Pushrod valvetrain may have their drawbacks but they thrive in some racing series. Wayne Ward looks at their design issues

Large pushrod V8 engines have formed the backbone of the passenger automotive industry for many decades in the US, so it is natural that such engines have been used widely for motorsport and it is likely that this will continue for some time. It is only recently that the top tier of US circuit racing, NASCAR’s Sprint Cup, has allowed engine manufacturers to race with engines not based on a production cylinder block [Fig. 1]. However, the rules governing the design of these bespoke pushrod race engines mean they still closely resemble the production blocks that preceded them, but are designed with modern principles in mind and using modern engineering methods such as CFD and FEA.

Many people in motorsport will have only ever seen or studied overhead cam (OHC) engines, so it is worth explaining the basics of the pushrod system. Most pushrod engines have two valves per cylinder, and one of the more appealing aspects of the valvetrain is the number of camshafts required and the position the camshaft occupies in the engine. A typical OHC vee engine, such as a current Formula One engine or the previous Champ Car engine, will have four camshafts: one for inlet and one for exhaust on each bank. These four camshafts are mounted at the very top of each bank, operating
the valves from above. A V8 pushrod engine uses a single camshaft, and this is mounted in the cylinder block, above and close to the crankshaft. Not only are there far fewer things to provide a half-speed drive to, but the distance over which the drive is taken is very short and the centre of gravity of the camshaft is low.

In a typical pushrod engine, the cams cause a follower to translate within a bore. In a race engine, these followers will generally through choice be a roller follower, so named because a roller at the tip of the follower is carried on a bearing. Flat-faced followers are used only by Sprint Cup racers because the rules dictate this. Into the follower is located the lower end of the pushrod, and this part of the linkage connects the bottom end of the engine to the top. The upper end of the pushrod operates a rocker. Typically the rocker will have a roller tip, but rollerless rockers have been developed and used with some success in NASCAR. There is a good picture of such a rocker in the previous ‘Focus’ article on the subject (1).

Pushrod valvetrain development continues apace within the tight confines of NASCAR, especially in its top two series, Sprint Cup and Nationwide. Sprint Cup, the top series, mandates the use of flat-faced lifters, while Nationwide allows roller followers. As we shall see, the type of lifter fundamentally affects the tuning potential of the engine.

Considerations in valvetrain design: stiffness and natural frequencies

The pushrod valvetrain is technically difficult to deal with, owing to several factors, chief among which are its inherent compliance and the fact that there are a number of points at which clearance may appear in the system. The complexity of the system lends itself very well to calculations that can be carried out by computer. Even in the 1960s, advances in computing power allowed engineers to treat the pushrod valvetrain as a far more complex multi-mass system than had been possible in practice before.

In his paper on the subject, Wagstaff (2) calculated the system response for a variety of degrees of freedom in the system, and included the effects of clearances within his calculations. The method by which he arrived at the main damping coefficients in the system was by practical experiment; it is probably still by the study of the vibration of practical systems that an accurate correlation between modern valvetrain simulation software and real-life engines can be arrived at. In common with the articles published in Race Engine Technology contributed by Blair et al (3), Wagstaff gave paramount importance to modelling the behaviour of the valve spring properly so that the predicted system response during resonant conditions (spring surge) was accurate. In their paper on the subject of modelling the Winston Cup valvetrain (Winston Cup eventually became Sprint Cup), McLaughlin and Haque (4) conclude that “the valve spring dynamics dominates the performance of the valvetrain”, especially in situations such as valve bounce.

So, in our calculations of the behaviour of a pushrod valvetrain system, we need to pay great attention not only to the design of the specialist valvetrain components such as pushrods, lifters, rockers and so on, but we should also take care to model the behaviour of the spring. The valvetrain engineer has to pay special attention not only to the natural frequencies of the spring, and their relation to both the fundamental frequency of the stimulus – that is, the cam profile – and its harmonics, but also to the natural frequencies of the associated components.

