Cylinder Block Layout and Design Decisions

8

8.1 Initial Block Layout, Function, and Terminology

In Chap. 5 the required engine displacement was calculated, and in Chap. 6 the number of cylinders, cylinder layout, and bore-to-stroke ratio were determined. These earlier decisions form the starting point for the discussion of this chapter. It is in this chapter more than any other that it will be necessary to limit the discussion to designs specific to automotive applications. Over the entire range of reciprocating piston internal combustion engines there is an extremely wide range of cylinder configurations and block layout and construction techniques. By limiting the discussion to automobile engines and heavy-duty engines in mobile installations, primary attention will be placed on in-line four, five and six cylinder engines, and vee six, eight, ten, and twelve cylinder engines. Casting the net a bit wider allows discussion of horizontally opposed four, six, and eight cylinder engines, and mention of the recently revived 'W-8' and 'W-12'. The cylinder block is the foundation of the engine, and supports the piston, cranktrain, cylinder head, and sometimes the valvetrain. It also houses the lubrication and cooling systems. It provides mounting points for the charging system, starting system, power take off (PTO), and typically has mounts which support the entire powertrain. The engine may be rigidly mounted as a structural member of the chassis, such as in a racecar or motorcycle. The cylinder block supports a variety of static, dynamic, and thermal loads, and must provide stiffness and alignment for many components. Because of the complexity of geometry, and complexity of loading, hand calculations are rarely used. Simplified finite element analysis (FEA) of a single power cylinder is usually the starting point, prior to analysis of the entire assembly.

Example cut-away illustrations of in-line and vee engines are shown in Figs. 8.1 and 8.2 respectively. In both of these engine configurations as used in automotive applications, the blocks are most often single-piece castings consisting of crankcase and cylinder sections. Engines in other industries are sometimes cast or fabricated with separate cyl-

K. Hoag, B. Dondlinger, *Vehicular Engine Design,* Powertrain,
DOI 10.1007/978-3-7091-1859-7_8

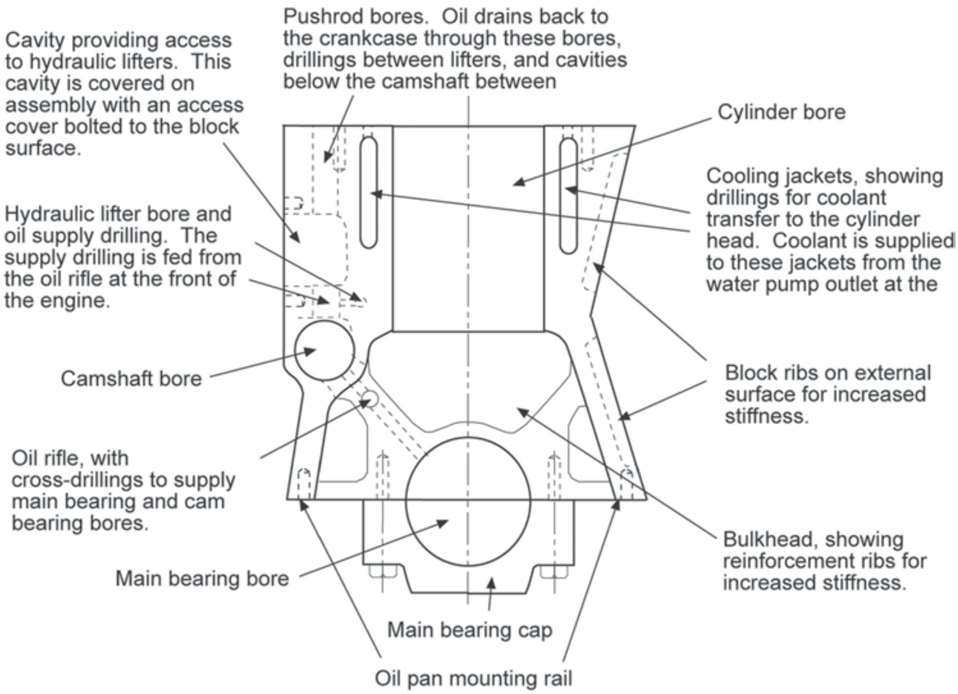


Fig. 8.1 Front-view of in-line automotive cylinder block, identifying typical features

inder block and crankcase sections. The crankcase section consists of a series of parallel bulkheads containing the main bearing journals in which the crankshaft spins. The front and rear bulkheads form the front and rear outside mating surfaces of the cylinder block. The bulkheads are tied together by outside walls generally referred to as the block skirts. The oil pan is affixed to the pan rails along the base of the skirts. Much of the lubrication system, to be discussed further in Chap. 12, is incorporated in the crankcase. One or more banks of cylinders are cast above the crankcase portion. The cylinder walls may be cast directly into the block, or inserted liners may be used. The upper surface of the cylinder section is termed the firedeck, and serves as the mating surface for the cylinder head as shown in Fig. 8.3. The cylinder section includes cooling jackets and plays an integral role in the cooling system. This system will be covered further in Chap. 13. The cylinder block may also contain the camshaft as shown in Fig. 8.1. The engine in Fig. 8.2 has overhead cams, mounted in the cylinder heads. The remaining sections of this chapter will take up each portion of block layout in greater detail. Durability development will be addressed in Chap. 10.

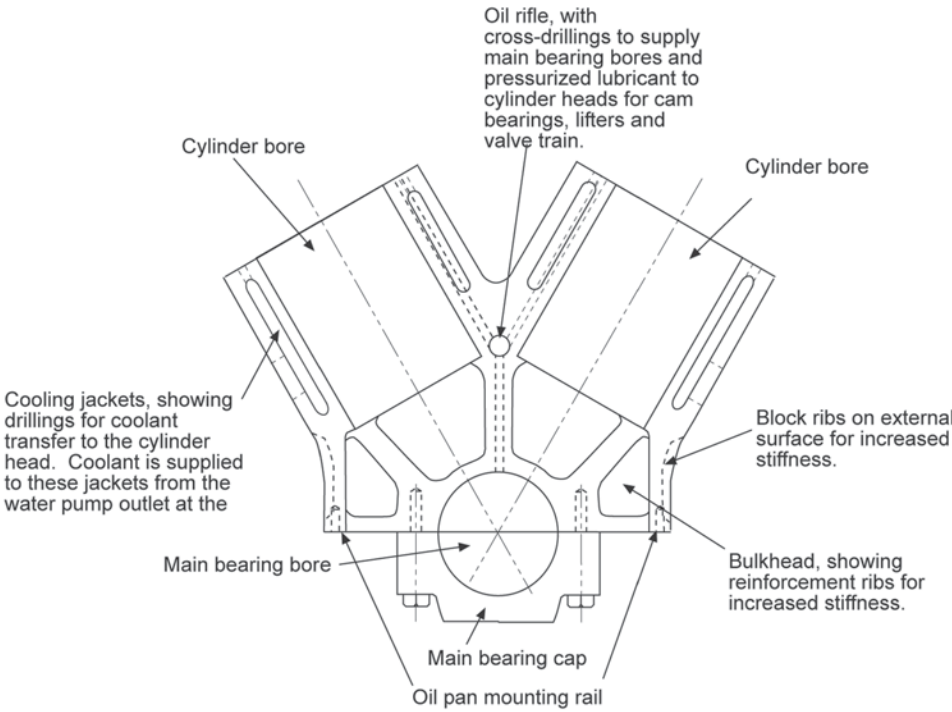


Fig. 8.2 Front-view of automotive cylinder block for vee engine, identifying typical features

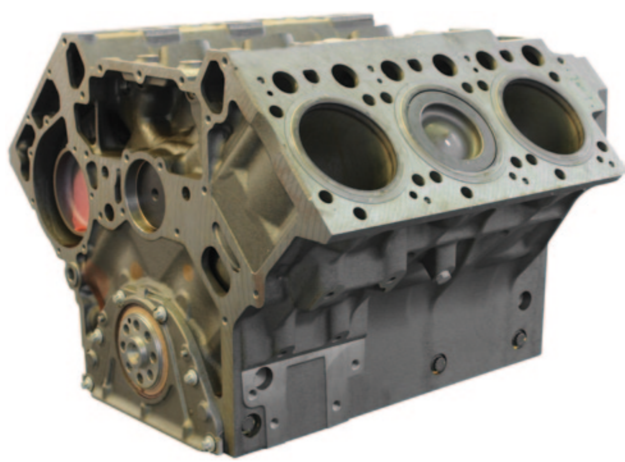


Fig. 8.3 Example V6 cylinder block showing exposed firedeck surface where cylinder head will be affixed

8.2 Main Block Features

The crankcase portion of the block is further depicted in a simplified three-dimensional representation in Fig. 8.4. As was described in the previous section, the crankcase consists of a series of parallel bulkheads tied together with the block skirts which make up the outside surfaces of the block. There is one bulkhead for each crankshaft main bearing saddle. In most in-line engines there is a main bearing journal between each cylinder—five main bearings in each of the four-cylinder engines shown in Figs. 1.6a, b and 1.8b; six in a five-cylinder engine; and seven in a six-cylinder engine. In some cases the engine is made smaller and lighter by placing a main bearing only between every second cylinder, but this practice is not typically used as it decreases block and crankshaft stiffness and increases noise emissions. In vee engines there is a main bearing between each pair of cylinders across the vee—four main bearings in a V-6; five in a V-8 (as shown in Figs. 1.6c and 1.8a).

