# NEW BUICK V-8 ENGINE

## • Durability
## • Fuel Economy
## • Easily Serviced
## • Light and Compact

BUICK has had a long and very successful history in the production of overhead-valve, in-line engines. Pioneering in the development of the overhead-valve principle, we have produced cars from 1903 to the present time powered by engines utilizing this design feature. Starting with a 2-cyl opposed engine, Buick has produced overhead-valve 4's, 6's, and 8's from 1903 to the present time.

The question naturally arises as to the reasons behind the decision to change to V-type engines in


**AIMS and objectives behind the decision to adopt the V-type design for the large Buick engine in 1953 models are set forth in this paper. From 1903 until the present time Buick produced cars powered by engines utilizing the overhead-valve principle. The authors cite four main reasons for departing from this long tradition:**

**1. Newer styling, based on extremely low lines, demands an engine proportioned to fit within allotted space under the hood.**

**2. A compact V-engine is inherently light in weight, especially in larger sizes. Weight reduction improves car balance, handling, and performance.**

**3. Improved combustion chambers and fuels permit higher compression ratios. Resulting higher explosion pressures call for a more rigid engine structure, which can be achieved on the V-type.**

**4. Great strides in engine manufacturing made old tooling obsolete. Since new tooling was needed on the larger engine, the advantage of a type change was indicated.**

**In conclusion, the authors state that the new engine offers durability, fuel economy improvement, low production costs, light weight, and is easy to service.**


### The Authors


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# OFFERS MANY ADVANTAGES

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the Series 50 and 70 Buicks at this time. In the first place, it should be emphasized that we do not believe that the V-engine has any inherent advantages over the in-line engine in regard to power or economy. As the head of our engine test department often says, "The cylinders don't seem to care whether they stand up straight or lean over." Increased performance in some of the newer cars has been accomplished with large displacement engines and by newer refinements which can be (and in many cases have been) adapted to the in-line engine.

Why then has the change been made, and why only on the large engine? Primarily for four reasons:

1. The proportions of the V-engine are more suitable for installation in cars with the newer styling and particularly in cars with the styling which General Motors believes will be standard several years from now. The cars of the future, the XP-300 and LeSabre, have extremely low lines and the 90 deg V-type engine was chosen for these cars because it was the only design which we could fit within the allotted space under the hood.

2. The V-engine, being more compact, is inherently lighter in weight than the equivalent in-line engine, especially in the larger sizes.

The many advantages of reduced weight and size are obvious. By reducing the engine weight, steering effort is reduced, while car balance, handling, and performance are improved. Weight is also a major factor in production cost, but to realize the maximum cost saving, any reduction in weight must be obtained by good commercial design, and not by the use of more expensive materials, or by the adoption of designs which are more complicated and difficult to manufacture.

3. The V-engine also is more rigid and is structurally more suitable to withstand the higher explosion pressures resulting from the higher permissible compression ratios obtained with improved combustion chambers and improved fuels.

4. The basic tooling of the Buick Roadmaster engine has been unchanged since the introduction of the justly famous Buick Century in 1936. New tooling was indicated at this time because of the great strides which have been made in engine manufacture in the last few years. The Buick 263 cu in. in-line engine on the other hand was retooled in 1950.

As has been mentioned, Buick has never before had a V-type engine in production, but extensive research work has been done dating from a 1931 experimental twin-6. Since 1944 a continuous program of high-compression V-engine development has been in progress. A special department was set up for the purpose of investigating the possibilities of this type of engine and many cylinder arrangements, V-angles, and combustion-chamber designs were tried. During this period 10 different types of engines were built and tested, and in all over 100 experimental models were made.

One promising line of investigation of 35-deg V-engines was abandoned, although these engines had many desirable characteristics, because the carburetor height could not be reduced sufficiently to

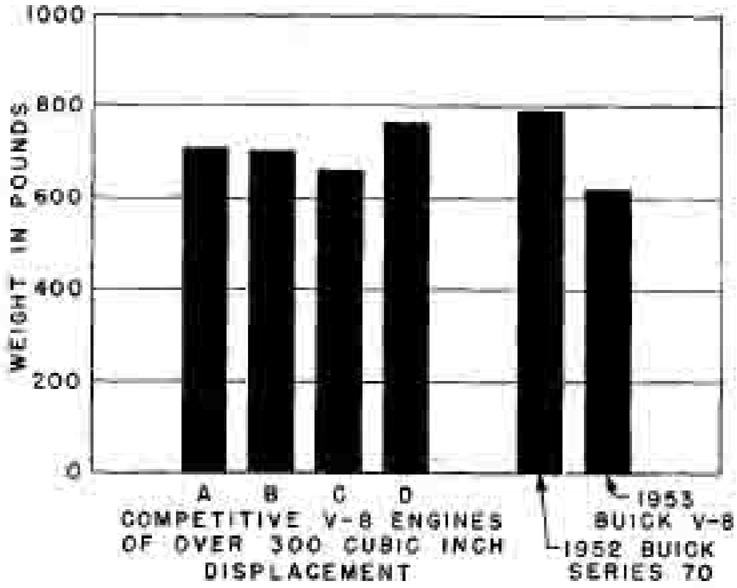


Fig. 1 – Engine total weight comparison

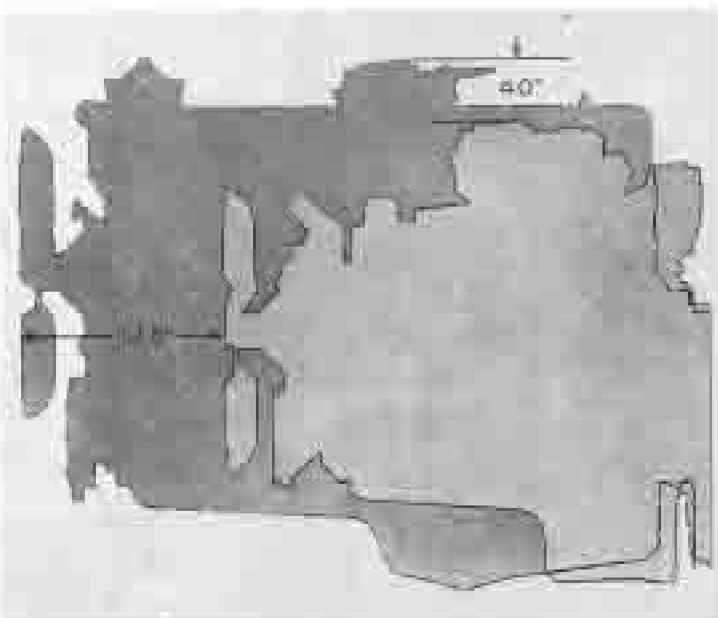

Fig. 2 – Engine size comparison. Black section indicates 1953 V-8 engine measured against gray background representing 1952 Series 70

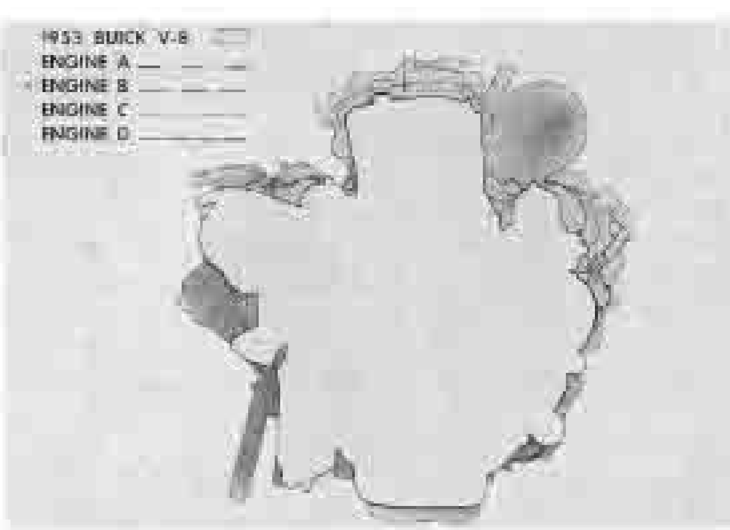


Fig. 3 – Engine size comparison

clear contemplated future hood levels. In fact, space requirements as dictated by the body stylist, are one of the most important, if not *the* most important factor, in determining the future trend of engine design.

The greatly reduced height and length of the 90-deg V-engine is a great help to the chassis designer and the body stylist, but the increased width is indeed a problem. The clearance condition is especially acute on the steering-gear side. The space between the extreme steering-lock position of the left front wheel and the left engine bank in many of the new designs leaves insufficient room for the frame side rail and steering gear, without very undesirable compromises in side rail section shape, steering-post angle and position, or turning radius; compromises which the Buick management chose not to accept. The Buick design, incorporating a stroke/bore ratio of 0.8 and a vertical, in-line positioning of the valves served to minimize the overall engine width. These engine design features accompanied with a 1-in. increase in front tread width permitted the continuation of all chassis improvements developed with the former in-line engine.

The new Buick V-8 engine has a 3.2-in. stroke and a 4-in. bore, resulting in the aforementioned stroke/bore ratio of 0.8 which is, we believe, the lowest used in any American production engine at the present time. The short-stroke, big-bore adopted in all the new V-8 engines has many real advantages, but the stroke/bore ratio of the new Buick engine was chosen chiefly because, from our investigation, it produced the minimum external engine size. The 322 cu in. displacement of the new engine as compared to the 320 cu in. displacement of the straight-8 Roadmaster, is considered ample in view of the power gains obtained from some of the newer design features to be discussed later.

The Fireball V-8 engine program was started in March, 1950, and, because of the difficult machine-tool situation, production was delayed until this year. The tool and building program was enormous, but it is now completed, and the new engines are in production. Needless to say, we are very happy with them.

## Objectives

Probably every designer on attacking a new project is fired with a desire to produce a finished product so perfect as to excel in every detail anything that has been done before. As the design progresses, invariably some balancing of objectives must be done, and the individual emphasis on the many design factors will determine the trend of the final design.

In planning the new Buick engine no compromise could be accepted in the requirements of a good powerplant; that it should be powerful, efficient, smooth, quiet, durable, and easily serviced.

However, these objectives were to be realized with the lightest and most compact package which we could produce, consistent with minimum manufacturing cost.

### Weight and Size

The greatest weight saving in the new Buick engine was obtained by adopting the inherently lighter 90-deg V-8 principle. But we went further. Every design detail was investigated for possible further weight reduction. Many of the old rule-of-thumb drafting room design standards were found to result in overdesigned and overweight parts. The reduction in size of lightly loaded flanges and bolt bosses, for example, provided worthwhile savings when applied to the entire engine.

In designing the stressed parts of the engine for minimum weight, consistent with adequate fatigue life, full advantage was taken of the fact that an automobile engine operates most of the time at part throttle. In any of the short stroke-large bore engines the calculated gas pressure loads and stresses are extremely high, and the inertia loads are correspondingly low. Luckily, experience has shown that even under severe road testing, fatigue life of the parts is more dependent on the inertia loading than on the gas pressure loading. This result is largely explained, of course, by the relative number of cycles of each type of loading which the engine undergoes in road operation.

In establishing the endurance standards for the new Buick engine we took the position that while the engine must be capable of withstanding our standard full-throttle high-speed dynamometer test, and not less than 200 hours of full-throttle power development running, the true gage of engine life was to be the behavior on the road.

The production-built Buick V-8 engines are 170 lb lighter than the 1952 Roadmaster engines (Fig. 1), and are lighter than any of the competitive V-8 engines of more than 300 cu in. displacement. The weights shown are based on the dry engine weight with all standard accessories, without transmission or clutch.

In addition to the compact design features discussed previously, the old standard design proportions and running clearances were re-examined, and where practicable, revised to save space. For instance, a very small length/diameter ratio of the piston, and its 1/16-in. running clearance with the counterweights aided in reducing the height and width of the crankcase.

The Buick V-8 engine is 4 in. lower and 13½ in. shorter than the former Roadmaster engine (Fig. 2), but the width is considerably greater. However, it compares very favorably with that of competitive V-8 engines as shown in Fig. 3.

### Power Output and Efficiency

As has been stated, in obtaining a light, compact unit, no sacrifice in engine output and efficiency

