Prof. Gordon Blair, CBE, FREng, Senior Associate, Prof. Blair & Associates discusses "Fundamentals and Empiricism in Engine Design"

Back to basics

In technical magazines contributors will sometimes use words to replace numbers when explaining design concepts; in their defence, this may be due to their information source refusing to numerically part with what is deemed to be confidential. As engineers design by numbers and not words it is in an attempt to illuminate some, or to refresh the memory banks of all, into the fundamentals of engine design that this paper is written. To the readers who are experts in engine design theory this paper could be 'old hat', so I apologise for boring them and wasting their magazine space. To engineering graduates of yesteryear with fading memories of their undergraduate course in 'ic engines', and yet others perhaps less theoretically agile, this paper should be welcome as it will permit them to compare design concepts with numbers rather than argument. To all designers, living daily with an often bewildering array of complex computer software, it should be a timely reminder that logic-based empiricism gives very effective guidance to engine design and development.

THE BASICS

An engine is a device with a number of cylinders (ncyl) each with a cylinder bore (B) and a stroke (S). This gives the engine a swept volume (Vsv) for each cylinder and a swept volume for the entire engine (Vtsv). The engine will have a bore to stroke ratio (Kbs). The calculation of this basic data is shown as Eqns.1-3 in Fig.1. If required, the units of any data value in all equations are shown as subscripts.

A further important mechanical design parameter is the mean piston speed (Cp), which is calculated by Eqn.4 in Fig.1. For racing engines, this limit parameter has hardly increased in numeric value in fifty years and that fact reflects the gradual improvement of cylinder design and lubrication technology since the 1950s. Then, an air-cooled and iron-linered Norton Manx cylinder running on Castrol R had a mean piston speed of 20 m/s at its engine speed for peak power. Today, a MotoGP engine at peak power with its liquid-cooled and silicon-carbide plated cylinder running on a synthetic lube oil has a mean piston speed of 25 m/s. One would be hard put to call that a technological breakthrough; at Castrol they will doubtless tell you that Castrol R was not easy to improve on!

"Racing engine mean piston speed has hardly increased in numeric value in fifty years"

THE FUNDAMENTALS

A firing engine produces a turning moment at the crankshaft, the TORQUE. Depending on the speed of rotation of the crankshaft (N) the engine produces a power output (POWER). The TORQUE is measured with the engine on a dynamometer. The computation of the POWER output is shown in Fig.2 in Eqn.5. There the unit is POWERkW (kW or kilowatts) but if you want horsepower (bhp) instead then divide POWERkw by 0.7457 to get POWERbhp.

Alternatively, in Eqn.6 in Fig.2, one can compute the POWER using the brake mean effective pressure (BMEP). However, and much more likely, having used Eqn.5 with the measured TORQUE to get the power output from the dynamometer data, you can calculate the value of the BMEP by back-calculation using a re-arranged Eqn.6, because all other

$$\text{Bore to Stroke Ratio} \quad K_{bs} = \frac{B}{S} \qquad (1)$$

$$\text{Swept Volume per Cylinder } V_{sv}\,(cm^3) = \frac{\pi}{4} \times \frac{B^2_{mm}}{100} \times \frac{S_{mm}}{10} \qquad (2)$$

$$\text{Swept Volume per Engine } V_{tsv}\,(cm^3) = n_{cyl} \times V_{sv} \qquad (3)$$

$$\text{Mean Piston Speed}\,(m/s) \quad C_p = 2 \times \frac{S_{mm}}{10^3} \times \frac{N_{rpm}}{60} \qquad (4)$$

Fig.1 Basic engine geometry equations, Eqns.1-4.

$$\text{POWER}\,(kW) = \frac{2\pi}{10^3} \times \text{TORQUE}_{Nm} \times \frac{N_{rpm}}{60} \qquad (5)$$

$$\text{POWER}\,(kW) = \frac{\text{BMEP}_{bar} \times 10^5}{10^3} \times \frac{V_{sv}}{10^6} \times \frac{N_{rpm}}{60} \times \frac{n_{cyl}}{2_{\text{revs per cycle}}} \qquad (6)$$

$$\text{POWER}\,(kW) = \frac{C_p \times \text{BMEP}_{bar}}{433.5} \times \sqrt[3]{n_{cyl} \times (K_{bs} \times V_{tsv})^2} \qquad (7)$$

$$\text{POWER}\,(bhp) = \frac{C_p \times \text{BMEP}_{bar}}{323.3} \times \sqrt[3]{n_{cyl} \times (K_{bs} \times V_{tsv})^2} \qquad (8)$$

Fig.2 Power related engine equations, Eqns.5-8.

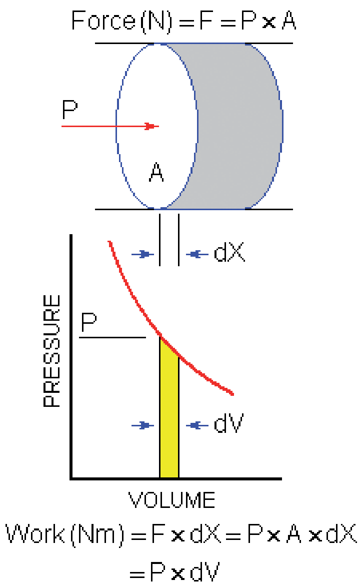

Fig.3 Definition of in-cylinder work.

