Oct. 5, 1965 J. E. WITZKY 3,209,736

ENGINE

Filed Jan. 14, 1964 4 Sheets-Sheet 1

Fig. 1

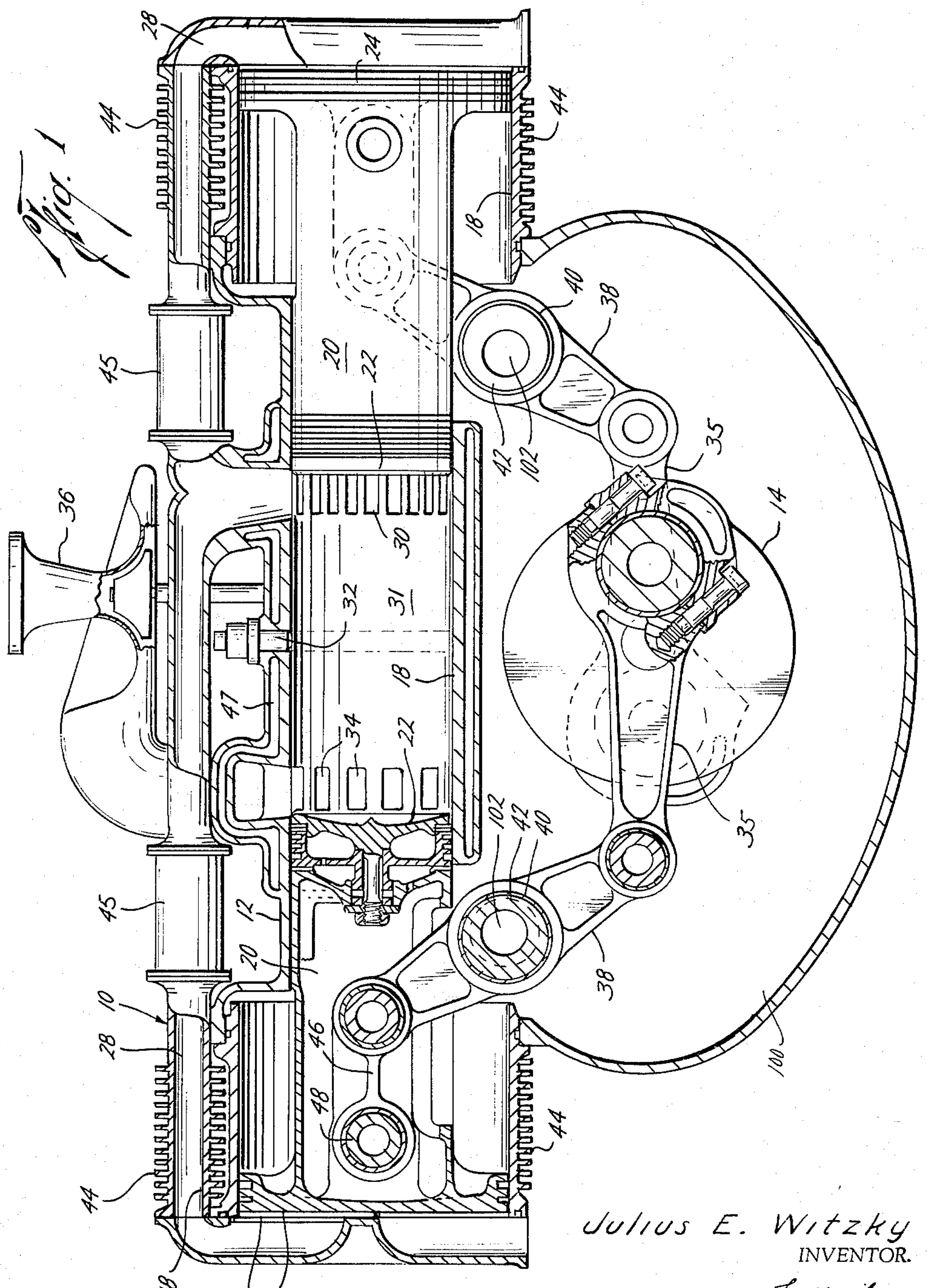

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Fig. 2

