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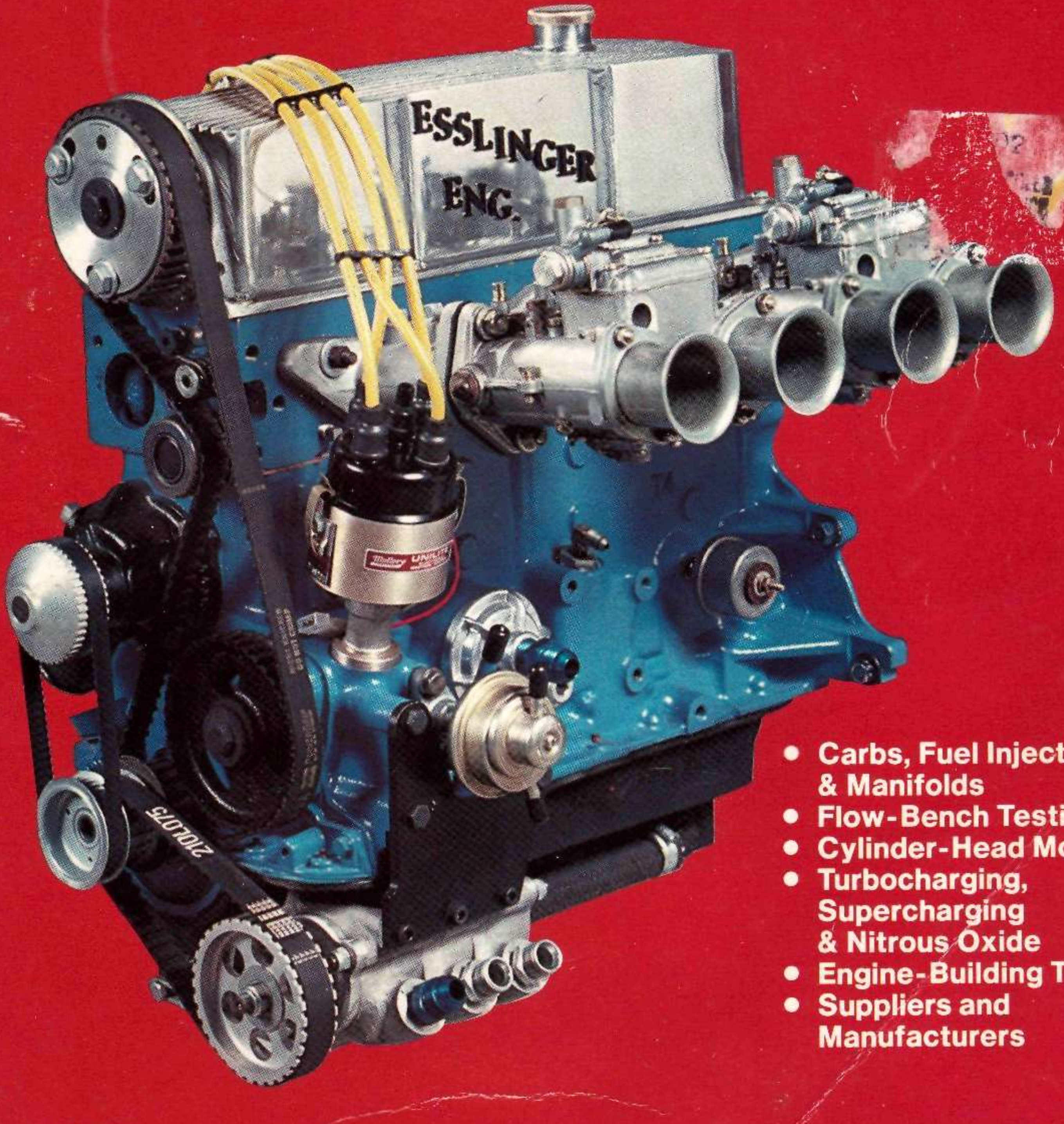

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2.0-LITER OHC FORD

David Vizard

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Contents

CREDITS

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To this I must add a special thank you to Daphne for one year's hard typing.

Introduction

The Ford Motor Company introduced their Single Over Head Cam (SOHC) engine in 1970, and since then they have been installing it in close on a million vehicles a year. This means that by now there are a lot of these engines around.

In Europe, this engine fulfils a number of duties ranging from everyday workhorse requirements to the higher performance needs of Ford's more sporting vehicles like the RS Escort. The same basic engine can displace 1300, 1600 or 2000cc. In some countries such as the USA, only the 2000cc engine was used, this being installed in Capris, Pintos and Bobcats. This is Ford's universal engine, a power unit designed to be put to as many uses as possible.

I had the opportunity to pull one of these engines apart not long after their introduction and from the racer's point of view, I liked what I saw. At the time I was writing a technical chat page for *Cars & Car Conversions* magazine in England and I did not hesitate to extol the virtues of this engine, as well as criticising some of the possible drawbacks as I saw them. I pointed out that this engine had all the ingredients for high horsepower outputs at modest cost. Its overhead cam, eight port crossflow head should, I said, be capable of allowing this unit to produce in excess of 100 bhp per litre in normally aspirated form, and twice that amount per litre in supercharged form. With all it had going for it I expected this engine to be an instant, overnight hit with the speed equipment companies and the general public alike, just as the British Leyland Minis had been eleven years previously.

You don't have to be a student of automotive history to know that my enthusiasm was not shared by all and sundry. After my initial acquaintance with this engine, many years were to pass before one came into my hands again. During those intervening years, one of the great unsolved mysteries of life for me was – why hadn't performance-minded people "discovered" this gem of an engine? True, a few enterprising folk did spend some time making them produce more power but few, very few, remotely approached anything like the true potential of an engine of this configuration. Those clever souls who did manage to produce reasonable horsepower from Ford's single overhead cam (SOHC) engine usually found a very limited response to their efforts by the public.

About 1976 I became re-acquainted with Ford's SOHC engine. I became the proud owner of a MK III 1600 Cortina GT. Although it was not quite the car I expected, I grew very attached to the machine, and as a result I slowly became more involved with the performance aspect of its engine.

As time passed, I began to formulate a theory as to why this engine had not achieved wholesale acceptance by motoring enthusiasts. Ford's European competition involvement is probably well known throughout the free world. Their principal competition engine is the Cosworth/Ford BDA four-valve-per-cylinder engine. This unit is available from Ford at displacements from 1100-2000.cc. Power output in excess of 280 horsepower normally aspirated can be achieved with this engine, though for long distance events such as rallies, 220 bhp is considered the norm. Since Ford already possessed a highly successful engine, it seemed to them to be pointless to develop its poorer, less-endowed cousin the SOHC engine. Having acquired the easy-to-come-by horsepower from the SOHC engine, they appeared, at least to outsiders, to have drawn the departmental line. At the time of writing (1983), if you built an engine using Ford parts, about 155bhp would be the limit you could reasonably expect. I am sure you will agree this is not a lot for an almost bullet-proof SOHC engine. The argument that since Ford has the Cosworth engine in its stable, why should they spend time and money on an alternative unit holds water except for one point: not everyone can afford the price of a four-value-per cylinder engine. On the other side of the coin, a lot of vehicles are already equipped with the SOHC engine.

If Ford has to date, declined to explore the true potential of the engine, what have the privateers done? Many people tried their hand at making the engine go but precious few, it seems, have actually found the key to unlock the true horsepower potential from this unit. You would think that the efforts of those who did make horsepower would be readily received, like exhaust into a vacuum. Unfortunately, this appears not to be the case. The situation seems more akin to a castaway on a Pacific island, reading the national news. Was this apathy on somebody's part? I think not. Ineffective lines of communication would describe it better.

My 'low acceptance' theory is based on what I have already said. It hinges on the fact that for an engine to be an instant performance hit, we must see leadership from the factory to constantly keep the unit in the public eye. Morever, it must be responsive to even the simplest modifications. The factory leadership we don't have, but this situation is changing with the increasing popularity of Group I now group A competition (the use of mass-produced vehicles with factory speed equipment). The engine itself is only semi-reponsive to simple modifications but highly responsive to the *right modifications*. Making horsepower is only a question of finding the weak link in the chain of power production events. Those few who did find the way to high horsepower outputs were not in a position to publicize their findings in a wholesale manner. Instead, the information filtered down through the ranks so that now, a decade or so after it introduction, the engine is only just beginning to achieve the status it deserves. The validity of this theory will be difficult to prove one way or the other but to my mind it contains enough seeds of truth to cause me to restructure the concept of this book. Normally I would start with simple, bolt-on modifications and from there, each subsequent chapter would delve deeper into the engine in the search for more and more horsepower and, of course, reliability.

Not so with this engine.

As I have already said, it is only semi-responsive to relatively simple, conventional modifications. This is part of the reason for its slow acceptance, and I do not intend to be found guilty of further retarding matters. Indeed, my intention is the reverse.

Whether you are looking for a big horsepower increase or a small one, you need to have a reasonable understanding of the device you are dealing with. Hopefully, this book will give you a good insight as to what you can expect the results of any modification to be. It should also allow you to get the best increase in performance for the money you intend to spend. If you are anything like me, economics play a vital role as to what you can do to your vehicle.

Bearing these factors in mind, I intend to go straight into the engine and deal with its idiosyncrasies first. When you have a greater understanding of the engine, we will then deal with the speed equipment, which falls into the more accepted bolt-on category. This and the more complex task of building a competition engine will be dealt with last. In other words I will deal with the sum total when I have dealt with the comprising parts.

How Do We Make Horsepower

I am going to stick my neck out and tell you that really there are no such things as speed secrets. If more power is required from an engine, then improvements must be made in one or more of the following areas: 1. Increased volumetric efficiency; 2. Increased thermal efficiency; 3. Increased mechanical efficiency.

Let's look at each of these three factors in turn and analyse them in a little more detail. First of all, increasing volumetric efficiency. In simple terms, this means improving the breathing efficiency of the induction and exhaust system. When you realize that at 7,500 RPM a typical engine has only six thousandths of a second to fill or expel the gases in the cylinder, you will realize the ease of doing so becomes important. To improve volumetric efficiency, we make changes to air filtration, carburation, manifolds, intake ports, valves, combustion chambers, exhaust ports, exhaust manifolds and silencers (mufflers). Into this cauldron of parts, throw the effect of the camshaft profile on engine performance, and you will begin to appreciate there are a lot parameters affecting the end product. While on the subject of cams, I should point out that high performance cams very often (but not always) increase high RPM breathing at the sacrifice of low RPM breathing. In other words, they trade low end performance for top end. Improving the breathing ability of the engine is the most important single factor affecting power output. Because of this, it's hardly surprising this book deals with various aspects of improving the volumetric efficiency in detail. Pay attention to that detail and you will find the extra power you are looking for.

The concept of volumetric efficiency is relatively easy to understand, but the term thermal efficiency, for many, is not. Let me explain: when a certain quantity of fuel is burnt, it releases a certain known quantity of heat. All forms of energy are interchangeable. If our engine converted all the heat energy to mechanical energy, it would have a thermal efficiency of 100% Remember, the fuel is burnt to heat the air in the cylinder so that it expands and pushes the piston down the bore. The more heat the air contains, the higher the pressure it reaches and the harder it pushes down on the piston. If, after burning the fuel, the heat is taken away from the air, it will not want to expand as much, so cylinder pressures will be lower and the power will be down. Typically 80% of the fuel we burn in our engine is wasted heating up the rest of the world. The remaining 20% is all that is converted to mechanical energy to propel the vehicle. Heat that is dissipated in the cooling system or goes out as hot exhaust is heat that the engine burned fuel to produce and did not convert to mechanical energy. The factors which affect thermal efficiency are important to those of you requiring fuel economy as well as horsepower. The principal factors affecting thermal efficiency are the quality and correct timing of the ignition spark, proper atomization or vaporization of the fuel in the airstream, correct cylinder to cylinder distribution, and correct calibration of the carburettor to deliver the optimum air/fuel ratio. The compression ratio is also a factor. The higher this goes, the better the thermal efficiency gets. Reducing heat losses to the cooling and lubrication system increases the thermal efficiency. Lastly, reduced frictional losses help thermal efficiency, but this really comes under the heading of mechanical efficiency.

As far as mechanical efficiency is concerned, the biggest step you can take to improve it is to build the engine to the finest tolerances possible. Things like con rod accuracy, crankshaft straightness, piston to bore clearances all affect the final frictional losses the engine will have. Care in selecting and establishing the right clearances when building the engine will produce higher mechanical efficiencies.

The overall concept of building a high performance engine is attention to every detail, big or small. In the following pages I will elaborate on the points that were touched upon here. I will give you the necessary details or acceptable ground rules so you can successfully build or modify an engine to your particular needs.

Heads for Power and Economy

Today, the challenge of coaxing extra horsepower from a modern cylinder head produces interesting (!) problems. Gone are the days when a little thoughtful use of the grinder, plus a set of larger valves, were all that was needed to get one step ahead of the competition.

No Sir.

These days sophisticated equipment is required to produce cylinder heads of advanced performance. Foremost among this hardware are the flow bench and dynamometer. But, unfortunately, such equipment is outside the financial means of most enthusiasts.

Fortunately, though, I have over the years acquired such equipment. Thus, in the following pages I will detail certain easier modifications, as well as many more exotic modifications which have all been developed on the flow bench and thoroughly dynamometer-tested. As a result, the changes will help the engine develop the amount of power one would expect from a SOHC, canted-valve engine.

ELEMENTARY PRINCIPLES

The production of horsepower depends to a large extent on air flow. If a head has insufficient air flow, it will *never* produce good horsepower but good air flow or an air flow increase does not guarantee a horsepower increase. Sometimes achieving extra air flow into the engine may upset some other aspect of the engine's functioning, leading to a situation where little or no gains are made. A rule which works almost 100 percent of the time is *if air flow increases and nothing else changes, horsepower will increase.*

Other factors affect horsepower apart from air flow, these being principally fuel atomization, combustion efficiency and heat losses. To optimize the cylinder head, attempts must be made to impove all these areas.

There is no doubt that cylinder head design is a very complex subject. It is often regarded by laymen as being a black art. As a result, many myths exist, and one of these is that polishing the ports is the trick to make a head work. Nothing could be further from the truth. *A polish does nothing to increase the power of an engine.* In fact on occasions it can reduce horse power.

Another myth you should dispel, especially with the Pinto, is that big ports produce flow. They do not; an excessively large port will sometimes flow less air than a smaller one.

The most important factor in cylinder head development is shape. This is the most important consideration with any cylinder head modifications. The shapes of the ports, combustion chambers and valves dictate just how effective that cylinder head will be. If you intend to grind your own cylinder heads, don't worry about a polished finish. A rough-ground finish is usually entirely adequate.

THE STANDARD HEAD

The first step toward improving a

FIG 2-1A

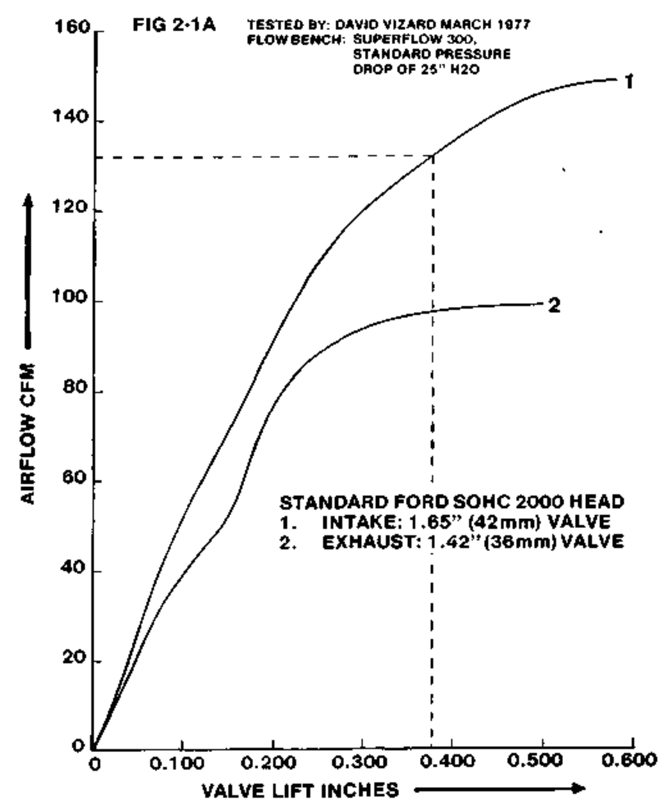
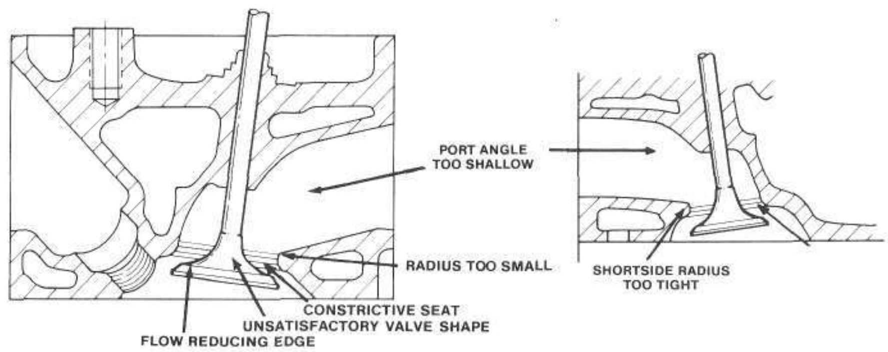
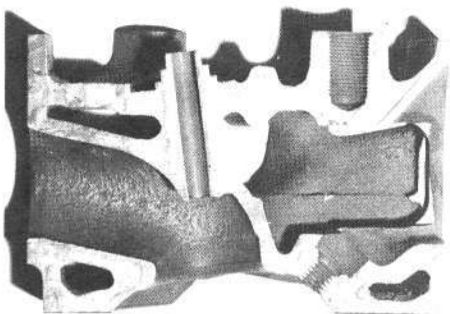


FIG 2-1B

EXHAUST PORT





2-1. A section through an unmodified 2000 head reveals that if nothing else, the intake port has plenty of area.

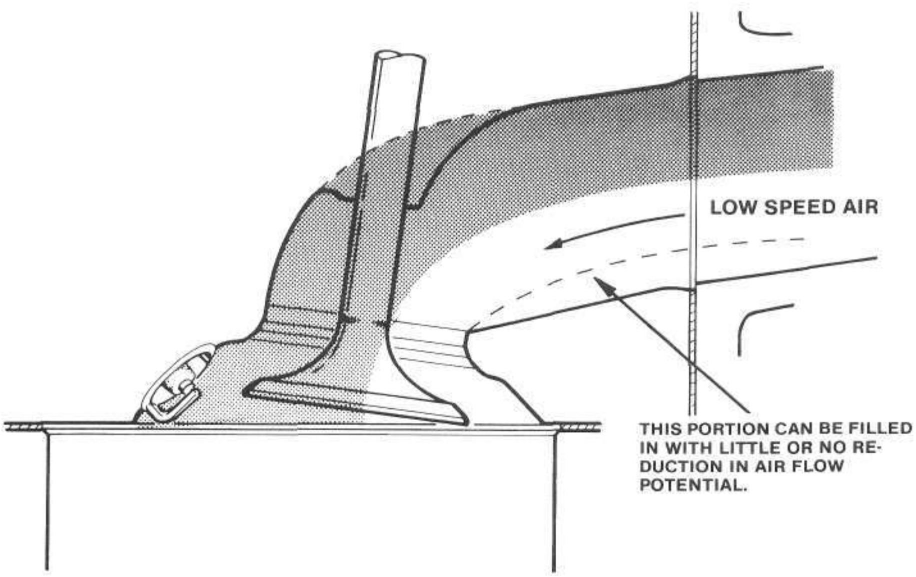


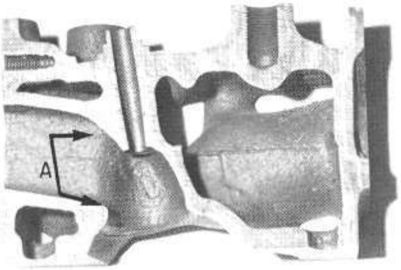
FIG 2-1C AIR FLOW PATTERN IN STANDARD INTAKE PORT

mechanical contrivance is to understand the nature of the device. To do this let us analyse the standard head in a little detail. Take a look at the graph Fig. 2-1A. This shows the amount of air that can be passed through the standard ports on a 2000 head. The dotted line starting on the horizontal axis of the graph is a typical maxium lift achieved with a standard camshaft. Follow it up until it meets the inlet flow curve. Then turn 90 degrees and follow it to the vertical axis. This indicates that the head flows 131.5 CFM at 375 thousandths lift. For a 1.65 inch diameter valve at this lift, this is not a very good showing. In fact the head, for all its promising looks, does not deliver the goods. As it comes from the factory it falls short on many counts as far as air flow and its power potential are concerned.

First of all, the valve seats, especially on the intake are very constrictive to flow. Secondly, the shape of the inlet valve is far from optimum. The port angle in the head is also disappointing, because it closely approaches the worst angle possible for flow. And last, the final approach to the back of the value is too short. In other words there isn't enough length of straight port behind the valve head to allow the air a more direct shot to the back of the valve. Interestingly, 1600 and 1300 heads don't suffer in this respect quite as badly as the 2000 head. Both 1600 and 1300 heads have shallower chambers and longer valves. This means the air can make a more favourable approach to the back of the valve, especially on the floor of the port around the tight turn just upstream of the valve seat. Fig. 2-1B shows the main restriction points. Apart from its breathing ability, the inlet port does suffer one other ailment: the port itself appears to be too large for the engine. The resultant slow gas speeds allow fuel to drop out of suspension easier than if the port were smaller. A study of the air flow pattern in the port reveals that most of the air flows at the top of the port, and the bottom of the port is almost redundant. Fig. 2-1C shows what I mean. Flow bench tests show that if the bottom of the port is filled in as much as ¼ inch, *almost no drop in air flow results.* Any modifications to this engine must be done bearing in mind that *fuel dropout* can occur. When fuel dropout takes place the engine will suffer a loss of horsepower, economy and throttle response. Any changes which help produce a more homogenous mixture entering the cylinders will usually improve the engine's performance in these areas. Straight away this should tell us two things about these heads: enlarging the ports is definitely out, and polishing them is not a good idea because a coarse finish is more likely to reintroduce puddled fuel back into the airstream. A shiny finish will allow fuel to stay on the surface as a liquid or drops which will run into the cylinder and subsequently pass through the engine unburnt.

EXHAUST PORT

The exhaust port suffers many of the

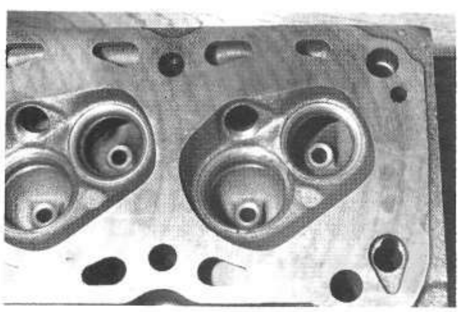


2-2. Too many abrupt changes in direction mean bad flow through the standard exhaust port. Arrows indicate the prime sources of inefficiency.

same ailments that the inlet port has. The valve seat geometry needs improvement. On the other hand, though, the exhaust valve shape, unlike the intake, is reasonably satisfactory. The exhaust port shape, however, is even worse than the intake port; its flow figures are way below those attainable by a highly developed port.

COMBUSTION CHAMBER

I am not suggesting this head is junk. Far from it. The factory designed the head to do a particular job, and this it does in a satisfactory manner. They did not design it for racers and for high performance it will need some redesigning. You may well ask if there is *anything* that doesn't need modifying? Certainly. The combustion chamber is a very good design and has many factors in its favour. Many production cylinder heads suffer from what is known as valve shrouding. This is the situation where, as the valve opens, the gap between the edge of the valve and the combustion chamber wall is insufficient to allow air out. On vertical valve engines, the cylinder bore will almost always cause shrouding. Although the Pinto engine has inclined valves, the inclination is not sufficient, nor is it in the right direction to obviate shrouding. Unnecessary shrouding by the chamber though, is virtually non-existent. The only shrouding suffered is caused by the proximity of the cylinder walls and there is little we can do about that.



2-3. Valve shrouding caused by the unnecessarily close proximity of chamber walls to valve heads is all but non-existent on the standard cylinder head.

IMPROVING THE HEAD

The prime factor of an engine's power characteristics, be it a well developed unit or not, is the cylinder head. Because of this, I will deal in depth with head modifications and how they affect airflow. I will show precisely what modifications are needed to produce the required results.

Improving airflow is finding the right combination of shapes. Two head modifiers starting off with the same basic head casting may arrive at two different combinations of parts and shapes to produce compatible airflow figures. And importantly, trying to combine the specifications of one with the other may well produce worse airflow figures than you started off with. A more precise example: let's say that somebody develops a trick valve shape which really turns the flow on with a standard port. There is no guarantee such a valve is going to work in the same manner if the port is steeply downdrafted. It all comes back to *combinations* of shapes. In other words, don't reckon on producing a super trick head by using what may appear to be the best points of several different heads. More than likely, whatever you do will be worse, as this is not an easy head to improve upon. The only way to find out what will work for sure is by testing on the flow bench and dynamometer.

SIMPLE MODIFICATIONS

My policy, when modifying an engine, is always to try the simple modifications first in an attempt to extract the most for the least. In the category of simple modifications, we have such things as multi-angle valve seat jobs, removing sharp edges from valves, plus going into the port with a grinder and just taking off any flash marks or sharp edges produced from machining in the port. Well, I have news for you; the 2000 head does *not* often respond positively to such moves. In fact, the first week I spent on the flow bench with one of these heads produced a large number of negative results. Whoever designed this head, made it so you have to fight all the way to get any flow increase.

Let's consider those multi-angle valve seats that are often touted as the trick thing for a few extra horsepower for next to nothing. I tried such valve seats on brand new heads, and believe me, the usual result is reduced airflow compared with the standard 30, 45, 60° seat that Mr. Ford puts on the head. This doesn't mean we can't improve the head; it just means that an elaborate 75, 60, 45, 30, 15 seat is not what's needed. Take a look at the chart (Fig. 2-2) and you will see the typical difference between a standard valve seat and a multangle one. To be truthful, taking the sharp edges off the valve makes things even worse. If you do any grinding on the back of the valve, it must be a substantial 30° cut as shown in Fig. 2-3. For such a simple modification, as you can see from the graph, the flow was substantially increased. This led me to test many valve shapes to determine which

FIG 2-2

Comparison between standard valve seat and multi/angle seat.

Valve Lift In Inches	C.F.M. Standard	C.F.M. Multi/Angle
0.025	12.0	11.4
0.050	26.6	25.9
0.082	44.8	41.6
0.165	76.1	72.9
0.247	108.3	110.8
0.330	125.7	128.7
0.412	137.6	134.6
0.495	143.4	140.6
0.577	147.3	144.5

NOTE: Standard width valve seat used in all tests. Test pressure drop 25" H_2O.

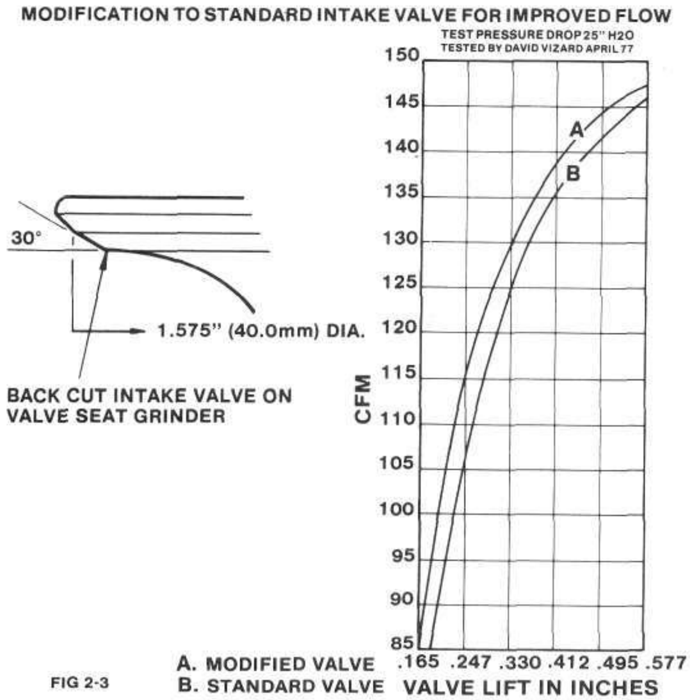
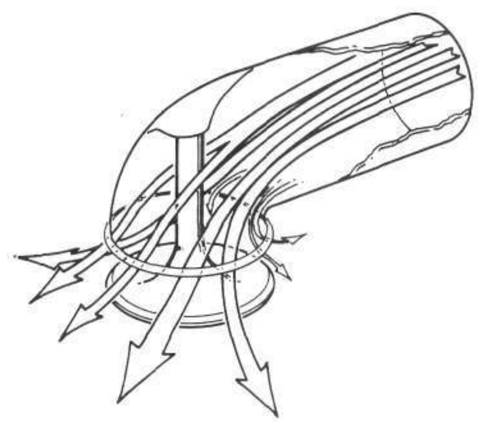


FIG 2-3

FIG 2-4
THIS SHOWS DIRECTION OF THE MAIN MASS OF AIR IN A 2000cc HEAD WHEN THE VALVE IS AT MID OR CLOSE TO FULL VALVE LIFT.



profile would be the best for this particular port. Flow testing quickly established that the flatter the profile on the back of the valve and the thinner the stem, the more air the port would flow. Velocity probing the port quickly showed the reason the flat valve worked best. If you look at Fig. 2-4 you will see that the airflow pattern is across the back of the valve, rather than in a downward direction toward the valve head and around its sides. This shows that most of the air entering the cylinder exits the port on the plug side of the valve; and little air exits the port on the opposite side. A valve with a large tulip section on the back presents a considerable obstruction to the airflow. A flat valve presents less obstruction and as a result mid-lift airflow is increased considerably as Fig. 2-5 shows. This trend in valve shapes predominates throughout all flow testing, where the basic port angle remained unchanged. In other words, unless we put in a new set of steeply downdrafted ports, the flat valve profile is the shape to have.

FIG 2-5

Valve Lift In Inches	C.F.M. Standard Valve	C.F.M. Flat Back Valve
0.025	13.3	12.9
0.050	27.1	25.9
0.082	42.6	42.5
0.165	74.3	89.1
0.247	108.9	119.6
0.330	127.5	135.6
0.412	135.6	142.6
0.495	145.7	148.5
0.577	148.9	152.5

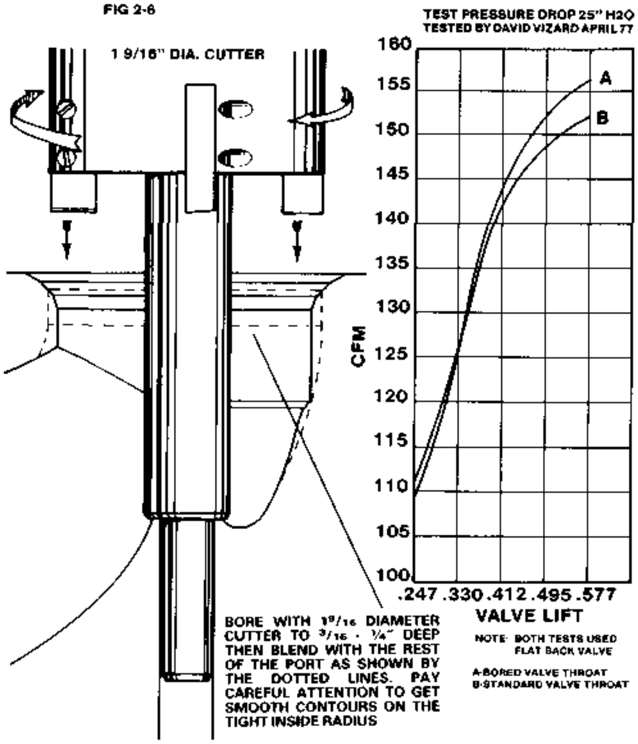
Test pressure drop 25" H_2O.

VALVE SEAT AREA

Any competent head modifier will tell you the most important part of any head modification is the area 1/4 inch before the valve seat to 1/4 inch after it.

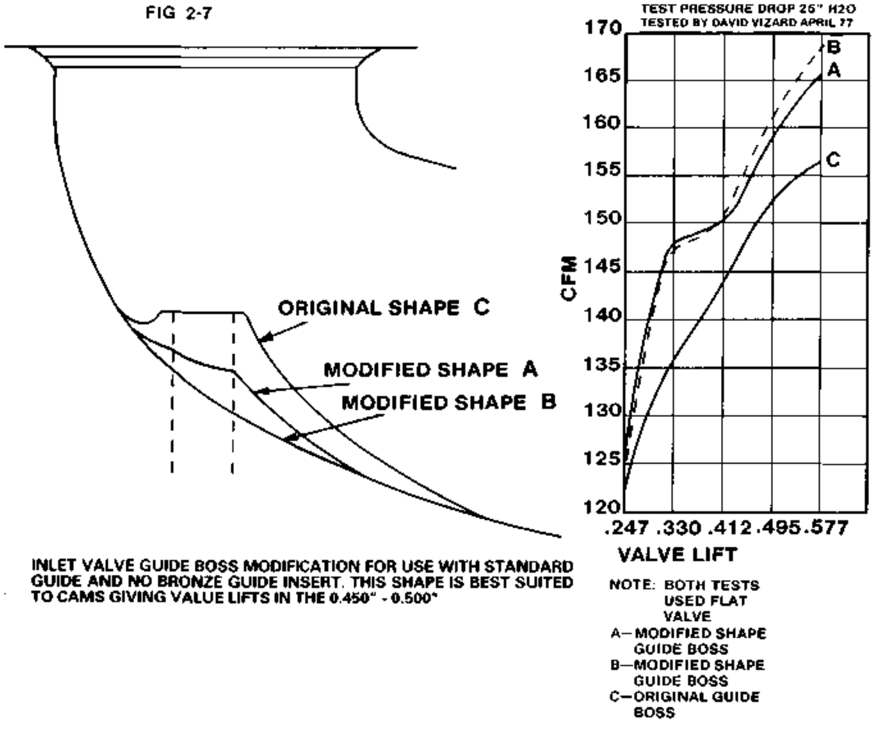
Because the multi angle valve seat proved singularly unsuccessful, a new line had to be tried. The obvious thing was to bore out the throat of the intake port. This would give a useful increase in the breathing area. Such modifications don't always result in extra flow because very often the efficiency of the port may drop due to the abrupt changes in shape in the vicinity of the seat. However, in the case of the 2000 head, this modification did work out. The easiest way to do this modification is to take a cutter of 1-9/16 inch diameter and bore the throat down to a depth of 1/4 inch or so. This leaves you with a seat width of approximately .070 inch, which is ample for the job. The step produced from boring should be blended into the rest of the port. After boring the port, it will be necessary to remove the sharp edges from the inside diameter of the seat. A needle file can be used for this, or a quicker job can be done by taking off the sharp edge with a 75° grindstone. The amount of chamfering required at this point is minimal. If you

FIG 2-6



MODIFICATION TO THE VALVE THROAT 2000CC ENGINE FOR INCREASED AIR FLOW

FIG 2-7

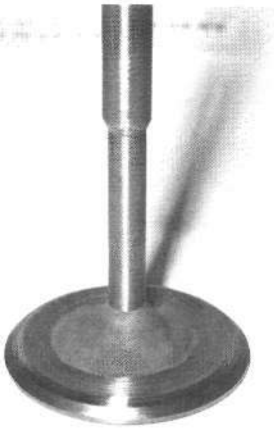


INLET VALVE GUIDE BOSS MODIFICATION FOR USE WITH STANDARD GUIDE AND NO BRONZE GUIDE INSERT. THIS SHAPE IS BEST SUITED TO CAMS GIVING VALUE LIFTS IN THE 0.450" - 0.500"

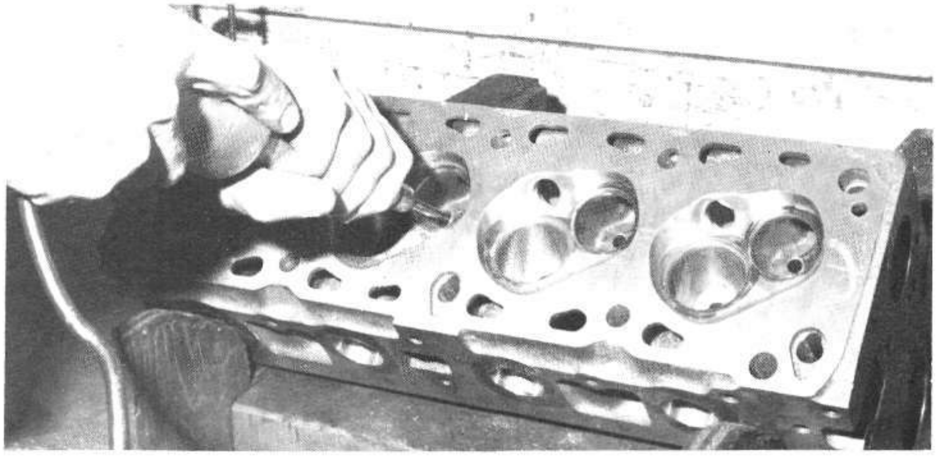
have broken the corner with a .010 inch wide cut, that is sufficient. The resultant increase in airflow is shown in Fig. 2-6. At this point, compare these figures with graph showing flow performance of a flat-back valve. All these modifications work using a more conventional valve with the back cut at the 30° angle described earlier. The proportional gain is similar but the results achieved with the more conventional valve shape are less than those achieved with the flat-back valve.

Moving farther down the port brings you to the guide boss. Here you must make a decision. The guide boss causes a flow restriction. If you are buying a head and you are paying big money for a so-called trick head, *be very wary of one that still contains the guide boss*, as the head will flow more air with the guide boss cut away. Just how much cutting away needs to be done depends on the lift of the cam you will run in the engine. Look at Fig. 2-7 which compares the relative flow increases for the original and two modified shapes in the guide boss area. Note that with no guide boss there is only a substantial increase in flow at lifts over .412 inches. It is unlikely you will see any benefit by cutting the guide boss completely away, unless you are running with a cam of at least .500 inch lift. Importantly, cutting away the guide boss does bring about its own problems. Because the guide length is shortened, valve guide wear increases drastically. The trick approach to running short guides without sacrificing guide life is to install bronze guide inserts. In the U.S.A., K-Line thin-wall bronze guide liners or Winona helical bronze inserts are commonly used. In Europe, especially in England, bronze guides are more commonly used. If you don't want to go to the expense of bronze guides, you can still shorten that guide boss and make an appreciable gain in flow. Shape A of Fig. 2-7 still gives sufficiently long guide life for most applications but heed a word of warning: the higher lift cam you use, the shorter guide life will be. If you are using a stock cam, modified shape A will probably be good for around 50,000 miles. As cams approach 0.500 inch lift, you will find guide life reduced to as little as 25,000 miles. Consider this guide life before you decide whether or not to use a bronze guide and the expense it involves.

Moving on to the parallel part of the port, you will find that this is critical. It is already too big for best results on the dyno; do not attempt to enlarge it. Also, don't waste time giving it a super high-polish finish. In fact, you can usually get the best results by *not* touching this part of the port. The main thing to remember when you are modifying the cylinder head is that it's shape that counts more than finish. The shape can make hundreds of percent greater difference in flow than polish. All your efforts at this stage should go on careful shaping in the area around the valve seat and the guide boss. When you've gone that far, stop.



2-4. This flow bench model valve gave almost as much airflow increase in a standard port as a complete porting job gave with a standard valve.

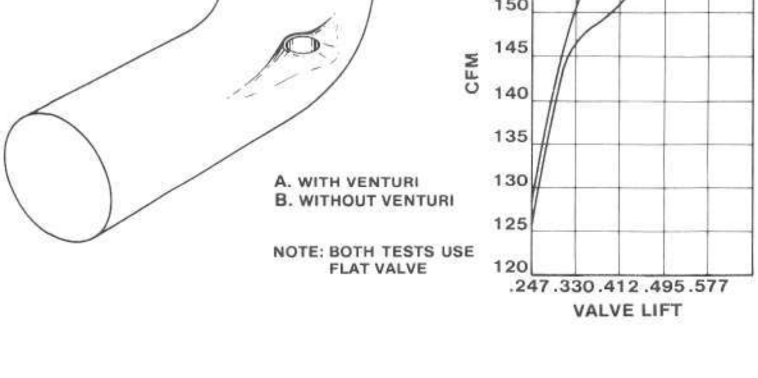


2-5. This head is being prepared at Shankle Automotive. A lot of attention is being given to the shape of ports and chambers. However, a chrome-like polish is nowhere to be seen. This is as it should be.

INTAKE PORT DEVELOPMENT

So far, simple modifications to the intake port have been considered. Now let's begin to get a little more exotic while retaining the standard size valve. If you are good with your cylinder head grinder, a venturi around part of the valve seat will pep up the flow. This works with either a re-profiled standard valve or a flat profile valve. Such a venturi shape will not generally increase peak flow but it certainly does help in the all-important mid-range flow. Remember, the valve reaches full lift only once in the lift curve but it reaches half-lift twice. With the venturi, substituting the conventional shaped valve for the flat profiled valve pays off with even greater flow. See Fig. 2-8.

FIG 2-8 INCREASE IN AIR FLOW WITH VENTURI VALVE THROAT

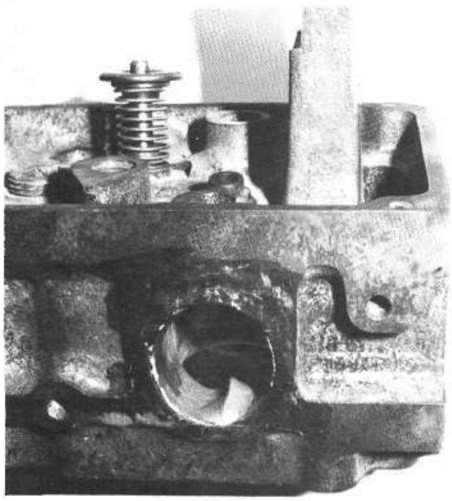


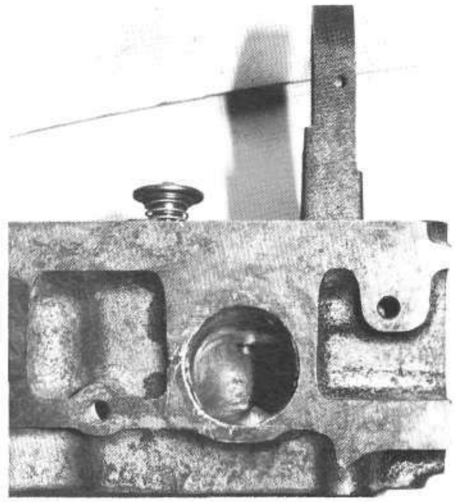
INCREASING PORT FLOW AND VELOCITY

So far, all the re-shaping we have considered has been done by removing metal only from the port. But let us consider this problem of the too large a port diameter. We can largely overcome this by adding material to the port, which on the inlet side is not too difficult to do. A material or bonding agent with metal in it, such as Devcon or an epoxy resin such as JB Weld, makes a good job of putting material back where it's missing. The point is, where should we add it and what sort of velocities do we need? If a comparison of the air velocities in the port of a 2000 engine is made with, say, a small-block Chevy, we find the 2000 head yields only about two-thirds the velocity. I will not spout lots of different numbers suggesting that you should have this or that port velocity. Port velocity is a subject which is very much misinterpreted. The thing to consider most when juggling with port velocities is that the prime objectives are to achieve a higher and more even port velocity so fuel stays in suspension better and increased inertial ramming of the cylinder is achieved. Dyno testing indicated that reducing the port size while retaining the same airflow in cubic feet per minute gives a worthwhile increase in horsepower. This is especially so if intake charge heating as is often encountered on factory manifolds is not present. Even if intake charge heating is present, we can still come out on the winning side with more power and economy by not aggravating fuel dropout.

Let me give you an example: Some time ago, I was testing an engine with engine builder David Ray. This engine had a very high specific fuel consumption. Its thermal efficiency was low. The head had fully re-worked and highly polished ports. This head was then substituted with another head with a rough intake port. Only the area in the immediate vicinity of the valve had been reworked. Airflow of both heads was about the same. The rough port head produced its best horsepower with a secondary barrel main jet *two* sizes smaller than required for maximum power jetting for the polished port head--this engine produced 1.5bhp more! Summing up: the rough-port produced *more* horsepower on *less* fuel.

Adding material to the port to increase velocity is all very well but the





2-6 & 7. The "ramp port" was a successful experiment as far as airflow and gas speed were concerned, but its shape was rather too critical to produce even on a four-off (one per port) basis.

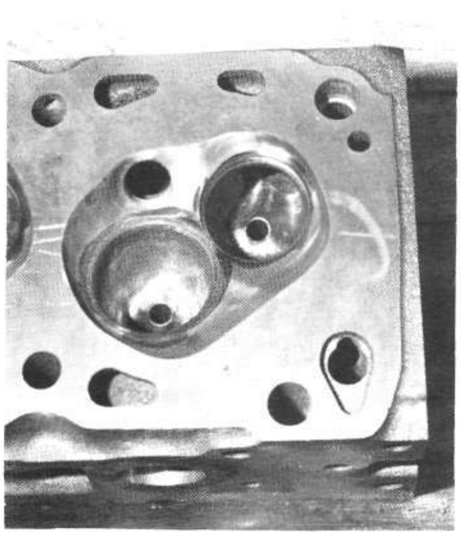
FIG 2-9
Comparison of conventional port shape V ramp port.
Tested by David Vizard Sept. 77

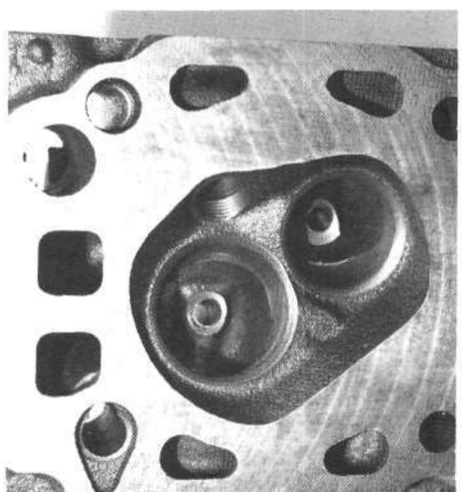
Lift Inches	Air Flow C.F.M.	
	Conventional Port	Ramp Port
0.082	41.7	42.7
0.165	86.9	87.3
0.247	125.9	125.6
0.330	146.9	149.4
0.412	151.1	161.4
0.495	159.7	172.3
0.577	168.5	177.2
0.650	173.2	180.8

Standard pressure drop 25" H_2O.
Valve Dia 1.65" (42 mm) with flat profile.

problem is, where to add it? Making the port into a D-shape by adding material to the floor will decrease the port size and gain the higher gas velocity required for better fuel suspension without significantly affecting the airflow. Usually, only a very small reduction in airflow occurs. However, it would be much better to see whether material could be added so as to increase airflow as well as increase port velocity. To do this we need to plot out where and in which direction the air is moving in the standard port. The drawing on an earlier page tells us most of the air is travelling along the ceiling of the port when the valve is at working lifts. And, as a result, air flowing at the top of the port makes good use only of the far side of the intake valve to feed the cylinder. If the core airflow can be moved so that it utilizes a greater amount of the intake valve, an increase in both port velocity and airflow are likely to result. Many hours experimentation were done along this line and it resulted in a port which, for want of a better name, I call a *ramp* port. With the ramp in the intake port as shown in the photographs, airflow was increased as shown in Fig. 2-9, and port velocity was evened out to a much more consistent figure. The curve on the ramp also tends to direct any fuel that is running low down in the port into the centre of the cylinder, rather than onto the cylinder wall where it is unlikely to burn. The addition of the ramp produced a port velocity which fell in line with many engines known to produce good specific fuel consumption figures. Making this ramp is a difficult job. To begin with, a rough form of the ramp needs to be added to the port, followed by the final shaping, which must be done in conjunction with the flow bench to make sure it's working right. It is not really a job for the amateur head modifier. Fortunately, the ramp does not need to extend upstream into the intake manifold. I tried this with a 45 Weber manifold and all I succeed in doing was reducing the total airflow.

My advice: if you intend to build a ramp port, do it as shown in the photos and leave it at that. To make sure you don't have some Devcon or JB Weld disappearing into the engine, be sure when you use it that the port is totally oil-free. Also grind some grooves in the

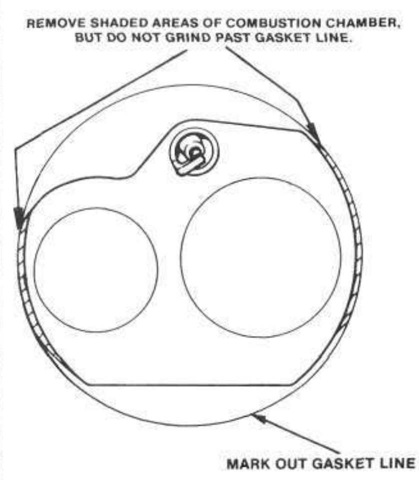




2-8 & 9. ONLY if larger valves are used does the combustion chamber require any significant reshaping. Even when ground to suit a larger valve, the chamber still retains a similar appearance to the standard shape. This photo of a virtually finished big valve chamber, demonstrates the point, when compared to the standard chamber.

FIG 2-10

SIMPLE COMBUSTION CHAMBER MODIFICATIONS



base of the port for the Devcon or JB Weld to grip securely. An engine which swallows a lump of the intake port isn't going to produce as much horsepower as it should.

COMBUSTION CHAMBER

As long as the standard intake valve is used, the combustion chamber suffers no unnecessary valve shrouding, apart from that caused in the area adjacent to the cylinder wall. Here, you should lay the head gasket on the cylinder head and mark out where the gasket comes to. Then cut the combustion chamber wall to that line. This will give the air the maximum chance of coming round on the chamber wall side. Fig. 2-10 shows where you should grind.

The same situation also applies to the exhaust valve and the same technique can be used to grind the combustion chamber to the limits of the gasket to relieve the exhaust valve of any unnecessary shrouding.

COMPRESSION RATIO

Some easy increases in power can be achieved by raising the compression ratio. These engines are available in a variety of compression ratios. Those with ratios in the range of about 8:1 respond very favourably even to only small increases in compression. As a rough guide, whichever engine you have, 0.045-0.050 inch (1.15-1.27 mm) off the head raises the compression ratio about ¾ of a ratio point, and 0.060-0.065 inch (1.52-1.65 mm) will bring it up about one whole point. Typically, these engines will respond with about a five-to-seven-percent increase in torque and horsepower. Because the power increase is brought about by more efficient use of the fuel, the economy is also increased.

In some areas of the world, such as the U.S.A., the octane rating of available fuels is little better than that of peanut butter. Once compression ratios start exceeding about 9.7-9.9:1 the engine is likely to run into detonation problems, even on premium-grade fuel. Where 100 octane fuel is available, compression ratios up to about 11.5 or 12.1 can be used.

As compression ratios increase, less metal has to be taken from the head to raise it a corresponding amount. For instance, if it took .065 inch to raise the compression ratio from 8.3 to 9.3, it will probably only take another .040 inch to raise it from 9.3 to 10.3. About the maximum that can be machined from a 2000 head is 0.165 inch. For now, we are dealing only with minimal compression ratio rises to obtain cheap and easy horsepower increases for normal street purposes. Later, we'll discuss the matters of maximum compression for maximum power.

From the factory, the compression ratio on these engines is altered through the crown height of the piston. RS 2000 engines already have a compression ratio a little in excess of 9:1. Forty thousandths off the head on these would put the compression ratio close to 10.1, and .060 inch will put it around 10.5:1. Don't be too greedy on compression ratios for street engines. Overdo it, and you may ultimately incur further expense by either measures to suppress detonation or *lowering* the compression ratio. One means of suppressing detonation with very high compression ratios is the use of water injection. This, and its effects on horsepower and emissions are dealt with in Chapter 9.

To get your engine's compression ratio raised it will be necessary to take the head to a motor machine shop and have the required amount machined off. Before reinstalling the head, be sure to remove the sharp corners from the edges of the combustion chamber left as a result of the machining operation. If this is not done, the possibility of pre-ignition is greatly increased. A needle file is the simplest means of taking off these sharp corners.

SIMPLE EXHAUST PORT MODIFICATIONS

So far, ways and means of filling the cylinder and effectively burning the mixture have been dealt with. Once the mixture is burnt, the gases must be expelled as efficiently as possible. Ineffi-

2-10. Machining the head face to raise the CR is an easy way of boosting power and economy from your Ford SOHC engine.

FIG 2-11

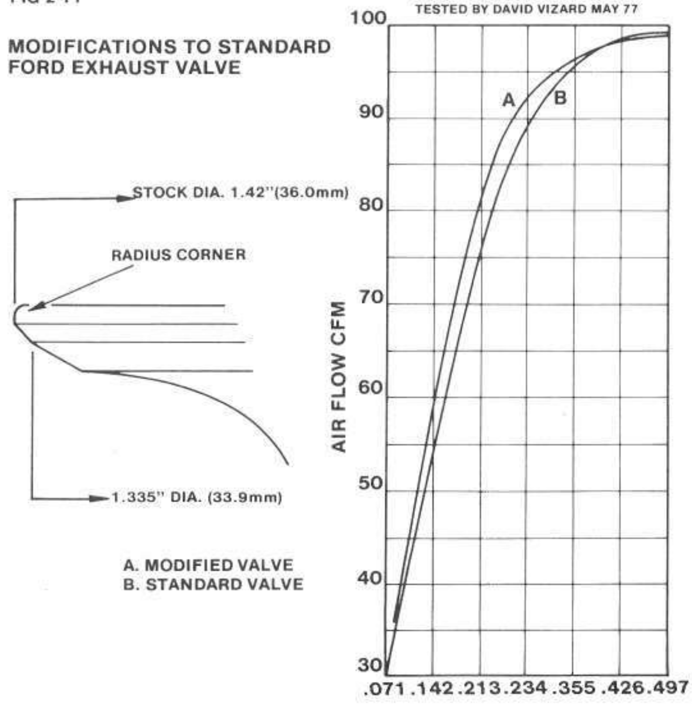


FIG 2-12

MODIFICATIONS TO STANDARD EXHAUST VALVE SEAT

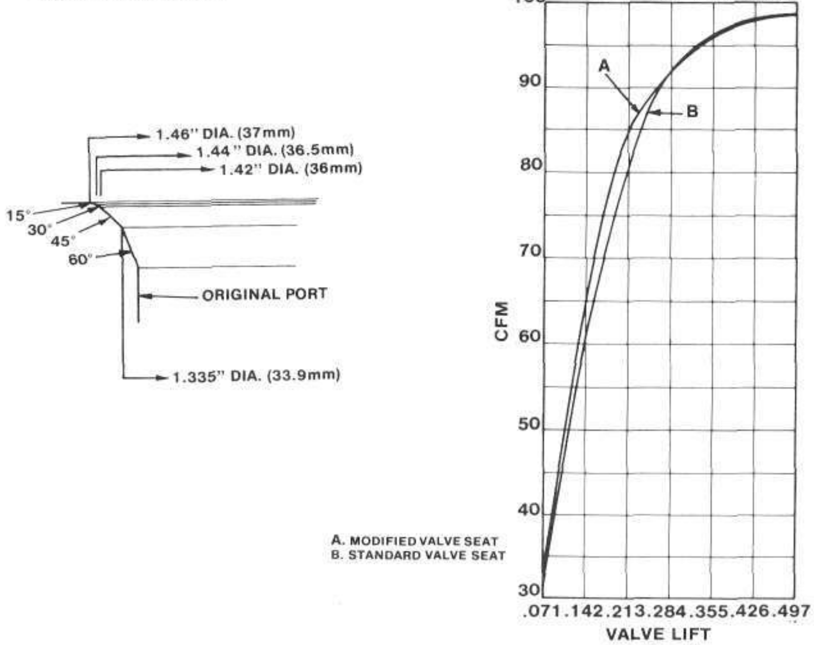
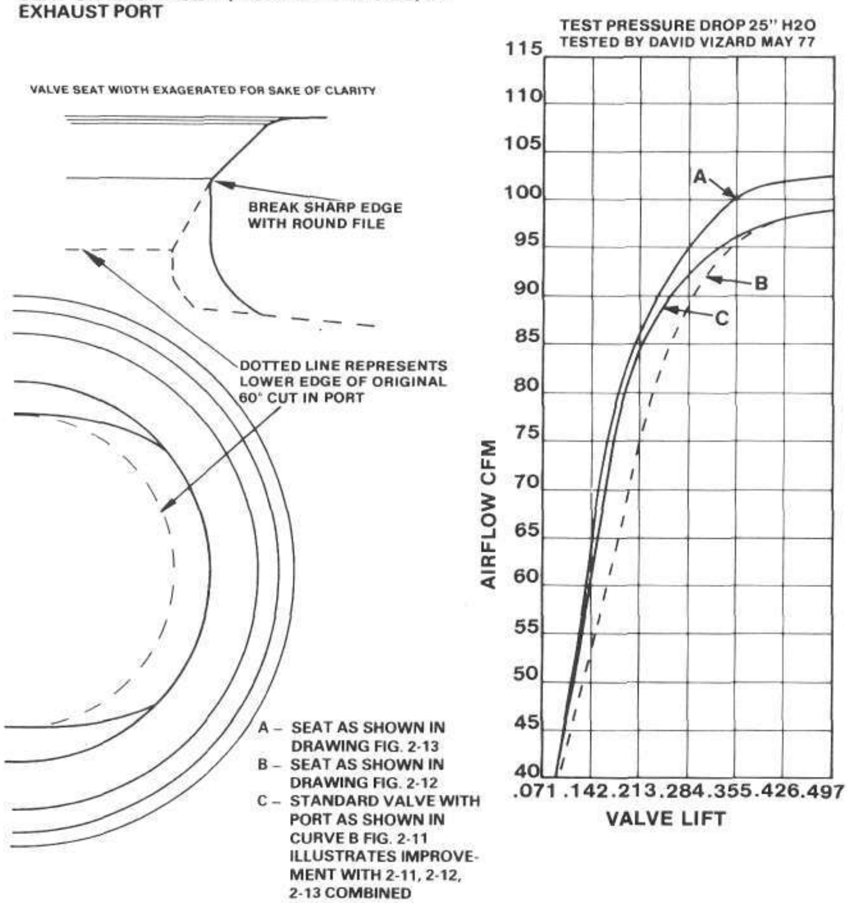


FIG 2-13

MODIFICATION TO PORT JUST BELOW VALVE SEAT ON SHORT SIDE (TIGHT RADIUS SIDE) OF EXHAUST PORT



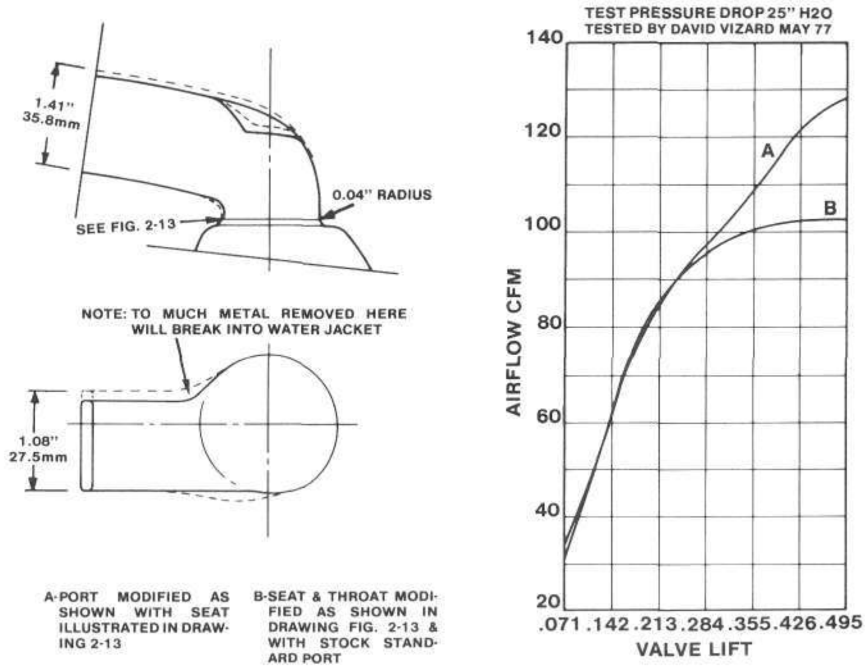
cient exhaust ports not only cost horsepower, but also cost economy. An exhaust port which does not flow very efficiently causes higher pumping losses on the exhaust stroke. This saps power previously generated by burning fuel on the power stroke.

The simplest, most effective modification to the exhaust port is to reprofile the valve as shown in FIG. 2-11. This involves radiusing the chamber side of the valve and back-cutting the back face of the valve on a valve seat grinder to the 30° angle shown in the drawing. This results in flow increases of up to eight percent in the mid-lift range of the valve lift cycle. Airflow over .400 inch lift changes little. This suggests that the port is limiting the flow at valve lifts of this height.

Although it's not the raving success one would wish for, a multi-angle valve seat job on the exhaust does help the situation a little. At lifts up to .330 inch, flow is increased a worthwhile amount. However, over .330 inch, flow suffers a little. The trade-off is definitely in favour of the multi-angle valve seat but it's certainly nothing to write home about. Fig. 2-12 shows the dimensions to cut the seat. If you have a high-speed electric drill or a small high-speed grinder, then five minutes' grinding in each exhaust port will considerably aid the flow. After the multi-angle valve seat has been cut, grind the short side of the port as shown in Fig. 2-13. I should point out at this stage that you need a steady hand. If you slip and gouge the valve seat, you will have to have it re-cut. The graph shows that this grinding operation pulls up the mid-range and top end flow considerably for such a simple modification. Basically, it's allowing more area for the air to pass out of the cylinder and it's making more effective use of the area available in view of the direction the exhaust gases are moving when higher lifts are attained. The tight curve on the short side of the port is so acute that the exhaust flow in this area at high lifts is minimal. At low valve lifts, where the speeds are highest right at the seat, the exhaust is able to make it round this corner without undue problems. Enlarging the throat area immediately under the seat helps the mid-range flow.

If further gains in the exhaust port are intended, you'll have to get deeper into the port and make some substantial changes to its shape. Looking down the exhaust port, you can see that the whole port has a kink in it. This kink and the proximity of the port wall opposite is responsible for flow loss, but to remove it effectively entails grinding quite a bit of metal out of the port. A profile you should aim for is in Fig. 2-14.

FIG 2-14
EXHAUST PORT RESHAPING FOR USE WITH STANDARD VALVE



A-PORT MODIFIED AS SHOWN WITH SEAT ILLUSTRATED IN DRAWING 2-13

B-SEAT & THROAT MODIFIED AS SHOWN IN DRAWING FIG. 2-13 & WITH STOCK STANDARD PORT

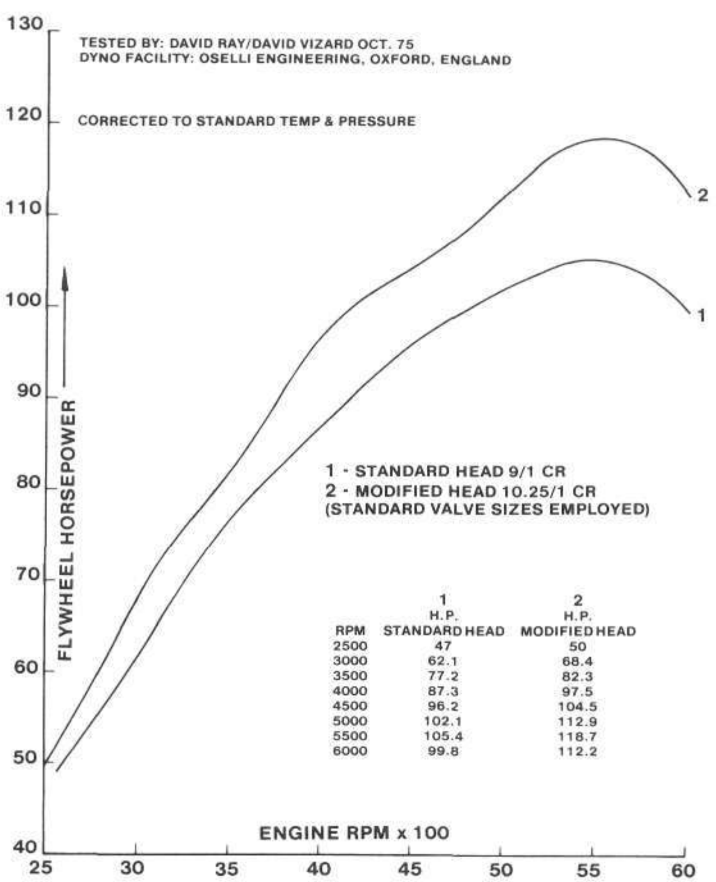
HOW MUCH HORSEPOWER INCREASE?

So far you have been presented with ways and means of increasing the airflow potential of the head. But when we get right down to it, it's not airflow that gets you down the road, it's how much horsepower your engine has. The question is, if you have done all the modifications suggested to improve the intake and exhaust ports, plus raised the compression ratio about one point, what sort of horsepower increase can you expect from a typical 2000cc engine? Fig. 2-15 gives this information. Curve number 1 is a standard Escort RS specification engine. The compression ratio is 9:1 and the engine ran on the dyno less the air filter element and with the dyno exhaust system. As a result the horsepower figures for the standard engine are slightly above those obtained in the as-installed condition. Curve number 2 is the horsepower output of a cylinder head utilizing all the simple modifications described so far. The compression ratio on this engine is 10.25. This requires 100 octane fuel, or water injection with lower octane fuel.

BIG VALVE HEADS

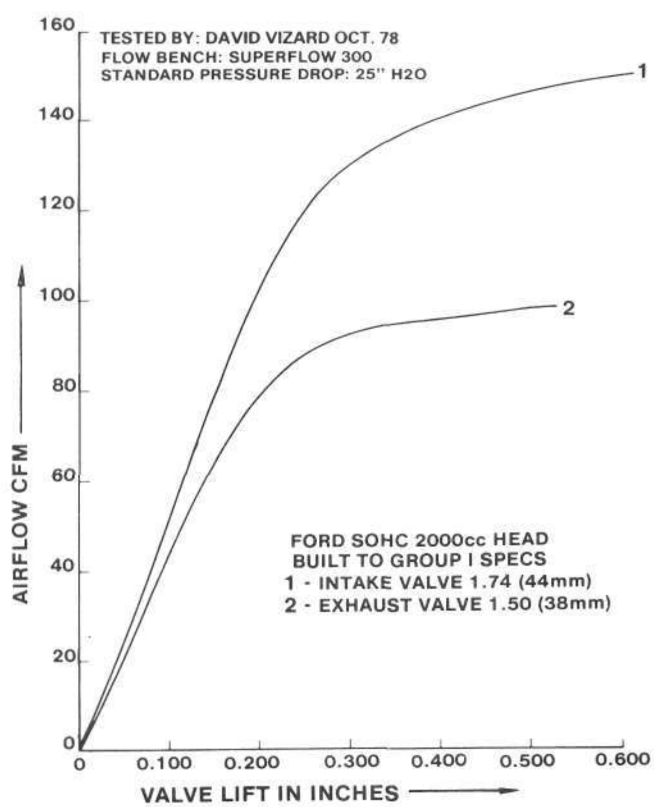
We will deal first with the Group 1 (or

FIG 2-15



RPM	1 H.P. STANDARD HEAD	2 H.P. MODIFIED HEAD
2500	47	50
3000	62.1	68.4
3500	77.2	82.3
4000	87.3	97.5
4500	96.2	104.5
5000	102.1	112.9
5500	105.4	118.7
6000	99.8	112.2

FIG 2-16



group A) cylinder head, available from Ford in England. This is a specially selected casting machined to accept larger valves. The valves employed in a Group 1 head are 1.74 inch (44mm) diameter on the inlet and 1.5 inch (38mm) diameter on the exhaust.

If you are going to compete in any sort of competition where Group 1 rules apply, you have little option but to run with this cylinder head. An out-of-the-box Group 1 head gave the flow figures shown in Fig. 2-16. A comparison with the flow curves given by the standard head shows that the increase is not very substantial. If this engine is to be competitive, then Ford will have to develop a better Group 1 head. My own flow and dyno testing has revealed a rule-of-thumb formula which is useful for predicting approximate maximum power output from a 2000cc engine, based on the airflow of the inlet valve: Horsepower 1.05 x intake cfm at 0.600 inch lift, when intake cfm is measured at 25 inches H_2O.

OR

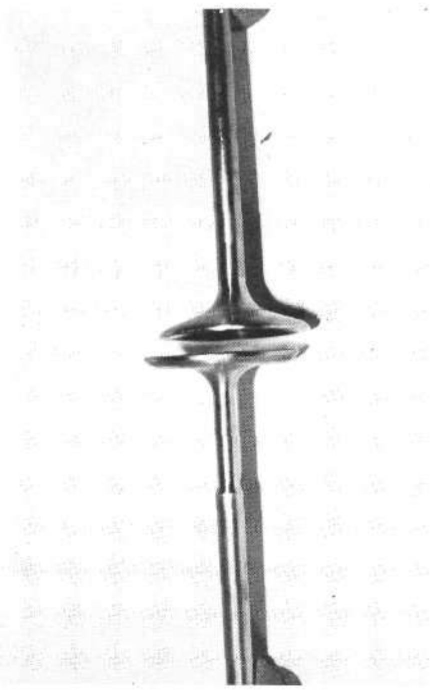
Horsepower = 1.659 x intake cfm at 0.600 inch lift, when intake cfm is measured at 10 inches H_2O

For this formula to be even close, several conditions must be met: first the exhaust flow should be about 70 percent of the intake flow at comparable lift. It also assumes that everything else connected with the production of horsepower is at least more than adequate for the job. By this I mean that there is enough carburation or fuel-injection breathing area, the cam has adequate lift and timing for the horsepower to be produced, the exhaust has correct length and zero back pressure, and so on. With the optimizing of all these items, the head should be left as the limiting factor in the equation. If it is, the formula works quite well and an example looks like this on a Group 1 engine: Airflow at 0.600 inch lift = 150 cfm at 25 inches H_2O drop. Using the formula, we have 150 x 1.05 = 157.5 horsepower. As it happens, even the best Group 1 engine builders are hard pushed to beat the 155 bhp mark.

Considering the fact that this is only an approximation formula, it has proved to be fairly accurate and rarely seems to be off more than about 10 horsepower.

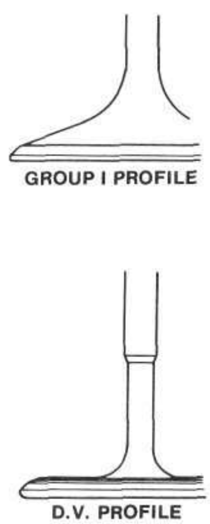
If you are going to build a legal Group 1 head, (rules as of 1978) then little can be done to increase its flow potential. Hand grinding, unless originally done by the factory, is specifically prohibited in preparation of the head. In other words, unless the factory does a porting job on the head, you cannot do one yourself and stay legal. You can, however, do machining work on the valve seats; by reducing valve seat width to a minimum, a little extra flow can be achieved at the expense of valve seat life.

The intake valve seats can be narrowed down to .040 inch (1.0mm) wide, and the exhaust ones to about .050 inch (1.2mm). If you prepare the exhaust valve seat and the exhaust valve with with the same machining techniques as described for the standard head, it is possible to achieve similar flow increases on the Group 1 head; exhaust flow increases noticeably in the all-important mid-lift range of the valve. As far as the intake is concerned, you may be able to use a substitute intake valve in some forms of Group 1 competition. Here you must check with your rulebook or tech official. The intake valve I recommend is one of my design manufactured by G & S Valves or the 'Rimflo' valve produced by Specialized Valves. This type of valve is available in the U.S. through such companies as Branch · Flowmetrics, Esslinger Engineering and Racer Walsh. To stay within Group 1 regulations this valve must be slightly reduced in diameter to 1.74 inch (44.2mm). As it comes from G & S, it is 45mm diameter. Substituting this valve for the Ford Valve in a Group 1 head gave flow increases as shown in Fig. 2-17. I tried many other shapes, plus proprietary brands of Pinto valves, and could not find one which could out-



2-11. This shot shows the radical shape difference between the factory Group 1 intake valve and the high flow valve developed by author David Vizard.

FIG 2-17



STANDARD PRESSURE DROP: 25" H2O

	AIR FLOW CFM	
LIFT INCHES	GROUP I VALVE	D.V. VALVE
0.087	45.7	45.9
0.175	92.3	94.1
0.263	123.8	129.4
0.350	136.9	140.8
0.438	142.8	146.2
0.525	148.0	149.2
0.613	150.3	150.0

flow either of these valves in any configuration of Pinto head where the standard port angle was retained.

NON-COMPETITIVE APPLICATIONS OF GROUP 1 HEADS

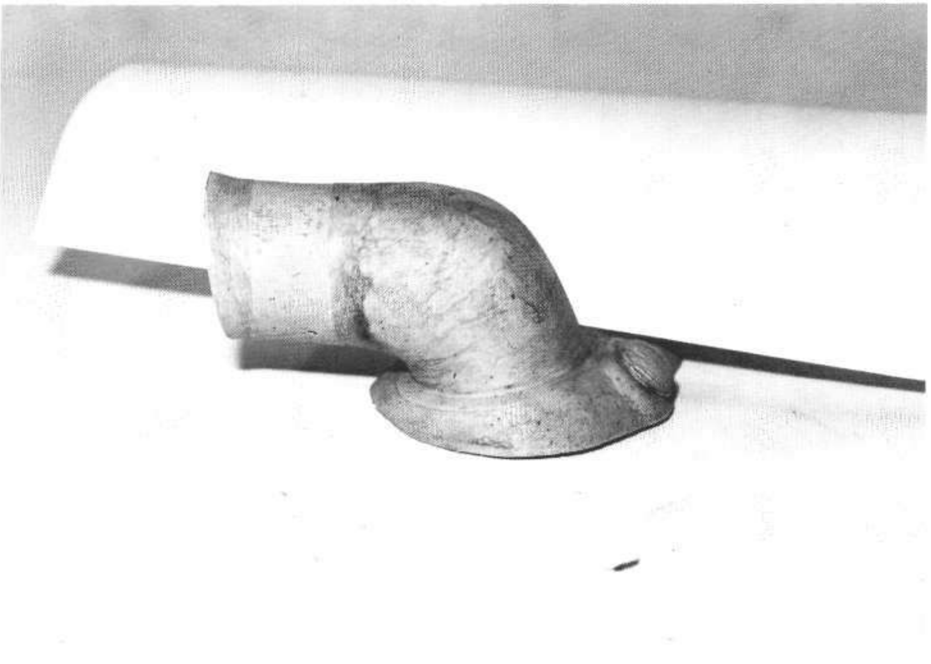
Many of you will no doubt be considering buying a Group 1 head for no other reason than to bolster the performance of your road- going machine. Here I should point something out: you have only to make a comparison of the flow curves to see the standard Group 1 head is inferior to reworking your existing head to even the simplest specifications described earlier. If you want to spend the sort of money involved in buying a Group 1 head take my advice--don't. Spend the money on valves and have your existing head reworked. You will get more airflow and more horsepower for your money.

Actually, there aren't many advantages to buying a Group 1 head for any application except Group 1 racing. The notable ones are: (1) You get a selected casting which may be slightly superior for modifying; it may be a stronger head casting. To some, even this benefit is of academic value only. (2) If you are modifying the head yourself, you will have less work to do in the valve seat area because the head is already machined for larger valves. If maximum flow is the object, a lot of work will have to be done on the valve seats but not as much as starting with a standard head.

BIG VALVE CYLINDER HEADS

Does a normal engine, which is only required for high performance on the road, require the ultimate in cylinder heads? When talking about the ultimate in cylinder heads, one should consider expense as well as practicality. My advice is, if your pocket book will stretch far enough, even for normal road applications, choose a big-valve head. You will lose nothing in low-end power, but you will gain top-end performance. (This assumes of course your engine is cammed in a suitable manner.)

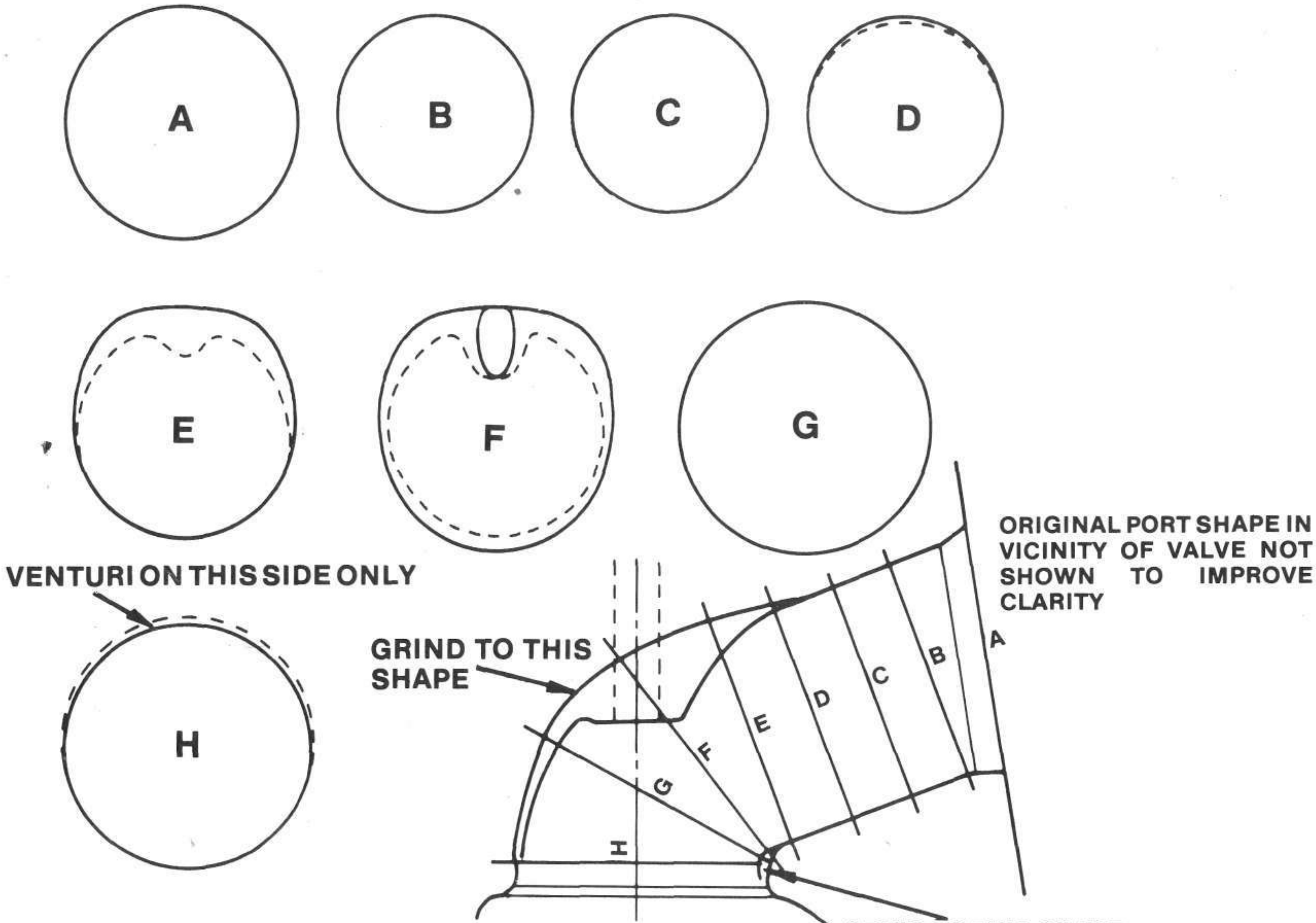
Let me give you an example. My normal road-going Pinto 2000 automatic uses one of my head configurations, which on the flow bench produced a lot more airflow than many current race heads. This engine works fine with the standard torque converter and produces almost double the horsepower at the wheels that the original engine did. This is achieved with head, carburation, cam and exhaust-manifold changes. This should bring home the point that Ford's SOHC engine needs every bit of breathing it can get. While producing this extra horsepower, the engine still met and even surpassed all its original emission standards. Moreover, it yields about 4 MPG more than it originally did. This rule of striving for maximum airflow should be considered valid only insofar as your finances permit. You can draw the line when weld material must be added to the ports. When you begin building completely new ports, then the expense is seldom justified for a road machine. You should look for other more economic ways to improve horse-



2-12. Here is a side-view of the intake port described in the text. It has been moulded in Blu-Sil silicone rubber which, once cured, is easily removed from the head. Note the large bulbous radius on the top side of the port.

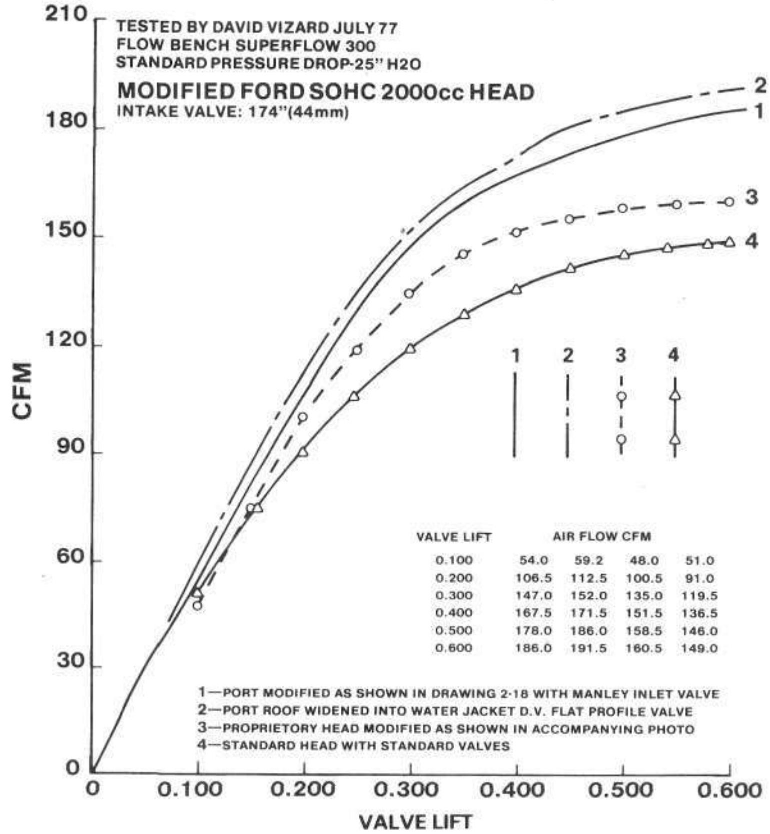
2-13. The widening of the intake port starts about 3/4" before the valve guide. Because of the proximity of the chamber wall, the general direction of flow needs to be biased towards the centre of the cylinder. This is partially achieved by taking a little more material from the cylinder wall side of the port than from the other side.

DRAWING FIG 2-18

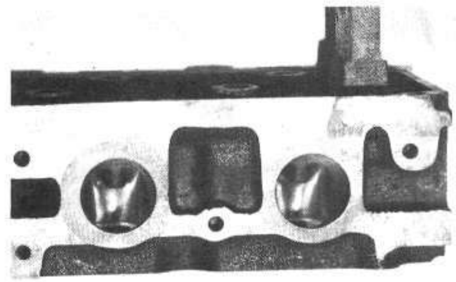


D, E & F: DOTTED LINES SHOW ORIGINAL PORT

FIG 2-18
DRAWING



VALVE LIFT	AIR FLOW CFM			
0.100	54.0	59.2	48.0	51.0
0.200	106.5	112.5	100.5	91.0
0.300	147.0	152.0	135.0	119.5
0.400	167.5	171.5	151.5	136.5
0.500	178.0	186.0	158.5	146.0
0.600	186.0	191.5	160.5	149.0

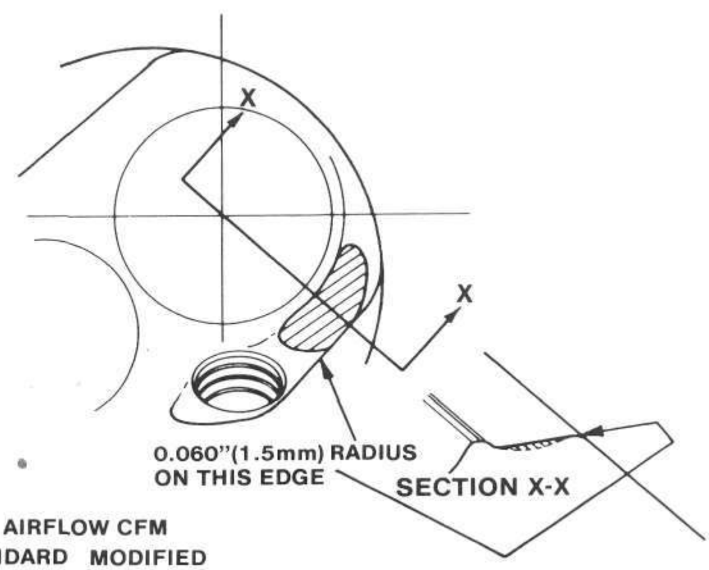


2-14. This head represents close to the ultimate in workmanship but little in the way of headmanship for horsepower as the text explains.

2-15. The Manley valve, flatter of the two shown here, proved an effective piece. Not only is its stem highly wear-resistant for long life, but its shape also proved superior to the high performance Ford item pictured below.

FIG 2-19

COMBUSTION CHAMBER MODIFICATIONS FOR OVERSIZE INTAKE VALVE



AIRFLOW CFM

VALVE LIFT	STANDARD CHAMBER	MODIFIED CHAMBER
0.174	47.6	48.3
0.261	97.3	98.2
0.349	130.7	132.3
0.436	172.3	173.1
0.523	180.2	180.0
0.610	187.3	186.1

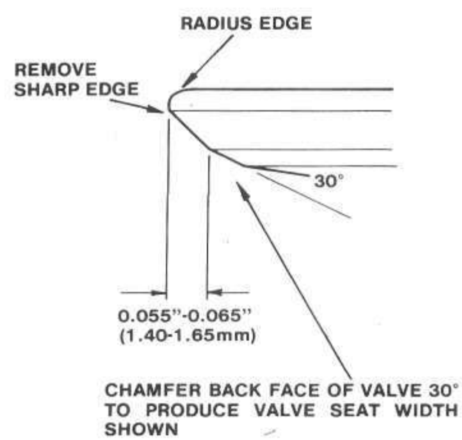
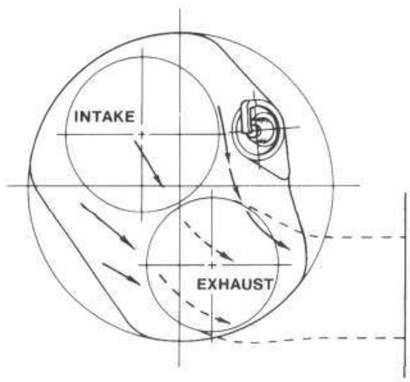
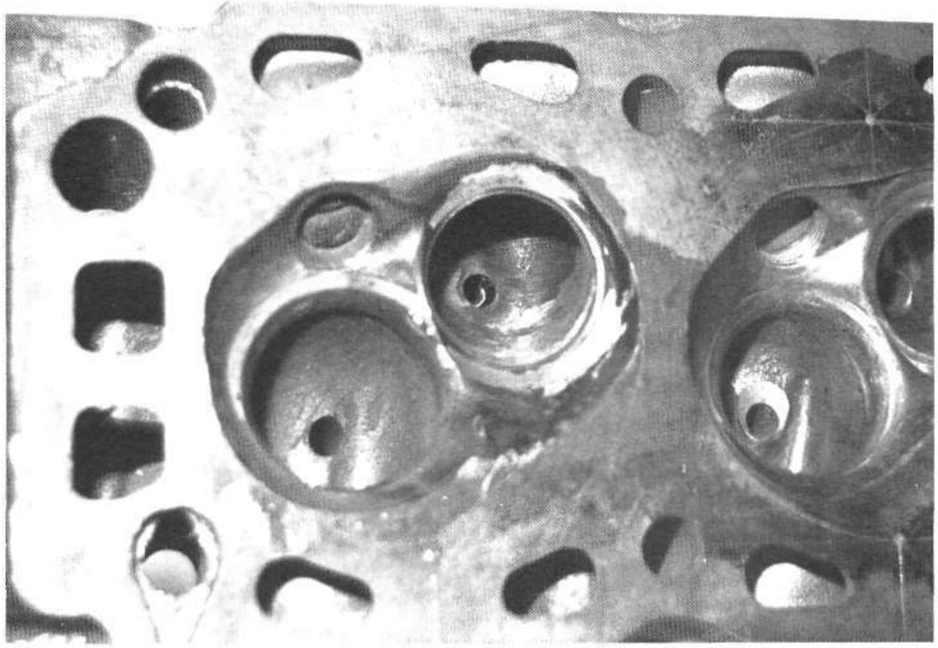


FIG 2-20
EXHAUST VALVE PREPARATION

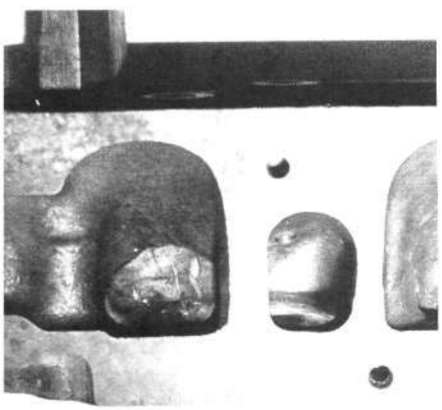
FIG 2-21
FLOW DIRECTION BIAS ON EXHAUST PORT



ARROWS SHOW THE PREDOMINANT DIRECTION OF AIRFLOW FROM COMBUSTION CHAMBER WHEN EXHAUST VALVE IS IN ITS MOST USEFUL WORKING RANGE. THIS IS NORMALLY BETWEEN 0.200" (5mm) TO AS HIGH AS 0.620" (16mm).

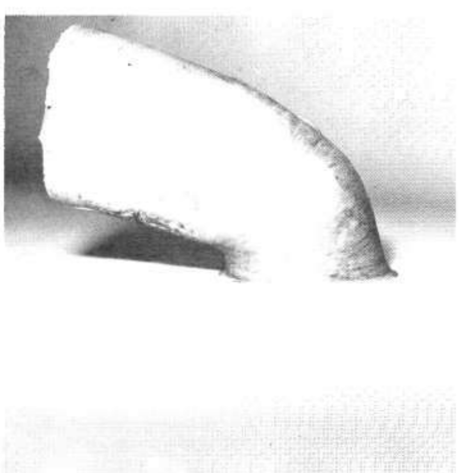


2-16. Best high lift airflow is achieved when most of the valve guide boss is cut away, as seen here.



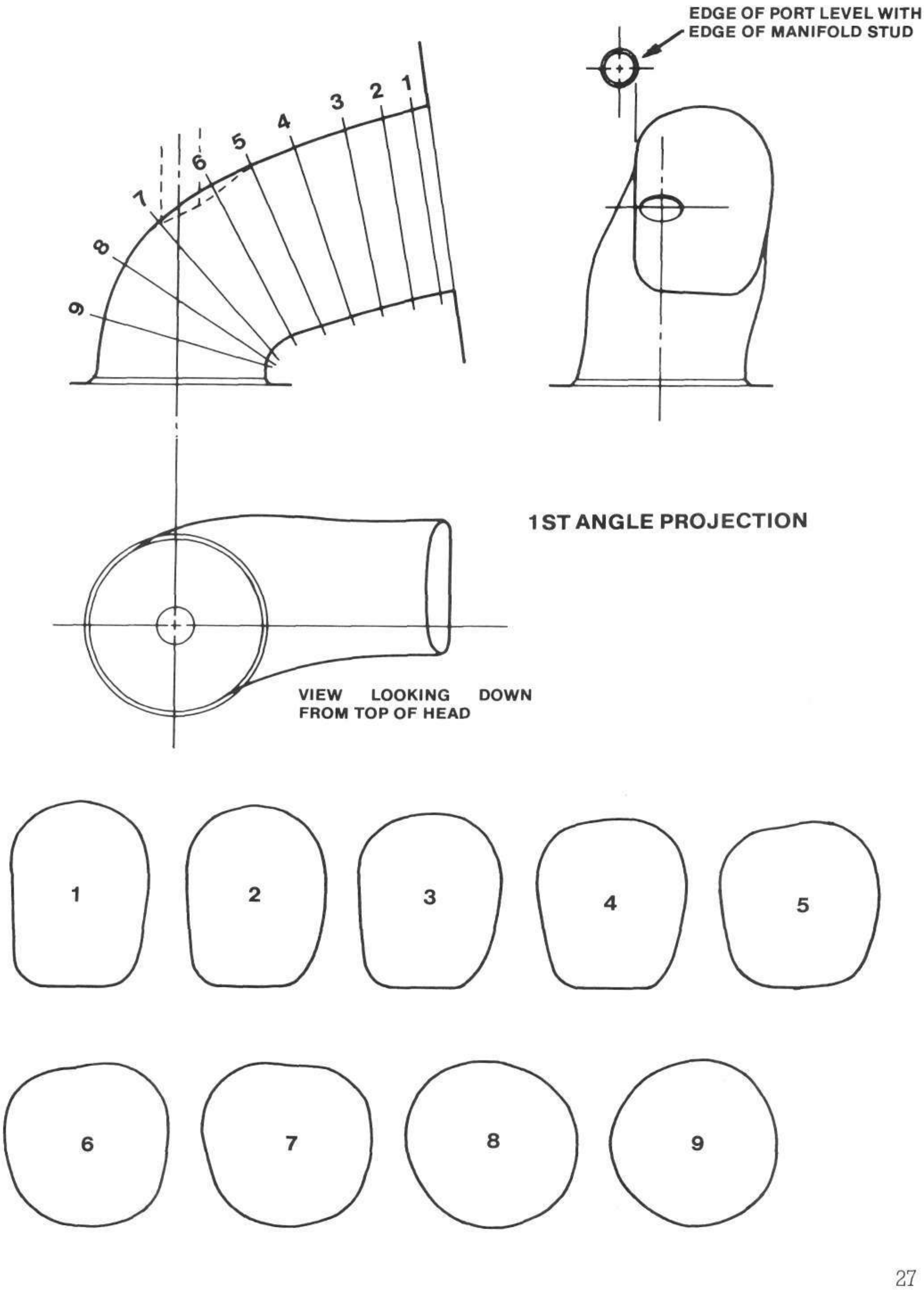
2-17. This is the shape of the exhaust port at the manifold face. Most of the flow takes place at the top and the righthand side of the port.

2-18. The centre of the cylinder is on the right of this exhaust port moulding. The flow direction bias shows up as a "lean" to the left.



2-19 & 20. These two photos show the side and top view of the port that gave the flow figures shown in Fig. 2-22B.

FIG 2-22 **EXHAUST PORT MODIFICATION FOR 38 or 39mm VALVE**



power. Turbocharging or adding nitrous-oxide injection to the engine just might be considered alternatives, as might more sophisticated carburation or even a cam change. Frequently, rules for competiton engines call for a specific stage of modification and a specific means of induction. For instance, if you are competing in a normally aspirated class then you must strive for the best ways possible for inhaling air, short of supercharging the engine. The rules sometimes call for heads which have no material added to the port which means your race head cannot be built with totally re-modelled ports, and you must make the best of removing metal where it will do the most good. All this means your head modification programs will follow several routes to comply with the various rules and regulations possible for various forms of competition.

39mm EXHAUST VALVE

The 1.5 inch valve does not represent the upper limit on exhaust-valve size. Indeed we can increase the exhaust valve size usefully to 1.54 inch (39mm) diameter. The valve I used here is a machined-down Manley 40mm Volkswagen valve. The port development process follows exactly the same routine as that of the 1.5 inch valve and so does the shaping of the seat and the head of the valve. The only caution suggested when using this size relates to the amount of lift possible before the valves touch each other. If your cylinder head is equipped with a 1.74 inch (44.2mm) intake valve and a 1.5 inch (38mm) exhaust valve, then the valves won't actually touch until lifted about .380 inch (9.6mm) off their seats. The Pinto cam hasn't yet been built with an overlap figure which is so much that valves are .375 inch off the seats at the overlap split point, so this is not a problem.

When the 39mm valve size is used, valves will touch at about .200 inch (5mm) lift. This is starting to get close to the overlap split lift figures that some really wild race cams use. Crane's P-322-10 cam has both valves .135 inch off their seats during the overlap period. This gives around .015 inch clearance between each valve head when they are at the same lift. I don't advise going any closer than this, because the valve heads can vibrate at high RPM, causing them to move closer together. I have seen vertical – valve engines with .015 inch static clearance show signs of valve contact after being turned to around 9000 rpm.

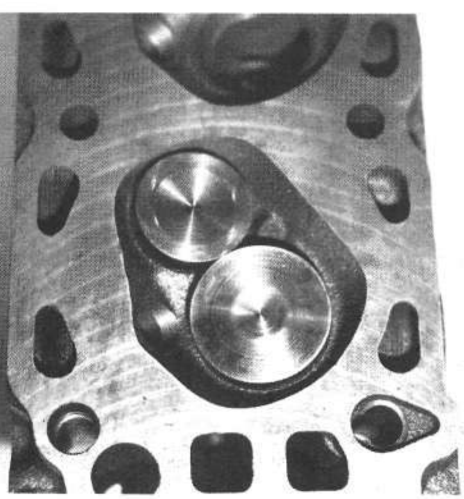
Our situation is a little more complex. With the Ford SOHC engine, the axes of the valves do cross. In a vertical-valve engine, if no vibration exists, valves never cross. The point to note here is, if you are going to use valves bigger than I have suggested, then you must ensure that clearance exists during the overlap period. This, of course, entails installing the cam and physically checking the clearance. Regard .015 inch between the edges of the valves as the minimum acceptable figure.

Okay, let's get back to our 39mm valve. The extra five percent of valve area is a help. It boosts low-end flow by about two percent and the top-end flow decreases by about one percent with the same port configuration as described for the 1.5 inch valve. A study of the flow figures in Fig. 2-22 shows that most cams will not get into the lift area where flow is reduced below that of the 1.5 inch valve, so this extra flow is a help, even if only marginally. This is as far as I had gone at the time of writing, but I do feel that with further development on the flow bench, a slightly more effective port could indeed be produced just using normal grinding techniques.

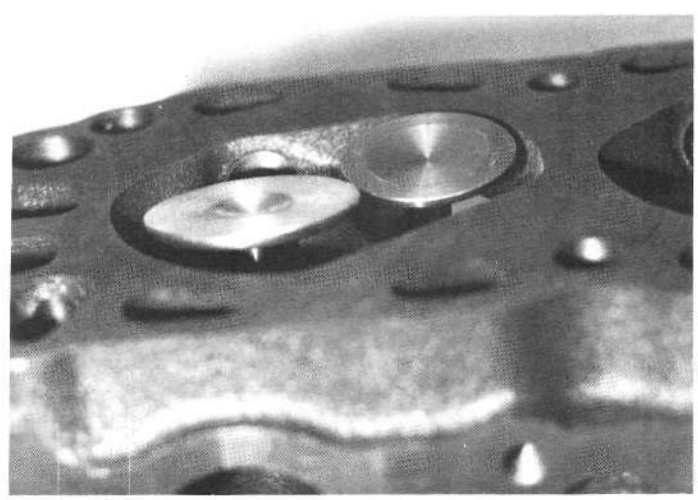
For any substantial gains in exhaust port performance, radical changes are going to be necessary. Such measures as relocating valve positions and welding the exhaust ports to achieve more favourable shapes will have to be considered. For the Ford SOHC engine, it's down the road a ways. However, if development is to go on, this is the route things must take.

HOW MUCH HORSEPOWER?

At the time of writing I have never tested a head prepared to a big-valve specification on an otherwise standard engine. I have used heads of this specification on numerous road- going engines, but they have always had other modifications on them as well. Dynamometer testing indicates just what we would expect from theoretical

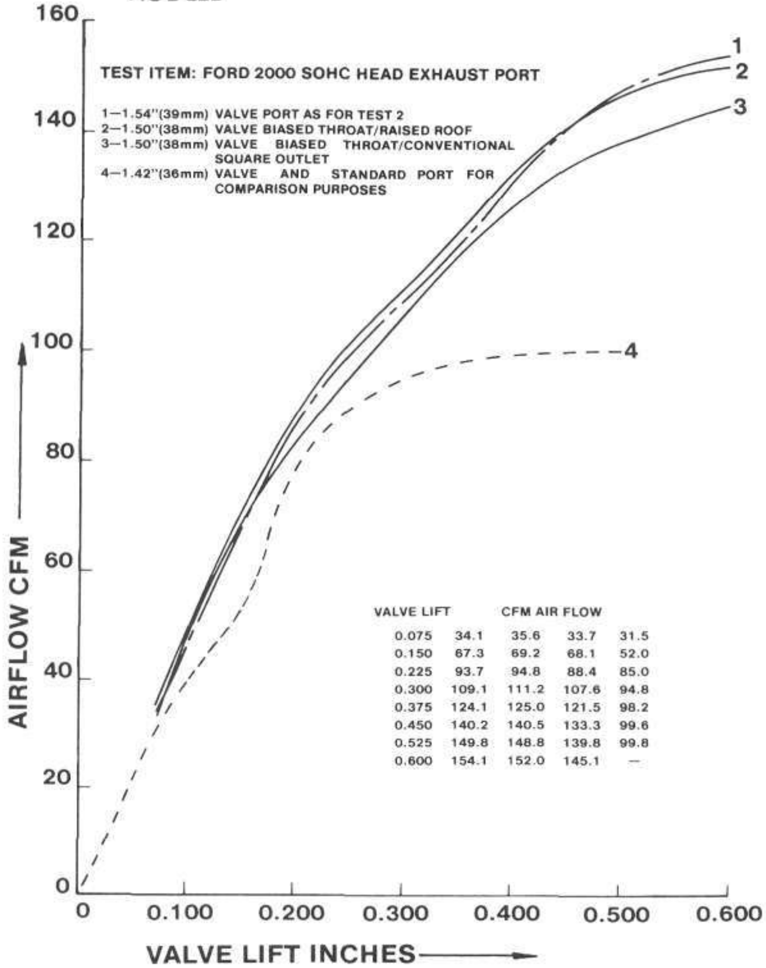


2-21. Because the valves are angled in relation to each other, they will touch if lifted high enough.



