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THE NEW RAMBLER SIX ENGINE —TORQUE COMMAND 232

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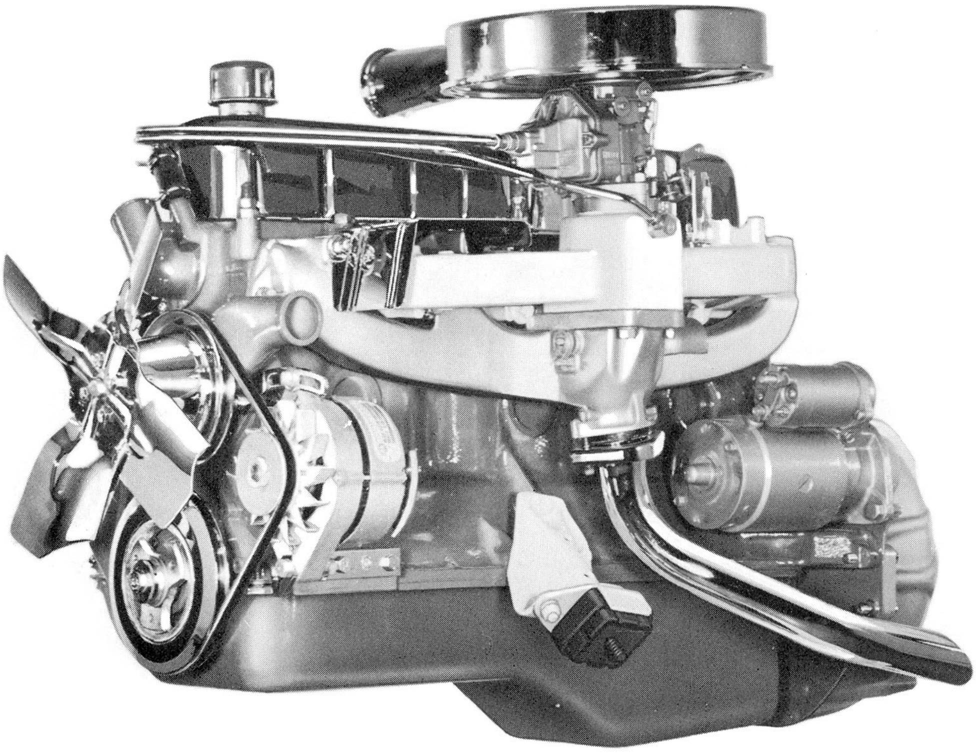
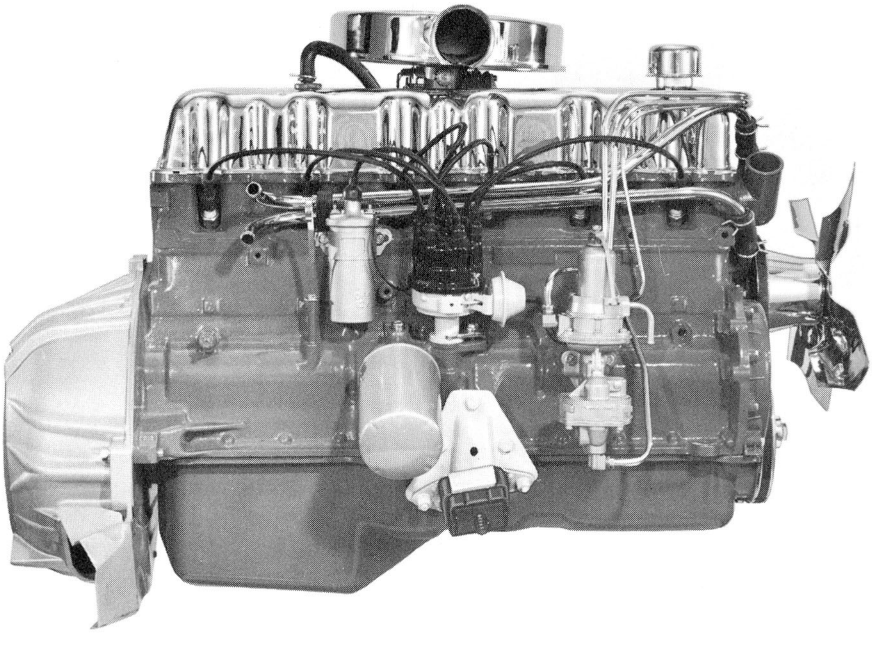




THE NEW RAMBLER SIX ENGINE— TORQUE COMMAND 232

American Motors' new "Torque Command" 232 cubic inch six-cylinder engine was recently introduced to meet progressively rising consumer standards of passenger car performance, smoothness, and durability — with emphasis on modest initial cost and economical operation.

In preliminary planning for the 232 engine program, market research studies were thoroughly analyzed and the following criteria established:

1. An exceptionally high level of reliability and durability consistent with present day long-term warranties.
2. Performance comparable to moderate displacement V-8's on the market.
3. Operating smoothness equal to that of a modern V-8.
4. Fuel economy at a level consistent with our present leadership in the field.
5. Operation on regular grade fuel, but with the capacity to accept higher compression ratios to fill future requirements.
6. Size and weight to permit flexibility of installation in the complete line of Rambler automobiles.
7. Ease of manufacturing and ease of service.

In achieving these objectives, we believed our present pre-eminent position in the six-cylinder engine field would be further enhanced by an engine having outstanding characteristics and broad consumer appeal.

This paper describes the design approach and development program which produced an engine meeting all the desired criteria, as subsequently proved in a two-million-mile test program under varying conditions.

DESIGN GOALS

A completely new engine design was envisioned, to be produced on all-new tooling. Design goals were:

1. *Performance:* 140 horsepower and 200 lb. ft. of torque, with emphasis on good "lugging" ability, for stop-and-go urban driving conditions.
2. *Quiet, vibration-free operation.*
3. *Fuel Economy:* Minimum B.S.F.C. of .45 lb./BHP-hr.
4. *Durability:* 100,000 mile design life; production engines must pass a 500 hour full load endurance run at 4000 RPM.
5. *Flexibility* in choice of displacement; no built-in obstructions to moderate increases or decreases in the future.
6. *Minimum Weight and Cost* consistent with the above five goals.

DESIGN FEATURES

After thorough evaluation of the various cylinder arrangements feasible in a six cylinder engine for front installation, the inline configuration was selected as the only one which could meet *all* of the design goals listed above. This decision was based on the results of design and test programs on both 60 and 90 degree V6 engines. In both cases, vibration from secondary couples prevented meeting our high standard of smoothness for this engine, and cost was also substantially higher than for the inline six.

The choice of a seven main bearing design was an integral part of the decision for inline construction. This permitted wide bore spacing, allowing stroke-bore ratios below unity over a wide range of useful engine displacements. Free breathing could readily be provided with the valve sizes possible in a "wedge" chamber of this diameter, as well as excellent water accessibility to valve seats and spark plug boss.

Wide bore spacing also made room for a crankshaft requiring a minimum of machining, with considerable cost savings. The short stroke reduced the crank arm and pin unbalance and thus minimized the amount (and weight) of counterweighting, as well as the counterweight outside radius.

Reduced counterweight radius, combined with short stroke, permitted a short connecting rod and resulted in a substantial saving in cylinder block height. The seven main bearing bulkheads improved crankcase stiffness, with benefits in quiet, vibration-free operation.

In sum, the benefits inherent in a large bore, short stroke, seven main bearing inline six coincided very well with our design goals as previously outlined. Added encouragement was provided by the success in past years of the "Ambassador Six" engine, which was also a seven main bearing design.

COMBUSTION CHAMBER

The combustion chamber, shown in Figure 1, is a conventional "wedge" design giving a compression

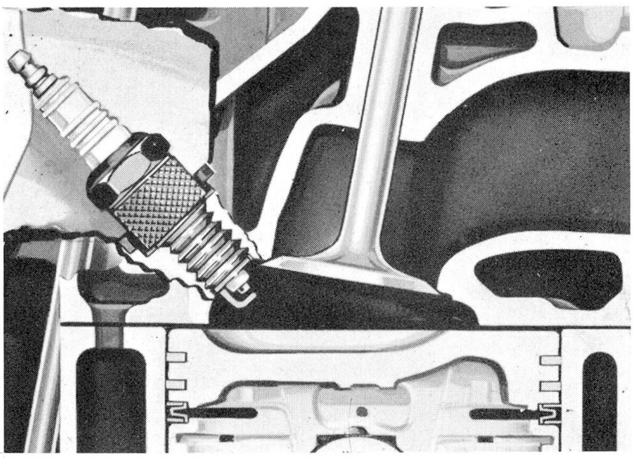


Figure 1 Combustion Chamber

ratio of 8.5 to 1. A portion of the chamber is formed by a depression cast into the piston crown. While this arrangement was adopted to provide greater freedom in "tailoring" combustion chamber volume distribution, it can also be utilized to change chamber volume for compression ratio or engine displacement changes, with no modification of the cylinder head or of piston machining.

