Number 479

THE NEW PACKARD V-8 ENGINE

by

W. E. SCHWIEDER

Packard Division

Studebaker-Packard Corp.

Detroit, Michigan

For presentation at the

SAE GOLDEN ANNIVERSARY PASSENGER CAR, BODY, AND MATERIALS MEETING

The Sheraton-Cadillac Hotel
Detroit, Michigan
March 1-3, 1955

Written discussion of this paper will be accepted at SAE Headquarters until April 15, 1955. Three double-spaced copies would be appreciated.

PREPRINT: Subject to revision. Permission to publish this paper, in full or in part, after its presentation and with credit to the author and the Society may be obtained upon request. The Society is not responsible for statements or opinions advanced in papers or discussions at its Meetings.

SOCIETY of AUTOMOTIVE ENGINEERS, Inc., 29 West 39th Street, New York 18, N. Y.

THE NEW PACKARD V-8 ENGINE

by
W. E. SCHWIEDER
Packard Division
Studebaker-Packard Corporation
Detroit, Michigan

For presentation at the
SAE GOLDEN ANNIVERSARY PASSENGER CAR, BODY AND MATERIALS MEETING
The Sheraton-Cadillac Hotel
Detroit, Michigan, March 1-3, 1955

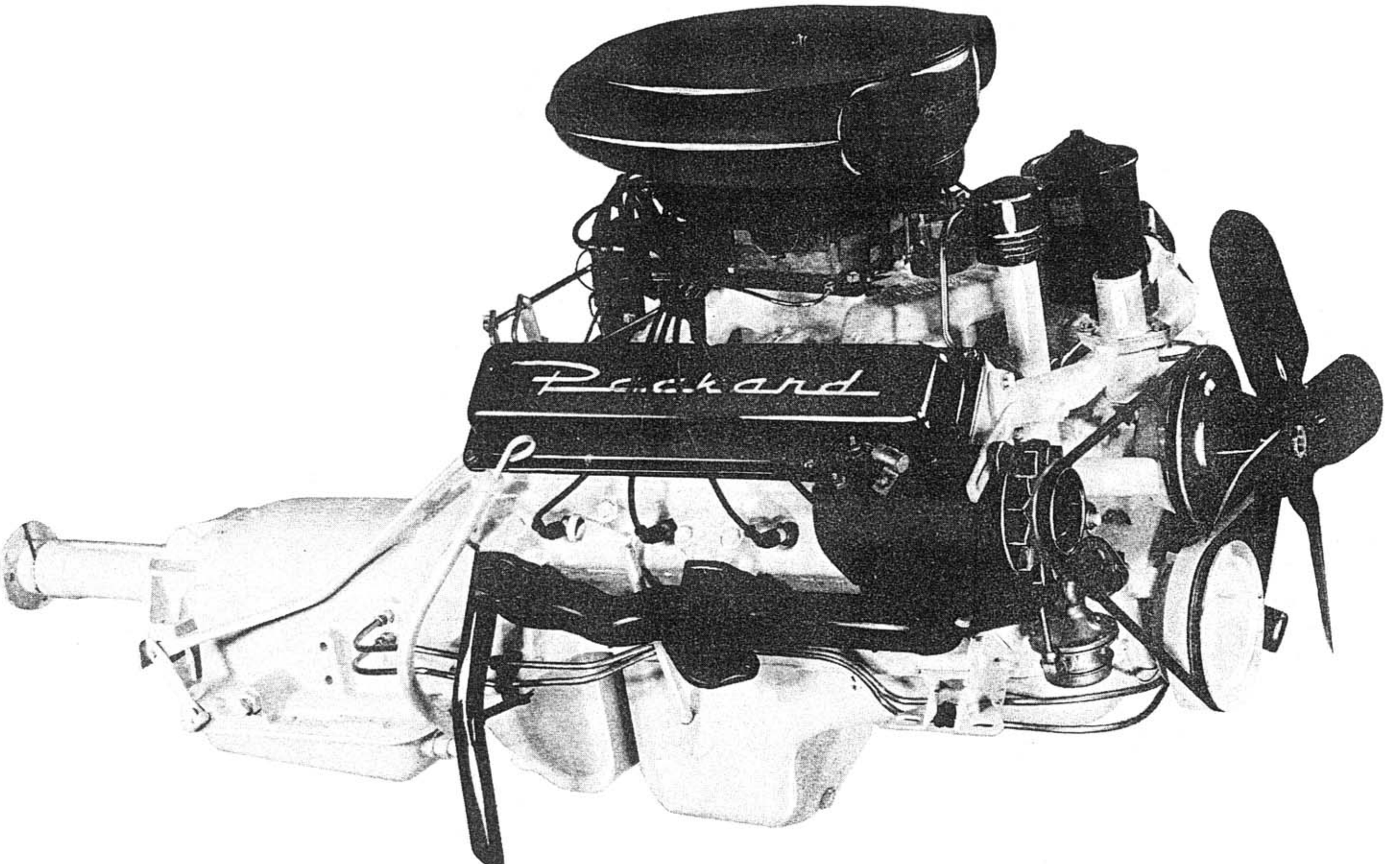


Fig. 1—The New Packard V-8 Engine

THE NEW PACKARD V-8 ENGINE

INTRODUCTION:

As a result of looking ahead and analyzing future requirements, an intensive engine development program was undertaken shortly after World War II. Its objectives were twofold, namely, (1) to fully exploit the potentialities of the Packard L-head, straight-8 engine for use during the interim period, and (2) to vigorously pursue the exploration of new and novel engine types and evaluate their suitability for use sometime in the future.

In 1949, it was becoming increasingly evident that the time was approaching when the familiar L-head engine would reach its limit of efficient utilization of high octane premium fuels. Studies indicated that an engine of valve-in-head design offered the most promise of meeting future requirements. The feasibility of redesigning the existing straight-8 was considered, but this approach was discarded, since any change would require a major design and tooling program. Moreover, the trend throughout the industry was toward V-8's, and the public was being educated to expect V-8's.

When the decision was made to proceed with the design and development of the new valve-in-head V-8, an important turning point was reached. Substantial sums of money had to be budgeted for engineering this new engine, as well as for its manufacture. Once the outlay has been made for such a program, the plant facilities and tooling must be usable without obsolescence and without much significant change for a long time to come. As a result, any new engine design must be approached on the basis of long range planning and looking ahead.

In planning the Packard V-8, the primary objective was to design into this new engine the best features that present technical knowledge and the state of the art permit, providing sufficient flexibility to incorporate future changes as better fuels become available and increasingly higher compression ratios become practicable. Instead of being concerned about designing to meet minimum requirements, the contrary approach was taken to insure that changing requirements could be satisfied without major retooling. Tentative specifications were first established for an engine that would meet these requirements, as well as those of the cars planned for 1955 production. Other major design objectives were, as follows:

a. **Basic Design** — Valve-in-head type to permit free-breathing and maximum volumetric efficiency.

b. **Physical Characteristics** — Size and weight consistent with long life; adequate structural rigidity to provide smoothness and quietness despite future operation at higher compression ratios.

c. **Displacement** — Approximately the same as the L-head straight-8 engine being superseded, and adaptable for enlargement without major changes in tooling.

d. **Combustion System** — Combustion control during the burning cycle to minimize octane requirements, and efficiently utilize improved fuels at increasingly higher compression ratios.

e. **Stroke-Bore Ratio** — Short stroke and large bore to increase mechanical and thermal efficiencies.

f. **Performance** — Capable of high specific output, maximum effective torque, and improved fuel economy.

g. **Durability** — Exceptionally long life, providing a new standard for passenger car application.

h. **Producibility** — Basic simplicity for ease of assembly and economy of manufacture.

BACKGROUND:

Vast experience and know-how have been gained since Packard began building automotive engines back in 1899. Over the fifty-six years, Packard has designed and developed nearly every conceivable type of internal combustion engine, including L-head, valve-in-head, overhead camshaft, and rotary valve types with radial, horizontal, vertical, in-line, 'V', 'W', and 'X' cylinder configurations.

For more than forty years, Packard has built V-type engines, starting with the Packard '299', '905', the twin six of the 1915 to 1923 era, the renowned Liberty engine of World War I, and the more recent classic twelve produced from 1932 to 1939. During World War II, Packard built the famous Rolls-Royce Merlin V-12 aircraft engine, and the equally famous Packard 4M-2500 V-12 marine engine which established such a fine reputation in torpedo boat operation. Being able to draw upon the vast reservoir of knowledge gained from the work conducted on these various types of engines has been invaluable during the design and development of the new Packard engine.

In planning the new engine, the performance requirements were projected into the future on the basis of existing scientific knowledge, as well as the predicted improvements in fuels over the years. For example, when work was first initiated back in 1949 on the design of the new Packard V-8 engine, it was anticipated that an engine, having a displacement of 269 cubic inches, would be sufficiently powerful. Subsequently, the displacement was increased first to 303, then to 333 and finally to the present 352 cubic inches, as it became increasingly evident that greater torque was required to satisfy the higher performance requirements of the new models planned for future production. Although horsepower ratings can be increased in smaller displacement engines by increasing the volumetric and thermal efficiencies, the only way to get adequate torque is by starting with an engine of the right size. The new Packard V-8 engine has more displacement and offers higher torque ratings than any other engine in current automotive production. In addition, by providing extra spacing between the cylinder bores, ample provision exists for even greater displacement without changing foundry or production equipment materially.

During the continuing development, the promise of unusually high performance offered by the new engine generated a great deal of interest in its potentialities. As a result, sufficient preliminary work was conducted to explore the most practical methods of obtaining still greater performance. To improve the breathing characteristics of the engine, changes were made in the combustion chamber to accommodate even larger valves than used in the production engine, various types of intake manifolds were investigated, special camshafts were tested, and the exhaust system was modified to minimize restriction. After determining the individual improvements obtainable by making these modifications, an investigation was made of the influence of variations in combustion chamber design and compression ratio.

Improvements in performance obtained during this work were sufficient to indicate that the new engine offers adequate capabilities of meeting future requirements for still greater horsepower and torque. An improvement of over 45% in power and 15% in torque is entirely feasible without major alteration of the basic engine. As work continues to extract still higher performance, the advanced designs resulting from these exploratory investigations will become future production realities.

DESCRIPTION AND GENERAL ARRANGEMENT:

As illustrated by Figure 1, the new Packard engine has a valve-in-head 90° V-8 configuration, and a large bore and short stroke, providing improved volumetric, mechanical and thermal efficiencies. Superseded by the new engine is the L-head straight-8 engine, having a displacement of 359 cu. ins. and a rating of 212 bhp, which was used in the 1954 Packards. Brief specifications for the new engine are given in Table I.