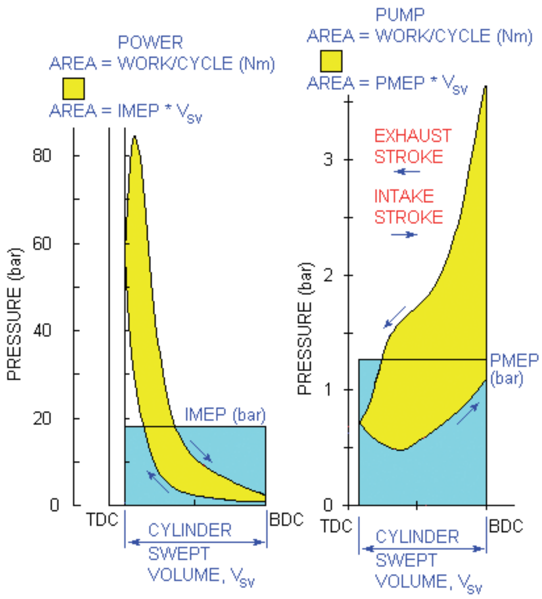

Fig.4 Cylinder pressure analysis for IMEP and PMEP.

equation parameters are known data values for the engine. You will note that the last term in Eqn.6 shows a division by 2 which reflects the fact that all mean effective pressure data are computed over a 360 degree crankshaft period and it takes two such crank degree periods to make up a four-stroke cycle. If we were discussing two-stroke engines that number 2 would be unity in Eqn.6. To finish that point, the word "cycle" refers to the "thermodynamic cycle" of events which in the 4-stroke engine takes 2 crankshaft revolutions; that cleverly efficient 2-stroke engine does it in one!

Later in this paper the concept of mean effective pressure will be discussed in more depth but at this point let it be said, somewhat like mean piston speed, this is another parameter which, for the naturally aspirated, spark-ignition, gasoline burning, four-stroke racing engine, has barely changed over the last fifty years. That Norton Manx racing motorcycle of 1955 attained a BMEP value of almost 14 bar. Today's MotoGP engine also has a BMEP value of about 14 bar. Some progress!

However, that 800 cc MotoGP engine does it at 17,000 rpm, whereas the 500 Norton did at 7100 rpm so the power differential is huge, i.e., 200+ hp versus 55 hp. How is that possible if two of the main performance parameters are virtually constant?

The explanation can be found through the onward manipulation of Eqn.6 to become Eqns.7 and 8. Consider two engines, each with a common BMEP of 14 bar. Take the Norton first: this 'square' 499 cc single-cylinder engine had a bore-stroke ratio (Kbs) of unity and a piston speed (Cp) of 20 m/s which data, when inserted into Eqn.8 gives 54.5 bhp at an engine speed (N) of 6978 rpm; the bore (B) and stroke (S) work out at 86 mm each. Now for today's 800 cc MotoGP four-cylinder engine with a bore-stroke ratio (Kbs) of 1.59 and a piston speed (Cp) of 25 m/s. That data entered into Eqn.8 reveals that the MotoGP engine will produce 201 bhp at 16,121 rpm and the engine bore (B) is 74 mm and the stroke (S) is 46.5 mm.

As the BMEP potential and the piston speed are such common parameters between engines, the Eqns.7 and 8 become very useful ready-reckoners as to the possible power performance of any engine. For example, the highest mean piston speed (Cp) I have heard of is 26.5 m/s and the maximum BMEP potential of the simple naturally aspirated, spark-ignition, gasoline burning, four-stroke racing engine at high piston speed is some 15 bar. Tuned at lower engine speeds and high compression ratios, BMEP figures above 16 bar are possible. If methanol or ethanol fuel is used, then add 10% to the potential BMEP. If the gasoline engine is turbocharged or supercharged, then the possible BMEP attainable is found by multiplying that exemplar 15 bar naturally-aspirated BMEP by the boost pressure ratio. If the engine is a 'diesel' the situation is a little more complex as account must be taken of both the lean air-fuel ratio and the very high compression ratio that it employs.

Nevertheless, Eqns.7 and 8 permit you to make quick design decisions as to what performance is potentially possible from any engine. It also permits you to numerically trip up those PR-based engine developers who grossly exaggerate their engine's power output as evidence of the application of their genius!

THE MEAN EFFECTIVE PRESSURE

Work is defined as the distance (dX) moved by a force (F). In the context of a piston in a cylinder, as seen in Fig.3, the force (F) on the piston is the product of the pressure (P) on it when applied over the piston area (A). Hence, the work done on, or by, the piston as it moves is the product of the pressure (P) and the cylinder volume change (dV) as it occurs. On the power stroke, as the volume increases that work is positive. On the compression stroke, as the volume decreases that work is negative, i.e., supplied by the engine to the piston.

During the power phase, from bottom dead centre (bdc) to the next bottom dead centre (bdc), or one turn of the crankshaft, the pressure-volume diagram of the in-cylinder events is shown in Fig.4. It is sketched from data for the MotoGP engine discussed in a previous issue of RET [2]. The net work (POWER WORK) on the piston during this process is the summation (integration) of all of the pressure- ▶

> "Those engine developers who grossly exaggerate their engine's power as evidence of their genius!"

$$\text{NET WORK}_{percyl} = \text{BMEP} \times V_{sv} = (\text{IMEP} - \text{PMEP} - \text{FMEP}) \times V_{sv} \quad (9)$$

$$\text{POWER} = n_{cyl} \times \text{NET WORK}_{per\ cyl} \times \frac{\text{REVS}_{per\ sec}}{\text{REVS}_{per\ cycle}} \quad (10)$$

$$\text{IMEP} = \frac{\text{POWER WORK}_{per\ cylinder}}{V_{sv}} \propto \frac{Q_{fuel\ per\ cylinder}}{V_{sv}} \quad (11)$$

$$\text{IMEP} \propto \frac{M^{fuel}_{per\ cyl}}{V_{sv}} \propto \frac{M^{air}_{per\ cyl}}{V_{sv}} \quad (12)$$

$$\text{TORQUE} \propto \text{NET WORK}_{per\ cyl} \propto \text{BMEP} \times V_{sv} \quad (13)$$

$$\text{BMEP} \approx\propto \text{IMEP} \propto \frac{M^{air}_{percyl}}{V_{sv}} = \text{DR} \quad (14)$$

Fig.5 WORK, POWER, IMEP, BMEP, and DR,, Eqns.9-14.

