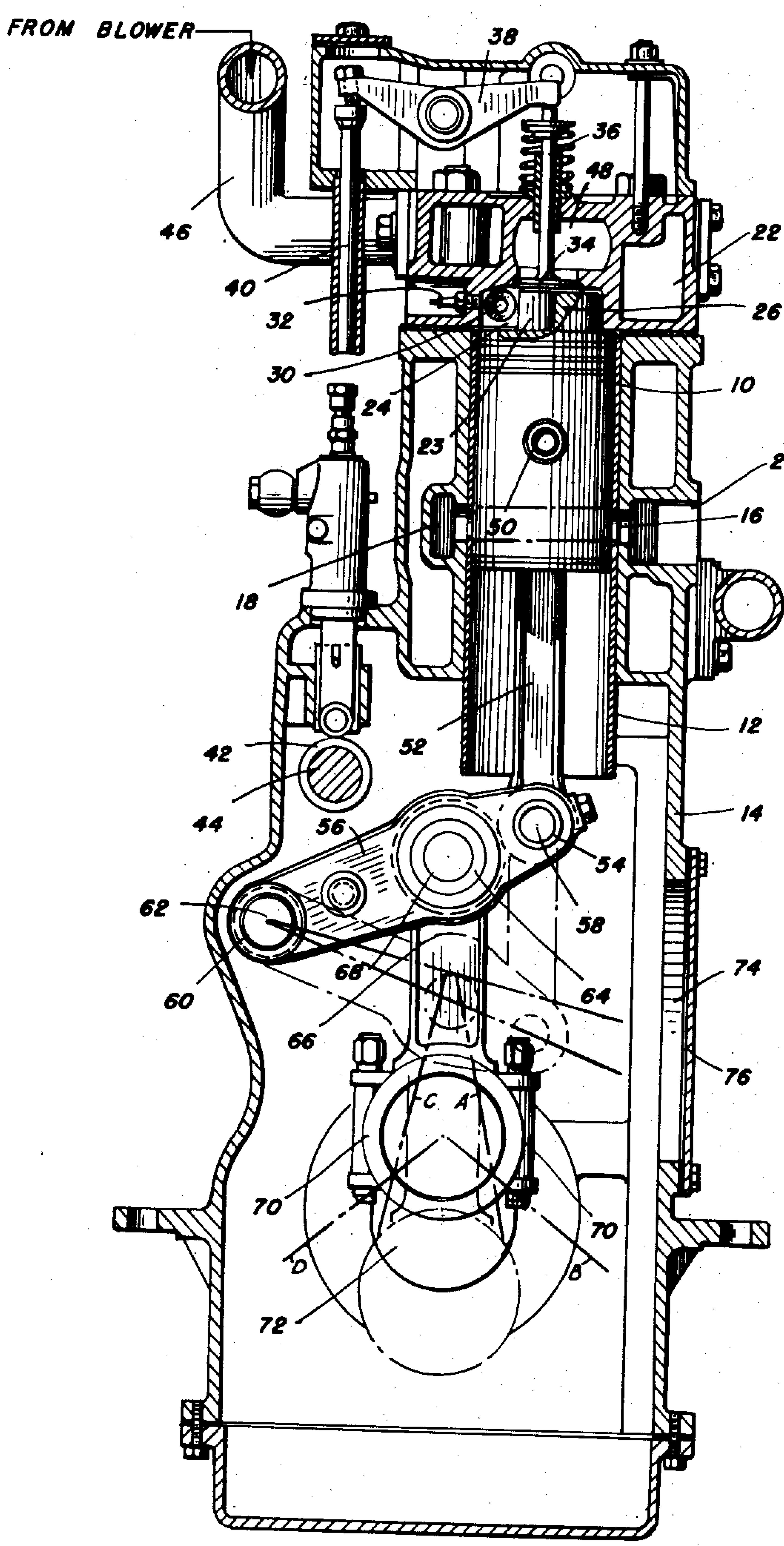
Nov. 17, 1953 C. C. CHRONIC ET AL 2,659,351

INTERNAL-COMBUSTION ENGINE

Filed Jan. 23, 1950



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Patented Nov. 17, 1953

2,659,351

UNITED STATES PATENT OFFICE

2,659,351

INTERNAL-COMBUSTION ENGINE

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Application January 23, 1950, Serial No. 139,986

2 Claims. (Cl. 123—65)

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This invention relates in general to an internal combustion engine, single or multiple cylinder, two or four cycle type, but is more particularly described as of a two cycle blower aspirated or super-charged diesel or semi-diesel type in which the crank shaft and piston are connected by a lever mechanism. This lever is pivoted and connected to the crank shaft and to the piston rod upon a substantially definite ratio and is so located that it lies approximately at a 90° angle to the cylinder bore at its mid-travel position and the piston rod is substantially parallel to the center line of the cylinder at each end of the stroke.

