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THE NEW GMC V-6 AND TWIN SIX ENGINES

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THE NEW GMC V-6 AND TWIN SIX ENGINES

This is a description of the new GMC engine family, a family which covers the complete range from the lightest to the heaviest gasoline engined GMC trucks. All members of this family are completely new; they have been designed from the ground up for commercial service. This exclusive design approach has led to unusual configurations, specifically to the use of the V-6 and twin six combination which is of advantage for truck applications. A new and thoroughly modern engine manufacturing facility has been set up to secure economical volume production.

The following table shows the new engine family has four members, three V-6's ranging from 305 to 401 cubic inches displacement, and the twin six with 702 cubic inches displacement. This table also shows that the stroke for all engines is the same and the displacement is varied by changing the bore and the number of cylinders.

Displacement	Type	Stroke	Bore
305	V-6	3.58	4.25
351	V-6	3.58	4.56
401	V-6	3.58	4.88
702	Twin 6	3.58	4.56

As may be surmised, additional engine displacements can be added with relatively small tool expense by adding intermediate bore sizes or by increasing the bore a little more than the present maximum size.

The 305 cubic inch engine is used in trucks of gross weights from 5,000 to 23,000 pounds. Three versions of the one displacement, called 305A, 305B and 305C, are used to cover this wide range. The "A" version is for light duty; the "B" version has heavy duty valves and other changes to suit the heavier load it will pull, while the "C" version is the "B" plus more power. This difference in power is shown by fig. 1.

Fig. 1 also shows that, while there is a large displacement gap between the 401 and 702 cubic inch engines, the gap in horsepowers is proportionately less than the difference in displacement because the 702 cubic inch engine is governed at the very low speed of 2400 rpm. Other factors enter into this situation, as will be explained in detail later.

At low speed, all of these engines deliver high torque which decreases rapidly as the speed increases. See fig. 2. This is done for two reasons: first, high horsepower output has been avoided because this means high temperatures, which shorten valve life. Second, there is a great deal of satisfaction in driving a truck with a torque curve which is high at low speed; as the truck is driven up a hill, it digs into the grades as its speed decreases, reducing the necessity for gear shifting. The information shown in figs. 1 and 2 is tabulated below:

Engine	Gross Horsepower	Maximum Torque
305A & B	150 at 3600	260 lb-ft
305C	165 at 3800	270
351	180 at 3400 (governed)	312
401	205 at 3200 (governed)	377
702	275 at 2400 (governed)	630

We use the large 702 cubic inch engine for a number of reasons: first, with an engine this size we can operate it at a rather low specific horsepower output which makes for durability. Second, because of the high torque developed with the large engine, we can operate it at low speeds, which provides durability and quietness. Most important, however, the engine will normally operate at less than its full power output, which further decreases temperatures and improves durability, and also increases fuel economy. Fig. 3 plots the engine power output against the power required to pull 60,000 GCW, both of these being on a net output basis at the flywheel. Contrary to the usual practice, the power required curve falls well below the power developed curve, consequently under stabilized conditions the engine will be run throttled. Fig. 3 also shows a series of lines of equal specific brake fuel economy, from which it may be seen the operation of the engine at a throttled condition along the power required curve will give better specific economy than if the engine were running at full throttle.

Still another advantage is that the large reserve power permits the use of a five speed transmission instead of the usual eight or ten forward speeds which reduces the necessity for gear shifting.

Fig. 4 shows one of the V-6 engines with typical accessories. The engine banks are at 60°, the left bank being staggered ahead of the right. Fig. 5 shows the twin six. It can be seen that a number of the external parts, such as cylinder heads and rocker covers, are the same as used for the V-6. There are many more interchangeable internal parts, as will be explained later.

The twin six is not a common cylinder arrangement and the V-6 is even more unusual. They were chosen after much study. They are well suited to commercial use and here are some of the reasons. First, the combination of six and twelve cylinder engines covered the required displacement range from 300 to 700 cubic inches. As few as six cylinders in the 300-400 cubic inch range can be used without getting excessive cylinder sizes. To reach up to the 700 cubic inch range, however, would have required excessively large cylinders for any number of cylinders less than twelve; this dictated the choice of the twelve for the larger engine.

Second, the combination of V-6's and twin sixes is ideally suited to tooling interchangeability, as shown by fig. 6. Both engines have the same vee angle and the same front and rear views, which means that common tooling can be used for any machining operations involving the external dimensions of the engines. For finishing cylinder bores and other boring or drilling operations parallel to the bores, the tooling also may be common. To bore the twin six, the front six cylinders are bored first, then, in effect, the block is indexed forward and the operation repeated.

Another reason for the choice of this engine family is that the V-6 is economical to build and to service. In rebuilding the engine and replacing worn parts, for example, there are only 6/8 as many parts to replace in a six as in an eight. Another reason, confirmed by experience, is that the V-6 makes a very rugged powerplant, as will be discussed in more detail later.

The fifth reason for using V-6's and twin sixes is that the engine dimensions thus achieved are well adapted to use in trucks. Fig. 7 shows the overall engine dimensions. The front view applies to both the V-6 and twin six engines; height and width dimensions shown for the six also apply to the twin six. The V-6 length dimension of 34-1/4 inches is overall, including the fan which is spaced well forward for good exit air flow. Due to the 60° vee angle, both engines are narrow, as shown by the 26 inch width dimension. The twin six is fairly long (although comparable to a line engine of the same displacement) but is both narrow and low for an engine of its size.

Figs. 8, 9 and 10 show how these engine dimensions help to fit the engines into vehicles. Fig. 8 shows how the V-6 fits under a short hood, thus reducing cab length and permitting longer load space.

The twin six is not short but it is low. This permits it to be used beneath a short cab, as shown by fig. 9. The engine projects behind the cab, however it is so low that the trailer or truck body extends over the top of the engine.

Both the V-6 and twin six with their 60° vee angle are narrow, thus either engine will fit between the front, or steering, wheels of a truck (see fig. 10).

And finally, the last reason we chose the V-6 and twin six family is parts interchangeability. Seventy-three major parts are interchangeable among the six cylinder engines and 56 parts are interchangeable among all the engines. Let's look at a few examples. The cylinder block assembly, shown by fig. 11, has interchangeable bearings, caps, bolts, studs, etc. The blocks themselves are not interchangeable due to differences in the bore diameters but can be produced over the same tooling.

A cylinder head is shown by fig. 12. One cylinder head is used for the 305 while a second head with larger combustion chambers and exhaust valve seat inserts is required for the 351, 401 and 702.

While we are on the subject of cylinder heads, fig. 13 will be of interest. It shows three sections illustrating our first, second and final attempts to secure a combustion chamber shape giving good mechanical octane numbers. The compression ratios are 7.75 in the 305 and 7.5 in the other engines; regular fuel may be used. In the final design, most of the combustion chamber is in the head. This is shaped as shown in cross section and, in plan view, consists of an elongated oval formed by two circles joined by tangent straight lines. A portion of the combustion chamber is in the saucer shaped depression in the piston.

Fig. 14 is a piston photograph which also shows the depression, while fig. 15 is a cross sectional drawing. Three sizes of pistons are used for the four engines. All pistons are tin plated; all have four rings, as shown by the figure. The top ring is chrome plated and the lower oil control ring is a three piece type with stainless steel expander and two steel chrome plated rails. All pistons also have a steel expansion control band and slots to separate the piston skirt from the head except above the pin bosses. The cross section also shows the grooves inside the piston pin bosses, which are for the retaining snap rings. The 351, 401 and 702 pistons have a steel insert in the upper ring groove which reduces ring and groove wear.

Fig. 16 shows another common part, the connecting rod which is identical in all engines. The piston pin floats in the piston and also in the steel backed bronze bushing in the upper end of the connecting rod. The bushing is lubricated by oil splashed into an opening in the top of the rod. Each end of each rod is balanced to plus or minus 1.75 grams. The lower end bearing is 2-13/16 inches diameter and has 2.52 square inches projected area; it is of heavy duty type, either copper lead or steel backed babbitt coated aluminum.

Fig. 17 shows the cast Armasteel crankshaft used in the 305 cubic inch V-6. Dimensionally and except for balance, the crankshafts for all V-6's are interchangeable but we use forged steel shafts for the larger engines. Our experience with the Armasteel material shows that it has a modulus of elasticity of about 26 million which approaches steel pretty closely and gives us the required rigidity.

Fig. 18 shows the forged steel shaft used for the 351 and 401 cubic inch engines. The pins and main journals are inducation hardened. The main bearings are also shown to illustrate the fact that they are the same for all engines. All main bearings are 3-1/8 inches diameter; the flanged thrust bearing has 4.22 square inches projected area; all others have 3.52 square inches. All upper main bearings have one circumferential groove to carry full pressure oil to the crankpins; the lower main bearings are grooveless. The upper and lower main bearing locating lugs are in different locations to prevent installing the bearing halves in the wrong place.

Since we use the same crankshaft for the 351 and 401 engines, the piston assemblies of these engines must weigh the same to suit the crankshaft counterweighting. This is accomplished by using a solid piston pin in the 351 and a tubular pin in the 401. See fig. 19.

Fig. 20 shows the twin six shaft with some more main bearings which, however, are the same as used in the V-6. The twin six crankshaft is similar in shape to that of the conventional line six except that two rods are side by side on each crankpin.

Fig. 21 shows the valve gear which is the same for all engines except that heavy duty engines have heavy duty exhaust valves. The rocker arm is a malleable casting mounted on a hardened steel shaft. The arm operating ratio is 1-1/2 to 1. Mechanical tappets are used in all V-6's. We use hydraulic tappets in the low speed, low temperature twin six. With either type tappet the body is one inch in diameter and is of hardenable iron; the flat lifter foot is hardened to 57-65 Rockwell C. The lifter is offset .060 inches to provide rotation. Both mechanical and hydraulic tappets are lubricated by pressure from longitudinal oil galleries shown in the engine cross section, fig. 43.

Although all engines will be operated at relatively low speeds, it was believed desirable to design the valve gear for high operating speeds to increase' the factor of safety and the durability. In particular the valve gear has a high order of rigidity as illustrated by fig. 22.

