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(54) **Title:** FOUR-STROKE GASEOUS FUEL ENGINE

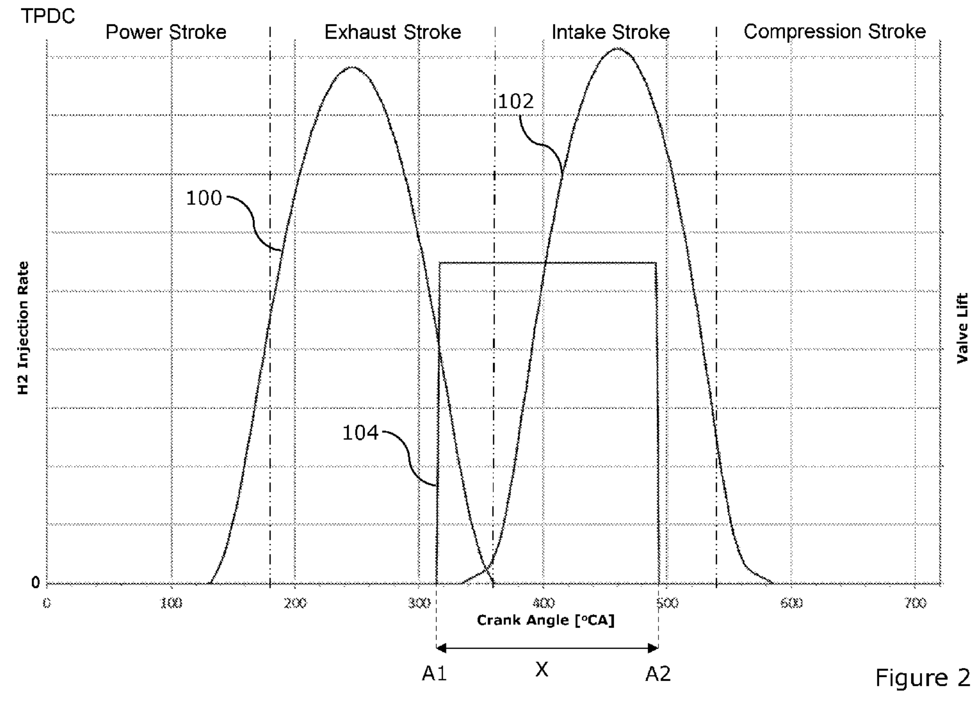

Figure 2

(57) **Abstract:** A four-stroke gaseous fuel engine comprising one or more cylinder assemblies, each cylinder assembly comprising : a cylinder including an inlet and an outlet selectively opened and closed by an intake valve and an exhaust valve respectively; a piston translationally movable within the cylinder;an intake runner leading to the inlet; and a fuel injector. The cylinder assembly is configured to selectively inject a gaseous fuel from the fuel injector into the intake runner. The engine is configured to supply gaseous fuel and air from the intake runner to the cylinder via the inlet during an intake stroke of the piston, and exhaust combustion gases from the cylinder via the outlet during an exhaust stroke of the piston. Within a four-stroke cycle of the engine, the cylinder assembly is configured to inject the gaseous fuel into the intake runner exclusively during an interval starting with a first crank angle of the piston and ending with a second crank angle of the piston. At a maximum interval that the cylinder assembly can inject the gaseous fuel into the intake

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runner, the first crank angle corresponds to the intake valve being closed and the exhaust valve being open during the exhaust stroke, and the second crank angle corresponds to the intake valve being open and the exhaust valve being closed during the intake stroke.

FOUR-STROKE GASEOUS FUEL ENGINE

FIELD

The present teachings relate to an engine, particularly a four-stroke gaseous fuel engine, a working machine, and a method for operating a four-stroke gaseous fuel engine.

BACKGROUND

Four-stroke gaseous fuel internal combustion (IC) engines are known in which a gaseous fuel, such as compressed natural gas (CNG) for example, is supplied to a cylinder of the engine via port fuel injection, in which the gaseous fuel is injected into an air supply passage upstream of the cylinder. This is in contrast to direct fuel injection, in which fuel is injected directly into the cylinder.

In such IC engines, a sufficient volume of the gaseous fuel is required to be injected into the air supply passage within each four-stroke cycle of the engine such that near-optimal combustion is achieved in the cylinder.

Hitherto, IC engines used in off-highway, and heavy commercial vehicle applications have typically used diesel as fuel, which is directly injected into the combustion chamber. To increase the efficiency of diesel and gasoline engines, it is known for intake valves to open at the end of the exhaust stroke before the piston reaches top dead centre (TDC) as it reduces the pumping losses of the engine. This potentially allows exhaust gases to flow into the air supply passage during the exhaust stroke within the cylinder causing gaseous fuel and air contained in the air intake passage to flow upstream away from the cylinder. If utilising hydrogen, for example, as an alternative fuel, and injecting the fuel into the inlet port there is an increased risk of the backflowing air and fuel in the inlet runner combusting outside the cylinder ("backfiring") or flowing into other cylinders of the engine, which may result in poor combustion and/or misfiring of those cylinders. It may also be costly to source fuel injectors capable of injecting gaseous fuel at a flow rate high enough to ensure that the required volume of gaseous fuel is supplied to the cylinder within each four-stroke cycle.

The present teachings seek to overcome or at least mitigate one or more problems associated with the prior art.

SUMMARY

According to a first aspect of the present teachings, there is provided a four-stroke gaseous fuel engine comprising one or more cylinder assemblies, each cylinder assembly comprising: a cylinder including an inlet and an outlet selectively opened and closed by an intake valve and an exhaust valve respectively; a piston translationally movable within the cylinder; an intake runner leading to the inlet; and a fuel injector, wherein the cylinder assembly is configured to selectively inject a gaseous fuel from the fuel injector into the intake runner, wherein the engine is configured to supply gaseous fuel and air from the intake runner to the cylinder via the inlet during an intake stroke of the piston, and exhaust combustion gases from the cylinder via the outlet during an exhaust stroke of the piston, wherein, within a four-stroke cycle of the engine, the cylinder assembly is configured to inject the gaseous fuel into the intake runner exclusively during an interval starting with a first crank angle of the piston and ending with a second crank angle of the piston, and wherein at a maximum interval that the cylinder assembly can inject the gaseous fuel into the intake runner, the first crank angle corresponds to the intake valve being closed and the exhaust valve being open during the exhaust stroke, and the second crank angle corresponds to the intake valve being open and the exhaust valve being closed during the intake stroke.

Advantageously, injecting gaseous fuel into the intake runner over such a maximum interval may help ensure a sufficient volume of gaseous fuel is supplied to the cylinder for near-optimal combustion, whilst also enhancing fuel-air mixing, which may promote enhanced combustion.

A maximum second crank angle at which the cylinder assembly can inject the gaseous fuel into the intake runner may be a crank angle of 495 degrees or less after a top power dead centre of the piston.

The maximum second crank angle at which the cylinder assembly can inject the gaseous fuel into the intake runner may be a crank angle of 490 degrees or less after a top power dead centre of the piston.

Advantageously, it has been found that such maximum second crank angles help to inhibit backflow of gaseous fuel and air within the intake runner away from the inlet, which may cause poor combustion and misfiring of the engine having a plurality of the cylinder assemblies.

A minimum second crank angle at which the cylinder assembly can inject the gaseous fuel into the intake runner may be a crank angle of 440 degrees or more, optionally 450 degrees or more, after a top power dead centre of the piston.

The gaseous fuel may be hydrogen.

Advantageously, hydrogen produces less harmful emissions when combusted relative to hydrocarbon gaseous fuels for example.

The maximum interval may be in the range of 160 to 200 degrees, optionally 170 to 190 degrees, for example approximately 180 degrees.

Advantageously, such an interval may help to ensure a sufficient volume of gaseous fuel is supplied to the cylinder for near-optimal combustion.

The interval may be a function of an output torque and/or an engine speed of the engine.

The maximum interval may be at a maximum output torque at a predetermined engine speed.

The predetermined engine speed may be in the range of 1600 to 2200 RPM, optionally, in the range of 1800 to 2000 RPM.

A minimum first crank angle at which the cylinder assembly can inject the gaseous fuel into the intake runner may be at a maximum output torque at a predetermined engine speed. The predetermined engine speed may be in the range of 1500 to 2600 RPM, optionally, 1800 to 2200 RPM, for example, approximately 2000 RPM.

A maximum second crank angle at which the cylinder assembly can inject the gaseous fuel into the intake runner may be at a maximum output torque at a predetermined engine speed. The predetermined engine speed may be in the range of 1200 to 1800 RPM, optionally, 1300 to 1700 RPM, for example approximately 1500 RPM.

For a constant output torque of the engine, the interval may increase as the engine speed increases.

Advantageously, increasing the interval with increased engine speed may help ensure that a sufficient volume of gaseous fuel is supplied to the cylinder as the duration of each four-stroke working cycle of the engine reduces.

The fuel injector may be configured to inject the gaseous fuel continuously between the first and second crank angles.

Advantageously, such a configuration of the fuel injector may help to ensure a sufficient volume of gaseous fuel is supplied to the cylinder for near-optimal combustion.

The fuel injector may be configured to inject the gaseous fuel at a substantially constant injection rate between the first and second crank angles.

Advantageously, such a configuration of the fuel injector may help to simplify implementation of the engine.

Gaseous fuel may be injected into the intake runner exclusively via the fuel injector.

Advantageously, such a configuration of the engine may help to simplify and reduce the cost of implementing the engine.

Within a four-stroke cycle, the intake valve may move from a closed position to an open position at a crank angle of the piston in the range of 310 to 350 degrees, optionally, 320 to 340 degrees, for example approximately 335 degrees, after a top power dead centre of the piston.

Within a four-stroke cycle, the intake valve may move from an open position to a closed position at a crank angle of the piston in the range of 570 to 610 degrees, optionally, in the range of 580 to 600 degrees, for example approximately 585 degrees, after a top power dead centre of the piston.

Within a four-stroke cycle, the exhaust valve may move from an open position to a closed position at a crank angle of the piston in the range of 350 to 390 degrees, optionally in the range of 360 to 380 degrees, for example approximately 375 degrees, after a top power dead centre of the piston.

Within a four-stroke cycle, the exhaust valve may move from a closed position to an open position at a crank angle of the piston which may be in the range of 110 to 150 degrees, optionally, in the range of 120 to 140 degrees, for example approximately 130 degrees, after a top power dead centre of the piston.

Within a four-stroke cycle: the exhaust valve may move from a closed to an open position at a crank angle of the piston of approximately 140 degrees after a top power dead centre of the piston; the exhaust valve may move from an open to a closed position at a crank angle of the piston of approximately 380 degrees after a top power dead centre of the piston; the intake valve may move from a closed to an open position at a crank angle of the piston of approximately 335 degrees after a top power dead centre of the piston; and the intake valve may move from an open to a closed position at a crank angle of the piston of approximately 580 degrees after a top power dead centre of the piston.

Advantageously, such a configuration of the engine has been found to minimise backflow of gaseous fuel and air in the intake runner away from the inlet, resulting in an improved volumetric efficiency of the engine.

The cylinder may comprise a flat roof arranged substantially normal to an axis of piston motion. The intake valve and the exhaust valve may be arranged to open and close via movement along respective axes that are parallel to said axis of piston motion.

The cylinder assembly may be configured to inject the gaseous fuel into the intake runner from the fuel injector at an injection angle to a mean flow direction of air flowing through the intake runner and/or the horizontal, in use, in the range of 0 to 25 degrees, optionally, in the range of 10 to 20 degrees, for example 14 degrees.

Advantageously, such injection angles have been found to provide good mixing between fuel and air, and to inhibit backflow of fuel entering the intake runner via the fuel injector, in use.

Each cylinder assembly may comprise an intake port leading to the inlet. An outlet portion of the intake runner may lead to the intake port. The cylinder assembly may be configured to direct the injected gaseous fuel from the fuel injector along an injection axis in a direction substantially towards the outlet portion. The intake port may have a width along an axis of the intake valve. The injection axis may intersect the axis of the intake valve at a position greater than 50%, optionally greater than 60%, optionally greater than 70%, of the width of the intake port from the inlet.

Advantageously, such configurations of the injection port and the injection axis have been found to provide good mixing between fuel and air in the intake port, and to inhibit backflow of fuel entering the intake runner via the fuel injector, in use.

