June 27, 1961 E. A. HULBERT 2,989,954

INTERNAL-COMBUSTION ENGINE

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FIG.1.

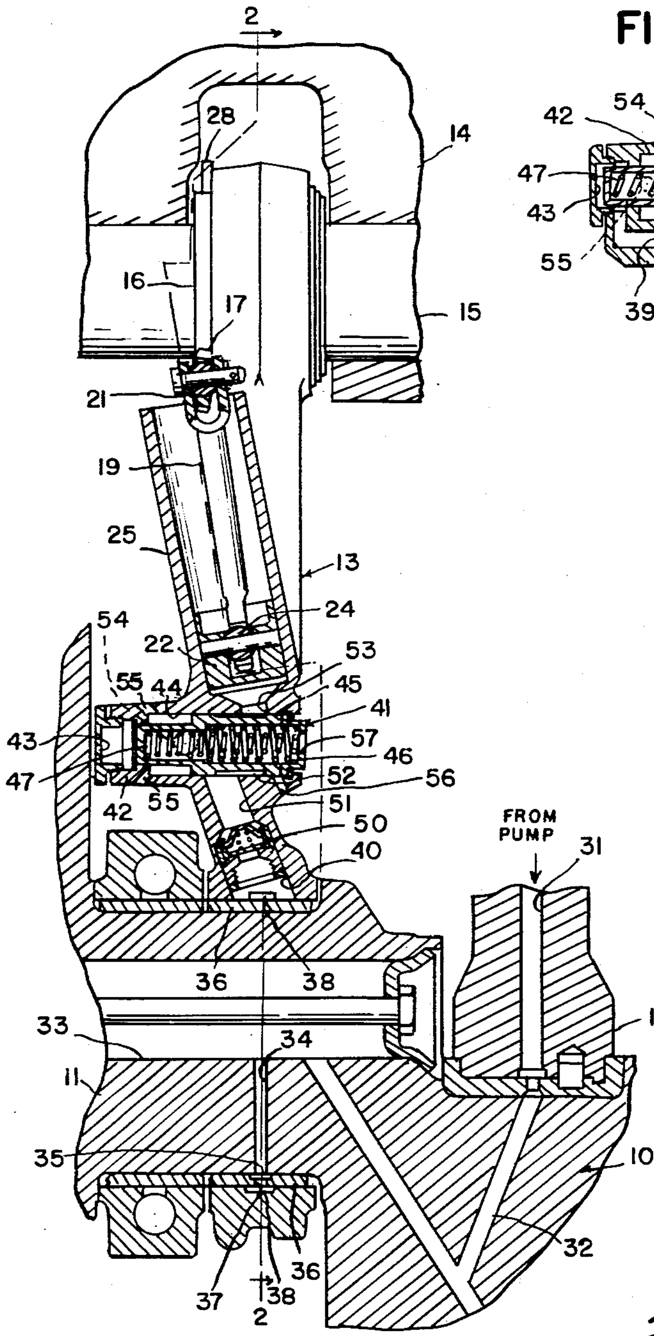

FIG.4.