The following table shows the exhaust valve equipment used in the various engines:

Engine Model	Exhaust Valves Head	Exhaust Valves Stem	Seat Insert	Rotation (Intake & Exhaust
305A	1.565 Sil XB	11/32	No	None
305B & C	1.565 Sil 10 hard faced	11/32	No	Positive
351	1.825 Sil XB hard faced	7/16 sodium cooled	Yes	Positive
401	1.825 Sil XB hard faced	7/16 sodium cooled	Yes	Positive
702	1.825 Sil XB hard faced	7/16 sodium cooled	Yes	Positive

The hard facing is a nickel-chromium-tungsten-cobalt alloy. Intake valves of the light duty 305A are of hardened chrome-nickel-molybdenum steel; the intakes of all other models are Sil XB.

We believe the camshaft drive story is interesting. Fig. 23 shows the gears used on the 401 and 702 cubic inch engines. The crankshaft gear is case hardened steel while the camshaft and idler gears are pearlitic malleable iron. The thrust and radial bearing surfaces of the gears are pressure lubricated. All gear teeth are shaved, and the idler gear is crowned slightly.

Fig. 24 shows the stub shaft used for supporting the idler gear. It is located by a 1-1/8 inch pilot boss fitting into a hole in the block. The idler gear stub shaft carries an attached hardened steel plate to take the thrust due to the helical gears. Incidentally, the thrust of the distributor and oil pump drive gears on the camshaft is opposite in direction to the camshaft drive gear thrust and thus reduces the net thrust force.

On our lighter duty 305B and 351 engines we do not use gears. A 3/8 inch pitch, double strand roller chain, shown by fig. 25, is used on the 351 and 305B and C, while the 305A, our lightest duty engine, employs the silent type of chain, shown by fig. 26. The center distances between the camshaft and the crankshaft were set up to suit the chain drives, then the gears were designed to fit. The stub shaft, shown in fig. 24, is not precisely on the vertical centerline, it is offset slightly to help in the solution of this problem.

Both chains and the camshaft drive gear teeth are lubricated by oil squirted from the camshaft bearing.

All camshafts are cast of high alloy electric furnace iron and have hardened and phosphate coated cam lobes. All V-6 shafts are identical. They are supported on four equally spaced journals. The twin six camshaft is supported on seven journals.

Fig. 27 shows a considerable number of gaskets which are interchangeable in all engines. This completes the interchangeability discussion.

Among the other engine features, the firing order and crankshaft arrangement of the V-6 may be of interest. Fig. 28 includes two diagrams, the one on the right is a plan view of the V-6 with its front toward the left. The cylinder numbers are inside the circles. The numbers beside the circles show the order in which these cylinders fire.

On the left side of fig. 28 is a diagrammatic front view of the crankshaft from which it will be seen that no two crankpins are in the same plane and that adjacent pins are separated by a 60° angle, thus there is considerable overlap providing rigidity.

Fig. 29 applies to the twin six. For each bank, the firing order is the conventional 1-5-3-6-2-4 commonly used on line six engines, but the firing alternates from one bank to the other. The left hand view shows that the crankpins are at 120° angles and that the front and rear pins are in a common plane, also the second and fifth, and third and fourth.

The following table gives valve timing information:

	V-6	Twin Six
Intake valve lift	.406	.378
Exhaust valve lift	.397	.378
Intake valve opens	14°BUDC	11°BUDC
Intake valve closes	58°ALDC	41°ALDC
Exhaust valve opens	64°28'BLDC	50°BLDC
Exhaust valve closes	32°14'AUDC	10°AUDC

The V-6 uses a six legged manifold, the plan view of which is diagrammed by fig. 30. This particular arrangement is for a single barrel carburetor. Some power increase may be secured from a dual carburetor, however the manifold is similar except for the shape of the chamber immediately below the carburetor. Since we have given a high priority to durability and feel this more important than securing the last bit of horsepower, no four barrel carburetors are used. This simplification also helps the service man.

The twin six manifolds are arranged as shown in fig. 31. Each manifold is similar to but not identical with the V-6 manifold, the difference being due to the amount of stagger between the right and left blocks which requires dimensional differences between the manifolds. The diagram does not show the pressure equalizing tube which connects the two manifold risers.

Because the V-6 is an unusual type, its inherent balance was investigated. We found that all primary forces, due to reciprocating masses, are balanced. Similar to a V-8 with 90° crankshaft, the rotating masses, due to the crankpins and connecting rod big ends, form a rotating couple which is completely counterbalanced. There are secondary forces, the vertical and horizontal components of which are given by the following formulae:

$$\text{Vertical} = \frac{1.5\, d\, W_i r^2\, \Omega^2 \sin 2\Theta}{gL}$$

$$\text{Horizontal} = \frac{1.5\, d\, W_i r^2\, \Omega^2 \cos 2\Theta}{gL}$$

in which:

d - main bearing spacing

W_i - reciprocating weight per cylinder

r - crank radius

Ω - angular velocity, radians/second

L - connecting rod length

Solving these two equations shows there is a rotating couple moving in a direction opposite crankshaft rotation and at twice engine speed. The following formula describes a counterweighted shaft of equal couple:

$$\text{Equivalent Counterweight} = \frac{1.5\, d\, W_i r^2}{4DL}$$

In this equation, D is the dimension between the opposed counterweights.

In the initial stages of the design, we anticipated the use of a counterweighted shaft to be located just above the camshaft and which would rotate at twice crankshaft speed. This shaft was omitted when it was found that the amount of unbalance did not require its use. This can best be illustrated by solving the equation for the engine movement due to the rotating couple. This is given by the following equation:

$$\text{Engine Movement} = \frac{K_{sc}\ W_i r^2 l}{4L\ J_{z-z}}$$

in which l is the dimension between motor mounts

K_{sc} is the ratio between the secondary rotating couple and the secondary inertia force for one cylinder

J_{z-z} is the engine moment of inertia about Z axis.

Solving this equation for our V-6 engine gives a linear motion of .0015. This motion is small enough so that it can be absorbed by the usual rubber mounts. We thought it desirable to confirm the small linear motion experimentally and mounted an engine on large rubber bellows, as shown by fig. 32. These bellows have an almost zero rate. Measurements made by displacement pick-ups showed motions of the same order prophesied by the mathematics.

Fig. 33 is a photograph of the cylinder block casting which has been inverted so as to show that the block rails are dropped three inches below the crankshaft centerline. This not only increases the rigidity of the block structure itself but provides a greater depth at the point of flywheel housing attachment which is desirable to reduce deflection at this point.

Fig. 34 is a section through a block giving further information as to the structure. The water jackets are full length. Each cylinder has six head bolt bosses evenly spaced and separated from the bore walls to minimize distortion. Ribs extend down the outer cylinder block wall several inches below the outer head bolt bosses. Also the oil drainback holes from the head to the crankcase are drilled through long bosses which add structural rigidity; there are four bosses in a V-6, and eight in the twin six.

The cross section also shows how each main bearing cap is located and held by its interference fit in the broached opening in the crankcase web. In addition, the thrust bearing cap (#3 in V-6's, #6 in the twin six) is doweled to locate it fore and aft. All bearing caps are held by two 9/16 inch bolts except the twin six center bearing cap which has four 9/16 inch bolts and all rear bearing caps with two 9/16 inch and two 1/2 inch bolts.

Fig. 34 also shows the pocket in which the camshaft operates. Greater detail is provided by fig. 35. When the engine is stopped, oil drains into the trough up to the height of the overflow drain holes. When the engine is restarted later, the initial lubrication is instantly available but after the engine is running, the oil is splashed out of the trough so that the cams do not continue to run submerged. This arrangement prevents tappet and cam scuffing during cold starts.

Valve temperatures are particularly important in a truck engine which has a high load factor. One important V-6 advantage is the inherent shortness of the engine which permits us to provide additional space in the cylinder head. Fig. 36 is a cross section through the head which shows the large water spaces which can be provided. Also, please note that the bridge between the intake and exhaust valves provides a good structure and allows room for the exhaust valve seat insert. No two exhaust valves are adjacent. The valve guides are integral with the head casting, a design we favor because of the improved heat transfer from the valve to the coolant.

Experimental experience with this engine and with other engines has convinced us that valve durability may be improved by providing high velocity water flow past the valves. This cooling effect is a matter of velocity rather than volume, although practically, of course, considerable volume is required to secure a desirable velocity. While the overall engine cooling is improved by increasing the water flow through the radiator, increasing the volume above that required to secure about a 10° temperature drop gets into an area of diminishing returns. Valve cooling requirements indicated a water volume giving only a 4° or 5° temperature rise through the engine, consequently about half the water is passed through the radiator and about half through a large fixed bypass. The bypass also provides for valve cooling and uniform temperatures during warm-up when the radiator thermostat is closed.

The chart of fig. 37 shows the water flow characteristics of the 401 cubic inch engine. The water flow for all engines is shown by the following table:

Total Water Flow, Thermostat(s) Open

Engine	Number of Thermostats	gpm at 2400 rpm	gpm at 3000 rpm
305	1	116	141
351, 401	2	150	182
702	3	200	-

You will note that while the water flow for the 702 dubic inch engine is greater than that of the 351 and 401 cubic inch engines, the volume has not been increased in proportion to the engine size. This is because, as previously stated, valve cooling is a function of water velocity rather than engine size and since cooling water enters the rearmost head on each side of the twin six and proceeds forward through the rear head and then through the front head, the twin six area is identical with that of the V-6, therefore the water velocity in the twin six will be the same as that in a V-6 at the same water flow. For valve cooling, then, no greater volume of water is required for the twin six than for the V-6. The larger engine has a greater total heat rejection, hence more water flow is provided through the radiator and, of course, the total flow is greater when the thermostats are open but the total flow is not increased in direct proportion to the engine size.

These large coolant flows are realized with normal sized water pumps and without high water pump horsepower by reducing the restriction to water flow; this is accomplished in two ways: first by the large by-pass previously described, second by opening up the bottle-neck of the system, the thermostat, which is accomplished by using two or three standard size thermostats in parallel.

Fig. 38 is a diagram of the cooling system. The water pump is at the front of the engine and delivers water to two outlets, one at the front of each block. The connection to the cylinder head is at the rear so that all the water flows past each valve, as previously mentioned for the twin six.

Lubrication features include large capacity oil pumps, full pressure to all rotating journals, large size full flow filters and generous oil sump capacities. See fig. 39.

Engine	Filter Size	Oil Sump Capacity	Oil Pump Output gpm at 60 psi 3400 rpm
305	1 qt	5	14
351, 401	2 qt	8	14
702	2 qt	14	17

The six cylinder distributor is at the rear of the engine near the center and is driven from the camshaft by the usual vertical shaft with the oil pump at the lower end. The twin six distributor is in the same location and has two sets of breaker points and two coils. To secure a wide terminal spacing, two rotors and two six cylinder bowls and distributor caps are used, as illustrated by fig. 40.

