

WATER-COOLED DIESEL ENGINE FOR USE IN AN OUTBOARD MOTOR

The present invention relates to a water-cooled diesel engine for use in an outboard motor.

5 Hitherto, 2-stroke petrol engines operable with a fuel mixture have been most popular for use in marine outboard motors, which are mounted on the stern of small-sized vessels, because this type of engine best meets the requirement for a light weight and a small size which are the essential requisites for marine outboard motors. 10 Thus, diesel engines have been used very seldom in marine outboard motors. The current rise in the price of petroleum, however, has given rise to a demand for the use of diesel engines in marine outboard motors. It will be noted that a distinction is here drawn 15 between an outboard "motor", by which term is intended the whole unit including engine, transmission, propellor, mounting clamp, controls etc, and an outboard "engine" by which is intended merely the propulsion unit of an outboard motor.

20 The use of a diesel engine in a marine outboard motor, however, involves overcoming various problems. Namely, a diesel engine can produce only a comparatively small power per unit weight because of the need to use cast iron as the material of the cylinder block and cylinder head. In addition, the overall number of engine parts is 25 large because the cylinder block and the cylinder head are

constructed separately from each other and joined to each other by means of bolts with a gasket interposed therebetween. In addition, the seal of the gasket tends to become imperfect.

5 An engine used in an outboard motor has to be swung manually for steering and has to be tilted up as desired. In addition, such an engine has to be carried when not in the boat, and mounted easily. Furthermore, an outboard motor engine is required to have a centroid on the neutral axis thereof. Namely, if the centroid is offset to the left or right, the manoeuvrability of the engine will
10 be impaired.

In order to obviate these problems, it is necessary to meet the requirements such as minimized weight of the outboard motor unit, including the engine itself, symmetry of the engine with respect to
15 a plane parallel to the intended direction of movement, minimized height, compactness of the engine, reduction in the number of the parts, simplified construction and reduced cost.

20 When a diesel engine is used for a marine outboard motor, the compactness of the engine is affected by the position of the exhaust gas outlet.

According to the present invention there is provided a water-cooled diesel engine for use as a marine outboard engine, comprising a
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cylinder block and a cylinder head formed as an integral unit from aluminium or a light-weight alloy of the aluminium group, the said cylinder head having intake ports and exhaust ports formed therein, the said exhaust ports communicating with an exhaust manifold
5 formed integrally in the said cylinder head.

Embodiments of the invention can provide a higher output power per unit weight of the engine and thus attain a higher propulsion power of the engine for a given weight.

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An engine formed as an embodiment of the invention may be made compact and with a reduced number of parts to facilitate the assembling.

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Two embodiments of the invention will now be more particularly described, by way of example, with reference to the accompanying drawings, in which:

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Figure 1 is a partly-sectioned side elevational view of an outboard motor having an engine embodying the present invention;

Figure 2 is a vertical sectional view of a diesel engine in accordance with a first embodiment of the invention;

Figure 3 is an enlarged view of a portion of the diesel engine shown in Figure 2;

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Figure 4 is a graph showing the relationship between the

radius of the corner of the combustion chamber and stress occurring therein;

Figure 5 is a schematic further enlarged view of a portion shown in Figure 3;

5 Figure 6 is a sectional view taken along the line VI-VI of Figure 2;

Figure 7 is a schematic, partly sectioned side elevational view of an outboard motor including a diesel engine formed as a second embodiment of the invention; and

10 Figure 8 is a sectional view taken along the line VIII-VIII of Figure 7.

In the accompanying drawings the same reference numerals have been used to identify the same or like components throughout.

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In the specification references to the orientation of the engine will be understood to relate to the engine in its intended upright position of use on an outboard motor for a small vessel. References to left, right, front etc will be construed accordingly.

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Referring to Figure 1, an outboard motor M has a diesel engine 2 accommodated by a cowling 1. The power of the diesel engine 2 is transmitted to a propeller 4 provided at the lower end of the body 3 of the outboard motor M to rotate the propeller 4 through a vertical transmission shaft (shown by two-dot-and-a-dash line)

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disposed in the body 3 of the outboard motor. As the propeller 4 is rotated, a small-sized vessel on which the outboard motor is mounted is propelled to run in the direction of an arrow A. The vessel is steered as the outboard motor M is swung to the left and right by means of a tiller 5 which projects forwardly from the outboard motor M.

