New Pontiac V-8 Engine

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THE new Pontiac engine is the culmination of nine years of design and development work. In 1946, it became evident that future styling requirements, coupled with the prospects for improved fuels, necessitated the eventual introduction of a more compact, more rigid engine, and an engineering program was initiated with those goals in mind. Other general design objectives established were:

1. Outstanding performance with assured adaptability to future fuels.
2. Proved durability, equaling or excelling its predecessor in all respects.
3. Ready adaptability to displacement increases without major tooling changes and with no compromise to engineering function.
4. Overall simplicity of design for complete ease of manufacturing and convenient servicing.

It is the purpose of this paper to show how these objectives have been realized in the overhead valve V-8 engine introduced in the 1955 Pontiac.

In conformance with the desire for simplicity of design, much attention was given to the effect of fundamental engine arrangement upon the location of all engine components. The general arrangement of the engine as evolved is shown in Fig. 1. The entire exhaust system, including the manifolds and connecting pipe, has been kept low to assure easy access to the spark plugs from the top of the engine, and also to minimize heating of the carburetor air. The generator has been placed on top of the engine, between the cylinder heads, where it is effectively separated from exhaust heat and readily accessible for servicing. This position has been attained with the simplicity of a single belt to drive the generator, water pump, and fan. Also evident in this photograph is the optional full-flow oil filter, which is in a vertical position for ease of servicing without oil spilling.

Fig. 2 illustrates the effect of cylinder bank arrangement upon the fuel pump and distributor drives. The combination fuel and vacuum pump is mounted in the low forward position on the left-hand side, resulting in minimum vapor lock tendency and optimum fuel line connection. The positioning of the right-hand bank of cylinders in the forward position made it possible to place the pump in this corner where it is close to the centerline of the engine, and where it can be driven directly by the eccentric on the camshaft. With the cylinder arrangement chosen, it was also possible to locate the distributor gear on the right-hand side (on the left in the illustration) of the camshaft without cylinder block complication. This location results

PONTIAC'S new V-8 engine, the authors say, fulfills well design objectives of durability, simplicity of manufacture and servicing, and ready adaptability to future fuels and displacement increases.

Important new features include:

1. Simplified casting of cylinder block and cylinder heads.
2. Valve train components designed for long valve gear life, quiet operation, and low cost.
3. A new gusher type of cooling system.
4. An engine ventilation system that scavenges harmful, corrosive diluents from all engine compartments.
5. A more-than-ample 12-v electrical system.
6. A wedge-shaped combustion chamber.

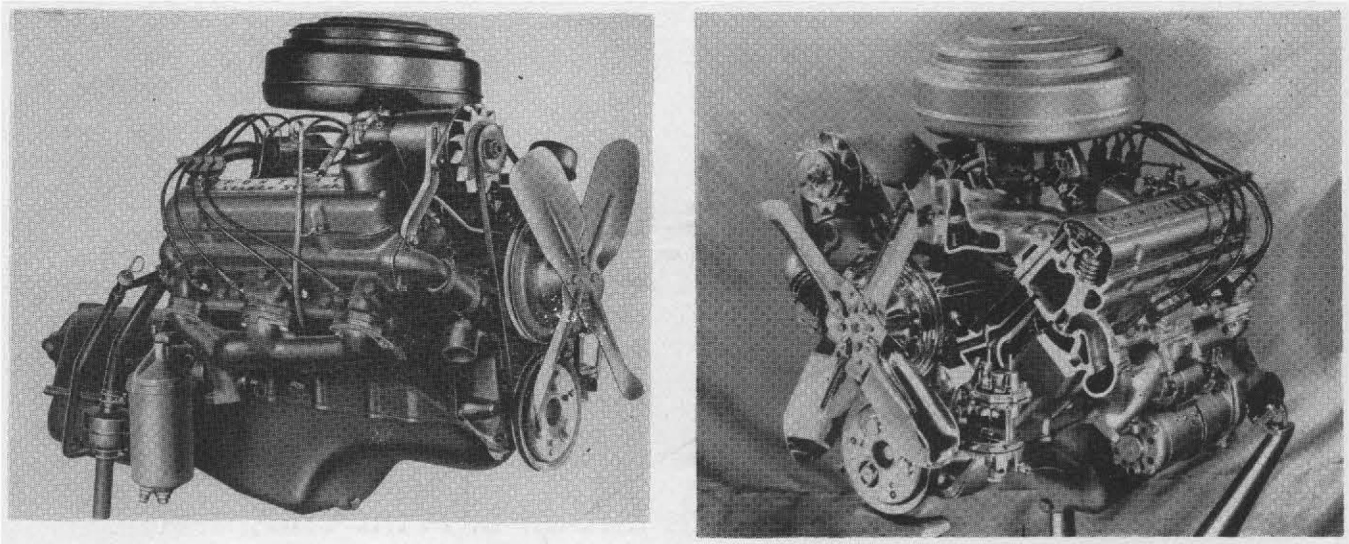

Fig. 1 – General arrangement of new Pontiac V-8 engine

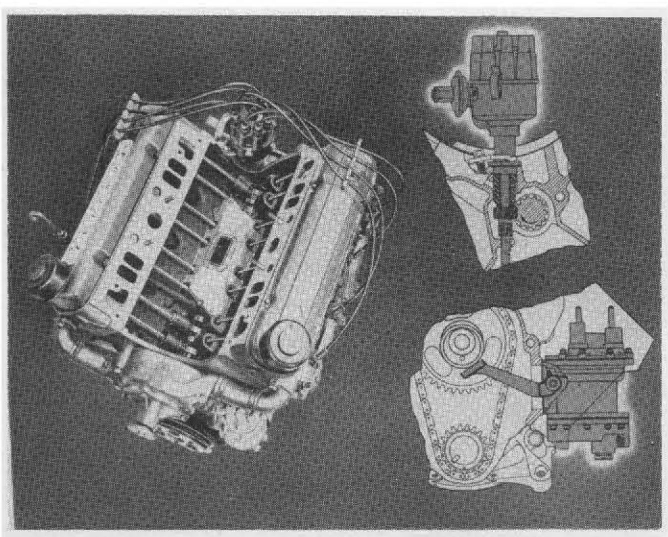

Fig. 2 – Effect of cylinder-bank arrangement on fuel pump and distributor drives

in an upward thrust on the driven gear, and makes a separate thrust surface in the block unnecessary.

Description of Engine

A transverse cross-section of the engine is shown in Fig. 3. The combination of 3.75 in. diameter bore and 3.25 in. stroke was helpful in attaining minimum external dimensions for the 287 cu in. displacement. In addition, the large bore made possible ample valve sizes, and the short stroke resulted in low friction characteristics. The aluminum alloy, steel strut pistons of slipper-type design are tin plated for proper break-in. Three piston rings are employed; the top compression ring is heavily chrome plated (0.004-0.007 in. thick) for durability, the second ring is lubrite coated, while the oil ring is of four-piece, chrome-plated, steel rail design for effective oil control. The four $\frac{1}{2}$-in. bolts per cylinder tie directly into the vertical walls of the block, and are separated from the cylinder walls to assure complete freedom from distortion due to tightening.

The curved sealing surface for the engine top cover made it possible to lower the exhaust heat passage of the intake manifold into this area between the cylinder banks and thereby effect a reduction of engine height. The distance from the centerline of the crankshaft to the top surface of the intake manifold is only 13.9 in. This, combined with a carburetor height of only 5 in., results in a very satisfactory value for the important engine height dimension.