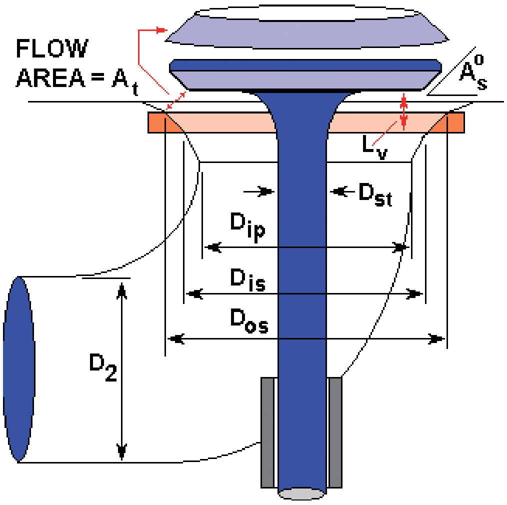

Fig.6 Valve, valve seat, port and manifold geometry.

volume increments over the period and is shown as the area coloured yellow in the diagram. If there were no other losses in the system, that would be the work delivered to the crankshaft; but there are.

The yellow area can be represented by the equivalent rectangular area shown in blue, which area has a height of IMEP and a width of the cylinder swept volume (Vsv). The value of IMEP is known as the indicated mean effective pressure. It is called 'indicated' as it is derived from the pressure transducer signal as measured in the cylinder head of the engine; in ancient times this signal was referred to as an 'indicator diagram'. The 'ancient times' I refer to are my student days where the only device available was an amazing instrument called a 'Farnborough' indicator; that should awaken a few memories among the octogenarian readership!

During the pumping phase that follows, from bottom dead centre (bdc) to the next bottom dead centre (bdc), or the next turn of the crankshaft, the pressure volume diagram is shown at the right of Fig.4. It too is for the MotoGP engine at 16,100 rpm [2]. Here, the work computation would elicit a negative value for the PUMP WORK, the yellow area on that diagram, as the opening (higher) line is of compression (negative dV) during the exhaust stroke. This yellow area can be equally represented by the equivalent rectangle of height PMEP and width Vsv and PMEP becomes labelled as the pumping mean effective pressure. Its negative numerical value indicates that the pumping work is supplied by the piston from the crankshaft; in short, it is lost work.

The rest of the engine work losses are lumped together as 'frictional' losses and can be expressed as a FMEP value, the friction mean effective pressure, again officially a negative number.

The upshot of this part of the discussion can be seen in Fig.5, containing Eqns.9-14. The net work per cylinder per cycle is shown in Eqn.9 where the brake mean effective pressure (BMEP) is observed to be the result of subtracting the (positive values of) the pumping mean effective pressure (PMEP) and the friction mean effective pressure (FMEP) from the indicated mean effective pressure (IMEP).

AS IMEP and PMEP data can only be determined from an analysis of measured cylinder pressure diagrams, but BMEP can be calculated from measured dynamometer data through Eqns.6-7, then one method of determining FMEP is through the re-arrangement of Eqn.9. The other method is to motor the engine on a dyno and measure the 'frictional' torque and calculate a 'FMEP', but there are complications here with the presence of another, and different, pumping loss during the motoring process. The ratio of BMEP to IMEP is known as the 'mechanical efficiency' of the engine and is normally in the 75 to 85% range for most racing engines.

The MotoGP engine [2], designed for a BMEP of 14 bar at 16,100 rpm, produced exactly that. It had a IMEP value of 18.52 bar, a PMEP value of 1.26 bar (see Fig.4) and a FMEP value of 3.26 bar; the mechanical efficiency was 75.6%.

> "The MotoGP engine designed for a BMEP of 14 bar at 16,100 rpm had a mechanical efficiency of 75.6%"

THE CONNECTION BETWEEN MEAN EFFECTIVE PRESSURE AND AIRFLOW

In Fig.5, Eqn.11, it can be seen that IMEP is the POWER WORK divided by the cylinder swept volume. Alternatively, that in-cylinder work is directly proportional to the heat released (Q) by combustion of the fuel trapped in the cylinder. The value of heat released (Q) will also be a function of compression ratio [1] but we will ignore this as it is a second order effect. In Eqn.12, this argument is advanced to relate the heat released (Q) to the mass (M) of fuel trapped in the cylinder. However, as air-fuel ratios for racing engines on gasoline are almost fixed at a 'lambda' value of 0.85, and with the calorific value of gasoline a virtual constant, Eqn.12 reduces to showing that IMEP is directly proportional to the mass of air (Mair) trapped in the cylinder.

It is but a short logical step in Eqn.14 to relate the BMEP to IMEP and the BMEP to the specific mass airflow rate into the engine, i.e., delivery ratio (DR). An even shorter logical step is found by linking Eqns. 13 and 14 to relate the engine TORQUE output per cylinder to BMEP and DR. In short, as BMEP and DR have only minor variations from one racing engine to another, BMEP and DR are far more useful numbers with which to compare the development level of differing engines than is the output TORQUE, because this number also incorporates the total swept volume of an engine. The bottom line, design-wise, is that brake mean effective pressure (BMEP) is inextricably linked with the specific mass airflow rate ratio, delivery ratio (DR).

In this discussion, you will note that I have not used or defined the term ‘volumetric efficiency’, which is a volume based specific airflow rate parameter. Imagine we have a race engine breathing air at sea-level and 20 deg.C. We take the same engine and run it at altitude where the air pressure is 90% of the sea-level condition but the temperature is still 20 deg.C. The air density at this altitude is 90% of that at sea-level. The engine will gulp exactly the same volume flow rate of air at altitude as at sea-level but only 90% of the mass flow rate of air. The engine will produce performance characteristics at altitude virtually pro-rata with air density. Hence, the volumetric efficiency of the engine at altitude is identical to that at sea-level whereas at altitude the delivery ratio is 90% of the sea-level value. It is obvious which parameter is the more useful as a guide for the engine power characteristics, and which is not and why it is ignored.

THE ONWARD CONNECTION BETWEEN AIRFLOW AND DESIGN

An engine inhales air through its intake valve(s) and exhales through its exhaust valve(s). The aperture area (At) through which this flow takes place at any valve lift (Lv) is shown in Fig.6. Also shown is the basic geometry of a valve seat, a valve seat angle, a valve stem, an inner port, and a duct size at the manifold. The physical dimensions are labelled as the valve seat angle (As), the diameters at the seat (Dis and Dos) and at the inner port (Dip), and at the manifold (D2). The manifold diameter (D2) may connect to a number of valves (nv) so, if so, the total aperture area for flow is obviously a multiple (nv times At) of that illustrated for one valve.

