



Making the Cam

This is the second of a three-
instalment Special
Investigation into valvetrain
design and it looks at the
production of cams and their
followers. Our guides
throughout this Special
Investigation are Prof.
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Hans Hermann Engineering.

THE FUNDAMENTALS

As we noted in the first instalment, when one opens up the program for 'cam design and manufacture' in the 4stHEAD software [1] the user is faced with the following quotation from the writers of this computer package. It is as follows: "There is no such thing as cam design, there is only valve lift profile design which requires the creation of a cam and follower mechanism to reliably provide this designed valve lift profile."

In Part One of this Special Investigation we described the creation of valve lift profiles. Here, in Part Two, we must describe the "creation of a cam and follower mechanism to reliably provide this designed valve lift profile" if we are not to be hoisted on our own petard!

CREATING THE CAM FOLLOWER MECHANISM

In Fig.1 is shown a cutaway picture of a direct-acting cam follower mechanism in the form of a bucket tappet acting on a valve restrained by valve springs. To "reliably provide" the designed valve lift profile this mechanism, like all other cam follower mechanisms, must do so for the duty cycle envisaged by the designer. That duty cycle may range from 1000 hours of urban driving by Joe Bloggs in his road car to a one-race scenario at full-throttle before replacing the entire valvetrain for the next race. In either case, the designer will designate permissible levels of stress to be imposed on all of the components of the mechanism and acceptable levels of lubrication between its moving surfaces.

The decision on which type of cam follower mechanism is to be used is not always a free choice for the designer. It may be

mandated by regulations, such as a pushrod system in NASCAR, or to be similar to that of the production vehicle if a Le Mans GT car. In any event, the geometry of the cam follower mechanism must be created and numerically specified in the manner of Fig.2 for a pushrod system, or similarly for finger followers, rocker followers, or the apparently simple bucket tappet [1]. Without knowing that geometry, the lift of the cam tappet follower and the profile of the cam to produce the desired valve lift diagram cannot be calculated.

THE HERTZ STRESS AT THE CAM AND TAPPET INTERFACE

As the cam lifts the tappet and the valve through the particular mechanism involved, the force between cam and tappet is a function of the opposing forces created by the valve springs and the inertia of the entire mechanism at the selected speed of camshaft rotation. This is not to speak of further forces created by cylinder pressure opposing (or assisting) the valve motion. The force between cam and tappet produces deformation of the surfaces and the "flattened" contact patch produces the so-called Hertz stresses in the materials of each. Clearly, the extent of this deformation depends on the materials involved and their physical properties.

Racing cams are normally made from hardened steel, but chilled cast iron is used in many industrial engines and 'plastic' cams with sintered iron tappets can be found there as well. The computation of the Hertz stress must take account of the physical properties of both the cam and the cam tappet surfaces and the permissible Hertz stresses then depend on the desired duty cycle for them.

A useful working 'rule of thumb' for the maximum value of the Hertz stress which can be tolerated in racing cams and cam tappets made from hardened steels is about 1250 MPa. Such steels are normally hardened to Rockwell 64C or 65C.

You should recall this discussion as we present here computed Hertz stress as a function of the design parameters.

LUBRICATION OF THE CAM AND TAPPET INTERFACE

The cam profile and the cam tappet interface are normally produced by grinding the surfaces. Typically, a simple grinding operation will produce the surface finish that is measured at 0.25 micron as a centre-line-average (CLA). This is the average height of the asperities on the surface. If the surface is first ground and then polished that surface finish will be improved to about 0.1 micron CLA. They are often further polished to a mirror finish at 0.01 micron CLA, particularly for flat tappets. The outer shells of production roller bearings supplied for cam tappets also tend to be polished to that same level.

There is a good reason for having a good surface finish on the cam and cam tappet profile as the lubrication film between them tends to be on the order of 0.4 to 1.0 micron thick. To put this in ►



Fig.1 The cam, cam tappet, valve, and valve springs.

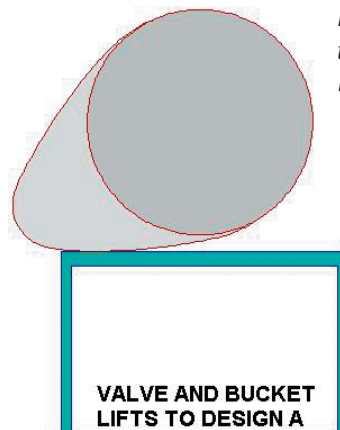


Fig.3 The cam and bucket tappet for valve lift Design A.

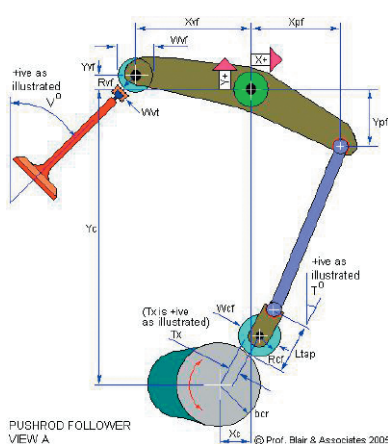


Fig.2 Geometrical data for a pushrod cam follower mechanism.