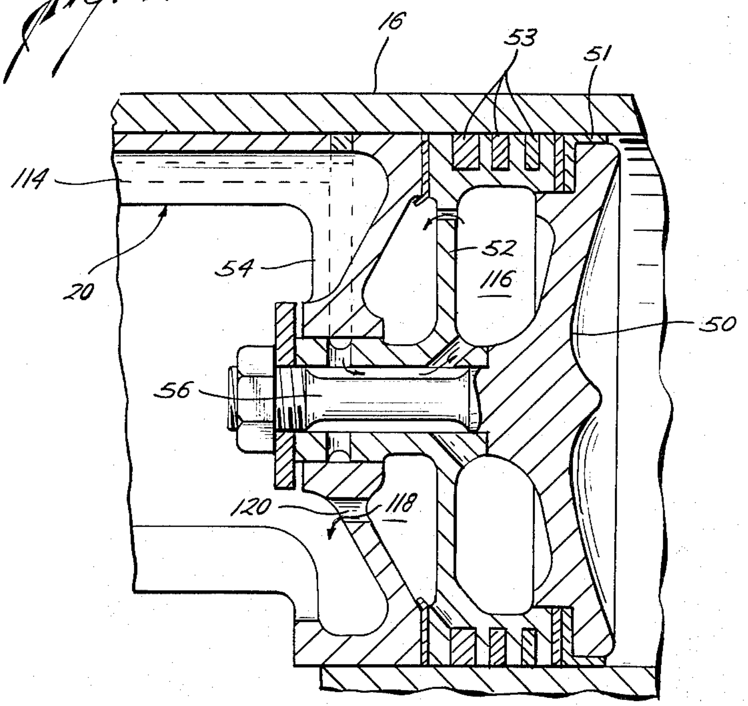


Fig. 3

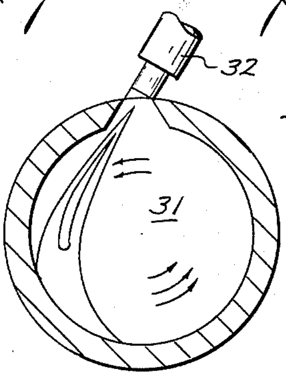


Fig. 4

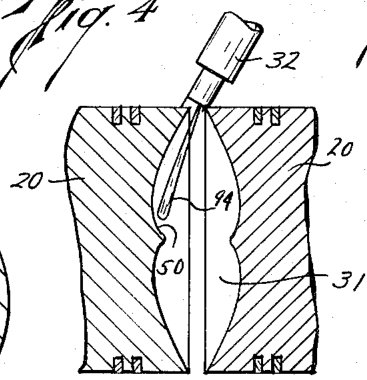
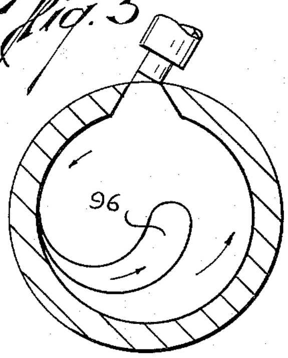


