Aug. 1, 1939. J. EHRLICH 2,168,096

TWO-STROKE ENGINE

Filed Dec. 3, 1937 3 Sheets-Sheet 1

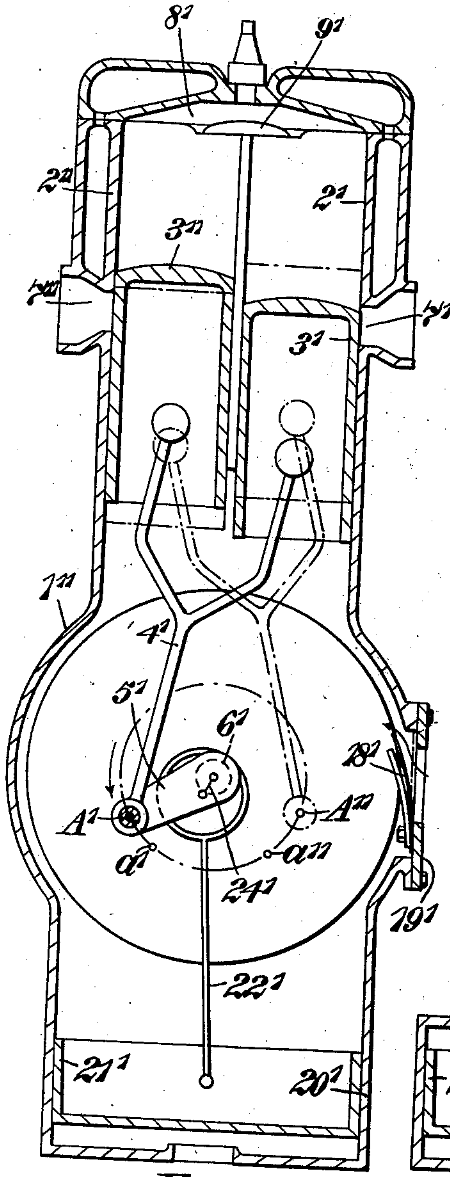

Fig. 1

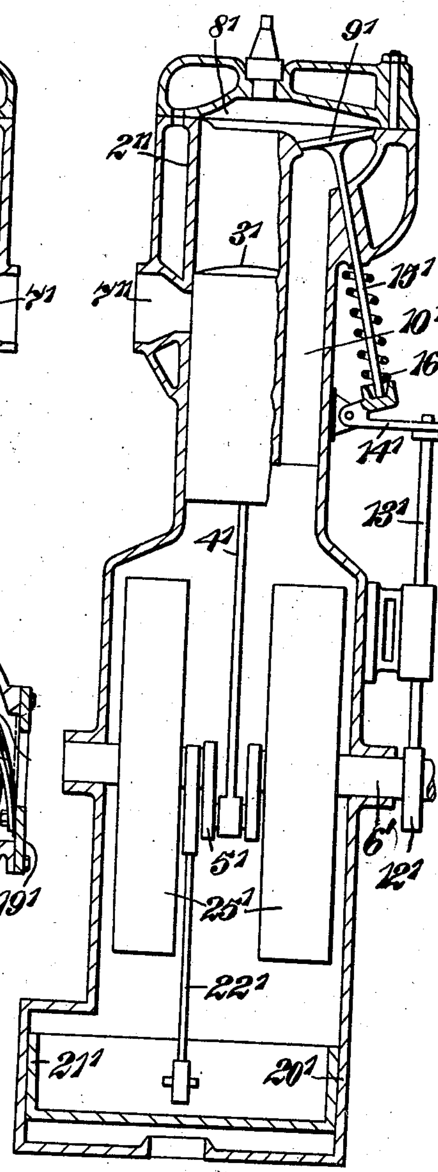

Fig. 2

J. Ehrlich
Inventor

By: Glascock Downing & Seebold
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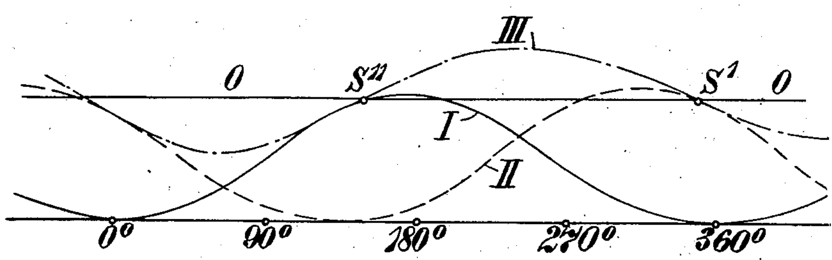

