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Chrysler Corporation's New Hemi Head High Performance Engines

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IN DECEMBER 1962 a request was made of engineering staff to develop an engine and vehicle combination capable of winning stock car competitive closed circuit track events throughout the country

It was further requested that a version of the engine suitable for use in supervised timed vehicle acceleration drag events also be made part of the engine portion of this program.

Both of these applications were to share the same basic engine design and development; thus a decision on the basic engine itself was one that required careful consideration.

However, the decision for Chrysler was obvious. The hemi engine had been a production engine from 1950 until it was discontinued in 1958. The background of development that preceeded the production version of the engine was available, as was experience in converting the engine for competition use. Chrysler hemi engines had been used previously in track events, and were still in use in many drag events. Engineering staff had also previously prepared a 271 cu in. fuel injected ram tuned version of the original hemi for use at the Indianapolis Speedway.

The design of the new hemi engine began in January 1963 with the Feb. 23, 1964 Daytona Beach, Fla. race set as the first target date.

A number of engine design avenues were initially explored in order to gain as many performance and durability features as possible while still retaining as much of the existing cylinder block tooling as practical.

The final design selected was deemed to have the potential necessary to win and the program was launched. The engine requirements were to include a single 4 barrel carburetor 426 cu in. track engine (Figs. 1A and 1B), a single 4 barrel carburetor 396 cu in. track engine and a two 4 barrel carburetor 426 cu in. drag engine (Figs. 2A and 2B). Immediately following the initial introduction of the engine, a production run of several hundred drag engines and cars were planned to be built. This would serve to give widespread usage of the hemi on the drag strips throughout the country and also serve as production backup to the track engine.

Experimental procurement of the 426 cu in. hemi engine was started in July 1963 and was completed by the end of November. The first engine ran under its own power on Dec. 6, 1963. This engine was built as a track engine; the drag engine version was built later the same month. The 396 cu in. track engine had subsequently been removed from the rules and the engine was never built.

This 426 cu in. track engine was used extensively throughout the 1964 season. The production of the several hundred drag engines was completed by the end of the 1964 model year. Another production run of several hundred drag engines was made for the 1965 model year automobiles, with a considerable weight decrease for the engines obtained by use of aluminum and magnesium components.

The 1965 track engine usage was restricted due to a change in eligibility rules which limited the scheduled race events in which the hemi engine was allowed to compete.

A detuned high volume street version of the 426 hemi was designed, developed, released, and tooled as part of the 1966 model offering (Figs. 3A and 3B). With this release,

ABSTRACT

This paper covers the design and development of a family of engines used for closed circuit track competition and acceleration trail competition. In addition, a detuned version of the engine suitable for normal street and highway driving is described. All these engines share a hemispherical combustion chamber using push rod operated valves.

full eligibility of the engine for 1966 track use was again established. A 404 cu in. version of the engine for use on certain tracks was also prepared for the 1966 season in compliance with the revised racing rules.

The design and development of the hemi engine in its four major versions -- 404 and 426 cu in. track, 426 cu in. drag, and a 426 cu in. street engine can now be described in detail.

DESIGN CONCEPTS

The 1950-1958 Chrysler hemi engine had used an included valve angle of 58.5 deg. This feature had had enough development that a decision was made to retain this angle for the new hemi (rounded off to 58 deg). Any sacrifice to the intake port was not to be considered, thus the best configuration known of bringing the port straight toward the bore from the intake manifold was used. On the previous hemi this had been no problem, since a four bolt head pattern was used. However, with the five bolt head pattern of the production cylinder block the inside bolt was directly in the way. This problem was solved by adding bosses in the tappet chamber of the cylinder block and providing a socket head screw that threaded into the floor of the intake port. This screw was later changed to a stud and nut but its location remained the same. Fig. 4 shows the original design layout. Fig. 5 shows cross-sections of the 1964 drag design of the engine, most of which is common to the other versions of the engine.

The valves do not have equal angles from the bore centerline in the transverse view, but are rotated toward the intake side of the cylinder head (Fig. 6). In laying out the geometry of the valve gear, the exhaust push rod was located as near the cylinder bore as possible. The length of the exhaust rocker arm was then determined by the angle of the exhaust valve. If the valve had been positioned further outboard, the rocker arm would have had to be made longer and valve false motion would have occurred at a lower speed.

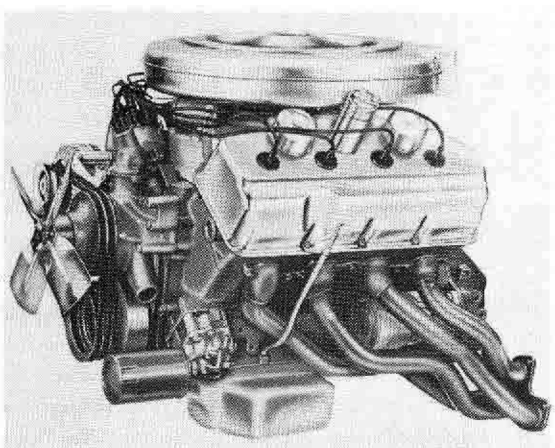
Fig. 1A - Hemi track engine

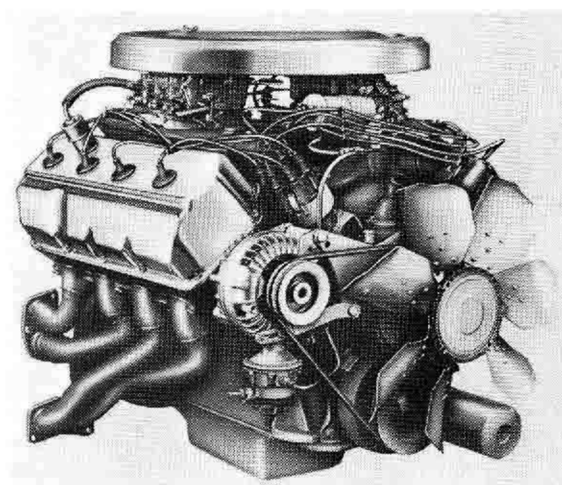
Fig. 2A - Carburetted hemi drag engine

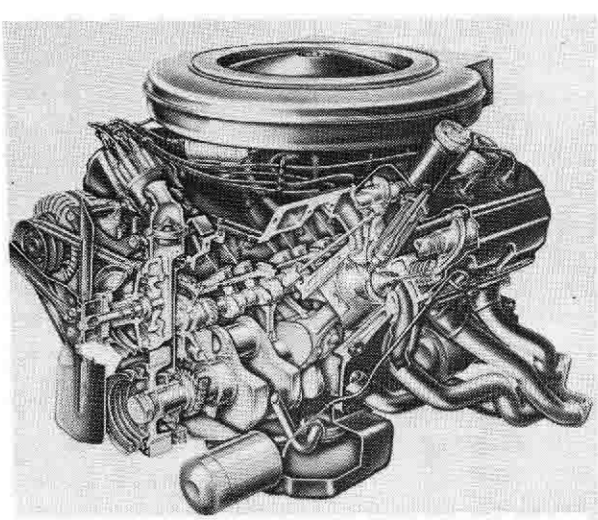
Fig. 1B - Hemi track engine

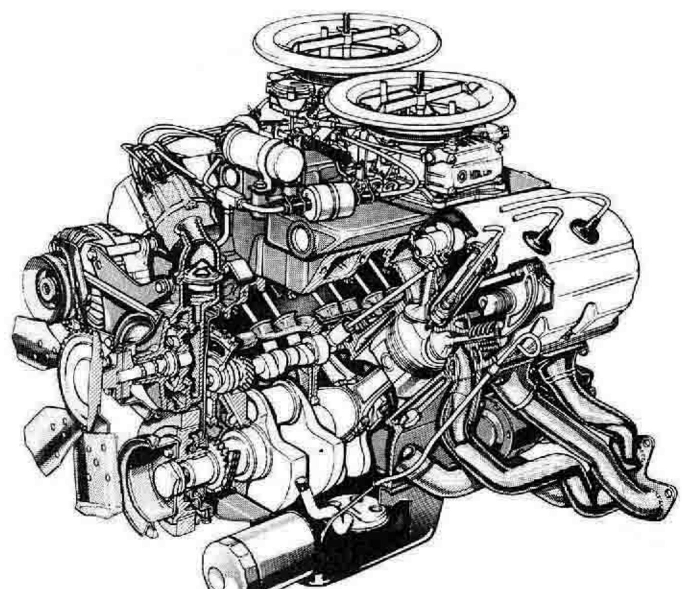
Fig. 2B - Carburetted hemi drag engine

If the valve had been positioned further inboard, the valve sizes would have had to be reduced.

The spark plug was located as close to the center of the chamber as possible. With the chamber and port design completed, the rest of the engine was redesigned in anticipation of the estimated power levels.

CYLINDER BLOCK

The cylinder block (Fig. 7) for the hemi engine shares basic machining dimensions with the wedge chamber engine cylinder block. Bore centers are 4.8 in. Height along the bore axis is 10.725. Vertical height from the crankshaft center is 10.875, and overall block length is 23.46. The front of the block has provision for mounting the distributor and the oil pump. The hemi engine requires an oil drain back hole in each corner of the cylinder block. Bosses extending into the tappet chamber for the cylinder head studs can be seen. The lower surfaces of these stud bosses are machined in order to provide a square seat for the stud nuts and to insure wrench clearance. The fillet of the machining cut is controlled to avoid a sharp corner where the nut seat joins the wall of the tappet chamber. Initial builups of the engine disclosed that the stud nuts were crushing into the iron under the torquing load. This was overcome by use of a hardened washer under the nut. A special torque wrench adapter is required for tightening these nuts inside the tappet chamber.

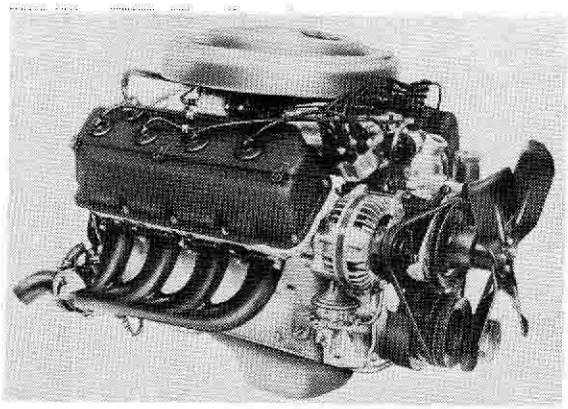

Fig. 3A - Hemi street engine

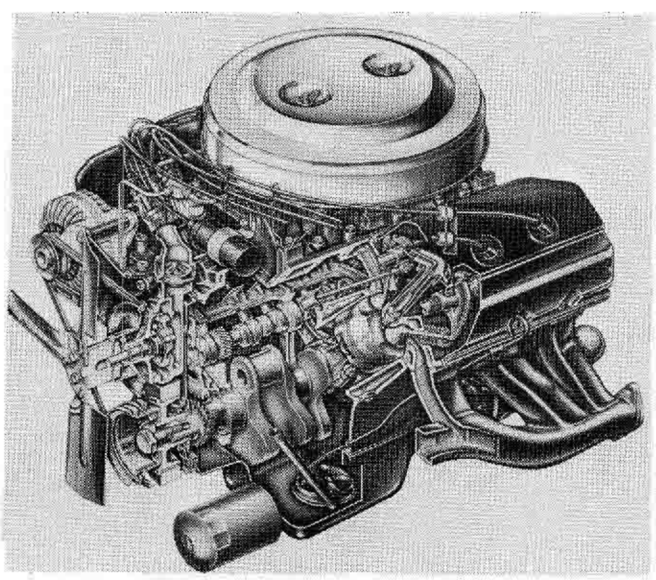

Fig. 3B - Hemi street engine

It was determined that as the connecting rod traveled through its path there was insufficient clearance to the bottom of the bore of the cylinder on the opposite bank. A machined notch was added to insure consistently satisfactory clearance.

Advantage was taken of the 3 in. deep skirt of the cylinder block to use 3/8-16 bolts to tie the skirt to the number 2, 3, and 4 main bearing caps (Fig. 8). This enables the skirt structure to help the bearing caps in resisting the horizontal components of the bearing loads (Fig. 9) that tend to rock the cap across the engine. The main bearing cap has a press fit of 0.002 at the parting line. At the tie bolt bosses, a loose fit of 0.002 per side is used. The skirt walls deflect to close the gap when these bolts are tightened to the specified 45 ft lb. By using hardened and ground washers underneath the main bearing screw heads and thread lubrication on assembly, an initial tension of 15,900 lb is achieved on each of the 1/2-13 screws with 100 ft lb tightening torque. This resists a maximum vertical separating load of 18,800 lb at 7200 rpm. The separating load causes the screw tension to fluctuate from 15,900 to 17,660 lb which is well within the fatigue limit of the grade 8 bolt material. This cap construction has proved to be virtually trouble free.

Shortly after the first engines were run at full power, several engine failures occurred due to vertical cracks in the thrust side of the right hand bank bore walls. A quick analysis showed that these block cracks were occurring in the bore wall opposite the piston pin pier (Fig. 10).

A reduction of this load concentration could have been obtained by increasing the piston cam so that the bore wall would be loaded more uniformly by adding load to the center of the bore while reducing it at the crack location area. The piston, however, had just undergone an intense development program of its own and further changes to it were ruled out. It was decided instead to increase the cylinder block bore wall to 0.3 in.

This problem, together with the uncovering of a residual casting stress in the bulkhead area that had caused random cracks, made it apparent that new cylinder blocks would be required to ensure satisfactory durability for the 500 mile Feb. 23, 1964 Daytona Beach race. A special casting run was made early in February to produce heavy bore wall blocks made by hand reworking water jacket cores. These blocks were then stress relieved by reheating to 1200 F with a slow cool down. The car that came in first on February 23 was equipped with a cylinder block cast on February 10.

