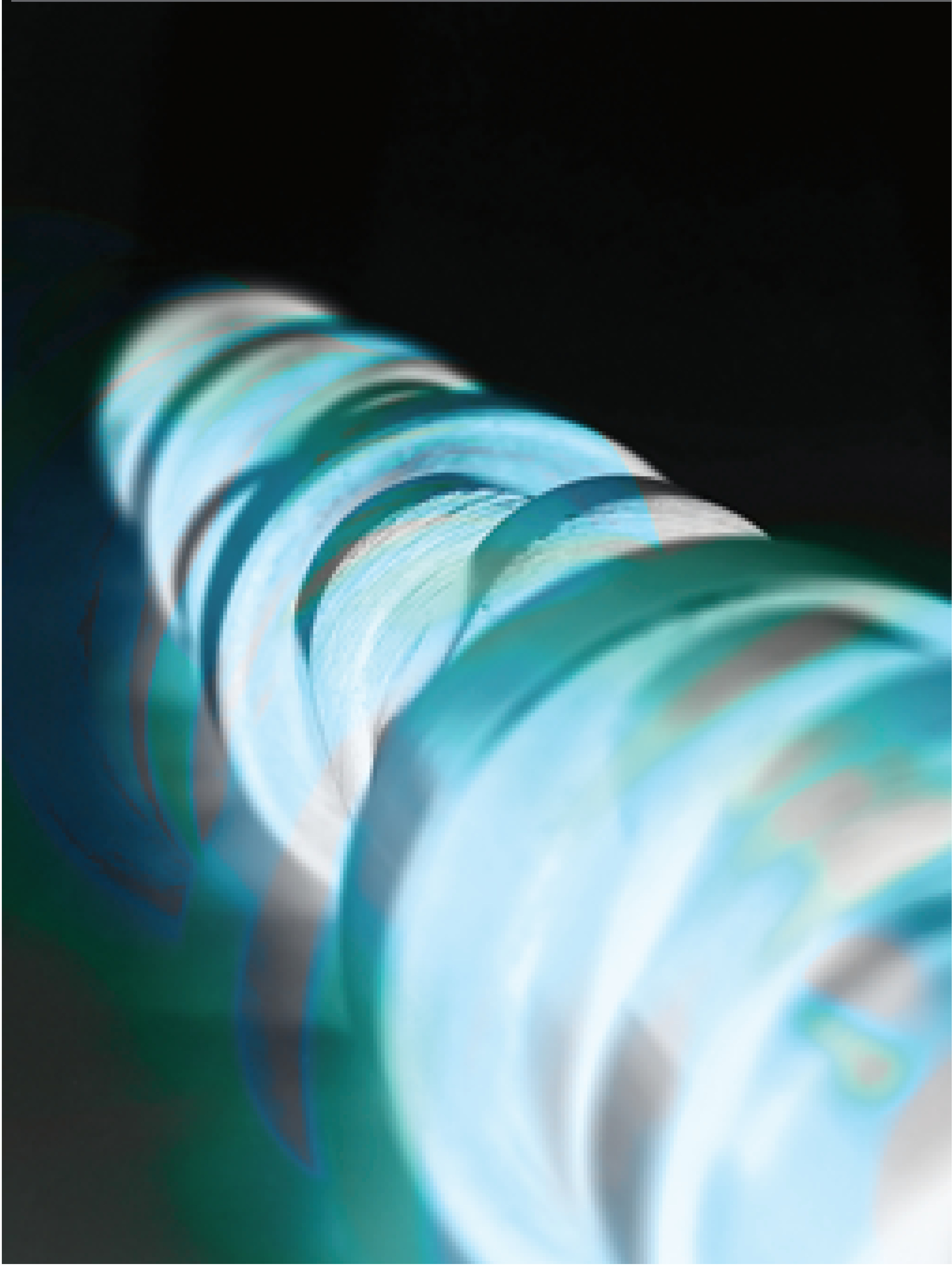


Pushrod Operation

This is ‘part B’ of the third section of our insight into valvetrain design. The third section looks at valvetrain dynamics and this part of it investigates valvetrain dynamics for a Pushrod Follower system. As ever our guides are Prof. Gordon Blair, CBE, FREng of Prof. Blair & Associates, Charles D. McCartan, MEng, PhD of Queen’s University Belfast and Hans Hermann of Hans Hermann Engineering

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.”

In Part One of this article in Race Engine Technology [2], we described the creation of valve lift profiles. In Part Two, we described the “creation of a cam and follower mechanism to reliably provide this designed valve lift profile” and in the process used a relatively simple dynamic analysis of the mechanism to compute the Hertz stresses and oil film characteristics at the cam and follower interface [3]. The analytic technique employed there was not sufficiently detailed as to ensure that the phrase ‘reliably provide this designed valve lift profile’ was completely satisfied.

Due to the sheer extent of this particular subject area it was not possible to cover it properly in any meaningful way in a single article in one issue of Race Engine Technology, as the reader was initially promised by the authors. So we split it into two parts, Part 3A and Part 3B. In Part 3A, we examined the valvetrain dynamics of the relatively simple case of bucket tappets, the most common of direct acting cam follower mechanisms [4]. Here, in this final article in Part 3B, we look in detail at the most difficult of all dynamic cases, the pushrod mechanism with particular reference to the NASCAR engine.

Hopefully, we have now fulfilled our charter to inform the reader how to ‘reliably provide this designed valve lift profile’.