Fig. 3

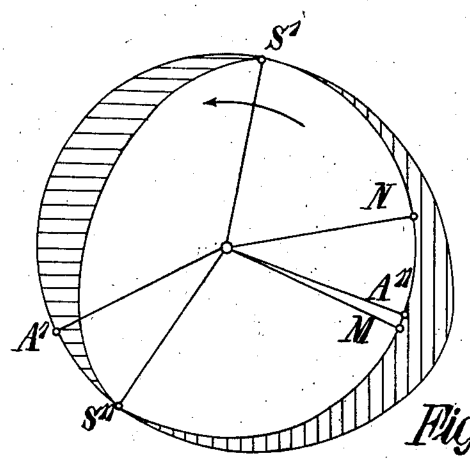

Fig. 4

J. Ehrlich
INVENTOR

By: Glascock Downing & Seebold
Attys.

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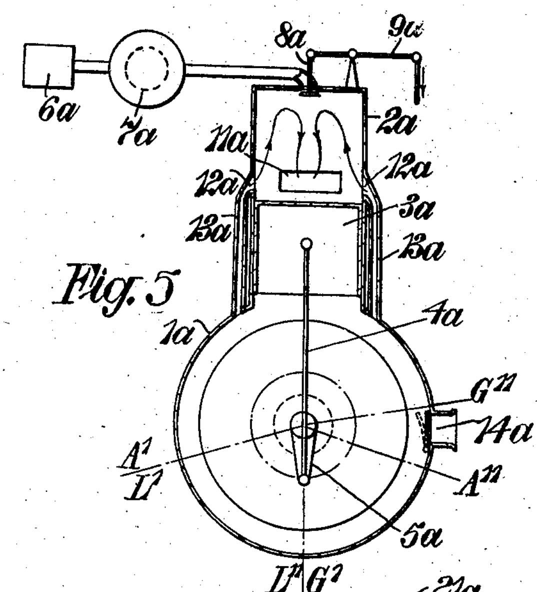

Fig. 5

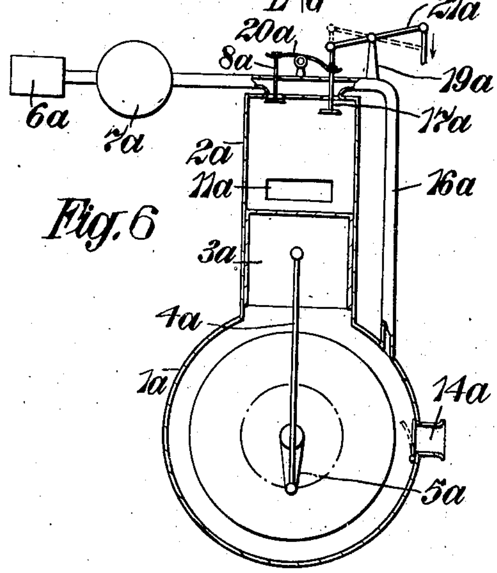

Fig. 6

J. Ehrlich
Inventor

By: Glascock Downing & Seebold
Attys.

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2,168,096

TWO-STROKE ENGINE

Josef Ehrlich, Klosterneuburg, Austria, assignor to Works Development Company, Limited, London, England

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4 Claims. (Cl. 123—53)

This invention relates to two-stroke internal combustion engines of the type which has twin cylinders connected with a common combustion space.

In known engines of this type inlet valves or ports are arranged to supply combustible mixture to one cylinder whilst the burnt gases after expansion are released by ports when the latter are uncovered by the piston in the other cylinder.

In another known engine of this type two pistons act upon the so-called "little" end of a common connecting rod and exhaust ports are controlled by the pistons in each cylinder in exactly the same manner and with identical timing.

The object of the present invention is to provide an engine of this type which is simple and efficient.

