Chapter-3

Cylinder heads, Cylinders & liners

Most modern automotive engines have all of their cylinders and the greater part of their crankcase poured in a single casting, so that cylinders and crankcase form a single unit. However, cylinders and crankcase perform different functions.

Separate Vs. Integral Cylinder Heads.

Cylinder heads now almost always are made separate castings, which are secured to the cylinder block with studs and nuts, with a gasket in between to ensure a gas-tight joint. The cylinder head can be cast integral with the block, and at one period in engine development that was the predominant practice.

With integral cylinder heads there is, of course, no machining of joint surfaces and no need for a gasket, but the cylinder casting is much more difficult to produce, and. besides, with the design which was usually employed, cooling of the combustion-chamber walls was less effective-the wall temperature of each combustion chamber being less uniform-than in an engine with a detachable head.

In the case of L-head engines with integral cylinder heads, the valves were introduced through openings in the head which were closed by threaded plugs generally referred to as "valve caps." These plugs presented to the hot gases in the cylinder a considerable surface which was not water-cooled, and which therefore formed "hot spots." It was customary to screw the spark plug into one of these "valve caps." Since the insulator of the plug naturally is a poor conductor of heat, and the additional threaded joint also formed an obstruction to heat flow, this further aggravated the situation with respect to "hot spots" and made it necessary to keep the compression quite low.

With the valve-in-head type of cylinder there are two alternate designs of integral heads. With one of these, exemplified in Fig, 1, the valves seat directly on the metal of the head, but this has the disadvantage that when they are to be reground, the whole block has to be removed from the car. With the other, use is made of so-called valve cages, that is, cylindrical sleeves which are set into bores in the cylinder head and retained therein between a shoulder and a ring nut. The valve seat is fom1ed on the inner end of the cage, and there is a port in the wall of the latter through which the gases flow from or into a valve passage cast in the cylinder head. The objection to valve cages is that they add another "joint" to the path for heat flow from the valve head to the jacket water, and therefore result in higher valve temperatures (particularly of the exhaust valve), which promotes detonation and makes the construction unsuitable for high speed, high-compression engines.

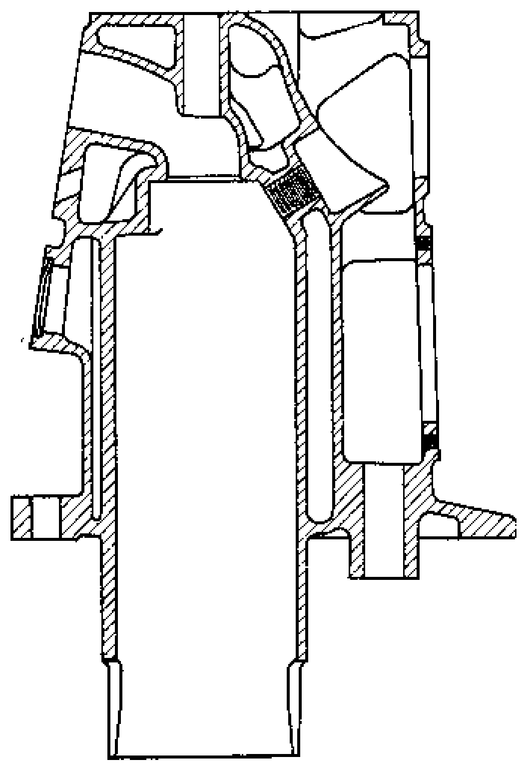


Fig.1. Cylinder with integral head

When the cylinder head is a detachable casting, the cylinder and jacket cores can be more securely supported in the mold, and the cylinder castings are likely to be more nearly true to pattern, with the result that after the cylinder is finished, its walls will be more nearly uniform in thickness.

With an engine having a removable head it is possible to thoroughly clean the combustion chamber of carbon, by scraping, after the head has been removed. If it is desired to locate the valves in the head, they may be seated directly on a water-cooled surface.

One reason for the continued, limited use of integral heads is that they avoid trouble due to distortion of the upper or outer end of the cylinder bore due to the drawing up of the cylinder-head retaining nuts. Such trouble is experienced occasionally, with detachable cylinder heads (blow-by past piston rings, leakage past valves, and excessive oil consumption), but it can be guarded against by performing the final finishing operation on the bore with a dummy cylinder head in place~ This produces a bore which is true when the retaining nuts are tightened.

Gaskets

Copper-Asbestos Gaskets.

Separate cylinder heads were rendered practical by the introduction of the copper-asbestos gasket. This consists of an asbestos sheet cut or stamped to the required form, which is armored with thin sheet copper. There is a copper sheet on each side of the asbestos sheet, and the two copper sheets lap along the outer edges of the asbestos sheet, so that the latter is completely encased. Copper grommets are inserted in the waterway openings and sometimes also in the combustion-chamber openings. In heavy duty engines the combustion-chamber grommet of the gasket may be reinforced by a copper-wire loop or a copper washer. In these copper-asbestos gaskets the copper provides the tenacity and the asbestos the compressibility needed in a packing. A gasket for a four-cylinder L-head engine is shown in Fig.2.

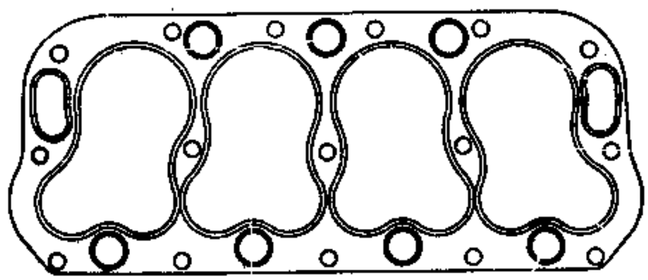


Fig. 2. Cylinder-head gasket for four-cylinder engine.

Steel-Encased and Other Gaskets.

Cylinder-head gaskets are made also of asbestos sheet encased in steel instead of copper. Cold-rolled, deep-drawing steel is used, and is rust-proofed to prevent trouble from corrosion. Among the rust-proofing processes applied to sheet steel for gaskets are tinning, electro-galvanizing, and terne-plating. Steel, being harder, does not have as good sealing properties as copper, and a sealing coat of some heat-resistant, non-hardening material is generally applied to the gasket, either in the manufacturing process or during installation. The edges of the steel sheet, of course, are not rust-proofed, and some steel-encased gaskets are fitted with copper grommets at the waterways. The principal advantage of steel- over copper-encased gaskets is that the production cost of the former is about 20 per cent less.

