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INTERNAL COMBUSTION ENGINE

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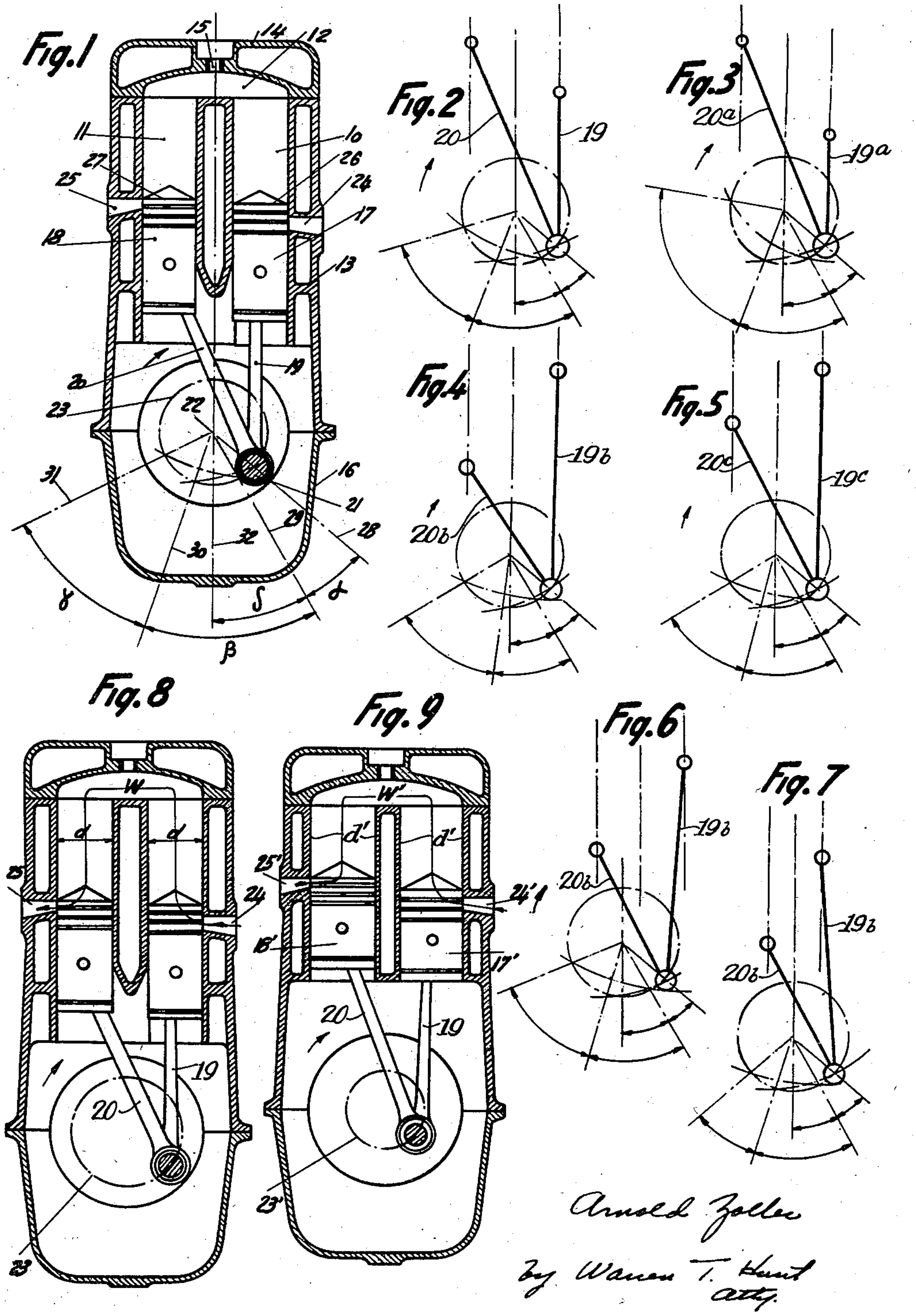

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INTERNAL COMBUSTION ENGINE

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This invention has for its object improvements in and relating to two stroke internal combustion engines.

In two stroke internal combustion engines in which the gases of the combustion are expelled out of the cylinders through the fresh charge of mixture introduced into the cylinders, losses are frequently caused by the fact that the exhaust ports are not yet closed when the burnt gases have already been expelled out of the cylinders. In such case parts of the fresh mixture escape into the atmosphere without having been burnt, i. e. without having been utilized, until a further escape of the fresh mixture is prevented by the closure of the exhaust ports.

