(19) Europäisches Patentamt
European Patent Office
Office européen des brevets

(11) EP 3 771 812 A1

(12) EUROPEAN PATENT APPLICATION

(43) Date of publication:
**03.02.2021 Bulletin 2021/05**

(51) Int Cl.:
***F02B 33/20*** *(2006.01)* *F01B 9/02* *(2006.01)*
*F02B 75/02* *(2006.01)*

(21) Application number: **20186473.3**

(22) Date of filing: **17.07.2020**

(84) Designated Contracting States:
**AL AT BE BG CH CY CZ DE DK EE ES FI FR GB GR HR HU IE IS IT LI LT LU LV MC MK MT NL NO PL PT RO RS SE SI SK SM TR**
Designated Extension States:
**BA ME**
Designated Validation States:
**KH MA MD TN**

(30) Priority: **01.08.2019 SE 1930242**

(71) Applicant: **Gustafsson, Fredrik**
**235 43 Vellinge (SE)**

(72) Inventor: **Gustafsson, Fredrik**
**235 43 Vellinge (SE)**

(74) Representative: **AWA Sweden AB**
**P.O. Box 5117**
**200 71 Malmö (SE)**

# (54) TWO STROKE HIGH PERFORMANCE PISTON PUMP ENGINE

(57) A two stroke combustion engine comprising at least one combustion cylinder (13) having a cylinder center axis (19), a combustion piston (12) movable within the combustion cylinder (13), the combustion piston (12) having a first compression direction (CD) along the cylinder center axis (19). The two stroke combustion engine further comprising at least one separate piston pump (3a) having a piston pump cylinder (3b) with a piston pump center axis (20b) and a pump piston (25) movable within the piston pump cylinder (3b). The separate piston pump (3a) is in movable mechanical connection (18) with the combustion piston (12). The two stroke combustion engine further comprising an air inlet (10) having an air inlet center axis (20a). The separate piston pump (3a), the air inlet (10) and the combustion cylinder (13) are in fluid connection with each other by a transfer channel (11), and the piston pump center axis (20b) intersects a cylinder plane (X-19) extending from the cylinder center axis (19), and the intersection defines a ray (19') extending from the intersection in the first compression direction (CD). A normal axis (N) extends perpendicularly from the cylinder plane (X-19), the normal axis (N) intersects the piston pump center axis (20b). An angle, measured from the intersection between the ray (19') and a point on the piston pump center axis (20b), located in the piston pump cylinder (3b), is between 30° and 160°.

EP 3 771 812 A1

## Description

### Field of the invention

**[0001]** The present invention relates to a high performance two stroke engine with a piston pump and crankcase lubrication.

### Background art

**[0002]** The current two stroke technology is characterized by engines with low weight and relatively few parts compared to other existing engine technologies such as the typical four stroke engine. Due to that the two stroke engine has twice as many combustions per engine revolution compared to a four stroke engine it has a high potential for a performance advantage. The relatively simple design of the two stroke engine makes it interesting for many applications where cost and available space are important factors. However, the current two stroke technology has some drawbacks limiting the further utilization of this technology. To lubricate a conventional two stroke engine, oil is needed to be added to the air/air-fuel mixture to lubricate the engine parts such as the bearings on the crankshaft and to act as lubrication between the piston and the cylinder walls. This added oil will give higher emissions. Due to the configuration of a two stroke engine using the crankcase for pressurizing the air/air-fuel mixture to act as an air pump it has been difficult to use a separate crankcase lubrication principle.

**[0003]** The conventional two stroke engine with the crankcase pump principle has an airflow path design from the air inlet into the combustion cylinder which is not favorable due to some important reasons: The air primarily enters the crankcase before entering the combustion cylinder and will have to make sharp angle turns and change of flow directions. The flow is also negatively affected by the moving parts in the crankcase such as the conrod. Furthermore, in a conventional two stroke engine the air flow, for each cycle into the engine, is normally much less than the crankcase volume. Consequently, the air is typically not transferred to the combustion cylinder until several additional cycles after entering into the crankcase and there is more time for the air to heat up inside of the crankcase. The surfaces in the crankcase are heated by the gearbox oil, conduction from the combustion cylinder, bearing friction and hot exhaust gases that enters the crankcase. The air/air-fuel temperature therefore rises and the density decreases which give a less efficient engine with lower power output.

**[0004]** Performance is to a large extent depending on the air mass flow into the engine and air mass flow is improved by higher air speed and increased air density into the engine. With a lower air inlet temperature and a higher pressure the air density is increased and consequently the power output.

**[0005]** Engines with piston pumps have been designed to make it possible to utilize the possibility of supercharging and crankcase lubrication and not having to add oil directly to the air/air-fuel mixture.

**[0006]** The prior art piston pump engines usually have a design where the piston pump is mounted relatively far away from the combustion cylinder and/or has the piston pump outlet mounted such that the outflow is not primarily towards the combustion cylinder/transfer ports. One of the reasons for this is that the piston pump is driven by a con rod mounted to the combustion engine crankshaft making it impossible to have the crown of the pump piston to be directed towards the combustion cylinder.

**[0007]** In addition, prior art solutions have long flow lengths, and most often different types of sharp bends. Also, valves and area restrictions are used when guiding the air flow into the combustion cylinder. This imposes flow resistance that gradually increases with the engine RPM and this limits the possibility to utilize the engine concept for high performance high RPM engine concepts.

**[0008]** Another problem with long flow lengths is the drawback that the air/air-fuel mixture has more time to heat up in contact with the channel/engine walls, with a following power output decrease.

**[0009]** Other solutions for piston pump engines have been to utilize the supposedly compressed air in the crankcase to move a separate piston to act as a pump. It is not likely that this type of arrangement will be successful in rendering a high performance high rpm engine. Somewhere along the rpm increase there will be a delay and/or resonance in the way the pump piston synchronization versus the combustion cylinder piston works and the effect will be lost power and efficiency.

**[0010]** Another problem is that it is doubtful there will actually build up enough pressure in the crankcase to effectively move the piston and the piston without mechanical guidance.

**[0011]** It is the purpose of the invention to make it possible to enable an improved power output compared to existing conventional piston pump two stroke technology.

### Summary

**[0012]** It is an object to mitigate, alleviate or eliminate one or more of the above-identified deficiencies in the art.

**[0013]** An object of the Invention is to enable increased power output levels compared to existing two stroke technology using solutions for isolating the crankcase from the air/air-fuel mixture using the piston pump principle.

**[0014]** According to one embodiment, the two stroke combustion engine comprising at least one combustion cylinder having a cylinder center axis, a combustion piston movable within the combustion cylinder, the combustion piston having a first compression direction along the cylinder center axis. The two stroke combustion engine further comprising at least one separate piston pump having a piston pump cylinder with a piston pump center axis and a pump piston movable within the piston pump cylinder. The separate piston pump is in movable me-

chanical connection with the combustion piston. The two stroke combustion engine further comprising an air inlet having an air inlet center axis. The separate piston pump, the air inlet and the combustion cylinder are in fluid connection with each other by a transfer channel, and the piston pump center axis intersects a cylinder plane extending from the cylinder center axis, and the intersection defines a ray extending from the intersection in the first compression direction. A normal axis extends perpendicularly from the cylinder plane, the normal axis intersects the piston pump center axis. An angle, measured from the intersection between the ray and a point on the piston pump center axis, located in the piston pump cylinder, is between 30° and 160°.

**[0015]** According to one embodiment, the angle between the first compression direction and air inlet center axis is less than 150°.

**[0016]** According to one embodiment, the air inlet comprises a one directional valve substantially allowing air to flow in an air inlet direction and hindering air from flowing in a direction opposite to the air inlet direction. This gives the possibility to enabling the potential of a high performance exhaust system to help the suction of fresh cold air/air-fuel through the one directional valve, enabled to be positioned close towards the combustion cylinder. This is also a solution used in a conventional two stroke engine but here the air flow will not be spread out into a crankcase volume and stay there for a number of revolutions before entering into the transfer ports. An alternative embodiment, for increased long term valve durability and tuning for a more limited RPM range compared to the one directional valve is to use an inlet valve that opens and closes when the combustion piston is at two specific positions in the cylinder. One such valve could be of the rotational type as known in the art.