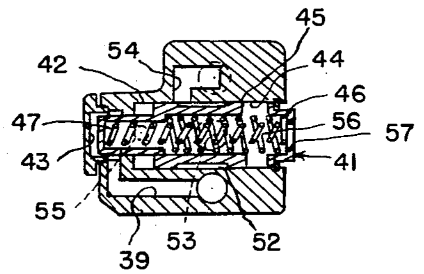

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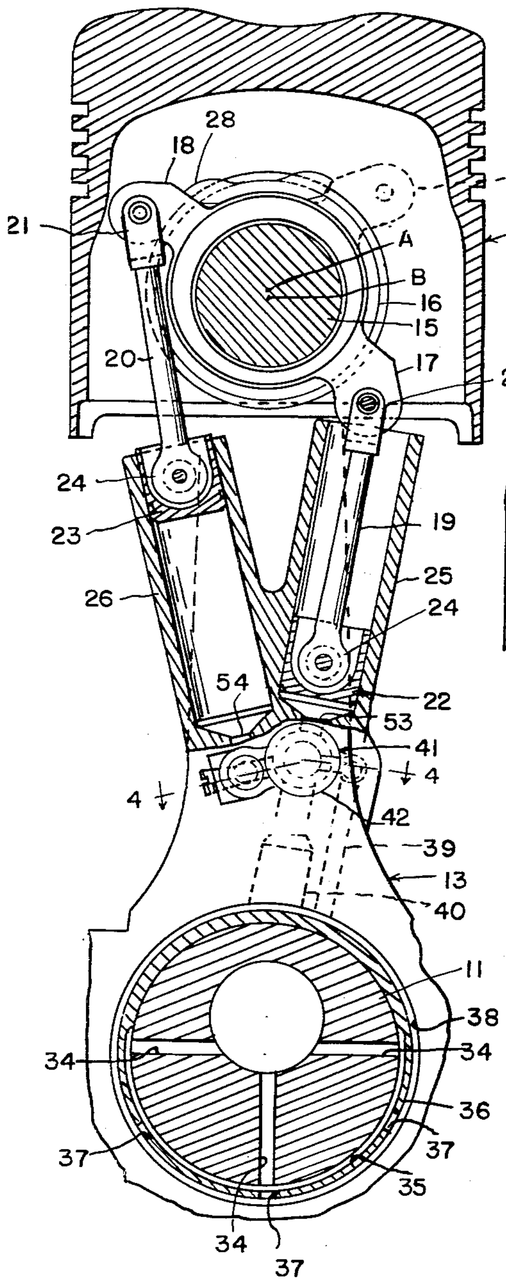
FIG. 2.



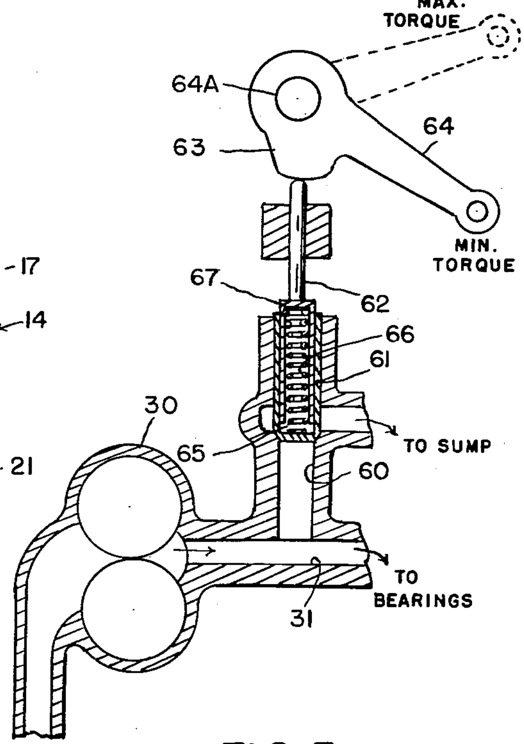


FIG. 3.

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INTERNAL-COMBUSTION ENGINE

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10 Claims. (Cl. 123—48)

My invention relates to internal combustion engines and more particularly to an engine adapted for compression ignition and having a variable compression means facilitating engine operation on various types of fuels including medium octane gasoline.

The present invention is of particular value for, but not restricted to, engines used in military vehicles in which the engines must be of low bulk and weight since the engine compartment must be protected with heavy armor plate. Each extra cubic foot of engine bulk results in a large increase in total vehicle weight which in turn requires greater engine power in order to provide the requisite maneuverability. This increases fuel consumption and to carry sufficient fuel to provide the necessary vehicle range means still greater vehicle bulk and weight.

Heretofore, it has been the practice to use spark ignition engines for combat vehicles, partially because of their lower bulk and weight. The need for low fuel consumption to increase vehicle range and to reduce logistical fuel requirements has led to increasing use of high economy compression ignition engines. However, conventional engines of this type are unsuitable for such use because of excessive bulk and weight and limitations on the attainment of high performance. Thus a need exists for a high performance, high economy engine for various applications.

The principal factor which limits the attainment of high performance in a compression ignition engine is the extremely high gas pressure developed in the cylinders on the firing stroke. Peak gas pressure is often more than twice as high as for a spark ignition engine of equivalent power, and the extremely high rate of pressure rise produces an impact effect which aggravates the destructive influence of high pressure on the engine structure.

For combat vehicles, it is highly desirable also that the engine operate on a wide variety of fuels including gasoline of medium octane which is available in greater abundance than other petroleum fuels. However, when gasoline is used in a compression ignition engine, the peak gas pressures and rate of pressure rise are still higher than with conventional compression ignition fuels.

