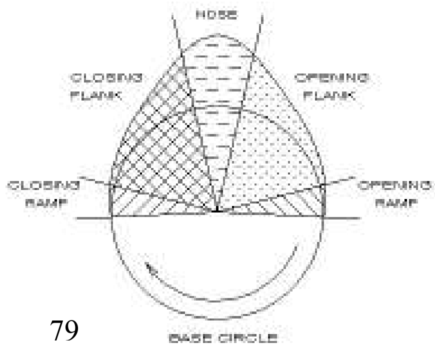
# CHAPTER 5.
# THE CAMSHAFT.

## CAMSHAFT CHOICE.

The camshaft is undoubtedly the most important single component to be selected when tuning the four-stroke engine.
The correct choice is often difficult to make and this chapter is principally concerned in setting out the problems and facts, in order that the correct decision can be made for an individual application.




79

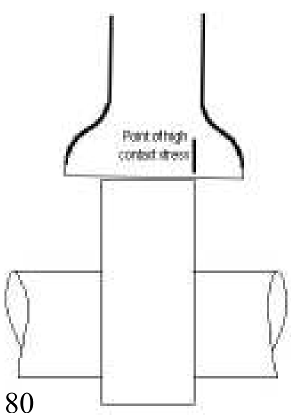
Recognition of the profile.
The cam lobe is made up of four essential elements (Fig. 79). The base circle or clearance circle, often called the heel, together with the ramp - the flank - and the nose.

The BASE CIRCLE is the area of the cam in which little or no contact takes place with the cam follower.
The centre of the base circle duration lies at approximately 180 degrees from the nose centre line and is the point at which valve clearances are normally set.

The RAMP is the area joining the base circle to flank, and is designed to take up valve clearance in a controlled manner, immediately prior to the start of valve lift.

The FLANK lifts the valve train with the spring in compression and accelerates to its maximum speed.

The NOSE takes over at this point and controls the valve train deceleration until it momentarily comes to rest at full lift, when the process reverses itself to the lower valve back to its seat, where the ramp will re-open the clearance.

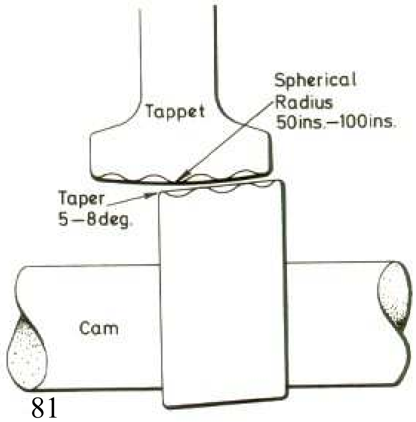



80

The whole procedure exerts an enormous strain on the components involved, sometimes ‘stacking-up’ contact stresses over the cam nose as high as 1300 Meganewtons/sq.m. (200 lbs/sq.in.), calling for a high degree of accuracy in design and manufacture, together with the need for great care and attention when fitting.

The cam nose stress stack-up is the result of a number of unavoidable design constrictions.
No matter how tight the engine manufacturing tolerances, there will always be a small amount of mis-alignment or ‘out-of-square’ mating between components, sometimes causing hairline or pressure point contact. (Fig. 80)

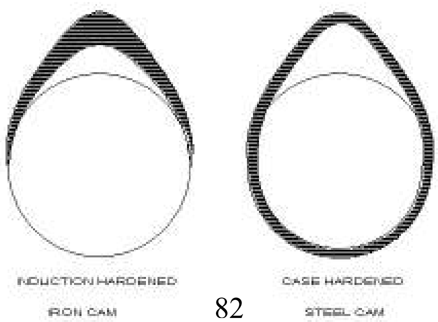
In order to minimise the effects of this possibility, cam lobes are often purposely machined with a taper to mate with a spherically ground tappet face (Fig. 81), creating an intentional but calculable high pressure footprint, slightly offset from the tappet centre-line to promote rotation of the tappet, thus improving overall service life.




81

This design feature means that individual cam lobes and tappets 'bed together' after an initial running period, and should therefore never be allowed to re-mate with other components during later re-assembly procedures.

Camshafts are made from either cast-iron or steel with the latter usually recognisable by their smooth forged or turned finish between the lobes as opposed to the rougher finish of the former which are left 'as cast' in this area.
In some cases they are also turned between the lobes to reduce the core size, usually done when an increase in lift dictates a smaller base circle diameter.
Cast iron shafts made of 'proferal' or 'K' iron, have the lobes and gears heat treated by flame or induction hardening processes, whereas 'chill cast' components are hardened in the vital areas during the casting process.