The cylinder may comprise a further inlet and a further outlet selectively opened and closed by a further intake valve and a further exhaust valve respectively. The intake runner may lead to the further inlet. The engine may be configured to supply gaseous fuel and air from the intake runner to the cylinder via the further inlet during the intake stroke of the piston, and exhaust combustion gases from the cylinder via the further outlet during the exhaust stroke of the piston.

The engine may further comprise an intake port including a downstream end leading to the inlet and the further inlet. The engine may further comprise an exhaust port including an upstream end leading from the outlet and the further outlet. The intake runner may lead to an upstream end of the intake port. The downstream end of the intake port may bifurcate to the inlet and the further inlet. The ports may be arranged in a tandem configuration.

According to a second aspect of the present teachings, there is provided a four-stroke gaseous fuel engine comprising: one or more cylinder assemblies; an intake assembly; a fuel tank for storing a gaseous fuel; a fuel supply conduit, such as a fuel rail; and a control unit. Each cylinder assembly comprises: a cylinder including an inlet selectively opened and closed by an intake valve; a piston translationally movable within the cylinder; an intake runner leading to the inlet; and a fuel injector configured to selectively inject a gaseous fuel into the intake runner. The engine is configured to supply gaseous fuel and air from the intake runner to the cylinder via the inlet. The engine is configured to supply a gaseous fuel stored in the fuel tank to the fuel supply conduit, such that a fuel pressure in the fuel supply conduit is independent of a fuel pressure in the fuel tank. The engine is configured to supply the gaseous fuel from the fuel supply conduit to each fuel injector, such that a fuel injection flow rate of each fuel injector is dependent on the fuel pressure in the fuel supply conduit. The intake assembly is configured to supply air to each intake runner. The control unit is configured to control a flow rate of air supplied to each intake runner via the intake assembly. Within a four-stroke cycle of the engine, the control unit is configured to control each fuel injector to inject the gaseous fuel into the intake runner exclusively during an interval starting with a first crank angle of the piston and ending with a second crank angle of the piston. For each cylinder assembly, the control unit is configured to: determine a first interval based on the fuel pressure in the fuel supply conduit; and in response to the first interval being greater than a predetermined maximum interval, reduce the flow rate of air supplied to the intake runner, and control the fuel injector based on a second interval less than or equal to the predetermined maximum interval.

When the fuel pressure in the fuel supply conduit drops below a normal operating pressure threshold (e.g. due to a sudden increase in operator engine power demand), there will be a corresponding reduction in the possible flow rate of each fuel injector. To ensure enough fuel is injected into each intake runner to provide an operator demanded engine power output, a fuel injection interval, starting with a first crank angle of the piston and ending with a second crank angle of the piston, may need to be increased. However, increasing the second crank angle beyond a maximum value may result in backflow of gaseous fuel and air within each intake runner away from the inlet, which may cause poor combustion and misfiring of the engine having a plurality of the cylinder assemblies. Moreover, a minimum first crank angle may be based on a computational scheduling of the control unit (i.e. a fixed crank angle within a four stroke cycle at which the control unit determines the fuel injection interval), and/or a computational speed of the control unit (i.e. how quickly the control unit can respond to operator demand), and therefore it may not be possible to reduce the first crank angle below this minimum.

Advantageously, reducing the flow rate of air supplied to each intake runner in response to the determined first interval being greater than a predetermined maximum interval helps to prevent the fuel mixture becoming too lean, and thus helps inhibit damage to the engine and/or abnormal combustion, whilst helping to ensure that the first crank angle does not drop below the minimum, and the second crank angle does not exceed the maximum.

The intake assembly may comprise a throttle. The control unit may be configured to control the flow rate of air supplied to each intake runner via the intake assembly by operating the throttle.

Advantageously, such a configuration helps provide precise control of the flow rate of air supplied to the intake runner(s).

The control unit may be configured to determine the first interval further based on an output power demand of the engine.

In response to the fuel pressure in the fuel supply conduit being less than a predetermined pressure threshold for a predetermined period of time or more, the control unit may be configured to output an alarm signal and/or maintain a reduced flow rate of air supplied to each intake runner irrespective of the determined first interval.

Advantageously, such a configuration helps to prevent damage to the engine when there is a persistent fault causing the fuel pressure in the fuel supply conduit to drop below the predetermined pressure threshold.

According to a third aspect of the present teachings, there is provided a four-stroke gaseous fuel engine comprising: one or more cylinder assemblies; an intake assembly; a fuel tank for storing a gaseous fuel; a fuel supply conduit, such as a fuel rail; and a control unit. Each cylinder assembly comprises: a cylinder including an inlet selectively opened and closed by an intake valve; a piston translationally movable within the cylinder; an intake runner leading to the inlet; and a fuel injector configured to selectively inject a gaseous fuel into the intake runner. The engine is configured to supply gaseous fuel and air from the intake runner to the cylinder via the inlet. The engine is configured to supply a gaseous fuel stored in the fuel tank to the fuel supply conduit, such that a fuel pressure in the fuel supply conduit is independent of a fuel pressure in the fuel tank. The engine is configured to supply the gaseous fuel from the fuel supply conduit to each fuel injector, such that a fuel injection flow rate of each fuel injector is dependent on the fuel pressure in the fuel supply conduit. The intake assembly is configured to supply air to each intake runner. The control unit is configured to control a flow rate of air supplied to each intake runner via the intake assembly. In response to the fuel pressure in the fuel supply conduit being less than a predetermined pressure threshold for a predetermined period of time or more, the control unit is configured to output an alarm signal and/or maintain a reduced flow rate of air supplied to each intake runner irrespective of the determined first interval.

Advantageously, such a configuration helps to prevent damage to the engine when there is a persistent fault causing the fuel pressure in the fuel supply conduit to drop below the predetermined pressure threshold.

According to a fourth aspect of the present teachings, there is provided a working machine comprising the engine according to the first, second or third aspects.

According to a fifth aspect of the present teachings, there is provided a method for operating a four-stroke gaseous fuel engine configured to supply gaseous fuel and air from an intake runner to a cylinder via an inlet of the cylinder during an intake stroke of a piston within the cylinder, and exhaust gases from the cylinder via an outlet of the cylinder during an exhaust stroke of the piston, the inlet selectively opened and closed by an intake valve, and the outlet selectively opened and closed by an outlet valve, the method comprising the step of:

within a four-stroke cycle of the engine, injecting a gaseous fuel into the intake runner exclusively during an interval starting with a first crank angle of the piston and ending with a second crank angle of the piston,

wherein at a maximum interval that the cylinder assembly can inject the gaseous fuel into the intake runner, the first crank angle corresponds to the intake valve being closed and the exhaust valve being open during the exhaust stroke, and the second crank angle corresponds to the intake valve being open and the exhaust valve being closed during the intake stroke.

According to a sixth aspect of the present teachings, there is provided a method for operating a four-stroke gaseous fuel engine configured to supply gaseous fuel and air from an intake runner to a cylinder via an inlet of the cylinder, the inlet selectively opened and closed by an intake valve, the engine comprising:

a fuel tank for storing a gaseous fuel;

a fuel supply conduit, such as a fuel rail; and

a fuel injector configured to selectively inject a gaseous fuel into the intake runner,

the method comprising the steps of:

supplying a gaseous fuel stored in the fuel tank to the fuel supply conduit, such that a fuel pressure in the fuel supply conduit is independent of a fuel pressure in the fuel tank;

supplying the gaseous fuel from the fuel supply conduit to the intake runner via the fuel injector, such that a fuel injection flow rate of the fuel injector is dependent on the fuel pressure in the fuel supply conduit;

within a four-stroke cycle of the engine, injecting a gaseous fuel into the intake runner exclusively during an interval starting with a first crank angle of a piston within the cylinder and ending with a second crank angle of the piston;

determining a first interval based on the fuel pressure in the fuel supply conduit; and

reducing a flow rate of air supplied to the intake runner in response to the first interval being greater than a predetermined maximum interval.

BRIEF DESCRIPTION OF THE DRAWINGS

Embodiments are now disclosed by way of example only with reference to the drawings, in which:

Figure 1A is a plan view of an internal combustion engine according to an embodiment of the present teachings;

Figure 1B is a cross-sectional view of the internal combustion engine along section X-X shown in Figure 1A;

Figure 1C is a cross-sectional view of the internal combustion engine along section Y-Y shown in Figure 1A;

Figure 2 shows plots of the movement of an intake valve and an exhaust valve, and an injection rate of a fuel injector, with respect to a crank angle of a piston, during a four-stroke cycle of the internal combustion engine of Figure 1A;

Figure 3 is a contour plot of a first (lower) crank angle of a fuel injection interval with respect to an engine speed and an output torque of the internal combustion engine of Figure 1A;

Figure 4 is a contour plot of a second (upper) crank angle of a fuel injection interval with respect to an engine speed and an output torque of the internal combustion engine of Figure 1A;

Figure 5 is a block diagram representation of an engine control unit of the internal combustion engine of Figure 1A;

Figure 6 is a flowchart representation of an exemplary method performed by the engine control unit of Figure 5; and

Figure 7 is a plan view of a cylinder, an intake port and an exhaust port of the internal combustion engine of Figure 1A.

DETAILED DESCRIPTION OF EMBODIMENT(S)

Referring firstly to Figures 1A to 1C, an embodiment includes an internal combustion engine 1. Figure 1A shows a plan view of the engine 1, Figure 1B shows a cross-sectional view of the engine 1 along section X-X shown in Figure 1A, Figure 1C shows a cross-sectional view of the engine 1 along section Y-Y shown in Figure 1A.

The engine 1 is a four-stroke gaseous fuel engine configured to be powered by a gaseous fuel, such as hydrogen, compressed natural gas (CNG), landfill gas or the like.

The engine 1 may be suitable for use as the prime mover in a working machine (not shown), such as a telescopic handler, a forklift truck, a backhoe loader, a wheeled loading shovel, a dumper, an excavator or a tractor, for example. Such working machines are suitable for use in off-highway industries such as agriculture and construction. In these industries they are generally configured to perform tasks such as excavation, load handling, harvesting or planting crops. The engine may also be utilised in a genset – a self-contained unit to provide electrical power at off-grid locations. As such the engine 1 is typically required to have certain characteristics such as a high torque output over a wide engine speed band, with peak torque occurring at a relatively low engine speed. which differ from light passenger vehicles, for example. In off-highway applications, this provides “torque backup” that enables working machines to continue to carry out working operations when encountering increased loads, or resistance to a working operation – e.g. an excavator encountering a particularly solid piece of earth to be excavated.

In this embodiment, the engine 1 has four cylinder assemblies indicated generally at 19. As configured the engine has a maximum power output of around 55kW, although it will be appreciated that the present teachings are applicable to engines with a wide range of power outputs. Each cylinder assembly 19 includes a cylinder 5 including two inlets 6 and two outlets 9, a piston 20 translationally movable within the cylinder 5, an intake runner 16 leading to the two inlets 6, and a fuel injector 22. Each cylinder assembly 19 is configured to selectively inject a gaseous fuel from the fuel injector 22 into the intake runner 16. Each inlet 6 is selectively opened and closed by an intake valve 7i, such that there are two intake valves 7i. Each outlet 9 is selectively opened and closed by an exhaust valve 7e such that there are two exhaust valves 7e.

The engine 1 is configured to supply gaseous fuel and air from the intake runner 16 to the cylinder 5 via the inlets 6 during an intake stroke of the piston 20, and exhaust combustion gases from the cylinder 5 via the outlets 9 during an exhaust stroke of the piston 20.

In alternative embodiments (not shown), the engine 1 may have more or fewer cylinder assemblies 19, e.g. 2, 3, 6, or 8. In addition, in other embodiments the cylinders 5 may be oriented in a "V" or boxer configuration rather than inline as in the disclosed embodiment.

In alternative embodiments (not shown), each cylinder 5 may include only a single inlet 6 and/or a single outlet 9. Alternatively, each cylinder 5 may include more than two inlets 6 and/or more than two outlets 9.

Each piston 20 is configured to translate along an axis C of the respective cylinder 5.

Figures 1B and 1C show the typical orientation of the engine 1 when implemented in a vehicle such as a working machine, in use, with the axis C of the cylinders 5 substantially vertical. However, in some embodiments the cylinders 5 may be orientated at an inclined angle with respect to the vertical.

Each cylinder assembly 19 includes an intake port 4. A downstream end 4d of each intake port 4 leads to the corresponding inlets 6 of the cylinder assembly 19 (see Figure 3). An outlet portion 16o of each intake runner 16 leads to an upstream end 4u of the corresponding intake port 4 of the cylinder assembly 19.