The aperture flow area (At) is considered as being the side area of a frustum of a cone and that cone shape changes position with lift

> “It must be persistently used to the point of pedantry in absolutely every aspect of the race engine design process”

(Lv)[3]. We decided to use this flow area convention in Belfast some forty years ago. It is not vital to employ this particular criterion as one could equally well select the At value as the side area of a simple cylinder comprising the inner seat diameter (Dis) with the height of the valve lift (Lv). What is vital, is that having decided on the use of a particular convention to acquire the aperture area (At) then, if all further analyses are to be accurate, it must be persistently used to the point of pedantry in absolutely every aspect of the design process from the experimental determination of discharge coefficients (Cd), valve flow time-areas, through to implementation within a theoretical engine simulation.

The theoretical computation of the airflow rate is conducted through the equations, Eqns.15-17, as illustrated in Figs.7 and 8. The opening statement of Eqn.15 repeats the last statement of Eqn.14 but continues on to show the fine detail of the computation of delivery ratio (DR) as a summation (integration), crankangle by crankangle, of small increments of the airflow rate. At any one step, the effective area of the aperture is the product of the discharge mass flow coefficient (Cd) and the area (At). The volume flow is found by multiplying that value by the particle velocity, and the mass flow rate by multiplying that product by the prevailing gas density (rho). The summation is conducted over the main part of the intake stroke from tdc to bdc. This is the complex step by step integration that proceeds incrementally within any computer-based engine simulation [3]. However, this computational approach for the delivery ratio (DR), seen in Eqn.17, will never be executed on your pocket calculator!

The main variables (Cd, rho, and c) vary dramatically during the

$$DR = \frac{M^{air}_{cycle}}{\rho_{ref} V_{sv}} = \frac{\dfrac{dt}{d\theta}\displaystyle\int_{tdc}^{bdc} C_d \rho c A_t \, d\theta}{\rho_{ref} V_{sv}} \qquad (15)$$

DELIVERY RATIO (dimensionless); DISCHARGE COEFFICIENT; DENSITY; PARTICLE VELOCITY; TIME; °CRANK; AIR DENSITY at STP; SWEPT VOLUME PER CYLINDER

Fig.7 Theory to compute delivery ratio (DR) Eqn.15.

$$DR = \frac{M^{air}_{cycle}}{\rho_{ref} V_{sv}} = \frac{\dfrac{60/N}{360}\displaystyle\int_{tdc}^{bdc} C_d \rho c A_t \, d\theta}{\rho_{ref} V_{sv}} \qquad (16)$$

$$= \frac{1}{6N}\,\frac{\displaystyle\int_{tdc}^{bdc} C_d \rho c A_t \, d\theta}{\rho_{ref} V_{sv}} \propto \frac{\displaystyle\int_{tdc}^{bdc} A_t \, d\theta}{6NV_{sv}} \qquad (17)$$

DELIVERY RATIO (dimensionless); APERTURE AREA; $\frac{dt}{d\theta}$; SPECIFIC TIME-AREA (STA) (units are s/m); ENGINE SPEED (RPM)

Fig.8 DR related to specific time-area (STA), Eqns.16-17.

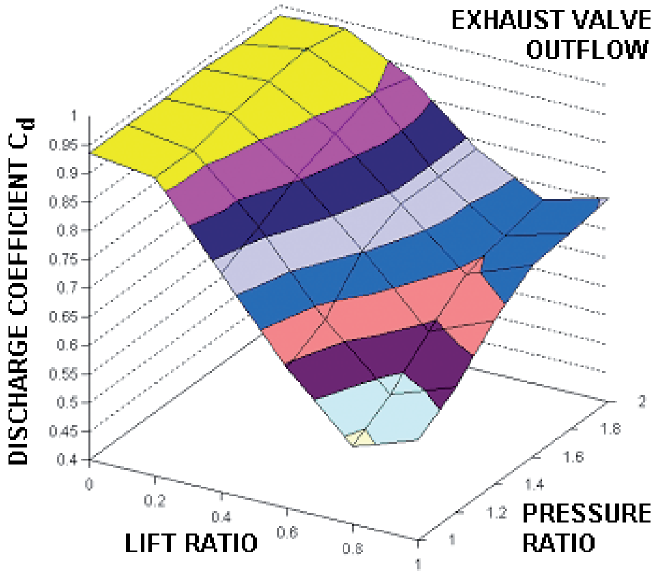

Fig.9 Discharge coefficient map (Cd) for exhaust outflow.

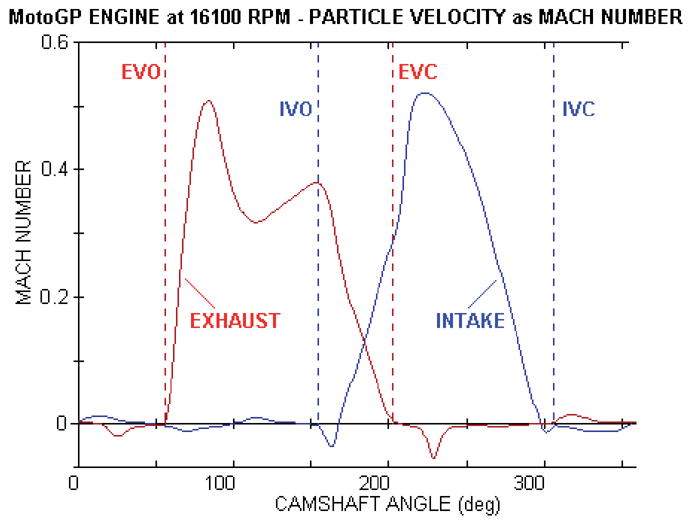

Fig.10 Exhaust and intake duct particle velocities.

summation process from tdc to bdc. For the discharge coefficient (Cd), a feel for the extent of that variation can be obtained in Fig.9 for exhaust valve outflow where it changes considerably not only with valve lift but also with the pressure ratio across the valve. In Fig.10 is shown the variation of the particle velocity (c) at the manifold diameter (D2) for both the exhaust and the intake processes; the value is plotted as Mach number which is particle velocity (c) divided by the local acoustic velocity. For the intake flow, from tdc to bdc, the particle velocity rises from near zero to a Mach number of about 0.5 (about 170 m/s and see below re Mean Gas Velocity).