THE DESIGN EXEMPLAR

The pushrod mechanism in the NASCAR Nextel Cup V8 engine is used as the design example. The Cup engine, as almost all readers will know, is a fascinating mix of the most ancient and the most up-to-date engine technology. Yesterday’s technology of pushrod followers, fuelling by a carburettor and ignition by a brainless distributor are mandated by regulation. However, there is nothing ancient in the technology within a near 5.9 litre V8 with a bore of some 106 mm and a stroke of some 83 mm which can produce some 850 hp at 9500 rpm or higher!

The mean piston speed at 9500 rpm is 26.3 m/s, which is right up there with the best of today’s Formula One or MotoGP engines. Each NASCAR cylinder has a swept volume of some 733 cm3 and breathes through a mandated two-valve layout, which cylinder is more than twice the individual cylinder swept volume in a Formula One engine and thrice that of a typical MotoGP unit. ➤

THE PUSHROD FOLLOWER MECHANISM

In Fig.1 is shown a snapshot from the movie in the 'cam manufacture' program of a pushrod mechanism in a NASCAR engine when the valve lift profile is Design A. Design A will be described below in detail. In Fig.2 is a close-up of the cam and tappet contact from that same movie, but with Design E as the valve lift profile and it too will be described below.

The geometry of the actual mechanism for one of the valves is shown in Fig.3 and the cylinder head into which they fit is illustrated in Fig.4. When all of this geometry and the two valve lift profiles are inserted into a single program, as in Fig.5, the relative motion of the valves and the piston can be observed and their proximity to each other and to that piston can be computed. Thus can the inter-valve clashing and piston cut-out dimensions can be determined. Clearly, this same basic geometry must also be numerically available in order to conduct an analysis of the dynamics of the valvetrain motion.

Within the numeric data shown in Fig.3, and graphically in Figs.1-2, are encapsulated the 'NASCAR rules' for the flat cam tappet. It must be 0.875 inch diameter (22.225 mm) and it must be 'flat' to within a specified tolerance which equates to a radius (Rcf) of some 1350 mm. This is actually computed and drawn as such in Fig.2 but the radius is so large as not to show up clearly even on the magnified graphics of this cam to tappet contact.

THE VALVE LIFT PROFILES

In earlier articles [2-4] we described the creation of valve lift profiles and the manufacture of the cams to work with all types of cam follower mechanisms. Somewhat didactically, we showed a valve lift profile as Design A and decreed it to be unsuitable to work with a pushrod follower. Similarly and equally didactically we showed a Design E and said that it would work satisfactorily with a pushrod follower. Here, we will examine these contentions closely and either prove the point or be forced to eat 'humble pie'.

In Fig.5, with our demonstration NASCAR engine working to scale, the intake valve is actually 53 mm diameter which, with an inner valve seat diameter of some 48 mm and using a lift ratio of 40%, yields a maximum valve lift of 19.26 mm. Setting at either end an acceleration ramp of magnitude 0.26 mm, and decreeing a valve lift duration of 180 cam degrees above that ramp, this provides the basic specification of all of the valve lift profiles used within this article. A 'hot valve lash' clearance of 0.23 mm is then assigned to all calculations within the valvetrain dynamic analyses so that the cam tappet indexes the cam just

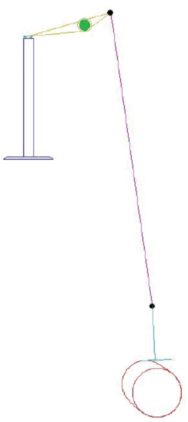


Fig.1 Movie snapshot of the NASCAR pushrod follower system.

Fig.2 Close-up of the cam to tappet contact.

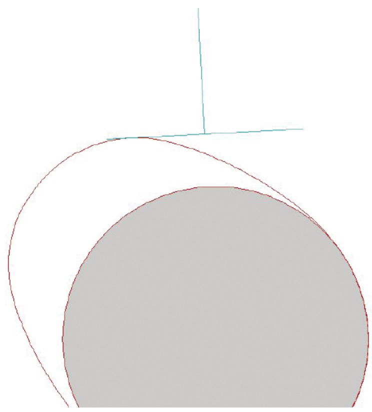


Fig.3 Input data requirements for the pushrod geometry.

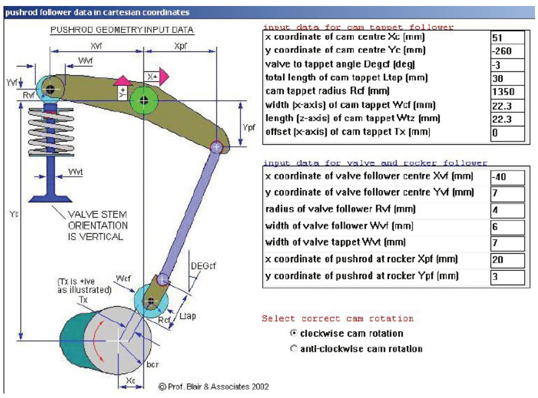


Fig.4 A modified wedge NASCAR combustion chamber.

Fig.7 The input data for valve lift profile Designs A-H.

