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(54) **INTERNAL COMBUSTION ENGINE**

VERBRENNUNGSMOTOR

MOTEUR À COMBUSTION INTERNE

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Description

TECHNICAL FIELD

[0001] The subject matter described herein, in general, relates to internal combustion (IC) engines.

BACKGROUND

[0002] Internal combustion (IC) engines find a variety of applications, for example, in automobiles, generators, marine applications, etc. The IC engines may be classified based on the number of strokes per working cycle of the IC engine. For example, IC engines are classified as two-stroke engines or four-stroke engines.

[0003] Conventionally in the two stroke engines, various phases of the working cycle, i.e., intake, compression, combustion and expansion, and exhaust, are accomplished in two strokes of a piston by performing the intake and exhaust strokes simultaneously during the end of the expansion, and the beginning of the compression strokes, respectively. To achieve the intake of charge during the intake stroke and to expel the combustion product during the exhaust stroke, the conventional two stroke engines are usually provided with ports in a cylinder wall. Due to layout of the ports in the cylinder wall and the simultaneous intake and exhaust strokes, a mixing of the charge and the combustion products may occur in the cylinder bore. Further, because of the overlapping intake and exhaust strokes, some amount of the charge may leak out from the cylinder bore during the exhaust stroke.

[0004] In contrast to the two-stroke engines, conventional four-stroke engines complete the phases of the working cycle in four strokes of the piston. The four-stroke engines have the ports provided in a cylinder head and have valves provided at the ports to regulate opening and closing of the ports. Usually, the ports and valves of the conventional four-stroke engines are associated with certain flow characteristics, which are prone to deficient flow. The flow characteristics of the ports and valves of the conventional four-stroke engines differ significantly from the flow characteristics of piston controlled ports of two-stroke engines.

[0005] JPH01200013A focuses on increasing the flowing quantity of new air being supplied to the combustion chamber and thus improving the output, by setting the internal diameter of each recessed groove formed at the inner wall surface of a cylinder head, at a particular size. The cylinder head has a plurality of inner wall surface portions which are respectively flat, and a pair of cylindrical recessed grooves are formed at one inner wall surface portion. Valve seating are respectively formed at the insides of cylindrical recessed grooves. Similarly air sucking valves are provided. The assembly is so set that the substantial supply of new air is started when each umbrella portion of each air sucking valve has passed out of each recessed groove.

[0006] DE1813906A1 discloses an exhaust valve arranged in the cylinder head of the two-stroke engine as a gas outlet in the cylinder chamber. The exhaust valve is provided by control cam having wide-ranging control timing. A valve lever is provided controlled by the cam. The fresh air inlet is arranged at the stroke end of the cylinder as usual.

[0007] US4907544A discloses a two-stroke cycle internal combustion engine without crankcase scavenging, which is able to operate in a four-stroke mode at cranking and idle speeds. The engine utilizes a turbocharger driven by exhaust gases to recharge the cylinder with fresh air or an air/fuel mixture. An auxiliary inlet valve is responsive to pressure within the cylinder and enables the engine to operate in a four-stroke mode, thus eliminating the need for an externally driven air pump when the engine is either idling, being started, or under light loads where turbocharger boost is too low to supply sufficient flow of air or an air/fuel mixture.

[0008] US1963780A discloses an internal-combustion-engine power unit including a four-cycle engine, an exhaust-gas driving turbine and an air-compressor driven thereby. A valve-controlled inlet opening at the upper end of the cylinder is adapted to admit part of the air during part of the suction stroke of the engine by the pumping-action of the engine-piston. Ports in the cylinder-wall is adapted to be uncovered by the piston near the end of the suction stroke for introducing the rest of the air only at the end of the suction stroke and the beginning of the compression stroke. Means for prematurely closing some of the air-admission members during the suction-action of the piston near the end of the working stroke is provided.

[0009] GB861080A discloses an internal combustion engine with direct fuel injection into a combustion chamber. The engine the form of a body of revolution arranged centrally in the working piston and which is so intersected by the top surface of the piston that no throttling action or only a slight throttling action is produced at the overflow opening. Means is provided at the air-admission valve for producing rotation of the air about the axis of the combustion chamber. The air-admission valve is countersunk or provided with a cylindrical portion above or below its seating surface and is controlled by a cam whose highest section extends over an angle of rotation of the cam of at least 30° so that the valve remains in its position of maximum opening over this angle.

SUMMARY

[0010] The subject matter described herein relates to an internal combustion engine. The internal combustion engine includes at least one port and a valve provided at the port to regulate opening and closing of the port. A valve seat is provided at the port for resting the valve. Further, a valve seat rim is provided at the valve seat for delaying the opening and advancing the closing of the port.

[0011] These and other features, aspects, and advantages of the present subject matter will be better understood with reference to the following description and appended claims. This summary is provided to introduce a selection of concepts in a simplified form. This summary is not intended to identify key features or essential features of the claimed subject matter, nor is it intended to be used to limit the scope of the claimed subject matter.

BRIEF DESCRIPTION OF DRAWINGS

[0012] The above and other features, aspects, and advantages of the subject matter will be better understood with regard to the following description, appended claims, and accompanying drawings where:

Fig. 1 illustrates a front sectional view of an IC engine, according to an embodiment of the present subject matter.

Fig. 2 illustrates a front sectional view of an exhaust port of the IC engine, according to an embodiment of the present subject matter.

Fig. 3 illustrates a front sectional view of an inlet port of the IC engine, according to an embodiment of the present subject matter.

Fig. 4 illustrates a valve timing diagram of the IC engine, according to an embodiment of the present subject matter.

DETAILED DESCRIPTION

[0013] Internal combustion (IC) engines usually include ports for intake of charge and for the expulsion of combustion products during a combustion cycle of the IC engine. Conventional two stroke IC engines, having ports in a cylinder wall for intake of charge and for expulsion of the products of combustion from a cylinder bore, are susceptible to thermal loading and distortions of the ports and of the cylinder wall. Such distortions may lead to high friction between a piston and the cylinder wall during movement of the piston in the cylinder bore. Further, these engines have the benefit of faster opening and closing of ports within a short duration resulting in effective gas exchange from the cylinder, i.e. the rate of opening and closing of the ports is high, which provides for the gas exchange from the cylinder.

[0014] In contrast, conventional four stroke IC engines have valves provided in the cylinder head for the induction of the charge, and for the expulsion of combustion products. The rate of opening and closing of ports is relatively gradual as compared to the opening and closing of the ports in two-stroke engine, for the same duration. Hence, it is difficult to ensure effective gas exchange in the available duration. Further, the usage of conventional valve train features and drive mechanisms of the typical four-stroke engine in the two-stroke engine to control the port area opening compromises the performance of the two-stroke engine due to constraints of valve train dy-

namics and the short available durations for gas exchange.

[0015] The embodiments of an internal combustion (IC) engine, interchangeably referred to as an engine hereinafter, disclosed herein will help address the aforementioned drawbacks in addition to providing several other advantages over the existing two stroke engines.

[0016] The IC engine includes one or more ports provided in the cylinder head. In one embodiment, the cylinder head includes at least one inlet port for intake of charge into the cylinder bore, and at least one exhaust port for discharge of combustion products from the cylinder bore.

[0017] Each of the ports is provided with a valve to control the opening and closing of the ports. Further, the ports are provided with a valve seat, corresponding to each valve. The valve lifts and rests at the valve seat during an open and a closed position, respectively, of the port.

[0018] According to an embodiment of the present subject matter, a valve seat rim is provided at at least one valve seat in the cylinder head. In said embodiment, the valve seat rim is provided at a circumference of the valve seat and is in the form of a cylindrical extension in the direction of the longitudinal axis of the valve.

[0019] The extension of the valve seat, as a result of the presence of the valve seat rim, delays the opening of the port and advances the closing of the ports by the valves. In case the valve seat rim is provided at an inlet valve seat, then the delayed opening and early closing of an inlet port provides a short duration for which the inlet port remains open. A shorter duration and rapid opening of the inlet port allows effective gas exchange from the combustion chamber in the available time.

[0020] In one implementation, the valve seat rim may be associated with the seat of the exhaust valve. In case of the rim being provided at an exhaust valve seat, a delayed opening of the exhaust port is achieved. The delay in the opening of the exhaust valve facilitates the engine in achieving a higher expansion ratio, and hence, higher fuel efficiency of the engine.

[0021] In addition, the provision of the valve seat rim at the valve seat facilitates in achieving a high rate of increase of effective flow area between the valve and the valve seat when the seat rim is uncovered by the valve. The rapid opening of the port area facilitates substantial increase in flow, for example, at the inlet valve, thereby overcoming the limitation of conventional valve train dynamics. The actual valve lift durations are high but the effective lift duration is made shorter by this delay feature. During the initial lift duration of the valve the port is not uncovered due to the valve rim and, hence, the actual flow of air does not occur. Further, the opening of the port takes place during the higher rate of valve lift, which helps efficient gas exchange. The same is true during the port closure.

