can impart work to the passing air. The casing portion **300** also defines a discharge outlet **306** for the radial compressor **226**.

Just beyond the discharge outlet **306** of the radial compressor **226**, the interior surfaces of the rear housing section **236** begin to curve immediately inwardly to provide a transition into the next adjacent section of the housing **212**, the exhaust cone **238**. In this fashion, the interior surfaces at the rear-most portion of rear housing section **236** and those of the forward portion of the exhaust cone **238** define internally a flow deflector **308**. In the preferred embodiment, the flow deflector **308** is closely and concentrically disposed about the outlet **306** of the radial compressor **226** such that the air being discharged from the radial compressor **226** does not have the opportunity to diffuse significantly prior to its arrival at the annular flow deflector **308**. The annular flow deflector **308** directs the output of the radial compressor **226** into the exhaust cone **238** by providing a smooth surface transition from the interior of rear housing section **236** to the interior of the exhaust cone **238**.

The exhaust cone **238** is a convergent, frustoconical section placed immediately downstream of the radial compressor **226** for receiving the output of the radial compressor **226** from the annular flow deflector **308**. In the preferred embodiment, the exhaust cone **238** is a single section of cast aluminum which is joined to the downstream end of the rear housing section **236** at a flanged joint **310**. Preferably, the exhaust cone **238** converges according to a half-apex angle of approximately 35° and defines the exhaust port **216** at its terminus. Threading **312** at the exhaust port **216** accommodates the attachment of the appropriate external ducting (not shown) leading to the intake of the engine.

Referring now to FIGS. **10**, **21**, and **22** the exhaust cone **238** includes a rear bearing support **314** which comprises members **316** which extend radially inwardly from the outer walls of the exhaust cone **238**. At a radial inward location close to the shaft **218**, the members **316** converge to form a cupped annulus which serves as a housing for the rear bearing assembly **320**. In the preferred embodiment, the members **316** are integrally formed with the walls of the exhaust cone **238**.

Surrounding the rear bearing support are a plurality of orifice holes **366**. In the preferred embodiment, the plurality of orifice holes **366** are closely and concentrically disposed near the outlet **216** of the exhaust cone **238** such that the air being discharged from the radial compressor **226** does not have the opportunity to diffuse significantly prior to exiting the exhaust cone **238**.

The exhaust cone **238** includes as step **368** located just inside of the flanged joint **310**. Incorporated into the step **368** is a plurality of threaded connection points **370** located in the step **368** for connecting the bell housing **372** to the exhaust cone **238**.

Referring now to FIGS. **10**, **17**, **18**, **19**, and **20**, the bell housing **372** is comprised of two pieces, a upper, semi-circular section **374**, and a lower section **376** which is bolted to the upper section. When the bell upper section **374** is attached to the bell lower section **376**, there is a void **378**, in the center of the bell housing **372**. The void **378** keeps the weight of the bell housing **372**, lower, to prevent the compressor **109**, from getting too heavy.

The apex of the bell housing semi-circular section **374** has an contoured depression **380** in which fits against the rear bearing support **314**. Additionally, the contoured depression **380** has a hole **382** in the center of the depression **380** for the shaft **218** to fit through.

When the bell housing **372** is assembled and attached to the exhaust cone **238** the assembly creates a small passageway **384** between the discharge outlet **306** of the radial compressor **226** and the plurality of orifice holes **366**. The incorporation of the bell housing **372** into the exhaust cone **238** prevents the air discharged from the radial compressor **226** from diffusing significantly prior to exiting the exhaust cone **238**,

Referring to FIGS. **10**, **11**, and **12**, the axial compressor **224** upon rotation draws air through the inlet **214** and imparts an initial amount compression to the air as it forces the air into the transition duct **296** of the rear housing section **236**. In the preferred embodiment, the axial compressor **224** comprises a hub **286** and a series of equally spaced, radially disposed blades **336**. Preferably, each blade **336** increases in cord from a root **338** to a tip **340** and includes a trailing edge **342** and a leading edge **344**, which edges are both slightly curved. The blades gradually increase in pitch from the root **338** to the tips **340**. However, the particular values of pitch and other geometrical aspects of the blades **336** might be varied in accordance with different operating speeds or other parameters as would be apparent to one skilled in the pertinent art and familiar with this disclosure.