Another type of gasket comprises a central steel core with a layer of .coated and graphited asbestos on each side thereof, the asbestos being bonded to the core by means of integral steel tangs clinched into it. These gaskets, which are used chiefly in the engines of low-priced passenger cars, generally are provided with steel grommets at the combustion-chamber and waterway openings, one manufacturer is using a cylinder head gasket consisting of a sheet of SAE No. 1010 steel 0.015 in. thick, which is corrugated around the openings therein, including those for the cylinder-head studs. The corrugations have a spring action. and the sealing properties of the gasket are further improved by applying a coating of a heat-resistant lacquer to both sides.

Cylinder-Head Studs.

To obtain a gas-tight permanent joint with a cylinder-head gasket it is necessary to make provision for an adequate number of studs distributed as nearly uniformly as possible. With L-head cylinders from 16 to 20 studs are used for a four-cylinder block, from 24 to 26 for a six-cylinder, and from 30 to 32for an eight-cylinder. With , valve-in-head cylinders only two rows of studs are required, instead of three, and the total number therefore is less, viz., 12 for a four-cylinder block, 16 for a six- cylinder, and 20 for an eight-cylinder. To prevent distortion of the casting by drawing up the nuts, there must be plenty of metal in the bosses for the studs, and the studs must not be too near the valve seats. In the design of the heads careful attention must be given to the avoidance of pockets which might form steam traps. It is not necessary to use very large water ports. Moderate-sized ports judiciously distributed, are better, as they make it easier to prevent leaks.

Cylinder Material.

In the past automobile-engine cylinders have been generally cast of close-grained gray iron approximating the following composition.

	Percent
Silicon	1.9 to 2.2
Sulphur	not over 0.12
Phosphorus	not over 0.15
Manganese	0.6 to 0.9
Combined carbon	0.35 to 0.55
Total carbon	3.2 to 3.4

The SAE has standardized five grades of cast iron, of which four are recommended for cylinder blocks and cylinder heads as follows: No. 111 for small cylinder blocks; No. 120 for cylinder blocks generally. No.121 for truck and tractor-, and No. 122 for diesel engine cylinder blocks. Pistons also are cast of these irons.

It was determined from tests conducted, that to obtain the better physical properties the total carbon & silicon contents must be reduced and the phosphorus content held to a lower limit.

Among other points usually covered in specifications for cylinder castings arc the following: Castings must be smooth, well cleaned and free from shrinkage cavities, cracks and holes, large inclusions, chills, excess free carbides and any other defects detrimental to machinability, appearance, or performance. They must finish to the size specified. When tensile tests are provided for, the portion of the casting from which the test piece is to be machined is usually specified. .

The use of steel for cylinders has often been suggested, and for racing and aircraft engines, cylinders are sometimes made from hollow steel forgings. Several American manufacturers use cylinder castings of semi-steel, more properly called high-test cast iron. This material is made by adding a certain percentage of scrap steel to the melt of cast iron, which results in a finer grain and in somewhat better tensile properties.

To make it possible to successfully cast a multiple-cylinder block with thin walls, the iron must pour well and have a "long life" (as the foundry men call it). These characteristics are strengthened, by high phosphorus content, but, unfortunately, this element tends to make the iron soft and less resistant to wear.

Nickel-Chromium irons.

Certain iron ore mined in Cuba contains small percentages of nickel and chromium, and the metal made from this are, known as Mayari iron, is sometimes added to gray iron for cylinder castings: Mayan iron therefore is a natural alloy. It is claimed that it is free from oxidation & has a lower solidification point, and that the "longer life" of the iron improves the "feeding" of castings when they are properly gated, in spite of low phosphorus content. Castings when sectioned -show sound metal even where there are heavy bosses and thick sections. Cylinder castings made of a mixture containing 10 per cent of Mayari iron showed a tensile strength of 36,740 psi, according to makers of the iron; a transverse strength of 4250 lb, and a Brinell hardness of 223-229. The same iron is also used for cylinder heads and pistons. Results similar to those from Mayari iron are being obtained by the addition of small quantities of nickel and chromium, and such alloy irons are now used not only for cylinder blocks, but also for pistons, particularly for heavy duty, commercial-vehicle engines.

The chief advantage of alloyed irons is that they possess greater hardness and wear resistance, and that without being harder to machine. The machinability of grey iron is dependent upon the absence of excess iron carbide of chilled or hard spots. Nickel acts to eliminate both, and so to improve machinability. In many cases the alloyed iron, although having a Brinell hardness from 30 to 40 points greater, is actually easier to machine than ordinary gray iron.

When nickel is used alone as an alloying element, the content usually ranges between 1.25 and 2.5%, whereas if it is used in combination with chromium, the nickel content ranges between o.50 and 1.50 % and that of chromium between 0.25 and 0.50 % it is claimed that a combed content of nickel and chromium of 1 per cent will give cast iron with a Brinell hardness of 207-217; of 2 per cent, 223-235, and of 3 per cent, 241-255.

Chromium and nickel, however, are not the only alloying elements purposely added to cylinder irons; others added to improve the fluidity of the molten iron, the resistance of the iron to wear, its machinability, or both of the latter qualities, include, molybdenum, vanadium and titanium.

Copper and Molybdenum Additions.

Copper is of value in cylinder irons in that it tends to prevent chill in thin sections and to give a finer grain structure in the heavier sections, thus acting the part of a stabilizer, It also increases the fluidity of the iron and acts as a "graphitizer"; it hardens and tightens up the matrix so that “sponginess” is reduced. The improvement due to copper is well shown in transverse tests, and these additions are particularly effective in the presence of high manganese and of nickel or chromium.

Molybdenum increases the resistance to wear of cast iron, especially at higher temperatures. This results from the refining action it has on the grain, and from the finer division of graphite which it brings about. It increases the Brinell hardness-although in this respect it is not as effective as an equal proportion of chromium and it accomplishes this without rendering the metal less machinable. It also increases the tensile strength and the toughness of the metal. Where there is a tendency for the castings to crack owing to faults in either the design or the foundry technique-molybdenum is often of benefit. It is mostly used in combination with either chromium alone or with both nickel and chromium.

Heat Cracks in Cylinder Walls.