DESIGN NAME	NEGATIVE ACC. EXPONENT Z	(UP RAMP) POSITIVE ACC. PERIOD I2	(UP RAMP) NEGATIVE ACC. PERIOD I3	(DOWN RAMP) POSITIVE ACC. PERIOD I2	(DOWN RAMP) NEGATIVE ACC. PERIOD I3	LIFT-DURATION ENVELOPE RATIO Kld
A	0.66	30	50	30	50	0.543
E	0.30	42	41	42	41	0.491
F1	0.30	44	41	40	41	0.493
F2	0.30	40	41	44	41	0.489
G1	0.45	42	41	42	41	0.494
G2	0.60	42	41	42	41	0.500
H1	0.30	40	43	40	43	0.498
H2	0.30	38	45	38	45	0.505

before the end of the ramp.

The upshot of the valve lift profile design process for Design A is shown in Fig.6. You will observe that the same attention to detail in the smoothing of the profile, as emphasised to the point of pedantry in a previous article [2], is still maintained here.

In Fig.6 the periods assigned to the various segments of the design process are shown; they are I (ramp), I1 (positive acceleration), I2 (transition acceleration) and I3 (negative acceleration). The numeric values for all valve lift profiles are shown in Fig.7. There are some common data numbers not listed. The ramp period I is 15 deg. The transition acceleration I2 for Design A is 10 deg but for all others, E through H2, it is 7 deg.

THE VALVE LIFT PROFILES FOR DESIGN A AND DESIGN E

The primary focus in this initial part of the article is on the valvetrain dynamics of our 'supposedly mythical' NASCAR engine when using valve lift profile Design A or Design E. In ➤

Fig.5 Movie snapshot of NASCAR valve and piston motion.

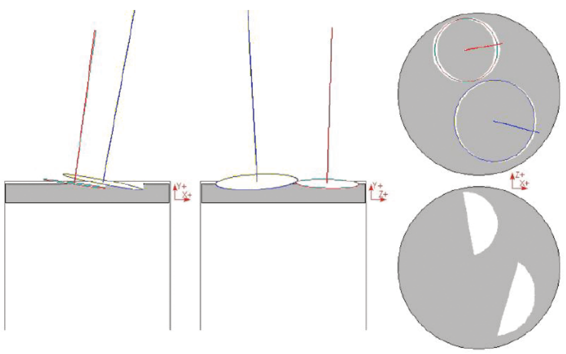


Fig.8 The valve lift for Design A and Design E.

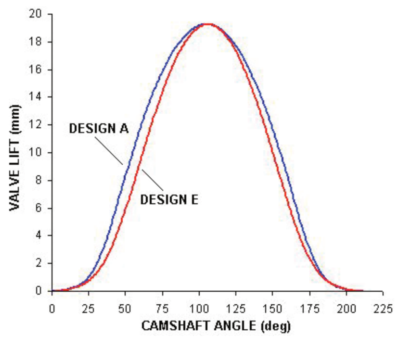


Fig.6 Period durations for valve lift profile design.

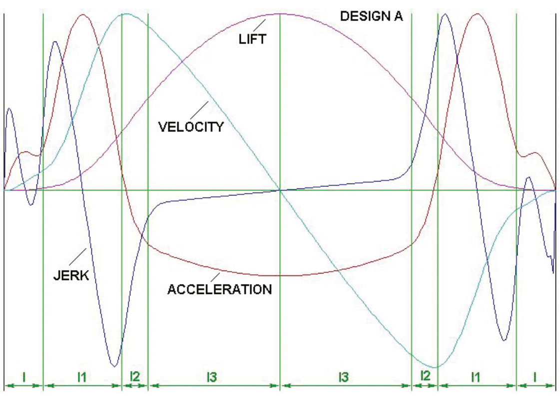


Fig.9 The valve acceleration for Design A and Design E.

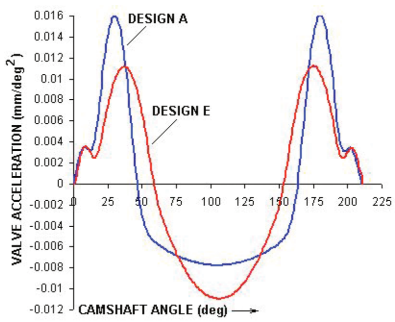


Fig.10 The valve velocity for Design A and Design E.

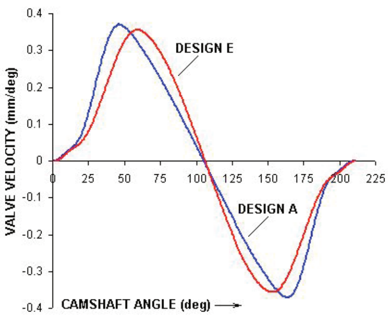


Fig.12 The cam profile for Design A and Design E.

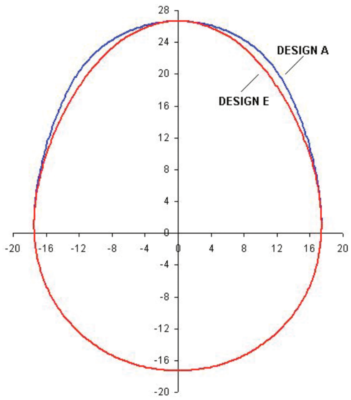


Fig.11 The valve jerk for Design A and Design E.

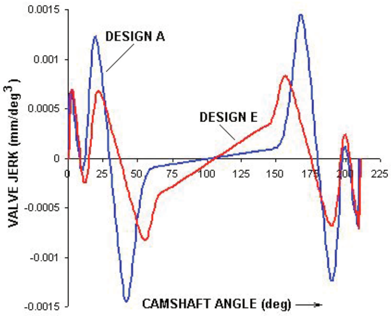


Fig.13 The mathematical model of a pushrod follower system.