After establishing the basic design of the engine, considerable effort was devoted to locating the various engine components and accessories. The general arrangement and details of construction are shown by the transverse and longitudinal cross sections of the engine, Figures 2 and 3, respectively.

As will be discussed later in this paper, such major components as the cylinder block, cylinder head, and intake manifold were all designed to facilitate casting, machining, and assembly. In the location and arrangement of these items, as well as the various accessories, maximum mechanical and functional simplicity were considered extremely important to allow for ease of servicing and economy of manufacture.

a. Size: The reduced height and length of the 90° V-8 engine greatly simplifies the chassis design and offers the body stylist

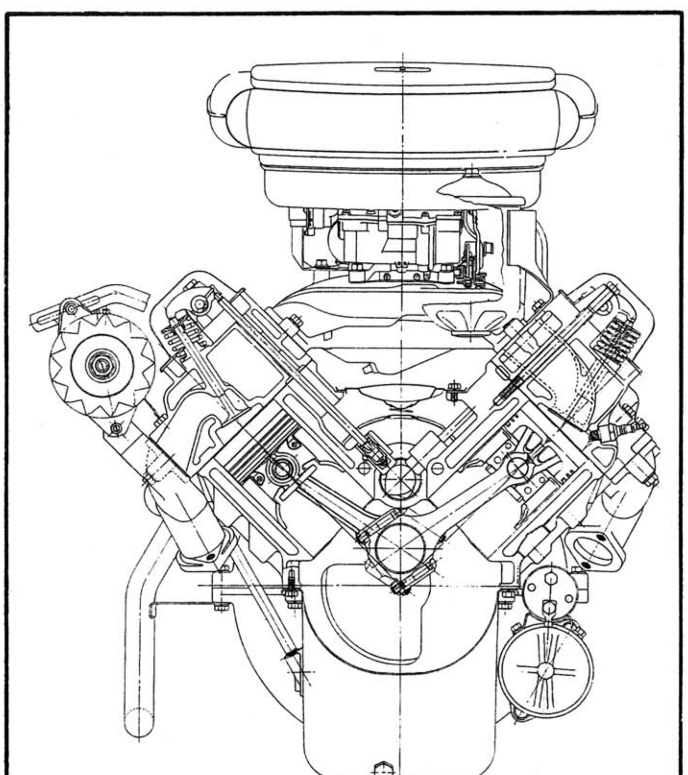


Fig. 2—Transverse Sectional View of the Engine

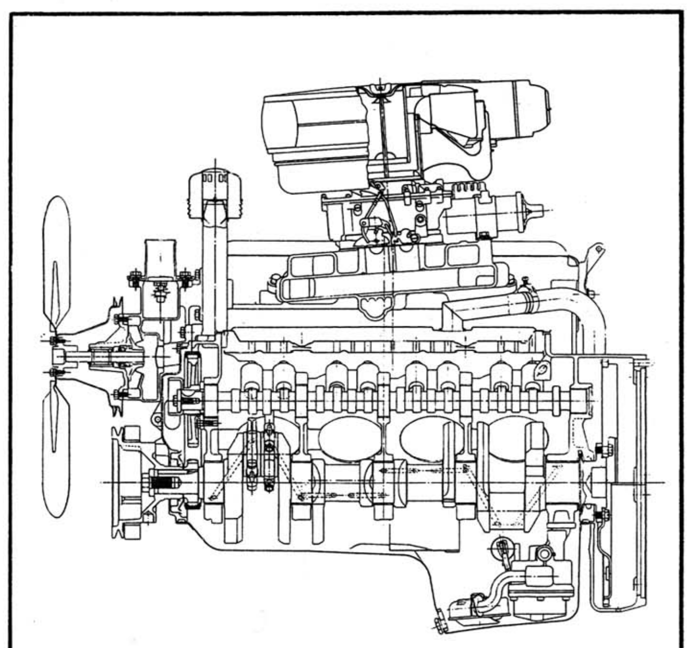


Fig. 3—Longitudinal Sectional View of the Engine

TABLE NO. I—GENERAL ENGINE SPECIFICATIONS

Type	OHV, 90° V-8
Bore, ins.	4.0
Stroke, ins.	3.5
Stroke-to-Bore Ratio	.875
Displacement, cu. ins.	352
Compression ratio	8.5:1
Designation of Cylinders	
Left bank, front to rear	1-3-5-7
Right bank, front to rear	2-4-6-8
Firing Order	1-8-4-3-6-5-7-2
Maximum Gross Brake Horsepower*	260 @ 4600 rpm
Maximum Gross Torque, lbs.-ft.*	355 @ 2400-2800 rpm
Maximum B.M.E.P., lbs./ sq. in.	152.1 @ 2600 rpm
Piston Travel, ft./ mile	1480
Engine Weight, lbs.	698

*Corrected to SAE standard conditions of 29.92 in. Hg. atmospheric pressure and 60° F. dry air.

TABLE NO. II—COMPARISON OF 1954 AND 1955 ENGINE WEIGHTS

Item	1955 V-8	1954 St.-8
Cylinder Block, Bare	210	283
Cylinder Head(s), Bare	128 (C.I.)	25 (Al.)
Crankshaft	56	109
Ultramatic Flywheel and Ring Gear Ass'y.	8	9
Connecting Rod Ass'y., Complete Set	14	19
Intake Manifold	28	24
Exhaust Manifold (s)	23	28
Camshaft	10	13
Valve Train w/o Camshaft	27	11
Engine, Complete Ass'y., Incl. All Accessories, Except Air Cleaner, Dry	698	752
Radiator, Complete w/Core and Tank	22	32
Engine and Radiator, Dry, Total Weight	720	784

somewhat more flexibility than was permissible with the 1954 straight-8 engine. However, the increased width of the 90° V-type of engine poses some clearance problems, particularly on the steering side of the engine compartment. Figure 4 shows a comparison of the 1955 V-8 engine and the 1954 straight-8 engine. The new engine is ½-inch lower and 8½ inches shorter than the 1954 engine. Although the 90° V-8 is somewhat wider, judicious arrangement of the other items also housed in the engine compartment eliminated any real difficulty from this standpoint.

b. Weight: Studies and investigations were conducted to insure that full advantage was taken of the inherently lighter 90° V-8 design. As shown by the weight comparison of the major components tabulated in Table II, a reduction of over 8% resulted from the introduction of the new engine. An even greater reduction in weight could have been obtained if compromises had been acceptable in the life and durability of this engine. Even so, it is evident that the new Packard V-8 engine compares very favorably with competitive V-8 engines on the basis of pounds-per-cubic-inch of displacement.

In addition to the direct savings resulting from the weight reduction, the new engine permits a somewhat more favorable weight distribution between the front and rear wheels of the car. Hence, by reducing the engine weight, an improvement has also been obtained in the amount of effort required for handling and steering the car.

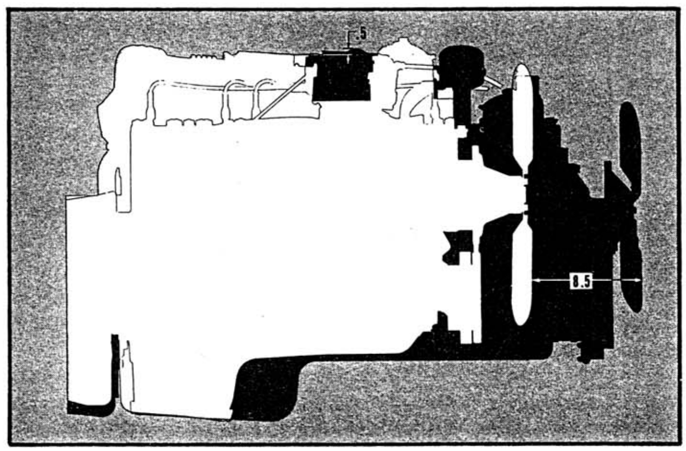


Fig. 4—Engine Size Comparison

COMBUSTION CHAMBER:

After extensive testing of the various types of combustion chambers and evaluation of their respective features, the high turbulence, wedge-type combustion chamber, having an elliptical shape in the plan view, Figures 5 and 6, was selected as being the most satisfactory for passenger car application. This design provides a lower burning rate of the charge and avoids

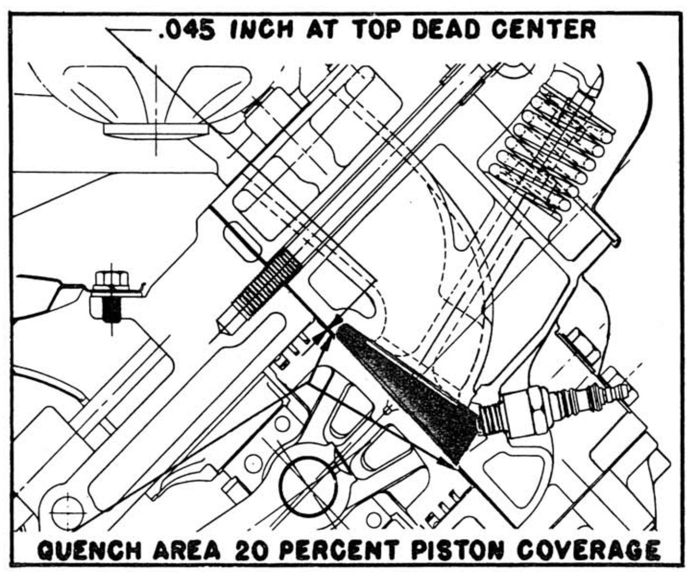


Fig. 5—Cross-sectional View of the Combustion Chamber

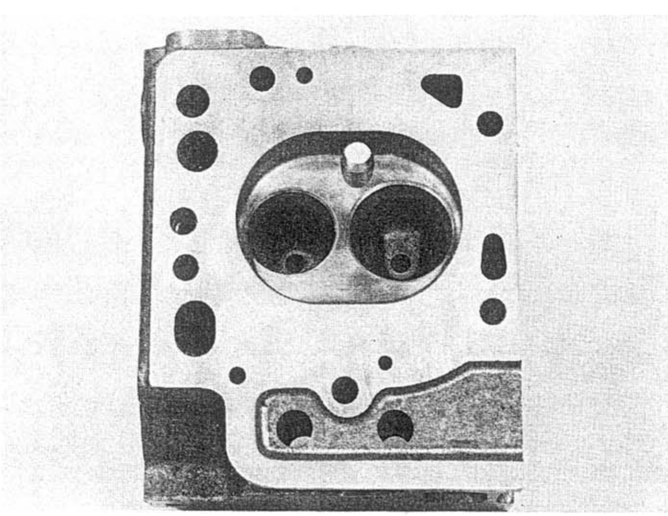
a high rate of pressure rise, resulting in freedom from objectionable combustion roughness. By cooling the last part of the charge to burn in the shallow section of the combustion chamber, called the quench area, combustion can be effectively controlled and the octane requirements are minimized. Another important factor in favor of this type of combustion chamber is its adaptability for increasingly higher compression ratios without sacrificing smoothness.