82

Whatever production method is used, cast-iron cam lobes finish up with a hardness pattern as in (Fig. 82), that is, about a quarter of an inch depth of hardness over the nose, tapering in depth down each flank.

This means that the base circle is usually relatively soft, which is acceptable because there is little or no load at this point. Because of this hardness pattern, these cams are particularly suitable for regrinding; only requiring final refinishing with a black, oil retaining, phosphate coating.

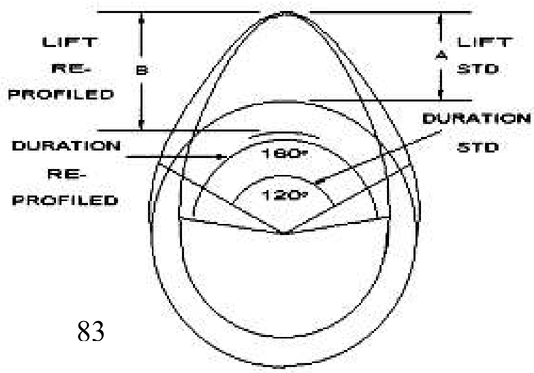
Steel cams, on the other hand, are case hardened, which means that they finish up with a thin hardened layer, usually about 1.0mm thick, which is penetrated when the cam is reground.
This necessitates heat treatment or hard facing to regain acceptable hardness after a regrind.
Lobe hardness over the nose should be 50-53 Rockwell 'C' on cast iron and 54-58 Rockwell 'C' on steel.

CAMSHAFT REGRINDING – HOW DOES IT WORK?

A few years ago, by far the majority of European high performance camshafts were produced by the process of re-profiling the standard cam.
Contrary to commonly held misconceptions, this procedure, if properly engineered, results in a product that is equal, both in reliability and performance, to the same component made from raw billet.
Although it would be ideal to make all camshafts from new billets to avoid the transportation problems of exchange units, this is just not practical in Europe, due to the vast variety of makes and models, often coupled with the non-availability of unmachined castings from the original manufacturers and the obvious poor economics of producing special casting to meet the small demands for one particular model.
This situation doesn't apply in the United States where a relatively small number of billets cover a very wide range of vehicle.

To many people the process of regrinding a camshaft is a black art, resulting in the inevitable question :
'How can you machine metal away from a cam, yet have it finish up with more duration and lift?'

It works like this…



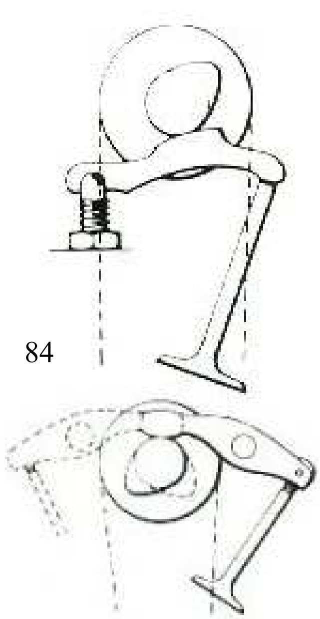
83

In (Fig. 83), the outer contour represents the original lobe shape and the cam lift is represented as dimension 'A', the difference between the base circle radius and the nose.

The duration of the lift is shown as 120 degrees at the cam, which would be 240 degrees at the crankshaft, because the camshaft rotates at half crank speed.

The inner contour line shows the cam shape after it has been reground and it can be clearly seen that the lift has now been increased to the dimension 'B', this time the difference between the new basic circle radius and the nose.
At the same time the duration has been increased to 160 degrees at the cam, that is 320 degrees at the camshaft.

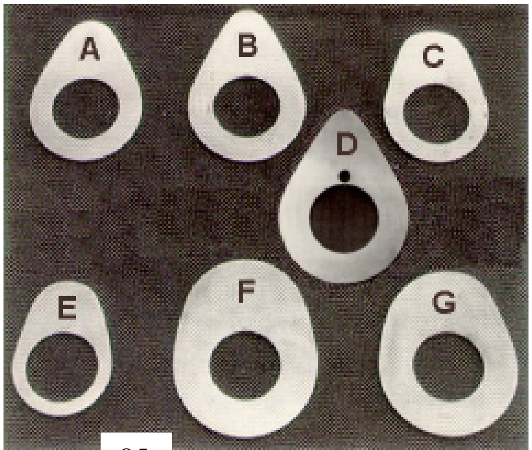
From this diagram it is also now possible to see why the cast iron cam lends itself so readily to regrinding. The deep hardness pattern over the nose is still fully effective.
As contemporary high efficiency engines are being evolved, the process of regrinding will no longer be acceptable for a number of reasons.
The increasing use of hydraulic tappets, fitted to automatically adjust running valve clearances to compensate for changes in engine dimensions due to heating and cooling effects, means that nominal base circle sizes may not be altered.
In this case, the replacement camshaft will need to be produced from a new billet or casting.
However, restrictions such as slide-in camshaft bearing diameters will often control the overall height of the cam lobe and thus mean that the new billet cam will have lobes no larger than those of a re-ground shaft.


