This is a Special Investigation into valve spring design for race engines. The authors are **Gordon P. Blair, CBE, FREng** of Prof. Blair & Associates, **Charles D. McCartan, MEng, PhD** of the Queen's University Belfast and **W. Melvin Cahoon, BSc** of Volvo Penta of the Americas

Uncoiling mysteries

This trilogy on the design of wire coil valve springs is meant to fill something of a vacuum on this topic in the international literature. It may well be that the modern manufacturer of such springs has distanced himself from his blacksmith origins and is now fully armed with computer design software which is either of his own creation or is the FEA outpourings as supplied by such as ANSYS [1]. If this latter assumption be true then these spring manufacturers are keeping their design 'cards very close to their chest' and are neither writing technical papers on the subject nor even posting their expertise on their websites. Even the latest textbooks written on the subject of 'engine valvetrains' provide no real design information for the reader and Wang [2], in a book of some 600 pages containing many hundreds of references, can only muster two with respect to valve springs and even those are dated from the fifties. We are not criticising Wang, merely using his book as an up-to-date example of the paucity of published information on the design of valve springs.

Many of the written texts on valve springs were produced some thirty or more years ago, i.e., before the advent of either computers and computer software, progressive wound springs, tapered springs, or springs wound with ovate wire. The classic book by Wahl [3] is a perfect example of this statement containing much fine theorising with assumptions that permitted a solution outcome using a slide rule and then only for simple 'parallel round wire' springs.

The paper by Ramezani and Shahriari [9] is stimulating from our standpoint. This paper is about the modelling of progressive coil springs for suspension units for vehicles, not the valvetrain of the engine, yet their theoretical approach, and their rationale for using their particular theoretical approach, is somewhat similar to ours. The presentation of experimental confirmation of their model is limited but helpful.

We have added three further papers below [10-12] at the end of the Reference section, as interesting background reading. While these authors claim to model valve springs, even progressive wound valve springs, none of these papers presents any experimental evidence of successfully calculating valve spring properties, such as load, stiffness, etc., and comparing that data with measurements.
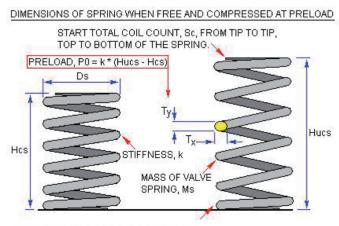
THE FUNDAMENTALS

It must also be remembered that the manufacture of valve springs is not a precision process in the same genre as, say, seven-axis CNC machining. It may be a slight overstatement to say that no two valve springs are ever identical, in terms of load and stiffness as a function of deflection, even if the manufacturer intended them to be. A spring is formed by winding a high-strength steel wire (hot or cold) on to a die and into the profile of a helical coil, which is then heat-treated, tempered, etc., and the end (or dead) coils closed to form a top/bottom base. These 'dead coils' are then ground flat to a set height. This rather brutal procedure is unlikely to produce physically identical specimens. Close, as is often heard said, but no cigar. What has improved dramatically in recent years is the quality and purity of the steel (normally Cr-Si) used to wind springs and this has lead to greatly improved reliability in these components in racing engines.

THE TRILOGY OF PAPERS ON VALVE SPRING DESIGN

In this first paper, Part One, we examine in detail the design of five springs: (a) the inner and outer springs for the intake valve of a NASCAR Cup engine; (b) the single intake valve spring from a large capacity V8 inboard marine unit; and (c) the inner and outer valve springs from a motorcycle engine. The springs (a) and (b) were

Fig.2 Information on basic valve spring geometry

Fig.1 Test springs; HM outer and inner, MM, KW outer and inner

SPRING TYPE	HM outer PR	HM inner PO	KW outer PR	KW inner PR	MM PR
Sc	5.5	7.625	6	7.75	6.875
Ms (measured)	87.9	53	42.5	20.1	65.9
Hucs	58.1	59.85	39.5	35.5	51
Hcs	?	?	?	?	?
Ds	38.5	27.5	29.3	20.8	31.18
Ts (Ty if ovate)	5.18	3.65	3.85	2.8	4.48
Tx (if ovate)		4.44			
k (measured)	56	36.5	33	17.6	40
Nd	1	1	1	1	1

TR = tapered round wire TO =tapered ovate wire
PR=progressive round wire PO= progressive ovate wire

Fig.3 Basic valve spring geometry for the five test springs

probably designed to be parallel, i.e., with equal pitch spacing of the coils but their real stiffness characteristics can only be theoretically determined if they are treated as being progressive springs with unequal coil pitch spacing. The springs (c) are genuine progressive springs and are employed here to highlight the difference in stiffness characteristics with the 'wannabee' parallel springs (a) and (b).