The development of the combustion chamber for the new Packard engine is based on studies of combustion phenomena and the influence of turbulence, quench area, squish, spark advance, and other related factors. This work has culminated in a design remarkably insensitive to combustion chamber deposits which are so troublesome in other designs, particularly under light load city driving conditions, causing knock, auto-ignition, pre-ignition and running-on. The combustion chamber shape finally evolved is machined for accurate volume control and now gives outstanding performance. The quench area amounts to 20% of the piston coverage, and the nominal squish clearance is .045-inch with the piston at top dead center, Figure 5.

COMPRESSION RATIO:

After working with engines at various compression ratios all the way up to 12 to 1, a compression ratio of 8.5 to 1 has been established for the new Packard engine, based on the octane rating of the premium grade fuels now available throughout the country. At this compression ratio, continued satisfactory performance can be obtained even in the presence of substantial combustion chamber deposits. Retarding the spark to permit acceptable operation after a few thousand miles of driving was not considered a suitable approach in establishing the compression ratio for the new engine, simply to be able to offer a somewhat higher compression ratio. The octane requirements were established during operation of cars at the Packard Proving Grounds under a variety of driving conditions. Their data indicated that the Ultramatic equipped cars could be satisfied with 94 octane fuel, even under the most severe prolonged light load city driving conditions. Obviously, the compression ratio has been conservatively established to avoid having to specially adjust or service the engine in those parts of the country still lacking, at this time, the super fuels required to obtain successful performance at somewhat higher compression ratios.

Fig. 6—Completely machined, high turbulence, wedge type combustion chamber, showing the side-by-side relationship of the valves and the spark plug location.



INDUCTION SYSTEM:

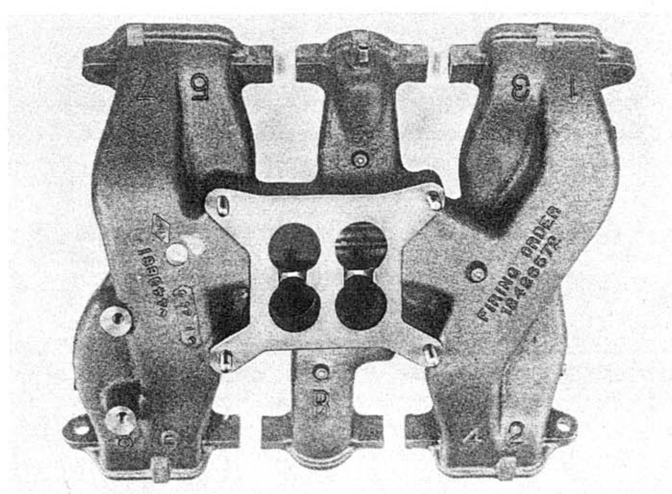
Since the power output of the engine is so largely dependent upon volumetric efficiency, considerable effort was devoted to the design and development of the induction system. Studies were made of the aerodynamic flow of the air-fuel charge through the induction system, resulting in the development of an exceptionally free-breathing engine. Extensive work was conducted to obtain equal distribution to all cylinders, to minimize restriction, and to promote turbulence only where desired for proper mixture of the charge entering the combustion chamber.

Early intake manifold designs were constructed out of wood, so that variations could readily be made to determine their influence on fuel distribution and general performance. As illustrated by Figure 7, the intake manifold finally evolved for the new engine has a 90° T-shaped configuration at the intersection of the branches. The passages are smoothly contoured and equal in cross-section area throughout, and generous radii are provided at all junctions.

The carburetor choke stove is located in the center exhaust cross-over passage of the intake manifold. The flow of the exhaust gases through this passage connecting the cylinder banks is controlled by the action of the heat valve located at the junction of the left-hand exhaust manifold and the exhaust pipe. By forcing exhaust gases through the cross-over passage underneath the risers during warm-up, sufficient heat is transferred to rapidly vaporize the fuel flowing through the intake manifold, and to the air being supplied the thermostatically controlled choke for positive operation. Adequate heat is not only applied to the intake manifold passages so that the choke comes off promptly during warm-up, but also to avoid a noticeable transition thereafter in the performance of the engine. However, the charge is only sufficiently preheated in the intake manifold to provide for successful ignition of somewhat leaner mixtures at part-throttle operation without adversely sacrificing octane requirements at full-throttle operation.

The passages in the cylinder head are likewise equal in cross-section throughout, possess generous radii at all junctions and are contoured to minimize restriction, control flow and provide optimum turbulence as the charge enters the combustion chamber. By taking advantage of the large bore offered by the new design, exceptionally large valves could be used, making an important contribution to the free-breathing characteristics of the engine.

Fig. 7—Intake Manifold



VALVE TRAIN:

As a result of studies made during the development of the new engine to obtain satisfactory dynamic valve motion, the cam profile was changed to decrease acceleration rates on both the opening and closing sides, and a lower rate valve spring was developed to reduce the cam nose loading. By using larger diameter, heavier wall tubing, the push rod rigidity was substantially increased, thereby minimizing deflection in the valve train. After making these changes, considerable improvement was obtained in the erratic behavior of the valve train formerly occurring at 4200 rpm, and the critical lifter "pump-up" speed was increased to over 5000 rpm. The valve train now used in the new engine is shown by Figure 8, and its design specifications are given in Table III.

Considerable work was conducted to obtain compatibility between the camshaft and lifters. Various types of materials and surface treatments were investigated in an effort to obtain freedom from excessive wear during the break-in period, and sufficiently high fatigue strength for long-life under subsequent operating conditions.

The precision molded alloy iron camshaft is hardened after casting, phosphate coated all over, mounted on five bearings, and chain driven. The cams are ground with a taper of 6 minutes, and are positioned .062 inches to the rear of the lifter centerline to avoid lifter over-run and insure positive lifter rotation.

The lifters are of the hydraulic type, having hardenable iron bodies. The lifter faces are ground to a spherical radius of 30 inches, hardened to a minimum of 54 Rockwell C, and Lubrited for improved break-in.

As a result of extensive testing, valves were developed for this engine that possess life equivalent to the other components in the engine. The exhaust valve, for example, is of the special flexible head design to assure maximum conformability of the valve face and seat. Figure 9 shows the 2112 austenitic steel exhaust valve which has a 1.687-inch head diameter and a 45° seat angle. Also shown in Figure 9 is the hardened and tempered Silichrome #1 steel intake valve, which has a 1.937-inch head diameter and a 30° seat angle. These large valves in combination with the lift of .375 inches are important factors in the high performance of the new engine.

Close coordination with the foundry resulted in obtaining generous cored passages around the valve seats, thereby assuring rapid and uniform heat dissipation to the coolant. Valve temperatures were further reduced by incorporating integral valve guides which eliminate the thermal barrier normally encountered with the separate valve guides, Figure 9. During the testing, it was found that the stem-to-guide clearances were extremely critical. As a result of this work, the stem-to-guide clearances were established at .001-.002 inches for the inlet valves and .002-.003 inches for the exhaust valves, requiring selective assembly in manufacture. Valve sticking results from deviation from the minimum limits, whereas inadequate oil control exists, particularly on the inlet valve, when the clearances are excessive.

The fuel pump is actuated by a hardened, chrome plated, stamped, steel eccentric bolted onto the front of the camshaft sprocket. The unbalance resulting from the eccentric has been compensated for by the non-symmetrical location of the openings in the camshaft sprocket.

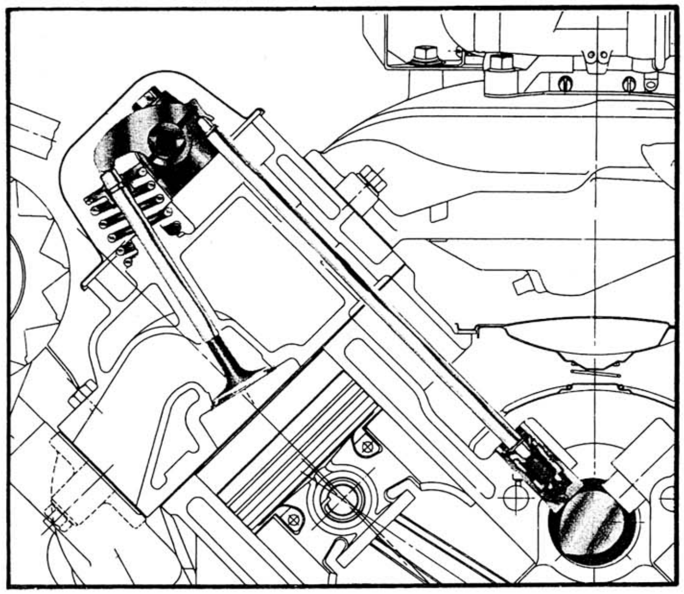


Fig. 8—Valve Train

TABLE NO. III—
VALVE TRAIN SPECIFICATIONS

Camshaft	
Material	Hardenable alloy iron
Bearings, No.	5
Drive, type	Chain
Width, ins.	1.000
Pitch, ins.	.375
Cams	
Width, ins.	.500
Taper, mins	6
Lifters	
Type	Hydraulic
Body	
Material	Hardenable alloy iron
Diameter, ins.	.904
Face, spherical radius, deg.	30
Push Rods	
Material	Steel tubing
Size	
Diameter and wall thickness, ins.	.375 x .065
Length, ins.	10.506
Ends, spherical radius, ins.	.250
Valve, Intake	
Material	Silichrome #1 steel
Head diameter, ins.	1.937
Stem diameter, ins.	.3725
Seat angle, deg.	30
Valve, Exhaust	
Material	2112 austenitic steel
Head diameter, ins.	1.687
Stem diameter, ins.	.3715
Seat angle, deg.	45
Rocker Arm	
Material	Pearlitic malleable iron
Ratio	1.60
Valve Lift, ins.	.375
Valve Spring Load	
Closed, lbs.	82
Open, lbs.	165
Valve Timing	
Intake opens, Deg. BTC	14
Intake closes, Deg. ABC	56
Exhaust opens, Deg. BBC	52
Exhaust closes, Deg. ATC	18