The distribution of chamber volume with respect to spherical flame travel from the spark plug gap was checked graphically. The percentage of chamber volume consumed during each increment of spherical flame travel was plotted against percent of flame travel (see Figure 2). By tailoring the cast-in cavity

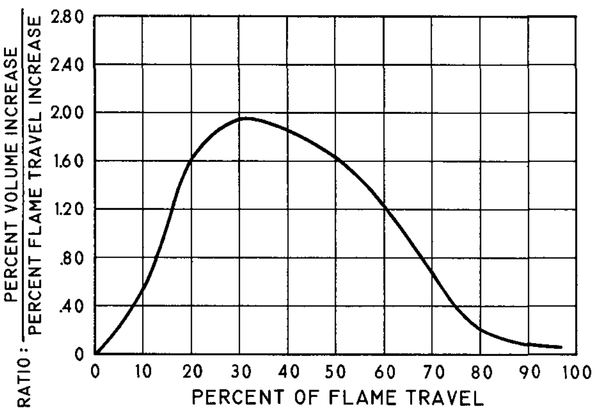


Figure 2 Combustion Chamber Volume Distribution

in the piston crown as well as combustion chamber height and contour, and making minor adjustments of spark plug gap location, combustion chamber volume was distributed in such a way as to provide rapid burning in the early stages of combustion, with a limit on maximum diagram height set by experience with previous designs, and tapering off uniformly to the end of combustion. This volume distribution, combined with turbulence from a swirl-producing intake port, resulted in measured maximum rates of pressure rise near 30 psi per degree.

Quench area equal to 22% of the bore area was provided for detonation control, with quench height held at .075 inch to promote end burning. This combustion chamber has responded well to developments connected with the control of exhaust emissions.

INDUCTION SYSTEM

A detachable intake manifold is used to permit changes in flow area for various applications and displacements. A single-barrel downdraft carburetor is standard.

VALVE TRAIN

Valve head diameters are 1.79 inches for the intake and 1.41 inches for the exhaust. A single valve spring, wound with variable pitch for damping effect, is installed. Polyacrylic rubber umbrella oil shields are provided on all valves. Cast pearlitic malleable iron rockerarms are used with solid pushrods.

Hydraulic lifters are standard equipment and contribute to quiet, maintenance-free operation. Lifter registration with the oil gallery occurs during 270 degrees of the cam's rotation, with registration interrupted during higher values of lift. Camshaft drive is by silent chain, with a sintered iron crankshaft sprocket and a die cast aluminum camshaft sprocket with molded nylon teeth.

CYLINDER HEAD

The cylinder head, illustrated in Figure 3, is a thin-wall alloy iron casting with a standard wall thickness

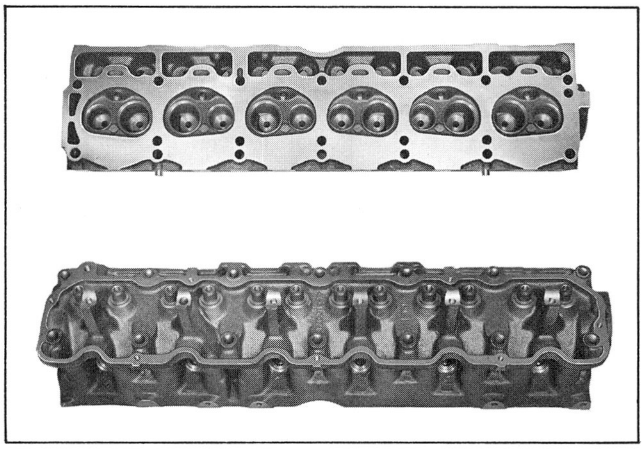


Figure 3 Cylinder Head

of .170 inch. The water jacket is cast with a two piece core. The intake and exhaust port core is one piece, as shown in Figure 4. This simplified coring assures

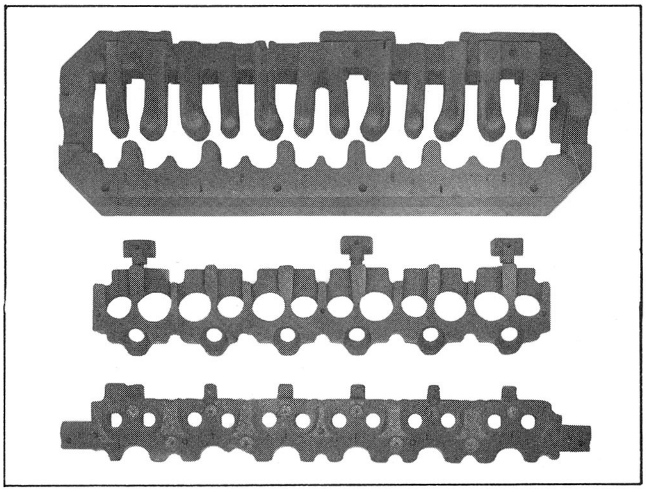


Figure 4 Cylinder Head Cores

proper wall thicknesses and high quality, uniform castings in production.

Intake ports are offset and shaped as shown in Figure 5 to produce swirl and increase turbulence in the cylinder.

Valve seats and guides are integral with the head to minimize operating temperatures. Use of a long reach, extended tip spark plug permits better cooling by reducing the spark plug boss diameter adja-

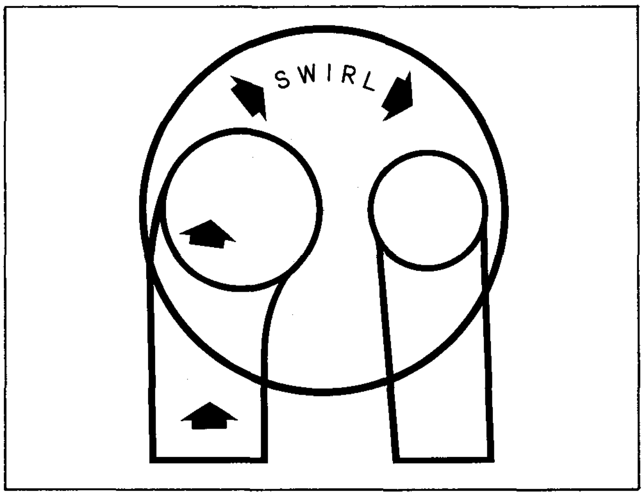


Figure 5 Port, Cylinder Bore & Swirl

cent to the chamber walls and near the exhaust valve seat area (see Figure 1). Considerable attention is given to cooling around both valve seats, as shown in Figure 6.

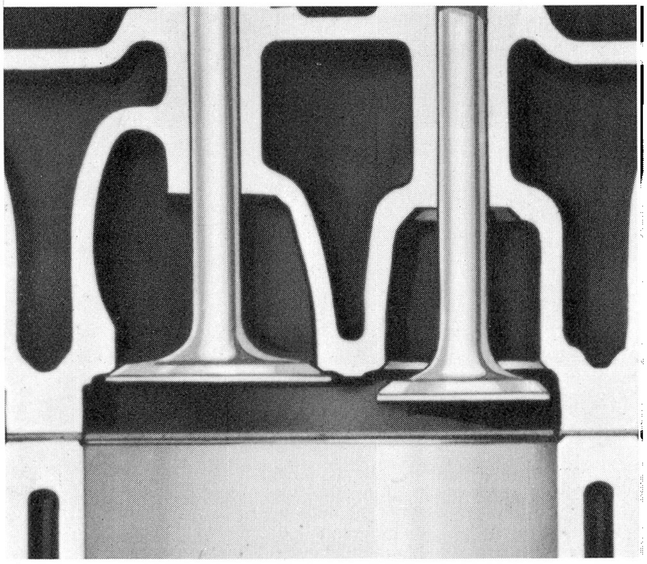


Figure 6 Valve Seat Cooling

Four ½ inch head bolts surround each combustion chamber, with torque specified at 85 lb. ft. A .015 thick embossed steel head gasket is used.

Integral rocker shaft pedestals provide positive location of the shaft, assuring proper rockerarm-to-valve geometry.

PISTON

Permanent mold, double trans-slot pistons are used, as illustrated in Figure 7.

The full-diameter skirt design, in combination with lubrication refinements mentioned later, results in excellent oil mileage. The skirt is elliptically turned, and a cast-in steel ring provides expansion control. Two extensions of the skirt on the thrust faces increase skirt bearing area and piston stability.

Generous piston pin projected bearing area of 1.62 square inches is provided by the pin currently used in our V-8 engine. The pin is pressed into the

Figure 7 Piston

connecting rod. A vertical drilling through each pin boss assures adequate lubrication.

CONNECTING ROD

The connecting rod, shown in Figure 8, is a pearlitic malleable iron casting. The center-to-center dimension is 5.875 inches.