Fig. 5



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ENGINE

Filed Jan. 14, 1964 4 Sheets-Sheet 3

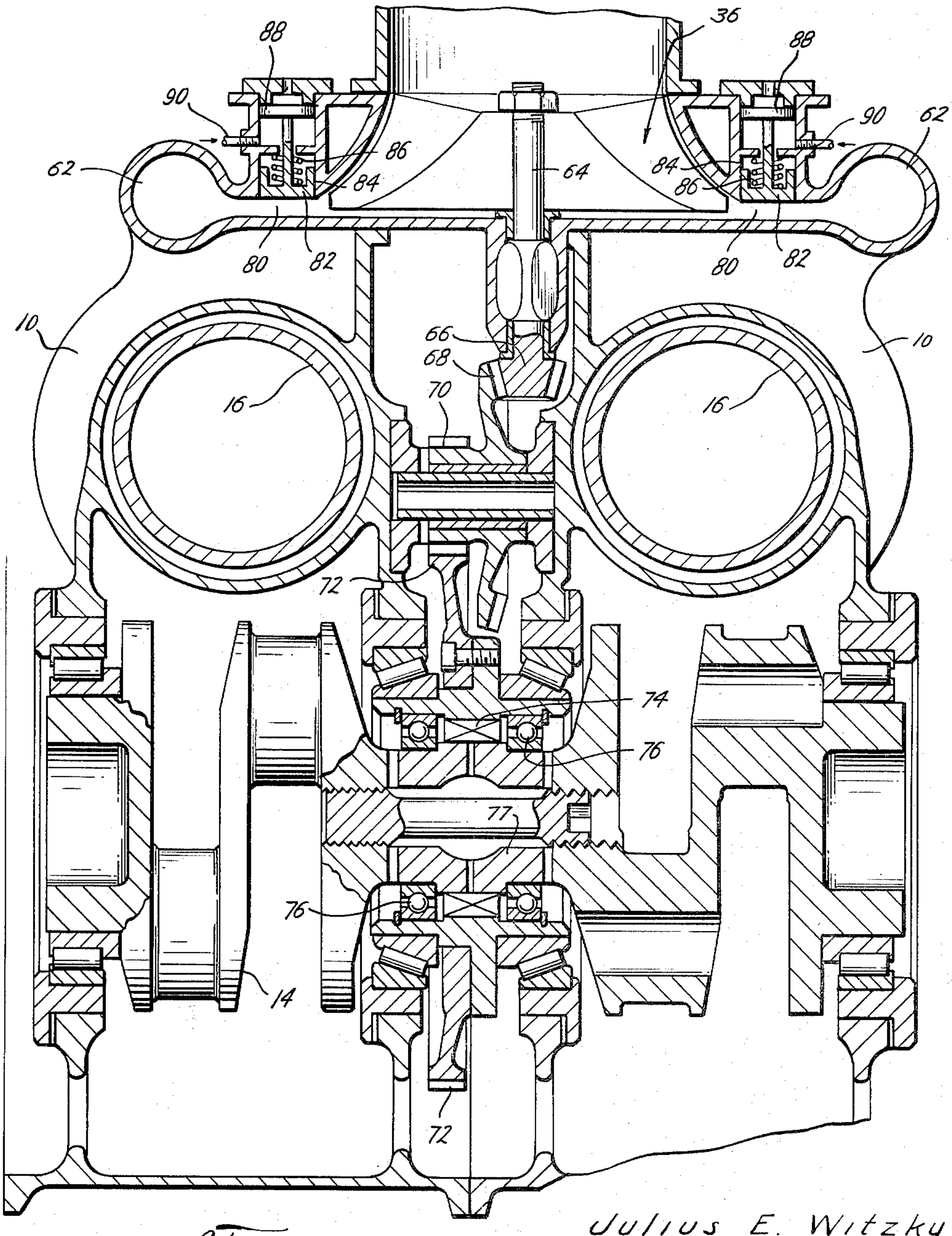

Fig. 6

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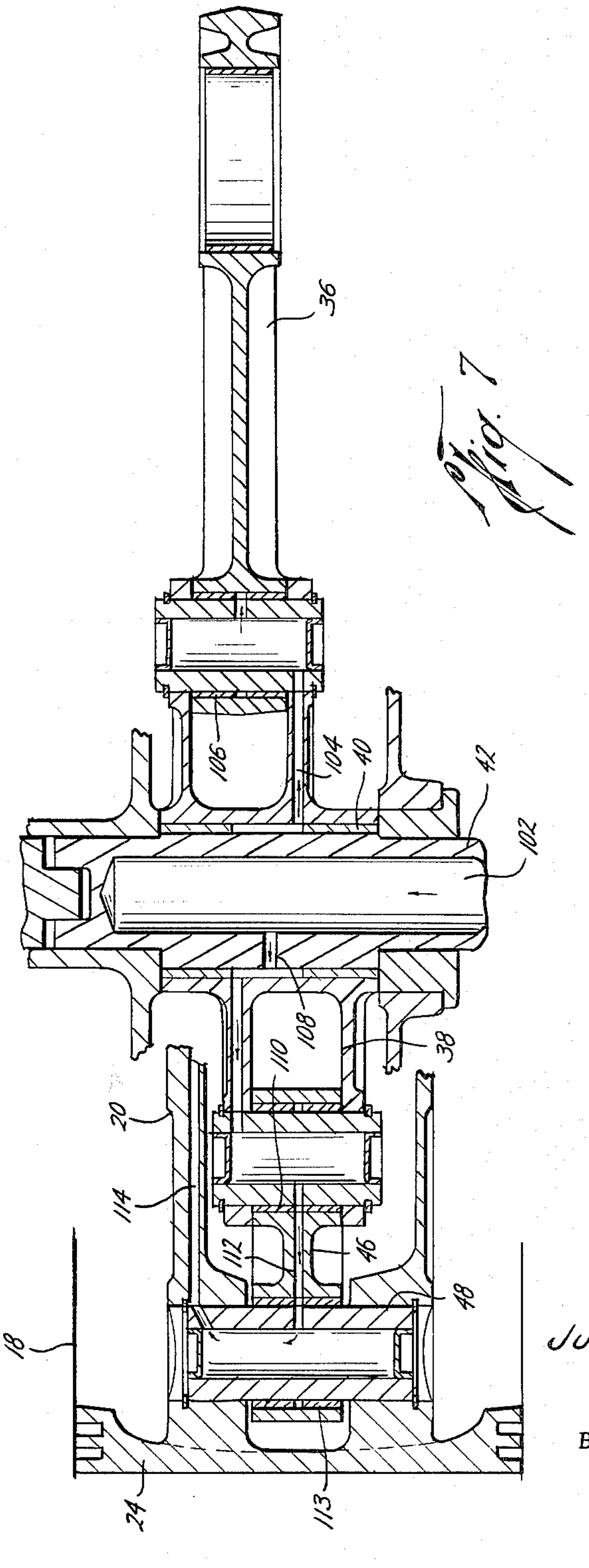
ENGINE

Filed Jan. 14, 1964 4 Sheets-Sheet 4



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3,209,736
ENGINE

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1 Claim. (Cl. 123—48)

The present invention relates to engines, and more particularly, relates to variable compression ratio type engines. In addition, the present invention is also directed to a piston-turbine compound engine.

The general idea of changing the compression ratio of an engine by rotating an eccentrically mounted shaft is generally described in my patent, Variable Compression Ratio Type Engine, No. 2,910,973. One feature of the present invention is the provision of a variable compression ratio type of engine where the eccentrically mounted rocker arms are connected to the pistons by rotatable connecting links. This has the advantage of eliminating the sliding bearing shown in my above patent and in addition allows the piston pin to be relocated to the cold end of the piston thereby permitting an advantageous redesign and improvement in the piston itself. This results in a piston which may utilize a separate piston crown and ring carrier which could then be fabricated out of a heat resistant material different from the main piston body and would allow round machining of the hot end components of the piston and avoid the characteristic running deformity.

