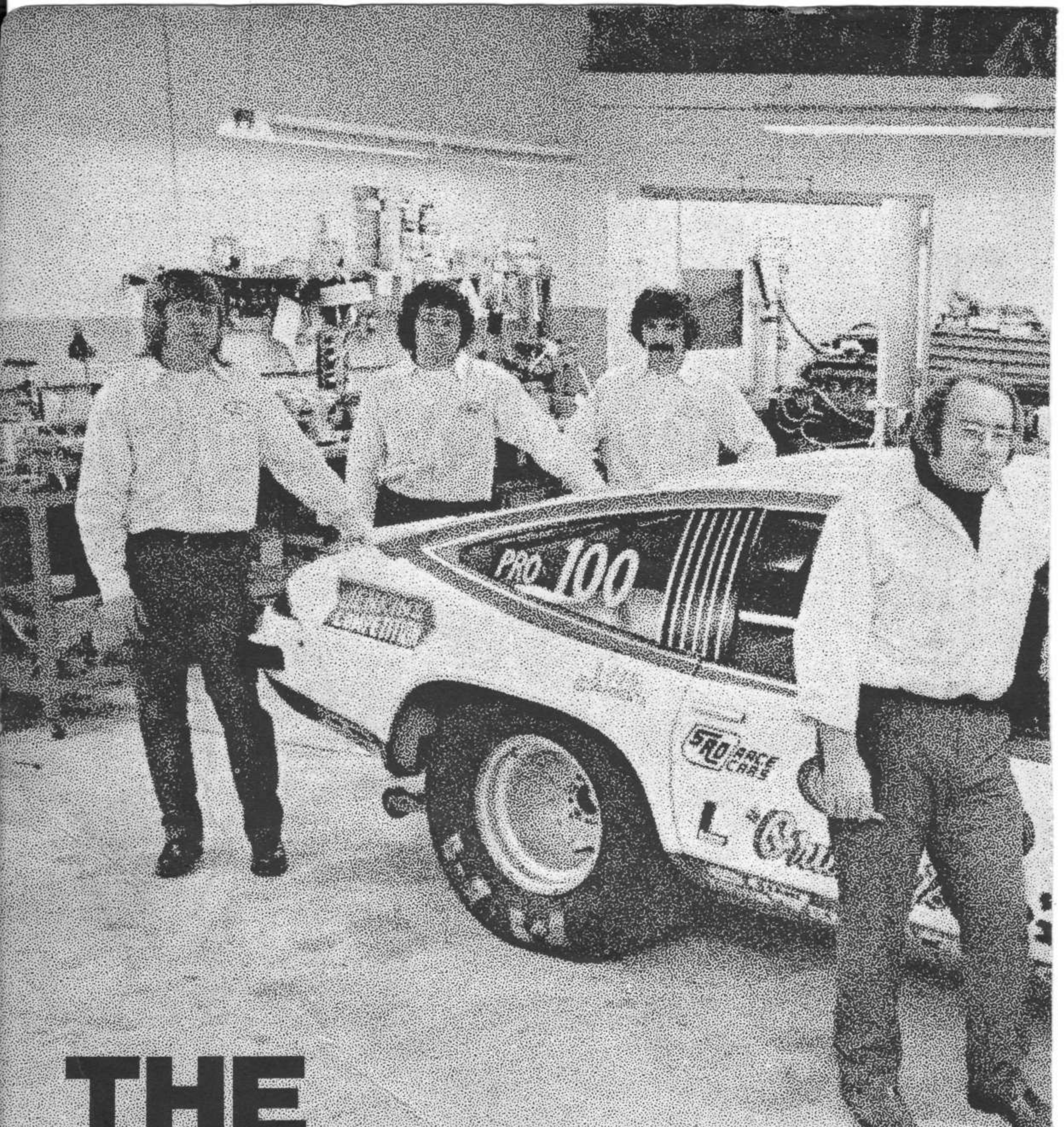
THE CHEVROLET RACING ENGINE

BY BILL JENKINS

WITH LARRY SCHREIB

$5.95



THE CHEVROLET RACING ENGINE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ABOUT THE AUTHOR

In 20 years of racing Bill Jenkins has stretched, squeezed, welded, machined, broken, bent, twisted, torn and molded the parts and pieces of a fairly complex mechanical device — known in familiar terms as the small-block Chevy — until today it produces power levels beyond all reasonable expectations. Since finishing his mechanical engineering studies at Cornell University and establishing his first "tune-up shop" in 1961, the Jenkins' technique and style has brought drag racing from a backyard sport to a quarter-million dollar professional level. During this period the Jenkins Competition shop has grown in size and capability, but the innocuous gray brick building in Malvern, Pennsylvania, gives very little clue to the prestige it represents in racing circles.

The Jenkins reputation first reached national prominence with an innocent-looking 1955 Chevy that dominated Stock Class drag racing in the early 1960's. It was a prelude to his unique approach of utilizing every component, every rule, and a full portion of imagination to build winning cars. In the following years he was to gain a small but dedicated following as various makes and models of national record-holding cars rolled out of his shop. In 1966 the first of the famous "Grumpy's Toys," an A/Stock 327-powered Chevy II appeared. As the stature of this awesome machine grew, the Bill Jenkins name became a household word among Chevy fans. It was soon followed by a string of incredible wheel-standing Super Stock Camaros. In 1970 the A/Super Stock number 6 car, one of the original legendary ZL-1 aluminum rat motor 1969 Camaros, began serving double-duty as a heads-up match race car with a dual-quad tunnel-ram induction. This car is considered to be the first "Pro Stock" racer ever to pass through the quarter mile timer. From this Jenkins-conceived heads-up match race formula the current NHRA and AHRA Pro Stock racing has grown. With the current "Toys" the Jenkins legend continues as a compelling force in the sport's most popular racing class.

Today the shop serves as home base for two fulltime racing machines. The Jenkins "Super Team" consists of seven specialists, and while Bill spends most of his time supervising the race car operations as well as personally preparing most of the research and development hardware, the Jenkins spectre is reaching even greater proportion. Each team member is hand-picked by Bill to fit a specific requirement. The right-hand man and shop foreman is Joe Tryson. Under his critical eye and personal touch the record-holding smallblocks are carefully detailed and assembled. With unwavering attention to endless fine points and organizational requirements, he juggles legal-competition engines with large displacement match race engines and highly-specialized test assemblies. During the racing season each of the race cars is supported by two men, a mechanic and a driver-mechanic. Putting in several thousand miles on the road each year, Ken Dondero and Ron Thacker keep the AHRA record books humming with one car, while Larry Lombardo and Rich Wright keep the Malvern-based machine ready for NHRA competition. During the heavy schedule each car races four to five times a week. To support the voracious engine and parts appetite generated by continuous competition, machinist George Areford and the team's newest member, Bob Rexrode, put in many hours of tedious behind-the-scenes work.

High on one side of the main shop a trophy shelf stretches several yards along the full length of a wall. It stands as quiet testimony to the full depth of the Jenkins reputation. From hundreds of trophies won over the years, the most prestigious crowd the shelf and spill over to the nearby bench. Interspersed among the brass and wooden statuettes are twenty coveted NHRA National Eliminator trophies. A few of these drag racing "Oscars" lie behind the bench covered with a layer of dust. While some men spend their lives struggling for one of these awards it is strange to see them treated in so casual a manner. But, it is an insight to the shop priorities. Not a single second is spent looking on past deeds — polishing, arranging and dusting trophies. All efforts, all thought, all imagination and creative energy, is turned toward winning. Every available resource is lavished on the all-important machines and toward creating their successors, the Grumpy's Toys that will carry the Jenkins Competition banner during the coming seasons.

Combined with seemingly unlimited personal energy, it is the ability to foresee, predict, evaluate new approaches and develop future techniques that is the essence of his success. From any viewpoint, the Bill Jenkins impact on drag racing will be measured for years to come.

THE EDITOR

The editor wishes to thank all the members of Jenkins Competition for their cooperation during the preparation of this publication.

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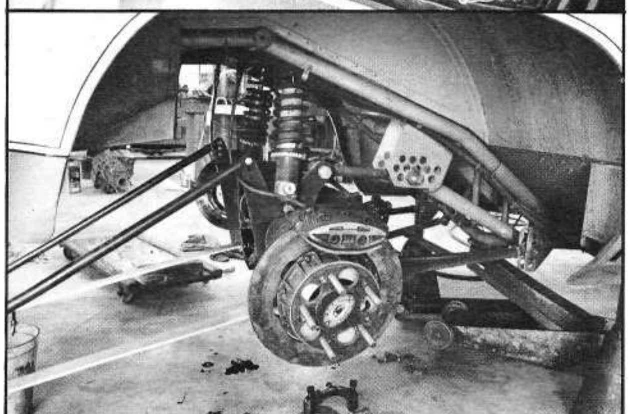
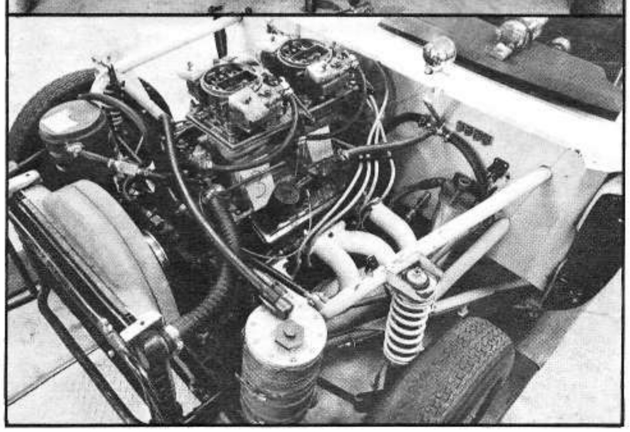
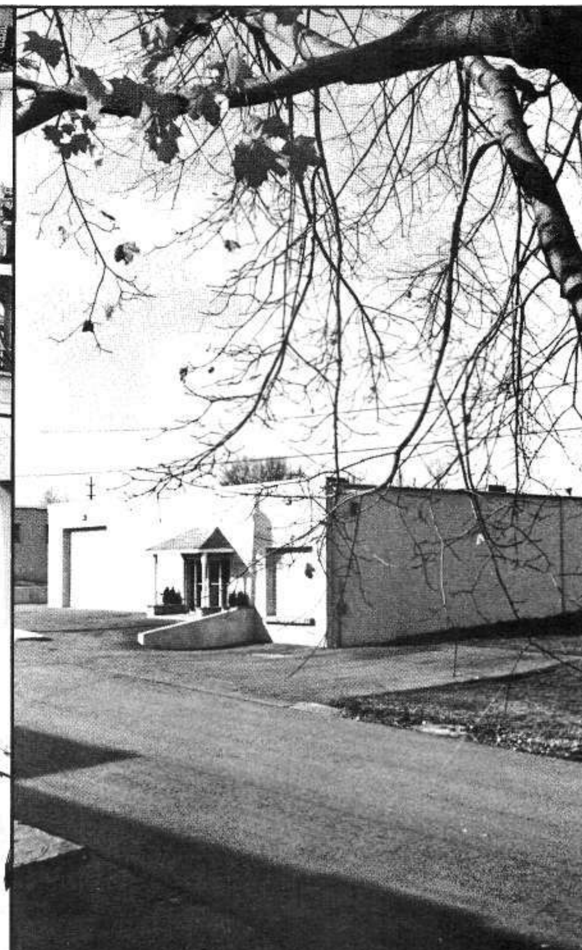
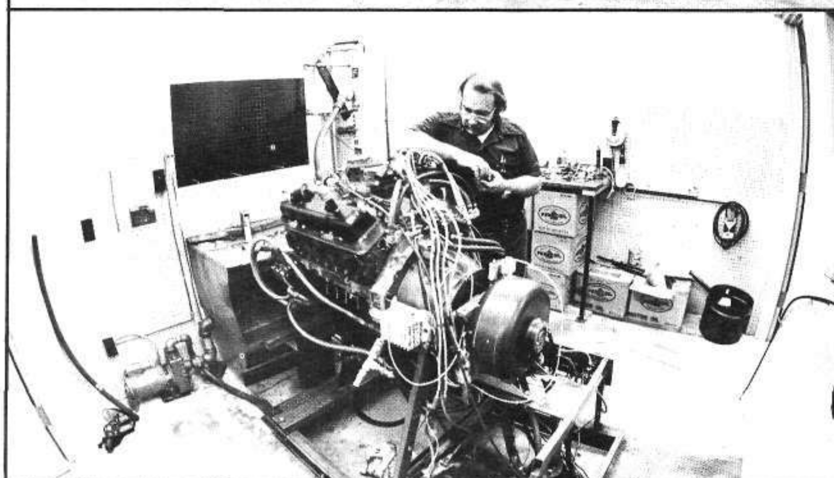





MOROSO
ACCEL
GUMOUT
PENNZOIL
HAYS
MANLEY






GUMOUT

INTRODUCTION

In the beginning we should make it clear what we intend to give to the readers of this book. The information herein is the result of more than twenty years racing experience with the smallblock Chevrolet V8 engine and it is intended as a basic study of racing engine design and assembly. The illustrations show typical techniques and hardware developed at Jenkins Competition for our own drag racing and Nascar stock car engines. Currently, we are talking about power levels of about 670 horsepower from our 330 cubic inch displacement Pro Stock drag motors and 580 horsepower from our Grand National four-barrel 354-inchers. In the original context we will be speaking entirely of very high horsepower "off-road" engines intended strictly for competition but the perceptive reader will find sound engineering principles which may be applied to any high performance internal combustion engine, regardless of make.

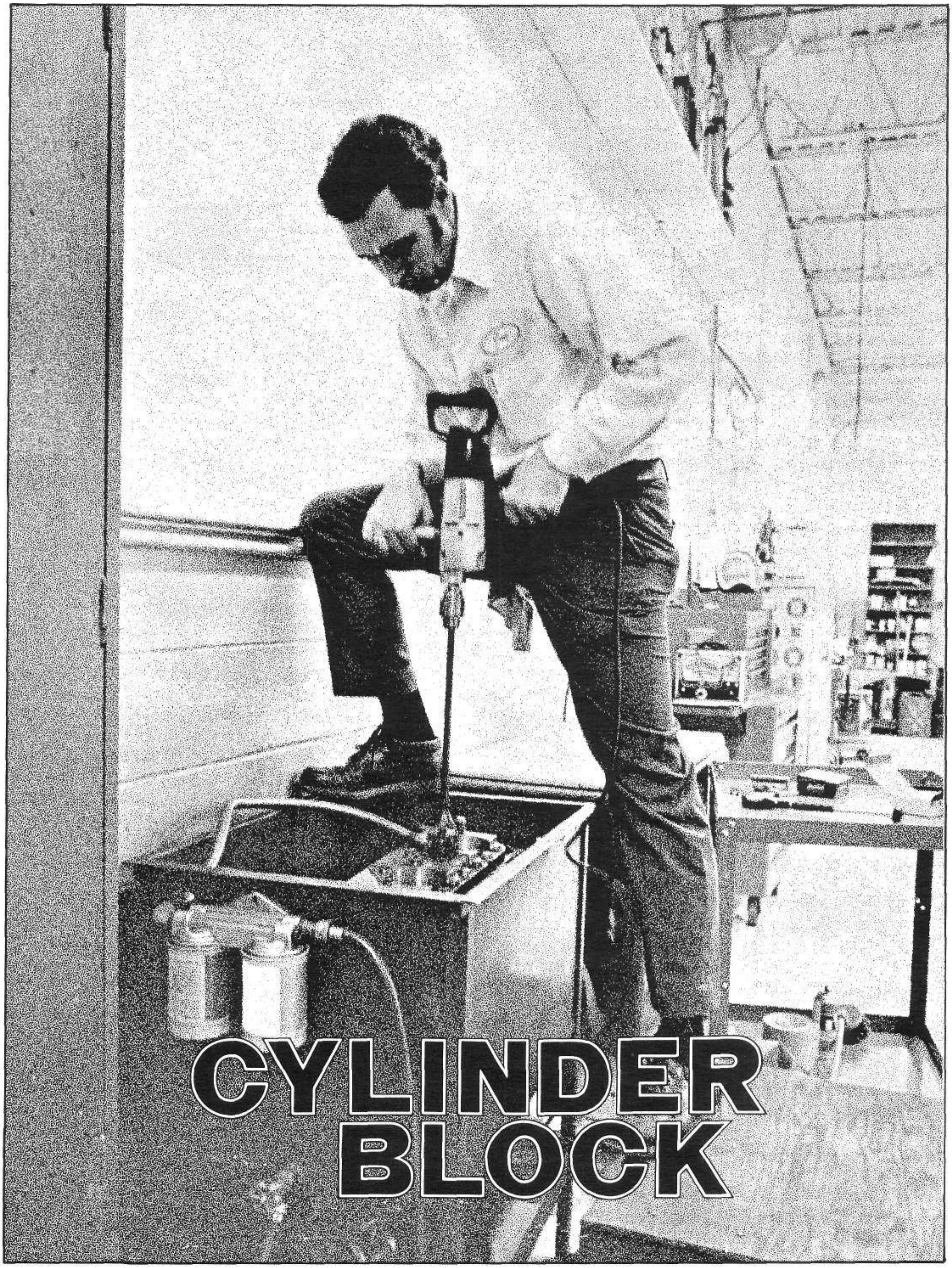
Historically, the development of the smallblock has been well documented in other publications. We won't cover old ground again, except for one comment. Considering the passenger car heritage of this engine we think the current power levels are incredible. Twenty years ago when we first started racing a showroom new '55 Chevy no one could have convinced us that the engine would eventually produce nearly 700 horsepower. We have discussed this phenomenal success with several knowledgeable people, some actually involved in the first design program, and the engine's long-lived reputation seems to be more a matter of good luck than engineering insight. Apparently such elusive factors as crank-to-block stiffness and the compatibility between component stiffness and weight factors has worked out better than anyone had expected or hoped. This is even more impressive when you realize that the basic block/heads run very nearly heads-up with engines like the late Cleveland Ford which was designed with the latest technology primarily as a racing engine and adapted for limited production passenger car use.

In terms of racing requirements the basic smallblock Chevrolet engine configuration provides excellent raw material. The overall outside dimensions are relatively small but the weight is not especially light in comparison to what could be achieved with the latest thinwall casting techniques. We have to remember that the basic block dimensions were established during research conducted by the Chevrolet engineers in the early 1950's. Construction requirements and techniques were different then. They introduced an engine with 265 cubic inches of displacement and developing approximately 195 horsepower. Today, the same basic engine block has been stretched 135 cubic inches to 400 c.i. and the power level has been multiplied by more than three times. At these levels the size and weight of the smallblock are truly impressive. The fact that substantial changes have not been made is very important, as far as we are concerned. The parts interchangeability and availability have kept overall costs relatively low and any racer who doesn't think replacement part costs are important either doesn't win too often or doesn't race very long. At Jenkins Competition we think it is one of the reasons we have been able to operate a successful racing enterprise, on the scale we do, for as long as we have.

This longevity is also important from another standpoint. Thousands of hours of research and development have been devoted to the smallblock Chevy by aftermarket parts manufacturers. In their zeal to produce new products to sell to Chevy enthusiasts they have helped little by little to extend the performance limits. At the same time, parts which can be sold to the huge number of Chevy racers and enthusiasts around the world result in cheaper production costs and lower retail prices. When you are watching the rev counter swing past the 9000 rpm mark every day this becomes more than just a side benefit—it can be the difference between success and failure. We know that only a small percentage of readers push an engine to this point but the same economic principles apply to the guy who swaps a carburetor or installs headers in his car while it's jacked up in the driveway as apply here at Jenkins Competition.

On the other hand, our current research has lead us to what we believe are the absolute physical limits of the basic block casting. The present 4-inch bore block is mass-produced for passenger cars and such things as manufacturing tolerance allowances, core shifts, material hardness and component strength limitations are problems the racer must overcome for his specialized use. We doubt that any future work will greatly increase the power output from this basic "case" as we are now using all the available inside space, strength and mass to the maximum. Of course, current work is concentrating on the lubrication system and the induction where more gains are to be had, whereas the most effective working mechanism of the block/crank assembly is fairly established at this point. Consequently, we are now looking at the possibilities of the 400 block which has a 4.125-inch cylinder bore. With success in this area we may find substantial gains in the future. Our research and development is an ongoing part of our race program and it is difficult to say exactly what will happen in the next six months or year. This publication, however, contains the essence of where we are—at this point in time. We hope all the readers find something of interest here.

BILL JENKINS


CYLINDER
BLOCK

BLOCK SELECTION

For all of our recent racing engines we have been using the late Chevy 4-bolt, 4-inch block. It can be obtained from local Chevy dealers as part 3970016. This block has the "good" features which are desirable for racing. The main webs which support the crankshaft are slightly thicker than on standard castings. The crank is supported from below by larger, wider, 4-bolt nodular main bearing caps. The rear crankshaft seal is made of neoprene rubber for improved oil control. At this point we prefer this over any other cylinder block casting. However, we usually want to use a case which has been run or "seasoned" in a previous application, like about 100,000 miles in a truck or something similar. A "green" block will move around quite a bit under racing stresses and in almost every instance the machined surfaces will not be true after a few hard racing passes, even if the case has been carefully prepped. This will hurt horsepower.

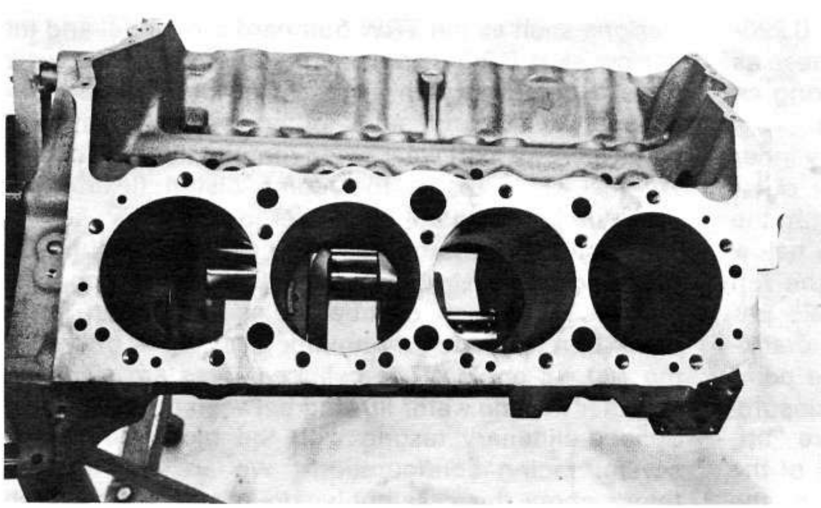
Before proceeding with further detailing we thoroughly clean the piece and examine for minor cracking or bore splitting. We seldom encounter any problems when the block has not been previously raced but a good "look over" is mandatory if you don't want to wind up spending money on a case that's already junk. This may sound a little trivial but it is the first step and is occasionally overlooked by people who really should know better. The cylinder bore condition is not important, as long as the lower portion does not have any cracking signs, because nearly all of our blocks are bored 0.020-inch oversize. It is particularly important that number 8 bore be closely examined.

If you are building an engine which requires a finish bore size smaller than 4 inches you must use an appropriate early casting. Most of these early cases had thicker cylinder bore walls and are therefore desirable. There are some oddball cases like the rare '57 number with a 0.250-inch thicker deck but these are no longer available and we feel the extra mass only adds weight as our current blocks hold the head gaskets in place quite adequately. The famous (or infamous) '62-'67 Chevy II Nova case with different filter pad location and unique clutch ball receptacle is also sought after by some but it does not have the 4-bolt caps. We feel searching for one of these cases is more trouble than it's worth.

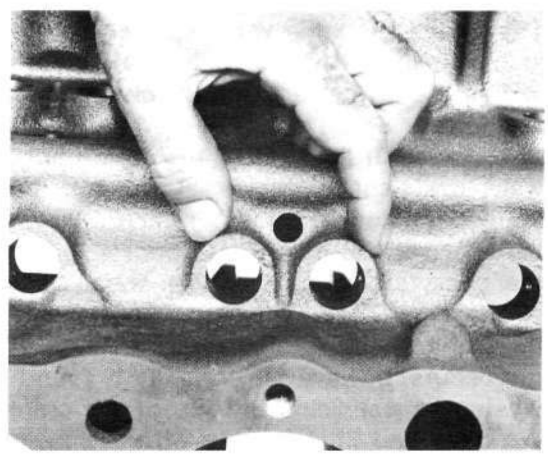
SONIC CHECKING

When seeking the best possible case for a race engine we do consider two important elements: *uniform wall thickness and material hardness.* Adequate cylinder wall strength to support the piston in a racing engine is absolutely essential. Current production methods and minor core shifting within the water jacket have made this a questionable matter. Since about 1970 we have been using an electronic instrument to sonic check the material depth of all our cases. The particular instrument we use has an ultrasonic probe which is held tightly against the metal surface and it gives a direct readout of the thickness on a dial gauge. When we receive a case for possible use in a race buildup we check the walls on the minor and major thrust side at a position about 2.5 to 3 inches below the deck. We want a case which has the thickest possible walls on the thrust side (passenger side) of each cylinder bank. We also like to see uniformity from front to rear and from one bank to the other. Thickness of the front and rear walls of the cylinder is less important, except to determine an unacceptable core shift.

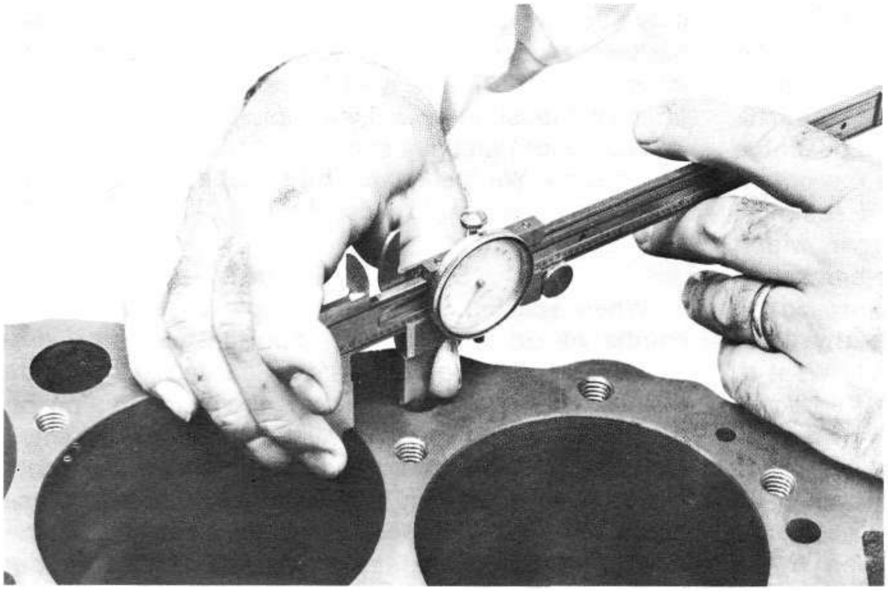
After about five years of checking cases we have found that the walls are generally thicker than the Chevrolet specification. According to our information they give an average spec of 0.223-inch on the thrust sides with a minimum of 0.125-inch allowed on the front and rear sides. We find an average reading taken on the minor/major sides of about 0.240- to 0.245-inch. We have located some which average as high as 0.270-inch and consider these pieces very desirable



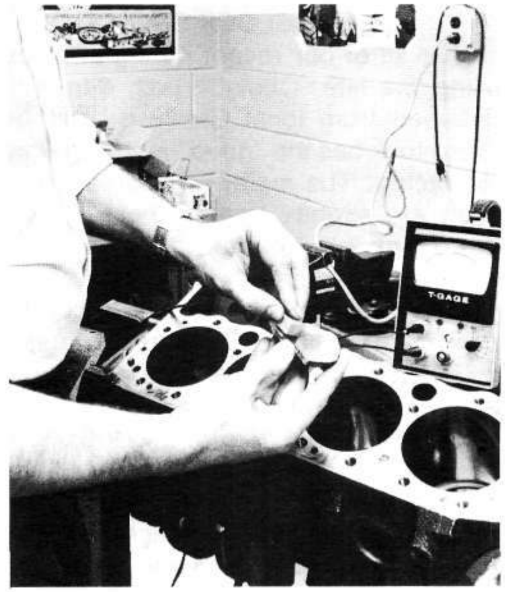
To build any racing or high performance smallblock engine with a finish cylinder bore diameter between 4.00 and 4.030 inches the best factory block casting to obtain is part 3970016. This case has the thick main webs and 4-bolt main caps for added crankshaft support.



The most desirable casting for a race engine will have a minimum production sand core shift relative to the machine finish. Visual inspection of the lifter bore drillings is a simple but inconclusive method for spotting shift. All drilled bores should be centered.



When absolute strength is not required some slight core shift is acceptable. Measuring between the as-cast core projection holes in the deck and the finished cylinder bores may locate severe shift. Hopefully, all of the measurements will be fairly equal. A wide variation indicates the cylinder hole axes are shifted relative to the water jacket coring and the thickness of the walls will vary at different points around the bore.



For absolute reliability we use an ultrasonic thickness gauge to inspect every block casting. A core shift toward the front or rear of the block is less important, but a shift sideways, along the major-minor thrust axis, can lead to wall distortion.

for racing. On the other hand, we have run some cases with a finish thrust wall size as thin as 0.205-inch on one hole and not had trouble with the engine but we prefer a minimum 0.215-inch (finished) major thrust wall in a race engine. One of the first places stress will show up in the 4-inch case is around the number 8 cylinder wall, therefore, we don't race any block which reads marginal in wall thickness at this location. For street use the finished walls may be as thin as 0.180-inch and will give good service.

When checking thickness of the walls in the area between the cylinders we usually get a smaller-than-average reading. In the 4-inch case the water jacket is designed with about 0.100-inch between the cylinder walls. There is a total of 0.440-inch between the nearest points of the waterside surfaces. This only leaves 0.340-inch total or about 0.170-inch per wall. The actual readings vary between 0.115- and 0.220-inch. Rarely are they as high as the latter but those as low as the former are still usable. That is, as long as the number 8 wall does not read thin.

Because of the draft in the coring box the cylinder walls are usually thicker at the top and the outer surface (water jacket side) tapers inward, with the thinnest section occurring about 4 to 5 inches below the deck, at the point where the wall joins the top of the crank box. The taper normally is 0.0025 per inch, and does not affect performance. If wall cracking occurs, however, it normally is found at the point where the wall joins the top of the crank enclosure. We have had little trouble with the jacket core "tipping" inside the case. In other words, the axis of the inner wall is very nearly parallel to the axis of the core wall. As a matter of interest, we found this to be a big problem on some of the Mark IV cases. The core would be tipped significantly and the walls would be, for instance, very thick on one side at the top and thick diagonally across the bore at the bottom while the walls diametrically opposite at the top and bottom would be thin. Needless to say, this is a bad condition and some of the big blocks we checked were very, very bad.

Some information we have received leads us to believe that Chevrolet production is presently making a concerted effort to reduce core shifting problems. Their work is very likely aimed toward gaining better control in order to reduce some section thicknesses. The resultant metal saving could be substantial over the production of thousands of smallblock castings.

In connection with this we will later discuss the importance of piston skirt design as related to cylinder wall endurance. Generally, the smallblock does not like pistons with extremely stiff, narrow skirts. Designs such as the TRW outboard pin model and the narrow skirt BRC or Manley pistons "push" the walls quite a bit. Even with a good wall configuration we have always found signs of severe wall distortion and excessive skirt wear when running this type piston. We feel, as a result, that some piston flexibility is essential for adequate block life!

Our latest research has been centered around the 400 block. It is virtually identical to the 4-inch case but has 4.125-inch cylinder holes and has a larger crank bore to match the unique cast crank offered in the 400 c.i. engine. The cylinder bores are siamesed together with no water flowing between them. We have done preliminary testing with the block built up to several racing configurations. We are still learning things about this case but we do know it is absolutely essential to sonic check them. Before race-prepping this block be sure to inspect the area around the steam holes between the cylinders very carefully.

Cracking is very prevalent in this area. They may crack across to the cylinder head bolt holes (very common) or from the head bolt holes into the cylinder walls (occasionally). Also, when sonic checking a 400 block, make certain to avoid reading directly under the head bolt holes. There are vertical clearance notches in the outer wall of the cylinder casting that will cause false readings.

If you are serious about engine building we recommend you sonic check any block considered for racing use. It may not be worth the cost to buy an instrument such as we use but similar equipment is available in most large cities and the cost to have a block checked is usually very small. This method is used widely in the aircraft industry and a good place to begin searching for such a service might be the local airport or aircraft engine rebuilding shop.

If you just can't find the facilities or simply won't be building enough power to be concerned with this detail, there are some ways to "eyeball" for core shifting. It is possible to measure across the deck from the sand core projections (the as-cast holes in the deck) to the adjacent cylinder walls. Hopefully, these distances will be relatively uniform from hole to hole. This indicates the core plug is well-positioned with respect to the drilled cylinder bore holes. Closely inspecting the lifter bores may also give you a clue. The holes should be drilled on-center with the casting bosses or "knobs" all along the valley. If you find the holes off-center as you look straight down toward the top of the bosses—be wary of core shifting problems.

HARDNESS CHECKING

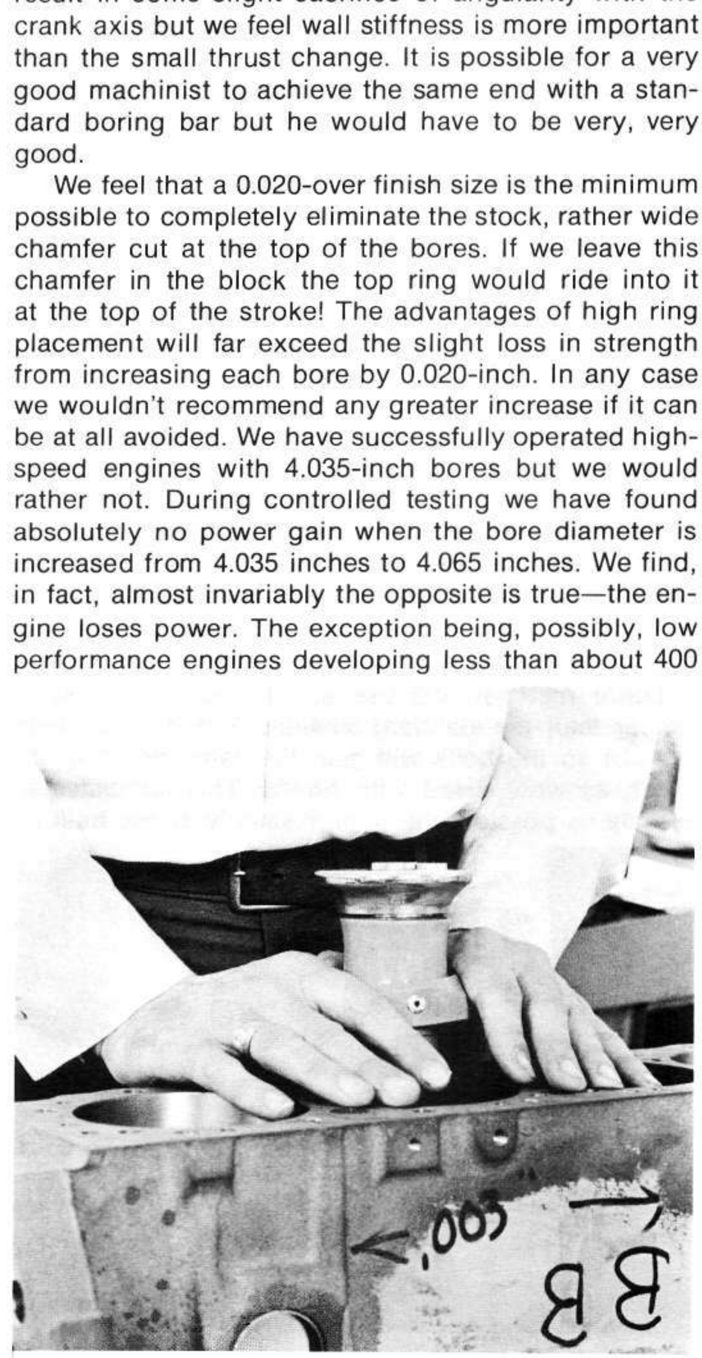
Only recently we have become aware of the importance of block material hardness. We are convinced that we can gain reliability and power by Brinell testing prospective race cases (Brinell hardness testing is a method of testing metal or alloy by hydraulically pressing a hard ball under a standard load into the specimen).

As far as we know the original block material specification has not been changed since 1955. It is possible, though, that the iron alloy mix has been adjusted slightly to make it as soft as practicable in order to gain maximum life from production tooling. Our initial testing has shown some appreciable difference in the cases. We are going to have to gather more data before we can make any specific recommendations. We are, however, looking for the hardest material possible to gain maximum stiffness throughout the case. We don't currently send our bare cases through any special treatments but we are convinced that anything we can do to locate a harder production sample or any method we can develop to make it harder will result in more power.

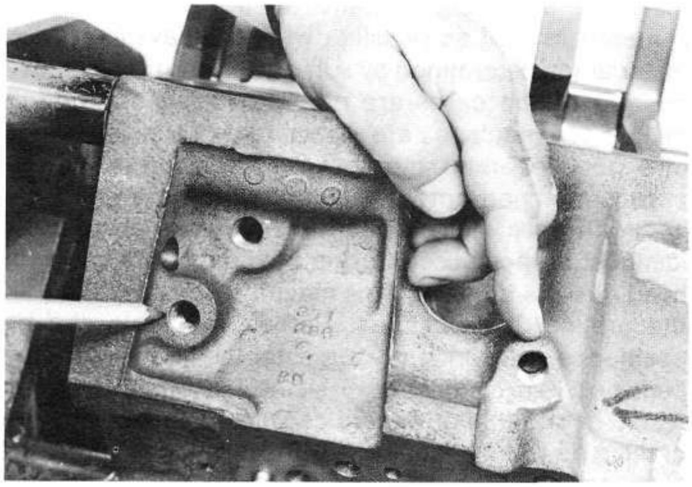
CYLINDER BORING

Most of our race cases are finished 0.020-inch oversize. By boring the cylinders we can make sure the holes are straight relative to the crank axis and centered as well as possible within the available wall material (as determined by sonic checking). To achieve this, all of the cases are bored on a special boring mill. These machines are rather rare but in the hands of a good operator they get the job done right. The case is clamped onto a mandrel with the crank caps and with deck plates bolted to the decks using bolts identical to those used during final assembly and with head gaskets in place. This machine does not read off the original bore centers. Rather, the table can be positioned independently to center the hole to gain the best possible cylinder wall sizing. At times this may result in some slight sacrifice of angularity with the crank axis but we feel wall stiffness is more important than the small thrust change. It is possible for a very good machinist to achieve the same end with a standard boring bar but he would have to be very, very good.

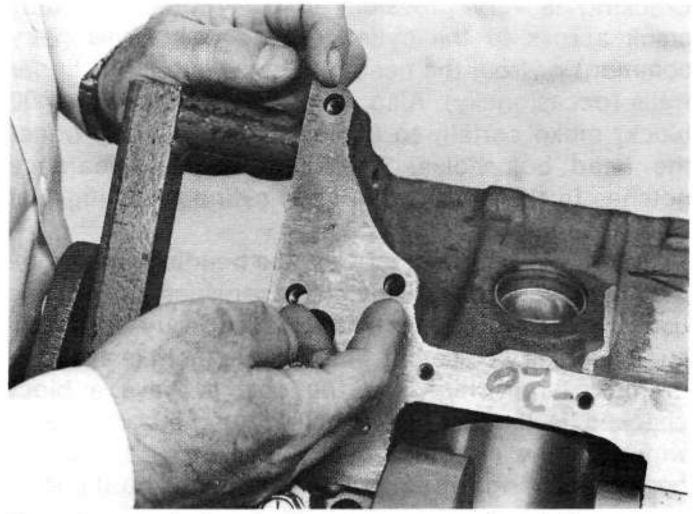
We feel that a 0.020-over finish size is the minimum possible to completely eliminate the stock, rather wide chamfer cut at the top of the bores. If we leave this chamfer in the block the top ring would ride into it at the top of the stroke! The advantages of high ring placement will far exceed the slight loss in strength from increasing each bore by 0.020-inch. In any case we wouldn't recommend any greater increase if it can be at all avoided. We have successfully operated high-speed engines with 4.035-inch bores but we would rather not. During controlled testing we have found absolutely no power gain when the bore diameter is increased from 4.035 inches to 4.065 inches. We find, in fact, almost invariably the opposite is true—the engine loses power. The exception being, possibly, low performance engines developing less than about 400



Our latest efforts have been directed toward hardness testing potential block selections. There is some variation in the production casting material and we use a portable Brinell tester to find "hard" blocks.



When selecting a new bare block some time must be spent checking all of the threaded holes. These clutch mounts may be important when the engine is in the chassis but are easily overlooked during inspection. Sometimes they are not completely drilled and threaded at the factory.



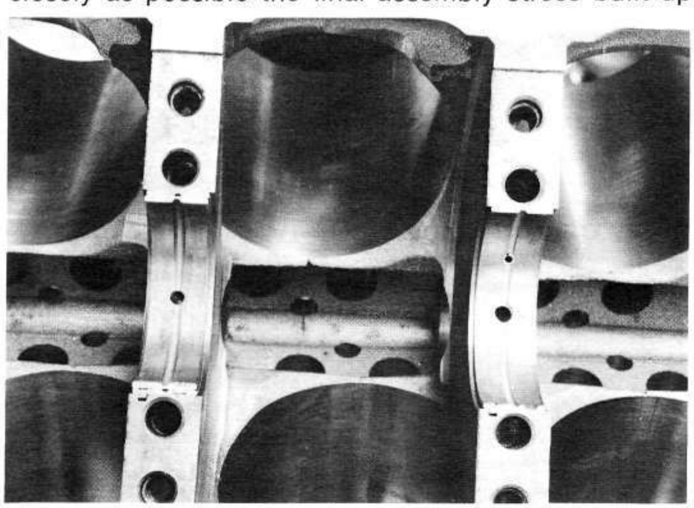
There is a variation in the starter mounting holes among the various smallblock applications. When a block casting is purchased these holes should be checked to insure the desired starter can be mounted to the block.

measured horsepower. A "sixty-over" bore would probably provide adequate stiffness for a street engine but certainly never in a race piece. Apparently the reduction in wall stiffness affects ring/piston stability enough to offset any power gain from increased displacement.

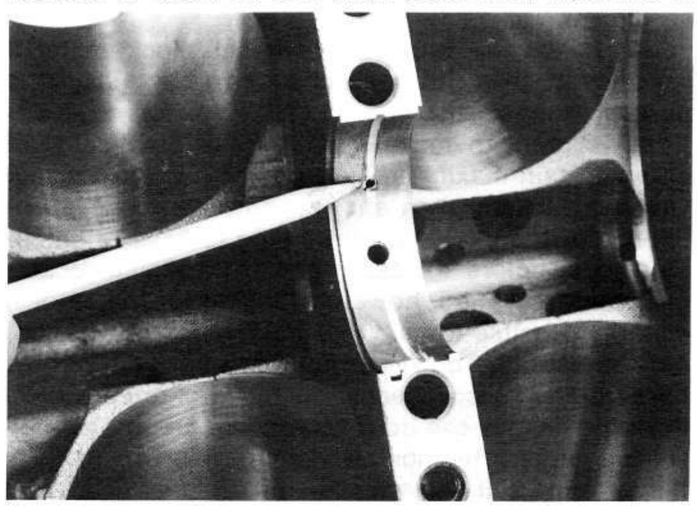
During all phases of bore prep a deck plate is fitted to the block. As far as we can tell the overall thickness of the plate is not important but the type of fastener used with the plate is very critical. The plates we use are designed for stock head bolts and when they are installed for bore prep we use stock bolts of the exact type to be used during final assembly. We will discuss fasteners in greater detail in a following chapter but we do feel the stock bolts are manufactured from a superior material. We use a bolt that is somewhat longer than the standard smallblock bolt. The plates are cut so the bolts will gain the same perch in the block as when used with heads. This simulates as closely as possible the final assembly stress built up in the upper portion of the bore. This is the area where the ring package rides and it is most important for ring seal that the bore be as round as possible during prep and running. When the head bolt holes are cleaned during preassembly they must not be cut oversize with the tap. After the deck is finish cut, the top of the holes must be chamfered *slightly* to prevent the top thread from being pulled above the deck when the bolts are torqued. A gasket of the same type used during assembly should also be installed between the block deck and the deck plate during all machine preparation.

ALIGN BORING

We are not big believers in reboring the crank housing. We do check to see the saddles are "reasonably" in alignment with each other but we don't get overly excited about it. Normally, we use a straight edge across the saddles to check for any possible trouble. If there is less than 0.003-inch variance at



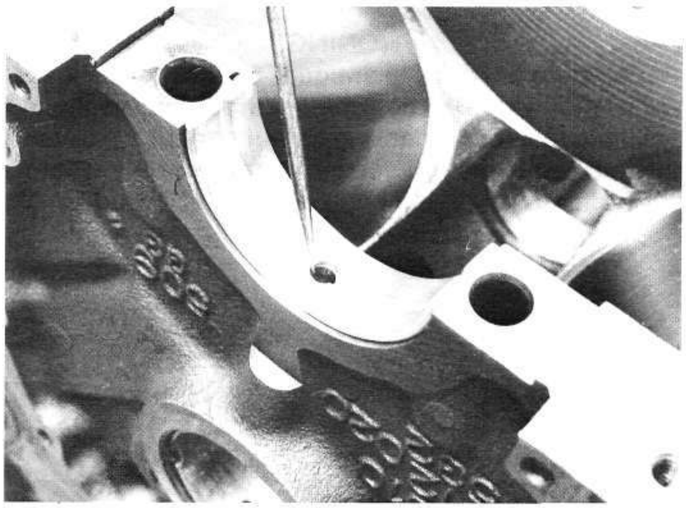
Because we normally sleeve the main bearing bore of the late block to the early 2.3-inch journal size, align honing or boring of the bare block is not necessary. We don't recommend aligning the main bore unless it is essential. Inaccurate machining causes more trouble than it solves.



To make the main bearing sleeves we use a set of ordinary 2.4-inch bearing inserts. These inserts are pinned in place and align honed to give a face diameter on the small side of the late journal tolerance. Note the roll pin must not interfere with the normal bearing installation.

any one position we are happy. Another useful method is to lay a known straight crank into the block with properly-clearanced bearings. If the crank turns smoothly without binding at some point in the rotation the block crank bore is straight enough. It seems a little simplistic in light of the current hullabaloo about align boring but with our building techniques we just haven't found it necessary.

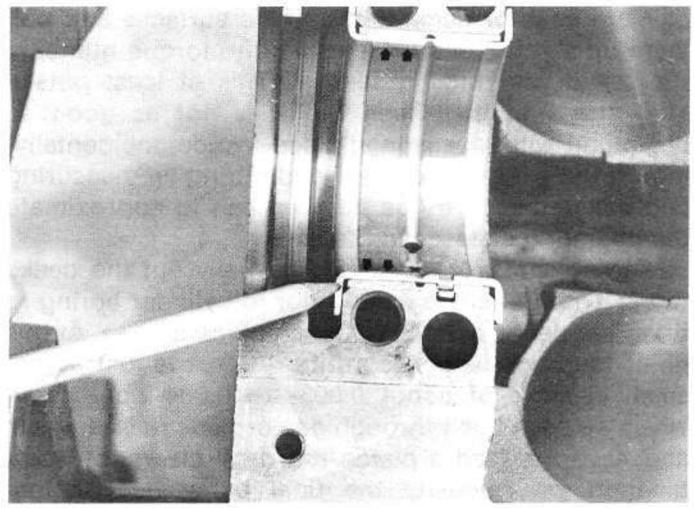
We do caution any engine builder to cast a careful eye at any block that has previously been align bored. It is possible for the technique to do more harm than good. Specifically, we have found trouble with the thrust cap on many bored cases. The biggest problem is getting the thrust face of the cap square and in the same plane as the thrust face on the block side. When the parting face of the rear cap is ground 0.005-inch prior to boring it is possible to wind up with the face at a different angle which results in the cap sitting on the block crooked. The thrust face will then not be flat. If we suddenly have a rash of thrust bearing failures this is the first thing we check. It is easy enough to use a small square to check this. If there is any doubt at all we blue the face of the bearing, install the crank, give it a few turns against the face and check for even wear patterns on both halves of the bearing.



After the insert-sleeve has been installed in the block the oil supply hole is drilled to 0.250-inch, to coincide with the main bearing oil supply passage in the main web. Note also that the face of the insert-sleeve has yet to be slotted in order to accomodate the bearing tang of the small-diameter insert

Although we avoid align boring, there are two possible exceptions. Naturally, if 4-bolt steel caps have been installed to a 2-bolt block the housing will have to be bored. In this case we prefer to finish the hole to final bearing size with a hone. We also inspect a brand new unused case very carefully. When you are starting with a well-used block you know a lot of things are in shape or else the engine wouldn't have been working. You don't have the same assurance with a new piece. We have seen some with all the caps installed backwards and one time we had a weird block that looked as if it had been bored from each end and the two holes didn't line up!

In our drag racing engines we use the early small-bearing 327 cranks and a spacer sleeve is required between the late block bearing saddle and the small journal bearing. This crank is an especially stiff forging and the smaller bearing diameter reduces bearing speed. Adapting this early crank to the late case is much simpler than it sounds. It is, in fact, extremely easy. To sleeve down the block we install the late (large diameter) bearing shells in the saddles and pin them in place with small roll pins as can be seen in the photos. These pinned bearings/sleeves are then align honed to the Chevy recommended saddle diameter for the early (small diameter) bearings. We usually shoot for the smaller side of specification range but never less. By setting to the "tight side" we gain a little more bearing crush to compensate for the crush in the sleeve and the additional gap between the backside of the sleeve and the case saddle. Before installing the small diameter inserts the oil supply hole in the sleeve is drilled to 0.250-inch to provide oil pressure into the main bearing. The thrust flange of the rear main sleeve must also be cut away before it is pinned to the block. New bearing tang slots are cut on the face of the refinished insert. When the Chevy bearings are installed they are placed in the normal manner, right on top of the pinned sleeves. Since we began using this method we have never had a bearing failure related to this procedure.

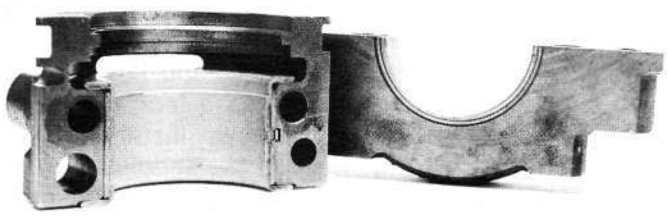


When the rear insert-sleeve is installed, the thrust flange must be cut away in order to allow the small-diameter bearing to fit over the sleeve. The parting edges of the bearing are also chamfered slightly to allow pressure oiling from the groove to the thrust flange face.

DECK FINISH

Most guys think of deck finishing in terms of deck height and compression but we have found the texture of the deck finish is important for head gasket life. We have a specific technique involving a number of steps which we follow. Since we evolved this manner of bringing the decks in we have had very few cylinder head gasket problems.

As soon as we receive the case we measure from the axis of the crank housing to the decks as precisely as



The fit of the bearing caps is very important. They must pilot correctly into the main web recesses and sit squarely on the webs.

we can. For this purpose we normally employ a large micrometer. If the case is seasoned, the decks will almost always show signs of being pulled around quite a bit. If the case is new we have to first mount a pair of heads or deck plates to the surfaces and pull them up tight with the fasteners and torque numbers we use during final assembly. This at least puts a little stress in the block but it is not as good as beginning with a seasoned piece. We do, incidentally, mount the caps to the lower end during all measuring and machining to try as best we can to approximate the operating stresses in the case.

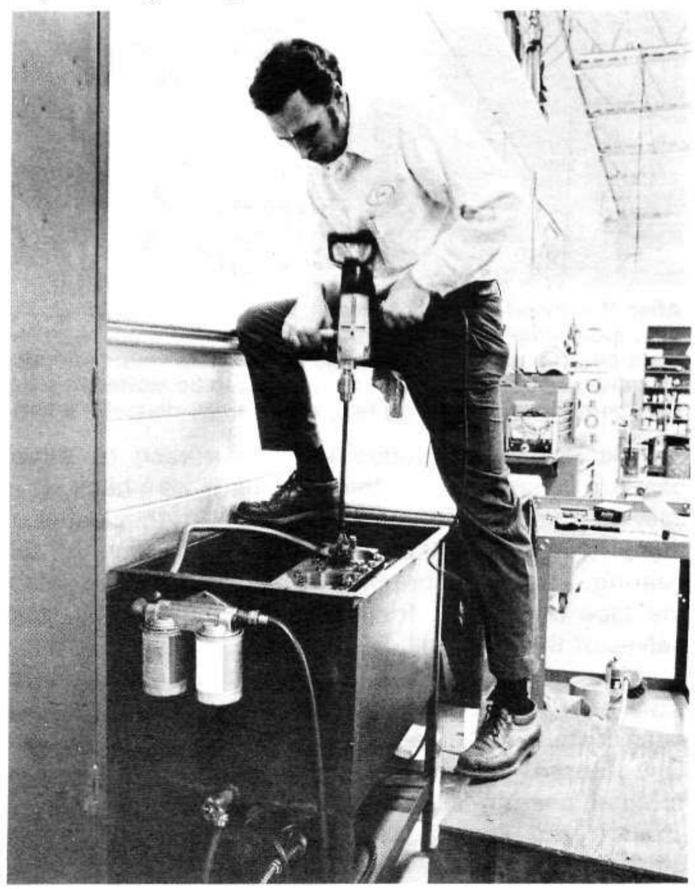
During the process we will usually cut the decks twice. The first cut is made prior to cylinder boring to bring the decks parallel to the crank centerline. At this point the deck height is arbitrary. This is just a very small cleanup of about 0.003- to 0.004-inch. Later, when we have been through one or two preassemblies and have finalized a piston/rod/deck clearance combination we calculate the final block deck height needed for the specific configuration and the deck is cut to this dimension. However, we often have several engines around the shop in various stages of assembly and during emergencies we have thrown together some pretty unusual combinations of rods from one engine, a crank from another and pistons from here or there. In these instances the deck height could be a restriction so we like to keep all the blocks fairly close. We don't normally finish down more than 0.020-inch though we have cut as much as 0.030- to 0.040-inch for odd combinations.

Deck height is important but we don't really find it ultracritical. We have several different gasket combinations which allow us to set the heads higher or lower as necessary. This greatly increases our flexibility to swap parts around and, since we never have gasket sealing trouble, it works out just fine. We install stock Chevrolet beaded steel gaskets and to make them work it is essential to have as smooth a deck/head surface as you can attain. We used to have the surfaces ground to our specifications but recently we have acquired a Van Norman mill which has an extremely slow feed. This has worked very well for us. It takes about one hour for the machine to make one pass down the 22 inches, leaving a very smooth finish. There is no such thing as a "too smooth" deck as far as we are concerned, at least with steel head gaskets. There should be no voids at all across the deck and when the gasket is cinched down between the surfaces the beads on the gasket will seal extremely well. Since we began smooth finishing decks we have never pushed a gasket. That is, we have never pushed one on a 4-inch block; the 4.125-inch block is a different story. We will discuss the 400 block more later.

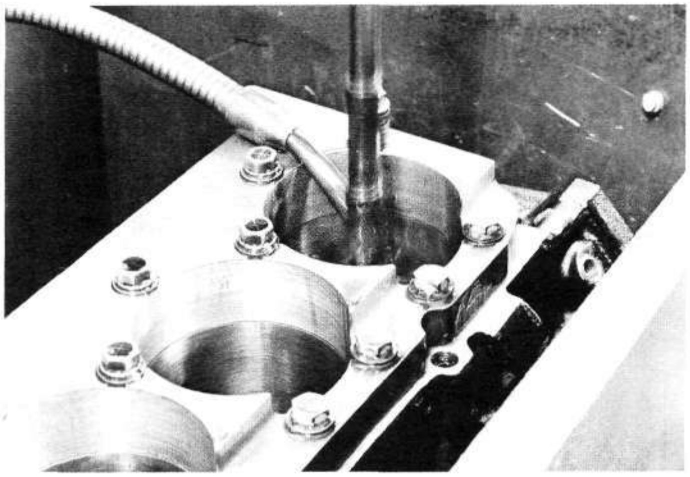
In some cases we will also lean the decks slightly during the final cut. The driver-side deck is especially worth leaning a very little bit to give the piston a little more clearance along the bottom side than at the top. This makes some allowance for the piston rock on the dead stroke (overlap) and is useful in a very high compression engine with tight piston-to-head clearances. We are talking about approximately 0.012-inch from one side of the cylinder to the other.

HONING

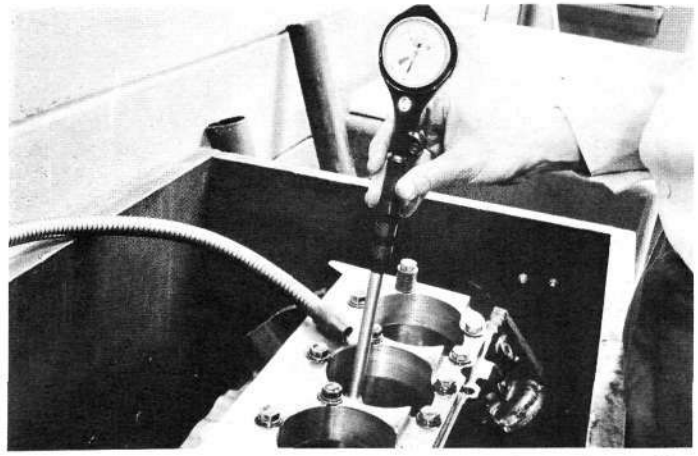
Our bore finishing techniques are not very unusual. We employ virtually the same methods and equipment available to every other engine builder but we follow a specific sequence religiously. Here is exactly what we do, nothing fancy. When the block is bored on the



Initial cylinder sizing can be completed in a machine such as a Sunnen CK-10 power hone but in our shop all finish work is slowly and carefully completed by an expert with a hand hone. Ours is not the most elaborate setup but it works. The walls are brought to within 0.0005- to 0.0007-inch of finish size with 400 grit J-85 stones. For moly rings the finish size is completed with 500 grit J-95 stones.



The hone adjustment must not put too much pressure on the walls. This is a slow process, requiring experience. Coolant circulation is continuously recirculated through an electric pump and filter arrangement.



The bores are measured several times throughout the process. We use a Sunnen dial bore gauge and read across the thrust faces at the top, the middle and bottom of the bore. A deck plate with gasket and selected fasteners is always mounted to the block.

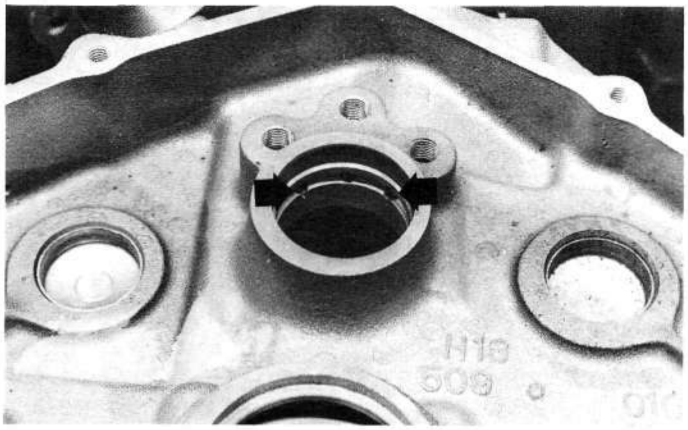
mill it is usually cut to within approximately 0.005- to 0.007-inch of the desired finish diameter. When it comes off the mill it is usually about 4.014 across the hole. The case with deck plates, gaskets and main caps is then mounted to a Sunnen CK-10 and power honed with 280 grit stones in the bar to within 0.0015- to 0.002-inch of the final. At this point we set it into our honing tank in preparation for final hand finishing. This tank is just an open box, like a parts washer, with a system for recirculating the honing fluid. A small pump pulls the fluid out of the bottom, pushes it through two ordinary automotive-type filters and out a dump tube. The tube can be adjusted so the fluid dumps into whichever one of the cylinders we are honing. There's nothing particularly spectacular about this but it makes the job easier. Much of the finish quality is a result of the talent and the experience of the individual operating the hone. By eliminating all of the distractions he can concentrate on what the hone is doing.

The first hand honing is made with 400 grit J-85 stones. We use Sunnen honing fluid, number MAN-845-1, in the tank. We normally adjust the stone pressure three or four times as the metal is cut away. The dimension is brought to within 0.0005- to 0.0007-inch in this step. We use a Sunnen bore gauge to measure the hole and the check is normally made between the thrust faces.

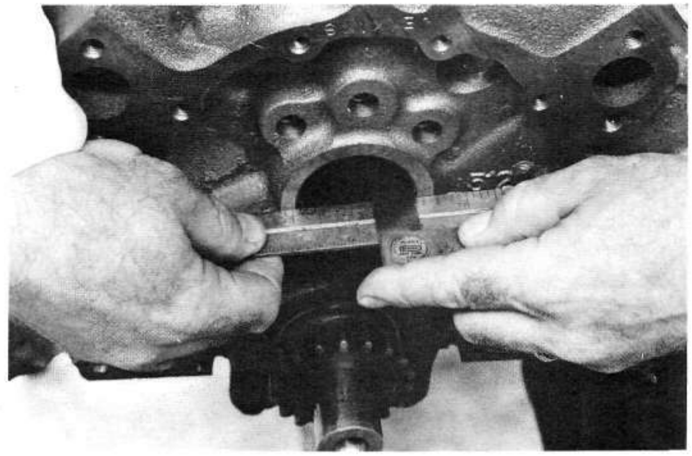
For the final finishing we use 500 grit J-95 stones to cut the last 0.0007-inch or less. You cannot cut more than this from the walls with a 500 stone. This stone simply won't be able to move that much metal. If you try, the stones will be cutting beyond the roots of the previous hone pattern and they will begin to fill and speckle. The binder won't be able to break down and the bore walls will quickly become burnished. This would be evidenced in a microscopic examination of the walls as a "flattening" of the tiny peaks and valleys formed by the honing process. The peaks are actually bent over sideways and they form beautiful little pockets in which dirt and miscellaneous debris is trapped. This grit will later be released when the rings operate because the small pockets can't be cleaned out during the normal cleaning process. A burnished bore will, more importantly, prevent proper ring break-in, especially with the moly-type rings. When the peaks are broken down, the surface just isn't able to break in the ring face. To prevent burnishing, don't try to cut more than 0.0007-inch with the fine hone and don't leave it in the bore too long. If the hone suddenly builds speed while you are running it in the bore, chances are good you have overdone it.

If we are running chrome-faced rings we generally shoot for a somewhat rougher bore. The same goes for wider, stock-type rings. We don't ever use a finish any finer than this though. We have tested engines with walls finished by 600 grit CO-3 "cork bond" hone inserts and have found out two things. One, the stones wear out very quickly, and two, the engine doesn't make any more power. If anything, it is less desirable because oil control becomes very difficult with this fine finish. We have found power gains and better sealing characteristics every time the wall finish is made finer, down to a finish of about RMS-7 (this is with a 0.043-inch wide, moly-sprayed top ring) as provided by 500 grit stones. If the surface is finished to a finer RMS-4 finish with the 600 stones we feel the compression seal may be so good that oil control becomes a severe problem (see rings) and upper cylinder oiling may result in lost power.

Initial bore honing is important but subsequent honing after run-in is critical. After any bore is refinished there will be a certain amount of metal shifting when the engine is again put into service. This will occur no matter how well it has been prepped. After the newly-assembled engine has been run some — a few hard drag passes, a few hard laps or a few dyno runs — it should be stripped down and rehoned with a fine stone. When the first refinish is made we always find an immediate horsepower gain, in some instances as much as 15 horsepower. Every time after this when we refinish and rering one of the racing engines we usually find a



At the rear of the cam bore we restrict the pressure oil feed up into the right and left lifter galleries. A 0.050-inch restrictor jet in each of these openings will keep oil pressure feeding into the center gallery and subsequently to the main bearings rather than bleeding excess oil to the valvetrain where it is not needed (provided roller rockers are used).



During machining the cam bore face must be checked. This surface must be perpendicular to the cam bore axis. This is especially important when racing springs are used and an iron cam sprocket is mounted against the face. It is also recommended in this instance that the face be machined to accept a thrust washer.

few more horses than before but the first time is the most important and shows the greatest gains. We find this wall shift occurs even in well-seasoned cases with over 100,000 miles.

CAM BORE

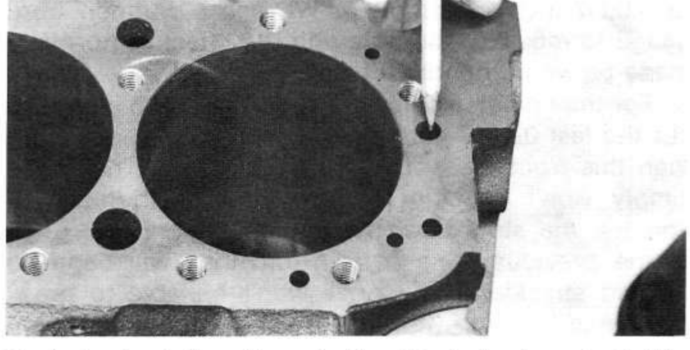
To be on the safe side (this is mandatory in a race engine) the cam bore should be inspected. Trouble is rare but we have found some inconsistencies. At times the cam saddles may be undersize and the cam won't fit into the hole with the bearings installed. It is then necessary to fit semi-finished cam bearings to the block and rebore through the bearings. It isn't necessary to spend a huge amount of time with the cam saddles and we actually have very little trouble with them. However, they must be straight. The condition of the bearings is less important. Slight nicks or the like aren't crucial. We just polish them up lightly with fine paper and run 'em.

The front face of the bore is also important. This surface, against which the cam drive sprocket rides, must be perpendicular to the bore axis. Production variations are not likely, but possible. We check the relationship using a small machinists' square and a straight edge. If it checks out with these simple tools it will probably not cause trouble. In the event it is sloped, the case must be faced off with as light a cut as possible (if a stock chain is to be used) or a little more metal can be cut away if a thrust washer is installed behind the cam sprocket (see camshafts).

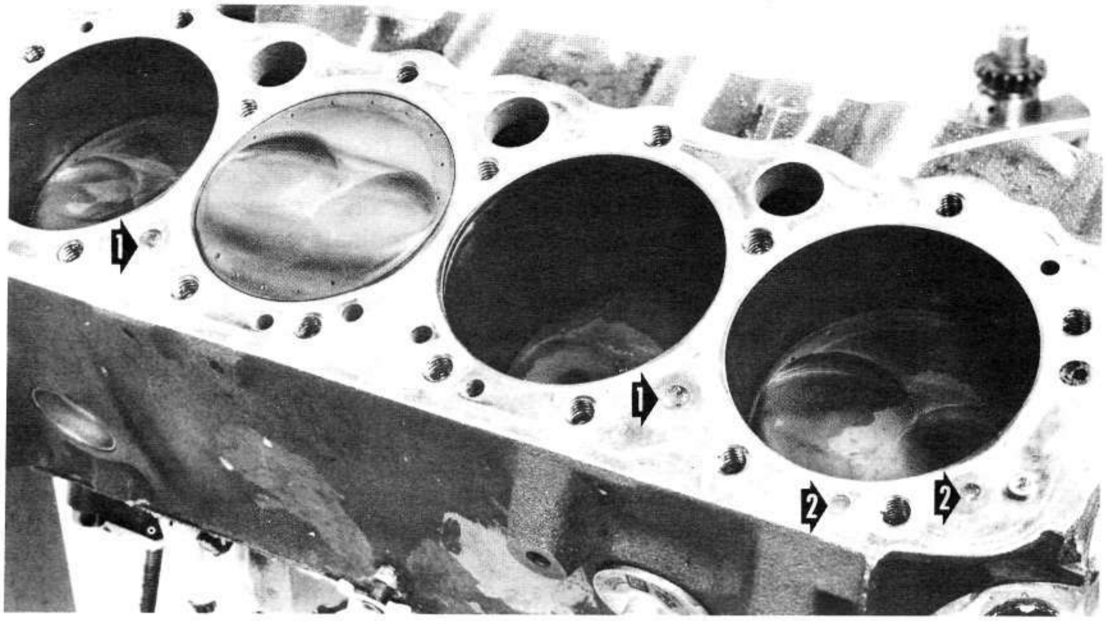
BLOCK OILING

The stock oil routing is completely adequate for any racing service. As long as all the passages are free from obstructions it will handle everything adequately. If the specific application allows for the use of roller bearing-mounted rocker arms it is possible to restrict some of the oil fed into the lifter/pushrod/rocker assembly. We definitely prefer to do this on our engines. The oil belongs down around the crank and rods and not up in the rocker boxes. The roller rockers can operate with less upper side oiling so we restrict the rear main gallery feeds to the right and left lifter banks. We tap the passages and use Carter carburetor screw-in jets with 0.050-inch orifices for this purpose. This setup works well with all types of lifters. It should not be used if stock rocker arms are installed as the reduced oil supply will lead to burning of the rocker balls.

Allowing full oil pressure to the valvetrain causes many problems. Drainback from the heads is usually inadequate so it is better just to keep the oil out of the heads. Full pressure to the lifters results in a tremendous amount of oil splash from the lifter bores. This oil volume is more than most racers imagine and it winds up bouncing around in the crank housing and upper engine space. It is a proven fact that oil resistance or "windage" around the crank is a detriment to power. Any reduction of oil raining down over the crank from upstairs means more power so we like to just keep the oil out of the top and in the bottom where we can control it better (more about this in a later chapter).



Coolant circulation through the block is important. We make an effort to equalize cylinder temperatures. The bypass transfer passage leading up into the right head from the water pump inlet can be a source of water leakage. We block this off with a threaded plug, and put a similar plug in the head opening.



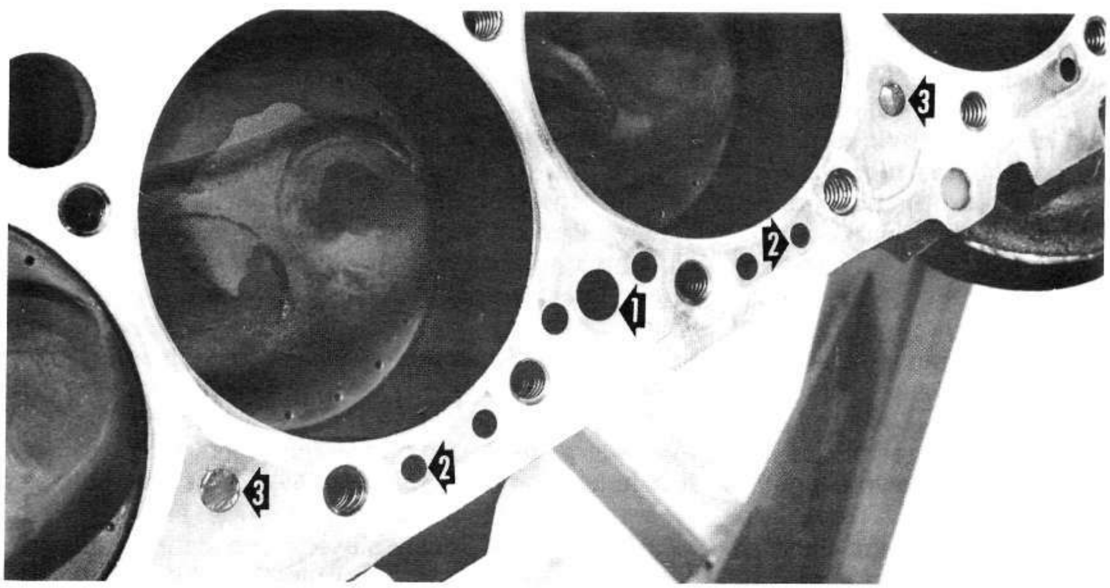
On the right deck we block off or restrict four other transfer passages in an effort to gain better heat distribution between the cylinders. The holes between the cylinders (arrow 1) are plugged to prevent cracking when the intake side of the chambers is cooled too much. Blocking the holes around number 2 cylinder (arrow 2) help warm this chamber, reducing the temperature spread in the right head.

We have thought of sealing the block above the cam and using an auxiliary pump, probably another scavenge stage on the dry sump pump, to pull the oil out of the rocker boxes and return it to the sump or reservoir. This might prove advantageous but we have not tried it yet and don't really know how effective it would be. It's certainly an interesting thought and it would be easy to do but it's questionable as to how much, if anything, would be gained. It would not solve the problem of splash from around the lifters. We think the best bet is to pump as little oil to the lifters as is practicable.

COOLANT CIRCULATION

Our dynamometer setup isn't as elaborate as some but we have found some definite cooling problems inside the case. Our tests show a temperature variation from one cylinder to the next. It might be as bad as 150° F from a good one to a bad one. We have tried as best we can to isolate this temperature problem from those of intake mixture distribution variations and exhaust pipe contour variations which will also cause cylinder-to-cylinder heat problems. It appears to be a built-in characteristic of the block construction and water circulation pattern. There are some variations in the pattern but normally the cylinders toward the front of the block operate cooler and those toward the rear hotter. As the water moves toward the rear of the block it becomes warmer. Consequently it does a less efficient job of cooling the rear cylinders. The tendency is especially pronounced on the left cylinder bank. When the water moves rearward some is routed upward into the cylinder heads through the matching water supply holes in the block and head decks. This causes some disruption of the heat distribution in the inboard four cylinders.

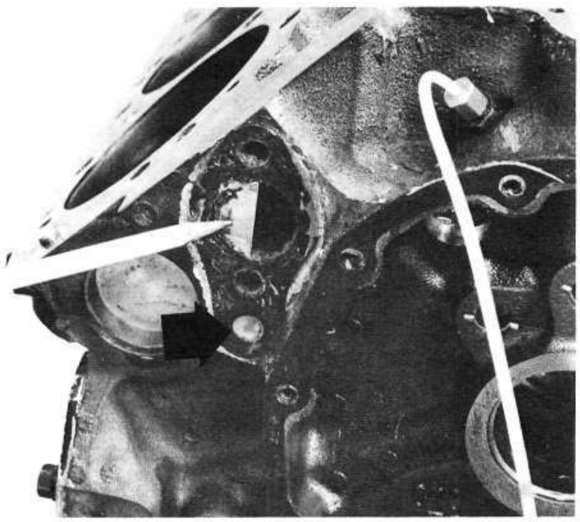
Obviously, pumping all of the coolant into the front of the cylinder block through the water pump inlets contributes to this uneven heat dispersion. We have tried several different methods to eliminate this disparity. To date we haven't found anything that has proven totally satisfactory. The general approach is to restrict some of the water feed into the front of the



Water control on the left includes drilling an extra hole between number 3 and number 5 cylinders (arrow 1) to get more water up between the exhaust ports for more cooling. Additional holes are drilled to feed the plug cooling holes, reducing detonation problems in this area (arrow 2). On this side of the deck we also plug the holes between numbers 1-3 and numbers 5-7 bores to suppress head cracking troubles (arrow 3).

block and/or to reroute some water externally to the rear of the case where it is fed into the side through the core holes or through drilled holes in the side of the block. The relative temperature of the combustion chambers has also been equalized somewhat by controlling flow from the block, into the heads, by restricting or blocking off some passages and drilling some new passages.

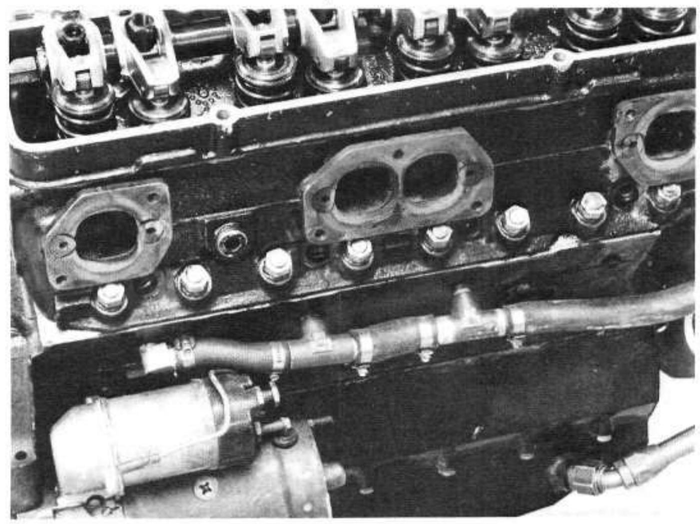
The approach is different for the 4-inch cases and the 4.125-inch blocks. In both instances we restrict the right water inlet to the block with a stainless steel metal partition. The restriction closes off approximately the outboard one half of the total inlet area. The water bypass passage is blocked off at the deck with a screw-in plug. We fill the rest of the passage with silicone by injecting the sealant upward from the passage opening in the block water inlet. This precaution prevents immediate water leakage into the number 2 cylinder if the head gasket seeps or fails. The bypass outlet happens to be the closest water source to any of the cylinders and is the most probable point of water contamination if the head gasket does not seal properly.



Inside the right water pump inlet we block part of the opening with a sheet metal partition. This is an effort to heat up the number 2 cylinder and force more water over to the other side of the block to cool down number 3 and number 5. Note also the bypass transfer passage leading to the right head has been completely filled.

In the experimental program with the 400 block we have had to resort to external water plumbing. Spacers are placed between the pump and the block inlets. Auxiliary outlet tubes are welded into the spacers to siphon some of the coolant flow away from the block inlet. On the passenger side this flow is directed through external hoses along the side of the block where it is put back into the block through a drilled opening approximately between the number 6 and 8 cylinders. On the driver side, the extra water is pumped into the front of the block through the core opening plug. We have also pumped water to the rear of the block on the left side but this has not shown a significant improvement in heat distribution. There are examples of different systems shown in the illustrations. The best of these has only produced a marginally acceptable heat distribution pattern in the siamesed-wall 400 casting and further testing may be necessary before a totally acceptable system is found.

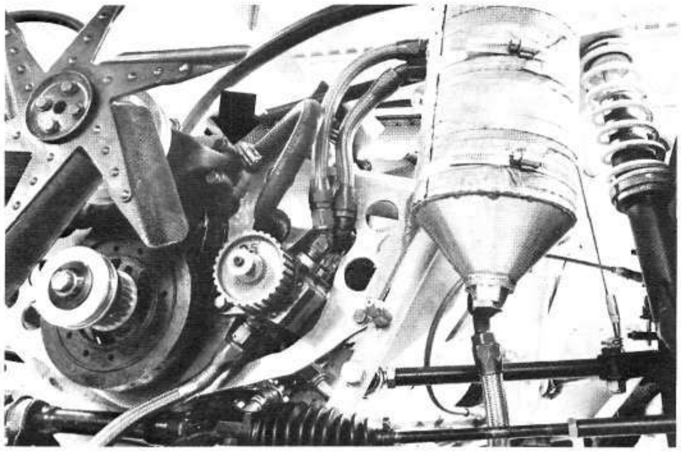
This temperature problem is not directly related to cylinder wall waterside heat. It is a consideration for efficiency in the upper cylinder and in the combustion chamber. For instance, since number 2 always runs very cool this would indicate the cooling system is drawing the heat out of the chamber too efficiently. For more power we want the heat to be retained to push the piston down in the hole and turn the rear wheels. The upper limit of heat retention is, of course, determined by the detonation limit of the chamber conditions and fuel. The spark advance and compression will be restricted by the hot cylinders and when there is a temperature spread between the chambers this means the cooler cylinders are not working to maximum capacity. What we want to do is reduce the spread so all eight holes will be running as near the detonation limit as possible when the engine is properly tuned. We want all eight working as hard as possible, just below the knock point imposed by the octane availability.



There is a heat distribution problem in the 400 cases when they are used for racing. We have not found the ultimate solution but we have spent many hours dyno testing different external coolant systems. This requires rerouting of water away from the pump, along the side of the block, and back into the case as needed.

In both the 4-inch and 4.125-inch cases we work toward this by slightly altering the water passages between the block and the head castings. We completely close off some and we enlarge others. This is all aimed toward equalizing the chamber temperature as best we can. The photos show very clearly which holes we are currently altering. On the left deck we plug two holes simply by injecting them full of GM silicone sealant and we block one with a pipe plug. We enlarge three holes as shown simply by drilling them larger. On the right deck we plug five holes completely. This may not be the ultimate answer but we feel it has helped our engines.

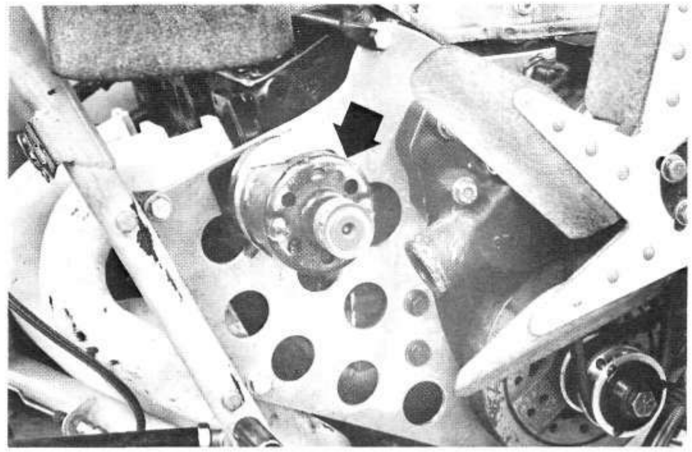
Regardless of what has been done to the circulation system it is interesting to note that the engine will



On the left side of the 400 cases we try to get more water into the front part of the case. Toward this end we put the extra water into the front core opening. The arrow indicates the spacer placed between the pump and block to serve as a source from which to draw pressure-side water. This coolant is then routed to external feeds in the case.

always run stronger when there is no water circulation at all. When the engine reaches the point of incipient boiling around the exhaust ports and chambers it makes bunches of horsepower. If you have done a lot of welding or brazing on the heads they will crack and the "solder" begins to fall out but when you need an all-out banzai run, it's one way to add power quickly. It can, however, be expensive.

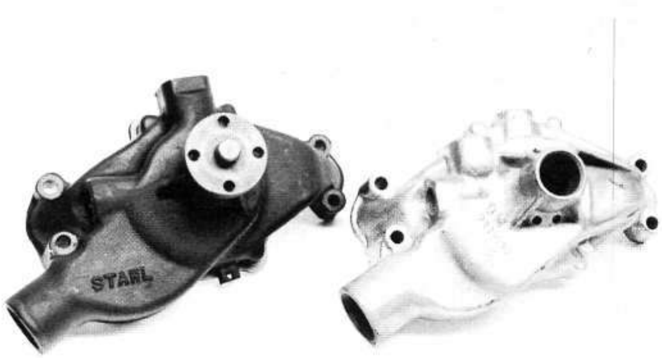
The number 8 cylinder runs fairly hot and we have seen some wall failures in our Grand National engines along the lower portion of the water jacket but we don't feel this is temperature related. At this point in time we are willing to attribute this to a vibration frequency response which is somehow concentrated at the forward wall of this cylinder. Vibration or natural frequency responses inside a superspeedway engine are extraordinarily severe and lead to many failures which we cannot track down on the dyno or in other controlled situations. They cause more problems than most people fully realize.



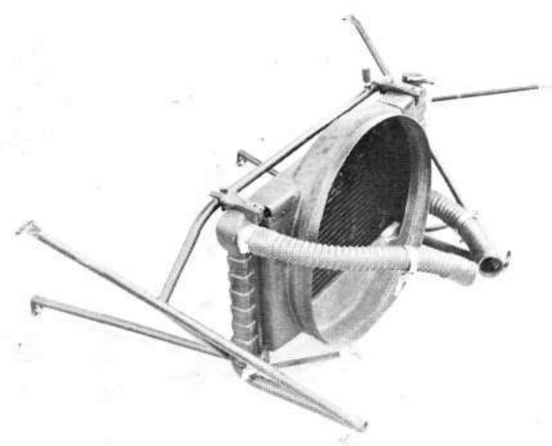
Creating very high temperature in the engine immediately prior to launching the car increases power. For many months we used a small electric motor to turn the water pump. When the car was staged on the line the driver could switch off the pump. This put more heat in the engine without risking overheating during pre-stage.

Overall temperature control is important because heat means horsepower. In our drag engines we like to see a water temp of about 180° to 190° F and preferably the temp will be fairly stable or equal around all the cylinders. At the time we launch the car it is absolutely essential to build heat in the cylinders as rapidly as possible. Naturally, you have to avoid boiling and we believe that 200° to 205° F is about the peak temperature point when leaving the starting line. This is with an all-iron engine. If the engine material is alloy aluminum in the heads or block the optimum temperature range is around 225° F. The increase is necessary to compensate for the very rapid heat dissipation of aluminum.

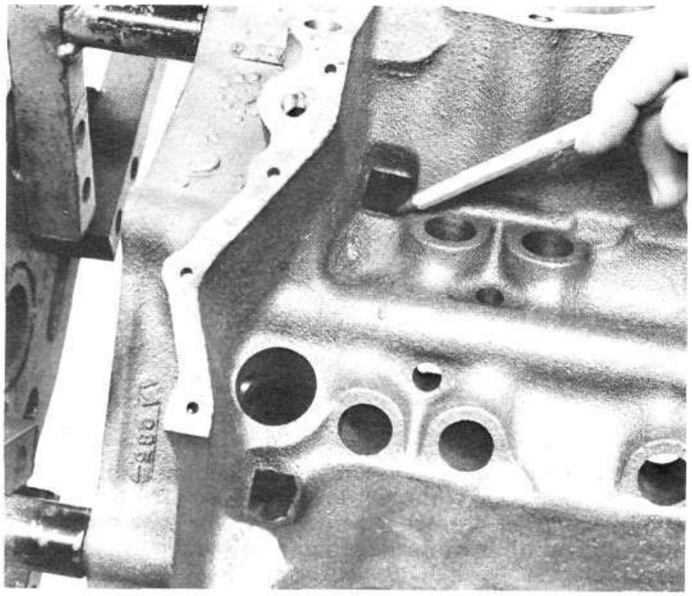
The picture is a little different with the circle track engines. On the super-ovals drafting is an important consideration. If we didn't have to allow for the possibility of overheating when the race car is running up tight behind another car, thereby cutting off the important air flow into the radiator, we could set the



On the left is the Stahl magnesium water pump that we prefer to use. The Moroso aluminum pump may also be used. The pump on the right is the very rare GM aluminum "military" pump. It has an additional pressure outlet, which may be helpful for some specialized applications.



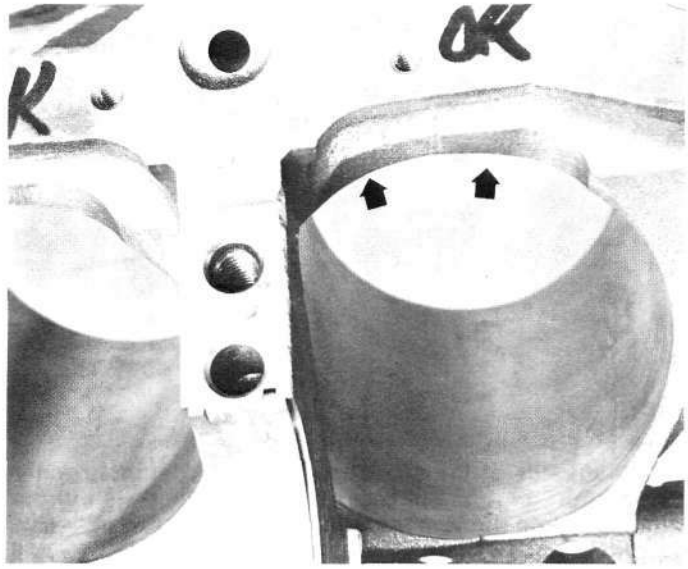
We use a standard Vega radiator for cooling. Because of the chassis construction we can utilize a very light forward support to hold the radiator and body in place. The entire structure can be removed by freeing the radiator hoses and pulling four quick-release pins.



Excessive grinding of the block is not necessary and we do not paint the internal surfaces. Cleanliness is the only important consideration during prefinishing. The area below this drainback hole is particularly troublesome. Casting "trash" is usually trapped beneath this ledge.

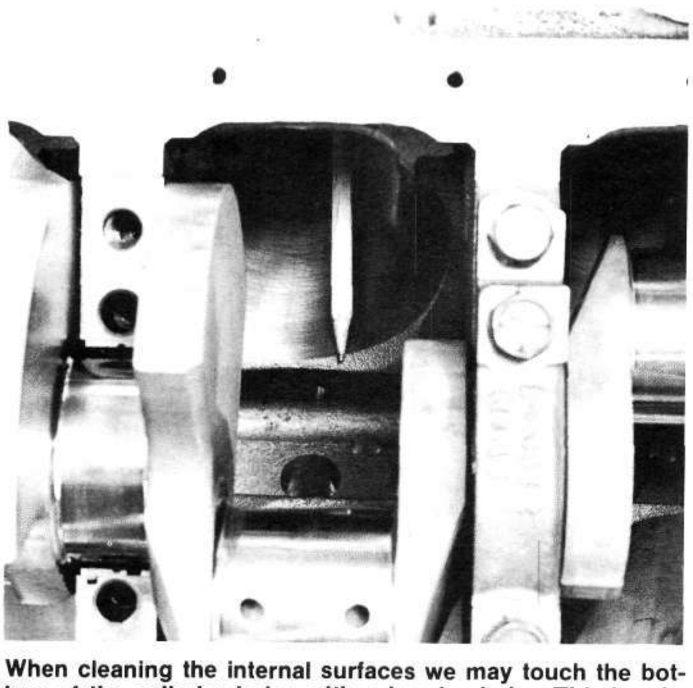
cooling system up to build a little more heat and power. However, the inevitable draft situation must be considered so we have to run the engines at 180° and build in a certain amount of overkill cooling to compensate for drafting. On short tracks the picture is much better and we would not hesitate to run the block hotter.

Speaking of temperatures, now is as good a time as any to discuss oil temperature. There is some power to be gained by controlling the oil temp properly! We can almost unequivocally say there is a flat one percent difference in power between 180°F and 220°F oil temp.

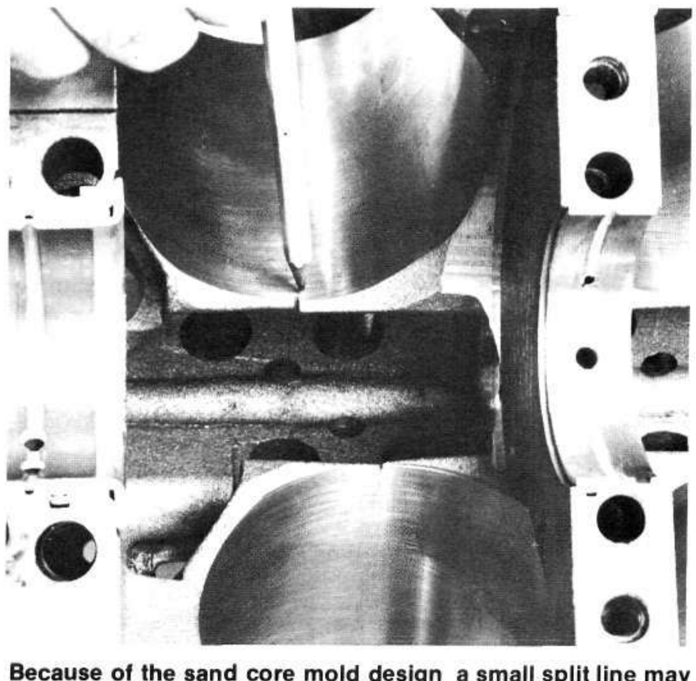


When long stroke cranks are employed, the lower edge of the cylinder bores must be ground for crank/rod clearance. The minimum amount of material should be removed to keep from shortening the cylinder bore walls. It is desirable for the walls to be as long as possible to support the pistons at BDC rock-over.

If we race any engine the oil temperature in the pan is 220°, period. This also eliminates any water which may leak into or condense in the oil. We know many people ignore this but we don't. At this temperature the operating mechanism seems to be somewhat insensitive to oil viscosity. We have tested 30 weight racing oil against 50 weight and we can find no measurable difference. This leads us to believe that this phenomenon is not viscosity related. It is more a matter of heat transfer to the lubricant. As a point of information we use 20-40 weight racing oil in our drag race engines and 40 or 50 weight in the track engines.



When cleaning the internal surfaces we may touch the bottom of the cylinder holes with a hand grinder. This breaks the sharp edge that may dig the piston skirts at BDC rock-over, but the wall should not be shortened.



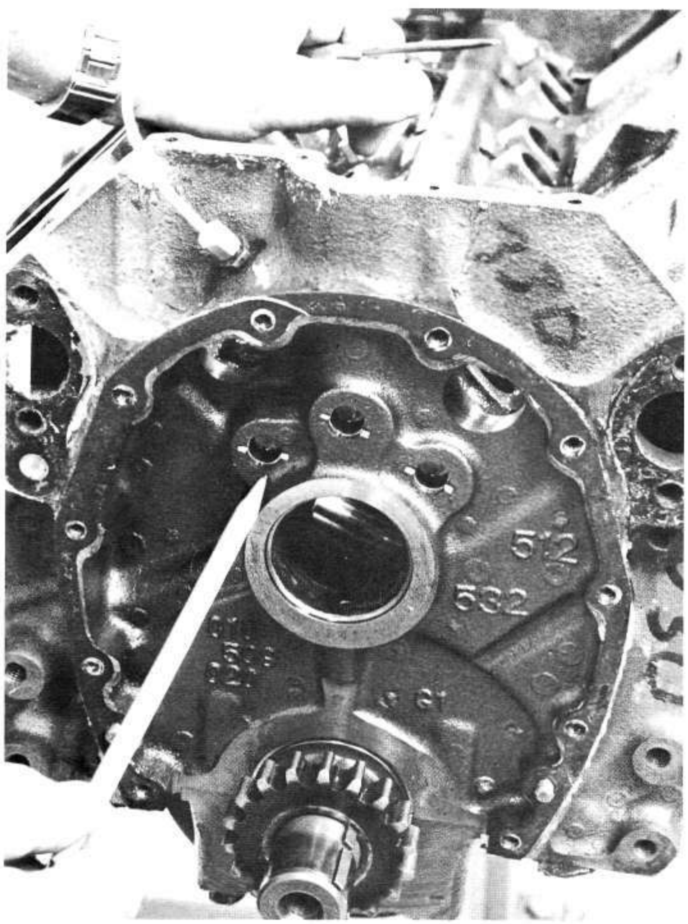
Because of the sand core mold design a small split line may appear at the bottom edge of the cylinder walls. We prefer to grind these notches away and the area should be inspected for minute cracks.

ASSEMBLY PREPARATION

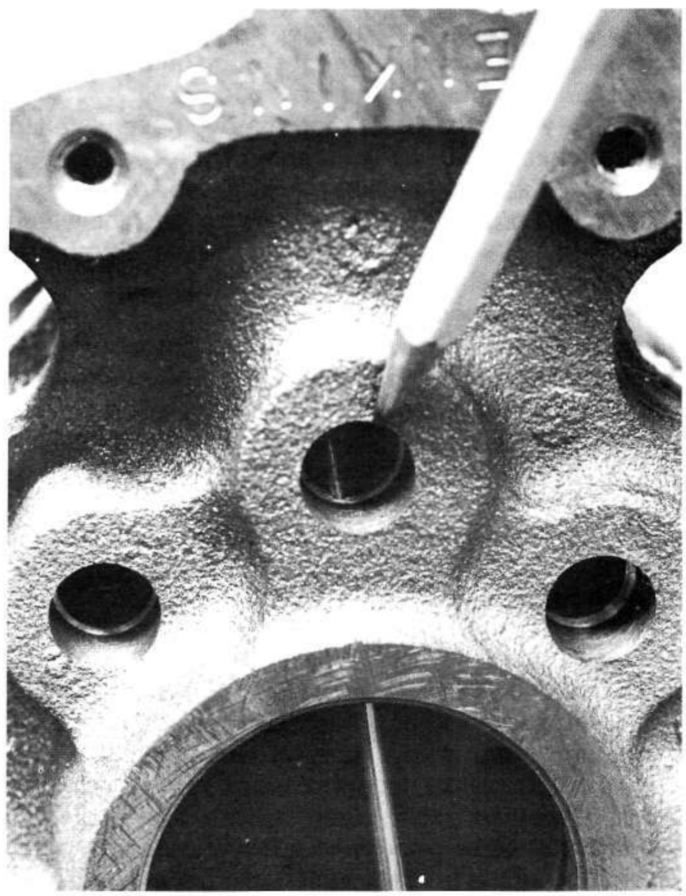
This is probably going to surprise some people and some may not even believe it, but we don't spend any more time than necessary massaging the block. We don't grind the hell out of it and we don't paint the inside. Some attention is given to knocking loose casting flash and the like but it isn't overdone. The most important thing is that the bare case be thoroughly cleaned. One specific area which has to be checked in every small case is the casting cavity below the oil drainback opening at the rear of the block. There's always a lot of crap down below the ledge formed around these holes. You have to get in there with some stiff brushes and knock it all out. This applies to a new or used block, any case that hasn't been well-prepped before. We also knock the cam bearings out of every new or used case as soon as we get it to make certain there aren't any unseen obstructions lodged in the annular oil supply grooves hidden behind the inserts. The main bearing oil supply comes down the center gallery over the cam, feeds downward through drilled passages into the annular grooves, around the grooves behind the inserts and then is forced down to the main bearing feed orifices drilled in the main webs. There is no need to modify any of these passages, just make sure they are all open to full pressure.

We run a brake cylinder hone quickly through each of the lifter bores to clean them up and we give a close inspection to make certain there are not any metal snags left over from the gang drills which form the lifter bores and oil galleries. The top edge of each cylinder bore is radiused *very slightly* and the bottom edge of each is also touched slightly to prevent any sharp edges from scraping the piston when it slides out of the bore at the lowest point of piston travel. The metal is thin here so some care is essential. Hitting this too hard with the grinder will substantially shorten the bore length and increase piston rock at bottom dead center.

When the block is ready to put in the working pieces we pound stock cup plugs back into the front oil gallery access holes and we install stock cup plugs into the core plug holes, except if we are planning to route water back into the block through one of the holes. We don't use screw plugs in the front of the oil galleries because it isn't necessary and is a lot of work to do properly. If by some chance you can't resist putting screw plugs in the ends of the oil tubes, make certain you don't run the center front plug in so deeply that it cuts off the oil supply passage running down to the front cam and main bearings. When the stock plugs are installed as prescribed in the factory manual they will not cause



At Jenkins Competition we always use stock core plugs in the front of the main oil galleries. They are carefully staked in place with a hammer and punch. We have never had trouble with these plugs and they are much easier to install than fancy threaded plugs.



Pressure oiling to the front main bearing feeds downward, behind the center gallery plug, into the drilled passage leading to the front cam bearing and down to the main. Note the shoulder inside the opening against which the stock plug is driven. This prevents it from going in deep enough to cut off oil to the main.

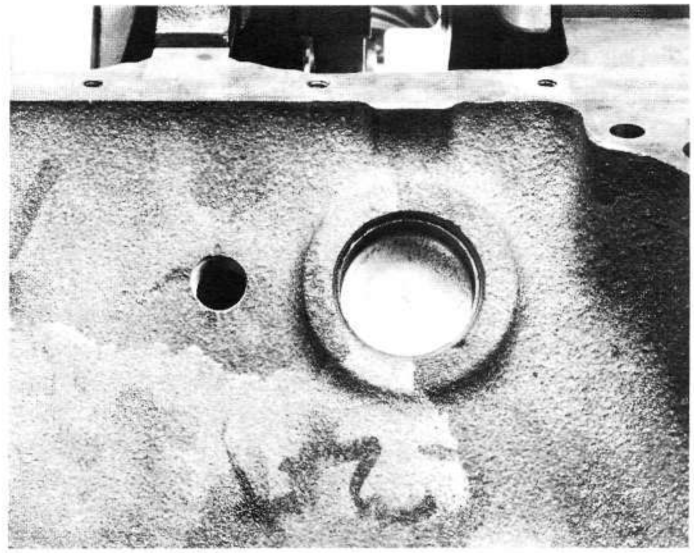
Bottom view of the front cam bearing seat, looking up toward the main feed from the oil gallery, shows the annular groove through which the main oil travels around behind the cam bearing. When threaded gallery plugs are improperly installed, this passage can be inadvertently blocked off.

any trouble. We have never had one fail. The same goes for stock core plugs. We pound 'em in and forget 'em. We have never found any reason to pin them in place or put any other sort of retaining strap across the holes. If you are putting water in through the hole we may weld a pipe fitting into the cup plug and since the plug must then carry some weight we do take the added precaution of pinning it in place.

You may note in the photos that we have installed block preheating devices in the forward core plug openings on each side of the block. These are standard GM preheaters available as options on engines which will be used in very cold climates. The part number is 735336. A few years ago there was a big splash about preheaters and keeping race blocks warm at all times. We don't think this is worth the effort but we do preheat engines prior to the first break-in run on the dyno. It does aid ring break-in, scuff prevention and sealing during initial start-up. Those first few seconds an engine runs and the initial ring run-in on the walls is super important. Anything to help assist proper break-in and increase sealing is well worth the effort. The GM heaters are inexpensive and easy to install. They operate on household 110-volt electrical source.