This was brought to my attention during a conversation on the subject with the late Prof Blair: his point was that it is very easy for people not to understand the system and to have a pushrod with a natural frequency that causes resonance in the valve spring. Any component will have a number of different stiffnesses depending on the direction; for example, a simple rectangular beam has two different bending stiffnesses, an axial stiffness and a torsional stiffness.

A part will also have a number of natural frequencies. An axisymmetric component such as a pushrod can have its lateral and longitudinal stiffnesses tuned independently so that their fundamental natural frequencies fall outside the operating range of the engine. As is the case with valve springs and the avoidance of surge, we need to ensure that significant harmonics from the cam do not excite resonances in the valvetrain components. This might entail designing the components such that their natural frequencies are four, five or more times higher than the basic natural frequency of the cam.

The consideration of component stiffness was also highlighted during discussions I had with component suppliers and technical experts at some of the top NASCAR engine suppliers. With the advent of valvetrain simulation software and the increasingly widespread use of valvetrain rigs, people have really begun to appreciate the value of improving component stiffness.

Consequently, we have seen some very strong trends in component design over recent years – pushrods have become thicker and heavier; valvetrain engineers are now more than happy to sacrifice some mass in order to significantly increase the stiffness of the components in the system, especially on the pushrod side of the rocker; and steel has begun to supplant aluminium for rockers, not only for greater stiffness but for lower moment of inertia too. More on these design and material trends will be expanded in the component discussions that follow.
**FOCUS : PUSHRODS, ROCKERS & LIFTERS**

**Camshafts**

Camshafts for pushrod engines differ significantly from those used in OHC engines, and not just because a single camshaft carries the cam lobes to open all of the engine’s valves. A striking difference is the fact that the largest features on the shaft are the cam bearing journal diameters [Fig. 2, page 57]. The camshafts slide into the block from one end of the engine; the radius from the camshaft axis to the top of the cam lobes is therefore limited.

In order to improve engine performance and control fuel consumption, development of camshaft profiles is very important. When the cam lobe radius (base circle radius plus cam lift) is at the maximum possible size, increasing lift necessarily means that the base circle diameter is reduced. This leads to increased flexing of the camshaft between the bearings, and this is the source of the first of the valvetrain engineer’s ‘headaches’. Higher lift camshafts with more aggressive levels of valve acceleration are controlled by heavier springs. The forces due to the spring and the acceleration of the masses and inertias in the system, combined with the lower stiffness of the camshaft, can lead to significant deformation.

NASCAR limits the camshaft journal diameter to 2.362 in (60 mm), preventing any scope for stiffening the camshaft by increasing bearing diameter. As ever, there are design options to provide more camshaft stiffness that the NASCAR valvetrain designer may use. Other race series don’t impose similar bearing size restrictions, and valvetrain designers may choose to take the associated rise in friction from the larger bearings to give more scope for performance improvement.

**Lifters**

Lifters (also widely known as followers or tappets) come in two basic types for racing – flat-faced and roller. Flat-faced lifters limit cam profile development because of the restrictions they place on the opening and closing velocity of the cam. Basically, the higher the opening or closing velocity, the larger the diameter of lifter required.

This can be overcome to an extent by using ‘mushroom’ lifters that have an increased diameter cam-contact face. However, NASCAR’s premier series, which enforces the use of flat-faced lifters, does not allow mushroom lifters and goes further than this in limiting the follower to be 0.875 in (22.22 mm) in diameter [Fig. 3]. As far as pushrod engines are concerned, flat-faced followers are an arcane technology, used only rarely in current production engines, despite the fact that they are much simpler and therefore less expensive, to produce. Given the focus placed on low production costs by car producers, for them to use roller lifters in preference to cheap, flat lifters there must be very real benefits to using roller lifters, even for production engines.