Cores for the first cylinder block castings were made with plastic core box equipment; however, the thinness of the sand bridges between the bores, particularly after the bore wall thickness was increased, caused the plastic equipment to be replaced with hot box equipment to increase substan-

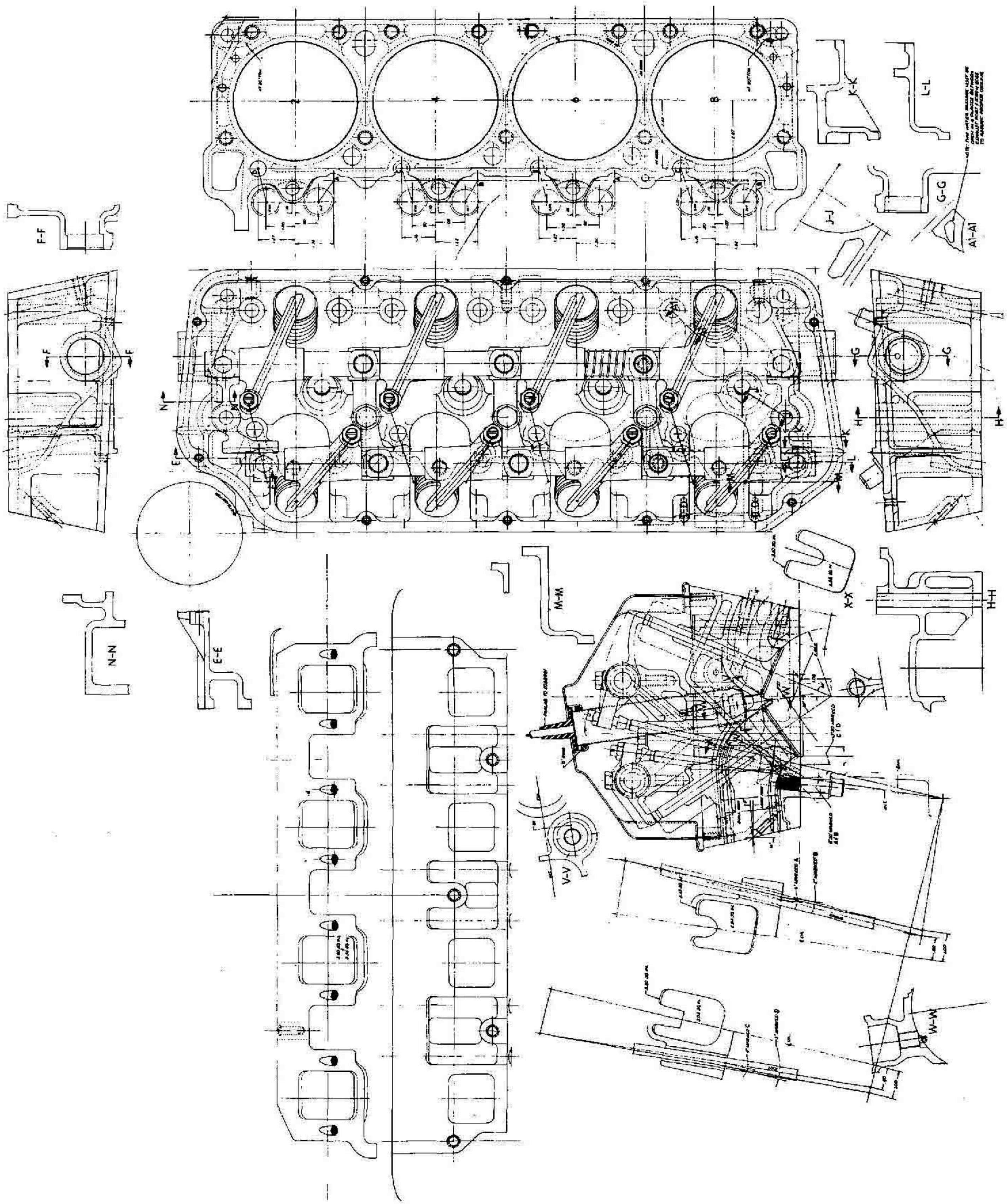

Fig. 4 - Original design layout

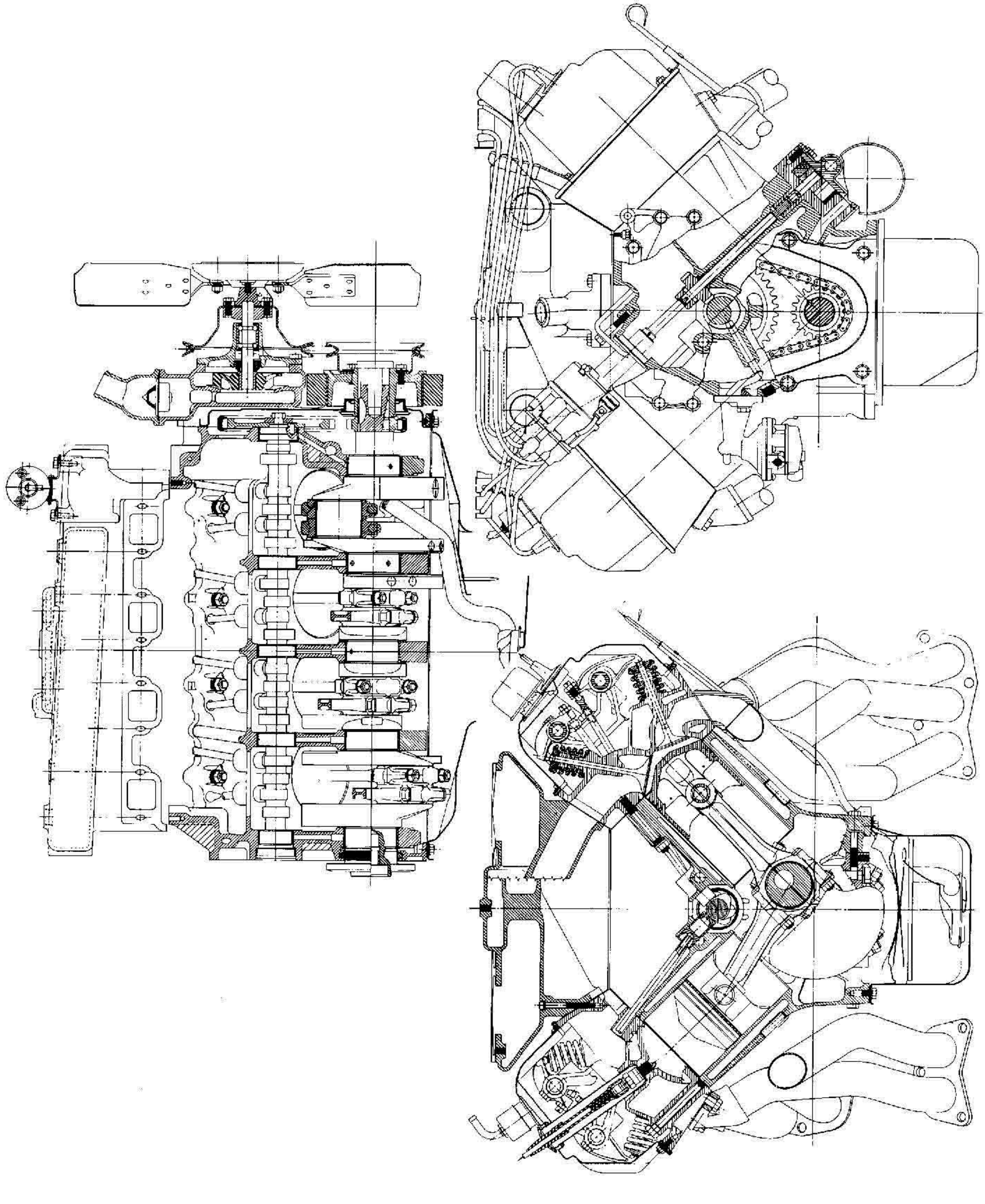

Fig. 5 - Cross-sections of 1964 drag design of engine

tially the strength of the sand core. A high temperature core wash is used on the sand bridges to prevent metal penetration during the casting process. The track, drag, and street engines all share the same tin-alloyed cast iron cylinder block.

CYLINDER HEAD

The cylinder head (Fig. 11) has a fully machined hemispherical chamber with the intake and exhaust valves located directly across the cylinder head from each other and with the spark plug located close to the center of the chamber. The spherical radius of the chamber is 2.420 and the chamber is 1.340 deep at the center. The intake port is led directly from the intake manifold face to the valve seat for minimum flow restriction. The valve push rods straddle the intake port and cause it to change from a circular section

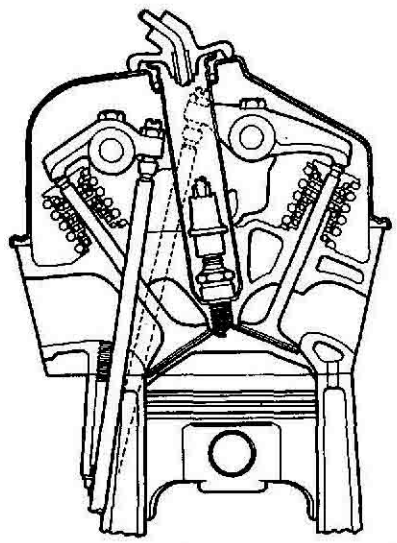

Fig. 6 - Hemi combustion chamber and related components

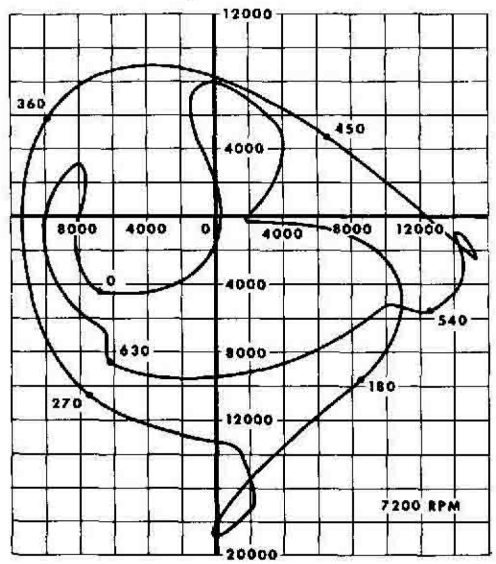

Fig. 9 - No. 4 journal loading

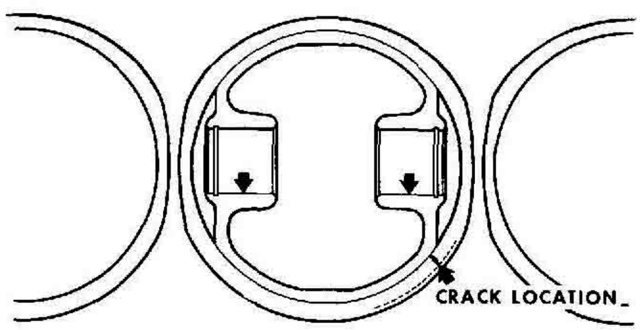

Fig. 10 - Bore wall development

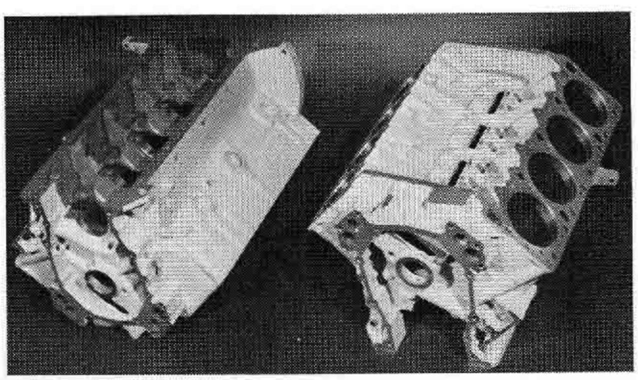

Fig. 7 - Cylinder block

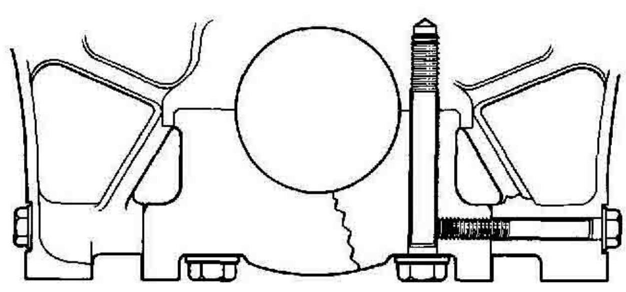

Fig. 8 - Main bearing cap

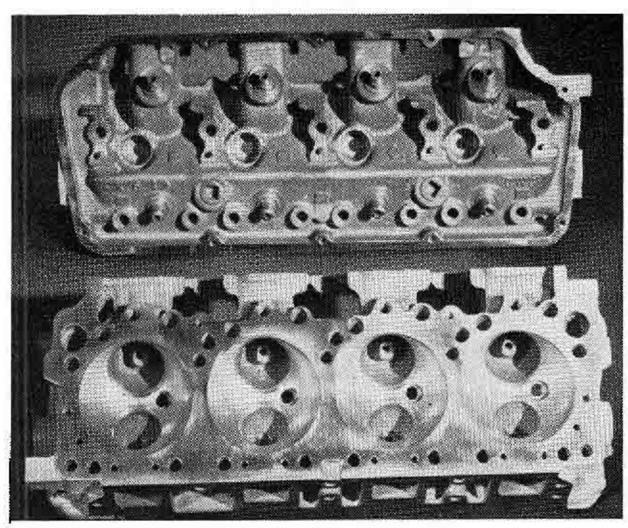

Fig. 11 - Cylinder head

at the valve head to a rectangular section at the push rod bosses. The intake port area is 3 sq in. This area was selected on the basis of valve size, engine rpm, and engine use. The exhaust port leads the combustion products directly from the exhaust valve seat to the exhaust manifold face. The 2.24 sq in. exhaust port has a pronounced bend in the transverse view to give a smooth entry into the exhaust headers which generally follow a vertically downward routing. Port airflow development was started prior to the running of the first engine. This was accomplished by use of port models. It is interesting to note that the improvements were obtained by filling in the port (Fig. 12) rather than by removing metal from the ports. The changes were then incorporated in the cylinder head pattern equipment as the patterns were being made. Fig. 13 shows the intake port airflow rate.

Water jacketing is provided completely around the exhaust valve seat and the exhaust valve guide boss. The spark plug seat and the intake valve seat are partially water jacketed while the intake valve guide boss does not have any water jacketing. Vaporization of the incoming charge of fuel maintains a low enough intake valve temperature to insure satisfactory life and undesirable addition of heat to the incoming charge is avoided. During the development

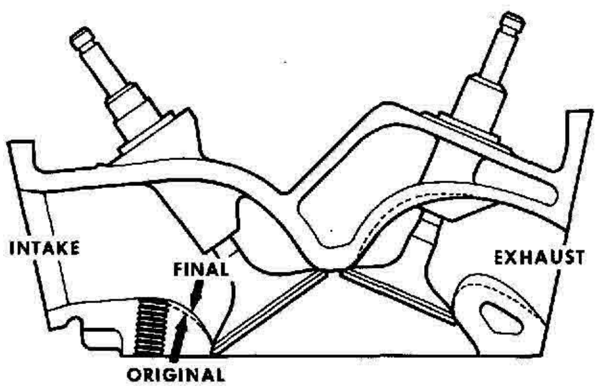

Fig. 12 - Port development

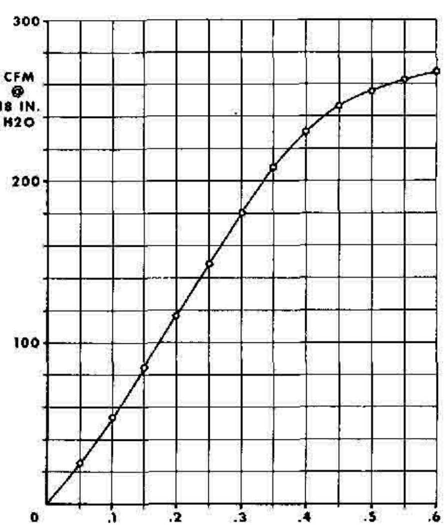

Fig. 13 - Intake airflow

of the engine, high stress areas in the dome of the chamber required a thickness increase from 0.28 to 0.36. Between cylinders, a 0.2 thick rib is located to prevent head deflection due to the gas loading. As soon as the development of the iron cylinder head was finished, an aluminum cylinder head project was started (Fig. 14). Valve seat inserts, valve guides, and steel shims under the valve springs to prevent frettage were added. Based on the stress work on the cast iron cylinder head, the thicknesses of the chamber dome and deck face were increased to 0.4. The diameter of the bolt bosses between the bores was increased from 0.88 to 1.0. Fillet radii were increased throughout the cylinder head and depths of threaded holes were increased. A 0.18 thick washer was used underneath the outboard head bolts.

These aluminum cylinder heads have been used on drag engines since the beginning of the 1965 season.