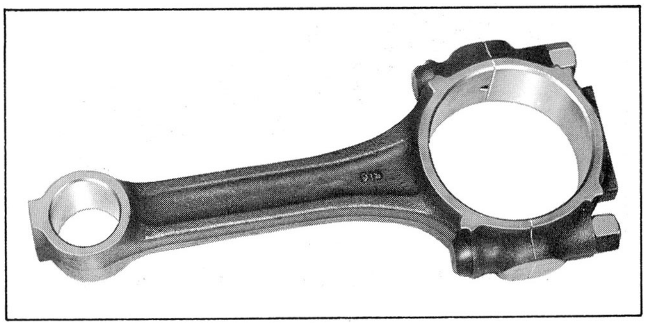


Figure 8 Connecting Rod

CRANKSHAFT AND BEARINGS

The seven main bearing crankshaft is a pearlitic malleable iron casting, which offers the advantage of better machinability. More significant, however, are the cost savings possible when bore center distance is great enough to make room for a .100 inch nominal cast clearance between the counterweight cheeks and the crankpin turning tools and grinding wheel (see Figure 9). In our case, this space is available, and we are able to realize the cost savings of non-machined counterweights. Only crankpins and main journals, nose and rear flange are machined.

The elimination of the masses of metal in the "kidneys" and center spacer (at the centers of the bays) of a four main bearing shaft, together with the short stroke, results in weight and cost savings which more than compensate for the three additional bearing shells and cylinder block bulkheads.

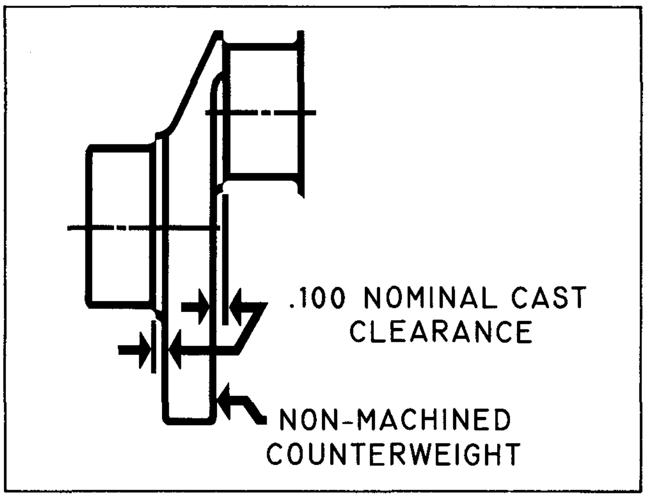


Figure 9 Cast Crank Cheek Clearance

Main journal diameters are 2.50 inches and crankpins are 2.095 inches. Main bearings are microbabbitt .98 inches long, and connecting rod bearings are copper alloy .88 inches in length.

End thrust is taken at the number three main bearing.

INITIAL CRANKSHAFT DESIGN: 4 Counterweights

Figure 10 shows the first crankshaft designed for this engine, which used four counterweights, placed so as to reduce main bearing loads below a mean level of 1200 psi, for microbabbitt bearing material. Excessive loads would occur with a non-counterweighted shaft, reaching mean pressures of 2200 psi at 4800 RPM in the center main bearing. Two counterweights adjacent to the center main reduced this load and all others below 1100 psi, and one counterweight at each end of the shaft restored dynamic balance and provided for balance drilling.

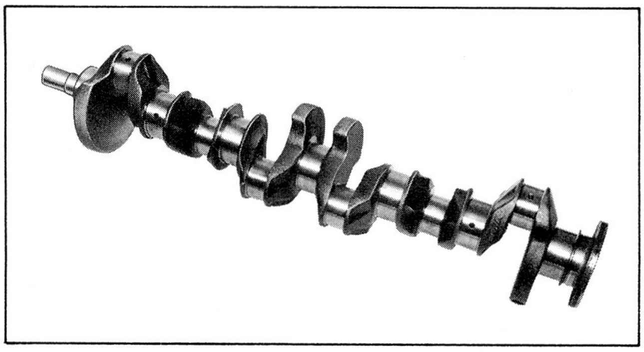


Figure 10 4 Counterweight Crankshaft

All counterweights lay approximately in a plane 34 degrees from the center-line of crankpins 1 and 6, counterclockwise when viewed from the front (see Figure 11).

This crankshaft was dynamometer tested in the early prototype engines, giving completely satisfactory journal and bearing life.

FINAL CRANKSHAFT DESIGN: 8 Counterweights.

Vehicle tests of prototype engines with the four counterweight crankshaft showed performance, economy and durability to be entirely satisfactory, but engine smoothness did not meet our goals.

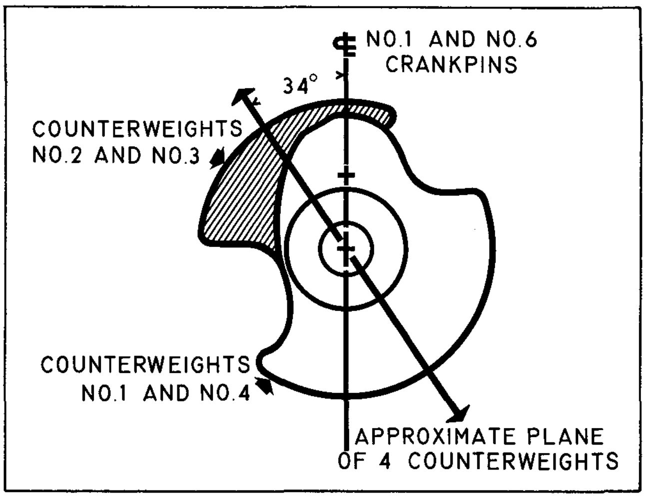


Figure 11 Counterweight Angles on 4 Counterweight Crank

Considerable design effort had been expended from the beginning to provide exceptional smoothness:

1. Crankcase rigidity was high.
2. The combustion chamber was tailored for smoothness.
3. Bearing clearances were modest.
4. Engine mounts were properly designed.
5. Flywheel inertia was well above typical values for inline six-cylinder engines of this size and output.

The next step was design and testing of a fully counterweighted crankshaft. The term "full counterweighting" as used here signifies a system of counterweights applied in such a manner that no bearing loads result when the bare shaft is spun in its bearings.

One possible procedure would have been to place a pair of counterweights straddling each crankpin and counteracting the unbalance of that crankpin and its two arms. This would have produced a shaft with twelve relatively small counterweights, being more difficult to cast and giving limited angular freedom for balance drilling in the end counterweights.

We chose another solution for this engine: full counterbalancing with eight counterweights. The counterweight angular locations are shown in Figure 12.

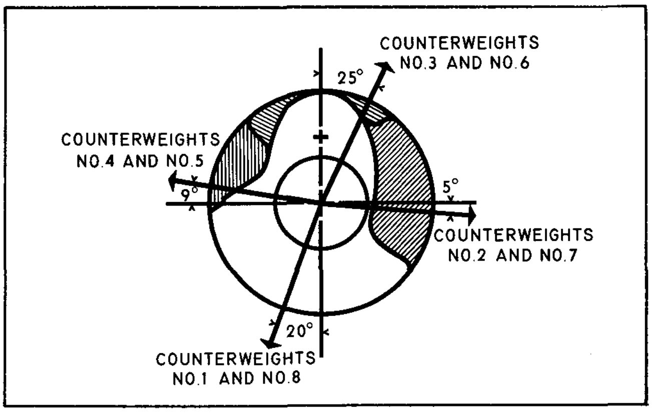


Figure 12 Counterweight Angles on 8 Counterweight Crank

These angles are approximate; exact angle and magnitude were determined by considering all crank-pin and arm unbalances tending to load main bearings 1 and 2, and tailoring counterweights 1 and 2 so that no loads occur. This procedure was repeated for bearing pairs 3 and 4, 4 and 5, and 6 and 7. The resulting crankshaft, when spun in its bearings without connecting rods and pistons attached, will produce no bearing loads, and thus fits our definition of a fully counterweighted shaft.

Of course, additional counterweighting could have been applied to compensate for the rotating portion of the connecting rod, thus further reducing bearing loads in the complete engine; but driving tests of the engine with the second shaft design revealed an exceptional degree of smoothness which was fully acceptable to all concerned. The final shaft is shown in Figure 13.

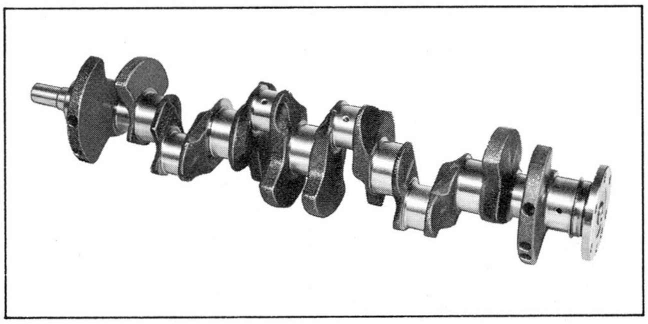


Figure 13 8 Counterweight Crankshaft

CYLINDER BLOCK

The cylinder block, shown in Figure 14, is a thin-wall alloy iron casting with .170 nominal wall thickness. Cylinder block rigidity is increased by the multiplicity of bearing bulkheads provided for the seven main bearings. Water jackets surround the entire ring travel area, and are full length except adjacent to the camshaft.