Exhaust gas E emitted from the diesel engine 2 flows through an exhaust passage 7 in the body 3 of the outboard motor and is discharged into the ambient water through an exhaust opening 7A which is positioned below the water level L during running of the vessel.

Referring now to Figure 2, a diesel engine 2 in accordance with the invention has a crankshaft 11 provided at its upper end with a fly-wheel 12. This engine, designed specifically for use as a marine outboard engine, is provided in a bore at the lower end of the crankshaft with an internal spline 13 by means of which the crankshaft is connected to an output shaft (not shown). In the illustrated embodiment, the engine 2 has two cylinders 14 and 15 which are arranged one above the other. Hereinafter two median longitudinal planes of these cylinders will be used for the purpose of explanation, namely a horizontal plane 0-0 (Figure 2) extending horizontally in the fore and aft direction of the engine, there being an individual such plane for each cylinder, and a vertical

plane 0₁-0₁ (Figure 6), which is common to both cylinders.

Numerals 16 and 17 denote pistons and connecting rods, respectively.

5 A cylinder block 18 and a cylinder head 19 are formed by casting as a unit from aluminium or an aluminium-base light-weight alloy. A cooling water jacket 20 is formed in the combined cylinder block 18 and cylinder head 19, which latter are integral with each other. The cooling water is introduced by a cooling water pump (not shown) into the cooling water jacket 20. Exhaust ports 21 and intake ports 22 are formed in the cylinder head 19. The cylinder head 19 is provided also with cylindrical bosses 27 for supporting the stems 25, 26 of the exhaust valves 23 and intake valves 24, as well as cylindrical bosses 29 defining mounting holes for the fuel injection devices 28. As shown in Figure 6, an exhaust manifold 30 is integrally formed in the cylinder head 19. Each of the cylinders 14 and 15 has one exhaust valve 23 and one intake valve 24. Thus, four valves in total, that is two exhaust valves 23 and two intake valves 24 are arranged in the vertical plane with their stems lying horizontally, and are adapted to be driven by a common cam shaft 31 through valve arms 32 in a manner which will be explained later. A casing 34 defining a valve arm chamber 33 accommodating the cam shaft 31 and the valve arms 32 is fixed to one end surface of the cylinder head 19 by means of bolts (not shown).

The diesel engine 2 has a crank case 36 which can be split into two parts: namely, a first part 37 adjacent to the cylinder block 18 and a second part 38 opposite to the cylinder block 18. The first part 37 is formed as a unit with the cylinder block 18 while the
5 second part 38 is fastened by bolts 39 to the first part 37. The plane at which the two parts 37 and 38 are joined contains the neutral axis C-C of the crankshaft and is perpendicular both to the horizontal and vertical median planes 0-0 and 0_1-0_1 of the cylinders. The part 38 of the crank case is made of a material
10 identical to or similar to that of the cylinder block 18.

A detailed description will be made hereinunder as to the construction of every portion of the diesel engine. Referring to Figure 3, which is an enlarged view of a portion of the engine
15 shown in Figure 2, dry liners 40 presenting the sliding surfaces for the pistons 6 are made from cast iron and are cast in the cylinder block 18. An alloy layer is formed along the boundary between the liner 40 and the cylinder block 18 and the liner 40 is joined to the cylinder block 18 through this alloy layer. The end
20 41 of the liner 40 projects towards the cylinder head 19 by a small distance ℓ_1 from the top ring 42 on the piston 16 at the top dead centre, and is comparatively spaced from the explosion surface 43 of the cylinder head 19.

25 Namely, the distance ℓ between the explosion surface 43 to the end

41 of the liner 40 is selected to be as large as possible without causing any obstacle to the sliding motion of the top ring 42. The reference numeral 44 denotes a corner portion near the outer periphery of the explosion surface 43. That is, the cylinder block 18 and the cylinder head 19 join each other at the corner 44. The corner surface 44a of the corner 44 facing the combustion chamber 45 has an arcuate cross-section of a radius R.

According to this arrangement, it is possible to obtain a mechanical strength of the corner 44 large enough to withstand the force produced by the combustion pressure in the combustion chamber 45. Namely, although the stress produced by the combustion tends to be concentrated to the corner 44, the stress can be dispersed by the roundness of radius R of the corner surface 44a so that the cracking in the corner 44 is avoided advantageously. Furthermore, since the length ℓ of the corner 44 is selected to be large, it is possible to obtain a large radius R of curvature so that the stress dispersing effect is enhanced to ensure a sufficiently high mechanical strength.