84

This in turn means that the hydraulic followers will have to be replaced with solid followers and a method of valve clearance adjustment introduced.
In any circumstances, it is desirable to switch to solid lifter operation wherever possible for efficient high performance engine conversion.

For the same reasons, the use of various types of overhead cam operating systems will have layout geometries that will not allow significant changes in cam dimensions. (Fig. 84)
The high performance replacements for these will, not only need to be manufactured from new billets, but will also need to use 'state of the art' profile design techniques to give the best results.

It is a general, but mistaken impression that it is possible to look at a cam shape and say whether it is for road or track.

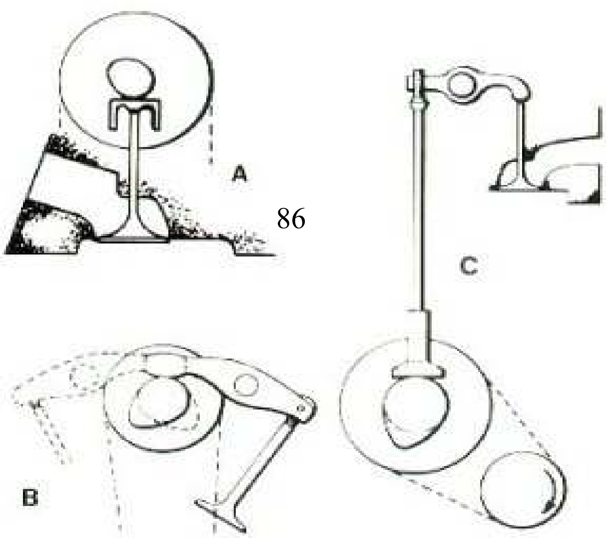



85

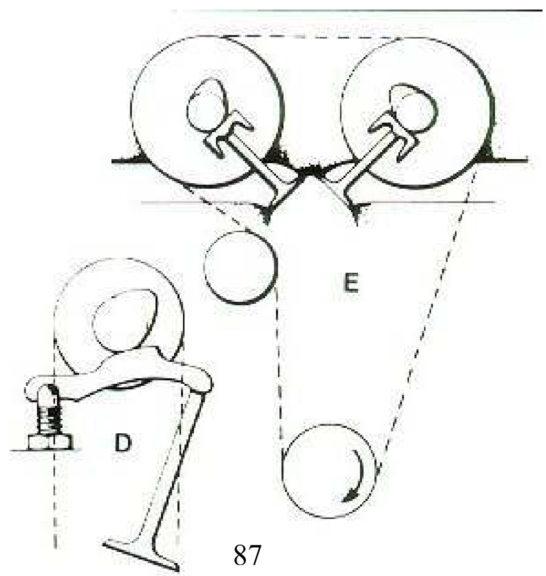
This is clearly illustrated in (Fig. 85), which shows a selection of racing cam shapes from a variety of engines.
‘A’ is a standard Fiat, ‘B’ is a racing XK140 Jaguar, ‘C’ is a racing Hillman Imp.
‘D’, which most ‘experts’ would define as a very mild cam, was the profile used in the contemporary Golf 1.8 GTI racing engines.
‘E’ is a standard Triumph Bonneville, ‘F’ is a standard Ford 2.0 OHC, ‘G’ is a racing Honda.

The shape of the lobe is entirely dependent on the design of the components that work with it and will vary hugely from engine to engine.

The technical requirements of successful high performance cam design require attention to a number of simple but vital engine component functions.