[0022] **Fig. 1** illustrates a sectional view of an internal combustion engine **100**, according to an embodiment of

the present subject matter. In one example, the internal combustion engine **100** is a two stroke internal combustion engine. The internal combustion engine **100**, referred to as engine **100** hereinafter, includes a crankcase **102** connected to a charging device (not shown in figure). The crankcase **102** houses a crankshaft **104**. Further, a cylinder block **106** having a cylinder bore **108** is mounted on the crankcase **102**.

[0023] In one embodiment, the crankcase **102** includes an induction valve **110**, for example, a reed valve. According to an implementation, the induction valve **110** allows a unidirectional induction of a scavenging fluid, such as fresh air or a lean composition of charge formed by mixture of air and fuel, into the crankcase **102**. In another embodiment, the cylinder block **106** includes the induction valve **110** to induct the scavenging fluid into the crankcase **102**.

[0024] During operation of the engine **100**, the scavenging fluid entering the crankcase **102** is inducted into the cylinder bore **108**. At the time of the induction of the scavenging fluid into the cylinder bore **108** from the crankcase **102**, some lubricant particles may be entrained along with the scavenging fluid. In an embodiment, a filter element (not shown in the figure) is provided in the cylinder block **106** such that the scavenging fluid inducted from the crankcase **102** into the cylinder bore **108** passes through the filter element before entering the cylinder bore **108**. The filter element filters any lubricant particles in the scavenging fluid in the crankcase **102** and prevents the entry of lubricant particles into the cylinder bore **108** that may otherwise be burnt during combustion. Hence, the provision of the filter element facilitates low pollutants in exhaust emissions from the engine **100**.

[0025] In one embodiment, the filter element may be integrated with the induction valve **110**, and the scavenging fluid is inducted into the crankcase **102** through the induction valve **110** and the filter element. In such a case, the stream of the incoming scavenging fluid may clean the filter element by drawing the lubricant particles in the filter element back into the crankcase **102**. This may ensure longevity of the filter element.

[0026] It may be understood that in other embodiments, the filter element may be provided separately from the induction valve **110** and in different locations, such as the crankcase **102** or in a passage connecting the crankcase **102** to the cylinder bore **108**.

[0027] Further, the cylinder bore **108** has a piston **112** disposed therein, such that the piston **112** is capable of reciprocating inside the cylinder bore **108**. It may be understood that the engine **100** may have more than one cylinder bore **108**. Further, the piston **112** is connected to the crankshaft **104** through a connecting rod **114** to drive the crankshaft **104**. Inside the cylinder bore **108**, the piston **112** has two extreme positions - a top dead centre (TDC) position when the engine **100** has completed a compression stroke and a bottom dead centre (BDC) position from where the piston **112** commences actuation at the beginning of the compression stroke.

[0028] According to an embodiment of the present subject matter, the cylinder bore **108** has a plurality of transfer ports **116** provided annularly along a periphery of a cylinder wall **118**. The cylinder wall **118** of the cylinder block **106** defines the cylinder bore **108** therein. In said embodiment, the transfer ports **116** are disposed in the cylinder wall **118** in such a way that the transfer ports **116** are closer to the BDC position than they are to the TDC position of the piston **112**. Each of the plurality of transfer ports **116** is connected to the crankcase **102** through an induction passage **120**. Through the transfer ports **116**, the scavenging fluid is inducted into the cylinder bore **108** from the crankcase **102** during movement of the piston **112** from the TDC position to the BDC position.

[0029] According to an embodiment, each of the transfer ports **116** is formed in the shape of a truncated cone, for example, a right cone or an oblique cone, having an apex angle α (not shown in the figure) and having a base at an opening of the induction passage **120** into the respective transfer port **116**. The apex angle α may be defined as the included angle of the lateral surface of the cone, the angle being measured on a plane passing through an imaginary apex and the base of the truncated cone. In said embodiment, the apex angle is about 10 to 30 degrees to provide a swirling motion to the scavenging fluid entering the cylinder bore **108**. Further, in said embodiment, an axis of each of the transfer ports **116** is inclined to the cylinder wall **118** of the cylinder bore **108** at an angle, the angle being measured in a horizontal plane. In another embodiment, the angle between the axes of the transfer ports **116** and the wall of the cylinder bore **108** may also be measured in a vertical plane.

[0030] In said embodiment, each of the plurality of transfer ports **116**, formed as a truncated cone, includes a first aperture (not shown in the figure) and a second aperture (not shown in the figure). The first aperture is provided at the cylinder wall **118** and the second aperture is provided at an opening of the induction passage **120** into the respective transfer port **116**. According to an embodiment, the diameter of the first aperture is smaller than a diameter of the second aperture.

[0031] The inclination and geometry of the transfer ports **116** with respect to the wall of the cylinder bore **108** provides a swirling motion of the scavenging fluid during the induction of the scavenging fluid into the cylinder bore **108**. The swirling motion of the scavenging fluid in the cylinder bore **108** facilitates in scavenging and purging the cylinder bore **108** of combustion products during the movement of the piston **112** from the BDC position to the TDC position. Further, opening and closing of the transfer ports **116** is controlled by the movement of the piston **112** in the cylinder bore **108**.

[0032] Further, a plurality of piston rings **122** are provided on the piston **112** for providing a seal between the piston **112** and the cylinder bore **108**. The piston rings **122** also facilitate scraping carbonaceous deposits off an inner wall of the cylinder bore **108**.

[0033] Further, a cylinder head assembly **124** is mount-

ed on the cylinder block **106** to close one end of the cylinder bore **108**. The cylinder head assembly **124** includes a cylinder head **126** and a cylinder head cover **128**. A main combustion chamber **130** is formed between the cylinder head **126**, the cylinder wall **118**, and a crown of the piston **112** when the piston **112** is at the TDC position inside the cylinder bore **108** during a combustion cycle. In an embodiment, the cylinder head **126** includes an inlet port **132**. In other embodiments, the cylinder head **126** may have more than one inlet port **132**.

[0034] Further, the cylinder head **126** has an exhaust port **134** for expelling combustion products from the main combustion chamber **130**. In other embodiments, the cylinder head **126** may have more than one exhaust port **134**. In an embodiment, the inlet port **132** is smaller in diameter than the exhaust port **134**. In said embodiment, a diameter of the inlet port **132** is about 10 millimeter (mm) and a diameter of the exhaust port **134** is about 27.5 millimeter (mm). The large size of the exhaust port **134** facilitates effective exhaust of the combustion products from the exhaust port **134** as well as scavenging of the combustion products from the cylinder bore **108**. The expulsion of the combustion products from the cylinder bore **108** is aided by the scavenging fluid entering the cylinder bore **108** through the transfer ports **116**. The provision of the exhaust port **134** in the cylinder head **126** of the engine **100** allows uniflow scavenging of the cylinder bore **108** by the scavenging fluid. Such a scavenging facilitates a thorough purging of the cylinder bore **108** and enhances fuel economy and efficiency of the engine **100**.

[0035] According to an embodiment, an auxiliary combustion chamber **136** is formed in the cylinder head **126** of the engine **100**. The auxiliary combustion chamber **136** is provided with an ignition element **138** to achieve combustion of the charge in the auxiliary combustion chamber **136**. In other embodiments, more than one ignition element **138** may be provided in the auxiliary combustion chamber **136** and one or more ignition elements **138** may be provided in the main combustion chamber **130**, to facilitate the combustion of the charge.

[0036] In one implementation, a partial combustion, referred to as first combustion, of charge takes place in the auxiliary combustion chamber **136**. The auxiliary combustion chamber **136** opens into the main combustion chamber **130** such that during the expansion stroke, a flame front produced by the combustion of charge in the auxiliary combustion chamber **136** spreads into the main combustion chamber **130** and the combustion of charge is substantially completed in the main combustion chamber **130**. The substantially complete combustion of charge in the main combustion chamber **130** is interchangeably referred to as second combustion of charge. In an embodiment, the auxiliary combustion chamber **136** is about 70% of the total combustion chamber volume and the main combustion chamber **130** is about 30% of the total combustion chamber volume.

[0037] During the movement of the piston **112** from the

TDC to the BDC, at about the end of the expansion stroke, the transfer ports **116** are uncovered and the swirling scavenging fluid scavenges the combustion products. Further, during the movement of the piston **112** from the BDC to the TDC, the scavenging fluid is compressed in the cylinder bore **108**.

[0038] In addition, the auxiliary combustion chamber **136** is in fluid communication with a fuel supply pump **140**. In one embodiment, the fuel supply pump **140** is a reciprocating type pump and includes an auxiliary piston **142** reciprocating inside an auxiliary bore **144**. In an embodiment, the auxiliary piston **142** has a small skirt length to reduce contact surface of the auxiliary piston **142**, and hence friction between the auxiliary piston **142** and a wall of the auxiliary bore **144**. Such a design of the auxiliary piston **142** ensures durability of the fuel supply pump **140**. The auxiliary bore **144** is formed in an auxiliary cylinder block **146** of the fuel supply pump **140**. In an embodiment, the auxiliary cylinder block **146** is formed in the cylinder block **106** of the engine **100**, and hence the fuel supply pump **140** is integrated in the cylinder block **106**. In another embodiment, the fuel supply pump **140** is integrated with the crankcase **102** of the engine **100**. In yet another embodiment, the fuel supply pump **140** is integrated with the cylinder head **126** of the engine **100**.

