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(54) **TWO-STROKE ENGINE TRANSFER PORTS**

ÜBERSTRÖMSCHLITZE VON ZWEITAKTMOTOR

ORIFICES DE TRANSFERT POUR MOTEUR DEUX TEMPS

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Description**BACKGROUND OF THE INVENTION**

1. Field of the Invention

[0001] The present invention relates to internal combustion engines and, more particularly to a transfer system.

2. Background Art

[0002] U.S. Patent 6,367,432 discloses a two-stroke cycle internal combustion engine which has a quaternary Schnurle-type scavenging system that is configured such that the capacity of a pair of second scavenging passageways are made larger than the capacity of a pair of first scavenging passageways, so that during the descending stroke of the piston, air is allowed to be introduced into the combustion actuation chamber from the second scavenging passageways prior to the introduction of the air-fuel mixture and at the same time, a relatively large quantity of air is allowed to be introduced into the combustion actuating chamber from the first scavenging passageways over a longer period of time as compared with the period of time in which air is introduced from the second scavenging passageways.

[0003] U.S. Patent 6,223,705 discloses a two-stroke internal combustion engine having a Schnurle scavenging system includes a pair of first scavenging ports and a pair of second scavenging ports. An inner horizontal scavenging angle formed close to an exhaust port and an outer horizontal scavenging angle formed remote from the exhaust port by a pair of scavenging flows blown out of the pair of the first scavenging ports are both set to an angle in the range of from 116 to 124 degrees. An inner horizontal scavenging angle formed close to the exhaust port and an outer horizontal scavenging angle formed remote from the exhaust port by a pair of scavenging flows blown out of the pair of the second scavenging ports are set to angles in the ranges of from 126 to 135 degrees and from 146 to 154 degrees, respectively.

[0004] WO96/31691 discloses a cylinder for a two-stroke combustion engine intended for a handheld working tool, wherein the exhaust port's mouth inside the cylinder as well as the cylinder's transfer ports are arranged symmetrically around a common symmetry plane which follows the cylinder's symmetry axis.

[0005] DE4447215 discloses an internal combustion engine with a first scavenging port which opens into the cylinder with an upward inclination, to direct a first volume of the scavenging gas upwards into the head section of the cylinder. A second scavenging port is positioned to direct a second gas volume at right angles to the cylinder towards a cylinder wall section which contains the first scavenging port.

[0006] DE19707767 discloses an engine including a cylindrical block. The cylinder block divides the exhaust path into two sections. Symmetrically formed scavenging exhaust paths are formed opposite to each other in the top and bottom portions of the cylinder block.

[0007] US5870981 discloses an engine in which there are a plurality of main channels and a plurality of support channels open into the combustion chamber for conducting fuel mixture into the combustion chamber. A first main channel and a first support channel are arranged on a first side of the center axis and a second main channel and a second support channel are arranged on a second side of the center axis. The first and second main channels have exit windows lying adjacent the exhaust window so as to cause the respective component main flows of fuel mixture to exit approximately horizontally toward the surface portion lying opposite the exhaust window.

[0008] US6223705 discloses a two-stroke internal combustion engine having a Schnurle scavenging system including a pair of first scavenging ports and a pair of second scavenging ports. An inner horizontal scavenging angle formed close to an exhaust port and an outer horizontal scavenging angle formed remote from the exhaust port by a pair of scavenging flows blown out of the pair of the first scavenging ports are both set to an angle in the range of from 116 to 124 degrees. An inner horizontal scavenging angle formed close to the exhaust port and an outer horizontal scavenging angle formed remote from the exhaust port by a pair of scavenging flows blown out of the pair of the second scavenging ports are set to angles in the ranges of from 126 to 134 degrees and from 146 to 154 degrees, respectively.

[0009] Because of increasing government pollution emissions standards, there is a continuing need to lower engine emissions in two-stroke engines. One of the sources of emission problems has been the discharge of unburned hydrocarbons due to short circuiting of fuel out of an exhaust port during an upward stroke of the piston before the exhaust port is closed. Thus, there is a need to minimize the loss of fresh, short circuit fuel exiting out of the exhaust. This minimization can result in lower hydrocarbon emissions and higher fuel economy.

SUMMARY OF THE INVENTION

[0010] In accordance with one of the present invention, there is provided a two-stroke internal combustion engine comprising; a cylinder; and a piston movably mounted in the cylinder, wherein the cylinder comprises an exhaust port and transfer ports, wherein the transfer ports comprise a first pair of the transfer ports disposed closer to the exhaust

port than a second pair of the transfer ports which are disposed further away from the exhaust port, wherein respective forward and rearward transfer passage side walls of the first pair of transfer ports are angled relative to each other at a first angle of 70 to 85 degrees and respective forward and rearward transfer passage side walls of the second pair of transfer ports are angled relative to each other at a second angle of 120 to 150 degrees; and the transfer ports extend outwardly from a main internal area of the cylinder into interior side walls of the cylinder and extend upward from proximate a bottom of the cylinder to a middle section of the cylinder; wherein directional discharge of scavenged air out of the transfer ports establishes a flow path for the scavenged air to minimize losses of fresh unburned fuel into the exhaust port.

[0011] Directional discharge of scavenged air out of the transfer ports establishes a flow path for the scavenged air to minimize losses of fresh unburned fuel into the exhaust port.

BRIEF DESCRIPTION OF THE DRAWINGS

[0012]

FIGURE 1 is a diagrammatic view of an internal combustion engine incorporating features of the present invention; FIGURE 2 is a cross sectional view of the cylinder of the engine shown in Figure 1; FIGURE 3 is a cross sectional view of the cylinder shown in Figure 2 taken along line 3-3; FIGURE 4 is a partial side elevational view of the side of the cylinder shown in Figure 2 showing the exhaust port; FIGURE 5 is a diagrammatic view of a portion of an internal combustion engine comprising an alternate embodiment of the present invention; FIGURE 6 is a cross sectional view of the cylinder shown in Figure 5 taken along line 6-6; FIGURE 7 is a cross sectional view of the cylinder shown in Figure 5 taken along line 7-7; FIGURE 8 is a diagrammatic view of a portion of an internal combustion engine comprising another alternate embodiment of the present invention; FIGURE 9 is a diagrammatic view of a portion of an internal combustion engine comprising another alternate embodiment of the present invention; and FIGURE 10 is a timing chart illustrating the exhaust and transfer port open area relative to piston position in crank angle degrees for the present invention compared to a prior art design.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENT(S)

[0013] Referring to Figure 1, there is shown a partial diagrammatic view of an internal combustion engine 10 incorporating features of the present invention. Although the present invention will be described with reference to the exemplary embodiments shown in the drawings, it should be understood that the present invention can be embodied in many alternate forms of embodiments. In addition, any suitable size, shape or type of elements or materials could be used.

[0014] The engine 10 is a two-stroke engine having a cylinder 12, a piston 15, a crankshaft 16, a crankcase 18, a fuel delivery system 20, and an ignition system 22. One type of specific application for the engine 10 could be in a small high speed two-stroke engine such as utilized in a hand-held power tool, such as a leaf blower, string trimmer, head trimmer, chain saw, etc.

[0015] The ignition system 22 generally comprises a spark plug 24 and an electrical generating system 26 connected to the spark plug 24. However, in alternate embodiments, any suitable type of ignition system could be used. The ignition system 22 is generally well known in the art.

[0016] The fuel delivery system 20 generally comprises a carburetor 28, an air filter 30, a main air inlet 32 into the cylinder 12, and a fuel and air inlet 33 into the bottom of the cylinder 12. However, in alternate embodiments, any suitable type of fuel delivery system could be used. For example, the fuel delivery system 20 could comprise a conventional fuel delivery system well known in the art. Alternatively, the fuel delivery system could comprise a fuel injection system or a newer type of efficient, fuel delivery system such as disclosed in US. Patent Nos. 6,295, 957; 6,293, 235; 6,286, 469; and 6,382, 176.

[0017] The piston 14 is movably mounted in the cylinder 12 and is operably connected to the crankshaft 16 in a conventional manner. Referring also to Figure 2, the bottom 40 of the cylinder 12 is connected to the crankcase 18. In addition to the inlet 32, the cylinder 12 also comprises an exhaust outlet 34 and transfer ports 36. A muffler (not shown) could be attached to the exhaust outlet 34. The cylinder 12 comprises a main internal area 38 which the piston 14 reciprocally moves in, and which forms a combustion chamber 42.

[0018] Referring also to Figure 3, in this embodiment the cylinder comprises two sets 44, 46 of the transfer ports 36. The first set of transfer ports 44 comprises a pair of first transfer ports 48. The second set of transfer ports 46 comprises a pair of second transfer ports 50. However, in alternate embodiments, the cylinder could comprise more than two sets of transfer ports, and each set of transfer ports could comprise more or less than two transfer ports each. The first set 44 of transfer ports are disposed closer to the exhaust port 34 than the second set 46 of transfer ports; which are disposed

further away from the exhaust port 34.

[0019] As seen best in Figure 3, the transfer passage walls of the transfer ports 36 are angled with respect to the cylinder axis 60 and the point of intersection 61 of the imaginary plane extending from the transfer passage walls. The first transfer ports 48 are angled relative to each other at a first angle 52. In a preferred embodiment, the first angle 52 is about 70 to about 85. In one specific form of embodiment, the first angle 52 is about 79. The second transfer ports 50 are angled relative to each other at a second angle 54. In a preferred embodiment, the second angle 54 is about 120 to about 150. In one specific form of embodiment, the second angle 54 is about 141.

