Die Entwicklungsalternativen für Grand Prix Motorradmotoren
Design Alternatives for MotoGP Engines

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1. Abstract

From 2002 the FIM (Federation Internationale Motocycliste) sanctioned the use of 990 cm^3, naturally-aspirated, spark-ignition, multi-cylinder engines operating on petrol for racing in the prestige and fastest Grand Prix category, i.e., MotoGP. This 990 cm^3 classification replaced what had been a 500 cm^3 class since the dawn of motorcycle racing and where in recent decades the two-stroke engine of some 3- or 4-cylinders and producing some 200 hp had eliminated all competition from four-stroke engines which would struggle to produce perhaps 150 hp from a 500 cm^3 engine. Consequently, the designers of racing engines had to learn new design techniques and much, if not all, of the theory of tuning learned the hard way on 2-strokes had to be rapidly unlearned. Many companies, such as Honda, Yamaha, or Ducati, to name but three, already competed in World Superbikes where all engines were historically 1000 cm^3 four-stroke twin-cylinder, or 750 cm^3 four-cylinder, or 900 cm^3 three-cylinder units. Within such organisations much experience already existed on designing and developing four-stroke multi-cylinder racing engines. However, in MotoGP, the choice of engine layout is much wider and the selection of the final design layout for MotoGP racing often devolved to personal choice, copying of that which is deemed to be currently successful, or to some long-held but as yet unproven theory. This paper attempts to set out some logic for any mechanical design to be selected for MotoGP racing and, in particular, shows the advantages and disadvantages of designs with two to six cylinders all nominally producing the same power output of some 230 hp at 15000 rpm.

2. The Geometric Layout Design Parameters

In a statement of the obvious, the engine has to fit into a motorcycle and be mated to a (probably six-speed mechanically foot-operated) gearbox. This means that width, centre-of-gravity locations, and weight of the engine are all significant design parameters. Almost without discussion, the engine will be liquid-cooled and also without excessive disagreement will probably have four-valves per cylinder in a pentroof-type combustion chamber. The valve operation will be double-overhead camshafts and the valve follower actuation may be desmodromic, finger followers, but more commonly will be flat direct-acting bucket tappets. The valve control may be desmodromic, but for bucket tappets and finger followers at this juncture in racing engine design this will be done with steel or titanium coil springs and the valves will also be of steel or titanium. The use of pneumatic or gas springs, as used in F1 car racing, may come along eventually to permit engine speeds in excess of 17000 rpm, but that stage would not appear yet to have been reached as 230 hp can be achieved at 15000 rpm in a 990 cm^3 engine and 230 hp is felt to be enough to win a MotoGP race at this time provided that the shape of the back-up torque and power curve gives competitive acceleration out of slow corners.

Consequently, if it is assumed that 15000 rpm is a 'comfortable' maximum engine speed for coil-spring controlled valve actuation and that an engine can run at a mean piston speed of 25 m/s without piston lubrication failure, and that a four-valve pentroof design can breathe air and fuel adequately, and burn it successfully, to produce some 14 bar bmep on gasoline at a 14:1 compression ratio, the following simple deduction can be made as to the basic engine dimensions and performance [1]. The cylinder stroke is 50 mm and the engine will produce 232 hp at 15000 rpm.

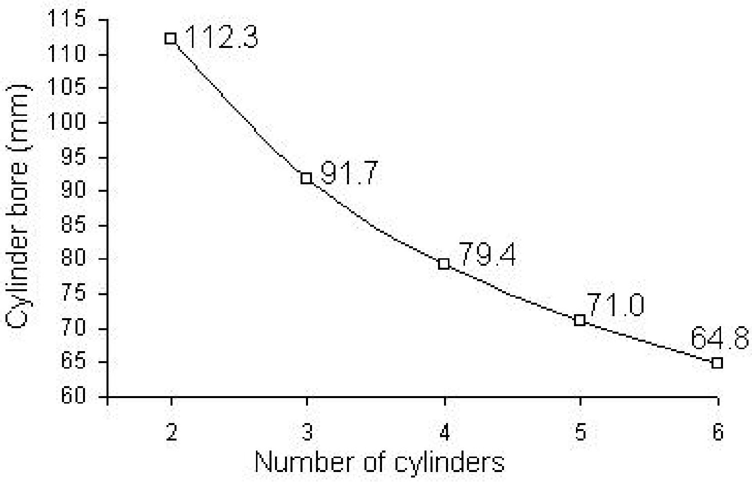

Figure. 1 Cylinder bore sizes for two to six cylinder MotoGP engines.

The bore sizes shown above correspond to bore:stroke ratios of 2.245, 1.836, 1.59, 1.42 and 1.3 for two- to six-cylinder units, respectively. A bore:stroke ratio of 2.245 is typical of a F1 car engine, although there bore sizes are a fraction less than 100 mm. The translation into engine bulk is best illustrated in Figure. 2 where the above dimensions are transposed into a plan scaled view showing in-line and vee formats.

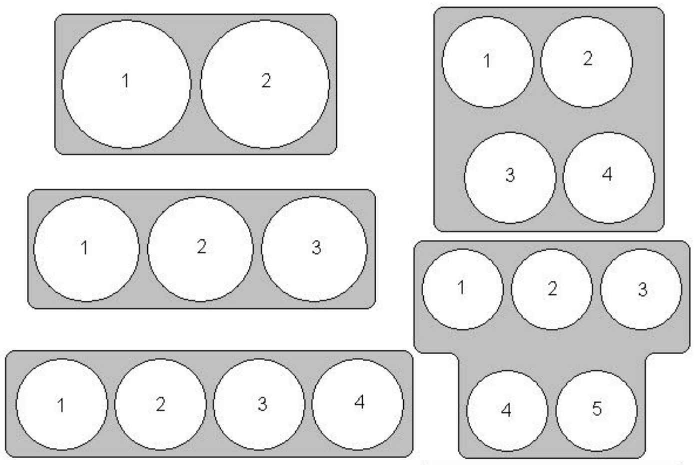

Figure. 2 The bulk of MotoGP engines with two to six cylinders.