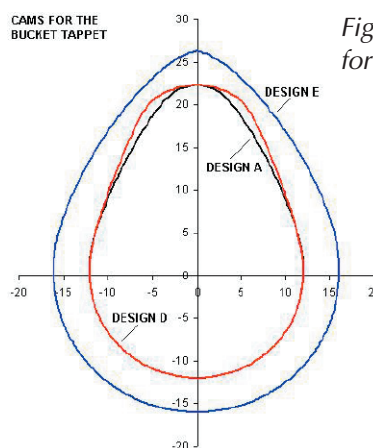


Fig.4 Bucket tappet cams for valve lift Designs A-E.

context, if the cam and cam tappet surfaces each have surface finishes of 0.25 micron and the lubrication film at its thinnest is 0.5 micron that implies that the asperities on both surfaces are just rubbing on each other. Even if they do not actually scuff or seize, the ensuing asperity-asperity polishing action gives extra friction loss to their motion and that is race-engine power one would prefer to have applied elsewhere.

You should bear in mind this discussion on the surface finish of cams and tappets as we present here computed oil-film thickness as a function of changes of the design parameters.

THE VALVE LIFT PROFILES FOR THE CAM MECHANISMS

You will recall, in Part One of this article [2] on valve lift profile design that among those presented were Design A, Design D and Design E.

These designs will be used here to numerically illustrate this article, so you can refer back [2] to find their characteristics of lift, duration, acceleration, velocity and jerk.

Design A was characterised, somewhat glibly, as one with a middle-of-the-road lift aggression characteristic best suited to all but pushrod followers. Design D was very aggressive and was mooted to be best used with bucket tappets or finger followers but opined as being dynamically unsuitable for pushrod followers. Design E had a lower lift aggression characteristic with a profile suggested as one that is typically used for pushrod follower systems.

CAM PROFILE DESIGN FOR MANUFACTURE

This is a complex, mathematical subject area as even a glance at two excellent textbooks written on the topic will confirm [3, 4]. The sheer intensity of the mathematical procedures requires a formal code written solution. A suitable scientific computer language must be adopted which facilitates comprehensive user and graphical interfaces. The simple spreadsheet does not easily satisfy these requirements.

It is obvious that numerical accuracy of the computation of the cam profile is vital and is typically presented to grinding machines with a tolerance of +/- 1.0 nm. It may be a little less obvious that a visual presentation of the rotating cam accurately meshing with its moving mechanism is at least as important. This is because there is considerable potential for obtaining what appears to be satisfactory numerical answers when the ensuing cam profile, with possible tappet and cam clashing, may yield a mechanical disaster. The 4stHEAD software used here is written in a comprehensive mathematical and scientific computer language with excellent graphics capabilities, which permits these twin design criteria to be met.

For those who wish to delve into this subject mathematically, you should study the books by Chen [3] and Norton [4] but before programming any of their equations you are strongly advised to theoretically prove each equation

Fig.5 Radii of curvature for the bucket tappet cams.

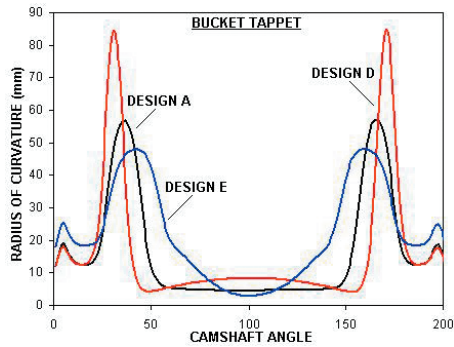
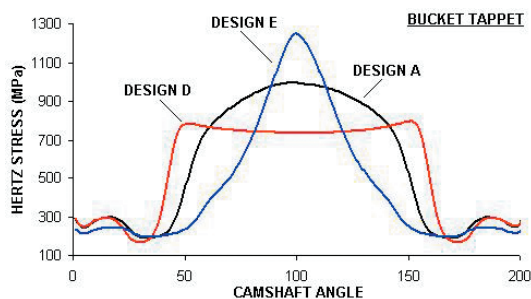


Fig.6 Bucket tappet Hertz stresses for Designs A-E.



out by itself. This way you should eliminate the possibility of an error due to an author's 'typo' that could prove very costly farther down the road of your cam design ambitions.

DESIGNS A-E APPLIED TO A BUCKET TAPPET MECHANISM

Within the 4stHEAD software [1] the data is entered for the physical properties of the materials (normally hardened steels) involved for the bucket and the cam. The oil at the interface is selected from a wide range of straight and multigrade oils. To illustrate the design of a bucket tappet when the valve lift profiles are Designs A-E the selected oil is SAE30 at an 80 deg C oil temperature. Appropriate values for the mass of the valve, the bucket, and the valve springs and the (combined if two) spring stiffness are also inserted as input data. The base circle radius of the cam is selected as 12 mm with a total valve (and bucket tappet by definition) lift of 10.3 mm. The bucket tappet is declared flat, although the computation permits a spherical or domed top to be employed instead. Among many other output data for the computation, an on-screen movie shows the turning of the designed cam with its bucket tappet lifting to the valve lift profile. The bucket is presented as having the minimum possible diameter to keep the declared width of the cam in full contact with the flat tappet surface at all times.