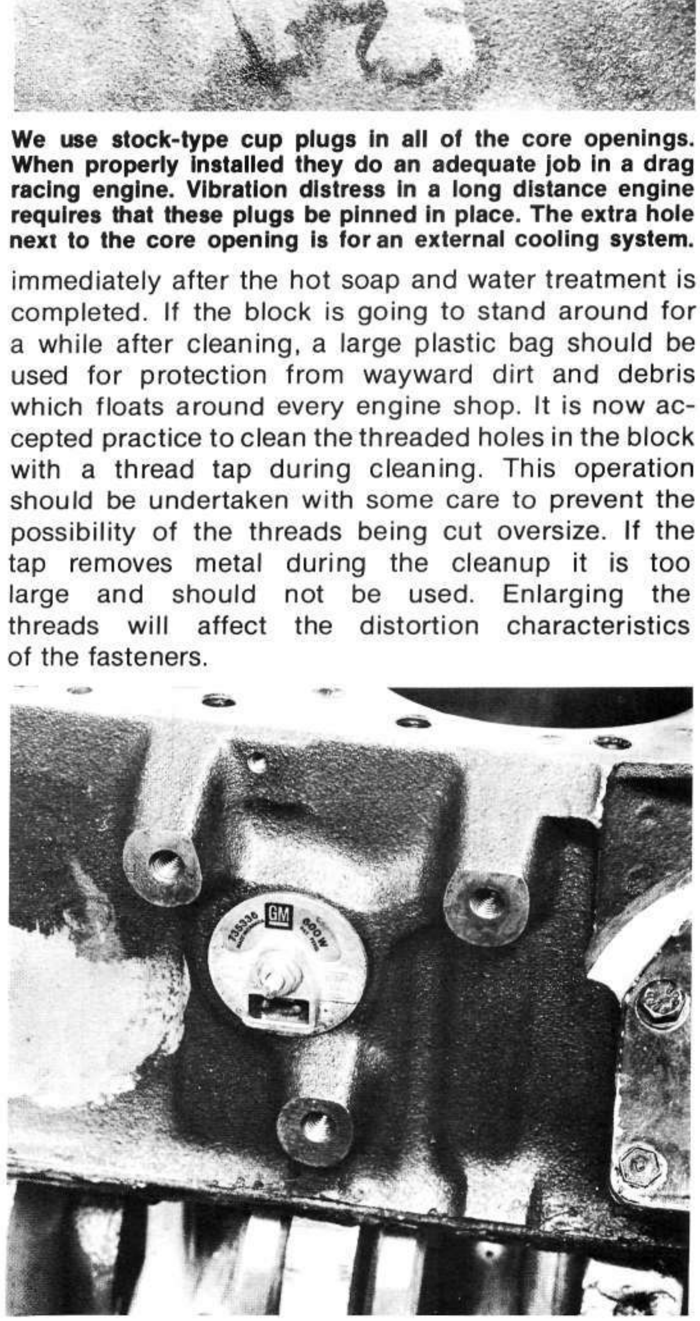
CLEANING

The last step before assembly is, quite naturally, the final cleaning. Much has been printed about the matter and there is very little we can add to the subject. It is impossible to get a block "too clean" and the only way to get it clean is with plenty of time and effort. It takes more than a few minutes to do the job right. Use the brushes which can get into the main oil galleries and all the little nooks and crannies. A high pressure water hose is a help and high pressure compressed air is a must to blow the water and debris out of the block when you are finished. The cylinder bores should be sprayed with a rust preventative

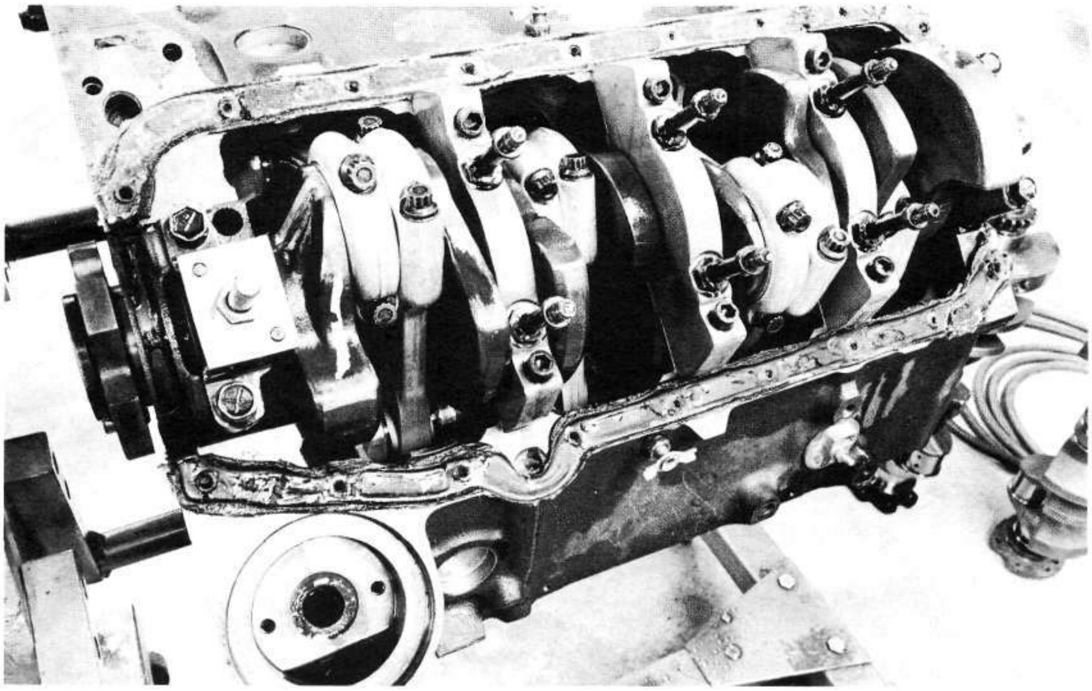


We use stock-type cup plugs in all of the core openings. When properly installed they do an adequate job in a drag racing engine. Vibration distress in a long distance engine requires that these plugs be pinned in place. The extra hole next to the core opening is for an external cooling system.

immediately after the hot soap and water treatment is completed. If the block is going to stand around for a while after cleaning, a large plastic bag should be used for protection from wayward dirt and debris which floats around every engine shop. It is now accepted practice to clean the threaded holes in the block with a thread tap during cleaning. This operation should be undertaken with some care to prevent the possibility of the threads being cut oversize. If the tap removes metal during the cleanup it is too large and should not be used. Enlarging the threads will affect the distortion characteristics of the fasteners.



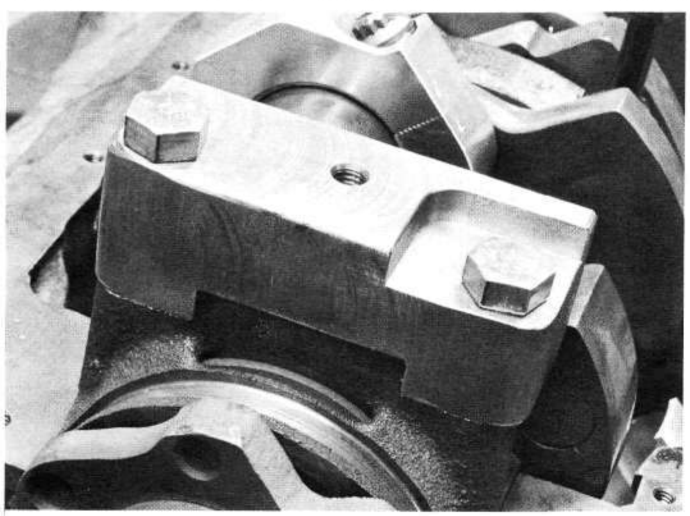
A standard GM block preheater is installed in the forward core hole on each side of the case. They are available as part 735336 and operate on standard household current. We use them only to preheat the case immediately prior to initial fire-up. This aids ring break-in.



If a 4.125-inch bore is desired, the best part to use is 330816. This is a 2-bolt case, and is preferred because the Chevy 4-bolt 400 block may suffer from cracking in the area where the outboard cap-bolts break out into the portion of the main webs that has been relieved for piston skirt clearance at the bottom of the stroke.

400 BLOCKS

We have discussed some of the 400 case eccentricities previously. They are available as part number 330816 (bare, 2-bolt) or as 3977676 (partial assembly, 4-bolt). If one is to be used for racing some additional work is required. First, the rear main cap must be reinforced. With the larger main bearing diameter used in this engine the caps are not any larger and there is practically no metal left in the rear cap. To beef up the rear main we machine a special steel support strut which bolts to the bottom of the cap using the cap bolts. We use longer, ½-inch bolts to compensate for the extra material depth. This might be a little more difficult with a wet sump engine as some provision must be made to mount and drive the oil pump.

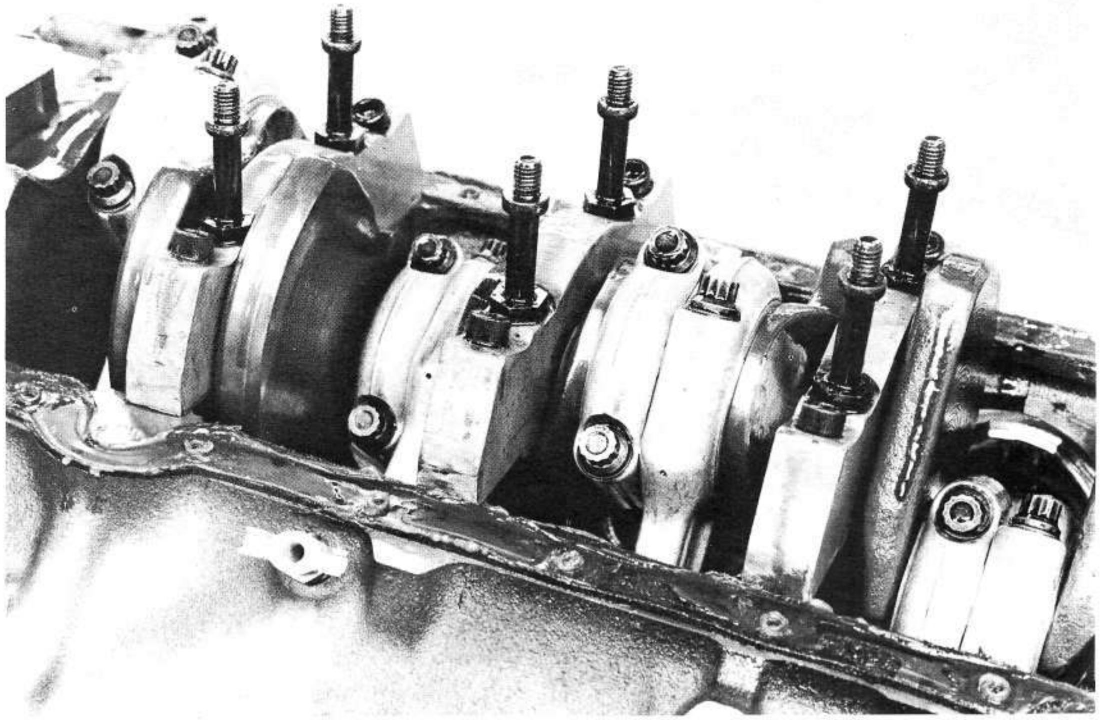


The rear cap of the 400 case is too weak for high output applications. When these caps are bored to accommodate the large-bearing 400 cast crank very little material is left to support the crank. This is especially crucial because all of the power generated in the engine is transmitted past this cap into the drivetrain. We always use a large reinforcing strap across the cap to keep it from rattling around under the load.

Since this is not a problem on our dry sump racing engines, the strap just reaches across the bottom of the cap, though we make provisions for a short stud to which the oil baffle is fastened.

At first glance it would seem that the 4-bolt block would be most desirable for a racing buildup. In fact, we prefer the 2-bolt block. When the 4-bolt caps are installed at the factory the drilled holes for the outboard cap screws break out into the relieved areas of the crank housing. These openings have proven to be a source of weakness for the main webs and some blocks have shown a tendency to develop severe cracks at this intersection. To eliminate this we have used 2-bolt caps with alignment dowels, similar to those used in Pontiac blocks. The cap mounting surface of the block is drilled to accept two steel dowels positioned outboard of the cap bolt holes. Stock two-bolt caps are drilled so the dowels will register correctly in the cap when it is installed. This should provide a better method to hold the caps in position without weakening the main webs. If this is insufficient the only solution is to install 4-bolt steel caps of the variety offered by the Summers Brothers, which have the outboard bolts angled outward along the approximate plane of the block outer wall.

We are still in the preliminary stages of our 400 block research but there seem to be several advantages to using this block in a racing buildup. By utilizing the larger bore it is possible to shorten the stroke to gain any specific desired displacement. This will result in less piston speed and allows for a much better rod-to-stroke ratio. We will discuss this in greater detail later but the rod ratio is an extremely important consideration for breathing efficiency. Besides this important point, breathing will also be improved in the chamber of a 400 block engine because the cylinder wall is moved an additional 0.060-inch away from the valves. The breathing picture is good on the intake

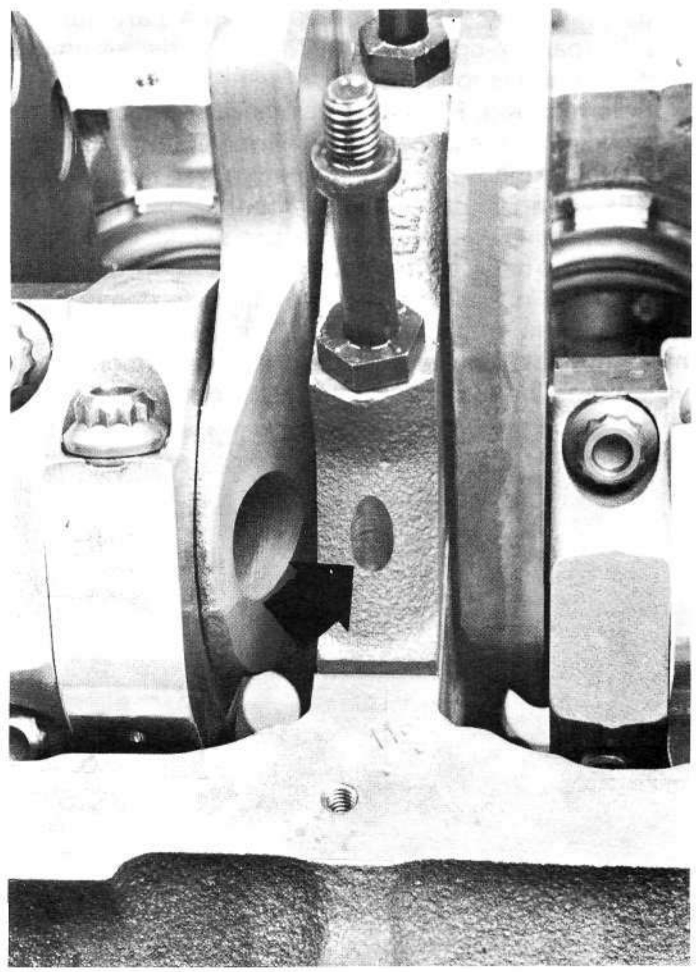


Whenever a non-stock 4-bolt cap is installed to the smallblock we prefer a nodular iron cap of the type that has the outboard bolts angled along the line of the outer block wall. This pulls the lower portion of the case together and reduces the possibility of cracking in the main webs.

side because the wall opposite the port exit is moved away from the valve head and flow into the chamber is very, very significantly improved. If we find that these benefits are not offset by possible wall flex problems or heat distribution trouble with the siamesed bores, we will probably be using the 400 block in more of our racing engines.

The one physical drawback that we definitely know exists at this point is the main bearing bore size. To counter crank flex problems in the 400 engine the Chevy engineers enlarged the bore size to 2.65 inches. This allowed them to use a beefed-up cast iron crank in this engine with great success. The increased diameter is, however, not an asset for a high-speed racing engine. To reduce bearing speed and fit a reasonable crankshaft into the big saddles we use a process

Whenever possible we always use stock Chevy hardware to bolt the engines together. These bolts are made from 300M steel and have a very desirable Young's modulus, preventing the threaded section from stretching excessively. When studs are required they can be obtained from several sources but must be mounted in epoxy for adequate thread perch.

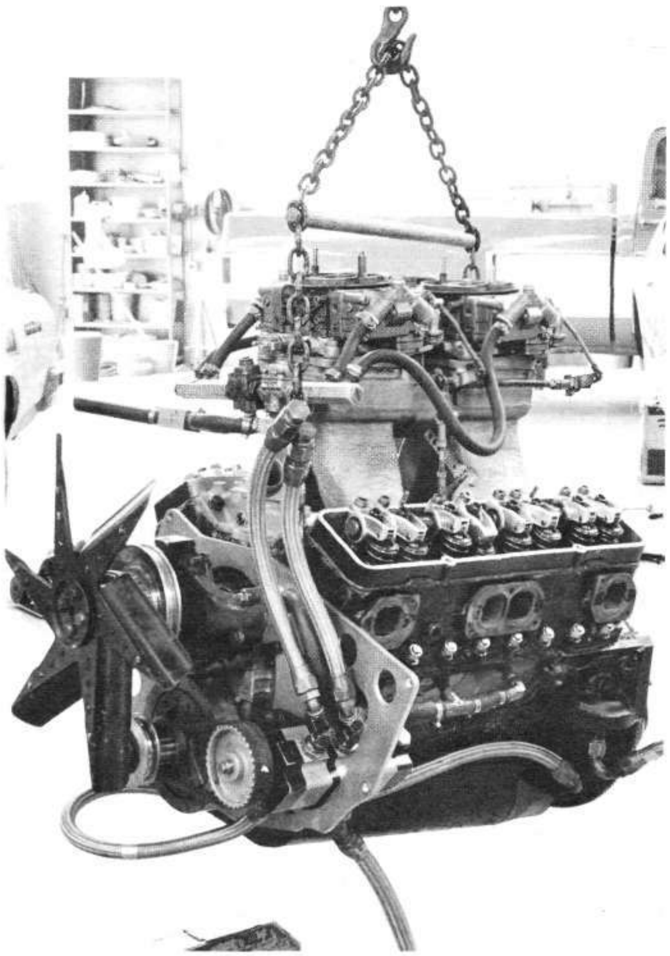


We have also experimented with 400 cases utilizing 2-bolt caps and pilot dowels, as used in the Pontiac engines, to hold the caps in alignment. We have only limited experience with this technique but it makes preparation simpler and has worked adequately in the experimental match race engines.

similar to that described earlier with the 350 block. The stock 400 bearings are pinned in place and honed to accept the 350 bearings which are just laid on top of the pinned sleeves.

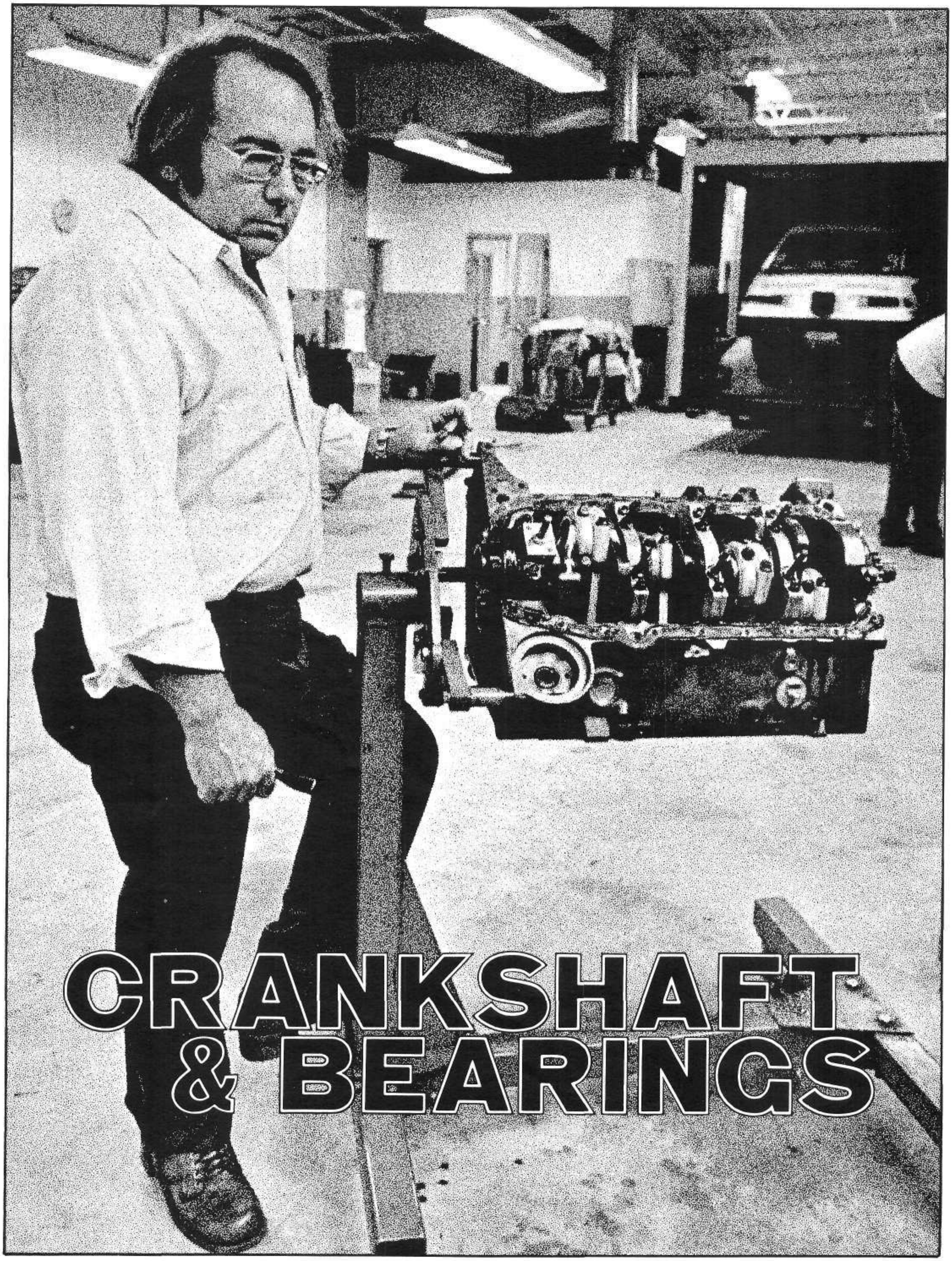
In one of our drag racing motors we would still prefer to use the early 327 crank with 2.30-inch nominal diameter main journals. This presents somewhat of a problem. We have tried two solutions and both have worked successfully. There are currently some "Space Saver" bearings available from TRW to drop the 350 crank right into the 400 case without requiring a sleeve. To accommodate the early 3.25 crank we have used one set of these bearings to make sleeves in exactly the same method as described during the 350 case discussion. Then we have installed stock small-diameter 265-283-327 bearings on top of the Spacer Saver sleeves (after align honing the sleeves to proper saddle size) and it has caused no problem. The alternative is to fabricate the sleeve entirely from aluminum material. A sharp machinist can probably accomplish this with very little trouble. The outer diameter of the sleeve would set into the block and cap saddles with some slight crush and would be pinned in place. The inner diameter would then be align honed to the required spec to seat the early 2.30-inch nominal diameter inserts with required interference crush. Again, we would probably shoot for the small side of the stock factory specs to allow for the additional crush from the aluminum spacers.

Our testing has also proved that some special attention must be paid to head gasket sealing. We don't normally employ studs to clamp the heads in place but in the 4.125-inch block they are essential. These special cases will be described more fully in the chapter on cylinder heads and head gaskets.



Since most of the competition are running match race engines displacing 400 or more inches we have been forced to test some large displacement engines and the results have been very, very satisfactory. We have even had some luck running 4.125-inch cases filled to within one inch of the deck with Devcon. Such techniques are only suitable for short duration drag racing.

There is very little mystery or "trick stuff" involved in the assembly of a small-block Chevy short block. Most of the effort is spent checking, inspecting and measuring critical areas. Once the clearances are properly set the basic working mechanism will operate to 10,000 rpm without the slightest difficulty.



CRANKSHAFT & BEARINGS

BORE-TO-STROKE RATIO

Generally, the mechanical efficiency in a racing engine, where higher crank speeds are not a problem, is better with a larger bore and smaller stroke. The short stroke results in less frictional loss as piston speed is reduced considerably. The breathing ability for a given cylinder displacement should be increased with a larger bore as there is simply more area across the bore in which to fit the larger valves. The shorter stroke will also allow relatively shorter rods and a shorter block deck height. All of which adds up to reduced engine weight. Of course, this is a very complex technical consideration as it is possible to have the rod ratio too high for the displacement, resulting in induction feeding difficulties. This is a matter of great balance and extremely fine design skill. As a point of interest, the late Leo Gossen, celebrated designer of the Offenhauser racing engine, probably had a better understanding of the overall design parameters in a racing engine than most people generally recognize. Success in any racing engine requires an ability on the part of the builder to visualize these complex interrelated principles. Frankly, this only comes with a good deal of experience and hard work.

STROKE-DISPLACEMENT SELECTION

Stroke length-versus-performance is always a good debating point. Stroke selection may be dictated by the rules of a specific sanctioning organization or by whatever suitable cranks are available from the factory. Beyond this, it is interesting to consider some of the power studies we have conducted and some of the results from other well-known evaluation programs.

The common factory smallblock stroke lengths are: 3-inch, 3.25-inch, 3.48-inch and 3.75-inch. In low speed engines there is some degree of performance gain with every increase in stroke, all the way through 3.75 inches, as used in the 400 smallblocks. There is not, however, a linear increase in output—power per inch—as the stroke goes up. There may be more torque and with considerable work it is possible to attain a pretty good specific power curve up to a stroke of 3.48 inches.

When maximum specific power is desired for racing purposes other considerations enter the picture. In this instance the stroke length must be dealt with in terms of the available rod length and piston design. These are important factors in the induction efficiency of the engine and, after all, this is where the game is won or lost. In simplest terms the reciprocating piston internal combustion engine can be viewed as a pump. The better it draws in combustible gases, burns them, turns the crankshaft and pumps them out, the more power it will produce. At the current time we are working day and night to improve the intake side of our "pump." We have found most of our recent gains through this research (as have many others).

Considering the requirements of *induction efficiency* as related to the stroke length, rod length and piston design, it is possible to gain a fairly decent rod length-to-stroke length ratio at any of the stroke lengths between 3 inches and about 3.50 inches. When the restrictions of piston design and induction system design enter the picture we feel the best "compromise" is somewhere in the middle ground - a stroke length of 3.25 inches. However, any of the common stroke denominations will produce suitable power curves if the overall engine is properly designed.

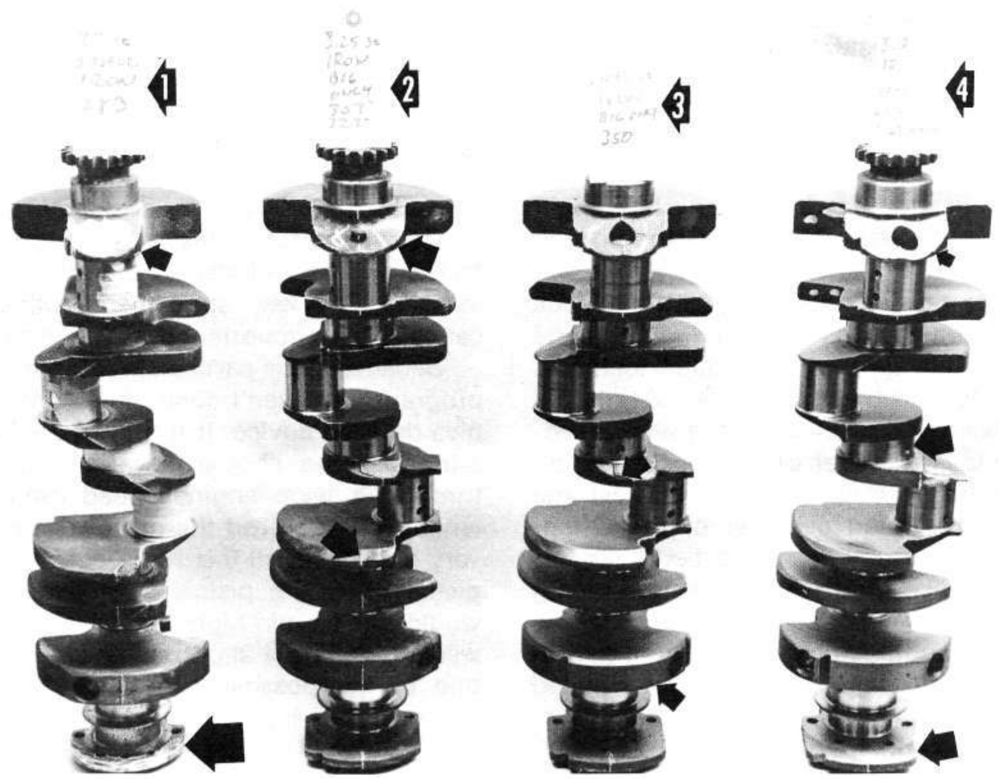
Detailed discussions of these many contributing factors will be included in the following chapters. We must touch briefly upon them in this discussion because of their importance to the crankshaft design.

Because of the particular requirements of our racing program we haven't done much 302 work so we can't give detailed advice. It would seem, however, that the 3-inch engine should have a high specific power through a wide engine speed range. With a stock length 5.70-inch rod the piston compression height is very high. With all this room between the pin and the piston deck, the piston is very heavy. A longer rod would put the pin higher in the piston and save some weight. This is a slight secondary consideration and one of the possible advantages of longer strokes. However, this increases the rod-to-stroke relationship and results in relatively restricted induction breathing with current manifolding and cam design.

A stroke length of 3.25 inches or slightly shorter provides an excellent mechanical compromise. The rod length can be nearly optimized without severely restricting the piston design. Currently-available tunnel ram-type induction systems will contain sufficient volume to feed the cylinder displacement. At the same time the cylinder head ports can be reworked so they do not become a very severe restriction (though this is still a big problem) and the camshaft design can be such that the cylinder feeding versus displacement is sufficient within the current mechanical limitations of the valvetrain actuating mechanism.

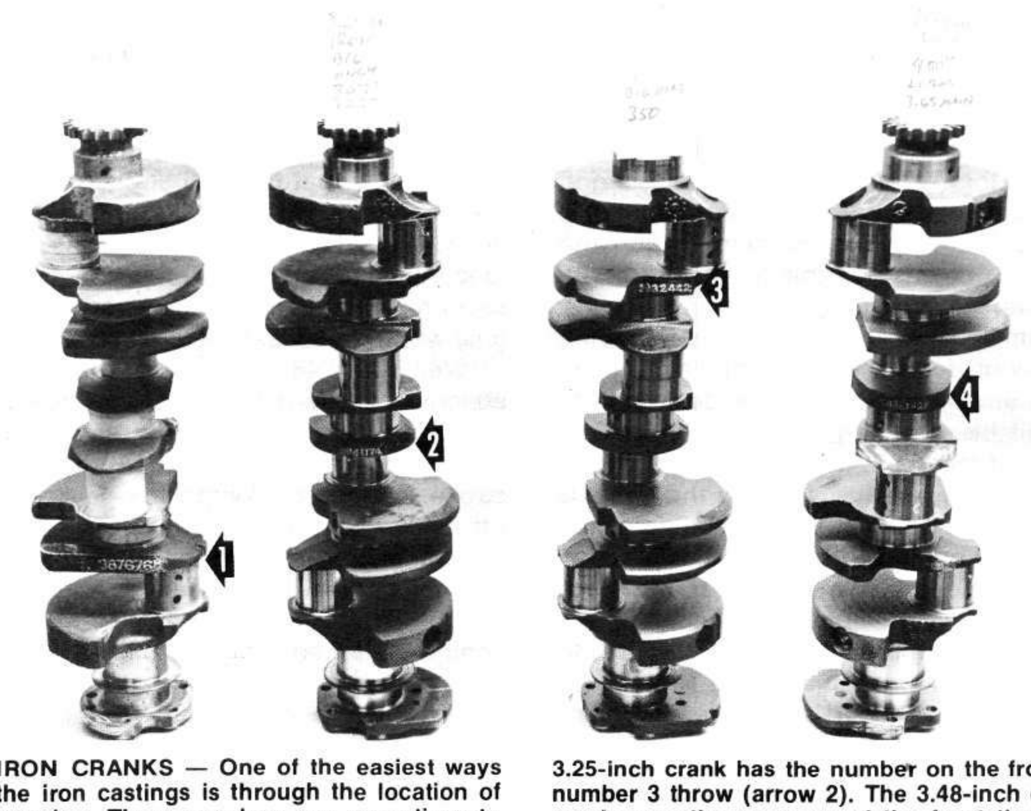
We know that the 3.25 stroke likes a rod length of 5.90 inches (a 1.8:1 rod ratio). This is what we use in our 330-inch Pro Stock engines. This rod is only slightly longer than the stock 5.70-inch connecting rod. It would seem reasonable that a stock-rod 327 cubic street engine would be an excellent all-around performer!

With the 3.48 stroke it is still possible to get an adequate intake manifold runner size to feed the engine but the piston design becomes a problem because of the high pin placement. Above this engine size/ stroke length/ rod length the peak power curve falls off rapidly beyond about 8000 or maybe 8200 rpm. At greater engine speeds the intake is just not able to feed the chambers adequately. As an example, with all else being equal, if a "good" 3.48-inch stroke 350 c.i. engine is stroked out to 370 inches, the optimum power point will drop from 8000 rpm to about 7600 rpm. We have some 3.48-inch stroke, 354-inch drag motors with 6.25-inch rods (ratio of 1.8:1) which we use as "iron-ball" tire test engines and they work quite well. In our good 3.48-inch, 354 c.i. Nascar engines we use 5.85-inch rods (a ratio of 1.7:1). Some of the Nascar builders are now using 6.0-inch rods but we favor the



COMMON IRON CRANKS—From left to right we have a comparison of 3-, 3.25-, 3.48- and 3.75-inch stroke cast iron cranks. The 3-inch, 2.30-inch journal crank (arrow 1) has a narrow front arm and a full circle flywheel flange. The 3.25-inch iron crank (arrow 2) is available only with 2.45-inch journals. The front arm is much wider than the 3-inch crank. Note the thin casting seam that readily identifies all cast iron cranks.

The 3.48-inch iron crank (arrow 3) is available only with the large journals. It is difficult to tell this apart from the 3.25-inch cranks. The counterweights are, however, generally wider and flattened in the center for piston clearance. The 3.75-inch cast iron crank (arrow 4) is the only long stroker available. It has unique 2.65-inch main journals. The casting has a slightly wider front arm and a larger flywheel flange weight.



COMMON IRON CRANKS — One of the easiest ways to identify the iron castings is through the location of the casting number. There may be some exceptions to these generalities but the number on the 3-inch crank is on the front arm of the rear rod throw (arrow 1). The 3.25-inch crank has the number on the front arm of the number 3 throw (arrow 2). The 3.48-inch crank has the number on the rear arm of the front throw (arrow 3). The 3.75-inch crank has the number on the rear arm of the number two throw (arrow 4).

slightly shorter rod and we have even built some 5.70-inch rod 350's which are almost as good as the long rod engines. The fuel specific curve isn't quite as good with these short rod engines, however, and the longer rod engines with the smaller volume single four-barrel intake manifolds that we are now using seem to be more flexible with better engine range, an important consideration in a circle track engine.

IDENTIFYING CRANKS

In most of our current racing engines we use brand new factory cranks. This is now just a matter of convenience, but like most racers when we first got started we had to economize in every way we could, and we often bought cranks from the local scrap or junkyard. During that time we developed an identification system to help quickly sort cranks and locate the specific stroke lengths and bearing sizes we use. With practice it is possible to pinpoint a crank at a quick glance and in some cases you can tell the difference even if the crank is still bolted into the case.

The accompanying photos show the major differences. Unfortunately, there are minor variations from one crank to the next within a specific category but usually there is more than one identifying characteristic or a combination of things which will make the i.d. positive.

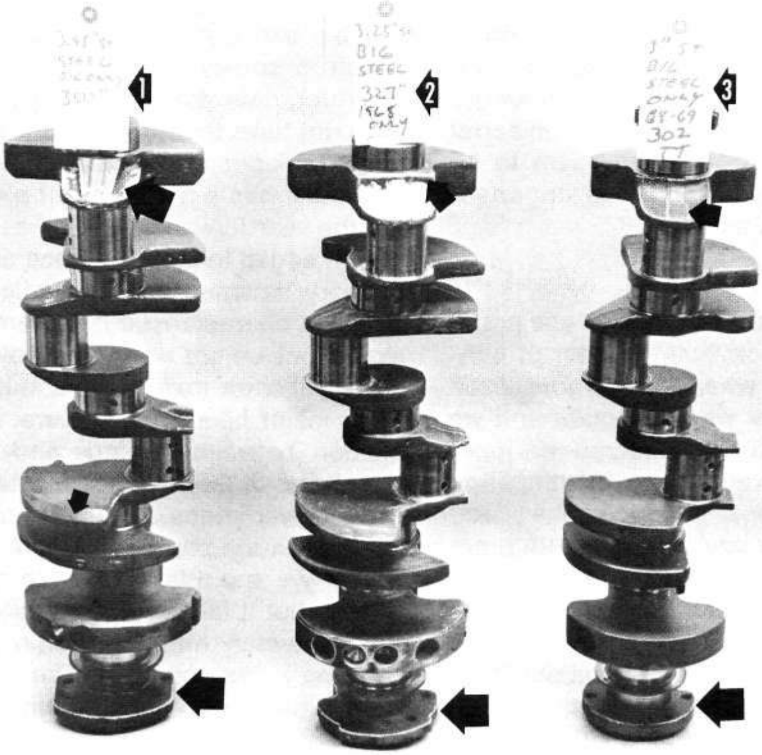
First, it is extremely simple to tell a cast crank from a forging. On all Chevrolet cast cranks the flash marks where the two halves of the mold meet is always evident. This flashing line assumes a characteristically peculiar convolution as it runs along the full length of the crank. All of the Chevrolet forged cranks are ground smooth after they are formed. There is no residual metal flash along the edges of the counterweights and throw arms. This is an instant way of identifying box-stock cranks if you are looking for a forging. In some rare cases a modified cast crank may have the flash line ground away but there is still a very distinct difference between the rough, somewhat grainy surface of the castings and the smooth even texture of the forging.

Within the family of forgings, the cranks that are of most interest to racers, it is fairly easy to tell the major stroke lengths apart by looking at the front arm of the number 1 crank pin and the flywheel flange. All the late 3-inch, 302 cranks have a round flywheel flange with a single notch for balance, the front pin arm is relatively narrow (similar to the 3.48, 350 crank), but the counterweights are semicircular and are not flattened in the center portion of the radius to gain piston skirt clearance at bottom dead center (as are all the cranks with strokes longer than 3 inches). The old 3-inch, 283 forgings are very rare but if you come across one it can be identified by the completely circular flywheel flange with no balancing notches. Most of the 3.25-inch, 327 cranks also have round flywheel flanges but there are two notches forged in the flange; the most distinguishing characteristic is the very wide front pin arm and, of course, the counterweights are slightly flattened for piston clearance. There is a notable exception in the 1968 big journal 327 crank which is somewhat similar to the late 350 forgings. However, this crank, as you can see in the photos, does not have the forged-in rib on the front arm of the front crank pin as does the 350 arm. The late 3.48-inch, 350 crank has a narrow front pin arm with reinforcing rib, the rear flywheel flange has a very large counterweight added to assist balance and the counterweights are very flattened in the middle for piston clearance. This last characteristic is extremely noticeable on the large front weight when it is viewed from head on.

We all know that Chevy cranks have been offered in three major bearing diameters. In addition to the distinction between forgings and castings and the different stroke lengths, some attention has to be given to these variations. When the cranks are next to each other as in the photos, the different sizes are apparent to the eye and it is possible to spot the differences by eyeball but it takes some experience. The early cranks had 2.30-inch (nominal) main journals and 2.0-inch (nominal) rod journals. The late 302, 327 and 350 cranks have 2.45-inch (nominal) mains and 2.10-inch (nominal) rods. The unique 3.75-inch stroke 400 c.i. cast crank has 2.65-inch (nominal) mains and 2.10-inch (nominal) rod journals. The 3-inch, 283-302 crank is available in 2.30-inch and 2.45-inch bearing *forgings.* All the 3-inch *castings* are of the 2.30-inch type and, as far as we know, there are no large journal 3-inch stroke cast cranks. There are large and small journal 3.25-inch stroke forgings but we know of no small journal 3.25-inch stroke iron cranks. All of the 3.48-inch stroke 350 cranks have the large 2.45-inch main journals, both steel and iron. There are no small journal 350 cranks. As we mentioned, the 400 crank is only offered in a cast iron version from Chevrolet with the very large main journals but several of the specialty crank grinders offer a forging or a welded forging with 2.45-inch bearings which can be ground to the 3.75-inch stroke or even longer.

SELECTING A CRANK

We use forged cranks in all of our race engines because they are readily available and suitable for our particular needs. Nodular iron cranks are, however, very interesting when endurance is a primary requirement. Chevrolet uses cast iron cranks in most of their current light truck engines and they serve quite adequately despite severe imposed loads. A cast crank does not have as much ultimate strength as a comparable forging, but when peak loading is not as severe they have an important advantage. Because of the inherent properties of the iron material they will dampen torsional vibration and frequency response problems. In a 500 mile Grand National engine this will possibly counteract the component failure rate which always plagues an engine operated under full load, at high rpm, for long periods of time. We feel this is not really an ultimate strength problem but is a natural frequency vibration response, which raises havoc



COMMON BIG-JOURNAL STEEL CRANKS—From left to right we have a comparison of the 3.48-inch, 3.25-inch and 3-inch steel forgings, all with 2.45-inch main journals. The easiest way to identify these cranks is through an inspection of the flywheel flange counterweight. The 3.48-inch crank (arrow 1) has a large squarish weight. Note that on the 3.48-inch and 3.25-inch crank some of the counterweight edges are tapered for piston skirt clearance. The 3.48 crank has a distinctive rib forged into the front rod throw arm. The rear flange counterweight on the 3.25-inch crank (arrow 2) is roundish but there is definitely some extra material added, compared to the completely round 3-inch crank flange. The front arm of the 3.25-inch crank does not normally have a rib and it has a squared-off face. The 3-inch crank (arrow 3) is very easy to spot. There are some forging marks on the front arm but no rib; the weights all have square edges and the flywheel flange is round.

COMMON BIG-JOURNAL STEEL CRANKS—A backside view of the big-journal cranks shows that these forgings are very similar. Again, we can see that the flywheel flange is the easiest variation to spot. The counterweight/throw arms of the 3.48-inch crank are wider and stronger as can be seen on the rear arm of the number 4 throw. On the forward arm of the number 4 throws you can see the wide forging line that easily identifies the forged cranks (provided they have not been ground smooth). The 3.48-inch steel crank is available only with 2.45-inch journals. The 3.25-inch steel crank with big journals was available only in 327 engines in 1968. The 3-inch big-journal steel crank was available only in the 302-inch Z/28 engines in 1968-69. It is a Tufftrided crank.

throughout the entire engine. It is not unusual to take an engine apart after one long race and find stress cracks all over the place. In any case, we only run our forged cranks for two five-hundred-milers at the most before we send 'em to the junk heap. It is interesting to note that the Ford guys are using nodular cranks in their track 351's with considerable success and we have no reason not to believe a 350 Chevy crank could do as well. In a high-rpm drag engine, however, a forged crank is a must.

As mentioned earlier we don't have a great deal of current experience with the 3-inch stroke engines. Our guess would be that the small journal forging from the '67 Z/28 302 engines would be a good place to start for a drag engine. The Formula 5000 guys who need a little more endurance are probably using the big journal late 302 Tufftrided crank which is a more durable piece. The early 283 forging is a very light crank, probably the lightest Chevy crank extant, but it is very near the borderline in strength; at least if you are building the power levels currently required in Modified Production and higher classes.

In the 3.25-inch family there are a number of part numbers from which to choose. The '68 big journal 327 Tufftrided crank is probably the second lightest Chevy forged crank ever made, next to the 283 steel arm. *It is four pounds lighter than the small journal 327 model.* The crank pin arms are necked down and the counterweights are cut down some. We don't really consider crank weight is very important as it just acts as flywheel mass within the engine, but for those who do think this is important this crank may be of interest. In the past we have drag raced this crank in various stroke lengths from 3.17 inches to 3.33 inches without any failures. We do not suggest that this crank be used in high-horsepower endurance applications such as road racing or circle racing. In these instances one of the heavier, beefier cranks must be used.

In our best 330-inch drag engines we use the '62-'67 *small journal* Tufftrided 327 cranks, number 3838495. This is the heaviest forged crank built by Chevrolet and for our drag racing purposes it is the strongest and best-suited. As you can tell from the photos this is a very "herky" piece. We have been using these cranks with only some slight modification and have had no trouble whatsoever with them. Specific modifications needed to race-prep the crank are detailed in the following section.

At this point there is a "rumor" afoot that this same basic forging may be reworked by the factory to add material on the mains and rod journals. The intention is to make a superheavy-duty semi-finished forging which can be ground to any stroke length from 3.10 inches to 3.60 inches *with the large diameter main and throw size.* This will obviously make an extremely nice crank for Nascar engines and, perhaps, some special drag applications. This is just an unconfirmed bit of speculation but there are a number of builders around the country who would love to get their hands on such a forging.

As a matter of speculation we find the Grand National possibilities with this proposed crankshaft quite interesting. Earlier we discussed the advantages to running a 4.125-inch bore block. Such a case with a 0.080-inch stroked 3.25-inch crank would put the displacement at around 356-358 inches, depending upon final bore size. We can't turn the good, little journal crank that far and right now such an engine would have to be built from a big journal 3.25 crank with offset ground small journals to lengthen the stroke sufficiently. It is doubtful, at least in our opinion, that such a crank could withstand a 500-miler. There's also certainly a good deal of skepticism surrounding the 400 case in such endurance applications. However, if we can bring the walls around in the big hole case and get a stout 3.25 plus 0.080 crank it would be a plenty sweet piece (on paper at least).

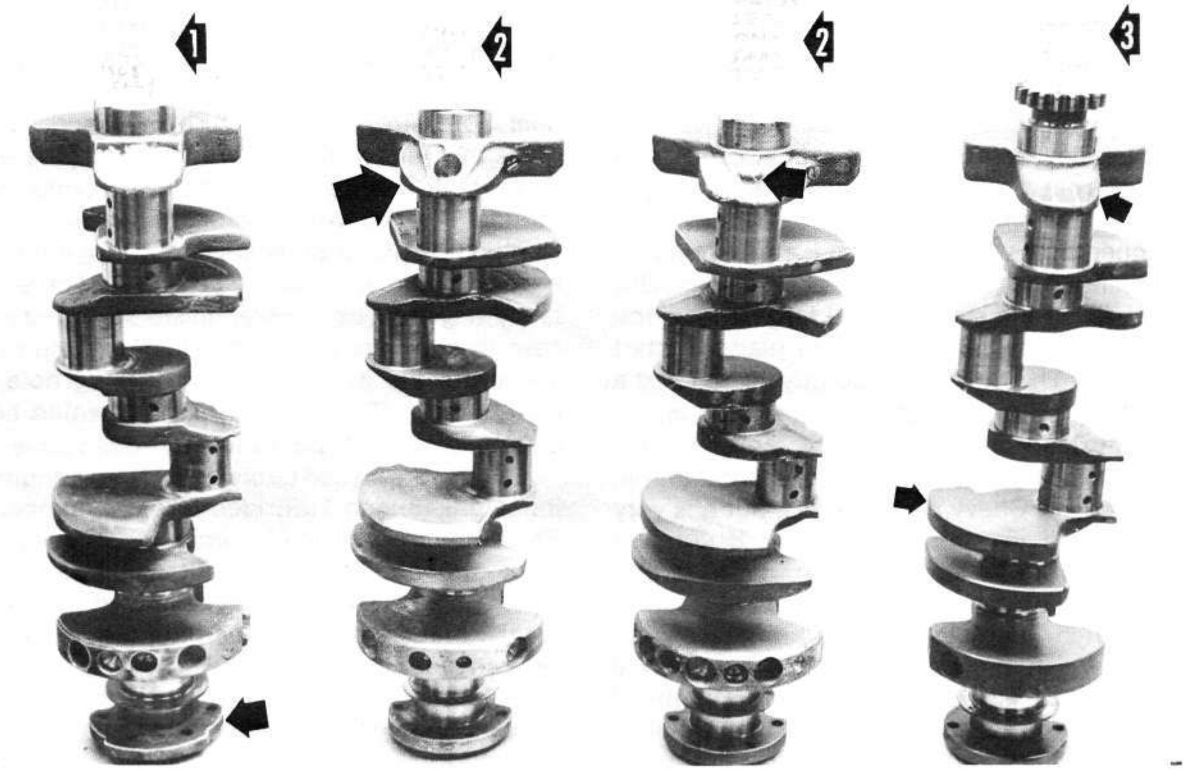
In our current 354 cubic inch Nascar engines we use the late 3.48-inch Tufftrided forging, number 3941184. Preparation is extremely simple and service has been adequate but a stronger crank would certainly be welcome. The 500 mile races really beat hell out of these cranks and we need all the strength in the lower end we can possibly achieve. We use the Tufftrided, finished crank "as is" most of the time. If we want to try something special we begin with a #3997748 semi-finished forging.

As to the long 3.75-inch stroke cast cranks we have no real opinion. Undoubtedly, it would make a good street crank. If it were cut down to fit a 4-inch case the engine would displace about 377 inches. With a special hydraulic cam and a little Holley carburetor such an engine would provide excellent low-speed torque and adequate life expectancy for even the most diehard street racer. The oil pan rail would have to be ground away some to get clearance with the long stroke but at least the gasket and cooling problems with the 4.125 case would not be a constant hassle.

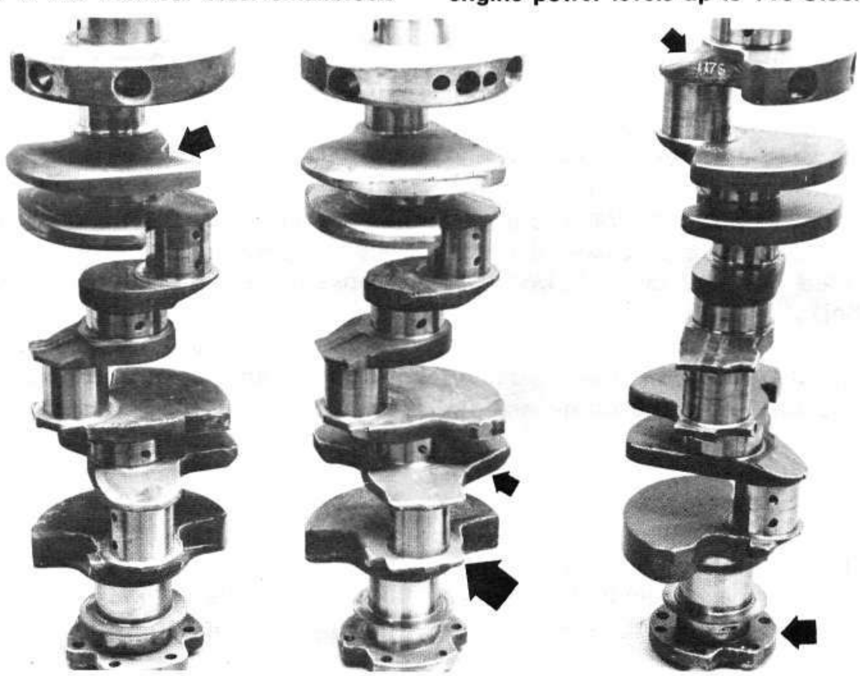
It is possible to grind a welded 3.48-inch forged crank to the 3.75-inch stroke without much trouble. Since several of the match racers have been running super big engines lately, we have been forced to stroker cranks to keep up with them. Right now we are putting together a 383-inch engine using a welded 3.75-inch forged stroker and a 4.020-inch bore case. If our work with the 3.48-inch stroke drag engines is an indication, this combo should produce some real suds. The only real trouble is getting the rod shoulders to clear the camshaft and the sides of the oil pan rail, but with some heavy work with a hand grinder everything will clear. And, if the "run whatcha brung" crowd wants to keep putting more inches between the fenders we figure the little engine can eventually be stretched a bit further by going to the 4.125-inch bore and there's even room for more stroke. Could be interesting!

PREPARING CRANKS

Designing a crank is a very difficult engineering problem. They can easily be made big and heavy if strength is the only consideration but this would make them much more expensive. Like most engine



HIGH PERFORMANCE STEEL CRANKS — These are the most common short stroke smallblock steel cranks. This comparison is intended to show the relative strength and bulk of the small-journal 3.25-inch crank (arrows 2) compared to the big-journal 3.25-inch crank (arrow 1) and the big-journal 3-inch crank (arrow 3). The small-journal 3.25-inch crank is the heaviest steel smallblock crank made by Chevrolet. A careful examination shows that the counterweights/rod throw arms are sturdier than the other examples. We have shown two of the small-journal 3.25-inch cranks to demonstrate the difference in the front rod throw arm. Some of them are drilled and some of them are not drilled. Either is suitable for any engine power levels up to Pro Stock applications.



HIGH PERFORMANCE STEEL CRANKS — Backside view of the big 3-inch and 3.25-inch cranks compared to the small-journal 3.25-inch crank again illustrates the relative strength of the small-journal crank. Note in particular that the rear portion of the small crank is very much heavier than either of the other examples. The rear counterweight/arm of the front throw is not relieved as is the big 3.25-inch crank. The 3-inch crank on the right has been turned slightly to show the forging number on the front arm, another way of identifying this crank.

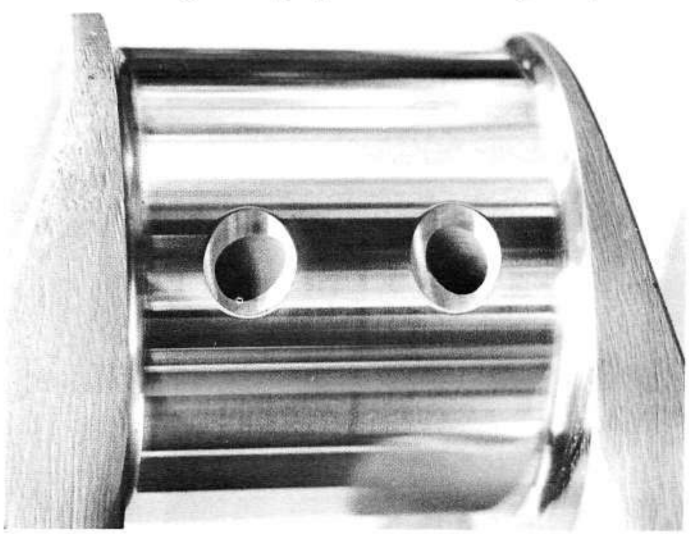
components they represent a compromise. All of the Chevy steel cranks are forged flat and twisted, after the metal has been removed from the forge die, to gain the right-angle relationship between the rod throws. The design also becomes progressively heavier from front to rear. The rear journal arm is reinforced heavily as all the power from the engine is transferred through this portion of the crank. Because of this, the rear of the crank is the most critical and must be inspected carefully. Crankshaft failure is not very common but what problems we have experienced over the years have occurred on the front or rear arm of the number 4 throw. We consider magnaflux inspection the mandatory first step in any crank buildup and these areas must be closely examined and found absolutely free of deep-seated imperfections.

Most of the early crank forgings were made from a 1046 steel alloy and were carburize heat-treated after the initial formation and cooling. This improved the strength considerably. The late cranks, including all of the 3.48-inch cranks, are made from a high carbon 1053 alloy which is air-cooled after forging and is not heat-treated thereafter. We feel that some extra work can make these cranks better for all-out racing. Any piece of iron alloy (steel) can be made a little stronger by heat-treating. The metal is raised to about 1475-1500° F and carefully brought back to normal temperature (normalized). This decreases the molecular grain size, increasing ductility and fatigue resistance. It also makes the metal somewhat softer and reduces the yield point slightly, but the overall strength gain is worth the effort as the crank will not crack or break as easily.

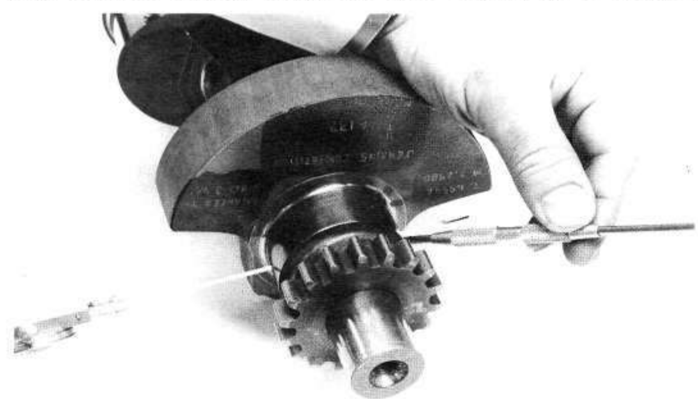
At this point we might mention that some of our cranks are prepared by Hank Bechtloff at Hank the Crank's in North Hollywood, California. Otherwise, when we receive them another 8-10 hours is spent on finish detailing. The preparation is fairly simple and any competent crank shop could duplicate what we do with very little trouble. Here we are forced to speak with some degree of generality because the details of the metallurgical work is proprietary information belonging to the individual crankshaft preparation shops. The specifics related here are an example of work completed at Hank the Crank's on one of our 3.25-inch cranks. After the core is magnafluxed and normalize heat-treated to Hank's specs, it is stress-relieved. This is an extremely expensive bit of work because of warpage. The main intent is to refine the internal grain structure to make the forging stronger.

The cranks that pass inspection must be able to be straightened so runout at the center journal does not exceed 0.0025-inch. Hopefully, the throws will not have to be reground to correct index and stroke length as this is an expensive, unnecessary operation unless the crank is severely warped or bent. We say, hopefully, because normally we don't want to reduce the diameter of the crank pins. If, however, we have a large diameter crank and are able to grind to the smaller rod bearing size this is not a problem. It is also possible to use undersize bearings with reground pins on the small crank, but we don't like to go this route. Prior to finish grinding, the factory-drilled balance holes in the outer circumference of the counterweights are welded up and the outside diameter of the weights is ground about 0.100-inch smaller to compensate. This is all aimed at reducing the windage produced in the crank housing by the spinning crank counterweights.

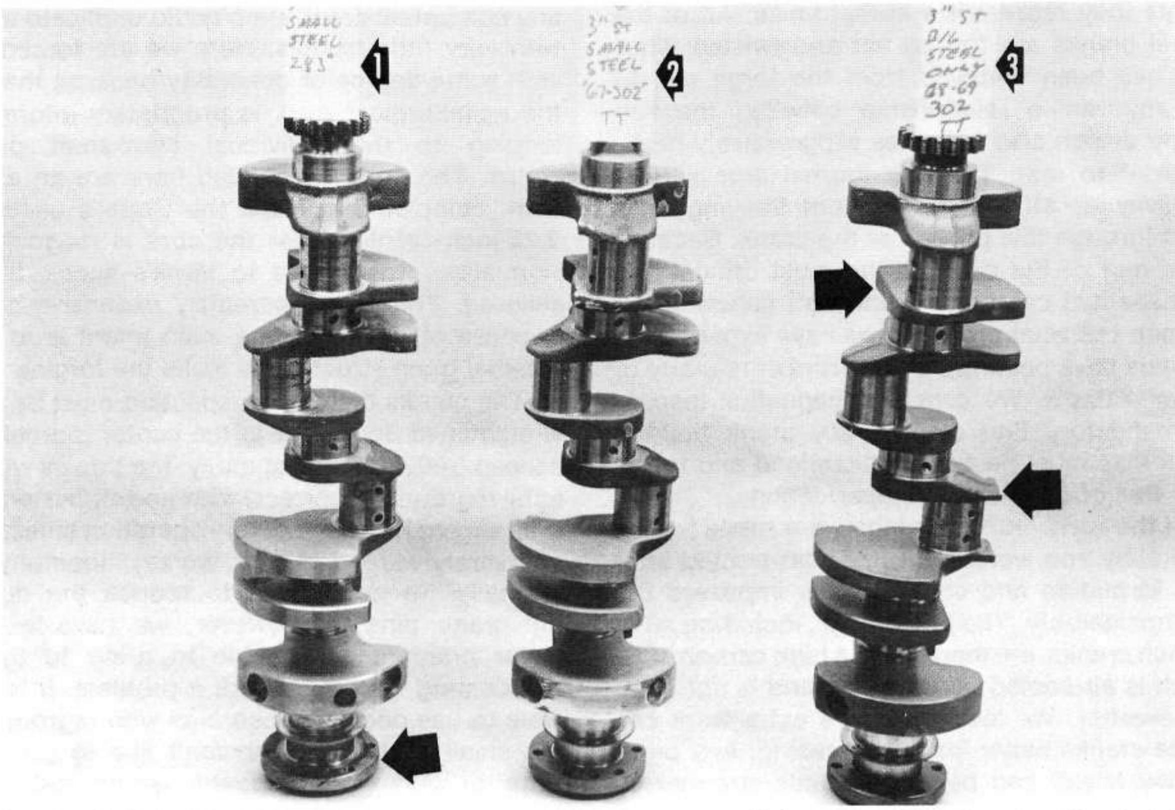
At this point the main journals are cross-drilled for 360 degree oiling. Recently some people have been questioning the necessity of this practice and we can't really say it is absolutely necessary, but we have been drilling cranks for many years and haven't experienced any problems. It is well-documented that the oil flow will increase with a cross-drilled crank running in half-grooved bearings versus a non-drilled crank in fully-grooved bearings. We never recommend grooving the crank or the installation of fully-grooved bearings as we feel this unnecessarily reduces the important bearing surface which supports the crank. In a normal



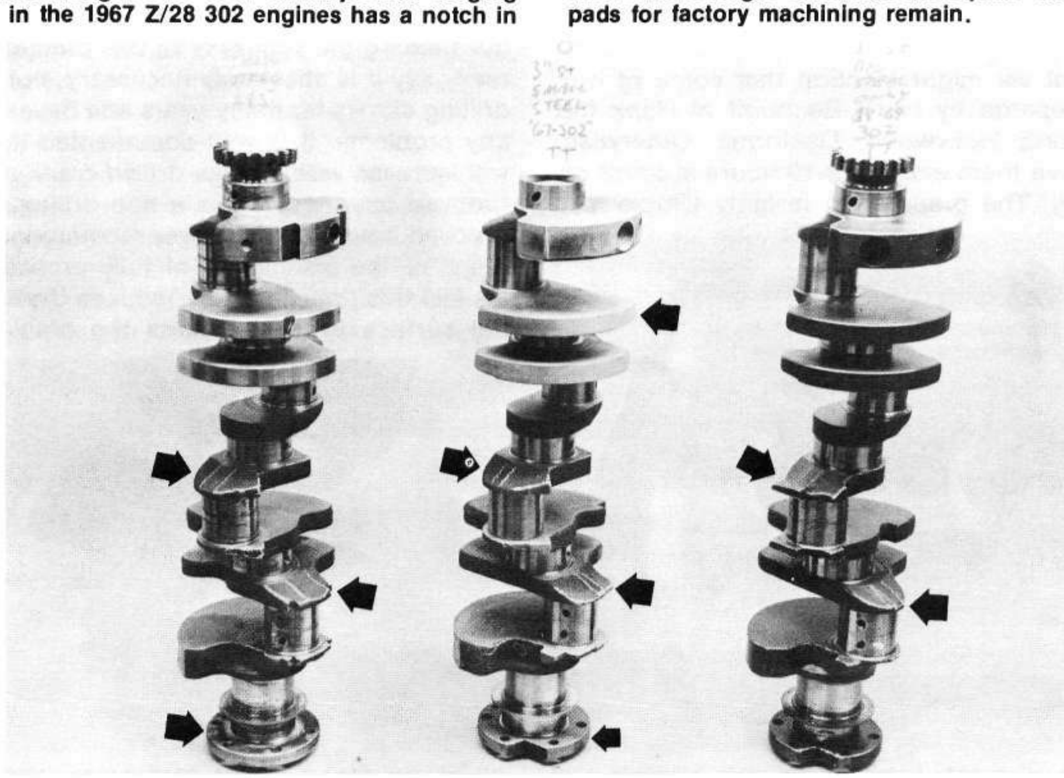
We chamfer the oil supply holes on the main journals and rod journals very slightly to provide increased oil distribution to the bearings. This must not be overdone. Opening the supply and increasing the side clearance unnecessarily just allows more oil volume to escape onto the cylinder walls.



All of our racing cranks have cross-drilled main bearing journals. We prefer this to fully-grooved bearings. Some people are beginning to question the necessity of cross-drilling, but at this point we still drill the cranks because it has not caused any trouble in the past and we feel it still provides a better oil distribution to the bearings.



3-INCH STEEL CRANKS — The family of cranks that is most difficult to tell apart is the 3-inch steel forgings. These photos indicate the notable visual differences. The old 283, small-journal, 3-inch crank (arrow 1) is one of the lightest cranks. It is the only one with a completely round flywheel flange. The later small-journal forging used in the 1967 Z/28 302 engines has a notch in the flywheel flange and is otherwise identical to the early crank except for some minor variation in the shape of the counterweights. The big-journal 3-inch steel crank from the '68-'69 302's also has a notch in the flywheel flange but note the counterweight on the rear arm of the front throw has straight sides and the square chucking pads for factory machining remain.



3-INCH STEEL CRANKS — This view of the 3-inch cranks shows the distinguishing notch ground in the flywheel flange of the late steel cranks. The forward sections of these cranks are virtually identical. Balancing notches and drilled holes will vary according to the individual forging variations. Major differences are found in the rear section. The forward arm of the number three throws on the small-journal cranks are rounded while this arm on the big-journal crank is square.

Crank straightness can be checked by dropping the crank into the block with only the front and rear bearings installed in the saddles. A dial indicator is then used to check runout of the center journal. If this runout does not exceed 0.0025-inch the crank is suitable for any racing or performance engine.

rpm application such as a street engine there is no reason to fool with the crank oiling at all. Do not cross-drill and do use stock half-grooved bearings. We even feel that cross-drilling may not be necessary in Grand National engines. The oil orifices on the journals are also chamfered slightly as is the current practice.

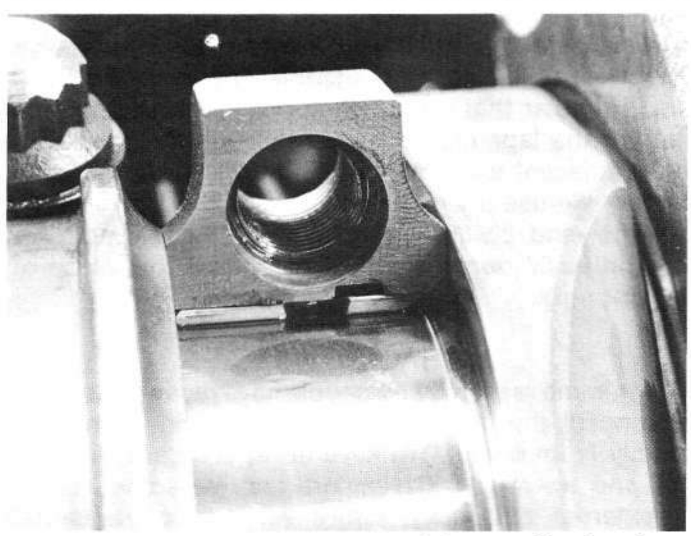
On cranks that are prepared by Hank, the processing includes a complicated double T-90 Tufftride surface heat-treat. The details of the double-treating process are part of Hank's bag of tricks. The last step prior to delivery is balancing. In finished form one of these cranks represents an investment of over $600. Fortunately, we very seldom have trouble with them.

There are some very definite reasons why we go through each of these operations. Most people are aware that we run the small journal cranks in our drag engines. This is simply because it is a stronger crank, and it also decreases the surface speed of the bearings and reduces the negative torque effect of the larger diameter bearings. We have never been able to read the difference on our dynamometer but then we have never really had controlled conditions for a good back-to-back test. As a matter of conjecture we do feel the small journals are worth perhaps 1-2% or something like 6-12 horsepower, even if we haven't measured it precisely. We have never found any indication that the crank is bending in our 330 engines as the bearings show little wear or eccentric loading. The Nascar picture is entirely different. We never run small bearings in these engines and we are looking for ways to reduce the torsional flexing evident in all of the long distance 354's. This usually shows up as distress in the harmonic balancer and timing chain. Either the rubber damping material begins to fall out of the thing or the outer rim begins to walk around the hub.

As another sidelight, we have also run some 3.48-inch large journal cranks in Pro Stock-type drag engines with considerable success. We haven't used them competitively nor have we spent much time trying to develop this combination. We use these big motors as our "iron ball" tire test and match race engines. They run slightly better than the 330's, with more durability.

We check crank straightness by hanging it in the end bearings and observing the runout of the center main as measured by a dial indicator. As long as the runout does not exceed 0.0025-inch, the crank is usable in any racing or high performance engine. Subsequent straightening should not be necessary. We have seldom found a crank that will not meet these standards but occasionally it will happen. As a last precaution we make a double check during final assembly, when all the preparation work and balancing has been completed. After this check, the caps are torqued in place and the crank should turn easily and smoothly throughout the entire 360° of rotation. In fact, with pistons installed, rings and all, and the whole assembly torqued to running specs, we can still turn the cranks on our drag engines by bare hand! Not everyone will believe this or try to accomplish the same thing, but it seems to indicate that things are going together pretty smoothly.

All of our racing cranks are factory Tufftrided or are Tufftride treated during preparation. We have always had good luck with this surface treatment. We feel it adds a high superficial surface hardness to the bearing surfaces without incurring the problems of chrome plating. Tufftriding provides an excellent surface for the bearings and will not scratch when dirt or metal particles find their way to the bearing interfaces. It



If the crank journals are reground for smaller bearings, the grinder may leave a fillet radius between the journal and throw cheek. The rod bearing must be installed with the relieved edge facing the fillet, and in some instances this fillet may be so large the bearing insert must be relieved further to prevent edge interference.

does not crack or peel as does poorly-applied chrome plating. Sometimes the cracks which form in the chrome can carry on into the crank and lead to eventual destruction. The Tufftriding is not any better or easier on the bearings than chrome, but we feel it is less dangerous and more reliable.

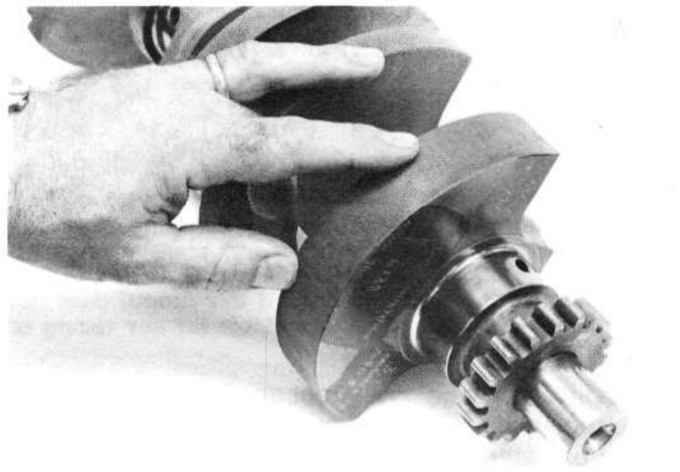
Chevrolet has been using Tufftride chemical heat-treating on their high performance crankshafts since about 1962 with much success. It is very peculiar stuff and all cranks which have been treated should be handled with care. Since it is only a surface hardening (it is not a plating process), it is very irregular. In some areas it may be as shallow as 0.0005-inch deep and in others as deep as 0.200-inch. Once a crank has been Tufftrided it isn't really suitable for resizing. This is the reason we sometimes prefer to begin our special crank work with a non-treated forging. A Tufftrided crank must also be polished with extreme care. We never use a new piece of abrasive tape when polishing such a crank. The surface quality is so irregular that it may be super-tough in one area and yet the tape may cut right through the treatment in an adjacent area. When using tape it must always be flat—we use a leather belt behind the tape to help support—and always go very easily. At the most we may hit each journal very lightly with an old piece of 600 grit tape.

It is possible to grind the forging line from the crank arms and remove the front arm rib without hurting the strength of the crank. This is wasted effort in a street or mildly reworked racing engine. We spend time smoothing a crank in order to reduce windage incurred by the spinning arms and counterweights.

BALANCING

For some reason many people have recently become convinced that internally balancing the crank assembly is important. We have never seen any proof of this and we even believe there may be some benefit in external balancing. We never add extra center counterweights to our racing cranks as this only increases the windage problem. We fill and smooth the counterweights in an effort to reduce wind turbulence inside the lower case. To help some with balance we have ground the rib off the front arm of the number 1 throw without affecting durability. We have also drilled the front throw on occasion but the hole must not be wider than the edges of the front rib. We do not remove the rib *and* drill the pin as this will usually lead to failure. At the rear of the cranks we always install the alignment dowel on the flywheel flange to help balance and we grind an additional notch in the flange, directly opposite the rear arm of the number 4 throw. This extra metal can be removed without affecting strength and as we said internal balance is not absolutely essential. On the 354 long distance engines we have even given some thought to an additional weight on the front of the crank, a possible assistance to damping torsional flex.

We know that the balance in a crankshaft assembly is created at the ends. Any intermediate loads imposed by rotating imbalance are transferred across the fulcrums of the main bearings to get to the ends. Such imbalance will therefore show up as eccentric loading on the bearings and we have never seen evidence that this is a problem in our high-rpm engines. Of course, reciprocating weight reduction is extremely important as it will also transfer across the bearings. It is obviously important to balance the reciprocating weight and, incidentally, it is advantageous to remove every bit of excess weight from the rod/piston. This helps reduce crankshaft stretch and flex which may show up as torsional twist and subsequent bearing loads.



When very high engine speeds are combined with small-volume dry-sump pans, some windage reduction may be gained by filling the balance holes drilled in the crank counterweights and smooth grinding the outer circumference. We may also reduce the radius of the weight for additional wind reduction.

There is nothing to absolute weight of the crank itself since this is just extra flywheel mass but there is some advantage to reducing the size of the counterweights and radiusing the sharp edges. This is largely a matter of reducing oil windage in the lower case. When a very shallow pan is used and the engine is set very low in the chassis, this may be the only way to quiet things down inside a very small oil pan. We have run competitive drag engines with cut down weights and without, and, frankly, we can see little difference. At this point it's one of those things you can do because it makes you feel better even though you aren't sure it's helping you. If you're on a tight budget, forget it!

When the crank is balanced we usually specify that it be "over-balanced" slightly. We don't do this to gain any performance advantage or to help endurance. We have found that over-balancing can help the bearings a little in the big block engines where the reciprocating weight is high. With light pistons and rods in the little block this may not be a significant factor. We do it just to help flexibility of engine assembly. By balancing the crank to a bob weight about 75 grams over the lightest reciprocating weight we ever use, it allows us to use the crank with many piston/rod combinations without requiring rebalance. We feel this more or less makes each

We do not add center counterweights to achieve internal crank balance. This is a waste of time and money in a drag or long distance engine. We do make some effort to assist balance by grinding a notch in the flywheel flange, directly across from the rear throw.

of the cranks "universal" so we can use them in several different engines if necessary.

MAIN BEARINGS

In all of our engines we use Chevrolet Moraine M400 bearings. These bearings are aluminum tri-metal in construction and available in standard undersizes as well as 0.001-inch oversize for added clearance on standard journals. We clean the lead irridium protective coating from the bearing surfaces with Scotchbrite before installing them to the block. This is to prevent the coating from packing up when the engine is started for the first time. The vertical clearances on big journal main bearings are always set at 0.0025- to 0.003-inch and on the little journal cranks at 0.003- to 0.0035-inch. The main journals are measured with a calibrated micrometer to determine exact diameter and the main bearings are measured for final inside diameter after they have been installed in the block and the caps torqued to running specs. A Sunnen dial bore gauge is used for these very important measurements.

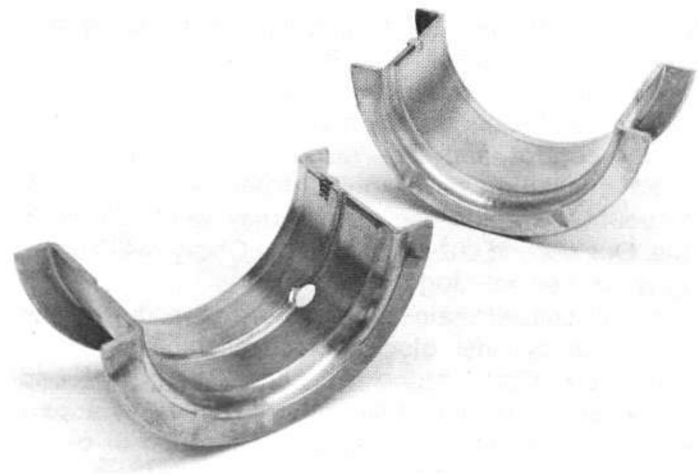
When the Moraine M420 bearings were available for the little engine we used them as they were more durable. Some of the readers may remember the old Moraine M500 truck bearings. These were absolutely the best bearings ever produced, in our opinion, but Chevrolet couldn't economically maintain the very stringent quality control and has discontinued them. They would absolutely never pull, blister, flake or chip. No doubt we would use them if they were still available. Our second choice behind the Chevy M400 bearing is the Federal-Mogul AP brand.

We discussed main bearing bore prep during the section on cylinder blocks. Most machinists will try to gain a perfectly round main bore. We have found that when a standard bearing is installed in a perfectly round saddle and the cap is torqued in place, the inside diameter should be bigger across the parting line than the vertcal measurement. The bearing shell

To assist balancing somewhat we also make certain the flywheel dowel pin is installed in a normal manner. By shaping the weights, grinding the balance notch, and using the stock dowel, it is possible to achieve a pretty good crank balance.

is designed so it will become round with normal loading. When a bearing is installed it normally squeezes in at the parting line but there may remain some variance. This will not cause trouble as long as the vertical clearance is within the limits previously specified.

Since there will usually be some variance between bearings and crank journals it is sometimes necessary to have the crank journals reground to gain the desired oil clearance even with 0.001-inch oversize bearings. This is one of the reasons we sometimes prefer to begin with a non-Tufftrided crank whenever possible. It is very difficult to grind a treated crank undersize because of the surface hardness variation. We don't really recommend polishing to bring the clearance to spec because it is difficult to keep the journal round with this process. Grinding to the low side of the clearance spec is best, then the crank can be Tufftrided (increasing the diameter slightly). If the variation is very great the crank must be ground to accommodate undersize bearings, 0.010-inch undersize being the most popular. It is important that the clearances be in the general neighborhood of the prescribed 0.0025- to 0.003-inch figure but it isn't necessary to



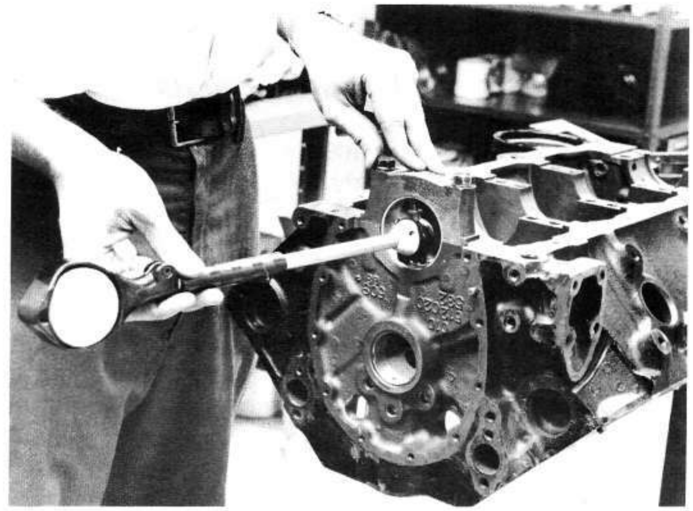
We use and recommend stock Chevrolet Moraine M400 bearing inserts. These bearings are of aluminum tri-metal construction and are available in standard undersizes and 0.001-inch oversize, making clearance adjustment relatively easy.

go overboard with this thing. If any drag or circle engine is within a half thousandth on the small size or a full one thousandth on the big side, it'll work just fine. We used to run our cranks a little looser with no trouble, but we don't really think it's wise to run them very much tighter than these recommendations.

Before the main bearings are laid in place for final assembly the outside edges may have to be chamfered slightly to clear any fillets which have been ground in the crank. Most grinders now use radius stones to give a smooth radius fillet between the journal and the connective structure in an attempt to eliminate any sharp edges where cracks may emanate and to help eliminate a reduction in strength resulting from the removal of metal. The chamfered bearings will prevent interference when the crank moves fore and aft in the bore. Don't reduce the bearing surface area any more than the minimum required and make sure the edge is polished smooth with Scotchbrite after it has been cut slightly with a bearing scraper.

The last thing we do is to scrape the edges along the parting line of the rear main bearing. This opens a

By utilizing the Sunnen dial bore gauge it is easy to gain exact bearing fit. When the running space is set as specified, the smallblock will easily turn 10,000 rpm with no difficulty, provided reciprocating weight is kept low.

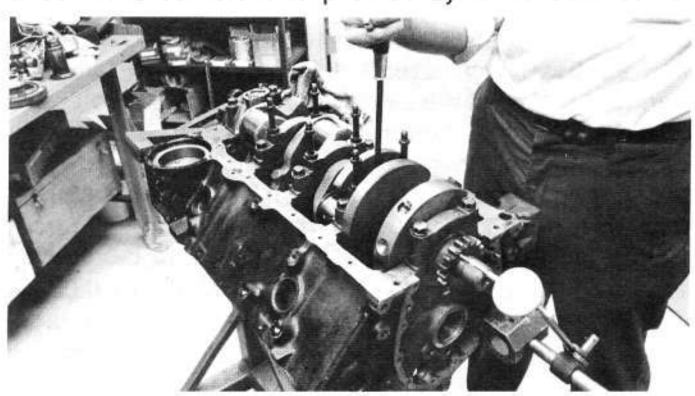


The vertical clearance on big-journal cranks is set at 0.0025- to 0.003-inch and on small-journal cranks at 0.003- to 0.0035-inch. There's no secret or trick to this. We measure the crank journals with a hand micrometer and the torqued inserts with a bore gauge.

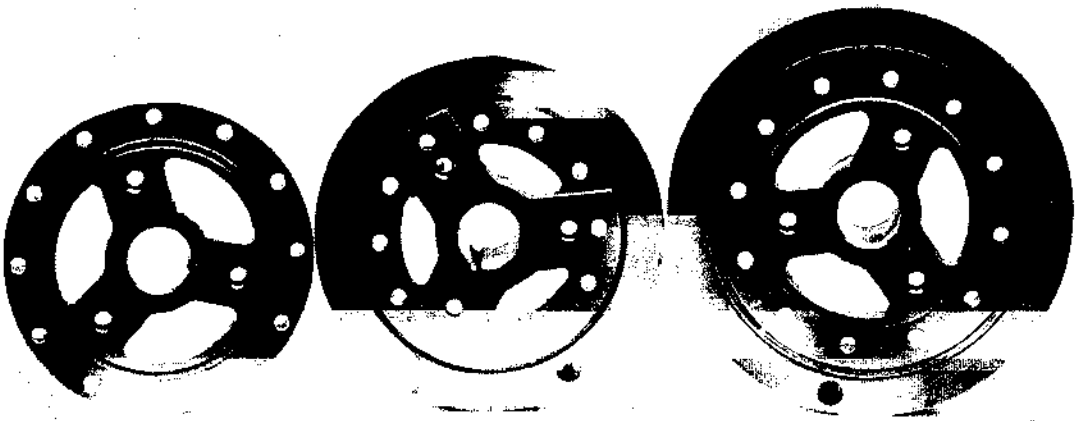
slight seam along the parting line to provide positive oil pressure feed from the rear main upper shell groove to the thrust flange interface between the crank and thrust surface of the rear main insert. If the rear cap is exactly perpendicular to the block and some oil supply is provided in this manner there should be no trouble with thrust face failure, even with extremely heavy clutch pressure.

HARMONIC BALANCER

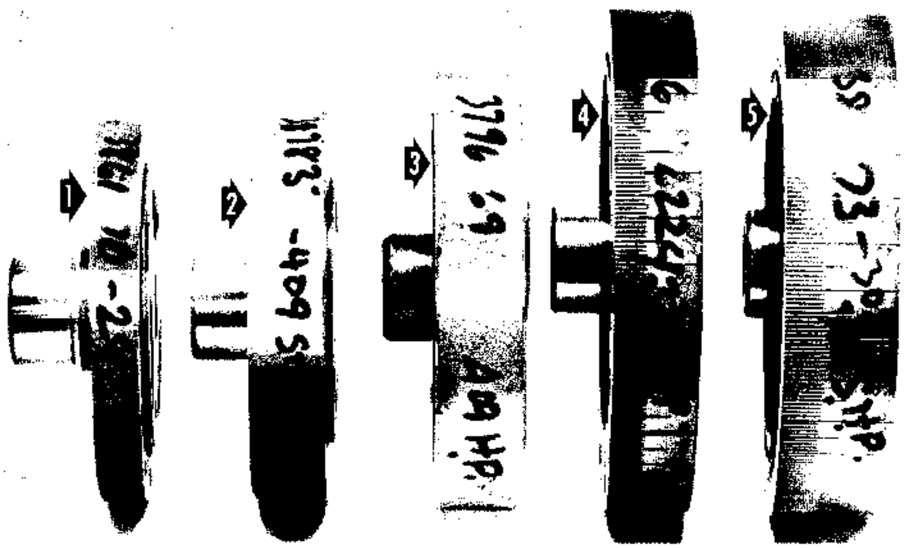
Every racing engine should be operated with a suitable harmonic balancer, sometimes called a torsional dampener, attached stoutly to the front of the crankshaft. There are several different sizes available from the local Chevy dealer. The accompanying photo shows all of the various sizes. We select the appropriate balancer according to the engine requirements. In the 330-inch drag engines we have found that the old 409 balancer, number 3796769, works well. It is smaller in diameter than the late hi po balancers but is not as small as the standard smallblock balancer. It is sufficient to provide dynamic balance for



When the crank is installed the end play must be measured to insure adequate thrust clearance. We recommend a minimum end play of 0.005- to 0.007-inch. The crank should be seated against the thrust face with a few light blows from a brass hammer before this clearance is measured.



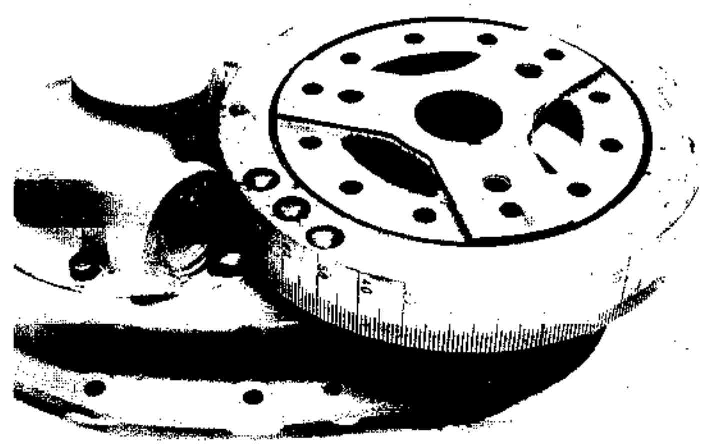
Several different balancers are available from Chevrolet. They are all interchangeable and provide the same front spacing for the pulleys. Iron cranks do not require much balancer weight. At most the hi po 350 balancer, 6272224, may be used. However, in any long distance endurance the largest, heaviest balancer available should be used to stabilize the front of the crank and timing chain. The 302-327 hi po balancer number 3817173 is the largest made.



For drag racing with small-journal cranks we use either of the 409 balancers—the special hi po model, part 3796769 (arrow 3), or the standard model, part 3878371 (arrow 2). On the track 354's with big-journals we use the late 350 hi po balancer (arrow 4). Experimental long-stroke drag engines with small crank diameters work best with the 3817173 special hi po 327 balancer (arrow 5) to dampen the flexing tendency. For drag racing a 3-inch crank with big journals, the 283 balancer number 3861970 (arrow 1), works adequately.

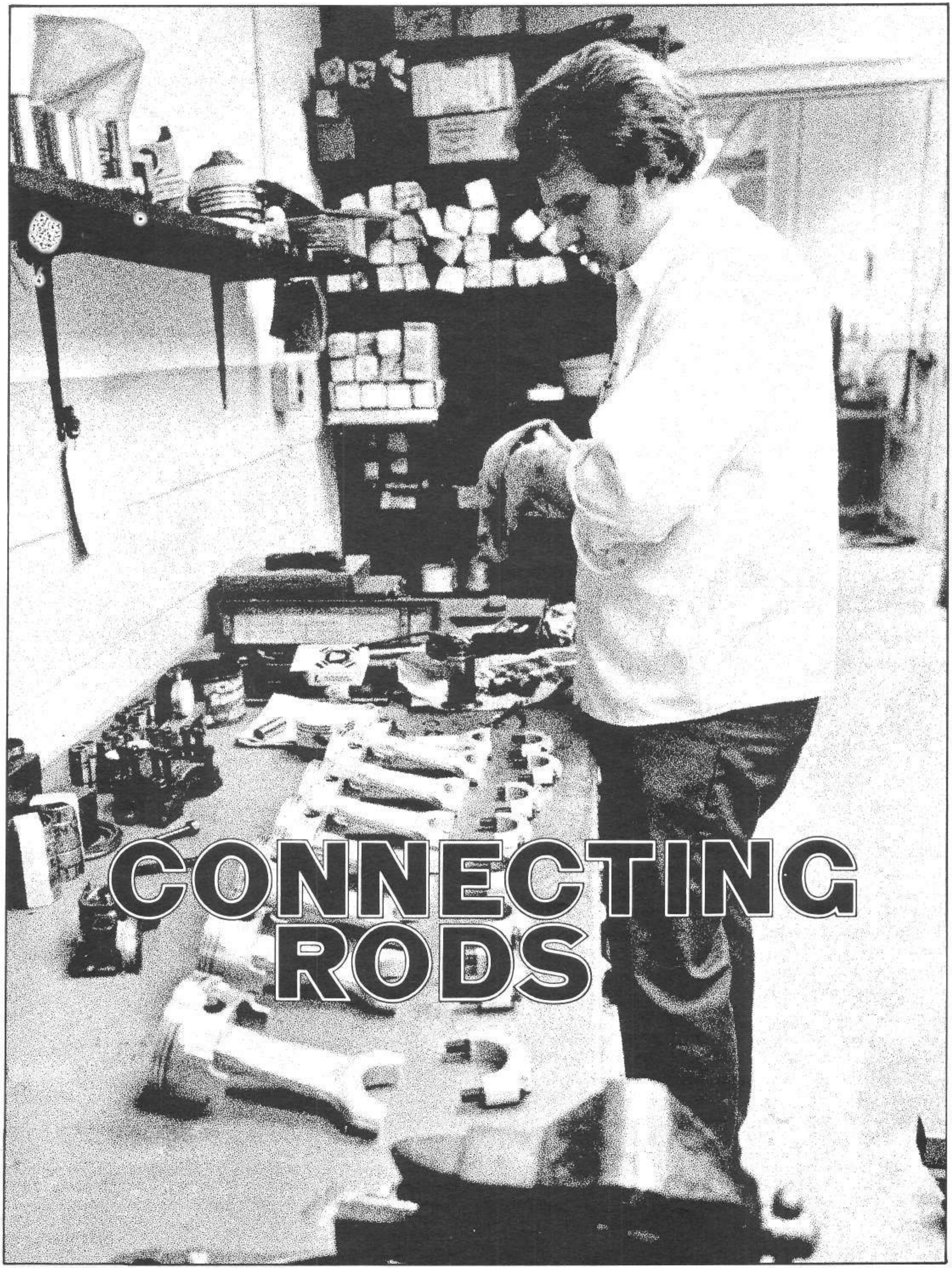
a short duration-type drag engine. In the long distance engines it is necessary to use the largest balancer available for the best possible dynamic dampening effect. It may not be commonly known, but the balancer from the old 327 Special High Performance engines is the heaviest and thickest balancer. It is available as part 3817173 and is better for the endurance engines than the late 8-inch balancer. The late hi po damper, number 6272224, is good for street and normal high performance applications or the nodular iron damper available from the off-road parts catalog as part 364709 may be used if desired.

We have the balancer completely degreed to provide for timing the engine. The balancer and the flywheel are always sent along with the reciprocating parts when final balancing is accomplished. On long distance engines we test the balancers to insure the outer rim is above 200 on the Brinell scale, the harder the better. We also mark the balancer rim and hub to use as a reference for checking if the rim is beginning to walk around the hub. This is common on GN engines due to vibration.



It is not absolutely necessary to have the damper degreed but it makes timing the engine and setting the valves much easier. We mark the balancer in a full 360° circle but it is possible to mark the TDC number 1 position through approximately 45° for total timing and only the other 90° locations for valve adjustment.

CONNECTING RODS



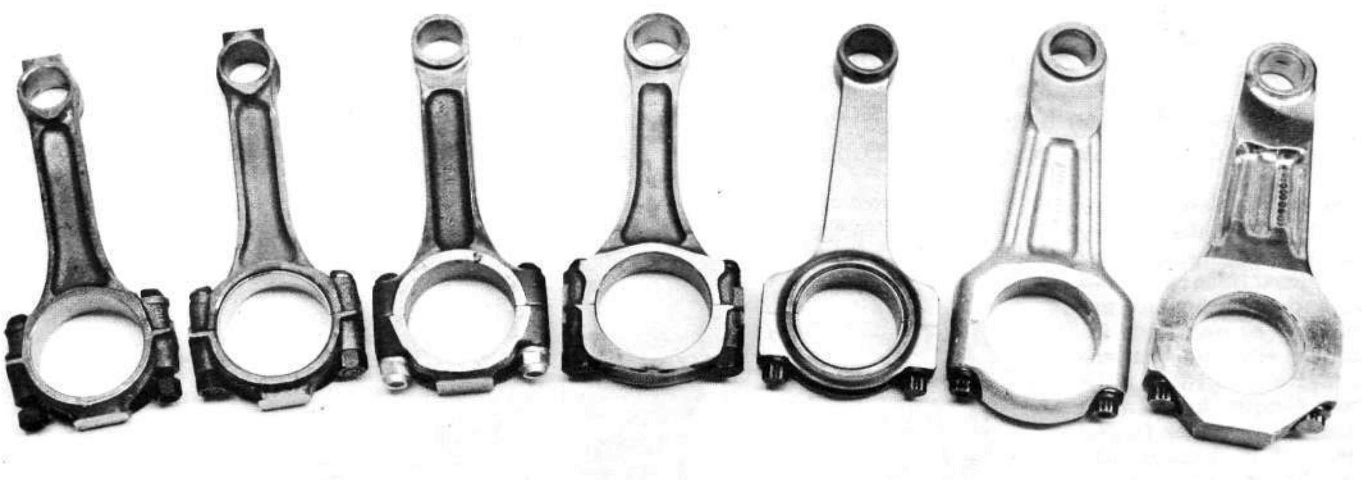
ROD RATIO

The length of the connecting rod will have a definite effect on the performance of the engine. Generally, this factor is expressed with relationship to the stroke length of the crank. In mechanical engineering texts this factor is referred to as the *l/r* ratio, where *l* is the length of the connecting rod center-to-center and *r* is the radius of the crank stroke. To make the figures somewhat less cumbersome and shorten the parlance required to discuss this theoretical figure we simply call it the *rod ratio* and we compute the numerical value by dividing the length of the connecting rod by the total length of the crank stroke (equal to twice the radius). As an example, the stock rod ratio for the 265, 283, and 302 engines is determined by dividing the stock rod length, 5.700 inches, by the length of the stroke common to these engines, 3 inches. The result is 1.9:1 which may be expressed in our specialized vernacular as a rod ratio of 1.9. Most current production V8 engines have rod ratios in the area of 1.6 to 2.0.

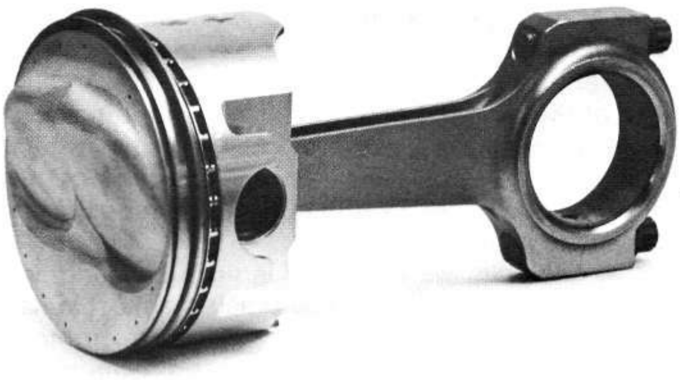
At the beginning of the chapter on crankshafts we briefly touched on the subject of stroke length and its relative importance to the connecting rod, camshaft and other engine parameters. For many years we have known there was a very important tie between these variables but for the particular requirements of all-out racing we have had to spend a great deal of research time chasing down the specifics. Back as far as 1969 we discovered the effects of rod length in the 427-430 Mark IV big blocks. We were searching for ways to get better breathing and more power. The Mark engines have good intake port breathing characteristics and they have a relatively small stock rod ratio—the rod was 6.135-inches long with a stroke of 3.76 inches, resulting in a ratio of 1.63. As time went on we became aware that this was an exceptionally small ratio for an engine displacement of this size. On looking back we now believe that when the engine is run in stock form this low ratio is actually an advantage. It offsets or, more accurately, *balances* the very good intake breathing and large port volume. By the time we discovered the possible importance of the rod ratio to intake breathing we were almost at the end of the program and were moving on to the smallblock development. However, as we began with the little engine we studied the rod ratio effect more closely and by late '74 we had a pretty good grasp on the whole picture.

At this point we feel there is no such thing as an *ideal rod ratio.* There is, to be precise, a most suitable rod ratio for a particular engine displacement, intake runner volume camshaft profile and exhaust port flow. This is true for both single four-barrel engines and dual-quad tunnel ram inductions. All of the factors we mentioned above must be balanced together for maximum breathing efficiency. There may also be other important factors but at this point these are the ones we have definitely isolated. And at that we are only fairly certain of the intake side because we haven't had time to thoroughly test exhaust tuning with respect to the rod ratio, but it seems logical that if there is a response on the intake side of the engine there should also be a response on the exhaust side. A direct comparison between two extremes like a 5.70-inch rod 350 and a 6.25-inch rod 350 would be very interesting. We haven't been able to make such a test but given the time, much could be learned. Of course, the results would require very careful interpretation.

For a minute let's ignore other physical/mechanical limitations and speak in theoretical or "philosophical"



A wide variety of steel and aluminum rods are available for the smallblock engine. From left to right we have two factory high performance rods (available with full-floating or pressed pins), a specialty steel rod produced formerly by the Mr. Rod Company, the heavy duty, semi-finished, Chevy rod based on the big block forging (shown here in finished configuration), the well-known Carrillo forged steel racing rod, the forged aluminum Superod and the forged aluminum BRC rod. The latter five rods can be ordered in custom lengths for non-conventional assemblies.



These are two of the common rod-piston assemblies we employ in Jenkins Competition engines. On the left is a typical assembly for a Nascar 354-inch competition engine. This is a TRW piston on a Carrillo steel rod. The rod length may vary from 5.70 inches to 6.00 inches. On the right is a typical drag racing assembly. To gain greater rod ratio flexibility a special BRC piston is used along with an aluminum rod that may vary from 5.85 to 6.250 inches.

terms. We have isolated three distinct induction tuning factors, the rod ratio, the intake volume and the camshaft lobe displacement angle (the angle between the centerlines of the intake and exhaust lobes). By making unit changes in each of these factors (not changing the others) it is possible to alter the power characteristics. This is not what you would call an exact relationship but there is a *tendency towards these results.* The smaller the rod ratio, the smaller the intake volume, and the smaller the displacement angle—the more low rpm power the engine will develop. The larger the ratio, the larger the intake volume, and the larger the displacement angle—the more high rpm power the engine will make.

The three tuning factors can be juggled somewhat provided the engine design is flexible enough. In reality, other factors will also be involved. The rod can only be stretched so far before the pin bore reaches the very top of the piston or shortened to the point that the piston becomes very heavy. Intake port volume can be enlarged or reduced only so much before other engine parts get in the way. And cam design is limited by certain material and mechanical restrictions. Each of these subjects will be taken up in greater detail at a later time but at this point we are only concerned with the theoretical effects possible from juggling the design parameters.

From what we know now, the intake manifold runner and port diameter or cross section and the overall runner/port length is extremely important. We call this total intake configuration/shaping the "intake volume," again just for simplicity purposes. We know that for a given displacement the intake volume must be balanced against the rod ratio. We have found that the longer the rod (the greater the rod ratio) the less intake volume the engine will require. Our tests have proven this but it is always relative to some known rod length and intake volume.