Another feature of the present invention is the compounding of a gas turbine with a piston engine. A gas turbine cannot handle the high initial temperatures and pressures required to produce high thermal efficiency. Therefore, the employment of a piston engine to handle the high pressure-high temperature phase of a combustion cycle and then further expand the exhaust gas by flowing the low pressure-low temperature gas from the piston engine through a gas turbine provides a compound engine in which each portion of the engine compensates for the deficiencies inherent in the other.

It is therefore an object of the present invention to provide an improvement in a variable compression type engine wherein a piston connecting link is rotatably connected to an eccentrically mounted rocker arm and is also rotatably connected to a piston pin located on the piston remote from the hot piston head.

A still further object of the present invention is the provision of an improved piston in a variable compression ratio engine wherein the piston crown in rotatably mounted on the piston body and is fabricated out of a heat resistant material and wherein the piston ring carrier is also rotatably mounted on but separate from the piston body.

Yet a further object of the present invention is the provision of a piston turbine compound engine wherein the power developed by the turbine is fed back to the engine crankshaft by a suitable arrangement such as a reduction gear, and an overriding clutch is provided to avoid driving the turbine.

A still further object of the present invention is the provision of a multi-cylinder, two stroke cycle, opposed piston engine compounded with a turbine wherein the exhaust ports are uncovered prior to the uncovering of the intake ports thereby allowing the high pressure gases in the cylinders to flow down into the turbine before the intake ports open.