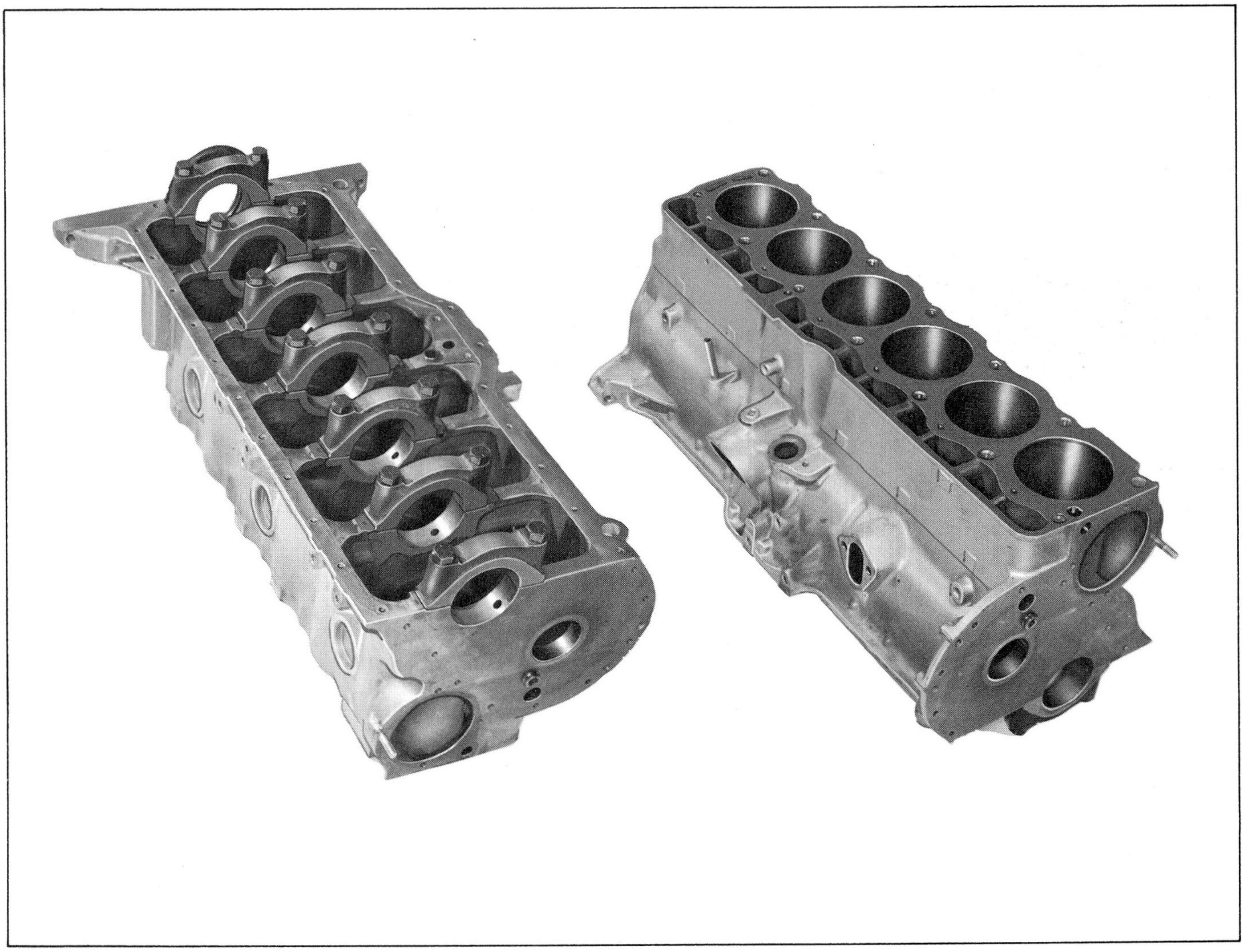


Figure 14 Cylinder Block

FLYWHEEL

As mentioned above, higher-than-average flywheel inertia was provided as one of the necessary requisites of a really smooth six cylinder engine. Flywheel polar moment of inertia is 1.97 lb. in. sec.^{2}

Using the cores shown in Figure 15, two blocks are cast in a single mold with their oil pan gasket faces adjacent. Each block requires a separate water jacket core, tappet chamber core and rear flange core. Figure 16 illustrates a typical double-ended barrel and body core which serves for one bay of each cylinder block in the mold. Six of these cores are used, giving a total of twelve cores in the mold or an average of six cores per cylinder block.

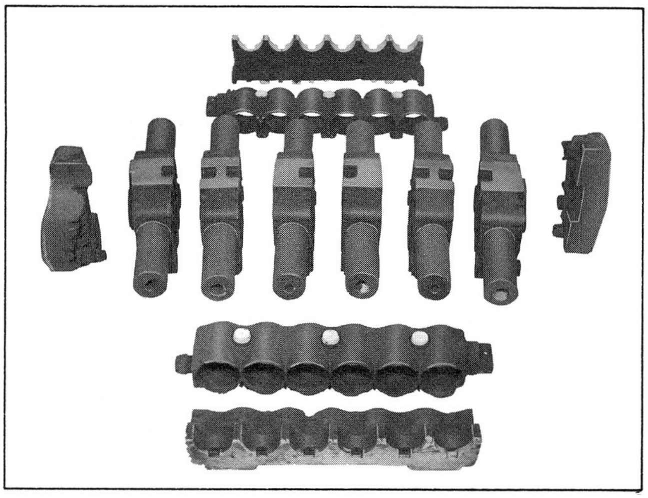


Figure 15 Cylinder Block Cores

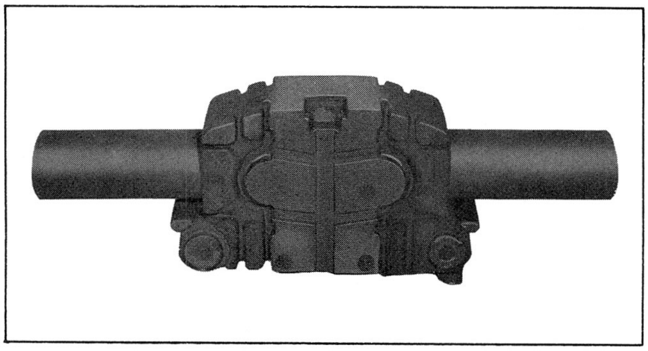


Figure 16 Cylinder Block Barrel and Body Core

Figure 17 shows one engine's water jacket and tappet chamber cores assembled to a set of barrel and body cores as they would be positioned in the mold. The rear flange core is not shown.

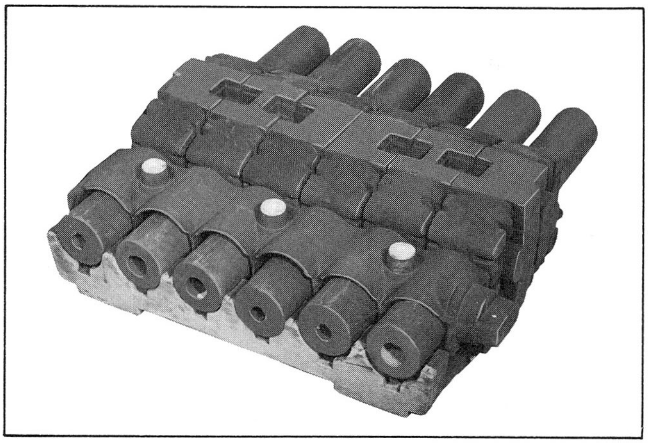


Figure 17 Partial Cylinder Block Core Assembly

LUBRICATION SYSTEM

The lubrication system is illustrated in Figure 18.

Oil regulated to 45 psi is supplied by a gear-type pump. The full-flow filter screws onto an adapter integral with the cylinder block. A filter bypass valve in the block opens at 9 psi differential pressure.

The oil gallery intersects the bores for hydraulic lifters, and individual cross-drillings from the main bearings to the gallery also intersect the camshaft bearings. A grooved bearing and a drilled passage in the front camshaft journal provide a constant supply of oil to the timing chain and sprockets.

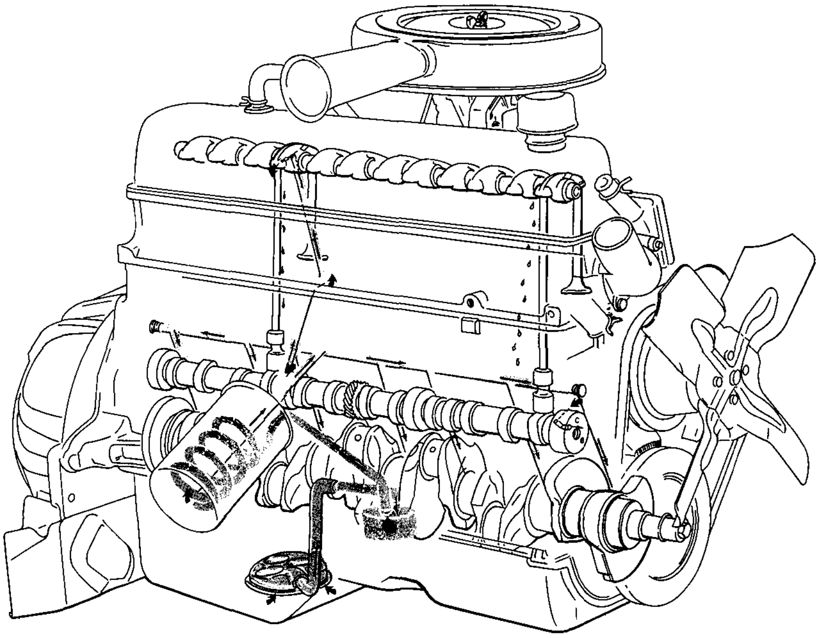


Figure 18 Lubrication System

A groove in the camshaft rear center bearing meters oil to a drilling which rises to the top surface of the cylinder block. Oil flows around a head bolt, through a drilled passage in the cylinder head, and around a rocker shaft bolt to reach the shaft interior. Oil pressure in the shaft is maintained at a low level by a bleed slot in the deflector at the top of the front rocker shaft pedestal (see Figure 19).