Another important consideration is the effect the rod length will have upon piston movement. As the rod is lengthened the piston will "reside" longer in the vicinity of top dead center. In other words the piston

Stock length steel rods are available from the factory and are satisfactory for high performance street use and some limited racing use. For maximum induction efficiency in endurance engines with stroke lengths longer than 3 inches, a longer forged steel rod is needed. Unlimited drag racing inductions normally respond to longer-than-stock aluminum connecting rods.

will remain up inside the combustion chamber for a slightly longer period of time while the rod/crank is moving through the uppermost portion of their travel. The rod length will also affect piston acceleration. As the rod is made shorter it will cause the piston to accelerate more quickly away from the TDC and more slowly from BDC. These things will affect the breathing but they also are important when compression ratio and cam design are considered. If the rod is shorter, relatively, the cam displacement angle can be stretched out and the engine compression ratio can be increased. Since the piston dome will stay in the chamber for a shorter period with the shorter rod the cam action is less restricted and the valve notches do not have to be cut as deeply in the piston for valve clearance. The dome will not reside in the chamber as long so the breathing at and near TDC will not be blocked off or restricted as greatly. It is because of these side effects (and others) that it is not a good idea to stretch the rod length any longer than is absolutely required by the induction efficiency.

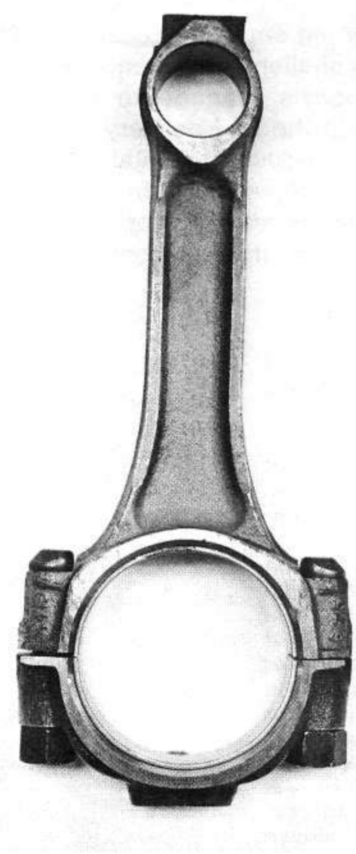
Of the three factors presented earlier, the camshaft is the least important when we consider what the overall induction effect will be. Getting the right cam into the engine is important, but at this point we pretty much know exactly what the different engines require and any reasonable change produces a very, very minor response. Generally, cam timing can help an induction which is inadequate for a specific displacement or rod ratio but it will do this only to a minor degree. If a combination is such that the intake volume is large and the rod ratio is high—a bad combination—the situation can be helped somewhat by decreasing the displacement angle of the lobe centerlines by about 2-3°. As a "fix" this is definitely third in line behind altering the rod ratio and the runner size. Balancing the ratio and volume will gain power much more effectively within the desired engine speed range than fiddling with the cam.

With respect to the smallblock Chevy we have found the following generalities. Given the 9.025-inch deck height of the block we feel the 5.700-inch rod provides fair breathing with a 3-inch crank stroke. There are some drawbacks from the slightly heavier piston required to gain high compression with this rod and stroke combination. With some work it is still possible to get the piston weight way down. We do know that many of the circle track and road racers use longer rods with the 3-inch stroke. This may be an attempt to compensate for the manifolding and camshaft restrictions imposed by the required engine range or the sanctioning rules. Some builders use rods as long as 6 inches in these engines with apparently a high degree of success.

In a 327-330 engine we like to use a 5.9-inch rod. This is not necessarily a perfect combination but it happens to work well with a 350 piston having a cut down deck. It is, therefore, economical to build such an engine utilizing a stock 350 forging. This is a very nice piston. The ring package is spaced very tightly even with the top ring moved very near the top as we like to see it; the piston is relatively short overall and it is impressively light.

In our "big" motors, the 354 tunnel ram inducted drag stuff, we lengthen the rod considerably, to 6.25 inches. This is essential for even "adequate" breathing with the big displacement. It does require a custom forging and the ring package is very tight. For circle racing the rpm range is restricted some by the intake manifolding and the engine may be better off with a shorter rod. In this case we put in 5.85-inch rods, and the piston shape remains reasonable. The ring package looks good and even though a slightly heavier, more durable piston is required in the long distance engines, they still look pretty good.

The larger rod ratios (around 2:1) may also be an advantage when isolated runner manifolds, 180° headers or 180° "flat" cranks are being used to tune engine torque peak. Here again, we aren't speaking from a great deal of experience but past experiments with I.R. manifolds have never quite lived up to expectations. It appears that the intake impulse signal (sometimes referred to as "reversion") is so severe in the manifold that it affects the fuel and torque curves. The result is "stand-off" or fuel loss out of the



Factory heavy-duty pressed pin rods provide excellent service for a relatively modest cost. These rods have an improved surface between the rod body and cap. They are heat-treated to a higher hardness and are magnaflux-inspected and shot-peened.

The good factory rods are usable in stock condition but may be ungraded somewhat by following the specs given in the *Chevrolet Power* book. For 2.0-inch journals use the pressed-pin rod, number 3927145, from the '67 Z/28. For 2.1-inch journals the best pressed-pin rod is number 3973386. If a large-journal floating-pin rod is desired, use number 3946841.

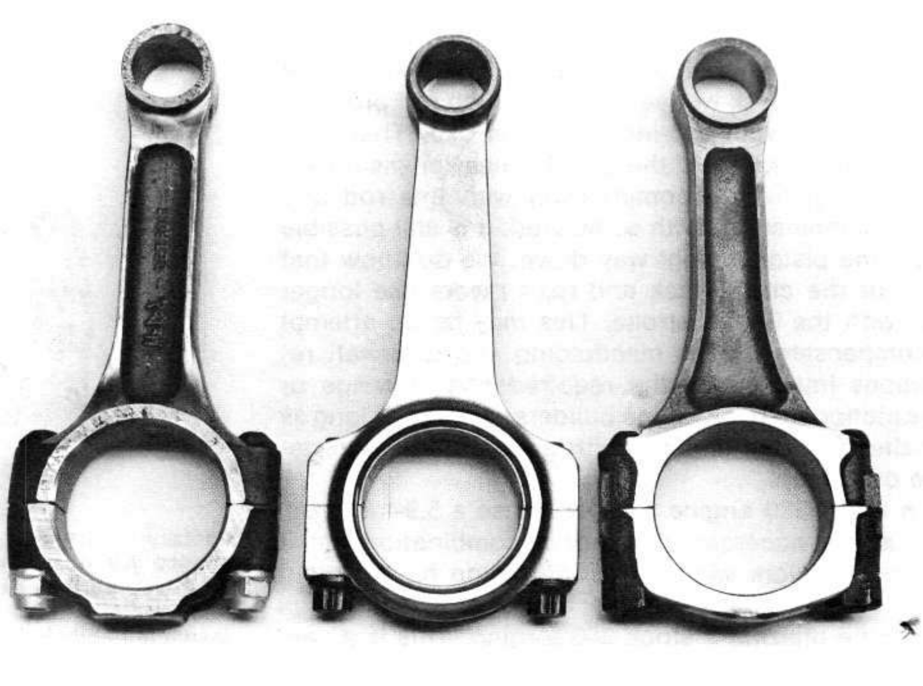
carburetors at certain engine speeds. The 180° header or crank causes a similar phenomenon when strong reverse exhaust signals manage to enter the intake system and stir up the fuel delivery. We think that a longer rod might reduce the intake signal and reduce the fuel loss problem. A testing program of a long rod engine with an extremely short, small volume I.R. manifold and 2.5-inch throttle bore carburetor might prove very interesting!

STOCK RODS

It usually isn't necessary or desirable to spend very much time or money on stock connecting rods. They aren't suitable for most racing uses and in those where they are required some restriction must be placed on the engine speed to prevent failure. No amount of special work will keep a stock-type rod together at extreme engine speeds or in long distance endurance applications. Currently, we don't do a great deal of engine building with stock rods but we have had past experience which should still be suitable for current engines.

As we outlined in the previous section, the stock 5.700-inch length is most suitable for engines in the 3-inch stroke range. The best stock rod available from Chevrolet is the late Z/28-LT-1 forging. It utilizes better metallurgy, has the good Boron material bolts, is shot-peened and is 100% magnafluxed at the factory to increase quality control. The part number is 3946841 and it is of the large journal variety (2.1-inch crank journal diameter). Our past experience has shown it is not a good idea to go to the smaller journal size (2.0-inch diameter) with stock steel connecting rods unless the sanctioning rules require this. An example might be an early 265 or 283 Super Stocker. In this case the '67 Z/28 rod is the best choice, part number 3927145. The later rod is still stronger and should be used where possible. There are some small journal rods which are lighter than the '67 rod but they are not as strong. Running them in a competitive engine would, in our opinion, be pretty risky. The engine speed with early rods should be limited to 6500 rpm (short duration) while the late rods can usually be operated to 8000 rpm (short duration). As a point of interest, back in '70 Gary Kimball was running one of our stock big journal rod 302 engines and developing about 520 horsepower. He was able to run the same rods for over a year and a half with absolutely no

Non-stock forged steel connecting rods to fit the smallblock Chevy are available from several sources. We have used nearly all brands successfully in circle racing engines. The former Mr. Rod design at left provides an economical steel rod that can be ordered in non-stock lengths. The highest priced forged non-stock rod is probably the Carrillo rod from Warren Machine. On the right is the semi-finished Chevy steel rod based on the big block rod. It can be finished to special lengths. Not shown but also available is the special forged steel rod produced by Hank the Crank.



problem. Recent technology could very likely boost the power level from a comparable engine so they might not last as long but the point is, the stock late forged rods are usable when properly prepared.

Because we don't fool with stock rods too much there is very little preparation advice we can give. Perhaps those specialty shops which rework them extensively can offer more helpful direction. For our money we don't massage stockers and wind up with expensive, inadequate pieces. We would rather spend the money buying the correct specially-built connecting rods which normally don't require any reworking. The exception is high performance street engines where stock rods are suitable.

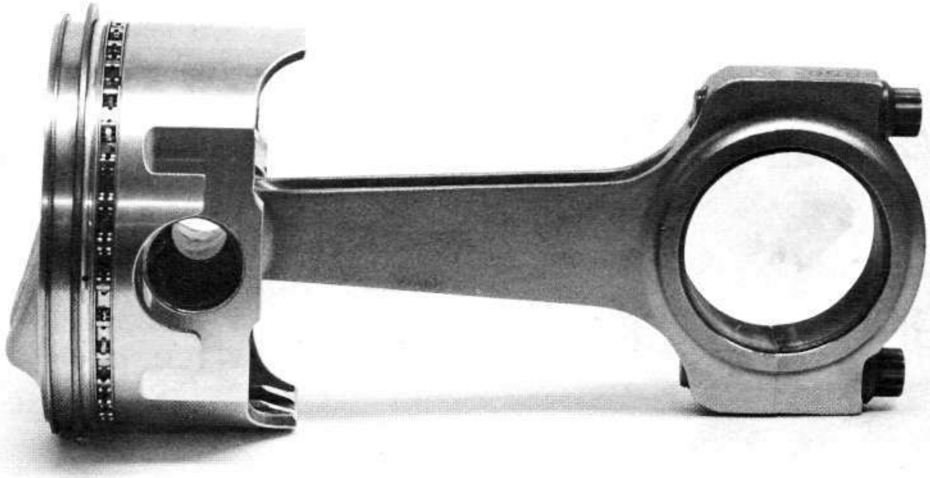
For those who must or insist on using stock rods some excellent guidance for special preparation is given in the Chevrolet high performance manual, *Chevrolet Power,* available from any dealer as part 3965775. Briefly, the Chevy engineers recommend a magnaflux inspection followed by a Rockwell hardness test or a Brinell test. On the Rockwell "C" scale they should check 28-34. The forging flash should be ground from the beams, removing the minimum amount of material possible. Afterwards, the beams, including bolt head and nut seats but not the bolt bores or cap interfaces, must be shot-peened to Almen 0.012-0.015A arc height with number 230 cast steel shot.

The big end weight can be reduced by machining or grinding as shown in the photos. The rod bolts should also be hardness tested to insure they fall within a Rockwell "C" range of 36-40. The rod bolt nuts should check within Rockwell "C" range of 28-34. The Chevy ⅜- or 7/16-inch connecting rod bolts should be center-drilled to a depth of 0.07-inch on each end with a number one drill and the torque checked by measuring bolt stretch to 0.006-inch.

The small end can be fitted with a thinwall bronze bushing to accommodate full-floating pins. Some of the stock Z/28 rods were full-floating without a bronze insert. A specially-developed plating process was applied to the inside of the pin bore. For maximum drag race durability we think these rods should be fitted with bushings. A 3/32-inch diameter oil supply hole should be drilled in the top of the full-floating rod and chamfered slightly to provide oiling. This is not a terribly effective way to oil the pins in the rod but it is adequate when the bushings have been properly installed. On an endurance engine it is a much better idea to oil the pin from below the pin bore, as we will discuss later, but this is more suitable to specialty rods and not necessary in short term drag racing engines. The bushing and top hole work fine in a street engine. The radial oil clearance between the pin and the bushing should be 0.0006- to 0.001-inch. Of course, for street engines a pressed pin is more than adequate since the rod-piston assemblies won't be taken down at frequent intervals as is the case in a racing engine.

The big end of any high-rpm steel connecting rod should be finished out-of-round. To achieve this out-of-round static measurement the rods must be prepped in an unusual manner. We have had some success by using shim stock between the body and the cap while finishing is completed on a rod resizing machine. If a 0.002-inch shim is placed in each parting face the resulting finished hole size will measure 0.002-inch (approximately) larger across the parting line axis than along the vertical axis when the cap is reinstalled without the shim in place. With experience it is possible to achieve any desired degree of out-of-roundness, vertical clearance and crush.

All connecting rods stretch a certain amount during the maximum acceleration points in the stroke. This results in a pulling and pushing action which distorts the big end of the rod. If the hole starts "round" it will be distorted out-of-round during this action, elongating the bearing bore along the axis of

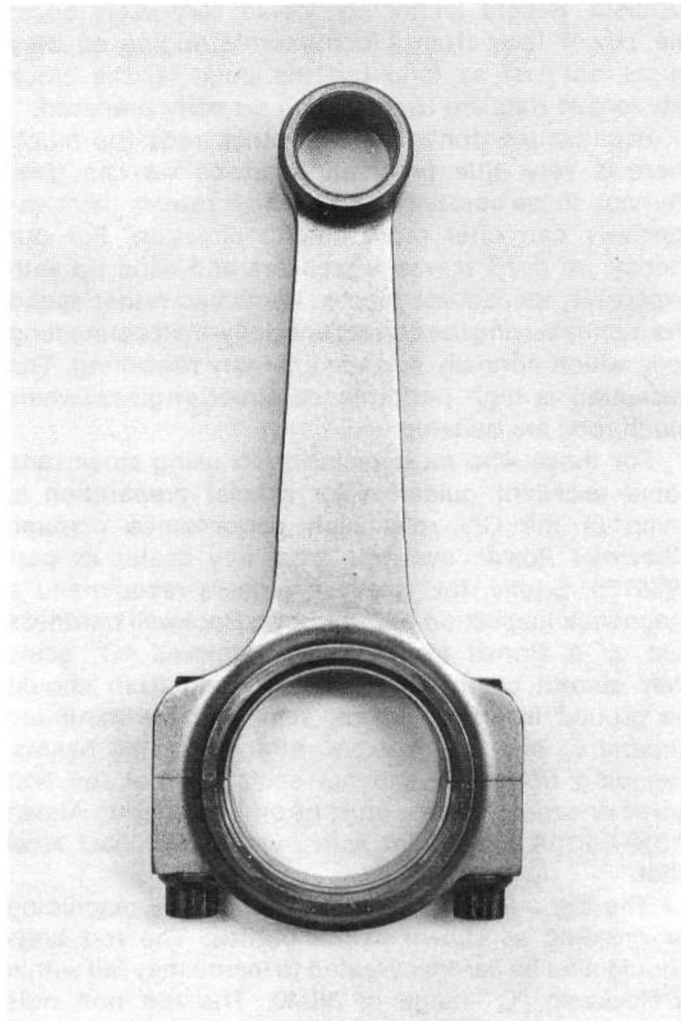


Even with the very highest quality steel connecting rods it is unreasonable to expect reliable performance at continuous in-service engine speeds much higher than 7500 rpm. For short term drag racing the speed limit may be raised to 8500 rpm. The stiffer the rod is, the more the shock loading it transfers to the parting line and the more difficult it is to hold the rod together. To accomplish a reasonable life expectancy, the piston/pin/ring weight must be as light as possible and the best rod bolts obtainable must be used.

the rod beam. When this occurs in an improperly prepped rod the ends of the bearing inserts will pull away from the bore and be squeezed inward. They then chisel the oil off the crank journal. The resultant lack of lubrication will cause a temperature rise, the surface of the bearing will begin to melt and the friction between the surfaces will increase. Eventually the friction will cause the bearing insert to spin in the bore and the game is all over. In our opinion this is the chain of events that most often occurs when an engine fails a rod bearing.

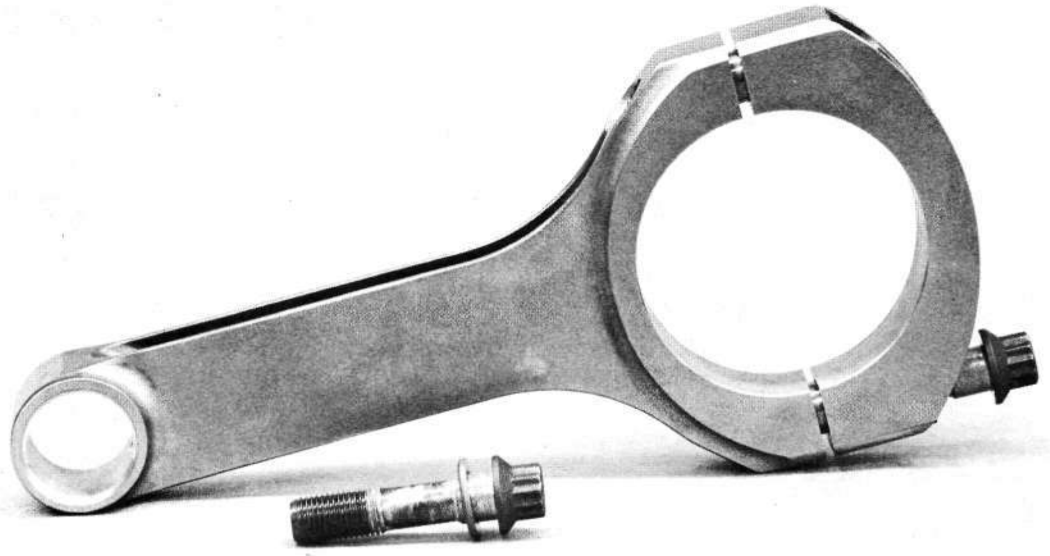
This is a particularly difficult problem in a high-rpm engine fitted with steel connecting rods. More than anything it is related to the strength of these rods in the area of the cap parting line. It is essential to keep reciprocating weight as low as possible in order to reduce the inertial loading which causes the rod to stretch. Given an aluminum rod and a premium steel rod, the aluminum rod will withstand higher engine speeds because of the greater bulk around the big end of the rod and the imparted strength across the parting line. Any engine speed over 8000 rpm is very dangerous with any known type of steel connecting rod. For short periods the premium quality (non-stock) custom rods can withstand 8500 rpm but anything in the vicinity of 9000 rpm is death on a steel rod. Even 8000 rpm, in a steady state dynamometer test approaching two horsepower per cubic inch is going to lead to bearing-related rod failure. Some of the best steel specialty rods are now being offered with more bulk in the cap area and it's certainly possible that these rods would live longer in a high-rpm application. We haven't tested the latest pieces so we couldn't say, but they do look better.

When finished and the cap is torqued in place, the bearing bore should be about 0.002-inch bigger across the parting line than when measured along the beam axis. This is providing the vertical clearance (beam axis measurement) is not less than 0.0025-inch. It



The superior machine finish of the Carrillo rod makes it a desirable choice for Nascar engines, despite the high cost. For an additional charge the rod can be fitted with S.P.S. 35N Multi-phase stainless rod bolts. These are currently the highest quality con rod bolts available. For ultimate endurance this is a good investment and they can be reused several times.

Side view of the Carrillo shows the excellent finish and the H-beam construction that makes these rods unique. They are machined from forged 4340 steel blanks. They can be ordered in finish center-to-center lengths from 4 inches to 10 inches and with 7/16- or ½-inch bolts. Finish diameter of small and big end can be set to any desired specification.



For an additional charge the Carrillo can also be fitted with a special heavy-duty cap. This adds approximately 100 grams weight to the rod, above the average 690 gram weight for a 5.700-inch model. Production finish includes X-ray inspection and heat-treating. As a result reliability is very high and the rod requires absolutely no further prep after it is received.

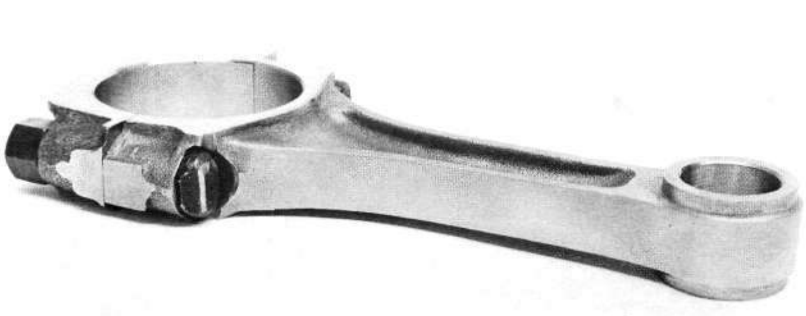
would be possible to run as light as 0.002-inch on the vertical clearance, especially on the big rod journals, with no problem as long as the parting line clearance is up to about 0.008- to 0.010-inch. Such a rod would pull the big end round with an average 0.004- to 0.005-inch clearance when the rod is dynamically stretched, without pulling the bearing tabs.

We also check side clearance in the normal manner. For drag or circle racing we use 0.015- to 0.20-inch. The latter figure is normally preferred to prevent oil trapping.

SPECIAL STEEL RODS

We have been skirting the subject for a while so it's probably best that we get on with it. In our GN engines we use special steel rods built to our specifications. We have used Carrillo rods and Hank the Crank rods with success. In the future we will be trying the new semi-finished big Chevy rod which can be machined to suitable Nascar smallblock specs.

As far as we can tell, all of the foregoing rods have been used in very competitive circle track engines. Our experience is very limited compared to some circle engine builders but we have had some success and, based on what we know at this point, we would evaluate the rods in this manner. The Carrillos are undoubtedly the class of the field but they are expensive. We have never had one fail in a GN engine but we have really used very few of them. Their finish is absolutely excellent though we feel that the area around the parting line is not as strong as it could be. They share this last shortcoming with every other brand of steel connecting rod known to us. From what we have heard lately, Carrillo is now offering a special rod with increased area across the parting line and a beefed-up cap.



A suitable alternative to the Carrillo is the new semi-finished Chevy forging. Available as part 343710, it can be finish-machined to any center-to-center from approximately 5.7 inches to 6.150 inches. The big end is already sized for the 2.1-inch throw journals. It utilizes the special 7/16-inch thru-bolt fastener used in the heavy-duty big block rods. At high speeds these rods have the greatest durability of any stock rod.

The finish rod weight of the semi-finished forging can be reduced to about 780 grams by lightening the cap in the manner shown here. By grinding material from between the cap ribs the finish cap should weigh no more than 160 to 170 grams.

When these rods are used with a crank stroke of 3.48 inches, some interference may occur between the rod bolt and the block and/or the camshaft. This can be corrected by grinding material from the block and installing 12-point aircraft nuts. To clear the cam lobes, some material may have to be ground from the head of the bolt.

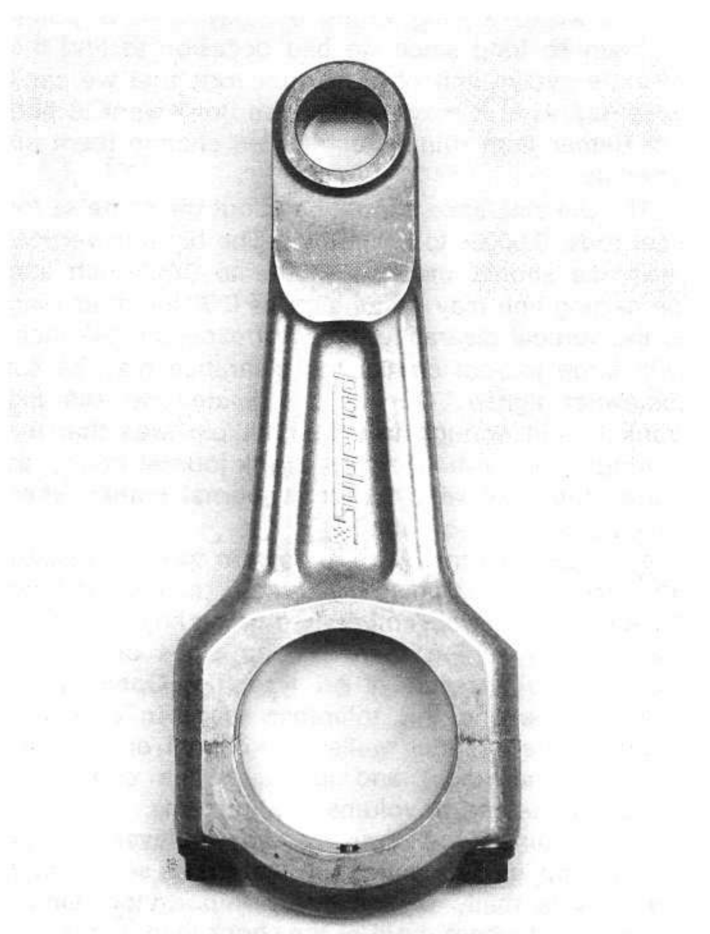
For an extra $112 Carrillo will sell his rod fitted with special S.P.S. 35N Multi-phase stainless rod bolts. These are the best bolts we have ever tested and are definitely worth the extra expense. Even at about $7 apiece (distributor's price) they are a worthy investment. It's just about impossible to imagine a better constructed bolt than this. They pull to torque "instantly" at 90 lb. ft. In most cases, the rod will fail before these bolts. We have no qualms about reusing them several times as they always torque exactly to stretch specs with no hint of trouble.

We also like Hank Bechtloff's special connecting rods. They are made of a good quality material and the overall finish is excellent. The I-beam construction is lighter than the Carrillo H-beam shape but should be very adequate for endurance engine speeds to 8000 rpm. These rods can also be ordered with the S.P.S. fastener and as an added measure of safety it might be a good idea to have these rods X-rayed.

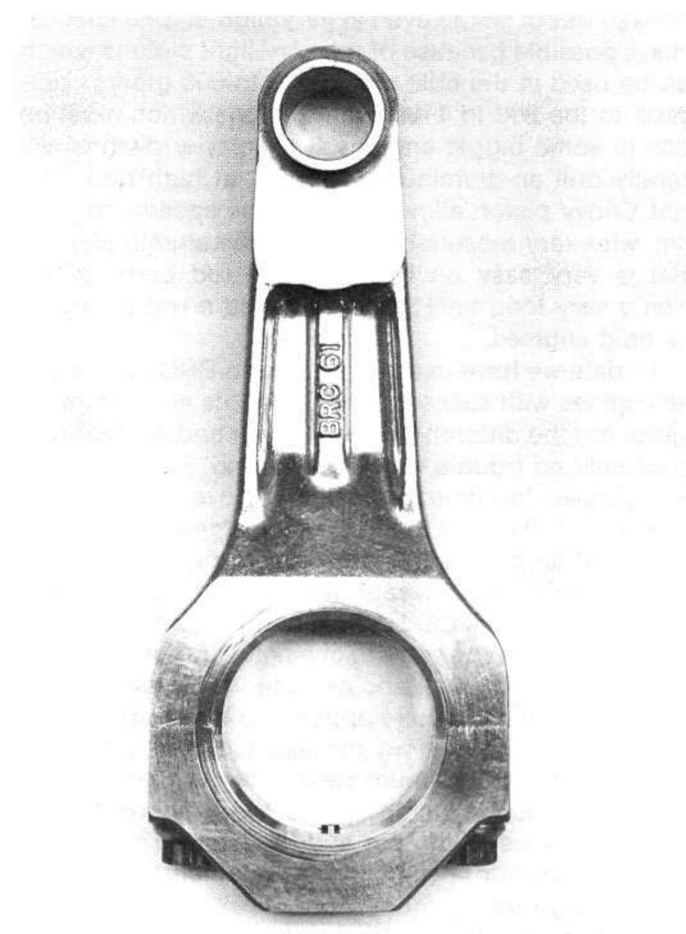
There has been some bad information circulating among the engine builders with reference to the special Chevy "big" forgings. These are rods which are forged from modified big block dies and use 7/16-inch diameter thru-bolts. They are available across-the-counter as semi-finished pieces. This allows the big rod pin end to be moved up or down to adjust center-to-center from 5.7 to 6.150 inches as desired. It is available as part 343710. It should make an ideal modestly-priced Grand National rod but there have been some reported reliability problems with them. We feel this is probably a result of improper preparation as the basic piece is more than adequate in appearance. We have no first-hand experience at this point but we feel certain that when the details are sorted out this rod will work with the best.

Any of these special steel rods should have the pin end finished in a similar manner to that described in the stock rod section. A bronze bushing should be fitted—tightly—but the oil supply should be provided from the bottom of the pin bore. Usually a pair of 3/32-inch holes are drilled up at about a 45° angle from the rod beam into the bottom of the pin

For high-speed drag racing engines aluminum rods are the ideal selection. They are light, inexpensive and do an excellent job of absorbing shock loading. When the piston weight can be held down to about 500 or 600 grams, aluminum rods provide excellent life expectancy and, when properly maintained, they should last for 70-100 runs (depending upon the maximum engine speed).



For many years we have used the Superod aluminum forging. These rods require minimal attention after they are received from the manufacturer. We may check side clearance on the big end and we check clearance between the small end and the insides of the piston pin bosses. We may lighten the small end some but there should be a minimum 0.200-inch material between the hole and the rod end.



In engine combinations requiring extremely long aluminum rods we have used the BRC forging with excellent results. These rods have proven reliable in lengths up to 6.250 inches and engine speeds to 9000 rpm. Aluminum rods are somewhat bulkier than their steel counterparts. Some grinding may be necessary to get adequate clearance with the block when long strokes are used.

bore. The lower opening of each hole is chamfered slightly. With the amount of oil splash present in a GN engine this should provide good pin oiling. Don't try oiling the pin bore in the rod from the top as there is some question as to the efficiency of this system in an endurance engine.

Now that we've told you that, we will get into a touchy subject. We hesitate to recommend this, unconditionally, but recent testing shows that it may not be necessary to bush the ends of GN steel rods. By eliminating the bushing it is possible to keep the metal around the pin thicker, subsequently increasing the strength of the pin end. To do this successfully we make sure the last hone in the bore is very, very fine and the bore is as straight as possible. The pin bore is oiled from below as described above. The bore is cleaned well prior to assembly and Sunnen pin assembly lube is used to coat the pin and bore. No special plating is used in the bore but the bore clearance must be held at 0.0008- to 0.0012-inch.

We have run such an engine for 500 miles and upon disassembly we could still see the bottom of the hone pattern. Apparently, there is enough splash in a *constant high-speed* engine to keep the pins lubricated adequately. We definitely don't recommend this for a drag racing engine where the engine speed fluctuates drastically and oil supply to the pin would be irregular.

Big end prep is less exotic in the Grand National engines. Since engine speed is not as high and the rods are stiffer around the journal than the stock steel rods, it is possible to finish them in a conventional round manner. We recommend an oil clearance of 0.0025-inch minimum. Nothing less is really suitable and it might even be possible to widen the dimension slightly across the parting line to allow for vertical stretch. We would not suggest the parting line measure exceed the vertical clearance by more than 0.001-inch. The side clearance should check at 0.015- to 0.020-inch.

ALUMINUM RODS

In drag racing engines we always prefer aluminum connecting rods. This type application is ideal for aluminum alloy. Where steel does many things wrong in a drag racing connecting rod, aluminum does so many things right. They have more bulk around the crankshaft, where it is needed, and they won't get knocked out of shape even at very high engine speeds. This is possible because of the very light pistons which can be used in the little block, 500 to 600 grams compared to the 800 to 1100 gram pistons which must be used in some bigger engines. The heavier pistons will literally pull an aluminum rod apart at high rpm. The light Chevy piston allows high engine speeds, to 9500 rpm, with very reasonable rod life. The aluminum material is very easy on bearings and rod bolts. It has been a very long time since we failed a rod in one of the drag engines.

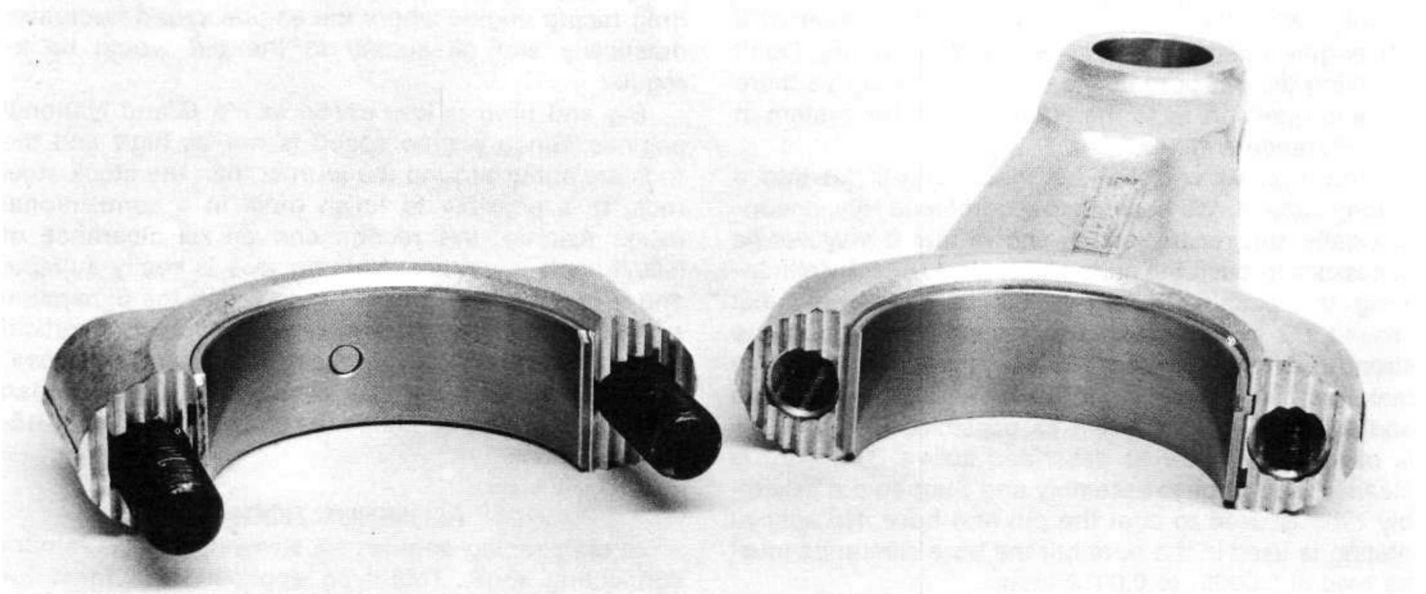
To date we have used Superod and BRC forgings in the engines with success. The Superods are somewhat lighter but the difference is very minor and we have had absolutely no trouble with either brand. Preparation is very simple. In some cases we remove a little metal from the small end of the rod if we feel the manufacturer has left an excess of material after the pin hole was bored. There should be about 0.200-inch of aluminum between the top of the pin hole and the end of the rod. Sometimes it is necessary to check the thrust clearance between the rod small end and the inside faces of the piston pin bosses. If there appears to be a possibility of interference here we cut the sides of the small end down. We prefer minimum clearance between the aluminum rod and the piston bosses. We always check the radial clearance to the pin and the bearing clearance. If these meet our specifications we cinch them to the crank and go racing.

The one well-known drawback is the rather low fatigue resistance of all aluminum alloys. There is simply no way to combat this other than chucking the rods into the trash at a decent interval, hopefully before they turn the rest of the engine into trash. At this time we never run our rods for more than 70 runs before scrapping them. Their cost is so reasonable that it's not worth fooling with them beyond this point. It's been so long since we had occasion to find the absolute cyclic limit of aluminum rods that we can't really say what it may be—and we don't want to find out! Rather than ruin an engine we change them on schedule.

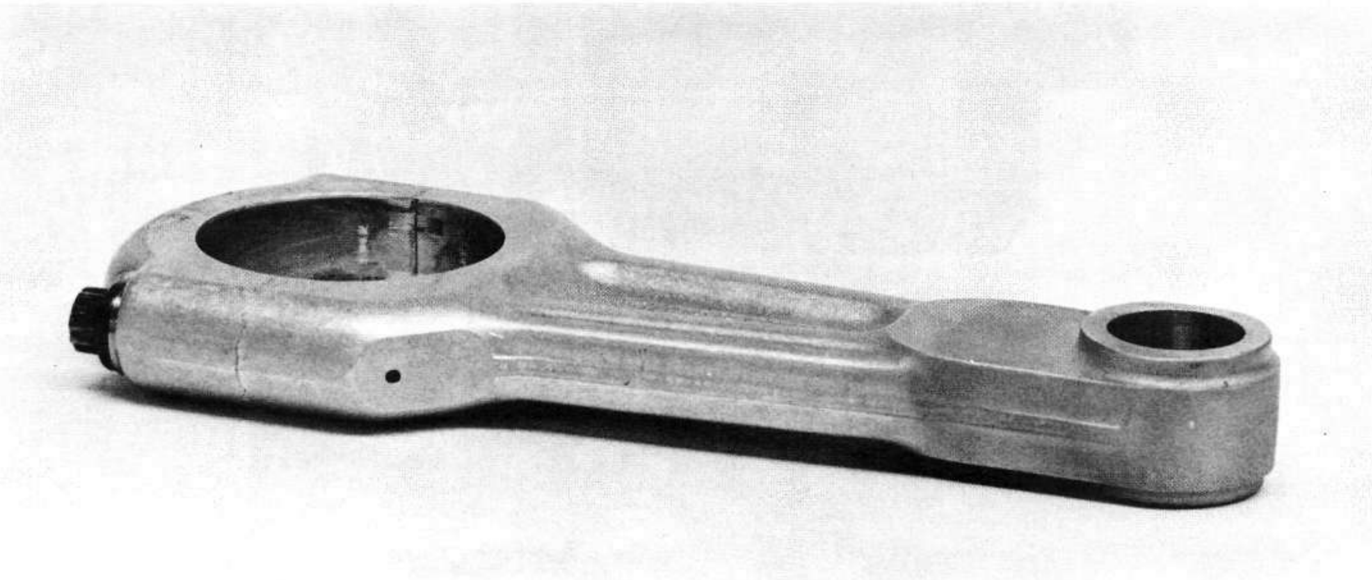
The pin clearance should be about the same as for steel rods, 0.0006- to 0.001-inch. The big end vertical clearance should measure 0.002- to 0.003-inch and the parting line may go as high as 0.008-inch as long as the vertical clearance does not exceed 0.002-inch. With large journal cranks the clearance may be set somewhat tighter. Our tests indicate that the big crank is stiff enough in the crank pin area that the bearings won't rub when the crank journal bends, as will be the case with the small journal cranks when the clearance is too tight.

For side clearance with aluminum rods we usually run a measured 0.018- to 0.020-inch. Testing indicates there is no power advantage in running any more than this. At this setting there are no signs of bearing problems as a result of oil trapping. Opening the clearance beyond this tolerance leads to excessive oiling to the cylinder walls (making oil control difficult for the rings) and increases the engine oil pump volume and/or volume requirements.

For fasteners we normally use whatever the rod maker supplies. The aluminum material is so forgiving that this is really of secondary importance. Unlike the steel rod where more of the shock load is transferred across the parting line of the rod by the rod bolt the aluminum dampens the shock so well that rod bolt failure in an aluminum piece is a very rare occurrence. When ratcheting the bolts in place we use the rod maker's recommended torque specs. We use nothing other than 30 weight oil or white lead on the threads.



The big ends of either the Superod or BRC can be finished for 2.0-inch or 2.1-inch journals. The bearing clearance for small journals should be 0.002- to 0.003-inch. To compensate for rod stretch the hole can be somewhat wider across the parting line. The parting clearance may go as high as 0.008-inch as long as the vertical does not exceed 0.002-inch. These specs can be reduced if the large crank journals are used.



With aluminum rods the radial pin clearance should be set at 0.0006- to 0.001-inch. The side clearance with two rods on the journal should be 0.018- to 0.020-inch. For drag racing purposes pin oiling in the rod is not important. In 99% of the cases we just inspect the rods for damage in shipment, check the clearances, bolt them in and race them.

The only thing to really remember when running aluminum rods is to change them at reasonable intervals. If you don't, something is going to pop.

ROD BEARINGS

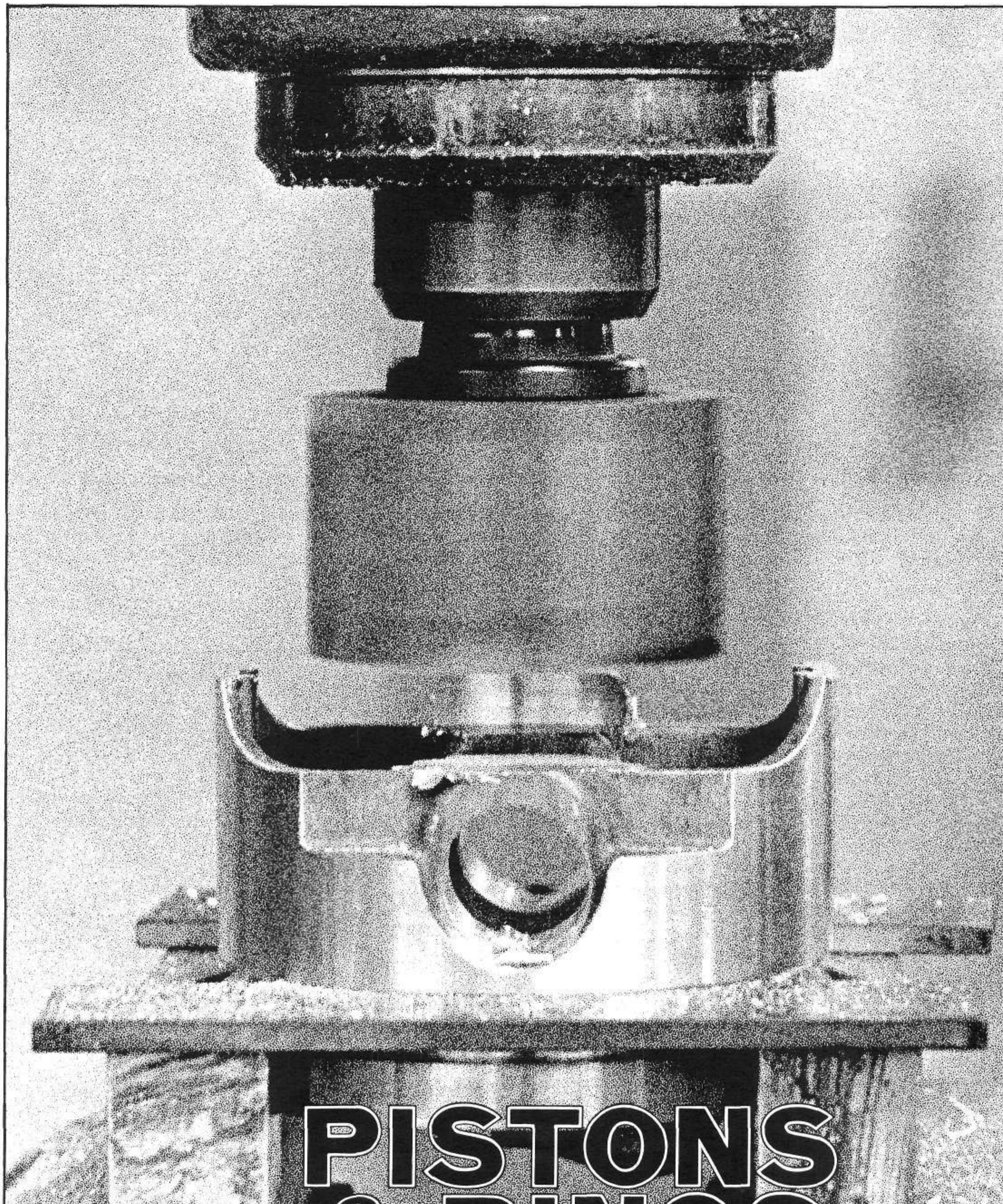
Inside of all the connecting rods we install Chevrolet bearing shells. For the reasons outlined in the crankshaft chapter we prefer to use Moraine M420 or M400 bearings. Our second choice is the Federal Mogul A220 or A200, their equivalent to the Chevy bearings. We don't use the very popular Clevite bearing because we haven't had good luck with them in the past. However, we know that several other builders use them with great success and swear by them.

In all cases we recommend that the bearings be inspected very carefully before installation. It is rare, but at times we have found some blistered bearings. For some reason one of the intermediate layers of the bearing has separated from the others or from the steel backing in very small localized areas and puckered upward. This condition will lead to early bearing failure if undetected. It is easy to spot if you polish the bearing with Scotchbrite and visually check the insert under a bright light for any small surface imperfection. Such a bearing should be thrown away.

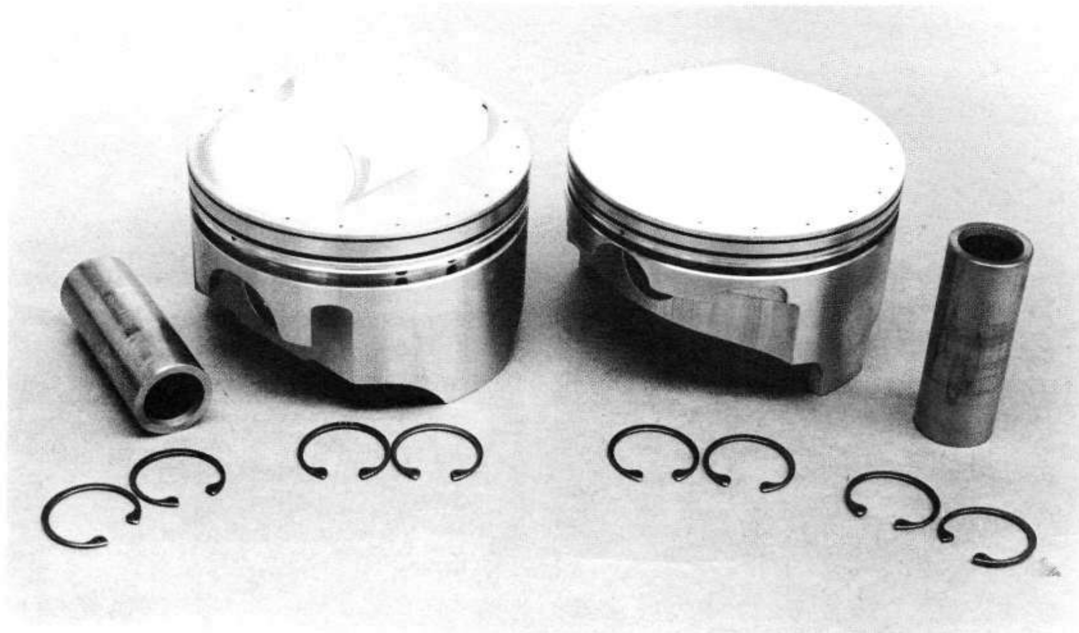
If the rod journal has been ground undersize for clearance or the counterweights have been side-ground and a fillet has been left, the bearing edges should be chamfered as described earlier. The lower bearing is pinned in the aluminum rods in the conventional manner. The Superods and BRC rods are furnished with the pins in place. Federal Mogul bearings are available pre-drilled but if it is necessary, the lower bearing shells must be drilled very accurately in a jig to insure it centers properly.

In drag engines we don't have to maintenance check the bearings at all because of the forgiving nature of the rods. We sometimes don't even have to use new bearings when the rods are changed. We only worry about them if we have experienced an oil pump failure or some similar problem which may have resulted in an oil supply disruption. In the GN engines we inspect and change the bearings after every long race, as is the usual practice.

For rod bearings we use Chevy Moraine M400 bearings. We drill them to accept the location pins installed in the aluminum rod caps. This must be done accurately to prevent edge interference with the crank journal fillets. Federal Mogul A200 or A220 bearings are also acceptable and do not have to be drilled as they are drilled at the factory. We remove the factory protective coatings from all bearings with Scotchbrite prior to installation.



PISTONS & RINGS



All of our competition engines are built with either TRW/Speed Pro or BRC forged aluminum pistons. The basic forgings are modified extensively in our shop for our specific demands. By the time machining has been completed, nearly every area of the piston has been recontoured, lightened or detailed in some special way. Much of this work is entirely unique and the result of extensive dyno and track testing.

PISTON SELECTION

There are many excellent racing and/or high performance pistons on the market today and suitable results can be obtained with nearly any of the reputable brands. For street applications the stock-type Chevrolet original equipment piston is more than adequate. It will provide some important advantages over pure racing-type pistons. Even in the roughest conceivable street use they will last a long time, provided the engine isn't constantly operated on the edge of detonation (as are most racing engines). The OE forged pistons can be run at 0.0045-inch skirt clearance. Current OE cast pistons can be set as tight as 0.0015-inch or as loose as 0.0035-inch (for racing only). This is a nice tight fit which helps to reduce piston rock, piston "slap" if you will, and the resultant shabby ring life. These pistons match the expansion characteristics of the block very well and they're hard to beat, even in a strong running "Saturday night special."

It goes without saying, however, that forged pistons are essential in a racing engine. If you are building the heat and pressure needed to make competitive horsepower, the stronger construction of the forged shaping is the only way to go.

For our particular racing program we must use special piston designs. In most of the 0.20-over engines, drag or circle, we use a TRW semi-finished piston which is not available for retail sale. It is built to our specifications and may not be of interest to the general reader. In the 0.030-over drag or circle engines we use one of the excellent Speed Pro semi-finished pistons. This design is very similar to ours, but without some of the features applicable only to Pro Stock racing. Whenever the specific requirements call for an unusual pin height or some other peculiarity that

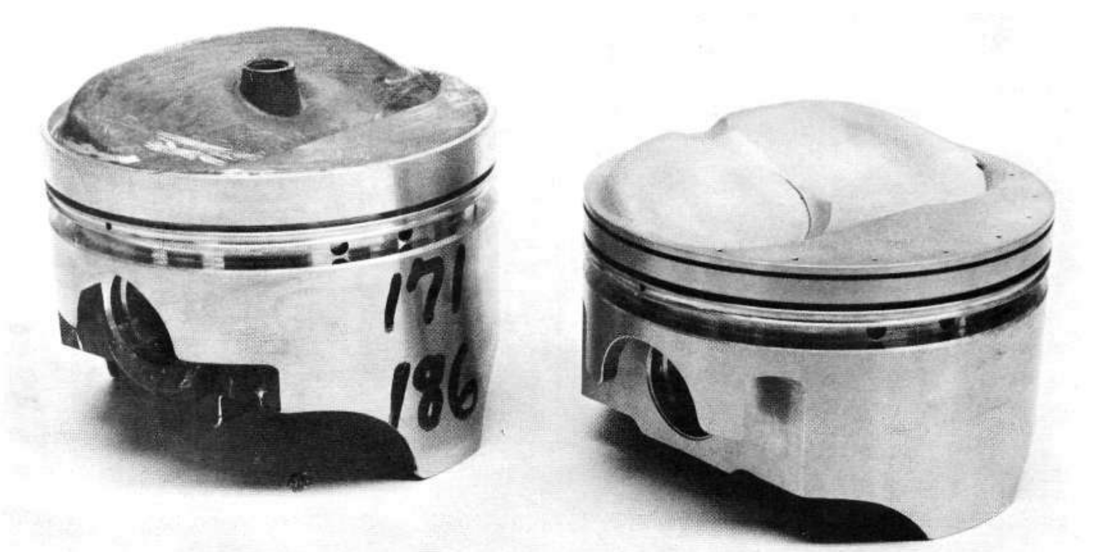
Because of our power levels and ring placement we prefer to use 0.020-inch oversize pistons in the late 4-inch cases. We use the 4.020 size because this much cylinder enlargement will remove the large chamfer from the top of the stock block. Our high ring placement would cause the ring to ride into this chamfer were it not removed. We don't like to go 0.030 oversize because this additional metal can unnecessarily weaken the cylinder walls.

On the left is one of our reworked TRW/Speed Pro pistons in ready-to-install condition for a Nascar 354-inch circle racing engine or in a 330- or 354-inch drag racing engine. On the right is a finished BRC forging for a 383- or 400-inch drag racing engine. This view readily shows the reworked deck and dome, the high pin placement, high top ring location, pressure pin oiling groove and the side-loading tabs.

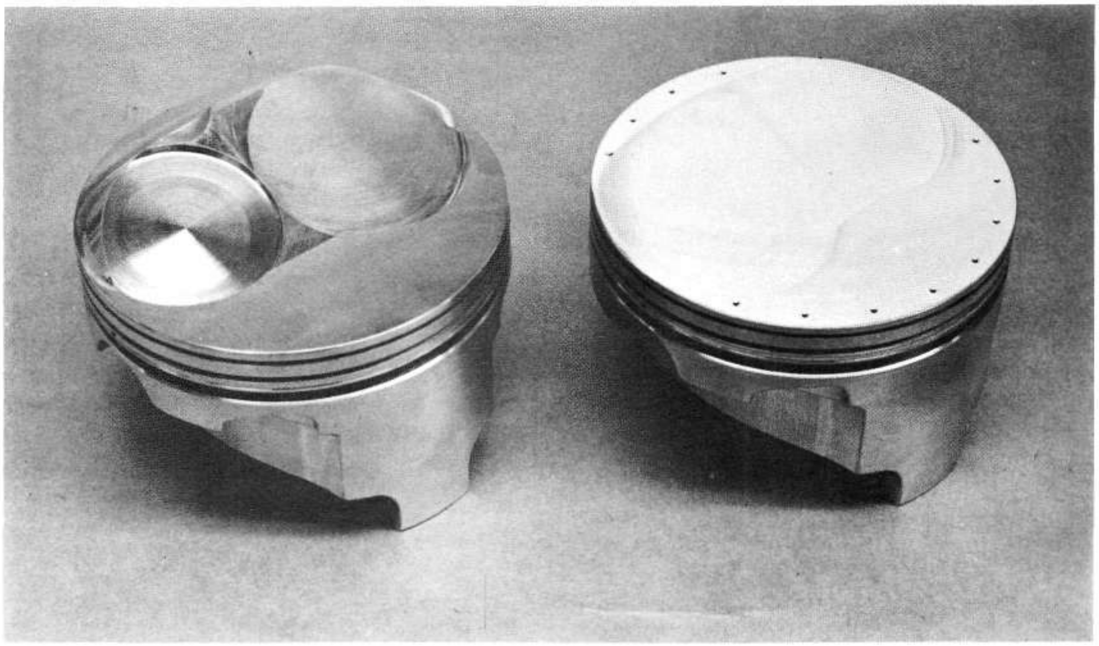
isn't available in a high volume production piston, we order custom forgings from BRC or one of the other specialty piston manufacturers. When any forged piston is properly designed and prepared for a racing engine, it should be suitable. Of course, when you approach the two horsepower-per-inch level there are some very special demands on the piston. In such cases, well-coordinated testing will be necessary to achieve maximum power output.

Selecting a basic forging, which in our case is only a springboard requiring several hours of finish machining and handcrafting, is never an easy task. Numerous factors will affect the final decision and certainly an entire text could be published about racing engine piston design, but we will try to point out some of the most important considerations. By the time the basic piece has been massaged to our specs, several man-hours will have been spent on each piston. In this light, the basic cost is really a small factor. For those who aren't operating a full-time racing program, some of the engineering discussed here and a good deal of the specific preparation may sound unreasonable. But the current power levels of our unlimited racing engines require nothing less than the most efficient piston we can produce.

Despite our "trick" work nearly any builder can get within a gnat's lash by using production parts. An example is our 333-inch drag engine. This particular combination is a 3.25-inch crank with a 4.030-inch bore. We don't necessarily like to go bigger than 0.020-inch on the bore but there aren't any production pistons that we know of for a 0.020-over bore, except the relatively heavy 0.020-over 302 piston. However, with a 30-over engine it is possible to use an economical and basically very sound production forging, either a TRW or Speed Pro, semi-finished, 3.48-inch stroke, 350



Getting the TRW/Speed Pro semi-finished forging in shape requires much more finish work. We do prefer this piston over the "California" forgings because it is made from a superior silicon-aluminum alloy. They seem to resist ring land damage better and they have excellent expansion characteristics. This blank is not generally available with a 4.020-inch finish size. By buying several hundred pistons at a time we can have them built to our own specifications. The 7544PS+.030 Speed Pro is nearly identical except for the 4.030 size, and has the same semi-finished deck.

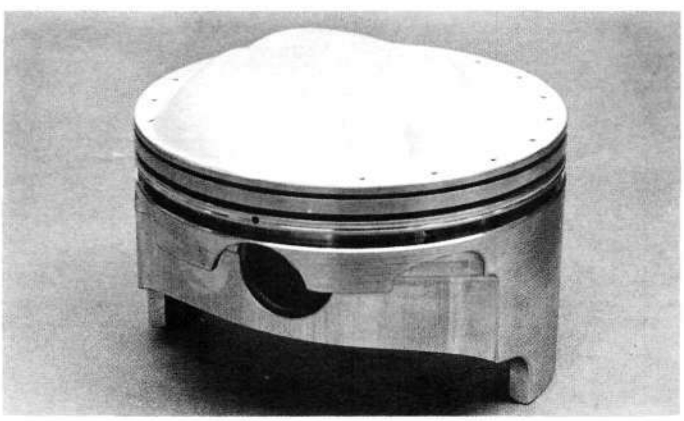


The BRC piston requires very little special work to get it ready for installation. Note that the pin oiling groove and the ring grooves are already completed on the factory sample (left). This is a fairly light piston as delivered and the factory machining for grooves is accurate. They can locate the grooves as desired by the customer and can cut them to any width. Most of our time is spent reworking the deck and dome to the specific application.

piston, in combination with a longer, 5.815-inch rod and a 3.25-inch crank. Happily, this gives a good rod ratio and a light piston with good ring groove location (the semi-finished piston has only the bottom and second ring grooves). Without pin and rings (and prior to further lightening efforts) this piston weighs 574 grams, at least the Speed Pro model does, and is available as part 7544PS+.030. The pin hole diameter is 0.9272-inch, the second ring groove is cut for a 1/16-inch ring and the oil ring groove is for a 3/16-inch three-piece ring.

All of our pistons have a conventional three ring groove design, except some short compression height pistons used in our 354 drag engines. These experimental engines use BRC forgings which have a 1.06 compression height and only two ring grooves. The top ring groove is cut for a 0.043-inch ring, as are all of our pistons, and the oil control ring is a ⅛-inch three-piece combination. This piston is used exclusively with a 6.25-inch aluminum rod on a 3.48-inch stroke crank. It is, however, entirely possible to put a three ring combination in this piston, which is the more usual case.

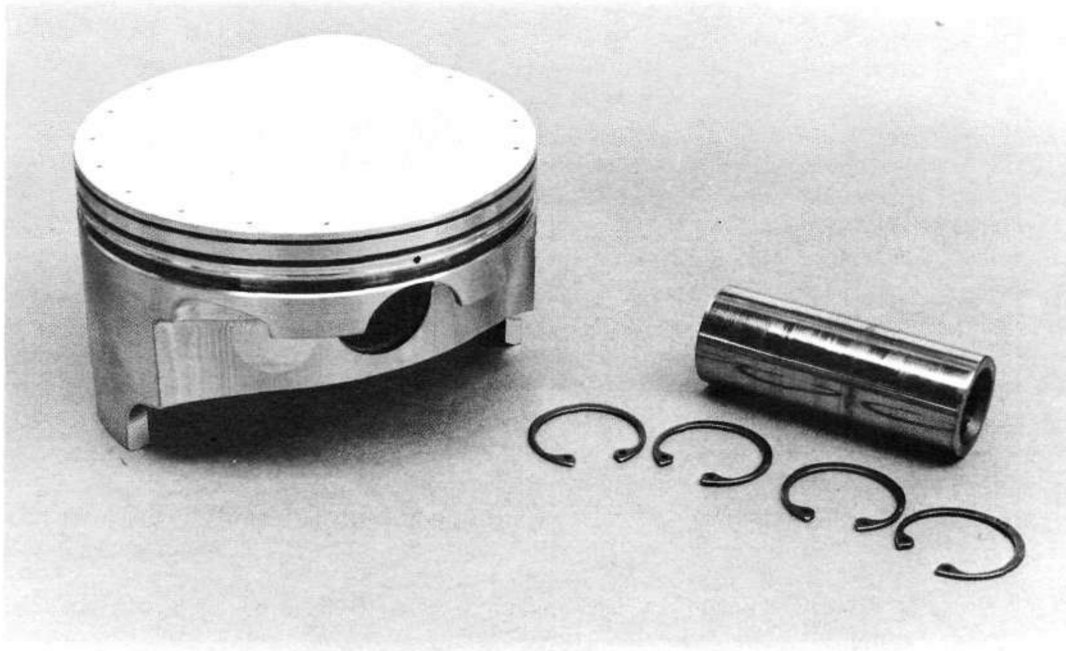
Any time the engine design will allow, we prefer to use a TRW or Speed Pro piston. In our opinion, they are manufactured from a superior silicon-aluminum alloy, compared to the somewhat less desirable 2018 alloy used in most "California" pistons. The ring lands stand up better under the punishment of a Pro Stock engine. This is exceptionally important with the unusual ring package we use. We find the top land will not pull as soon but this may not be important if the



This is a BRC for a 3.48-inch stroke, 354 cubic inch, match race drag engine with a 6.25-inch rod. The compression height is approximately 1.06 inches, squeezing the ring grooves into a tight package at the top of the piston. The top groove is cut for an 0.043-inch wide Speed Pro moly ring, the second is for a 1/16-inch ring and the bottom accepts a ⅛-inch, three-piece oil ring. This short deck height requires that the piston overall length be quite short, otherwise the loading on the skirts may create cracking at the point where the extension tab joins the thrust face.

For a drag racing 330-incher we prefer to use the TRW pistons because they give a better rod-to-stroke ratio for the smaller displacement and still allow considerable latitude for ring placement. In this case we combine the 1.565-inch deck height with a 5.815-inch rod and a 3.25-inch crank or we may shave the deck some and lengthen the rod a corresponding amount to lighten the piston. The ring package is a 0.043-inch, 1.16-inch, 3/16-inch combination. Though many builders prefer a 6-inch rod in their 354-inch Nascar engines we feel this same piston with a shorter rod will produce comparable power to any trick stuff.

In a very high speed engine or an endurance engine every effort must be made to get the pistons as light as possible. We spend several hours reducing the weight of each piston by even a few grams. However, it is dangerous to reduce the mass enough to endanger durability, especially in the area of the pin bosses and the deck. The BRC piston can be made as light as 360 to 380 grams, and with a 120 gram BRC, 0.9272-inch diameter, 2.5-inch long pin, rings and double locks the whole affair can be kept at approximately 500 grams. For in-car service a 70 gram pin can also be used.

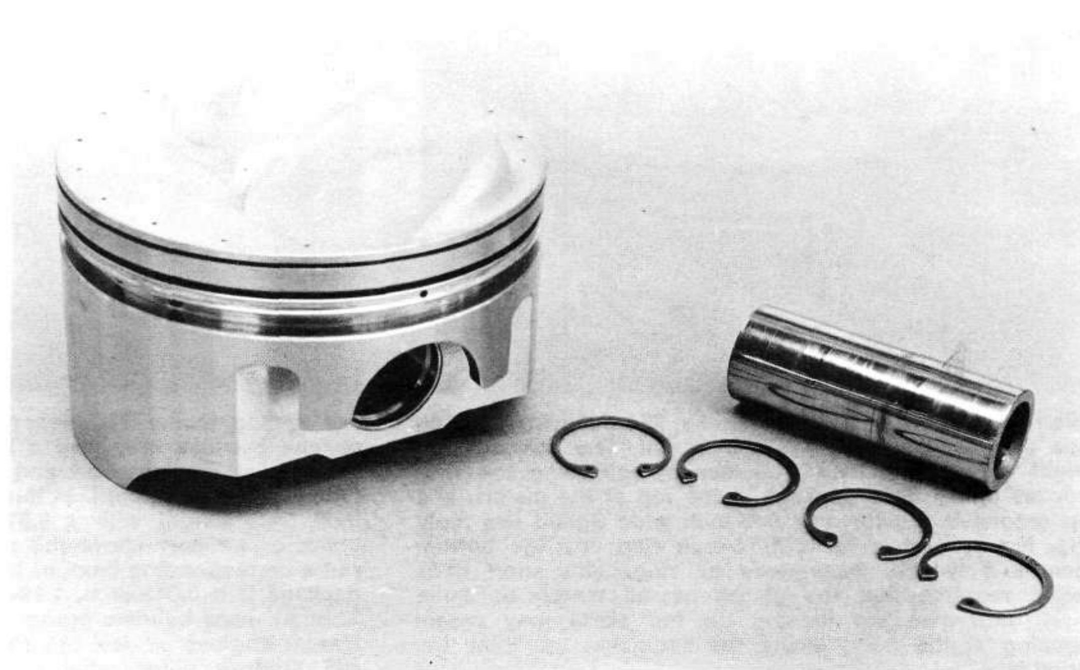


top ring is not placed as close to the top of the piston as we prefer. They also have a somewhat less tendency to "stick in the bore" compared to the West Coast stuff. We can run the TRW/Speed Pro pistons at 0.008- to 0.0085-inch wall clearance with no problem while we are forced to use a good deal more running room with the California pistons to prevent scuffing.

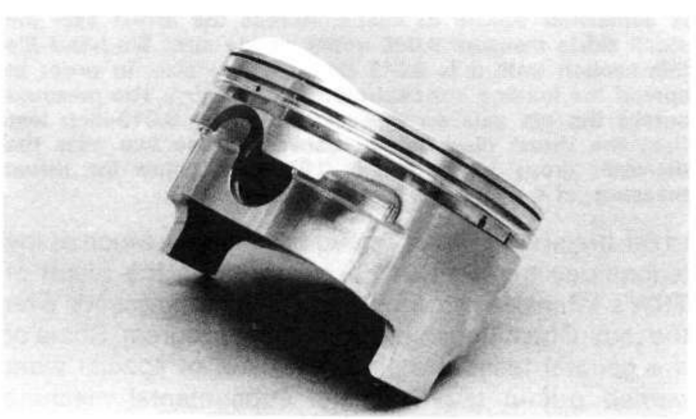
Currently, all the pistons we use are of the typical slipper skirt design. In recent months we have taken a second look at the conventional full-skirted piston and may be experimenting with them in the near future. Cylinder wall loading has been plaguing most of our very high output engines. We constantly fight wall life now and as power approaches 700 hp there is every reason to believe it will become an even greater problem. The Arias and Venolia full skirt pistons, just to name two, are quite light in weight and the nice round skirts should spread the bore loading sufficiently to reduce the problem. Conversely, we know the increased area will increase frictional drag. However, if we could get an edge on the loading problem it would be a good trade off. Since the inside punch die can go all the way up into the dome around the full skirt, we have noted the weight saving possible with these pistons. This, too, is an intriguing prospect.

Since piston weight is so important we might as well get into some specifics here before we go on to a detailed discussion of special work. In our long rod/short compression height drag engines the BRC pistons usually weigh about 360-380 grams and the pin about 120 grams. In the TRW-equipped 330 drag engines the piston normally averages about 480

The longer compression height of the TRW forging necessarily makes it a heavier assembly. For drag racing this piston may be as light as 480 grams with a 70 gram pin plus rings and locks. This gives a total in the neighborhood of about 550-560 grams. For endurance engines we leave a little more deck in the piston, but we have successfully run them as light as 530 grams. With a stock Chevy, 0.9272-inch diameter, 2.90-inch pin weighing 245 grams, this gives a total of about 785 grams. We have run pistons as heavy as 575 grams on this pin, totalling as much as 830 to 840 grams, without any trouble.



grams with a 120 gram pin. This works fairly well on the dyno but for in-car use we can substitute a lighter 70 gram pin. The light pin simply won't live in a dyno engine. But we normally use the heavier pin so we won't have to pull the engine down to change the pins if we want to drop a competition engine on the dyno for a few check-out runs. The 50 gram difference isn't worth that much work and we like the flexibility of getting an engine out of the car and on the dyno in a short time. Inside the super-speedway Grand National engines we use a 530 gram piston with a heavier, more durable 145 gram pin. In certain instances we may leave the tops of the pistons in a circle engine a little thicker, resulting in a piston weighing about 575 grams.



Piston body/skirt shaping is very important for ultimate performance. When the piston reaches operating temperature it must expand to the proper shape and dimension to give maximum support of the piston rings. When it reaches this hot shape it must also do an adequate job of transfering the heat of combustion away from the chamber and into the walls and cooling system. Because of the different heat concentrations and the various material concentrations, the piston will not expand at the same rate in all directions. There must be an allowance in the cold shape to compensate for these variations.

SKIRTS

The details of piston skirt shaping are very important when ultimate horsepower is the goal. This is certainly a critical factor in stock-type engines, but in very high rpm powerplants it is crucial. It is essential that when the piston has been heated to operating temperature it expands to a mechanically efficient configuration. In other words, when it's hot it must ride smoothly inside the bore with a minimum of rocking or side slapping; it must support the piston rings precisely as they perform the all-important task of sealing the combustion chambers from the crankcase; and it must be durable enough to withstand the adverse effects of heat and detonation pounding.

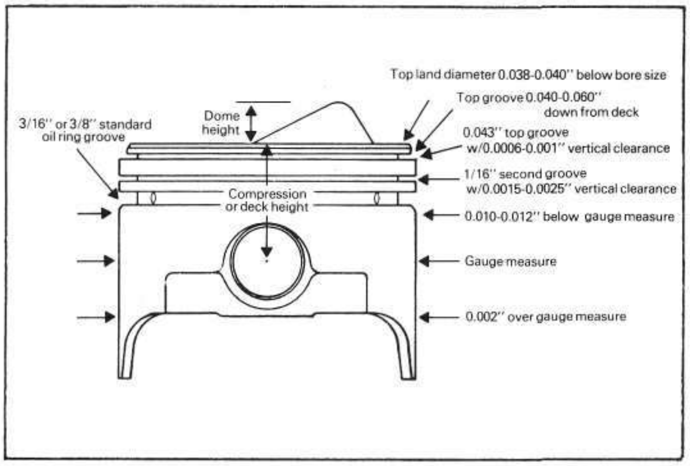
Because most current pistons are made from aluminum alloy they will expand and change dimension considerably when they reach operating temperature. Unfortunately, the temperature will vary greatly from the top of the piston to the bottom of the skirt and cause the metal in different areas to expand at different rates. The unique shape of a racing piston skirt has to be designed to accommodate these variations. If the piston does not expand to a precise heated shape, it may operate too loosely in the bore, causing excessive "rock" and destroying the ring seal, or it may expand too much and scuff or seize in the bore. Some consideration must be given also to the fact that when the engine is first fired the piston will be cold and it will take some time before it reaches operating temperature. During this time if the cold shape is too radical to provide sufficient support for the rings, the engine will not be very long-lived, piston ring life will be very short, and it is even possible that fractures may develop in the piston as it rattles around in the cold bore.

Typically, a cold piston is not completely round when viewed from above. It is elliptical or "cam

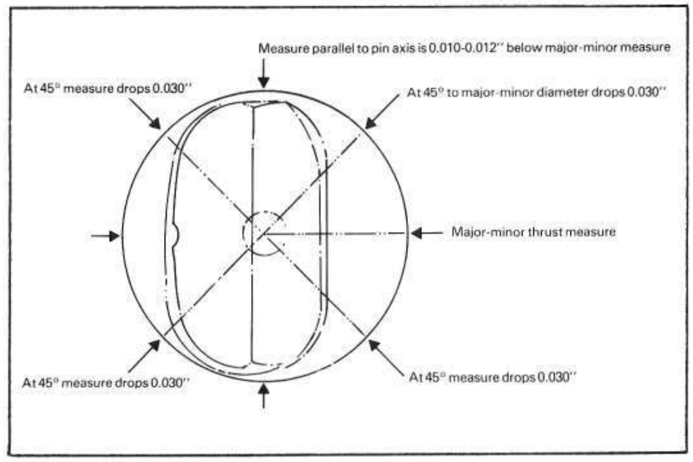
Our pistons have a cloverleaf or "square" cross section at the skirt line. The measure across the thrust faces and across the pin axis line will be larger than a comparable diameter measured on a 45° axis to these principle dimensions. This gives the piston more support during cold start-up and reduces sideslap but allows it to expand adequately when it is hot. It will also have "barrel-shaped" skirts. In this case the measure across the thrust axis, at a point immediately below the ring grooves, is smaller than a comparable measure on the gauge point. Normally, the skirt will also pick up some size below the gauge point.

ground" in the skirt area; the dimension across the skirts where the thrust loading is imparted is larger than the dimension across the pin bosses. This allows the skirts to provide some support during cold start-up, yet provides room for the piston to expand when it gets hot. As the piston expands along the pin bore axis, the pin bosses move apart and the pin locks move away from the ends of the piston pin. This condition, combined with the side loading imparted to the piston by the crank thrust, leads to problems or failure with the pin locks. The cam shape must, therefore, be carefully designed.

The barrel shape is also important for reliability. The most critical areas are where the skirt begins and the pin bosses are located. Since the heat of combustion will be concentrated in the uppermost portion of the piston, the greatest amount of expansion will occur here. The head dimension must always give sufficient room for the rings to "breathe" without significantly reducing the support which the ring lands impart to the rings. Most of this absorbed heat will be transferred from the pistons into the cylinder walls through



SIDE VIEW — The barrel shape measures approximately 4.020-inches at the gauge line. Moving above this line the skirts drop off in measure to about 0.006-inch under gauge at the bottom of the lower ring groove. From here up the head is straight except for what reduction may be necessary in the ring land circumferences to keep them from riding on the walls. From the gauge point downward the skirts pick up measure to about 0.002-inch over gauge at the lower tips of the skirt extensions.



TOP VIEW — The cross section at the very top of the skirt is somewhat square in shape. Across the thrust axis the stock skirts measure 0.006 under gauge size. We hand-file this section until it is 0.012 under gauge size, in order to spread the loading concentrated at this point. The measure across the pin axis on this same plane is 0.010-inch less than the thrust diameter. In between these two axes the diameter drops off until it is 0.030-inch below the thrust measure, at a 45° angle to the two axes.

the rings. However, a good portion of the heat is not moved through the rings but is concentrated in the upper portion of the skirts where it is then transferred to the walls. This heat causes a greater area of contact, decreasing the unit loading. It is important to observe *used pistons* carefully to determine if the size and shape is correct to balance this heat expansion and load concentration. Improper loading can be observed as signs of scuffing. These indications are usually found up under the oil ring or down at the lower corners of the skirt.

All of our competition 330 drag engines and 354 GN engines currently are fitted with TRW forged pistons. These are parts which fulfill our specific requirements but, in most major aspects, they are similar to off-the-shelf TRW or Speed Pro pistons. Much of the technology behind this design shape is the result of TRW's intensive research and development work with the Indy-Championship Offenhauser program. Some of the general ideas are also the result of special work carried out in the Chevrolet experimental machine shop. Through researching these and other sources we developed our own design specs, to which these pistons were built.

This piston has a barrel shape, as is common with most TRW racing pistons, but looking down at the piston from overhead, the skirt has a cloverleaf shape rather than the more conventional single ellipse cam shape. The "square" design is wider at two points,

Our most recent efforts with piston design have been aimed at reducing side movement of the piston and rings. This arises from the thrust of the crank and rod assembly bearing against the piston pin. It can adversely affect ring seal and is a contributing factor to lock ring failure in GN endurance engines. By leaving side support on the pistons and, perhaps, shimming the rod small end inside the piston boss we feel we can eliminate this mechanical problem.

When using short pins we like to mill as much metal as possible from the outside of the pin bosses to lighten the piston. Cutting this material away entirely aggravates ring support problems and pin lock ring troubles. Leaving the support tabs reduces the frequency of lock failure in a long distance engine and increases the quality of the ring seal in both the drag and long distance engines.

one at each end of an imaginary axis parallel to the pin bore axis but slightly above the pin bore, and it is wider at two points on either end of the major thrust centerline. In between these four "high spots" the skirt measures slightly smaller in diameter. The longer dimension across the thrust faces is typical of a racing forging. The wider measure across the pin axis is less usual. It is an effort to reduce side slap of the piston which has become a problem, especially in the long distance engines. The measure across the pin is not as large as that across the major faces.

Describing the shape in detail is difficult, but it's something like this. First, the barrel of the body begins undersize at the top of the skirt, the point just under the lower ring groove and picks up about 0.006-inch at the gauge point plane, which coincides with the horizontal plane of the pin centerline. On down the skirt it picks up more to 0.002-inch over gauge point at about the lower end of the side-loading tabs (seen in the photos).

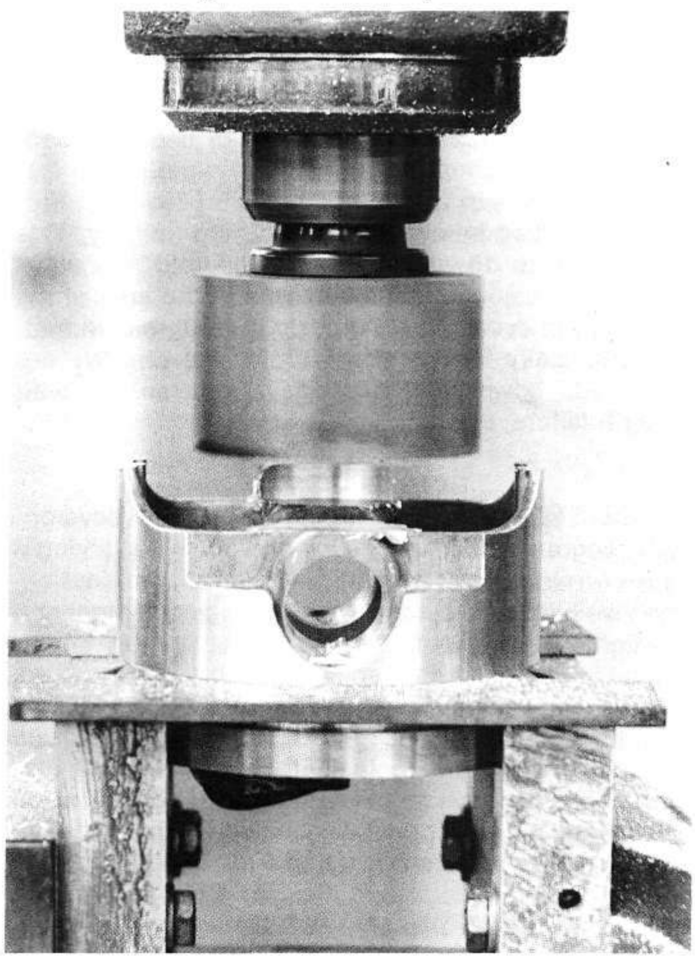
The cloverleaf shape of the skirt drops off 0.030-inch at a point 45° between the thrust axis and the pin axis. It picks back up, on around to the pin axis, until it is within 0.010-inch of the major thrust measure. We have found that with this much cam there is still a pretty strong load concentration area just under the oil ring on the major thrust face. In the past we have been hand-filing a small triangular area down under the ring and this has helped. Now we drop the diameter in this area to 0.010- to 0.012-inch under gauge point and let it pick up gradually to the major thrust point. The load pattern looks very good with this shape. In the future to eliminate this handwork we may increase the barrel shape by reducing the overall diameter up under the oil ring by 0.010-inch.

Another unique feature is the side-loading tabs. Previously, we cut this material away entirely to reduce piston weight. The result was increased ring wear trouble from side loading. Now, we cut in behind the piston boss protrusions with a rotary mill to create a side-loading tab. These short flexible tabs cushion and protect the rings from fore-and-aft piston loading created by crankshaft whip.

In addition to the tabs, we have given some thought to installing shims between the rod small end and the insides of the piston pin bosses. With small aluminum (or similar material) shims on the pin we hope to reduce the force with which the pin pushes against the pin locks. They would minimize the slack between the pin bosses, and when the rod pushes the pin sideways it would not bang into the locks. Instead, it will contact the shims and they, in turn, will bear against the pin bosses. This transfers the side load directly to the piston, instead of to the locks. We know the entire load could be transferred directly to the cylinder walls by using pin buttons. However, aluminum or Teflon buttons can add as much as 200 grams to the piston assembly weight. This is too much of a trade off in weight for a slight gain in ring/pin lock stability.

The system is only theoretical, as we have not tried to build an engine like this, but it would work well with the side-load tabs and the unique cloverleaf piston shape. Once the loading is effectively dispersed to the walls, piston pin lock failure and ring wear trouble associated with "side slap" is eliminated. This may not be tremendously helpful in a short duration drag engine but the effects could be very beneficial in a long distance Nascar engine. These effects are always troublesome in this type engine. The constant high speeds and the somewhat heavier pistons used in these engines increase crank whip and side-loading troubles. If this system would work, part of the GN smallblock endurance problems may be reduced.

This is all very esoteric stuff which has little value to the average guy because he can't get pistons from TRW like this. It is possible to get something similar from BRC if you know exactly what you want. They have always done a very good job of putting together special pieces for us, however, it may require some additional filing along the upper portion of the skirt to get them loading properly. We are not absolutely convinced there is power to be had from this piston shape but we have seen some encouraging signs. In a recent test engine we suddenly found 12-15 horse-



Piston lightening can be affected by cutting as much material from the lower part of the skirts and pin bosses as they can stand. The bosses can be reduced in length until only 0.120-inch is left between the pin bore and the bottom of the boss.

power when using these pistons in a block with well-centered bores. Upon takedown we found that the walls looked better than in any racing engine we had ever previously inspected.

In the drag racing engines which are equipped with TRW/Sealed Power pistons we have also found it necessary to do some machining in the skirt and pin boss areas. The TRW piston is a little longer from the deck to the lower edge of the skirt than the BRC piston but the BRC is longer from the bottom of the oil ring groove to the tip of the skirt, giving it a little more support. With the high pin TRW/Sealed Power pistons we have had some problems with skirt cracking, on either side of the skirt tails. We have solved this by shortening the skirts. It is difficult to give an exact recommendation here because cutting the tails will also reduce the piston stability. If cracking does not occur they need not be cut away, but if they are shortened they should be reduced the very minimum possible to eliminate the cracking.

It is also possible to remove material below the pin bores. This is strictly a matter of reducing piston weight. We usually leave 0.180-inch between the bottom of the pin bore and the bottom of the pin boss. In any case this dimension should probably never be less than 0.120-inch in a high speed drag engine. In our application the average pressure on the pin is reduced considerably. There is really little likelihood of a failure at this point. At the top of the dead stroke there is only a small amount of negative loading. Every pin failure we have ever experienced has been with the pin under compression loading. Frequency-induced failure is another thing entirely. This has nothing to do with maximum loading. Any small crack progression which may begin in the area of the pin boss will develop very quickly to a full-blown fracture, especially if it is down under the pin. We are convinced, however, that this is not an ultimate strength failure, as such.

RING GROOVES

If there is any one area of our research and development program which has proven personally satisfying it is the current piston ring and ring groove combination. When we began studying very high ring placement and very tight vertical groove clearance, many highly-knowledgeable and respected engine builders predicted we would meet with failure. Past studies based loosely upon the same theory had not been successful for one reason or another and logically there seemed little reason for our program to prove fruitful in light of these precedents. We were convinced, however, that once the problems were resolved we could substantially improve ring life with this approach. Despite other failures, we knew that similar designs had been operated very successfully in one and two cylinder motorcycle engines. We feel our current package is highly efficient and with some degree of immodesty we are quite proud of the results.

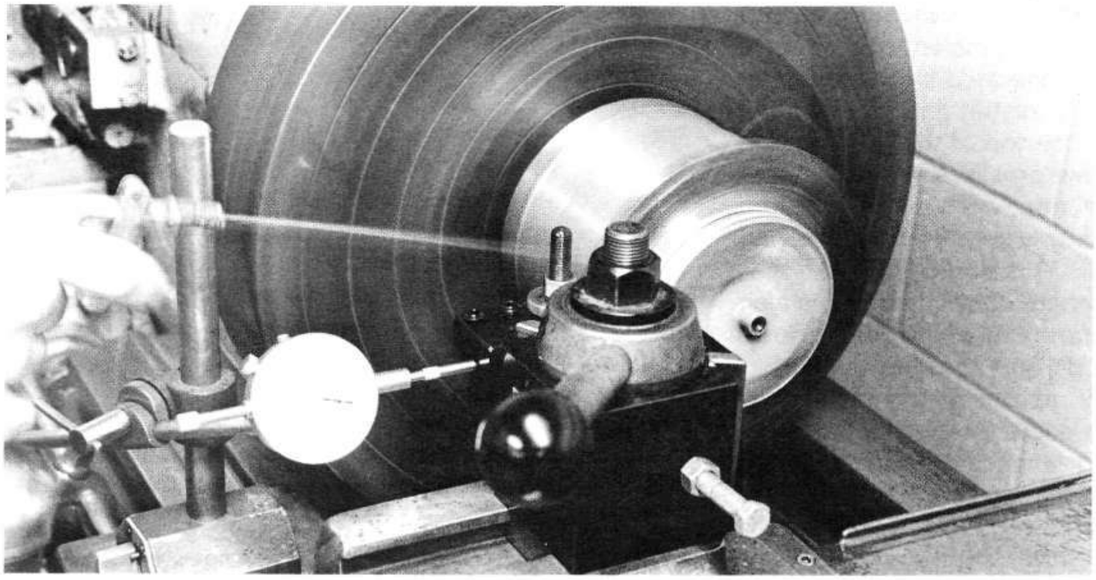
In simple terms we have taken a fairly standard racing ring package and placed it very near the top of the piston. The top ring is most important in any combination. In our particular case we have reduced the vertical clearance between the ring and the top and bottom of the groove. If this clearance is reduced too much the ring will not operate properly because it relies very heavily upon the gas pressures developed during combustion for correct sealing. To compensate for the reduced area for gas operation, we drill gas ports in the top of the piston to insure the gas pressure has sufficient area to actuate the ring. The object of this entire reduced-clearance program is to increase ring life. We have never been able to prove that there is any horsepower increase, per se, in this arrangement. We do know, however, that laboratory testing with small displacement motorcycle engines has verified power improvement from the high top ring placement. Theoretically, the same should be true in larger V8 engines. *And in fact, once we worked out the total combination of high ring placement, small vertical clearance, and piston deck gas ports, we found a definite power improvement in our drag race engines.*

If there is any one area of racing engine operation where widespread misunderstanding exists, it is piston ring functioning. For the rings to operate efficiently, a little blow-by is absolutely essential! A zero blow-by engine cannot stop oil from reaching the combustion chambers — exactly the opposte of what many believe.

It is essential to understand the specific mechanical functioning of the piston rings and their interrelationship with the piston and piston grooves. The best way to explain why our ring configuration works is to examine this chain of events. Most standard piston ring packages utilize three rings. The top ring, often called the compression ring, is the primary barrier between the hot combustion and high pressure conditions developed in the chambers and the relative lower pressure in the lower case. It must seal the piston to the walls so the pressure of combustion cannot escape, resulting in lost power. In addition, it transfers the greatest amount of heat from the piston into the walls where it can be carried away by the coolant system.

The second ring serves a similar function but to a lesser degree. In effect, it is only a backup. However, it provides an additional intermediate stage or sealing point which effectively reduced the pressure drop between the top (high pressure) and the bottom (low pressure) of the compression ring. This pressure differential is extremely important to ring functioning. The greatest differential occurs across the top ring while a significantly smaller differential exists across the second ring and the lower ring. The second ring also assists the lower ring in controlling the oil distribution on the cylinder walls. Likewise, it assists the top ring by providing additional area to absorb the load transferred to the rings by the motion of the piston.

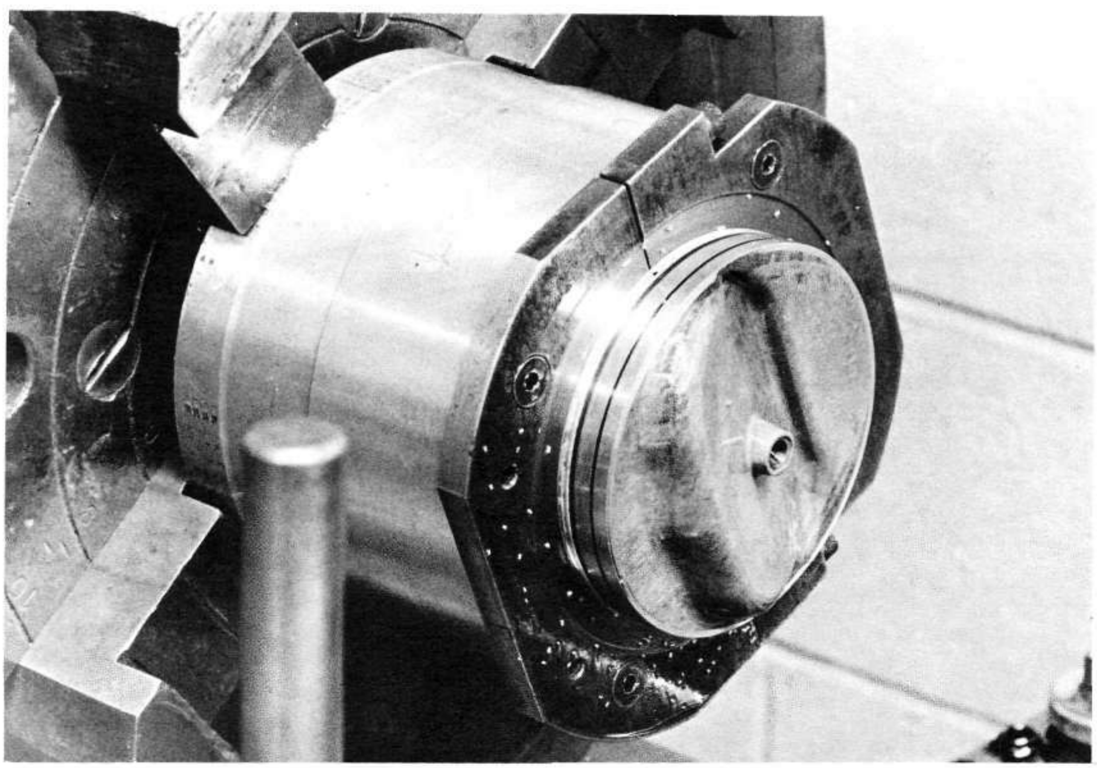
The bottom ring, usually called the oil ring, is most often described as an oil scraper. In reality it does not act as a true "scraper." This indicates that the ring scoops up the oil residing on the wall and dumps it



We always cut our own top ring grooves or have the specialty manufacturer cut them to our specs. We place the groove close to the deck to reduce the dead space around the piston circumference. The groove is also very narrow to reduce the vertical clearance to the ring. This technique cuts down ring flutter but must be combined with the drilled ports in the deck to gain correct ring pressurization.

back into the lower case. There is some slight scraping action, but with current technology we know the pressure differential between the area above the ring and the area below the ring (exposed to normal crankcase pressure) is responsible for the evacuation of the oil from the cylinder walls. This is where the blow-by phenomenon becomes extremely important. When the area above the ring receives pressure from combustion it becomes a positive pressure "chamber" relative to the lower pressure in the crankcase (this is becoming an increasingly greater differential with the new vacuum-extraction systems currently in use). This differential picks the excess oil off the walls and blows it through the oil evacuation holes drilled in the lower groove. If this positive pressure evacuation system does not operate correctly the engine will be plagued by upper end oiling and in case anybody is not aware of the fact—chamber oiling is absolute death in a racing powerplant. It's the surest way we know of to trim 10 or 15 percent off the peak horsepower figure. In an engine which has an absolute pressure seal at the top ring, upper end oiling always occurs because the lower ring cannot possibly "scrape" the oil off the walls sufficiently, and the excess causes the upper rings to stand up and skip along the wall. The leftover oil winds up in the chambers. The blow-by also helps keep the upper rings from hydroplaning over the walls by blowing the oil off the face of the ring and forcing it down toward the evacuation area.

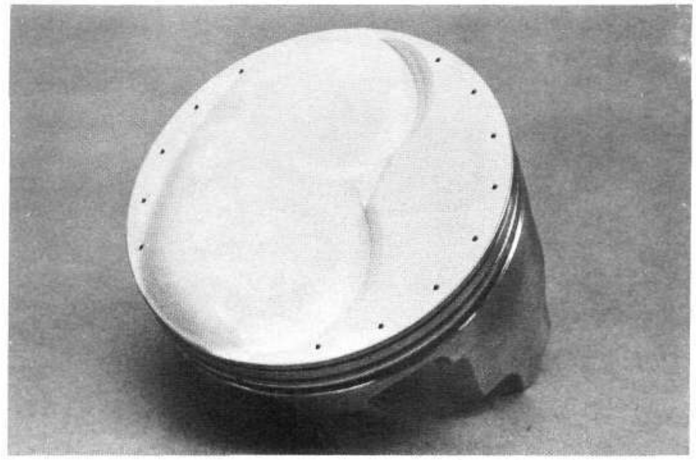
Gas pressure from combustion is most important to the functioning of the top ring. The seal provided by



Cutting the top ring groove is not difficult when a fixture such as this is used. It squares the piston off the bottom ring groove and acts as a chuck to hold the forging in a lathe. With a 0.043-inch wide ring the groove should give 0.0006- to 0.001-inch vertical clearance. For drag racing it should be down 0.060- to 0.080-inch from the deck. We prefer the face depth to be 0.000-inch (net) or 0.004-inch below the land, at the most.

the compression ring is not totally created by the radial tension of the ring. The tension does help a little but most of the seal is generated by the pressure developed in combustion. The piston ring groove must provide clearance between the top of the ring and the roof of the groove. This allows pressure from the chamber to bleed down to the ring, across the top, and into the back face of the ring, where it forces the ring to expand against the wall. This vertical clearance is essential for the ring to operate correctly but it is also a detriment. When the piston is moving up and down in the bore this clearance permits the ring to "flutter" in the groove. Specifically, as the piston rocks over TDC in preparation for the power portion of the cycle, the ring continues upward when the piston changes direction until it slams into the top of the groove. This seals the back of the ring from gas pressure and the overall ring-to-wall pressure is reduced. At the same time, combustion pressure is building above the ring. These circumstances can combine to break the ring seal completely, allowing excessive blow-by across the face. We said earlier that some controlled blow-by was essential for oil control, but leakage across the face of the ring is not what we had in mind. It causes more heat buildup in the piston and ring (due to the loss of heat transference between the ring and wall).

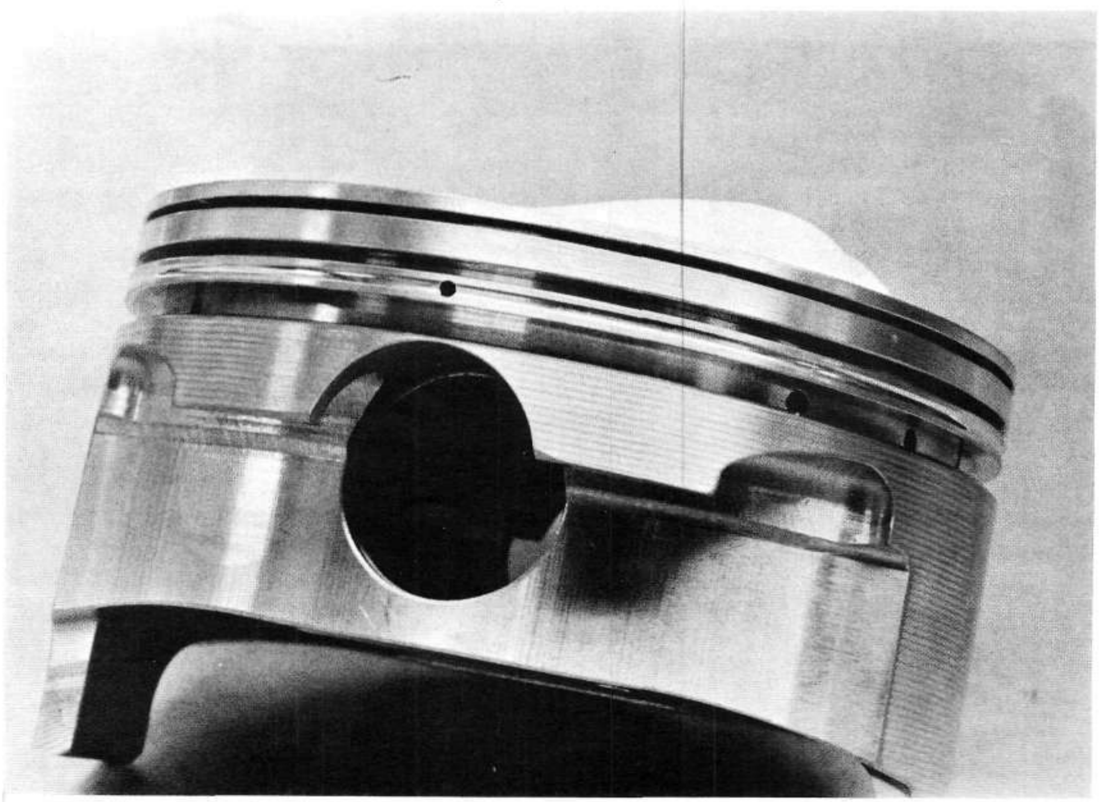
The Dykes ring was developed to resist this ring flutter problem but it was still a compromise to some degree. A fairly large vertical clearance is still required for the Dykes ring to function correctly and, even though it is a much lighter ring, flutter can still occur. When we set out to reduce this limitation it was obvious that some other means had to be found to apply pressure to the backside of the ring. It would then be possible to close the vertical clearance and eliminate, or at least greatly reduce, the flutter. The gas port was a natural solution. Drilling small holes right through the deck of the piston into the rear of the top ring groove provides the pressure needed for the ring to operate.

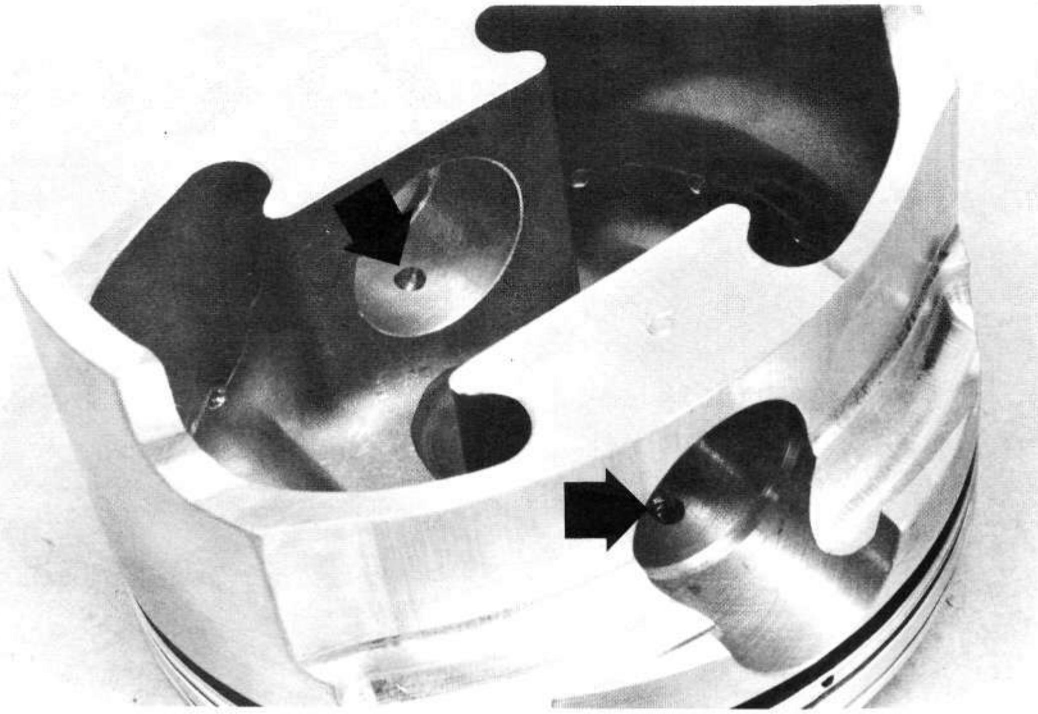


The top ring relies on the pressure of combustion for expansion sealing, and not on the static radial tension. When the vertical groove clearance is reduced to keep the ring from bouncing up and down in the groove, some method must be provided for the gas to pressurize the backside of the ring. The easiest and simplest method is to drill small gas ports through the deck of the piston to open into the back area of the top groove.

This technique has been used in small-bore motorcycle racing for many years but in large-bore automotive V8 engines the rings were supposed to heat up and stick in the bore. It never happened to us. Right from the beginning, in our engines, the gas ports worked like crazy, beyond our hopes. Even during full power 30-second loaded runs on the dyno we have

Oil ring functioning is sometimes misunderstood. It does not "scrape" oil off the cylinder walls. When a vacuum extraction crankcase system is used, the pressure differential across the oil ring will draw the oil from the walls. It is collected in the groove and pulled into the evacuation holes drilled in the back of the oil ring groove. We use this pressure differential to oil the pin bore in the piston. A groove is ground on the land between the bottom and second ring. A passage is drilled back into the piston boss and down to the top of pin bore. The crankcase vacuum will provide pressure oiling from the collection groove, down to the pin bore.