Although these simple pictures do not tell the whole truth about bulk and weight, some interesting conclusions can be drawn. The twin-cylinder engine, be it in-line or vee, is compact and simple. The in-line three-cylinder is wider, but not by much, and is certainly narrower than the in-line four (a very popular Japanese format). The vee-four is narrower than the in-line three-cylinder engine and also narrower than the vee five- or vee six-cylinder units. Interestingly, the vee-six is potentially narrower than the vee-five, which latter is currently the engine of fashion in MotoGP. From an exhaust tuning standpoint, collections of exhaust runners from engines firing evenly give the best tuning when there are three or more cylinder exhausts being so collected [1]. This leaves the in-line four cylinder as the number one option, followed by the in-line three and vee-six engines. The vee-four is also in good shape for exhaust tuning if the crankpins are stepped and the cylinders fire at 180 degree intervals. The vee-five is the most awkward to ‘exhaust tune’ as it is imperative that the front three cylinders fire at 240 degree intervals and the rear two cylinders fire at 360 degree intervals. This imposes design constraints on the crankshaft and the ensuing torsional oscillations must be carefully designed out by the almost-inevitable addition of a torsional damper at the non-drive crankshaft end.

3. The Gas Flow and Combustion Design Parameters

The next design step is to determine the physical dimensions of the valves and their location within the cylinder. A four-valve pentroof arrangement will be employed for all cylinders of each engine. The software package used to do this is called 4stHEAD [2]. The outcome is a determination of the valve sizes which can logically fit within a chamber; which chamber volume provides the required compression ratio; the surface area of this chamber is determined for future engine simulation requirements [3]; the valves can be moved on-screen dynamically with piston motion and the valve cut-out depths on the piston crown determined. For all of the engines types analysed here a valve seat plan width of 1 mm, and a piston to head minimum clearance of 0.75 mm, is assumed. The maximum lift of the valves is set at 37% of the inner seat diameter. The upshot of this design process is shown for the two-cylinder engine below. It is drawn to scale and the actual dimensions and lift of each valve are shown on the sketch, Figure. 3 below, for the two-cylinder engine.

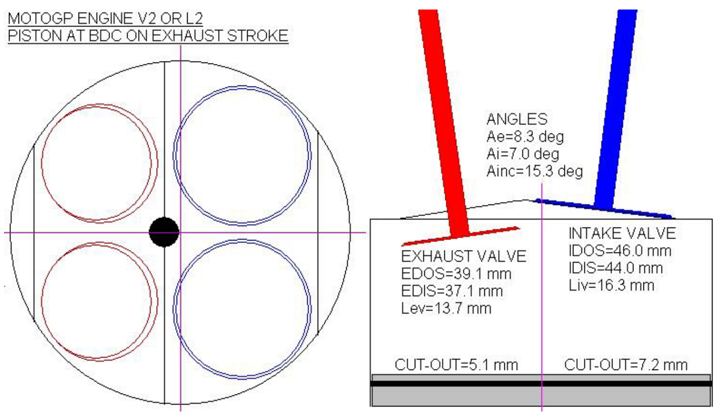

Figure. 3: The two-cylinder engine with the piston at bdc.

The maximum depth of the cut-out in the piston for the outer seat edges of the exhaust and the intake valves are shown on the sketch but the valve face extension beyond those seats must be added to the dimensions shown above so as to determine the full depth of the relevant cut-outs. The angle of inclination of each valve on the pentroof, and the included valve angle, is also illustrated on the sketch. The following Figures, Figures. 4-7, show the equivalent cylinder layouts for engines with up to six cylinders. When examining Figures. 3-7 remember to take into account the optical illusion of the printed Figure; it is the stroke which is common at 50 mm and not the cylinder bore!

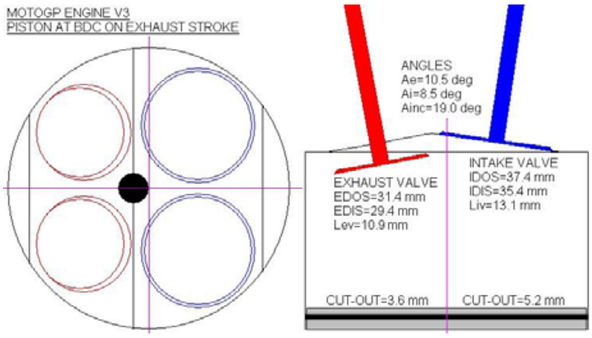

Figure. 4: The three-cylinder engine.

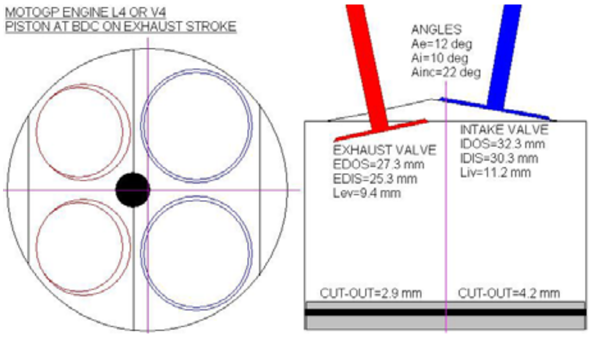

Figure. 5: The four-cylinder engine.

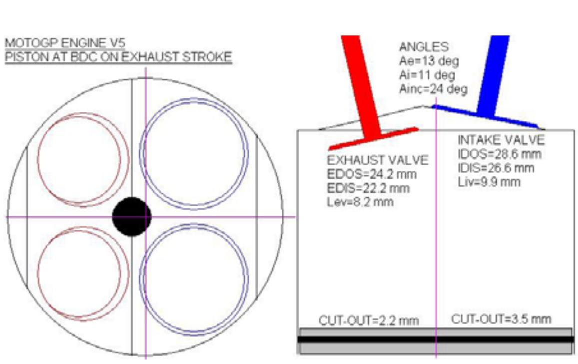

Figure. 6: The five-cylinder engine.