Oil drain is accomplished by using holes in the extreme outboard corners of the cylinder head that match with similar holes in the cylinder block.

Shell molded intake and exhaust port cores are used on both the cast iron and aluminum cylinder heads to obtain smooth port walls free of fins and metal burn in. The coring system on the cylinder head casting is conventional. Oil sand cores are used for the upper and lower water jackets, the lightener cores between the intake ports, and the perimeter core. The cope and drag are in green sand. Because of the presence of the large ports and the dome of the combustion chamber, the lower water jacket core has many thin sections that cause it to be fragile. Careful handling together with inspection of the casting to ensure open water passages around the exhaust valve seats is required.

Cylinder head covers are formed of 0.040 thick stamped steel with oil baffles welded to the inside. Four holes for spark plug tube entry are provided so that the spark plugs and spark plug tubes can be replaced without removal of the cylinder head covers.

Ten 1/4-20 studs are provided in each cylinder head for attachment of the cylinder head cover and retention of the cover gasket.

The street engine uses a conventional oil fill cap on the

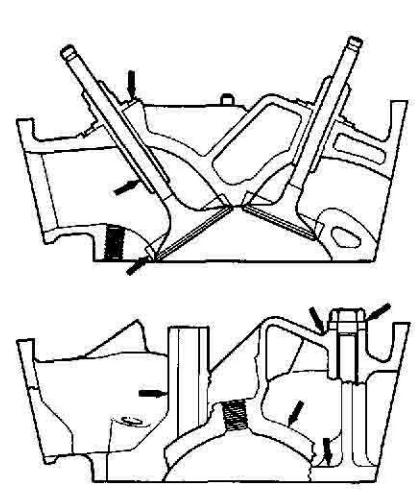

Fig. 14 - Aluminum cylinder head features

left cover and a crankcase ventilation valve cap on the right cover. Solid oil fill caps are used on the track engines. Crankcase blowby pressure is relieved on the track engines by use of tubes from the cylinder head covers to remote breather caps to avoid loss of oil through the breather system during turning maneuvers.

For appearance purposes chromium plated covers are used on the drag engines and black crackle finish covers are used on the street engines.

Cylinder head gaskets are shown on Fig. 15. The iron cylinder head uses a stainless steel embossed head gasket made from 0.020 stock (lower view). An early problem was that of sealing around the large water opening at each end of the cylinder head. This was initially overcome by applying rubber cement on the beads of that portion of the gasket. Later it was determined that the sealing problem was caused by the inboard corners of the cylinder head being lifted during assembly of the intake manifold. Reduction of the screw torque of the two intake manifold screws at each corner of the manifold from 6 to 4 ft lb minimized the cylinder head deflection and this, together with changes to the bead loading pattern, permitted the discontinuance of the rubber cement. The aluminum cylinder head uses an embossed gasket (upper view) similar in dimensions and thickness to the stainless steel gasket but made from a copper sheet. The copper hardness in Rockwell 15T 80-86.

PISTONS

The pistons are impact extruded from an aluminum slug (Fig. 16). The head plate of the die forms the contoured top of the pistons. In order to obtain the 12.5:1 C.R. used by the 426 track and drag engines the piston head is required to protrude 0.755 into the combustion chamber. The 404 track engine piston protrudes 0.772 inches.

Valve timing events are such that at tdc between the end of the exhaust stroke and the beginning of the intake stroke, both intake and exhaust valves are partially open. This requires a circular depression in the piston for valve head clearance. At the closest point, the piston is 0.070 from the valve.

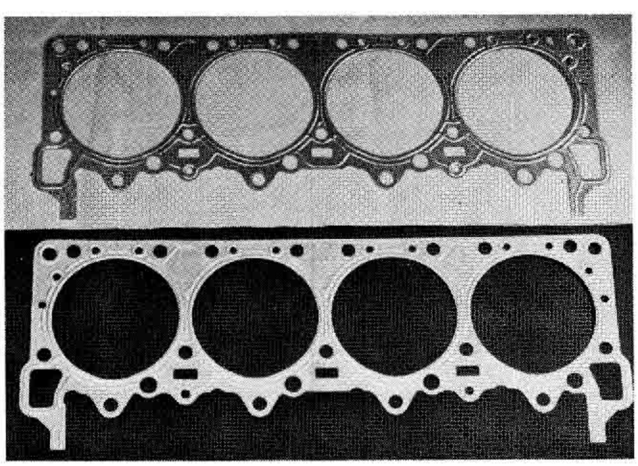

Fig. 15 - Cylinder head gaskets

The extrusion punch is designed to give a 0.305 in. thickness to the piston head. The extrusion process prevents lightening the piston above the piston pin pier as can be done with pistons made by the permanent mold process. A machining cut is used in this area to reduce the piston weight.

The stress problems of the piston largely stem from inertia loading at tdc under high rpm conditions and compression loading due to chamber pressure. Fig. 17 shows areas where cracks developed and where metal had to be added to reduce the localized stress. The fillet between the head of the piston and the top of the piston pin pier had to be increased in radius. The diameter of the lightening cutter had to be reduced to increase the area on either side, and overall width of the piston pin pier had to be increased to strengthen the pier area at the piston pin bore centerline. A chamfer was added to the piston pin bore breakout inside the piston to avoid edge loading of the pin bore. In spite of continuous development on the piston, enough cracked pistons have been observed in dynamometer operation to realize that piston durability is of a critical nature.

A very rigid inspection procedure is used for the competition engine pistons with any surface imperfections in the highly stressed areas being cause for rejection. The original piston design had the piston pin centerline offset 0.06 toward the minor thrust side of the bore. The net effect of gas loading, rod angularity, side reaction load distribution, and friction over the entire cycle caused a reduction in piston drag up and down the bore and an increase in power. The gain in power output was rather modest as measured in the laboratory, however, and when weighed against the disadvantage

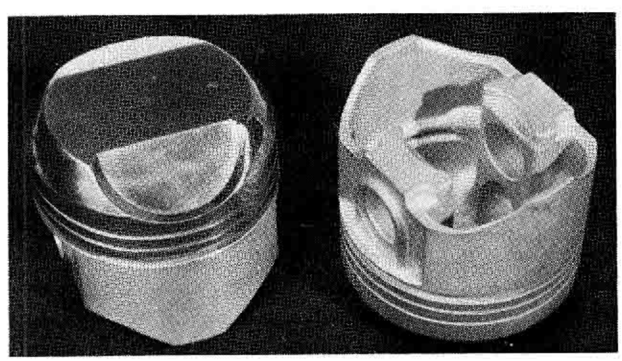

Fig. 16 - Piston

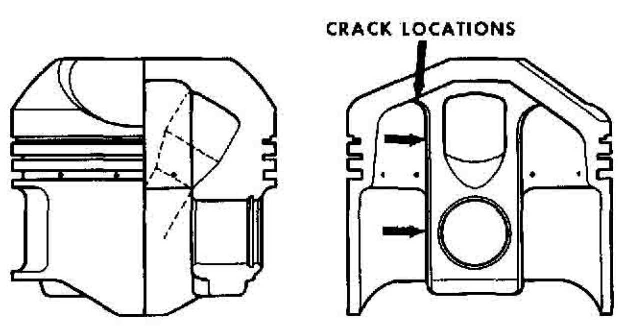

Fig. 17 - Piston development

of having to maintain field supplies of top grade pistons of two types -- a right and a left bank -- rather than one type if the pin were placed on center, it was decided to use the on-center pin for the 1966 track engine.

The 1966 track piston was also modified to reduce piston weight by shortening the piston pin piers. This was accomplished by the use of a longer connecting rod which raised the piston pin bore. This piston is common to the 426 and 404 cu in. track engines.

The street engine also uses an impact extruded aluminum piston somewhat similar to the competition engine except that the compression ratio is lowered to 10.25 by reducing the protrusion of the piston to 0.515. A slot is used to ventilate the oil ring instead of the drilled holes used by the competition pistons. This slot permits the street engine piston to be fitted with less skirt to bore clearance for improved low temperature noise control. This is possible since the slot provides freedom of the skirt from the thermal changes of the piston head. The piston pin is offset 0.06 toward the major thrust side for piston slap noise control, with the vertical location of the piston pin determined by the street engine's use of the original rod length. There are also differences in the piston ring grooves.

PISTON RINGS

Two types of ring line-ups have been developed for use in the engine (Fig. 18). A narrow land extra low friction line-up for use in the track engine is shown on the left and a standard land low friction ring line-up for use in the drag and street engines is shown on the right.

The ring tension of the top ring of the narrow land line-up is 1.5 lb measured at 90 deg to the gap. The ring is 0.062 thick and is made of piston ring iron with a 0.004-0.007 chrome layer on the bore face. The second ring is also 0.062 thick and made of piston ring iron with the face tapered and tin-plated. A 6.4 lb load is required to close the second ring to the 4.25 bore size.

The oil ring is composed of three parts -- two steel rails 0.018 thick, and a stainless steel spacer. These are designed to fit in a 0.126 high groove in the piston. The bore face of the rails is chrome-plated.

The standard land ring line-up has a top ring 0.078 thick with a 10 lb load specified. The ring is iron with a chrome-plated bore face. The chromium surface is slightly crowned to give it a barrel shape. This is done to reduce break-in time. The second ring is also 0.078 thick with a tapered tin-plated bore face and a reverse twist. It is made of piston ring iron with a 9.6 lb load specified. The oil ring uses two 0.024 thick steel chrome faced rails with a stainless steel spacer. These are designed to fit in a 0.188 high groove in the piston.

PISTON PIN

A 1.031 diameter full floating piston pin was initially designed for the engine. Trouble arose in the form of occasional retaining ring breakage due to pounding of the piston pin. A spiral type of retaining ring was used at the time. Tests were immediately started using a 1.0936 diameter pin pressed into the small and of the rod. This pin then became the source of repeated pin scuffing evidently occurring when the engine was being run at the low speed points on the performance curve with full throttle torque. When operating at the rpm range scheduled for the high speed tracks the pin seemed to be free of trouble and accordingly this design was used for the first two seasons for both track and drag racing. Believing that the floating pin was still the best design if the retaining ring breakage could be overcome, development continued, and finally by reducing the end clearance from 0.024 to 0.0145, using the type of retaining rings shown with the identification marks omitted, and reducing the OD chamfer at the ends of the piston pin, a successful design was obtained (Fig. 19).

The identification mark was found to be a stress riser and its elimination helped to eliminate the breakage. Reduction of the OD chamfer caused the piston pin to bear against the retaining ring closer to the ring groove where it had solid support. In addition to being full floating, the 1966 track pin uses the 1.0936 OD with a 0.75 ID. The 0.75 ID is not full length but extends only for the center 1.25 length of the pin and then tapers up to 0.86 ID at the open ends. The ta-

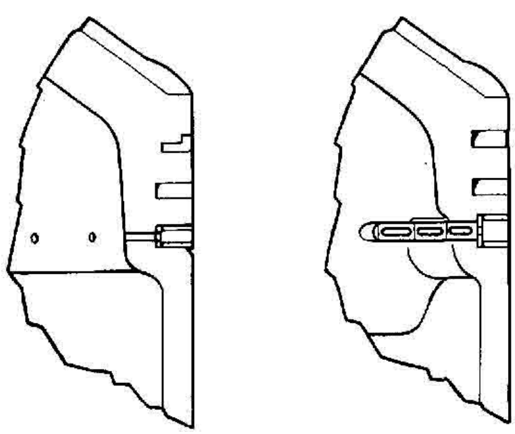

Fig. 18 - Piston rings

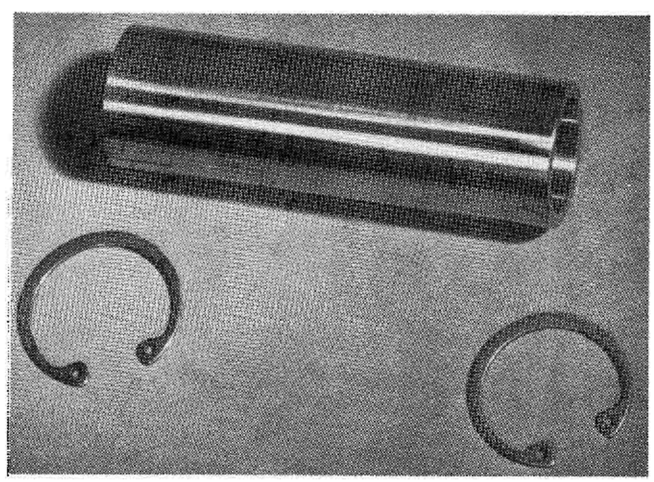

Fig. 19 - Piston pin and retaining rings

pered ID is used to minimize the weight of the piston pin. A 30 Mu in. finish is specified for the ID of the piston pin in order to avoid stress risers due to machining marks. A 1.03 diameter full floating piston pin is used for the street engine with a straight 0.685 ID.

CONNECTING ROD

The connecting rod designed for the original track engine had a distance of 6.861 between the piston pin and the crank pin bores, a machined all over small end, and an I Section of 0.35 min. square inches. The development problems of the connecting rod were mainly those associated with prevention of the inertia load of the piston and rod itself at top dead center during high rpm from distorting the bearing bore of the crankpin end and overcoming the cap bolt clamping force.

The bearing load diagram (Fig. 20) shows that the 813 gr piston plus the other parts that make up the reciprocating weight of the 1966 track engine impose a load of 16,000 lb that must be resisted once every other engine revolution with the engine running 7200 rpm. This load tends to elongate the bore in the direction of the long axis of the connecting rod while reducing the bore size across the parting line between the rod and cap. This reduction of the parting line axis is accompanied by a bending inward of the bolt at the parting line and the adjacent structure of the rod and cap. Any appreciable bending of the bolt, however, results in localized high stress conditions that cause it to fail prematurely.

The initial design (Fig. 21, upper left) was thought to be robust enough with its two 7/16 bolts loaded to 16,300 lb tension apiece. The tension loading was determined by using a micrometer to measure a bolt stretch of 0.008/0.0085 from the top of the head to the threaded end extending beyond the nut. Since the bolts are loaded very close to their yield point, the normal torque versus bolt tension could not be used due to variation of friction between the nut and the bolt threads, and the friction between the nut face and the nut seat in the cap. Failures of the rod bearings occurred with the initial design. The first fix used was that of tapering the shell thickness at the parting line to provide additional clearance across the parting line to compensate for the closing in of the bore. This change made the rod assembly satisfactory enough for the 1964 and 1965 track and drag engines and for the street engines. Sporatic failures under engine operation beyond 7000 rpm continued to occur so development continued.

The first redesign (Fig. 21, upper right) explored to improve the rod bore condition was a design using screws threaded into the rod for attaching the cap rather than the through bolts. The screw size was increased to 1/2-20 and the dimension across the parting line was increased by 0.24 to 3.96. The theory behind this parting line increase was that inward bending of the screw under the inertia load at tdc could be reduced and the screw would be subjected more to pure tension if the outermost contact point between the rod and the cap were moved further away from the screw.

Laboratory tests of the threaded rod design disclosed early failure of the screws at the first threads. A number of design variations of the screw design were attempted but the early fatigue failure could not be overcome. Evidently the remaining deflection of the big end caused the screw still to be subject to bending and the screw being shorter than the bolt together with the notch effect of the threads caused even higher stress concentrations on the screw than on the bolt.