One object of my invention is to facilitate cold weather starting of engines, particularly of the compression ignition type, by providing a means for varying the compression ratio of the engine.

Another object of the invention is to provide for more stable and smoother operation of engines at light loads by providing a means of automatically varying compression ratio in response to engine torque changes.

A further object of the invention is to reduce stresses and loads on the component parts of an engine when operated at high output by providing an improved variable compression means.

Yet another object of the invention is to facilitate construction of a high output multi-fuel engine having light weight and low bulk and yet provide long life under severe service conditions by utilizing an improved variable compression means actuated by automatically regulated changes in oil pressure.

Still another object of the invention is to provide an improved variable compression means for engines by utilizing an eccentric piston wrist pin-to-connecting rod connection operated by a hydraulic actuating means carried entirely by the connecting rod.

A still further object of the invention is to provide a simplified variable compression means for engines by constructing an integrally formed connecting rod and hydraulic actuating means operable to vary the piston-to-crankshaft dimension while maintaining constant stroke.

For a more complete understanding of the invention, reference may be had to the accompanying drawings illustrating a preferred embodiment of the invention in which like reference characters refer to like parts throughout the several views and in which:

FIG. 1 is a fragmentary cross-sectional view of an engine crankshaft, piston and connecting rod assembly incorporating a preferred hydraulic means embodying the invention.

FIG. 2 is a fragmentary cross-sectional view taken substantially on the line 2—2 of FIG. 1.

FIG. 3 is a diagrammatic cross-sectional view of a preferred fluid pressure regulating system, and

FIG. 4 is a fragmentary cross-sectional view taken substantially on the line 4—4 of FIG. 2.

In FIGS. 1 and 2, a portion of an internal combustion engine crankshaft **10** is illustrated as having a crank **11** and as carried by a bearing assembly **12**. A connecting rod **13** is rotatably carried by the crank **11** and supports a piston **14** having a wrist pin **15**.

The piston-to-crank distance is adjustable by means of an eccentric element **16** rotatably carried by the connecting rod **13** on an axis A and rotatably supporting the piston wrist pin **15** on an axis B. The eccentric element **16** has a pair of arms **17** and **18** to which are respectively secured a pair of rods **19** and **20**, preferably by means of universal joint connections **21**. The rods **19** and **20** are actuated by plungers **22** and **23** respectively, being connected thereto preferably by means of universal joint connections **24**. A stop element **28** is preferably mounted on the connecting rod **13** to limit movement of the arms **17** and **18**.

The plungers **22** and **23** respectively reciprocate in single acting hydraulic cylinders **25** and **26** which are integrally formed with the connecting rod and which are inclined outwardly as shown to facilitate drilling and reaming the cylinder bores. In operation, fluid pressure is directed to one and vented from the other of the cylinders **25** and **26**, and to actuate the eccentric **16**, shifting the position of the piston wrist pin axis B to alter the effective piston-to-crank dimension, thus varying the compression ratio of the engine. It will be seen that in the position shown in FIGS. 1 and 2, the plunger **22** is retracted and the plunger **23** extended, such that the piston **14** is at its lowest point, giving a relatively low compression ratio for normal engine operation. For starting and idling, however, fluid pressure will be directed to the cylinder **25** while cylinder **26** is vented, causing the plunger **22** to extend and plunger **23** to retract. The eccentric arm **17** will then be in the illustrated dotted line position, and the piston wrist pin axis will be raised or extended relative to the crank a distance of about ⅛ inch, which preferably effects an increase in compression ratio, in the present case, from about 15 to 21. The long stroke of the pistons will be found to give adequate leverage against operating forces acting on the piston **14** tending to alter the position of the eccentric **16**, and will be further assisted by the operation of a check valve hereinafter described.

Fluid pressure may be directed to the cylinders **25** and **26** by a hydraulic system such as the oil pressure system of the engine. In the preferred embodiment of the system, an oil pump **30**, as indicated in FIG. 3, supplies oil under pressure through suitable passages **31** to the bearing **12**, shown in FIG. 1, from whence it flows through passages **32** provided in the crankshaft **10** to a chamber **33** in the crank **11**. The oil then passes through passages **34**, at least one of which will register during rotation of the crank **11** with an arcuate groove **35** provided on the inner surface of a connecting rod bearing **36** opposite the shank of the connecting rod. The oil will thus pass through radial passages **37** through the bearing **36** to an annular groove **38** in the connecting rod and thence to passages **39** and **40** which are connected to a preferred shuttle valve mechanism **41**.