For racing, where engineers have a choice, roller lifters are used [Fig. 4]. As mentioned, flat-faced lifters suffer in limiting lift velocities, and the relationship between lift velocity and follower radius is widely documented [1]. There are two real advantages to using roller followers. The first is that the design and development engineers have much more freedom with cam profiles and the resultant valve lift profiles. Care must be exercised here though; the paper by Prof Blair (2) published in Race Engine Technology gives examples of cam profiles for flat-faced lifters, but the lessons of tempering the ‘aggression’ of cam profiles apply equally to all types of followers.

If we ignore the matter of contact (Hertzian) stresses, the limiting factor for a roller follower is pressure angle. This is the angle between the axis of the follower translation – that is, the lifter bore axis – and the normal to the cam/follower contact. There is a critical angle by which the follower will tend to bind in the bore rather than translate smoothly. High pressure angles can also lead to increased friction and wear. The maximum pressure angle is a function of follower velocity and also of acceleration at the point of maximum pressure angle. In general, roller-lifter pressure angle is influenced by the base circle diameter of the camshaft and the follower roller radius, lift and any eccentricity (lifter bore axis offset from camshaft axis) with larger base circle diameters having smaller pressure angles for a given lift profile. Offsetting the lifter bore axis so that it doesn’t intersect the camshaft axis can be an effective way to control pressure angles. Rothbart (5) and Chen (6) both provide the mathematical formulae for pressure angle for roller followers.

The level of surface finish achieved and any surface treatments or coatings applied will have a direct effect on frictional losses. The surface finish will dictate the lubrication regime, specifically the extent to which the operating cycle at any engine speed the cam-lifter contact operates in a fully hydrodynamic/elastohydrodynamic regime, and what proportion of the cycle is a mixed regime. A better surface finish leads to a lower proportion of the cycle operating in a mixed lubrication regime: the greater the proportion of the cycle...
during which the cam-to-follower contact is one of hydrodynamic (or elastohydrodynamic) lubrication, then the lower the frictional losses. A low-friction coating means that any time operating in the mixed lubrication region is subject to a lower coefficient of friction, and this in turn lowers the overall frictional losses.

Where we have the correct conditions for hydrodynamic lubrication, the tangential friction component is negligible, and the subsurface stress field is very similar to that of a static contact. Subsurface fatigue leads to pitting, the fatigue crack being initiated at or close to the site of maximum stress. The depth of this maximum stress depends on the geometry of the contacting bodies, the loads and the materials in use.

Where there is a more significant friction component, as is the case with mixed lubrication, there are two main effects on the subsurface stresses. First, and most important, the maximum stress is increased. The second effect is to reduce the depth at which the maximum stress occurs in the material. Where lubrication is inadequate and sliding motion is dominant, then the maximum stress is very close to the surface, and surface damage can be confused with conventional wear mechanisms rather than subsurface fatigue.

In terms of roller followers, the same grade of material may be more highly stressed if used for a roller (compared to a flat follower) without suffering from surface wear or subsurface fatigue, as the contact should be one of pure rolling rather than sliding. This is not to say that rolling contacts won’t suffer from fatigue, but it will happen at a higher level of stress.

In aiming for maximum system stiffness, we might expect there to be a trend towards shorter pushrods, and this could be achieved with a taller lifter. However, the angularity of the pushrod and its effect on frictional losses between the lifter and its bore need to be considered. Wear of the lifter bore may become a problem when angularity is too great. One supplier of the complete range of pushrod valvetrain components, from cams to rockers, said its philosophy is to keep the pushrod seat in the lifter as low as possible in order to minimise angularity.

**Pushrods**

The vast majority of pushrods are made from steel. NASCAR is quite specific on the types of materials it will allow for its race series, and much of the rest of the market benefits from the same materials and manufacturing techniques.