**[0017]** According to one embodiment, the movable mechanical connection between the piston pump and crankshaft comprises at least one of: a belt drive, gear drive and a chain drive.

**[0018]** According to one embodiment, the movable mechanical connection is based on a Skotch yoke design. Compared to a conventional piston-conrod-crank pump the Skotch yoke has lower side forces and less friction surface area to the pump cylinder wall. In addition, the Skotch yoke principle gives the possibility to a more compact engine

**[0019]** According to one embodiment, the movable mechanical connection is based on a crankshaft-conrod-piston design.

**[0020]** According to one embodiment, the movement of the combustion piston and the pump piston is synchronized by means of at least one of; electrical motor, electrical solenoid, hydraulic motor, hydraulic solenoid, air motor, air solenoid or a combination thereof.

**[0021]** By using a separate piston pump positioned and designed to be enabled to push air from the piston pump outlet into a less resistance air flow path towards the transfer channels and transfer ports, into the combustion cylinder. This will increase the flow efficiency and power output. At higher RPMs it is increasingly important to decrease flow resistance through an engine to improve performance and power output. The piston pump mechanical drive secures that it is always is synchronized with the combustion piston position such that after one complete engine revolution the pump piston comes back to the starting point. A variable phasing mechanism may also be enabled to change the engine characteristics.

**[0022]** Having both the piston pump and the one directional valve close to the combustion cylinder the system can be enabled to be further optimized for a short air/air-fuel to hot surface contact time. This reduces the heating of the air/air-fuel and increases its density. This contributes to enable higher power outputs.

**[0023]** The invention may enable a higher primary compression, in combination with a lower primary compression volume, which may be used at certain RPMs to enable a delay of the air/air-fuel mixture flow into the combustion cylinder such that it has less time to escape out through the exhaust port. This delay can be enabled by allowing a higher combustion cylinder pressure at transfer port opening. This could for example be done by decreasing the timing of the exhaust port such that it opens later. This will also allow that more usable work is created before the exhaust opens and the pressure in the combustion cylinder is released.

**[0024]** The higher primary compression can decrease the level of hot exhaust gas spill out into the transfer channels at certain RPMs and load levels. This will further assist to keep the temperature of the fresh air/air-fuel lower and thereby increase its density.

**[0025]** It should be noted that elements from the described embodiments can be used in combination as long as this is not clearly contradictory. The description of the elements in connection with different embodiments should be seen as a way of facilitating the understanding and not as limiting to the ways the elements can be combined.

## Brief descriptions of the drawings

**[0026]** The invention will by way of example be described in more detail with reference to the appended schematic drawings, on which:

Fig. 1. Exemplary layout of the invention design principle,

Fig. 2a. Exemplary layout of the invention design principle using a belt driven Skotch yoke type piston pump,

Fig. 2b. Definition of Piston pump compression stroke projection towards the combustion cylinder,

Fig 3. Exemplary layout of the invention design principle using a belt driven piston-con rod-crank type piston pump,

Fig 4. Exemplary embodiment with a design principle of a Skotch yoke piston pump mechanism,

Fig. 5. Layout of the Lotus Engine Simulation software v.5.05 used for simulating how a more straight flow path can increase the power output for an embodiment of the invention,

Fig. 6. Simulation results showing an increase of power output as the piston pump compression stroke flow is better directed towards the transfer channels and transfer ports.

Detailed description

**[0027]** The present invention will now be described more fully hereinafter with reference to the accompanying drawings, in which currently preferred embodiments of the invention are shown. This invention may, however, be embodied in many different forms and should not be construed as limited to the embodiments set forth herein; rather, these embodiments are provided for thoroughness and completeness.

**[0028]** Although the invention is described hereinafter with more particular reference to a single cylinder engine the invention is not in any way restricted thereto and is also applicable to for example two stroke in-line or V-engines with more than one cylinder as well as engines with two cylinders with pistons sharing a common combustion chamber.

**[0029]** The invention can enable a higher primary compression pressure, and a comparably lower temperature of the air/air-fuel mix to other conventional two stroke engines and with a possibility to at certain RPMs and load levels enable a faster inflow with higher air/air-fuel density into the combustion cylinder. At higher RPMs it is increasingly important to decrease flow resistance throughout an engine to improve performance and power output.

**[0030]** The invention uses a lubrication principle found in four stroke engines, and piston pump two stroke engines, with a better direct access for the oil to the cylinder walls and the piston compared to conventional two stroke engines. This makes it possible to enable a higher power output due to the possibility for better sustained engine lubrication. The definition "piston pump engine" is generally used here for engines with a separate piston pump that is used for pumping air into the engine and to the combustion cylinder instead of using the lower side of the combustion piston as a "pump piston" instead of the upper pump piston crown.

**[0031]** The invention has a piston pump mounted close to the combustion cylinder and is characterized as being mounted in direct contact with the transfer channels by means of having its own separate piston pump driving mechanism. In **Fig. 1.** a schematic exemplary embodiment of the engine layout can be seen. In **Fig. 2a** and **Fig.3** exemplary piston pumps **21,24** of the invention design are shown mounted directly to the combustion cylinder **13.** In addition to eliminating the crankcase **26** flow for the air/air-fuel mix, where it spends several cycles before moving into the transfers channels **11** as the volume of the crankcase is normally several times the combustion cylinder volume, this position of a piston pump enables the possibility to use a very efficient flow path for the air as it is pushed towards the transfer channels and transfer ports **47** in an lower (straighter) flow resistance air flow path compared to prior art. Longer air flow paths, contractions and bends as seen on prior art piston pump engines can be avoided and thereby the air/air fuel volume has a shorter contact time with the hotter engine surfaces and this lowers the intake temperature which gives a higher power output, compared to prior art conventional and piston pump two stroke engines.

**[0032]** The interface 1b between the pump piston **25** crown face **27** and the transfer channels **11** is characterized in that, in addition to that the flow length between piston pump and combustion cylinder can be minimized, it can be enabled such that it has no airflow restriction induced by a valve or a pipe section of reduced area to the pump piston crown face area. For a lower power embodiment of the invention a restriction plate or other means of narrowing the area of the interface between the pump piston crown and transfer channels can be utilized.

**[0033]** In another embodiment of the invention the piston pump and combustion cylinder and crankcase are arranged with one or more liquid cooling circuits, this will further assist to lower the induction temperature.

**[0034]** The piston pump is giving one compression and one suction stroke per combustion cylinder revolution.

**[0035]** Not all air/air-fuel volume goes to and from the separate pump cylinder **3** before entering the transfer ports **47.** Flow can pass directly from the air inlet, passing the transfer channels **11** towards the transfer ports **47** and a tuned exhaust pipe **5** may assist this flow, as is known in the art.

**[0036]** A Skotch yoke type piston pump design may be enabled for less friction losses and a more compact design. Other solutions that could be enabled can for example be a piston-con rod-crank design. Any mechanism that precisely can, without variations depending on engine revolutions per minute, move the pump piston in a 1:1 relation to the combustion piston motion could potentially be utilized.

**[0037]** Compared to a conventional piston-conrod-crank pump the Skotch yoke has less side forces and less friction surfaces to the pump cylinder wall. In addition, the Skotch yoke design principle gives the possibility to a more compact engine as seen comparing the design of **21** and **24.** The scotch yoke mechanism gives essential an axial movement, with limited side forces, on the pump piston **25** in the piston pump cylinder. This reduces the friction work for the piston pump. The pump is in one embodiment of the invention enabled to be lubricated by oil/oil mist from the crankcase using a tube arrangement from the crankcase. In another embodiment of the invention the piston pump can have its own internal lubrication system.