In a multi-cylinder engine it is necessary to split each main bearing journal radially in order to install the crankshaft. The caps are separately cast and bolted to the block to complete the bearing journal. Because the primary cap loading is tensile it is quite common for the caps to be made of a different alloy than the block to which they mate. For example, ductile iron caps are frequently used with gray iron blocks, especially in engines seeing high load factors and for which durability expectations are high. A recent exception to separately casting the main bearing caps is seen when the block is cast from compacted graphite. The much higher tensile strength allows the caps to be made from the same material as the block. This in turn allows the caps to be cast integral with the block, and then "cracked," using the same process that has become the norm with connecting rods

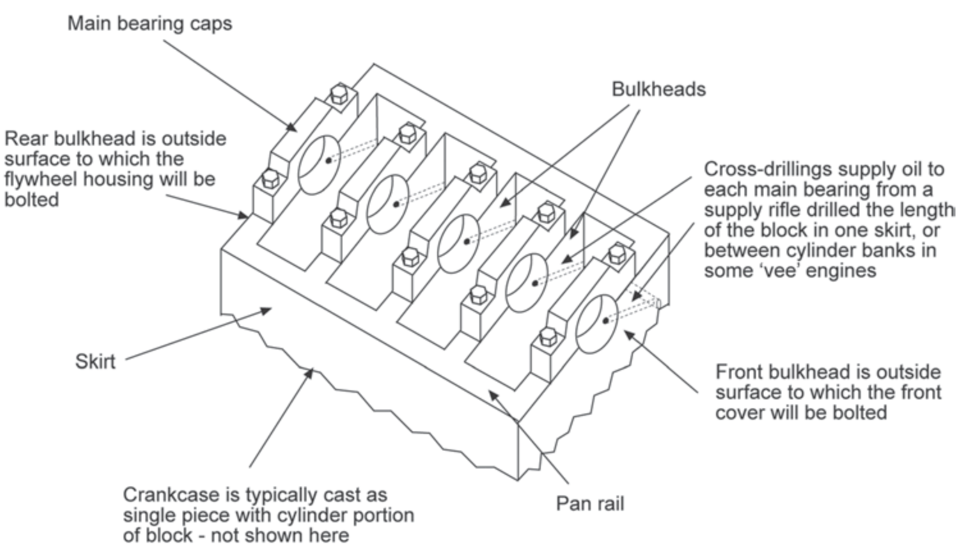


Fig. 8.4 Simplified depiction of crankcase, showing main design features

and their caps. The process is further discussed in Chap. 16 where connecting rods are discussed.

At the base of the skirts are the pan rails—pads of material running along the length of the skirts which are machined to mate with the oil pan, and which contain a series of tapped and drilled holes to secure the pan. Near the front or rear of the block an additional pad may be used immediately inboard from one of the pan rails to which the oil pump will be mounted. Alternately, the oil pump may be mounted co-axial with the crankshaft using a gerotor style pump, and with the housing integrated within the block or front cover. The oil pump, which will be discussed further in Chap. 12, is a positive displacement pump used in conjunction with a pressure relief valve and bypass passage back to the oil pan. In some cases the pressure relief valve is included in the lube pump housing, while in other cases it is incorporated into the block casting. An example of the latter approach is detailed in Fig. 8.5.

A supply drilling from the oil pump will intersect a horizontal oil rifle or gallery drilling running the length of the block to supply pressurized oil throughout the engine. The gallery drilling is cut through the entire length of the engine. Because of the length of the drilling, it is often made in progressively smaller steps from both the front and back of the block. The additional cost of the multi-step process is minimized by placing the successive operations at consecutive stations in the block machining line. Threaded plugs seal the rifle drilling at both the front and rear bulkheads. Cross-drillings through each bulkhead are then made to distribute pressurized oil to each main bearing. Further cross-drillings must be made at various locations in the crankcase to supply pressurized oil to the camshaft and valve train. If piston cooling nozzles are to be used in the engine the same gallery is often used. Other engines use a second gallery drilling devoted to these nozzles, also fed from the lube pump. Locations for the piston cooling nozzles must be designed into the block

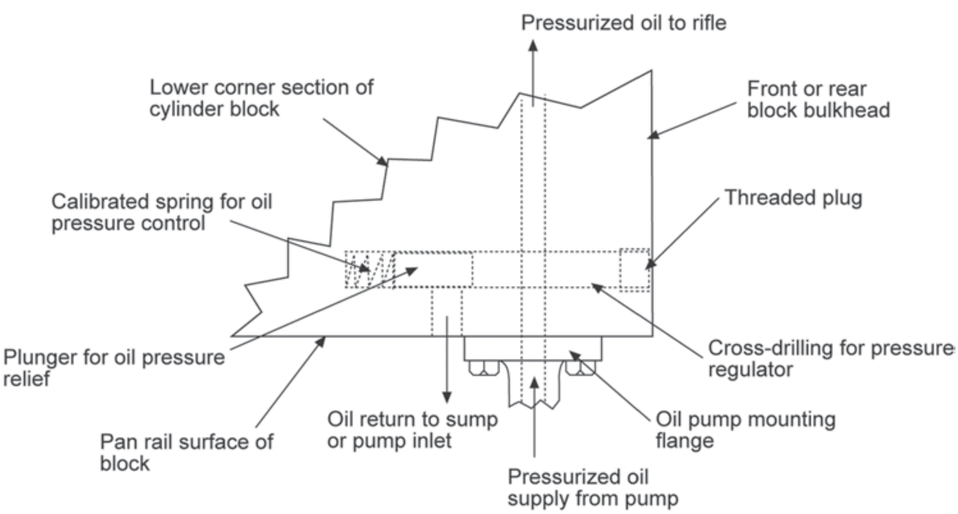


Fig. 8.5 Details of cylinder block showing oil pump mounting and pressure regulation

Fig. 8.6 Piston cooling nozzles

at each cylinder. These generally consist of machined holes cross-drilled from the gallery into which the nozzles are pressed, and flat surfaces on which the nozzles seat, and to which they are bolted as shown in Fig. 8.6. If the nozzles are designed to provide directed sprays, the location is critical and the machined pads may include dowel pins by which the nozzles are located. Piston cooling nozzles can also be seen in Figs. 1.6a, b, 1.8a and b.

The relatively open crankcase sections allow the casting cores to be held from the outside. In sand cast designs each cylinder in an in-line engine (or pair of cylinders in a vee engine) uses a single core around which the cylinder and crankcase bulkheads are cast. These open sections also lend themselves well to die casting, and this process has become increasingly popular with high-volume aluminum blocks.

As was stated in the opening section of this chapter, the front and rear bulkheads also serve as the front and rear surfaces of the block. In most automotive engines, the cam drive system is located immediately outside the front surface. This is the case for each of the engines of Chap. 1; the reader is referred especially to Fig. 1.7 where the cam drive system and its mounting to the front bulkhead can be clearly seen. The side-views of the engines in Figs. 1.6 and 1.8 also provide a look at front and rear bulkhead design. The majority of the front surface is milled flat, and the cam bearing plate, any auxiliary gears, and belt or chain tensioners are mounted to this surface as shown in Fig. 8.7. A stamped or cast cover then encloses this drive system, and is bolted to the block face. The cover must incorporate a gasket or seal against oil leakage, and in many engines it will also include sealed coolant transfer passages between the water pump and the cylinder section of the block. The crankshaft nose protrudes through the cover, and it is convenient to drive the water pump from the crankshaft nose, and therefore to mount it in front of the cam drive cover. The rear bulkhead is also milled, in this case mating with the flywheel housing or transmission casing as shown in Fig. 8.8. A semi-circular arc of bolts surrounds the upper half of the flywheel. Because of the criticality of shaft alignment between the engine and transmission this bolt pattern is usually supplemented by a pair of dowel pins on this mating surface,

Fig. 8.7 Cylinder block front bulkhead and timing cover mounting face

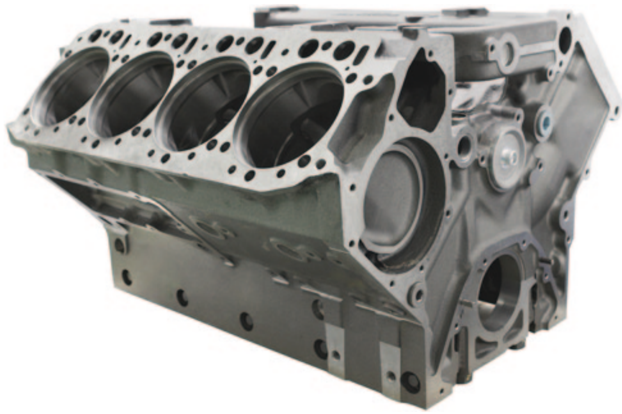


Fig. 8.8 Cylinder block rear bulkhead and bell housing mounting face

one on either side of the crankshaft. The dowel pins may not be necessary in transverse installations, where a chain drive replaces the direct alignment with the transmission shaft.

The brief discussion of this section introduces the main design features of the cylinder block. A discussion of block ribbing, primarily along the skirts and in some cases on the bulkheads will be deferred to Chap. 10, where block durability and noise are considered. The discussion now proceeds to critical crankcase layout dimensions. Various options for approaching further specific features of the design will then be covered.

8.3 Main Block Design Dimensions

A front-view layout showing critical crankcase dimensions for a vee engine is shown in the sketch in Fig. 8.9 and the example photograph of Fig. 8.10. While of necessity this book must cover engine design in a linear fashion it is important to recognize that much of the design work actually occurs along parallel paths. Evidence of these parallel paths can