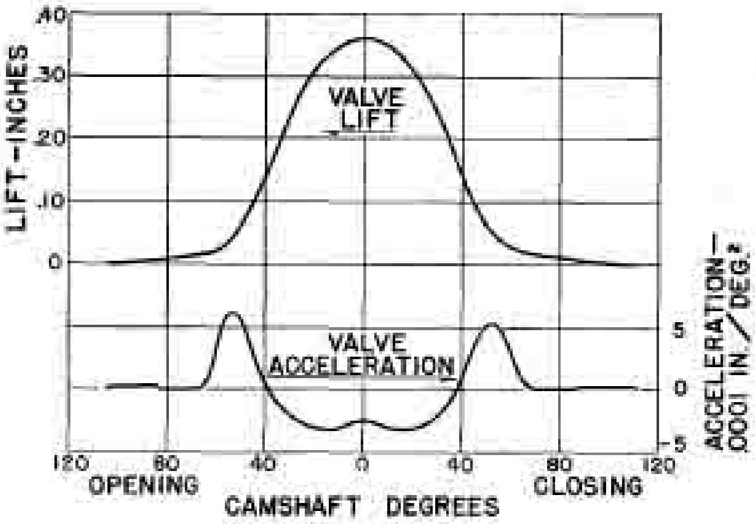

Fig. 4 – 1936 cam design

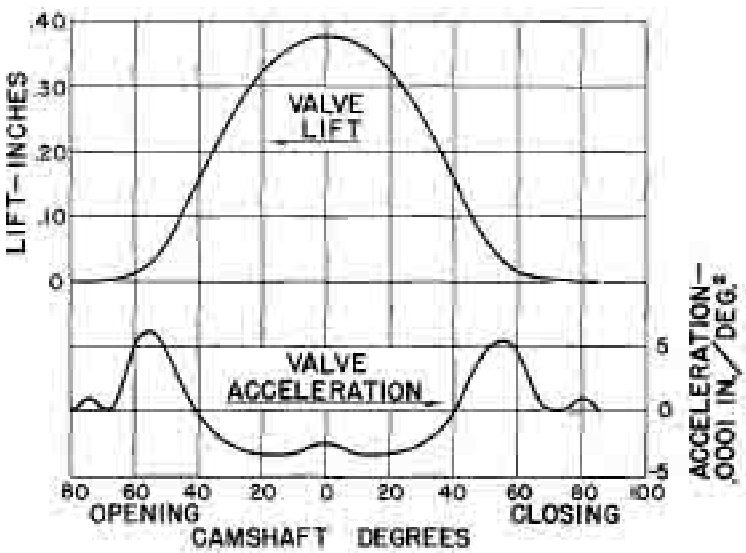

Fig. 5 – 1953 cam design

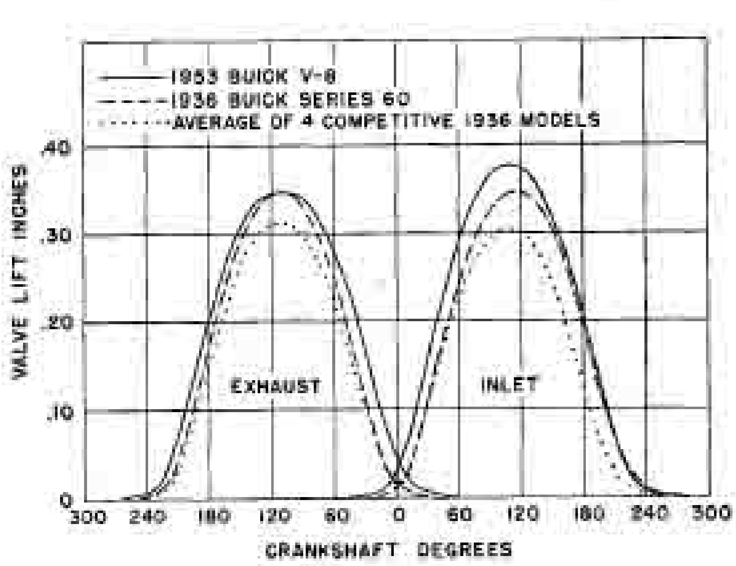

Fig. 6 – Valve lift comparison

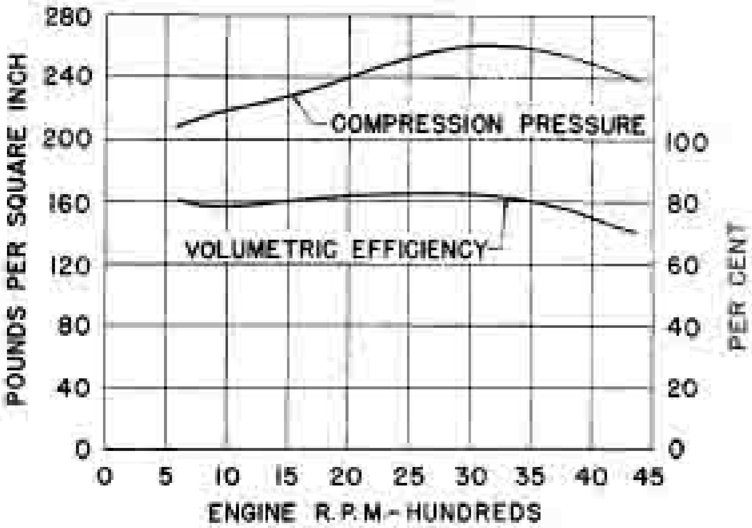


Fig. 7 – Compression pressure and volumetric efficiency

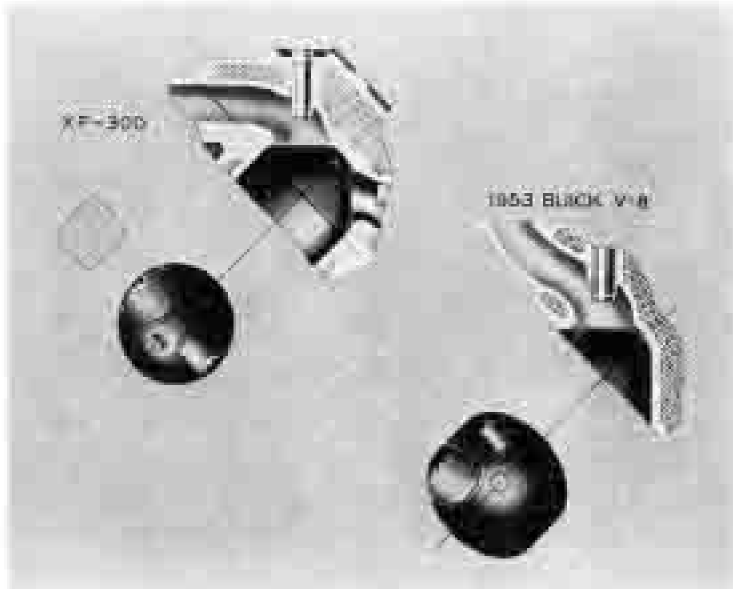


Fig. 10 – Inlet valve port comparison

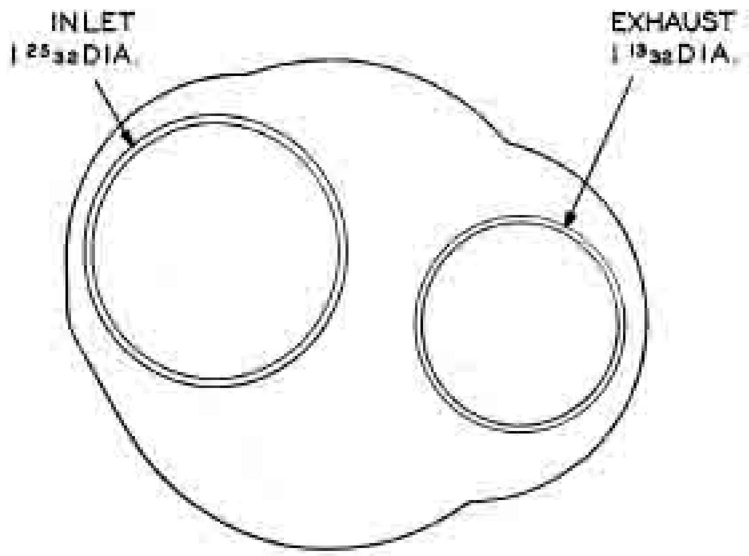


Fig. 8 – 1918 valve size comparison

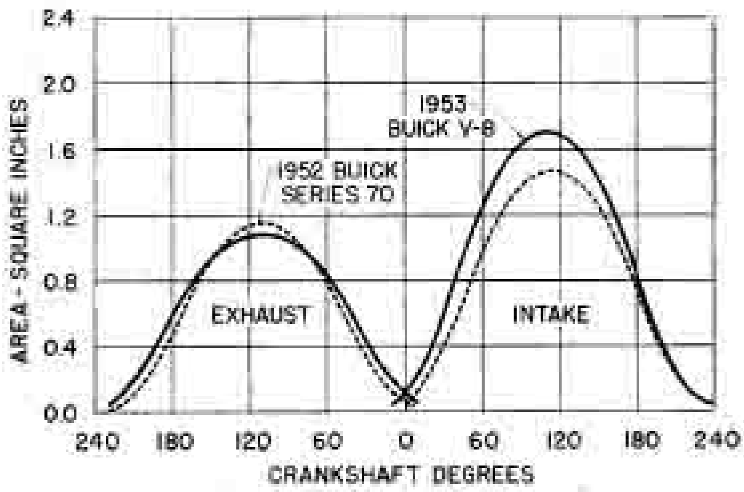


Fig. 11 – Effective valve opening area comparison

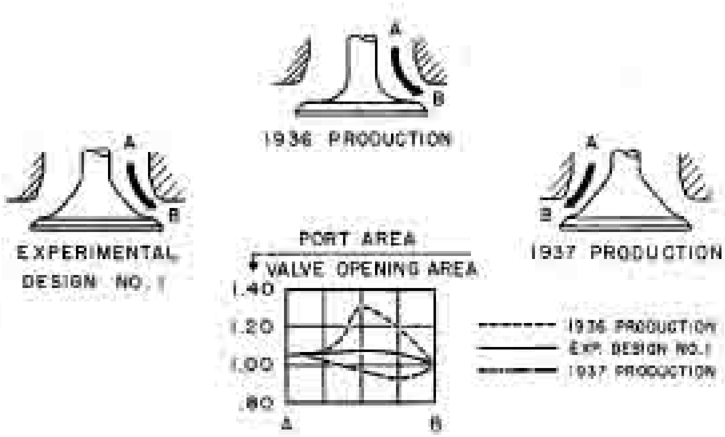


Fig. 9 – Streamlined inlet valve development

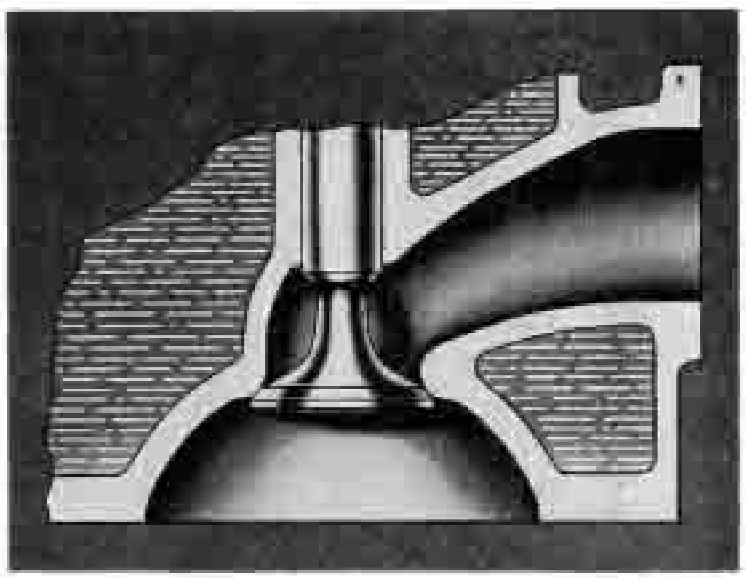

Fig. 12 – 1936 streamlined exhaust port

could be accepted. There has been much discussion on the relative merits of designs which emphasize high volumetric efficiency and those which improve the mechanical octane rating and permit the use of higher compression ratios, with consequent higher power and economy. It is difficult to see why there should be anything antagonistic in these two approaches to higher engine power and efficiency. Buick has carried out a continuous development program on improving the overhead-valve engine along both lines, and has made many important engineering contributions. Some of these improvements have been claimed as new discoveries by companies which have recently adopted the overhead-valve principle, notwithstanding the fact that many of these same features have been in production at Buick for many years.

Some of the earlier Buick developments which improved volumetric and combustion efficiency and which influenced the Buick V-8 design will be discussed briefly. Later the engine will be described in more detail with some discussion of the engineering problems encountered.

### Cam Design and Valve Timing

Push-rod-operated overhead-valve engines require much more highly developed cams than do L-head engines operating in comparable speed ranges, because of the greater deflection of the valve mechanism parts under operating conditions. Buick has always been confronted with this problem and, as engine operating speeds increased, cam and valve mechanism design had to be improved to meet the requirements.

By 1936 Buick had developed a new type of cam profile (Fig. 4) which has been used in production since that time. All recently developed overhead-valve V-engines have adopted cams of similar contour. Briefly, the Buick cams are designed with an acceleration curve of generally sinusoidal form with all changes in acceleration, and therefore in load, as smooth and gradual as possible.

The cam profiles of the Buick 90-deg V-engine (Fig. 5) have been carefully tailored to suit the elastic characteristics of the valve mechanism and to minimize spring surge. The pump-up speed of the hydraulic lifters is 5500 rpm for the exhaust valves and 5300 rpm for the inlet valves. These speeds are high enough above the normal top operating speeds to provide an ample factor of safety, and are quite gratifying considering the high inlet valve lift of 0.378 in. and the relatively low valve spring loads of 62 lb closed and 144 lb open.

Buick developed a high-lift high-speed timing for the 1936 Century models (Fig. 6), and the basic principles of this timing are still used. The Buick V-8 has an even wider timing with the higher inlet valve lift already mentioned. The resultant volumetric efficiency is excellent as is shown by the volumetric efficiency and compression curves (Fig. 7). Note that the volumetric efficiency is above 75% at 4000 rpm.

### Inlet Valve and Port Design

As early as 1909 Buick recognized the value of an inlet valve of much larger size than the exhaust valve and the Buick-Bug race car which exceeded 105 mph in 1910 utilized this feature. Buick adapted this principle to its 1918 production engines (Fig. 8) with a 1 25/32-in. diameter inlet valve and 1 13/32-in. diameter exhaust valve which is very close to the size and proportion used in most of the new engines today.

Throughout the middle '30's the Buick engineering department was very active in investigating the effect of manifold, port, and valve configuration on flow resistance over the full range of valve openings. Best results were obtained by maintaining a nearly uniform velocity in the whole induction system from the carburetor to the exit side of the inlet valve seat. The valve throat area is very important, and a considerable reduction in flow resistance has been obtained by reducing the inlet throat area to approximately the maximum valve opening area. The Buick streamlined inlet valve (Fig. 9), adopted in 1937, with its conical section at the junction of the head and stem was designed to avoid a sudden enlargement in cross-sectional area at this point with consequent energy loss.

The in-line valve arrangement was retained in the 1953 Buick engine chiefly to make the engine more compact, to save weight, and to facilitate manufacture.

Surprisingly enough, the flow restriction of the Buick V-8 inlet port and valve was no greater than that of the experimental engine used in the XP-300 and LeSabre cars, under the same conditions (Fig. 10) with the same size valve and valve openings. This experimental engine was designed for high output and had both valves on the transverse centerline of the combustion chamber. Upon investigation, the curving inlet port of the Buick V-8, which was so shaped because of clearance conditions in the head, was shown to have a slight advantage over the straight-in port of the XP-300, because of the effect on flow of valve stem and guide interference.

### Exhaust Valve and Port Design

The size of the Buick V-8 exhaust valve was based on earlier investigations, but its adequacy has been thoroughly proved by further testing of different valve sizes in the V-8 engine. The most critical part of the exhaust event is the blowdown period, that part of the cycle from the time the valve opens to a little past bottom dead center. A valve opening and port cross-section area sufficiently large to handle this portion of the cycle has been shown by test to be adequate for the remainder of the exhaust stroke.

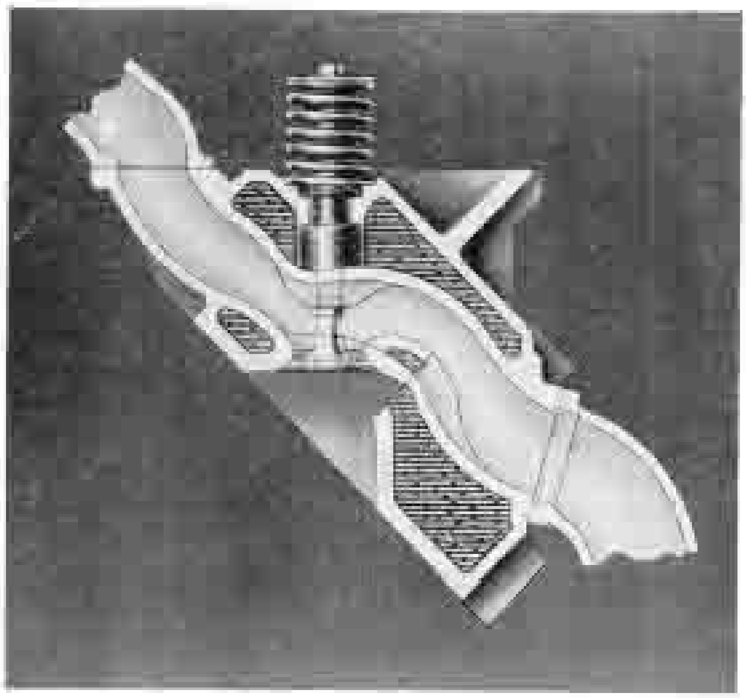

Fig. 13 – Exhaust port at cross-over

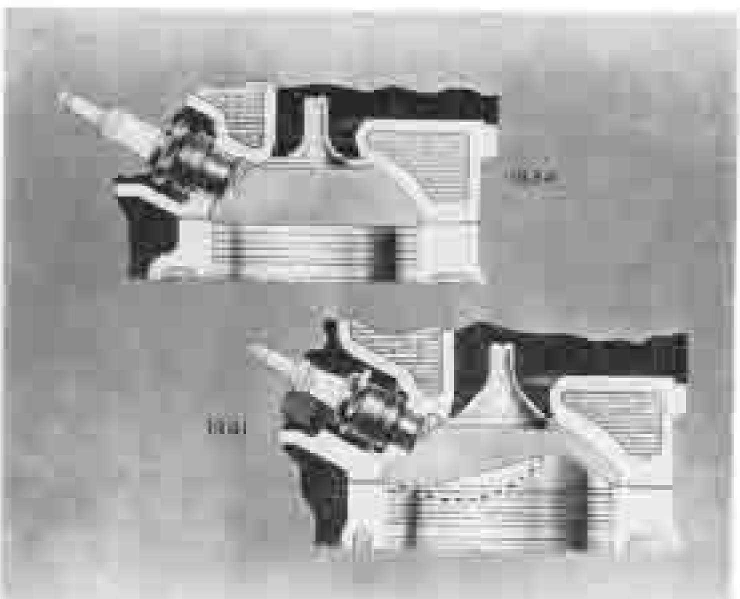

Fig. 14 – Earlier Buick combustion chambers

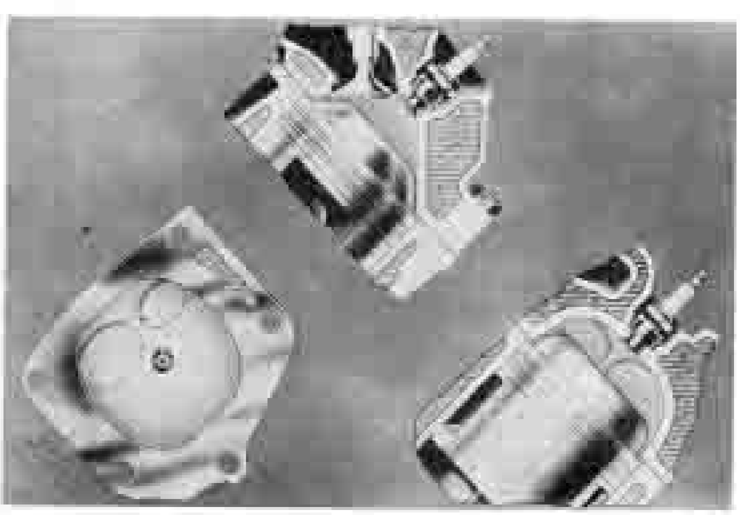

Fig. 15 – Buick V-8 combustion chamber

At bdc the exhaust valve in the average engine has opened less than half of its maximum lift. Therefore, a smaller valve opening earlier and higher can duplicate, through the critical range, the opening area of a larger valve (Fig. 11) with the added advantage that flow efficiency is maintained by avoiding the sudden expansion into a large port.

The use of this relatively small exhaust valve made a more compact combustion chamber possible, improved valve cooling, and reduced cost, without entailing any loss in power. The reduced valve weight brought the exhaust pump-up speed well above that of the inlet valve, thus providing insurance against exhaust valve breakage.

As shown by the 1936 port design (Fig. 12), Buick engineers have long recognized the benefits of streamlined exhaust ports. Considered as the exit side of a venturi, they are designed with a gradually increasing cross-sectional area from the valve throat into the main section of the manifold to obtain all the pressure recovery possible.

Because of the valve arrangement adopted for the Buick V-8, the exhaust ports (Fig. 13) must cross the combustion chamber, but the area of the port exposed to water is minimized by the reduced width of the head, the angle at which the exhaust flanges are placed, and by the relatively small diameter of the ports.

### Combustion-Chamber Design

Although many valve arrangements and head shapes had been tested previously, the best combustion chamber developed at Buick by 1934 (Fig. 14) had a semihemispherical shape in the transverse section but was necessarily elongated in the longitudinal section to accommodate the valves. The piston top was flat and no quench area was provided. A further improvement in the combustion chamber was obtained with the turbulator piston adopted in 1938, modified in 1941, and used in production up to the present time. This design provided a considerable increase in turbulence and brought the bulk of the charge closer to the spark plug. All the combustion-chamber investigations made at Buick have shown the great importance of short flame travel and high turbulence in reducing octane requirements, although increased engine roughness in past models has prevented full benefit from being obtained from these factors.

In planning the new large bore engine a centrally located spark plug was considered mandatory. In addition to the flame travel considerations, there is considerable evidence that the centrally located spark plug provides improved part-throttle ignition characteristics.

In the production design (Fig. 15), the Buick combustion chamber is of symmetrical inverted-V form with the spark plug at the apex and the valves in line longitudinally at a 45 deg angle to the cylinder axis. The spark plug is more nearly centrally located than in any other American automobile engine.