**[0038]** The piston pump center axis **20b** is positioned at an angle position to the combustion cylinder center axis **19** at a for the specific engine layout suitable angle **B,** as seen in **Fig.2a.** in the range of 30°<B°<160°. Preferably the angle **B** should be in the range of 70°<B°<140°. The air inlet center axis angle **20a** is positioned to the combustion cylinder center axis **19** at a for the specific engine layout suitable angle A° to the combustion cylinder center axis in the range of 0°<B°<180°, preferably in the range of 0°<B°<150°.

**[0039]** In **Fig 2b.** the definition of the piston pump center axis **20b** and the projection of the piston pump **61** compression stroke **P** is defined; the piston pump center axis **20b** intersects a cylinder plane **X-19** extending from the cylinder center axis **19,** and the intersection defines a ray **19'** extending from the intersection in the first compression direction **CD,** and a normal axis **N** extends perpendicularly from the cylinder plane **X-19,** the normal axis **N** intersects the piston pump center axis **20b,** characterized in that an angle **B°,** measured from the intersection, between the ray and a point on the piston pump center axis **20b,** located in the piston pump cylinder **3b,** is between 30° and 160° but preferably in the range of 70°<B°<140°.

**[0040]** As stated above the crankcase **26** volume as a part of the primary compression volume is eliminated in the invention and this makes it possible to enable a higher primary compression. The primary compression is defined as: (Piston pump cylinder volume+ Volume downstream of air inlet to the transfer ports)/ (Volume downstream of air inlet to the transfer ports).

**[0041]** The higher primary compression that can be enabled, due to the close mount of the piston pump and one directional valve to the transfer channels, can be used to enable a higher pressure at the transfer ports and transfer channels **11** representing the primary compression volume. The comparably small volume of the same, makes it more difficult for a higher combustion cylinder **13** pressure, at transfer port opening, to cause a severe combustion gas to spill deep into the transfer channels and cause a more uncontrollable mix with the fresh air/air-fuel mix before the pressure in the transfer channels and cylinder is equalized and the spill outflow can stop and the flow reverses. If the combustion gas mix with the fresh air/air-fuel it will be heated up and will get a decreased purity. It will also heat up the transfer channel surfaces causing heat up of the air/air-fuel mixture in the next coming cycles. This leads to a less efficient engine function. The embodiment can allow combustion gas in a more controlled way into a part of the transfer channels volume, pressurizing the smaller total primary compression volume more quickly and to a higher pressure. The gas volume closest to the transfer port in the narrow transfer channels closest to the transfer ports will therefore at certain RPMS and load levels, have a high combustion gas content and this gas will at reversal flow be first to go out into the exhaust in case of short cutting when the gas enters the cylinder. This will limit the amount of short cut of fresh air/air-fuel volume and more of its volume will stay inside of the combustion cylinder and increase the power output.

**[0042]** The increase of the pressure in the transfer channels due to the relatively more controlled spill out of a part of the combustion gas will increase the pressure and could be enabled to assist to a quicker reentering of the following air/air-fuel mixture. The high pressure will cause a faster reentering and the higher speed will cause a higher turbulence and better mix of air and fuel. In addition, as the flow is faster, a stronger suction pulse behind may enable a comparably lower pressure in the transfer channels at the point when the transfer ports are closing. This will then assist in the next cycle to give a stronger suction pulse signal to the one directional valve 9 which will flow better and more efficiently fill the transfer channels and piston pump cylinder volume with fresh air/air-fuel volume. The effect may be further assisted by an efficient tuned exhaust system.

**[0043]** During the delay of the flow from the transfer ports into the combustion cylinder, the exhaust gas in the combustion cylinder will have a longer time to leave the cylinder before the air/air-fuel mixture starts to flow. As stated earlier, the first gas from the transfer ports can at certain RPMs and loads have the possibility to be enabled to be of high combustion gas content.

**[0044]** The enabling of a high primary compression in combination with a lower primary compression volume may allow for optimizing the exhaust port configuration compared to prior art. Due to the earlier explained delay of outflow of air/air-fuel into the combustion chamber the exhaust port 6 can be enabled to open later and the port 6 size can be made smaller. This is because there is more time for the exhaust to leave the combustion chamber before air/air-fuel is entering. This improves the fuel efficiency and power output of the engine due to that more air/air-fuel is trapped in the combustion cylinder but also due to that more usable work is created before the exhaust port opens and the pressure in the combustion cylinder is released.

**[0045]** Due to the design of the invention where the piston pump drive is not using the combustion cylinder conrod or an additional conrod mounted on the engines crankshaft, or any believed unpredictable gas crankcase pressure, it is easy to adjust the phasing position between the combustion cylinder and pump piston. The phasing can alter the pumps suction and pushing timings versus for example transfer ports openings such that the power and characteristics of the engine can change. When we have lower load and relatively lower RPM on the engine it can for example be to an advantage to decrease the pressure in the transfer channel at transfer port opening to delay or decrease the inflow of the air/air-fuel flow into the combustion cylinder such that it has less time to short circuit out through the exhaust port and exhaust channel and be lost for combustion work.

**[0046]** A phasing mechanism can be utilized in an embodiment of the invention such that the phasing can be changed continuously in the interface between a piston pump crank **34** and, in one embodiment, a piston pump crankshaft drive wheel **38.** One possibility for such a gear function is to use the centrifugal force to push weights radially in between two attached wheels, a driving wheel and a driven wheel, where both wheels have angled tracks such that the wheels are phased differently depending on rotational speed, when the weights are moving in and out radially and to some extent axially. To each wheel a drive shaft is mounted such that the angle phase between the shafts changes. Such a solution is presented in patent application US-2016/0010517-A1 "Valve gear for engine", Suzuki Motor Corporation®

**[0047]** In another embodiment of the invention the piston pump cylinder position, the pump piston interface **1b,** can be manually or continuously and automatically moved in or out in relation to the transfer channel **11.** This will make the primary compression variable such as to have the possibility to enable an optimized efficiency and power output for each RPM and load level. For example it can be to an advantage to decrease the primary compression to make engine start easier as the primary compression is lower. In such an embodiment, the piston pump cylinder needs to be possible to slide in and out in relation to the transfer channel. The piston pump drive mechanism must also be made such that it can comprehend the piston pump movement.

**[0048]** In another embodiment of the invention, as an alternative, or a compliment to an embodiment with the variable position of piston pump cylinder and mechanism, at least one separate chamber is added to the transfer channels volume. Such a volume chamber is regulated with a valve to let air flow in and out. The purpose of the chamber is to change the primary compression volume to further optimize performance characteristics and power output. It can also be used to decrease the force needed for starting up the engine. Such a system is described in for example patent EP-1018596-B1 "Method of modulating the power of a two stroke internal combustion engine with a resonator connected to the crankcase." The term crankcase shall be interpreted as the "transfer channels" used in the current invention.

### *Simulation of the power output of one exemplary embodiment of the engine*

**[0049]** To verify the importance of the positioning and alignment of the piston pump for the power output the simulation program Lotus Engine simulation v.5.05 has been used. The simulation includes the mounting of the piston pump to the combustion cylinder with the piston pump outlet directions variated to show the power output improvement for a less bent and more straight air flow path towards the combustion cylinders transfer channels and transfer ports.

**[0050]** In **Fig.5** the schematic simulation set up is visualized with graphics taken from the simulation program.

**[0051]** As can be seen from the diagram the power output increases when the bending angle on the section between the piston pump outlet and the transfer channels, leading to the cylinder, decreases and the flow path straightens. The flow loss decreases as the piston pump has a straighter line to push the air through, towards the transfer ports. The improvement is most noticeable at the higher RPMs where an improvement in power output of up to 40-45% can be seen.