The longitudinal section of the engine (Fig. 4) shows other details of engine construction. The inherent structural rigidity of the cylinder block made possible the placement of the oil pan flange at the centerline of crankshaft. The slanting surface of the cast timing chain cover furnishes a flat sealing contact for the oil pan gasket. This construction also permits the use of a thrust-type seal at the front of the crankshaft. At the rear, a slinger and rope-type seal are employed to prevent oil leakage. The crankshaft has main bearing journal diameters of 2.5 in. and crankpin diameters of 2.25 in., resulting in an overlap of 0.76 in. Its natural frequency of torsional vibration is 350 cps. The five main bearings are supported by rigid bearing caps, doweled in position, and attached to the ribbed bulkheads of the cylinder block.

The mounting surface for the clutch housing is shown in Fig. 5. All of the six attaching screws, including the lower two, enter rigid portions of the cylinder block. The large span between the screws in both the vertical and horizontal directions makes possible a firm support for the clutch housing and transmission. Notice that the two upper screws have been raised approximately 2 in. above the flywheel diameter to increase the important vertical span. This method of attachment provides ample

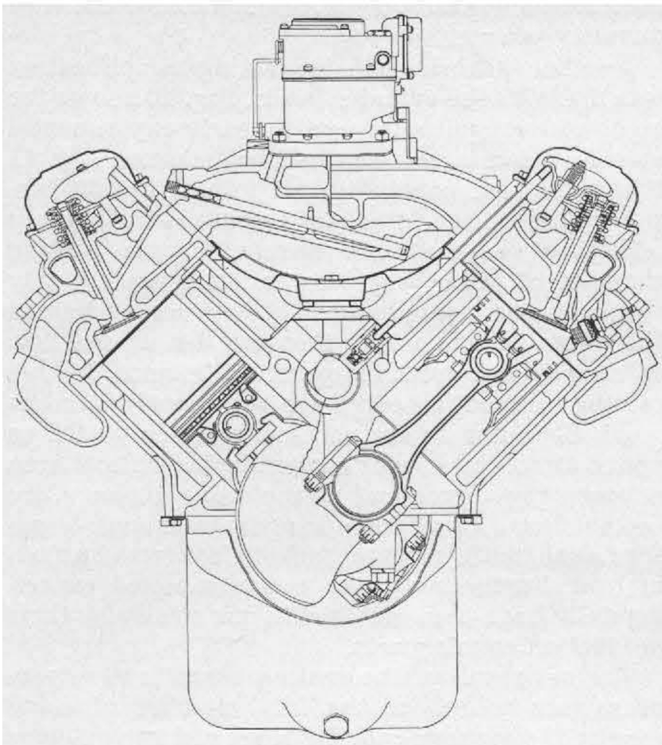

Fig. 3 – Transverse cross-section

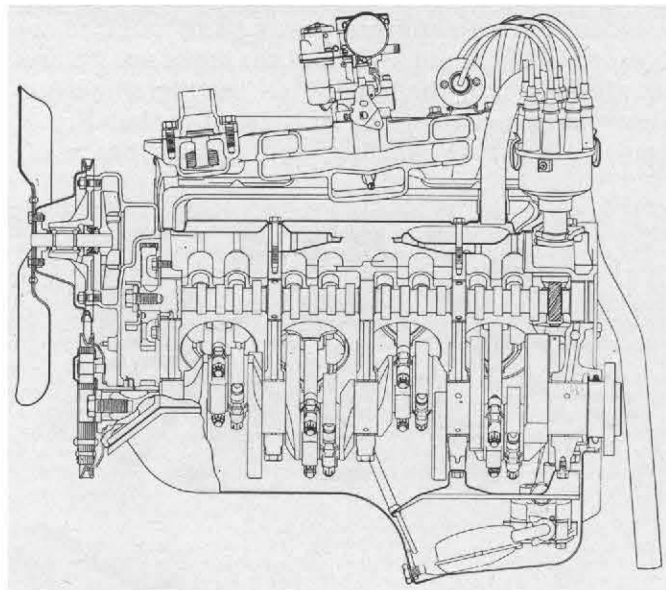

Fig. 4 – Longitudinal cross-section

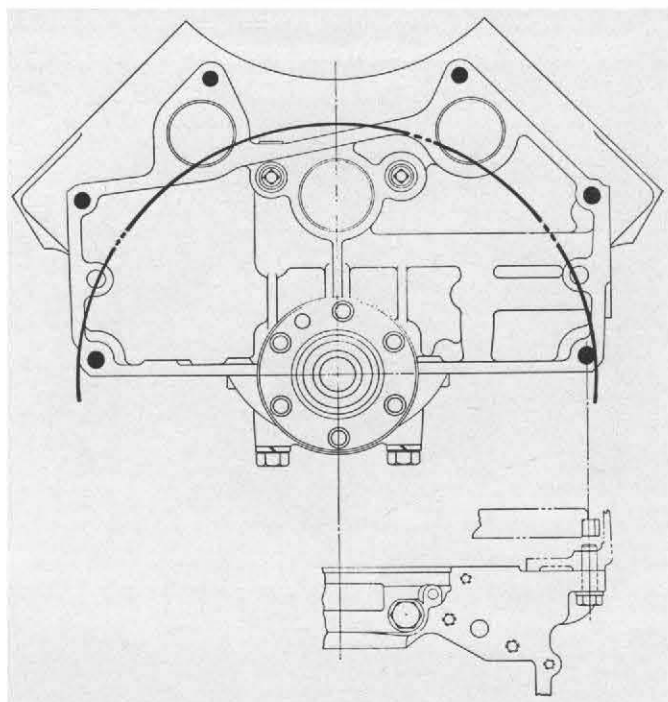

Fig. 5 – Mounting surface for clutch housing

rigidity at the transmission, and no resonant vibration is encountered between drive line components throughout the engine operating speed range.

Casting Design

An exceptionally important part of the engineering history of the Pontiac V-8 is the simplification of the casting of the cylinder block and cylinder heads. It is well known that these units are among the most complex items cast by mass production methods. However, because of the many diverse requirements which must be considered during the formative stages of engine design, the tendency is to design for conventional casting procedures.

The 18 cores required to cast an early experimental cylinder block in the conventional manner are shown in Fig. 6. In contrast (Fig. 7), only eight cores are required to cast the block as released for production. This reduction in cores was made possible by careful integration of a new casting concept into the design of the engine.

As shown in the transverse cross-section through the cylinder block mold (Fig. 8), there are no flanges which project into the center area at the top of the cylinder block. This makes it possible to use green sand to form the entire top surface. Also significant is the fact that the barrel cores, which form the cylinders, are an integral part of the crankcase cores.

The longitudinal section through the mold (Fig. 9) shows more details of the casting method. The dry sand core at the rear of the block is the only

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core required to form an external surface. Separate core assemblies requiring pasting and nailing have been virtually eliminated, and the cores are fixture-set directly into the mold. This casting simplification was found to result in reduced probability of core shift during pouring, and in turn has made possible the lowest weight castings consistent with quantity production.