**2-22. With a 1.74" (44.2mm) intake valve and a 1.5" (38.0mm) exhaust valve, contact is made at about 0.38" (9.6mm) lift.
That's just about the depth of the standard chamber.**

FIG 2-22B



VALVE LIFT	CFM AIR FLOW			
0.075	34.1	35.6	33.7	31.5
0.150	67.3	69.2	68.1	52.0
0.225	93.7	94.8	88.4	85.0
0.300	109.1	111.2	107.6	94.8
0.375	124.1	125.0	121.5	98.2
0.450	140.2	140.5	133.3	99.6
0.525	149.8	148.8	139.8	99.8
0.600	154.1	152.0	145.1	—

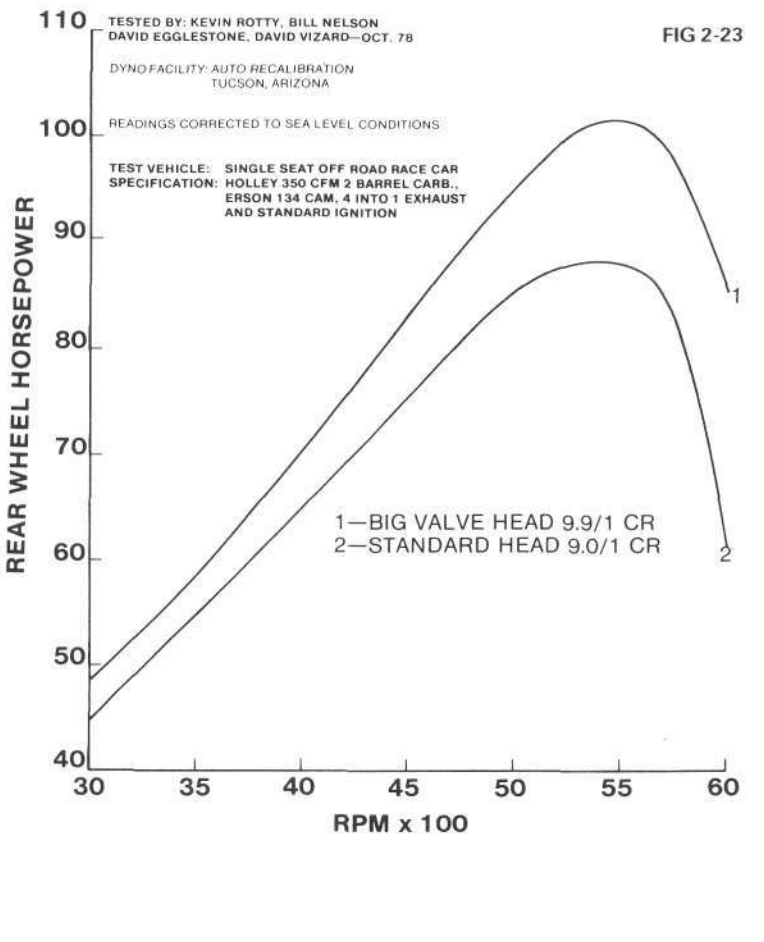
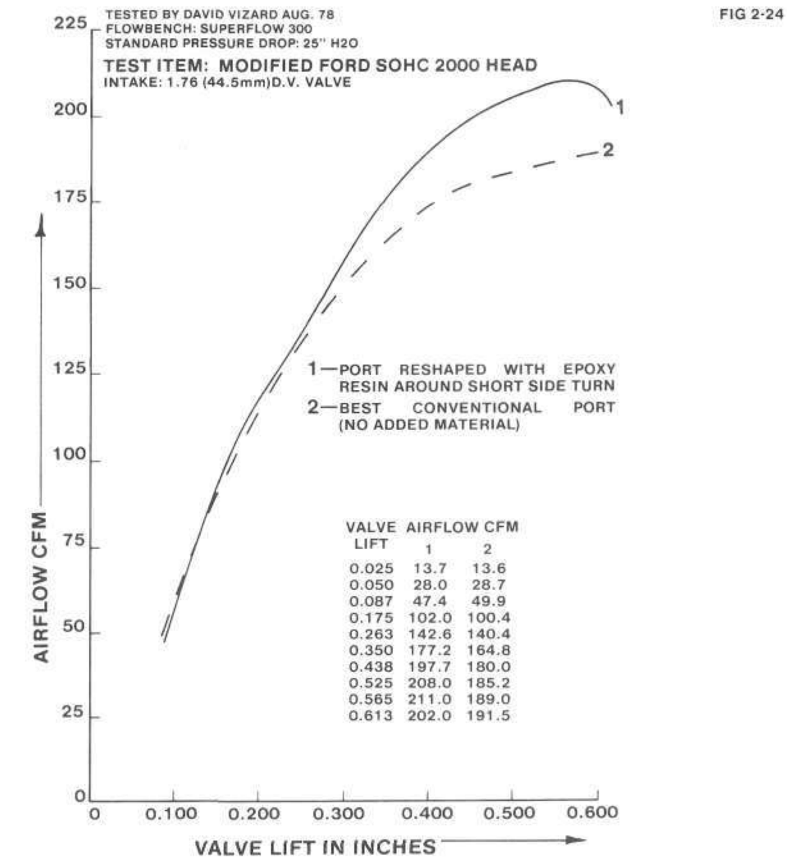


FIG 2-23

projections, namely that with the cylinder head developed to as high a degree as possible, every other modification we do to the engine pays off in a more beneficial manner. Those carbs that give 10 extra horsepower when bolted onto a standard engine give 12 to 15 more horsepower when fitted to an engine with a well-modified head. In Fig. 2-23 you see the horsepower curve on an engine first equipped with a standard head; then with a modified head. The rest of the engine is to specifications I have built several times with pleasing results. The engines gave satisfactory power curves for normal road use for off-road vehicles or vehicles with automatic transmissions. The engine produced strong horsepower from mid- range right through to about 7000 rpm. The only modifications on the test engine (Fig. 2-23) prior to installing the head were a Sig Erson 134 cam, a four-into-one exhaust manifold and a 350-cfm two-barrel Holley carburettor mounted to the standard intake manifold with an adaptor plate. The increase in horsepower coming from replacing the standard head with a modified one on this engine was very satisfying.

This test engine was installed in an off-road race car. A chassis dynamometer was used to measure output. Cars equipped with typical off-road tyres when tested on a chassis dynamometer, can exhibit a sudden drop off in the power curve. This is due to high power absorption by the tyre above a certain critical speed. You will notice that both horsepower curves drop off very rapidly. This is partially due to the fact that tyre horsepower absorption is increasing rapidly at the wheel speeds encountered at the high rpm levels the engine was tested at. What you should consider more than the fact that peak horsepower went up from 88 bhp to 101.5 bhp is the fact the horsepower at 6000 increased no less than 24 at the wheels. Looking at it another way, peak horsepower went up by 15 percent and the horsepower at 6000 rpm went up 39 percent. In this particular application the off-road vehicle was able to pull a very strong top end in situations where high speeds could be run. On the other hand, lower down the rev range, it had sufficient boost in torque to be able to accelerate quickly out of slow turns.

2-23. Here are the three most common types of intake valve used. Left to right, they are Manley, Ford Group 1 and DV anti-reversion valve.



MAXIMUM FLOW PORTS

If the valves are to occupy all the available space in the combustion chamber, it is generally accepted that for maximum power the valve diameters be apportioned so that the exhaust flow is about 70% of the intake. You may remember that the exhaust port discussed flowed 154 cfm at .600 inch lift. The best intake port so far investigated flowed 190 cfm at 600 inch lift. This means the exhaust flow is 81 percent of the intake. If the head we were dealing with had super-efficient ports that could hardly be improved upon, the next move would probably be to increase intake-valve diameter, even if this meant reducing exhaust-valve diameter. The head in question does not have highly efficient ports; throwing away some of our exhaust flow for very minimal gains brought about by using a larger intake is unlikely to produce positive results in terms of bhp. The only practical route to follow is to increase inlet port flow efficiency.

The easiest way to change port shape is to add epoxy-based material such as Devcon JB Weld or equivalent. In early experiments, the ramp port worked well but the critical nature of its shape led me to the investigation of more conventional port forms. This brought about the development of the port shape in Fig. 2-24. To negotiate the short side curve, the port floor is raised, and to maintain a reasonable cross-section area, it is widened just before it goes over the hump. Looking down such a port, you will see it forms an apple-shape just upstream of the turn toward the back of the valve. Check the drawings and photos. These show that the epoxy buildup is higher on one side of the port than the other. This biases flow toward the centre of the cylinder. The extra flow of this port starts to pay off from lifts of .300 inch lift upwards. It doesn't do much for lifts less than .300 inch because below this level the available flow area between the valve and the seat in the head is the limiting factor, not the port. At .565 inch lift the efficiency of this port (based on valve diameter) reached 64 percent compared with 57 percent at the same lift with a "metal-removed-only" type port. Increasing flow in this manner means exhaust flow is 73 percent of the intake. This indicates intake-to-exhaust diameter bias is about right for maximum output.

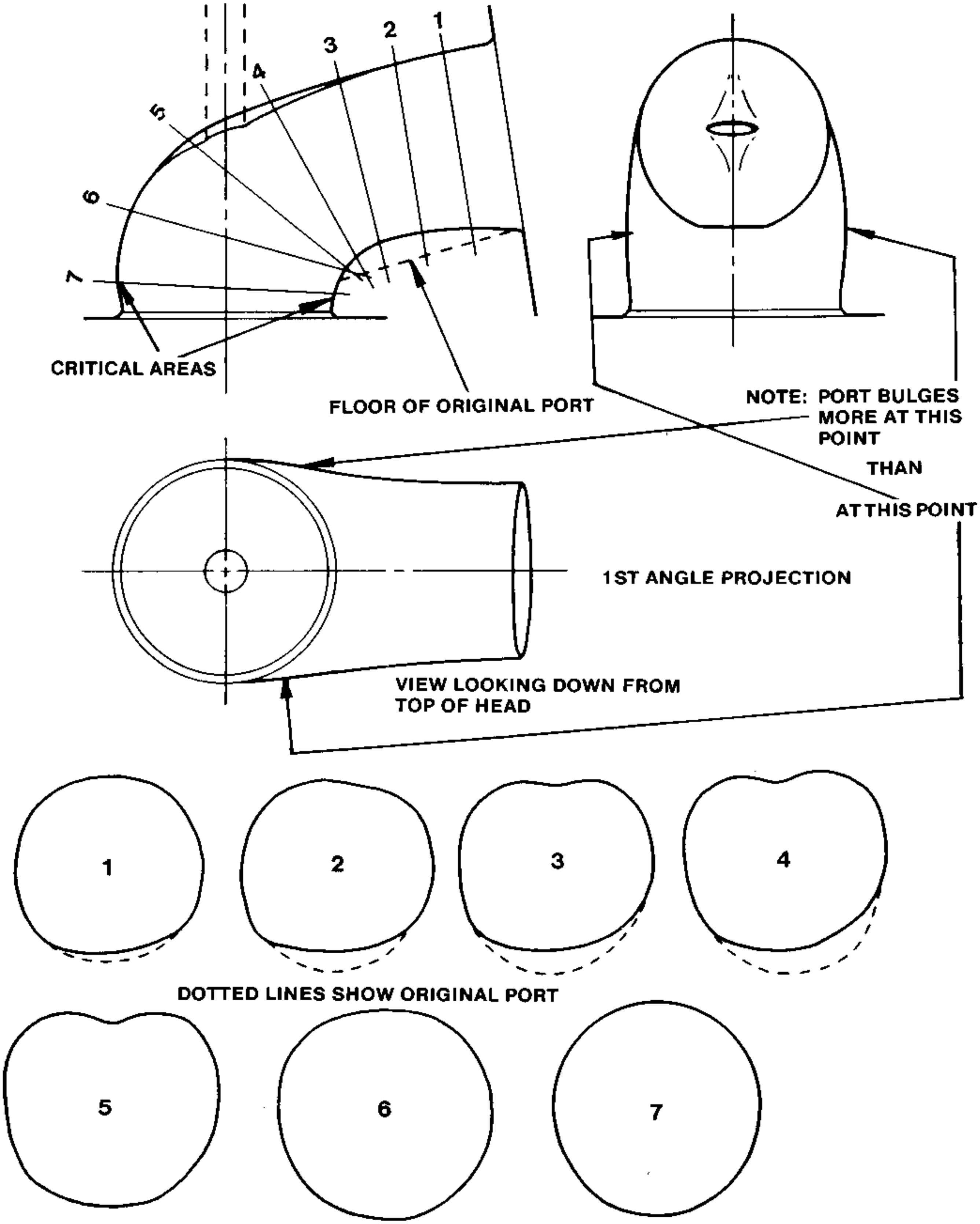
Another interesting pcint with this port is that unlike most of the ports so far shown, the flow actually starts to drop off with very high valve lift. This port does not need such a tremendously high valve lift to make it work as do the more conventional ports, where flow improvements are achieved by removing metal only.

DOWNDRAFT INTAKE PORTS

In the early 1970s, when the 2000 engines seemed stuck on a horsepower shelf of around 180, Ford Motor Company U.S.A. approached Jerry Branch of Flowmetrics with the idea of developing a superior head. Ford commissioned Branch to develop the necessary head modifications and design an induction system to give the engine a 200 bhp capability. Testing on Jerry's flow bench quickly established that the standard port had chronic breathing problems. Experience also told him that it might take a long time to make the stock port work. (Oh, how right he was!) This, and other relevant factors brought about the decision to put in a new set of ports at a more favourable angle. After a number of experiments, 29° between the valve and port centreline was decided upon. Port

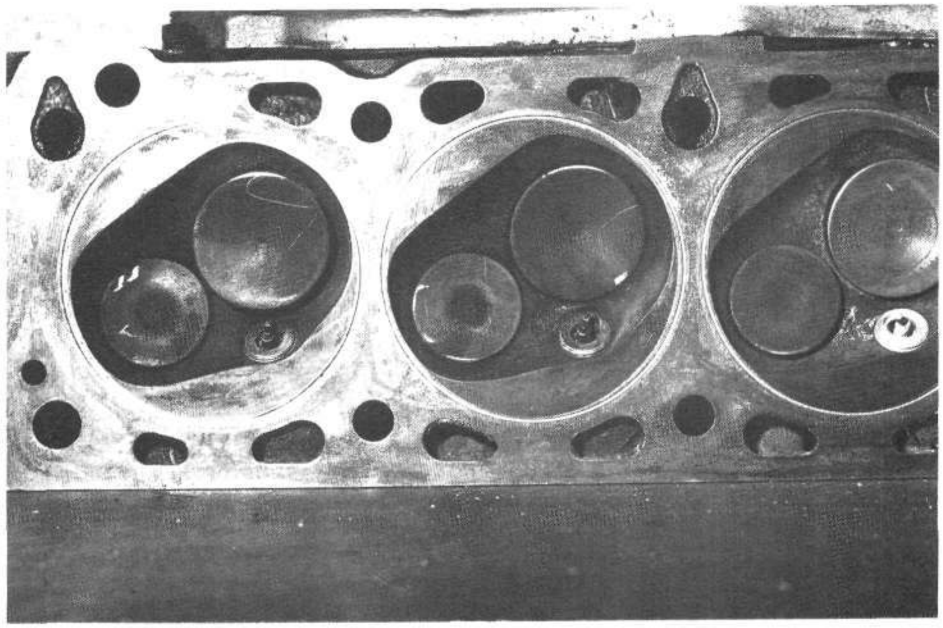
FIG 2-24

INLET PORT WITH ADDED MATERIAL KNOWN AS AN 'APPLE PORT' BECAUSE OF THE CROSSECTIONAL SHAPE



Right

2-25. Smooth contours of the combustion chambers on the Branch developed head are evident here. The slightly different chamber shape used could be expected because of the different port entry angle.

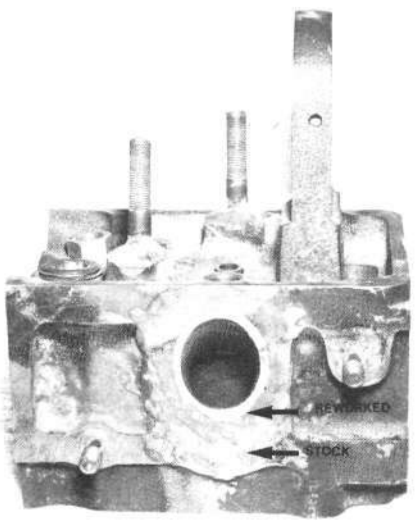


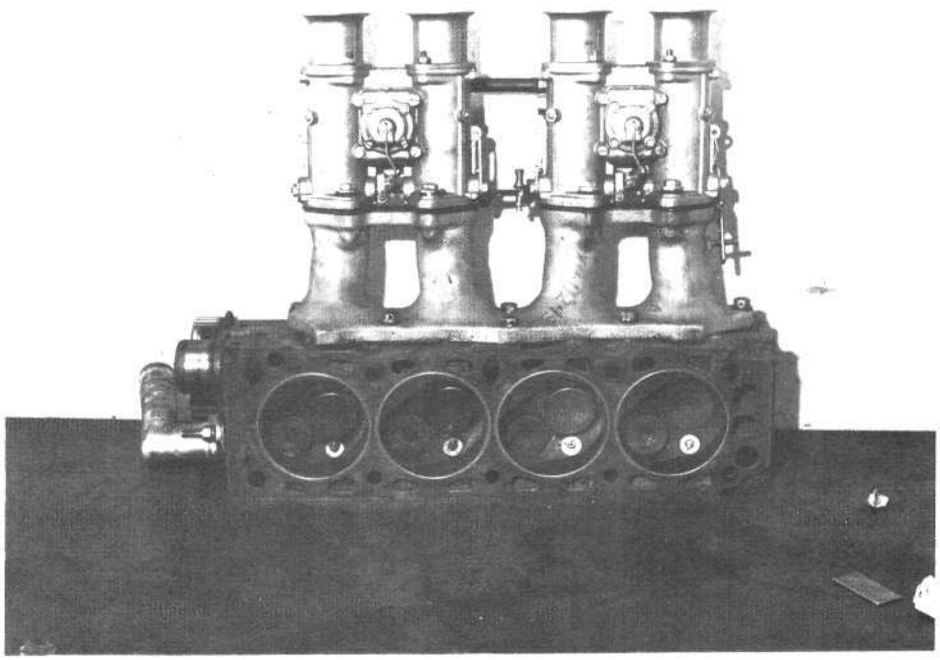
Below left

2-24. Jerry Branch's flow bench model used to develop an effective downdrafted head. This head design was responsible for producing one of the first genuine 200 hp, 2000 cc engines.

Below right

2-26. The downdrafted Branch head utilized a brace of 45 DHLA Dellorto carbs. Because of the new port positions, a special manifold had to be made up.





diameter was smaller than standard to maintain fuel suspension. The result of Branch's thorough airflow work, plus the efforts of the engine builder and cam designer, resulted in 203 bhp and an almost-sheet drivable rev range. This was probably one of the first true 200 bhp-plus Pinto engines to be built. Although some people had claimed 200 bhp prior to this, they were largely unjustified claims. Figures on the Branch-headed engine were substantiated by Ford engineers. Ford of U.S.A.'s subsequent racing ban ended this very promising development. The head Jerry did back in '73 is now dated. Heads with ports at the normal angle can be made to produce more flow and more power. What should not be overlooked, though, is that as early as '73 Jerry Branch had realized the problem preventing this engine from producing good power. In less than three months, he designed, developed and produced a cure. Most of the rest of the world took four years to catch up and some haven't caught up yet. Although Jerry's original head may be out of date today, his concept is still as valid as it was back in '73. This concept dictates that to achieve maximum flow, the port must be moved to a more favourable angle and position. As the photos of the clay mock-up show, ports on the Branch head were moved up considerably, requiring a custom-made intake manifold to adapt 48 DHLA Dellortos carburettors to the head. This is fine for an experimental engine but hardly convenient for the average enthusiast engine builder.

A downdraft head made by Manx Racing in England strikes a happy compromise. The port is raised only by about .1875 inch at the manifold face. This means that a little filing allows using a manifold with the original bolt pattern. Although the downdraft angle is only about 4.5° steeper than standard, flow figures show a useful gain at high valve-lifts. Fig. 2-25.

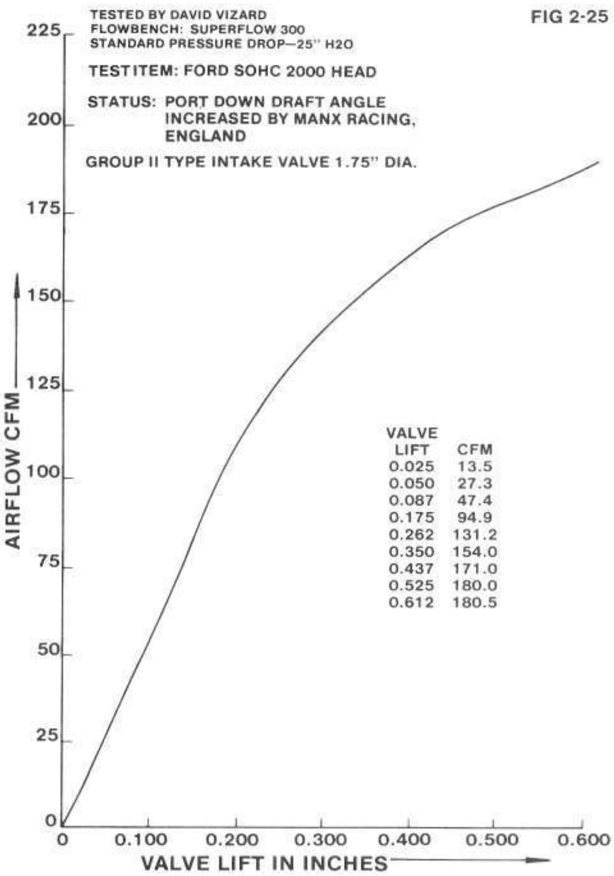
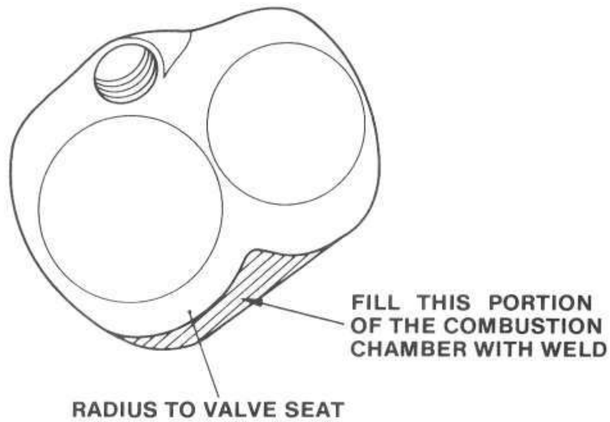
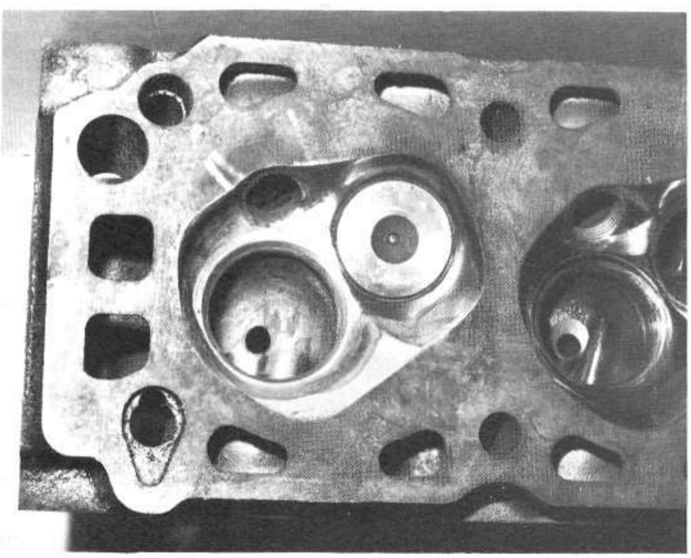
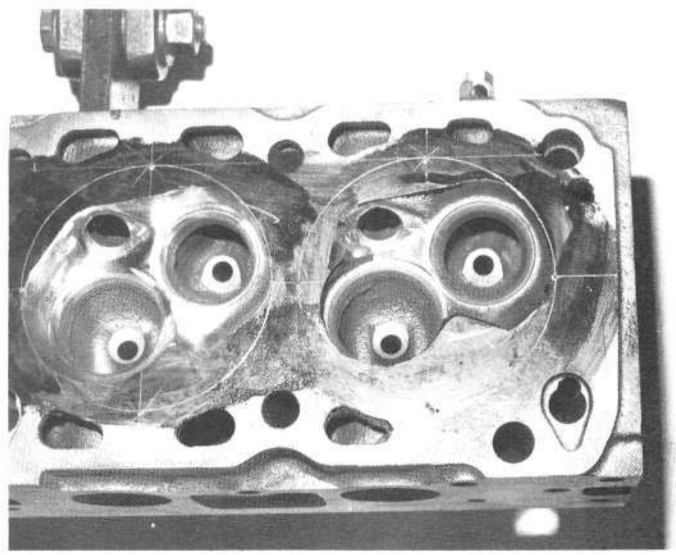


FIG 2-26
SHAPE OF WELDED CHAMBER





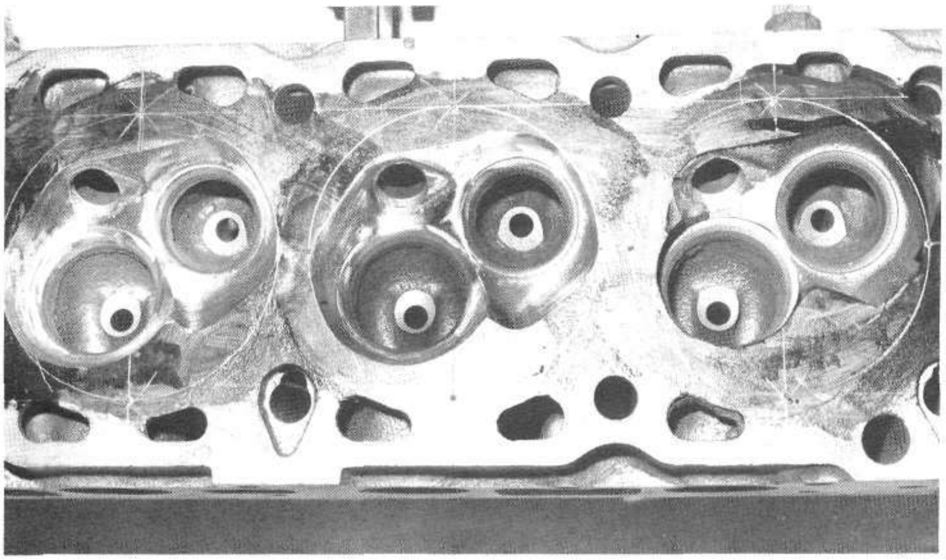
2-27. Here the clay flow bench model of the chamber shape shown in Fig. 2-26. The added material accounted for a flow increase of between 3 and 5 cfm.



2-28. Here is what a welded combustion chamber looks like during the early stages of construction. The only operation so far has been to mill off excess weld from the head face and start the layout marking.

WELDED CHAMBERS

Earlier on, I said little shrouding existed in the combustion chamber, and from this point of view, the chamber was a good basic design. This does not mean, however, that it is as good as it can be for making horsepower. The original design was to produce adequate power consistent with satisfactorily low emissions. If your intention is to build a no-holds-barred race engine, however, some horsepower is available by modifying the combustion chamber. The first reason for modifying the chamber shape to that shown in Fig. 2-26 is that it increases airflow. Secondly, there is a small reduction in flame travel distance due to the more compact chamber. Thirdly, it produces more squish (quench) area



2-29. At this stage the welded chamber is rough-ground to shape.

on the side of the chamber opposite the spark plug.

Let's look at this in more detail to see why these modifications work. One of the principal reasons why airflow past a typical valve is characteristically inefficient is that there is a sudden change in the cross-section area right after the valve. If you inspect a venturi, you will notice attention is paid not only to the approach angle and shape of the venturi, but also the exit angle. As far as most valve seats are concerned, streamlining after the hole is non-existent. At this point it is important to note that flow gains can sometimes be made by using the chamber wall to control the air as it comes out past the valve seat, so it's not a completely lost cause. As it happens, in the Pinto engine, the chamber around the long side of the port is just about where it should be, so that this area is fairly efficient. The short side of the port however is not, and in this area modifications can be made to increase flow. The drawing shows that the chamber wall moved closer to the valve. Remember, not a great deal of air comes out here, and what does come out is a highly turbulent flow. Placing the chamber wall closer to the airstream steadies the flow and produces more airflow from this side of the valve. Flow tests show that with either a conventional port or a port which has had the floor raised by some means or other, will usually flow between three and five cubic feet more with the chamber shaped as shown.

Dealing with the second point, it's a well-known fact that a large combustion chamber will generally detonate at a lower compression ratio than a small combustion chamber. This is primarily due to the length of flame propagation time and the amount of compression and heat radiation the unburned charge receives from the advancing flame front. The longer this unburned charge is exposed to the advancing flame front the more likely it is to detonate. By making the combustion chamber smaller, we can reduce the engine's tendency to detonate at any given compression ratio. Alternatively, the engine can use a higher compression ratio before detonation occurs.

The third point is that this engine tends to require less ignition advance if it has more quench area on the side of the chamber opposite the plug. At the same time, it has also shown a tendency to produce more power when the quench area on the plug side is reduced. The simplest way to reduce quench area on the plug side is to file a generous radius on the edge of the quench area. A similar but smaller radius is also beneficial on the quench area around the edge of the exhaust valve. This radius allows easier out flow of spent gases toward the end of the exhaust stroke. After initial development work on the flow bench, the particular head you see in the photos was built by Carl Schattilly of C & G Porting, Tucson. The exacting standards to which Carl Schattilly reproduced the flow bench model, show the art of head modifying at its best.

VALVE MODIFICATIONS

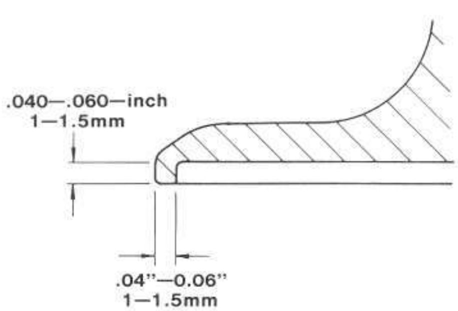
One drawback of a cam with too much overlap and duration is that it sacrifices low-speed engine performance. For a race car, even if low crankshafts speeds are seldom used, it's nice to be able to drive around the pits or paddock without worrying about stumbling to a halt. For a street-driven machine, it's an advantage to be able to choose a slightly longer cam for more top-end bhp, if you can do so without compromising low-end power and flexibility for good street manners. Simple modifications to the intake and exhaust valves provide some flexibility increases, although the result is not totally without drawbacks. The principal factor affecting flexibility at low rpm is valve overlap. When both valves are open, the exhaust may want to flow back out through the intake valve rather than the exhaust valve. Also, any adverse shock waves or pulses in the exhaust tract, may cause exhaust gas which has already left the chamber to return just prior to exhaust-valve closure. If the intake and exhaust valves are modified to flow well in the correct direction and to have less ability to flow in the reverse direction, then loss of flexibility at low-rpm can be significantly reduced.

Recessing the inlet valve face as in Fig. 2-27 reduces the valve's reverse-flow efficiency. It also lightens the valve, which is a fringe benefit. To have any effect on reverse flow, the undercut in the valve face should be 0.03 inch (0.75mm) or more. Valves of this design are now available from G & S Valves and Specialized Valves in England. A similar experiment was tried with the exhaust valve. The back face of the valve was undercut to de-streamline it. The ridge formed at the seat reduces exhaust flow back into the combustion chamber. This proved very effective with standard ports. Reverse flow was reduced as much as 15 percent compared to the standard valve. When this type of anti-reversion valve was used with a modified port, results were singularly unimpressive. Reversion was reduced a negligible amount and a penalty was paid in terms of reduced flow in the normal direction. The anit-reversion concept on the exhaust appears good, but for this application more development work is obviously needed.

16 VALVE CYLINDER HEADS

Holbay Racing Engines Ltd., are well-known for designing and producing race-winning engines. To give the 2000cc engine every possible chance of doing this, they produced in conjunction with a design specialist a cylinder head with four valves per cylinder. The idea of using four valves per cylinder is to accommodate more valve area in the confines of the cylinder. Also, because the head is intended for maximum power, port shapes, valve angles, and other factors are designed for as much airflow as possible. The original two-valve-per-cylinder head was not produced with ultimate power in mind, hence the lack of attention to producing tracts with adequate air flow.

To use the 16 valve head, pistons and rods, plus everything above the head gasket must be changed to parts compatible with the head. The use of this head also requires special camshaft-drive components. As far as induction systems go, Holbay make a slide-plate throttle, timed fuel-injection system or a manifold to accept sidedraft carburettors, such as Webers or Dellortos. Depending on the application, cams can be supplied for road or race specifications for use in this head. Possible applications for the Holbay 16 valver are wide and varied. It could be used to produce a flexible but powerful, street-drivable engine or for race applications such as midget racing in the U.S. At the time of writing, this type of racing is dominated by Volkswagen-powered machines. A 16-valve conversion, together with a stroked crank and a bigger bore to produce 2300cc should give this engine a lot more reliable bhp than the Volkswagen.

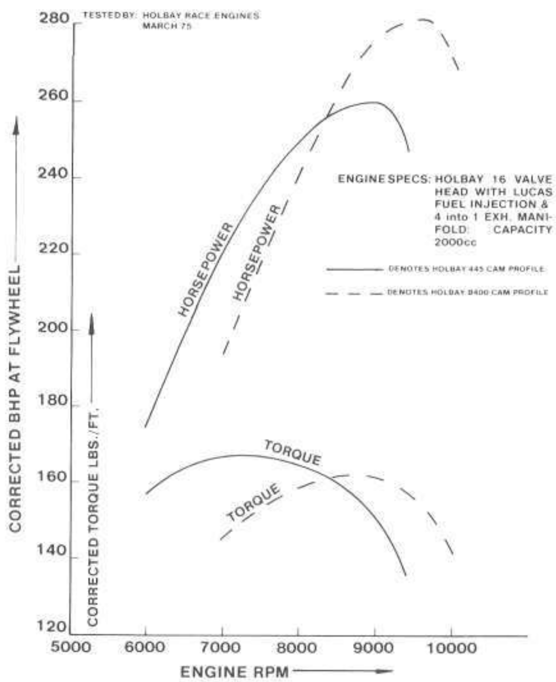


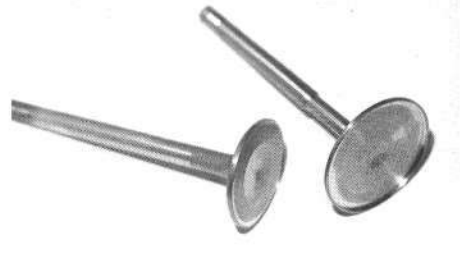
MODIFICATION TO INLET VALVE TO REDUCE REVERSE FLOW DURING OVERLAP

There are many other applications which fall between these two extremes, ranging from rally cars to fuel dragsters. You only need some imagination to see its potential.

I would like to have shown a comparison between the airflow of a 16-valve head and a modified 8-valve head but unfortunately none of this exotic equipment has ever come my way. To give you some idea of the potential of a 16-valve, two-litre engine, just look at the power output achieved on Holbay's dyno, Fig. 2-28.

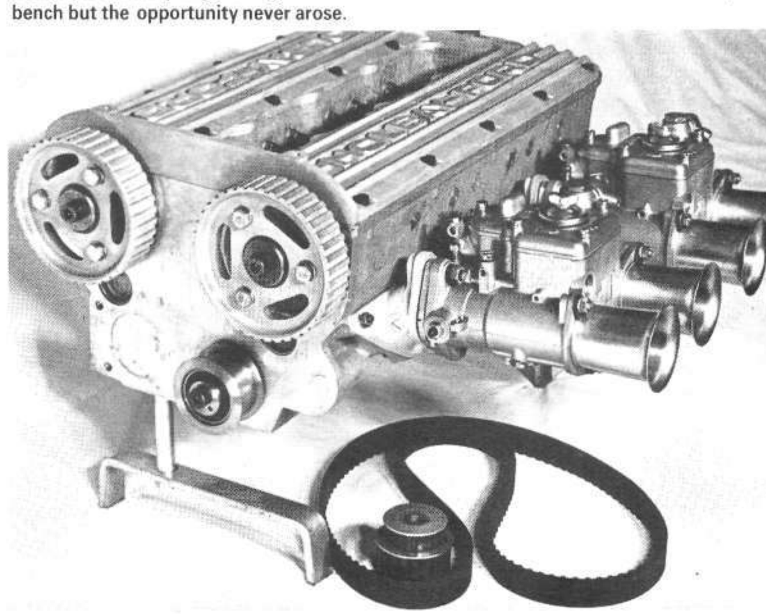
FIG 2-28

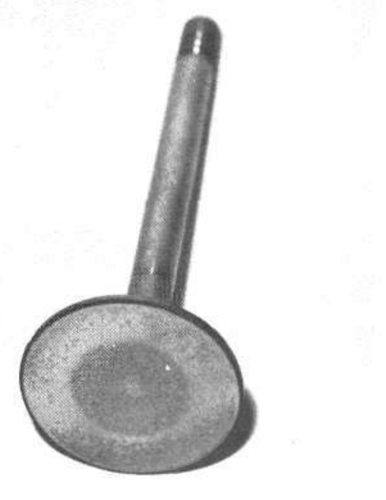




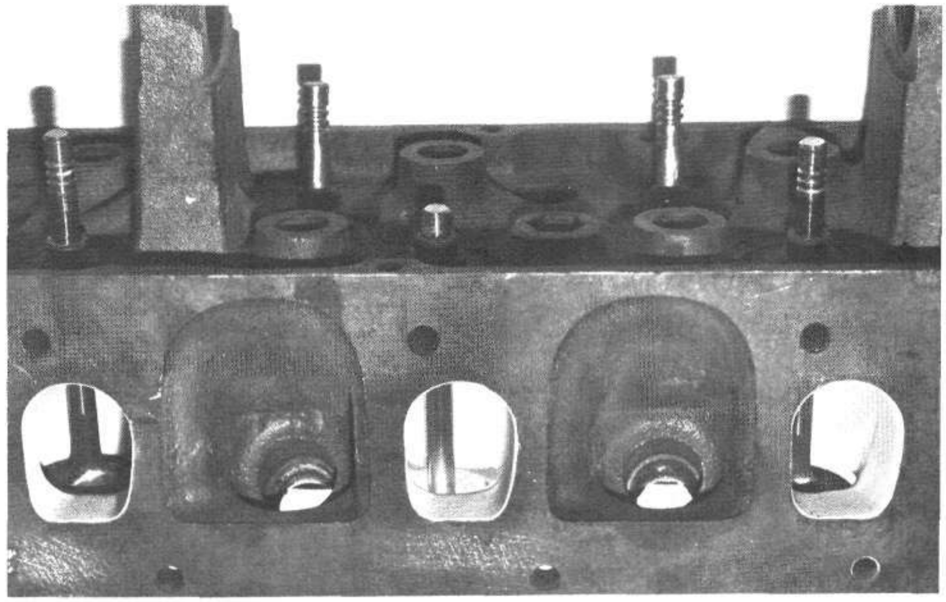
2-30. The recess on the intake valve acts as a means to reduce reverse flow, in short an anti-reversion lip. The shallow undercut to the face of the exhaust valve is intended to accommodate a thermal barrier coating.

2-31. Here's Holbay Engineering's 16 valve head. I would like to have run this over my flow bench but the opportunity never arose.





2-32. To stop the ceramic thermal barrier chipping around the edge of this exhaust valve, the valve face was machined, leaving a 0.015" (0.38mm) lip around the edge.



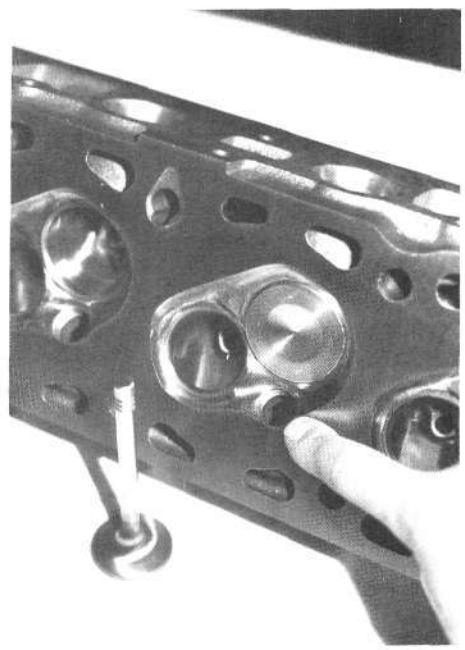
2-33. There is considerable rejection of heat through the exhaust port walls. Applying a thermal barrier here effectively eliminated cooling problems that occur on high hp Ford SOHC engines.

THERMAL BARRIER COATINGS

An engine develops its power from the rapid expansion of gases trapped in the cylinder. This gas is expanded by virtue of the heat produced as a result of burning fuel within the gas. Heat which excapes from the gas by conduction out through the combustion chamber walls and the piston crown is nothing less than discarded horsepower. If the heat can be kept in the cylinder where it can do work, cylinder pressures for a given quantity of fuel burned and a given compression ratio will be higher. Result: more horsepower and less heat rejection to the coolant and oil. At the time of writing, the only company I know of that applies heat-resistance finishes on automotive parts is Heany Industries. A ceramic-base coating known as Heanium is sprayed onto the components to approximately 0.012 inch (.3mm) thickness. As far as ceramic materials go, this coating is tough, but like most ceramics, it tends to crack at the edges. Cracking is not normally a problem but it is easy to cure. All that need be done to stop edge cracking, is to machine the surface to which the coating is to be applied so a lip of 0.012--0.014 inch (.3-- .35mm) is left around the edge. Photos of exhaust valves to be coated show what I mean. The same technique should also be applied to pistons. After the coating has been applied, smooth it with 400-grit or finer emery cloth or paper but take care. The function of the coating depends on it retaining as much thickness as possible. The smoothing operation must only remove the barest minimum of coating.

Analysing the advantages given by this coating reveals some interesting possibilities. Most obvious should be a horsepower increase. Coating the pistons alone can be worth between three and five percent more bhp. If the pistons and combustion chamber faces are done, add another one or two percent on top of that. If you have a race engine that you are struggling to get over 200 bhp, then just coating these parts could be worth up to another 10 bhp; that's something worth thinking about. The bhp advantage is the obvious one but not so obvious is what happens when this coating is applied to the exhaust valve. It will have a far lower thermal load. This means it won't have to dissipate as much heat through the valve seat to the head. Result: a narrower exhaust valve seat can be used. In turn this can lead to more exhaust flow which, of course, this head badly needs.

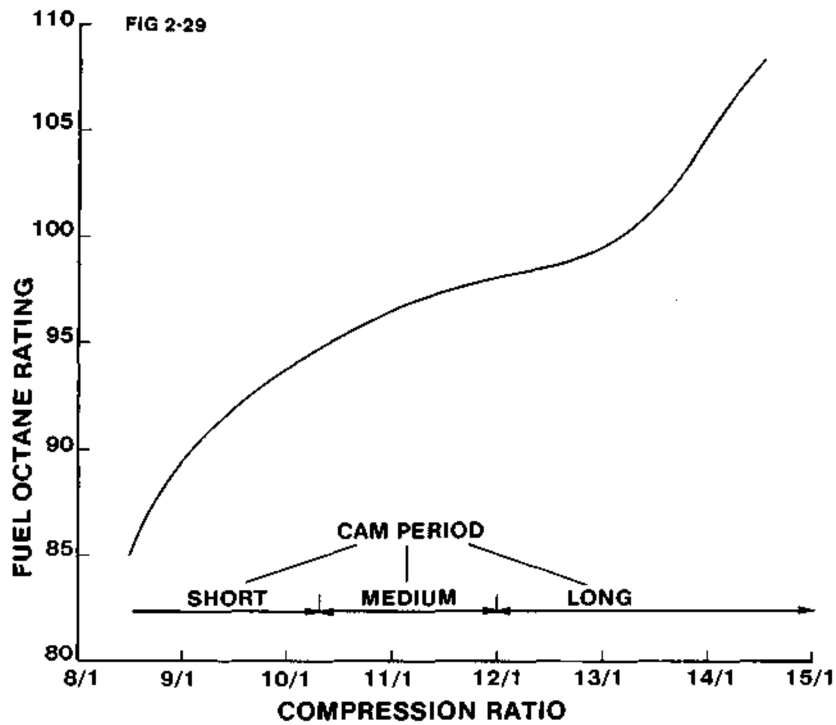
On the inlet side, another advantageous situation is arising. It's not commonly realized that a great deal of the heat picked up by the inlet charge as it enters the cylinder is done as it passes an intake valve. It can be running at temperatures up to 600° f. Coating the combustion-chamber side of the inlet valve can significantly reduce the temperature of the back side of the valve. As a result, a cooler, denser change can enter the cylinder. Because less heat is transmitted through the coated piston crown, the piston temperature is far lower. Aluminium alloys tend to lose strength very rapidly with increasing heat. Heat reduction on the piston is so significant that piston



2-33B. When large amounts of metal are faced from the head, the edge of the spark plug hole gets very close to the face. Do not break into the plug hole or leave sharp edges here.

weight reduction could theoretically be achieved using a thinner crown and adjacent areas, so as to produce a lighter piston. In turbocharged engines, the heat load through the piston is tremendous. This coating will reduce the heat load to about, or even below that seen by a normally aspirated engine, despite the fact that heat energy above the piston may be double that of a normally aspirated engine. If detonation starts, a normal piston can be eroded through in a matter of seconds; pistons coated with Heanium have shown they can withstand detonation for maybe 30 times as long before serious damage is incurred. In some instances, the use of this coating has meant the oil cooler has been an unnecessary item because so great a reduction in heat transfer from a piston to oil has been made.

FIG 2-29



COMPRESSION RATIOS

I have already dealt with small changes in compression ratio when discussing simple head modifications. I am now going into this very important subject in detail, so anything I said earlier must be tempered by what I have to say here.

First of all, let's establish one point of reference, namely the higher the compression ratio that can be used before detonation sets in, the greater the torque and power output of this particular engine. The question is, just how high can the compression ratio be taken before destructive detonation occurs? The principal points in question are: 1) fuel octane rating; 2) camshaft timing; 3) combustion-chamber design; 4) combustion- chamber temperature; 5) ignition timing.

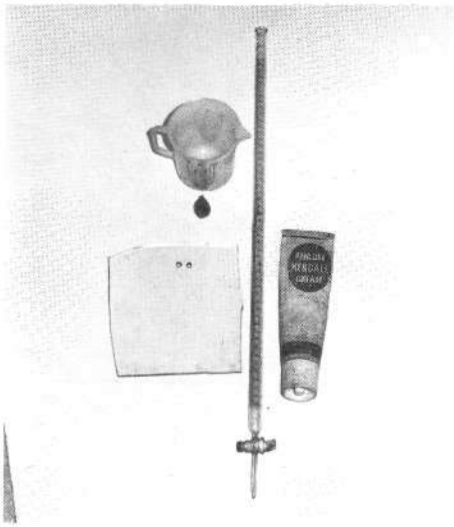
FUEL OCTANE

The octane rating of petrol used can range from the low 80s up to 105 plus octane of racing fuel. The availability of high-octane fuel is diminishing all over the world. 100-octane fuel is no longer available in the US or England, and no new passenger vehicles have been built in years for which its use is essential. Whatever compression ratio you elect to use in your engine, it must reflect the available octane rating. Because variables other than fuel-octane rating are involved, it is not possible for me to give you exact figures on just how much compression ratio can be used with any particular octane of fuel. The best I can do is to give you some guidelines. Fig. 2-29, covers possible compression ratios on a standard or near-standard engine at one end of the scale to race engines at the other. This graph is not made up of tests performed specifically on one engine; points on the graph were determined from a number of my own engines run at various compression ratios and fuel octanes. Use the curve as a guide. Don't expect to pinpoint the highest compression ratio your engine can use without problems.

CAMSHAFT

When we talk about the amount of compression ratio an engine will withstand before detonation sets in, what we are really trying to do is to numerically describe prevailing cylinder pressure just before ignition. Take a standard engine as an example: a 2000cc unit in good shape with a 9:1 compression ratio shows about 170 psi on a compression gauge. That same engine with 11:1 CR would probably show around 220 psi. The compression pressure that occurs depends not only on the compression ratio but *also* on the cam timing. The standard cam closes the intake valve only a short time after BDC. As a result, a cylinder which is close to being full, is compressed into the cumbustion chamber. A longer duration high-performance cam may not close the valve until the piston is 20 percent of the way up the bore. Result: only 80 percent of the total capacity of the cylinder is being compressed into the chamber, thus giving a lower pressure prior to ignition. The net result of a longer duration cam is that the effective compression ratio is substantially reduced by comparison with the geometric compression ratio.

With long-duration cams in a race engine there is a certain amount of dynamic filling of the cylinders stemming from the ram effect produced when engine rpm coincides with existing sonics in intake and exhaust tracts. As a result of this extra cylinder filling, the effective compression ratio moves nearer to the theoretical compression ratio. If the engine achieves a full 100 percent volumetric efficiency, the effective and theoretical compression ratios would coincide. To date I don't know of any Ford SOHC engines that have achieved 100 percent or more volumetric efficiency. On very sophisticated engines, the current state of the art produces volumetric efficiencies of around 94 percent at best. This means with long-duration cams, plus the less than perfect breathing ability of the head, the compression ratio must be very high to make any decent bhp. Duane Esslinger of Esslinger Engineer-



2-34. You will need these items to measure your engine's C.R. Starting at the lower left and working our way around clockwise, items shown here are : a piece of 1/4" thick clear plastic with two 1/4" dia. holes in it; a small bottle of food colouring dye, and a mixing jug; a burette and some grease to seal around the plastic sheet when it's placed on the head.

ing successfully uses 14.8:1R ratio on his championship-winning and record-breaking engines. This ratio would, of necessity, require 105-octane fuel. If you are running very long duration cams in your engine, unless the compression ratio is very high, your engine will not make any good bhp. A change of only one ratio may make the difference of 15 bhp with a 320° duration cam.

2-35. If you are cc-ing the chambers to determine the C.R., set the head CLOSE to level using a bubble level as demonstrated here by Carl Schattilly of C & G Porting.

When using a race-type cam, I suggest the use of one, perhaps one and a half, ratios higher than would be used with a street cam. Cams which are hotter than stock also require compression ratios proportionately higher than stock. For instance, you might expect a typical hot street cam to require a compression ratio of one-half to three-quarters of a point higher than shown in Fig. 2-29. It is also important to note that these figures are general in nature and can vary significantly with regard to cams, fuel octane, spark timing and pistons, as well as the subtler influences of intake air temperature, barometric pressure and such.

As a corollary to this, there exists such a thing as a cam too radical for a given compression ratio. For example, let's assume that you live in an area where 90-octane fuel is the best available. Under these circumstances, your engine cannot utilize a compression ratio high enough to take advantage of the cam's wilder characteristics. In short, you can't run enough compression to use what the cam offers. If this is so, your engine might well deliver more power with a camshaft having less duration, because of better cylinder filling at relatively lower speeds. In my judgement, you should figure on 97-100 Research octane fuel with a compression ratio of around 13.0:1 before attempting to use a grind of 300 degrees or more duration.

2-36. Smear grease around the edge of the chamber, then position the plastic sheet so that the holes are on the high side. This allows trapped air to escape while filling the chamber.

COMBUSTION CHAMBER DESIGN

Certainly one of the most significant determinants of maximum usable compression ratio is combustion chamber temperature. Now the heat values here are not always static or, to the eternal consternation of some, predictable. A cold engine having a compression ratio of, say, 15.5:1 might burble along happily indeed, without a hint of detonation. However, as that same engine warms up, its combustion chamber temperatures will rise as well, thus opening the door to detonation. Furthermore, an engine that gets along well with its compression ratio under normal conditions will likely prove highly susceptible to detonation when it becomes overheated, as when, for example, the cooling system malfunctions. Or, another slant would be the engine that operates acceptably under partial throttle openings but detonates destructively at larger throttle settings, preventing anything resembling full-throttle operation.

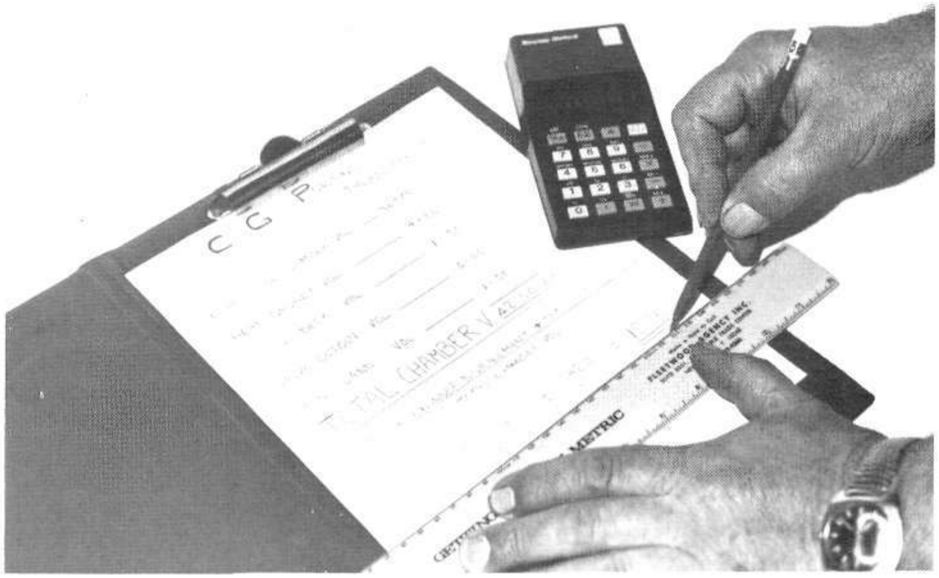
On the other hand, a chamber can be too cold for efficient combustion and power. Fuel within a relatively cold chamber can condense on wall surfaces, or it might never realize atomization in the first place, neither condition helps engine power, while causing exhaust pollutant levels of unburned hydrocarbons to soar.

An excessively cool chamber can also delay the vaporization of fuel until too late in the combustion cycle, which also contributes to poor engine efficiency and high pollutant readings at the exhaust pipe.

And if all this weren't bad enough, a cold combustion chamber also hurts efficiency by exacting from the ignited charge a disproportionate amount of heat, which results in compromised power.

Another factor affecting maximum usable compression ratio is intake-charge temperature. This in turn depends largely upon how much heating of the intake port, manifold and carburettor has taken place. A cool charge in a hot chamber will stand more compression than a hot charge in the same temperature chamber. Intake-charge temperature depends mostly upon engine running time. A 12-second drag-race engine isn't likely to heat the intake system appreciably. On the other side of the coin, the engine of a road race or rally car may be run hard for hours on end. On a long-distance race engine, figure on a compression ratio about one-half to three-quarters of a ratio lower than its drag racing equivalent.

Squish or quench area affects the amount of compression that can be used before detonation occurs. Indications are that too much squish exists in the 2000cc head. Remembering that our aim is to produce bhp, not compression, it is permissible to cut away some of the squish if it results in more power. Reducing the squish area as detailed earlier, when I dealt with chamber modifications, achieves the desired effect with the minimum amount of metal removal. Removing the minimum amount of metal from the chamber is an important factor if you are aiming for an ultra-high compression ratio. On 2000cc engines, to get more than 14:1, the head face may need machining up to a safe maximum amount of 0.165 inch (4.2mm). If too much metal is removed from the chambers, you may not be able to get the ratio you desire. Heads with welded chambers, as per the design shown earlier, are not such a problem. To achieve 14.8:1, such heads generally need 0.120--0.130 inch (3.05--3.3mm) machined off the head face.



2-37. To make sure you calculate your C.R. correctly, make out a sheet like this just as the professionals do.

2-38. The tricky part of determining how much has to come off the head is measuring the distance between the head face and the fluid. Molecular attraction makes accurate measurements difficult.

ECONOMY

With few exceptions, modifications which work well to produce extra bhp from the engine can also help economy. One of the best and simplest modifications to improve economy, is to raise the compression ratio. Most driving is done at part-throttle. Raising the C.R. about two points might increase fuel economy by as much as three mpg. Big exhaust valves and free-flowing exhaust ports also help. Such valves and ports reduce *pumping losses*. If your engine has a restrictive exhaust system, and by this I mean the entire exhaust system from valve to tailpipe, the effort required by the piston to push out exhaust represents horsepower lost from the power stroke. In other words, fuel is being burned simply to get rid of the exhaust. With a free-flowing exhaust system, pumping losses can be reduced to a bare minimum.

One of the few areas of head performance modifications that doesn't usually enhance fuel economy is the inlet port. Although I have not tried it out on the Ford 2000cc engine, my experience with a lot of other engines has been that fully reworked intake ports cause a small reduction in fuel economy. This means the best economy intake port is probably a standard intake port and valve. All those rough lumps, bumps and edges are good for breaking up the fuel and that's just what they do. And that helps economy.

Further economy gains may be possible if inlet port size is reduced. Remember, the bottom part of the port is not a working area. Filling in the bottom of the port could help economy, but I have not tried it.

MODIFIED HEADS & EMISSIONS

Most people are aware that the octane rating of fuel has been coming down over the past few years, especially in the U.S. Few people, however, realize exactly why. Basically there are three pollutants government require manufacturers to control: These are carbon monoxide (CO); hydrocarbons (HC) that's unburned fuel; and oxides of nitrogen (NOX) The first two pollutants can be controlled relatively easily by attention to design detail of the engine and regular attention to keep the engine in a good state of tune. Carbon monoxide is dependent on the mixture ratio. If the engine is run with a lean mixture, carbon monoxide can be almost eliminated. Hydrocarbon pollution is not quite as easy to cure. It depends on how effectively the engine burns fuel. Large, cool surfaces in the combustion chamber tend to cause fuel deposited on these surfaces to evaporate too late in the combustion cycle to be burned. As a consequence, these evaporated fuel droplets are emitted as unburned hydrocarbons. Now, running the engine hotter than normal (so the combustion chamber surfaces are much hotter, too) usually corrects this. Ceramic-coated pistons and chambers are said to be a help in this area, because they tend to transfer less heat away from the chamber.

But probably the most dramatic improvements can come from using a *completely evaporated* air/fuel mix. While achieving this can be somewhat complex in practice, the results are too good to ignore, for in some tests unburned hydrocarbon levels have been reduced to almost nothing at all.

Also, while not promising to eliminate unburned hydrocarbons, carburettors that will deliver a well balanced, highly atomized mixture have prove to be so effective that their use as original equipment is becoming more and more commonplace.

Reduction or removal of CO and HC pollutants is not totally compatible with high-performance requirements. However we can live with de-toxing requirements without sacrificing too much power. To a degree, there are ways and means of making up for certain power losses. In terms of road vehicles at least, meeting tight HC and CO requirements can indeed help fuel mileage.

Unfortunately, the big problem as far as high-performance engines are concerned is the oxides of nitrogen or NOX. This pollutant is formed in the combustion chamber as a result of high temperatures and pressure, which incidentally, are just the conditions commonly required by a high-output engine.

Indeed, to get best power requires relatively high compression ratios in our engines. So, while the dictates of horsepower production say we must, the government says we mustn't. And if that's not enough, there is another thermo-chemical nail into the high-compression coffin. The compound our engines need to operate with high compression ratios is a fuel additive called tetraethyl lead. It is the performance buff's misfortune that this compound is woefully incompatible with the heaviest gun in the anti-smog arsenal, the catalytic convertor. Tetraethyl lead poisons the convertor so it no longer works. The catalytic convertor is essential for scrubbing NOX from exhaust gases, so the lead's gotta go!

High-performance, high-compression engines cannot run on low octane lead free fuels. As performance enthusiasts, we need to look around for an alternative means of utilizing high CRs, an alternative which we should, as a matter of course, install on our engines to keep pollution levels down. Fortunately there is another means of doing this and that is by water injection. Before building yourself a high-compression, road-going engine, for use in areas with stringent emission requirements, read Chapter 9 which deals with water injection.

HEAD MODIFICATIONS FOR RELIABILITY

VALVE GUIDES

A modification shown earlier which definitely helps airflow involves shortening the intake valve guide. Unfortunately, though, this also has the undesirable effect of shortening the guide life. Indeed, shortened guides working with even moderate cam profiles will last 10,000, maybe 20,000 miles before they are completely worn out.

There are several techniques for maximizing guide life that are used by various racing emporiums with comparable success. One method favoured by such firms as Manx Racing, Swaymar Race Engines and Janspeed is to install bullet-tapered, bronze valve guides. Using these units there is a nominal trade-off of airflow for increased life but this trade-off is largely problematical. Only a small airflow reduction occurs at very high valve lifts.

Importantly, there is another technique (used mostly by U.S. builders) that works well and has no trade-off disadvantages. It is based upon the use of the thin-wall K-line guide liner. This is a bronze sleeve of either .015 or .030 inch in thickness. The existing guide is enlarged to accept the new sleeve, and then the sleeve reamed to final size. (The use of these guide liners is dealt with in my H.P. book *How To Rebuild Your Ford SOHC Engine*. These units can be fitted prior to grinding the port. The end of the guide conforms exactly to the port profile which is normally at an angle of about 30 ° in this area.

It is important to note that, when cutting valve seats, the angled end of the guide will throw off concentricity if a press- fit pilot is used.

You can rectify this by removing a few thousandths of an inch of material from the bore of the valve guide, so that the sized portion of the guide ends horizontally.

Performing this simple but often overlooked chore ensures that the pilot is true to guide centre line and seat concentricity is constant. Fig. 2-30 explains.

VALVES

Profuse quantities of oil are required to lubricate the 2000cc engine's top end. There is a lot of oil sloshing around up here, which contributes to a potential problem of oil control in this area. To prevent oil consumption, very tight valve seals are used. To a degree, this can lead to increased valve wear.

If you want maximum life, choose a valve with a chrome-plated stem. Many oversize valves have stems longer than standard. Typically .050 inch is added to the stem. This is done to maintain acceptable rocker adjustment with the small base circles commonly encountered on replacement performance camshafts. Extra valve length reduces valve lift and also reduces *effective* duration of the cam. As a result, the valves may have to be shortened to achieve the lift and duration figures the cam grinder intended. (Before assembling your head, read my comments on valve train geometry in Chapter 3.)

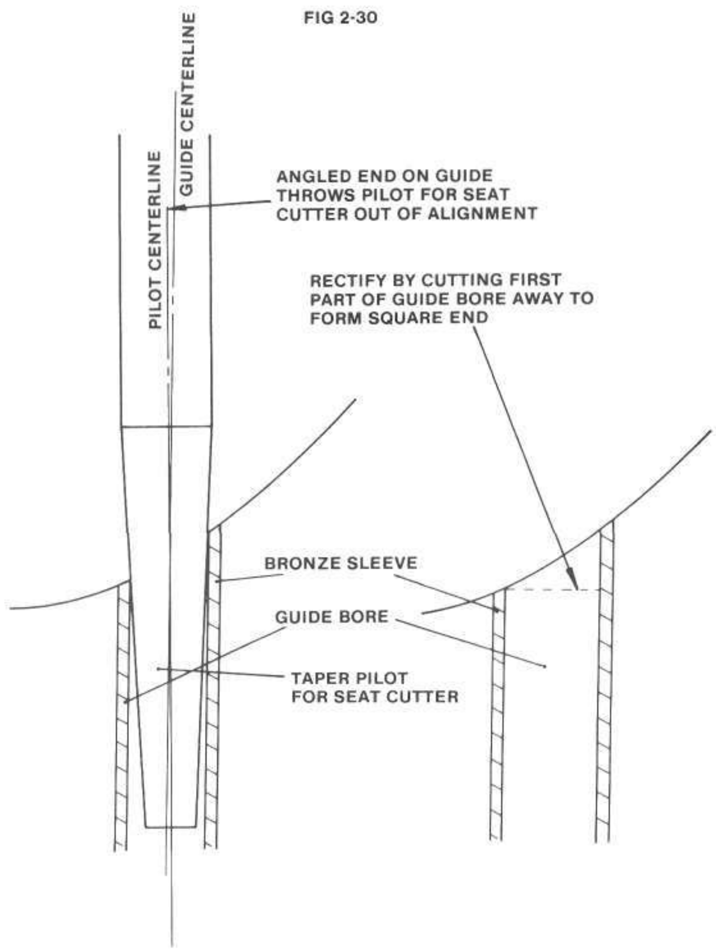
ROCKET PIVOT BOSSES

Installing a high-lift cam in a Ford SOHC engine often means installing a cam with a smaller-than-stock base circle; increasing the lift reduces the base circle. This means the rocker pivot pillar must be adjusted upward and each time you adjust it up, there remain fewer threads engaged in the head. By the time you reach valve lifts of about .500 inch (12.7 mm) there are too few threads engaged to secure the pillar. In turn, side- loads on the pillar pound the threads, which leads to prying open the threads in the top of the boss. The upshot of all this is that the lock-nut has an unfortunate tendency to come undone.

With a lift of about .550 inch, there may be only two or three threads engaged. These will pull out the first time the engine is started if stiff valve springs are used.

The cure is to use Racer Walsh press-in inserts. These are installed in the head by boring out the existing thread and pressing in these inserts. Caution: do not use more than .0005 inch (.03mm) press fit, or the boss will crack. This modification raises the height of the boss by about .200 inch and ensures sufficient thread engagement for cams right up to .620 inch lift.

FIG 2-30

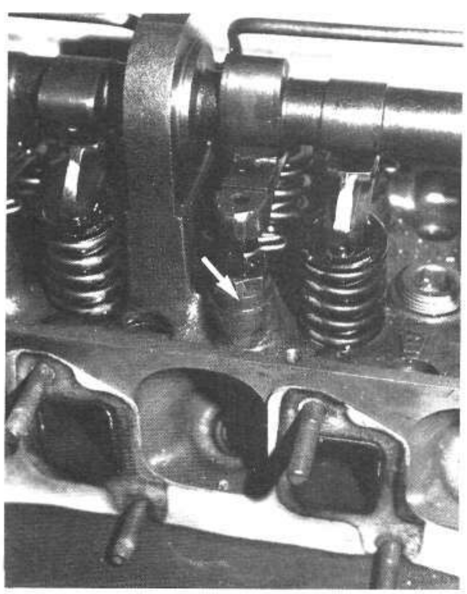


CAM BEARINGS.

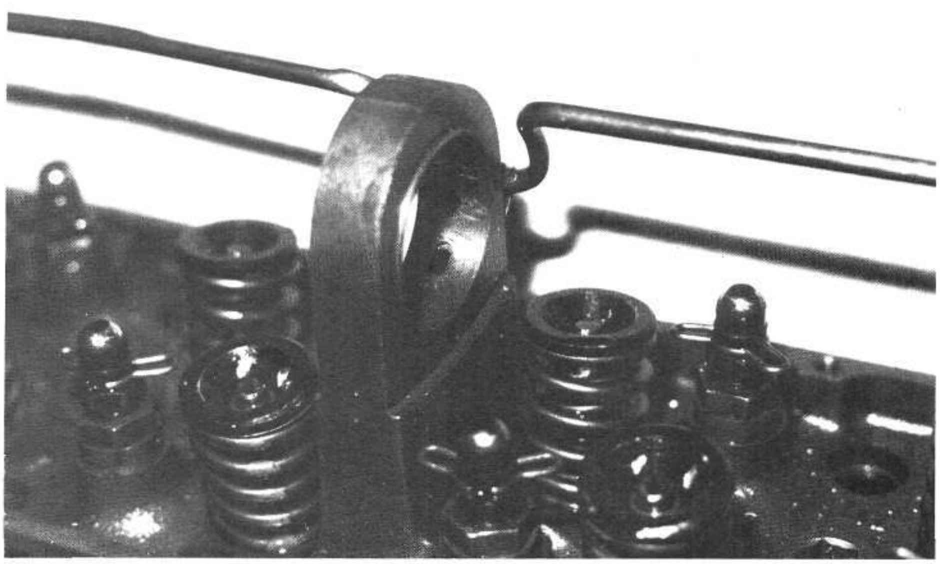
The centre cam bearing on these engines takes a beating. Inspection of a standard bearing in a standard engine with 30,000 to 40,000 miles will find that in 9 out of 10 cases it has worn quite a substantial groove in the top side of the bearing. Using a high-lift cam and heavier springs aggravates the problem.

There are numerous cam bearings on the market and most of them are

2-39. When high lift cams are installed, most, if not all of the rocker pivot pillar adjustment is used up. This can be comensated for by raising the height of the thread into which it engages by means of these threaded inserts.



2-40. This head has a bronze centre cam bearing installed. Such bearings give much longer life when high lift cams and heavy springs are used.



hard pushed to last 10,000 miles in an engine having even a moderately warm cam the valve train. The best bearing, I feel, is the standard Ford bearing. Buy it from your Ford dealer and make sure it's in a Ford box. This bearing will generally run 40,000 to 50,000 miles in a sanely cammed street machine.

For race engines, this cam bearing can be a source of problems. Normally, it's only the centre cam bearing that wears, because it carries twice as much load as the other bearings. For a circuit car with a high-lift cam, the standard bearing may last only a race or two. If really high valve lifts and heavy poundage springs are used, that is, 0.600 inch with springs of 120 pounds seat and 300-350 full lift loads then under such trying circumtances you can expect the bearing to last sometimes as little as 20 seconds.

However, there is a reasonably simple solution to this problem: make the bearing from phosphor-bronze. This will provide a life of several hours with high lifts and heavy valve springs.

Now, for most racing applications, this is fine. Nevertheless, certain precautions must be taken. Chief among them is the fact that you have to use an oil or oil additive with zinc dithiophosphate in it, otherwise the bearing has a good chance of seizing.

One of the best additives you can get for this application is General Motors' Engine Oil Supplement, otherwise known as EOS. Ford makes an equivalent called Ford Oil Conditioner and Crane or Piper cam lube will also get the job done. Such additives are essential to *both* cam and bearing life.

If you want the ultimate in cam bearings, then your best bet is to use the Holbay roller bearing conversion. This involves line boring the camshaft bearing housings in the head and installing roller bearings. To be successful, roller bearings need to run on a reasonably hard surface. To do this means running with a steel-billet cam. As you would expect, Holbay make a steel billet cam for use with their roller conversion. In case you feel limited to the use of a Holbay cam profile if you intend going with roller bearings, let me point out that companies such as Crane Cams, Competition Cams, Crower Cams and Sig Erson Cams, to name but a few, all make steel billet cams to special order. Before installing roller bearings in the cam towers of a Ford 2000c head, you will have to decide whether you are going to run conventional finger followers or roller followers. If roller followers are used on the cam profile, then the requirement for copious quantities of oil in the top end of the engine is eliminated and the oil feed to the cam tower can be restricted so that there is little more than oil mist lubrication. If you intend to use roller bearings in the cam towers but conventional finger followers, then you will need to supply just as much oil to the finger followers as was previously the case, even though the roller bearings could get by with a minimum of oil. With the amount of oil that comes out of the finger follower lubricating holes in the spray bar, there is more than enough mist flying around to lubricate the roller bearings.

The easiest way to restrict the oil supply to a roller bearing is to grind a dimple in the slot in its outer diameter about half an inch on both sides of the hole, which connects the outside of the bearing with the inside. The dimple is there merely to key a blob of resin so it doesn't move. If it does move, oil pressure will force it into the bearing and there goes a roller bearing. Once the groove has been filled with epoxy resin, and this must be done in totally grease-free conditions to ensure adhesion, file the outside of the resin so it conforms to the curvature of the bearing case.

When installing the bearing, do so with the oil hole on the side opposite the spray bar. The bearing slot joins the oil-supply hole with the spray bar feeding hole, so oil can be supplied to the rockers.

2-41. Roller cam assembly on Holbay modified head gets around the excessive wear problem of the centre cam bearing.

VALVE-STEM OIL SEALS

Because of the tremendous quantity of oil required to lubricate the cam lobes and followers, the valve-stem oil seals have a far harder job controlling oil than on most other engines. Problems occur when oil seeps past the guide, contaminating the intake charge with sufficient oil to cause detonation. If the engine is required to cover high mileages, it can also cause carbon deposits to accumulate on the back of the valve. This can seriously impede port flow, and thus power output.

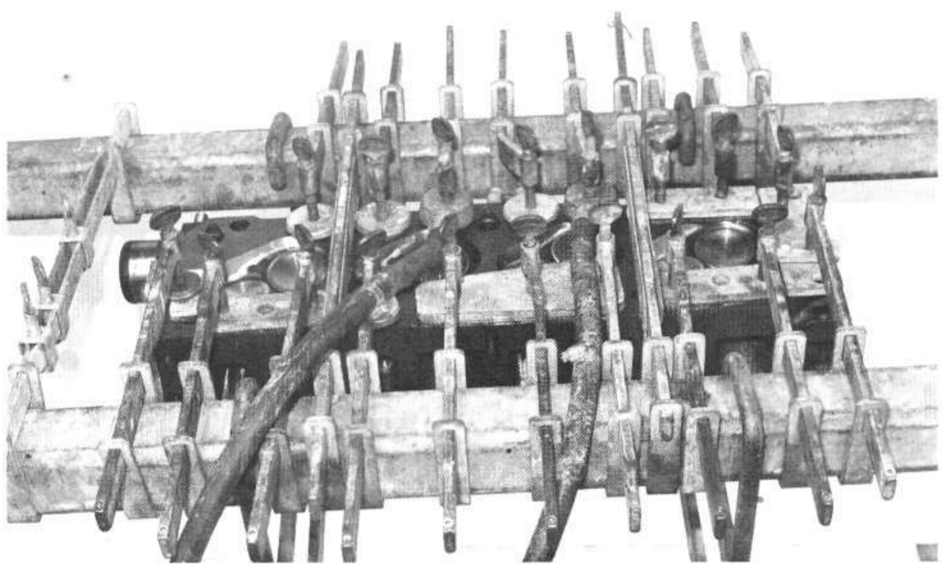
That is the first problem valve-stem oil seals have to cope with. The second one is associated with valve lift. When valve lifts get up to and around .500 inch, the seal can be pounded by the underside of the spring retainer. This quickly pulverizes the oil seal into uselessness and the guide-oiling problem returns instantly. Many seals made for the Ford 2000cc do an adequate job as far as oil control is concerned, but most, however, have an installed height that makes if difficult to use high-lift cams without interference. Felpro offers one of the few stem oil seals compatible with high-lift cams. These oil seals add very little to the height of the guide; it is possible to use valve lifts up to .580 inch before the underside of the valve spring retainer is close enough to cause oil-seal damage.

HEAD GASKETS

In my experience, the standard Ford head gasket is one of the least reliable such units available. I have had these blow on engines with as little as 130 bhp, although that may have been a chance occurrence. I have had several fail on engines with outputs around 150-160 bhp. All the aftermarket gaskets I have tried have done better than this, but I have not tried every available gasket. I've found the most reliable head gasket to be the Felpro item. It appears to be quite capable of holding power outputs of about 180 or so without drastic unreliability. It also appears to be good for withstanding 11 or 12 psi boost on a turbocharged engine. However, at about 15 pounds, even the Felpro gasket lets go. Felpro gasket aren't readily available in England but could be purchased through such establishments as John Woolfe Racing. Holbay Engineering market a gasket which they claim is entirely satisfactory up to a compression ratio of 10.3:1. I have never used this gasket but the Holbay name is for some engine builders sufficient to guarantee quality merchandise.

When an engine reaches internal pressure levels where all conventional gaskets fail, we have to look at alternative means of sealing cylinder pressures. The simplest way is to 0-ring the head or, preferably, the block. Ideally, this should be done with the engine out of the car and stripped to the bare block so that all machining swarf can be cleaned out after the job has been done. However, this is not always convenient, so some companies will groove the head to take 0-rings rather than the block. This works almost as well but sometimes you can run into problems with gasket misalignment. As a result, the 0-rings do not bear in the precisely right position on the head gasket. The type of 0-ring most commonly used is a piece of 0.030 inch (0.075mm) diameter copper wire. This is installed in a machined groove which

2-43. This is the Holbay head gasket which is claimed to be reliable up to 10.3/1 on normally aspirated engines.



2-44. Under this conglomeration of bars and clamps lurks an extensively modified 2000 head. The reason for all this paraphernalia is that the head is being pressure-tested to make sure it won't leak due to the large amounts of metal removed from certain areas.

is 0.015 inch (0.38mm) deep. The protruding part of the 0-ring bites into the gasket, causing a very high local pressure and this almost eliminates gasket blowing.

If you prefer the grooved head route you will have to take the head along to a machine shop to have the work done. If it's a machine shop that doesn't normally do such jobs, take along a head gasket as a template for positioning the 0-rings to suit the head gasket.

If you decide to 0-ring the block, then Iskenderian makes a tool which makes grooving the block a simple job for the home mechanic. (See Chapter 7) When selecting the diameter of an 0-ring groove, it has been normal practice to cut the groove to coincide with the centre of the head-gasket bead (fire-ring). Recent experience indicates however, that a better seal is possible when the 0-ring groove coincides with the outside diameter of the head gasket bead around the bore.

So far as reliability is concerned, an 0-ringed head gasket irrespective of the head gasket, involved, seems capable of handling around 250 HP. With Felpro head gaskets, reliable performance in excess of 350 bhp is possible.

On their own race engines, Holbay use yet another sealing system of proven reliabilty. It is an outgrowth of the 0-ring method. Instead of using a conventional head gasket as a basis for sealing, they use a head gasket which seals only water and oil passages. The areas around the bores are omitted. Sealing of the high pressure gases in the cylinder is achieved by a gas-filled stainless-steel ring commonly known as a *Wills ring*. Either the block or the head can be machined to accept these rings. Holbay can supply the special head gasket and the four rings required to use this sealing system. Along with this, they also supply a drawing showing the required machining dimensions. If specified tolerances are adhered to, this system appears to be the nearest thing to 100 percent reliable.

BUYING A MODIFIED HEAD

To avoid buying a dud head, there are a number of things you can do. There are many good head-grinding companies which do not know about SOHC Ford heads simply because they have no experience with this type of engine. In this instance, take this book along to such a company and have them grind your cylinder head to your specifications

If you are buying a head not necessarily produced to any of the specifications in this book, buy only from companies with a proven record on these engines. If you find you cannot readily locate a company of proven capability then deal with one of the companies mentioned in this book. Companies such as Esslinger Engineering, C & G Porting, Holbay, Swaymar, Manx Racing and a few others know what is needed. But be warned, many more do not. Also, just because I have not mentioned a particular company within these pages does not mean it is incapable. What it does mean is that I am leaving you to decide whether they are capable of producing what you require.

A FEW FACTS ABOUT AIRFLOW MEASUREMENT

Many of you will not be familiar with flow bench techniques and you may wonder why two formulas were used to estimate the horsepower output from the intake-port airflow potential. One formula was for 25 inches standard pressure drop and the other was 10 inches standard pressure drop.

What is meant by standard pressure drop? When we attempt to measure airflow, it should be taken into account that we are dealing with a highly expansible and compressible gas. The situation is much like trying to measure a piece of elastic with a ruler. You will notice that the standard pressure drop is always quoted on the graph or chart. This is the amount of suction or pressure used to make the air flow through the port being tested. More air pressure or vacuum, depending on whether it's an intake or exhaust being tested, would flow more air. But it must be remembered that the idea is to get more air through the ports due to port improvements, rather than to see how much we can squeeze through due to increased flow bench output. Going back to the elastic analogy, we can say that fixing our pressure drop is about the same as putting in a certain amount of tension into the elastic. This means our ruler measurements will reflect whether we have a longer or shorter piece of elastic, not its ultimate elastic capability.

The procedure for air-flowing heads is far from standardized, and test pressure ranging from five inches H_20 to five inches mercury (Hg) (67.7 inches H_20) are used. Generally speaking, the vast majority of flow benches operate between five inches and 30 inches H_20. If you want to compare cfm figures done on the bench at one test pressure drop with another bench at a different pressure you must correct all figures to one standard test pressure drop. This is done by applying the following formula:

$$\sqrt[2]{\frac{\text{test pressure being used}}{\text{test pressure being converted.}}}$$

Let's do an example. Say we want to compare cfm figures of one head measured at 25 inches and one measured at 10 inches. Convert the 10 inch figure to 25, or the 25 to 10. For the purpose of this example I will change the 10 to 25. So we have

$$\sqrt[2]{\frac{25}{10}} = \sqrt[2]{2.5} = 1.58$$

If all flow figures measured at 10 inches are now multiplied by the 1.58 factor, they will be directly comparable to those measured at 25 inches.

HOW TO MEASURE AND SET YOUR COMPRESSION RATIO

What is the compression ratio? It is the ratio of the *total* volume above the piston at BDC divided by the total volume above the piston at TDC. To define the total volume above the piston at BDC we must first establish the swept volume of one cylinder. This is calculated from the formula

$$\frac{B \times B \times 0{\cdot}785 \times \text{Stroke}}{1000}$$

where all dimensions are in millimetres. After determining the displacement of one cylinder, your next job will be to calculate the volume remaining in the block above the piston when the piston is at TDC. If you make all the measurements in inches, the formula will be

$$B \times B \times 0.785 \times DH \times 16.39$$

and the answer this produces will be in cubic centimetres. If you take all the measurements in millimetres, the formula to use to get the answer in cc will be

$$\frac{B \times B \times 0.785 \times DH}{1000}$$

In these formulas, B = the bore diameter and DH = the deck height, that is the amount the TDC piston is below the block face.

The next step is to calculate the volume of the gasket. To do this we use the same as previously, but B will be the bore of the gasket and DH will be the *compressed* thickness of the head gasket. The only other calculable volume remaining is the volume between the top edge of the piston crown and the top ring. This is normally 1–1½ cc, depending upon the piston design. To give yourself room for measurement error, assume this ring land volume to be 1 cc. If it's slightly more, your compression ratio will be fractionally lower than you calculate it to be. This puts you on the safe side of the fence.

We have now covered all the easily calculable volumes. From here on, a burette or pipette will needed. Volume measurements will be done by pouring fluid into the spaces to be measured. Here, accuracy is essential. At the highest ratios an accumulated error of 1 cc could mean as much as 0.4 of a ratio off the intended figure. Effects of surface tension can upset measurements unless precautions are taken. An easily brewed mixture for the burette or pipette to minimize errors is 80 percent water and 20 percent alcohol and two or three drops of food colouring. If you have to fill the measuring instrument by sucking it in your mouth, *forget the poisonous alcohol* and substitute liquid soap.

Turn your attention to the cylinder head. Install the spark plugs, then set the head on blocks such that it is just a little way off being level. Apply a smear of grease to the valve seats to effect a seal and install all the valves in the correct position. Smear grease around the chamber to be checked, then press a sheet of glass or plastic with a hole in it over the chamber to be checked. Position the hole at the highest point of the chamber so air displaced by the entering fluid can easily escape. Fill the chamber to the bottom of the hole and note how many cubic centimetres of fluid it took to do so. Do this with all the chambers to see they are all the same within ½ cc. That ½ cc is a little more than the limit of accuracy with this method of measurement. If any greater difference exists on a finished head it should be rectified by enlarging the smaller chamber to the volume of the largest. Once the chamber volume is established, use the same technique to measure the volume of any valve cutouts the pistons may have. With this chore done, you are in possession of all the figures you need to make final compression ratio calculations from the following formula:-

$$CR = \frac{\text{Cylinder displacement} + \text{total combustion chamber volume}}{\text{total combustion chamber volume}}$$

The cylinder displacement we have already calculated. The chamber volume is the sum of all the small volumes that were measured or calculated. This figure is obtained by adding together the volumes of the combustion chamber, head gasket, piston-to-deck clearance volume, valve cutouts and the ring land volume.

MACHINING THE HEAD FACE HOW MUCH TO REMOVE

So far all you have been told is how to measure the existing compression ratio. The burning question for most hotrodders is, "How much must I machine from the head to raise compression to a new level?" Here the problem must be worked the other way around.

Okay, so you know the ratio you want. The unknown factor and the one you wish to change is the volume of the combustion chamber. As in the previous case, you need to know cylinder displacement and the volumes contained in the valve cutcuts, piston-to-deck clearance and head gasket. These *parts* of the *total* chamber volume are *fixed*. Subtract them from the *total* chamber volume required. This leaves the volume the combustion chamber must contain to give the desired ratio. First of all, the forumla for the total chamber volume:

$$\text{Total chamber volume} = \frac{\text{cylinder displacement}}{\text{compression ratio} - 1}$$

Example: an 11:1 compression ratio is required on a 1600cc engine Total chamber volume to acheive this is

$$\frac{1600\text{cc} \div 4}{11 - 1} = \frac{400}{10} = 40\text{cc}$$

Say that 12cc of this volume is contained in the piston/deck volume, valve cutouts, etc. To achieve the desired ratio, combustion chamber volume must be 40 – 12 = 28cc. The head must be machined until it contains 28cc in the combustion chamber to produce the required 11:1 compression ratio.

To determine how much must be removed from the head face, we must first set the head as close to perfectly level as possible; a spirit level is a valuable tool here. With plugs and valves in place, adjust the head level with shims until the head face is level. Using a burette or pipette, carefully fill the largest chamber with fluid of the exact volume required of the finished chamber. Place a rule or straight edge over the chamber and measure from the edge of the rule to the surface of the fluid with the tip of a vernier calliper as shown in the photo. Subtract the width of the rule from the measurement just taken. The figure left is the amount the head face must be machined. The procedure is simple but be aware of two things: when the end of the dial or vernier calliper gets within 0.005–0.007 inch (0.13–0.18mm) of the fluid surface, the fluid will rise to meet the end of the calliper. This phenomenom is known as "Molecular attraction". This means precise measurements are difficult. In practice, you usually end up takeing a speck less off the head, because in reality the measurement comes out a little short of what it should be.

On a road machine, where the very last bhp isn't that important, use this factor as a safety margin. On a race head where the compression ratio is required to be on the dot, have the head machined by the minimum amount indicated by measuring. After measuring, check the ratio again. If it's less than you require, machine a few more thousandths from the head face and recheck volume. If you are attempting to produce very high ratios without overstepping the mark, accuracy is *essential*. As little as 0.002 inch (0.05mm) can make a 1/10 ratio change. If you are already working on the limit, that tenth of a ratio could mean severe detonation problems. Whatever time it takes to check and double-check, do so – it may save you a lot of time and trouble later on.

Cam Drives, Cam & Valve Trains

In this chapter I begin my discussion at the crankshaft and work my way up the engine through the cam-drive system to the camshaft, and then ultimately to the valves themselves.

Let's start with drive belt reliability. The belt that drives the cam rarely breaks, but this doesn't mean it never does. Three things, or a combination thereof, cause belt breakage. First is age. If the belt has seen high mileage, then it could be ready to break depending on the climate. I have seen belts break at 20,000 miles in hot climates while in cooler climates, belt breakages may not occur until 80,000 miles or so.

The second factor affecting belt breakage is valve spring pressure. Heavy non-standard valve springs impose more stresses on the belt and lead to earlier breakage.

A third factor is engine location. Many Ford OHC engines find their way into off-road machines, most using a Volkswagen transaxle with the engine mounted at the rear. If this is the case *you must* fit some sort of deflector to prevent sand, grit and stones from being caught between the crank sprocket and the belt. Do not take this point lightly. You may not make 20 miles in a competitive off-road event without adequate belt protection. And when the belt breaks, bent valves result. It is quicker to fabricate a shield to prevent this than it is to pull the head and replace valves and maybe a piston or two.

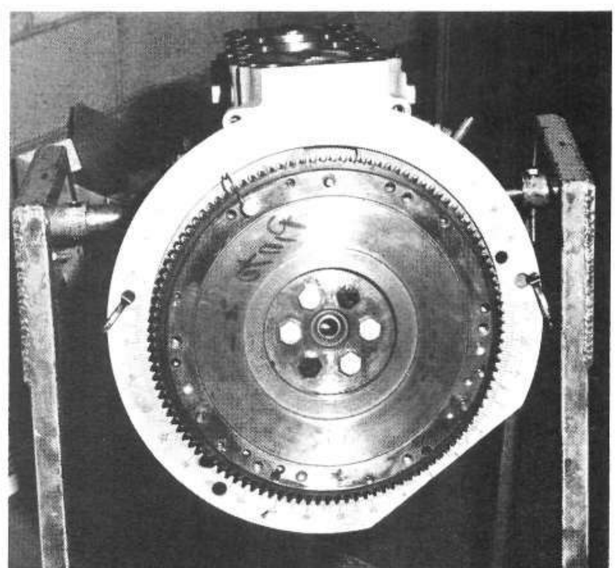
Based on my experience, I suggest you use the following recommendations as a guide. The standard Ford belt is the best quality available and I have tried most brands but not all. Of the brands I've tried some have had a disappointingly short life on race engines, on occasion as short as 10 minutes. Granted the engines in question did have valve spring pressure in excess of 300 lb. at full open, but this is the sort of spring typically used on a high-performance SOHC Ford engine. If you live in a very hot climate such as the southwest U.S.A. or central Australia, then on street engines with valve springs any stronger than standard, it's advisable to change belts at 30,000 miles. If it's a race engine, do yourself a favour and change belts every 1000 miles in such hot climates.

In cooler climates such as in England, belt breakage seldom occurs except when very heavy spring pressures are used.

CAM SPROCKET.

The cam sprocket offers an easy way to "dial in" the cam. This can produce the best overall cam timing for your particular engine tune. Cam timing can be altered by filing the slot in the sprocket and taking up excess slot width with shims, or by using an adjustable cam sprocket. I have personally used Burton Engineering, Holbay, Esslinger Engineering and Racer Walsh adjustable sprockets. I have also used one or two types which were of flimsy design and shoddy workmanship. When selecting an adjustable cam sprocket, assume that the bulkier they look the better they will withstand repeated cam timing alterations.

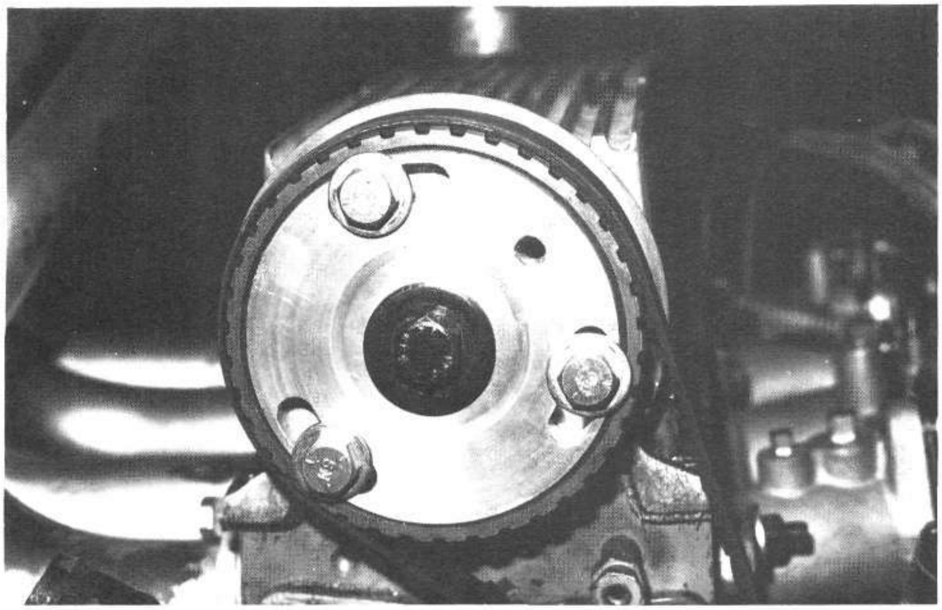
Sprockets produced by Burton Engineering are produced to a very high standard and utilize a vernier design to allow cam timing changes in precise 3/4

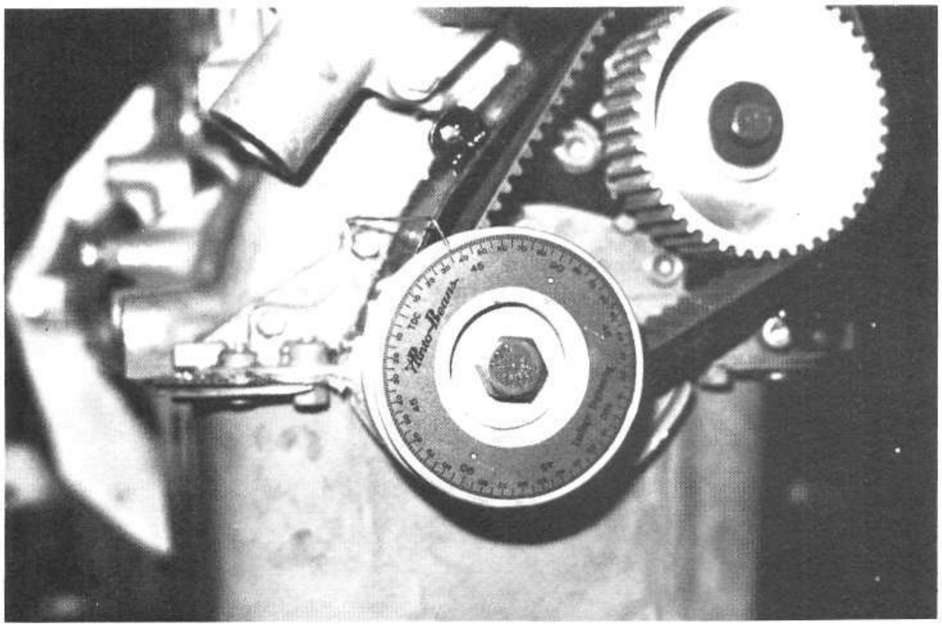


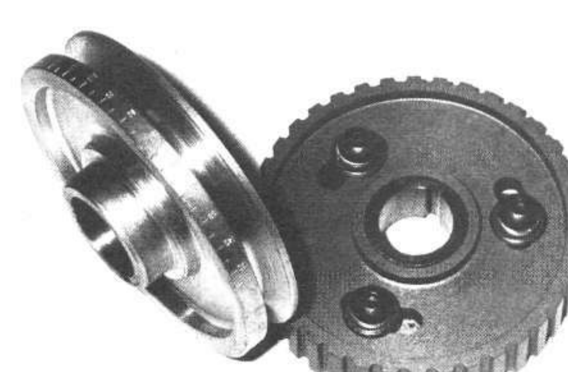
3-1. An adjustable cam sprocket is a wise investment any time a cam change is made. On occasions, the standard cam can be far enough outright from the factory to warrant using an adjustable sprocket for correction. Some adjustable sprockets will not clear the standard belt casing so check this out before you buy. Item shown here is made by Holbay.

3-2. John Shankle at Shankle Automotive uses this large protractor to set up cams on his clients' race engines. Here TDC has just been established.

Shown below is an Esslinger adjustable cam sprocket. This sturdy design has much to commend it.







3-3. Precision degreed pulley in the middle photo is marked in increments of one degree. Produced by Esslinger Engineering, is a very useful item for the serious engine builder. Once you have adjusted your cam and ignition timing for maximum power, you can accurately determine the settings. When rebuild time comes round, there will be no need to guess where the cam or ignition timing should be. Lower photo shows adjustable sprocket and degreed crank pully from Manx Racing.

DEG crank angle increments. This precision requires about two minutes longer to alter the cam timing than more conventional adjustable sprockets. I have used an Esslinger Engineering sprocket on may cams and with many cam timing changes, and it always operate as smoothly new every time

CAM TIMING FOR POWER.

Pinto or Capri fanatics will likely have read in performance magazines that advancing cam timing does all sorts of wonderful things for engine power. Well, I hate to disappoint you, but this is only partially true. I have tested the effects of advancing cam timing on at least three different two-litre engines in stock or near-stock form. A standard cam was installed, *set to factory specifications* and a power curve was taken off the dyno. Then the cam timing was advanced. (See Figure 3-1.) Normally, advanced cam timing yields a little more low-end power at the expense of high-rev power. On a manual transmission car, where engine speed can be kept high, standard or near-standard timing seems best. However, in an automatically shifted car, low-end torque can be considered more important. Advancing the cam six to eight degrees from standard timing will normally shed. .10 second or so in ET, while raising quarter-mile trap speed by one or two mph.

In some respects then, the advantages of advancing cam timing seem slender, suggesting that the adjustable sprocket is of questionable value. Well, this would indeed be true if production tolerances didn't influence the equation. You see, often the cam timing is not where it's supposed to be. I have heard of cam timing being as much as eight or nine degrees out and six is not at all uncommon. Under these circumstances, an adjustable cam sprocket can result in a worthwhile power increase. Typically going from, say, six degrees retarded to somewhere between stock and two-degrees advance can yeild up to 11 bhp.

Although it can provide increased horsepower, the true worth of an adjustable cam sprocket is its use with a high performance camshaft. A particular cam may require different timing for various engine specifications. If you make a change in carburation, exhaust or cylinder-head specification, you may find optimum cam timing needs change too. Cam timing giving best results may be considerably different from timing obtained by lining up the timing dots or setting the timing to the cam grinder's specification. This last point is probably the most important aspect so far as an adjustable sprocket is concerned. For instance, when testing a Ford cam of U.S. origin, I found the best power curve with the cam occurred eight degrees away from the dot aligment position. When initially installing the cam, I used the timing dots as a starting position and began power testing. I found out that power achieved by optimizing cam timing can often be as much as 25 bhp more in certain parts of the rev range, as compared with merely aligning the dots. (See Fig 3-1A.)

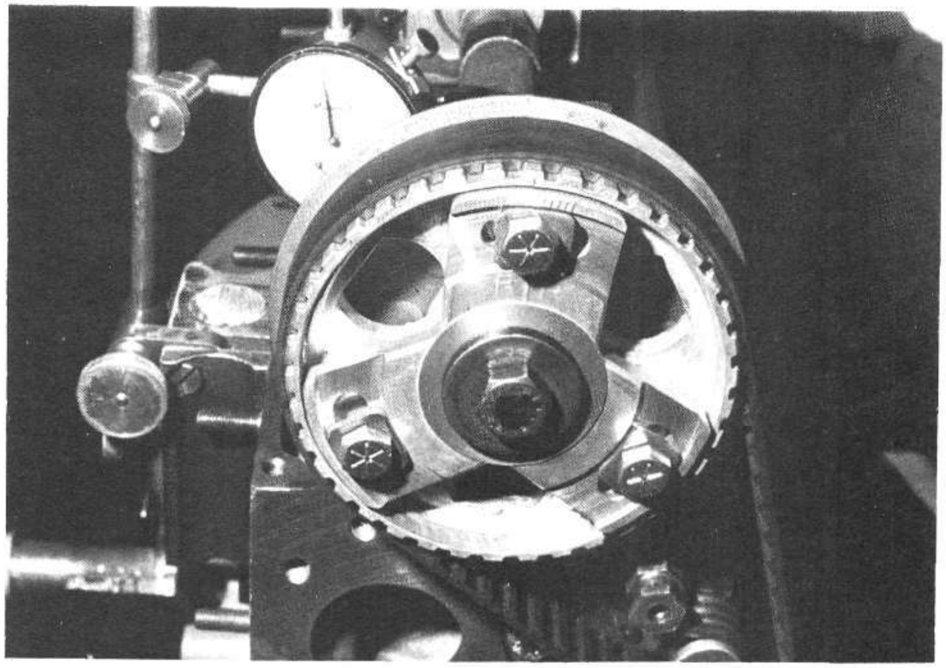
VALVE TRAIN GEOMETRY

If there is one aspect of this engine which produces a love/hate relationship, it is valve train geometry. Although its concept appears simple and straightforward, the geometry of a pivoting rocker and SOHC is anything but simple.

Five factors affect maximum valve-lift and the shape of the valve-lift envelope. To see how each factor relates to valve-lift, assume the *lift profile* of the cam remains unchanged. If this is the case, valve-lift is affected by pivot- pillar height (arrow 1 of Fig. 3-2), cam base circle (arrow 2), radius of pad on follower (arrow 3), radius of pad bearing on the valve tip (arrow 4), and the valve length (arrow 5).

Dealing with point 1, increasing the pivot pillar height within a reasonable range of the normal, produces increased lift. However this is not of great value because pivot height depends upon the cam's base circle, the relative positions of the two pads on the follower and the position required for correct running clearances. Keep in mind that increasing the pivot pillar height increases valve lift.

Moving on to arrow 2, the pad on which the cam profile bears is critically important. Wear life of this item is dependent not only on the metallurgy of the cam and follower combination but also on the relative geometry of these two components. Geometry-wise, three things can happen to the pad to



3-4. The typical range of adjustments for adjustable cam sprockets is about 10° either side of the "straight up and down" position. On the engine shown here, the sprocket needed adjusting to 1½° advance to put the cam in the correct position.

FIG 3-1
EFFECT OF CAM TIMING ON POWER OUTPUT

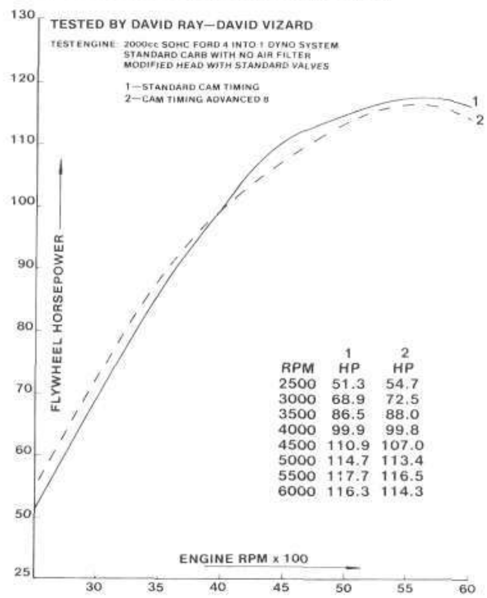
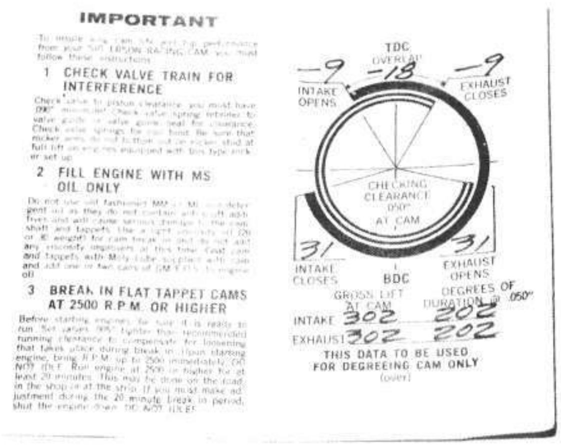


FIG 3-1A

EFFECT OF CAM TIMING ON HORSE POWER

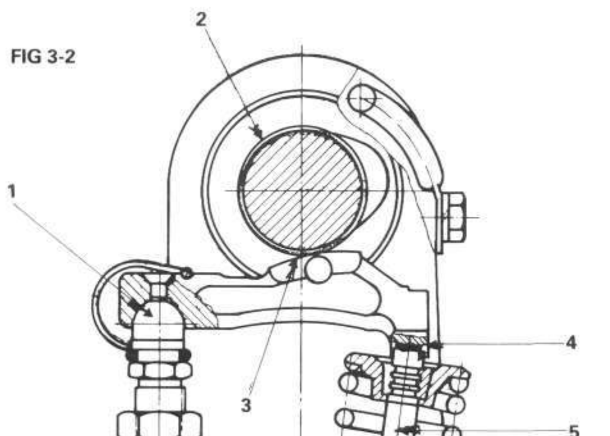
R.P.M	1 TIMING DOT ALIGNED HP	2 ADVANCE 4 HP	3 ADVANCED 8 HP
2000	27.6	32.3	33.2
3000	54.3	56.9	64.7
4000	95.6	100.0	104.3
5000	117.5	125.0	129.3
6000	108.0	129.5	132.2

NOTE: **With timing dots aligned cam timing was due to manufacturing tolerances approx 5° retarded. Final timing as in column 3 was 3° advanced over cam manufacturers specifications and produced best power curve.**



3-5. When you buy a cam from any reputable cam company you will get a cam data sheet like this. READ IT.

FIG 3-2



DRAWING COURTESY FORD MOTOR CO.

affect lift. If the pad radius remains unchanged, but the position of its surface is moved up or down, a change in valve lift occurs. If the pad is higher on the follower, valve lift is reduced because the pivot pillar has to be adjusted down to achieve the necessary clearance. This loses lift. If the pad is lower on the rocker, the reverse applies because the pillar must be adjusted upwards to achieve clearance between cam and follower.

The next factor subject to change on the follower is the pad radius. The smaller the radius, the slower the valve opening rates. The larger the pad radius, the faster the valves open. If the ends of the pad remain untouched, but the entire pad is machined flat, more lift is achieved, because a flat surface can be regarded as a radius of infinite size. If a flat pad is used on the standard follower, the pillar adjustment moves up, causing more lift. Fig. 3-3 shows the effect on the lift curve when a flat-profile follower versus a curved follower is used against a standard cam. The lift curve produced is typically as radical as a mild street grind cam, or if you prefer to call it such, a Stage 1 cam.

Shifting from theories to practicalities for a moment, let me say that I have never tried running a flat follower against a standard cam to see the effect on reliability or horsepower. You may like to try such a move, so a few words on both the pros and cons are in order.

The obvious advantage of the flat follower is the increased lift and opening rates. Directly opposing these advantages in the fact that the increased opening rate imposes higher surface stresses on both the cam profile and follower. This will wear out these items quicker, but offsetting that is the fact that the surface stress seen by either the cam or the follower is partially dependent on the instantaneous radius of curvature of the two components. Flattening the follower cuts down the instantaneous radius of curvature at any point in the lift cycle. If all other things are equal, surface stresses will be lower. This factor tends to offset the increased stresses and increased acceleration rates of the valves. Whether or not it will entirely offset it is another question. I don't believe it would, but it would certainly help the situation.