With the valve lift profile employed as Design A, a snapshot from this on-screen movie is shown as Fig.3. Such movies are

Fig.7 Bucket tappet to cam forces for Designs A-E.

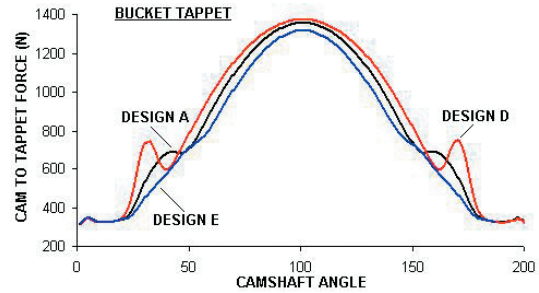
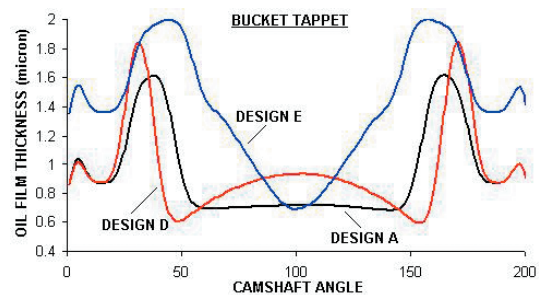


Fig.8 Bucket tappet oil film thickness for Designs A-E.



comforting to the designer, not to speak of the code writers, for if inter-surface clashing does not visually occur then the designer has great confidence that the cam can be accurately manufactured and will be successful in its operation.

When the alternative valve lift profile Designs, D and E, are used instead within the software the differing cam profiles are computed and graphed in Fig.4. The more aggressive profile for the cam to lift the valve is Design D (in red). Its shape makes that quite obvious. The profile for Design E (in blue) is very different because a larger base circle radius of 16 mm had to be used to prevent a very sharp nose developing on that particular cam. Actually, at a base circle radius of 12 mm, as used for the others, it virtually had a point for a cam nose with an associated and impossibly high Hertz stress.

Cam profiles for (flat) bucket tappets are convex and relatively easy to grind with a free choice of grinding wheel diameter as the radius of curvature of the profile is always positive. The radii of curvature of the three cam profiles are shown in Fig.5. The dip towards zero at the cam nose for Design E can be clearly seen even with the use of a larger base circle radius.

The Hertz stress levels for Designs A, D and E are plotted in Fig.6 when the selected camshaft speed was 3000 rpm (6000 rpm at the engine crankshaft). Hertz stress characteristics are a function of camshaft speed because so are the inertia forces. Nevertheless, the most aggressive lift profile, Design D, does not yield the highest Hertz stress whereas the sharper nose of ▶

Fig.9 Bucket tappet oil film thickness and temperature.

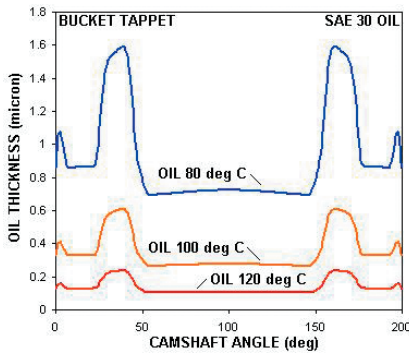
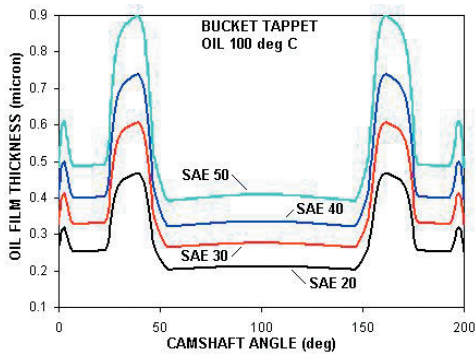


Fig.10 Bucket tappet oil film thickness and oil viscosity.



the least aggressive valve lift profile, Design E, does and reaches the nominal 1250 MPa limit.

That the Hertz stress levels are principally related to the sharpness of the profile of the cam, i.e., the radius of curvature of its shape, can be seen when the cam to tappet forces are plotted in Fig.7. They are found to be somewhat similar for all three valve lift profiles and are highest around the nose of the cam. The Hertz stress is a function of this force but the 'indent' at, and the area of the contact patch,

Fig.11 Input geometry data for rocker and finger mechanisms.