A still further object of the present invention is the provision of a multi-cylinder, two stroke cycle, opposed super charged engine in which the intake ports are designed to create a high rate of swirl of combustion air about

the cylinder axis between the two pistons and an injection nozzle is provided which injects the fuel in high energy jets to impinge preferably upon the hot piston crown adjacent the exhaust ports where a controlled rate of evaporation can be used to slow the rate of energy released to the combustion process.

A further object of the present invention is the provision of a variable compression type engine wherein internal oil passageways are provided through the rocker arm and the piston connecting link to lubricate the bearing surfaces and then to the interior of the piston head to cool the piston crown.

Yet a further object of the present invention is the provision of a piston turbine compound engine wherein a variable nozzle is provided in the turbine so that the turbine is efficient over the range of operating speeds of the engine and the area of the nozzle may be controlled as a function of the exhaust gas pressure from the combustion cylinders.

Yet a further object of the present invention is the provision of a piston turbine compound engine which provides a compact or "folded" configuration by providing connecting rods parallel to the cylinders which in turn allows the use of a lighter overall main engine structure and permits the advantageous uses of both ferrous and non-ferrous materials for the main engine.

Other and further objects, features and advantages of the invention will be apparent from the following description of a presently preferred embodiment of the invention, given for the purposes of disclosure and taken in conjunction with the accompanying drawings, where like character references designate like parts throughout the several views, and where:

FIGURE 1 is an elevational view, in cross-section, illustrating the apparatus of the present invention with the pistons approximately at the end of the firing stroke,

FIGURE 2 is fragmentary elevational view, in cross-section, illustrating the structure and connection of the piston crown and piston ring carrier to the piston body and the provision of cooling passageways therein,

FIGURE 3 is a schematic elevational view, across the combustion chamber, illustrating fuel injection in the direction of the air swirl,

FIGURE 4 is a fragmentary schematic elevational view, longitudinally along the combustion chamber showing the injection of fuel on the hot piston crown,

FIGURE 5 is a fragmentary elevational view, across the combustion chamber, showing the swirling combustible mixture of the fuel and air,

FIGURE 6 is a fragmentary elevational view, in cross-section across dual cylinders, illustrating the connection of the turbine to the engine crankshaft, and

FIGURE 7 is a fragmentary elevational view, in cross-section, across the connecting rod, rocker arm, connecting link and piston showing the lubrication system.

Referring now to the drawings, and specifically to FIGURE 1, the reference numeral **10** generally designates the engine of the present invention which generally includes a block **12** and a crankshaft **14**. Preferably, the engine is a multi-cylinder, two stroke cycle, opposed piston super-charged diesel, only a single cylinder being shown for convenience of reference. Thus, horizontally formed in the block **12** is power or combustion cylinder **16** which is in communication with a pair of compressors cylinders **18** at opposite ends of the combustion cylinder **16** for supplying supercharging air to the cylinder **16**. Reciprocally mounted in the cylinder **16** are a pair of double ended pistons **20** each one of which has a power portion **22** disposed in the power cylinder **16** and a compressor piston **24** disposed in the adjoining compressor cylinder **18**. Each of the compressor cylinders **18** communicate through a valve **26** and duct **28** to an annular series of intake ports **30** whereby the two compressor pistons **24** provide super-charging air for the cylinder **16** through the intake ports **30**. The power pistons **22** face each other in the power cylinder **16** and form together with the cylinder a combustion chamber **31** which receives air under pressure from the air intake ports **30** and fuel through an injection nozzle **32**.

The exhaust gases from the combustion chamber are discharged through exhaust ports **34** to a gas turbine **36** as will be more fully described hereinafter.

Conventional cooling fins **44** are provided about the compressor cylinders **18** and air intake ducts **28** to cool the incoming air. In addition, air intercoolers **45** may be provided if desired. For cooling the combustion cylinder **16**, conventional water passages **47** are provided.