Cracks in L-head cylinder castings (especially in large ones) sometimes start at the sharp edge formed by the cylinder bore and the valve-passage wall. This edge reaches a very high temperature, because the hot gases pass over it during the exhaust period, and a crack naturally starts easily at a sharp edge. Rounding off this edge has been found a good preventative against heat fatigue cracks. Cracks may start also at either the inlet- or exhaust-valve seat. It was shown that such cracks usually are the result of pre-ignition. The latter causes local overheating of the combustion-chamber wall, and the crack forms when the overheated metal cools again. By installing a "hot" spark plug in one cylinder and then running the engine under full load at from 3000 to 3500 rpm, cracks could be produced at will. The "hot" plug causes pre-ignition, and usually one 10-minute run under these conditions resulted in the formation of a crack, though sometimes several such runs were required.

Cylinder Wear.

The characteristic which is most important in judging cylinder irons is their resistance to wear under engine- operating conditions. As the cylinder bore wears, the engine loses power, consumes excessive quantities of oil, and gives off smoke in the exhaust. In fact, the rate of oil consumption is usually taken as an index of the state of wear of the cylinder bore.

It was observed many years ago that the wear of cylinder bores is very non-uniform. It is greatest at the top end of piston travel (under the topmost ring with the piston at the end of its up-stroke), and decreases rather rapidly from there down. (Fig. 3.) It has been pointed out that cylinder wear is due to three separate causes, viz.,

- Abrasion, which is due to foreign particles in the oil film;
- Erosion, which is due to metal-to-metal contact between the cylinder wall on the one hand and the piston and rings on the other; and
- Corrosion, which results from chemical action on the cylinder walls by the products of combustion.

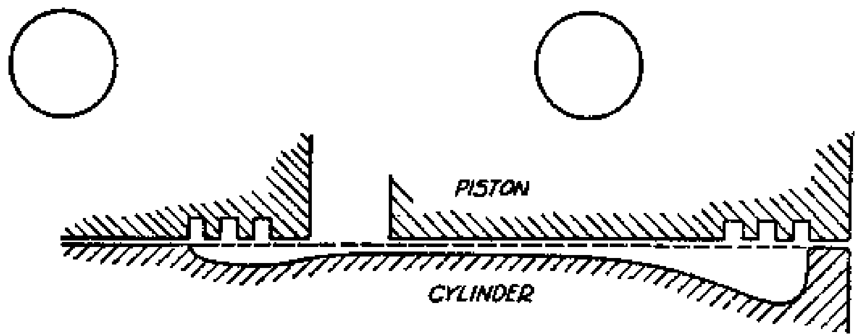


Fig. 3. Diagram showing distribution of cylinder wear.

The order of importance of the three causes varies with conditions of operation. That corrosion may play an important part in the wear of cylinder bores, it was found that accelerated cylinder wear occurs at low cylinder temperatures and is attributable to corrosion resulting from deposition of acid-bearing moisture on the cylinder walls. The reasons for assuming corrosion to be responsible were briefly as follows:

1. The pitted and discolored appearance of the cylinder walls and piston rings after low-temperature operation.
2. The fact that increased wear begins just below the calculated dew point.
3. The detection of acids in the water of combustion.
4. A large reduction in the rate of wear obtained with hydrogen fuel.
5. A reduction in wear obtained when using corrosion-resisting materials.

The research work showed that corrosion is largely due to carbonic acid formed by the solution of carbon dioxide, a product of combustion, in water condensed from the gases of combustion. When hydrogen is used as fuel there is no carbon dioxide in the exhaust, so that no carbonic acid can form.

Effect of Cylinder Material on Rate of Bore Wear.

The result of the Brinell test is generally regarded as bearing some relation to the rate of cylinder wear. That hardness is a factor in wear resistance is indicated by the fact that heat-treated liners of alloyed iron with a Brinell hardness of slightly over 500, have been found to require reconditioning of the bore (by re-grinding) only one

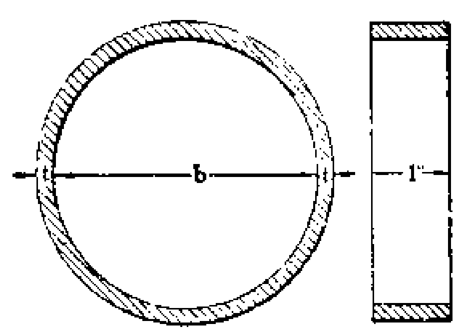


Fig. 4. Cylinder-wall-strength diagram.

third as often as the bores of gray-iron cylinder blocks with a Brinell hardness of around 200. Cylinders with soft or "porous" spots which are readily detected by the Brinell test, usually show a high rate of wear, but differences in hardness within the usual range specified for gray-iron cylinder castings, say. 180 to 230 Brinell, have little effect on the resistance to wear.

Cylinder Stress and Wall Thickness.

With the usual compression ratio of between 7 and 8 (for passenger-car engines) a maximum explosion pressure of about 700 psi may be figured with. Now consider a section of a cylinder of b in. bore and 1 in. long, as represented in Fig. 4. The pressure developed in the cylinder by the explosion tends to rupture the wall along lines parallel with the cylinder axis and at opposite ends of a diameter. With a maximum combustion pressure of 700 psi the rupturing force on the section of the cylinder considered is 700b lb. If the wall has a thickness t and the material has a tensile strength of 35,000 psi, the resistance to rupture of the two sections 1 in. long and t in. thick is 70.000t lb and the. factor of safety then is

$$f = 70000t/700b = 100t/b$$

For a factor of safety of 4 the ratio of wall thickness to bore then evidently must be 1/25

This rule when applied to cylinders of small bore gives values for the cylinder-wall thickness which, while large enough so far as withstanding the stresses of a normal explosion is concerned, would be too small from the standpoint of shop production. If the water jacket is cast integral, as it usually is, the cylinder can be machined only on the inside, and the minimum thickness of the wall then depends upon the accuracy with which the cores are set. Some allowance must be made for inaccurate core work, and a good value for the wall thickness is

$$t = (b/25) + 0.10\text{in}$$

This formula can be safely applied to the whole range of sizes of automotive engines with cast-iron cylinders.

The cylinder head must be quite stiff in order to resist the stresses of detonation. The wall itself is usually made slightly thicker than the cylinder wall. In the case of an overhead-valve engine, the Wall is normally stiffened by the vertical walls of the valve pockets. A similar stiffening effect is usually obtained in the heads of L-head cylinders from the walls of spark-plug wells, but if there are any extended flat surfaces in these heads, they should be stiffened by ribbing.

Details of Water Jacket.

