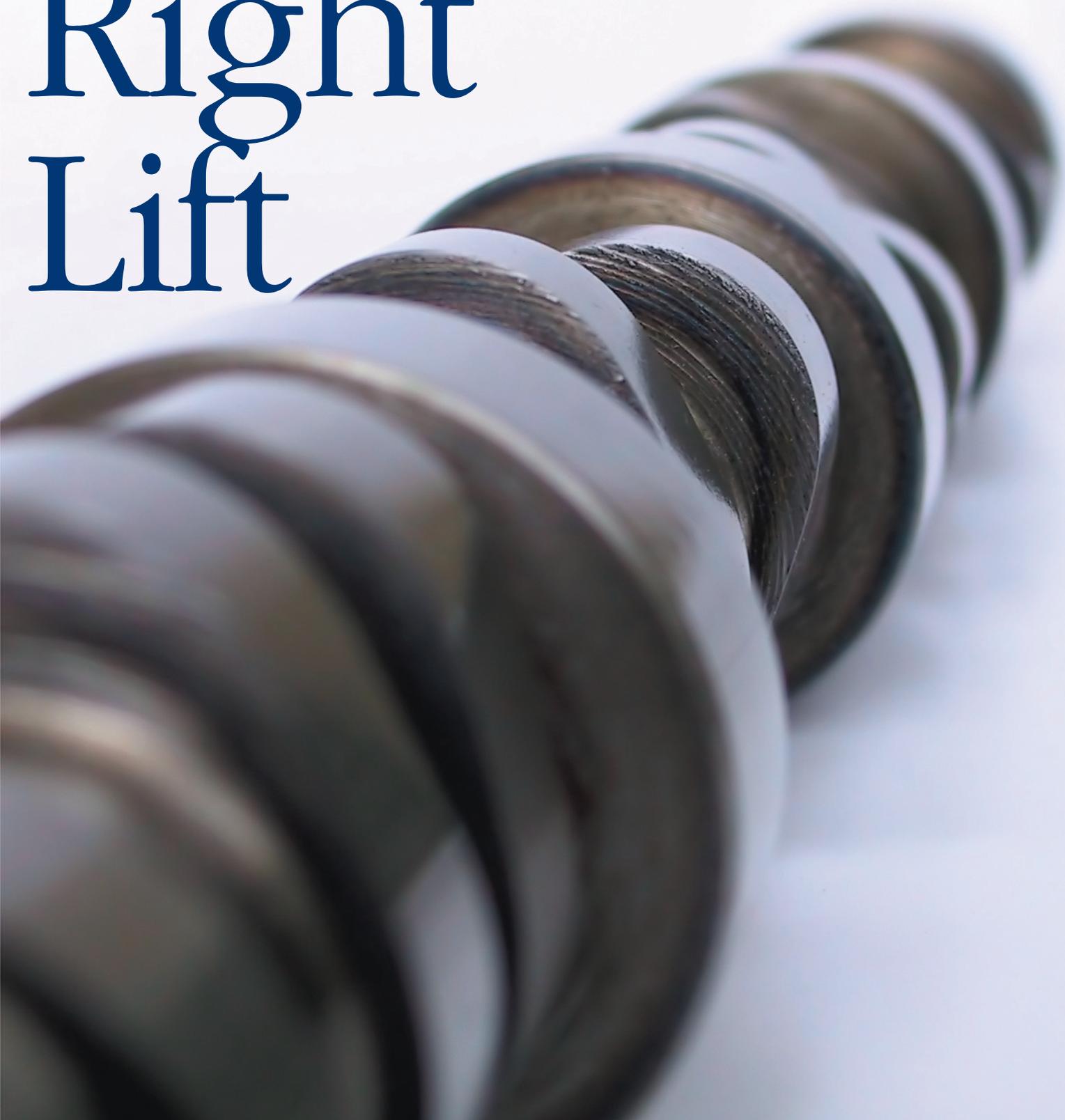


# The Right Lift



This is an insight into valve lift profile design and it constitutes the first segment of a three-installment Special Investigation into the subject of valvetrain design. Our guides throughout are Prof. Gordon Blair, CBE, FEng of Prof. Blair & Associates, Charles D. McCartan, MEng, PhD of OPTIMUM Power Technology and Hans Hermann of Hans Hermann Engineering. The aim of the work is to shed light onto a subject that is central to the performance of the racing engine yet which is not properly understood by all who have responsibility for racing engine design and development.

## THE FUNDAMENTALS

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."

This bald statement sets the tone for all that a client designer executes within the entire software suite known as 4stHEAD created by Prof. Blair & Associates [1].