Taking this one step further, if the flat follower helps the lift profile on a standard cam what will it do to a more radical cam, profile? This is where I should warn you to tread very carefully. I plotted the lift curve using a flat follower on a Crane P332-10 cam. The result was almost instantaneous valve lift just off the seat. The surface stresses and stresses in the valve train here would be so severe that the cam would annihilate itself. My advice here is: use flat followers with caution if you intend trying them at all.

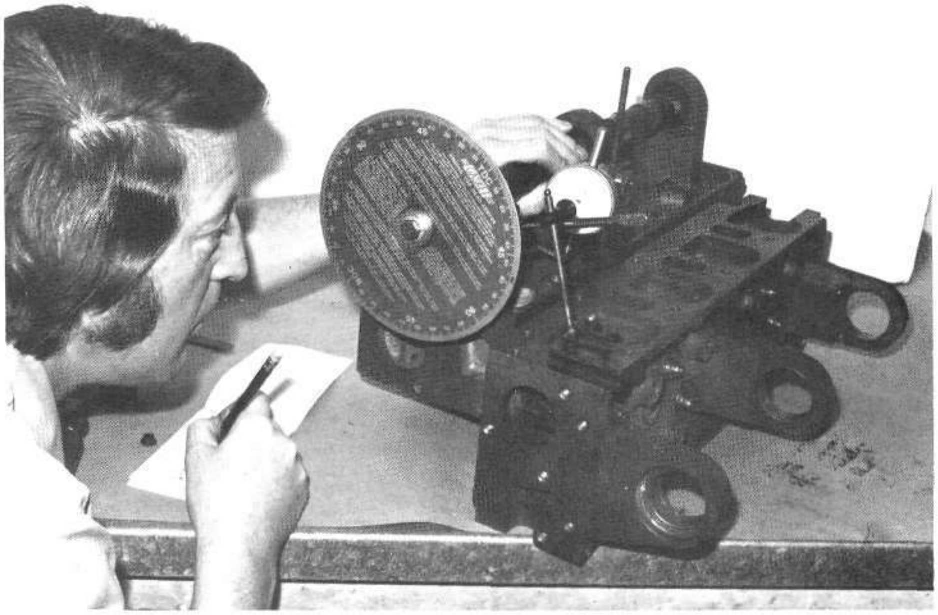
The third factor which can affect valve lift and opening envelope is the relative angle of the pad. Look at a pad and you will see it's tipped at an angle to the rocker. If the pad is flattened or increased in steepness relative to the position of the pad on the tip of the valve and the ball pivot socket, it will alter cam timing. Tipping the pad will cause the valve to open later or earlier. Imagine rotating the follower pad about its centre so the pad end nearest the valve is lower, and the end nearest the pivot point higher under these circumstances. The cam has a slightly quicker off-the-seat acceleration and the valve will open sooner. This in effect advances cam timing, If the pad slope is altered in the other direction, the opposite situation will apply.

Variable number 3 — only controllable by the cam grinder, is the cam's base circle diameter. If the base circle gets smaller, lift starts to increase, and at a certain point, no further gains are made. Further reductions in the base circle reduce the valve lift. The point at which this occurs is smaller than you are likely to use.

While we are on the subject of base circles I should point out that as the base circle is reduced, all radii on the cam are correspondingly reduced for any given cam profile. The result is higher surface stress — one thing you do not need for this cam and valve train as it only decreases reliability. This is one of the major arguments against cams reprofiled from a used cam.

Moving on to the valve-tip pad, as indicated by arrow 4, there is only a limited effect due to changing in profile. The curve on the pad is designed to give a rolling action across the end of the stem. If the pad were flat, it would cause the valve to open quicker. It would also create a tremendous side load on the valve, greatly accelerating guide wear. The standard valve pad on the follower has curvature across its width as well as down its length. This allows the follower a certain amount of side-to-side rocking movement. This in turn allows the cam-follower pad to sit squarely on the cam profile without being influenced by parallelism errors between the tip pad and the valve stem. Fig. 3-4 shows this. It's only a small point to consider if you are looking for long life from your engine but it is important.

Valve length, arrow 5 is the most important aspect so far as the engine builder is concerned. The effect on lift and acceleration caused by valve-length



3-6. Checking out the valve train geometry to establish basic parameters doesn't need a lot of equipment. If you are careful, meaningful measurements can be made with the simple equipment shown here.

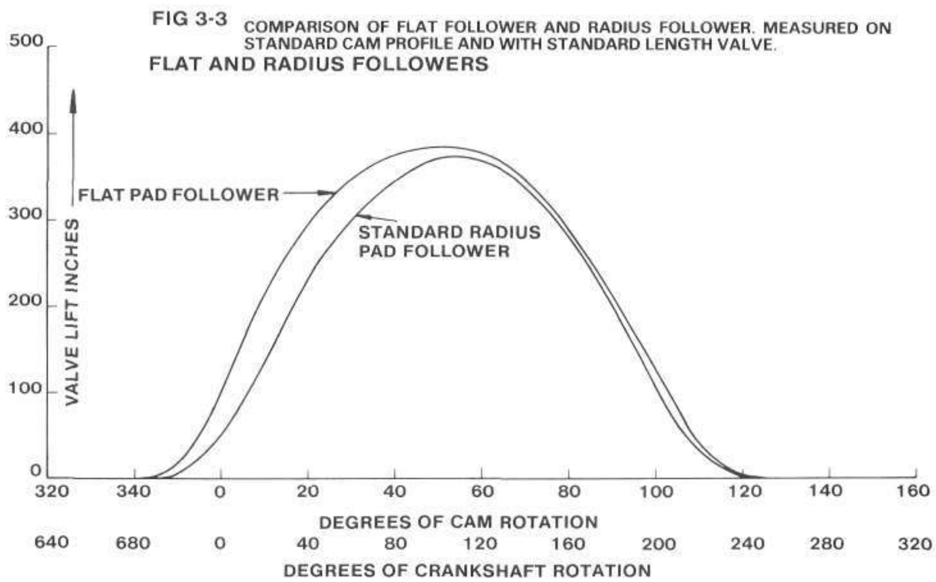
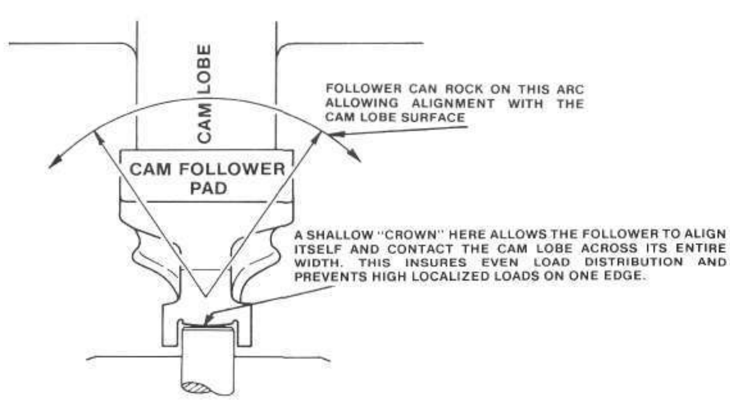


FIG 3-4



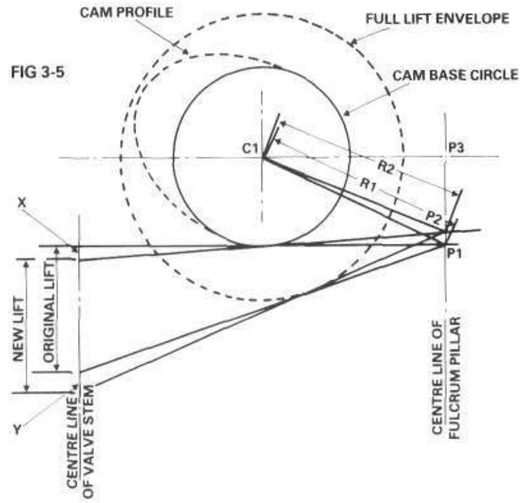
changes is not commonly realized. A small change significantly affects how a valve lifts. The shorter the valve becomes, the more valve lift is increased. This is partially due to the valve tip moving down and partially due to the pivot point (the pivot pillar) moving up on the other side of the cam. The geometry involved is difficult to appreciate, but Fig. 3-5 should help.

When you have digested that, you should be in good position to understand for yourself why Ford's SOHC valve train is not as simple a system as it first appears.

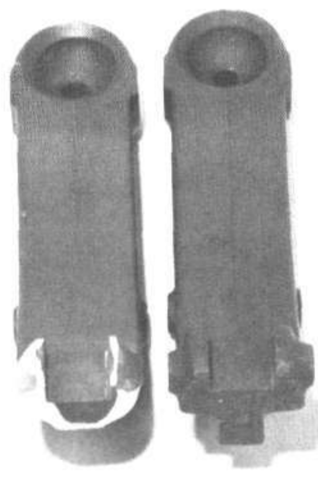
SETTING UP THE VALVE TRAIN

From the engine builder's point of view the most important variable in valve train geometry is valve stem length. Although all the other factors may have an influence, most of them are beyond our control since most of us do not have the necessary machinery to change these factors. Valve-stem length is another matter. Any lathe or machine shop valve-stem facer can be used to set the valve lengths to whatever is required. On an engine using standard-length valves, the practical limit for valve shortening is around 0.075 inch (1.9mm). Shortening the valves this amount increases lift of approximately 0.025 inch (0.63mm). Generally speaking, you can figure on a 3-1 ratio between the amount the valve is shortened and the amount the lift is increased. The 0.025 inch (0.63mm) extra lift represents around a seven percent increase in area under the curve (See Fig. 3-6). This graph also shows the lift envelope biased toward the opening side of the curve, which means faster opening ramps with slightly less increase on the closing side. If anything, this is more advantageous to us than an even spread, as it gets the valve off the seat quicker and allows the air an easier passage into the cylinder during the intake stroke. To shorten standard-length valves by as much as 0.075 inch (.0029mm) requires other modifications to ensure compatibility with the rest of the valve train. The most obvious problem is that the ears or tabs on the side of the valve stem pad on the follower contact the valve spring retainer. If they do, it may cause the spring retainer to separate from the valve, and a valve could drop into the engine. To avoid this, you must keep a close watch on the clearance

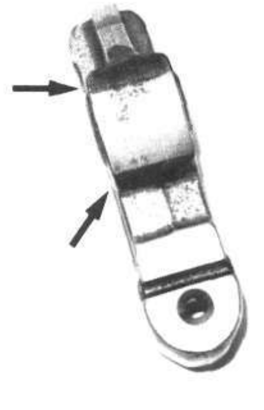
FIG 3-5



Left: Here is the geometry of the increased lift due to valve stem shortening. As the stem is shortened, so the pivot point of the rocker on the fulcrum must go up. As it goes up, so the distance between the camshaft centre-line C1 and the fulcrum pillar pivot point becomes less. Take the two pivot points P1 and P2. With pivot P2 the radius R2 is shorter than the radius R1 given with pivot P1. If the pivot point was moved up to P3 (impossible in practice) then the distance C1 to P3 can be seen to be much shorter than C1 to P1. By shortening this distance we are increasing the rocker ratio which in turn gives us more lift. In this schematic example, it can be seen that if we shorten the valve by the amount X we gain at the other end of the valve travel the greater amount Y. The difference in X and Y is the gain in valve lift.

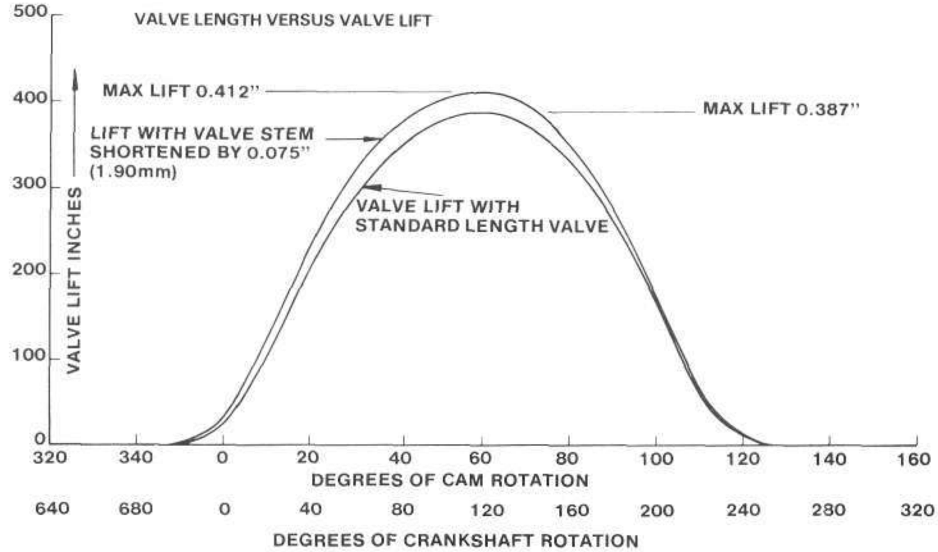


3-7. When the valve stem is shortened to get the design lift of the cam or more, the "ears" on the follower may need to be shortened. If you have to do this, you may as well turn the job into a lightening operation as well. Rocker on the left shows the general form the end should take.



3-8. This rocker has been blued, set up with a 0.550" lift cam. Note how the wiped area between the two arrows is towards the fulcrum end of the pad. Any more shortening of the valve could have caused the wiped area to come off the pad. This would have rapidly modified the cam profile to a low lift form!

FIG 3-6 THIS GRAPH SHOWS HOW VALVE LIFT IS AFFECTED BY VALVE LENGTH. MEASURED ON INTAKE VALVE WITH 0.008" CLEARANCE.



between the ears or tabs and the spring retainer. It is permissible to shorten these ears, but regard 0.09-0.1 inch (2.2-2.5mm) as the minimum length of the shortened ear. Moving the valve tip closer to the valve spring retainer may also cause the follower to hit the spring retainer.

The last detail you need to bear in mind when shortening a valve, is to keep an eye on the wiping patch of the cam lobe on the cam follower pad. As the valve is shortened, the wiping patch moves progessively nearer the fulcrum point. If a valve is shortened too much, the cam will actually come off the end of the pad and edge-ride on the corner of the pad. This will quickly destroy the cam and follower. When shortening a valve, do so progressively. Reinstall it between each shortening operation to find out how much farther it is possible to go before problems arise. Most difficult to compensate for if you go too far is edge-riding caused by moving the wiping area too far back on the pad. To avoid this, blue the cam-follower pad and check the area that the lobe is wiping. When it gets within about 0.015 inch (.006mm) of the end of the pad, stop shortening the valve stem.

HIGH-PERFORMANCE CAMS AND VALVE LENGTH

I've touched on this subject briefly in the previous chapter; now I will deal with it in more depth. Many replacement, large diameter valves have a longer-than-standard valve stem. This reduces the effective lift of any cam. In simple terms, the more radical the cam you intend using, especially from the lift point of view, the greater percentage of

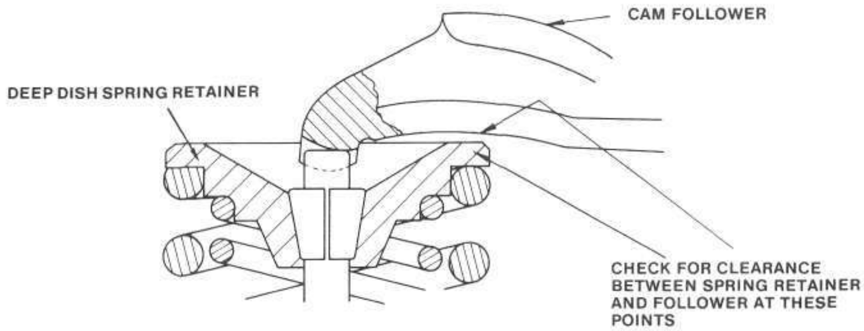
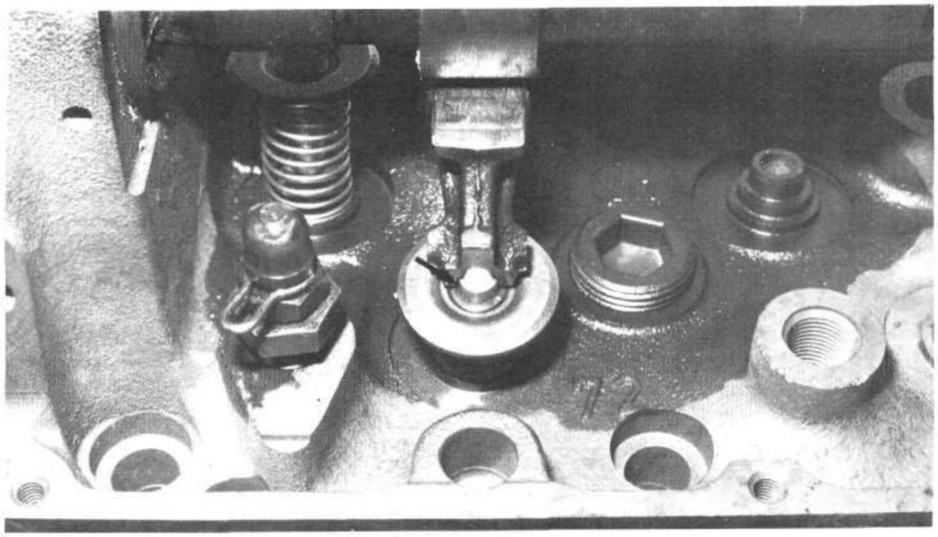
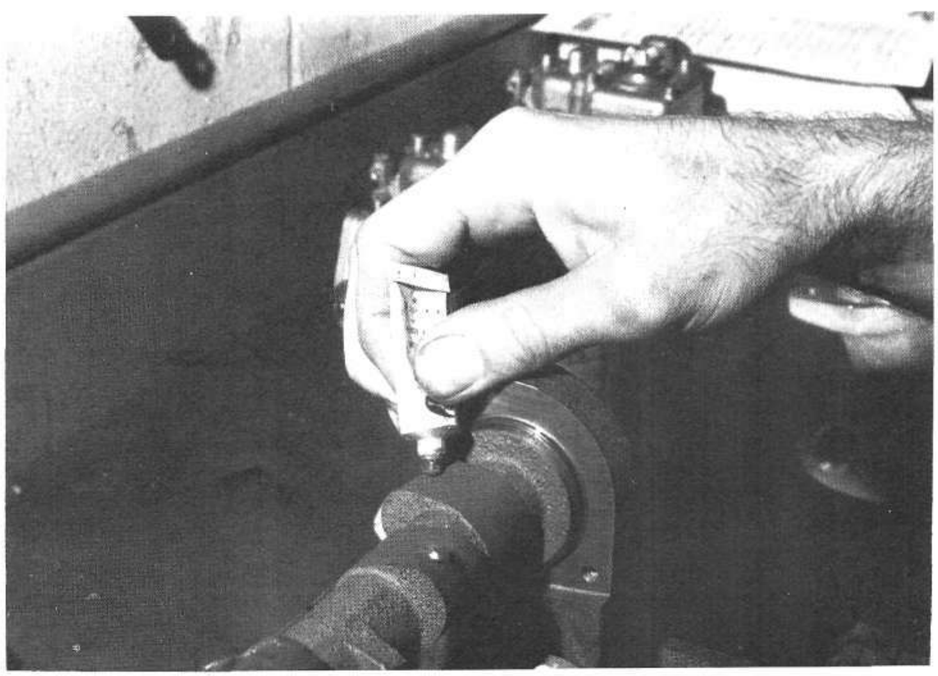


FIG 3-7 CAM FOLLOWER TO SPRING RETAINER CLEARANCE



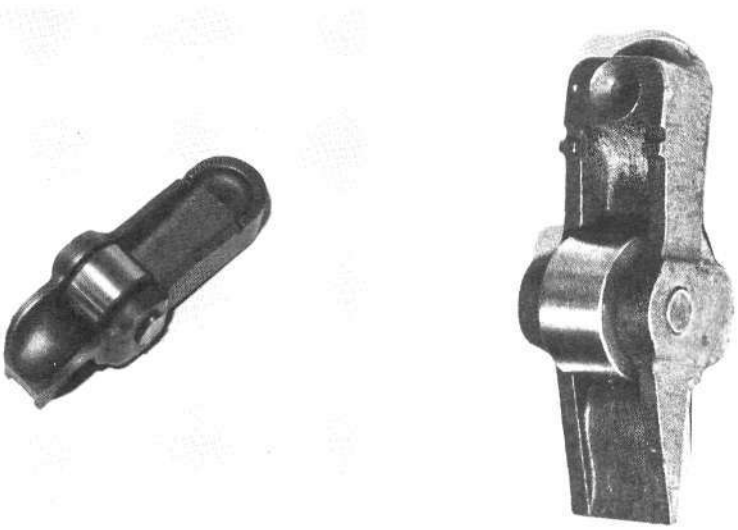
3-9. When very high lift cams are used, the rocker starts to move off the end of the valve as shown here. Apart from wearing the outer end of the rocker a little quicker, this doesn't seem to have any other bad effects.

3-10. These "lash caps" are used to give added valve length so that the correct cam/rocker clearance can be set up without running out of threads on the rocker pivot pillar when adjusting. The drawback to using "lash caps" is they reduce valve lift.



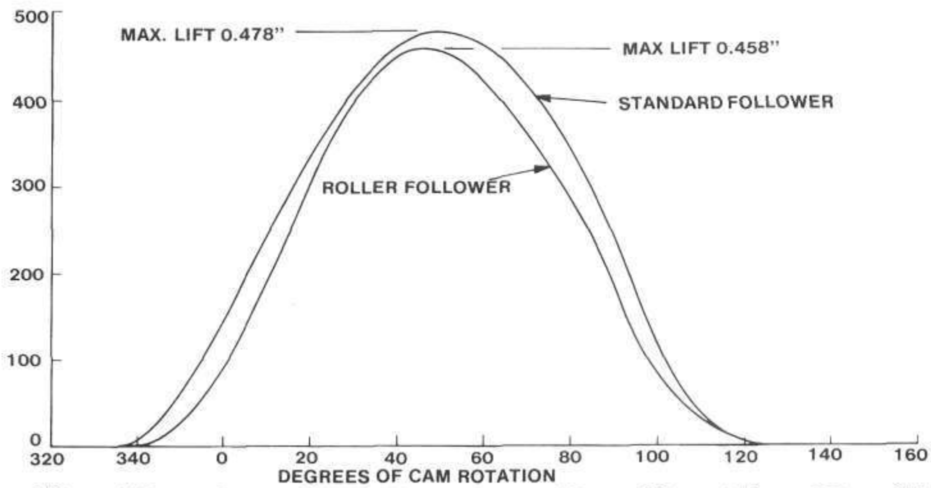
3-11. Cam pre-lube is VERY important if the cam is to function without wiping itself out. Special lubricants are usually supplied in the cam kit, so use them.

valve-lift loss there will be due to a longer valve stem. A longer valve stem has been used in most cases to allow setting the correct running clearances with a cam having a smaller base circle. Without this extra valve length it is possible to run out of pivot-pillar adjustment. With some very high-lift cams, typically in the .550 inch (.15mm) and above range, the base circle is so small that the pivot pillar runs out of adjustment before the correct running clearances are achieved. Some cam companies market *lash caps* which fit onto the valve stem to effectively lengthen it. Although these may give you pillar adjustment, they effectively reduce valve lift. In other words, lash cap are really only a crutch to construct an operational valve train. The correct way to tackle the problem of insufficient pivot-pillar adjustment with small base circle, high-lift cams is to install Racer Walsh press-in inserts. Using these inserts instead of lash caps can make the difference between your cam lifting 0.520 inch and 0.590 inch, (13.2mm and 15mm), depending on which route you go. When using such high-lift cams, you will probably be using deeply dished spring retainers. These, even more than standard spring retainers, suffer cam- follower clearance problems typically at the points indicated in Fig. 3-7. Also, with high-lift cams it is easier for the cam contact patch to run off the end of the cam-follower pad. As a matter of course, check every one to make sure it is not edge-riding.



3-12 & 3-13. Manx Racing roller cam follower eliminates the cam and follower wear problem. Some changes in valve train geometry occur but these are easily accommodated as the nearby test explains. Early type shown on left, current type on right.

FIG 3-8
COMPARISON OF LIFT CURVE GIVEN WITH STANDARD FOLLOWER AND ROLLER FOLLOWER.
CAM PROFILE: RACER WALSH STAGE III



CAM FOLLOWERS

Cam followers are a sore point with many SOHC Ford owners. The follower and cam-lobe wear problem has spoiled many an engine. But it is a problem which can be overcome. You only need to follow a few rules to almost guarantee a reliable cam and follower setup. 1. Whenever you use a new camshaft, use new followers. 2. Ford has made several follower types over the years. Latest is best. The latest followers, as of 1979 are easily recognized by their shiny tumbled finish. These rockers also have a slightly longer pad than the earlier black-oxide-finished followers, so if you are using a high lift cam, these are a definite must.3. Lubrication is also a key factor to the life of the cam profile and the followers. Since using General Motors Super Engine Oil Supplement (E.O.S.) or Ford Oil Conditioner, I have not suffered a cam-to follower wear problem on a race engine, and that's using cams of up to 0.650 inch (16.5mm) lift.

The last step and probably the most important is to hardness check each follower. Generally they range from 49 to 55 Rockwell C. Avoid anything under 51-52 and if choice permits, 54-55 RC is preferable.

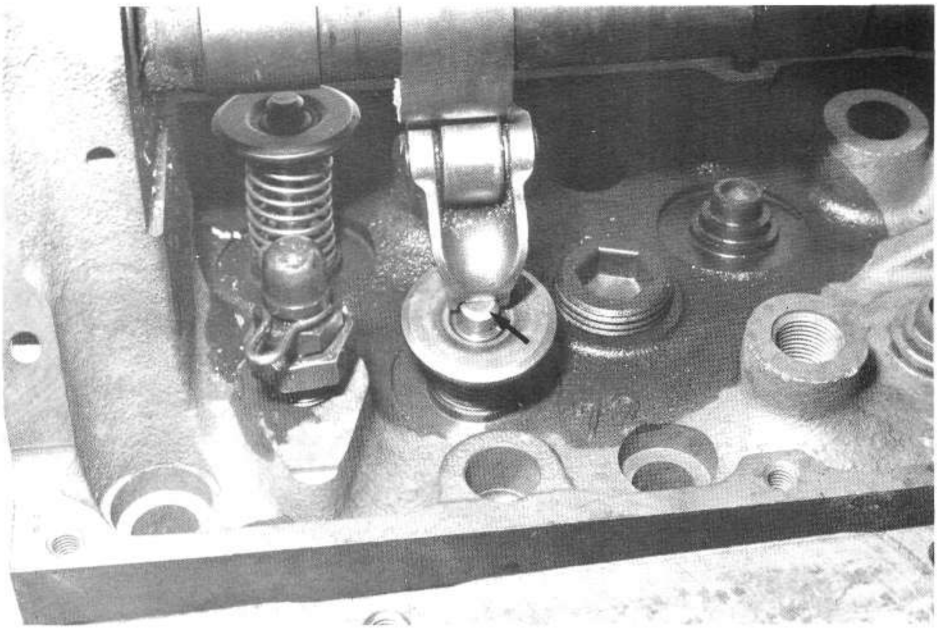
The rest of the precautions to ensure reliability have already been covered in the section on shortening the valve stem. Heed these points, and your engine will probably not have cam wear problems. If you want to be 100 percent certain of maximum possible life, there are a couple of routes you can go. Of the more conventional follower design, Piper produces one with a stellite-type pad. These are virtually indestructible, but good lubrication must still be supplied because although the follower may survive, the cam may not.

Still on the subject of cam profile and follower reliability, the latest cams from Ford are bored hollow and a small oiling hole is drilled into the loaded flank of each cam lobe. This applies oil to the follower just prior to its maxiumum load point. Hollow cams, in conjunction with the late tumbled-type follower appear to have minimized the wear problem.

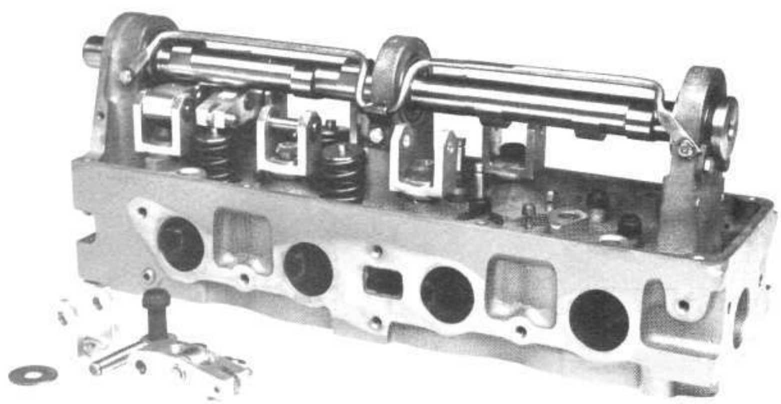
ROLLER FOLLOWERS

The surest way to eliminate wear at the cam-lobe and follower interface is to use a roller follower. This also allows reducing the quantity of oil to the top end. Remember, most of the oil up there is to lubricate the cam and follower when a conventional follower is used. The simplest roller-tappet conversion uses Manx Racing followers. Everyone I know who has used these rollers has found they are 100 percent reliable in street or race applications up to about 0.520 (13.5mm) lift. These followers can be used as a direct replacement for the standard items, but a direct swap loses lift and effective timing. This is because the roller has a smaller radius than the standard follower pad. Fig. 3-8 shows the typical loss of lift when a *direct* swap is made. Ideally, when you change from a conventional follower to a Manx roller follower, you should shorten the valve to restore the lift to that obtained with the conventional follower. Then the roller lift curve is similar to the standard follower except that valve acceleration just off the seat is marginally slower. A test on the dyno revealed no discernible difference in the horsepower output due to the slightly slower opening rates of the roller follower. With roller followers the torque require to turn the cam is noticeably less than with a conventional follower. Dyno results suggest the reduced friction compensates for the slightly reduced valve-opening rates.

Although an effective means of enhanced reliability at any stage of the game, the roller followers really come into their own with wild cam profiles and profiles designed especially for roller lifters. Some of the big Crane cam profiles lifting to 0.580 inch (14.73mm) or more, fall into the "wild" category. These profiles are sometimes regarded as "shaky" for a road race event of any length but can usually be made to live for a race or two with roller followers. Some companies such as Racer Welsh, Competition Cams and Manx Racing in England offer profiles designed to work with these roller followers. As yet, the Manx Racing profiles are somewhat more conservative in lift than their U.S. counterparts. In 1979 Racer Welsh had profiles giving over 0.600 inch (15.24mm) lift with the roller followers.



3-14. When attempting to use Manx roller followers with super high lift cams (0.600" or more) check to see that the tip does not pass the centre of the valve as seen here (arrowed). If it does, check with Manx Racing for ones with longer valve pads.



3-15. As can be seen here, the Holbay roller follower differs considerably from the Manx item and is designed specifically for use with the Holbay roller cam.

HOLBAY ROLLER FOLLOWERS

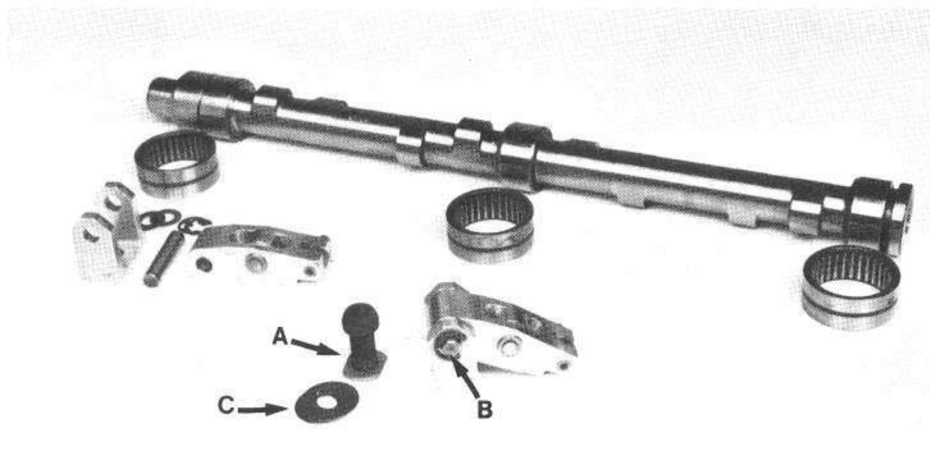
Equally effective but more complex and costly are the roller followers produced by Holbay Racing. These are designed to go with their own roller bearing steel camshaft. Installing the Holbay roller bearing conversion entails replacing the total valve train assembly. Setting the cam follower running clearance or valve lash is unconventional. The near by photo shows the roller follower conversion. Stud (A) is screwed into the head. Pivot-centre height adjustment in the pivot pillar (B) is achieved by placing the required number of shims (C) over the stud in the head. The shims set the clearance close enough to allow the final fine setting achieved by adjusting the eccentrically mounted roller (D) and locking it in position once clearance is set. The Holbay system appears reliable. The only snag is that as of 1979 Holbay had no really high-lift cams. Although the cams they offer work well, I feel, in view of the head's flow characteristics, they need a lot more lift than 0.460 inch (11.68mm).

SPRING RETAINERS

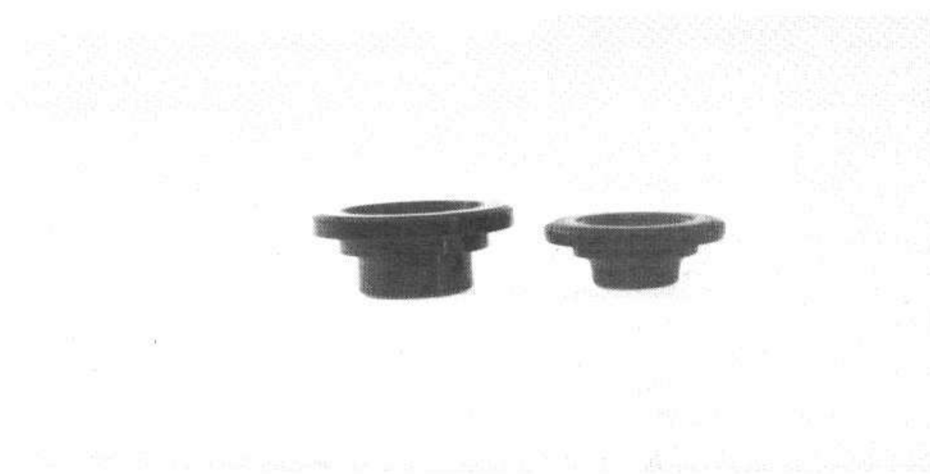
When considering valve spring retainers, the following three points must be borne in mind: lift available before the valve spring becomes coil bound, reciprocating weight, and cost. The first is most important. When an increase in lift or rpm or both is required, heavier valve springs are needed. To achieve the increased spring pressure, thicker gauge wire or an inner spring or both are used. This inevitably *increases* the stacked or fully compressed height of the valve spring. Depending on the type of spring used, at valve lifts around 0.480--0.500 inch (12.2--12.7 mm) the spring can become coil bound. Attempt to run the engine under these conditions and you will wipe out the cam and followers within seconds. If there is any doubt as to whether coil-binding will occur, check the coil-to-coil clearance of the spring with the valve at full lift. Coil-to-coil clearance should be about 0.010 inch (0.25 mm) minimum. When coil-binding becomes a problem, two courses of action are open: first you can machine the spring platform.

Material under the spring platform is typically 5/16--3/8 inch (7.8--9.4 mm) thick, so it can be machined 1/8 inch (3.2 mm). As an alternative, dished retainers can be used to give a greater installed height to the spring. This is usually preferable to machining the spring platform as dished retainers are usually made of aluminium, which is lighter than the standard steel retainers; and are cheaper to buy than machining your spring platforms. If the valve spring that must be used will not fit the standard retainer you may have to buy other retainers. When buying a cam which is anything more than a mild regrind, it is a good idea to buy the springs and retainers with the cam.

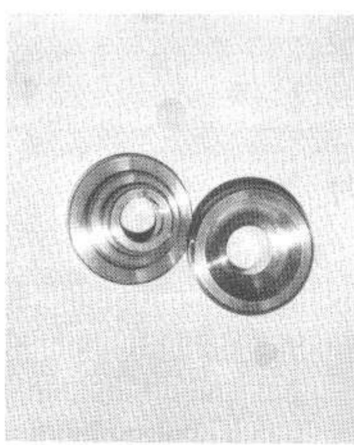
Moving on to point 2, reciprocating weight, this is not as important as it would appear. Reciprocating weight of the follower and spring, plus total valve weight accounts for more than 90 percent of the reciprocating mass in the valve train. Less than 10 percent is accounted for by the spring retainer. If you could design a weightless spring retainer, you would only reduce reciprocating weight by something less than 10 percent. This does not mean lightweight retainers don't have their place; it's just a question of perspective. A lightweight spring retainer helps but it

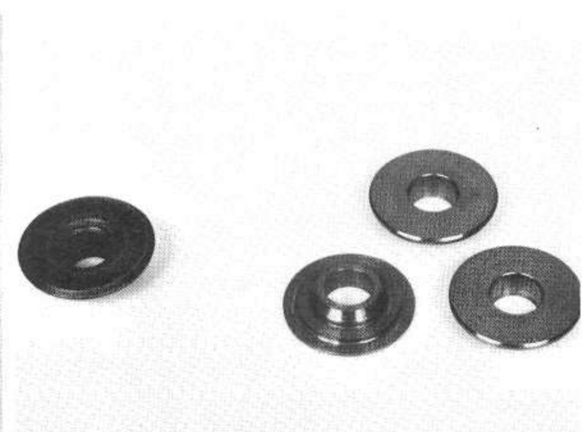


3-16. The Holbay roller kit comprises of these parts. Refer to the nearby text for an explanation of the installation.



3-17. If spring coil binding is to be avoided when high lifts are used, it is imperative that the correct spring retainers are used. The Crane retainer on the left gives about 1/8" more installed spring height than the retainers on the right. This allows greater valve lift before spring coil binding occurs.





Above left

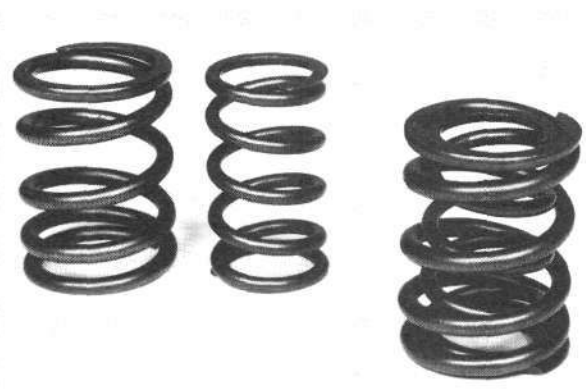
3-18. These Sig. Erson retainers are available at a very reasonable price. They are 30% lighter than standard and are suitable for cams up to about 0.480"-0.500" lift with no modifications to the valve spring seats.

Above right

3-19. Titanium retainers offer a considerable weight saving over the standard steel item. The three Holbay retainers on the right, weigh the same as the one standard one on the left. Unfortunately these retainers are not dished enough to allow spring heights that will work with high lift cams. This could be partially remedied by machining the spring platform on the head.

won't give as many extra rpm before valve float occurs as some think. Using a live example to illustrate the point, if you have an engine which runs to 9000 rpm before valve float occurs with steel retainers, then using aluminium retainers of half the weight may give another 150 rpm before valve float. If you use titanium retainers one-third the weight of steel ones an extra 200 rpm may be available.

Looking at cost, figure that gold, black or silver spring retainers look fancy, but do nothing for the power output of an engine which peaks out at less than 7000 rpm. Such items on an engine like this are a waste of money. The standard steel spring retainers are as good as any.



Above left
3-20. When spring pressures of 180 lbs or LESS over the nose of the cam are required, single springs with a flat wound damper appear to be perfectly satisfactory.

Above right
3-21. Double springs with opposite hand winding on the spring coils tend to damp out unwanted vibrations by virtue of the friction between them. When over the nose spring loads of 200 lbs or more are required, always go for double springs.

VALVE SPRINGS

The standard valve springs are typically 45--55 lbs. (20.45--25kg) pressure on the seat and 100-110 lb. (45.5--50 kg) open. With the standard camshaft, they are good for about 7000 rpm before valve float. Compared with pushrod engines, overhead cam engines will rev higher for a given spring poundage and valve lift because of reduced reciprocating weight and increased valve train stiffness. Many cam manufacturers take advantage of the valve train's inherent stiffness and design profiles that lift and lower the valves at much faster rates than are possible in a pushrod engine. Controlling the valve motions produced by such cam profiles, requires high-poundage springs. An example can be seen in a popular street cam with 10 ° more duration than the standard cam and a lift of about 0.480 inch (12.0 mm). For normal applications it uses valve springs giving 90-100 lb. of seat pressure and 180 lb. (81.8 kg) pressure at full lift. Valve float occurs at about 7500 rpm. The extra 80 percent or so spring pressure gave only 7 percent increase in rpm before valve float simply because of the higher valve accelerations imparted by the more radical cam profile. A reasonably conservative road/race cam will use 100 lb. (45.4 kg) on the seat and 250--285 lb. (113.6--129.5 kg) at full lift--providing about 8500 rpm before valve float. For an all-out drag-race cam, these figures would run in the order of 100--120 lb. (45.5--54.5 kg) seat pressure and 320--350 lb. pressure at full lift. Rev capability for such a cam and valve spring combination would be around 10,000 to 11,000 before valve float.

I have given these figures so that you will be aware of the sort of spring pressures required. When buying valve springs, don't think you can out-guess the cam manufacturer. Always seek his advice and go along with his recommendations. If a flat wound spring damper is recommended, be sure to use it. Leaving it out can cause spurious vibrations which may allow valve float at lower rpm than would occur with the damper. If you have a choice between using variable rate springs and conventional springs, choose variable rate ones, because they usually require slightly less spring pressure to reach a desired rpm. Looking at it another way, slightly higher rpm can be achieved with similar overall spring poundages. The reason the variable rate spring works better is that it is not so prone to resonate at a specific vibration frequency as a conventional spring. You can recognize a variable rate spring because the coils are spaced differently over the length of the spring. The close-wound coils go next to the head with this type of spring.

3-22. High lift and long timing are the ingredients for this Crane 333B cam. The components shown are supplied when the entire cam "kit" is ordered. The Crane Super-lube in the bottle, contains the seemingly vital ingredient Zinc Dithiophosphate which is an aid to curing rapid cam and follower wear.

CAMSHAFTS

Choosing a cam is often a major stumbling block for the amateur engine builder. An error of judgement made here can produce the wrong type of power curve, making the engine less than suitable for the intended application. Having some idea of the engine's likes and dislikes helps to make a more effective cam choice. Many of the camshaft's required characteristics are influenced by the cylinder head. Refer to the flow curves in the cylinder head chapter. For conventional ports, airflow continues to increase with valve lift. For use with some of my early heads, I had cam profiles ground to 0.750 inch (19.05 mm) lift. Fortunately, developing the "apple" intake ports produced effective results with less lift. On such ports, regard the maximum valve lift required as being in the 0.600-0.650 inch (15.2-16.5 mm) range. On ports when no material is added, work on the assumption that the more lift, the better the resulting power output. Unfortunately, this requirement must be tempered by the fact that as valve lift is increased, so is the mechanical stress on the components involved. If a cam is substituted for another having the same timing but more lift, you run into a less than desirable circle of events. To allow operation at the same rpm as with the previous lower-lift cam, a new high-lift cam must be used with heavier springs to avoid valve float. Increased lift results in better high-rpm breathing and the engine has a higher rpm capability. Therefore it needs even stronger springs. Unfortunately, the spring pressure required is not proportional to the rpm required, but proportional *to the square* of the rpm. To see how this works, equate the original rpm capability to 100 percent. Next let's assume you require the engine to turn 41 percent more rpm. The new rpm will be 141 percent. But the valve spring pressure will be 1.41 x 1.41 greater. The answer to that minor bit of maths turns out to be 2 or more accurately, 1.988. In other words, twice the spring pressure will be needed for 41 percent higher engine rpm. As you can appreciate, with extra valve lift requiring very much more spring pressure, the valve train can quickly become grossly unreliable. Because of this, the valve lift must take into account both the metallurgical and dynamic stress limits of the valve train.

By extending the overall valve-open period of the cam, the amount of time available to open and close the valve is likewise increased. In the long run this benefits us by allowing comparatively less spring pressure to achieve a given rpm or a given spring pressure to achieve a higher rpm. On race cams this is fine. On street cams, an increase in opening period is detrimental to the power curve at low rpm. The implication is that although the engine could benefit from a street cam 0.600 inch (15.3 mm) lift, the short opening period would impose stresses too high for reliability. On the other hand, a race cam with its longer duration and lesser need of reliability can successfully use as much as 0.650 inch (16.5 mm) lift. Equally important as lift are the acceleration rates imparted to the valve by the cam profile. Because high lift is what this engine needs, it also follows that the sooner the valve achieves full lift, the better. This means cams with high acceleration rates. The higher the acceleration rates, the more the stress and the less the reliability. By good design a cam manufacturer can minimize the destructive effects of dynamically induced acceleration stresses, yet still allow the cam to make the most of rapid valve opening and closing rates. Because of the *critical* and complex nature of a cam profile which makes the best of all these variables, most competent cam manufacturers use a computer calculated profile.

CAM TIMING

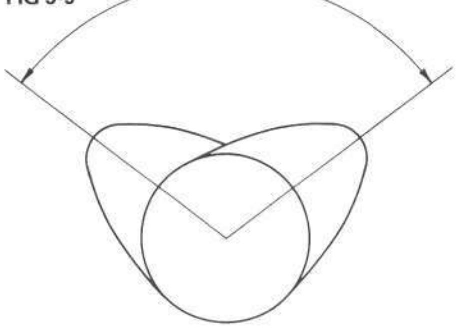
Varying the point at which the valves open and close in relation to the crankshaft motion has a profound effect on the power curve. As with almost any engine, the longer the duration of valve opening, the less low end power the engine produces. In relation to the rpm the engine will turn, the Ford SOHC engine *does not need* as much valve duration as a typical pushrod engine. This is mainly due to the faster acceleration rates possible with an OHC layout. One of the biggest mistakes made by amateur engine builders is to *put too much cam duration* into these engines. Although it may thrive on all the lift you can give it, tread very carefully when it comes to duration. Depending on cam lobe centres, which will be discussed shortly, increased duration yields increased overlap. If you are looking for low speed drivability, avoid cams with much more overlap than standard cam. Overlap destroys low-speed drivability and power quicker than any other factor in these engines, especially if the head breathes well. Where bhp is the major factor, increased overlap brought about by increased opening can be an asset. If a tuned-length exhaust system is used, the exhaust extraction effect can start to pull in the fresh intake charge through the intake valve even before the piston has started the induction stroke. Known as *chamber scavenging*, this is very beneficial as far as power is concerned. Many competition vehicles do not have exhaust systems which provide any worthwhile extraction effect. A typical example is a rally car which must pass certain stringent noise level tests due to regulations. Usually, silenced exhaust systems have a certain amount of back-pressure. This back-pressure exaggerates the effect of the overlap because exhaust blows through the intake port prior to the start of the induction stroke. When the induction stroke starts, it draws in a mixture of fresh charge and exhaust *and* that's not good for bhp. If back-pressure is likely to be a problem, think in terms of *less* overlap. When building an engine strictly for race applications it is best to choose a cam timing which will take advantage of this engine's high-rpm potential. For drag, road and most other forms of competition on asphalt, you can use a duration sufficiently long to allow the engine to produce peak horsepower at 8000 rpm or higher. Such cams typically have 300° or more of duration measured at the seat.

For off-road race vehicles, you should choose a cam with a lot less duration than for its road-race counterpart. Most off-road drivers agree that engine flexibility is a very important matter when racing in the rough. Under such conditions, some of the hotter *street* cams produce the quickest machines.

LOBE CENTRES

The subject of *lobe centres* is rarely dealt with in performance books, because its effect on the power curve is not generally understood. Although less dramatic than the effect of duration, the effect of changing lobe centres is important. A definition of lobe centres

FIG 3-9



DEFINITION OF LOBE CENTRES

or displacement angle, is probably important here. Look at Fig. 3-9: *lobe centres* or *displacement* is a good description. The term means the angle between the full-open position of the inlet and the full-open position of the exhaust. This is the only valve event measured in cam *degrees*. All other valve-timing figures are given in *crankshaft degrees*..

For the strongest power curve, lobe centres around 104°-106° appear ideal. With these lobe centres on anything more than street-type duration figures, the engine comes "on the cam" very sharply. Precisely where it comes on the cam depends on duration. Spreading the lobe centres makes the engine more flexible so it can produce power at relatively low rpm. On the negative side, this increased flexibility is paid for in reduced power output. The basic reason why flexibility is increased is because spreading the lobe centres reduces *overlap*. As I said previously, overlap is probably the single biggest factor affecting low speed drivability. If there is any unavoidable back-pressure in the exhaust system or if you are going to run an automatic-transmission vehicle, then slightly wider lobe centres can be beneficial. As a guide, the following should help you with your selection of a cam. Where power alone is the main consideration, lobe centres between 104° and 106° work best. If any exhaust back-pressure is present due to any form of exhaust restriction, choose 108° to 110° lobe centres. On occasions, cams with 112° centres have produced satisfactory results but more power was gained by making the exhaust more efficient and staying with tighter lobe centres. Automatic-transmission vehicles, especially those with low-stall-speed torque converters need power right off idle. Cams with only a moderate increase in duration on tight lobe centres tend to develop a hole in the power curve just off idle. An automatic-transmission car can fall right into this hole and the result is a nighmare street machine. This tendency can be reduced by using lobe centres close to 110°. Some cams, especially those produced by the factory have lobe centres between 112° and 116°. Although cams on these centres can be made to work, they seem to give away too much power for the sake of a smooth idle and increased low end flexibility. The standard 1.3 and 1.6 cams from the factory have 106° lobe centres and the 1.6 GT/2000 cams have 113°. The 113° lobe centres probably account for the good low- speed manners of the 1.6 GT/2000 engines. On vehicles I owned I could floor the throttle at 1200 rpm in high gear. Acceleration was slow but unfaltering.

SELECTING A CAM

The most difficult machine to select a high- performance cam for is a small-displacement, automatic- transmission vehicle. An automatic Pinto, Cortina, Capri or whatever else your SOHC Ford engine may propel, usually falls into this category. For such vehicles, useful power must be delivered from just above idle to as far up the rev range as possible. To give you some idea of what you are looking for, let's take the standard 1600 GT/2000 cam as a starting point. This cam has a theoretical lift of 0.399 inch (10.1 mm) but in pratice it is usually nearer 0.387 inch (9.4 mm). Duration is 268° for both inlet and exhaust. Lobe centres are 113° and timing is: inlet opens 24° BTDC, closes 64° ABDC; exhaust opens 70° BBDC and closes 18° ATDC. At 0.050 inch (1.27 mm) lift, the measured timing is 7° ATDC to 33° ABDC. Exhaust is 39° BBDC to 13° BTDC. Duration of both inlet and exhaust at 0.050 inch (1.27 mm) lift is 206. Lift when both inlet and exhaust are open the same amount at or close to the TDC point is 0.030 inch (.76 mm). A cam for an automatic-transmission vehicle should closely resemble these figures on all counts except lift. Avoid too much duration like the plague and consider 10°, or at the very most, 15° extra duration measured at the seat as maximum. More important is the amount of extra duration at 0.050 inch (1.27 mm) lift. My own experience has been that cams of around 200-210° work out well. It is possible to make cams with as much as 225° duration at 0.050 inch (1.27 mm) run but the vehicle is lazy off the line. As stated before, 110° lobe centres work well with an automatic.

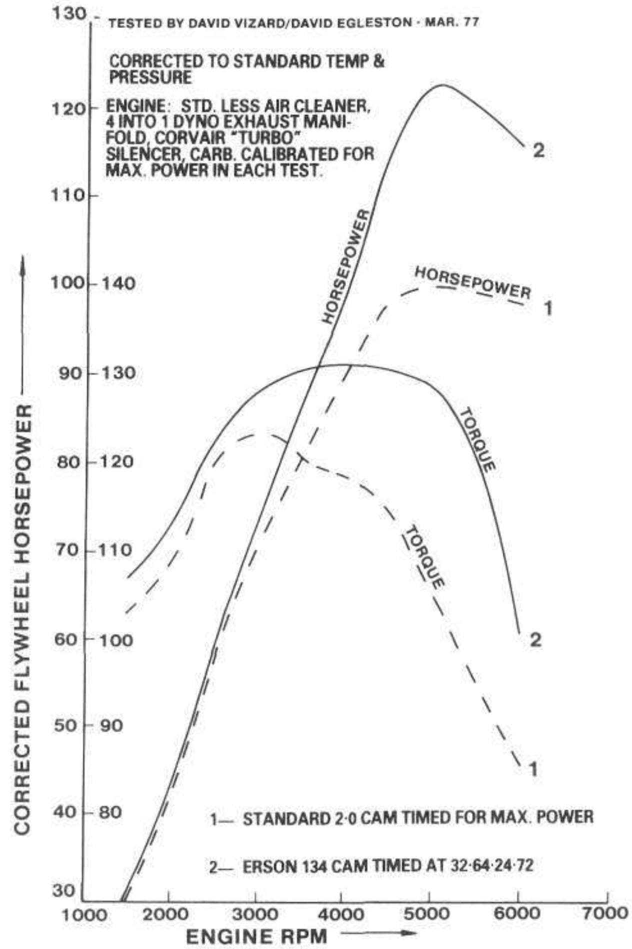
Do not pick a cam which has both valves open too much during the overlap period. The amount of lift when both valves are open the same amount should not exceed 0.050 inch (1.27 mm) and ideally it is best to keep it nearer the figure of 0.030 inch (.76 mm) of the standard cam. When it comes to lift, go

3-23. The Erson 134 grind cam. Proof that high lift short cams really do work for street use. Refer to fig. 3-10 for a before and after power curve.

for everything you can get. Because of the short duration involved, mechanical reliability considerations limit the maximum feasible lift for a street cam to around 0.500 inch (12.7 mm). If that seems like a lot of ground rules for choosing a cam for an automatic-transmission application, it's not without good reason. Deciding on a cam for such an application is one of the most difficult choices to make. This is further complicated by the fact that there are not many cams that fit this bill. One such cam that fulfils the requirements is Sig Erson's number 134 grind. This grind was designed to cover such applications and has proved very successful. As a result of the generous increase in both torque and bhp, this cam has found its way into many manual-transmission street machines and off-road race cars. Fig. 3-10 depicts a back-to-back test of the standard two-litre cam versus the Erson number 134 cam. Torque increase is between four and 15 lb./ft. from 1500 to 6000 rpm and peak bhp is up from 98.2 to 122.3 The design has only 202° duration at 0.050 inch (1.22 mm) lift. That's *less* than the standard cam. The lift when both valves are open the same amount during the overlap period is 0.030 inch (.76 mm), the same as the standard cam. Valve lift on the test engine was a true 0.490 inch (12.45 mm) as opposed to the true 0.387 inch (9.83 mm) lift of the standard cam it was tested against. Fig. 3-10 shows tests are on a relatively standard engine. On engines where big-valve modified heads and additional carburation have been used, power outputs well in excess of 150 bhp have been achieved, with peak power occurring in a 6200-6400 R.P.M range.

The number 134 grind has been used in street-driven manual-transmission vehicles with equal success when good low-speed torque and power up to 6000 rpm have been required. If a free-flowing exhaust system is used, I recommend having this cam ground with 106° lobe centres as opposed to the 110° lobe centres for an automatic transmission. For a good all round street cam the Piper Magnum 270 HR is to be recomended. It produces 15 or so extra horsepower with a useful torque increase throughout the rev range. Along with this cam and follower life seems to be excellent. If you don't mind buzzing up and down the gears and are prepared to sacrifice power below 2000 rpm,

FIG 3-10



then you can use a longer duration cam with a manual-transmission vehicle. Under such circumstances, consider 290° of duration as maximum. A Piper Magnum 285 HR fits this bill and results in a really high reving but street drivable power unit. Translated into typical duration figures at 0.050 inch (1.27 mm) lift, a cam of this ilk this runs out about 230° – 235°. Any more duration than this and you will quickly grow tired of driving your car in the city rush hour. At this stage of the game you must still consider valve lift as a more important factor so far as producing usuable power is concerned.

STREET/COMPETITION USE

Choosing a cam for a dual purpose application is always a little dicey because it almost always falls into the category of "Jack of all trades, master of none". The disadvantages of such a cam are that it is never drivable as a true street cam, nor does it produce the power of a true race cam. On the plus side, with the right cam and other engine mods to suit, you can build an engine with 160 to 170 bhp. Figure on 300° measured on the seat or 245° at 0.050 inch (1.27 mm) lift. Duration about maximum. Even in the range of cams with this sort of timing, many do not have adequate lift. If you narrow your choice to a few cams, use the one with the most lift. A cam with 280° (seat) duration and 0.500 inch (1.27 mm) lift, almost always makes more power than one with 0.460-0.470 inch (11.7-11.9 mm) lift and 300° duration. The longer the cam-opening period, the more static-compression ratio the engine will need. A 300° cam *needs a ratio of 11:1 or more* before it starts to perform properly. Such high CRs are a problem because of fuel availability, so you may want to change your ideas about the type of cam you are going to use. If you have lots of high-octane fuel available, then you can use high CRs.

RALLY/OFF-ROAD

I've grouped these two categories together because they are essentially both off-road type events. Cam requirements are somewhat different because each races on a different basic terrain. European-style rallies are usually run on rough dirt roads and relatively high speeds are maintained. Engine rpm should rarely drop below 4000 on a correctly geared car, so power below this is not absolutely necessary. At the other end of the rev range the engine needs good overspeed capability where the driver dare not let go of the steering wheel to grab another gear. For an engine like this, pick a cam with 295-305° duration and around 240-260° at 0.050 inch (1.27 mm) lift. Whatever cam you choose, ask the cam manufacturer if the cam can be used with springs to give about 8000-8200 rpm before valve float. This will typically require springs giving 250-280 lb (113-127 kg) at full lift. The golden rule is: the more lift the better! Within this range of cam timing, cams are available with valve lifts up to about 0.550 inch (14 mm) with a minimum of 0.500 inch (12.7 mm). Again, CR must be high in the region of 11 or 12:1. Minimum-restriction exhaust is required, as well.

For an off-road vehicle, the terrain is usually much more severe than that confronting a rally driver. Speeds are often down to little faster than walking. As a result, torque output over a wide rpm range, typically from as little as 1500-2000 rpm up, is more important than peak power. The more successful Pinto-powered machines are using some of the best street-cam profiles. With the rest of the engine modified, this produces all the bhp they can normally use, together with good low-speed lugging ability.

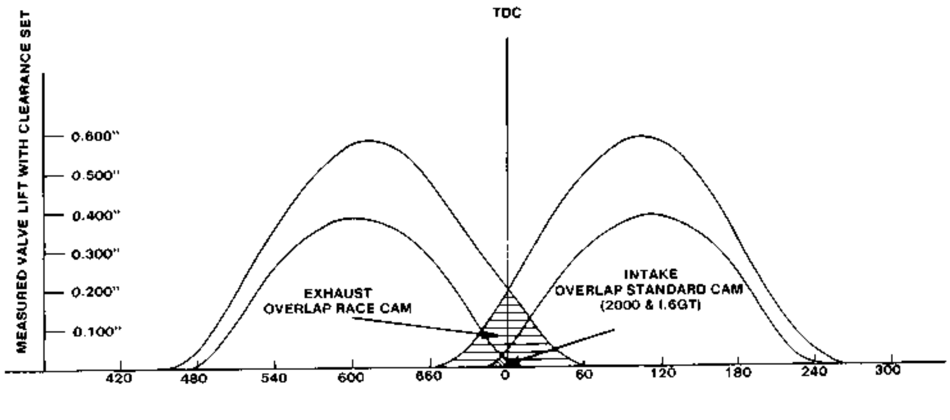
DRAG AND ROAD RACE APPLICATIONS

The only difference between road and drag racing is the length of the race. Both situations demand maximum power. The only problem is, the more radical the cam, the less reliability it may offer.

When all things are considered, if a cam profile is properly designed, the two reliability-limiting factors are lift and spring pressure. Fortunately you have a certain amount of control over both, so it is possible to tailor the cam to suit a particualr requirement. The one factor you don't have much control over is cam duration, so for an all-out race cam, choose one which has a measured-off-the-seat duration of 300- 330°. This can work out to 260-285° at 0.050 inch (1.27 mm) lift.

FIG 3-11

THIS COMPARISON OF THE LIFT CURVE OF A RACE CAM (CRANE P332-10) AND THE STANDARD 2.0/1.6 GT CAM SHOW JUST WHERE AND BY HOW MUCH A RADICAL RACE PROFILE CAN DIFFER FROM A CONVENTIONAL FACTORY PROFILE.



So far as lift is concerned, select one with an advertised lift of at least 0.525 inch (13.3 mm) but preferably nearer 0.550 inch (13.9 mm). I have tried a number of race cams, and three I recommend are the Erson 308 AS grind and two Crane grinds, numbered P332-10 and P322-10.

Let's take the P332-10 Crane cam as an example, and I will explain how to tailor it to your race requirement. The basic spec of the cam is: inlet opens 63° BTDC, closes 89° ABDC. Exhaust opens 95° BBDC, closes 57° ATDC. That is off-the-seat-timing. Timing at 0.050 inch (12.7 mm) lift is 38° BTDC. 62° ABDC for the inlet, and for the exhaust 70° BBDC. 30° ATDC. Lift is a theoretical 0.586 inch (14.9 mm) measured at zero valve lash.

Assume you intend to run a long-distance race, say, 300-500 miles. Having to race these distances, you may be concerned that the cam life will be a problem. The simple solution is to set the valve-stem length to give, say, 0.520-0.540 inch (13.2- 13.7 mm) lift, decreasing the loads on the cam follower and cam profile and hopefully extending profile life.

For the shorter road race, say 25-50 miles, you may choose valve lengths resulting in a higher lift, say 0.540-0.560 inch (13.7-14.2 mm). For drag racing, you have only to make enough passes of the strip to win the final round, in which case careful selection of cam followers and valve lengths can result in lifts as much as 0.620 inch (15.75 mm) from this cam. For what it's worth, my experience with the P332-10 suggests it is capable of at least 50 racing miles when set to produce about 0.580 inch (14.7 mm) lift. Correctly set up cams producing around 0.525-inch lift seem to be good for up to 1000 racing miles. All I have just said assumes using the standard cam followers. Indications are that roller followers, on cams up to 0.550 (14.0 mm) valve lifts, have a long racing life.

CAMS FOR TURBOCHARGED ENGINES

I will relate some of my own experiences, plus those of Duane Esslinger of Esslinger Engineering. Duane's experience of building many 500 bhp-plus turbo engines puts him in a position where he can best be described as having very valid opinions.

Dealing with street applications first, we can say that whatever works well in a normally-aspirated engine also seems to work well in a turbo engine. If you are choosing a cam for a street turbo engine, get one that produces good low-end torque. Don't worry too much about top-end power, the turbo will take care of that. To get low-end torque means using a cam with short duration. Because of the turbo, back pressure is produced in the exhaust manifold and it's a good idea to keep overlap to a minimum. Wider lobe centres will, for a given duration, reduce overlap. Figures between 110° and 114° between lobe centres have been used with success. With a turbo engine valve lift is important, but not so much as with its normally-aspirated counterpart. The main reason is a good turbo installation builds a lot of power, typically 70 bhp or more over the standard output. Struggling for another five bhp by means of .030 inch or so extra valve lift isn't quite as important. When choosing a cam for a normally aspirated engine, if cam A gives 15 bhp and cam B gives 20 bhp by virtue of a higher lift, which will you go for? The turbo engine on the other hand, will probably supply you with all the top-end power you want or can use. An extra five bhp at 6000 rpm is almost irrelevant.

So far as recommending street cams is concerned, both of the standard Ford cams work well with a turbo, so don't assume it essential to rush out and buy a trick cam. Generally, an alternative cam with increased valve lift and possibly more duration will help, only in circumstances where you are attempting to extract the most power for the least boost. It may also be worth considering an alternative cam if boost is likely to be limited because of the fuel octane considerations. If you do want to pull up low-speed torque and get more top-end power as well, then Esslinger recommends the Crane number 274 turbo grind. But for my own part, I'll once more recommend the Erson number 134 grind. Should you need a power curve to fall between street and race, then try a cam with the same sort of timing you would expect to use in a normally-aspirated engine. When using a longer duration cam in a turbo engine, don't expect to see the same proportion of power increase that a cam change in a normally-aspirated engine produces. A friend, Doug Somerville, built a turbo engine for his sand car. Once built, the engine was tested on the dyno with different cams. Results were not totally conclusive but some valuable information was gathered. Tests were conducted with cams from 270° to 300° duration. In essence, these ranged from a hot street grind to a mild race grind. The 270° cams started to run well from 2000 rpm. On the other hand, the race cam did nothing but splutter up to 4000 rpm. At 5000 rpm with boost figures the same in both cases, the race cam produced only seven bhp more. At 6000 rpm the two cams differed by approximately 15 bhp. This is about the same difference one could expect to see between two similar cams in an unblown engine. The moral of the story is, too much duration drops bhp at low rpm without necessarily giving a corresponding increase in power at the top end. Making a statement like this could be unwise, but let me qualify it further by saying that future developments in intermediate street cam profiles for turbocharged engines may well prove this statement wrong.

If you intend to go racing, then choose a cam along more or less the same lines as I previously described for a normally-aspirated engine. On Esslinger's recommendation, I tried a Crane P332-10 cam in a turbo engine running on petrol/gasoline and it produced what can be best described as dazzling results. However, I know that Competition Cams also has a well-developed race turbo cam. In the Esslinger Engineering turbo alcohol engines, a Crane P332-10 cam is used with 45 psi boost; these engines run to over 10,000 rpm and produce over 500 bhp.

COMPARING CAMS

Wading through cam manufacturer's specifications, comparing one cam with another to see what will best suit your needs is tiresome. Comparison of some aspects of the specifications is easy; others are not. So you can make such comparisons; a brief rundown on the way various cams are measured will help.

I'll start off with an easy one; making comparisons of lifts is generally straightforward. It's just a question of looking at the number and deciding whether it's bigger or smaller than another one. The only snag you are likely to run into is, some cam manufacturers quote the lift with zero running clearance and some quote it with running clearance. Also, some manufacturers assume the rocker ratio on a Pinto is 1.6 and others assume 1.66. So even though its easy to compare the numbers, they may, in fact, differ by as much as the difference in rocker ratio used and the difference in the clearance ramp.

Next in line are timing figures. Most cam manufacturers quote off-the-seat timing figures, so you can compare these without any great difficulty. Unfortunately, comparing off-the-seat timing figures gives you no indication of how *fast* the cam is lifting the valve but a comparison of timing figures at 0.050 (1.27 mm) lift will give you a much better idea of what is going on. So the next stage is to make a comparison of timing figures given at 0.050 inch (1.27 mm) lift. The longer these are, the more the effective duration. Generally speaking, a racy cam with quick opening ramps will have the 0.050 inch (1.27 mm) duration figures shortened by 22 °-25 ° as compared with off-the-seat figures.

Cams with milder acceleration ramps, where noise during operation is also a factor considered by the cam designer, may have the 0.050 inch (1.27 mm) timing figures shortened by as much as 31 °-34 °.

Off-the-seat duration is usually given in the manufacturer's specification but if you want to, it's an easy one to work out. Let's say, for example, you want to calculate duration. Take the figure the inlet valve opens BTDC and add this to the figure the inlet closes at ABDC. Add 180° to this number. The same technique is applied to the exhaust. Add the degrees BBDC the exhaust valve opens to the degrees ATDC that it closes, and add 180°. That's the duration of the exhaust opening. To calculate the angle between lobe centres is more long-winded but it's still straightforward. Basically what you do is determine how many *degrees ATDC on the induction stroke* that the inlet valve is *fully open*. After that you determine how many *degrees BTDC on the exhaust stroke* the exhaust valve was *fully open*. You add these two figures together, divide by two and voila, you have the lobe-centre angle. Here's an example:

Cam timing is I.O. 44 ° I.C. 76° Intake duration = 44 + 76 + 76 + 180 = 300° Maximum valve lift will occur at Intake Duration divided by 2 = 150° Crank angle at which intake is fully open = 150° – 44° = 106 ATDC.

Ex. 0. 78° Ex. C. 42° Ex. duration = 78 + 42 + 180 = 300° Maximum valve lift will occur at Ex. duration divided by 2 = 150° Crank angle at which Ex. is fully open is 150° – 42° = 108° BTDC

$$\text{Lobe centre} = \frac{106 + 108}{2} = \frac{214}{2} = 107^\circ$$

Well, that's about it. Armed with the information I've set out, you should be better able to choose the right camshaft for your application.

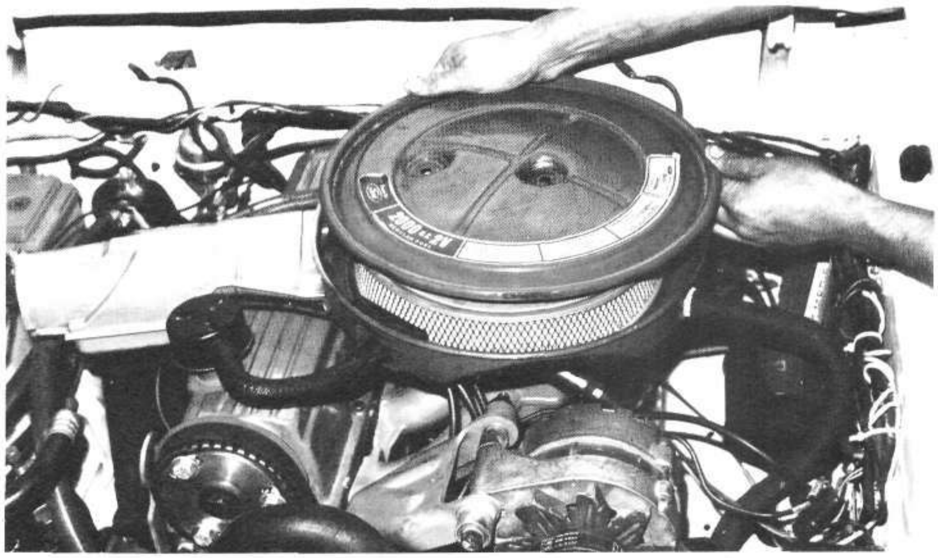
Air Filters, Carburation & Manifolds

AIR FILTERS

The standard air filter assembly can be a point of great air flow restriction. Having some prior knowledge of this you may choose to change the stock air filter case and element for one of the replacement units. Choose the wrong one here and you will achieve little more than a lighter wallet and less horsepower than you had originally. I have found that the air flow abilities of air filters vary considerably from type to type and manufacturer to manufacturer. Testing every applicable air filter on the market would be a mammoth job. I have tested many and those discussed here are a representative sample of the ones tested. But remember. Just because something has "Speed Equipment" stamped on it, doesn't necessarily mean it is a go-fast item.

CLEAN FILTER POWER POTENTIAL

The obvious function of an air cleaner is to clean the air. Under normal circumstances, the longer the air cleaner performs this function, the less its air flow ability is. As the air filter gets dirty, so the maximum power is reduced. For example, a filter was taken from a Pinto which had some 4000-5000 miles on the filter since it was installed. This filter was then mounted in a standard air filter case on a standard carburettor which, in turn, was mounted on an adaptor on the flow bench. Flow figures were then taken through the air filter. This filter was then replaced with a new air filter of the same brand and the same air flow tests performed. If you refer to the chart Fig. 4-1, you will see that the dirty air filter, in spite of only



4-1. The air filter and case on many Ford SOHC engine installations can be a measurable source of air flow restriction. The very least precaution you should take here is to use a clean filter element of a type that is free flowing.

FIG. 4-1 FILTER COMPARISONS

FILTER FLOW EFFICIENCY MEASURED WITH STOCK CARBURETOR AND FILTER CASE

* USED	* NEW	MOTORCRAFT	FRAM	AC
94%	97.8%	99.2%	99.2%	97.2%

*These filters were purchased from a discount parts store and the manufacturer could not be indentified. Apart from this, such stores buy from the cheapest source so the origin of filters so acquired will vary.

FILTER POWER OUTPUT COMPARISON ON CLAYTON CHASSIS DYNOMOMETER

R.P.M.	HORSEPOWER* USED FILTER (94% EFF.)	HORSEPOWER NEW MOTORCRAFT FILTER (99.2% EFF.)
2000	19.0	19.0
3000	29.0	31.0
4000	41.5	42.5
4500	40.0	44.5
5000	33.5	37.5

Test vehicle 2000 cc Pinto 1973 automatic transmission all figures taken in second gear. Tested by Joe Antonelli/David Vizard, January 1977.

having covered a maximum of 5000 miles, was only 94 percent efficient. This efficiency, by the way, is measured by comparing the air flow of the filter being tested, and no air filter at all in the case. The comparison of the two numbers obtained then gives an idea of just how much the air filter impedes the flow.

A new air filter of the same brand as the used one tested, proved to be 97.8 percent efficient. To give you some idea of what can be achieved, I have added figures for other brands as well. Another note: since measuring the air flow efficiency of the Fram air filter, a design change was made that incorporates a cotton belt fitted around the outside of the paper filtering element. This effectively reduced the Fram filter from being joint best, with the Motorcraft filter, down to the same flow efficiency as the AC filter. Thus, if you are going to replace the air filter with a new paper one, use Motorcraft filters (available in the U.S.), or Cooper filters (available in England).

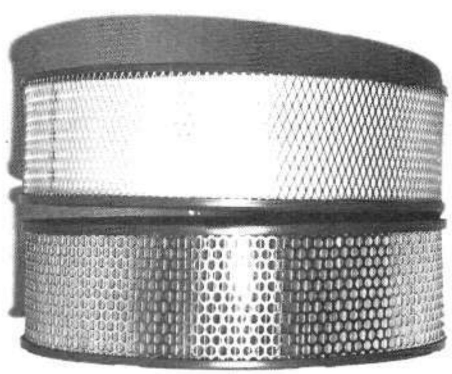
ALTERNATIVE FILTERS

On many vehicles employing the Ford SOHC engine, the proximity of the filter case lid to the mouth of the carburettor is a restriction. Very often, however, this restriction can be substantially reduced by selecting an air filter about ¼ inch taller than the case. This does two things: it yields a bigger area through which to draw in air. Secondly it allows air to come into the filter case around the annular space at the edge of the filter case. Let's take the Pinto as an example. With the standard Motorcraft air filter installed, air flow measured at 25 inches of water pressure drop was 199 cfm. With the air filter removed and the lid of the filter case still in position, the air flow was 200.5 cfm. With the lid of the case removed, the air flow was 227 cfm, thus showing the air filter lid was responsible for a flow loss of 26.5 cfm. A filter ¼ inch taller was installed in the air filter case. The air flow given with this taller filter was 220 cfm. An interesting thing to note here is that if you take the whole air filter case off and just use the carburettor bare without any case surrounding it at all, air flow is only 213.8 cfm, thus showing that the base of the air filter case aids air flow into the carburettor.

TRICK FILTERS

Beware of the so-called trick filter. It is easy to bolt on a negative bhp change. I will give some positive guidelines to steer you clear of modifications that won't work. These rules will apply essentially to almost any two-barrel down- draft carburettor, be it the standard carburettor, a 350 or 500 Holley, a different kind of Weber, of whatever.

As far as filtering elements are concerned, foam element filters are less efficient than the same size paper filter. Indeed, some foam element filters have such an open cellular structure of foam that they do *not* filter. Therefore, if you insist on running with a foam filter, you must have at least 20 percent more area of foam to filter than the original paper element. The only advantage to a foam filter is that it does not clog as quickly and it can easily be cleaned and reused. Beware of the open cell foam filter; all it will do is reduce your engine life. If you want to buy an alternative filtering element, the K & N filter is about the best you can buy. (See chart 4-2.) This filter is also superior as far as resistance to clogging is concerned. On a typical Pinto, installing a K & N filter means the filter could run the entire life of the car without cleaning and still flow better than a brand new paper element filter.



4-2. The Motorcraft filter shown on the left consistently proved to be the best flowing PAPER element filter tested. The cotton belted Fram filter on the right was the worst flowing filter the author could find!

4-3. Many old hands at hot rodding use this simple trick of turning the filter lid upside down to increase airflow to the carburettor. Making this move with a typical Pinto filter case as seen here achieves little, as the now concave form of the inverted lid restricts the flow at the mouth of the carburettor.

The size of the filter and the case dimensions all have an effect on the amount of air flow into the carburettor. Tests have shown that in general, filters under eight inches diameter, irrespective of the air filter area involved, reduce flow drastically. There seems to be a turning point at about nine inches. At this size and above they can be made very efficient. The rule here is: never use a filter less than eight inches diameter: in fact regard nine inches as

FIG. 4-2 AFTER MARKET FILTERS

FILTER FLOW EFFICIENCY WITH 8" AFTER MARKET CASE ON STOCK CARBURETTOR

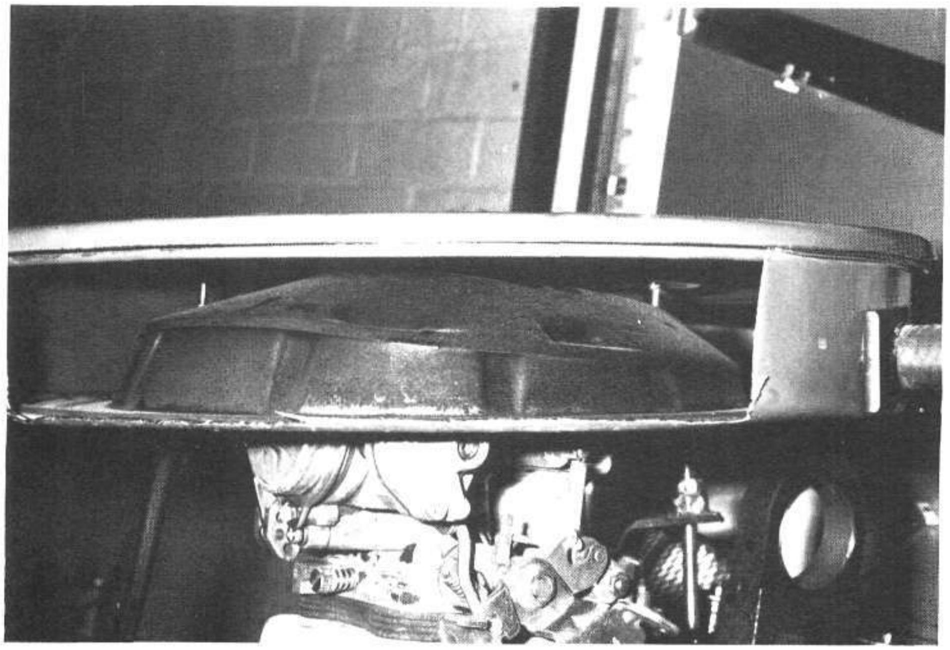
FOAM ELEMENT	* TALL PAPER ELEMENT
97.5%	98%

* This filter gave the same filter area of the stock filter but flow is still below best achieved with the stock filter.

FILTER FLOW EFFICIENCY ON 350 C.F.M. 2 BARREL HOLLEY

8" CASE.FOAM ELEMENT	* 8" CASE.PAPER ELEMENT	9" C.P. CHROME CASE PAPER ELEMENT	9" C.P. CHROME CASE K & N FILTER ELEMENT
92.2%	93.1%	99.0%	No measurable flow loss.

* This is the same tall element as used for the test with the stock carb.



4-4. To an extent this is already slighty restricted when the lid is up the right way as is illustrated by this cutaway filter case.

the minimum. An oval filter 8-9 inches along its principal length will work fine.

If you intend using a Holley four-barrel carburettor on your engine, there are a few effects you can take advantage of in the air filter department. Because a four-barrel carburettor draws air over a wide area due to the spacing of each barrel of the carb, it is not so sensitive to the proximity of the air filter lid, as far as shrouding is concerned. In fact, if the air filter lid is positioned at a certain distance away from the carburettor mounting face, the air filter top actually *aids* flow into the carburettor. The placement of the lid, relative to the carburettor throats, does not appear to be that critical, and typically any low-profile air filter helps the flow. However, it is sensitive on diameter. You should regard 10 inches as about the minimum. It is possible to get away with nine inches but you are starting to lose out. As far as the air filter element is concerned, I recommend either the standard Motorcraft or Cooper paper element filter or best yet, a K & N.

On engines equipped with one carb barrel per cylinder, the air filter situation becomes critical. While testing on Oselli Engineering's dyno, engine builder David Ray found that on a Group 1 Escort RS engine equipped with Solex carburettors, the air filter system was responsible for a drop of 10 bhp. Such a large drop in power almost invalidates the use of the one barrel per cylinder-type carburation. This substantial drop in power, due to the air filter system, seems peculiar to Solex carburetted Group 1 engines. It would appear that the later Group 1 setup employing downdraft Weber carburettors and a redesigned air filter box, overcame most of these problems. If you are installing side-draft Webers or Delortos, the quickest and simplest air filters are the foam element filters that bolt directly to the carburettors. Although I have not tried every brand on the market, those I have tried range in performance from average to useless. Further, most, if not all, suffer from one basic flaw in design; the carburettor ram pipes have to be removed to allow the installation of the air filter. *This is bad.* Whatever filter you use, try to incorporate some form of curved entry into the carburettor. Having a sharp entry loses air flow. This can lead to erratic carburation at lower revs. This may not be felt on a manual transmission car, but it certainly does show up on an automatic transmission car, spoiling the whole drivability of the machine. If you are going to the trouble and expense of side-draft carburettors, take all steps to equip them with a large air box, or large K & N air filters and K & N Stub Stack ram pipes.

The ideal setup for a side-draft carburettor is an airbox on the carburettor and a remote air filter drawing cool air from the front of the car to feed the air box. So long as the air box is of reasonable dimensions it is possible to have an air filter system on your engine which actually increases the bhp. Aerodynamically, having an air box with proper volume gives the shock waves within the impression that they have struck open air. In turn, this yields more positive filling of the cylinders at certain parts of the rpm range.

The Manx Racing air box is an example of an air box big enough to get the job done. This particular box is designed for remote mounting of the air filter.

HOW BIG?

The next question is, how big an air filter is needed for any given situation? Formulas exist which tell you the minimum size necessary to give a 99 percent efficient air filter element. Sure, you can make the filter bigger than the following formulas describe, and it will be to your advantage. But any gains that may be made in power are subject to the laws of diminishing returns with respect to overall size. These formulas also only give the effectiveness that an air filter will have when it is perfectly clean. If your driving will be in very dusty conditions, double the size of the air filter that the formula suggests. Further, for off-road competition, *do not use a paper element filter* because it *will* clog far too quickly. Go straight to a K & N filter; don't bother with anything else.

Be aware that this formula is not infallible. For example, when dealing with engines of lesser bhp, it may suggest an air filter smaller than the minimum specified earlier. If this is so, ignore the calculation and regard the aforementioned diameters as the ones to use.

Now for the formula. For a *good* paper element air filter; the filtering area

$$A = \frac{CID \times RPM}{20{,}900}$$

CID is the cubic inch displacement of the engine. If you know your capacity in cubic centimeters, then divide this by 16.39 to yield displacement in cubic inches. RPM is the engine speed at which you estimate peak HP to occur.

If you intend to use a foam element filter, then the formula will be:

$$A = \frac{CID \times RPM.}{18{,}870}$$

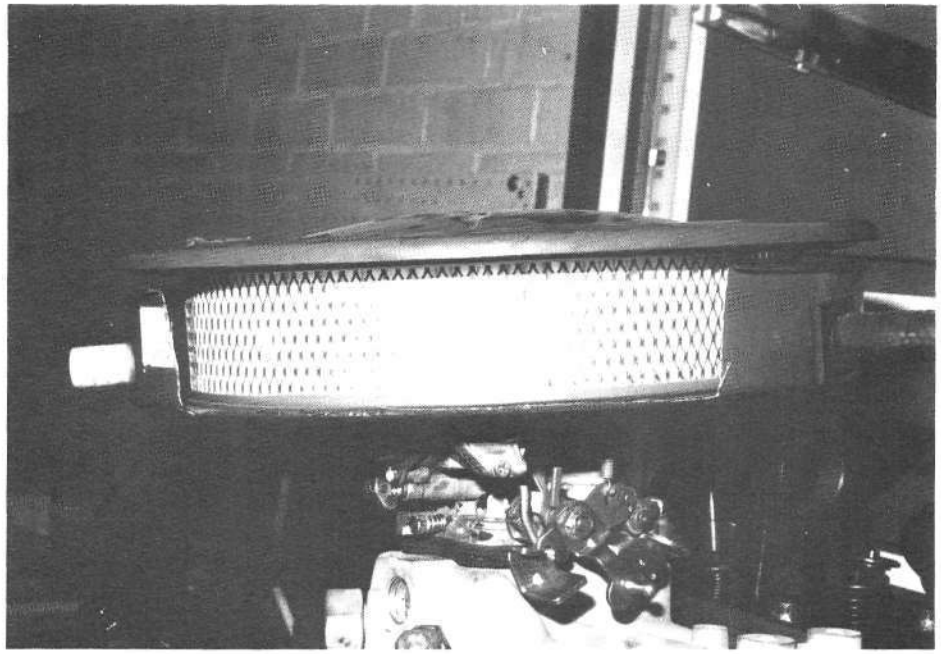
If you use a K & N filter, the formula is,

$$A = \frac{CID \times RPM}{25{,}500}$$

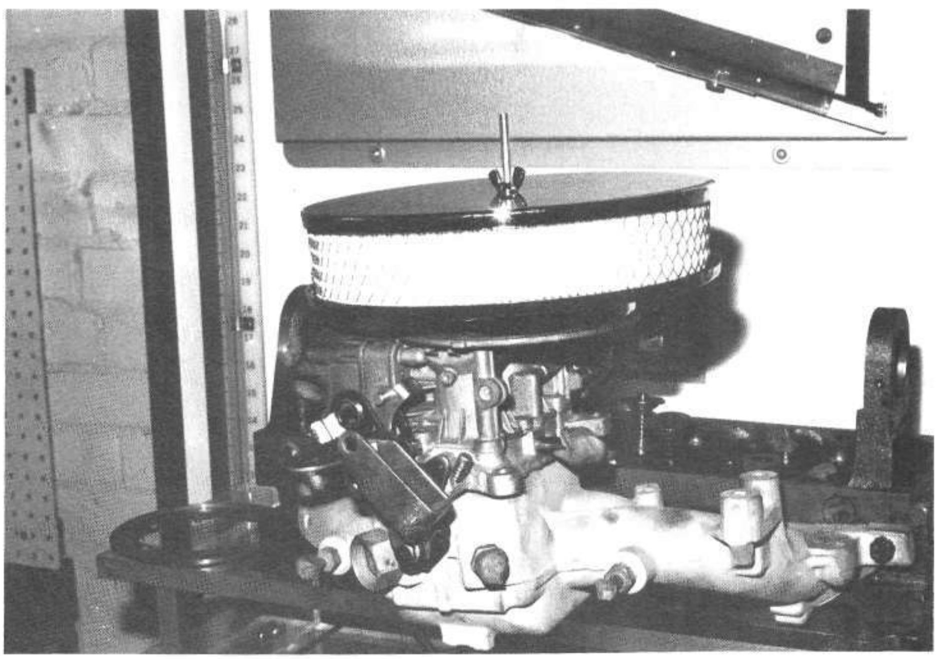
To realize the *minimum* air filter diameter, you will need to use the following formula:

$$\frac{A}{AFH} \times 3.142.$$

A is the area you have just calculated from one of the above equations. AFH is the Air Filter Height you intend to use. This height is measured between the rubber sealing rings of the air filter element, not the overall height. Generally speaking, the effective height of an air filter is 1/4 - 3/8 inch less than the overall height



4-5. By using an air filter 1/4" taller, three air flow restriction points prior to the carburettor mouth are relieved, these being: the snorkel restriction, as much of the air now bypasses it. The air filter size is increased, so helping flow and the lid is moved 1/4" further away from the carb mouth.



4-6. Before installing an air filter like this, I advise checking out the flow figures in Fig. 4-2. This should demonstrate the fact that it is strictly a cosmetics-only item.

CARBURETTORS & MANIFOLDS

The standard two-litre engine can greatly benefit from increased carburettor air flow. Given all the carburation it can use, a standard two-litre engine would produce about 12-14 bhp more. Smaller engines can also benefit from extra carburation, especially those with a restrictive single-barrel carb. If your vehicle has a single-barrel carb your best plan, if cost is a prime factor, is to replace it with the standard Ford two-barrel unit. This, complete with manifold and linkage, can usually be bought at a breaker's yard.

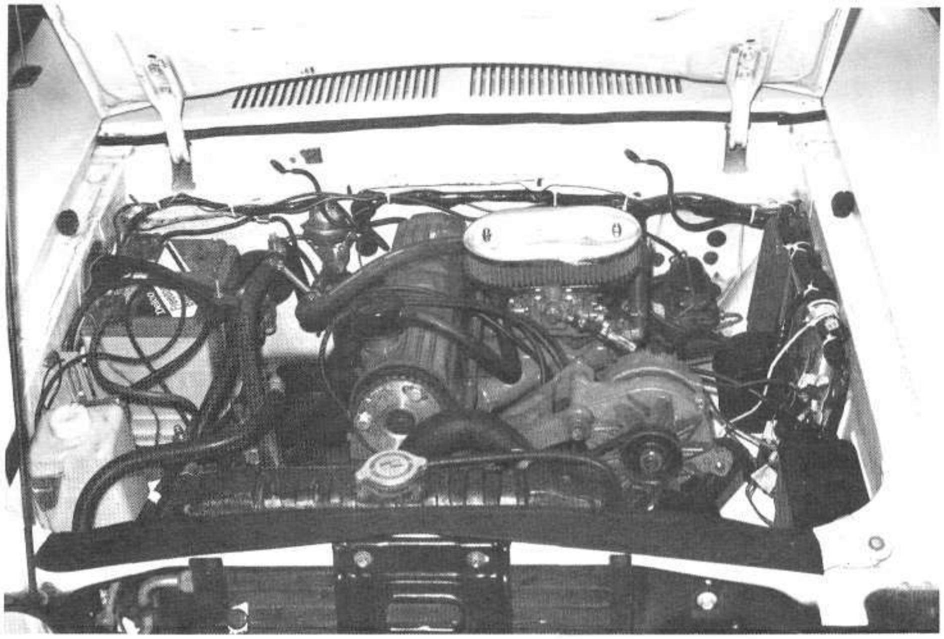
As the engine becomes more radically modified, the carburettor's effect on the power curve becomes more critical. At each stage of the game, the key to producing *the best power curve for money spent* is to have carburation *adequate* for the job in hand. Spending large sums of money on carburettors having huge air flow capacities does not necessarily produce that much *more* power than an *adequate* carburettor. Indeed, in some cases it may produce less. If you are after all the bhp possible obviously you must use the best carburation you can afford. Most hotrodders must modify their engines with some sort of budget in mind. I will start with the cheapest effective modifications first and progress to the expensive ones later.

UTILIZING STANDARD PARTS

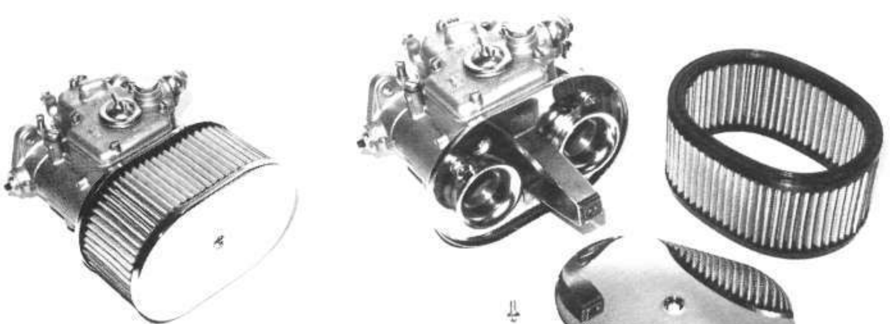
The standard intake manifold utilizing the Holley/Ford/Weber carburettor, is good. Flow bench and dyno tests show it is capable of performance superior to almost all the other two-barrel manifolds on the market. It is also capable of producing more power than some of the four-barrel carb/manifold combinations. In standard form, the intake is about 93 percent efficient. This figure is arrived at by measuring flow with and without the manifold. Flow distribution, – how well one runner flows in relation to another, – is also good. Typically all runners are within 1 ½ percent of each other. With a few minutes' work, this manifold can be improved further by radiusing the areas indicated in the accompanying photo.

The main factor limiting air flow in the standard intake assembly is the carburettor itself. Since the standard manifold works so well, we would expect that any increases in air flow made at the carburettor would show up as a bhp gain. Testing proves this out; even small increases in airflow made at the carburettor readily show up on the dynamometer.

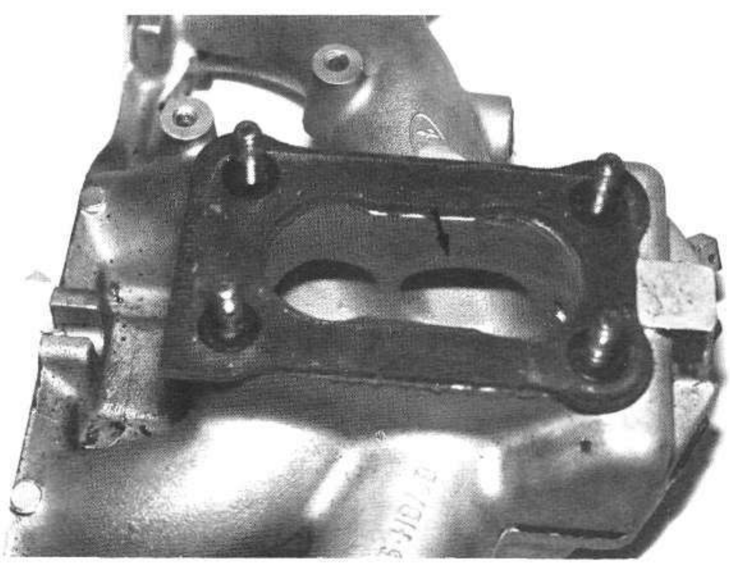
Working on the premise that modifications costing little, yet producing worthwhile extreme power, are the



4-7. Apart from being highly functional, this K & N filter case and element made a smart looking installation on the author's Weber carburated Pinto.



4-8. A & B Finding a satisfactory filter setup for a twin side-draft carburettor installation can be difficult as most range from indifferent to useless. This K & N setup, however, is one I can recommend. Not only does it allow high airflow without measurable restriction but it also allows the carb ram pipes to be used.



4-10. By increasing the radius on each port runner at the junction with the plenum, the flow of this already efficient manifold can be increased. A large radius on *the top edge produces the greatest gains in flow.*

FIG. 4-3

All measurements at 25" H_2O

	1	2	3	4	5
C.F.M.	227	237	239	262	265

1. Standard carburettor mounted on adaptor on flow bench. Base of standard filter in place. Filter and filter case lid removed.

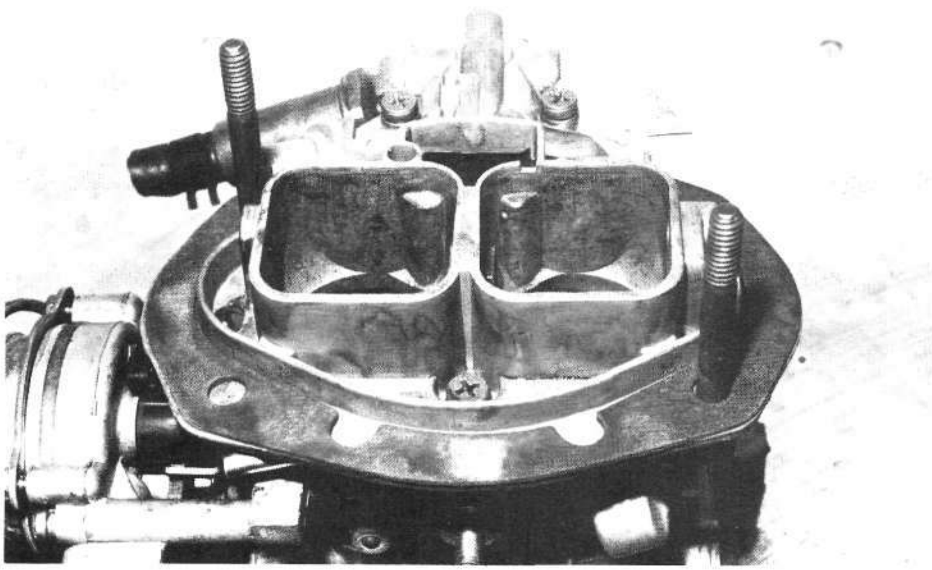
2. As for test 1, but butterfly shafts narrowed and butterflies knife-edged.

3. All 'flash' marks removed from booster (auxiliary) venturi and edges of air horn rounded off.

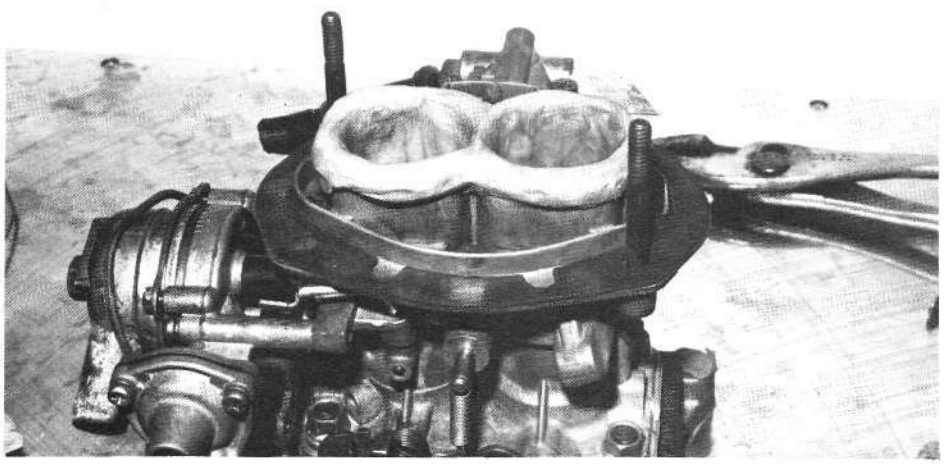
4. Primary barrel venturi bored to 30.0 mm and secondary to 31.0 mm

NOTE: If boring out venturis breaks through casting, it can be easily repaired with any epoxy resin adhesive.

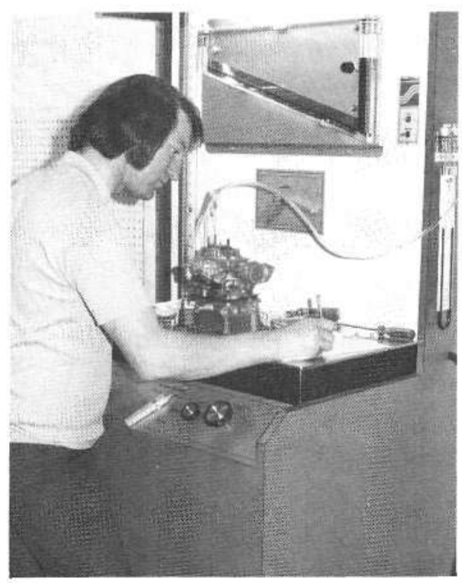
5. Choke butterfly removed.



4-11. Removing the choke butterfly was tried at various stages of carb modifying. The flow bench showed that it was generally worth 3 cfm at whatever stage it was tried.



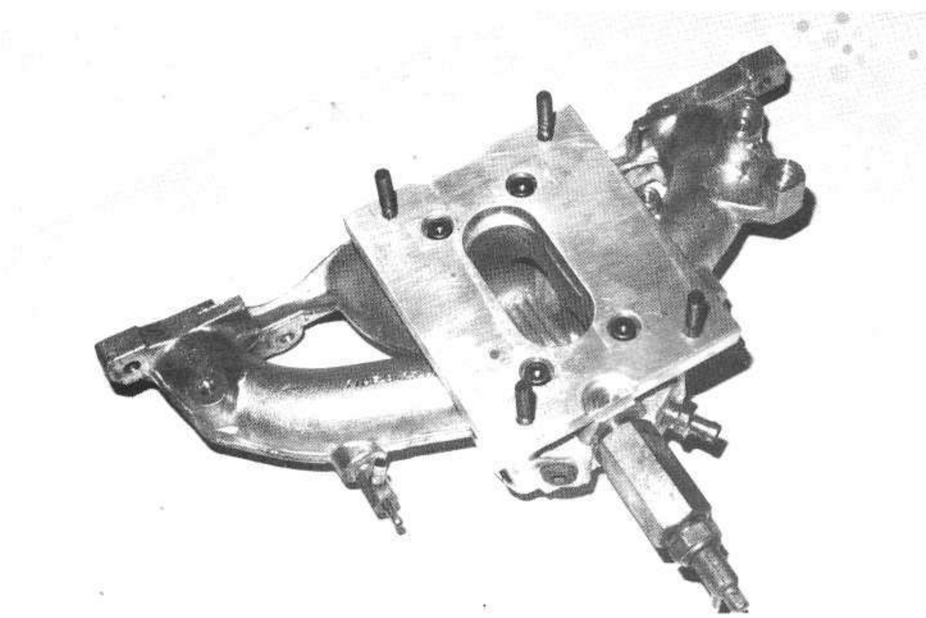
4-12. A radius on the carb intake, smoothed out airflow but produced very little in the way of extra airflow. The dyno, however, tells a different tale. A generous radius prior to the carb mouth almost always gave about 2 hp increase.



4-13. Here David Vizard checks the "signal strength" to various jets after carb modifications have been made. This is done by making tappings into the various circuits of the carb and measuring the pressure drop that occurs when a given amount of air is passed through the carb.

most desirable, let's consider modifications to the standard carburettor. Fig 4-3 lists modifications that increase air flow from 227 cfm to 265 cfm. As usual, this is measured at 25 inches water. You may want to compare these air flow figures with most U.S. (2 BRL) carburettors, which are typically measured at three inches of mercury. To do this, multiply the figures in chart Fig. 4-3 by 1.27. Interestingly, measured at the conventional three inches hg. pressure drop normally used for two barrel carburettors, air flow increases from 288 cfm to 336 cfm.

More important than air flow gained solely through the carburettor is air flow gained through the *total* induction system of the engine. Taking the induction system on a standard two-litre engine, the flow loss at a selected representative valve lift is 21.6 percent. If you performed the air filter modifications I mentioned earlier, that is, using a taller element, preferably the K & N type, total air flow loss will be down to 19.4 percent. If you now modify the carburettor along the lines previously described, the total air flow loss will be down further to 14.2 percent. On the dyno, these flow gains represent a solid gold increase on a standard engine.



4-14. A 350 or 500 cfm two barrel Holley carb on the standard manifold makes a good alternative to the standard carb. An adopted plate as shown here is required. In the past I have obtained mine from Racer Walsh or Spearco.

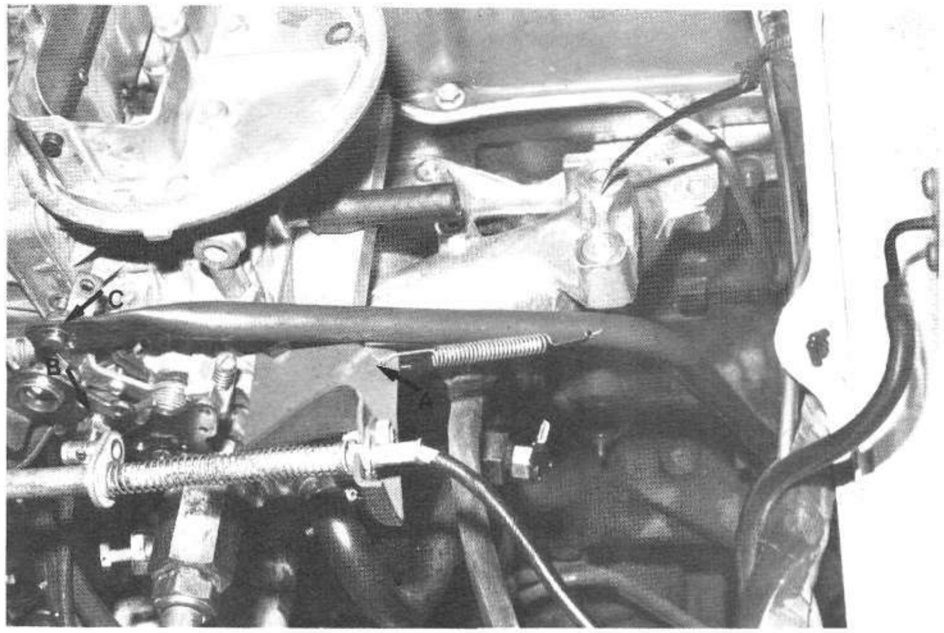
4-16. If you intend using a two barrel Holley on a Pinto with emission equipment, you will need to alter the carb linkage a little in many cases. For instance it is often necessary to cut part of the original linkage at the point indicated so that it clears everything when the throttle is opened.