Previously it was stated that mechanical tappets were used on all V-6 engines. Tappet adjustment is facilitated by the use of self-locking screws, as illustrated by fig. 41. Another unusual feature of the engine is the ability to remove the valve push rods and valve tappets from the top without removing the cylinder heads. The tappet is pulled up through the large push rod hole, as shown by fig. 42.

And, finally, figs. 43 and 44 show the 401 cubic inch engine cross and longitudinal sections. The cross section illustrates the crankcase ventilation tubes passing from each rocker cover to the intake manifold. This positive ventilation system is used on all models except the 305A in which the road draft type is used. Please note the spark plugs on the inside of the vee, accessible from above, also the wiring is cooler because it's separated from the exhaust manifold.

The longitudinal section provides additional information about the engine.

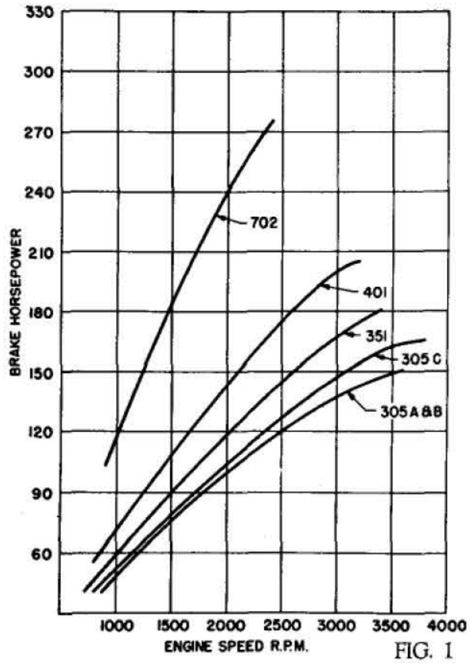


FIG. 1

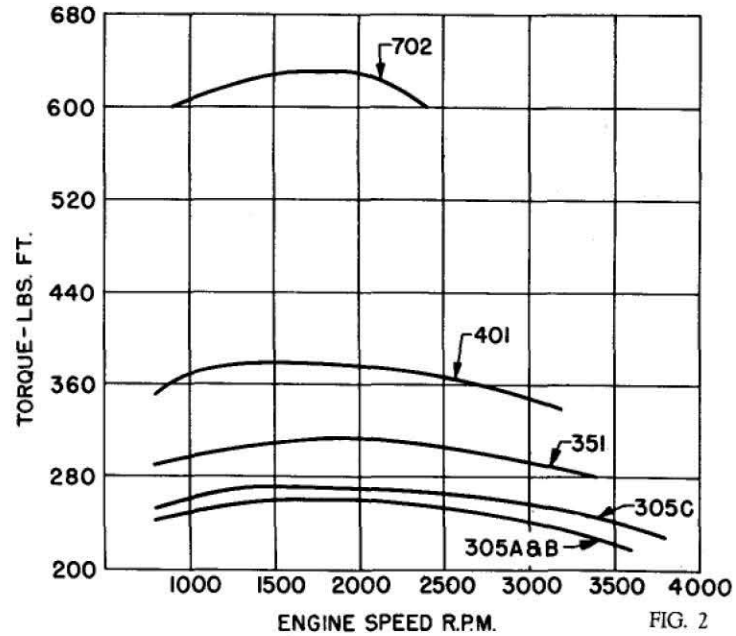


FIG. 2

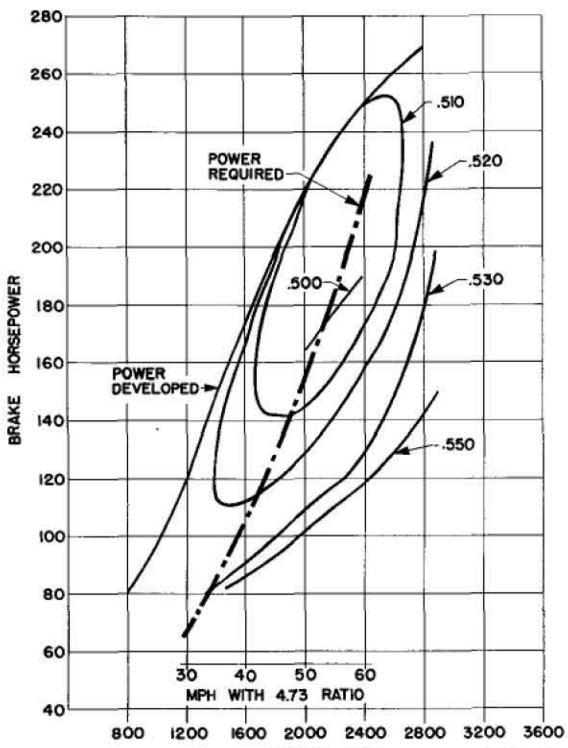


FIG. 3

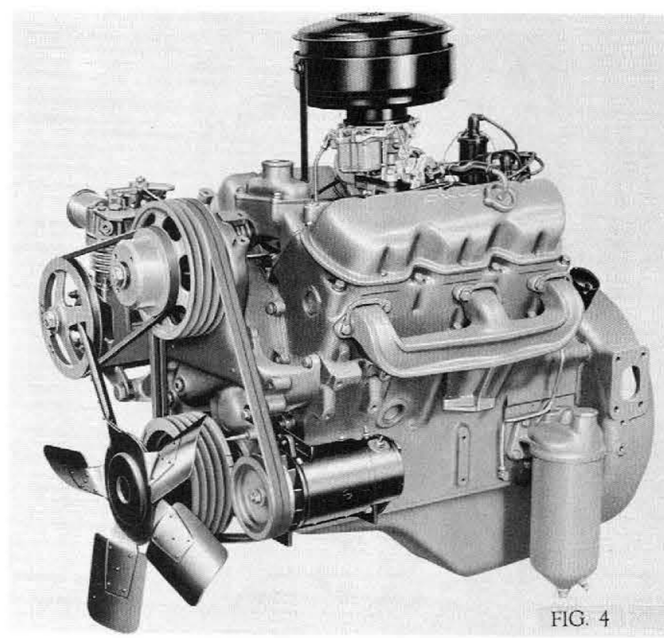


FIG. 4

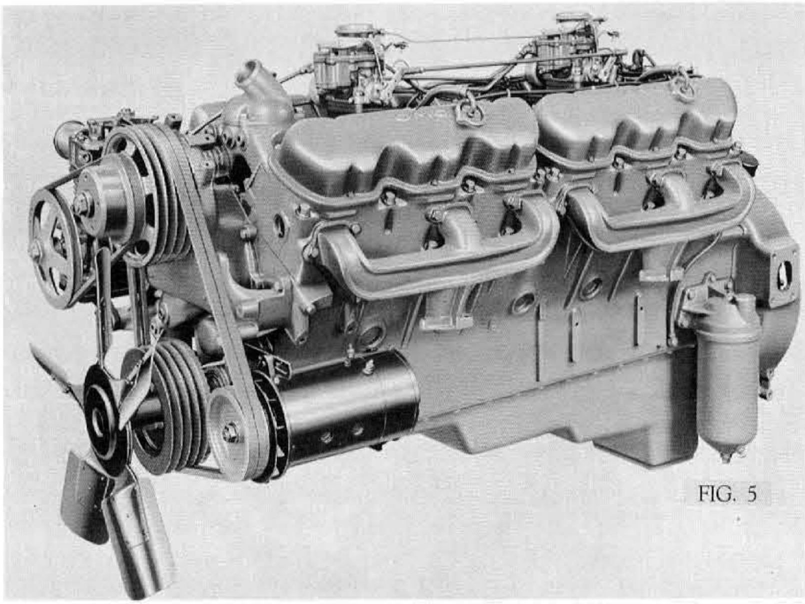
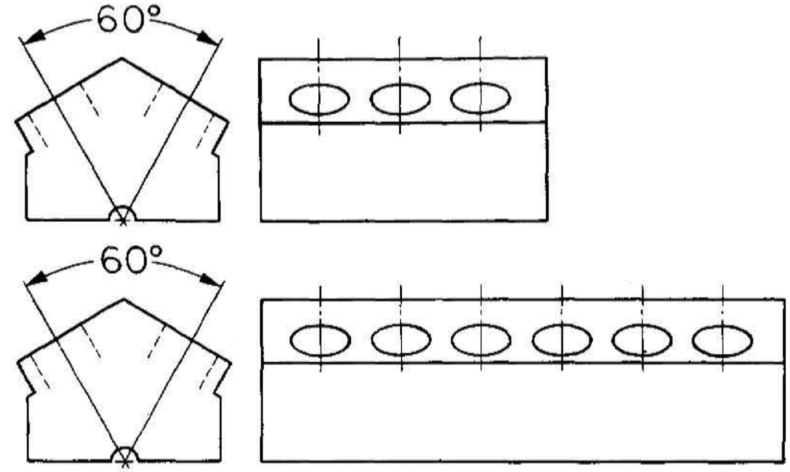