The fundamental point being made is that the user is designing a four-stroke cycle engine to inhale air and exhale exhaust gas and this function is carried out by poppet valves whose lift and duration at the design speed are the real objective of the design exercise if the engine is to provide the required performance characteristics of power and torque. In short, one firstly designs the valves and valve lift to satisfy an engine breathing requirement and then secondly the entire cam mechanism to do that job successfully at the design speed.

At this point the reader is probably sagely nodding their head at such logic. This reader would be equally surprised to know that the world is apparently still full of 'cam design gurus' who

carry out this procedure in the reverse order, often re-shaving and re-shaping cam profiles with scant regard for the non-linear geometry of the mechanism involved nor the ensuing potential havoc to be wreaked on the valvetrain under dynamic conditions at speed.

There will be three Parts to this paper on valvetrain design: Part One on 'valve lift profile design', Part Two on 'cam design for manufacture' and, Part Three on 'valvetrain dynamics'. Although we have written 'firstly' and 'secondly' above as if these design functions are conducted in a linear fashion, the reality is that it is an iterative to-and-fro process between these three 'Parts' as the inevitable engineering design compromises must be made to overcome the design problems uncovered in each segment.

## CREATING A VALVE LIFT PROFILE

The creation of the requisite valve lift, duration and size characteristics for an engine has been written about extensively, of which references [2-5] are but a sampler, and the use of an accurate engine simulation [6] for that procedure is recommended as the only logical final design route. So, this article is written on the assumption that the designer knows what valve lift and duration is required but must now create that valve lift profile and cam mechanism to mechanically achieve it. It is not quite as simple as that, as hinted at above and as later evidence in Parts 2 and 3 will demonstrate, but we will proceed on that basis for the present. All of the designs and graphs in this discussion are produced by the 4stHEAD software [1] with many of them reproduced directly from its on-screen graphics.

In Fig.1 is a series of valve lift designs, A-E. They show five differing designs of lift profile with identical total lift and total durations of 200 degrees rotation of the camshaft or 400 degrees rotation of the crankshaft. Within this paper, all angles subsequently reported will be in degrees camshaft.

In Fig.1, it can be seen that the first, and last, part of the lift is very shallow. This is known as the 'ramp', which is designed to take up the valve tappet clearance, the so-called 'valve lash'. In this case it is 25 degrees long and rises to 0.3 mm. The valve lash, when the engine is running hot, will be set to about 0.2 or 0.25 mm, which will translate to an equivalent 'cold' valve lash setting which must be determined as a function of the relative expansion characteristics of the cylinder head and the cam follower mechanism.

If the engine uses hydraulic tappets the 'lash clearance' is effectively zero and so the ramps would normally be reduced in both amplitude and duration. The objective is to take up the valve tappet clearance smoothly and progressively and, as will be seen in Fig.3, the normal racing engine practice is to do so at 'constant velocity'.

There are also more aggressive 'acceleration' and 'cosine' ►

types of ramp design, often found in diesel engine practice, which can be alternatively indexed from within the software. All of the designs A-E have a lift of 10 mm above this 0.3 mm ramp and, with 25 degrees allocated to the opening and closing ramps, the actual valve lift duration is nominally 150 degrees.

It can be seen that the designs A (black line), B (red line) and C (blue line) are somewhat similar in valve lift profile but that of design D (cyan line) is more aggressive and design E (green line) is less aggressive in profile. To judge the ‘aggression’, or otherwise, of a valve lift profile a lift-duration envelope ratio, Kld, is defined as shown in Fig.2. It is the area under the valve lift curve divided by that of the rectangle in which it sits and, while it can be deduced at any valve lift,

Those of you with a mathematics bent will realise that the numerical differentiation of the valve lift-degree profile will produce a velocity-degree curve; the differentiation of the velocity-degree curve gives an acceleration-degree profile; and the differentiation of the acceleration-degree curve gives the jerk-degree characteristics. (Just as the velocity of a car is the rate of change of a distance-time curve and acceleration is the rate of change of velocity).

It will also be remembered that, courtesy of Isaac Newton, acceleration is directly related to force and hence jerk, which is the rate of change of force, is an ‘impulse’ or ‘hammer blow’.

The velocity, acceleration and jerk characteristics for designs A-E are shown in Figs.3-5, respectively. The lift-duration

“What we must pay for the aggressive nature of design D now becomes clear. How we obtain design D also becomes obvious”

normally it is determined for the valve opening duration which in this case would be at a lift of 0.3 mm or over the 150 degrees of actual valve opening.

The characteristics of each of the designs A-E are summarised in the following table. The profile is symmetrical about maximum lift and the periods (in degrees) of positive acceleration (PACC), transition acceleration (TACC) and negative acceleration (NACC) are shown in the table, making up 75 degrees in all cases. There is also a negative acceleration exponent Z (NAEXPZ) to be discussed but the ensuing lift-duration envelope ratio, Kld, for each of the designs shown in the table can be seen to describe perfectly the ‘aggressive’ nature of the several valve lift profiles shown in Fig.1.