A test result shows that the strength of the corner 44 can be increased to a sufficiently high level when the radius R is selected to be about 2% of or greater than the cylinder inside diameter but smaller than the distance ℓ to permit the movement of the piston 16. More specifically, referring to Figure 4,

representing the ratio of the radius R to the cylinder inside diameter by γ (%), the stress δ is drastically decreased as the ratio γ is increased when the ratio γ is smaller than 2%. However, the decrease of the stress becomes not so appreciable when the ratio γ is increased beyond about 2%. It is, therefore, possible to obtain a sufficiently high mechanical strength at the corner 44 shown in Figure 2 by selecting the value of the ratio γ to be about 2% or greater.

Referring now to Figure 3, the upper portion 47 of the cylinder block 18 surrounding the end portion 46 of the liner 40, as well as the cylinder head 19, is expanded outwardly beyond the intermediate portion 48 of the cylinder block 18 as indicated by arrow S. By thickening the upper portion 47 of the cylinder block 18, it is possible to prevent the upper portion 47 from expanding radially outwardly when subjected to the explosion pressure and, hence, to avoid undesirable formation of a gap between the upper portion 47 and the liner 40. Consequently, leak of the gas into the crank chamber through the gap between the liner 40 and the cylinder block 18 can be avoided so as to prevent deterioration of the engine performance attributable to such a leak of gas. It is not necessary to thicken the wall of the intermediate portion 48 of the block 18 adjacent to the crank case 36 because such intermediate portion is not subjected to a high explosion pressure. Thus, it is possible to reduce the weight of the engine without causing any

problem concerning the strength, by thinning the wall of the intermediate portion of the cylinder block 18. The thickening of the upper portion 47 increases the cross-sectional area of the upper portion 47 so that the force in the axial direction of the cylinder produced by the pressure acting on the explosion surface 43 can be borne by a greater area of the upper portion 47. Consequently, the stress in the corner 44 can be decreased correspondingly.

Referring now to Figure 5, the inner surface of the liner 40 is finished by honing. The surface 50 finished by the honing extends between the portion of the inner surface of the liner 40 adjacent to the crank chamber to the end portion 51 of the same. More specifically, the end portion 51 is located at a position which is offset by a distance l_2 which is about 1 to 4 mm from the position of the top ring 42 on the piston 16 at top dead centre towards the explosion surface 43, so that the top ring 42 slides only within the area of the surface 50 finished by honing. The portion of the inner peripheral surface of the liner 40 between the end portion 51 and the extreme end 41 is recessed radially outwardly by a distance l_3 which is about 0.1 to 0.2 mm from the surface 50 finished by honing, so as to provide a recess 52 serving as a relief for the honing. This arrangement offers the following advantage. When the honing tool is inserted into the liner 40 from the crank chamber it is not desirable for it to touch the material of the cylinder head

19 or cylinder block 18 because of the risk of clogging, for which reason it is advisable to avoid honing up to the extreme end of the liner. If the liner 40 before honing had a constant inside diameter up to the end 41, a boundary or step of honing could be left in the end portion of the inner peripheral surface of the liner 40 after honing if the honing tool did not match perfectly with the end of the liner. Since the inside diameter of the liner would then be less at the end portion of the latter as compared with the surface 50 finished by the honing, the piston 16 could catch against such end portion of reduced inside diameter. Such problems, however, can be obviated in the engine of the invention because of the presence of the honing relief portion 52. In addition, the provision of the honing relief portion 52 in the liner 40 effectively prevents the honing tool from acting on the cylinder block 18 which is made from aluminium or its alloy so that the unfavourable clogging of the honing tool is avoided advantageously.

Referring back to Figure 3, the ceiling wall 53 of the cylinder head 19 presents at its one side the aforementioned explosion surface 43 while the other side of the same faces the cooling water jacket 20. The thickness of the ceiling wall 53 is greater at the central portion thereof than at the peripheral portion, so that the portion 20a of the cooling water jacket 20 adjacent to the centre of the cylinder is spaced by a greater distance than the outer

peripheral portion 20b of the same. By varying the thickness of the ceiling wall 53 in the manner described, it is possible to reduce the weight of the engine through reducing the mean thickness of the ceiling wall, while ensuring sufficient strength to the explosion force. In the illustrated embodiment, the thickness of the ceiling wall is changed in a stepped manner. This, however, is not exclusive and the thickness of the ceiling wall 53 may be progressively and linearly increased towards the centre of the latter. By so doing, it is possible to further enhance the strength of the ceiling wall 53.