A second through bolt design was then prepared (Fig. 21 lower). The bolt was increased to 1/2-20 and the foot of the rod increased again to 4.12 (future increases will be limited by the 4.25 bore through which the rod must pass). A bolt stretch of 0.0095-0.010 is specified which results in 18,200 lb tension in each bolt. At 0.010 stretch unit tensile

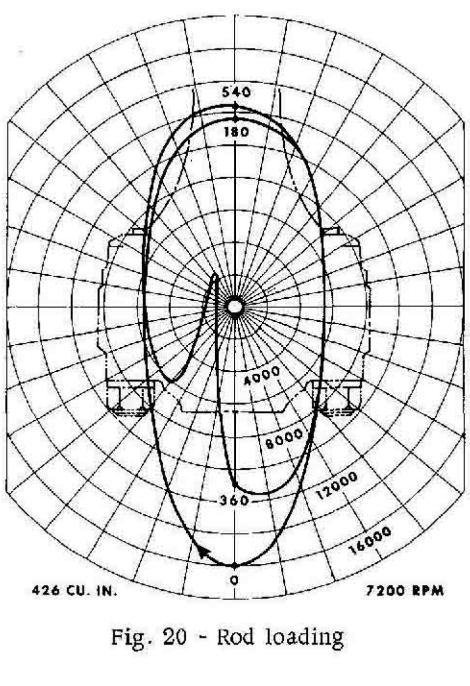

Fig. 20 - Rod loading

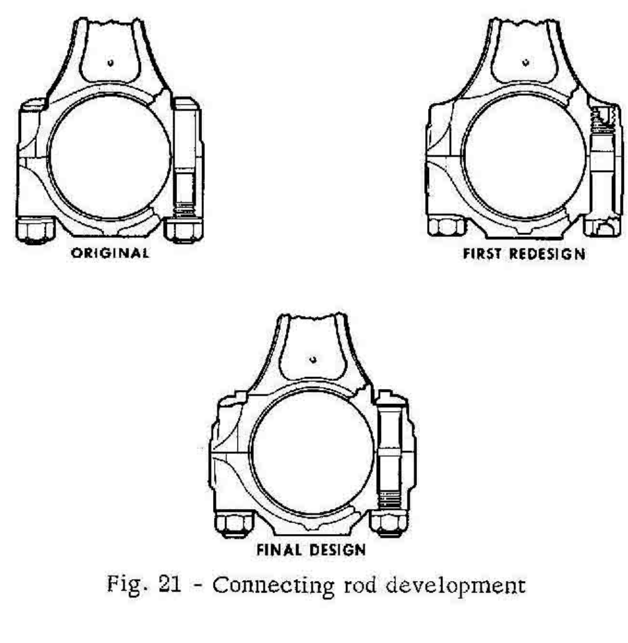

Fig. 21 - Connecting rod development

loading on the bolt is 148,000 psi. Bolt yielding starts at 0.011 stretch. The bolts are made from SAE 8640 steel.

The 1/2 in. through bolt rod design is used in the 1966 track engines. The 426 cu in. engine uses a rod with 7.061 centers (Fig. 22) and the 404 cu in. engine uses a rod with 7.174 centers.

CRANKSHAFT

The street, drag, and 426 cu in. track engines use a forged steel crankshaft with a 2.75 diameter main journal, a 2.375 diameter crankpin journal, and a stroke of 3.750 (Fig. 23).

A compound radius undercut fillet is used between each journal and the connectors. This permits the journals to be ground to a true cylinder within 0.0005 taper and 0.00025 hourglass while the larger radius portion of the fillet (0.07 at the mains and 0.06 at the crankpins) distributes the surface stress to an acceptable level. The crankshaft flange uses eight 1/2-20 UNF screws for attaching the flywheel or torque converter drive plate.

Both main journals and crankpin journals were originally finished to 15 Mu in. but in the course of development this was reduced to 5 Mu. This was required to prevent oil film breakthrough under the high inertia loadings of the higher speeds. The oil drilling in the crankshaft is very conventional with each rod bearing being lubricated by a single 1/4 in. diameter hole which intersects the surface of the main journal. A circumferential groove is cut into the center of the main bearing shells so that the rod oil holes have a constant supply. During development problems with connecting rod bearings, the oil supply became a subject of investigation. Cross drilled holes in both the crankpins and the mains, and oil reservoirs using hollow crankpins were tried. The results were inconclusive. The parts that were developed -- a more rigid connecting rod big end to maintain a rounder bearing bore, an increase in the clamping pressure of the connecting rod cap bolts, and reduction in the microinch finish of the crankpins -- together with a very careful journal and fillet inspection of each crankshaft selected for competition use appear sufficient for dependable race usage.

The crankshafts are heat treated prior to machining, then machined with the exception of the finish grind of the journal surfaces. At this point the crankshafts are shot peened all over. The journals are then ground. Following this, the entire crankshaft is surface hardened by a nitride immersion process. Finally, the journals are lapped, with care being taken to not lap through the nitride layer on the journals.

The 404 cu in. track engine uses a crankshaft similar to the 426 engine except that the common forging is machined to a 3.558 stroke.

BEARINGS

Tri-metal type of bearings are used for both the mains and rods. These bearing shells have a steel backing layer with a bronze inner layer and lead-tin alloy face layer. Selective assembly is used to obtain a main bearing clearance of 0.002-0.003 and a rod bearing clearance of 0.0025-0.0035 on the track engines. Selective assembly is also used to obtain a clearance of 0.0015-0.0025 on both mains and rods of the street engine.

VIBRATION DAMPER

The track and drag engines use a vibration damper with a 9.5 lb steel inertia member (Fig. 24). This annular inertia

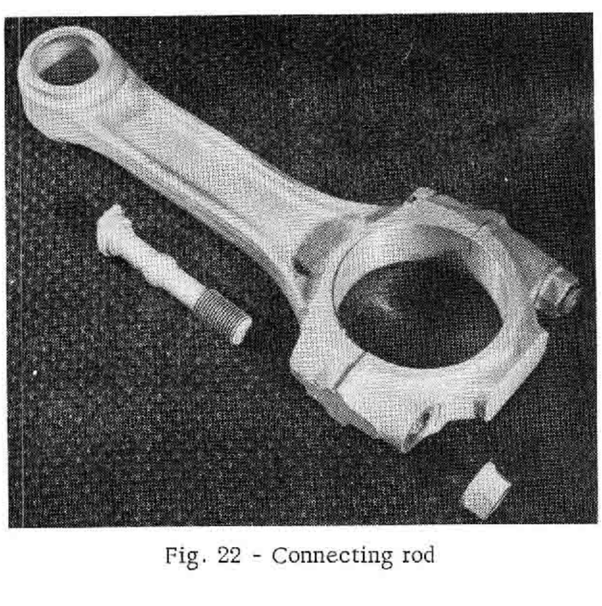

Fig. 22 - Connecting rod

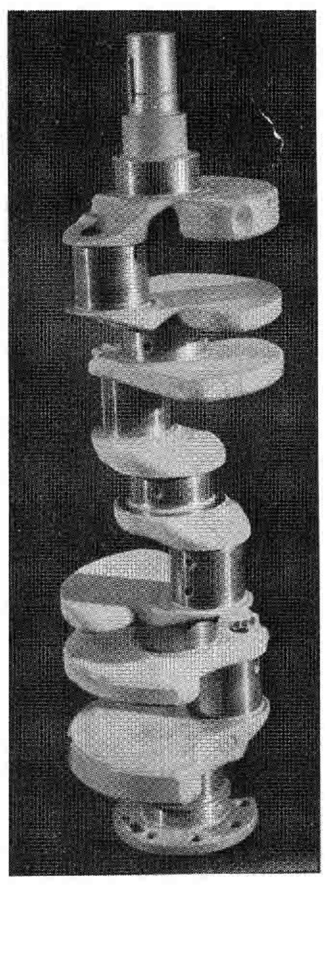

Fig. 23 - Crankshaft

member dampens the crankshaft torsional vibrations by motion through an annular rubber ring which is compressed between the inertia member and the vibration damper hub. The damper has a tuned frequency of 280 cps. A safety lip is provided on the hub to contain the inertia member against the chain case cover should a rubber failure occur. The timing mark is extended down the face of the damper onto the hub section so that any rotation of the inertia member relative to the hub can be easily determined. The street engine vibration damper is identical to the track and drag except that malleable iron is used optionally with steel as the inertia member material.

VALVE TRAIN

The system of intake and exhaust valves, valve springs, rocker arms, push rods, tappets, camshaft, and other parts related to the valve train is designed to operate smoothly throughout the speed range and is designed to move large quantities of air in and out of the combustion chamber at high engine speeds (Fig. 25). The weight of the large intake and exhaust valves requires higher load valve springs, steel rocker arms, large diameter hollow push rods, and hollow tappets to assure a system with maximum stiffness and minimum weight so that valve false motion will not occur within the normal engine speed range. Placing the intake valve at 35 deg from the bore axis and the exhaust valve at 23 deg from the bore axis requires a double rocker shaft arrangement for the rocker arms.

The intake valve has a cone shaped head of 2.25 diameter with a 0.309 diameter solid stem (Fig. 26). The conical surface of the upstream side of the valve head was determined by airflow testing. The head thickness is 0.135 and the seat angle is 45 deg. Valve stem scuffing was a development problem that was solved by increasing the stem clearance and chrome plating the stems. Oil control for the intake valve stems is provided by seals mounted to the top end of the valve guides in the cylinder head.

The intake valve used in the aluminum cylinder head has a head diameter of 2.23. This 0.02 difference is required because of the valve seat insert used with the aluminum cylinder heads. The intake valves are made of silchrome XB steel and use a single bead lock.

The exhaust valve has a cone shaped head with a solid 0.308 diameter chrome plated stem. The head diameter is 1.94. A single bead lock is also used on this valve.

The valve material is 21-4N chrome-manganese steel with a stellite faced seat surface.

Dual valve springs with dampers are used on both intake and exhaust valves. Spring open loads of 384 lb are used on the track and drag engines and 275 lb on the street engine. The installed height of the outer spring is 1.86 and its ID is 1.09. The spiral spring damper is located between the inner and outer springs and has an interference fit with both springs.

The aluminum cylinder head uses seat inserts for both the intake and exhaust valves, valve stem guides, and steel shims for the valve spring seats. The intake seat insert is

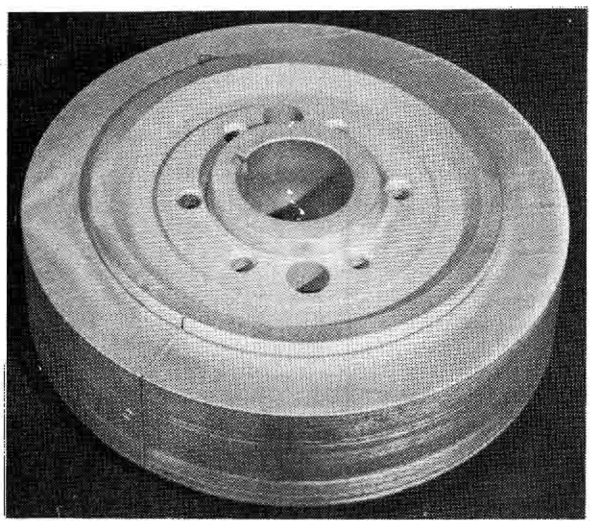

Fig. 24 - Vibration damper

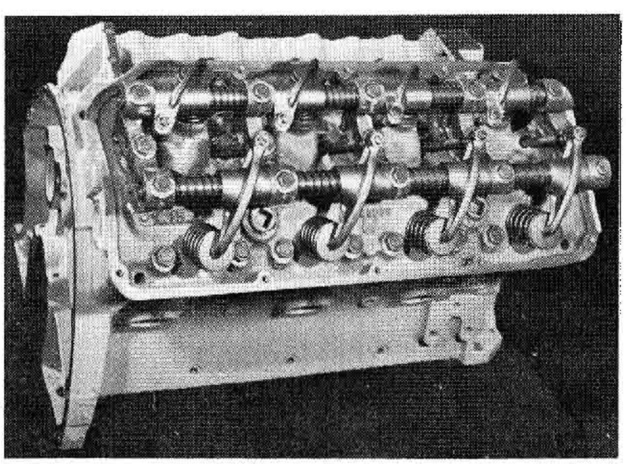

Fig. 25 - Valve train components

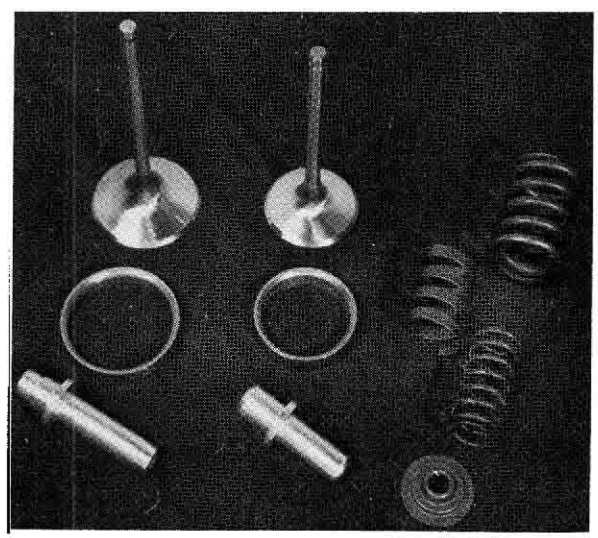

Fig. 26 - Valves, valve seats, springs, and guides

made of cast iron and the exhaust seat insert is made of steel. Both intake and exhaust valve stem guides are made of cast iron. A shrink fit of 0.005 is used for the intake valve seat insert and 0.0045 for the exhaust valve seat insert. The valve stem guides have a 0.002 shrink fit.

The installation of both the seats and the guides is accomplished by heating the cylinder head to 300 F and cooling the seats and guides in liquid nitrogen.

ROCKER ARMS

Both intake and exhaust rocker arms are made from forged steel with a full length steel backed bronze bushing pressed into the bore (Fig. 27, right and left). The bushing bore is finished after assembly. The valve tip pad is hardened and ground to 30 Mu in. maximum. 1/8 drilled holes route oil from the rocker shaft supply to the valve tip and the push rod end of the rocker arms. The adjusting screw and its tapped hole have 3/8-24 UNF class 2 threads. The lock nut has 3/8-24 UNF-3B threads. Twenty-five foot pounds of torque are used on the nut to lock the screw.

ROCKER SHAFTS

0.872 OD x 0.60 ID steel tubing is used for the four rocker shafts. 1/8 drilled holes lead oil from the ID of the shaft to the surface at each rocker arm. A diagonal groove cut in the shaft at the exit of the 1/8 hole distributes the oil along the length of the rocker arm bushing. The original 0.001 diametral clearance to the rocker arm was increased to 0.0022 after several scuffing failures occurred. This clearance has proved satisfactory. A light helical spring is used on the shaft to keep the rocker arms in place against the adjacent brackets where the thrust of the push rod is resisted. The rocker shaft is hardened at the location of each rocker arm.

ROCKER SHAFT BRACKETS

The malleable iron rocker shaft brackets are attached to the cylinder heads by using five of the cylinder head screws. A solid dowel is used to locate accurately each rocker bracket. 3/8-16 screws locate and clamp the rocker shafts to the brackets.

In the interests of reducing engine weight, aluminum brackets were also tested. A slight reduction in valve float speed resulted, however, so the material remained unchanged.

Holes are drilled in the rocker shaft brackets to lead lubricating oil from the rocker shaft bracket mounting surface to each of the rocker shafts.