Designs A-C are within 2% of each other but design D is some 10% higher than A-C, while design E is some 20% less aggressive than design D. See also Fig.6 for this Kld data drawn as a bar chart. One could simplistically argue that design D might just breathe 20% more air as an intake valve, or pump 20% more exhaust gas as an exhaust valve, by comparison with design E.

DESIGN	PACC	TACC	NACC	NAEXPZ	Kld
A	20	10	45	0.66	0.5435
B	20	10	45	0.30	0.5305
C	20	10	45	0.90	0.5507
D	10	10	55	0.90	0.5981
E	30	10	35	0.30	0.4849
F	2	2	64	0.01	0.596

envelope ratios, Kld, for designs A-E are compared in Fig.6. What we must pay for the aggressive nature of design D over the others now becomes clear in Figs.3-5. How we obtain design D, by comparison with design E, also becomes obvious, particularly from the acceleration diagrams in Fig.4 and from the numbers in the table above.

It can be seen in Fig.4, and from the table above, that the totality of the transition and negative acceleration periods for designs A-C is 55 degrees, so the portion of the acceleration lying below the zero line is nominally 110 degrees. The difference in profile is controlled by the negative acceleration exponent, Z, which is 0.66 for A, 0.3 for B and 0.9 for C. Design C is marginally the most aggressive lift profile, so the higher is the exponent Z then the flatter is the majority of the negative acceleration period and the more aggressive is the lift profile.

The converse of that statement can be seen in design B.

However, to produce the most aggressive design D, it is necessary to reduce the ratio of the positive to the negative acceleration periods, and vice-versa to create the least aggressive design E. You may well ask why one would not always try to create design D as this must be a much better valve lift profile to fill a cylinder with air or empty it of exhaust gas. The answer lies in the mass of the cam and follower mechanism to move this valve at the engine speed at which it is doing it.

The acceleration in Fig.4, and hence the force, to move design D is nearly three times that of design E and nearly

Fig.1 Valve lift characteristics for designs A-E.

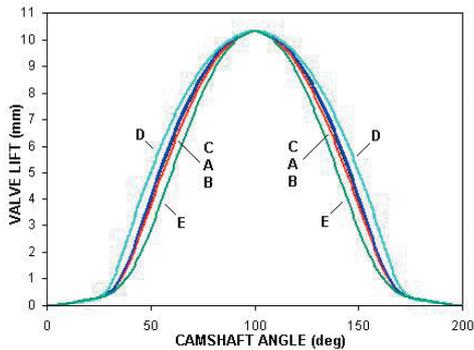
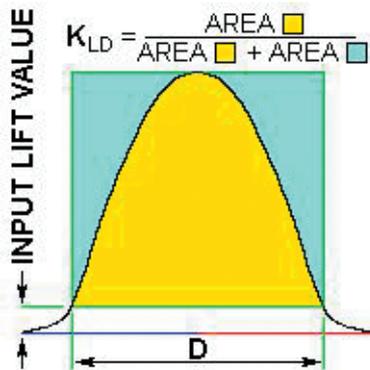


Fig.2 The lift-duration envelope ratio,  $K_{LD}$ .



twice that of designs A-C. The jerk in Fig.5, and hence the impact between cam and cam follower, to move design D is some four times that of design E and about twice that of designs A-C. Depending on the cam follower mechanism involved, the mechanical constraints involved in using design D may not be sustainable under dynamic operation.

If the cam follower mechanism is relatively light and stiff, such as a finger follower or a direct-acting bucket tappet, then it is possible to use a valve lift profile such as design D. If the system is a single ohc rocker system with a potentially higher inertia, then one of the designs A-C will almost certainly be more appropriate. If the cam follower mechanism is a pushrod ohv rocker device then design E is almost certainly the type of profile that will successfully permit the movement of the greater masses, and more flexible components, involved.

The design goal is to use a valve lift profile that is as aggressive as possible so as to achieve superior engine breathing characteristics, but it must be done within the mechanical limitations of the engine design. For example, the use of a lift profile such as design D in a two-valve NASCAR Cup pushrod engine at 9500 rpm is not a realistic option.

The comment passed earlier, regarding the use of a 'constant velocity' ramp lift to 0.3 mm for all designs A-E, can be seen to be borne out in Fig.3 and the acceleration curve 'blip' profile required to attain it can be seen in Fig.4 and, in a similar fashion, later in Figs.12-13.

Fig.3 Valve velocity characteristics for designs A-E.

