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PNEUMATIC DIFFERENTIAL PISTON FOR INTERNAL COMBUSTION ENGINES

Filed March 15, 1935

2 Sheets-Sheet 1

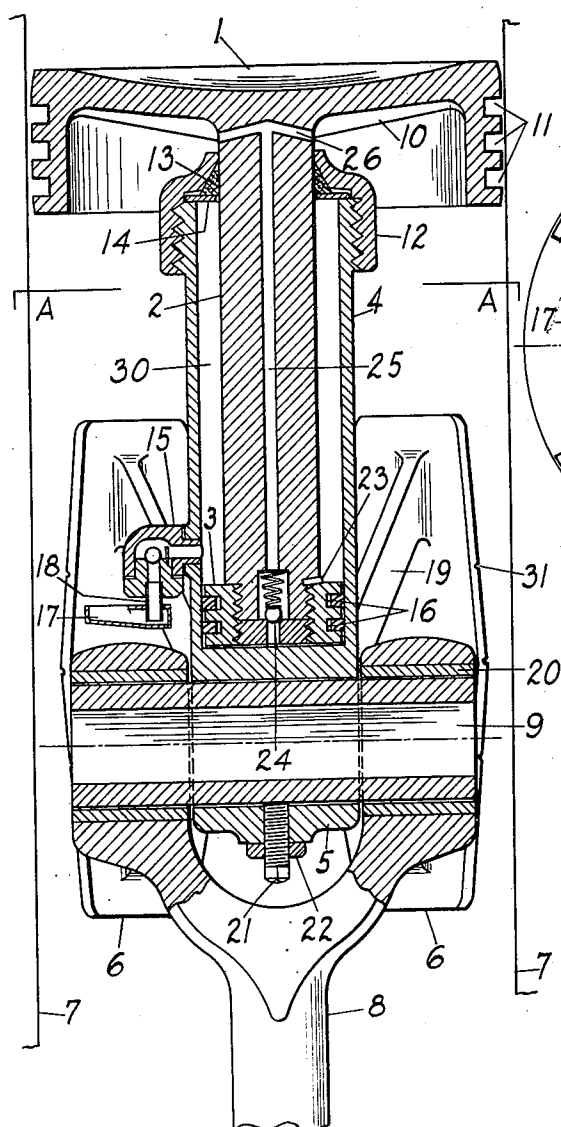


FIG. 1

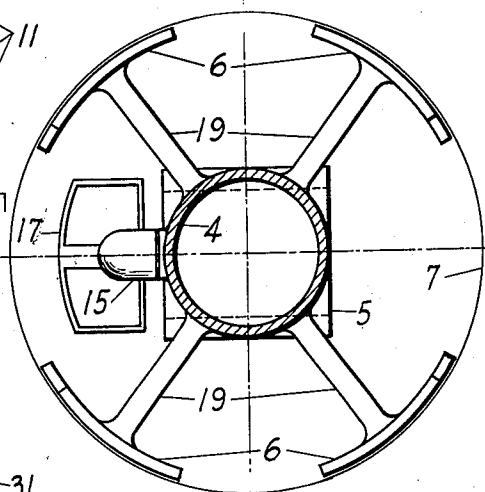


FIG. 2

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2 Sheets-Sheet 2

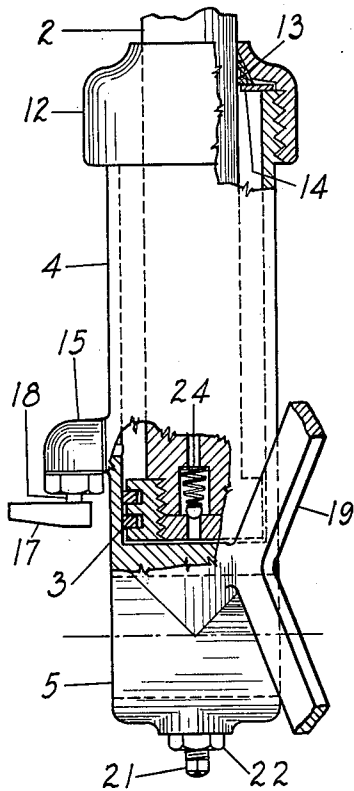


FIG. 3

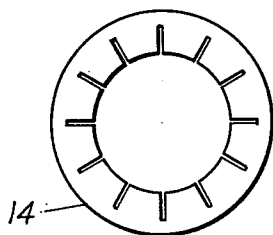


FIG. 5

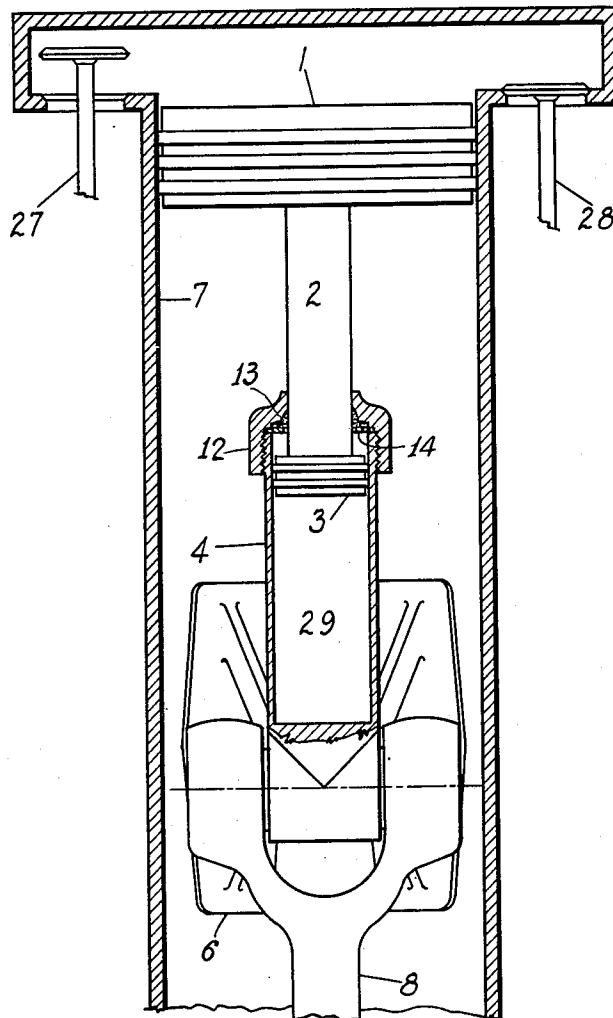


FIG. 4

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UNITED STATES PATENT OFFICE

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PNEUMATIC DIFFERENTIAL PISTON FOR
INTERNAL COMBUSTION ENGINES

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2 Claims. (Cl. 123—78)

This invention relates to pistons for internal combustion engines of the four and six stroke cycle type, and in particular, relates to a piston, a dash-pot device, a cross-head and a connecting rod, which, operating together, serve to produce strokes of variable length.

Briefly described, the invention consists of a piston having a piston-rod cast integral therewith, with a pneumatic double-acting piston at the rod's other extremity. Said piston, piston rod and pneumatic piston have no mechanical or fixed attachment to the cross-head, connection being had through pneumatic means alone. The pneumatic piston reciprocates within a pneumatic cylinder which is a part of the cross-head, said cross-head being constrained to move within the engine cylinder by guides. The cross-head is journaled to the connecting-rod by means described hereinafter and shown in the drawings.

The purpose of this piston, known as the pneumatic differential piston, is to enable a more complete expansion of the initial charge, down to atmospheric pressure if desired. As is well known, the most serious deficiency in internal combustion engines to date is the lack of sufficient expansion. This results in a number of undesirable effects, among which are, wasted energy through the exhaust; back pressure through the exhaust amounting to approximately 5% of the net output; excessively high exhaust temperatures resulting in overheating and burned valves; noisy operation requiring elaborate muffling devices, and, not the least important is the incomplete combustion of the fuel, resulting in carbon deposits and air pollution. These effects can largely be overcome by more complete expansion and a very large saving in fuel can thus be accomplished.