Although internal combustion engines embodying the leverage principle have been built and used experimentally, the prior use resulted in the use of the lever as an end in itself. It did not alter in any way the source of the power potential which is determined by the volumetric efficiency. The basic improvement resulting from the prior art was the reduction of friction between the piston and the cylinder wall and the tendency to a mechanical advantage in concentrating a longer stroke into a much shorter crank pin travel.

This invention exceeds and improves the prior art in the construction whereby the leverage hookup as disclosed to make its operation practical, is merely a means used, together with other novel structural features, to alter and improve the conditions essential to a greater power output at the source which is directly reflected in a superior scavenging cycle in two cycle diesel operation. Furthermore the prior art is objectionable because the weight combinations, the driving ratios and proportions have not been adopted and set forth in accurate relationship for satisfactory results. This prevented the accomplishment of the true potential of this engine in the way of power output and unrestrained operation. The leverage principle, because of the mass and weight involved in the linkage is not compatible with that of an inherently high speed construction such as the typical gasoline engine which attains its higher power output at high crank shaft speeds. The critical inertia force inherent in the operation of a lever at high crank shaft speeds as well as the reciprocal unbalance caused by the added weight produces a combination that results in excessive vibration or breakage of parts.

Another objection heretofore encountered has been the inability to recognize and overcome the need to reduce the size of the connecting rod

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upper end bearing which would in turn reduce the mass and weight of the lever itself as well as the mass and weight of the connecting and piston rods. This prior construction held to the theory of obtaining maximum piston travel to minimum crank throw which necessitates a very large upper end connecting rod bearing with still excessive bearing pressure at this point causing a restrained action in the entire cycle of operations.

To overcome these serious objections, the ratios and proportions of the entire power linkage are changed in this invention so that the resulting action may reflect a pronounced power increase and a sharp reduction of the inertia forces, removal of excess bearing pressures, and producing a free and unrestrained operating action.

Among the advantages of the present lever construction are the practical elimination of piston slap caused by the angularity of the connecting rod in the conventional internal combustion engine, the practical elimination of the wear on the piston, piston rings, and cylinder walls caused by pressure of the piston on the cylinder walls as occurs in the conventional engine, and the reduction of the friction by relieving the wall pressure thereby further aiding in the development of a high even torque over a broad range of speed; and improving the accessibility and ease of servicing.

A primary object of this invention is to secure maximum volumetric or scavenging and mechanical efficiency by obtaining more advantageous piston travel when the crank pin is in the best position to transmit this power, to produce minimum movement of the piston while the exhaust ports or valves are open, as it applies to the two cycle diesel type, to obtain superior scavenging, and to obtain maximum volume of intake air when the crankpin is likewise in the most advantageous position. No internal combustion engine can have a greater power potential than that which is made possible by the amount of air it can trap within the cylinder to support combustion. In two cycle diesel operation, this is called scavenging. It is the displacement of burned or burning gases with fresh oxygen bearing air. The present construction takes advantage of a longer dwell of the piston at the bottom of the stroke. By so doing, a complete reversal of construction practice is made in which the ports at the bottom of the cylinder are used for the exhausting while a valve is used in the head for the air intake and is so timed that a longer and practical scavenging cycle is thereby an accomplished fact. This construction permits a higher scavenging efficiency than heretofore known and thereby results directly in a superior power development.

By making the cylinder ports the exhaust outlet and the valve in the head the air intake, it follows that the life of this valve is greatly increased over that of a valve used to release burning exhaust gases, and at the same time a marked advantage is further obtained by such placement of the air intake that the incoming and cooler air cools the vital parts which thereby reduces detonation and gives longer life to these critical parts.

Another primary object of this invention is to improve combustion process cycles in two cycle diesel operation.

A further object of prime importance is a sharp reduction of the high inertia forces inherent in the leverage principle.

Another necessary object is to produce an unrestrained action in the leverage movement.

A further object of the invention is to secure more efficiency in supercharging and to gain additional volumetric efficiency.

In obtaining these objectives and other advantages, this invention sets forth a new and improved set of ratios, proportions and combinations of the integral parts of the leverage mechanism and other pertinent functioning parts of the engine to give the most advantageous engine performance, to relieve stresses and strains within the engine and to obtain low bearing pressures.