For a long time it was the general practice to extend the water jacket down the cylinder wall only to the level of the top of the piston when at the bottom of the stroke. As the lower part of the cylinder is not contacted directly by the hot gases, it does not reach an excessive temperature, and therefore does not seem to require water-jacketing. However, in modern high-speed engines the crankcase oil often reaches an excessive temperature, which reduces the load-carrying capacity of the oil film in the bearings, and may cause the latter to fail in hard service. It has been found that by extending the water jacket all the way down the cylinder, the temperature of the oil in the crankcase under extreme conditions may be lowered by as much as 50 Fahrenheit degrees, as compared with an engine with "half-length" jackets, and "full-length" jackets have come into general use.

Some designers taper the jacket down from the top to the lower end, so as to place a larger body of water around the compression chamber, where most of the heat must be absorbed. In most engines, however, the depth of the water jacket is uniform from top to bottom. This depth varies somewhat in different designs, but usually is equal to about one-eighth the cylinder bore. Certain parts of the jacket which directly affect the over-all dimensions of the block can be made smaller in depth, including the space between adjacent cylinders and that between a cylinder and a valve pocket or a tappet housing. Liberal water spaces have the advantage that the core sand can be more effectively removed from the casting. In engines of special design, such as those with "wet" liners, the jacket depth can be made less.

The jacket wall generally is made as thin as the foundry process permits. It can be made thinner, of course, in a small cylinder than in a big one, because in the former the area is smaller. Average practice with regard to jacket-wall thickness is as follows:

Cylinder bore, inches	3	4	5	6
Thickness of jacket wall inches	5/32	3/16	7/32	1/4

Jacket walls must be made heavier when cylinder liners (especially the "wet" kind) are used and the tensile stresses due to the force of explosion are sustained chiefly or wholly by these walls.

On the cylinder head the water jacket is usually made of somewhat greater depth than around the cylinders, so as to provide adequate heat-storage capacity over the area where most of the waste heat enters the cooling water. There should be water spaces between all adjacent valve pockets (instead of common walls), and the water should come quite close to the valve seats, as it is only in this way that uniform cooling of the valve seat can be assured, and distortion and consequent leakage prevented. Cylinder heads must be so designed that no steam pockets can form in them; that is, it must be possible for the water to flow from any part of the jacket to the outlet along a continuously rising path. Trouble from overheating is most likely to arise at the exhaust-valve seats, and it is therefore desirable that the cooling effect of the circulating water be most intense at the valve pockets. This can be assured by inserting a distributing header in the water jacket, the header connecting with the water entrance to the jacket at the front of the block and having an outlet adjacent to each exhaust-valve pocket. The header is usually made of sheet metal and set into the mold. Two arrangements are illustrated in Fig. 5.

With valve-in-head cylinders the location of the water outlet presents some difficulty: because the valve mechanism on top of the engine is usually provided with a cover. One solution of the problem consists in forming a number of outlet bosses on the head over to one side, so they come outside the valve cover, and using a water-return manifold. While this tends to promote uniformity of circulation, it makes for dissymmetry of appearance, which is the more objectionable because the manifold is located very prominently on top of the engine. The more common plan is to have an outlet at the front end of the head, just outside the valve cover, and usually oblong in form, with the long diameter across the engine, so as to minimize the overhang.

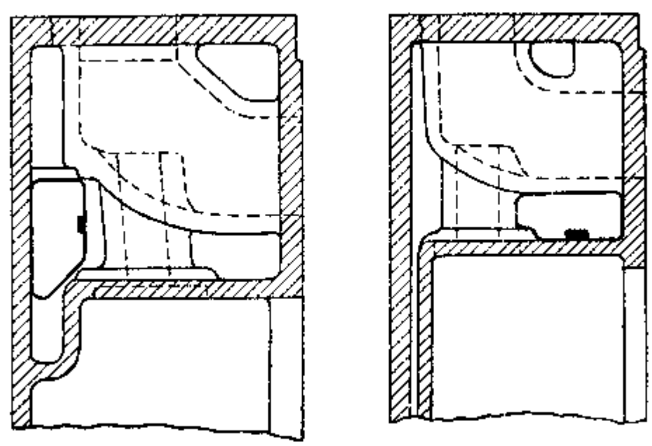


Fig. 5. Two arrangements of water distributor tube in L-head cylinder block.

In cylinders provided with "full-length" jackets, the central portion of the barrel lacks the reinforcement which with "half-length" jackets is provided by the flange that forms the bottom of the jacket. If the barrel also happens to be of minimum thickness its central portion will have very little rigidity and will distort easily, particularly if during machining operations the tool strikes a “hard spot:” This makes it almost impossible to obtain a true cylindrical bore. Conditions can be improved in this respect by providing the barrels of such engines with one or two circumferential ribs at intermediary points of their length.

While the flange around the cylinder at mid-length in engines with half-length water jackets has the advantage of affording the rigidity of structure desirable during machining operations it is detrimental under certain operating conditions. For instance, when an engine is being run under full load immediately after a cold start, the piston heats up much more rapidly than the cylinder block and is apt to get tight in the cylinder and scuff. It has been observed that in engines with half-length jackets such scuffing occurs particularly at the level of the water-jacket bottom flange, which latter prevents the cylinder from expanding.

Guarding Against Cylinder Distortion.

It has been pointed out already that a frequent source of trouble in operation is distortion of the cylinder bore which results in blow-by overheating and excessive cylinder wear. Cylinder distortion may he due to either mechanical or thermal causes. Mechanical distortion is most likely to result from tightening of the cylinder-head nuts, if the anchorages for the cylinder head studs are not properly supported. It .has been suggested that these anchorages be either located in a wall which extends straight down to the cylinder bottom flange so that the pull of the stud produces pure tensile stresses on the material of the block, or else be cast on the jacket wall rather than on the cylinder wall, as illustrated in Fig. 6. To further reduce cylinder-wall distortion, this wall is thickened near the top, while the thickness of the deck around the cylinder wall is reduced.

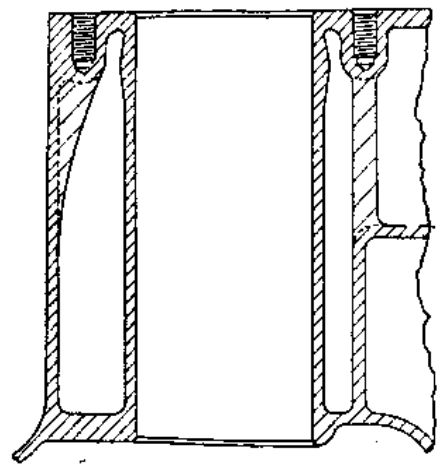


Fig. 6. Design minimizing mechanical cylinder distortion.