[0020] In one type of embodiment, the main internal area 38 of the cylinder 12 has a diameter of about 1.375 in (3.5cms). Flows from the transfer ports 36 can be directed towards an inner most general area 61 of the intersection which is spaced at a distance 66 from the cylinder axis 60. For the diameter of about 1.375 in. (3.50cms), the distance 66 can be about 0.3 inch (0.76cms) to about 0.412 inch (1.05cms).

[0021] The transfer ports 36 are angled towards a front of the cylinder in a direction away from the exhaust port 34. The transfer ports 36 extend upward from the bottom 40 of the cylinder to a middle section of the cylinder. The transfer ports 36 extend outward from the main internal area 38 into the interior side walls of the cylinder 12. The transfer ports 36 are preferably wider at their base, proximate the bottom 40, then at their top ends 56, 58. The top ends 56, 58 are substantially flat.

[0022] However, in alternate embodiments, the top ends could have any suitable type of shape.

[0023] As seen best in Figure 2, the top ends 56 of the first transfer ports 48 are shorter than the top ends 58 of the second transfer ports 50. The transfer ports 36 are opened and closed relative to the combustion chamber 42 as the piston 14 moves up and down in the main internal area 38 of the cylinder 12. Because of the difference in height between the top ends 56, 58 of the first and second transfer ports 48, 50, there is a differential in timing of opening of the second transfer ports 50 relative to the first transfer ports 48 as the piston moves downward in the cylinder towards its bottom dead center (BDC) position. More specifically, as the piston 14 moves downward in the cylinder, 12, the second pair of transfer ports 50 are opened into the combustion chamber 42 before the first pair of transfer ports 48 are opened. As the piston 14 continues to move towards its bottom dead center position, the second pair of transfer ports 50 are subsequently opened. Because the second transfer ports 50 are located further away from the exhaust port 34 than the first transfer ports 48, the transfer ports located furthest away from the exhaust port 34 open first the combination of the sequential opening of the different types of transfer ports and the angled shaped of the transfer ports combine to help prevent short circuiting of fresh unburned fuel from exiting the exhaust port 34.

[0024] Unlike conventional two-stroke engines, the front and rear pair of transfer ports have a phase difference in timing of their opening. As the piston moves downward towards a bottom dead center position, the piston uncovers the front ports, i. e. , the second pair of ports 50 about four to eight degrees sooner than the rear ports, i. e. , the first pair of transfer ports 48 are uncovered. During the early scavenging process, the front ports 50, which opened sooner, discharge live charge (fuel and air) into the cylinder, away from the exhaust port 34 due to directional discharge characteristics of the ports, the charge that is discharged furthest away from the exhaust port enters the cylinder first and, also travels the longest distance. The earliest entering charge is also the fraction of the total charge that is most likely to be lost into the exhaust 34. Even though the charge that enters through the second transfer ports 50 enters first, it has to travel the farthest and is the least amount of charge entering from the two sets 44, 46. Thus, the fractional loss is also minimal.

[0025] The early opening of the front two 50 of the four transfer ports helps to establish a flow path for the charge that follows in such a way that it may result in a near-perfect displacement scavenging. Thus, flow pattern and staggered discharge of live charge helps minimize the loss of fresh fuel into the exhaust which results in lower emissions and higher fuel economy.

[0026] The top ends 58 of the second transfer ports 50 can be located below the top end of the exhaust port 34. The width of the second transfer ports 50 can be smaller than the width of the first transfer ports 48. The use of a tapered shape along the height of the second transfer ports 50 can also reduce the side of the opening of the second transfer ports when the second transfer ports 50 are uncovered by the piston 14. It is believed that narrow opening of the front ports late during the blow-down process can increase the discharge velocity, which helps mixing. Low short circuit loss of fresh charge combined with improved mixing reduces significantly the exhaust emissions.

[0027] Referring also to Figure 4, in the embodiment shown the exhaust port 34 comprises a general chevron shaped wall. More specifically, in the embodiment shown, the top side 62 of the exhaust port 34 has a chevron shape, the top side 62 of the exhaust port 34 has a chevron shape, and the bottom side 64 has an opposite chevron shape. As the piston 14 uncovers the exhaust port 34, the initial opening of the exhaust port 34 is relatively small because the apex of the upper chevron wall is merely uncovered. As the piston 14 continues to uncover more of the exhaust port 34, the opening into the exhaust port is enlarged. The chevron shaped exhaust port provides a stepped flow area which can result in optimum blow-down performance. The engine could be provided with the transfer port feature described above alone, or in combination with the chevron shaped exhaust port as shown in Figure 4.

[0028] Tests of an engine incorporating features of the proposed invention has demonstrated emissions below 2004

EPA Phase II emission levels without the use of a catalytic converter. Implementation of the present invention into a conventional engine design is relatively simple and existing hardware (such as pistons, etc.) Can be used with the redesigned cylinder described above. Tooling cost to implement the features of the present invention is minimal. The following table shows results of such a test and variations of port configurations on a 30cc engine. Similar testing on a 25cc engine has demonstrated low emission levels.

	Transfer Port Timing in Degrees	Exhaust Port Timing in Degrees	Power	HC & NOx
#1 cyl. Version 1	137 (all)	118	0.74 hp @ 7500 rpm	66.96@7500 rpm
#1 cyl. Version 2	134, 129 (staggered)	118	0.90 hp @7500 rpm	53.33 @ 9000 rpm
#2 cyl.	129 (all)	118	0.91 hp @ 7500 rpm	57.90 @ 8500 rpm
#3 cyl.	134, 129 (staggered)	118	0.90 hp @7500 rpm	60.85 @ 8500 rpm

[0029] Referring now to Figures 5-7, an alternate embodiment of the present invention will be described. In this embodiment the engine 70 comprises a fuel delivery system 72 with an air filter 74 and an inlet 76 extending into the cylinder 78. The cylinder 78 also comprises an exhaust outlet 34 and four transfer ports 80. The transfer ports 80 comprise a first set of first transfer ports 82 and a second set of transfer ports 84.

[0030] Pairs of the transfer ports, on each side of the cylinder, comprise a common bottom channel 86 extending into the side wall of the cylinder in a bottom portion of the cylinder, and separate respective top channels which form two of the ports 82, 84. The cylinder 78 comprises a partition wall 88 which extends between the two ports 82, 84 to form the two separate top channels. In the embodiment shown, the partition wall 88 comprises a general triangular cross section. However, in alternate embodiments, the wall 88 could comprise any suitable cross sectional shape. The wall 88 has a height that is about two-thirds the heights of the ports 82, 84. In the embodiment shown, the forward and rearward sides of the bottom channels 86 are angled relative to each other at angles 94 and 96. In one embodiment, the angle 94 is about 80° and the angle 96 is about 130°. However, in alternate embodiments, any suitable angles could be provided. This embodiment can be formed the same angles 52, 54, shown in the embodiment of Figure 3. The top ends 90, 92 comprise top surfaces which are angled downward in a direction of the exhaust port 34. The second transfer ports 84 each comprise a top surface at the ends 92 which is at least partially higher than a top surface of the first transfer ports 82 at th ends 90 such that the second transfer ports open before th first transfer ports as the piston moves towards a bottom dead center position.

[0031] There is provided a progression of discharge angle 98 due to curvature of the piston. The partition walls 88 need not extend all the way to the piston 14. One of the features of this embodiment, is that the pairs of transfer ports 82, 84 can be provided in a relatively compact area. This allows features of the present invention to be used in relatively small size cylinders. In an alternate embodiment, the top ends of the transfer ports could be substantially straight and horizontal, and the top surface of the piston could be angled to allow a stepped progression of entry of a charge into the combustion chamber. In another alternate embodiment, the top surfaces of the transfer ports might not be straight, but could be non-straight.

[0032] Referring now also to Figure 8, another alternate embodiment is shown. In this embodiment, the cylinder 100 comprises transfer ports with a first type of transfer ports 102 and a second type of transfer port 84, the first and second transfer ports 102, 84 comprise a common bottom channel 86. A partition wall 88 is located at a top of the bottom channel 86 and separates the two ports 102, 84 from each other. This embodiment differs from the embodiment shown in Figure 5 in that the top end 104 of the first transfer port 102 is substantially straight and horizontal. However, the top end 92 of the second transfer port 84 is inclined downward.

[0033] Referring now also to Figure 9, another alternate embodiment of the present invention, another alternate embodiment of the present invention is shown. In this embodiment the engine 110 comprises nearly two transfer ports 112 located on opposite sides of the cylinder. Each of the transfer ports 112 comprise an angled top surface 114.

[0034] The following tables illustrate the exhaust and transfer port areas as a function of piston position in crank angle degrees with 0 representing piston top dead center (TDC) and 180 representing piston bottom dead center (BDC). Four engines W through Z, ranging in displacement from 25 to 40 cc. have been evaluated having a four transfer port design as generally illustrated in Figures 1-4. A prior art standard two-stroke cycle engine having a 30cc displacement and a single pair of transfer ports is provided for comparison purposes.