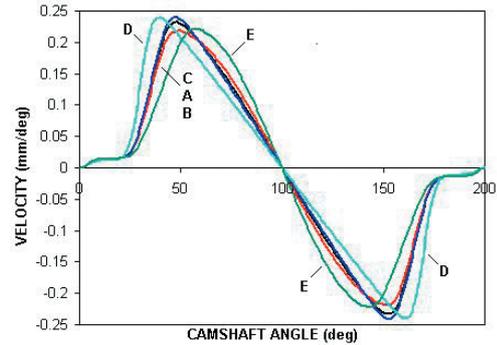
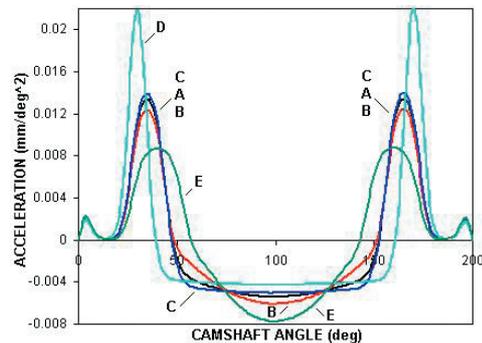


Fig.4 Valve acceleration characteristics for designs A-E.



### SMOOTHING IS VITAL IN VALVE LIFT PROFILE DESIGN

One of the three valve lift profile design methods within the 4stHEAD software suite is called the HMB method. The initials stand to denote that it is the Hermann, McCartan and Blair technique. It uses up to 11th order polynomial functions to connect the various phases of the lift and the acceleration diagram. It is a most powerful method as one inserts the demand valve lift and cam angle periods into the software and one then manipulates the segment and exponent data, as in the above table, until the designed profile is that which is required, such as a design E rather than a design A.

The alternative approach, illustrated later in this paper and very visible on the web [1], is the GPB method where one manipulates the acceleration diagram directly by dragging it on-screen with the computer mouse into any desired shape at the whim of the designer, which shape is then automatically integrated into velocity and lift diagrams and differentiated to the jerk diagram. You may readily imagine that this GPB technique is a most popular method with the 'cam design gurus' of this world who have cam profile whims aplenty.

Whichever method is employed, it is vital that the various segments of the lift and/or acceleration diagrams are connected up smoothly otherwise the ensuing forces and impacts on the cam follower mechanism will be considerable. In other words, a good mathematical smoothing technique ►

Fig.5 Valve jerk characteristics for designs A-E.

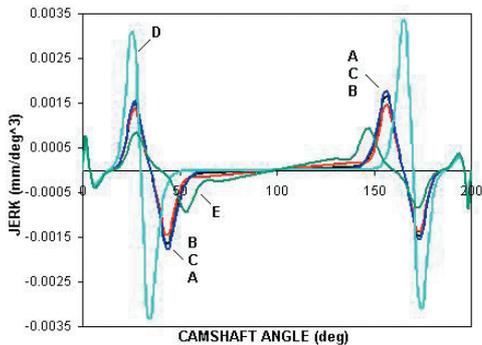
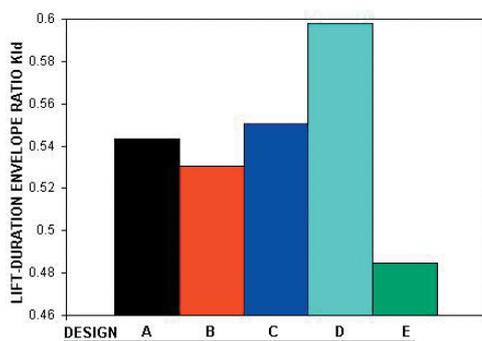


Fig.6 Lift-duration envelope ratios for designs A-E.



within the valve lift design process is absolutely essential. This is the fundamental difference between the best ‘valve lift design’ methods and the ‘also-rans’. Always remember that if the valve lift profile design method, and its smoothing technology, is not of the highest quality then neither can be the ensuing ‘cam design’.

To provide an example of the effects of not applying a good mathematical smoothing process, the design A described above is repeated within the HMB design program in the 4stHEAD software, but with the smoothing process deliberately bypassed. The results for valve lift, velocity, acceleration and jerk are shown in Figs. 7-10, respectively.

In Fig.7, the difference between the smoothed and the unsmoothed valve lift is barely visible and amounts to lift differences of no more than +/- 0.01 mm at several locations. The effect on the lift-duration envelope ratio, Kld, barely registers at the fourth decimal place. The engine, apart from the rapid wear rate and impact pits on the cam and cam follower interface, and the higher valvetrain noise, from a breathing standpoint could not tell the difference between the ‘smoothed’ or the ‘unsmoothed’ valve lift designs. This also assumes that no high-speed valvetrain dynamic bouncing effects interfere with this simple comparison.

In Fig.8, the unsmoothed valve velocity (in red) appears briefly, and barely, at the beginning and end of the opening and closing ramps and at the locations of maximum velocity.

Fig.7 Valve lift with and without smoothing for design A.