NASCAR mandates the use of magnetic steel materials for pushrods. Since most steel materials have very similar elastic moduli, the stiffness for a given geometry is not greatly affected by the choice of steel. So unless another property such as strength or toughness is required, the pushrod designer and manufacturer need not venture into exotic and expensive materials. If using a steel material, the stiffnesses and natural frequencies of the pushrod are a function of geometry alone. Where steels are required for extreme use, especially in drag racing applications, the choice of materials may extend to tool steels for the main body of the pushrod. Compared to more conventional steels, these offer improved strength, toughness and impact properties.

Some race series do not limit the use of materials to steels. In such series, both aluminium and titanium pushrods are used, but more adventurous producers have looked into other types of materials.

Those who produce titanium and aluminium pushrods for competition say they are chosen primarily by motorcycle competitors.

Composite materials are perhaps not ideal candidates for pushrods, but if they are allied to more conventional metallic materials, they become more practical. In an ideal world, one might reasonably design a pushrod using a composite material with fibres running in a predominantly axial direction; this would provide the maximum axial and bending stiffness. However, the tangential stiffness would be poor, as there would only be resin providing stiffness in this direction. Careful thought is required here in order to provide a cylindrical pushrod tube with excellent axial stiffness per unit mass combined with sufficient tangential stiffness to prevent splitting of the tube.

There are options to ‘fit’ a thin sleeve around a CFRP (carbon fibre reinforced polymer) inner, but the only company willing to talk about development projects involving composites for use in pushrods does not take this approach; instead it uses a metal matrix composite with a carbon fibre reinforcement. Its comment on the carbon-reinforced material is that it provides a lot of extra stiffness in compression, but the properties of the material in bending are not optimal. The highly directional properties of fibre reinforced composite materials would apply to short-fibre metal matrix composites to an extent, especially if the material had any significant extrusion during processing. One pushrod manufacturer admits to investigating aluminium beryllium materials in the past.

One thing is certain in terms of materials selection, and that is the choice of the type of material for the ends of the pushrod. Where a single-piece component is specified, the ends are clearly going to be of the same material as the body, and the body will therefore be subject to any surface treatments required for the ends. With nitriding being mentioned as a common surface treatment for pushrod ends in order to improve wear resistance, the body of the pushrod will benefit from the improved corrosion resistance imparted by this process. Where three-piece pushrods are specified, the ends will generally be much harder than the body. As we have mentioned earlier, there is often no advantage to using an exotic steel in the body of the pushrod, and so the ends use a material that is much harder and wear resistant. Tool steels are common choices here for many applications, although the grade of steel used is not universal. Suppliers who gave more detailed answers on materials selection admitted to using both shock-resisting and hot-work tool steels.

In terms of pushrod manufacture, the ability to react quickly to
orders is vital. As we shall see later, engine suppliers are now far more likely to use ‘fixed’ rockers, as in without adjusting screws. This requires that they are able to quickly source pushrods of the correct length to suit each valve, unless they are to hold a large inventory of different lengths. This may mean resorting to measures such as hard turning, especially where pushrods are made of hardened and tempered steels. Hard turning lends itself to materials such as tool steels. One supplier was very proud to say that it can ship solid (one-piece) tool steel pushrods of any length within 24 hours of the order being received.

In many cases, the pushrod will be the most flexible single component in the valvetrain. Great strides in valve control have been made since valvetrain test rigs became more widely available [Fig. 5]. Such test machinery has helped point the way towards stiffer valvetrains, and improving the stiffness of pushrods has certainly been an important part of this [Fig. 6].

In terms of axial stiffness, where materials are essentially fixed in terms of modulus – as is the case with NASCAR’s ferrous magnetic rule – the valvetrain specialist will look to increase pushrod cross-sectional area, and the trend in recent years has been towards larger-diameter pushrods in order to improve valve control. Such pushrods clearly come with a significant weight penalty, but the importance of stiffness outweighs considerations of pushrod mass generally. When questioning the senior engineers at some of the NASCAR Sprint Cup engine suppliers, one of them noted that they are currently at the limit of what is possible by increasing pushrod diameter, as the bore in which it operates is the restriction to further increases in diameter and stiffness.