The combustion chamber is fully machined for more accurate volume control. The piston crown is raised and is shaped to conform closely to the head shape, with a flat top surface. The resultant combustion chamber is very compact with minimum flame travel from the spark plug to the extreme edge of the effective portion of the combustion space. The close clearance space provided around the lower part of the piston crown results in high turbulence in the combustion chamber during the latter part of the compression stroke. The excellent combustion-chamber characteristics of the Buick V-8 have made possible the use of the very high compression ratio of 8.5/1, using premium fuel, with remarkable freedom from combustion harshness.

General Motors Research collaborated with Buick in developing the new Buick combustion chamber and piston-top shape, and tested many cylinder-head and piston-top designs for the new Buick engine, including two without quench areas, before the final design was approved. The General Motors research experience and new techniques in combustion-chamber testing have been extremely valuable. Their work on the combustion process has been most thorough for many years, and recently they have undertaken a very comprehensive testing program in which practically every production combustion chamber as well as a great many which are still experimental have been evaluated.

### Covers and Exhaust System

The compact design of the 1953 V-8 engine is immediately apparent (Fig. 16). The horizontally mounted rocker-arm covers and vertically mounted spark-plug covers carry on the traditional Buick arrangement. The spark-plug covers protect the ignition wires from moisture and excess radiant heat from the exhaust manifold. These covers are also effective in reducing television interference.

The exhaust manifolds are of the 4-port type, to avoid the exhaust overlap periods resulting from siamesed ports, with the left-hand manifold outlet directed forward to shorten the crossover exhaust pipe which passes under the front of the oil pan, and to move this source of radiant heat away from the starter. The heat control valve is located in the left-hand manifold at the outlet flange. The right manifold outlet (Fig. 17) is directed backward and outward so that the exhaust pipe will clear the full-flow oil filter at the rear of the engine.

### Generator

The generator is mounted on the right exhaust manifold, in the position adopted by other General Motors engines. The relative location of the generator and spark plugs is such that all plugs can be serviced without disturbing the generator (Fig. 18).

Fig. 16 – Left side of Buick V-8

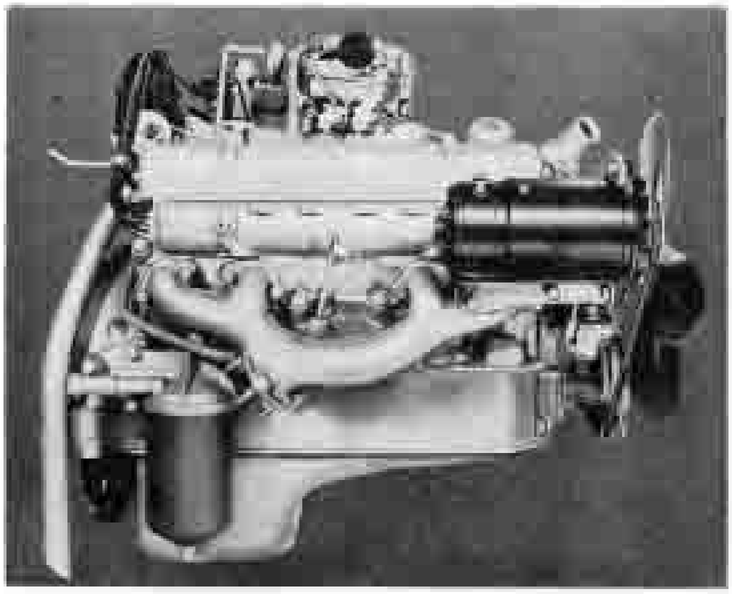

Fig. 17 – Right side of Buick V-8

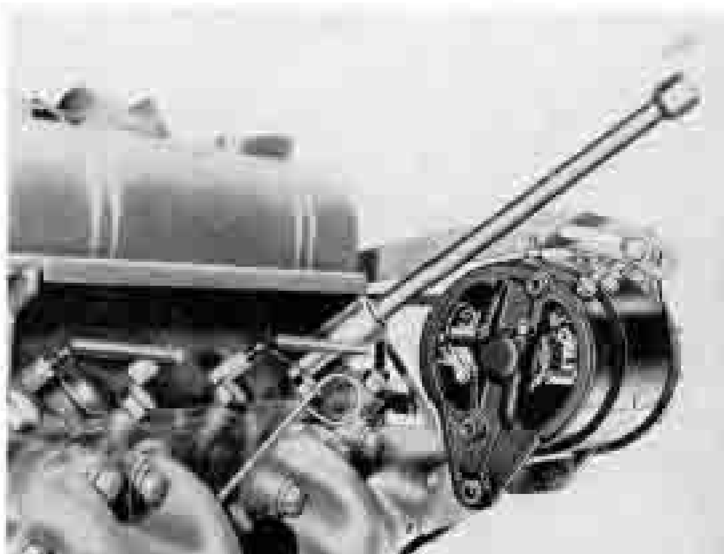

Fig. 18 – Spark-plug removal clearance with generator

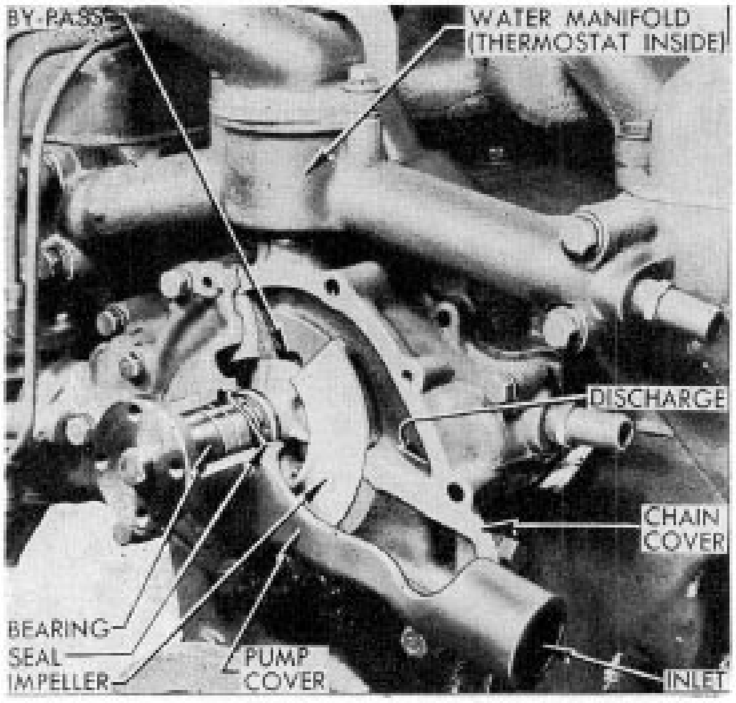


Fig. 19 – Water pump section

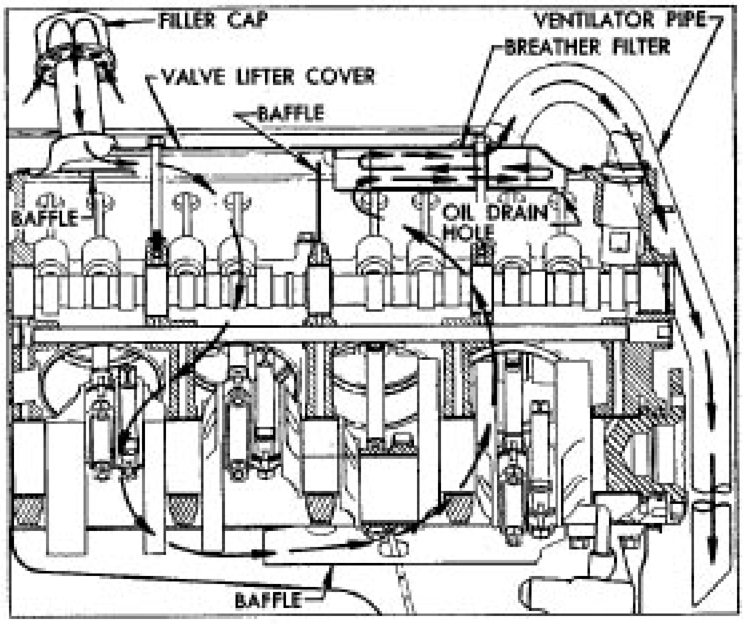


Fig. 20 – Crankcase ventilating system

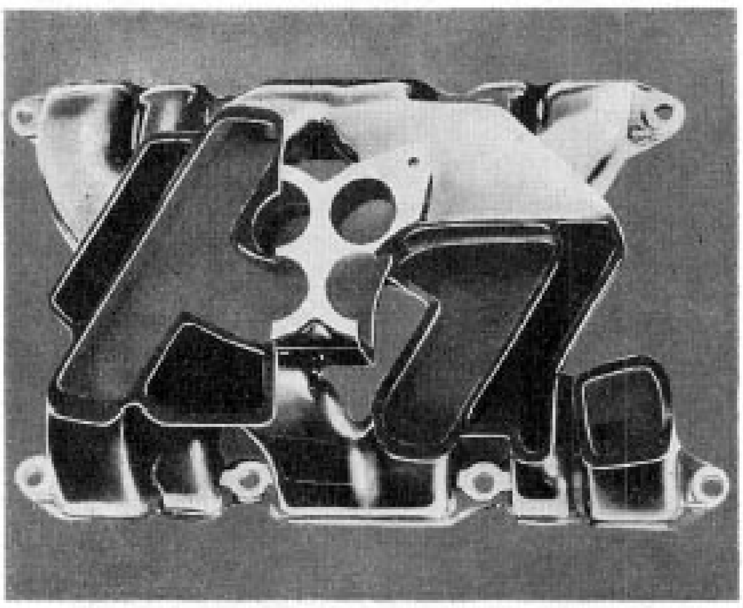

Fig. 21 – Inlet manifold section

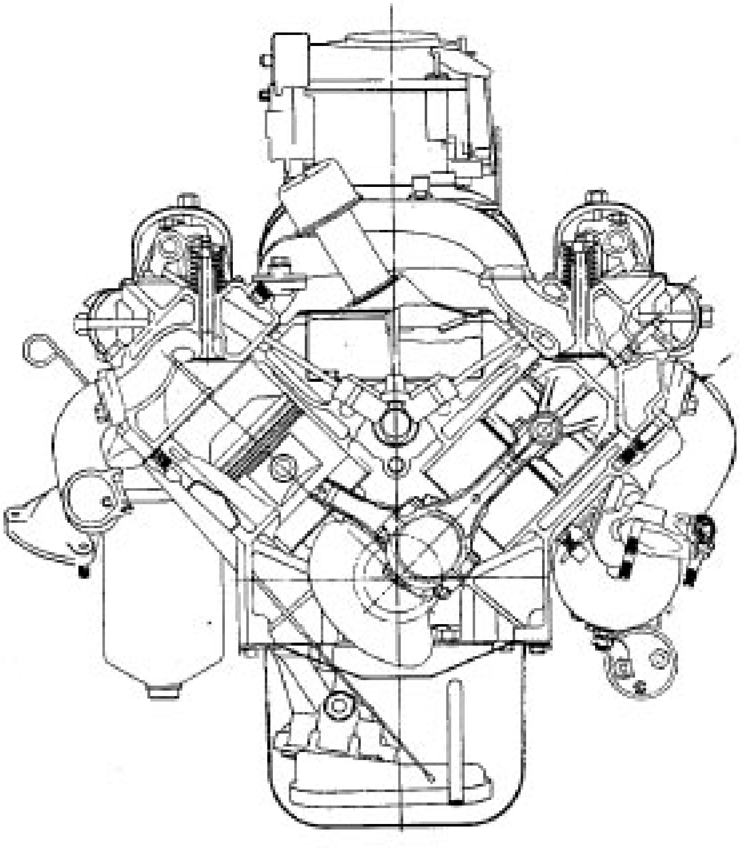

Fig. 22 – Transverse cross-section of Buick V-8

### Cooling System

A cast timing-chain cover is used, with the water-pump impeller housing and discharge passages recessed in the front face (Fig. 19). The water-pump cover forms the front half of the pump housing and carries the inlet pipe and the impeller bearings and seal, with one side of the inlet passage formed by the water-pump cover and the other by the timing-chain cover and the impeller disk. The water flows from the inlet side through six holes in the hub to the vane side of the impeller. This construction saves considerable space and opens up the passages so that these parts can be die cast in aluminum alloy. At present they are being cast in grey iron because of economic considerations.

Discharge passages from the water pump are provided to supply cooling water to each bank of the cylinder block. The standard Buick straight-through cooling system is used with connecting water holes at the rear of each cylinder head and cylinder block. A water manifold carrying the thermostat housing is bolted to the water outlet passages at the front of the cylinder heads. To provide a bypass connection, the water manifold is spigotted into a hole in the timing-chain cover leading to the pump inlet, with a rubber O-ring seal. The use of a separate water outlet manifold simplifies servicing as the intake manifold may be removed without breaking any water connections, and the heads may be removed without disturbing the water pump.

### Fuel Pump

The fuel pump is mounted on the timing-chain cover. This low position, in line with the fan blast, is helpful in avoiding vapor-lock troubles.

### Crankcase Ventilation

The combined crankcase ventilator inlet and oil filler cap is located in the front (Fig. 20), and the multiple pass outlet is located in the rear of the lifter cover. A transverse baffle separates the front and rear sections of the lifter compartment and directs the flow of air in the ventilating system.

### Carburetor and Manifold

A 2-barrel carburetor is provided in the 1953 Buick Series 50 line, and a 4-barrel carburetor for added power in the Series 70. Except for the carburetor flange the two manifolds are identical. The Buick inlet manifolds (Fig. 21) provide 90-deg T-shaped sections at all branching points. The individual exhaust porting is carried to the heat crossovers, providing direct hot spots at each of the 4 horizontal T-sections of the manifold, and extensions of these hot spots surround the carburetor riser section.

### Piston

Fig. 22 shows a transverse section through the engine. The piston, although much larger in diam-