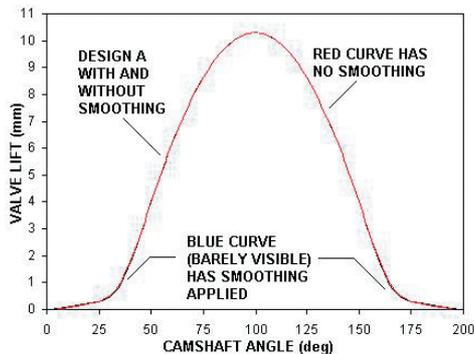
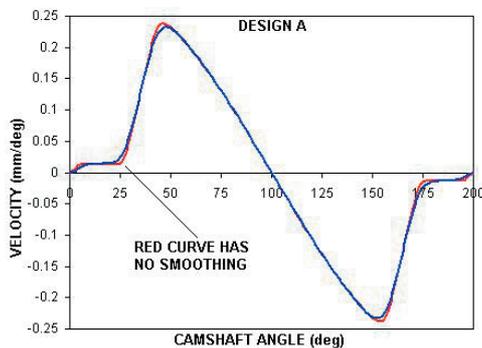


Fig.8 Valve velocity with and without smoothing for design A.



It is in Fig.9 that the problems begin to appear at these same ramp locations with a 40% higher acceleration at the very start of the ramp, where a hydraulic tappet would surely notice it. The maximum positive acceleration is also increased by 5%.

Remember always that acceleration translates to ‘force’ under dynamic conditions. However, it is Fig.10, the jerk curves, in which the problems with the unsmoothed profile really show up. Here, the jerk at the end of the ramps and at the beginning of the true valve lift, which are the normal locations of maximum jerk, are now increased by 90%. Remember also that jerk translates to an impulse and excessive impact ultimately leads to scuffed and pitted cam followers.

In racing, one might say ‘so what’ to damaged followers, so long as the valvetrain survives the race then all is well. Not so in all automotive fields, racing or otherwise; higher forces and impulses not only lead to long-term wear and tear but also to higher-than-normal valvetrain noise levels from the high-frequency chatter of the cam and cam follower.

Even in racing, viz Formula One with its new two-race engine rule, the DTM with its season-long engine rule, 24-hour sports car races such as are held annually at Le Mans, Daytona, Spa and the Nurburgring, and so forth, engines have to withstand long-term high stress levels and badly-smoothed valve lift profile designs are simply not good enough to meet such stringent requirements.

Fig.9 Valve acceleration with and without smoothing for design A.

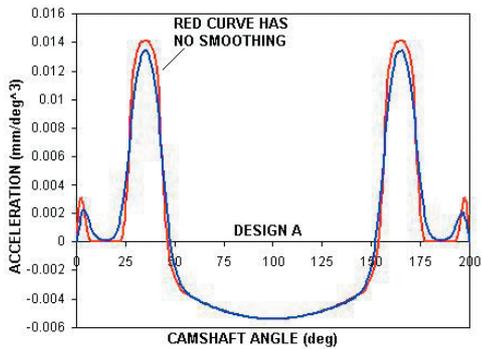
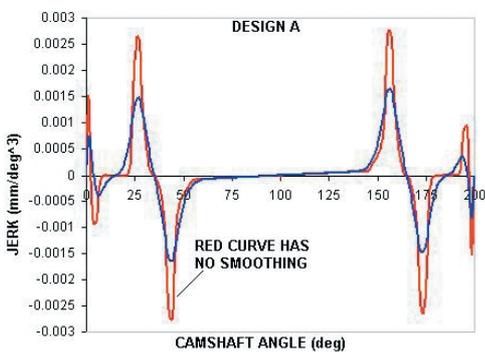


Fig.10 Valve jerk with and without smoothing for design A.



### SMOOTHING IS CRITICAL IN VALVE LIFT DESIGN FOR RACE ENGINES

One of the most difficult areas in a valve lift diagram, to create a smooth transition from one lift phase to another, is at the very beginning of the positive acceleration. An example of this problem is to be highlighted for a valve lift profile design for a 'touring car championship' engine as created by some very well-known specialists in cam design and manufacture.

One of the programs within the 4stHEAD suite permits the user to import a measured valve lift curve and mimic it closely with a user-created valve lift profile. The user may also import instead a measured cam profile, such as that acquired on a Cam Doctor or Cam Pro Plus machine or their equivalent, together with the geometry of the cam follower mechanism, and again mimic that with a user-created valve lift profile.

There are three versions of this particular program (see website [1]), in HMB, GPB or GPBv2 formats. The one being used here is the HMB version so as to be consistent with the previous discussion.