VALVE TAPPET AND PUSH RODS

A hollow fabricated tappet is used with an extruded steel body brazed to an iron face (Fig. 28).

The face of the tappet is flat. A clearance of 0.0015-0.0028 is used between the tappet bore in the cylinder block and the tappet body so that the tappet can tip slightly in its bore to obtain a line contact rather than a point contact with the camshaft lobe. The camshaft lobes are tapered to cause tappet rotation. The tappet body diameter is 0.903-0.9035.

The push rods are made of hollow steel tubing with hardened steel inserts pressed and welded to the ends. The tubing has an OD of 0.375 and a wall thickness of 0.083. A 12 deg taper section is used to reduce the OD of the tube to a 0.33 diameter at each end. The first tubes had a short cylindrical section of 0.33 diameter instead of the taper but collapsing of this section took place under prolonged operation. Use of the taper overcame the problem. A 65 maximum microinch finish is maintained on the rubbing surfaces of the steel inserts.

CAMSHAFT

The development of the valve operating characteristics together with the intake and exhaust manifold tuning largely determines what can be obtained from any given engine.

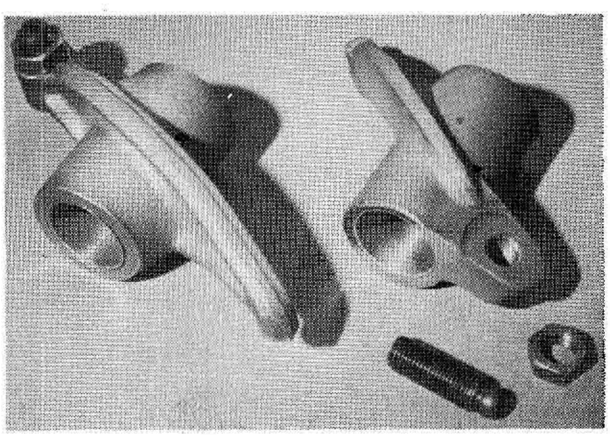

Fig. 27 - Rocker arms

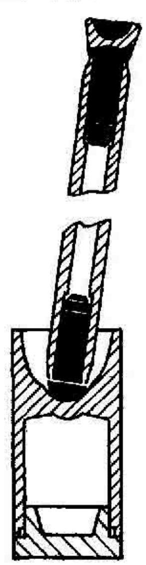

Fig. 28 - Tappet and push rod

The track and drag engines require high speed power, so long duration high overlap camshafts were accordingly tested. A spectrum of durations above 300 crankshaft degrees and a spectrum of phasing of the valves relative to the piston motion were explored. The camshaft used in the 1964 season had an intake and exhaust valve duration of 312 deg with an overlap of 88 deg and a lift of 0.54 (Fig. 29). Further development brought the duration up to 328 deg with a 112 deg overlap and lift of 0.565 for the 1966 season. The use of the street engine required the engine to be driven at lower speeds with considerably less requirement for high speed power. Accordingly a camshaft giving a 276 deg duration for each of the valves, 52 deg overlap, and a lift of 0.46 is used which is still considerably higher than a 256-duration 0.42 lift standard cam. A comparison of the valve acceleration diagrams of the 1966 track engine and the street engine (Fig. 30) shows that the longer duration camshaft has a longer period of positive acceleration with a lower peak acceleration level.

The camshaft is made of hardenable cast iron and is supported by five bearings in the cylinder block 5.15 in. above the crankshaft centerline. The bearing shells are babbitt on steel. An eccentric for the fuel pump push rod is ground in the camshaft behind the number one journal. A gear for driving the distributor and oil pump is cut in the camshaft behind the fuel pump eccentric. All wearing surfaces except the gear are hardened.

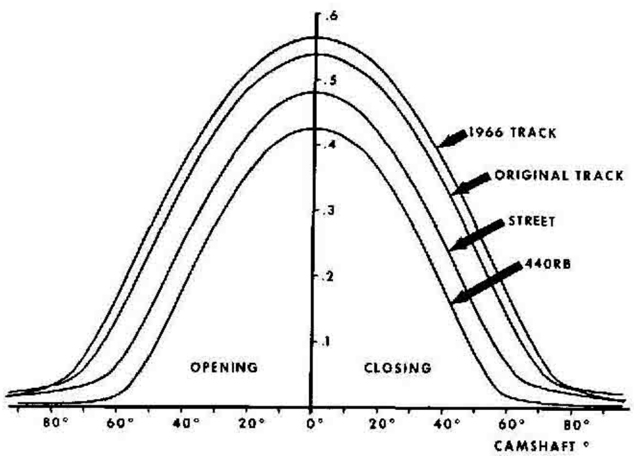

Fig. 29 - Valve lift

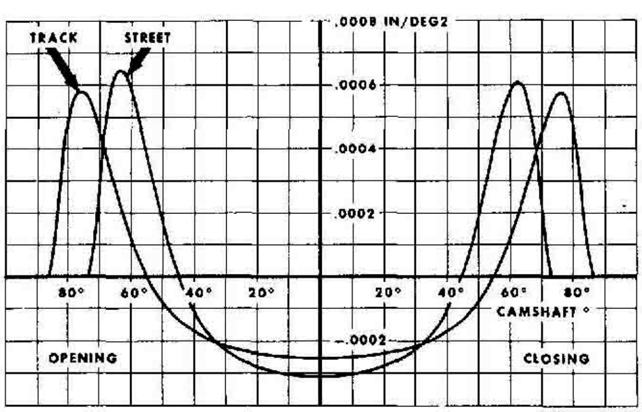

Fig. 30 - Valve acceleration

CAMSHAFT DRIVE

A double roller timing chain is used to drive the camshaft on the track, drag, and street engine (Fig. 31).

The chain is 0.38 pitch and 66 links long. The crankshaft sprocket is made of steel with 25 teeth and has a pitch diameter of 3.007. The camshaft sprocket is made of cast iron and has 50 teeth with a pitch diameter of 6.014. The camshaft sprocket was originally attached to the nose of the camshaft by a single 7/16-14 screw with a heavy washer. Occasional screw loosening failures occurred until the attachment was changed to three 3/8-16 screws.

VALVE GEAR DYNAMICS

Valve gear dynamics have been one of the most important aspects on the hemi competition engine. The present engine has operated to 8000 rpm successfully on manual transmission drag cycles and 7500 rpm with automatic transmission vehicles. The valve gear fixture is a vital part of the test program to enable the testing of higher intensity camshafts, valves, springs, and other valve train parts. The valve gear fixture is a standard hemi cylinder block with a crankshaft but without pistons. The camshaft is geared one to one with the crankshaft thus enabling the fixture to be used in rpm limited test cells. The technique allows the use of stroboscope and oscilloscope equipment to trace the valve gear motion for frequencies, surge characteristics, and vibration periods.

The same piece of equipment is used for durability evaluations of the valve train system components for sustained high speed operation prior to installation in the actual engines.

INTAKE MANIFOLDS

Track rules require the use of a single four barrel carburetor. The first track engine had a manifold design of a

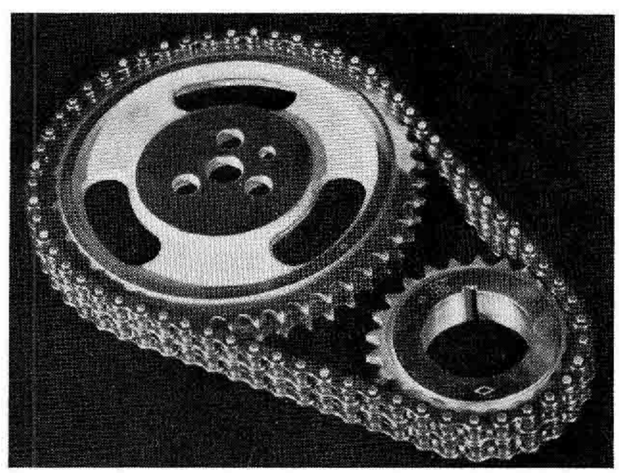

Fig. 31 - Timing chain

conventional style (Fig. 32, lower right) using a two level runner section at the center of the manifold with each level, being fed by a primary and secondary carburetor bore and each level, in turn, feeding four cylinders of the engine (Fig. 33). Various interconnecting slots were cut between the upper and lower level bores of the manifold until optimum fuel distribution and power were obtained.

This manifold had an average branch length of 11.28 in. measured from the intake valve seat to the branch opening.

The 1966 season track engine single four barrel intake manifold is shown in Fig. 32, upper left and Fig. 33. It has a lower body containing the branches (Fig. 34) and a cover section which has the carburetor mounted to it. This manifold has an intake branch length of 14.38 in. The performance development of this manifold was initiated by removing the top from a dual four barrel drag ram intake manifold and adapting a plate with a four barrel carburetor. The satisfactory power and fuel-air distribution results indicated further investigation was warranted. The ensuing period of time was then devoted to working with a fabricated and then a prototype cast manifold. The fuel mixture distribution was inherently uneven and unstable due to the carburetor pad being necessarily lower due to hood height specifications and due to the large open plenum chamber in which the mixture was introduced.

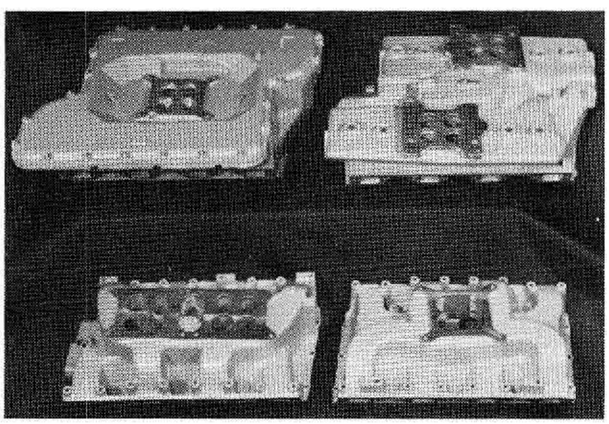

Fig. 32 - Intake manifolds

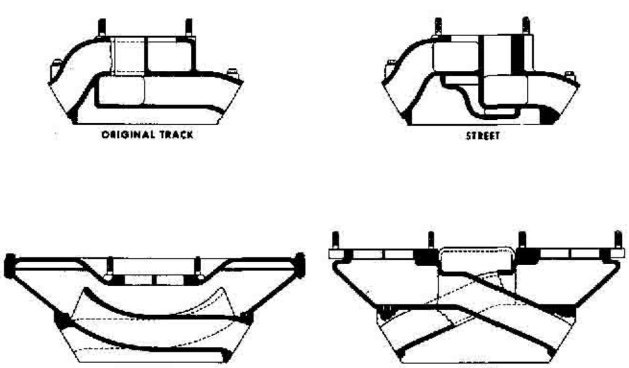

Fig. 33 - Intake manifold sections

The final version owed its success to the installation of the two outer ribs which, in a fashion, formed runners in each side of the plenum. The fuel-air mixture traveled fore and aft similar to a standard type four barrel intake manifold. The results of this modification provided a very acceptable fuel-air distribution pattern to obtain stable operation throughout the speed range of 3200-7200 rpm. The volumetric efficiency of the system provides 102% at 5600 rpm.

It was felt that the ribs might cause an unacceptable power loss due to higher restriction, therefore, tests were run with the manifold equipped with fuel injection. The ribs were then removed and reinstalled; the results were quite satisfactory with only approximately 1% loss in power being found. The first track use of the manifold was the Daytona Beach race of Feb. 27, 1966.

The drag engine uses two four barrel carburetors (Fig. 32, upper right) because of the difference in rules. Each carburetor feeds a plenum chamber that feeds the four cylinders on the bank opposite to the carburetor. An interconnection is cast between the two plenum chambers. Aluminum was used for the 1964 drag engine intake manifold; for the 1965 model season the material was changed to magnesium for weight reduction. This drag manifold has a total branch length of 12.40 in.

The street hemi uses two four barrel carburetors mounted in a fore and aft tandem arrangement (Fig. 32, lower left). Two levels are provided in the manifold so that each level is fed by a primary and secondary bore from each carburetor. This arrangement was selected over the drag type of two four barrel intake manifolding so that staging of the carburetors could be accomplished. The street hemi required warmup heat for the floor of the manifold in order to make the engine drivable at the lower ambient temperature ranges This was accomplished by routing two steel pipes from the right hand exhaust manifold and exhaust pipe (Fig. 35).

The pipes attach to the rearmost face of the intake manifold from where the gas flows through a double floor in the intake manifold underneath the carburetor risers of the rear carburetor. A heat valve routes the gas to the manifold floor when needed.

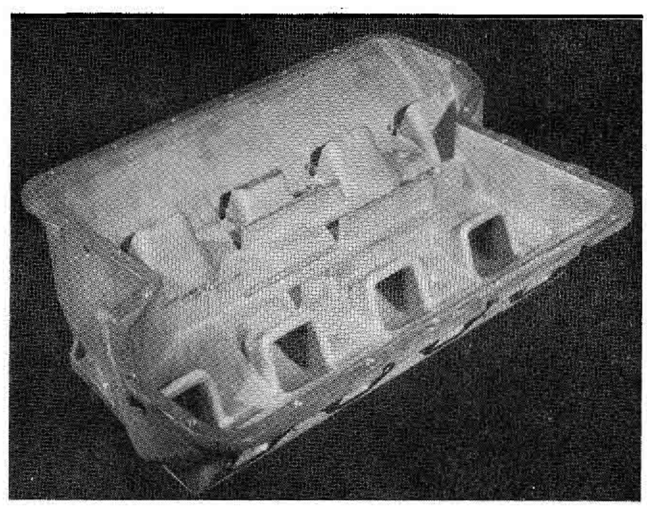

Fig. 34 - 1966 track manifold lower body

EXHAUST HEADERS

Exhaust headers made from 2 in. steel tubing welded into a steel casting were provided on the first track and drag engines. The casting bolted to the cylinder head and provided the transition from the rectangular opening in the cylinder head to the circular opening of the tube (Fig. 36, lower). Dynamometer testing showed that these engines preferred a 45 in. tube length for each exhaust valve. Each of these tubes discharged singly into a 4 in. collector.

Further development disclosed that if the four tubes were shortened to 30 in. and grouped together in a four leaf clover arrangement for discharge into the collector (Fig. 36, upper) a power improvement in the higher rpm range could be made.