Engine W displacement 25.4 cc

[0035]

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Crankshaft rotation 0=TDC	Exhaust Area	Transfer Port I	Transfer Port II	Total I+II
118	0.0	0.0	0.0	0.0
119	1.3	0.0	0.0	0.0
120	3.1	0.0	0.0	0.0
121	5.7	0.0	0.0	0.0
122	8.6	0.0	0.0	0.0
123	11.7	0.0	0.0	0.0
124	14.7	0.0	0.0	0.0
125	17.8	0.0	0.0	0.0
126	20.8	0.0	0.0	0.0
127	23.8	0.0	0.0	0.0
128	26.8	0.0	0.0	0.0
129	29.8	0.0	0.0	0.0
130	32.7	0.0	0.0	0.0
131	35.5	0.0	0.3	0.3
132	39.8	0.0	0.9	0.9
134	43.8	0.0	1.6	1.6
135	47.8	0.0	2.3	2.3
137	51.7	0.4	3.0	3.4
139	56.5	1.6	3.9	5.5
141	61.1	2.8	4.7	7.5
143	65.4	3.9	5.5	9.4
145	69.5	5.0	6.2	11.3
147	73.2	6.1	7.0	13.0
150	78.3	7.5	7.9	15.5
153	82.8	8.8	8.9	17.7
156	86.5	10.0	9.7	19.7
159	89.4	11.1	10.4	21.5
164	92.9	12.5	11.4	23.9
169	95.1	13.6	12.1	25.7
174	96.1	14.3	12.6	26.8
179	96.3	14.6	12.8	27.4
180	96.3	14.6	12.8	27.4
All area measurements in sq mm				

Engine X displacement 25 cc

[0036]

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	Crankshaft rotation 0=TDC	Exhaust Area	Transfer Port A	Transfer Port B	A+B
	118	0.0	0.0	0.0	0.0
5	119	0.7	0.0	0.0	0.0
	120	2.4	0.0	0.0	0.0
	121	4.2	0.0	0.0	0.0
	122	6.1	0.0	0.0	0.0
10	123	8.0	0.0	0.0	0.0
	124	10.0	0.0	0.0	0.0
	125	12.2	0.0	0.0	0.0
15	127	16.0	0.0	0.0	0.0
	128	20.1	0.5	0.0	0.5
	130	24.5	1.4	0.0	1.4
	131	28.9	2.4	0.1	2.5
20	133	34.8	3.6	1.7	5.3
	135	40.5	4.8	3.4	8.2
	137	46.2	6.0	5.0	11.0
25	139	51.6	7.1	6.6	13.7
	141	56.8	8.2	8.1	16.3
	144	64.2	9.6	10.2	19.9
	147	71.1	11.0	12.2	23.2
30	150	77.3	12.3	14.0	26.3
	153	82.9	13.4	15.7	29.1
	158	90.7	15.1	18.0	33.1
35	163	96.8	16.4	19.9	36.3
	168	101.2	17.4	21.3	38.7
	173	104.0	18.0	22.2	40.3
	178	105.3	18.3	22.7	41.0
40	180	105.4	18.4	22.7	41.1
All area measurements in sq mm					

Engine Y displacement 30 cc

[0037]

	Crankshaft rotation 0=TDC	Exhaust Area	Transfer Port A	Transfer Port B	A+B
50	118	0.0	0.0	0.0	0.0
	119	1.0	0.0	0.0	0.0
	120	2.7	0.0	0.0	0.0
55	121	4.6	0.0	0.0	0.0
	122	6.6	0.0	0.0	0.0
	123	8.7	0.0	0.0	0.0

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(continued)

Crankshaft rotation 0=TDC	Exhaust Area	Transfer Port A	Transfer Port B	A+B
124	10.9	0.0	0.0	0.0
125	13.2	0.0	0.0	0.0
127	17.1	0.0	0.0	0.0
128	21.4	0.0	0.0	0.0
130	25.9	0.0	0.0	0.0
131	30.3	0.7	0.0	0.7
133	36.3	1.9	0.0	1.9
135	42.2	3.1	0.0	3.1
137	47.9	4.2	0.4	4.7
139	53.4	5.4	2.0	7.3
141	58.7	6.4	3.6	10.0
144	66.3	7.9	5.8	13.7
147	73.2	9.3	7.9	17.2
150	79.6	10.5	9.8	20.4
153	85.3	11.7	11.5	23.2
158	93.5	13.3	14.0	27.4
163	99.8	14.7	16.0	30.7
168	104.5	15.6	17.5	33.1
173	107.5	16.3	18.5	34.8
178	108.9	16.6	18.9	35.5
180	109.0	16.6	19.0	35.6
All area measurements in sq mm				

Engine Z displacement 40 cc

[0038]

Crankshaft rotation 0=TDC	Exhaust Area	Transfer Port A	Transfer Port B	A+B
118	0.0	0.0	0.0	0.0
119	0.0	0.0	0.0	0.0
120	1.0	0.0	0.0	0.0
121	3.0	0.0	0.0	0.0
122	5.1	0.0	0.0	0.0
123	7.2	0.0	0.0	0.0
124	9.4	0.0	0.0	0.0
125	12.1	0.0	0.0	0.0

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(continued)

Crankshaft rotation 0=TDC	Exhaust Area	Transfer Port A	Transfer Port B	A+B
127	17.0	0.0	0.0	0.0
128	22.3	0.0	0.0	0.0
130	27.7	0.6	0.0	0.6
131	33.1	1.7	0.0	1.7
133	40.1	3.0	0.2	3.3
135	47.0	4.4	2.5	6.9
137	53.6	5.6	4.8	10.4
139	59.9	6.9	6.9	13.8
141	66.0	8.0	9.0	17.0
144	74.5	9.7	11.8	21.5
147	82.3	11.2	14.5	25.7
150	89.5	12.6	16.9	29.5
153	95.9	13.9	19.1	33.0
158	105.0	15.7	22.3	38.0
163	112.0	17.2	24.8	42.0
168	117.1	18.2	26.7	44.9
173	120.3	18.9	27.9	46.9
178	121.8	19.3	28.5	47.8
180	121.9	19.3	28.5	47.9
All area measurements in sq mm				

Standard Engine displacement 30 cc

[0039]

Crankshaft rotation 0=TDC	Exhaust Area	Transfer Port A
111	0.000	0.000
112	1.434	0.000
113	5.103	0.000
114	8.918	0.000
115	12.802	0.000
116	16.721	0.000
117	20.654	0.000
118	24.584	0.000
119	28.499	0.000
120	32.389	0.000
121	36.244	0.000

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(continued)

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Crankshaft rotation 0=TDC	Exhaust Area	Transfer Port A
122	40.057	0.000
123	43.822	0.000
124	47.531	0.000
125	51.181	0.000
126	54.771	0.000
127	58.301	0.000
128	61.770	0.000
129	65.178	0.000
130	68.524	0.000
131	71.808	0.000
132	75.030	0.000
133	78.189	0.000
134	81.285	0.638
135	84.317	2.267
136	87.285	3.946
138	93.030	7.348
140	98.516	10.729
142	103.742	14.034
145	111.087	18.777
148	117.817	23.199
151	123.907	27.260
155	131.009	32.079
159	136.947	36.195
163	141.741	39.598
167	145.427	42.287
171	148.054	44.260
176	149.938	45.718
180	150.390	46.075
All area measurements in sq mm		

[0040] To better illustrate the relative size and timing of the transfer ports and the exhaust port area of the present invention in contrast to the prior art, a port area versus crank angle timing diagram is provided in Figure 10. The standard prior art two-stroke engine is represented by exhaust port area curve 120 and transfer port area curve 122. Engine Y, is a comparably sized engine utilizing the present invention. Engine Y has an exhaust port area versus crank angle degree curve 124. Relative to standard exhaust port area curve 120, the present invention is not only slightly lower in maximum area, but is shifted approximately at 10° later in time. Quite subtly, but important, is the shape of the exhaust port area curve 120 as it initially opens. The exhaust port area initially increases more gradually than the prior art due to the chevron shaped exhaust port described previously.

[0041] The exhaust port of engine Y has a blow down region which is 20 % to 30 % of the total port area which has a reduced circumferential length relative to the remaining port region resulting in a more gradual port opening and port closing. This small size blow down region allows for the intake charge to be effectively trapped while still allowing efficient exhaust blow down and discharge so that engine power is not compromised. Preferably, the exhaust blow down region

will have a circumferential port length of about 50 % of the maximum circumferential length from the remainder of the exhaust port.

[0042] As further illustrated in Figure 10, as well as the accompanying timing charts for engines W-Z, the preferred exhaust port opening occurs between 116°-121° after TDC and preferably, 117°-120° after TDC. Most preferably, the exhaust port opens 118°-119° after TDC.

[0043] In addition to delaying exhaust port opening and port opening geometry, engines of the present invention open the transfer ports relatively early. The combined area of the transfer ports result in a more gradual transfer port opening. In Figure 10, the second transfer port opens initially, as illustrated by curve 126, while the first transfer port area is illustrated by curve 128. The combined areas of the two transfer ports is illustrated by curve 130. As shown graphically in Figure 10, as well as in engine tables W-Z, the maximum area of the first transfer ports at BDC is greater than that of the second transfer ports at BDC. Preferably, the second transfer ports will have a BDC area which is less than 90 % of the BDC area of the first transfer ports at BDC. More preferably, the second transfer port area will be 65 %-90% of the first transfer port area at BDC and most preferably, 80%-90% of the second transfer port area at BDC.

[0044] The relative timing of the opening of the first and second transfer ports are likewise illustrated in the Figure 10 graph as well as tables W-Z. The second transfer port opens over 3° prior to the first transfer port, preferably 3°-10° before the first transfer port, and most preferably, 4°-8° before the first transfer port.