[0039] The fuel supply pump **140** further includes an auxiliary crankshaft **148** housed in an auxiliary crankcase **150**. The auxiliary piston **142** is operably coupled to the auxiliary crankshaft **148** through an auxiliary connecting rod **152** such that a rotational motion of the auxiliary crankshaft **148** is transformed into a reciprocatory motion of the auxiliary piston **142**.

[0040] In an implementation, the auxiliary crankshaft **148** of the fuel supply pump **140** obtains a drive from the crankshaft **104** of the engine **100**. In one embodiment, the auxiliary crankshaft **148** obtains the drive from the crankshaft **104** through a chain drive assembly (not shown in the figure). Similarly, other drive mechanisms may also be used to provide a drive from the crankshaft **104** to the auxiliary crankshaft **148**.

[0041] In case of a chain drive assembly providing the drive to the auxiliary crankshaft **148**, an end of the auxiliary crankshaft **148** has an auxiliary sprocket wheel (not shown in the figure) mounted and fixed thereon. In said embodiment, a crankshaft sprocket wheel (not shown in the figure) is mounted on an end of the crankshaft **104** to provide a drive to the auxiliary sprocket wheel of the fuel supply pump **140**. A phase difference may be provided between the drive of the crankshaft **104** and the drive of the fuel supply pump **140** so that the induction of the charge into the auxiliary combustion chamber **136** is achieved substantially before the piston **112** reaches the TDC position in the cylinder bore **108**.

[0042] Further, the fuel supply pump **140** includes an auxiliary cylinder head (not shown in the figure) mounted on the auxiliary cylinder block **146**. The auxiliary cylinder head includes an auxiliary inlet port **154** and an auxiliary exhaust port **156**. In an embodiment, the auxiliary inlet

port **154** is connected to a fuelling device (not shown in the figure), such as a carburettor, and receives a charge of fuel and air from the fuelling device through a charging valve **158**. In one example, the charging valve **158** is a reed valve, which allows a unidirectional flow of the charge from the fuelling device into the auxiliary bore **144** of the fuel supply pump **140** through the auxiliary inlet port **154**. In one implementation, a small quantity of the charge is received by the fuel supply pump **140** from the fuelling device. In said implementation, the charge is richer in composition as compared to the corresponding stoichiometric composition of the charge for an operation, for example, full load operation or part load operation, of the engine **100**.

[0043] In another embodiment, the auxiliary bore **144** of the fuel supply pump **140** includes a fuel inlet port (not shown in the figure) connected to the fuelling device for the induction of charge. In said embodiment, the fuel inlet port is formed in the auxiliary cylinder block **146** slightly above a BDC position of the auxiliary piston **142** in the auxiliary bore **144**. The charge is inducted into the fuel supply pump **140** when the auxiliary piston **142** is approaching the BDC position in the auxiliary bore **144**. Hence, the reciprocating motion of the auxiliary piston **142** regulates the opening and closing of the fuel inlet port for the induction of charge.

[0044] The charge entering the auxiliary bore **144** is pressurized by the reciprocating motion of the auxiliary piston **142**. The pressurization of the charge facilitates substantial atomization and disintegration of the charge before it is supplied into the auxiliary combustion chamber **136** for combustion. Such pressurization assists in complete burning of the charge and hence improves the efficiency of the engine **100**. Further, in one embodiment, the fuel supply pump **140** is a low-pressure pump and has a compression ratio of about 3:1. The compression ratio may be understood as ratio of volume of auxiliary bore **144** when the auxiliary piston **142** is at the BDC position to the volume of the auxiliary bore **144** when the auxiliary piston **142** is at the TDC position. In an embodiment, a volumetric capacity of the fuel supply pump **140** is about 10% of a volumetric capacity of the engine **100**. In said embodiment, the capacity of the fuel supply pump **140** is in a range of about 10 cubic centimeters (cc) to 15 cc.

[0045] Further, the auxiliary exhaust port **156** is connected to an inlet passage **160** leading to the inlet port **132** of the engine **100**. The fuel supply pump **140** inducts a pressurized charge of air and fuel for combustion into the auxiliary combustion chamber **136** through the inlet passage **160** and the inlet port **132**. In one embodiment, the inlet passage **160** is formed in the cylinder head **126** in substantial proximity of the main combustion chamber **130** and the auxiliary combustion chamber **136**. In said embodiment, a substantially complete vaporization of the fuel in the pressurized charge is achieved in the inlet passage **160** because of the proximity to the combustion chambers **130** and **136**.

[0046] In one embodiment, the auxiliary exhaust port **156** includes a discharge valve (not shown in the figure) to regulate the induction of pressurized charge from the fuel supply pump **140** into the inlet passage **160**. In an implementation, the discharge valve is a pressure-regulated check valve and is calibrated in such a way that it allows the pressurized charge to enter the inlet passage **160** when the pressure of the charge is above a predetermined value. During the induction of charge from the fuelling device into the fuel supply pump **140** and during the pressurization of charge, the discharge valve remains closed.

[0047] In operation, the fuel supply pump **140** inducts the charge into the auxiliary combustion chamber **136** when the piston **112** is compressing the air inducted in the main combustion chamber **130** through the transfer ports **116**. During this phase, the piston **112** is said to be in a compression stroke and is approaching the TDC position in the cylinder bore **108**. Accordingly, during this phase, the main combustion chamber **130** has the scavenging fluid, such as fresh air or a lean composition of charge, while the auxiliary combustion chamber **136** has a small quantity of relatively rich composition of the charge as compared to that in the main combustion chamber **130** creating layers or strata of the charge with different compositions of air and fuel.

[0048] Due to this stratification of charge, a plurality of strata of charge is formed in the main combustion chamber **130** and the auxiliary combustion chamber **136**. The different strata of charge differ in the composition of the air and fuel. This allows the engine **100** to operate on an overall lean composition of charge. Hence, the overall fuel consumption of the engine **100** is low.

[0049] It will be understood that the engine **100** may include only the main combustion chamber **130** for the combustion of the charge. In such a case, the fuel supply pump **140** inducts charge into the main combustion chamber **130**.

[0050] In addition, the cylinder head assembly **124** includes a valve train assembly **162**. In one embodiment, the valve train assembly **162** is provided in the cylinder head **126** and housed inside the cylinder head assembly **124**. In one implementation, the valve train assembly **162** includes an inlet valve **164** and an exhaust valve **166**. In said embodiment, the inlet valve **164** is provided at the inlet port **132** in the cylinder head **126**, and the exhaust valve **166** is provided at the exhaust port **134**.

[0051] An inlet valve seat and an exhaust valve seat (both not shown in the figure) are provided at the inlet port **132** and the exhaust port **134**, respectively. The valves **164** and **166** rest at their respective valve seats during a closed position of the respective ports **132** and **134**. By lifting and seating at their respective valve seats, the inlet valve **164** and the exhaust valve **166** regulate the opening and closing of their respective ports, i.e., the inlet port **132** and the exhaust port **134**, at which they are provided.

[0052] According to an embodiment of the present sub-

ject matter, a valve seat rim (not shown in the figure) is provided at least at one of the exhaust valve seat or the inlet valve seat in the cylinder head **126**. In said embodiment, the valve seat rim is cylindrical, provided at a circumference of the valve seat and extends in a direction of a longitudinal axis of the valve **164** or **166** towards the cylinder bore **108**. In another embodiment, the valve seat rim is provided at the inlet valve seat as well as the exhaust valve seat.

[0053] Further, in one embodiment, the valve seat rim is formed integrally with the cylinder head **126**. In said embodiment, the valve seat rim is formed as a bore in the cylinder head **126**, extending to the valve seat. In another embodiment, the valve seat rim may be formed separately from the cylinder head **126** and disposed at the valve seat.

[0054] The provision of the valve seat rim at the valve seat facilitates in achieving a high rate of increase of an effective flow area between the valve and the valve seat, thereby achieving a substantially rapid lifting and seating of the valve. Further, the extension of the valve seat, as a result of the presence of the valve seat rim, delays the opening of the port and advances the closing of the ports by the valves. The valve seat rim is described in further detail in **Fig. 2** and **3**.

[0055] Further, the valve train assembly **162** includes a camshaft **168** to actuate the inlet valve **164** and the exhaust valve **166**. The inlet valve **164** is operatively coupled to the camshaft **168** via a first push rod **170** and a first rocker arm **172**. Similarly, the exhaust valve **166** is operatively coupled to the camshaft **168** via a second push rod **174** and a second rocker arm **176** (partially shown in **Fig. 1**). Further, the first rocker arm **172** and the second rocker arm **176** include a first follower **178** and a second follower (not shown in figure), respectively. In one embodiment, the first follower **178** and the second follower are roller type followers.