In valve-in-head engines the bases for the brackets carrying the rocker arms must he well supported, so they will not yield unduly under load which would make the engine noisy.

Removable Liners.

In most engines the pistons hear directly on walls forming part of the cylinder block, hut in some-and particularly in engines with large cylinders-removable liners are used. There are two types of these liners:

A "dry" liner is one which is in contact with metal of the block over its whole length, or nearly its whole length, while a "wet" liner is one which is supported by the block over narrow belts only, and is surrounded by cooling water between these belts.

In the United States "wet" liners came into use first, especially in the engines of farm tractors and commercial vehicles. Aside from the fact that any liner when worn or damaged can be replaced at relatively low cost, the construction offers the advantage that because of their uniform wall thickness (being machined inside and lout) and because they are very little affected by the tension of cylinder-head studs, separate liners distort less in service than the integral barrels of conventional cylinder blocks.

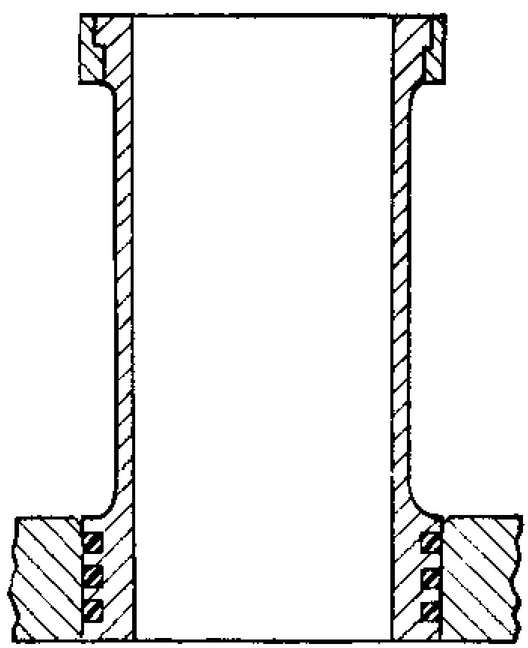


Fig. 7 "Wet" cylinder liner with packing rings.

At first the liners were made of the same gray iron that was used for cylinder blocks, but in the course of time materials of greater wear resistance were developed, and as most of these were more expensive than ordinary gray iron, they lent themselves particularly to use in liners. One method of installing a removable "wet" liner in a cylinder block is illustrated in Fig. 7. At the top the liner is provided with an external flange which enters a counter bore in the cylinder. The top of the liner is flush with the top of the block, and the joint is sealed by the cylinder-head gasket. In some cases and especially in Diesel engines-the hole in the gasket is made slightly larger than the cylinder bore, and a ring or loop of copper is inserted to reduce the pressure on the gasket.

At the bottom the liner is enlarged in diameter and has three grooves for packing rings cut in it. Instead of in the liner, the grooves may be cut in the block. These packing rings are made of synthetic rubber, which is more resistant to mineral oil and other petroleum products than natural rubber. The packing rings may be made of circular section, of a diameter slightly larger than the width of the grooves, and insertion of the liner then will deform them so that they substantially fill the grooves. To permit easy insertion of the liner, either it or the bore of the block is chamfered, depending on which part contains the packing rings.

Inaccuracies in the section diameter of these packing rings are said to have been the cause of some trouble. If the diameter is too small there may be leakage, whereas if it is too large the pressure exerted when the liner is forced into place may crack it. To overcome this difficulty, a cork-synthetic rubber composition of greater elastic compressibility has been developed. Packing ring of this material are molded with a square section, and when inserted project slightly above the surface of the part in which the grooves are cut. Insertion of the liner compresses them flush with that surface. Single and two packing rings also are used, and in the case of two rings, a third groove sometimes is cut between the two containing the packing rings, to collect any oil or water that may seep past the rings and allow it to drain off.

"Dry" liners, which in Great Britain were used practically exclusively from the beginning, seem to have gained the ascendancy over the "wet" type in this country after World War II. A typical "dry-liner" installation (in a GMC engine) is shown in Fig. 8. In this engine the cylinder block and crankcase are separate castings, and the liner extends some distance into the crankcase. It is held in position by a flange. at the top. In some other engines with dry liners and a separate crankcase the retaining flange on the liner is near the bottom and is held between the cylinder block and the crankcase. A British manufacturer of Diesel truck engines (Albion) copper-plates the dry liners on the outside. The copper is said to act as a lubricant, facilitating the insertion of the liner, and also to improve the heat flow from liner to cylinder wall.

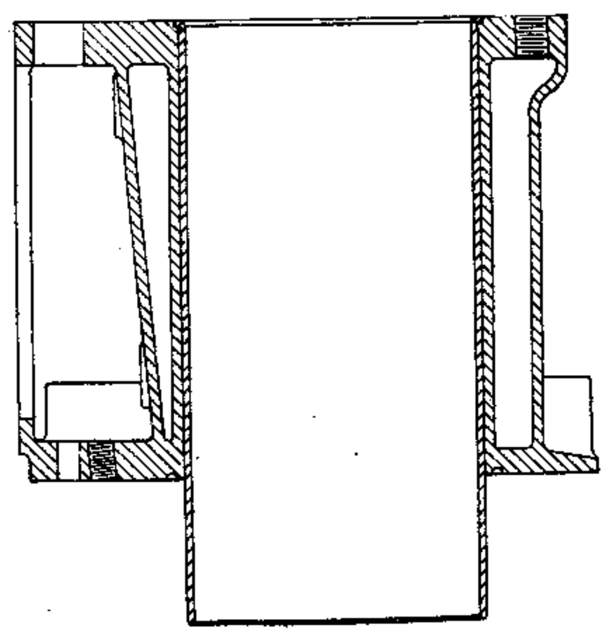


Fig. 8 "Dry" cylinder liner in position.