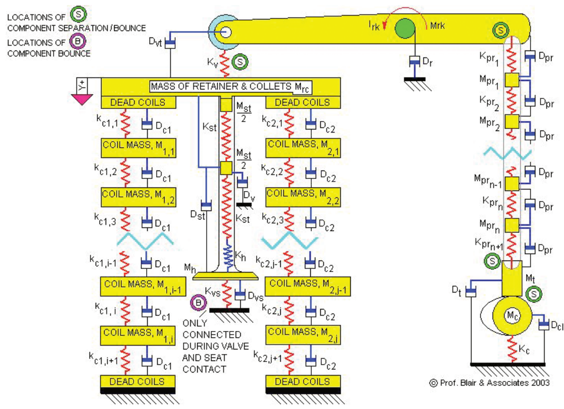


Fig.8 is shown the valve lift profiles for Design A and Design E, their acceleration profiles in Fig.9, their velocity profiles in Fig.10 and their jerk profiles in Fig.11. Although the actual valve lift here is higher at 19.26 mm, these diagrams have exactly the same relative 'shape' as those presented in the original article [2 in Figs.1-5]. So too are the numeric values of their lift-duration envelope ratio, Kld, the measure of 'aggression' of a valve lift profile defined earlier [2 in Fig.2].

When the cam is manufactured within the 4stHEAD software, using the follower geometry of Fig.3, the ensuing cam profiles clearly demonstrate the more aggressive nature of Design A, as seen in Fig.12.

THE MATHEMATICAL MODEL OF THE PUSHROD VALVETRAIN

In Part 3A [4] of the valvetrain investigation we discussed the significance of simple and complex mathematical models of a valvetrain. A realistic mathematical model of a pushrod valvetrain is, by definition, even more complex than that of a bucket tappet. It is shown in Fig.13. While the cam and camshaft, valve head, valve stem and valve springs are treated similarly as for a bucket tappet system, now the mass, stiffness, inertia, etc., of the rocker and pushrod must be added to the model. So too must be added the possibility of separation and bounce of more components, such as at the interfaces between rocker and valve stem, rocker and pushrod, pushrod and cam tappet, and cam tappet and cam.

In Part 3A of this article we promised to comment on the damping coefficients used for the spring-mass-damper systems within the computation. Damping coefficients are not dimensionless but have the units of force/velocity, or Ns/mm in the case of the 4stHEAD software [1].

The literature on valvetrain dynamics is almost entirely bereft of experimental data to use as input data numbers for

"A realistic mathematical model of a pushrod valvetrain is, by definition, even more complex than that of a bucket tappet"

Fig.14 Experimental work at Cork Institute of Technology.

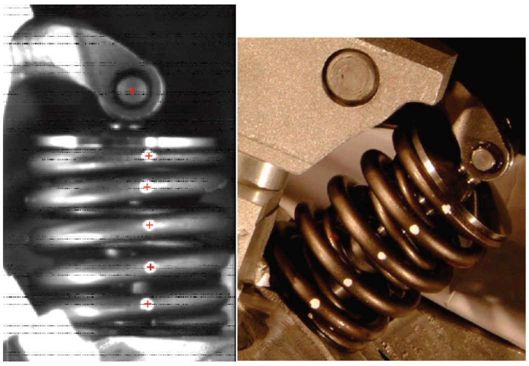


Fig.16 Motoring dynamic valve lift for Design A.

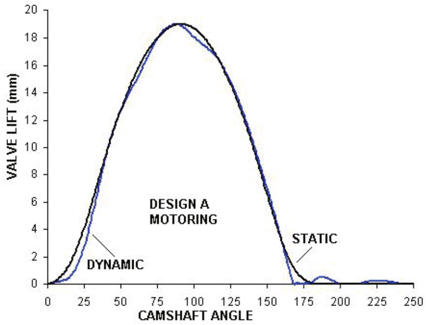


Fig.15 Input data requirements for the pushrod.

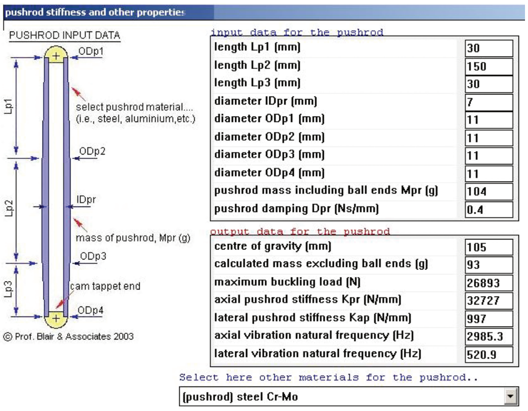
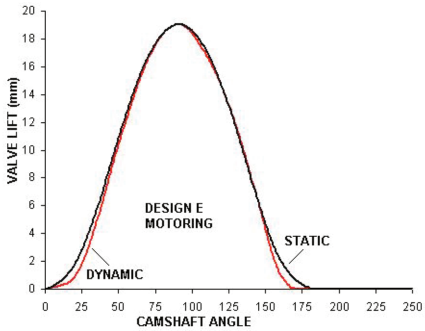


Fig.17 Motoring dynamic valve lift for Design E.



damping coefficients. We have established a collaboration with Dr Keith McMullan at Cork Institute of Technology in Ireland to measure and quantify many of these input data numbers. In Cork, Dr McMullan and graduate student Mr Michael Noonan have a high-quality motoring and video rig from which the movie snapshot in Fig.14 of the valve springs of a pushrod follower system is acquired.