The shuttle valve mechanism **41** preferably comprises a casing **42** formed integrally with the connecting rod **13** and having a control chamber **43** and a distributing chamber **44** as illustrated in FIGS. 1 and 4.

A shuttle valve **45** is carried in the distributing chamber **44** and has one end in the control chamber **43**. Oil under pressure is ported to the control chamber **43** from the passage **39** and operates against the compression of springs **46** and **47** disposed in the shuttle valve **45** and tending to move the shuttle valve **45** to the position of FIG. 4. When oil pressure overcomes the springs **46** and **47**, the shuttle valve **45** takes the position shown in FIG. 1. Oil under pressure will then flow from the passage **40** through a non return check valve **50** into a passage **51** and to an elongated annular groove **52** provided on the periphery of the shuttle valve **45**, from which it is directed through a passage **53** to the cylinder **25**. A passage **54** connected to the other cylinder **26** is openly connected through the chamber **44** to vent openings **55** to relieve the pressure from the cylinder **26**. This position of the shuttle valve thus gives the higher compression ratio, the eccentric being moved to the position indicated by the dotted arm **17** position.

When oil pressure is lower, the springs **46** and **47** will move the shuttle valve **45** to the position of FIG. 4 in which the passage **54** leading to the cylinder **26** is registered with the annular groove **52** of the valve **45**, and oil pressure from the other cylinder **25** is vented through the other end of the distributing chamber **44** out vent ports **56** and **57**. It will be noted that since the shuttle valve **45** is finely balanced between oil and spring pressure, its axis must be parallel to the axis of the piston wrist pin **15** in order to prevent oscillating and reciprocating inertia losses of the connecting rod **13** from biasing the shuttle valve **45** in either direction.

It will be apparent that the present system preferably requires two distinct oil pressures to provide for two distinct positions of the piston **14**. The two oil pressures are provided by utilizing a pressure regulating means as shown in FIG. 3, in which the opening of a bypass passage **60** connected with oil pressure passage **31** and vented to the engine sump is selectively regulated by means of a regulating valve **61**, said regulating valve **61** having a cylindrical cavity **65** in which is inserted a compression spring **66**, the upper end of which reacts against a cup shaped piston **67** slidably mounted in cavity **65** of valve **61**. A push rod **62** presses downward on piston **67** and is actuated by a cam **63** provided on a fuel control lever **64**. The fuel control lever **64** is actuated by any desired linkage from conventional throttle controls and through a rotatable shaft **64A** operates the engine fuel metering valve (not shown) from the minimum torque position represented by the position of the lever **64** shown in solid lines to the maximum torque position represented by the dotted line position of the lever **64**. In the minimum torque position; that is, when the engine is idling or being started, the spring **66** urges the valve **67** toward a closed position so that a relatively higher oil pressure is needed to open the valve **61**, and the higher oil pressure will be directed to the shuttle valve mechanism **41**, in which case the force of the shuttle valve springs **46** and **47** is overcome and the oil pressure directed to the cylinder **25** to produce the higher compression ratio as described previously.

In the maximum torque position of the lever **64**, the spring **66** is lesser compressed and exerts a lower force on valve **61**, venting more of the pumped oil to the sump so that the oil pressure reaching the shuttle valve mechanism is lower than the force of the shuttle valve springs **46** and **47**, and the oil pressure is directed to the cylinder **26** to produce the lower compression ratio.

The cam **63** is preferably contoured with a sharply sloping curve as shown so that the oil pressure change will be relatively rapid on moving the fuel control lever **64** and can be effected at a desired intermediate stage between minimum and maximum lever torque positions.

It will also be noted that the use of the check valve **50** between passages **40** and **41** serves to lock the plunger **22** or **23** to which oil pressure is directed in its maximum upper position, and in fact inertia forces acting on the piston and connecting rod will be assisted by the check valve **50** when tending to actuate the piston **14** to its regulated position and opposed by the check valve **50** when tending to actuate the piston **14** out of the actuated position.

Although I have described only one preferred embodiment of my invention, it will be apparent to one skilled in the art to which the invention pertains that various changes and modifications may be made therein without departing from the spirit of the invention or the scope of the appended claims.