However, while these variations are significant and would inhibit the 'pocket calculator' solution for DR at the penultimate term of Eqn.17 in Fig.8, the pattern of all these variations from engine to engine is really quite similar. Hence, citing these similarities, we can solve for the very last term in Eqn.17 by declaring that the value produced is proportional to, but not equal to, delivery ratio (DR).

The last term in Eqn.17 is known as the specific time area (STA) with units of s/m; as it is for the "intake pumping period", or intake stroke, it is labelled as STAip. In Fig.11 is shown the graphical result of solving separately the top line of the last term of Eqn.17; it is the area coloured blue in the diagram which is the integration of the intake aperture area from tdc to bdc. The entire intake valve period extends from opening (IVO) to closing (IVC), but the STAip data refers only to the main intake pumping period from tdc to bdc.

In Fig.12 is sketched the result of the equivalent calculation for the exhaust pumping period, the exhaust stroke from bdc to tdc for the exhaust valve and is labelled as STAep. By definition, any air mass induced into the engine inevitably becomes the exhaust mass post-combustion (plus the added fuel mass) and which requires to be expelled from the engine. Therefore, there is an obvious proportionality connection between the STAip and STAep values.

In a racing engine with tuned intake and exhaust systems, scavenging of the trapped exhaust gas during the valve overlap period from the small space that is the clearance volume is a vital part of effective engine design. The pressures that force this process can be seen in Fig.13; this picture is taken from a simulation of the MotoGP engine at 16,100 rpm [2]. It can be seen that the direction of the pressure difference across the cylinder during the majority of the valve overlap period is from the intake side to the cylinder and onwards towards the exhaust. This permits a through-draught of fresh charge to scavenge the cylinder of its exhaust gas and fill it with fresh charge. If carried out effectively say, in an engine with a compression ratio of 11, it means that the induction process can begin with a delivery ratio of some 10% before the downward piston motion even begins to suck in air.

At the risk of being accused of pedantry, 10% extra DR can be 10% extra BMEP, 10% extra TORQUE and 10% extra POWER. The scavenge process will only be successful, always assuming that the intake and exhaust tuning is as well organised as in Fig.13, if the phasing of the opening intake valve and the closing exhaust valve apertures is effective. This phasing is well-expressed pictorially in Figs.14 and 15 by the specific time areas for the overlap valve periods for the exhaust valve(s) (STAeo) and the intake valve(s) (STAio) as the red and blue coloured areas, respectively. If either value, or both, is numerically deficient then, even with perfect pressure wave tuning, the throughflow scavenge process will be impaired, as will the engine POWER. In a two-door room with both doors shut, there are no draughts on a windy day.

Another valve area segment to be considered is the period from the opening of the exhaust valve(s) to the bdc position, i.e., the exhaust blowdown period. The specific time area for this period (STAeb) is shown, coloured red, in Fig.16. If this value is numerically inadequate then the cylinder pressure at bdc will be high as a sufficient mass of exhaust gas has not been bled from the cylinder. Hence, the ensuing exhaust pumping process from bdc to tdc will be conducted with higher than normal cylinder pressures giving increased pumping losses (PMEP) and may even promote excessive exhaust gas backflow up the intake tract as the intake valve opens. If this latter situation occurs, even a well-designed scavenge process could be negated because it would be conducted with backflow exhaust gas and not fresh intake charge; a situation guaranteed to invisibly and inexplicably reduce power output, raise the trapped charge temperature and encourage

> "A situation guaranteed to encourage detonation and make a tyro designer believe in chaos theory!"

detonation, and make a tyro designer believe in chaos theory.

The final valve area segment to be considered is the period from the bdc position on the intake stroke to intake valve closure at IVC, i.e., the intake ramming period. The specific time area for this period (STAir) is shown, coloured blue, in Fig.17. The higher is the required delivery ratio (DR), i.e., the higher required BMEP and STAip values, then so too must be the need for effective intake ramming which requires sufficient valve aperture and time at any engine speed. That a well-designed and phased intake system will give the correct direction of pressure differential to encourage a ramming action can be observed from bdc to IVC in Fig.13.

THE CONNECTION BETWEEN SPECIFIC TIME AREA AND DESIGN

Many years ago [4] I established the effectiveness of the STA-BMEP connection for two-stroke engines and adapted it for the design of four-stroke engines [3]. For the four-stroke units, I analysed many engines [3], ranging from high performance racing engines to lawnmowers, all at their engine speed for peak horsepower, and discovered that there was indeed a logical numerical connection between their individual STA values and the BMEP attained by them.

There was, naturally, scatter in this plethora of data from so many sources, but the trends were very clear. A theoretical connection between STA and BMEP was established and reduced to equations; these are the Eqns.18-23 seen in Fig.18. It cannot be emphasised too strongly, as these equations were determined at the engine speed for peak horsepower, that they can only be applied in reverse for another engine as design criteria at the required speed for peak power for that engine. Another important point to note is that this is empiricism and so, in the design mode, while one should match the six individual STA values as closely as possible to their target values for a required BMEP, it is not critical to match them to the last 0.001%. What is important is not to have any one STA value seriously deficient of its target value as that will make the design 'unmatched' and the engine will breathe badly.

Although the Eqns.18-23 can be solved on your pocket calculator, it is too complicated to produce the actual STA values for a given engine. This requires the numerical integration of the six segments of the two valve lift curves and their aperture areas. This is really only possible on a computer with a spreadsheet, e.g., in MS Excel, or hard-coded into a computer program.

Today, in the 4stHEAD software [5] the STA analysis for the empirical design of a new engine, or analysis of an existing one, is conducted within a dedicated computer program. It should also be stated that the Eqns.18-23 really only apply to naturally aspirated, spark-ignition, gasoline burning, four-stroke engines as those were the engine types analysed for their creation [4]. In the 4stHEAD software, the original STA-BMEP equations of Fig.18 have been enhanced and extended to cope with both spark-ignition and compression-ignition engines; with the use of gasoline, kerosene, methanol and ethanol fuels; with the employment of differing compression ratios; and the use of supercharging or turbocharging.