Materials for Cylinder Liners

For the engines of public-service vehicles, which latter run up enormous mileages in the course of a year, it has been found advisable to use alloy iron for the liners and to heat-treat them. General Motors Truck & Coach Division, for instance, uses such hardened liners in all of its larger engines, the material being a nickel-chromium iron of the following composition:

	Percent
Total carbon	3.10-3.40
Combined carbon	0.75-0.90
Manganese	0.55-0.75
Phosphorus	0.20 max.
Sulphur	0.10 max.
Silicon	1.90-2.10
Nickel	1.80-2.20
Chromium	0.55-0.75

In the "as cast" condition the liners show a Brinell hardness of 212-241, a transverse strength of 2400 lb on A.S.T.M. arbitration bars (bars of 1.2 in. diameter and 18 in. between supports), a transverse deflection of 0.20-0.30 in., and a minimum tensile strength of 37,000 psi on test bars machined from-the casting. A hydrostatic test also is applied to the liners, which must withstand 1500 psi for a wall thickness of l/8 in. and. bores of 4-5 in. To increase their wear resistance, the liners are hardened, by being heated to. 1540- 1560 F for 30 to 40 minutes and quenched in still oil. After this they must show a Brinell hardness of at least tensile 512 while the strength must range between 28,000 and 36,000 psi and the transverse strength between 2700 and 2900 lb for the arbitration bar. With these liners the mileage between cylinder overhauls is said to be practically trebled, as compared. With solid cylinders of gray cast iron showing from 230 to 240 Brinell. A minor disadvantage is that it takes up to 5000 miles for the piston rings to wear in fully, hence the oil consumption is rather high during the early part of the life of the liner.

Nitrided Cylinder Liners.

A process for nitrogen hardening or “nitriding” cast iron was developed in Europe. The process consists in exposing cast-iron objects to be case- hardened to a current of ammonia vapor at about 900 F for a considerable length of time, and then quenching. At this high temperature the ammonia breaks up into its constituents. Nitrogen and hydrogen, and the nitrogen penetrate into the surface of the casting & combines chemically with the metallic elements, forming very hard nitrides.

A Special alloy iron containing aluminum must be used. The liners are exposed to the ammonia vapors for 65 hours at 950 F and then have a hardened case of 0.015 in. depth, the hardness tapering off from the outside, where it is somewhere between 800 and 1000 Brinell.

A slight "nitride fuzz" produced on the surface of the liners during the process is removed before they are shipped to engine builders. Some distortion is caused, and the effects of this are eliminated by honing after the liners are inserted into the block, for which purpose an allowance of 0.002 in. on the diameter is made. Nitriding also produces a slight "growth," of the order of 0.001 in., and this, too, is allowed for in advance. Liners are installed in blocks with a press fit, an interference of 0.0015 to 0.0025 in. being allowed, depending on the bore.

Chromium Plating.

Another method of reducing the rate of wear consists in chromium plating the bore. The process differs radically from that of chromium plating for ornamental purposes. .It gives a "porous" coating which holds oil, while the so called bright plating process gives a dense coating to which oil will not adhere & which for this reason is readily is scored in service. From 200 to 500 times as much chromium as in conventional decorative plating is deposited per unit of area. If slightly too much should be deposited, so that the bore is undersize by from 0.0005 to 0.001 in., the excess can be removed by honing.Wear tests made on a plain gray-iron cylinder of 241 Brinell hardness and a similar cylinder plated indicated that chromium plating reduces the rate of cylinder wear approximately in the proportion of 7:1 and that the wear on the top piston ring is coincidentally reduced about 4:1.

Such methods as nitriding and chromium plating of cylinder bores are applicable particularly to bus and railcar gasoline engines and to Diesel engines, which have a much longer service life than passenger-car engines. Cylinder bores in plain cast iron must be reconditioned about every 50,000 miles, and with either a nitrided or chromium-plated bore, if reconditioning is required at all, it will be required only after a much longer interval.

Length of Bore

In most modem engines of both the L-head and I-head type the combustion chamber is formed in the cylinder head and at the end of the up-stroke the top of the piston is flush with the finished top surface of the cylinder block. One reason for not making the piston overrun the end of the bore is that that would bring the top ring beyond the upper end of the water jacket at the end of the up-stroke, where it would not be so effectively cooled, in the ring groove. The lower end of the piston generally is made to overrun the end of the bore slightly.

The total length of the finished bore evidently is equal to the length of stroke plus the length of the piston minus any overrun of the piston at both ends, the overrun being considered negative when the piston does not come quite to the end of the bore. To facilitate getting the piston rings into the cylinder, the bore is chamfered at the end from which the piston is entered

Production of Engine Blocks

In the design of engine block or cylinder block it is well to consult with the foreman of the pattern shop, because a casting of this kind is a difficult piece of mold, and the advice of an experienced mechanic may obviate trouble later.

Cylinders must be molded with the head downward, for the reason that blowholes, porous spots, etc are most likely to occur near the top of the casting, & the head of the cylinder, which is the working end, must of necessity be of sound metal.

When the castings have cooled the core sand is removed, the seams etc., are chipped off, & the castings are then put through a cleaning process. [Either by pickling & neutralizing or by blast cleaning (blast cleaning by sand or small granules of chilled iron or steel) & then normalizing & cleaning]

Further the cast & cleaned blocks would undergo other operations in sequence like Milling, Drilling, Cylinder boring, Precision boring, Finish of bore, Honing, Lapping, followed by measurement of quality of surface finish, Water test Finishing of valve seats & guides & surface broaching.

Transfer machines were adopted since world war-II to perform the operations automatically.

DESIGN OF CYLINDER AND CYLINDER HEAD

Cylinder should be

- designed to withstand the high pr. & temp. conditions.
- be able to transfer the unused heat effectively so that metal temp. does not approach the dangerous limit.

The Cylinder wall is subjected to gas pressure & the piston side thrust.

-Piston side thrust tends to bend the wall but the stress in the wall due to side thrust is very small & can be neglected.

-The gas pressure Produces 2 types of stresses;

-longitudinal and circumferential, which act at right angle to each other & the net stress in each direction is reduced. The longitudinal stress is usually small & can be neglected.

$$f_l = \text{longitudinal stress} = \frac{force}{area} = \frac{\frac{\pi D^2}{4} p_{\max}}{\frac{\pi\left(D^2{}_O - D^2\right)}{4}}$$

D=cylinder diameter,

D_O= cylinder outside diameter,

$p_{\max}$ =max. gas pr.

f_c =circumferential force= $\frac{p_{\max} \times D}{2t}$

Net $f_l = f_l - \frac{f_c}{m}$, &

Net $f_c = f_c - \frac{f_l}{m}$,

where $\frac{1}{m}$ = poision's ratio = $\frac{1}{4}$

CYLINDER WALL THICKNESS

The Wall Thickness is usually calculated by applying the formula for a thin cylinder,

thus $t = \dfrac{p_{max} \times D}{2f_c} + k$

Where t=wall thickness, mm,

p_{max} = max. gas pr.,N/mm^2 (3.1 to 3.5N/mm^2),

D=cy. bore, mm,

f_c =max. hoop stress and is equal to 35 to 105 N/mm^2 depending on the size and material, larger values are used for smaller bores,

Cylinder bore, mm	75	100	150	200	250	300	350	400	450	500
k =reboring factor, mm	1.5	2.3	4.0	6.0	7.5	9.5	10.5	12.5	12.5	12.5

The thickness of the cylinder wall usually varies from 4.5mm to more than 25mm, depending upon the cylinder size.