The engine 1 comprises a cylinder block 2, a cylinder head 3, and an intake assembly 10. The cylinder block 2 includes the cylinders 5. The cylinder head 3 includes the intake port 4 and a downstream portion 16d of the intake runner 16 of each cylinder assembly 19. The intake assembly 10 includes an upstream portion 16u of the intake runner 16 of each cylinder assembly 19. Each upstream portion 16u leads to one of the downstream portions 16d.

The cylinder head 3 is mounted to the cylinder block 2. The intake assembly 10 is mounted to the cylinder head 3. The engine 1 is configured such that the intake assembly 10 supplies a mixture of air and fuel to the intake ports 4 of the cylinder head 3 via the intake runners 16. The air-fuel mixture is then supplied from the intake ports 4 to the corresponding cylinders 5 of the cylinder block 2 via the inlets 6. As such, the engine 1 is a port fuel injection engine (i.e. fuel is provided to the cylinders 5 via port fuel injection).

The intake assembly 10 includes an intake manifold 11. The intake manifold 11 includes a first plenum 12, a second plenum 14, and the upstream portions 16u of the intake runners 16. The first plenum 12 includes an intake manifold inlet 13. The intake runners 16 are fluidly connected to, and extend from, the second plenum 14. In use, air supplied via the intake manifold inlet 13 travels sequentially along the first plenum 12, the second plenum 14, and the intake runners 16 towards the inlets 6.

In alternative embodiments (not shown), air may be supplied to the intake runners 16 via any suitable arrangement.

The intake assembly 10 includes an air supply passage 15 (part of which is shown schematically as two parallel lines in Figure 1A) connected to the intake manifold inlet 13. The air supply passage 15 includes an air supply inlet 15i open to the external environment. The air supply passage 15 supplies ambient air from the air supply inlet 15i to the intake manifold inlet 13 for supply to the intake runners 16.

The intake assembly 10 includes a throttle 17 along the air supply passage 15 for controlling the flowrate of air from the air supply inlet 15i to the intake manifold inlet 13 when the engine 1 is running. The throttle 17 may be manually operated via a control input device, such as an accelerator pedal (e.g. via a throttle by wire system), and/or automatically operated by a control unit, as discussed more below.

The intake assembly 10 includes a compressor 23 along the air supply passage 15 (e.g. a turbocharger, a supercharger, or any device suitable for compressing engine intake air). In the illustrated embodiment, the throttle 17 is downstream of the compressor 23, but may be upstream of the compressor 23 in alternative embodiments.

In the illustrated embodiment, each fuel injector 22 is supplied with gaseous fuel via a fuel supply system 50 including a fuel supply conduit 52, which in the illustrated embodiment is a common fuel rail 52, in fluid communication with each fuel injector 22. The fuel supply system 50 supplies the gaseous fuel from the fuel supply conduit 52 to each fuel injector 22 such that a (e.g. maximum) fuel injection flow rate of each fuel injector 22 is dependent on the fuel pressure in the fuel supply conduit 52.

The fuel supply system 50 of the engine 1 includes a fuel tank 51 for storing the gaseous fuel, and a supply line 53 (represented as a dashed line in Figure 1A) for supplying the gaseous fuel from the fuel tank 51 to the fuel supply conduit 52. A pressure regulator arrangement 55 in the supply line 53 is configured to maintain a substantially constant fuel pressure in the fuel supply conduit 52 irrespective of the fuel pressure in the fuel tank 51 under normal operating conditions. As such, the fuel pressure in the fuel supply conduit 52 is independent of the fuel pressure in the fuel tank 51. The pressure regulator arrangement 55 helps maintain the fuel injection flow rate of each fuel injector 22 as the fuel pressure in the fuel tank 52 decreases.

As the engine 1 utilises a gaseous fuel, e.g. hydrogen, a spark is required to initiate combustion. Thus, each cylinder assembly 19 comprises a spark plug 21 (illustrated schematically) mounted intermediate the intake and outlet ports 6, 9 in the cylinder head 3. If a hydrogen fuelled engine uses carry-over parts from a diesel engine (which does not require a spark plug) as a way of minimising costs, an advantage of port fuel injection is that the space for an injector in the cylinder head 3 is made available for a spark plug

21 instead, by repositioning the fuel injector 22 away from being directly above the corresponding piston 20.

The engine 1 includes a crankshaft (not shown) coupled to each piston 20. The engine 1 is configured such that translational movements of each piston 20 is converted into rotational movement of the crankshaft around a rotational axis thereof. Rotation of the crankshaft drives a camshaft (not shown), which moves the intake valves 7i and the exhaust valves 7e between open and closed positions. A rotation of the crankshaft by one degree is referred to as a degree of crank angle.

With reference to Figure 2, the engine 1 repeats a working cycle including four strokes of each piston 20. These four strokes are: a power stroke; an exhaust stroke; an intake stroke; and a compression stroke. The four-stroke working cycle of the engine 1 includes two complete revolutions of the crankshaft and thus 720 degrees of crank angle. Exactly one four-stroke working cycle of the engine 1 is shown in Figure 2.

Each piston 20 is translationally movable between a bottom dead centre and a top dead centre. Within each four-stroke cycle, there are two different types of top dead centre. A first of these types is the so-called top charge changing dead centre (TCDC), which occurs between the exhaust stroke and the intake stroke, and corresponds to a crank angle of 360 degrees in Figure 2. The second type is the so-called top power dead centre (TPDC), which occurs between the compression stroke and the power stroke, and corresponds to a crank angle of 0 degrees in Figure 2. Each piston 20 is at a bottom dead centre between the power stroke and the exhaust stroke at a crank angle of 180 degrees in Figure 2, and between the intake stroke and the compression stroke at a crank angle of 540 degrees in Figure 2.

Figure 2 shows exemplary plots of the movements of the intake valves 7i and the exhaust valves 7e, and the fuel injection rate of the fuel injector 22, with respect to the crank angle of the respective piston 20 during a four-stroke cycle of the engine 1 for each cylinder assembly 19.

The exemplary plots of the movements of the intake valves 7i and the exhaust valves 7e are typical for a conventional diesel engine. As such, if reconfiguring an existing diesel engine to operate with a gaseous fuel, e.g. hydrogen, or seeking to commonise parts across engines operating with different fuel, a common camshaft for controlling valve timing may be utilised for both diesel and at least hydrogen fuelled engines. Such engines may also use common engine blocks, cylinder heads, crankshafts, crankshaft connecting rods, bearings, lubrication systems, gear trains, cooling systems, flywheels, and parts of the valve train such as pushrods and rocker arms.

Figure 2 shows a first plot 100 of the amount of lift of each exhaust valve 7e away from the respective outlet 9 with respect to the crank angle of the piston 20. The outlets 9 are closed when the lift of the exhaust valves 7e away from the outlets 9 is zero, and open, at least partially, otherwise.

Figure 2 further shows a second plot 102 of the amount of lift of the intake valves 7i away from the respective inlets 6 with respect to the crank angle of the piston 20. The inlets 6 are closed when the lift of the intake valves 7i away from the inlets 6 is zero, and open, at least partially, otherwise.

The term "open" includes any position of the valves that is not closed – i.e. in a maximum open position or partially open.

In the exemplary embodiment, the maximum valve lift of each exhaust valve 7e away from the respective outlet 9 is in the range of 5 to 15 millimetres, and the maximum valve lift of each intake valve 7i away from the respective inlet 6 is in the range of 5 to 15 millimetres.

Figure 2 further shows a third plot 104 of the injection rate of gaseous fuel from the fuel injector 22 into the intake runner 16 with respect to the crank angle of the piston 20. In the illustrated embodiment, the gaseous fuel is hydrogen. Note that in Figure 2, the scaling of the first plot 100 and the second plot 102 is different to the scaling of the third plot 104.

Within a four-stroke cycle of the engine 1, the cylinder assembly 19 is configured to inject the gaseous fuel from the fuel injector 22 into the intake runner 16 exclusively during an interval X starting with a first crank angle A1 of the piston 20, and ending with a second crank angle A2 of the piston 20.

The third plot 104 in Figure 2 shows the maximum interval X that the cylinder assembly 19 can inject the gaseous fuel into the intake runner 16. For this maximum interval X, the first crank angle A1 corresponds to the intake valve 7i being closed and the exhaust valve 7e being open during the exhaust stroke, and the second crank angle A2 corresponds to the intake valve 7i being open and the exhaust valve 7e being closed during the intake stroke.

Advantageously, injecting gaseous fuel into the intake runner 16 over such an interval X may help ensure a sufficient volume of gaseous fuel is supplied to the cylinder 5 for near-optimal combustion using the single fuel injector 22. An extended angular interval X (and therefore an extended time proportional to engine speed) may also enhance the fuel air mixing, also promoting enhanced combustion.

In the illustrated embodiment, the first crank angle A1 and the second crank angle A2 are both a function of an engine speed and an output torque of the engine 1. As such, the interval X is a function of the engine speed and the output torque of the engine 1. The output power (P) of the engine 1 is a function of the engine speed (S) and the output torque (T) of the engine 1; i.e. $P = 2\pi TS/60$, where P is in watts, T is in newton-metres, and S is in revolutions per minute. As such, in the illustrated embodiment, the first crank angle A1, the second crank angle A2, and thus the interval X, are a function of the output power of the engine 1.

In alternative embodiments (not shown), the first crank angle A1 and the second crank angle A2, and thus the interval X, may not be a function of the engine speed and/or the output torque of the engine 1. For example, the first crank angle A1 and the second crank angle A2, and thus the interval X, may be a function of the output torque or the engine speed of the engine 1 only.

In alternative embodiments (not shown), only one of the first crank angle A1 and the second crank angle A2 may be a function of the output torque and/or engine speed of the engine 1. For example, the second crank angle A2 may be substantially constant.

Figure 3 shows an exemplary contour plot of the first crank angle A1 with respect to the engine speed and the output torque of the engine 1. Similarly, Figure 4 shows an exemplary contour plot of the second crank angle A2 with respect to the engine speed and the output torque of the engine 1. The contour plots in Figures 3 and 4 were determined from on an optimisation of the first and second crank angles A1, A2 with respect to the engine speed and the output torque of the engine 1.

With reference to Figure 3, a minimum first crank angle A1 at which the cylinder assembly 19 can inject the gaseous fuel into the intake runner 16 is at a maximum output torque of the engine 1 at a predetermined engine speed in the range of 1500 to 2600 RPM, optionally, 1800 to 2200 RPM, for example, approximately 2000 RPM. In the illustrated embodiment, the minimum first crank angle A1 is approximately 316 degrees. In alternative embodiments (not shown), the minimum first crank angle A1 may be in the range of 300 to 330 degrees, optionally, 310 to 320 degrees.

By ‘maximum output torque at a predetermined engine speed’ it is intended to mean that the output torque of the engine 1 is maximised for the given engine speed, and not necessarily that the output torque is at the maximum value capable of being produced by the engine 1 (i.e. the engine is operating at a maximum load). For example, in Figures 3 and 4, at an engine speed of 2000 RPM, the maximum output torque of the engine 1 is approximately 270 Nm. Whereas the maximum output torque capable of being produced by the engine 1 is approximately 450 Nm.

With reference to Figure 4, a maximum second crank angle A2 at which the cylinder assembly 19 can inject the gaseous fuel into the intake runner 16 is at a maximum output torque of the engine 1 at a predetermined engine speed in the range of 1200 to 1800 RPM, optionally, 1300 to 1700 RPM, for example approximately 1500 RPM.

The difference in engine speeds at the minimum first crank angle A1 and the maximum second angle A2 in the exemplary contour plots of Figures 3 and 4 is due to the optimisation, which primarily determined the second crank angles A2 that optimise for fuel efficiency and emissions, and determined the first crank angles A1 that ensure that sufficient fuel is injected into the cylinder 5 within each four-stroke working cycle of the engine 1, whilst integrity of the engine components are maintained.

In the exemplary plots shown in Figure 2, it can be seen that the intake valves 7i are open during the exhaust stroke, before the initiation of the intake stroke. As will be appreciated, during operation of the engine 1, this may result in a portion of the exhaust gases in the cylinder 5 flowing through the inlets 6 and into the intake runner 16. In diesel engines, opening the intake valves during an exhaust stroke can increase the efficiency of the engine as it reduces the pumping losses of the engine.

A problem with opening the intake valves 7i during the exhaust stroke in the engine 1 is that there is the potential for the exhaust gases flowing into the intake runner 16 to push air and fuel contained in the intake runner 16 upstream away from the inlets 6. In some cases, air and fuel may backflow into other intake runners 16 via the inlet manifold 11, causing misfiring of the corresponding cylinders 5 and/or poor combustion.