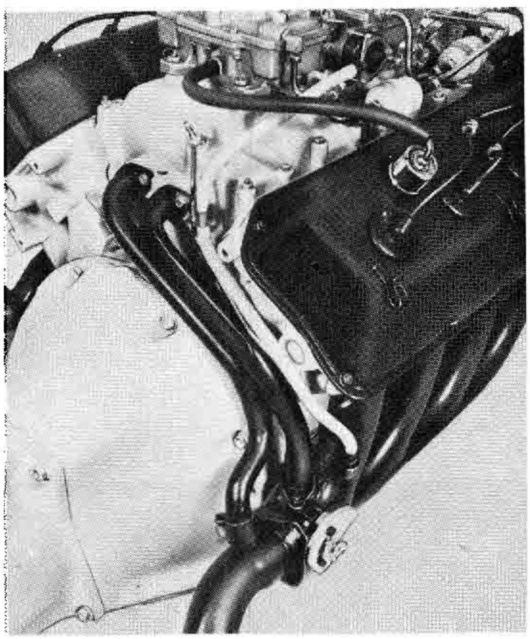

Fig. 35 - Street engine heat tubes

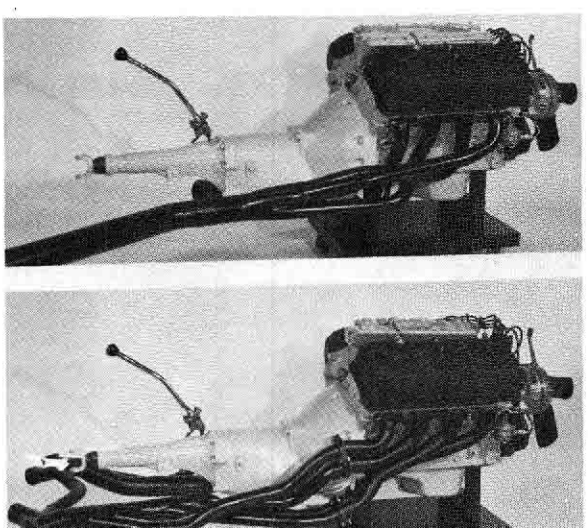

Fig. 36 - Exhaust headers

The street engine is provided with cast iron headers that are well streamlined and have a considerable branch length for each valve (Fig. 37). The anticipated use of this engine did not require the use of tuned pipes and, in addition, it was decided that normal assembly plant procedures would be used in building cars using this engine. Accordingly, the exhaust pipe flanges are located near the back of block, and the manifolds are designed to provide clearance for the body drop operation where the front end longitudinals must pass vertically over the outermost engine surface. The use of cast iron permits the heat valve to be mounted in the outlet flange of the right hand header.

OIL PUMP AND LUBRICATION SYSTEM

The oil pump is driven by a shaft located diagonally at the left front side of the cylinder block. This intermediate shaft is gear driven from the camshaft. The intermediate shaft gear is shaved. The hexagon shaped lower end of the shaft that drives the oil pump is surface hardened to maintain durability for the track engines under the higher oil pump drive load.

An increase in oil flow is desirable in the competition engines, particularly between the surfaces of the connecting rod and main bearings and their respective crankshaft journals. This increased oil flow maintains the bearing temperatures at an acceptable level and is accomplished by increasing the bearing clearances over the 0.0002/0.0022 used on standard passenger car engines.

Work even prior to the hemi disclosed that a very considerable increase in pump capacity could be obtained by lowering the restriction on the suction side of the pump. This was accomplished by using a larger than standard pump suction pipe located in the normal location and adding a second suction pipe (Fig. 38, lower left).

The pump is externally mounted and oil normally enters from the suction pipe through a drilled hole into the block side of the pump. This suction pipe was increased from 0.50

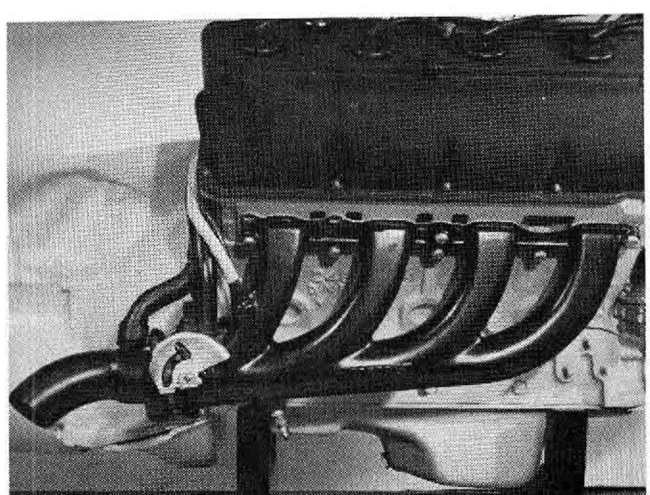

Fig. 37 - Street engine exhaust manifold

to 0.64 ID and the drilled holes were increased from 0.547 to 0.688 which resulted in a 63% pump capacity increase. The second inlet was accomplished by fastening another pipe in the oil pan and having it pass through the wall of the pan. This oil was then led to the cover of the pump by a heavy wall rubber hose. A pump cover casting change was required and a drilled hole was added to provide an oil passage leading from the cover to the suction side of the rotors. This resulted in an additional 62% flow improvement.

This double suction system has been used on both track and drag engines. Another pump suction system (Fig. 38, lower right) developed particularly for road racing applications uses a swinging strainer that is free to rotate about a vertical pivot. The pivot consists of a hollow tube rotating vertically inside a reamed hole in a fixed block of steel. Two oil lines, one internal and one external, lead the oil from the fixed block to the suction side of the pump. The original swinging strainer occasionally became stuck on dead center. This would result in the strainer being at the opposite end of the pan from the oil supply. A weight added to the strainer gave some improvement in reliability to the strainer. However, the final solution came with an oil vane welded to the strainer at a slight angle to the symmetrical centerline. The moving oil gives an initial push to the strainer and guides the strainer to the location of the deepest oil supply.

The single large suction pipe is used in the street engine (Fig. 38, upper) because its oil requirements do not have the same severe conditions as do the competition engines.

Fig. 39 compares a standard oil pump (left) with a track engine oil pump (right). High pressure oil from the pump is routed to a paper element full flow filter mounted to the oil pump cover. In the drag and street engines the oil is next routed to the right hand oil gallery in the cylinder block. In the track engine, oil is routed from the filter through the external tubes shown to an oil cooler mounted to the front end sheet metal, where adequate air circulation past the cooler is obtainable. The cooled oil is then also routed to the right hand oil gallery in the cylinder block. Oil is led from this gallery to the main bearings by 1/4 drilled holes. The camshaft journals are lubricated by vertical holes drilled from the main bearings to the camshaft bearings. The right hand gallery intersects the right side tappet bores. The left side tappet bores are lubricated by a left oil gallery that is connected to the right hand gallery by drilled holes at the rear of the cylinder block.

The number four camshaft journal has both two intersecting holes and a shallow circumferential groove to feed oil to two holes that lead from the camshaft bearing surface to each head face of the cylinder block. The clearance holes for the cylinder head bolt between the No. 5 and 7 cylinders on the left bank and the No. 6 and 8 cylinders on the right bank are used to transfer the oil through the cylinder head to the rocker shaft bracket. The rocker shafts receive the oil from the cylinder head bolt holes through more drilled holes for final distribution to the rocker arm, valve tips, and push rod sockets.

The track engine oil pan (Fig. 40, right) has a requirement that restricts it in two parameters. The first parameter is the ground clearance at the oil pan which is specified by track regulations. This establishes the sump depth. The second parameter is the pan capacity. Track experience has established that a minimum 10 qt capacity is necessary in order to permit some oil to be used during the longer races without lowering the level to a point where the strainer might be uncovered during car maneuvers. In order to provide the 10 qt capacity without having the oil level interfere with

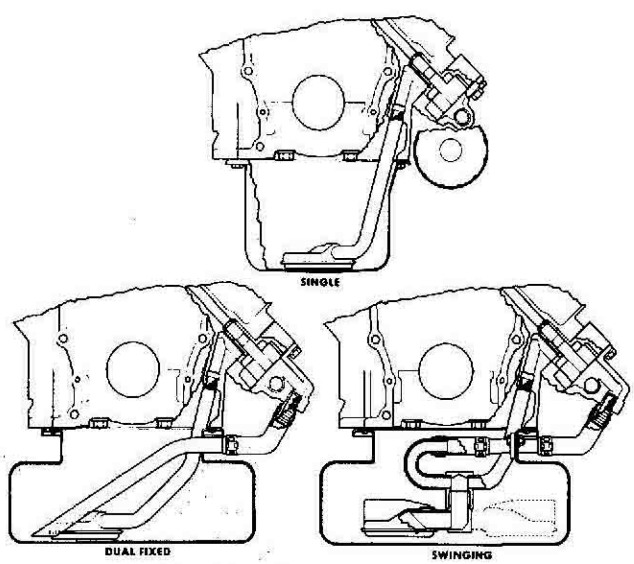

Fig. 38 - Oil suction systems

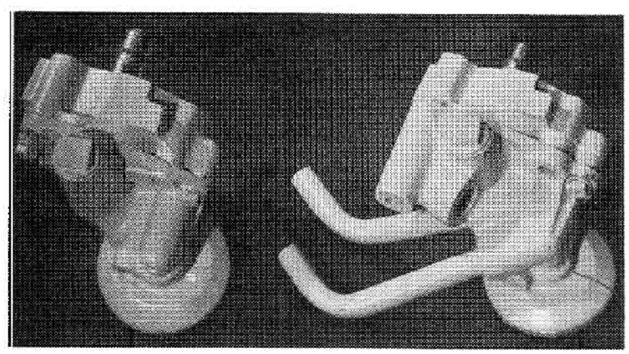
Fig. 39 - Oil pumps

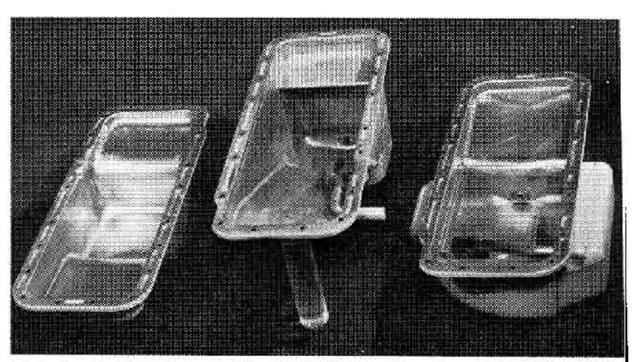
Fig. 40 - Oil pans

the connecting rod travel, an oil pan with bulges in the sides is required. Baffles are welded to the inside of the oil pan and a windage tray is bolted to underside of the main bearing caps to restrict oil movement and reduce power loss to the oil.

The drag engine is located as high above the ground as practicable so that the mass of the engine will produce a force couple during vehicle acceleration that will have the effect of adding weight to the rear wheels. This high engine location together with an absence of restrictions on the ground clearance permit the use of a very deep oil pan (Fig. 40, center). The position of the sump at the rear favors the acceleration period of the engine. The oil level is deliberately kept at a considerable distance from the connecting rod travel to avoid a very sizable horsepower loss that increases as the oil surface is raised. Baffling can reduce this loss but so far has not eliminated it.

The left pan in Fig. 40 is the pan used on the street engine. This pan has a 5 qt capacity and uses a conventional configuration.

COOLING SYSTEM

The circulation of the coolant through the engine follows basically a series flow pattern with the coolant entering the front of the cylinder block from the high pressure side of the water pump housing, flowing past all the cylinder barrels, transferring from the cylinder block to the cylinder head through large holes at the rear of the block, and then flowing forward in the cylinder heads.

Four additional holes are provided near the outboard edge of each head face of the cylinder block to flow coolant directly from the water jacket of the cylinder block to each exhaust valve seat. From the front of the cylinder heads, the coolant flows through a short passage in the cylinder block connecting from the head face to the front face of the block. An upper passage in the water pump housing leads the coolant to the thermostat for transfer to the radiator.

Other than the aluminum water pump housings used on the 1965 drag engines all engines have been built with cast iron housings which are the same as the housings used on the standard passenger car engines. The water pump impellers are modified in accordance with the engine usage. The track engine pump (Fig. 41) has a 3.32 diameter six vane impeller whereas the drag and street engine pump uses a 3.67 diameter six vane impeller. Idle cooling is not a problem on the track engine and the reduction in water pump horsepower is desirable. All the impellers are made of molded plastic with a sintered iron hub.

CLUTCH HOUSINGS

Four clutch housings are required to cover the usage of the engine. An aluminum housing (Fig. 42, left) is used where weight is a dominant factor. A cast steel housing (Fig. 42, center) with 0.3 thick wall sections and a bolted on cast steel pan is used where required by drag rules. A cast iron housing (Fig. 42, right) machined for a 10.5 in. clutch is used on the track engines with a similar housing machined for an 11 in. clutch used on the street engines.

IGNITION SYSTEM

In order to secure an accurate high voltage spark, particularly at the high engine speeds encountered in track and drag applications, a transistorized ignition system is used in conjunction with a dual breaker point distributor on all competition engines (Fig. 43).

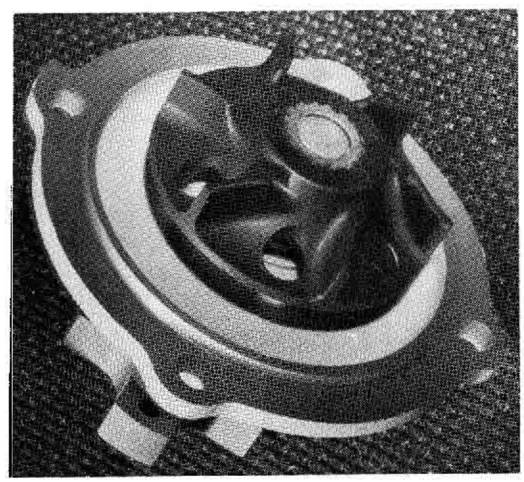

Fig. 41 - Water pump

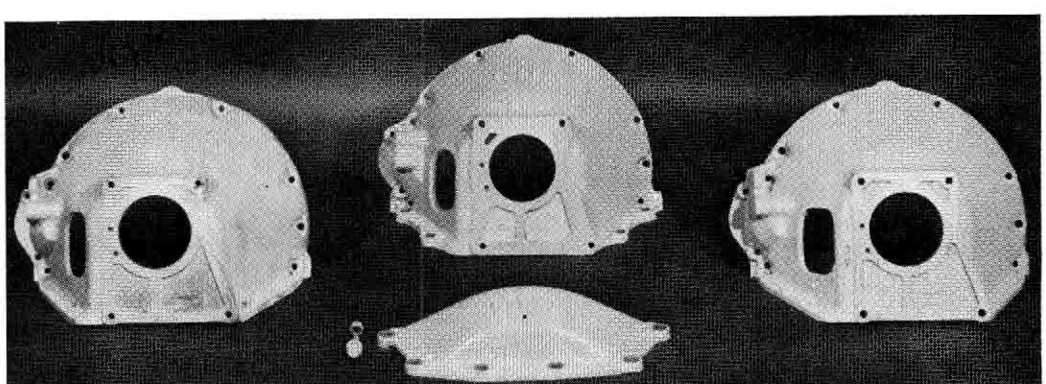

Fig. 42 - Clutch housings

In this circuit the opening of the ignition coil circuit creating the secondary voltage is accomplished through a germanium transistor. The transistor is triggered to perform this function by the opening of the distributor points. Diodes and resistors are added to the transistor circuitry to give circuit protection and to cause a very clean cutoff in the secondary circuit when the points close. The street engine uses a conventional ignition circuit in which the primary current flows to ground through the distributor points.

The track and drag engines use spark plug cables made of a 9 mm silicone jacket with an inner glass braid and a stainless steel core. The street engine uses a resistance conductor type cable with a hypalon jacket.

A long, high resistance plastic connector (Fig. 44) is attached to the spark plug end of the wire so that the wire can be snapped onto the spark plug with the plug being located at the bottom of the spark plug tube. A circular shoulder on the connector assists in centering the connector for assembly onto the spark plug.

Three-quarter inch reach 14 mm spark plugs are used in all the engines. The track engines use a cold heat range plug with the electrodes buried down into the open end of the plug. The drag engines use a spark plug with the electrodes extended to help prevent fouling under light loads. The street hemi uses a N-9Y spark plug with extended electrodes. This plug has a higher temperature heat range than do the track or drag spark plugs.