86

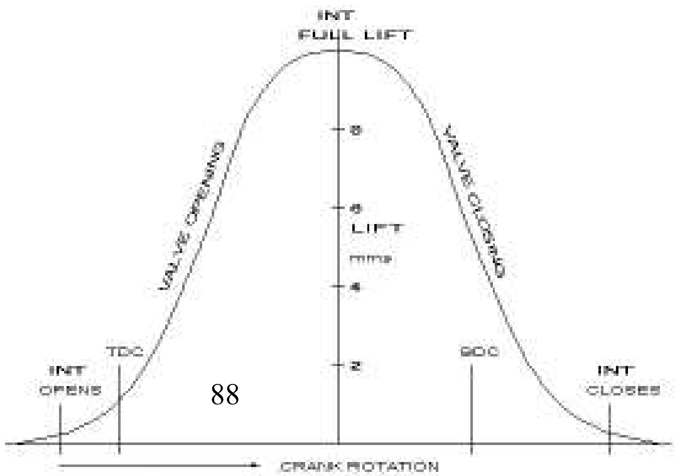



87

The vast majority of production engines have valve train mechanisms that generate ‘symmetrical’ motion at the valve. (Figs. 86 & 87)

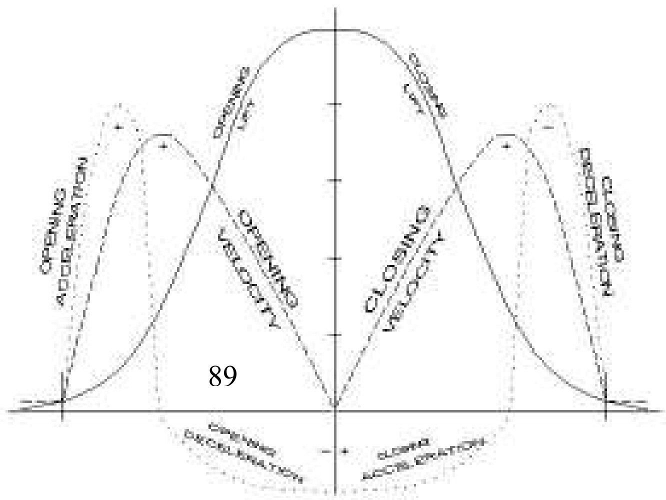
This is achieved by either directly moving the valve with conventional ‘direct operating’ cam technique (Fig. 86a), in which case a symmetrical cam profile imparts a symmetrical valve motion, or by moving the valve via a rocker mechanism, which imparts symmetrical valve motion by compensating for the constantly varying rocker ratio by using an asymmetrical cam profile.

However, modern computer design and analysis techniques allow for the development of cams that impart asymmetrical motion to the valve, thus taking advantage of part of the generous safety margins that are built into standard engine valve train dynamics.




88

If we study the movement of a valve during its' opening and closing cycle as a graph (Fig. 88), then the actual valve lift is shown as a single smooth curve on the upper, or positive side of the zero line.




89

The velocity is seen as two sharper positive curves, representing the speed of the valve as it lifts to full lift, pauses momentarily, then speeds up once more as it starts to close. (Fig. 89)

Finally, the acceleration is shown as positive as the valve accelerates to maximum velocity, then negative as it decelerates to rest at full lift, followed by a reverse cycle of acceleration and deceleration as it closes.

The design of this acceleration pattern is the point at which all cam profile design starts.

Normally this pattern is symmetrical and is controlled by the maximum deceleration that occurs at full lift, from which the valve spring dimensions are calculated, the speed at which the valve is lowered back onto the valve seat in order to avoid valve bounce, and the amount of ‘wind-up’ that occurs in the valve train due to flexibility of the components.
This last factor is of course the reason why overhead direct acting valve trains can be made to operate at far higher speeds than systems using rockers or push-rods, hence the evolution of the twin OHC engine.

Using computerised design techniques and high precision manufacturing methods, the valve dynamics can be modified by distorting the symmetry of the acceleration, velocity and lift curves, to give a greater proportion of valve open, or breathing time, within any given valve duration.

The result of this strategy is that, for any given valve duration (i.e. time of valve leaving then returning to its' seat), the actual open area through which gases can move is increased.
This means that a relatively short duration cam, giving good low-end performance, can also allow enough breathing area for high speed volumetric efficiency and consequently increase power.

These poly-dynamic cam profiles represent a significant step towards the all-purpose high performance camshafts, but it should be remembered that the high valve opening accelerations impose stresses that are eroding the normal standard production safety limit margins.