With reference to Figure 4, to help solve this problem, the maximum second crank angle A2 at which the cylinder assembly 19 can inject the gaseous fuel into the intake runner 16 is a crank angle of approximately 495 degrees after a top power dead centre of the piston 20. In alternative embodiments (not shown), the maximum second crank angle A2 may be a crank angle of less than 495 degrees after a top power dead centre of the piston 20. For example, in some embodiments, the maximum second crank angle A2 at which the cylinder assembly 19 can inject the gaseous fuel into the intake runner 16 may be a crank angle of approximately 490 degrees or less after a top power dead centre of the piston 20.

Advantageously, it has been found that providing such an upper limit on the second crank angle A2 helps to inhibit backflowing of gaseous fuel and air within the intake runner 16.

In the illustrated embodiment, a minimum second crank angle A2 at which the cylinder assembly 19 can inject the gaseous fuel into the intake runner 16 is a crank angle of approximately 440 degrees after a top power dead centre of the piston 20. In alternative embodiments (not shown), the minimum second crank angle A2 may be a crank angle of more than 440 degrees, for example 450 degrees or more, after a top power dead centre of the piston 20.

The maximum interval X between the first crank angle A1 and the second crank angle A2 that the cylinder assembly 19 can inject the gaseous fuel into the intake runner 16 is in the range of 160 to 200 degrees, optionally 170 to 190 degrees. In the illustrated embodiment, the maximum interval X is approximately 180 degrees.

Advantageously, such a maximum interval X may help to ensure a sufficient volume of gaseous fuel is supplied to the cylinder 5 for near-optimal combustion.

The maximum interval X is at a maximum output torque of the engine 1 at a predetermined engine speed, which may be in the range of 1600 to 2200 RPM, optionally in the range of 1800 to 2000 RPM.

For a constant output torque of the engine 1, the interval X increases as the engine speed of the engine 1 increases. Advantageously, this may help ensure that a sufficient volume of gaseous fuel is supplied to the cylinder 5 as the time duration of each four-stroke working cycle of the engine 1 reduces with increased engine speed.

With reference to Figure 5, the engine 1 includes an engine control unit 200 configured to determine the interval X for each fuel injector 22 within each four-stroke cycle of the engine 1, and control each fuel injector 22 to inject fuel into the corresponding intake runner 16 exclusively during the determined interval X. The engine control unit 200 may determine the interval X based on input engine speed and a torque output of the engine 1, for example using a look-up table or map stored in associated memory. The engine control unit 200 may determine the engine speed and the torque output of the engine 1 based on one or more signals received from an engine sensor arrangement 202, e.g. including one or more of an engine speed sensor, a throttle position sensor, an intake air flow rate sensor, a coolant temperature sensor, an exhaust gas oxygen concentration sensor, an exhaust manifold pressure sensor, and an intake manifold pressure sensor. Additionally or alternatively, the engine control unit 200 may determine the engine speed and/or the torque output of the engine 1 based on a signal received from a fuel pressure sensor 206 (discussed more below).

As shown in Figure 2, the fuel injector 22 is configured to inject the gaseous fuel continuously between the first crank angle A1 and the second crank angle A2. Advantageously, such a configuration of the fuel injector 22 may help to ensure a sufficient volume of gaseous fuel is supplied to the cylinder 5 for near-optimal combustion. In alternative embodiments (not shown), the fuel injector 22 may not be configured to inject the gaseous fuel continuously between the first crank angle A1 and the second crank angle A2. For example, the fuel injector 22 may be configured to inject the gaseous fuel as a series of pulsed injections within the interval X.

As shown in Figure 2, the fuel injector 22 is configured to inject the gaseous fuel at a substantially constant injection rate between the first crank angle A1 and the second crank angle A2. In the exemplary embodiment, the fuel injector 22 injects the gaseous fuel at an injection rate in the range of 0.0005 to 0.0025 kilograms per second. Advantageously, such a configuration of the fuel injector 22 may help to simplify implementation of the engine 1.

In alternative embodiments (not shown), the fuel injector 22 may be configured to inject the gaseous fuel at a varying injection rate between the first crank angle A1 and the second crank angle A2.

In the illustrated embodiment, gaseous fuel is injected into the intake runner 16 exclusively via the fuel injector 22.

In alternative embodiments (not shown), gaseous fuel may be injected into the intake runner 16 via a plurality of the fuel injectors 22, or via the fuel injector 22 in addition to any other suitable means.

Within a four-stroke cycle of the engine 1, each intake valve 7i moves from a closed position to an open position at a crank angle of the piston 20 in the range of 310 to 350 degrees, optionally in the range of 320 to 340 degrees, after a top power dead centre of the piston 20. In the illustrated embodiment, each intake valve 7i moves from a closed position to an open position at a crank angle of the piston 20 of approximately 335 degrees after a top power dead centre of the piston 20.

Within a four-stroke cycle of the engine 1, each intake valve 7i moves from an open position to a closed position at a crank angle of the piston 20 in the range of 570 to 610 degrees, optionally in the range of 580 to 600 degrees, after a top power dead centre (TPDC) of the piston 20. In the illustrated embodiment, each intake valve 7i moves from an open position to a closed position at a crank angle of the piston 20 of approximately 585 degrees after a top power dead centre of the piston 20.

Within a four-stroke cycle of the engine 1, each exhaust valve 7e moves from an open position to a closed position at a crank angle of the piston 20 in the range of 350 to 390 degrees, optionally in the range of 360 to 380 degrees, after a top power dead centre of the piston 20. In the illustrated embodiment, each exhaust valve 7e moves from an open position to a closed position at a crank angle of the piston 20 of approximately 360 degrees after a top power dead centre of the piston 20. In alternative embodiments (not shown), each exhaust valve 7e may move from an open position to a closed position at a crank angle of the piston 20 of approximately 375 degrees after a top power dead centre of the piston 20.

Within a four-stroke cycle of the engine 1, each exhaust valve 7e moves from a closed position to an open position at a crank angle of the piston 20 in the range of 110 to 150 degrees, optionally 120 to 140 degrees, after a top power dead centre of the piston 20. In the illustrated embodiment, each exhaust valve 7e moves from a closed position to an open position at a crank angle of the piston 20 of approximately 130 degrees after a top power dead centre of the piston 20.

Although not shown in the figures, in some embodiments, the engine 1 may be configured such that, within a four-stroke cycle of the engine 1:

- the exhaust valve 7e moves from a closed to an open position at a crank angle of the piston 20 approximately 140 degrees after a top power dead centre of the piston 20;
- the exhaust valve 7e moves from an open to a closed position at a crank angle of the piston 20 of approximately 380 degrees after a top power dead centre of the piston 20;
- the intake valve 7i moves from a closed to an open position at a crank angle of the piston 20 of approximately 335 degrees after a top power dead centre of the piston 20; and
- the intake valve 7i moves from an open to a closed position at a crank angle of the piston 20 of approximately 580 degrees after a top power dead centre of the piston 20.

Advantageously, such a configuration of the engine 1 has been found to minimise backflow of gaseous fuel and air in the intake runner 16, which may result in an improved volumetric efficiency of the engine 1.

As shown in Figure 2, within a four-stroke cycle of the engine 1, movement of each of the exhaust valve 7e and the intake valve 7i from a closed position to a maximum open position, and from the maximum open position to the closed position, is substantially symmetric with respect to the varying crank angle of the piston 20.

In alternative embodiments (not shown), movement of one or both of the exhaust valve 7e and the intake valve 7i from a closed position to a maximum open position, and from the maximum open position to the closed position may be asymmetric with respect to the varying crank angle of the piston 20.

With reference to Figures 1B and 1C, each cylinder 5 has a flat roof 30. In the illustrated embodiment, the flat roof 30 includes a face of the cylinder head 3 facing the cylinder 5. The roof 30 and the piston 20 define a combustion chamber therebetween within the cylinder 5 when the piston 20 is at the top power dead centre position. The flat roof 30 is arranged substantially normal to the cylinder axis C. The intake valves 7i and the exhaust valves 7e are arranged to open and close via movement along respective axes that are parallel to the cylinder axis C. In the illustrated embodiment, the exhaust valves 7e and the intake valves 7i extend through the flat roof 30. Such arrangements are typical of diesel engines, so retaining this configuration may also allow for a common cylinder head, and certain valve train parts such as pushrods and rocker arms.

In alternative embodiments (not shown), the cylinders 5 may have e.g. a pent-roof or hemispheric roof and the valves may have axes that are inclined with respect to the axis C.

With reference to Figures 1B and 1C, the cylinder assembly 19 is configured to inject the gaseous fuel into the intake runner 16 via the fuel injector 22 at an injection angle to a mean flow direction F of air flowing through the intake runner 16, in use, and/or the horizontal, in use, in the range of 0 to 25 degrees, optionally in the range of 10 to 20 degrees, for example 14 degrees.

Advantageously, such injection angles have been found to provide good mixing between fuel and air, and to inhibit backflow of fuel entering the intake runner 16, in use.

In the illustrated embodiment, the fuel injector 22 is received in a bore 40 in the intake assembly 10. The bore 40 is in fluid communication with a fuel injection passage 42 in the intake assembly 10. The fuel injection passage 42 includes a fuel orifice 44 within the intake runner 16. Fuel injected from the fuel injector 22 travels along the fuel injection passage 42 and into the intake runner 16 via the fuel orifice 44. The fuel injection passage 42 is configured to direct fuel exiting the fuel orifice 44 substantially along an injection axis A.

In alternative embodiments (not shown), fuel may be injected into the intake runner 16 from the fuel injector 22 via any suitable arrangement. In some embodiments, the fuel injector 22 may be arranged in the cylinder head 3.

The cylinder assembly 19 is configured to direct the injected gaseous fuel from the fuel injector 22 along the injection axis A in a direction substantially towards the outlet portion

16o of the intake runner 16. The intake port 4 has a width W along an axis of the intake valve 7i closest to the intake runner 16. The injection axis A intersects said axis of the intake valve 7i at a position greater than 50%, optionally greater than 60%, optionally greater than 70%, of the width W of the intake port 4 from the corresponding inlet 6.

Advantageously, such configurations of the injection port 4 and the injection axis A have been found to provide good mixing between fuel and air in the intake port 4, and to inhibit backflow of fuel entering the intake runner 16 via the fuel injector 22, in use. Further, by having an injection axis orientated in this way, it may be less likely for gaseous fuel, particularly hydrogen that has a low autoignition temperature (around 500°C), to come into contact with hot exhaust gases, combust outside the cylinder and cause a backfire event.

During operation of the engine 1, the fuel pressure in the fuel tank 51 will decrease as the gaseous fuel stored in the fuel tank 51 is supplied to the fuel injectors 22. As previously discussed, the pressure regulator arrangement 55 maintains a substantially constant fuel pressure Pr in the fuel supply conduit 52, and thus maintains the flow rate of the fuel injectors 22, irrespective of the fuel pressure in the fuel tank 51. However, occasionally, the fuel pressure Pr in the fuel supply conduit 52 may temporarily drop due to, for example, a sudden increase in operator engine power demand, a temporary blockage in the supply line 53 (e.g. in a fuel filter), and/or a temporary fault in the pressure regulator arrangement 55. Such a temporary drop in the fuel pressure Pr results in a corresponding reduction in the possible flow rate of the fuel injectors 22. To ensure enough fuel is injected into the intake runners 16 to provide the operator demanded engine power output, the interval X needs to be increased. However, increasing the second crank angle A2 beyond the maximum second crank angle A2 (e.g. 495 degrees after a top power dead centre of the piston) may lead to the problem of backflow of gaseous fuel and air, previously discussed. Moreover, the minimum first crank angle A1 may be based on a computational scheduling of the engine control unit 200 (i.e. a fixed crank angle within a four stroke cycle at which the engine control unit 200 determines the interval X), and/or a computational speed of the engine control unit 200 (i.e. how quickly the engine control unit 200 can respond to operator demand), and therefore it may not be possible to reduce the first crank angle A1 below this minimum.

As discussed more below, to inhibit the engine 1 running too lean as a result of a temporary drop in the fuel pressure Pr, which may result in engine damage and/or abnormal combustion, the engine control unit 200 is configured to de-rate the engine 1 by reducing the flow rate of air supplied to the intake runners 16.

With reference to Figures 5 and 6, the engine control unit 200 determines the fuel pressure Pr in the fuel supply conduit 52 based on a signal received from a suitable fuel pressure sensor 206 (e.g. mounted to the fuel supply conduit 52).