ALTERNATIVE CARBS

Since the factory manifold is so good, the limiting factor becomes the maximum abilities of the standard carburettor. And past this there comes a time when the carb cannot be easily modified to yield any more air flow. At this point adapting a higher air flow carburettor to the standard manifold, is the most straightforward solution.

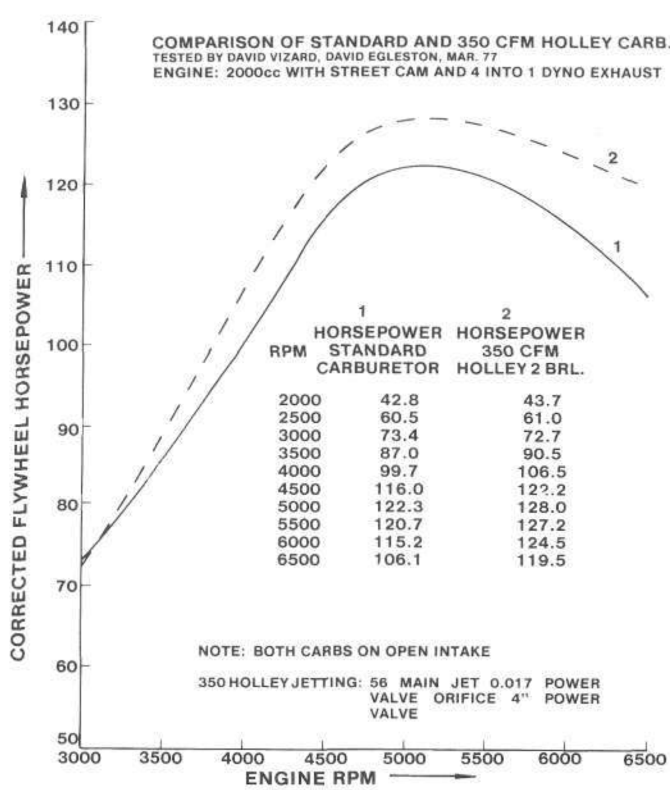
Two carburettors which have proved highly successful are the 350 and the 500-cfm Holley two-barrel carburettors. Many Pinto specialists in the U.S.A. stock an adaptor plate to convert the stud mounting on the standard manifold to the bolt-hole pattern on the 350 and 500 Holleys. Calibrating a 350 or 500 Holley for a two-litre engine is very simple if you bear in mind one thing: most of these carburettors are built for engines of at least double the displacement. As such, the power valve restriction orifice is normally way too large for a two-litre engine. For most street applications on a Pinto engine or any two-litre Ford OHC engine this orifice needs to be around 0.017-0.020 inch (0.43-0.51 mm) diameter. If it is too large, when the power valve comes into operation, the mixture will be very rich and power will take a nose-dive. And, if you lean-out the main jet to compensate, you run into drivability problems at part throttle. Apart from this, calibrating the Holley for use on the Pinto engine seems a straightforward operation. Not only does it achieve good bhp gains, it also provides with regard to emissions, a very clean running engine. On my own engines I have found I have been able to meet emission standards easier with a 350 Holley than the standard Holley/Weber carburettor. To give you some idea of how much power the 350 Holley is worth over the standard Holley/Weber carburettor, see Fig. 4-4.

If a 350-cfm carburettor is good, how much better can a 500- cfm carburettor be? Well, among other things, this depends on the engine's state of tune. If you have a cam which pulls fairly high rpm, a modified head and a free flowing-exhaust system, then you possibly can utilize the 500. Again, in my experience there appears to be little or no disadvantage in using the 500 on mildly tuned engines. It's just that if an engine doesn't need it, it doesn't gain much bhp. If you expect your engine to make over about 130 bhp and you expect to rev it up to 7000 rpm or more, then the 500 cfm/Holley pays off. See Fig. 4-5, which shows the effect of substituting a 350 Holley for a 500. This particular engine had a Ford Stage II cam in it and a modified head. It had been developed to the stage where the extra cfm could be utilized. Note that there is virtually no difference in the power curves of these two carburettors until about 5500, but from here on up, the power curves diverge. Importantly, if you have previously run a 350 Holley and you decide to use a 500 Holley, it will be necessary to modify the adaptor plate so that the larger butterflies on the 500 Holley clear the adaptor plate.



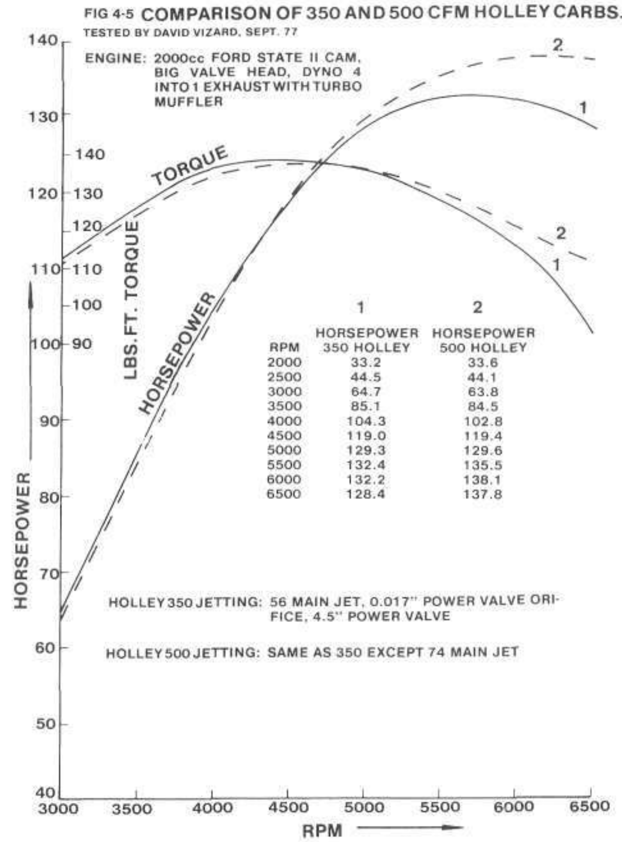
4-17. If you are going to use a 2-barrel Holley on an automatic Pinto there will be some linkage fabrications to do but the effort is well worth it. The throttle cable bracket (arrow A) has been cut out of the original bracket. Compare what you see here with your own throttle cable bracket and modify it accordingly. Arrow B indicates the cable connection to the throttle lever and arrow C the transmission "kick-down" attachment point.

FIG 4-4



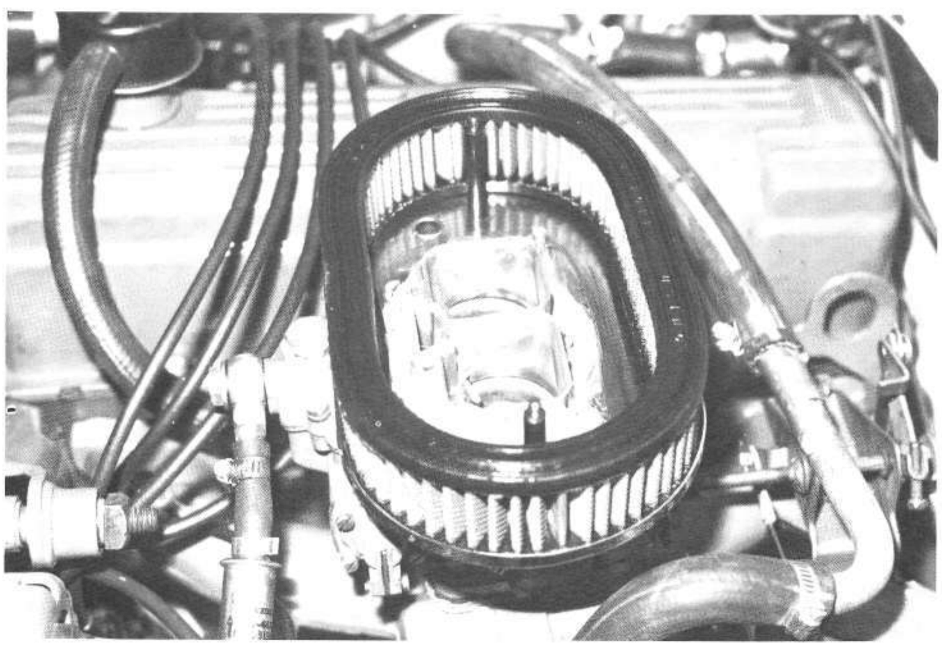
RPM	1 HORSEPOWER STANDARD CARBURETOR	2 HORSEPOWER 350 CFM HOLLEY 2 BRL.
2000	42.8	43.7
2500	60.5	61.0
3000	73.4	72.7
3500	87.0	90.5
4000	99.7	106.5
4500	116.0	122.2
5000	122.3	128.0
5500	120.7	127.2
6000	115.2	124.5
6500	106.1	119.5

FIG 4-5



RPM	1 HORSEPOWER 350 HOLLEY	2 HORSEPOWER 500 HOLLEY
2000	33.2	33.6
2500	44.5	44.1
3000	64.7	63.8
3500	85.1	84.5
4000	104.3	102.8
4500	119.0	119.4
5000	129.3	129.6
5500	132.4	135.5
6000	132.2	138.1
6500	128.4	137.8

4-18 & 4-19 It will be necessary to make up your own transmission "kick-down" rod when installing a two barrel 350/500 Holley on your Pinto. These two shots show the form it should take.



4-20. I tried a Weber 40 D.F.I.5 Weber both in standard form and modified on a 2-litre Pinto. It produced good hp but there were many drawbacks as the text explains.

ALTERNATIVE WEBERS

Although I have not tried it myself, I have heard reports that the factory engineers have achieved very good results with a 38 DGAS Weber on two-litre engines. Drivability is said to have been improved and power increased by about five bhp. This carburettor will bolt directly to the standard intake manifold, so apart from installing an air cleaner (K & N) and throttle linkage, it is a straightforward conversion. For my own part, I have tried a 40 D.F.I.5 Weber on a 2000 engine. This carb flows about 330 cfm and was originally intended for use on engines having one barrel per cylinder. I believe this carburettor was normally used on Ferraris. It bolts directly onto the Pinto manifold. As far as maximum power is concerned, it works very well, but it does suffer from one drastic problem: it has no power valve and therefore must be calibrated very rich. The result is a tremendous amount of exhaust pollution, very high fuel consumption and bad drivability until close to full throttle is reached. At full throttle, however, it produces a commendable power curve. For my own part, I advise you to steer clear of this carburettor, not because there is anything particularly wrong with it but because it is unsuitable for our purposes. It was never designed to go onto a setup whereby more than one cylinder per barrel is used.

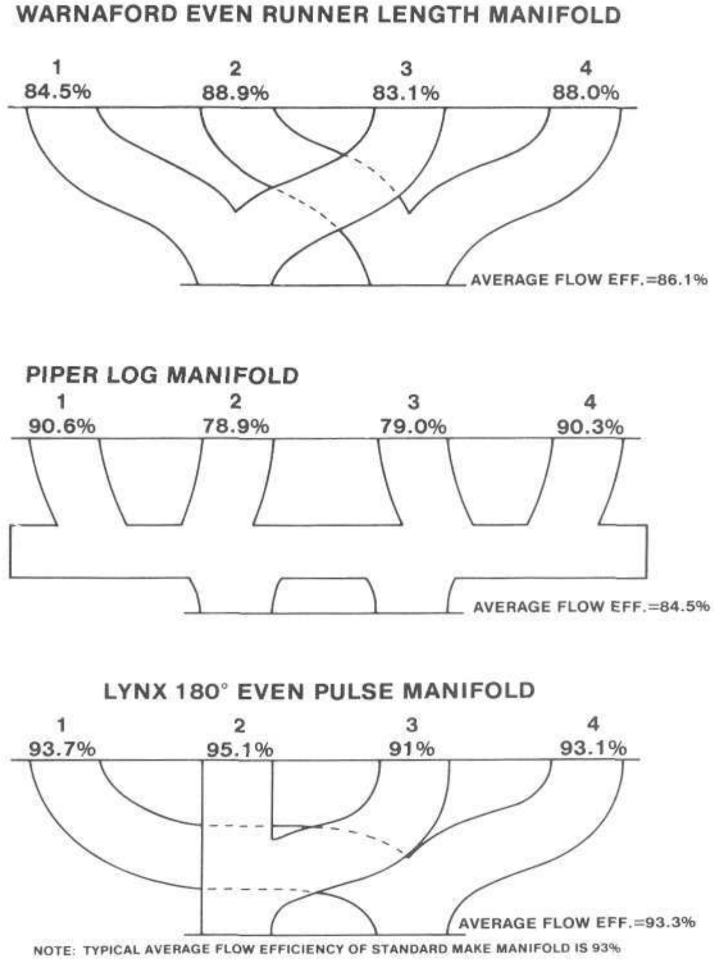
SINGLE SIDEDRAFT MANIFOLDS

There may be more, but I only managed to track down and try three single side-draft manifolds. Each has a different mode of operation. If you refer to Fig. 4-6 you can see the basic port layout of each type. The Australian-made Warnerford manifold is an equal-length type. As can be seen from the photos, the passages cross over each other, and in doing so, there is a considerable change in cross-sectional shape and area. As a result, this manifold has poor air flow. Some of the ports have flow efficiencies as low as 83 percent. The average flow efficiency of this manifold is 86 percent, *considerably less than the standard manifold.* It is possible to achieve more power with this type of manifold and a single-side draft Weber than with the standard manifold and the carburettor. Here is the reason why: on the standard manifold/carburettor setup, the carburettor is the limiting factor, not the manifold. With the Warnerford manifold/side-draft carburettor combination, the carburettor will flow considerably more air than the manifold, and the *total* efficiency of the carburettor/manifold combination may be up on the standard carburettor and standard manifold combination. However, if you compare 500 cfm of carburation on both the Warnerford manifold and the standard manifold, the standard manifold will give more air flow and more power.

Next, the Piper log manifold. In essence, the theory behind its design is sound but the manifolds itself is poorly executed. Of the three manifolds this had the lowest average flow efficiency, only 84.5 percent. The biggest hangup is caused by the two centre cylinder runners being acutely angled away from the direction of air flow. This means the air has to make a very sharp turn to enter these port runners. Result: very poor flow efficiency as you can see from the numbers on Fig. 4-6. Again, this manifold failed to produce as much power as a typical 350 Holley on the standard manifold.

Last, we come to the Lynx 180° even-pulse manifold. This is one of the few manifolds that I've tested apparently having some serious thought done on it. It is also the only one I've tested having flow efficiencies which were as good or better than the standard manifold. Since it was the only manifold of the group that really did *work*, it bears looking at in more detail. First of all, why the 180° setup? Would it not have been easier to have the right barrel of the carburettor feeding the two right-hand cylinders and the left barrel feeding the two left cylinders? Surely the air flow could have been made better under these circumstances. The answer to that is "Yes, it could." Unfortunately, however, this means each throat gets two pulses and then a long idle period where air flow is minimal. As a result, a mixture mal-distribution problem arises. There is a tendency of each cylinder fed by each barrel to have an *incorrect mixture in both cylinders.* But the Lynx manifold was designed for normal street use. It was designed to give good mileage and good throttle response even at low revs while also taking emission requirements into account. By having pulses spaced 180° apart on each carburettor barrel, mixture distribution problems are minimal

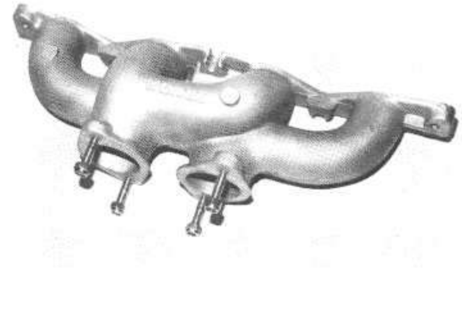
FIG 4-6

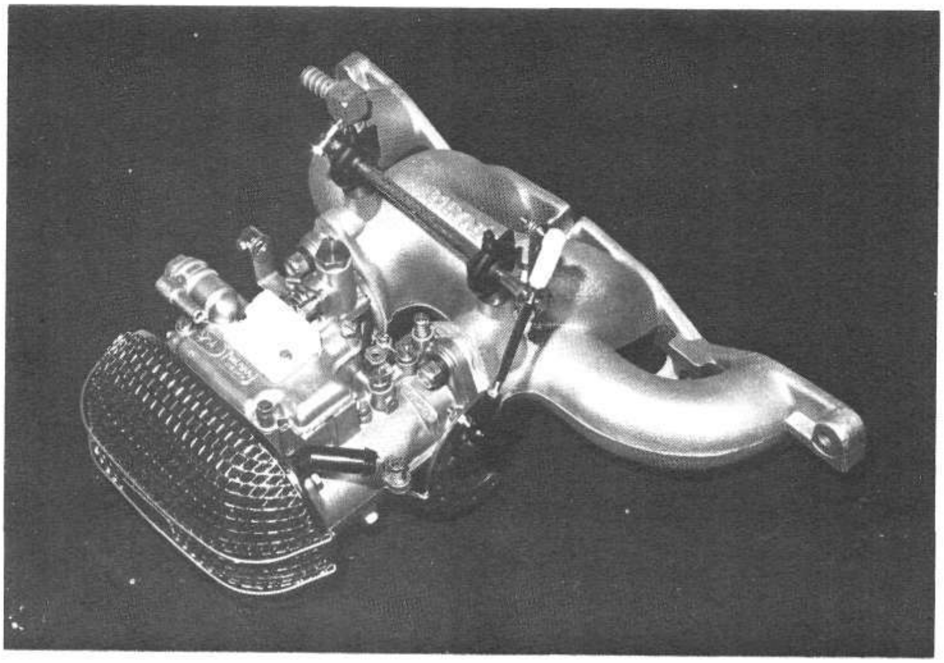


4-21. The flow bench showed the Warnerford single side draft manifold was FAR LESS efficient than the standard Ford intake manifold. The dyno showed corresponding results.

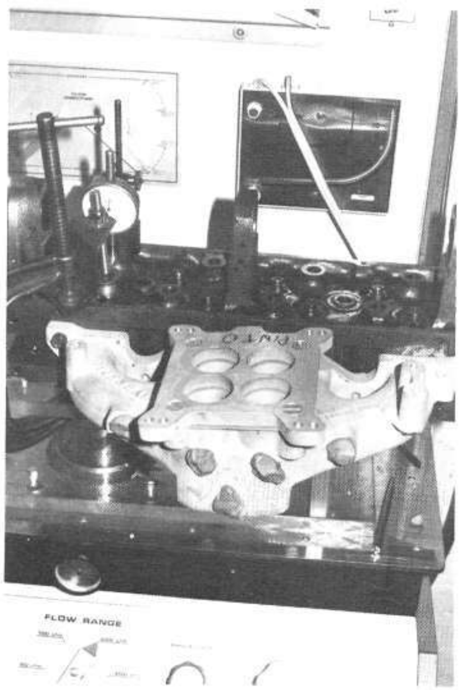
4-22. With carburettors of equal flow potential, the Pipe "Log" Manifold produced less air flow hp than the standard Ford intake.

4-23. My own choice of single side draft manifolds would be this Lynx unit. It was the only one I tested which produced better flow efficiencies than the standard manifold.





4-24. Here is the single Dellorto kit as it comes from Lynx. This setup proved to be just about the best induction setup for torque in the low and mid speed range, thus making it ideal for an automatic transmission or towing vehicle. Top end hp was similar to that produced by a 350 cfm Holley. For optimum results the only change I would make to this setup would be to use a large K & N filter and 50 mm (or thereabouts) long ram pipes.



4-25. This split port Offenhauser manifold worked well on the flow bench and on the strip. The principle of separate ports WITHIN EACH PORT for the primary and secondary barrels works well, probably because of excessive port size in the head. The primary barrels introduce the fresh charge into the fast moving air at the TOP of the intake port in the head.

as each cylinder operates under more or less identical pulse conditions.

Although this manifold works well with a Weber carburettor, I suspected that it might work better with a Dellorto carburettor due to the latter type of carburettor's better fuel atomization. A call to Lynx's John Bruderlin in Australia confirmed this. Their findings have been that the Dellorto not only gives better low-end throttle response (important on an automatic transmission car), but also gives a few more bhp at peak revs. Talking of RPM ranges and automatic transmission cars lead me on to this subject in particular. However, you should know that this manifold is not a high rpm unit. Depending on the cam and the head-work that's been done, my own experience has been that the power curve flattens at about 6000 rpm. Other people have reported this, and Lynx themselves claim this manifold is not meant for high rpm applications.

With respect to automatic transmission torque demands, this is one of the best manifolds for such use. Dyno testing has shown that in the 1300-2300 rpm range, this manifold gives more torque than twin side- draft manifolds. Again, this may be due to better atomization, because two cylinders draw from each barrel rather than one. However, due to the 180° layout, each carburettor barrel is unaware, at least in the lower rpm range, that it is feeding anything else but one cylinder. Because manifold runner volume is small, minimal reduction of the induction pulses due to plenum chamber action takes place.

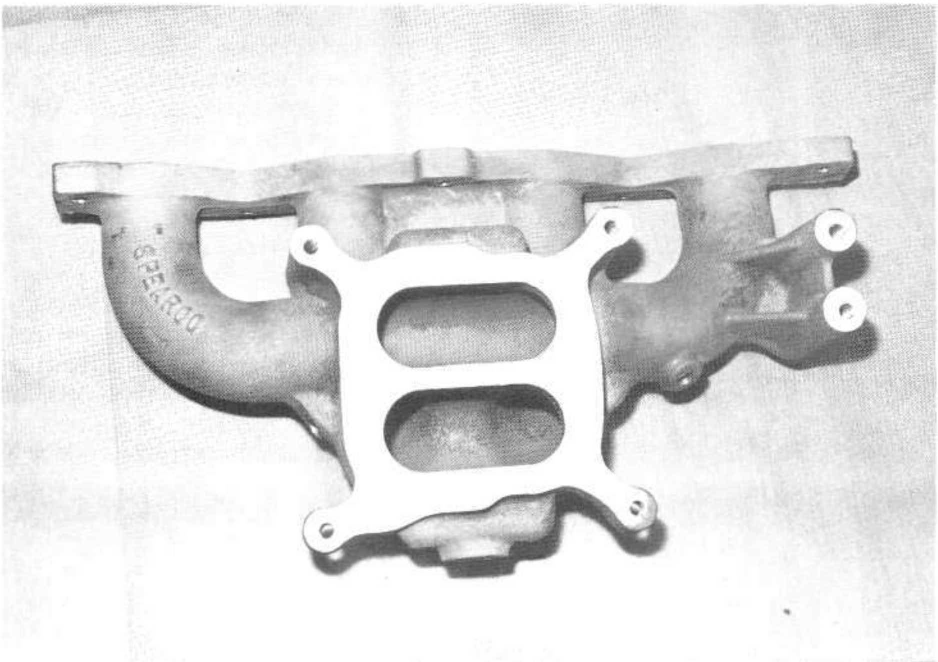
If you are contemplating an alternative carburettor/manifold for an automatic transmission car, a Dellorto carb on this manifold is well worth a try. As far as maximum power is concerned, figure on a power curve marginally superior at top revs to that of a 500 cfm Holley.

FOUR-BARREL CARBURETTORS

Airflow comparisons made between two-and four-barrel carburettors relate directly to manifold flow efficiency. Some four-barrel manifolds divide carburettor flow into two barrels for two cylinders. Flow bench testing is normally carried out by opening the intake valve on one cylinder then drawing air through the valve, manifold and the carburettor. Evaluating a four-barrel carburettor in such a manner has the effect of halving the effective capacity of a carb, thus a 390cfm four-barrel carb acts like a 195cfm two barrel.

Flow efficiencies for four-barrel manifolds typically range between 90-95 percent, even for those with divided runners or divided plenums. Most, if not all, four-barrel manifolds available from such companies as Spearco, Racer Walsh, Weiand, etc., are intended for use with a Holley 390 cfm four-barrel. carburettor. This carb has two small primary barrels giving good atomization at part-throttle operation. The larger secondary barrels are vacuum-operated and come into operation only as the engine demand warrants. In an automatic transmission Pinto I owned, I used a Spearco exhaust manifold and a Spearco intake manifold mounting a four-barrel 390 cfm Holley and one of my own modified heads. This combination raised the rear wheel horsepower from 51 to 69. However, a 350 two-barrel on the standard manifold also produced 69 horsepower at the rear wheels. This result seems to be fairly common when comparing two- and four-barrel installations, and the four barrel advantage is, *in most cases* marginal. Where four barrels do score over the big single two barrel, is in drivability and smoothness. This seems to be most evident on engines equipped with the standard cam with little or no head-work. I favour the small four-barrel carb for use on a mildly modified engine, especially with auto transmissions.

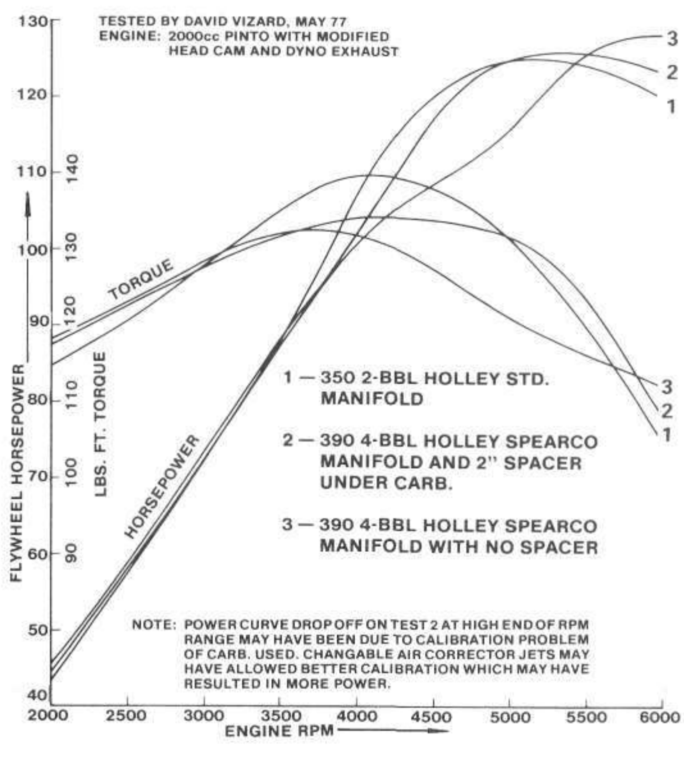
Short duration, high-lift street cams and ported heads can produce sharp induction pulses. In turn these produce air flow velocities in a 350 Holley two-

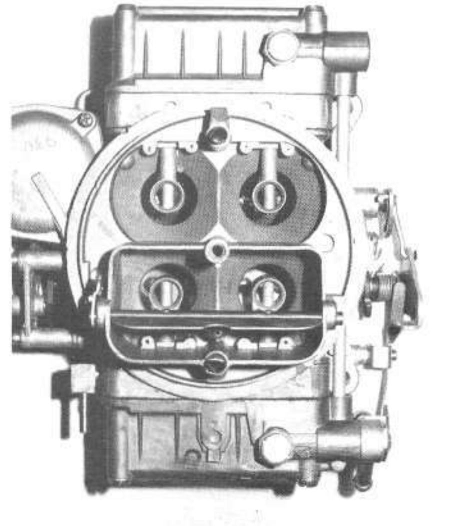


4-26. This Spearco manifold does, in effect, divide a single four barrel into two two-barrel carbs of half the capacity. Adding a 2" spacer . . . . .

4-27. . . . . . as seen here allows all four cylinders to draw from all four barrels of the carb. See Fig. 4-7 for the effects on the power output when used with and without spacer.

FIG 4-7
COMPARISON OF HOLLEY 350 CFM ON STANDARD MANIFOLD AND 390 4 BRL HOLLEY ON SPEARCO MANIFOLD WITH AND WITHOUT 2" TALL SPACER UNDER CARB.



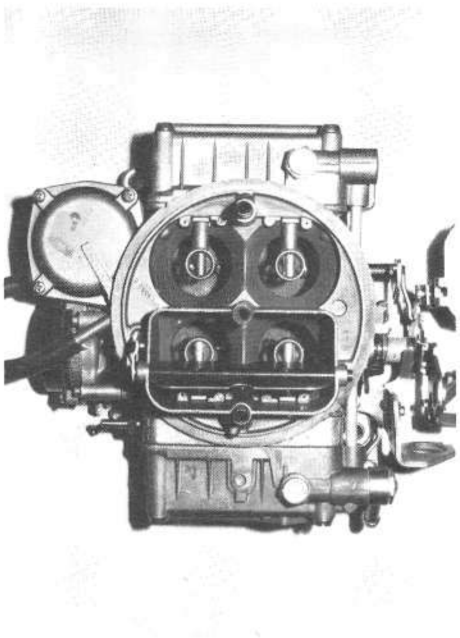


4-28. This 390 cfm Holley four barrel was designed specifically for a small displacement engine. The size of the venturis compared with its . . . . .

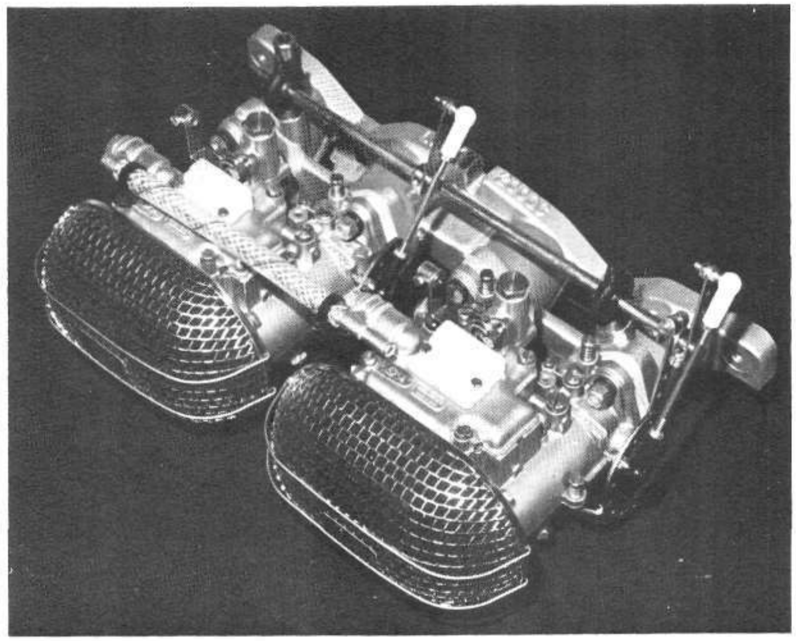
barrel carb, high enough to compensate for drivability problems stemming form large throttle bores. But cam timing is touchy. If duration is just a little too long, drivability may suffer. With some hot street cams and two-barrel installations, drivability can be *seriously affected* by as little as 4 ° too much advance on the *camshaft*. This situation tends to be less critical if a four-barrel carburettor is used.

COMPETITION FOUR-BARREL CARBURATION

Virtually all the four-barrel manifolds available for the Pinto are intended mainly for street use. At the time of writ-

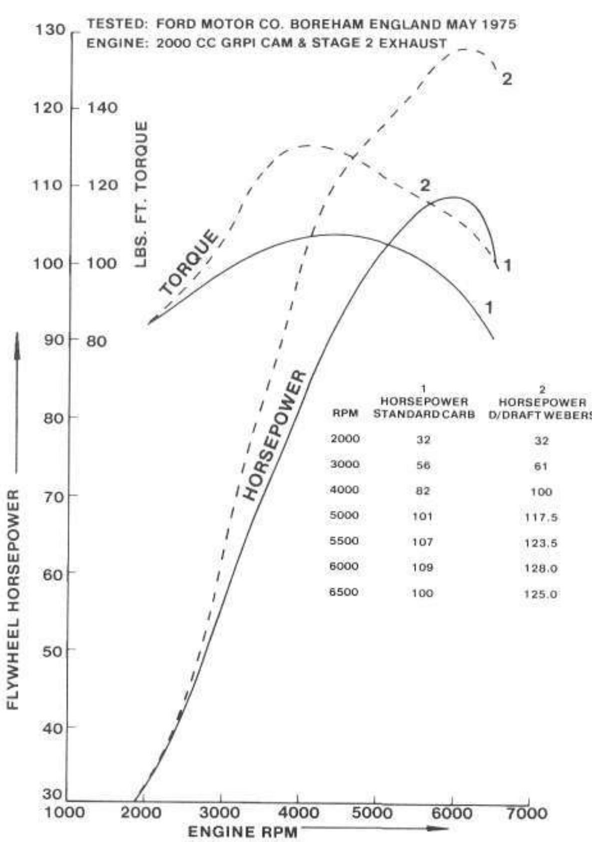


4-29. . . . . bigger brother, the 600 cfm Holley, can be seen here. A really high winding, well developed engine could probably just about utilize a 600 cfm Holley.



4-31. This excellent twin side draft Dellorto setup is available as a kit from Lynx in Australia. Basing your carburation on Dellortos is a good starting point for any high performance Ford SOHC engine.

FIG 4-8



RPM	1 HORSEPOWER STANDARD CARB	2 HORSEPOWER D/DRAFT WEBERS
2000	32	32
3000	56	61
4000	82	100
5000	101	117.5
5500	107	123.5
6000	109	128.0
6500	100	125.0

ing, no manifolds exist that employ the tunnel-ram principle to its fullest extent. Since this manifold design has produced such effective results on V8 engines, it is surprising that no one has applied it to a small, four-cylinder engine, utilizing a single four-barrel carburettor. Although the manifolds previously discussed will normally mount 390 cfm four-barrel Holleys, they can be just as easily equipped with Holleys all the way up to 850 cfm. It is doubtful whether even a radical banzai 2000cc engine could utilize anything flowing more than about 600 cfm. However, I feel if a good manifold were designed, this would be the way to go. To put this sort of manifold into perspective, dyno testing on V8 engines has shown that some tunnel ram manifolds are even superior to independdant runner fuel injection installations.

ONE BARREL PER CYLINDER

Let's start off with the factory manifolds. These are the ones you can get from Ford's Competition Department. The Solex carburettors that were first offered as Group 1 items are best avoided and that's really all I am going to say about those. If you have to run Group 1, then the Weber setup that subsequently replaced the Solex, looks a good bet, as the power curve Fig. 4-8 shows. Apart from the factory-available downdraft Webers, manifolds are available to adapt sidedraft Webers or Dellortos to most installations. When Webers and Dellortos are mentioned in the same breath, the almost inevitable question comes up as to which is better. The Dellorto carburettors appear to have finer atomization of the fuel and since we are in a situation where fuel dropout can easily occur due to the large port, the Dellortos would appear to be better. My preference is for Dellortos, and many of the quickest, normally aspirated engines are so equipped. Dellorto carburettors are now readily available in U.S.A. The company you need to get in touch with is 'Claudes Buggies' in Farmersville California. In Australia, try Lynx. In England, Contact Developments is the company to deal with. You can find Weber distributors in every major city, but if you should have trouble, try contacting B.A.P. or Red Line in U.S.A. In Australia, Warnerford would be a good one to contact, and in England, try Weber U.K. Carburettors of Sunbury-on-Thames.

HOW BIG?

When you are shopping for sidedraft carburettors, just how big should you go? Should you use 40s, 42s, or 45s or even bigger. A simple rule helps here. Although not 100 percent accurate it will give you a guide. If you expect your engine *realistically* to produce over 155 bhp, figure on 45s. If you anticipate less than 155, figure on 42s or 40s. Forty-millimeter throats are good up to about 150 bhp and 42s to about 155. Over that you will need the bigger carburettors. Should you be overly ambitious and decide to go for 45s when not needed you will actually lose bhp. Interestingly, a 42 DCOE Weber with 32 mm chokes will flow more air than a 45 DCOE with 34 chokes. And if 35 mm chokes are used the 45 starts to out-flow the 42. The point I am trying to make is, don't be over-ambitious. When building a street engine, if anything, err on the small side for better drivability. If you intend to build an all-out race engine, 45 mm carburettors are too small. You'll need at least 48 mm carbs, and here Dellortos prove to be the favourite.

MIKUNIES

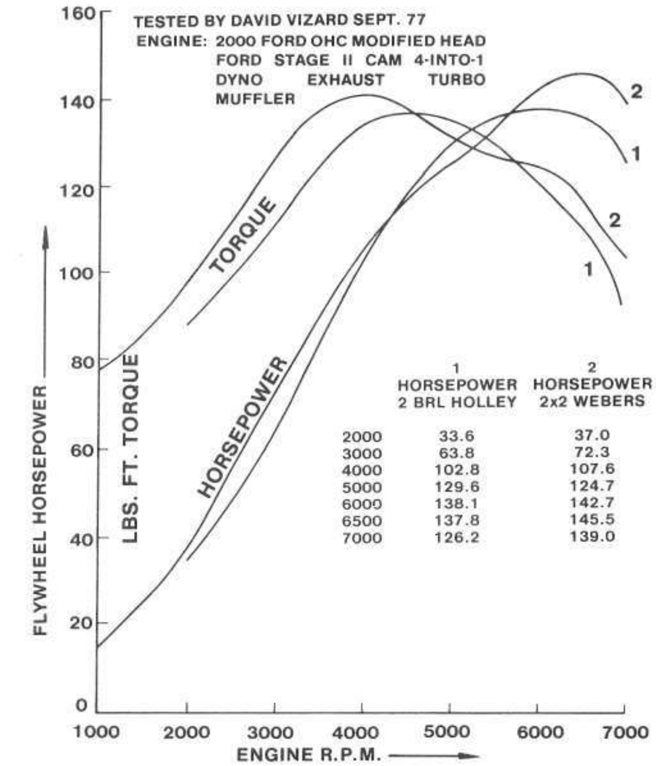
Another carburettor well worth considering for use on an independent runner system (I.R.) is the Japanese Mikuni carburettor. This has excellent flow traits and for out-and-out racing yields good power.

JETTING

If you think you can buy a pre-jetted carburettor for your application, you

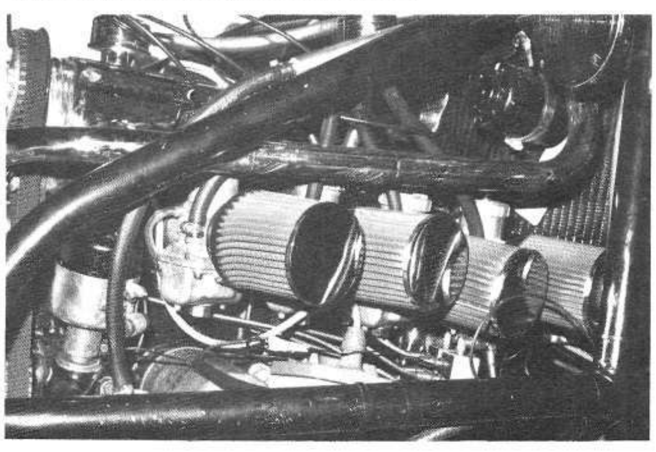
4-32. Four 40 mm Mikuni carbs grace this Ford powered off-road car. They proved capable of good power output and are insensitive to the bumping and jarring experienced in off-road conditions.

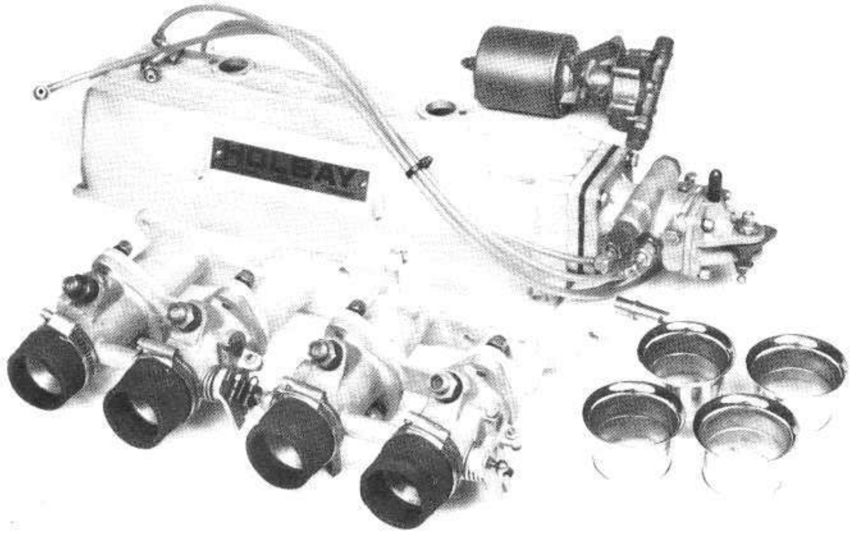
FIG 4-9



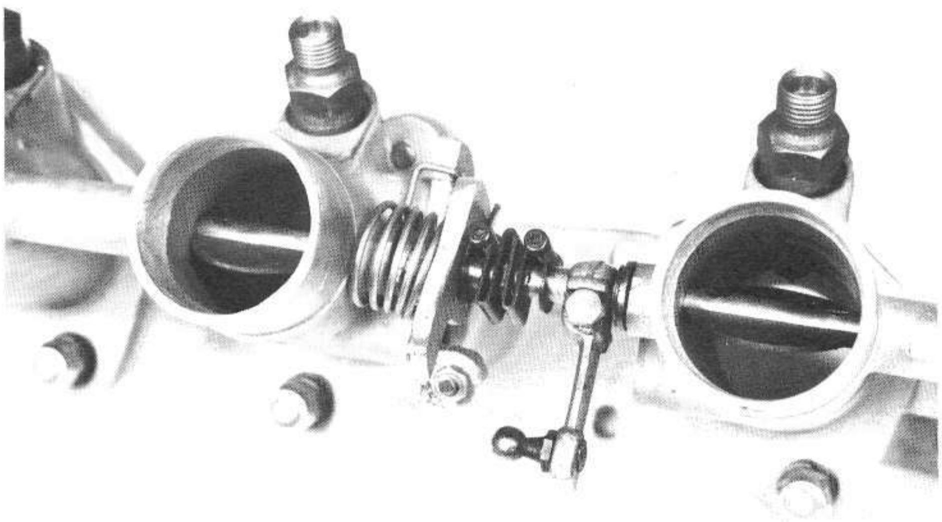
	1 HORSEPOWER 2 BRL HOLLEY	2 HORSEPOWER 2x2 WEBERS
2000	33.6	37.0
3000	63.8	72.3
4000	102.8	107.6
5000	129.6	124.7
6000	138.1	142.7
6500	137.8	145.5
7000	126.2	139.0

4-33. Filter requirements for Mikuni carbs are easily satisfied by these high flow K & N clip-on filters.

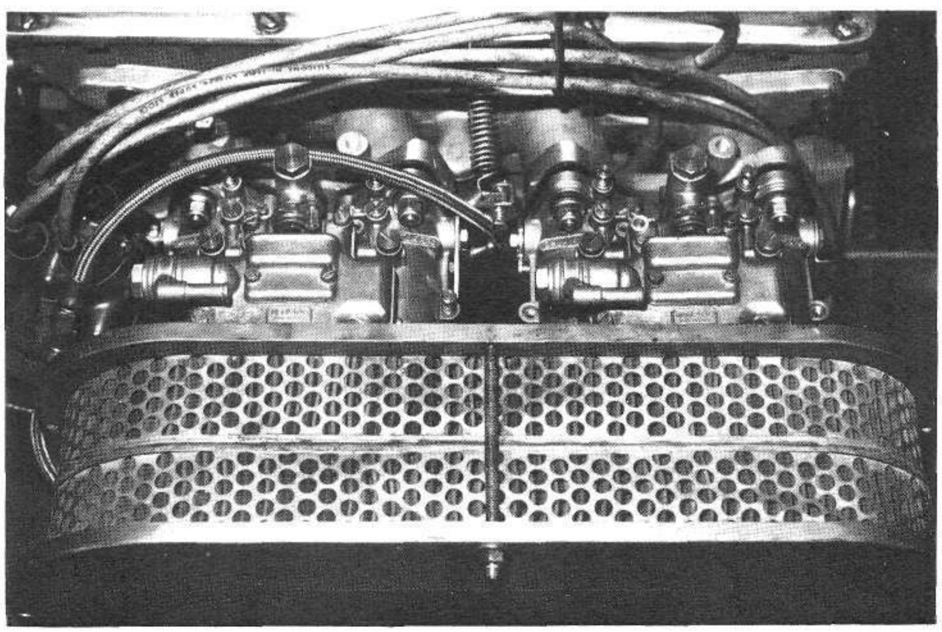




4-34. This fuel injection system is offered by Holbay Engineering. It is intended for use with petrol/gas as opposed to the alcohol systems offered by Esslinger Engineering, Racer Walsh and others.



4-35. Many high performance fuel injection systems employ slide plate throttles but the Holbay setup uses butterflies. This would appear to be adequate for most installations.



4-36: When dusty conditions prevail, large air filter must be used. Shown here is the large element on Dwaine Esslinger's normally aspirated sand dragster.

are not being realistic. In all the years I have been building high-performance engines, I have only had *one set of carburettors jetted optimally at the outset.* Typically, a set of carburettors jetted by a carburettor manufacturer or supplier will be between 5 and 10 bhp down from what they are ideally capable of. On these engines we are typically expecting a ten, sometimes a 15 bhp increase through precise jetting. The golden rule is, having purchased your carburettors, *always* use a chassis dynamometer for correct carburettor calibration.

CARBURETTOR LINKAGES

This is an area where many home mechanics fall prey to problems. If you buy a carburettor kit, be sure it contains linkages compatible with your particular vehicle. Some carburettor linkage kits cannot be used with automatic transmissions inasmuch as they have no provision for a kickdown linkage. If you intend using a four-barrel setup on a Pinto and you want to retain the kickdown linkage I suggest you use the Spearco setup. If you intend using a two-barrel Holley on an automatic and maintain the kickdown, take a look at the nearby photo and see what I cobbled up through modifying the original brackets. The only other item necessary was a piece of light gauge tube from which to fabricate the kickdown rod.

As far as making a linkage for your carburettor installation, I offer a couple of suggestions. Racer Walsh has some linkages available for sidedraft Webers, be they single or twin, and for the two- and four-barrel Holleys. For unconventional conversions, Lynx produces a carburettor adaptor kit containing all manner of gadgets, rods, ball joints and fittings. I have used this kit on a couple of occasions not only for Ford SOHC but other vehicles as well, and it has proved to be very useful. In England most of the carburettor kit manufacturers provide some linkage parts.

Magard outlets and Chris Montague Carbarettor Co are a good source of supply for excellent twin side draft linkages, if your carb kit is not linkage equipped. However, not all do, so do investigate before you buy a carburettor kit which may seem to have a super knockdown price.

Exhaust Manifold & Exhaust System

In spite of much research, exhaust systems are still a combination of science and trial and error. There is no doubt a great deal is still to be learned and on top of that, there are a few things to be *unlearned*. Getting the best power curve possible from a normally aspirated engine is a question of juggling with pressures. On a silenced, street-driven engine, the lowest-back pressure possible over the entire rpm range used is our target. On an open-exhaust race engine we attemped to use shock wave and an inertially produced pressure drop to draw the exhaust from the combustion chamber. This effect is especially beneficial on engines which have large overlap cams. Here is how it works: if a pressure below atmospheric can be created in the exhaust port during the overlap period, the contents of the combustion chamber will tend to flow from the chamber into the exhaust port. By the same token, the contents of the intake will tend to flow from the port into the combustion chamber. When this happens, we score in two ways. Under normal circumstances the piston cannot push exhaust from the combustion chamber. As a result, this residual charge of exhaust dilutes the incoming charge. If the chamber is scavenged with the incoming air/fuel mixture, not only has the exhaust charge been eliminated, but the combustion chamber is filled with a fresh charge. This begins to take on more significance when some demonstrative numbers are thrown in.

Let us assume the engine is a 2000cc unit with a 14:1 compression ratio with a chamber volume 38cc. If each cylinder pulls in another 38cc of *fresh* charge, the power is going to improve in about the same manner as increasing engine capacity by 152cc (4 x 38). If exhaust tuning is really dialled in and the cylinder scavenging rpm coincides with intake shock wave tuning, yet more increases in power are possible and one more factor of significance starts to show up. If the scavenging action is great enough, a fresh charge will not only be pulled into the chamber but also through into the exhaust port. If it gets this far, it's wasted. Under these circumstances it may be necessary to *lower* the CR so that the increased charge volume has room to stay in the chamber. Under these conditions *change retention always gets preference over the geometric CR.*. Although the geometric CR may come down, the effective CR is staying largely unchanged because of increased cylinder filling.

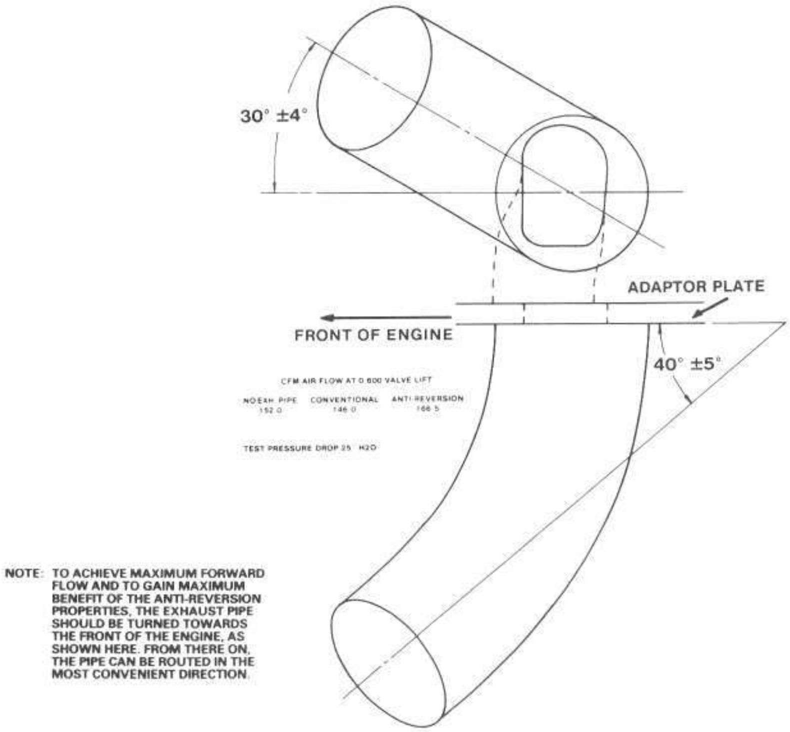
With an exhaust system for the street, shock wave and inertial extraction become significantly more difficult, due to the presence of the muffler. The biggest enemy power has in a silenced system is back-pressure. Some experimenters have claimed that in certain instances, a little back-pressure in a street system has helped power. My contention here is that the experiments were not closely controlled enough and the small back-pressure was in fact compensating for some maladjustment in mixture or timing. In my experience, when *everything* that I can think of has been optimized, the lowest back-pressure gives the best power output.

An approach that seems to be taking increasing importance, is the designing of an engine's entire air through-flow system having the minimum back flow possible. This is a relatively new concept for most high performance engine builders. The basic concept is very simple.

Deciding on exhaust pipe dimentions is never an easy task, and it is further complicated by the fact that there are essentially two different configurations to consider. Systems can be four into one (4/1), or four into two into one (4/2/1). The 4/2/1 system seems popular, but I have not seen as much power with this type as with the 4/1 type. Another aspect of at least equal importance for a street-driven machine is that a 4/1 system seems better at ironing out undersirable characteristics of relatively long-timing road cams. To give an example, a rally engine equipped with a 300 ° cam would pull right down to 1000 rpm with a long 4/1 system, but with a 4/2/1 system it would positively refuse to take full throttle even as high as 2800-3000 rpm. Moreover, it imposes a power loss of 4 to 15 bhp between 5000 and 7000 rpm. The only point against four-into-one systems is that they generally do not produce the mid-range torque that a 4/2/1 system can. This is the principal reason that many tubular factory manifolds are of this configuration and the fact the 4-1 system sometimes cannot be installed due to space limitations. Another point is that a manufacturer is hardly likely to install an exhaust system yeilding high-rev power if the engine is redlined at, say, 5000 rpm.

The biggest drawback to long-period, high-overlap cams, is the flow reversals which occur at low rpm. These flow reversals diminish as rpm rises, because increased gas speeds have more momentum and less time to reverse. If ports and valves are designed to flow well in the correct direction but badly in the reverse direction, undesirable low-speed characteristics of these cams are *substantially* reduced. Positive and negative shock waves travelling within the exhaust system present a similar situation. Also current system designs as shown in this book, which backflow badly, also tend to allow easier passage to a useful negative shockwave than for a power-reducing positive shockwave. Exhaust systems which a good extraction effect at a designated rpm will have a power reduction effect at other speeds. This is of little consequence if it occurs outside the normal operating range of the engine. But in many instances, a high-performance street engine, for example, a power loss anywhere in the rev range is to be considered a bad thing. Under such conditions, exhaust systems with bad back flow are a definite asset. Although the term "back flow" is easily understood, it is somewhat unweildy. Systems which are deliberately designed to have bad back flow are becoming known as "anti-reversion" systems so this is how they will be referred to from now on. In England *CV* (controlled vortex) systems operate on much the same principle and can be considered as performing basically the same function.

FIG 5-1
CORRECT MISMATCH AND EXHAUST PIPE LEAD OUT DIMENSION FOR MAX FORWARD FLOW AND 'ANTI-REVERSION' PROPERTIES.



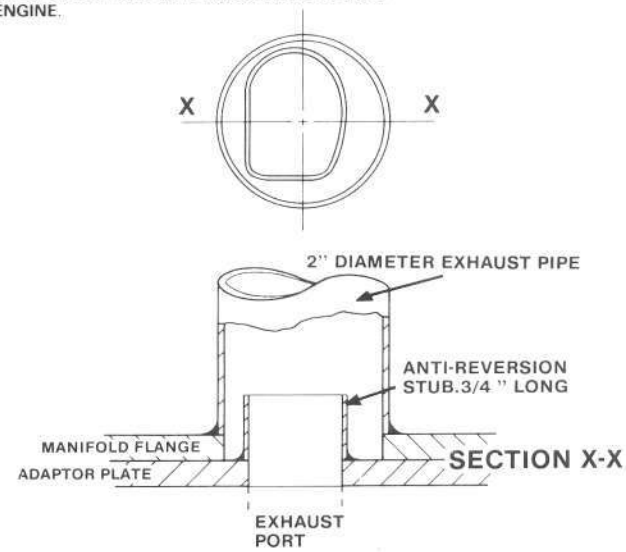
HEAD/MANIFOLD INTERFACE

With the advent of anti-reversion or controlled vortex exhaust systems, the head-to-manifold interface takes on much importance. Indeed, I am going to take a long-standing internal combustion engine rule and throw it out of the window. How many times have you heard or read that it is very important to match the exhaust manifold to the exhaust port in the head?

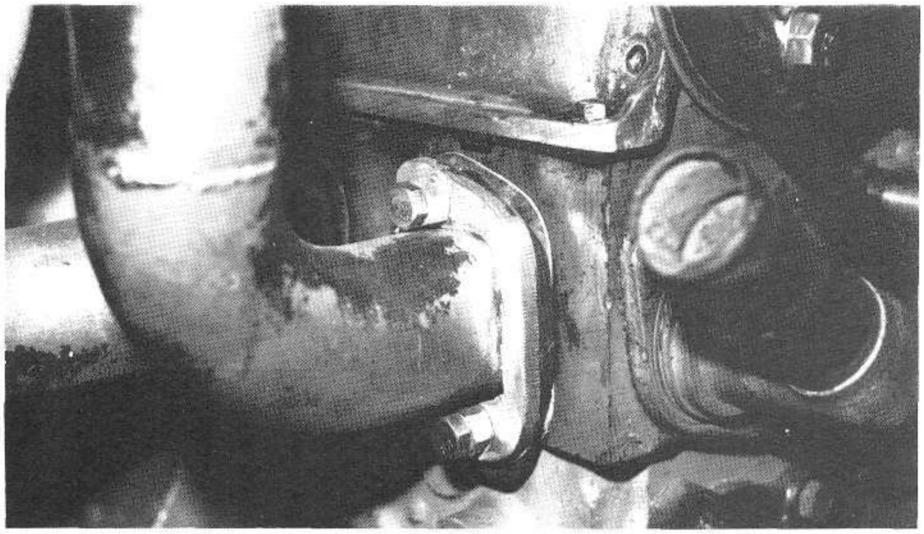
Plenty of times, no doubt but the *really* important point is to have the correct mismatch. On the Ford OHC engine, substantial gains in both flow and power have been achieved by a *radical mismatch* of the port in the head and the manifold. (See Fig. 5-1.) A second aspect is that a highly developed racing engine generally needs large diameter exhaust pipes. Even in street form, this engine can respond to pipe diameters larger than expected. The large diameter pipe (5-1) in conjunction with the surrounding manifold face at the head, provides a highly unstreamlined path for exhaust to go *in* the exhaust port. As far as forward flow is concerned, the mismatch actually increases proper flow, thus bringing about two very real advantages: 1) better forward flow and 2) better anti-reversion properties.

If you have a cylinder head in front of you and some two-inch exhaust pipe to hand, it won't take long to realize that using a two-inch bore is not as easy as it looks. There isn't a large enough expanse of flat face around the exhaust port or the head to position such a pipe. Nor is there room for many of the studs in the existing positions. The way around this is to make an adaptor plate about 5/16-3/8 inch thick. This adaptor is secured to the manifold face by means of countersunk socket fasteners screwing into the original manifold stud holes. New manifold securing holes, either for studs or screws, can be positioned in the adaptor plate. The seal between the adaptor and the manifold needs to be gas-tight. The simplest way of achieving this is to use *Silicone* sealer between the plate and the head. The plate becomes a permanent fixture. After the plate has been secured, the exhaust holes in the plate must be cut to exactly match the exhaust ports. As an anti-reversion feature, the stubpipe as shown in Fig. 5-2. can be added.

Fig 5-2
DETAILS OF MANIFOLD ADAPTOR PLATE AND ANTI REVERSION STUB ON 2000 ENGINE.



MANIFOLDS AND STANDARD CAMS



5-1. Here's the 2" diameter exhaust pipe used on the Vizard/Esslinger 2000 cc engine. Note the plate between the manifold flange and head. This plate . . .

5-2. . . . . houses the anti-reversion stubs described in the text and drawings.

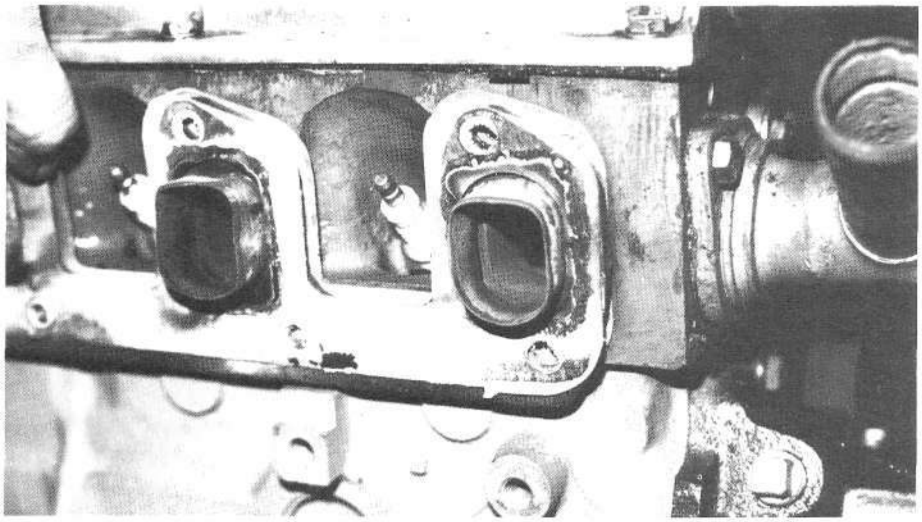
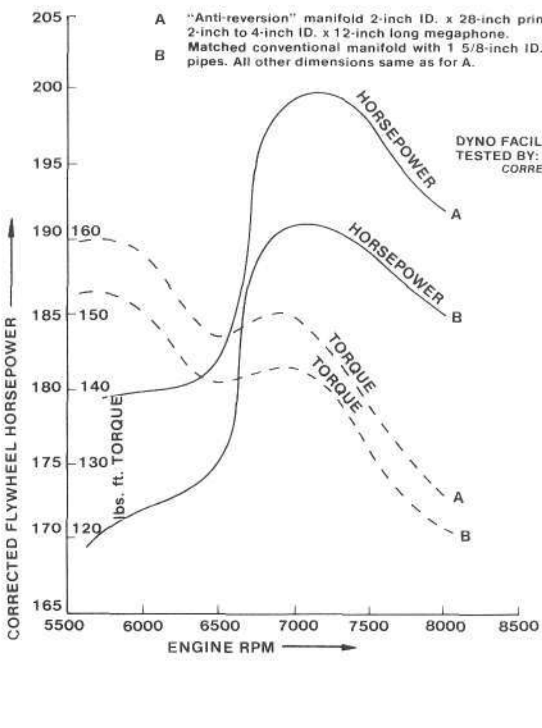


FIG 5-3



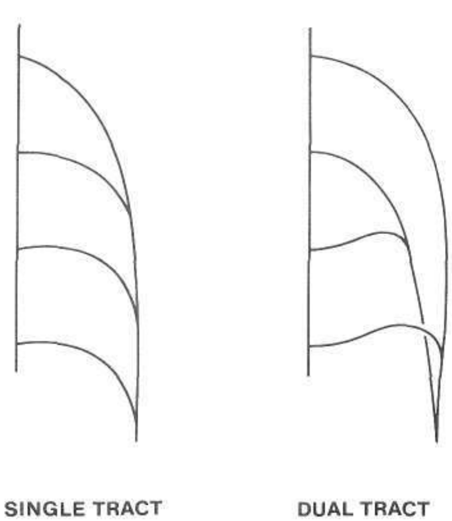
Many performances enthusiasts, especially those operating on a low budget, will be tempted to install a custom exhaust system yet retain the standard cam. This will also be the case for Formula Ford and Sports 2000 competitors, both are racing classes requiring standard cam profiles. Under these conditions what sort of power increase can be expected?

Let's start by discussing the two types of cast iron exhaust manifolds: single-tract and dual-tract. A look at Fig. 5-4 shows the general routing of internal passages. The dual-tract manifold has a two down divider which is found near the exhaust pipe flange. Dual tract manifolds are used on virtually all performance two-litre engines; single-tract manifolds are used on 1.3 and 1.6-litre engines. The only exception to this is the 1600 GT engine, which we will deal with later.

Installing a dual-tract manifold will, due to the longer separation of the exhaust gases, produce a little more power when it is used as a replacement for a single-tract unit. Fig. 5-5 shows typical results of just such a swap on a 1.6-litre engine which in all other respects has been brought up to GT specification.

On some engines, such as the 1600 Cortina GT, the factory installs a tubular 4/2/1 exhaust manifold. Although less

fig 5-4 COMPARISON OF EXHAUST ROUTING ON SINGLE AND DUAL TRACT MANIFOLDS



SINGLE TRACT DUAL TRACT

than the ultimate, this manifold gives more power over both cast iron items, as the power curve in Fig. 5-6 shows. This same graph also shows that with the standard cam, an aftermarket 4/2/1 high-performance manifold is only slightly superior *in this instance*.

Moving on to the 4/1 system, we find that the same rules apply. The 4/1 system tuned for a standard cam produces the same sort of change on the power curve as mentioned before, namely a little less mid-range torque than a 4/2/1 but substantially more top-end power. This is especially noticeable from 5000 rpm upward (See Fig. 5-7). At 6000 rpm we see as much as 20 bhp more than that yielded by the cast iron manifold. This sort of manifold is to be preferred if outright performance with the standard cam is required. Also, this is essentially the same reason why this type of manifold is used for formula racing.

You will note that none of the manifolds tested prior to 1978 were anti-reversion types. Work on the manifold for the 2000 did not start until late 1977 and most of the testing on these was performed by Duane Esslinger at Esslinger Engineering. I have little doubt that all manifolds will, to a greater or lesser degree, perform better when modified to anti-reversion designs.

fig 5-5 COMPARISON OF SINGLE AND DUAL TRACT CAST IRON ORIGINAL EQUIPMENT MANIFOLDS

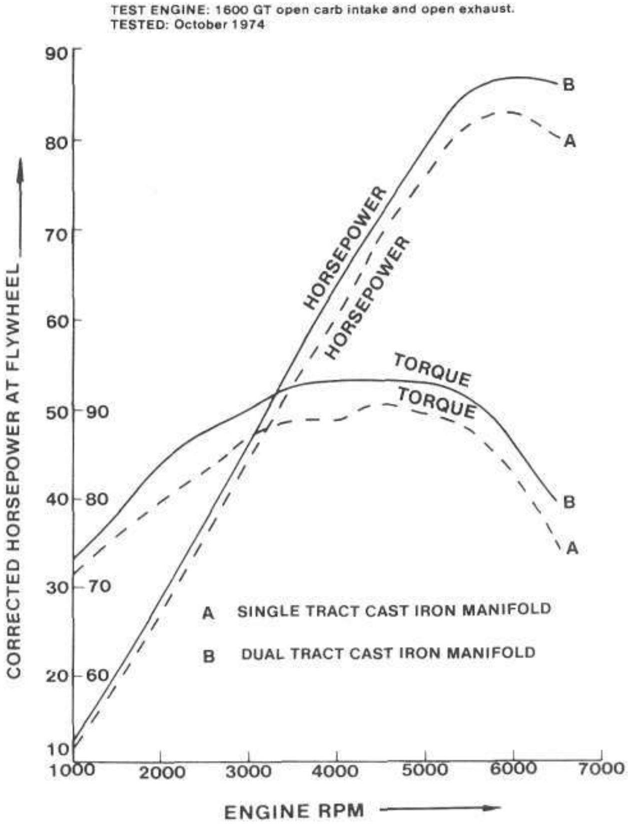
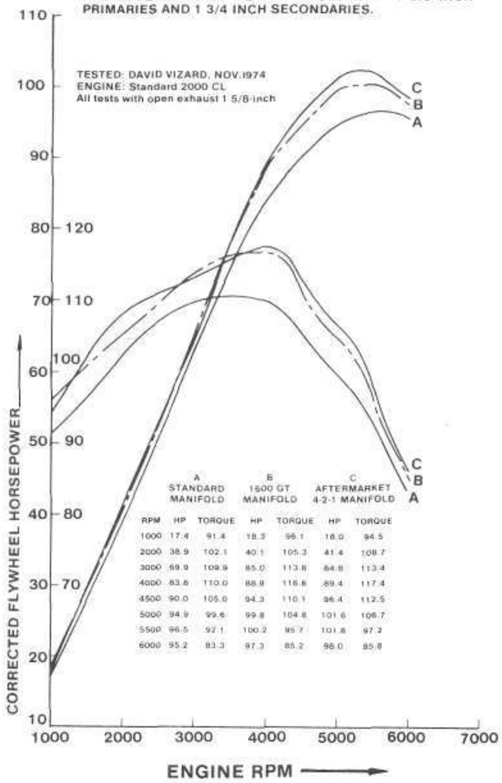
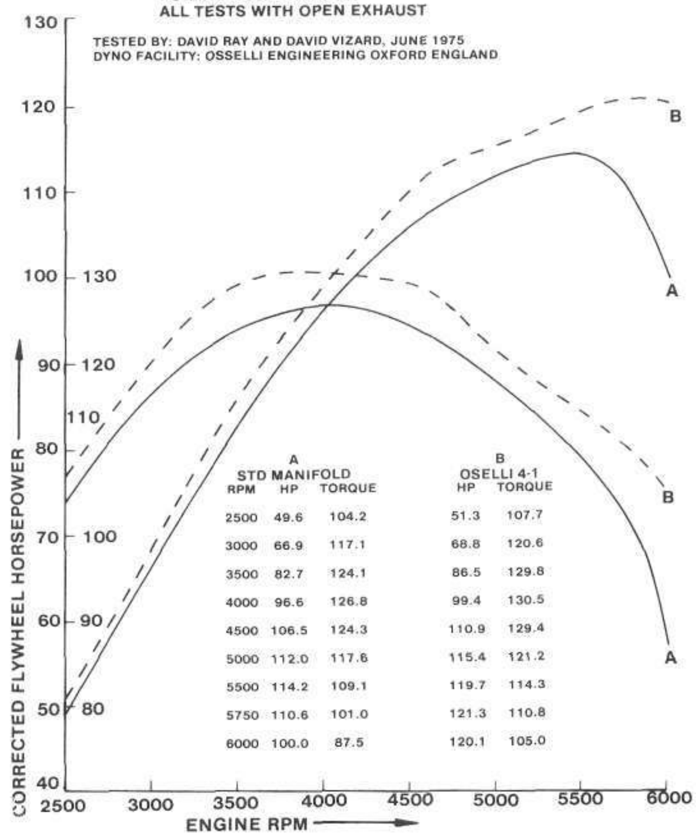


FIG 5-6 COMPARISON OF STANDARD CAST-IRON, DUAL-TRACT EXHAUST MANIFOLD WITH FACTORY 1600 GT MANIFOLD AND AFTER MARKET 4-2-1 MANIFOLD WITH 1 5/8 INCH PRIMARIES AND 1 3/4 INCH SECONDARIES.

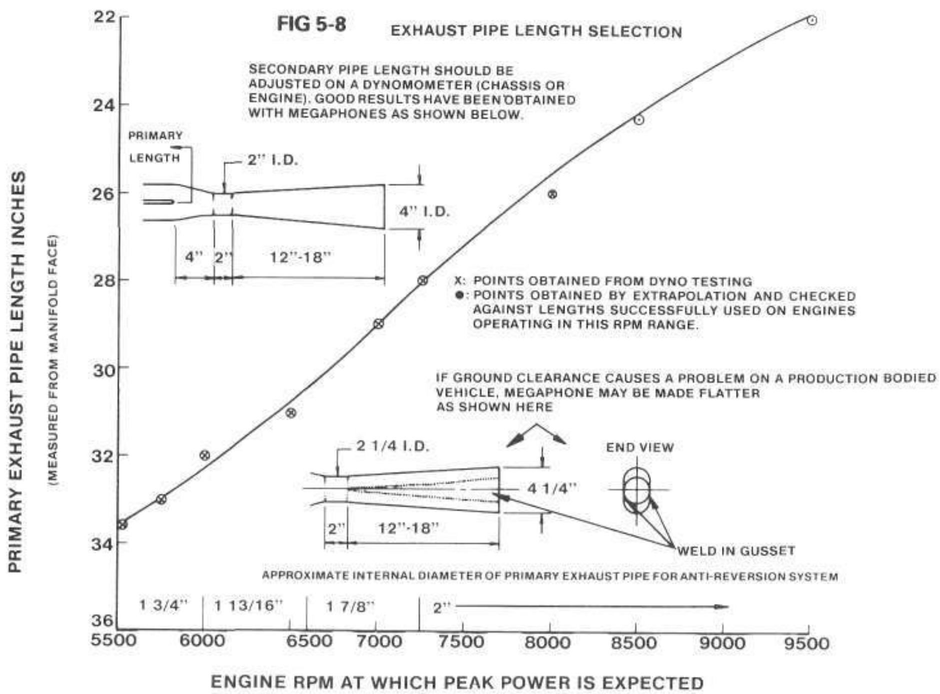


	A STANDARD MANIFOLD		B 1600 GT MANIFOLD		C AFTERMARKET 4-2-1 MANIFOLD	
RPM	HP	TORQUE	HP	TORQUE	HP	TORQUE
1000	17.4	91.4	18.3	96.1	18.0	94.5
2000	38.9	102.1	40.1	105.3	41.4	108.7
3000	69.9	109.9	65.0	113.8	64.8	113.4
4000	83.8	110.0	88.8	116.8	89.4	117.4
4500	90.0	105.0	94.3	110.1	96.4	112.5
5000	94.9	99.6	99.8	104.8	101.6	106.7
5500	96.5	92.1	100.2	95.7	101.8	97.2
6000	95.2	83.3	97.3	85.2	98.0	85.8

FIG 5-7 A COMPARISON OF STANDARD DUAL-TRACT EXHAUST MANIFOLD AND OSELLI 4 INTO 1 MANIFOLD USED ON FORMULA SUPER FORD. ALL TESTS WITH OPEN EXHAUST



	A STD MANIFOLD		B OSELLI 4-1	
RPM	HP	TORQUE	HP	TORQUE
2500	49.6	104.2	51.3	107.7
3000	66.9	117.1	68.8	120.6
3500	82.7	124.1	86.5	129.8
4000	96.6	126.8	99.4	130.5
4500	106.5	124.3	110.9	129.4
5000	112.0	117.6	115.4	121.2
5500	114.2	109.1	119.7	114.3
5750	110.6	101.0	121.3	110.8
6000	100.0	87.5	120.1	105.0



5-3. An unusual system for the 2000 cc engine is made by Hedman headers. It is a 4 into 2 into 1 system that almost becomes a 4 into 1 due to the very short secondary length. Though it doesn't incorporate all the advanced features mentioned in the text, it does work well for a cheap off-the-shelf-unit.

5-4. When you buy an exhaust manifold (header) check all the welds. On the item shown here, it was necessary for me to have ALL the flange-to-tube joints rewelded. What you see here is how they should be.

EXHAUST MANIFOLD DIMENSIONS

When it comes time for selecting an exhaust manifold, remember, the exhaust system dimensions that your engine requires depend on many variables. Essentially, you must select the exhaust manifolds to complement the basic shape of the cam's power curve. Take a look at Fig. 5-8; this relates pipe lengths and diameters using anti-reversion designs. If a conventional matched manifold is used, reduce the diameter by about 15 percent. I hasten to point out that the figures given in Fig. 5-8 are approximate figures only. I have no control over the specification of your engine; you may find lengths and diameters either side of what I suggest to be superior depending on tune. Only your own experimentation can establish this for sure.

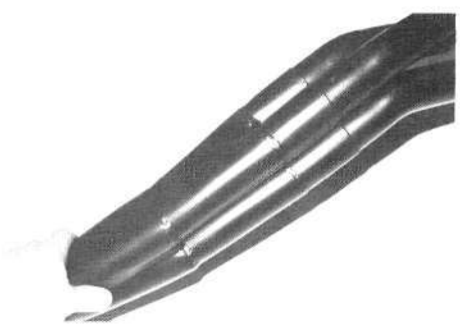
SILENCING

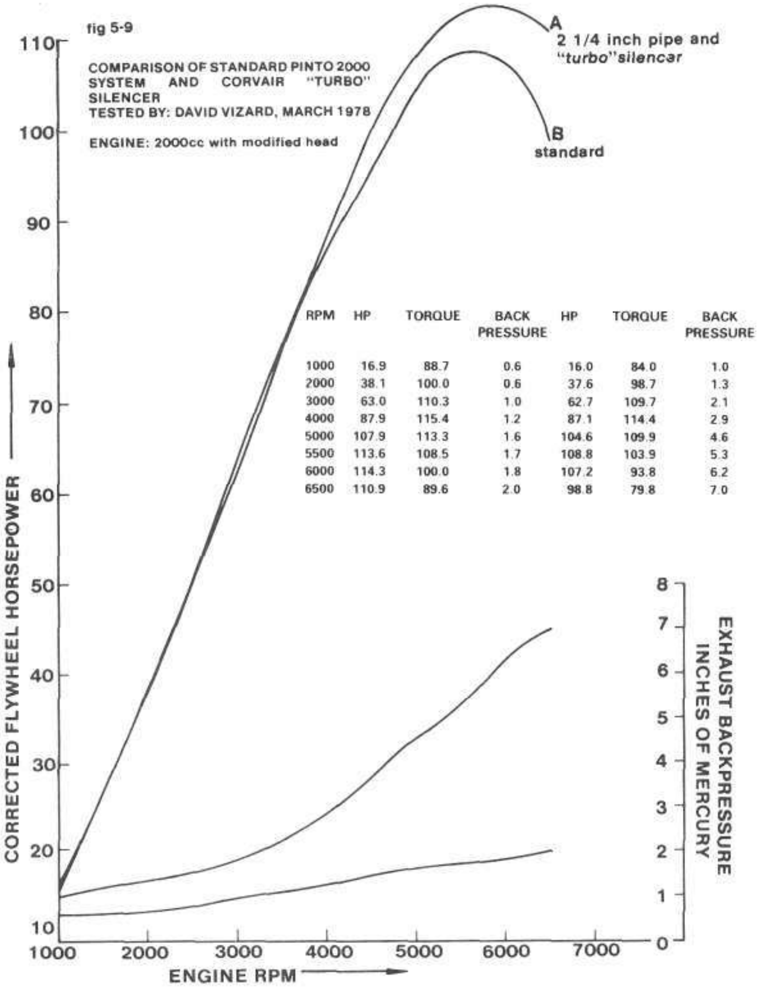
In a silenced, high-performance exhaust system the main enemy of engine power is *backpressure*. All those super-trick pipe-lengths will produce a power curve far from ideal, unless the backpressure is between minuscule and zero. Based on simple theoretical considerations, 3 ½ psi backpressure, which is not at all unusual, can reduce power output at 6000 rpm by 4.4 bhp on a 1300, 5.4 on a 1600 and 6.7 on a two-litre engine.

MEASURING BACKPRESSURE

Before ripping off your car's silencing system, it would be helpful to know how much backpressure it has. The easiest way to do this is to use a large scale fuel

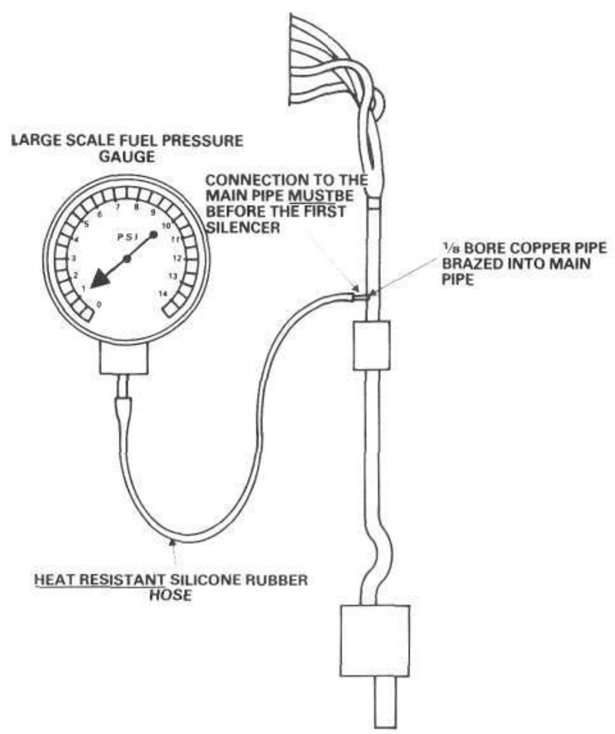
5-5. Some exhaust manifolds use slip joints, supposedly to ease installation. Though they may help installation they may also take a while to seal up. Quicker sealing can usually be achieved by the use of silicone sealer on the pipes BEFORE assembly. This works okay if exhaust pipe temperature is taken up slowly.





RPM	HP	TORQUE	BACK PRESSURE	HP	TORQUE	BACK PRESSURE
1000	16.9	88.7	0.6	16.0	84.0	1.0
2000	38.1	100.0	0.6	37.6	98.7	1.3
3000	63.0	110.3	1.0	62.7	109.7	2.1
4000	87.9	115.4	1.2	87.1	114.4	2.9
5000	107.9	113.3	1.6	104.6	109.9	4.6
5500	113.6	108.5	1.7	108.8	103.9	5.3
6000	114.3	100.0	1.8	107.2	93.8	6.2
6500	110.9	89.6	2.0	98.8	79.8	7.0

FIG 5-10 MEASURING BACK PRESSURE IN THE EXHAUST SYSTEM



THIS DIAGRAM SHOWS A TYPICAL LAYOUT UTILIZING A CORVAIR 'TURBO'SILENCER

5-6. A nicely finished system can enhance the appearance of your engine. This is a shot of one of my Pintos with the ignition wires removed for a better shot.

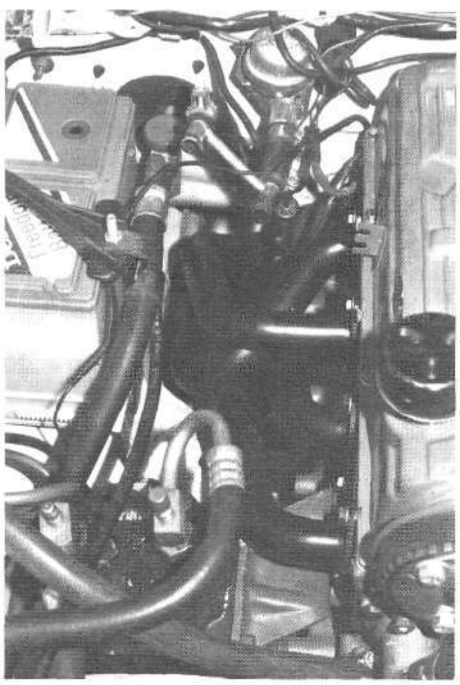
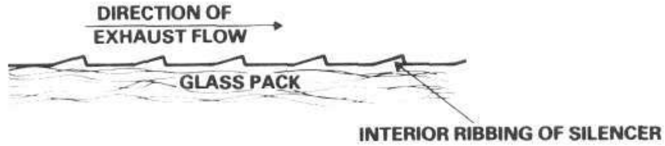


FIG 5-11

NOTE: MOUNTING THIS TYPE OF SILENCER THIS WAY IS BACKWARD TO THE NORMAL WAY BUT FOR AN APPLICATION WHERE A SECOND BIGGER FREE FLOW SILENCER IS USED IT PRODUCES LOWER BACK PRESSURE FIGURES WITHOUT A NOTICEABLE NOISE PENALTY



NOTE: AIR FLOW IN THE DIRECTION SHOWN IS TYPICALLY 90% GREATER THAN AIR FLOW IN THE REVERSE DIRECTION.

pressure gauge connected to the exhaust system as shown in Fig. 5-10. Measurements are best taken in third gear in vehicles which have manual transmissions or second gear if it has an automatic transmission. The object of the exercise is to reduce backpressure to as low as possible.

PERFORMANCE EXHAUST SYSTEMS

There are numerous so-called performance silencers on the market which do little other than make subdued sporty noises. Our aim is to produce maximum power consistent with

Fig. 5-11B

COMPARISON OF 'TURBO' MUFFLERS

BAND	PART NO.	BORE	FLOW AT 25" H_20
THRUSH	Turbo 500	2½	252.8 CFM
THRUSH	Turbo 501	2¼	207.5 CFM
I.P.C.	H.P. 250	2½	279 CFM
* SUPREME	Super 'C' 111224	2½	355 CFM
CYCLONE(American exhaust Industries)	40122	2½	180 CFM
CASSLER	98 MH	2¼	243 CFM
CASSLER	90	2½	267 CFM
MAREMONT	Quiet Q	2½	260 CFM
GENERAL MOTORS	3869877	2½	293 CFM
MIDAS	7051	2½	279 CFM
WALKER	RED LINE Z	2½	275 CFM
* ARVIN INDUSTRIES	TURBO	2½	332 CFM
GORLICK	TURBO	2½	261 CFM
CYCLONE	TURBO SONIC	2¼	335 CFM
CYCLONE	TURBO SONIC	2½	405 CFM

*** Mufflers marked thus * are actually the same but sold under a different trade name. Difference in flow probably represents normal spread due to manufacturing tolerances.**

minimum noise. A combination that I have found works very well when used in conjunction with a standard manifold or high-performance tubular manifold, is a 2-2¼ inch I.D. pipe from the manifold back to a short-large-bore,straight-through silencer. Before you mount the silencer take a look inside. If it's the kind that has louvered projections in the exhuast stream, mount it so exhaust flow is as shown in Fig. 5-11. From this silencer, run the exhaust to a turbo silencer and then out the back of the car. Such a system will be quiet and efficient. If you own an R.S. 2000 you are lucky, Competition Silencers produces a quiet rally system, that gives *NO* power loss on engines up to 160 B.H.P.

If you intend putting together a low-backpressure system along these lines, a few additional hints will be of help. If the exhaust pipe is to make tight bends (which it often does in the vicinity of the axle) use a minimum of 2¼ inch I.D. pipe or better yet 2½ inch I.D. pipes to reduce restriction at these bends.

There are many imitation Corvair turbo silencers on the market. I have tested almost every brand of these, most possess a low backpressure, although some are a little noisier than others. As of 1982 the highest flowing turbo by a large margin, is the top line item produced by American Exhaust Industries (Cyclone). Their 2¼ turbo is capable of outflowing the opposition's best 2½. Added to this super high flow efficiency is a noise level that equals the quietest turbo available. For the sporty car, a desirable feature of the Cyclone turbo is its customized sound signature (C.S.S.). The exhaust note produced is a deep, mellow powerful-sound. Just the ticket!

Ignition Systems

Some engines are relatively insensitive to ignition quality, while others lose power rapidly as spark quality declines. The Ford 2000 SOHC engine falls into the second category. It takes little spark degradation to produce a measurable drop in power and economy. If the engine is to stay in top tune, the quality of the spark must be maintained. With this in mind, let's start our look at the ignition system right at the spark plug.

SPARK PLUGS

The first consideration is that the spark plugs be clean. Even in the case of a standard engine, I advise cleaning the plugs at intervals no longer than 3000-3500 miles. To clean the plugs, have them lightly sandblasted, then blow them out with compressed air. Next wash them in a solvent such as acetone. This ensures that all grit, plus any fine particles of metal which may be lurking inside the insulator area, are washed out. If this isn't done, it is possible for the plug to start tracking across the insulator, thus costing you a spark. Once the plugs have been sandblasted, take a fine needle file and file the end of the centre electrode so it has a sharp edge. Then file the side electrode so any rounded edges disappear. This is done because a spark can jump easier between sharp edges than it can between rounded edges or flat surfaces. Once the spark plug has been suitably dressed, set the gap on it. If you are using a standard coil ignition system, use the standard recommended gap of 0.025-inch (0.64 mm). When the plugs have done 10,000 miles, replace them; their useful life is over.

If you have a stock or near-stock engine that burns no oil, you can change to a cooler plug and pick up a slight increase in power. Typically the gain is around 2 bhp. Fig. 6-1 indicates the sort of plug to use. If the engine has a tendency to oil-foul its spark plugs, you may find that a cooler plug will cause the engine to oil up the plugs more rapidly, especially if slow, stop/start traffic driving is done.

If you have a modified engine, then you should figure on using the coolest running plug the engine will stand without misfiring due to fouling.

In the case of an electronic ignition system, the greater available voltage means that the engine can use a much colder plug without misfiring due to fouling. Again, for your modified motor, Fig. 6-1 gives you a good starting point on plug selection. Once you have mentally digested the contents of the charts, consider these few additional points: highly tuned engines for events of any distance will develop high plug temperatures very easily. As a result, the plug, if it cannot dissipate the heat fast enough, becomes the prime source of detonation, especially where a compression ratio of 14:1 or more is used. To avoid serious engine damage, use a suitably cool plug. However, if water injection is used, combustion temperatures are reduced. In such cases, overheating of the plug is not such a problem.

Fig. 6-1

% POWER INCREASE

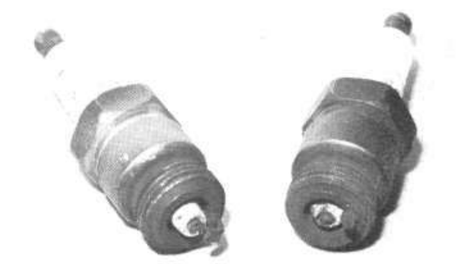
PLUG BRAND	0-10	10-30	30-60	60-80	80 → ON UP
Motorcraft	BRF 42 BF 42	BRF 32 BF 32	BF 22		
Autolite	45 35	45 34	33		
Champion	RF11Y F11Y	RF9Y F9Y	F7Y. F83Y	F62Y F81Y	F60Y, F79, F60R, F57R, F54R
A.C.	R84TS 84TS	R83TS 83TS			
Bosh	MA125T7 MA95R7 MA95T7 MA125TR7 MT55P	MA145T7 MA6BC MA145TR7 MT65P			
K.L.G.	GT5T MT75P	GT6T	MT85P		
N.G.K.	AP5FS APR5ES APR5FS11	AP4F	AP6FS		
Lodge	CTNY	HTNY HT18 HPTNY	2HTNY		

Hotter ← Plug Heat Range → Cooler

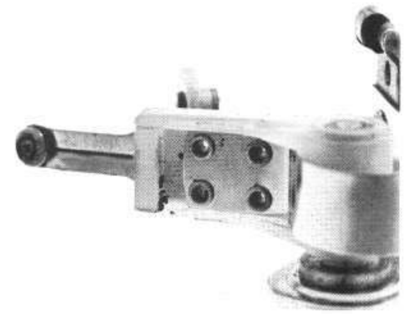
NOTE: This chart gives a good starting point for plug selection. At power increases up to 30% it will often be found beneficial to use one range cooler. If alcohol fuel is used, try a range hotter or two ranges hotter than expected on petrol (gasoline). If nitromethane or nitrous oxide is used, select as for petrol or one, maybe two ranges cooler. Better radio interference suppression especially on FM is achieved with resistor plugs. These are normally denoted by the inclusion of an R in the port number. For Autolite numbers an even first digit means a resistor plug; odd means a non-resistor plug.

Nitrous oxide, which we will talk about in detail in a later chapter, can present severe spark plug problems. The use of this oxidant can cause combustion temperatures to climb way above that seen by a conventional engine. For use with nitrous oxide, select a plug two or maybe three ranges cooler than Fig. 6-1 suggests. A plug incapable of dissipating heat, will last around 15 seconds while the nitrous oxide injection is in effect, after which point its electrodes will melt. Often it's the side electrodes which burn. If this is the case, a retracted side electrode racing plug must be used.

Alcohol-fuelled engines normally run cool because of the relatively low heating value of alcohol and its high cooling ability. As a result, plug requirements are not so critical. Indeed it is sometimes a good idea to use a plug a step or two hotter than indicated by the chart. However, if nitromethane is used as fuel, spark plug requirements favour the cooler end of the spectrum. The reason for this is that nitromethane is highly prone to detonation. If heavy nitro loads are used, the nitro- burning engine can destroy itself quickly when excessive spark plug temperatures lead to detonation.



6-1. Choosing the right heat range of plugs is important. The plug on the left was a few grades too hot for the application. As a result, the centre electrode shows signs of burning and the porcelain is chipped. The plug on the right which was about standard heat range, has been used in a nitrous oxide injected engine. Fifteen seconds of N_20 injection annihilated both electrodes. To avoid this, a very cold running plug is needed. This should be used with a good electronic ignition to combat fouling under light load conditions.



6-1a. Contact breaker points pitted as these shown here can reduce power output by 5-10 horse power.

SPARK PLUG CABLES

If you have an engine equipped with carbon string plug cables, either dump them or be prepared to replace them every year. This type of lead has a high resistance when new, and as it ages this resistance increases further. For a competition engine where radio interference is not a prime factor, copper-core or monel wire leads are a much better bet. Do not attempt to use any ignition system without some electrical resistance (in the form of radio interference suppression) between the coil and the plug. If no resistance is used, you will actually decrease the energy of the spark. The near-ideal resistance for the hottest spark of the longest duration is usually around 6000 ohms. This can be attained by installing inline radio suppression resistors in the plug leads and then using a plain, unsuppressed cap on the plug. Good sources of performance leads are N.G.K. MOROSO, BORG-WARNER AND SPARKRITE.

COILS

My own dynamometer testing has not shown a clear-cut advantage to using a high output sports coil, as opposed to the standard item. This is not to say sports coils don't work but it does suggest that a power increase from this source is too small to be effectively separated from experimental error. Tests on two engines, one near-stock, the other an extensively modified road engine, indicate that a high-output coil in conjunction with a standard contact breaker system may be worth about one bhp between 6000 and 7000 rpm. This gain, incidentally, was achieved in conjunction with a .030-inch plug gap instead of .025. Again, the increase is very small. But, as many of you will appreciate, in classes where the engine modifications are strictly limited, such as Formula Super Ford or Sports 2000, one bhp may make the difference between winning and being an also-ran.

Obviously an increase in spark intensity is a move in the right direction, but it could be that the energy from a high-output coil is still insufficient to deliver all that the engine requires. This tends to be substantiated by the fact that high-output electronic ignitions are clearly beneficial on these engines.

CONTACT BREAKERS

Testing has shown that incorrect or substandard contact breaker action has a distinct effect on power output. Three factors come into action here. First of all, the contact breakers need to be set to give the correct dwell angle. For those of you not familiar with the term, the dwell angle describes the time that the contact breakers are closed during their rotational cycle. During the time they are closed, the coil charges up its magnetic field. When the contact breakers open, the collapse of this magnetic field generates a high, reverse voltage and this is what fires the plug. If the contact breakers are not closed for a sufficient length of time, there may not be a fully developed magnetic field in the coil.

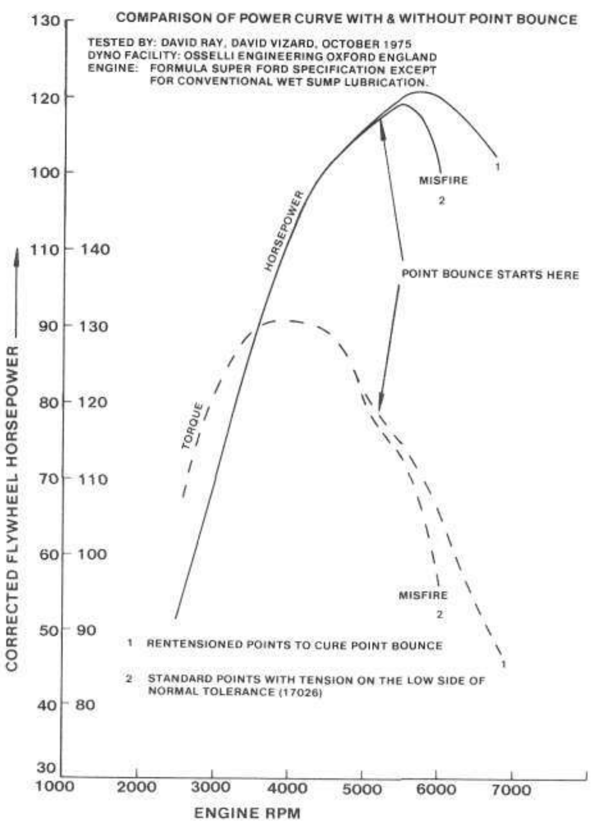
If an engine is to spend the majority of its time at high rpm, it is advisable to set dwell at the greatest permissible angle for a given distributor or maybe even five degrees extra. This may result in less than optimum ignition at low rpm. However, this is of little concern if peak power is the sole requirement. (See Fig. 6-2 for recommended dwell angles).

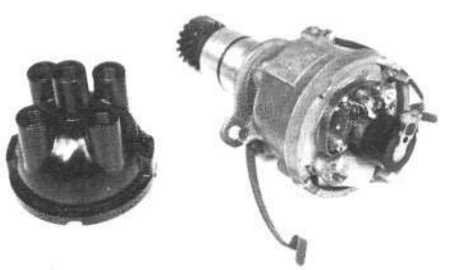
FIG. 6-2

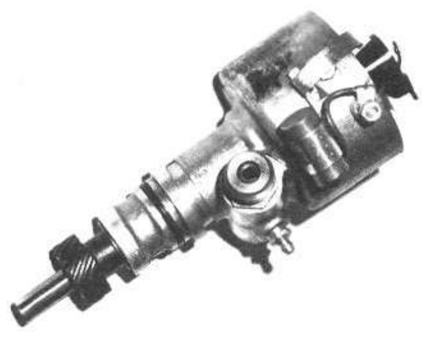
DWELL SETTINGS FOR HIGH RPM USE ONLY

DISTRIBUTOR	STANDARD DWELL ANGLE	HIGH RPM DWELL ANGLE
Motorcraft	48°-52°	52°-57°
Bosch	48°-52°	52°-57°

FIG 6-3







6-2. Holbay offer a Lucas competition distributor for Ford's S.O.H.C. engine. The points tension/cam dynamics combination will allow a claimed rpm as high as 9000 before point bounce occurs. The advance characteristics of this distributor are also tailored to suit a competition application.

6-3. If you need a mechanical tacho drive, this competition Bosch distributor is for you. Alternatively you . . .
6-4. . . . . . can use a drive off the end of the cam cover. or . . .
6-5. . . . .a drive taken from the distributor jack shaft through the position originally occupied by the fuel pump.

The second important factor we must deal with in a breaker-type ignition system is point bounce. Point bounce occurs when rpm has reached such a high level that the spring tension in the points is insufficient to control their motion. As a result, the closure of the points is not a clean and quick operation, rather the contacts bounce as they strike each other. This eats into the dwell time and reduces the spark intensity. The amount of power that can be lost through point bounce is almost unbelievable. A simple point bounce at, say 6000 rpm, can easily cost 20 bhp. Unfortunately the points available for both Bosch or Motorcraft distributors can vary a great deal in quality and spring tension. It is therefore a good idea to equip yourself with a points tension gauge like the one manufactured by Sun. If you must use a contact breaker-type ignition, test the points' installed tension. A typical set of points will usually have a tension of around 19 or 20 ounces. Sometimes they are as high as 22 or 23 and as low as 16. If points are on the low side, your distributor may have point bounce as low as 5000 rpm. But even when they are on the high side, they are not likely to be effective much over 6500 rpm. You can often increase point tension by bending the spring in the reverse direction to its normal curvature prior to installation. This works for a short period of time but the spring will quickly fatigue and weaken and the result is less tension than if you had left them as they were. This means that if you want to use this tweak to achieve better point control you will have to replace your contact breaker set at very close intervals, 50-100 miles at the most. Now this might seem a lot of trouble to go to just to increase points tension, but take a look at Fig. 6-3 and decide for yourself.

The last power robber we will consider here is pitted points. When the points become pitted, the separation rate of the points during the opening phase is reduced. To achieve an effective spark, the opening rates ideally should be instantaneous. Of course this is mechanically impossible. But standard points in good condition come close enough to this ideal to provide the necessary results. However, when the separation rate of the points is reduced, spark intensity at low rpm is seriously hurt. This is one of those faults that has more effect on low-rpm power than high-rpm output.

The upshot of all this is that even on a street machine *if top performance is to be maintained*, it will be necessary to replace the contact breakers at regular intervals usually not greater than 5000-7000 miles. This really applies to both the Bosch and the Motorcraft distributor.

6-6. The best ignition system I could find for Ford's S.O.H.C. engine was the Autotronics M.S.D. The street version seen here performed just as well as the all-out M.S.D.7 race unit. . . .

6-7. . . . shown here. However, the race unit was generally more convenient to use because of easier hook-up and electronic tacho drive provisions. It also can be had with a built-in Soft Touch rev limiter.

DISTRIBUTORS

The standard distributor, be it Ford or Bosch, is only as good as it need be for the job it's intended to do. The most common problem, and this relates principally to the Motorcraft distributor, is a worn top distributor shaft bearing. The main shaft is supported by two bearings, one in the distributor body and the other down in the engine block. The engine block bearing rarely, if ever, wears, but the bearing in the distributor does. When wear occurs here, the shaft can orbit around the bearing. This causes the timing, dwell angle and contact breaker motion to change erratically. This means the worn bearing is distorting the three factors essential to peak performance. If the top bearing is worn and spark scatter occurs, replace the distributor body and/or main shaft to cure the problem. If you want a permanent cure, Manx Racing modifies distributors by installing a needle roller top bearing. For race engines, Holbay Engineering supplies a Lucas racing distributor adapted for use in the SOHC engine. When used in contact breaker form, this distributor is good for about 8500, maybe 9000 rpm. If you are looking for a purpose- built distributor then, your best bet is Aldon Automotive in England. They produce custom built units, which are priced, believe it or not, at about the level of stock units!

The last sort of distributor worth mentioning is the extensively modified one from Esslinger Engineering. This item is a very compact unit, having no mechanical or vacuum advance and is intended principally for use on alcohol-burning engines (midgets, alcohol dragsters, hydroplanes, etc.)

So far the discussion concerning distributors has hinged upon wear occurring of the top bearing. By replacing the contact breaker with some other contactless form of spark trigger, the bearing wear rate is substantially reduced and the problem is virtually alleviated. This route is by far the most sensible route to take, as we will now see.

ELECTRONIC IGNITION SYSTEMS

All too often, big claims are made for electronic ignition systems by their manufacturers. By and large, such claims need to be taken with a substantial amount of salt. Even though manufacturers' claims may be gross overestimates, do not, on your part, underestimate the value of a good electronic ignition system. To sort fact from fiction and explain what may at first appear to be a contradiction, let's look at the real tried and tested advantages of *contactless* electronic ignition systems.

The *first and most* obvious advantage is the elimination of one of the biggest sources of trouble in a contact breaker system, the breaker points. The second advantage is that an electronic ignition system is capable of delivering a much higher voltage at the plugs. And, in most instances, it also delivers more energy. Indeed, the Ford SOHC engine likes to have a good, big fat spark. If it's more accurately timed, as is usually the

FIG 6-4

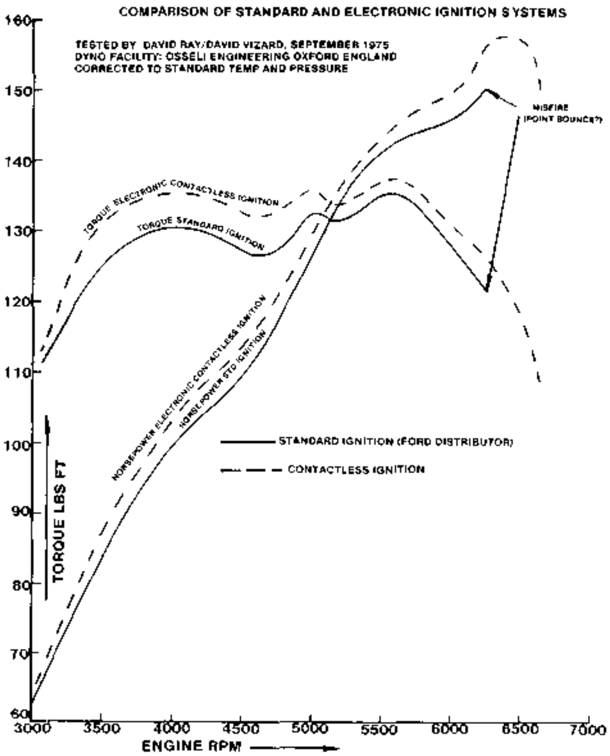
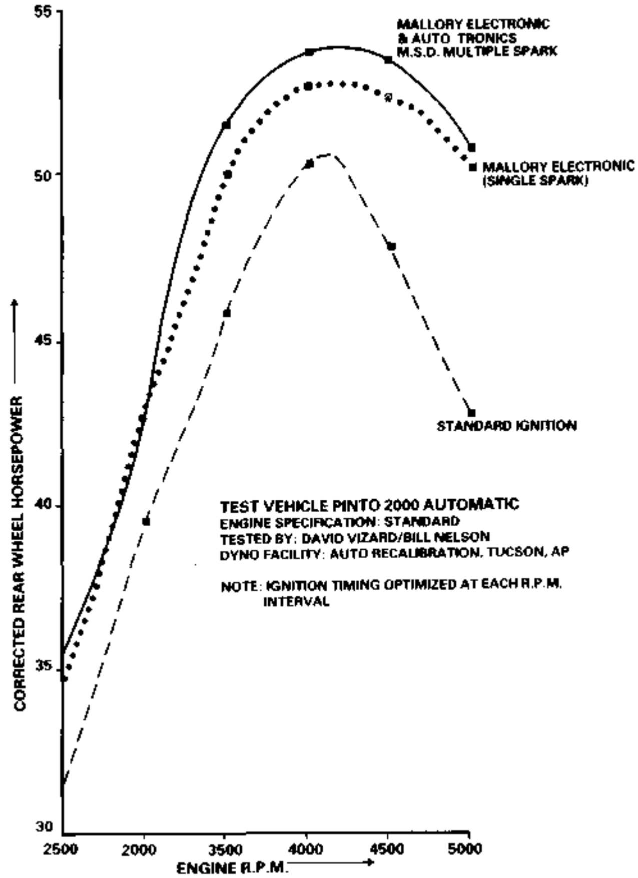


FIG 6-5
COMPARISON OF STANDARD IGNITION AFTER 5000 MILE USE SINGLE SPARK ELECTRONIC IGNITION (MALLORY) AND MULTIPLE FIRING IGNITION (MALLORY & M.S.D)



case with sound electronic systems, so much the better.