As will be seen from Figure 2, the cylinder head 19 is provided therein with exhaust valves 23, intake valves 24, exhaust ports 21 and intake ports 22, so that the overall height H as measured from the explosion surface 43 to the end surface 35 is inevitably large. Furthermore, a multiplicity of ribs 55 are formed in the cylinder head 19 so as to form walls of the cooling water jacket 20, exhaust ports 21 and intake ports 22, as well as cylindrical bosses 27 mentioned before. Thus, the cylinder head 19 itself exhibits a sufficient strength because it has a large overall height H and because it is stiffened internally by a multiplicity of cylindrical bosses 27 and ribs 55.

Referring now to Figure 6 which is a sectional view taken along the line VI-VI of Figure 2, the cylindrical boss 29 for the fuel

injection nozzle 28 also contributes to the enhancement of the strength of the cylinder head 19. The cylindrical boss 29 extends from a position near the centre of the ceiling wall 53 substantially along the axis of the cylinder, so that it provides a greater effect of stiffening the cylinder head 19 as compared with other cylindrical bosses 27 shown in Figure 1 and ribs 55. The fuel injection device takes the form of an injector unit 57 in which a fuel injection nozzle 28 and a fuel pump 56 are constructed as a unit with each other. This injector unit 57, when fitted in the cylindrical boss 29 as will be explained below, serves further to increase the rigidity or strength of the cylinder head 19.

The construction of the injector unit 57 will be explained briefly. The injector unit 57 is composed of a fuel injection pump 56 having a body 91 and a fuel injection nozzle 28 having a sleeve 98 connected to the end of the body 91 of the fuel injection pump 56. As a plunger 93 moves reciprocatingly within a barrel 92 mounted in the body 91, fuel is supplied from the fuel injection pump 56 to the fuel injection nozzle 28 through a pressurized fuel passage 94 formed in the body 91. The injector unit 57 is provided with an annular step 95 between the end portion of the body 91 and the sleeve 98. With the annular step 95 pressed onto the inner peripheral step on the cylindrical boss 29, the portion of the body 91 projecting above the cylindrical boss 29 is fastened to the cylinder head 11 by means of a metal retainer 96. Therefore, an

initial compression force overcoming the explosion force is applied to the centre of the ceiling 53 by means of the injector unit 57. This also contributes to the increase of the strength of the cylinder head 19 so that the amount of deflection of the ceiling wall 53 is decreased effectively.

The portion of the injector unit 57 constituting the fuel injection pump 56 has a larger diameter than the nozzle portion 28, and also fits in the cylindrical boss 29. Therefore, the cylindrical boss 29 serves as a large reinforcement having a large diameter and, hence, contributes to the strength of the cylinder head 19.

The fuel injection nozzle 28 has a nozzle port which is opened when a high pressure is applied by the fuel delivered by the fuel injection pump 56. As stated before, in the injector unit 57, the fuel injection pump 56 and the fuel injection nozzle 28 are connected to each other solely through a short pressurized fuel passage 94 formed in the body 91, so that the fuel pressure established in the fuel injection pump is correctly transmitted to the fuel injection nozzle 28. It is, therefore, possible to inject the fuel from the fuel injection nozzle 28 at exact correspondence to the pressure in the fuel injection pump 56 without any secondary injection over the entire region of the variance of the fuel injection rate. Furthermore, since the pressure drop along the short pressurized fuel passage 94 is small, it is possible to

obtain a high fuel injection pressure in the fuel injection nozzle 28 which in turn promotes the atomization of the injected fuel. Furthermore, the injection from the fuel injection nozzle 28 takes place without a substantial time lag with respect to the pumping operation in the fuel injection pump 56. It is possible therefore to maintain an optimum condition for combustion and to make full use of the engine performance, partly because secondary injection and injection time lag are avoided and partly because atomization of the fuel is promoted. The avoidance of delay in fuel injection also contributes to improve the performance of the engine during high-speed operation. It is remarkable that these advantages are brought about with a simple construction which does not require a timer as has been found indispensable in conventional engines for adjusting the timing of the fuel injection.