In the illustrated embodiment, the engine control unit 200 automatically controls a flow rate of air supplied to each intake runner 16 by operating the throttle 17. In alternative embodiments, the engine control unit 200 may additionally or alternatively control the flow rate of air supplied to each intake runner 16 via the intake assembly 10 by any suitable means (e.g. by operating one or more further valves in the intake assembly 10).

In an exemplary embodiment, the engine control unit 200 is configured to de-rate the engine 1 via the following method, which is represented as a flowchart in Figure 6. In the following, the method is carried out for each cylinder assembly 19, and thus refers to, for example, "the fuel injector 22".

In step S1, the engine control unit 200 determines the fuel pressure Pr in the fuel supply conduit 52 via the fuel pressure sensor 206. Moreover, the engine control unit 200 determines an output power demand of the engine 1 (e.g. determined based on an input from a control input device such as an accelerator pedal).

In subsequent step S2, the engine control unit 200 determines a first interval X1 based on the determined fuel pressure Pr and output power demand of the engine 1. The first interval X1 may be a function of an output torque and/or an engine speed of the engine 1, as described above in relation to Figures 3 and 4.

In subsequent step S3, the engine control unit 200 compares the determined first interval X1 to a predetermined maximum interval. In some embodiments, the maximum interval may be determined based on a minimum first crank angle A1 and a maximum second crank angle A2 as previously discussed. If the first interval X1 is less than or equal to the maximum interval, the method proceeds to step S4. Otherwise, the method proceeds to step S5.

In step S4, in response to the first interval X1 being less than or equal to the predetermined maximum interval, the engine control unit 200 controls the fuel injector 22 based on the first interval X1. The engine control unit 200 may control the throttle 17 based on the first interval X1. Subsequently, the method returns to step S1.

In step S5, in response to the first interval X1 being greater than the predetermined maximum interval, the engine control unit 200 determines a second interval X2 and a de-rate throttle position. The de-rate throttle position is determined by the engine control unit 200 so as to reduce the flow rate of air being supplied to the intake runners 16, so as to prevent the air-fuel mixture from becoming too lean. The second interval X2 and/or the de-rate throttle position may be a predetermined (e.g. user input) parameter, or may be dynamically determined based on one or more engine operating parameters (e.g. the fuel pressure Pr and/or the output power demand of the engine 1). The second interval X2 is less than or equal to the predetermined maximum interval.

In subsequent step S6, the engine control unit 200 controls the throttle 17 based on the determined de-rate throttle position, and controls the fuel injector 22 based on the second interval X2. Subsequently, the method returns to step S1.

Advantageously, the foregoing method helps to prevent the fuel mixture becoming too lean, and thus helps inhibit damage to the engine and/or abnormal combustion, whilst helping to ensure that the first crank angle A1 does not drop below the minimum, and the second crank angle A2 does not exceed the maximum.

In some embodiments, in response to the fuel pressure Pr in the fuel supply conduit 52 being less than a predetermined pressure threshold for a predetermined period of time or more, the engine control unit 200 is configured to output an alarm signal and/or maintain a reduced flow rate of air being supplied to the intake runners 16, e.g. irrespective of the fuel pressure in the fuel supply conduit 52 and/or the determined first interval X1. For example, an audible and/or visual alarm may be activated (e.g. in a vehicle cab) based on the alarm signal, informing a user that there is an engine fault so that they may take mitigating action. Advantageously, such a configuration helps to prevent damage to the engine 1 when there is a persistent fault affecting the fuel pressure Pr.

With further reference to Figure 7, each cylinder assembly 19 includes an exhaust port 8 including an upstream end 8u leading away from the outlets 9. Each exhaust port 8 is arranged to transport exhaust gases away from the corresponding cylinder 5 towards a downstream portion 8d of the exhaust port 8. The upstream end 8u of the exhaust port 8 bifurcates to the outlets 9. Put another way, the upstream end 8u of the exhaust port 8 bifurcates into two branches, each branch leading to one of the outlets 9. In other embodiments (not shown) a non-bifurcating exhaust port 8 and a single outlet 9 may however be utilised.

The downstream end 4d of the intake port 4 bifurcates to the inlets 6. Put another way, the downstream end 4d of the intake port 4 bifurcates into two branches, each branch leading to one of the inlets 6. One branch is longer than the other so that the ports have a so-called "tandem" configuration. In this configuration the ports are arranged transverse to an axis defined by a crankshaft of the engine. The outlet ports are similarly arranged.

Such an arrangement is typically used to generate a swirl motion of air in the cylinder that is desirable for efficient compression ignition combustion in diesel engines.

Spark ignition engines typically use a cross-flow configuration to generate a desired tumble motion in the cylinder for efficient combustion. It has however been found that the combination of port fuel injection of hydrogen, the valve and injection timing described above and this tandem configuration efficient combustion can nevertheless be achieved

In other embodiments (not shown) a non-bifurcating intake port 4 and single inlet 6 may however be utilised.

As described above it is desirable to utilise common parts from pre-existing diesel engines for hydrogen fuelled engines to minimise tooling costs and inventory costs, as well as costs associated with repackaging an IC engine into a particular application. The use of common parts may similarly retain certain characteristics of diesel engines that are advantageous for off-highway and/or heavy commercial vehicle applications, in particular high and broad torque band starting at a low engine speed.

CLAIMS

1. A four-stroke gaseous fuel engine comprising one or more cylinder assemblies, each cylinder assembly comprising:

a cylinder including an inlet and an outlet selectively opened and closed by an intake valve and an exhaust valve respectively;

a piston translationally movable within the cylinder;

an intake runner leading to the inlet; and

a fuel injector, wherein the cylinder assembly is configured to selectively inject a gaseous fuel from the fuel injector into the intake runner,

wherein the engine is configured to supply gaseous fuel and air from the intake runner to the cylinder via the inlet during an intake stroke of the piston, and exhaust combustion gases from the cylinder via the outlet during an exhaust stroke of the piston,

wherein, within a four-stroke cycle of the engine, the cylinder assembly is configured to inject the gaseous fuel into the intake runner exclusively during an interval starting with a first crank angle of the piston and ending with a second crank angle of the piston, and

wherein at a maximum interval that the cylinder assembly can inject the gaseous fuel into the intake runner, the first crank angle corresponds to the intake valve being closed and the exhaust valve being open during the exhaust stroke, and the second crank angle corresponds to the intake valve being open and the exhaust valve being closed during the intake stroke.

2. The engine of claim 1, wherein a maximum second crank angle at which the cylinder assembly can inject the gaseous fuel into the intake runner is a crank angle of 495 degrees or less after a top power dead centre of the piston; optionally, 490 degrees or less after a top power dead centre of the piston.

3. The engine of claim 1 or claim 2 wherein a minimum second crank angle at which the cylinder assembly can inject the gaseous fuel into the intake runner is a crank angle of 440 degrees or more, optionally 450 degrees or more, after a top power dead centre of the piston.

4. The engine of any preceding claim, wherein the gaseous fuel is hydrogen.

5. The engine of any preceding claim, wherein the maximum interval is in the range of 160 to 200 degrees, optionally 170 to 190 degrees, for example approximately 180 degrees.

6. The engine of any preceding claim, wherein the interval is a function of an output torque and/or an engine speed of the engine.

7. The engine of claim 6, wherein the maximum interval is at a maximum output torque at a predetermined engine speed; optionally, in the range of 1600 to 2200 RPM, optionally in the range of 1800 to 2000 RPM.

8. The engine of claims 6 or 7, wherein a minimum first crank angle at which the cylinder assembly can inject the gaseous fuel into the intake runner is at a maximum output torque at a predetermined engine speed in the range of 1500 to 2600 RPM, optionally, 1800 to 2200 RPM, for example, approximately 2000 RPM.

9. The engine of any one of claims 6 to 8, wherein a maximum second crank angle at which the cylinder assembly can inject the gaseous fuel into the intake runner is at a maximum output torque at a predetermined engine speed in the range of 1200 to 1800 RPM, optionally 1300 to 1700 RPM, for example approximately 1500 RPM.

10. The engine of any one of claims 6 to 9, wherein, for a constant output torque of the engine, the interval increases as the engine speed increases.

11. The engine of any preceding claim, wherein the fuel injector is configured to inject the gaseous fuel continuously between the first and second crank angles.

12. The engine of claim 11, wherein the fuel injector is configured to inject the gaseous fuel at a substantially constant injection rate between the first and second crank angles.

13. The engine of any preceding claim, wherein gaseous fuel is injected into the intake runner exclusively via the fuel injector.

14. The engine of any preceding claim, wherein, within a four-stroke cycle, the intake valve moves from a closed position to an open position at a crank angle of the piston in the range of 310 to 350 degrees, optionally in the range of 320 to 340 degrees, for example approximately 335 degrees, after a top power dead centre of the piston.

15. The engine of any preceding claim, wherein, within a four-stroke cycle, the intake valve moves from an open position to a closed position at a crank angle of the piston in the range of 570 to 610 degrees, optionally in the range of 580 to 600 degrees, for example approximately 585 degrees, after a top power dead centre of the piston.

16. The engine of any preceding claim, wherein, within a four-stroke cycle, the exhaust valve moves from an open position to a closed position at a crank angle of the piston in the range of 350 to 390 degrees, optionally in the range of 360 to 380 degrees, for example approximately 375 degrees, after a top power dead centre of the piston.

17. The engine of any preceding claim, wherein, within a four-stroke cycle, the exhaust valve moves from a closed position to an open position at a crank angle of the piston in the range of 110 to 150 degrees, optionally in the range of 120 to 140 degrees, for example approximately 130 degrees, after a top power dead centre of the piston.

18. The engine of any preceding claim, wherein, within a four-stroke cycle:

the exhaust valve moves from a closed to an open position at a crank angle of the piston of approximately 140 degrees after a top power dead centre of the piston;

the exhaust valve moves from an open to a closed position at a crank angle of the piston of approximately 380 degrees after a top power dead centre of the piston;

the intake valve moves from a closed to an open position at a crank angle of the piston of approximately 335 degrees after a top power dead centre of the piston; and

the intake valve moves from an open to a closed position at a crank angle of the piston of approximately 580 degrees after a top power dead centre of the piston.

19. The engine of any preceding claim, wherein the cylinder comprises a flat roof arranged substantially normal to an axis of piston motion, and wherein the intake valve and the exhaust valve are arranged to open and close via movement along respective axes that are parallel to said axis of piston motion.

20. The engine of any preceding claim, wherein the cylinder assembly is configured to inject the gaseous fuel into the intake runner from the fuel injector at an injection angle to a mean flow direction of air flowing through the intake runner and/or the horizontal, in use, in the range of 0 to 25 degrees, optionally in the range of 10 to 20 degrees, for example 14 degrees.

21. The engine of any preceding claim, wherein each cylinder assembly comprises an intake port leading to the inlet, wherein an outlet portion of the intake runner leads to the

intake port, wherein the cylinder assembly is configured to direct the injected gaseous fuel from the fuel injector along an injection axis in a direction substantially towards the outlet portion, wherein the intake port has a width along an axis of the intake valve, and wherein the injection axis intersects the axis of the intake valve at a position greater than 50%, optionally greater than 60%, optionally greater than 70%, of the width of the intake port from the inlet.

22. The engine of any preceding claim, wherein the cylinder comprises a further inlet and a further outlet selectively opened and closed by a further intake valve and a further exhaust valve respectively, wherein the intake runner leads to the further inlet, and wherein the engine is configured to supply gaseous fuel and air from the intake runner to the cylinder via the further inlet during the intake stroke of the piston, and exhaust combustion gases from the cylinder via the further outlet during the exhaust stroke of the piston.

23. The engine of claim 22, further comprising:

an intake port including a downstream end leading to the inlet and the further inlet; and

an exhaust port including an upstream end leading from the outlet and the further outlet,

wherein the intake runner leads to an upstream end of the intake port, the downstream end of the intake port bifurcating to the inlet and the further inlet, and

wherein the ports are arranged in a tandem configuration.