### *Detailed description of the engine and engine parts of one embodiment*

**[0052]** An embodiment of a piston pump two stroke internal combustion engine shall now be described in more detail in conjunctions with **Figs. 1-5.** As illustrated in e.g. **Fig. 1-5** the engine comprises:

On top of the cylinder **13** a cylinder head **8** is positioned. The cylinder head has a spark plug. In another embodiment, at least one direct injection valve could be mounted in the cylinder head to substitute or assist a carburetor **4** or fuel injection valves, or elsewhere upstream of the exhaust port **6.** In the cylinder wall of the combustion cylinder are transfer ports **47** positioned. At least two transfer ports are distributed around the circumference of the combustion cylinder inner wall and are arranged in an annular configuration.

**[0053]** A piston **12** movable within the combustion cylinder between, as seen in **Fig. 2a,** a top position; the top dead center **14** and a bottom position; the bottom dead center **15.** In the top dead center, a cylinder volume of the combustion cylinder has a minimum value and in the bottom dead center, the cylinder volume has its largest value, as is known in the art.

**[0054]** The piston has at least one circumferential groove for a piston ring seal **22** and at least one annual positioned lubrication hole below this piston ring where oil from the crankcase using crankcase pressure can protrude to lubricate the cylinder wall.

**[0055]** The groove with the piston ring seal **22** is positioned at a higher part of the combustion piston, i.e. near the piston crown of the piston. One or a second seal and piston ring grooves are added to the lower part of the piston at **23** to force excessive oil down from the cylinder walls during the piston down stroke and limit the blow-by of oil and gases.

**[0056]** The lower part of the cylinder wall will receive oil moisture from an oil bath in the crankcase. One or more than one transfer channels **11** envelops the cylinder and connects to the cylinder transfer ports. At a suitable distance downstream of the one directional valve **9** the transfer channel is divided into several separate transfer channels, each leading to at least one transfer port **47.** The transfer channels connect to at least one air inlet port **10.** The transfer channels and air inlet are separated to the crankcase **26.**

**[0057]** The one directional valve **9** is mounted in the air inlet **10.** To the one directional valve the carburetor **4** is mounted. As an alternative, or complement to the carburetor, in an embodiment with a fuel injection, a throttle valve can be mounted into the air inlet. A fuel injection also needs an air speed/flow or air mass flow metering function.

**[0058]** The one directional valve is characterized by the function to allow the air/air-fuel only to essentially pass into the engine. When the pressure is higher upstream of the valve than downstream, air moves into the engine. When the pressure is higher downstream of the valve, in the transfer channels volume, than upstream of the valve, the valve function principle is to not let air move in the upstream direction, out of the engine.

**[0059]** At least one piston pump **3a,21,24** is directly linked to the volume around the cylinder that the transfer channels **11** create.

**[0060]** The piston pump sucks air through a one directional valve on its up stroke, as seen in **Fig. 2a,** moving towards its highest position **17** and compresses and sends a pulse of air/air-fuel mix towards the transfer channels and transfer ports on its down stroke towards its lowest position **16.**

**[0061]** The air on the underside of the combustion piston **12** is therefore not used for direct scavenging of the air fuel mix into the combustion cylinder **13.** The underside of the combustion piston is the surface of the piston which is faced away from the earlier mentioned cylinder volume, i.e. is not the surface of the piston which is in direct contact with the combustion cylinder combustion volume.

**[0062]** The piston pump is positioned such that its full piston face in one embodiment of the invention can be enabled to be in direct contact with the combustion cylinder enveloping transfer channels **11** which have the transfer ports **47** leading into the combustion cylinder. To enable a lower power output the interface area **1b** can use a contraction to limit flow.

**[0063]** The piston pump **21** can be enabled using a scotch yoke mechanism. An alternative can be a piston-con rod-crank solution and/or include a pneumatic-, hydraulic-, electrical or other alternative mechanical mechanisms.

**[0064]** The piston pump is driven by a synchronous belt drive **18** connecting the piston pump and crankshaft and the combustion piston crankshaft in a 1:1 relation. It can be expected that the belt drive has an efficiency of about 96-98% which causes losses that are well below the expected level that would significantly influence the power output advantage to compared prior art. As an alternative to a synchronous belt drive for example a gear drive or chain drive can be used.

**[0065]** The Skotch yoke principle pumping phasing and characteristics can in one embodiment be changed by changing the path of the track slope **31 Fig.4** for the mechanism.

**[0066]** The pump piston has a piston crown **27** and one or more sealing ring grooves with a seal **29.**

**[0067]** At least one exhaust outlet port **6** with a separate exhaust outlet channel **7** is provided which extends through the cylinder to the exhaust system **5.** The exhaust port opens before the transfer ports such that the combustion (exhaust) gases flows out from the combustion cylinder to the exhaust.

**[0068]** The bottom end is essentially mechanically as per a conventional high performance two stroke engine but with a crankcase lubrication as seen by prior art piston pump engines.

**[0069]** The crankcase of the engine and the piston pumps are well ventilated such as to minimize pumping losses.

***Principal function of the exemplary embodiment***

**[0070]** The crankshaft drives the synchronous belt drive **18** which drives the piston pump **21.** When the pump piston is rising inside the piston pump cylinder towards its highest position **17** from its lowest position **16** the air pressure in the transfer channels **11** and in the piston pump cylinder is decreasing. At the same time the combustion piston **12** is moving from the region of the bottom dead center **15** towards the top dead center **14.** When the pump piston has moved some distance into the piston pump cylinder the transfer ports **47** are covered by the raising combustion piston. The low pressure being produced, by the moving piston pump in the transfer channels, causes the ambient air to flow through the carburetor **4** and through the one directional valve **9.** A negative pressure pulse from the exhaust could also contribute to a lower pressure in the transfer channels before closing of the transfer ports.

**[0071]** When the pump piston has continued to move to the highest position **17,** and the combustion piston is close to its top dead center, the air/air-fuel mix is ignited by the spark plug and is combusted. When the pump piston then continues and moves towards the lowest position **16** it compresses the air/air-fuel mix in the piston pump cylinder and the transfer channels. And the efficient pumping action and straight flow path towards the transfer channels and transfer ports contributes to efficiently fill up the volume there with air/air-fuel mass.

**[0072]** The air/air-fuel mix can due to flow inertia effects still be flowing through the one directional valve for some time after the pump piston has started to move down. As the compression is rising this flow will finally stop. At one point during the descending of the pump piston towards its lowest position **16** the combustion piston has moved down and the exhaust port **6** opens and the combustion gas flows out through the exhaust port and through the exhaust outlet channel **7** and into the exhaust system **5.** When the piston pump is closer to the lowest position **16** the combustion piston opens the transfer ports **47** into the combustion cylinder and depending on the load level and RPMs the flow into combustion cylinder will start at a different time after opening. The compressed air/air-fuel mix begins to flow into the cylinder **13** when the pressure becomes lower in the cylinder compared to the transfer channels and the outflow, due to inertia forces, from the cylinder has stopped. The air/air-fuel mixture, and any combustion gas, is helping to push out the combustion gas from the cylinder and fills it with air/air-fuel mixture. The combustion piston is now moving towards the lowest position **16** and a new cycle can begin.

***Figure reference identifications***

**[0073]**

1a. Combustion cylinder piston crown
1b. Pump piston interface to transfer channel
2. Combustion cylinder wall
3a. Piston pump
3b. Piston pump cylinder
4. Carburetor
5. Exhaust system
6. Exhaust port
7. Exhaust outlet channel
8. Cylinder head
9. One directional valve
10. Air inlet
11. Transfer channels/transfer channel volume
12. Combustion piston
13. Combustion cylinder
14. Combustion piston Top dead center (TDC)
15. Combustion piston Bottom dead center (BDC)
16. Pump piston lowest position
17. Pump piston highest position
18. Crankshaft to Piston pump driving mechanism
19. Combustion cylinder center axis
20a. Air inlet center axis (angle A)
20b. Piston pump center axis (angle B)
21. Piston pump (Skotch yoke)
22. Combustion piston upper seal
23. Combustion piston lower seal
24. Piston pump (piston-con rod-crank type)
25. Pump piston
26. Crankcase
27. Pump piston crown
28. Pump piston body
29. Pump piston seal
30. Pump piston body sliding surface
31. Track slope for slider ring
32. Skotch yoke track slider ring
33. Bearing
34. Crank
35. Bearing pin
36. Balancing wheel
37. Crank bearings
38. Belt drive wheel
47. Transfer ports
60. Exemplary Skotch yoke pump mechanism layout
61. Piston pump compression stroke P projection on XY-Z Plane
19'. Ray from intersection point 61 in the CD direction
CD. Combustion piston 12 compression direction
P. Compression direction for the pump piston
X-19. The intersection plane for the piston pump compression direction P to combustion cylinder center axis 19.
N. A normal plane to plane X-19.