Other objects of the invention will appear in the specification and will be apparent from the accompanying drawing in which the single figure is a sectional view of an internal combustion engine, in accordance with this invention.

Referring now more particularly to the drawing, an internal combustion engine having many more or less conventional parts is shown in the drawing and comprises a piston **10** reciprocable in a cylinder shell **12**, suitably mounted in a vertical casing **14**. Intermittent the ends of the cylinder are discharge openings **16** communicating with a discharge manifold **18** and having a gas discharge opening **20** at one side of the casing.

At the top of the cylinder is a cylinder head **22** suitably secured to the casing in which is a recess **24** for receiving a projection **26** of the piston which also has a recess **28** at one side thereof to cooperate with a portion of the recess **24** to provide a reduced combustion chamber. Cooperating with the combustion chamber are a conventional fuel inlet (not shown), an energy cell **30** and a starting glow tube or spark plug **32** in accordance with our application for Turbulence Chamber for Internal Combustion Engines, Serial No. 126,824, filed November 12, 1949.

At the top of the combustion chamber is an air intake valve **34** having a stem **36** engaged by one end of a pivoted rocker arm **38**, the other end of the arm having operating means including a rod **40** actuated in a conventional manner by a cam **42** on an engine cam shaft **44**. An air inlet pipe **46** is suitably connected to a chamber **48** outside of the valve **34** to admit air through the valve to the combustion chamber in a well known manner.

Pivotally connected internally to the piston by a bearing **50** of moderate size is a piston rod **52**. The lower end of the piston rod is connected to a suitable bearing **54** at the outer end of a lever **56** on a center line **58** near the outer end of the lever. The other end of the lever is connected by a bearing **60** mounted in the casing and upon an axis **62** such that when the lever is in its substantially horizontal position, the piston and the piston rod are substantially at the center of their strokes.

Connected to the lever **56** intermediate the ends and near the outer end is a bearing **64** for the upper end of a connecting rod **66** which is located upon an axis **68** parallel to the axes **58** and **62** of the piston rod and lever respectively. The lower end of the connecting rod **66** is connected to a bearing **70** suitably mounted upon an engine crank shaft **72** in a well known manner, the position of the connecting rod being substantially vertical when the piston is at both opposite limiting positions.

In one side of the casing **14**, opposite the movable end of the lever **56** is an opening **74** normally closed by a plate **76** through which the lever and its connections with the ends of the piston rod **52** and the connecting rod **66** are accessible for insertion, removal, adjustment and repair.

In the two cycle diesel engine, to obtain a more advantageous ratio between the piston movement during the power stroke and during the scavenging interval as well as the movement during the interval of air intake, to the rotation of the crank while the exhaust is open, the proportions of the reciprocating parts are established which likewise accomplishes the result of obtaining maximum pressure on the crank pin when it is in the most advantageous position to accept and transmit this thrust, that is, during the power stroke.

At the bottom of Fig. 1, line A represents the approximate angular position of the connecting rod **66**, and line B the angular location of the crank shaft relation to its center and this position represents the location of the top of the piston at the top of the exhaust opening **16** prior to its opening. The line C represents the angular position of the connecting rod and the line D its angular connection with the motor crank at the time of the exhaust valve closing. The angle subtended by the lines B and D is approximately 103½° of the crank travel. During this time, the piston has moved only about 15% of its total movement while the exhaust is open approximately 29% of the crank rotation. This ratio permits the attainment of a greater volumetric efficiency by permitting a longer power stroke followed immediately by a pronounced dwell of the piston while the crank is in movement of 103°. Because of this pronounced piston dwell, the exhaust ports or valves open for more efficient scavenging which in turn provides greater capacity for the entering oxygen bearing air through the intake valve or ports.

By receiving the air intake through a valve in the head and discharging the exhaust near the bottom of the cylinder, a remarkably free and unrestricted exhaust exit is made possible because of the unusual and novel timing of the air intake valve and the freedom of the exhaust gas exit, the exhaust cycle can be reduced to a new and superior minimum duration. This reduction in the exhaust cycle is automatically reflected in an increased duration of the power cycle which is also of highly superior duration. This extra length of power stroke permits a more complete burning within the cylinder before exhausting. This condition is reflected in a remarkable non-stalling characteristic of the engine.

These highly superior and novel timing ratios of approximately 110° (as interpreted upon the crankshaft) for the scavenging stroke, 120° for the compression, 130° for the power stroke and 103° for the exhaust. Such highly desirable timing ratios have not heretofore been obtainable.