THE CONNECTION BETWEEN THE VALVE APERTURES AND THE DUCTS

The high cylinder pressure during exhaust blowdown, and the low cylinder pressure during the induction stroke, creates compression waves and expansion (suction) waves in their respective ducts. It is the reflection of these waves at the exhaust pipe end (or mid-section ▶

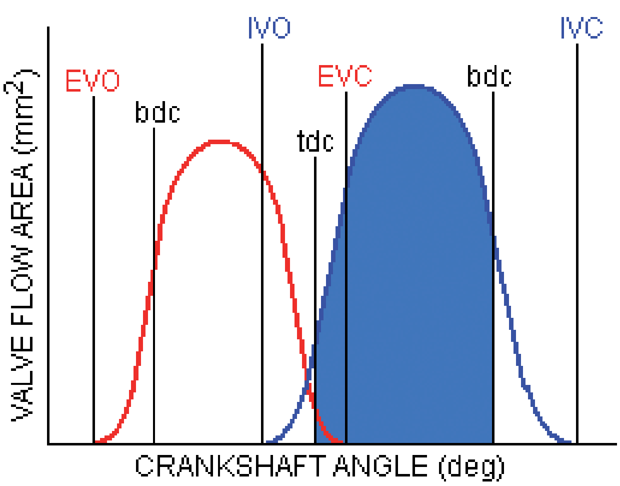

Fig.11 Intake pumping specific time-area, STAip.

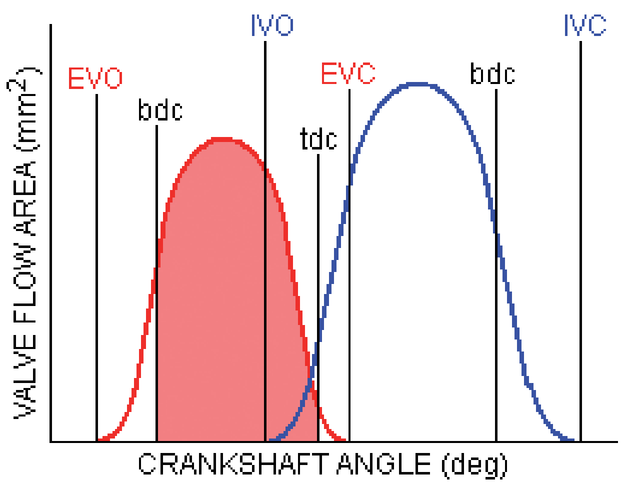

Fig.12 Exhaust pumping specific time-area, STAep.

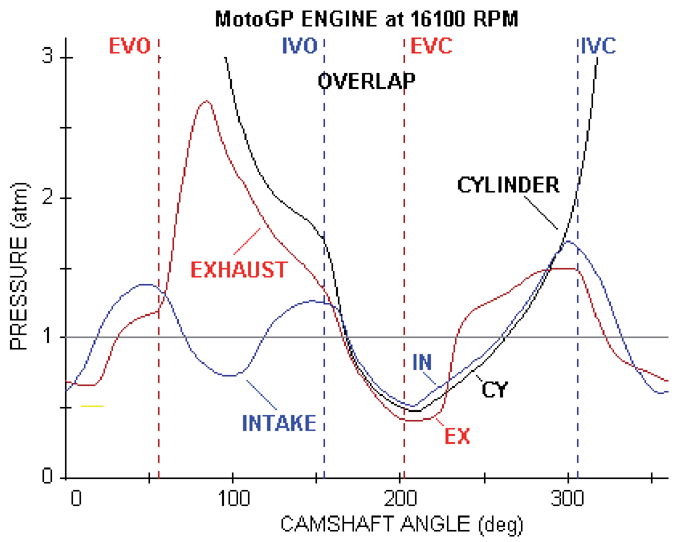

Fig.13 Cylinder, exhaust and intake pressure diagrams.

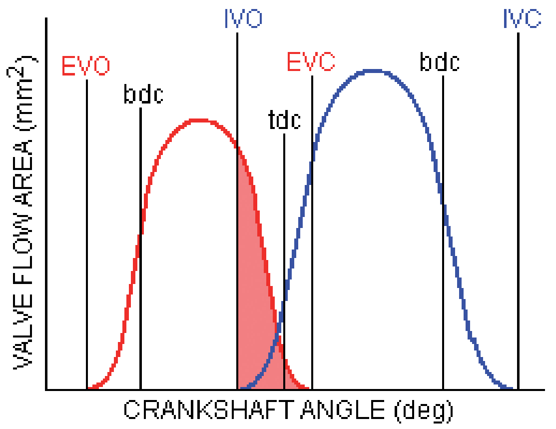

Fig.14 Exhaust overlap specific time-area, STAeo.

expansions at a branch or collector), or at the bellmouth of the intake, which provides the pressure differential characteristics to conduct cylinder scavenging during the exhaust overlap period [4]. In the case of the intake, that tuning length also needs to be set correctly to aid the ramming process [4].

Apart from designing in the correct lengths, the empirical design of which is more than adequately covered elsewhere [4], the size of the ducts at the manifold is a most important design consideration and one which is rarely, if ever, emphasised in published empirical theory. If the ducts are too large then the pressure waves will be weak except at the very highest speeds and if too small they will yield waves of excessive amplitude except at the lower engine speeds. Upon pipe end reflection, weak waves give less effective pressure differentials for the scavenge or ramming processes and waves of excessive amplitude friction-scrub themselves along the pipe walls with an inevitable reduction in their strength giving the same outcome.

Analysis of many engines, both empirically from their physical geometry and theoretically using complete engine simulations, yields empirical design criteria for the optimum size of these ducts. The empirical criteria relate the manifold duct size to their valve apertures and the number of valves (nv) providing them [4]. The calculation of these empirical design criteria for the intake and exhaust pipes is given in Eqn.24 in Fig.19 for the manifold to port area ratios (Kem and Kim) as are the limit values recommended for their use.