By tracking the motion of each spring coil and comparing it to the predictions given by the 4stHEAD software, it is possible to derive damping coefficients for the coils of the valve springs. Similar methods are employed for other components such as the valve, the pushrod, etc., but component specific rigs are also used to obtain data such as stiffnesses, damping coefficients and natural frequencies, when that particular component is isolated from the others.

THE VALVES AND VALVE SPRINGS FOR THE PUSHROD VALVETRAIN

The methodology of input data creation for valves and valve springs were discussed in Part 3A [4] and we will not repeat it here except to say that, by definition, these intake and exhaust titanium NASCAR valves are larger at 53 and 43 mm diameter, heavier at 71.5 and 57.8 g, respectively, but not any stiffer either in the stem or the head. The inner and outer valve springs are also made with thicker wire and are larger in outside diameter, with stiffness levels at 57 and 23 N/mm, with outside diameters of 41 and 26 mm, each with a wire thickness of 5.5 and 3.5 mm, with 5.5 and 7.5 coils, each with a mass of 106 and 41 g, and with identical natural frequencies of 441 Hz, respectively.

THE PUSHROD

The input data for the pushrod must be very detailed in terms of its dimensions and its material to acquire the necessary data to execute the mathematical model in Fig.13. The relevant input data page is shown in Fig.15. Of interest here is the computed lateral (whipping mode) stiffness of only 3% of its axial stiffness and a lateral natural frequency of just 521 Hz. Clearly, it would be illogical to design the natural frequency of the valve springs and the pushrod to be identical, as a device for the encouragement of mechanical resonance could hardly be bettered.

THE ROCKER

The input data for the rocker is equally detailed and requires not only the geometry of Fig.3 but also its mass, inertia and stiffness. This data can be obtained from a CAD system but it can also be measured and the relevant input data page describes how to do just that. ➤

Fig.18 Motoring valve acceleration for Design A.

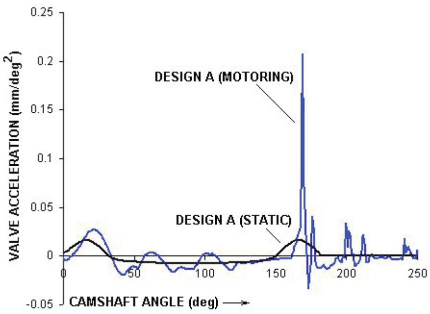


Fig.20 Motoring valve head force for Designs A and E.

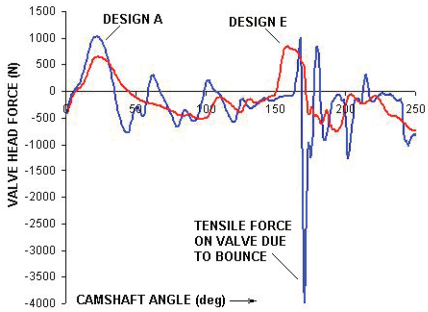


Fig.19 Motoring valve acceleration for Design E.

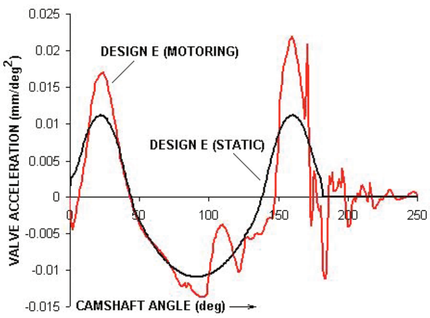
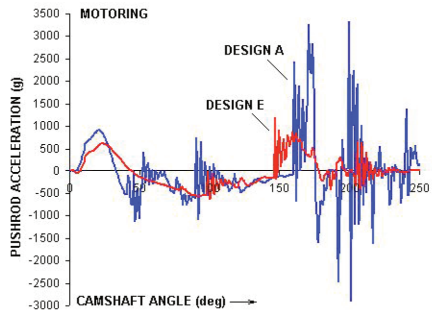


Fig.21 Motoring pushrod acceleration for Designs A and E.



> "Motoring a valvetrain is the normal experimental procedure adopted by NASCAR teams"

DYNAMIC ANALYSIS OF DESIGNS A AND E WHILE MOTORING AT 9500 RPM

The dynamic analysis is conducted while theoretically motoring the valvetrain at an engine speed of 9500 rpm, or 4750 rpm at the camshaft. Motoring a valvetrain is the normal experimental procedure adopted by the racing industry, and the NASCAR teams are no different. All calculations described, and all Figures presented, below are conducted at this camshaft rotational speed of 4750 rpm.

In Figs.16 and 17 are plotted the dynamic valve lift characteristics for Design A and Design E together with their static lift values for comparison. It can be seen that the dynamic lift of Design A contains valve bounce up to 0.5 mm at the valve closure point. Design E shows almost no inclination to provide valve bounce at the same location; in actual fact it does bounce but only to 0.018 mm.

That the valve bounce of Design A is unacceptable becomes clear when examining the dynamic acceleration of the head of the valve shown in Fig.18. The equivalent picture for Design E is shown in Fig.19. The acceleration of the valve head rises to 0.22 mm/deg2 whereas that for Design E is only 0.02 mm/deg2. As acceleration is linearly related to force, that raises the force on the valve head for Design A to an order of magnitude above that for Design E.