24. A four-stroke gaseous fuel engine comprising:

one or more cylinder assemblies;

an intake assembly;

a fuel tank for storing a gaseous fuel;

a fuel supply conduit, such as a fuel rail; and

a control unit,

wherein each cylinder assembly comprises:

a cylinder including an inlet selectively opened and closed by an intake valve;

a piston translationally movable within the cylinder;

an intake runner leading to the inlet; and

a fuel injector configured to selectively inject a gaseous fuel into the intake runner, wherein the engine is configured to supply gaseous fuel and air from the intake runner to the cylinder via the inlet,

wherein the engine is configured to supply a gaseous fuel stored in the fuel tank to the fuel supply conduit, such that a fuel pressure in the fuel supply conduit is independent of a fuel pressure in the fuel tank,

wherein the engine is configured to supply the gaseous fuel from the fuel supply conduit to each fuel injector, such that a fuel injection flow rate of each fuel injector is dependent on the fuel pressure in the fuel supply conduit,

wherein the intake assembly is configured to supply air to each intake runner,

wherein the control unit is configured to control a flow rate of air supplied to each intake runner via the intake assembly,

wherein, within a four-stroke cycle of the engine, the control unit is configured to control each fuel injector to inject the gaseous fuel into the intake runner exclusively during an interval starting with a first crank angle of the piston and ending with a second crank angle of the piston, and

wherein, for each cylinder assembly, the control unit is configured to:

determine a first interval based on the fuel pressure in the fuel supply conduit; and

in response to the first interval being greater than a predetermined maximum interval, reduce the flow rate of air supplied to the intake runner, and control the fuel injector based on a second interval less than or equal to the predetermined maximum interval.

25. The engine of claim 24, wherein the intake assembly comprises a throttle, and wherein the control unit is configured to control the flow rate of air supplied to each intake runner via the intake assembly by operating the throttle.

26. The engine of claims 24 or 25, wherein the control unit is configured to determine the first interval further based on an output power demand of the engine.

27. The engine of any one of claims 24 to 26, wherein, in response to the fuel pressure in the fuel supply conduit being less than a predetermined pressure threshold for a predetermined period of time or more, the control unit is configured to output an alarm signal and/or maintain a reduced flow rate of air supplied to each intake runner irrespective of the determined first interval.

28. A working machine comprising the engine of any preceding claim.

29. A method for operating a four-stroke gaseous fuel engine configured to supply gaseous fuel and air from an intake runner to a cylinder via an inlet of the cylinder during an intake stroke of a piston within the cylinder, and exhaust gases from the cylinder via an outlet of the cylinder during an exhaust stroke of the piston, the inlet selectively opened and closed by an intake valve, and the outlet selectively opened and closed by an outlet valve, the method comprising the step of:

within a four-stroke cycle of the engine, injecting a gaseous fuel into the intake runner exclusively during an interval starting with a first crank angle of the piston and ending with a second crank angle of the piston,

wherein at a maximum interval that the cylinder assembly can inject the gaseous fuel into the intake runner, the first crank angle corresponds to the intake valve being closed and the exhaust valve being open during the exhaust stroke, and the second crank angle corresponds to the intake valve being open and the exhaust valve being closed during the intake stroke.

30. A method for operating a four-stroke gaseous fuel engine configured to supply gaseous fuel and air from an intake runner to a cylinder via an inlet of the cylinder, the inlet selectively opened and closed by an intake valve, the engine comprising:

a fuel tank for storing a gaseous fuel;

a fuel supply conduit, such as a fuel rail; and

a fuel injector configured to selectively inject a gaseous fuel into the intake runner, the method comprising the steps of:

supplying a gaseous fuel stored in the fuel tank to the fuel supply conduit, such that a fuel pressure in the fuel supply conduit is independent of a fuel pressure in the fuel tank;

supplying the gaseous fuel from the fuel supply conduit to the intake runner via the fuel injector, such that a fuel injection flow rate of the fuel injector is dependent on the fuel pressure in the fuel supply conduit;

within a four-stroke cycle of the engine, injecting a gaseous fuel into the intake runner exclusively during an interval starting with a first crank angle of a piston within the cylinder and ending with a second crank angle of the piston;

determining a first interval based on the fuel pressure in the fuel supply conduit; and

reducing a flow rate of air supplied to the intake runner in response to the first interval being greater than a predetermined maximum interval.

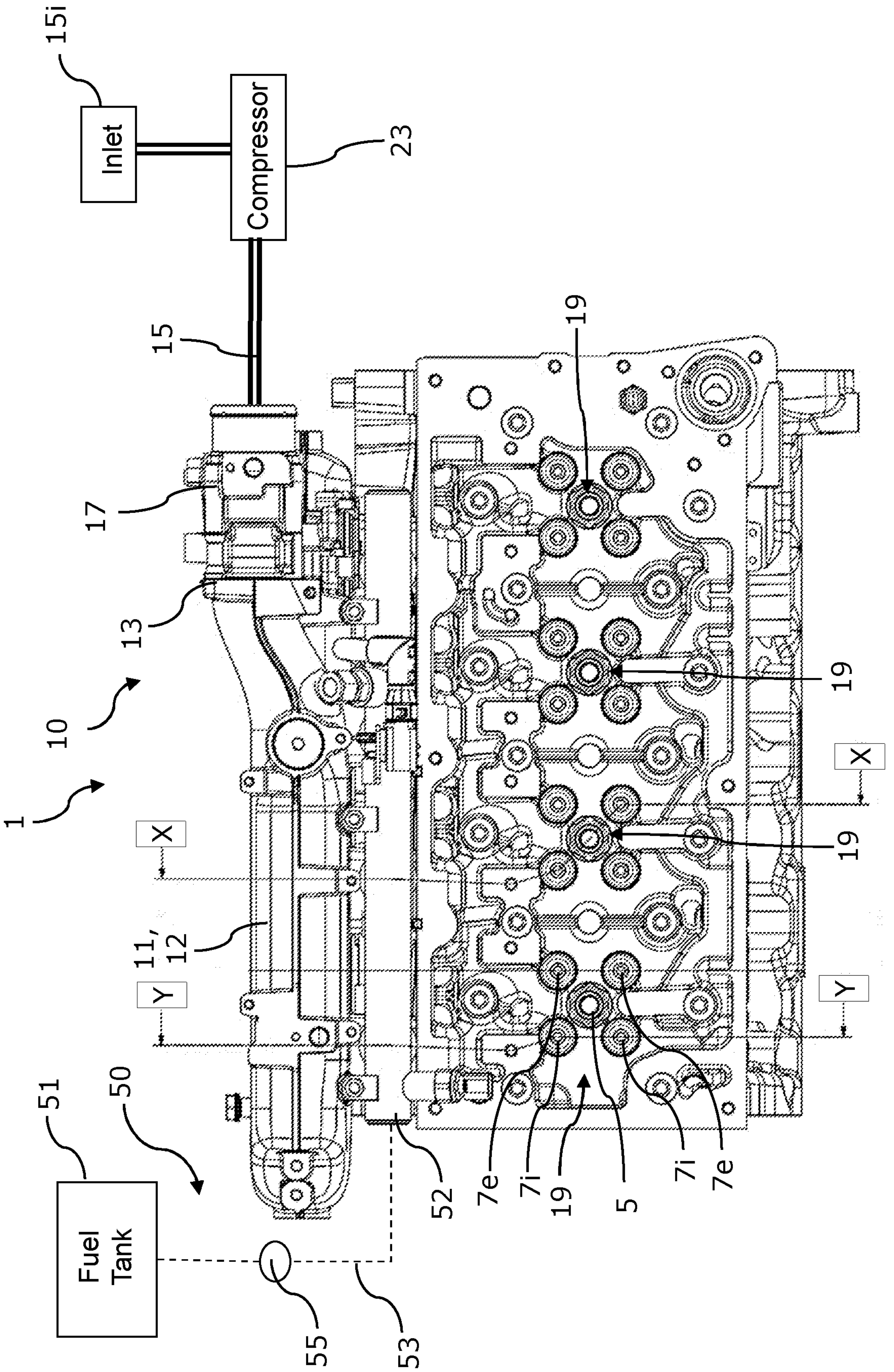
15i
Inlet
Compressor
23
15
17
13
10
1
X
11,
12
Y
19
19
19
X
Y
51
50
Fuel
Tank
52
7e
7i
19
5
7i
7e
55
53