DESIGN USING EMPIRICAL CRITERIA

I have already highlighted the design of a MotoGP engine previously presented in considerable detail in RET [2]. This discussion extends the design information for that engine. It was initially designed using the 4stHEAD software against all of the empirical criteria debated above [5]. The numerical evidence behind that statement is shown in Fig.20 for the specific time areas (STA) which were incorporated into the design at 16100 rpm for a piston speed (Cp) of 25 m/s and a BMEP of 14 bar with the exhaust and intake duct sizes (D2) labelled as standard in Fig.20. You will note that it is not possible to meet all STA target values precisely, nor is it vital as empiricism is not an exact science! The reason one cannot precisely mesh actual and target STA values is that one must work within the confines of real valve lift profiles that must also survive without failure the 4stHEAD analyses of valvetrain dynamics and cam design and manufacture [2].

This data for the MotoGP engine is presented to an accurate engine simulation [3] and run over a speed range from 12,000 to 17,000 rpm with the three differing sizes of exhaust and intake ducts shown in Fig.20. The results for POWER and airflow rate (DR) are shown in Figs.21 and 22. The largest duct pairing gives the highest power and airflow at the higher engine speeds but loses out at 12,000-14,000 rpm. However, the larger duct pairing also exceeds the designed power output of 202 bhp (14 bar BMEP at 16100 rpm) but at the expense of a 'peaky' power curve and an even 'peakier' airflow curve; the latter may provide on-track difficulties in fuelling smoothly. The standard duct sizes match the design power criterion exactly. The smaller duct pairing wins out at the lower speeds but loses power at the design speed of 16,100 rpm and above. The behavioural forecast for the empirical Km criteria is seen to be justified.

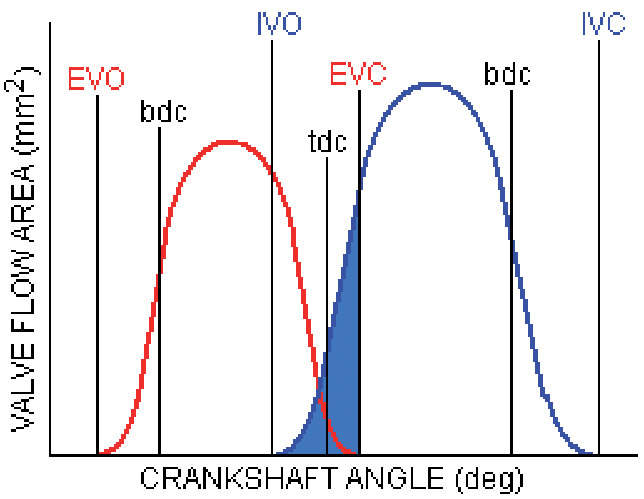

Fig.15 Intake overlap specific time-area, STAio.

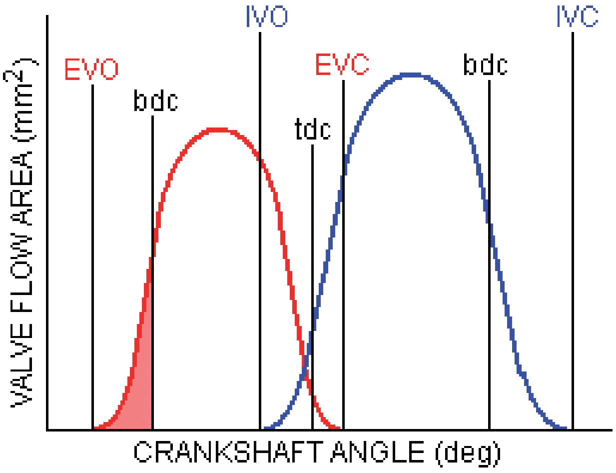

Fig.16 Exhaust blowdown specific time-area, STAeb.

As seen in Fig.20, the applied Km criteria reveal that the duct size varies from the 'standard' value by about +/- 1 mm for 'acceptability' and that 'acceptability' is well-nigh proven in Figs.21 and 22. As these Km criteria exhibit a very narrow dimensional tolerance, this evidence should provide a cautionary tale for those who may somewhat arbitrarily size their engine ducts and are even now pondering the reasons behind either 'peaky' power curves or 'inadequate' peak power curves when, by their design lights, they ought to have been 'perfect'.

A further insight into the design thinking behind the Km criteria shown in Fig.19 is provided by the graphs of the Mach number (the particle velocity) in the ducts of the MotoGP engine in Fig.23. With the three duct sizes as input data, these are computed by the engine simulation at the design point and peak power speed of 16,100 rpm. The 'standard' data (STD) has already been shown in Fig.10 but is repeated here. A 'perfect' design is considered to have a maximum particle velocity in the exhaust and intake ducts where the Mach number is 0.5.

It can be seen in Fig.23 that the standard data does exactly that, thereby justifying the numerical selection of Kem at 1.2 and Kim at 0.95 as an optimum. Here, the larger duct has lower Mach numbers than 'standard' but will assuredly rise to the 0.5 level at speeds above 16100 rpm, thereby giving more airflow and BMEP and POWER at those speeds. The smaller duct has high peak Mach numbers at 16,100 rpm which basically equates to too-strong exhaust and intake pulse amplitudes, while the highest exhibited value of the exhaust particle velocity at intake valve opening (IVO) almost certainly indicates that the exhaust pumping loss, and exhaust gas backflow into the intake tract, with this smaller pipe is greater than the others.

In short, a well-executed design using the STA-BMEP parameters can be negated to some extent by an incorrect sizing of the intake and the exhaust ducting.

"A well-executed design can be negated to some extent by an incorrect sizing of the intake and the exhaust ducting"

SIMPLER EMPIRICAL THEORY

The literature is full of simpler empirical design theories. You will find a selection of references on the topic at the end of Chapter 6 of a textbook [3]. I will examine but one here and one that is often quoted, i.e., the Mean Gas Velocity (Kl) or, as describes it better, a 'mean intake gas velocity' criterion. The basis for its calculation can be found in Fig.24 using Eqns.25 and 26.