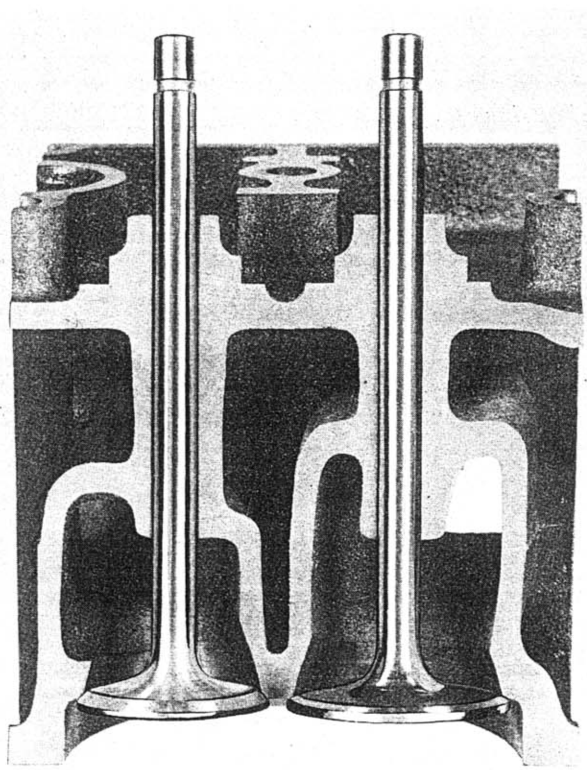
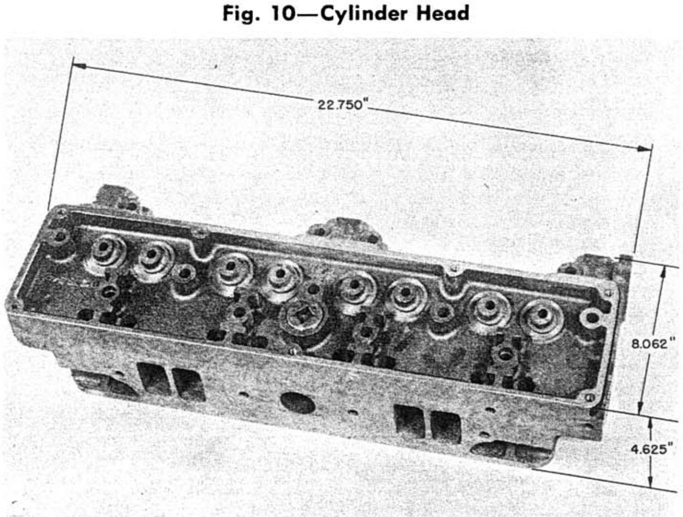


Fig. 9—Valves and Valve Guides

CYLINDER HEADS:

The cylinder heads are conservatively designed, having greater height, width and length than other competitive designs using a similar type of combustion chamber, Figure 10. Generous water jackets have been provided for adequately cooling the valve seats, valve guides and gas passages, Figure 11. Conventional casting methods are used in making the cylinder heads, and the experimental design has been successfully translated into production. The cylinder heads are interchangeable and are attached to the cylinder block by 15 screws appropriately spaced so that 5 screws surround each bore, Figure 14. Tension loads are transferred to the bulkheads in the cylinder block by this arrangement, permitting the use of a thin, one-piece, embossed, steel gasket.

Fig. 10—Cylinder Head



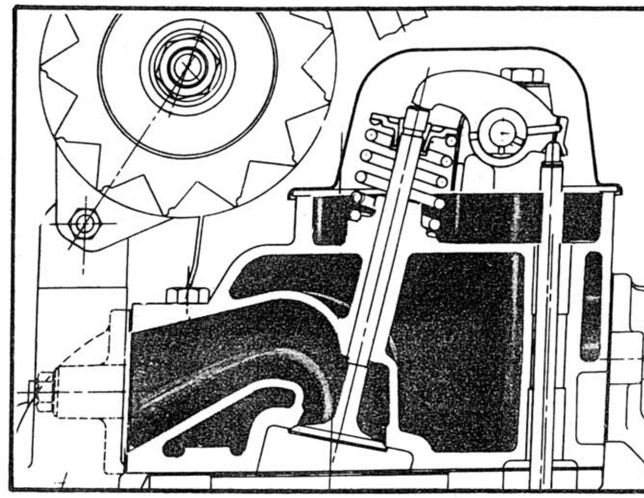
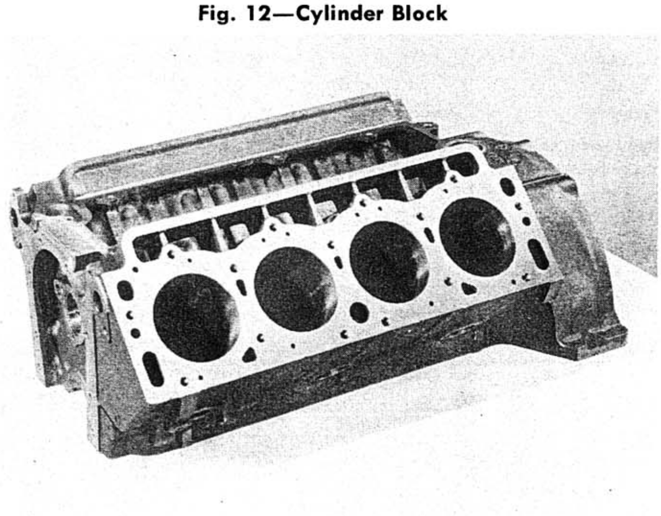


Fig. 11—Cross Sectional View of the Cylinder Head

CYLINDER BLOCK:

Taking full advantage of one of the inherent characteristics offered by the 90° V-8, an exceptionally rugged, durable cylinder block has been developed for this engine, providing adequate rigidity to support the loads, Figure 12. The loads are distributed evenly throughout the entire structure by five transverse bulkheads which tie the two blocks into a single rigid unit, made from a one-piece casting of high grade alloy iron. After various stress coat and strain gauge studies, as well as dynamic tests, it was not considered necessary to extend the casting below the centerline of the crankshaft. To satisfactorily locate the main bearing caps, however, longitudinal rabbets are broached .125 inches above the oil pan gasket surface, Figure 13. The cylinder head screw bosses are tied into the water jacket wall structure by vertical ribs to avoid stressing the individual cylinder barrels. Substantially larger water jackets have been provided around the full length cylinder walls than offered by any other competitive design, lowering the temperature of the working parts and giving unusually long life. The center-to-center distance of the cylinder bores is 5 inches and the over-all length of the block is 27.750 inches. The upper half of the flywheel housing is cast integrally with the cylinder block, offering improved support for the transmission and propeller shaft by minimizing the deflection resulting from the attachment of these items, Figure 14.

Fig. 12—Cylinder Block



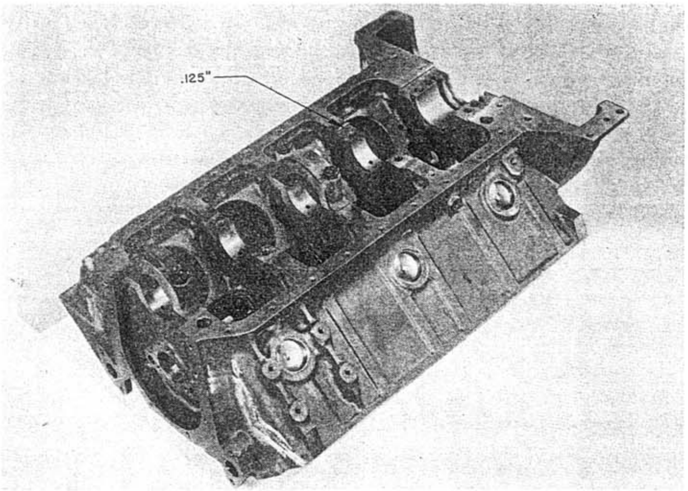


Fig. 13—Underside of Cylinder Block

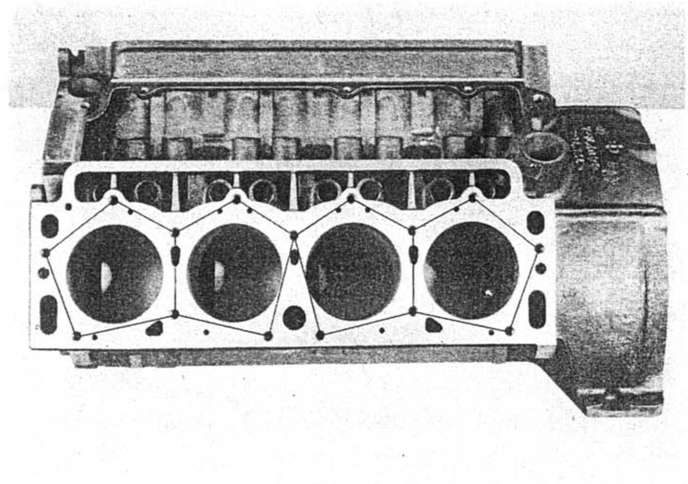


Fig. 14—Right Side of Cylinder Block

CRANKSHAFT:

As a result of an unbiased evaluation of the relative merits of both the cast and forged types of crankshafts, a cast-steel crankshaft is used in the new Packard V-8 engine, Figure 15. Cast steel offers a sufficiently high modulus of elasticity as well as density to provide substantial savings in weight without sacrificing rigidity or stiffness. This type of crankshaft also permits disposition of the counterweights for maximum balancing effectiveness, and allows coring of the crank pins to minimize the amount of unbalance that must be compensated for by the counterweights. A physical description of the crankshaft is given in Table IV.

Either type of crankshaft could have been selected without the penalty of increased over-all length, since the conservatively designed cylinder block already offered adequate spacing to accommodate the uncheeked counterweights and still provide grinding wheel clearance at each crank arm. As indicated by Table IV, only 9 pounds of metal are removed during the processing of the cast crankshaft, or less than half the amount that would have to be removed from the forged type. By using the cast crankshaft, not only has a substantial weight reduction been obtained, but excellent manufacturing economies have also been realized.

By using a non-bonded rubber harmonic balancer, a suitable reduction has been obtained in the amplitude of crankshaft deflection, providing satisfactory performance throughout the normal operating speed range. Although a fifth order occurs at approximately 3400 rpm, its amplitude has been dampened to .22 degrees double amplitude and is imperceptible in the car. Other torsional disturbances do not occur in the normal driving speed range. As shown by Figure 16, the torsional vibration characteristics of the new engine are greatly superior to the 1954 engine. The increased stiffness offered by the shorter crankshaft accounts primarily for the improvement, despite the higher output of the new V-8.