The pressure pin oiling is drawn off the ring land through a 0.075-inch drilling. A connecting 0.125-inch passage channels the oil down to the top of the pin bore.

never found subsequent signs of distress or blow-by increase.

We usually cut our own top ring grooves. Depending upon the rod length, we may cut as much as 0.115-inch off the decks of some pistons. Until this is finished in our machine shop we can't properly locate the top ring groove. If we use special order BRC's, often we have them cut the grooves. The general configuration is established and we don't have to shave the decks down, except in rare cases where the most we may have to remove is 0.030- to 0.040-inch.

The drag pistons look something like this. The top groove is 0.060- to 0.080-inch down from the deck, and is never down more than 0.100-inch. Between the deck and the top of the groove the piston circumference is reduced 0.038- to 0.040-inch from bore size. This clearance may be smaller if the distance from the top of the skirt to the deck of the piston is very short. The groove is cut for a 0.043-inch wide ring. The vertical clearance is about 0.0006- to 0.001-inch. Experienced builders will recognize this is an extremely tight fit. The standard racing forging normally is supplied with 0.0015- to 0.0025-inch vertical ring clearance. Some standard OE pistons call for as much as 0.002- to 0.003-inch clearance.

To compensate for the difference between conventional 0.002-inch clearance and the 0.001-inch or less that we run, we drill 13, 14 or 15 balance holes, each 0.040- to 0.043-inch in diameter, in the head of the piston. The holes open into the rearmost area of the top groove. The drilling should not cut into the back wall and *must not cut into the floor of the groove.* It is a straight-cut groove, with the back perpendicular to the top and bottom. The top and bottom are "flat," perpendicular to the cylinder wall.

Ring back clearance is extremely important to insure proper ring functioning, but in our particular case it is doubly important. We like to see the ring face 0.000- to 0.004-inch below the land surface when the ring is fully depressed in the groove. We would like to hold it near 0.002-inch but this can be difficult. If too much back clearance is allowed, the ring will be sluggish to respond as gas pressurizes the backside. With less space the ring will expand or relax much more quickly as the pressures vary inside the combustion chamber. Increasing the space beyond this limit will significantly increase the possibility of slight blow-by occurring across the face of the ring. This makes a measurable difference in the amount of ring life and the quality of the seal. The depth of the groove will also have some bearing on the strength of the top land. The deeper the groove, the weaker the top land will be, and the less resistant to "pulling."

Some sanctioning organizations take a dim view of drilling holes in the deck of the piston. Through some rather circuitous reasoning they contend this alters the shape of the piston top surface. Be that as it may, there are several other ways to provide auxiliary back pressure without drilling gas ports in the piston deck. They all work equally well but they are somewhat more difficult to achieve. One method is to machine a recess in the top ring land from which gas holes can be drilled on a diagonal downward toward the backside of the top groove. The height of the deck from the top of the groove may restrict this approach. Another method, used extensively by some racers, is to cut small grooves in the roof of the groove itself. It entails going into the groove with a little 0.040- to 0.050-inch end mill to cut half-round slots in eighteen or so positions around the circumference of the groove roof. The slots should be about 0.025- to 0.030-inch deep and extend all the way to the rear of the groove, but it is important that the mill not cut into the rear wall or cut into the floor of the groove. This approach cannot

Into the piston grooves we fit Sealed Power nodular iron rings. The top is a 0.043-inch wide ductile iron ring with a moly plasma-coated face. The second is also ductile and moly-coated, but is 1/16-inch wide. The bottom is either a ⅛-inch or 3/16-inch stainless expander with low tension SS-50U rails. End gaps are, from top to bottom: 0.014-inch, 0.010- to 0.012-inch and 0.025-inch.

be detected when the bore piston is observed from above, but it is a very tedious and difficult task.

Between the top and second groove the land clearance is reduced 0.032-inch from bore size. This, again, may vary but it really doesn't hurt to ride this land on the wall some. It is much less critical than the top land which must not rub the wall. The second land is cut on a slight taper, toward the bottom of the piston.

The second groove is cut for a 1/16-inch ring with a 0.0015- to 0.0025-inch vertical clearance. As we said, this ring is just along for the ride and acts as a safety barrier to catch the fire and compression if something happens to the top ring. It also transfers some of the heat out of the dome and acts as an oil scraper. We like to see about a minimum of 0.100-inch between the bottom of the top groove and the top of the second groove. The face depth here should also be 0.000- to 0.015-inch. This back clearance is not nearly as critical as the top groove.

The bottom groove is also fairly standard to accept either a 3/16-inch or ⅛-inch three-piece oil ring. We prefer the wider ring, where practicable, because it provides better oil control. The most unique or unusual feature at this point is the pin oiling system. An annular groove is ground completely around the piston on the third land. Above each of the pin bores a 0.075-inch hole is drilled on the same level as the groove, inward toward the area of the pin boss immediately above the pin bore. A 0.125-inch passage is then drilled up from the pin bore to connect with the first passage. This is the best way we know to oil the pins within the piston bosses. The pressure differential which we spoke of earlier is the fundamental force behind the system. A pressure differential will exist between the higher pressure above the third land and the significantly lower pressure in the crankcase. Oil scavenged from the walls and accumulated above the oil ring will be driven into the groove and around to the oil passages where the pressure differential draws it down to the pin bore. The groove and oiling passages can be clearly seen in the accompanying photos.

In the Grand National engines we felt there would be less advantage to using gas ports and narrow top grooves. The engine speeds are not nearly as severe as in the drag engines. We didn't feel, therefore, that ring flutter was a problem. However, all of our GN engines have used high rings, gas ports and close vertical clearance with surprising success. We move the top ring down slightly, 0.080- to 0.100-inch, from the deck. There has been no indication of pulling or "squashing" the lands as long as detonation doesn't occur. Detonation is the real problem with high ring placement. All you have to do is rattle the engine pretty hard just once and the thin top land lifts away from the ring. When running this setup you have to keep track of your "tune-up" very closely or you wind up changing the pistons a lot. We have been able to run some GN engines 800 miles without seeing any signs of wear. In fact, these pistons looked better than some drag pistons with four or five runs on them. We could find no signs of face wear or vertical groove wear.

The gas port system is largely an aid to ring life but we were very surprised to find some horsepower advantage in recent tests. It is hardly enough to measure (one or two percent) in a dyno engine. We think the stabilized top ring may be worth a tiny percentage. The real advantage probably comes from the thin width of the rings. This is difficult to explain or prove but we feel there is a definite power loss from wider rings. As the rings become more and more seated to the bore and the interacting surfaces grow smoother, there develops a "bond" through the oil film. This film begins to act as a glue, increasing frictional drag. As the surface area increases from break-in, the glue tendency raises and a very discernible power loss is observed. This is one reason, we believe, why an engine can have less leakage or blow-by after 25-30 runs than at 10 runs, yet when you rering the engine at this point it will produce more power. The new rings provide less ring-to-wall surface and the oil bond will have less of a tendency to adhere to each other. Our explanation is a bit simplistic but it's the best we can devise. This same condition has been observed at Chevrolet engineering and they don't have any better explanation than ours.

RINGS

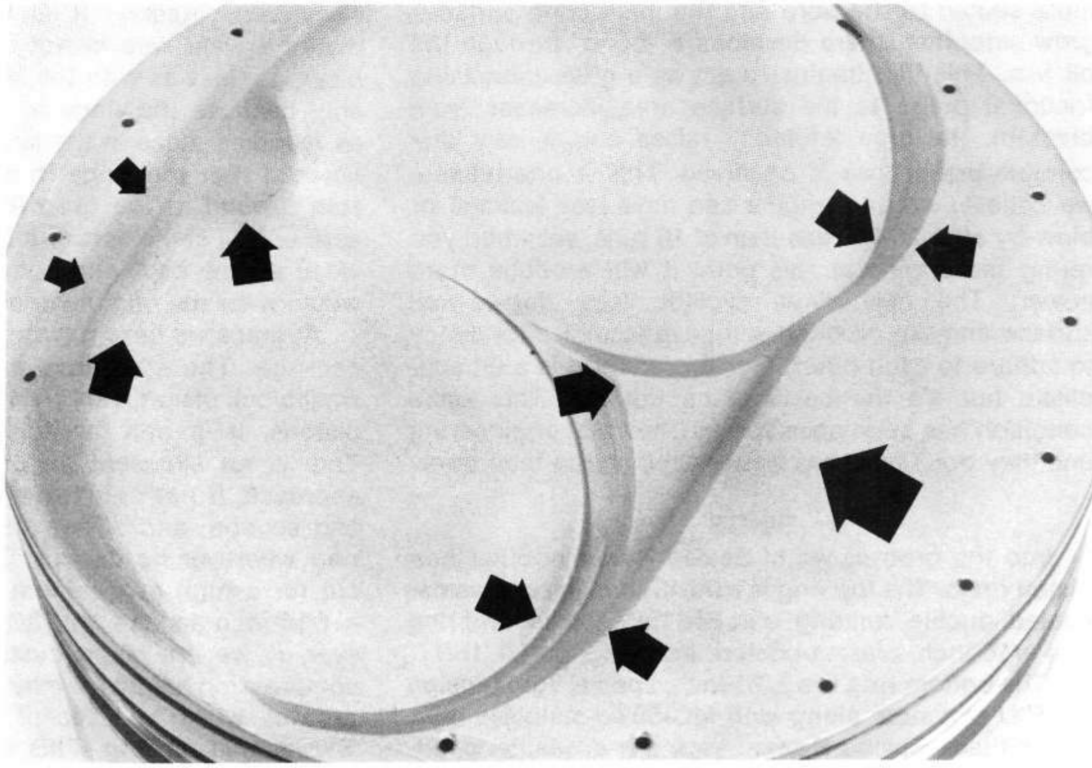
Into the grooves we fit Sealed Power nodular iron piston rings. The top ring is a 0.043-inch, moly plasma-coated, ductile iron ring, part BR-18P. The second ring is a 1/16-inch, plasma-coated, iron ring, part 1/16BT-10. The bottom ring is a 3/16-inch, special light tension SS-50U expander along with MD-50SU stainless rails. All of these Sealed Power rings are available in different sizes for bore diameters other than the 4.020-inch we prefer. As we have gotten into other methods of oil control (vacuum) we have steadily decreased the oil ring tension from the stock 15 pounds, with no adverse reactions.

When using this ring package we set the end gaps as follows: top—0.014-inch, second—0.010- to 0.012-inch and bottom — 0.025-inch (though end gap on the bottom ring really isn't crucial). It is absolutely essential when gapping the rings, especially the top and second ring, that a deck plate be mounted to the block with correct fasteners. It is dangerous to run end gaps this low in a block that has not been prepped with deck plates, and using the plates during the end gapping procedure helps pull the cylinder round in order that end gap measuring may be as precise as possible. If tight gaps are run in blocks which have not been bored and honed with deck plates, the rings will get up on the high spots at the top of the bore, dance around, and slam the ends together. This allows blow-by to occur across the face of the ring and the face gets hotter, further decreasing the end gap thermally. Instead of the ring being dampened by contact with the wall it will jump around and begin hammering itself to death. We have seen rings gapped at 0.020-inch and operated on walls which had not been prepped with plates, that looked like someone had been using a sledgehammer on the ends. We can run as tight as 0.014-inch with plate-prepped walls and they show no signs of trouble. We are not sure how much the gaps could be closed beyond this without incurring some interference. We have used some stainless rings as close as 0.009- to 0.010-inch without trouble, but we don't like stainless rings because they are not compatible with the aluminum material in the pistons. The aluminum material will wear the stainless away very quickly. It is worth mentioning briefly that the wear rate is not as severe with the smallblock engines as with the Mark engines. This is probably because the rings in a smallblock don't rotate as much as those in the larger bore engines. We discovered that the rings in a big engine would really spin around in the grooves with surprising velocity and, unless some sort of lubrication plating were provided on the back and bottom of the rings, the lands would wear the ring material away at a ferocious rate.

At times we have run some variations on this basic package. The standard Sealed Power semi-finished smallblock piston, part 7544PS+0.030, is similar to our pistons. It is only available in the 4.030 diameter. This is an excellent piston for the more practical approach. It has only two piston ring grooves, bottom and second, and allows the buyer to place the top ring wherever he desires. This piston can easily be cut for a high groove with the gas ports. It requires a 1/16-inch second and 3/16-inch bottom ring. However, if we are using custom forgings with a short compression height we may have to cut ⅛-inch bottom grooves which will accept the ⅛-inch Sealed Power low tension oil ring. This ring is very adequate with the vacuum oil control system.

In the instance when we use exceptionally long rods, such as the 6.25-inch rod 350 engines, we have used BRC pistons with a 0.043-inch, ⅛-inch combination, stacked very close. We have also used a two-ring TRW piston which has worked surprisingly well. We're still working on this setup, however, and cannot at this time recommend it unconditionally. The engines we have run to date have sealed up with no sign of gas passing the top ring seal. We don't feel it seals quite as well as our best 330 drag stuff, at least at this point. The TRW ring package was a 1/16-inch, moly-faced top ring with gas ports and tight clearance and a 3/16-inch, three-piece low tension oil ring. The annular groove and pin oiling system were also used and indications show there was enough pressure differential to drive the excess wall oil into the pin supply hole.

A few last words of advice before moving to another subject. We always use an unusable piston to work out the machining details of a new groove shape. Instead of destroying a good new piston we try out all the machining on something we can afford to throw away. It usually takes a few tries before the tooling is exactly right. Also, when you have an air cleaner over the carb(s), you can run the engine for quite some time with a bigger top land running lightly on the wall. On a race engine just a little dirt between the head and wall will dig the wall up terribly and cut the land to bits. This is less of a problem with a very skinny top land, but it is something to watch constantly and prevent as much as possible. Lastly, our experience has shown that a moly-filled ring will go back in and operate pretty well on an unhoned wall, whereas a moly plasma-coated/sprayed ring seems to require a hone "rough-up" whenever the engine is reringed.



Close-up shows the important features of the dome. The valve notches are plunge cut. The edge of the quench pad bordering the exhaust should be sharp, the intake edge should be radiused even more than can be seen here. The small peak between the plunge cuts is radiused. The peak of the dome is rolled over with a ¼-inch radius and a small ditch is cut in front of the plug. The backside of the dome opens smoothly to prevent secondary combustion formation along the ledge of the head casting. The quench pad is also cut on an angle, opening toward the dome to achieve any pumping gain outward from the quench area toward the chamber.

DECK AND DOME

Nearly all forged racing pistons are delivered from the manufacturer in semi-finished condition It is necessary to spend some time (often several hours) getting the piston to final configuration. The following steps are typical of a race prep for one of our high compression Pro Stock engines. This list shows the extent of our preparation and each step will be discussed in greater detail in the following text. If lower compression (less than 12:1) and a stock length rod is used for street or stock-type racing classes, the work is less extensive. First, the dome height and deck height must be finalized. In the particular instance of the TRW/Sealed Power forging we machine the piston deck away to lighten the piston and increase the height of the dome. This will also reduce the compression height of the piston. The main intent is to make the piston as light as possible. Cutting the deck is the easiest way to greatly reduce the amount of material in the piston. Once the deck is cut to the minimum, the compression height will be reduced such that a rod length of 5.9 to 5.92 inches will be required for our 3.25-inch stroke drag race engines. At this time we will also machine the underside of the forging, in the area of the pin bosses and backside of the dome, to reduce the weight further. On the topside we plunge mill the piston notches for correct clearance with the valves. A small ditch is cut across the dome to unshroud the plug. As required, the dome may be recontoured to assure it fits properly into the chamber. Then, the piston ring grooves are cut and other machine work for oil drainback or pin oiling is finished. Finally, the dome and deck are hand-finished to remove sharp corners and gain a smoothly-radiused contour. The deck and dome are bead blasted and a coating of VHT white heat-resistant paint is baked on with infrared heat lamps. Currently our pistons probably represent more work and time than any other parts in the engine, except the heads.

When the TRW/Sealed Power decks are lightened, about 0.115-inch can be safely removed. If a suitable valve notch is cut for a high lift Pro Stock cam this will only leave 0.090-inch of material in a very small area beneath the exhaust notch. This little spot doesn't seem to cause trouble but we really don't like to see small areas get any thinner than 0.090-inch. The minimum overall average deck should be 0.200-inch. It certainly is possible to live with the decks as thin as 0.180-inch in a drag race. However, if you rattle the engine hard, they will probably give up. Depending upon how much material has been cut from the top of the deck, as dictated by the compression height, we may also go up inside bottom with a ball mill to cut away material in an effort to lighten the piston. This must be done carefully because of the thin area beneath the valve notch. It is important when cutting on the pistons to insure the machine tools leave radiused corners and do not form rough spots where stress risers might begin. With any piston some careful measuring will be needed to insure the decks are adequately thick.

We have an ace up our sleeve, too, with the head gasket setup. By using different combinations we can space the heads further away from the block or down closer as the situation may demand. This flexibility is very helpful when sorting out the details of deck height and piston-to-head clearance. In the drag engines with aluminum rods we shoot for about 0.060- to

0.070-inch static clearance between the deck of the piston and the quench area of the head chamber. With steel rods and lower engine speeds, such as typically observed in GN engines, this static clearance can be reduced to about 0.050- to 0.060-inch. In a low speed street engine with steel rods this quench clearance can be closed to almost contact the head with the piston. Rod stretch usually isn't a problem in a 6000 rpm engine. However, there should always be at least 0.010-inch high-speed running clearance in the quench area in order to allow for carbon buildup. In the racing engines our testing has shown the aluminum rods will stretch enough to close a 0.050-inch gap and give a 0.000 (net) running piston-to-head clearance, with our particular pistons and rods. Therefore, we always run at least 0.060-inch static to allow the extra 0.010-inch margin beyond "smacking clearance."

We also believe there may be a slight "pumping gain" by leaving some clearance across the quench pad. Test results from outside sources indicate that power can be gained by providing quench clearance, up to a certain point. Of course, all this is dependent upon the c.r. requirements. This gain is probably due to the greater ease with which residual fluid gases can be pulled out and pumped back into the quench as the piston changes direction. The indications seem to be that this pumping gain is increased as the running quench clearance is increased, all the way up to about 0.030-inch. We know there is definitely horsepower to be gained by using the 0.010-inch running clearance over a completely closed-off quench, despite the loss of compression ratio. We have operated engines with as much as 0.070-inch static clearance at the quench and haven't seen any power loss. We have not gone beyond this figure and have no idea how power would be affected with more clearance.

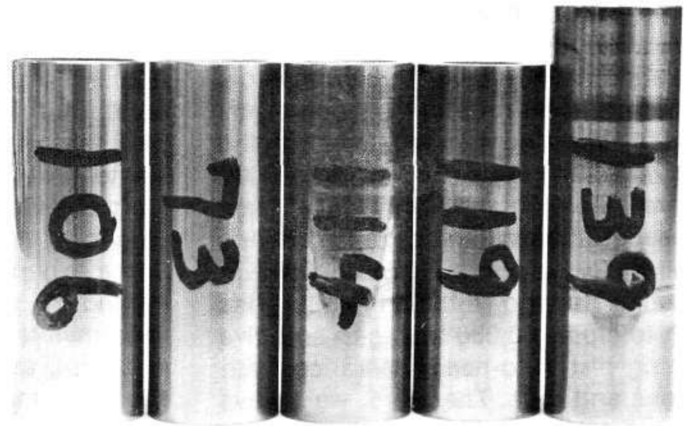
Up on the peak of the dome projection we fiddle with the lump to give plenty of clearance across the top and around the sides so the cylinders have room to breathe with the piston at TDC. This always winds up as a trade off. If we leave a lot of the lump, it will hinder breathing, especially exhaust breathing, with the piston high in the chamber. If we cut the dome down to open the chamber we naturally lose some of the compression battle. We cut away as much as possible without reducing the c.r. beyond our desired limits. In the Grand National engines we shoot for 12.5:1 compression. In the drag engines we usually achieve a measured 13 or 13.3:1. These are about the optimums. We have used engines with c.r. as high as 13.5:1 and, though it hasn't hurt the elapsed times in the drag racer, our tests have shown there is a definite power loss. We're not sure at this point if this is an octane requirement problem or something to do with breathing restriction.

When working the plug side of the dome it is desirable to get the piston as tight against the low side of the chamber as possible as long as there aren't any blocked-off pockets created where secondary combustion can develop. Other than this, there isn't any problem with pushing the dome up quite tight under the plug. As far as we can tell, there is no gain or loss in swirl flow with the piston lump up high under the plug and it's possible to all but hit the head here. Across the face of the dome and down toward the bottom of the front-side most conventional racing pistons do a pretty good job of filling the chamber. We like to see about a ¼-inch radius across the top of the lump and we cut a very small ditch right in front of the plug to let the flame spread away from the plug rapidly. Some experts insist there should be ¾-inch clearance around the plug. This is almost impossible to achieve and maintain the required compression ratio. A small cut, as can be seen in the photo, is more than adequate and it won't hurt the compression situation. The ditch must be radiused smoothly.

To help the compression picture we plunge spot cut the valve notches so they tightly fit the valve heads. If you don't need the compression, a fly cutter can be used. We don't think there is any advantage to the channel notch which opens the area between the valves. It doesn't hurt if you can stand the compression loss. With the individual pockets it is important to radius the little peak between the notches. Where the notch lip meets the flat quench, we radius the lip at the edge of the intake valve notch and leave the lip with an abrupt angle at the edge of the exhaust valve notch (see photos). When the piston is changing direction at the top of dead stroke it will tend to draw some flow back out of the exhaust port. If the quench edge is radiused this backflow will be promoted. Therefore, a sharp edge is desired to reduce this action. Conversely, a radiused quench edge is desirable around the intake valve to gain better flow into the chamber. Anything in the road will cut flow capacity as the piston starts downward because the valve is dropping into the piston pocket, following the piston as it recedes. A sharp edge along the quench pad will cause a great deal of disturbance as the incoming flow passes across the back of the valve and along the roof of the chamber.

To determine intake valve-to-piston clearance the engine is mock-assembled with light valve springs and positioned with the piston at 8° (crank) ATDC. At this point the intake valve (properly lashed) is theoretically at the nearest point to the piston. By opening the valve against the light spring it is possible to measure exact minimum valve-to-piston clearance. The valve pocket depth must be increased to gain the desired minimum dimension if adequate clearance is not present. We make certain the intake VTP clearance is at least equal to the piston-to-head clearance. With aluminum rods and the intake valve properly lashed this figure should be in the neighborhood of 0.055- to 0.060-inch. This is an absolute minimum to allow for rod stretch. The exhaust valve is checked in the same manner with the piston at 8° (crank) BTDC. Minimum allowable at this point is 0.085- to 0.090-inch. If the valve gear is over-reved, the exhaust valve may bounce upon return to the seat, so the added margin

Here is a sampling of piston pins which may be selected for the little engine. From left to right we have: a 2.5-inch BRC tapered wall pin weighing 106 grams; a 2.5-inch, 0.090-inch wall, 73 gram, H-11 BRC; a 2.5-inch, 0.140-inch wall, 114 grams, BRC H-11; a 2.5-inch, 0.160-inch wall, 119 gram Chevy stock pin; and a 2.90-inch, 0.160-inch wall, 139 gram Chevy stocker. For most purposes we use the 0.140-inch wall, 114 gram, BRC H-11 pin.



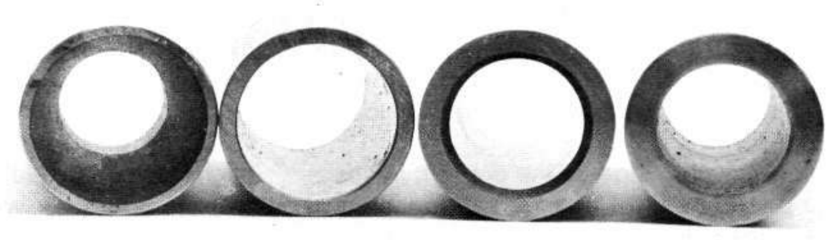
of clearance is insurance. The exhaust valve stem will also expand quite a bit more because of the greater heat induced in the exhaust valve. Under certain engine speed and load conditions it is conceivable that the valve lash will be reduced considerably, reducing the VTP clearance. This is possible even if the valves have been adjusted with the engine hot and running at idle. However, nobody knows if this is actually true or not. Under controlled conditions it is possible to run the exhaust valves closer, maybe as tight as 0.075-inch, but this is very touchy in a race engine.

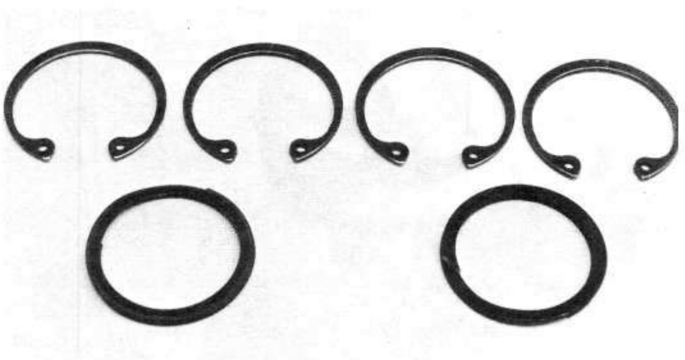
When all the handwork is completed, we tape off the barrel and ring land area of the piston and bead blast the dome surface. It is important that the ring lands be carefully protected during this procedure. If the grooves are blasted even slightly, the rough surface texture imparted by the bead shot will destroy the essential ring seal which must be created between the bottom surface of the ring and the smooth lower surface of the ring groove.

The last procedure is an application of Sperex VHT white paint to the dome surface followed by a 8-10 hour heat bake with infrared lamps. This is, more or less, a shield to provide faster heat buildup when the car is leaving the line. On the dynamometer we haven't found any power increase from this technique after the heat has stabilized. We know that an engine which has been assembled with bright, shiny, new combustion chambers requires a different "tune-up" setting until the chambers have become coated with a fine carbon layer. The VHT paint acts similar to a thin carbon layer and helps promote carbon buildup. Without using the paint it takes 6-10 hard passes before we can get back to our race tune-up. By using the paint we can go straight from the shop to the strip with a brand new motor, fire the engine, and hit the first pass with our race tune-up. The carbon buildup, incidentally, is very important in a drag engine. It acts as a heat dam or absorption layer inside the chamber and on the head of the piston. During operation it reaches glowing-hot temperatures and assists heat buildup in the chambers during initial full throttle application — quickening response in the first few, ultra-critical feet of the race. We use the same VHT coating on the inside surfaces of the big-block aluminum head chambers for the same reason. There are other commercially-available coatings which are designed to do the same thing. We haven't tried any of them. The VHT does an adequate job it is inexpensive, and it's very easy to apply.

The coating also helps considerably to read the domes of the piston because of the rapid carbon depositing. When the engine is running properly there should be a smooth gray-black deposit across the entire piston dome. The deposits should extend all the way to the edges where some light oil scum carbon should be visible.

End view of the pins shows structural differences. The pin on left is the light BRC tapered wall, next is the 0.090-inch wall BRC, then the 0.140-inch wall BRC we generally use and on the right is a 0.160-inch wall stock pin.





To lock the floating pins in place we use either a pair of double Tru Arc snap rings per side or one of the double-wrap 0.075-inch thick Chevy Vega Spirolocs on each end.

PINS

We usually look for the lightest pin available which will not crack or pull out of shape at the intended engine speed or length of usage. Reducing pin (and piston) weight to the bare minimum is very important. All this weight is transferred down through the connecting rod to the crankshaft. In the process it stretches the connecting rod, creating fatigue, and it bends the crank, producing lost heat energy and disturbing the engine balance. For drag racing and dyno testing we use BRC pins. In all cases, the pin diameter is 0.9272-inch, the standard Chevrolet size. The drag pin length is 2.5 inches and it is constructed from H-11 steel material. Wall thickness is 0.140-inch. This gives us a fairly light but strong pin, weighing about 120 grams. We have never failed one of these pins in a drag race engine or on the dynamometer. With this pin and a reworked piston we normally have a total of 580 to 600 grams hanging on the small end of the rod in a drag engine. In the long-distance endurance engines which run at constant high speeds we prefer a stock Chevy, heavy duty, 2.9-inch long, 0.9272-inch diameter, 145 gram pin.

We have tried experimental thinwall pins which weigh as little as 70 grams although we feel they may be a little dangerous and they don't suit our needs. About two years ago we were using such pins in the race engines without apparent trouble. However, when using them on the dyno, we began experiencing trouble with them "squashing." We found that the pin would pull oblong at the point where it passes through the connecting rod. The assembly would then lock up and rip the top off the rod. We became nervous about their reliability in the race engines and, since it was a hassle to change the pins whenever we wished to pull an engine out of the car to test it on the dyno, we discontinued use of the 70 gram pins.

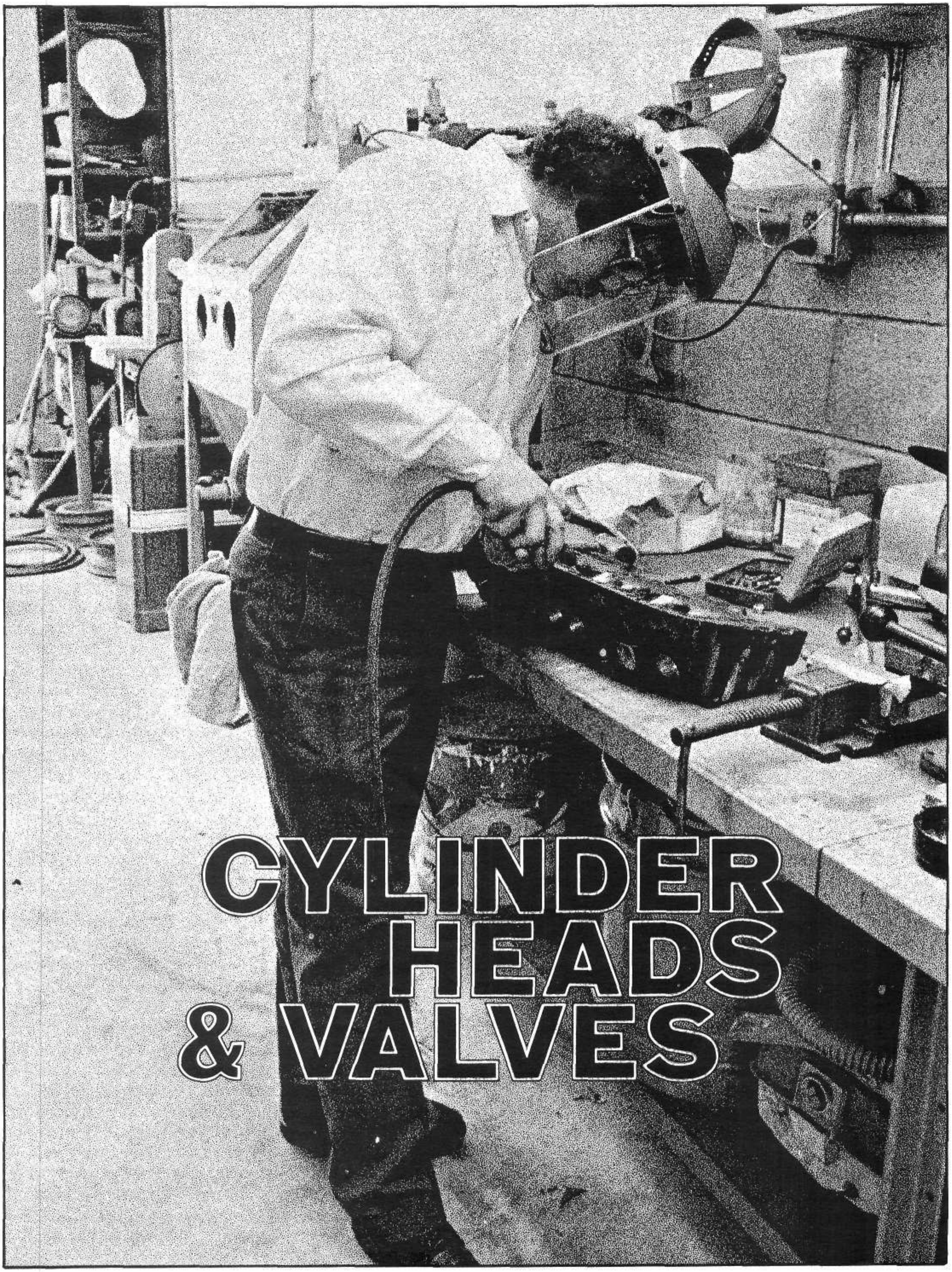
Unfortunately, the H-11 steel material isn't too compatible with aluminum and the small ends of our drag rods will wear out. Chrome plating the pins would add considerably to the life of the rod but we don't like chrome pins. Our experience has proven them to be unreliable. Cracking is always a problem when chrome plating is used and we find that cracks which originate in the chrome will often pass on into the subsurface material. As far as we know there aren't any chrome pins which can be considered safe. The steel pins aren't a real problem anyway. Considering the frequency with which we change the rods, pinhole wear does not become a factor. If someone were to stretch the time between rod changes they would be wise to check the little ends carefully, for elongation.

We used the TRW/Speed Pro pistons with short, 2.5-inch pins by cutting some material away from the outer sides of the bosses and machining new pin retainer grooves. This works fine with the BRC pin but we do not recommend shortening the standard Chevy high performance full-floating pin. In our shop we have successfully taken 0.050-inch off these pins by carefully grinding them while using a good supply of coolant to keep heat buildup under control. We have found out that if this is not done correctly, or even if you just happen to be unlucky, the pin will eventually shatter. These are case-hardened pins and incorrectly shortening them will lead to internal cracking which always emanates from the end of the pin which was shortened. We don't even consider this anymore and don't recommend it.

LOCKS

To hold the pins in place we use double locks. In the case of the drag engines we install two Tru-Arcs per side, always making certain the "flat" side of the stamping is against the retaining groove in the piston. This causes the sharp edge of the lock to dig into the groove, preventing it from bouncing out. As mentioned earlier, we have had considerable problem with piston side loading and natural frequency pounding the retainers out of the pistons in the GN engines. We have finally counteracted the trouble by using the side-tab extensions and installing the double-wrap, 0.072-inch thick, Cosworth Vega Spirolocs in the retaining grooves. After installation we cut the tails off so they cannot lead out of the grooves. It's nearly impossible to get these retainers out once they are in place. The end play should be about 0.003- to 0.006-inch, or as tight as possible without risking the possibility of them "sticking" when the assembly is cold started.

The radial clearance of the pin to the rods and pistons is usually about 0.0006- to 0.001-inch with the floating pins. This works well with the pressure oiling from the groove between the bottom and second ring. Inside the rods, the pins are oiled from a single hole in the top of aluminum rods or from below with two holes in steel rods, as described earlier in the connecting rod chapter. We have also used an additional pin oiling hole in the oil collection groove, next to the passages, which supplies oil to the pin bores in the piston. The second hole is drilled completely through to the inside surface of the piston. The pressure differential will force oil into the hole where it sprays out into the area directly above the connecting rod small end. This system works well with a single hole drilled in the top of the small end and has proved successful with full-floating GN rods which have not been bushed.



CYLINDER HEADS & VALVES

In the next few pages we will discuss some of the most important information in this book. There is absolutely no doubt that the current power levels developed in Pro Stock engines and, to a degree, in Grand National engines is the result of recent cylinder head developments. We feel this is the core of modern racing engine preparation. There's more to be gained by correctly working the heads than by any other single thing you can do to the engine. Between the very best short block preparation and fairly shabby work there is no more than 15-20 horsepower difference. On the other hand, when you take a pair of out-of-the-box smallblock heads and finish them up correctly, you're fooling around with about 100 horsepower! Most of this work is subtle in nature and difficult to understand. At the same time it is some of the most difficult technical material to describe in words or pictures.

Currently, we feel it is very tough, if not impossible, for an unknowing person to attain acceptable cylinder head performance unless he has access to a flow bench or at least has a very close working relationship with one of the reputable specialty shops that base their work on thorough flow studies. It is easy to spend a lot of money on a set of "professionally prepared" cylinder heads that may not be worth a damn! To separate the meaningful work from cosmetic "eye gloss," you must know what to look for. On the other hand, a good pair of Pro Stock heads is probably the best horsepower buy, per dollar, on the market.

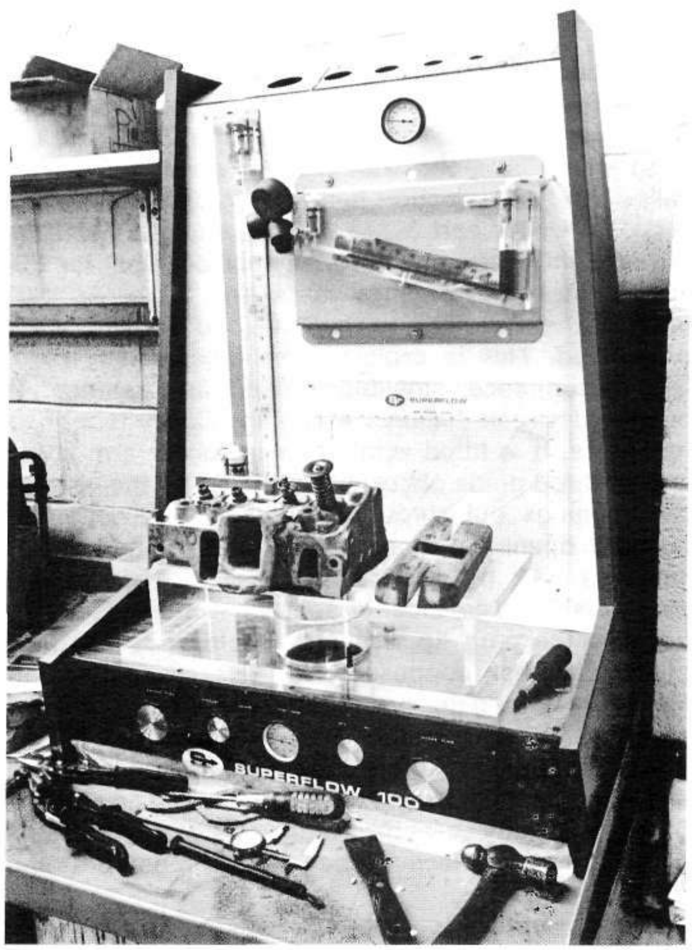
Our basic castings are initially prepared by Air Flow Research in Van Nuys, California, and we pay a pretty penny for them without regret. We feel it's money well-spent because they know what to do and can finish the time-consuming foundation work. After the heads get to our shop we spend an additional 40-50 man-hours on each pair before they are ready to go on a race engine. That totals a lot of money and work, but it's essential to be competitive. In our entire shop the only two items we consider irreplaceable are some early 292 cylinder heads which work better than anything else we have. They do everything right, and we will do anything possible to keep them in service as they simply cannot be replaced or duplicated at this point. But we're getting ahead of ourselves; let's go back to the beginning.

SELECTING CYLINDER HEADS

In discussing the various Chevrolet cylinder heads we use a unique reference system. In our specialized terminology we identify a basic casting by the last three digits of the *casting number* rather than by using the more cumbersome and less specific GM part number. For instance, the early Chevy fuel-injection heads were all based on casting number 3782461. We refer to it as the "461" head. *This always indicates the same basic chamber and port configuration,* though in various years between 1961 and 1967 this casting was fitted with different size valves and spring packages. In each of these cases a separate part number was assigned to the complete head assembly.

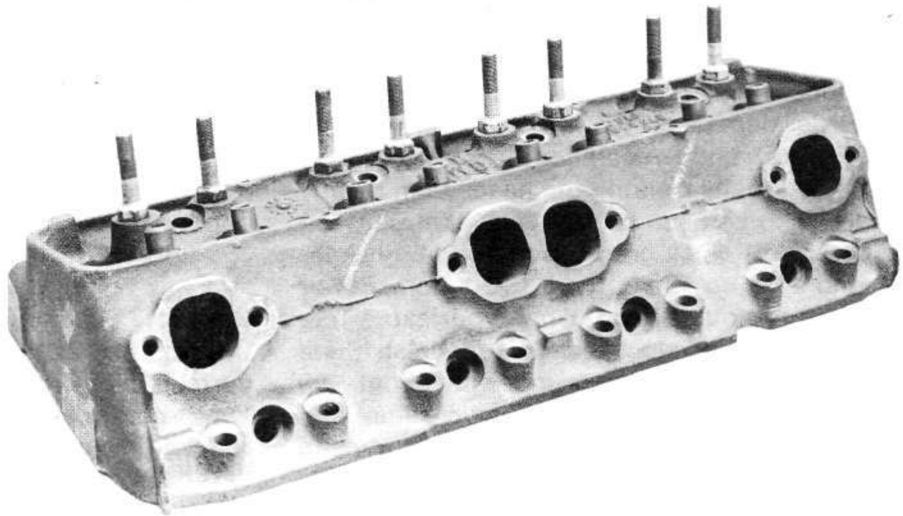
The 461 casting is most often sought for use with the 1.94-inch intake and 1.50-inch exhaust valves, used in the 275 hp and 315 hp, 283 engines; and in the 300 hp, 340 hp and 360 hp, 327 engines. In later years it was fitted with the 2.02-inch intakes and 1.60-inch exhausts to use with the 350 hp, 365 hp and 375 hp 327's. In these engines the valve size was enlarged, but the port dimensions remained the same. They are, in fact, the same size ports as used on the Z/28, LT-1 and all the way through the current L-82 engines. These heads were certainly a large contributing factor to the legendary street reputation earned by the smallblock engine.

This casting is still used by builders of small displacement Modified Production engines because the port, chamber and small valves are compatible with the 3⅞-inch bore engines. The chamber is not so large that it hangs over the bore wall, forming a sharp ledge to restrict airflow around the heads of the valves. The spring seats can be machined to accept



The current power levels in Pro Stock racing and Nascar circle track racing are the result of extensive improvements in cylinder head flow. To be competitive in any racing class where cylinder head modifications are legal, flow bench development is an absolute must. In most cases, the cylinder head and port prep will be the difference between success and failure.

When cylinder head selection is not restricted by the sanctioning rules, Chevrolet cylinder head number 3965784, based on casting number 340292, is the best head to use. This is the newly-revised off-road casting that is often *incorrectly* called the "Turbo Head." This casting has very little in common with the non-quench chamber turbo-charged smallblock cylinder head and we do not use this misnomer. To differentiate this casting from the earlier high performance head, we simply call it the "292 head," from the last three digits of the casting number.

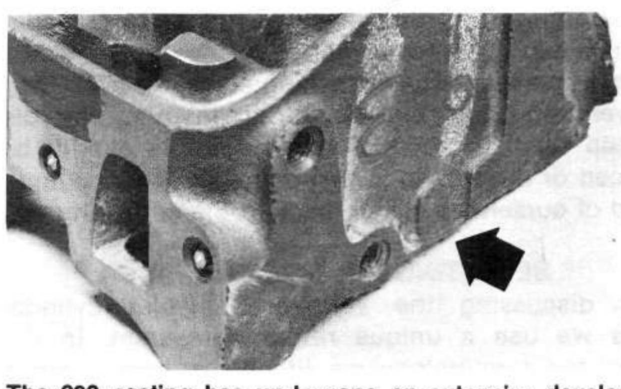


big diameter valve springs without much fear of breaking into the water jacket (a problem with later heads). These combustion chambers do not have the cast-in "carbon crunch" pockets as do the later pieces, thereby helping the compression picture. We don't currently use any of these casting in our race engines but they work very well in small engines, displacing up to about 287 inches, or thereabouts.

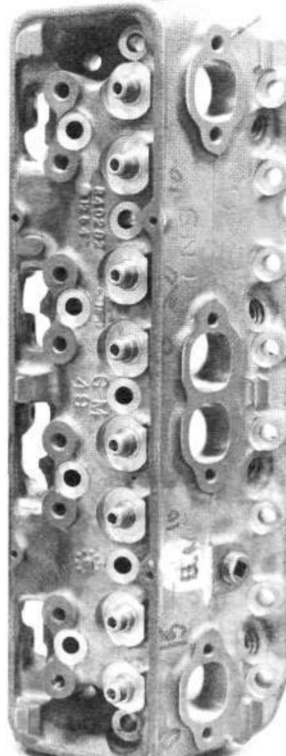
Another number which is important to many racers is casting 3991492. This is the Z/28 or LT-1 head which is, along with the 461 head, sometimes called the "fuel-injection head." This casting has two versions. Under part 3987376 this casting is listed as the *straight spark plug* "service replacement" for the special high performance 302's and 350's. As part 336746 it was revised to the "off-road" *angled spark plug* head. This is probably the most widely-used, high performance, smallblock head in existence. It has the desirable features which any Chevy racer can recognize. It is fitted with screw-in rocker arm studs and pushrod guide plates, necessitated by the ventilator openings cut through the heads, between the pushrod openings. In all cases it is machined to accept the 2.02 by 1.60 valve package. The angled-plug version is considered somewhat more desirable because the plug tip was moved closer to the roof of the chamber, supposedly to increase combustion efficiency and gain power. In published Chevrolet reports this modification was worth 10-12 horsepower, though details of the engine configuration and speed range were never specified. The current belief is that these angled-plug, service replacement heads are most effective when very high compression ratio is desired.

Early samples of the 492 casting in both the straight- and angled-plug version had one notorious shortcoming. Many problems occurred when racers attempted to machine the spring seats for larger diameter valve springs. Apparently the water core had been enlarged by removing some metal from the area immediately to the outside of the outboard spring seats. Cutting too large a spring pocket would result in breaking through to the water at this point. We have heard that the latest angled-plug castings have been modified to rectify this condition. They can be machined for large springs without trouble. It must be noted, however, that this operation should always be approached with care or avoided, unless the specific requirements of the racing program demand the larger springs.

For street applications the 492 should be very adequate with any sort of reasonable single four-barrel induction. With racing dual-quad inductions, we think it would be limited to smaller engines in the 283 to 287 c.i. range. At this displacement the ports would probably not have to be enlarged very much. In stock form they have about as much volume or maybe a little more than these small engines will need. This will help flow enough to allow a little less cam. It should also help balance displacement to rod ratio with the small runner/high velocity intake manifolds that the engine would prefer. Of



The 292 casting has undergone an extensive development since the first pieces were cast in 1971. It shares the common features of other high performance heads, but it can be distinguished most easily by the complete removal of the heat crossover passage from between the two intake port pairs. This new one-bump designator appeared on the later castings and is now common on all 292 castings. Earlier pieces did not have this designator and can be identified by the casting number and/or casting group date on the underside of the intake runner.

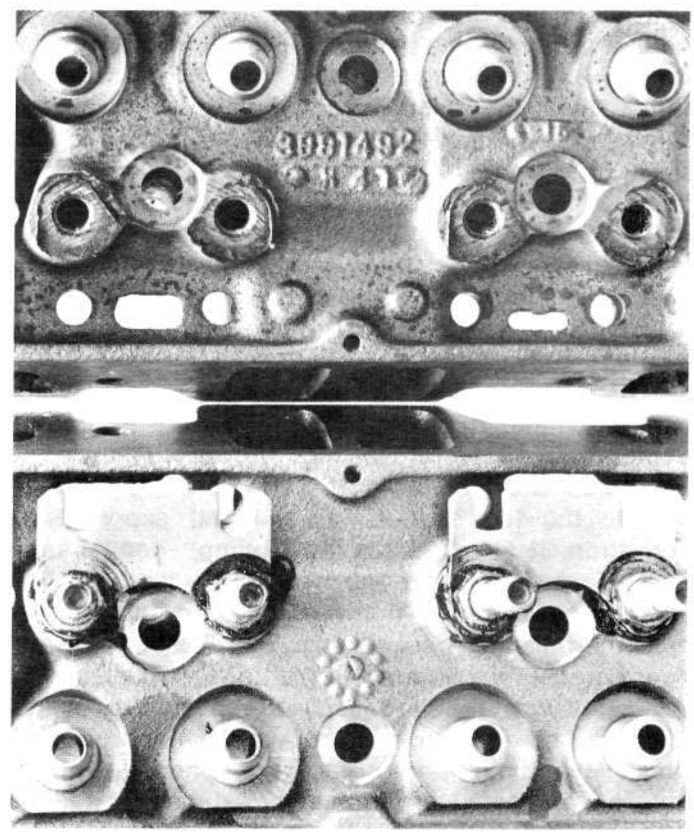


On the top side the 292 head has provisions for the screw-in rocker arm studs, the spring seats have additional metal allowing more machining for large-diameter valve springs, and the pushrod/ventilator openings are extremely large. They provide a potential performance increase over all other smallblock heads because they have larger intake ports and can be ported to a larger volume than any other casting.

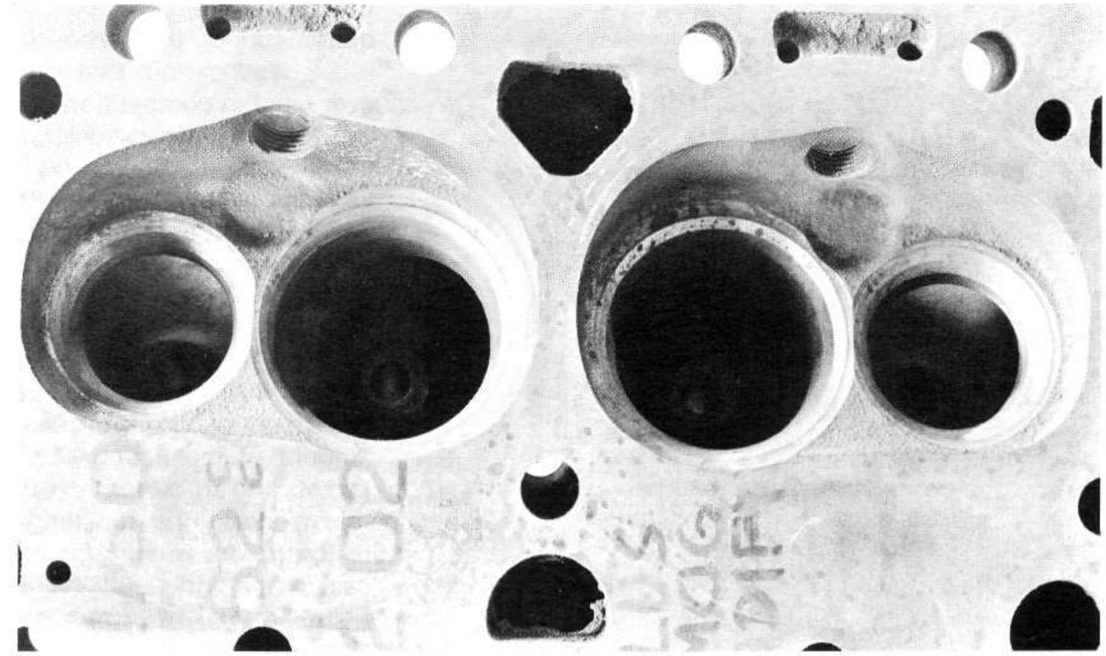
course, this will ultimately depend upon the other variables discussed in the connecting rod chapter. In all cases, the pistom dome must be carefully contoured to prevent secondary combustion trouble with the carbon crunch pocket. Some cylinder head shops fill these pockets with weld material, but we think this may be a rather expensive solution compared to simply selecting a properly-designed piston.

This brings us to the currently accepted race piece, the "292" head. It is available as a cylinder head assembly under part 3965784, based upon cylinder head casting 340292. This casting has been erroneously dubbed "the turbo head." As far as we can tell, the currently available 292 casting has very little in common with the head castings developed for the fuel-injected, turbo-charged, alcohol, smallblock engine, studied briefly for experimental purposes in about 1971. We believe that about 20 of these genuine tubo heads were built, and none were released for use outside of Chevrolet engineering, except to those sources which were contracted directly for the turbo-charged smallblock study. These experimental pieces had ports of the same size as the 292 casting and shared the other special features of the conventional high compression head, except that the combustion chamber was of a non-quenched design to accommodate the low compression combustion requirements of the turbo-charged engine. We are almost dead certain that none of these castings were ever used in competition drag or GN engines.

The first experimental 292 off-road/racing heads were cast in September of 1971. To date, five versions of this head have been built. The differences between these castings are very minor. We know the following general features are readily attributed to the 292 head, while the specific variations are listed in the discussion of each individual casting group. The as-cast intake ports were larger than the 461 or 492 heads. To allow this enlarged volume, the floor of the port was lowered, reducing the amount of metal between the runner and the water jacket, and in current production heads the roof was raised along the full length from the port opening to the guide. In early prototypes the ports were also cast much wider. On the exhaust side, a very significant improvement was made by pulling the backside wall, in the pocket area immediately above the valve, in toward the valve guide, to gain better high-lift exhaust flow. To facilitate factory machine requirements, the divider wall separating the siamesed intake ports was enlarged to 0.24-inch, over the more desirable 0.18-inch dimension. It is possible to cut back the wall in each port to gain the desired thickness (see the *Chevrolet Power* manual for details).



This close-up comparison of the topsides of a 292 head (below) and a 492 head shows the enlarged spring seats on the stock 292 casting. They can be remachined to take the largest spring that can be fitted next to the head bolt hole (provided there is very little core shift).



The stock 292 combustion chamber measures approximately 66 cubic centimeters. It is virtually unchanged from the 492 head except that on later models the inlet valve unshrouding cutter diameter was reduced to 2.34-inches. This reduces cylinder bore overhang and leaves more material on the backwall of the chamber. It does not affect air flow. Valve spacing remains the same as on early heads.

Other minor changes include the complete elimination of the heat cross-over passage and the enlargement of the ventilator between the intake ports, in order to improve lower case venting and pushrod clearance (varies on some casting groups). On the topside of the casting the rocker stud bosses were enlarged to facilitate offset relocation of the studs. The valve spring pocket was enlarged from 1.28 inches to 1.44 inches for the same reason. Additional metal was put into the water jacket-side, immediately around the pocket, to allow even larger spring seats to be machined, if desired. Inside the combustion chamber a cast relief was added to the side of the angled-plug hole to give an even thread break-out for the plug tip. In the very latest castings, those currently available as part 3965784, the inlet valve unshrouding cutter was reduced from 2.40 inches to 2.34 inches. Otherwise, the chambers remain virtually identical to the conventional quench-type 461/492 castings.

The 292 head has had a rather interesting evolution. A limited number of castings were poured in September, 1971, for testing and evaluation. The '71 batch did not have the 292 casting number, despite the fact that it was the first experimental derivation. It still had the "two bump" identification mark on each end of the head. This was the first attempt to increase porting capacity, but factory machine-finishing caused problems. We understand considerable welding and additional work around the seats was required to bring them into shape. The pushrod and ventilator cut was not incorporated, thus allowing the port entry to be widened, an important advantage. Unfortunately, extensive welding was required deep inside the ports to bring the flow number to a respectable range, considerably negating the advantage of the enlarged entry.

The next batch, produced in February, 1973, was virtually identical to the '71 batch, except some minor changes had been made, based on the information gained from work with the first pieces. The two-bump identifier remained. Only about 20 of these pieces were molded.

After more testing and evaluation with this group of pieces, another run was made in June of 1973. *These are absolutely the best smallblock Chevrolet cylinder heads ever sold.* At least that's our opinion. These are the heads we cannot replace or duplicate and, needless to say, these pieces are very dear to us. It is extremely sad that only about 100 of these castings were released. We believe that most of them were used on Nascar engines and subsequently destroyed. They were immediately put into extra heavy duty service and the strain took its toll. The constant loading of the GN engines showed that certain section thicknesses had been reduced a little too much, and a characteristic crack would develop across the chamber roof, below the plug, and into the internal surfaces of the casting. Many of the heads were then scrapped as useless but, for drag racing purposes, they would still be fully acceptable. Despite the fact that ours have been hammered many times, we continue to weld them as necessary and they work just fine.

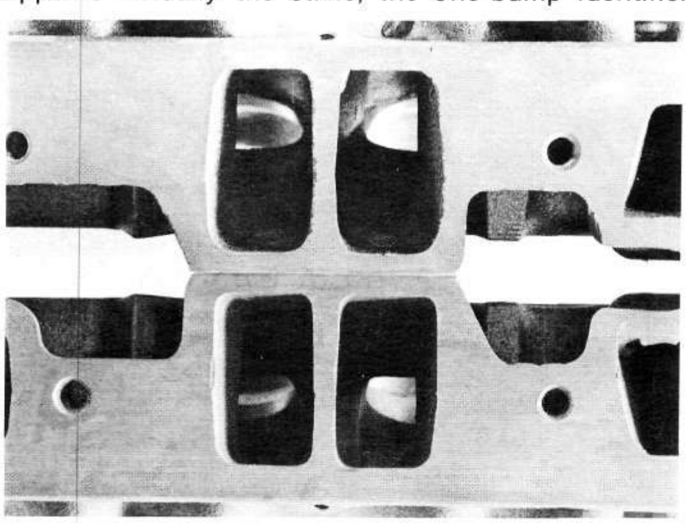
The June heads had somewhat poorer intake and exhaust ports, from a shaping standpoint, but they were *BIG*. The longside wall was moved over far enough to make a notable improvement, but the shortside floor corners had to be filled with weld or Devcon to get the proper approach angle to the valve hole. Some welding was also required to get the intake port entry as wide as we like. We really believe this is where the "welded head" work began to evolve into the preposterously complicated work we are now seeing. With this work and a fullhouse port job, these intakes would flow like gangbusters practically throughout the valve lift range, up to 0.750-inch lift. The exhaust floors were also dropped too far in the original casting and they had to be built up with weld to get the exit numbers where they belong. This batch of casting could be identified by the 340292 casting number, which emerged for the first time on these parts. The June '73 date appeared directly below the casting number, though the two-bump identifier was still used.

Following the June batch a relatively large number of castings, about 2000, was produced in November, 1973. Though the June '73 castings were available to anyone aware they were for sale and willing to pay the price, it was the November group that became widely available. These heads still had most of the porting capacity of the earlier casting groups, but from this point on the porting capacity was reduced because of water core changes which removed metal from the waterside port surfaces. These ports were not as good as the June ports because you cannot move the longside intake port wall (the exhaust-side wall) quite as far. This results in a small, but significant, volume reduction. For GN use these heads and the current group are better because the cracking problem has been greatly reduced. It is still somewhat troublesome but, through alloy specification changes and the core plug alterations mentioned above, the trouble has been significantly reduced.

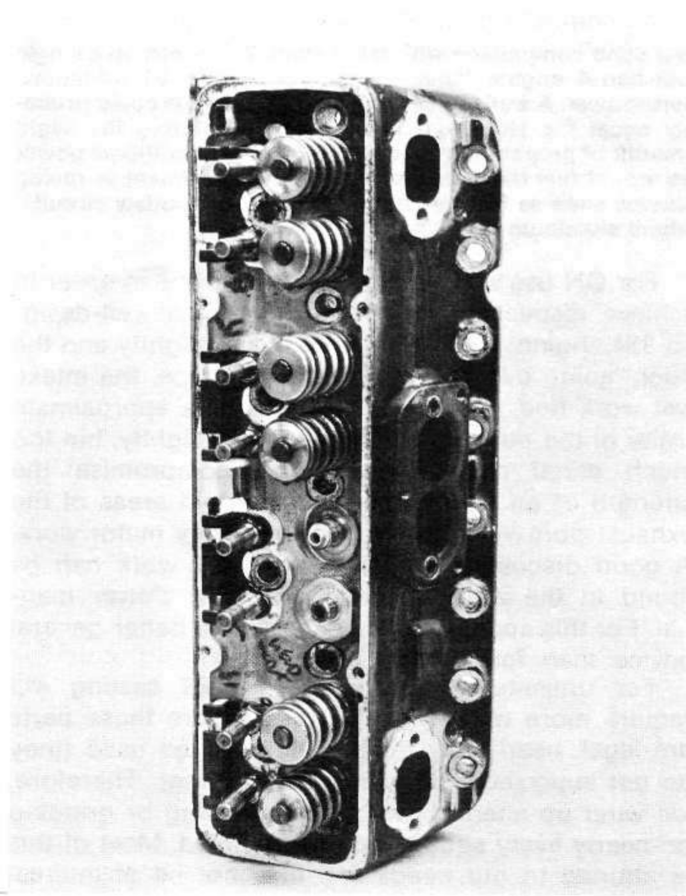
This casting had the 292 number and/or the casting date, making it easily identifiable. Also, on these castings the one-bump marking first appeared on each end of the head. A slight change was also made in the way the molds fit together, resulting in the mold parting line coming up around the outside corner of each piece. This part may still be serviced by some dealers who have heavy-duty parts in stock. It's virtually identical to the current batch except the valve unshrouding cutter diameter is still 2.40 inches in diameter.

The current batch, dating from November, 1974, is very similar to the preceding part. Externally, it appears virtually the same; the one-bump identifier is still used, but only the 292 casting number appears without an accompanying casting date. The only ostensible difference is the reduction of the diameter of the intake valve counterbore tool. The small change from 2.40 inches down to 2.34 inches leaves more metal on the outer wall of the chamber, next to the intake valve opening, for hand shaping. This should eliminate any possibility of the chamber hanging over the wall of the 4-inch bore block. It is always very desirable that the wall along the intake valve be matched carefully to the cylinder wall contour to prevent a ledge from impeding the inrushing mixture.

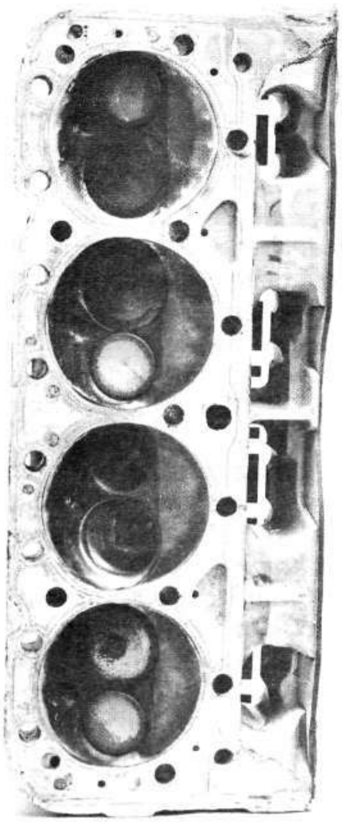
The big drawback of this head compared to the "good" '73 castings is the reduction of the size of the intake ports. The later castings are made with newly-revised core boxes and, in our opinion, metal has been taken out in some very critical areas. The most crucial being that space between the water-side port wall and the head bolt bosses. Reducing this section severely limits the degree to which the port can be widened. In the June '73 casting more waterside metal was existent in this area. By grinding a wider port section here and pulling the pushrods apart with offset rockers, the June '73 head allowed us to get almost as much volume in the port as we would like.



From the outside, the intake ports of a 292 head (top) don't appear that much larger than a 492 casting. They are, nonetheless, larger, and more importantly, they allow the total volume to be increased through normal port grinding techniques.



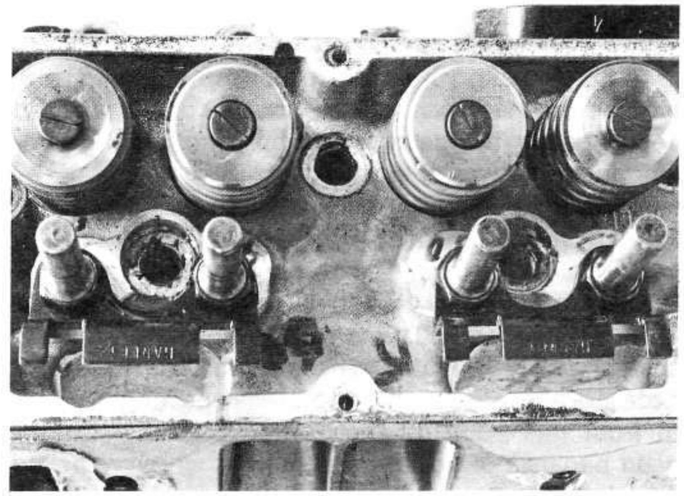
Where aluminum cylinder heads are allowed, the alloy castings from the Brownfield Company, Burbank, California, are worth considering. These heads are similar to the standard off-road smallblock casting but the intake and exhaust ports are significantly better.



In a dyno comparison with fairly stock 292 heads on a single four-barrel engine these heads were worth 51 additional horsepower. A well worked-out set of 292 heads could probably equal the aluminum heads, but considering the slight amount of preparation required to gain this additional power we would feel these heads are a good investment in racing classes such as the so-called circle racing "outlaw circuits" where aluminum heads may be legal.

For GN use very little needs to be done in order to achieve respectable performance from a well-desiged 354 engine. If the roof is laid back slightly and the floor "going over the hill" is put in shape, the intake will work fine. The "neck down" at the approximate plane of the pushrods can be opened slightly, but too much metal removal here may compromise the strength of an endurance head. Certain areas of the exhaust port will also require some very minor work. A good discussion of this elementary work can be found in the well-illustrated *Chevrolet Power* manual. For this application we cannot give better general advice than found there.

For unlimited drag racing the 292 casting will require more work. In any class, where these parts are legal, nearly any modification can be used (they do not supersede the 492 or 461 casting). Therefore, we wind up altering, welding, machining or grinding on nearly every square inch of the head. Most of this is attuned to our needs and may not be of interest to the average racer. For instance, we do remachine the spring seats to fit the particular springs we like to run. This may not have to be done if different springs are selected. Most of the important general work will be outlined in the succeeding sections.

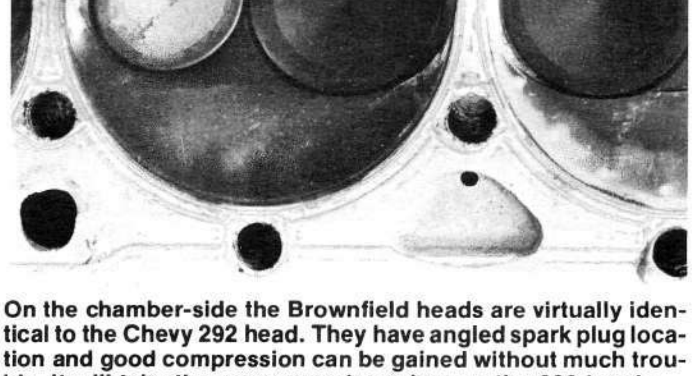


On the Brownfield aluminum heads the rocker arm studs are already moved away from the sides of the intake ports, allowing the exhaust-side walls to be spread apart. This allows the minimum cross-sectional area to be larger than on the cast iron heads. Such an advantage on a high-rpm or larger displacement smallblock is worth a good deal of power.

ALUMINUM HEADS

For those who race under sanctions allowing aluminum cylinder heads, we have recently completed an interesting test program with the alloy castings being sold by Brownfield, Co. This work was not directly related to our own racing program, but it was a chance for us to evaluate some of these pieces on our engine dynamometer. The testing was not fully under our control, yet we found the results interesting and report them here as a matter of interest.

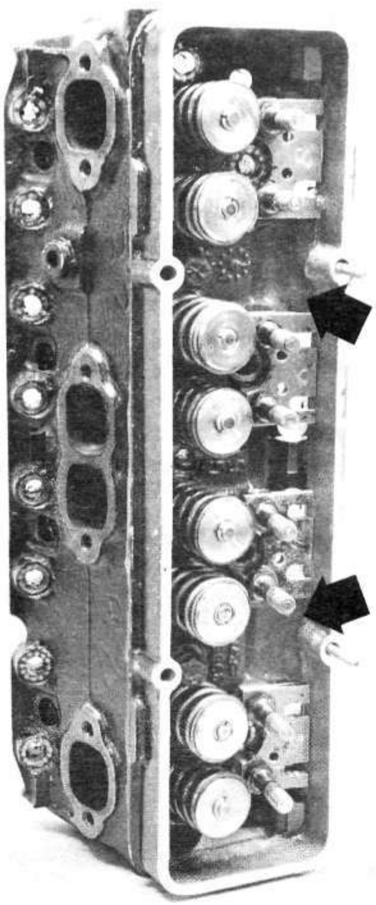
Externally, the Brownfield heads appear to be very similar to the Chevrolet off-road heads. They can be fitted with the standard screw-in studs and pushrod guide plates. The spring seats will easily accept the biggest springs you can fit between the head bolt bosses. On the bottomside, the chambers



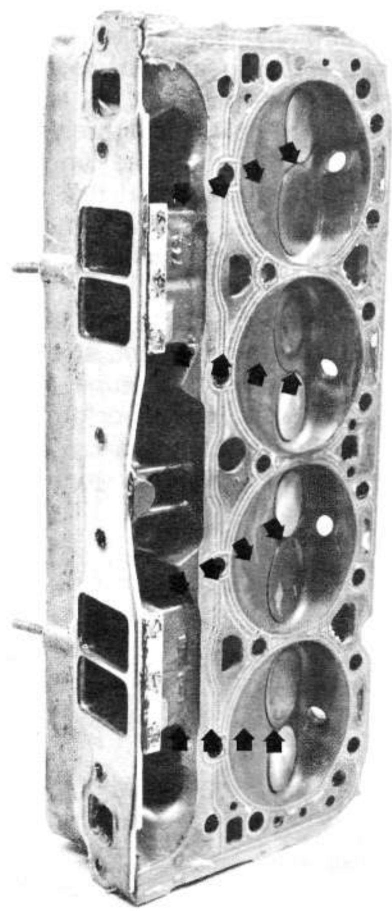
On the chamber-side the Brownfield heads are virtually identical to the Chevy 292 head. They have angled spark plug location and good compression can be gained without much trouble. It will take the same oversize valves as the 292 heads.

look like the Chevy 292 heads. However, the resemblance ends there. Inside the ports things have been changed around quite a bit. They look to be much larger and very efficient. We have done some preliminary flow testing and both the intake and exhaust ports, in stock form, flow much better than the stock Chevy ports. In fact, they appear to do pretty well when matched against good reworked factory ports. We aren't saying that in stock form they are better than a Pro Stock prep, but they are quite impressive. With some modification we suspect they would be among the best anywhere.

The results of our dyno test highlight how good they really are. The test engine was a typical Nascar-type, Super Modified, single four-barrel 354. It was not one of our engines but had been built in another shop and brought here for testing. We undertook a series of tests of which the most meaningful was an A-B comparison between the Brownfield heads and a set of 292 heads. Each set of heads had out-of-the-box ports with only a good competition valve job, equalized combustion chambers, and only some slight port matching to the manifold. Admittedly, the 292 head isn't all that charming with only this little bit of work, but the Brownfield heads had not been worked over to any greater degree. We verified that under these stark conditions, on a four-barrel engine, the Brownfield heads showed a power increase of 51 horses over the 292 heads. That's a good bunch of power from a simple head swap. It's more difference than you can usually find when going from a single four-barrel to dual-quads on a tunnel ram. With a little more fiddling in the heads we could maybe find another 20 horsepower without seriously compromising their strength. So, we are talking about adding approximately 20 "estimated" horsepower to the tested 51, giving an overall power gain of 70 horsepower. A really good set of 292 heads can be reworked to produce equal performance, but if we were racing a circle-track sprint car or in some other racing class where aluminum alloy heads were legal, these would look like a very good deal. Considering that they outrun the stock smallblock iron heads by a respectable margin and require only a minimal amount of work, they could well be one of the best horsepower buys in the whole racing market.



This is one of our most highly-modified smallblock cast iron heads. Big springs, titanium retainers, fabricated aluminum pushrod guide plates and a rocker cover spacer to clear the stud tie bar are fairly standard. Note, however, that the area over the intake port pairs has been built up into the rocker cover area. This is done to raise the roof of the intake port as high as possible to gain a short roof length from the port pocket straight into the manifold runner.



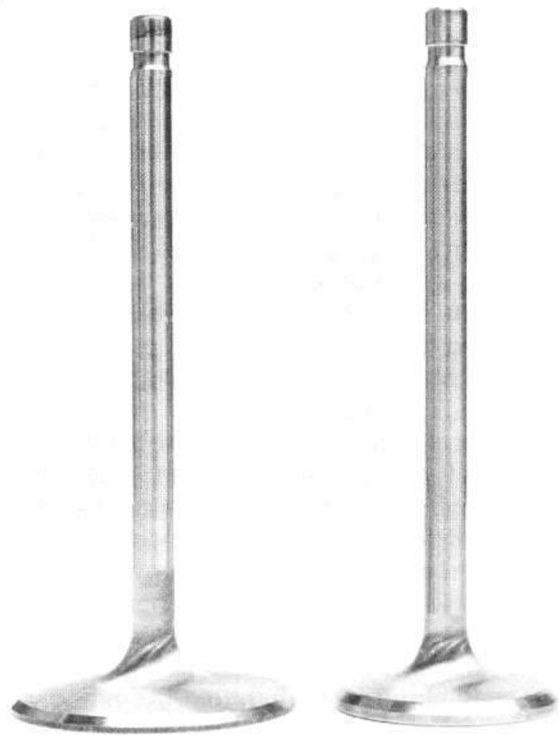
A bottom view of our big heads. The floor material has not been cut away but the roof has been moved up into the rocker cover area and, more importantly, the exhaust-side walls have been cut completely away and new material has been welded into place to gain more volume inside the port. This wall has been moved over until it lays against the upper cylinder head bolt boss. These heads may possibly be too big for the current 330's but they work well on the large match race engines.

GENERAL MODIFICATION CONSIDERATIONS

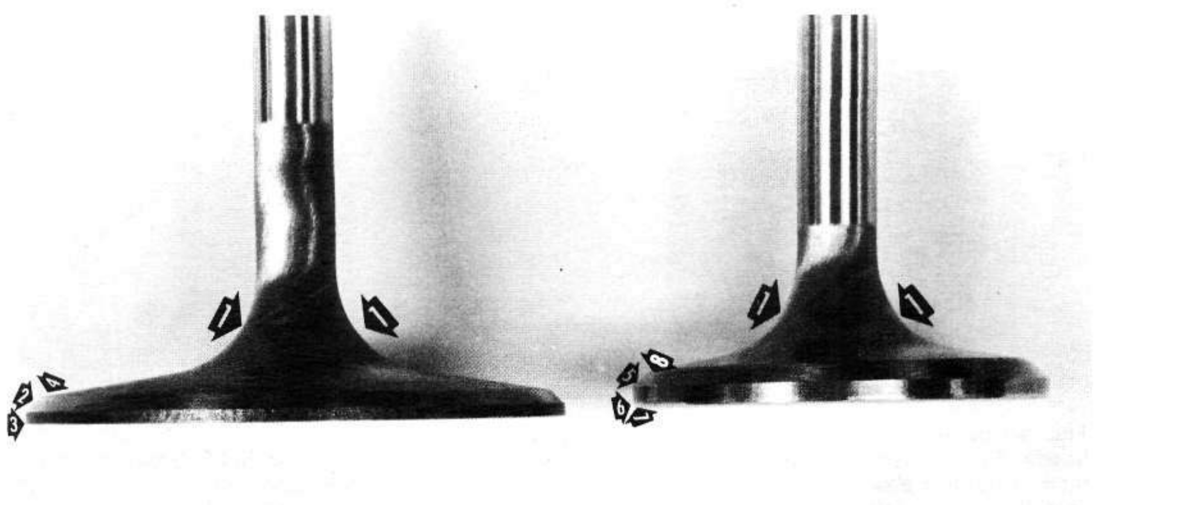
Since most of the finish work is accomplished here in our shop, we have been able to develop some of our own opinions about head porting. By utilizing our flow bench, we get a much clearer picture of what is important inside the intake and exhaust ports and what is not so important. Through these studies we have learned that the most crucial portion of the entire port is that area immediately at the valve seat. There is no room for anything less than absolute care when finishing the seats. These details affect high- and low-lift flow. As the velocity goes up, their effect becomes increasingly pronounced! The next most important area is the ⅜-inch immediately behind the seat and up into the bowl. Beyond this, the next 1½ inches must be carefully shaped. The rest of the port is of less importance, in a similarly diminishing fashion, other than for volume considerations.

Work in the port entry, well away from the seat, will affect the "available volume" the port provides to the cylinder. It generally has very little determinable effect in a flow test. However, you must always watch what happens to the flow when you are chasing down volume changes away from the seat area. The intake manifold port size must also be considered when the volume is being altered, as the manifold will influence the flow numbers. It is difficult to state specifics, but we know it is entirely possible for the shape and volume of a manifold runner to significantly influence the flow characteristics of the intake port. For instance, we have seen a ¾-inch change in the length of one manifold runner sidewall change the combined wall length sufficiently to alter the measurable flow through the head port. Such a change inside the manifold plenum, at the leg entry, or in the length of an injector tube, may have a very drastic effect on the engine performance. Often a port may be so large in certain areas that the flow looks dead when you are only checking it with a bare flange entry, but it really comes to life when you put a full intake runner on the flange. By increasing the length of only one side or of the floor, the flow may lose velocity in a localized area, allowing a greater volume of flow to make the turn over the hill more effectively. Presto! The combination works like gangbusters when the port, by itself, was relatively dead.

To counteract or allow for this phenomenon, we try to include in our testing some flow studies which will emulate the manifold runner and what effect it will have on the intake ports. This is sometimes very difficult to accomplish, but the results are often grati-



All our race pieces are fitted with Manley 5/16-inch stem "flat head" valves. These valves or something with a similar backside radius between the stem and the head must be used to keep from blocking up the port at the lower lift ranges.



Valve face prep is critical. Note the small backside radii (arrow 1) The intake valve must have a 45° face, 0.085- to 0.090-inch wide (arrow 2). The margin (arrow 3) must be 0.050-inch wide with a square top corner. We always insist upon a 30° undercut (arrow 4), the width is unimportant. The exhaust valve also has a 45° seat, 0.060- to 0.080-inch wide (arrow 5). The face must be 0.010- to 0.015-inch wider than the seat. The margin should be 0.080- to 0.090-inch tall (arrow 6) with 0.020-inch top radius (arrow 7). A 30° undercut (arrow 8) is also used on the exhaust.

fying. In our opinion, the best way to develop a good intake port is to work on a flow bench that allows the port to be studied with a manifold runner attached to the flange. Of course, carrying this to the extreme, we can see some benefit to flowing the port with a runner attached along with a complete plenum chamber and carburetor(s).

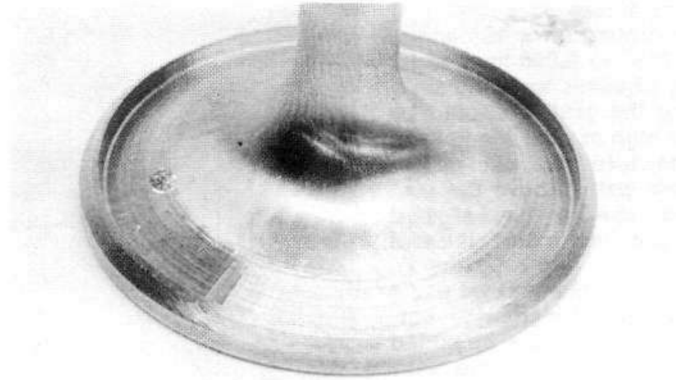
VALVES

In all of our racing heads we install Manley intake and exhaust valves. We like these valves because they have 5/16-inch stems and the backside radii are small. This is important to avoid blocking the Chevy port with the valve. The 5/16-inch stem, though it is only a small difference over the 11/32-inch stem, makes a measurable increase in flow. The short backside radius has the same effect. By opening the area across the backside we find a definite advantage at midlift. We have found the Manleys to be satisfactory in every way, except that they may be a little heavy for speeds in excess of 9000 rpm. With the springs currently available, controlling them becomes difficult at very high speeds.

We have investigated titanium valves, but they have too much "tulip" or radius on the backside. They block up the port badly at low lift. It is possible to rework the existing titanium valves to a shape similar to the Manley. This works pretty well if the bowl shape is also reworked to compensate. In the future Manley will be offering a "flat head" titanium valve similar to their current stainless number. It will probably be available with a 5/16- or 11/32-inch stem and should be a big help for anyone running a high-rpm valvetrain. They will be expensive but the benefits will far outweigh this consideration.

At this point we use 2.06-inch diameter intakes and 1.615-inch diameter exhausts in our drag race heads. To run this combination, the valve guide centerlines have been moved apart. To keep the valve heads from clashing, we move the exhaust guide away from the intake guide and over toward the exhaust-side wall of the chamber. We have found it is possible to move the exhaust valve guide as much as 0.040-inch without trouble. This is necessary to get the larger intake valve into the chamber, although it has the additional advantage of placing the exhaust valve more in line with the center of the exhaust port. It is possible to use a 2.06-inch, 1.60-inch combination without moving the valves, provided the stock guide centerline-to-centerline condition is about average.

We also have experimented with a larger combination which has 2.100-inch intakes and 1.625-inch exhausts. We have to move both guides toward the exhaust-side wall to get this setup into the chamber. This is a lot of work for very little improvement. In fact, with these larger valves we have seen some evidence that the heads will not perform as well as with the 2.06 by 1.615 arrangement. The 2.100 intake would only help low-lift flow "slightly," but the



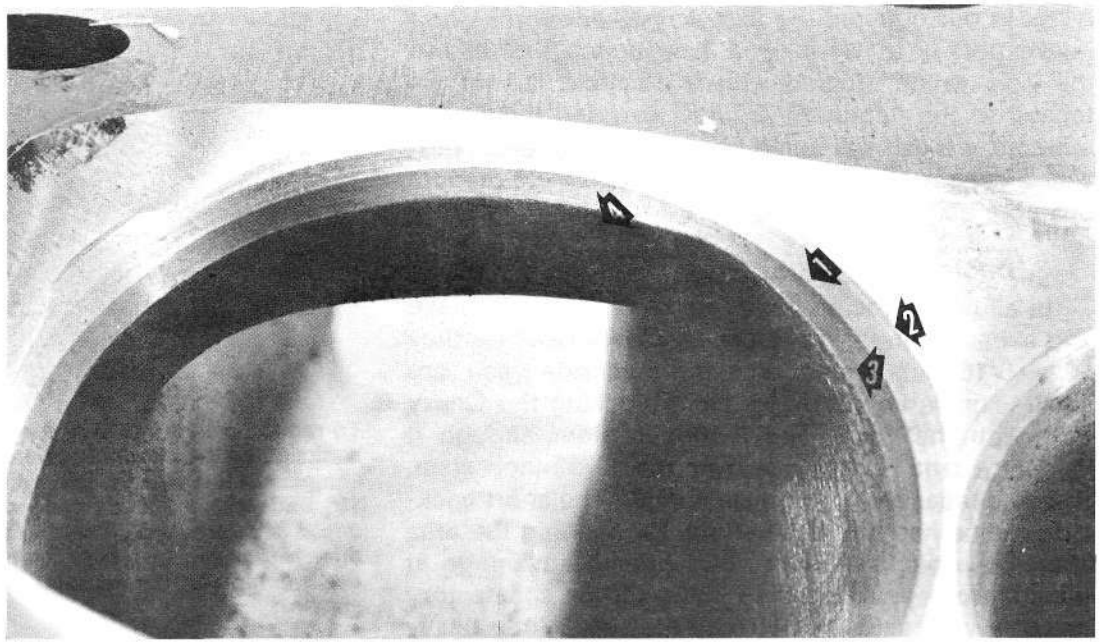
To reduce backflow out of the exhaust port we ditch cut the backside of the exhaust valves. This happens to be a rough sample but it shows the general idea. At the inside edge of the 30° undercut, a 90° machine cut is made into the head to a depth of approximately 0.050-inch. The bottom edge of the ditch is then radiused back toward the stem tulip section. After this is completed the backside should be reswirl-polished (not shown).

effect is minor. It can actually be a disadvantage because we don't want the port to really take off until the piston has moved past TDC and downward on the intake portion of the cycle. This is closely tied in with the cam design, but it seems now that we really want very limited flow at low lifts. Then, we like to see it build very rapidly at about 0.250- to 0.300-inch of valve lift, and continue to gain substantially as the valve moves further from the seat.

Our best drag heads are set up to operate with the big valve combination, 2.100 by 1.625, but we install the smaller valves. The results are better with the smaller valves because we feel we can gain an improved "mouth" or bowl shape immediately behind the seat. With the very big valves it is difficult to work this area properly. To fit the valves into the guides we use bronze insert bushings designed for the 5/16-inch stems. We run the valves quite tight in the stems with about 0.0015-inch clearance and we check them often because they will pound out the seats very quickly if they get loose. The valves must be inspected carefully during preparation. If we find any small imperfections or nicks in the stem, radius, or tulip section, we consider that valve unusable, unless we can swirl polish the surface completely smooth. Failure to do so is a sure invitation for it to self-destruct.

SEATS AND FACES

The most important work in the entire head preparation is the correct shaping of the seats and faces. On the intake valves we use conventional 45° faces which are 0.085- to 0.090-inch wide. We have run them as wide as 0.100-inch with no trouble. If it is not possible to gain a 0.085-0.090 face width with some break left below the face, the valve is not suitable. The intake valve must have a 0.050-inch wide top margin with a dead-square top corner. The Manleys are usually very good in this respect although they must be checked. The "squarer" the top

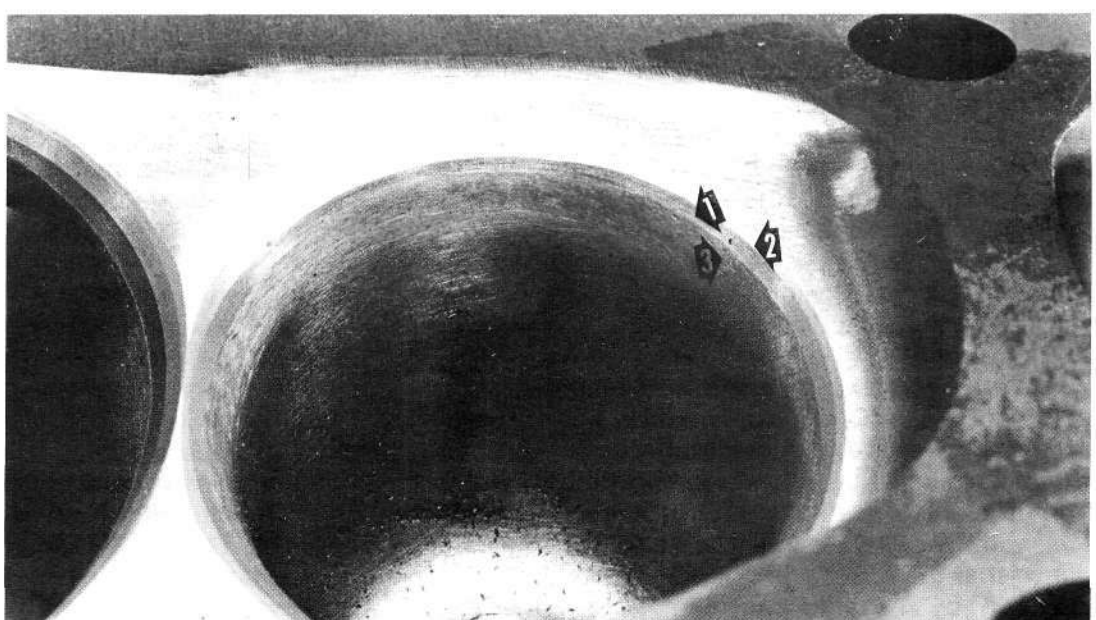


The intake seat (arrow 1) is ground on a 45° angle, 0.085- to 0.090-inch wide. In all cases we avoid sinking the seat. It should be as high as possible to keep the valve up out of the flow path. Above the seat we use a bowl-shaped stone that runs into the seat at a 30° angle (arrow 2). Beneath the seat we use a 60°, 0.100-inch wide bottom cut, (arrow 3) butted directly against the seat. Below this cut a 90° vertical (arrow 4) must drop toward the port for a minimum of 0.100-inch on the short radius side before the turn is initiated. This is a critical feature and using a 2.060- or 2.100-inch valve makes this vertical easier to achieve.

edge, the better. This portion of the valve acts more or less as a chisel to reduce backflow entering the intake port. A margin must be present, as a knife edge above the face will hinder inward flow. Putting some margin surface between the top of the face and the top of the valve gives the flow "direction" as it comes around off the seat. We have checked several different margin widths. Flow always improves by increasing the margin width, up to the recommended 0.050-inch spec. There is no subsequent improvement from 0.060-inch on up to 0.125-inch.

We insist on a slight break or undercut on the intake valves. A 30° surface is preferred, but the width is unimportant. This also helps to stabilize flow. Sometimes it doesn't really seem to make a measurable difference. In other instances, it helps ever so slightly. In any event, it doesn't hurt, so we always use the undercut.

The intake seats are ground on a 45° angle and are 0.085- to 0.090-inch wide. We avoid sinking the valve, preferring to keep it as high as possible, out of the flow path. Above the seat we use a bowl-shaped stone which runs into the seats at a 30° angle. If the seat is very high, the intersection angle may be reduced to 15°. On the port-side a 60° bottom cut which is about 0.100-inch wide must be butted against the seat. Directly below this cut a 90° vertical falls into the port *a minimum of 0.100-inch before the turn is initiated.* This is sometimes difficult to achieve with stock valves. Installing the larger valves will increase the diameter of the seats and allow a full 0.100-inch wide vertical drop into the bowl because of the additional metal which lies below the seat at this point. All of the numbers listed here are extremely important for maximum intake efficiency. There should never be a radius



Exhaust seat preparation is simple. The 45° seat is 0.065- to 0.070-inch wide (arrow 1). If there is room for a top cut, it should be a 30° angle (arrow 2), as narrow as possible to keep the valve up out of the port bowl. Below the seat a radiused departure (arrow 3) leads directly into the bowl. This departure must start exactly at the lower edge of the seat and must be ground very carefully to keep from getting into the seat.

leading directly from the intake seat into the mouth of the port — never!

The exhaust valve faces are 45° measuring 0.060- to 0.080-inch wide. For best results the face should be 0.010- to 0.015-inch wider than the seat. The vertical margin from the edge of the face to the top of the valve must be 0.090-inch wide. At the top of the margin a 0.020-inch radius joins the margin to the top flat of the valve head. These are important features. We have tested several different radius dimensions on this edge but the 0.020 works better than anything else. The height of the top margin on the exhaust is also critical. Lengthening the margin increases the amount of flow, all the way up until the margin is 0.080- to 0.090-inch wide (the top edge radius may be included in the overall margin measurement). This is especially important if the exhaust valve is close to the chamber wall, as is the case when the valve has been moved over to accommodate a larger intake valve or when the chamber is very small for use on small-bore engines. On the other side of the coin, increasing the margin beyond this point is not necessary and adds weight. Reducing this margin down to 0.060-inch will not be very detrimental. There should never be a constant radius from the top edge of the face to the top edge of the valve. The perpendicular surface of the margin is essential for the exhaust to "work." In the mid-lift range it could mean as much as 10% to 15% more flow.

Below the face, a 30° undercut should always be used. The width is not critical, but it must butt directly against the face. We have also done some testing with exhaust valves having ditch cuts on the backside of the valve head. By ditching the backside we have found that backflow into the chamber can be significantly reduced. The reduction amounts to about 25% at 0.100-inch lift and 10% at 0.200-inch lift. When testing these valves in our dyno engine there was no discernible increase in top end power, much to our disappointment. But they did show us 1% more at 7000 rpm and 1.5% at 6500 rpm. We believe the engine responds much better from off-idle. To date we have not tried these valves in a GN engine, but in the drag engines this modification has not caused distress, "tuliping" or edge overheating.

We leave the 30° undercut about 0.025-inch wide. At this point we make a 90° machine cut down into the back of the valve head to a depth of 0.050-inch. The inside edge of the ditch is then radiused smoothly into the tulip section. This can be seen clearly in the accompanying illustrations.

The exhaust valve is seated on a 45°, 0.065- to 0.070-inch cut. The exhaust top cut, if there is any, should be 30°. Below the seat a radiused departure begins immediately at the edge of the seat and leads directly into the bowl. This must be done very carefully. It is worth mentioning again that the radius must start *exactly* at the edge of the seat.

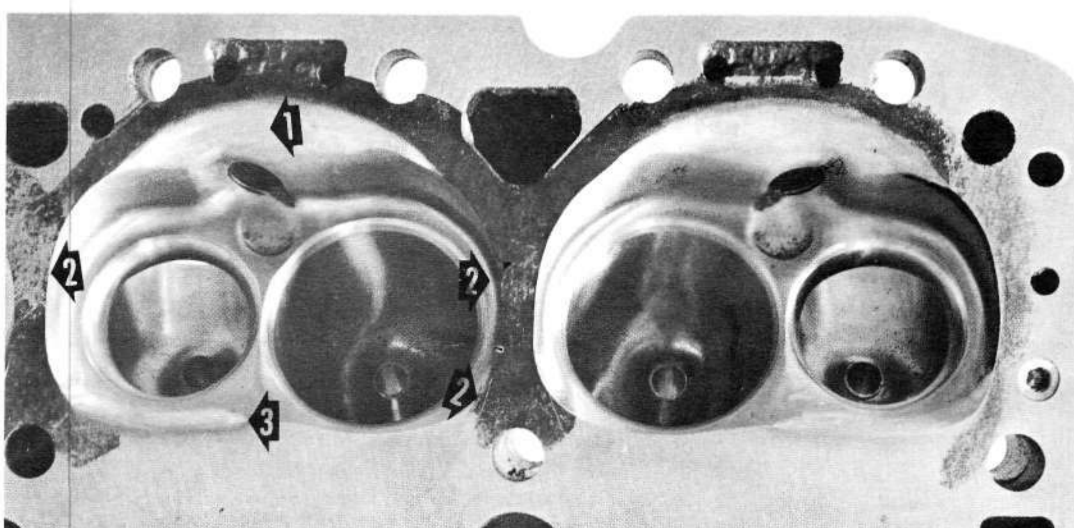
When working the seats on a 292 casting, it should be remembered that they are induction-hardened, a carry-over from lead-free gas development. This is an advantage for circle-track and alcohol applications but any welding or intense heating will destroy the treatment.

COMBUSTION CHAMBERS

Very little work is required inside the 292 chambers. With valves seated and plug installed, the nominal chamber volume is 66 cubic centimeters. Excessive metal removal will possibly cause a loss of compression ratio which cannot be regained through milling the head. It will also be undesirable to increase the dome height to recover compression; therefore, it is best to be conservative inside the chamber.

We prep the quench pad edge in an opposite manner to the piston valve notch. That is, we radius the edge of the pad in the area of the exhaust valve. This assists exhaust flow from the quench area toward the open exhaust valve. On the intake side we leave the edge sharp to discourage reverse flow from the quench area of the chamber into the port during the initial stages of overlap.

The only major alterations in the rest of the chamber should be a slight opening of the walls directly



The ledge above the backwall (arrow 1) is opened toward the roof of the chamber in order to prevent any secondary combustion (detonation) pockets. Around the intake and exhaust openings some grinding is required to bring the wall out to the gasket line (arrow 2). The radius where this wall joins the roof, around the valve opening, should be smooth and the approach to the exhaust must not be lowered. The ledge between the exhaust opening and the quench roof is radiused (arrow 3).

adjacent to the exhaust valve. It is best to spray machinist blue on the head surface and use the selected head gasket as a pattern to scribe the gasket bowl circumference on the head deck. At the point where the chamber wall passes closest to the valve head a grinder can be used to lay the wall back to the scribed line. Again, only a small amount of metal needs to be cut away. This is another method to promote flow toward the exhaust opening.

If the compression requisite permits, we will do some grinding on the vertical or "back-wall" directly across from the intake port exit. At the very least, the top edge of the wall should be blended over to the small backside squish area, cast in the roof below the spark plug. It is possible to create secondary combustion pockets up in the roof squish area if the chamber shape in this area and the piston dome are not properly coordinated. The intake-side of the back-wall must be opened to give at least a 0.275-inch radial clearance to the valve edge. Induction mixture crossing the back of the wide open valve can then gain flow area as it leaves the port. Opening the chamber more than this is just throwing away compression ratio. The wall will not have to be cut as deeply on the exhaust side as all efforts on this side of the chamber are directed toward smoothing the entrance to the seat, particularly from the quench area. Removing metal from the far wall of the exhaust will only further reduce compression. A slight roll along the exhaust-side edge will reduce this and help the swirl action of incoming mixture.

Finally, with the slant plug heads we insure there aren't any break-out threads left exposed inside the chamber. Some touch-up grinding may be needed to remove extra threads left by the factory tooling. The edge of the carbon crunch pocket can also be rolled over a little. All of this is intended to remove sharp metal edges which might become superheated during combustion and create pre-ignition or secondary combustion detonation problems. This was more pronounced on the early slant-plug heads, but the thread break-out area of the 292 head has been improved considerably and this is now not much of a problem.

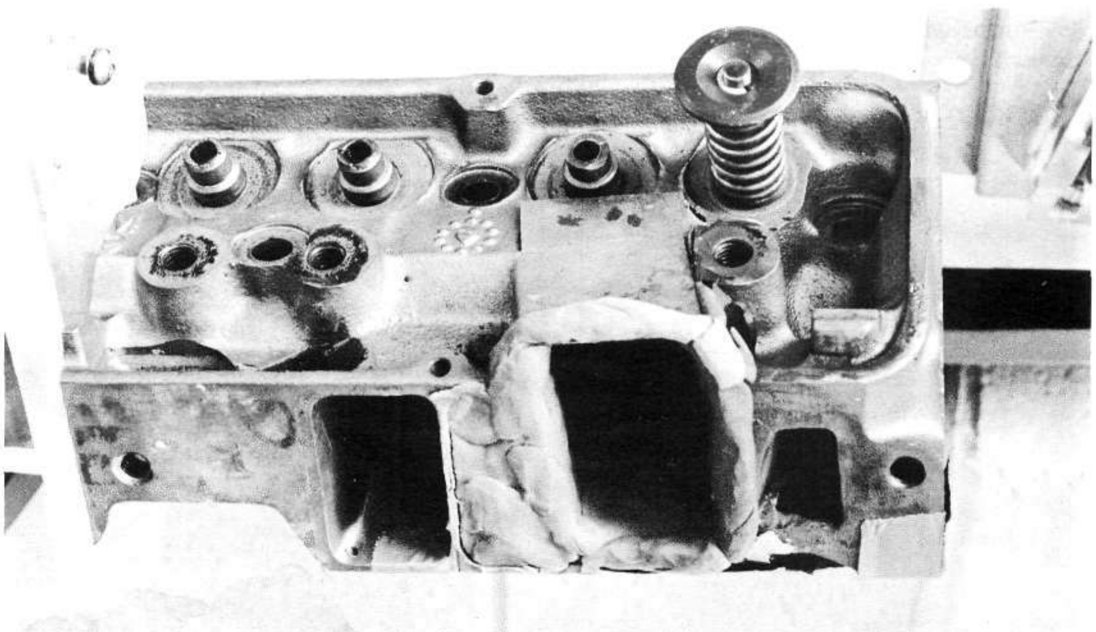
INTAKE PORT MODIFICATIONS

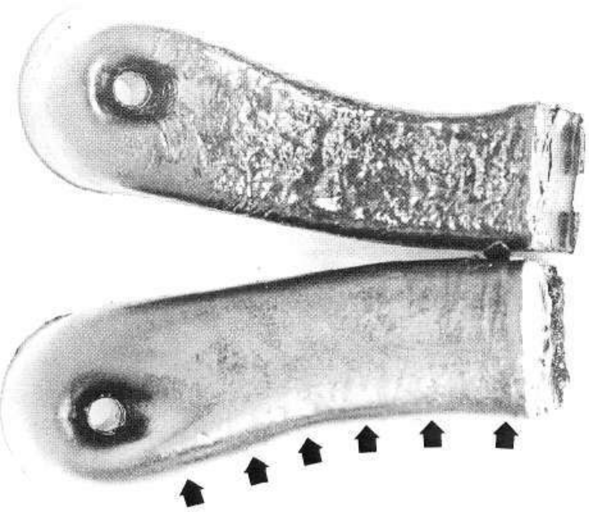
For moderate street/racing performance requirements (as compared to Pro Stock drag racing) some very minor port preparation will considerably improve the intake flow in the production 292 cylinder head. This work is described in the *Chevy Power* publication. Much of the work outlined there is similar to the preparation of our racing heads. The only major difference is the cutting and/or welding that may be necessary for increased port volume in a very high speed engine.

Beginning immediately above the valve seat, as described earlier, there is a 60° bottom cut and a 90° vertical before the radius turns of the bowl are initiated. Each of these surfaces should be 0.100-inch wide. We cannot overstress the importance of this configuration! Once this has been achieved, the long radius of the roof and the short radius of the floor can begin. On the other side of the bowl, that which forms the roof when it leaves the bowl area, the long radius should blend right into the 0.100 vertical without a distinct break.

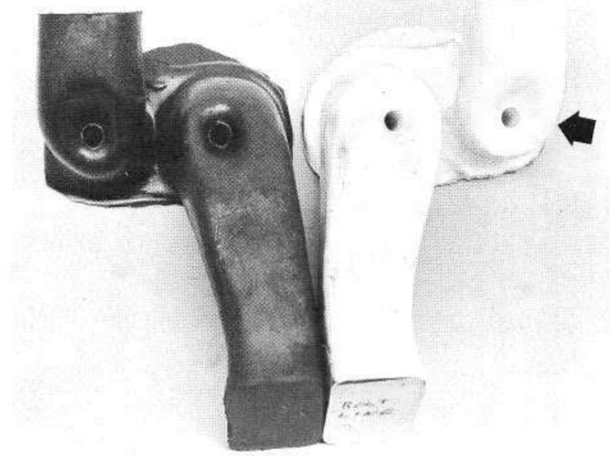
Higher in the bowl this wall will form the protrusion for the valve guide support. We do not cut the guide completely away. Considerable metal can be removed here to decrease the cross section width of the protrusion, but the guide definitely should not be shortened. Metal can be cut from either side of the protrusion, as viewed from the intake flange, to give the guide a narrower profile. A minimum of metal should be removed from the section that extends along the vertical plane of the valve centerline. This provides support for the valve and guide, especially important with radical valve action. Besides, this

Unfortunately, even the very best stock smallblock cast iron heads have a difficult time keeping up with cylinder heads that have been developed with a strong emphasis on racing applications. We have, however, explored some extremely radical smallblock configurations.