With an electronic ignition, your first move should be to increase the plug gap. Most electronic ignition systems are quite happy with gaps of around .035 – .040 inch (0.89-1 mm). The bigger gap helps the power output by allowing more energy to be dissipated in the gap. Being able to fire fouled and very cool plugs is sometimes worth a little extra power. In the higher rpm ranges, where normal contact breakers are starting to malfunction due to point bounce and erratic operation, the contactless ignition system can continue unflustered. Another power bonus is realized from this. Often, a breakerless ignition system with larger plug gaps can achieve maximum power with slightly leaner mixtures. This is a general rule rather than a specific one, though, for it depends on a number of other factors.

Having regarded manufacturers' claims dubiously, you may ask what sort of power increase can be expected. However, to answer the question we must look at the matter from two aspects. First of all we need a comparison of just how much extra power is possible when comparing a perfectly set up contact breaker system with an equally perfectly set up contactless system.

The second, which is specially relevant to vehicles driven on the road or long-distance events, is how much extra power can we expect to gain by having a contactless system which does not go out of tune, versus a contact breaker system which does go out of tune. In other words, what will the difference in power be 5000 miles down the road? Fig. 6-4 shows that the power output given by the standard ignition looks fairly respectable up to 6250 rpm, at which point the engine starts misfiring. And this misfire has every indication of being caused by chronic point bounce. The power curve given by the standard ignition in this instance was optimized in every way possible. In other words it was running with new plugs, new points, the dwell angle was set correctly, and so on.

The electronic ignition was then installed. After widening the plug gaps and resetting ignition timing, the power curve shown by the dotted line was achieved. As can be seen, it falls far short of many advertising claims but nevertheless is a worthwhile power improvement.

Fig. 6-5 depicts the difference between a standard ignition system which has run untouched for 5000 miles and an electronic system, the dotted line being the one given by a conventional, single-spark ignition. Here a much greater difference in power output is seen, which shows that the true worth of a transistor ignition system is maintaining peak spark efficiency after prolonged periods.

The third curve on Fig. 6-5 is for a more unusual ignition system. Instead of the spark occurring once on the firing stroke, it occurs many times, spaced about two degrees of crank angle apart. This system, known as the MSD system is manufactured by Autotronics is El Paso, Texas. Even on a standard 2000 engine, this multiple spark ignition showed its superiority over single-spark ignition. On more radical engines, it seems to be consistently worth two to three bhp more than a single-spark ignition. I feel that you can also expect the MSD-type ignition to work very well on engines which use alcohol or alcohol and nitro-methane fuel.

ADVANCED CHARACTERISTICS

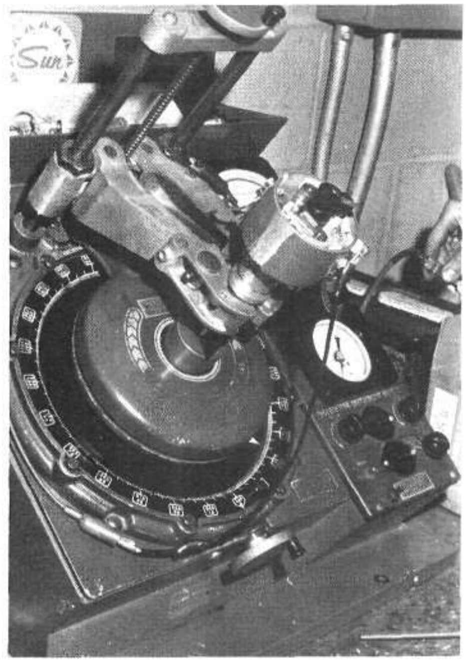
The only really effective way of getting the ultimate out of your motor is to have the distributor set up on a distributor machine as part of the overall tuning of your engine. Although it would be possible to quote springs and weights part numbers for different distributors, these can vary so much from one distributor to another that you may be no nearer the required curve with the recommended weights than with those you had originally. There are a couple of suggestions to be made in a broad sense but you will still need that dyno and distributor machine to dial that advance curve in precisely. (Though a custom built Aldon unit can often, for all practical purposes, be close enough with regard to advance curve characteristics).

I am now going to give you a few general rules to help put you on the right path.

Due to varying specifications, namely in compression ratio and camshaft, the *total* advance used by standard engines varies from a little over 30° to as high as 46°. Certain changes in engine specification produce certain changes of requirement in ignition timing. For instance, if the compression ratio is raised and nothing else is changed, the burning time will be speeded up. This means the engine will require less ignition advance than an engine with a lower compression ratio. Modified heads with higher compressions do two things: they squeeze the air in the cylinder tighter and secondly, they draw in more air in the first place. So you can say that generally speaking, a modified head will require less ignition timing for maximum power than that required by a standard head with a lower compression. Typically, going up two ratios on a well-modified head will dictate that ignition timing comes back on most engines about four degrees. If you are not sure just what the original ignition timing should be for your vehicle, try 32 degrees full advance as a starting point. Understand, that figure of 32 degrees is with all the mechanical advance out; that is not 32 degrees static. To set this ignition timing, you will have to run the engine up to about 4500-5000 rpm in neutral, then adjust the distributor with a degree-reading timing light. Or, you can do the old mechanic's trick of using a battery and a flashlight bulb plus a dab of white paint on your engine's crankshaft pulley at the required number of degrees of advance.

ADVANCE CURVE

If a cam change is made, a drastic change in advance requirements may come about, but this depends very much on the nature of the cam. Cams which have short timing and high lift, that is, cams designed to give good torque at low rpm, in general give better low rpm cylinder filling than a standard camshaft. Also, they will yield better high-speed cylinder filling. This means that such cams require the engine to have less total advance and possibly a less rapid advance curve at the lower revs. In other words if your engine has 12 degrees advance at 1500 rpm, a short-timing, high-lift cam may mean that it runs best with only seven degrees advance at 1500 rpm. This generally means that the curve has to be altered so that the ignition advance comes in slower and total advance is less. On the other hand if long timing cams are used, that is, those which yield poor low end power due to extensive overlap, the engine will need a distributor having the centrifugal advance coming in much quicker. The reason is that the engine has much less cylinder filling at low rpm. Thus the *effective* compression ratio is lower, so combustion rate at lower revs is slower; and thus more time is needed to complete the combustion cycle. As I said earlier, in most cases you are going to have to get your distributor curve set up on a distributor machine. One possible exception to this is the 1600 GT Motorcraft distributor; it has a very useful curve for 2000 engines using some of the hottest road cams, that is, those with up to a maximum of about 300 degrees duration. The 1600 GT distributor has a very fast initial advance. But do not fall into the typical hotrodder's trap of assuming that the more advance the engine has, the more power it will make. In fact, nothing could be further from the truth. An engine which uses the minimum amount of ignition advance to make its maximum power is generally the most efficient engine in terms of combustion dynamics, volumetric efficiency, burning effectiveness and mixture quality. With your engine dialled in correctly, you will find that a typical engine producing a power output of about 80 bhp per litre on premium fuel will need around 36-38 degrees total advance. Very often this amount of total advance can be shortened by two to four degrees by utilizing a multi-firing ignition, rather than a single-shot system.



6-8. Don't try and guess the advance characteristics of your distributor. What's stamped on the weights and what the advance really is, were often two completely different things. A distributor machine is the only sure-fire way to check and, if need be, correct your distributor's advance curve.

Another mistake is to assume that the vacuum advance is going to cost performance. This is not so. Removing the vacuum advance on a street machine will do nothing except lose mpg. The vacuum advance characteristics may need to be altered along with mechanical advance characteristics as you make changes to the engine. The most important part of the vacuum advance curve is that which occurs at the speeds that you most often use. Only on an outright competition machine should one need to consider removing the vacuum advance. If this operation is necessary, it is easily done on both Motorcraft and Bosch distributors. It's just a question of removing the vacuum unit itself, and disconnecting it from the baseplate of the distributor. Once you have done this, the base plate needs to be anchored firmly in position so that only the mechanical advance operates. Removing the vacuum advance canister will leave a hole in the side of the distributor. It's a good idea to make a small metal plate to cover this hole, otherwise you will find dust can intrude and cause the advance mechanism to stick or become erratic.

IGNITION SYSTEMS FOR TURBOCHARGED ENGINES

During its operating cycle, a normally aspirated engine sees a pressure range varying from as low as five psi up to the normal atmospheric of 14.7 psi. This means an ignition to cope with a little less than 10 psi difference. On the other hand, a turbocharged engine's ignition will have to deal with a much wider pressure range. A turbocharged engine can see much the same manifold vacuum as a normally aspirated engine, ie, down to about five psi absolute. At the other end of the scale, a turbo engine may have 10 psi boost so the ignition has to cope with as much as 20 psi difference variation in total manifold pressure. This means we have to have a spark timed to suit these variable conditions. Under normal circumstances, it is necessary, as the boost comes on, to have a form of pressure retard. You can view this as a means of backing off the vacuum advance until there is no advance at zero boost or vacuum, and then continue to back off the advance as the boost comes in. One takes over where the other leaves off. Achieving pressure retard is quite straightforward. The simplest way is to use the U.S. available dual-diaphragm distributor as is found on many stock 2000 powered vehicles. When used in conjunction with a turbocharger, the dual-diaphragm distributor utilizes only the outer part of the vacuum can, that is the section further away from the body of the distributor. Briefly, the diaphragm in this part of the vacuum can is free-floating; it can be pulled one way by vacuum and pushed the other way by pressure. The other part of the vacuum system is disconnected and not used. The dual-diaphragm distributor will typically start to retard at about 3 psi boost and continue to retard 10-12 crankshaft degrees, up to about 10 psi boost. This system works very well on an engine tuned for 10-12 psi boost with a compression ratio of 8:1 and possibly up to 10:1, if charge cooling is employed.

Because of their extensive experience in building turbocharged Pinto engines, Esslinger Engineering has compiled much useful information concerning ignition timing on boosted Pinto engines. First of all, if boost levels of 15-20 psi are anticipated as well as a compression ratio of, say, 7:1, a fixed distributor works very well. This is a distributor with no mechanical or vacuum/pressure advance whatsoever. It is fixed at 28 degrees. The low compression allows the engine to start alright with 28 degrees static advance. And, as boost and rpm come up, the fixed spark advance remains quite suitable indeed. If fuel consumption is a problem, Duane Esslinger recommends use of a distributor with no mechanical advance, but a vacuum advance yielding 10 distributor degrees. Such a distributor would still be set at around 28 degrees static advance and the extra vacuum advance which would occur at light throttle openings would increase fuel economy.

A great deal of interest is being shown in the 2000 Pinto engine for use in quarter- and half-mile midget racing, as well as formula hydroplane racing. Both these classes use alcohol fuels. Alcohol has different needs characteristic when it comes to selecting ignition timing. Normally, because of the slow burning characteristics of alcohol, a lot of ignition advance is needed. However, it has been found by extensive dynamometer testing at Esslinger Engineering, that as combustion efficiency is improved and compression ratios are raised, the amount of advance required tends to diminish. Typical spark lead for a less radically tuned alcohol engine is around 45 degrees total advance. A 2000 Pinto midget engine with a 15:1 or 16:1 compression ratio, a relatively short cam (for good power off the corners) seems to produce the best power with a timing between 25 and 30 degrees advance. This, in many cases, is less than is used in engines running on pump fuel from the service station.

Blocks, Pistons, Rods, Cranks & Flywheels

If you are pulling apart a SOHC Ford powerplant, you will notice that in most of these engines the pistons do not come anywhere near the top of the block at TDC. Since heads are standardized within a capacity group, varying compression ratios are achieved by varying piston crown height. For instance, on a 2000 engine, the compression height varies between 1.587 and 1.640 inch. If you are rebuilding an engine and desire higher compression while retaining standard pistons, go for the tallest piston possible. Fig. 7-1 shows you the various compression heights of standard pistons. Even with the tallest pistons, you will find that the crown is still short of the top of the block by a considerable margin. Higher compression ratios are achieved by milling the block. For example, .010 inch taken off the block on a 2000cc engine will reduce combustion chamber volume by 1.62cc. On the 1600 engine, .010 inch off the block will reduce this volume by 1.5cc and on the 1300 by 1.22cc

On the other hand, let's say you have a good set of standard pistons but they aren't the highest compression ones available. How much can be machined off the top of the block? The answer depends greatly on how much power you want from the engine and how fast you will turn it. Usually, taking .040, and sometimes .060 inch off the top of the block causes no problem. But there have been rare instances when problems arose where the block has cracked around the cylinder head bolt holes. This seems to occur principally on long-distance race engines, that is, engines which run races 50 miles or more and turn over 8000 rpm. Granted not many people will be using standard pistons in a long distance race engine and standard low compression pistons at that. I think for most practical purpose .050 inch off the block for a street driven engine is not going to present any problems.

O-RINGING THE BLOCK

In this next section I am going to quote some very large bhp levels, numbers which you may find out of proportion to the size of the engine. However, as you get further through the book, you will realize how these high power outputs are generated.

The purpose of O-ringing is to retain the intergrity of the head gasket when very high cylinder pressures are encountered. The sort of cylinder pressures I am talking about are those which generate in excess of 250 bhp and go clear up to over 500. Such high unconventional power figures are, of course, generated by somewhat unconventional methods. e.g. nitrous oxide injection and turbocharging. Both these subjects will be dealt with later on, but for now we will deal with maintaining head gasket intergrity when these large power outputs are developed.

The easiest way to hold a head gasket is to put a copper wire ring on the block's surface so that line contact loads exerted on the head gasket are greatly increased. This prevents the gasket blowing out. The method used for O-ringing blocks (which has been very successful on my own engines, as well as the Esslinger Engineering turboed, alcohol-burning dragster engines) is shown in accompanying photos. The tool used is an Iskenderian O-ringing tool. The normal technique for O-ringing blocks is to set the O-ring diameter concentric with the fire ring diameter on the head gasket. This yielded very promising results. However, Duane Esslinger found that on his *very* high horsepower engines, head gaskets became a problem once again. In an effort to overcome head gasket problems, many things were tried. Among them was a subtle rearrangement of the O-ring. This entailed locating the O-ring outside the fire ring on the head gasket. This appears to have solved the problem, so the moral here is, set the O-ring's diameter to the outside diameter of the fire ring, not its geometric centre, which is the more conventional practice.

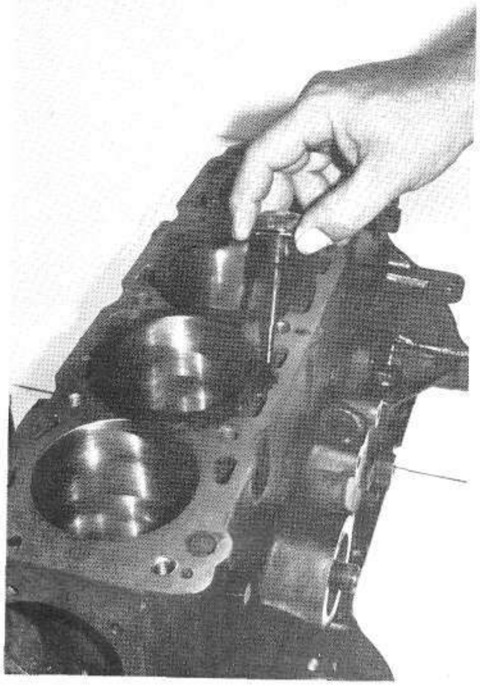
FIG. 7-1

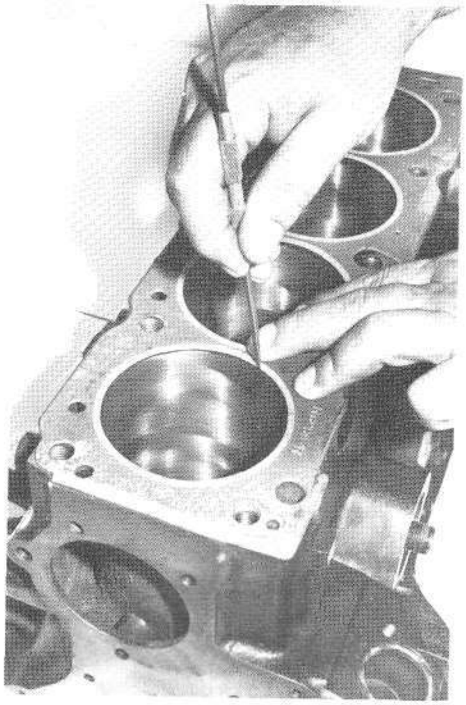
PISTON HEIGHTS

C.C	C.R.	PISTON HEIGHTS
1300	8.2	1.7453" (44.32 mm)
	9.2	1.7788" (45.19 mm)
1600	8.2	1.5776" (40.07 mm)
	9.2	1.6221" (41.20 mm)
2000	8.2	1.5886" (40.35 mm)
	8.6	1.6106" (40.91 mm)
	9.2	1.6398" (41.65 mm)

MAIN BEARING CAPS

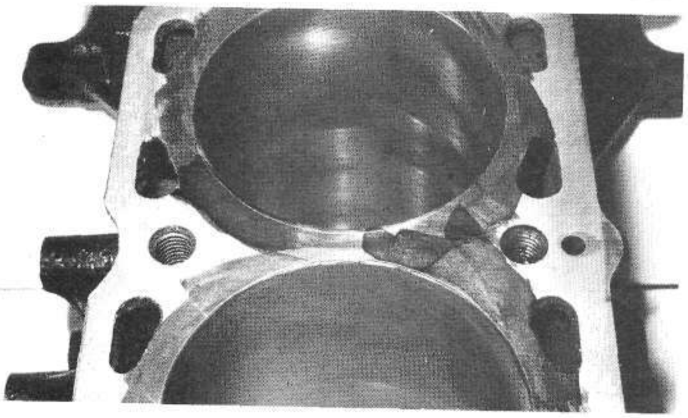
One problem that has been the concern of many racers on the small pushrod Ford engines is the strength of the main bearing caps. However, this does not appear to be a problem with the overhead cam engine. The block is essentially a very strong item. Steel mains caps are, to all intents and purposes, unnecessary. If you intend building a fuel-burning drag race engine and intend to boost it to umpteen pounds per square inch, then a block problem may arise. At about 450 bhp, block distortion sets in with a vengeance. Various means can be used to reduce the effects of this. One system employed by



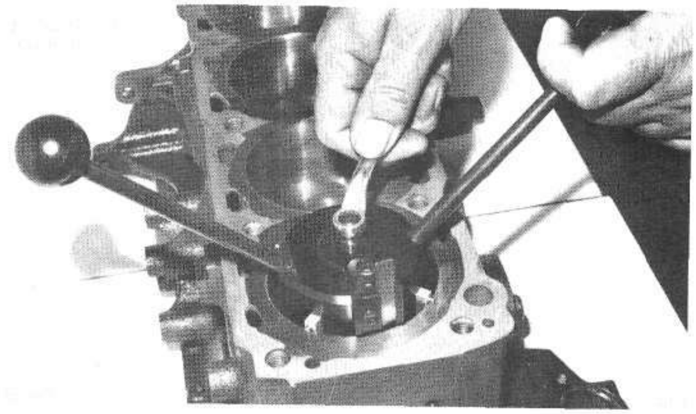


7-1. To O-ring the block, the first step is to blue the cylinder head deck face around the bores.

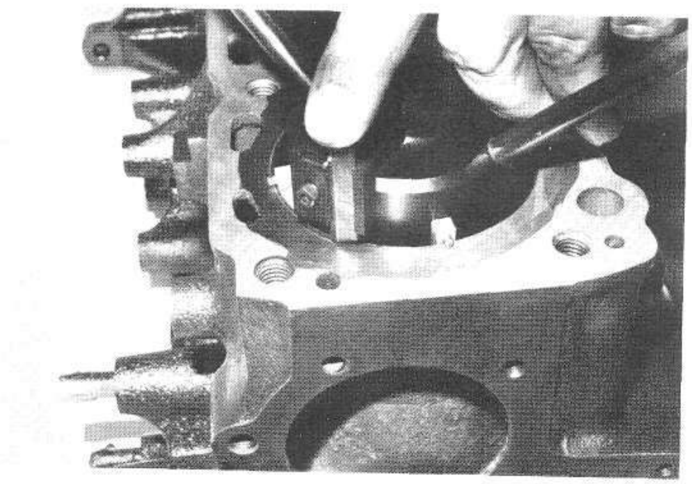
7-2. Mark out the gasket bore on the block.



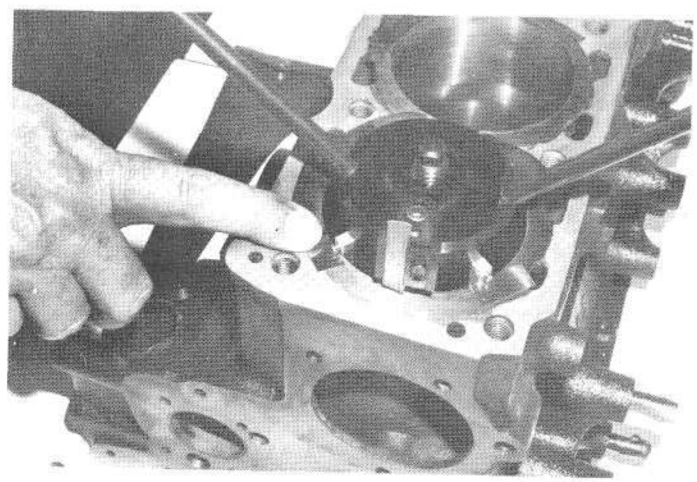
7-3. Here is the gasket bore clearly scribed out. This will be used as a guide to set the diameter of the cutter for the O-ring groove.



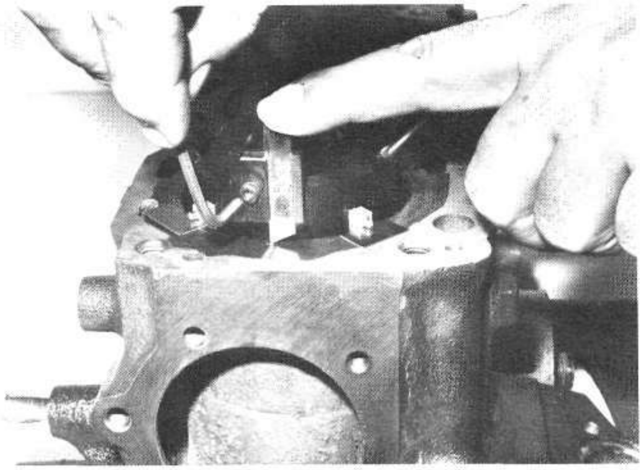
7-4. Set the Iskenderian O-ring cutting tool in the bore and adjust the cat's paws until they are located on the edge of the bore as shown here.



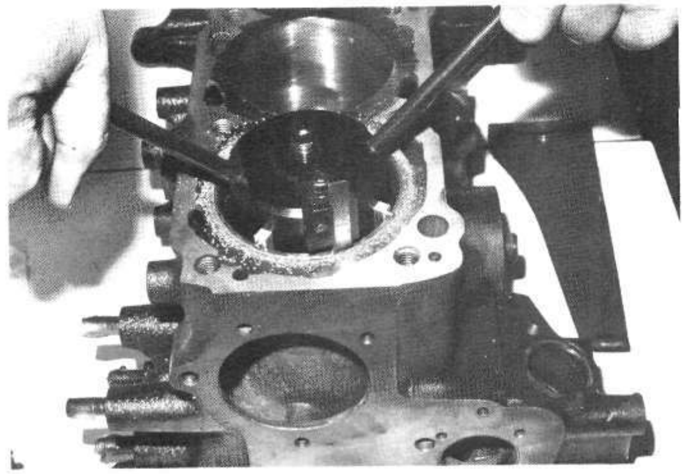
7-5. Slacken off the Allen key clamping the cutting tool and press the cutting tool down until it just touches the top of the block.



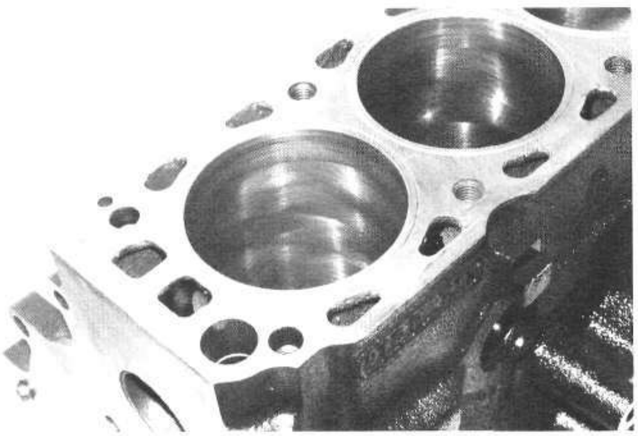
7-6. Place the depth setting pads under the pegs on the cat's paws. In this particular instance, 15 thousandths shims are being used.



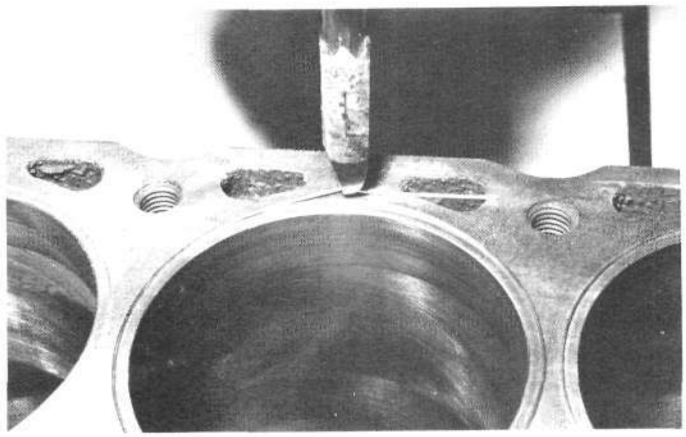
7-7. While the shims are still under the pegs on the cat's paws, slacken off the Allen securing screw holding the tool in place. Press the tool down onto the deck face of the block and lock it in place.



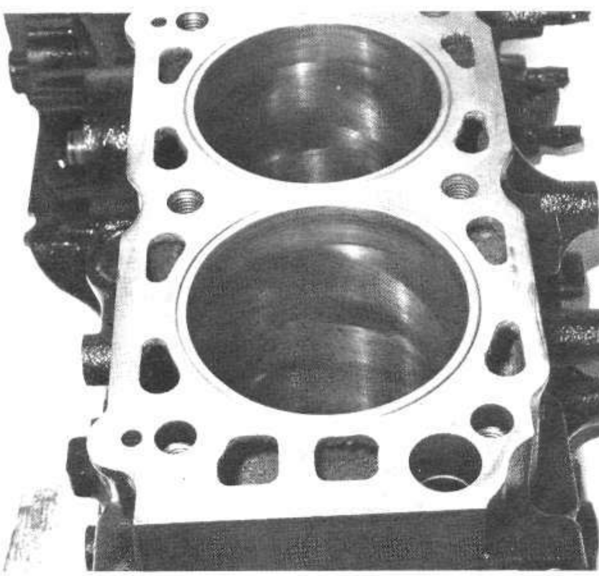
7-8. Remove the depth setting shims and commence cutting the O-ring groove as shown here.



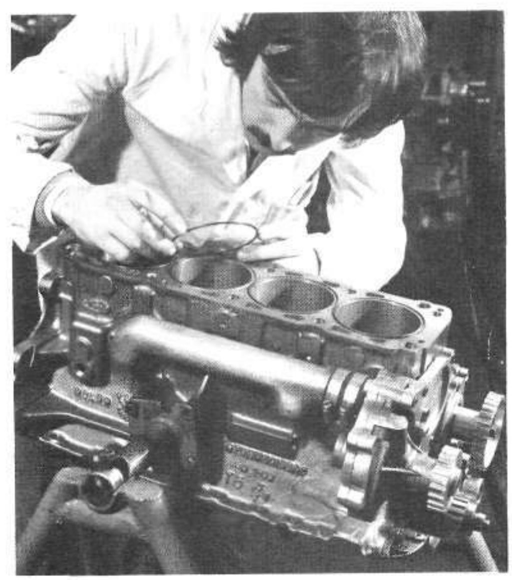
7-9. This shot shows the O-ring groove completed. . .



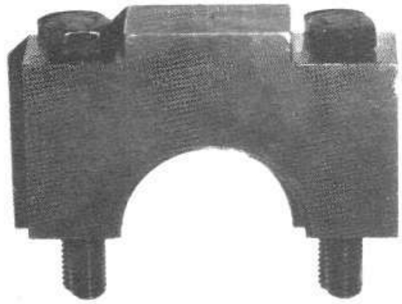
7-10 . . . and the next step is to install the copper wire in the groove. The most important part about installing the copper wire is that the ends of the wire must butt together without leaving a gap. If a gap is left, this could be the source of gasket failure.



7-11. Here is the completed job on the block.



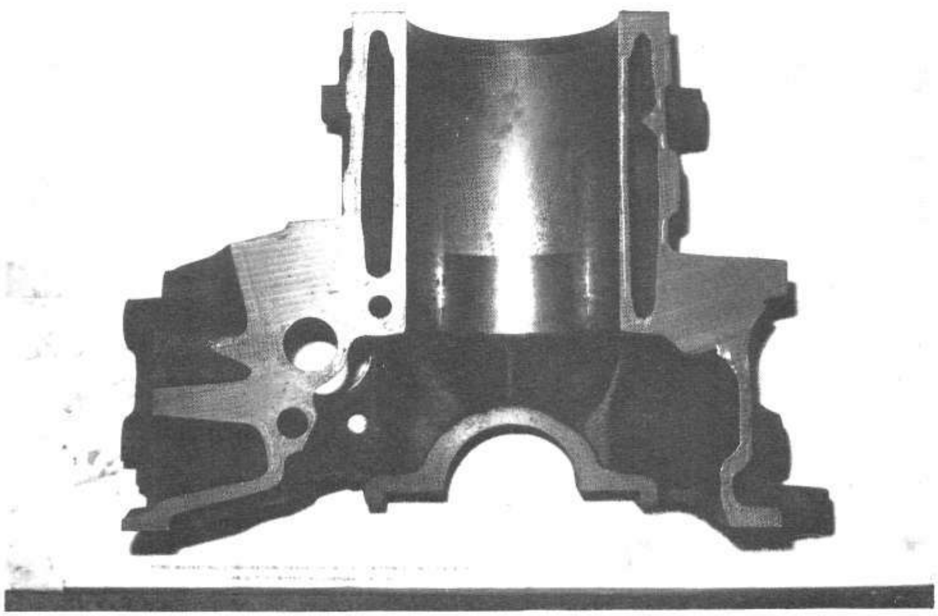
7-12. Instead of a plain copper ring, the Holbay O-ringing system employs the hollow, gas-filled Wills rings. These are laid into a semi-circular groove cut in the top of the block.



7-13. Steel main bearing caps are available for Ford's S.O.H.C. engine as can be seen by this example here. However, the standard main bearing caps are very robust, so steel items is really a case of gilding the lily.

Below

7-14. In this shot you can see just how much material there is supporting the cylinders. This particular block has a slight core shift, as can be seen by the fact that the material is thinner on one side of the bore than the other.



7-15. Here, Denny Wyckoff runs through the final stages of checking out a block for a hot street machine. Not only are the bores accurately sized but they have also been given the correct honing finish, as can be seen. The block has also been decked and a very important step, the tops of the bores have been chamfered to allow easy installation of the piston and ring assembly.

Esslinger Engineering is to cast a special resin or plastic into the block and run without the aid of any water cooling. (Until the early 1960s, concrete was used with reasonable success.) This appears to be quite satisfactory for a quarter-mile blast or, if you are sand-dragging, a hundred yards, as the engine is only under full power for a very short time.

Another, more expensive alternative, is to use the stiffer block a few of which were manufactured by Ford UK, and, if you are lucky, available from Holbay Engineering. This block is better able to withstand the stresses in engines producing 400 bhp or more.

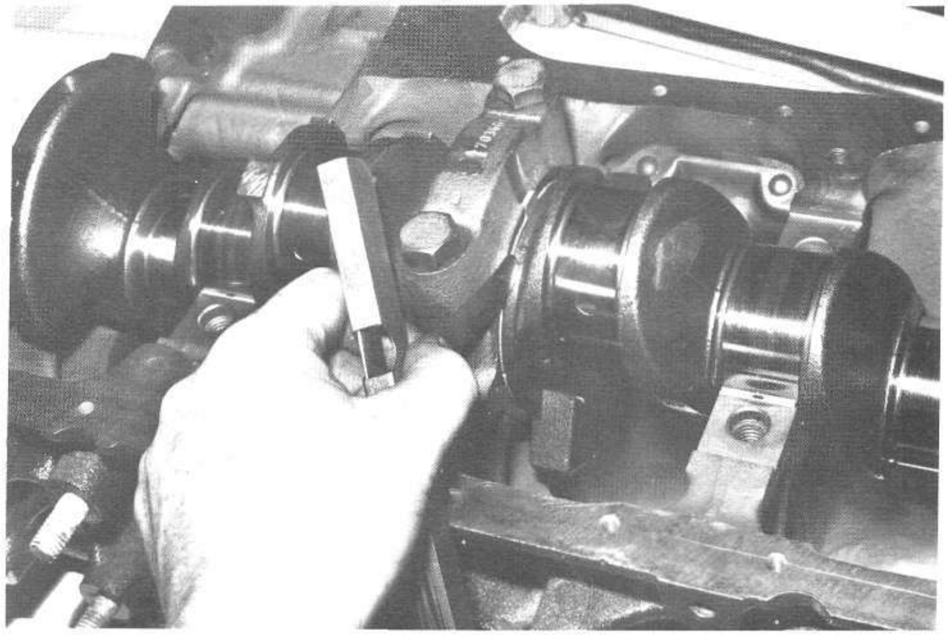
PISTONS, RINGS & PINS

To optimize precise piston assembly, there are three factors we must consider: the ring's gas sealing capability, the strength of the piston and how light it can be without losing reliability.

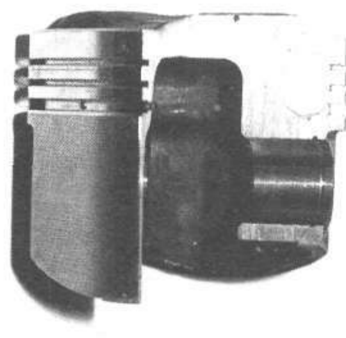
First of all, the standard piston is a good, strong piston for most purposes except N_2O injection. With 25-30 H.P. increase due to N_2O injection the top ring land will collapse. Another major point against the stock piston, especially those in the 2000 engine, is the width of the rings. The engine is capable of such high speeds that acceleration rates on the piston will cause ring flutter. This may begin as engine speeds approach about 8000 rpm. Another factor against the stock piston is its weight. Typically, a stock 2000 piston will weigh in around 630 grams. Of this, about 152 grams is the weight of the wrist (gudgeon) pin. With all this weight being hurled up and down the bore, a violently high stress is put into the con rod. From the performance and reliability points of veiw, it is best to get rid of all unnecessary reciprocating weight. The reduction of reciprocating weight does not mean the engine will have any more power in a steady-state condition. What it will have is more power in the acceleration mode. Since most high-performance cars are operating in the acceleration mode, the lighter the internals can be made, the more acceleration the vehicle will have. It may be a small factor, but it is nevertheless measurable on a quarter-mile dragstrip. By lightening components to their limit, we can see a reduction in quarter-mile times of around .10 second.

If you are not pursuing the ultimate degree of performance from your engine, but are more concerned with making a reliable high-performance street machine, the standard piston is as good a bet as any. Moreover it is capable of withstanding fairly high boost pressures, if turbocharging is the route you intend to take. The weight of the piston, of course, only starts to become a problem at really high rpm. A good alternative to this standard piston, is the TRW forged, high-performance piston. This piston is readily available over the entire U.S.A. but it is probably only available from companies importing speed equipment from the U.S.A. in England or Australia.

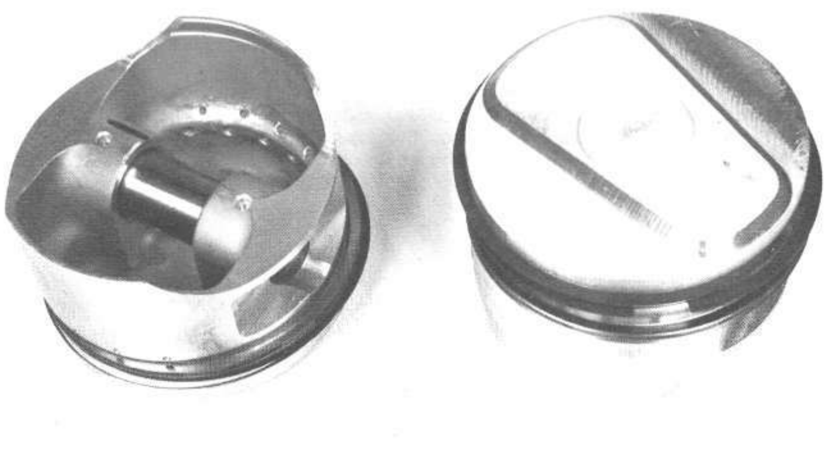
The TRW piston has a great deal to recommend it. It has proved to be as bullet-proof as a piston can be. It's relatively light and uses thinner rings than standard. As it comes from TRW it has a dome on it which the engine builder can machine to achieve the necessary compression height. This may need some amplification here to put you in the picture. Earlier we discussed compression ratios as high as 15:1. On alcohol engines, this figure may go even higher–16 or 17:1. There comes a time when it is not practical to cut the head any more. And at this point it becomes necessary to have a non-standard piston dome to achieve the compression. However, it is always best to get as much of your desired compression increase from milling the head, before resorting to high-domed pistons. Flat-top pistons have proved more reliable than domed pistons at any given compression ratio, so a dome is only used as a *last resort*. The TRW piston gives the engine builder complete freedom as to the size of dome he needs to use. Indeed there is enough material on the piston crown to machine the top flat. For most applications, this is what will be done, as sufficient compression can be achieved in most instances, by machining the head and decking the block. For all-out race applications, TRW pistons or indeed *any piston using a low expansion aluminium alloy*, hone out the wrist pin bores to give 0.0005-inch clearance on the pin at normal room temperatures. This much clearance produces a "drop through" piston-to-pin fit without any detectable "slack". If you want to pay for the best, both in machining and metallurgical quality, then the range of pistons offered by Cosworth is



7-15B. One of the clearances that really doesn't need changing from stock is the end thrust clearance on the crank. Irrespective of the stage of modification involved, just set the end float or end play to the standard setting.

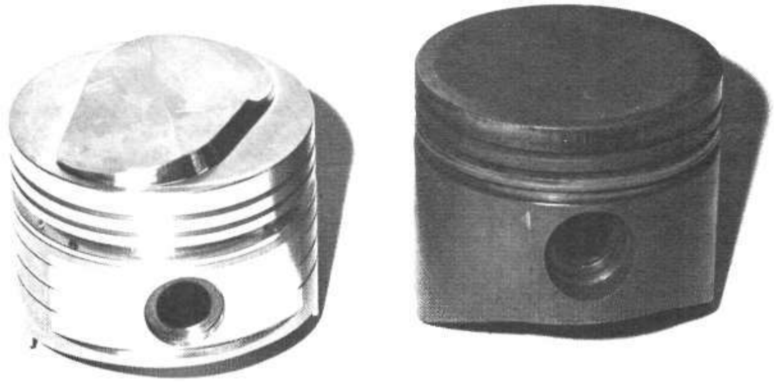


7-16. Here is a cut-away of a forged T.R.W. piston. As you can see it's a stoutly made piston. This proves to be the case in practice. Even when the dome is machined off, these pistons are quite capable of being used successfully in highly boosted, turbocharged engines.



7-17. At 506.5 grams, the Holbay forged piston for the 2000 cc engine is a fairly light piston. It is machined from a Holbay forging, and as received from Holbay, has the dome on it as seen here. If used in a Holbay engine, this dome does not need to have valve clearances cut as the Holbay camshafts lift only a little over 400 thousandths. If this piston is to be used with a more radical cam, then piston/valve reliefs will need to be machined into the dome.

7-18. This design of Aries-made piston was used in some of the author's most powerful normally aspirated engines. Special rings made by Childs & Albert were used to combat ring flutter at 10,000 rpm. The top ring is a very thin section Dikes ring. whereas the second ring is a conventional thin-section ring. The all-up weight of this piston was 508 grams, 65 of those being due to the super-light tool steel Esslinger piston pin. Although at first sight this piston looks heavier than the Holbay item remember that it is 1/4" longer due to the lower pin height on it. Although it's slightly heavier, it would be used in conjunction with a rod 1/4" shorter and therefore the reciprocating weight of the small end of the rod would be correspondingly less. Couple these two facts together and the piston, plus the reciprocating weight of the rod comes out at about as low as it's possible to get. Had this piston been made with a higher pin, it would have undoubtedly been 30 or 40 grams lighter.



7-19. Another custom piston, this time from Venolia. Compared with the standard piston on the right, you can see that this Venolia piston has a pretty healthy dome. This piston was built for a road-race engine, and as such, has too much dome, if only a 13 - 13.5/1 compression is to be used. In practice, much of this dome will be machined away and the compression built by machining the head face down to its maximum.

hard to beat. These full-skirted pistons require minimal clearance, use thin rings, and can be had, if required, with unfinished crowns. They are ideal for long distance events.

CUSTOM PISTONS

Apart from the Cosworth pistons, which can be "custom" or in stock, off the shelf designs, there are other companies that can supply pistons made to order. In an effort to reduce reciprocating weight in my engine to the barest minimum, I had some special pistons made up by Arias Pistons. Much discussion with the engineers at Arias resulted in a super-lightweight piston and pin combination. As you can see from the accompanying photograph, the pin bosses extend farther inward than on a conventional piston. This allows the use of a much shorter pin thus yielding a substantial weight saving. Weight of this piston, together with its short pin, was 508 grams, 24 percent lighter than the original. This, in turn, results in 24 percent less load on the small end of the rod at a given rpm. Looking at it another way, this gives the engine the ability to run 11.5 percent higher rpm before the load on the pin end of the rod is the same as it would have been with the heavier piston.

Another high-performance piston worth considering is the one produced by Holbay Engineering. This piston is unusual. It has 1/4-inch less compression height than the standard piston. It is built specifically for use with the Holbay connecting rod, which is 1/4-inch longer between centres than the standard rod. The pin used in the Holbay piston is of .8125-inch diameter instead of the 24 mm (.946 inch) diameter used in the standard piston. As it comes from Holbay, the piston weighs 506.5 grams, which makes it one of the lightest pistons you can get. Tests in turbocharged engines have shown that the piston is strong and reliable under extreme conditions. As you would expect from a race-bred piston, the compression rings are thin, to avoid ring flutter at high rpm. The standard ring width with the Holbay piston is 1 mm. The oil control ring is 2-1/2 mm wide.

RINGS

If you are going to have custom-built pistons, there are a few things you should know about rings. At the time of writing, gas-ported pistons are much in vogue. Some time ago I considered using such pistons. But a few simple calculations showed that if they worked, they could also cause a drop in power due to increased ring-to-wall friction. Discussions with engineers at TRW, Sealed Power and Arias revealed that gas-ported pistons cause severe wear right at the top of the bore. Moreover, it has yet to be proved that they indeed help power. Some racers claim power increases but the situation as yet is not clear. One thing can be stated for sure. A narrow, Dykes-type ring for the top groove does work. It will seal the piston well without causing exaggerated bore wear.

For an all-out race engine, I recommend 1/32-inch thick compression rings, the top one being a Dykes ring with 1/16- inch wide face. The second rings needs to be just a plain 1/32-inch wide ring. The material for the rings should be stainless steel for good wear resistance. As far as the oil ring is concerned, a conventional, double-rail ring with a stout support rail between will do the job. You may have problems obtaining rings of this size. A company I can recommend here is Childs & Albert. This company made a set of rings to my own specification at a very reasonable price. The only factor influencing my design was the fact that their limitations occur on diameter rather than ring thickness. It seems they have rings of pre-set bore sizes and it was necessary for me to have pistons and rings to suit a 3⅝ inch bore (92.075mm).

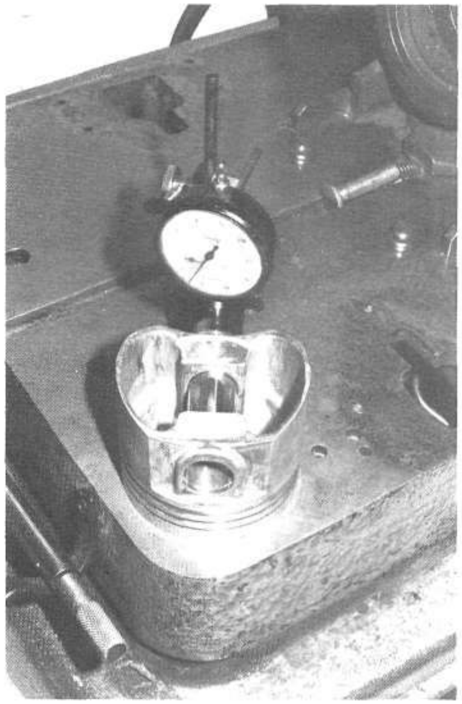
VALVE-TO-PISTON CLEARANCE

In 2000 engines, cams with less than 0.450-inch lift don't seem to require clearance notches in flat-top pistons which come flush to the block deck. When valve lifts approach .500 inch, valve reliefs may be needed if compression ratios exceed 13.5:1. Past .500 lift, you should check the valve-to-piston clearance regardless of compression ratio. Certainly, when .600-inch lift is used in engines with 14.5:1 compression, considerable valve cut-out is needed. Because of reduced chamber depth all smaller displacement engines will run into a piston / valve clearance problem that much sooner, so it's always wise to check this out.

If you are using a cylinder head with welded chamber, as detailed in the cylinder head section, the need for valve notches is reduced. The welding in the head that for a given compression ratio, the chamber is about .060-inch deeper, so this gives you a safety margin before the valve strike the pistons.

CERAMIC PISTON COATING

The piston is one of the greatest sources of heat loss in an engine. Being alumnium, it conducts heat readily and this heat has to be dissipated through oil sprayed on the underside of the piston, then allowing the oil to dissipate the heat through the rest of the engine or through an oil cooler. But remember that the heat being dissipated is also potential power. The coated piston keeps heat in the cylinder where it is working to give more power. Another plus is that it gives the piston greater strengh. Aluminium loses its strength rapidly as temperatures rise; the ceramic coating can make enough reduction in



7-21. When setting the deck height of your block, it is imperative that you know the compression height of the piston. An easy way to check this to see that they all are the same is to lay the piston on a flat machine bed and check each one with a dial gauge as shown here. Small differences in piston pin height can be compensated for by selected assembly with the rods.

7-21B After selective assembly of pistons and rods, each rod and piston set can be checked out for compression height in the bore. This is just what I am doing in this shot here. I tried each rod and piston assembly in each bore. There is no need to put the cap on the rod when doing this check. As long as the crank is lubricated and you turn the crank to push the piston up to TDC, you will do fine. Make sure you always check the piston to deck clearance at a consistent spot on the piston and it must be above the centre-line of the crank, otherwise you will also measure a degree of piston rocking.

7-22. Here I am just about to install a T.R.W. piston into a high performance street engine. The golden rule with the assembly of any engine is to give the rings adequate lubrication before installing them in the block. To make sure I put the piston in the right way, I mark a large arrow on the piston face, as can be seen here. Note that this T.R.W. piston has had the dome machined off it.

operating temperatures that the piston strength is dramatically increased. This suggests that a coated piston can be of far lighter contruction than a non-coated piston.

Another matter is bore clearance and the type of piston you are using. There are two types of alloy that we deal with; low-and high-expansion alloys. Many custom pistons are of high-expansion alloys and they must have relatively large bore clearances when measured cold. If these pistons are coated, they will expand far less and the reduced expansion could impair the effectiveness of the rings. If you are going to coat your pistons, consult your piston supplier on what clearance you should use. My own findings are that the pistons can run with about .0015 inch less clearance than they would uncoated. These figures are what have worked for me, but they are subject to further testing. They are not immutable. Typically, pistons made of high-expansion alloy need .006 to .007 inch clearance, and this can be effectively reduced to .0045 to .005 when the piston is coated.

PISTON-TO-BORE CLEARANCES

At one time or another, I have run the gambit on bore clearances and although different engines like different clearances, there is one sure thing I have found. The smaller clearances you can get away with, the better. I have heard of engines built with clearances as high as .010 inch. The builders of these powerplants aren't cranks putting together engines that don't work; these are people who build engines and then go and set records. At the other end of the scale though, I have seen engines built which were equally successful with as little as .0027 inch clearance. It's worth noting that a full skirted piston of the Cosworth style can run with less clearance than a slipper type cutaway skirt piston, even if the material from which they are made is the same.

I have found that for a low-expansion alloy piston, without a coating, .004 is fairly good for race applications. If a coating is applied, then reduce the clearance by about .001 on a low-expansion piston. If the piston is a high-expansion, California-type piston, then figure about .0005 less. If you are building a street motor, with standard pistons, go for top limit to top limit plus .0005. A standard bore clearance is typically .0023 to .0025 (0.066 mm).

PINS

During early development of the SOHC Ford engine, some pin lubrication problems were experienced. These problems caused galling of the piston pin bores. The problem was alleviated by putting an oiling slot in the pin bore and drilling an oil hole in the base of the pin boss. This works fine but it is still necessary to use relatively large pin-to-piston clearances as stated eariler.

If you choose custom-built pistons, be sure you have an oil- groove slot in the pin bore to eliminate this piston pin seizing tendency. Pistons normally bought out of stock, such as the TRW piston, already have oiling grooves in the pin bosses. This oiling groove becomes more important on coated pistons. The reason for this lies in the connecting rod design. If you look at the standard rod you will see it has an oil hole in the side. This hole is there partially to lubricate the bore walls but principally to cool the underside of the piston by spraying a jet of oil on it. This oil hole ensures plenty of oil up around the piston pin. When the piston is coated with a ceramic, this oil hole becomes unnecessary. In fact, blocking this oil hole can mean that the oil rings have less work to do. In a high-rpm engine, there is always plenty of oil mist around to lubricate the bore, so there is no reason to make the oil ring's life more difficult. If the hole in the rod is blocked because the piston is coated, there will be less oil to lubricate the pin. Under these circumstances, if adequate oil grooves aren't in the pin boss bores, seizure is almost sure to occur.

Finally, there is the matter of piston pin weight. It is difficult to lighten a piston radically for the simple reason that aluminium is a lightweight material and removing a great deal of it lightens the piston only a small amount. The pin, on the other hand, is made of steel and the removal of a small amount of material from it can yield a tremendous reduction in weight. Most pins installed in rods currently available are overweight. If pins are made of super-high-grade steels they can be reduced substantially in weight. The standard pin in the standard piston weighs a tremendous 152 grams. The pins available from Esslinger Engineering for the TRW piston or the standard piston, weigh 71 grams! In case you are worried about the strength of these pins, Duane Esslinger uses such pins in his 500-bhp, 10,000-rpm turbocharged engine and has not yet had a failure in his or any of his customer's engines.

BLOCKS

If a piston having short pin bosses is used, such as the piston built for me by Arias, a pin with the same thin wall design, but shorter, can be utilized. Such a pin is around 60 grams in weight. This is a big drop from the stock 152 gram pin.

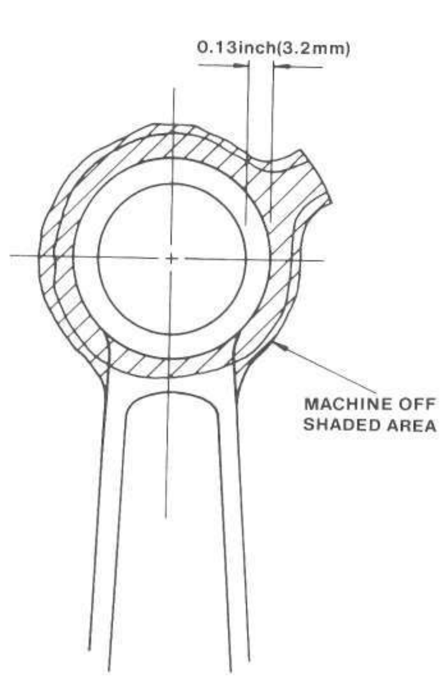
Let's look again at the Holbay piston. Although this was a super-light piston, the pin was relatively heavy at 109 grams. It would be simple to reduce the weight of this piston by around 30 grams by using a lightweight pin of the type used in Esslinger Engineering engines. This could bring the overall weight of the Holbay piston down to 470 grams, making it more than one-third lighter than the standard piston. Much of this extra lightness is because the piston is ¼-inch shorter than standard. Everything else being equal, the rod will be slightly heavier. However, the trade-off is in favour of the shorter piston as we will see in our discussion of the connecting rods.

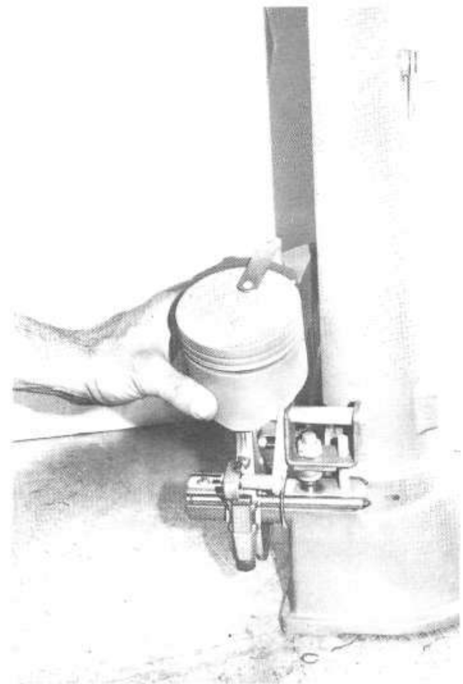
CONNECTING RODS

My first impression of the standard connecting rod was that it would prove to be the Achilles heel of the engine. Fortunately, time, experience and a number of people with a heavier right foot than mine, have proved me wrong. For all its frail appearance, the standard rod is very robust and in most instances is able to take fairly heavy loads. Being suitable for moderate race applications, I feel the rods sturdy enough for about 190 bhp normally aspirated or about 250 bhp turboed, plus about 8000 rpm for a circuit race engine and about 8500 rpm for a drag race engine. If the occasional gear is missed, these rods will stand going to about 9500 and some people have reported going as high as 10,000 in missed gearshifts on drag race engines without breaking a rod, which speaks well for the its strength. Of course the lighter the piston you hang on the rod, the more revs it can withstand. What will eventually kill the rod is fatigue. At 8000 R.P.M. a fully prepared rod is good for about 2 hours or so of racing, and at 8,500 about 30 minutes, but that may add up to a season's drag racing.

With this in mind, let's start at the top of the rod and work our way down to see exactly what can be done to improve the rod for high-performance use. A substantial amount of excess weight is in and around the pin area. It is possible to reduce the wall thickness around the pin to as little as 0.125 inch (3.2 mm) without causing excessive weakening. Removing this much metal is best done on a rotary table with an end mill in a milling machine. Fig. 7-2 shows what should be done. After removing weight from the small end, consider next the type of pin retention to use. In the U.S.A. the fully floating pin is favoured. Many racers make the pin fully floating in a rod, not really understanding why it should (or shouldn't) be that way. The press-fit actually yields a stronger pin assembly. By having the pin a press-fit in the rod, the strength of the pin in bending is radically increased. But it is not often realized by many racers that pin deflection can result in piston cracking as quickly as any other factor. By utilizing a press-fit, the beam strength of the pin can be increased sufficiently in a borderline case to prevent piston cracking. The only real justification for fully floating pins is in instances where piston changes are made regularly. Taking the pin out of a fully floating rod is, of course, a straightforward job and does not necessarily involve the destruction of the piston to remove it. Of course if piston replacements are having to be made because of failures in the pin boss, it's a sign that the fully floating pin may be causing the problem. To minimize the effects of piston distortion, and to avoid galling the bore of the pin boss in the rod, I have found that it's a good idea to set the interference fit to .0005-.0007-inch (0.013-0.018 mm)

Fig 7-2
CONNECTING ROD LIGHTENING DETAILS





7-23. A good way to lose hp on any engine is to have a piston and rod assembly which is bent. ALWAYS check that this assembly is true before installing in the engine.

7-24. This breakage caused the annihilation of an engine. The culprit is arrowed. The head of the rod bolt parted company from the shank of the rod bolt. As a consequence, the whole assembly let go. The rod bolt is usually, but not always, the first point of failure.

BEAM FLANKS

The easiest parts of the standard connecting rod to rework are the flanks of the beam. But there are several things you should be aware of when you are working in this area. First of all, do not grind across the width of the rod. Grind up and down its length. This reduces the likelihood of flaws being created by scratches across the width of the rod.

The next factor to consider is the oil hole boss. This is step-drilled from the journal bore. The initial drill size is relatively large and followed by about 3/4 mm (.030 inch) diameter. It is permissible to remove the oil hole boss so long as you do not break into the large diameter of this oil passage. If you do, you will lose the metering ability of the hole. Such a move may also result in a lower than desirable oil pressure on the top rod journal bearing. If, when you are grinding on the rod, you inadvertently grind into the larger diameter of the oil hole, the rod is not scrap. The easiest way to remedy the problem is to fill the hole with Araldite, allow it to harden, then re-drill the oil hole to the correct metering size. The object of the oil hole is to lubricate the bores and to cool the underside of the piston crown by spraying oil. If you have coated pistons, there is no real necessity for this cooling function to be employed. If you are building an engine last many, many miles and not necessarily a super high-performance engine, leave the oil hole to perform its secondary function, which is lubricating the thrust side of the bore. If you are building a race engine, where 20,000 miles of bore life is more than adequate, then fill in the oil hole and allow the bores to be splash-lubricated. This will give the rings the minimum amount of work to do. *Do not* fill in the oil hole if *you are not using coated pistons* as the piston crown is likely to overheat.

As far as the rest of the flank of the rod is concerned, remove only enough material to clean up the rod and remove any nicks that may be possible stress risers. Always use a generous radius at the point where the flank of the rod joins the pin end and the journal end.

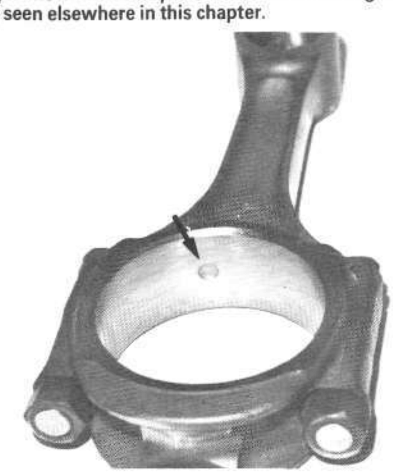
BOLT SEATS

One area that uninitiated engine builders seem to overlook when reworking a set of connecting rods, is the bolt seats. These seats are, in fact, very important. Often if a breakage is going to occur, it will occur across the inner radius of the bolt seat. See Fig. 7-3. Careful work here can reduce this likelihood. Take a needle file of about ⅛-inch diameter (3.2 mm) and remove all the rough machining marks in the radius of the bolt notch. Having done this, wrap fine emery cloth around the needle file and give that radius a good polish. Try not to remove any more material than possible from the flat surface the head of the bolt sits on. If this surface is out of true with the bolt hole, it can cause undue stresses on the head of the bolt. This can lead to the head snapping off at loads a lot lower than it is capable of. When the head snaps off, the bolt comes through the rod and you lose a rod. What happens from there on is anybody's guess.

Ideally, the bolt head surface needs to be touched up on a surface grinder to set the surface dead square to the bolt hole in the rod. If you are having this

7-25. If the rod bolt doesn't fail, the rod usually fails at the shank, about the place shown here. This failure occurred at about 9.500 rpm.

7-26. In this photo you can see just how much the pin end of the connecting rod has been lightened. Compare it with photos of standard rods seen elsewhere in this chapter.



7-27. This rod is to be used with a coated piston, and as such, the oil spray to cool the underside of the piston will not be required. The oil hole has been blocked up with epoxy resin (arrowed).

done, also have the machinist dress a radius on the grinding wheel so that the radius in the corner of the bolt face can also be ground. This leaves you very little to do except to finally polish the radius with 400-or 600- grit emery.

THE SPLIT LINE

When reconditioning connecting rods, many companies will grind the split line of the connecting rod at a very small angle so that when the bolts are tightened it will pull the sides of the rod in slightly. This is done so that when the rod is honed out to resize it, it will have a honing finish around the entire diameter. If the end of the rod and the cap were cut square, the honing operation would leave an unrefinished patch in the rod journal bore. This may cause their customers to think the job had been done shoddily, hence the reason for angle cutting.

As far as connecting rods for high-performance engines are concerned, *angle cutting the split line is out*. By angle cutting the split line, a bending load is applied to the bolt causing a significantly higher tensile stress in the outside of the bend. At high rpm this decreases their ability to take the loads they normally have to cope with. If you suspect that the con rods you intend using in your high-performance engine have been reconditioned, junk them. When having your rods reconditioned, have your friendly machine shop cut the split line exactly horizontally on both the rod and the cap.

CONNECTING ROD CAPS

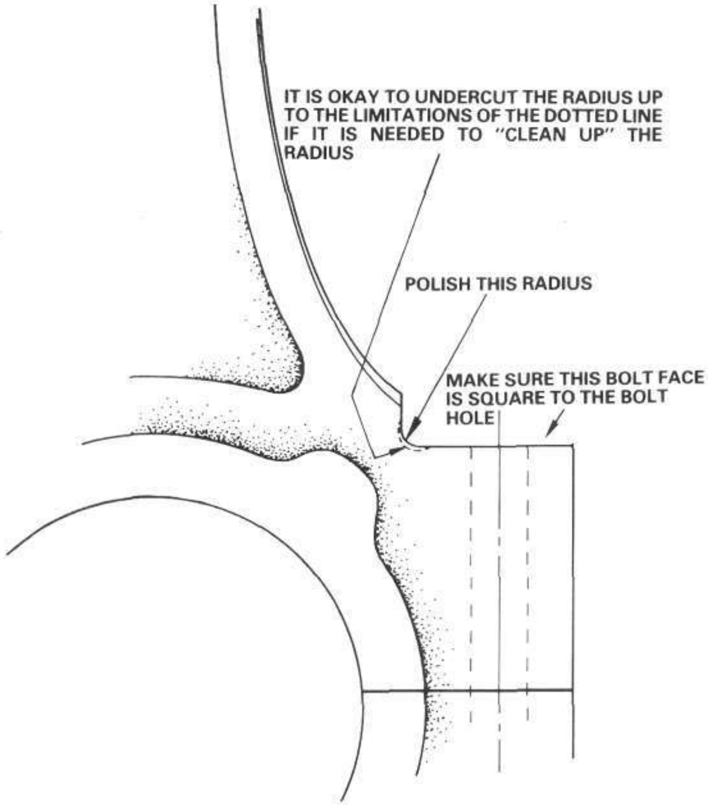
The balance pad on the connecting rod cap is another source of excess weight and can be machined off. This should be done carefully to minimize the loss of stiffness of the cap. The easiest way to do this is by using a ball-end mill and machining a slot in the end of the cap. When performing this operation, be sure to leave enough material so the balancing operation on the rod can be carried out without producing a weak cap.

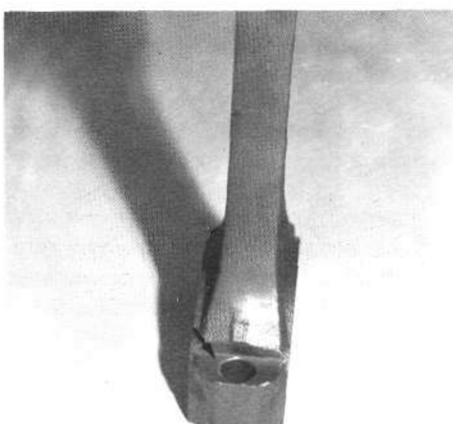
BEARING RETENTION

On high-rpm, high-output engines, it is possible, when utilizing standard connecting rods, to spin the bearings. If the bearing does spin, it will, of course, wipe out the connecting rod, and, if not caught soon enough, lead to severe failures in other quarters. The steps one takes to avoid this will depend on the usage of the rods. For a typical circuit race engine where peak revs are around 8000-8500 rpm, conventional straightforward precautions are adequate. These precautions involve sizing the *rod journal bore to the bottom limit*. This means that the pre-load or "crush" on the bearing shells is at its highest. Having sized the rod journal bore to its bottom limit, *do not chamfer the inner edge of the split line any more than is necessary to just remove the sharp edge*. In other words, a .002-inch-wide chamfer is adequate. If the chamfer is too large, in time the bearing shells will lose their "crush" fit by being swaged into the chamfer. This, in turn, will allow the bearing to spin.

When lightening the rod in the area of the split line, it is important that you retain as much surface area as possible at the split line, especially outboard of the bolt. When I say "retain as much surface area as possible" I am referring to

FIG 7-3





7-28. The radius between the rod flank and the bolt head seating face is a very important point. If there are any flaws in this area, it can be subjected to fatigue stresses which will eventually break the rod. When preparing the rod, be sure to polish this radius to remove any rough machining. Also check to see that the rod bolt seating face is square to the rod bore.

mating surface area. Any metal extending outside of the mating area is only along for a free ride, so it can be ground off. Ideally, these rods could use a little more metal outside the bolts, because metal here helps the journal bore to stay round at high rpm. If metal is removed outboard of the bolt, the rod journal bore could become oval along the rod's lengthwise axis. This causes the rod bolts to bend inward, a stress they are not designed for. As a result, the rod bolt may fracture on the outside of its bend. In this way, such stress can cause bolt failure long before loads are high enough to break the bolt were it in pure tension.

If you intend building a high-rpm turbocharged engine, it becomes necessary to prevent spinning of the bearings by pinning them. The following method has proved satisfactory on my own engines, and those of Jim Flynn (Flynno)! It can be used on all normally aspirated engines, and on turbocharged engines, up to about 400 bhp. The first thing you need to do is make a drill jig. This is done by obtaining a cap and bearing from an old or scrap rod. Drill a hole in the centre of the cap 3.1 mm (.127 inch) in diameter. This is your drill jig hole. Drill it straight through the bearing shell as well as the cap. Using old rod bolts, fasten the cap you have just drilled to the cap you intend to modify. The new cap must have its bearing in place. Torque up to about 20 ft./lb., then drill the 3.1mm hole through the new bearing into the rod cap. Do not drill all the way through, but 1/4 to 5/16 inch deep into the rod cap. Once the cap has been drilled, split the two caps and take out the new bearing and place it on one side. Carefully deburr the hole and clean out any cutting oil you may have used to drill the hole. Next, take a 1/8 inch diameter steel roll pin, dip it in Loctite and drive it into the drilled cap. More than likely, the roll pins you obtain will be too long for the job but that's of no consequence at this time. Before the Loctite hardens, wipe off any excess adhesive. This is important – if you omit this, a thin film of Loctite might be trapped behind the bearing shell, which may cause the bearing to be tight on the journal. Stand the caps in an up-right position so that all Loctite drains to the bottom of the hole. When the Loctite has hardened, grind any excess material from the roll pin so it protrudes from the surface of the cap by .040 to .050 inch. Once you have ground the pin to the correct size, give the end of the pin a good wire brushing to deburr it and round off the corners.

Once you have drilled the caps and located the pins, turn your attention to the bearing. Expand the hole in the bearing from its previously drilled 3.1 mm size to 3.2 mm, that's .1299 inch. This is meant to provide clearance for a 1/8 inch (.125) diameter. Do not make the bearing hole exactly the same diameter as the one in the rod. It needs to have just sufficient clearance so it can centre itself in the rod. Once the holes have been drilled, deburr the bearing on both sides.

Roll pins installed in this manner have proved adequate. However, I must point that should a hardened steel roll pin come loose, it can score the crank journal. To counter this, Duane Esslinger makes the bearing retaining pins to bronze. Thus, should a pin come adrift for any reason, the bronze will not cut the journal.

SURFACE TREATMENT

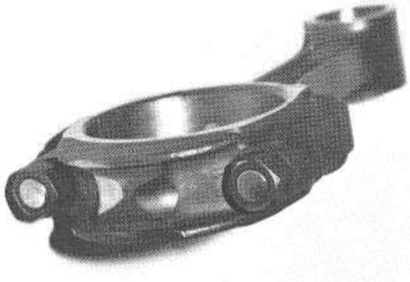
Under this heading we are going to discuss tuftriding and shot-peening. The two processes are vastly different, but they produce a similar result. They allow a connecting rod to operate much closer to its stress limit before fatigue fracture occurs. It is rare that a connecting rod breaks through pure overload. Instead, it is usually a matter of overload and time. If you fully reworked a set of standard rods and installed them in an engine, you could probably run that engine at 11,000 rpm for a few seconds without breaking them. On the other hand, if you ran the engine continuously at 9000 rpm, the rods would undoubtedly *eventually* break, but certainly not as quickly.

Tuftriding and shot-peening extends the period that the rods can safely be used, by virtue of the fact that they increase the rods' resistance to fatigue. Tuftriding does this by chemically changing the surface of the rod. Shot-peening achieves a similar thing by compressing the surface of the rod. After shot-peening, the surface of a connecting rod is actually under compression. This means a certain amount of tension must be applied to the rod before the surface of the rod reaches zero stress. Most rods break due to tensile loads, rather than compressive ones. By compressing the surface, we reduce the possibilities of a flaw fracture occurring, because such fractures normally only occur during tensile loading. The strength of an item is very much dependent on surface charactoristics. Cracks rarely propagate from the inside out. Indeed, in 99 percent of the cases, it is from the outside of the rod in. Once a crack spreads inward, the cross-sectional area of the rod is vastly reduced and, of course, the end of the crack represents a focal point for stress. By having a compressive surface, it is more difficult for a crack to start, hence the increased resistance to fatigue.

If you are going to tuftride the rods, it should be done almost as a last operation. When you send the rods for tuftriding, they should be complete, except for grinding the split line and honing and sizing the big end. All other operations should be done, however, such as polishing, radiusing the bolt notch, finishing off the bolt surface and lightening. If a rod is to be shot-peened, it also needs to be completely finished prior to peening treatment. Point out to the shot-peening operator the important areas of the rod, namely the flanks of the rod, the area immediately under the pin boss and the radius in the bolt slot. Make sure shot peening takes place adequately in the bolt face radius. It's a good idea to temporarily install smaller bolts with plenty of clearance around the head of the bolts so that the peening shot strikes the radius and does the job it is supposed to.

The next question will likely be: "why not combine the benefits of tuftriding and shot-peening together?" How much is to be gained by this one-two punch is uncertain, but if the processes are to be combined it must be done in a certain order. If you shot-peen the rods first, tuftriding will cancel out the shot-peen, for the tuftriding process also acts as a stress-relieving process. This means that if the two processes are to be utilized, the rods must first be tuftrided. After this has been done, they should be brushed all over with a stiff nylon brush to move the dusty deposit left by the process. Once the rods have been cleaned to a dull sheen, the appropriate surfaces can then be masked. The cap and rod should then be temporarily bolted together with bolts having small heads, so as to give shot access to the bolt face/beam radius.

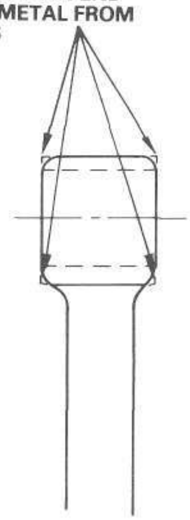
7-30. To avoid spun bearings on high rpm engines, it's necessary to make sure the bearing retention is good. One of the simplest ways of doing this is sizing the rod journal bore (that's the big end) to the minimal size permissible within the tolerance allowed. When I cut the rod as shown here, the bottom limit on the bore diameter was represented by a zero reading on the dial indicator. As you can see, this rod is smack on the bottom limit.



7-29. A considerable amount of material can be removed from the balancing pad of the rod cap. The two conical cut-outs in the balance pad of this rod shown here, represent the amount of material that had to be removed during the balancing operation on the rod.

FIG 7-4

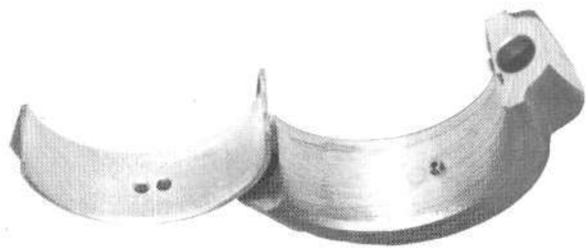
BALANCING LIGHTEN STANDARD ROD



BOLTS

The standard bolt seems to be about the right strength to fail about the same time as the rest of the rod. In other words, it's neither excessively strong nor particularly weak. There are a few things you should know, though, if you are to avoid bolt-breakage. At high-rpm use, the standard torque value is too low, especially if you are hanging a fairly heavy piston on the far end. To avoid breakage, the rod nuts need to be torqued to 40 ft./lb., not the standard 29-32 ft./lb. Select the rod bolts carefully. Check each one for scratches and nicks in the shank. Extra security can be had through crack-testing, be it Magnaflexing or a similar process. Also make sure the threads are well oiled before assembly, to minimize thread frictions.

If you are totally committed to the upmost in security, I suggest installing S.P.S. 180,000lb bolts, type intended for use on the big-block Chevy engine. These bolts are larger than the standard Ford rod bolts, so the holes in both the cap and rod require reaming out to size to suit the knurled section of the bolt shank. Tighten them to 50 lb/ft. torque. In England these bolts may be difficult to obtain. However, John Wolfe Racing should be able to get them if needed.

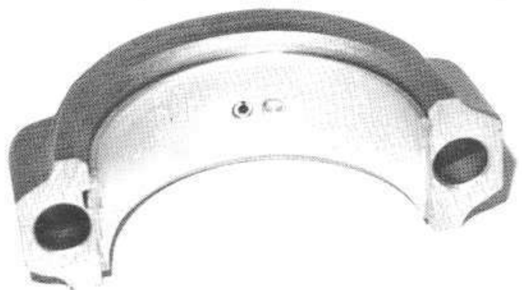


7-31. Here is the roll pin used to retain the bearing installed in the rod cap. Note the extra hole required in the shell to suit this pin.

BALANCE & ACCURACY

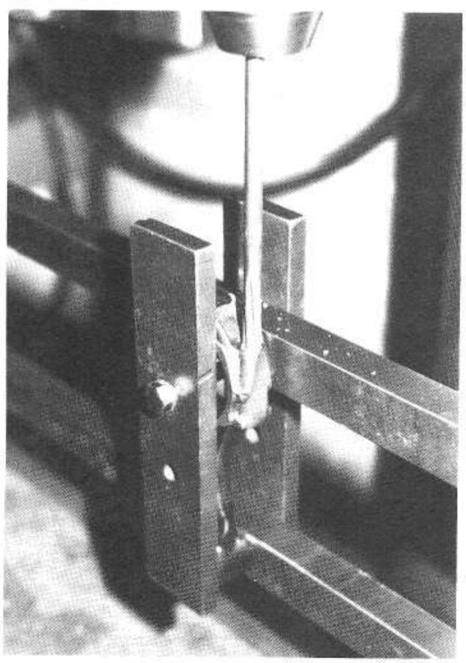
We are now down to the last few operations for complete connecting rod preparation. At this stage we must make sure the rod is not bent or twisted. This is especially important if it has been head-treated. Most machine shops have facilities for checking the accuracy of rods either on or off the piston.

Once the rods are determined accurate, the last operation is to have them balanced as a set. If you have taken quite a bit of material from the rod cap, your balancing man is not likely to be able to find too much metal to remove to even up the balance. Fortunately metal can be removed from the ends of the bolts, that's the nut end of the bolt, so this gives a little leeway. To achieve balance on the pin end, metal should be removed from the area shown in Fig. 7-4. If you have diligently lightened the rods as those shown in the photo, you should have units weighing around 565 grams and are good for 8000 rpm,

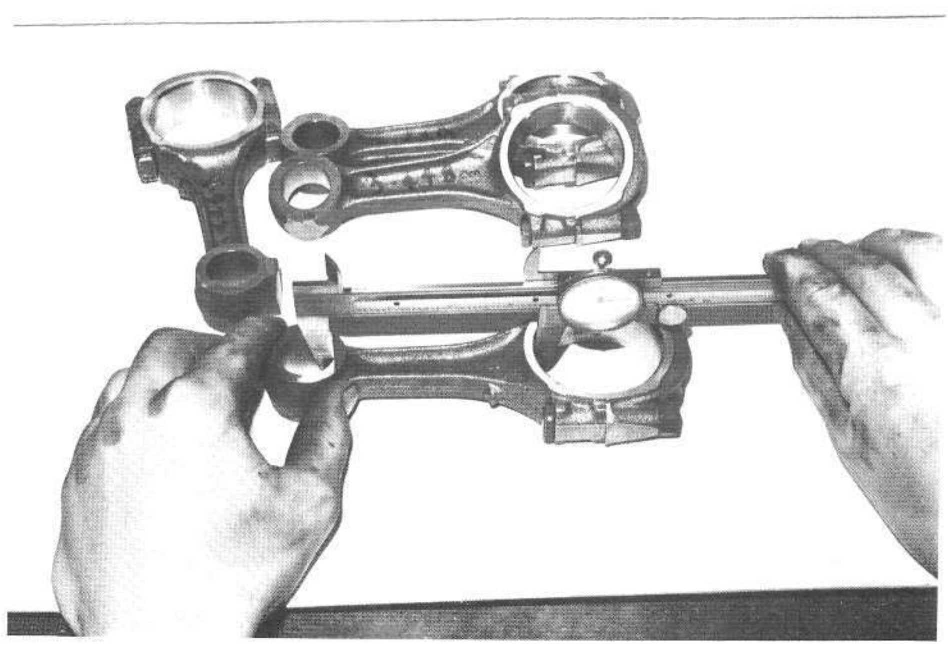


7-32 This is what the rod, cap and bearing look like when assembled. Be sure the pin does not stick through the bearing surface, otherwise it will just eat the journal alive.

7-33. If you want to completely eliminate a rod failure due to a bolt breakage, then the installation of a big block Chevy rod bolt of S.P.S. manufacturer is a good move. As can be seen here,not only is it slightly bigger than the standard rod bolt but it is also made of a better material.



7-34. To install the S.P.S. big block Chevy rod bolt, it is necessary to ream out the existing rod bolt hole to suit the size of the new rod bolt.



7-35. When it comes time to assemble your engine, select your rods to suit the pistons. If you use the longest rod with the piston having the shortest compression height, you will get the most even piston heights when the pistons and rods are installed in the block. Of course the crank may have slight variations in stroke length, but if you swop pistons around at this point, put the longest rod and piston combination with the shortest stroke, and you should end up with fairly even compression heights in most instances.

under circuit race conditions with excursions up to 9000 rpm permissible.

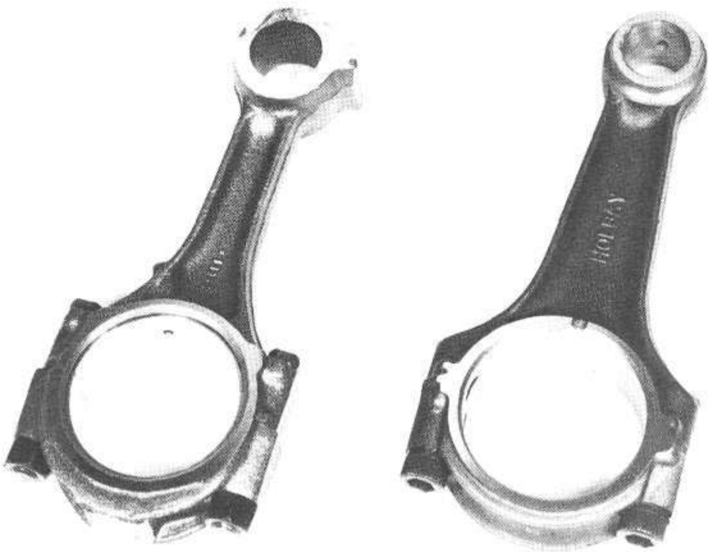
ALTERNATIVE RODS

If you are in the market for pure, high-performance connecting rods, there are several you may chose from. As expected, a very nice connecting rod is made by Holbay Engineering. This longer rod is 5¼ inches centre-to-centre, which yields a more favourable rod length-to-stroke ratio. Although no back-to-back tests have been done, it would appear that this longer rod helps power output, as engines built with this rod seem to perform better than those built with shorter rods. As far as weight is concerned, a Holbay rod is relatively light, although heavier than that achievable by lightening the standard rod. As it comes from Holbay, it weighs around 595-600 grams. However, it is entirely practical to lighten the pin end of the rod by another 20 grams or so. This can be done, either by removing metal from the outside face of the pin end, if the small pin diameter is to be retained, or if you intend using a set of specially made pistons, the pin bore can be made 24 mm diameter thus achieving a weight saving here.

In overall terms, total reciprocating weight can be reduced by the utilization of this longer rod and shorter piston. Another alternative to the standard connecting rod is the steel rod produced by Fred Carillo. This rod is 5⅛ inches long and is primarily intended for the small-block Ford V8. The only snag is that it's very heavy: 700 grams. The great weight of this rod would, for me, prohibit its use unless it was a last resort.

Another rod is the 289/302 small-block Ford V8 rod. As expected, it's 5⅛ inches centre-to-centre and can be lightened to put it fairly close to the weight of a standard Pinto rod. If cost is no object and you are after the ultimate rod, the lightest, strongest one you can get is the Jet Engineering part which is made of titanium alloy. This rod is based on the forging used for the Cosworth V8 titanium rod, and as such, it has a centre-to-centre distance around 5¼ inches. Since these rods are all special-order items, you can order the small variations in the centre-to-centre distance to suit your requirements. This means the rod could be sized from 5.150 to about 5.3-inches centre-to-centre. This rod, in conjunction with a suitably light piston, will give you just about the lowest reciprocating weight possible by current technology, and it will also give your engine's bottom end a 12,000 rpm capability.

Another rod available for the 2000 engine – not nearly as light as the titanium rod, but far lighter than the steel one – is the aluminium rod made by Super Rods. This has a weight of 495-500 grams, lighter than any of the steel rods by a significant margin. These rods are capable of more than 10,000 rpm and priced at a figure well within the reach of many racers. The only snag to aluminium rods is that their fatigue life is relatively short, compared to steel or titanium. The fatigue life has proved sufficient for a typical season's



7-36. A comparison of the standard rod and Holbay rod shows that the Holbay rod has the metal in places where it will do some good. The rod weighs in typically at 595 - 600 grams.

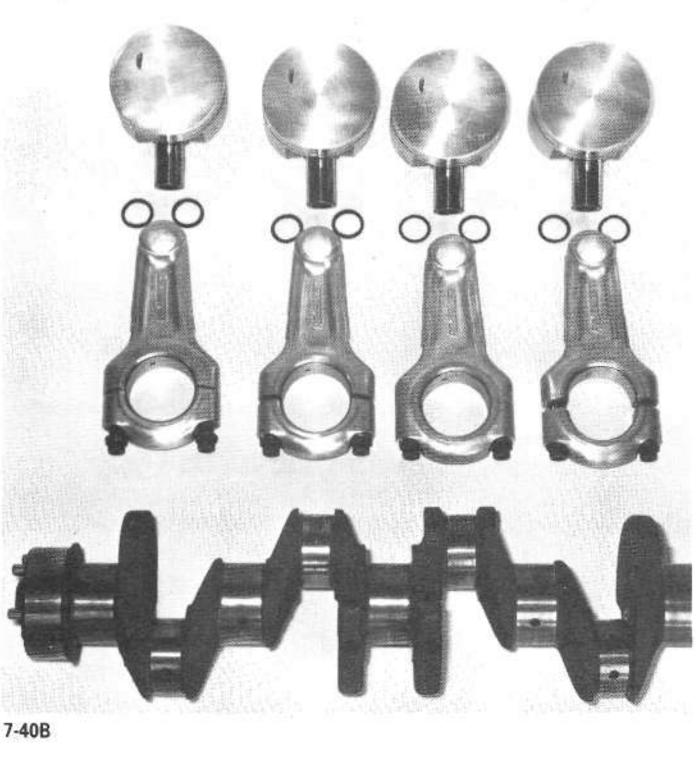
7-37. As with most high performance rod designs, the Holbay rod utilizes high tensile, screw-in bolts rather than the through bolts typically employed by the standard rod.

7-38. The rod shown here is a Corillo rod and it would appear to be based on the rod used in the Ford small block V8. It has 1/8" longer centre-to-centre distance than the standard 2-litre Ford rod. Although well-made, its biggest drawback is the fact that it weighs 700 grams.

7-39. The lighest, strongest rod that can be used in Ford's S.O.H.C. engine is the Jet titanium rod, of which an example is shown here. This rod weighs in at a little over 300 grams and is good for about 12,000 rpm.

7-40

7-40A



7-40B

7-40. This aluminium super rod is a type I have used in my own 2-litre engines on occasions. They have been run as high as 10,000 rpm on the dyno with no problems, but their fatigue life may be too short for a season's circuit racing, although they are ideal for drag racing. The all-up weight of this rod depends on the centre-to-centre distance involved, but typically about 490 grams is about as heavy as they will get.

7-40A. TRW pistons, aluminium Super Rods and a standard crank with suitably sized bearing journals proved to be robust enough for a reliable 385 BHP turbo engine.

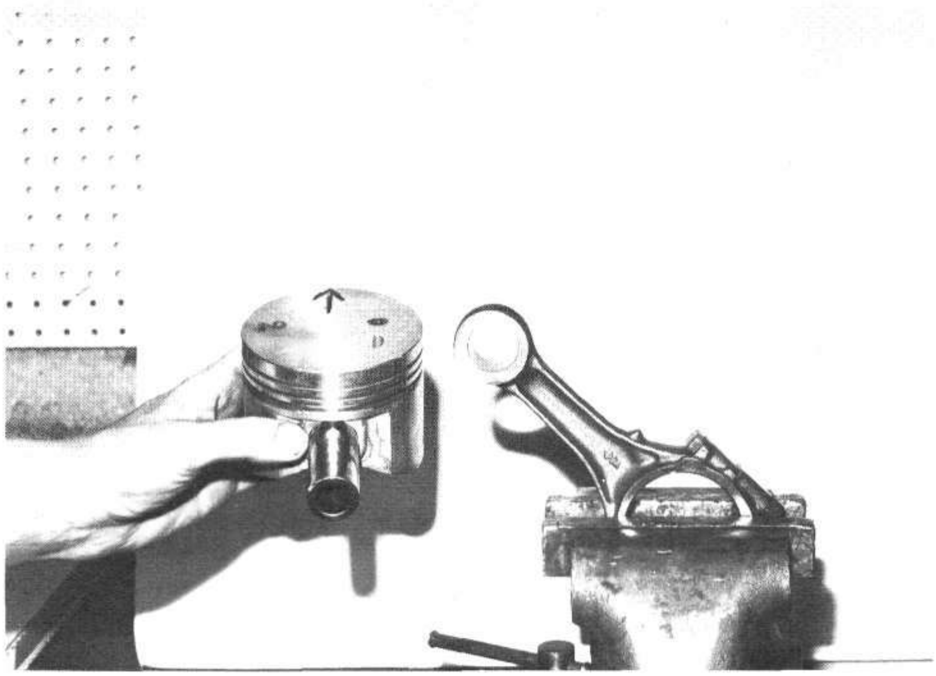
7-40B. Shown here is the bottom and assembly utilizing aluminium Super Rods Because of the bigger bulk the block has to have clearance notches ground into it.

drag racing, but it may be inadequate for the circuit racer who attemps to run a season on them.

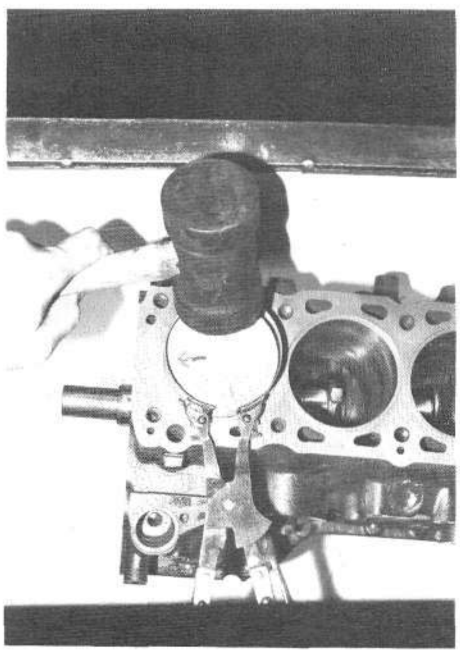
CRANKSHAFTS

Fortunately, the standard crankshaft is a very strong item, though fatigue will eventually cause it to break. It will cope with most applications in its stock form. It can for short periods, take the punishment dished out by a 210 bhp circuit race engine spinning 8000 rpm. For shorter periods, such as drag racing, it can safely withstand 25 psi boost and 10,000 rpm. For applications this severe, though, it's always best to take every precaution possible. The first thing to consider is bearing clearances. Once again, Esslinger Engineering's high rpm turbocharged engines have brought problems to light which may not otherwise have come to the forefront for some time. During development of this engine, Duane Esslinger found that bearings were starting to spin in the rod journal bore. This problem was alleviated to an extent by increasing the bearing clearance. Because of the tremendous capacity of the standard oil pump, the increased bearing clearances did not seem to unduly affect the oil pressure. Duane has experimented with bearing clearances on the rod journals as wide as .010 inch. Even with these large clearances, the standard oil pump proved still capable, even at an awesome 500 bhp and 10,000 rpm. As a matter of fact, Esslinger achieved longer bearing life under these circumstances than with much smaller bearing clearances. Fig. 7-5 suggests bearing clearances which may seem to be on the large side to many of you yet I have built engines using some of these larger bearing clearances and have found them to be entirely satisfactory. It might be worth noting at this point that if you are using a dry sump system, compare the size of the pressure side of the pump with that of a standard unit. Some dry sump units use a smaller pressure pump and therefore cannot be run with wide bearing clearances without causing oil pressure to suffer.

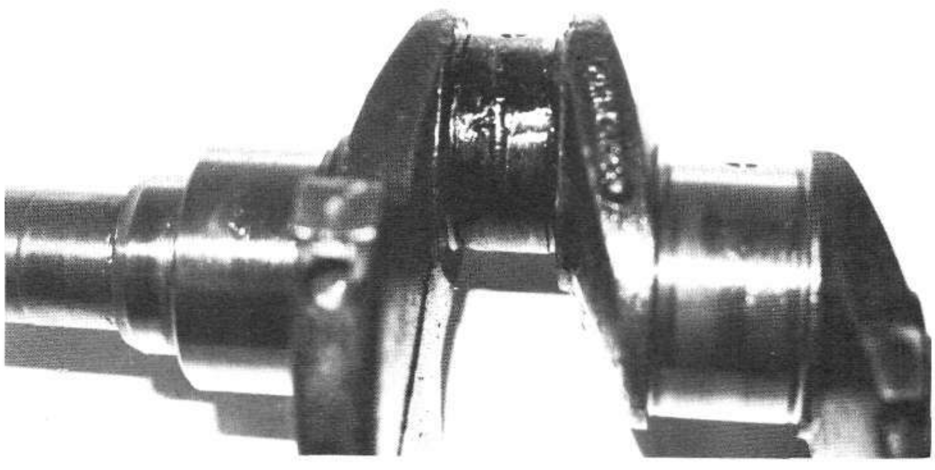
One final aspect worth considering is the way the bearings are assembled into the engine. Tests indicate that



7-41. When you assemble the rod and piston, be sure you assemble them the right way round. The arrow on the top of this T.R.W. forged piston indicates the front of the engine. Note that the rod is held in the vice with the oil hole in the beam of the rod upwards.



7-42. When you install the rod and piston assembly into the block, treat it for what it is, namely a delicate operation.



7-43. Another for the Hall of Horrors. In spite of a rod letting go at 9.500 rpm, this crank stayed in one piece although it severely mutilated the journal as can be seen here.

7-44. Before installing the bearings in either the main bearings journals or the rods, check the bearing thicknesses. The easiest way to do this is with a roller, not a ball and a micrometer. In this shot I am using a centre drill, as it was convenient. When the bearings are installed in the rods, use the thicker of any pair of bearings in the top half of the rod. In the main bearings, reverse the situation. Use the thicker bearing in the cap half of the main bearing.

FIG. 7-5

STANDARD WET SUMP LUBRICATION

	APPLICATION	R.P.M RANGE	CLEARANCE RODS	CLEARANCE MAINS
NORMALLY ASPIRATED	Street Rally/off road Race Drag Race	7,500 8,000 9,000 10,000 +	Top limit standard 0.0025" - 0.003" 0.0028" - 0.0032" 0.0035" - 0.004"	Top limit standard 0.0025" - 0.003" 0.003" - 0.0035" 0.0035" - 0.004"
TURBOCHARGER	Street Rally/off road Race Drag Race	7,500 8,000 9,000 10,000 +	Top limit standard 0.003" - 0.0035" 0.0038" - 0.0043" 0.0050" - 0.0065"	Top limit standard 0.0025" - 0.003„ 0.004" - 0.005" 0.0045" - 0.0055"
DRY SUMP LUBRICATION				
NORMALLY ASPIRATED	Rally/off road Race Drag Race	8,000 9,000 10,000 +	Top limit standard 0.0025" - 0.003" 0.0030" - 0.0035"	Top limit standard 0.0025" - 0.003" 0.003" - 0.0035"

If the engine is turbocharged, add 0.0005" to 0.001" to the clearance figures shown immediatly above The pressure pump on dry sump lubrication systems needs to be of adequate capacity for the large bearing clearances involved.

bearings are much more able to withstand high loads if the radical clearance in the loaded half of the bearing if less than the unloaded half. This means you should measure all your bearings, then put the thickest bearing in the *rod half of the rod* or in the *cap half* of the *main bearings*. This gives better bearing life and enhanced mechanical reliability.

The crankshaft has what are known as rolled fillet radii. If you take a look at one of these cranks, you will see a generous radius between each of the journals and the cheek of the crankshaft web. This provides a compressive stress on the surface, which increases the crankshaft's resistance to fatigue. This is important if you are considering tuftriding the crank. This rolled fillet, you see, helps achieve the same thing that tuftriding achieves. And, in the case of the SOHC Ford crank, tuftriding does not extend the crank's fatigue life significantly over stock. Moreover, if any heat-treating is done on the crankshaft, the benficial effect on the rolled fillet radius is lost, because the stress is tempered out of it. Thus, it is a good idea, if tuftriding is done, to re-roll the fillet radius; this should give a measure of the best of both worlds.

Regrettably, you will know when your engine really starts to make power because it will begin shearing flywheel bolts. This starts when power exceeds about 160 or 170 in conjunction with a suspension system which really takes a bite on the road. If you intend building a circuit race car with engine power approaching the 180 mark, it is definitely desirable to take steps to prevent the flywheel coming off. For drag-racing, I suggest that anything more than 160 bhp requires extra flywheel security.

There are two ways of retaining the flywheel for arduous conditions. The first: drill out the existing bolt holes and install bigger, heavier bolts. The second: put a dowel between each bolt. If you are going for really high horsepower and high revs, the second method seems to be far more effective. Typically, installing a 5/16 inch dowel between each of the bolts allows power outputs up to about 400 or 500 to be used without having the flywheel part company from the crank.

STEEL CRANKSHAFTS

If you want the best possible crank, there is no substitute for a good, steel crank. Several companies will make custom-built cranks. Holbay, for example, can supply steel crankshafts for the 2000 engine, as can Race Engine Components. A unit such as this, should give you a good, reliable 10,000 rpm engine. So long as nothing ridiculous happens, such as a rod breaking or a piston breaks up, the crank should last indefinitely.

LONG-STROKE CRANKS

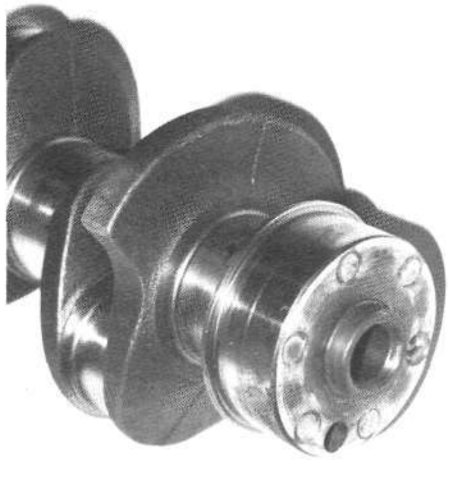
Speaking of steel crankshafts, I should point out that both previously mentioned companies can supply crankshafts with the stroke increased from 77 mm to 84.6 mm. This, in conjunction with the standard bore size, stretches the two-litre engine out to 2.2 litres. If you use this with a 1.5 mm oversize piston, displacement expands to 2264cc. The longer stroke will, of course, have to be compensated for as far as piston deck height is concerned, either by the piston having a lower compression height or a connecting rod of lesser centre-to-centre length. When making this choice, bear in mind that the five-inch centre-to-centre length of the standard rod is barely adequate for the 77 mm stroke. Therefore it's best to consider the shorter piston compression height to achieve the correct assembled deck height.

If you are going for all the capacity you can, then there is a limit to how much you can overbore the engine. It would appear that around .100 inch is about maximum. But even so, head gasket failures have been reported at this figure. A more moderate .080 inch (2 mm) oversize may be a happy compromise here. If gas-filled Wills O-rings are used to seal the head to the block instead of a conventional head gasket, the problem of head gasket failure is removed and the block may well withstand even further boring. Whatever the situation is, you will need special pistons. I suggest in U.S.A. you contact a company such as Arias for the pistons and Childs & Alberts for special rings to go with these special pistons. In England Omega Pistons can produce specials.

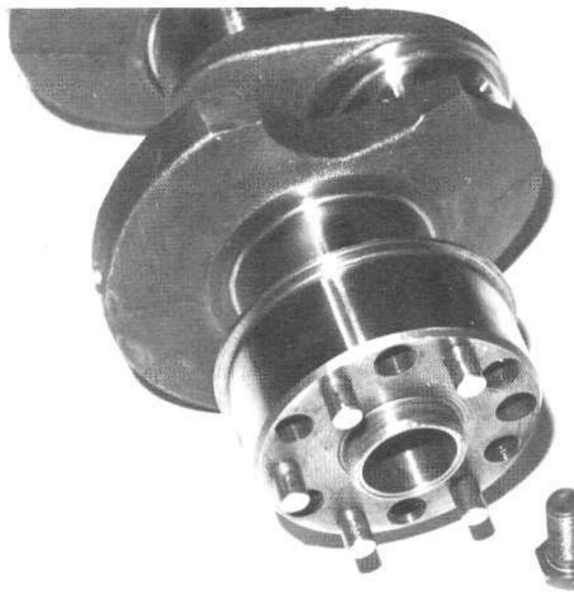
FLYWHEEL

I am not going to say too much about flywheels here except to reiterate that above a certain horsepower, they tend to part company with the crankshaft. The second point I want to make is that it is not advisable to spin a cast iron flywheel at very high rpm, especially if you are inclined to dump the clutch on it and induce further stresses. I would not use the cast iron flywheel on a drag race motor if it were to turn over 8000 rpm, even if the rules allowed it. If I were using it for a circuit race car, I would draw the line at 8500 rpm because a circuit car does not have the severe starting shock that a drag car has, so it has a little more safety margin.

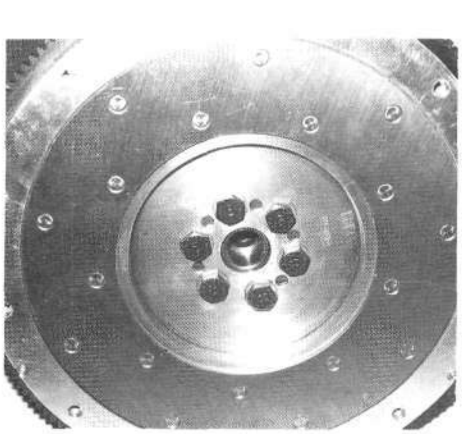
Many drag race cars use the weight of the flywheel to help launch the car. Unfortunately, once the car is rolling, that heavy flywheel is a disadvantage. If you are thinking of drag racing a car with a two-litre engine, weighing about 1800-2000 lbs., I suggest you look at rear end gearing very carefully. Figure on using a flywheel and clutch assembly around 35 lbs. You will need an overall starting gear ratio of about 22-23:1. This is assuming you are using tyres around 26 inches tall. To achieve this 22-23:1 overall ratio, you will probably have to find an axle ratio around 7:1 and run this with a first gear ratio of around 3.2:1. To launch the car, you will have to dump the clutch at around 8000 rpm. Ideally, a five-speed box is needed for drag racing with these engines. With this sort of



7-45. So you thought that those flywheel bolts were indestructible? Well here's a set of sheared off bolts in the crankshaft of an off road engine belonging to a friend of mine. As you can see, they have snapped off as clean as a carrot.

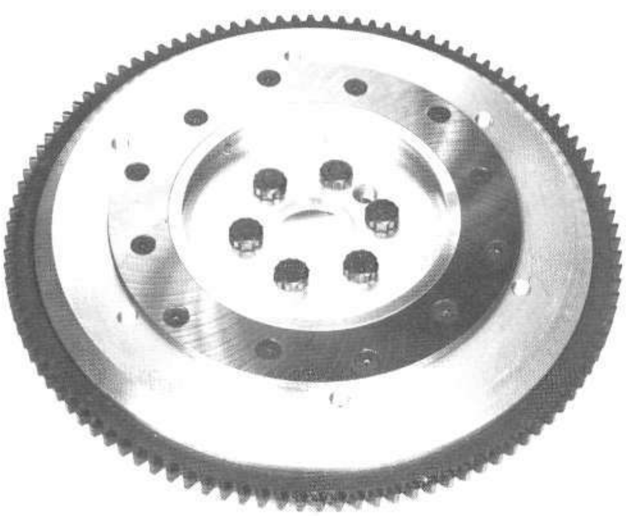


7-46. Here's the fix for flywheels parting company from crankshafts.
A 5/16" dowel between each bolt hole, that's the first reinforcement. Secondly each bolt hole in this instance has been drilled out and tapped to a larger scale size. The two bolts in the picture demonstrate the difference between the larger bolt used and the standard bolt.



7-47. Here is what the assembly looked like with a 30lb Hayes flywheel bolted up to it. This flywheel was used in a turbocharged, drag-race Pinto that much of the development work of this book was done on.

7-48. Here is an example of a steel crankshaft. This particular item is produced by Holbay Engineering. It's available in standard and long stroke. In conjunction with the standard bore, the long stroke version will give the 2-litre unit the capacity of 2.2 litres.



7-49. For road racing this Holbay flywheel has a lot to commend it. It's light weight makes it ideal for rapid gear changes etc. For drag racing you may need something heavier, especially if you have to launch a heavy car. In that case, such companies as Hayes, Schaeffer, etc., can produce flywheels to order.

gearing you should expect the car to run around 10,000 rpm in the lights. Of course, this is assuming you have an engine of 200+ bhp.

BALANCING

I am going to discuss only the rudiments of balancing here, for it is a field outside the scope of the home mechanic. Balancing, especially of the crankshaft, needs some fairly exotic equipment. Similarly, although it is possible to balance your own pistons and connecting rods, for the price you are likely to pay, it is hardly worth acquiring the necessary equipment.