In the second paper, Part Two, we examine in detail the design of three tapered springs; (a) and (b) round wire springs from two (speedway racing) motorcycle engines and (c) an ovate wire spring from a large capacity vee-twin motorcycle power unit.

In the third paper, Part Three, we examine in detail the design of four round wire progressive springs; the inner and outer intake valve springs from an automobile engine and (b) the single intake and exhaust valve springs from a five-valve motocross racing engine.

There are twelve springs in total making up this three-part investigation and they cover all examples of modern spring design from low to high speed engines, with (supposedly) parallel, progressive and tapered springs, and springs wound with either ovate or round wire. All springs are measured from free height to near coil bind for their load-deflection and stiffness-deflection characteristics. Also, all springs are measured physically and the geometry-based data are computed for their load-deflection and stiffness-deflection characteristics [4]. Some of the springs are modelled in FEA software [1] for these same data values. In all twelve cases, the measured and computed data are compared numerically and graphically and the physical geometry of every spring is numerically presented so that others, e.g., designers of valve springs, maybe including some makers of valve springs, can compare their theories with our measurements.

DATA FACTS

Throughout these three papers the mechanical properties of Cr-Si wire for valve springs are assumed to be:

Shear modulus = 77.2 GN/m2
Young's modulus = 203.4 GN/m2
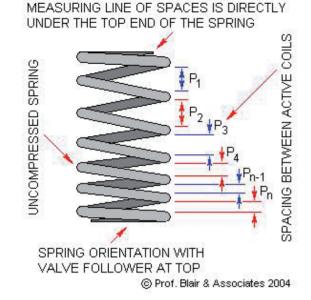Poisson's Ratio = 0.29
Density = 7833 kg/m3

MEASURING LINE OF SPACES IS DIRECTLY UNDER THE TOP END OF THE SPRING

UNCOMPRESSED SPRING

SPACING BETWEEN ACTIVE COILS

SPRING ORIENTATION WITH VALVE FOLLOWER AT TOP

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Fig.4 Information on valve spring coil spacing

SPRING TYPE	HM outer PR	HM inner PO	KW outer PR	KW inner PR	MM PR
coil space P1	8	5.15	5.2	3	3.81
coil space P2	8.75	5.56	5.3	3	4.89
coil space P3	8.7	5.84	4.7	3	4.86
coil space P4	5.3	5.84	1.3	3	4.9
coil space P5		5.92		2	3.5
coil space P6		4.12		0.8	

Fig.5 Valve spring coil spacing for the five test springs

THE VALVE SPRINGS (PART 1)

In Fig.1 is a photograph of the five valve springs. From left to right are the NASCAR Cup springs labelled here as 'HM outer' and 'HM inner'; the large V8 inboard engine intake valve spring labelled as MM; and the (Kawasaki) motorcycle engine intake valve springs labelled as 'KW outer' and 'KW inner'.

In Fig.2 is the software [4] information page explaining the data symbols for the basic geometry of a spring and in Fig.3 are the actual data values for the five springs in question. You should note that one spring (HM inner) is made with ovate wire whereas the other four are wound with round wire. It is fairly obvious from Fig.1 that the pitch spacing of the coils for 'HM outer and inner' are almost equal, and were intended to be 'parallel' (equal coil pitch spacing) spring designs, whereas the other three springs have decidedly unequal coil pitch spacing and are therefore deemed to be progressive springs. What is not visually obvious from Fig.1, or indeed obvious even with the spring in one's hand, is that the wire for 'HM inner' is ovate; those who analyse valve spring designs should take careful note of this potential geometrical pitfall!