This can be seen in Fig.20 where the forces on the valve head are plotted. The maximum/minimum forces for Design E amount to +600/-700 N but for Design A that rises to +1000/-4000 N. The valve head experiencing the Design A lift profile is being bent or stretched with an impact force of up to 4000 N (408 kgf or 913 lbf) and on a 53 mm diameter titanium intake valve it is most unlikely to keep doing it for very long.

Further evidence of the unsuitability of Design A with a pushrod follower system is shown in Figs.21-23. Here are plotted the pushrod acceleration, pushrod forces and cam tappet forces for both Designs A and E. Although the computation calculates this data at all locations along a pushrod, we plot here the acceleration at its centre, and the

Fig.22 Motoring pushrod force ratio for Designs A and E.

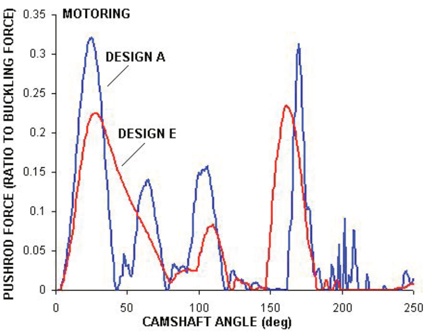


Fig.23 Motoring cam-to-tappet force for Designs A and E.

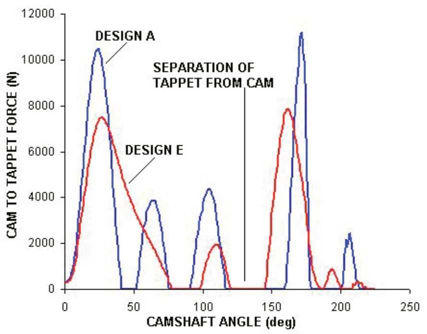


Fig.24 Firing dynamic exhaust valve lift for Design A.

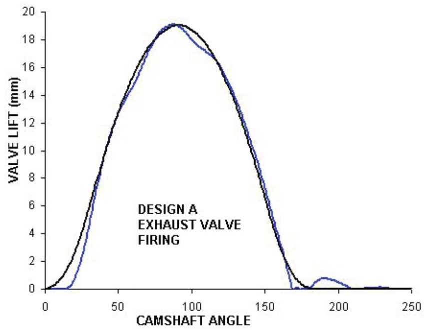
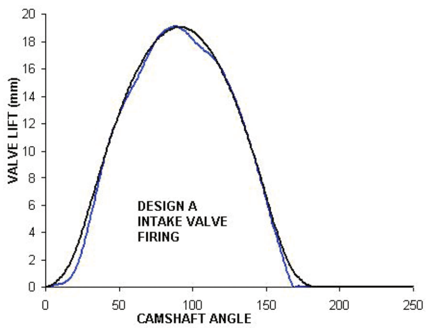


Fig.25 Firing dynamic intake valve lift for Design A.



force at its ‘attachment’ point to the rocker, where that force is plotted as a ratio to that which would cause a catastrophic buckling of the pushrod.

The violent shaking of the pushrod when actuated by Design A can be clearly seen in Fig.21, not only at the valve closure point but also at two other locations around maximum valve lift. The origins of these locations can be identified in Fig.22 where the force (ratio) goes to zero twice for Design A, whereas it does so for Design E only around valve closure.

The significance of the force at the top of the pushrod going to zero is that it does so by separating its contact with the rocker, thereby releasing all of the stored energy in the pushrod to shake it at its natural frequency. We can plot the same force data anywhere along this pushrod but it is more informative to plot it at its cam tappet end to see if it has similarly waltzed away into free space and at the same junctures in the cam angle diagram!

In Fig.23 we show the cam-tappet forces and, sure enough, the cam tappet with Design A has separated its contact from the cam at the same intervals as the top of the pushrod with the rocker, but for even longer periods at the cam tappet end. Even Design E is not immune from separation of the cam tappet from the cam. The cam-tappet forces with Design A are some 25% higher than with Design E which means that the Hertz stresses at the cam and tappet interface at those junctures have been raised by the same proportion.

DYNAMIC ANALYSIS OF DESIGN A AND E WHILE FIRING AT 9500 RPM

The effect on the valvetrain dynamics of the exposure of an exhaust or an intake valve to the cylinder pressure was debated earlier in Part 3A of this investigation [4]. Here we repeat the analysis for the pushrod follower system and, to make comparisons even more relevant, we will use exactly the same cylinder pressure diagram [4 in Fig.25]. This is quite logical as the brake mean effective pressure (bmep) in both example engines are virtually identical as are the exhaust and intake tuning behaviour.

The results for the dynamic lift of the exhaust and intake valve for Design A are shown in Figs.24-25 and similarly for Design E in Figs.26-27. As a pushrod system is normally much less stiff than a bucket tappet device, the exhaust valve opening is even further delayed. The delay on exhaust valve opening shown in Figs.24 and 26 is 11 degrees at the camshaft which is 22 degrees at the crankshaft whereas that of the bucket tappet was just 4 camshaft degrees [4].

The wobbly nature of the dynamic valve lift around the maximum valve lift for Design A is even further exaggerated and the valve bounce at valve closure with Design A has increased to 0.75 mm. Design E is no longer immune to valve bounce under ‘firing’ conditions but it is contained to just 0.07 mm. ➤

Fig.26 Firing dynamic exhaust valve lift for Design E.