**[0074]** Additionally, variations to the disclosed embodiments can be understood and effected by the skilled person in practicing the claimed invention, from a study of the drawings, the disclosure, and the appended claims.

**Claims**

**1.** A two stroke combustion engine comprising:

at least one combustion cylinder (13) having a cylinder center axis (19), a combustion piston (12) movable within the combustion cylinder (13), the combustion piston (12) having a first compression direction (CD) along the cylinder center axis (19), a closed crankcase (26) sealed off from the combustion cylinder (13) and the transfer channels (11) by the combustion piston (12),

at least one separate piston pump (3a) having a piston pump cylinder (3b) with a piston pump center axis (20b) and a pump piston (25) movable within the piston pump cylinder (3b), wherein the separate piston pump (3a) is in syncronized movable mechanical connection (18) with the combustion piston (12), and

an air inlet (10) having an air inlet center axis (20a), wherein:

the separate piston pump (3a), the air inlet (10) and the combustion cylinder (13) are in fluid connection with each other by a transfer channel (11),

the piston pump center axis (20b) intersects a cylinder plane (X-19) extending from the cylinder center axis (19), and the intersection defines a starting point (61) of a ray (19') extending from the intersection (61) in the first compression direction (CD), and

a normal axis (N) extends perpendicularly from the cylinder plane (X-19), the normal axis (N) intersects the piston pump center axis (20b), **characterized in that** the pump piston has its compression direction towards the plane (X-19), and towards the combustion cylinder, and wherein an angle (B), measured from the intersection between the ray (19') and a point on the piston pump center axis (20b), located in the piston pump cylinder (3b), is between 30° and 160°.

**2.** The two stroke combustion engine according to the claim 1, wherein the air inlet comprises a one directional valve (9) substantially allowing air to flow in an air inlet direction and hindering air from flowing in a direction opposite to the air inlet direction.

**3.** The two stroke combustion engine according to any one of claims 1-2, wherein the movable syncronized mechanical connection (18) between the piston pump (3a) and crankshaft comprises at least one of: a belt drive, gear drive and a chain drive

**4.** The two stroke combustion engine according to claim 3, wherein the movable mechanical syncronized connection (18) is connected to a piston pump driving mechanism based on a Skotch yoke design.

**5.** The two stroke combustion engine according to claim 3, wherein the movable mechanical syncronized connection (18) is connected to a piston pump driving mechanism based on a crankshaft-conrod-piston design.

**6.** The two stroke combustion engine according to any one of claims 1-5, wherein the movement of the combustionpiston (12) and the pump piston (25) is synchronized by means of at least one of; electrical motor, electrical solenoid, hydraulic motor, hydraulic solenoid, air motor, air solenoid or a combination thereof.

**7.** The two stroke combustion engine according to any one of the preceding claims, further comprising a variable phasing mechanism.

**8.** The two stroke combustion engine according to any one of the preceding claims, wherein the angle (B), measured from the intersection between the ray (19') and a point on the piston pump center axis (20b), located in the piston pump cylinder (3b), is between 70° and 140°.

**9.** The two stroke combustion engine according to any one of claims 1, 2 and 4 - 8, wherein the air inlet comprises an opening and closing valve, wherein the opening and closing valve is adapted to:

open when the combustion piston is in a first position in the combustion cylinder, and
closed when the combustion piston is in a second position in the combustion cylinder.

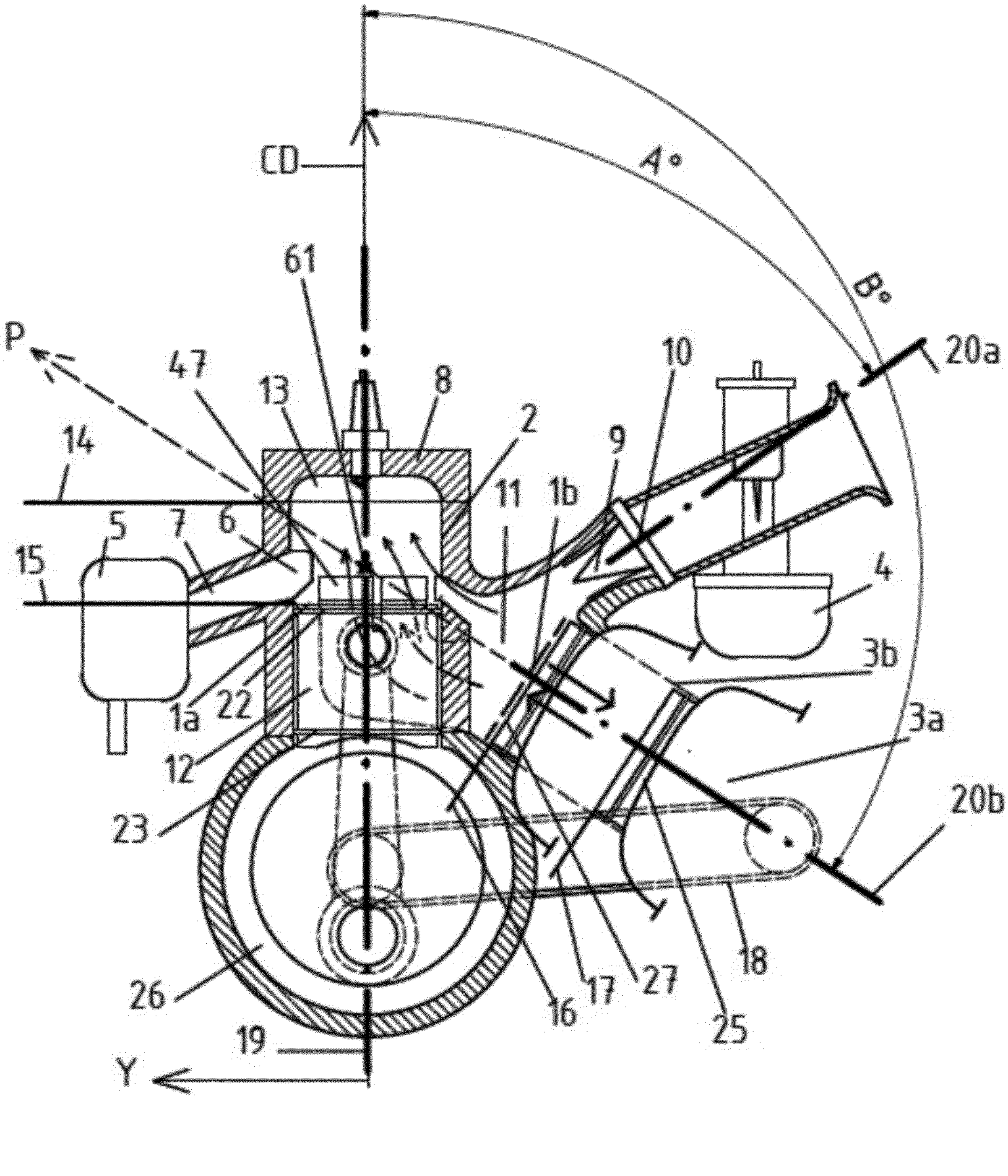
CD
A°
B°
61
P
47
13
8
2
10
9
20a
14
11
1b
5
7
6
15
4
3b
1a
22
3a
12
20b
23
18
26
16
17
27
25
19
Y

Figure 1.