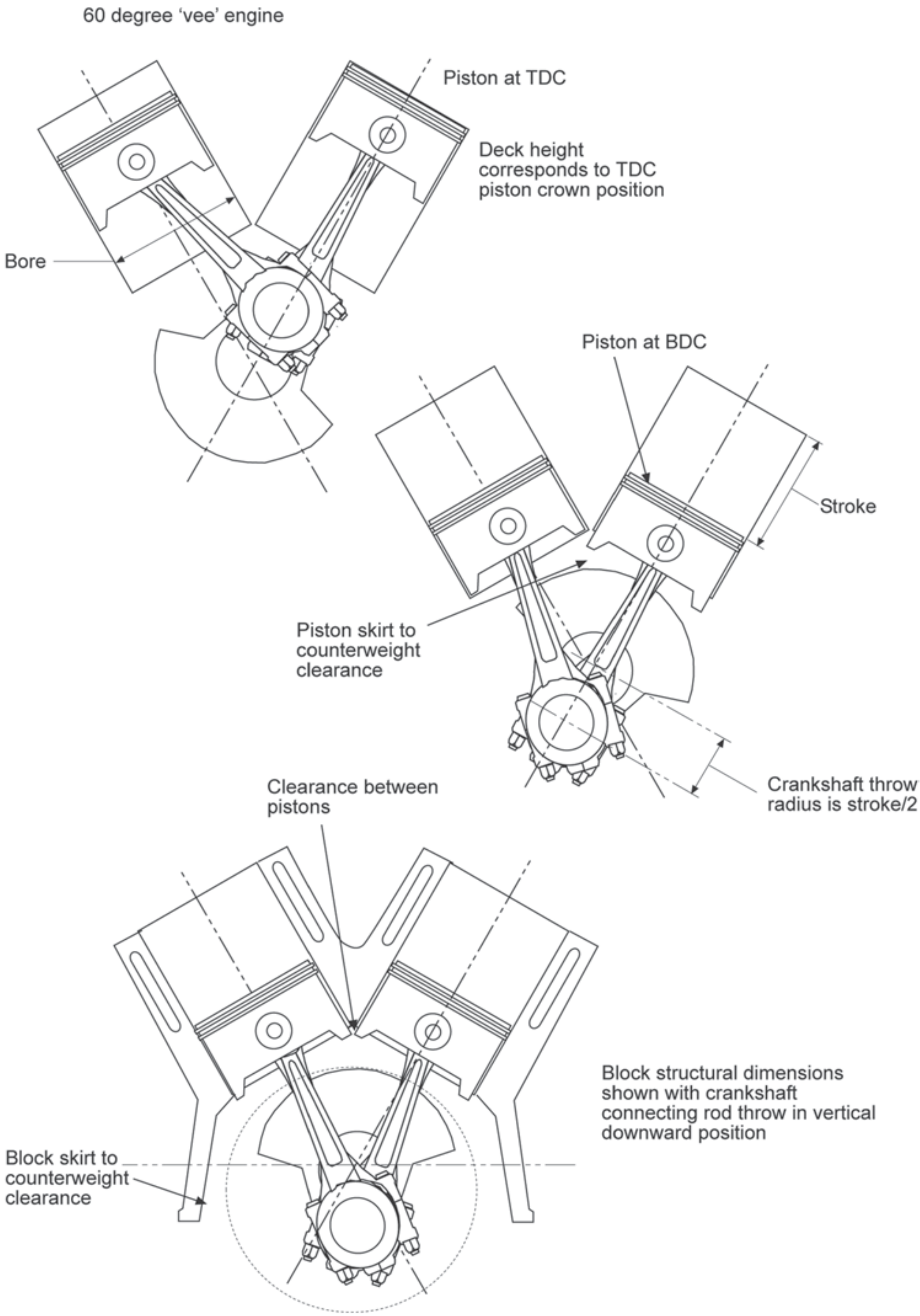


Fig. 8.9 Critical layout dimensions defining minimum package size for a vee engine

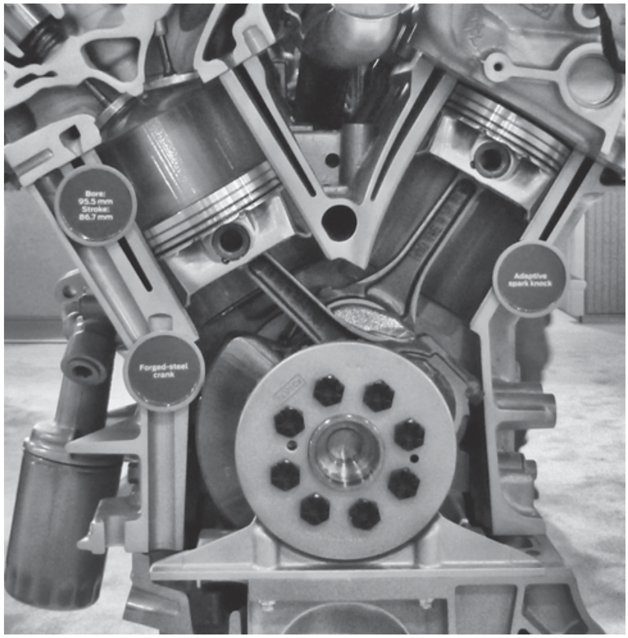


Fig. 8.10 Connecting rod and counterweight to cylinder block clearance

now be identified in Fig. 8.9. The main and rod bearing diameters will be determined as part of the crankshaft design and bearing sizing processes. Crankshaft counterweight diameter is an element of the engine balance and crankshaft design processes. The width of the crankcase at the pan rails will be further impacted by design features of the large end of the connecting rod. As the design layout moves from the crankcase to the cylinder section of the block critical dimensions will be determined by decisions regarding connecting rod length and piston design. The design begins by establishing deck height, then vee angle is considered, and finally cylinder spacing. This design loop is iterated as necessary.

8.3.1 Deck Height

A critical block layout dimension is the ***deck height***. The dimensional stack-up that defines deck height is illustrated in Fig. 8.11. The deck height is the distance from the crankshaft main bearing centerline to the firedeck surface, where the cylinder head mates to the block.

Combustion chamber design will be discussed further in the next chapter where cylinder head design is presented. It will become apparent in that discussion that for both diesel and spark-ignition engines the piston rim very closely approaches the firedeck surface as it reaches TDC as shown in Fig. 8.12. This being the case, the static deck height is determined by the sum of one-half the stroke, plus the connecting rod length from rod bearing centerline to piston pin centerline, plus the piston height from its pin centerline to rim (compression height) as shown in Eq. 8.1.

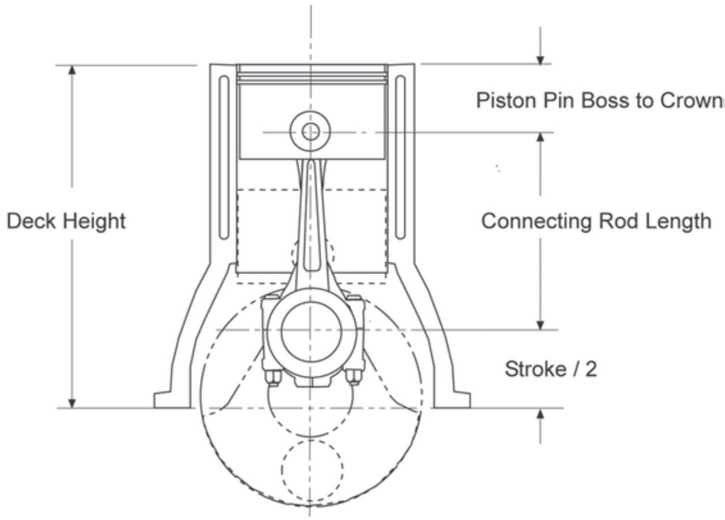


Fig. 8.11 Layout variables determining the deck height

$$\text{MinimumDeckHeight} = \frac{\text{Stroke}}{2} + \text{Connectingrodlength} + \text{Pistoncrownheight} \quad (8.1)$$

A compact design requires minimizing the deck height, which means reducing the stroke, connecting rod length, or piston crown height. Reducing the stroke to minimize deck height requires increasing the bore, which will affect the bore-to-stroke ratio, and also possibly the vee angle. The minimum piston height will be based on the need to fit the ring pack above the piston pin, the need to control ring temperature, and the need to ensure structural integrity of the piston. In the diesel engine the need to incorporate the combustion chamber in the piston bowl will further impact the minimum height. These considerations will be further detailed in Chap. 15 on piston and ring pack development. The final variable determining the static deck height is connecting rod length. Reducing the connecting rod length reduces both rotating and reciprocating forces, but increases side forces (and thus friction and wear) and the second order component of the reciprocating forces. As the connecting rod length is reduced, a further consideration is counterweight-to-piston skirt clearance. This is shown with the BDC view in Fig. 8.11. As the piston approaches BDC the counterweights located at some of the crankcase throws swing an arc that approaches the piston. It should be noted that the relationship between counterweight and piston locations is dependent on both the engine configuration (number and layout of cylinders) and on counterweight design. The reader is referred back to Chap. 6 for a discussion of counterweight design and position. The minimum connecting rod length for counterweight-to-piston skirt clearance is then determined by the necessary skirt length. In most automobile engines today the skirts are cut back to allow shorter connecting rods and the resulting deck height reduction. In heavy-duty engines such cutouts are not used

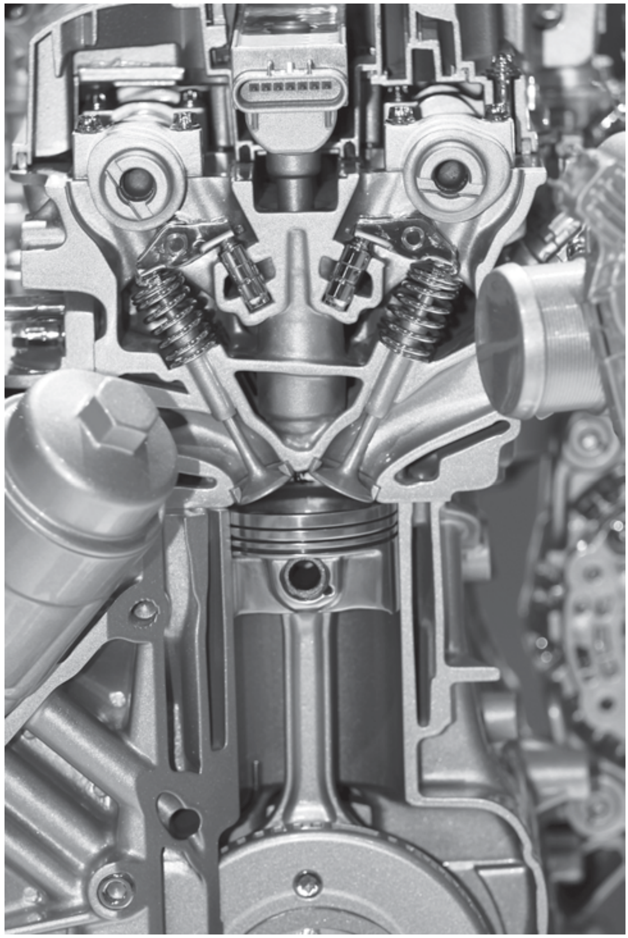


Fig. 8.12 Piston at TDC

in order to maximize skirt durability. Looking at the production engines in Chap. 1, those in Fig. 1.6a and c have skirts cut-outs and shorter connecting rods. The spark-ignition engine in Fig. 1.6b, and all of the diesel engines in Fig. 1.8 use full piston skirts and longer connecting rods.

The equation presented above is for all components in their static state. It is important to realize that while the engine is running, there are dynamic forces that will affect the piston-to-head clearance, and must be accounted for. There will be dynamic deflection of the crankshaft, connecting rod, and piston due to inertia forces. Also, the piston will rock in its bore around the piston pin due to secondary piston motion, and thus may cause the leading or trailing edge of the piston to be higher than the cylinder deck.