## CAMSHAFT POSITION AND DRIVE

Referring again to Figs. 86&87, the camshaft and associated valve train component layouts covering 95% of modern engine designs are shown, together with their applications, virtues and disadvantages as listed below :

| KEY | USED BY | VIRTUES | DISADVANTAGES |
|---|---|---|---|
| (A) S.O.H.C. Direct Operating | Rover Gp. V.W. Audi SAAB Honda many others. | Single Camshaft Simple Drive Belt or Chain | Valve size Restriction due to in-line layout. Bore size controls sum of inlet and exhaust seat diameters. Ports often too long. |
| (B) S.O.H.C. with Rockers | B.M.W.,Mazda Colt,Moskvich, BL,Peugeot, Porsche, Toyota Honda, Yamaha, | Allows better valve placing, still simple drive. | Spark plug position is restricted due to rockers and shafts. |
| (C) Pushrod | Mostly OHV industrial manufacturers. | Ease of servicing. | Flexibility and weight of a long train of components. |
| (D) SOHC Loose follower finger. | Datsun (Nissan), Fiat,Ford, Lada, Vauxhall. | Ease of servicing. Choice of valve position. | Flexible by normal OHC standards. Excessive overall height. |
| (E) Direct operating. | AlfaRomeo, Aston Martin, Ferrari, Honda, Lancia, Lotus, Maserati, Jaguar, Toyota, Suzuki, Kawasaki Many others. | The only true Twin Cam. racing layout. Allows freedom of valve and plug position. | Costly to manufacture. Difficult to service, but becoming easier with hydraulically adjusting bucket followers. |

## CAMSHAFT INSTALLATION.

As stated earlier, the camshaft is the most highly stressed component in the engine and therefore requires particular care when being fitted.

The majority of cast-iron cams have a black phosphate coating.
The purpose of this is not, as many people think, a surface hardening process but it is for oil retention during the early life of the cam. It carries out this function admirably but unfortunately also retains any dirt that is brought into contact.
Even just handling a camshaft with grubby hands while fitting, can implant enough tiny particles of grit to seriously shorten its life.

## THE RULES OF SUCCESSFUL CAMSHAFT INSTALLATION.

Research indicates that the majority of cams that wear out, start to fail during the first few moments of operation.
Many cams are irreparably damaged, even before the engine is started, because the basic rules of camshaft break-in have not been followed.

The cause of premature cam and tappet failure is metal-to-metal contact between the tappet and cam lobe. Should this contact occur due to lack of proper lubrication, or excessively high pressure due to valve train interference shearing the oil film, then 'galling' will take place.
When this happens, metal is transferred from the tappet to the cam or vice versa in a process comparable to welding. Microscopic high spots, which are present on all machined parts, become overheated due to friction and pressure bond together, tearing sections loose from the tappet or lobe. These pieces of metal remain attached to the mating part, creating further local overheating during the following revolutions of the camshaft, leading to ultimate failure of the affected components.

Listed below are the mistakes that lead to premature failure:

1.Inadequate lubrication during the initial rotation of the camshaft with full spring load applied.

2.Interference in the valve train due to improper installation and failure to check for interference.

Valve spring coil boxing, spring collar to guide contact, valve to valve contact and valve to piston contact are the main problems.

3.Installation of used tappets with a new camshaft.
No matter how good tappets look, new tappets must be used with a new camshaft!!!
Beware of reclaimed tappets; they are usually ground flat whereas geometric relationships between cam and follower, often involve the use of a tapered cam lobe working with a spherically radiussed tappet foot.

Fig. 81 (above), shows an exaggerated view of this condition, which essentially comprises a cam lobe taper of around 6-10 minutes! That's about 0.025mm over 13mm and a spherical radius on the tappet foot of 1500-2500mm!
The centreline of the tappet is also offset from the lobe centreline by about 1.0mm.

4. Water, petrol or other contaminant in the oil that can lower film strength, or create abrasion.

5.Excessively long cranking on the starter. Oil will not reach cam lobes until engine is running.

6.Low idle speeds during break-in. Cam lobes in pushrod engines usually depend on all thrown from con-rods for lubrication. Oil delivery will not be sufficient at idle.

A word of warning !

Cam profiles, particularly high performance profiles, have sensitive areas of dimensional tolerance at the point where the flank joins the ramp, and the contour over the nose.
Even at the point of original manufacture from the high precision master cam, small but acceptable contour discrepancies occur.
However, any attempt to further copy the camshaft will degrade the profile contour, just as videotape degrades with copying.
In order to avoid the problems that this causes in the valve train dynamics, together with the subsequent reductions in performance and reliability, you should confirm that your camshaft supplier is the original designer and production originator, rather than a machine shop specialising in producing copies of other manufacturers' original designs.

Reputable camshaft manufacturers and suppliers will themselves take several other precautions by:

1. Supplying cam profile designs that are not overloaded or highly stressed.
2. Provide cams with the correct machined finish.
3. Phosphate or otherwise treat cams to assist oil retention.
4. Supply or recommend special oil for assembly.