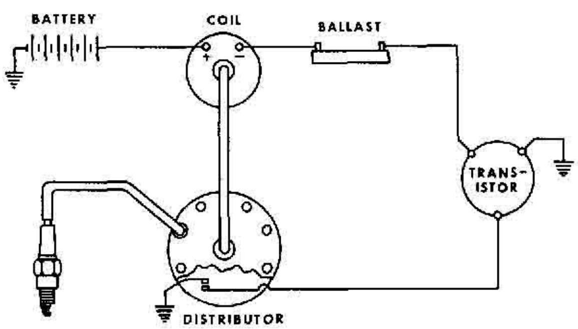

Fig. 43 - Ignition circuit diagram

AIR CLEANERS

The track engines use a 23.5 in. diameter air cleaner (Fig. 45, left) using a 16 in. OD circular paper element filter. A 62.2 sq. in. rectangular opening at the rear of the cleaner is connected to an opening in the heater plenum section of the dash panel with a flexible rubber boot. Air enters the heater plenum section of the dash panel through the louvered slots in the cowl section directly ahead of the lower edge of the windshield. Tests have shown that a positive air pressure is present at this point. The drag engines are equipped with an air cleaner (Fig. 45, right) having a single oval shaped paper element filter. Two 5.03 diameter holes in the lower support plate of the cleaner connect to the mounting flanges of the two carburetors. In actual competition the air cleaner is generally removed and replaced with two ideal entrances which are sealed against the hood opening beneath the hood scoop. All air for the engine enters the front opening of the hood scoop. The street engine uses a 19.44 diameter air cleaner cover and an 18 OD circular paper element filter (Fig. 45, center and Fig. 46) with two 4.24 diameter holes in the lower support plate for mounting the air cleaner to the carburetors. The air cleaner profile is sufficiently low so that a hood scoop is not required. For appearance reasons the street engine air cleaner is chrome plated.

CARBURETORS

The street hemi incorporates an intake manifold and carburetor combination which differs from the competition engines in several respects.

The wide range in engine flexibility required for both city driving and high-speed performance is achieved, in part, by using two specially designed four-barrel Carter carburetors connected by a staged throttle linkage.

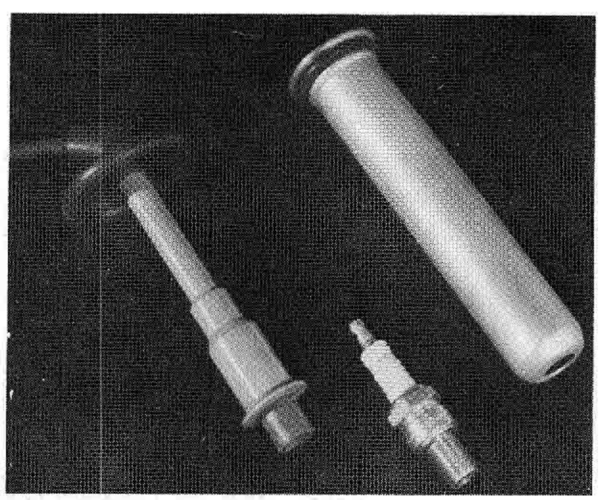

Fig. 44 - Spark plug, tube, and connector

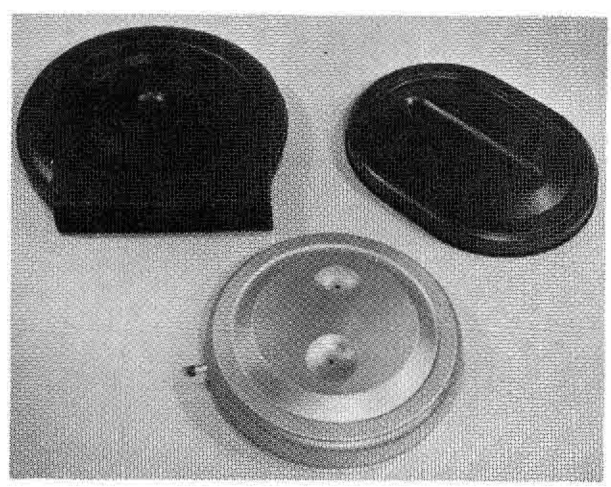

Fig. 45 - Air cleaners

The throttle bore size was selected as 1-9/16 in. primary and 1-11/16 in. secondary which allowed the engine performance to be optimum at wide open and still enable good response at part throttle conditions.

The rear carburetor is equipped with a conventional automatic choke unit. Unlike the front carburetor, it is provided with manifold heat to facilitate engine warmup.

The throttle linkage is staged (Fig. 47). This means that the primary throttle blades of the forward carburetor do not begin to open until the primary throttle blades of the rear carburetor are advanced approximately 40% or 30 deg of their travel. From this point, both sets of primary throttle blades operate together to reach wide-open position at the same time. Thus the engine runs at low and intermediate speeds on the two barrels of the rear carburetor only.

The secondary barrels of both carburetors are velocity-controlled and actuated by the intake airflow of the engine. The secondary velocity valves are weighted to remain closed until the pressure drop across the carburetor is great enough to offset the counterweights and open the secondaries.

The dual four barrel drag intake manifold incorporates the use of two 1-11/16 x 1-11/16 in. throttle bore Holley carburetors. These are high airflow capacity type carburetors that flow 770 cfm at 1.5 in. Hg for maximum airflow at high engine speeds.

The secondary blades are operated by a mechanism actuated by a diaphragm spring unit. As the airflow is increased through the primary venturi, the venturi vacuum increases which actuates the diaphragm and opens the secondary blades. The carburetor secondary opening is generally calibrated for full throttle at about 4000 rpm.

The single four barrel track intake manifold also incorporates the use of a Holley carburetor. This carburetor, as used on the closed circuit tracks, is a 1-11/16 x 1-11/16 in. throttle bore in conformance to the present rules. This is again a high airflow capacity unit which also incorporates the vacuum operated secondary system. This carburetor is equipped with center inlet high capacity fuel bowls which prevent fuel starvation on high banked race tracks. A 5 in. carburetor air horn is used with a low restriction air cleaner which obtains its air through a duct from the cowl opening of the vehicle.

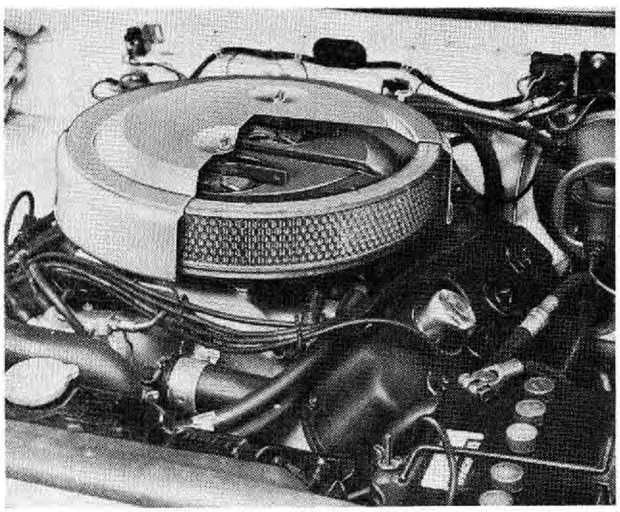

Fig. 46 - Street engine air cleaner cutaway

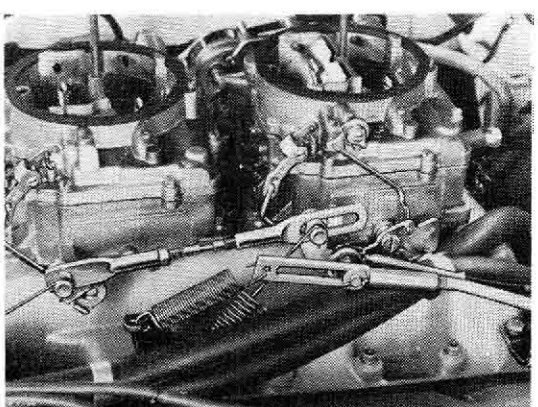

Fig. 47 - Staged throttle linkage

FUEL INJECTION

Due to the interest of AFX type drag cars for exhibition, fuel injection was a natural evolution for improved performance. The system allowed a greater flexibility for optimum ram tuning plus lower airflow restrictions than normally found with cast manifolds and carburetors.

A manifold with throttle controls was procured from Fuel Injection Engineering Co. with pump, lines, and nozzles. This led to extensive laboratory and vehicle testing of various combinations to provide an optimum package (Fig. 48).

The tuned lengths of inlet pipes were finally selected as 16.25 and 23 in. with the longer being used by the automatic

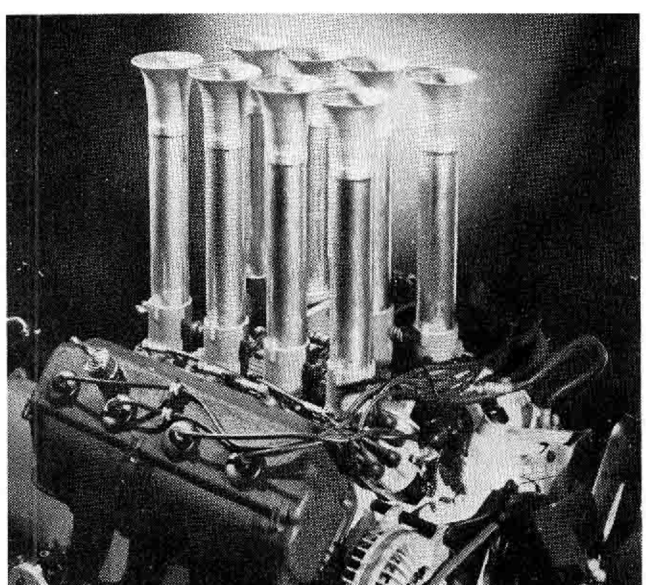

Fig. 48 - Fuel injected engine

transmission vehicles due to their requiring more torque response. Fig. 49 illustrates the final released power package available for the manual transmission. Note the horsepower increase due to the ram tuning effect of the shorter tube system.

A schematic diagram (Fig. 50) illustrates the general fuel injection system as used on our competition vehicles.

ENGINE PERFORMANCE

Fig. 51 illustrates the WOT performance characteristics of the hemi engine in various forms. The street engine is rated at 425 brake horsepower at 5000 rpm with a maximum torque of 490 lb ft at 4000 rpm.

The specific fuel rate for the engine is 0.45 lb/bhp/hr at 4000 rpm. The volumetric efficiency is 90% at 4000 and 5000 rpm.

Fig. 51 also illustrates the competition drag type engine with dual four barrel ram intake manifolds and with fuel injection. Both of these versions depict the possible potential of the engine with all optimum parts calibrated in one package. Both use higher duration camshafts of 328 deg duration and 112 deg overlap, and optimum tuned tubular header exhaust system. Various combinations of header systems have been used which can affect the torque characteristics of the engine in the lower speed ranges. The selection of camshafts for the competition engines were basically for top end performance. Many outside procured camshafts have been tested and where their application is best suited, have been recommended to the field.

The horsepower and torque curves of the 1966 track 404 cu in. displacement engine using the new four barrel plenum ram manifold is also shown.

Normal dynamometer testing in the laboratory requires a period of stabilization at each speed to record fuel, beam, temperatures, fuel-air measurement, and other essential information with the engines operated up to 7200 rpm. This type of operation is quite satisfactory from the endurance standpoint of track type engines since they are operated at relatively WOT most of the time. Many experimental mechanical parts are tested along with performance orientated items in this manner. On drag type development, however, the steady state tests are not completely valid since a drag vehicle is operated over relatively short period of time such as 10-14 sec during drag strip competition events. Tests have been initiated for drag work on an inertia wheel equipped dynamometer with the equipment to measure torque, cycle time, and other data related to acceleration operation. The results have been very useful.

Endurance schedules of track engines are simulated on the dynamometer as close as possible to the particular track in question. The track test is duplicated as a lap time, variations of load and speed during the lap cycle, and the number of pit stops required. Both 404 and 426 cu in. engines have run many successful schedules of this type and durability has been quite satisfactory.