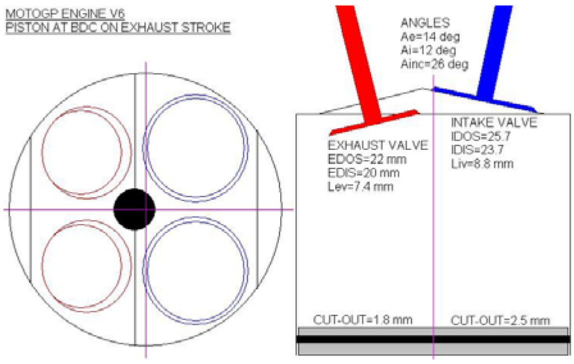

Figure. 7: The six-cylinder engine.

The surface area of the head, the squish band and the piston crown are computed by a fine mesh procedure and calculated as a surface area ratio, K_{sr}, and a surface to volume ratio, K_{sv} [2]. The graphical output for each engine cylinder head is shown here and set very approximately and relatively to size on the printed page.

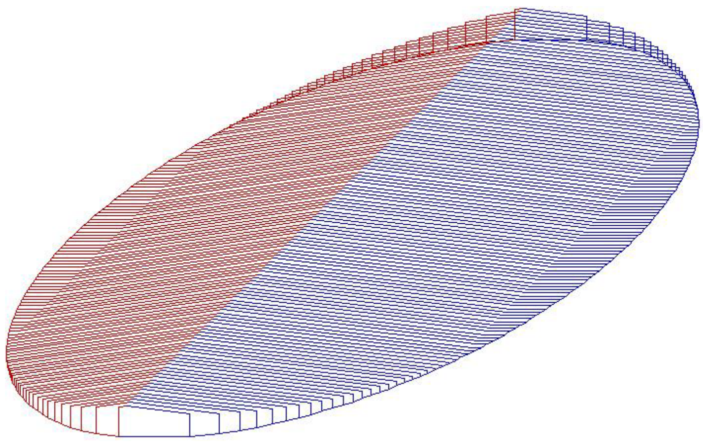

Figure. 8 The cylinder head of the two-cylinder engine.

The surface to volume ratios, K_{sv}, for each of the engines of two to six cylinders is 0.545, 0.554, 0.558, 0.565, and 0.567, respectively, giving an overall shift of 4%

reflecting the increasing included angle of the pentroof with engines having larger cylinder numbers. On the other hand, it is quite clear from these pictures, Figures. 8-12, that combustion in the six-cylinder engine will be much more rapid and efficient by comparison with that in the two-cylinder engine and will most probably be reflected in a shift of ignition advance by as much as 15 deg, and a reduction of burn period by some 10 deg, over that same cylinder range [1]. The in-cylinder tumble characteristics will also be enhanced in engines with decreasing bore:stroke ratios.

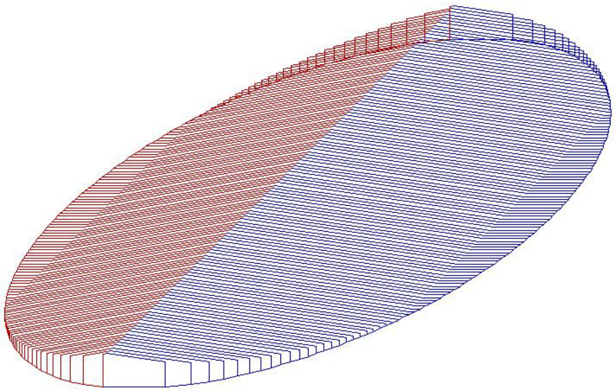

Figure. 9: The three-cylinder head profile.

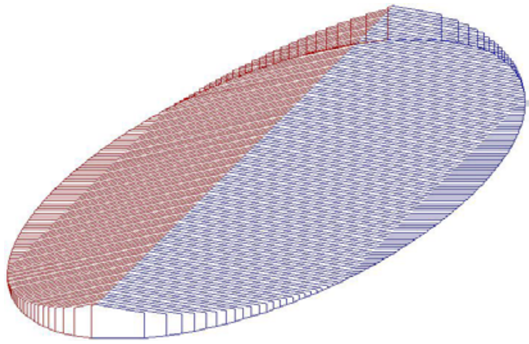

Figure. 10: The four-cylinder head profile.

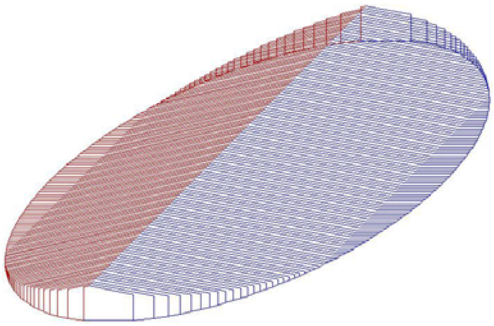

Figure. 11: The five-cylinder head profile.

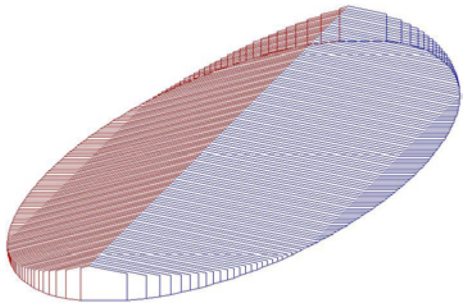

Figure. 12: The six-cylinder head profile.

Good combustion is not aided by excessive valve cut-outs in the piston crown and from Figures. 3-7 it can be seen that the two-cylinder engine has cut-outs of some 5 and 7 mm deep. The depths decrease with the number of engine cylinders with the result that in the six-cylinder engine they are minimal at only 1.8 and 2.5 mm.
The general and approximate conclusion from the above comments on combustion is that under conditions of equal charging efficiency there could be a 10% increase of indicated torque over the range of cylinders from the two- to the six-cylinder engine.