[0045] The flow of the intake charge into the cylinder in the four transfer port embodiments initially comes from the second transfer ports which are oriented at an included angle of 120°-150° relative to one another as illustrated in Figure 3. As the piston moves down and opens the first transfer ports, the additional intake charge is introduced into the cylinder and a more pronounced angle relative to the transfer center line with the included angle between the first transfer ports being in the 70°-85° range as illustrated in Figure 3. The flow through all four transfer ports converges in a transfer port convergence zone 63. The transfer port convergence zone 63 is located along the transverse centerline between the cylinder axis 60 and the cylinder front wall opposite the exhaust port 34. Ideally, the convergence zone is spaced from the bore axis 60, a distance greater than .4 times the cylinder radius, preferably, 4-9 times the cylinder radius and most preferably, .5-.8 times the cylinder radius in the four point embodiment of Figures 1-4.

[0046] In the alternative embodiments shown in Figures 6 and 7, the transfer port convergence zone is located slightly closer to the cylinder wall opposite the exhaust port. It should be appreciated that whether the four port design shown in Figure 3 is used or the alternative port designs shown in Figures 6 and 7 are used, the intake charge initially entering the cylinder is introduced at a greater included angle between the opposed ports than when the charge which is introduced later in the intake cycle when the transfer ports are fully opened. This design serves to maximize scavenge efficiency and intake turbulence while limiting intake charge short circuit losses. The combined benefits of the exhaust and transfer port timing and shape, enables significant improvements in emissions to be achieved without the use of expensive add on emission remediation hardware.

Claims

1. A two-stroke internal combustion engine (10; 70) comprising;
a cylinder (12; 78); and
a piston (14) movably mounted in the cylinder,
wherein the cylinder comprises an exhaust port (34) and transfer ports (36; 80), wherein the transfer ports comprise a first pair (48; 82) of the transfer ports disposed closer to the exhaust port than a second pair (50; 84) of the transfer ports which are disposed further away from the exhaust port, wherein respective forward and rearward transfer passage side walls of the first pair of transfer ports are angled relative to each other at a first angle of 70 to 85 degrees and respective forward and rearward transfer passage side walls of the second pair of transfer ports are angled relative to each other at a second angle of 120 to 150 degrees; and the transfer ports (36; 80) extend outwardly from a main internal area (38) of the cylinder into interior side walls of the cylinder and extend upward from proximate a bottom (40) of the cylinder (12; 78) to a middle section of the cylinder;
wherein directional discharge of scavenged air out of the transfer ports (36; 80) establishes a flow path for the scavenged air to minimize losses of fresh unburned fuel into the exhaust port (34).
2. A two-stroke internal combustion engine (10; 70) as in claim 1 wherein the second pair (50; 84) of transfer ports (36; 80) comprise top surfaces (58; 92) which are at least partially higher than top surfaces (56; 90) of the first pair (48; 82) of transfer ports (36; 80) such that the second pair of transfer ports open before the first pair of transfer ports as the piston (12) moves towards bottom dead center position.
3. A two-stroke internal combustion engine (10; 70) as in claim 2 wherein the top surfaces (92) of the second pair (84) of transfer ports (80) each comprise an inclined surface which is angled downward on a side closest to the exhaust

port (34).

4. A two-stroke internal combustion engine (10; 70) as in claim 3 wherein the top surfaces (90) of the first pair (82) of transfer ports (80) each comprise an inclined surface which is angled downward on a side closest to the exhaust port (34).
5. A two-stroke internal combustion engine (10; 70) as in claim 3 wherein the top surfaces (56) of the first pair (48) of transfer ports (36) are not inclined towards the exhaust port (34).
6. A two-stroke internal combustion engine (10; 70) as in claim 1 wherein two of the transfer ports (80) comprise a common bottom channel (86) extending into a side wall of the cylinder in a bottom portion of the cylinder and separate respective top channels, wherein the cylinder comprises a partition wall (88) extending between the two separate top channels to form the two ports (80).
7. A two-stroke internal combustion engine (10; 70) as in claim 1 or claim 2 wherein the exhaust port (34) comprises a general chevron shaped top wall (62) which has a stepped flow area.
8. A two-stroke internal combustion engine (10; 70) as in any one of the preceding claims wherein the first angle is 79 degrees.
9. A two-stroke internal combustion engine (10; 70) as in any one of the preceding claims wherein the second angle is 141 degrees.
10. The two-stroke internal combustion engine (10; 70) as in any one of the preceding claims wherein the maximum area of the second pair (50; 84) of transfer ports (36; 80) is less than 90% of the maximum area of the first pair (48; 82) of transfer ports when the piston is at bottom dead centre 'BDC'.
11. The two-stroke internal combustion engine (10; 70) of any one of the preceding claims wherein the maximum area of the second set of transfer ports is between 65 %-90 % of the maximum area of the first pair of transfer ports when the piston is at BDC.
12. The two-stroke internal combustion engine (10; 70) of any one of claims 3 to 10 wherein the maximum area of the second set of transfer ports is between 80%-90% of the maximum area of the first pair of transfer ports when the piston is at BDC.
13. The two-stroke internal combustion engine (10; 70) of any one of the preceding claims wherein the second pair (50; 84) of transfer ports (36; 80) opens over 3 crankshaft degrees before the first pair (48; 82) of transfer ports.
14. The two-stroke internal combustion engine (10; 70) of any one of the preceding claims wherein the second pair of transfer ports opens over 3 to 10 crankshaft degrees before the first pair of transfer ports.
15. The two-stroke internal combustion engine (10; 70) of any one of the preceding claims wherein the second pair of transfer ports opens over 4 to 8 crankshaft degrees before the first pair of transfer ports.
16. The two-stroke internal combustion engine (10; 70) of any one of the preceding claims wherein the flow of intake charge converges in a central convergence zone located along a transverse center line of the cylinder and between the bore axis (60) and the front region cylinder wall opposite the exhaust port (34),
17. The two-stroke internal combustion engine (10; 70) of claim 16 wherein the center of the transfer port convergence zone is spaced from the bore central axis by an amount greater than four times the cylinder radius.
18. The two-stroke internal combustion engine (10; 70) of claim 16 or claim 17 wherein the center of the transfer port convergence zone is spaced from the bore central axis by an amount equal to 4 to 9 times the cylinder radius.
19. The two-stroke internal combustion engine (10; 70) of any one of claims 16 to 18 wherein the center of the Transfer port convergence zone is spaced from the bore central axis by an amount equal to 5 to 8 times the cylinder radius.

Patentansprüche

1. Zweitakt-Verbrennungskraftmaschine (10; 70), die Folgendes umfasst:

5 einen Zylinder (12; 78) und
 einen Kolben (14), der beweglich in dem Zylinder angebracht ist,
 wobei der Zylinder einen Auslass-Schlitz (34) und Überströmschlitze (36; 80) umfasst, wobei die Überström-
 schlitze ein erstes Paar (48; 82) der Überströmschlitze, die näher an dem Auslass-Schlitz angeordnet sind als
 10 ein zweites Paar (50; 84) der Überströmschlitze, die weiter entfernt von dem Auslass-Schlitz angeordnet sind,
 umfassen, wobei jeweilige vordere und hintere Überström-Durchgangsseitenwände des ersten Paares von
 Überströmschlitzen im Verhältnis zueinander in einem ersten Winkel von 70 bis 85 Grad abgewinkelt sind und
 jeweilige vordere und hintere Überström-Durchgangsseitenwände des zweiten Paares von Überströmschlitzen
 15 im Verhältnis zueinander in einem zweiten Winkel von 120 bis 150 Grad abgewinkelt sind und sich die Über-
 strömschlitze (36; 80) von einem inneren Hauptbereich (38) des Zylinders aus nach außen in die inneren
 Seitenwände des Zylinders hinein erstrecken und sich von nahe einer Sohle (40) des Zylinders (12; 78) aus
 nach oben bis zu einer Mittelsektion des Zylinders erstrecken,
 wobei eine gerichtete Abgabe von Spülluft aus den Überströmschlitzen (36; 80) eine Durchflussbahn für die
 Spülluft herstellt, um die Verluste von frischem unverbranntem Kraftstoff in den Auslass-Schlitz (34) auf ein
 Minimum zu verringern.

20 2. Zweitakt-Verbrennungskraftmaschine (10; 70) nach Anspruch 1, wobei das zweite Paar (50; 84) von Überström-
 schlitzen (36; 80) obere Flächen (58; 92) umfasst, die wenigstens teilweise höher sind als obere Flächen (56, 90)
 des ersten Paares (48; 82) von Überströmschlitzen (36; 80) derart, dass das zweite Paar von Überströmschlitzen
 vor dem ersten Paar von Überströmschlitzen öffnet, wenn sich der Kolben (12) zu der unteren Totpunktstellung hin
 25 bewegt.

3. Zweitakt-Verbrennungskraftmaschine (10; 70) nach Anspruch 2, wobei die oberen Flächen (92) des zweiten Paares
 (84) von Überströmschlitzen (80) jede eine geneigte Fläche umfassen, die auf einer dem Auslass-Schlitz (34)
 nächsten Seite nach unten abgewinkelt ist.

4. Zweitakt-Verbrennungskraftmaschine (10; 70) nach Anspruch 3, wobei die oberen Flächen (90) des ersten Paares
 (82) von Überströmschlitzen (80) jede eine geneigte Fläche umfassen, die auf einer dem Auslass-Schlitz (34)
 nächsten Seite nach unten abgewinkelt ist.