The axial compressor **224** is preferably constructed from a single, cast aluminum section with the faces of the hub **286** being machined for purposes of achieving accurate, axial positioning of the axial compressor **224** on the shaft **218** relative to the housing **212**. Additionally, the outer periphery **350** of the axial compressor **224** is machined to assure uniform clearance between the shroud and the adjacent interior surfaces **48** of the axial compressor duct **234**. The axial compressor **224** is preferably secured to the shaft **218** by an interference-fit onto the shaft **218**.

Dynamic balance test machines of the conventional type may be used to test the balance of the axial compressor **224** prior to its installation. If an imbalance is detected, material can be removed at the outer periphery **350** of the axial compressor **224** so as to achieve proper balance.

Referring now to FIGS. **10**, **13**, and **14**, the radial compressor **226** is constructed from a single section of cast aluminum and includes a hub **356** and curved blades **358**. Interposed between each pair of blades **358** are a second set of blades **360** which terminate short of the intake **362** of the radial compressor **226** so that the intake **362** is not crowded by both sets of blades. Accordingly, the radial compressor **226** features both a large total number of blades and an intake of relatively small diameter, which features enhance the performance of the compressor **226**. In the region of the intake **362**, the blades **358** present leading edges **364** and undergo a twist into the direction of rotation so as to prevent a favorable angle of attack at the intake **362**.

The shaft **218** is constructed from a hardened steel and is threaded at both ends **230** and **332** for receiving nuts **290** and **328**, respectively.

In operation with the compressed gas engine of the current invention, the compressor **109** is suitably connected at its outlet **216** to the line **111** and a check valve **113** to the compressed gas storage tank **23**, with the drive belts **135** from the engine being attached to the pulley wheel **228**. Then, as the engine is operated, torque is transferred by the drive belts **135** to the pulley wheel **228** for driving the compressors **224** and **226**. Upon rotation, the axial compressor **224** draws air from the line **107** through the inlet **214**, imparts an initial amount of compression to the air and discharges it into the transition duct **296** with a swirl. If the air supplied by the line **107**, is insufficient, additional make up air can be drawn into the compressor **109**, through the pulley opening **136**.

As the discharged from the axial compressor **224** is caused to leave the axial compressor duct **234**, the deflector blade **386** in the transition duct **296** is causes the air discharged from the axial compressor **224** to swirl in the opposite direction. Then action of reversing direction of the swirl of the air has the positive effect of making the radial compressor **226** more efficient. Since the axial compressor **224** and the radial compressor **226** rotate about the same shaft **218**, without any turbulence in the flow, the air velocities with get excessively high when the compressor **109** is run at high rotational speeds, causing the compressor **109** to stonewall, or choke. Running the air between the axial compressor **224** and the radial compressor **226** through the deflector blade **386** prevents the air velocity from exceeding the stonewall point, maximizing the volume of air passing through the compressor **109**.

Upon leaving the transition duct **296**, the air enters the inlet **298** of the radial compressor **362** and then into the compressor **226** itself. In passing through the radial compressor **226**, the air is turned and whirled such that the airflow is centrifugally discharged with a substantial radial velocity component, whereupon the resultant flow is abruptly turned by the annular flow deflector **308** and caused to enter the exhaust cone **238**. Compressed air travels through the exhaust cone **238** through the passageway **384** created by the bell housing **372**, the plurality of orifice holes **366**, and exits through the outlet **216**. The small cross-sectional areas of the passageway **384** and orifice holes **366** prevent the air from expanding as it exists the annular flow deflector **308**, keeping the air compressed at the high pressures necessary to operate the engine. When the compressor is turning over at approximately 5,000 rpm, the compressor output is approximately 25 cfm at 450 psi.