The present invention consists in a two-stroke internal combustion engine comprising two power cylinders, a common combustion chamber, a piston in each of said cylinders and exhaust ports overrun by the piston in each of said cylinders, wherein the strokes of the pistons are so arranged that the total period of opening of the exhaust ports is greater than the period of opening of the exhaust ports opened by one of the pistons.

Preferably the exhaust periods of the two cylinders are caused to overlap to such an extent that the total open exhaust period is at least 25 per cent. greater than the open exhaust period of each cylinder for which purpose the two pistons may be connected to a common crank arm on a crankshaft.

In order to improve the efficiency, precompressed air, which may be provided by crank case compression in the case of a pair of cylinders or a blower when more than two cylinders are concerned, may be caused to flow into the cylinders, at a very low pressure, from the commencement of the exhaust period until the commencement of charging the cylinders with combustible mixture.

The combustible mixture is brought by a compressor, driven by the engine itself or by independent means such as an electric motor, to such a pressure that it enters the cylinders both during the exhaust period and after the exhaust ports are closed.

The invention may be carried into effect by way of example as will now be described, reference being had to the accompanying diagrammatic drawings.

In these drawings:—

Figures 1 and 2 show sectional elevations at right angles to each other through a form of engine according to the invention.

Figures 3 and 4 show diagrams of the volume displacement in the crank case of the engine shown in Figures 1 and 2.

Figures 5 and 6 show diagrammatically in section a modified form of the engine shown in Figures 1 and 2 wherein the crank case compresses air for interposition between the burnt gases and the incoming charge whilst the combustible mixture is fed from a compressor into the head of the cylinder, the auxiliary piston in the crank case being omitted for the sake of clearness.

Referring to the drawings the engine according to Figures 1 and 2 has a crank casing **1''**, two blocked cylinders **2'** and **2''** with pistons **3'** and **3''** guided therein, which pistons are connected by a forked connecting rod **4'** to the crank arm **5'** of the crank shaft **6'**. **7'** and **7''** are the exhaust ports provided in the cylinders **3'** and **3''**. The admission of the live gas takes place in the space **8'** of the common cylinder head through the valve **9'** and the connecting duct **10'** which leads to the crank casing. The control of the valve **9'** is effected by a cam **12'** mounted on the crank shaft **6'** which cam acting through the push rod **13'** of the bell crank **14'** forces the valve spindle **15'** at the desired moment upwards into the open position whilst the spring **16'** effects the return movement into the closed position. The diaphragm **18'** controls the suction of the live gas from the carburettor into the crank casing, its extreme swing being limited by a supporting plate **19'**. The position of the suction valve **18'**, shown in Figure 1, is the open position, corresponding to a partial vacuum in the crank casing, in which position the live gas enters the crank casing around the diaphragm in the direction of the arrows. When there is a pressure in the crank casing, the diaphragm **18'** will occupy the closing position indicated by a dot and dash line. At the bottom of the crank casing **1''** is provided an auxiliary cylinder **20'** in which an auxiliary piston **21'** slides, the stroke volume of which is approximately equal to or somewhat greater than the total stroke volume of the two power pistons. The connecting rod **22'** of the auxiliary piston is connected to a crank arm **24'** of the crank shaft, which is so offset with respect to the crank arm **5'** that the greatest displacement of the auxiliary piston (top dead centre position) lags behind the greatest displacement through the power cylinders (bottom dead centre position) by about 135°, which, in view of the oppositely directed motion of the pistons, corresponds to an actual offset of about 45°. **25'** are two fly wheels.

This engine operates in the following manner: After the ignition and explosion have taken place, the piston **3'** will lead in its downward motion and in the position A' (shown in full lines in Figure 1) will open the exhaust port **7'**; at the point *a'* the exhaust port **7''** is opened by the lagging piston **3''**. During the subsequent upward stroke the piston **3'** first closes the port **7'** in the point *a''* and the final closure is effected in the point A'' by the piston **3''** (position shown in dot and dash lines). Through this overlapping of the exhaust periods the total exhaust period is increased by about 25% as compared with the exhaust period with a single cylinder alone. Consequently the height of the exhaust ports themselves may be reduced and thus an increased power stroke volume can be obtained with otherwise the same dimensions.