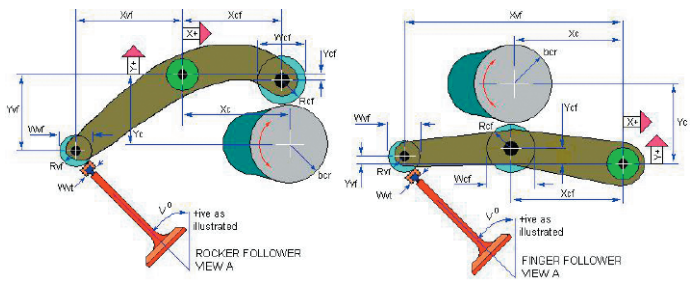
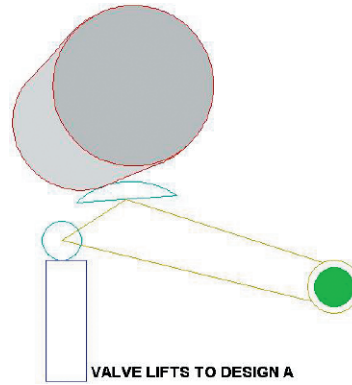


Fig.12 Finger follower cam for valve lift Design A.



micron, the locations of the thinnest film point(s) are different in the valve lift period. At about 0.6 micron the minimum oil film thickness, with the oil declared as SAE30 grade and at 80 deg C, seems safe enough but is very much a function of the viscosity of the oil and its temperature at the cam and tappet interface. This is illustrated by Figs.9 and 10 for the valve lift profile Design A with the bucket tappet mechanism.

Fig.9 shows the variation of the oil film thickness with

“The Hertz stress levels are principally related to the sharpness of the profile of the cam, i.e., the radius of curvature of its shape”

between cam and tappet is primarily a function of the radius of curvature of each of the three cams if the forces are somewhat similar. The bucket tappet is flat, the maximum force at this camshaft speed is at the nose, and so the maximum Hertz stress occurs at the nose of that cam with the smallest radius of curvature, which is the one for Design E.

The graphs of oil film thickness are presented in Fig.8. While the minimum values are very similar at about 0.6

temperature and the same SAE30 oil and Fig.10 shows the variation at a fixed oil temperature of 100 deg C when the oils selected for the computation are SAE20, SAE30, SAE40 and SAE50. In Fig.9, with an SAE30 oil, a rise of just 20 deg C in oil temperature halves the oil film thickness. In Fig.10, at a common temperature of 100 deg C, an SAE20 oil gives an oil film that is only half as thick as that given when using an SAE50 oil.

Fig.13 Rocker follower cam for valve lift Design A.

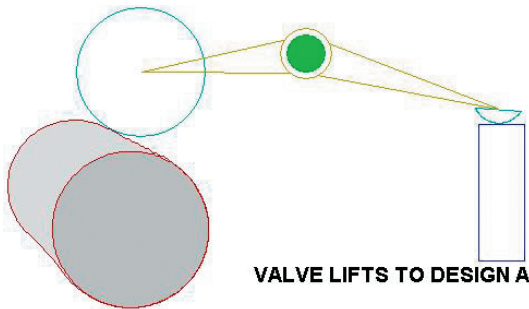


Fig.14 Pushrod follower cams for valve lift Design A.

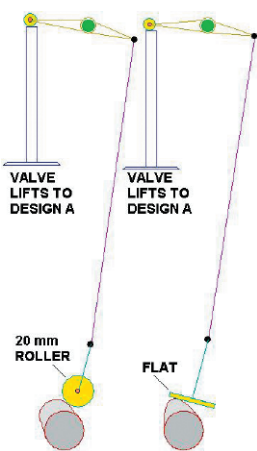


Fig.15 Roller tappet with pushrod: cam profiles for Designs A-E.

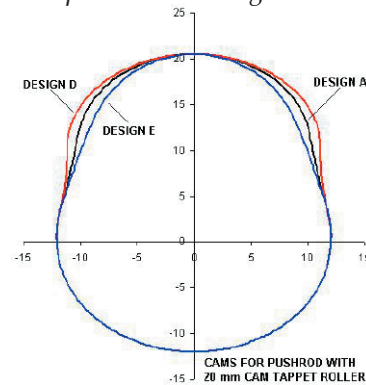
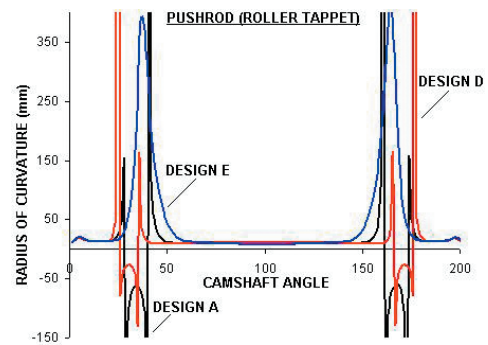


Fig.16 Roller tappet with pushrod: cam curvature for Designs A-E.



DESIGN OF FINGER AND ROCKER MECHANISMS

A similar design procedure for other follower mechanisms can be pursued within the software. The mechanism geometry is acquired for the finger or the rocker using similar data input procedures as in Fig.2 but as shown specifically in Fig.11. Similar physical data as described above for the bucket tappet is employed and the cam profile generated.