Up to the present time engine manufacturers, experimenting with engines using greatly increased expansion of the initial charge, have found that the theoretical gains in output and economy have been more than offset by the increase in friction losses. This being the main point to overcome in a practical manner, I will outline a means for overcoming same with a theoretical engine, having the pneumatic differential piston.

Assuming a conventional four stroke cycle engine, in which all four strokes per cycle are of equal length, (hereinafter called engine #1), drawing in a given volume of fuel at each suction stroke and developing a given horsepower at a given number of R. P. M.

Then assume a second four stroke cycle engine

(hereinafter called engine #2) having the pneumatic differential piston, with an arbitrary cylinder area of, say, twice the cylinder area of engine #1, and drawing in the same volume of fuel at each suction stroke as engine #1. Both engines have the same ratio of compression. It will then be seen that the intake or suction stroke of engine #2 will be one-half the length of stroke in engine #1. Then, since the pneumatic differential piston makes short suction and compression strokes and long expansion and exhaust strokes, the burned gases in engine #2 are expanded to approximately twice the volume of the admitted charge. Since the expanding gases in engine #2 act upon a piston having twice the area of the piston in engine #1, the horsepower developed per cycle in engine #2 will be considerably greater than that developed in engine #1. Then, in order that engine #2 develop a horsepower output equal to that of #1, the R. P. M. of #2 will be proportionately less than that of engine #1 to achieve that result. In any case engine #2 will operate with greater economy and less friction per horsepower produced than engine #1.