FIG. 5



CYLINDER BLOCKS

FIG. 6

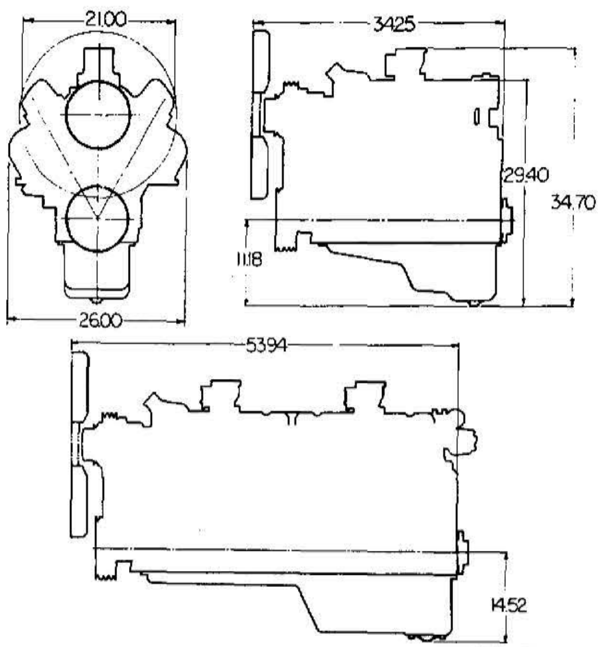


FIG. 7

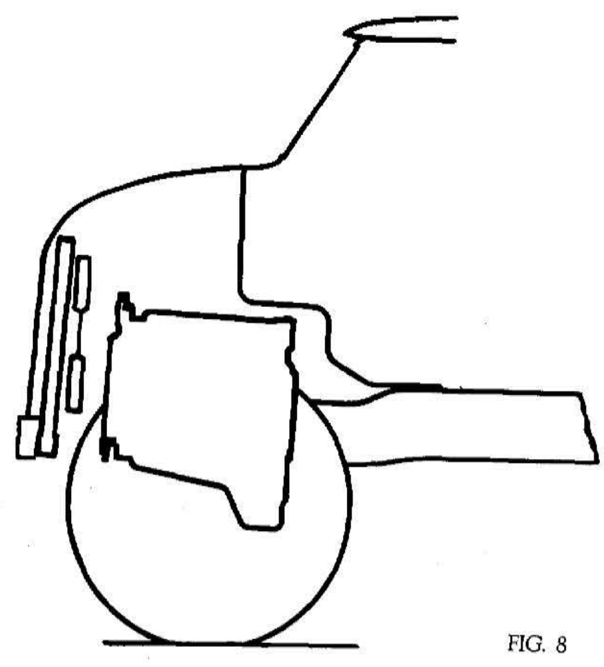


FIG. 8

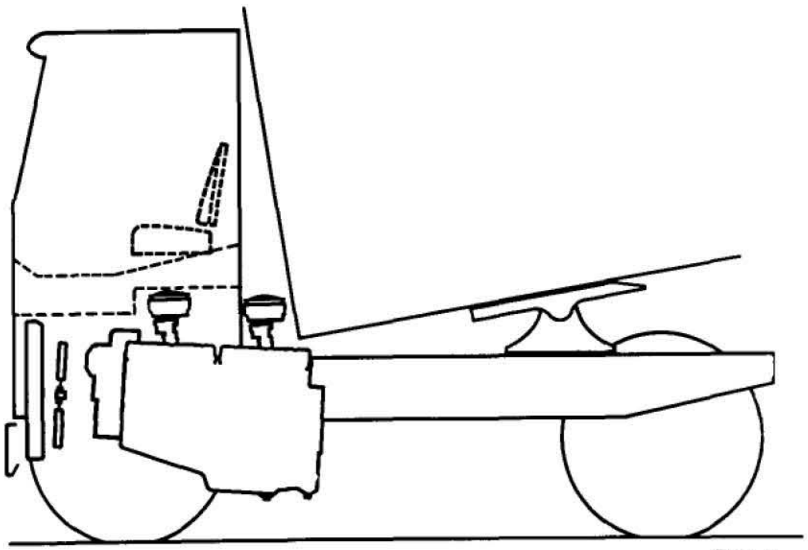


FIG. 9

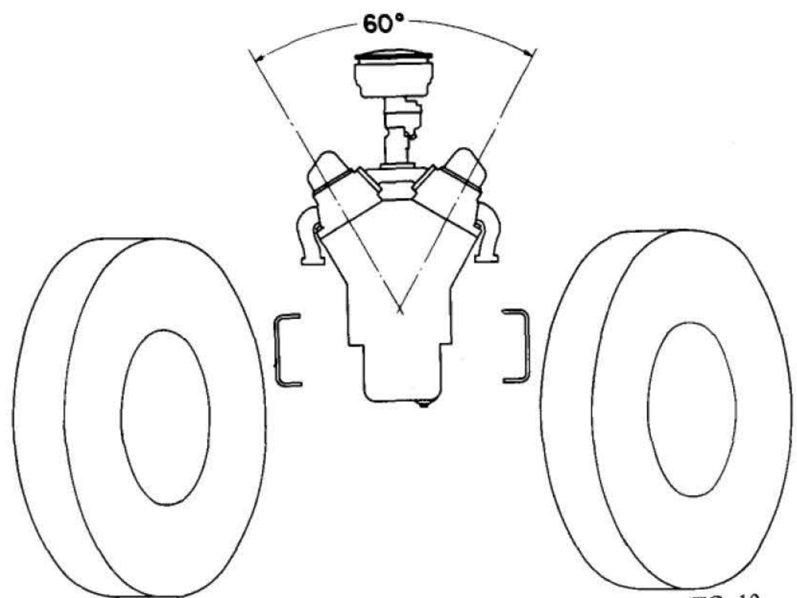


FIG. 10

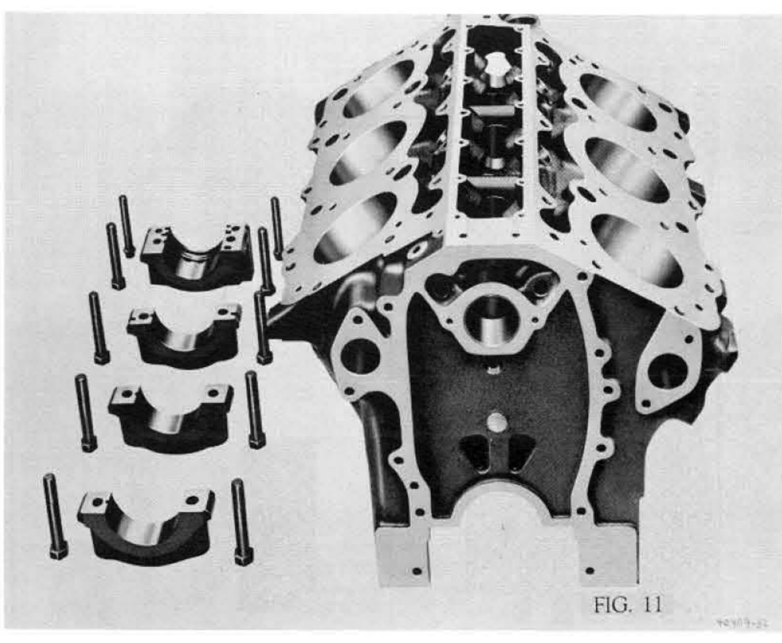


FIG. 11

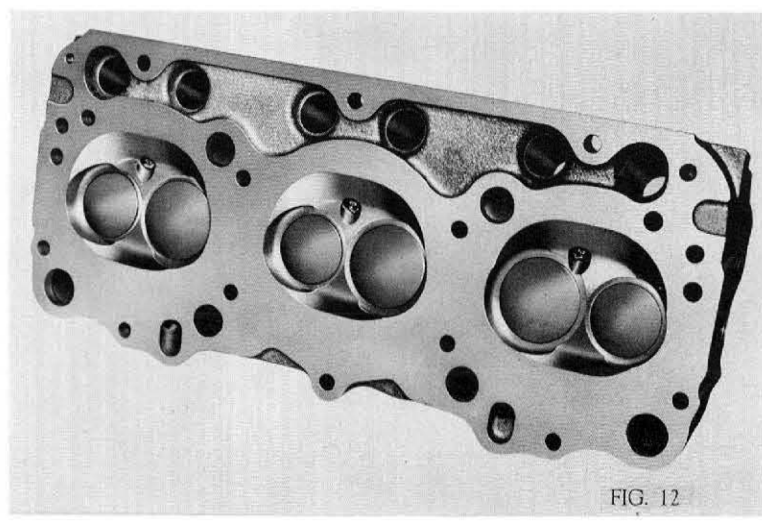
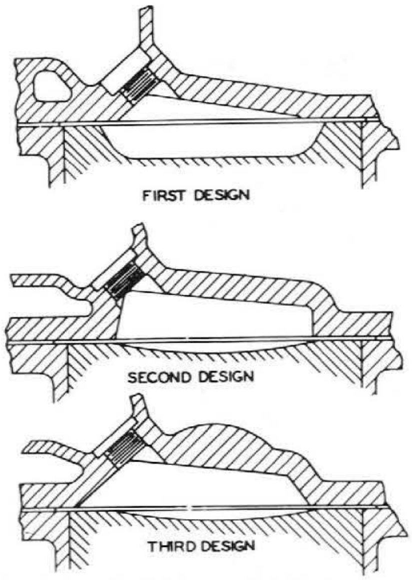