The end of the fuel injection nozzle 28 is exposed to the combustion chamber 45 substantially at the centre of the explosion surface 43, and no vortex flow chamber (pre-combustion chamber) is provided in the cylinder head 19. Since the engine is of a direct injection type as stated before, it is possible to support the injector unit 57 by the full height H of the cylinder head 19. Consequently, the injector unit 57 can be held stably and the distance between the explosion surface 43 and the other end (protector 71) of the injector unit 57 can be decreased to permit a reduction in the size of the engine as a whole. Thus, although the

injector unit 57 inherently has a large length, it is nevertheless possible to keep the size of the engine small by designing the engine as a direct injection type engine. For comparison, if a vortex flow chamber were to be formed in the cylinder head 19, the distance between the injector unit 57 and the explosion surface 43 would have to be increased by an amount corresponding to the size of the vortex flow chamber, so that the size of the cylinder head 19 and the size of the case 34 would have to be increased.

Direct injection offers also the following advantage. Namely, in an engine having a vortex flow chamber, a heavy thermal load is imposed on the inner surface of the communication passage between the vortex flow chamber and the combustion chamber 45 during propagation of the flame from the former to the latter. By contrast, in a direct injection type engine having no vortex flow chamber, no local thermal stressing takes place in the cylinder head 19 nor in the cylinder block 18. Therefore, the cylinder head 19 and the cylinder block 18 are never damaged by heat even though they are made from aluminium or an aluminium alloy having small resistance to heat.

Aluminium or its alloys exhibit a high heat conductivity at the cost of small resistance to heat. Therefore, the heat transferred to the cylinder head 19 and the cylinder block 18 from the combustion chamber 45 can be delivered promptly to the cooling

water in the cooling water jacket 20, so that the cylinder block 18 and the cylinder head 19 are protected against overheating. Thus, the engine is protected against thermal damage even in this respect.

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The piston 16 is provided in the top surface thereof with a recess which defines a part of the combustion chamber 45. When the piston 16 is position at the top dead centre in its stroke, the portions of the top surface of the piston other than the central recess
10 closely approach the explosion surface 43 or the rounded corner around the explosion surface 43 without leaving any substantial gap or clearance.

The explosion surface 43 may be conically concave although it is
15 flat in the described embodiment. In this way, it is possible to enhance the strength of the cylinder block 18 and the cylinder head 19 because of the greater strength of the arch structure formed by the cylinder block 18 and the cylinder head 19 in the illustrated cross-section, that is the arch structure constituted by the side
20 walls of the cylinder block 18 and the ceiling portion provided by the cylinder head 19.

The following explanation relates to the mechanism for actuating the fuel injection pump 56, as well as the exhaust valve 23 and the
25 intake valve 24 (Figure 6 shows only the valve 24). Besides the

cam shaft 31 and the valve arms 32 which were mentioned before, the valve arm chamber 33 accommodates a valve arm 59 for the fuel injection pump 56. In addition, valve stems 25 and 26 of the exhaust valve 23 and the intake valve 24 respectively, as well as the fuel injection pump 56, project from the cylinder head 19 into the casing 34. The valve stems 25 and 26 are located, for example, on the left of the vertical longitudinal median plane 0_1-0_1 , that is the plane containing the axis of the cylinder: the terms "left" and "right" will be understood in relation to the normal orientation of the engine as illustrated in the drawings, in which, in Figure 2, the engine is shown in an upright orientation viewed from the left, so that forward motion is from right to left as indicated by the arrow A of Figure 2, which indicates the direction of motion of a vessel having the engine fitted thereto.

The ends of the valve stems 25 and 26 are located comparatively close to the head end surface 35. The valve stems 25 and 26 are provided at their ends with protectors 60 which are contacted by adjusting screws 62 held in position on respective ends of the valve arms 32 by means of lock nuts 61. Each valve arm 32 is provided at its intermediate portion with a tappet 63 adapted to be actuated by a cam 64 on the cam shaft 31. Four valve arms 32 are supported at their other ends by a common valve arm shaft 65 which is vertical and supported by the case 34. The valve arm shaft 65 is offset leftwardly from the longitudinal vertical median plane

0₁-0₁, and the outer peripheral surface thereof is somewhat spaced from the end surface 35 of the cylinder head 19. The axis 65a of this shaft is nearer the end surface 35 of the cylinder head 19 than the protectors 60 on the ends of the valve stems by a distance of about 2 to 3 mm (about 1/3 of the valve lift) when the valves are closed.