According to an empirical relation,

For liners of oil engines,

$$t \geq \frac{D}{15}$$ near the top portion & through 20% of the stroke.

For dry liners,

The total thickness't' is the thickness of the liner & that of the cylinder wall.

The thickness of the Dry liner is given as $t' = 0.03D$ to $0.035D$

The thickness of the inner walls of the automobile engine cylinders is usually given empirically as

$$t = 0.045D + 1.6\text{mm}$$

The thickness of Jacket wall is given as $= \dfrac{1}{3}$ to $\dfrac{3}{4}\, t$, larger ratio for smaller cylinder

or $= 0.032D + 1.6\text{mm}$

The water space between the outer cylinder wall & inner jacket wall is =10mm for a 75mm cylinder to about 75mm for a 750mm cylinder

or $= 0.08D + 6.5\text{mm}$

CYLINDER DIAMETER AND LENGTH

The o/p of a given cylinder can be written as - Power$= \dfrac{pLAn}{60}, W$

Where L=stroke in m,

A=piston area, mm^2,

n=no. of working strokes per minute= N for 2 stroke engines and $\dfrac{N}{2}$ for 4 stroke engines

p=imep-if power is indicated &

bmep if o/p is in brake power, N/mm^2

** As a guide, the max. gas pr. can be taken as 9 to 10 times the bmep*

CYLINDER FLANGES AND STUDS

The cylinder is either cast integral with the upper half of the crankcase or attached to it with the help of flanges, studs and nuts.

The cylinder flange is made thicker than the wall of the cylinder.

Flange thickness should not be less than 1.1 to 1.25t

Common value for flange thickness = 1.2 to 1.4t

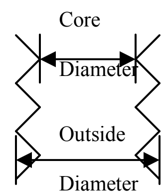
Or =1.25 to 1.5d where d =bolt diameter, nominal

The distance of the end of the flange from the center of the stud or bolt should not be $< d$ +6mm, and not $> 1.5d$.

The use of studs decreases the bending stress at the flange root since the moment arm can be made very small.

The material of the studs or bolts is usually nickel steel with a yield point of 630 to 945MPa.

The diameter of the bolt or stud is calculated by equation of the gas load to the area of all the studs at the root of the threads multiplied by the allowable fibre stress.



$$\therefore, \quad \frac{\pi}{4} D^2 \times p_{\max.} = z \times \frac{\pi}{4} d_c^2 \times f_t$$

$$\therefore \quad D^2 \times p_{\max.} = z \times d_c^2 \times f_t$$

$$\therefore \quad d_c = D\sqrt{\frac{p_{\max}}{z \times f_t}},$$

where f_t = allowable fibre stress, 35 to 70 N/mm^2,

d_c = core diameter

Low value of f_t is taken since there is already high stress in the studs due to tightening of the nuts.

The number of studs $'z'$ may be taken as $\left(\frac{D}{100}+4\right) to \left(\frac{D}{50}+4\right)$, D in mm

Or the pitch of the bolts may be taken as $19\sqrt{d}$ to $28.5\sqrt{d}$, where d is in mm.

In practice d generally varies from ($\frac{3}{4}$ to 1) times the thickness of the flange.

In no case d should be $<$ 16mm

CYLINDER HEAD

Usually a separate cylinder cover or head is provided with all but the smallest engines. A box type section is employed of considerable depth to accommodate ports. The general design of the cover is governed by the following factors along with the strength consideration.

- Air and gas passages
- Accommodation of valves and their gear
- Accommodation of the atomizer at the centre of the cover in the case of the diesel engines.

Cylinder head is the most difficult part to be designed and manufactured. The cylinder heads are usually made of close grained cast iron or alloy cast iron containing nickel, chromium and molybdenum, for small and medium sized engines, while for large engines, the material is low carbon steel.

The thickness of the cylinder wall ranges from about 6.5mm for small engines to proportionately larger values for large engines. The thickness depends on the shape of the head. If the cylinder head is approximately a flat circular plate, the thickness can be determined by the relation:

$$t = D\sqrt{\frac{Cp_{\max}}{f_t}}$$

Where C=const., in this case equal to 0.1, f_t =allowable stress, taken to be 35 to 56 N/mm^2

A low value of 'f_t' is taken because both pr. & temp. stresses are induced in the cylinder head and the above equation is based upon only the cylinder pressure. The heat transfer through the head is about 5 to 13 times as much heat per unit area as the cylinder walls, depending on the design and amount of cooling.

• Example - 1

Determine the thickness of a cast iron cylinder wall & the stresses for a 300mm petrol engine, with a maximum gas pressure of 3.5N/mm^2

• Solution :

Given

D = cylinder bore = 300mm, p_{max} = max. gas pr. = 3.5N/mm2

Wall Thickness is usually calculated by applying formula for a thin cylinder,

Thus Wall Thickness, $t = \frac{p_{max} \times D}{2 f_c} + k,$

where, D = cy. bore, mm, p_{max} = max. gas pr., N/mm^2 (3.1 to 3.5N/mm^2),

fc = max. hoop stress and is equal to 35 to 105 N/mm^2

depending on the size and material, larger values are used for smaller bores,

Cylinder bore, mm	**75**	**100**	**150**	**200**	**250**	**300**	**350**	**400**	**450**	**500**
Reboring factor, mm	**1.5**	**2.3**	**4.0**	**6.0**	**7.5**	**9.5**	**10.5**	**12.5**	**12.5**	**12.5**

$$From\ above\ table\ k = 21.5mm,$$

$$Assume\ f_c = 45N/mm^2,$$

$$Wall\ Thickness,\quad t = \frac{p_{max} \times D}{2 f_c} + k \quad = \frac{3.5 \times 300}{2 \times 45} + 9.5 \quad = 21.5mm$$

Now apparent longitudinal stress,

$$f_l = \frac{force}{area} = \frac{[(\pi D^2/4) \times p_{max}]}{[\pi(D_o^2 - D^2)]/4} = \frac{p_{max} \times D^2}{(D_o^2 - D^2)}$$

Where, D = cylinder diameter,

D_o = cylinder outside diameter & p_{max} = max. gas pr.