FIG. 12



CROSS SECTION OF THREE BASIC COMBUSTION CHAMBERS EVALUATED

FIG. 13

FIG. 14

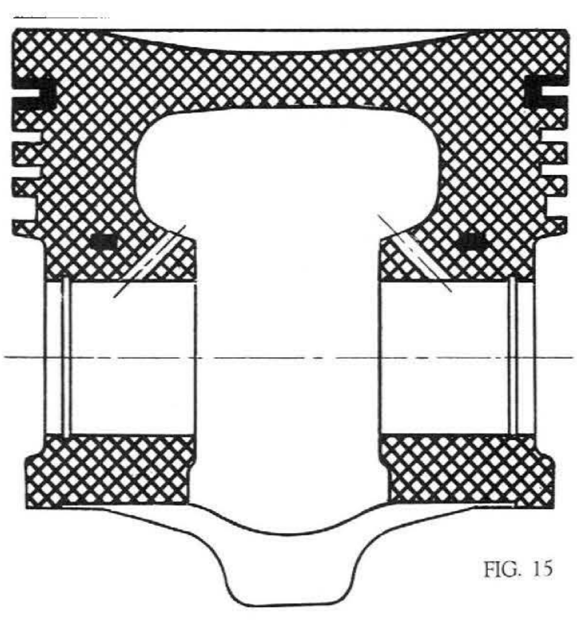


FIG. 15

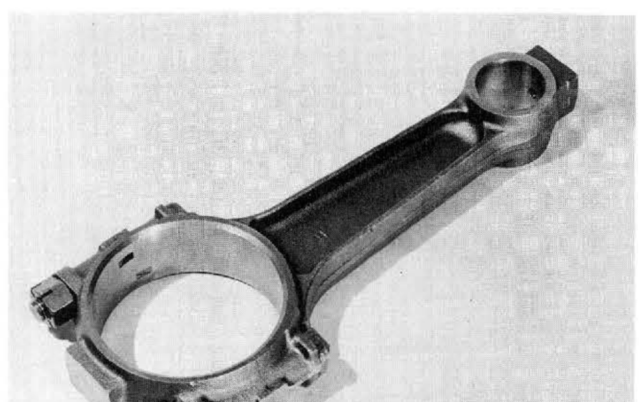


FIG. 16

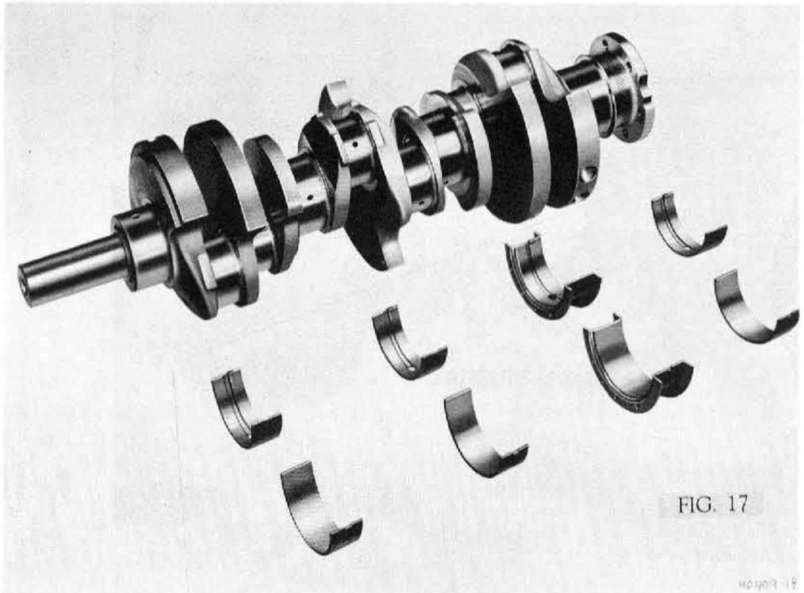


FIG. 17

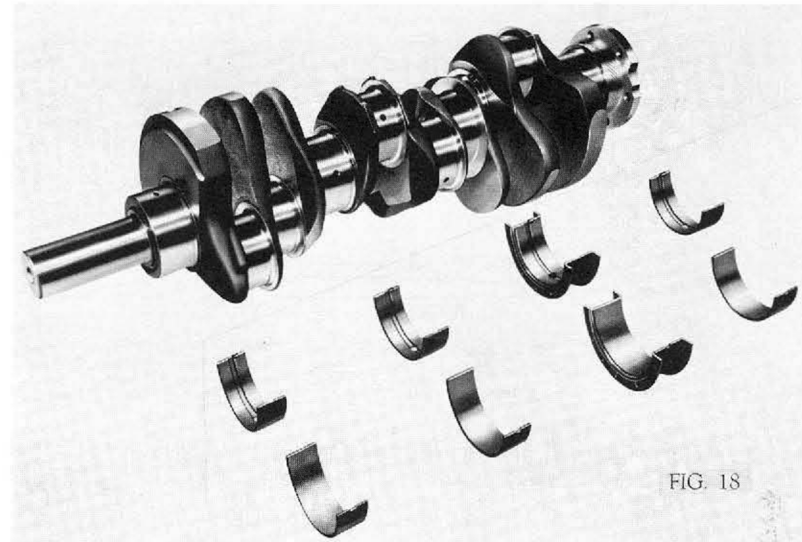


FIG. 18

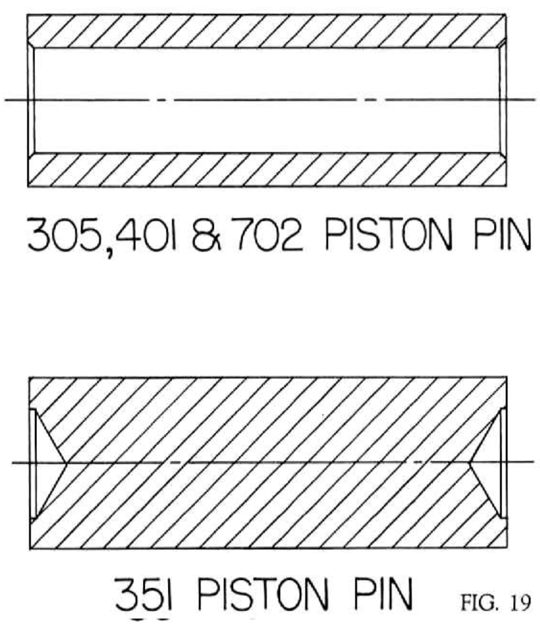


FIG. 19

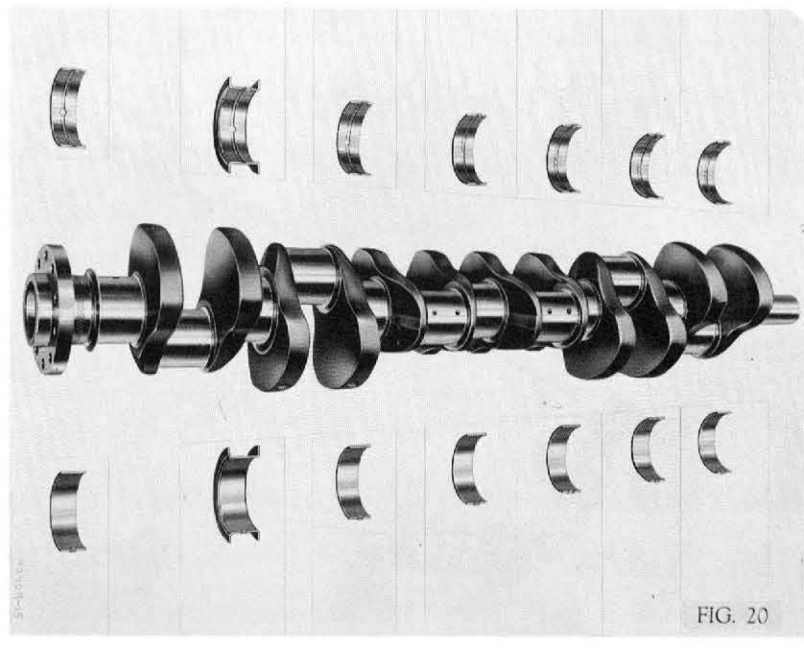


FIG. 20

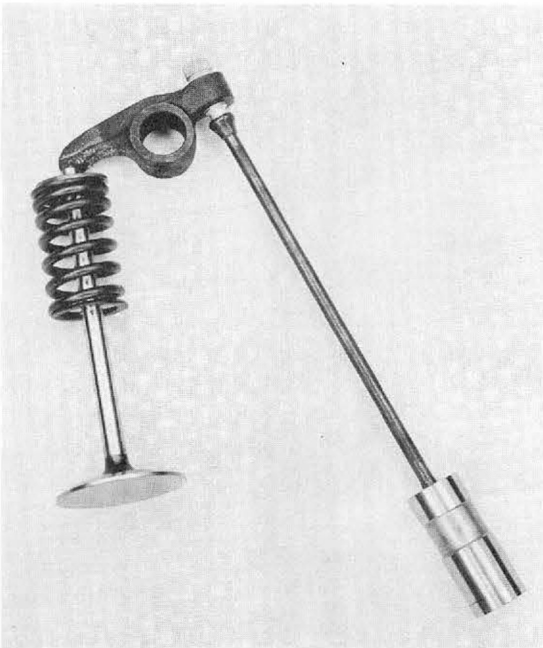


FIG. 21

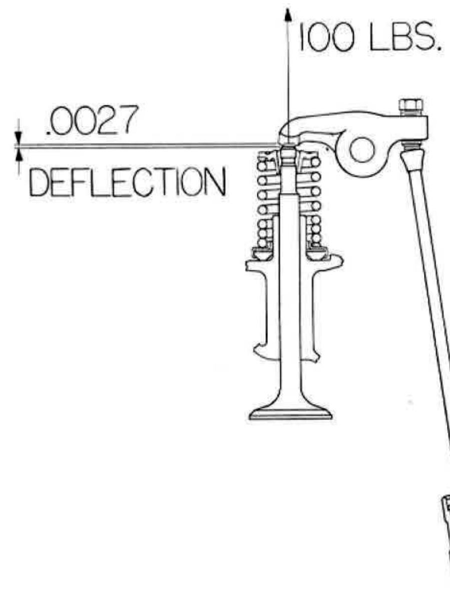