Snapshots in Figs.12 and 13 from the on-screen movie output when the valve lift profile is Design A shows the cam profile and the mechanism geometry for examples of finger and rocker followers, respectively. We will not dwell further on these designs at this point except to say that (a) it is vital that the cam and valve tappet followers can be treated by the design system as either sliding pads or rolling element bearings because the oil film thickness profile is different for rolling or sliding motion and (b) output data from the design of these two followers will be discussed when the analysis of a pushrod follower mechanism is discussed below.

DESIGN OF A PUSHROD MECHANISM

Using mechanism geometry to the format shown in Fig.2 and with valve mass data, valve spring stiffness data, and material physical properties as specified for the bucket tappet but with further realistic data added for the pushrod and the cam

tappet, and employing the same three valve lift profiles Designs A-E, the cam profiles are computed for the two common types of cam tappet. The first cam tappet example is flat and the second example is a 20 mm diameter roller. A snapshot of each example, taken from the on-screen movies when the valve lift profile is Design A, is drawn to scale and shown in Fig.14. The two cam tappets provide very different shapes for the cams which drive them.

The cam profile with the roller follower is concave, i.e., it is 'hollow-flanked'. This is more easily observed in the graphs for the cam profile in Fig.15 and the radii of curvature graphed in Fig.16. In Fig.15 the cam profile for design E is visibly convex, unlike that for Designs A and D. This is backed up by Fig.16 where the graph for Design E (blue) is always positive whereas the minimum negative curvature for Design A is some 60 mm and that for Design D is about 30 mm.

Quite irrespective of the view (offered in Part One of this article) that Designs A and D are not really suitable for use with a pushrod mechanism (you must await Part Three on valvetrain dynamics to be convinced of the accuracy of that opinion), cam profiles with negative radii of curvature have serious implications for the practicality, or otherwise, of grinding them. For the three profiles seen in Figs.15 and 16, the cam grinder would need a 120 mm diameter grinding wheel to make the cam for Design A and a 60 mm diameter wheel to produce that for Design D; he is most unlikely to accede to your request to have them ▶

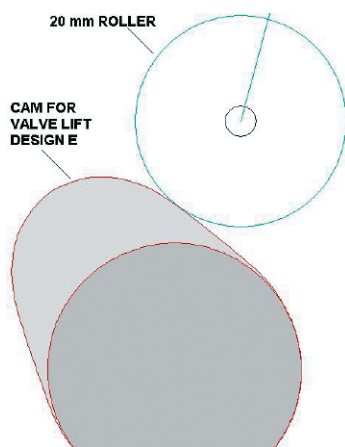


Fig.17 Roller tappet with pushrod: cam close-up for Design E.

Fig.19 Flat tappet with pushrod: cam profiles for Designs A-E.

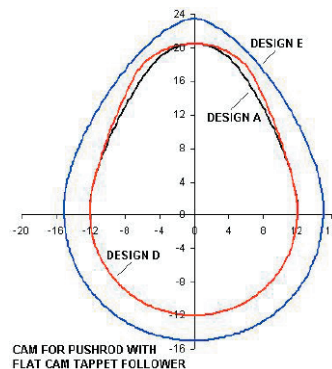


Fig.18 Roller tappet with pushrod: oil films for Designs A-E.

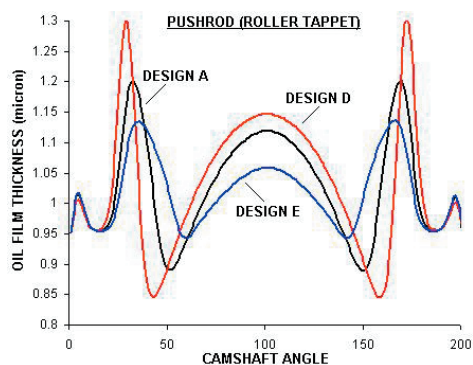
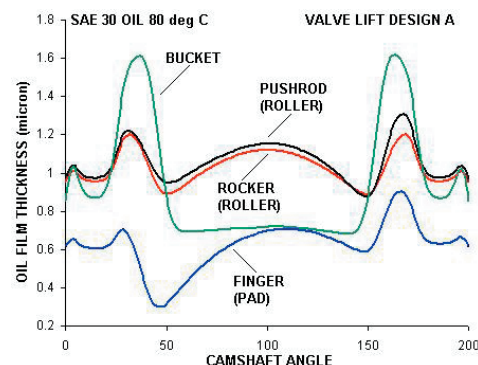


Fig.20 Oil film thickness for all followers for Designs A-E.



manufactured as the smallest grinding wheel he possesses is most likely about 200 mm diameter! The cam profile for Design E presents no problem for the cam grinder. It is convex, which shape is emphasised in Fig.17, a zoom-in snapshot from the on-screen movie of its rotation and that of the pushrod mechanism. In Fig.18, it can be seen that the Design E provides the most consistently thick oil film of the three valve lift design cases.

When the cam tappet on the pushrod mechanism is flat, see Fig.14, all cam profiles are convex, as was the example of the flat bucket tappet. The actual cam profiles for valve lift Designs A-E are shown in Fig.19 and the problem already found with Design E, with the bucket tappet, reappears. The base circle radius must be increased to 15 mm from 12 mm otherwise the nose of the cam for Design E becomes too sharp which would raise the local Hertz stress beyond any usable limit.