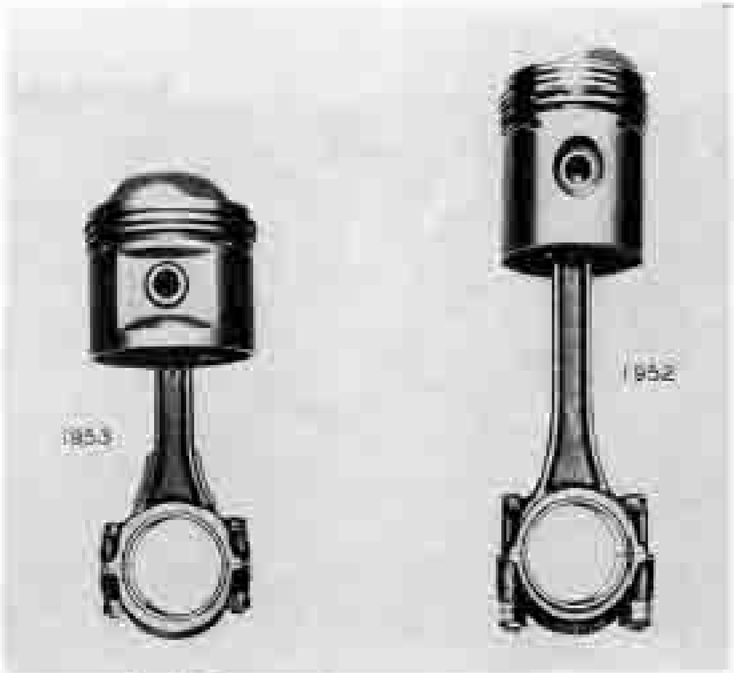


Fig. 23 – Piston and rod assembly comparison

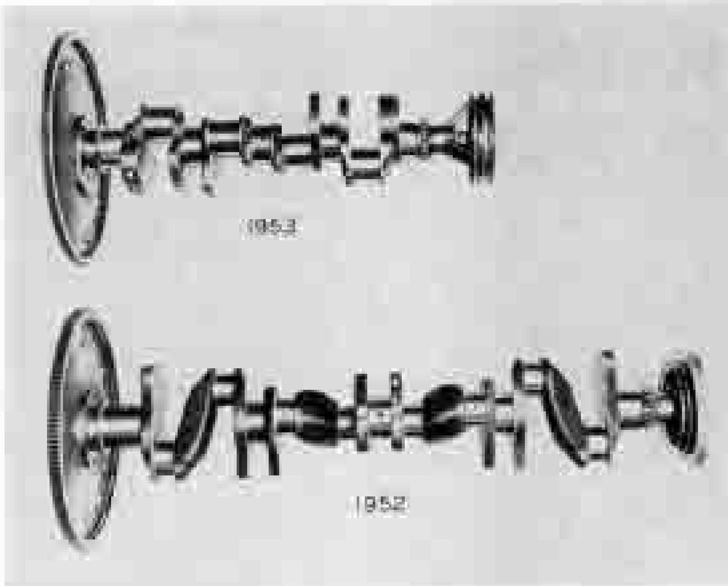


Fig. 24 – Crankshaft comparison

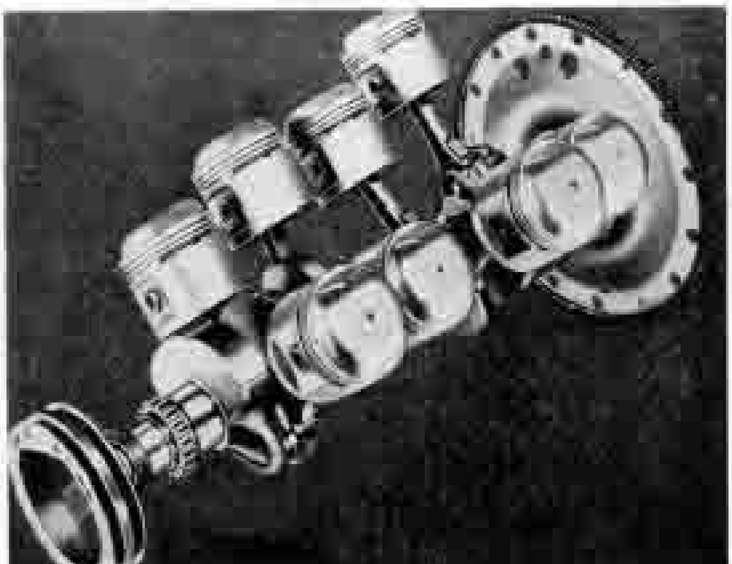

Fig. 25 – Power train assembly

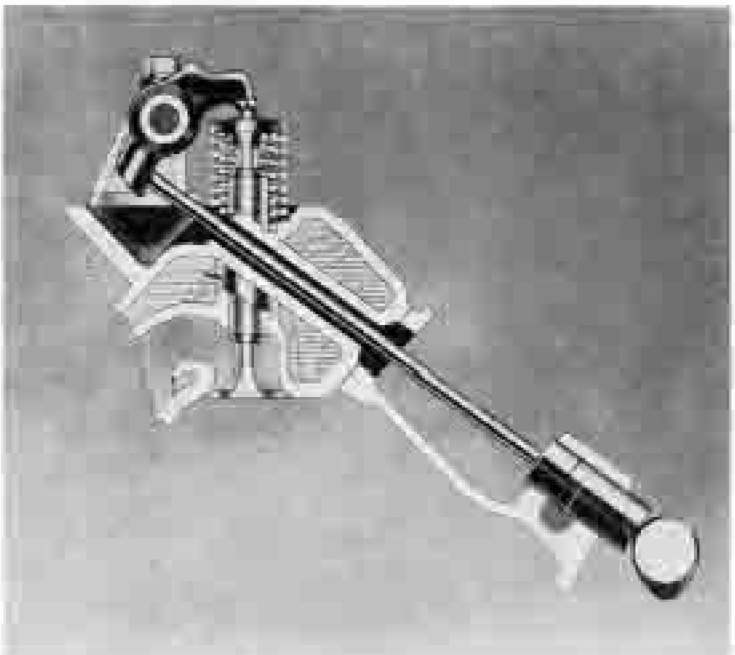

Fig. 26 – Valve-actuating mechanism

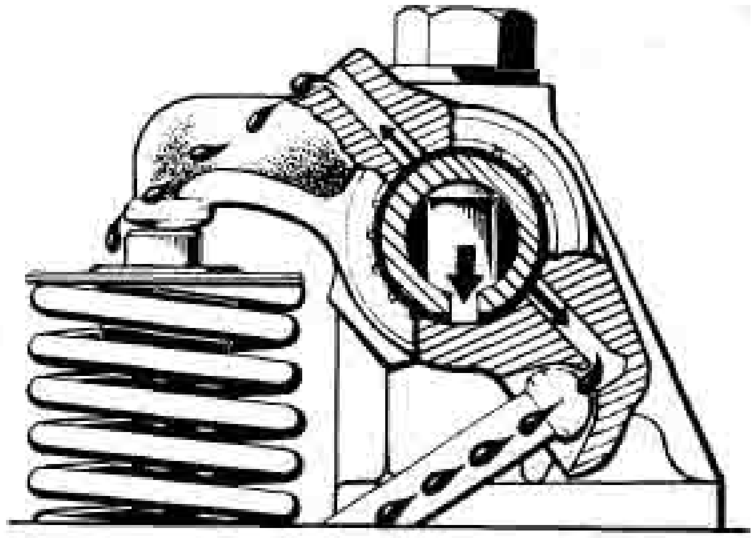

Fig. 28 – Rocker-arm lubrication

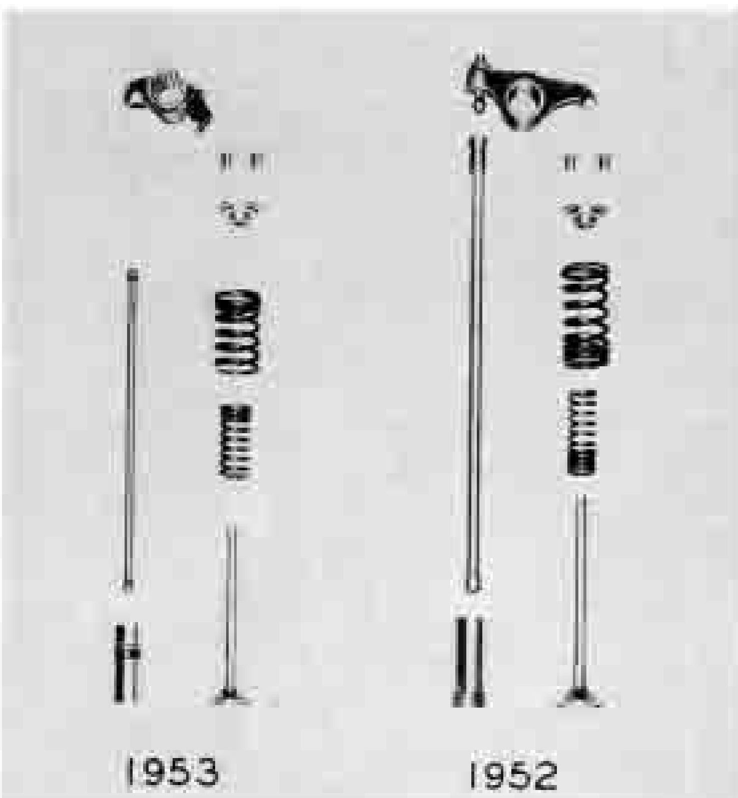


Fig. 27 – Valve train comparison

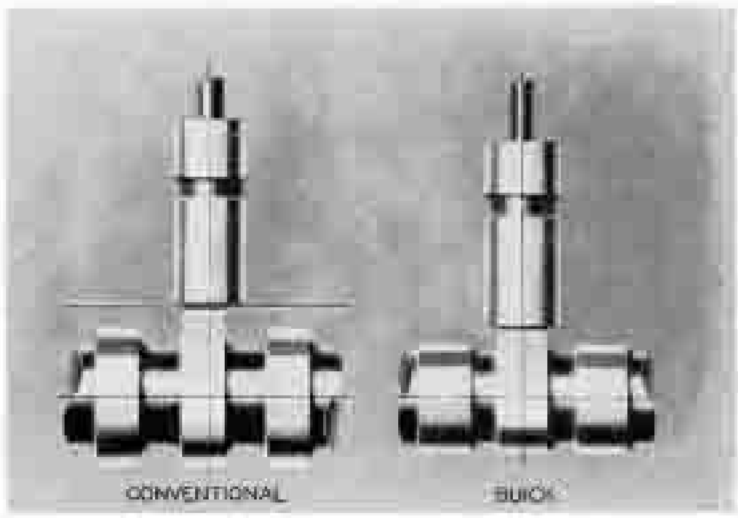


Fig. 29 – Vave lifter and camshaft alignment

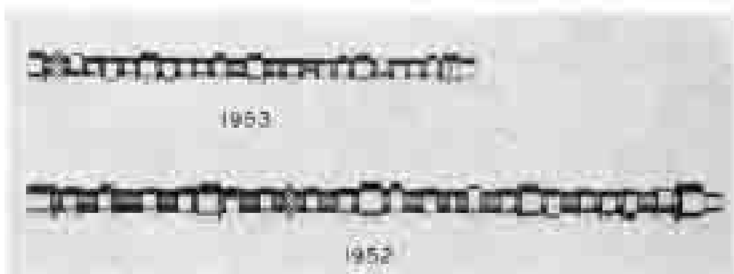


Fig. 30 – Camshaft comparison

eter than in previous models, follows time-tested Buick practice with 1-piece aluminum alloy construction and full skirt trans-slot design. Two conventional 5/64-in. compression rings and one 3/16-in. flexible steel oil ring are provided. The piston pin is not offset.

### Connecting Rod

The connecting rod is very short (Fig. 23), only 6 in. between centers, which in conjunction with the short stroke gives a rod length/stroke ratio of 1.875, a very conservative figure. The rods are brought to uniform weight by machining off the required amount of stock from lugs located near the center of gravity of the rod. Our computations show that commercial tolerances in balancing the connecting rods will be less detrimental to overall balance in the new engine than in the former in-line engine with its balance lugs at each end of the rod. The production Buick-method of clamping the piston pin in the connecting rod is retained, because we have found nothing superior in quietness, durability, and replacement cost.

### Crankshaft and Bearings

The crankshaft (Fig. 24) has five main bearings with the rear bearing flanged to carry the thrust load. The main and connecting-rod bearings are of the replaceable liner type of Durex 100A material as used in previous Buick engines. The exceptionally great bearing journal overlap, of over ¾ in., contributes to the outstanding rigidity of the crankshaft, which from our tests is greater than that of any other American automobile engine now in production. The natural frequency of the crankshaft coincides with the engine firing frequency at 5300 rpm, well above the maximum engine operating speed. The weight of the crankshaft is 56 lb which is lighter than that of any of the new engines of more than 300 cu in. displacement, and less than half the weight of our 1952 Roadmaster crankshaft.

Due to the exceptionally short stroke and short connecting rod the space left for counterweighting is severely limited (Fig. 25). The outer surface of the counterweights is cam-turned to maintain maximum useful counterweighting with a constant minimum clearance with the bottom of the piston skirt. The radius at the tips of the counterweights is limited by the clearance with the cylinder barrels, and this barrel clearance radius is blended into the piston clearance contour, with the final shape so calculated as to permit high-speed camturning in production. A small amount of counterweighting is carried in the crankshaft pulley and in the flywheel. This method of completing the counterweighting is very effective from a weight standpoint because of the long span from the pulley to the flywheel. It also makes possible a correction for any future piston weight changes without affecting the crankshaft counterweight tooling or balance.

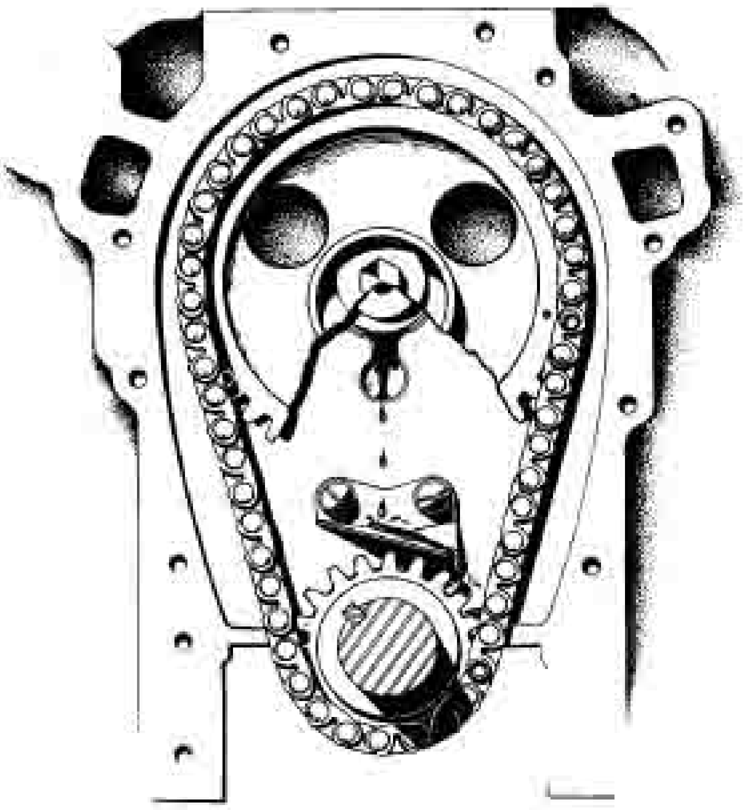

Fig 31 – Timing-chain lubrication

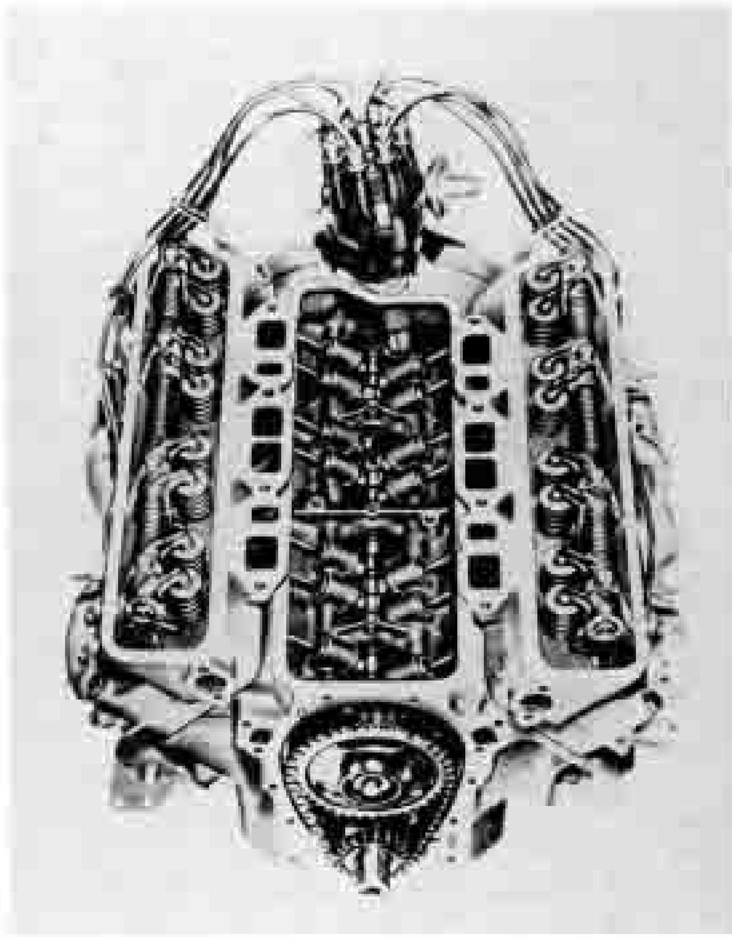

Fig. 32 – Top of Buick V-8 with manifold and covers off

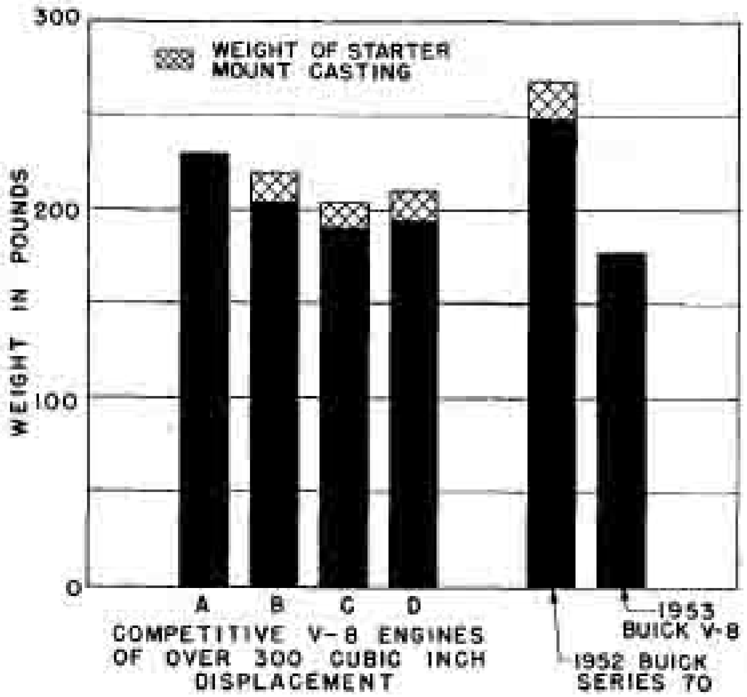


Fig. 33 – Crankcase weight comparison

### Valve Train

The arrangement of the valves, and valve-actuating mechanism (Fig. 26), is unique in the Buick 90-deg V-8.

The valves are arranged side by side at an angle of 45 deg to the cylinder axis, and therefore in a vertical plane with respect to the ground. The reversed position of the rocker arms with the centerline of the rocker shaft outside the line of valves, and with the push-rods crossing the valve stems, results in a very compact cylinder head. The design also makes possible a common horizontal gasket surface for the rocker-arm cover and for the intake manifold, which simplifies machining, and even more important, greatly reduces the sealing problem at these critical points. The valves are ⅞ in. shorter than the straight-8 valves (Fig. 27), and the dual valve springs are 7/16 in. shorter than past production, thereby reducing valve operating inertia as well as reducing material cost.

The pearlitic malleable iron rocker arms are very small and are of the nonadjustable type (Fig. 28). The rocker-arm bearings are broached with eight longitudinal grooves, which not only serve to control the amount of oil metered to the valve stems and push-rods, but have proved to be so effective in preventing rocker-shaft scoring that the antiscuff coating formerly used on the rocker arms is no longer necessary.

The push-rods are of ¼-in. diameter solid rod, upset to provide ⅜-in. spherical diameter ends to eliminate excessive wear at these critical points. The exceptionally short 8⅜-in. push-rod is made possible by the compact engine design.

Hydraulic valve lifters of the diesel equipment type are used. The bodies are cast iron with chilled wearing faces ground flat and ferrox-coated. The steel camshaft has cams 9/16-in. wide and ground without taper. No attempt is made to spin the lifters, although some of the lifters turn as a result of manufacturing alignment variations (Fig. 29). We have found consistently longer life with the lower unit load resulting from a wide parallel contact of the lifter face on the cam.