4. The Breathing Design Parameters

The breathing of air into the cylinder, or of exhaust gas out of it, is very much a function of the number of valves, the valve and seat dimensions, the valve lifts and durations, the cylinder dimensions, and the compression ratio. While the valve sizes and lifts are also numerically shown in Figures. 3-7, they are graphed separately below in Figures. 13-14. It can be seen that the valve size, graphed as outer seat diameter EDOS, decreases with number of cylinders and the maximum valve lift, calculated at 37% of the inner seat diameter, EDIS, has the same trend. On Figure.

14, there is also a graph showing exhaust valve lift as a function of 40% of the inner seat diameter and its inclusion will be made clear later in this section of the paper.
All of the valves in all of the cylinders of all of the engines are, as an initial design gambit, assumed to open and close according to the following schedule at a set cam tappet clearance of 0.3 mm; the exhaust valves open, EVO, at 80 deg bbdc and close, EVC, at 50 deg atdc, for a lift duration of 310 deg; the intake valves open, IVO, at 60 deg btdc and close, IVC, at 90 deg atdc for a lift duration of 330 deg.

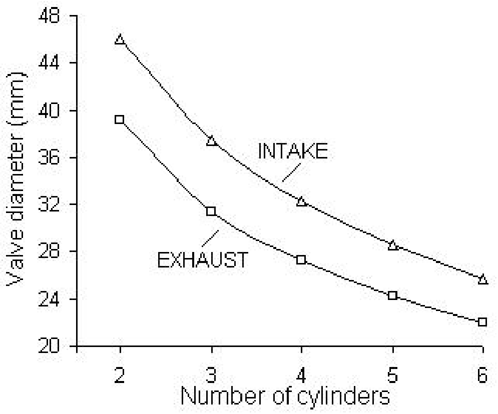

Figure. 13: Valve diameters.

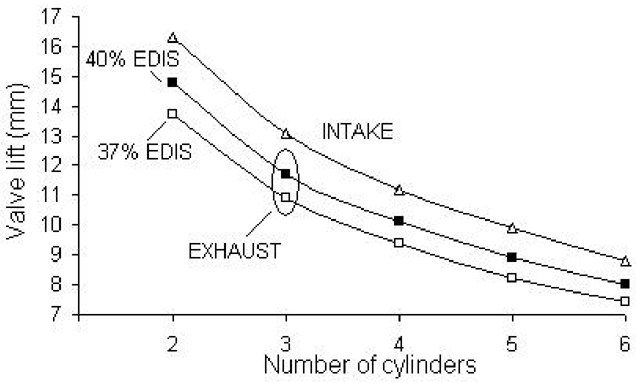

Figure.14: Valve lifts.

The relative intake to exhaust valve size ratio, K_{ie}, is set at between 1.17 and 1.2 for all engines [1, 2]. The valve lift profiles are set to reasonable equality for each engine and the actual profiles for each of the exhaust valves are shown as graphed by the preparation software [2] in Figure. 15 below; that for the intake valves is almost identical but is 10 deg longer in camshaft angle terms. The valve ramp lifts are each 0.3 mm. It will be observed, by those experienced in the art and science of valve lift design, that the profile of the acceleration diagram selected is quite mild in terms of the ensuing profile of the jerk diagram it creates, particularly around the nose of the cam which will provide it to the valve.

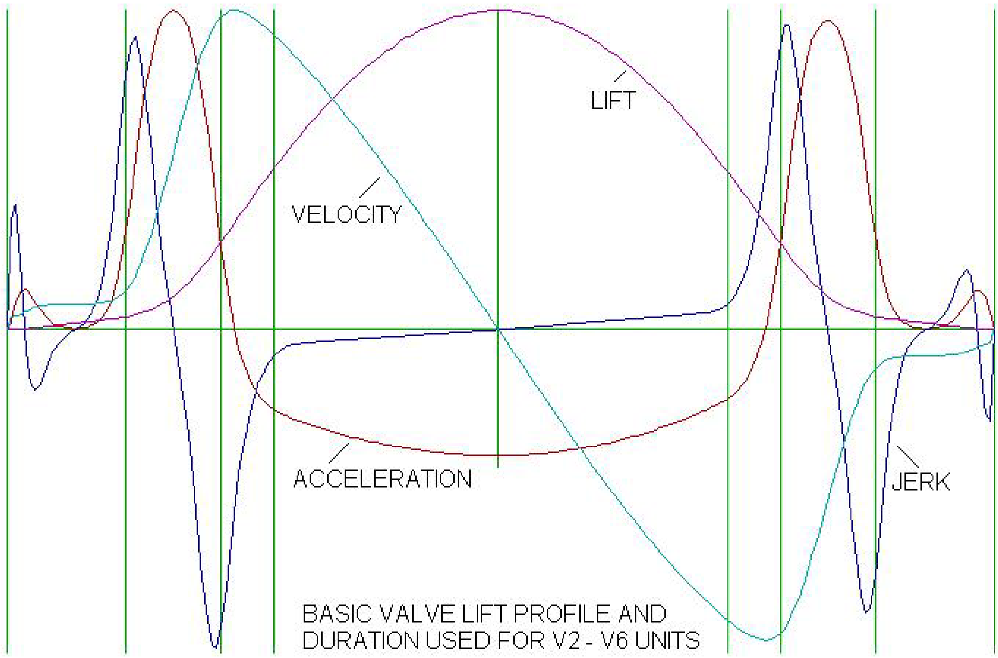

Figure. 15: Valve lift characteristics for all exhaust valves.

To estimate the breathing capabilities of these engines, before inevitably turning to an accurate engine simulation [3] for a definitive answer to the question, the time-areas per unit swept volume have been shown to be a valid and realistic empirical means of conducting such a study [1].
For the exhaust breathing process, the basic elements of time-area for the 'exhaust blowdown' period, the 'exhaust pumping' period, and the 'exhaust overlap' period are shown in Figure. 16 below.