The other bank of the V-8 engine has its exhaust ports arranged with adapter plates **103** similar to those on the first bank. However, the exhaust from this bank of the engine **21** is not collected and circulated through the compressor **109**. In a preferred embodiment, a portion of the exhaust is collected in a line **159** and fed to an enlarged chamber **161**. A second fluid is fed through a line **163** into the chamber **161** to be cooled by the cool exhaust emanating from the engine **21** in the line **159**. The second fluid in the line **163** may be either transmission fluid contained in a transmission associated with the engine **21** or a portion of the oil used to lubricate the engine **21**. A second portion of the exhaust from the second bank of the V-8 engine is removed from the line **159** in a line **165** and used as a working fluid in an air conditioning system or for any other suitable use.

It should be noted that the particular arrangement utilized for collecting and distributing the gas exhausted from the engine **21** would be determined by the use for which the engine is employed. In other words, it may be advantageous to rearrange the exhaust tubing such that a larger or smaller percentage of the exhaust is routed through the compressor **109**. It should also be noted that since the exhaust lines **105** are plastic tubing, a rearrangement of the lines for a different purpose is both simple and inexpensive.

In operation of the engine of the present invention, the engine **21** is started by energizing the solenoid valve **29** and any suitable starting device (not shown), e.g., a conventional electric starter as used on an internal combustion engine. Compressed gas from the full tank **23** flows through the line **25** and a variable amount of the compressed gas is admitted to the distributor **33** by controlling the regulator valve **31** through the linkage **34** and the operator actuated throttle linkage **35**. The compressed gas is distributed to each of the lines **37** which lead to the individual cylinders **20**. The compressed gas is admitted to each of the cylinders **20** in timed relationship to the position of the pistons within the cylinders by opening the valves **39** with the valve actuator **41**.

When it is desired to increase the speed of the engine, the operator moves the throttle linkage **35** which simultaneously admits a larger quantity of compressed gas to the distributor **33** from the tank **23** by further opening the regulator valve **31**. The timing of the valve actuator **41** is also advanced through the linkage **104**. Still further, as the speed of the engine **21** increases, the effective length of the rotating contact **83** (FIG. **4**) or **95** (FIG. **6**) increases thereby electrically contacting a wider portion of one of the stationary radial contacts **85** to cause each of the valves **39** to remain open for a longer period of each engine cycle to admit a larger quantity of compressed gas to each of the cylinders **20**.

As can be seen, the combination of the regulating valve **31**, the mechanical advance **104**, and the valve actuator **41**, combine to produce a compressed gas engine which is quickly and efficiently adaptable to various operating speeds. However, all three of the controls need not be employed simultaneously. For example, the mechanical advance **104** could be utilized without the benefit of one of the varying valve actuators **41** but the high speed operation of the engine may not be as efficient. By increasing the duration of each engine cycle over which each of the valves **39** remains open to admit compressed gas to each of the cylinders **20** as the speed increases, conservation of compressed gas during low speed operation and efficient high speed operation are both possible.

After the compressed gas admitted to the cylinder **20** has forced the piston **22** downwardly within the cylinder to drive the shaft **139** of the engine, the piston **22** moves upwardly within the cylinder **20** and forces the expanded gas out through a suitable exhaust valve (not shown) through the adapter plate **103** (if employed) and into the exhaust line **105**. The cool exhaust can then be collected in any suitable arrangement to be compressed and returned to the tank **23** or used for any desired purpose including use as a working fluid in an air conditioning system or as a coolant for oil.

When using the apparatus and method of the present invention to adapt a ordinary internal combustion engine for operation with compressed gas it can be seen that considerable savings in weight are achieved. For example, the ordinary cooling system including a radiator, fan, hoses, etc. can be eliminated since the compressed gas is cooled as it expands in the cylinder. In addition, there are no explosions within the cylinder to generate heat. Further reductions in weight are obtained by employing plastic tubing for the lines which carry the compressed gas between the distributor and the cylinders and for the exhaust lines. Once again, heavy tubing is not required since there is little or no heat generated by the engine of the present invention. In addition, the noise generated by an engine according to the present invention is considerably less than that generated by an ordinary internal combustion engine since there are no explosions taking place within the cylinders.