### Camshaft and Drive

The camshaft (Fig. 30) is carried in five bearings, and is unusually small and light. The timing-chain drive (Fig. 31) is lubricated with the overflow oil from the front camshaft bearing, directed by means of a small stamping to the lower crankshaft sprocket and the inside of the chain.

The fuel pump eccentric is a hardened and chrome-plated cup-stamping driven from the front of the camshaft sprocket.

The distributor and oil pump are driven by a gear at the rear of the camshaft in the orthodox manner (Fig. 32), except that the distributor and oil pump are located on the right side of the camshaft. This arrangement puts the distributor gear thrust upward and eliminates the need for an additional bearing in the crankcase to take the thrust load, and results in some weight and cost saving.

### Crankcase

Since the crankcase is the heaviest single part in the engine, it should offer the greatest opportunity for weight saving (Fig. 33). In the Buick design the starter mounting is carried on the flywheel housing, which is an integral part of the crankcase and eliminates the separate starter mounting casting used in other V-engine designs. The crankcase flange is far enough below the crank centerline to provide a flat, continuous oil-pan gasket surface with consequent sealing advantages. Contrary to accepted opinion this construction reduced weight. The dropped flanges (Fig. 34) not only stiffen the crankcase proper, but because of the greater vertical height of the attaching flanges, greatly increase the rigidity of the attachment of the engine to the transmission bell-housing.

The cylinder bores extend 1 5/16 in. below the water jackets. Attachment of the extended portions of the bores to the bulkheads is avoided by casting 1¾ in. by 3 in. windows in the intermediate bulkheads which effect a further weight saving. Substantial reinforcing ribs tie the water jackets into the main bearing bolt bosses on each side of the crankcase "windows." The center of each intermediate main bearing is reinforced vertically by a Z-shaped wall, which carries the main bearing oil passages and ties into the center oil gallery and camshaft bearing structure.

Five head bolts per cylinder (Fig. 35) are so placed as to carry the major part of the gas pressure loading on the cylinder head into the water jacket walls, rather than into the cylinder barrels.

The bolt spacing conforms to the structural characteristics of the head and block, with closer spacing across regions of greater deflection. The design has proved to be very effective in reducing gasket and cylinder distortion troubles.

A one-piece embossed steel gasket is used with a double bead around the combustion chambers and ample gasket area at all critical locations.

The main oil gallery, which supplies the main bearings and camshaft bearings, is a steel tube cast in the block. This construction was used primarily to open up the center section of the crankcase casting with consequent easing of strains during cooling in the foundry, and secondarily as a means of saving weight.

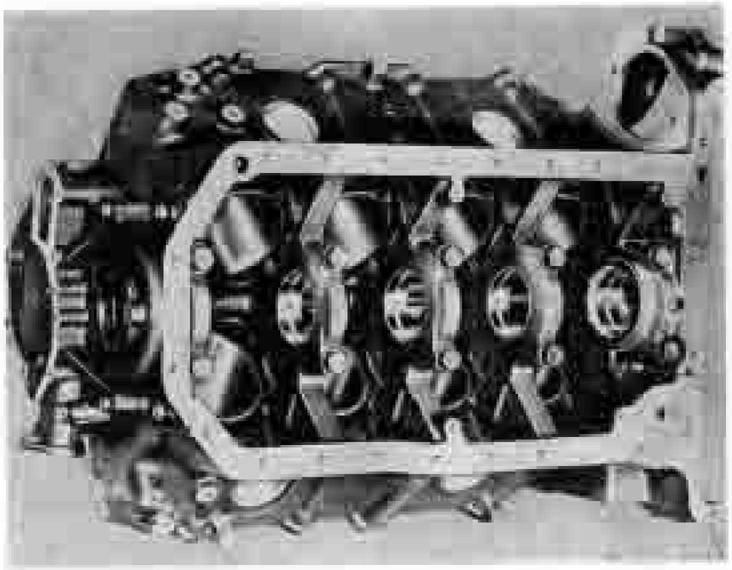

Fig. 34 – Bottom of crankcase

### Lubricating System

As automobile engines become more highly developed the demands on the lubricating system have become more exacting. Closer clearances, thinner babbit overlays, higher power outputs, and higher unit loads have all increased this problem. Hydraulic lifters, which have a pronounced sensitivity to dirt, air bubbles, varnish, and sludge, have not reduced the engine designer's difficulties, although the oil companies have done much to relieve the situation with improvements in oils and additives.

In designing the new Buick engine every effort was made to improve the lubricating system (Fig. 36). The oil pan is deep, with the oil screen near the bottom to provide the maximum depth of oil to minimize splashing and foaming, and to insure a constant supply of oil even during fast acceleration or turns. The fixed screen has an area of 19 sq in. to reduce the inlet flow velocity and the tendency to carry foreign material through the screen.

A horizontal baffle extends over the entire sump area and reduces aeration of the oil by preventing crankshaft oil fling-off from churning the sump oil, and oil in the sump from being thrown against the crankshaft. The oil pump is of standard Buick design and mounts on the crankcase flange rail, a position made possible by the dropped rail design. Oil is carried by drilled passages to a full-flow oil filter in which the filter element is mounted vertically, a feature which we feel certain will be greatly appreciated by anyone who has tried to change a filter element in one of the horizontal or angle mounted units.

From the filter the oil is delivered to the rear of the centrally located main oil gallery. This gallery supplies all the camshaft and main bearings, and from the grooved main bearings it feeds the connecting-rod bearings in the conventional manner.

Two other longitudinal oil galleries are used, drilled to intersect the lifter guide holes. These galleries are supplied with oil at reduced pressure through drilled passages registering with a metering groove in the front camshaft bearing. Drilled passages from the front end of these oil galleries lead to the rocker-arm assemblies.

The separation of functions of these oil galleries has several advantages. Pressure in the main gallery is not affected by lifter clearances or leakage, since the oil feed to the lifters is controlled by the camshaft groove. Oil velocities are reduced in the lifter galleries, since they feed only the lifters. The lower oil velocity and lower oil pressure aid in the elimination of air bubbles at the front end of the gallery before the oil reaches the lifters, the air being vented through the rocker-arm and shaft assemblies.

### Electrical System

The present trend toward higher volumetric efficiency and higher compression ratios has cre-

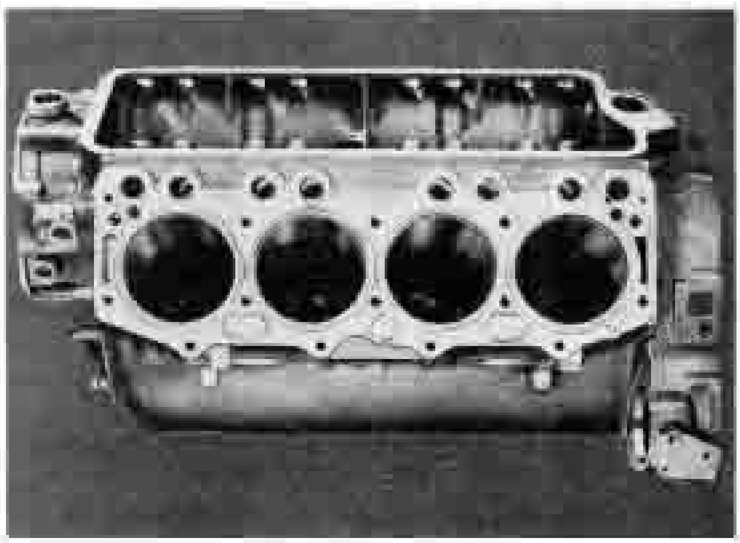

Fig. 35 – Cylinder-head gasket and crankcase

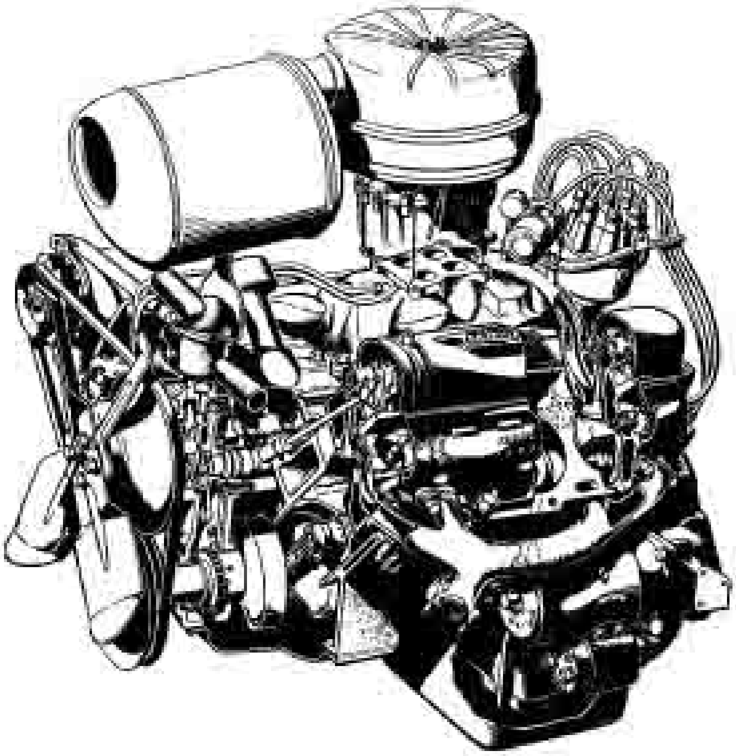

Fig. 36 – Engine lubrication

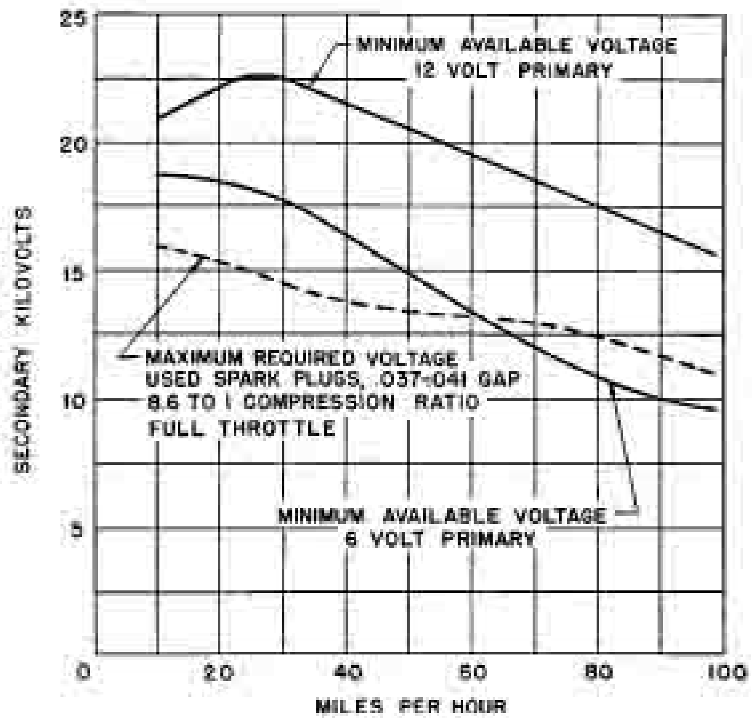


Fig. 37 – Secondary voltage comparison with full electrical load

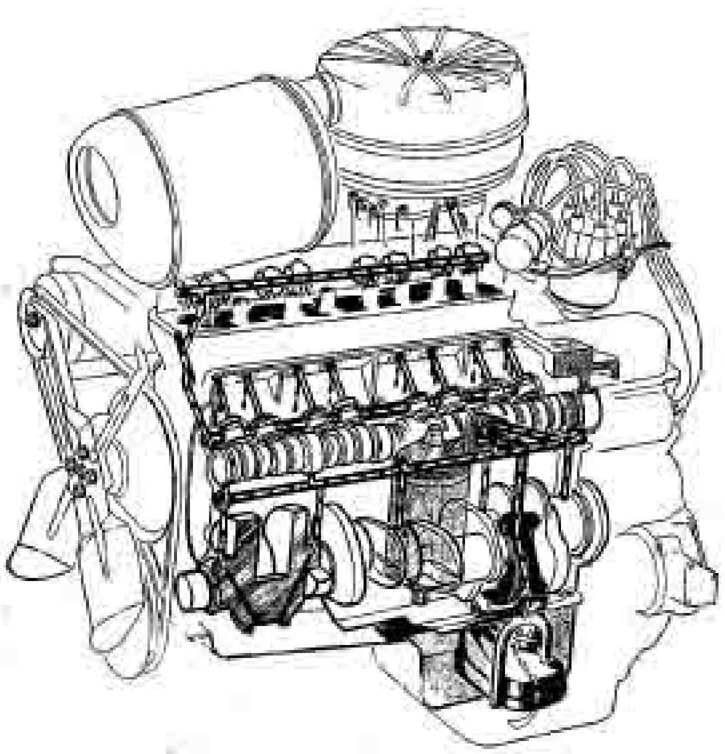

Fig. 38 – Sectional view of engine

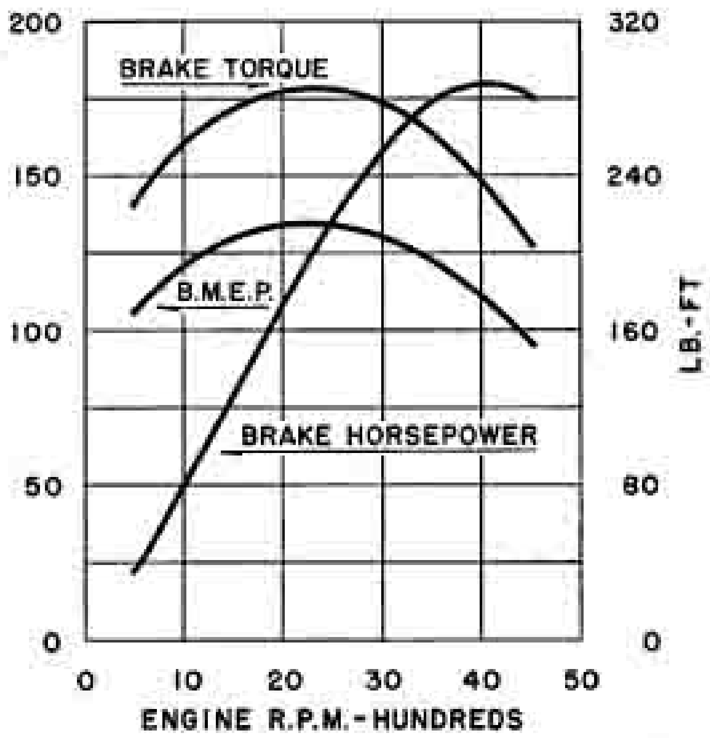


Fig. 39 – Full-throttle maximum power corrected to 100 F

ated a problem for the electrical engineers. The required ignition voltage (Fig. 37) has now become of such a magnitude that the secondary voltage obtainable with a 6-v primary system is closely approaching the minimum firing voltage of the spark plugs, even under optimum conditions.

Several attempts which have been made in an effort to increase the secondary voltage while retaining the 6-v primary system have met with at least a temporary success. However, in view of the ever-increasing secondary voltage requirements, Buick decided that the most satisfactory long range solution was the adoption of a system with a higher primary voltage, and has adopted the 12-v electrical system on the V-8 engine for 1953.

### Endurance

The Buick 90-deg V-engines (Fig. 38) have undergone 10,000 hr of dynamometer testing, and over 1,000,000 miles of road testing in cars running at the General Motors Proving Ground. The usual quota of corrections have been made as faults developed, but it is of interest that the second experimental engine, assembled in the fall of 1950, ran on full-throttle high-speed endurance at 4200-4500 rpm several hours longer than any of the notably rugged straight-8 Roadmaster engines have ever run in the same standard test. Road tests have been equally satisfactory and it is safe to say that in its final form the more powerful Buick 90-deg V-8, although 170 lb lighter, is equal to or better than the Series 70 straight-8 Buick engine in durability.

### Performance

Because of the current contest in advertised horsepower, we approach the subject of engine output with some misgivings. The problem of whether to try to out-exaggerate the field or to quote actual figures, which may not seem sufficiently high to some, is a difficult choice. We have tested four of the new competitive V-8 engines in our engine test department according to the General Motors test code. We have access to the results obtained on these and other makes of engines, which have been tested in other divisions of General Motors, at General Motors Research, and at the General Motors Technical Center. None of the tests on 1952 engines have shown higher bmep figures, hp ratings, or better specific fuel curves than we have obtained with our new engines under the same test conditions.

We believe the purpose of this paper is best served by quoting actual figures obtained under test No. 7 of the General Motors test code, which is run without muffler, and is corrected to 100 F carburetor air temperature (Fig. 39).

The 60 F correction factor commonly used for advertising purposes will increase the hp to 188, the maximum torque to 298 lb-ft and the bmep to 139 psi.

A gain in performance, which is often overlooked, is due to the lower engine weight, and to the lower inertia of the moving parts.

A gain in fuel economy results from the reduced engine friction, and to the higher compression ratio of the 1953 Buick V-8. Also, it should be remembered that the resulting increased efficiency is evidenced in improved performance as well as better fuel economy.

### Conclusions

After we have been in production for a year we will be in a better position to state just how well we have accomplished all our aims and objectives in the design of the new Buick V-8. At the present time we believe we can safely draw the following conclusions about the new engine:

1. The Buick V-8 is as durable as our 1952 line of straight-8's, with sufficient strength and rigidity to permit increasing compression ratios and power output as advances in fuels and combustion-chamber design permit.
2. From our tests the fuel economy improvement will be up to 8%.
3. Our service department states that the Buick V-8 is the easiest of the new engines to service.
4. Our production department estimates that the production cost of the new engine will be 7% less than for the Roadmaster in-line engine. One-half of this saving is due to weight reduction.
5. To the best of our knowledge the Buick V-8 is the lightest and most compact automobile engine for its power output now in quantity production in America.