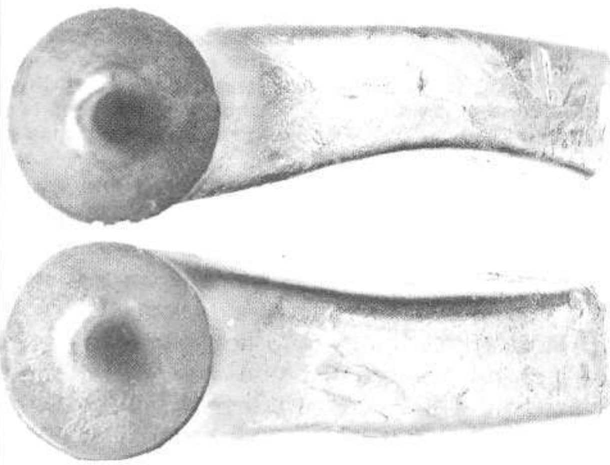




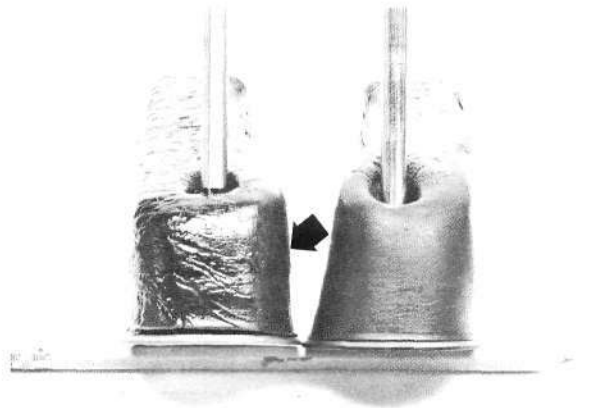
For sufficient feeding volume in a very high speed racing engine the intake port volume may have to be enlarged. From the intake flange into the port approximately 3 inches it is possible to rework extensively to gain volume, without hurting flow. We have cut the exhaust-side wall away and welded in new material to gain more port width (mold on the bottom) as compared to a stock 292 port (mold on the top).



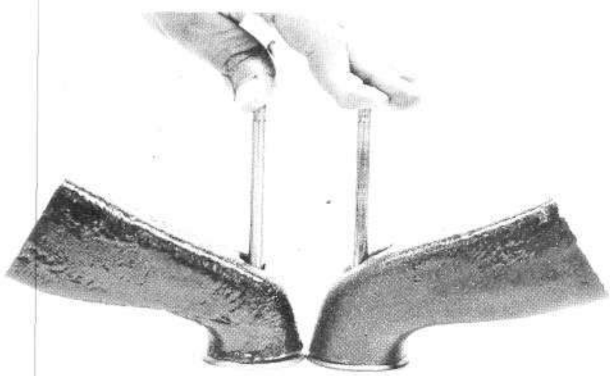
The port pair on the left is the best 292 head we have ever had without cutting and welding the port walls. The intake port has been enlarged as much as possible without hitting water and the exhaust has a good backside shape for mid- to high-lift flow. The pair on the right is one of our very early 292 efforts. The "fuller" backside shape of the exhaust port worked well at low- to mid-lift but died at high lift.



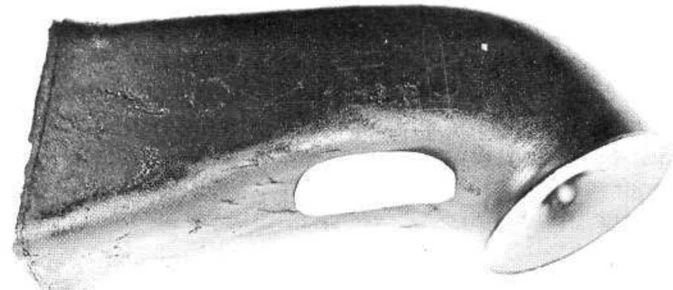
Bottom view of the stock 292 port mold (top) and the welded port (bottom) shows how much volume can be gained by moving the exhaust-side wall away from the port centerline. We have gone even further with this wall than the comparison shown here, but these efforts put too much volume in the port.



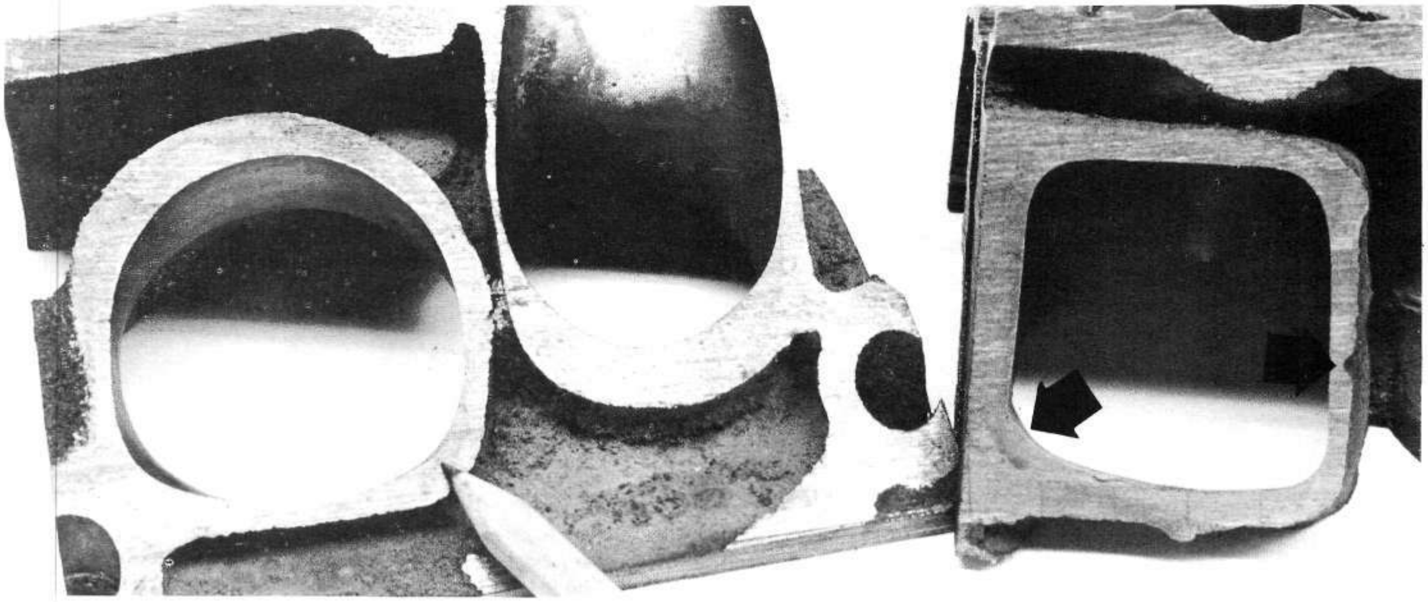
On the left is a stock Brownfield aluminum intake port mold, the one on the right is a good 292 mold. This comparison shows how the bowl shape of the Brownfield head feeds the chamber better than a 292 head. Note the wall shape on the long or exhaust-side of the bowl.



On the left is a Brownfield as-cast intake, on the right is a worked-out 292 (without welding). There is considerable difference between the two, along the backside radius of the bowl, up near the valve guide. The 292 has more volume after grinding than the stock aluminum port and may work better at very high engine speeds, but the width of the Brownfield bowl shape feeds very well at lower engine speeds



The white patch on this 292 intake port mold indicates the area where the divider-side floor-to-wall radius should be filled and enlarged. Filling this corner will give the floor a distinct slope from the divider side, downward toward the exhaust side. The exhaust-side floor-to-wall radius should be smaller. This modification will help the port pull harder on the exhaust side.



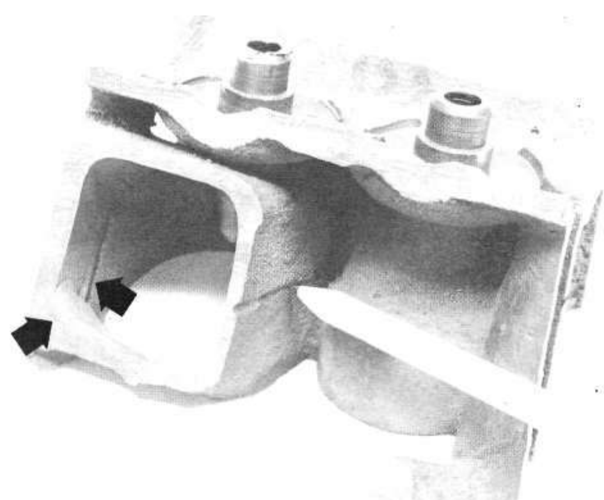
These are two sections of the same port. The section at right sits on top of the left one. The face of the right section is cut approximately 3 inches into the port, the face at left is approximately through the center of the port pocket. We have put some clay in the corner of vertical section to show the area of the divider-side floor that must be filled and raised. The floor then takes a distinctive slope toward the exhaust-side wall. The radius indicated by the pointer should be reduced more than shown here to gain the correct slope. Note the "paste ooze" line along the water-side surface. This occurs occasionally when the cores are put together during the molding process. When you grind the port walls this area may break through. Since it is only a localized shallow spot it can usually be fixed easily. On the other side of the port there appears to be more metal for enlargement grinding. This is the divider wall between the two ports. Grinding in this area is dangerous because you can break into the head bolt boss area.

portion does not present a restrictive section to the direction of flow.

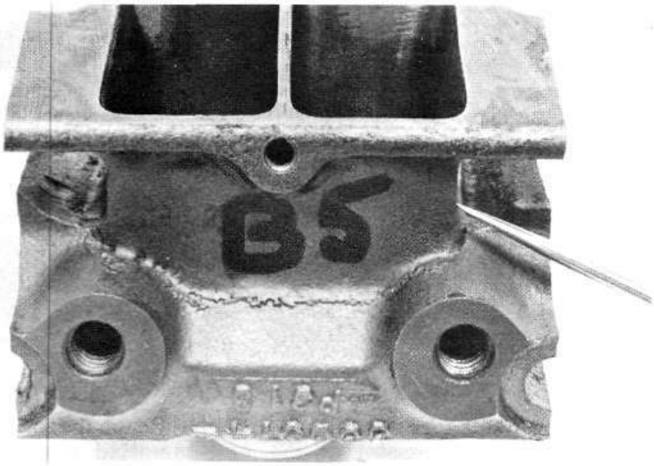
The shortside radius of the floor profile, leading up to the highest point (crown) of the floor, should not be reduced. Reducing this radius or lowering the floor at the crown will hurt flow throughout the lift range. Work at this point should be concentrated on the corner radius where the floor joins the sidewall. We have a somewhat unique preparation in this area. When viewing the port from the flange, it is easy to view the radius where the floor joins with the divider wall and the corresponding radius on the other side of the floor where it joins with the exhaust-side wall. We do not make the radius of both these corners equal. Previously the plan has been to make these corner radii as small as possible, thus effectively widening the floor and making a tiny 90° blend upward to the sidewalls. We follow this general idea on the exhaust-side wall. However, across the floor on the divider side, we prefer a longer radius and we have added material in this area to raise the floor. The floor then, at the crown, assumes a slope away from the divider wall and falls toward the exhaust-side wall. This technique has markedly improved all of our intake ports.

At the approximate plane of the high point, the cross section approximately 3 inches from the manifold flange, it is possible to begin widening the port for increased volume. The runner sidewall should be blended smoothly into the sides of the bowl. Care must be taken with the divider wall. In the area of the floor crown, a thinwall bolt boss and drilling pass down between the adjoining sidewalls of the siamesed intake ports. Cutting the wall away to gain width here will not be of benefit. Once the wall has formed a straight plane from the flange entry to the bowl, all further effort at gaining volume should be directed toward the exhaust-side wall.

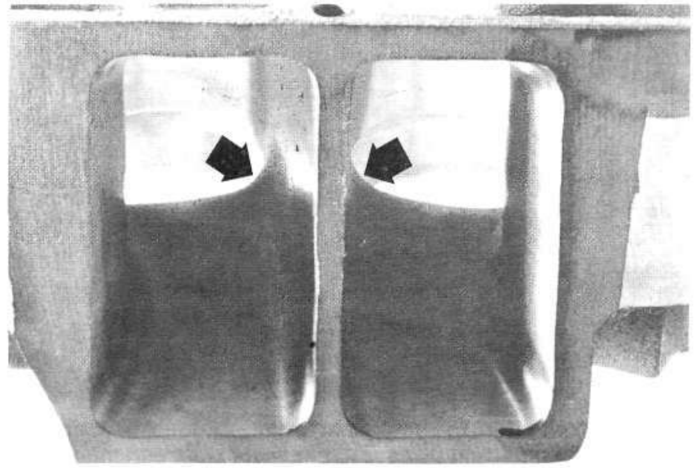
Most of the volume and flow gain can be achieved by moving the exhaust-side wall. Metal can be removed from the side of the bowl all the way out to the flange. Unfortunately, the production 292 casting has less metal on the waterside surface of this wall than we would like. A head bolt boss/drilling passes through the head in the close vicinity of this



Another view of the section shown on the right in the photo above. Note the filled corner and the paste ooze line. It is also possible to see the results of too much grinding on the divider wall. Cutting this wall more than necessary to gain a straight line from the bowl to the entry is dangerous.



When more volume is needed inside the port entry the place to begin is with the port roof and sides. This section has been cut completely away and a new roof is welded in place to get the section width taller. The sides of the entry have been filled with weld so the section width can be increased.

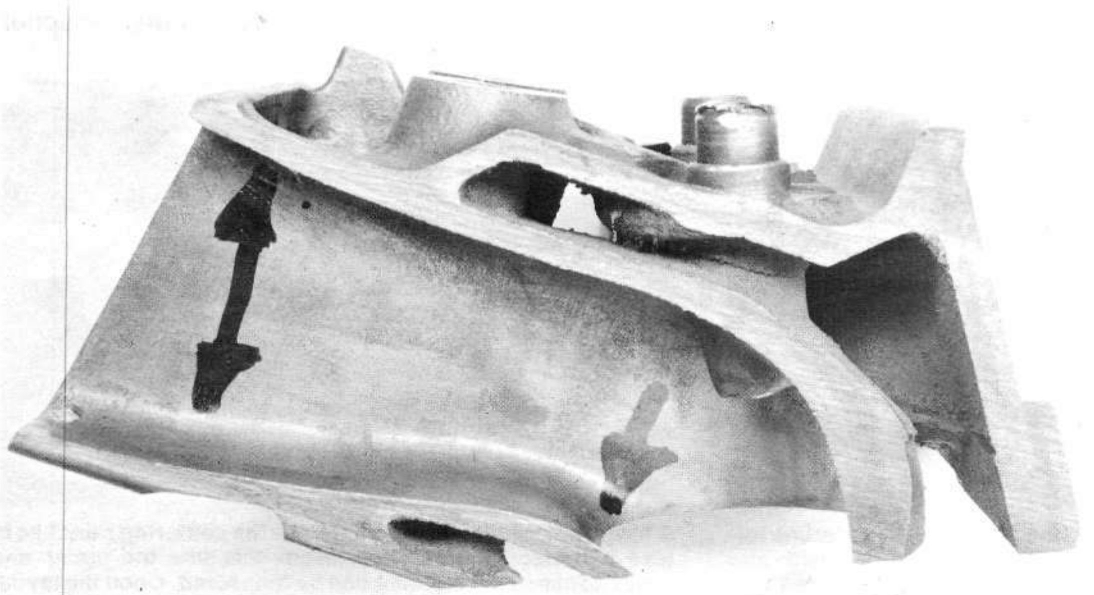


These ports are well-prepped 292's that have not been welded. Note the longer radius on the divider-side of the floor blend. It is difficult to see here but the blend on the opposite of the floor is much smaller, all the way along the exhaust-side wall (also called the long wall).

wall. Current core molds call for some water space between the waterside surface of the port wall and the bolt boss. If we had our way, this space would be eliminated and the intake port exhaust-side wall could be pushed over right next to the bolt boss.

A Pro Stock 330 needs all the grinding this wall will take and more. We have gone so far as to mill this complete wall away, all the way into the bowl, and to weld in a new piece of metal. It is possible to move the wall right against the bolt boss when you do this. With grinding and shaping this makes a very large port and the high-rpm engines seem to respond to this treatment, up to a point. Recent testing indicates it is possible to move the wall too far away from the centerline. When the wall is moved, the roof is raised, and the floor is lowered; it is possible to gain too much volume for the engine displacement. We are not certain exactly where the fall-off point is, but our best 330's can use as much as 2.85 square inches of area at the crown (this is the measured area of a cross section taken perpendicular to the major axis of the port, at the high point of the floor).

The other major restrictions on the exhaust-side walls are the pushrod openings on either side of the port pair. The intake port flow will be improved by opening this "neck-down" as much as possible. Grinding the wall away, until it becomes as thin as the casting and your nerves allow, will help. For our engines this is still not enough area. All of our ports have been welded along this side wall and ground to the widest cross section achievable. In conjunction with this work we offset the pushrods to move them further away from the outside walls of the port. We can see nothing but better results by opening the port more and more at this section. As new valvetrain gear, such as rocker shafts, make it possible to spread the pushrods, we will continue to weld and widen the ports as much as we can. Of course, it may be possible to make the entry too wide, just as we have been able to move the exhaust-side wall over too far, but it



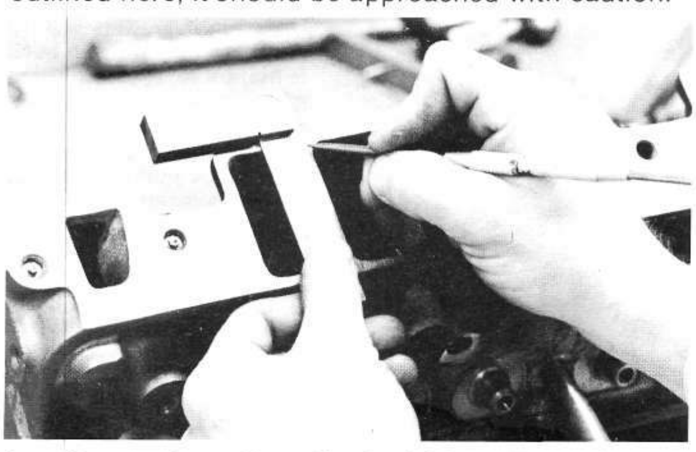
This is a 292 intake port section, cut roughly along the centerline of the port. Unfortunately, this isn't a particularly good port but it is helpful for locating some of the areas we have been discussing. The marking on the left indicates the portion of the port that can be modified extensively for increased volume. The roof or the floor can be ground or cut away and reformed to gain as much volume as you care to achieve at this point. The marking on the right shows the divider-side floor blend that should be filled and/or ground to increase the radius.

For maximum effect the intake port openings must be matched to the intake manifold ports. To make all heads and manifolds interchangeable we try to lay out all port openings to one common length and width. For special cases the openings must be matched to specific manifolds.

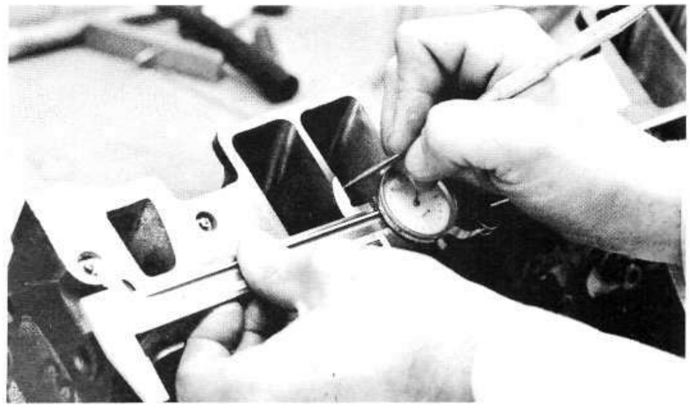
seems we are still quite a way from this point.

Volume may also be gained by raising the roof or lowering the floor of the port. The smallblock cylinder heads are designed for use with relatively low single four-barrel manifolds. When a tunnel-ram manifold is used, there must be some angularity between the head port and the manifold port as it rises up to the plenum. By angle-milling the head this angularity can be reduced. Raising the roof of the port will further reduce this angularity as will welding/grinding the roof of the manifold port. Our largest ports have been raised to the point that the roof protrudes up into the valve cover gasket surface. These, again, are the ports that may have been a little too big for the engine.

Lowering the floor is also largely an attempt to gain more volume. The crown must not be lowered as this feature is important for the initiation of the flow downward into the cylinder. If the port floor is dropped, this should be accomplished well in front of this point. Between the flange and a point about 2 inches into the port it is possible to drop the floor a good deal. Past this area it should rise toward the high point. Shaping of this "pot belly" is very important and, like all of the radical surface treatments outlined here, it should be approached with caution.



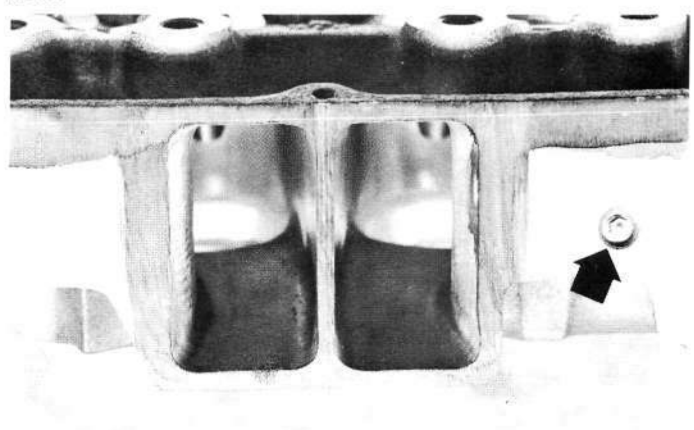
A small square is used to scribe the divider-wall centerline reference on the manifold face. From this reference the inside and outside vertical sides of the opening can be measured and scribed, parallel to the centerline.



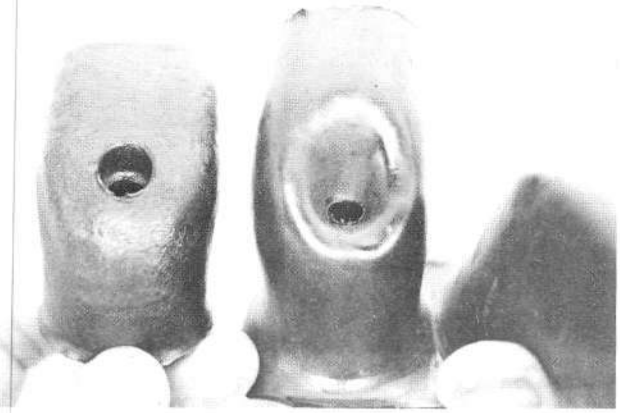
The layout begins by locating the centerline of the divider wall and the centerpoint of the end manifold mounting holes. The centerlines of 1-3 and 6-8 pairs are 5.03 inches from the end hole. The 2-4 and 5-7 pairs are 5.01 inches from the corresponding holes.

At the flange we make certain the port will match with our intake manifold(s). The port entries are laid out on the flange of each head, and they are identical from one to the other. This is done by locating the centerline of each cylinder-pair divider wall (giving a vertical reference) and the centerline of the manifold bolt holes (giving a horizontal reference). Centering punches are threaded into the manifold mounting holes and the divider centerline (as given on factory blueprints) is measured from these points. The divider centerlines of the 1-3 and 6-8 pairs are 5.03 inches from the centers of the end holes. The divider centerlines of the 2-4 and 5-7 pairs are 5.01 inches from the corresponding end holes. With these reference lines established it is possible to lay out the port openings in a precise manner.

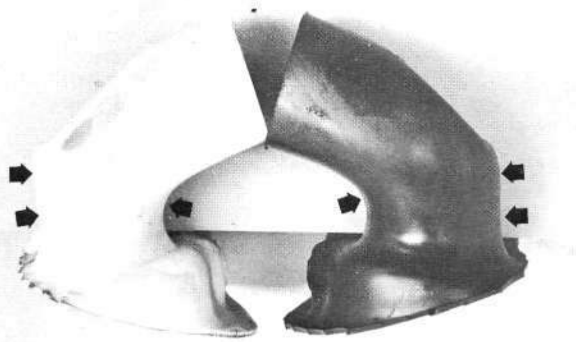
Because of variations in port volumes/inlet openings it is difficult to gain an exact match between all of the heads and manifolds. In all instances it is important that the roof of the manifold match as closely as possible with the roof of the intake port. Any mismatch, especially if the head port hangs down below the manifold port (forming a ledge to impede the incoming flow), will distinctly hurt induction flow.



A long straightedge is used between the centering punches to give a horizontal reference. From this line the upper and lower edges of the opening can be measured. Once the layout is completed the opening is ground to the scribed lines.



This is another comparison between a Brownfield aluminum port (left) and a well-worked Chevy 292 exhaust (right). Our tests indicate that the Brownfield port is an extremely good port at all valve lifts. The port is centered much better on the valve than the 292 port. The water core of the 292 head will not allow the port to be centered as well.



The port on the left is from a 492 casting, and on the left is a 292 port. The vertical drop from the guide to the bowl as seen on the left mold will flow well at low- to mid-lift of the valve. When the long radius is pulled out toward the port centerline, as on the right, the port flows better at mid- to high-valve lift.

EXHAUST PORT MODIFICATIONS

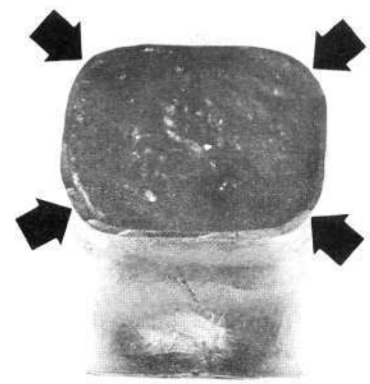
The most important work in the exhaust port is the roll of the shortside (floor) radius. The as-cast radius need not be altered drastically but the floor should be as "wide" as possible. The corner radii can be shortened to facilitate this spread. It is particularly desirable to widen toward the intake-side wall. Once the floor has passed the radius leading out of the exhaust bowl, it is not necessary to spend more effort increasing the width. At this point the shape of the floor will have little effect.

Inside the exhaust port, shaping of the backside wall, the long radius (roof), determines the flow characteristics of the port. If the wall has a fairly vertical drop from deep behind the valve guide to the seat, the port will flow well at low- to mid-lift. If you straighten this wall out, filling in behind the guide, so the wall drops down from the backside of the guide straight to the inside radius of the bowl, the port will not flow well at low- to mid-lift but it really takes off at high-lift. Depending upon the engine requirements, some shape between these two extremes will give the best performance. Finding the right compromise on the backside is the secret to getting the exhaust port to flow at all lifts.

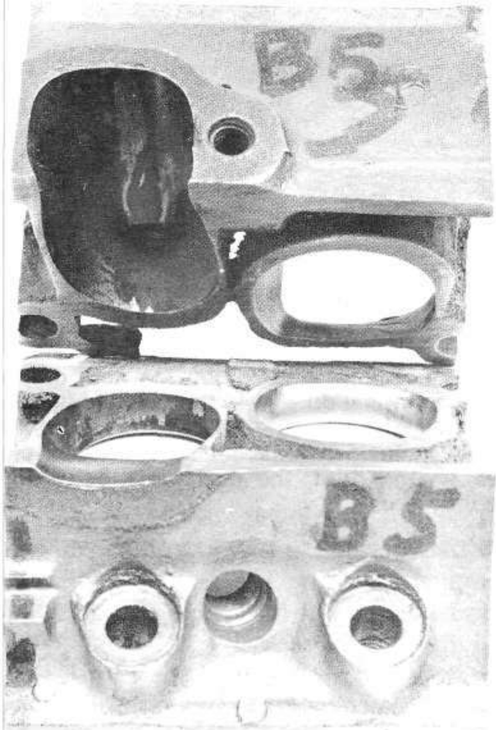
Once the flow has moved past the turn it will tend to rise closer to the roof. From this point outward the as-cast shape should be retained, but metal may be removed to increase the size/volume. There is not a great deal of extra material on the waterside surface of the 292 exhaust port, so it is wise to approach this grinding with some caution unless you can afford to pay the price when the wall gets "too thin." There is often a "paste ooze" line along the side of the exhaust port. This occurs during factory casting. If you grind through at this area it is easy to repair.

We have not done extensive welding and wall movement in the 292 exhaust port. The basic casting works very well and it is not really necessary to do this expensive work. We don't recommend a definite round shape to the port exit as we feel the square shaping with short corner blends provides the best

A side view of the Brownfield exhaust port mold. The short radius and "spread of the floor" give this port the ability to move the exhaust flow very efficiently. The long radius seems to be an excellent compromise to give extraordinary all-around flow.

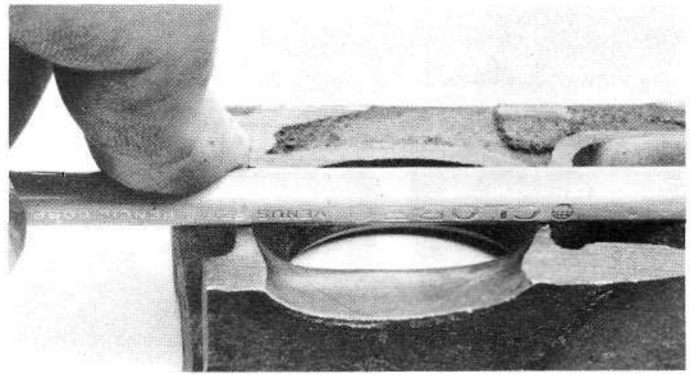


Looking from the flange end of the Brownfield mold we can see how well the port is centered on the valve opening. Note also that the port is definitely square with fairly small corner radii. It would be extremely difficult to get a 292 exhaust port to out-perform this port.



This section of a 292 casting has been cut open to show the interior of the ports and the relative shaping of the water coring.

flow. We do not make an attempt to match the port exit to the exhaust pipe. It is better to have a mismatch at this point to reduce exhaust backflow. There may be some slight gain, however, in raising the centerline of the exhaust pipe above the centerline of the head port. With the tube elevated some, the activity near the roof of the port will exit more nearly in line with the center of the header.



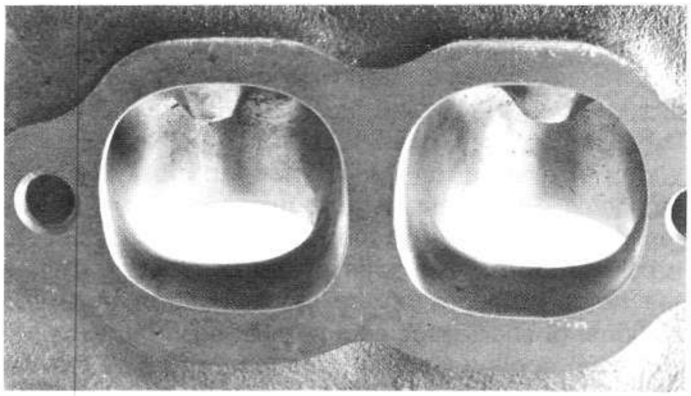
A close view of the exhaust port sections shows what we feel is an almost ideal floor shape. The floor is relatively flat but not absolutely flat. The corners are spread apart and the radii are shortened. If the floor is dead flat with more spread than shown here the port flow will have pulsing characteristics at valve lifts in the neighborhood of 0.300- to 0.400-inch.

MACHINING DETAILS

If the stock valves are to be used (for street application only) they can be fitted into the stock stem guides as long as the measured clearance is within 0.0015- to 0.0025-inch. If the clearance caused by excessive wear is more than this, the guides should be bushed or otherwise repaired. In the case that small-stem valves are desired, the guides will have to be sleeved down. In either instance we prefer to use bronze inserts of the popular Manley/K-Line replaceable type. It is possible to run tight stem clearance with this guide and they can be replaced if the clearance opens up or if the guides are damaged by some valvetrain mishap.

If oversize valve springs are to be used, the valve spring seats will have to be milled larger than stock. The latest 292 heads have enough material under and around the seats to accommodate a very sizable spring. When machining the pockets on early

When the 292 port is split open and the upper and lower halves are placed side by side, the relative thickness of the vertical walls can be observed. In nearly all of these castings the walls are very thin. Enlarging the port will often lead to breakout into the water jacket. There is usually more metal on the wall opposite the intake port. This is the area to grind if you want to open the port a bit.



Looking into a completed port, from the flange, it is possible to see the desired shape of the floor. The port must not be "round." Spreading the floor, back in the port, by the short turn is important to keep from squeezing the flow toward the port centerline. As the flow gets closer to the flange it will move across the port, toward the roof. Spreading the roof near the flange is, therefore, important.

heads they should not be cut deeper, as there is very little metal in this area. The pockets on the 292 heads may be cut deeper, but only 0.135-inch and no more. At this depth the maximum diameter to which the pockets on the 292 heads can be cut is 1.55 inches. Care must be taken that the pocket cut does not serve as a point for cracks to begin. The bottom corner of the pocket cut should have a 0.025-inch wide by 45° chamfer or there should be at least 0.010-inch radius left on this circular edge by the mill cutter.

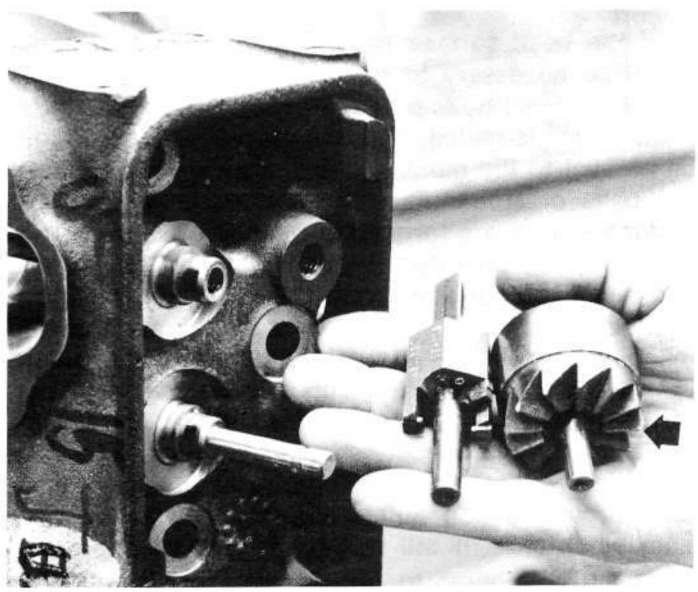
The large intake springs we prefer to use (see camshafts) require considerable enlargement of the seats. We have not had any trouble with the spring pads breaking. Some springs may also require that the stem guide extension be reduced slightly to insure proper inside clearance with the spring. It is important that the retainer does not contact the guide extension at full valve lift/spring compression. If possible, we do not run springs seated directly against the head surface. Were there sufficient room we would run a hardened-steel cup under the springs to center them around the guide. Unfortunately, there generally isn't enough room on the smallblock head to use such cups. In lieu of the cups, we try to use at least one hardened-steel shim under each spring to insure that the inner damper will not cut into the spring pad of the cast-iron head. In the Pro Stock racing engines we have a very difficult time obtaining sufficient valve springs, and it is essential that every precaution be taken to gain spring fitting in the best manner possible. When machining the seats for big springs, some check should be made to guarantee that the spring has sufficient clearance inside the pocket when it is compressed and expanded.

While working over the top side it may also be necessary to spend some time on the rocker arm bosses. The 292 heads and most of the other Chevy high performance heads are already drilled and tapped to accept screw-in studs. These ⅜-inch smallblock studs are adequate for all but the most severe racing purposes. However, in our Pro Stock and Grand National engines we replace them with the 7/16-inch studs from the Chevy big block.

In most cases we opt for a 1.55:1 or a 1.65:1 rocker arm ratio. It is possible to go as high as 1.75:1 on the rocker ratio without trouble. Most of the specialty manufacturers are now making high-ratio rockers that can be used with the stock stud and valve locations. If spring interference is not a problem, there is no reason to relocate the studs away from the valve centerline. We do relocate the valves sideways in order to move the pushrods away from the sides of the intake ports. It is possible to move the stud sideways 0.050-inch by milling the side of the 7/16-inch threaded stud hole and filling it with a thread coil. We have often moved the studs more than this. In this case, the technique is to fill the stock hole with some sort of threaded material (we use a 7/16-inch bolt and cut the head off) and redrill a new hole in the desired location.

When additional clearance is required between the underside of the rocker arm and the edge of the spring or spring retainer, the rocker arm may have to be moved up on the stud. When possible, it may be desirable to use a rocker which has a longer overall length, but this may lead to clearance problems with the rocker covers and other mechanical interference. Using a longer arm will reduce the amount of angularity through which the arm must operate. However, in these cases (when a longer overall rocker arm length is selected), the rocker stud will have to be relocated to gain proper rocker tip-to-valve stem contact.

We have found that there is a performance gain by decreasing the valve angle with relation to the

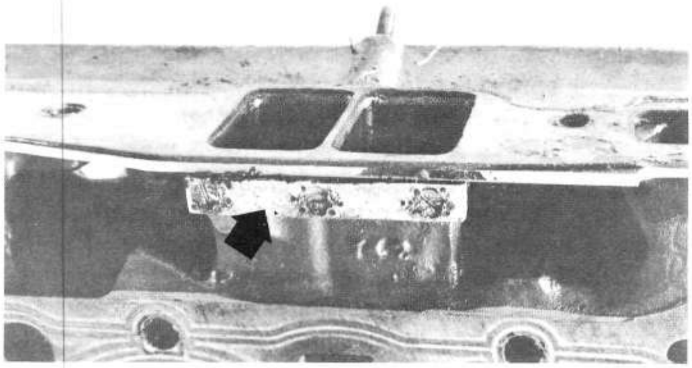


Machine preparation of the head must be coordinated with the particular spring and valve selection. We fit the valve guides with bronze inserts to fit the 5/16-inch stem valves. If large diameter springs are used (a necessity for Pro Stock racing), the spring seats must be enlarged. The edge of the cutter must leave a radiused corner at the bottom of the spring pocket to prevent cracking. The stem guide must also be shortened if valve seals are used with extreme lift cams.

cylinder centerlines. In other words, by angle-milling the head or block deck surfaces so more metal is removed from the exhaust-side than from the intake-side, the valves will stand more upright and the intake port angle is better. It is possible to cut as much as 0.150-inch from the exhaust-side of the deck, but this is very dangerous. If this much is removed with an angled cut, nothing at all can be taken off the intake-side of the deck. It is possible to angle mill the head without making a cut that reaches from one side completely across to the other. Several of our heads are milled this way. Often the cut will only reach as far as the approximate line of the upper row of head bolt bosses. When cutting on a severe angle this greatly reduces the possibility of breaking through the deck surface, but any cut removing more than 0.100-inch from the exhaust-side of the 292 head decks can be edgy.

further outboard on the head. This isn't too critical on the drag racing heads as it is certainly possible to run the heads for a short time with no plug cooling at all. However, it would be smart to have at least the exhaust-side hole open if this is possible. There is plenty of material on the 492 head to move the cooling holes out a considerable distance. There is less material in this area on the 292 head. It is possible to break through the outside of the head when drilling the holes outboard of the stock location. A Grand National or short-track engine must have the exhaust-side plug cooling holes in functional order to prevent detonation.

Whenever the head decks are angle-milled we also mill the intake manifold mounting surface to bring it back to the stock angle. Chevy blueprints call for the top of the manifold surface to lean away from the deck perpendicular at a 10° angle. As the deck is



When the floor of the port is cut down toward the lower edge of the casting, there will be very little gasket bearing area remaining to hold the intake gasket from blowing down toward the valley. We use a small aluminum extender block screwed to the bottom of the runners to increase the gasket surface.

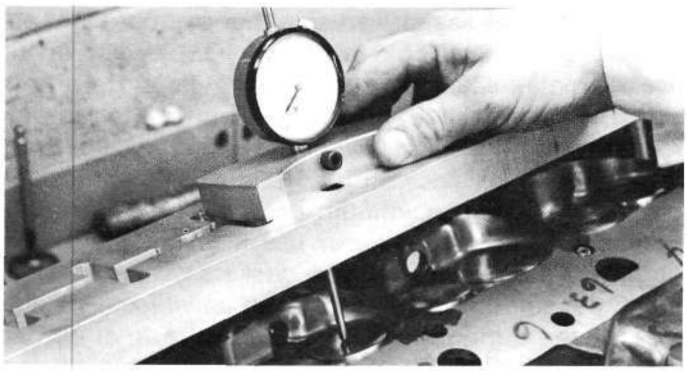
We often switch heads from one block to another as new combinations go together or as last-minute preparations are made. It is important to keep records to know which pieces will fit together. We have built a special head checking fixture to make this job easier.

If the head surface has been milled on an angle it will be necessary to reface the bolt-boss surfaces where the head bolts seat against the heads. When the head is angle-milled, these surfaces will no longer be parallel to the block decks. If they are not refaced, the bolts or stud nuts will not seat flat against the surface as they are torqued into place. As a result, the block will be distorted. Whenever the head surface has been angle-milled, or in the case that the block deck has been angle-milled, it is necessary that the machinist record the angular deviation from standard in order that an equal but opposite-sloping angle can be used for spot facing the bolt pads. This is important when the heads are used on a 4-inch case, but it is super-critical on a 4.125-inch case because block distortion is a constant struggle with the big-bore smallblock case. It will also be a good idea to redrill the head bolt bores to bring them back in line with the bolt bores in the block. When the proper head bolts or studs are used, there isn't very much clearance inside the bolt bores and misalignment with the block bolt holes will put an undesirable side load on the studs.

If the head is angle-milled very severely, the spark plug cooling holes may have to be filled and redrilled

angle-milled, the relative angle with the perpendicular will increase. We do not machine the manifolds to compensate for this variation. This would decrease the interchangeability and create all sorts of mounting problems. We remachine the head to bring this angle back to 10° but the two head surfaces will be further apart because of the metal removed at this time. To get the manifold and head surfaces properly mated, we often use two thick manifold gaskets between them. We have also used thin aluminum "stuffing plates" mounted to the head surfaces or simply sandwiched between the manifold and head. The stuffing plate also gives us additional material to form a transition between the manifold runner and head port. This may be particularly advantageous in a single-plane manifold where there is a severe angle in the transition from the runner to the head port.

One of the most difficult parameters to keep track of when the heads are angle-milled is the valve-to-piston. When the valve stem angle assumes a new relationship with the cylinder and consequently the piston head, the valve head will approach the piston dome at a different angle. If you are building only one engine this is not much of a problem, and the piston domes can be machined to accept the valve head with sufficient clearance to eliminate clash. However, we must have flexibility in our shop because cylinder-head assemblies are often swapped back and forth from one engine to the other. To make this possible, we have built a special fixture which allows us to compare the valve head and piston dome angles and total clearance in just a few minutes. It consists of an aluminum surface plate and mounting pedestal assembly that can be fitted directly on the head or block. Eight crisscrossed slots are cut in the plate, directly above the valve head locations. A dial indicator assembly slides on the top of the surface plate with the indicator arm extending down until it touches the valve head. As the indicator moves back and forth across the head in an end-to-end or side-to-side direction, the relative distance or angle to the surface plate can be read and recorded (the exact



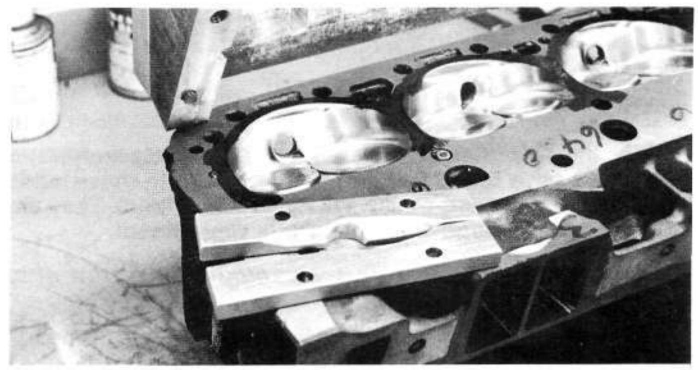
The fixture bolts directly to the cylinder head deck, allowing a sliding dial indicator setup to read directly off the valve heads. We can read all of the valve head heights, relative to the plane of the checking fixture and determine the valve head angles, relative to the fixture plane.

figure is unimportant as we are only comparing the two surfaces to the common plane of the surface plate). The plate locates to the head with dowel pin holes at either end, and adaptor plates allow the plate dowel pins to locate the plate assembly on the block deck. With the piston at TDC, the dial indicator is again used to read the relative angle and height of the valve pockets cut into the piston dome. With each comparable valve and dome angle measured relative to the surface plate, it is easy to compare the sets of figures and determine the exact corresponding angle and valve-to-piston clearance between the two surfaces. We measure each head/valve assembly as soon as it is finished, correct as necessary, and record the results for future reference. It is, therefore, possible to verify if a head assembly and block will be compatible. And, when building new short blocks, we can make certain the piston domes will be compatible (interchangeable) with all our head assemblies.

HEAD GASKETS

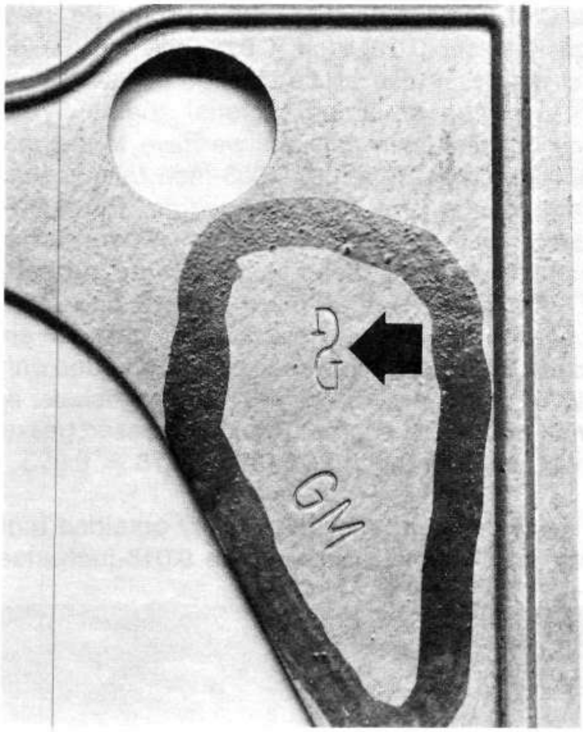
We haven't found any satisfactory gaskets other than the Chevrolet factory gaskets. With the 4-inch, 4.020-inch and 4.030-inch bores, one of the three available Chevy beaded steel gaskets should be used. These gaskets are 0.016-inch, 0.018-inch and 0.022-inch thick and they can all be stacked or used separately in any drag or Grand National engine. When using any of these shim gaskets we have found that they will "compress" to about 0.003-inch *thicker* than *their metal gauge size.* In other words, the 0.016-inch gasket will measure 0.019-inch when it has been installed and the head fasteners are properly torqued. The same is true for the 0.018 and 0.022 gaskets. However, when two of these gaskets are stacked, the 0.003-inch allowance need be added only once. When using two of the 0.018-inch gaskets, as is common on the GN engines, the compressed gasket height will be 0.039-inch (0.018 + 0.018 + 0.003 = 0.039).

The most common and most easily obtained high performance Chevrolet gasket is the 0.018-inch steel



The adaptor plates allow us to bolt the fixture onto the deck of a block assembly to measure the angle of the piston dome. When the heads have been angle-milled, the valve heads will approach the piston dome at a different angle and the calculated valve-to-piston clearance will not be a true.

shim, available as Chevy part 3916336. This is the gasket listed in the Chevy Heavy Duty parts program. It is manufactured for GM by the Victor Gasket Company and can be identified by the small square cutout in the upper right-hand corner. The 0.022-inch gasket is available as part 3830711. The gasket is made for GM by three different vendors. When you order this number, about 95% of the time you will get the gasket which is made for GM by Detroit Gasket. We only use the gasket made by Detroit Gasket and don't recommend those made by any other vendor. This 0.022-inch gasket is supposedly serviced in the rebuild gasket sets, but generally the gasket included in the kit is not the Detroit Gasket part. The Detroit Gasket unit is easily identified by the "DG" letter identifier stamped in the gasket tails. The 0.016-inch gasket is sometimes difficult to obtain but should be available as part 3783631. This is a gasket once used by the GM marine engine division and is not generally serviced to the automotive division. We obtain it through special negotations with GM Parts and Accessories Division, but we have heard that at one time it could be ordered through Chris Craft Marine dealers as a replacement gasket for marine 327 engines. This gasket is also manufact-



This is the identifier stamping for a Detroit Gasket-brand gasket. We have found these gaskets superior to those made by other suppliers and we always recommend them. They are only available through Chevrolet Parts Department.

ured by Detroit Gasket and we recommend that only the DG-make be used.

By using the gaskets individually or in various combinations it is possible to gain a wide degree of shim height variation when installing the heads. With the 4-inch bore bowl cutout in the gasket, every 0.005-inch of installed gasket height will increase or decrease the chamber volume by one cubic centimeter. With the permutations available we can vary the shim height from 0.019- to 0.047-inch (compressed) in nine different increments.

For the smaller bore engines we recommend the 307 c.i. steel shim gasket, available as part 3995633. This gasket is 0.016-inch thick, 0.019-inch compressed, and is suitable for the 3⅞-inch through 3-15/16-inch bore engines. It is available as a Detroit Gasket brand and should not be used with the 4-inch (or larger) bore engines and heads as it may overhang into the chamber opening and restrict breathing. There are no other steel shim gaskets suitable for the small bore engines as far as we know. It is commonly known that cylinder head sealing isn't a problem with the small bore engines, and the reason is clear in the gasket comparisons. The 307 gasket has a noticeable increase in material between the adjacent cylinder bores, decreasing the likelihood of blowout failure.

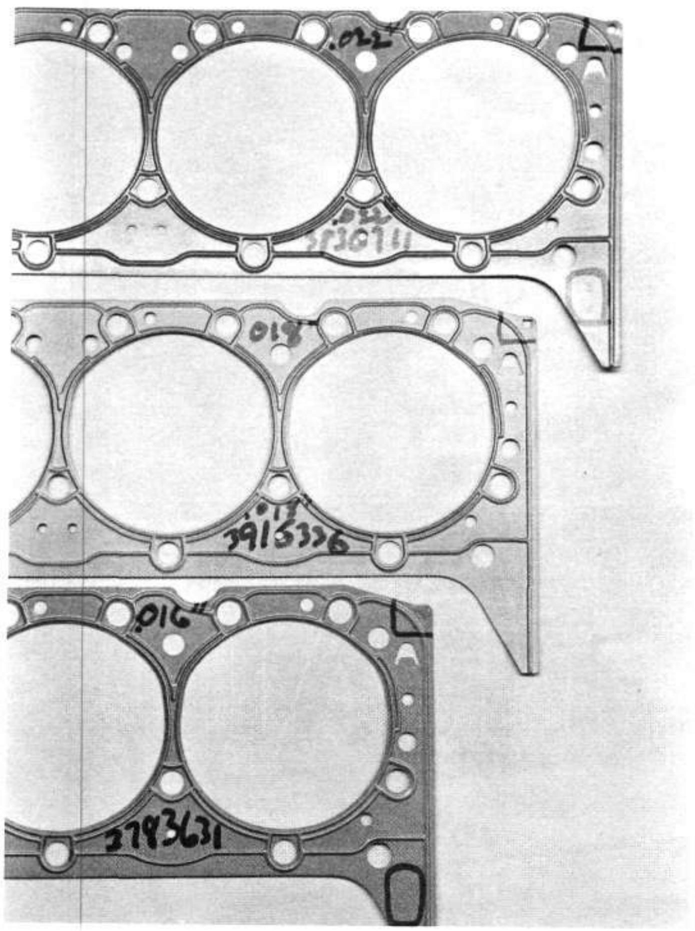
Chevrolet offers only one usable gasket for the 4.125 bore cases. This is listed as part 3965790, and is a Victor Unitorque, composition, semi-sandwich gasket of steel and asbestos. It compresses to 0.038-inch, though that's not what it measures out-of-the-box. We don't feel this is as good a gasket as the steel shim numbers but it is the only gasket available for the big-bore engines. Supposedly, a great deal of testing and development has gone into the piece. We have only done limited testing with this gasket, but in some of our match race engines they have performed very well, except in the one instance when we drove the engine into severe detonation and blew the ends right out of the gasket. With big cylinder bore holes there just isn't very much material left between the cylinders or on the ends of this gasket. This will almost always result in marginal sealing. We, therefore, have tried using studs with these blocks and so far the combination has not given us cause for much concern. We have not gone to O-rings but it may be necessary if chronic trouble begins. This gasket should never be used in a 4-inch case. It will leave a thin slot all the way around the bore where heavy dead air can collect and it will not seal as well as the standard 4-inch gaskets.

For a gasket sealer we use either standard GM accessory sealer available from any GM dealer or we use Permatex 300. We use a sparing amount on both sides of each gasket and install them in the factory-recommended fashion. With the 4.125 gasket or a slightly rougher deck finish, it is possible that the old trick of using aluminum paint for a sealer might be acceptable, but we never use it on the steel shim gaskets. When using the steel gaskets we also recommend the smoothest possible head and case deck surface. If the surfaces are ground on the final cut, so much the better. Remember, here we are talking about very high compression racing engines which are subjected to unreasonable stress. For stock or street high performance engines a normal surface preparation will be adequate. If you aren't currently having any trouble with head gaskets, there is little reason to change procedure. If you are having trouble, we would recommend that you check the quality of the deck surface (the smoother, the better), the type of gasket and the type of sealer used, before resorting to studs and/or O-rings.

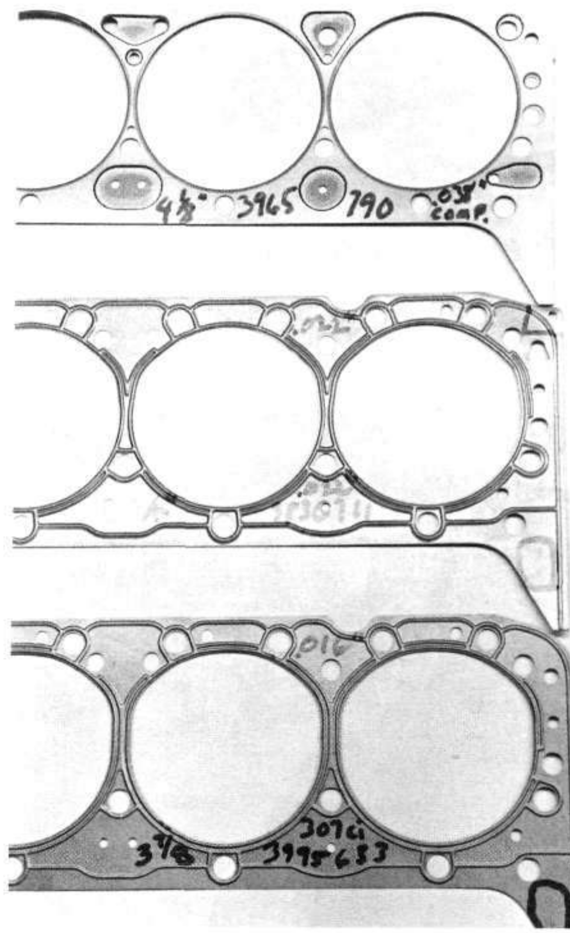
We use stock Chevrolet 1041 steel case-hardened bolts to hold the heads on the 4-inch cases. GM calls this material 300M steel and it has a very straight stress-strain curve. The stock bolts are easily identified by the 4-point cross on the head. Because of the excellent material, these bolts will not stretch in the threaded section before the rest of the bolt begins to stretch. We have never found any specialty bolt that will do a better job than these bolts. We use washers under the head bolts and, as outlined earlier, we make certain the bolt pads are spot-faced absolutely perpendicular to the threaded bores in the block. When using head studs in the 400 case, we always install them into epoxy-filled threads. When the heads are fitted in place over the studs, they will almost always have some interference, pulling the studs off at an angle. The stud threads will not have a full-length perch in the female block thread. This effectively defeats the purpose of using studs. To gain a full perch, we fill the block threads with Devcon epoxy and install the studs until they are hand tight in the block. The heads are immediately put in place and the stud nuts are tightened down to about 5 ft-lb while the epoxy is allowed to cure. Hopefully, this method will prevent anything from cracking when the heads are torqued and will distribute the load along the full length of the threads.

It is also possible that the threads of a head bolt or stud will interfere with the outer surface of the 400 block cylinder wall if the subject bolt/stud is screwed too deeply into the case. If this happens and the bolts/studs are given a side load as the heads are in place, they will certainly push the thin cylinder walls out of shape. To prevent this, we recommend that the bolts/studs be shortened so they don't extend down into the block any deeper than the head deck. If you are using studs on a racing engine, make a mock-up fit to insure that the row of short studs along the outboard edge of the head will not interfere with the header flange when the nuts are fitted to the ends of the studs. They may have to be shortened some. We have found it is best to do this before setting the studs into the head with epoxy!

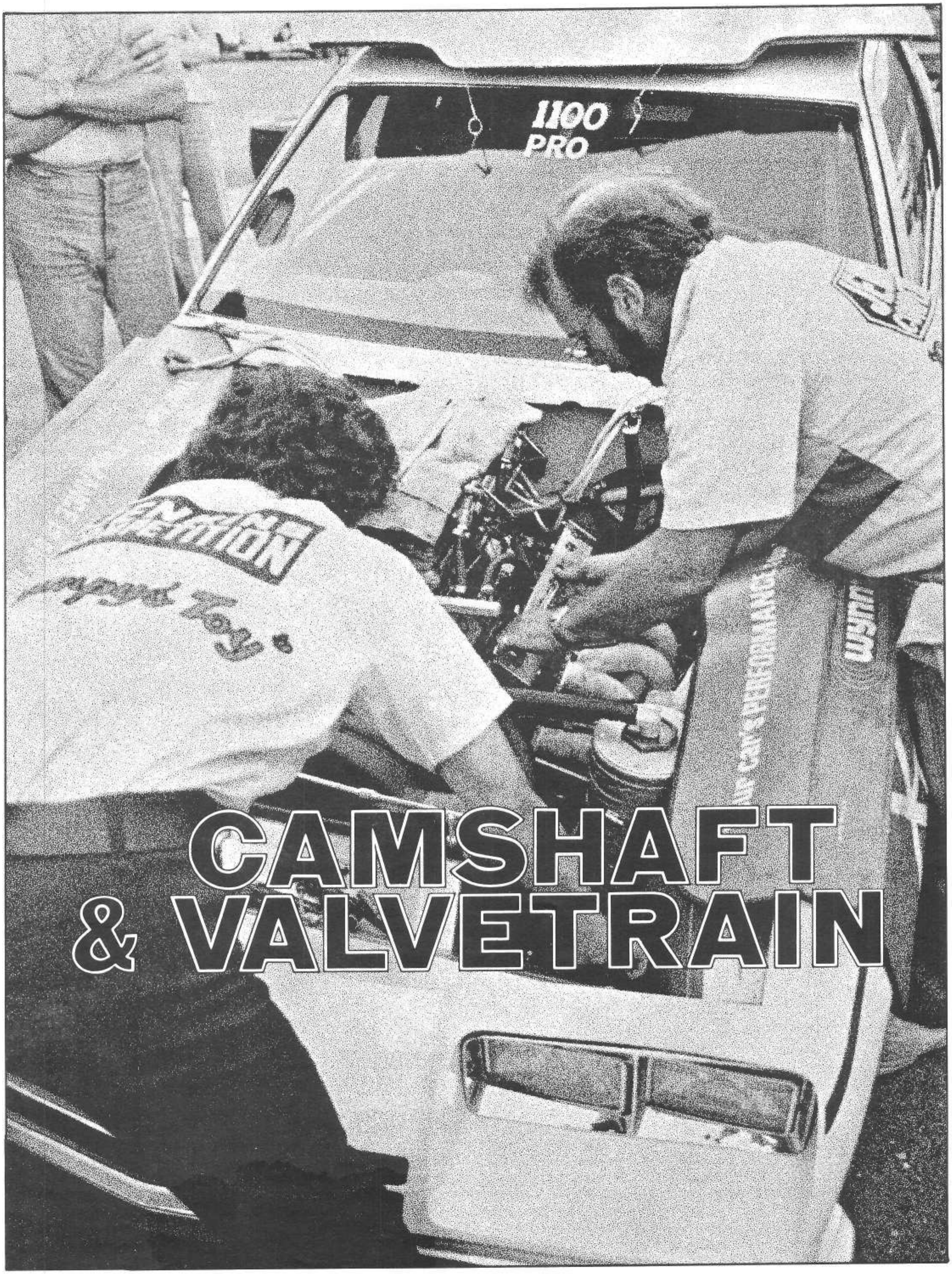
Even with the best precautions we have found that the little-block decks can get pulled around pretty easily. It's especially troublesome if double-stacked head gaskets or some sort of "double throw-down" trick gaskets are used. When this happens you might as well count on the decks looking like the Rocky Mountains after the heads are torqued in place. This will probably cause subsequent gasket leakage trouble, especially between the middle two cylinders or up into the valley. When you are right "on size" and don't want to recut the deck, this can be trouble. We have, however, been able to successfully cut these little humps down with a hand-filing technique. We use a 10-inch, fine-tooth hand file laid across two of the pulled bolt bores. By carefully filing the little humps flat, it is possible to regain an adequately smooth surface without resorting to another machine cut. This must be done carefully and the file should always be drawn across at least two bolt bores in order to keep it flat with the deck.



These are the three steel shim gaskets available for the 4-inch bore engine. At the top is a 0.022-inch service ***replacement*** **gasket, part 3830711. It has a round identifier hole in the corner. In the center is the 0.018-inch stainless shim built by Victor, available as part 3916336. On the bottom is the OE service 0.016-inch gasket used in factory assemblies. This piece has part number 3783631, but is not available as an order item except, possibly, through Chris Craft marine dealers as a replacement for marine 327-inch engines. These gaskets can be used individually or stacked as required.**



This is a comparison of gaskets used for different bore sizes. ***At the top is the Victor Unitorque composition gasket for the*** **4.125-inch bore. It is available as part 3965790 and is the only Chevrolet gasket offered for this bore size. It should not be used on smaller bore. Next is the 0.022-inch standard gasket for the 4-inch bore. On the bottom is the shim gasket currently-available and recommended for 3⅞-inch bore blocks. It is available as part 3995633 and should not be used with larger bore blocks as it may overhang into the bore and impede air flow into the chamber.**



CAMSHAFT & VALVETRAIN

As the cam requirements become more radical, special grinding techniques must be used. When lift is increased the diameter of the base circle must be decreased to obtain the required lobe height. Consequently, the overall strength of the cam is decreased. With current valve spring loads the cam can pick up some torsional twist, in addition to the harmonic vibration transferred to the cam by the timing chain. To compensate we use semi-relieved and non-relieved grinding techniques.

Selecting the proper cam design and valvetrain gear is one of the most critical decisions any engine builder faces. The ultimate choice will be very closely tied to the induction efficiency achieved by the carb, manifold/port work and the rod ratio of the engine. Exhaust efficiency also enters the picture, but in most cases there is far less tuning work on the exhaust side of the engine. The main concern will be to use the cam design in whatever way is necessary to balance the intake and exhaust efficiency.

At this point in our particular racing program we have a good idea what the drag race engines need to make everything else work. We have some new ideas which have yet to be explored but what gains may be left in the cam design are really very minimal. The Grand National picture is a little less obvious. In the long distance engines we face a much different set of problems. It is certainly possible that better cam and valvetrain design will show considerable gains in the circle track smallblocks.

CAM SELECTION

The basic design selection may be dictated by the particular sanctioning organization rules. In the NHRA and IHRA legal drag engines and the match race engines we are allowed to run cams with roller followers. The Nascar engines are restricted to flat followers. Most of the sport and sedan racing categories require flat-tappet designs, with the notable exception of the SCCA Formula 5000 series which currently allows rollers.

We much prefer to use a roller cam in any unlimited class engine. The initial expense is slightly greater, but there are many advantages to using roller-tipped followers in a racing engine. Generally, a roller cam can be ground with greater maximum valve velocity than a flat follower cam. This allows the builder to gain more area under the curve with less duration. The phrase "area under the curve" refers to the total area under the valve displacement curve, a plotted curve of the valve opening and closing action. Any time the area is increased through lengthening, raising, or reshaping the curve, the theoretical breathing capacity of the engine should be improved. In practice this may not be totally true, for a variety of reasons, but for our purposes it can be considered a valid generality.

This doesn't necessarily mean that running rollers is a bed of roses. They also have some inherent design problems. Roller designs have trouble with side loading in the tappet bores and a limitation on positive acceleration. There are many controversial philosophies of racing cam design and it isn't our intention to compare them. But the nature of a roller-tipped lifter makes it not want to cope with extremely high positive acceleration rates during the valve opening sequence. It's possible to really smack a flat lifter with the cam lobe and impart a very high positive acceleration to the follower. It is also possible to hit a roller lifter pretty hard with the side of the lobe, but

Close view of a typical full non-relieved cam shows how the lobes are ground wider in order to use the overlap of the material for additional strength. This reduces but does not totally eliminate camshaft twisting and flexing.

Manufacturing a non-relieved cam is an expensive and difficult process. The 8620 steel core must be semi-finished, straightened before final finish shaping. Such prep is only required when lobe height reaches 0.450- to 0.470-inch. If the requisite valve lift calls for more lobe than this, we feel the rocker ratio should be raised and the base circle of the core should not be reduced any further, despite non-relieved grinding.

this will require a difficult hollow-ground lobe technique. Since the lobe will be acting against the side of the roller tip in this action, it also puts a side load on the follower body and roller pin. However, once the lifter/lobe mechanism moves past the opening contact point and constant velocity ramp (if one exists), it can have a higher maximum velocity than a solid-lifter design (discounting mushroom-lifter designs). In the same manner, all of these forces act on the backside of the displacement curve during the negative/positive acceleration of the closing sequence.

Other problems can spring from incorrect cam/valvetrain design. If the negative-acceleration portion at wide-open valve is too severe for the counterbalancing force of the valve spring, the valvetrain may pull apart. As a result, the follower loses contact with the lobe and the entire mechanism goes into the condition we call uncontrolled "valve float." This is not very common with modern camshaft designs. A similar effect, called "valve bounce," may occur as the valve is returned to the seat. In this instance, the "spring rate" of the entire valve-actuating mechanism may be too low and combine with the positive acceleration portion of the closing sequence, allowing the valve to hit the seat too hard and bounce away again. This bounce is also the result of the natural elasticity of the metal used in the valve construction.

The roller designs are also much less susceptible to wear. With any cam, the shape of the nose radius is very critical. A roller cam can have a higher nose height, and greater negative acceleration than a flat tappet design. When maximum power is being sought at extreme engine speeds, negative-acceleration rates are very critical. To attain the negative rates we need, the valve-spring pressure must be very high and this would lead to very high wear failure with conventional flat followers. This is true despite the fact that it is now required practice to use light break-in springs when high-rate flat tappet designs are used in a racing engine. Subsequent failures are still more prevalent with flat-follower cams, despite the most meticulous break-in.

We will go into more specifics about springs later, but at this point it is worth mentioning that one of the single most common mistakes with roller-lifter cams is not running enough spring tension. Sometimes you find yourself in the situation where things start breaking up in the valvetrain. It's natural to think this might be the result of "too much spring." In fact, it's exactly the opposite. Whenever we begin breaking lifter buttons, pulling valve heads, and shearing keys, we inevitably find the spring tension has fallen off. Such failures are the result of the valvetrain "springing apart" at high speed and hammering back together when the spring finally catches up with the action. If these symptoms are encountered with springs that check to spec, you better look for some better springs and/or get the tension higher.

In some cases the type of material from which the cam core is made may have a bearing on the engine performance. We are currently fighting a very severe torsional twist problem with our drag race cams. This trouble manifests as 4-6° timing variations from one cylinder to the next with a camshaft that is otherwise perfect. It's easy to find this with a timing light. In one instance we checked an engine in which we had a thoroughly-checked camshaft. The timing was set at 40° at 3000 rpm and the engine was run through a speed cycle. At 5500 the timing read 36° and at 8500 it jumped up to 44°. Then we put the timing light on another cylinder and ran the same test. The timing "scatter" or pattern was entirely different throughout the range.

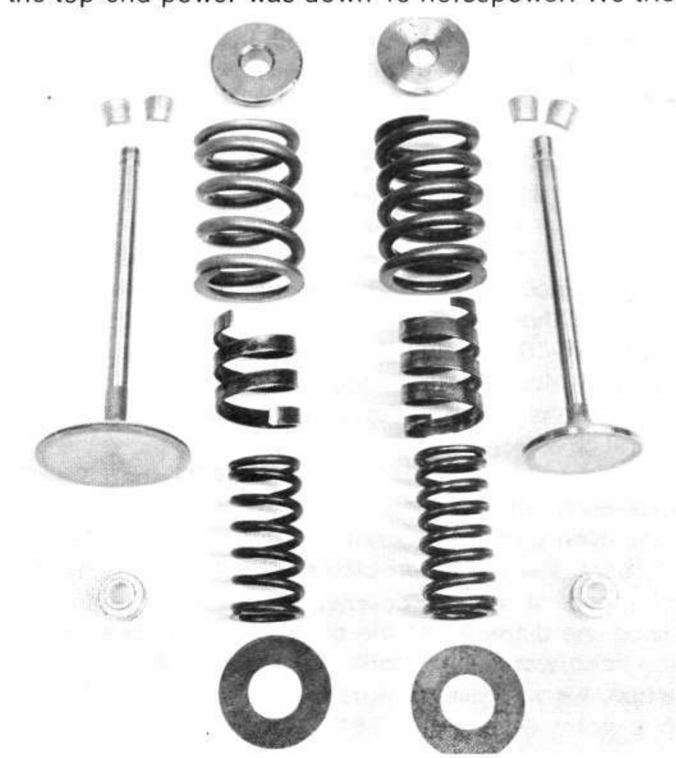
In another test we had a complete engine on the stand in the assembly room with a good cam installed. We could turn the crank through exactly 180° and the back end of the cam would turn only 86°. Then, we would pull the crank through another 180° and the back would turn 94°. In many tests it was not uncommon to find this "windup and let go" in the range of 3-4°. The problem is caused by the very small base-circle size used on the high-lift cams combined with the extreme spring loading. However, we never encountered the problem with the older nodular-iron cores. So, it may be that the current trend to 8620 steel-billet cores has at least aggravated the torsional problem.

We chased this around for quite some time, but the only satisfactory solution has been to use non-relieved lobe grinding. This technique has proven extremely satisfactory to dampen the torsional vibration in our drag racing 330- and 354-inch engines. In any case, we definitely recommend using some sort of cam-drive mechanism that will help reduce the amount of vibration transferred from the crank to the cam. The non-relieved and semi-relieved 0.875-inch wide lobe cams pictured here were especially ground for our shop by General Kinetics and Cam Dynamics. These are very expensive cams to produce and they take a long time to prepare. The 8620 steel cores must be semi-finished to very near final size. The cam is then heat-treated and straightened. Then, the journals and lobes are all carefully finish-ground to the final diameters and shapes. This technique has helped considerably and we are reluctant to reduce the base circle any more than that currently used. In the future this limitation will force us to use larger rocker-arm ratios if we need more net valve lift, or increase the cam-journal size, so the required lobe lift can be ground on a larger base circle.

DURATION

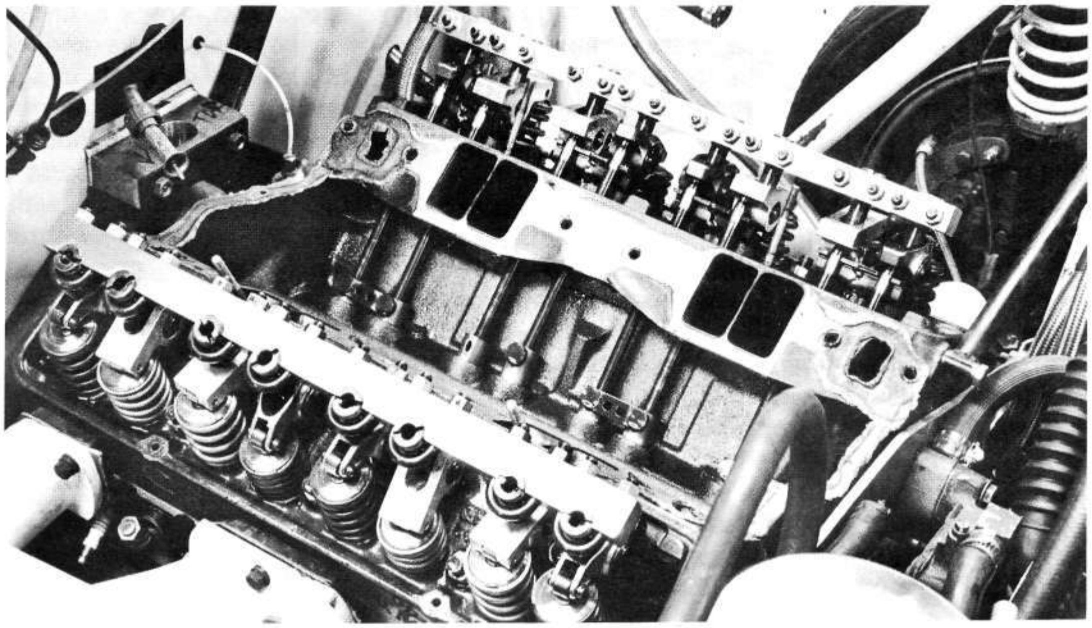
There are many things to be considered when selecting a cam duration for a specific engine size and application. There is always a need for some balancing between intake and exhaust flow conditions and what valve duration is selected. Generally, the flow quality in the mid-lift range will give the strongest indications of what duration is required. Heads with lazy mid-lift intake ports (0.300- to 0.400-inch valve lift) that come back with good flow in the 0.600- to 0.700-inch range will require more duration. If you are not getting good low- and mid-lift flow from the heads, you're just not going to hurt the engine with a lot of timing. This was most dramatically illustrated in a series of tests we ran early in the 330-inch development program. We were testing flat-tappet cams on the dyno and we found a substantial top-end power increase when the right ports and cam timing were combined. We had a set of heads with very bad mid-lift flow and very good high-lift flow. The exhausts were also good at high lift. When we put a General Kinetics grind in the engine with about 286° duration, measured at 0.050-inch lifter travel, the engine really perked up. At that point in the game this was considered to be quite a lot of duration. The cross combination, good mid-lift, bad high-lift flow and short duration was a total disaster, but it got us to thinking. Any combination with bad flow throughout the lift curve or bad high-lift flow could be eliminated. However, importance of mid-lift flow efficiency, or rather the lack of it, was beginning to emerge. So, in the same series of tests we tried a set of heads with better mid-lift flow, good high-lift flow and a cam with 10° less intake duration. This produced a flatter power curve and more torque but the top-end power was down 15 horsepower. We tried

All components of a very high-rpm valvetrain must be carefully selected. A Pro Stock smallblock must turn 9,500-10,000 rpm and such speeds put exceptional demands on every single piece. Big springs, roller rockers and lifters are required.



Two of the most important components are the valve springs and valves. The valves must be as light as possible and large diameter, double springs of high-strength material with good durability are important.

The secret to producing power from a racing smallblock lies in the proper preparation of the cylinder heads and induction. The valvetrain is a key part of this formula. Modern design has allowed valve action far beyond past performance. However, increased instability and strain are constant enemies. Note here that one of the tie links has broken. Constant inspection is essential to prevent minor mishaps from becoming major disasters.



a long duration cam in the engine. This killed the bottom end completely and we never could get the horsepower to come back. In the last attempt we tried the heads with low mid-lift and good high-lift flow with a short duration cam. This brought the low speed torque up but we wound up losing 25 horsepower.

Cam lobe duration isn't a particularly accurate parameter for cam design comparison, but it is the most common. Any duration figure is meaningless unless that point in the lift curve at which the duration figures were taken is known. This is a long-standing argument among cam grinders and engine builders, an argument which probably will never be resolved. Nearly every major maker has a cam or two that falls pretty close to what we use in our competition engines, but finding it may be a little difficult if we aren't all talking in the same terms.

We feel the most effective way to compare cam duration is by using the timing at 0.100-inch net valve lift (lashed). As far as we know, no one else uses this figure. Yet, it is a very realistic figure to use in cam design discussions. Below this point in the valve-lift curve there is virtually no flow in the port. As a result, any timing which occurs before this point in the net valve-lift curve and is included in the duration figure is meaningless. In our tests the flow numbers at 0.050-inch lift, versus wide-open valve lift, are about 2 out of 125, which is less than 2%. At 0.100-inch lift the flow is still only about 7-8% of total, even with a very good low-lift port.

There are many problems inherent in using this reference. It is not popular, and for absolute precision the diameter of the base circle and the rocker-arm ratio would have to be considered. As a consequence, we will give all duration figures with reference to a point defined as 0.017-inch cam lift above the suggested lash point on a roller or solid lifter lobe profile. This point can be considered to roughly coincide with the currently popular reference known as the "effective 0.050-inch timing point."

Using this reference, our long-rod 354-inch drag race engines use about 290° intake duration and 300° exhaust. In the 330-inch engine we drop about 4° off each of those figures, giving suggested durations of 286° intake and 296° exhaust. Dropping ,down another 30 inches to the 302-inch engines calls for further reduction of 4° on each lobe to about 282° intake and 292° exhaust. *In general terms we look upon duration as a function of engine displacement and peak power engine speed.* As the displacement is increased the net timing can be increased. Likewise, as peak power is moved higher in the range the duration can be increased. However, the usable power band will be narrower and we are assuming a severe intake port backflow problem is not created by induction/rod ratio design.

At a big Grand National track like Talladega or Daytona, a four-barrel engine can use timing of 282° intake and 286-288° exhaust. You will note that the timing spread on this cam is less pronounced than on the drag race timing. We almost always use 10° more on the exhaust timing of our drag race cams to, in effect, over-exhaust the engine as a balancing factor for the higher efficiency of the induction systems. On the GN cams the current tendency seems to be toward increasing the "feed" to the engine with less exhaust scavenging and overlap. In these instances fuel mileage has become very important, and over-exhausting the engine at high speeds will tend to raise the fuel specific in this range as a result of blowing fuel out the exhaust. We even feel that with a very good, highly-efficient exhaust port it may be possible to run a single-pattern cam with the same duration on intake and exhaust curves.

The figures given in the previous discussion pertain to an engine equipped with a good set of 292 heads. In any single four-barrel engine which is equipped with the 461 or 492 stock casting, such as required in the current NHRA Super Stock classes, you would probably want to increase each of the given figures by 4° or more on each valve. Most four-barrel engines will not require as much spread between the intake and exhaust timing. In these cases you will want to cut down on the exhaust timing or overlap because there won't be as much "back flow" impeding the intake charge, and exhaust blowdown may result in less efficient induction. Just as in the GN example, it may be possible to run a single pattern cam in a four-barrel drag race motor if the exhaust is working very well (good ports and well-designed tube headers) and/or the engine has a small carburetor. With a very, very small carb you would also reach the point where it may be necessary to spread the lobe centers, reducing the overlap timing. In the instance that small valve heads are required by the racing restrictions (an example is the lower NHRA Super Stock classes), a small engine in the 283- to 292-inch range may call for even more duration timing beyond that previously recommended.

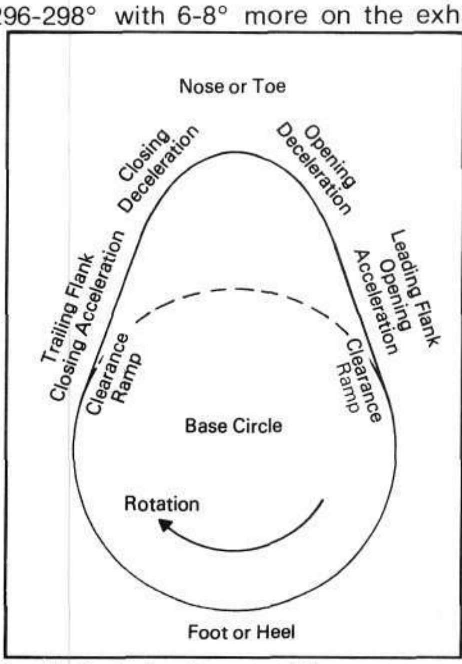
In the road race engines which require good engine range, such as the 305-inch Formula 5000 engines, the timing should reflect a little less duration than recommended for the drag cars. Depending on the course, gearing, and how low the engine rpm drops in the corners, the reduction may amount to 4-8° on each lobe.

Recently we have been running some very big match race engines with displacements in the high 300-inch range. Our data is as yet indefinite, yet these engines have a fairly small rod ratio because of the space limitations of the block; however, we have run cams successfully with as much as 300-302° at 0.050. We haven't been able to find any more power beyond this point, at least with the current revamped 292 heads. The best results have been in the range of 296-298° with 6-8° more on the exhaust lobe. There are also some indications that the big engine wants more lobe spread, to about 110-111°, than the 109° we normally use. We don't completely understand why this is helpful, though it probably is a result of the rod ratios.

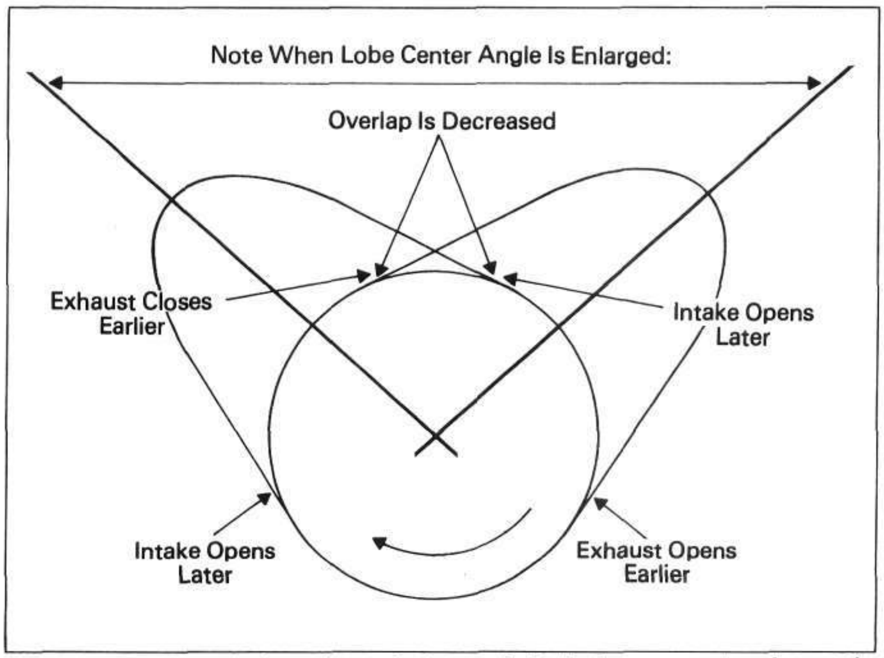
LOBE CENTERS

The displacement angle between the lobe centers is an important design parameter related to the lobe durations and the valve events. By spreading the lobe centers, the exhaust events will occur earlier relative to the crank position and the intake events will occur later. This results in less overlap period (intake opening and exhaust closing move closer together). When the centers are moved closer together all these effects will be reversed. At this point we have to assume that the reader has some understanding of camshaft functioning, and a detailed discussion of the lobe and event relationships is not necessary. Nonetheless, we may be able to shed some light on what we consider important about these relationships when a camshaft is being ground for a high-speed racing engine.

All of our unlimited racing cams are ground on 109° centers. We have experimented extensively with different designs and almost every configuration conceivable. No matter what we do with the other variables, we always find our unlimited smallblock engines will give the best all-around power when the cam lobes are ground on 109° centers. Generally, changing the centers will alter the engine characteristics to some extent. If the centers are spread, the engine range is reduced and shifted toward the high end of the scale. Cams used in stock engines with high-rev potential are often ground with centers as wide as 114°. In the case of the opposite extreme, when the



Camshaft design is one of the more complicated aspects of the high performance or racing engine. For unlimited racing the lobe is contoured to gain the maximum area under the valve displacement curve.



The lobe center angle is an important camshaft design parameter. Increasing or decreasing the angle will affect all four valve events. Spreading the angle will decrease overlap, open and close the exhaust valve earlier with respect to the crank and piston, and open and close the intake later.

centers are closed together, the range will be wider and lowered in the rpm scale. However, this also leads to increased overlap and the resultant increase in exhaust scavenging during the overlap may have a detrimental effect on the fuel specific curve, because raw fuel is drawn across the chamber and out the exhaust.

Moving the center angle may help gain some desired engine characteristic when another approach is not allowed by the rules; however, in an unlimited situation, the 109° figure should give the best results if everything else is in proper shape. We have chased this all around the engine, and with rod ratios in the 1.7-1.9 range and a good induction, there is no other choice. In our opinion this response may be related to the design of the smallblock chamber and the piston dome. Exactly why this is so is still uncertain, but through a process of elimination we have ruled out every other variable we can isolate.

Traditionally, there have been three methods to gain bottom-end power: reducing the duration, reducing the center angle, and advancing the cam. We know that reducing the duration will definitely hurt power. Reducing the center angle may work in some instances, although this is very restrictive, and if fuel consumption is important, this is out of the question. The only remaining possibility is advancing the cam by 2-4°. This will really not have much effect with an unlimited cam, but it may help.

We feel the best alternative in this predicament, and one that is seldom explored, is to play with the intake runner volume. The idea is to use the smallest runner size consistent with the specific engine combination without incurring a top-end power loss. You can look at it another way; to gain the best engine range you need the largest runner volume consistent with low-end power requirements. You cannot have both, and you cannot go after one with total disregard for the other. This often requires testing in the chassis as well as on the dyno to determine "driveability" factors which cannot be verified in static bench (dyno) testing alone.

This brings us to another important concept related to the valve duration. Of the four valve events, we have strong feelings about only one—the intake closing. This event is universally accepted as the most important parameter, compared to the relative insensitivity most engines display for precise placement of the other events. This point must be finely balanced to gain the greatest amount of cylinder filling before the valve is closed. We know the intake valve can be closed well after the piston has passed bottom dead center on the intake stroke and is actually moving up on the compression stroke. It takes some time before the upward piston movement begins to create enough pressure in the cylinder to create compression and initiate backflow into the intake port (if it were still uncovered). By delaying the closing point we can take advantage of this extra area under the curve for more induction to create more power. Closing it late will also take advantage of any velocity built up in the intake runner; however, this velocity will only be a major factor at high engine speeds. If the valve is closed earlier, there will be less blowback loss up the intake and the pressure will begin building more quickly in the cylinder; an important factor for low-speed torque.

The foregoing effects have to be considered when the duration and lobe centers are selected. And, it is possible to see how the rod ratio and induction efficiency can have a bearing on selecting the maximum point location (remember the rod ratio will affect piston acceleration away from BDC and TDC and the amount of time the piston resides in the relative BDC and TDC location).

This location of the intake closing is the effect felt when the cam is advanced or retarded with respect to the crankshaft. When the cam is advanced,

Drag racing is very abusive to the valvetrain. In between rounds we must adjust the valve clearance as a matter of course. There is a more important reason for pulling the covers. Valve spring and lifter failure is so prevalent that we must check after every run to locate these potentially destructive calamities before they incur severe damage.

the lobes are displaced slightly in the direction of rotation so the events will all occur earlier. The intake closing occurs earlier and the bottom-end performance is enhanced. When the cam is retarded, the lobes are moved in the direction opposite of rotation. The intake closes later relative to crank (and piston) movement. This results in better breathing at high engine speeds and the consequent power gains in the upper range is felt. Normally, this effect cannot be found with a change of less than 2°.

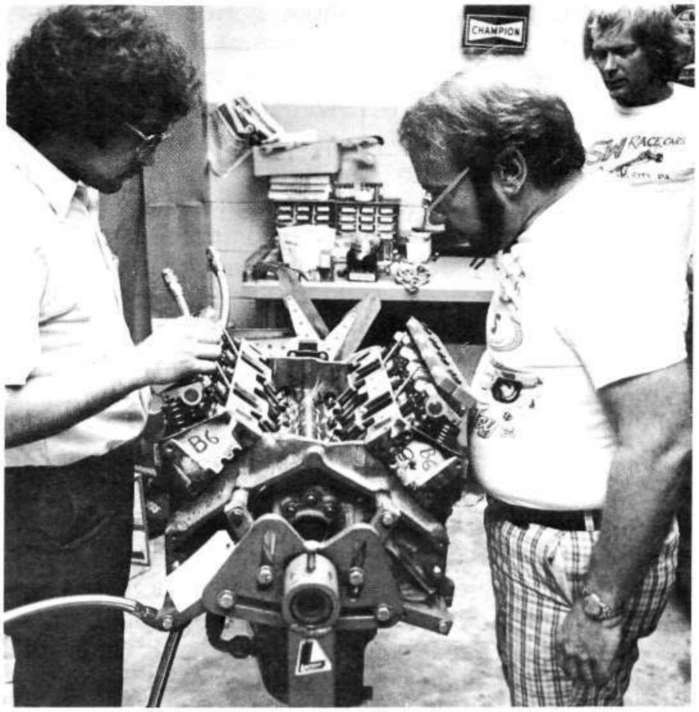
The exhaust valve opening will also be important in rare instances. Some years back when we were involved in the Mark engine program we discovered unexpected gains by opening the exhaust very early. This engine did not have extremely good exhaust ports, and opening the valve early helped initiate exhausting through blowdown from the pressure, rather than relying entirely upon port/header scavenging during maximum lift. The limiting feature here is the lost power which may escape out the port with blowdown. This is not a problem with the smallblock because our experience has shown that the smallblock port is proportionately better at mid-to-high lift of the exhaust valve. At this time (the lower part of the stroke), the piston is still moving downward with considerably more velocity in relation to the engine speed (because of the better rod ratio) as compared to the big block, where the piston is virtually stopped at the bottom of the stroke for some time.

LIFT

Currently we are running cams with approximately 0.700-inch net lift at the valve on a lobe reading 0.730- to 0.740-inch theoretical lift. The valve gear isn't stable enough to withstand more lift than this, though there may be some power on up toward 0.800-inch lift. At least we haven't been able to utilize this much lift reliably with the current stud-mounted rockers. To gain this sort of valve lift, the cam lobes must produce a lifter rise of 0.472-inch with 1.55:1 rocker ratio and 0.440-inch with 1.67:1 ratio.

There's really no trouble with running as much lift as mechanically possible in an unlimited racing engine. Finding a valve spring to handle all the lift may be difficult, but that's another subject. We find the main physical limitations are how small the base circle can be reduced and possible lobe interference with the connecting rods.

Within the journal limitations of the cam core, it is necessary to reduce the diameter of the base circle to gain the amount of lobe height desired (the lobe radius height always equals the radius of the cam journal, while "lobe height" or lifter rise equals the difference between the radius height and the base circle radius). When the base circle is reduced, the overall torsional rigidity of the finished cam will be diminished. We are reluctant to reduce the base circle diameter beyond that required to gain the 0.472-inch lobe height. It would be possible to go to the big



Modern racing competition requires a continuous research and development program. Whatever we know today will be inadequate in six months. We must spend at least 50% of our time working on new ideas and techniques that may (hopefully) or may not keep us competitive in six months or even a year beyond the present.

block cam journal size and use a larger core, saving some base circle size if desired, but the problem with rod clearance is not so easily solved.

In certain cylinders the connecting rod may interfere with the cam when the crank journal is closest to the cam and the corresponding lobe is in a downward orientation. With the current lobe size, the cam will clear Superod aluminum rods on a 3.250-inch stroke crank. In the 3.48-inch stroke engines we must relieve the somewhat beefier BRC rods because of this interference problem. The situation is nearly impossible with the 3.750-inch match-race engines. This is only troublesome on certain lobes. In a long stroke, high cam-lift engine, the minimum clearance between all of the connecting rods and cam lobes should be checked to insure approximately 0.060-inch at the closest intervals. This must always be considered in long-stroke engines using steel rods with thru-bolts such as the Chevrolet 710 connecting rod or the bulky aluminum rods.

ROCKER RATIO

At this point we must get into a discussion of rocker arm ratio and correlative effects. The stock smallblock rocker arm ratio is 1.5:1 (theoretical). Increasing the rocker ratio beyond this will, if all else in the valvetrain remains the same, increase the effective net lift at the valve, increase the effective duration at the valve, and increase the valve acceleration and maximum velocity. Some of these resultants are easy to visualize, others are less obvious.

The increased net lift at the valve is self-explanatory. The lobe height or, more correctly, the lifter rise must be multiplied by the theoretical rocker ratio to obtain the theoretical lift at the valve. Whenever the ratio is changed, a new computation must be made to determine the revised theoretical lift. Here we are using the term "theoretical lift" because in actual practice, with the extreme high tension springs required to control the current components at high engine speeds, there is a great deal of flexing induced, backward through the valvetrain. This is one of the negative side effects of high-ratio rocker arms. All physical properties transferred across the fulcrum point of the rocker will be multiplied by the rocker ratio. The strain imposed on the follower and pushrod by the valve spring will be increased when the rocker ratio is increased. This compression or deflection of components accounts for a noticeable loss in the net valve lift versus theoretical valve lift (other factors are involved in the relatively high-percentage loss in our particular racing valvetrains, but they will be discussed later). This is why our "net static valve lift" is significantly reduced from the theoretical figures.

When the ratio is increased, the effective valve timing will also be changed. Our tests show that when the ratio is increased from 1.5 to 1.70:1 the net valve timing at 0.050-inch valve opening will be increased by about 5°. The initial point will not be altered, yet by increasing the velocity and acceleration of the valve, all the intermediate effective timing measurements will be spread further apart. It is theoretically true that the duration at the lobe has not been increased but, in fact, the effective duration at the valve will be increased. This assumes that the shape of the lobe has not been changed to accommodate the higher ratio.

When changing the rocker ratio on a cam lobe that has not been altered for more ratio, the first problems usually arise from the positive acceleration increase and the contact velocity increase. When the ratio goes up, the effective length of the clearance ramp on the cam lobe is decreased (if the clearance remains the same). Should the related components not be able to absorb the subsequent rise in contact shock, some failure may result. At times the running clearance can be decreased to compensate for the change in contact velocity. But changing the lash figure can be a little dangerous, especially if you don't really know what you're doing. We don't recommend changing the rocker ratio on a lobe without first consulting the cam maker.

Since the ostensible purpose of altering the rocker ratio is to increase the area under the curve, the question arises as to whether it might be better to accomplish this through a redesign of the lobe. *In our experience, when attempting to gain the same valve displacement curve through altering either ratio or the lobe, we have always found it better to use the rocker ratio, resulting in a lower lobe height for the same theoretical valve lift.*

We have done comparison displacement traces

The traditional flat or "solid" follower is an excellent choice when durability is a prime requisite. In some instances roller lifters may not be legal or are otherwise undesirable. We prefer to use the Eaton-manufactured inertia valve lifter offered as part C2L-A by General Kinetics, though similar lifters are offered by other distributors.

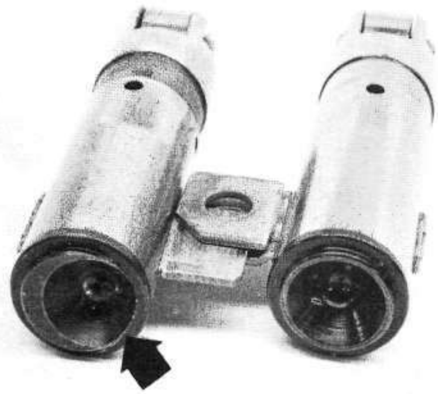
with different ratios. At times we have found that it may not be possible to successfully grind as much positive acceleration into a low-ratio lobe as can be attained through an increased rocker ratio. The problem is uniquely inherent in the design of a roller cam. It is possible to impart a terrific positive acceleration to a flat lifter by hitting it very hard with the leading flank of the lobe. This technique will not work with a roller. The mechanical effect of the roller tip won't allow such a lifter to be "kicked" open by the lobe. Things have progressed to the point that to gain the required curve with a roller cam and 1.5:1 ratio rockers, the side of the lobe has to be hollow ground. In the current lift ranges and acceleration values this can normally be avoided by going to a higher rocker ratio.

LIFTERS

In the flat-lifter engines with cast camshaft cores we use the Eaton-type lifter available from General Kinetics as part C2L-A. This particular lifter has the inertia-valve insert to meter oil flow up the pushrod to the upper valvetrain. It has slightly less oil-relief groove ground in the body and is less susceptible to breaking here than some of the other brands. The model sold by GK can be identified by the copper-plated snap ring used to retain the valve in the lifter body. The lifter does not control the oil as well as the Chevy edge-orificing type. Therefore, we restrict the oil flow by jetting the lifter galleries. This will prevent the lifters from supplying too much oil up to the rockers. The inertia-valve lifter is an excellent unit as long as it is kept clean. In the instance that full oil pressure is allowed to the lifters and dirt happens to lodge in one of the metering valves, the valve stays open and the lifter floods the pushrod with oil. As long as the oil is changed often and is well-filtered, this will not be a problem.

We drill a small hole in each of the front oil gallery plugs to eliminate air-locking the gallery. This fix originated during the Mark engine program when we

Satisfactory roller lifters are available from several makers. We have used many brands, all with equal success. These Crower rollers are typical. They have been drilled in order to allow quick cam changes without removing the manifold and have also been chrome plated, but are otherwise stock.



At times we have useo the unique high-button Isky lifters. They allow the engine to be fitted with shorter and smaller diameter pushrods. Note also that the offset button provides pushrod offset, giving added clearance between them for the intake ports.

had trouble with hydraulic lifters draining down during shut-down periods. At start-up, air would become trapped in the gallery and the lifters would not get oil properly. If a problem arises with lifter wear, we may increase the rear orifice size to let more oil in the galleries, although we also bleed more off the front. When balanced properly the pressure level will not be unnecessarily high, yet the flow rate will be increased as required to lubricate the lifters. The lifter galleries don't need a great deal of pressure in a racing engine. We don't want the oil upstairs as the roller-bearing rockers require very little lubrication. Any excess pressure fed into the galleries will only cause excess oil splash around the lifters and down into the cam and crankshaft housings.

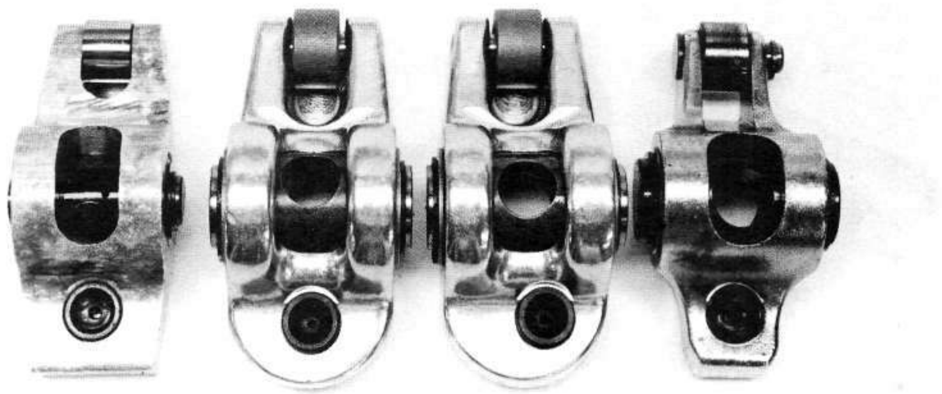
When we elect to use roller followers, the choice is usually a standard Crower steel roller that has been chrome-plated. Lifters similar to the Crower model can also be purchased from Engle, Sig Erson or General Kinetics. We have also used the old-style Crower aluminum body lifters and the discontinued Isky high-button steel model. All of these various brands give adequate service. However, we use that phrase advisedly. Without some proper care, roller lifters in a Pro Stock engine can be troublesome. The springs and lobes we use create a tremendous side load in the lifter bores. We have seen a plain roller lifter broach the side of an iron lifter bore to death in just 15 minutes of dyno running!

We experimented with aluminum-bodied lifters to prevent this wear. They worked well, but once the hard-coating wore down, the lifter would become chewed-up very quickly. If the spring tension drops off, these lifters will also break up fairly easily. We have also experimented with other anti-friction coatings and they worked well. Right now, however, the simplest and cheapest method is to just have the lifter bodies chrome-plated. This prevents heat build-up and scuffing in the bores.

We have found that the Crower roller tips are somewhat easier on the camshaft lobes. The radius-faced roller won't edge-dig the lobes and they will survive well on a somewhat softer core. The one thing to watch is the link-and-pivot arrangement that keeps the lifters from rotating. The Crower's are no worse than any other lifter using this system, but we have found that the most prevalent failure in our engines, next to the valve spring problem, is the breaking of these pivots and/or links. When this happens, the lifter rotates in the bore and destroys the cam (along with several other parts, if it isn't discovered quickly).

Some of the engines pictured here are equipped with the high-button Iskenderian lifters. These are no longer available, but they have one unique advantage. If large port volume is a requirement, we may still use these lifters. With the steel insert button placed very high in the roller body, the pushrod can be shorter and smaller in diameter. Using offset buttons, it is possible to gain more port clearance with this setup than almost any other combination we have. We have also had somewhat less trouble with the fork-and-blade interlocks used in this design, but they must be combined with sufficient spring tension or they can float away from the lobes, and the pins will break out of the roller tips very quickly.