5. Zweitakt-Verbrennungskraftmaschine (10; 70) nach Anspruch 3, wobei die oberen Flächen (56) des ersten Paares
 (48) von Überströmschlitzen (36) nicht zu dem Auslass-Schlitz (34) hin geneigt sind.

6. Zweitakt-Verbrennungskraftmaschine (10; 70) nach Anspruch 1, wobei zwei von den Überströmschlitzen (80) einen
 gemeinsamen unteren Kanal (86), der sich in einem unteren Abschnitt des Zylinders in eine Seitenwand des Zylinders
 40 hinein erstreckt, und gesonderte jeweilige obere Kanäle umfassen, wobei der Zylinder eine Trennwand (88) umfasst,
 die sich zwischen den zwei gesonderten oberen Kanälen erstreckt, um die zwei Schlitze (80) zu bilden.

7. Zweitakt-Verbrennungskraftmaschine (10; 70) nach Anspruch 1 oder Anspruch 2, wobei der Auslass-Schlitz (34)
 eine im Allgemeinen winkelförmige obere Wand (62) umfasst, die einen abgestuften Durchflussbereich hat.

8. Zweitakt-Verbrennungskraftmaschine (10; 70) nach einem der vorhergehenden Ansprüche, wobei der erste Winkel
 79 Grad beträgt.

9. Zweitakt-Verbrennungskraftmaschine (10; 70) nach einem der vorhergehenden Ansprüche, wobei der zweite Winkel
 141 Grad beträgt.

10. Zweitakt-Verbrennungskraftmaschine (10; 70) nach einem der vorhergehenden Ansprüche, wobei die maximale
 Fläche des zweiten Paares (50; 84) von Überströmschlitzen (36; 80) weniger als 90 % der maximalen Fläche des
 ersten Paares (48; 82) von Überströmschlitzen beträgt, wenn sich der Kolben an dem unteren Totpunkt (UT) befindet.

11. Zweitakt-Verbrennungskraftmaschine (10; 70) nach einem der vorhergehenden Ansprüche, wobei die maximale
 Fläche der zweiten Menge von Überströmschlitzen zwischen 65 % und 90 % der maximalen Fläche des ersten
 Paares von Überströmschlitzen beträgt, wenn sich der Kolben an dem UT befindet.

12. Zweitakt-Verbrennungskraftmaschine (10; 70) nach einem der Ansprüche 1 bis 10, wobei die maximale Fläche der zweiten Menge von Überströmschlitzen zwischen 80 % und 90 % der maximalen Fläche des ersten Paares von Überströmschlitzen beträgt, wenn sich der Kolben an dem UT befindet.
- 5 13. Zweitakt-Verbrennungskraftmaschine (10; 70) nach einem der vorhergehenden Ansprüche, wobei das zweite Paar (50; 84) von Überströmschlitzen (36; 80) mehr als 3 Kurbelwellengrade vor dem ersten Paar (48; 82) von Überströmschlitzen öffnet.
- 10 14. Zweitakt-Verbrennungskraftmaschine (10; 70) nach einem der vorhergehenden Ansprüche, wobei das zweite Paar von Überströmschlitzen mehr als 3 bis 10 Kurbelwellengrade vor dem ersten Paar von Überströmschlitzen öffnet.
- 15 15. Zweitakt-Verbrennungskraftmaschine (10; 70) nach einem der vorhergehenden Ansprüche, wobei das zweite Paar von Überströmschlitzen mehr als 4 bis 8 Kurbelwellengrade vor dem ersten Paar von Überströmschlitzen öffnet.
- 16 16. Zweitakt-Verbrennungskraftmaschine (10; 70) nach einem der vorhergehenden Ansprüche, wobei der Fluss von Einlassladung in einer mittigen Konvergenzzone konvergiert, die längs einer quer verlaufenden Mittellinie des Zylinders und zwischen der Bohrungssachse (60) und der Vorderbereichszylinderwand gegenüber dem Auslass-Schlitz (34) angeordnet ist.
- 20 17. Zweitakt-Verbrennungskraftmaschine (10; 70) nach Anspruch 16, wobei der Mittelpunkt der Überströmschlitz-Konvergenzzone um ein Maß von der Bohrungsmittelachse entfernt angeordnet ist, das größer als das Vierfache des Zylinderradius ist.
- 25 18. Zweitakt-Verbrennungskraftmaschine (10; 70) nach Anspruch 16 oder Anspruch 17, wobei der Mittelpunkt der Überströmschlitz-Konvergenzzone um ein Maß von der Bohrungsmittelachse entfernt angeordnet ist, das gleich dem 4- bis 9-fachen des Zylinderradius ist.
- 30 19. Zweitakt-Verbrennungskraftmaschine (10; 70) nach einem der Ansprüche 16 bis 18, wobei der Mittelpunkt der Überströmschlitz-Konvergenzzone um ein Maß von der Bohrungsmittelachse entfernt angeordnet ist, das gleich dem 5- bis 8-fachen des Zylinderradius ist.

Revendications

- 35 1. Moteur à combustion interne à deux temps (10 ; 70), comprenant :
- un cylindre (12 ; 78) ; et
un piston (14), monté de manière mobile dans le cylindre ;
le cylindre comprenant un orifice d'échappement (34) et des orifices de transfert (36 ; 80), les orifices de transfert
40 comprenant une première paire (48 ; 82) d'orifices de transfert agencés plus près de l'orifice d'échappement qu'une deuxième paire (50 ; 84) d'orifices de transfert, agencés plus à l'écart de l'orifice d'échappement, des parois latérales de passage de transfert avant et arrière respectives des orifices de transfert de la première paire étant inclinées l'une par rapport à l'autre à un premier angle compris entre 70 et 85 degrés, des parois latérales de passage de transfert avant et arrière respectives des orifices de transfert de la deuxième paire étant inclinées l'une par rapport à l'autre à un deuxième angle compris entre 120 et 150 degrés ; les orifices
45 de transfert (36 ; 80) s'étendant vers l'extérieur à partir d'une zone interne principale (38) du cylindre, dans des parois latérales internes du cylindre, et s'étendant vers le haut à partir d'un point proche d'une partie inférieure (40) du cylindre (12 ; 78), vers une section médiane du cylindre ;
la décharge directionnelle de l'air de balayage s'échappant à travers les orifices de transfert (36 ; 80) établissant
50 une trajectoire d'écoulement pour l'air de balayage, pour réduire au minimum les pertes de carburant frais non brûlé dans l'orifice d'échappement (34).
- 55 2. Moteur à combustion interne à deux temps (10 ; 70) selon la revendication 1, dans lequel les orifices de transfert (36 ; 80) de la deuxième paire (50 ; 84) comprennent des surfaces supérieures (58 ; 92), au moins en partie plus hautes que les surfaces supérieures (56 ; 90) des orifices de transfert (36, 80) de la première paire (48 ; 82), de sorte que les orifices de transfert de la deuxième paire sont ouverts avant les orifices de transfert de la première paire lorsque le piston (12) se déplace vers la position du point mort inférieur.

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3. Moteur à combustion interne à deux temps (10 ; 70) selon la revendication 2, dans lequel les surfaces supérieures (92) des orifices de transfert (80) de la deuxième paire (84) comprennent chacune une surface inclinée, inclinée vers le bas sur le côté le plus proche de l'orifice d'échappement (34).
- 5 4. Moteur à combustion interne à deux temps (10 ; 70) selon la revendication 3, dans lequel les surfaces supérieures (90) des orifices de transfert (80) de la première paire (82) comprennent chacune une surface inclinée, inclinée vers le bas sur le côté le plus proche de l'orifice d'échappement (34).
- 10 5. Moteur à combustion interne à deux temps (10 ; 70) selon la revendication 3, dans lequel les surfaces supérieures (56) des orifices de transfert (36) de la première paire (48) ne sont pas inclinées vers l'orifice d'échappement (34).
- 15 6. Moteur à combustion interne à deux temps (10 ; 70) selon la revendication 1, dans lequel deux des orifices de transfert (80) comprennent un canal inférieur commun (86), s'étendant dans une paroi latérale du cylindre dans une partie inférieure du cylindre, et des canaux supérieurs séparés, le cylindre comprenant une paroi de séparation (88) s'étendant entre les deux canaux supérieurs séparés pour former les deux orifices (80).
- 20 7. Moteur à combustion interne à deux temps (10 ; 70) selon les revendications 1 ou 2, dans lequel l'orifice d'échappement (34) comprend une paroi supérieure (62) ayant en général une forme en chevron (62), comportant une section d'écoulement étagée.
- 25 8. Moteur à combustion interne à deux temps (10 ; 70) selon l'une quelconque des revendications précédentes, dans lequel le premier angle correspond à 79 degrés.
- 30 9. Moteur à combustion interne à deux temps (10 ; 70) selon l'une quelconque des revendications précédentes, dans lequel le deuxième angle correspond à 141 degrés.
- 35 10. Moteur à combustion interne à deux temps (10 ; 70) selon l'une quelconque des revendications précédentes, dans lequel la surface maximale des orifices de transfert (36, 80) de la deuxième paire (50 ; 84) représente moins de 90% de la surface maximale des orifices de transfert de la première paire (48 ; 82) lorsque le piston se trouve au point mort inférieur «PMI».
- 40 11. Moteur à combustion interne à deux temps (10 ; 70) selon l'une quelconque des revendications précédentes, dans lequel la surface maximale des orifices de transfert du deuxième groupe représente 65% à 90% de la surface maximale des orifices de transfert de la première paire lorsque le piston se trouve au niveau du PMI.
- 45 12. Moteur à combustion interne à deux temps (10 ; 70) selon l'une quelconque des revendications 1 à 10, dans lequel la surface maximale des orifices de transfert du deuxième groupe représente 80% à 90% de la surface maximale des orifices de transfert de la première paire lorsque le piston se trouve au niveau du point BDC.
- 50 13. Moteur à combustion interne à deux temps (10 ; 70) selon l'une quelconque des revendications précédentes, dans lequel les orifices de transfert (36 ; 80) de la deuxième paire (50 ; 84) sont ouverts à plus de 3 degrés de vilebrequin avant les orifices de transfert de la première paire (48 ; 82).
- 55 14. Moteur à combustion interne à deux temps (10 ; 70) selon l'une quelconque des revendications précédentes, dans lequel les orifices de transfert de la deuxième paire sont ouverts à plus de 3 à 10 degrés de vilebrequin avant les orifices de transfert de la première paire.
15. Moteur à combustion interne à deux temps (10 ; 70) selon l'une quelconque des revendications précédentes, dans lequel les orifices de transfert de la deuxième paire sont ouverts à plus de 4 à 8 degrés de vilebrequin avant les orifices de transfert de la première paire.
16. Moteur à combustion interne à deux temps (10 ; 70) selon l'une quelconque des revendications précédentes, dans lequel l'écoulement de la charge d'admission converge dans une zone de convergence centrale agencée le long d'une ligne médiane transversale du cylindre et entre l'axe de l'alésage (60) et la paroi de la région avant du cylindre opposée à l'orifice d'échappement (34).
17. Moteur à combustion interne à deux temps (10 ; 70) selon la revendication 16, dans lequel le centre de la zone de convergence des orifices de transfert est espacé de l'axe central de l'alésage d'une distance représentant plus de