A related issue associated with deck height is the potential need to increase the stroke of the production engine at a later date. Referring again to Fig. 8.11, if the stroke is increased and the deck height is to remain unchanged, the connecting rod must be shortened or the piston pin must be raised in the piston. Since deck height is a key block layout dimension it must remain fixed in all but the lowest volume engines or the retooling costs will be huge. The least expensive approach is typically to raise the piston pin height, but this may not be possible. In summary, if there is any possibility that the engine's stroke will later be increased, design margin should be included in either the piston or the selection of connecting rod length.

8.3.2 Vee Angle

The decision pertaining to connecting rod length is further complicated in the vee engine configuration, based on the combination of desired angle between the banks and the chosen bore-to-stroke ratio. If both connecting rods in the vee share the same crankpin, near BDC the adjacent pistons come close to touching as shown in Fig. 8.9 previously. To address this, the vee angle may be increased or the deck height may be increased. Increasing the deck height moves the working portion of the cylinder bore away from the crank centerline, which increases connecting rod length for a given stroke. Finally, the bore-to-stroke ratio for a given displacement can be changed.

Since the engine block accounts for up to a third of engine weight, it is desirable to minimize the weight of this component. A more compact cylinder block with a short deck height, enables a lighter weight engine. Several combinations are shown in Fig. 8.13 and the critical clearances are identified. Larger bore, shorter stroke, wider vee angle designs generally package better in order to maintain acceptable deck height dimensions in vee engines. This can be seen in the figure, where each of the engines shown has the same displacement per cylinder. A larger vee angle drives a shorter deck height, for a constant bore and stroke, as shown in Fig. 8.14.

8.3.3 Cylinder Bore Spacing

In order to realize the most compact longitudinal cylinder block design, whether I-4 or V-8, it is important to minimize the cylinder bore spacing. In automobile engines the key limiting dimension is head gasket land width between cylinder bores. This can be influenced by cylinder liner choice, and will be discussed further in Sect. 8.5, Cylinder Design Decisions.

Another key decision is whether the cylinders will be separate or ***siamesed***, which means they are cast as a unit and there are no cooling jacket between cylinders as shown in Fig. 8.15. Separate cylinders will allow a coolant passage between cylinders, which improves cooling uniformity, but increases block length. Siamesed cylinders will increase block strength and reduce block length, at the sacrifice of cooling between cylinders which may increase thermal bore distortion.

The combination of main and rod bearing width and crankshaft web thickness are also cylinder spacing considerations, and their roles in defining the engine layout are seen in

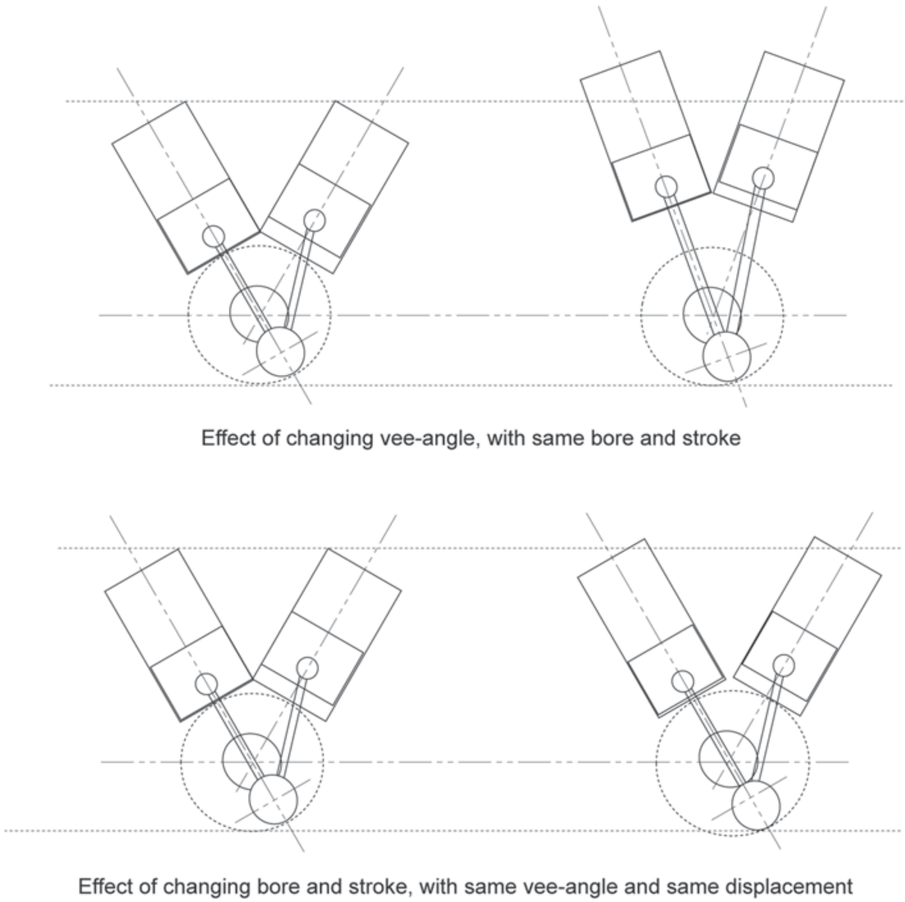


Fig. 8.13 The effect of vee angle and bore-to-stroke ratio on the deck height of a vee engine

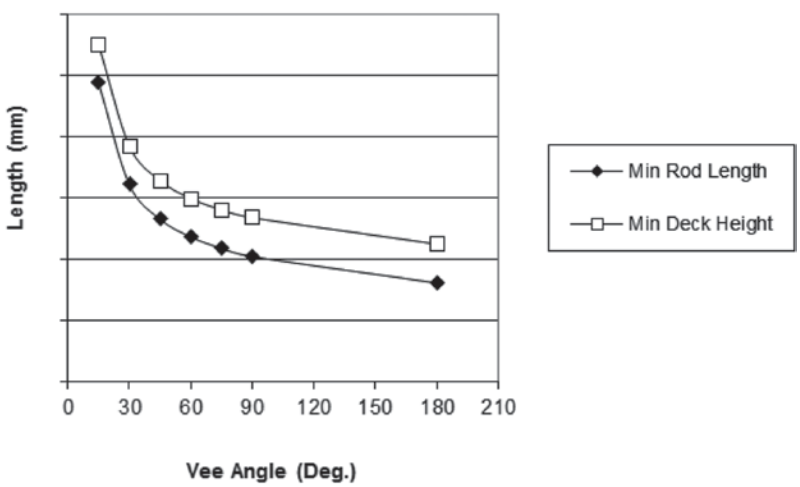


Fig. 8.14 Deck height as a function of vee angle for a constant bore/stroke

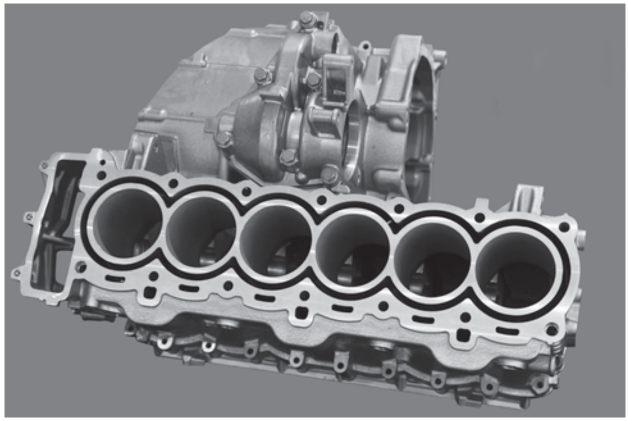


Fig. 8.15 Example of siamesed cylinders

Figs. 8.16 and 8.17. The bottom view of the cylinder block is shown for an in-line engine in Fig. 8.18. Note that two connecting rod bearings share a single rod throw on the vee engine crankshaft. One bank is therefore placed one rod bearing width behind the other from front to back. This can also be seen in the production V-8 engines shown in Figs. 1.6c and 1.8a. This benefits the deck height, as the vee can be narrower for a given connecting rod length since the bores are not directly in line. An additional manufacturing consideration is the need for the cylinder hone to run out below the cylinder bottom for complete machining, and clearance must be left from the tool to the crankshaft bearing bulkhead.

Another consideration in this layout of the engine is the need to include a thrust bearing surface at one of the crankshaft main bearings. While the majority of the loads seen by the crankshaft are perpendicular to the main bearing axis, some loading is seen along this