We check all roller lifters to determine if the roller pin is on the centerline of the lifter body. This can affect camshaft timing. The easiest way to accomplish this check is with a flat plate and a dial indicator. We lay the lifter on the plate with one side of the roller upward. It is passed under the indicator to determine a reference reading. The lifter is then flipped over and rechecked. If there is a variation between the two readings, the roller axis is not centered with the lifter body. After some little experience it is possible to determine a known reference reading and the lifters can be checked by just passing them under the indicator one time. Most commercially-made lifters will not be exactly centered. As long as they



We have used BRC, Crane, General Kinetics and Norris rocker arms with success. All of them are available in ratios between 1.5:1 and 1.7:1 and with longer arms to give additional spring clearance. They are also available with offset pushrod sockets for additional intake port clearance.

are within about ±0.004-inch, the timing of the cam will not vary by more than ±¼°.

LIFTER-BORE PROBLEMS

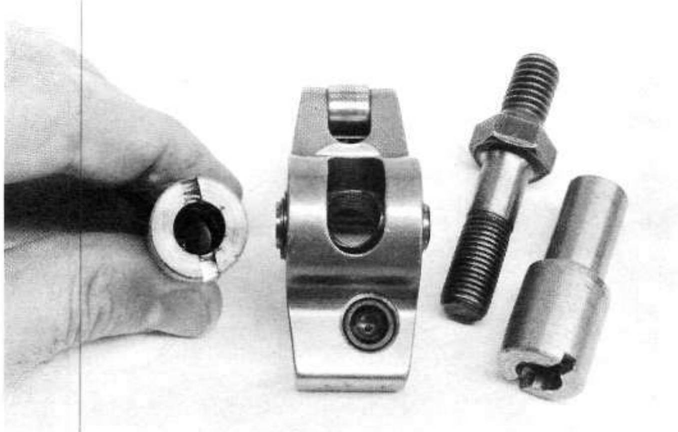
For production engines the lifter-bore centerline tolerances are not extremely critical. They certainly should be within the factory specifications, but, for racing, these tolerances must be much closer. We have seen some blocks come from the factory with lifter-bore "lean" problems so acute that flat tappets may rotate too fast, or rotate in the wrong direction, or they won't rotate at all. In some cases, they ride up on the taper edge of the cam lobe or even ride on the wrong edge of the taper (this taper is built into the lobe to make the lifters rotate). This type of variation is unacceptable in a racing engine. Whenever it is equipped with a flat tappet follower.

There are four definable problem areas. They are caused when the factory production gang-drilling procedure is not accurate. The frequency of one or all of these troubles is common. More than likely they will exist to some degree in every production block. Factory prints call for the lifter-bore axes to intersect with the axis of the camshaft. All of the lifter-bore axes should lie in the same plane and they should be perpendicular to the cam axis. Also, the bores should be correctly spaced, fore-and-aft, as required, over the cam lobes. If any of these four specifications are severely violated, some sort of lifter or cam problem may occur. Factory tolerance allows for 0.0006-inch out-of-perpendicular and 0.005-inch out-of-plane "lean."

First we will consider flat tappets. It is imperative that all the lifter-bore axes are perpendicular to the cam axis and they must be properly spaced fore-and-aft. These parameters determine whether the lifter rides correctly on the lobe and rotates as required. The axes may lean out-of-plane, the upper end of the axes may tilt toward the centerline of the engine or away from the center, as long as the axes all still intersect the centerline of the cam. There will be some timing variation as a result of this "lean," but there will not be any mechanical action to severely restrict the engine operation. We have noted that it is common to find the bore axes off-the-perpendicular. If a problem does occur, it will not often affect the entire bank of bores. More than likely, only one of the gang drills will hit a particularly hard spot in the cast iron, walk sideways and drill a single bore off the perpendicular.

The original block casting was not designed to work with a roller-tipped lifter. Some racers may not take this into consideration when they buy a set of rollers and slap them into the block. However, the mechanical interaction of the roller with the cam lobe makes them much more sensitive to some of the problems previously described. We don't need to make provisions for the roller lifter to rotate in the bore, but the roller contact with the lobe is important for proper timing. The rollers are most sensitive to a bore axis being entirely out-of-plane with the cam axis, although any of the eccentricities mentioned above can also cause trouble. If the axis is out-of-plane but still intersects with the axis of the cam, timing will be affected, but the roller will still operate adequately. However, if the entire axis of the bore is out-of-plane with the cam, the roller will not properly contact the lobe. In this instance, the side load imparted to the lifter may be much higher.

To check the bores we use the assumption that the block was designed to be used with flat tappets. Finding variations from one lifter bore to the other is accomplished by checking all of the lobe centerlines using a flat tappet on the roller-tappet lobes. Then, we put a well-centered roller lifter in each of the bores and read the centerlines again. If there is a wide variation, something is wrong with that bore. In rare cases we have seen this variation as much as



All of our rocker arms are equipped with trunnions to fit over the 7/16-inch studs from a Mark engine. These studs fit in place of the stock smallblock studs. Note the Jomar adjusting nut has been drilled and tapped to accept a locking screw.

3°. It is not possible to tell exactly what is wrong with the bore from this method, but it is a means for locating the existence of a problem.

If you are in a hurry, the easiest way to circumvent (ignore) this problem is to install a roller cam on-center by using a flat lifter in any cylinder, picked at random. The factory block finishing methods are designed to work with a relatively simple flat (or hydraulic) lifter. As a result the flat lifter will not be as sensitive to lifter bore inconsistencies as will a roller lifter. In particular, they will not pick up an axis out of plane problem as will the roller. Therefore, you will be more likely to get the cam on center (or nearly so) with a flat lifter for reference

For light duty application there is little recourse to solve this problem. We can only suggest that you check the lifter bores and hope for the best, or perhaps check several blocks to find the best one. A better solution is to drill the bores oversize and sleeve them to the correct axis angle and location. By using aluminum-bronze sleeves the side-load induced lifter wear could also be reduced. Unfortunately, few machine shops are equipped to perform this job correctly but it is the ultimate solution if exact cam timing is required.

ROCKER ARMS

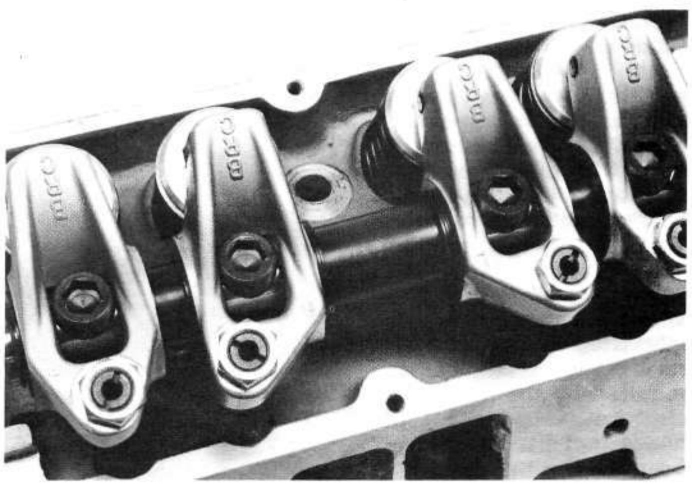
All of our racing engines are equipped with roller-tipped, roller-bearing rocker arms. The latest engines are still using stud-mounted rockers, but we are extensively testing shaft-mounted rockers. We will discuss rocker-shaft assemblies in a later section, but we feel there should be a definite advantage with shaft-mounted rockers over stud mounts.

In our engines we have used rocker arms manufactured by BRC, General Kinetics, Crane and Norris. They vary from stock ratio to larger ratios with 0.050-, 0.100- and 0.140 longer arms in right or left pushrod offsets. The offset may be as much as 0.170-inch. All of the rocker arms are inspected, the ratio checked, marked, and divided into matched sets as soon as we receive them. Our program requires a large number of rocker arms to keep all the racing, testing and back-up engines ready to go. Considering the loads, we don't have much trouble with any of the rockers.

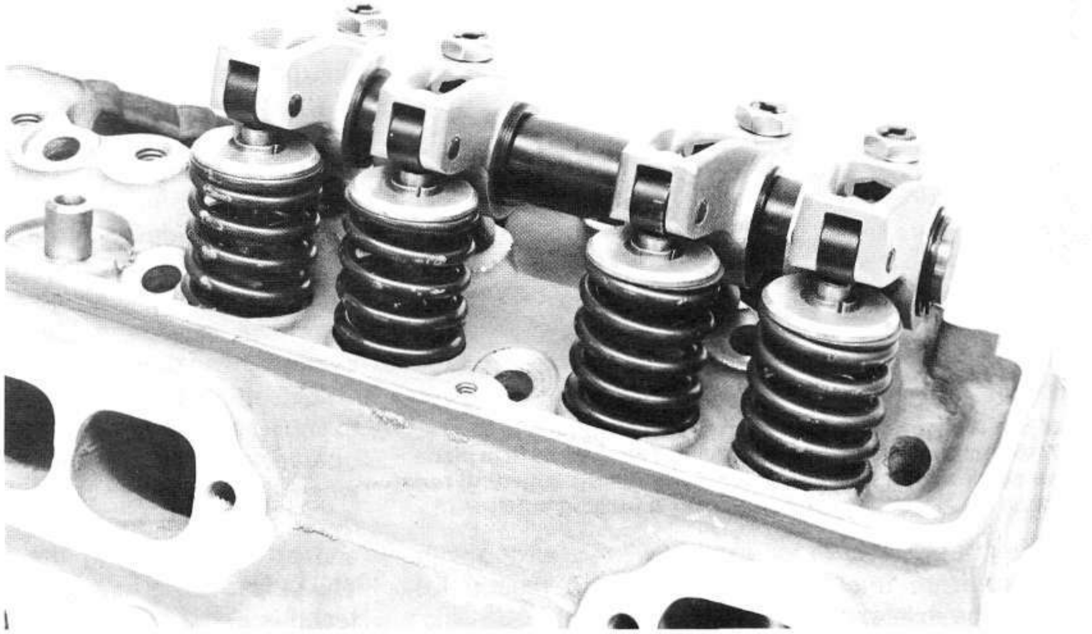
If a longer arm is used, the stud is usually moved away from the valve centerline to gain spring clearance and maintain the correct geometry between the rocker tip and the valve stem. Also, if the valves have been moved sideways in the chamber and if the pushrods are offset to gain more port clearance, the studs may also be moved sideways to increase clearance and salvage a little bit of the geometry (see cylinder heads).

Offsetting the push rod causes the rocker arm tip to move sideways across the end of the valve during operation. This doesn't hurt the valve stems but, when the pushrod socket is moved sideways a great deal, the rocker may develop wear failures in the tip and/or start breaking the trunnions out of the side. The side loading may also aggravate pushrod flex. We feel it is better to move the stud as far as possible and reduce offsetting the pushrod socket as much as possible. Regardless, any offset of the rocker or socket will cause the tip to move sideways on the valve end. Up to a point, we haven't had excessive problems in the drag engines, but this technique is hardly suitable for a Grand National engine. Using shaft-mounted rockers will eliminate this problem.

We have run some experimental 1.70:1 ratio rocker arms but most high-lift roller cams don't need this much rocker. In the drag engines we run rockers 1.63:1 "loaded." There's a normal loss of 0.03 to 0.04 ratio in a competitive valvetrain. If you exceed this figure you're just kidding yourself, because everything is just bending and flexing instead of moving the valve. We check the load-deflection loss by installing the valvetrain with light valve springs, cycling the cam and measuring the net lift at the valve. Then, we put the racing high-pressure springs on and recheck the net lift. With only one valve assembly



Using shaft-mounted rockers instead of the stud-mounted type offers several distinct advantages. Offsetting the pushrod sockets on a stud-mounted rocker creates a side load on the valve tip during rocker action. A shaft mount eliminates the problem.

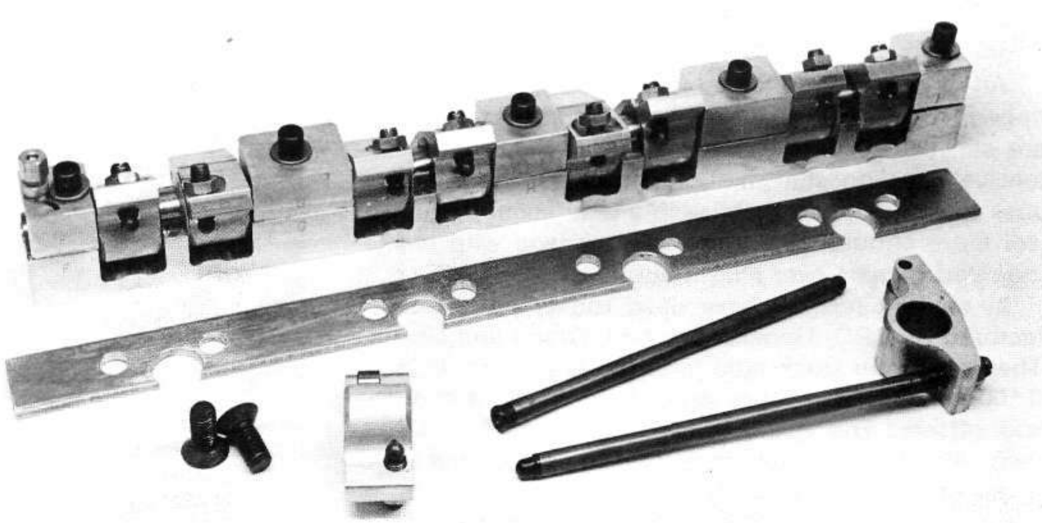


We have not tested all of the different rocker shafts extensively, but the BRC shaft demonstrates some of the important features. The shaft is large in diameter for stability. The rockers are large and stiff, though they may be overly heavy. The rockers ride on roller bearings, requiring no pressure oiling. The pushrod can be offset a great deal and the clearance is easily adjusted. We would like to see more mass in the arms where they pass around the shaft and the roller tip pin could be larger.

working, and providing that lobe-deflection and lifter-bore problems have been eliminated, any resultant loss in valve lift can be attributed to load deflection. If the deflection loss is less than 0.03, it is quite good, less than 0.04 is acceptable, and over 0.05 is very edgy. Anything more than that is totally unacceptable and countermeasures must be taken. No matter what it takes, bigger studs, larger stud-tie bars, locking and adjusting nuts to the studs, thicker pushrods or anything else, the valvetrain must be able to match the required valve springs for the desired engine speed.

In the drag engines we have used the General Kinetics extruded-aluminum rockers, BRC forged-aluminum rockers and the Norris investment-cast steel arms. The big fat GK rockers allow for a lot of pushrod offset. We haven't been able to isolate any problem with the aluminum rockers flexing or see any difference in deflection between the aluminum and the steel. However, with the current 600 lb-plus Pro Stock spring loads and offset "everything," we occasionally break the bottoms out of the rockers or break up the trunnions. The steel rockers hold the trunnions in very well (though the trunnions will break). For economy the extruded aluminum designs have small bearings that cause trouble. Larger bearings in these rockers would be a welcome improvement. We would also like to try a larger roller on the tip of the arms. This would add unwanted mass to the end, but they would cover the tip of the valve better, reducing the tendency for the rocker to pound the end of the valve to death.

In the Grand National engines we have always used the Crane extruded aluminum rockers and they have never given us any trouble. Of course, they don't have to cope with more than about 350-lb spring



Another shaft design with excellent potential is the Exact Performance assembly. Rocker and spring maintenance is somewhat more difficult with this shaft and it requires external pressure oiling to the plain bearing-mounted rockers. To date our test engines have shown this shaft works as well as any. The shim plate shown here was fabricated in our shop to raise the assembly as required by the machined stud bosses on a set of heads that had previously been fitted with studs.

pressure and 0.600-inch lift. This type of endurance engine does not take kindly to offset geometry and we don't try to make these drag racing techniques work on the circle tracks. Remember, the offset technique gains clearance so the port can be welded and widened. Endurance heads definitely do not remain very "durable" after the strain of excessive heating in localized areas.

ROCKER SHAFTS

There are several advantages to a well-designed rocker-shaft assembly. They will accept the side loading of offset pushrods and eliminate the problem of the tip wiping the end of the valve. A good example of this is the Chrysler "Trans-Am" 340 head, which allowed effective increase in the port size of the smallblock Chrysler by angling the pushrods away from the ports and offsetting the pushrod adjusters in the shaft-mounted rockers. There is the additional benefit of a rocker assembly designed, hopefully, to withstand the special demands of a high-rpm engine with very high spring loadings. In the past, attempts to produce shaft assemblies for the little engine have not lived up to expectations.

One rig we will be testing is the Exact Performance assembly. This arrangement bolts to the head in the original stud receptacles. The aluminum rockers use ball-type pushrod adjustment to control valvetrain clearance. The shaft requires separate oil pressure feed to the assembly for lubrication of the plain bearing rockers. For our particular needs they may not be the best answer. One thing that bothers us is the trouble this setup causes when maintenance is required. To change springs or perform any other service work on the top of the head, we have to remove the entire assembly. As things now stand, we often have to replace the springs in a hurry. This extra work could be a real disadvantage at the race track. We are also not very happy about the requirement to feed extra oil to the shaft to lube the rocker bushings. This looks like more paraphernalia to get in the way or cause trouble.

The other assembly we are testing is the BRC unit. This assembly bolts to the head with large-diameter pedestals, piloted into the stock stud holes. The shaft is very large and the rockers are roller bearing mounted to the shaft. All the individual components appear to be very strong. Unless some unseen weakness springs up, these shafts and rockers should handle any demands the current spring designs can impose. We especially like the roller bearings which will not require pressure lubrication. Like conventional roller bearing rockers, these aluminum extrusions should operate just fine with no oil supply except the oil cloud floating around inside the engine. The pushrod socket can be offset a large amount, and the roller tip should provide excellent contact with the valve end. Servicing the rockers or springs should be easier and somewhat quicker than with other shaft assemblies.

Before changing the subject we should also mention the various geometrical advantages of rocker shafts. Though some might consider this to be a very small consideration, we have noted that by installing the adjusting screws at an angle, using different length pushrods, and altering the height of the shaft above the head, it is possible to make changes in the rocker ratio and the positive-negative acceleration values in the valve displacement curve. This is difficult to explain. But if you study a good drawing of a shaft assembly and valvetrain, it becomes clear that by moving the shaft (the centerline of the rocker arm fulcrum) up or down with relation to the valve ends, it is possible to reduce lift and increase the valve "action" at certain points in the curve. The opposite effect can also be achieved, more lift with less action in the midsection of the curve. It is easy to verify the ratio change that occurs when the adjuster screw is angled outward and/or different length pushrods are used in the valvetrain. The ratio change is going to be small, yet we have seen minor changes produce startling results in other cases; the same might be true of small changes in the rocker

These are our current spring choices. The GK 2017 (left) works well to 0.750-inch net lift with a light exhaust valve. The GK Pro Stock spring (center) or the Norris "Battleship" winding made from Carpenter steel will work on the intake. For Nascar the GK 2001 or 2002 will do.

For reasonable life the springs must be inspected and properly prepped. We take them apart, check for minute nicks or scratches, shorten the dampers and bead blast the inner and outer wire windings.

During operation the springs rattle around pretty good at 10,000 rpm. It is essential to install a hardened steel shim under every spring, otherwise the bouncing damper may saw a hole in the spring seat.

Despite the best design, selection and preparation the current springs will only last 60 to 70 runs in a Pro Stock engine. This is the eventual result. If not spotted in time the spring will load sideways, break the valve and/or the stem guide.

ratio (dependent upon the valve displacement curve shape).

SPRINGS

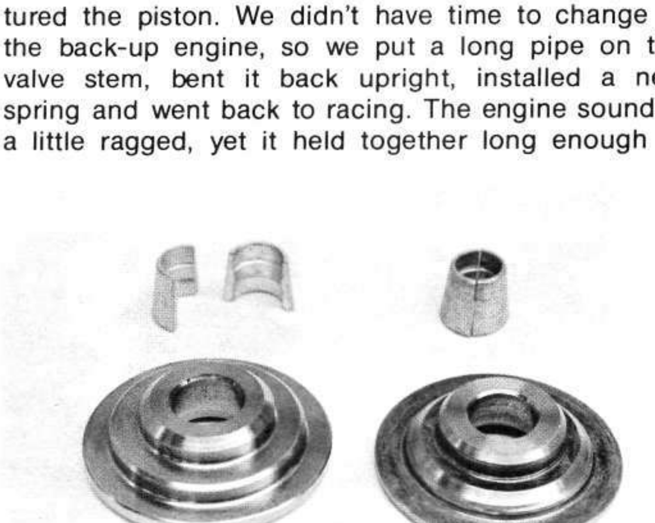
Valve springs are the biggest single problem in Pro Stock drag racing! None of the major teams, Chevrolet or otherwise, can honestly say they have a true handle on the situation. The future doesn't look any brighter. Our problem centers around a lack of space. The Chevrolet heads are very compact with everything tightly spaced. This makes for a light and efficient production head casting, but it can be a drawback from a racing standpoint. We have pushed everything around as much as we can, yet we still wind up limited to a net valve lift of 0.750-inch (at most). Beyond this, everything just bends! We have discussed this with the country's leading designers and it appears that with the current materials and available space, it's not possible to wind anything better.

In our engines we use a General Kinetics or a Norris Carpenter steel spring on the intake valves and a General Kinetics 2017 spring around the exhaust valves. These springs are similar to the Chrysler Hemi chrome-silicon spring. We know that other cam makers offer variations of this design which may work just as well, but some of them will fold up quickly in a competitive Pro Stock engine. For net valve lift in the neighborhood of 0.700-inch, the Norris damper is loaded too tightly in the outer coil and the material is not wide enough. When you really lean on them the damper begins to unwind and "tail out" of the outer winding. As it gets further out, it acts as a shim and forces the spring to load sideways on the valve. Normally, this breaks the valve stem and splits the guide boss.

The Carpenter steel material does have good surface stress acceptability so the outer spring seldom breaks. It is possible to delay the inevitable somewhat by removing the dampers, shortening them, and cutting the tails square. We inspect them carefully for nicks or imperfections, bead blast them, put them back together and pray. With luck they will last 60-70 runs after this "tune up," but we have to watch them like a hawk or one of the dampers will unwind and drop a valve with no warning at all.

The material will lose tension fairly quickly. We set the new springs at 180 lb on the seat and that usually gives us about 600 lb wide open. They go away in a few runs and fall to as little as 155 closed and 540 open. At this point we shim them back up to 180 lb. Once they have "sagged" to this point, they seem to hold tension pretty well. When they fall off again it's time to get rid of them. They will work with as little as 130 to 140 lb on the seat. It's easy to tell when the springs are dead. With everything fresh and up-to-date, the engines pull 9500 rpm with no trouble. They lay down at 8800-9000 when the springs retire.

It's dangerous to run valve springs too long when the cam action is very severe. Many times we have pulled the rocker covers off after a pass and the pieces of a spring have fallen out onto the ground. Once, the valve stem was bent but the valve had not punctured the piston. We didn't have time to change to the back-up engine, so we put a long pipe on the valve stem, bent it back upright, installed a new spring and went back to racing. The engine sounded a little ragged, yet it held together long enough to



We hook the springs to the valves with General Kinetics titanium retainers and Isky machined steel locks. The titanium retainer is only necessary for extreme duty, otherwise aluminum will suffice. Isky locks would be a worthwhile investment in any engine.

finish out the day. However, the damage is normally more "final."

To install these intake springs on the smallblock heads, the valve pocket must be cut to fit the 1-9/16-inch outer diameter "tightly" and a hardened seat must—absolutely must—be installed under the spring to keep the damper from sawing a hole through the head as it bounces around inside the outer winding. This is a short spring and with stock length valves it stacks at about 1.800 inches, leaving plenty of room even with the highest net lift we have ever used. It works fine with 1.67:1 rockers and a lobe lift up to 0.440-inch.

We hang the valves with General Kinetics titanium retainers. They fit tightly to the inner and outer springs. To fasten the retainers in place we always use Isky, machined, hardened steel, valve locks. They cost more than the stamped steel locks but they are the only ones that will hold everything together in our Pro Stock engines. Under less severe drag racing conditions a stamped lock and aluminum retainers would be acceptable.

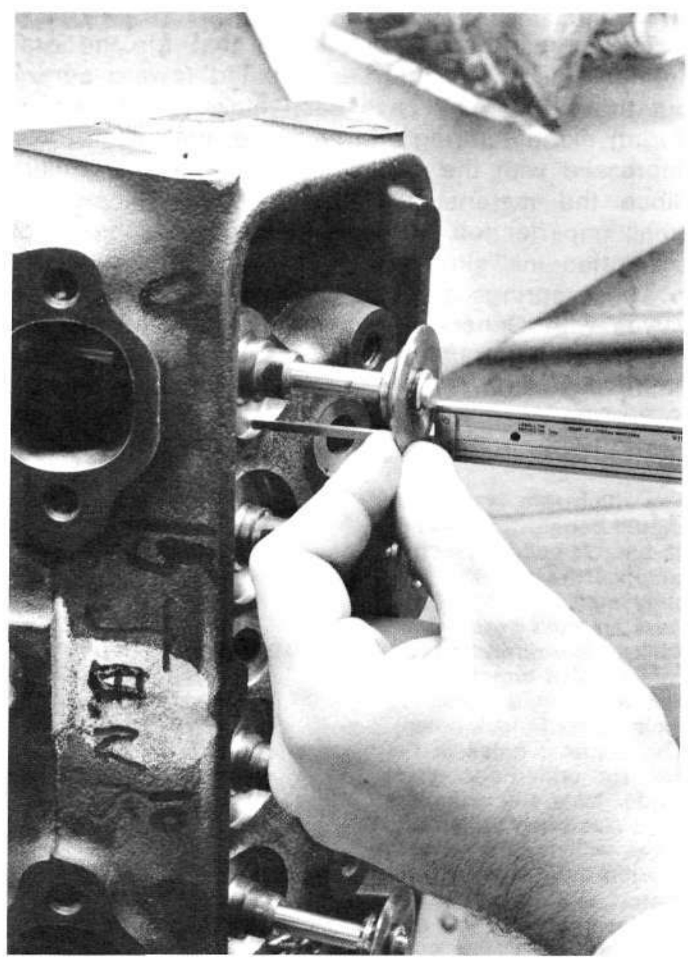
The GK 2017 causes very little trouble with any reasonable roller profile giving up to 0.625-inch net lift or 0.650-inch theoretical. With engine speeds in the range of 8000 rpm, they would be reliable on either valve, although they definitely won't hold the big intake valves down at higher engine turns. They do an "adequate" job on the lighter exhaust valves in our Pro motors at 9500 rpm with 0.730-inch theoretical/0.700-inch net action. They live OK and they don't break.

To make these windings work in the drag engines, we take them apart and inspect them closely. This may sound like a lot of trouble, except we find it easier to look for flaws when the spring is in one piece, before it breaks in half and drops the valve through the piston dome. We have never quite gotten used to scraping a piston off the floor of the pan.

When the springs have passed inspection they are installed to the exhaust valves to provide 160 lb pressure on the seat. They will lose a good bit of tension after a few runs, but we keep after them and shim them as necessary. They will have to be scrapped at about the same interval as the intake springs. Fortunately, they do not unwrap like the intakes so we seldom have any total failures with the exhaust springs. This is largely due to the lighter weight of the 1.600-inch valves. We also use a General Kinetics titanium retainer and the Isky machined locks on these valves.

For Grand National competition we use a General Kinetics 2001 spring straight across. It is good for about 0.600-inch net lift with reasonable endurance.

The intake springs are installed with about 180 lb on the seat to give 600 lb over the nose. They sag after initial installation and require reshimming. The exhaust springs are set at 140 lb.



Once the seat and valve-open pressure heights are determined they must be compared with the seat-to-retainer height of the selected valves to determine the amount of seat shim required.

These windings don't have to withstand the severe lift and acceleration of a drag cam, yet most of the competitive teams are still having trouble with valve springs. The 0.600 lift, or more, in a 500-mile engine wears the springs out very quickly. When a spring lets go on a GN engine, the day is over—there's no second chance. It is, therefore, important to inspect the springs. In this type of application you absolutely cannot overstress the material, whereas in a drag racing engine the springs are overstressed immediately.

And one thing most GN engine builders really dread is the pit road drag racing which occurs in any close race. The springs and cams are not designed for engine speeds beyond 7500. An over-anxious driver can push the engine too far and the valvetrain suffers very quickly. When a spring is forced into a valve float condition it will burn itself up in a matter of seconds. After that, failure is an inevitability.

The General Kinetics number 2002 spring is good for a little more lift. However, we still haven't found a spring that will live at anything near 0.625-inch lift for 500 miles at 7200 rpm. The track engines could definitely use more lift than that, but if you become hungry for power and slip in a little more cam or rocker ratio, you're going to get a spring—plain and simple.

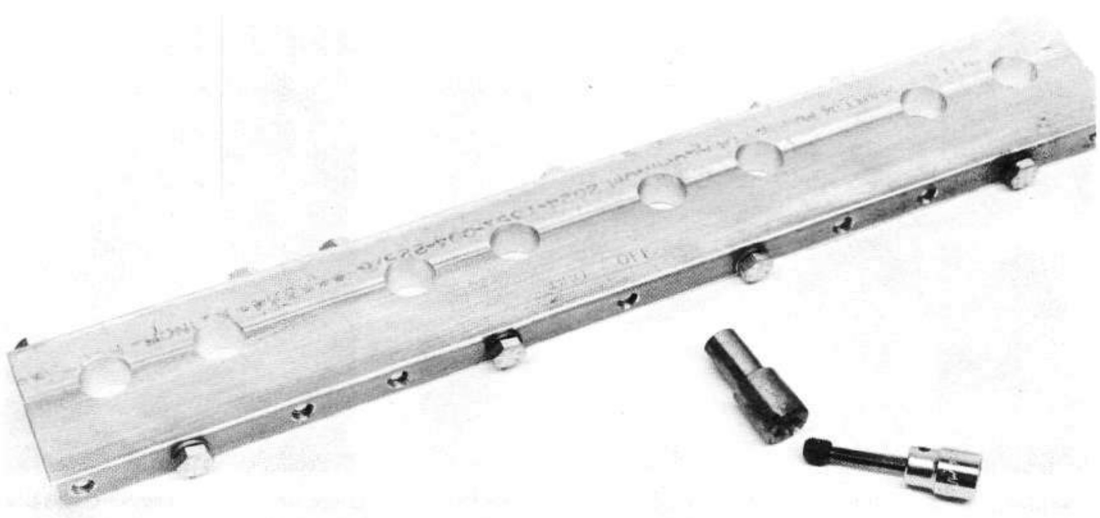
The preparation is similar to that for the drag springs. All of the effort is directed toward sorting out those that may have been nicked or scratched during manufacturing and shipping. We are not very impressed with the quality control of the suppliers. Since the material is largely chrome-silicon, any small imperfection will lead to an instant "break on the dotted line" situation.

These springs are tied to Manley 5/16-inch stem valves with General Kinetics titanium retainers and Isky locks. When held within reasonable engine speed limits, this setup works just fine. We shim them to 140 lb on the seat, recheck them after the engine has been broken in, and reshim as required. After a slight initial fall off, they hold tension, as long as the cam rate and lift are reasonable.

An all-out valvetrain will require 3/8-inch pushrods to offset the strain of high-pressure valve springs and large-ratio rocker arms. When stock stud location is maintained a kicked-up Manley guideplate can be used. When studs are offset, the guideplate should be cut apart, relocated under the studs, aligned and welded back together.

PUSHRODS & GUIDEPLATES

In the drag race engines we use Manley 3/8-inch pushrods. They have proven adequate as long as they are dead straight before installation. We have never failed one unless some other factor contributed to the problem. Depending upon the rocker arm ratio and location, special pushrods may have to be made longer than stock. We use them in lengths from stock to 0.100-inch over.



Any high-rpm valvetrain should benefit from a stud tie-bar. We build our own to increase the size and stability imparted to the studs. Ours are built from 3/4-inch thick aluminum and the two bars are tied together with hex-head bolts (making them easier to lock down). The adjuster holes in the bar are drilled for offset studs. Note the adjusters have also been drilled and fitted with lock screws to lock the entire assembly together. Similar newly-released designs are now available from the commercial suppliers.

Grand National engines can be fitted with a double-row chain available from Chevrolet truck parts or from aftermarket suppliers. The iron cam sprocket can cut into the face of the block under pressure from racing springs. We use a simple brass washer between the faces to act as a bearing.

In the drag engines we prefer to use a stock pre-'67 Chevrolet timing chain and a teflon thrust button. This link-belt chain is wider than the late model, and the aluminum sprocket with nylon teeth does an excellent job of damping crankshaft vibration that might otherwise transfer to the cam.

When we use the high-button Isky lifters in a drag engine, a shorter-than-stock pushrod is required. We prefer to use Smith Brothers 4130 steel, 5/16-inch diameter, 0.65-inch wall pushrods which have polished stems. The short reach of this combination makes the smaller diameter acceptable. We don't recommend 5/16-inch pushrods in any high-rpm drag setup other than this.

For an engine with stock stud location, we recommend the Manley guideplate that has a kicked-up section to accept the ⅜-inch pushrods. For the drag engines with offset pushrods there isn't anything in the speed equipment market that will work. We have to contend with offset studs, moved away from the springs or sideways for port clearance. There just isn't any way a stock piece will work. If we use the ⅜-inch pushrods, we use the kicked-up Manley guide. It must be cut apart, so each half can be mounted under the respective studs, and welded back together. This can be a bit tricky, but there won't be any problem as long as you make certain the plate will pilot the pushrod into the rocker arm socket. Of course, these guideplates are made from hard steel and must be used with pushrods that are fitted with a hard-steel insert to prevent galling. The short 5/16-inch Smith polished pushrods don't have hard inserts, so we fabricate our own guide plates from ¼-inch stock aluminum. They look pretty much like the stock plates, as can be seen in the photos, but allow for pushrod offset.

The pushrods and guide plates in a Grand National engine cause little concern. We use the stock Chevy 5/16-inch pushrods with the hardened end inserts and the stock Chevy guideplates.

STUDS & GIRDLES

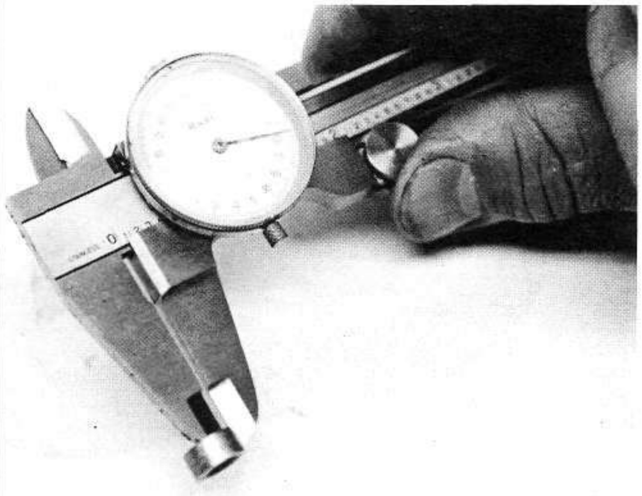
All of our racing engines are fitted with the stock 7/16-inch screw-in studs, originally fitted in the Mark engines. We know that some engine builders prefer the Diamond-Elkins version of this stud. It has a larger radius between the nut section and the threaded portion. This will make it stronger but, then, we have never failed one of the stock Mark studs.

Any competition engine with stud-mounted rocker arms can benefit from some sort of stud tie-bar to stabilize the valve gear, eliminating timing and valve clearance fluctuation. The most popular commerically-made stud bar is the well-known Jomar stud girdle which is now available from several sources. We use our own version of this heavily-patented device, a necessity because of the offset studs. We have experimented with different thicknesses and fasteners until we have come up with something that works very well.

Our version uses two ¾-inch thick bars. One is 1-inch wide and the other 1.5 inches. They are clamped together around the adjusting nuts by straight bolts rather than the U-bolts used in the original design. This makes the unit a super-stiff beam to support the studs and stud nuts. We also make our own version or modification of the long adjuster nuts. The original design calls for an adjuster that floats on the stud and locks only to the bar. We use a similar nut, but we also lock it to the studs. The adjusters are drilled completely through and small socket head lock screws are put in from the top of the adjuster. When the adjustment is set, the lock screws are tightened down against the top of the studs just as on a typical "Poly lock" type adjuster. When all the valves have been adjusted and the adjusters locked, the tie bars are cinched up tight to lock the whole upper valvetrain into a single solid structure.

CAM DRIVES

As far as we are concerned there is never any reason to drive the cam with anything other than a stock-type chain. We never use any sort of gear drive. In the past we have been involved in testing and design work with camshaft gear drives. The top-quality drives are very well made and they do "tie" the cam and crank together very solidly, as they are



When the cam is installed it must be timed with the crankshaft. Advancing or retarding through the use of offset bushings is the recommended procedure. Never trust the numbers stamped on the commercially-made bushings. Measure them and install as needed. Moving the drive pin 0.005-inch alters the timing by 1°.

When the Chevy pre-'67 chain assembly is used, we always substitute a crank sprocket from a Borg-Warner set for the stock gear. The B-W sprocket is steel and will not crack as does the stock Chevy sintered iron sprocket.

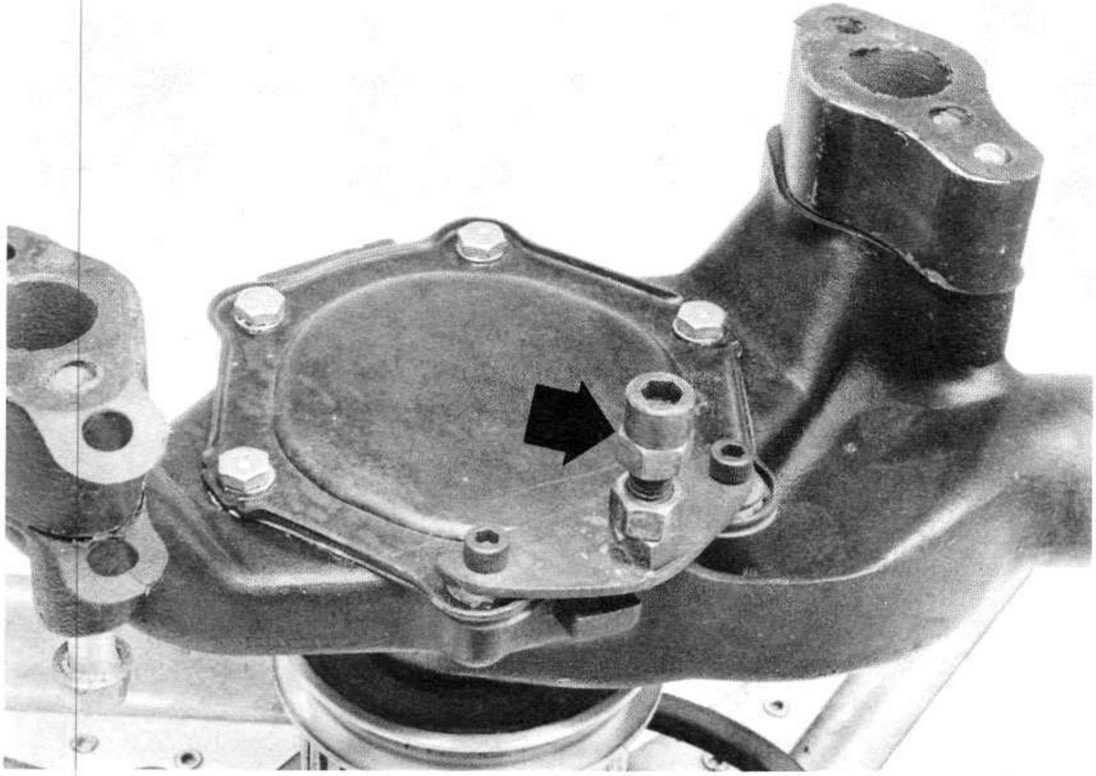
supposed to do. We currently feel this may be a bigger liability than advantage. When the crank and cam are joined by a third gear or a pair of gears, every bit of harmonics and vibration developed in the crank will be transferred directly into the camshaft and valvetrain. As it is, we are having trouble holding the valve gear stable, or reasonably so, and using a solid connection to induce more trouble isn't the way to go.

A chain acts as a damper to isolate the valvetrain from all the banging and rattling in the crank, especially the stock chain with an aluminum cam sprocket and nylon drive teeth. Even if there is some slack in the chain and the cam lags a little behind the crank, this can be compensated for when the cam is timed with the crank. However, just to be safe, we don't run a chain to death. They are inexpensive and we carefully inspect the sprockets and replace the chain every time the chain cover comes off during routine engine inspection.

All our Pro Stock engines are fitted with a standard Chevrolet Morse pre-'67 silent chain. This chain is wider than the later models. The cam sprocket is a stock pre-'67 aluminum casting with nylon teeth. It is checked for wobble and eccentricity before installation. The center hole is enlarged to accept a Teflon cam bumper, and the bolt holes and drive pin hole are drilled out to accommodate offset index bushings. The Teflon bumper is retained inside the sprocket by a washer that fits under the sprocket bolts. The crank sprocket is not a stock Chevy gear. All of the stock gears are now made from sintered iron and they will break. We use the sprocket from a Borg-Warner pre-'67 replacement set, as it is made from steel. This setup has never failed or caused one bit of trouble, but the chain must be serviced regularly when used in an engine with very high valve-spring pressure.

One important tip to remember when indexing the cam, do not trust the little numbers stamped on the offset bushings. They are never correct. We measure the bushings to check the exact thickness on the drive surface. Moving the cam drive pin 0.005-inch will change the timing 1° (crank). Determine where you want to move the centerline through one of the accepted methods and calculate how much the pin will have to be shifted. The pin is moved counterclockwise to retard and clockwise to advance the timing. We always recommend that the bushing be installed so the cam pin is driven by the thickest or thinnest section of the bushing, not one of the "sides." The bushing and sprocket should be snug against the cam face *before* the gear bolts are torqued tight. This prevents the bushing from twisting around as might happen if the sprocket is pulled onto the cam with the bolts. The bushing must be staked in place with a small center punch.

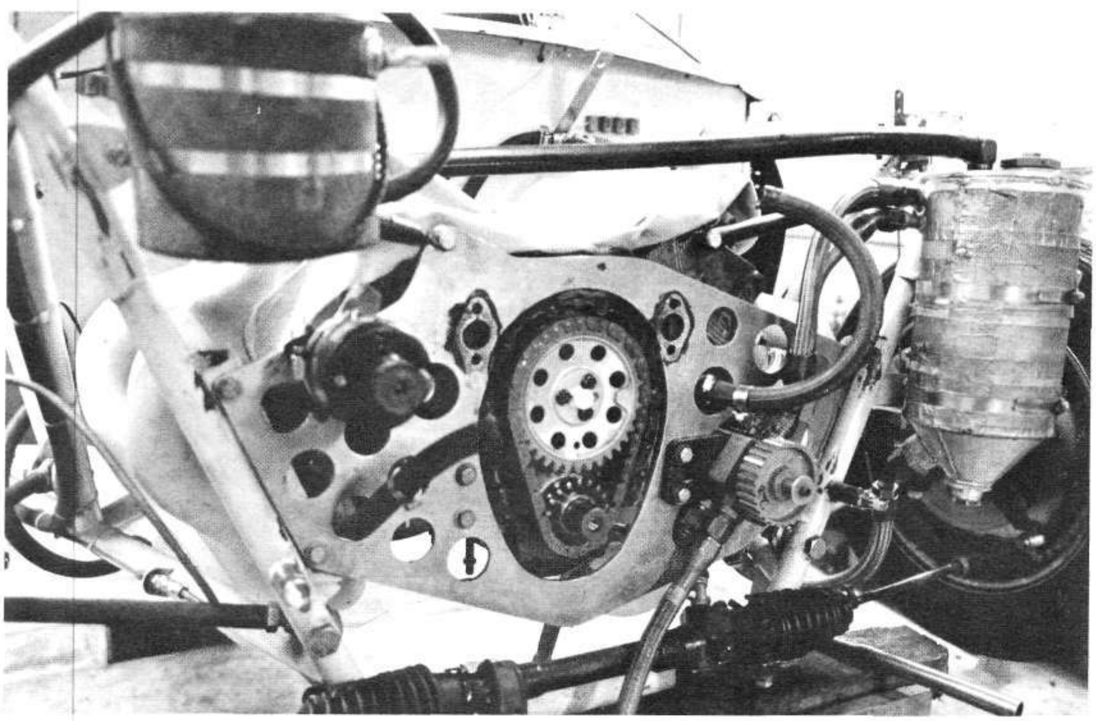
To drive the cams in our Grand National engines we use the double-roller heavy-duty Chevy truck chain and sprockets. It is more durable and heavier than the silent chain. However, the iron sprockets aren't compatible with the iron face of the block, and racing valve-spring pressure may cause it to start eating the block away. To prevent this, we mill the face of the block, behind the cam sprocket, to accommodate a 0.030-inch brass washer. The back face of the sprocket is also squared, and both surfaces are polished smooth. The washer is punched or cut from flat brass stock to act as a simple thrust bearing, eliminating galling between the two similar metal surfaces. There is also a Torrington bearing setup that can be used for this purpose, but it is more expensive. In any case, something should be used between the cam sprocket and block because the heavy spring loads of a racing cam and the taper ground on a flat tappet lobe will generate enough end load



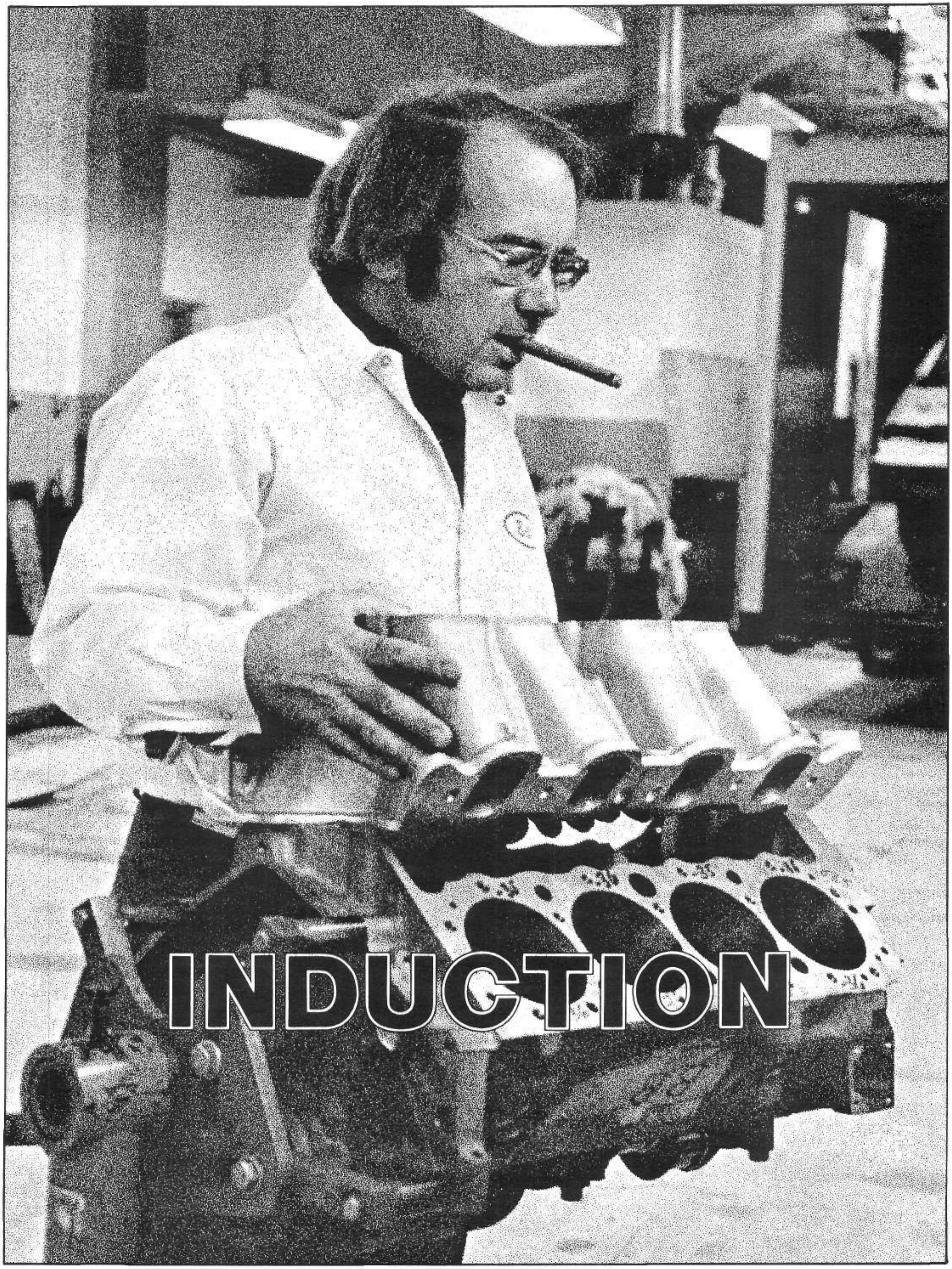
A camshaft thrust bumper is always used on the drag engines. A teflon plastic button is installed into the camshaft drive chain sprocket. With high-tension racing springs the button will just bend the thin stamped metal chain cover. The bumper is fastened to the back of the water pump and is positioned so it can be adjusted to counter the forward thrust of the cover, button, chain sprocket and camshaft.

to cause real trouble unless some precaution is taken.

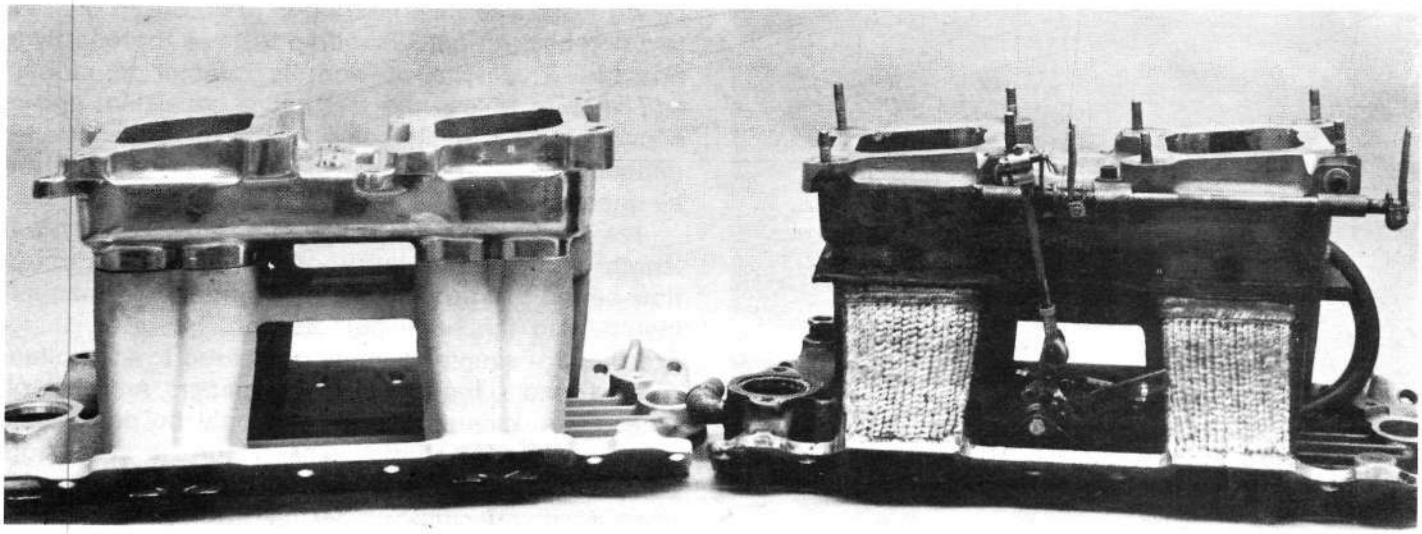
From the photos, it is obvious that we don't use rev kits with our roller cams. They're more trouble than they are worth. They help keep the lifters in the bore and save the cam if a pushrod or rocker breaks, but they make it impossible to change the cam without taking the engine apart. We feel maintenance is always an important consideration; you can't win races unless the engine is running. Toward this end, the photos show that the roller lifters have been drilled through from front to rear. We carry a pair of long rods with us. By inserting them through the oil drainback holes in the front of the block and through the drilled holes in each row of lifters, we can raise the lifters up off the cam, out of the way of the journals (the rockers have to be taken off the engine). This allows the cam to be pulled out of the engine without removing the manifold. When the new cam is in, the rods are pulled out, and the lifters fall back onto the lobes. The rockers are then put back in place and the valve clearance adjusted. In a tight fix we can get the whole job done in 10 or 15 minutes.



We never use a gear drive to propel the camshaft. Any solid gear connection will tie the cam timing very solidly to the crankshaft, but at the same time it will transfer and multiply resonant and vibration harmonics from the crankshaft to the camshaft. We feel this is too great a disadvantage to pay for the timing benefit. The chains must, however, be inspected and replaced often as the strain from racing valve springs will pull them out of shape quickly.



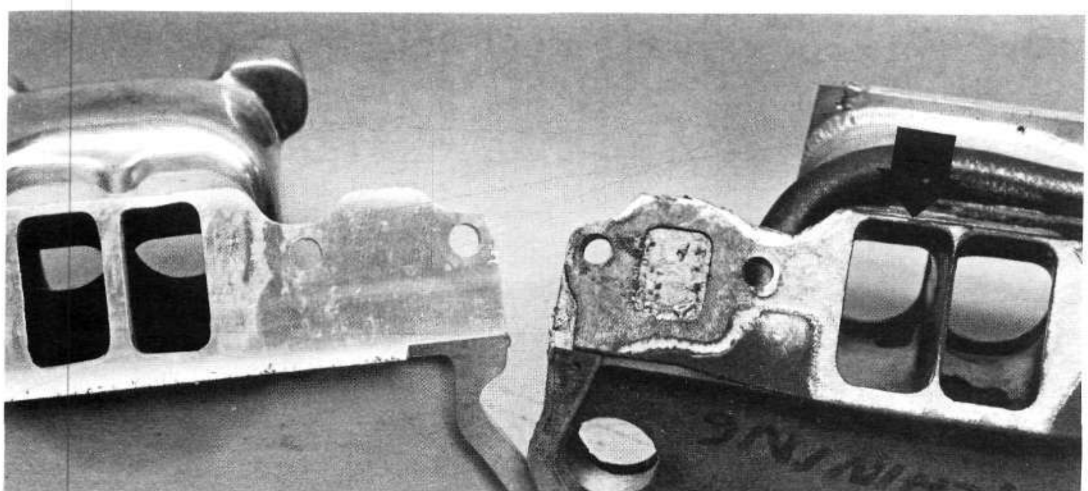
INDUCTION



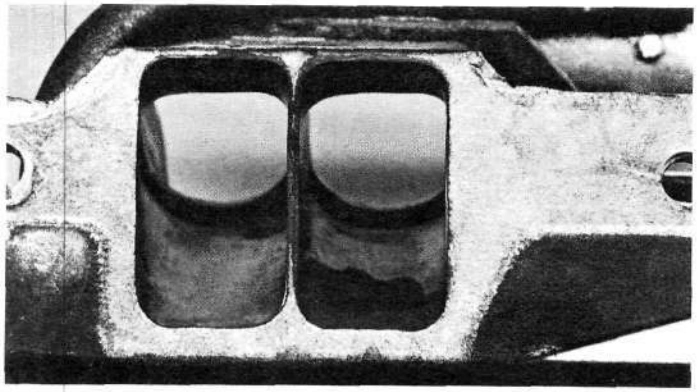
Tracking down elusive "driveability" factors is centered largely around the manifold and carburetion system. In the past we have spent many thousands of hours modifying stock tunnel rams like the Edelbrock TR-1Y (left) to be compatible with the drag race camshaft and rod ratio configurations. Enlarging runner cross sections, shortening the runner length, varying plenum volumes and shaping the runner entries is critical for off-the-line response.

Along with the cylinder head work discussed earlier, the induction system design is some of the most critical work involved in any racing engine development. Unfortunately, the work is also among the most difficult to successfully isolate and evaluate. We are in a fairly stagnant period with the current system because it works well, all things considered, and we are able to find substantial gains in other areas for the same investment of time and money. For instance, we have been able to find 10-15% horsepower in some recent oil system modifications—a very substantial improvement—but the very most we can imagine squeezing from the same effort on the induction would be 2% or 3%. Sometimes it is possible to spend an incredible amount of effort chasing down a 1% power gain and in the end we never know if such a dyno reading is legitimate, because it is on the fringe of our measurement error. Therefore, most of the information in this chapter is somewhat dated, yet these systems work well with the present competition engines. We have some unproven theories which may be worth more development but, unless some radical breakthrough in manifold or carburetor design appears in the future, what you see here will be the standard hardware for some time to come.

Occasionally we use the term "driveability." There are several definitions of this nebulous factor. In our testing we have stumbled across certain engine elements that cannot be proven on the dynamometer, but when tested in the car make a repeatable improvement in the elapsed time or trap speed. We feel these are subtle, unexplainable effects that may be key factors in making other variables in the overall picture "click." Certainly the experience and talent of the driver is a part of this gray area. It also includes such things as flywheel weight, engine "throttle response," the driver's acceptance of the throttle feel, accelerator pump discharge characteristics, carburetor intermediate-circuit mixture, idle-circuit mixture and others. We know there are many, many, fine-tuning devices which have to legitimately fall into this category, although they also exist inside that 1% unmeasurable-error field. Unknown phenomena are also often thrown into the driveability category, for lack of a better explanation. It is impossible to spend the effort to chase all these possibilities to the



Comparison of a stock TR-1Y (left) and our Number 2 manifold shows the extensively reworked runner openings. We have verified that there is a direct correlation between the displacement/operational speed of the engine and the volume of the runner. For extreme-rpm smallblocks the runner size must be increased in order to have enough mixture volume to feed the chamber adequately. Runner length and plenum volume will also enter the picture, as will the engine rod ratio and camshaft capabilities.



A big engine operating at 9500 rpm needs a lot of runner volume and big cross-sectional areas. The divider wall is as thin as possible, the walls and floor are as thin as the casting permits, and the top is welded so the roof can be brought up considerably.

ultimate end. Several of these elements are linked to the induction and how it functions, relative to the way the car chassis transfers the engine power to the ground and the way the driver controls this process. Given unlimited resources, it would be possible to squeeze some power gains from the current induction systems but, for the reasons outlined above, it would be prohibitively expensive at this time.

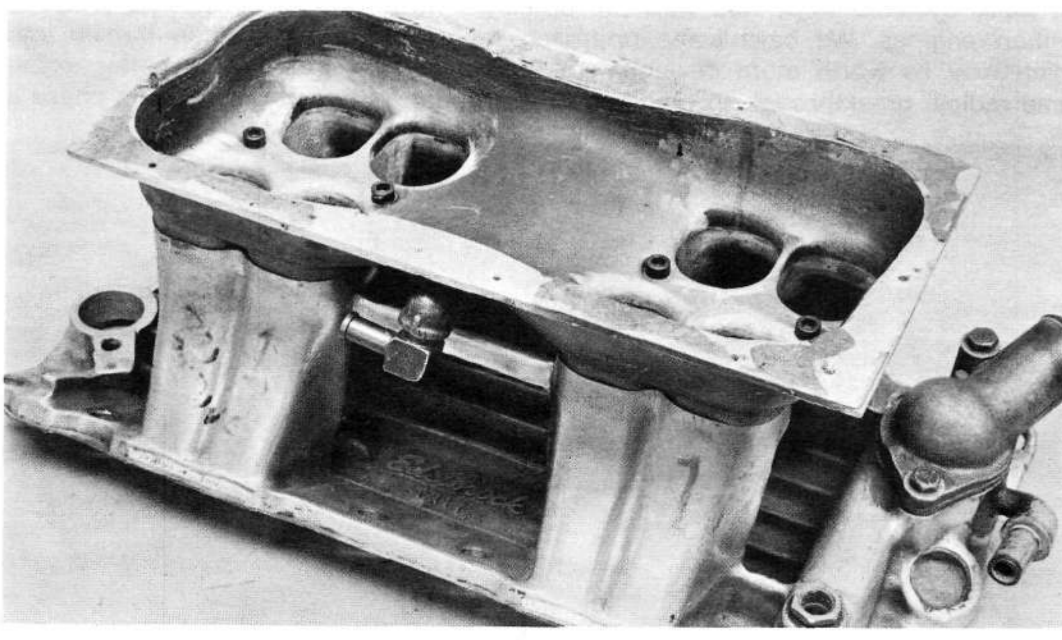
TUNNEL-RAM MANIFOLDS

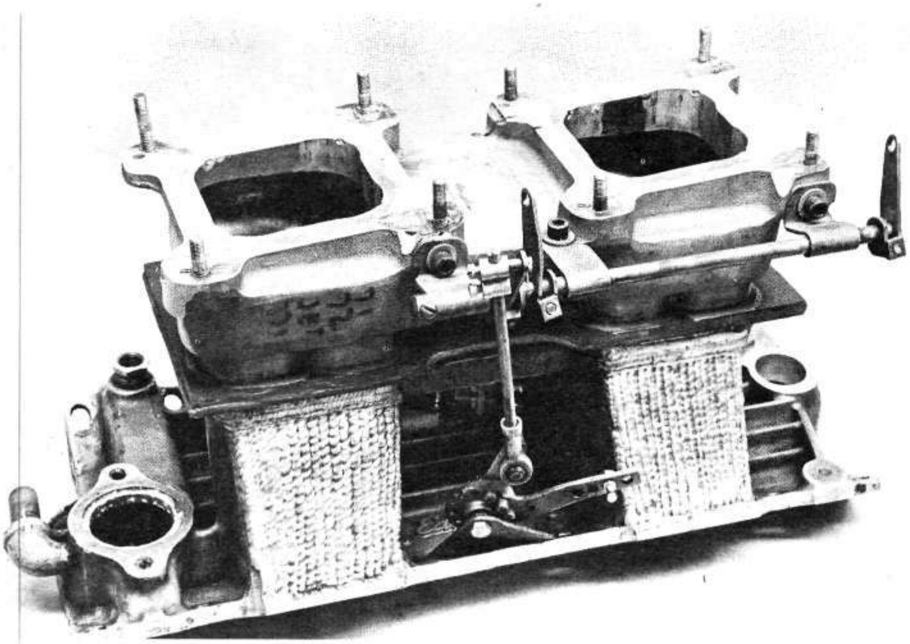
Our current Pro Stock smallblocks are all fitted with either Edelbrock TR-1Y or Pro Ram II intake manifolds. These have been modified to suit our specific requirements. Depending upon the engine induction-related variables, we may have to reduce the plenum volume, rework the plenum entry to the runners, shorten the runners or increase the volume of the runners. We have spent a great deal of time chasing these variables around and, in most cases, it is hard to give specific recommendations because of the elusive interrelationships involved. Most of the knowledge we have gained is the result of extensive strip testing. It is part of the driveability factor discussed earlier and cannot often be found on engine bench tests. The best method is to go out to the strip and flog back-to-back manifold tests as thoroughly as possible. This type of work is difficult to monitor and analyze unless the test car is consistent and as many outside variables as possible can be controlled (ambient air temperature, strip surface temperature, air density, etc.).

We can track down the effects of runner wall length and plenum entry or exit shaping through flow-bench testing with the carburetors and manifold mounted to the head port section. However, things like this are simple to study compared to the volume of the runners, the length of the runners, and the volume of the plenum. These can only be determined through dynamic studies. The picture is even more complicated when you realize these variables will change with the camshaft design, the rod ratio, the engine displacement and speed range. So, we are often happy to just "get in the ball park." Toward this end we have established a procedure that works adequately, even if it isn't too scientific. We have three different manifolds which can be swapped back and forth during testing. This gives us some idea of how the test engine is going to respond to vaguely defined variables. The three units vary in plenum size, runner length and runner volume. One is fairly small in all three areas, one is about average, and the other one is large in size. Thus we have a "little," a "medium," and a "big" manifold. As things now stand, we don't believe any of these manifolds represent the ultimate for a specific engine design, but we have isolated "tendencies" related to the aforementioned parameters. Given time and resources we could probably nail this down to more detailed specifics, but further effort would net minor gains in comparison to the required expenditure of resources.

The stock Edelbrock TR-1Y has a plenum of approximately the correct volume for a hard-working 350-inch engine. It may be a bit large for smaller engines, and a stuffing block between the front and

Our "arsenal" of test manifolds includes three modified TR-1Y Edelbrock manifolds. Each has a unique combination of tuning parameters. The number 1 piece has stock length runners and the plenum volume is stock. The plenum has, however, been cut apart to ease the task of radiusing the runner inlets. The inside volume of the runners has been increased to the limits of the stock casting material. It is most suitable for larger engines with lower rpm capabilities.



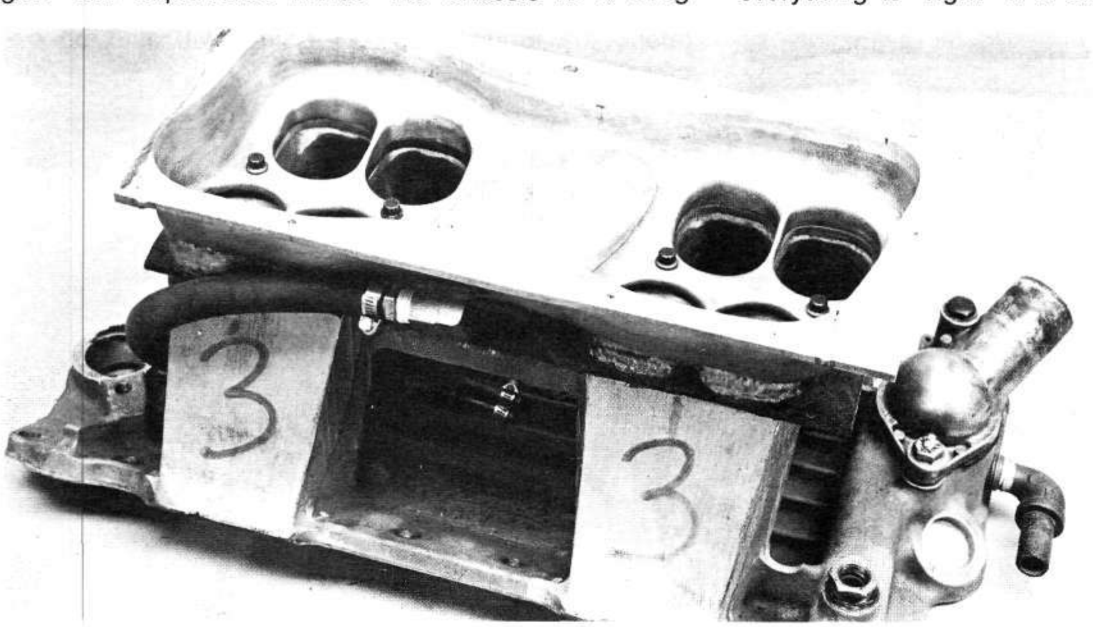


The number 2 manifold is reworked a good deal more. The runner length is shortened 0.60-inch for high engine speeds. It has been fitted with the "X-type" mini-plenum to improve accelerator pump response. The runner entries have been radiused only slightly. A heat isolator is placed between the plenum and the runner base. The base is welded to raise the outer wall and the runner volume is larger than the number 1 manifold.

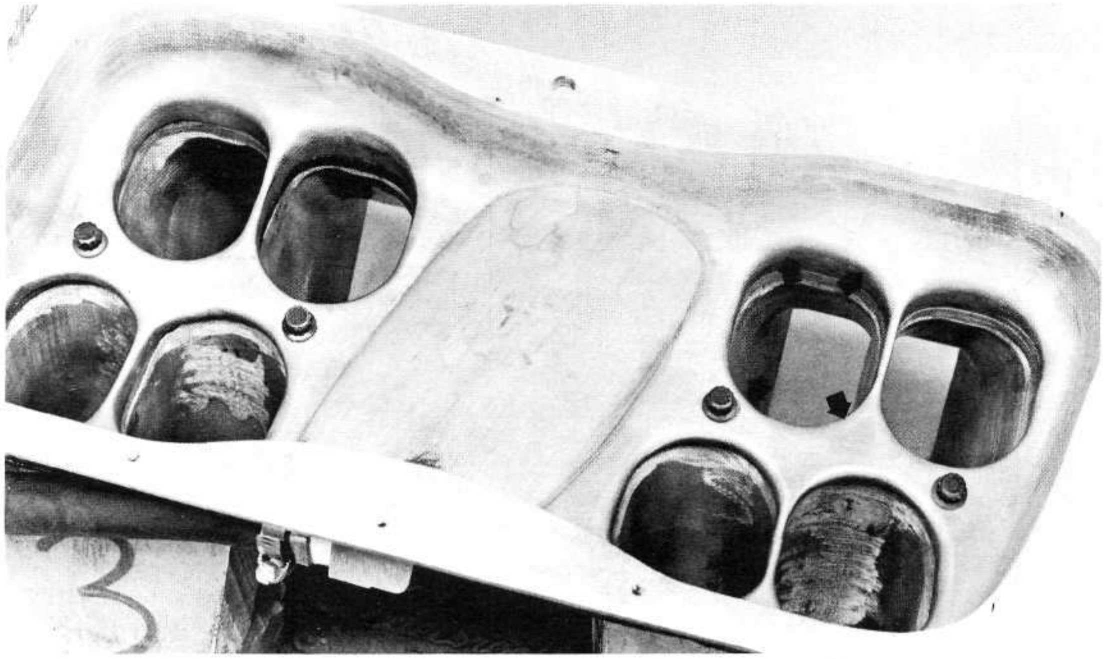
rear runner openings should improve the driveability somewhat. This assumes an unlimited camshaft similar to what we run, a good set of 292 heads, and a rod ratio in the neighborhood of 1.8 to 1.9:1. The stuffing blocks as shown in the photos work adequately and serve to reduce "puddling." We also cut the plenums apart to gain access to the interior. By using bolt flanges and gaskets between the upper and lower halves of the plenum, spacers may be used for some measure of fine tuning of the plenum volume. We make these spacers of ordinary sheet fiberboard. They are easy to cut and shape. The fiberboard also acts as a heat barrier or insulator to keep engine heat away from the carburetor bodies.

If there is any one variable which must be worked out in the car, it is plenum volume. This is almost totally a driveability factor and is difficult to find on a dyno. We simply can't totally duplicate the same running conditions inside the dyno room that the engine will experience inside the chassis at a drag strip. Most of our dyno testing is performed at a maximum engine speed of 8500 to 8750 rpm. At the drag strip we easily run the machine to 9500 rpm. Therefore, some high speed problems as experienced on the track won't show up on the dyno. The same is true of sudden throttle applications under loading. You can't "feel" how the engine takes throttle, under acceleration load, when it is bolted to the bench. This is something the driver has to sense. We feel a good place to start is with the plenum slightly smaller than the smallest volume possible without incurring power loss on the dyno. An overly-large plenum will require a tremendous amount of accelerator pump squirt to get things going. This is sometimes difficult to achieve if the plenum is too large, and usually leads to poor response problems.

Any change in the plenum height is also going to alter the jet requirement. Reducing the volume generally allows you to run a smaller jet size. When everything is "right" it is easy to get consistent plug



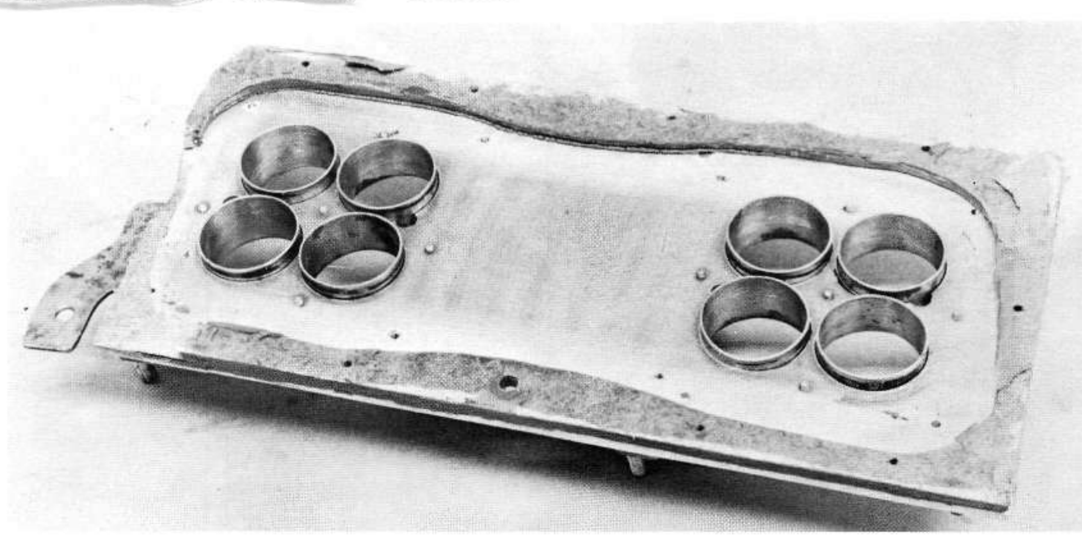
The number 3 manifold is our largest. It has an early plenum but a stuffing block has been added between the front and rear pairs to reduce the overall inside volume. The runners have been welded and the inside grinding raises the roof considerably along the full length of the base. The outer wall of the short runner extensions in the plenum have also been welded to allow the inner surface to be brought outward gaining even more volume.



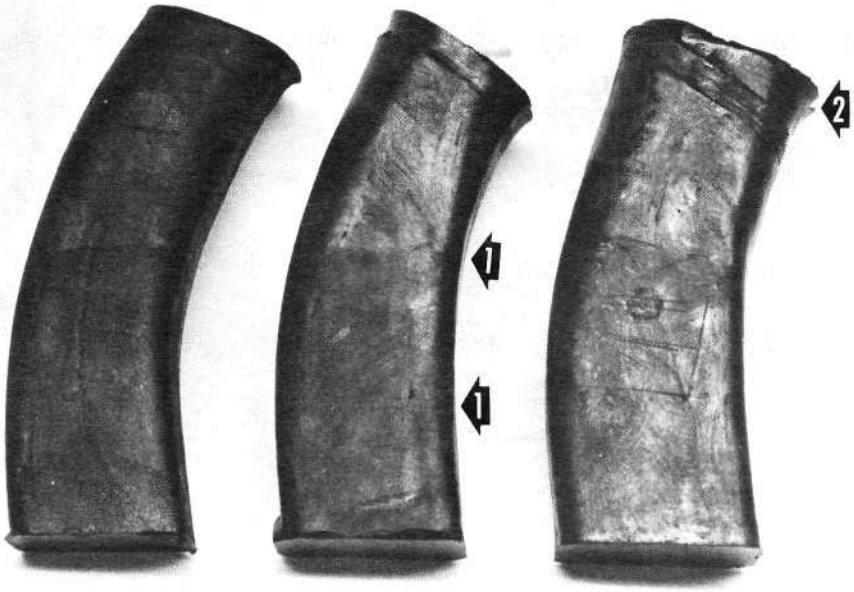
Before the current mini-plenums were available we had to modify the existing plenums a great deal to reduce the volume. This helps the engine pick up the effect of the carburetor accelerator pump discharge as the clutch load is dumped on the engine at the starting line. Lowering the roof of the plenum and bringing the carbs nearer the runner entries helped, but we also used this varnished wood stuffing block to reduce the volume even further.

readings. An overly-large plenum can make it extremely difficult to get good readings.

Runner length and runner volume are closely related factors. They are not nearly as difficult to track down as the plenum size, but this is not to say they aren't important to driveability. At least they produce some response on the dyno, although in-car testing is still the final test. The runner length is a function of the approximate engine speed at which maximum power is achieved. This "tuned induction" effect is not new. As the desired power point is raised in the speed band, the overall length of the induction from the runner entry to the backside of the valve should be reduced. Of course, the corollary is true. As the speed is reduced, the induction length should be lengthened. But, and this is a big but, it is not advisable to just chop the runner length off and go racing. There is another effect to be considered. As the runner length is reduced, the resultant volume of the runner is also reduced, provided all else remains the same. Unless some sort of compensatory action is taken, the reduced volume may offset any gains from the tuned length. We feel the intake runner volume requirement is a function of the total engine displacement, rod ratio (actually we are considering a single cylinder so the runner volume is tied to the volume of the single cylinder it feeds) and the camshaft considerations discussed in an earlier chapter. Therefore, if an optimum runner volume has been achieved, but the overall runner length is shortened in an effort to get a higher tuned speed range, it may be necessary to enlarge the cross section of the manifold and/or port runner to regain the required volume. The volume requirement appears to be a direct relationship with the cylinder volume. As the cylinder displacement gets larger, the required runner volume becomes larger, and vice versa. This assumes the rod ratio and cam remain the same. As discussed earlier, if the rod ratio is changed, the required intake volume will be affected. As the ratio is increased, the required intake volume will be decreased. And, if the cam displacement angle is increased for some minor tuning effect, the required intake volume will also be decreased.



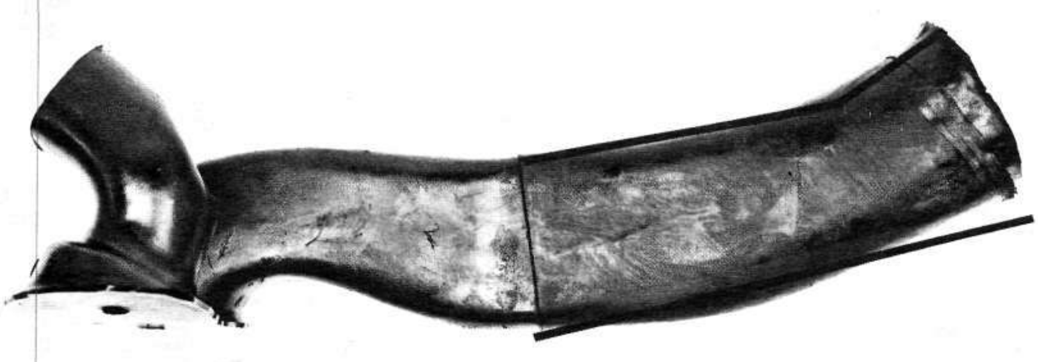
In conjunction with the reduced-volume plenums we found the engine displayed better "driveability" response at the track with bore extension tubes fixed beneath the carb throttle bore openings. These extensions have the effect of bringing the throttle bores closer to the runner entries without actually reducing the plenum volume. This is particularly helpful at launch. It gets the accelerator pump fuel into the runners very quickly and minimizes the dispersion that occurs when the pump fuel enters the turbulence in a wide open plenum.



These are molds from each of our test manifolds. At left is the smallest, number 1, and at the right is the largest, number 3. It is easy to see the volume differences in this comparison. The number 2 runner (middle) shows the effects of welding and grinding the outer wall of the runner base (arrow 1). Carrying this even further with the number 3 manifold, we gained a great deal of volume, but carried the technique further up in the runner and into the plenum extensions (arrow 2). Each runner type is best suited for a specific rod ratio and engine displacement.

With these general principles in mind we use the following general specs for our racing engines. The Pro Stock 330-inch engines with 1.8 rod ratios seem to prefer a plenum volume 3-4 cubic inches smaller than a stock TR-1Y, a runner length 0.6 inches shorter than stock, and a related increase in runner volume. A 354-inch drag engine likes a stock plenum, a runner length 0.6 inches shorter, and a larger volume. Engines varying in displacement or engine power band should benefit from proportional variations, according to the guidelines in the preceding text.

Once the basic volume and length requirements have been worked out and the plenum volume is determined, the manifold is cut up and welded as required to gain these dimensions. This gets very hairy around the area where the runners join with the plenum floor and where the runners mate to the head surfaces. If the runners have been drastically shortened, some fiberglass and epoxy fabrication may be required to get the runners and plenum matched. When volume enlargement is required, the outer wall of the runners must be welded up, and many hours of grinding inside the runner are necessary to get the outer wall raised (the outer wall becomes the roof port inside the head and the inner wall becomes the head-port floor). This work must be coordinated with work performed on the head castings to make the transition from the manifold to the head as smooth as possible. In this respect, the stuffing plates can be a big help. When properiy designed they will get the manifold in a correct relationship with the port, and they provide extra material to make the flow transition between the two components. This is particularly important along the manifold outer wall and head-port roof of the tunnel ram manifolds where the majority of the high velocity flow activity is centered. This gets very touchy and gasket sealing is often a problem. It may even be necessary to weld additional material above or below the port in order to gain sufficient sealing surface for the gaskets. It is also possible, as shown in the photos, to just add small aluminum shims, screwed into the floor of the runner, down below the port opening, to act as a gasket-bearing surface. Of course,



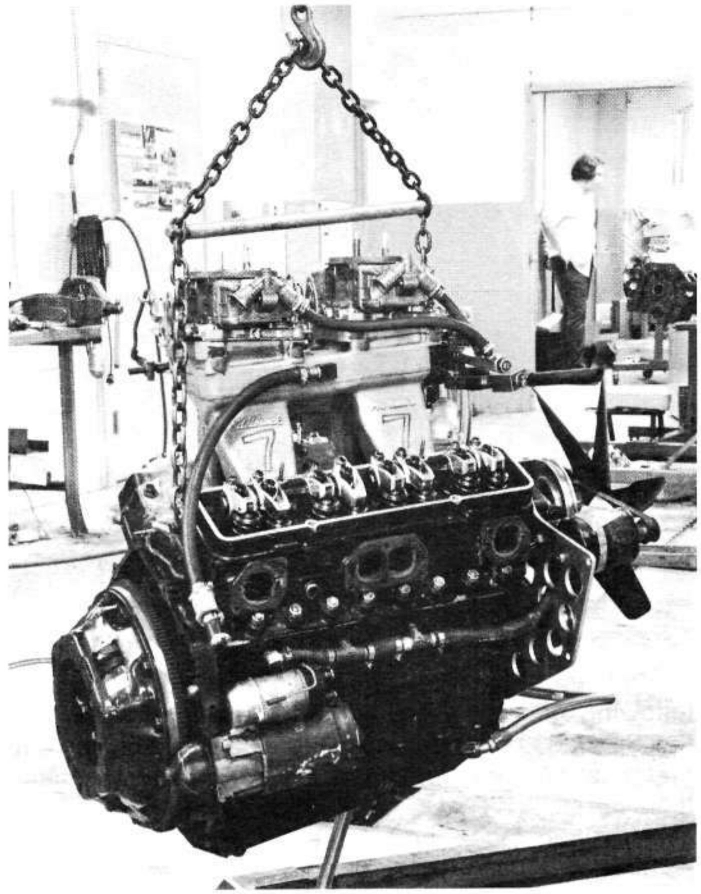
The matchup between the manifold runner and the head port is very important. These two molds demonstrate a good transition from the runner to the port. This is our number 3 manifold with our largest 292 ports. Recently we have been running the Edelbrock Pro Ram II or "Z manifold" with excellent results. It has a radically different runner shape. The roof is nearly straight and makes a sudden, sharp turn up near the runner entry. At the turnpoint the roof is very fat and wide. The floor is also very straight, as indicated in the illustration.

adding material to the top of the head-port runner can be very difficult because the rocker arm covers get in the way.

All of this may become much easier in the near future. We are now using Edelbrock's Pro Ram II. It is somewhat shorter than the current TR-1Y. The outer walls of the runners are raised some and they have additional material thickness. This should make it easier to gain the required runner volumes without excessive welding. Gasket sealing is also improved and it appears to have all the other good features of the TR-1Y. It should be an excellent piece to use as a starting point for an engine like our 330-inchers.

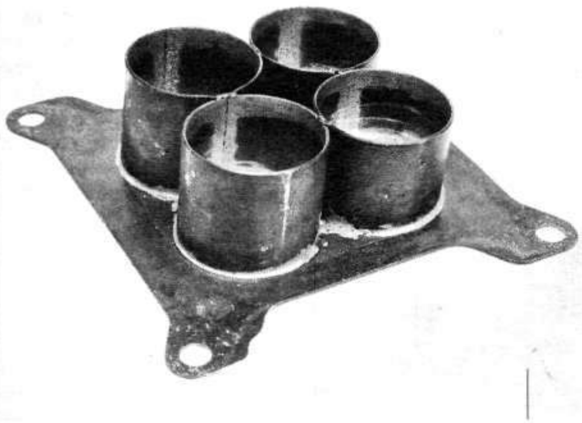
As far as flow efficiency goes, we find that most of this work must be done in conjunction with head port development. Exact shaping can only be determined through head port/manifold/plenum/carb flow studies as a unit on the flow bench. Barring this, most of the current tunnel ram-type manifolds can benefit from some slight touch-up at the point where the runners leave the plenum. The radius of the departure from the plenum into the runner is very important. The total length of each side or wall is also important, relative to the total runner/port length and the dynamic effect this will have on the flow over the hill, into the valve seat/exit area. The details will vary from engine to engine as the cam/rod ratio/displacement/engine speed requirements vary. For the lack of better expression we can only say the entry to the runner must be smoothly radiused and any transitions from the plenum to the runner and from the runner to the head port must be absolutely smooth. We do not use any mismatch to reduce the so-called "reversion effect." Everything is as aerodynamically "slick" as possible.

When we cut the plenums open we replace the upper half with a flat bolt-on plate, to which the carburetors are mounted. We also use this plate as a convenient way to add mounting tabs for the throttle linkage and other gear. Under the carb throttle-bore openings in the plate we use bore extensions. Note



The Pro Ram II is a very radical departure from the previous Edelbrock tunnel rams for the smallblock. The floor of the plenum is angled toward the heads and is not flat. The entries are nearly perpendicular to the centerline of the runners and as the runners rise toward the plenum they get wider. The plenum volume and shape seem to work very well with 330-inch and larger engines, but we are still gathering data on this manifold.

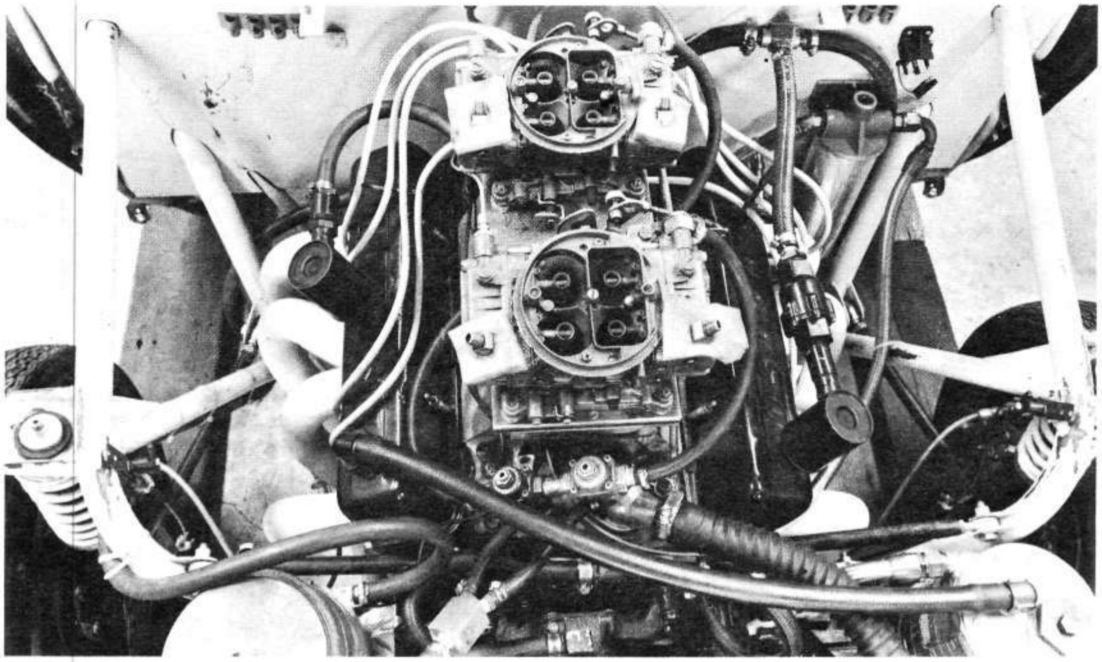
that the individual extensions match the bores of the carburetor throttle. These extensions are an attempt to get the bore openings closer to the runner openings in the plenum, without lowering the carbs so much as to reduce the plenum volume more than necessary. These plates or similar production plates are now being sold by Moroso. They are called "reversion plates." In actuality, they have little to do with the reversion or manifold backflow phenomenon, but for lack of a better name they are now commonly called reversion plates. They may possibly reduce the runner pulse signal to the carb somewhat, although we feel the big advantage is in getting the fuel exit from the carb down closer to the entry area. This reduces turbulent fallout or acceleration inertial pull toward the rear of the plenum. When the bore extensions are installed, the engine will usually require an increase of one jet size in all the carb feeds. This is another one of those hazy effects that we can't track down on the dyno, yet we feel the car gets a little more "push" in low gear when they are used.



The bore extensions will not show improvement on the dyno but they help launch the car. The diameter of the extension tube must match the carburetor throttle bore diameter. Similar plates are available from Moroso.

DRAG RACE CARBURETORS

We have never found any carburetor that runs better on the TR-1Y plenum than the number 4224 Holley 600-cfm center-squirters. We have done some

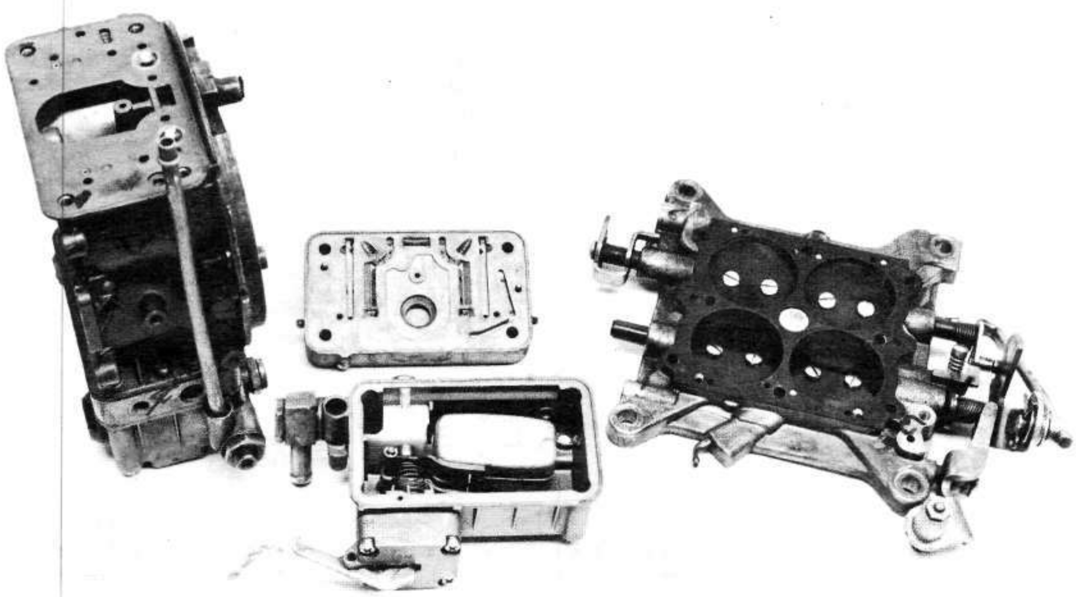


On all the Pro motors we use Holley number 4224, 600-cfm, center-squirter carburetors. We have never used any other carb that gives the driveability of this model. Some of the 750-cfm double-pumpers and the 4500 Dominators will very nearly equal the center-squirters for absolute power, but we have always been able to get the car down the strip quicker with the 4224's. These carbs will, however, require some reworking before they are totally suitable for unlimited drag racing.

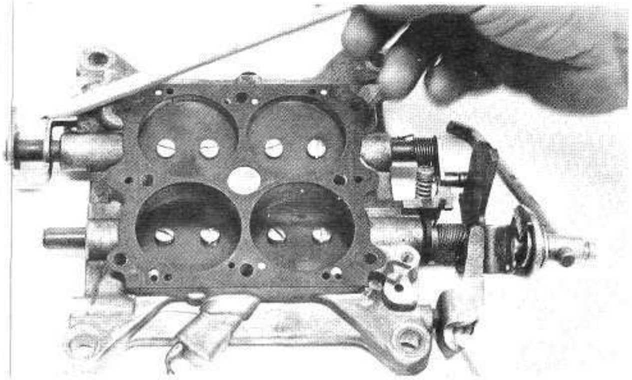
work with the big 4500-series carbs. This will be described in the following section. We have also dyno tested the 750-cfm and 850-cfm double-pumpers, but the power was about the same, with some minor variation. However, the fuel curve was not as good, with the top end showing a definite rich fuel specific. We never ran these carbs on the car, so we can't make a knowledgeable judgement of their driveability, although we suspect this could be a problem. It certainly could be worked out with some effort, but we are currently pretty happy with the 660's and don't see a need to flog the bigger carbs. We have also tested the new Carter Competition Series four-barrels on tunnel ram inductions and they show about the same peak power as the good center-squirters. Here again, we think working out the driveability could be troublesome and we don't have the time to run this sort of thing down when there may be nothing to be gained relative to the required effort.

To get the 660's working on a Pro Stock engine requires a little special attention, although this isn't anything too complicated. Briefly, we cut off the choke horn, tune up the accelerator pump, get the vents and floats in shape, clean up the idle and mid-range circuits, and adapt the 850-cfm throttle body to the main body. The details as described below are the same as the carbs used on our record-holding Pro Stock engines.

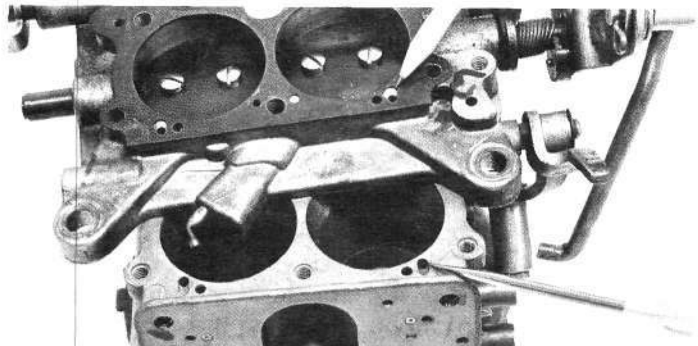
All of our present carbs are fitted with double-pumper throttle bodies that measure 1¾ inches across the throttle bores. This change improves both the driveability and the power of the engine. The body can be obtained from the R-4296-AAS Holley 850-cfm double-pumper as used on the 1969 L-88 Corvettes or from the R-4223-AAS 800-cfm center-squirter carbs. The current body from the carbs of the 4770 series will not work because the linkage requires a mounting boss on the side of the main body. This boss is not present on the 4224 main



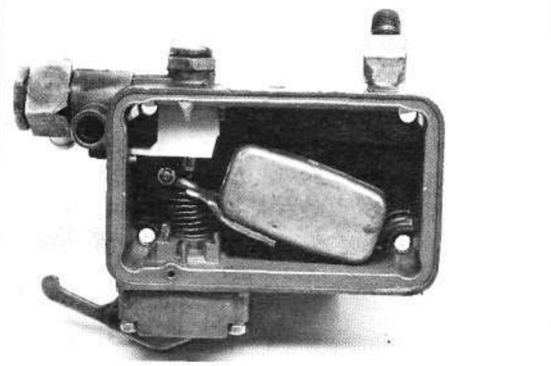
The 4224 is an excellent drag racing carburetor. We have found that with the current induction requirements of an extreme-rpm 330-incher they must be modified slightly for more air flow. We have a few other little changes that help fuel control under the somewhat erratic conditions created in a Pro car.



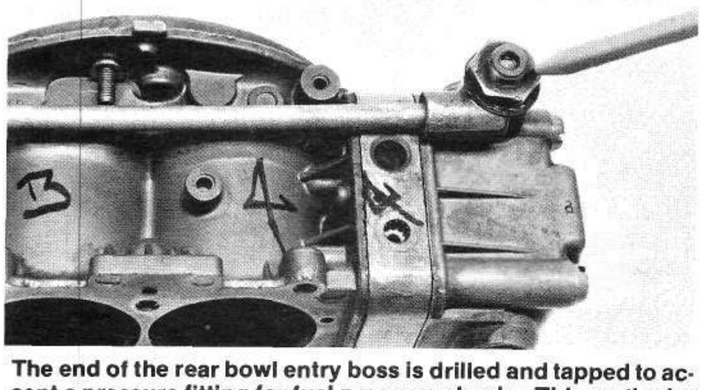
To increase throttle bore size we install the 1¾-inch throttle body from an *800 or 850 double-pumper to the 4224 main* venturi assembly. The second pump is not required.



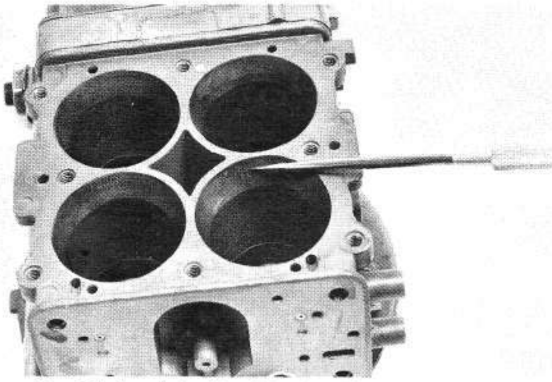
The curb-idle feed channel in the 850 throttle body will not align *with the corresponding passage* in the 4224 venturi body. We grind a small slot in the throttle body to join them together.



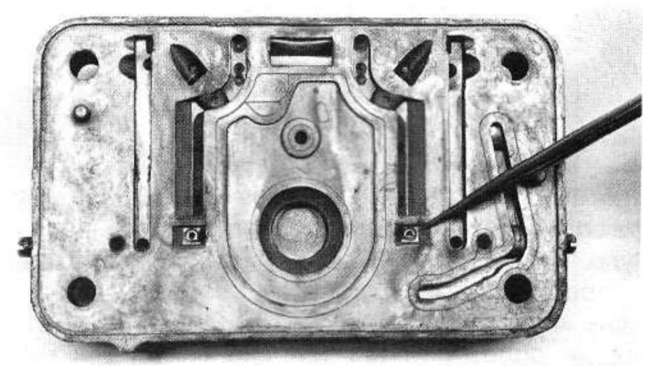
We use the brass floats from a number 1849 or 1850 Holley in the late side-inlet bowls. They give better bowl filling and control during hard acceleration. Note 50 cc pump, new vent.



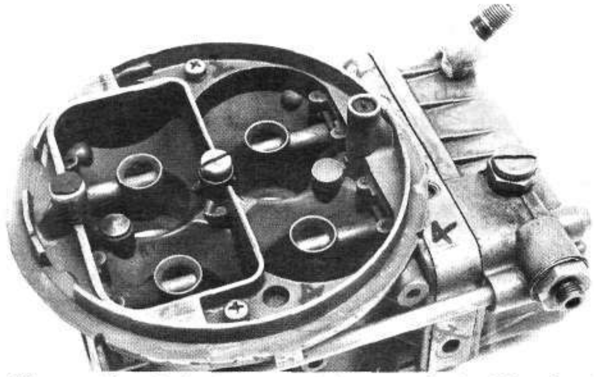
The end of the rear bowl entry boss is drilled and tapped to accept a pressure fitting for fuel pressure checks. This particular fitting has been plugged as it is not in use.



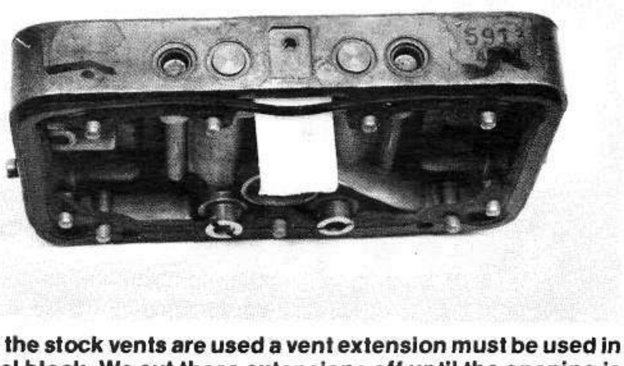
When the large throttle body is fitted to the 4224 main body, the *bottoms of the main body venturi bores must be chamfered to* match with the larger 1¾-inch openings in the throttle body.



High idle speed settings in a drag engine often get the throttle blades up into the idle transfer slot, enrichening the mixture excessively. We restrict the idle jet with a 0.017-inch wire.



We mill away the choke horn and chamfer the inside edge. The air cleaner bracket holes are blocked with screws. The stock vents are blocked and a revised vent is used.



When the stock vents *are used* a vent extension must be used in the fuel block. We cut these extensions off until the opening is approximately in the center of the bowl.

body. This is a bolt-on swap except for enlarging the body throttle bores and matching the curb-idle feed holes. The feed holes on the big-bore throttle body is outboard of the corresponding feed hole in the 4224 main body. To match the holes, a small feed channel must be ground in the 4224 body (see photos) and the big-bore throttle body gasket must be cut out to match the small feed channel. The bottom portions of the 4224 main body must also be bored larger to match with the 1¾-inch throttle openings.