Figure 1A

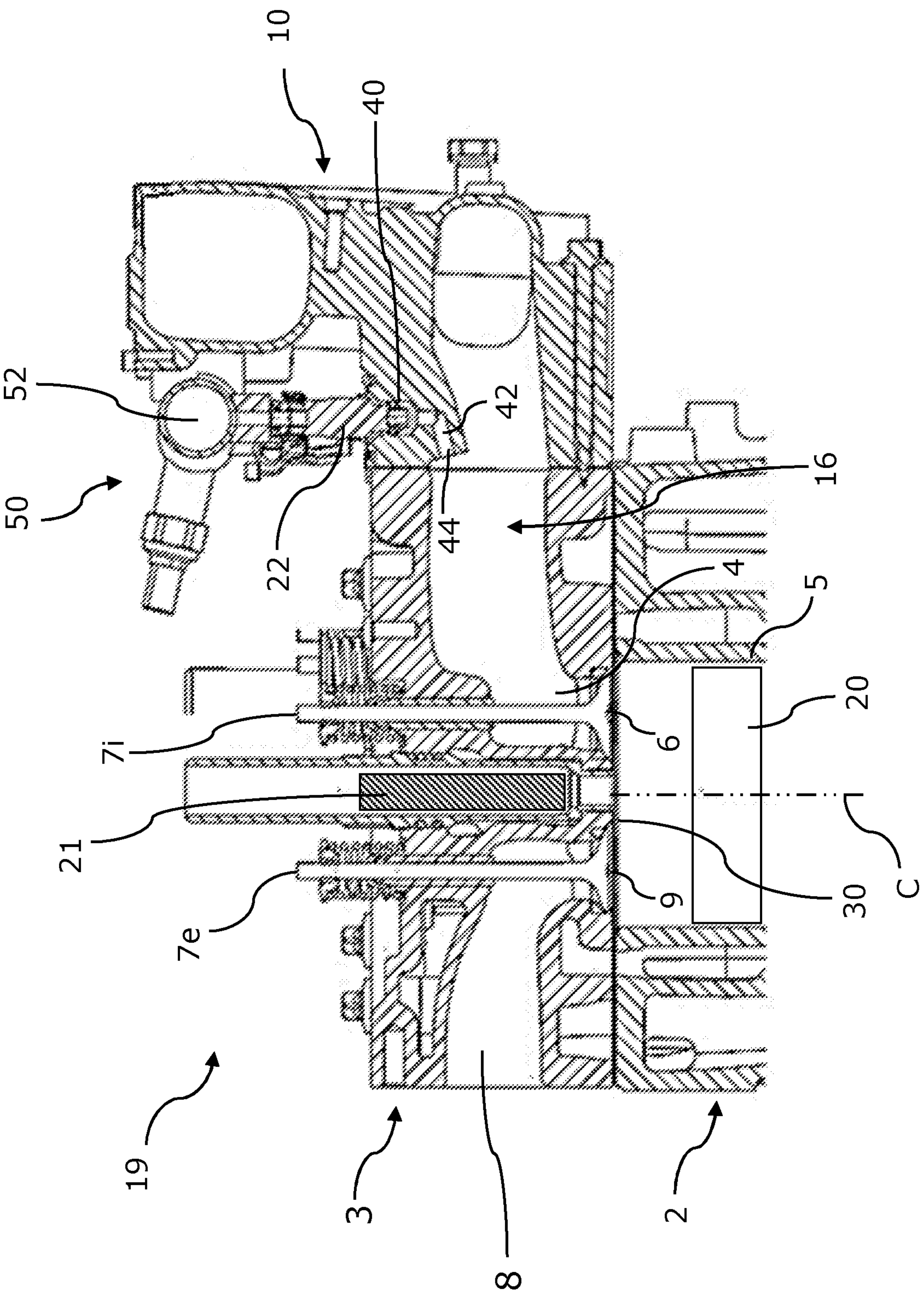

Figure 1B

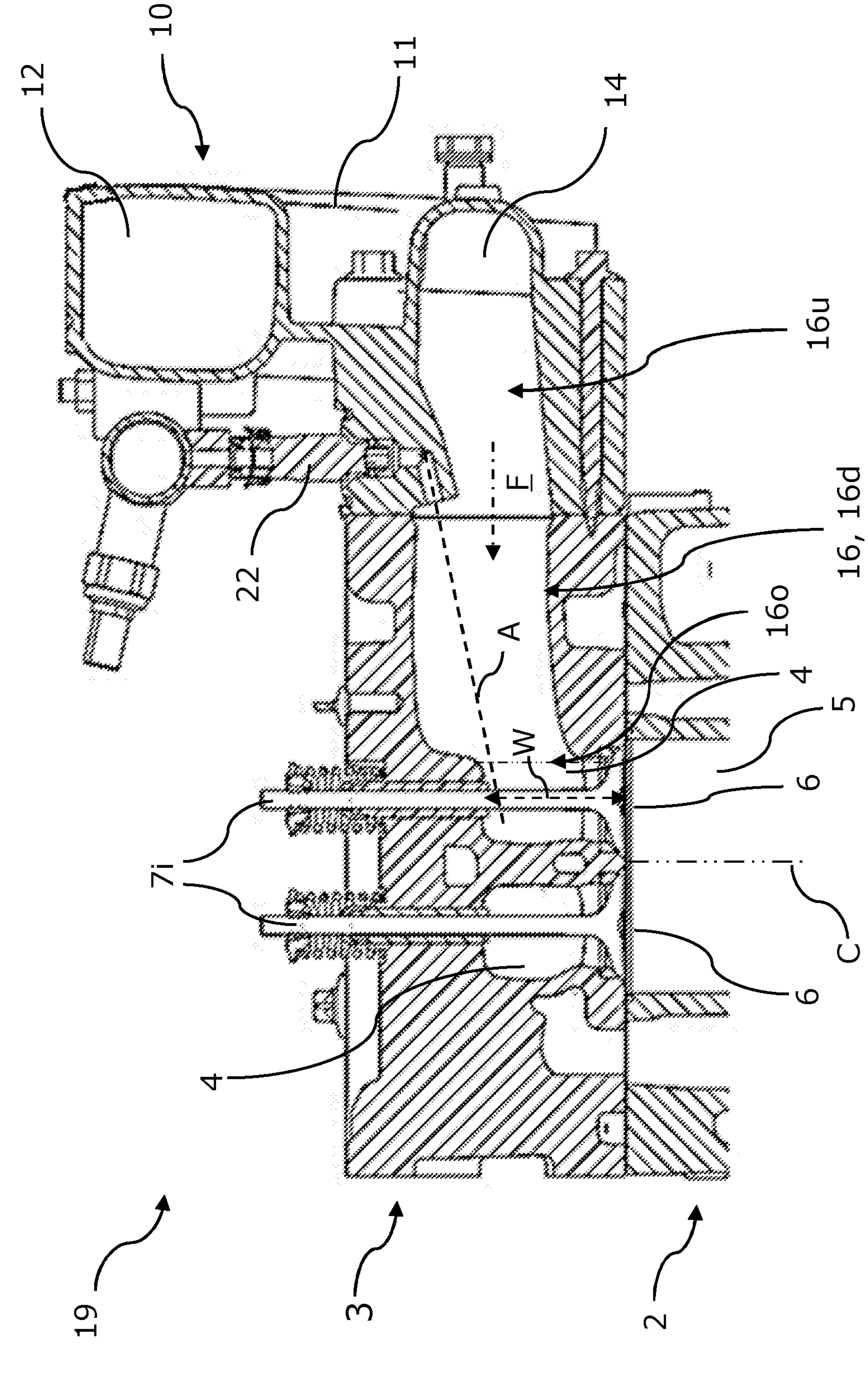

Figure 1C

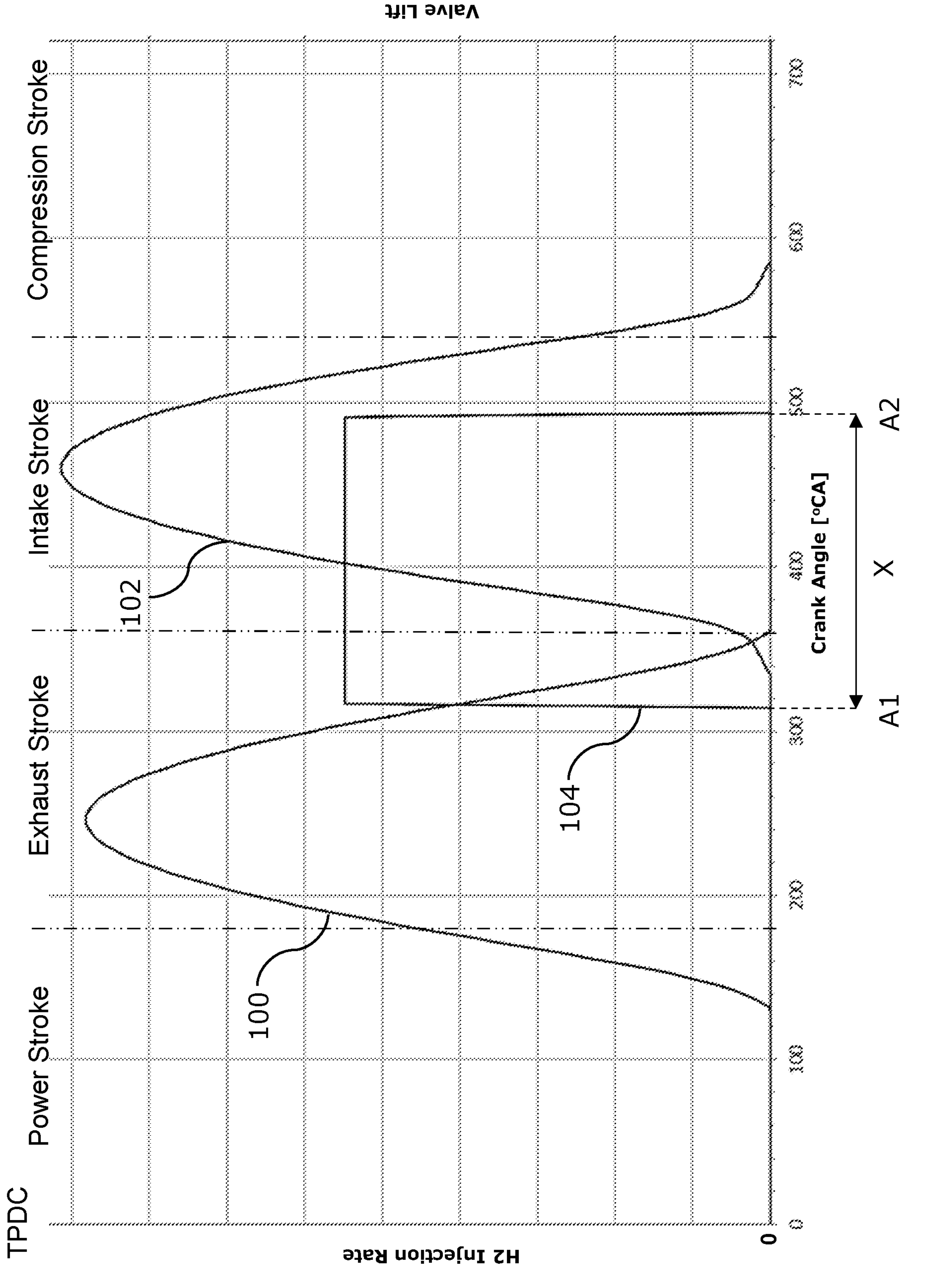
TPDC
Power Stroke
Exhaust Stroke
Intake Stroke
Compression Stroke
Valve Lift
H2 Injection Rate
0
Crank Angle [°CA]
0
100
200
300
400
500
600
700
100
102
104
A1
A2
X

Figure 2

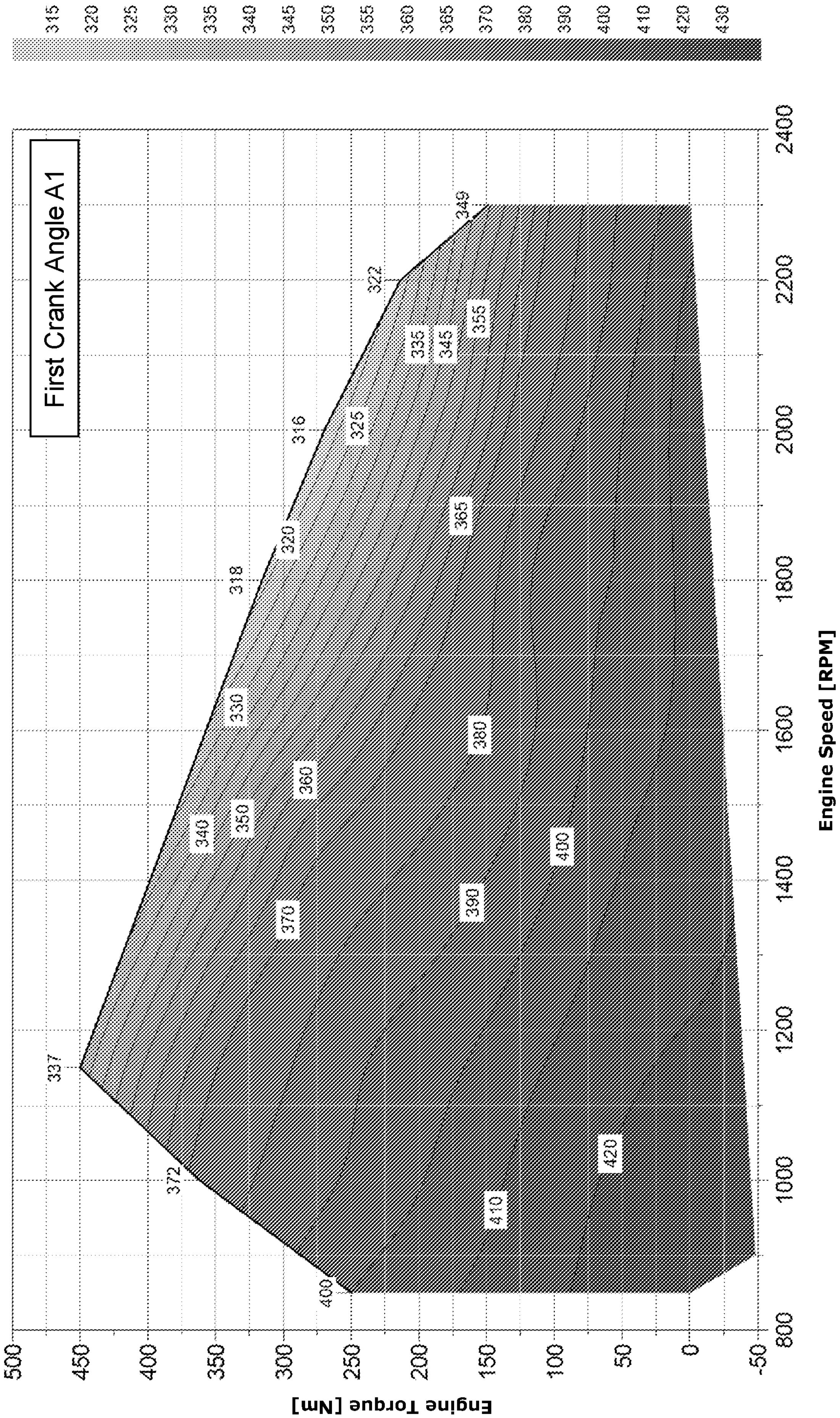
First Crank Angle A1
Engine Speed [RPM]
Engine Torque [Nm]
800 1000 1200 1400 1600 1800 2000 2200 2400
500 450 400 350 300 250 200 150 100 50 0 -50
315 320 325 330 335 340 345 350 355 360 365 370 380 390 400 410 420 430
400 372 337 318 316 322 349
320 325 330 335 340 345 350 355 360 365 370 380 390 400 410 420