FIG. 22

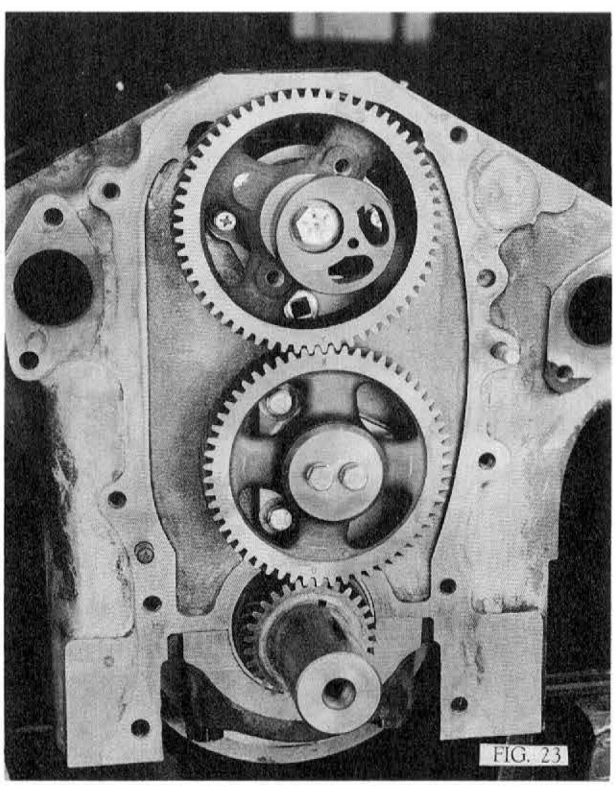


FIG. 23

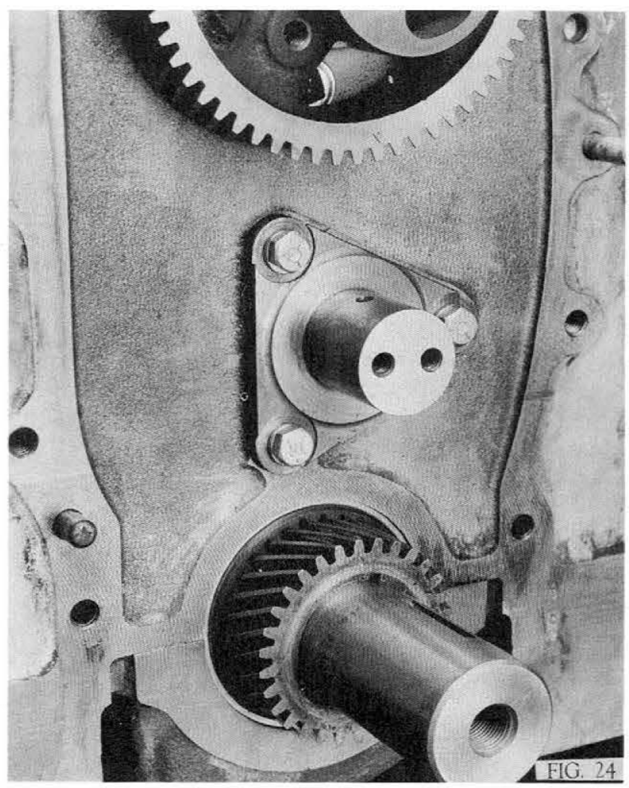


FIG. 24

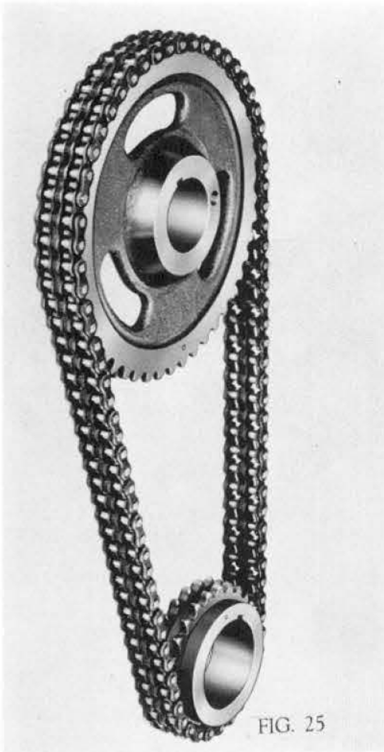


FIG. 25

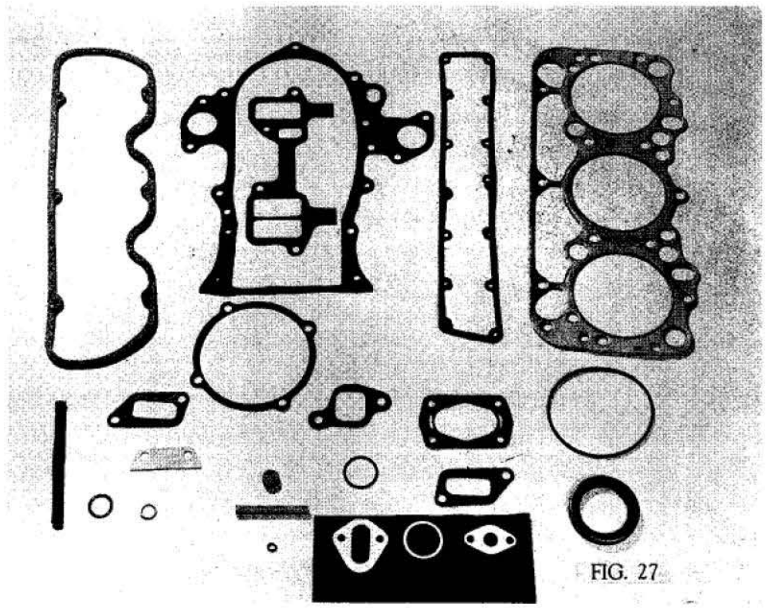


FIG. 27

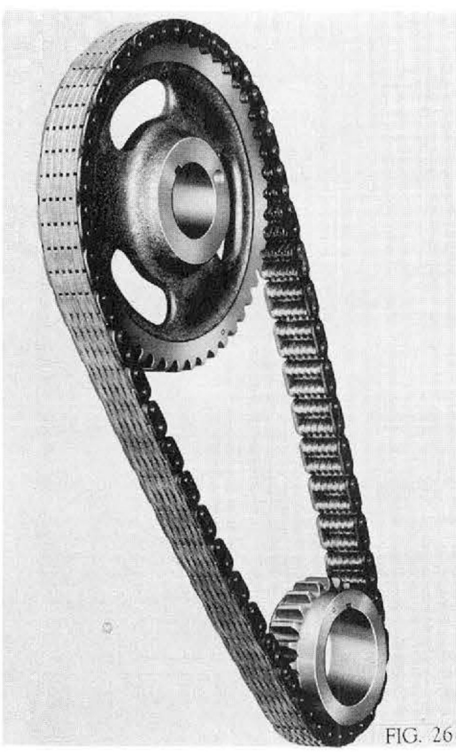


FIG. 26

V-6

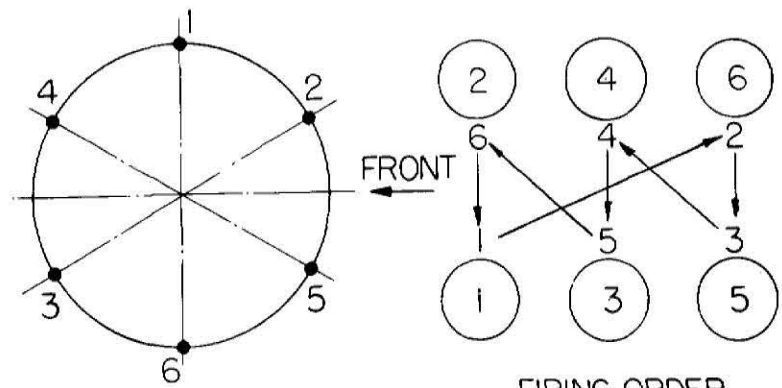


FIG. 28

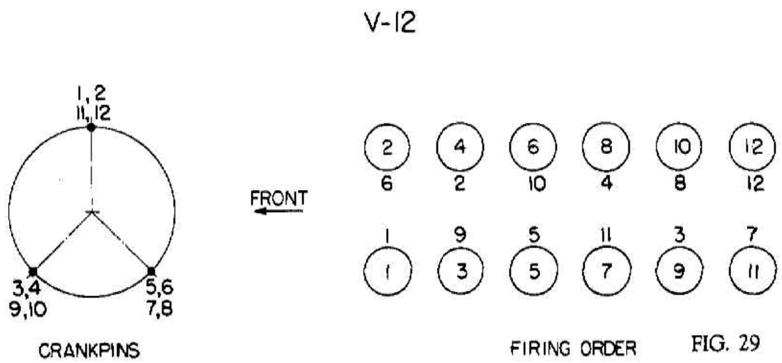


FIG. 29

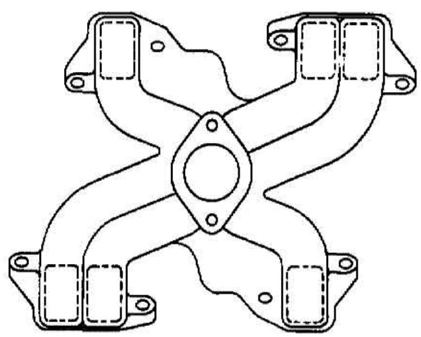


FIG. 30

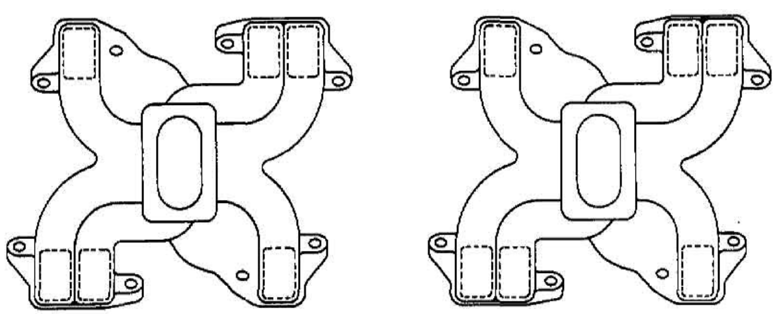