The auxiliary piston **21'** effects on the one hand an increase of the pressure produced in the crank casing which, however, does not signify an increase in power as the energy required for this purpose must be taken from the crank shaft. On the other hand, however, it produces a displacement of the maximum volume displacement produced by the pistons and consequently of the pressure maximum in the crank casing, as illustrated in Figure 3. In this diagram is plotted on the abscissa axis the path of the crank pin of the power piston and on the ordinate axis the volume displacement, the outer dead centre position being given as to be equal to a volume displacement of the value zero. The curve I gives the volume displacement of the power piston and the curve II of the auxiliary piston both in dependence on the path of the crank pin of the power pistons. The curve III shows the total volume displacement of the two pistons. It is assumed that the power pistons have a stroke volume of 250 cm. and the auxiliary piston a stroke volume of 270 cm. In the total volume displacement it will be seen that the maximum has a lag of about 66° with respect to that of the power pistons. Assuming that the lower half of the curve III corresponds to the suction and the upper half to the charging, we get below the line 00 the suction period and above the line the charging period. For the sake of clearness the crank circle of the power piston is again shown in Figure 4 to twice the scale and the course of the curve of the volume displacement III is shown, as far as there is a suction action, within the crank circle and, so far as there is charging action, outside the crank circle in the form of the curve IV, the ordinates being in this case plotted radially above and below the straight line 00. It would be seen that the increase in pressure, on the exhaust ports first opening, commences at the point S'', reaching its maximum about 66° after the inner dead centre position of the power pistons, and when about at the point S' the suction action begins. If this curve be examined with respect to the scavenging and charging of the engine, the following will be seen:—

In the period between the points S' and S'' the suction of the live gas into the crank casing through the diaphragm-controlled valve **18'** takes place. The exhaust commences independently thereof in the point A' and continues to the point A'' (see Figure 1). From the point S'' onward, the pressure in the crank casing increases and reaches its maximum in the point M of the crank circle. At the same moment, that is just before the closing of the exhaust ports, the charging valve **9'** opens, whereby, owing to the high pressure in the crank casing, a blowing out of the remaining gases will take place, which lasts only a short time, but is very strong. Losses of live gas are prevented through the rapid subsequent closing of the exhaust at the point A''. The valve **9'** remains open up to the point N, as, in spite of the closing of the exhaust ports, the live gas will continue to flow across until there is a higher pressure in the crank casing than in the cylinders (subsequent charging). Only in the point N will the ascending power pistons cause the increasing pressure in the cylinders to equal that in the crank casing and charging will then cease.

The advantages obtained through this displacement of the pressure maximum in the crank casing are therefore as follows:—

An extremely long exhaust period, a rapid scavenging of the remaining exhaust gases during the last portion of the period during which the exhaust ports are open, without any loss of live gas, and a prolongation of the charging period for a considerable time after the closing of the exhaust ports. As tests have shown, these effects together produce a considerably improved efficiency.

It is also possible to effect the suction of the live gases by piston-controlled ports and to control the inlet valve by a slide valve or to use simply a pressure-controlled non-return valve.

The two-stroke internal combustion engine according to Figure 5 has an air-tight crank casing **1a**, the cylinder **2a** forming a continuation upwards of the same. The piston **3a**, the piston rod **4a** and the crank **5a** are in this case shown in the inner dead centre position. The live gas passes from the carburettor **6a** to a compressor **7a** and thence to the inflow valve **8a** which is actuated in the usual manner by control members **9a**. In the cylinder, exhaust ports **11a** are provided in the usual way and at the same height as these ports there are further inlet ports **12a** which are in communication through ducts **13a** with the crank casing **1a**. On the crank casing there is provided a non-return valve **14a** which operates automatically with a membrane for the admission of the outside air.

The mode of operation is as follows:—

After the ignition and explosion the piston opens towards the end of the power stroke simultaneously the exhaust ports and the admission ports **12a** for the scavenging air at the point A'L' of the crank circle, A' indicating the commencement of the exhaust and L' the commencement of the admission of air. As the air in the crank casing is compressed during the downward stroke, this air will flow, owing to the inclined direction of the ducts **13a**, simultaneously with the exhaust of the exhaust gases towards the middle upwards into the cylinder and will cause a thorough whirling or eddying through and scavenging of the cylinder. When the bottom dead centre position is reached, the inflow of scavenging air through the ports **12a** ceases, as is indicated by the reference letter L'', since the pressure in the crank casing now begins to drop again. At the same time the control **9a** opens the valve **8** and allows the live gas to flow in, which is indicated by the reference letter G'.