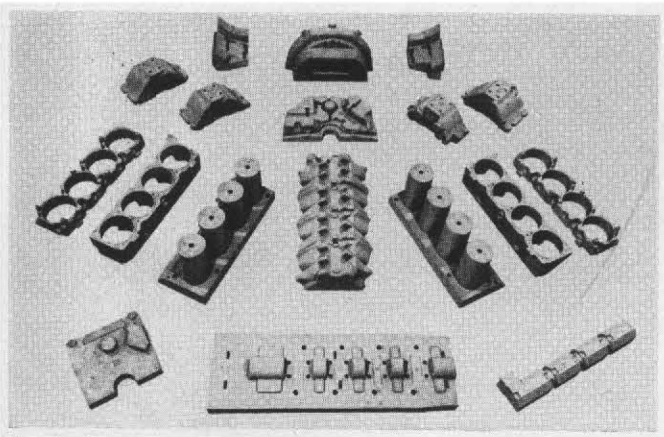

Fig. 6 – Eighteen cores required to cast experimental cylinder block in conventional manner

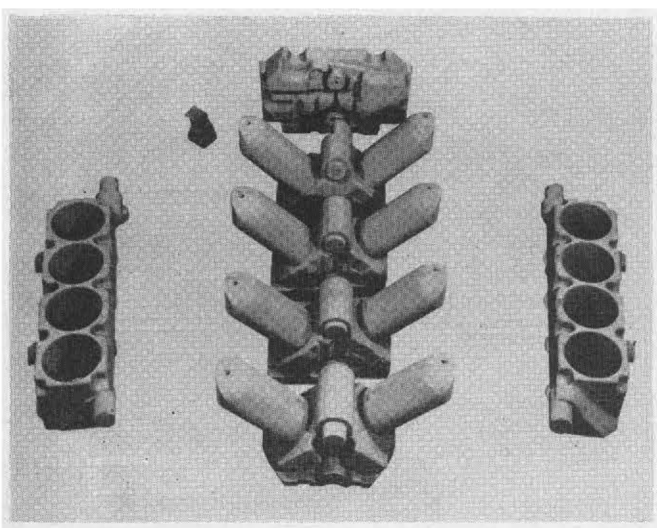

Fig. 7 – Only eight cores required to cast production cylinder block

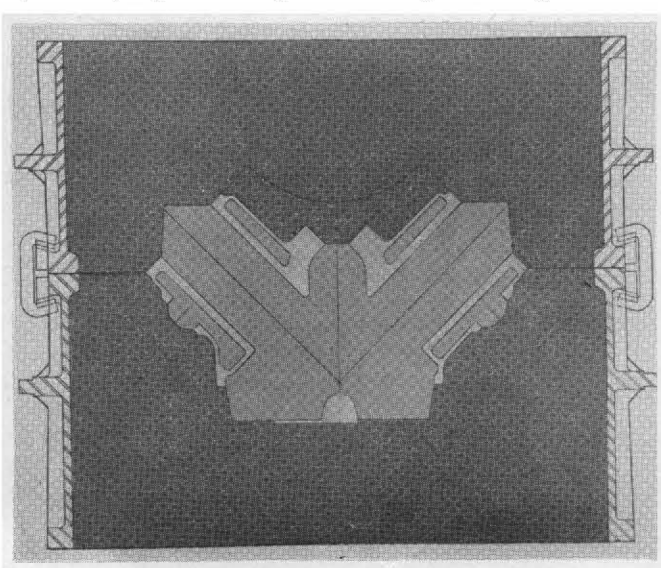

Fig. 8 – Transverse cross-section through cylinder-block mold

Another achievement in casting simplification was made on the cylinder heads. Fig. 10 shows the eight cores required to cast an early experimental cylinder head in the conventional manner. Fig. 11 shows the four cores required to cast the production cylinder head. A salient feature is the absence of separate core assemblies requiring the pasting together of water jacket cores. Such pasting may result in the unpredictable occurrence of fins in the water jackets, where they can disturb the flow of coolant and upset temperature balance. In this casting method, all cores are set directly into the mold. Such fins as are necessary are so placed as not to affect the flow of coolant in the critical area between the valves and at the valve guides. Core sections were in all cases kept large enough to assure clean, uniform water jackets free from burned-in iron. Furthermore, the castings tested experimentally have been matched in uniformity by their production counterparts.

The net result of the casting simplification program was to reduce the total number of cores required to cast one cylinder block and two cylinder heads from 34 to 16. The total weight of the dry sand cores was reduced from 320 lb to 158 lb. The ease of casting resulting from this program has resulted not only in manufacturing economies, but also in more uniform castings and a more functional design.

Valve Train

Early in the engine development program it was resolved that every effort would be made to develop valve train components which would result in long valve gear life, quiet operation, and low manufacturing costs. The type of valve train used on the new engine was designed by Pontiac in 1948, and has been successfully proved in the many engines tested since that time.

The ball pivot rocker arm is shown in Fig. 12. The ball is located on a stud which is pressed into the cylinder head. The cyanide-hardened, stamped steel rocker arm pivots on this ball and is able to

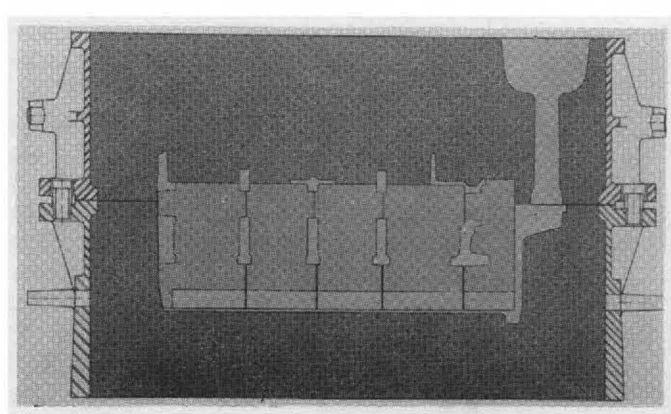

Fig. 9 – Longitudinal cross-section through cylinder-block mold

square itself on the end of the valve stem. This eliminates all misalignment of the rocker arm with the push-rod and valve stem, and minimizes the tendency toward valve cocking. As illustrated, the valve, stud, rocker arm, and push-rod all lie in a single plane for geometrically correct valve gear motion, and sidewise bending moments on the rocker arm are eliminated.

The complete valve train is shown in Fig. 13. The contact of the spherical inner surface of the rocker arm is essentially at the lower surface of the ball, and the entire depth of the rocker arm flanges is available to carry the valve gear load. Positive, pressure lubrication is provided for the ball pivot by oil entering the ball from a drilled hole in the stud which connects with the oil gallery in the head. The hollow, cyanide hardened push-rod is pressure lubricated at each end by oil from the valve lifter.

Hydraulic valve lifters were chosen for the new engine to provide the advantages they afford. It has been possible to use optimum valve timing for high engine output, while maintaining good low-speed idling without engine noise under all operating conditions. The use of dual valve springs and rigid valve gear components has resulted in a valve lifter "pump up" speed of 5000 rpm.

The use of tapered valve guides (Fig. 14) has been continued on the new engine because tests proved their advantages were even more important on the overhead valve engine. The small end of the guide inside diameter is at the top or cool end of the valve. The valves are selectively fitted to the guides to a diametral clearance of 0.0003-0.0006 in. at this point. This close clearance in combination with the shield is helpful in preventing oil flow down the intake valve guide under high vacuum conditions and has contributed to part-throttle-driving oil economy. The larger clearance at the lower end of the valve then allows for expansion of the hot valve stem. In addition, it permits a degree of stem freedom to assure proper valve seating. This self-centering freedom, combined with the reduced cocking tendency of the ball pivot rocker arms, has resulted in reduced valve stem and valve guide wear and has made possible desirable valve seating characteristics.

The intake valve head diameter is 1.781 in. and has a 30-deg seat angle. The exhaust valve diameter is 1.500 in. with a 45-deg seat angle. These valve sizes and the 0.367 in. valve lift have made an important contribution to the engine performance.

Cooling System

The desire for long valve life required low valve seat temperatures, and this consideration had an important bearing on the choice of cooling system for the new engine. Pontiac was the first to introduce the "gusher" type of cooling system to the L-head engine. This system was subsequently adopted by many L-head engine manufacturers.