The cam shaft 31 is positioned to the rear of the valve arm 32, that is on the side remote from the end surface 35, and to the left of the lock nut 61. The shaft 66 for the valve arm 59 extends vertically on the right hand side of the central plane 0₁-0₁, and is disposed to the right of the cam shaft 31. The shaft 66 is also supported by the valve arm chamber casing 34. Two valve arms 59 (only one of them is shown) are carried at their intermediate portions by the common shaft 66 and are each provided at one end with a respective cam follower 67 in contact with the rear side of an associated cam 68 on the cam shaft 31. Adjusting screws 70 are fixed to the other ends of the valve arms 59 and held in place by means of lock nuts 69. The ends of the adjusting screws 70 contact the protectors 71 on the ends of the plungers 93. The adjusting screws 70 are located to the right of the longitudinal vertical median plane 0₁-0₁ of the cylinders. Each injector unit is inclined to this plane 0₁-0₁ and approaches it towards the end of the fuel injection nozzle 28.

An opening 72 is formed in the rear wall of the valve arm chamber casing 34 so as to extend from the right end towards the left end portion of the rear wall. A cover 73 for closing the opening 72 is secured by bolts 74 to the casing 34. A lever-type decompression
5 mechanism 75 is secured to the left of the case 34. The decompression mechanism 75 is for manually operating valves into opening positions when the engine is to be started. The valve arm 32 is provided with an arm 76 which is adapted to be actuated by the decompression mechanism 75.

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In the engine structure described above, the rotation of the cam shaft 31 causes cams 64 to actuate exhaust valves 23 and intake valves 24 through the action of valve arms 32 and, at the same time, the cam 68 actuates the plunger 93 of the fuel injection pump
15 56 through the action of the valve arm 59. The operation of the exhaust valves 23, the intake valves 24 and the fuel injection pumps 56 can be adjusted by varying the positions of the adjusting screws 62 and 70. Since the adjusting screw 70 and the lock nut 69 face the opening 72 in the valve arm casing 34, access to them can
20 be easily gained for adjustment through the opening 72 by removing the cover 73. The adjusting screws 62 and lock nuts 61 are also accessible for adjustment through the opening 72 via a gap between the cam shaft 31 and the shaft 66 because these shafts are on opposite sides of the adjusting screws 62 and lock nuts 61.

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As shown in Figure 2, the cam shaft 31 is supported at both ends and at an intermediate portion by the casing 34. For each of the cylinders 14 and 15, the cams 64 for actuating the valves and the cam 68 for actuating the fuel injection pump are spaced vertically along the camshaft 31, with the cam 68 being positioned at the same level as the horizontal longitudinal median plane 0-0. To the lower end of the camshaft 31 is connected a pump shaft 79 of a lubricating oil pump 78 which is fastened to the lower side of the casing 34 by means of bolts (not shown). The inlet of the lubricating oil pump 78 communicates with an oil pan (not shown) through an oil passage drilled through the casing 34, the cylinder head 19 and the cylinder block 18. The oil pan is formed by a case which encases the output shaft (not shown) extending downwardly from the lower end of the crankshaft 11. The outlet of the lubricating oil pump 78 is connected to every portion of the engine through an oil passage (not shown) drilled through the casing 34, the cylinder head 19 and the cylinder block 18.

The upper end portion of the cam shaft 31 projects above the casing 34. A pulley 81 fixed to the projecting part of the upper end of the cam shaft 31 is drivingly connected to another pulley 82 fixed to the upper portion of the crankshaft 11 by means of a timing belt 83. A generator 84 is connected to the fly-wheel 12 above the pulley 82. A ring gear 85 on the outer periphery of the fly-wheel 12 is adapted to be driven by a starter 86 secured to a portion 30 of the crank case 36. An oil filler duct 87 projecting

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obliquely upwardly is attached to the upper portion of the bottom wall (front wall) of the crank case 36. A governor 88 is attached to the cylinder block 18 while a fuel injection rate increasing device 89 which operates during starting up of the engine is
5 secured to the cylinder head 19. The governor 88 is adapted to be driven through the timing belt 83. The governor 88 and the fuel injection rate increasing device 89 are connected to the plunger 93 of the fuel injection pump 56 shown in Figure 6 through a lever mechanism 90.