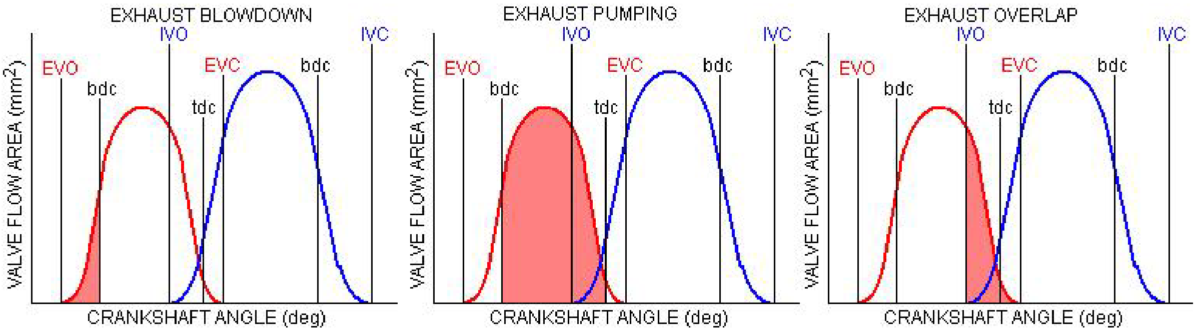

Figure. 16: The exhaust time-areas for blowdown, pumping, and overlap valve periods.

For the intake breathing process, the basic elements of time-area for the 'intake ramming' period, the 'intake pumping' period, and the 'intake overlap' period are shown in Figure. 17 below.

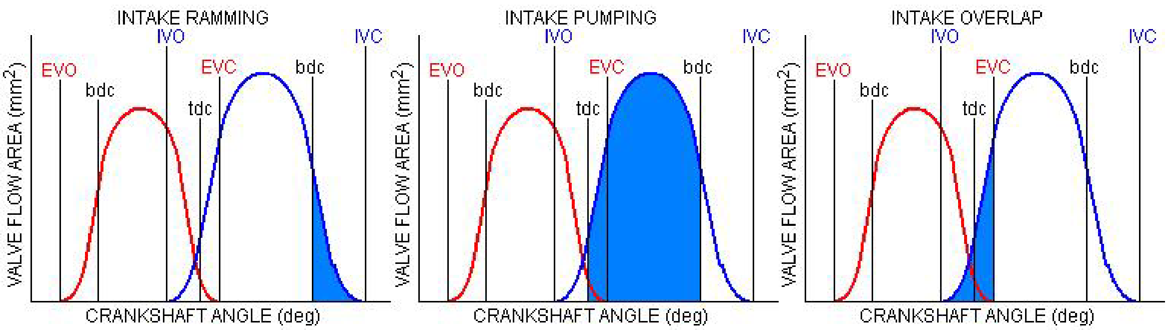

Figure. 17: The intake time-areas for ramming, pumping, and overlap valve periods.

While the sketches, Figures. 16-17, show the relevant total valve areas in mm^2.deg units, they are converted to 'time-area values per unit swept volume', TASV, in s/m units by dividing by SV (the swept volume of any one cylinder) and the relevant ratio of crankshaft angle to absolute time with respect to the crankshaft rotation rate (rpm). It is shown [1] that TASV values are directly related empirically to the 'delivery ratio' proportion of gas which will flow through the valves, to or from the cylinder, during the periods indicated. In short, it is a much more sophisticated means of empirical comparison of engine breathing characteristics than the traditional 'mean gas velocity' or 'mean Mach index' or 'Lovell factor' values seen elsewhere in the literature because each time-area element for each segment of the valve duration for each valve must have a numerical value within a specific numerical range. If this is not the case, the required airflow/gasflow breathing will be stunted within that segment and hence affect the total breathing behaviour through that valve of the engine. Hence, all segmental numerical criteria must be satisfied or the desired torque (as bmep) will be unattainable, or so runs the empirical argument.

4.1 The Intake Valve Breathing Design Parameters

The time-area characteristics at the nominal peak power engine speed of 15000 rpm for all engines for the intake valves are shown in Figure. 18; the plot units are s/m x 10^4. It is clear from discussion elsewhere [1] that these intake time-areas have a strong correlation with airflow delivery ratio and hence bmep at the engine speed selected for peak power.

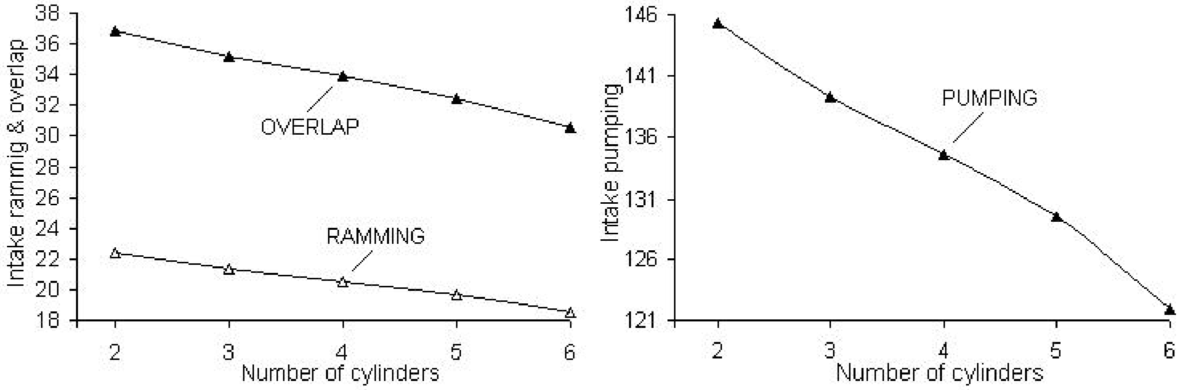

Figure. 18: The intake valve time-areas for 'ramming' , 'pumping' and 'overlap'.