[0056] In said embodiment, the inlet valve **164** and the exhaust valve **166** are spring loaded valves. In said embodiment, the camshaft **168** includes a plurality of cam lobes (not shown in the figure), corresponding to each of the inlet valve **164** and the exhaust valve **166**, to actuate the valves **164** and **166**. To drive the inlet valve **164** and the exhaust valve **166**, the camshaft **168** may obtain a drive from the engine **100**, for example, through a chain drive assembly. In such a case, a camshaft sprocket wheel (not shown in the figure) mounted on the camshaft **168** receives a drive from the crankshaft **104** through the chain drive assembly.

[0057] In one embodiment, the chain drive assembly is a step-less chain drive assembly, that is, the crankshaft **104** provides a drive to the camshaft **168** for actuating the inlet valve **164** and the exhaust valve **166** and also provides a drive to the auxiliary crankshaft **148** for driving the fuel supply pump **140**, without any reduction in the drive ratio. This facilitates synchronization of the rotation of the crankshaft **104**, the opening and closing of the inlet valve **164** and the exhaust valve **166**, as well as the ro-

tation of the auxiliary crankshaft **148** of the fuel supply pump **140**, for effective functioning of the engine **100**.

[0058] The chain drive assembly may include a plurality of chain tensioners (not shown in the figures). The chain tensioners maintain proper tension in the chain to reduce vibrations and noise generated by the chain during operation. The chain tensioners achieve a compensation for the wear that the chain undergoes and, hence, facilitates avoiding errors, such as tooth-skip, during the transmission of drive through the chain. The chain tensioners also help in reducing synchronization errors between the crankshaft **104**, the camshaft **168**, and the auxiliary crankshaft **104**, which may occur due to loss in tension of the chain.

[0059] In another embodiment, the inlet valve **164** and the exhaust valve **166** may be electronically actuated valves.

[0060] **Fig. 2** illustrates a magnified sectional view of the exhaust port **134** of the engine **100**, according to an embodiment of the present subject matter. As shown in **Fig. 2**, an exhaust valve seat **202** is provided at the exhaust port **134**. The lifting and the seating of the exhaust valve **166** at the exhaust valve seat **202** regulate the opening and the closing of the exhaust port **134**. According to an embodiment, an exhaust valve seat rim **204** is formed at a circumference of the exhaust valve seat **202**, extending along a longitudinal axis **206** of the exhaust valve seat **202** in the direction of the cylinder bore **108**. In said embodiment, the exhaust valve seat rim **204** is cylindrical in shape. As mentioned earlier, the exhaust valve seat rim **204** may be formed integrally with the exhaust valve seat **202** in the cylinder head **126**, or may be formed separately and disposed at the exhaust valve seat **202**. In one embodiment, the exhaust valve seat rim **204** has a length of about 5% to about 15% of a valve lift of the exhaust valve **166**. The valve lift of a valve can be understood as the maximum distance, measured from the respective valve seat, traversed by the valve when lifted off the valve seat during operation. In an example, the length of the exhaust valve rim **204** is about 1 millimeter (mm) measured from the exhaust valve seat **202**.

[0061] The operation of the exhaust valve **166** may be understood with respect to **Figs. 1** and **2**. The actuation of the second follower of the second rocker arm **176** by the corresponding lobe of the camshaft **168** causes the second push rod **174** to actuate the exhaust valve **166**. On such an actuation, the exhaust valve **166** is lifted off its seat in the cylinder head **126** to open the exhaust port **134**. The opening of the exhaust port **134** allows the expulsion of the combustion products from the cylinder bore **108**.

[0062] With the provision of the exhaust port **134** and the exhaust valve **166** in the cylinder head **126**, the piston **112** can be of a relatively small skirt length. Since, the piston **112** of the engine **100** has a small skirt length and has less area of contact with the cylinder wall **118**, thereby reducing friction between the piston **112** and the cylinder wall **118** and ensuring longevity and durability of

the engine **100**. Further, the piston **112** with small skirt length has a small weight and therefore less inertia. Hence, the engine **100** expends less thrust of combustion in overcoming the inertia of the piston **112**. In addition, with the provision of the exhaust port **134** in the cylinder head **126**, the thermal loading and distortions due to the combustion products occurs in the cylinder head **126** and not at the cylinder wall **118**, hence, ensuring durability of the engine **100**.

[0063] Moreover, due to the presence of the exhaust valve seat rim **204**, a delay in the opening and an advance in the closing of the exhaust port **134** is achieved, which provides a short duration of opening of the exhaust port **134**. The short duration of the opening of the exhaust port **134** facilitates expulsion of the combustion products from the cylinder bore **108** of the engine **100**, such as in the case of two stroke engines where the exhaust stroke is completed in a short duration. Further, the delay in the opening of exhaust port **134** helps the engine **100** in achieving a high expansion ratio, thereby allowing the engine **100** to extract large amount of energy from the combustion of the charge. The engine **100**, hence, achieves high fuel efficiency.

[0064] The high expansion ratio also results in low pressure inside the cylinder bore **108** at the beginning of the exhaust stroke of the piston **112**. As a result, a pressure gradient between the cylinder bore **108** and an exhaust manifold (not shown) is also low, facilitating less noise during expulsion of the combustion products from the cylinder bore **108** through the exhaust port **134**.

[0065] Fig. 3 illustrates a magnified sectional view of the inlet port **132** of the engine **100**, according to an embodiment of the present subject matter. As mentioned earlier, the inlet port **132** is formed in the cylinder head **126** and the opening and closing of the inlet port **132** is regulated by the inlet valve **164**. The lifting and seating of the inlet valve **164** at an inlet valve seat **302** achieves the opening and closing, respectively, of the inlet port **132**.

[0066] According to an embodiment, an inlet valve seat rim **304** is provided at the inlet valve seat **302**. In said embodiment, the inlet valves seat rim **304** is formed at the inlet valve seat **302** in a similar manner as the exhaust valve seat rim **204** is formed at the exhaust valve seat **202**. In an embodiment, a length of the inlet valve seat rim **304** measured from the inlet valve seat **302** may be about 25% to about 40% of a valve lift of the inlet valve **164**. In an example, the length of the inlet valve seat rim **304** measured from the inlet valve seat **302** is about 1.5 millimeter (mm).

[0067] The operation of the inlet valve **164** is explained, as follows, in relation with Fig. 1 and Fig. 3. The first follower **178** is operably coupled to the corresponding lobe of the camshaft **168** for actuating the first rocker arm **172** and the first push rod **170**. The actuation causes the first push rod **170** to push the inlet valve **164**. As the inlet valve **164** is pushed, it is lifted off the inlet valve seat **302** in the cylinder head **126** to open the inlet port **132**. As

the inlet port **132** is opened, the pressurized charge from the fuel supply pump **140** enters the auxiliary combustion chamber **136** through the inlet passage **160** and the inlet port **132**.

[0068] The provision of the inlet valve seat rim **304** delays the opening and advances the closing of the of the inlet port **132**, which provides a short duration of the opening of the inlet port **132**. Such a short duration of opening of the inlet port **132** allows an appropriate amount of charge to be inducted into the auxiliary combustion chamber **136**.

[0069] Further, the inlet valve seat rim **304** facilitates in achieving a high rate of increase of effective flow area between the inlet valve **164** and the inlet valve seat **302**, when the inlet valve **164** uncovers the inlet valve seat rim **304**.

[0070] Fig. 4 illustrates a valve timing diagram **400** of the engine **100**, according to an embodiment of the present subject matter. The description of the valve timing diagram **400** is provided in relation to Fig. 1, 2, and 3 for a two stroke internal combustion engine. The positions of the various components of the engine **100** and the events are described in terms of degrees of rotation of the crankshaft **104**. As the piston **112** reciprocates between the TDC position and the BDC position in the cylinder bore **108**, the crankshaft **104** rotates. Hence, the displacement of the piston **112**, at any position between the TCD and the BDC in the cylinder bore **108**, is expressed in terms of an angular rotation of the crankshaft **104** in degrees. For the purpose of description of the valve timing diagram of the engine **100**, the TDC and the BDC have been taken as reference positions. The TDC position and the BDC positions are depicted as positions **A** and **D**, respectively, on the valve timing diagram **400**.

[0071] At about 90 to 110 degrees after the TDC position of the piston **112**, the exhaust valve **166** starts to lift from the exhaust valve seat **202** to open the exhaust port **134**. However, due to the exhaust valve seat rim **204**, the actual lifting of the exhaust valve **166** and opening of the exhaust port **134** is achieved at about 110 to 150 degrees after TDC position, depicted as position **B** in the valve timing diagram **400**. With the opening of the exhaust port **134**, the combustion products, which were formed during the immediately preceding phase of the working cycle of the engine **100**, are discharged from the cylinder bore **108** through the exhaust port **134**. The discharge of the combustion products is aided by a pressure difference between a pressure inside the cylinder bore **108** and a pressure in the exhaust manifold. The opening of the exhaust port **134** marks the beginning of the exhaust stroke.