Thus such a two stroke internal combustion engine works with the minimum of losses when the exhaust ports are closed by the control gear at the time when all the burnt gases have been expelled out of the cylinders and when the fresh mixture just reaches the exhaust ports.

The time which elapses or the angle which the crank describes from the beginning of the admission to the moment where the fresh mixture reaches the exhaust ports is a function of the scavenging or charging pressure, which itself is a function of the size of the admission and exhaust ports, on the one hand, as well as of the dimensions of the cylinders and, on the other hand, of the pressure lasting on the mixture before the latter enters the cylinder or cylinders through the admission ports. Another dependency relation is present between the size of the admission sections and the time or the crank angle during which the admission ports must still remain open after the closure of the exhaust ports if an overpressure of the charge in the cylinder equal or corresponding to the overpressure existing in advance of the admission ports before the beginning of the compression is desired.

With the known arrangements with predetermined sizes for the exhaust and admission ports the optimum can be obtained only for a predetermined speed or only for a very small speed range of the engine, because when the speed increases the so-called time-sections, that is the products of time and size of the sections of the exhaust or admission ports which are momentarily free, become smaller. Thus, if the sections of the ports are very large, i. e. if they are designed for great speeds, fresh mixture will escape through the exhaust ports at small speeds, thus producing losses of mixture, the engine having then a relatively small power and working uneconomically in the range of small speeds. On the other hand, if the sections of the admission and exhaust ports are small, the engine being then designed in order to work with a high commercial efficiency at small speeds, the charge resistances will be very high at high speeds with this result that the pressure of the charge or the weight of the charge in the engine cylinder will be small or that the power consumption of the charging pump will be very high.

Arrangements with control gears which are controllable during the operation and through which the above mentioned inconveniences may be avoided are less reliable in operation and more expensive than the normal control gears of simpler design.

One object of the invention is to avoid the inconveniences of the known control gears which are uncontrollable in operation for two stroke internal combustion engines, as much as possible, and more particularly to provide in a so-called U-shaped engine, i. e. an engine of the known type with twin cylinders connected together through a common combustion chamber, the one of which comprising the admission ports and the other the exhaust ports, large or normal time-sections without substantial losses of fresh gases at small speeds and without subtantially larger losses of charge at high speeds.

Another object of the invention is to provide a larger supercharging angle in an engine of the above mentioned kind and in spite of this a closure of the exhaust ports in due time and consequently an early beginning of the compression.

A further object of the invention is to avoid the inconveniences of engines of the above mentioned kind by means of connecting rods of different lengths for the two pistons of the twin cylinders, but with the aid of equal crank circles for the two connecting rods with the same spacing of the two cylinder axes from the crank shaft axis.

Still another object of the invention is to provide an improvement by designing an engine of the above mentioned kind so that the ratio: piston diameter to piston stroke is reduced to 1:4 or less with respect to the usual ratio of approximately 1:3, which may be achieved by making the stroke of each piston of the twin cylinders about 1½ times as great as its diameter.

A form of execution of the invention is shown by way of example in the appended drawing, in which

Figure 1 is a sectional view of an internal combustion engine with a twin cylinder according to the invention through the middle of the cylinder, all the parts which are not essential for the invention being left away.

Figures 2 to 5 are diagrammatical views showing the different sizes of the scavenging and charging angles for different length ratios for the connecting rods.

Figure 6 is a diagrammatical view corresponding to Figure 4 but with this difference that the middles of the cylinders are unequally spaced from the middle of the crank shaft axis.

Figure 7 is a diagrammatical view corresponding to Figure 4 but with this difference that the middles of the cylinders in Figure 7 are at a smaller distance from the crank shaft axis than in Figure 4.

Figure 8 is the same figure as Figure 1 in opposition to

Figure 9, in which the ratio: piston diameter to piston stroke has the usual value of approximately 1:3 in the whole.