For those involved in NASCAR racing with their pushrod mechanisms, the need for large base circle radii on the cams driving the flat cam tappets in Nextel Cup cars and the relative simplicity of creating cams to work with roller cam tappets in the Busch series, this discussion should strike something of a responsive chord.

MORE ON LUBRICATION OF CAMS FOR ALL MECHANISMS

For all of the cam mechanisms discussed above, bucket, finger, rocker and pushrod (with roller tappet) the oil film thickness profiles are plotted in Fig.20. For each follower mechanism the common

input data for the oil is specified as SAE30 grade at 80 deg C and the valve lift profile is Design A. The bucket tappet (as in Fig.3) and the finger (as in Fig.12) operate with sliding friction whereas the pushrod (as in Fig.17) and the rocker (as in Fig.13) experience rolling friction around the cam. The oil film thickness for all but the finger follower can be considered as quite safe even for a simple ground, but not polished, cam, but the finger follower has an oil film thickness profile that requires comment.

It is quite common for a finger follower to have a curved pad as a cam tappet whereas if a roller follower is used instead it makes the finger follower heavier. On the other hand, the oil film would be thicker because it is then rolling friction. The reason for the dip in oil film thickness, on the opening lift flank of the cam with the finger follower in Fig.20, is because the tappet and the cam surface are momentarily travelling in the same direction and, as the oil entrainment velocity is reduced at that point, so is the oil film thickness.

Clearly, this is a feature of cam profile design for finger followers that must be executed most carefully and for those cam design packages that cannot properly compute the oil film thickness their prognostications should be treated with great suspicion. Strictly speaking, that comment should be extended to cover cam profile design for all cam follower mechanisms. The mathematics and fluid mechanics of lubrication is demonstrably not a simple technology, as an examination of the ‘tribology bible’ will clearly demonstrate [5].

Fig.21 Grinding the cam for the bucket tappet for Design A.

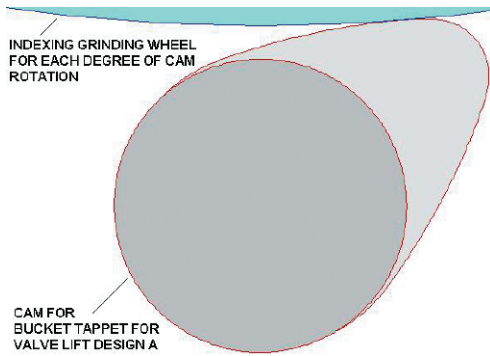


Fig.23 Snapshot of the cam profile check process for Design A.

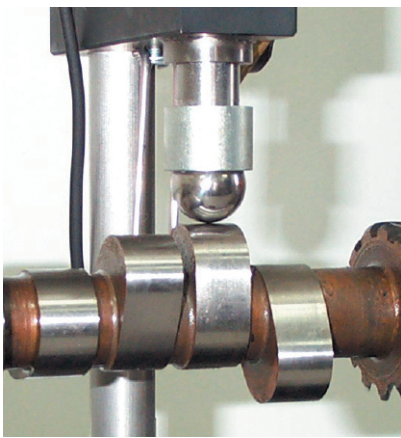
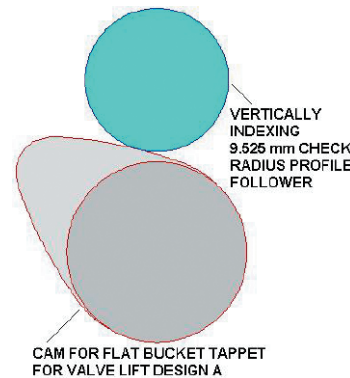
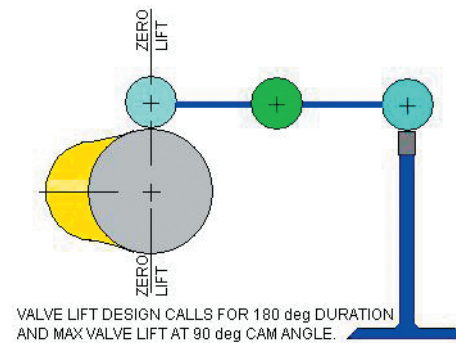


Fig.22 A cam profile measuring apparatus.

Fig.24 Definition of cam nose twist angle; at start of valve lift.



GRINDING THE CAM PROFILE

When the design of a cam profile is complete, it is obvious that the output data from the design process must allow the grinding process for that cam to be accurately executed. The 4stHEAD software automatically outputs the cam profile data in many standard manufacturing formats, in both metric and imperial units, for production grinding machines such as those made by Landis-Lund, Toyoda, etc. It also provides manufacturing output data for the cam for an in-house CNC grinding process where the user can insert the actual grinding wheel diameter and can

such observations. The Fig.21 shows a snapshot from the on-screen movie of the cam being ground for the bucket tappet of Fig.3 in order to create the valve lift profile Design A.