[0072] At about 10 to 15 degrees after the opening of the exhaust port **134**, i.e., at about 130 to 150 degrees rotation of the crankshaft **104** after the TDC position, depicted as position **C** in the valve timing diagram, the piston **112** crosses and uncovers the transfer ports **116**. In another embodiment, the transfer ports **116** are uncovered at about 135 to 145 degrees of rotation of the crankshaft

104 from the TDC position. The opening of the transfer ports **116** marks the beginning of the intake stroke. With the opening of the transfer ports **116** the scavenging fluid, such as air or a lean charge of air and fuel, enters into the cylinder bore **108**. The scavenging fluid is transferred from the crankcase **102** to the cylinder bore **108** through the induction passages **120** and the transfer ports **116** as a result of the movement of the piston **112** towards the BDC position and the resulting compression of the charge in the crankcase **102**.

[0073] Further, due to the inclination of the transfer ports **116** to the cylinder wall **118** and as a result of the geometry of the transfer ports **116**, the scavenging fluid entering the cylinder bore **108** is in a swirling motion. The scavenging fluid entering the cylinder bore **108** facilitates the scavenging of the combustion products through the exhaust port **134** in the cylinder head **126**, thereby achieving uniflow scavenging and thorough purging of the cylinder bore **108**.

[0074] At the BDC position (position **D** in **Fig. 4**), i.e., at a 180 degrees rotation of the crankshaft **104** after the TDC position, the exhaust port **134** and the transfer ports **116** are both open. As a result, the intake or induction of the scavenging fluid occurs simultaneously along with the expulsion of the combustion products from the cylinder bore **108**. The piston **112** crosses the BDC position and moves towards the TDC position due to inertia, and as a result the rotation of the crankshaft **104** continues beyond 180 degrees.

[0075] At about 200 to 210 degrees after the TDC position, i.e., about 20 to 30 degrees after the BDC position, the exhaust port **134** is fully open and the transfer ports **116** are open as well. This condition is depicted as position **E** in the valve timing diagram **400**. As the exhaust port **134** is fully open, the removal of the combustion products takes place at a high rate.

[0076] Further, at position **F** in **FIG. 4**, during the motion of the piston **112** from the BDC position to the TDC position, the piston **112** covers the annular transfer ports **116** at about 210 to 230 degrees of rotation of the crankshaft **104** after the TDC position. Hence, the transfer ports **116** are closed at about 30 to 50 degrees of rotation of the crankshaft **104** after the BDC position. With the closure of the transfer ports **116**, the intake stroke of the engine **100** is concluded. However, due to the open exhaust port **134**, the exhaust stroke continues.

[0077] Further, at about 240 to 280 degrees of rotation of the crankshaft **104** after the TDC position, i.e., at about 60 to 100 degrees of rotation after the BDC position, the exhaust port **134** closes. In another embodiment, the exhaust port **134** closes at about 260 to 280 degrees of rotation after the TDC position. According to an aspect, the exhaust stroke takes place for about 120 to 160 degrees of the rotation of the crankshaft **104**. Shown as position **G** in the valve timing diagram **400**, the closure of the exhaust port **134** marks an end of the exhaust stroke and a commencement of the compression stroke.

[0078] During the compression stroke, the movement

of the piston **112** towards the TDC position compresses the still swirling scavenging fluid in the main combustion chamber **130** being formed between the piston **112**, the cylinder head **126**, and the cylinder wall **118**. The compression of the swirling scavenging fluid causes turbulence in the main combustion chamber **130**. Such turbulence facilitates mixing and combustion of the charge as will be explained later with reference to the combustion phase. During the compression stroke, an increase in a temperature and pressure of the scavenging fluid occurs. As can be seen from the valve timing diagram **400**, the compression stroke takes place for about 60 to 110 degrees of the rotation of the crankshaft **104**.

[0079] At about 240 to 265 degrees of rotation of the crankshaft **104** after the TDC position, i.e., at about 60 to 85 degrees of rotation after the BDC position, the inlet port **132** starts to open. At this instance, the inlet valve **164** starts to lift from the inlet valve seat **302**. The provision of the inlet valve seat rim **304** at the inlet valve seat **302**, however, delays the opening of the inlet port **132**. The actual opening of the inlet port **132** occurs at about 260 to 290 degrees of rotation of the crankshaft **104** after the TDC position, i.e., at about 80 to 105 degrees of rotation after the BDC. This instance is depicted as position **H** in the valve timing diagram **400**.

[0080] With the opening of the inlet valve **132**, the charge from the fuel supply pump **140** starts to enter the auxiliary combustion chamber **136**. The charge entering the auxiliary combustion chamber **136** may include a small quantity of a rich composition of charge. Further, the turbulence created in the main combustion chamber **130** by the compressed scavenging fluid may spread into the auxiliary combustion chamber **136**. The turbulence helps in thorough mixing of the charge in the auxiliary combustion chamber **136** and facilitates good combustibility of the fuel in the charge.

[0081] Further, as a result of a small quantity of rich composition of the charge in the auxiliary combustion chamber **136** and the scavenging fluid, such as air or a lean composition of charge, present in the main combustion chamber **130** facilitates in formation of strata of different compositions of air and fuel in the charge in the main combustion chamber **126** and the auxiliary combustion chamber **136**, thereby achieving stratification of the charge. The stratification of the charge allows the engine **100** to operate on an overall lean composition of charge. Hence, the overall fuel consumption of the engine **100** is low.

[0082] In another embodiment, on opening of the inlet port **132**, the charge enters directly into the main combustion chamber **130** and the combustion of charge is achieved in the main combustion chamber **130**.

[0083] At about 300 to 305 degrees of rotation of the crankshaft **104** after the TDC position, i.e., at about 120 to 125 degrees of crankshaft rotation after the BDC position, the inlet port **132** is fully open and the induction of the charge takes place at a fast rate. This is shown as position **I** in the valve timing diagram **400**. According to

an embodiment, the intake of charge takes place for about 60 to 90 degrees of rotation of the crankshaft **104**.

[0084] Further, at about 280 to 330 degrees of rotation after the TDC position, i.e., at about 140 to 145 degrees of rotation after the BDC position, the inlet port **132** closes. In an embodiment, the inlet port **132** closes at about 320 to 325 degrees of rotation after the TDC position. Shown as position **J** in the valve timing diagram **400**, the closure of the inlet port **132** concludes the injection of the charge into the auxiliary combustion chamber **136**. According to an embodiment, the injection of charge into the auxiliary combustion chamber **136** takes place for about 20 to 45 degrees of rotation of the crankshaft **104**.

[0085] Further, an end of the compression stroke takes place at about 325 to 360 degrees of rotation and is depicted by position **K** in the valve timing diagram **400**.

[0086] Subsequently, the ignition element **138**, such as a spark plug, provided in the auxiliary combustion chamber **136** fires at about 325 to 360 degrees of rotation after the TDC position, i.e. at about 10 to 30 degrees of rotation before the TDC position. In another embodiment, the ignition element **138** fires at 340 to 350 degrees of rotation of the crankshaft after the TDC position. The ignition element **138** ignites the charge in the auxiliary combustion chamber **136**. In the auxiliary combustion chamber **136**, a partial combustion of the charge occurs and as a result of the compact and small size of the auxiliary combustion chamber **136**, a flame front of the burning charge travels quickly into the main combustion chamber **130** where the combustion of the charge is substantially completed.

[0087] The combustion of the charge in the auxiliary combustion chamber **136** and the main combustion chamber **130**, shown as position **L** in the valve timing diagram **400**, marks the beginning of the expansion stroke. The combustion of the charge produces expanding combustion products that propel the piston **112** towards the BDC position.

[0088] In another embodiment, the expansion stroke begins at about 10 degrees of rotation of the crankshaft after the TDC position.

[0089] As mentioned earlier, at position **B** shown in the valve timing diagram **400**, the exhaust port **134** opens and marks the end of the expansion stroke. In one embodiment, the expansion stroke ends at about 120 to 140 degrees of rotation of the crankshaft from the TDC position. The expansion stroke, as is clear from the valve timing diagram **400**, takes place for about 110 to 150 degrees of the rotation of the crankshaft **104**.

[0090] The previously described versions of the subject matter and its equivalent thereof have many advantages, including those which are described herein.

[0091] The provision of the exhaust valve seat rim **204** at the exhaust valve **202** provides a delay in the opening of the exhaust port **134** on the completion of the expansion stroke. As a result, the expansion stroke is of a longer duration as compared to the compression stroke, which leads to a high expansion ratio of the engine **100**. The

high expansion ratio facilitates the engine **100** in achieving high fuel efficiency and low pressure of the combustion products in the cylinder bore **108** at the beginning of the exhaust stroke. The low pressure of the combustion products helps in substantially noise-less expulsion of the combustion products from the cylinder bore **108** during the exhaust stroke.

[0092] Further, the provision of the inlet valve seat rim **304** results in high rate of change of the effective flow area when the inlet valve **164** uncovers the inlet valve seat rim **304** and, hence the flow deficiency associated with the conventional inlet valves is substantially prevented. Further, the provision of the inlet valve seat rim **304** provides for a short duration of effective opening of the inlet port **132**.