The air inlet valve is timed to open approximately 5° of crank shaft travel after the exhaust ports are uncovered by the movement of the piston and closes approximately 16° after the exhaust ports are closed. This designed lag in the inlet valve closing permits a blower or a supercharger to force additional air into the cylinder after the exhaust ports are closed. In this way, supercharging may be effected if desired.

The operation of this engine is in a well known two cycle of the diesel type, each down-stroke is a pressure stroke and each up-stroke is a compression stroke. The proportions of the operating parts for the most satisfactory results should have the following limits: The length of the lever **56** expressed in distance between the centers **62** and **58** of the extreme end bearings should be held within the limits of 1.25 to 1.60 times the length of the piston stroke. The ratio in the present form of the invention is approximately 1.34 times the piston stroke. This ratio may be changed within the limits mentioned depending upon the performance characteristics desired.

The distance from the center of the bearing on the fulcrum end of the lever to the center **68** of the bearing for the upper end of the connecting rod should not be less than 58 percent of the total distance between the bearing centers **62** and **58** located at the extreme ends of the lever. In the present form, the center **68** is two-thirds or 66⅔ percent of the distance between the centers **62** and **58**.

In the two cycle blower aspirated or supercharged type of diesel engine, the length of the piston rod should not exceed 1.90 times the stroke of the piston nor be less than 1.40 times the stroke of the piston. In the present form, the ratio is 1.70 times the piston stroke. This may vary within the limits depending upon the design of the cylinder ports and of the piston itself. To further minimize any restraint in the complete leverage linkage, the included angle formed by the connecting rod approximately at the extreme horizontal positions of the crank throw should not be more than 34° nor less than 25°. In the present form, the included angle is 29½° which may be varied within the specified range depending upon variations of design.

The lever linkages proportioned within the foregoing limits overcome the three major handicaps predominant in the prior art. In the present engine, this gives (1) a greater freedom in rotative and reciprocating movement; (2) a greater reduction and equalization of bearing pressures; and (3) a smaller and more compact lever which has an appreciable reduction in reciprocating weight. The use of these proportions and ratios produces a free running and more powerful engine per cubic inch displacement than heretofore produced by such an engine with or without benefit of the lever combination. A supercharged or blower aspirated two cycle diesel engine would thus prove more advantageous for use of the leverage mechanism than a 4 cycle type of engine as there is downward pressure

upon the piston at all times thereby minimizing the extreme stresses caused by the heavy inertia forces. Because a diesel engine is essentially a heavy duty type working under very high pressures, the mass is heavier and more rugged, thereby accommodating to a better advantage the greater inertia and reciprocating weight forces inherent in the lever linkage. For the industrial type engine, the low and medium speed range may be taken as up to approximately 1600 crank shaft revolutions per minute.

Thus all the factors surrounding the principle, cycle and construction of the diesel type of engine makes an ideal combination for the low and medium speed lever construction as described and shown herein producing thereby a power plant of high horse-power output with very smooth running characteristics.

While a lever engine having the construction and specified limits of certain of the operating parts has been described in some detail, it should be regarded by way of example and illustration rather than as a limitation or restriction of the invention as various changes in the construction, combination and arrangement of the parts as well as metallurgical advancements such as the use of titanium alloys permitting higher speeds may be made without departing from the spirit and scope of the invention.

We claim:

1. In an internal combustion engine of the lever type having a piston with an extending pivoted piston rod, a crank shaft having a crank with an extending pivoted connecting rod and a lever pivotally mounted at one end and pivotally connected to the ends of the piston rod and the connecting rod, the connecting rod being pivoted to the lever at a distance from the pivot of the lever substantially two-thirds of the distance of the pivot of the piston rod from the pivot of the lever, the arrangement being such that the connecting rod and the piston rod are substantially parallel to each other and to the axis of the piston at both limits of the stroke of the piston and the axis of the crank shaft is in the vertical plane of the connecting rod when said rod is at either of the limits of the stroke of the piston an air inlet valve in the cylinder head of the engine, and exhaust port openings in the lower portions of the cylinder and uncovered when the piston is near the lower end of its stroke at which position a slower piston movement is obtained to provide a longer scavenging cycle, a relatively shorter exhaust cycle, and an expansion stroke of greater duration than the compression stroke.

2. In an internal combustion engine of the lever type in accordance with claim 1, in which the distance from the center of the bearing at the pivoted end of the lever to the center of the bearing for the upper end of the connecting rod on said lever is not less than 58% of the total distance between the said bearing centers at the extreme ends of the lever.

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