A connecting rod **35** is connected to the crankshaft **14** for each of the pistons **20**. A rocker arm **38** is connected to each of the connecting rods **35** and extends through the cylinder **16** and is eccentrically mounted on a rocker arm bearing **40** by an eccentric shaft **42**. The rocker arm bearings **40** are mounted on the eccentric shafts **42** and the shafts may be rotated while the engine is in operation. Rotating the eccentric shaft **42** changes the pivot point of the rocker arms **38** which in turn changes the effective compression ratio of the piston and compressor cylinders. Preferably, the eccentric shaft **42** would be linked together (not shown) and operated by a single lever which would be controlled either by a pressure signal or scheduled on an r.p.m. and/or injection pump rack setting.

The rocker arms **38** are not connected directly to the pistons **20** but each are pivotally connected to a piston connecting link **46** which is in turn rotatably connected to a piston pin **48**. The piston connecting link **46** takes care of all the oscillating bearing motion that occurs and avoids the sliding bearing with its lubrication and loading problems as shown in my above named patent. Secondly, the use of the piston connecting link **46** allows the piston pin **48** to be positioned at the cold end or compressor end of the piston **20** thereby permitting the use of an improved piston as will be more fully described hereinafter. With the piston link **46** the length of the piston stroke remains constant, but the outer and inner limits of the stroke can be moved as the variable compression ratio eccentric shaft **42** is rotated.

Referring now to FIGURE 1, the positioning of the piston pin **48** at the end of the piston body **20** remote from the hot head **22** results in several improvements. By moving the pin **48** and boss to the cold end of the piston the disadvantageous characteristic of conventional piston design of requiring an elliptical machined piston skirt so as to make the piston round in the running condition is avoided. The positioning of the pin **48** allows round machining of the hot end components and greatly relieves the stress level in the piston which is both thermally and mechanically the highest stressed part in an opposed piston engine. Referring now to FIGURE 1, the piston crown **50** and the piston ring carrier **52** are preferably made separately and mounted to the piston body **54** with a bolt bearing arrangement **56** such that the crown **50** and carrier **52** are free to rotate and seek their best balanced load running position. By making the crown **50** and piston ring carrier **52** out of separate pieces these parts may be made out of a heat resistant material while the piston body **54** may be fabricated of aluminum. Thus, the heat transfer rate out of the crown **50** would be reduced thereby holding a high energy level in the combustion chamber and reducing the piston cooling requirements. A fire ring **51**, made of steel, is mounted on the crown **50** and protects the piston rings **53**.

Referring now to FIGURE 6, the connection of the turbine through the crank shaft is best seen. Preferably, a plurality, such as two diesel single cylinder engines **10** are compounded with a single exhaust gas turbine **36** which is mounted between and above the two cylinders **16** with a horizontal plane of rotation. The exhaust manifold **62** from each cylinder **16** feeds approximately half the scroll around the turbine nozzle ring. The power developed by the gas turbine **36** is fed back to the main engine crankshaft **14** through a turbine shaft **64** by means of a speed reducing, overriding, gear and clutch arrangement. For instance, a first reduction gear **66** is connected to the turbine shaft **64** and the turbine output power is fed to a spiral bevel gear **68**. However, since the turbine rotates at such a higher speed than the crankshaft **14**, a second reduction gear is provided. Thus, gear **70** which is connected to gear **68** will mesh with and drive gear **72**. Incorporated in this turbine to crankshaft power train is an overrunning clutch **74** which is connected between the reduction gears and the crankshaft **14**.

Clutch **74**, which is positioned between crankshaft **14** and gear **72**, will prevent any feedback of power from the engines **10** to the turbine **36** and also will prevent shock loading of the gearing and will transmit torque only when the turbine **36** develops power. Suitable bearings **76** are provided to rotatably support gear **72** about the crankshaft.

Any suitable means may be used for varying the turbine nozzle as a function of the exhaust gas pressure from the piston combustion chamber. For instance and referring to FIGURE 6, the area of the nozzle **80** should vary in size, depending on the speed of the engines **10**, so as to be efficient over the entire speed range. That is, the nozzle should be small at low speeds and large at high speeds. The nozzle **80** is positioned between the exhaust manifolds **62** from the combustion cylinders **16** and the turbine **36**. A movable valve element **82** is provided which is slidable in a cylinder **84** and is thus movable into and out of the nozzle **80** to control the area of the nozzle. A spring **86** normally urges the element **82** downwardly to close the nozzle. A piston **88** is connected to the valve element and an inlet **90** is provided in communication with the exhaust manifold **62** to provide a pressure on the back side of the piston **88** to move the valve element **82** upwardly, increasing the nozzle area, as the exhaust gas pressure increases. The exhaust air inlets **90** are connected to exhaust gas manifolds of the engines **10**. Thus, as the speed and thus the air pressure in the exhaust manifold **62** of the engines **10** increases, the area of the nozzle **80** will increase thereby providing for an efficient operation of the turbine **36**.