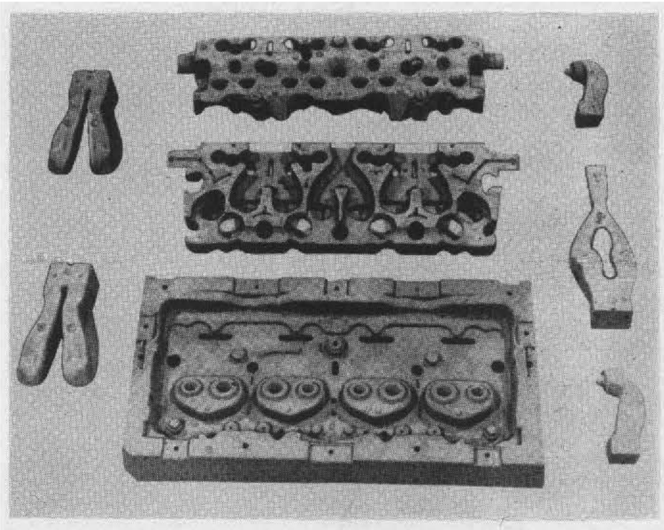

Fig. 10 – Eight cores required to cast experimental cylinder head in conventional manner

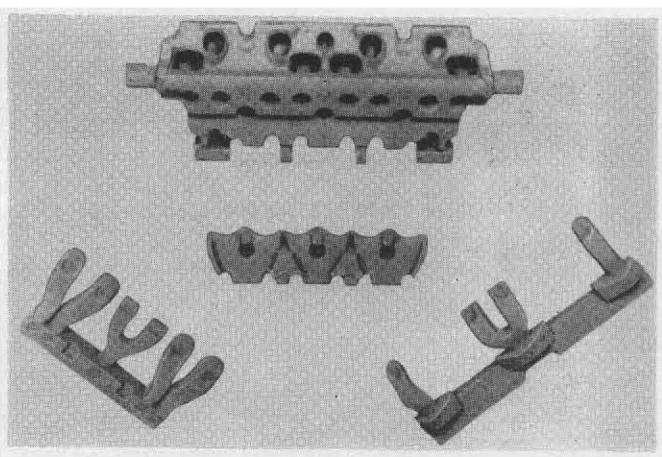

Fig. 11 – Only four cores required to cast production cylinder head

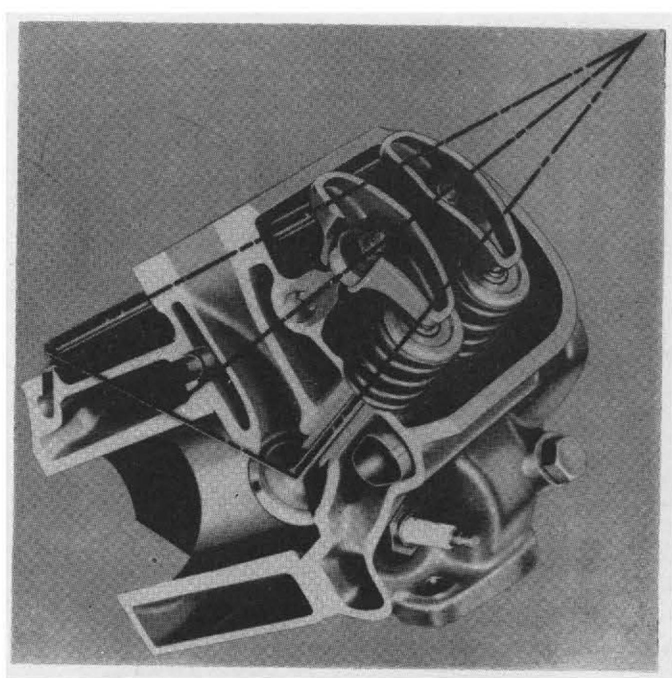

Fig. 12 – Ball pivot rocker arm

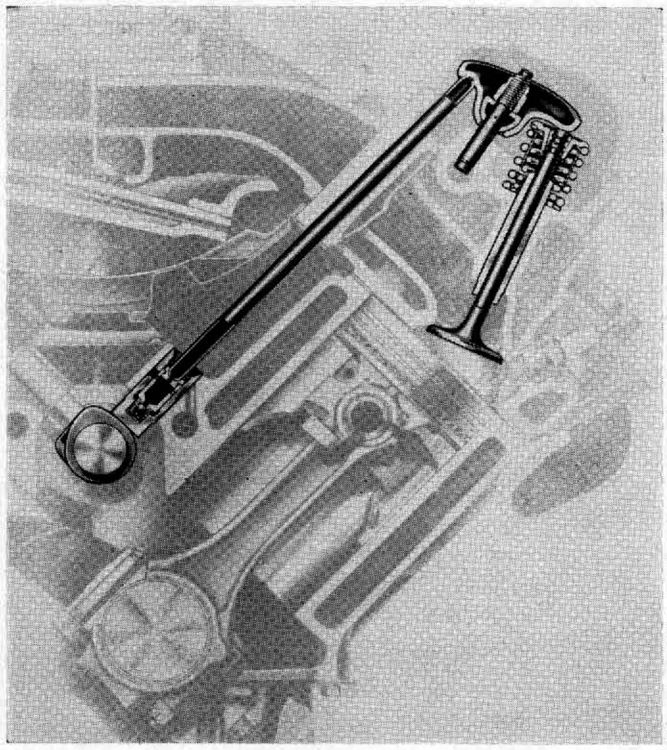

Fig. 13 – Valve train

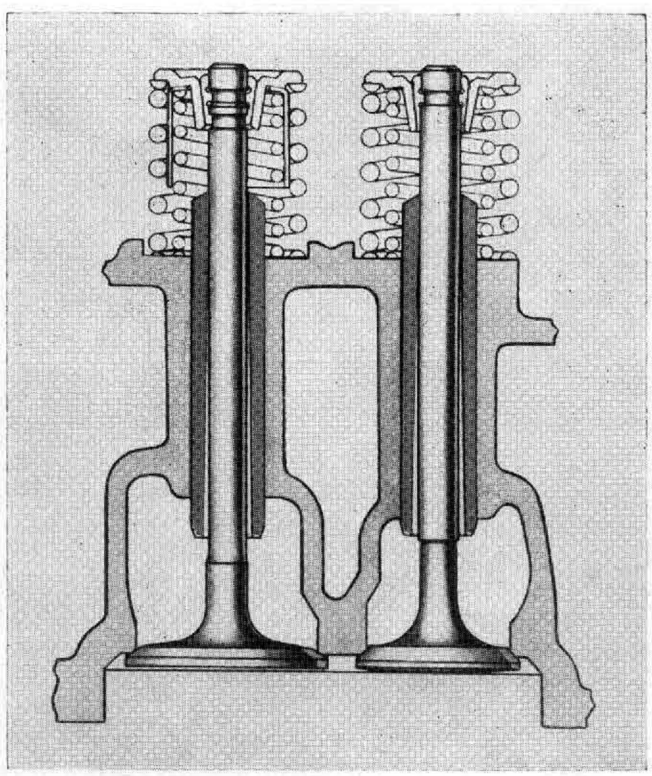

Fig. 14 – Tapered valve guides

The very satisfactory results attained therefrom have prompted the introduction of this feature on the new overhead valve V-8.

The valve cooling is shown in Fig. 15. Cold water is ejected at high velocity from holes in the distributing tube in each cylinder head. This water is directed toward the exhaust valves, where it scours the surface of the valve ports around the seats and absorbs heat therefrom. The possibility of hot, stagnant areas developing around the valve seats is thus completely avoided. In this system, the complete flow from the water pump is used to cool positively the hottest and most critical regions in the engine.

Details of the cooling system are shown in Fig. 16. The single, high-capacity water pump delivers coolant through two outlets which are connected to the front of each cylinder head. From here the water enters the brass distributing tubes which extend the full length of the heads. After the water has been ejected from the holes in the tubes, and has cooled the valve areas, a portion flows downward into the cylinder block through holes surrounding each cylinder. Therefore, no cold water is thrown directly onto any one cylinder at the expense of the others, and virtually complete temperature balance is obtained around the cylinder bores. Through recirculation holes at the front of the block, this water then enters the induction side of the pump. The water which did not enter the block from the heads returns to the radiator by way of a connecting passage at the front of the engine. This passage houses the single thermostat and is cast integral with the intake manifold.