In Fig.4 is the information page explaining the data symbols for the pitch spacing of spring coils and in Fig.5 are the actual data values for the five springs in question. Throughout the paper and on all Figures, the units of load are Newtons (N), length is in millimetres (mm), spring stiffness is in N/mm, and mass is in grams (g).

It should be noted that the pitch spacing of the HM springs are indeed almost identical. They were clearly intended or designed to be even- ▶
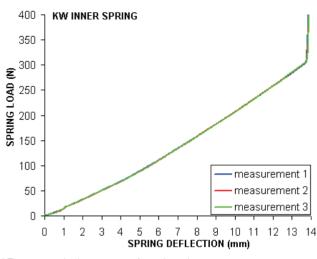
Fig.6 Three separate load measurements for a valve spring

Fig.7 Three separate stiffness measurements for a valve spring

spaced parallel spring coils but the manufacturing process precluded that precision. For the 'HM outer' such 'parallel' coil spaces would have been each 9 and finally 4.5; for the 'HM inner' the coil spaces would have been each 5.74 and the bottom space 4.3. If you compare this data with the first two columns of Fig.5 you can see the differences to be only some tenths of a millimetre. However, later in this paper it will be found that these apparently minor differences are significant should the valve lift profile extend the spring to near coil bind.

MEASUREMENT OF THE VALVE SPRING LOAD AND DEFLECTION

Each spring is installed on a Lloyds tensile/compression test machine and its load-deflection characteristics measured for 1000 steps from its free height until coil bind. The measurement process is both accurate and detailed. The load-deflection characteristics are almost identical for repeated measurements, as shown in Fig.6 for the 'KW inner' spring. The numerical differentiation of the load-deflection data yields the stiffness-deflection characteristics and the same three measurements of Fig.6 are so derived and plotted in Fig.7.

The repeatability of the load-deflection and stiffness-deflection graphs is quite remarkable and, although not shown here, all the data for all of the twelve test springs could be produced to show that such repeatability is quite universal. What is more remarkable is the profile of the stiffness graph in Fig.7. It goes up in a series of steps which are not visible in the load graph in Fig.6. The 'KW inner' is a progressive spring so its stiffness is expected to rise with deflection but, prior to such measurements, it was expected to do so 'smoothly' as the helix is presumed to wrap itself smoothly, coil upon coil, on top of the bottom dead coil. Not so, as will become clear as we examine the experimental evidence of the remaining springs.

A FEA COMPUTATION FOR SPRING DEFLECTION UNDER LOAD

We suppose the obvious answer, in this twenty first century to the question of the selection of the computation method for the load-deflection characteristics of springs is to employ one of the many available FEA packages. In this case, we modelled some of our test
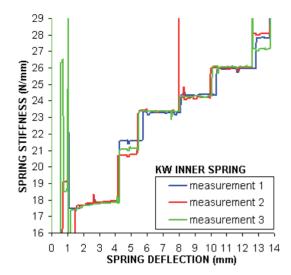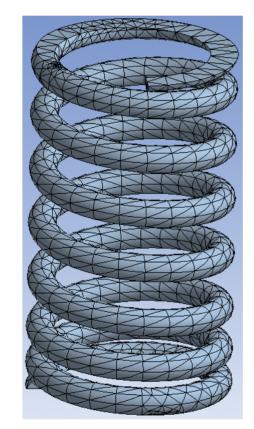

Fig.8 ANSYS FEA mesh model of the 'KW inner' spring

springs in the ANSYS [1] FEA software, including the 'KW inner' spring. The model was set up using the data from Figs.3 and 5. The computational mesh for it is shown in Fig.8. Depending upon the number of elements involved, in this case with some 4150 elements for the entire spring, the computation moves in about 0.1 mm deflection steps from free height to near coil bind. For the 'KW inner' spring this takes some 1.86 hours on a 3.0 GHz single processor running Ansys Workbench v11.0 on a Windows XP 64 bit system.