When balancing an engine, the first chore is to ensure all the pistons weigh the same. Once the pistons' (with pins and rings) weights have been equalized, the next parts to balance are the connecting rods. To balance a connecting rod, all the rod journal ends must be weighed and adjusted to give the same weight, ditto all the pin ends. This means the rods are balanced so that all small ends match each other, all rod journal ends match each other and all overall weights match each other.

The next job is balancing the rotating parts of the engine. This is where a balancing machine is needed. To balance the crankshaft, it is spun up to speed on a set of rollers and the out-of-balance loads are electronically detected. The operator then removes material from the heavy spots to get the crankshaft to run perfectly balanced. After this, the flywheel is added to the relevant end. Then the out-of-balance effects of the flywheel are dealt with. Then the crankshaft pulley is added and the same process is repeated. Lastly, the clutch assembly is installed on the flywheel and the clutch assembly balanced. The balancing ensures the engine will run as smoothly as possible with the minimum amount of vibration. This is where its strong point lies. Unfortunately, it has often been quoted that balancing is worth 500 rpm in added crankshaft strength before breakage will occur. This is not so. More than likely the extra rpm realized through balancing is of the order of 5-10 rpm at most. This is not to decry balancing for it does have the effect of smoothing out the engine sufficiently well so that parts don't fall off the car quite as often.

Lubrication & Cooling

LUBRICATION & COOLING

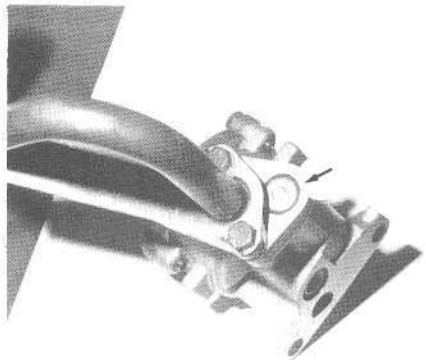
The very survival of the engine depends on how well it is lubricated and cooled, so this chapter, though short, is still very important. When working on either of these two systems, remember that overkill is far less likely to cause problems than underkill. With that in mind, let's start off by taking a look at the standard oil pump.

STANDARD OIL PUMP

Fortunately for us, the standard Ford oil pump has plenty of capacity and pressure capability. For all practical purpose, nothing need be done to the pump except to make sure that it is in perfect working order.

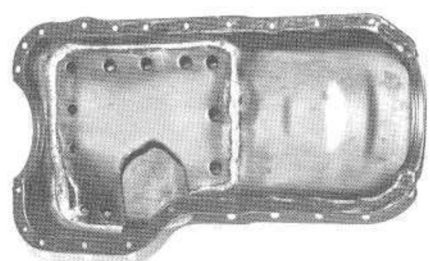
OIL PRESSURE

On most standard pumps the pump pressure release valve is set to give about 40 psi when the engine is hot. A few pumps, such as those fitted to the Capri are sprung on the pressure release valve to give a little more pressure than this. For most purposes, this oil pump pressure is entirely adequate. If you are running a race engine, you may like to raise the oil pressure a little by shimming the oil pump pressure release valve spring. About 60 psi is as high as you need go for hot oil pressure. When shimmed to give 60 psi hot, the oil pump will normally give about 80-100 psi when the engine is cold. If you are running a turbocharged engine, avoid excessive oil pressure. On a couple of occasions I have experienced oil escaping past the turbocharger oil seal when oil pressures reached 90psi or more. This can cause the engine to detonate due to oil lowering the octane value of the fuel. Whether the excess oil pressure going by the turbocharger oil seal was a fluke occurrence or not, I could not say, but since the problem is easy to avoid in the first place, it's best to do so. The other point to consider is the engine does not need such high oil pressure in the first place. Even in a turbocharged engine with a great deal of boost, 50-60 psi is as much as is needed.

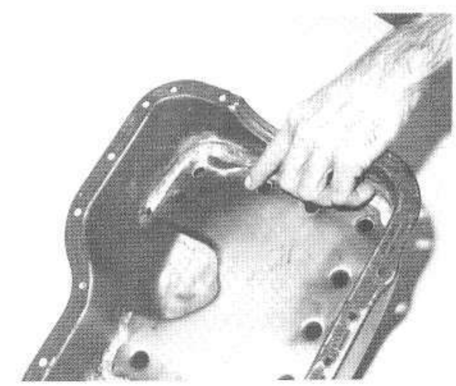


8-1. If a little more oil pressure is needed, the spring under the cap arrowed, can be shimmed.

8-2. Rather than spend time cranking the engine over on the starter on my own engines, I use the method shown here. This entails removing the distributor and driving the oil pump with an electric drill. When oil pressure has been achieved, everything is put back ready to run.



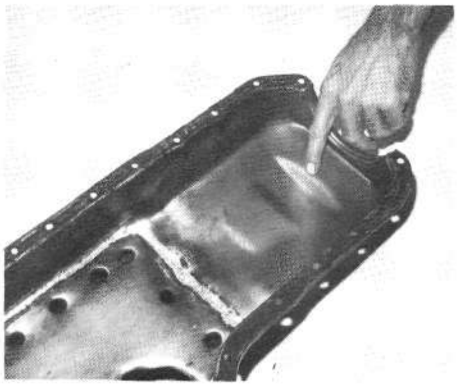
8-3. This baffled sump (pan) here shown overall, proved very successful on one of the test engines I worked on for a friend's off-road engine.



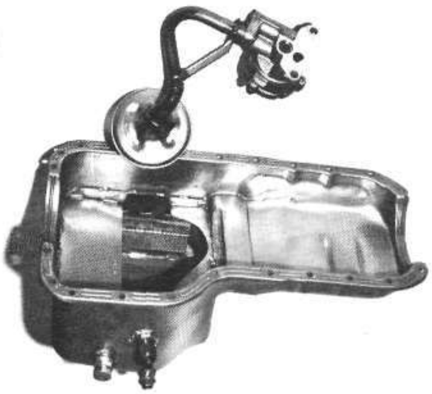
8-4. The part of the baffle indicated by the anonymous finger is angled off to give clearance to the pump.

SUMP BAFFLES

For almost any form of competition, from mild to wild, problems due to oil

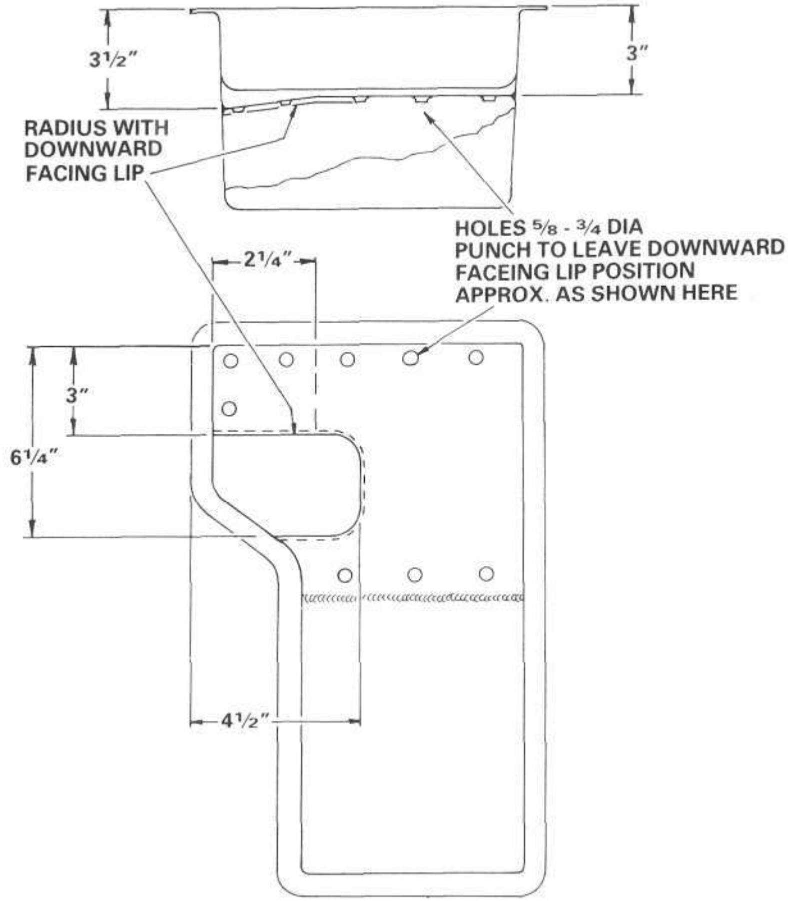


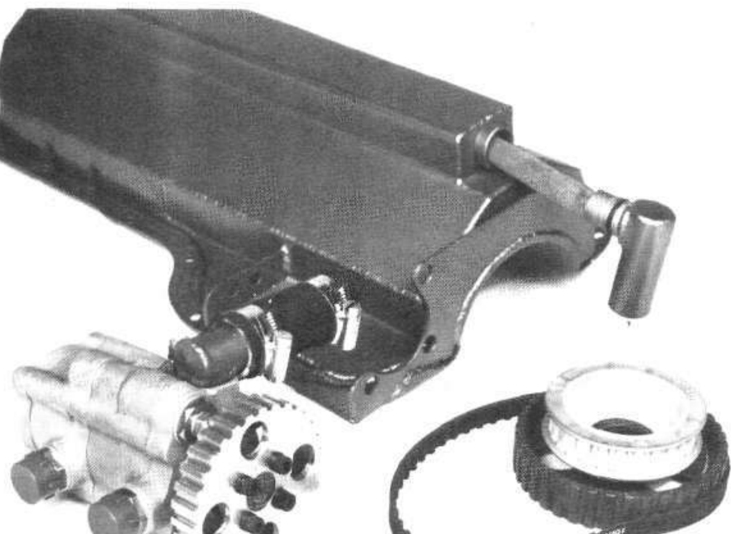
8-5. This valley is to give clearance to the big end. Be sure that the horizontal baffle is no higher than this.



8-6. Here is a baffled anti-surge sump made for a road racer by Shankle Automotive. This pump was used together with a slightly repositioned pick-up pipe.

FIG 8-1 TYPICAL BAFFLE SYSTEM FOR WET SUMP





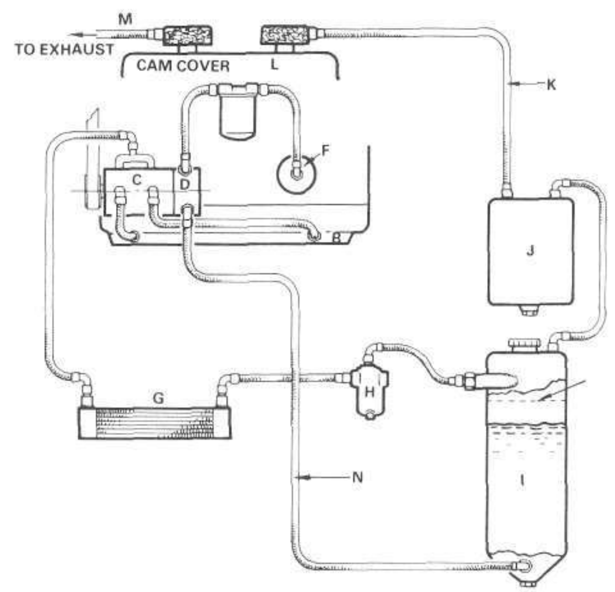
8-7. This particular dry sump unit is from Holbay Engineering. In the U.S.A. similar systems can be acquired from David Bean Engineering of Santa Barbara, California. If large bearing clearances are used, go for pump capacities of 2.2 gallons/hour/1000 rpm rather than the eariler type of only 1.35.

surge may crop up. About the only type of vehicle that isn't going to get oil surge will be moderately modified, street-driven machine having normal street tyres. Such a vehicle is not likely to be able to generate sufficient Gs to cause oil surge. Almost all other categories of vehicle can run into this problem, and for the moderate expense and effort it takes to baffle the sump, it is well worth doing, as it can easily save an expensive competition engine. Vehicles which come into the category of requiring sump baffling are those used for off-road racing, rallying, circuit racing, hill climbing, sprinting, drag racing and auto tests, or slaloms as they are known in the U.S.A.

The easiest way to obtain a baffled sump, although it will cost, is to buy the special aluminium sump for the RS 2000 engine from Ford. It may not be usable in every application, so the next option is to baffle the sump you already have. Fig. 8-1. shows a typical baffle system for the sump of these engines. One of the mistakes people often make when baffling a sump is to install the baffles vertically. The problem here is that the oil easily flows over the top of the baffle and after a few moments, the effect of the baffle is completely lost. By installing a horizontal baffle just above the normal oil level, oil is prevented from climbing the side of the sump.

If you want the ultimate in lubrications systems, a dry sump design is the way to go. It can allow the engine to deliver more power due to reduced oil splash in the sump itself, while also eliminating the possiblity of engine damage due to oil surge. Moreover, the dry sump is important to the circuit racer because it allows the engine to be set lower in the chassis, thus lowering the centre of gravity of the car. As far as power advantages are concerned, there do appear to be some with a well-designed system. I have heard increases claimed as high as six bhp. My own limited experience of dyno testing dry sump systems indicates that two bhp with a well-designed system may be a realistic figure. There are a number of good dry sump systems on the market, and for what they cost, I would advise anyone to buy the complete kit. There is very little the amateur can make without spending a great deal of time which could usefully be used in other areas. While is may be practical to make your own sump, it's a minor part of the entire system. A possible exception to this is the tank itself. It appears that if problems occur with dry sump systems, they are often connected with the design and placement of the tank. If you intend to make your own tank, there are two things you should consider. First, the tank must be capable of de-foaming the oil effectively. Second (and this applies when racing in the colder parts of the year) the oil tank must have big lines with as short a run as possible to the engine. My experience on dry sump units has been that racing in cold weather causes oil to flow so slowly through the pipes from the tank that it can take a considerable time to see even a few pounds of oil pressure on the gauge. It thus was necessary to install very large oil lines to overcome this problem. Here, synthetic oils would be of great assistance, but we will talk about that later.

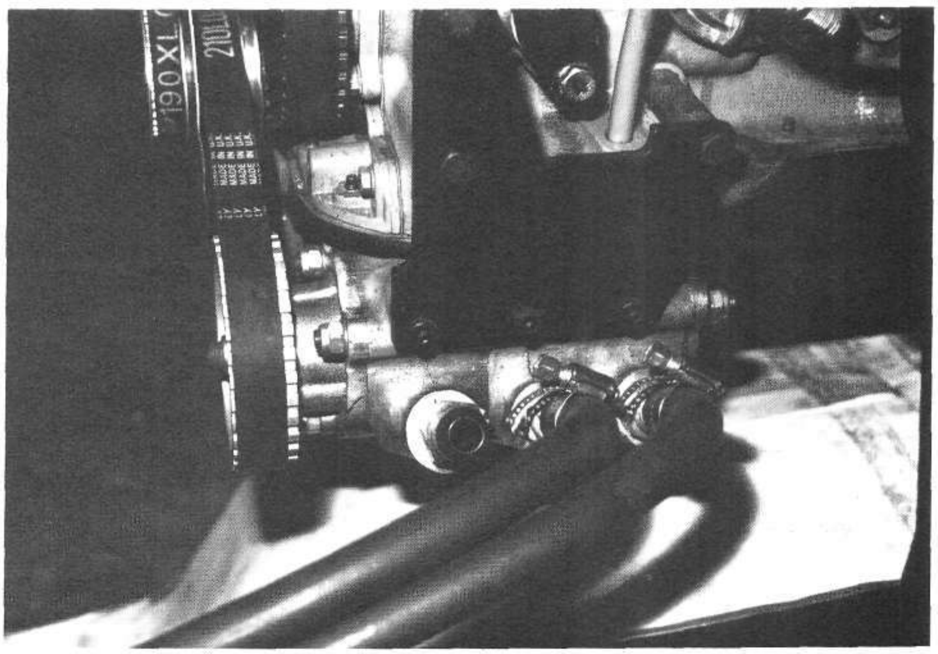
FIG 8-2 **EVACUSUMP SYSTEM**



A. SUMP
B. SCAVANGE PICKUP PIPES
C. DOUBLE SCAVANGE PUMP
D. PRESSURE PUMP
E. REMOTE FILTER
F. BLOCK UNION
G. OIL COOLER
H. SCREEN FILTER
I. RESERVIOR TANK
J. OVERFLOW TANK
K. OVERFLOW VENT
L. OIL/AIR SEPERATOR
M. OIL/AIR SEPORATOR TO EVACUSUMP SYSTEM
N. PRESSURE PUMP PICKUP PIPE.

To ensure the most effective operation of the dry sump system shown here, a few additional points will be of use. First of all the scavenge pickup points in the sump B should have a screen mesh to stop engine debris going into the scavenge side of the pump. The oil cooler can be put on the pressure side of the pump. This has the advantage that the oil in the reservoir tank will be hotter and will de-airate slightly quicker. A slight disadvantage is that the pump has to push the oil through the cooler and as a result, a small pressure loss will occur. The filter H is there to remove debris large enough to damage the pressure pump. It does not have the fine filtering ability that is required of a normal oil filter.

The plumbing of the overflow tank J is critical. If an evacusump system is used, the air space in the overflow tank *and* the air space in the reservoir tank must be at crank case pressure. If this can't be done, the higher pressure at 3 can force foaming oil into the engine. If an evacusump system is not used, line K should be vented to the atmosphere. To prevent dirt getting into the system, a small air filter should be used on the vent.



8-8. One of my test engines used a dry sump kit from Burton Engineering. This shot shows how the pump is mounted on the block.

Let us go back, for a moment, to the problem of de-foaming the oil. To do this, direct the oil return pipe to the tank so oil flows around the diameter of the tank, thus centrifuging air from the oil. Then have the oil drop through a very fine wire screen; this breaks up oil bubbles and further aids de-foaming. Make the tank tall; it height should be at least 2½ but preferably 3½ times its diameter. Figure on a minimum of 1½ gallons total system capacity, with as much as two gallons to be on the safe side. An oiling system of insufficient capacity is more susceptible to foaming problems than one with a larger capacity. The longer the oil spends in the tank allowing air bubbles to rise out of it – the better. Fig. 8-2, shows how the dry sump system should be plumbed. This should give you a good guide, whatever type of vehicle you are using it in.

Before we leave this topic, there is one point I want to emphasize: be sure the pressure pump is compatible with your engine's bearing clearances.

Some dry sump systems do not have enough volume on the pressure side to cope with large bearing clearances. If there is any doubt, check with the manufacturer of the system.

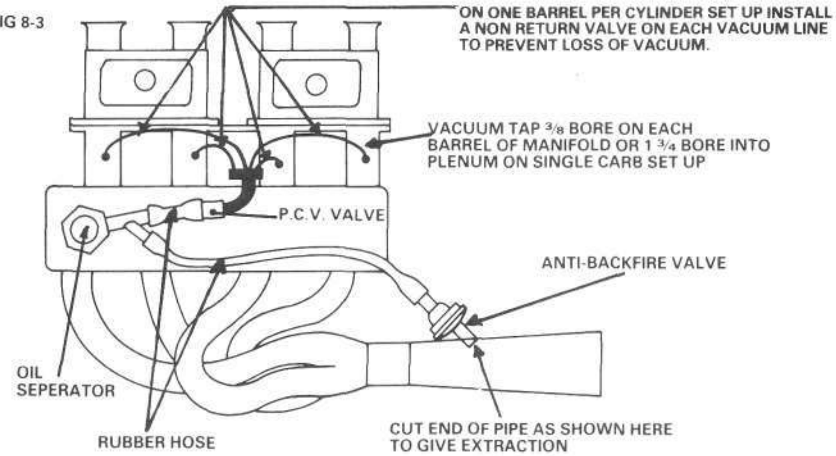
PRESSURE-BALANCED LUBRICATION SYSTEM

This practice is relatively new and, as far as I know, it originated in the U.S.A. A pressure-balanced lubrication system is a means of reducing the air pressure in the crankcase so the rings and valve seals have less tendency to pass oil up into the combustion chambers. With air pressure reduced in the crankcase, the reverse tends to happen. The higher pressure gases in the ports and combustion chambers tend to come down by the rings or seals and end up in the crankcase. Now at first this may seem to be the wrong way of doing things but it does have several advantages.

Consider: in order to get these engines to run strongly we need very high compression ratios. Oil contamination of the charge reduces the octane rating of the fuel. This, of course, can lead to detonation. However, with the pressure-balanced system, oil consumption and contamination are markedly reduced. It further allows low-tension oil rings to be used and this helps engine power. For a dry sump engine, the pressure-balanced system allows the oil to de-foam much quicker, so all in all, it has numerous advantages.

Here's how the pressure-balanced system works: at low rpm, when considerable amounts of manifold vacuum exist, the pressure in the crankcase is reduced by the engine drawing the fumes and air out via a PCV valve. With vacuum in the manifold, our pressure-balanced system is working as designed. However, opening the throttles reduces the manifold vacuum, one hopes, to zero. Under these conditions, an exhaust extraction effect occurs. A pipe connected to the crankcase of the engine is ducted to an anti-backfire valve which is installed in the tailpipe of the exhaust system. The exit of high-speed gases past the anti-backfire valve causes a low pressure to be generated. Thus, crankcase fumes are sucked out of the crankcase and expelled through the exhaust pipe. In a nutshell, when manifold vacuum extraction leaves off, exhaust extractions takes over. Apart from the lubrication benefits, there is a slight power increase with this system. It is possible to pull as much as 11-15 inches of water in the sump during full-throttle use. To see how the system is connected to various points of the engine, refer to Fig. 8-3.

FIG 8-3

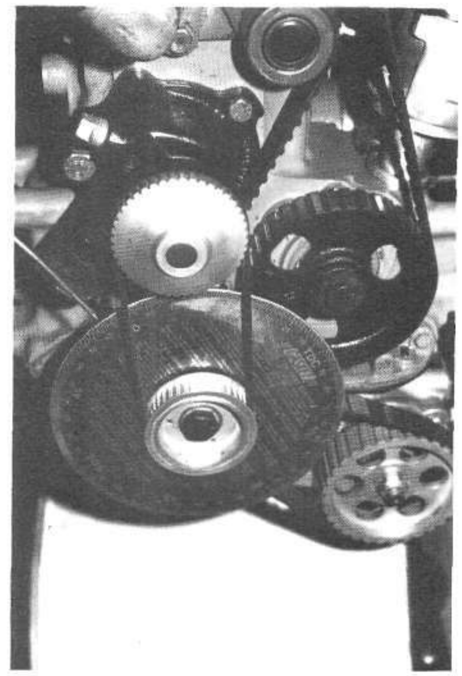


Now for the areas where you can and cannot use this pressure balance system. It can be used in a well baffled wet sump system as well as a dry sump system. It cannot be used where a silencer is installed in the exhaust system, unless the silencer presents no back pressure whatsoever. It is possible to connect an exhaust evacuation line to the exhaust system after the silencer but here the exhaust gases have cooled off and the velocities are much slower than they are closer to the engine as would be the case with an open exhaust pipe. If it is connected after the silencer, much of the benefit of this system may be lost. If the system is required for use with a turbocharged engine, the line from the crankcase must be coupled to the exhaust pipe downstream of the turbo. If it is connected upstream of the turbo, the anti-backfire valve will close to prevent the high pressure exhaust going into the crankcase, and of course, nothing will happen except you may not have adequate crankcase ventilation.

This brings us to crankcase ventilation. On very high-rpm turbocharged engines, ventilation of the crankcase becomes super-important. Unless there is space for plenty of air to move in and out of the crankcase, the engine is likely to blow an oil mist out, covering the engine compartment with oil. It will also drop oil on the track and your fellow competitors will not thank you for this.

OIL TEMPERATURE

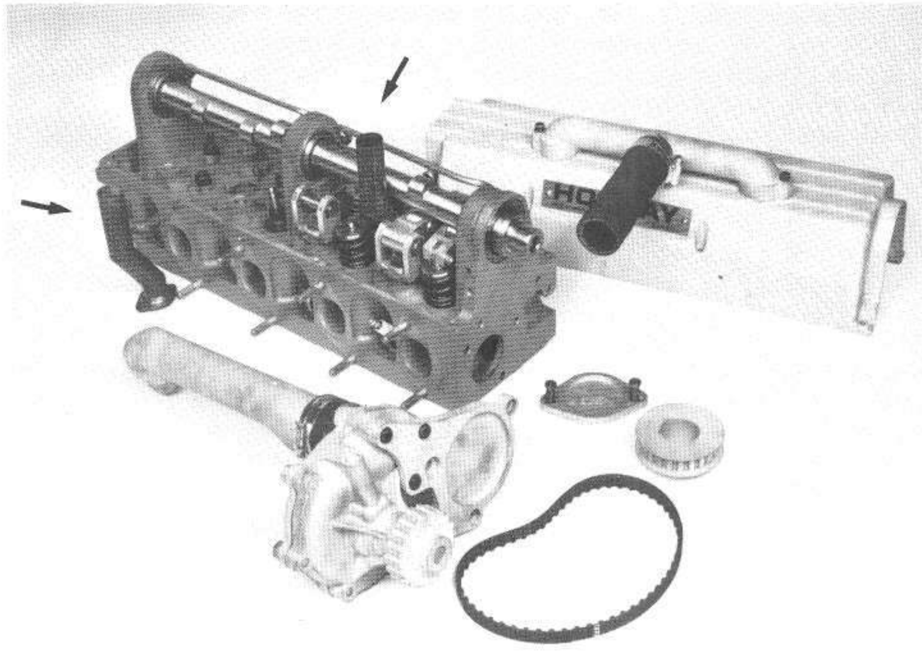
For a high-performance engine, especially one that is to see arduous use over prolonged periods, oil temperatures should be kept around the 95-degree C mark. Lower than this, and the viscosity is a little too high; higher than this and the oil film can break down and lead to rapid wear. Just about all race engines having any sort of distance to go will require oil coolers. With drag race machines, the distance is too short to worry about. Anyway, with this kind of racing you usually break something long before you wear it out.



8-9. Here's how the water pump drive looks on a typical dry sumped engine. On this engine, a cam degree plate was adapted to the crank so that cam and ignition changes could be easily measured.

Most of the heat picked up by the oil is from the underside of the pistons. And, as I stated earlier, pistons which have heat-resistant coatings on tend to transfer far less heat to the oil than uncoated pistons. If you have coated pistons, it is unlikely, in many instances, that an oil cooler will be needed.

There are several companies marketing devices which act as oil temperature thermostats. By wary about using these. Many of them will not flow enough oil to keep up with the demands of a race engine. They may be perfectly adequate for street engines which never turn over 5000 rpm and even when they are, it's only for a short period. With race engines running as high as 9000 rpm they may quickly lose control of the situation; although your oil may be at the right temperature, its flow might be inadequate. If you find that an oilstat has inadequate flow, try coupling two in parallel, so the oil can go through two oilstats rather than one. Apart from flow considerations, an oilstat is a good idea, as it allows quick warm-up of the oil and prevents over-cooling in cooler climates.



8-10. Here you see the extensive redesign job that Holbay did on the water pump and block water routing to give even temperatures in the water jacket. The original pump is dispensed with as a cog belt-driven side mounted pump is used. This feeds water into the two core (freeze) plug holes in the side of the block. The water moves up through the head out of the two stand pipes arrowed, and into the water manifold on top of the cam cover. From here it's routed back to the radiator.

TYPES OF OIL

Almost every motorist has his favourite brand of oil. Racers in particular can be brand-conscious, although they are ready to change in an instant if there is some clear-cut advantage to another type of oil, or if they experience some misadventure which can be remotely connected with the type of oil used. When things go wrong, it is easy to blame something as simple as the oil. Almost all the top brands of oil today are extremely good. Deciding whether one is better than another can be a very difficult task. Moreover, if a component fails on one oil, it will more than likely fail on another, unless the particular problem can be cured by a special brew of oil. For example, a lubrication problem exists in the Pinto valve train that can be cured by the addition of certain elements in the oil. These days, the question I am most often asked is, are synthetic oils any better for race engines? I have to admit that I have not tried them, though the French Motul oil V300 and V2100 look good (sold by T.W.R. in England). The principal reason behind my own cautious approach being that the Pinto's cam problem may be aggravated by the use of some synthetic oils. I use the word "may" here because caution must be struck on such a judgment. When you find something that works you tend to stick with it. Certainly the oil mixes I have been using in Pinto engines appear to have minimized rapid cam wear, even on cams with lifts over 0.600 inch. If a roller lifter assembly is used, thus sidestepping the cam lubrication problem, synthetic oils would seem to have numerous advantages. Among them, excellent temperature stability. From test figures presented in various SAE papers, it would appear that these oils are much more capable of sustained high temperatures without degradation, as well as being much more fluid at low temperatures. The lower viscosity at lower temperatures means that the engine oil temperature is not so critical and if it did drop on the low side during a race on a cold day, it would mean little in the way of lost power due to extra oil drag. All I can say at this stage in the mineral-versus-synthetic oil argument, is that if you have roller cams, then synthetic oil should theoretically be your best bet. If you are using a conventional cam and followers, mineral oil with General Motors Super Engine Oil Supplement (E.O.S.) or Ford Oil Conditioner, seem to work well.

To be perfectly frank, after having struggled this far to get an oil combination that works, I think I will sit back and let somebody else discover whether or not synthetic oils will do just as effective a job.

COOLING

Let's discuss the temperatures you should run your engine at. If you have an engine built for long distance racing – actually anything but drag racing – figure on running the engine at around 70 degrees centigrade (160°F). Anything above that temperature will cause power to drop off. If you are running with alcohol fuel, such as for an oval track midget, you may want to run the engine just a little hotter. This is because alcohol likes to have a fairly hot combustion chamber, but has very pronounced cooling properties. Thus, a little extra heat in the chamber tends to work out better. If you have a drag race engine, try this technique: start the engine and warm it so that oil and water temperatures are over 80 degrees C. Next, drain off the water, put in cold water and push the car to the start line. Do the tyre burnout and try to gauge things such that when you leave the start line engine temperature is 70-75 degrees C. with oil optimally around 80 degrees C.

According to tests done at Holbay Engineering, the Ford SOHC engine encounters cooling problems past the 160 bhp mark. John Reid of Holbay Racing Engines commented that "The standard arrangement gives rise to localized overheating on the exhaust side of the combustion chamber. Number four cylinder overheats, but this can be overcome by rearranging the controlling gasket holes. Unfortunately, this then aggravates the first problem. The overheating causes the usual combustion maladies and early head gasket failure." To overcome this problem, Holbay has designed a cooling system which mounts an all-new water pump low on the block. Water leaving the pump enters a log-type manifold along the side of the block. From this manifold, water enters the two freeze plug holes in the side of the block, then circulates in an upward direction toward the cylinder head. On engines utilizing Holbay cam covers, extra cooling can be given to the head. This is done by removing the two large socket screws between the cam towers of the head and installing a pipe that comes up from the head and through the cam cover. Water from here is then taken and fed back to the radiator.

My own experience of this overheating problem is limited. I must confess, it is a problem I have not been looking for. On the other hand, the number of engines I have built developing over 160 bhp is limited. If the problem is as severe as Holbay indicates, it would appear that ceramic coating of the exhaust port combustion chambers should rectify the problem, by preventing the heat getting into the cooling system in the first place. But while I feel this is fine for a normally aspirated engine, a long-distance turbocharged engine may encounter cooling problems, due to the time available for heat to soak through the ceramics. My suggestion here is to employ overkill. Coat the piston, combustion chambers, valves and exhaust ports with ceramic insulation material and then utilize the Holbay revised cooling systems.

AUXILIARY COOLING EQUIPMENT

Let's talk about equipment used to dissipate the heat that the engine accumulates. First of all, never skimp on a radiator. If you are designing a car from scratch, always start off with a radiator that is *too* big for the job. If you find it overcools the engine, reduce its size. Secondly, don't let the engine generate any more heat than is necessary. This may seem a silly thing to say, but such things as mis-timed cams and mistimed ignition can cause the engine to make more heat than it should without necessarily producing an equivalent amount of power. As far as coolant is concerned, use anti-freeze in the race car's system. The anti-freeze increases the boiling point of the water. It acts as a safety margin just in case you should have some mishap. On the other hand, the specific heat of the coolant with anti-freeze in it is lower. But to compensate for this, anti-freeze will keep the cooling system clean, so although it might not dissipate all the heat that a pure water system will when new, down the road it will perform better by preventing the radiator and block from scaling up.

Talking of blocks, here is another aspect: it is a good idea to have a block thoroughly hot-tanked in a caustic solution if you are using a block which has seen previous use and has rust scale on it. That rust scale acts as a heat insulator.

The radiator cap is another factor you should be aware of. The higher the pressure the radiator cap blows off at, the higher the boiling point allowed in the cooling system. Radiator caps are now available up to 21 psi blow-off pressure. This raises the boiling point of the coolant by about 10 degrees C. Exactly how much it increases depends on the mix of anti-freeze and water you intend using. But with the right combination, you can raise your engine boiling point to about 115 degrees C. If you build the right system for your racing car, you will be able to run without a fan of any sort. For street applications, the situation may be a little different. Normally you won't have the heat dissipation problems on a street-driven motor simply because it's not as powerful. On the other hand, you may decide to dispense with the standard fan and use an electric fan. This is fine in cooler climates. In warmer climates, however, be sure you get a fan of adequate size. Many aftermarket fans are just not big enough for the job. My suggestion here is that you try something like a Honda Civic fan, as it has good airflow and may get the job done.

But if you have an overheating problem, do not expect an electric fan to cure it. The most likely fan to cool an engine that has serious overheating problems is an engine-driven one. In my experience, electric fans simply do not have enough power to pull enough air through the radiator.

Fuels & Water Injection

One of the first things to consider before any modifications are done on the engine is the fuel you intend to run in your engine. Low-octane fuels cost less than high, but high-octane fuels used together with the highest useful practical compression ratio will normally take you further down the road on a given amount of money. High- octane fuels also allow greater power but only if the compression ratio is raised to suit.

Two factors are causing the fuel we buy at the pumps to have lower than desirable octane values. First of all, tetraethyl lead, which is used to increase the octane valve of a fuel, poisons catalytic converters. Because of this, non-leaded fuel was introduced. Over the coming years, leaded fuels will be phased out.

The second factor was the energy crunch of 1973-4. Consequent studies have shown a need to blend fuels and develop cars which will give the most miles per gallon of crude oil. Refining high-octane fuel requires the use of higher refinery energy, less of the total crude input to come out of the refinery as automobile fuel. Studies have shown that 87-octane fuel (R + M/2) produces the best miles-per-barrel from the original crude oil. As a result, modern cars are tending towards lower compression ratios to make use of 87-octane fuel. It doesn't require much thought to realize that 87-octane fuel is way too low for a high performance engine. The situation in America has become acute as far as high-octane fuels are concerned. Nationwide surveys show that between 1971 and 1977 premium fuel dropped, on average, 1½ octane. Between 1977 and 1978 it dropped 1 more octane point. It could be that in five years, the highest octane fuel that can be bought in the U.S.A. will be approximately 90 octane. In Europe, the octane crunch hit over a very short time span. Over a period of about 3 years premium fuel vanished, leaving Europe in the same situation as U.S.A.

If we are to modify our engines to use high compression ratios, we will have to increase the octane of the fuel we feed to our engines. There are many courses open to us. Some of these courses may not be possible in England. Others may not be possible in the U.S.A., so you may have to make your choice based on locality as well as convenience.

OCTANE MEASUREMENT

Before we delve into all the ways and means of increasing the apparent octane value of our fuel, let's look at how octane is measured. For testing purposes, a single-cylinder, variable compression ratio engine is used. Tests performed with this engine are done in two ways: one method of testing results in the research octane number (RON). The other results in the motor octane of the fuel (MON). In most cases, the octane number quoted at the fuel pump will be the research octane number plus the motoring octane number divided by two. This will normally be written

$$\frac{R + M}{2}$$

FINDING THE RIGHT PUMP FUEL

Both in the U.S.A. and Europe 100 plus octane fuel has already slipped from the scene, so it is necessary to make use of the next best available fuel. But the octane values of fuels available at pumps vary dramatically. A survey showed that premium fuels could vary in octane from 95 to 91.5. A similar test with regular fuel showed that octane could vary from 93 octane down to 87 octane. Just because you are buying premium does not mean you are necessarily buying the highest octane available. What it does mean is that you can try various brands of fuel until you find one that your engine runs on without detonating. I suggest you refer to the graph in the cylinder head chapter for a rough idea of how much octane your fuel will need to avoid detonation in your engine.

Another point: if you live in the U.S.A. fuel brewed for sale in high-altitude regions tends to be of lower octane than fuel used on coastal areas. The reason is that with the more rarified atmosphere, it is more difficult to make an engine detonate and so a lower grade fuel is sold at the pumps. The moral here is, do not transport fuel from a high altitude to a low altitude; you may run into severe detonation problems.

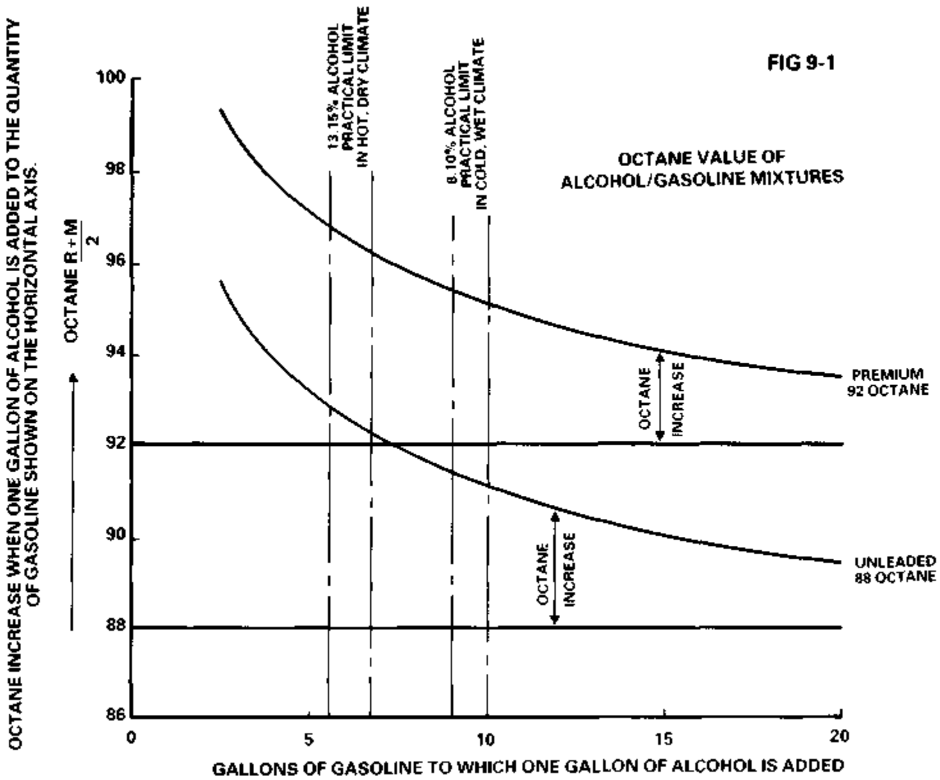
BLENDING YOUR OWN FUEL

If you cannot get enough octane out of the pump, then you will have to especially buy or blend your own fuel mixture. High-octane fuel is made at the refineries by adding tetraethyl lead to the basic pertoleum mixture. Tetraethyl lead is highly toxic and as a result, it is difficult to obtain. My advice here is, don't even try to get any. This is the one route to high octane fuel that you should avoid like the plague.

A far safer alternative is to buy super-high octane racing fuel such as Daeco, H & H or VP. Most of the fuels available from these companies have an R + M octane number ranging from 102 to about 118 or so. Dyno tests performed at Esslinger Engineering indicate that a normally aspirated, two-litre engine seems to produce best power on Daeco fuel. Buddy Ingersol, who, at the time of writing, ran the world's fastest gas-burning turbo Pinto, used the very highest octane brew of H&H racing fuel in his car. This allowed his engine to run almost 28 psi boost with a compression ratio of about 7.5/1 with no detonation problems.

Without a doubt, racing fuel is expensive and that trick 10.5:1 road-driven engine of yours may not need more that 98 or so octane to run without detonation. Remember, your engine only needs a fraction more octane value than the first octane value that won't detonate. Excessively high-octane fuel *does not help power output* in any way, so there is no point spending money on unnecessary octane. Bearing this in mind, a very practical way to boost the octane of service station pump fuel is to mix it with super-high octane racing fuel. By mixing in suitable ratios, you can come up with a blend which will be detonation-free in your engine. When blending your own fuel in this manner, you should consider the convenience, or lack of it, and the cost. If you live close to a bulk dealer of racing fuel, the solution is practical in both respects. The way to figure out cost effectiveness is to look at how much extra octane over and above your basic pump fuel you are buying. To do this, subtract the octane value of the basic pump fuel from the octane value of the racing fuel. Then subtract the cost of the pump fuel from the racing fuel. Now divide the difference in cost into the difference in octane. The answer tells you how many octane extra you buy per pound or dollar spent. If you have a choice of racing fuels in your area, go for the one that gives you the most extra octane. As far as the octane achieved by blending pump and racing fuel together is concerned, this is an easily calculable figure. To work out this value, use the following formula: (gallons of pump fuel × octane number) + (gallons of racing fuel × octane number) ÷ total gallons of fuel = octane of mixture.

An example: say we have six gallons of 93-octane pump fuel and two gallons of 105.8-octane racing fuel (Daeco), what would be the octane rating of the mixture? Using the formula we have (6 × 93) + (2 × 105.8) ÷ 8 = (558 + 211.6) ÷ 8 = 96.2. The mixture is now 96.2 octane as opposed to 93 octane of the original pump fuel. In fact these figures are conservative as the "blending value" of racing fuel, used in this way is often a lot higher than its bulk measured value.



OCTANE-BOOSTING ADDITIVES

While blending high-and low-octane fuel may get the job done, it can be inconvenient. Also, the hassle of hauling around high-octane fuel can be a problem for many. Certainly you should never store the extra fuel in the boot (trunk) of your car: a true fire hazard. An alternative to fuel mixing is to use one of the octane-boosting additives on the market. Some examples are made by Aldon Automotive, Vortex, H & H, Atlantic Coast Engineering and Moroso. According to Dr. Dean Hill, consultant to H & H racing gasoline, most octane boosters consist mainly of analine. It has long been known that this substance will raise the octane value of fuel, but unless it's purified to a high level, it causes a very obnoxious smell in the exhaust. The additives we discuss here will typically boost the octane of a typical premium grade. U.S. fuel by 3 to 4 points when used in the recommended ratios. One of the advantages of these octane boosters is the small amount of additive needed to give the desired results. Using U.S. measures, a one-quart can is often sufficient to treat 20 gallons of fuel. On a cost-per-gallon basis, the additive method works out about the same to quite a bit cheaper (depending upon booster cost) than is achieved by mixing racing fuel and ordinary pump fuel.

ALCOHOL BLENDS

Alcohol fuels have much to commend them. They run cool and can withstand tremendously high compression ratios without detonating. Further, alcohol can be mixed to our base fuel to increase its octane rating. Essentially the two types of alcohol that are commonly available are methanol and ethanol. The most commonly used one, methanol, is, in terms of performance, more advantageous to use than ethanol, but it is not without its snags, so we will deal with methanol first.

As far as its ability to increase the octane is concerned, see graph Fig. 9-1. The curve on this graph represents the octane increase by adding one gallon of methanol to the base-line gallonage shown on the graph. In other words, point A is the octane figure you would get if you added one gallon of methanol to 35 gallons of 90-octane fuel. As you can see, the line gets steeper and steeper as the methanol content becomes greater. As far as boosting octane goes, methanol gets the job done. Now for the snags: first of all, methanol does not contain the heat energy that petroleum distillates do (that's petrol or gasoline, depending on which side of the Atlantic you live). This means that although you may have raised the octane value of your fuel, you will have lowered its ability to produce horsepower. However, this can be more than offset by the higher energy yielded by a correct mixture charge and the high compression ratios that can be used. So at this point we are still on the winning side with methanol.

Another disadvantage of adding methanol to the fuel is that above a certain proportion methanol tends not to mix well with the base fuel. Under typical conditions, about 15 percent methanol-to-fuel ratio is about as high as you should try to go. At temperatures around 40 degrees F or less, all but a small percentage of methanol will separate from the base fuel. All factors considered, 10 percent tends to be a practical percentage of methanol to use under most conditions.

The next problem that will be experienced with methanol petrol/gasoline mixtures is the effect methanol has on the air/fuel mixture ratio. The correct fuel/air ratio for petrol/gasoline for complete combustion is around 15:1, richer for maximum power. The correct fuel/air ratio for methanol for complete combustion is around 6.5:1, maybe 5:1 for best power. With petrol/gasoline jetting, the more methanol you add to your fuel the leaner your fuel/air ratio will become. With small amounts of methanol, little or no jetting changes are necessary, but as percentages become larger, some re-jetting may be required. As a rough rule of thumb, figure that the area of the jet needs to go up by one percent for every one percent of methanol used in the base fuel. As an example, if you have a pair of Webers jetted correctly for straight pump fuel with 150 mainjets, this engine would typically require 157 mainjets if a pump fuel and 10 percent methanol mix were used.

If you live in a wet or damp climate, methanol may cause a problem: it tends to attract water. When methanol is mixed with fuel, and water is introduced to the mixture, a phase separation occurs. The fuel as a whole will become divided into three distinct levels. The top level of fuel is the one which contains a little alcohol dissolved in it. The second level contains a higher percentage of alcohol while at the bottom, the water which acted as the contaminant in the first place, resides. Dissolved in this water are the most water-soluble constituents of the mixture as a whole. A phase separation like this in your carburettor means that when you come to start the vehicle in the morning, you will be trying to start it mostly on water from the bottom of the float bowl instead of fuel. To an extent this phase separation can be overcome. If 2-or 5 percent acetone or methyl ethyl ketone are added, the problem can be substantially reduced in proportions.

One further problem, the last of any consequence, is that methanol can cause corrosion of many of the materials used in fuel systems. Any alloy which contains lead, zinc, magnesium or aluminium is, to a greater or lesser degree attacked by methanol. As it happens, most of these materials are used at one point or another in the construction of your fuel system. If used in strong enough concentrations, the methanol will, over a period of time, attack these materials and cause a build-up of corrosion in the fuel system. If any water gets into the fuel due to the presence of the alcohol, the corrosion rate seems to increase. Fortunately with low concentrations of methanol, the corrosion problem does not seem to be too severe. Methanol concentrations up to 10 percent do not appear to have caused more than slight corrosion in fuel systems over periods as long as three years.

Again, the addition of acetone seems to reduce the effect of corrosion. This additive is very useful when used with methanol because not only does it make the methanol a more usable anti-detonant, but it also contributes to the octane rating of the fuel itself. This chemical is highly volatile and in cold climates it can assist starting of the engine. Methanol in cold climates can make cold starting more difficult. Also, methanol has a lower boiling point than conventional pump fuel. As a result, in hot climates it can cause vapour locks to occur easier than would otherwise be the case.

Now let's look at ethanol. This is closely related to methanol and achieves almost the same effect on boosting the octane. It is sometimes harder to get, possibly because it's roughly the same sort of alcohol that is found in drink. If you are to buy this type of alcohol, it has to be bought in its denatured form. As far as use for an additive goes, it has more to commend it than methanol simply because it does not have so many unwanted side-effects. First of all, it dissolves readily in petrol/gasoline. Secondly, it does not corrode the fuel system quite as quickly, and thirdly, it is not so prone to phase separation if a little water gets into the fuel system. Unfortunately it is more expensive than methanol in most places, so these extra little items of convenience will cost you more money. If you live in a cold climate, then, like methanol, it is best to add one or two percent of acetone to help eliminate temperature related problems.

AN A.D.I. SYSTEM

A.D.I. stands for anti-detonant injection and under this heading we will consider the feasibility of water injection and water/alcohol injection. Water injection has fascinated the motoring enthusiast for almost as long as the car has been in existence. Claims made of what water injection will do are many and varied, and the great majority bear no relation to fact. When considering water injection, bear this one very relevant fact in mind; water does not burn. If you don't believe me, call your local fire department. Possibly because it cools and doesn't burn, it has the ability to suppress detonation. Indeed, if you inject enough water into an engine, you can use really low-grade fuel on compression ratios as high as 17 or 18:1. That's the good side of water injection. The bad side is that you don't necessarily gain all the horsepower that you might expect from those compression ratios.

Let's look at this another way. Suppose you have an engine in which you have the ability to raise the compression ratio to any level. You check power HP output at each ratio until detonation sets in. Just as detonation occurs, you initiate water injection. As soon as the water is turned on, horsepower will drop slightly, so you will have to raise compression a little more to recover the power the engine was delivering prior to water injection. To get a substantial increase in power, it will be necessary to go one or two ratio points higher before any increase in power will be seen. In simple terms, this means that if your engine doesn't need water injection, using it will cost you power and *economy under full-throttle conditions.*

At this point, water injection doesn't look like a good bet. Well, so far we have not looked at the whole picture. In the cylinder head chapter, I pointed out that high compression ratios not only give good power output but they also increase *part-throttle economy* a substantial amount. Under part-throttle conditions, the engine is not likely to detonate simply because it is not completely filling the cylinder. So our high compression ratio under these conditions will give good part-throttle economy. Therefore we can achieve the economy benefits of high compression ratios while using water to suppress full-throttle detonation. It was also pointed out in the cylinder head chapter that high compression ratios are needed if long- duration cams are used because the valve overlap causes much of the incoming charge to be pushed back into the intake duct at low rpm. This leads to a "soggy" feel of the engine. By utilizing a high numerical compression ratio the cam can be made of work much more effectively. The water injection may not be helping peak power as such. Its effect may cancel out benefits of the high compression ratio *but the two used together will allow that long-duration high-lift cam to be used optimally.* The net result is that an engine properly set up for water injection may be able to produce more power. It does not score by just adding it to an engine which did not detonate previously. This is the big thing to remember with water injection. There are many companies marketing what they claim to be water injection units. Some even say they will make a quart of water last a thousand miles. To assume such a system is well designed because it is economical on water is a fallacy. Unless the water is going into the engine, it will not suppress detonation. Therefore a system which does not use water under full-throttle conditions is obviously not doing anything. The water in a nearby lake would be no more effective than the water under the hood/bonnet if it is not going into the engine.

One of the few systems which really does get the job done is the Spearco water injection system. I have used several of these units on my own vehicles and have found that compression ratios as high as 12.5:1 can be used with 87-90 octane fuel. The unit can be likened to a simple fuel injection unit, inasmuch as it can be calibrated to suit the needs of your engine. It's not necessary to go into detail on how to calibrate the unit here, as it comes with very comprehensive instructions; it is simplicity itself to install. The only factor I should point out, though, is that you must avoid injecting too much water into the system. This can cause steam corrosion of the exhaust valves as well as causing a drop in power. When calibrating the system use the minimum amount of water necessary to stop detonation. Any more than that and it will cost you performance.

WATER-ALCOHOL INJECTION

An alternative to straight water injection is water alcohol injection. Here we have the scope to make more power as well as suppress detonation. Many tests done over the years, especially in the

9-1. Here is a Spearco unit of the type I used on almost all my test engines. In three years' testing with numerous units, the only failure I experienced was the splitting of the reservoir tank.

9-2. On a four barrel with vacuum operated secondaries the water injection nozzle that I'm indicating with my finger dumps into the primary side of the carb.

aircraft industry during World War II, have shown that a mix of 50-50 water and methanol is very beneficial for power output and detonation suppression. In a sense, the injection of alcohol in a mixture with water is better than mixing the alcohol with fuel. The basic reason for this is that if the alcohol is mixed with the fuel, *you are using the increased octane value whether it is needed or not*, and at part-throttle it is not needed. When alcohol and water are mixed for injection to prevent detonation, the injection takes place only when it is needed to suppress that detonation. Therefore the alcohol is not used under part-throttle conditions when it is not needed. The use of alcohol and water instead of straight water can actually give small increases in HP. Another important factor is that the alcohol prevents the water from freezing during winter, which is important if you live in cold climates. If an engine is to be built and used with water-methanol injection, the compression ratio can be very high. Even with a short period cam, a ratio as high as 14:1 can be employed.

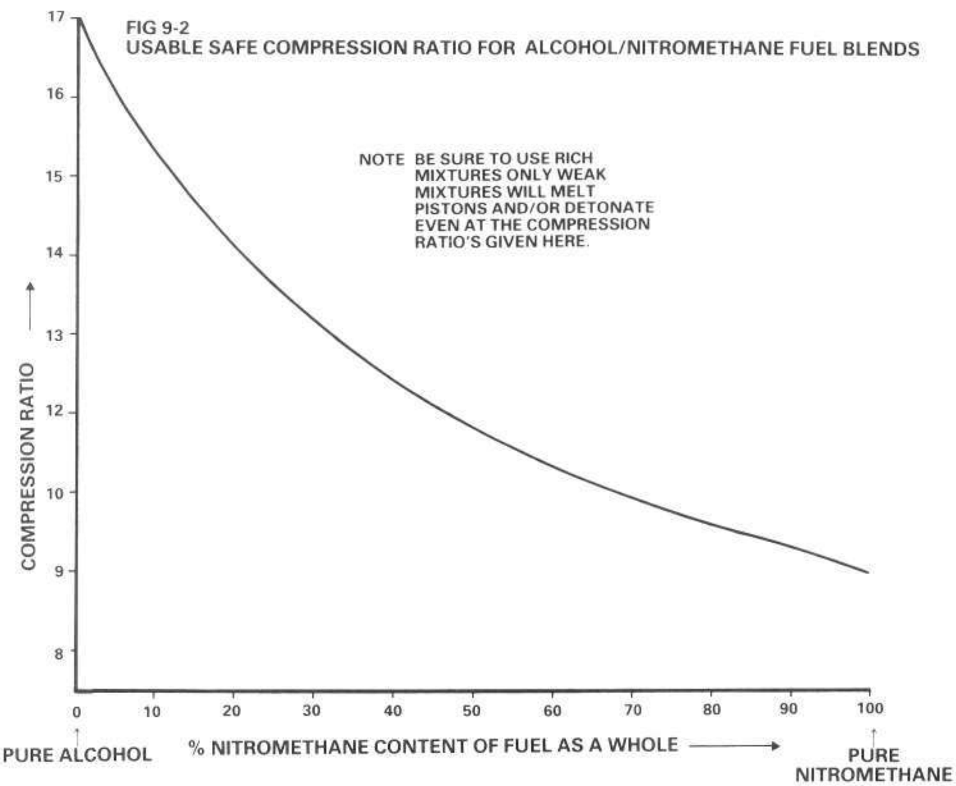
ALCOHOL & NITROMETHANE

In certain forms of racing, the regulations allow, and in some cases demand, the use of fuels other than petrol/gasoline. The most commonly used alternative fuel is alcohol. This may be either methanol or ethanol. The more common of the two is methanol simply because it produces higher heat energy when burned with a pound of air. This means, all other things being equal, it will make slightly more power. Apart from that, it's usually cheaper to buy, a factor which figures in anybody's race budget. Making the change from petrol/gasoline – type fuels to alcohol is not difficult so long as one basic factor is borne in mind. Earlier in this chapter I indicated that the general proportions of fuel to air are much richer for methanol than for petrol/gasoline. This means the alcohol fuel delivery system must cope with 2½ to 3 times as much fuel as it does when petrol/gasoline is used. Because of this many carburettors end up incorrectly calibrated. They physically cannot handle the needed quantity of alcohol to give the correct fuel/air ratio. This does not mean a carburettor cannot be used. But it does mean that a carburettor may need considerable modifications to successfully utilize alcohol fuel. It isn't just a question of opening up jets, because fuel passages in the carburettor body itself may not be large enough. Also, in many instances, the needle valve in the float bowl has insufficient flow. If the needle valve cannot pass enough fuel, no amount of jetting changes will rectify the lean mixture produced. Most 40 and 45 mm carburettors will prove marginal when used on alcohol engines of 200 bhp or so. If you expect an engine to produce more than this, consider the use of 46 or 48 IDA downdraft carburettors. These carburettors were built with large engines in mind and can flow much more fuel through their passageways. Of course, the IDA Webers are not usable without modification. But they offer much greater scope for modification.

It is because of the fuel flow problem that many alcohol-burning engines are used with fuel injection rather than carburettors. Fuel injection normally employed for alcohol-fuelled engines is simple in nature and is commonly known as a "speed-density" system. Calibration is relatively crude and the only reason it can be used successfully is that alcohol-fuelled engines can run with rich mixtures without losing noticeable amounts of power. This gives the injection system a wide margin of fuel/air ratios to work within. That same fuel injection run on petrol/gasoline would fare very badly because of this same lack of accurate calibration. Thus the use of an injection system greatly simplifies the use of this fuel. It also gives the user scope for adding other horsepower- producing elements into the basic fuel mixture. The most commonly used of these is nitromethane. Trying to introduce nitromethane into a carburetted, alcohol-burning engine is a very unsatisfactory route to go. If you think jet sizes are large when straight alcohol is used, jet sizes go out of sight when nitromethane is added to the brew. If you do intend running alcohol nitromethane mixes, be sure to start off with an injection system and save yourself a lot of trouble. Such systems are available from companies such as Esslinger Engineering and Racer Walsh and Hilborn Engineering. Their price is barely more than that of carburettors, so it doesn't really make sense to run anything else.

To get the best from an alcohol fuel, figure on compression ratioes between 15 and 17:1. But as nitromethane is added, the compression ratio must come down, because nitromethane is highly prone to detonation. A 15:1 alcohol engine can usually withstand 10 to 15 percent nitromethane if it's jetted suitably rich to start with, but past this compression must be reduced. Fig. 9-2

FIG 9-2
USABLE SAFE COMPRESSION RATIO FOR ALCOHOL/NITROMETHANE FUEL BLENDS



should give you an approximate idea of the ratio to use in any given brew.

When building an alcohol-burning engine, achieving suitably high cylinder pressure may be difficult if long-duration cams are used. Tests at Esslinger Engineering show that it pays off to shorten the cam timing. In this way, valve cutouts in the pistons can be made smaller and higher cylinder pressure achieved. Very often this means that an engine will produce, say, 200 bhp at 6500 rpm, rather than 200 bhp at 7000 rpm. In other words, the engine's torque output has increased but the shorter cam has limited top end power. To offset this, the higher compression ratio has increased bhp; the two have traded off. The net result is less rpm to produce the same power. If welded combustion chambers are used, the trade-off does not necessarily work in this fashion, because the required compression ratio is otherwise achieved. This makes it possible to utilize a slightly longer cam than is conventionally used in methanol motors and realize more power. A good fuel-injected, methanol-burning engine will, when everything is going for it, produce around 220 bhp at around 7000 rpm. An engine built on more of a budget but with the right components, can make a reliable 200 bhp at about 6500. When nitromethane is added to the methanol, the power increase is almost proportional to the percentage of nitromethane used. A 30 percent load of nitromethane will increase the power output by about 25 percent. This extra power shows up on the dynamometer as extra torque throughout the usable rpm range. In other words, adding nitromethane to the fuel gives the driver the impression that the engine has grown in displacement.

MIXING & USING ALCOHOL AND NITROMETHANE FUELS

If you have never used alcohol or nitromethane fuels, here are a few words of advice. First of all, most types of exotic fuels are poisonous. Do not let either of these fuels come into prolonged contact with the skin.

Do not concoct your fuel mixtures then store them for prolonged periods. Some components of the fuel evaporate faster than others. What you mix will not be necessarily what you run, if there is a long time period between the two. Always make a note of the jetting you use for any particular ratio of alcohol and nitromethane. This will avoid many jetting problems. If you run heavy nitro blends lean parts of the engine can burn out in a matter of seconds. Also, when high nitro percentages are used, ignition timing needs to be retarded from that used for petrol/gasoline. The more nitromethane used, the more retarded the ignition timing needs to be. Best power is achieved usually just prior to detonation. Try to approach this ignition setting from the safe side rather than from the detonation side.

The use of nitromethane makes for a very noisy exhaust because the fuel breaks down thermomechanically in two stages, and the second stage of its chemical breakdown occurs late, and usually happens in the exhaust pipe, hence the terrific increase in exhaust noise. As with any open-exhaust race engine, you will not hear detonation over the noise of the exhaust. If detonation occurs on a nitromethane engine, expect the pistons to last no more than a few seconds. Caution and some notes based on previous successfully used timing and jetting may save you an engine.

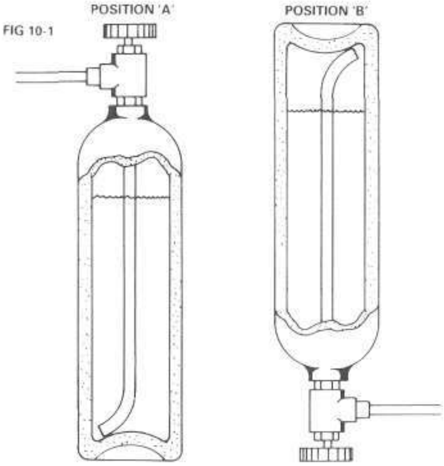
Nitrous Oxide

WHAT IS NITROUS OXIDE?

Nitrous oxide is a compound that has the ability to make you laugh (sometimes known as laughing gas) and put people to sleep (it can be used as an anaesthetic). However, neither of those qualities is of any concern to those of us delving into its performance potential. What concerns us most is that nitrous oxide (chemical symbol N_2O is made up of nitrogen and oxygen. The important part, its oxygen content, is by weight 36.3 percent of the whole. At normal temperatures N_20 in a free state is gas. If it is kept under sufficient pressure, it can be stored as a liquid even at normal temperatures. The pressure under which the N_2O is stored in liquid form is known as the vapour pressure. At 75 degrees F. the vapour pressure is 800 psi, and at 97 degrees the vapour pressure is up to 1034 psi. If you read a pressure gauge on a N_2O bottle, the pressure you read will be a result of the temperature, not the amount of N_2O the bottle contains. The pressure gauge will read the same pressure on a full bottle as it will one nearly empty. Not until the last vestiges of liquid N_2O are gone from the bottle, will the pressure gauge start to drop.

When nitrous oxide is released from the cylinder, it can come out in one of two forms, depending on the attitude of the bottle. Take a look at the drawing, Fig. 10-1. When the bottle is in position A, the pickup pipe is submerged in nitrous oxide liquid. Opening the valve causes the high pressure gas above the liquid to force the liquid out through the pickup pipe. In this instance the discharged nitrous is in the form of liquid. In drawing B, we have the situation where the pickup pipe is above the liquid, so opening the valve causes the discharge of gaseous N_2O. In either case the action of releasing N_2O causes the pressure in the cylinder to drop. The boiling point of a liquid is dependent on the pressure it is under, and liquid N_2O is no different from any other liquid in this respect. The resultant lower pressure due to releasing N_2O from the cylinder causes the liquid to boil and the vapour given off from boiling tries to re-establish the pressure in the cylinder back to the vapour pressure for the particular temperature the bottle is at.

FIG 10-1



THE POSITION OF THE N_20 BOTTLE DICTATES WHETHER GAS OR LIQUID IS INJECTED. POSITION 'A' PICKS UP LIQUID POSITION 'B' PICKS UP GAS

Two things need to be considered here. First, when N_2O boils, it absorbs energy. Secondly, when allowed to expand from the confines of the cylinder to open air N_2O becomes super-cooled. This in turn causes two things to happen: the storage tank temperature drops, typically by 10 to 15 degrees F. Also liquid nitrous when released through a jet almost instantly turns to a gas and drops in temperature down to as low as −128 degrees F. The tremendous drop in temperature can be very important from the horsepower point of view, as we shall see later.

How does nitrous oxide produce the dramatic power increases claimed of it? And why? Let's go back to an earlier statement. N_2O is 36.3 percent by weight oxygen. By comparison, the atmosphere we breathe is just short of 20 percent oxygen. The implication here is that if we were to run an engine in an atmosphere composed of N_2O it would produce proportionately more power, about 81 percent more in fact. Putting it another way, it would be equivalent to having 12 psi supercharger boost. Tests on aircraft engines have shown that engines run on N_20 gas, either partially or completely, produce just about the theoretically expected power increases. The only snag is that using N_2O in gaseous form for anything but relatively small power increase is inconvenient. Handling a large number of cubic feet of gas is difficult and presents its own set of problems. The logical step then is to inject liquid N_2O, as this has a ton of advantages over its use in gaseous form.

10-1. Here is a typical N_20 bottle. Seen alongside is an electric fuel pump. This pump supplies the fuel to go with the oxygen content of the N_20.

I will start the list of advantages for liquid N_2O with a really fancy term, namely the Displaced Charge Factor, or DCF for short. To explain what this means, let's go back to the all-gaseous N_2O situation. Remember, if we were to run an engine totally on N_2O gas, it would achieve an 81 percent power increase on 100 percent N_2O; the original atmospheric charge being completely replaced by the N_2O charge. This means the normal atmospheric charge is contributing 0 bhp to the engine's total output because the N_2O gas has completely displaced the original charge the engine normally gets from the atmosphere. In other words, the Displaced Charge Factor is 100 percent.

Now let us inject N_2O into the engine at exactly the same rate, but this time in liquid form. A pound of liquid N_2O only takes up 1/606 of the space that 1 lb. of gaseous N_2O does. This means that if the N_2O stays liquid right up until it's in the cylinder, the DCF is only 0.15 percent. In other words, the engine can go on producing 99.85 percent of the power it produces from a normally inhaled charge, plus the power the liquid N_2O is going to give. Putting numbers to it, injecting the same amount of N_2O in liquid form will give a power increase proportional to 181-0.15 percent 180.85 percent. In practice, the liquid won't stay as a liquid form very long; the moment it leaves the jet, it starts to turn into a gas. How much turns to a gas before it hits the cylinder depends on a number of things: how close the N_2O jet is to the intake valve, how hot the engine is and how many revs the engine is turning. All these things cause the N_2O to expand, thereby increasing the DCF. But there is an action taking place within the manifold which tries to reverse this. Remember, as the N_2O is discharged from the jet it becomes super-cooled (down to as low as -128 degrees F). This cooling action causes the normal atmospheric charge already in the manifold to shrink and become denser. This tends to increase the charge weight of normally aspirated air and thus *reduce* the DCF. Under normal conditions, the DCF will be between 3 and 15 percent for a well-designed system. Once in a while a system will, as a result of an educated guess rather than design, have everything at optimum, such as just the right amount of N_2O, the best angle of injection, the best distance from the valve, the optimum volume of N_2O, etc. Under these "ideal" conditions, it is actually possible to get a negative DCF. Experience with one particular engine on the dyno indicated that carburettor airflow went up about 10 percent when N_2O injection was taking place.

Having outlined the reasons, let me sum up the case for injecting liquid N_2O. Because of the small amount of space it takes up, substantial quantities of liquid N_2O can be injected into an engine to produce additional bhp. So long as distribution problems are not encountered, it is entirely possible to inject enough N_2O to achieve 200 bhp per litre, if no other limitations on power exist.

At this stage it all looks very promising. We can certainly get enough oxygen into the cylinder to produce big power numbers. But we also have to consider the "if no other limitations exist" criterion. Any action which involves burning is brought about by the chemical combination of two components, an oxidant and a fuel. The N_2O is the oxidant. It will not burn on its own, so fuel must be added to go with the N_2O content of the charge.

Using chemical equations, the chemically correct N_2O-to-fuel ratio can be calculated. Unfortunately, a typical petroleum-based fuel consists of a blend of numerous compounds, making the use of any chemical equations exceedingly difficult. Fortunately a typical fuel can be well represented by iso-octane whose chemical composition is C8 H18. This has a specific gravity of 0.72 and belongs to the paraffin hydrocarbon series.

The atoms of carbon and hydrogen will, when combined with oxygen, form carbon dioxide CO_2 and water H_2O so the balanced chemical equation looks like this:

$$\underset{\text{fuel}}{C_8H_{18}} + \underset{\text{nitrous oxide}}{25N_2O} \rightarrow 8CO_2 + 9H_2O + 25N_2$$

In other words, the hydrocarbon fuel combines with the oxygen content of the nitrous oxide and during combustion forms eight molecules of carbon dioxide and nine molecules of water, plus 25 diatomic molecules of nitrogen N_2.

The atomic weights of the compounds we are dealing with are:

Hydrogen 1
Carbon 12
Nitrogen 14
Oxygen 16.

Our nitrous to fuel ratio by weight = $\frac{\text{mass oxidant}}{\text{mass fuel}}$

This works out to be:

$$\frac{\overset{\text{nitrogen}}{(25 \times 14 \times 2)}}{\underset{\text{Carbon}}{(8 \times 12)}} + \frac{\overset{\text{oxygen}}{(25 \times 16)}}{\underset{\text{Hydrogen}}{(1 \times 18)}}$$

$$\frac{700 + 400}{114} = \frac{1100}{114} = 9.649$$

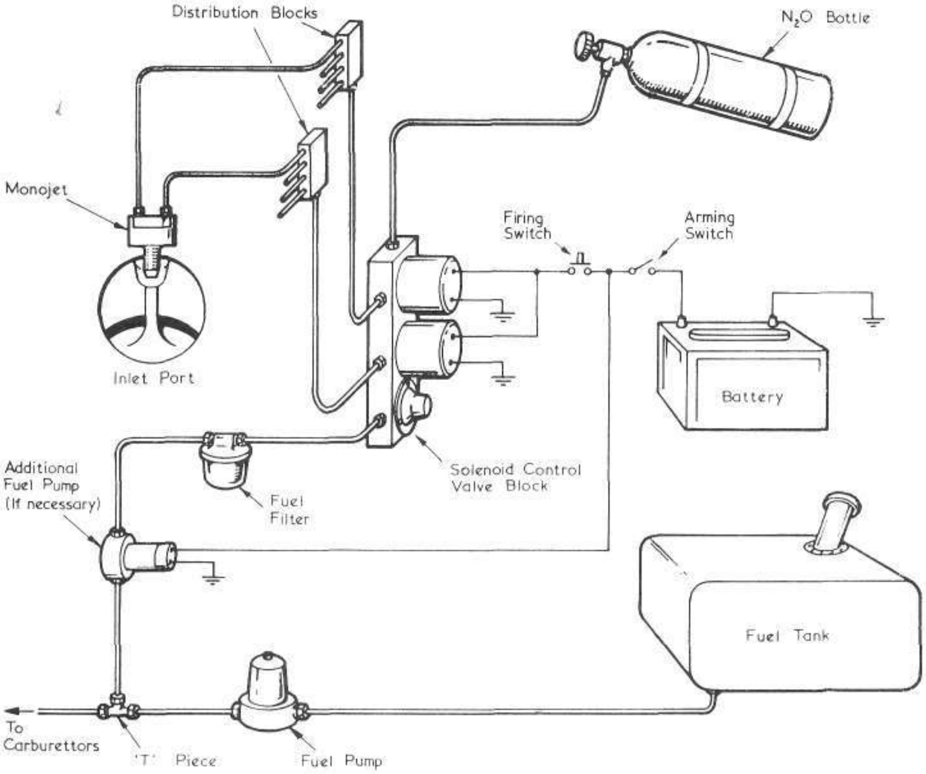
This shows that for complete combustion of the oxygen available and the fuel with nitrous oxide/fuel mixes, a ratio of one lb. of fuel to 9.645 lbs. of nitrous oxide is needed. Unfortunately things aren't quite this simple and there are numerous reasons why this chemically correct ratio is not used. To see why, we'll need to look at a basic property of combustion.

As stated earlier, our atmosphere contains almost 20 percent oxygen, and apart from small amounts of impurities, the other 80 percent is nitrogen. Nitrogen is a relatively stable compound and doesn't readily form chemical combinations except under high temperatures and pressures, when it will break down and form oxides of nitrogen (NOX). As far as we are concerned here, its most important contribution is that it acts as a temperature damper. If the nitrogen content of the total charge is reduced, combustion temperature will climb very rapidly.

N_2O is 36.3 percent oxygen and 63.7 percent nitrogen. If an attempt is made to double the power output of an engine by injecting N_2O, then the nitrogen content of the charge will typically drop from 80 percent to around 68 percent. This causes combustion temperatures to climb substantially. Unfortunately it doesn't stop there. To double the power, the amount of oxygen put into the engine has to be doubled. Not only does this mean twice the potential power but the engine must also somehow dissipate at least twice the waste heat. This extra heat leads us to one very important conclusion, namely that if no additional means of cooling is employed, the engine will run hotter

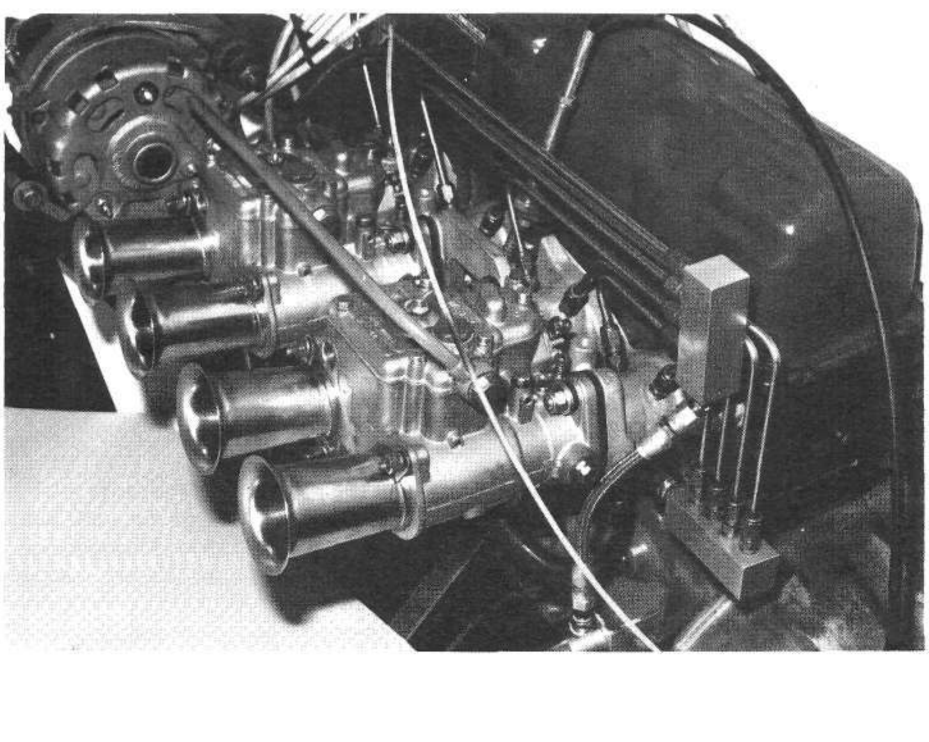
FIG 10-2.

This schematic shows the basic operating components of a nitrous-oxide injection system. The nitrous-oxide and the enrichment fuel can be injected into each individual manifold port or into the intake-manifold plenum. The simplest method is to use a spacer between the carb and the manifold, with the relevant plumbing installed in the spacer to inject the nitrous and enrichment fuel into the air stream as it leaves the base of the carb. This works fine for single carb installations but where twin side draft carbs are used port injection is necessary.



NOTE: NITROUS AND FUEL MAY BE INJECTED IN EACH INDIVIDUAL PORT RUNNER OR INTO MANIFOLD PLENUM IF APPLICABLE.

10-2. All this elegant plumbing is to feed the fuel and N_2O to each individual port. On this experimental installation a separate jet and jet holder was used for fuel and N_2O supply. Later type used by author, such as the Aldon Oselli system employed mono jets as depicted in the drawing above.



and become more prone to detonation. Unless steps are taken, overheating and detonation will limit the power output possible long before any limitations brought about by the amount of N_2O we can cram into the cylinders.

On the face of it we have a nasty problem, but fortunately it's one that can, to a large degree, be overcome. The key to the whole problem can be summed up in three words, Excess Fuel Factor (EFF). Essentially the detonation and overheating problem is why a chemically correct N_2O to fuel ratio is not used. By using a much richer mixture, the excess fuel acts both as a means to cool the combustion chamber and piston, and to suppress detonation. This is, in fact, why N_2O doesn't end up melting the engine. The knowledgeable racer might, at this point, ask, "An air fuel mix that is way too rich causes a substantial drop in power on a normally aspirated (non-N_2O injected) engine. Won't running a-way-too rich mixture to cool the engine and cut detonation rob us of a potential power increase?" Fortunately, petrol/gasoline with N_2O can tolerate running on the rich side of the chemically correct mixture. In this respect, petrol/gasoline/N_2O is similar to alcohol/air.

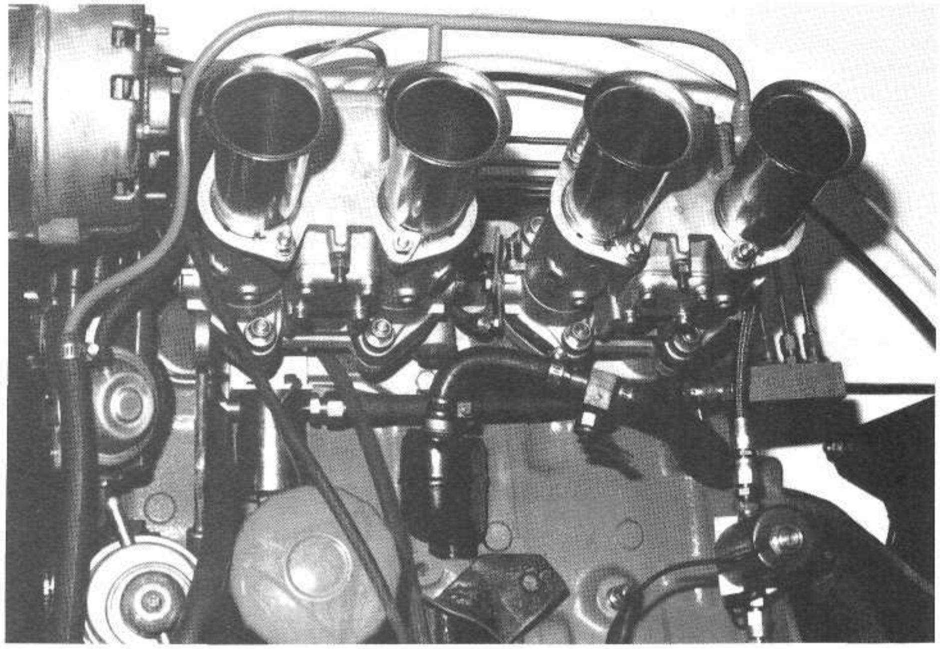
Looking at the other side of the coin, what would happen if the mixture became lean when the N_2O was injected? I've seen racers find this one out the hard way. If, when the nitrous is being injected, the mixture becomes lean/weak, then the increased heat plus the surplus oxygen will literally burn the chamber, plug, valves and piston crown to a crisp. What you are doing in effect is turning your engine into a self-consuming petrol/gasoline/N_2O cutting torch. And the amount of warning you get that this is happening is limited. I have seen a highly loaded nitrous engine literally digest its innards in as little as two or three seconds from the time the button was punched. This was on a unit that the racer "adjusted" himself with no real knowledge of what was going on. Nitrous kit manufacturers are very aware of this situation and go to great lengths on their calibrations to prevent such overheating.

There comes a time when increasing the EFF does little or nothing to reduce an engine's tendency to detonate any further. This is where we come to the end of the line on power increases. Past this point EFF serves only to cool the combustion chamber.

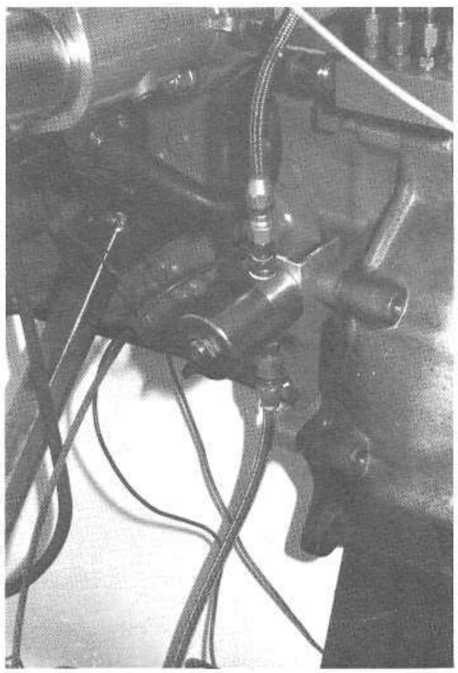
But the bottom line is this: how much power increase can we expect? My own experiences indicate that a conventional N_2O kit with no exotica can produce a relatively easy 25-30 bhp per litre increase. If the engine is strong and can dissipate heat well, figure on as much as 50 bhp per litre with little in the way of extra modifications. If you want to go to town and build the engine especially to cope with heavy loads of nitrous, then as much as 70-80 bhp per litre are possible, but you had better have a bulletproof unit to handle this much.

I have talked about practical power increases in terms of numbers, but these numbers alone fail to convey one very important point concerning N_2O injection. The use of N_2O causes the power to go up due to a substantial increase in engine torque. For instance, if your Ford SOHC 2000 produces 100 bhp maximum at 5000 rpm, a typical 50-bhp (approx.) N_2O kit would put the power up to approximately 150 bhp at 5000 rpm. In other words, torque at that rpm would be up from 105 lb./ft. to 157.6 lb./ft. In simpler terms, adding 50 bhp with N_2O has the effect to adding 1000cc to the engine's capacity. With this sort of impact on the power curve, "hitting the button" has a profound effect on acceleration, at least until the N_2O runs out.

Just how long will the nitrous supply last? As you would expect, nitrous consumption is fairly proportional to the amount of power increase it is supposed to produce. Fortunately, an easy-to-remember relationship generally describes how long the N_2O consumption lasts. As a rule one pound of N_2O will produce 100 horsepower for 10 seconds. Half the power increase will make that one pound of N_2O last twice as long. Similarly, twice the power and it will last half as long. Although bottles can be had in a wide range of sizes, 10- or 15-lb. bottles seem to be the most popular. This suggests a 50-bhp system will give five minutes running time on a 15-lb. bottle.



10-3. Here's how the plumbing looks on the underside. *The rubber hose carries* fuel at 6 psi and the braided stainless steel lines carry N_20 at 750 p.s.i.



10-4. Here's a close-up of the N_20 valve. This valve must be free-flowing otherwise the resultant pressure drop across it will cause the line to freeze. Just for reference, the 'freezing' action is required in the intake manifold. This is just what happens if the principal restriction is the metering jet in the manifold.

SYSTEMS OPERATION

So far, we have talked about how Nitrous Oxide can give us more HP, now let us go into the actual mechanics of actually getting the Nitrous Oxide into the engine when it's required.

Fig. 10-2 shows how a basic system operates. As you can see, the basic system is essentially straight-forward, but there are a few pitfalls which must be avoided. First of all, the actuating solenoids must be mounted as close to the injection nozzles as possible. This is especially so as far as the fuel line is concerned. If there is a long run between the solenoid and the jet, the fuel can be slowly drawn out of the line, and when the Nitrous button is operated the Nitrous, at 750 lbs or more per square inch, will arrive at the jet and get injected into the engine before the fuel does. This means that for a moment there is no extra fuel for the extra oxygen. The result is that you can start eating away at pistons, plugs, etc. in that brief time before the fuel arrives. Secondly, Nitrous valves must be capable of flowing a considerable amount more Nitrous than the jet passes. If there is too much pressure drop across the Nitrous valve, the line can freeze up. All the pressure drop should occur at the jet and nowhere in the line. The fuel system as you have probably already gathered, is important, and if anything happens to interrupt the extra fuel supply needed to go with the Nitrous, then trouble starts, so precautions must be taken to prevent this. The first and most obvious precaution is to fit a fuel filter in the fuel line so there is no possibility of dirt blocking the fuel jet. Secondly, when the system is produced by a manufacturer, the jet size will be pre-set in the Nitrous injection kit, but the jet size for fuel will have been set with a particular fuel pressure. This means you must install a fuel pressure gauge and a fuel pressure regulator in your fuel line to set the fuel pressure for the jet to pass the correct amount of fuel for the amount of Nitrous involved. This aspect is important if a U.S. built Nitrous injection system is imported to England. The reason for this is that essentially European fuel pumps operate at a much lower pressure than U.S. fuel pumps. Most Nitrous manufacturers build their Nitrous kits to operate on 5–6 PSI fuel pressure. Most European cars are operating on considerably less pressure than this, typically 3 PSI. This means you either have to get the fuel jet made larger to compensate, or install a high pressure fuel pump. Whichever the case, it will be necessary to have the correct pressure for the jet size involved, otherwise you could destroy your engine. If you intend building your own unit, the actuating switch must be a push-button switch. If you use a toggle switch that stays in one position or the other, then if anything happens you will not be able to turn the Nitrous off. This means the engine will continue to run, short of turning off the ignition.

IGNITION REQUIREMENTS

The blend of nitrous oxide and fuel/air charge does not have the same rate of combustion as does a normally aspirated engine without nitrous oxide injection. Thus, the ignition timing curve needs to be adjusted to make the most use of nitrous oxide. Unfortunately, this can mean that the timing curve might be incompatible with day-to-day driving. It is best not to push the button until at least 3000 rpm is seen on the tach. If nitrous oxide injection is started before this, the engine will probably start to detonate. Above about 3000 rpm, the ignition timing may require retarding back two degrees or three degrees when nitrous oxide injection of around the 50 bhp is used. If yours is a competition engine and you are only concerned with the peak power using 100 to 150 bhp worth of nitrous, ignition timing may require from four to six degrees less advance.

Generally, ignition requirements are less severe with nitrous oxide. The extra oxygen makes the charge easier to ignite. However, it is still advisable to use a good transistorized ignition. This is because there can be occasions when the nitrous oxide causes the plugs to burn out. If one plug burns out and fails to spark, your engine will dump a load of very combustible mixture into the exhaust manifold. If this burns in the exhaust manifold or even further down in the exhaust pipe, the engine will produce a tremendous backfire, far more severe than a backfire from a normally aspirated engine. If you have a silencer on this car, it will almost certainly blow it clean off. To avoid such eventualities, it is a good idea to install an electronic ignition that will fire a plug come what may.

Since the spark plug tends to be the hottest operating component in the combustion chamber short of the exhaust valve, it will be necessary to run a much cooler plug to avoid the electrodes being burned off. On a typical two-litre engine, the spark plug heat range may have to go down as much as three ranges, if a 50 bhp nitrous kit is used. If a 100 bhp nitrous kit is used, it will be necessary to use retracted nose plugs or some other means to cool the combustion chamber.

COOLING IT

Although the Excess Fuel Factor is the most often used means of controlling combustion chamber temperature, there comes a time when with large enough quantities of nitrous oxide injection, it is no longer adequate. Water injection works well in conjunction with nitrous oxide. It allows as much as 50 percent more nitrous oxide to be injected, smooths out combustion noticeably and adds extra control for detonation. The simplest way to use water injection is to wire the water and N_2O systems so that they are injected simultaneously. The water injection system can be based on a windshield washer bottle and pump. The typical pressure from a windshield washer pump is around 40 psi. This allows the use of some fairly fine jets which spray water in a well-atomized form. When calibrating the water injection system for a nitrous oxide unit, start off with way too much water. If you are injecting water into a Weber-carburetted setup with a windshield washer pump, then try .015-inch jets into each intake port as a starting point. If you are injecting into the plenum chamber of a single-carburettor setup, two jets diametrically opposed to each other of around .021 inch are a good starting point. These jet sizes assume you are using nitrous oxide injection which will give around 100-125 bhp more. The area of the water jets should be altered proportionately for higher or lower quantities of nitrous oxide.

I have mentioned it before in this book, but the use of a heat-resistant coating is to be seriously considered if nitrous oxide injection will be used for large power outputs. The coating has the ability to keep the pistons and combustion chambers intact when the engine runs into detonation, should it occur. This means that instead of destroying the engine, the effects of detonation will be held in check for maybe as much as a minute. Often this will give you time to find out exactly what's happening. Under pronounced detonation, spark plugs will melt. But the demise of the spark plugs is infinitely preferable to the demise of the pistons. In this instance, the spark plug acts as a safety valve. The only thing you need to remember is to take careful spark plug readings.

It is easy to be greedy, when it comes to nitrous oxide injection. And the results of being greedy are usually disappointing, to say the least. For normal road or street use, I recommend nitrous oxide injection up to about 50 bhp, no more. The stock cast pistons will NOT take any N_2O so forged pistons MUST be used. If you are building a competition motor which you need to last a reasonable amount of time, figure on 100 bhp as about the maximum. If you are going to drag racing and your only interest is to get to the other end of the track in as short a time as possible, even if it consumes an engine, nitrous oxide injection up to about 200 bhp is possible, although engine life with this much nitrous is very much dependent on how well you set up the engine to control the heat generated.

N_2O & ALCOHOL FUELS

Nitrous oxide and alcohol make excellent partners. In fact, nitrous oxide is to be preferred to nitromethane. It can make as much power as nitro, but is not quite as detonation-prone as nitro. The use of alcohol instead of petrol/gasoline reduces engine temperatures substantially. It is unlikely, except in extreme circumstances, for water injection to be needed when alcohol fuel is used. For an alcohol race engine using 100-200 bhp worth of nitrous oxide, use a compression ratio substantially lower than used with straight alcohol. Total experience seems to be limited in this area, but my guess is that alcohol and a 150 bhp nitrous system would require a ratio of around 10:1, but be prepared to experiment when building an engine such as this.

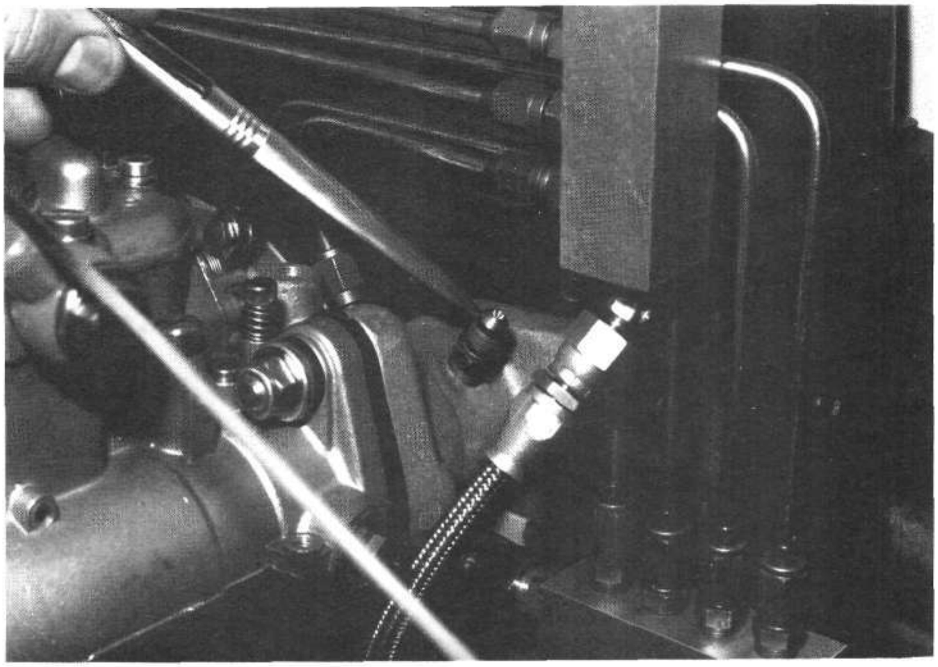
N_2O & THE TURBOCHARGER

Nitrous oxide injection and the turbocharger were made for each other. Each complements the other's good points and cures the bad ones. The results gained by putting these two together are nothing less than sensational. By combining the two, it is possible to build a 270-300 bhp turbocharged engine for the street without any throttle lag. If that intrigues you, turn to the turbocharger chapter.

SOME NUMBERS

To prove the effectiveness of nitrous oxide injection on a normally aspirated engine, I tested a unit built for use with one-barrel-per-cylinder carburation. A similar, but more advanced port injection system with adjustable power levels, is being developed in England by Aldon Automotive and Oselli Engineering. I have tested prototype units from these companies and results have been excellent. However, back to the first unit.

For the particular tests here, the unit was jetted to give theoretical 50-, 100- and 150-bhp increases. However, there was a problem with testing at higher power levels. Unlike the normal throttle action, nitrous oxide cannot be brought into action slowly. When you hit the button, the power is there instantly. The resultant changes in torque output are so large and so abrupt that the engine would very quickly overrev. This is in spite of the load being increased on the dyno prior to hitting the button. The nitrous tests only include a full power curve on the 50-bhp jets and one spot reading on the 100-bhp jets. Taking the one spot reading was about as much as my nerves would stand. To achieve this one figure, it was necessary to lug the engine down to 3000 rpm. Hitting the nitrous button instantly caused the engine to run to over 7000 rpm before I managed to apply enough load to stabilize it. Rather than destroy an engine which I had planned to put in a car, I cut short the dyno testing. The point was proved. Nitrous oxide makes big horsepower. Fig. 10-3 demonstrates the effect it has on the power curve.

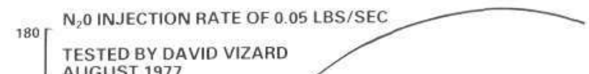


10-5. To test the practicability and reliability of various hp levels, this experimental system was equipped with replaceable jets. This allowed adjustment of power levels and calibration of N_2O to fuel ratios.

FIG 10-3

TEST OF NITROUS OXIDE INJECTION

ENGINE: 2000cc FORD WITH MODIFIED CYLINDER HEAD AND STAGE II CAM WITH 2x40 DCOE WEBERS



	WITHOUT N_2O		WITH N_2O	
R.P.M.	HP	TORQUE	HP	TORQUE
3000	72	126	101	177
4000	108	141	140	184
5000	125	131	171	180
6000	143	125	181	159
7000	139	104	180	135

Turbocharging & Supercharging

In this chapter I am going to deal with forced induction. But before going too far, let me clarify some terms that we'll encounter. "Turbocharging" is really short for *exhaust-driven turbine supercharging*. I mention this specifically so you don't fall into the same trap that some racing officials did. The local half-mile oval track expressly forbade the use of a supercharger for a particular class. So two enterprising guys turned up with a turbocharged car, claiming that it was turbocharged as opposed to supercharged. Before the start of the race, the track officials were not made aware that a turbocharger is just a specific type of supercharger. They thought that the racers had, in fact, found a loophole in the regulations: the car was allowed to run its race. With a substantial power advantage, the turboed car blew all the opposition into the weeds. Right after the race, the track officials found out the true definition of turbocharging from a number of irate drivers.

When we say supercharging we refer to a mechanically driven pump to force a charge into an engine. A turbocharger specifically refers to superchargers driven by exhaust gases. There are pros and cons for each method. Generally speaking, turbocharging gives the highest horsepower potential, although the latest mechanically-driven supercharger technology may have reduced this margin. The biggest, most often cited drawback to turbocharging is the dreaded throttle lag. If we are to use correct terminology, this should be called boost lag. The fact that a mechanically driven supercharger does not have boost lag is one of the reasons for its continued use, especially in drag-racing. Its instant throttle response often means arrival at the far end of the drag strip ahead of its possibly more powerful turbocharged equivalent. With almost instant response now possible for turbos and higher compressor efficiencies being realized for mechanically driven superchargers, the performance gap between the two forms of supercharging is reduced. Making a choice between the two will not be easy. At this point, my preference is for the turbocharger; it has the power advantage over a mechanically driven supercharger.

TURBOCHARGING – WHAT IS IT?

A turbocharger consists of two turbines mounted on a common shaft. Each of these turbines is encased in a separate housing. One turbine and housing constitutes the intake turbine, and the other turbine and housing constitutes the exhaust turbine. When this unit is mounted on the engine, exhaust gases from the exhaust ports are routed to the exhaust turbine housing. As exhaust flows through the housing, the exhaust turbine will rotate. The intake turbine, since it is on the same shaft, will also rotate. And when the turbine reaches a high enough speed, the intake turbine will start to draw charge or air into its housing and then force it into the engine. Fuel may be introduced into the air either by sucking air through the carburettor prior to its entry into the intake turbine, or passing the air from the outlet side of the intake turbine into the carburettor and then into the engine. A third method is to fuel inject the engine at the intake ports in the cylinder head.

Look at its function in the following manner. Imagine your car is being driven down the road at about 30 mph. Under these conditions, it will have only a small amount of throttle opening, because the power required to drive the vehicle at 30 mph is low. As a result, there won't be much air passing into the engine, nor will there be much exhaust coming out of it. Flow through the exhaust turbine is sufficient only to cause the turbine to windmill somewhat idly. When, you require increased power, the throttle is opened fully. This allows more air into the engine and as a result, more exhaust is produced. This increased quantity of exhaust impinges on the exhaust turbine, causing it to spin faster. Since the intake turbine is on the same shaft, it will also spin faster and draw air into the engine. This extra induced air means more exhaust is produced and that extra exhaust will cause

11-1. Here I am making some minor adjustments on the engine before a power run is taken. Note the white exhaust housing on the turbo. This is because it was finished with a coating of Heanium to retain the heat.

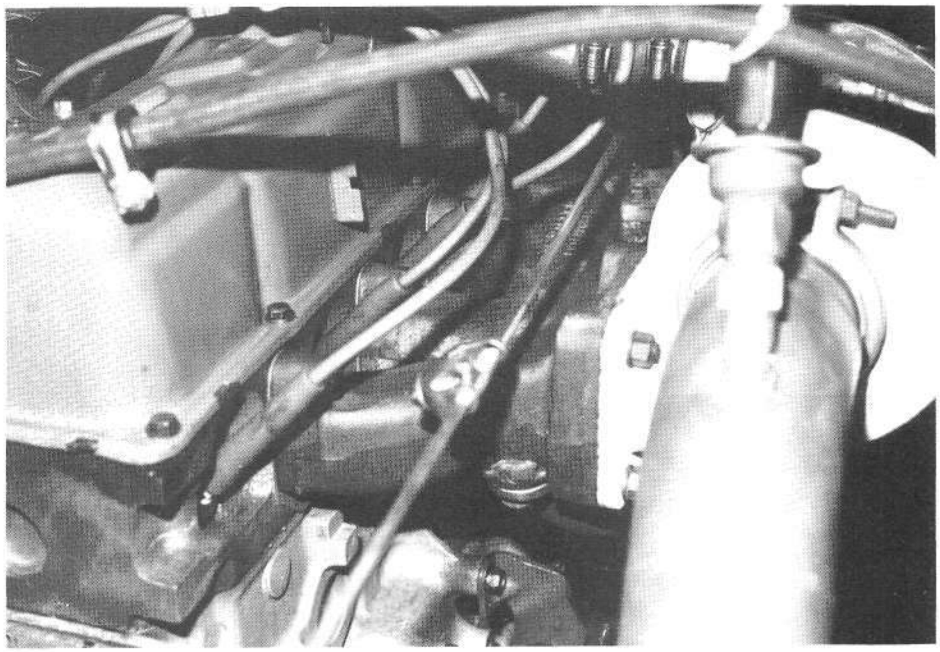
the exhaust turbine to spin even faster. The intake turbine will now spin faster and it may now be going fast enough to actually start drawing in more air than the engine would of its own accord. This means the manifold pressure now starts to rise above atmospheric. As you can see, this leads to boost being produced in the intake manifold.

In practice, on the road, this boost building process can take a second or so. This is what is normally called throttle lag. As you can see, it is actually boost lag. The opening of the throttle on a turbocharged engine does not produce instant boost; this is the turbocharger's main stumbling block especially for a drag racing or a circuit car where throttle response from the start line or out from slow corners is important.

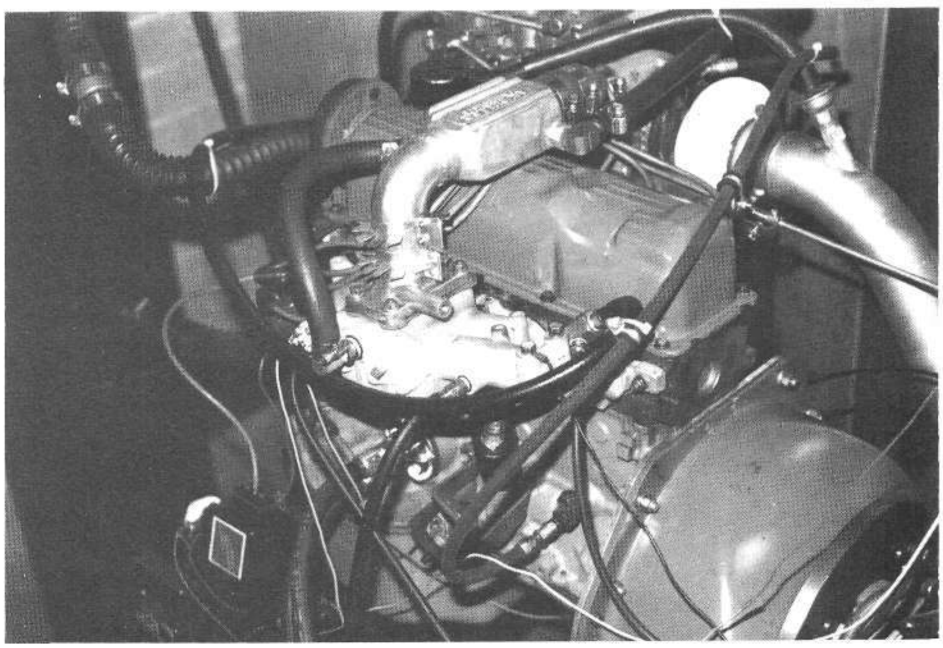
You may ask: "Why doesn't boost go on increasing forever, since it seems to be a closed cycle of events?" Well, the restriction of flow past valves, through ports, the turbine housing, carburettor, exhaust manifold exhaust turbine housing, eventually clogs up the system. The turbocharged engine will reach some stable level of boost. Just where this stable level of boost is will be determined by the factors just mentioned.

If the overall system is at all free-flowing, the boost will stabilize at such a high level that the engine will blow itself to bits long before such a situation is reached. It is therefore necessary to exercise some form of boost control on a turbocharged engine. Boost control can take many forms. Generally it is achieved through use of the right size turbines and housings, plus the addition of a positive boost control or some method of limiting airflow. Positive control can be brought about by a wastegate which bleeds off excess exhaust when a certain boost level is reached. On the other hand, limiting boost by limiting airflow can be had by the use of carburettors of reduced size so the airflow is eventually restricted by the carburettor. The third method is to restrict the exhaust flow out of the engine. This limits the turbocharger's ability to draw on the intake side. All these methods have their pros and cons, which we will discuss later.

Fundamentally, the boost on a turbocharged engine or one with a mechanically driven supercharger is governed by the same principle, namely, how fast the compressor spins.



11.2. Most (if not all) the turbo engines I worked with used the Ak Miller exhaust manifold shown here.



11-3. The cross-over pipe is also an Ak Miller item. When used for road-going applications, heated water from the engine is passed through the jacket of the rectangular section above the cam cover. For race applications, alcohol, super-cooled by dry ice, can be fed through to act as an intercooler.

Supercharger boost is controlled by selecting pulleys or gears to give the desired ratio between engine speed and supercharger speed. Boost given by a mechanically driven supercharger is much nearer a constant value throughout the rpm range whereas a turbocharger gives a boost of ever-increasing value unless it is restricted by one of the means we have just discussed. The result of this on the power curve is that mechanically driven superchargers generally give a lot more low-down torque than a turbocharged engine while top-end power is usually less than that given by a turbocharged engine. However, these circumstances can be drastically modified, as we will see later.

CHOOSING YOUR TURBOCHARGER

There are enough turbocharger kits on the market today that it is possible to buy a setup for your particular engine tailored to the power output required and the size of the engine. The importance of sizing a turbocharger to the application cannot be over-stressed. Unless the turbo is correctly matched and sized, performance will be poor. Most turbo manufacturers are aware of this; much work is done in this area. When purchasing a turbocharger kit, be sure it will be suitable for its intended purpose. For instance, a turbocharger for road use will likely be smaller than one for racing, since power requirements for the road will be less than they are for racing. For racing, the turbocharger will be sized to give a higher peak horsepower and, of course, stress the engine much nearer to its ultimate limits. Two brands of turbochargers, are popular for use on the Pinto engine, these being made by Airesearch and Rajay.

As far as the intake side of the turbo is concerned, the type of turbine required is based upon how much power we expect the engine to produce. For applications where bhp outputs are expected to be less than about 220-250, an Air Research TO4 with an E1 trim compressor will get the job done. For very high horsepower levels, especially those needed in all-out competition, a larger compressor is necessary, typically an Air Research TO4B with a V1 trim. This can handle power outputs up to about 550 bhp. For equivalent sizes in Rajay, figure on a B-flow for a typical 150-175-bhp turboed, street two-litre engine, and F-flow for 175 to about 350, maybe 400 bhp. An E-flow should be used for anything more than that.