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quatre fois le rayon du cylindre.

- 5 **18.** Moteur à combustion interne à deux temps (10 ; 70) selon les revendications 16 ou 17, dans lequel le centre de la zone de convergence des orifices de transfert est espacé de l'axe central de l'alésage d'une distance représentant 4 à 9 fois le rayon du cylindre.

- 10 **19.** Moteur à combustion interne à deux temps (10 ; 70) selon l'une quelconque des revendications 16 à 18, dans lequel le centre de la zone de convergence des orifices de transfert est espacé de l'axe central de l'alésage d'une distance représentant 5 à 8 fois le rayon du cylindre.

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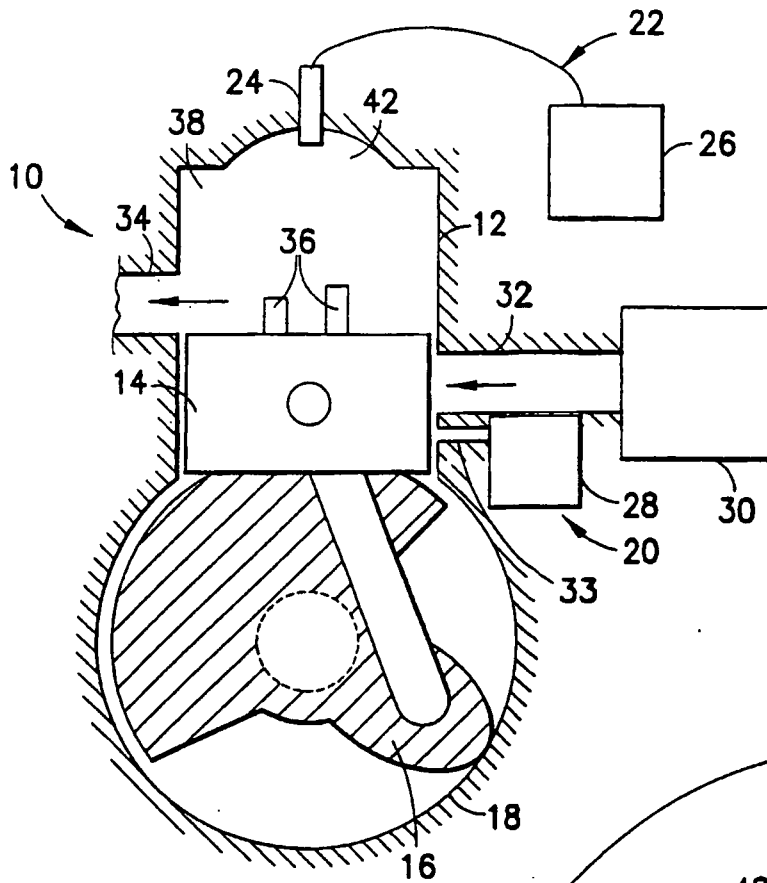