Now Do = D + 2t = 300 + 2t = 300 + (2 × 21.5) = 343mm

Apparent longitudinal stress $f_l = \frac{3.5 \times 300^2}{(343^2 - 300^2)} \quad = 11.45 N/mm^2$

Now apparent circumferentinal stress,

$$f_c = \frac{force}{area} = \frac{p_{max} \times D}{2t} = \frac{3.5 \times 300}{2 \times 21.5} = 24.4 N/mm^2$$

Net $f_l = f_l - \frac{f_c}{m}$, where $\frac{1}{m}$ = poision's ratio = $\frac{1}{4}$

Net $f_l = 11.45 - \frac{24.4}{4} = 11.45 - 6.1 = 5.35 N/mm^2$

& Net $f_c = 24.4 - \frac{11.45}{4} = 24.4 - 2.86 = 21.54 N/mm^2$

Example 2

A vertical 4 stroke CI Engine has the following specifications:

Brake power = 4.5kW, Speed = 1200rpm, imep = 0.35N / mm², η_{mech} = 0.80.

Detrmine the dimensions of the cylinder.

Solution :

$$Since \quad \eta_{mech} = \frac{Brake\ Power}{Indicated\ Power}$$

$$\therefore Indicated\ Power = \frac{Brake\ Power}{\eta_{mech}} = \frac{4.5}{o.8} = 5.625kW$$

$$Indicated\ Power = \frac{P_{imep} N/mm^2 \times L\ m \times A\ mm^2 \times n\ rpm}{60}\ Watt$$

$$[1Watt = 1N\frac{m}{s}]$$

$$n = \frac{n}{2}\ for\ single\ acting\ 4\ stroke\ Engine = \frac{1200}{2} = 600$$

$$\therefore \quad 5.625\times10^3 watt = \frac{0.35\times L\times A\times 600}{60}$$

$$or \quad L\ m\times A\ mm^2 = \frac{5.625\times10^3\times60}{0.35\times600} = 1.608\times10^3$$

$$or \quad L\ m\times\frac{\pi\times D^2}{4}\ mm^2 = 1.608\times10^3$$

$$Now\ assuming\ \frac{Stroke}{Bore} i.e. \frac{L}{D}\ ratio\ as\ 1.35,\ or\ L = 1.35D$$

$$\therefore \quad 1.35D\ m\times\frac{\pi\times D^2}{4}mm^2 = 1.608\times10^3$$

$$or \quad \frac{1.35D}{1000}mm\times\frac{\pi\times D^2}{4}mm^2 = 1.608\times10^3$$

$$or\ Bore\ Diameter \quad D = 115mm,$$

$$\therefore\ Stroke\ Length \quad L = 1.35D = 1.35\times115 = 155mm$$

$$Now\ Length\ of\ Cylinder = Stroke + clearance\ on\ both\ sides$$

$$= Stroke + 10\ to\ 15\%\ of\ the\ stroke$$

$$\therefore Length\ of\ Cylinder = 155 + (155\times0.15)$$

$$= 178.5mm$$

Example 3

Determine the thickness of a plain cylinder head for 0.3m cylinder.

The maximum gas pressure is approximately 3.2N/mm^2*. Design the studs also for the cylinder cover.*

$$Solution: \quad Thickness\ of\ cylinder\ cover = t = D\sqrt{\frac{C \times p_{\max}}{f_t}}$$

$where D = 300mm, \quad p_{\max} = 3.2N/mm^2$ $\{C = constant = 0.1, \ \&\ f_t = allowable\ fibre\ stress = 35\ to\ 56N/mm^2\}$

$assuming\ f_t = 42N/mm^2,\ for\ good\ grade\ cast\ iron$

$$\therefore \quad t = 300\sqrt{\frac{0.1 \times 3.2}{42}} = 26.2mm$$

<u>*Studs*</u>

The gas will actually act upon the p.c.d. of the studs, but as the stud diameter is not known initially, the pressure may be assumed to be acting the cylinder diameter. Or, it is a common practice that the centre of the stud should be $1.25d$ *to* $1.5d$ *from the the inner wall of the cylinder.*

$(d = nominal\ bolt\ diameter, \quad d_c = core\ diameter)$

$\therefore$ *Pitch circle diameter* $D_p = D + 3d = 300 + 3d\ mm$

$$\therefore \quad Load\ on\ the\ stud = \frac{\pi \times D_p^2}{4} \times \max.\ gas\ pressure$$

$$= \frac{\pi \times (300 + 3d)^2}{4} \times 3.2$$

$$But\ load \ = Z \times \frac{\pi \times d_c^2}{4} \times f_t,$$

$where\ f_t = 35\ to 70N/mm^2\ \ \&\ d_c = core\ diameter,\ Z = No. of\ studs$

$let,\ core\ diameter, d_c = 0.8 \times nominal\ diameter = 0.8 \times d$

$$Now\ No. of\ studs\ Z = \left(\frac{D}{100} + 4\right)\ to\ \left(\frac{D}{50} + 4\right) = \left(\frac{300}{100} + 4\right)\ to\ \left(\frac{300}{50} + 4\right) = 7\ to\ 10,$$

$Let\ Z = 8 \qquad \& \qquad f_t = 63N/mm^2$

$$\therefore \quad \frac{\pi \times (300 + 3d)^2}{4} \times 3.2 = 8 \times \frac{\pi \times (0.8d)^2}{4} \times 63$$

$By\ trial\ \&\ error,\ we\ get,\ d = 43mm$

$\therefore D_p = D + 3d = 300 + 3d = 300 + 3 \times 43 = 429mm$

$$\therefore Pitch\ of\ the\ studs = \frac{\pi \times D_p}{Z} = \frac{\pi \times 429}{8} = 168.5mm$$

Now minimum pitch should be $3d = 3 \times 43 = 129mm$ *and maximum pitch lies between* $19\sqrt{d}$ *to* $28.5\sqrt{d}$

i.e., $124.5mm$ *to* $187mm$, $\therefore$ *Both conditions are satisfied*

References:

1. High Combustion Engines – P M Heldt
2. M/C Design –Sharma & Agarwal