Water flow characteristics are shown graphically in Fig. 17. The flow rate through the radiator core of the 1955 engine has been increased approximately 30% over that of the 1954 in-line engine. The total pump flow, or the amount of water directed toward the valve areas in the cylinder heads, is approximately 90% more than the flow through the radiator core. This difference between the two upper flow curves on the diagram represents the actual amount of recirculation through the cylinder block and induction side of the pump. Cooling of the cylinder block by recirculation of preheated water from the cylinder heads has resulted in water temperatures which are approximately 5 F higher in the block than in the heads. Especially important is the fact that during warmup, when the thermostat is closed, the large amount of recirculation quickly raises the temperature of the cold cylinder walls. Condensation is thus decreased, and lubrication is made more effective during this critical period of operation.

Engine Ventilation

The engine ventilation system (Fig. 18) was not merely designed as a crankcase ventilation system, but has been engineered to scavenge harmful, corrosive diluents from all compartments of the engine. Two air inlet caps, which also serve as oil filler caps, are provided. These caps are located at the front of each rocker arm cover, where they are directly in line with the fan blast. Fresh, filtered air is forced directly into the rocker arm covers, and its only exit from there is through cast openings connecting to the crankcase at each end of the cylinder heads. Therefore, a portion of the admitted air travels the entire length of the cylinder heads and effectively removes vapors which may have escaped past the valve stems. If positive ventilation is not provided here, the inner surface of the rocker arm covers becomes an ideal place for condensation of such vapors.

After ventilating these compartments, the air passes into the front and rear bulkhead compartments of the crankcase from both cylinder banks. Rotation of the crankshaft assists in thorough ventilation of this area, and also creates a pressure difference at the front which assists in the movement of air through the timing chain cover from a hole at the top of the block.

From the lower part of the crankcase, the air flows upward through the two center compartments, where it exhausts past a cast baffle in the block, into an opening in the center baffle. This opening connects to a large area, low-velocity chamber in the engine top cover. Here, oil is given an opportunity to settle out before the air enters the outlet tube. The high position of the bend in the outlet tube further encourages oil separation, and, in addition, discourages "chimneying" or draw back of dusty air during the cool-down period.

With this system, it has been possible to secure positive, pressurized airflow under all engine operating conditions, with minimum oil loss. Especially important is the fact that the double air inlets permit ample air to enter even during idling and slow-speed operation. These conditions are often encountered during engine warmup, and proper ventilation at this time has been assured.

Lubrication System

The lubrication system of the new engine is shown in Fig. 19. An oil pan baffle completely covers the sump area and minimizes aeration by oil thrown from the crankshaft. Directly under this baffle is the floating oil inlet screen which picks up oil at a controlled distance beneath the top of the oil level. Oil is delivered from the pump to the optional full-flow oil filter, and thence to the left-hand oil gallery. Main crankshaft bearings receive oil through holes drilled into this gallery. Connecting-rod bearings are fed from main bearing journals through drilled holes in the crankshaft. The camshaft journals are lubricated by vertical holes intersecting the main bearing supply holes. The right hand gallery (to the left in the diagram) receives oil from another hole intersecting the two holes at No. 1 main bearing. The two parallel oil galleries supply oil to holes in the lifter bosses, which in turn furnish oil to the valve lifters. Indexed holes in the camshaft journals supply oil at reduced pressure to each cylinder head oil gallery, from which positive lubrication is supplied to the valve train surfaces.

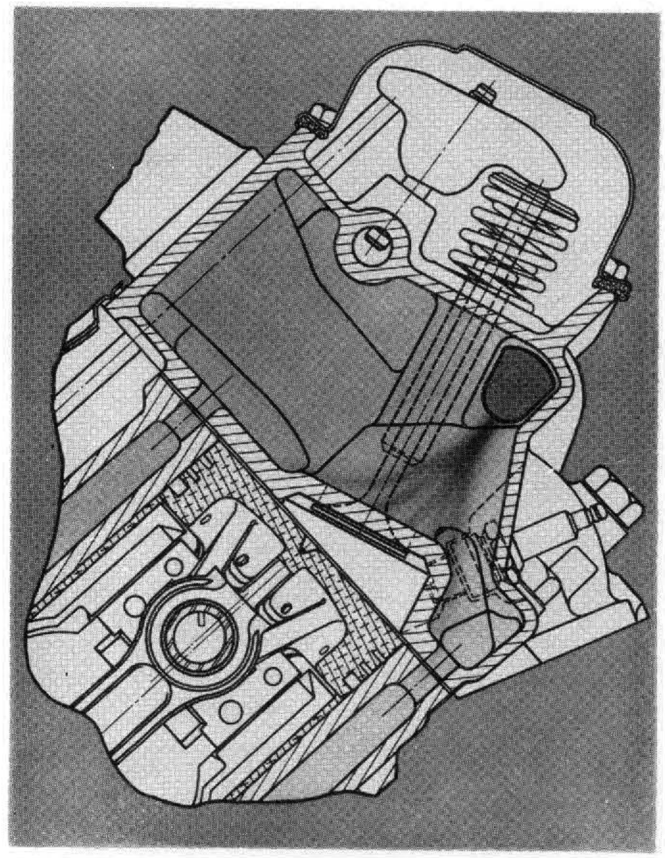

Fig. 15 – Positive valve cooling

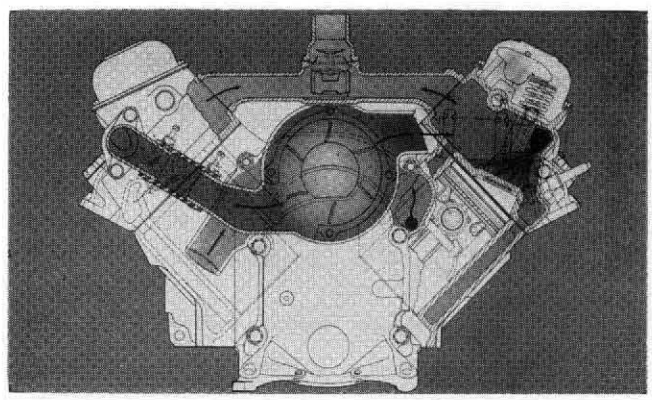

Fig. 16 – Cooling system

The distributor lubrication and oil pump drive are shown in Fig. 20. Lubrication of the distributor lower bushing and drive gear is accomplished by gravity feed. The oil returning from the right-hand cylinder head drains into a hole in the block which aligns with a hole in the distributor pilot. The overflow oil then flows past a flat on this pilot downward to lubricate the distributor drive gear.

The oil pump is driven by an intermediate shaft, which transmits the rotation of the distributor gear through a tongue and groove at each end. Thus, any slight misalignment in the drive line can be tolerated without binding. The swaged projections on the shaft prevent accidental withdrawal through the guide hole when the distributor is removed.

The oil pump utilizes coarse pitch helical gears for high capacity and quiet operation.

Oil pressure regulation (Fig. 21) has been developed to such an extent that very little variation in pressure occurs throughout the speed range. This has been accomplished with the trouble-free operation of a ball relief valve.

Cylinder wall lubrication is accomplished as shown in Fig. 22. A stream of oil is directed onto the upper side of the cylinder wall from the connecting rod of the corresponding cylinder in the opposite bank. This jet of oil is ejected through a groove across the rod at the split line of the connecting rod. This groove indexes with the oil supply hole in the crankpin. Effective lubrication is thereby provided for the cylinder walls immediately after a cold start. The oil also sprays the interior of the piston and lubricates the piston pin.