The inflow of live gas continues not only up to the closing of the exhaust ports in the point A'' of the crank circle, but continues owing to the pressure produced by the compressor 7a to the point G''. The position of this final point of the inflow is determined by the degree of compression in the compressor.

Since in this mode of operation a layer of fresh air is always blown in between the burnt gases and the freshly flowing in live gas, a good scavenging of the cylinder of exhaust gases is insured and losses of live gas are avoided. Whereas the greater part of the air escapes out of the cylinder again between the points L'' and A'' through the exhaust ports, the smaller part will become mixed with the entering live gas. It should be noted that the scavenging air is heated both by the compression in the crank casing and while flowing through the ducts 13a, which has an accessory effect, constitutes a desired cooling of the cylinder, the air through being mixed with the live gas being thus capable of completely vaporising any particles of fuel which have not yet been vaporised and thus bringing about a complete utilisation of the fuel.

In the constructional example shown in Figure 6, the analogous parts of the engine bear the same reference numerals as in Figure 5. The difference of this construction as compared with that of Figure 5 consists in this that the scavenging air is supplied to the cylinder not through piston-controlled ports but through a control valve 17a provided at the cylinder head. The valve casing is therefore connected by way of a passage 16a with the crank casing. The control 19a actuates both valves 8a and 17a with the aid of a spring (hairpin spring) 20a which connects the two valve spindles with one another over a fixed point of rotation.

The engine works in the same way as in the constructional form shown in Figure 5, the only difference being that the succession of the inflow of scavenging air and live gas is effected through the valves 8a, 17a by means of the control 19a. In the position shown in the figure, the valve 17a is open whilst through the tension of the spring 20a the valve 8a is pressed firmly on to its seat. On the bottom dead centre position being passed the control lever 21a will move into the position shown by a broken line, in which position it closes the valve 17a and allows the valve 8a to open under the pressure produced in the compressor 7a. At the point G'' of the crank circle, the lever 21a moves into the intermediate horizontal position in which both valves are closed and are pressed by the initial tension of the spring 20a sufficiently firmly against their seating.

It will be obvious that the manner of control of the admission of the scavenging air may be varied in many ways without exceeding the scope of the invention.

I claim:

1. A two-stroke internal combustion engine comprising two power cylinders provided with an inlet port and directly connected by a common combustion chamber, a piston in each of said cylinders, exhaust ports overrun by the piston in each of said cylinders and means connecting the pistons to a common crankshaft, the arrangement being such that the total period of opening of the exhaust ports is greater than the period of opening of the exhaust ports opened by each of the individual pistons, means for compressing the working mixture and means for introducing the compressed mixture into the cylinders both during and after the period during which the exhaust ports are open.

2. A two-stroke internal combustion engine comprising two power cylinders, a common combustion chamber, a piston in each of said cylinders, exhaust ports overrun by the piston in each of said cylinders and means connecting the pistons to a common crankshaft, the arrangement being such that the total period of opening of the exhaust ports is greater than the period of opening of the exhaust ports opened by each of the individual pistons, a blower for compressing the working mixture and means for introducing the compressed mixture into the cylinders both during and after the period during which the exhaust ports are open.

3. A two-stroke internal combustion engine comprising two power cylinders, a common combustion chamber, a piston in each of said cylinders, exhaust ports overrun by the piston in each of said cylinders and means connecting the pistons to a common crankshaft, the arrangement being such that the total period of opening of the exhaust ports is greater than the period of opening of the exhaust ports opened by each of the individual pistons, and a blower driven independently of the engine for introducing the compressed mixture into the cylinders both during and after the period during which the exhaust ports are open.

4. A two-stroke internal combustion engine comprising two power cylinders, a common combustion chamber, a piston in each of said cylinders, exhaust ports overrun by the piston in each of said cylinders and means connecting the pistons to a common crankshaft, the arrangement being such that the total period of opening of the exhaust ports is greater than the period of opening of the exhaust ports opened by each of the individual pistons, crank case compression being used to cause precompressed air to flow into the cylinders at very low pressure between the commencement of the exhaust period and the commencement of the charging period.

JOSEF EHRLICH.