Referring now to FIGURES 1 and 3, the intake ports **30** are designed to impart a high rate of swirl to the incoming air about the cylinder axis. This air swirl will be utilized in the combustion chamber **31**, formed by the pistons **20** and cylinder **16**, to both mix the air and fuel and to limit the rate of combustion. The injection nozzle **32** is located at the periphery of the cylinder at the center point and injects the fuel in a sharp high energy jet to impinge upon the piston crown **50**. This will permit most of the fuel to be deposited on the piston crown where a controlled rate of evaporation can be used to slow the rate of energy released to the combustion process. Preferably, as best seen in FIGURE 4, the fuel **94** from the fuel injector **32** may also be injected at an angle along the longitudinal axis of the combustion chamber **31** and against only the drown of the piston **20** which is at the exhaust end of the combustion chamber **31**. This piston will be hotter as it is not cooled by the incoming air and thus better fuel evaporation can be obtained. In any event, injecting the fuel with the air swirl, as shown in FIGURE 3, will provide a swirling mixture of fuel and air **96**, as best seen in FIGURE 5, and a thorough mixing of the fuel and air.

One important characteristic to note about the intake and exhaust ports **30** and **34** is the location along the cylinder axis with respect to the engine center line. The ports would normally be placed so that the outer end of their lengths coincide with the outer dead point of the pistons. Because of the variable compression ratio of this engine, the lengths of the ports have these effective values at a single compression ratio setting and as the compression ratio is increased the effective length of the ports would be decreased. However, when the compression ratio is increased this would occur at only lower engine speeds and therefore while the effective lengths of the ports is shortened the actual elapsed time the ports remain open would tend to increase because of the effect of slower speed.

As shown, the connecting rods **35** are normally connected to the crankshaft **14** at 180 degrees apart. However, it is contemplated that the connecting rods **35** will be phased such that the piston which uncovers the exhaust ports **34** leads the piston uncovering the intake ports **30** by a predetermined amount, for example approximately 15 degrees. This allows the high pressure gases in the diesel cylinder **16** to blow down into the turbine scroll before the intake ports **30** open. It also closes the exhaust ports **34** first which reduces blow through of the charging air and achieves better supercharging through the differential closing of the ports.

It is also desirable that the piston-turbine compound engine have a "folded" or compact shape. This is accomplished by positioning the connecting rods **35** parallel and close to the axis of the cylinder **16**. In addition, this also allows the use of a lighter overall main engine **12**. The connecting links **46** have only tension forces applied to them, the connecting rods **35** are subject to only compression loads, and the cylinder **16** is required to resist circumferential tension loads of the combustion pressure plus the relatively low tension load imposed by the compressor pistons. This will permit the use of non-ferrous materials for the main structure and the use of stronger ferrous members between the pivot points **42**.

Referring now to FIGURES 2 and 7, the lubrication system is best seen. To provide a full pressure lubricating system, a conventional gear-driven oil pump (not shown) may be located close to the oil sump **100** (FIGURE 1). The opening **102** at the rocker arm fulcrum is pressure lubricated and drilled passages are provided in the rocker arm, piston connecting link, and piston **20**, which are interconnected and will supply lubrication to the bearings and will also provide oil cooling for the hot end of the pistons if desired. Thus, lubricant is supplied under pressure to opening **102** and flows in one direction through drilled passageway **104** in the rocker arm **38** to lubricate bearing **106** which connects the rocker arm **38** and the connecting rod **36**. In addition, lubricant flows from opening **102** through passageway **108** in the rocker arm **38** to lubricate the bearing **110** about the rocker arm pin and then through passageway **112** to lubricate the bearing **113** at the piston pin **48** which connects the connecting link **46** to the piston **20**. From the passageway **112**, the lubricant flows through passageway **114** in the piston **20** to supply oil for piston cooling if desired. Referring now to FIGURE 2, the lubricant passageway **114** is in communication with chamber **116** to cool the hot piston crown **50**. The lubricant flows then to a chamber **118**, through a passageway **120**, and behind the piston head and back to the sump **100** (FIGURE 1).