The design criterion for all of the 990 cm^3 engines is for a bmep of 14 bar and examination of the numerical criteria published [1] shows that the relevant values of intake time-areas required for this torque output at peak power are 19 (+/- 1.0), 128 (+/- 5.0), and 39 (+/- 2.0) s/m for the intake 'ramming', 'pumping', and 'overlap' valve periods', respectively (again quoting in the more convenient units x 10^4). Examination of the graphs in Figure. 18 shows that the two-, three-, and four-cylinder engines meet or exceed these time-area criteria but that the five- and six-cylinder engines are more marginally acceptable. The engines with less than five cylinders could be easily re-optimised, with slightly lesser valve lift, or period duration to precisely meet the criteria, and vice-versa for the engines with five and six cylinders. However, it should be remembered that this is empiricism and such decisions of fine detail are best left to optimisation within an accurate engine simulation program [3]. Nevertheless, and in a reversal of the conclusions of the combustion discussion, the basic trend for superior intake valve breathing is for the engines with fewer cylinders, or, perhaps to re-state it more technically, for engines with the greater bore:stroke ratios.

4.1 The Exhaust Valve Breathing Design Parameters

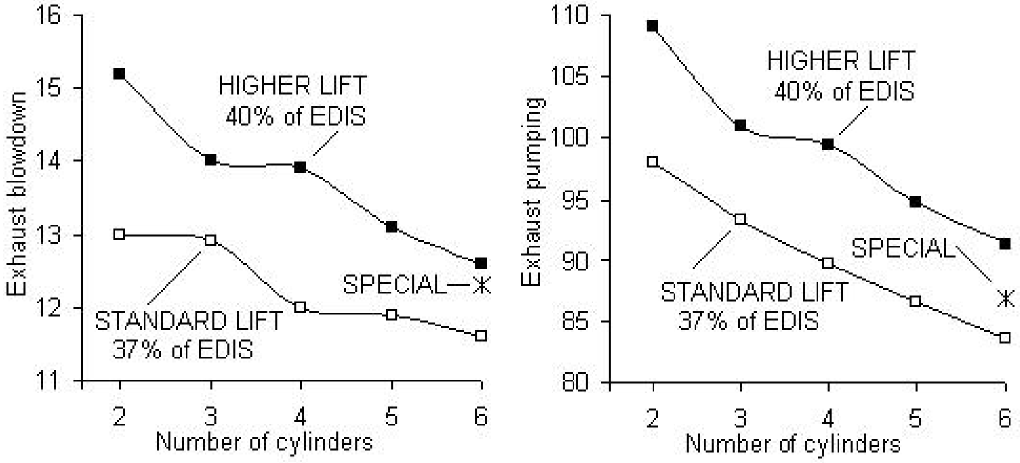

Figure. 19 (a) & (b): The exhaust valve time-areas for 'blowdown' and 'pumping'.

The time-area characteristics at the nominal peak power engine speed of 15000 rpm for all engines for the exhaust valves are shown in Figures. 19 (a-c); the plot units are s/m x 10^4. The design criterion for all of the 990 cm^3 engines is for a bmep of 14 bar and examination of the numerical criteria published [1] shows that the relevant values of exhaust time-areas required for such a torque output at peak power are 15.5 (+/- 1.5), 100 (+/- 5.0), and 31 (+/- 3.0) s/m for the exhaust 'blowdown', 'pumping', and 'overlap' valve periods', respectively (again quoting in the more convenient units x 10^4).

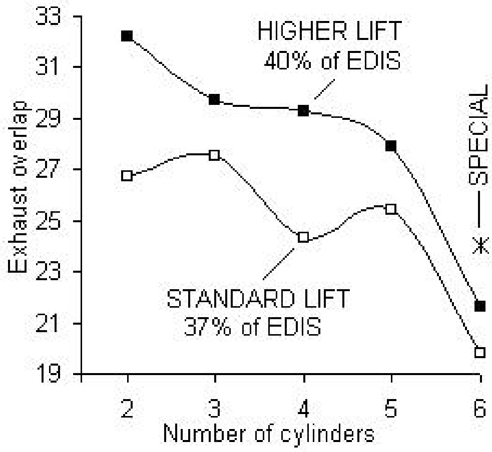

Figure. 19 (c): The exhaust valve time-areas for the 'overlap' valve period.

Examination of the graphs in Figure. 19 (a-c) shows that the two-, three-, and four-cylinder engines exhibit the same general trend as with the intake valve. When the valve lift is at the 'standard' value of 37% of the inner seat diameter, EDIS, and the valve lift profile is that shown in Figure. 15, then only the engines with the greater bore:stroke ratios approach the required empirical flow criteria. In particular, the six-cylinder engine is deficient in valve flow area. To redress this, new time-area criteria are calculated when the exhaust valve is lifted higher, to the same lift profile and duration as in Figure. 15, but now to a maximum value of 40% of the valve inner seat diameter. The outcome is also graphed in Figure. 19 (a-c) and it can be seen that this improves the situation considerably towards the acceptable for all engines, with the exception of the 'overlap' time-area in Figure. 19 (c) for the six-cylinder engine.