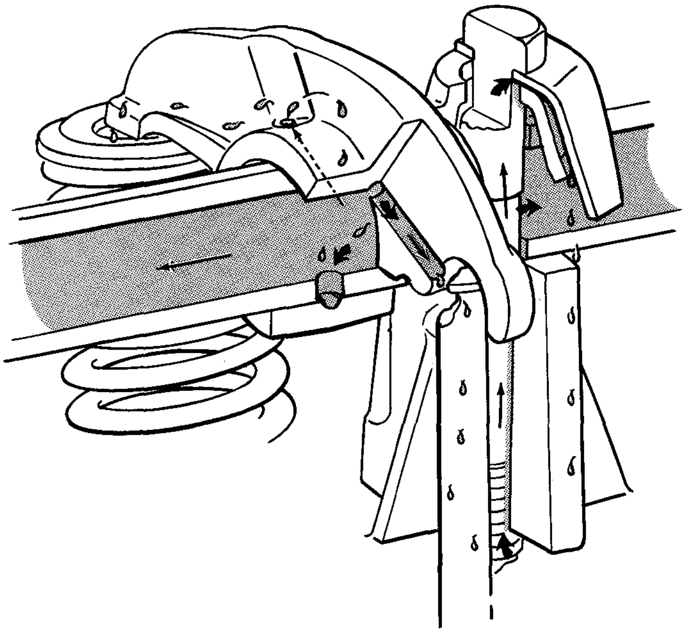


Figure 19 Overhead Oiling

Drilled holes in the rocker shaft deliver oil to a groove in the rockerarm bore. A drilling in the rocker intersects this groove and carries oil to the pushrod socket. A second hole drilled through the rockerarm bearing boss breaks out into the bore near the oil groove. Oil metered through the rocker-to-shaft clearance seeps out this hole, wetting the rocker exterior and providing controlled lubrication to the valve stem and rockerarm tip.

COOLING SYSTEM

The water pump delivers coolant through a curved blade impeller directly into the front of the cylinder block water jacket. Coolant circulation is in series across the cylinder walls to the rear of the block and then into the head, returning to the front of the engine through the cylinder head water jacket. Small holes in the head gasket meter water to each spark plug boss.

The thermostat is held in place by an aluminum die-cast elbow which provides connections for the radiator hose and the thermostat bypass to the heater.

CRANKCASE VENTILATION

Crankcase emission is controlled by a ventilation system which introduces air into the rocker cover through a filter. Crankcase fumes are then removed through a rubber tube running from the valve cover to a point in the intake manifold just below the carburetor. A modulating valve is inserted in the line to control the flow of blowby gases and ventilating air.

Carburetor metering is, of course, compensated for the flow introduced downstream of the carburetor.

ENGINE PERFORMANCE

Figure 20 depicts some of the WOT performance characteristics of the engine. The maximum rated power is 145 brake horsepower, occurring at 4300 RPM. The torque curve is fairly flat, with excellent low speed torque. The maximum rated torque is 215 lb. ft., occurring at 1600 RPM. The fuel consumption characteristics are very satisfactory, indicating that the manifold, combustion chamber and carburetor are well matched designs. The low friction horsepower characteristics shown are partially the result of the oversquare design (low piston velocities), and partially the result of the low pumping losses (efficient induction system).

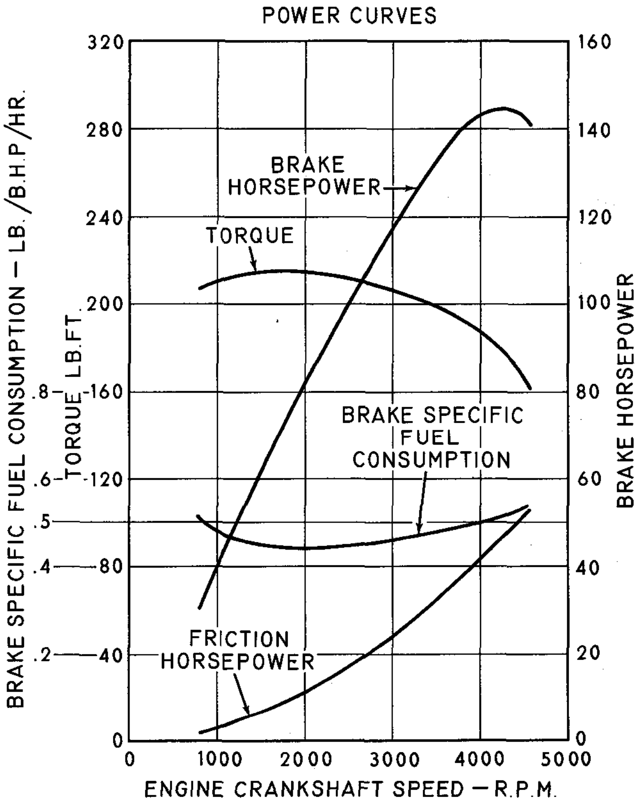


Figure 20

Figure 21 shows the full-load spark timing characteristics. Maximum power at speeds above 1400 RPM was obtained without detonation. At speeds lower than 1400 RPM only a slight loss in power was required with the spark timing curve which provided operation without even trace detonation.

Figure 22 shows heat rejection to the radiator in B.T.U.'s per minute versus engine speed at both road load and at wide open throttle conditions.

Figure 23 shows the oil pressure versus engine speed. The high oil pressure at idle and the flatness of the oil pressure curve at the higher speeds indicate the adequacy of the pump for both capacity and regulating valve control.