The mechanic handling the installation bears the greatest responsibility for break-in of the camshaft.
The following outlined steps will help ensure long and trouble-free life from the camshaft and associated components :

1. Coat the cam lobes and cam face of the tappet with lubricant.
If a proprietary cam lube containing Zinc-Dio-Thio-Phosphate (ZDTP), like Piper Cam Lube, is not available, then an E.P. 140 or 90 Hypoid rear axle oil is the next best alternative.

2. Check entire valve train for interference before attempting to start engine, and particularly check that the cam eccentricity or wipe path across the follower, does not run off the edges. High velocity cams will wipe across a much wider face path than standard cams.

3.Set pushrod engine valve clearances 0.003in to 0.005in smaller than specified for initial start-up.

4.Before starting any engines, prime the oil by turning oil pump manually. Fill carburettor with petrol, fill radiator and ensure correct ignition timing. Engine must start right away and not be subjected to a long grind on the starter.

5.Do not idle engine during the first twenty minutes of operation.
Rpm should be kept at 2500 or above.
In pushrod engines oil throw-off from the crank may not be sufficient to lubricate the cam followers. Also contact stresses at the nose of the cam are very high at low speed. Engines may be run in the shop or on the road or strip. If adjustments need to be made during the twenty minutes break- in period, shut the engine down. DO NOT LET IT IDLE!

If the engine is dismantled for repair, maintenance or inspection, after any running at all, it is important that the tappets be kept in order.
Each tappet will have mated to a cam lobe and swapping tappets may cause failure.

CAMSHAFT - SELECTING, CHECKING AND SETTING TIMING.

Guide to correct camshaft selection.

Std:Average standard engine figures.

Stage 1:Normal commuting.

Stage 2:Fast road use with lightly modified engines.

Stage 3:Ultimate fast road. Mild rally. For use in modified engines.

Stage 4:Rally. Short circuit, loose surfaces.

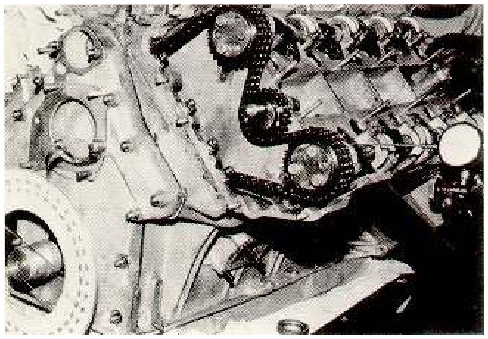
Stage 5:Circuit race tarmac. Ultimate short circuit loose surface.

Stage 6:Ultimate tarmac. Drag racing.

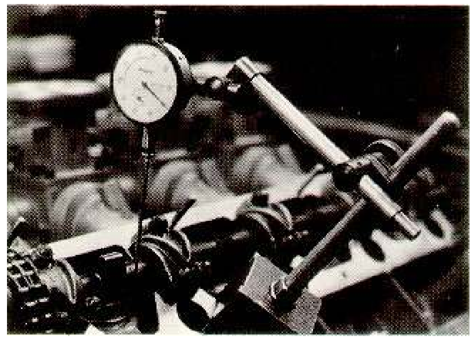
| Stage | Cylinder Capacity (ccs) 250 | | | | | 350 | | | | | 500 | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | IO | IC | EO | EC | VL | IO | IC | EO | EC | VL | IO | IC | EO | EC | VL |
| STD | 20 | 45 | 45 | 15 | 7.6 | 20 | 45 | 45 | 15 | 8.6 | 20 | 45 | 45 | 15 | 9.9 |
| OR | 1000-6000 | | | | | 1000-6000 | | | | | 1000-5500 | | | | |
| ST.1 | 30 | 60 | 60 | 30 | 8.9 | 25 | 65 | 65 | 25 | 9.4 | 25 | 65 | 65 | 25 | 10.1 |
| OR | 2000-7000 | | | | | 1500-7000 | | | | | 1500-6500 | | | | |
| ST.2 | 38 | 66 | 68 | 36 | 9.6 | 40 | 76 | 76 | 40 | 9.6 | 34 | 62 | 64 | 32 | 10.1 |
| OR | 3000-7000 | | | | | 2000-7000 | | | | | 1500-7000 | | | | |
| ST.3 | 45 | 73 | 73 | 45 | 9.6 | 44 | 78 | 78 | 44 | 10.1 | 40 | 74 | 74 | 40 | 10.7 |
| OR | 3500-8000 | | | | | 2500-7000 | | | | | 2000-7000 | | | | |
| ST.4 | 48 | 78 | 78 | 48 | 9.6 | 48 | 78 | 80 | 46 | 9.6 | 48 | 78 | 80 | 46 | 11.2 |
| OR | 4000-8000 | | | | | 3500-7500 | | | | | 3000-7000 | | | | |
| ST.5 | 54 | 86 | 86 | 54 | 9.6 | 54 | 86 | 86 | 54 | 9.6 | 54 | 86 | 86 | 54 | 11.7 |
| OR | 5000-9000 | | | | | 4500-8000 | | | | | 4000-7000 | | | | |
| ST.6 | 58 | 88 | 88 | 58 | 9.6 | 60 | 86 | 88 | 58 | 9.6 | 60 | 90 | 90 | 60 | 12.7 |
| OR | 8000-10,000 | | | | | 7000-9000 | | | | | 7000-9000 | | | | |