In Figure 1 both two cylinders 10 and 11 have the common chamber of combustion 12. The cylinders 10 and 11 are enclosed in a common jacket 13, which comprises, in its upper part, the head 14 with an aperture 15 for inserting the spark plug (not shown) which eventually has to be inserted there, and to the lower part of which the crank case is adapted in the usual manner.. In the cylinders 10 and 11 are slidably mounted the pistons 17 and 18 which are operatively connected with the crank pin 21 through the connecting rods 19 and 20. 22 is the axis of the crank shaft, not shown, and 23 is the crank circle. 24 is an admission port in the cylinder 10, through which the fresh mixture may be introduced into the cylinder through any suitable means, not shown, and 25 is an exhaust port in the cylinder 11, through which the burnt gases may escape into the atmosphere. 26 is the edge of the piston 17 which may free and close the admission port and 27 is the edge of the piston 18 which can free and close the exhaust port when the pistons reciprocate through the movement of the crank pin 21 on the circle 23.

28 is the radius of the crank circle 23 on which the middle of the crank pin is, when the edge 27 begins to uncover the exhaust port 25; 29 is the radius corresponding to the opening of the admission port 24 through the edge 26 and 30 and 31 are the radii which correspond to the closure of the openings 25 and 24 respectively. The radius 32 corresponds to the lowest position of the crank pin 21.

In the range of the angle alpha between 28 and 29 there takes place thus only an exhaust of the burnt gases; in the range of the angle beta between 29 and 30 the scavenging takes place through fresh mixture simultaneously entering through the opening 24, while in the range of the angle gamma between 30 and 31 there is only an introduction of mixture into the cylinder and the supercharging takes place.

As shown, in Figure 1 the exhaust connecting rod 20 is longer than the admission connecting rod 19 and in the example shown the relation is 1:0.9.

The following Figures 2 to 5 show how the values of the scavenging angle beta and of the supercharging angle gamma vary when the relation between the lengths of the two piston rods 20 and 19, *20a* and *19a*, *20b* and *19b*, *20c* and *19c*, is a different one. For this purpose in all the Figures 1 to 5 the values of the angle alpha between the radii 28 and 29 and of the angle delta between the radii 29 and 32 are the same, so that the openng of the exhaust takes place 50° in advance of the opening of the admission 30° in advance of the lowest position of the crank. Thus, the angle alpha has a value of 20° and the angle delta has a value of 30°. The middles of the cylinders are disclosed at 19' and 20'.

In Figures 6 and 7 the length relations between the two piston rods *20b* and *19b* are the same as in Figure 4, but in Figure 6 the middles of the cylinders are differently spaced from the crank shaft axis, while in Figure 7 the distance between the cylinder middles is smaller than in Figure 4 in order to show also the influence of this variation.

The resulting relations are thus the following ones:

Fig.	Length of piston rod		Scavenging angle	Supercharging angle
	Exhaust	Admission		
1	1	0.9	49°	44.5°
2	1	0.7	49°	52°
3	1	0.5	49°	79°
4	0.7	1	35°	53°
5	0.85	1	44°	44°
6	0.7	1	47.5°	51° Cylinder middles differently spaced from crank shaft axis
7	0.7	1	46.5°	32° Distance of cylinder middles reduced

Therefore, the more different from another the lengths of both rods 19 and 20 are made within certain limits, the greater will be the supercharging angle which can be obtained under substantially similar conditions and particularly with equal time-sections for admission and exhaust. On the other hand, a relative increase of the exhaust piston rod 20 will entail an increase of the scavenging angle beta. An increase of the scavenging angle may also be obtained by arranging the middle of the exhaust cylinder 11 nearer to the crank shaft axis 22 than the middle of the admission cylinder 10, or by reducing the distance of the middles of the cylinders 10 and 11 from each other.

Comparing then for instance the arrangements according to Figures 1 and 5, there will be seen that for the same charging pressure of the mixture and the same time-sections for admission and exhaust, the supercharging will be also approximately the same for all speeds, because the supercharging angles are also approximately the same in both cases. But since the scavenging angle in Figure 5 is smaller than in Figure 1, in the engine shown in Figure 5 gases of the mixture will be able to escape through the exhaust, which is still open, only at speeds which are lower than the speeds at which gases of the mixture may already escape in the case of Figure 1. On the other hand, an increase of the supercharging angle through a relative diminution of the length of the admission piston rod with respect to the exhaust piston rod, as for instance according to Figure 3 with respect to Figure 1, will be preferable if it is desired, above all, to reduce the power consumption of the device for pressing the mixture into the cylinder.