CHECKING AND MEASURING THE CAM PROFILE

After a cam profile is ground, either in-house or by an external cam grinding establishment, it is clearly important that this manufactured cam profile is precisely that which was designed. While one could take cam grinding accuracy on trust, not checking a manufactured cam profile is hardly a

“One can scrutinise an on-screen movie to ensure that there is perfect contact between the grinding wheel and the cam profile”

then store the motion of, and the coordinates of, the grind wheel centre. Further, and just as importantly, one can scrutinise an on-screen movie of that cam being ground to ensure that there is perfect contact at all times between the motion of the specified grinding wheel and the designed cam profile. Many an expensive mistake has been eliminated by

procedure to find favour under ISO9000. Cam profile data can be created by the software and checked by a measuring machine such as the one (a Cam Pro Plus device) shown in Fig.22. The check device on the measuring head can be a spherical ball or a flat component, although the ball is probably the more accurate of the two. ▶

Fig.25 Definition of cam nose twist angle; at maximum valve lift.

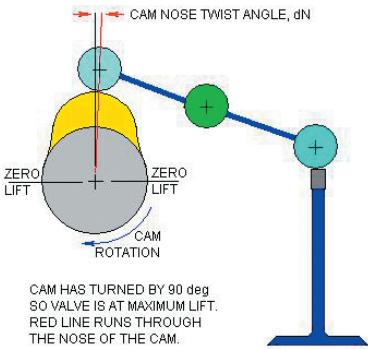
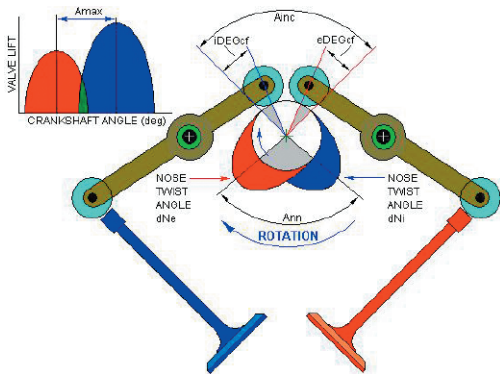


Fig.26 The disposition of cam noses on a camshaft: the sohca case.



The 4stHEAD software, with the check ball radius inserted as input data, a zero is used if a flat check follower, mathematically rotates the designed cam and not only outputs the check follower lift at each camshaft angle of rotation but shows a ‘movie’ of the process as seen in Fig.23. The movie snapshot is for the cam for the bucket tappet (of Fig.3) when the valve lift profile is Design A and the check ball has a 9.525 mm radius (0.75 in diameter). Needless to add, in this case the profile check lift data is different from the valve lift data as the former uses a spherical ball while the cam is being indexed around but the latter moves a flat-surfaced tappet.

After the actual cam is manufactured for Design A with a bucket tappet, that camshaft can be mounted into a measuring machine such as in Fig.22, a 9.525 mm radius ball is then inserted into the measuring head, and the cam profile data is experimentally measured. The as-designed and as-machined cam profiles can then be numerically compared and the cam grinder congratulated or admonished as appropriate.

GRINDING THE CAMSHAFT

Within the cam profile design process the 4stHEAD software also provides precision information regarding the cam nose twist angle, dN. For flat cam tappets, and direct acting and pushrod followers with no tappet offset, dimension Tx in Fig.2, the value of cam nose twist angle is zero. For most finger and

Fig.27 Torque requirements to drive a cam for valve lift Design A.

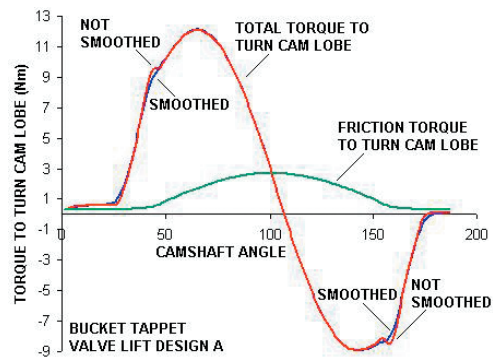
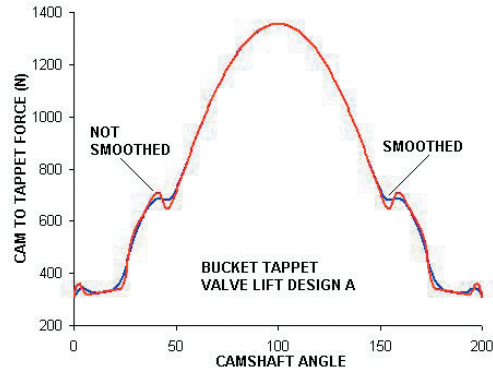


Fig.28 Cam to tappet forces with and without valve lift smoothing.



rocker follower mechanisms it is not zero and its numerical significance for the correct orientation of a series of cams along a camshaft when grinding it, is critical.

However, we should first describe and define ‘cam nose twist angle’.