FIG. 1

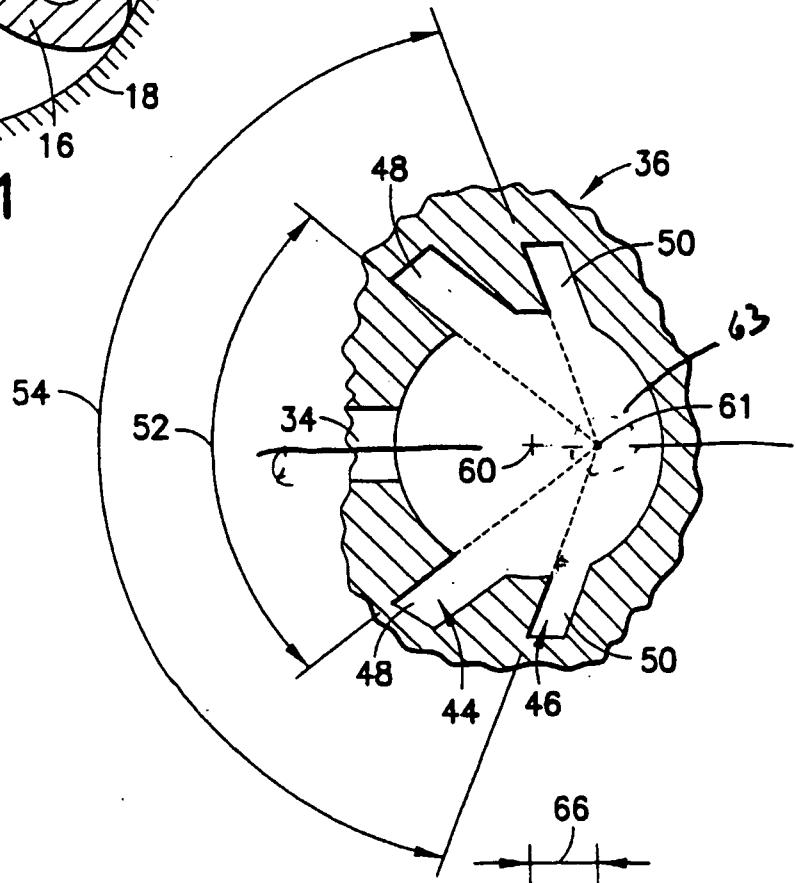


FIG. 3

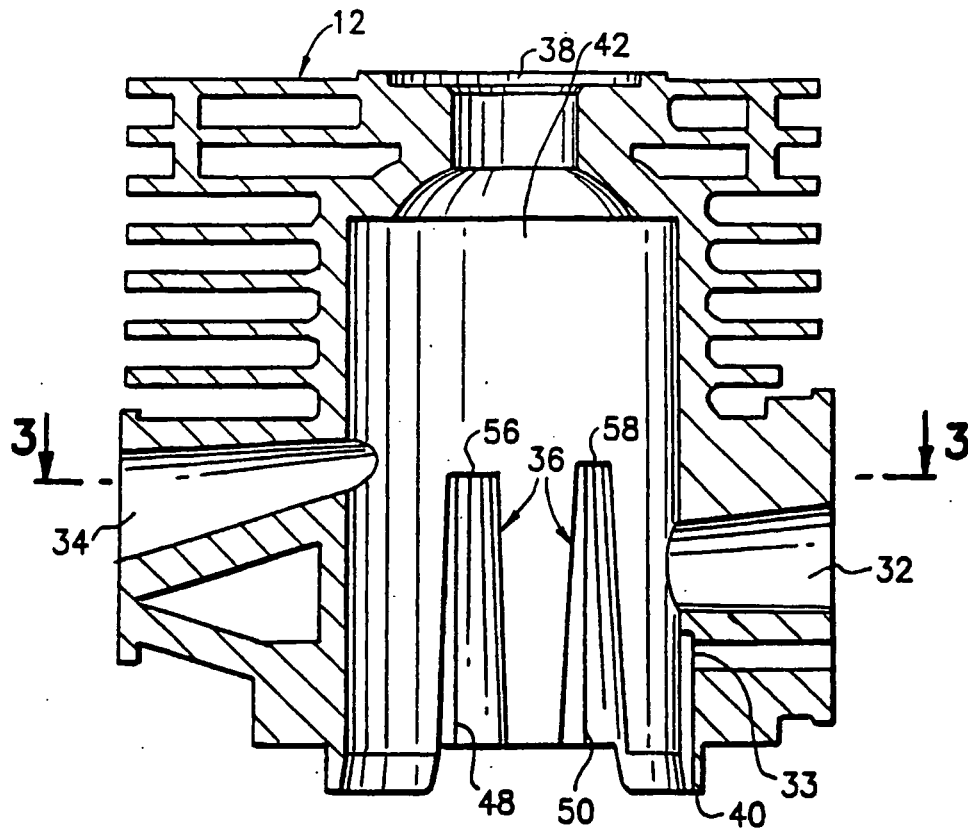


FIG. 2

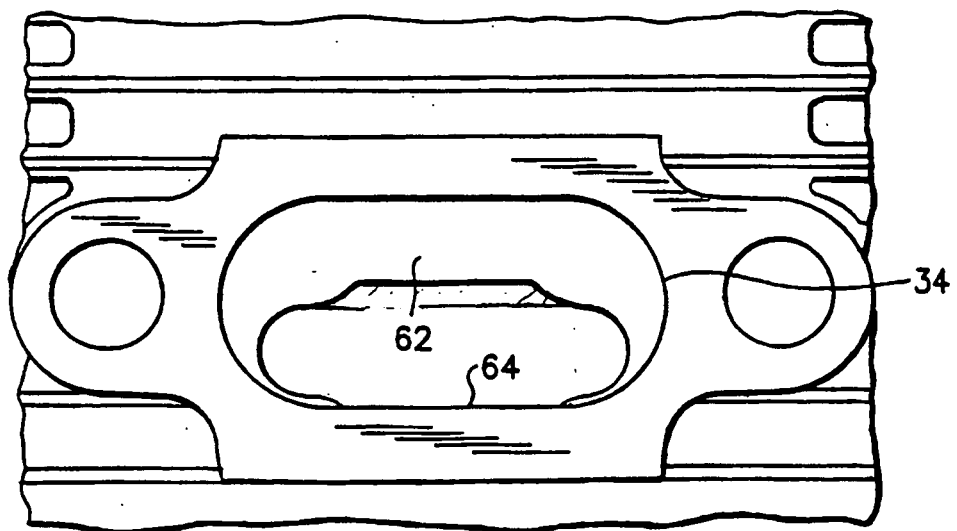


FIG. 4

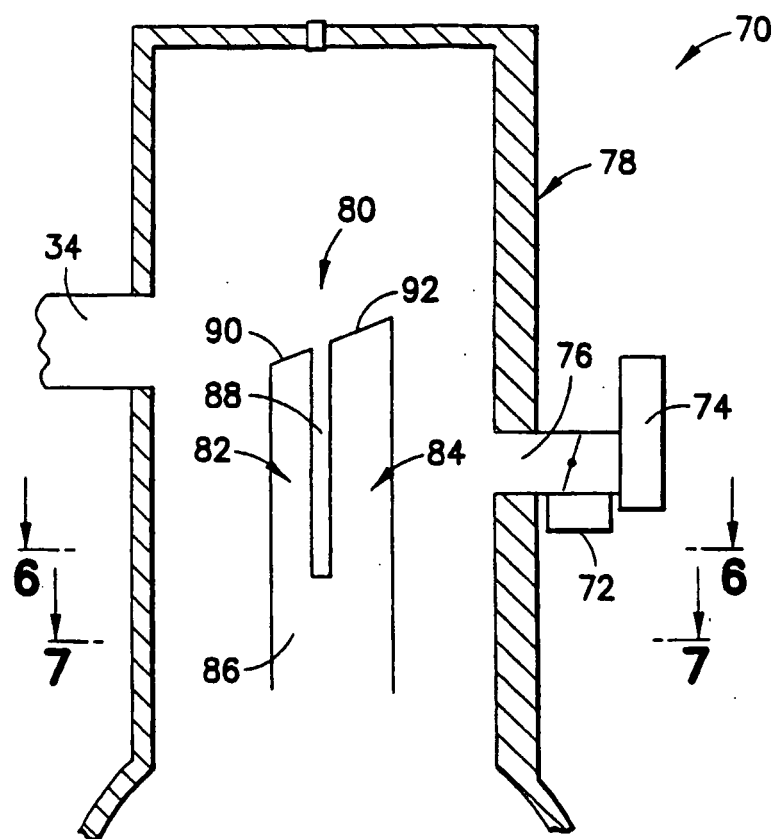


FIG.5

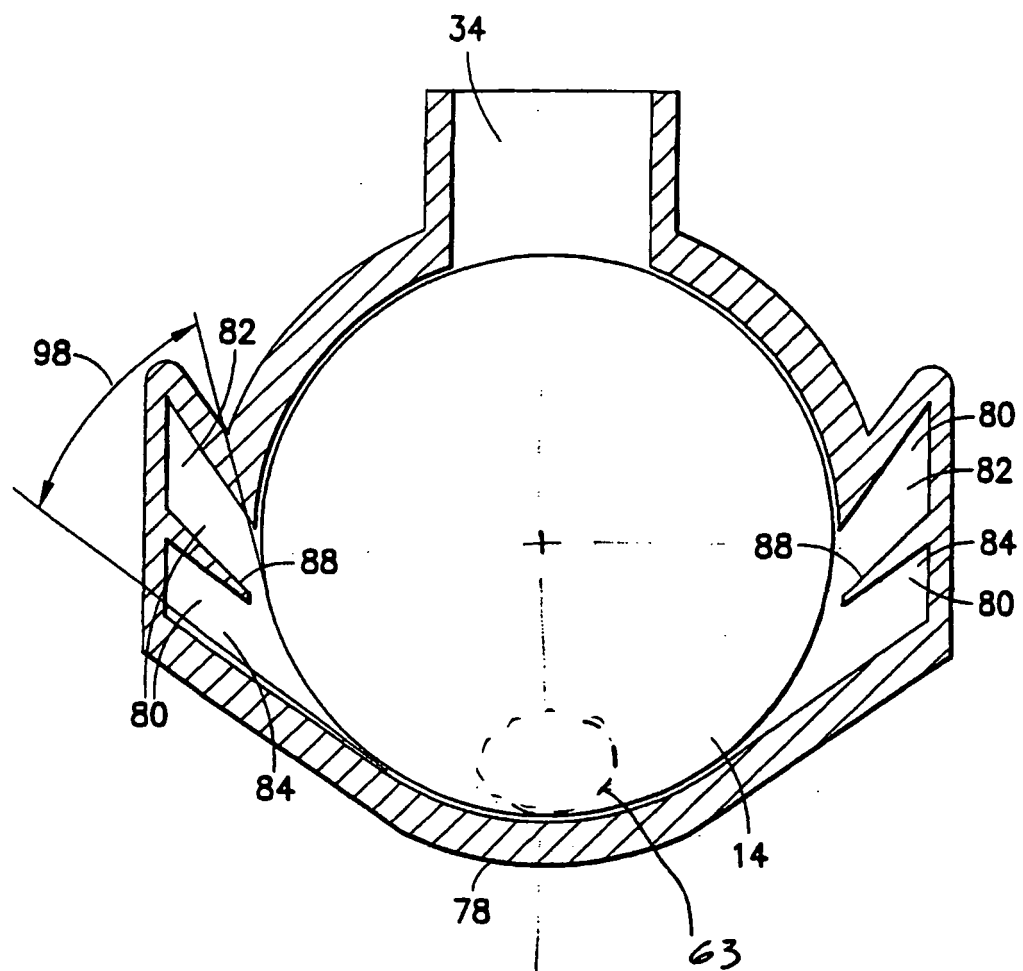
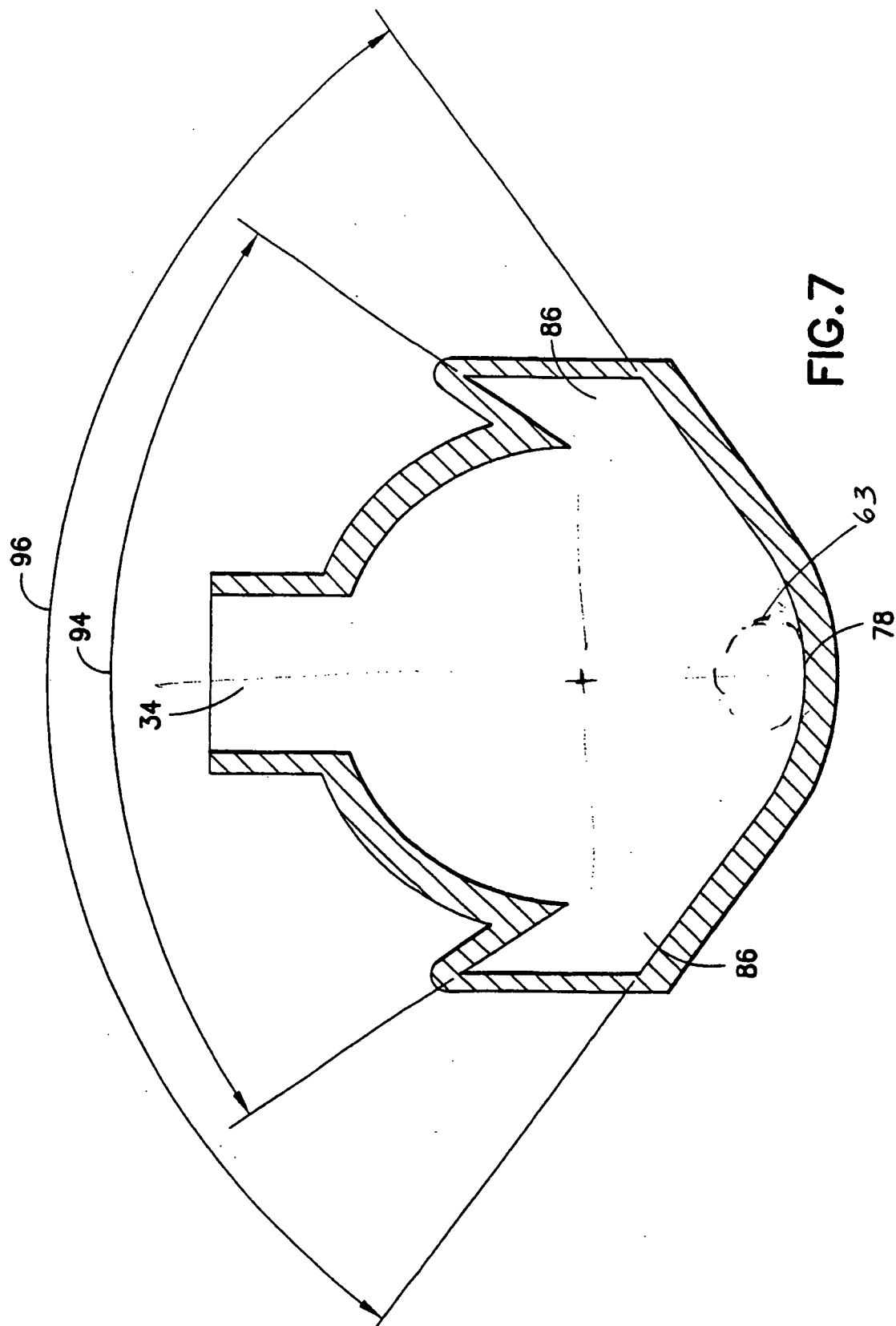