The lubrication of the timing chain and fuel pump eccentric is shown in Fig. 23. Oil from the camshaft journal is fed intermittently to a groove in the camshaft thrust plate. This groove registers with a groove in the block which directs a stream of oil downward onto the crankshaft sprocket. Positive lubrication of the hardened, stamped-steel fuel pump eccentric is effected by a hole in the camshaft thrust plate, which squirts a stream of oil forward through an indexed hole in the camshaft sprocket.

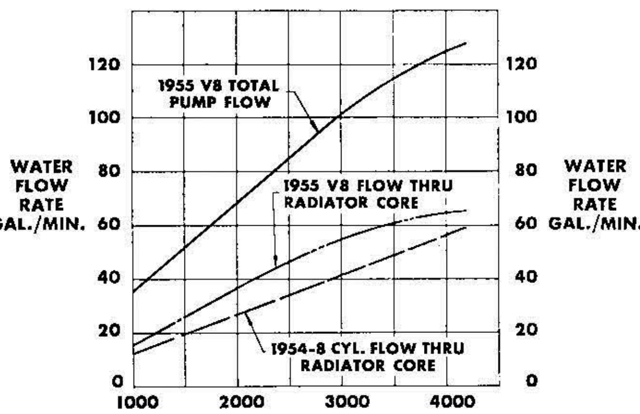

Fig. 17 – Water flow characteristics

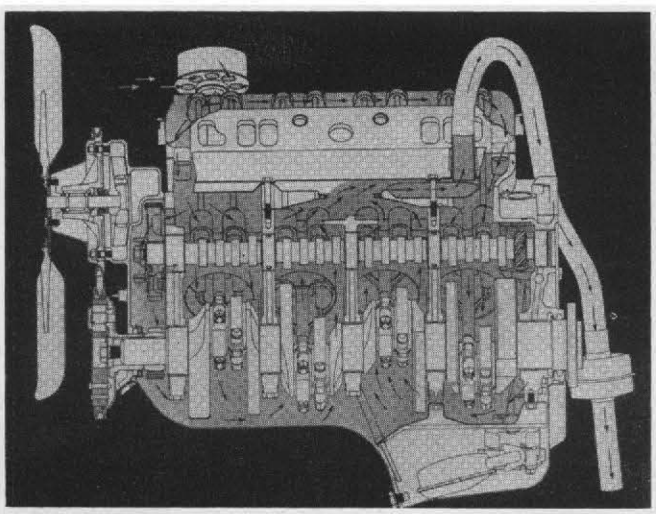

Fig. 18 – Engine ventilation system

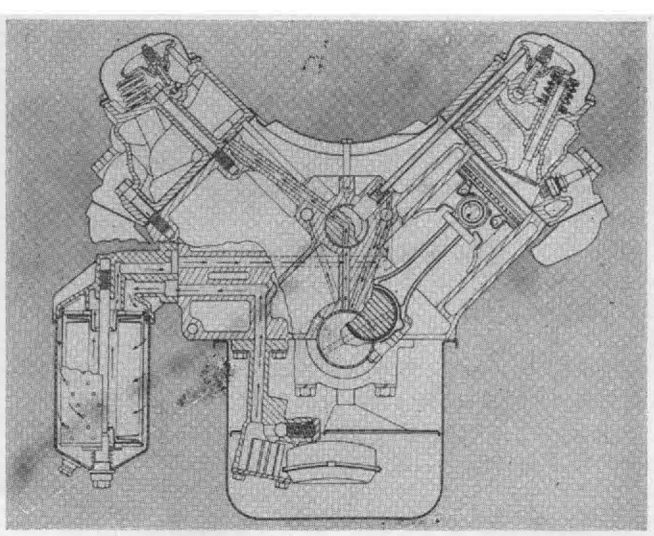

Fig. 19 – Lubrication system

Twelve-Volt Electrical System

The 12-v electrical system has been adopted for the new engine because of its recognized advantages. How well this system has met the demands of the high-output engine is shown in Fig. 24. This illustration compares the secondary voltage required to fire the spark plug with the available voltage, under the conditions shown. Note that the available voltage is always ample to assure proper

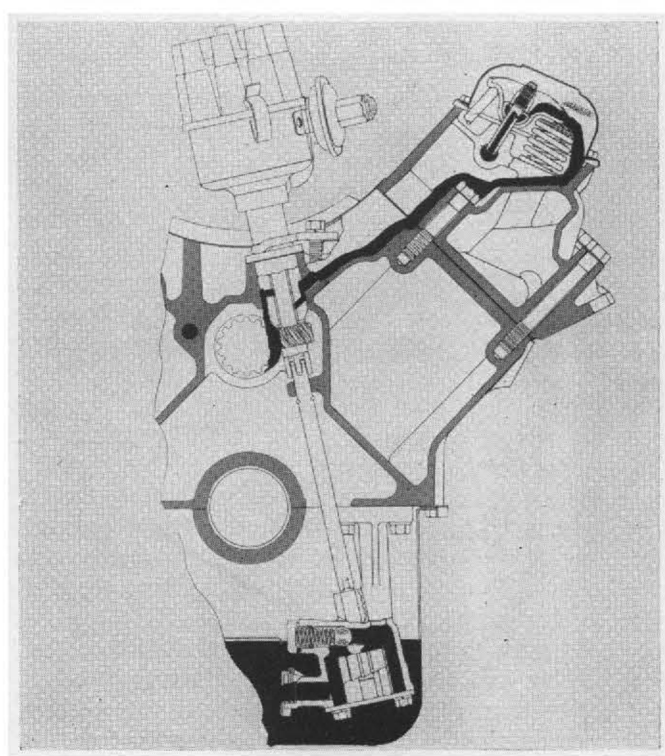

Fig. 20 – Distributor lubrication and oil pump drive

ignition of the charge.

The left-side view of the engine (Fig. 25) shows other details of the electrical system. The coil is mounted on top of the engine in a position close to the distributor. The secondary wires are unshielded, which contributes to the high voltage available at the spark plug, and are well separated to assure immunity from cross-firing. Also evident is the effect of the low position of the exhaust manifold in reducing the proximity of exhaust heat to spark plugs and secondary wiring. The separation of exhaust heat from these parts has negated any possibility of reduced life expectancy from this cause.

The solenoid-actuated starting motor is also visible here. The use of the 12-v system has made possible, at −10 F, a 16% increase in cranking speeds over a 6-v motor of the same size. Statistics concerning the generator further emphasize the advantages of the 12-v system. Power output has been increased from 270 w, developed by the 1954 6-v generator, to 300 w obtained from the new 12-v generator. This 11% increase is accomplished from a generator of the same diameter, but 1 in. shorter than the 1954 6-v model.

Intake Manifold and Carburetor Choke Stove

The intake manifold (Fig. 26) has been designed and developed to obtain optimum distribution of the fuel to all cylinders. Runners are approximately equal in length and are of ample size to match the breathing capacity of the entire induction and exhaust system. During warmup, exhaust gases are forced through the center exhaust passage under the risers to assure vaporization of the fuel. The flow of exhaust gas is regulated by the action of a thermostatically controlled valve located in the right-hand exhaust manifold.

The carburetor choke stove (Fig. 27) is shown in the cross-section through the center exhaust passage of the intake manifold. The tube pressed into this passage transfers heat from the exhaust gas to the air being supplied to the choke by engine vacuum. The use of a helical baffle inside this tube increases the heat transfer to this air, thereby providing sufficient heat to the choke to assure positive operation. In addition, other refinements in the choke itself have increased the precision with which it is made inoperative after warmup. Also evident in this diagram is the compact design of the duplex carburetor. Shown too is the de-icing passage under the idling jets. Exhaust heat is supplied here to prevent ice formation under low temperature, moist air conditions.