In Figure 8, which corresponds to Figure 1, the diameters d of the pistons are smaller than the diameters d' of the piston in Figure 9, while the crank circle 23 in Figure 8 is greater than in Figure 9. Therefore, the way w which the gases of the mixture have to travel over from the admission port 24 is longer than w'. Therefore if it is necessary, for some reason, to have a great scavenging angle, it will be well to make the engine with a long stroke (Figure 1 or 8) contrary to the usual engine with a short stroke according to Figure 9, in which the ratio: diameter d' to the whole stroke of both pistons 17' and 18' does not exceed 1:3. Then, for the same pressure of the mixture, the gases of said mixture reach again the exhaust port, which is not yet entirely closed, at speeds which are lower than they would in the engine with a short stroke as in Figure 9, and it is thus possible to avoid losses of fresh gases with or without supercharging. But it is thus possible to obtain a greater power with a greater commercial efficiency which is thus profitable to the acceleration capacity of, for instance, a motor vehicle driven by the engine. For this purpose it has been found convenient to make the ratio between the whole piston stroke, that is the sum of the strokes of both pistons, to the piston diameter not smaller than 4:1 per se or in combination with piston rods of a convenient length.

Many possibilities of changes will be evident for those skilled in the art. More particularly, the various numerical values have been given only by way of examples and it is not intended to limit the scope of the invention to said values, reference being had, in this respect, to the appended claims.

I claim:

1. In a two stroke internal combustion engine of the U-shape kind with twin cylinders connected together through a common combustion chamber, a crank shaft having crank pins, one of said cylinders having an admission port for the scavenging and charging mixture and the other an exhaust port for the burnt gases, means for controlling the opening and closure of the admission and exhaust ports, including pistons having connecting rods pivotally connected with the same crank pin, the piston rod of the piston for the cylinder with the exhaust port having a greater length than the piston rod of the piston for the cylinder with the admission port.

2. In a two stroke internal combustion engine of the U-shape kind with twin cylinders connected together through a common combustion chamber, one of said cylinders having an admission port for the scavenging and charging mixture and the other an exhaust port for the burnt gases, pistons in the cylinders controlling the admission and exhaust ports, piston rods connected with the pistons and acting on the crank shaft in the same size crank circles, the piston rod of the piston for the cylinder with the exhaust port having a greater length than the piston rod of the piston for the cylinder with the admission port, and the middle line of one of the cylinders being at a greater distance from the crank shaft axis than the middle line of the other cylinder.

3. In a two stroke internal combustion engine of the U-shape kind with twin cylinders connected together through a common combustion chamber, one of said cylinders having an admission port for the scavenging and charging mixture and the other an exhaust port for the burnt gases, pistons in the cylinders controlling the admission and exhaust ports, piston rods connected with the pistons and acting on the crank shaft in the same size crank circles, the piston rod of the piston for the cylinder with the exhaust port having a different length than the piston rod of the piston for the cylinder with the admission port, and the middle line of one of the cylinders being at a greater distance from the crank shaft axis than the middle line of the other cylinder.

4. In a two stroke internal combustion engine of the U-shape kind with twin parallel cylinders connected together through a common combustion chamber, one of said cylinders having an admission port for introducing the scavenging and charging mixture to both cylinders and the other having an exhaust port for discharging the burnt gases from both cylinders, pistons of the same diameter in the cylinders controlling the admission and exhaust ports, a shaft having a crank pin, and piston rods of different lengths pivotally mounted about the axis of the crank pin, the ratio between the sum of the strokes of both pistons to the diameter of the pistons being not smaller than 4:1.

5. In a two stroke internal combustion engine of the U-shape kind with twin cylinders connected together through a common combustion chamber, one of said cylinders having an admission port for the scavenging and charging mixture and the other an exhaust port for the burnt gases, pistons in the cylinders, connecting rods pivotally connected with the pistons and the same crank for controlling the admission and exhaust ports, the piston rod of the piston for the cylinder with the exhaust port having a greater length than the piston rod of the piston for the cylinder with the admission port, the ratio between the sum of the strokes of both pistons to the diameter of the pistons being not smaller than 4:1.

6. In a two cycle internal combustion engine having a crank shaft, a pair of parallel cylinders connected by a common combustion chamber, the axes of said cylinders being on opposite sides of the crank shaft axis, one of said cylinders having an intake port, the other of said cylinders having an exhaust port, a shaft having a crank pin, pistons in said cylinders for controlling said ports, and connecting rods of unequal lengths connected to said pistons and pivotally mounted about the axis of said crank pin, the longer of said rods being connected to the piston controlling the exhaust port.

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