The result of the computation for the stiffness-deflection characteristics is given in Fig.9 and compared with measured data. The error from calculation to measurement is consistently about 20% over the range of deflection. The mass of the spring is satisfactorily computed by ANSYS at 19.5 g, whereas the measured value is 20.1 g (see Fig.3). ▶
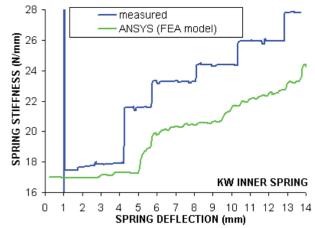
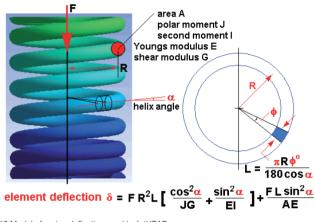Fig.9 Measured and FEA computed stiffness of 'KW inner'

element deflection $\delta = F R^2 L \left[\dfrac{\cos^2\alpha}{JG} + \dfrac{\sin^2\alpha}{EI} \right] + \dfrac{F L \sin^2\alpha}{AE}$

Fig.10 Model of spring deflection used in 4stHEAD

A MATHEMATICAL MODEL FOR SPRING DEFLECTION UNDER LOAD

The less obvious answer, to this same question, is to write one's own code [4] for the load-deflection characteristics of a valve spring. It becomes a truly logical approach if this computational procedure is destined to be directly linked to a sophisticated calculation of the dynamics of the entire valvetrain.

The fundamental analysis of the deflection of an element of a valve spring is shown in Fig.10. In our analysis in the 4stHEAD software [4] we use an element angle of 5 degrees, giving 72 elements per coil. The theory for the bending, torsion and compression of an element of a helical spring, resulting in its deflection due to a force as shown on Fig.10, is based on theory described by Benham et al [5].

In Fig.11 is shown a snapshot of the computation in action for a rather extreme example of a progressive spring where the top, middle and bottom coil spaces are equal but smaller than the rest. The result, during spring deflection, is that the top, bottom and middle spaces at some point are reduced to zero and these elements of the spring are 'trapped' with a zero deflection. The complete spring model is shown at the left with 9 coils total and, with 72 elements per coil, contains 648 computation elements.

The computation proceeds, from a free spring position at zero force, in increasing force increments to give deflection steps of some 0.05 mm. Any top or bottom active coil element becoming trapped by a dead coil is then drawn in 'red' and any central active coil element becoming similarly trapped is plotted in 'blue'. All such trapped elements have, by definition, a deflection of zero and automatically increase the stiffness of the spring, i.e., the load has
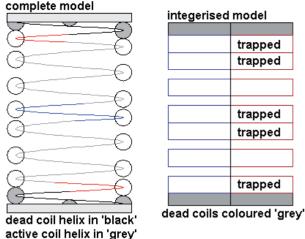

Fig.11 Screen snapshot of both 4stHEAD computation models

risen incrementally but the deflection has not.

While this complete spring model can be operated in a stand-alone manner, it is also automatically linked to the valvetrain computation where it is necessary to create an integerised model of the coil spring. In Fig.11 at the right, is the integerised model of this same 'extreme' progressive spring and the top, bottom and middle coils can be observed to be trapped in a similar manner to the complete model at the left.

The issue, as far as accuracy of spring modelling is concerned is (a) how close can the complete 4stHEAD spring model predict the measured data, (b) how close is the integerised 4stHEAD spring model to either the measured data or the complete theoretical model, and (c) how closely can the ANSYS (FEA) spring model predict the measured data.

For the Kawasaki inner spring, 'KW inner' we have seen the correspondence for criterion (c) in Fig.9. If we add the complete model analysis to Fig.9 and plot that in Fig.12, we get an answer for criterion (a) and if we add the integerised model analysis to Fig.12 and plot that in Fig.13, we get an answer for criterion (b). To complete the analysis for the Kawasaki valve springs, the (complete and integerised 4stHEAD) analysis for 'KW outer' is given in Fig.14; an ANSYS computation was not conducted for this spring. The basic conclusions are that the complete and integerised 4stHEAD models of the Kawasaki springs are a very acceptable mimic of the measured data, at least as good if not actually superior to the ANSYS model, and the integerised models of the springs can be used with some reasonable confidence in any dynamic analysis of the entire Kawasaki valvetrain.