Combustion Chamber

The wedge-shaped combustion chamber, Fig. 28, is completely machined for accurate control of compression ratio. The thin-quench area covers 35% of the piston-head area; turbulence is thus promoted, and effective flame travel is shortened. This type of chamber was chosen because it presented the best combination of output and fuel economy of any of the many types of chambers evaluated for this engine. Tests also indicate that this wedge shape offers maximum adaptability to

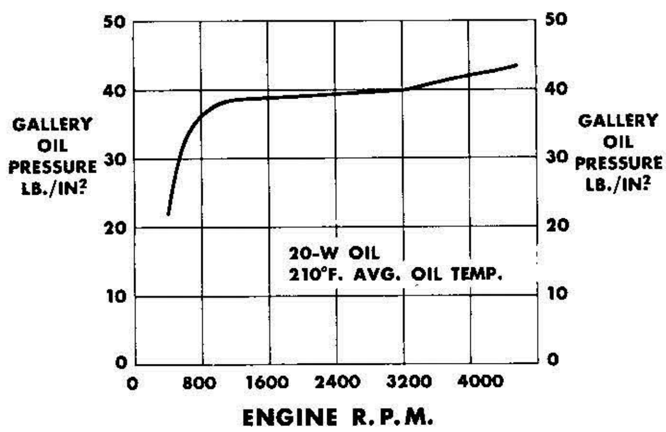

Fig. 21 – Oil pressure regulation

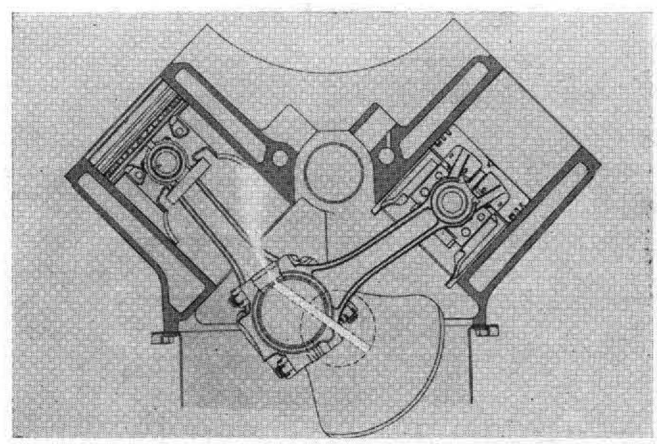

Fig. 22 – Cylinder wall lubrication

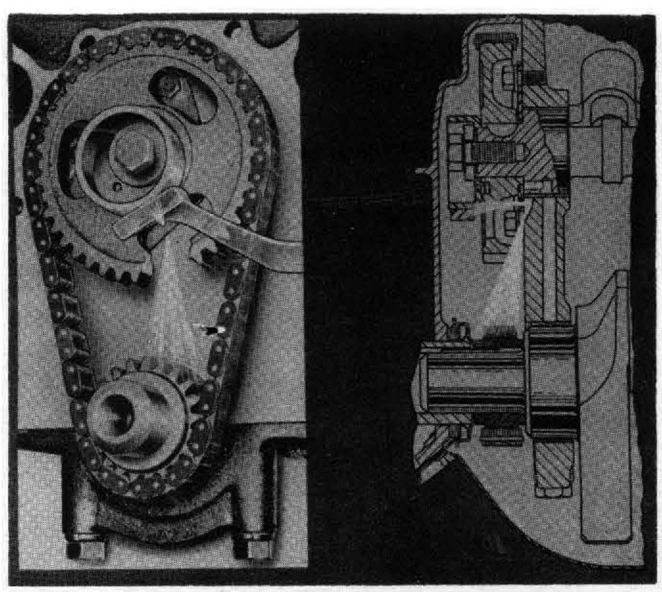

Fig. 23 – Lubrication of timing chain and fuel pump eccentric

future compression ratio increases without combustion harshness. This diagram also shows the large sections of core around the spark plug which assure positive cooling of this unit.

Fig. 29 shows the octane requirements of the 8.0:1 compression ratio engine when used with the Hydra-matic transmission. The operating conditions before evaluation of requirements are such as to encourage deposit buildup. A city-suburban schedule is driven at a speed not exceeding 50 mph at any time. An oil is used which is average in its deposit-forming tendency. The curve shows that in the test range of 3000-7500 miles from clean, 80% of the requirement observations were made without knock when using 93-Research-octane-number fuel. When 96-octane fuel was used, 100% of the requirement observations were made without knock. Since premium fuels in critical areas now have Research octane rating of from 93 to 96, the new engine is well matched to the fuel available. It is felt that a realistic compromise has been reached in the choice of compression ratio which will result in minimum annoyance from abnormal combustion of all types, under varied driving and climatic conditions, while still taking advantage of the potentialities of today's fuels.

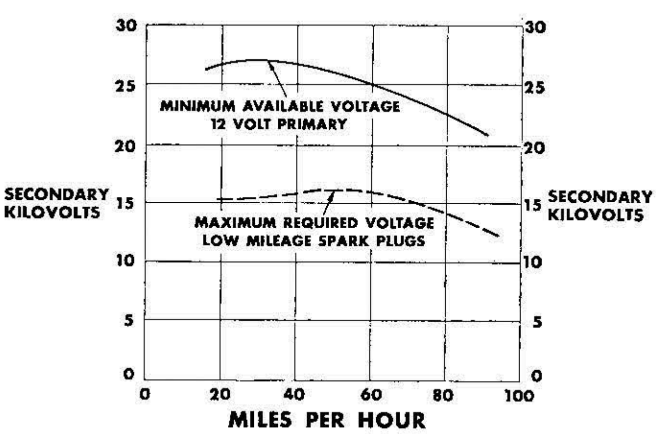

Fig. 24 – Comparison of required and available secondary voltages

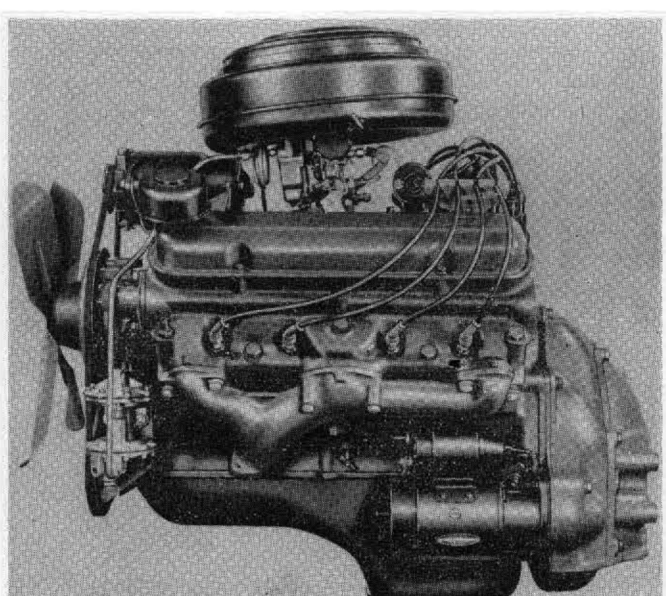

Fig. 25 – Left-side view of engine

Performance

Full-throttle performance of the Pontiac V-8 is shown in Fig. 30. The data presented are from test No. 20 of the General Motors Automotive Engine Test Code. This test is run with spark adjusted for best torque, and the results are corrected to SAE standard conditions of 29.92 in. of Hg atmospheric pressure and 60 F air temperature. As shown, a maximum full-throttle output of 180 bhp is attained at 4600 rpm. The maximum torque of 264 lb-ft at 2400 rpm is developed by a bmep of 139 psi at this speed. The maximum specific output is 0.627 bhp per cu in. of displacement. Minimum brake specific fuel consumption is 0.45 lb per bhp-hr.