TABLE NO. IV—
CRANKSHAFT SPECIFICATIONS

Main Bearings, No.	5
Counterweights, No.	6
Over-all Length, ins.	27.344
Connecting Rod Journals, dia., ins.	2.250
Connecting Rod Effective Bearing Area, sq. in.	52.8
Main Bearing Journals, dia., ins.	2.500
Main Bearing Journal Effective Area, sq. ins.	38.6
Bearing Journal Overlap, ins.	.625
Rough Casting Weight, lbs.	65
Machined Casting Weight, lbs.	56

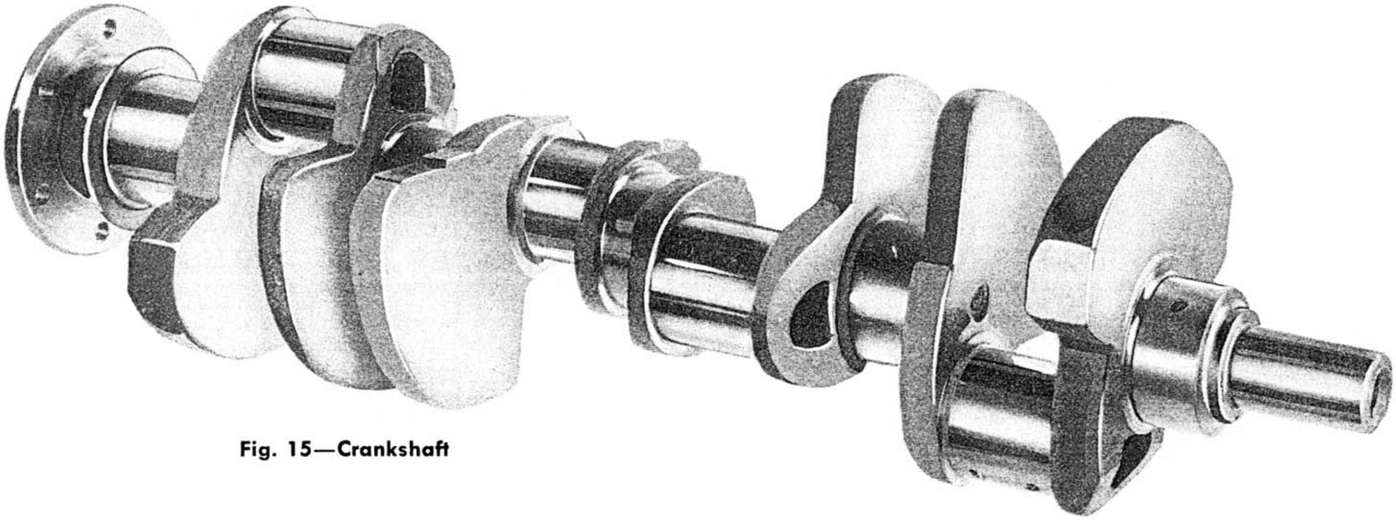


Fig. 15—Crankshaft

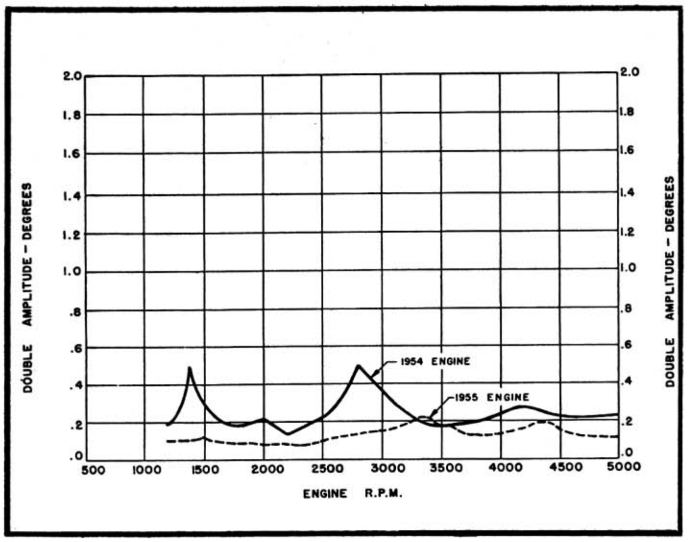


Fig. 16—Comparison of Crankshaft Torsional Vibration Characteristics

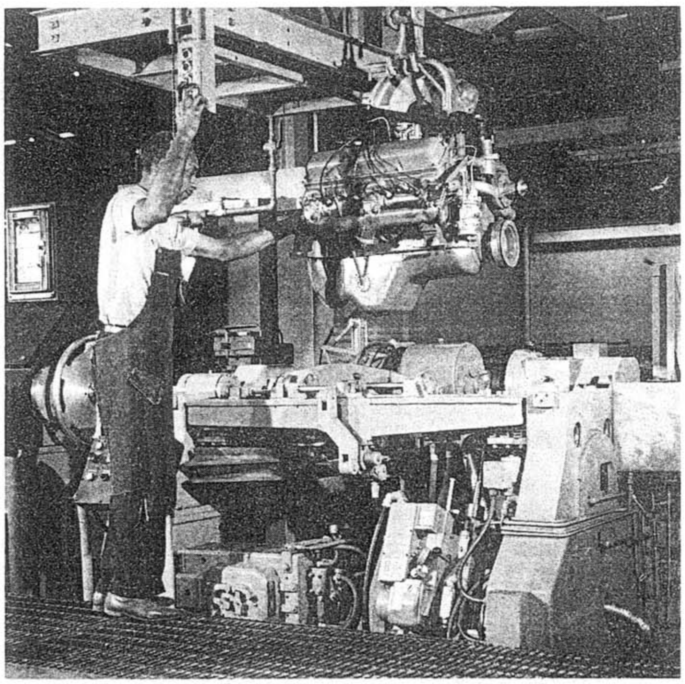
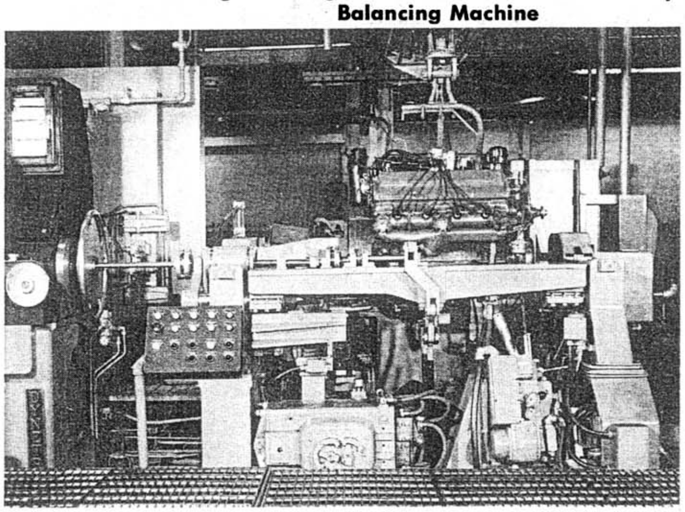


Fig. 17—Lowering Engine for Final Assembly Balancing

Fig. 18—Engine Mounted on Final Assembly Balancing Machine



After assembly of all the components, which have been individually balanced for use in production or in service, the engine is mounted on a specially designed machine and balanced as a unit, Figure 17. In performing this operation, the engine is motored sufficiently to indicate the amount of unbalance, at which point rotation stops and the prescribed amount of unbalance compensated for by automatically drilling into the crankshaft pulley and welding a slug onto the flywheel, Figure 18. As a result, the stack-up of tolerances that otherwise might accumulate during the assembly of the engine is reduced to not more than ¼-inch-ounces, thereby insuring precision balancing of every engine.

BEARINGS:

All of the main bearings are of the same diameter. The thrust load of the crankshaft is taken on the rear main bearing. Steel-backed micro lead-babbitt is used for the connecting rod bearings, and overplated copper-lead is used for the main bearings. Both types of bearing materials perform equally as well on the cast-steel crankshaft without hardening of the journals. Both connecting rod and main bearing loads are sufficiently low so that the bearing materials now used give exceptional durability and long life.

CONNECTING ROD:

The connecting rod now used in the new V-8 has been conservatively designed and extensively tested under loads substantially in excess of those imposed under any driving condition, Figure 19. Table V gives brief specifications for the connecting rod.

In an effort to reduce the reciprocating and rotating weight, the original connecting rod was redesigned by reducing the column section and crank pin journal end of the rod. Subsequent testing indicated that some improvement was obtained in the natural frequency of the mass elastic system without sacrificing any of the structural rigidity of the connecting rod. Balancing lugs are located at each end of the rod assembly.

PISTON AND RINGS:

Following previous Packard practice, the new V-8 also uses autothermic aluminum alloy pistons, Figure 20. Brief specifications for the piston and rings are given in Table VI.

Piston slap has been eliminated by study of the various factors related to the piston-to-bore clearance and the manner in which the piston moves across the bore near the top of the firing stroke, resulting in an offset of the piston pin of .062 inches toward the major thrust face. The fire-wall or top of the piston is .280 inches thick, allowing for trouble-free operation at compression ratios well above the currently used 8.5 to 1. Extensive tests were conducted to establish the caming and surface finish of the piston skirt, and complete freedom from scoring has resulted.

As shown in Figure 20, two thick wall, alloy iron, 5/64-inch wide, taper face compression rings are used in combination with an open slot, ventilated, alloy iron, 3/16-inch wide, oil ring equipped with a hump type expander. Complete freedom from scuffing during initial engine operation has been obtained by using a .004-.007-inch chrome plated top compression ring along with the Ferrox coated second compression ring. No undue difficulty was encountered in obtaining satisfactory oil control under various driving conditions, both in urban and rural traffic. As the manifold vacuum increases, oil control became somewhat more troublesome, however, the variation in oil consumption between high and moderate vacuum conditions was relatively small. Although smoking from the exhaust may be observed under certain driving conditions, for example, after long periods of idling, comparison with competitive engines has been quite favorable.

The full-floating type of piston pin was selected, since past experience with the combination of the autothermic piston and this type of piston pin has been unusually satisfactory. Table VI gives additional information on the piston pin.

LUBRICATION SYSTEM:

As illustrated by Figures 21 and 22, lubrication for the new engine is provided by a full-pressure system. Brief specifications for this system are given in Table VII.

After entering the pump through a screened floating inlet at a controlled distance beneath the top of the oil level, oil is delivered through the rear bearing cap to a vertical oil header. From this vertical header, oil is forced (1) through drilled passages to lubricate the rear main bearing and adjacent connecting rod bearings, (2) through an intersecting right longitudinal gallery, and (3) through a connecting header to feed the rear camshaft bearing and the left longitudinal gallery.

Through connecting drilled passages, oil flows from the right main gallery to the rocker arm shaft and the valve lifter guides on the right bank, and also to the center and front main bearings. The center and front camshaft bearings, respectively, are lubricated from the center and front main bearings, as well as the adjacent connecting rod journals.

Oil flowing through the left main gallery lubricates the valve lifter guides and the rocker arm shaft on the left bank, the two intermediate main bearings, and all adjacent connecting rod bearings. The two intermediate camshaft bearings are lubricated through drilled holes from the intermediate main bearings.

Forty-five degree angular headers, located at the rear of the right main gallery and at the front of the left main gallery, supply oil upward through the cylinder block and head to the rocker shafts, Fig. 23. From the rocker shafts which act as oil galleries, oil is metered to the rocker arms and thence flows onto the valve stems and push rod sockets. The surplus oil collects in the deep trough adjacent to the rocker cover rail and returns through the cylinder head drain-back passages and hollow cylinder head dowels into the crankcase.