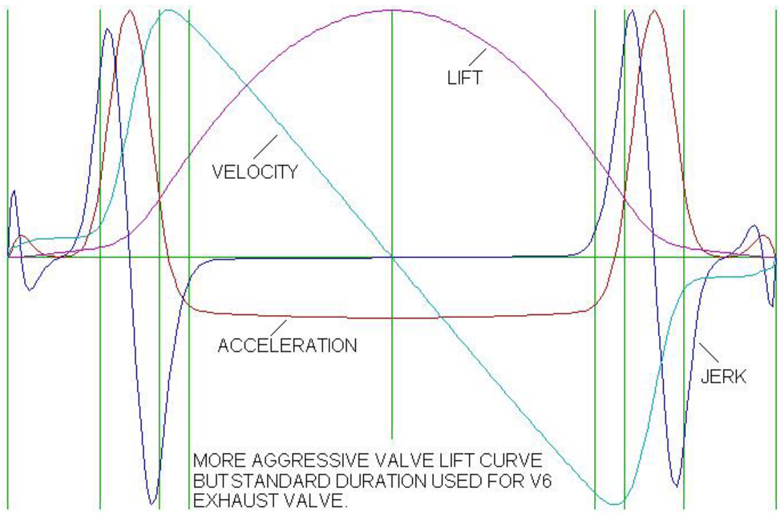

Figure. 20: More aggressive exhaust valve lift profile used for the six-cylinder engine.

To show that all engines can be eventually raised to an acceptable exhaust time-area level prior to commencing engine simulation studies, a more aggressive valve lift profile is employed for the exhaust valves in the six-cylinder engine. This while retaining the same 37% of EDIS as a maximum lift value and also the same opening and closing valve timings of 80 EVO/50 EVC already specified above. The outcome of the analysis for the six-cylinder engine only is shown plotted in Figure. 19 (a-c) above and marked as 'special'. The more aggressive valve lift profile to achieve this effect is provided through the 4stHEAD software [2] and is shown in Figure. 20. The valve lift profile in Figure. 20 raises the valve lift-duration envelope ratio, K_{id}, from 0.557 to 0.595, i.e., a gain of 7% which translates into the roughly equivalent time-area changes for the point marked as 'special' in Figure. 19 (a-c) [2].

The intake and exhaust valve lift profiles can be directly exported to engine simulation software for the immediate commencement of simulation studies of any of the engines.

4.2 The Intake and Exhaust Duct Design Parameters

Effective intake and exhaust tuning is only possible if the ducts have the correct lengths and diameters [1]. The first essential criterion is to set the area of the relevant duct some few diameters away from the valves, usually at the manifold face at the cylinder head, to such a value as will provide the correct amplitude of pressure wave to be propagated away from the cylinder through that duct towards the atmosphere. This pressure wave, of expansion in an intake duct or compression in an exhaust duct, will subsequently be reflected to provide the optimum strength of intake ramming or exhaust suction as to constitute effective engine tuning [1].

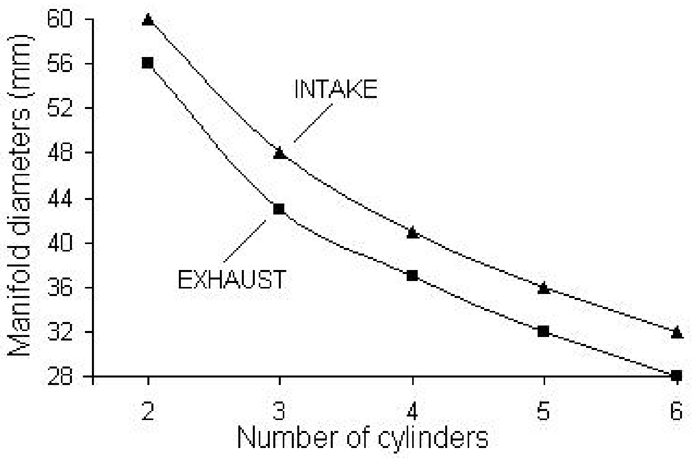

Figure. 21: Intake and exhaust manifold areas (as equivalent diameters).

For all of the engines studied here, the intake manifold-valves area ratio, C_{im}, and exhaust manifold-valves area ratio, C_{em}, are set to 0.95 and 1.3, respectively, using the advice on such numerical values published elsewhere [1]. It is calculated by the 4stHEAD software as the time-area analysis described above proceeds [2]. The outcome for all engines is shown above in Figure. 21 where the area required is expressed in terms of an equivalent diameter. The general trend, as could be expected, is for reducing engine duct sizes with increasing number of cylinders.

Advice on exhaust and intake duct lengths to tune the engine effectively at 15000 rpm, with collector exhaust systems where appropriate, for all of the engines with two to six cylinders is also available but will not form part of this discussion [1].

5. The Mechanical Design Parameters for the Valves

The design debate above makes the major assumption that the valves designed for all cylinders can be operated by overhead camshafts actuating tappets and valves for the lifts and durations already specified and using coil springs to control the dynamic movement of the valves. The Figure. 22 below shows a snapshot of the cam design for manufacture from the 4stHEAD software for the exhaust valve of the six-cylinder engine at its higher 8.0 mm lift specification. It is shown actuating a flat direct-acting bucket tappet. For this particular design, using the mass of the valve, its tappet, its spring retainers, and the coil springs stiffnesses that will control it, the software predicts that at 15000 rpm, and using a SAE 10W-40 oil at 90 deg C, the minimum oil film thickness will be 0.93 micron and the maximum Hertz stress at the cam and follower interface will be some 780 MPa. In short, it will operate successfully and such a cam could be ground directly from the relevant 4stHEAD software output files. However, such success may not be possible for bucket tappet actuation and coil spring control of all of the valves in all of the engines discussed.

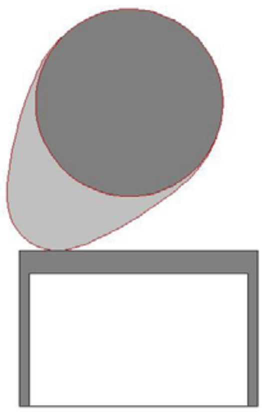

Figure. 22: The six-cylinder exhaust cam and its bucket tappet.