Conclusion

The development and testing program made important contributions to the durability characteristics of the new engine. The magnitude of this program is reflected by statistics concerning the number of miles accumulated by experimental engines. Over 3,300,000 miles were run on road test and dynamometer installations prior to the start of production. An additional 300,000 test miles were accumulated on production-built engines be-

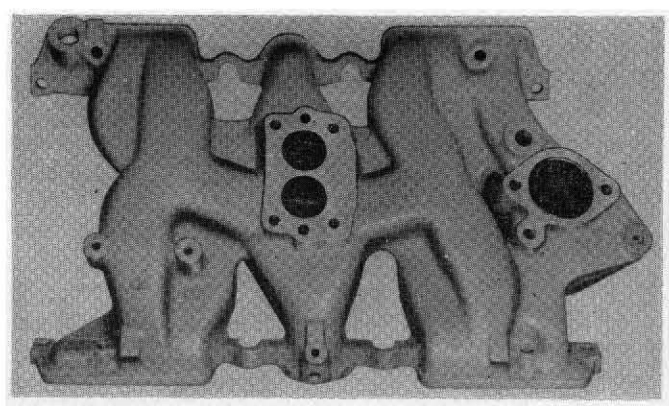

Fig. 26 – Intake manifold

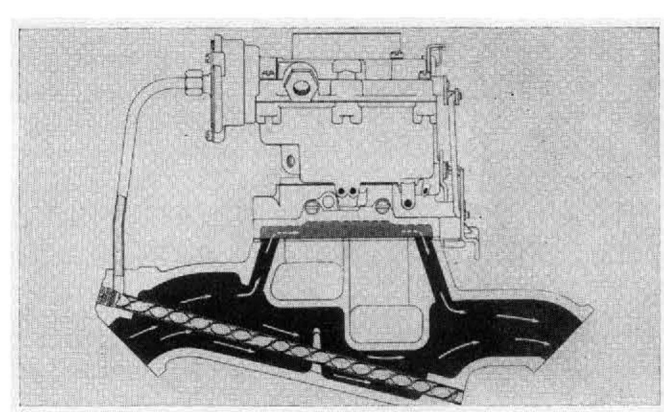

Fig. 27 – Carburetor choke stove

fore new model car assembly was initiated. The final result of this program of design, development and testing is the new Pontiac V-8 engine, which we believe represents, in a large measure, the attainment of our original design objectives.

Appendix

Bore, in.	3.75
Stroke, in.	3.25
Stroke-Bore Ratio	0.866
Displacement, cu in.	287.2
Cylinder Number	
(Front to Rear):	
Left Bank	1-3-5-7
Right Bank	2-4-6-8
Firing Order	1-8-4-3-6-5-7-2
Compression Ratio	8.0/1
Maximum Bhp	
Corrected to 60 F (at 4600 Rpm)	180
Maximum Torque, lb-ft (at 2400 Rpm)	
(Corrected to 60 F), lb-ft	264
Crankshaft:	
Weight, lb	5.80
Number of Main Bearings	5
Main Bearing Journal Diameters and Lengths of Bearings	
No. 1, 2, 3 and 4	2.4985 x 0.938
No. 5	2.4985 x 1.560
Bearing Taking Thrust	No. 4
Projected Crankshaft Bearing Area, sq in.	13.22
Connecting Rod:	
Length (Center to Center), in.	6.625
Crankpin Bearing Diameter, in.	2.2493
Crankpin Bearing Length, in.	0.881
Pistons, Rings and Pins:	
Piston Material	Aluminum alloy (tin plated)
Number of Compression Rings	2
Width of Compression Rings, in.	0.078
Width of Oil Control Ring, in.	0.186
Piston-Pin Diameter and Length, in.	0.980 x 3.02
Type of Piston Pin	Floating
Mean Piston Speed (at 4000 Rpm), fpm	2165
Camshaft:	
Material	Alloy iron
Type of Drive	Chain (3/8 in. pitch x 1 in. wide)
Valve Train:	
Valve Head Diameters	
Intake, in.	1.781
Exhaust, in.	1.500
Valve Seat Angles, deg.	
Intake	30
Exhaust	45
Valve Stem Diameter, in.	0.341
Rocker Arm Ratio (Mean)	150/1
Valve Lift, in.	0.367
Valve Spring Load, lb.	
Valve Closed—Outer	58
Inner	26
Valve Open—Outer	107
Inner	61
Valve Timing	
Intake Opens, deg btc	22
Intake Closes, deg abc	67
Exhaust opens, deg bbc	63
Exhaust Closes, deg atc	27
Mean Gas Velocity at Intake Valve (at 4000 Rpm), fpm	14,900

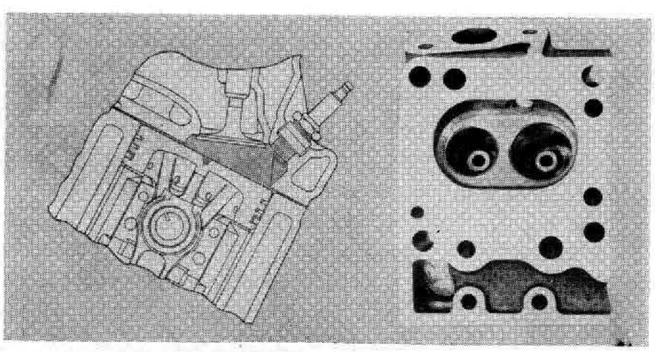

Fig. 28 – Completely machined wedge-shape combustion chamber

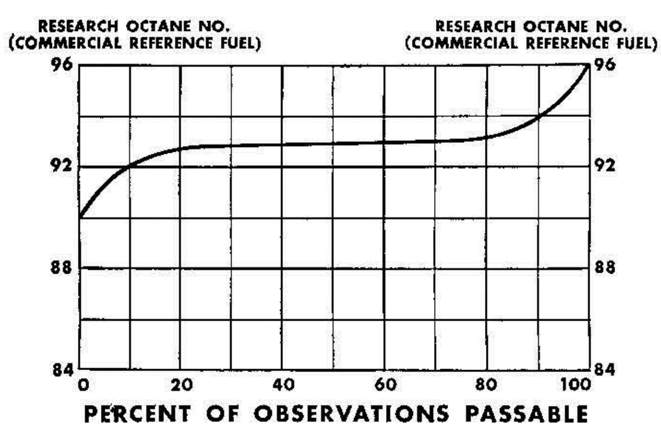

Fig. 29 – Octane requirements

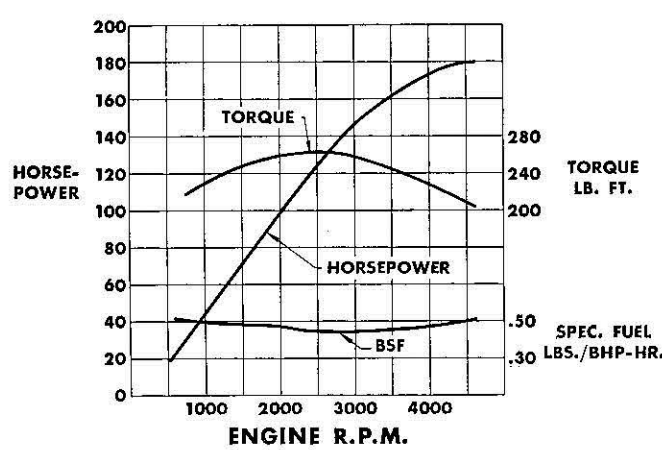

Fig. 30 – Full-throttle engine performance