It will be found that the piston rings (the point of greatest friction in the cylinder) of engine #2 pass over but 12% greater area per cycle than the rings of #1 per cycle without, however, being subject to side thrust.

This is due to the fact that the cylinder area traversed in engine #2 by a stroke equal in length to the stroke of engine #1 would be approximately 1.4 times the area traversed per stroke in #1. The suction and compression strokes of #2, being one-half the length of stroke in #1 will traverse an area approximately 0.7 times that traversed during those strokes in #1. The expansion and exhaust strokes in #2, being slightly longer than the corresponding strokes in #1, will traverse an area approximately 1.54 times that traversed during those strokes in #1, thus leaving a net traversed area per cycle in engine #2 of approximately 1.12 times that traversed in engine #1. The cross-head guides of #2, although comprised of arcuate segments, also pass over a larger area per cycle than the skirt in #1, but there is opportunity for more efficient lubrication in #2, which therefore, has less friction. In both engines an equal quantity of fuel mixture is compressed but in engine #2 compression does not start until the piston has passed more than half way up in the full stroke of the cross-head and the angularity of the connecting rod is diminishing, therefore, friction in the cylinder and other related engine friction during compression is less in #2 than

in #1. In consideration of all this and other hereinafter mentioned sources of friction economy the total friction loss per cycle of engines #1 and #2 should very nearly balance, either way. Since the expansion in engine #2 is carried farther than that in engine #1, the M. E. P. of #2 will be the lesser, but, since for an equal horsepower output engine #2 operates with less friction and exhausts against a lesser back pressure, it will have a greater mechanical efficiency than engine #1, which will be reflected to the benefit of its M. E. P.

In addition, a practice now made necessary by high exhaust pressure, that of opening the exhaust valve a very considerable time before the piston reaches bottom dead center, need not be followed. The lead could be considerably shortened, according to the design.

The above illustration is no doubt an extreme application of the pneumatic differential piston, due to the assumed greatly increased cylinder area and assumed greatly reduced speed making necessary a much larger engine than one to be had by a more moderate design. Such a moderated design would achieve approximately proportionate results to the above by using a more moderate increase in cylinder area, a more moderate decrease in R. P. M. and a reduction in fuel consumption per cycle in comparison to the conventional engine.

For most efficient operation, the force of the expanding charge should be delivered through the piston to the connecting-rod, in a direct line, axially of the piston and with a minimum of frictional resistance by all parts. This is accomplished by my invention. The piston and piston rings are not subjected to side thrust at any time, and the piston is free to rotate naturally within the cylinder, thus allowing for self-adjustment to any irregularities of the cylinder walls and promoting equal wear.

The working piston, piston rod and pneumatic piston are of aluminum alloy, the cross-head and guides preferably of steel, although any other suitable metals may be employed. The cross-head guides are provided with a plurality of peripheral oil-grooves on their bearing surfaces and may be machined to very close limits of cylinder clearance since they are separated from direct contact with the piston. The guides, as will be seen, are formed in pairs as a letter H, each pair receiving thrust at alternate strokes. By transmitting the thrust through the struts, each at an angle, an equal projected bearing area may be had with less material and weight than can be had with the usual type of trunk piston skirt, and a better distribution of wear of the cylinder walls is obtained.

Recommended embodiments of my invention are illustrated in the accompanying drawings, in which

Figure 1 is a vertical section of the piston assembly.

Figure 2 is a transverse section of the cross-head pneumatic cylinder at A-A without the connecting rod being shown.

Figure 3 is a fragmentary view of the cross-head.

Figure 4 is a diagrammatic view of the relative position of the piston during the suction stroke, at the time of opening the fuel inlet valve.

Figure 5 is a plan view of the packing spring washer.

Referring to the drawings:

Figure 1. The working piston 1 is provided with a plurality of peripheral grooves 11 for the reception of piston rings (not shown) and a plurality of reinforcing ribs 10, the contour of the face being optional according to the engine design. The piston-rod 2 is cast integral with the piston 1 and has secured to its other extremity by screw threads, a pneumatic double-acting piston 3 locked in place by a cotter 23, and provided with a plurality of grooves 16 for the reception of piston rings. The piston-rod also has a spring ball release valve 24 fitted within the rods extremity and connecting with drilled passage 25, said passage 25 leading to a plurality of ducts 26.