We fit the secondary side of the carb with a metering block and the long transfer tube as supplied in the Holley 85R-3548 conversion kit. We remove the stock floats and substitute the old-style square brass floats from early 1849 or 1850 carbs. We have played around a lot with float design and shape in an effort to gain adequate bowl filling and stable fuel level during hard acceleration. These floats are about the best stock-type floats available. The center of buoyancy is slightly ahead of the bowl center and the pivot of the needle and seat is located to give better bowl fill upon acceleration. At the same time, we block the stock bowl vents at the point where they open into the main body air entry. We just pull the stand pipes out and fill the openings with a gasoline-resistant epoxy. To vent the bowls, we drill a small opening in the upper corner of the bowl roof, across from the fuel inlet. This is the forwardmost corner of the bowl when the carbs are mounted sideways. We thread the opening and screw in a small 45° brass fitting until it just barely reaches into the air space above the standing fuel level in the bowl. These vents must extend up into the sealed-off area of the hood air scoop, and must be aimed in such a manner that they are not pressurized by the air velocity moving down the scoop. This is one method to eliminate the problem of bowl spewing or inadequate pressure equalization. We have also used long hoses, clamped to the stock vents, that extend up into an area of the hood scoop where there is less air velocity. They work as well as the brass fittings but are somewhat more troublesome because they have to be removed whenever the hood seal is taken off the car.

The 4224 idle circuit requires some "fixing" in order to achieve a clean idle with a radical engine. The idle circuit in the number 5913 primary metering block supplied with the 4224 was originally designed for a carb with a power-valve system. The 4224 doesn't have a power-valve circuit and the idle circuit may be very rich with the normal race engine idle settings. This is a matter of cleaning up the off-idle settings rather than the absolute closed-throttle condition. However, we normally have to get the throttle open a little bit before the engine gains an adequate idle speed. This gets the throttle blades into the idle transfer slot and the mixture gets rich despite adjusting the curb-idle screws as lean as possible. In stock shape the mixture goes rich as soon as you crack the throttle.

We have successfully countered this problem by increasing the idle-feed restriction to the idle well (decreasing the size of the idle jet, at the point where idle fuel is fed from the main well into the idle well). Best results are obtained when this restriction is increased by the area approximately equal to the cross section of a 0.017-inch diameter wire. This is best accomplished by removing the small brass jet pressed into the idle-well feed channel entry in the metering block, and replacing it with another jet that can be drilled to the required smaller restriction area. This is very difficult to accomplish without considerable experience and practice. We have obtained the same results by installing a 0.017-inch wire into the stock restriction hole. We insert the wire into the feed restriction and loop it into the fuel feed from the main well to hold it in place. This works fine as long as you are careful when removing the metering block gaskets; otherwise the wires may fall out.

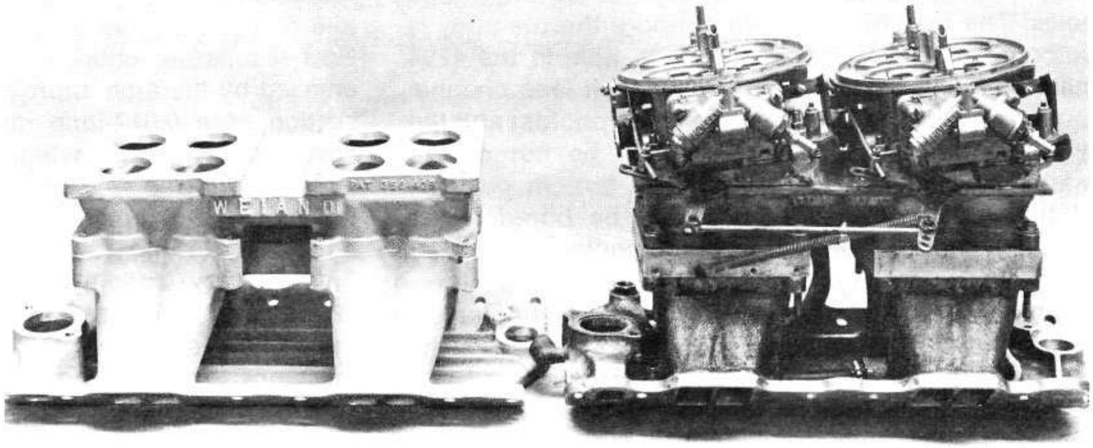
If the carbs are used with a fairly large manifold plenum, the accelerator pump discharge nozzles will have to be enlarged. We usually prefer to drill another discharge hole alongside the existent hole rather than just enlarging the single hole. This improves the atomization and will pick up the response. On the 660 carbs we will enlarge each of the existent center-squirter discharge passages up to 0.037-inch in diameter. If the engine requires more discharge area than this, we go to a second hole. We may double them up with a diameter of 0.025- to 0.027-inch each. Along with this work we spend some time getting the accelerator pump cam in shape. The best advice we can give is to just look for some dwell before the pump discharges completely, in order to have some left when you dump in the load at the starting line.

We have fooled around with different pick-up points for the secondary throttles. Despite great effort we could never find any real improvement in this area. Some secondary delay is required, as one-to-one throttles are totally undriveable.

Jetting will vary depending upon engine size and manifold, although they should fall in the range from number 69 to 75. We cut the choke horn extension off the primary side to gain additional clearance over the top of the carbs for the hood scoop. This increases the flow capacity of the carb marginally and may help fuel mixture distribution in the engine somewhat. We read our fuel pressure from the secondary carb bowl entry. The intent is to get the reading at the point where the greatest pressure drop exists. This will be the entry to the bowl that is furthest away from the pump. The 4224 makes this very easy. We simply drill the end boss of the transfer tube receptacle on the secondary bowl of the rear carb. The hole is tapped and a small adaptor is put in to monitor the delivery pressure just upstream of the needle and seat.

DRAG RACE DOUBLE-PUMPERS

We have not been able to substantiate this on our 330 engines, but other sources indicate that, with the



At various times we have researched alternative inductions to the 660 setup but none have been able to produce determinable improvements. During extensive dyno testing we developed some of the pieces that were to become the forerunner of the Weiand manifold for the Holley Dominators.

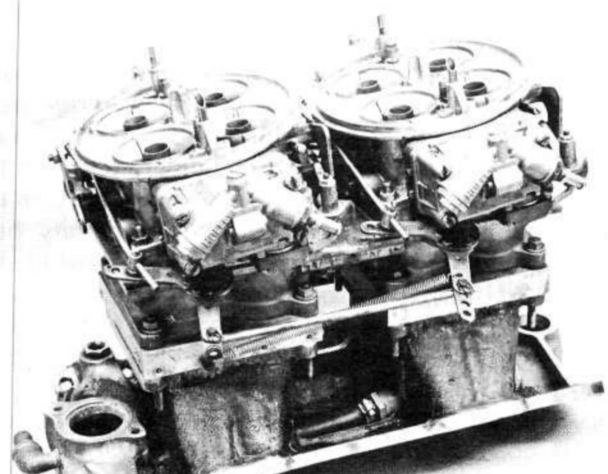
short plenum manifolds, a pair of 700- or 715-cfm double-pumpers may prove to be slightly better than the 4224 center-squirters. We still feel the 750-cfm double-pumper is, at best, a standoff with the 660.

We have only limited experience with double-pumpers on a tunnel ram manifold, but they don't appear to require idle-circuit work as do the 4224's. We don't feel these carbs need the big 50cc pump on the secondary side. This is too much accelerator-pump volume for most manifolds. With 0.037-inch primary squirters and 0.033-inch secondary squirters and the right pump cams, this carb gives respectable driveability. Of course, you must use the cathedral float bowls, unless you go to the effort to fabricate a secondary pump on the side-hung bowls. With the carbs mounted sideways, we normally cut the plastic "whistle" vent extensions off so they open in about the center of the bowl. When the carb is mounted sideways this gives better fuel control going around corners. Otherwise, fuel control isn't much of a problem with the big capacity float bowls.

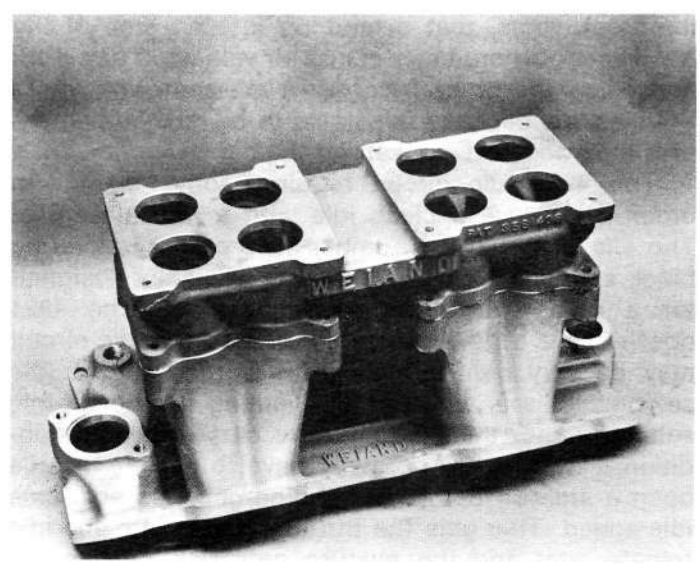
INDIVIDUAL-RUNNER MANIFOLDS

For our specialized drag racing requirements we feel the best alternative to the 4224 carbs may be the Holley racing series 4500 carbs on an individual-runner manifold. We briefly flogged this program a few years back, but we ran into several problems. Now, we feel the picture looks much better, and if there are gains to be had from a dual-quad tunnel-ram induction, they will come from renewing this research. Holley has a new model 4500 carb with greater flexibility and we have made improvements in the cam/short block combination that may broaden the "peaky" power curve associated with non-plenum induction. The manifold picture is still pretty grim although this is something we can overcome. With hardware similar to the experimental manifold shown here, we have a good place to begin. These pieces are the forerunners of the Weiand Holeshot manifolds and could conceivably be converted to IR use.

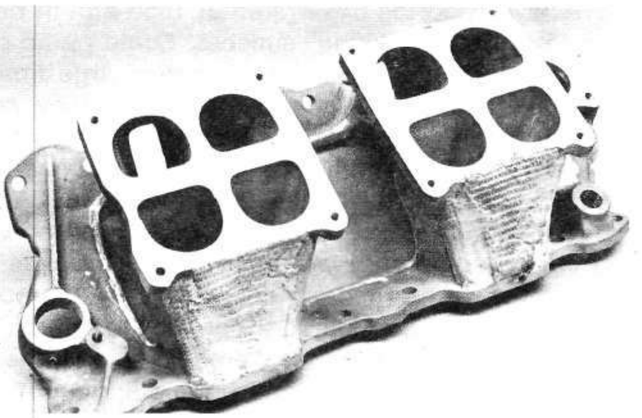
When all of the interest was first generated in the IR inductions, we ran into the same problems every-



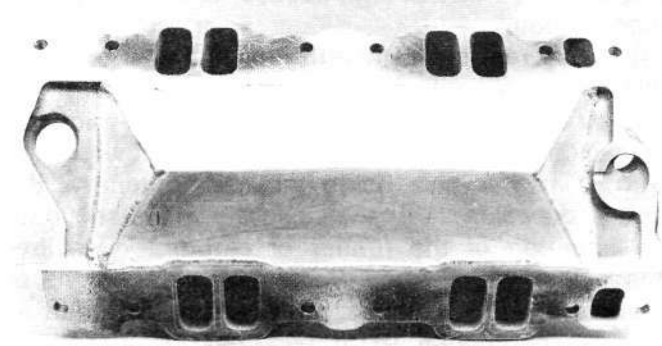
The big Holleys cause some problems with overall induction height and current hood scoop limitations. They are, nonetheless, excellent racing carburetors. With very short runners and some sort of mini-plenum they produce impressive dyno figures on 350-360 drag engines.



At the drag strip the 4500/Holeshot combination would not get the car moving fast enough. This could be worked out but we have found greater gains in other areas and have been forced to discontinue induction testing because of our limited resources.



The best application for big carbs like the 4500 is on individual-runner manifolds. We have successfully built some prototype IR manifolds for this purpose but the venturi area of the current 4500 cannot feed a 330-inch Pro engine.

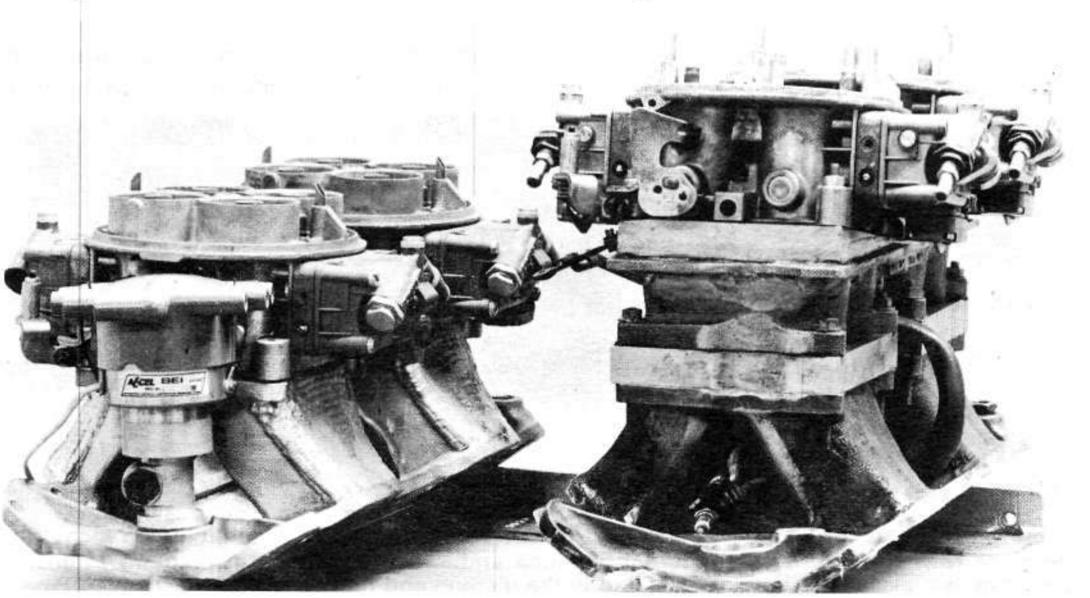


This is a very old manifold and was one of our first efforts with isolated valley baffles. Similar construction techniques are now used on all racing manifolds.

one else experienced. As the engine speed got up around 8000, the signal back to the carbs was so intense that they began flowing backwards and blowing fuel out the tops. This "standoff" problem contributed to a high-speed lean out which became increasingly severe as the engine speeds went higher. We had made progress toward finding a solution, but finally got sidetracked to more lucrative research. Basically, we found improvements by using a short manifold with a very long, 6.250-inch rod, in the 3.25-inch stroke engines. This setup worked quite well with peak power very comparable to the plenum/660 combination. It still needed some tuning work to get the top end in shape. At that time we were working with IR carbs with high-speed bleed wells to add air to the top-end fuel. Upon retesting we are going back with non-bleed well metering to give solid-fuel flow at WOT. One of the main ways to correct the tendency for carbs to go rich at the top end is to increase the air-bleed correction. More air bleed lets the fuel level drop lower in the emulsion well as the air velocity goes up. The fuel delivery rate drops off accordingly. We believe eliminating or at least cutting down the air bleed would be the best way to boost high-speed fuel delivery and reduce the lean-out problem.

The major problem with the old program was getting the carbs as low on the engine as necessary. The ignition gets in the way when the carbs are lowered. This will no longer be a limitation because front-drive ignition mechanisms can be used to put the distributor out where it won't interfere. We know from the previous work that the engine wanted extremely short IR runners so we can at least solve this one problem without much difficulty.

We also know the backflow problem can be reduced immensely by cutting down the piston action. This wouldn't be too hard to accomplish. It looks as if a very long rod, short-stroke engine would be the fat answer. Considering the currently available pieces, something like a 322 cubic engine looks good. With a 3.17-inch stroke, a 4.020-inch bore and a 6.4-inch rod, the signal back to the carb would be significantly reduced, helping the carbs to hold the fuel. This would naturally affect the required runner volume. With the ignition out of the way, it would then be possible to raise or lower the runner length as needed to tune in the bottom end and get some



With our current rod ratio and camshaft configurations we feel a low profile IR manifold may be the only viable alternative for future induction improvement. The engine rod ratio would have to be high, the manifold runners short and the carb venturi/throttle would have to be extremely large. Using a front-drive ignition would solve clearance problems.

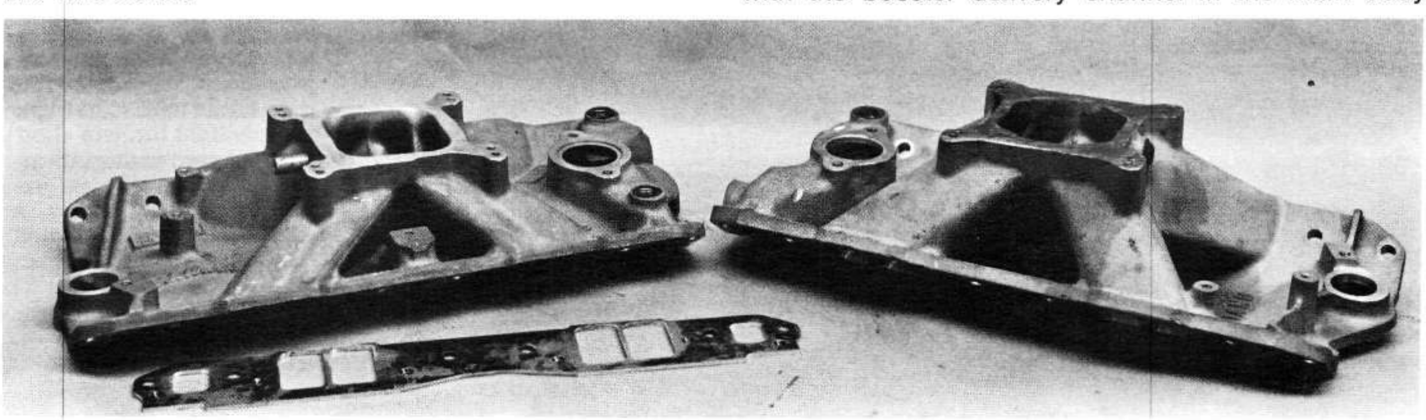
driveability back into the system, without kicking the fuel out of the carbs at high speeds. Of course, this is all speculation.

The only real problem with any IR setup is going to be the upper limit of the cylinder displacement imposed by the carburetor size. At WOT the only volume from which the cylinder can draw fuel is the runner length from the valve up to the carb entry. As the cylinder draws in fuel, the absolute cross-sectional area limitation of the runner or carburetor will become a restriction. After preliminary calculations of the areas inside a manifold and carburetor, such as our old experimental IR, it becomes clear that the throttle and venturi of the carb aren't big enough to feed anything other than a very small chamber. For instance, we feel that a standard 6214 Dominator rated at 1150-cfm with 1-13/16-inch venturis and 2-inch throttles will just about be able to fill a cylinder displacing approximately 35 cubic inches. This assumes there is absolutely no plenum area under the carb. Multiplying this figure by eight cylinders, we can see the total engine displacement would be limited to about 280 inches. If larger throttle/venturi IR dominators were available, the engine size could be pushed up some. With 2.125-inch throttles it might be possible to get the total displacement up to 302 inches. With 2.250-inch throttles the picture would be very good for a 320-inch Pro Stock engine. Since this appears to be an unlikely possibility, some sort of mini-plenum will have to be used to provide sufficient breathing, dimming the prospect of an effective true individual-runner induction in the foreseeable future. Just as an aside observation, we have also noted that a workable IR might have some other benefits. With an extremely short manifold/carb height, the engine could be raised in the chassis, at least to the limits imposed by the existing hood scoop requirements, and a bigger, more efficient oil pan could be put around the crank.

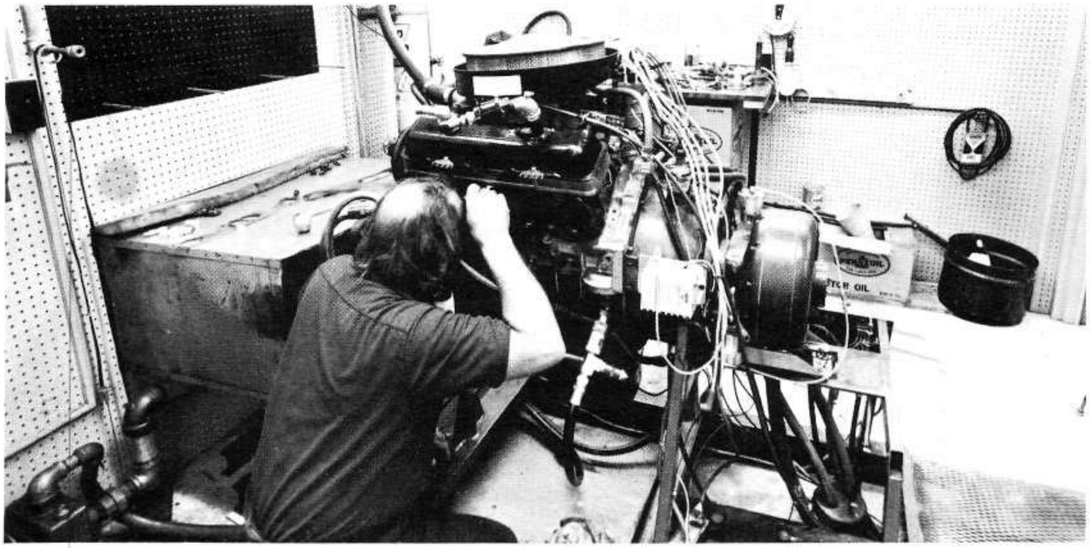
Though we don't use them on our 330-inchers we have tested the Dominators on 354 engines with Weiand's Holeshot manifold and they do quite well. On the dyno they pump out better numbers than the 660's, but they don't drive down the strip as well as the little carbs.

We have used the 6464 rated at 1050-cfm in near-stock form with "qualified" success. Some pump cam work is required to get the proper discharge timing. We double-drill the discharge nozzles if they need more pump area. The exact size varies according to the engine requirements, but with 50cc pumps, two holes, each 0.027- to 0.033-inch, will work fine. Along with this, we cut a small notch in the top of the venturis so the pump shot won't hit them. This gets the fuel into the plenum more quickly. To help the hood-scoop clearance problem, we cut the horns off. We use the stock floats as delivered in the cathedral bowls. Jetting may vary some depending upon the case in point. When the carbs are mounted sideways, there may be some very slight variations because of inertia effects upon the metering channels as the car accelerates forward. There will also be some acceleration inertia effect upon the fuel as it moves across the plenum or down the runner legs. The larger the plenum, the more the fuel has a tendency to flow toward the rear of the manifold. To counteract any of these problems, some stagger jetting may be required. This usually calls for 1 to 3 numbers richer toward the front, although the overall average Holley jet size should still be about 83 to 86 for a 354 engine.

On the dynamometer we have successfully operated 360-inch and larger engines with the Holley 6214 "IR" Dominator rated at 1150-cfm. These carbs are fitted with special boost venturis to dampen the induction-pulsing associated with IR inductions. They can be used on mini-plenum manifolds, but they will have to be reworked a great deal. The 6214's use the Holley metering blocks with small idle-feed tubes inserted into the main wells. The tubes reduce the overall diameter of the main well and restrict the amount of fuel the carb flows at WOT (these same blocks are used in the 6464 but they don't cause problems there). As the jet size is enlarged above a number 92 jet, the normal progression of fuel enrichment falls off. For each step above this point, the resulting delivery increase is relatively diminished because of the restriction created by the idle-feed tubes. To counteract this restriction, the metering block main well and the angle channel that lines up with the booster delivery channel in the main body



All of our current data on single four-barrel inductions is the result of our recent past involvement with Nascar Grand National racing. During this program we evaluated both Weiand and Edelbrock single-plane manifolds on many different 354-inch endurance engines. Both require extensive working in the plenum and runner entries .



Generally speaking, we feel the Edelbrock Tarantula core cast in a Scorpion isolated-runner body works well on an engine with 5.85- to 6.00-inch rods on a 3.48-inch stroke. The Weiand X-pert seems to do very well on a 354 with 5.70-inch rods. Either manifold must be fitted with a carb spacer and the entry radius at the top of the runners must be smoothed and reshaped for adequate flow transition from the carb throttle bore to the runner.

must be drilled larger. This is a hell of a lot of work and should probably be avoided unless you know what you're doing and have a great desire to run these carbs.

Rather than fiddle with this, it looks like the best bet would be to try Holley's new 4500, available as list number R-7320. We have not conclusively tested this carb although it appears to have better idle characteristics and more responsive fuel feed. They might prove to be the fat answer on a semi-plenum with short runners and a numerically large rod ratio. We tend to be conservative and stick with proven parts, but someone with the time to sort out this setup might find some power.

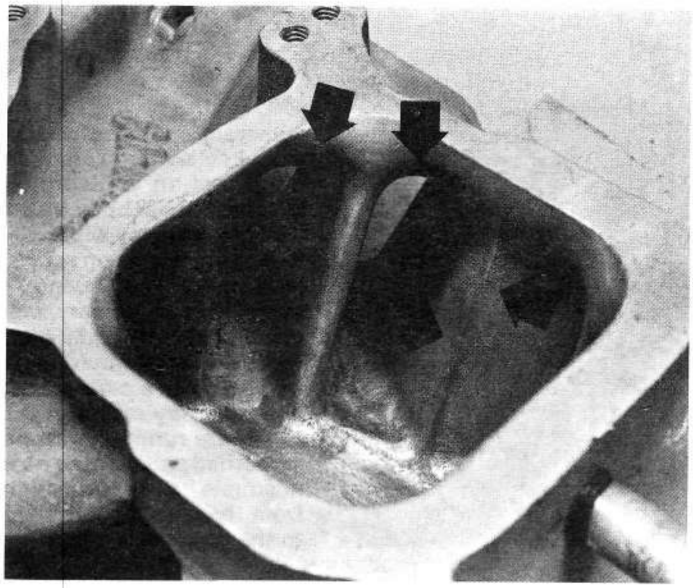
SINGLE FOUR-BARREL INDUCTIONS

Most of our recent single four-barrel work stems from the Grand National stock car development for Donny Allison and DiGard racing. This sort of information becomes dated very quickly but it should have some application to current Super Modified and similar drag engines. The parts and pieces may change, but some of our "theories" should still be valid and, with some imagination, they could be applied to nearly any class of racing where the induction is limited to a single four-barrel on a stock or stock-type manifold.

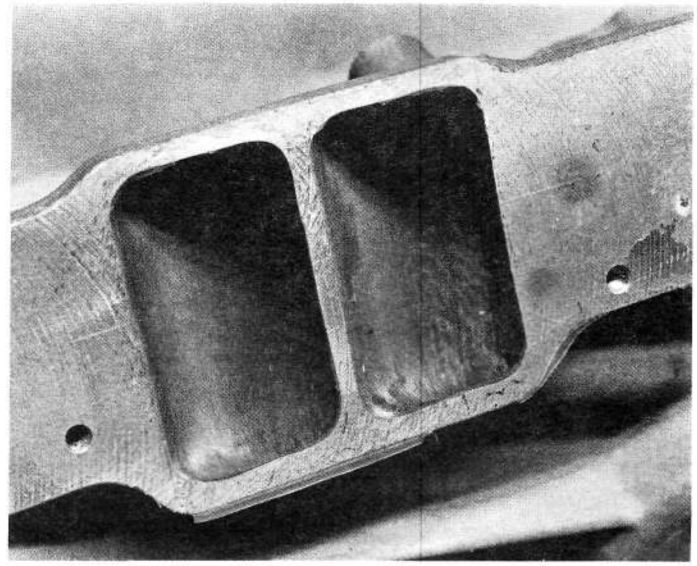
We have tested competitive engines which have been fitted with either Edelbrock or Weiand single-plane manifolds. Each manufacturer offers several variations of the same basic idea, and if the specific model is selected to properly match with the engine displacement and rpm operating range, the results should also be adequate in a near-stock engine. By this we mean that the performance gain should be good, provided the correct carburetor is selected. However, it is unrealistic to expect horrendous horsepower gains from a simple manifold swap. There's a lot more to it than that. Induction efficiency is always important, and the best engine in the world will fall flat if the manifold and carburetor are not very carefully examined, tested, tuned and maintained. It is, therefore, nearly impossible to simply bolt a new manifold onto an "unmatched" combination and win competitive class racing. They provide an excellent place to start, but getting the combination "right" may require additional grinding or welding or other specialized work to get the engine breathing compatible with the rod ratio and camshaft. Also, we can't forget the head ports; they are an integral part of the induction "system" and each part of the system, the carb, the manifold and the head ports, will affect the manner in which the other components contribute to the overall performance.

At this point it's likely that better performance results will be obtained from one of the aftermarket single-plane manifolds. We feel in a single-plane manifold, it is important that each runner/port combination flow as nearly like the others as possible. A tunnel-ram manifold is not very sensitive to the effect one runner may have on the others. However, the single-plane manifold has relatively small runners and the plenum is small. This allows the runners to rob back and forth and/or affect the balance between the individual cylinders. We have found that the best way to counter this problem is to flow each cylinder runner/port combination on the bench to get the base numbers as equal as possible. This usually gives the best overall performance. There are so many details to talk about, we hardly know where to begin.

Both the Weiand and Edelbrock competition manifolds need some sort of spacer between the top of the manifold and the carburetor. We haven't had any experience with the Holley manifolds so we can't say what they would be like. However, the same problem should be investigated when testing them. The problem being, there is a very short turn radius coming out of the bottom of the carb throttles into the entries of the manifold runners. Getting the carb up higher cuts down the velocity of the intake, letting it turn the corner more easily. The Scorpion/Tarantula needs at least a ½-inch rise and possibly a full one inch. The Weiand must have at least a one-inch plate



Most of our good Nascar manifolds have been sold but this is one of our well-worked Tarantulas. A lot of metal has been removed at the entry roof. The divider wall is radiused as much as possible without hurting flow and the divider-side radius is raised to give a flat approach. The inboard-runner common wall should be radiused more than shown here.



The runners are wider and taller to gain more volume. We do not go along with the reversion step idea and cut all this away in an effort to enlarge the manifold. The runner opening must match exactly to the head port, without any gasket overhang. We feel the Tarantula runner meets the head port at too severe an angle so we always use a stuffing plate to continue the inside curve from the runner to the port.

and maybe more. The corners of the Weiand must also be filled with weld and hand-blended carefully to get a smooth entry radius. The Weiand is a fairly tall manifold to begin with and adding this material makes the total combination very tall, so much so that hood clearance becomes a problem in some cars.

The Edelbrock Tarantula or the Scorpion body with the Tarantula core is about the best racing piece available. We think the Tarantula works a little better than the Scorpion; the Scorpion runners have more "twist" to them as they approach the flange and this may slow the flow down a little. They will be adequate on a small displacement drag engine or a long rod (5.85- to 6.00-inch) Grand National 350. The major work required is to get the runners wider and taller to get the volume up where it belongs. It's just not possible to get the runners of a Tarantula too big, at least for use on a 354-inch GN engine with an unlimited camshaft. We definitely do not agree with the Edelbrock reversion step or port mismatch theory. We eliminate these cast-in steps and make every effort to blend the manifold runner smoothly into the head port. It is especially important that an overlap does not exist on the divider-side wall of the inboard ports. This restricts the area in these ports and a mismatch here is death on the flow numbers.

The turn radius of the transition from the manifold to the head ports is not particularly good on these manifolds. To help this turn we use a ⅛-inch stuffing plate to continue the radius of the manifold turn slightly further, giving it a better blend into the port openings of the head. The plate also gets the manifold runners and the head ports aligned. We raise the manifold with the stuffing plates until the roof of the runner matches the roof of the port. The bottom and sides of the runner are ground away until they meet the entry of the port. It's pretty simple, but many people ignore this. The only matchup which may not be critical is the long or outboard wall of the end ports. This wall takes a reverse turn immediately after it enters the head to go around the push rod. The flow activity is relatively low at this point.

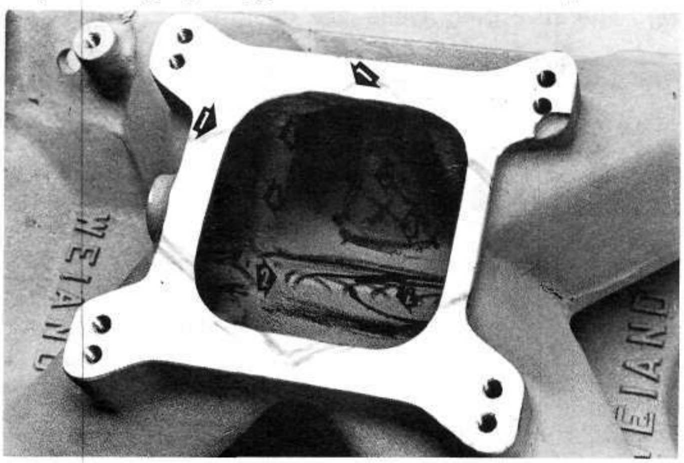
The entry to the runners will also need some additional attention. Earlier we mentioned the work required to gain a good departure from the carb pad into the roof of the runners. This is always the most critical area in manifolds of this design, but it is not the only area that may cause trouble. The lengths of the various walls and the radius of the entry at the roof and at the side walls will also have a surprising affect on the total flow capabilities of each port.

The Weiand X-pert manifold is also a good basic piece to use for a race buildup. It allows the runner volume to be increased more than most other manifolds. The tall height of the manifold may give a little better transition from the manifold runner to the head port. However, Weiand makes progressive changes as new batches of these manifolds are made, so it is difficult to make absolute recommendations. In stock form all the turning radii down where the runners meet the head are very good, and in dressed-up form it should work pretty well on a small-port head (non-292) or something like an NHRA Super Modified engine. For a Grand National engine it must be hogged-out considerably to gain port matchup, especially with a big-port 292 head.

The hardest work on this manifold is getting the plenum in shape. And this can be a formidable task. The floor must be flattened to prevent fuel from washing down into the inboard cylinders. We prefer to keep the floor completely flat or grind a small ditch with fuel dams down the middle to help get fallout fuel to the end runners. The end runner usually runs lean with this manifold and this fix will definitely help distribution. On the sample castings we currently have in the shop, the runner-pair divider walls and the runner/plenum sidewall dividing the forward runner pair from the rear runner pair require some major surgery. The sidewalls protrude too far into the plenum, making the inboard walls of the middle runners too long. These walls become the long walls down in the head ports, and making them longer by extending the manifold runner walls is not the thing to do. In the area where these port walls meet to form the sidewall of the plenum, we chop the whole section away from the original casting. Then we weld in a new plate to enclose the plenum, enlarge the plenum volume, and shorten these walls. The divider wall is also too long up near the roof of the port entry radius, and it sticks way out into the plenum. This gives the inside radius from the carb pad into the runners a very strange shape. We weld up and radius the corners to get an acceptable shape here so the flow has a wide, smooth approach to the roof of the runners (see photos).

With all of this preliminary work and several dozen hours on the flow bench, we can get this manifold to flow as well as our drag race tunnel rams. This is, of course, without a carburetor on the manifold, and really means only that one of the runners of the single-place manifold will flow as well as one of the runners of a tunnel ram.

Unfortunately, all of our well-worked out single four-barrel manifolds were purchased by the DiGard racing organization and we don't have any examples to show the full state of development. It would, however, be an understatement to say they were very thoroughly flogged. In this class of competition a



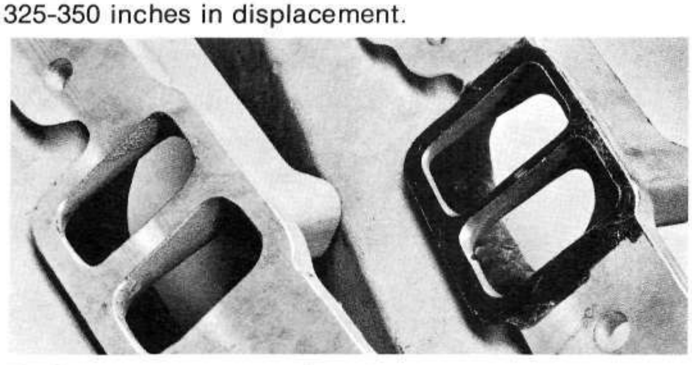
The Weiand X-pert calls for extensive entry work. A 1-inch riser is needed to get the carb high enough for a smooth transition. This sample is not modified but the entry area (arrow 1) must be welded and squared with the runner roof. A large-radius entry must be built to allow the flow a smooth approach. The floor should be flattened and ditched (arrow 2) to channel fallout fuel toward the end runners. The inboard runner common wall (arrow 3) should be cut away and pulled back to shorten them. This is the best model X-pert to use as it has the divider wall that extends well into the plenum (arrow 4).

full-time flow bench and dyno study is the only way to get competitive.

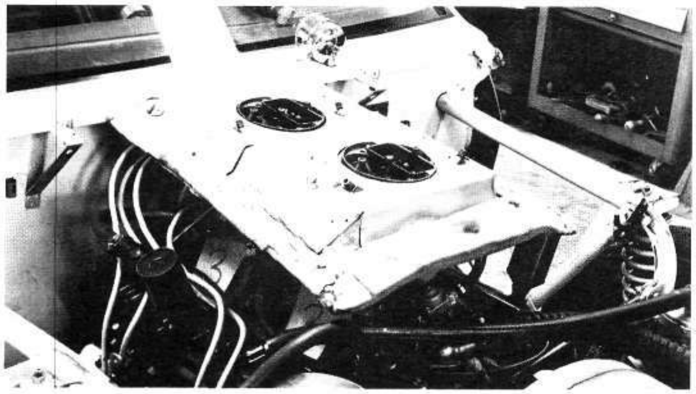
Any of the Holley double-pumpers will work well on either the Weiand or the Edelbrock manifolds. For unlimited classes the 850-cfm model is hard to beat on a 327-inch or larger engine. With some proper attention to camshaft selection, manifold volume, and rod ratio, it would also work on smaller displacement. If the class requirements call for a smaller carb, some problem may arise with fuel fallout on the floor of the plenum. This problem is related to flow velocity out of the bottom of the carb. With small carb bores the fuel is moving so fast as it exits the carb at high rpm that it will not make the turn into the runners properly. Consequently, it accumulates on the floor of the manifold and begins to affect distribution. The only way to prevent this is with fuel dams or ditches to channel the fallout into the lean cylinders. In the case of an average single-plane manifold, these will be the end cylinders — one, two, eight and seven. This problem was very troublesome on the big block Nascar engines when they were required to run with restrictor plates under the carburetor. It may still exist in some NHRA Super Stock classes where a small carburetor is used with unlimited cams, headers, and a single-plane manifold.

The double-pumpers are very easy to tune. You just follow what the engine tells you it wants. We have never had to fool with the idle circuits or anything else. We cut the choke horn off to get flow into the front bores balanced with the flow into the rear bores. The last time we raced one we just got the pump cams to work with the staging technique, set the floats, and jetted it out to the engine requirements.

Below 4000 rpm a good two-plane manifold like the Edelbrock C3-B and a Holley 715-cfm carb will run right along with the best of the single-plane setups, although cylinder-to-cylinder distribution will still be very ragged. We wouldn't want to have to live off the difference between this and an Edelbrock Torker, but, back when we were running this sort of induction, the C3-B/715-cfm combination seemed to be an ideal match for smallblocks of about 325-350 inches in displacement.



The X-pert runners approach the head port at a good angle and stuffing plates are not needed for flow control. The runners have been ground extensively for increased volume. However, for an engine smaller than 350 inches or an engine with longer rods, the runners wouldn't have to be enlarged as radically.



Hood scoop design is important for correct mixture distribution. The hood must provide a minimum of 2 inches clearance over the carb and the walls diverge to slow the incoming air velocity. The pan seals the bottom of the scoop to the carbs. Note the direction of the bowl vents to prevent pressurizing the bowls.

If you don't have time to fiddle with the manifold, you should still be aware that some carb work will be necessary when switching from a dual-plane manifold to a single-plane. In nearly all cases the carb jets will have to be enriched. The air activity inside a single-plane manifold is always somewhat less pronounced than in a two-plane. The signal back to the carb is much weaker. To compensate for the calmer air, the carb must be "fatter" to get the correct fuel delivery.

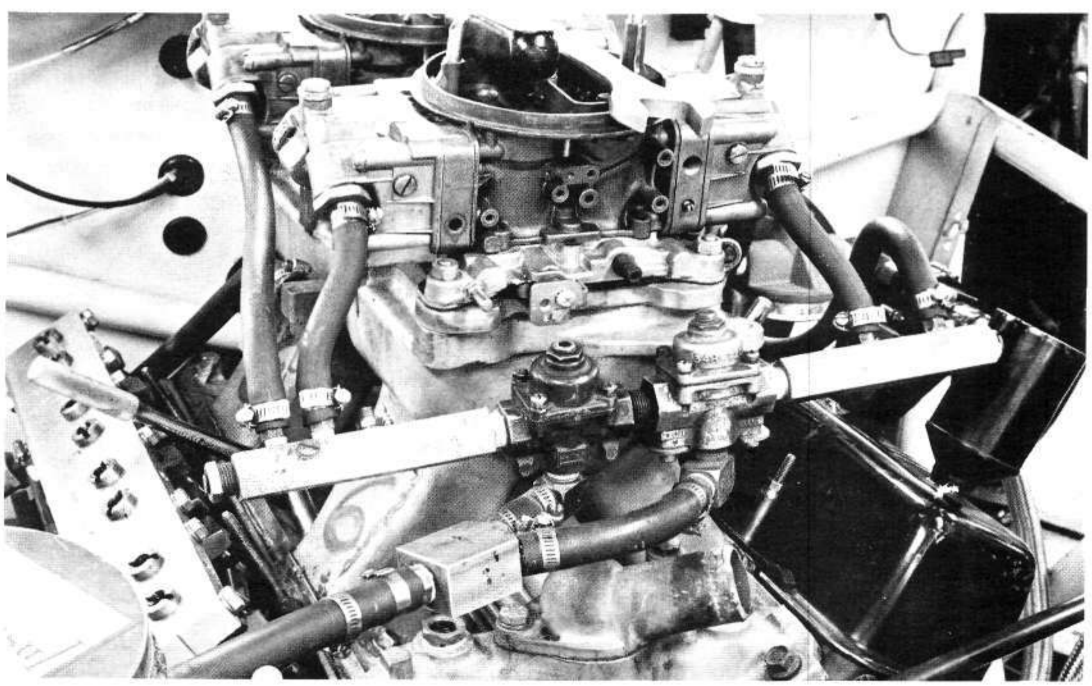
HOOD SCOOPS

We are currently using the snorkel-type hood scoop. Ours have sidewalls that diverge toward the rear of the scoop. It is difficult to find a ready-made scoop built like this, but ours has about a 7° diversion, front to rear. This allows the air to slow down after it enters the opening, so most of the pressure recovery occurs inside the hood scoop plenum. We seal the floor of the scoop to the carb openings with a simple aluminum sheet. There is still some airspeed over the front carb that will require compensation, but there is virtually no velocity by the time it reaches the rear carb. Engine height is often a problem with tall plenums, and hood-scoop limitations may restrict the overall manifold/plenum height, especially when some thought is given to the required space above the carbs for smooth air entry. If there are no scoop limitations, a tall stack with a vertical slot, as used in the Formula 5000 cars, is about the best bet. This directs the air down over the carbs in a vertical pattern, gaining the most pressure recovery and least amount of stagnation loss in the plenum. And this design does not blow air across the carb entries.

In any case, the area of the opening should be as small as practicable. It is only necessary to have enough area for acceptable flow with the car sitting still and the engine at full power. In a Pro Stock car this area will be about 2.5 times the area needed to feed the engine at 150-plus mph. A little bellmouth flare around the opening will help get the air into the snorkel. Not much is needed, and the radius should be less than ¼ the diameter of the "hole." Flow bench work has shown that this simple radius bellmouth can effectively double the flow capacity of a tube (this is true at least in still air).

The best way to determine the optimum size of the opening is to observe for signs of reverse flow out of the opening. This is the first thing that happens if the scoop is too large or the bottom of the scoop is too near the boundary layer of the hood. The velocity gradient across the opening will be such that if the scoop is taking in more air than the engine can accept, the stacked-up air will be pushed backwards along the floor and out the opening. We do not have any pressure relief holes in the rear of the scoops. This just aggravates the velocity across the carb opening. The diverging walls are designed to relieve in-

Two pressure regulators in parallel must be used for adequate fuel delivery. The lines to the regulators are number 8 and to the carb number 6. The regulators give 5.5 lb. line-out pressure.



ternal velocities, and you definitely want as little velocity across the carbs as possible to eliminate fuel siphon-off.

PUMPS & PRESSURE REGULATORS

In certain classes of racing, Nascar being the most notable, electric pumps are not allowed and some sort of stock engine-driven mechanical pump must be employed. In this instance, most of the teams are using the Carter racing pump, PL-4594S. In the past I have used the high-pressure pumps supplied with the early fuel-injection Corvette systems and some crossbreeds of W-engine high-pressure bodies and Mark pump arms. It is impractical to use many of these special pumps because they are not now available, so the Carter may be the best bet. Chevrolet also offers a heavy-duty pump for the smallblock as part 6415325. We believe this is a two-piece pump that was used on the late high-performance 327's and is still available. Either of these should be adequate, although the only way to be absolutely certain is to install a fuel-pressure gauge *and read it religiously.* Failure to do so has caused many disappointments and some foolish-sounding excuses.

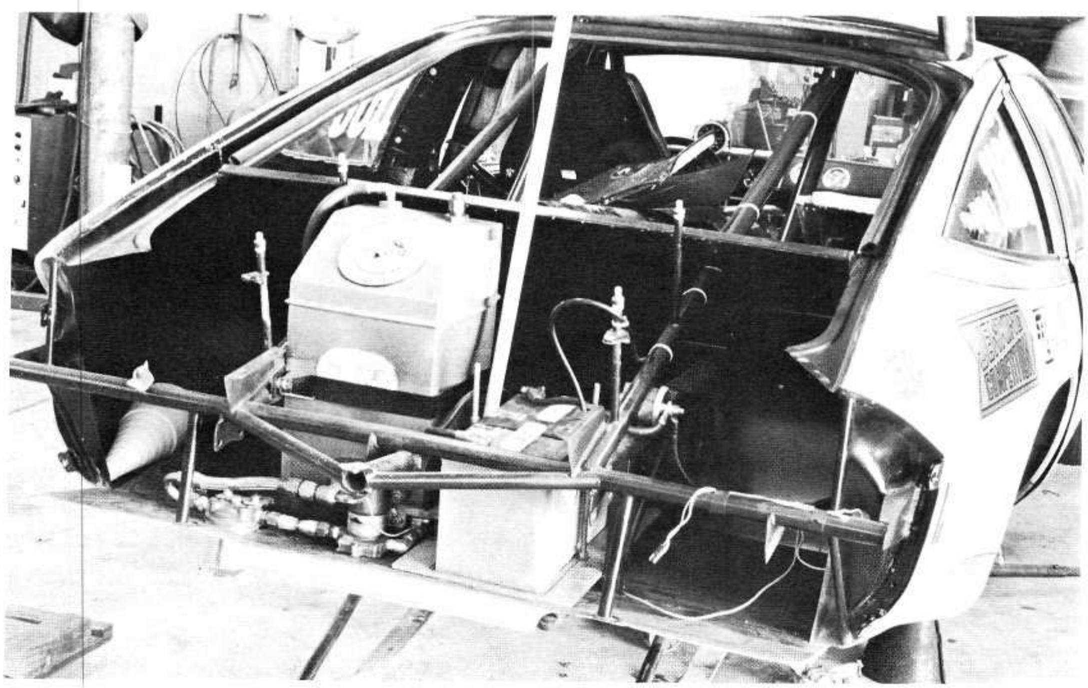
In any case, we recommend on a competition car that the pump be taken apart and the fuel valve be inspected and restaked very firmly in position. We have seen the valves come loose, at least on some Nascar applications, and cause pump failure. Such a circumstance plagued one of our competition engines and, unfortunately, it was not discovered until "too late." Such things can try the patience of even the most die-hard racer.

The design of drag racing fuel-supply systems has been well-covered in other standard references. We just make certain the supply pressure to the bowl entry furthest from the pump and regulator is a minimum of 4 to 4.5 lb, under full acceleration. The electric pump entry must be lower than the tank for gravity feed to the suction side, and we always use a high capacity filter between the tank and the pump to prevent dirt from entering the pump. An internally regulated pump is very susceptible to dirt that may enter the pressure regulator and upset functioning. When the pump is installed we shim-adjust the line-out pressure to give a pressure greater than 18 lb.

The fuel line leading from the pump to the pressure regulators is a number eight. We use two regulators, in parallel, with each set to 5.5 lb line-out pressure. The delivery lines to the bowl inlets are number six. You must use two regulators, as flow through one is insufficient to supply two racing carbs at full-power fuel requirement, when hard acceleration cuts the high-side pressure to 4 or 5 lb.

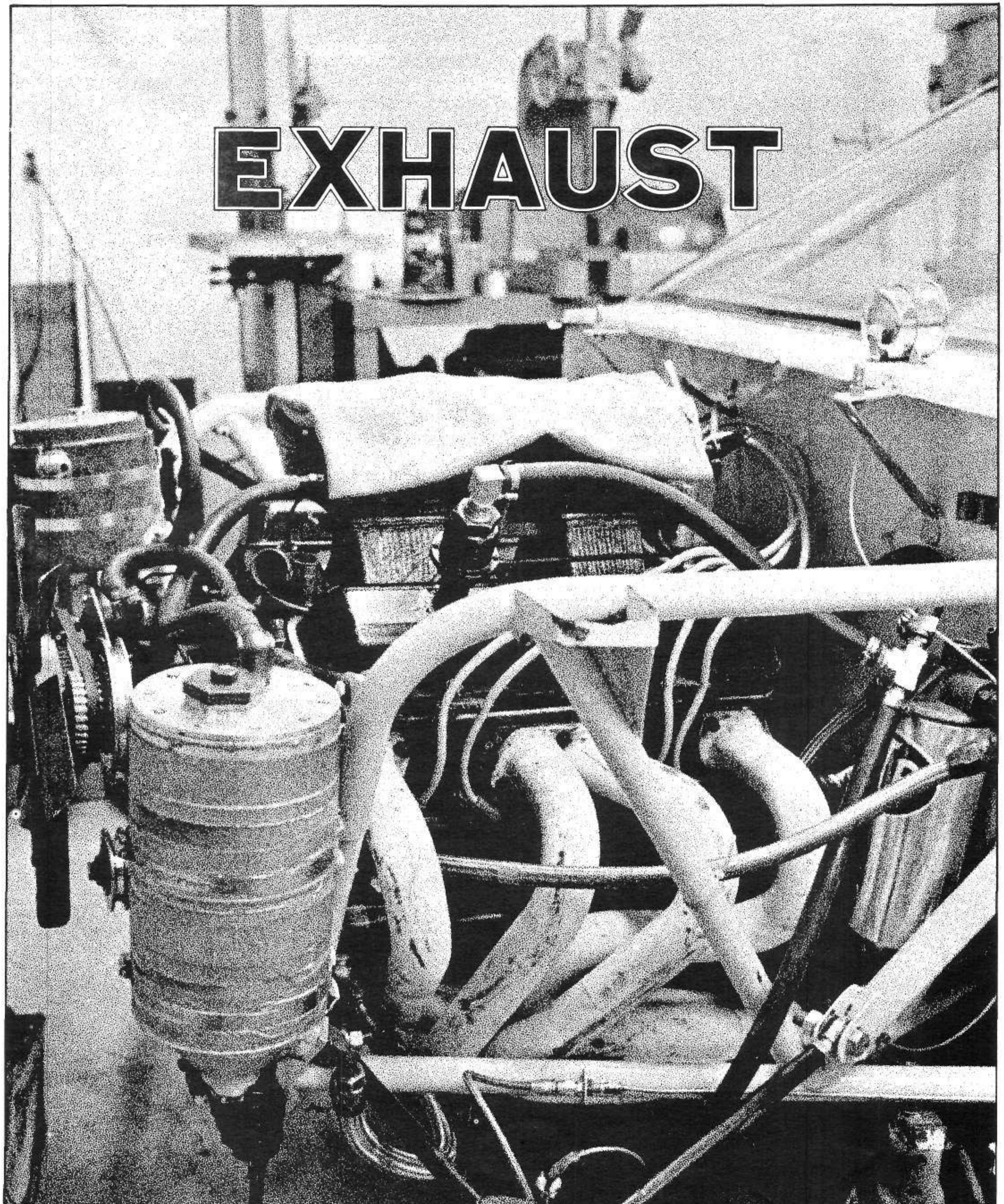
FUEL REQUIREMENT

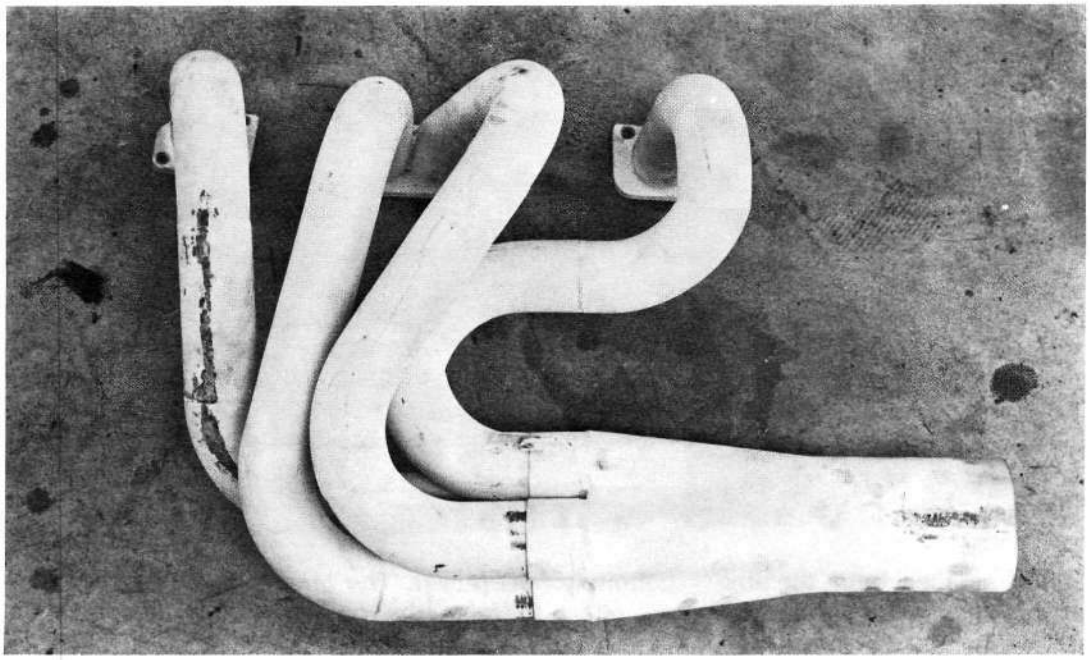
In the drag racing engines we use gasoline fuel that measures a minimum 96.5 Motor octane. The readily-available H&H racing gasoline is supposed to check at 96.5, and the straight Union racing gasoline used on the Nascar circuits is 96.5. This may vary some from outside contamination. Mixing the Union racing blend with about 10% (by volume) 100/130 octane aviation fuel cuts the weight some and raises the octane reading to about 97.5. If you fall short of the Research, you can always get it back by adding a very slight amount of an anti-knock chemical additive such as Toluene. We have also been able to successfully run the Sunoco summer blend 260 with 33% (by volume) 115/145 avgas. This cuts the weight from 0.760 to about 0.740, and gets the Motor from 93 to about 97 and the Research from 103 to 104.5. The combination reduces detonation, but the summer blend 260 is hard to find (we go to Florida to buy it), and the 115/145 avgas is also becoming difficult to obtain.



In the rear, gravity feeds fuel from the cell to an electric pump. The pump is shim-adjusted to give a delivery pressure greater than 18 lb. Before the fuel enters the pump it is filtered through a large capacity filter of the type commonly used in marine applications. This prevents dirt from entering the pump where it can cause damage.

EXHAUST





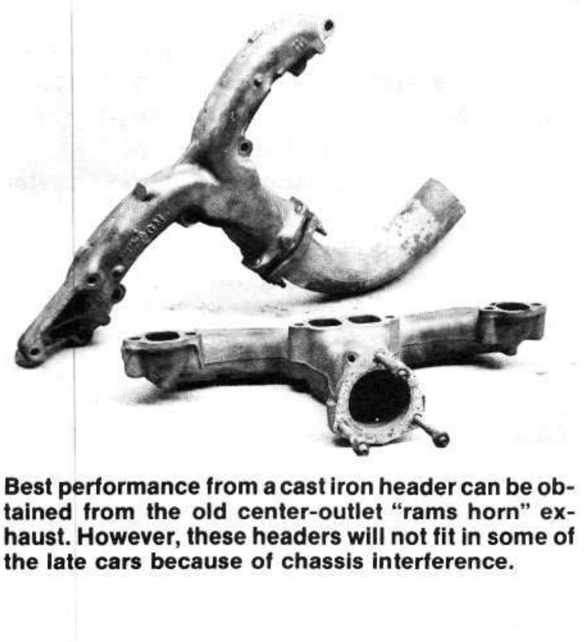
The simplest component on our racing engines is the exhaust system. We feel there could be some definite power advantages to try-Y or 180° systems but each poses construction difficulties. We still use conventional 4-into-1 headers because of their overall flexibility. For a drag racing 330 the primaries are 2⅛ inches in diameter by 28-29 inches long. Larger displacement drag race engines will need something 2½ inches by about 30 inches. The collectors should be 4 inches in diameter, cut to whatever length the engine rpm range needs.

DRAG RACE HEADERS

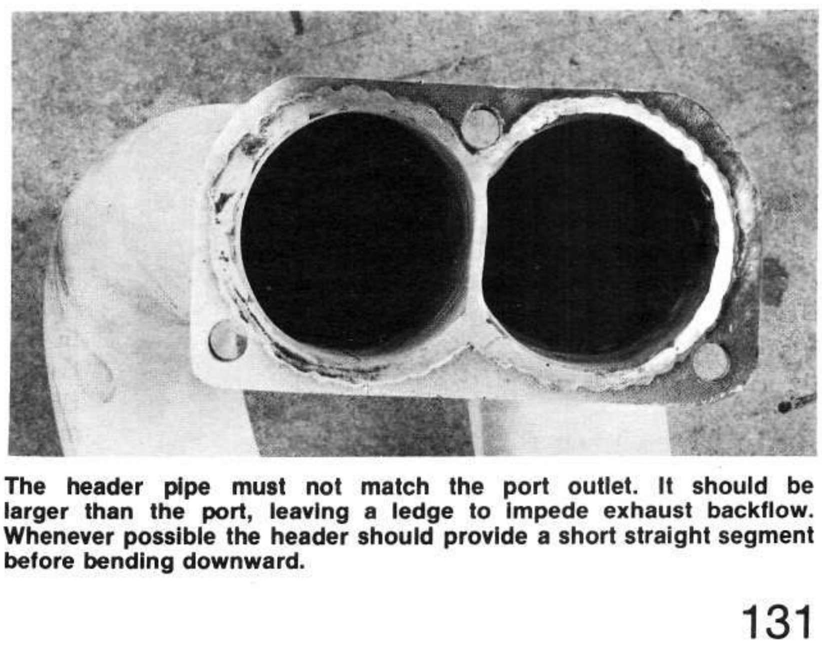
Our drag racing header systems are not very unique. They are "tuned" for peak torque at 7000 rpm. They have always been of the conventional 4-into-1 design. On the drag racing 330's the primaries are 2⅛ inches in diameter and about 28 to 29 inches long. Engines in the range of 350 inches still like the 2⅛-inch pipes, but the primary length should be shortened to 26 inches. For larger match-race engines in the range of 380 to 400 cubic inches, something approximately 2¼ inches by 30 inches long seems to do the trick. Smaller engines of about 300 inches displacement will work best with 2-inch pipes, 28 inches in length. All of the drag race engines favor a 4-inch diameter collector. We have tried smaller 3½-inch collectors on experimental systems designed to improve the efficiency of the case vacuum extraction system, but this was not entirely successful. The length of the collector should be adjusted to gain the best performance from the application in question.

During past testing programs we have investigated tri-Y and 180° designs. There may be some advantages with these systems at certain engine speeds and with specific engine sizes. An engine which requires a broader power band may benefit from a well-designed tri-Y header. The most difficult header to work with is the 180° design, although it can provide significant power increase, over a much wider engine range. This is an attractive possibility, but engines with 180° headers or "flat" cranks are extremely sensitive to tune, and routing the pipes inside a drag race chassis can be impractical.

We can't shed any new light on conventional header design, but there are a few details we think should be reiterated. First, there must be a sharp step between the exit of the exhaust port and the entrance to the header pipe. This is important to reduce backflow of exhaust-gas pulsing out of the header pipe and into the head ports and combustion chamber. The greatest benefits are derived when the

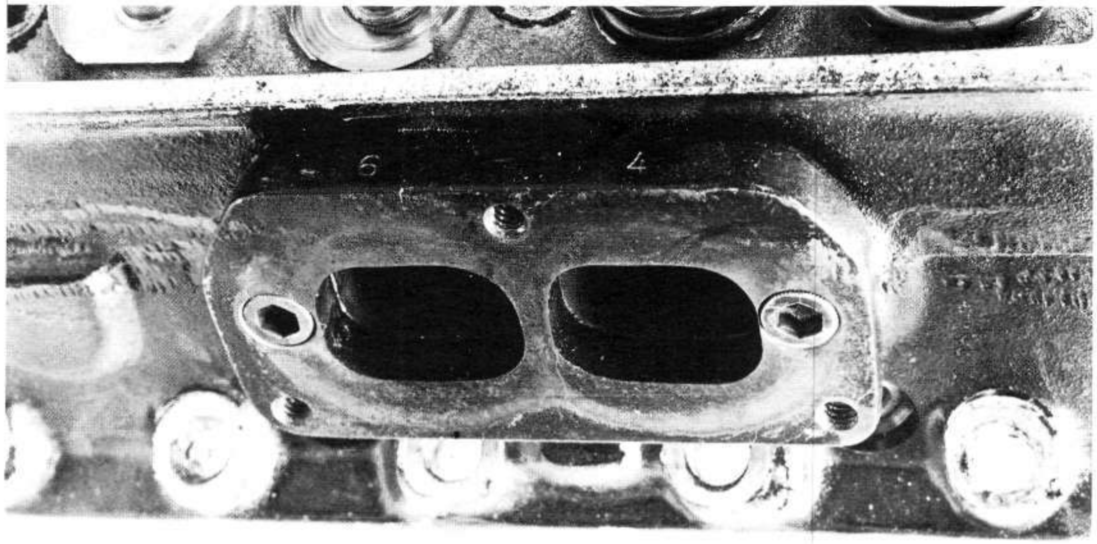


Best performance from a cast iron header can be obtained from the old center-outlet "rams horn" exhaust. However, these headers will not fit in some of the late cars because of chassis interference.



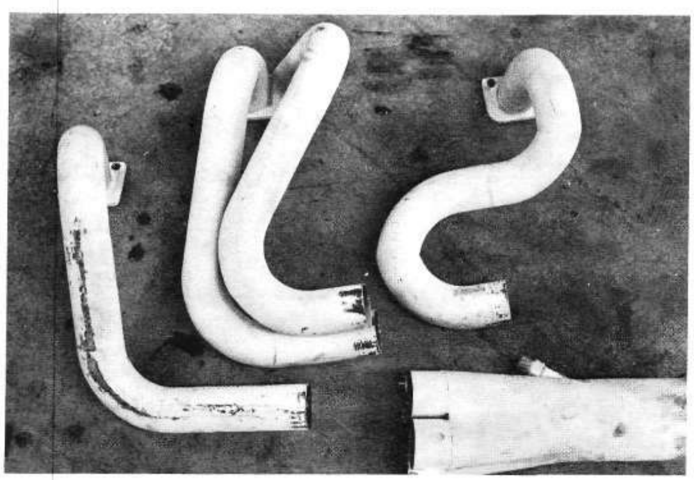
The header pipe must not match the port outlet. It should be larger than the port, leaving a ledge to impede exhaust backflow. Whenever possible the header should provide a short straight segment before bending downward.

To provide room for mounting large pipes we use special flanges and adaptors. The adaptor fastens to the head through the original bolt hole location. The large flange on the header can then be bolted to the adaptor with any desired bolt pattern and the pipes don't have to be "squashed" at the flange joint. The inside shape of the opening matches the port outlet but must *not* be shaped to form a smooth transition.



head port exit is *substantially* smaller than the inside diameter of the header pipe. It was noted in the cylinder head chapter that we do not use round exhaust ports. This is a great deal of wasted work that has virtually no advantage to exhaust-port flow. Since we do not enlarge the port very much and we maintain a "squarish" shape with the corner radii fairly small, it is easy to gain the desired step effect.

The biggest problem arises from the two center-exhaust ports. It is difficult to get large pipes centered over the port exits and the stock header mounting bolt holes on either side of the stock-port exits are too close together to get the pipes side-by-side. We prefer not to squash the pipes and force them to fit between the stock bolt holes. It is possible to fill the stock holes and redrill new holes, offset away from the port mouths. However, we like to use an adaptor plate between the head port and the header flange. The opening in the plate matches exactly to the inside shape of the exhaust port, and the backflow step is formed between the plate and the pipe openings in the header flange. There is no attempt to use the plate as a funnel to make the transition from the port to the header smooth! Such an effort will



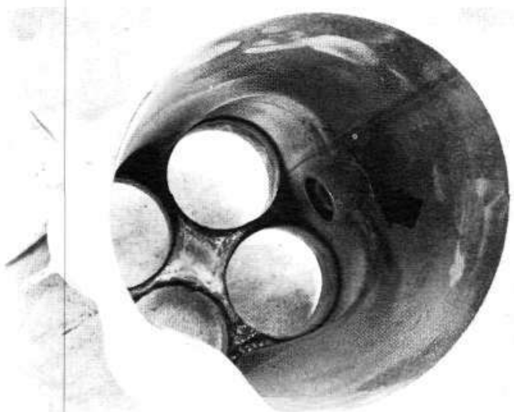
To save maintenance and repair time we must be able to quickly and easily remove the engine from the chassis. Individual tubes with a slip-on collector makes this possible.

definitely hurt power. The photos show that we make no effort to raise the port outlet in the head, but there may be some advantage to raising the header pipe centerline above the port centerline. It is best for the floors to approximately match and the greatest mismatch to occur at the roof. Note that the plate should be thick enough for the header flange bolts to gain a good thread perch but not so large as to extend the port any further than the absolute minimum. We also go to a three- or four-bolt pattern between the header flange and the adaptor plate in order to increase the holding strength.

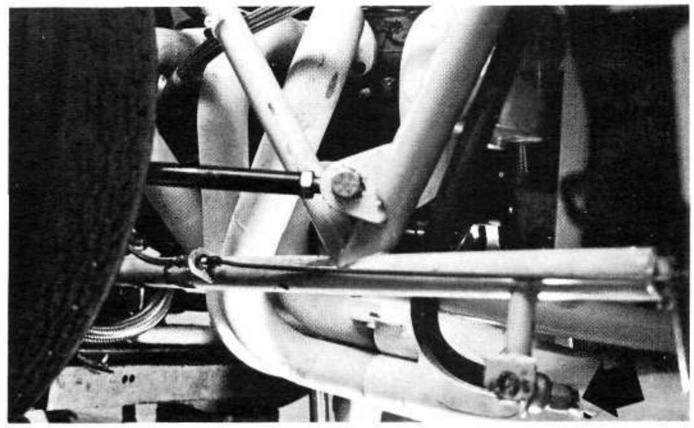
The second thing to watch is the header tube departure angle. The exhaust flow will be enhanced if the axis of the header tube is perpendicular to the plane of the header mounting flange. Any cylinder with a measurable off-angle at the flange will tend to run hot. We haven't been able to prove or disprove any disadvantages in putting a bend immediately past the pipe entry, but it certainly wouldn't hurt to leave a short straight section before the first downward bend, especially if flow is any kind of an indication. If this means there will a sacrifice of some short parallel section of the primary pipe immediately before it enters the collector, don't worry about the bend at the flange, and try to gain the parallel section on the collector end. The latter feature is more important and should be observed in any header design.

CRANKCASE VACUUM SYSTEM

Our intake/exhaust vacuum-extraction system has proven to be very successful. It has caused some misunderstanding but the functioning is simple. The system is an adaptation of two separate ideas, neither of which was developed in our shop. The manifold extractor was first conceived by Dick DiBiasse and the exhaust system was developed by Bill Million. We more or less took their basic research and combined it into the system seen on our current drag race engines. The overall concept is intended to provide a relative pressure depression in the lower case under idle, staging and full-throttle conditions. This cannot be achieved by either system separately.



The reed valve outlet for the crankcase vacuum extractor enters the collector on an angle and is cut off on a 45° angle, parallel to the moving gases. We still use a 4-inch collector diameter.



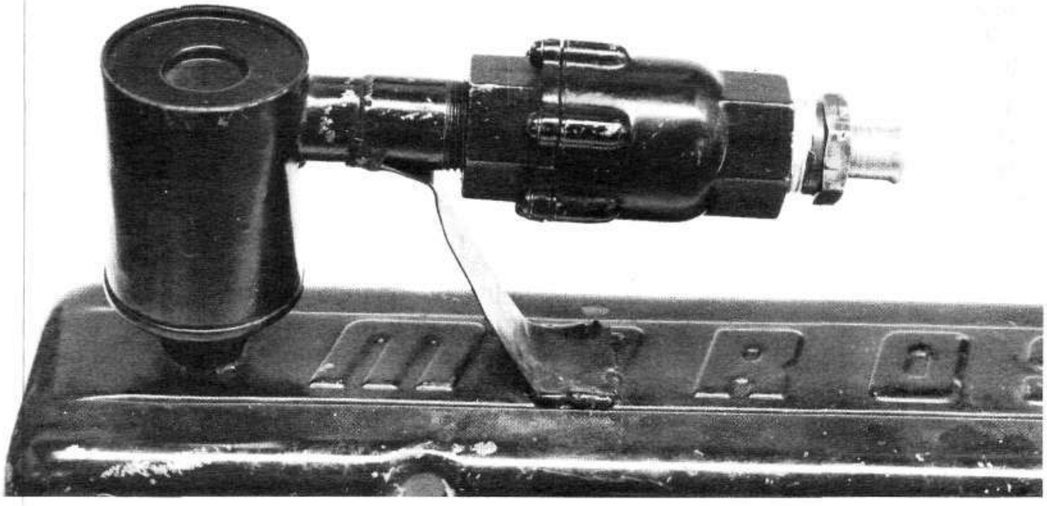
Our chassis construction is designed for maximum header clearance. The engine pan can be removed with the headers still attached to the engine. Note vacuum reed valve, partly obscured by frame member.

When they are combined in the way we have adapted them, the desired effect is obtained.

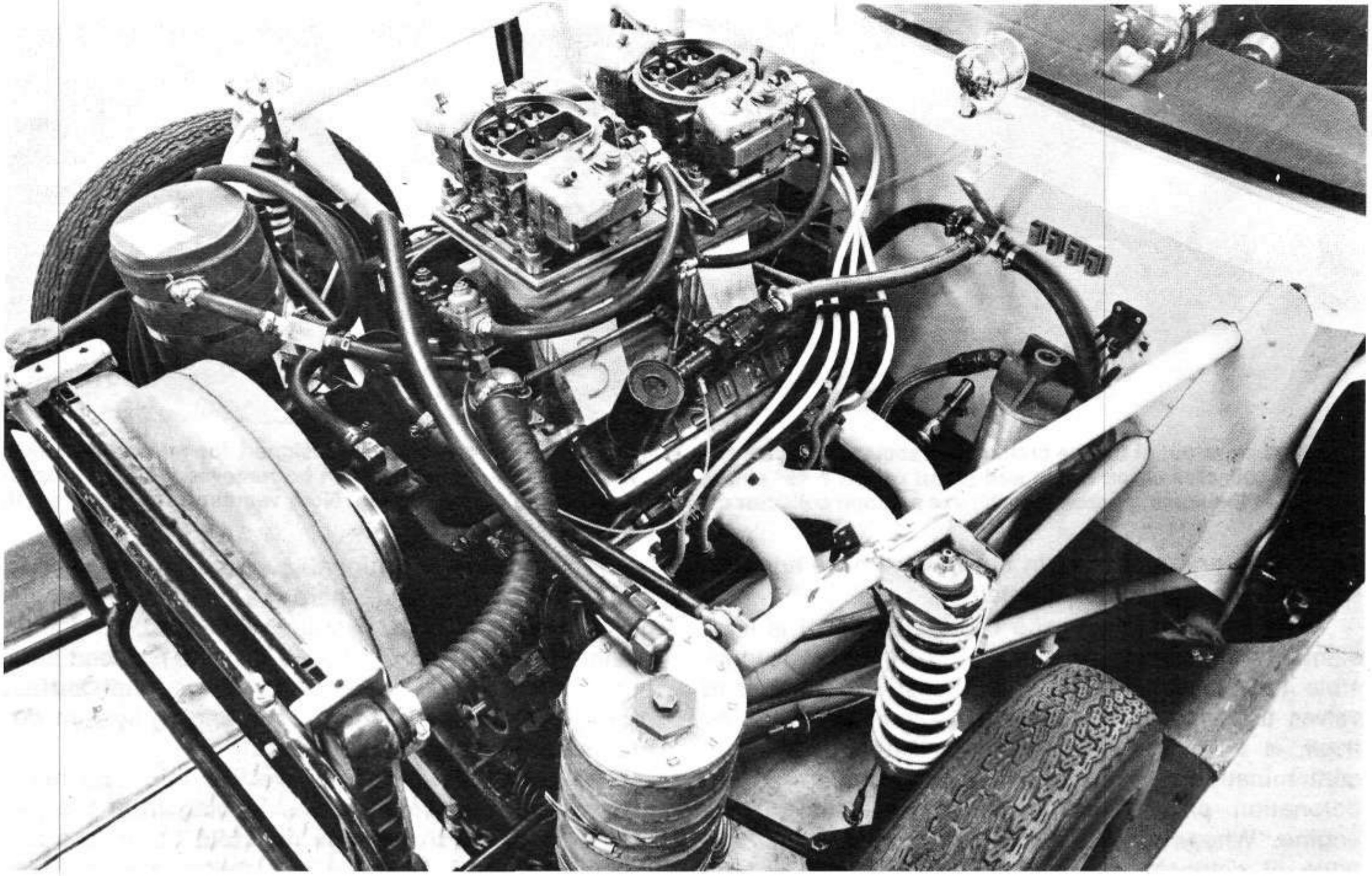
With a depression in the case, oil control is substantially improved. By this we mean that it is possible to totally eliminate oiling past the rings and valves that may contaminate the chambers. This, by itself, is an extremely important advantage. Any oil contamination at all will immediately create a severe detonation problem in a high-compression racing engine. When the engine is run right on the thin edge of detonation, as are our racing engines, the slightest leak will kill power.

There is possibly a secondary advantage of reduced oil aeration and increased oil control inside the sump. We don't have a full understanding of this phenomena. But according to dyno tests performed by Bill Million, there is some definite power advantage when the case vacuum goes above 55-60 inches (water). In this particular test the vacuum was taken from a 3½-inch balance pipe joining two extremely long 3½-inch collectors/tailpipes, such as is currently used on a Grand National superspeedway engine. According to the data, there was a readable power gain when the case vacuum exceeded 60 inches (water). We can only speculate that the reduced aeration may allow oil to flow off the internal surface more readily and lets the oil scavenging system pull oil out of the sump more efficiently.

The combined system works like this. The manifold extractor is simply a line leading from the plenum area of the tunnel-ram manifold down to a baffled oil-drainback chamber and check valve mounted to one of the rocker covers. The only trick here is to make certain the drainback does not allow any oil to enter the manifold during deceleration, the time when manifold vacuum is the highest. At light throttle the manifold system will provide up to 18 inches (mercury) of vacuum. This is sufficient to prevent oil from entering the valve guides and/or get by very weak oil rings. With the check system any air getting into the case must come by the crank seals or blow by the rings.



The vacuum "suction" is applied to the crankcase through the rocker covers. An oil separation baffle is used to keep from pulling oil out of the engine. We happen to use the separator from an early Chevy road draft vent system but other suitable separators are available. A check valve is used upstream of the separator to prevent a backfire in the headers from accidentally igniting the crankcase.



Overall view of the left shows the vacuum system as used in the number 13 car. From each collector the vacuum hoses meet at a junction block on the firewall and a single line runs to the check valve. Each of these lines is tightly sealed with a hose clamp. To check the effect of the system we have a vacuum gauge hooked into the pipe between the separator and the check valve. At wide open throttle in the speed traps the system can provide up to 4-5 inches of vacuum (mercury).

The exhaust-extractor system is intended to maintain the depression during the WOT operation conditions (when manifold vacuum is very low). The end of an extractor tube extends into each collector. The velocity of the escaping gas going past the tubes will create a slight suction and the pulsing condition of the exhaust will help build more "vacuum effect." Our system uses a reed-type valve, near the end of the extractor, to trap this vacuum in each of the tubes. A line from each extractor runs up to a common connector where they join together. Then, a single line runs from the connector to a check valve/ oil separator mounted on the other rocker cover.

At peak power on the dyno or when going through the ¼-mile traps under full throttle, this system will develop 4-5 inches (mercury) of vacuum, when a 4-inch collector is used and the extractor tubes are properly designed. The check valve acts to close off the tube when the vacuum drops below a certain point and, again, we have a situation where any air getting back into the case must come by the seals or by the rings.

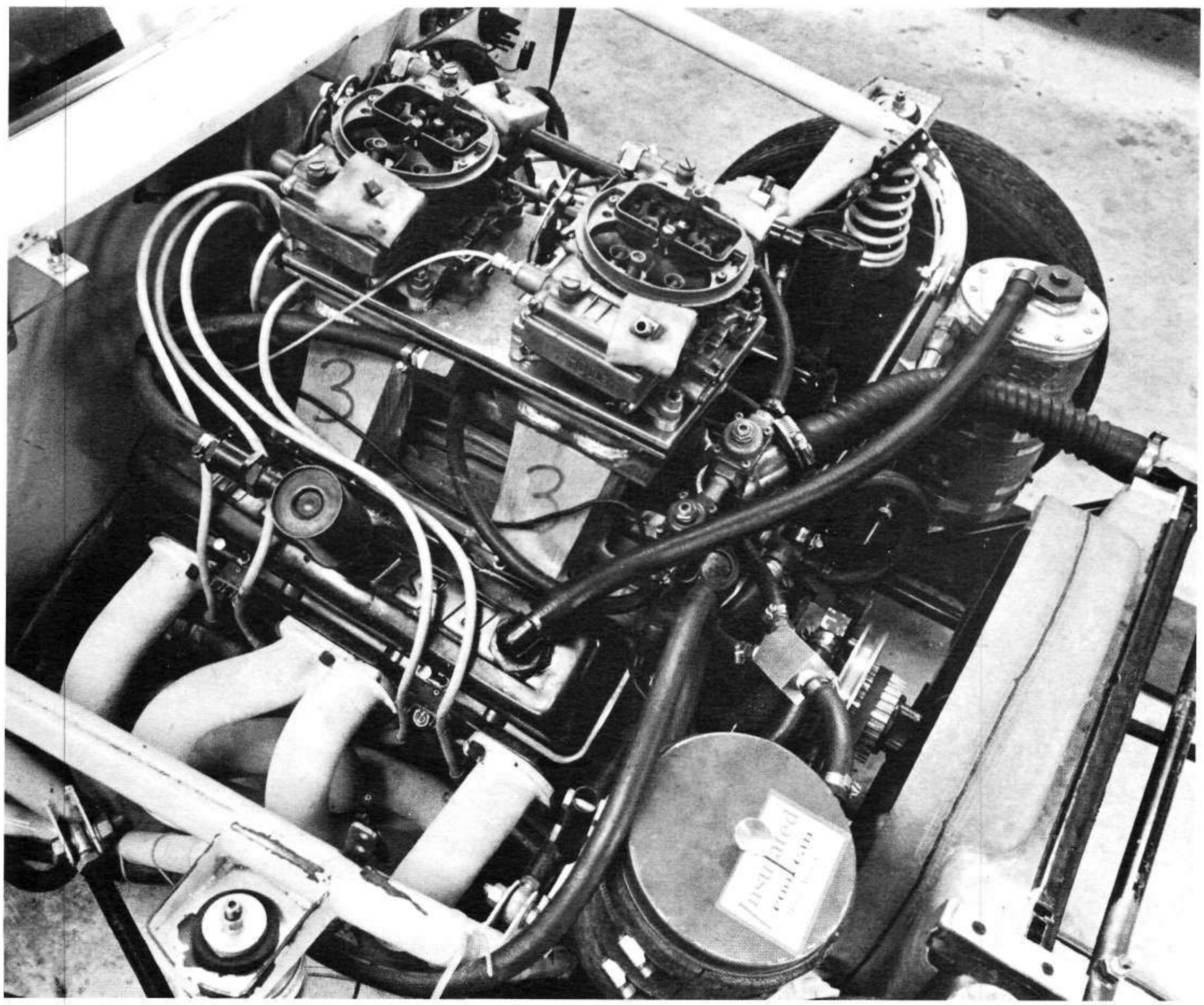
The two systems work in a complementary fashion to very effectively eliminate oil contamination to the chambers. It would be possible to create more exhaust extraction by reducing the collector size to 3½ inches (increasing the velocity past the tube), but this would cause us to lose some top-end power. Already it seems we have created a problem with lack of oil to lubricate and cool the valve stems. The current system seems to do such an effective job that the stems run dry, when stem seals are installed, and the valves have a tendency to stick in the bronze guides. To stop this, we have begun running the valves with no seals whatsoever.

Any effective increase will also depend upon developing a better crank seal. The stock seals do a good job of holding oil and air pressure inside the case, but they don't effectively keep air from entering the case in the opposite direction. We have tried running with the stock seals reversed, and this works well until a little bit of dirt gets into them. When this happens, they break down quickly, and the engine just rains oil whenever the vacuum system isn't operating.

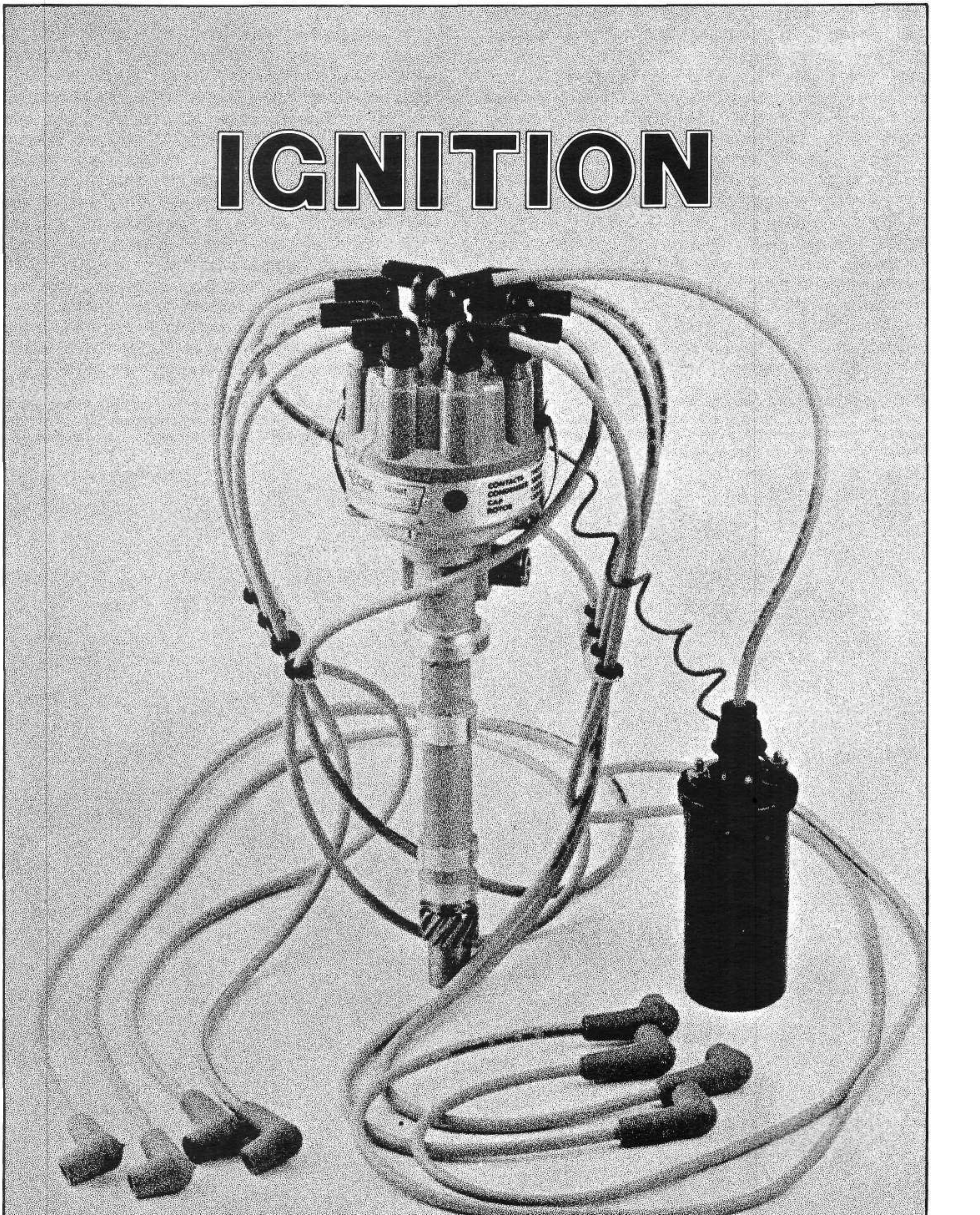
Through working with the system we have discovered some ways to gain useful information from the crankcase vacuum-pressure condition. On the dynamometer we can determine the exact condition of the compression rings. With a known-volume pan and a vacuum-pressure gauge installed, we can watch the gauge while the engine is running with the vacuum system hooked up and good crank seals in place. If the throttle is opened suddenly and the needle drops like a ton of bricks, we know the case is probably being pressurized past the rings (blow-by) rather than sealing tight. Otherwise the exhaust system will hold the depression pretty well and the dropoff is gradual until the exhaust velocity builds up and the gauge reading stabilizes. We can then double-check by shutting the engine down and watching the reaction of the gauge needle. The rate the needle drops will indicate the normal seal leakage. A comparison of the two rates tells us if the rings are sealing up well. If it drops faster when you hit the throttle than when you normally shut the engine down, the case is being pressurized by blow-by. When things are really working right (no blow-by) in an engine with intake/exhaust extraction, we can balance the engine at WOT against a load to get the engine speed on the peak torque point and just watch the gauge. It will drop to the crank-seal leakage point, about 5 inches (mercury), and just lay there as solid as a rock.

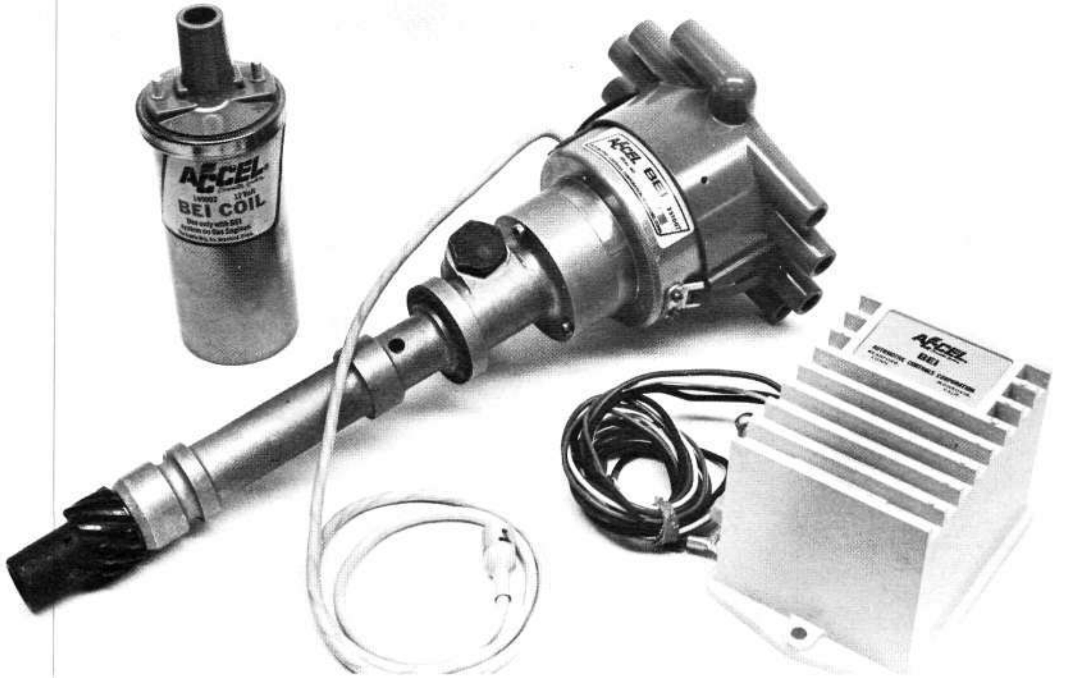
The system has allowed us to make other minor changes in the engine. We haven't been able to prove there's power available from very high vacuum depressions, but we do know that the current levels will allow us to reduce the oil ring tension, and this does help the power picture. Right now the tension is down to almost nothing in the good drag racing engines using the intake/exhaust extraction combination outlined here. Of course, the manifold line must be carefully baffled for oil drainback, and the check valve cannot allow any oil to be sucked up into the manifold at high vacuum. So far, we have been successful with our system and we find absolutely no sign of oiling on the backside of the valves. Since it appears the system will allow us to run without any valve seals, we can run the valve guides 0.200-inch longer and this should help support the valves.

On the right we see the second part of the vacuum system, the manifold extractor. At idle there is not very much vacuum created in the headers but there is a high vacuum in the intake manifold. We can use this to provide the depression in the case. A separator is used in conjunction with a check valve, preventing a manifold backfire from grenading the pan. At idle and light throttle the manifold can provide up to 18 inches vacuum (mercury).



IGNITION



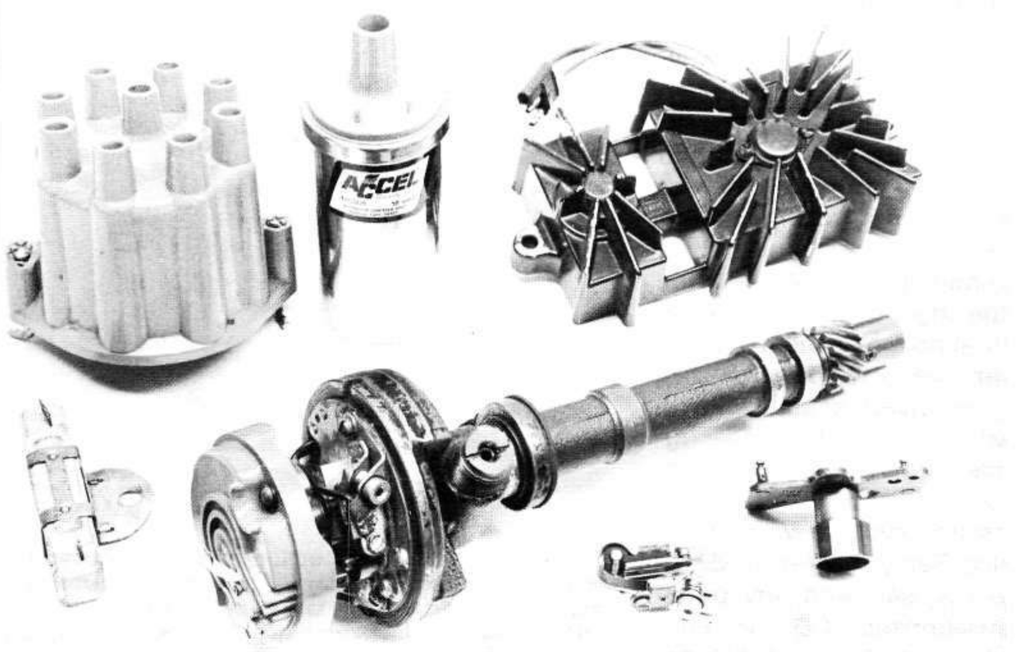


All current Jenkins Competition smallblocks are equipped with Accel BEI breakerless electronic ignition systems. This ignition is fully adequate for a 10,000 rpm Pro Stock engine. At times we have modifed the basic pieces to achieve a special effect, such as for a high-load retard, but once the advance mechanism is set up for the required ignition mechanical advance it does a very admirable job. We have also found power increases by opening the plug gaps to 0.060-inch. To make the drive compatible with steel cams we replace the stock iron drive gear with a bronze gear.

DISTRIBUTOR SELECTION

There are high demands upon an ignition system required to operate at 9000 rpm-plus engine speeds. Not every system can answer these demands but most of the better designs will work, when they are sorted out properly. All current Jenkins Competition drag race smallblocks are equipped with Accel BEI breakerless electronic ignitions. For reasons we will discuss later, they have replaced the Prestolite 201-252 point-triggered transistor systems that we used for several years. We have not explored the possibilities of multiple-spark systems or long-duration capacitor discharge ignitions. Systems of these types have been used successfully by other racing programs and there is no reason to believe they would not be adequate on a high-rpm smallblock. The only other system we are studying at this time is the Delco-Remy HEI breakerless ignition, currently available as an OE ignition on some GM engines.

When we first began work with the BEI system there was no measurable power improvement over the Prestolite. In fact, we had tested several of the highly-touted electronic ignitions and none of them could show actual "power" gains. At least we didn't find anything that could be isolated from the one percent reading error on a typical dynamometer test. All of this testing was with the spark plug gaps around 0.032- to 0.035-inch, fairly standard settings. At that time the Accel engineers and several other engine experts led us to believe that there was no



For point-type triggering we feel the Prestolite 201-252 transistor package with a Delco-Remy 1110985 cast iron distributor is the way to go. With dual Mallory 102X points and some work to reinforce the advance plate this system provides adequate spark through 10,000 RPM. We recommend the point cam be replaced with the 1970240 cam from a '67-'69 Z/28 distributor. The distributor has a ball bearing-mounted centershaft and a mechanical drive for the tach. The bronze drive gear is required to prevent damage to the cam during high speed operation.

At times we have used a specially-built BEI with two LED triggers. With a switching system we can provide slightly-retarded timing in high gear to compensate for the increased engine loading at the bottom of top gear. Several different switching techniques are possible but we use a simple manual toggle switch mounted in a convenient spot for the driver to operate it. This distributor has also been modified with an extremely unique advance system to gain the desired constantly-varying mechanical advance.

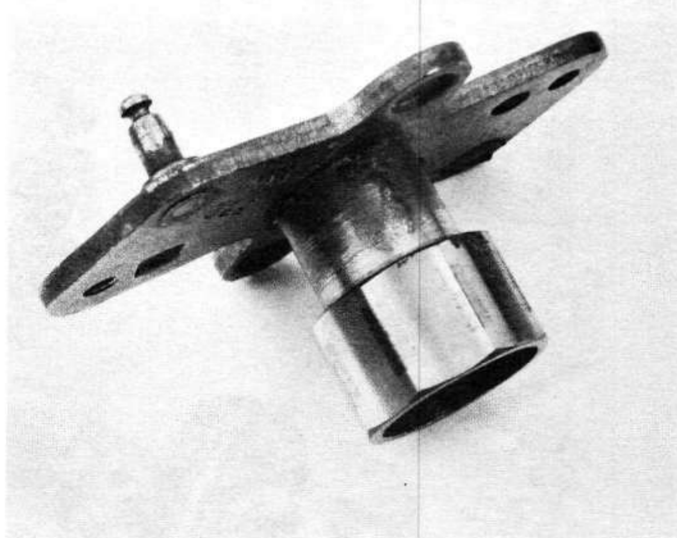
advantage to opening the plug gaps beyond this point. Despite their convincing arguments, we ran several tests with wider gaps and some power began showing up on the dyno curves. In one case we positively found 12 horsepower, with no change other than opening the plugs from 0.035-inch to 0.060-inch. From this we have drawn a conclusion: in this particular application, any power to be found in the ignition is gained through opening the plug gaps. This will be true as long as the system can fire these wide gaps at high rpm and is still able to start the car.

The BEI system works well in completely stock form, except for the advance curve. For our specific needs we modify the curve (see the following section). The stock iron distributor drive gear is replaced with a bronze gear that has been deburred. The electronics package is installed as is, along with a standard Accel coil.

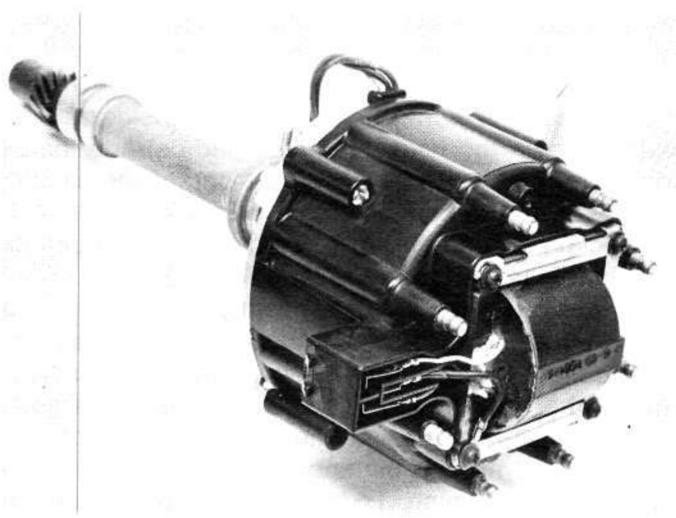
In some engines we have used a specially-built BEI distributor that has a spark-retard capability. This is basically a stock BEI unit with two light-emitting diodes mounted inside. By allowing the ignition to be retarded in high gear, some advantages may be realized (this will be explained in the section on ignition advance). The unit we tested was built for us by the factory and is not generally available. It could be duplicated with very little ingenuity, or the same effect can be achieved with almost any other type or brand of ignition. However, we should emphasize that we have successfully operated smallblocks in competition drag racing without using high-gear ignition retards (this depends largely upon the gear spread going into high).

With the Prestolite transistor system we use the GM number 1110985 Delco-Remy cast-iron distributor. This spark sorter works well with any point-triggered ignition — transistorized CD or conventional. It's not really the best Delco distributor ever made, but it is one of the best still available. In the past we have raced some '58-'61 competition fuel-injection distributors that may have been a little better. Inside these F.I. distributors the breaker points are bolted directly to the cast-iron body instead of being mounted to a separate plate. The center shaft is also full-floating in upper and lower ball bearings. These unique features give them point stability even at 10,000 rpm. Unfortunately, this distributor is no longer serviced by replacement parts and can only be found, if at all, in a junkyard bin.

The 985 distributor can be "tuned up" to nearly match the old F.I. unit, but it takes some work. This



High rpm breaker point action will be enhanced if the performance breaker point cam is used. It has smoother ramp action and provides slightly less point lift. This could increase the rate of breaker point oxidation but it is not a severe problem as long as the points are changed regularly.



The first Delco-Remy magnetic breakerless system has been superseded by the HEI (high energy ignition) breakerless system. It is an ignition module with coil and circuitry built into the cap. Despite the unwieldy size it could be a good wide-gap system.

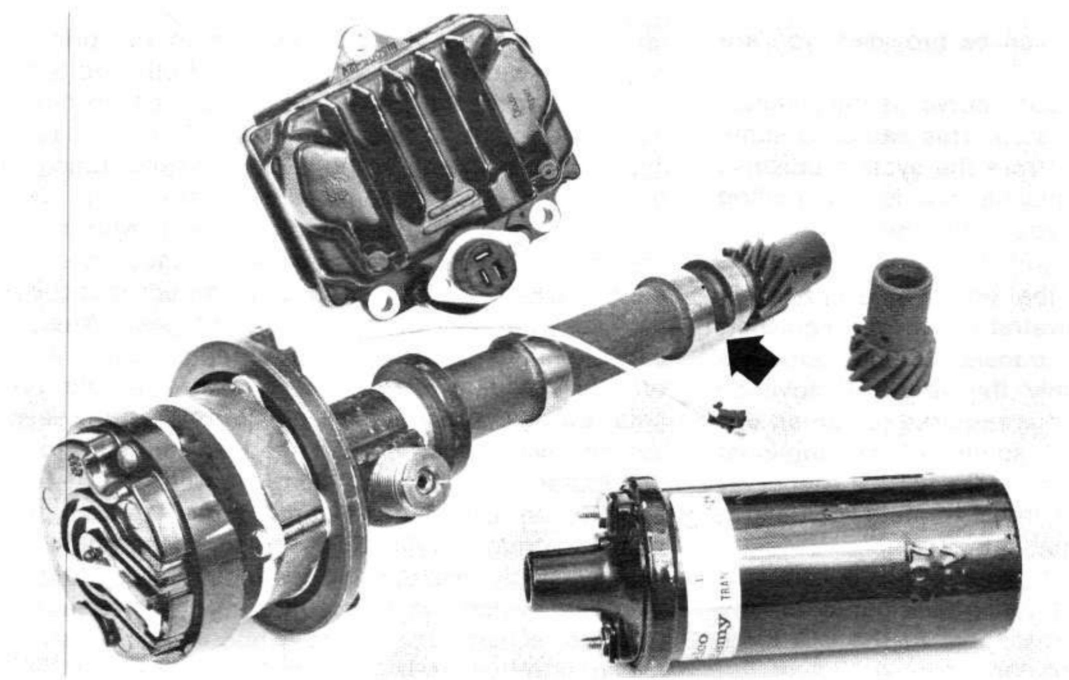
is the same casting used for the vacuum-advance distributors. There is no vacuum-advance point plate in this model, but the points are still fastened to a rather flimsy stamped-steel plate.