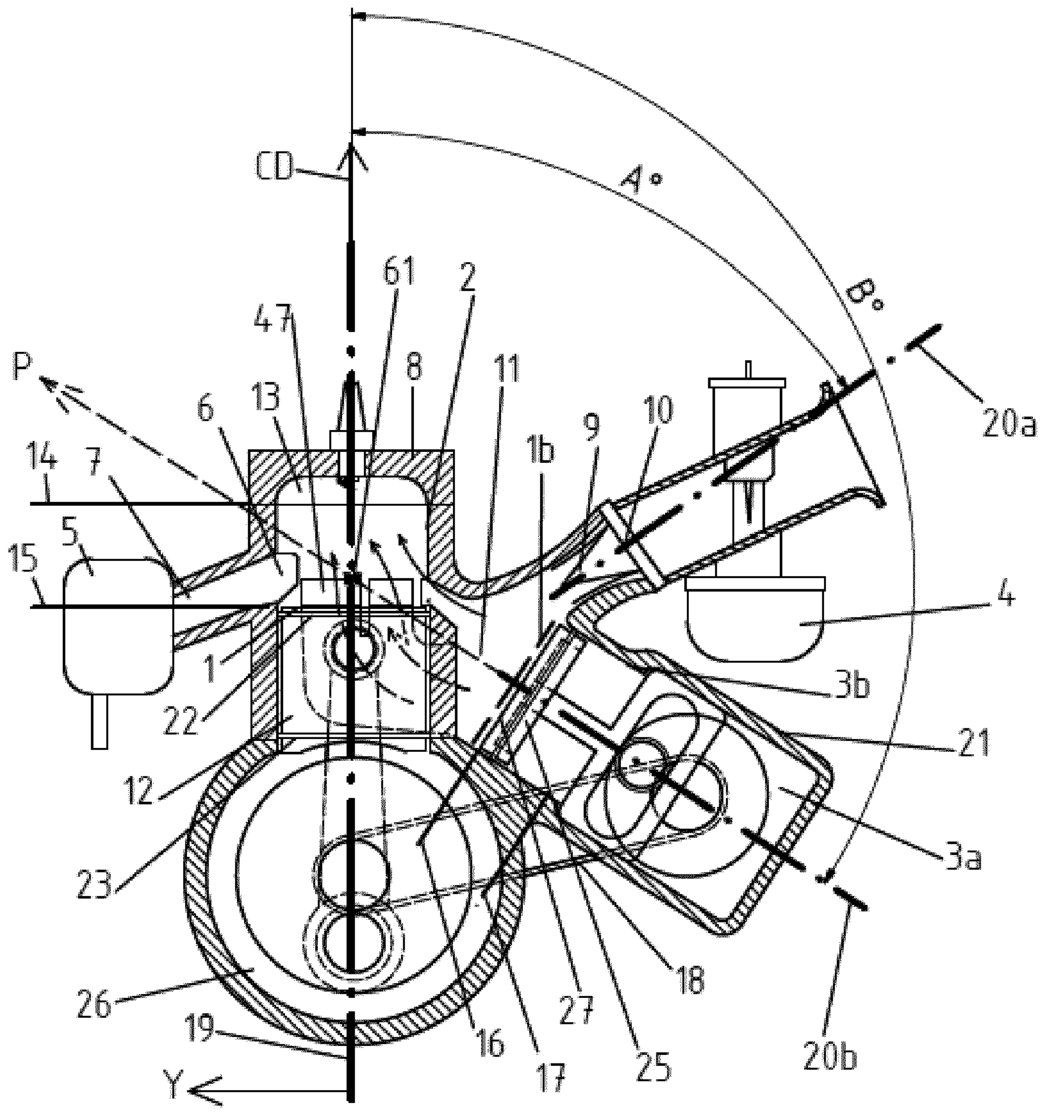
CD
A°
B°
P
47
61
2
8
11
13
6
14
7
5
15
1
22
12
23
26
19
Y
1b
9
10
20a
4
3b
21
3a
18
27
25
16
17
20b

Figure 2a.

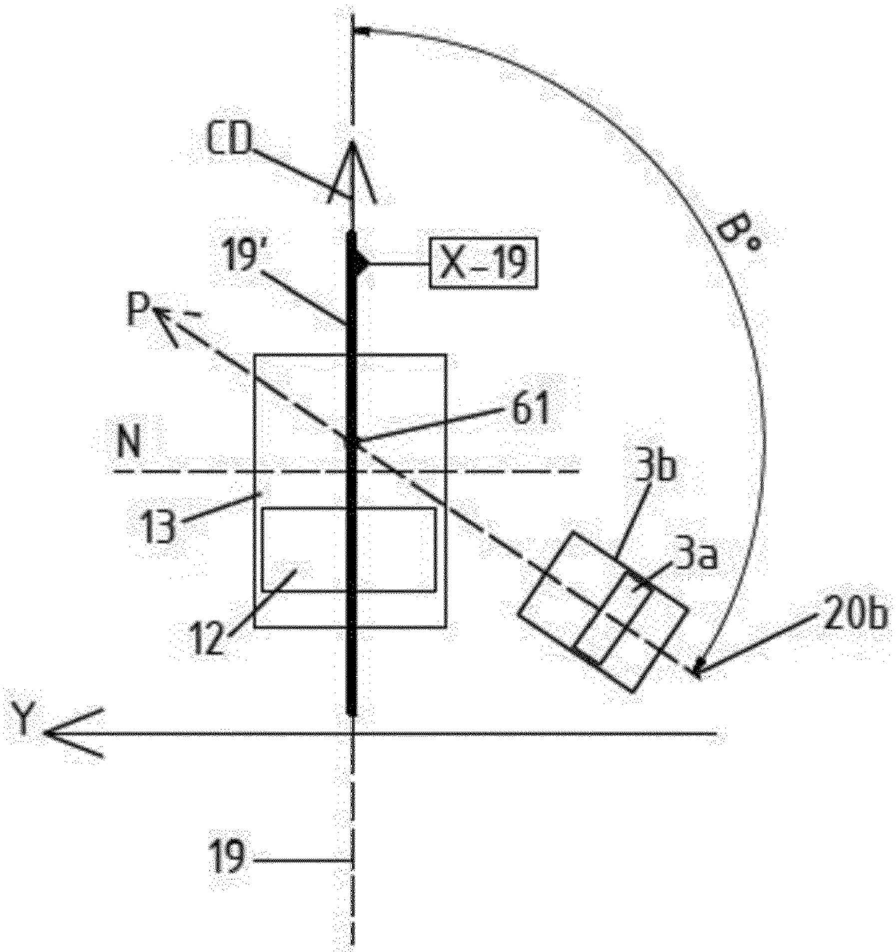


Figure 2b.