Oil is delivered to the partial flow filter through a drilled and tapped hole that intersects the angular header which lubricates the left rocker shaft and rocker arms. The filtered oil returns through a tapped hole, located in the cylinder block near the oil filler tube, to the timing chain compartment and thence to the sump.

Figure 21 shows the method of lubricating the distributor and oil pump drive. The distributor drive gear is splash lubricated, and its thrust face is lubricated by a combination of splash and gravity feed. Oil flowing through a 5/16-inch diameter hole in the distributor housing provides splash lubrication of the distributor gear bearing.

The oil pump is driven by an intermediate floating shaft coupled to the distributor drive gear, Figure 21. A tongue and groove arrangement on the oil pump end of the drive shaft allows for slight misalignment. Continuing previous Packard practice, the oil pump uses coarse pitch straight gears, providing freedom from pulsation and quiet operation. Variation in oil pressure has been reduced throughout the speed range, as a result of

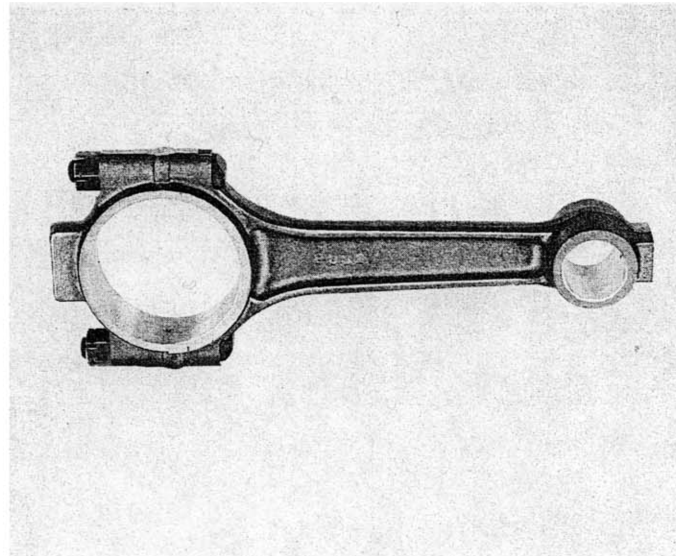


Fig. 19—Connecting Rod

TABLE NO. V—
CONNECTING ROD SPECIFICATIONS

Center-to-Center Distance, ins.	6.781
Length-to-Stroke Ratio	1.937
Material	SAE 1041 steel
Weight w/ Piston Pin Bushing, gms.	757

Fig. 20—Piston Assembly

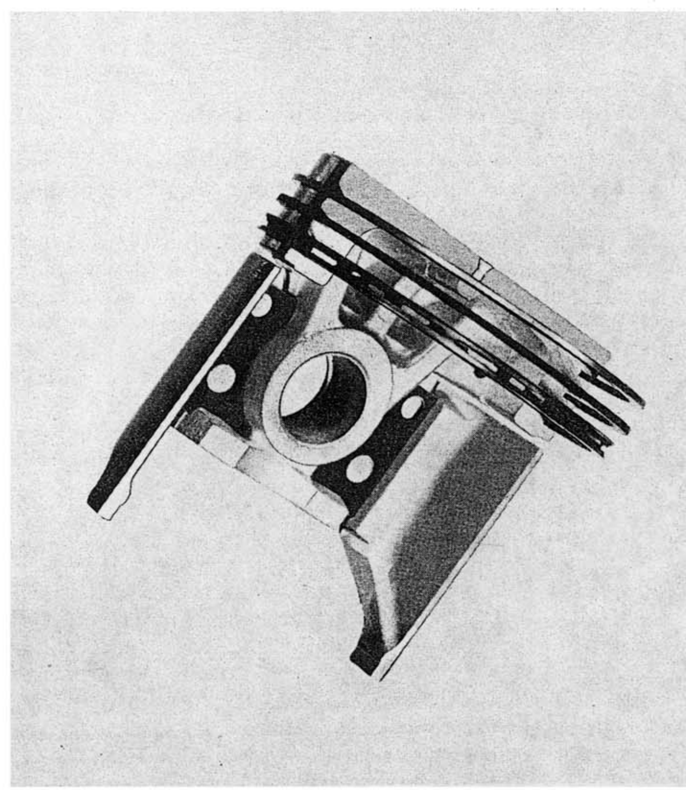


TABLE NO. VI—SPECIFICATIONS OF PISTON, RINGS AND PINS

Piston	
Type	Autothermic, cam ground, slipper skirt
Material	Aluminum alloy
Weight, gms.	702
Piston Rings, Compression	
Number	2
Type	Thick wall, taper face
Material	
Top	Alloy iron, chrome plated
Intermediate	Alloy iron, Ferrox coated
Width, ins.	.078
Piston Rings, Oil	
Number	1
Type	Open slot, ventilated, w/expander
Material	Alloy iron
Width, ins.	.186
Piston Pin	
Type	Full floating
Material	SAE 1117
Size, dia. and length, ins.	.980 x 3.250

TABLE NO. VII— SPECIFICATIONS OF LUBRICATION SYSTEM

Lubrication System, type	Full-pressure
Oil Pump, type	Gear
Oil Intake, type	Floating
Oil Pressure, max., psi	50
Oil Filter, type	Partial flow
Crankcase Capacity, Less Filter, qts.	5

work on the relief valve which re-circulates the oil whenever the pressure rises above the predetermined maximum of 50 psi.

To control cold scuffing and piston slap, the cylinder walls are lubricated by spraying a fine stream of oil from the connecting rod of the corresponding cylinder in the opposite bank, Figure 23. By providing a groove at the split line of the connecting rod, the best trajectory of the oil stream can be obtained without incurring objectionable stress risers in the highly loaded tension section of the rod. Ample lubrication is thereby provided on the critical areas of the upper sides of the cylinders, whereas gravity supplies oil to the lower sides. In addition to providing beneficial lubrication of the cylinder walls during cold starting, the oil sprayed from the connecting rod lubricates the piston pin and effectively cools the piston under hot operating conditions.

As illustrated by Figure 22, lubrication of the timing chain is accomplished by oil supplied from the left main gallery to a groove in the camshaft thrust plate. Oil passes through a radial hole, intersecting this groove, which directs a continuous stream of oil downward onto the timing chain. The fuel pump eccentric is lubricated by oil passing through a hole in the camshaft thrust plate, which communicates with the right main gallery and registers with a hole in the camshaft sprocket, allowing an intermittent stream of oil to spray onto its surface.

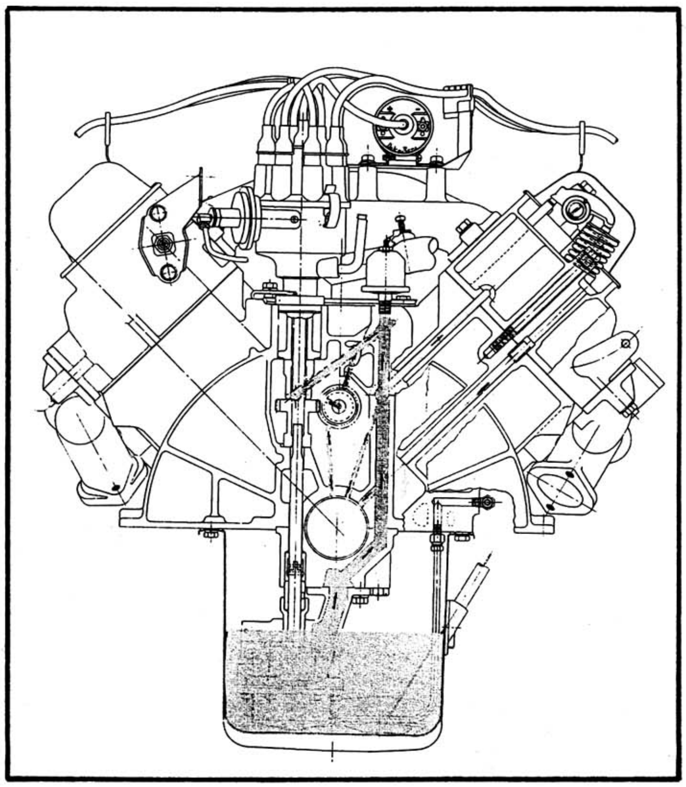
VENTILATION SYSTEM:

Positive circulation has been provided to insure thorough ventilation of the engine under all driving conditions. With this system, air enters the engine through the combined crankcase ventilator inlet and oil filler cap which is located at the front of the engine, directly in line with the fan blast.

After passing downward through the timing gear compartment into the crankcase, the fresh filtered incoming air becomes thoroughly intermixed with the corrosive blow-by gases. The air then passes upward through the push rod clearance slots in the stamped steel baffle, designed to separate the crankcase from the valve lifter compartment and to prevent oil pull-over. As the air flows through this relatively large, low velocity chamber, oil vapors are encouraged to condense before entering an opening near the front of, and leading into, a channel in the valve lifter compartment cover. From this channel, the air flows into the road draft tube, located at the rear of the valve lifter compartment cover.

During idling of the engine, the exhaust from the windshield wiper rotary vacuum booster pump, attached to the underside of the oil pump, creates a pressure difference which assists in the circulation of the air through the crankcase. When the car is in motion, vacuum created by the air flowing past the road draft tube increases the circulation of air, thereby assuring positive ventilation of the engine under all operating conditions.

Fig. 21—Transverse Cross-section of Lubrication System



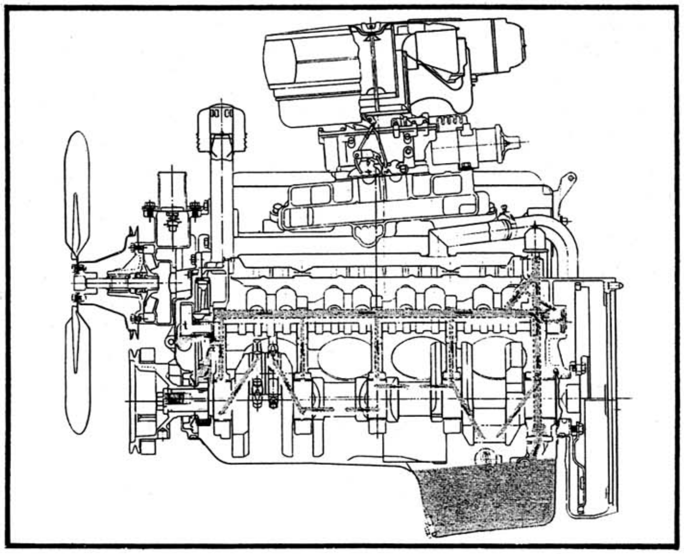


Fig. 22—Longitudinal Cross-section of Lubrication System

COOLING SYSTEM:

The cooling system employs a single, high capacity water pump, centrally mounted at the front of the cylinder block. The pump discharges directly into an equalizing chamber, supplying a balanced flow and assuring even distribution of coolant to both cylinder banks. Upon entering the cylinder banks, the coolant circulates through the passages around the cylinders and upward through cored holes to the cylinder heads. As the coolant flows through the large, generous passages, permitted by the conservative spacing of the cylinder bores, the critical regions around the cylinders, combustion chambers, valves, and spark plugs are thoroughly and adequately cooled. After leaving the cylinder heads through the outlet at the front, the coolant flows through a header, which houses the single thermostat. By restricting flow when the engine is cold, the thermostat controls the temperature of the coolant for prompt warm-up.