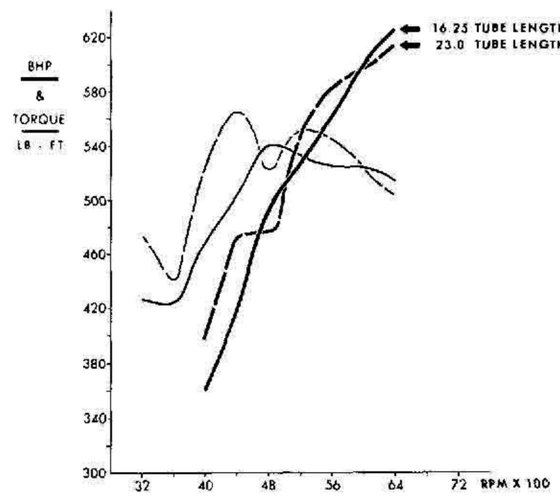

Fig. 49 - Fuel injection engine performance

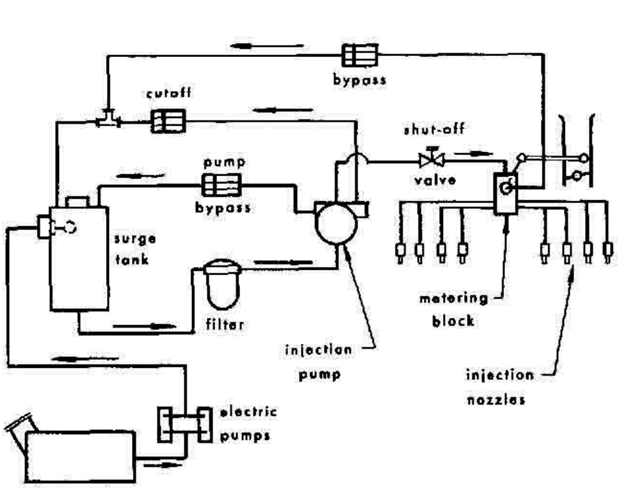

Fig. 50 - Fuel injection system

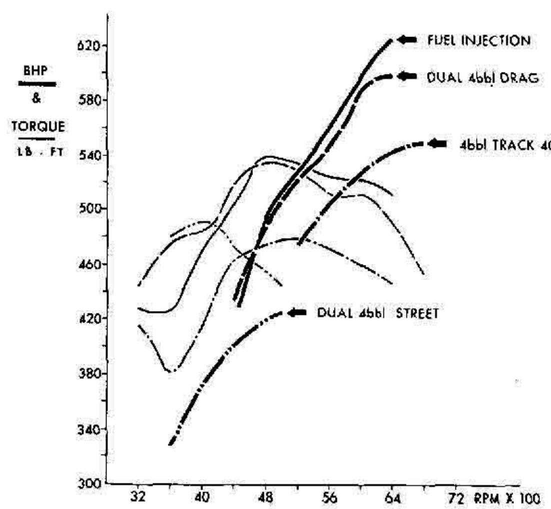

Fig. 51 - Engine performance

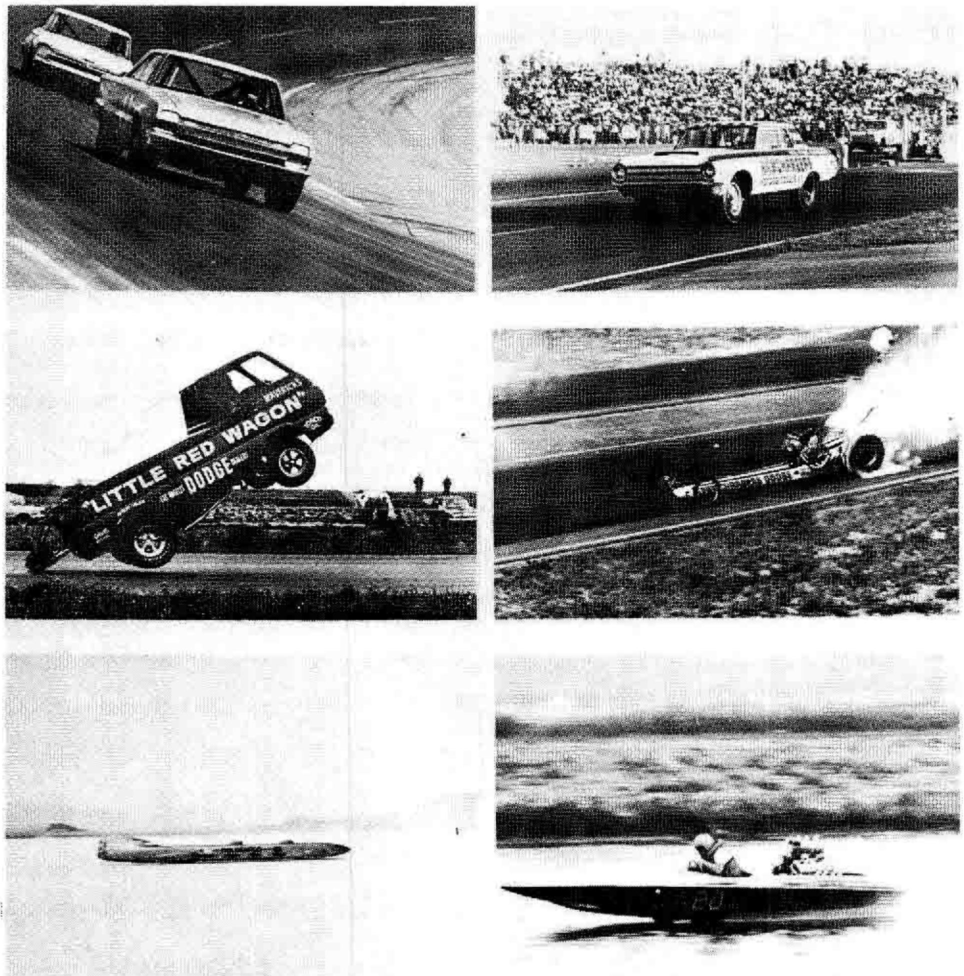

Fig. 52 - Hemi engine competition usage

CONCLUSION

Chrysler Corp.'s new hemi engine has been designed and developed to utilize fully the inherent strengths of the hemispherical combustion chamber. The variety of successful high performance usages of this engine shown in Fig. 52 attest to accomplishments made to date. The overcoming of the many problems associated with high speed operation together with the continued development for increase of specific output levels have again added to our engineering knowledge of internal combustion engines.

Appendix will be found on next page

APPENDIX A

LAND SPEED RECORD VEHICLE

The Summers' Bros. of Ontario, Calif. requested technical assistance of Chrysler Corp. in 1964 in preparation of hemi engines for use in a new land speed record wheel driven vehicle with which they hoped to exceed the existing record and bring it to the United States.

The entire design for the vehicle was based on maximum aerodynamic efficiency which necessitated an extremely low body silhouette and therefore compact engine compartments. The plans called for four essentially stock hemi engines with several important modifications. The body shell was 28 in. from the ground and required 5 in. road clearance. Since the profile was so low, it required that a fuel injected induction system be developed which would not extend above the valve covers. In addition, the road clearance specifications required an oil pan of only 2-5/8 in. depth below the cylinder block oil pan rail.

The goal of performance was set for 600 bhp on gasoline within the speed limitations of the engine. A standard Fuel Injection Engineering Co. fuel injection system was installed and several lengths of 2.06 ID intake tubes were run to verify the power required. A fabricated manifold was made which tilted the throttle bodies inboard to the centerline of engine. The tubes to be attached to the throttle bodies were interlaced due to the tight fit. This test provided sufficient information so procurement could proceed on actual cast throttle bodies and tubes (Fig. 53).

The four engines were placed in line and coupled in pairs, back to back. The tight clearance problem of the manifold housing with respect to the tube entrances was carefully checked and indicated only about 1% power loss.

Two air scoops on the hood of the vehicle supplied the ram airflow required by the four engines. The front scoop supplied the front two engines while the rear (which was offset from the front) provided the air for the rear pair.

Due to the severe space limititions of the oil sump, a dry sump was considered as the only practical system to provide the lubrication required plus keeping the friction losses to a minimum.

Several external scavenge pumps were tried but the means for driving were complicated and bulky; therefore, it was decided to use an extension of the present oil pump drive unit. The standard oil pump was known to be adequate for

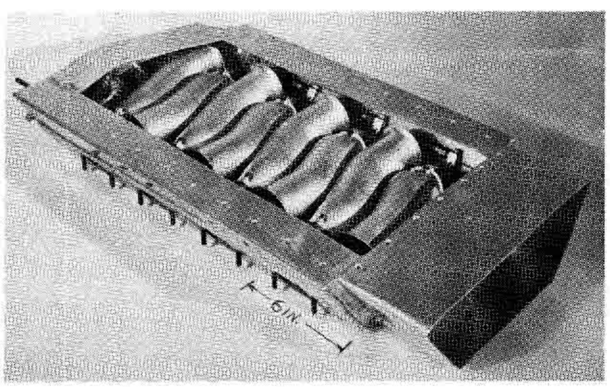

Fig. 53 - L. S. R. intake manifold

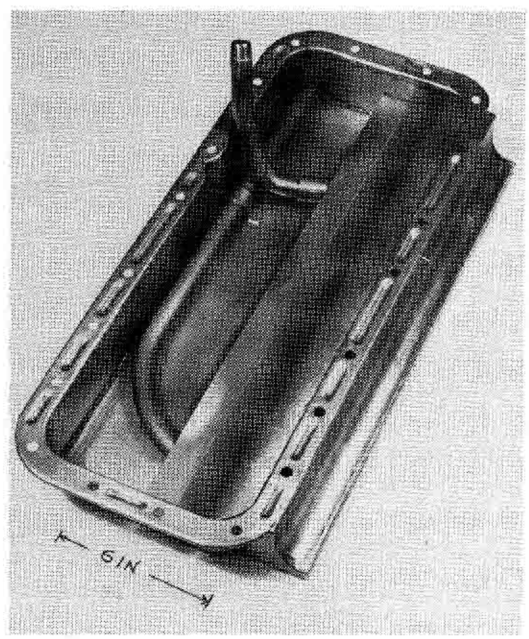

Fig. 55 - L. S. R. oil pan

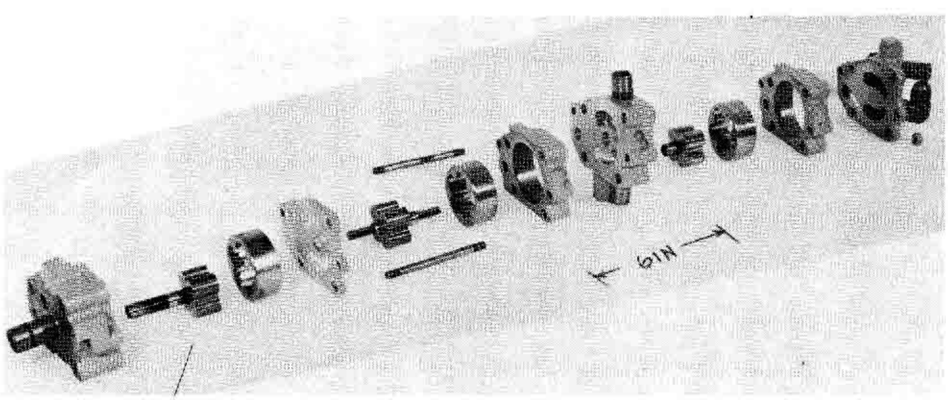

Fig. 54 - L. S. R. oil pump

the lubrication requirements but would require a scavenge unit. This was provided by adding two standard rotor units to the standard pump with a spacer, such that each rotor unit worked as a separate scavenge pump as to suction and had a common outlet to the surge tank. Each scavenge pump had its own pickup in the oil pan, one forward and the other to the rear, to ensure coverage of the pickups during acceleration and deceleration. Each engine had its own surge tank which was of various configurations as dictated by the vehicle space limitations.

The surge tanks were connected by two 1/2 in. ID lines to the pressure side of the pump, one in the pump cover and the other to a special aluminum spacer housing. A remote filter was connected to the pump cover for full flow filtering prior to the oil entering the engine. Fig. 54 illustrates the exploded view of the pump.

The oil pan was of special interest due to its tight vehicle environment. The right side had an extension built on to allow the oil from windage to collect. A special scrapper type baffle was installed to also aid in reducing oil whip back up on the right side. Fig. 55 illustrates the pan as released. The scallop on the extension was for chassis drive clearance.

On Nov. 12, 1965 the vehicle was driven to a two way record of 409.277 mph on the Bonneville salt flats of Utah, thus accomplishing the desired objective.

APPENDIX B

COMPARATIVE ENGINE SPECIFICATIONS

	1964-1965 Track	1966 Track	1964 Drag	1965 Drag	1966 Street
Displacement	426	426-404	426	426	426
Bore	4.25	4.25	4.25	4.25	4.25
Stroke	3.75	3.75-3.558	3.75	3.75	3.75
Comp. ratio	12.5	12.5-12.0	12.5	12.5	10.25
Cylinder block	Cast iron stress relieved	Cast iron stress relieved	Cast iron stress relieved	Cast iron stress relieved	Cast iron stress relieved
Bearing caps	Mall. iron tie bolted	Mall. iron tie bolted	Mall. iron tie bolted	Mall. iron tie bolted	Cast iron tie bolted
Crankshaft	Forged steel shot peened and nitride hardened 15 Mu in. journals	Forged steel shot peened and nitride hardened 5 Mu in. journals	Forged steel shot peened and nitride hardened 15 Mu in. journals	Forged steel shot peened and nitride hardened 15 Mu in. journals	Forged steel shot peened and nitride hardened 15 Mu in. journals
Main bearings	Trimetal	Trimetal	Trimetal	Trimetal	Trimetal
Main journal dia.	2.75	2.75	2.75	2.75	2.75
Crankpin dia.	2.375	2.375	2.375	2.375	2.375
Piston	Impact extruded aluminum	Impact extruded aluminum	Impact extruded aluminum	Impact extruded aluminum	Impact extruded aluminum
Weight, gm	852	813	852	848	843
Top of skirt to bore clearance		0.008			0.003
Piston pin offset	0.060 toward minor thrust side	0.000	0.060 toward minor thrust side	0.06 toward minor thrust side	0.06 toward major thrust side
Piston pin OD	1.0936	1.0936	1.0936	1.0936	1.0311
ID	0.751	0.75 0.86 taper	0.751	0.751	0.685
Type	Pressed	Floating	Pressed	Pressed	Floating
Connecting rod	Forged steel	Forged steel	Forged steel	Forged steel	Forged steel
Centers	6.861	7.061-426 7.174-404	6.861	6.861	6.861
Intake valve	Silchrome XB	Silchrome XB	Silchrome XB	Silchrome XB	Silchrome XB
Head diameter	2.25	2.25	2.25	2.23	2.25
Stem diameter	0.309 solid	0.309 solid	0.309 solid	0.309 solid	0.309 solid
Stem finish	Chrome	Chrome	Chrome	Chrome	Chrome

(con't)

Appendix B (con't)	1964-1965 Track	1966 Track	1964 Drag	1965 Drag	1966 Street
Exhaust valve	21-4N	21-4N	21-4N	21-4N	21-4N
Head diameter	1.94	1.94	1.94	1.94	1.94
Stem diameter	0.308 solid	0.308 solid	0.308 solid	0.308 solid	0.308 solid
Stem finish	Chrome	Chrome	Chrome	Chrome	Chrome
Valve springs installed height					
Outer	1.86	1.86	1.86	1.86	1.86
Inner	1.64	1.64	1.64	1.64	1.64
Valve closed load					
Outer	85	85	85	85	105
Inner	40.5	40.5	40.5	40.5	50
Valve open load					
Outer	280	288	272	280	184
Inner	94	96	92	94	91
Wire size - Outer	0.216	0.216	0.216	0.216	0.187
Inner	0.128	0.128	0.128	0.128	0.128
Water pump body	Cast iron	Cast iron	Cast iron	Cast iron	Cast iron
Impeller dia.	3.32	3.32	3.67	3.67	3.67
Water pump housing	Cast iron	Cast iron	Cast iron	Aluminum	Cast iron
Oil pump body	Cast iron	Cast iron	Cast iron	Aluminum	Cast iron
Oil pump cover	Cast iron with cooler tubes	Cast iron with cooler tubes	Cast iron	Aluminum	Cast iron
Oil suction pipe	Dual-fixed and swinging	Dual-fixed and swinging	Single	Single	Single
Intake manifold type	Aluminum conventional single 4 bbl	Aluminum plenum-ram single 4 bbl	Aluminum plenum-ram dual 4 bbl	Magnesium plenum-ram dual 4 bbl	Aluminum two level tandem 4 bbl
Manifold heat	None	None	None	None	Exhaust gas
Exhaust headers	Steel casting and tubes	Plate and tubes	Steel casting and tubes	Plate and tubes	Cast iron manifolds
Carburetors	Single Holley	Single Holley	Dual Carter	Dual Holley	Dual Carter
Choke			Manual	Manual	Automatic hot air
Rod bolts	7/16-20	1/2-20	7/16-20	7/16-20	7/16-20
Bolt load	0.008/0.0085 stretch	0.0095/0.010 stretch	75 ft lb	75 ft lb	75 ft lb
Cylinder head	Cast iron machined hemisphere	Cast iron machined hemisphere	Cast iron machined hemisphere	Aluminum machined hemisphere	Cast iron machined hemisphere
Chamber radius	2.42	2.42	2.42	2.42	2.42
Chamber depth	1.34	1.34	1.34	1.34	1.34
Chamber volume	172.7	172.7	172.7	170.4	172.7
Camshaft	Hardenable cast iron	Hardenable cast iron	Hardenable cast iron	Hardenable cast iron	Hardenable cast iron

(con't)

Appendix B (con't)

	1964-1965 Track	1966 Track	1964 Drag	1965 Drag	1966 Street
Cam sprocket attachment	Single 7/16-14	Three 3/8-16	Single 7/16-14	Single 7/16-14	Three 3/8-16
Timing chain	Double roller	Double roller	Silent	Double roller	Double roller
Valve Events (crankshaft degrees)					
Intake duration	312	328	300	312	276
Intake max open	112 atdc	106.5 atdc	114 atdc	112 atdc	106 atdc
Exhaust duration	312	328	300	312	276
Exhaust max open	112 btdc	109.5 btdc	110 btdc	112 btdc	114 btdc
Overlap	88	112	76	88	52
Intake valve lift	0.54	0.565	0.52	0.54	0.48
Exhaust valve lift	0.54	0.565	0.52	0.54	0.46
Rocker ratio					
Intake	1.57	1.57	1.57	1.57	1.57
Exhaust	1.52	1.52	1.52	1.52	1.52