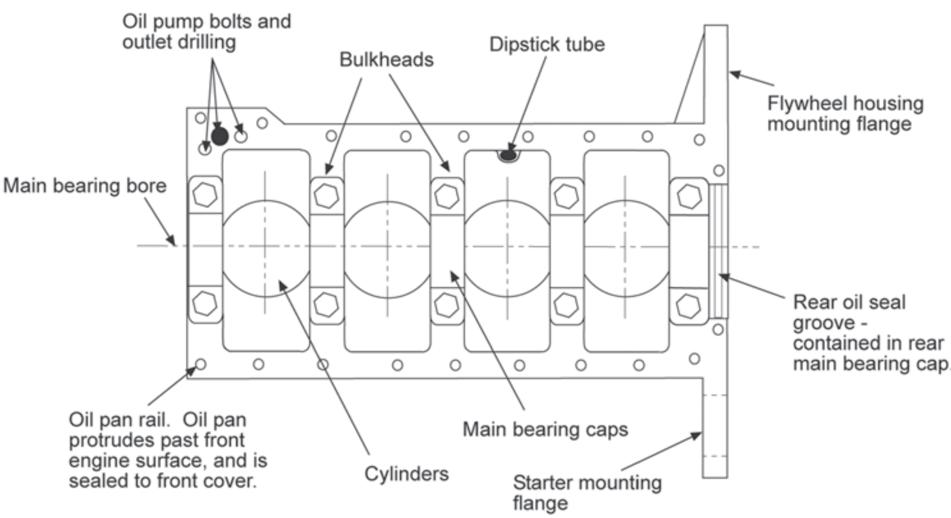


Fig. 8.16 Bottom view of the crankcase of an in-line engine

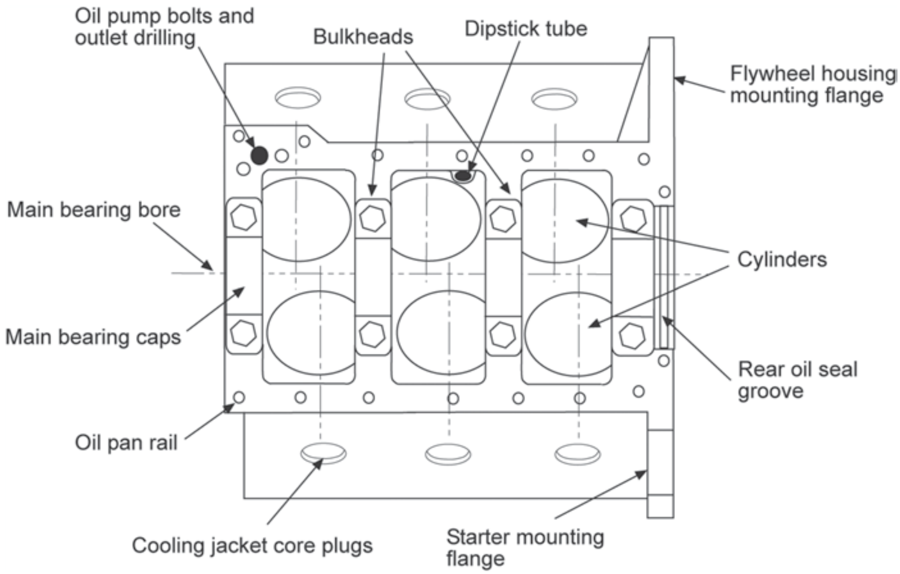
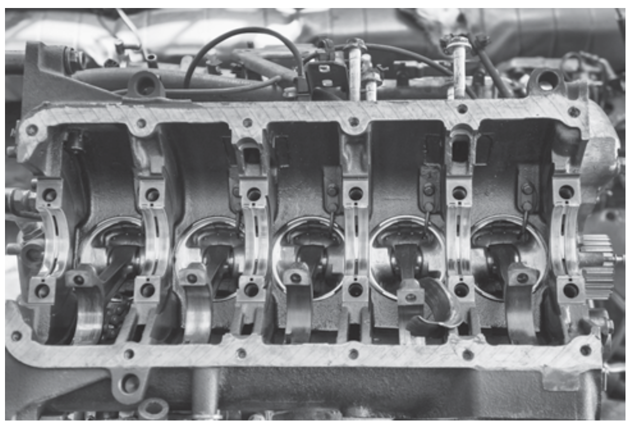


Fig. 8.17 Bottom view of the crankcase of a V-6 engine

Fig. 8.18 Bottom view of an inline engine



axis as well. The greatest load along the axis is seen when the clutch in a manual transmission application is disengaged. Other loads along the axis are contributed by the torque converter in automatic transmission applications, helical gears driven by the crankshaft (manual transmission gears, or gear-driven camshaft and accessory drives), bending of the crankshaft throw, block deflection, and dimensional misalignments. The thrust bearing is most often placed at the second main bearing from the rear of the engine. Loads along the crankshaft axis are highest at the rear of the engine, but the rear bulkhead must also accommodate the crankshaft seal. In engines having especially stiff crankshafts the thrust bearing may be placed at the center bulkhead. The thrust surface provides the fore and aft datum for crankshaft machining, and centrally locating this surface allows the crankshaft machining tolerance to be split evenly between the front and back cylinders, allowing closer dimensional control. The thrust bearing requires a wider main bearing, and if the journal bearing area is not to be compromised at this bearing engine length must be increased slightly. Finally, the front and rear of the crankshaft must accommodate oil seals. While looking at the bottom view of the engine layout it is important to consider oil pump and pick-up location as well. On any given engine the oil pump will be placed in a location convenient to be driven from the crankshaft. Although the pump location remains constant a given engine may be designed with the option of either a front or rear oil sump for different vehicle applications. If this is the case, a symmetric pan rail bolt pattern may allow a single oil pan to be used, with different pick-up tubes for the different sump locations.

8.3.4 Other Block Dimensions

Other critical clearances when laying out cylinder block are:

- Connecting rod cap and cap bolts to pan rails clearance throughout the stroke,
- Connecting rod to cylinder bore clearance throughout the stroke,
- Crankshaft counterweight to the cylinder bottom,
- Crankshaft counterweight to the piston bottom at BDC,
- Crankshaft counterweight to the operating oil level in a wet sump,
- Piston to piston clearance in a vee engine at BDC, and
- Piston to cylinder head clearance at TDC

When laying out the engine, it is important to consider the fact that under loading components will deflect, and this must be accounted for in the design. Additionally, components have machining tolerances and casting tolerances that may adversely stack up.

The minimum crankcase width at the pan rails is determined by the combination of the engine's stroke and the necessary counterweight size and connecting rod motion envelop. Counterweight diameter is invariably reduced by splitting the counterweight mass across a given connecting rod throw, as well as by wrapping the counterweight around a greater arc. As this is done a greater total crankshaft mass is required to achieve the required balancing. This was previously discussed in Chap. 6. When determining pan rail location, the designer must also consider whether there could be a need for increased displacement at some later time. If the stroke is ever to be increased this must be considered, and design margin must be allowed in the pan rail width dimension to ensure that the larger stroke crankshaft (and correspondingly larger counterweights and connecting rod orbit) will still clear the pan rails. The pan rails are often used for fixturing during block machining operations. With in-line engines the pan rails are generally the widest portion of the block, and the block machining line will be designed to clear the pan rails, but with little further margin. Both of these facts mean that increasing this dimension later would result in extremely expensive tooling modifications.

8.4 Crankcase Bottom End

Among the early crankcase design variations that may be considered is that of block skirt length, as depicted in Fig. 8.19. The extended skirt design adds weight to the block and may increase the overall engine height. However, it also allows the main bearing caps to be tied more rigidly to the remainder of the block, and it simplifies oil pan design and sealing as the entire oil pan sealing surface can be made flat. With the short skirts machining complexity is minimized if the oil pan mating surface is at the main bearing centerline so that the bearing caps are mounted on the same machined plane as the oil pan. This requires the oil pan to include semi-circular openings or angled surfaces (and resulting complex sealing surfaces) at its front and rear in order to clear the crankshaft. The long skirt design is typical in heavy-duty engines as shown in Fig. 8.20, while most passenger car engines utilize the shorter skirt design.

Another important consideration is that of main bearing cap constraint. Most automobile engines use two-bolt caps, an example of which is shown in Fig. 8.21. Also shown in the figure are various four-bolt cap designs. The parallel bolt design is commonly seen on high performance automobile engines with short block skirts. The cross-bolt design significantly increases cap rigidity and block stiffness, at the sacrifice of load transmission to outer surfaces potentially increasing NVH emissions. It is especially common in extended skirt vee engines, where the vee configuration results in firing forces having significant horizontal components. The angled bolt design allows similar cap rigidity with the shorter block skirts. However, with this bolt geometry, machining and dimensional control of the cap are especially difficult. Occasionally on inline engines, the cylinder head bolts go all the way through the block and anchor the main bearing cap. While making assembly and servicing more difficult this approach places the majority of the block in compression, improving management of firing forces. In Fig. 8.22, a ladder frame or bedplate bearing cap is shown. In this case the entire cylinder block splits along the crankshaft main bearing centerline, and a single piece replaces the individual main bearing caps. This design