FIG. 31

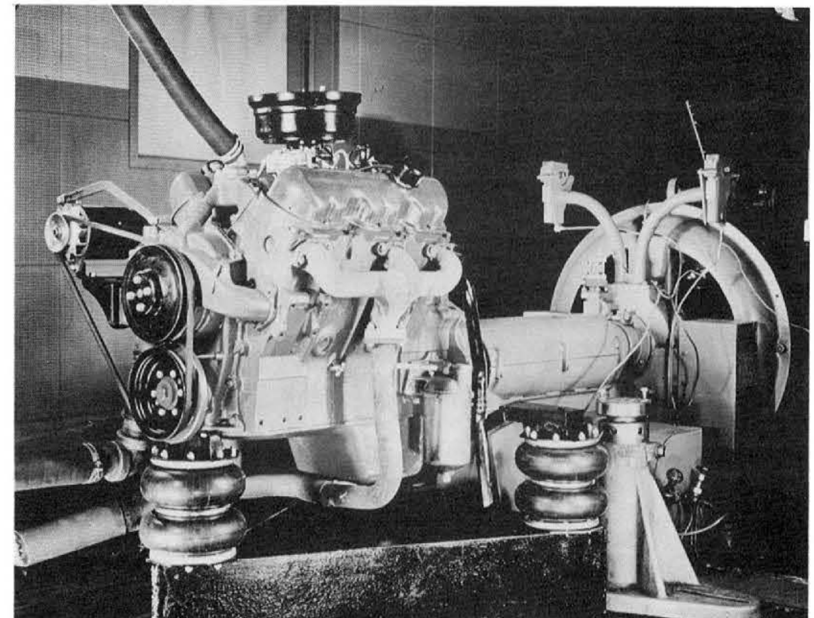


FIG. 32

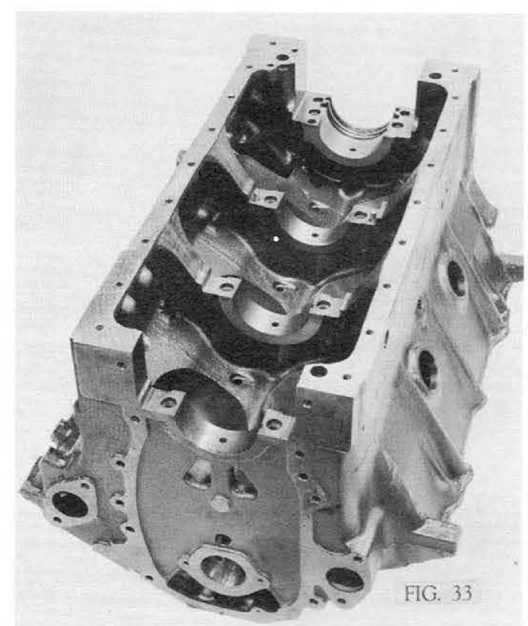


FIG. 33

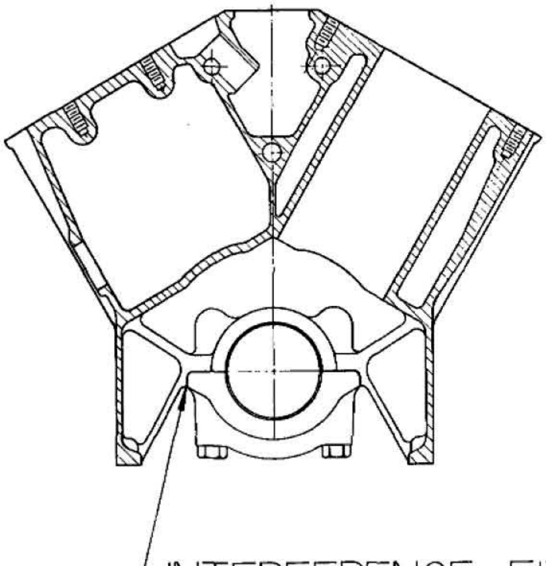


FIG. 34

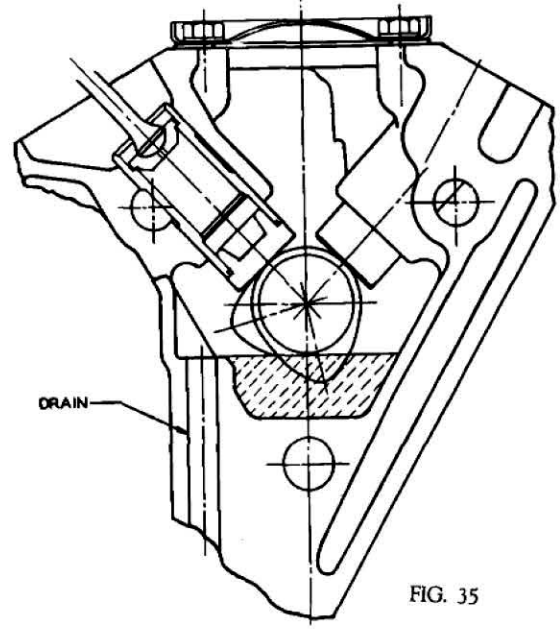


FIG. 35

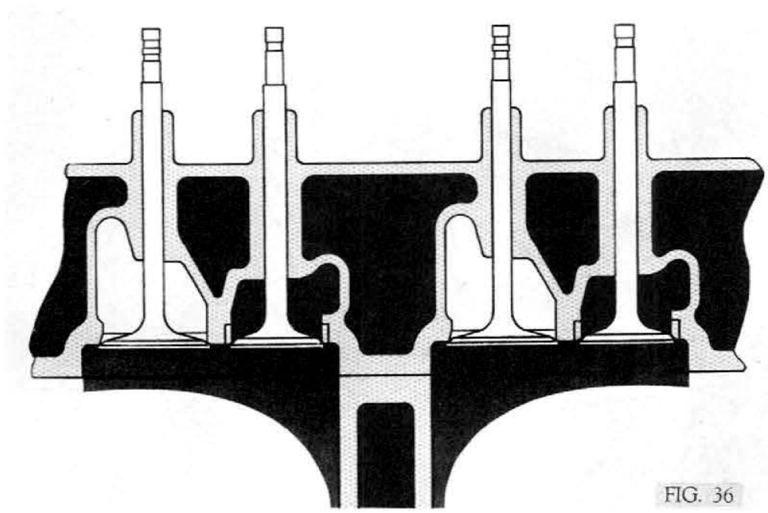


FIG. 36

351 & 401 ENGINE

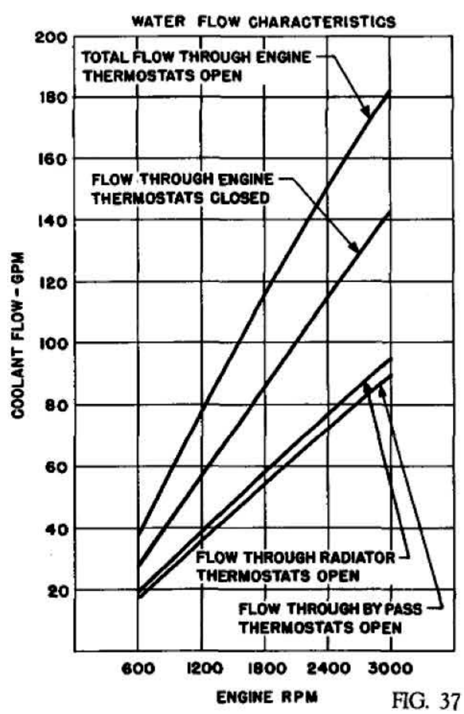


FIG. 37

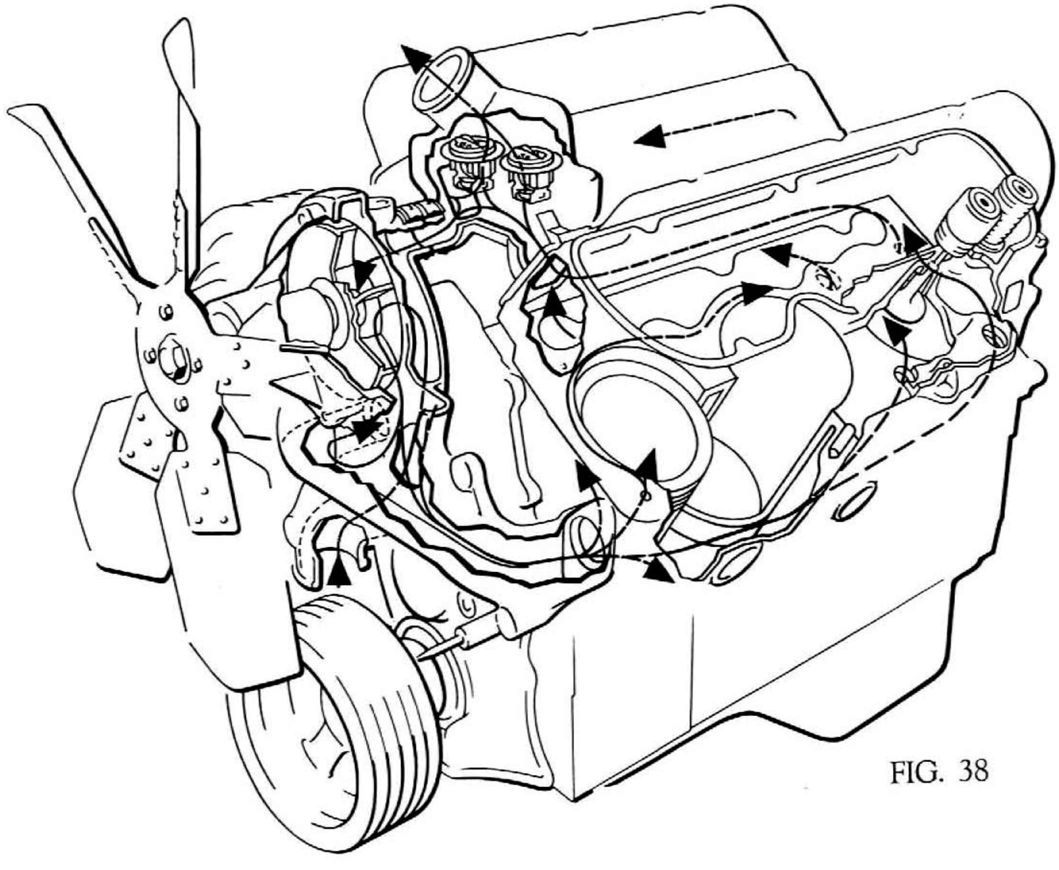


FIG. 38

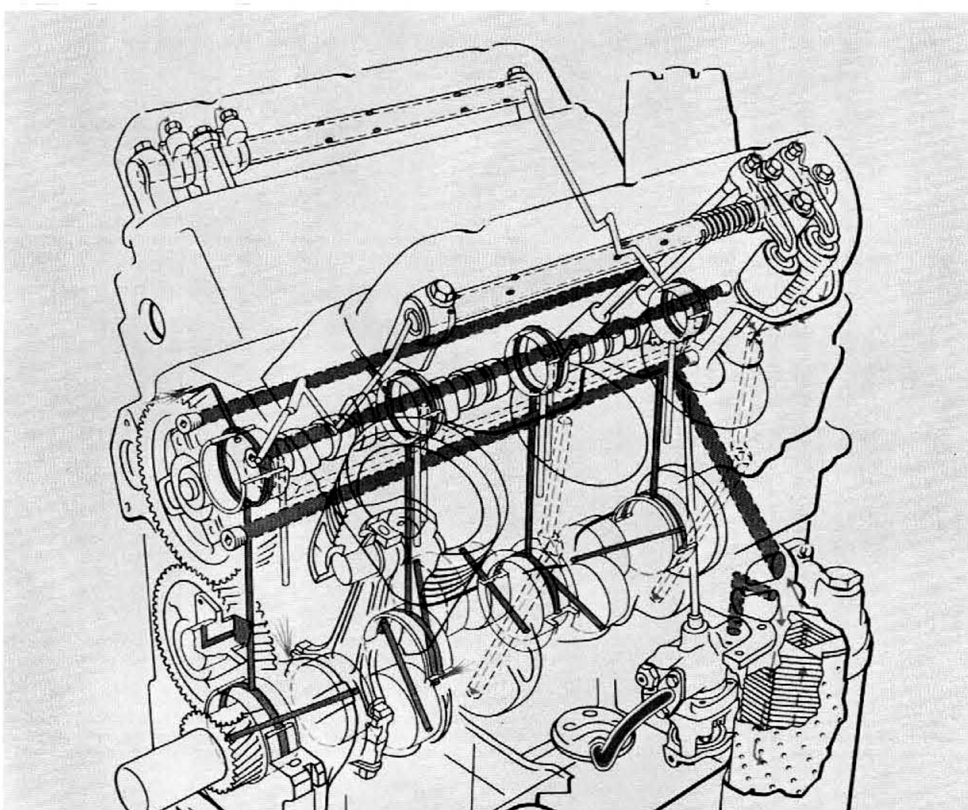


FIG. 39