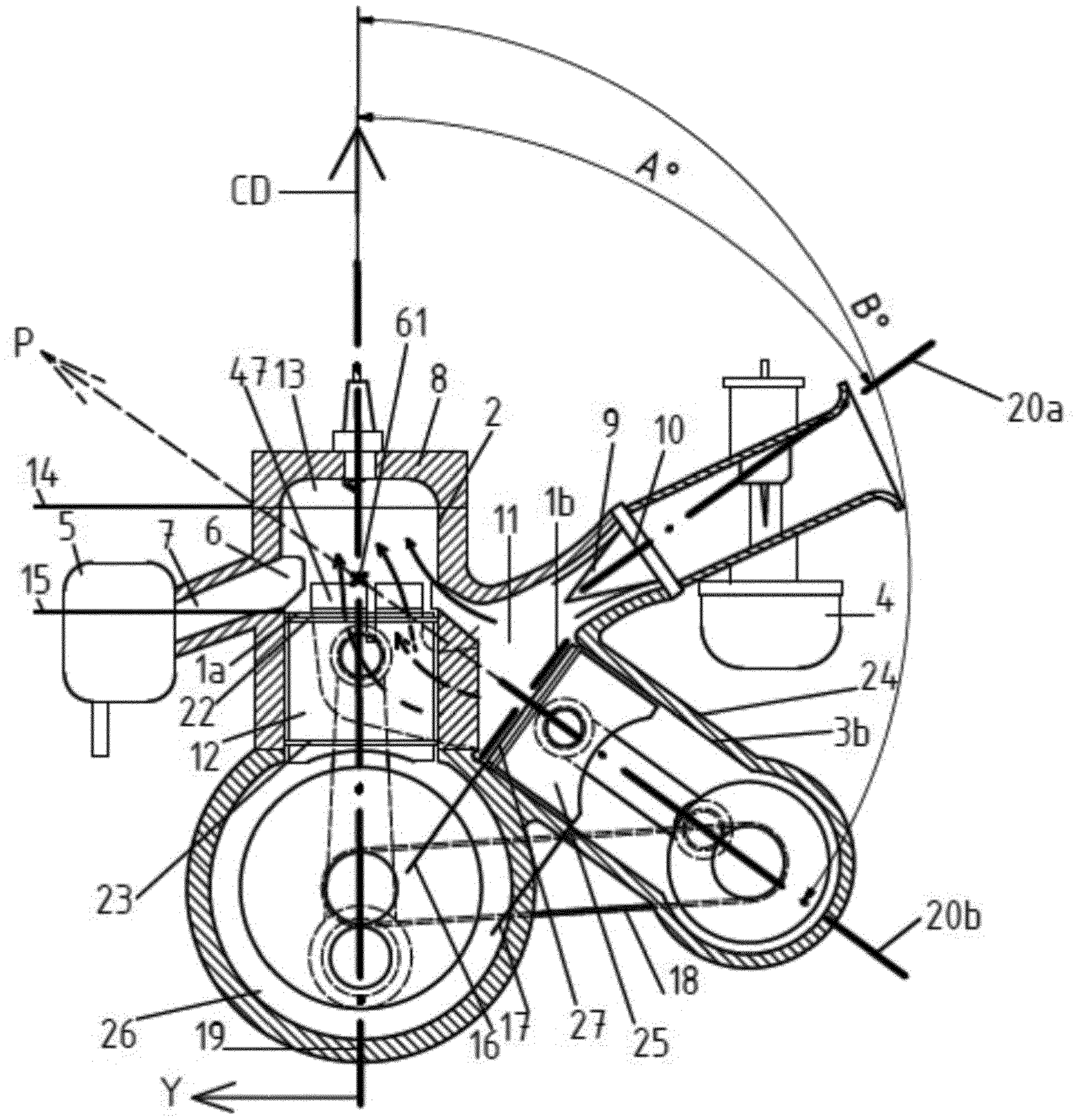
CD
A°
B°
P
61
47
13
8
2
9
10
1b
11
20a
14
5
7
6
15
1a
22
12
4
24
3b
23
26
19
16
17
27
25
18
20b
Y

Figure 3.

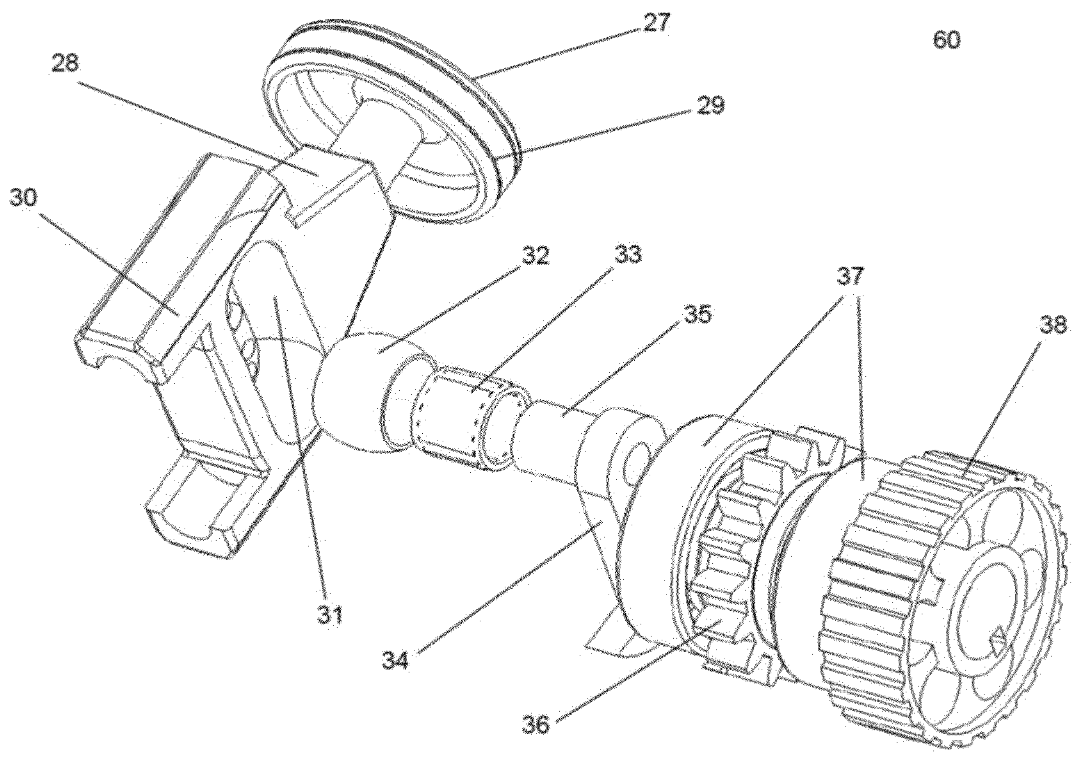


Figure 4.

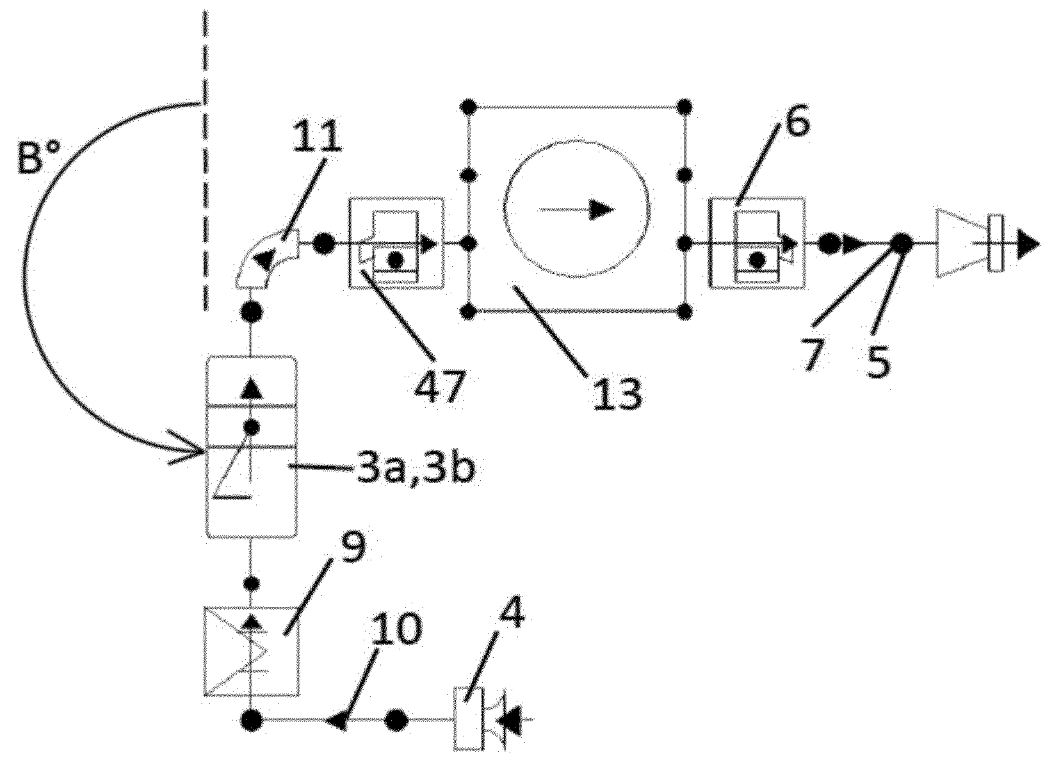


Figure 5.