In this context, in Fig.15 are shown snapshots of the 'KW inner' spring at 7 mm deflection during the static modelling (at the left) and during dynamic modelling (at the right) of the entire Kawasaki valvetrain at 12,000 (engine) rpm [6,7]. The dynamic model (at the right) shows the integerised spring snapshot drawn in two halves where the left half shows the static coil positions, which are identical to the static model at the left, but the right half shows the inertia due to the mass of each coil delaying the lift of each coil.

From a user standpoint, spring design and analysis can proceed apace, because the speed of computation of the complete and integerised 4stHEAD model is much faster than the ANSYS model as the former takes only a few seconds to complete rather than a few hours for the ANSYS. The further advantage is that the full valvetrain dynamics analysis, to fully investigate the implementation of each design iteration of the valve springs, is only a mouse-click away.
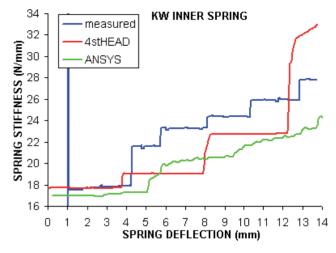
Fig.12 Measured and computed stiffness by FEA and 4stHEAD

Fig.13 Stiffness modelling by FEA and both 4stHEAD models

STATIC ANALYSIS OF THE NASCAR VALVE SPRINGS

The two valve springs for the NASCAR Cup engine are shown in Fig.1 and the input geometry data are given in Figs.3 and 5. The measured data for load and stiffness of the 'HM outer and inner' springs are plotted in Figs.16 to 18 (blue line). The computation by 4stHEAD of each spring as a progressive spring is shown on all Figures (red line) and on Figs.17 and 18 as a parallel spring (cyan line). There is a satisfactory correlation between the measured load and stiffness and their calculations.

It has already been observed that the coil spacing of both springs is quite even, as is clear from the Fig.1 picture and the Fig.5 data. The perfect 'parallel HM outer' spring would have equal coil spacings of 9 and lastly 4.5, and similarly 5.74 and 4.3 for the 'HM inner' spring. One can observe, in Fig.5, just how numerically close these springs are to being 'parallel'. One suspects that the design was intended to be for 'parallel' springs and this is as close to that intention as the spring maker could manage.

In the 4stHEAD software, with a single mouse-click, we can declare these springs to be parallel and so induce the 'perfect' coil spacings declared above. The spring stiffness characteristics are then plotted in Figs. 17 and 18 as the cyan lines. It is to be expected that a parallel spring will have almost constant stiffness with deflection and this is what one finds in Figs.17 and 18. However, the stiffness rises by some 10% but only in the last 10% of the spring deflection and the computation using the true coil spacings (red line) is seen to be more accurate than declaring the springs to be parallel (cyan line). This near constant stiffness for the HM springs contrasts markedly with Figs. 13 and 14 for a true progressive spring where the stiffness virtually doubles over the full spring deflection.

The rise in spring stiffness in the latter stages of spring deflection can cause problems for the designer. In a V8 NASCAR engine the valvetrain is very highly stressed and the last straw is to create variable valve lofting caused by variable spring stiffness at maximum spring deflection, i.e., at maximum valve lift. A typical NASCAR engine will have an intake valve lift of some 20 mm and a preload of some 6 or 7 mm, giving a possible spring deflection of 27 mm. You will note,
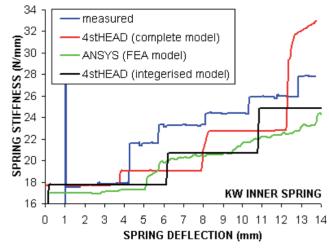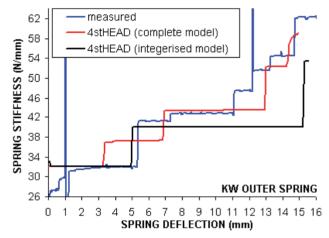

Fig.14 Measured and 4stHEAD computed stiffness of 'KW outer'

from Figs.17 and 18, that this is exactly where the 'progressive' spring calculations predict a rise in stiffness and, depending on how the spring maker produces springs, one can expect that the variation of their characteristics will lie somewhere between the red and cyan lines on Figs. 17 and 18. This is unsatisfactory from the critical viewpoint of engine durability and perhaps one can find a NASCAR engine builder testing batches of valve springs in order to select those springs which have equal load characteristics at maximum deflection.