Consider the very simple rocker mechanism shown in Fig.24 where the cam is just about to lift the cam tappet roller on a valve lift profile that has a 180 degree duration with maximum lift at 90 degree of cam rotation from zero lift. The centre of the cam tappet roller lies directly above the camshaft centre and the valve is vertical. In Fig.25 the cam has turned 90 degrees and so the valve is at maximum lift but the nose of the cam is no longer positioned directly above the camshaft centre. It lies off the vertical at a value defined as the cam nose twist angle, dN, because the centre of the cam tappet roller has moved to the right as the rocker oscillates and the valve lifts. However, the cam nose must lie on a direct line between the cam tappet centre and the camshaft centre, as the tangents to both the nose and the tappet must be at right angles to this line. The computation of this simple example provides the entire cam profile but it will be a profile which on the opening flank is 89 degrees from zero lift to the cam nose, and 91 degrees on the closing flank from the cam nose to zero lift if dN is, say, +1.0. If this cam is ground to this profile and installed in an engine with this same rocker mechanism, maximum valve lift will be at 90 degrees after the commencement of valve lift.

The person grinding your camshaft knows nothing of this mathematical and geometrical scenario when grinding a series of cam lobes along a camshaft. All they have is a shaft of metal and the information the designer gives them on the grinding profile of each lobe along it and the angular disposition of each nose of each cam lobe with respect to a fixed marker point at the end of the camshaft. As cam nose twist angles can be as high as 10 degrees, particularly for finger followers, it is obvious that there could be a considerable phase error caused by ignoring the cam nose twist angle data when arranging the nose-to-nose disposition of cams along a camshaft.

The situation in practice is even more complex than that, as illustrated by the sketch in Fig.26 of a single-overhead-camshaft rocker mechanism for the exhaust and intake valves. The engine designer will create valve lift profiles for both valves and, to optimise engine performance, will want to place them in the crank-angle diagram so that the maximum valve lift positions are A_{max} degrees apart; this is $A_{max}/2$ degrees in terms of camshaft angle. The valves lie within the engine cylinder head at an included angle of A_{inc} degrees. The cam tappet followers for the exhaust and intake valves lie at cam tappet follower angles, $eDEG_{cf}$ and $iDEG_{cf}$, with respect to the line of their valve motion. The cam nose twist angles predicted by the software for the exhaust and intake cams are dNe and dNi , respectively. The software design system will predict the manufacturing cam profiles for each cam lobe in terms of its mechanism geometry. What the designer must know, and which vital piece of information he must transmit to the cam grinder, is the nose-to-nose angle between the cams, Ann . Without calculating that for its succession of exhaust and intake cam lobes, the camshaft cannot be manufactured. The 4stHEAD software provides the designer with the information to directly calculate the cam nose-nose angle, Ann . We will not belabour the point but the numerical value of Ann is different, in terms of the symbolism of Fig.26, if the camshaft rotation is anti-clockwise.

TORQUE REQUIRED TO TURN A CAM LOBE

The torque required to turn a cam lobe is the addition of that torque to overcome the spring and inertia forces and that of the friction between the cam and tappet. On the simplistic assumption, which will be corrected in Part Three of this article, that the cam and tappet stay in continuous contact with each other, the valve spring forces will provide assistance in turning the cam after the point of maximum valve lift. The friction force always opposes motion and always requires torque to keep turning the cam. The friction force is a function of the cam-tappet follower force, the area of the contact patch between them and the viscous force normal to this direction due to the local prevailing viscosity of the oil film. The mean torque to turn the cam lobe is the cyclic average of the

instantaneous value of the total torque signal. The power to turn the cam lobe is then a function of the camshaft speed.

Using the same computation data for the bucket tappet seen in Figs.3-8 and using valve lift Design A, Fig.27 shows the computation by the 4stHEAD software of the total torque to turn the cam lobe at 3000 rpm (cam) and also the friction torque. The normal valve lift profile Design A, which is inherently smoothed by the 4stHEAD software and about which there was much debate in Part One of this article regarding poor quality smoothing, is drawn in 'blue'. The results when the 'lift profile smoothing' is deliberately removed from valve lift Design A are also graphed in Fig.27, in 'red'. The ripple on the torque required to turn the cam lobe is very visible.

This ripple is equally visible when the cam-tappet forces from this same computation are graphed in Fig.28 with the same colour coding. From Part One of this article you will recall that jerk, or cam-to-tappet 'chatter', will be the derivative of these curves. The 'chatter' from the non-smoothed curve in 'red' will be very much higher than that for the normal Design A, when the standard 4stHEAD smoothing technique is applied, in 'blue'.

CONCLUSIONS

Cam profile design and cam design for manufacture is a specialist topic. With the advent of mathematically complex but accurate, user-friendly and highly visual computer software it, as with valve lift profile design, can be professionally executed by the engineer who normally designs the power-producing cylinder-head components of the engine. However excellent those design techniques may be for cam design and manufacture, the quality of the ensuing cam profile design and its subsequent behaviour within an engine is only as good as that of the valve lift profile design, and the quality of its inherent smoothing techniques, which precede it.

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