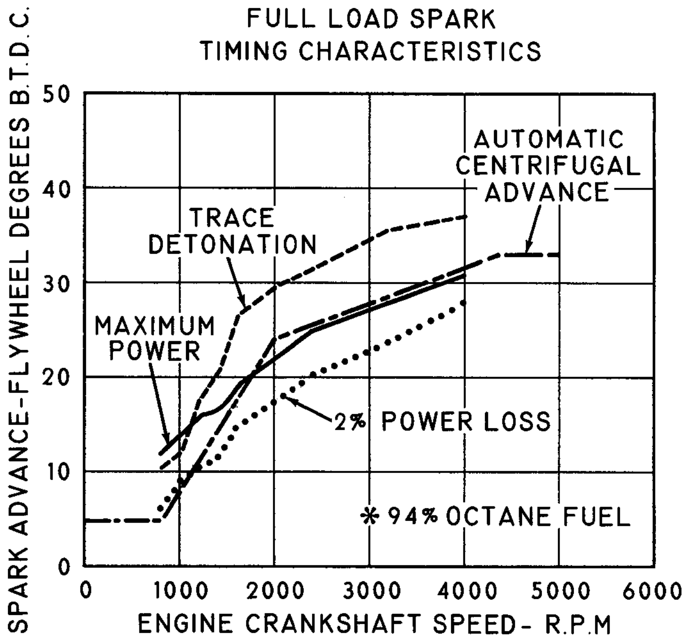


Figure 21

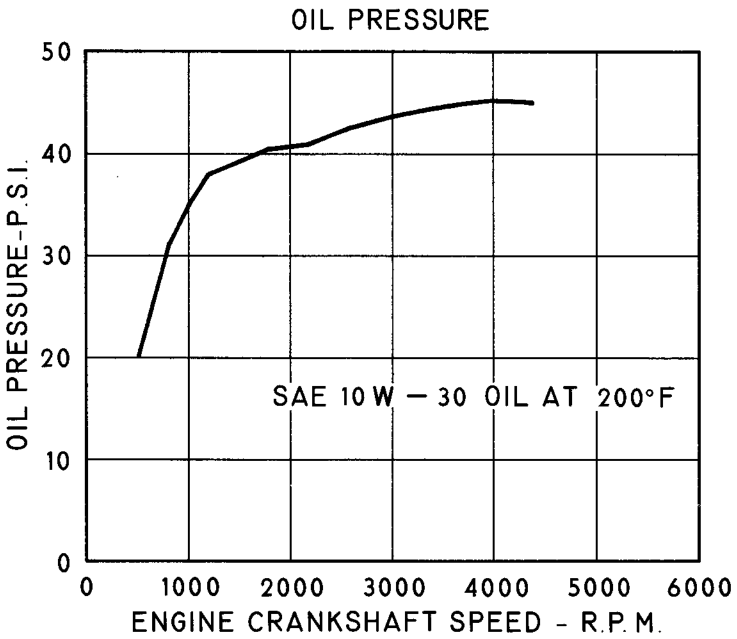


Figure 23

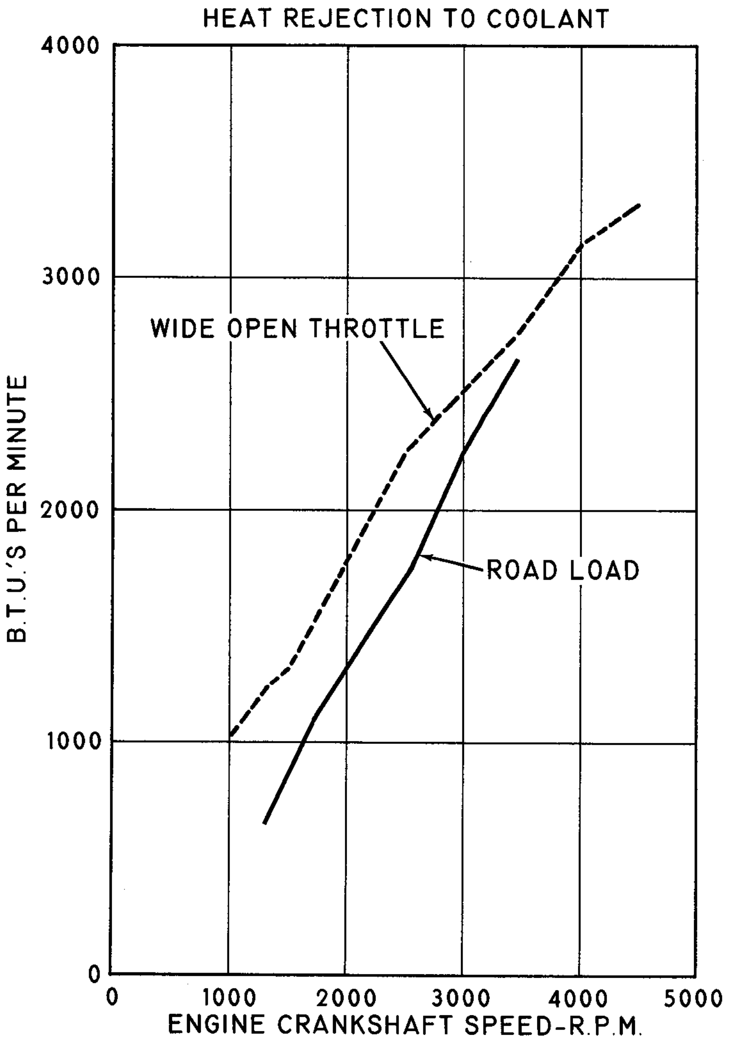
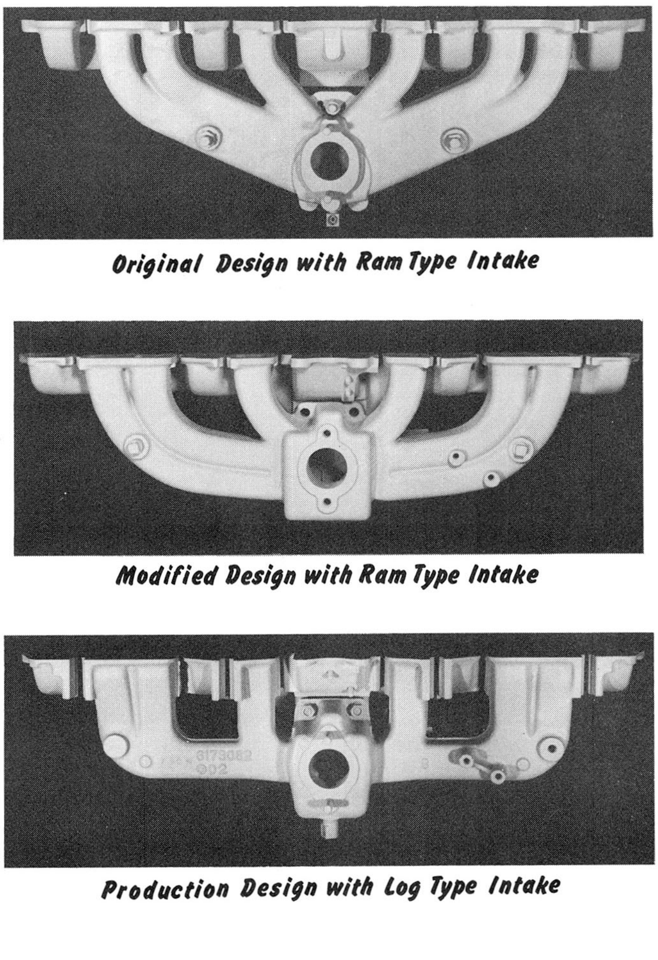


Figure 22



Original Design with Ram Type Intake

Modified Design with Ram Type Intake

Production Design with Log Type Intake

INTAKE & EXHAUST MANIFOLD ASSEMBLIES - TOP VIEW

Figure 24

DEVELOPMENT PROGRAM

MANIFOLDS

Intake — The original intake manifolds under consideration were of two basic designs, the ram-type and the log-type (Fig. 24). The early ram-type manifold (Fig. 24—Top) had to be abandoned for two reasons:

1. Engine compartment revisions reduced the space available.
2. There was a problem of air distribution to the cylinders.

The modified ram-type manifold (Fig. 24—Center) was then tested. Major driving problems (low end WOT stability especially in cold weather) occurred with this manifold. The problems were traceable to the box section of the manifold which was of necessity very large, having all the individual runners originating at this point. Even though a considerable amount of exhaust heat had been directed to the riser area of the intake manifold to aid in proper vaporization of the fuel, the problems were not completely corrected.

The log-type manifold (Fig. 24—Lower), although initially presenting problems, responded well to minor changes in the box section and runner areas. These included the sizing and location of the fuel distribution ledges in the floor and walls of the manifold. See Fig. 25 for a comparison of air fuel ratio variation from cylinder to cylinder at 1000 RPM with all three manifolds.

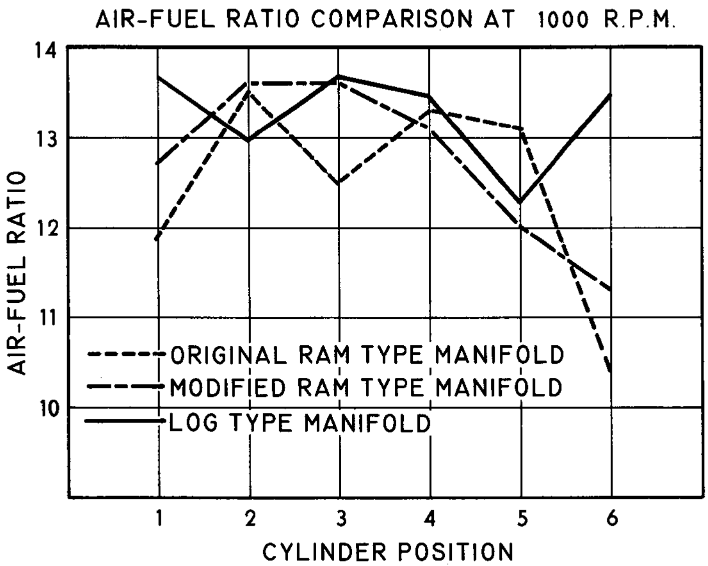
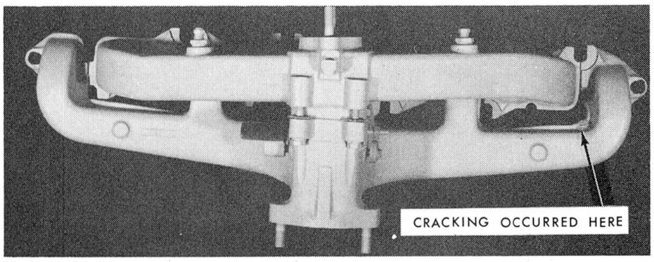
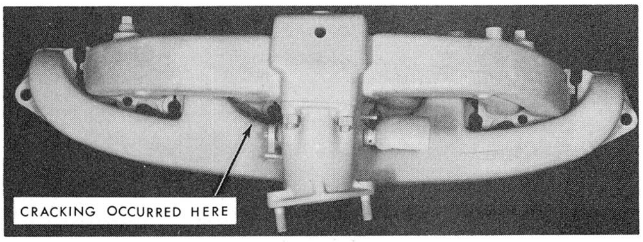


Figure 25

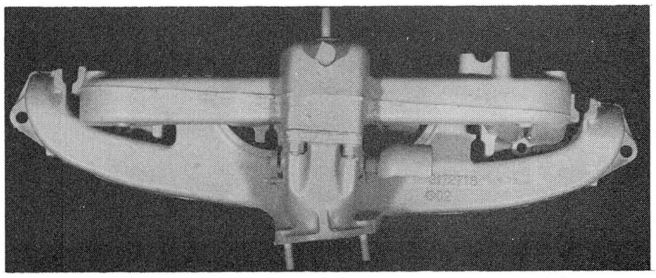
Exhaust — The original design exhaust manifold (Fig. 26—Upper) had square corners at number 1 and 6 cylinders. It was found that under severe power changes on dynamometer test, the growth and shrinkage of the runners due to temperature changes resulted in cracking in the areas depicted. A redesign (Fig. 26—Center) was made to reduce the high stresses in the area of the sharp bend, thereby distributing the stresses along the manifold. This was successful in completely eliminating this problem. However, after extended operation cracking appeared at the saddle between cylinders number 4 and 5



Original Exhaust Manifold



Non Ribbed Exhaust Manifold



Ribbed Exhaust Manifold (Production)

INTAKE & EXHAUST MANIFOLD ASSEMBLIES - SIDE VIEW

Figure 26

and occasionally between cylinders number 2 and 3. Reinforcing ribs (Fig. 26—Lower) eliminated cracking in these areas. Occasional instances of cracking in and around the box section were eliminated by increasing fillet radii and metal thickness in the problem areas.