You will also observe in Figs.17 and 18 that the measured spring stiffness for the first 5 or 6 mm is some 10% below the (almost constant) spring stiffness from about 8 to 20 mm deflection. If one looks closely at Fig.1 for the HM springs, you can see that the end of the dead coil tang is not resting on the first (or last) active coil, as is the case with the other springs there. In short, the 'dead' coil is not truly 'dead' and softens the actual spring until the dead coil becomes closed after about 6 or 7 mm of spring deflection. This is an unusual design feature and, while the preload deflection is normally more extensive than this 'soft dead coil and tang closing' period, it is another potential troublesome variable for the preload setting that an engine builder does not need.

You can see that the 'HM inner' spring is wound with ovate wire, 3.65 mm by 4.44 mm in the radial direction. The equivalent round wire diameter, for the same spring mass, is 4.05 mm. If this spring is computed as a round wire spring with all other data retained as ▶
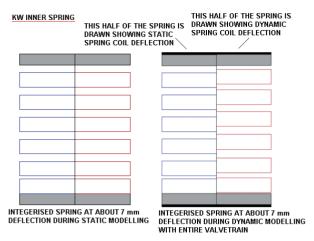
Fig.15 Static and dynamic modelling by 4stHEAD of 'KW inner'

Fig.16 Measured and 4stHEAD computed loads of HM springs

Fig.17 Measured and computed stiffness of 'HM outer' spring

unchanged, the ensuing spring mass and stiffness are also virtually unchanged (35 N/mm and 54.8 g) but the deflection of the spring from its free height to coil bind is reduced from 32.73 mm to 30.08 mm. The use of ovate wire for the (normally less stiff) inner spring gives greater design flexibility without sacrifice of spring deflection before coil binding; this is often a real valve spring design issue for those engines with high valve lift characteristics.

STATIC ANALYSIS OF THE INBOARD MARINE ENGINE VALVE SPRING

This is a single intake valve spring, it is shown in Fig.1 and, as the middle spring of the five shown, is labelled as MM. Its geometrical details are given in Figs. 3 and 5. Its measured stiffness data is shown in Fig.19 along with its computation by the 4stHEAD software not only as a progressive spring but also as a parallel spring with equal coil spacings of 4.2 mm. In Fig.5, the three actual coil spacings in the middle of the spring are basically 4.9 mm and the two end spacings are basically equal at some 3.65 mm.

History does not record if the spring was designed deliberately with this curious form of progression or if it was designed instead as a parallel spring but an imperfect manufacturing process to close the dead coils at each end influenced the spacing of the top and bottom active coils. In any event, it is clear that the progression of the spring stiffness is more pronounced than the NASCAR 'parallel' springs and the computation of the spring as progressive (red line) better fits the measured data (blue line) than that computed as if parallel (cyan line). The increase of spring stiffness from free height to coil bind is some 50%.

In Fig.20 is a snapshot from the 4stHEAD model of the MM computed as a progressive spring. The model has been halted at 16 mm deflection and the trapped elements of the helix, shown in red, illustrate that the increase of stiffness at 16 mm deflection, be it measured or computed data, is caused by the top and bottom coils becoming trapped by the dead coils. In short, the lesser coil spacing at top and bottom of the spring is taken up before the more widely-spaced middle coils become trapped.

We are very familiar with the entire MM valvetrain and can confirm that the valve spring preload setting is 7.6 mm and the maximum valve lift is 12 mm. Hence the total valve spring deflection is 19.6 mm at maximum valve lift which means that the valve spring stiffness (see
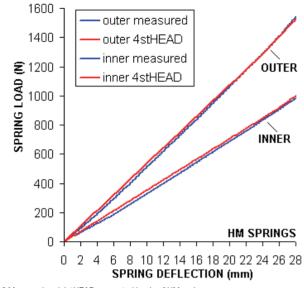
Fig.19) is some 60 N/mm, i.e. some 50% higher than at preload level, to inhibit valve lofting. This engine uses hydraulic tappets where valve lofting and valvetrain component separation is an absolute design 'no-no' because hydraulic tappet 'pump-up' will occur. This form of valve spring progression is a preferred design alternative to using a parallel spring with the same peak lift stiffness because the cam-tappet forces and valvetrain friction are 50% lower at the preload level. Whether the MM engine designer and his spring maker knew all of this design information at the design stage, or arrived at this satisfactory conclusion as the result of continuous experimentation over the years, is not known.