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2. *SAE Journal* (Transactions), Vol. 50, June, 1942, pp. 213-220; 252: "Air Flow Through Intake Valves," by G. B. Wood, Jr., D. U. Hunter, E. S. Taylor, and C. F. Taylor.
3. SAE Transactions, Vol. 6, April, 1952, pp. 290-303: "Economic Relationship of Engine-Fuel Research," by C. L. McCuen.
4. SAE *Quarterly* Transactions, Vol. 1, October, 1947, pp. 669-679: "More Efficient Utilization of Fuels," by C. F. Kettering.


## APPENDIX I

### Buick Weight Comparison

| General Motors Uniform Parts Classification | | Series 70 1952 | 1953 | Difference |
|---|---|---|---|---|
| 6A1 | Cylinder block and crankcase | 248.67 | 176.52 | −72.15 |
| 6A2 | Cylinder head | 93.34 | 105.99 | +12.65 |

| | | | | |
|---|---|---|---|---|
| 6B | Flywheel housing | 20.40 | 1.03 | —19.37 |
| 6C1 | Crankshaft | 114.21 | 55.91 | —58.30 |
| 6C2 | Crankshaft balancer | 13.68 | 0.00 | —13.68 |
| 6C3 | Flywheel | 12.05 | 9.44 | — 2.61 |
| 6D | Connecting rods, pistons, and rings | 30.74 | 28.59 | — 2.15 |
| 6E | Oil pan | 13.60 | 9.36 | — 4.24 |
| 6F1 | Oil pump and drive | 5.82 | 5.15 | — 0.67 |
| 6G1 | Oil distribution system | 0.12 | 0.13 | + 0.01 |
| 6G2 | Oil filler | 0.26 | 0.50 | + 0.24 |
| 6G4 | Oil gage rod | 0.11 | 0.12 | + 0.01 |
| 6G5 | Oil filter | 5.09 | 5.20 | + 0.11 |
| 6H | Engine ventilating system | 3.13 | 1.75 | — 1.38 |
| 6J | Engine front covers | 6.89 | 12.02 | + 5.13 |
| 6K1 | Fan and drive | 4.67 | 9.52 | + 4.85 |
| 6K2 | Water pump and drive | 13.26 | 7.66 | — 5.60 |
| 6K3 | Thermostat and engine cooling parts | 4.15 | 6.16 | + 2.01 |
| 6L | Intake and exhaust manifolds, and heat controls | 46.40 | 50.61 | + 4.21 |
| 6M1 | Carburetor | 9.81 | 10.43 | + 0.62 |
| 6M3 | Air cleaner and silencer | 9.89 | 13.11 | + 3.22 |
| 6M4 | Fuel and vacuum pump | 5.94 | 5.88 | — 0.06 |
| 6Q | Powerplant mountings | 4.65 | 4.55 | — 0.10 |
| 6X1 | Camshaft and drive, valve springs, and lifters | 48.54 | 31.47 | —17.07 |
| 6X3 | Valve rocker arms, shafts, and covers | 16.95 | 13.75 | — 3.20 |
| 6Y1 | Generator | 24.85 | 22.24 | — 2.61 |
| 6Y2 | Starting motor and control | 25.83 | 23.92 | — 1.91 |
| 6Y3 | Distributor | 6.13 | 6.43 | + 0.30 |
| 6Y4 | Spark plugs, ignition coil, and wires | 4.93 | 6.49 | + 1.56 |
| | Total engine dry | 794.11 | 623.93 | —170.18 |

## APPENDIX II

### 1953 Buick V-8 General Specifications

| | | |
|---|---|---|
| Bore, in. | | 4.0 |
| Stroke, in. | | 3.2 |
| Displacement, cu in. | | 322 |
| Numbering System, Front to Rear | | |
| Left Bank | | 2-4-6-8 |
| Right Bank | | 1-3-5-7 |
| Firing Order | | 1-2-7-8-4-5-6-3 |
| Taxable Horsepower | | 51.2 |
| Compression Ratio | | 8.5/1 |
| Maximum Brake Horsepower, Corrected to 100 F | | 180 at 4000 rpm |
| Maximum Brake Torque, Corrected to 100 F, ft-lb | | 285 at 2200 rpm |
| **Piston, Rings, and Pin** | | |
| Piston Material | | Aluminum alloy |
| Features of Piston | | Cam ground, full skirt, transverse slot, anodized |
| Number of Compression Rings | | 2 |
| Width of Compression Ring | | 0.078 |
| Number of Oil Rings | | 1 |
| Width of Oil Ring | | 0.187 |
| Piston-Pin Diameter and Length | | 0.94 x 3.40 |
| Piston-Pin Locking Method | | Clamped in rod |
| **Crankshaft** | | |
| Material | | Forged steel |
| Weight, lb | | 55.91 |
| Bearing Taking Thrust Load | | No. 5 |
| Number of Main Bearings | | 5 |
| Main Bearing Journal Diameter and Effective Length, in. | | |
| No. 1 | | 2.4985 x 1.220 |
| No. 2, 3, and 4 | | 2.4985 x 1.250 |
| No. 5 | | 2.4985 x 1.765 |
| Crankshaft Main Bearing Material | | Durex 100A |
| Crankpin Journal Diameter, in. | | 2.2495 |
| **Connecting Rod** | | |
| Length, Center-to-Center, in. | | 6.00 |
| Connecting-Rod Bearing Material | | Durex 100A |
| Effective Connecting-Rod Bearing Length | | 0.881 |
| **Camshaft** | | |
| Material | | Forged steel |
| Type of Drive | | Chain |
| Number of Bearings | | 5 |
| **Valves and Operating Mechanism** | | |
| Type of Lifters | | Hydraulic |
| Rocker-Arm Ratio | | 1.5/1 |
| Valve-Seat Angle, deg | | 45 |
| Spring Pressure, lb | | |
| Valve Closed | Outer | 37.5–42.5 |
| | Inner | 19.5–24.5 |
| Valve Open | Outer | 85–91 |
| | Inner | 53–59 |
| Valve lift, in. | Inlet | 0.378 |
| | Exhaust | 0.350 |
| Valve Head Diameter | Inlet | 1.750 |
| | Exhaust | 1.250 |
| Valve Stem Diameter | Inlet | 0.3720 |
| | Exhaust | 0.3714 |
| Valve Timing | | |
| Inlet Opens, deg btc | | 25 |
| Inlet Closes, deg abc | | 77 |
| Exhaust Opens, deg bbc | | 70 |
| Exhaust Closes, deg atc | | 42 |
| Timing Point | | Valve 0.004 in. off seat |
| **Lubrication** | | |
| Type of Lubrication | | Full pressure |
| Oil Pump Type | | Gear |
| Oil pressure, Maximum, psi | | 40 |
| Type of Oil Intake | | Stationary |
| Oil Filter Type | | Full flow |
| Capacity of Crankcase, Less Filter, qt | | 6 |

# DISCUSSION

## Development of New V-8 Combustion Chamber Reviewed

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Research Laboratories Division, GMC

THE authors have presented an excellent introduction to the features of the new Buick engine. We are glad to discuss our part in the development of the combustion chamber for this engine.

Fig. 15 shows design details of the head and piston. As pointed out by the authors, this design has an intake and exhaust valve at one side of the combustion chamber in a plane 45 deg to the normal. This provides an easy entrance into the combustion chamber and results in high volumetric efficiency. In addition, the flat 45 deg portion of the piston comes within 0.080 in. of the surface of the combustion chamber, resulting in turbulence during the compression stroke. Another important feature of this combustion-chamber configuration is the location of the spark plug. As the authors have pointed out, it is within less than ⅛ in. of the centerline of the cylinder bore. This obviously is a location which results in minimum possible flame travel length.

In order to better appreciate the various features which are incorporated in the design of the new combustion chamber, it might be of interest to review some of the laboratory's earlier work. For many years, this laboratory has directed much of its effort toward improving the efficiency of internal-combustion engines. This work and, of course, the work of a great many others in both the automotive and petroleum industries, has resulted in a continuous improvement in power and economy over the past half century. In 1947 C. F. Kettering pointed the way to further progress in engine efficiency by demonstrating that compression ratios as high as 12.5/1 are practical in automotive engines if sufficient consideration is given to design of the engine structure. Mr. Kettering also indicated that there are at least two ways of achieving higher compression ratios: (1) By means of chemical octane numbers built into the fuel at the refinery by the petroleum technologists. (2) By means of mechanical octane numbers built into the engine by the engine designers.

The petroleum industry has an enviable record of progress in improving the antiknock quality of gasoline over the years, and we have no reason to believe that they will not continue this trend as long as they find it economical to do so.

There are many factors which contribute to an engine's mechanical octane rating or its ability to operate more efficiently on fuel of a given octane level. They include spark advance, carburetion, valve timing, volumetric efficiency, and combustion-chamber design. These factors are all interrelated and have a bearing on the selection of the compression ratio at which the engine will operate. The relation between these factors in the final engine represents the best compromise the designer can make, in view of the wide range of operating requirements which the automotive engine must meet. Of these many factors which influence the compression ratio at which the engine operates, we found that the combustion chamber offered a very promising field for development.

Since the octane requirement of an engine is affected by the weather, in addition to all these other factors, the first step in the evaluation of various combustion-chamber designs, in terms of octane requirements, was to develop a procedure which would allow accurate reproduction of engine operating conditions over a long period of time. Fig. A shows some of the apparatus which was used to control the variables which influence octane requirements. As an example, the humidity, pressure, and temperature, were held to definite values so that variations in the air entering the carburetor would not influence the tests. Such mechanical variables as valve timing, cam design, valve sizes, and compression ratio were also held at definite values.

It was decided to maintain a constant compression ratio of 9/1 for this combustion-chamber development for two reasons:

1. To simplify evaluation of the various designs.
2. Our past experience in this type of work has indicated that we should be able to achieve 9/1 compression ratio with present commercial gasolines.

In addition, the air/fuel ratio was adjusted for maximum knock for each fuel used during the tests. Standard reference blends of isooctane and normal heptane were used to determine the octane requirement of the various combustion chambers. However, since engines are not operated in the field on reference fuels, ratings were also obtained using commercial regular and premium grade gasolines. The present rating of the regular grade gasoline used for the Buick combustion-chamber development is 86 octane research and 80 octane motor, while premium grade now has a rating of 92 research and 82 motor.

Finally, to eliminate the deposit variable, all the designs were rated clean, even though it is a recognized fact that commercial engines operated in the field always have a certain amount of combustion-chamber deposits present. Of course, this factor must be taken into account in selecting a compression ratio for the production engine and, naturally, this has been done.

It should be quite apparent that all this attention to detail demonstrates the importance of taking every possible precaution to make sure that, insofar as is possible, combustion-chamber design was the only variable.

An example of a typical fuel rating test of a combustion chamber is shown by Fig. B. The shape of the chamber is shown by the scale cross-sections. At any given speed, the procedure consists of first obtaining a spark fishhook. This is a plot of power as expressed in terms of indicated mean effective pressure versus spark advance. The spark fishhook is obtained using a fuel which will permit the engine to develop its maximum imep without knock which, of course, indicates the maximum possible power output of the design under detonation-free conditions. Next, the borderline-knock spark advance values of primary reference blends and commercial gasolines are obtained. As an illustration, 80-octane reference fuel has a border-line spark advance of 5 deg and the premium grade gasoline used in these tests has a borderline spark advance of 13 deg. It can be observed that these values are superimposed upon the spark fishhook. When combined in the manner shown by Fig. B, these data can be interpreted by the observer as he desires. As an example, it can be seen that it requires about 92.5-octane fuel for this chamber to operate detonation free at 99% of maximum power. It can also be seen that premium gasoline will permit operation at almost 99% maximum power. Another conclusion that can be drawn from this chart is that regular gasoline, as rated by this combustion chamber, is equivalent to approximately 86-octane fuel.

Since octane requirement depends upon power output when other factors are held constant, it is obviously very important to obtain power measurements in addition to

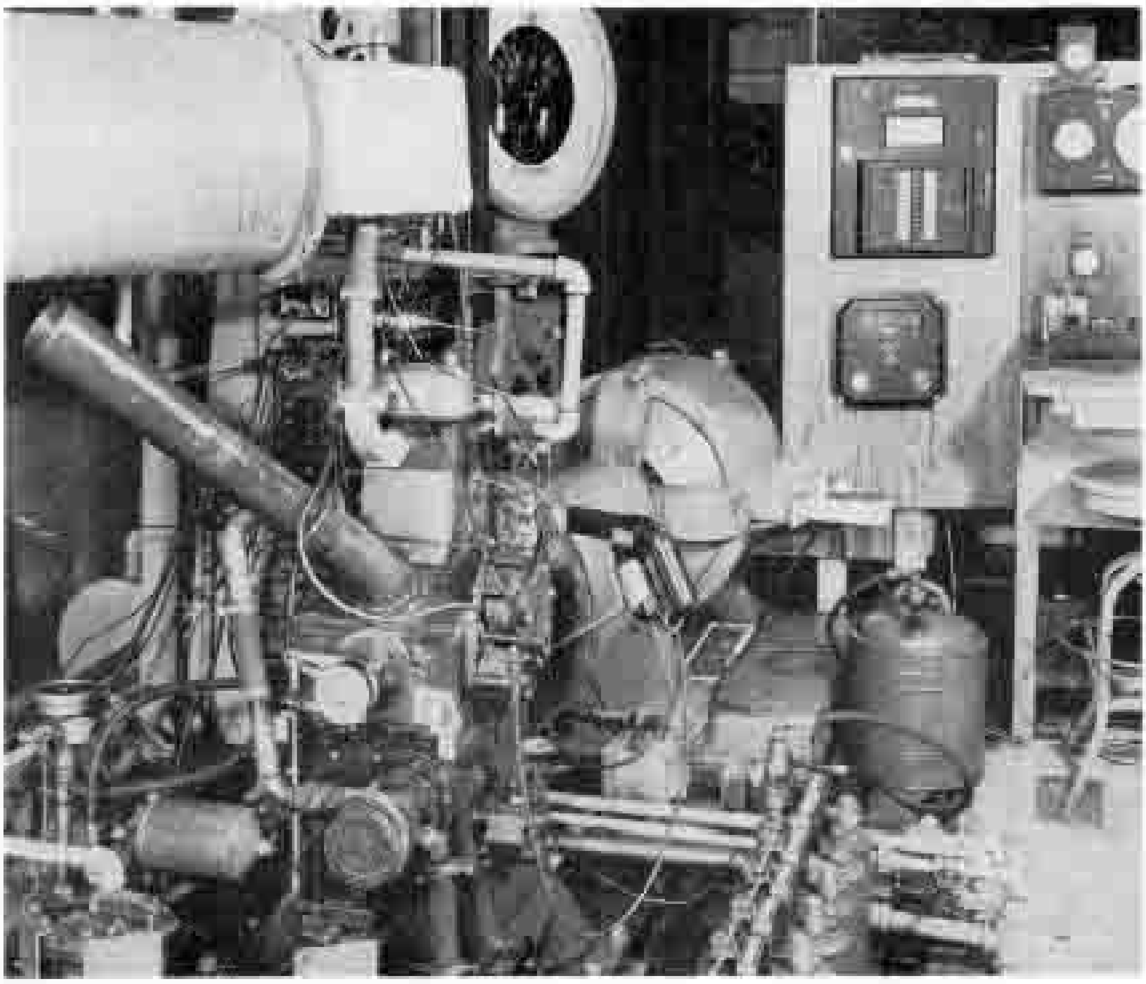

Fig. A – Apparatus used to control variables which influence octane requirements

the fuel ratings. We think this type of presentation is extremely valuable since it presents important factors such as gasoline rating, octane rating, and power measurement on one sheet, where they can be conveniently examined.

Similar tests for each chamber design evaluated were performed and the results are summarized as shown by Fig. C. We refer to Fig. C as an octane tree. In this case, it was made up by taking the octane rating for all designs at 99% of maximum power, 1000 rpm, 9/1 compression ratio, and arranging them along a vertical octane number scale. In addition to the octane number arranged in the vertical direction, the research ratings of regular and premium gasolines are also labeled. As can be observed, these ratings are 86 octane for regular and 92 for premium fuel.

The various combustion chambers are identified by the scale cross-sections of each design. As an example, design A shows cross-sections in two planes through the combustion chamber. As has already been mentioned, the valves are located in the 45 deg plane. The spark plug is located in the center of the engine cylinder. It can be seen that almost all changes in combustion-chamber shape for this particular investigation were made by simply changing the configuration of the pistons. It is apparent that this Buick head design with central spark plug location lends itself very well to a study of this nature. It also possesses the added advantage that future increases in compression ration can be achieved by a relatively simple piston change.

More detailed examination of this figure shows that chamber A, the original design for the Buick engine, requires 96½-octane fuel to operate at 99% output; while chamber J, the design selected for the production engine, requires only 88½-octane fuel. This reduction in octane requirement of eight numbers is a fine example of a mechanical octane improvement. Other designs can be selected and examined in a similar manner. As an example, design F, a dome-shaped combustion chamber that was introduced by Buick back in 1934 shows an octane requirement of 95½. This, as was pointed out by Mr. Turlay in his presentation, was the best that Buick had available at that time, but was abandoned in 1938 in favor of the turbulator piston which is shown as design G with a requirement of 93 octane. This, of course, allowed the compression ratio to be increased while still using the same fuel.

These two early designs were revived and modified for testing at 9/1 compression ratio to see how they would fit into the overall picture.