**Rockers**

Where rockers are concerned, there are two main aims: the reduction of moment of inertia around the rocker pivot, and an increase in stiffness, both of which are valuable to the valvetrain designer. With lower moment of inertia, we find reduced forces and, as is the case when using a lighter valve, a lighter spring is required, reducing forces and therefore reducing frictional losses. Stiffness helps with the control of the valve. If the valvetrain development engineer wants the valve to dance to his tune, then rocker stiffness is an important part of his method of achieving this.

With pushrods we find that people are prepared to sacrifice lightness to gain stiffness, while with rockers they are prepared to sacrifice ease of use to gain stiffness and reduce inertia. In sacrificing ease of use, I’m referring to deleting the adjusting screw and locknut on the pushrod end of the rocker. These allow, within reason, one pushrod length to be used for a number of valves, with any fine adjustment taken up by the screw and locknut. However, if the engine supplier uses steel rockers and is prepared to hold a larger inventory of pushrods, or is able to have pushrods made quickly to his exact requirements, he can machine a hemispherical socket directly into the body of a rocker. This strategy is not an option for those who continue to use aluminium rockers.

To a large extent, steels have supplanted aluminium for expensive applications where the design has been optimised for stiffness and inertia [Figs. 7 and 8]. There are a number of suitable alloys in terms of having sufficient fatigue strength, but one company mentioned that it favours 7000 series alloys in terms of stiffness. However, we should not assume that the aluminium rocker body is static in terms of materials selection. In discussing rocker materials for this article, it was pointed out that, in terms of elastic modulus (stiffness), all aluminium alloys are “not created equal”. There are some that would offer a very significant improvement on 2000 and 7000 series alloys in terms of stiffness. Some aluminium alloys, whose unconventional manufacturing methods impart some truly impressive strength and stiffness properties, may prove to be very successful. However, the machining and subsequent handling of parts made from them will prove to be crucial, as such materials often have very low ductility. Any accidental ‘dings’ or hard contacts can initiate fatigue cracks.

Where the rocker contacts the valve lash cap, we most often find a roller bearing. The roller bearing, providing that it rolls, has very low orders is vital. As we shall see later, engine suppliers are now far more likely to use ‘fixed’ rockers, as in without adjusting screws. This requires that they are able to quickly source pushrods of the correct length to suit each valve, unless they are to hold a large inventory of different lengths. This may mean resorting to measures such as hard turning, especially where pushrods are made of hardened and tempered steels. Hard turning lends itself to materials such as tool steels. One supplier was very proud to say that it can ship solid (one-piece) tool steel pushrods of any length within 24 hours of the order being received.

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friction as it sweeps across the top of the valve. However, because it is required to be supported on a shaft, the rocker loses stiffness in this area. The rollerless rocker, which was discussed in the previous Race Engine Technology article on pushrod valvetrains (1), is aimed squarely at maximising stiffness while minimising inertia. However, because there is some sliding action of the rocker tip across the lash cap, frictional forces tend to load the valve laterally. To minimise this frictional force, the profile of the tip is not a simple radius, but aims to replicate the rolling action of an involute. Rollerless rockers are not in widespread use, and although they have been developed for NASCAR Sprint Cup, they have still found only limited acceptance among engine suppliers.

Another area of the rocker assembly that has an effect on stiffness is the pivot. Opinions vary as to whether a plain bearing or rolling element bearing is the best compromise here between friction and stiffness. The plain bearing option is felt to offer greater stiffness, but rolling element bearings are said to offer lower friction. The plain bearing option also requires a feed of pressurised oil to each rocker pivot, and this can involve significant complication. Rockers that use rolling element bearings at the pivot do not require a pressurised oil feed and can rely on an oil mist or splash lubrication.