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As will be seen from Figure 6, the governor 88 is disposed rearwardly of the crank case 36 and on the right of the cylinder block 18. A lubricating oil filter 100 is disposed on the left side of the cylinder block 18. As explained before, the fuel
15 injection pump 56 is mounted in the cylinder head 19 as a unit with the fuel injection nozzle 28. Thus, the fuel injection pump 56 is carried on the cylinder head 19, while the governor 88 and the lubricating oil filter 100 are carried on the cylinder block 18. Therefore, the engine as a whole exhibits a substantially
20 oval or egg-like form when viewed from the upper side as shown in Figure 6, which shape is quite suitable for use as a marine outboard engine. Needless to say, the engine as a whole is covered by a cowling 1 (Figure 1) of the outboard motor M.

25 An air intake pipe 101 is connected at one end to the intake port

22 on the right side of the cylinder head 19 while the other end is open near the forward end of the crank case 36. The air intake pipe 101 extends along the right side of the cylinder block 18 and the crank case 36 and the inlet end portion thereof turns round to the front side of the crank case 36 to reach a position near the longitudinal vertical median plane O_1-O_1 . By adopting such a long air intake pipe, it is possible to enhance the intake inertia effect to improve the performance of the engine.

As has been described, since the intake pipe 101 extends along the right side of the crank case 36 and since the starter 86 is positioned on the left side slightly forwardly of the crank case 36, it is possible to obtain a certain rough balance and approximate symmetry of the engine as a whole with respect to the longitudinal axis thereof. Furthermore, a certain balance and symmetry within the valve arm casing 34 is attained as can be seen from Figure 6 by positioning the injector units 57 inclined to the right while the cam shaft 31 and the valve arms 32 are disposed to the left.

Figures 7 and 8 show a 4-stroke water-cooled diesel engine formed as a second embodiment of the invention for use in a marine outboard motor.

In operation of this second embodiment, fuel is injected from injector units 57 mounted in the cylinder head 19, and two pistons

16 in cylinder liners 40 in the cylinder block 18 are reciprocated in the horizontal directions thereby to drive the vertical crankshaft 11 which in turn drives the propeller 4 through the thrust shaft 6.

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As shown in Figures 7 and 8, fresh air is introduced as indicated by an arrow S through intake ports 22 communicating with respective combustion chambers 45. The gas generated as a result of the combustion is discharged through respective exhaust ports 21 as indicated by arrows E. These two exhaust ports 21 are connected through an exhaust manifold 30 to a common exhaust passage 7 formed in the body 3 of the outboard motor as shown in Figure 1.

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The exhaust manifold 30 leads to an outlet 30a at the interface between the diesel engine 2 and the body 3 of the outboard motor and, indeed, the first embodiment has a similar configuration in this respect. In both embodiments this outlet 30a is partly located in the cylinder block 18, that is towards the crank

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As has been described, in the diesel engine 2 of both the first and the second embodiment, the exhaust manifold 30 is formed as a unit with the cylinder head 19 and the cylinder block 18, so that it is not necessary to use a separate exhaust manifold unlike conventional engines. The elimination of a separate exhaust

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manifold affords a compact construction of the engine as a whole and reduces the number of parts, as well as the number of steps of the assembling process.

- 5 In the case of a marine outboard motor, since the engine as a whole is covered by a cowling, it is important to surround the exhaust manifold by water to prevent an excessive temperature rise in the cowling 1 due to the heat derived from the exhaust manifold which is heated to a high temperature. The provision of a
10 separate water-cooled exhaust manifold would increase the weight as well as the number of parts.

- In contrast, in the engine of the invention, the exhaust passage passes through the cooling water jacket 20 formed in the cylinder
15 head 19 and the cylinder block 18 so that the water-cooled manifold can be formed without requiring any additional provision of the water chamber around the exhaust passage. This arrangement is superior to the conventional one because it contributes to the reduction in the weight, number of parts and the size.

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- The arrangement of the parts of the engine of the described embodiment may be inverted with respect to the neutral axis of the engine. The invention does not exclude an arrangement in which the portion 37 of the crank case 36 is formed separately from the
25 cylinder block 18, and the invention can be embodied as an engine having three or more cylinders. It is also possible to provide a

vortex flow chamber (pre-combustion chamber) within the cylinder block 18. It is also not essential to form the fuel injection nozzle 28 and the fuel injection pump 56 as a unit. Namely, the fuel injection nozzle 28 and the fuel injection pump 56 may be
5 constructed separately from each other and connected to each other through a high-pressure tube.