HEAT VALVE

A section view of the intake and exhaust manifold assembly is shown in Fig. 27. The path taken by exhaust gases during the warmup period of operation is traced by arrows. The design is such that an almost complete envelopment of this area of the intake manifold by the exhaust gases is obtained. This provides intake manifold heat for quick fuel vaporization to insure good driveability during warmup. It also permits early opening of the automatic choke, which is important in obtaining good fuel economy during short trip driving.

Several problems arose during the development of the heat valve. After extended operation, primarily during very cold weather, there was a definite lack of engine response to sudden throttle openings. Examination disclosed that a 1/8 inch thick carbon deposit had built up on the floor of the intake manifold. This deposit did two things:

1. It insulated the hot floor of the intake manifold so that the fuel provided by the sudden opening of the throttle did not immediately flash into vapor for fast engine response.

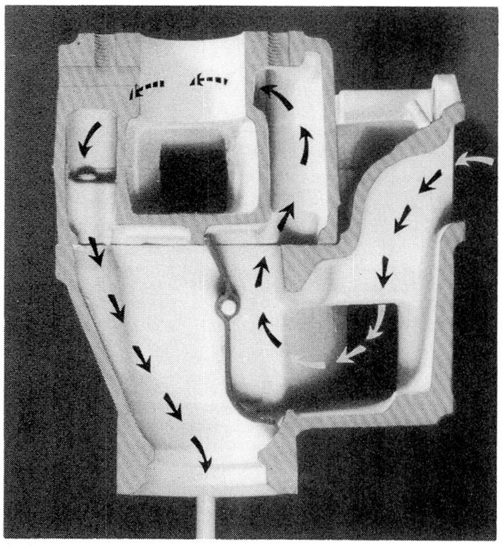
2. It destroyed the intake manifold floor configuration, generally disturbing the air fuel mixture distribution and resulting in rough engine operation at low speed.

Laboratory analysis showed that the deposits contained both fuel and oil residue.

Two changes were made to eliminate the oil residue problem:

1. The rocker cover baffle was revised to minimize oil passing through the crankcase ventilating system into the intake manifold.
2. The vacuum booster section of the fuel pump was revised to seal against oil passage to the intake manifold during windshield wiper operation.

Tests determining the effect of intake manifold floor temperature on fuel deposit build-up rates indicated that the build-up rate was very slight as long as the floor temperature did not exceed 300°F. Controlling the manifold floor temperature was accomplished by changing the characteristics of the bimetal spring operating the heat valve. The decreased floor temperatures obtained with the modified bimetal spring eliminated the deposit problem in the intake manifold without deteriorating the satisfactory warmup performance.



INTAKE & EXHAUST MANIFOLD ASSEMBLY

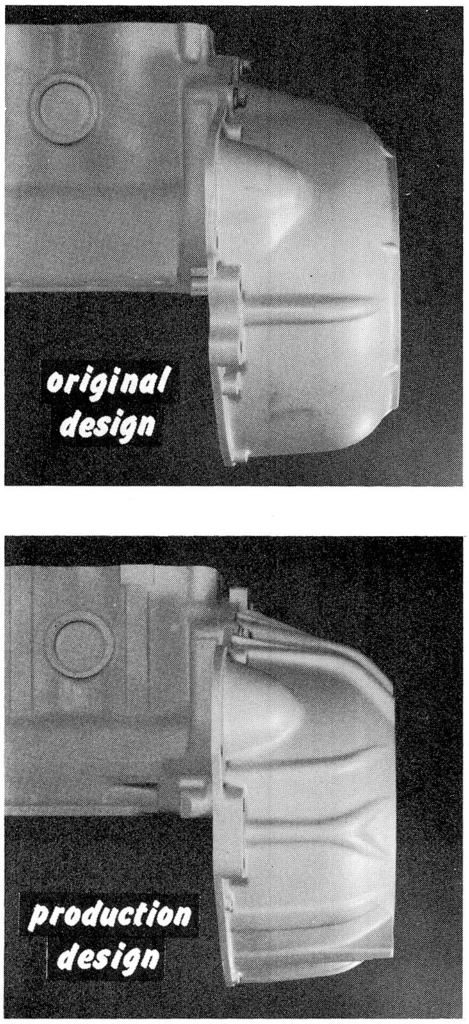
section view through heat valve

Figure 27

POWERPLANT VIBRATION PROBLEMS

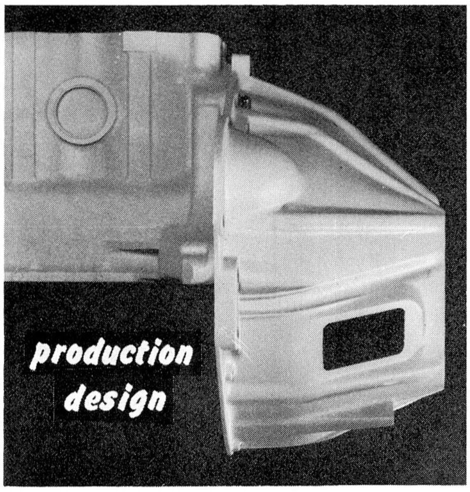
One vehicle vibration problem was encountered at approximately 3000 RPM. Engine load in this case was not a factor. It was determined that the cause was vertical bending vibrations of the engine and transmission assembly. A low vertical stiffness of this assembly resulted in a natural frequency of 60 cycles per second. This noise problem was eliminated by raising the natural frequency to 85 cycles per second by the following changes (see Fig. 28):

1. Thickness was added to the lower dowel bolt bosses on the block, and generous ribs were added between the bosses and the pan rails.



Converter Housing & Cylinder Block (Attaching Area)

Figure 28



Clutch Housing & Cylinder Block (Attaching Area)

Figure 29

2. The converter housing was modified by adding ribs, increasing the transmission mounting flange fillet radii and increasing the wall thickness from .160 to .200.

The information gained by working out the problems with the converter housing was successfully applied to the clutch housing design used on manual transmission cars (see Fig. 29).

Other vehicle noise and vibration problems were determined to be due to lateral vibrations of the cylinder block. The engine front mounts are located in an area of high lateral amplitude. This amplitude was reduced and the problem was corrected by increasing the metal thickness of the oil pan rails and widening the webs at the oil pump flange area (see Fig. 28).

PHOTOSTRESS ANALYSIS

The stress level and distribution of stresses in the cylinder block were investigated using photostress analysis techniques. (For more details regarding this procedure see SAE Paper 471-J, entitled "Photostress Analysis of Automotive Structures" by L. J. LaDouceur, of American Motors Corporation.)

A nominal endurance limit of 22,000 psi was determined to be representative for the cylinder block material based on a modified Goodman-Soderberg diagram for non-reversed stresses.

The maximum stress of approximately 22,500 psi in the head bolt boss area was reduced to approximately 19,000 psi with modifications to the cylinder block, as follows:

1. Wall thickness was increased.
2. Fillet radii were increased.
3. Internal ribbing was added.
4. The head bolt diameter was increased from 7/16″ to ½″.

This stress reduction was obtained in spite of a bolt torque increase which accompanied a change in the head gasket design.

The core plug area, originally critical for stress concentrations, showed a substantial stress reduction as a result of some of the previously mentioned structural modifications.

A force of 7,400 pounds was applied to the crankshaft by means of a hydraulic load cell placed in the cylinder between the head and the crankshaft to simulate combustion loading. A very safe maximum stress of 13,500 psi was located at the base of the main bearing bulkhead in line with the bearing cap bolts.

The maximum cylinder block stress encountered as a result of simulated inertia loading of the crankshaft was 13,000 psi.

A force of 1,500 pounds on the thrust bearing resulted in a maximum stress of only 8,000 psi at the base of the bulkhead in line with the bearing cap bolts.