THE DESIGN IMPLICATIONS OF THE BASIC SPRING THEORY

The Fig.10, or reference [5], shows the basic spring theory. For the benefit of readers who have not previously thought about spring theory too deeply, an examination of the equation given there for the deflection of a spring element yields some useful design conclusions.

1. The more spring elements there are, with all terms on the right-hand side held constant, then the greater is the deflection (d) of the spring for a given force F and the lesser (softer) is the spring stiffness. In ▶
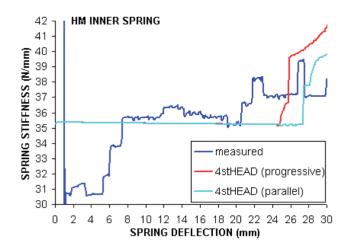
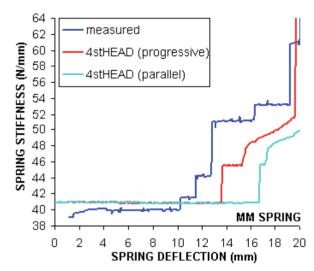Fig.18 Measured and computed stiffness of 'HM inner' spring

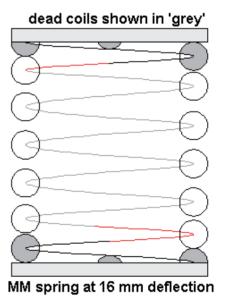Fig.19 Measured and computed stiffness of MM spring

Fig.20 4stHEAD model of MM spring at 16 mm deflection

COMPUTATION BY 4stHEAD					
SPRING	HM outer	HM inner	KW outer	KW inner	MM
TYPE	PR	PO	PR	PR	PR
Ms (measured)	87.9	53	42.5	20	65.9
Ms (calculated)	89.4	53.5	41.5	20.1	67.9
k (measured)	56	36.5	33	17.6	40
k (calculated)	53.9	35.4	32.1	17.7	40.85
DEF to bind (meas)	30.6	33.05	17.1	14.5	22.4
DEF to bind (calc)	30.75	32.73	16.5	14.8	21.96

Fig.21 Measured and computed basic data for the five test springs

4. As can be seen in Fig.10, in a progressive spring the helix angle normally tends to get smaller towards the bottom of the spring. As the cosine of the helix angle is the dominant term in the equation, it means that the stiffness of the bottom coils of a spring are normally (slightly) lower than at the top.

5. The conclusion is that, as all terms on the right-hand side are variables for each element of a spring, the 4stHEAD software must track the numeric value of each one at every level of deflection for every conceivable type of spring, spring geometry, wire geometry, and the material properties of that wire.

BASIC DESIGN DATA FROM THE COMPUTATIONS

It is possible today to theoretically model the load and stiffness characteristics of the helical spring typically used in engine valvetrains, not only with some reasonable degree of accuracy but also reasonably quickly on a desktop PC, using software [4]. Apart from the many graphical correlations we have presented of the measured and computed load and stiffness characteristics of the five springs of Fig.1, using only their physical geometry as given in Figs.3 and 5, further useful design data is made available through 4stHEAD and is tabulated in Fig.21. Here is compared the measured and calculated spring mass, stiffness at zero load, and the spring deflection from zero load to coil bind. It can be seen that the correlation is very good for these basic design parameters.

As the answer computed by FEA software ANSYS [1] for the mass of the 'KW inner spring' is 19.5 g, and 16.95 N/mm for the stiffness, and it took 1.86 hours to calculate it by comparison with 20 seconds for 4stHEAD [4], at this stage of our report the computational theory in 4stHEAD would appear to be somewhat more effective for valvetrain design. However, in later Parts of our trilogy on spring design we explain why neither ANSYS nor the 4stHEAD software precisely mimics the measured spring stiffness data.