In comparison with the 1954 engine, the heat rejection of the new engine to the coolant was greatly reduced, as shown by Figure 24. At 4000 rpm, for example, the heat rejection amounted to 6900 btu/min. for the 1954 engine and 5400 btu/min. for the 1955 engine, providing a 21% improvement. As indicated in Table II, the reduction in the heat losses to the cooling system permitted a saving of 10 lbs., or 31% in the weight of the radiator required for the new engine.

ELECTRICAL SYSTEM:

Although early development of the new engine was conducted with the 6-volt electrical system, it became evident that higher voltages were necessary to satisfy the demands of the engine, as the compression ratio was progressively increased and the general over-all performance was improved. However, prior to the final selection of the electrical system, an analysis was made of the electrical requirements of the new engine, as well as those of the cars planned for 1955 production. As a result, the 12-volt electrical system was adopted for the new engine, since its higher output was considered essential for both the present and long range requirements. With this system, adequate voltage is always available to satisfy the requirements of the engine, as indicated by the comparison of the available voltage and the required voltage under the conditions shown by Figure 25.

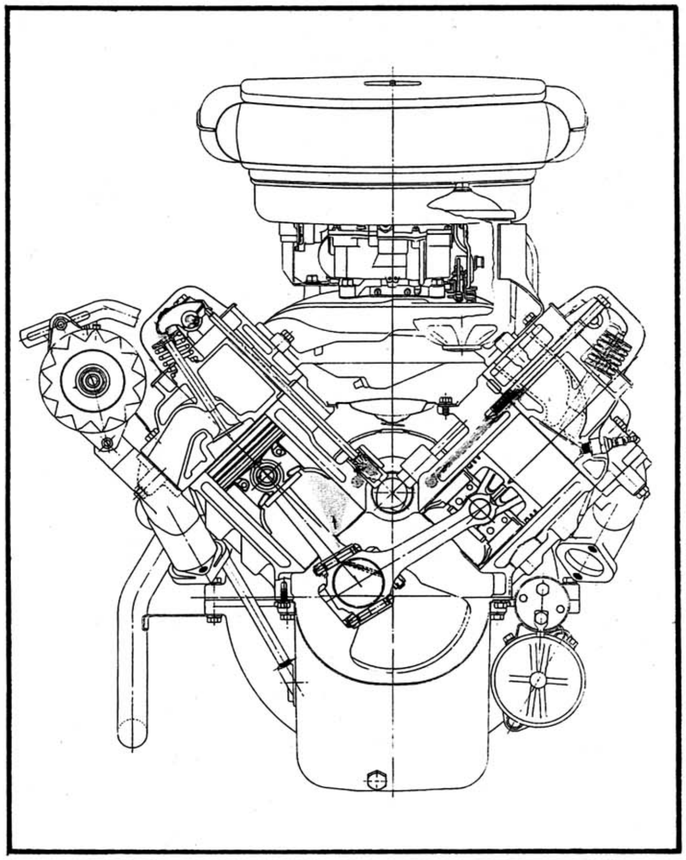
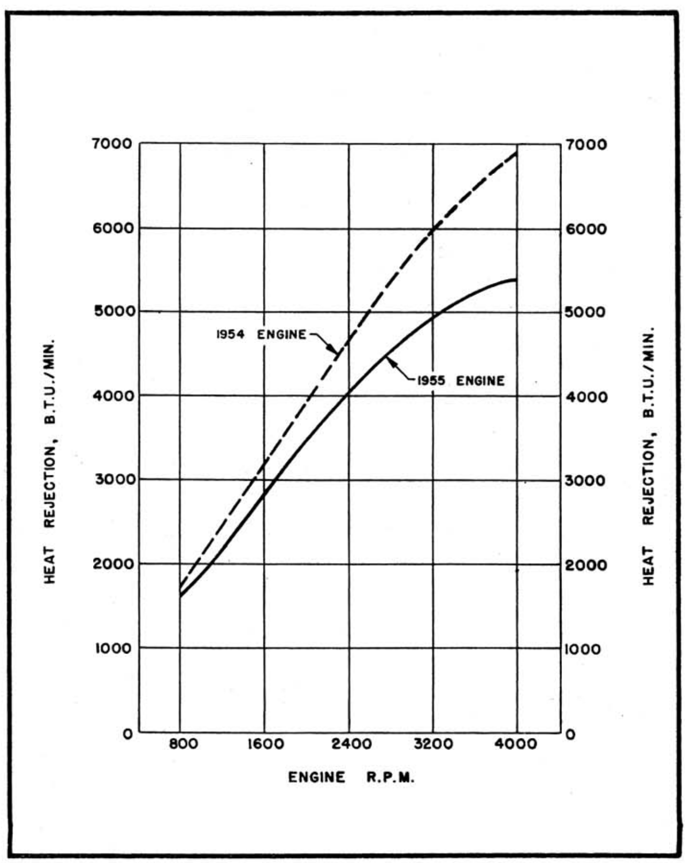


Fig. 23—Lubrication of Cylinder Walls and Valve Train

Fig. 24—Comparison of Full-Load Heat Rejection



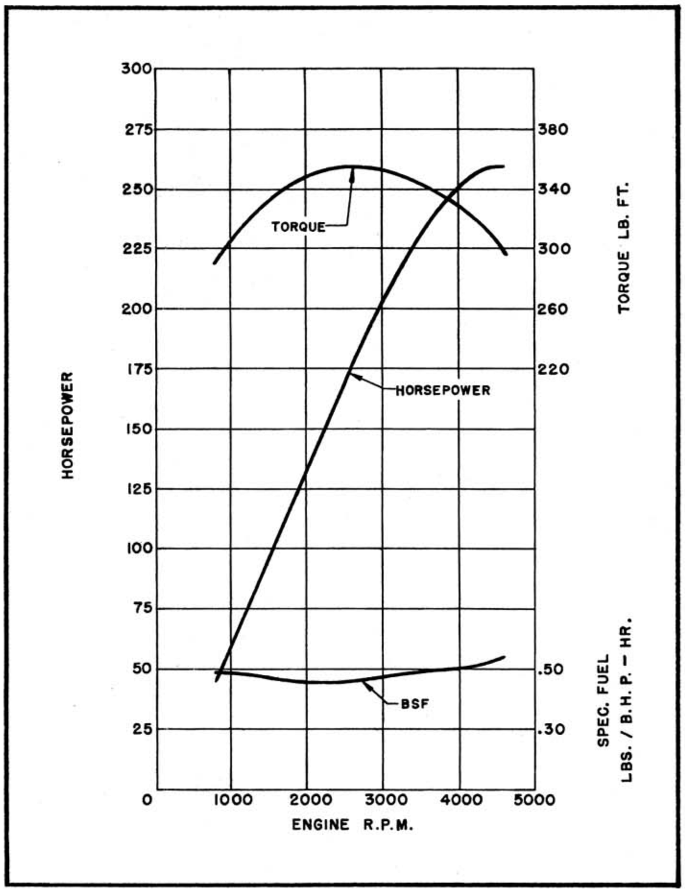
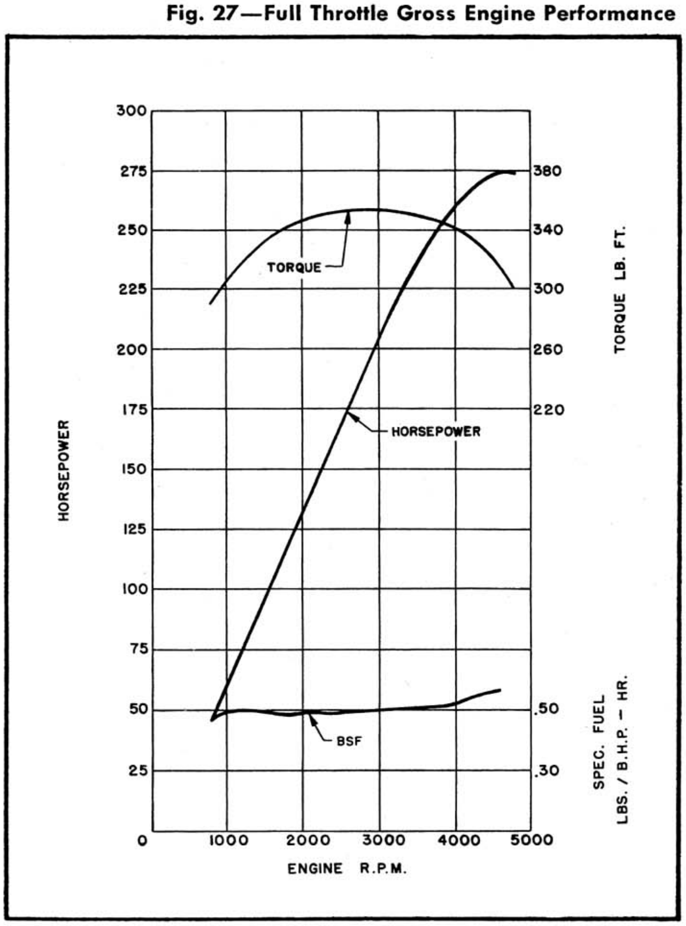


Fig. 26—Full Throttle Gross Engine Performance

Fig. 27—Full Throttle Gross Engine Performance



Adoption of the 12-volt system offered the following major benefits: (1) Increased available voltage, providing an adequate reserve to fire the spark plugs under all driving conditions, and permitting greater gap growth before servicing is required; (2) higher generator output to meet the ever increasing loads, particularly at slow engine speeds; and (3) improved cranking motor performance to satisfy the cranking requirements under both cold and hot weather starting and operating conditions.