The cross-head 5 is provided with guides 6 and struts 19 cast integral. The guides 6 have a plurality of peripheral oil grooves 31 for the maintenance of proper lubrication. The cross-head 5 is further provided with a pneumatic cylinder 4 cast integral. The cylinder 4 is fitted near its lower end with an automatic ball intake valve 15, said valve 15 being provided with a suggested means for lubrication consisting of an oil duct 18 and an oil reservoir 17 supplied by the splash system. At the upper end of the cylinder 4 there is a packing cap 12, packing spring washer 14 and metallic packing 13. The spring washer 14 has radially cut slots extending from its inner circumference half-way to the outer circumference and exerts a pressure upward against the packing 13. The packing cap 12 is secured as shown by screw threads.

The connecting rod 8 has bushings 20 in its forked ends and is articulated with the cross-head 5 through a wrist pin 9, said wrist pin 9 being held firmly in place by a set screw 21, and lock nut 22. It will be evident that the wrist pin 9 may thermally expand and contract at will without causing distortion of any other part, a fault common to pistons using wrist pins held at the ends, thus eliminating cylinder scoring due to piston distortion.

In use, the piston operates as follows:

At the start of the intake, or suction stroke, with the piston 1 and the cross-head 5 in close proximity (as shown in Figure 1) at top dead center, the fuel inlet valve 27 will remain closed and the exhaust valve 28 will be closing. The cross-head 5 will then descend, but since both valves 27 and 28 are closed, a decrease in pressure will result in the compression space, due to the piston 1 being drawn downward by the vacuum or suction caused between the piston 3 and the cross-head 5. When the drop in pressure in the compression space has become sufficient (only a slight drop being necessary) to overcome the vacuum behind piston 3, the main piston 1 will cease its descent. With the cross-head in motion downward the entrapped air within the air chamber 30 will be compressed, imparting a smooth silent and increasing effect to the movement of the pistons 1 and 3. When the piston 3 passes the inlet port from valve 15 the vacuum will be broken and air chamber 29 will fill with air, said air drawing in lubricating oil from reservoir 17. When the cross-head 5 has descended to the relative position as shown in Figure 4, with an arbitrary air pressure in chamber 30, the fuel inlet valve 27 will then open, admitting the fuel mixture. Piston 1 will then descend in the usual manner drawing in a charge of fuel mixture. It will be seen that due to the design of the packing cap 12 and washer 14, the packing 13 will afford a tight fit about the piston rod 2 during

the compression stroke of the piston 3 in the chamber 30, but will not bind on the return stroke. The length of stroke made by the piston 3 within the cylinder 4 is determined by the desired ratio of expansion.

During the following compression stroke the piston 1 will remain virtually idle while the cross-head 5 ascends, compressing the air within chamber 29 until a sufficient pressure has developed to open the release valve 24. This pressure does not require any additional work on the part of the engine since it is but the resultant of the compression pressures within the engine cylinder, and the escaping air will have a cooling effect on the piston rod 2. The cross-head 5 is brought into easy, silent contact with piston 3 and the entire assembly then proceeds, with the piston face 3 in close proximity to the cross-head 5, to the end of the compression stroke. Any loss of air in chamber 30 is then replaced through inlet valve 15, which again carries lubricating oil. The expansion or power stroke and the following exhaust stroke are carried out in the usual manner, with the piston 1 descending the full stroke with the cross-head 5. Upon completing the exhaust stroke, the piston 1 is restrained from leaving the cross-head 5 (due to its inertia) by the aforesaid vacuum, or suction developed between piston 3 and the bottom of the cylinder 4 integral with the cross-head 5. Said vacuum is aided by the resistance of the exhaust gases. If there should be a slight movement by the piston 1 into the compression space, it would be but momentary and would aid the scavenging process.

It will be seen that the piston 1 can be removed from the cylinder by removing the cylinder head and gland cap 12, thus avoiding any disturbance of the oiling means or the connecting rod. Suitable means for inserting a removing tool can be provided in the piston face.

I am aware that prior to my invention pistons have been made with piston rods, to operate with cross-heads, and I, therefore, do not claim such a combination broadly, but

I claim:

1. The combination, in an internal combustion engine, of a working piston, and a second reciprocatory member, a dash-pot device comprising a pneumatic cylinder, a piston rod extending through one end of the cylinder and connected at one end to the working piston and at the other end to a double-acting piston adapted to reciprocate in said cylinder, and cooling means for said rod comprising a relief valve suitably disposed at one end of the piston rod.

2. The combination, in an internal combustion engine, of a working piston, and a second reciprocatory member, a dash-pot device comprising a cylinder, a piston rod extending through one end of the cylinder and connected at one end to the working piston and at the other end to a double-acting piston adapted to reciprocate in said cylinder, and single means for the admission of air to either side of the double-acting piston whereby said piston is under restraint of compressed air during strokes in both directions, and means for the release of air from said cylinder.

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