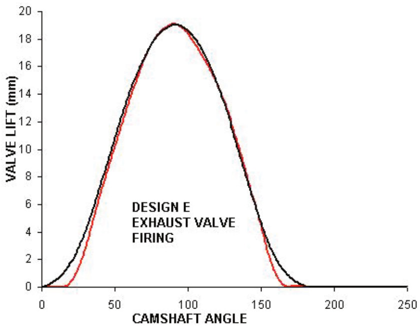


Fig.27 Firing dynamic intake valve lift for Design E.

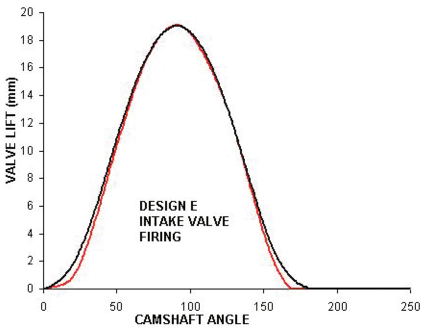


Fig.28 Static valve acceleration diagrams for Designs E and F.

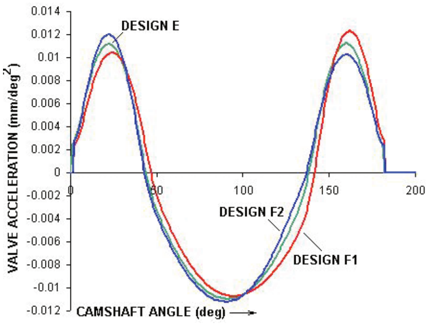
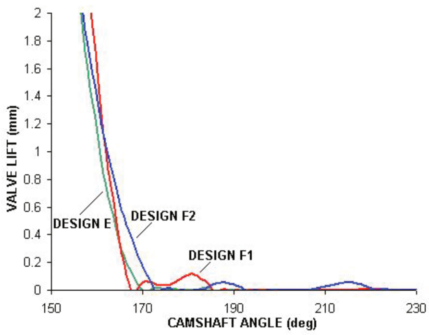


Fig.29 Motoring dynamic valve lift for Designs E and F.



The case of the intake valve under firing conditions continues the pattern seen previously [4]. With Design A the dynamic lift 'wobble' at maximum valve lift is still present but somewhat reduced as is the valve bounce at valve closure which is now just 0.065 mm. For Design E, the valve bounce is identical to the motoring case at 0.02 mm.

THE DESIGN IMPLICATIONS OF THE VALVETRAIN DYNAMIC ANALYSES

Quite irrespective of the vindication of our opinion [2] that a valve lift profile like Design A is unsuitable for use with a pushrod actuated valvetrain, as in this NASCAR engine exemplar, the fundamental message to the designers of such engines is that there is a limit to the aggression that can be applied to the valve lift profile in such engines. Design E is acceptable, but how much more aggressive can it be made before it too becomes unacceptable? That is the real design question.

The subsidiary information from the above analyses of the firing case, perhaps even more vital, is that the 'acceptable' valve lift profile for an exhaust valve is not the same as for an intake valve and the optimum profile design for either valve is not going to be found by conducting analyses under motoring conditions either theoretically on a computer or experimentally with a motoring rig.

A further implication, and a major complication, concerns the use of an engine simulation [5] by the designer to optimise the performance characteristics of an engine. Apart from loading in all of the geometrical and thermodynamic data for the engine to that simulation, an extensive exercise in itself, the valve lift characteristics for the intake and exhaust valve(s) must also be inserted.

Clearly the only relevant data for this purpose is the actual valve lift profile under dynamic/firing conditions at any given engine speed otherwise the simulation cannot possibly predict the instantaneous gas flow into, through, and out of, the engine cylinders during the engine cycle. The static valve lift is clearly useless for this purpose but the motoring data is little better as a true representation of how the valve(s) move under firing conditions. So what is the solution to this problem?

Experimentally, one must measure the dynamic valve lift under firing conditions at every engine speed and use that within the engine simulation for further theoretical optimisation of the engine; this is an almost impossible task. Theoretically, if one uses a valvetrain dynamic analysis which correlates well with experimental data acquired experimentally while motoring the valvetrain, it should be capable of being programmed to accurately predict the dynamic valve lift under firing conditions which data can then be exported to the engine simulation. Even so, it must

Fig.30 Static valve acceleration diagrams for Designs E and G.

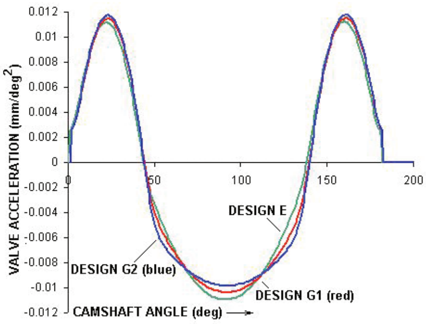


Fig.32 Static valve acceleration diagrams for Designs E and H.

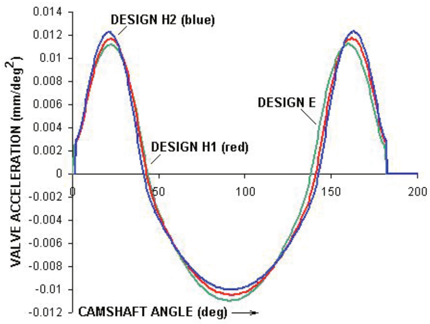


Fig.31 Motoring dynamic valve lift for Designs E and G.