FIG. 6



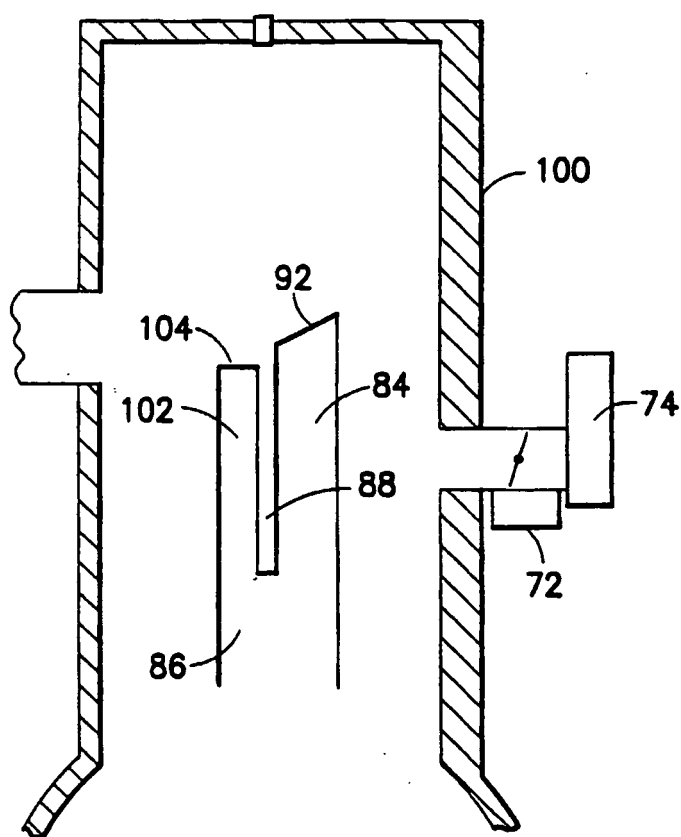


FIG.8

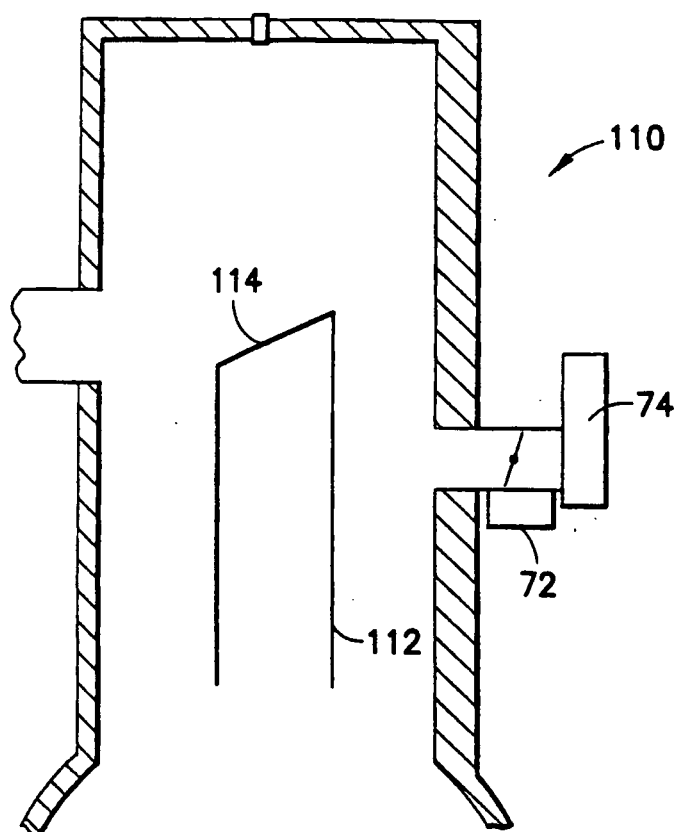


FIG.9

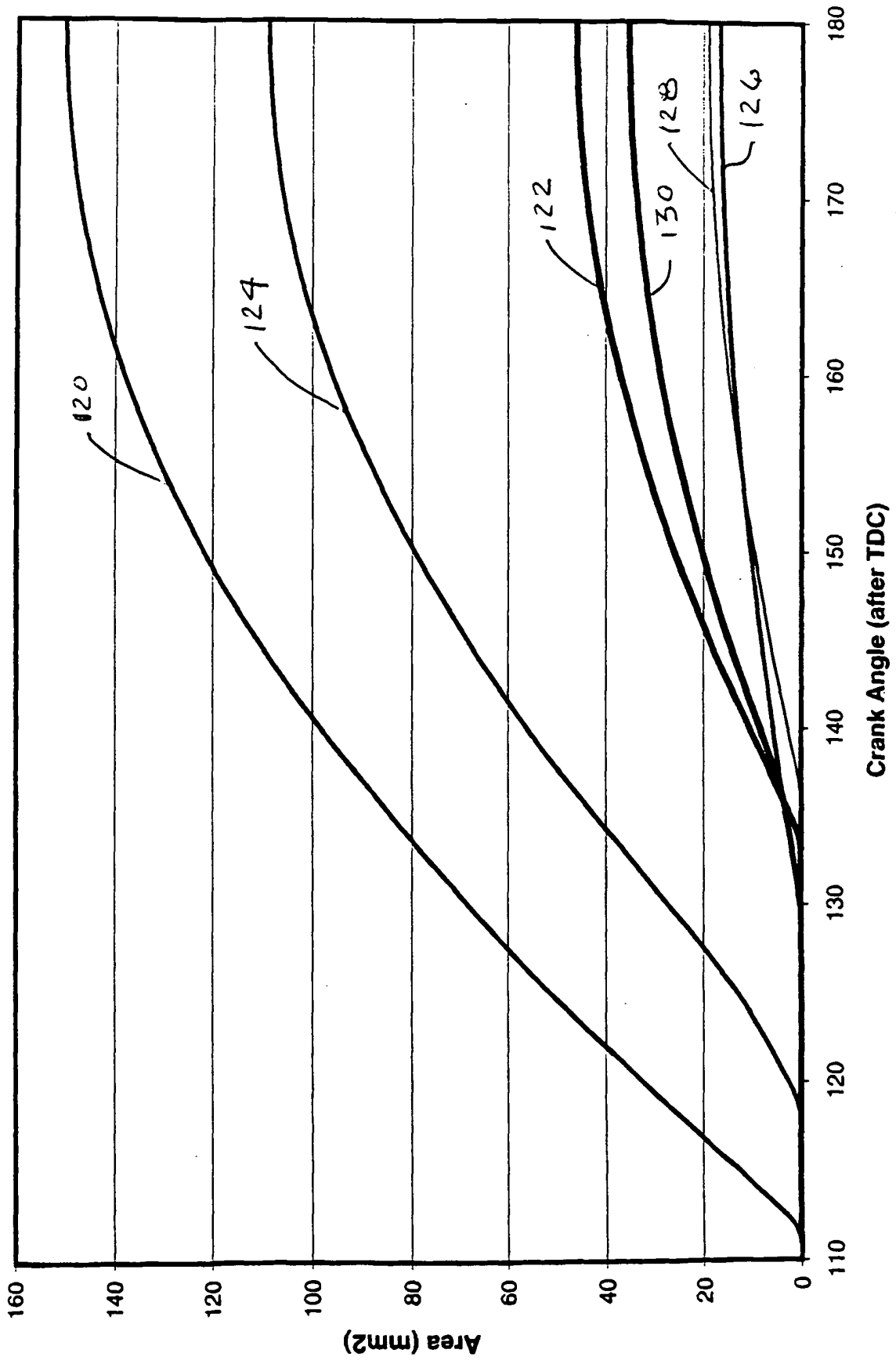


FIG. 10

REFERENCES CITED IN THE DESCRIPTION

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