One of the questions which may come to mind at this point is: what are the factors which the designer must build into a combustion chamber in order to obtain mechanical octanes?

Our past experience had indicated that if we could shorten the combustion process the charge would complete burning before detonation had time to occur. Therefore, at the start of this development we deliberately set out to speed up the rate of burn, by designing chambers with high turbulence and the shortest possible flame travel.

Referring back to Fig. C and comparing designs A and J; it can be observed that chamber J has a much shorter flame travel and, because of its more effective piston coverage, has much greater compressive turbulence than design A.

The fact that shorter combustion time is accompanied by a decrease in octane requirement can be demonstrated by comparing the spark-advance values shown by Fig. D. This chart compares the test results obtained for the original design (A) and the final production design (J). It can be seen that the spark advance value at 99% maximum power for chamber A is 20 deg and for design J it is 13 deg. This represents a decrease in burn time of at least 7 deg.

As an additional illustration of the effectiveness of short flame travel in reducing octane requirements, Fig. E shows,

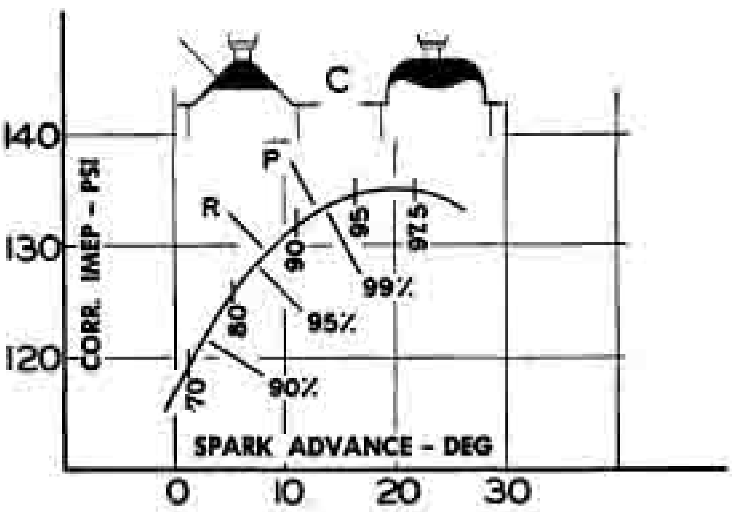


Fig. B – Fuel rating test of combustion chamber – 1000 rpm and 9/1 compression ratio

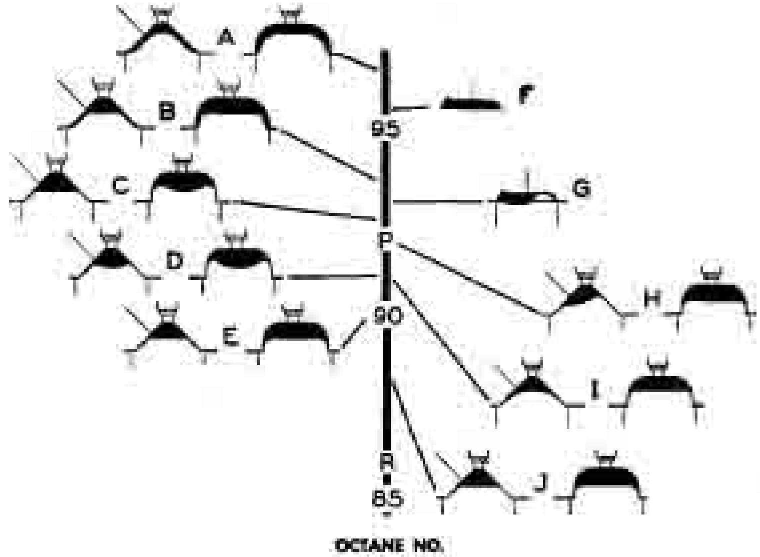


Fig. C – Single-cylinder octane requirements at 99% maximum power, 1000 rpm, and 9/1 compression ratio

the effects of two different spark-plug positions in the same chamber. By comparing the octane requirements at 99% maximum power, which is approximately 92 octane for the central plug position and about 98.5 octane for the point of ignition on the exhaust valve side of the head, it is apparent that Buick's central plug position alone accounts for a 6.5 octane improvement. Further examination shows that at 95% maximum power, which is more in line with the usual production spark settings selected by most manufacturers, Buick's central plug location reduces the octane requirement from 94 to 79 octane, an improvement of 15 numbers in this range of the octane scale. In addition, it can be observed by comparing the 99% maximum power spark advance values that there is a decrease in combustion time of at least 14 deg for the central plug position. That is, from 24 deg to 10 deg.

Another interesting question which may be asked in connection with combustion chamber design is, what effect does it have on power output and thermal efficiency?

As shown by Fig. F, we have compared the imep and indicated thermal efficiency values obtained for the combustion chambers included in the Buick development with the same valve and bore dimensions. We consider the differences shown to be nothing more than random variations within the limits of error of measurement. Therefore, we can conclude that, within the scope of the Buick program, combustion-chamber shape has shown no significant effect on power or thermal efficiency.

In addition to the Buick development program, we have carried on an extensive investigation of present commercial combustion chambers as well as a great many experimental chambers for overhead-valve engines. Our tests have shown no significant difference in thermal efficiency between any of them. However, many have exhibited a wide spread in mechanical octane characteristics, a feature which can be exploited to improve efficiency through the medium of increased compression ratio.

In summarizing this discussion, we have shown that the

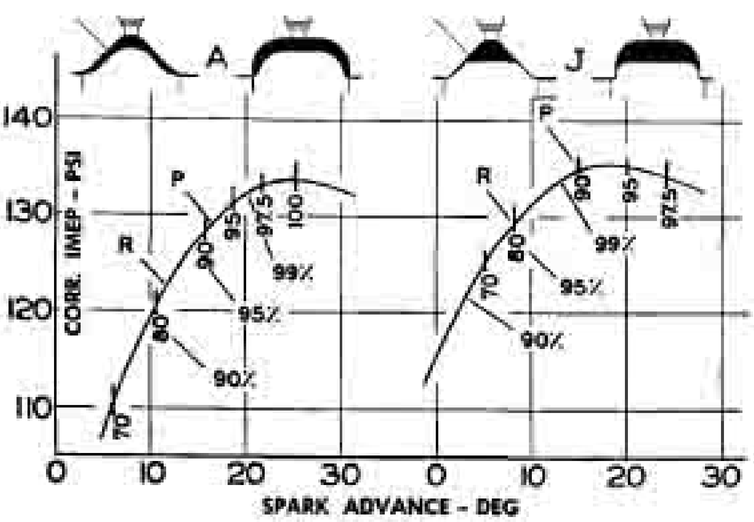


Fig. D – Comparison of test results for original design (A) and final production design (J)

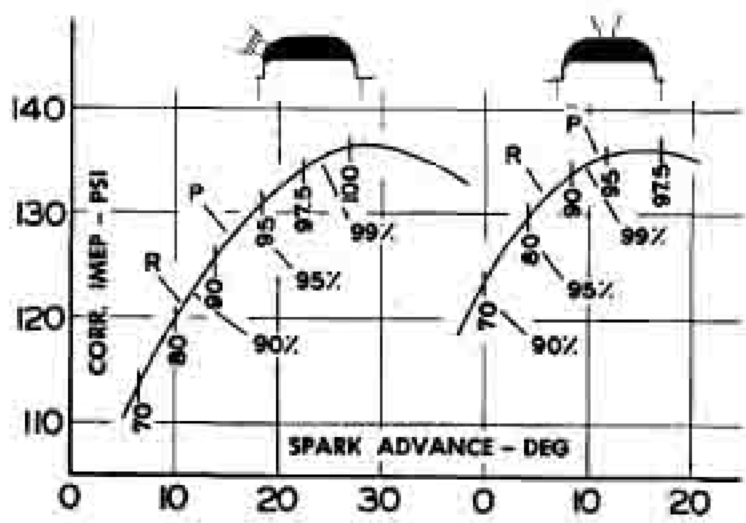


Fig. E – Effects of two different spark-plug positions in same chamber

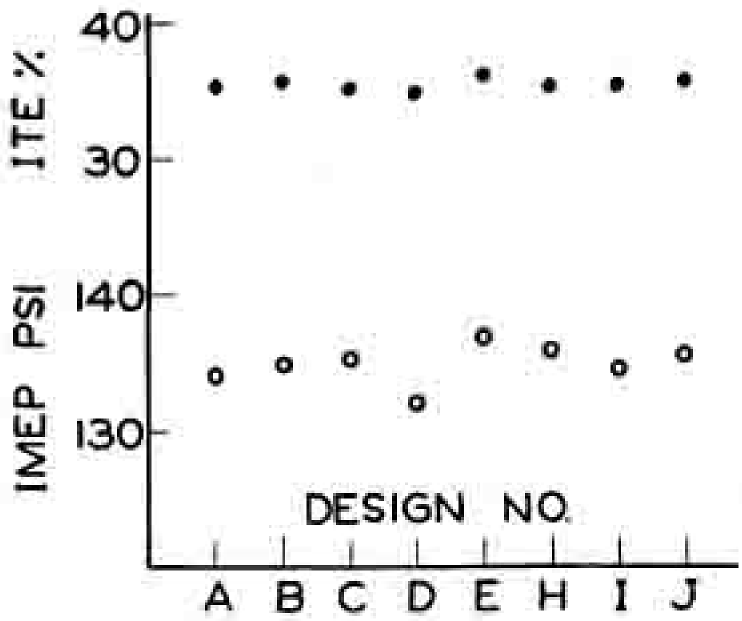


Fig. F – Effect of combustion-chamber shape upon combustion efficiency

combustion chamber for the new engine is the result of a systematic mechanical octane development program. A very realistic gage of the extent to which this program has paid off is demonstrated by the ability of the production engines to operate at 8.5 compression ratio on premium gasoline.

## New Engine Elicits Pertinent Questions

– J. R. MacGregor
California Research Corp.

THE authors have performed a herculean task in describing the numerous and intriguing design changes incorporated in the new Buick engine. Not being an engine designer and not having had any experience with this particular engine, it would be presumptuous for me to discuss any of the mechanical details from the design standpoint. However, based on experience with other engines there are several questions and comments that may be pertinent.

The presentation of the background for the valve-timing diagram makes no mention of the modest change in the timing of the events. Is this change simply a byproduct of the modification in the lift diagram, or is it an indication of a desire to fit more nearly the requirements imposed by the hydraulic type of transmission rather than the syncromesh?

The combustion-chamber design is interesting, to say the least. It would seem that the necessity of keeping deposits from the small clearance areas would be very great if interference is to be avoided and quenching retained. Has experience shown that the turbulence is adequate to scour these areas or is it anticipated that other means will be required for periodically removing carbon deposits?

Extensive experience in the removal of deposits from the combustion chambers has shown that not only can the octane requirement of an engine be materially reduced but also an incidental and almost equally important return of lost power can be obtained. The magnitude of this power recovery is dependent upon the amount and type of deposits and the design of the combustion chamber. The power recovery has varied from practically nothing to approximately 16 per cent. A cursory examination of the new Buick combustion chamber suggests that power loss through the laying down of deposits may be rather great. Has any information been obtained on this subject that is available for release?

In view of the influence that breather gases have on the deposits found on the outside of the carburetor and within the induction system it would seem that trouble may be experienced from this cause during those extensive periods of idling sometimes encountered when thermal forces cause the gases to move in the opposite direction from those shown by the arrows in the diagrams. This item will bear close watching.

With reference to the drive for the distributor it would appear that the arrangement which causes friction to oppose gravity might introduce a serious fluttering of the ignition with its attendant undesirable effect on octane requirement. Has some precaution been taken to avoid any uncontrolled vertical play of the distributor shaft?

The lubrication requirements of modern engines have been generally becoming more exacting and it would appear that those of the new Buick engine may even be critical. The large diameter piston with a crown design that suggests high heat transfer both by radiation and conduction; the short stroke which tends to permit more oil throw-off to reach the piston and its crown; and the short water jackets all seem to indicate high crankcase oil temperatures. In addition, the crossover of the exhaust pipe would appear to contribute to the elevation of oil temperature. It would be appreciated if advance information can be given as to the lubricating oil temperatures that are to be expected, particularly during extended high-speed driving under desert conditions.

The manner used to present the power data is very interesting and we shall look forward with great enthusiasm to the acquisition of one of these cars for chassis dynamometer and road test purposes. This will permit the establishment of our own views on power and some of the other operational features that so importantly influence the motorist in his evaluation of both the engine and the petroleum products he uses.

## Fuels and Lubricants Sought for New Designs

– D. P. Barnard
Standard Oil Co. (Ind.)

DISCUSSION of this paper is difficult for an oil man, particularly as this oil man has had no experience in the design or the development of mechanical devices – including any kind of an engine. These remarks endeavor to record those first impressions which the paper makes on such a representative of the petroleum industry.

First, adherence to overhead valves for 50 years. This is most reassuring to the layman as it dramatizes the extreme reluctance with which changes have always been contemplated in a perfect product. It not only helps to know that the familiar overhead valves are still with us, but it adds to our feeling of security to realize that, throughout the years since Louis Chevrolet laid down the first Model 10, the valves have remained virtuously upright. The cylinders may lean to the right or left, but the valves – no. (I hope someone can point out the significance of this observation over and above its relation to the installation problem.)

The new design aims at the best possible utilization of the improvements which have been made in fuels and lubricants. By taking advantage of all of the know-how currently available, this utilization will be far ahead of anything that could be accomplished by modification of the earlier design. We recognize other advantages (which are so clearly pointed out in the paper), but from our side of the table we are particularly interested in what the new job will do with and to our products.

The compression ratio of 8.5 fills us at once with admiration and concern: admiration for the strides in engine development which justify placing such a compression ratio in the hands of the general public; concern over whether the petroleum products which our industry makes – and will be able to make during the foreseeable future – will be entirely satisfactory. To us, the $64 question is "how is it done?"

The fuel manufacturer is especially interested in the effects of deposit allowances in holding the allowable compression ratio down, and the rather high maximum torque speed in permitting such a high compression ratio on commercial premium fuels. Our misgivings do not reflect defeat but merely recognize troubles as an inevitable part of the price of progress. Every increase in compression ratio brings up pictures of more combustion-chamber deposit problems. Of course, we realize that the modern engine at its compression ratio is less susceptible than its ancestors of a generation or more ago. Nevertheless, even the most modern gasoline engine design is, as was once pointed out by Bill James, wanting in ash-handling capacity. This will always worry us, since there is no practical way of achieving present octane number levels – and increasing octane numbers still further – without at the same time accepting necessity for ash handling. And not all combustion-chamber deposit difficulties are chargeable to the fuel or even to the lubricating oil. Some of this material is just plain adventitious dirt.

Recognizing that we will never be completely free from combustion-chamber deposits, what to do about them is likely to remain a major problem. The new designs with which we are already familiar have contributed much toward minimizing their effects, and we hope that the new Buick design will extend this know-how. In the meantime, we think that we have been able to help some and, just as the engine builder keeps seeking to improve his designs, we in the oil industry well keep trying to improve our products.

The quest for higher octane numbers has raised levels to the place where the going is getting difficult. Current premium grade octane numbers imply a marked reduction in the number of hydrocarbons available to the fuel manufacturer. Each octane number increase is reflected in the necessity for more complicated procedures, higher costs, and the inevitable processing losses than was the case when levels were appreciably lower. Of course, a 1-unit increase in the range above 90 is worth more than a similar increase in, say, the 70's, but the costs per octane number go up much more rapidly, too, and the inevitable implications must not be disregarded. Lest some of us become too impatient to achieve the magic level of 100 let it be noted that the effective knock-rating improvement in going from present average levels to 100 is at least as great as the sum of all octane improvements made up to the present.

We in the petroleum industry are most appreciative of the frank, accurate, and complete presentations in this and other papers describing the new engine designs. We may not fully understand the many considerations underlying these new jobs, but at least we can welcome the clearly competent efforts to base them on the best information available in order to deliver the most performance and service, with the greatest of reliability, and within the practical limits of economics. In the meantime, we in the oil industry should gladly invoke all of the facilities available to us in our efforts to provide these new designs, together with their ancestors, with fuels and lubricants.

## Cylinder Performance in New Engine Discussed

– S. D. Heron
Consulting Engineer

THIS paper is of decided interest from the standpoint of cylinder performance. The Ethyl Corp. has for several years studied cylinder behavior in respect to knock and preignition-limited performance and also in regard to fuel economy and air utilization. The knock-limited behavior of the new Buick engine is obviously very good since it uses a compression ratio of 8.5/1 with premium fuel, and does so with excellent volumetric efficiency. In appraising knock-limited performance, volumetric efficiency must be considered as well as compression ratio. In this connection Socony-Vacuum laboratories showed, several years ago, that a passenger car of about 7/1 compression ratio would, at road load, operate without knock on zero octane number fuel up to about 60 mph.

The combustion-chamber design of the new engine is of distinct interest to the student of cylinder performance. This chamber enables a very short stroke to be used while maintaining a compact shape of the charge at the point of ignition. With a high compression ratio and a short stroke it is difficult, with some combustion-chamber layouts, to avoid layers of charge which are difficult to ignite satisfactorily at light load.

The new chamber permits the use of a very considerable quench area without interference with the flow path of the gas leaving the intake valve. With overhead valves, a large quench area often involves considerable interference with the gas leaving the intake valve. The authors appear to believe that the combustion benefits of quench area, as they have used it, are a result of turbulence rather than due to quench.