Claims

1. An internal combustion engine (100) comprising:

a cylinder block (106) having at least one cylinder bore (108) defined by a cylinder wall (118), the cylinder block (106) comprising,
a piston (112) reciprocating in said cylinder bore (108) to drive a crankshaft (104);
a plurality of transfer ports (116) provided annularly in the cylinder wall (118) along a periphery of the cylinder wall (118); and
a cylinder head (126) mounted on the cylinder block (106), the cylinder head (126) comprising:

at least one inlet port (132);
at least one exhaust port (134);
an inlet valve (164) provided at the inlet port (132) to regulate an opening and closing of the inlet port (132);
an exhaust valve (166) provided at the outlet port (134) to regulate an opening and closing of the exhaust port (134);
an inlet valve seat (302) provided at the inlet port (132) for resting the inlet valve (164);
an exhaust valve seat (202) provided at the exhaust port (134) for resting the exhaust valve (166);

characterized by an inlet valve seat rim (304) provided at the inlet valve seat (302) to allow delay opening and to advance closing of the inlet port (132); and
an exhaust valve seat rim (204) provided at the exhaust valve seat (202) to allow delay opening and advance closing of the exhaust port (134), and wherein

a maximum valve lift of the inlet valve (164) from the inlet valve seat occurs during the movement of the piston from a bottom dead centre position to a top dead centre position at about 120 to 125

- degrees of rotation of the crankshaft (104) after the bottom dead centre position, wherein a maximum valve lift of the exhaust valve (166) from the exhaust valve seat (202) occurs during the movement of the piston (112) from the bottom dead centre position to the top dead centre position at about 20 to 30 degrees of rotation of the crankshaft (104) after the bottom dead centre position, and wherein the piston (112) uncovers the plurality of transfer ports (116) to induct the scavenging fluid at about 130 to 150 degrees of rotation of the crankshaft (104) after the top dead centre position and covers the plurality of transfer ports (116) at about 210 to 230 degrees of rotation of the crankshaft (104) after the top dead centre position, wherein the inlet valve (164) remains closed while the plurality of transfer ports (116) is open.
2. The internal combustion engine (100) as claimed in claim 1, wherein the inlet valve seat rim (304) and exhaust valve seat rim (204) are provided at a circumference of the inlet valve seat (302) and the exhaust valve seat (202) respectively and wherein the inlet valve seat rim (304) and exhaust valve seat rim (204) are cylindrical, extending along a longitudinal axis of the inlet valve (164) and exhaust valve (166) respectively.
 3. The internal combustion engine (100) as claimed in claim 1, wherein the inlet valve seat rim (304) and exhaust valve seat rim (204) extend to a length of about 5% to about 40% of the inlet valve lift and exhaust valve lift respectively.
 4. The internal combustion engine (100) as claimed in claim 1, wherein the inlet valve seat rim (304) and exhaust valve seat rim (204) extend to a length of about 1 millimeter to 1.5 millimeter measured from the inlet valve seat (302) and exhaust valve seat (202) respectively.
 5. The internal combustion engine (100) as claimed in claim 1, wherein the inlet valve seat rim (304) and exhaust valve seat rim (204) are provided at the inlet valve seat (302) and the exhaust valve seat (202) respectively.
 6. The internal combustion engine (100) as claimed in claim 1, wherein the inlet valve (164) opens the inlet port (132) of the internal combustion engine (100) to induct a charge at about 240 to 290 degrees of rotation of the crankshaft (104) after the top dead centre position and closes the inlet port (132) at about 280 to 330 degrees of rotation of the crankshaft (104) after the top dead centre position.
 7. The internal combustion engine (100) as claimed in claim 1, wherein the exhaust valve (166) opens the exhaust port (134) of the internal combustion engine (100) to discharge combustion products at about 110 to 140 degrees of rotation of the crankshaft (104) after the top dead centre position and closes the exhaust port (134) at about 240 to 280 degrees of rotation of the crankshaft (104) after the top dead centre position.
 8. The internal combustion engine (100) as claimed in claim 1, further comprising an ignition element (138) to ignite a charge, wherein the ignition element (138) fires at about 325 degrees of rotation of the crankshaft (104) after the top dead centre position.
 9. The internal combustion engine (100) as claimed in claim 1, wherein an expansion stroke of the internal combustion engine (100) begins at about 10 degrees of rotation of the crankshaft (104) after the top dead centre position and concludes at about 120 to 140 degrees of rotation of the crankshaft (104) after the top dead centre position.
 10. The internal combustion engine (100) as claimed in claim 1, wherein an expansion stroke takes place for about 110 to 150 degrees of rotation of the crankshaft (104) and a compression stroke takes place for about 60 to 110 degrees of rotation of the crankshaft (104).
 11. The internal combustion engine (100) as claimed in claim 1, wherein the plurality of transfer ports (116) is formed in the shape of a truncated cone having an apex angle α and having a base at an opening of the induction passage into the plurality of transfer port (116), wherein the apex angle is the included angle of the lateral surface of the cone, and wherein an axis of each of the plurality of transfer ports (116) is inclined to the cylinder wall (118) of the cylinder bore (108) at an angle, wherein the angle is measured in a horizontal plane.
 12. The internal combustion engine as claimed in claim 11, wherein the apex angle α is about 10 to 30 degrees.

Patentansprüche

1. Verbrennungsmotor (100), umfassend:
 - einen Zylinderblock (106), der mindestens eine Zylinderbohrung (108) aufweist, die von einer Zylinderwand (118) definiert wird, wobei der Zylinderblock (106) umfasst
 - einen Kolben (112), der sich in der Zylinderbohrung (108) hin und her bewegt, um eine Pleuellwelle (104) anzutreiben;

eine Vielzahl von Durchlasskanälen (116), die in der Zylinderwand (118) ringförmig entlang eines Umfangs der Zylinderwand (118) bereitgestellt sind; und
einen Zylinderkopf (126), der auf dem Zylinderblock (106) angebracht ist, wobei der Zylinderkopf (126) umfasst:

mindestens einen Einlasskanal (132);
mindestens einen Auslasskanal (134);
ein Einlassventil (164), das am Einlasskanal (132) bereitgestellt ist, um ein Öffnen und Schließen des Einlasskanals (132) zu regeln;
ein Auslassventil (166), das am Auslasskanal (134) bereitgestellt ist, um ein Öffnen und Schließen des Auslasskanals (134) zu regeln;
einen Einlassventilsitz (302), der am Einlasskanal (132) bereitgestellt ist, zum Absetzen des Einlassventils (164);
einen Auslassventilsitz (202), der am Auslasskanal (134) bereitgestellt ist, zum Absetzen des Auslassventils (166);

gekennzeichnet durch

einen Einlassventilsitzrand (304), der am Einlassventilsitz (302) bereitgestellt ist, um zu ermöglichen, das Öffnen des Einlasskanals (132) zu verzögern und das Schließen desselben vorzurücken; und
einen Auslassventilsitzrand (204), der am Auslassventilsitz (202) bereitgestellt ist, um zu ermöglichen, das Öffnen des Auslasskanals (134) zu verzögern und das Schließen desselben vorzurücken, und wobei
ein maximaler Ventilhub des Einlassventils (164) vom Einlassventilsitz während der Bewegung des Kolbens von einer unteren Totpunktlage zu einer oberen Totpunktlage bei etwa 120 bis 125 Grad Drehung der Kurbelwelle (104) nach der unteren Totpunktlage erfolgt, wobei ein maximaler Ventilhub des Auslassventils (166) vom Auslassventilsitz (202) während der Bewegung des Kolbens (112) von der unteren Totpunktlage zur oberen Totpunktlage bei etwa 20 bis 30 Grad Drehung der Kurbelwelle (104) nach der unteren Totpunktlage erfolgt, und
wobei
der Kolben (112) bei etwa 130 bis 150 Grad Drehung der Kurbelwelle (104) nach der oberen Totpunktlage die Vielzahl von Durchlasskanälen (116) freilegt, um das Spülfluid einzuleiten, und die Vielzahl von Durchlasskanälen (116) bei etwa 210 bis 230 Grad Drehung der Kurbelwelle (104) nach der oberen Totpunktlage bedeckt, wobei
das Einlassventil (164) geschlossen bleibt, wäh-

rend die Vielzahl von Durchlasskanälen (116) offen sind.