From the software, for all engines and for all valves, it is possible to calculate the valve mass and the mass of the bucket tappet and the spring retainer. Assuming that the bucket tappet and the spring retainers are in steel, but that the valve could be either steel or titanium, the information so gathered is plotted in Figure. 23 below. Needless to add, the data plotted in Figure. 23 is not definitive in that the numerical band will vary from designer to designer and their choice of valve stem thickness, tappet wall thickness, etc., etc., but it is doubtful if that variation is more than +/-10% of that quoted in Figure. 23. With that caveat in mind, it is clear from Figure. 23 that the variation in valve and bucket tappet and spring retainer mass over the range of engines from two to six cylinder is very considerable, as is the range of their valve lifts, to the point where it is highly improbable that the valves of the two-cylinder or three-cylinder engine could be operated in this fashion. It should not be forgotten that it is the combined mass of the valve, tappet and spring retainer which reciprocates and must be controlled by the coil spring. From the two- to the six-cylinder engine, the higher is the reciprocating mass and the valve lift, then the thicker in wire section and the larger in outside diameter becomes the wire of the springs to control that movement. The upshot is that some of the coil spring designs will have excessive dynamic movement at the 15000 rpm operating point and high surface stresses leading to failure.

The experimental evidence from various racing engines, and corroborated by theoretical advice from the 4stHEAD software, is that a total reciprocating mass of some 50-60 g can be successfully controlled at 15000 to 16000 rpm by coil springs.

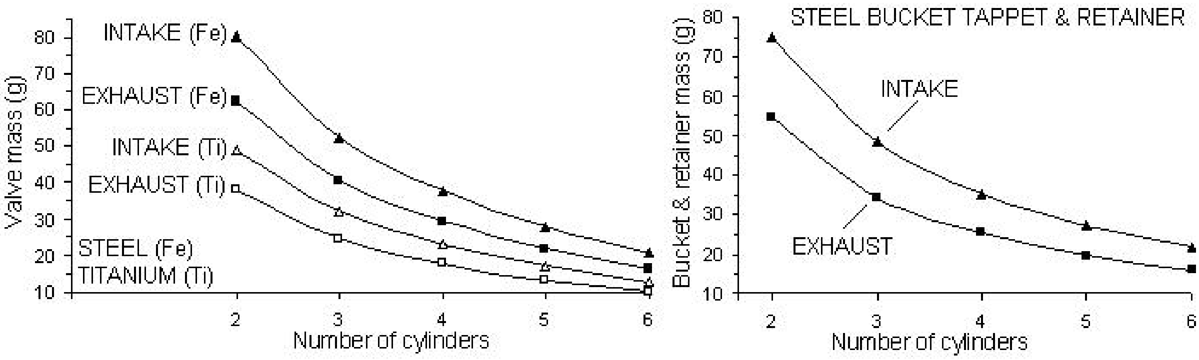

Figure. 23: The mass of valves, bucket tappets and spring retainers for all engines.

By this criterion, and taking data from Figure. 23 above, it is probable that the five- and six-cylinder engines would dynamically survive operation at 15000 rpm using steel valves and bucket tappets. The four-cylinder engine valvetrain might survive the same engine speed by using a steel exhaust valve but would need to use titanium for the intake valve. Close to the 50g mass borderline, the three-cylinder engine would probably need to use titanium for the intake and exhaust valves; however, successful bucket tappet actuation now becomes less probable due to its relatively high mass but the designer could opt for (lighter) finger followers for the valve actuation. For the two-cylinder engine, it would appear from Figure. 23 that coil spring control of titanium exhaust and intake valves, which already alone weigh some 40 and 50 g, respectively, by whatever valve follower method, is really not a design option. Consequently, the use of desmodromic valve actuation or valve control by a pneumatic valve spring is the only way forward for the successful operation of the valvetrain of that particular power unit at 15000 rpm. However, desmodromic valve actuation does come burdened with high friction losses at very high engine speeds and high valve lifts, so it may not be so attractive an option as it might appear to be at first sight.

6. Summary and Conclusions

From the discussion above it is clear that the best combustion characteristics are to be found as the number of cylinders increases and, as this analysis evolves it, to the engines with decreasing bore:stroke ratio. Breathing, which is the ability to induce fuel and air to high levels of delivery ratio from which torque and power are maximised, is more easily accomplished in engines with higher bore:stroke ratios, i.e., as this analysis evolves it, to the engines with fewer cylinders. As usual, the design engineer is faced with a compromise which is equally heavily influenced by engine weight and bulk and also power spread over a wide speed range. However, even here a further compromise appears as the engines with fewer cylinders will, in general, have the least bulk and weight. On the other hand, the engines with better combustion and a lower bore:stroke ratio will normally breathe, and certainly 'burn', their charge better over a wide speed range. However, and this is a most important point, it is shown above that one can change the valve lift and valve lift profile, without even changing valve lift duration, to raise the time-area or breathing characteristics of any engine to another, higher, level. There is no equivalent

procedure to rectify the inherently inferior combustion characteristics of engines with high bore:stroke ratios. Finally, as if the above was not enough to worry about, there is always a future racing season for the designer to bear in mind as peak power demands inevitably rise and the engine speed to attain them does likewise.
It is not the purpose of this paper to declare any particular engine layout as the ultimate design to win MotoGP races. Indeed, the design brief is to examine the potential for two- to six-cylinder engines to produce the same power output of 230 hp at the same engine speed of 15000 rpm and it is hoped that the reader has observed that the design path is not one suited '*to personal choice, copying of that which is deemed to be currently successful, or to some long-held but as yet unproven theory*', expressed above as methodologies seen to be much in vogue. If nothing else, this paper illustrates that today's designers have available software tools to aid the design debate in a fashion that was not possible when four-stroke engines last raced in MotoGP some thirty or more years ago. As the great man once said, "..until you can put numbers on your problem, you are not yet at the beginnings of a science..".
However, history is also a great teacher and the history of motorcycle racing provides the circular argument that ultimately shapes racing engines; 'more power always wins'; 'more power is produced at higher engine speeds'; 'higher engine speeds come from engines with more but smaller cylinders'; 'lightweight valve control mechanisms permit the highest engine speeds'. Basically, with numbers attached, that is what this paper says.

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