I claim:

1. A variable compression means for an internal combustion engine having a crankshaft, pistons each having a wrist pin, connecting rods respectively operably connecting said piston wrist pins to said crankshaft, and a fluid pressure system, said variable compression means comprising an adjusting means carried by each connecting rod and operable to vary the effective crankshaft-to-piston dimension whereby to vary the compression ratio of said engine while retaining constant piston stroke, hydraulic means actuating said adjusting means and operated by said fluid pressure means in response to variations in pressure thereof, and means regulating fluid pressure relative to engine torque, said connecting rod having an eccentric element rotatably carried on said piston wrist pin, said hydraulic means being connected with said eccentric element and being operable to rotate said eccentric to thereby vary the crankshaft-to-piston dimension, said hydraulic means comprising a hydraulic cylinder carried by said connecting rod, a plunger reciprocable in said cylinder and operably connected with said eccentric element to rotate same, means connecting said hydraulic cylinder with said fluid pressure means and having a valve operable to control fluid pressure flow to and from said hydraulic cylinder.

2. The variable compression means as defined in claim 1 and in which said hydraulic cylinder is integrally formed with said connecting rod.

3. The variable compression means as defined in claim 2 and in which said hydraulic cylinder is inclined outwardly with respect to the axis of said connecting rod to permit easy boring of the inside of said hydraulic cylinder.

4. A variable compression means for an internal combustion engine having a crankshaft, pistons each having a wrist pin, connecting rods respectively operably connecting said piston wrist pins to said crankshaft, and a fluid pressure system, said variable compression means comprising an adjusting means carried by each connecting rod and operable to vary the effective crankshaft-to-piston dimension whereby to vary the compression ratio of said engine while retaining constant piston stroke, hydraulic means actuating said adjusting means and operated by said fluid pressure means in response to variations in pressure thereof, and means regulating fluid pressure relative to engine torque, said connecting rod having an eccentric element rotatably carried on said piston wrist pin, said hydraulic means being connected with said eccentric element and being operable to rotate said eccentric to thereby vary the crankshaft-to-piston dimen-

sion, said hydraulic means comprising a pair of hydraulic cylinders carried by said connecting rod, a single action actuating plunger reciprocable in each hydraulic cylinder, said actuating plungers being operably connected to opposite sides of said eccentric element to rotate same in opposite directions, means connecting said hydraulic cylinders with said fluid pressure means and having a valve operable to control fluid pressure to and from said cylinders.

5. A variable compression means for an internal combustion engine having a crankshaft, pistons each having a wrist pin, connecting rods respectively operably connecting said piston wrist pins to said crankshaft, and a fluid pressure system, said variable compression means comprising an adjusting means carried by each connecting rod and operable to vary the effective crankshaft-to-piston dimension whereby to vary the compression ratio of said engine while actuating constant piston stroke, hydraulic means actuating said adjusting means and operated by said fluid pressure means, said connecting rod having an eccentric element rotatably carried on said piston wrist pin, said hydraulic means comprising a hydraulic cylinder, an actuating plunger therein connected with said eccentric and operable to rotate said eccentric to thereby vary the crankshaft-to-piston dimension, said hydraulic means including fluid pressure passages in said connecting rod and connecting said hydraulic cylinder with said fluid pressure means, and a valve carried by said connecting rod and operable to control fluid pressure flow to and from said hydraulic cylinder.

6. The variable compression means as defined in claim 5 and having check valve means in said fluid pressure

passages operable to trap fluid in said hydraulic cylinder to lock said engine piston in adjusted position against unfavorable inertia forces of said engine piston during operation.

7. The variable compression means as defined in claim 5 and in which said hydraulic cylinder is integrally formed with said connecting rod.

8. The variable compression means as defined in claim 7 and in which said hydraulic cylinder is inclined outwardly with respect to the axis of said connecting rod to facilitate easy boring of the inside of said hydraulic cylinder.

9. The variable compression means as defined in claim 5 and in which said valve comprises a casing integrally formed with said connecting rod and in which said hydraulic cylinder is integrally formed with said connecting rod.

10. The variable compression means as defined in claim 5 and in which said valve is disposed parallel to said wrist pin whereby motion of said valve is not biased by reciprocation and oscillation of said connecting rod.

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