The measured valve lift curve from that 'touring car championship' engine is that labelled as design F in the table above, with extremely short positive acceleration and transition acceleration periods but a shallow 'low Z exponent' negative acceleration period. The lift above a 0.573 mm high ramp is 11.416 mm. The design is quite aggressive with a lift-

Fig.11 Matching measured valve lift for racing design F.

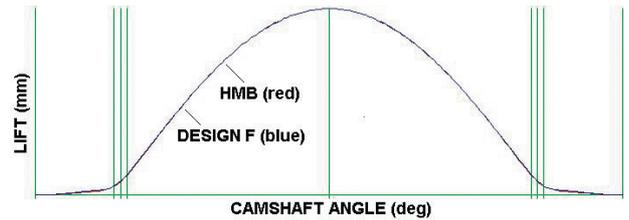
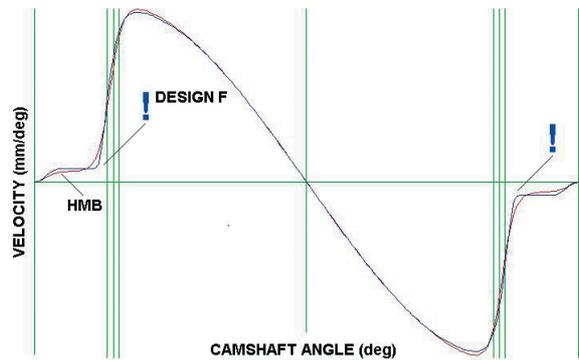


Fig.12 Matching measured valve velocity for racing design F.



“This technique constitutes quite a ‘detective’ exercise and yields much information on the designs”

duration envelope ratio, Kld, of 0.596.

The output from the HMB mimic program in Figs.11-14 are actual on-screen snapshots from this computer program and are shown for lift, velocity, acceleration and jerk, respectively. The blue line in each case is the measured data and the red line is that created by the user to mimic the measurement. As you can imagine, this technique constitutes quite a 'detective' exercise and yields much valuable information on the designs, design approach, software smoothing quality, and design techniques as



used by other 'cam design gurus', manufacturers, competitors, and other assorted experts and specialists.

In Fig.11, it can be seen that the measured valve lift and lift profile has been closely matched to better than 0.01 mm just about everywhere to the point where the (thin) red and blue lines are almost indistinguishable. In Fig.12, the velocity curve, the lack of smoothing at the very beginning of lift is beginning to show itself on the (blue) measured line.

In Fig.13, this same lack of smoothing at this critical point (marked by a !) results in a positive acceleration which is nearly 20% higher than is given by the HMB method for the very same valve lift at that juncture. This means that 20% extra force has to be available from the valve springs to contain the valve at this location and this implies that more power and torque is required to turn it. Note that the HMB method inserts a smooth transition from the horizontal to the vertical at this very point.

The upshot of this lack of smoothing is shown clearly on the jerk diagram in Fig.14 where it is now twice that which would be created by the HMB method and that, it must be re-emphasised, for precisely the same valve lift profile. To reiterate, jerk equates to impulse and to cam wear, pitting, and scuffing. Note also that, even at this third differentiation of the lift profile as a jerk curve, the HMB smoothing process is quite intact and continues to provide a smooth horizontal to vertical curve transition at this difficult (marked by a !) location. Even the secondary jerk peak, at the positive to the transition acceleration location, is some 20% less than that of the measured profile.

We have considerable sympathy with the famous cam design specialists who created design G for not being able to incorporate smoothing at this most difficult of all profile transitions because we had to work long and hard to incorporate that degree of smoothing into the 4stHEAD software. It is critical to do so, however, as this particular profile transition is normally the location of maximum jerk on all valve lift profiles for high-performance engines.

### SOME GUIDANCE FOR THE DESIGN ENGINEER

Pointers have already been issued in the discussion regarding designs A-F with respect to the creation of suitable valve lift profiles that could be applied to various types of cam follower mechanism. Design A, an atypical valve lift profile, follows the simplistic design rules for the actual opening period above the ramp where (a) the maximum positive acceleration is numerically some three times that of the (absolute) value of the maximum negative acceleration and, (b) the duration of positive acceleration is some 35-40% of that (negative) period of acceleration which is less than zero. More aggressive valve lift profiles are created by reducing the period of positive acceleration at the expense of increasing the period of negative acceleration and also by flattening the profile of that

Fig.13 Matching measured valve acceleration for racing design F.