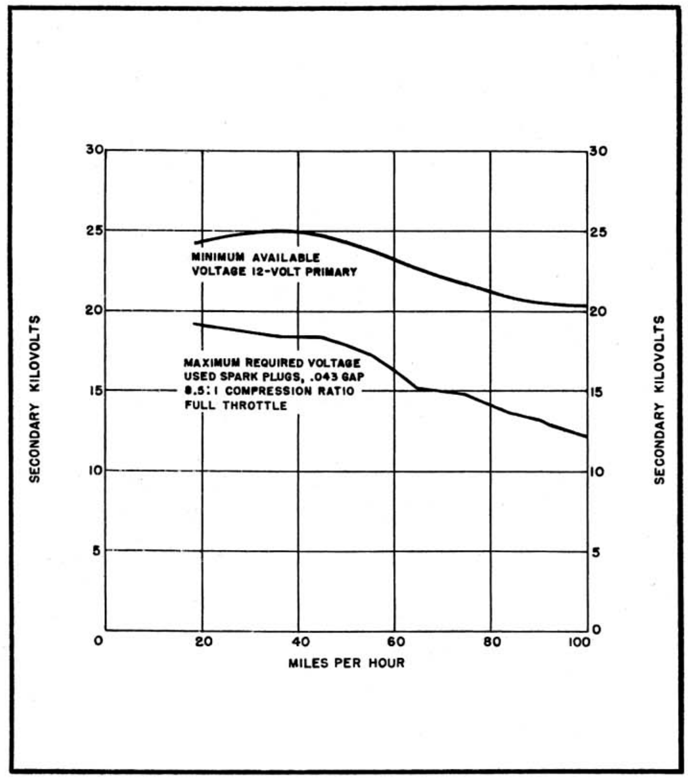


Fig. 25—Full Electrical Load Comparison of Secondary Voltages

PERFORMANCE:

Considerable improvement in the performance characteristics of the new engine was obtained as a result of continuing development since the inception of its design. Through close coordination of the design, development and testing, the outstanding performance now offered by this engine has been realized.

Figure 26 shows the full throttle gross performance of the new Packard V-8 engine equipped with a single 4-barrel carburetor. The gross performance of the engine is based on the test results obtained with manual fuel and optimum spark, corrected to SAE standard conditions of 29.92 inches of Hg. atmospheric pressure and 60° F. air temperature.

The new V-8 develops 260 bhp and 355 lbs.-ft. torque, whereas the L-head straight eight developed 212 bhp and 330 lbs.-ft. torque. Although a substantial gain in horsepower has been obtained, the really significant achievement is the improvement in effective torque, which, after all, means the translation of engine output into actual car performance. For the new engine, the maximum specific output is .739 bhp/cu. in. of displacement, and the maximum bmep is 152.1 lbs./cu. in. at 2600 rpm. The friction horsepower has been reduced from 76 at 4000 rpm for the 1954 engine to 54 at the same speed for the 1955 engine, giving an improvement of 29%. The specific fuel consumption of the new engine is lower over the entire speed range, reaching a minimum of .452 lbs./bhp-hr.

For still greater output, the new engine equipped with dual 4-barrel carburetors is available in the Packard Caribbean models. As shown by Figure 27, this combination develops 275 bhp at 4800 rpm. The torque remains unchanged at a maximum of 355 lbs.-ft., although it is available over a broader speed range, namely, 2400 to 3200 rpm. Both engine friction and specific fuel consumption are slightly higher, however.

DURABILITY:

From the inception of the design, outstanding durability was considered a basic requisite for the new engine. This objective was achieved primarily through the somewhat more conservative criteria used in designing the cylinder block and cylinder head, thereby providing exceptionally generous cooling for the critical regions in the engine. Rather than satisfy only minimum requirements, the contrary approach was made in an effort to achieve durability and life hitherto unknown for passenger car application. During the development of this engine, thorough investigations and tests were undertaken to prove out the durability of the components, both in the laboratory and on the road. Information and data obtained from this work resulted in various changes in design to provide still greater improvement. Subsequent testing indicated that the new engine was capable of successfully completing a full-throttle, high-speed dynamometer test of 250 hours duration. Based upon these unusually favorable results, the next step was to determine the engine life on the road. As demonstrated during the recently completed 25,000-mile endurance run on the 2½-mile track at the Packard Proving Grounds, which was supervised by the American Automobile Association Contest Board, the new engine offers greatly improved performance and exceptional durability. According to the AAA certification, the prototype 1955 Packard V-8 powered sedan averaged 104.737 mph for the 25,000-mile run, including all stops, thereby surpassing every recognized stock car record.

ACKNOWLEDGMENTS:

Since work was first initiated on the new Packard V-8 engine, various individuals have been responsible for making a significant contribution to its design and development. Naturally, any undertaking as important to management as the design and development of a new engine, and the large expenditures required for its manufacture, involves the team play of many people in a large organization. Although a number of men might be mentioned for the value of their work, the two who most certainly deserve special recognition for their invaluable contributions are Messrs. J. R. Ferguson, retiring Chief Engineer, and E. A. Weiss, Assistant Chief Engineer, for the Packard Division of the Studebaker-Packard Corporation.

SUMMARY:

After the original objectives were established, work was initiated in the fall of 1949 and thousands of hours have since been expended on the design and development of the new Packard V-8 engine, Figure 28. Prior to production, experimentally built engines were exhaustively tested, not only on dynamometers in the laboratory, but also in cars on the road at the Proving Grounds and elsewhere throughout the country. Since the initiation of production, testing has continued, of course, on production built engines. How successfully the new Packard V-8 engine meets the original objectives is summarized, as follows:

a. Basic Design: Exceptionally free breathing through painstakingly designed passages, and overhead valves 25% larger in head area than those used in the 1954 engine to provide maximum volumetric efficiency; and incorporates short stroke, low friction characteristics to increase mechanical and thermal efficiencies, culminating in higher performance and greater economy and longer life.

b. Physical Characteristics: New configuration offers mechanical and functional simplicity, allowing more flexibility for future styling trends and providing accessibility for servicing; and greatly increased rigidity, amply providing for future operation on improved fuels at compression ratios above 12 to 1.

c. Displacement: About the same as its predecessor, and potentially capable of being enlarged beyond the displacement obtainable with any other 1955 automotive engine, indicating extremely conservative design in anticipation of possible future requirements for still greater power and torque.

d. Combustion System: Highly turbulent, wedge-type, elliptically shaped combustion chamber, permitting knock-free operation on improved fuels at compression ratios over 12 to 1.

e. Stroke-Bore Ratio: Extremely favorable stroke-bore ratio of .875 gives a reduction in piston travel of 22% and accounts primarily for the higher power output of the 1955 Packard engine, since the displacement and compression ratio are about the same as the 1954 engine.

f. Performance: Overall performance greatly improved in comparison with the 1954 engine of essentially the same displacement and compression ratio, as follows: maximum bhp 22% and torque 7% higher, fhp 29% less at 4000 rpm, and bsfc 10% lower over the normal driving speed range—potentially capable of meeting future requirements for higher output and more efficient operation without major alteration.

g. Durability: Ruggedness and durability proven under every conceivable condition, including a 25,000-mile endurance run during which all previous stock car records were eclipsed, setting a new standard for automotive engines.

h. Producibility: Designed for manufacture by the most modern, efficient and economical methods, including automated machining, assembly and testing.

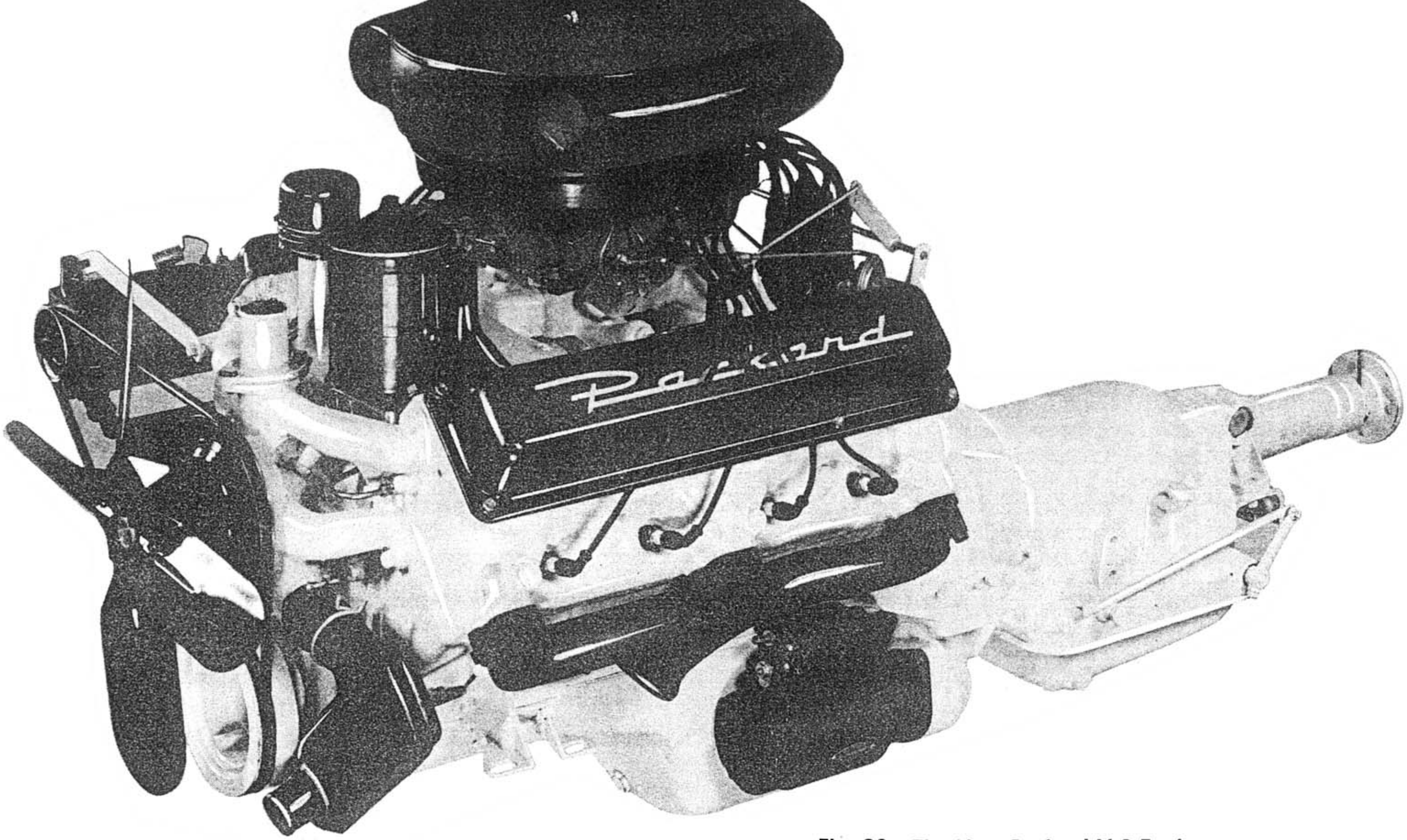


Fig. 28—The New Packard V-8 Engine