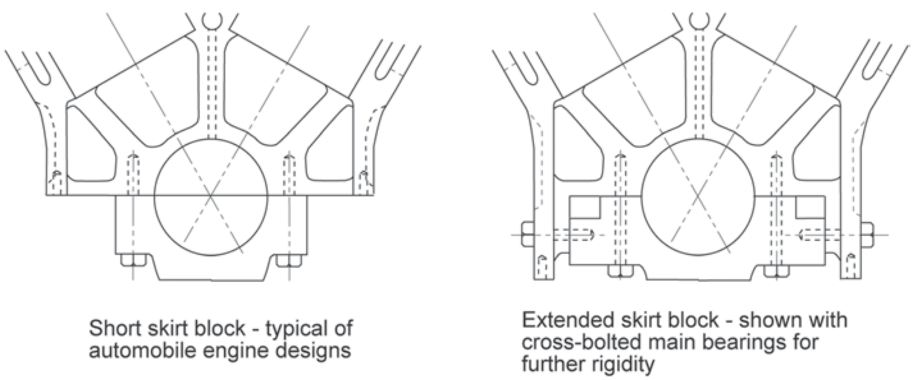


Fig. 8.19 Skirt design options with conventional main bearing caps

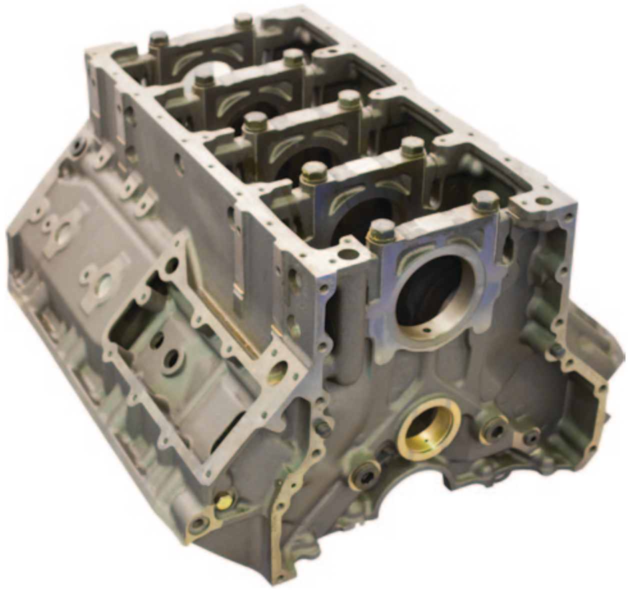


Fig. 8.20 Cross-bolted main bearing caps

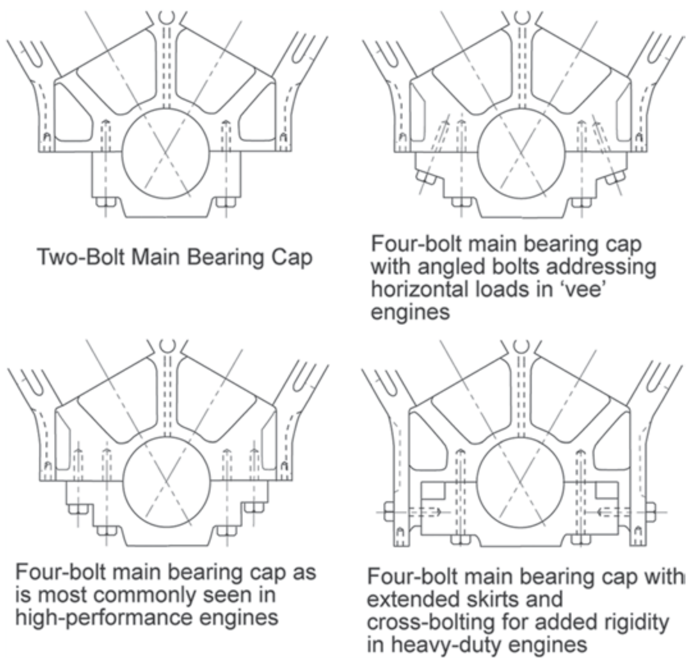


Fig. 8.21 Main bearing cap design variations

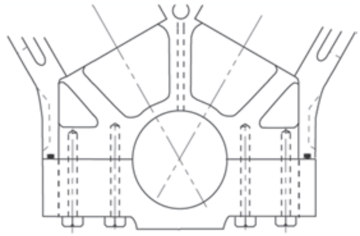


Fig. 8.22 Ladder frame construction for the crankcase of an automotive engine

significantly increases cylinder block stiffness, and minimizes or eliminates cap separation. However it can be difficult to seal because it requires the use of sealant as the use of a gasket would impact bearing clearance. It is especially common in engines with aluminum blocks where design modifications to increase stiffness are required. Examples of this design are shown in Figs. 1.6b and 8.24.

In blocks made of aluminum, the main bearing cap is frequently made of cast iron for strength and rigidity. Gray iron limits bearing clearance growth. Additionally, an upper main bearing insert made of cast iron is occasionally cast into the main aluminum engine block to form a composite casting. No matter what material they are made from, main bearing caps and crankcase main bearing bores are always machined as a set, and identification is usually provided to make sure each cap mates to its original bore upon reassembly.

The horizontally opposed engine requires significantly different crankcase geometry, and the basic features of two designs are shown in Fig. 8.23. In both of these designs the cylinder sections are cast separately and bolted to the crankcase. This is not absolutely necessary, but is typical in order to reduce casting size and complexity. One challenge with this design is tightening the connecting rod fasteners with the block assembled. The design on the right shows the entire crankcase splitting at the main bearing centerline. In the design on the left main bearing caps similar to those used with in-line and vee engines are used.

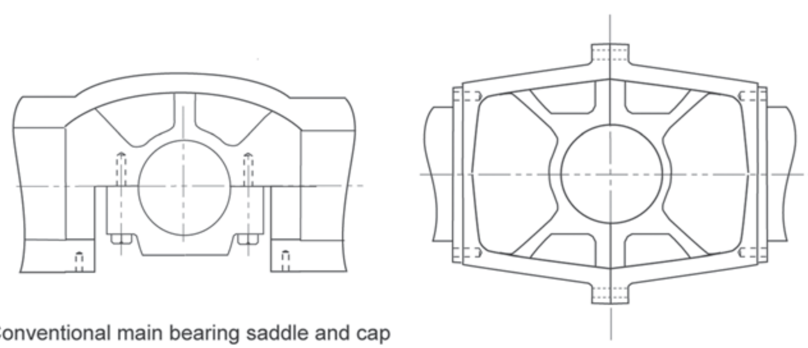


Fig. 8.23 Main bearing design options for a horizontally-opposed engine

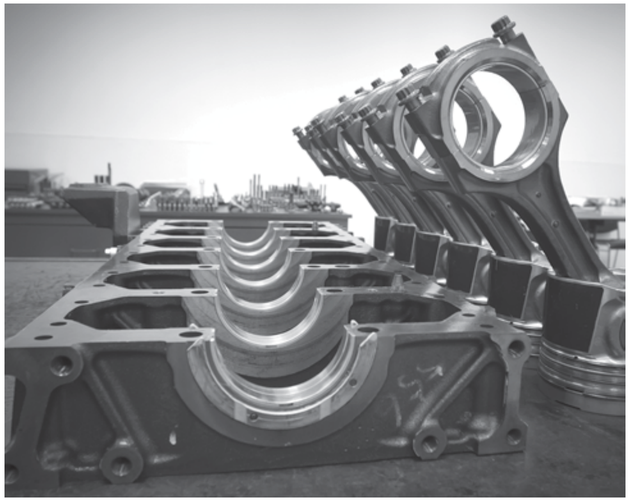


Fig. 8.24 Ladder frame construction in a medium-speed stationary engine

8.5 Cylinder Design Decisions

The discussion now returns to the cylinder section of the block. The goal of the cylinder liner is to provide a good wear surface for the piston rings, and to maintain cylindricity to aid piston ring sealing. If the liner deviates from round through machining, static head bolt loads, dynamic firing loads, or thermal stresses, the engine will experience higher oil consumption, higher blow-by, and higher engine friction. There are three general approaches to the cylinder liner: ***integral liner, dry liner,*** or ***wet liner***. Depending on the cylinder architecture, the block may be ***closed deck*** or ***open deck***. At the top of the cylinders the cooling jacket may be sealed with an upper block surface referred to as the ***firedeck,*** on closed deck engines. The cylinder head is then bolted to this surface. In open deck engines, this surface is eliminated—the cylinder head is sealed at the top surface of the cylinder walls and at the outer wall, and coolant makes direct contact with the cylinder head. Referring to the production engines introduced in Chap. 1, those shown in Figs. 1.6, 1.7, and 1.8b are all parent bore designs. The engines in Figs. 1.6a and 1.8b have open decks, while those of Figs. 1.6b, c, 1.7, 8.1, and 8.2 have closed decks.

8.5.1 Integral Cylinder Liner

For automobile engines the most common approach is the parent material bore or integral liner design, in which the cylinder walls are cast integrally with the cylinder block. This is most common in cast iron blocks and closed deck applications. The cylinders are surrounded by a cooling jacket, which in turn is surrounded by the outer wall of the block. This design minimizes the cylinder bore spacing, and shortens the total length of the block, as well as providing the shortest possible axial cylinder length. It has the further advantages of good heat transfer from the cylinder surface to the coolant, and reduced cylinder bore distortion compared to dissimilar materials used in other applications. An integral cylinder liner is the least expensive to manufacture, but offers the least flexibility for engine overhaul.

Alternately, a coating or thin film may be applied to the parent bore cylinder walls on an aluminum cylinder block to increase wear resistance. While running a parent bore liner may save as much as 0.5 kg per cylinder, the challenges include added cost and higher scrap rate due to adhesion problems. Specifically, coatings will not adhere to porosity exposed by machining of the parent bore.

8.5.2 Dry Cylinder Liner

A dry cylinder liner is one that is made from separate material than the block, and is inserted or cast into the crankcase. This technique can be used on both open and closed deck blocks. No contact with the coolant occurs, as this is completely encased in the parent block material. This reduces the opportunity for leaks, but reduces the heat transfer from the liner through the block to the coolant. A dry liner variation is the removable liner that cold be replaced during an engine rebuild. This approach was common in earlier truck and tractor engines but is seldom seen today. Controlling temperature is extremely difficult, leading to high piston ring temperatures and bore distortion.

If the block is cast from aluminum, cylinder liners made of a higher hardness material may form the running surface, with the block cast around these liners, as shown in Figs. 8.25 and 8.26. Material choices are typically iron, steel, or high silicon aluminum alloy, and it is critical to make sure the aluminum adheres to the liner so that proper thermal contact is maintained. This is achieved with outer surface features and by heating the liners to over 200 °C prior to casting the rest of the block around them.

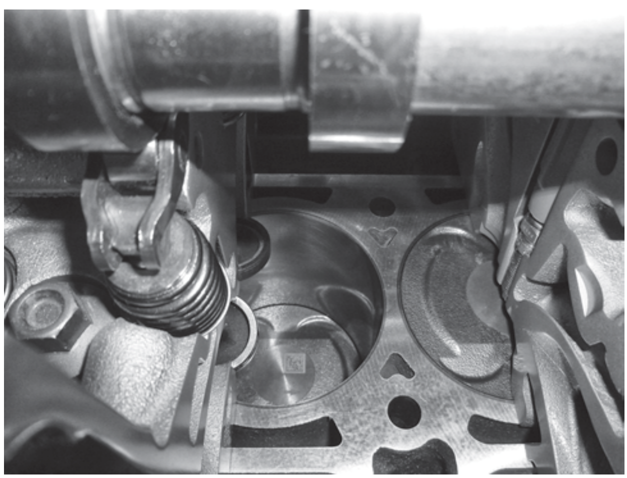


Fig. 8.25 Cast in cylinder liner

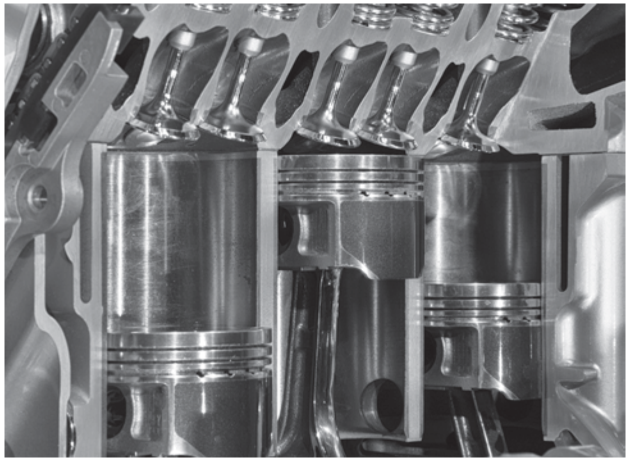


Fig. 8.26 Cast in cylinder liner

8.5.3 Wet Cylinder Liner

Most heavy-duty engines and some passenger car diesel engines in production today use wet liners, as shown by the engine in Figs. 1.8a, and those in Figs. 8.27 and 8.28. In such engines, the cylinder liners are pressed or slip fit into the surrounding structure such that they are in direct contact with the engine coolant, and sealed at the top and bottom. In order to provide sufficient structural rigidity, the block casting includes walls separating the cooling jackets for each cylinder. The wet liner engine holds the advantages of having the cylinders easily replaced when the engine is rebuilt, often with the engine in place in the vehicle, and direct contact of the cylinders with coolant in individual jackets for optimum temperature control. However the cylinder spacing and thus the overall length of the engine must be increased and the possibility of coolant leakage is increased. Care must be taken with dissimilar materials, and differential thermal growth will affect both the radial fit, and axial length of the cylinders. Due to manufacturing variation, the protrusion heights of the sleeves at the cylinder deck may be different and the head gasket must tolerate this. Alternatively, the top of the cylinders can be machined after installation, to guarantee the same protrusion height. Of course engine cost is also substantially higher than that of a parent bore design.