It has provisions for a mechanical-tachometer drive. The advance mechanism is above the points, as on all late Delco units. In any point-type distributor we prefer to use the stiffly-sprung Mallory 102X breaker points. These points will operate to 10,000, but the spring holds the rubbing block against the centershaft cam with a terrific force. This pushes the centershaft around in the bearings and forces the mounting plate to bend or "wobble." None of this is particularly helpful for high-rpm timing stability.

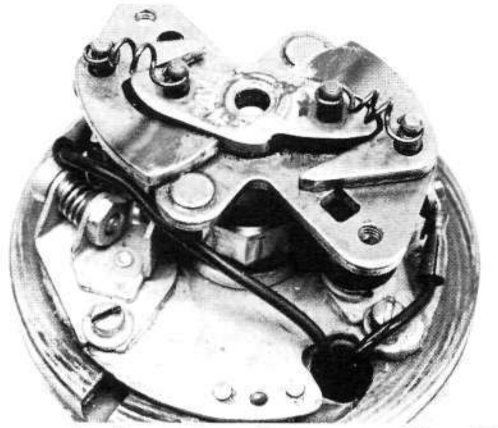
We counteract this by changing the breaker-point cam and increasing the stability of the mounting plate. Any breaker-point action in a Delco distributor can be improved by the installation of the number 1970240 cam from a '67-'69 Z/28 distributor. This design has a milder action and imparts less velocity and lift to the point arm. Of course, opening the points slower and lifting the contact slightly less, with the same amount of dwell, causes more electrical arc across the contacts. With a conventional ignition this results in decreased point life. Yet the arcing should not be insufferable in a street or race engine because this breaker cam was used successfully in the stock '67-'69 Z/28 engines. However, it would be wise to frequently check and replace any breaker points operated against this cam. In a transistorized setup there isn't enough arc to worry about.

Getting the point plate calmed down is a little tougher. Some racers have replaced the stock plate with that from a '57 Corvette dual-point distributor. This plate is thicker and sturdier. It can be used in place of the stock plate or can be mounted right over the stock plate. We feel a better solution is to build up the open area under the breaker plate with Devcon epoxy. The space can be filled entirely, up to the level of the stock plate. The plate is mounted right on top of the epoxy and long screws are used to hold the points to the plate and epoxy filler (they extend right through the plate and into the epoxy below). When the points are installed we use a silicone sealant between the bottom surface of the point assembly and the plate.

This is a plain-bearing distributor but it will give acceptable service to 8500 rpm with these fixes. It is a little edgy at 9000-plus; things began to shake and rattle, but the dwell loss is usually no more than about 3° at this speed. It must be well-greased with a heavy lubricant. If a crankcase vacuum extraction system is used, the distributor must be removed and greased regularly. This system sucks the lubrication out of the distributor rather quickly.



The Nascar Chevy teams are restricted to "standard" ignitions. Most of the current builders prefer the Delco-Remy magnetic breakerless system. The cast iron distributor with tach drive is available as part 1111263. It has a reverse gear for gear-drive cams. This gear must be replaced with a standard iron or bronze gear and the segment of the casting joining the two lower support collars must be machined away for adequate oil circulation around the base of the distributor.



We use a constantly-varying advance, providing 1½° of advance per 1000 engine rpm. To gain this action the weights must be trimmed and very heavy springs are installed. At maximum engine speed the advance mechanism must not rest against a solid stop. Note the open area beneath the advance plate has been filled with Devcon.

IGNITION ADVANCE

The ignition advance is set to give 1½° additional timing per one-thousand engine-rpm increase, with no fixed limit at the top end of the operating range. Once the mechanical curve has been built into the distributor, we adjust the crank timing at some predetermined rpm, allowing for the 1½° per thousand. For the sake of fixing range, we generally use something like 36° at 6000 rpm, 39° at 8000 rpm, and 42° at 10,000 rpm.

There is a controversy over the need for an engine-speed related variable advance. We feel there is some indication that a curve is desirable. The "ideal ignition advance," will follow a curve, but is difficult to find the exact requirements. Normally, you can find a timing-requirement difference in engine reaction over a 2000 rpm speed range. At times it may be very, very difficult to find any difference at a single point in the range and at other times you may be able to find a gain quite readily. In the final analysis, we always feel that if a curve can be provided, you are better off to do it.

An important feature of our "curve" is the elimination of a high-rpm limiting stop. This dampens some mechanical spark "scatter" from the system because the mechanical advance mechanism is not resting against a solid stop. If you lock the mechanism against a metal stop (held against the stop by the high-rpm centrifugal action of the weights), every bit of torsional chatter in the valvetrain, from the camshaft drive chain on back, will transfer directly into the breaker point cam. Certainly the very stiff advance weight counterbalance springs required to run an unlimited curve will transfer some of the torsional vibration, but it's not as severe as with a rubber or plastic stop. In any case, a metal-to-metal contact at the advance stop is absolutely taboo.

With some mechanical ingenuity it is possible to build a constantly varying, unlimited curve in any distributor with advance provisions. In the BEI units we use rubber bumpers in the holes drilled in the weights. This just takes some fiddling until the right counterbalance between the bumpers and weights is achieved. In the Delco-Remy distributors we reduce the size of the centrifugal weights considerably by cutting down the heavy end (see photos) and installing extremely high-rate springs. None of the existing factory or hod rod replacement springs will do the job. A specially-wound spring or some sort of adaptation from another application is required.

In drag race cars with a healthy percentage drop going into the top gear, an advance retard may also be required. At this point the load factor is considerably higher than in any of the other gears. It is going to more nearly represent the conditions of load we see on the dyno and the engine will probably require somewhat less timing. To begin with, if we can run on the bench at 38° advance without detonating under load in the 9000 rpm range, when we put the engine in the car we usually bump it up to about 40-41° (giving 2, 3, or more degrees of "total" advance at any point in the curve), to get better driveability response in the lower gears. We discovered this effect several years ago through experiences on ⅛-mile tracks. On short tracks the elapsed times were always better with 2-4° more timing. However, this same advance would hurt the time slips on ¼-mile tracks. It gradually became apparent to us that the gearing requirements on the long tracks will change the engine loading and, consequently, the timing requirement. In the lower gears the engine will stand more timing, like that used on the ⅛-mile tracks, but when the engine gets into high gear it won't swallow as much advance successfully. After some fooling around, we found the engine prefers 40-42° total in the lower gears and 38-40° in top gear.

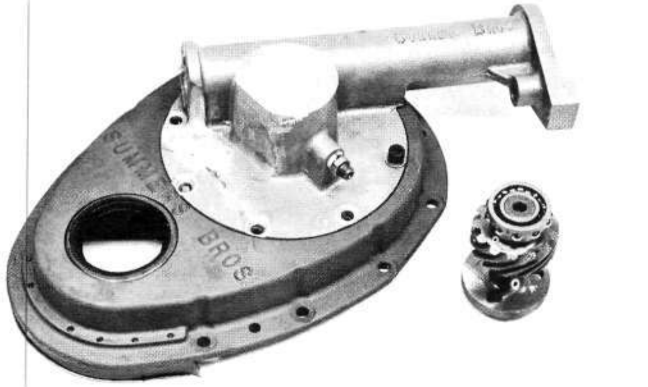
To gain the 2-4° variation, two triggering mechanisms are usually required. The secondary trigger is mounted to an adjustable plate so it may be indexed a few degrees out-of-phase with the primary trigger (at a 90° or 180° position relative to the primary). A simple switch is used to change from one trigger to the other, at the right time in the race, to change the firing point in the distributor. This is easy to accomplish with any of the LED, magnetic-transducer or breaker-point systems. There are many other methods to obtain the switching retard. With a vacuum-advance distributor, a separate vacuum source may be used to hold the plate in an advanced position through first, second, and third gear. When the car is shifted into high, the vacuum source is cut off, the vacuum diaphragm relaxes and the point plate swings to a retarded position. We have seen a simple mechanical linkage hooked directly to the distributor or magneto that allowed it to rotate slightly, on command, retarding the ignition point. The two-trigger system can also be sophisticated somewhat by making the switching automatic. An electrical switch can be mounted on the transmission lever to actuate the retard when the shift lever is moved into high gear.

ALTERNATIVES

We have studied various other alternatives. Along this line, we feel there are many problems with the current magneto ignitions. They won't start the car when the plugs have very wide gaps. This is just throwing a little bit of horsepower down the drain. All magnetos fire four of the plugs in reverse polarity. Spark plugs don't like to be fired in a reverse direction because ionization is less pronounced. With four plugs more reluctant to fire than the other four, the engine is less efficient. It is also very difficult to achieve any advance curve with some magnetos, much less gaining the curve we use.

Some competitive engines have recently been operated with crankshaft-mounted magnetic triggers. We don't think too much of this idea. All such devices have a fixed advance, except for the induction retard that almost always occurs with this type of triggering circuit. As the engine speed goes up, the timing retards because of the circuit characteristics of magnetic inductors. Electronic engineers claim they can now produce induction elements that don't suffer from this problem, although we have never really seen this improvement. Crank-triggering sounds like a great idea but we wonder if it is as accurate as claimed. There is certainly quite a bit of crank twist and/or harmonic balancer twist when the engine is running under load at very high speeds. This should affect the timing. We have wondered what would happen if you hooked up a crank trigger and then rechecked the timing with a timing reference back at the flywheel. We would be willing to bet that a great deal of variation exists between the two. In any event, we feel you sacrifice too much, in return for very little, with these systems.

There may be some potential in the new General Motors HEI breakerless ignition. This is a replacement for the old Delco-Remy magnetic breakerless system. It is a unitized package with the coil and amplifier circuit built into the distributor. The design was originally conceived to fire plug gaps as wide as 0.100-inch. We know that cold-start and secondary-wire leakage can be a problem with this much gap. The GM engineers are now recommending gaps of about 0.055- to 0.060-inch. Potentially, all of the operational problems can be solved, but there are physical limitations caused by the size of this ignition. It is a very bulky package, and in the conventional mounting position it causes interference problems with almost any tunnel ram. In classes where the engine can be set back, closer to the firewall, it may get in the way. In fact, where the engine is already close to the firewall, it is so bulky that it cannot be used to replace some conventional distributors.



New front-mount ignition drives have solved clearance problems.

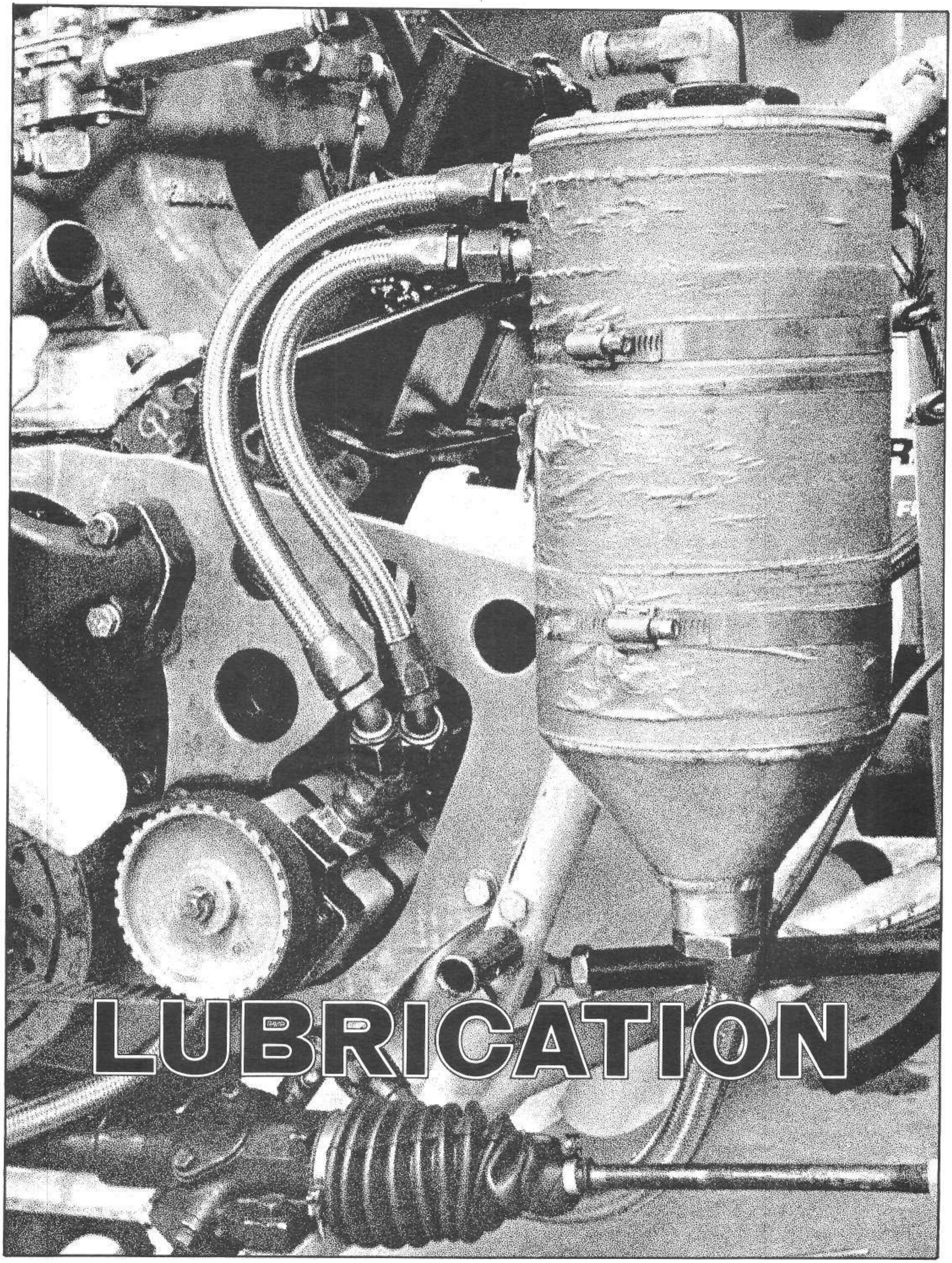
The simple solution to this problem is to relocate the distributor. One of the currently available front-drive mechanisms could be used. We have worked with the Summers Brothers drive, but we aren't completely happy with it, at least in stock shape. Like all of these units, there is difficulty with chatter, and camshaft end-play is very critical. This could be solved with a semi-resilient damper between the cam sprocket and distributor drive. In the final picture, however, we still prefer the stock mounting location for most distributors. You can usually create enough clearance room back there, and it allows you to relocate any related trivia (coil, amplifiers, heat sinks, etc.) inside the driver's compartment, where it won't get too hot.

For our money the "ultimate" system would be a distributor assembly driven off the crank, at some point close to the front bearing, by a gilmer belt. This would eliminate the problem with balancer index variations; the gilmer belt would isolate the distributor from crank torsional problems; and the distributor would retain advance curve capabilities. We have never used a system such as this, but we have seen two other adaptations based roughly on this concept, and they both looked very slick.

NASCAR IGNITION

Nascar rules call for some sort of "stock" ignition. This is a rather vague requirement and the inspectors don't look very closely when the cars go through tech checkout. An OE Delco-Remy distributor must be used, however, as none of the aftermarket hot rod units are accepted. But the inspectors don't look up under the dash, and you can run about anything you like as long as the distributor is stock or "stock-appearing."

Most teams are using the Delco magnetic distributor, available as part 1111263 (cast iron, ball-bearing mounted centershaft, reverse drive gear for gear-drive cams, no vacuum advance and mechanical tach drive provision). With the stock electronics this is not a good high-gap ignition. The spark plugs must be limited to 0.030- to 0.035-inch gap. Most of the racers don't really worry about this. They feel that if they can start the engine and get rolling, everything will be OK. The constant high speed keeps the plugs dusted off and they have few operational troubles. Some teams do carry an extra transistor circuit mounted under the dash as a safety measure. If the primary system suffers from heat or vibration failure, they can switch immediately to the back-up unit.



LUBRICATION

GENERAL CONSIDERATIONS

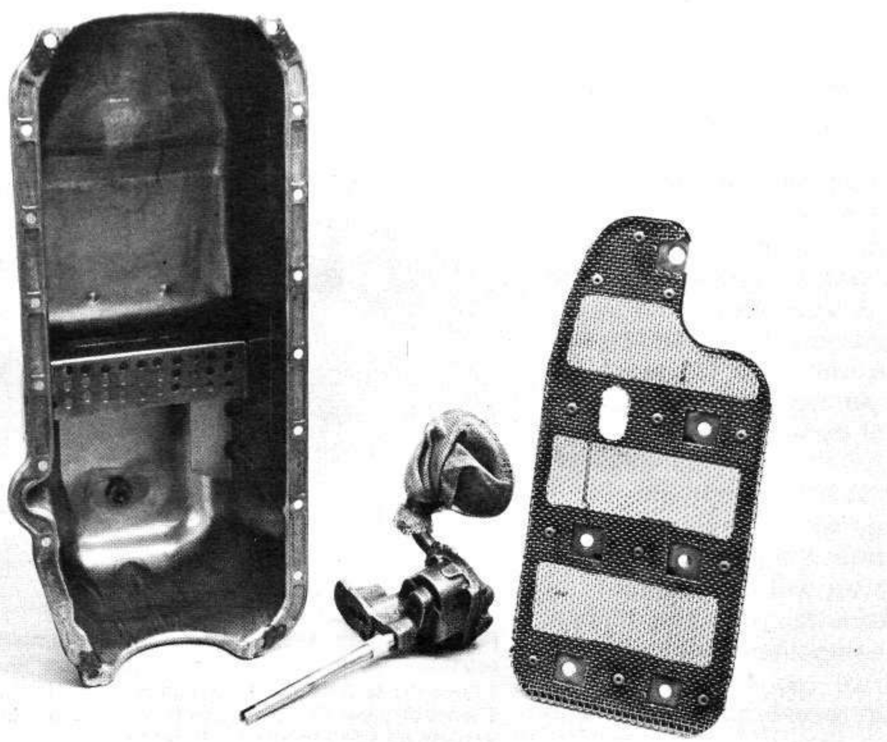
On the surface it seems that the lubrication system has only to perform a simple function. There's far more to it than meets the eye. We feel that a well-designed system can actually add to engine horsepower output. Our tests have verified at least a 10% increase over an average stock system. Some quick arithmetic shows this may mean as much as 60 horsepower in a Pro Stock drag race engine. However, comparing a 330-inch Pro engine to a factory 327 isn't very realistic. We would be willing to bet, though, that any smallblock racing engine now equipped with an average wet-sump could pick up 30 to 40 horsepower with an efficient dry sump. A bigger engine would benefit even more because of the larger contact area on the crank.

We have previously touched upon the efficient nature of the smallblock oiling. It provides very adequate lubrication, even at 10,000 rpm engine speeds. A big advantage the Chevrolet system has over other designs is that it performs this task with a relatively small volume of oil and at relatively modest line pressures. This is especially true when the block has been modified to reduce oil flow to the lifters and up into the rocker arm boxes. With less oil bouncing around in the engine, there is less resistance around the spinning crank, and it is easier to scrape/scavenge residual oil away from the crank.

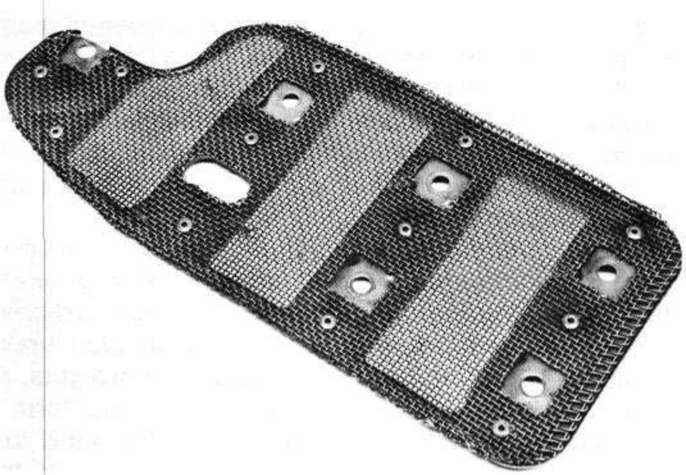
Certainly the system's primary purpose is to supply oil up to the critical mechanical parts of the engine, but consideration must be given to what happens after the oil passes out of the supply passages and leaves the bearing interfaces. At this time the oil falls back toward the pan but it finds the thrashing crank mechanism in the way. Once it does manage to get to the bottom of the pan, the oil pump picks it up for redistribution into the supply system. These secondary stages of operation — drainback and collection — cannot be overlooked. This is where a surprising amount of power can be absorbed through frictional losses.

Observations of the action inside the crankcase/oil pan at high engine speeds are simply beyond belief. The oil hangs together like a rope wrapped around the crankshaft. This acts like a giant brake shoe, resisting the rotation of the counterweights. At times the loose oil droplets will collect and form a little ball. These balls float around in the wind turbulence created by the crank, often following in the air voids behind the crank pins as they thrash around in the enclosed crankcase. When the ball gathers enough volume, it may fall away from the assembly. If the pan is close to the crank, the ball will actually bounce off the pan surface and spring back up into the crank assembly where it smashes into the spinning mechanism. This occurrence creates more friction and absorbs power.

The biggest contribution any oil system can make to the engine efficiency is to keep the oil down, out of the upper engine where it doesn't do anything but splash around and create trouble. It is difficult to imagine exactly how much effect this residual oil can have on the engine. The concept of keeping excess oil away from the spinning crank was first discovered by our competition. Like most of the other racers, when we first heard about this "windage" problem, we listened, with tongue-in-cheek, and didn't think they were gaining as much as they claimed. However, our



Oil system functioning is closely tied with parasitic power loss in the lower case and spinning assembly. Many detrimental efforts are directed at providing an excessive volume of oil to the crank and valvetrain. This only causes oil control problems and power loss. In a racing engine the oil pressure and circulating volume should be kept to the absolute minimum, without incurring mechanical damage. Dry-sump systems can accomplish this in a better fashion but for wet systems the 6-quart Corvette pan along with a semi-screen baffle and modified pressure pump is adequate for most drag engines.



In restricted-capacity pans a semi-screen baffle should be used to isolate the crank windage area from the sump oil. A solid baffle reflects oil back into the assembly, but a semi-screen construction breaks the windage oil into small droplets that fall into the sump.

own tests have shown that they were right on, straight across the board, and these techniques save as much or more power than they claimed.

A good racing system can reduce this oil-related power loss. In one recent test we had an indication of what an efficient oil system should be able to do. This was a Pro Stock 330 with a Weaver dry-sump pump, one of our competition dry-sump pans, a screen-type baffle, and about 8 quarts of oil in the system. We had a sight gauge on the reservoir tank to check the amount of oil in the tank during the test, giving us a good indication how much oil would be retained in the engine. With everything warmed up and running at 2000 rpm, the engine would pull about one-half quart out of the reservoir, leaving 4½ quarts in the tank, two quarts in the filter, and one quart in the pump and lines. *At 9000 rpm the sight gauge still indicated the engine was only retaining one-half quart of oil!* We consider this to be very good performance from the sump scavenging system and the results can be seen in the dyno readings.

Our studies show that certain elements of the pan/baffle assembly are very important, but we can't say that we have the final answer. At times we have actually seen the in-car dry-sump system beat the deep wet-sump dyno pan for horsepower, and this is very difficult to do. A surprisingly simple change in one part of the combination will cause the assembly to retain more oil. As of yet we don't have sound explanations for why some of these things happen.

WET-SUMP SYSTEMS

In the preceding sections we expressed some of our general observations about the smallblock oiling system. Within this context we will discuss some of the acceptable wet-sump techniques. It has been a long time since we ran a competition engine with a wet-sump oil system, but we can generalize some interesting theory from our recent discoveries with the dry-sump equipment.

First of all, it is absolutely impossible to build a wet-sump pan "too big." Every effort must be made to get more volume in the pan. With this you also run the minimum amount of oil and isolate the crank assembly from the lower sump. We think the difference between a chassis that allows a large volume pan and a restricted chassis could be a very important factor in selecting a drag car/chassis for competition. An excellent example of how this could be put to advantage is the NHRA Super Modified class. Many of the current cars are built from the '67-'69 Camaros. This is a terrible car, from the standpoint of oil pan design. It has a huge front frame crossmember running underneath the front half of the pan, severely restricting how much the pan can be enlarged. A better choice would be one of the early Chevy II chassis. These cars had no restricting structures under the engine and they can be fitted with as large a pan as can be put under the engine without hitting the ground. A pan 10-12 inches deep would certainly be possible. In our estimation this could be worth 10-15 horsepower in these cars. Another example that comes to mind is the late-model Super Stock Corvettes. When these cars suddenly emerged as "killers" in the category, it was interesting to note that they had huge oil pans under the engines. This chassis is fairly open in this area and some knowledgeable racers saw this advantage.

Any time the pan makes a circular wrap around the crank, such as the forward portion of a standard

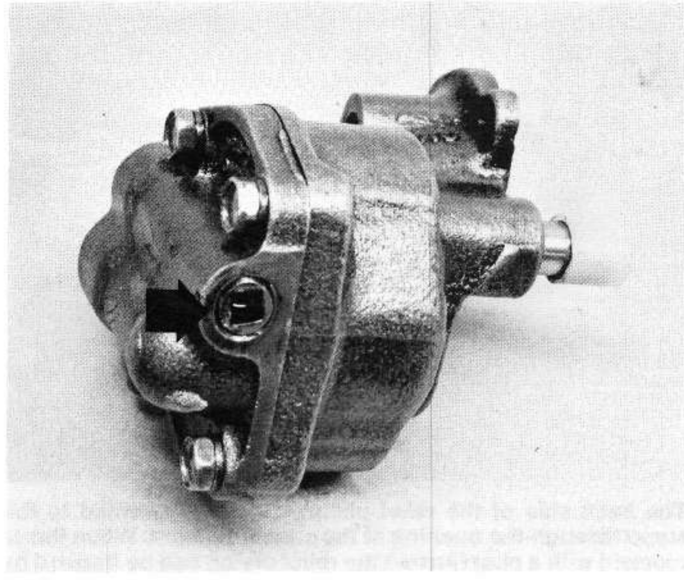
For wet-sump systems a standard smallblock oil pump provides more than adequate pressure and volume if the valvetrain and running clearances have been properly set up. We modify the pump to give increased stability at high engine speeds and to increase reliability.

wet-sump pan, it is prone to bounce oil back up into the assembly. In ordinary applications people believe all of the oil is going to be pushed to the rear of the pan, under acceleration, and the front portion isn't going to be important. This is not true. A good deal of the oil will be held in the front of the pan, blocked by the terrific turbulence created around the spinning assembly.

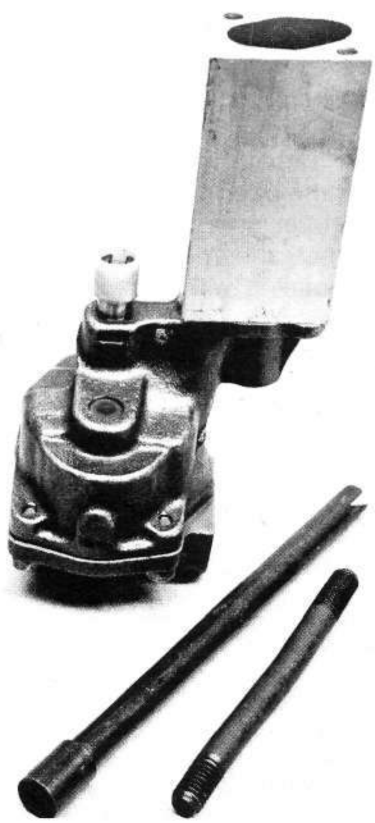
Because of this action we don't recommend solid baffles or windage trays under the crank. Such a baffle mounted directly below the crank gives the oil a nice flat springboard to bounce away from and get right back up into the assembly. If a very deep pan can be used, such as the 16-inch pan we have on our dyno engines, no baffle whatsoever is required. When the pan floor gets up nearer to the crank, something must be done to buffer oil coming off the assembly. We prefer some sort of screen baffle/grate. They will break up the oil balls as they dance around, and force the oil to return to the lower sump. We fabricate our own baffles from ⅛-inch screen mesh, as nothing suitable has been commercially available in the past.

The pan should also be as wide as possible, but this is difficult to achieve. With the starter hanging on the right side of the engine and the header clearance required in most cars, there isn't much room left to put side sumps on the pan. However, if it is possible to get a big, wide-bottom wet pan on the car, some provisions will be required inside the pan to channel oil toward the pickup. This doesn't need to be too complicated. The pickup should be all the way to the rear of the pan. Under acceleration the oil on the floor will move this way, so supply isn't a big problem initially. However, during braking some baffling should be provided to keep the oil from surging forward away from the pickup. It's also possible to shut the engine off during deceleration to prevent operation under oil-starved conditions. Swinging pickups might also be useful in this respect, but we have no experience upon which to base a specific suggestion. It seems wise to give some consideration to the starter location. By using a large-diameter flywheel and offset starter, it is possible to get more clearance for the side of the pan. The larger flywheel also has a separate advantage. It will give greater inertia-per-pound of weight, allowing the wheel to be slightly lighter without losing the all-important inertia required to launch the car at the line. As another sidelight to the chassis considerations outlined earlier, we have also noted it is much easier to inspect and maintain an engine when there isn't any chassis or steering linkage hanging under the pan.

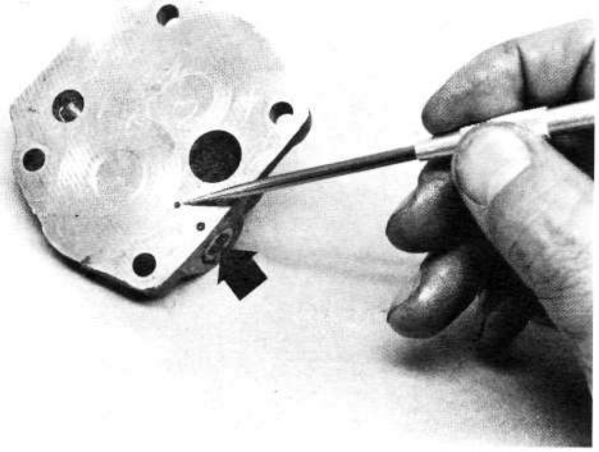
Enlarging the pan is essential, but you can't expect to find power with this change alone. We have operated a 9-inch deep, flat-bottom, wet-sump pan on the dyno and found hardly any improvement over the stock 5-quart Corvette pan. The next key is the windage baffling and scraping system. When properly designed, these implements get the oil away from the crankshaft and down into the sump. When the oil level in the wet sump can be kept 5-6 inches away from the lowest point in the crankshaft travel, a screen-type "grate" will be much better than a solid baffle. If the clearance is less than this, a solid baffle may be required to keep the crank windage from



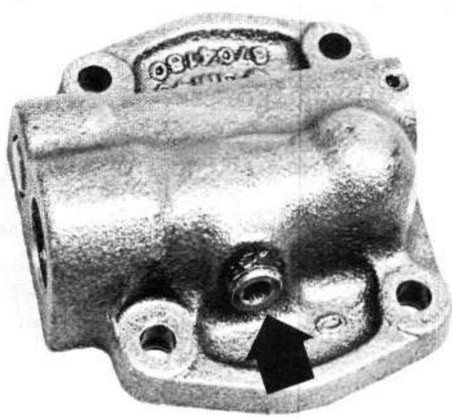
We have seen the oil pressure bypass piston and spring blow right out of the pump body. To totally eliminate this disaster we replace the roll pin retainer (arrow) with a screw plug. The spring and piston must be shortened a commensurate amount to allow for the plug.



When a deep sump is used to lower the oil level away from the crank assembly, a pump extension mount must be used to lower the pump body into the oil. Extended pickups are generally not acceptable.



The back side of the relief piston is normally vented to the sump through the opening at the end of the bore. When this is blocked with a plug (arrow) the relief piston can be trapped by oil buildup between the piston and plug. We drill a bleed hole back into the gear housing.



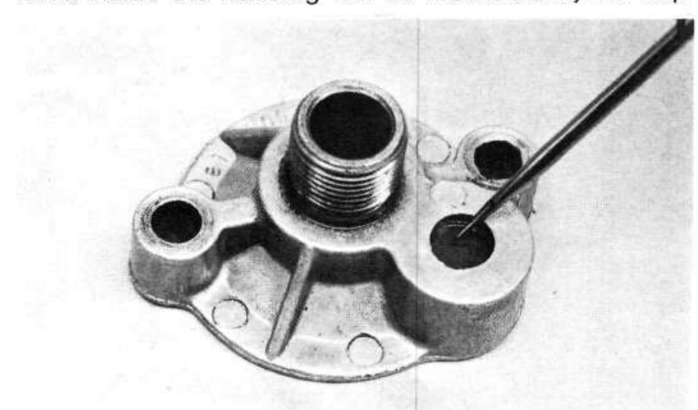
To provide adequate bypass volume during cold-start and during high engine speeds we increase the size of the bybass pressure-relief channel. The stock plug is removed and the passage is drilled to ⅜-inch. A larger screw plug is then installed in the access hole.

whipping the standing sump oil into the turbulence. Beyond this level, or something close to this, a screen grate will break up the windage sufficiently and will do a better job of grating oil away from the assembly. A screen will peel oil away from the encircling cloud, break it into slower droplets, and let them fall back into the lower sump.

Along the right siderail (the upwind side of the windage pattern) a profiled windage/oil scraper can be sandwiched in between the block and the pan rail. This is a piece of narrow sheet steel extending inward toward the spinning assembly. It is shaped to give minimum clearance to the individual components of the connecting rod-crankshaft assembly as they swing past this point. It scrapes off any residual oil that may be riding around with the assembly, hanging off the edges, but not thrown away because of the turbulent wind-created voids in certain areas. Shaping of the edges of the counterweights may help this throw-off. Some tapering of the weight outer circumference will direct hang-on oil to the outer edge where it can be more efficiently removed by the profile scraper(s). These techniques are more beneficial at very high engine speeds and we feel they are right on the edge of readable error. One thing that does show up quickly, however, is the condition of the piston skirts. When we began fitting the engines with profile scrapers and screen grates, we found much less evidence of metal "flak" in the piston walls and piston skirts. Apparently controlling the oil and keeping it down out of the assembly gives the metal flakes (existent in almost every engine) a chance to settle out of the oil.

The last thing to watch in a wet-sump system is the oil pump. When we competed with a wet-sump system we ran a standard smallblock pressure pump, but we spent a little time with it to add some reliability. Early in the program we had some problems with the regulator valve springs and pins. We actually had one of the pins shear, leaving the engine with zero oil pressure. Needless to say, we lost that engine. This might have been a freak accident, but when these things happen we take steps to prevent them from occurring again. The stock method of retaining the pressure regulator spring in the housing is with a pin. A simple method to increase the strength is to remove this pin and replace it with a bigger hard pin.

A much better method is possible, but it takes more work. We prefer to put a plug in the end of the pressure regulator housing, in place of the stock pin. This closes the housing off completely, eliminating any possible chance of the regulator assembly coming out of the pump body accidentally. This does create another problem though. The pressure regulator spring and valve assembly is balanced to crankcase pressure through the open end of the housing. This is a necessity for proper pressure regulator functioning. If there is no relief, the movement of the regulator valve inside the housing will be restricted by oil trap-



The standard pressure-relief at the filter mount blows open at 12-15 lb pressure. As a result, oil in the pan is constantly circulated to the engine without filtering. This puts leftover metal into the running assembly. We always block off this relief. It is possible to use a standard filter without this relief as long as the pump bypass pressure is reasonable. We prefer the long Chevy truck filter on a remote mount. It will handle full oil circulation without any relief.

ped in the passage. The plug installation closes off the end of the passage that normally provides the pressure balance. It is possible to drill a small hole in the plug to regain the pressure relief. But most oil systems have the pressure-regulator piston balanced to the low pressure side of the pump rather than to some other source (the value of this may be academic). To duplicate this, it is simple to drill a small bleed hole from the area behind the piston, into the low-pressure side of the pump gear assembly (see photo).

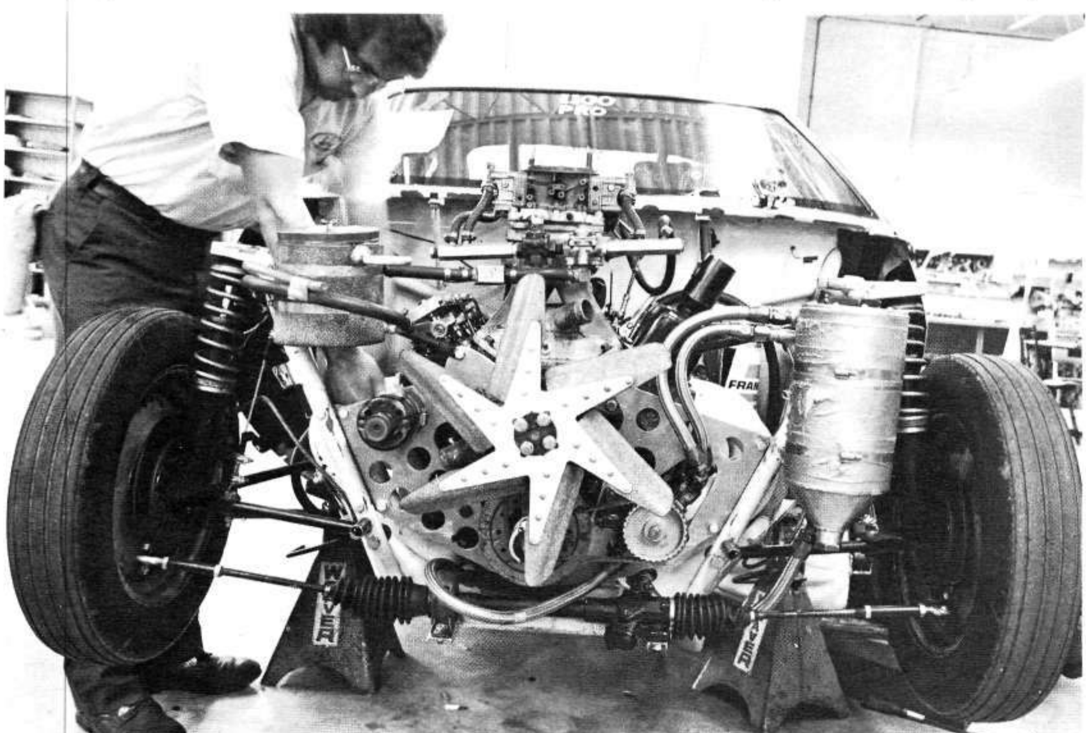
When the plug is installed in the end of the pressure regulator housing, it may shorten the length of the housing (the plug shown here extends deeper into the opening than the location of the drilled holes for the roll pin). To gain proper operation the pressure regulator spring must be shortened to make up this difference. The regulator valve must also be trimmed to insure it will not close off the relief passage leading back to the gear housing. This is a delicate balancing act. The regulator action must give the right reaction to the pump pressure, and when the pressure fully opens the regulator valve against the spring/plug, the relief passage must remain completely open. Otherwise, the valve will seal off the passage and proper pressure relief to the backside will be prevented.

We balance the relief to give 55-60 lb hot oil pressure. With a racing engine this can drive the cold-start pressure to 80-100 lb. We have experienced problems with wear in the distributor-oil pump drive gear from this much loading. The production steel gear may wear quickly or the drive gear machined in the cam may begin to give way. This dumps a lot of metal flakes into the engine, and will affect the ignition timing. We always suggest that an aluminum-bronze gear be installed to the distributor. It will still wear, but not as quickly.

There is also some slight problem with oil pump "cavitation" and subsequent "spark scatter" created by the load fluctuations of the pump gears. The problem is not as bad in the smallblock as in the Mark engine. The situation is particularly severe when the engine is cold. Under these conditions the engine can't use the volume of cold, sluggish oil coming through the pump. Consequently, the oil pressure inside the pump builds up in the manner described previously. As this occurs, the pressure regulator valve in the pump cover is pushed wide open and the excess oil is directed into the pressure-relief channel joining the pressure regulator valve housing to the pick-up passage in the pump cover. The size of this bypass channel controls the amount of oil flowing back to the pickup. When the pressure is high, the velocity through this channel is also very high. As the oil dumps into the pick-up channel with this velocity, it disturbs the oil coming into the pump. This causes an interruption of the oil supply and pump cavitation.

In the past we have used stabilizing grooves ground into the pump body and cover to reduce this chatter. We now use a slightly different approach. The easiest solution is to enlarge the size of the bypass channel. In stock form it is approximately 5/16-inch in diameter. By removing the access plug (see photo) in the cover, it is possible to drill the passage to about ⅜-inch without difficulty. The access hole is then replugged with a ⅜-inch threaded pipe plug.

When mounting the pump in a deep pan we recommend using an extension spacer to lower the pump down to the floor of the pan. However much the pan floor has been dropped, will dictate the length of the extension. Leaving the pump up on the cap and extending the pickup down to the oil is not an



In any unlimited racing situation a properly-designed dry-sump system will provide a performance increase. It would not be unreasonable to find 10% more horsepower merely by switching from a wet sump to the dry system. It is especially beneficial when clearance below the engine requires a very small oil pan. In the same way going to a dry sump with a very small pan allows the engine to be lowered in the car, permitting lower front end profiles and more clearance on top of the engine for the induction.

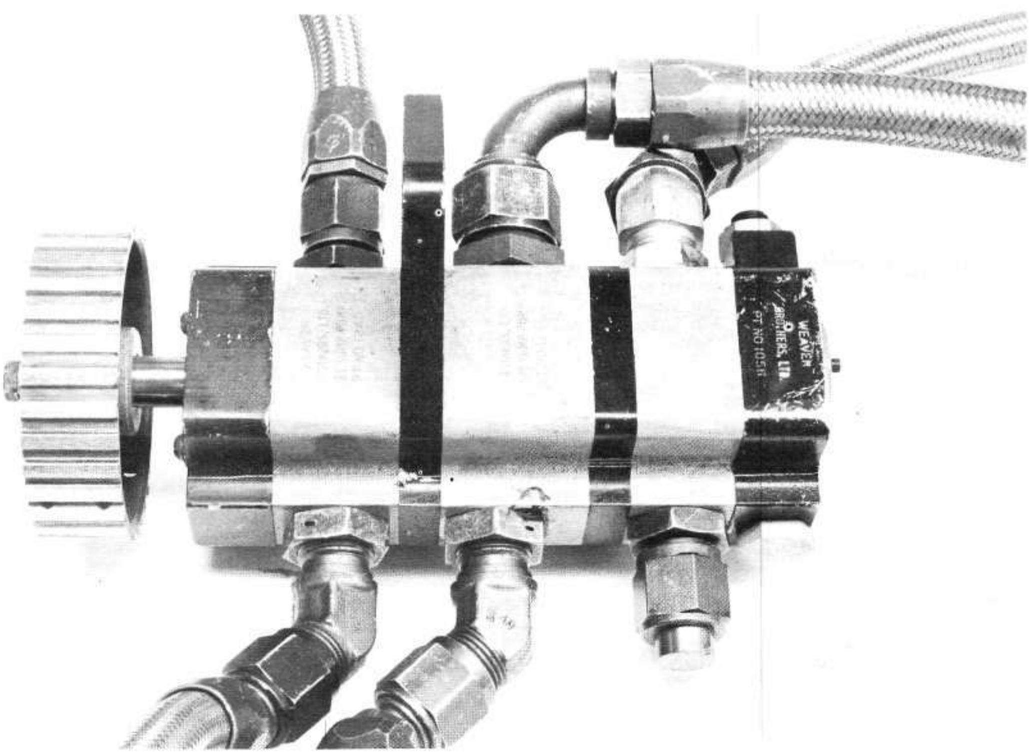
acceptable method. This leads to problems priming the pump and if the pick-up tube is very long, oil cavitation or starvation is possible. The stock pick-up shroud works very well. If space is a problem it is possible to fabricate a small cone or box-shaped shroud on the end of the pick-up tube. In this instance a fine-mesh screen should be used to protect the pump from metal flak and as an aid to break up vortex action created by the pump suction.

DRY-SUMP SYSTEMS

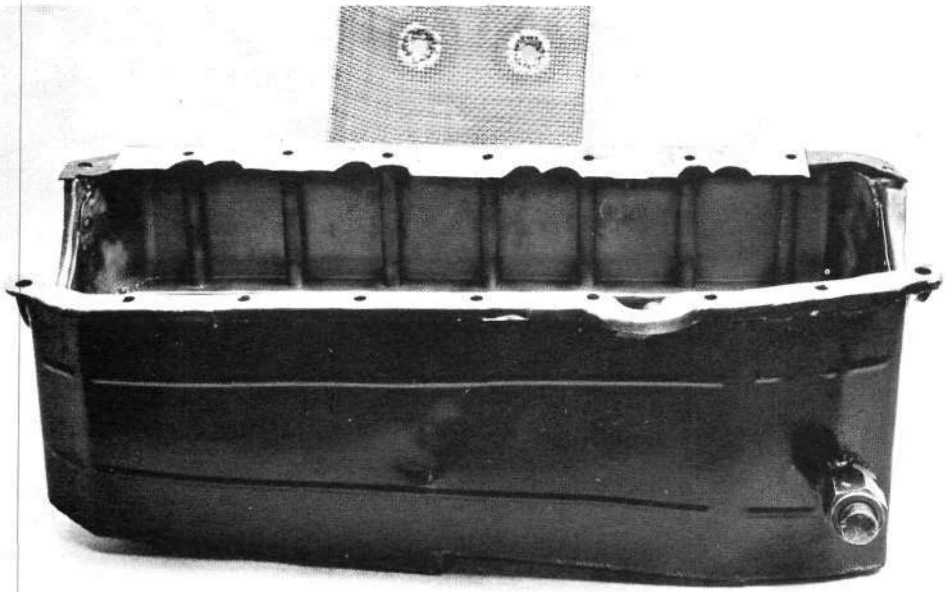
We feel the dry sump has even greater power advantages in the car than indicated just in the dyno tests. It's fully possible to have the dry-sump pan nearly empty during acceleration (if it and the pick-ups are well-designed). In a wet sump you normally have 6-7 quarts of oil inside the pan. No matter how many trick swinging doors and baffles you have in the pan, nobody knows exactly what happens to that oil when the car is propelled forward at the starting line. It is generally conceded that it will "stack up" in the back of the pan and may even get up into the spinning assembly if the pan is not deep enough and not well-baffled. Whatever the combined effects of wind and acceleration may be, we don't think it's doing the engine any good.

It is dangerous to approach the installation of a dry-sump system without some careful thought. We know racers who have bought and installed one of the commercially-available dry-sump systems and *lost 30-40 horsepower.* This doesn't surprise us. When an inadequately-designed dry-sump system is "bolted on" a smallblock engine, there will be a power loss. The trick is in the design of an oil pan/baffle/grate/pickup combination that will effectively scavenge the oil away from the crank assembly. A dry-sump system can achieve this much more effec-

Our reservoir is about minimum size, 14 inches tall, 6 inches in diameter. It holds 4 quarts and is internally baffled for air separation. Number 10 lines feed from the line-out sides of the pickup sections of the pump into tangential dump tubes at the top of the tank. A number 12 line feeds oil from the tapered bottom of the tank to the line-in side of the pressure section.



The heart of the dry sump is a three-stage Weaver Brothers gear-type pump. The 1.45-inch front scavenge section draws from the forward half of the pan. The 1.7-inch center scavenge section draws from the rear portion of the pan. The rear section is 1.2 inches wide and provides pressure feed into the remote-mounted filter and on to the engine. The line-out hot oil pressure is 55 lb and the line diameter is number 12. The pumps run at 55% of crank speed. The pump speed should be limited to 5000 rpm as they don't like to wind tighter than this. The gear pump is also susceptible to ills from ingested metallic leftovers. Every effort should be made to protect them from swallowing engine garbage.



The absolute key to finding horsepower in a dry-sump oil system lies in the design of the pan and the scraper/grate combinations. An inadequately-designed pan will cause the engine to lose power. The difference between a bad pan design and a good one could be over 30 horsepower! Within the limits of the engine/chassis, the floor should be as far away from the crank centerline as possible. The main desire is to scrape and grate the residual oil away from the incredible turbulence created by the spinning assembly. Once it is returned to the pan floor it must be drawn out as quickly as possible.

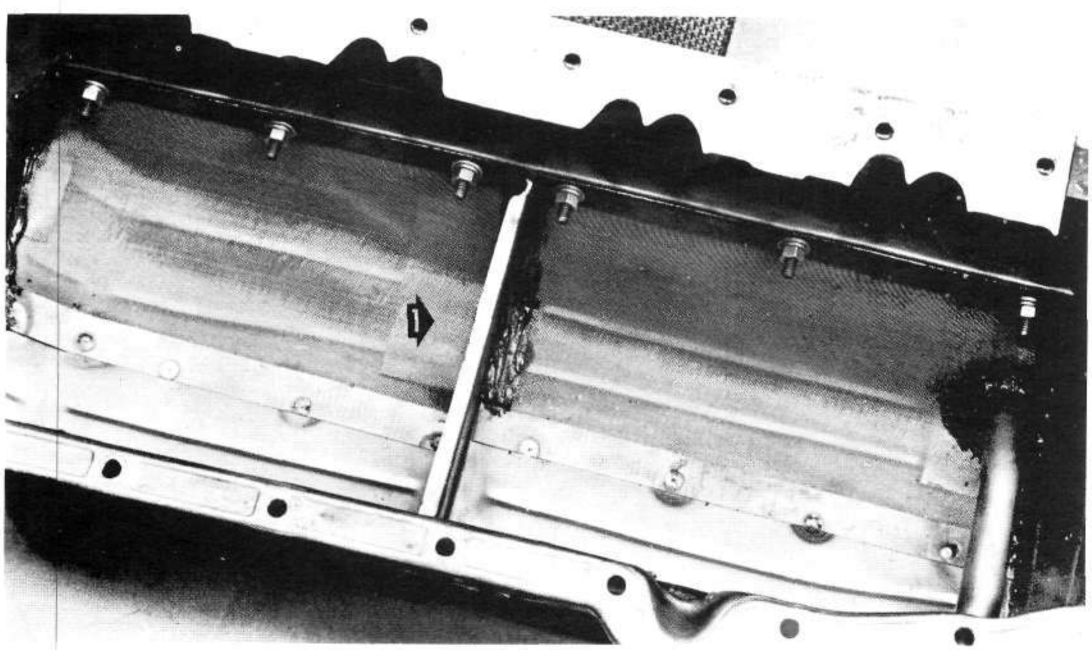
tively than a wet-sump system. As in most engine modifications, it's not what pieces you put in the engine, it's how effectively you use those pieces.

One of the key factors in a dry-sump system is the design of the pan. In past years we tested most of the commercially-available pans and found none of them were doing an adequate job of controlling the oil. We spent countless hours building and testing pans to find some improvement. As mentioned earlier, we have a system that now leaves only one-half quart of oil in the engine, at all speeds from 2000 to 8500 rpm. When you achieve this sort of oil scavenging in a smallblock, you're going to definitely see a jump in the dyno numbers.

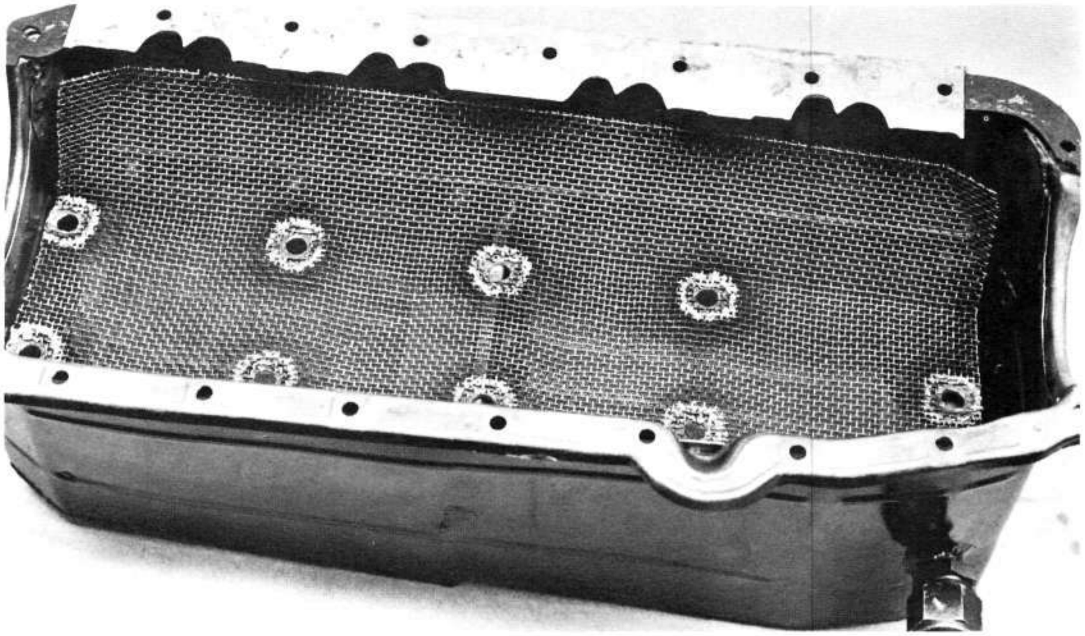
In a very low ground-clearance application, such as the typical dry-sump installation, where you can't get the floor away from the crank, you must make a side sump on the right side of the pan. The scraper or scoop arrangement is then used to channel the oil over to that side, away from the crank.

The side sump collects the oil and lets it drain down to the pickups. Tubes can be welded through the side sump to allow access for the fasteners holding the pan up against the block. All-Thread or long bolts reach through the tubes.

In our very latest designs we have tried curving the left side down into the floor to help direct the oil toward the side sump. It must, however, not be too close to the crank assembly. This might cause problems with the "ropes and balls" of oil hanging around the cranks, but the intent is to use this surface to direct oil over toward the pick-up area. We may grind some of the block material away along the left side-rail to let the wind generated by the crank blow oil down toward this surface, where it is channeled across to the pick-up section.



The floor of the pan is divided into two sections. It slopes to the rear and toward the right-rear corner. A one-inch pick-up tube leads forward from the right corner of the center divider (arrow 1), out the right side of the pan where it joins the scavenge line to the front pump section. A similar tube scavenges oil from the right-rear corner of the rear compartment. Each pickup is covered by a solid metal baffle and a fine-mesh screen keeps metal away from the pickups and acts as a secondary oil-bounce cushion.



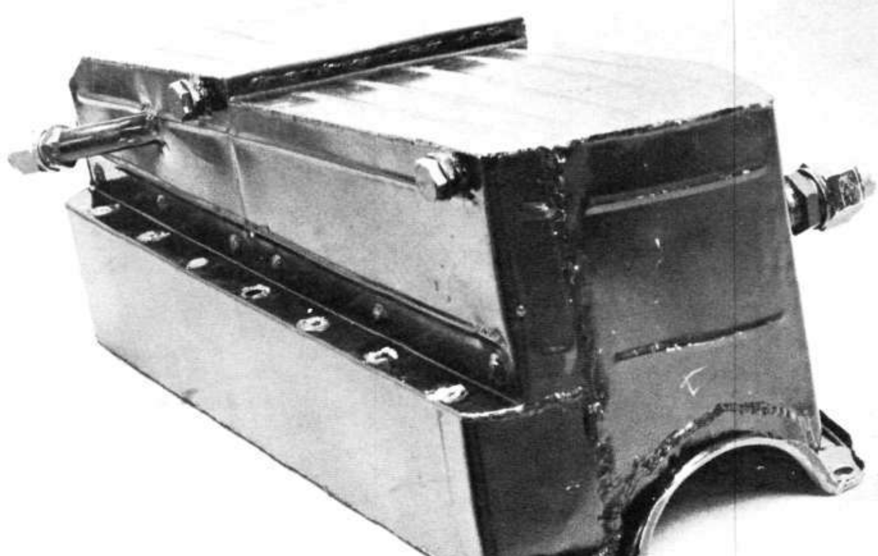
When the oil pan is nearly empty during operation, as in a dry-sump pan, a simple screen grate is much better than a solid or semi-solid baffle. The screen is approximately ⅛-inch mesh and must extend the full length of the pan. It is held in place with stock Chevy extended studs. A profiled scraper is also used along the right pan rail. The screen breaks the oil up into small droplets rather than providing a flat surface to act as a springboard for resilient oil action created by the severe air tubulence.

let the wind generated by the crank blow oil down toward this surface, where it is channeled across to the pick-up section.

You will note in the photos that the example pan is pretty deep. This was an attempt to discover if a deep dry-sump pan might provide more power. We couldn't find anything on the dyno, therefore, we believe a dry sump may not be depth-sensitive, as long as the pump is pulling the oil out of the pan quickly. The deep section also extends all the way forward to the front lip seal. Fabrication may be simpler if the sump only extends to the front of the forwardmost counterweight and none of the efficiency will be lost. The floor of the pan is not "flat" with respect to the crank. It slopes downward, toward the right, and back, toward the right-rear corner. Currently, we feel this slope should be no less than 1-1½ inches from the left-front corner to the right rear. The sample shown here also has a step in the floor. This is only an attempt to increase the average overall pan depth. The forward pickup is placed in front of the step and a small baffle wall runs across the inside of the pan right at the step. This effectively isolates the front floor of the pan from the rear. Dividing the oil reduces sloshing somewhat during braking action. We are not completely convinced this is the ultimate answer for a smallblock dry-sump pan, but it's better than anything else we have tried.

A fine-mesh screen covers the pickup-side of the pan floor. The screen extends the full length of the pan and is sealed around the edges with some sort of RTV (we use the GM brand because it is very resistant to heat and chemicals). This is a secondary shock baffle to prevent windage from disturbing the oil flowing into the pump pickups. It also filters metal flak out of the oil entering the pumps. We have found

This is one of our earlier experimental pans. The step in the floor was an effort to increase the overall average depth but we have found this is unnecessary. The right side-sump can be seen here. Long bolts or All-thread studs must be used to hold the right side in place. The more room you can provide on this side of the pan, the better. You can see that the floor slopes toward the right-rear corner. Even more slope than is shown here is desirable. Note the left side of the pan is square. Our later designs use a rounded section here to channel oil toward the right side.



that the Weaver dry-sump pumps don't like to ingest metal "leftovers." Solid metal baffles cover the pick-up entries to gain further protection from the windage.

Currently, we use one-inch tubing to form the pick-ups. Some method must be used to prevent vortexing as the oil rushes into the pick-up opening. We use a sloping or tangential cut on the end of the tube. A better method might be to run the pick-up tube into a 1½- to 2-inch diameter can-like collector with a slot milled along the bottom and a screen over the end. This would reduce the velocity gain as the oil enters the tube and prevent the fluid from building a vortex at the mouth of the tube. Anything similar to the current stock GM pickup, except maybe smaller, would work well.

In any case, the end of the tube must be covered to prevent windage from disturbing the fluid entry and some provision must be made to reduce vortexing, otherwise the oil will not leave the pan as rapidly as is required, and as the air is sucked up with the spiral of oil it will have to be separated from the oil in the reservoir tank. In the current setup we run a number ten line to the front scavenge pickup and a number 12 to the rear pickup. When the pumps are properly selected this seems to handle the oil flow adequately.

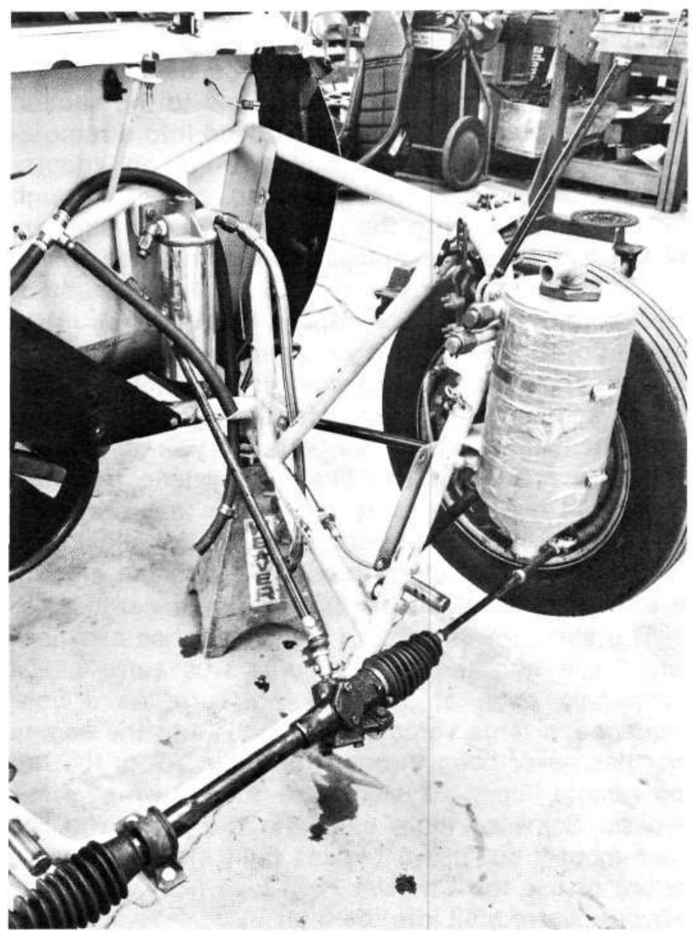
The large mesh screen grate is mounted below the crank with the GM extended studs. Note that the screen is completely full length, from the front all the way to the rear of the pan. A profiled scraper is used along the right pan rail. We have also investigated profiled scrapers mounted down along the extended studs, above the screen grate.

We use a three-stage Weaver pump in all of our drag racing systems. Two sections scavenge the pan and one section provides pressure. We have seen some systems on bigger engines with as many as four scavenge stages. In these cases two of the sections were used to pull oil out of the rocker boxes. If oil collects in the covers, this may be a good idea. We haven't found it necessary to do this because we don't put any more oil into the covers than the absolute minimum required to make the rockers survive.

The two scavenge sections are 1.7 inches wide (rear scavenge) and 1.450 inches wide (front scavenge). The pressure section is 1.200 inches wide. The pump is turned at 55% crank speed with a standard gilmer belt setup. We have run smaller sections at faster speeds with success, but the pumps don't like to turn faster than about 5000 rpm. Depending upon the design of the pan, it may be necessary to enlarge the size of the section pulling from the rear portion of the pan. We think this is a better solution than going to a third pan scavenge section. There's nothing at all "trick" about the Weaver pumps. We use them in stock condition and they give no trouble as long as the speed limit is observed. The pressure stage is adjusted to give 55 lb hot oil pressure.

The reservoir tank we use is about as small as possible, and is acceptable in a drag race car (it is too small for a road race car). It is about 14 inches tall and 6 inches in diameter. Some provision must be made to get the air separated out of the oil while it is inside the tank. If vortexing is reduced at the pickup, this is less of a problem, but it must always be considered when a dry-sump reservoir is designed and built. The tank should be cylindrical in shape with the entry tubes coming in at a tangent to the wall. This allows the oil to flow around the walls of the tank, letting the air escape up the middle toward the centrally-located vent. The taller and thinner the cylinder, the better this effect will be. The vent is hooked to the valve cover of the engine with a large-diameter line. This is important to balance the pressure inside the tank to the crankcase pressure (remember the crankcase is only vented to atmosphere through the vacuum extraction system). Our tank has three horizontal baffles inside to help control sloshing and assist air separation. It would be possible to use less oil in the system if we had a third scavenge section with a pick-up positioned in the pan to accept oil under braking

The long Chevy truck filter is used for 100% oil filtration. The entire system carries about 8 quarts of oil. The reservoir holds 4 quarts. The lines, the pump, and the filter hold about 3½ quarts and when the engine is running at wide-open throttle the reciprocating assembly requires only one-half quart of oil. Note the fitting in the top of the reservoir that provides venting to the valve covers, balancing the air space above the oil level to the vacuum conditions in the crankcase.



Several different filter possibilities are open to the Chevy racer. The remote mount for a single filter (left) must be used with a Chrysler/Ford filter, having a built-in pressure relief valve. The double filter mount (center) also requires bypassing filters. Neither will give 100% oil filtration. In fact, under high pressure they filter less than half of the oil passing through them. The AC 832 Chevy truck filter (right) used with a Traco remote mount gives 100% filtration with no pressure-relief bypass. Several block adaptors are available to route pressure oil into the block through the stock filter pad. The stock filter mount (center below) must have the pressure relief blocked.

conditions. We don't want the power loss from a third section, and the four-quart reservoir usually provides enough oil volume to feed the engine during the normal deceleration period at a drag strip.

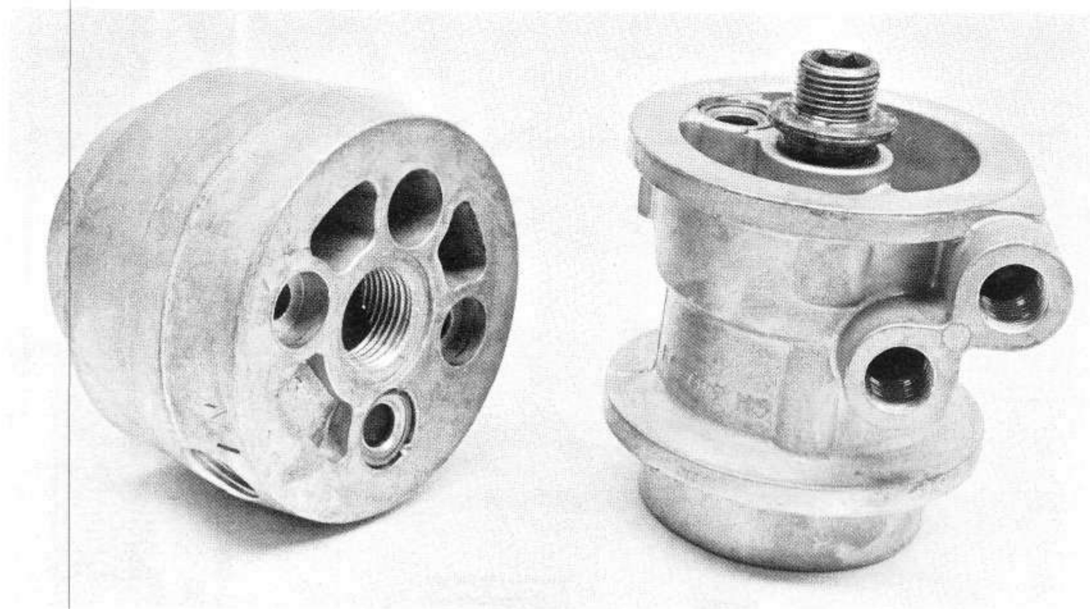
Oil routing through the system is similar to all dry-sump designs, except that we don't use any external oil coolers. The oil leaves the bottom of the tank through a single number 12 line to the suction side of the pressure stage, is pumped into a remote-mounted filter, into the block through an adaptor mounted to the stock oil filter pad, passes through the engine, down into the pan where it is picked up by the two scavenge pumps, and is returned to the top of the reservoir tank. We deviate from the standard in only one major aspect. Our system filters every drop of oil, there is no operational bypass circuit in either the pressure-side filter or filter mounts. This is accomplished easily with the right selection of components, but can be dangerous if you don't know what you are doing. We like this system, however, because it gives complete protection to the engine. All oil entering the pressure-side of the block has gone through the filter, greatly reducing the amount of foreign material forced through the clearances.

The standard smallblock oil system has a bypass valve built into the filter mount. This valve blows completely open at 12-15 lb pressure. As a consequence, a large volume of oil goes into the engine that has never been through the filter. All of the hot rod remote-filtering systems are built to work with a bypass. Some of them eliminate the bypass at the filter mount, but use a bypass built into the remote mount or use the Chrysler/Ford type filter which has a bypass valve built into the filter.

Most spin-on filters must be used with some sort of bypass system because they can't flow enough oil to handle 100% of the pump output/engine demand. The one exception we know of is the huge 12-inch long AC 832 Chevy truck filter. We have tested this filter and it will handle enough oil flow for any racing smallblock without restricting volume or pressure. With the filter mounted right to the engine we ran one run with the bypass locked open and one run with the bypass locked closed. There was absolutely no difference in oil pressure at the block.

With this filter, building a 100% filtering system is possible. The bypass valve in the stock filter adaptor should be removed and the passage blocked with silicone sealant or a small bolt and washer can be threaded into the opening. It is also possible to use an adaptor like the Transdapt 69018 which replaces the stock filter adaptor/bypass valve assembly. This is a nice piece because it does not have a right-angle bend in the passages to restrict the oil flow, and it will fit early or late blocks. The outboard passage in this adaptor will have to be blocked off as it isn't needed with a dry-sump system. The original feed hole leading from the stock oil pump mounting boss on the rear main cap into the filter mount will also have to be effectively blocked to prevent bleed-off of the entering oil pressure. Weaver also offers a flat plate with a single entry that can be used to route pressure oil into the block.

To hold the AC 832 filter in place we use a Traco remote filter mount. This is the only filter mount we know of that accepts the stock non-bypassing Chevy filter. Finding room for this filter may be a problem, but it does an excellent job as long as water is not



These block adaptors are available from Chevrolet to use external coolers along with the stock filter and wet-sump pump. They sandwich between the block filter pad and the filter. Oil is routed in and out of the adaptor and to the cooler on the pressure side of the pump. Both adaptors retain stock-type pressure relief systems. Number 340258 (left) has forward-facing hose connections and number 326098 (right) has connections facing the side.

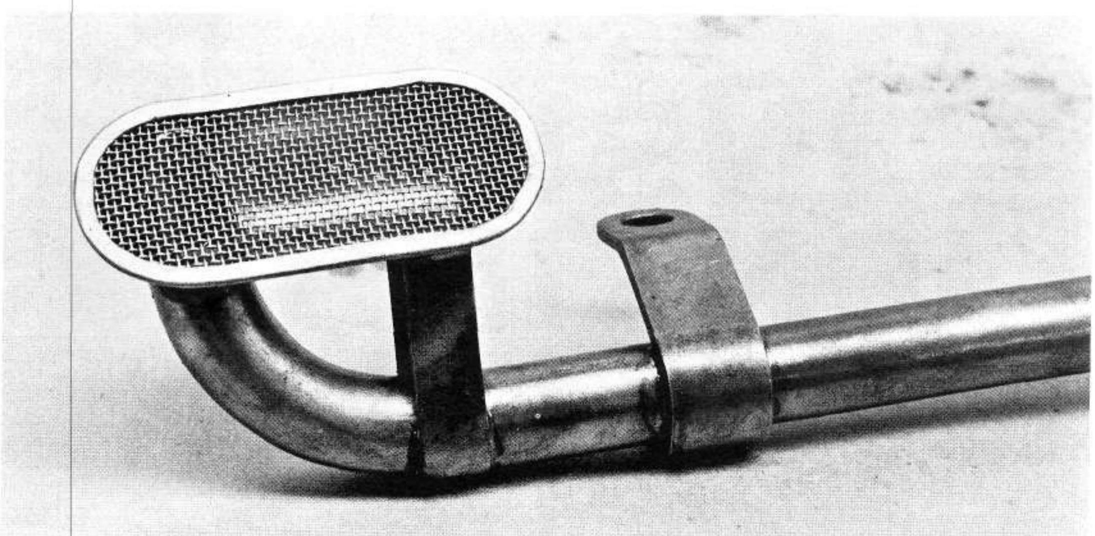
allowed to enter the system. Like all paper element filters, water will clog this filter very quickly and reduce the oil pressure to the engine. As long as the engine oil temperature is kept up to 220-230°, water shouldn't be much trouble, as this is usually enough temp to boil the water off and yet is not so much as to cause oil breakdown problems. Replacement filter elements are available from both AC and Fram.

NASCAR LUBRICATION

The Nascar rules allow for dry-sump oiling systems and most of the successful teams are running variations of the dry-sump theme. At times we have seen some unusual crossbreeds, but we don't think much of them. In one case we tested a superspeedway 354 with a Chevy L-88 mechanical pump as a scavenge pump and a single-stage Weaver external pump to provide pressure. We felt this system was leaving a lot of oil in the engine. In a direct dyno test we replaced this combination with a dry-sump system as described in the drag racing section and "found" 16-18 horsepower. This was a static test and in a race situation some changes may be required in the pan (compared to what is shown here), but there is power to be had in an efficient dry-sump system whether it's on the drag strip or superspeedway.

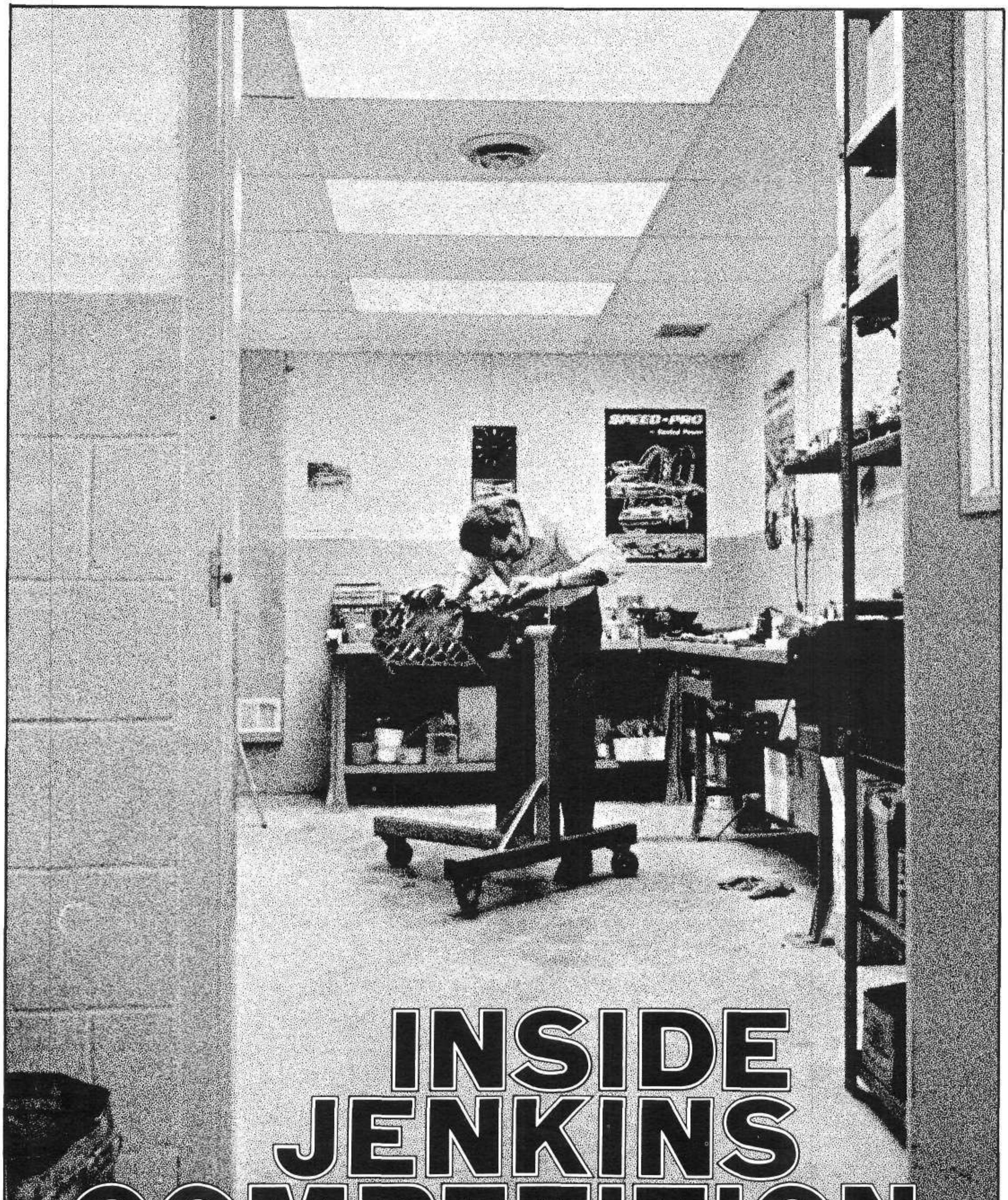
The basic approach is very similar to the drag race systems. The pickups must be located for best scavenging on the outboard side of the turns or a swinging pickup may be desirable if the car is used on a road course. In certain instances the chassis and oil pan design may require a third scavenge stage on the oil pump. The long distance engines will also call for a much larger oil reservoir. Some of the tanks are as large as 25 quarts.

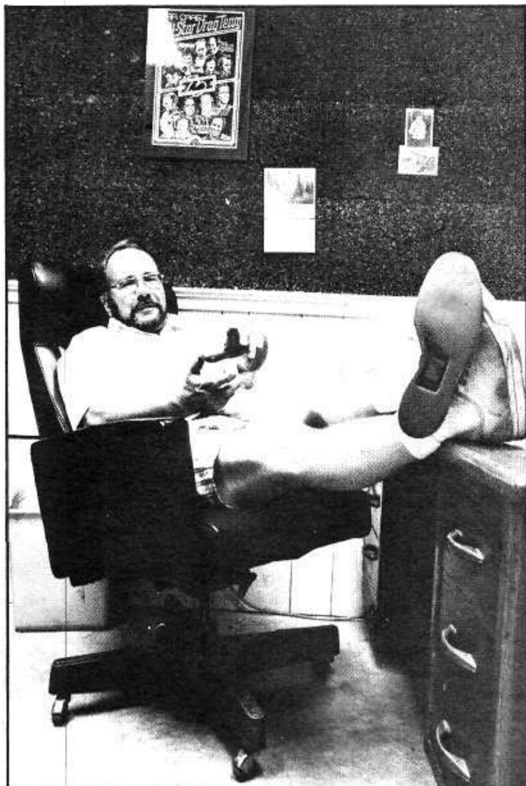
In those cases when oil must be routed through external coolers while the stock wet sump pump is used, Chevy offers two excellent adaptors to make this conversion. The adaptors sandwich between the oil filter and the stock filter mount pad. Adapator number 326098 has side outlets (as may be required to clear obstacles) and number 340258 has the outlets facing forward. Both have provisions for oil bypassing should this feature be desired.

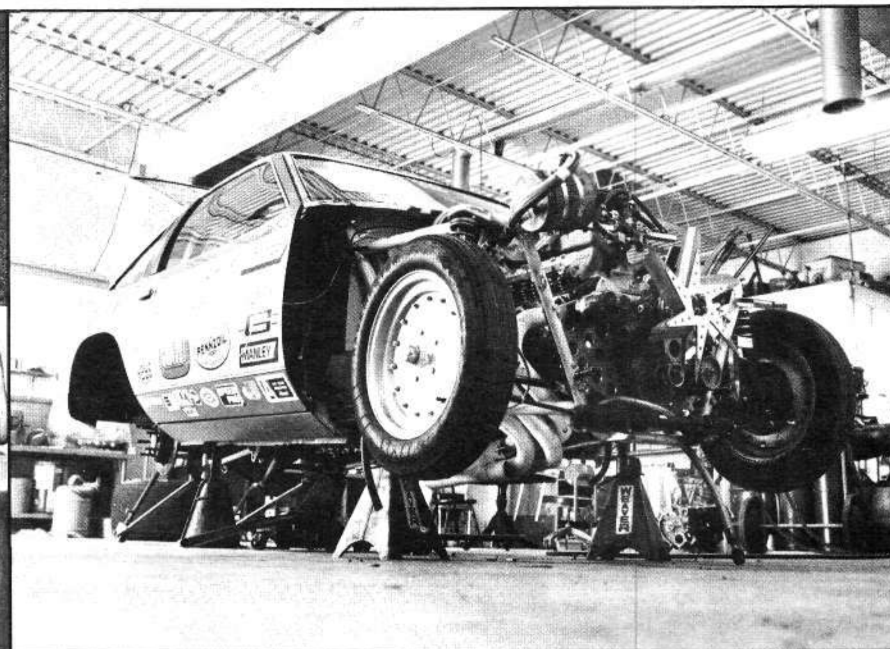


In the dry-sump systems we use either an open end pipe which has been cut off at an angle to reduce vortexing or we build a small pick-up shroud such as seen here. It is similar to a stock shroud and has a screen across the opening to break up vortexing at the entry and to prevent oil aeration as it moves up the pipe.

INSIDE JENKINS COMPETITION



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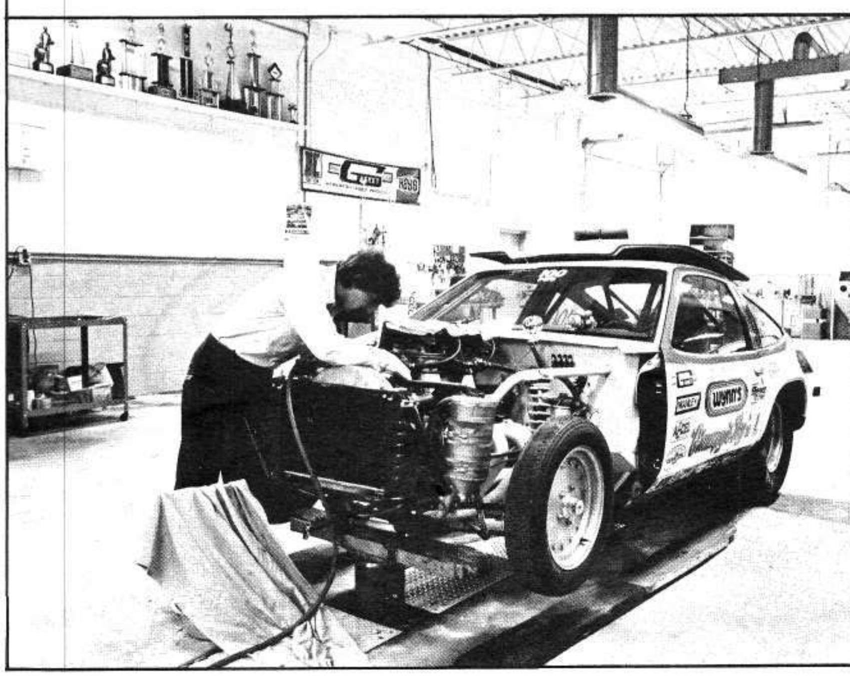
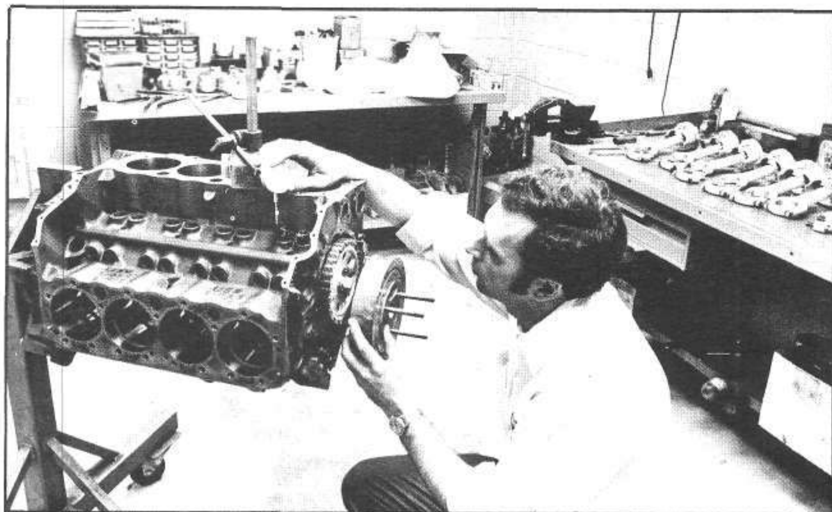
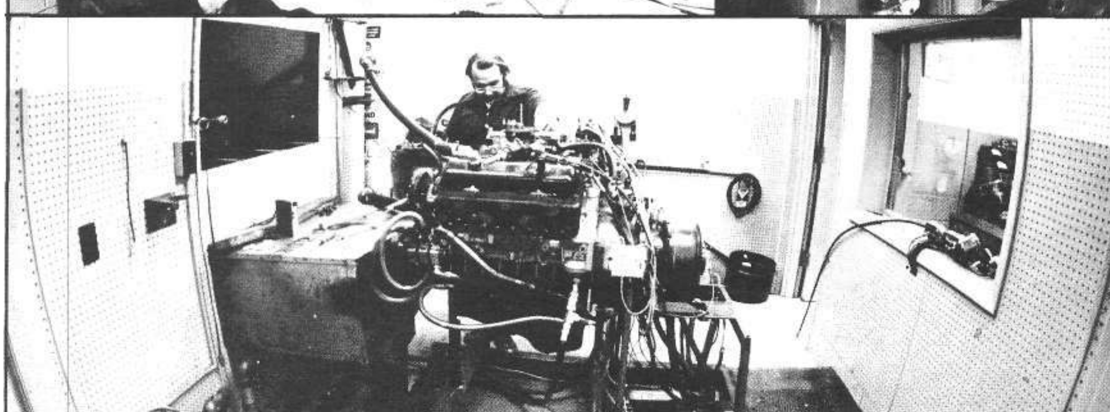


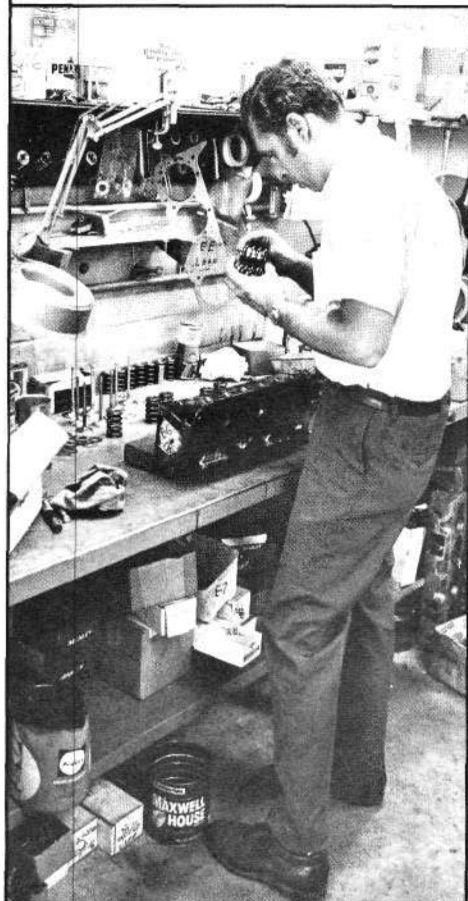


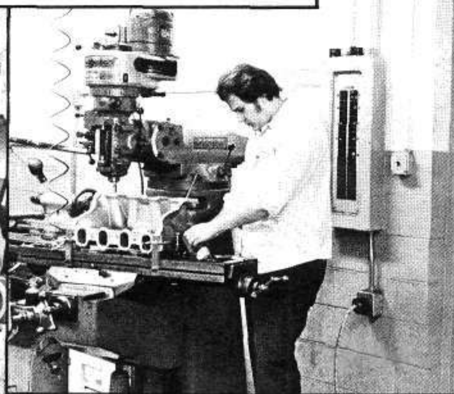
RESEARCH & DEVELOPMENT

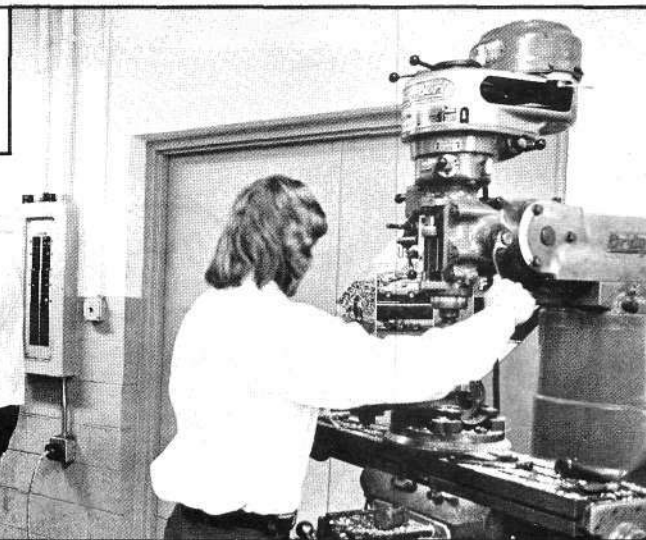


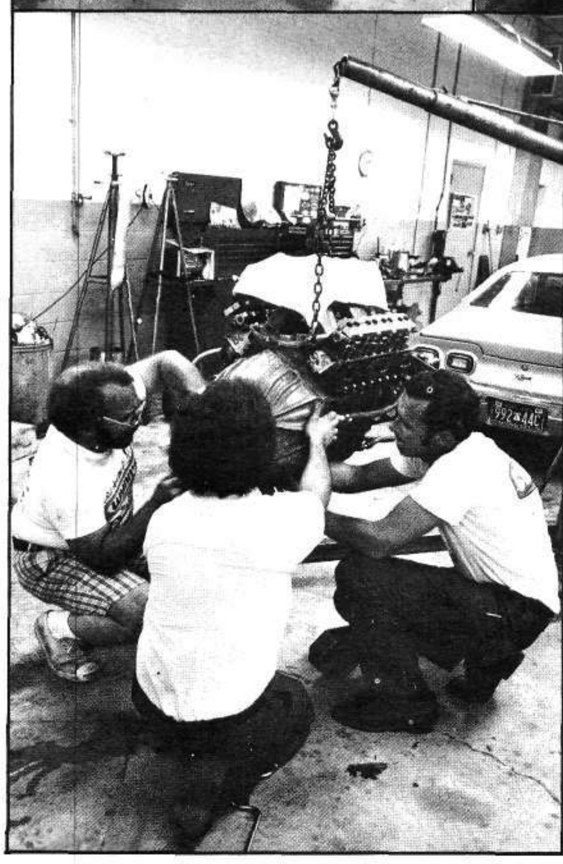


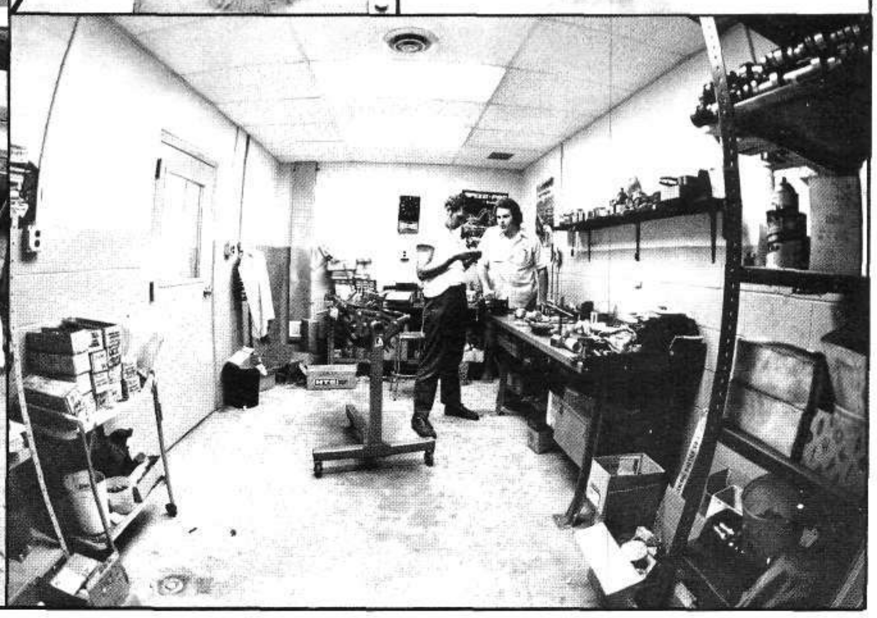

ASSEMBLY

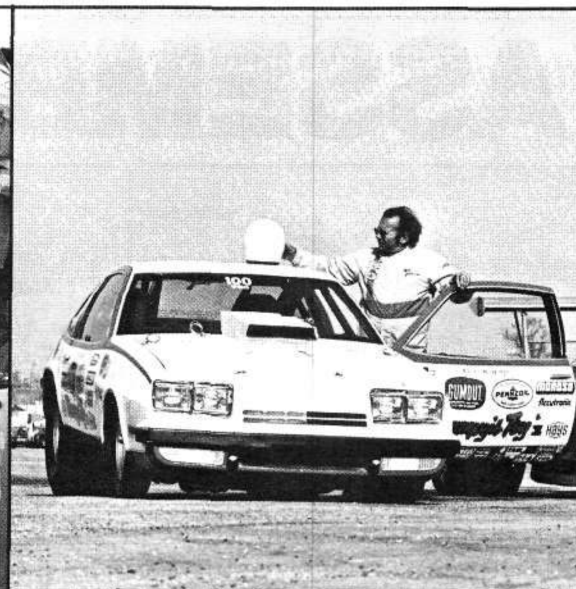


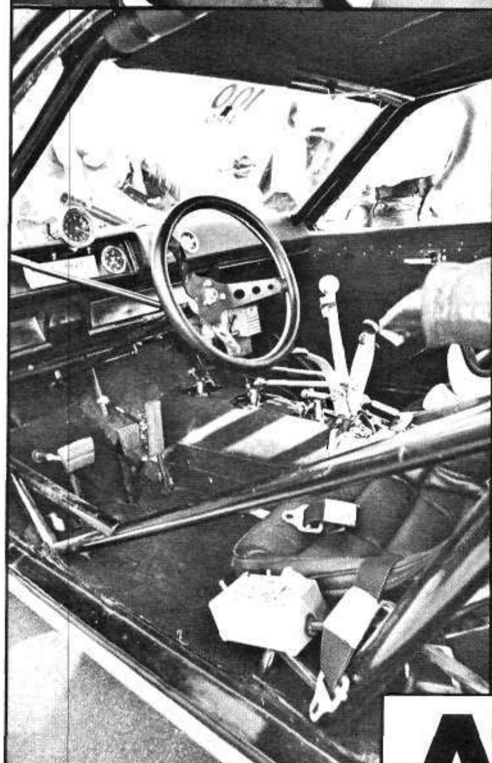




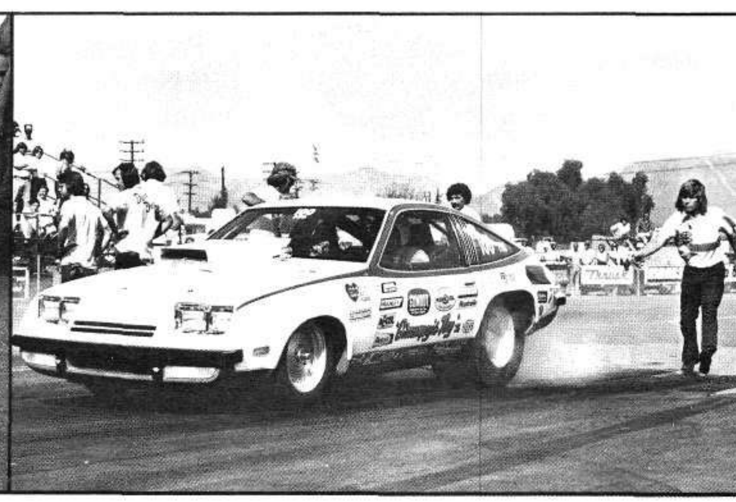


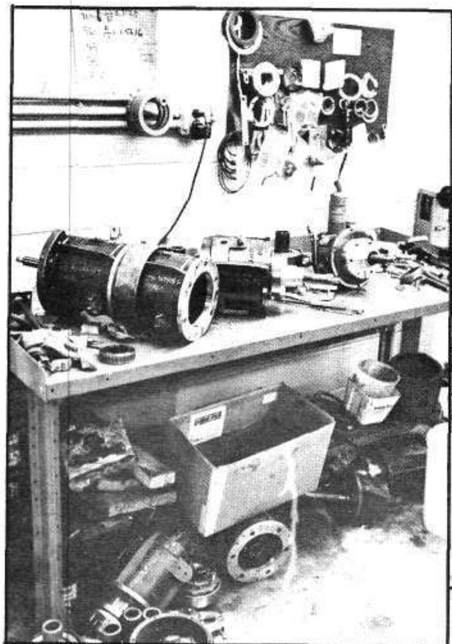


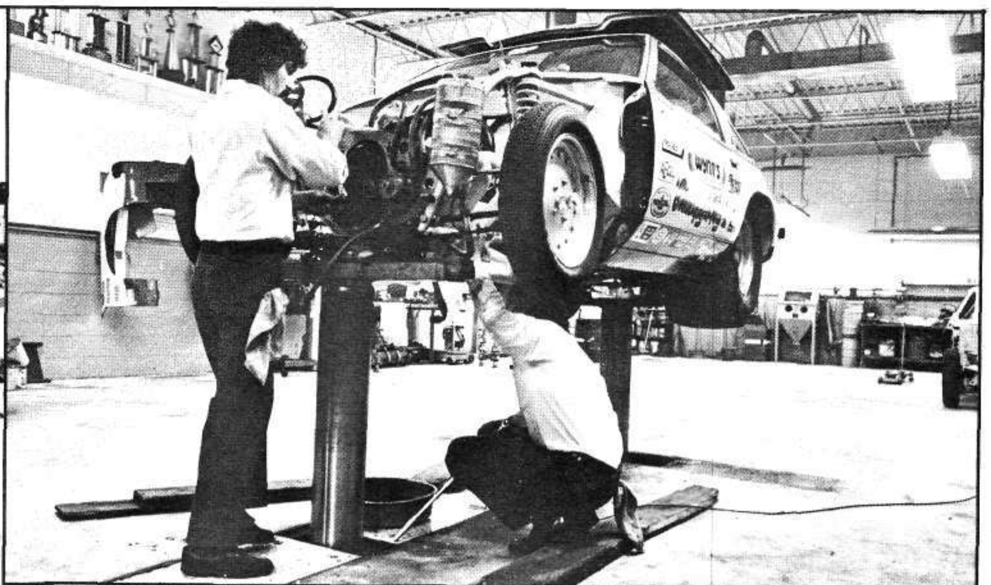


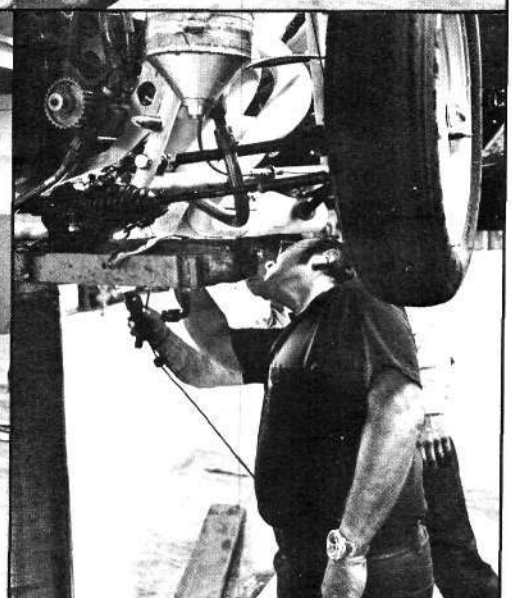


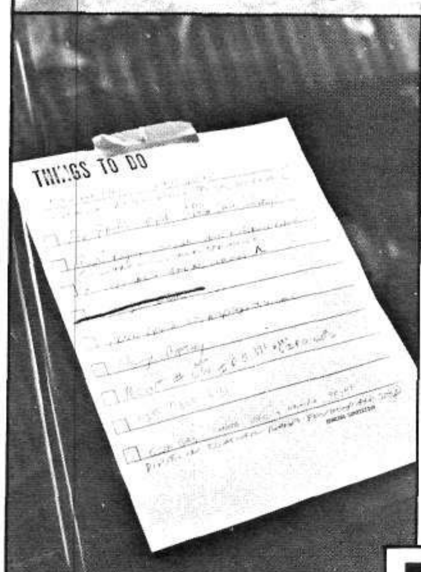
AT THE TRACK

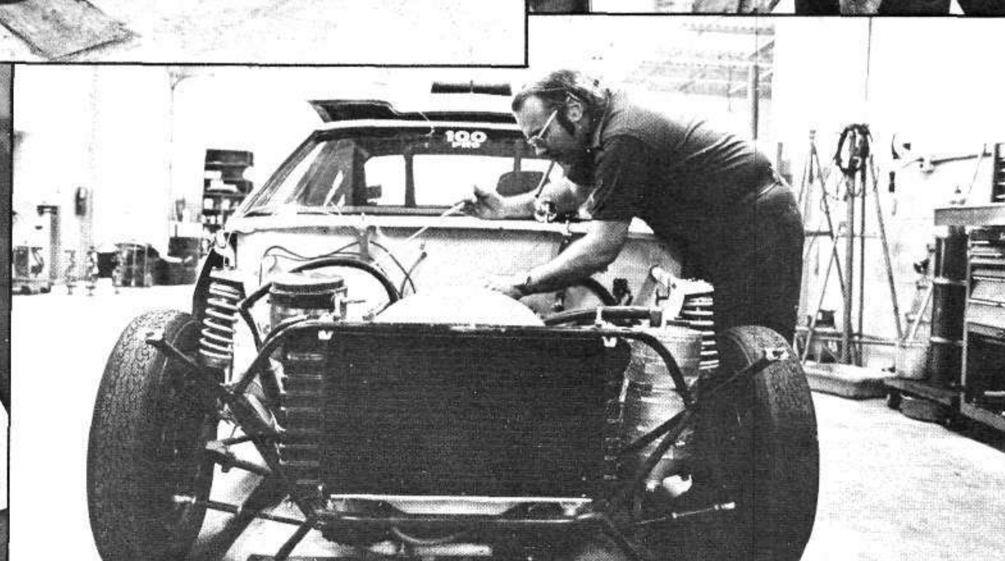












MAINTENANCE