IO: Inlet opens before T.D.C. IC: Inlet closes after B.D.C. EO: Exhaust opens before B.D.C.
EC: Exhaust closes after T.D.C. VL: Valve lift (mms)
OR: Operating R.P.M.range (Generally for car engines. Small capacity multi-cylinder motorcycle engines will run to appreciably higher figures).

Having selected a camshaft of suitable duration and lift from the sections on inlet and exhaust characteristics and the chart above, it is now necessary to ensure that the timing is zeroed in to the specified figures.



90



91

92

To accurately set or check timing it will be necessary to use a protractor bolted firmly to the crank nose, and a dial gauge attached to the cylinder head, registering motion of the appropriate valve. (Figs. 90 - 92)

First zero the protractor at T.D.C.
The most accurate way of doing this is to use a modified spark plug, with a fixed probe in place of the electrode, that stops piston motion a small amount before T.D.C.
Rotating the crank slowly backwards then forwards will enable accurate zeroing between the two points at which the piston is stopped. After this, all further timing checks must be carried out in the correct direction of rotation to ensure that all valve train 'slack' is taken up in the normal way.
As an example, let's suppose that a camshaft has been selected with the following characteristics:

| | | |
|---|---|---|
| Inlet | 40 B.T.D.C. - 72 A.B.C.D. | Valve lift 10.0mm. |
| Exhaust | 76 B.B.D.C. - 36 A.T.D.C. | Valve lift 9.7mm. |

Method 1.

Set valve clearance to a known figure, say 0.25mm. (0.010in.)
Zero the dial gauge with the valve closed.
Turn the crank slowly and watch for a reading of, say, 0.1mm (0.005in.) of valve lift and note the protractor reading.
Continue to turn the crank through the opening and closing phases of lift until the dial gauge returns to the same reading and note the protractor reading again.
Repeat the procedure with the other valve and then convert the protractor readings to their appropriate 'before and after T.D.C. and B.D.C.' status.
This method is often not very satisfactory due to the fact that uncontrollable valve train movement takes place, caused by clearance or wear in rockers and shafts or tappets and guides etc., which tends to give a distorted indication of start and finish of valve motion.

Method 2.

If the valve lift at a given crank angle is known, say at T.D.C., then the timing can be set accurately by the following procedure.
Install the camshaft on standard timing marks with the valve clearances set to the correct running valve.
Zero the dial gauge with the chosen valve on its seat.

Turn the crankshaft to T.D.C. and note the valve lift. If it is not correct then disconnect the cam drive and rotate the camshaft alone until the correct lift is achieved.
Re-couple the drive using one of the methods discussed below to accommodate the need for a small amount of angular change.
Twin-cam engines will need a lift figure for both inlet and exhaust valves.
Always start by adjusting the shaft that comes first in the drive-line and always ensure that the drive belt or chain is held at running tension while readings are being taken.

Method 3.

Valve full lift position.
This is probably the most accurate but tedious method, and requires that the full lift position of the inlet and/or the exhaust valve be worked out in the following way.
Referring to our 'example' cam at the start, the inlet timing is :

40 B.T.D.C. - 72 A.B.C.D.

From these timing figures we can say that the total timing duration is :

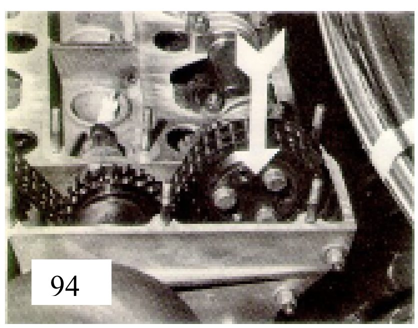
40 + 180 + 72 = 292 degs.

The full lift of the valve will occur halfway through this period, which is 292 divided by two, or 146.
So, if full lift occurs 146 after the start of valve motion, and valve motion itself starts at 40 before T.D.C., then full lift must occur at 146 minus 40, or 106 degs. after T.D.C.
Similarly, the full lift position of the exhaust valve can be calculated to be 110 degs. before T.D.C.