The mounting of the rocker and the choice of pivot style are also important considerations. Shaft-mounted rockers are favoured where maximum stiffness is required, and the rocker stands to which the rocker shafts are attached are designed with stiffness in mind. Stud-mount rockers remain popular, although they require pushrod guide plates in order to keep the rockers in the correct orientation [Fig. 9].

Pushrod valvetrain alternatives: the Ilmor-Mercedes 500I

Mercedes is not a company known for its pushrod engines in recent times, but in 1994 the Ilmor-designed Mercedes-Benz 500I engine – designed and manufactured by British race engine specialist Ilmor – took Indianapolis by storm. The rule makers had provided pushrod engines with a capacity and boost pressure advantage in an effort to keep such engines competitive when ranged against small, bespoke OHC race engines. This was ostensibly to keep the production-based pushrod engine projects involved at Indianapolis.

The Indy race was USAC-sanctioned, and ran to different rules from the rest of the CART races; it was therefore an Indy-only exception that was granted to pushrod engines. However, there was no stipulation that the engine had to be production-based. Sensing an advantage, a new engine was designed in secret by Ilmor which produced prodigious power, and which was to take a stunning, if controversial, victory in the 1994 Indy 500. The engine was a two-valve-per-cylinder 3.43 litre pushrod V8, designed to minimise the disadvantages of the pushrod layout as far as valvetrain stiffness and other limitations were concerned. The camshaft was mounted as high in the vee of the engine as was practical, minimising the pushrod length.

The most striking difference from any other pushrod engine is the use of a type of cam follower more normally seen in an OHC engine. A finger follower was chosen, providing a very direct coupling between the cam and the pushrod. This is a very stiff component between the two surfaces contacting the cam and pushrod. It is lighter than a comparable conventional flat-faced lifter, and suffers none of the complexity and lack of stiffness that can be associated with a roller lifter. Furthermore, compared to a conventional flat-faced cylindrical lifter, it is not limited by having to increase lifter diameter to increase lift velocity. While the finger follower’s lower pad, which contacts the cam, may have to increase in size in order to accommodate aggressive profiles, this never becomes limited by having to house two cylindrical lifters within a confined space. Depending on your preference for the relative widths of cam and finger follower, the finger pad needs to be only slightly narrower, or wider, than the width of the cam lobe.

In the engine’s single Indianapolis outing, it used its power advantage over the four-valve-per-cylinder OHC V8s to charge to victory in what can be argued wasn’t a great car. The same Penske PC23 cars failed to qualify for the race a year later, this time without the considerable power of the Mercedes 500I [Fig. 10, page 64], which was said to produce more than 1000 hp. Sensing that the Mercedes was too powerful to allow other engines to compete, USAC dropped the boost pressure for pushrod engines immediately after Indy, and then again later in the year. It would also later retrospectively
SOME EXAMPLES OF PUSHROD, ROCKER & LIFTER (TAPPET) SUPPLIERS

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USA
Alan Johnson Performance Engineering
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Bullet Racing Cams
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CAM FX
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Koerner Racing Engines
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LSM System Engineering
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Manton
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Performance Forged Products
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Procomp Electronics
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PRW Industries
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Scorpion Performance
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Smith Brothers
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Stage V
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T&D
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ban purpose-designed pushrod race engines from competing. If it hadn’t taken such measures, it was inevitable that the quad-cam OHC engines would need to be granted a substantial performance boost, or engine manufacturers would have had to design engines to compete with the 500I at Indy.

Summary
The pushrod valvetrain presents a number of difficulties to the valvetrain development engineer when compared to an OHC system. While it can appear arcane to an engineer, overcoming its inherent disadvantages offers an interesting engineering challenge. Almost 20 years ago, Ilmor showed what can be done within an open set of rules. Most other competition is more strictly governed, but there are still significant opportunities for improvement, as shown by the relatively recent trends towards stiffer components throughout the valvetrain.

References
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