For most medium-to-high power levels, the turbocharger unit most chosen will be TO4 in V1 trim. It is worth noting here that if too much boost is attempted with this turbo, together with a cam which is too short, the turbo will run into a surge condition. This occurs because too much boost is being generated for too little airflow through the engine. A cure can be accomplished in one of two ways. Either turn down the boost or put in a bigger cam. An example: an extensively modified turbo engine was equipped with a Ford Stage II cam. At 70 inches (hg. absolute) boost and 7000 rpm it ran into a surge condition. Just before surging took over control of the engine, it cranked out about 255 bhp.

Changing the cam shaft to the longer timing, higher lift Crane 322 B cam,

FIG 11-1A
FLOW MAP FOR AIRESEARCH TO4 B TURBO IN V-1 TRIM

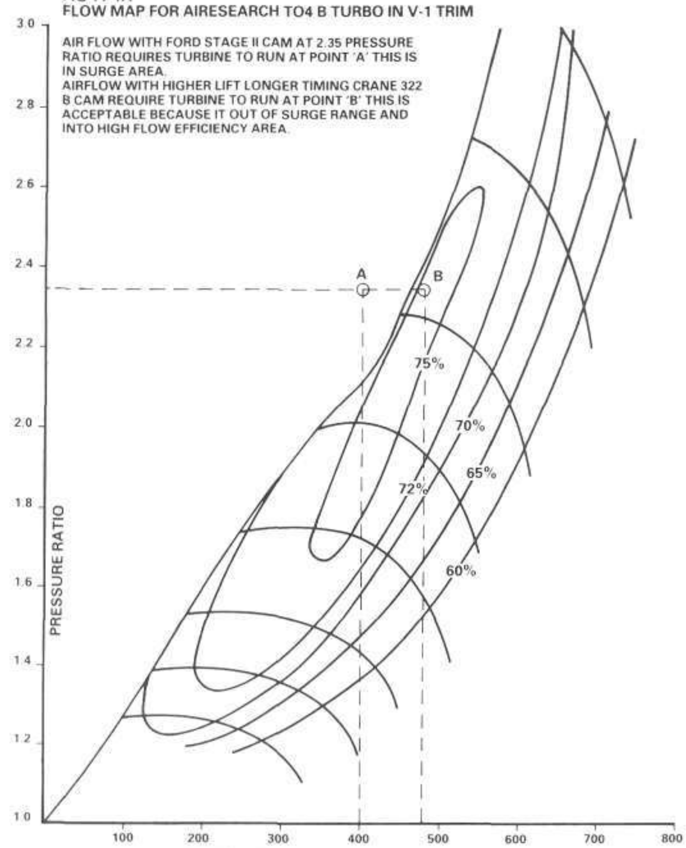
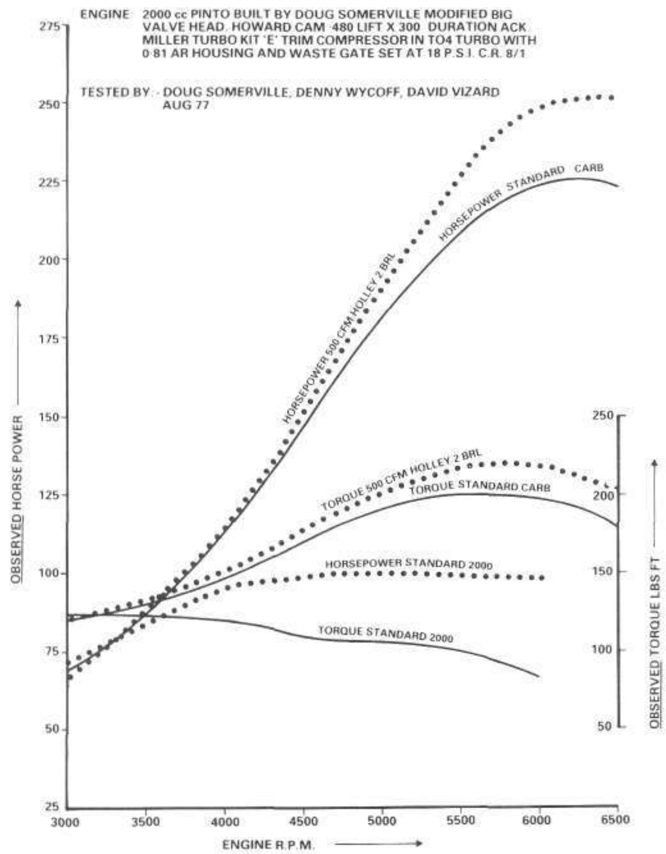


FIG 11-1B



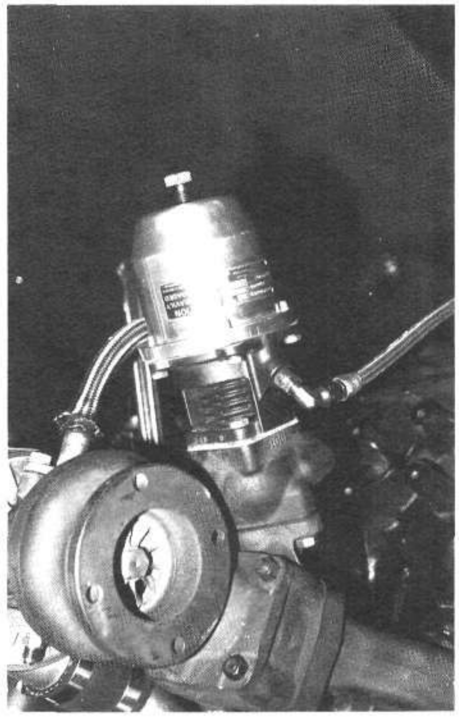
caused the surging to be eliminated. Horsepower at 70 inches of boost at 7000 rpm, rose from 255 to 306. Fig. 11-1A may help illustrate why this is so. With a small cam and a pressure ratio of 2.35 (this is what our 70 inches proved to be) the point on the map where these two co-ordinates intersect is on the left side of the surge line. Installing the bigger cam caused some 490 cubic feet of air to flow into the engine at our 2.35 pressure ratio. This put the point at which these two co-ordinates intersect on the map just to the right of the surge line right in the island of maximum efficiency.

If you find installing a big cam leaves your engine with too little low-end power and you are still looking for big boost numbers, then you probably picked the wrong turbo. You should really go down to a TO4 in E1 trim.

Although selecting the size of intake housing and turbine combination that will give the desired flow for your particular power requirements is important, it is even more important to choose the right exhaust housing. Selection of the exhaust housing dictates where in the rev range and how much the boost will be. The exhaust housing sizes are measured as a ratio of the area of the input to the housing and the approximate centre of this area is the central shaft. This is known as the area radius ratio. The smaller the number, the quicker the boost comes on and the higher the boost will go. A typical two-litre street Pinto will use an exhaust housing with an A/R ratio of around 0.58 or .69 for an Airesearch-type turbo or 0.40 for a Rajay. The more power and the higher rpm the engine is expected to yield, the larger the exhaust housing A/R ratio will have to be to prevent overboosting at peak revs and power. Unfortunately, the larger the exhaust housing, the less boost there is at low rpm. The ways and means of limiting boost, however, affect the size of turbo housing used, so we will deal with that subject now.

LIMITING THE BOOST

The simplest way of limiting maximum boost of a turbocharged engine is by flow limitation. This limitation can take the form of restriction in the intake flow or exhaust flow. A method used extensively on turbo kits for the Ford SOHC engine is the Impco valve. This valve fits in the turbo output side between the turbo and the intake manifold. When the boost reaches a certain level, the valve starts to close off, therefore limiting the air flow to the engine. Although effective in its function, it does cause intake charge temperature to rise dramatically.



11-4. Waste gates come in many sizes, both big and small. There are numerous cheap (and nasty) waste gates on the market. Airesearch makes a good, though expensive, waste gate, an example of which is shown here. Another quality unit is produced by Environmental Products in Los Angeles.

As suggested earlier, a simpler, but less effective means of limiting the boost is to deliberately use a carburettor too small for the job. There comes a time when even the turbo is unable to suck any more air through the carburettor and therefore the boost is limited by this restriction.

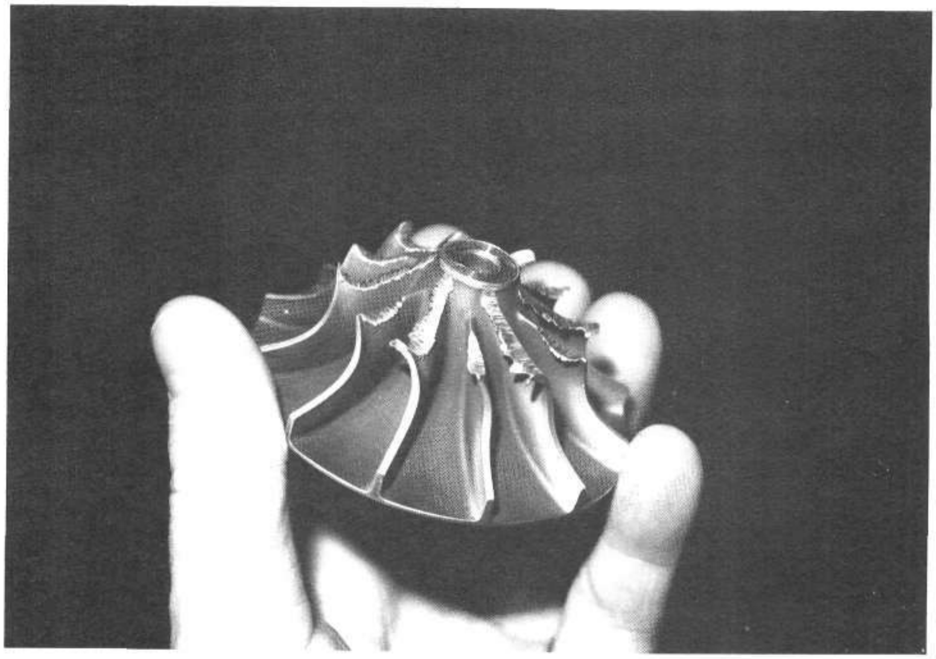
The exhaust can be restricted as well as the intake. The speed with which the exhaust turbine is spun is dependent to a large extent upon the pressure differential that exists across the exhaust manifold and the exhaust pipe downstream of the turbo. The more pressure drop across the turbine itself, the faster the turbine will spin. If some form of restriction is placed downstream of the turbo, then the pressure drop which occurs across the turbo is reduced. This effectively limits the maximum boost. Exhaust restriction can take the form of a restrictive silencer/muffler or a specially sized orifice plate just downstream of the exhaust turbo. The advantage of using a sized orifice is that the size of the hole can be altered until the desired boost levels are reached.

The means so far described of limiting boost are simple and they get the job done. However, the most effective way of limiting boost is to use a wastegate. A wastegate is a valve which senses manifold boost pressure and when the boost pressure reaches a certain pre-determined level, the wastegate opens and bypasses excess exhaust around the exhaust turbine. This means the turbo only delivers the desired amount of boost and excess exhaust is simply dumped overboard.

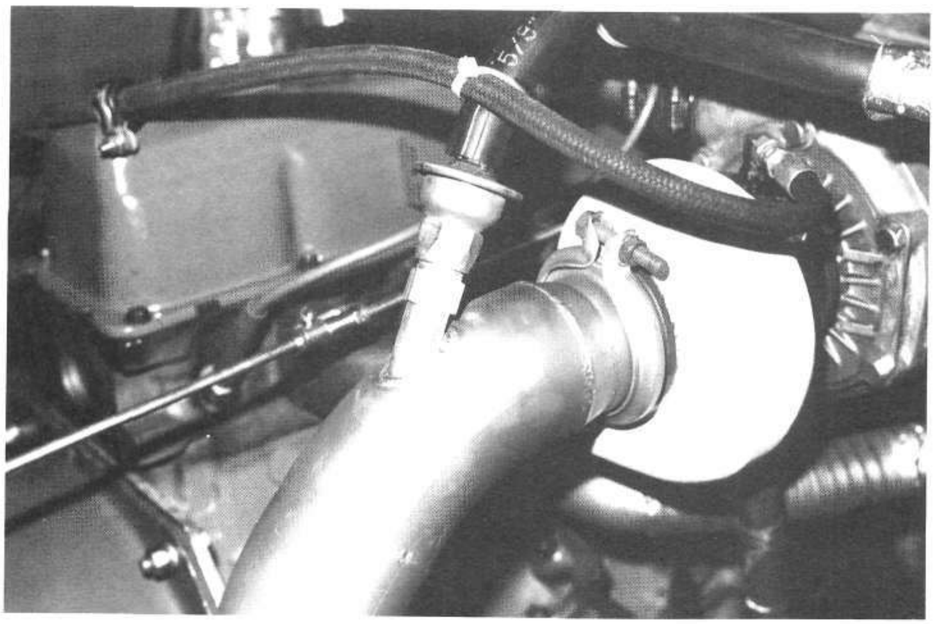
This method has the advantage of allowing the exhaust housing to be one, maybe two, sizes smaller than would otherwise be the case. This brings the boost on sooner and allows for a more responsive turbocharger installation.

Until recently, the big argument against wastegates was that they were prohibitively expensive. The tremendous upsurge in turbocharger usage, however, especially in the automotive field, has meant mass production of wastegates; they are now at a price which makes their use a much more feasible, even desirable, proposition.

The other means of limiting boost pressures, though simple, do have their drawbacks. Just how much these disadvantages may affect you depends on the applications you intend your turbo machine for. Let's look at the exhaust backpressure limitation. This method works fine up to a point, but if exhaust



11-5. Here is what can happen if a small foreign body meets with the intake impeller. If a large object had been involved, the chances are you wouldn't find anything. The moral here is always run an air filter and make sure it's of adequate flow capacity.



11-6. An evacuated sump system proved less effective on a turbo engine than a normally aspirated engine. Plumbed in as seen here, crank case pressures at full throttle were only fractionally below atmospheric pressure.

backpressure runs too high, it causes the response of the turbo to suffer in the mid-and upper-power ranges, as well as making throttle response feel mushy. But a certain amount of exhaust restriction works fine; this is something you will have to experiment with if you are putting together your own turbo installation.

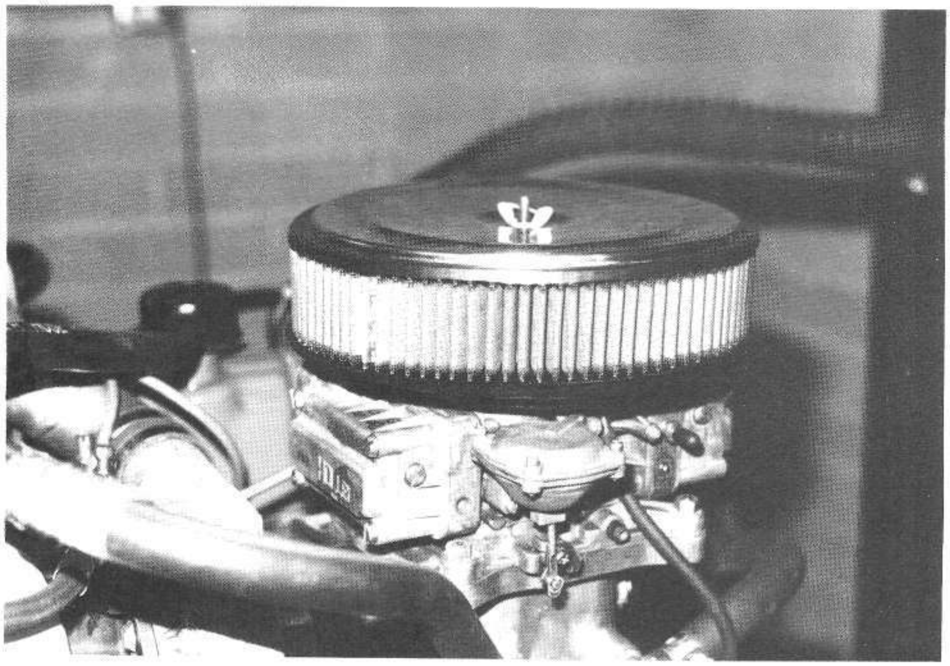
Limiting boost by means of intake restriction does not produce the mushy feel even if done to excess. What happens here though is that power is limited, but if restriction is too high, the turbocharger can run into a surge condition which can actually damage the turbo.

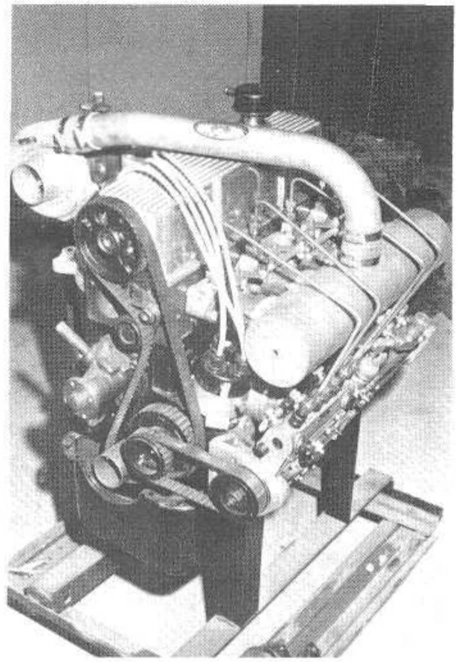
The Impco valve has the advantage of leaving the motor almost unaffected until it comes into operation. This means that exhaust backpressure can be reduced to a minimum and a restrictive intake due to carburettor down sizing is elliminated. However, the Impco valve itself presents considerable restriction to flow and so some of its assets are cancelled right there. Also, as it begins to limit boost by closing, it causes the intake charge to be severely heated. On engines which are boosted to 12-15 psi, the Impco valve seems to work fairly well, but for boost levels above this, either a restriction-regulation or a wastegate is preferable.

DRAW-THROUGH OR BLOW-THROUGH

The carburettor can be mounted in one of two positions on a turbocharged engine and a great deal of controversy exists as to which is the better. The carburettor can be mounted such that it feeds air into the turbo and the turbo then feeds it into the engine. This is known as a draw-through system. Under these conditions, the carburettor operates more or less as it would do on a normally aspirated engine. Another school of thought says that the carburettor should be mounted between the turbocharger and the intake manifold. This means the turbocharger draws fresh air and pumps it into the carburettor which then feeds the fuel/air mixture to the engine. The claimed advantages of this method are that the turbocharger is always operating with its intake at atmospheric pressure for quicker response and, secondly, it does not need an oil seal between its intake turbine and housing. The reason for the seal is to prevent oil being sucked into the vacuum that exists at part throttle when the carburettor is upsteam of the turbo. The principal snag with the blow-through system is that carburettor calibration becomes difficult. The carburettor is a velocity-sensitive device; it does not know about pressure changes. This means that although the engine may receive vastly increased quantities of air, the fuel supplied to the engine may not go up in proportion. The result is that the engine runs weak and is in danger of detonating and melting pistons, unless some often intricate complex carburettor mods are made.

Another snag to the blow-through system is that the carburettor must be enclosed in a pressure box and the fuel supplied to the carburettor must be at a suitable pressure above the boost pressure. This means the fuel pressure has to be regulated in accordance with the boost pressure used at all times. My



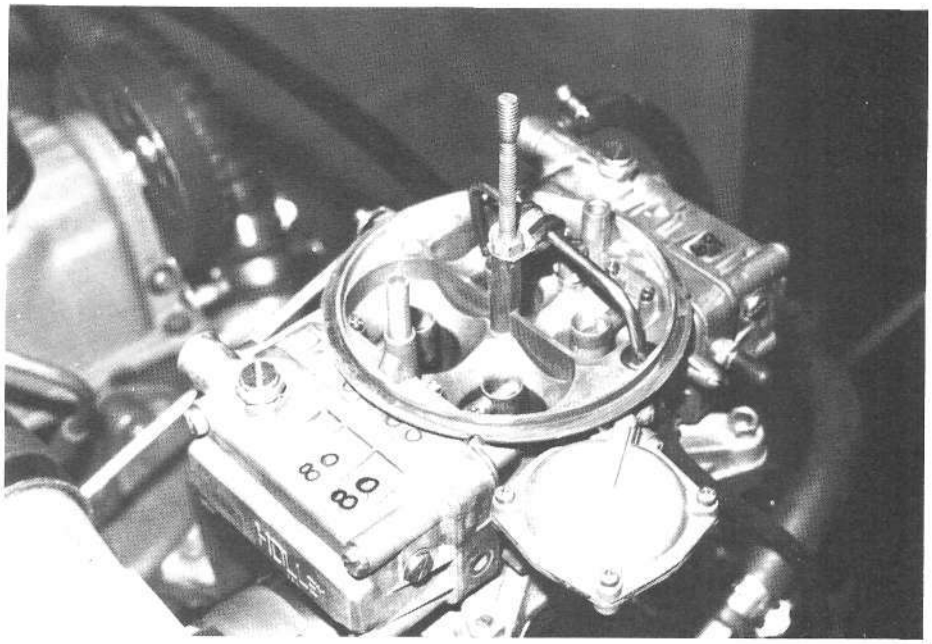


Above

11-7. On a race engine, choose an efficient air filter for the carb. A K & N filter is mounted on a 600 cfm Holley on this draw-through system. At 99.5% efficiency, the filter flowed enough air for about 425 bhp.

Above right

11-8. This turbocharged, fuel injected 2000 cc engine was built by Ak Miller. It features port injection downstream of the throttle butterflies. A butterfly is housed in each port runner. By taking this route, Ak eliminated the seemingly perennial argument of blow-through versus suck-through.



Right

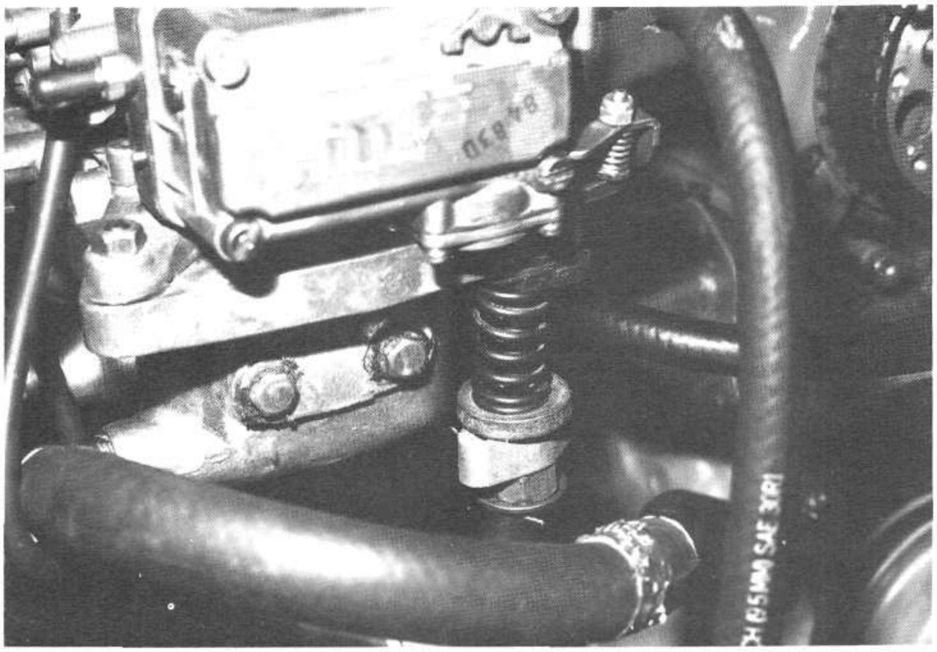
11-9. Another Ak Miller product allows the mounting of a Holley four barrel such as this onto a TO4 type turbo.

own personal preference to date has been for the draw- through system and this is largely substantiated by other people in the industry who are running very high-powered turbocharged cars. Almost all are using draw-through systems. However, as I've said before, the speed equipment industry does not stand still. A new development with blow-through systems could change the picture. The specially designed for blow through turbo application Dellorto side draft carbs may well be that new development.

A third possibility exists: fuel injection. Here, the injection system has to be designed to measure the amount of air going into the turbocharger by whatever means. It then injects the correct quantity of fuel into the engine at whatever point is convenient. The butterflies for throttling the engine may be situated either prior to the turbo or after it, depending on ease of installation layout. If the injectors are situated in such a position that they inject into a high-pressure boosted environment, then the injection pressure must be compensated for by whatever pressure the boost level has reached. Fortunately most fuel injection systems have ample fuel pressure to cope with boost levels, as high as the 45- 50 psi used in alcohol burners.

FUEL INJECTION OR CARBURATION

One of the principal reasons for fuel injecting an alcohol engine is that the quantities of fuel involved are so much higher than they are with a petrol/gasoline engine. But generally, for use with petrol/gasoline, a carburettor makes a lot more sense than fuel injection. This is not to say that fuel injection won't give you more horsepower, but even with the airflow restrictions of carburettors, the engine can still develop enough power to blow itself to smithereens. It therefore seems of little point to spend that much more money mak-



11-10. Unless adequate precautions are taken, vibration can cause all sorts of carburation problems. Here is an example of one of the ways in which carb vibration can be reduced. The carb adaptor itself is connected by a piece of rubber hose to the turbo so it has a flexible vibration absorbent joint at this point. The weight of the carb is then supported on a spring as seen here.

11-11. Without taking the necessary steps to damp out vibrations, such problems as cracked float arms can arise. This and fuel foaming leads to an almost total lack of carburettor calibration.

ing something with the potential for more power when engine strength, not airflow, is the limiting factor.

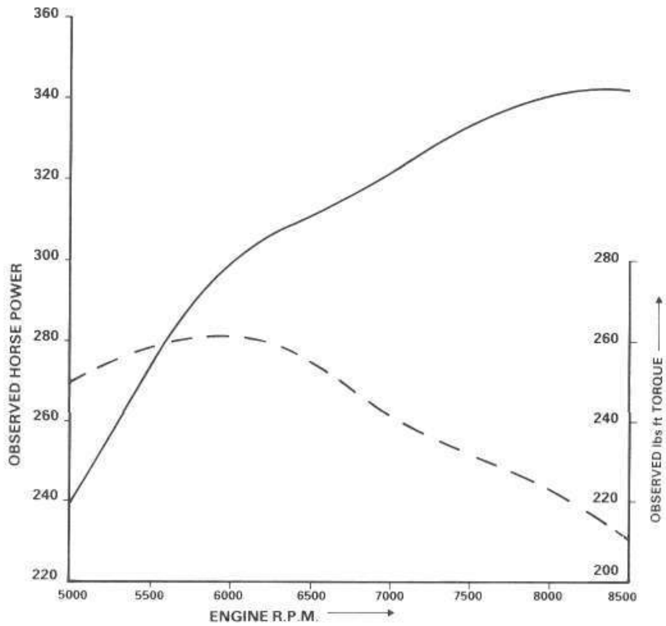
Many of the kits on the market, especially those designed for road use, utilize the standard carburettor. But at anything over about 180 bhp, the standard carburettor becomes a prime source of restriction and limits horsepower. A good substitute for the standard carburettor is a 350-or 500-cfm Holley two-barrel. An adaptor plate can be used to mount the Holley.

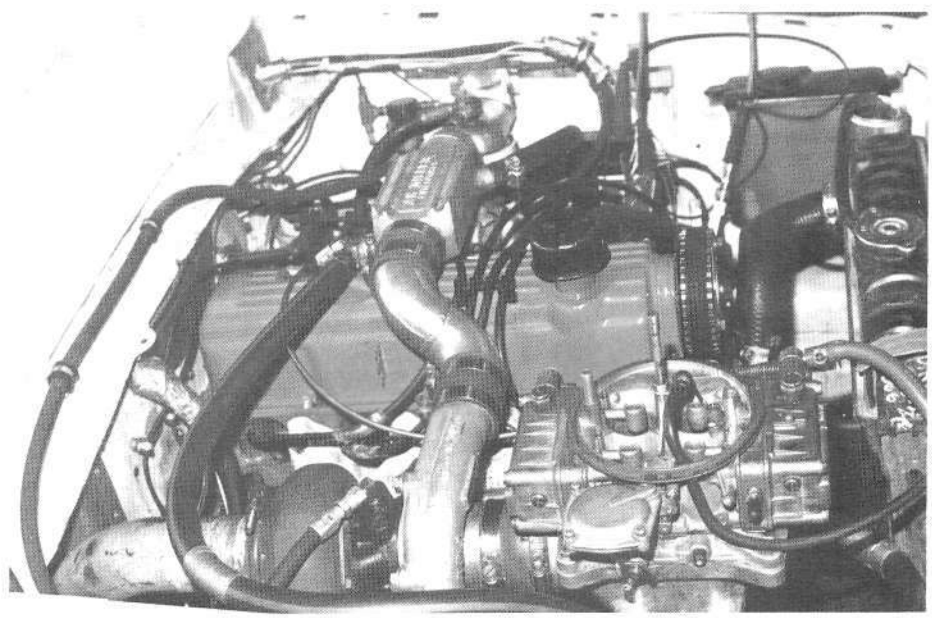
Another useful way to go is to use a four-barrel Holley with vacuum secondaries. Ak Miller produces a manifold for this carburettor, which allows it to be mounted onto a TO4 type turbo. A precaution I should mention here is that this type of carburettor, when used on a four-cylinder engine, can be prone to vibration problems. And this situation gets worse if solid engine mounts are used. To avoid this problem, it is necessary to couple the carburettor in such a way that it is divorced from sources of vibration. This usually means rubber-mounting the carburettor both to the turbo intake manifold and to the body of the vehicle. Another problem with using the Holley four-barrel on a highly boosted (20 psi or more) installation is that it is often difficult to calibrate, especially at the top of the rpm range. This means the carburettor must be modified to incorporate replaceable air correctors/air bleeds. Braswell Carburation can perform this modification, as can many other carburettor specialists throughout the U.S.A.

Those of you who are intending to use as much boost as possible through the standard carburettor may find that a high-speed enrichment problem occurs. This is due primarily because the carburettor is run too near its airflow limit. This is what happens: air speed through the carburettor becomes so high (far higher than ever intended for a naturally aspirated engine) that it causes the high-speed enrichment circuit to come into operation long before full power is reached. Consequently, when the turbo engine does reach full power, this high-speed bleed is well and truly in action. The result is that it feeds far too much fuel. Usually the high-speed bleed only comes into action on a normally aspirated engine right at the top of the rpm range. To cut down this gross enrichment it will be necessary to restrict the high-speed bleed. If you are running into this situation, it's good to remember that you may end up with a substantial increase in power by going up on carburettor size, if indeed extra power is what you need. If the carb is too small for the application, enough vacuum can be created at the base of the carb, to fool it into thinking it is operating at part throttle. Under these circumstances the power valve can switch "Out" and your engine will more than likely burn down. To avoid this situation, select a carb big enough for the job and a power valve that switches in at a vacuum, higher than it will see under full throttle conditions. Fig. 11-1 compares the power of a turbocharged engine with a standard carburettor and a turbocharged engine with a 350 Holley two-barrel. The difference in power further down the boost range would be insignificant, but on engines producing more than about 175 bhp, the bigger carburettor starts to pay off.

The particular engine tested required water injection from 12 psi boost upwards, and was run on 100-octane fuel. As we already know, hundred octane fuel is virtually non existent, thus it would be necessary in many instances to start the water injection at boost levels lower than the 12 psi used in this case. Obviously the water injection need not be started until just before the point of detonation. An adjustable pressure-sensitive switch can be used to trigger water injection. Usually a fixed jet, constant-volume injector operated from a pressure switch is entirely adequate for a turbo engine. But better than injecting just water is a combination of water and methanol. In some forms of competition, water injection is not allowed but the use of racing petrol/gasoline is. Under these conditions, it is necessary to use as high an octane fuel as is possible. Two fuels which seem to cope well with high boost are Daeco Racing Fuel, at around 106 octane, and H&H blue, which is around 115 octane. With compression ratios of about 6.5:1 these two fuels could be used with boost pressures as high as 28 psi without water. With a selection of the right cam, headwork, and such, it is entirely possible to generate 330-400 bhp with the 2000cc engine. As is usual with turbocharged engines, the limiting factor is the point at which detonation starts. The use of alcohol fuel gives a bigger margin of safety here from two aspects. First, the alcohol has a much higher latent heat of evaporation and is able to cool the intake charge much more than petrol/gasoline. The other aspect is that alcohol is highly detonation-resistant. In Fig. 11-2 we see the power curve given by an Esslinger Engineering alcohol-turbo engine. Note that the wastegate is only set at 15 psi and yet the engine produced close on 350 bhp. This type of engine, when used in competition, can have the boost screwed up appreciably to win. Duane Esslinger's competition experience has shown that the engine will go up to as high as 50 psi. Although it has never been dyno-tested at such high boost levels, because the engine life is relatively short, it is estimated that at 50 psi the engine gives in excess of 500 bhp.

FIG 11-2
POWER CURVE OF ESSLINGER ENGINEERING DEVELOPED ALCOHOL TURBO CHARGED 2000 ENGINE. PORT INJECTION CRANE P-332 CAM BIG VALVE HEAD AIR RESEARCH TURBO VI TRIM 0·96 AR EX HOUSING WASTE GATE SET TO 15lbs BOOST





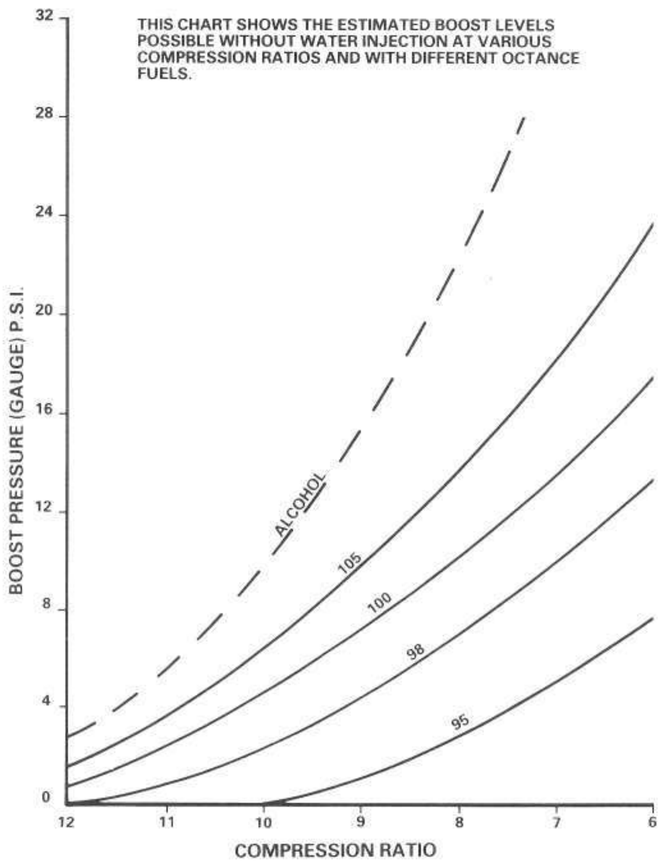
11-12. This was one of the early test engines I built together with Pinto racer Jim Flynn. It cranked out about 350 bhp at 8000 rpm on petrol (gasoline). This was enough to propel a 2200 lb car 0 – 118 in 12.09 secs. The car was equipped with slicks.

CYLINDER HEADS

The cylinder head used on a turbocharged car engine is of less consequence than that on a normally aspirated engine, but it still has a bearing on how much power the engine produces. A well modified head makes no difference to power at low rpm but starts to produce superior results above about 3000 rpm. Tests have shown that from about 4000 rpm up, more power can be built on the same boost, or more boost can be built with the same exhaust manifold backpressure. This means that less energy is consumed forcing the air through the engine.

To give you some idea of what a modified head is worth, it has been found that an engine developing about 200 bhp with a standard head, would give 220 bhp on the same boost and same compression ratio with a modified head. With exhaust back- pressure at its original level, the pressure gauge showed two psi boost more and power up to about 228 bhp. Interestingly, in lower rev ranges a modified head can often yield less power; this is seemingly due to two reasons, the first being the effect of fuel puddling. The rotary action of the impeller tends to centrifuge the fuel out of the air. This, of course, assumes it is a draw-through system. The second is that at low rpm, the higher exhaust backpressure which is present in the exhaust manifold of a turbocharged engine causes a greater amount of exhaust to cross from the exhaust manifold, through the combustion chamber into the inlet manifold. This situation occurs only under little or no boost conditions.

FIG 11-3



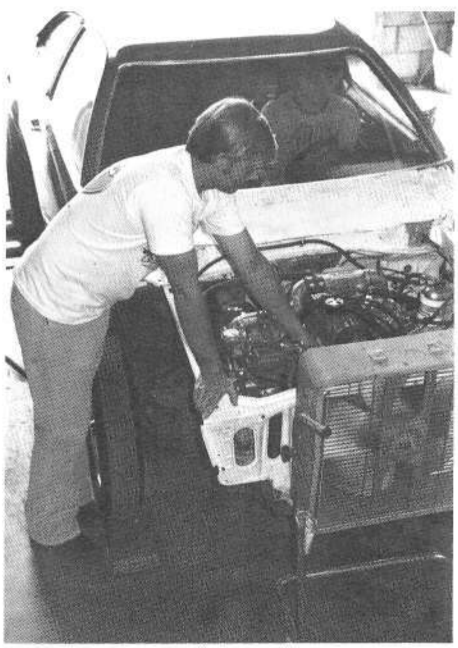
The effect of a modified head on low-end power is even greater when using a cam of a longer opening period than a standard cam. If you are concerned with limiting boost to a reasonable level, say 15 psi, a modified head to help increase the top-end power, together with a camshaft having reasonable lift but short timing to help the bottom end, works well. The subject of cams for turbocharged engines is covered in greater detail in Chapter three.

THE COMPRESSION RATIO

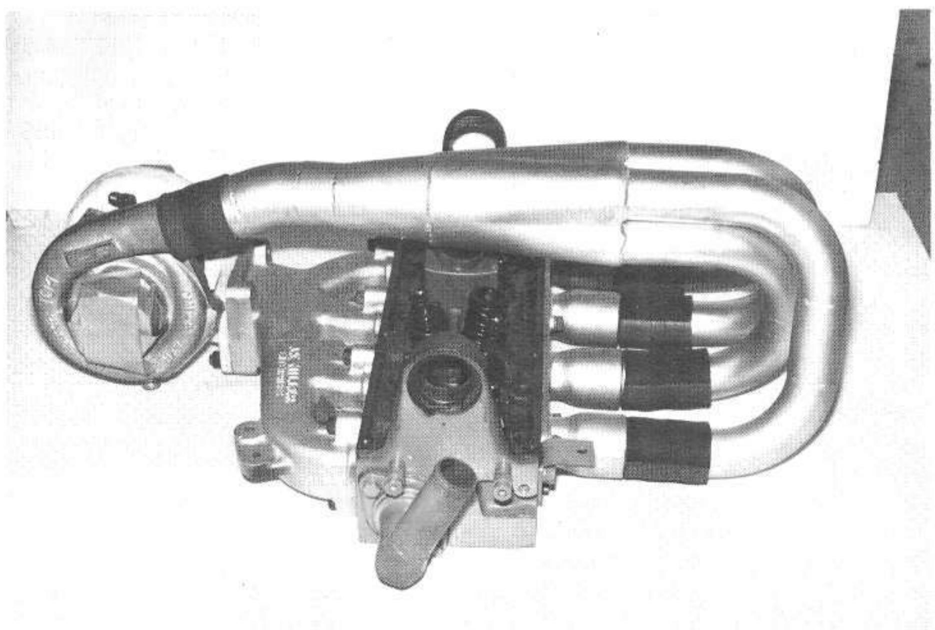
A typical Ford SOHC engine will stand about 10 psi boost with a compression ratio of 8.3:1 of 95-octane fuel before detonation sets in. Once detonation becomes a limiting factor, steps must be taken to cure the problem, or no further increase in power can be achieved. As higher boost is required, so the compression ratio must be lowered. Fig. 11-3 gives you some idea of the boost that can be used with different compression ratios and different octane fuels.

INTERCOOLING

To date, few people have taken advan-



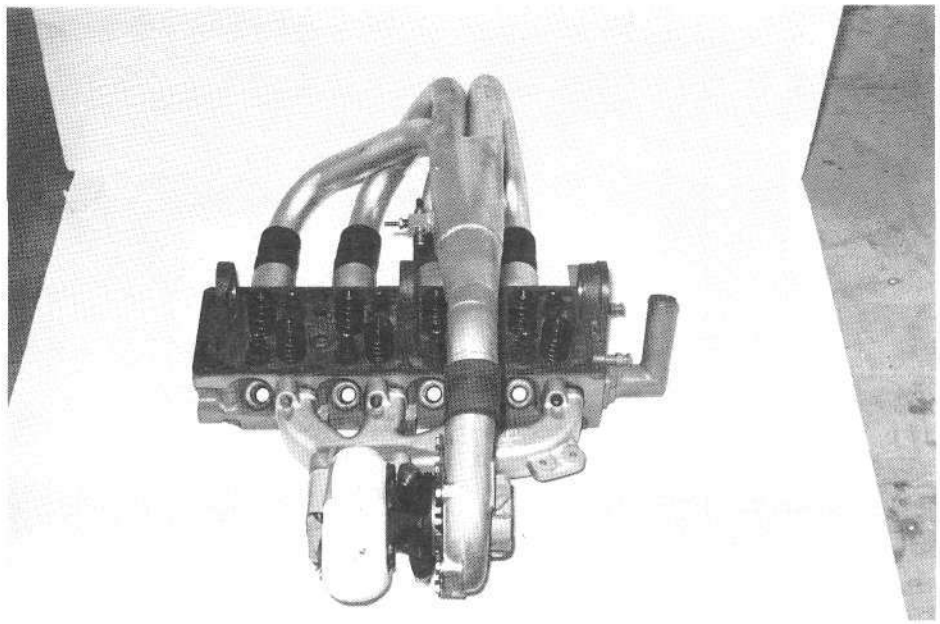
11-13. Jim Flynn's turbo pinto undergoes a full check out on the chassis dyno. Full power runs were not possible as the car would climb right off the rollers at about 150hp at the wheels. Unfortunately this power occurred at part throttle, just at the point when the engine came up on the cam. In spite of this, a great deal was found out about idle, transition and pump circuits. After dialling in these parameters, the car could be driven around just like an ordinary road car.



11-13a. This manifold, was an effort to overcome some of the deficiencies that the production two-barrel Holley/Weber manifold has. Under very high boost conditions, a mixture distribution appears. If alcohol fuels are used, the problem is compounded. Apart from this, long intake lengths from the plenum are used to increase drivability and power in the lower RPM range i.e. around 3500 – 4000 RPM.

tage of charge intercooling. It could well be that intercoolers present installation problems, due to their relatively bulky nature.

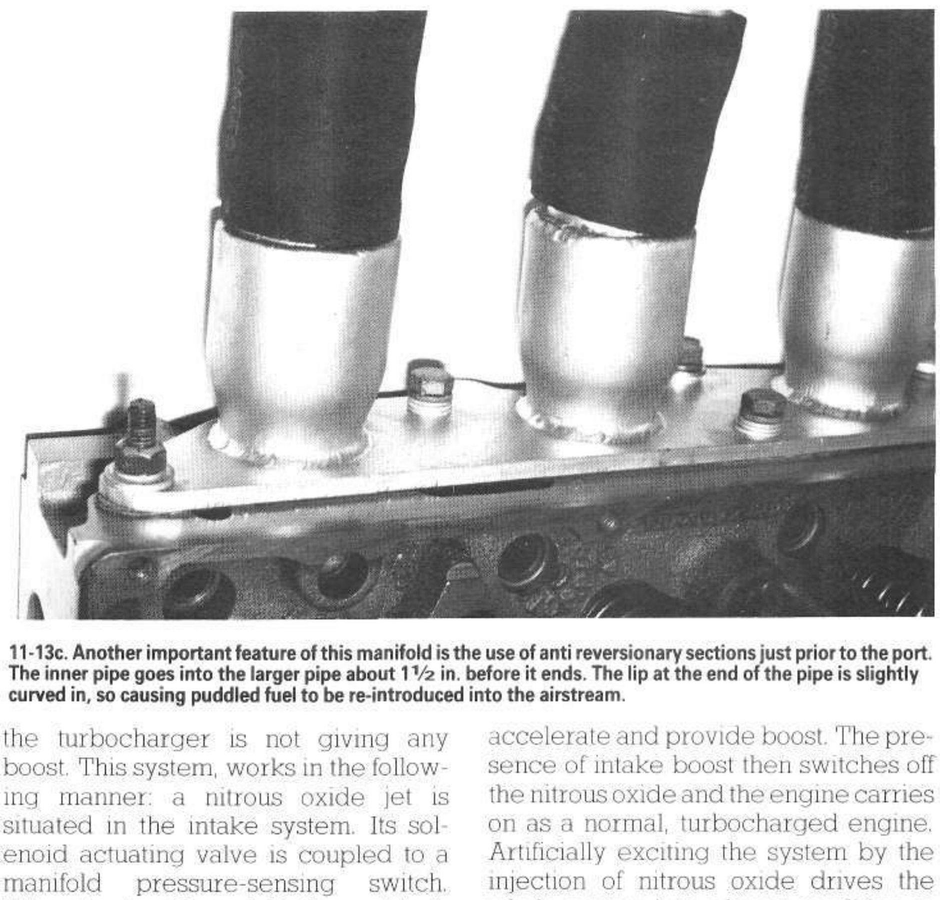
Another factor is the availability today of super-high-octane fuels; these effectively postpone detonation to the point of mechanical failure not from detonation but from the tremendous cylinder pressures produced. Trying to develop more power under these conditions is almost a waste of time until reliability problems are cured. Obviously if you are going for super-high power levels you must use the strongest components available such as steel cranks, titanium rods and specially strengthened blocks. At these levels, the blocks themselves are not able to withstand the tremendous cylinder pressures. The bores are literally ballooning and the blocks are bending, due to the loads imparted to the main bearing saddles.



11-13b. The placement of the tubes into the plenum is in rotational order. From this view, the induction pulses are counter clockwise.

TURBOCHARGER BOOST LAG

So far I have talked about the vast quantities of horsepower available by using a turbocharger. I have made no mention of ways to overcome boost lag. For the drag racer this is very important. So often a turbocharged engine fails to produce the results on the dragstrip simply because it will not launch the vehicle from the line fast enough to keep up with its normally aspirated or mechanically supercharged brethren. There are several ways around this. Some racers using Pinto engines are employing very heavy flywheels. I have heard of weights in the region of 90 lb. for the flywheel and 40 lb. for the damper on the front of the crankshaft. To launch the car, they are turning engines as high as 11,400 rpm and using kinetic energy to launch the car from the line. Needless to say, this is very hard on clutches and the bottom end of the engine. It has also been known for such engines to shed their 40-lb. crankshaft dampers. But there are other ways of dealing with lag. The two methods I propose here are ones I have had personal experience with and can vouch for their effectiveness.

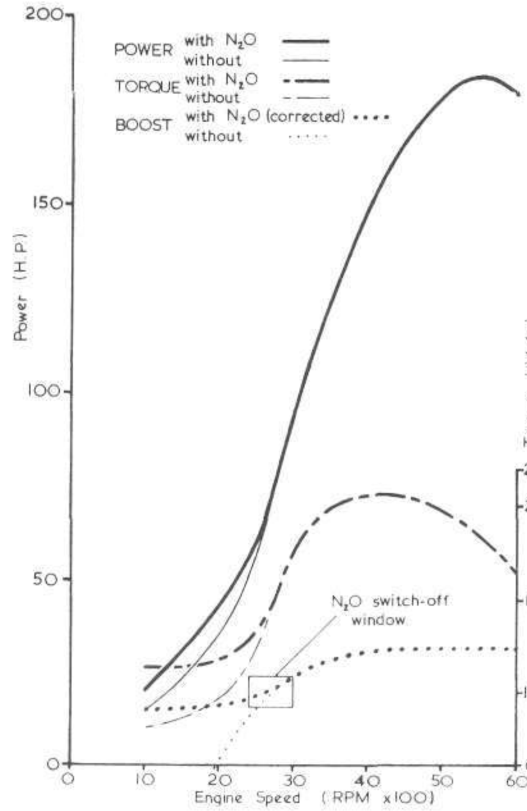


11-13c. Another important feature of this manifold is the use of anti reversionary sections just prior to the port. The inner pipe goes into the larger pipe about 1½ in. before it ends. The lip at the end of the pipe is slightly curved in, so causing puddled fuel to be re-introduced into the airstream.

N_2O BOOST LAG

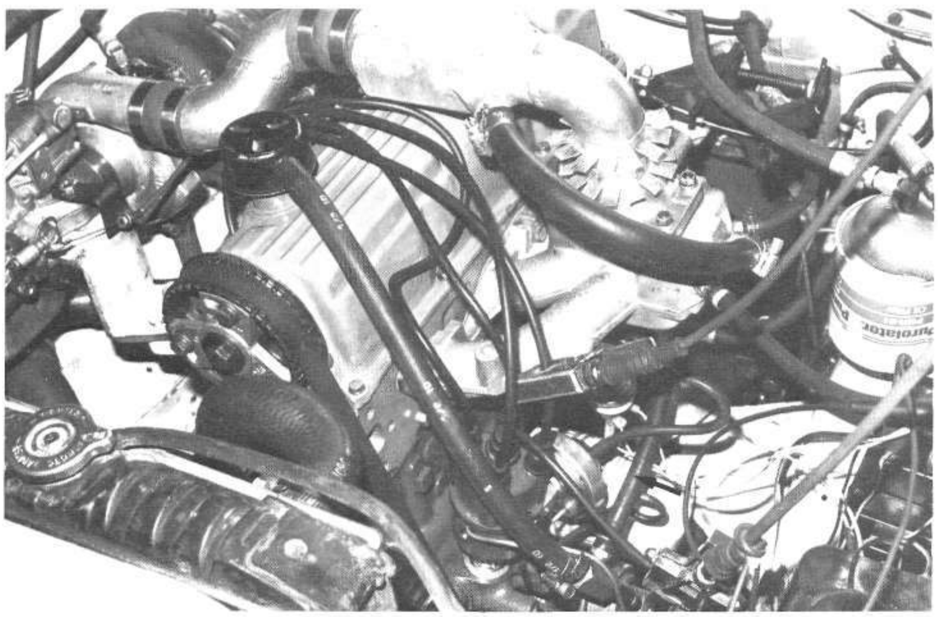
The first involves the use of nitrous oxide injection to cover the time when the turbocharger is not giving any boost. This system, works in the following manner: a nitrous oxide jet is situated in the intake system. Its solenoid actuating valve is coupled to a manifold pressure-sensing switch. When the throttle pedal is floored, both extra fuel and nitrous oxide are injected into the engine. While producing almost instant power, it also produces copious quantities of exhaust gas. When this hits the exhaust turbine, it causes it to accelerate very quickly. This causes the intake turbine also to accelerate and provide boost. The presence of intake boost then switches off the nitrous oxide and the engine carries on as a normal, turbocharged engine. Artificially exciting the system by the injection of nitrous oxide drives the whole system into a boost condition so quickly that the response time is *indistinguishable from that of a normally aspirated engine.*

Boost and backpressure measurements with and without nitrous oxide injection show what happens when the nitrous oxide is used to overcome turbo

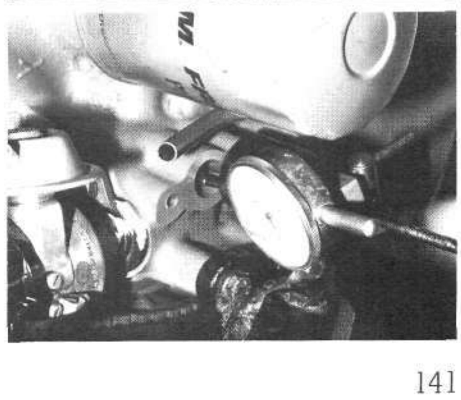


lag. Fig. 11-4 portrays what happens to unassisted and N_2O-assisted boost-equivalent. The boost-equivalent is the minimum boost required to equal the power output given by the N_2O injection and resultant boost. Remember, nitrous oxide has the same effect as boost. Our particular test engine used 0.05 lb. per second which provided the same power increase as 2.5 psi boost. All things considered, for a typical two-litre engine, 0.02 lb. per second of nitrous oxide injection would be equivalent to about one psi of supercharger boost.

An important advantage of the nitrous-assisted system is, that the wastegate becomes unnecessary. The reason is simple: since nitrous oxide takes care of turbocharger response, a larger size exhaust housing can be employed. This gives several advantages, among them lower back pressure for any given power level. This means part-throttle economy is better, plus it usually allows the production of a little more bhp per pound of intake boost in the higher rpm ranges. Since

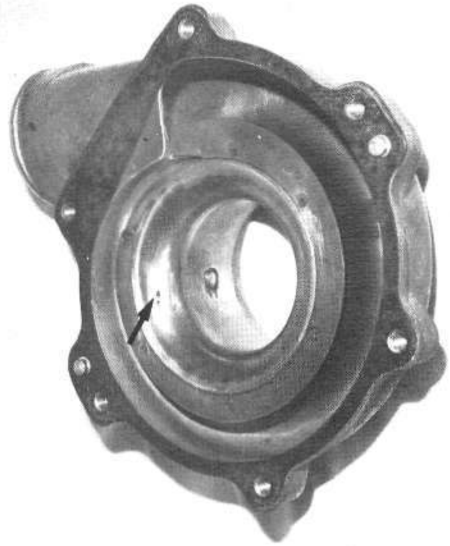


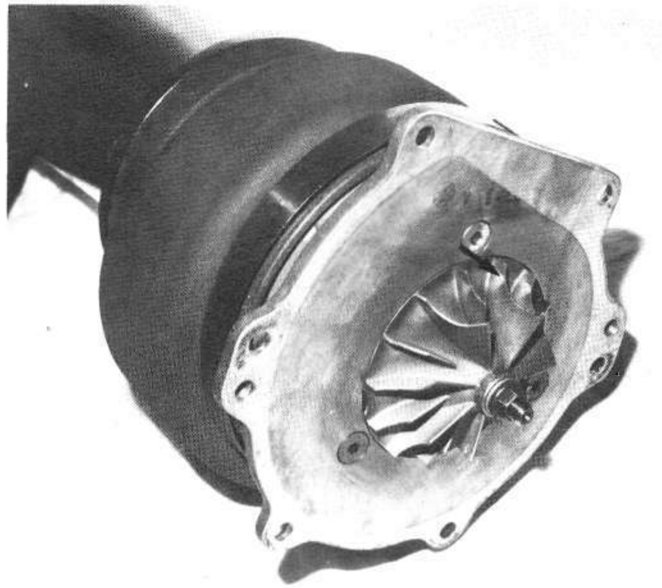
11-14. The distributor used here is a duel diaphragm unit using only the outside diaphragm. This gives 10° – 12° of retard as boost pressure builds up.



11-15. Duane Esslinger's turbo sand dragster. This alcohol engine cranked out 550 bhp. Later engines with port runner injection are producing even more hp than this and figures in the region of 620 have been spoken of. This dragster has an enviable track record and has twice been a major championship winner.

11-15a. The standard mechanical fuel pump has enough capacity to supply the needs of a 250 hp turbo engine. In some instances the full stroke of the fuel cam may not be utilized due to a spacing gasker which is too thick or a pushrod which is too short. On my own engines I set up the pushrod length so that the full travel of the fuel cam is transmitted to the fuel pump.

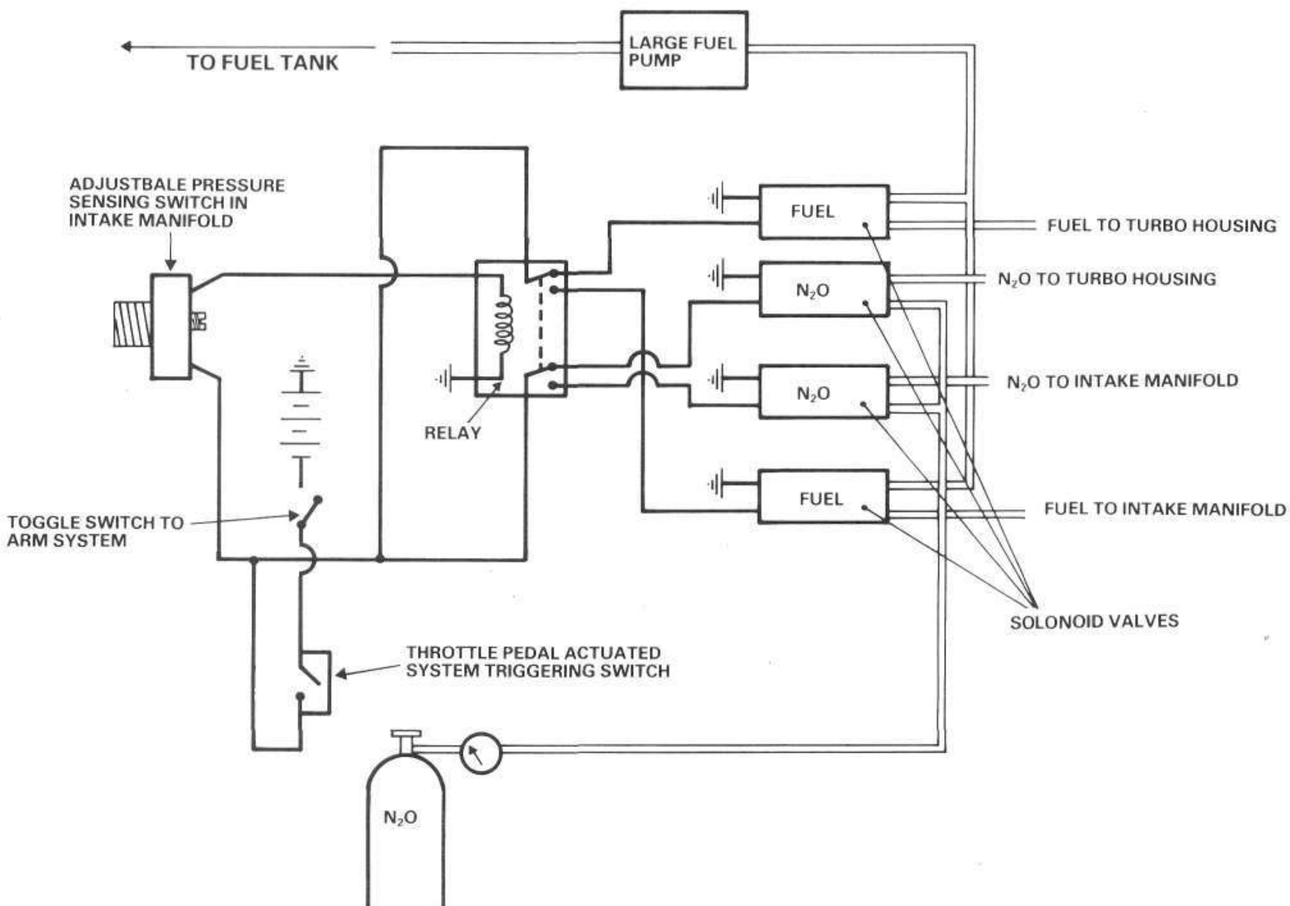


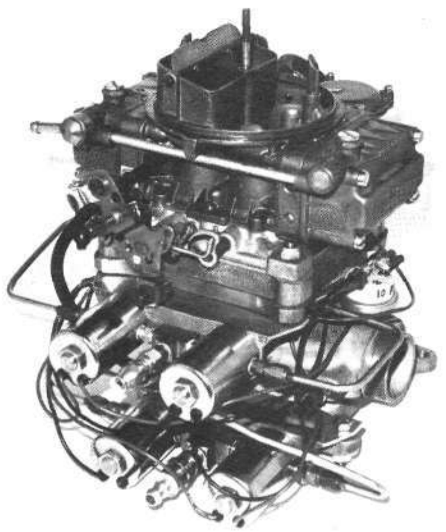


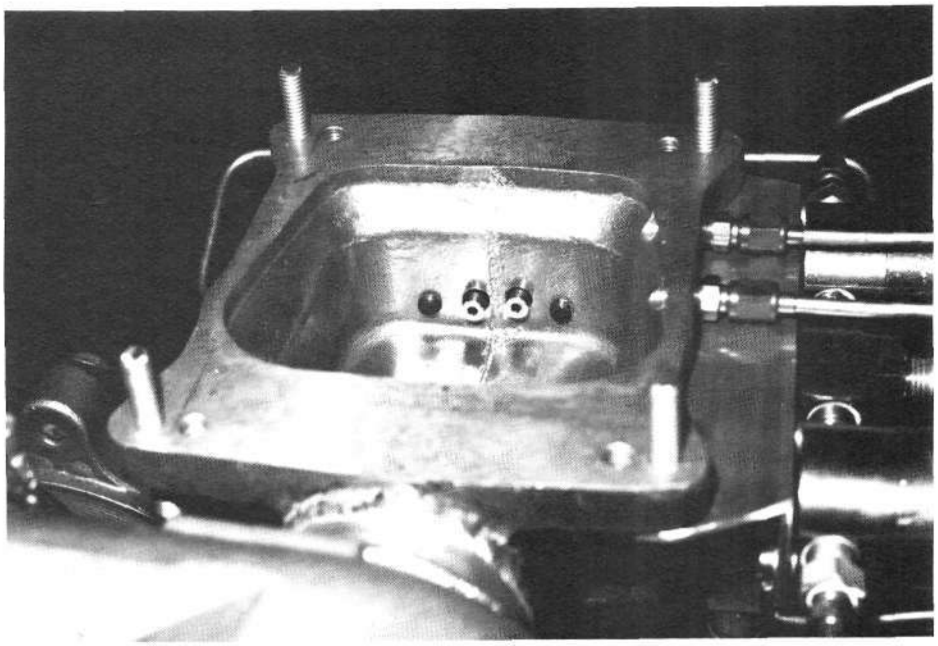
11-16. The best place to inject N_2O is into the intake turbo housing so that it impinges on the blade. Arrowed are the exit holes in the turbo case.

11-17. The light patch on the back side of the turbo blades show the point at which the steam of N_2O has been impinging.

FIG 11-5 DIAGRAM SHOWING WIRING & PLUMBING FOR NITROUS ASSIST SYSTEM

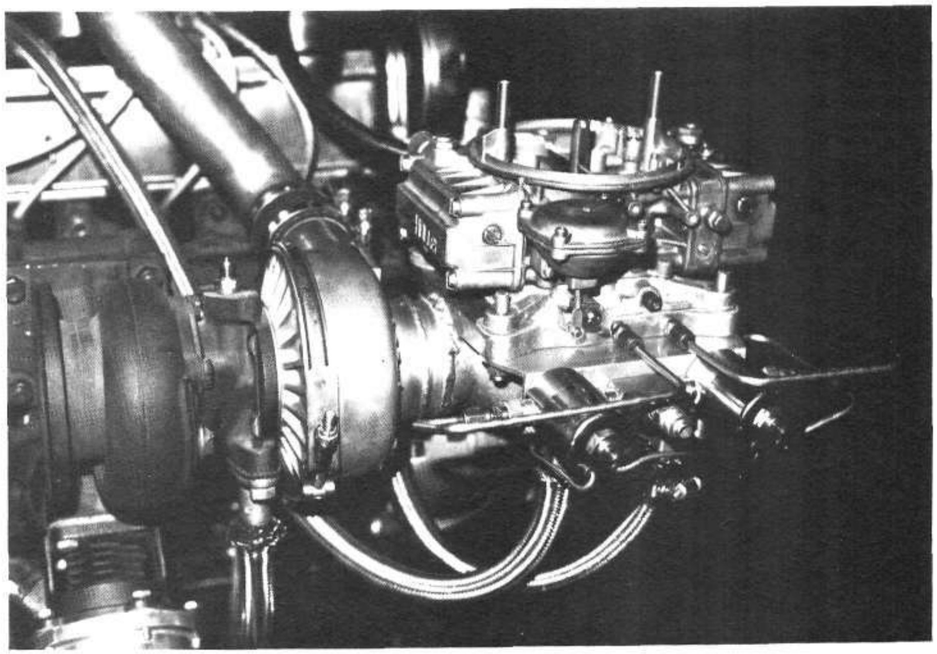






Above
11-18. An alternative to N_2O injection into the turbo blades is injection into the carburettor manifold. This seemingly complex setup is built to switch out the N_2O injection at 10psi. Dyno testing showed that this setup can produce the torque curve of a typical 5-litre (302 CID) engine. Fuel economy when driven normally, is nearer that of a 2-litre engine.

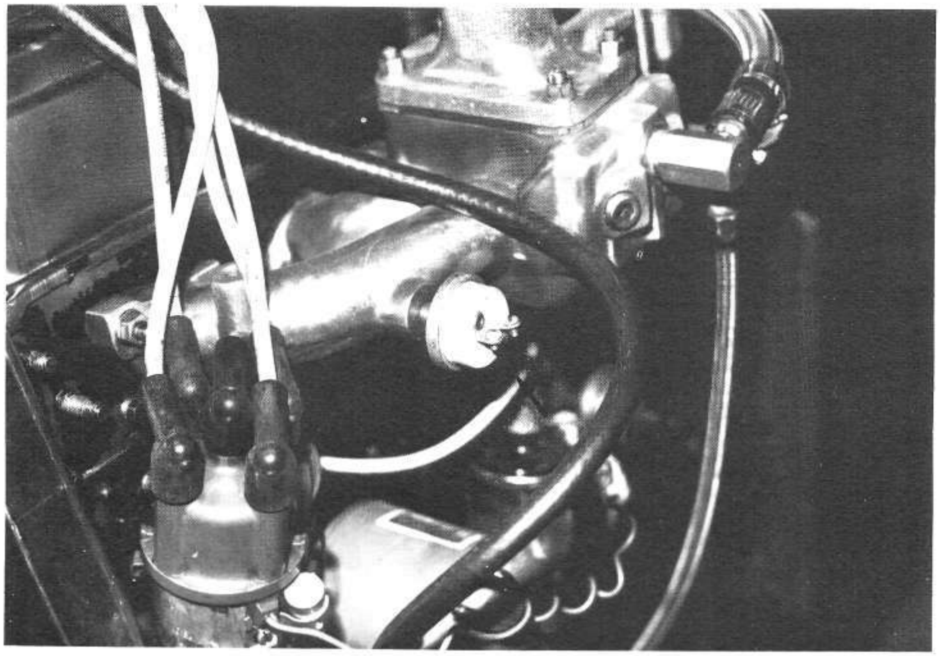
Above right
11-19. On this engine, the N_2O injection took place in the carburettor manifold only. On some race units when N_2O is on full time, injection is directly into the port runners along with fuel from the injection system. The injection system is calibrated to give enough "extra" fuel to go with the oxygen content of the N_2O.



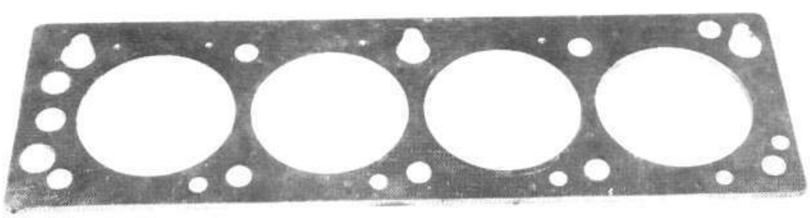
Right
11-20 Here is an N_2O assisted system all bolted to the engine. With 0.05lbs/sec injection and 20 lbs of boost and alcohol fuel, 350 ft lbs of torque can be seen with ease.

throttle response and low-end power are covered by the nitrous, the sizing of the exhaust housing need only take into consideration the boost required at peak rpm. This means, if you decide to go for a nitrous-assist kit, you can deduct the cost of a wastegate from your budget.

Apart from enabling the engine to develop boost instantly, nitrous oxide injection also has another advantage, even when boost is present. As has been previously stated, nitrous oxide, when released from its jet, is at very low temperature. The nitrous oxide injection can thus act as a charge cooler, and a very effective one at that. Even a small amount of nitrous oxide injected downstream of the turbocharger can have a profound effect on power output, more so than the same amount of nitrous oxide injected into a normally aspirated engine! The quantity of nitrous oxide that would give a 25 bhp increase on a normally aspirated engine will usually yield between two and three times the bhp increase on a turbocharged engine. When used to extract high power from a normally aspirated engine, the rate of nitrous oxide consumption is high by comparison with that used in a turbocharged engine. Typically, one lb. of nitrous oxide will produce 100 bhp increase for 10 seconds and, within reason, power outputs above or below this figure are proportional. On a turbocharged engine, the amount of nitrous oxide used is considerably reduced. If the nitrous oxide is used only to overcome turbo lag, then a 10-lb. bottle will last between 400 and 500 full-bore runs down the drag strip or about 60 laps of a typical race track. Fig. 11-5 shows an N_2O assisted system in schematic form.



11-21. Cut-out switch for the N_2O can be inserted into this already tapped hole in the standard manifold.



11-22. There comes a time when the CR cannot be lowered by any other simple means. At this point it is necessary to use a spacer plate such as this one shown here. This item is available from Esslinger Engineering.

ELECTRONIC BOOST LAG ELIMINATOR

This one hit me at one o'clock in the morning, and it's so simple I can't understand why no one else has thought of it. (Since putting pen to paper various magazines have erroneously accreditted this method of turbo lag ellimination too two other performance equipment companies. Fortunately both they and I know the true originator of the idea.) I have since used the system on a drag race Pinto. It actually pre-boosts the engine before the car leaves the line. In other words, while the orange light is on, there is boost showing on the gauge ready to launch the car down the strip.

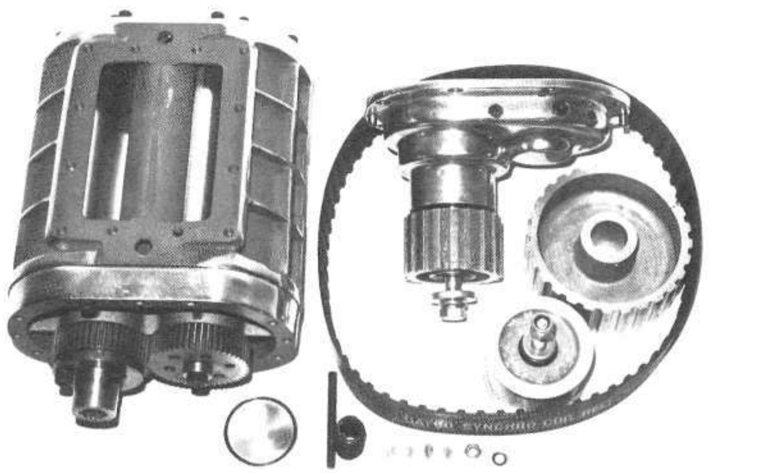
Here is how it works: a two-stage, electronic rpm-limiter is wired into the ignition system. The type used is the Autotronics Soft Touch unit. This brand is preferred because it limits rpm by degrading and retarding the ignition, rather than switching it off. This is prefered for the successful operation of the unit. Normally the Soft Touch unit is preset by means of a plug-in module to limit revs at one point. However, some changes are made here; an adjustable pressure switch is installed in the manifold. At a pre-set boost pressure, this switch changes the limiter from a lower-rpm-limiting mode to a higher one. The two rpm figures selected are (a) the one we wish to leave the line at, and (b) the max rpm for the engine. Next, a manifold pressure switch is preset to switch when boost increases, from the first rpm limit to the second, when the car leaves the line. Here is what happens: to leave the line, the driver merely depresses the throttle *right to the floor*. The engine starts to rev up in the normal fashion and then hits the limiter. Ignition is degraded or retarded as necessary but the spark plugs are still sparking. When the engine stabilizes at the rev-limiter's primary stage, all throttle butterflies will be wide open. Air is going through the engine at almost the same rate *that would occur if the car were actually running down the strip*. This means the engine has enough gases impinging on the exhaust turbine to actually cause boost, even though the car isn't going anywhere. The prototype unit allowed the engine to see eight psi boost in neutral; this is sufficient to launch the car in grand style. The manifold pressure switch is adjusted to change the limiter from low to high mode at just a little more than starting line launch boost. Under these circumstances at 8000 rpm going down the strip, our test car would show 15 psi boost. But on the line, at 8000 rpm in neutral, it would show only about eight. The principal reason is that the degraded ignition doesn't allow the volume of gas necessary to spin the turbine to make 15 psi. Dumping the clutch reduces rpm slightly, thus causing the ignition to come back to normal. This causes the efficiency (it changes from zero thermal efficiency to a finite number) of the engine to drastically increase. The volume of gas goes up and the next instant, the engine is producing 15 psi. This immediately switches the rev-limiter from the start-line mode, 8000 in our case, to the maximum rpm limit. The rev-limiter now operates as a normal unit to stop over-revving the engine. Basically, the whole process of achieving boost before we leave the line is done by degrading and retarding the ignition. The only snag I have found so far with this unit is that it is very sensitive to ambient atmospheric conditions. On cold, humid days, it makes considerably less pre-boost than on dry, warm days.

Another factor which affects it greatly is the air/fuel ratio when it is in the start-line pre-boost mode. The wide-open-throttle rich mixture used to cool the exhaust reduces the effectiveness of the ignition retard. In some cases this problem can be simply overcome. On the test car, a Holley four-barrel carburettor with vacuum secondaries was used. Vacuum was tapped from the primary barrel and routed to the power valve via a solenoid valve. Like the ignition primary rev limiter, the solenoid valve was coupled into the manifold pressure sensing switch. When the throttle was opened venturi vacuum would be routed to the power valve, thus keeping it closed. This fed the engine on a much leaner mixture. This produced at least four psi more pre-boost.

When the clutch is realeased, engine rpm will slow down slightly, ignition quality and timing would return to its normal state, and boost will come up, all causing the rpm limiter to switch onto its second mode. At the same time the solenoid valve ceases functioning, thus leaving the power valve to sense boost pressure. In turn, this allows the power valve to come into operation and richen the mixture for maximum power.

The system just described works very well for a drag race car. It produces so much starting line power that instead of turbo lag hampering the car, the engine produces so much low-end power that coupling it up to the track has proved to be difficult. So the problem of turbo lag for the drag racer appears to be one of yesterday. But just how much this affects the lot of circuit and rally racers remains to be seen.

The electronic turbo lag eliminator I have described works only in a sequence beginning on the starting line. It could not be directly applied to a circuit car but there is a technique that it may be possible to use, although I have not, as yet tried it. If the distributor were modified so that the base plate could turn maybe 70° or 80°, it could be actuated by a cable to retard the ignition. In turn, this would be coupled to the accelerator pedal. The throttling necessary to control the vehicle could now be achieved by advancing and retarding



11-23. Quality engineering goes into the Magnusson "Magnacharger". Relatively high compressor efficiencies have been realized with this unit and it has achieved notable success as far as many world records are concerned.

the ignition, not by opening and closing the carburettor. *Once out on the race track*, the throttle would stay open all the time. The amount of power produced by having the ignition retarded 70° or 80° is minimal to zero, but the boost may still stay around seven or eight psi. Slamming the ignition back to about 28° instantly produces horsepower, almost instantly causing the boost to build to wastegate pressure. Although this system is untried, I'm sure it will work. Further, I should point out that on very fast tracks, Silverstone for example, turbocharger boost lag is really of no consequence. It's only on twisty courses where the driver is continually on and off the throttle, very busy with steering and gear-changing that boost lag becomes a time-loser. Under these conditions, I am sure the boost lag eliminator would make the engine respond exactly like a very powerful and torquey normally aspirated engine.

MECHANICALLY DRIVEN SUPERCHARGERS

I have had virtually no personal experience with mechanically driven superchargers on the Ford SOHC engine. I haven't tried them simply because of the success of turbo units I have been involved with. The mechanically driven blower has been around many years but has been superseded, to a large extent, by the turbocharger. This does not mean that a mechanically driven supercharged SOHC Ford cannot successfully be built. It's just a question of choosing the supercharger for the job and its application. There have also been new developments in positive-displacement superchargers which make their application a little more desirable than just a few years ago. Also, consider that such an installation may prove to be a viable proposition in terms of cost and ease of use. All these factors have to be taken into account.

Another point worth considering is that a positive-displacement supercharger works better than a turbocharger when used in conjunction with an automatic transmission. This is because a positive-displacement supercharger can provide lots of extra torque low down. Though this is possible with a turbocharger, by one of the means I have outlined previously, it is by comparison cumbersome and involved.

Let's just look at a few superchargers. First there is the Allard-Shorrock supercharger. This vane-type of supercharger has been around many, many years. It can best be described as a blower giving moderate bhp and torque increases without increasing the rpm at which it occurs. Specifically, torque increases between 25 percent and 40 percent have been seen. The Shorrock supercharger, however, should be avoided for high-rpm use; it needs considerable doctoring to avoid scoring the case and the vanes. But it does a reasonably good job if a rev limit of about 6000 rpm is observed.

A Rootes-type supercharger kit for Pintos is distributed by Speedway Motors in Lincoln, Nebraska. It is a very comprehensive assembly and well worth looking at. This kit boosts torque and horsepower by about 40 percent and contains all the parts necessary to make the conversion. Costing around 80 percent of what a turbocharger costs, it has a good horsepower-per-dollar ratio.

A third supercharger is the Magnuson unit, designed by Jerry Magnuson in California. This supercharger looks very promising in that it is very efficient for either a vane-or Rootes-type supercharger. Due to subtle and sophisticated design within the supercharger, the efficiency of the Magnuson unit is claimed to approach 60 percent. The results achieved by some racers using this supercharger are nothing short of outstanding. I suspect that a 300-bhp petrol/gasoline-burning, two-litre Pinto engine is possible, using the Magnuson supercharger. This unit is built to very close tolerances with a high standard of workmanship. It's capable of withstanding as much as 18,000 rpm, and it's expensive, with just the blower costing in the neighbourhood of $2000. (£1400)

Unlike turbocharging, a positively driven, positive-displacement supercharger will give instant power, regardless of engine speed. In fact a seven-or eight-psi boosted engine will feel more like three-litres rather than two-litres. As with a normally aspirated or turbocharged powerplant, a blown engine using a positive-displacement supercharged responds more or less to the same modifications, namely cam requirements, head requirements, ignition and so on. The main requirement of a positive displacement-type supercharger is that the exhaust system needs to be free-flowing. As a matter of fact, it is sometimes beneficial to make the exhaust valve maybe .060 inch larger at the expense of a similar amount taken from the intake valve. The reason for this is that there is a lot more exhaust to get rid of.

Big exhaust pipes are a must. And as far as length is concerned, determine that which will allow the exhaust system to work optimally at the rpm the engine is designed for.

Building a C4 Automatic Transmission for a High Output Engine

By Keith Roof

Introduction by David Vizard.

*I think it will be evident to most readers by now, that my own field of endeavour is engines. However, I felt there were so many cars using the Ford SOHC engine that also utilized automatic transmission, that it would be a failing of the book not to include transmission modifications which would help the owner of such a vehicle to make the best of a more radical engine. In many places I have talked of modifications which would allow an engine to produce more power yet still use the standard automatic transmission. However, the situation must be looked at for those of you who wish to go a step or two further than this. Personally, I am not an expert on automatic transmissions, and so to cover this aspect I asked **Keith Roof** of Arizona Transmissions to look into ways and means of making the standard C4 automatic transmission able to give us more by being adapted and modified to suit the characteristics of more radical engines. Keith's experience in this field made him an ideal candidate for this task. He has built numerous transmissions for race and championship winner cars requiring or needing an automatic-type/transmission of one sort or another. He made short work of developing the C4 transmission into a bullet-proof one for a Ford SOHC powered car, and so at this point I am going to hand the show over to Keith and let him tell you what needs to be done and how you should do it. D.V.*

BASIC CONSIDERATIONS

When modifying an automatic transmission, there are two basic considerations we try to work with. The first is to tailor the transmission's characteristics to suit the power curve of the engine. The second is to build in reliability. In our case, these two subjects are so tightly integrated that we will deal with them virtually concurrently as we go through the C4 transmission. Many of you may not be familiar with the workings of an automatic transmission. I will tell you all the modifications necessary to allow your transmission to function correctly with a high performance engine. This will entail completely rebuilding the transmission. This book is not intended to be an overhaul manual, so if you are not familiar with automatic transmissions, I suggest you get a workshop manual on the C4 transmission. I will discuss only the modifications necessary in a step-by-step rebuild of the transmission.

THE TORQUE CONVERTER

Any time peak torque is moved up the rpm range, the required torque converter characteristics also change. If the torque converter has too low a stall speed (the maximum speed it will run to if you put it into first gear and load it against the brakes) it will prevent the engine from getting "on the cam". This results in a vehicle that is slow from the lights. This has the same effect as letting out the clutch at too low an rpm in a manual transmission car. This bogs the engine into an rpm range where it doesn't produce enough torque to suitably accelerate the car. A more *slippery* or *loose* torque converter must be used. Such a unit allows the engine to produce enough torque to launch the car properly. A higher stall speed is usually achieved through using a smaller torque converter. Many manufacturers modify existing torque converters from other vehicles to produce higher stall speeds. But there is a limit on just how small they can go to produce high-stall-speed torque converters for small-engined cars such as the Pinto. Indeed, at the time of writing, it is not practical to build a torque converter having a stall speed higher than about 3400 rpm behind a strong, normally-aspirated engine. Of course, stall speed also depends on the torque output of the engine; if the engine is turbocharged then its stall speed will likewise be higher. Several high-stall torque converters were tried in one of David Vizard's test Pintos and the results gained from this was worth noting.

Automatic transmissions and small engines are not as good a combination as automatic transmissions and big engines. The reason is that small engines cannot afford to give away any horsepower to drive the transmission. Big engines, however, don't really notice the loss of 20 or 30 bhp absorbed by the transmission.

To make effective use of a high-stall-speed torque converter, the rear axle ratio must be lowered or the car lightened considerably. This is because the ratio normally used with an automatic transmission causes the engine to run "on the converter" for a considerable time. In other words, when the throttle is floored the engine comes up to a little over stall speed and then hangs there until the speed of the car approaches converter lockup. Besides causing transmission oil to heat up considerably, it doesn't accelerate your car very quickly. The cure is to lower the axle ratio. Typically it would be changed from, say, 3.45 down to around 4.7:1. If the car is intended strictly for racing, then the ratio may go lower than that, in extreme cases as low as about 6.1.

Another snag with high-stall torque converters is that they are not very economical on the road. At normal driving speeds they can still be in their high-slip mode. As a result, fuel economy can drop drastically. Couple this with the fact that the overall gear ratio may also have to be lowered, and mileage may suffer by as much as six or eight mpg. The increase in performance, however, is dramatic.

There are two ways of increasing the stall speed of a torque converter. One way is acceptable, the other way is not. Some companies will increase the stall speed of the torque converter by cutting the case, opening up the converter and spacing the turbine and stator a little farther apart. This increases the stall speed by virtue of a reduction of coupling efficiency. This method is not the best way to go.

Another way of increasing stall speed is to adapt parts from a smaller torque converter to your specific purposes.

This was done with one of the torque converters used for experiments in the project car. A custom torque converter was built by A1 Transmissions in California with a stall speed of about 3200 rpm behind a hot street engine. This stall speed was selected with the engine's torque curve in mind. In this case the stall speed is around 300-500 rpm less than the torque peak. If you have a two-litre engine, probably the longest cam you could use with a 3400-rpm converter would be about 295 degrees duration. Aim for a cam here that will give maximum torque at about 3700-3900 rpm, maybe 4000 rpm at most.

Installing a high-stall torque converter is a straightforward job but a few precautions should be taken. First, always flush out the new torque converter thoroughly. Make sure every little granule of welding grit is out of the torque converter; sometimes particles are trapped in there. Remember, it takes only one particle to go through your transmission to clog a valve or ruin a bearing. One particle of grit can wreck a whole automatic gearbox.

Secondly, use a dial indicator to check the snout of the torque converter for truth and concentricity, otherwise seals will wear quickly. If the snout is not true, send the torque converter back to the manufacturer and ask for a replacement. Apart from these two precautions, installing your high-stall torque converter is merely a matter of replacing the standard one.

THE GEARBOX

As it comes from the factory, the C4 transmission fitted to the Pinto is a pretty stout unit. If you intend to run a normally aspirated engine up to around 170 bhp, the transmission is strong enough to cope with the job. If you intend to run an all-out, normally-aspirated engine, or a turbocharged engine having much more torque than a normally aspirated engine, you should modify the transmission accordingly. What I suggest here may be viewed as overkill. But I assure you that if you build the gearbox exactly as shown here, it will be able to withstand even the wildest turbocharged two-litre engine. A transmission like this has proved perfectly adequate behind a 500-bhp V8 engine, so it should be able to take anything that a two-litre engine can put out.

I am going to assume that you have removed the transmission from the car, thoroughly cleaned the outside of the case and then stripped it down. At this point, I will describe how to reassemble the transmission to make it as bulletproof as possible. Trying to second-guess where all the parts go in an automatic transmission is almost impossible, if you have no experience. This is because the function of all those parts is not apparent until the operation of the transmission is completely understood.

For those who are not transmission experts, I suggest that you work closely with the accompanying photographs. Follow the rebuilding sequence as described in the text and the numbered photographs.

REBUILDING THE TRANSMISSION

Your very first job, before attempting to put any parts together, is to go out and buy a master transmission rebuilding kit for a C4 transmission.

MAIN CASE

Photo 1. Your first job is to inspect the main case for cracks and stripped screw threads. Next, inspect the bushing indicated by my pen in photo 1. These bushings seldom wear out, unless the car has been towed incorrectly. If it is worn, replace it. In photo 1, you will see that some components are already installed on the case, namely the Park-Pawl arm, and not so easily seen, the outer ring of the sprag. This is directly under the bearing I am indicating inside the case. Check this item for an signs of wear. The last check: make sure that all mating surfaces are free of burrs and, of course, the whole case must be *immaculately clean*, as with *everything* concerned with automatic transmission rebuilding.

REAR SERVO

Photo 2 shows the installation of the rear servo piston. Before installing it you must make sure that the bore the piston runs in is not scored. The piston itself must also be in perfect condition. If there are any wear marks on its edge, it should be replaced. The fit in the bore must be good enough to allow only the tiniest of air leaks. An easy way to check the piston fit is to smear automatic transmission fluid around the edge of the piston and the bore and then push

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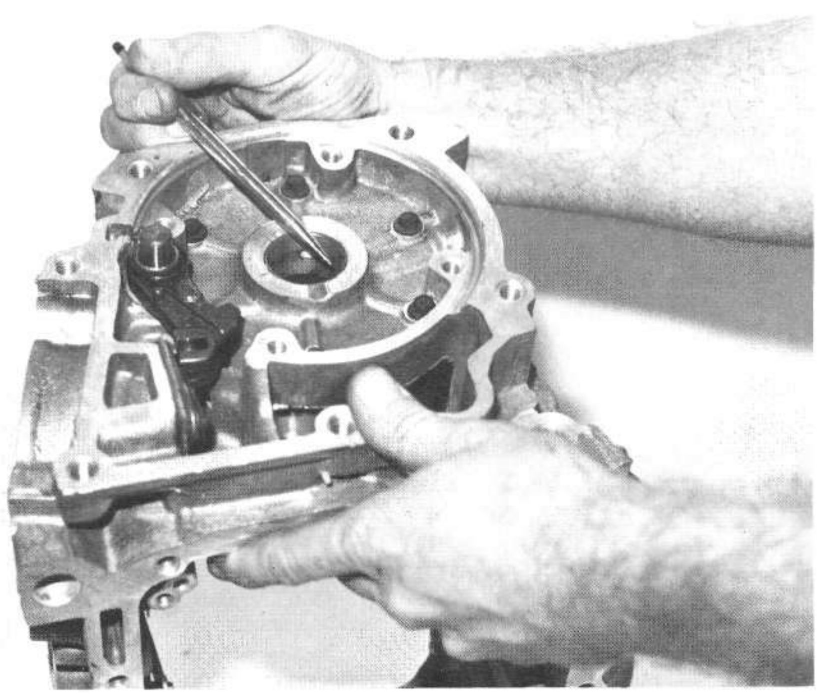

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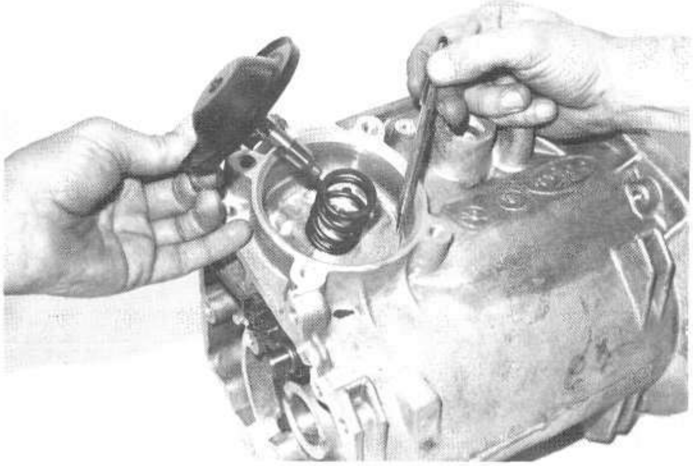

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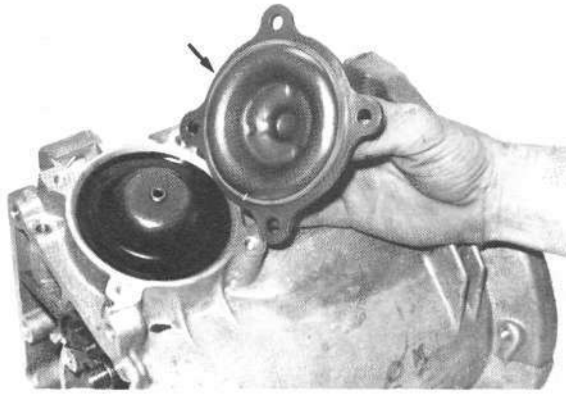

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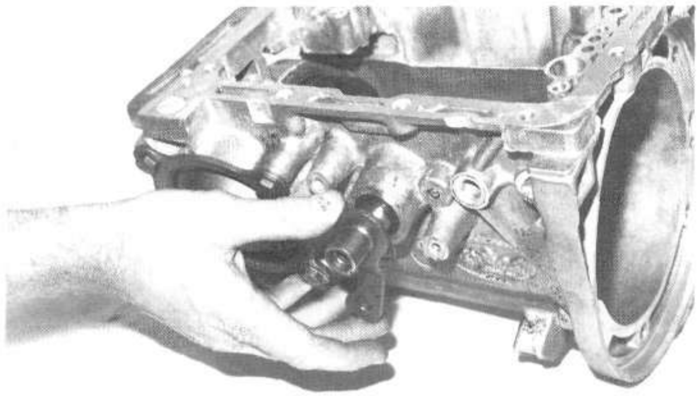

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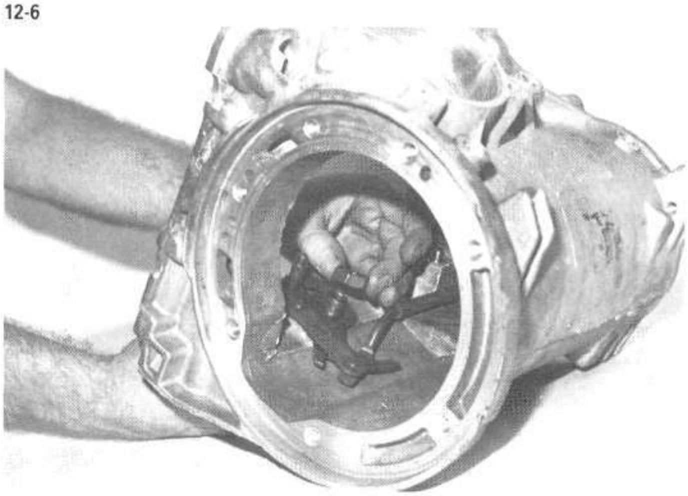

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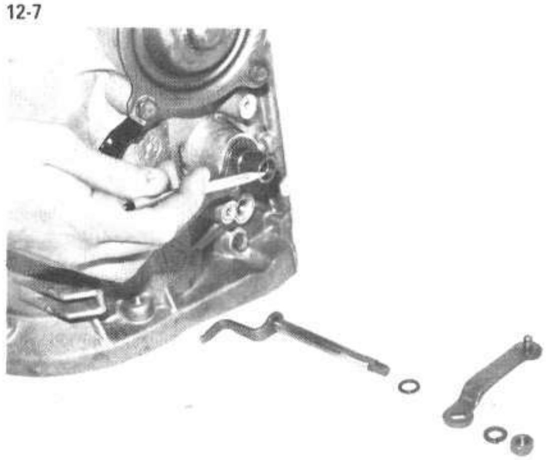


the piston into its bore. Next, cover the exhaust port with your finger and try to push the piston out. Because of its good seal, it should be very hard to push out.

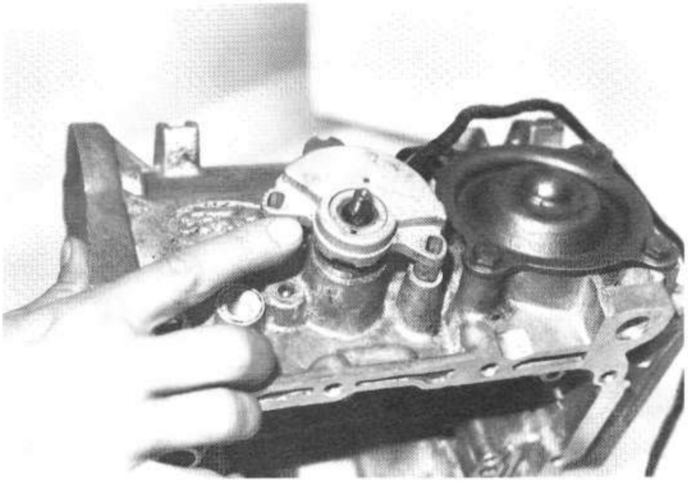
Photo 3 shows the installation of the rear servo cover. This must be installed with a new seal, which is indicated by the arrow. Do not use any sealer here.

Install the linkage seal as shown in photo 4. A little silicone sealer around the edge of the seal is a help. The seal can be installed with thumb pressure, the big groove going downward. Push the seal in until it is just below flush with the edge of the casing. About .010 inch down is just right. If you push the seal too far, use a wide-blade screwdriver to pull it back up. Remember, it's imperative not to damage the seal, otherwise your transmission will have a serious leak.

Insert the first part of the transmission linkage as shown in photo 5. Photo 6 shows what the linkage looks like inside. Install the nut shown here with a little Loctite to ensure that it stays put. Tighten the nut to about 30-40 ft./lb.

Photo 7 shows the order in which the kick-down linkage goes into the main part of the linkage you have just instal-

12-8



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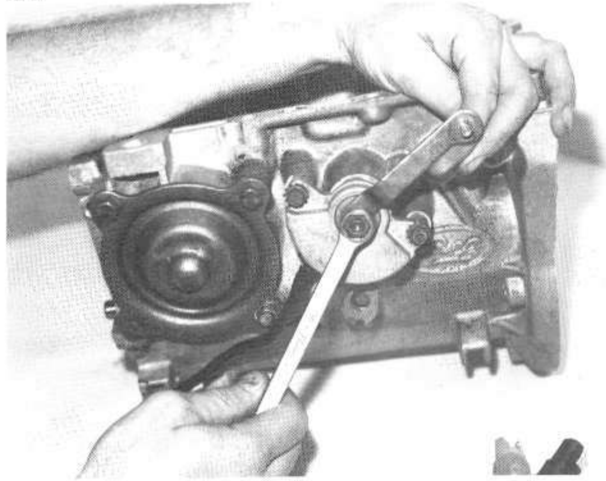

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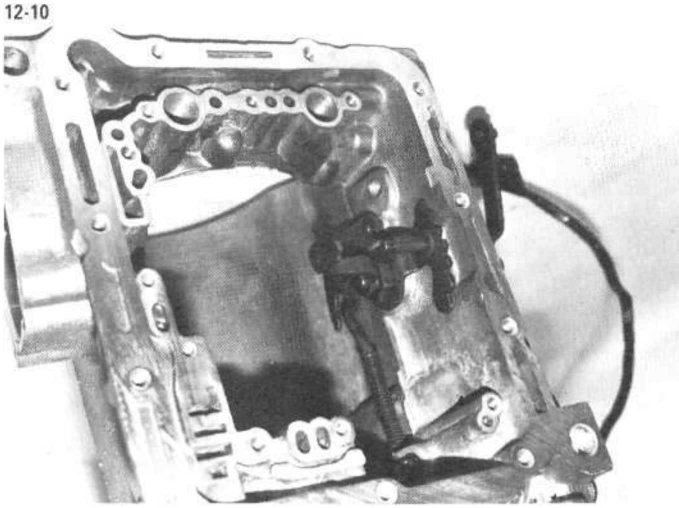

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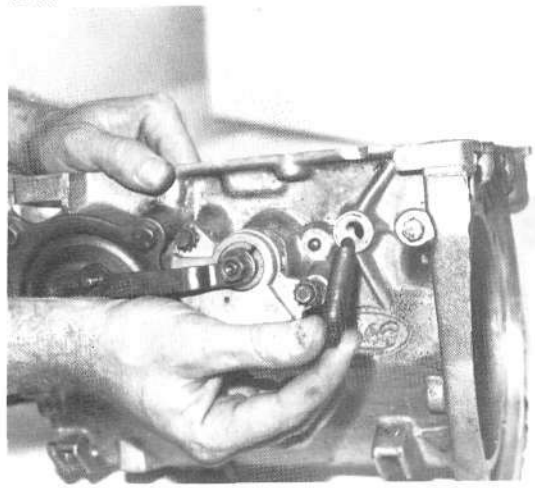

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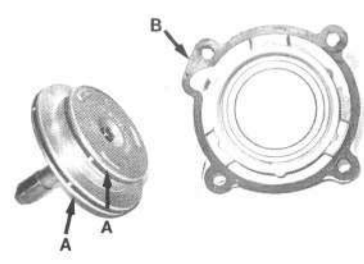

led. But before you install the lever on the shaft, be sure to put on the neutral safety switch as shown in photo 8. Do not tighten the switch, just place it in position, then install the kick-down arm as shown in photo 9. The neutral safety switch will need to be adjusted in position later on, so the two screws securing it need only be finger-tight at this stage.

Photo 10 shows what the linkage should look like inside the case.

FRONT BAND ADJUST

Screw in the front band adjusting screw as shown in photo 11. Do not install the locknut yet. This should be done when it comes time to adjust the bands themselves.

FRONT SERVO

There are many types of gaskets and seals in a master rebuilding kit, so your first job is matching of the old seals with new so you are sure you have the right ones.

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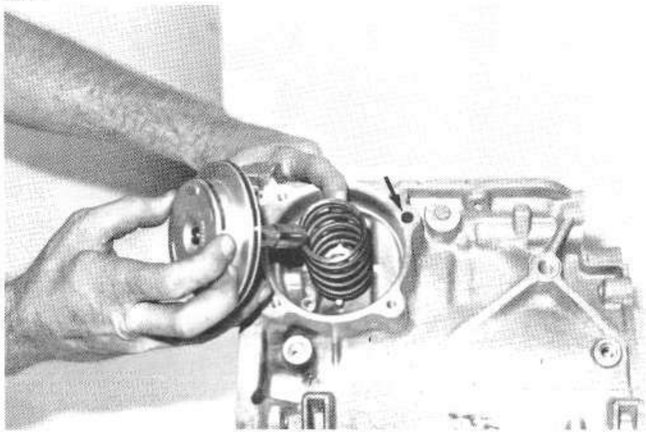

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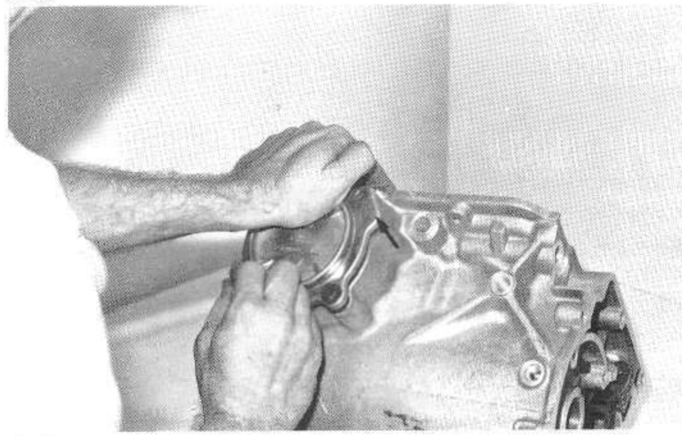

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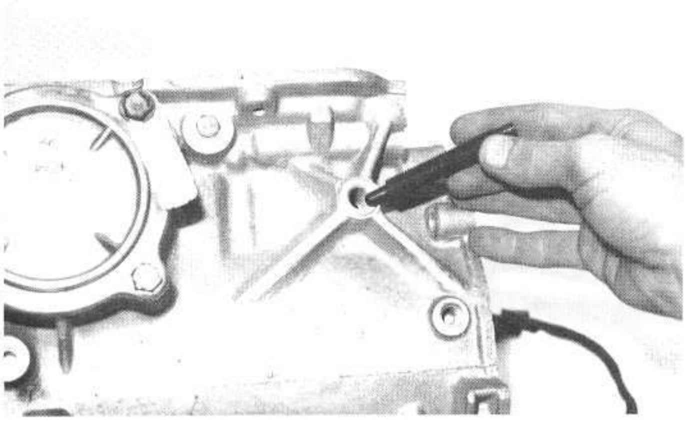

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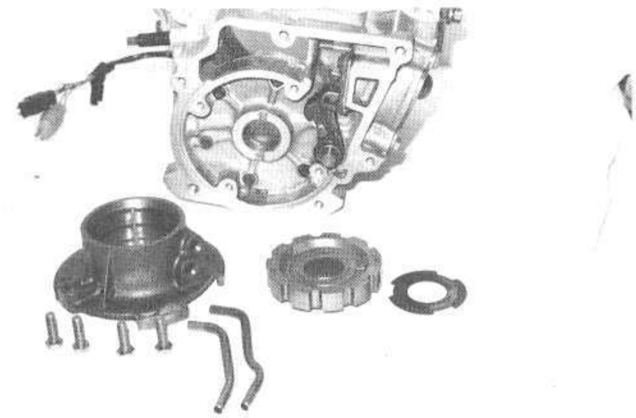

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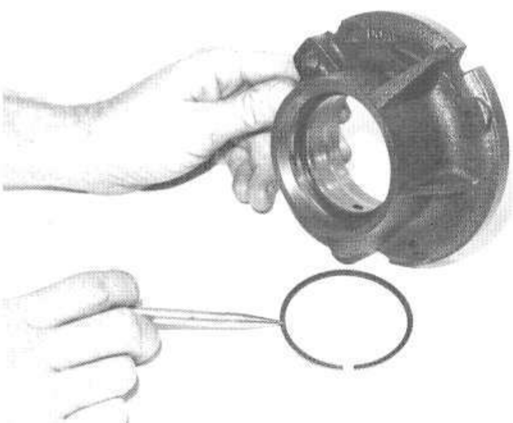

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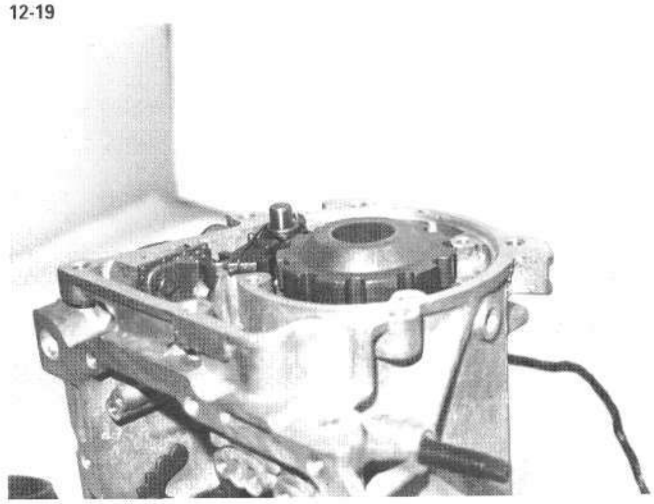


Refer to photo 12. This is the first point where we deviate from standard rebuilding procedure. The front servo we are going to use is the type used in the C4 transmission that backs up a 302 V8 engine. We use the seal cover and piston assembly from the C4 transmission, plus the 302 C4 spring. Photo 12 shows the difference between the C4 front servo for a four-cylinder application (left) and the C4 front servo from 302 V8 applications that we will be using (right). Install the seals on the servo piston as indicated by arrows A in photo 13 and locate the gasket on the cover, arrow B of the same photo. Lubricate the servo seals with automatic transmission fluid.