The principles of preferred embodiments of the present invention have been described in the foregoing specification. However, the invention which is intended to be protected is not to be construed as limited to the particular embodiments disclosed. The embodiments are to be regarded as illustrative rather than restrictive. Variations and changes may be made by others without departing from the spirit of the invention. Accordingly, it is expressly intended that all such variations and changes which fall within the spirit and the scope of the present invention as defined in the appended claims be embraced thereby.

What is claimed is:

1. An apparatus for operating an engine having at least one cylinder and a reciprocating piston therein comprising:

a compressor that supplies compressed gas to the engine said compressor comprising an axial compressor comprising angled axial compressor blades upstream of a radial compressor, a compressor pulley, a shaft connecting the axial compressor, the radial compressor and the pulley, a housing enclosing the axial compressor, the radial compressor, the pulley, and the shaft, said housing having an inlet section, and pulley opening, and a hollow, cone shaped exhaust section with an insert inside the exhaust cone creating a passageway between the insert and the exhaust cone preventing compressed gas exiting the radial compressor from expanding before the compressed gas exits the compressor;

distributor means connected with the compressor for distributing the compressed gas to the at least one cylinder;

valve means for admitting the compressed gas to the at least one cylinder when the piston is in approximately a top dead center position within the cylinder;

varying means for increasing the duration of each engine cycle over which the valve means admits compressed gas to the at least one cylinder as the speed of the engine increases;

exhaust means for exhausting gas as the piston subsequently approaches approximately the top dead center position;

a rotating engine output shaft driven by the engine with an engine pulley at one end; and

a belt connecting the compressor pulley and the engine pulley through said pulley opening, where rotation of the engine output shaft drives rotation of the axial and radial compressors.

2. The apparatus of claim **1**, said compressor further comprising a stationary deflector blade comprising a plurality of angled diffuser blades and veins, said diffuser blades angled in the opposite direction of said axial compressor blades and said deflector blade located downstream of axial compressor and upstream of the radial compressor.

3. The apparatus of claim **1**, further comprising at least one line connecting said exhaust means for exhausting gas and said compressor inlet whereby supply gas for the compressor is primarily provided from said exhaust means and makeup gas is provided through the pulley opening.

4. An apparatus for operating an engine having at least one cylinder and a reciprocating piston therein comprising:

a compressor that supplies compressed gas to the engine said compressor comprising an axial compressor comprising angled axial compressor blades upstream of a radial compressor, a stationary deflector blade comprising a plurality of angled diffuser blades and veins said diffuser blades angled in the opposite direction of said axial compressor blades and said deflector blade located downstream of axial compressor and upstream of the radial compressor a compressor pulley, a shaft connecting the axial compressor, the radial compressor and the pulley, a housing enclosing the axial compressor, the radial compressor, the pulley, and the shaft, said housing having an inlet section, an exhaust section, and a pulley opening;

distributor means connected with the compressor for distributing the compressed gas to the at least one cylinder;

valve means for admitting the compressed gas to the at least one cylinder when the piston is in approximately a top dead center position within the cylinder;

varying means for increasing the duration of each engine cycle over which the valve means admits compressed gas to the at least one cylinder as the speed of the engine increases;

exhaust means for exhausting gas as the piston subsequently approaches approximately the top dead center position;

a rotating engine output shaft driven by the engine with an engine pulley at one end; and

a belt connecting the compressor pulley and the engine pulley through said pulley opening, where rotation of the engine output shaft drives rotation of the axial and radial compressors.

5. The apparatus of claim **4**, said compressor exhaust section further comprising a hollow conical shape with an insert inside the exhaust section creating a passageway between the insert and the exhaust cone preventing compressed gas exiting the radial compressor from expanding before the compressed gas exits the compressor.

6. The apparatus of claim **4**, further comprising at least one line connecting said exhaust means for exhausting gas and said compressor inlet whereby supply gas for the compressor is primarily provided from said exhaust means and makeup gas is provided through the pulley opening.

7. The apparatus of claim **5**, further comprising at least one line connecting said exhaust means for exhausting gas and said compressor inlet whereby supply gas for the compressor is primarily provided from said exhaust means and makeup gas is provided through the pulley opening.

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