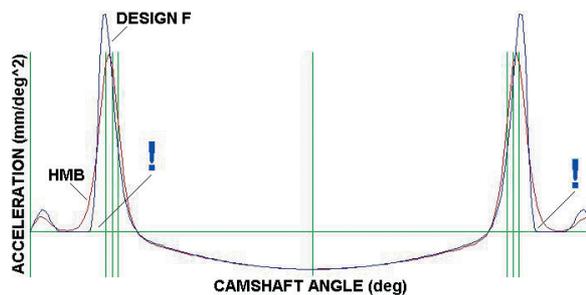
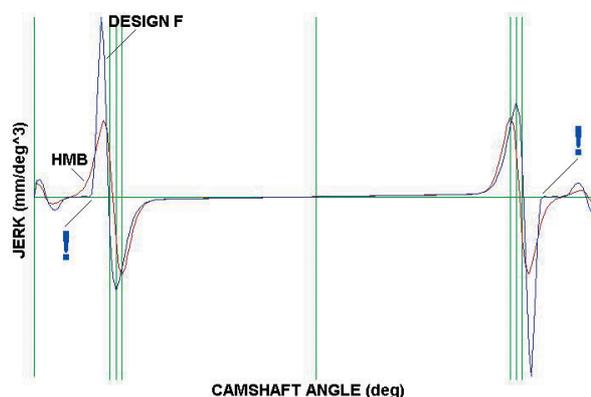


Fig.14 Matching measured valve jerk for racing design F.



negative acceleration. Less aggressive valve lift profiles apply this design philosophy in reverse.

Further design assistance from the 4stHEAD software comes in the form of an on-screen graphic of a Fourier analysis of the acceleration diagram. In Fig.15 is a graph of the outcome for Designs A, D, and E where the colour coding of the bars is as used previously. Each of the harmonics is shown up to the 28th, although the software presentation graphic actually does so up to the 36th harmonic. Design D, the more aggressive lift profile, clearly demonstrates that it has high-frequency components of a large amplitude which implies that it will shake the valves, springs and other components of its cam follower mechanism to the greatest extent. The harmonic components for design D do not significantly diminish in amplitude until about the 14th harmonic, but there is a significant dip at the 9th harmonic.

Let us assume for the sake of argument that the engine will spend much time at 6000 rpm, i.e., 3000 rpm at the camshaft, which translates to a frequency of actuation of the cam and follower mechanism of 50 Hz. The 9th harmonic is then 450 Hz and a wise designer would attempt to design his valve springs to have a natural frequency of 450 Hz so that the cam actuation system will have the least potential of sending the valve springs into a resonance mode.

On the other hand, design A has a minimum amplitude at the 7th harmonic so for design A the valve springs should be designed to have a natural frequency of 350 Hz if it is to be

Fig.15 Fourier analysis of acceleration for designs A, D and E.

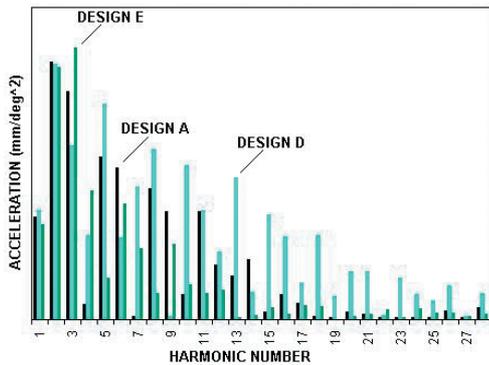
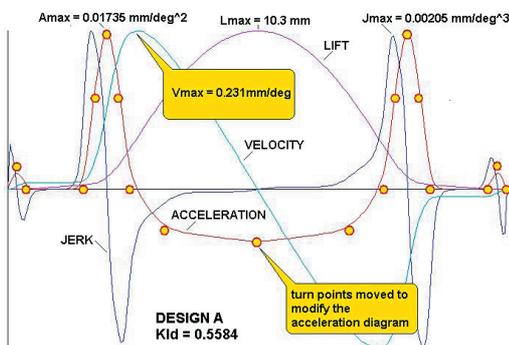


Fig.16 Valve lift profile design by the GPB method for design A.



used on the same argument engine. The same thinking applied to design E will use valve springs with a natural frequency of 400 Hz as it has a dip at the 8th harmonic.

Broadly speaking, the valve springs tend to have a natural frequency in the 350 to 600 Hz range, so one can see the cogency of this discussion, although in Part Three of this article the debate will be greatly extended. The old “rule of thumb” in this matter was to instruct the designer to employ a valve lift profile that had a low 7th harmonic in its Fourier

Fig.17 Valve lift profile design by the GPB method for design G.