Figure 3

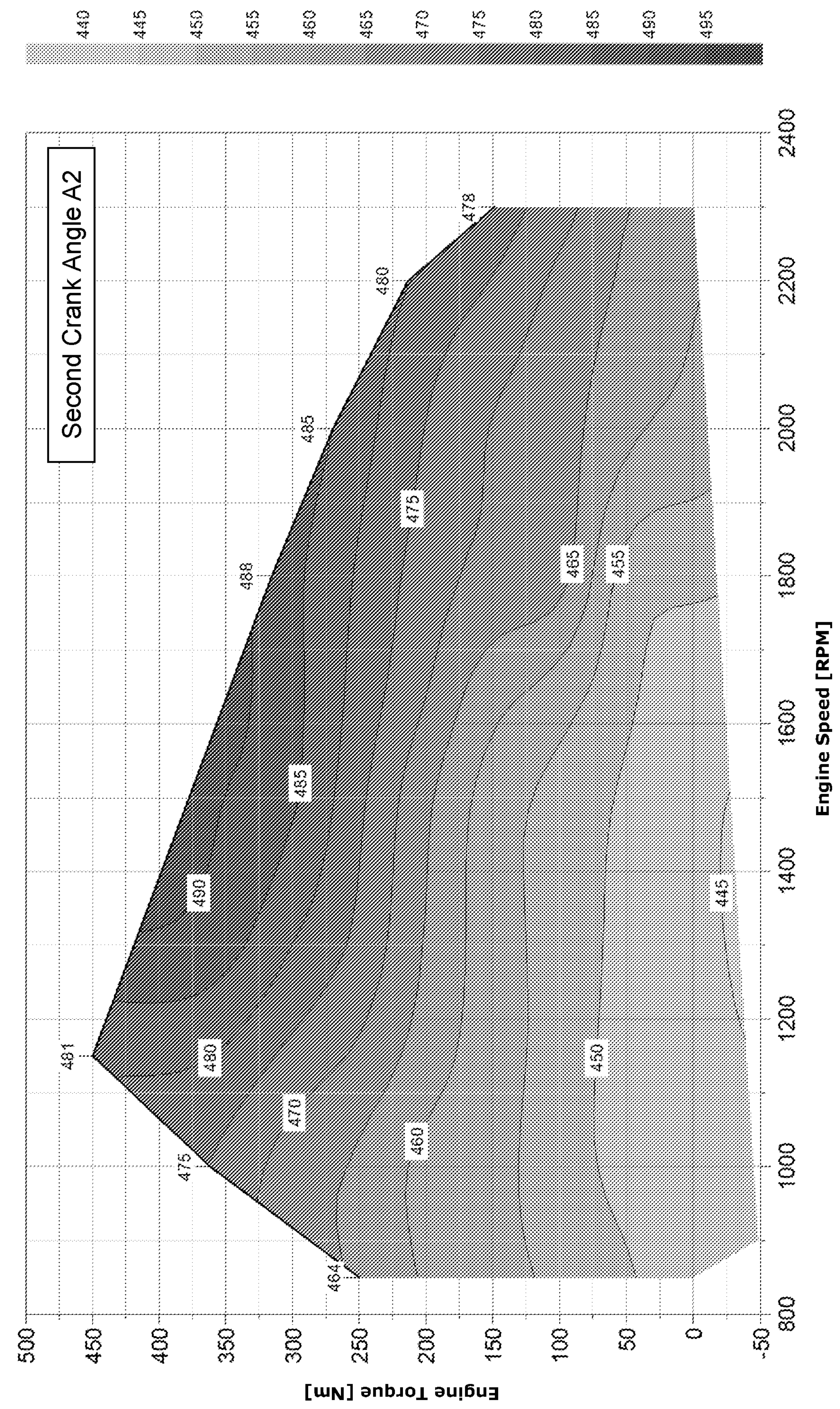
Second Crank Angle A2
Engine Torque [Nm]
Engine Speed [RPM]
500
450
400
350
300
250
200
150
100
50
0
-50
800
1000
1200
1400
1600
1800
2000
2200
2400
440
445
450
455
460
465
470
475
480
485
490
495
464
475
481
488
485
480
478
490
485
480
470
460
475
465
455
450
445

Figure 4

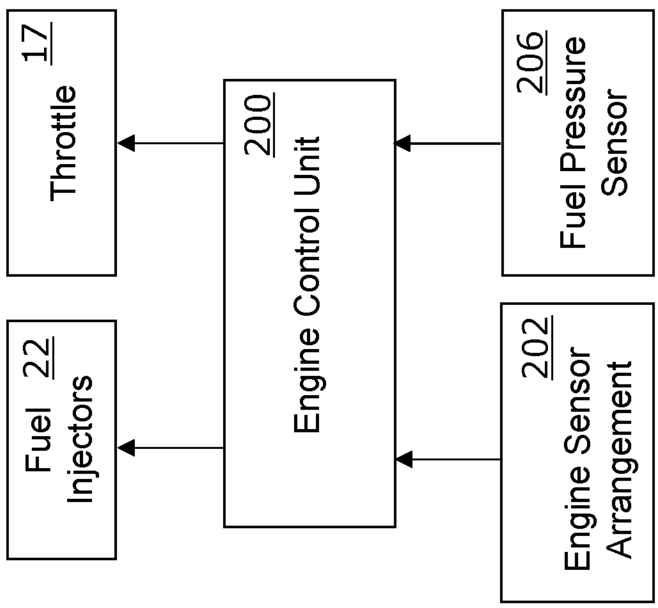
Fuel Injectors 22
Throttle 17
200
Engine Control Unit
202
Engine Sensor Arrangement
206
Fuel Pressure Sensor

Figure 5

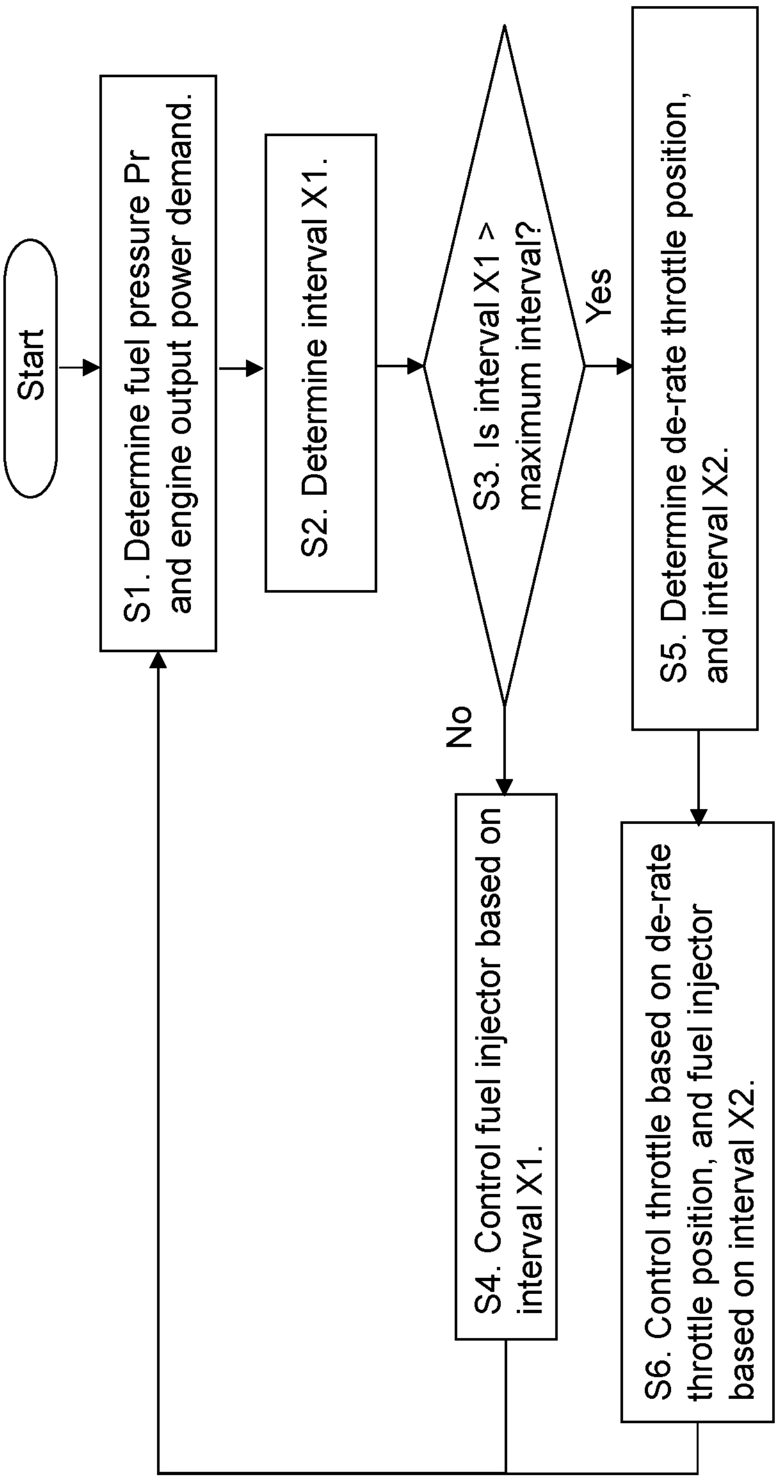
Start
S1. Determine fuel pressure Pr and engine output power demand.
S2. Determine interval X1.
S3. Is interval X1 > maximum interval?
Yes
No
S4. Control fuel injector based on interval X1.
S5. Determine de-rate throttle position, and interval X2.
S6. Control throttle based on de-rate throttle position, and fuel injector based on interval X2.

Figure 6

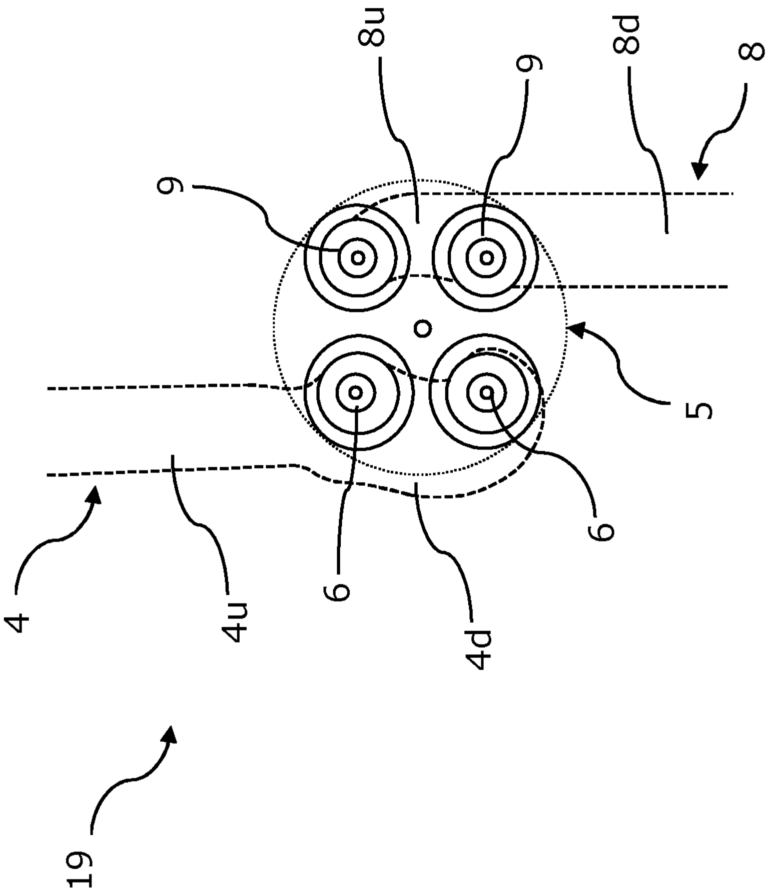

Figure 7

INTERNATIONAL SEARCH REPORT

International application No
PCT/GB2023/052815

A. CLASSIFICATION OF SUBJECT MATTER
INV. F02D41/00 F02D41/34
ADD.

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
F02D

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

EPO-Internal

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X Y	US 2010/318284 A1 (SURNILLA GOPICHANDRA [US] ET AL) 16 December 2010 (2010-12-16) abstract figures 1, 2, 2B claim 1 paragraph [0018] paragraph [0024] -----	1-18, 24-30 19-23
Y	WO 2014/093030 A1 (CATERPILLAR INC [US]) 19 June 2014 (2014-06-19) abstract figure 2 -----	19
Y	US 6 609 499 B2 (FORD GLOBAL TECH LLC [US]) 26 August 2003 (2003-08-26) abstract figures 1-6 -----	20-23

☐ Further documents are listed in the continuation of Box C.

☒ See patent family annex.

* Special categories of cited documents :

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"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

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"Y" document of particular relevance;; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art

"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
31 January 2024	21/02/2024

Name and mailing address of the ISA/	Authorized officer
European Patent Office, P.B. 5818 Patentlaan 2 NL - 2280 HV Rijswijk Tel. (+31-70) 340-2040, Fax: (+31-70) 340-3016	Kämper, Fabian

1

INTERNATIONAL SEARCH REPORT

Information on patent family members

International application No
PCT/GB2023/052815

Patent document cited in search report		Publication date	Patent family member(s)		Publication date
US 2010318284	A1	16-12-2010	CN 101922361	A	22-12-2010
			CN 103256136	A	21-08-2013
			US 2010318284	A1	16-12-2010
WO 2014093030	A1	19-06-2014	CA 2892945	A1	19-06-2014
			CN 104854328	A	19-08-2015
			DE 112013005285	T5	13-08-2015
			US 2014158088	A1	12-06-2014
			WO 2014093030	A1	19-06-2014
US 6609499	B2	26-08-2003	NONE		