2. Verbrennungsmotor (100) nach Anspruch 1, wobei der Einlassventilsitzrand (304) und Auslassventilsitzrand (204) jeweils an einem Umfang des Einlassventilsitzes (302) und des Auslassventilsitzes (202) bereitgestellt sind, und wobei der Einlassventilsitzrand (304) und Auslassventilsitzrand (204) zylindrisch sind und sich jeweils entlang einer Längsachse des Einlassventils (164) und Auslassventils (166) erstrecken.
3. Verbrennungsmotor (100) nach Anspruch 1, wobei sich der Einlassventilsitzrand (304) und Auslassventilsitzrand (204) jeweils auf einer Länge von etwa 5 % bis etwa 40 % des Einlassventilhubes und Auslassventilhubes erstrecken.
4. Verbrennungsmotor (100) nach Anspruch 1, wobei sich der Einlassventilsitzrand (304) und Auslassventilsitzrand (204), vom Einlassventilsitz (302) und Auslassventilsitz (202) gemessen, jeweils auf einer Länge von etwa 1 Millimeter bis 1,5 Millimeter erstrecken.
5. Verbrennungsmotor (100) nach Anspruch 1, wobei der Einlassventilsitzrand (304) und Auslassventilsitzrand (204) jeweils am Einlassventilsitz (302) und dem Auslassventilsitz (202) bereitgestellt sind.
6. Verbrennungsmotor (100) nach Anspruch 1, wobei das Einlassventil (164) den Einlasskanal (132) des Verbrennungsmotors (100) bei etwa 240 bis 290 Grad Drehung der Kurbelwelle (104) nach der oberen Totpunktlage öffnet, um ein Frischgas einzuleiten, und den Einlasskanal (132) bei etwa 280 bis 330 Grad Drehung der Kurbelwelle (104) nach der oberen Totpunktlage schließt.
7. Verbrennungsmotor (100) nach Anspruch 1, wobei das Auslassventil (166) den Auslasskanal (134) des Verbrennungsmotors (100) bei etwa 110 bis 140 Grad Drehung der Kurbelwelle (104) nach der oberen Totpunktlage öffnet, um Verbrennungsprodukte auszustoßen, und den Auslasskanal (134) bei etwa 240 bis 280 Grad Drehung der Kurbelwelle (104) nach der oberen Totpunktlage schließt.
8. Verbrennungsmotor (100) nach Anspruch 1, weiter ein Zündelement (138) umfassend, um ein Frischgas zu zünden, wobei das Zündelement (138) bei etwa 325 Grad Drehung der Kurbelwelle (104) nach der oberen Totpunktlage auslöst.
9. Verbrennungsmotor (100) nach Anspruch 1, wobei ein Expansionstakt des Verbrennungsmotors (100) bei etwa 10 Grad Drehung der Kurbelwelle (104)

nach der oberen Totpunktlage beginnt und bei etwa 120 bis 140 Grad Drehung der Kurbelwelle (104) nach der oberen Totpunktlage endet.

10. Verbrennungsmotor (100) nach Anspruch 1, wobei ein Expansionstakt über etwa 110 bis 150 Grad Drehung der Kurbelwelle (104) stattfindet, und ein Verdichtungstakt über etwa 60 bis 110 Grad Drehung der Kurbelwelle (104) stattfindet. 5
11. Verbrennungsmotor (100) nach Anspruch 1, wobei die Vielzahl von Durchlasskanälen (116) in der Form eines Kegelstumpfs gebildet sind, der einen Scheitelwinkel alpha aufweist und eine Basis an einer Öffnung des Einleitdurchgangs in die Vielzahl von Durchlasskanälen (116) aufweist, wobei es sich beim Scheitelwinkel um den eingeschlossenen Winkel der Seitenfläche des Kegels handelt, und wobei eine Achse von jedem aus der Vielzahl von Durchlasskanälen (116) in einem Winkel zur Zylinderwand (118) der Zylinderbohrung (108) geneigt ist, wobei der Winkel in einer horizontalen Ebene gemessen wird. 10
12. Verbrennungsmotor nach Anspruch 11, wobei der Scheitelwinkel alpha etwa 10 bis 30 Grad beträgt. 15

Revendications

1. Moteur à combustion interne (100) comprenant : 30

un bloc cylindre (106) présentant au moins un alésage de cylindre (108) défini par une paroi de cylindre (118), le bloc cylindre (106) comprenant, 35

un piston (112) se déplaçant alternativement dans ledit alésage de cylindre (108) pour entraîner un vilebrequin (104) ;

une pluralité d'orifices de transfert (116) prévus en anneau dans la paroi de cylindre (118) le long d'une périphérie de la paroi de cylindre (118) ; et une tête de cylindre (126) montée sur le bloc cylindre (106), la tête de cylindre (126) comprenant : 40

au moins un orifice d'entrée (132) ;

au moins un orifice d'échappement (134) ;

une soupape d'entrée (164) prévue au niveau de l'orifice d'entrée (132) pour réguler une ouverture et fermeture de l'orifice d'entrée (132) ; 50

une soupape d'échappement (166) prévue au niveau de l'orifice d'échappement (134) pour réguler une ouverture et fermeture de l'orifice d'échappement (134) ; 55

un siège de soupape d'entrée (302) prévu au niveau de l'orifice d'entrée (132) pour le

repos de la soupape d'entrée (164) ;

un siège de soupape d'échappement (202) prévu au niveau de l'orifice d'échappement (134) pour le repos de la soupape d'échappement (166) ;

caractérisé par

un bord de siège de soupape d'entrée (304) prévu au niveau du siège de soupape d'entrée (302) pour permettre le retard d'ouverture et pour avancer la fermeture de l'orifice d'entrée (132) ; et

un bord de siège de soupape d'échappement (204) prévu au niveau du siège de soupape d'échappement (202) pour permettre le retard d'ouverture et avancer la fermeture de l'orifice d'échappement (134), et dans lequel

un levage de soupape maximum de la soupape d'entrée (164) du siège de soupape d'entrée apparaît pendant le mouvement du piston d'une position de point mort bas à une position de point mort haut à environ 120 à 125 degrés de rotation du vilebrequin (104) après la position de point mort bas, dans lequel un levage de soupape maximum de la soupape d'échappement (166) du siège de soupape d'échappement (202) apparaît pendant le mouvement du piston (112) de la position de point mort bas à la position de point mort haut à environ 20 à 30 degrés de rotation du vilebrequin (104) après la position de point mort bas, et

dans lequel le piston (112) découvre la pluralité d'orifices de transfert (116) pour admettre le fluide de balayage à environ 130 à 150 degrés de rotation du vilebrequin (104) après la position de point mort haut et couvre la pluralité d'orifices de transfert (116) à environ 210 à 230 degrés de rotation du vilebrequin (104) après la position de point mort haut, dans lequel la soupape d'entrée (164) reste fermée alors que la pluralité d'orifices de transfert (116) est ouverte.

2. Moteur à combustion interne (100) selon la revendication 1, dans lequel le bord de siège de soupape d'entrée (304) et le bord de siège de soupape d'échappement (204) sont prévus à une circonférence du siège de soupape d'entrée (302) et du siège de soupape d'échappement (202) respectivement et dans lequel le bord de siège de soupape d'entrée (304) et le bord de siège de soupape d'échappement (204) sont cylindriques, s'étendant le long d'un axe longitudinal de la soupape d'entrée (164) et la soupape d'échappement (166) respectivement.

3. Moteur à combustion interne (100) selon la revendication 1, dans lequel le bord de siège de soupape d'entrée (304) et le bord de siège de soupape d'échappement (204) s'étendent à une longueur

d'environ 5 % à environ 40 % du levage de soupape d'entrée et du levage de soupape d'échappement respectivement.

4. Moteur à combustion interne (100) selon la revendication 1, dans lequel le bord de siège de soupape d'entrée (304) et le bord de siège de soupape d'échappement (204) s'étendent à une longueur d'environ 1 millimètre à 1,5 millimètre mesurée depuis le siège de soupape d'entrée (302) et le siège de soupape d'échappement (202) respectivement. 5
5. Moteur à combustion interne (100) selon la revendication 1, dans lequel le bord de siège de soupape d'entrée (304) et le bord de siège de soupape d'échappement (204) sont prévus au niveau du siège de soupape d'entrée (302) et du siège de soupape d'échappement (202) respectivement. 10
6. Moteur à combustion interne (100) selon la revendication 1, dans lequel la soupape d'entrée (164) ouvre l'orifice d'entrée (132) du moteur à combustion interne (100) pour admettre une charge à environ 240 à 290 degrés de rotation du vilebrequin (104) après la position du point mort haut et ferme l'orifice d'entrée (132) à environ 280 à 330 degrés de rotation du vilebrequin (104) après la position de point mort haut. 15
7. Moteur à combustion interne (100) selon la revendication 1, dans lequel la soupape d'échappement (166) ouvre l'orifice d'échappement (134) du moteur à combustion interne (100) pour évacuer des produits de combustion à environ 110 à 140 degrés de rotation du vilebrequin (104) après la position de point mort haut et ferme l'orifice d'échappement (134) à environ 240 à 280 degrés de rotation du vilebrequin (104) après la position de point mort haut. 20
8. Moteur à combustion interne (100) selon la revendication 1, comprenant en outre un élément d'allumage (138) pour allumer une charge, dans lequel l'élément d'allumage (138) s'allume à environ 325 degrés de rotation du vilebrequin (104) après la position point mort haut. 25
9. Moteur à combustion interne (100) selon la revendication 1, dans lequel une course de détente du moteur à combustion interne (100) commence à environ 10 degrés de rotation du vilebrequin (104) après la position de point mort haut et se termine à environ 120 à 140 degrés de rotation du vilebrequin (104) après la position de point mort haut. 30
10. Moteur à combustion interne (100) selon la revendication 1, dans lequel une course de détente se déroule pendant environ 110 à 150 degrés de rotation du vilebrequin (104) et une course de compression se déroule pendant environ 60 à 110 degrés de ro- 35

tation du vilebrequin (104).