The cam timing can now be set by following the initial procedure described in 'Method 2' and then, turning the crank to the appropriate position say in the case of the inlet, 106 A.T.D.C.
At this point, the cam drive can be disconnected and the camshaft rotated slightly, until the inlet valve is fully open, at which point the drive should be reconnected.

Don't rely on eyeball judgement of the dial gauge to decide the position of full lift. Velocity in this area is so low that there is an apparent period of dwell and it would be inaccurate to try and guess the centre point of this period.

To be safe, take a point each side of the full lift, say 0.5mm before and after, note the protractor readings at these two points, add the two together and divide by two. This will give you the true angular reading.


Having checked and corrected the timing, it will probably be necessary to use an offset dowel or key (Fig. 93) to couple up the drive in the correct position, although some of the more sophisticated engines are built with a vernier cam drive adjustment, which makes life a lot easier. (Fig.94)

Kawasaki, Honda, Yamaha, and other centre drive motorcycle cams, will have to be advanced one complete tooth and the drive bolt holes elongated in the direction opposite to rotation until the correct position is reached.

Hydraulic Tappets.

Due to the ‘leak-down’ feature of their design, hydraulically adjusted tappets or rocker pedestals should be temporarily replaced with modified ‘solid’ replacements, whilst carrying out camshaft setting operations.
This need only be done for one cylinder to set the cam timing and any of the above timing methods may be used.
After setting is completed, the ‘solid dummies’ should be replaced with the working components for final assembly.

## TWO - STROKE PORT TIMING.... CHECKING AND MODIFYING.

Although not generally recognised, there is a close affinity between the two-stroke and four-stroke engine in many respects.
Although the two-stroke does not have a camshaft as such, its function is performed by the piston, opening and closing the valves at the correct moment in relation to crank rotation.

As described earlier, these are in effect sleeve valves, which do not have efficient flow capability and therefore have to be proportionately larger than poppet valves in relation to a given cylinder size.

Whilst it is relatively easy to change the valve timing of a four- stroke by the simple process of changing the camshaft, it requires a lot more skill and sensitivity to efficiently change the timing of a two-stroke.

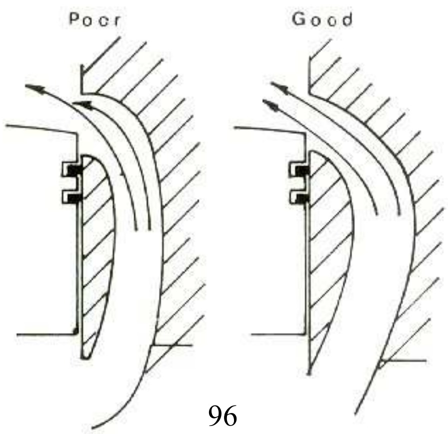
As with all engines, the first step in accurate checking is again to fit a protractor to the crank, with a firmly mounted pointer that will give an exact and repeatable indication of crank angle.
It should be zeroed at T.D.C. by the piston stop method described earlier in the four stroke timing section.

Inlet timing can be measured by looking straight into the inlet port and trapping a thin, say 0.05mm 0.002in. feeler gauge between the bottom of the piston skirt and the lower edge of the port.
Take a note of the protractor reading.
Rotate the crank so that the piston rises to T.D.C. and traps the feeler again as it is coming down, at which point take a further note of the angle.
The intake duration is the sum of the angles to and from TDC, typically 140 to 180 degrees, depending on level of tune.

Production engine timing often relies on casting and coring accuracy and can normally be considerably improved by attention to detail in the port mouth and to the respective area of the piston.

Transfer port timing can be measured by visually checking piston motion while looking down into the cylinder.
The lower lip of the port should align with the piston crown at B.D.C.

Internal modifications to the transfer area are much more difficult than those to the other ports and should be carried out by experienced personnel using the right equipment.
General rules are to maintain gentle port curvature. (Fig. 96)
Any sudden changes in shape mean loss of energy charge.




96

Transfer timings will vary from around 75 degs. each side of B.D.C. in road engines to about 62 degs. each side of B.D.C. in full race units.

Unlike four stroke characteristics, increased two-stroke transfer timing duration lowers maximum power R.P.M.

If the transfer timing duration is decreased then the R.P.M. at which maximum power is produced will be higher.

This is due to the fact that crankcase pressure is increased and maintained, either by the addition of a disc or reed valve, or by high-speed induction ram.

However, in general, transfer timing alterations do not have a great effect on power output and provided sensible matching and smoothing is carried out, time is better spent on other tuning details.