8.5.4 Cylinder Cooling Passages

The most significant thermal loading on the block is near the top of the cylinder, and between adjacent cylinders. Cooling jacket design will be covered in greater detail in Chap. 13, but a few observations should be noted here. In the case of the integral liner and dry liner engines the cooling jacket is an open passage surrounding all of the cylinders. Coolant supply and return locations are carefully optimized to achieve as even a flow distribution

Fig. 8.27 Wet liner and piston

around each cylinder as possible. The wet liner engine requires separate jackets surrounding each cylinder. These jackets must be fed from a separate header, increasing complexity but resulting in relatively even flow distribution. A simple design rule for gasoline engines is to size the coolant passages around the cylinder assuming that 10% of the fuel energy supplied the engine is rejected to the coolant at the cylinders as shown in Eq. 8.2.

$$Q_{coolant} = \frac{q_{heat}}{c_p \cdot \Delta t_{coolant} \cdot \rho_{coolant}} \tag{8.2}$$

Where:

$Q_{coolant}$	Volumetric flow rate of coolant (velocity * area)
q_{heat}	10% of fuel Lower Heating Value (LHV)
c_p	Specific heat of coolant
$\Delta t_{coolant}$	Desired temperature rise of coolant
$\rho_{coolant}$	Density of coolant

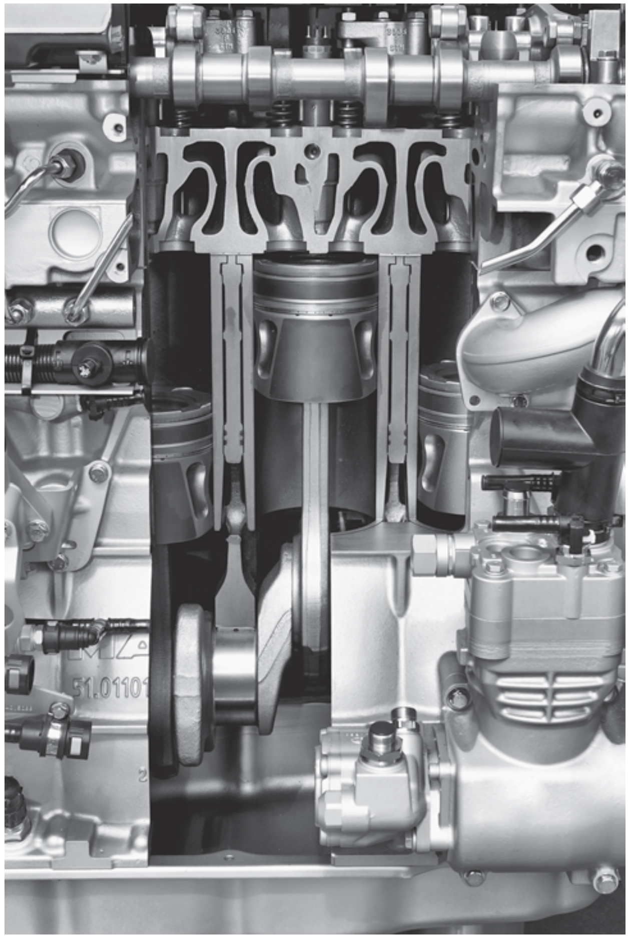


Fig. 8.28 Wet liner installed

The cross sectional flow area is the width of the coolant jacket multiplied by the height. The height of the coolant jacket around the cylinder ranges from 30 to 60 % of the piston stroke for an aluminum block and 50–100 % of the stroke for a cast iron block due to the difference in thermal conductivity. Cylinder liner temperature is typically highest near the cylinder head, as this portion is exposed to combustion. The combustion pressure and temperature are highest when the piston is near its TDC position. As the piston approaches BDC, temperature and pressure have dropped; this portion of the cylinder is exposed for a shorter period of the operating cycle. The coolant velocity can be obtained from pump performance data. It is important to keep the coolant jacket small for engine packaging, and also to keep the velocity of the coolant high. It is also beneficial to minimize the total amount of coolant in the engine, as this reduces engine warm up time and improves catalyst light-off.

If the cooling jackets are to be sand cast, core prints must protrude through the outside surfaces of the block to locate the core. This will be necessary both to hold the cores in place and to remove the core sand after casting. A lost foam casting will have the same requirements. It is general practice to make the resulting openings in the outside surface of the block circular. They are then machined and stamped steel ***freeze plugs*** are pressed in during block assembly to seal the cooling jackets. The open deck design identified earlier in this section allows the cooling jackets to be cored from the top surface of the block, and core prints or freeze plugs are not required. In engines designed with wet liners the cooling jackets are cored as part of the cylinder openings, and separate cooling cores are not needed. In these engines a water header may be cored along an outside surface of the block to feed coolant to the individual jackets around each cylinder.

Simplified top- and side-view layouts of cylinder banks for parent bore and wet liner engines are provided in Fig. 8.29. The layout in Fig. 8.29a depicts the key design features that determine the required length of the cylinder bank for the parent bore engine. In the previous section main and rod bearing width and crankshaft web thickness were identified as important parameters defining the required cylinder spacing; this can clearly be seen in the figure. Another important parameter is cooling jacket design. The jacket shown in the figure must be of sufficient width that the casting dimensions can be maintained in production. If the block has been sand cast or cast with the lost foam process, sand clean out must also be readily achieved. The casting thickness at the cylinder wall is determined by

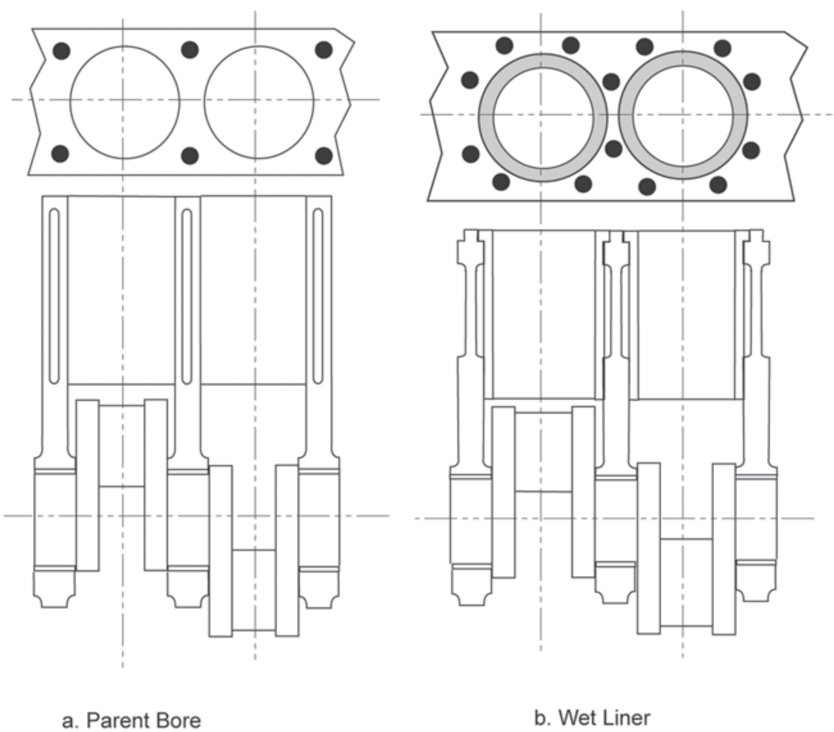
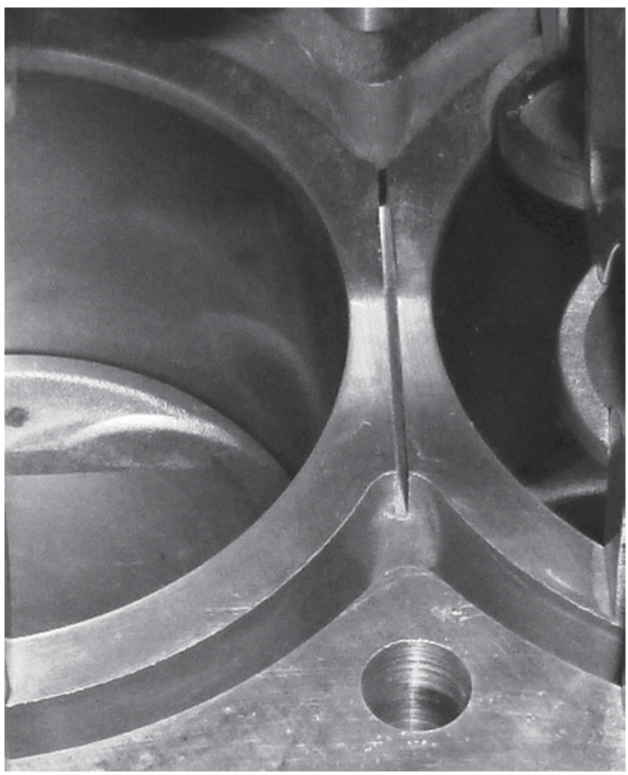


Fig. 8.29 Layout variables determining cylinder bore spacing in a parent bore and a wet liner engine

casting dimensional control, structural requirements, and by expectations that the cylinder may be re-bored during engine rebuild. In order to minimize cylinder spacing the cylinder walls may be cast siamese, assuming sufficient crankshaft bearing area can be maintained. The cooling jackets are placed only around the perimeter of the cylinder bank. In addition to reducing cylinder spacing the siamese casting increases structural stiffness, but the resulting temperature profiles must be carefully assessed. This may be improved by machining a slit or small cross drillings near the head gasket, to allow a minimal coolant flow between cylinders at the hot spot, as shown in Fig. 8.30. Finally, it should be noted that four head bolt bosses have surrounded each cylinder in the parent bore engine. This is typical practice in automobile engines (although a few examples can be found using five bolts per cylinder). The bolt bosses are placed at each corner, where they have little or no impact on either cylinder spacing or intake/exhaust port restriction. It is beneficial to move the threaded region for the head bolts deep in the block, below the firedeck, by using a long counterbore. This has the advantage of enabling longer head bolts, reducing the stress concentration of the threads, and reducing the amount that the thread distortion impacts cylinder roundness. It also moves the threads away from the firedeck, were thermal creep relaxation may be an issue.