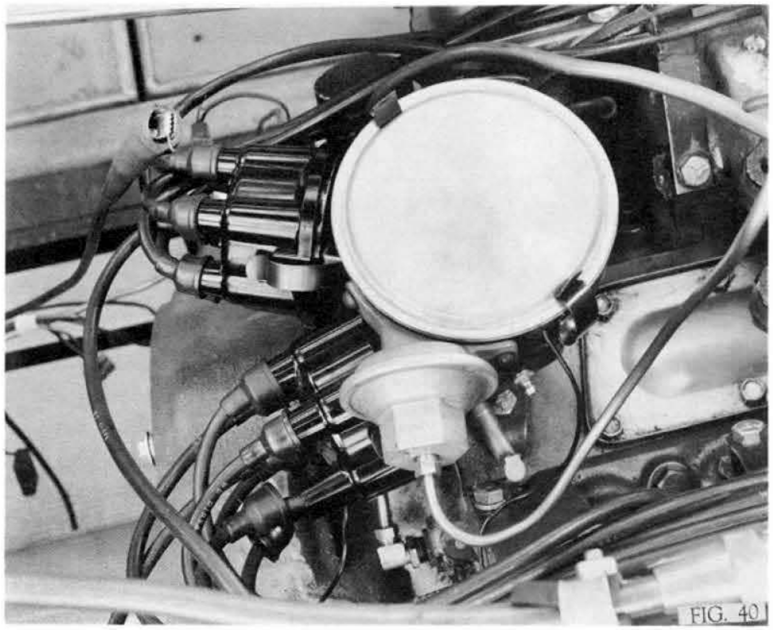
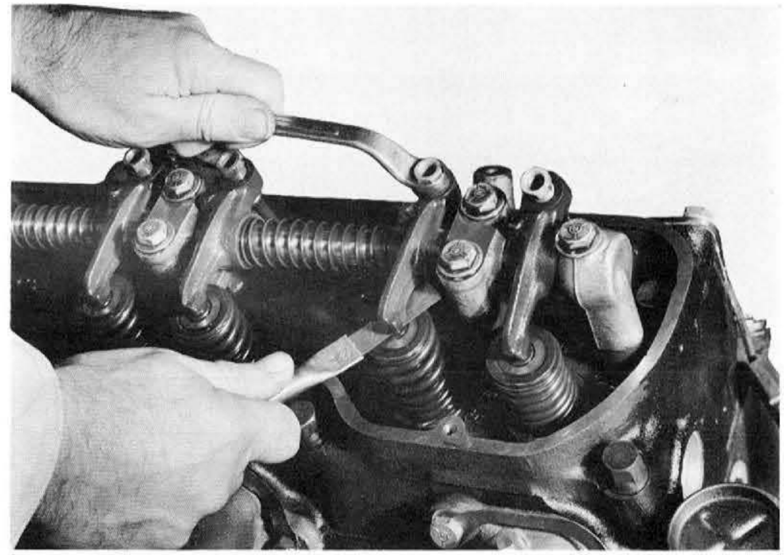
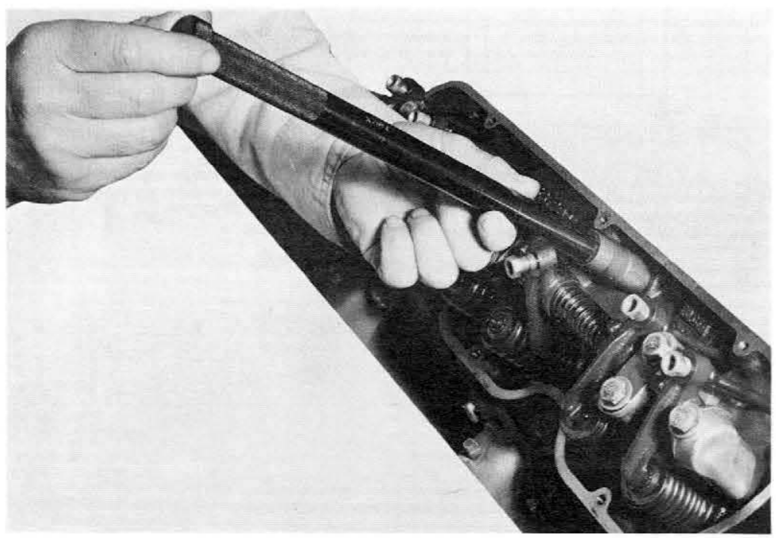


FIG. 40



VIEW SHOWING EASE OF ADJUSTING VALVE LASH

FIG. 41



VIEW SHOWING REMOVAL OF VALVE LIFTER THROUGH CLEARANCE HOLE IN CYLINDER HEAD USING LIFTER REMOVAL TOOL

NOTE THAT IT IS NOT NECESSARY TO REMOVE AIR CLEANER, CARBURETOR, AND INTAKE MANIFOLD

FIG. 42

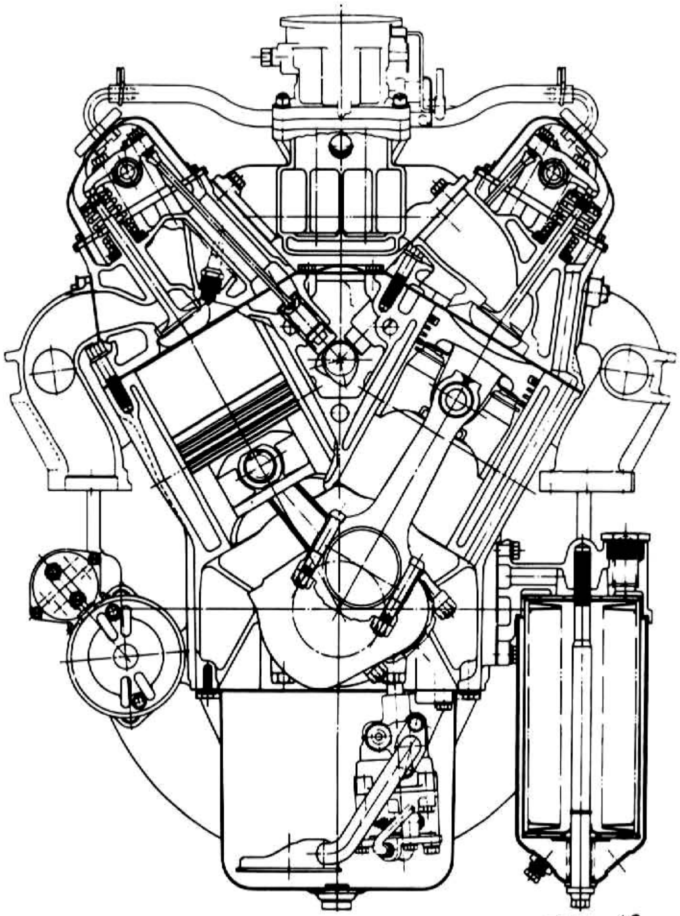


FIG. 43

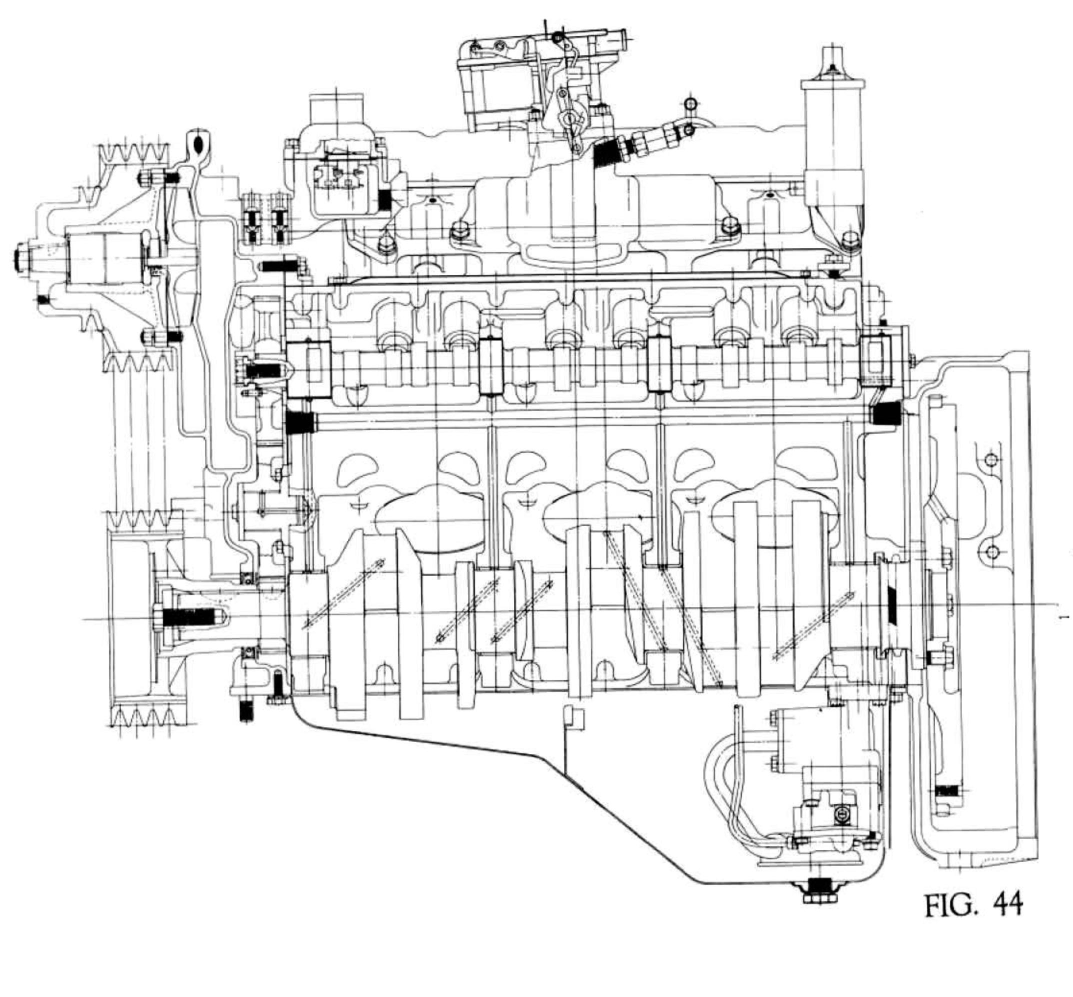


FIG. 44