In use, the turbo-compounded, multi-cylinder, opposed piston, two cycle engine with a variable compression ratio provides a compactness ratio of about 25 horsepower per cubic foot of installed volume. This compactness ratio is in marked contrast to standard diesel compactness ratios between 5 and 10. The variable compression ratio feature will make possible exceptional cold starting ability while allowing combustion pressures to be maintained at any desired level. This feature makes the engine a true multifuel unit while still designed for lightweight construction.

The compression ratio may be changed by rotating either or both of the eccentric shafts **42** which changes the pivot point of the rocker arms **38** since the rocker arm journals are eccentrically mounted on the shaft **42**. This in turn changes the effective compression ratio of the combustion chamber **31**. Preferably, the eccentric shaft **42** for each of the pistons **20** would be linked together and operated by a single lever (not shown) which would be controlled either by a pressure signal or scheduled on a r.p.m.-rack setting ratio.

The use of the piston connecting link **46** provides a desirable connection to the piston **20** at the piston pin **48** which is remote from the hot piston crown **50**. While still providing the necessary movement required as the compression ratio is changed, the linkage **46** simplifies the piston **20** construction by allowing the use of a separate piston crown **50**, and a separate piston ring carrier **52**, both of which are connected to the piston body **54** with a bolt bearing connection **56** such that the crown **50** and carrier **52** are free to rotate and seek their best balanced load running position and in addition can be made of a heat resistant material.

As best seen in FIGURE 6, a plurality such as two diesel cylinders or more may be compounded with a single exhaust gas turbine **36** whereby the power developed by the gas turbine is fed back to the main engine crankcase **14** by a high speed reducing gear arrangement and an over-running clutch **74**. Preferably the turbine is a one stage radial inflow turbine. In order to provide a steady inflow condition to the turbine with the exhaust gas from the cylinders **16** being expanded to approximately supercharger level in the exhaust manifolds **62**, the diesel cycle should be adjusted to allow maximum practical open time for the exhaust ports in an effort to reach the theoretical optimum of each of the two cylinders having its exhaust ports open for 180 degrees of crank angle. As a practical matter, it is preferred to permit the exhaust ports in the cylinders to be open for approximately 135 degrees of crank shaft rotation. In addition, the diesel is designed such that the piston which uncovers the exhaust ports **34** leads to the piston uncovering the intake ports **30** preferably by approximately 15 degrees. This allows the high pressure gases in the diesel cylinder to blow down into the turbine scroll before the intake ports **30** open.

And as previously described the intake ports are designed to impart a high rate of swirl to the incoming air about the cylinder axis. This air swirl will be utilized in the combustion chamber, formed by the two pistons **20** and the cylinder walls of cylinder **16** to both mix the air and fuel and to limit the rate of combustion. The jets from the injection nozzle **32** may impinge on the crown **50** adjacent to the exhaust ports and produce a combustion mixture as shown on FIGURE 5.

The present invention, therefore, is well suited and adapted to attain the ends and objects mentioned herein as well as others inherent therein. While a presently preferred embodiment of the invention is given for the purpose of disclosure, numerous changes in the details of construction and arrangement of parts may be made which are within the spirit of the invention and the scope of the appended claim.

What is claimed is:

An engine comprising,

an elongated cylinder

a pair of two ended pistons slidably mounted in opposed relation in said cylinder to form a combustion chamber therebetween,

each of said pistons including a piston body, a piston ring carrier, and a crown, the ring carrier and crown being rotatably connected to the piston body,

a crankshaft carried by the engine,

a rocker arm for each piston passing through the cylinder and having one end pivotally connected to said crankshaft,

a connecting link pivotally connected to each piston and to one end of the rocker arm, said connecting link being connected to the piston solely at the end remote from the combustion chamber, and eccentric means pivotally supporting each of said rocker arms for shifting the pivot axis of the rocker arms whereby the compression ratio of the engine may be varied.

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