GENERAL TEST PROGRAM

Over two million miles were accumulated during the development of the "Torque Command" engine. Included in these vehicle tests were evaluations of general engine performance at various conditions of altitude and climate, as follows:

Altitude	Climate	Location
High	Hot and Dry	Albuquerque, New Mexico
Low	Hot and Dry	Phoenix, Arizona
High	Cold and Wet	Denver, Colorado
High	Medium	Smoky Mountains
Low	Cold and Damp	Bemidji, Minnesota
Low	All	Kenosha, Wisconsin Detroit, Michigan Burlington, Wisconsin

Over 15,000 hours of performance and endurance testing were carried out on dynamometers in Kenosha, Wisconsin and Detroit, Michigan.

Over one million miles were accumulated in and near San Antonio, Texas on ten cars with engines built both on the production line and by the Engineering Department. Mileage at the rate of 1,000 miles per car per day was accumulated regardless of weather. The vehicles were driven on a route which included the following:

Roads	Test Conditions	Percent of Route
1. Turnpike	70 mph	35%
2. Farm Roads Hill Country City and Town	Dusty and Rough Hills and Curves Stop and Go	36%
3. Test Track	80 mph	29%

The first car to complete 100,000 miles did so without the necessity of removing the cylinder head or oil pan from the engine. At 107,000 miles (at which time the engine was removed from the car for inspection) the oil economy was still better than 1,000 miles per quart.

SUMMARY—CONCLUSIONS

The design goals set forth at the beginning of the 232 engine program were met as indicated below:

	Design Goal	Final
Horsepower	140	145
Torque, lb. ft.	200	215
B.S.F.C., lb. per BHP-hr.	.45	.45
Service life, miles	100,000	100,000
Dynamometer endurance, hours	500	500
Weight, with flywheel, pounds	450	441

Our concept was to provide an engine which would approach the driving "feel" of a moderate displacement V-8, with the operating economy and cost savings of a six. We felt that very modest concessions in cost and weight, as compared to normal six cylinder practice, could produce the desired results.

By judicious choice of basic engine parameters and by designing to take advantage of the most modern fabrication techniques, the price paid to meet our higher standards was kept minimal and our weight target was met.

And at the conclusion of the development program, comments from non-technical test drivers indicated that the V-8 "feel" was indeed there.

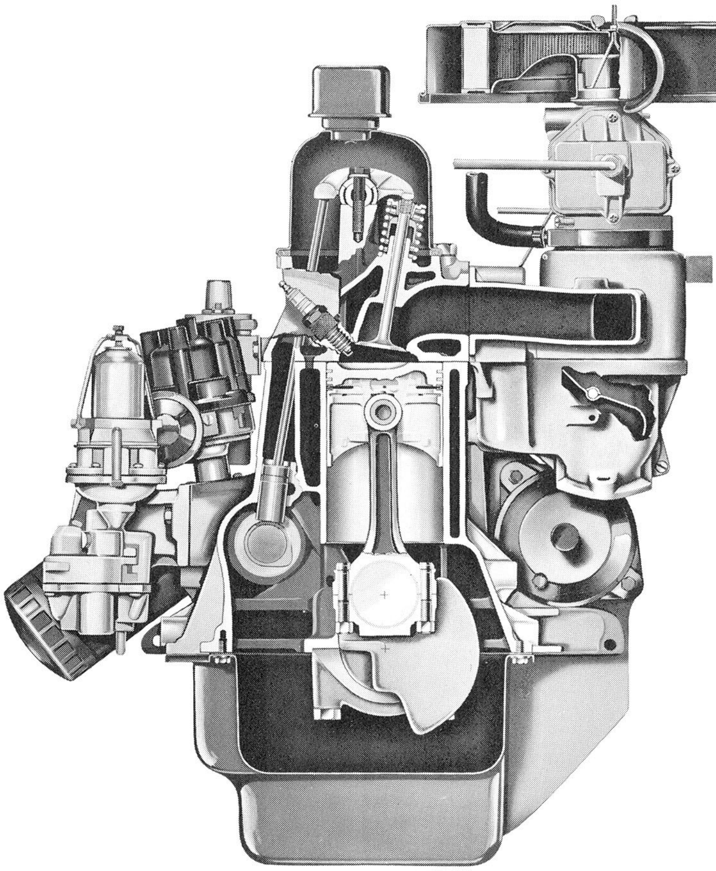
ACKNOWLEDGEMENTS

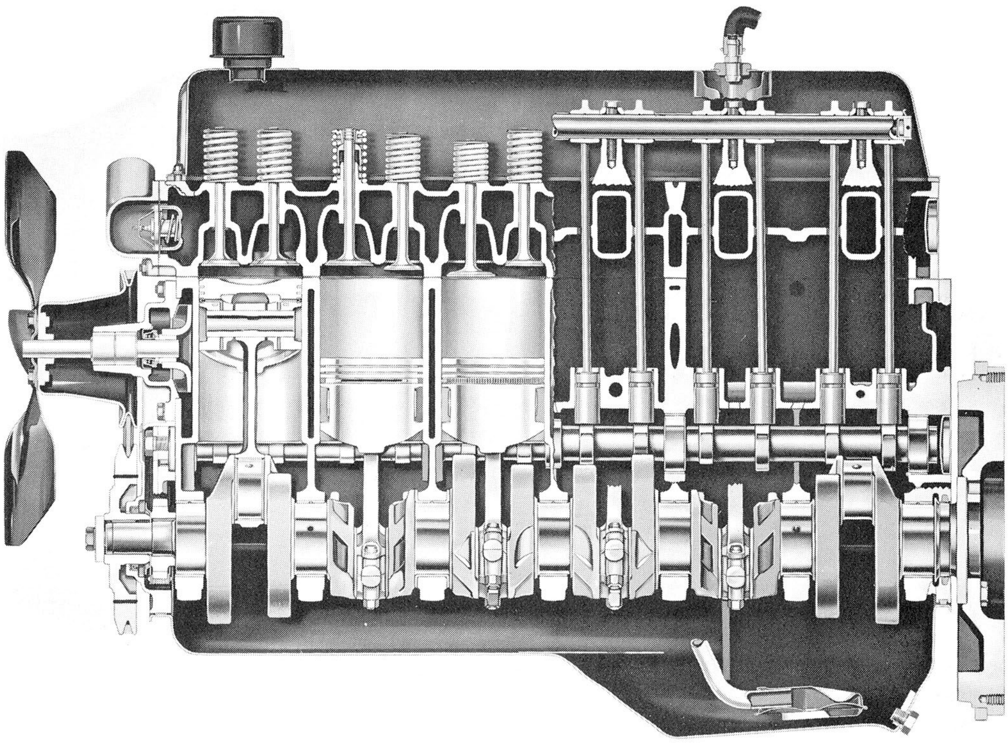
The success of the engine is due, of course, to a number of dedicated individuals and their expenditure of time and energy on this program. The team play of the personnel involved is the only means by which a new engine program of this magnitude can succeed.

Space does not permit mentioning all of those directly concerned with this project, but a few who deserve special merit must be noted. The complete program was under the direction of Ralph H. Isbrandt, Vice President of Automotive Engineering and Research, and through his guidance the project was jointly developed by both our Kenosha and Detroit Engineering Departments. Design and prototype development was under the direction of Wallace S. Berry, Director of Automotive Research, and Carl E. Burke, Chief Development Engineer. The project was accomplished with the close cooperation of John F. Adamson, Director of Engineering, Joseph A. Seidl, Executive Engineer, Kenosha Operations, and his assistants, E. J. Pobar and S. O. Wahamaki.

GENERAL SPECIFICATIONS

Number of Cylinders	6
Displacement, Cu. In.	232
Bore, In.	3.75
Stroke, In.	3.50
Stroke-Bore Ratio	.933:1
Bore Spacing, In.	4.38
Gross HP @ RPM	145 @ 4300
Gross Torque @ RPM, lb. ft.	215 @ 1600
Compression Ratio	8.5:1
Carburetor and Choke	1-Bbl., Auto.
Valve Lifters	Hydraulic
Valve Head Diameter, In.—Intake	1.79
—Exhaust	1.41
Valve Seat Angles—Intake	30°
—Exhaust	45°
Valve Lift, In.—Intake	.375
—Exhaust	.375
Valve Timing—Intake Opens, Degrees BTC	12°30′
—Intake Closes, Degrees ABC	51°30′
—Exhaust Opens, Degrees BBC	50°30′
—Exhaust Closes, Degrees ATC	10°30′
Valve Spring Load, lb.—Valve Closed	88
—Valve Open	155
Rockerarm Ratio (Nominal)	1.5:1
Crankshaft Material	Cast Malleable Iron
Crankshaft Main Bearings	7
No. of Counterweights	8
Main Bearing Diameter, In.	2.500
Crankpin Bearing Diameter, In.	2.095
Wristpin Diameter, In.	.931
Connecting Rod Material	Cast Malleable Iron
Oil Gallery Maximum Pressure, PSI	45
Engine Oil Capacity, Including Filter, Qt.	5
Firing Order	1-5-3-6-2-4





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