Install the spring, piston and cover as shown in photos 14 and 15. Make certain that the exhaust port in the case is aligned with the cutout in the cover as indicated by the arrow in photo 15.

Install the rear band adjusting screw (photo 16).

PARK ASSEMBLY

Our next move is to install the components shown in photo 17. These comprise of the thrust washer, park gears and oil distribution sleeve assembly. Refer to photo 18. Install the rings in the oil distribution sleeve and check them for end gap and sideface wear. The end gap should be no more than 0.006- 0.010-inch. If there is any visual sideface wear replace them.

Grease the washer and locate it in the case, then install the park gear, as shown in photo 19.

Install the oil distributor sleeve and oil pipes, as shown in photo 20.

SPRAG ASSEMBLY

The next components to go into the case are those shown in photo 21. Before assembly it will be necessary to check on some of these components. First, check the thrust washer for wear. If there is anything but a minimal amount of wear, replace it. Next, look at the accordion springs for fatigue cracks. If there are any, replace them. Now check the outside diameter of the roller race and the rollers for cracks, scuff marks, scores or burn marks. Unless they are in perfect condition, replace them.

The back of the thrust washer is smeared with grease and installed facing the inside of the transmission. Insert the race cage, then insert the race with the relieved side down. Position the rollers and install the springs. When you have done all this, the assembly should look the same as that shown in photo 22. Once assembled, lubricate with automatic transmission fluid.

MAIN SHAFT

Check the oil filter screen on the flange of the mainshaft to see that it is free of debris and damage, as shown in photo 23.

Photo 24 shows the next phase of rebuilding the mainshaft, installation of the new rings. These rings do not need gapping. After they have been installed, lubricate them with automatic transmission fluid. Install the mainshaft in the oil distribution body as shown in photo 25.

GOVERNOR

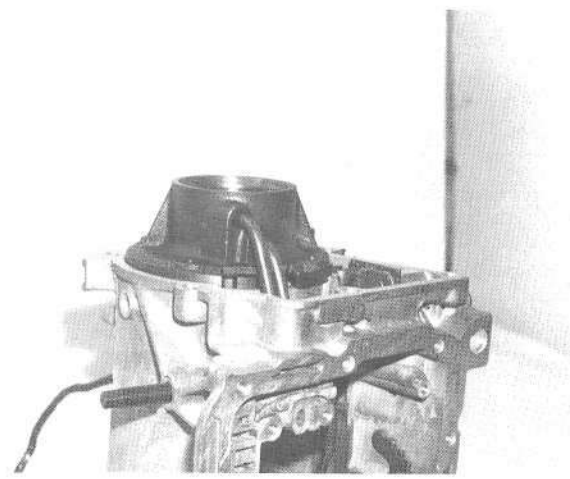
Photo 26. Here you see. one of the springs, retainer clips and governor weights removed from the governor housing. This particular one has been removed from the left side of the governor. The one in the right side is the standard weight spring assembly. The one to go on the left side should be changed from the stock one to a C4 302 Mustang secondary governor spring and weight. Before assembling this unit, polish the valves and then reassemble it. Lubricate the governor with automatic transmission fluid and make sure all weights are free to slide in the housing.

Install the governor assembly on the mainshaft as shown in 27.

REVERSE, LOW BRAKE DRUM & PLANET GEAR ASSEMBLY

Sort out the components that are shown in front of the case in photo 28, from your pile of parts. Make sure the outside

12-20



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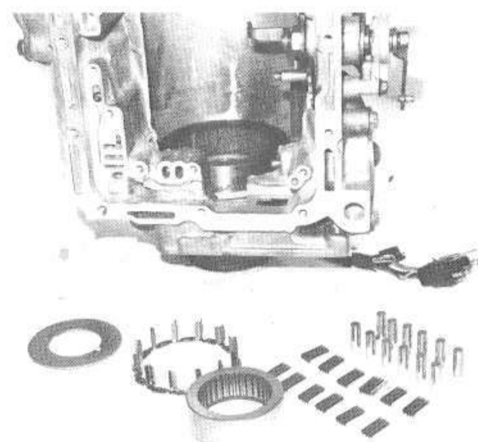

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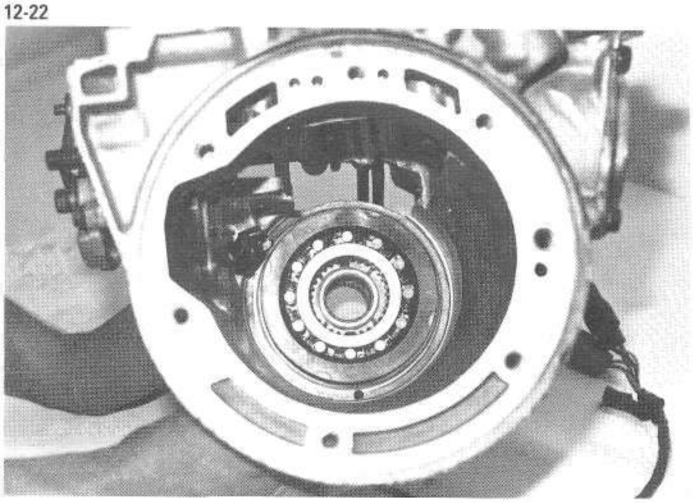

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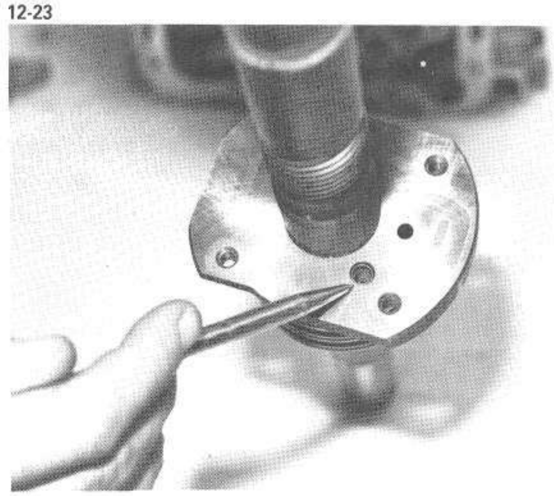

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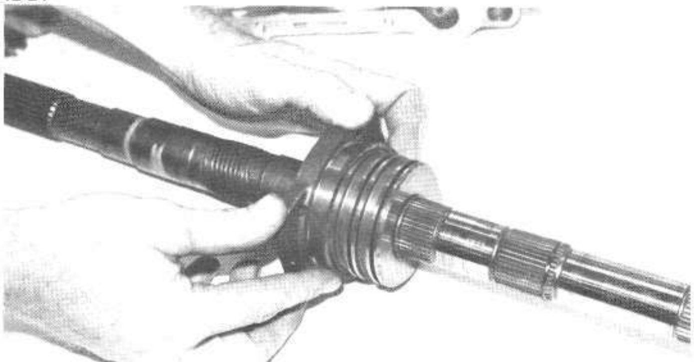

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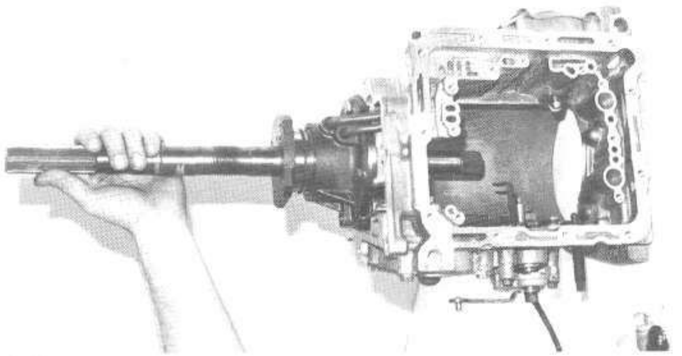

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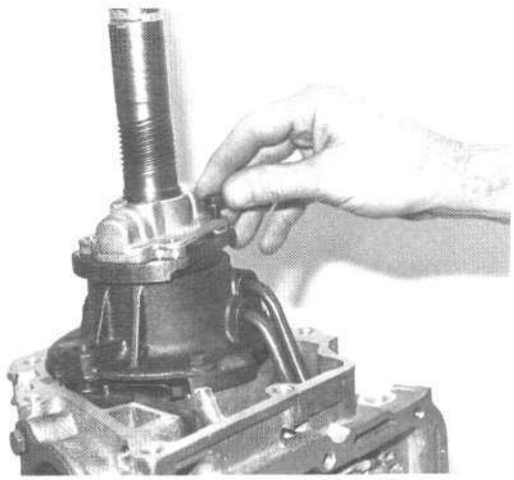
12-27



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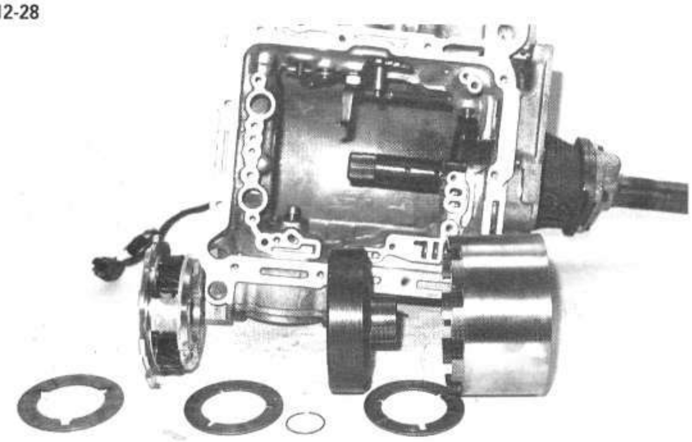

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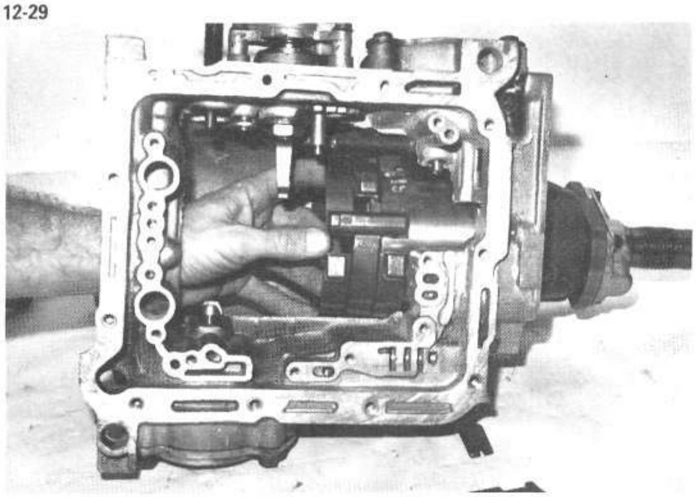


diameter of the drum is smooth and not heat-checked. Assemble the parts into the drum in the order they are shown in photo 28. Use a smear of grease to hold thrust washers in place while you are assembling it.

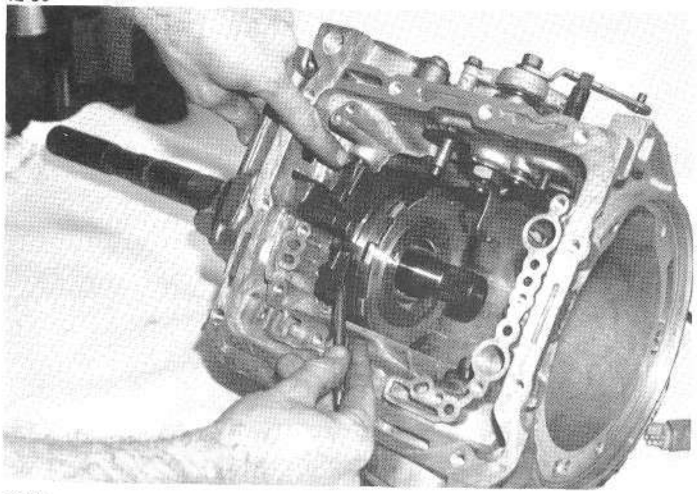
Check the brake band for wear, heat-cracks and distortion. If the grooves are gone in the friction face, it's worn out, so replace it. Install the brake band, as shown in 29. Note that the single anchor on the centre strip of the brake band goes toward the side with the linkage on it, whereas the part of the brake band with two anchor points goes towards the front servo side. Install the anchor-side first; that's the side indicated by my pen in photo 30.

Install the Sun gear input driving shell after you have inspected bushings and thrust faces for good working condition. Photo 31 shows how it is installed.

EXTENSION

Photo 32 shows the end of the gearbox extension shaft. Both the bearing and the oil seal that are in the shaft must be replaced. These bearings wear out

12-30



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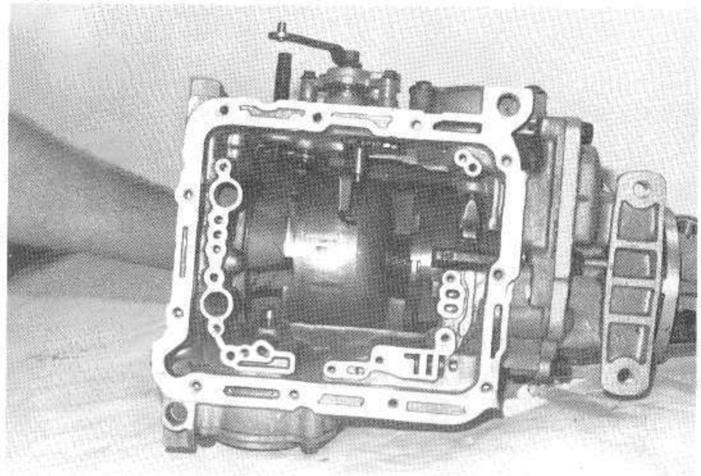

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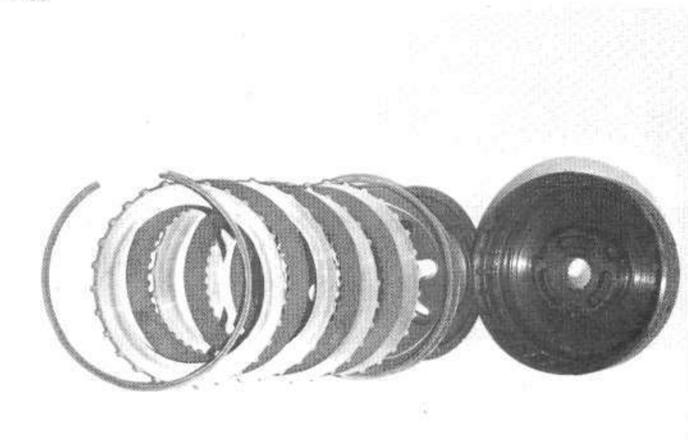
12-33



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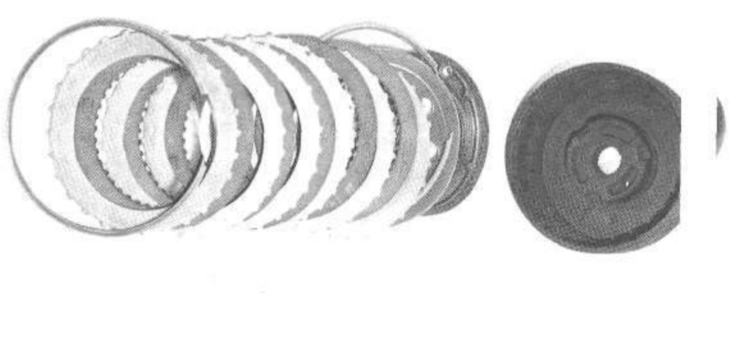

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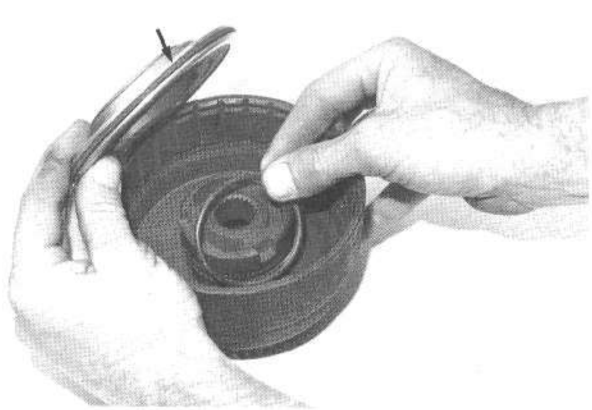

very quickly on Pintos, so it is best to start with a brand new bearing and seal. Part of the reason they wear out quickly is minor out-of-balance of the driveshaft. While you have the transmission out it is a good idea to take your driveshaft to a transmission specialist who can balance it spot-on.

SUB-ASSEMBLIES

At this point we are going to make another change in the transmission. In the C4 four-cylinder transmission, the forward clutch pack has three friction plates and two steel plates in it, as shown in photo 33. We are going to use the drum and clutch pack assembly from a 302 C4 transmission. As can be seen in photo 34, this has five friction plates and four steel plates. All the parts go into the drum in the order as shown in photo 34.

To assemble the forward clutch drum, first install the O-ring on the hub

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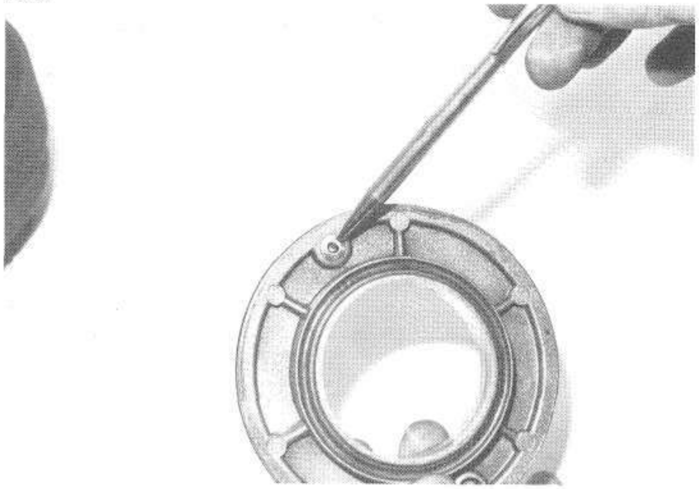

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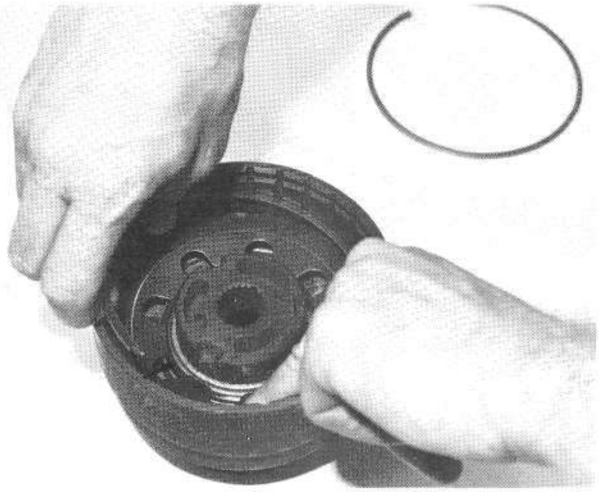

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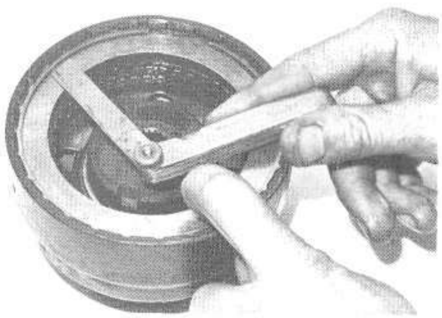

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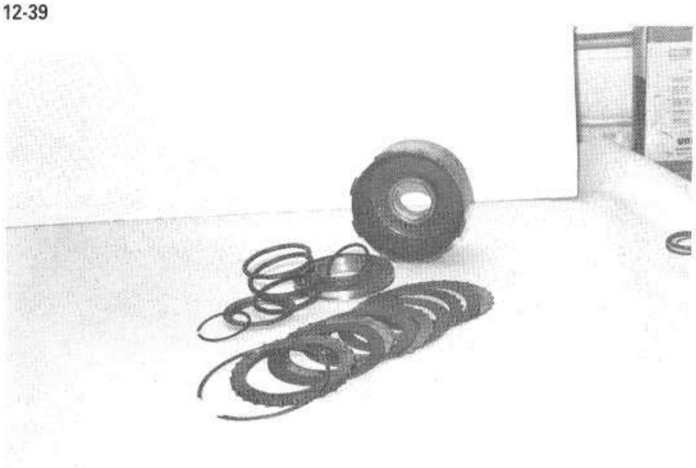

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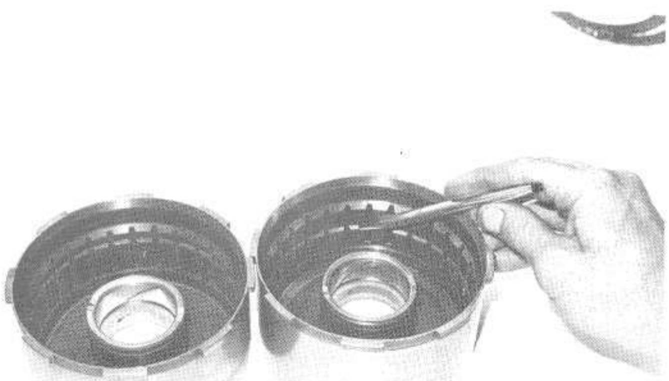

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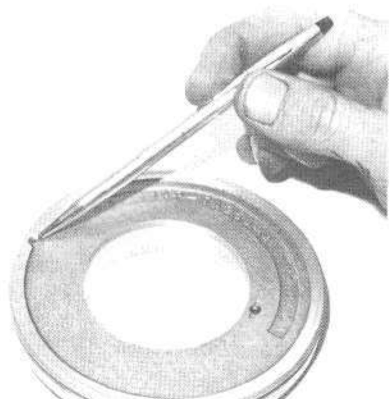


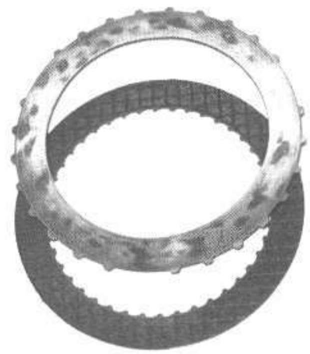
of the cylinder and the square-section O-ring on the piston. Lubricate these with automatic transmission fluid.

Before you install the piston in the forward clutch cylinder, make sure the check ball indicated in photo 36 is free. Then press the piston into the drum with heavy thumb pressure.

Install the Belleville return spring pivot ring, (snap ring) on the piston, as indicated in photo 37. Install the return spring concave side up. This must be in perfect condition and it's best to use a new one if any doubt exists. Next, install the large snap ring, as shown in photo 37.

Install the rear pressure plate, then install the friction and steel clutch plates alternately. Finally, install the front pressure plate and snap ring. Check the clearance between the snap ring and the front pressure plate, as shown in photo 38. The clearance should be between .025 and .040 inch on plates which have not been de-fuzzed. The tight end of the tolerance is preferable. Select a snap ring to set the clearance to the desired tolerance.

12-42



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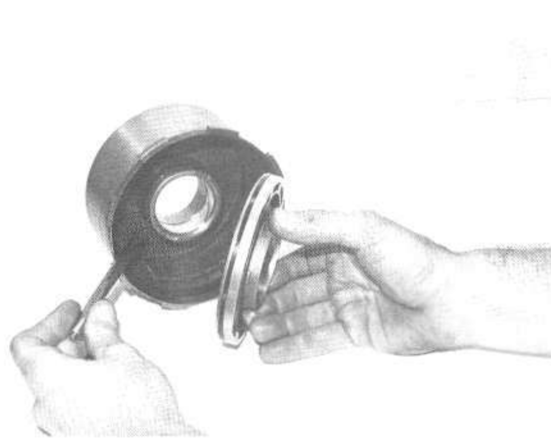

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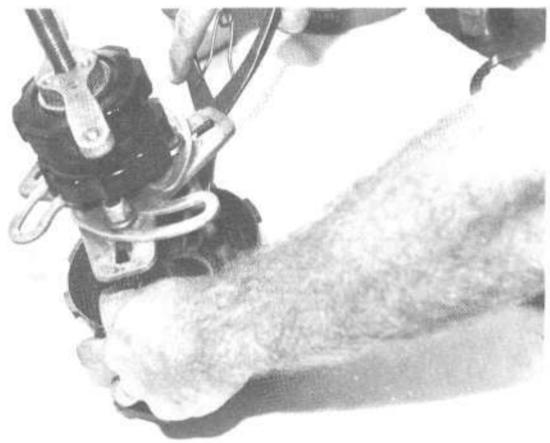


12-45

DIRECT/REVERSE CLUTCH DRUM SUB-ASSEMBLY

At this point we are making another change from the standard setup. The clutch drum and clutch pack to be used will be from the 302 V8 C4 transmission. Parts in the order they are assembled are shown in photo 39. This clutch drum and clutch pack assembly has four clutches, as opposed to the four-cylinder C4's two clutches.

Photo 40 shows the difference between the drum used in the Pinto and one used in a 302 Mustang. The principal difference is the position of the snap ring groove within the housing. My pen indicates the snap ring groove used in the Pinto. Note that it is much farther down the case than the groove of the 302 C4 Mustang to the left of it.

Take the apply piston shown in Photo 41 and block off the hole indicated by my pen. This hole can be blocked off with epoxy resin. But before you apply any adhesive, check that the component is totally oil-free or the adhesive (Araldite works well here) will not be secure. Blocking this hole gives much quicker and firmer pressure to the clutches.

Just for reference, I have included this photo. Photo 42 shows the result of trying to transmit 250 bhp through a standard Pinto automatic transmission. This transmission was only used for a couple of weeks. It would not have lasted long with such high power inputs.

Install the seals on the hub of the cylinder and on the piston, as shown in 43. Lubricate them with automatic transmission fluid and assemble the piston into the drum.

Install the spring, retainer and snap ring. For this you will need a special tool, as shown in photo 44. Probably your best bet is to take this sub-assembly to your local automatic transmission shop and they should assemble this for a minimal charge.

When the clutch pack has been assembled into the drum, it should appear as shown in photo 45. Check the clearance between the pressure plate and the snap ring. This should be between 0.045-and 0.060-inch. Clearance is checked in exactly the same manner as on the previous clutch pack. If the clearance is not within the specified limit, select a snap ring that will put it within a specified limit.

ASSEMBLING FRONT PLANETARY, FORWARD RING GEAR, CLUTCH HUB ASSEMBLY AND FORWARD CLUTCH DRUM ASSEMBLY

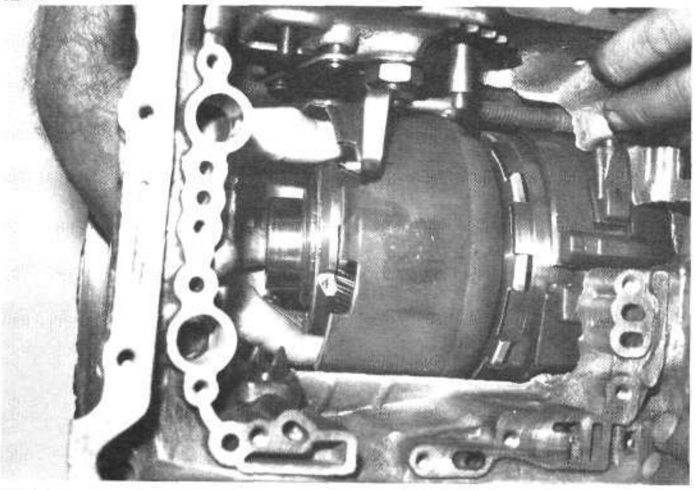
Check the spline on the planetary gear assembly (photo 46). The spline must be in perfect order. If not, replace the assembly.

Install the three-tang thrust washer on the planetary gear assembly; hold

12-46

12-47

12-48



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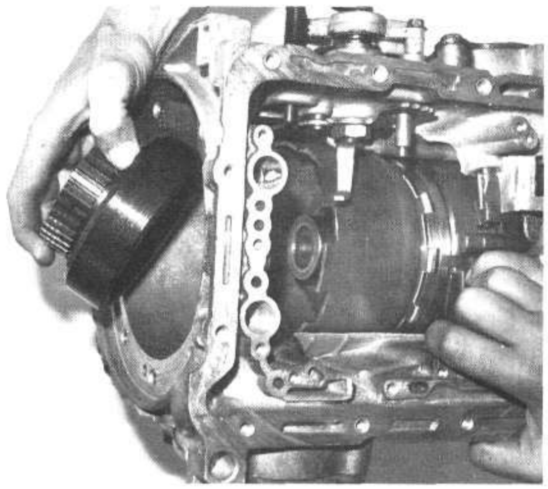
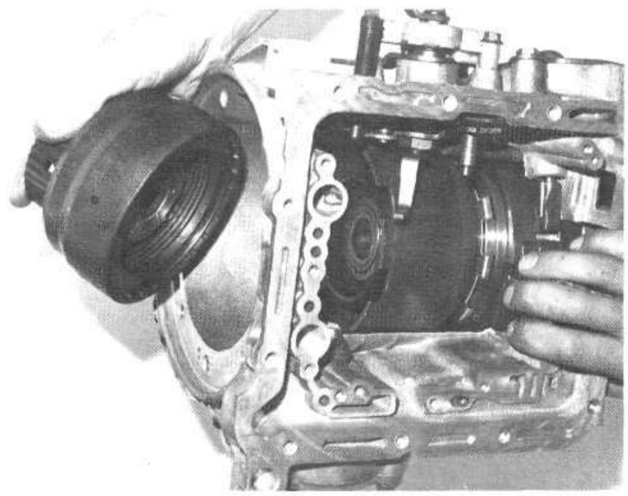


12-50

12-51



the washer in place with a smear of grease as shown in photo 47.

Install the planetary gear assembly, as shown in photo 48.

Install the forward clutch hub and ring gear, as shown in photo 49.

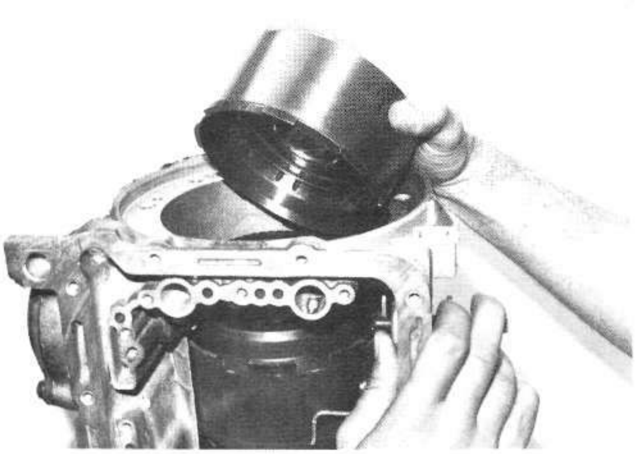
Smear grease on the back of the thrust washer and locate it in the forward clutch drum assembly, as shown in photo 50. Install the forward clutch drum assembly into the main assembly (photo 51). To ensure all clutch teeth line up, wiggle the clutch drum back and forth until it drops all the way home. Check this by raising the clutch assembly and allowing it to drop. It should hit with a solid sound.

Install high and reverse clutch drum, as shown in photo 52, then insert the input shaft into the spline of the front planetary (photo 53). Use the input shaft to position and align the clutches so that the drum drops into position.

INTERMEDIATE BAND INSTALLATION

The intermediate band, shown in photo 54, must be replaced. Before installing the band, soak it for a few minutes in automatic transmission fluid.

12-52



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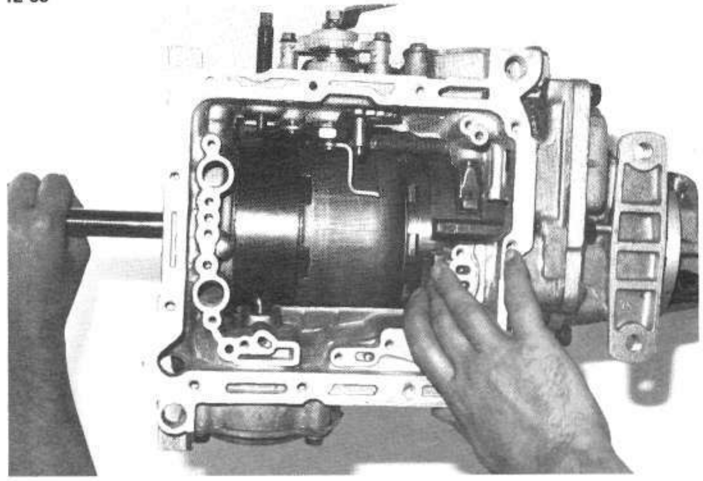

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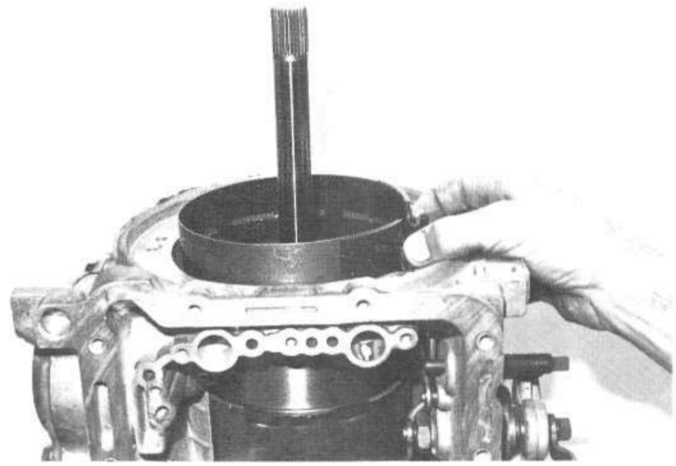
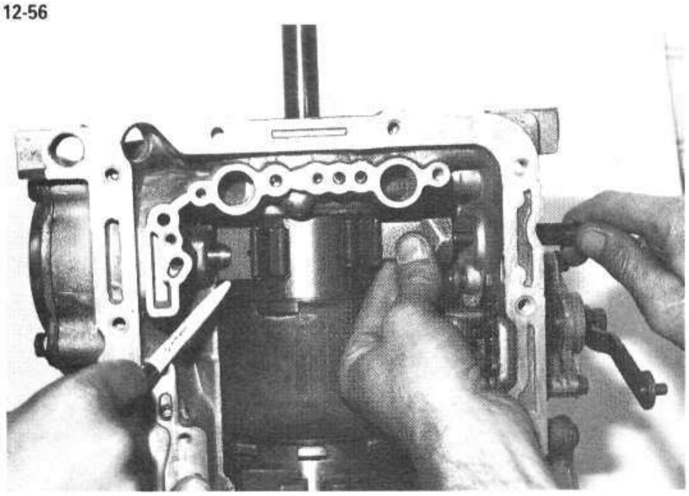
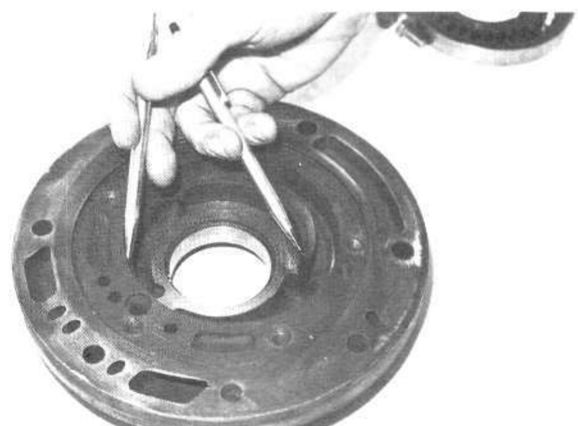
12-55



12-56



12-57



Install the band in the front of the case, as shown in photo 55.

When the brake band is in the position shown in 56, install the anchors. I am holding one in my left hand and the other is indicated by the pen. Then install the band adjusting screw. I am installing it through the right side of the case. Screw in the adjustor just sufficiently to hold the anchors in position so they don't fall out.

FRONT PUMP HUB ASSEMBLY

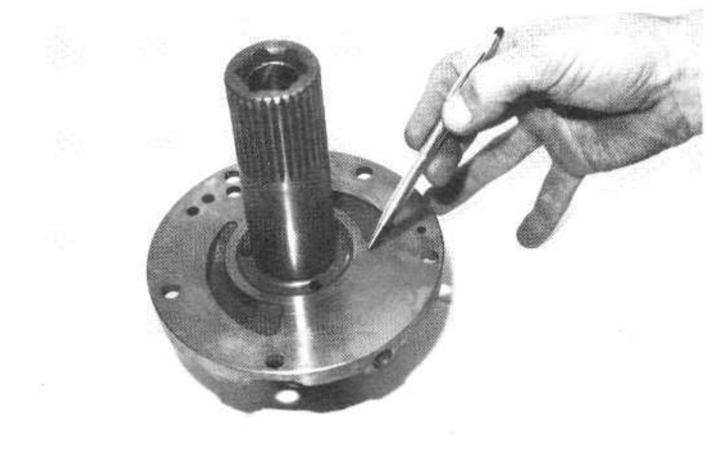
Inspect the pump body for wear in the areas indicated by the pens in photo 57. Also replace the bearing in this housing. Inspect the pump gears (photo 58). These gears must be in perfect condition. Inspect the teeth, the outside diameter and the flats of the smaller gear's inside diameter. If there is any doubt as to the serviceability of these items, get a replacement at your local transmission shop. (Not serviced at a Ford dealership).

Check the pump stator for wear on the thrust face as indicated by my pen in photo 59. If it's scored, it will need replacement. Also check the bearings in the bore and replace as necessary. Check the ring areas arrowed in photo 60.

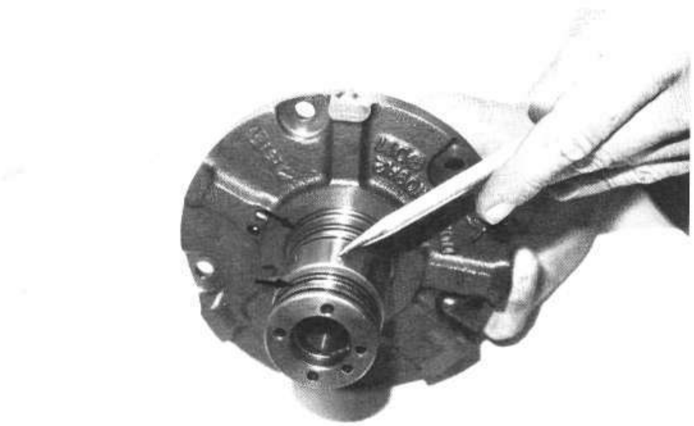
Check that the diameter indicated

12-58

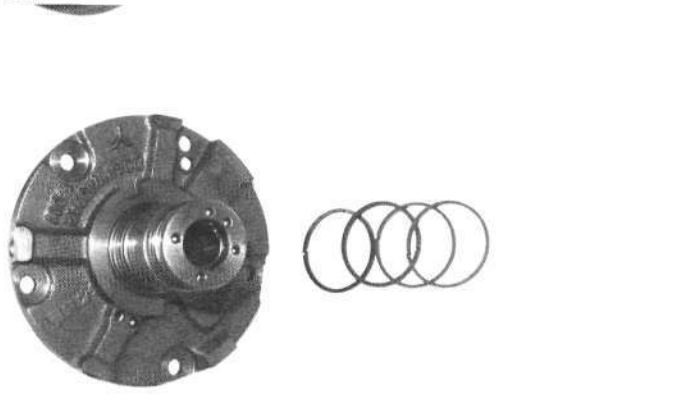
12-59



12-60



12-61



12-62

12-63

by my pen in photo 60 is in perfect order. It should be highly polished and free of any scores.

Next, install the rings shown in photo 61 onto the snout of the pump stator. There are two sizes, the larger ones being installed nearest the flange and the smaller ones towards the front of the snout.

Photo 62 shows the rings in position.

Install the oil seal in the pump body. Use a smear of silicone sealer on the outside diameter of the seal before installation.

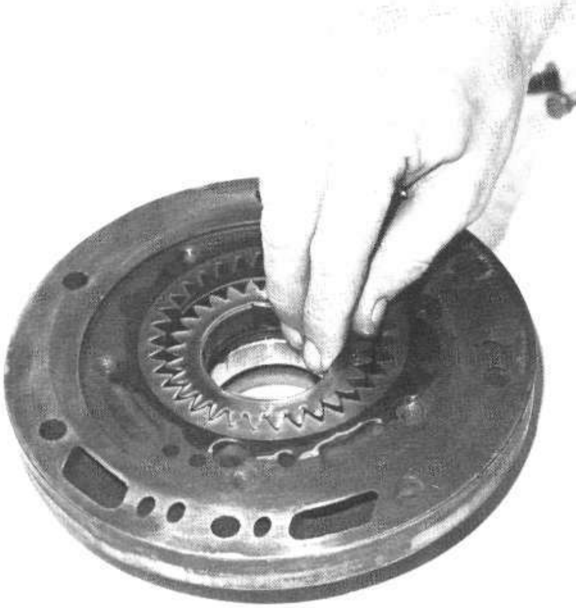
Smear the bearing in the oil pump body with grease. Install the gears and lubricate with automatic transmission fluid, as shown in 64.

Install the pump stator into the pump body.(Photo 65).

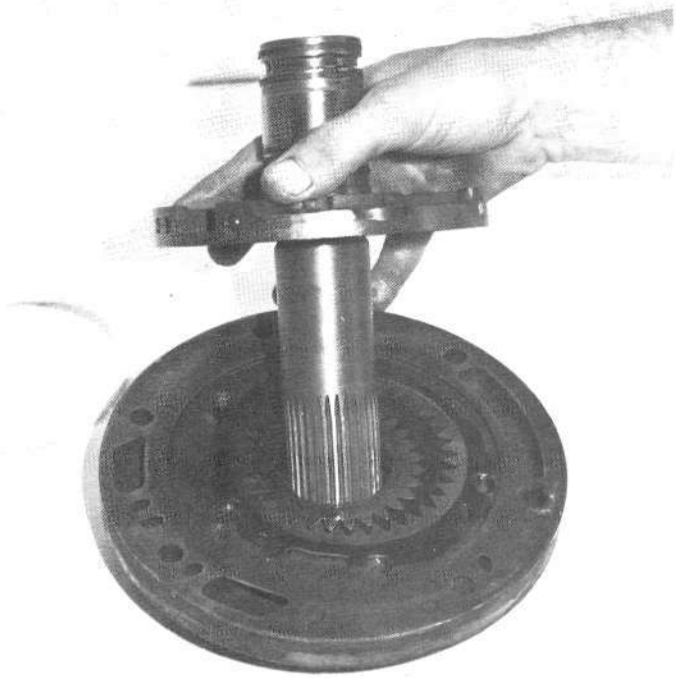
Check that the pump turns freely by dropping it onto the torque converter hub and rotating the pump body. (Photo 66).

Install the two thrust washers indicated in photo 67. These thrust washers are number-stamped on the back so as to enable the clearances to be set up. When one thrust washer is changed, the other will also have to be changed.

12-64



12-65



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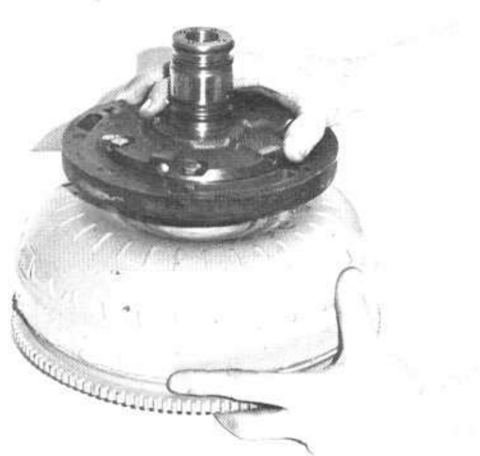


12-67

These will be selected later to suit the case. For now, just install the ones you have from your old transmission, as in all probability they will be the ones you will finally use.

Position a pump gasket on the face of the pump. Grease the gasket lightly to hold it in place. Lubricate the rings on the pump stator shaft and install the assembly into the transmission, as shown in photo 68. Then bolt on the bellhousing to compress the gasket.

Mount the dial indicator gauge on a magnetic stand, as shown in photo 69. Lever the sun shell backward and forward and check the end play on the shaft. It should be between 0.0042-0.008- inch. If it's not between these two limits, select alternative sized thrust washers shown in photo 67 to set the end float to the desired limits.

ADJUSTING THE BANDS

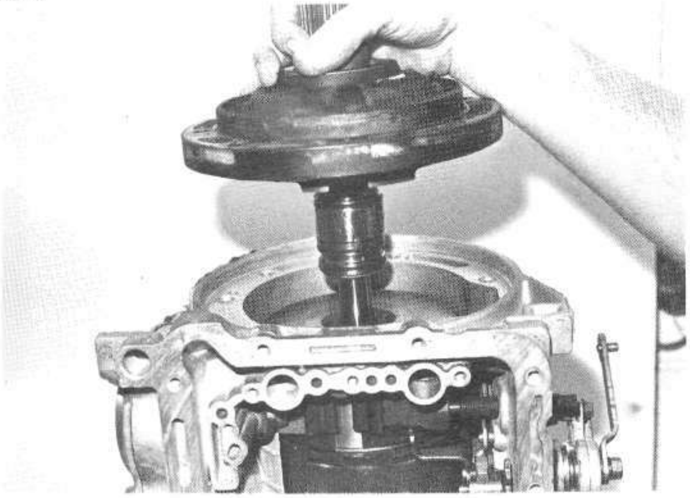
Adjust the front band (photo 70) until it is tight on the drum, then back it off 1½ turns. Lock the band adjusting screw in this position with a new lock nut.

Adjust the rear band adjusting screw in the same fashion, but back it off three turns instead of 1½. Again, lock it in place with a new lock nut, as shown in photo 71.

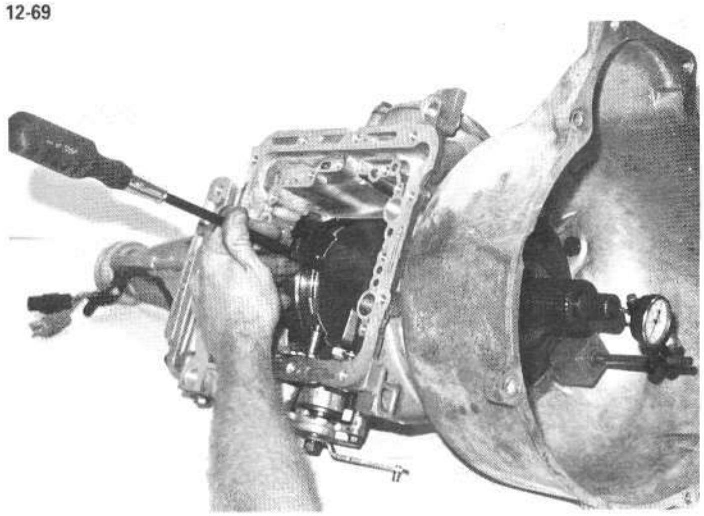
VALVE BODY

Completely disassemble the valve body and polish all valves. To reassemble the valve body, refer to photo 72. At this point we are going to make some more changes and these changes will depend upon the type of shift you want and the specific application of the transmission. If you want the gear change to be a normal, relatively soft shift, then reassemble all the components in the valve body just as they came out, as per photo 72. If you wish to make the shift slightly harder, pull out the spring and valve from the position shown in photo 73 and remove the spring, assembling the valve back in the body without the spring. If you want a really hard shift for competition applications or if you have a turbocharged engine of 230 bhp or more, remove the spring from its normal position, put the valve back in the

12-68



12-69



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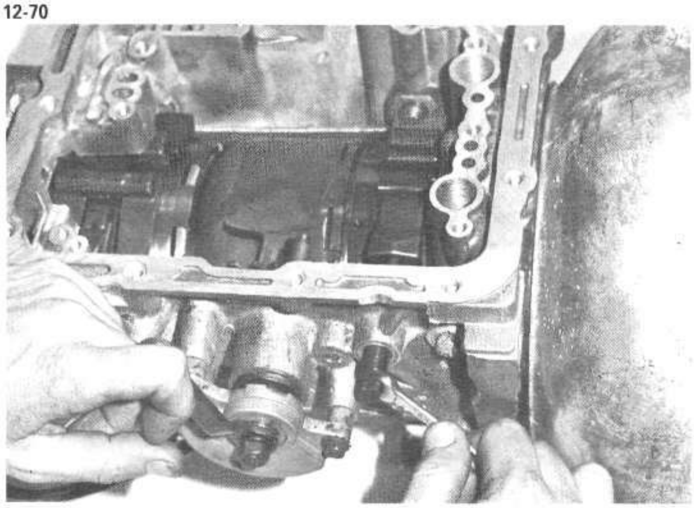

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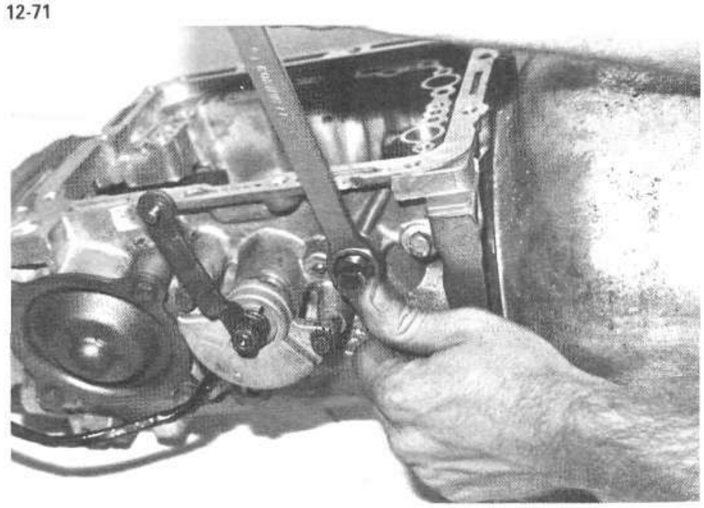

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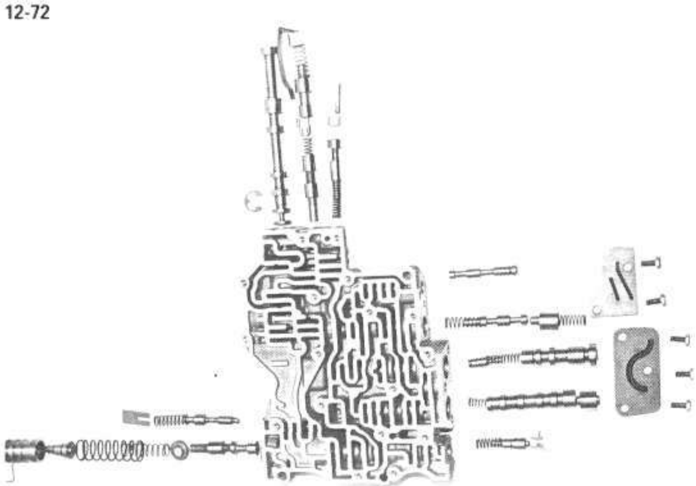
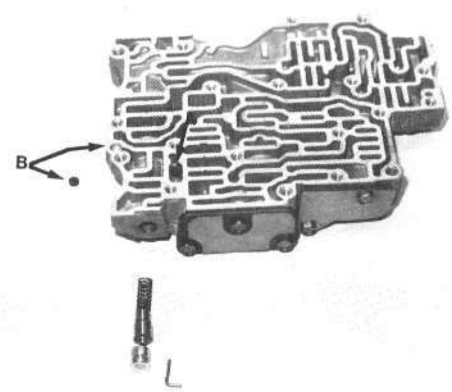


12-73

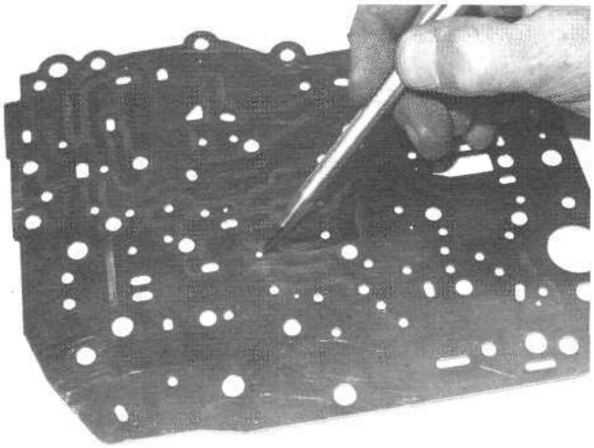


bore of the valve body and install the spring *behind* the valve. Also leave out the check valve. The check valve and its location in the body are shown by the arrow B. When assembling the valve body to the separator plate, do not forget to install the ball valve, arrow A.

VALVE BODY SEPARATOR PLATE

Photo 74 shows the valve body separator plate. A change must be made here. The nature of the change depends on the type of shift you require. If you are going to leave the shift as standard, then nothing need be done to the plate. If you are upgrading the shift by leaving out the valve spring, you will need to drill the hole in the separator plate indicated by the pen to 0.086 inch diameter. If you are going for the really hard shift, which involves

12-74



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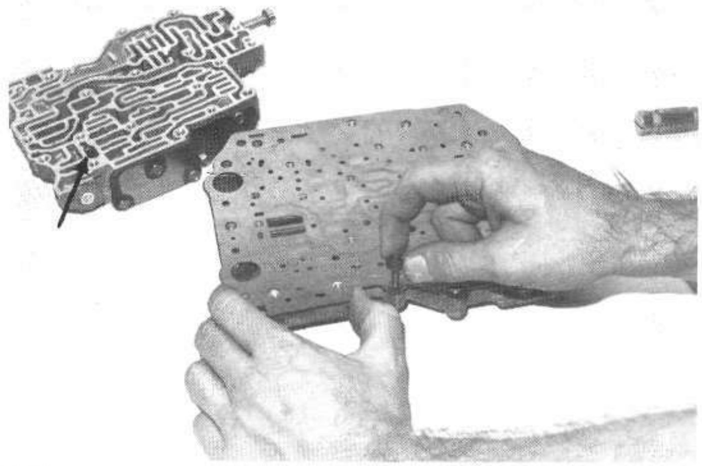

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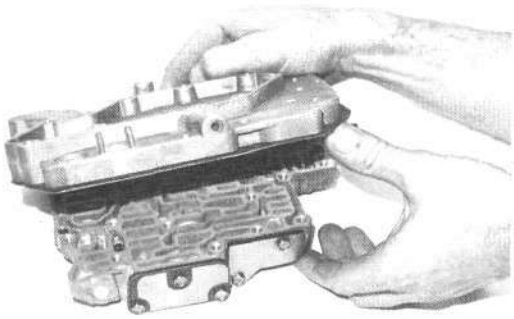

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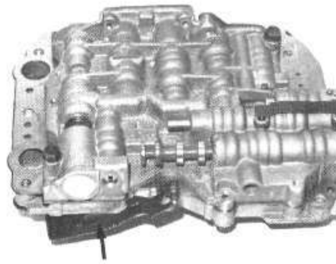

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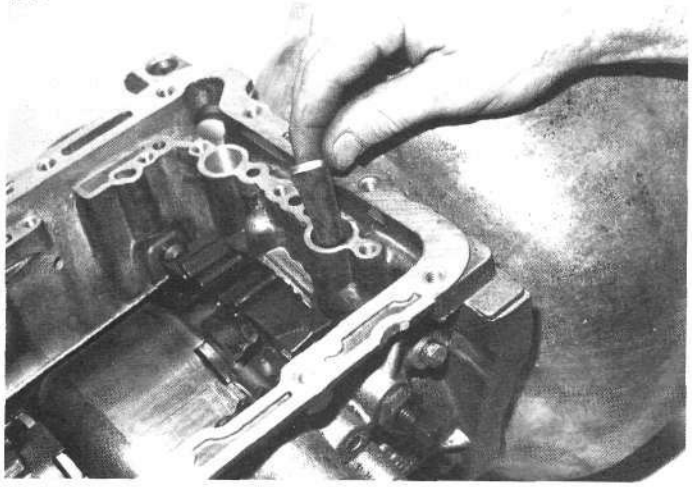

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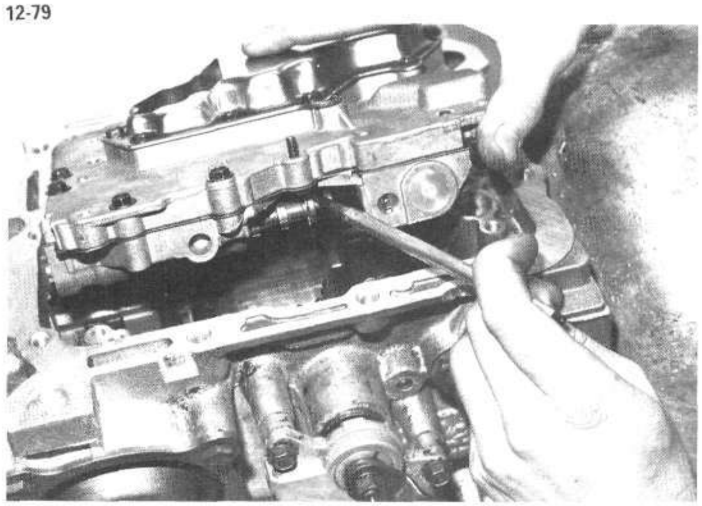


locating the spring behind the valve and leaving out the check valve, you will need to drill the hole to 0.101-inch diameter. Install the two neoprene check balls as arrowed in photo 75. Install the separator plate to the lower section of the valve body shown in photo 75. Don't forget to use a gasket between these two components.

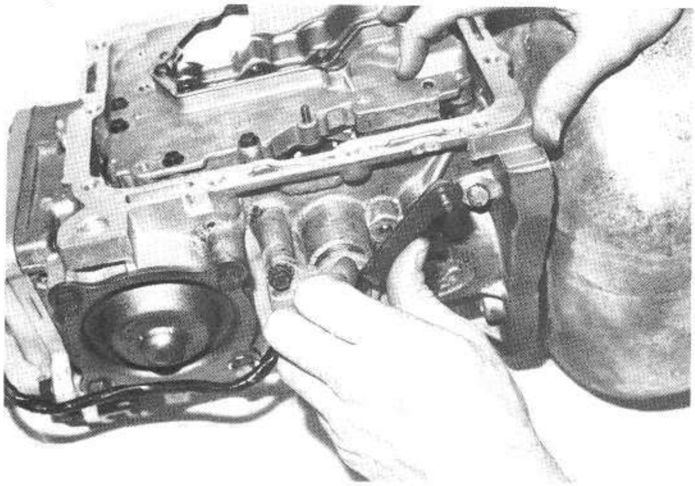
Put the two halves of the valve body together (photo 76) and tighten the bolts down to 40-50 lb./in. Work from the centre outward and pull the bolts down in increments of 10 lb./in. until maximum torque is reached. Doing this avoids distortion of the valve body. Add the filter to the lower side of the valve body.

Photo 77 shows the assembled valve body from the top.

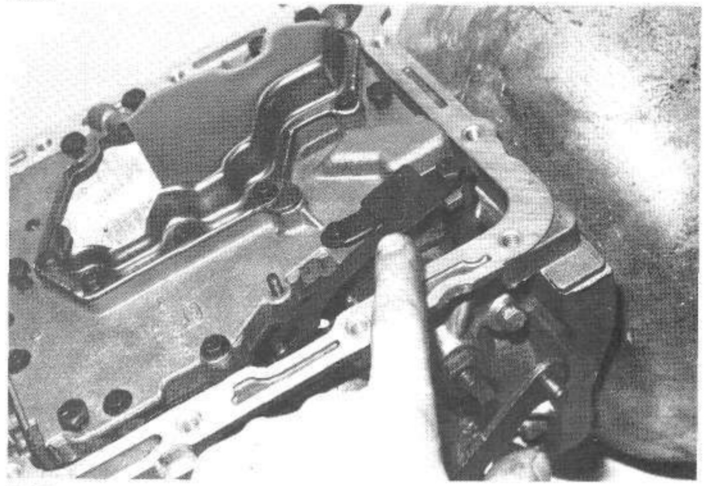
Replace the filter in the oil return hole in the transmission case, as shown in photo 78.

Engage the pin on the gear actuator with the manual selector valve in the position indicated by the pen in photo 79. As the valve body is lowered into the transmission case, make sure that the kickdown lever is operating the kickdown valve correctly and is in the

12-80



12-81



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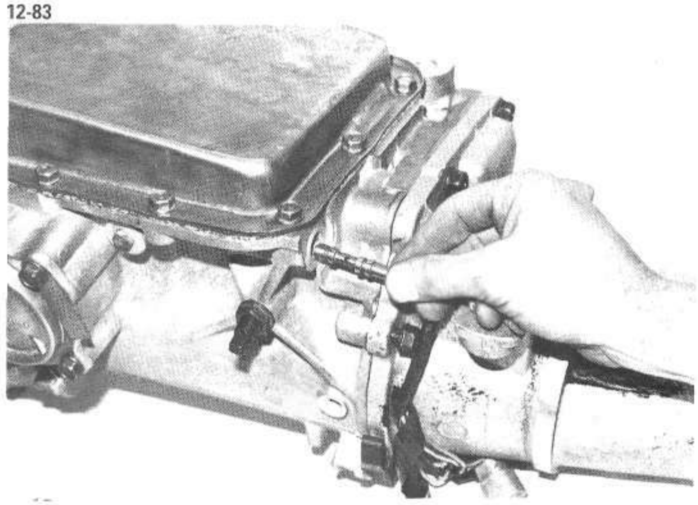
12-83



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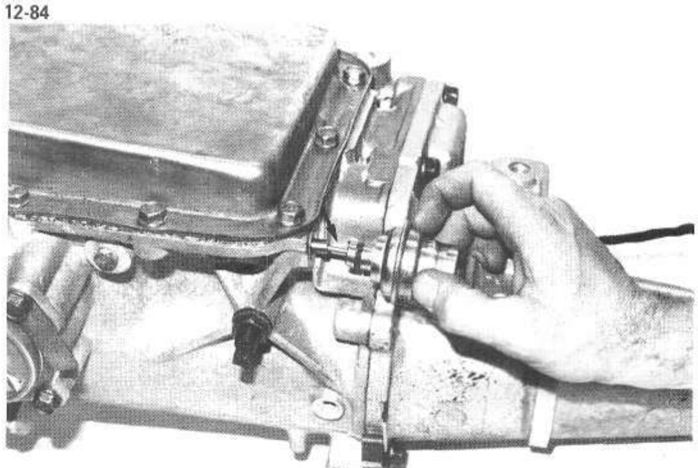


correct position. It should be in the position shown in photo 80. When the kickdown is operated, it should move an inch or so back towards the tailshaft. Install the selector detent spring, as shown in photo 81. Ensure that the area around the pan securing bolt holes is flat so the bolts will draw the pan down evenly. Position the pan gasket on the bottom face of the transmission case. Place the pan on the gasket and *progressively* tighten the securing bolts to 10lbs./ft. as shown in photo 82.

MODULATOR

Install the modulator valve pin (arrowed in photo 84) and modulator, as in photographs 83 and 84. With the installation of these parts your transmission is finished. Final chore: the vacuum hookup.

Some control of an automatic transmission comes from intake manifold vacuum. The automatic transmission uses this vacuum to sense engine load. When the engine load is light, the vacuum is high and the transmission shifts gently. When the throttle is opened fully and vacuum goes away, the transmission shifts harder to make sure that clutch slippage doesn't occur. Vacuum also determines the point at which the transmission will shift in the rpm range.

If a turbocharged engine is used, the transmission can see a different set of circumstances. If the tapping for the transmission vacuum is taken from the intake manifold downstream of the turbo, then the vacuum line to the transmission will see not only vacuum which occurs at part-throttle, but boost pressure which can occur at part-and full-throttle. However, the modulator was not originally designed to take boost. As a result, boost must be bled off from the modulator before it pressurizes the cannister. Several valves are available to install in the vacuum modulator line when tapping into the intake manifold to prevent pressurizing the modulator. These valves are available from companies such as Spearco, Rotomaster and B.A.E. The only snag with such a system, and it's of minimal important, is that the transmission is under the impression that the engine is delivering full power when no vacuum exists in the manifold. Unfortunately, full power is produced when maximum boost is produced at the manifold. But positive manifold pressure is not sensed by the transmission, so the transmission shifts at part-throttle as if the engine were operating at full-throttle, making the shifts fiercer than they need be.

An alternative method is to install the vacuum line downstream of the carburettor but upsteam of the turbo. At this point the vacuum module senses the vacuum existing immediately below the carburettor which provides more accurate measure of engine load. If a stock transmission is used, this approach may cause difficulties. This is because a vacuum can exist under the carburettor at part-throttle but sufficient air is coming by it to cause boost to occur downstream of the turbo. Thus, the transmission thinks that the engine is at part load, whereas the engine is in fact producing more power than a stock engine does at full-throttle. Under these conditions the transmission clutches are loaded up for part-throttle operation but torque output in excess of a stock engine full-throttle output is being generated. With the stock transmission this could cause rapid clutch wear. Fortunately our modified transmission more than takes care of this. The increased clutch area and the increased loading applied to the clutches by our uprated servos and such make the transmission well able to withstand the torque output. If the transmission has been uprated as described, it will operate perfectly satisfactorily under these conditions.

CHAPTER THIRTEEN

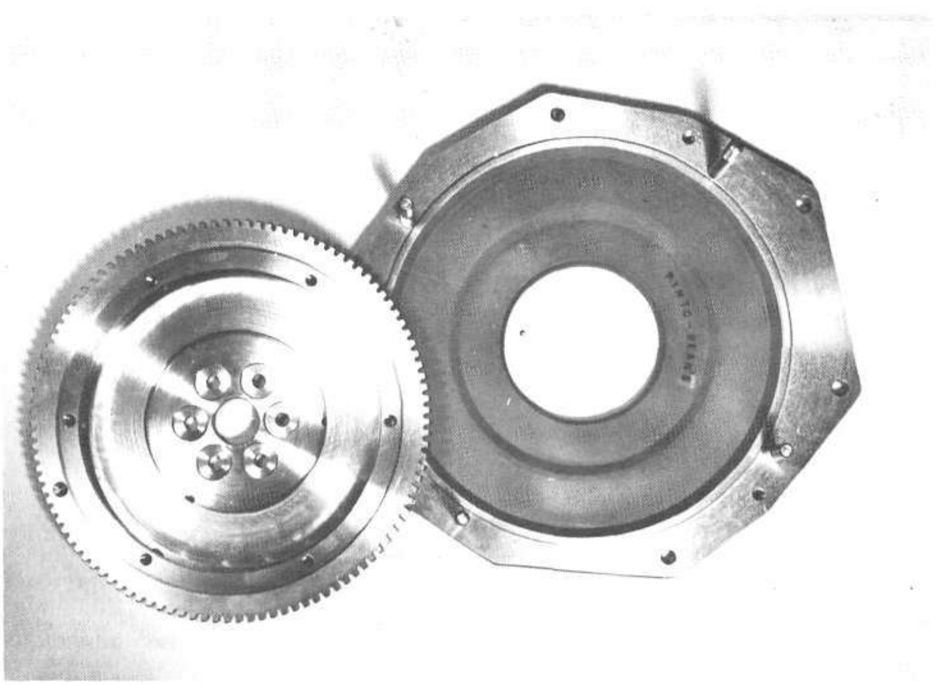
Off-Roading the Pinto Engine

In the U.S.A., our favourite engine is rapidly gaining popularity for use in off-road applications. It may never replace the Volkswagen engine but it is finding favour with those who want reliable, powerful engines with high-revving potential with fairly standard components. Getting 150 bhp from a Pinto engine is a lot cheaper than trying to get 150 bhp from a Volkswagen engine. The disadvantage of the Pinto engine is that it is heavier and is water-cooled. Offsetting this is the potential of more power for less money.

ADAPTORS

There may be others on the market, but the adaptor kit I am most familiar with is the one manufactured by Esslinger Engineering. This consists of an adaptor plate and flywheel. The adaptor plate allows a Pinto engine to be bolted up to a VW transaxle, and the flywheel allows the regular VW starter and related hardware to be used. There are two different types of starters — six-and 12-volt, and the flywheel gear must be cut to suit whichever starter is used. Also, there are two options as far as the adaptor plate is concerned. Which option you choose depends on whether you're building a mid-engined or rear-engined car. (A rear-engined car mounts the engine behind the transaxle. Mid-engine means the engine is aft of the cockpit but ahead of the rear axle.)

Certain measurers must be taken for mid-engine usage. When using a VW gearbox, the transmission must be turned upside-down, and then fitted with an adaptor plate allowing the engine-to- gearbox mating to take place in reverse. An alternative is to strip down the VW transaxle and take the ring gear from one side of the pinion and install it on the other. As it happens, the transaxle case is symmetrical, so this is a straightforward job. This reverses axle rotation out of the gearbox.



13-1. This is the Esslinger Engineering adaptor kit to mate a Ford S.O.H.C. engine to a VW transaxle. Flywheel takes all VW parts either standard or heavy duty race items.

To sum up, when you are buying an adaptor plate be sure to state whether you are using the gearbox normally or turned upside-down.

FLYWHEEL

For most off-road applications, the flywheel in the Esslinger Engineering kit is entirely adequate for an off-road engine with a good torque spread. However, if you are building a vehicle to be used for anything involving high horsepower and revs, plus subjecting transmission to violent shock, it is necessary to observe the flywheel-securing precautions mentioned earlier on in the book. You must decide just how severe the usage is. But remember that you can save the cost of maybe a complete transmission, possibly an engine and certainly a crankshaft, by making sure the flywheel never parts company from the crank. I advise that a minimum of two 5/16 inch dowel pins be inserted between any two of the bolts. If you intend using this flywheel with a 400-bhp turbocharged engine, put a pin between each bolt.

EXHAUST SYSTEMS

If you are building a rear-engined car, the exhaust system can present quite a problem for the amateur builder, since most do not have tube benders or the necessary sheet-metalworking skills to construct an exhaust system. As it happens, Esslinger Engineering builds an exhaust manifold for either a rear-engined or a mid-engined car. An alternative to this, if you happen to live in the land of road-racing (England), is to use a Formula Super Ford exhaust system. However, it should be pointed out that the lengths of the Super Ford pipe are more suited to the characteristics of the standard cam. If you are using a hotter cam, then you will need shorter pipe lengths, as detailed in the exhaust manifold chapter.

CARBURATION FOR OFF-ROAD

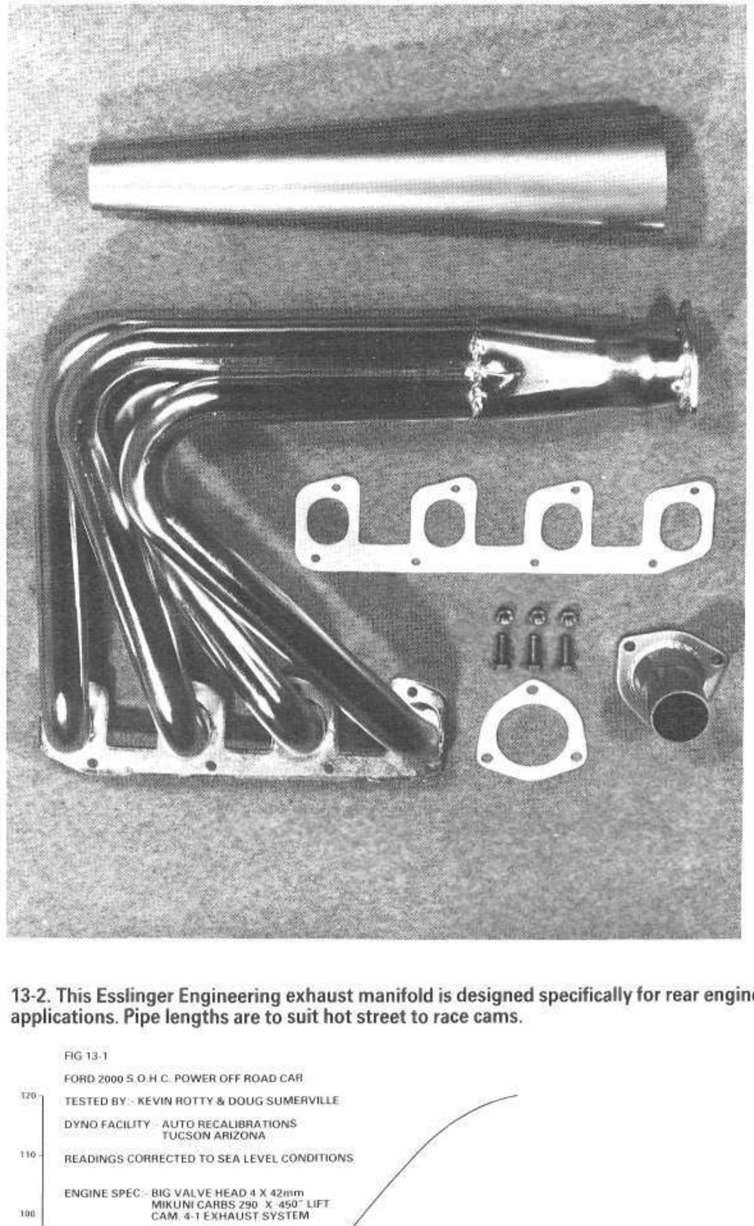
If the vehicle is used in very bumpy conditions, as in a Baja-type event, carburation can be a critical factor. Most carburettors will not stand the continual pounding. As a result, they will either flood or lean out from one moment to the next. The engine may grind to a halt or sputter just when you need the power. This is not conducive to putting on a competitive performance, so it's necessary to consider what can be done to avoid the problem. First, a carb that will work on off-road conditions right out of the box is the Mikuni motorcycle carburettor. It seems able to stand severe jolts and acute angles of lean without significantly altering its metering. My negative experience with the Mikuni, however, is that it lacks drivability at low rpm. On the other hand people who have used the Mikuni on many more engines than I have, state categorically that it will run over a very wide rev range when correctly set up, and the lack of flexibility is due to an incorrect setting up of the carbs. So be it. I bow to superior wisdom.

As far as power is concerned, the Mikuni fares pretty well, not quite as well as the Dellorto, but there again little does. To give you some idea of just how much performance a Mikuni can deliver, see Fig. 13-1. This particular engine was not a high-dollar engine. It was just a well-prepared conventional one with a 290-degree, 450 lift cam, well-modified head with no welding done on it, Mikunis and a four-into-one exhaust system.

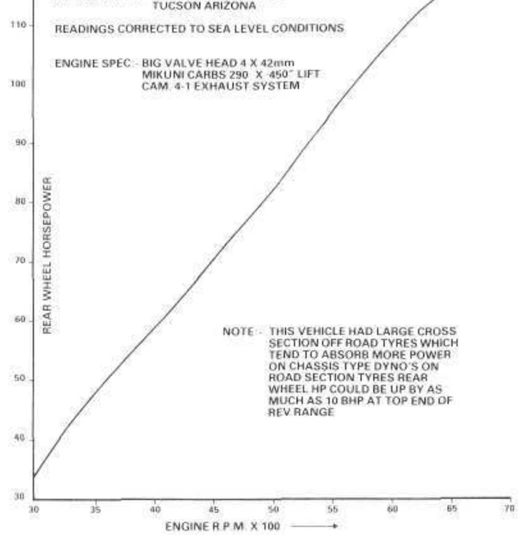
ALTERNATIVE CARBURATION

Maybe you're not interested in using Mikuni carbs; maybe you're looking for the smoothness given by a pair of Dellortos, Webers or even the surprisingly smooth drivability given by a two-barrel Holley on this engine. The Holley is a carb that is cheap to install and is very effective on engines up to about 150 bhp. Let's discuss its off-road application.

One of the things the Holley cannot cope with is bouncing around. It seems to be more sensitive to being bounced than most other carbs. As an off-road carb out of the box, it is "useless". Holley makes a special off- road bowl and float conversion for this which helps the situation slightly but is by no means a cure. It yields about 10 percent improvement subjectively speaking, but still leaves 90 percent of the problem present. The off-road bowl may be adequate for such applications as sand-dragging or relatively smooth off-road work, but cer-



13-2. This Esslinger Engineering exhaust manifold is designed specifically for rear engined off-road/sand car applications. Pipe lengths are to suit hot street to race cams.



tainly the normal type of off-road event will leave you parked somewhere with a flooded engine. However, there is a way around the fuel slosh problem in the Holley, or *indeed, in any other carb.* See Fig. 13-2. Here you see the schematic for a scavenged, floatless fuel bowl on a carb. This system employs two fuel pumps. One delivers fuel to the float bowl, where it is prevented from sloshing by having fuel tank filler foam inside. A standpipe in the float bowl has its top edge at the fuel level. Fuel rising above this level pours down the standpipe and is drawn back to the fuel tank by a second electric pump. This system is very simple yet extremely effective. A friend's off-road car fitted with this arrangement could be stood on its nose and the engine would run as if the car were horizontal. Fuel slosh was reduced to the point that the engine would run under any jolting and bouncing conditions that the driver could endure. Carburation was clean and precise despite racing conditions so violent that the driver suffered nosebleed and bloodshot eyes.

To make sure the system is as effective as possible, there are a few points you should note. First an electric pump is preferable to use as the bowl-to-tank scavenge pump, keeping the mechanical pump to feed the carb. If you try things the other way around, you will have flooding upon starting the engine. Second, it is helpful if the incoming jet, located where the needle valve and float used to be, is sized so that only just sufficient fuel is passed into the fuel bowl at full throttle. This reduces the amount of fuel returned to the tank to a minimum. Fig. 13-3 gives you an idea of jet size to use at various fuel pump pressures.

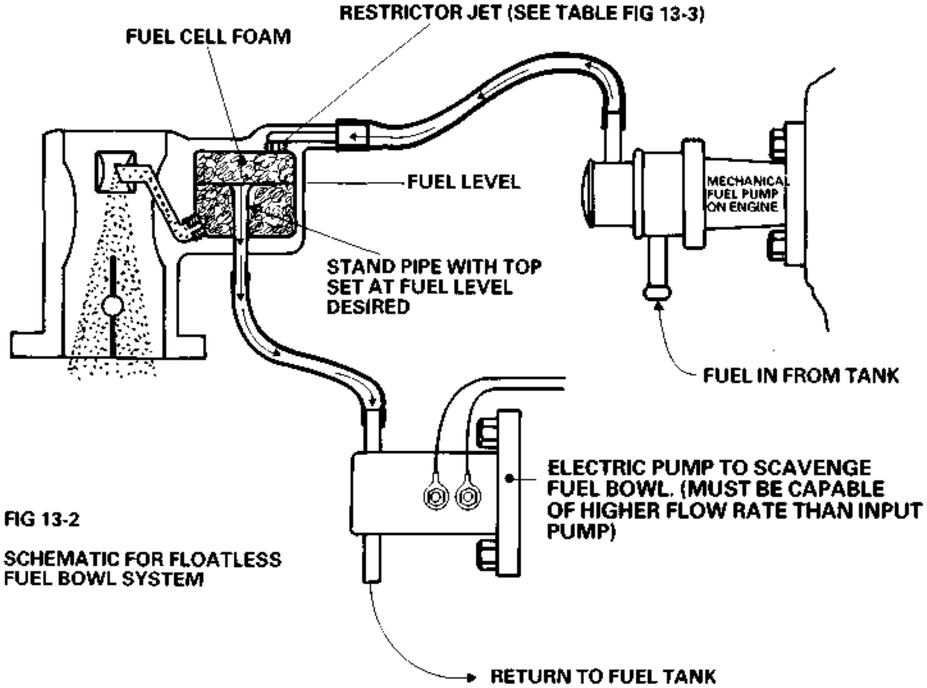
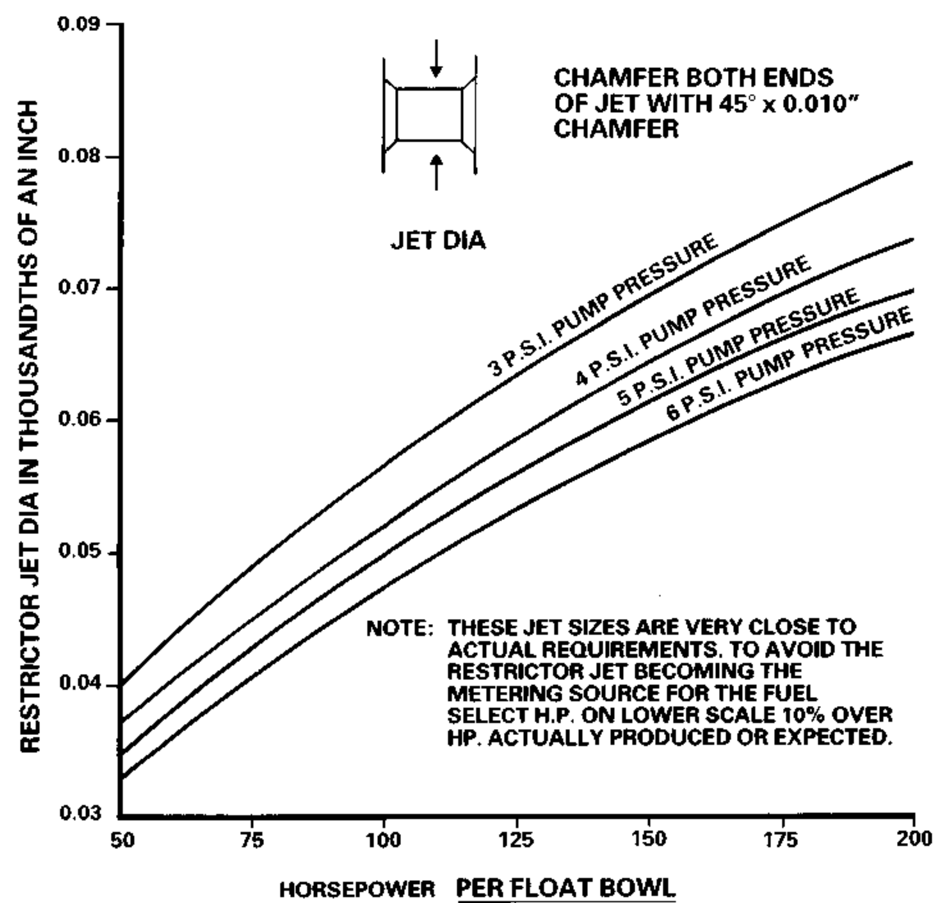


FIG 13-2

SCHEMATIC FOR FLOATLESS FUEL BOWL SYSTEM

FIG 13-3

RESTRICTOR JET SIZE SELECTION CHART FOR FLOATLESS FUEL BOWL



Building the Engine for a Purpose

When it comes time to decide on the specification of your engine, two factors loom large. In spite of being very important, these two factors are often over-looked. They are the purpose for which you intend the engine to be used and the cost of producing an engine for that purpose. Of course the problem is much diminished from the financial point of view if you have enough money to build the engine to the exact specification required. Few people are in this position and so some sort of decision must be taken as to where to spend the money for the best results.

CHEAP TRICKS

Let's go back to the very beginning and assume you have yet to raise a wrench to your engine. Face facts: unless it's perfectly fresh, you're going to have to rebuild it. There is no point in modifying an engine which is partly worn out. Modifying a worn-out engine usually leads to disappointment, as the engine becomes more unreliable and the performance gained is often minimal. The golden rule, then, is always to work on an engine that is in sound condition. And if producing an engine in sound condition means rebuilding it, there are many simple measures to be taken to increase power without necessarily incurring a lot of extra expense.

The first concerns reboring. Most of the expense of a re-bore on your Ford engine is the cost of the pistons. Pistons of +.040 or +.060 cost no more than those at +.020 oversize. By having the engine re-bored +.060-inch over (92 mm bore), the capacity of a 2000cc engine is increased by 53cc. This is worth approximately three ft./lb. more torque and about three bhp more with an otherwise standard engine.

While the engine is down, consider having the block or cylinder head milled to raise the compression ratio. Raising the ratio by 1½ points can give up to five ft./lb more torque and about another three or four bhp. In the head chapter, we looked at elementary cylinder head modifications extensively. Performing some of these modifications will also help power. Again, the amount can vary, but figure on around three to five extra bhp.

Another three bhp can often be gained usually by going through the valve train and making sure each valve opens the exact amount. Also, ensure that there is minimal friction in the engine, thus maximal power at the flywheel. Don't forget to check that rods and crankshaft are true and that piston clearances are exactly where they should be.

If you don't have the money for balancing the engine, don't worry too much. Balancing will not give any more performance or rpm. All it will do is make your engine run smoother. The standard engine, as it comes from the factory has an adequate balance for most applications. The only exception to this rule is if you have lightened your connecting rods, the chances of them all being the same or even close, are fairly remote.

Acceleration can be helped somewhat, especially in heavier cars, by having the flywheel lightened. A few pounds off the flywheel will make the engine much more responsive, and it isn't usually that expensive to have done. However, no extra power will be gained by having a light flywheel.

There are many other simple changes available, too. One of the simplest is to block the water heating passage on the intake manifold. However, while this works fine in summer weather, it can necessitate prolonged used of the choke in the winter. So, for cold weather, it's not really a good idea. In hot climates, blocking this water heating passage can account for about a three-bhp increase. A tall air filter will yield around two more horsepower.

WHAT ORDER ?

On a dollar-spent-per-horsepower-gained basis, there are four measures of modification having excellent value. These are a high performance air filter, a special exhaust system, a cam and a modified cylinder head. Anyone of these can pull 10-15 bhp more from a standard 2000cc engine.

But all of these measures are also closely interrelated; each has its own practical considerations and drawbacks. For instance, an exhaust manifold, plus auxiliary hardware, may take a day to install. Most exhaust manifold manufacturers claim a bolt-in fit for their products. However, I have installed manifolds that have taken two day's work because of inaccurate manufacture. Then on top of the cost of the exhaust manifold, you must add the cost of the rest of the exhaust system; if you retain a restrictive standard exhaust system, many of the advantages of your exhaust manifold will be lost.

How about a modified head? If you do the cylinder head yourself, this can work out fairly cheaply. But this, of course, requires considerable skill and knowledge. To the cost of head modification you must also add the cost of a gasket set. Of course, if you don't have a spare head, you have the vehicle's time off the road to consider as well.

And the camshaft? Well, it's cheap enough by itself but in order to install it, you will need to remove the cylinder head. Also, to get the best from a cam change, you may also have to fork out on an adjustable cam sprocket. More dough. And to remove the head, you will also have to remove the exhaust manifold; and this is halfway to installing a high-performance exhaust manifold. With all these things interacting, it's hardly surprising that cam, manifold and head changes are commonly all done at once. It makes both practical and economic sense. The only exception to this is the air filter. A change here involves a little cash and ten minutes work installing the filter and making a jet change. For minimal effort your engine can produce 8-10 B.H.P. more at the wheels.

Of key importance to street-rodders is the question, just how much power can be extracted from this engine without impairing drivability? This is especially important if the vehicle has an automatic transmission. To give you some idea of what can be achieved, see Fig. 14-1. This particular vehicle, a Pinto automatic 2000 has 65 percent more power than the production vehicle, yet it would deliver four more mpg under normal driving conditions, as well as a 20 percent higher top speed.

If you intend using an engine-transmission combination for road work as well as for the occasional trip down the dragstrip, remember that automatic transmission cars tend to be very much over-geared for acceleration work. The Pinto's final drive ratio needs to be reduced at least two, possibly three ratio points to even come close to what's needed. But remember every time you drop final drive ratio, you will also drop mpg. That may hurt your finances more than you want.

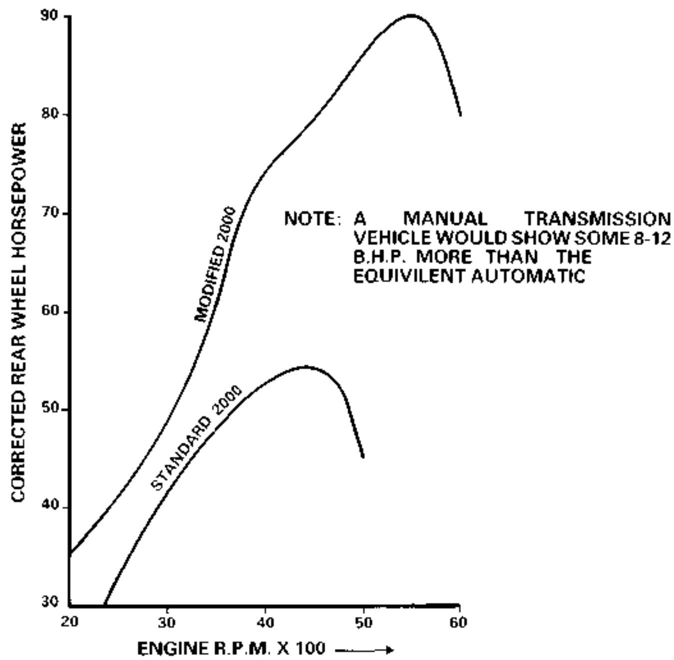
Let us assume you want to go one stage further than the specifications shown in Fig. 14-1. Where can you go from here, especially if your vehicle is automatic transmission? It is entirely possible to use twin sidedraft carbs on such an engine. If correctly calibrated, they will work low down the rpm range just as well as, say a single Holley 350 or even the standard carb. My personal preference here is for Dellortos, as they seem to have better bottom-end response. That's important for an automatic transmission car.

If the vehicle does not have an automatic transmission, the cam can get wilder than the Erson 134 profile used for auto- equipped vehicles.

FIG 14-1 DYNO TEST OF 2000cc PINTO WITH AUTOMATIC TRANSMISSION

TESTED BY:- DAVID VIZARD
DYNO FACILITY:- AUTO RECALIBRATION TUCSON ARIZONA

ENGINE:- 2000cc BIG VALVE HEAD 10/1 C.R. ERSON 134 CAM AND ADJUSTABLE SPROCKET. 4-2-1 RACER WALSH EXHAUST MANIFOLD SUPREME SUPER C 111224 TURBO SILENCER HOLLEY 350 CFM 2 BRL CARB K & N AIR FILTER M.S.D. MULTI SPARK IGNITION.



RACE ENGINES

Trying to put together a race engine on a budget is a nightmare. Unless you have power figures at least close to your opposition, you are going to be blown into the weeds. And if you are not racing to come in first, why are you racing? If you are attempting to put together a race engine as cheaply as possible, start with the engine's breathing. Always go with a good cylinder head. If necessary, spend the time carving your own. It will be a lot cheaper than having a head specialist do it, although it may take you a hundred hours to produce if you have never done one before. Once you have a good cylinder head, consider your next priority item is a cam that will work with that cylinder head and produce the power curve most suited to the type of competition you intend doing, be it off-road work, drag racing, hill climbing, circuit racing, etc. Pay close attention to your cam selection. Remember, big cams with high lifts do tend to be a little less reliable than slightly more moderate cams. Bear this in mind if engine breakage is something that may put you out of the race for financial reasons.

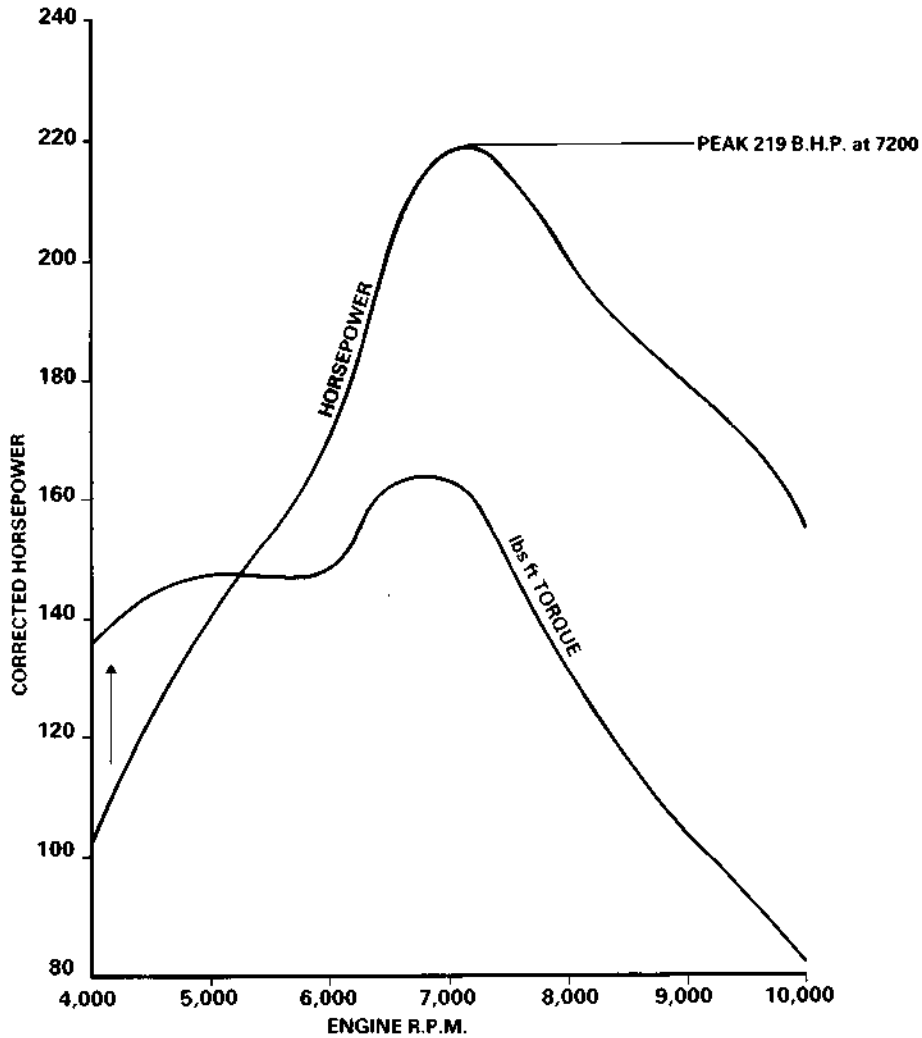
After you have the head and cam, consider the exhaust system. A good exhaust system will be the next necessity. If you can't afford twin side draft carbs, then a 500 Holley on the stock manifold will do a good job, and HP figures in excess of 170 can be achieved on such carburation. This makes a fairly healthy unit. Once you find you have the finances for twin side draft carburettors, that is the next stage to go for.

After this you may consider that such things as special pistons, special rods and cranks, etc., are a necessity. Until you can afford all the equipment for the breathing part of the engine, then trick pistons, rods, etc., are a case of gilding the lily. You will be spending money that could have been spent more usefully in other directions. In most instances, a turbocharger kit represents good value for money, but when you buy one, if it is for competition purposes, be sure that the turbo unit can deliver enough air for the sort of HP you are looking for. Some turbo kits have their compressors and housings sized for high-performance street applications. These will not make as much HP as a normally-aspirated engine with the right equipment on it. On the other hand, a turbocharger kit for race applications, can turn a 100 HP production unit, into a 200 HP racer straight out of the box.

So far, I have spent all the book up to this point making recommendations on this, that and the other. Hopefully, at each stage, the modifications I have described have been totally justified or substantiated. Although I have talked about many types of engines, and ways and means of getting HP, probably the most used type of racing engine will be normally-aspirated unit. This being the case, just how much HP can be produced by screwing an engine together, utilizing all the specifications I have given in this book. Fortunately, that's not too difficult to establish. I made up or had made as many of the "ultimate" parts as possible. These were then assembled at Duane Esslinger's shop in California, and the engine run on Esslinger's dyno. The power curves for this engine are as shown in Fig. 14-2. To give you some sort of standard by which you can judge the merit of these figures, it may be a good idea for me to point out that very few people in the world have built 2-litre, SOHC Ford engines that will even make 200 HP let alone surpass it. Many claims for over 200 have been made, but very few of these ever get substantiated when independently tested on an accurate and unbiased dynamometer. All you need to do to build an engine giving the HP as shown here, is to follow the specification to the letter. If funds had permitted on our test engine, there would have been a few small changes I would have liked to have made. For instance the exhaust pipe lengths were too long on the test engine for the high R.P.M. at which peak power occurs. The test engine had the unusual situation of peak torque and peak H.P. occurring at the "same" R.P.M. Analysing the results of the dyno test indicates that the primary length needs to be reduced from about 28 inches to 21-22 inches. This I am sure, would push power levels close to the 230 B.H.P. mark. Unfortunately that must be something to test on another occasion.

FIG 14-2 DYNO TEST OF PROJECT ENGINE.

TESTED BY: DUANE ESSLINGER OF ESSLINGER ENGINEERING
ENGINE SPEC: APPLE PORT SUPER HEAD WITH COATED CHAMBERS & EX. PORT CAM: CRANE 322B CARBS. 48 DHLA DELLORTO'S 2" EX PIPE 5" C. TO C. SUPER RODS. ARIAS PISTONS. CHILDS & ALBERT RINGS. BURTON ENGINEERING DRY SUMP SYSTEM.



Dyno Tuning

When all is said and done, the final question always is: "How much power will my engine produce?" Unfortunately, building a good engine is not just a question of having the right parts. It's a question of having the right *combination* of parts, then maintaining them, through tuning adjustment, in precise harmony with one another.

Let's consider a few of the things that are adjustable or selectable on a relatively unmodified SOHC engine: ignition timing, spark plugs, plug gaps, ignition advance curves, cam timing, main jets, idle jets, emulsion tubes, air correctors, pump jets and high-speed bleeds. All these items are readily adjustable. If we consider a race engine, we can add to that list auxiliary venturis (booster venturis), main venturis (chokes), exhaust tailpipe length, induction length, and if it's a turbo unit, throw in an exhaust housing or two, and a wastegate and water injection settings. The effect of any one of these items on its own, can be small. When each item causes such a small change, you may ask yourself, why worry? But the fact is, there are so many small items affecting power, that the sheer number eventually add up to a large power loss. Thus, these adjustments must be optimized. The only place to do this is on the dynamometer. This horsepower-measuring device can take one of two forms. Probably the most common type of dyno is the one which measures the power at the rear wheels. This is known as a chassis or rolling road dyno. The other type, most often used for pure engine development work, is the engine dynamometer. It measures power directly off the power plant.

The value of setting up an engine on a dyno cannot be over-emphasized if performance and, in the case of a road vehicle, good fuel consumption, are of importance. Because an engine is completely unmodified, does not mean it need never see a dyno. Even a standard engine can benefit from a good dyno tune. Most, if not all, chassis dynos are used in conjunction with electronic engine analysers. This allows major as well as minor system faults to be detected. It's amazing how little things such as a condenser on the way out, or faulty contact breaker action can go totally undetected by any other means, yet these seemingly minor things all serve to pull the power down.

Importantly the electronic engine analyser also has the ability to measure an engine's air/fuel ratio. This makes for easy trouble-shooting of carb calibration problems and allows the jetting of carbs to be dialled in right on the money.

If you think setting up an unmodified engine to workshop manual specs will give your engine all its potential performance, you have a lot to learn. A little thing called "production tolerances" knocks that theory down flat. As far as maximum performance is concerned, every engine coming off the production line is unique. It is not uncommon to take a car with factory settings on every changeable item, set it up on the dyno and see 10-15 percent increase in power. Such setting up includes jetting the carb to exactly meet the engine's requirements, setting the ignition timing for maximum power, selecting the ideal heat range for the plugs and so on.

When we consider modified engines, it's a whole new ballgame. There are so many combinations of speed equipment available that the undisciplined technical mind can boggle. There is no book to refer to (not even this one) which will tell you what main jet should be used with what carb, when so-and-so's cam, together with what's-his-name's head is used. Whichever way you look at it, the dyno must be the last stage of your engine's modification program. Try as I might, I find it almost impossible to convince some people of this. They will spend a grand on their engine and get only a fifth of that worth in terms of extra power, because they won't spend 50 notes on a good dyno tune. It's not uncommon, on an extensively modified engine, to bolt the thing together and have no more power than a standard or near-standard engine. Admittedly, if persevered with, a lot of the extra potential power can be wrung out of the engine, but it may take 10 race meetings to do it. Even then, you can't be sure all the power has been found. Doesn't it make sense to set the engine up on the dyno and get it right the first time around? Doing things this way, you are sure to beat the guys who don't know about dynos, even if they do have more highly-modified engines.

There is also another aspect to consider: an engine which is set up wrongly is potentially self-destructive. A little too much ignition advance or a little too lean a mixture may cause the engine to eat its innards. Trying to save the cost of a dyno session and losing an engine in the process is a poor bargain.

Turbocharged competition engines are the worst offenders here. They have the potential for so much power from such a small unit that they will literally burn themselves out unless everything is just right. Not only are maximum power settings critical, but so are part-throttle, idle and transition settings of both carb and ignition. The moral here is, don't skip the dyno session. It is just as important for your engine as the bolts that hold down the cylinder head.

AND IN CONCLUSION...

Although an enormous amount of information was gleaned from various engines, I still feel just a little thwarted. Why? Because racing and race engine development is a pursuit of the ultimate. In pursuing the ultimate, it is necessary to extend your resourcefulness to the ultimate as well.

If money were absolutely no object in the developmental program, I could feel confident that the limits of power and reliability could have been taken even further.

However, money was an object here. Thus, there was some changes, mostly very expensive, that were left untried. For example, I would like to have tested a set of titanium connecting rods that were maybe 1/4 inch longer than the ones used in our testing. Also, I would have liked to have explored the performance effect of a two-ring piston design having the top ring a little closer to the piston crown.

Unfortunately, though, these and other aspects of Pinto thermomechanical dynamics must be tested on other occasions. But, believe me, even though I and many others are continually chasing the ultimate, we'll never quite make it. Why? Because the ultimate cannot exist outside the pages of a philosophy book. Even though the development of this engine has reached a fair degree of sophistication and performance, it would be foolish to think it has been taken all the way. Indeed, even as you read this, new things are being tried and evaluated.

In the racing world, nothing stands still....

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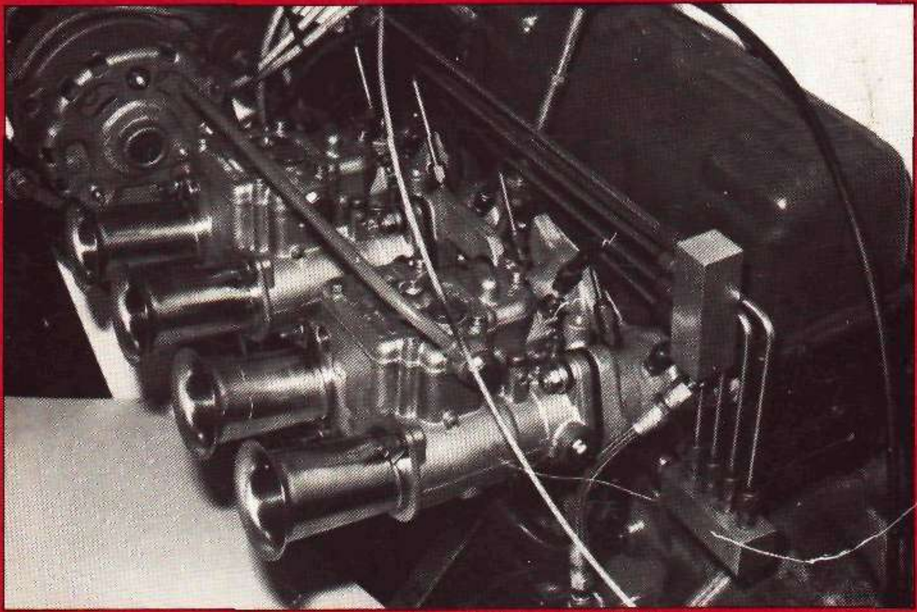
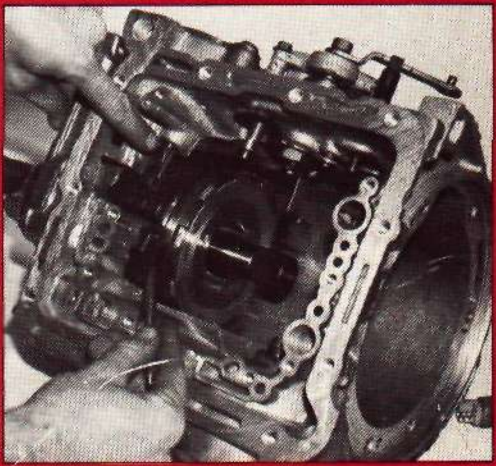


Side-draft Weber carburetors and nitrous oxide, page 127.



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