11. Moteur à combustion interne (100) selon la revendication 1, dans lequel la pluralité d'orifices de transfert (116) est formée sous la forme d'un cône tronqué présentant un angle d'apex alpha et présentant une base au niveau d'une ouverture du passage d'admission dans la pluralité d'orifices de transfert (116), dans lequel l'angle d'apex est l'angle inclus de la surface latérale du cône, et dans lequel un axe de chacun de la pluralité d'orifices de transfert (116) est incliné vers la paroi de cylindre (118) de l'alésage de cylindre (108) à un angle, dans lequel l'angle est mesuré dans un plan horizontal. 40
12. Moteur à combustion interne selon la revendication 11, dans lequel l'angle d'apex alpha est d'environ 10 à 30 degrés. 45

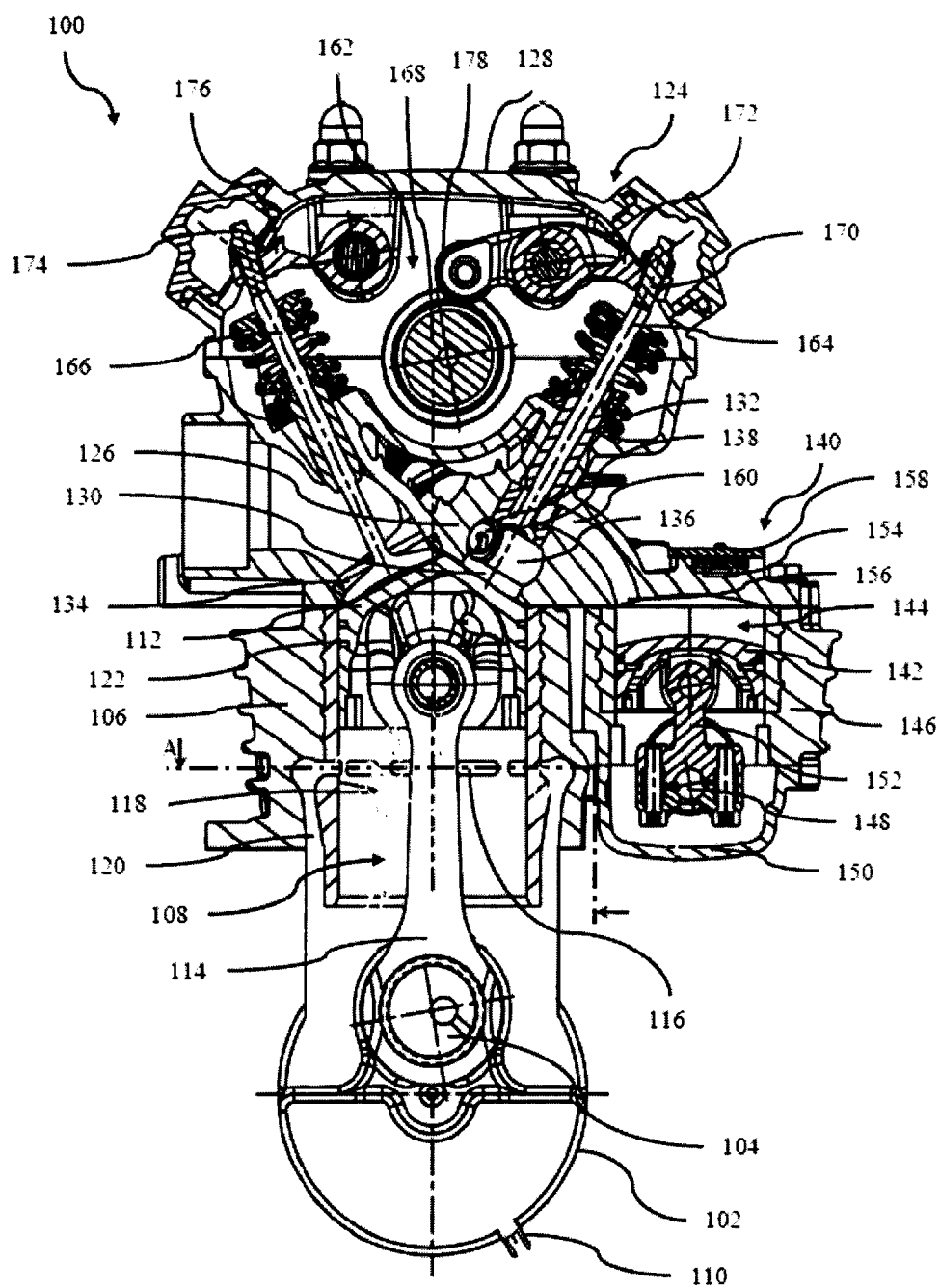


Fig. 1

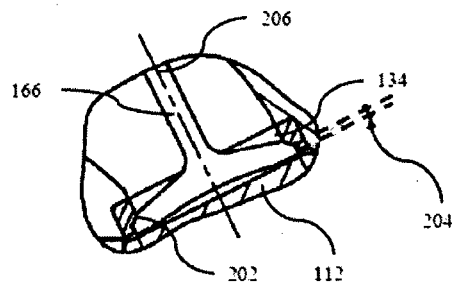


Fig. 2

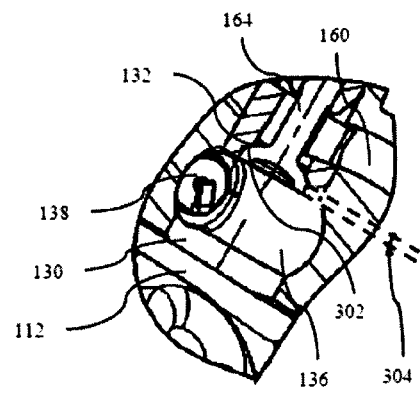


Fig. 3

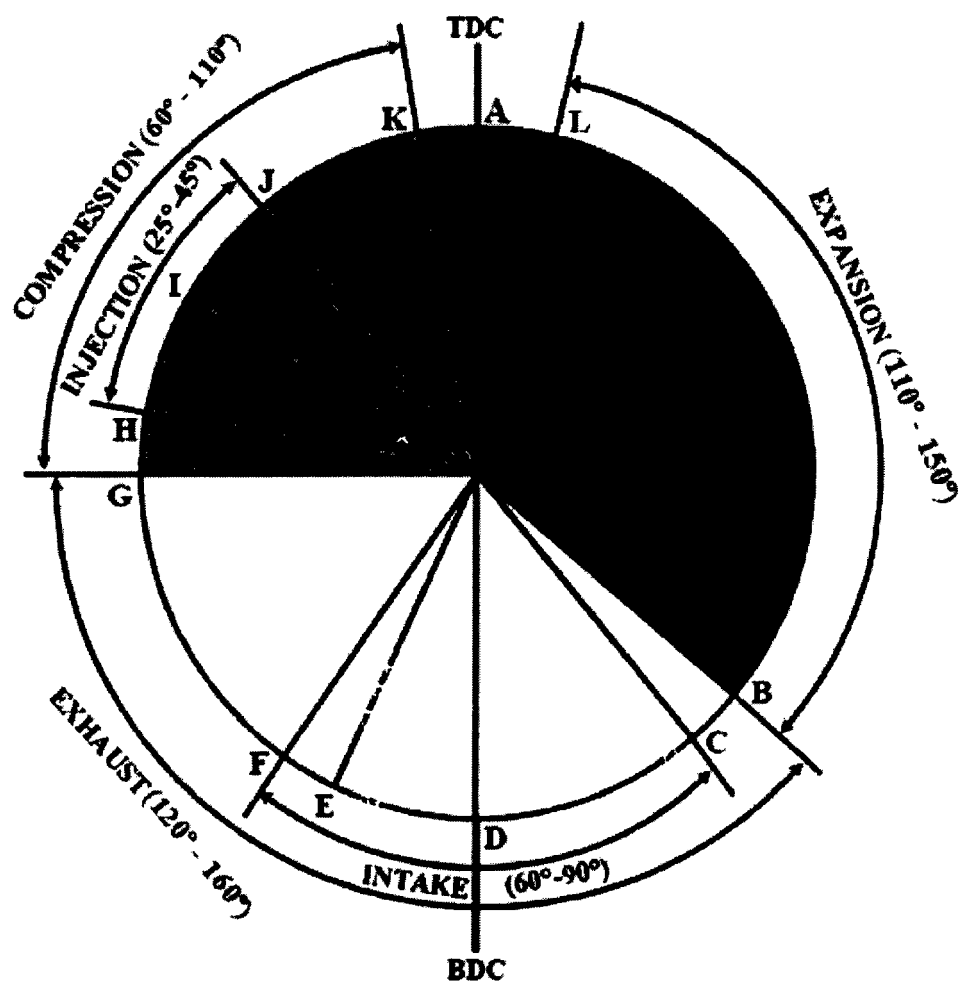


Fig. 4

REFERENCES CITED IN THE DESCRIPTION

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