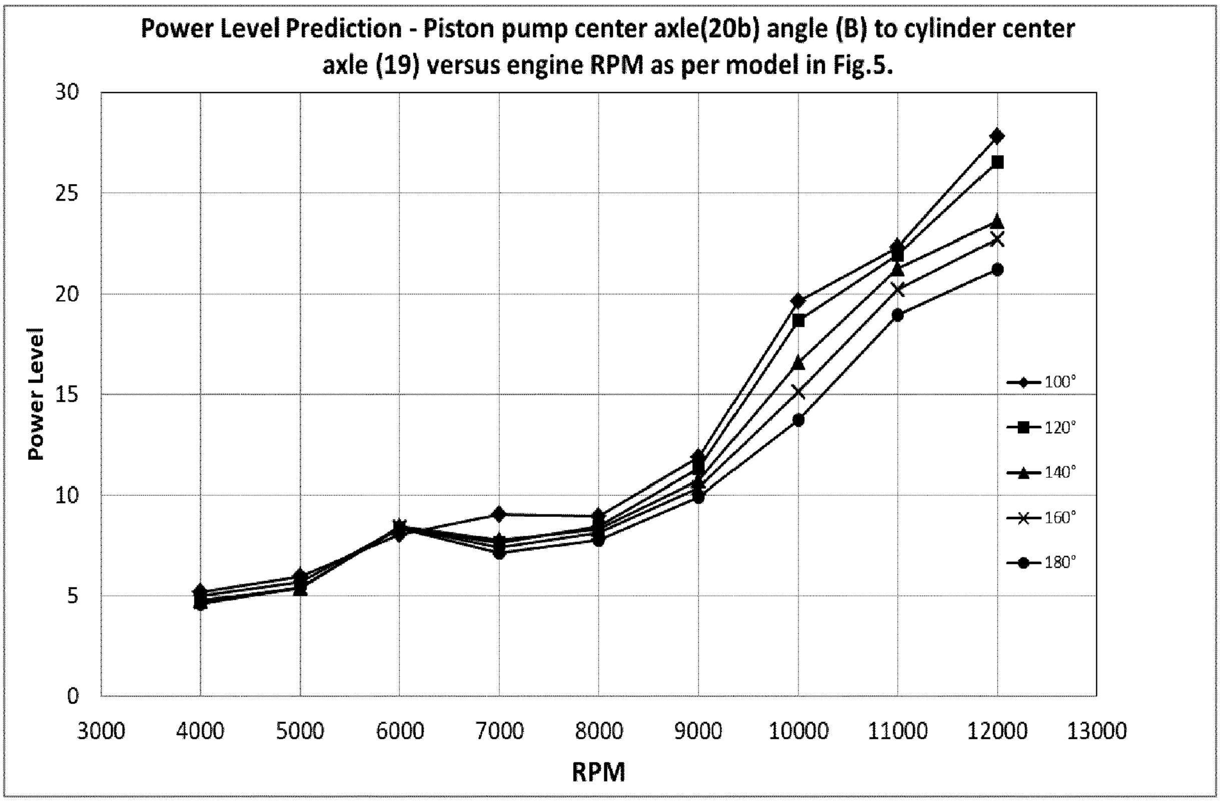
Power Level Prediction - Piston pump center axle(20b) angle (B) to cylinder center axle (19) versus engine RPM as per model in Fig.5.
Power Level
0
5
10
15
20
25
30
3000
4000
5000
6000
7000
8000
9000
10000
11000
12000
13000
RPM
100°
120°
140°
160°
180°

Figure 6.

Europäisches Patentamt
European Patent Office
Office européen des brevets

# EUROPEAN SEARCH REPORT

Application Number
EP 20 18 6473

## DOCUMENTS CONSIDERED TO BE RELEVANT

| Category | Citation of document with indication, where appropriate, of relevant passages | Relevant to claim | CLASSIFICATION OF THE APPLICATION (IPC) |
|---|---|---|---|
| X<br>Y | GB 101 531 A (CHÉDRU, GUSTAVE E) 7 December 1916 (1916-12-07)<br>* page 1, line 23 - page 2, line 6; figures 1-3 *<br>* page 2, line 51 - page 3, line 2 * | 1,2,5,8,9<br>4 | INV.<br>F02B33/20<br>ADD.<br>F01B9/02<br>F02B75/02 |
| X<br>Y | US 2019/170055 A1 (HOFFMAN RONALD J [US]) 6 June 2019 (2019-06-06)<br>* paragraph [0027]; figure 4 * | 1,2,5,8,9<br>4 | |
| Y | EP 1 039 113 A2 (K SEVEN KK [JP]) 27 September 2000 (2000-09-27)<br>* figures 15,17,18 * | 4 | |
| A | US 6 352 057 B1 (DRECQ DANIEL [FR]) 5 March 2002 (2002-03-05)<br>* column 7, line 53 - column 8, line 38; figures 5-7 * | 1-9 | |

TECHNICAL FIELDS SEARCHED (IPC)

F02B
F01B

The present search report has been drawn up for all claims

1

| Place of search | Date of completion of the search | Examiner |
|---|---|---|
| Munich | 29 October 2020 | Tietje, Kai |

CATEGORY OF CITED DOCUMENTS

X : particularly relevant if taken alone
Y : particularly relevant if combined with another document of the same category
A : technological background
O : non-written disclosure
P : intermediate document

T : theory or principle underlying the invention
E : earlier patent document, but published on, or after the filing date
D : document cited in the application
L : document cited for other reasons

& : member of the same patent family, corresponding document

EPO FORM 1503 03.82 (P04C01)

# ANNEX TO THE EUROPEAN SEARCH REPORT ON EUROPEAN PATENT APPLICATION NO. EP 20 18 6473

This annex lists the patent family members relating to the patent documents cited in the above-mentioned European search report. The members are as contained in the European Patent Office EDP file on
The European Patent Office is in no way liable for these particulars which are merely given for the purpose of information.

29-10-2020

| Patent document cited in search report | Publication date | Patent family member(s) | Publication date |
|---|---|---|---|
| GB 101531 A | 07-12-1916 | NONE | |
| US 2019170055 A1 | 06-06-2019 | CN 109973212 A | 05-07-2019 |
| | | EP 3495637 A1 | 12-06-2019 |
| | | US 2019170055 A1 | 06-06-2019 |
| EP 1039113 A2 | 27-09-2000 | AU 2245000 A | 28-09-2000 |
| | | EP 1039113 A2 | 27-09-2000 |
| | | JP 2001082157 A | 27-03-2001 |
| US 6352057 B1 | 05-03-2002 | AR 022211 A1 | 04-09-2002 |
| | | AT 292236 T | 15-04-2005 |
| | | BR 0007418 A | 16-10-2001 |
| | | CN 1377442 A | 30-10-2002 |
| | | EP 1018597 A1 | 12-07-2000 |
| | | FR 2788307 A1 | 13-07-2000 |
| | | JP 2003516490 A | 13-05-2003 |
| | | KR 20010089789 A | 08-10-2001 |
| | | US 6352057 B1 | 05-03-2002 |
| | | WO 0040845 A2 | 13-07-2000 |

EPO FORM P0459

For more details about this annex : see Official Journal of the European Patent Office, No. 12/82

## REFERENCES CITED IN THE DESCRIPTION



### Patent documents cited in the description


- US 20160010517 A1 **[0046]**
- EP 1018596 B1 **[0048]**