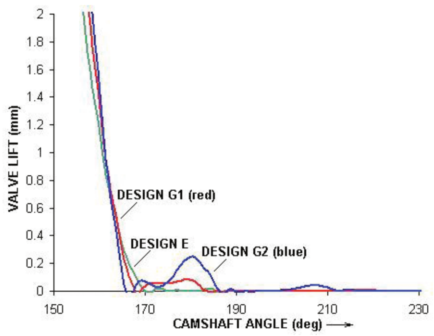
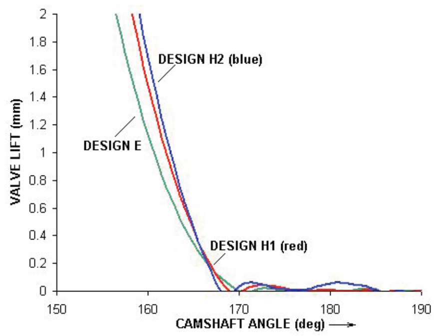


Fig.33 Motoring dynamic valve lift for Designs E and H.



needs be an accurate engine simulation [5], a point we have emphasised, nay belaboured, many times [6].

OPTIMISATION OF A VALVE LIFT PROFILE IN A PUSHROD ENGINE

As we write above, 'Design A has proven to be unacceptable and Design E is acceptable, but how much more aggressive can it be made before it too becomes unacceptable?'

As seen in Fig.7, the lift-duration envelope ratio Kld for Design A is 0.543 while that for Design E is 0.491. That is a gas flow time-area disadvantage for Design E of almost 11%, which could be crudely interpreted to infer a potential 11% reduction of torque or bmep or power accruing to its use, compared to employing Design A.

In Fig.7, we show the data for the creation of valve lift profile Designs G1 and G2, F1 and F2, and H1 and H2, almost all of which marginally raise the Kld value of the profile over that for Design E. As these latter Kld values exhibit only marginal gains over Design E, it could be argued that they are not worth the investigation time involved. Not necessarily so because (a) the NASCAR designer will cheerfully tell you that he will 'kill' for but a few more horsepower, (b) the intake valve can tolerate more lift profile aggression under firing conditions than the exhaust valve and one of these latter designs F-H may fall into that category of

"We never cease to be amazed at the minute differences which exhibit 'better' valve bounce behaviour"

'success' and (c) as we never cease to be amazed at the minute differences in valve lift profile which exhibit 'better' or 'worse' valve bounce behaviour in what is fundamentally a highly non-linear mechanical system then one of these F-G profiles could well prove worthy of designer-time to click that mouse but a few times more.

In Fig.28 are the acceleration diagrams of Design E (shown as a green line in all of these final Figures) and Design F1 and Design F2. The objective is to vary the positive acceleration periods I2 while retaining all other characteristics but rendering the lift diagram to be asymmetrical by a few degrees. The dynamic valve lifts under motoring conditions ➤

for the Designs E and F are shown expanded around the valve closure period in Fig.29. Both Design F1 and F2 bounce more than Design E, but Design F1 might have some potential for an intake valve profile under firing conditions.

In Fig.30 are the acceleration diagrams of Design E and Design G1 and Design G2. The objective is to retain all other characteristics of Design E but increase the negative acceleration exponent Z in order to increase their Kld values. The dynamic valve lifts under motoring conditions for the Designs E and G are shown expanded around the valve closure period in Fig.31.

"Such a realistic valvetrain computation is capable of detecting the effect of even a minor change to the input data"

Both Design G1 and G2 bounce more than Design E, Design G2 unacceptably in amplitude terms, but Design G1 might also have some potential as an intake valve profile under firing conditions.

In Fig.32 are the acceleration diagrams of Design E and Design H1 and Design H2. The objective is to retain all other characteristics of Design E but progressively decrease the positive acceleration period I1, and correspondingly increase the negative acceleration period I3, to increase the Kld values for Designs H1 and H2. The dynamic valve lifts under motoring conditions for the Designs E and H are shown in Fig.33.

Both Design H1 and H2 bounce more than Design E, Design H2 unacceptably if not in amplitude at least in duration, but Design H1 might possibly make an intake valve profile under firing conditions.

In short, our minor changes to the valve lift profile have not produced any 'magic' improvement in the valve bounce behaviour over the 'successful' Design E, which is an experience not unknown to the NASCAR experimenters on their motoring valvetrain rigs while testing a multitude of camshafts! What it does emphasise is that a realistic valvetrain computation that includes separation and bounce of the components is capable of detecting the effect of even a minor change to the input data.

Put rather more bluntly, more elaborate computation models have been produced with valves, valve springs, rockers and pushrods rendered in the FEM manner, and they are admittedly very pretty to watch. However, if they cannot incorporate component separation and bounce everywhere and cannot incorporate cylinder pressure conditions facing the valves, they do not constitute a design tool.

CONCLUSIONS

It is possible today to model valvetrain dynamics with some considerable accuracy provided that the mathematical model is sufficiently extensive. We regret that it is not possible, due to space limitations here, to describe the effects on valvetrain dynamics of a host of other input data variables. Needless to add, a minor variation of the numeric value of almost every single input data value in this non-linear system can have a profound effect on the outcome.

We hope that the tyro has found these four articles on valvetrain design to be educational and possibly the experienced designer has also found them useful through the discussion of a particular topic from a viewpoint outside that experience. We enjoyed writing them so we can but hope that you enjoyed reading them. ■

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