The computation by 4stHEAD [4] rapidly yields the natural frequency of the spring as it is deflected under load. This is a more difficult and time-consuming computation for FEA software. The 4stHEAD software automatically tracks at each incremental spring deflection the number of spring elements that are still active (can be deflected) and those that are bound or trapped (cannot be deflected). Thus the software memory retains the mass of active coils Ma (kg) at each increment of spring deflection as well as load, stiffness k (N/m) and even the stress (s) in the wire [5,8]. The natural frequency f (Hz) at any computation point is then given by:

$$\text{natural frequency} \quad f = \frac{1}{2}\sqrt{\frac{k}{M_a}}$$

For the five test springs, their natural frequency characteristics are plotted in Fig.22 and their stress levels graphed in Fig.23.

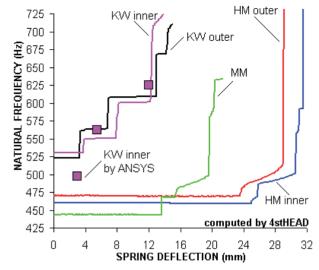In Fig.22 the rapidly rising natural frequency values as the spring

short, the longer the wire in the entire spring, the less stiff it will be.

2. The greater is the mean radius (R) of the spring then the greater is the deflection (d) of the spring for a given force F and the lesser (softer) is the spring stiffness.

3. As all terms which contain the 'diameter', or area (A) of the wire, such as the polar moment (J) and the second moment of area (I), are all listed on the denominator of the equation, this leads to the rather obvious conclusion that the larger is the spring wire 'diameter' then the lesser is the deflection and the stiffer is the spring.

Fig.22 Natural frequencies of the five test springs

Fig.23 Stress levels for the five test springs

deflection nears coil bind reflect the declining values for the mass of the active coil elements. Consequently, one can logically neglect from technical consideration any natural frequency values in the last millimetre (or so) of spring deflection. The two HM springs have very similar natural frequency levels for most of the valve lift in the range of 440 to 460 Hz. The important design issue here is that one should arrange for the valve lift profile for the HM engine to have low excitation acceleration harmonics in this frequency range. A similar logic applies to the KW springs, and to a lesser extent the MM spring, but with progression these springs exhibit a much broader range of potential excitation frequencies.

Also plotted on Fig.22 are three points for the natural frequency of the 'KW inner' spring as computed by ANSYS; the correspondence with the equivalent data computed by 4stHEAD is quite good with the worst difference being about 6% for the very first point at a 3.5 mm deflection. You may well ask why this ANSYS computation is not conducted over the entire deflection range; it is a separate computation procedure within the ANSYS software and it takes much longer than the others!

In Fig.23 the stress levels exhibited by the NASCAR Cup springs (HM) show values of some 1200 MPa as they approach coil bind. We have already commented above that the valve lift profile for such a NASCAR engine will almost certainly give spring deflections close to coil bind. A shear stress level of 1200 MPa is approaching the normal durability limit for Cr-Si steel springs of some 1250 MPa and illustrates just how near to the design limits a NASCAR engine designer must operate. The other three springs are clearly designed at lower stress levels below 1000 MPa for greater durability and, doubtless, the need to satisfy commercial/financial requirements to be able to use a lesser quality/purity Cr-Si spring steel. That would definitely not be the case for the HM springs where only the very best Cr-Si steel will permit the NASCAR engine valvetrain to last its 500 punishing full-throttle miles.

CONCLUSION

The use of software [4] to design the valve springs, and to link them directly with a valvetrain dynamics computation, permits the designer not only to analyse existing valve springs for suitability within his engine design but also to create new, more optimised, spring designs to improve the behaviour of his engine's valvetrain. ∎
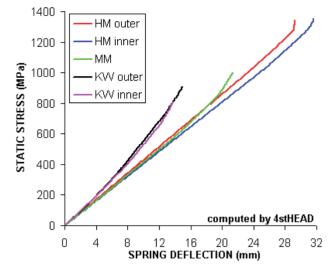
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