As has been described, according to the invention, there is provided a diesel engine in which the cylinder block and the
10 cylinder head are formed as a unit from aluminium or an aluminium alloy, and intake and exhaust ports are formed in the cylinder head. Thanks to the structural features as summarised above, the diesel engine of the invention offers the following advantages.

(a) The weight of the engine can be reduced as compared with
15 conventional engines in which the cylinder block and the cylinder head are formed from cast iron. The reduced weight in turn affords an increased output power per unit weight of the engine as compared with conventional engines.

(b) The number of parts is decreased, as well as the weight, and
20 assembly is facilitated because it is not necessary to unite the cylinder block and the cylinder head by bolts through a gasket or a like member, unlike conventional engines.

(c) A higher cooling effect can be attained because of the
elimination of the gasket serving as a heat insulator between the
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cylinder head and the cylinder block.

(d) Undesirable deformation of the liner is prevented because of the elimination of the necessity for tightening of the cylinder head by head bolts, so that the life of the liner is prolonged and the maintenance of the same is facilitated.

(e) Although a high pressure is produced in the cylinder as in the case of ordinary diesel engines, the leakage of gas, cooling water and lubricating oil can be avoided perfectly because the cylinder block and the cylinder head are constructed as an integral unit. It is therefore possible to increase the combustion pressure to attain higher output power of the engine.

(f) The weight of the engine can be decreased thanks to the elimination of thickened portions around the jointing surfaces of the cylinder head and the cylinder block in the conventional engines.

(g) The combustion chamber defined by the unitary structure of the cylinder block and the cylinder head can have various forms which enhance the mechanical strength of the engine.

(h) The freedom of selection of the positions of the exhaust ports and intake ports is increased thanks to the elimination of the head bolts.

Specification No. 84.32733 (Serial No. 2150221) is divided from this Application and contains a disclosure of the invention defined in the following Claims.



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CLAIMS

1. A water-cooled diesel engine for use as a marine outboard engine, comprising a cylinder block and a cylinder head formed as an integral unit from aluminium or a light-weight alloy of the aluminium group, the said cylinder head having intake ports and exhaust ports formed therein, the said exhaust ports communicating with an exhaust manifold formed integrally in the said cylinder head.
2. A water-cooled diesel engine as claimed in claim 1, wherein the said engine has a fuel injection device including at least one injector unit.
3. A water-cooled diesel engine as claimed in claim 1 or claim 2, wherein the corner portion of the combustion chamber formed by the said cylinder block and said cylinder head is rounded to have an arcuate cross-section.
4. A water-cooled diesel engine as claimed in any of claims 1 to 3, wherein the said cylinder block and the said cylinder head are provided with a continuous cooling water jacket.
5. A water-cooled diesel engine as claimed in any preceding claim, wherein at least one cylindrical boss for the fuel injection nozzle is formed in the said cylinder head.



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6. A water-cooled diesel engine as claimed in any preceding claim, wherein the said engine has at least one dry liner mounted in the cylinder block and having an inner peripheral surface finished by honing, and wherein a radially outwardly recessed honing relief portion is formed in an end portion of the inner peripheral surface of the said liner adjacent to the end of the combustion chamber, whereby to avoid honing the material of the cylinder block.
- 10 7. A water-cooled diesel engine as claimed in any preceding claim, wherein the combustion chamber formed by the cylinder head and the cylinder block is a direct-injection type combustion chamber.
- 15 8. A water-cooled diesel engine as claimed in claim 6 or claim 7, wherein the said dry liner is united with said cylinder block by casting and the end of said liner is positioned below the rounded corner portion of the combustion chamber and above the position occupied by the top ring on a piston in the liner when positioned at top dead centre.
- 20 9. A water-cooled diesel engine as claimed in any preceding claim, wherein the said exhaust manifold formed in the cylinder head has an outlet opening in the surface of said combined cylinder head and cylinder block at which it is joined to the body of an outboard motor.
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10. A water-cooled diesel engine for use as a marine outboard engine substantially as hereinbefore described with reference to and as shown in the accompanying drawings.