Fig. 8.30 Slit machined between siamesed bores



The simplified top- and side-views are repeated for a wet liner engine in Fig. 8.29b. As with the parent bore engine the crankshaft webs and main and rod bearing widths are parameters that may limit the minimum cylinder spacing, but in this engine several additional factors must also be considered. It should be noted that even though these further factors lead to greater cylinder spacing the design may still be limited by the crankshaft parameters. This can be explained by the typically higher loading and greater durability expectations imposed on the wet liner engine. The outside diameter of the cylinder must include a step that positively seats the liner in the block. The step may be located either above or below the cooling jackets, but in either case adds to the spacing required between cylinders. As was previously described, the wet liners usually require an additional wall cast between each cylinder, with a cooling jacket on either side of the wall. The cooling jackets (or at least the portion between each cylinder) may have machined surfaces, allowing narrower passages and thus reducing the required cylinder spacing somewhat as compared to the cast jackets discussed in the case of the parent bore engine. The combination of bore diameter, peak cylinder pressure requirements, durability expectations (specifically head gasket life), and cylinder head stiffness determine the clamping requirements at the critical head gasket seal. With the larger bore diameters in combination with high load factors and long durability expectations it is typical to use between six and eight head bolts per cylinder. The head bolts must be relatively evenly spaced, so it can be expected that head bolt boss location may also impact cylinder spacing.

8.6 Camshaft Placement Decisions

While there is increasing interest in the "cam-less" engine, virtually all four-stroke engines in production today continue to use a camshaft and mechanical valve actuation. It thus continues to be important to consider where to place the camshaft in the engine layout. The trade-offs that go into determining camshaft placement will be detailed in Chap. 17, where the camshaft and valvetrain are discussed in detail. Briefly summarizing that discussion, high performance and high engine speed make overhead camshafts—placed above the valves, in the cylinder heads—attractive. In some heavy-duty diesel engines separate lobes on the camshaft are used to actuate the fuel injectors; in these engines high camshaft loading and high stiffness requirements also make overhead cams attractive. However, overhead cams result in complex drive systems, make the engine taller, and are typically more expensive than cam-in-block systems. The remaining discussion in this section will address the details of camshaft placement in a cylinder block, as is typical with overhead valve (OHV) engines.

The first consideration is that of where to place the camshaft within the block. If the camshaft is to be placed in the block it will be desirable to place it as close to the crankshaft as possible in order to simplify the drive system. The simplest drive system is a direct gear coupling, with a gear on the nose of the crankshaft meshing directly with one having twice as many teeth on the nose of the camshaft. As camshaft placement moves further from the crankshaft the gear diameters increase until their sizes become prohibitive and intermediate gears or a belt or chain-drive is substituted. With in-line engines the camshaft will be placed above the crankshaft along one side of the block. With vee engines it is most convenient to use a single camshaft placed in the vee between the two banks of cylinders. Whereas the camshaft placement in the in-line engine allows direct gear drive, placement in the vee necessitates greater distance between the camshaft and crankshaft, generally requiring one of the alternative drive systems. A few exceptions should be noted: First, as a means of gaining some of the increased stiffness and reduced inertia of the overhead cam engine while maintaining a simpler drive system, the camshaft may be placed near the firedeck to keep the pushrods short. Second, on some vee engines separate camshafts are used for each bank of cylinders. In these cases the cams are placed on the outside of the block as there will not be sufficient space for two camshafts in the vee.

The camshaft is designed to spin in a series of journal bearings under fully hydrodynamic lubrication. The bearings are placed at the same positions relative to the front and back of the engine as the crankshaft main bearings. This allows the same bulkheads to carry the bearings, and in many cases allows the same oil cross-drillings to be drilled to reach both the main bearings and the cam bearings. Unlike the crankshaft bearing the camshaft bearing shells are continuous, and are pressed into the journals. This requires that the radius of each bearing be made larger than the radius of the cam lobe at maximum lift, such that the camshaft can be installed from the front or back of the engine. It is important to consider possible future valve lift increases and cam lobe radii for future engine performance enhancements.

The camshaft bearing bore is cut through the entire length of the engine. In most engines the camshaft is installed from the front of the engine, and the bore is sealed at the rear, typically with a pressed-in plug. In a few cases the camshaft is driven from the rear of the engine and the installation is reversed. The camshaft must be held in place against fore and aft motion. This is most often accomplished with a bearing plate placed behind the cam gear. The plate is then bolted to the block surface through access holes in the gear, and provides a thrust bearing surface against fore and aft motion.

The cam lobe acts on a solid tappet, a hydraulic lifter, or a roller follower. Each tappet or hydraulic lifter is clearance-fit into a bore in the block. Pressurized oil must be supplied to each tappet bore, both for lubrication of the part sliding in the bore and in the case of hydraulic lifters, supplying oil for valve lash adjustment. Lifter design will be further covered in Chap. 17. Pressurized lubricant is also supplied to the cam bearings. Through drainback from these locations, and other locations above the camshaft, enough lubricant must be supplied to provide a hydrodynamic film at the cam lobe interface to the lifter. However, it is also important not to “bathe” the camshaft in oil as this would significantly increase friction as well as foaming of the oil. It is therefore important to provide sufficient openings below the cam lobes for rapid drainback to the crankcase.

8.7 Positive Crankcase Ventilation

If the example of a single cylinder engine is considered, the movement of the piston up and down in its bore can be seen to significantly change the volume in the crankcase cavity below the piston from BDC to TDC. If the crankcase were completely sealed, this cyclical volume change would have little effect on total work over the 4-stroke cycle, as the crankcase volume would act as a spring absorbing work and then returning it. Once a second cylinder is added to the same crankpin, two different scenarios can play out depending on vee angle. If the vee angle is very small (0–45°) this change in crankcase volume is magnified, if the vee angle is large (between 90 and 180°) the effects start to cancel as air and oil mist are transferred back and forth between cylinders across the crankcase cavity. As additional pairs of cylinders are added to form a V-4 arrangement, the movement of air is driven from one pair of cylinders in a bay to the other. As further engine cylinders are added to the vee arrangement, the percent change in crankcase cavity volume from TDC to BDC is reduced. The transfer of air and oil mist across main bearing bulkheads is termed ***bay-to-bay breathing.*** As engine speed is increased the restriction to flow between the bays becomes increasingly important, and needs to be considered for its impact on parasitic losses.

Returning to the single cylinder engine, the crankcase is not actually perfectly sealed. The cyclical crankcase pressure will blow oil past the crankshaft seals near BDC, and will draw air past the piston rings near TDC. The flow of air past the rings depends on what the crankcase cavity pressure is with respect to pressure above the piston rings. As pressure builds in the combustion chamber, a percentage of combustion gases will leak past the piston rings into the crankcase cavity below the piston, which is known as piston ring ***blow-by***. Over repeated cycles of combustion, positive pressure will eventually build up in the crankcase cavity. The solution to this issue is a one way check valve that allows ventilation of the gases that build up in the crankcase, known as the ***positive crankcase ventilation* (PCV)** valve.

In addition to blow-by gases in the crankcase, the mass movement of air within the crankcase entrains oil mist in the air, and adds hydrocarbons to the blow-by gasses. Early engines allowed this positive crankcase pressure to ventilate to the atmosphere. However, this combination of gasses is now considered an environmental pollutant. To limit the amount vented to atmosphere, these gasses are now typically vented back into the intake tract of the engine, so that they may be combusted prior to being released. This lessens the environmental emissions, but creates a new problem as the presence of oil in the intake air can reduce knock margin. However, this still releases some emissions to the atmosphere, hence efforts to make the piston rings as tight as possible to prevent blow-by, and devices known as air/oil separators are used to filter as much entrained oil from the air as possible.

Typically, these devices to separate the air and the oil have a torturous path that the air can navigate but the heavier oil cannot. Sometimes a coalescing media or sponge-like material is used to help separate the two by causing the oil entrained in the air to condense, and drain back to the sump. Usually air/oil separators are plumbed into a still cavity in the oil deck or valvetrain cavity of the cylinder head to isolate this flow from the more turbulent crankcase area, and to allow maximum time for the oil to separate from the air on its own. This drives the need for communication passages between the crankcase cavity and the valvetrain cavity of the cylinder head. For convenience, the oil drain backs from the head are typically used for this function also. It is important to consider the non-steady state air flow that will be moving back and forth in these passages, when sizing the diameter. As oil attempts to drain from the cylinder head, positive crankcase pressure may blow it back into the head if the passages are sized too small.

As the piston moves through its stroke, the piston and rings may move opposite and out of phase with each other due to piston motion or gas flow into and out of the piston rings. This relative movement of the rings is known as ring flutter, and significantly affects piston ring sealing. Drawing a slight vacuum in the crankcase cavity helps to stabilize the rings against the bottom of their grooves in the piston, and improves sealing. This improved sealing also improves in-cylinder emissions, as it is more difficult for combustion gases to flow into and out of the piston crevice volumes between the rings, which leads to unburned hydrocarbons.

8.8 Recommendations for Further Reading

The following paper is an excellent recent example of cylinder block design and analysis. The emphasis is on weight reduction through material removal, and is intended to be applicable regardless of the base material selected (see Osman 2012).

Originally published in German, the AAM-Applications series made available through Verlag Moderne Industrie includes sect. 1 on Powertrain. The focus is on aluminum parts, and includes excellent material regarding design approaches to the cylinder block, cylinder liners, pistons, and cylinder heads. English translations are available on the internet. At the time of this writing the articles could be found at the following link: http://www.pdfengineeringbooks.yolasite.com/resources/Aluminum_Applications_Power_train.pdf

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