The term, quench area, is very generally used in this country in contrast to the term, squish, which is generally used in Europe. It is really unimportant which term is used since either term is descriptive and generally understood. The term quench does, however, imply that the combustion effect is known. Some two years ago J. B. Macauley initiated a study to determine if quench area produced beneficial effects as a result of quench or as a result of turbulence. One phase of the study came up with a quite clear answer to the effect that the benefits were due to turbulence. Another phase, conducted under conditions where quench area produced marked improvement in knock-limited performance, gave no clues as to whether quench, turbulence, or any other cause, was responsible for the improvement.

The Ethyl Corp. studies of cylinder performance are, it is hoped, without prejudice since we do not design combustion chambers except for experimental studies. The types designed have mostly been rather bad although this, in part, has been due to trying to cover too wide a range. Attempting to cover wide ranges of compression ratio and turbulence in one cylinder head often produces a result analagous to trying to produce a musical instrument which will give the combined effects of a bassoon and a violin.

These studies of knock-limited performance of necessity are based upon somewhat limited data and further data might revise some of the present conclusions which are rather destructive of some currently accepted ideas. For instance, it appears that the knock-limited performance of very small cylinders is no better than that of those of 6-in. bore. Also, it appears that current large bore aircooled aircraft engines are just as good at 3000 rpm as are very small water-cooled cylinders at the same speed. The studies have covered a fuel range from zero to above 100 octane number and while academic in many respects do lead to the conclusion that one is liable to be the victim of one's preconceived notions.

## Blowby Characteristics Of Engine Requested

– T. A. Scherger
Studebaker Corp.

THE authors have covered their subject so concisely and thoroughly that it is quite difficult to make any constructive additions to the material presented. In general the engine appears to be of rugged, stiff construction, with ample displacement to perform the assigned task. With the short stroke and high overlap journals the crankshaft should be very stiff and should contribute toward smooth engine operation. In mentioning the balance it was stated that a small amount of counterweighting is carried in the crankshaft pulley and in the flywheel. This brings up the question as to the method of balancing. Is the crankshaft balanced in production with the flywheel and crankshaft pulley assembled? If this is the case, what provision is made for servicing the crankshaft pulley in case it is damaged in the field? Doesn't this type of balancing give somewhat higher bearing loads on the front and rear bearings or do the authors feel that with the high overlap of the bearing journals this is not a factor to be considered.

I note that the compression ratio of this engine is 8.5/1. No mention was made of the spark advance. I am wondering whether the spark advance as used is for maximum torque at relatively low speeds or whether some sacrifice in full-throttle power was made for octane numbers, thereby obtaining the benefit of high compression ratio for part-throttle fuel economy.

The ratio of sealing volume per cubic inch of piston displacement is unusually high for this engine. No mention of blowby was made. If the data is available, what are the blowby characteristics of this engine?

## New Engine Designs Reflected In Future Gasoline Picture

– A. J. Blackwood
Standard Oil Development Co.

I AM pleased to have this occasion to discuss the paper on the Buick engine, for although I have been in the midst of a petroleum technology for many years, I spent my first 18 months out of college doing engine research and development work with the Mack truck people. Consequently, I found the paper particularly interesting. For an engineer who has been away from the details of engine development work for over 25 years, many of the problems discussed still have a familiar and pleasant ring. From my present position in the petroleum product research and development field, it is certainly gratifying to see some of the familiar problems being so effectively solved.

I would no longer presume to be an automotive engineer capable of offering sound comments on this paper insofar as design details are concerned. I shall, therefore, leave such comments to others who are far better qualified. I would, however, like to offer a few thoughts on three or four points directly related to the paper, and then to offer some general comments on the impact which engines such as this may have on the future gasoline picture.

I was interested in the effort made to have all combustion-chamber volumes exactly the same, and to maintain a uniform induction system velocity from carburetor to exit side of the inlet valve. This should result in uniform filling – an important consideration involved in making all cylinders have the same antiknock requirement.

Mounting the fuel pump on the timing-chain cover and in the direct fan blast seems a step in the right direction to minimize vapor lock. It would be even better to have it in a direct stream of outside air which would be some 10 to 15 deg cooler than radiator discharge air.

I was interested in the approximation of about 100 miles of road test per hour of laboratory test to bring the development to a successful conclusion. In a current research project in our own laboratories, we have been averaging about 25 miles on the road for each laboratory engine test hour. Our work, however, is not seeking to establish any mechanical endurance factors.

The anticipated gain in fuel economy of 8 per cent is most praiseworthy. Part is attributed to a reduction in engine friction. We feel this friction effect on fuel economy has been overlooked in the past. In England, operators have been reluctant to use anything but heavy crankcase oils. Three or four years ago, one of Esso's technical men interested a couple of fleet operators in trying out a light oil of the 5W type. One of the benefits was a marked improvement in gasoline mileage. Now the light oil idea is spreading very rapidly in England.

We were pleased to note the special attention devoted to getting a good lubricating system on this new engine. Other features which we hope some day to find in a lubrication system are an instrument panel oil level gage; an oil pump system which compensates for wear and hot oil so that it maintains pressure; and better seals and gaskets so the garage floor stays reasonably clean.

And now I should like to discuss the fuel situation. We have made many studies of the economics of compression ratio and gasoline quality. Also, we have always been concerned with economic balance between processing losses of crude oil and added mileage to be gained by higher octane fuels and higher compression ratio engines and have recently made a further study of this point. In 1925, a 60-octane gasoline was typical, while by 1952 the octane increase over the years had amounted to about 30 points. During this time both the volume demand and percentage required from crude increased. This increase was obtained by thermal and catalytic cracking of higher boiling fractions. The processing losses from cracking are chargeable to the required increase in volume. Beyond this, however, there have been processing losses chargeable to octane increase alone. These result from applying thermal reforming and catalytic reforming to a small fraction of the lowest gasoline octane constituent, namely heavy naphtha.

On the basis of the requirement in 1952 of 39 per cent gasoline, 34 per cent distillate, 14 per cent residual fuel, and 17 per cent lubricating oil, asphalt, wax, and fuel for refinery processing, and, in the process of making a leaded pool gasoline of 90 research octane number, the losses due to reforming amount to only 2.4 per cent on the total gasoline produced.

This however is not the important point. Based on the increased efficiency resulting from the higher compression ratios which could be used with the 90 research octane number gasoline available in 1952 as compared to the limited compression ratio to operate on the 60-octane gasoline of 1925, it was calculated that about 67 per cent more work could be obtained from a barrel of gasoline of the higher octane number. Adjusting for the 2.4 per cent gasoline loss, and if we take full credit for the entire octane number increase from 60 to 90, the net gain is about 65 per cent. In other words, 2 gal today gives roughly the mileage of 3 gal in 1925 as a result of a better quality gasoline and engine performance. It seems clear to us that over the past 30 years the efforts to increase octane number and the higher compression ratios which such fuels permit, have been constructive from the standpoint of conservation of petroleum resources.

Now what about the future? Another comparison was made between the 1952 assumed 90 research octane number leaded pool and a future pool of 96 research octane number. In this second study, production of products other than gasoline was held at 1952 levels. Also, in this case catalytic reforming replaced thermal reforming and more of the low octane virgin naphtha was catalytically reformed in the 96-octane number case. The final comparison is on a slightly different basis than in the first study. It was determined that with high compression ratio engines designed for 96 research octane number, it would be necessary to process 98 barrels of crude to get the same amount of work as is obtained from 100 barrels of crude processed for 90 research octane number gasoline. In other words, less crude would be necessary, less gasoline would be made, but the work obtained from the higher compression ratio engines would be the same. There is still room in the future for improved gasoline quality and engine adjustment that will take advantage thereof.

In my closing remarks I would like to observe that both the engine manufacturers and the petroleum industry have done an excellent job of giving the public more economical and better performing engines over the years and that trend will no doubt continue. However, it should be recognized that any improvements in economy which can be gained by better so-called mechanical octanes, through better piston design, better valves, better induction systems, better cooling systems, and so forth, provide improved economy even at the current octane number levels. On the other hand, since there is a 40-million pool of cars already on the road, and because the petroleum industry cannot make octane increases to satisfy specific cars which have higher octane requirements, the economic advantage is going to be slower if we rely entirely on chemical octane improvement.

## Seeks Additional Data On Fuel Requirements

– C. J. Livingstone
Gulf Oil Corp.

AMONG the many features of the new Buick V-8 engine, I am impressed by the authors' report of their evaluation of the engine. I am sure the severity of their testing has been adequate to predict the endurance of the engine when it is delivered to the customer. However, as one who will be concerned with its fuel requirement later on, I am disappointed that the authors did not see fit to present more data on this phase. I am sure there are a good many oil people who are interested in the fuel requirements of the engine as installed with the new transmission in the automobile.

Our experience with other high-compression ratio engines indicates that the most critical fuel requirements occur when the car is operated with extremely light-load factor. In my own case, I have been driving one of the newer V-8s with 8/1 compression ratio. It has wonderful performance and I experienced practically no detonation or autoignition with present premium fuel. A few months ago it became necessary for my wife to take over the driving of this car for a period of 90 days. Before the three months was up the engine was detonating on premium fuel and was in trouble from autoignition. Consequently, I am more interested in fuel requirement data when the car is driven as my wife would drive it, than when it is put through the vigorous paces by the test engineers.

When an engine gets into trouble from detonation with premium fuels we can correct the trouble by pulling back the spark even though we may have to sacrifice performance, but when we encounter autoignition there seems to be no solution at present. Consequently, I believe the oil people would be interested in any data in fuel requirements after deposit accumulation at light load, that the authors might be able to present. We would also be interested in learning how the new engine transmission combination rates fuel in respect to the research octane number. If the new engine at 8.5/1 compression ratio can be operated without objectionable knock with present-day premium fuels, and is free from autoignition after light-load deposit accumulation, Buick has done a remarkable job in the development of this engine.

## Fuel and Combustion Reactions Considered

– E. F. Miller
Socony-Vacuum Oil Co., Inc.

THE authors have centered their paper largely around mechanical design and construction, and while we are interested in many of these features, we will comment only on those items which relate to fuel requirements and combustion.

From a fuel requirement standpoint the outstanding feature of the new design is the combustion chamber. We are glad to see the amount of effort being devoted to combustion-chamber shape, on the part of the various engine manufacturers.

We recognize that turbulence and spark-plug location and quench areas influence the combustion process. The design in this case must produce considerable turbulence.

Do the authors attribute the ability to achieve 8.5/1 compression ratio on current premium gasoline largely to the combustion-chamber design or is some of this increase a result of spark advance compromises? Any quantitative indication would be of considerable interest.

Some of our experience shows that turbulence exerts some control on the formation of combustion-chamber deposits and we wish to ask whether, in the course of their development work, the authors developed information of this type.

Because of the various design influences, any observations on our part as to how well this design is satisfied by premium grade gasoline must await our experience with the cars in the hands of customers.

By now I believe most of us are familiar with the fact that cars of a given make and model usually differ in octane number requirement by 15 to 20 units as a result of driving habits, and mechanical, maintenance, and deposit variations.

The importance of this 15- to 20-octane-number variation in requirement can be appreciated when it is compared with the difference in quality between housebrand and premium gasolines which ranges from 5 to 10 octane numbers at the present time.

The machining of combustion chambers we assume is an attempt to minimize one of these mechanical variations. Any reduction in the variation of requirements would constitute a distinct step forward.

While the paper is not entirely clear to us in its reference to the extensions of the hot spots to the carburetor riser, we assume this was done for the purpose of controlling carburetor icing.

While we believe the application of heat is the most effective means of controlling icing, we recognize that the amount of heat to be applied is a difficult choice for the designer. Too high a temperature in the region of the carburetor may produce objectionable deposits.

Another factor of the design related to combustion that interested us was the stroke/bore ratio of 0.8. From our studies of this design factor, we have learned that the improvement in engine efficiency of some of the new engine designs results from the compactness of the combustion chamber in reducing heat losses.

While we have no reason to believe that going under one in the stroke/bore ratio by a modest amount hurts engine efficiency, we became worried, however, when the authors said, ". . . space requirements, as dictated by the body stylist, are one of the most important, if not the most important factor, in determining the future trend of engine design."

In a second place the authors say, "the short stroke – big bore adopted in all the new V-8 engines has many real advantages, but the stroke/bore ratio of the new Buick engine was chosen chiefly because from our investigation it produced the minimum external engine size."

We hesitate to hold the authors rigidly to these statements but we had not realized that style controls such fundamental engine design items as stroke/bore ratio.

We long since have learned in this country that engineers are required to make design compromises in the interest of making a product more attractive and therefore more salable to the public, but we hope there is a limit beyond which we as engineers will not go, for fundamentally, in this instance, we are selling transportation.

As a further question pertaining to the control styling exerts, we wish to ask whether underhood space limitations are becoming such as to restrict ventilation, thereby producing high temperatures with their adverse effect on the vapor-locking tendency of the fuel system.

## Requests Additional Data On Fuel and Performance

– A. C. Sampietro
Willys-Overland Motors, Inc.

I WONDER whether the authors would be willing to supply performance data in addition to that given in Fig. 39. Specific fuel consumption and heat-to-coolant data for both a clean engine and for a dirty engine, with the spark retarded to avoid knock, would be of pronounced interest.

## Describes Fuels and Effects Of Carbon Accumulation

– Henry W. Boylan
Buick Motor Division, GMC

I WILL try to sum up all the questions that have been asked regarding the Buick combustion chamber, the fuels required for this chamber and the effect of carbon accumulation, in a general statement covering the results of our tests and a comparison with our past in-line engine.

We feel from our tests that the Buick V-8 engine will operate as satisfactorily on premium fuel as our past model in-line engine. On our carbon accumulation schedule for wide-open throttle operation, which is the equivalent to 24 hr on power runs from 600 rpm to 4500 rpm, we take a torque loss of 2½% from the clean condition at maximum torque. At peak horsepower we take a loss in torque of 3½% from the clean condition.

On our carbon-accumulation schedule which consists of 1 min at 600 rpm closed throttle, and 3 min at 1500 rpm open throttle for a 24-hr period, we have a loss in torque of 4½% at 1000 rpm and 5% at 2000 rpm. This is comparable to the loss on the in-line engine which amounts to 4½% at 2000 rpm.

Our road tests are made at the General Motors Proving Ground on their octane rating schedule. This schedule consists of entirely low-speed, part-throttle operations. Using this schedule and primary reference fuel for determining requirements, this engine requires 88.5 octane fuel for the engine clean; and a 91.5 octane fuel for the engine not clean.

During the operation of the car on the slow-speed carbon-accumulation schedule, the engine is regularly checked for autoignition. Our engines showed no sign of autoignition during this period. When rating the engine not clean for the various fuels, a slight amount of autoignition was experienced at 1800 rpm and 2000 rpm while rating Red Crown 86-80 and primary reference fuels with an octane rating of 82.5 and 85.0.

At the end of the carbon-accumulation schedule we had trace knock on the Uniontown method at 1800 and 2000 rpm – the spark timing 3 deg btc. Compression ratio will have an average buildup for the engine of ½ ratio.

The question has been asked as to how much power loss we take at the low end due to retarded spark. We take approximately a 5% loss at 600 rpm, with this loss diminishing to 1% at 2200 rpm. From this speed, and on out, the spark is set mbt (minimum for best torque). This power loss taken at the bottom end of the power curve is no different than the loss taken on most of the new engines on the market today and is particularly inconsequential in the Buick car where a Dynaflow transmission is used. The Dynaflow does not allow the engine to operate at wide-open throttle much below 1700 rpm.

In the case of the synchromesh-transmission equipped cars, it is necessary to reduce the compression ratio to 8/1 and use premium fuel to obtain satisfactory operation from a detonation standpoint. With Buick's new-type pistons and combustion-chamber design, it is necessary to maintain the design dimensions of the quench or squish area even when reducing compression ratios. For this reason, in changing from a 8½/1 ratio to a 8/1 ratio it is necessary to install new pistons rather than just increase the gasket thickness.

Mr. Scherger asked a question on blowby about which I did not quite understand. I can say this – the rate of blowby in cubic feet per minute on the new V-8 engine is not any more, and averages slightly less, than our past in-line engine with a comparable displacement.

## Authors' Closure To Discussion

**In reply to Mr. MacGregor** – The distributor used in the new engine is of the same general construction as used in past production models, in which the thrust is also upward. The lower distributor bushing is designed to take the thrust load. We believe this construction results in less possibility for variations in end clearance than with the same of the other designs in which the thrust is downward and is carried on a bearing in the crankcase.

**In reply to Mr. MacGregor and Dr. Barnard** – Buick has used a high-speed timing for some time, as was stated in the paper. Such a timing, compared to more conventional timing, gains at high speed and loses at low speed. The low-speed octane requirements are reduced which permits a small increase in compression ratio, but not enough to gain back all the low end loss. Of course, the compression increase gives an additional gain at high speed.

In working out the timing for the V-8 advantage was taken of the fact that with the Dynaflow transmission low-speed torque is not as important as with other transmission types. Some changes in the straight-8 timing events were also necessary because of the different requirements caused by engine design differences such as valve position and manifolding arrangement.

**In reply to Mr. Scherger** – The crankshaft, pulley, and flywheel are not balanced as an assembly. The crankshaft and the pulley are balanced separately, as in the past production engines. The engine is balanced at final assembly as a unit, the flywheel being drilled to correct any out-of-balance condition. These operations are the same as used for past production engines and should cause no new service problems.

**In reply to Mr. Miller** – The extensions of the hot spots are intended to reduce icing troubles. Other V-engine manifolds have a hot spot under the vertical riser section, but none at the horizontal branches. We chose to provide hot spots at the T-section branches, and provided some additional heat to the riser section with the hot-spot extensions.