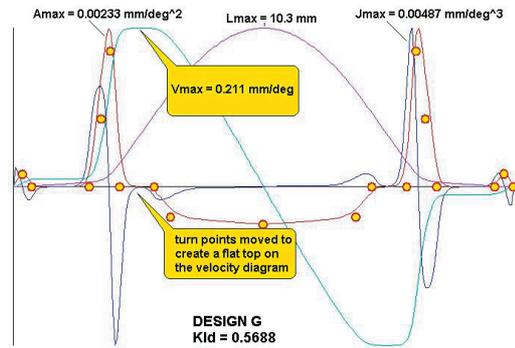
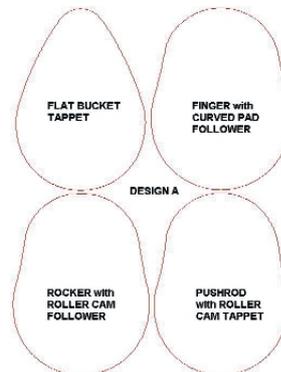


Fig.18 The bucket tappet, finger, rocker and pushrod follower cams to provide the design A valve lift profile.



cam which will no longer stay in full contact with the existing cam tappet on the engine. Make a larger tappet, you say! If it can be done, then all well and good but often such items are limited in size by the writers of race regulations in the interests of retaining close (fair?) competition or even in reducing or controlling maximum power output within a particular racing class. The retention of a cam on a tappet is principally controlled by the maximum velocity level on the valve lift profile, consequently the designer has to find a way around

“The retention of a cam on a tappet is principally controlled by the maximum velocity level on the valve lift profile”

acceleration spectrum. Like all “rules of thumb”, with Murphy and his Law forever active, they rarely apply to a particular case and here it would fortuitously work well for design A, but not for designs D and E!

Often the race engine designer finds that his creation of a new valve lift profile, usually one that is more aggressive and/or with more lift, finishes up with a design for the actual

such a predicament should it arise.

Let us illustrate the solution by using the GPB method of valve lift profile design. In Fig.16 is shown the on-screen software output for a valve lift profile that is a reasonably good mimic of design A as the durations are the same, the ramp lift is 0.3 mm and the maximum total lift is 10.3 mm. It shows the graphs of lift, velocity, acceleration and jerk and ►

on the actual output screen of the software there also appears the numeric values of all maxima and minima. They have been added here in larger lettering so that the reader may comprehend the points being made.

The actual lift-duration envelope ratio, Kld, is 0.5584 in this GPB method, cf 0.5435 reported earlier with the HMB method. Let us assume that the cam design has to move a bucket tappet to give this valve lift profile and that the 'cam manufacture' program (to be discussed in Part Two of this article) reports that the required bucket tappet diameter for full cam contact is 30 mm but the actual bucket tappet available is only 27.5 mm, or 8.3% less. The maximum valve velocity, currently shown on Fig.16 as 0.231 mm/deg must perforce be reduced by 8.3% down to 0.212 mm/deg. Naturally, the designer would like to retain all of the advantages of design A, but keeping the cam on the tappet must take the higher design priority. The solution is seen in Fig.17 for the re-think labelled as design G.

In the GPB method, one drags the turn points with the mouse

mechanisms to be described in Part Two, in Fig.18 are the (unscaled) shapes of the cams which give that same design A valve lift profile in each case when the cam follower mechanism is variously a bucket tappet, a finger follower, a sohc rocker follower, and a pushrod-rocker follower system. The difference in profile of each cam is obviously a function of the particular cam follower mechanism. The valve lift profile provided by each cam and cam mechanism is design A in each case. This is where 'cam design for manufacture' really starts.

## CONCLUSIONS

With accurate, comprehensive, and user-friendly software, valve lift profile design is no longer the exclusive province of the specialist cam designer but can be professionally executed by the engineer who normally designs the power-producing cylinder-head components of the engine. It is critical that such software employs the highest-quality smoothing techniques if

# “Valve lift profile design is no longer the exclusive province of the specialist”

to modify the acceleration diagram, and in Fig.17 it can be seen that two points are moved to create a 'flat' on both the acceleration and the velocity diagram with a cap created on the maximum velocity of just less than the target value of 0.212 mm/deg. It will be observed that the now-traditional 4stHEAD smoothing quality is unaffected by this process. The same maximum lift of 10.3 mm is retained and, magic even, the Kld value has been increased to 0.5688 or some 2% of an increase in profile aggression.

Nothing comes for nothing in the world of mechanical design, as the maximum acceleration has gone up from 0.001735 to 0.00233 mm/deg<sup>2</sup> and the maximum jerk has risen from 0.00205 to 0.00487 mm/deg<sup>3</sup>, which are increases of 34% and 237%, respectively. Further design work with the software is clearly required to reduce these penalties should they be deemed to be unacceptable. The main re-design task has been accomplished; the cam will now stay in full contact with the bucket tappet.

## DESIGN OF CAMS AND MECHANISMS TO PROVIDE THE VALVE LIFT PROFILE

In Part Two of this article, we deal with the design of the cam and cam mechanisms which provide a valve lift profile created in the manner discussed above. Here, to illustrate the point, consider the valve lift profile detailed above as design A. Without going into the fine geometric detail of the cam follower

these same power-producing components are not to be subjected to unnecessarily high stress levels nor will they require unnecessarily high power levels to drive them.

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