First, one calculates the Inlet Valve Area ratio, the ratio of the area exposed by the intake valve(s) to the cylinder bore area, using Eqn.25. In Eqn.25, whether one should use the outer seat diameter of the valves (Dos), or the outer diameter of the valve itself (Dv), or the inner port diameter (Dip), is not clear and is certainly not a subject of clarification in any 'technical' paper that I have read. Having acquired the value of Kiv, this data is used in Eqn.26 where the mean piston speed (Cp) is divided by Kiv to determine the Mean Gas Velocity (Kl); it has the units of velocity (m/s).

Consider the geometry of the MotoGP engine at peak power at 16,100 rpm; the mean piston speed (Cp) is 25 m/s and the cylinder bore is 74 mm. The outer and inner seat diameters (Dos and Dis) of each of the two intake valves are 30 and 28 mm, respectively. There is no throat, so the inner port diameter (Dip) is also 28 mm. Taking the two possible 'intake valve diameters' into Eqn.25, the intake valve area ratios (Kiv) are 0.323 and 0.286, respectively. In Eqn.26 that yields Mean Gas Velocities of 87.4 and 76 m/s.

To some of my readers, these numbers may have great significance. ▶

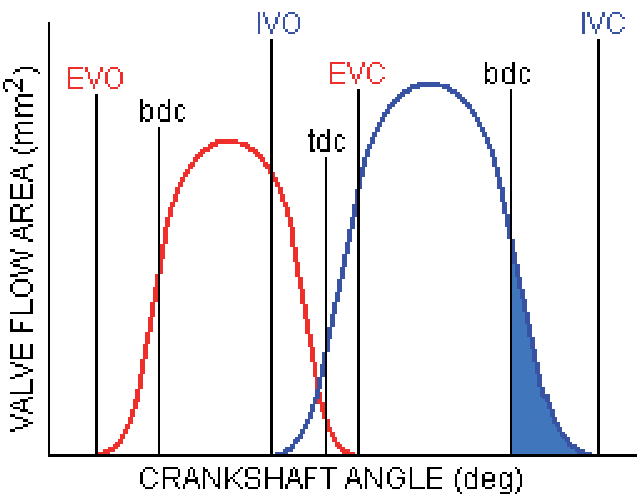

Fig.17 Intake ramming specific time-area, STAir.

$$\text{Intake Pumping} \quad STA_{ip}\,(s/m) = \frac{5.02 \times BMEP_{bar} + 57.78}{10000} \quad (18)$$

$$\text{Exhaust Pumping} \quad STA_{ep}\,(s/m) = \frac{1.7775 \times BMEP_{bar} + 74.822}{10000} \quad (19)$$

$$\text{Intake Overlap} \quad STA_{io}\,(s/m) = \frac{4.1185 \times BMEP_{bar} - 17.985}{10000} \quad (20)$$

$$\text{Exhaust Overlap} \quad STA_{eo}\,(s/m) = \frac{3.0296 \times BMEP_{bar} - 11.363}{10000} \quad (21)$$

$$\text{Exhaust Blowdown} \quad STA_{eb}\,(s/m) = \frac{1.6329 \times BMEP_{bar} - 7.1871}{10000} \quad (22)$$

$$\text{Intake Ramming} \quad STA_{ir}\,(s/m) = \frac{2.4022 \times BMEP_{bar} - 14.57}{10000} \quad (23)$$

Fig.18 STA relationship with BMEP defined, Eqns.18-23.

$$\text{Manifold} - \text{Port Area Ratio} \quad K_m = \frac{A_{manifold}}{A_{port}} = \frac{D_2^2}{n_v \times (D_{ip}^2 - D_{st}^2)} \quad (24)$$

$$\text{Exhaust criterion} \quad 1.2 < K_{em} < 1.4 \quad \text{standard value of } K_{em} = 1.3$$

$$\text{Intake criterion} \quad 0.9 < K_{im} < 1.0 \quad \text{standard value of } K_{im} = 0.95$$

Fig.19 Manifold to Port Area Ratio (Km) defined, Eqn.24.

800 cc MotoGP engine at 16100 rpm
units of STA values are s/m x 10000

		actual	target
Exhaust	STAeb values	14.4	14.4
	STAep values	109.8	98.8
	STAeo values	29.3	28.3
Intake	STAip values	130.1	122.3
	STAir values	17.7	18.2
	STAio values	32.8	36.8
Exhaust	duct diameters	D2 (mm)	
	Kem = 1.4	36.2	
	Kem = 1.3	34.9	standard
	Kem = 1.2	33.5	
Intake	duct diameters	D2 (mm)	
	Kim = 1.0	39.2	
	Kim = 0.95	38.2	standard
	Kim = 0.9	37.2	

Fig.20 STA and Km data computed for a MotoGP engine.

I regret to say that, while this is a design criterion for the dimensions of an intake valve and is doubtless helpful in that regard, further assistance is not forthcoming for the rest of an engine design.

However, if one replaces the mean piston speed with the maximum piston speed (Cp) in Eqn.26, i.e., the above-used number 25 would virtually double to about 50 m/s, the Mean Gas Velocities then double to 168.8 and 152 m/s, respectively, the first of which is not a million miles/hour away from the Mach number optimum of 0.5 (170 m/s) debated above. This reasonable correlation, between Mach number and a Kl value based on maximum piston speed, lends theoretical credence to its usefulness as a basic method to size an intake valve.

Moreover, if one extends Mean Gas Velocity thinking to the exhaust valves of an engine, where the speed of sound in the elevated temperatures of exhaust gas is some 600 m/s, there the Mach number criterion of 0.5 translates to (if computed at maximum piston speed) a Mean Gas Velocity of 300 m/s. For the exemplar MotoGP engine, using 50 m/s as the maximum piston speed and 22 mm which is the inner port diameter (Dip) of each exhaust valve, the exhaust valves area ratio (Kev) is 0.177 from Eqn.25, and an exhaust-based Mean Gas Velocity becomes 284 m/s from Eqn.26; and that is a pretty good match for the supposedly required value of 300 m/s. Hence, it seems feasible to extend the Mean Gas Velocity concept to the exhaust valves as well; this is important as the relative sizing of the exhaust and intake valves is a critical design factor which has been previously discussed [6].

However, while the basic sizing of the valves in any given design may well be guided by using the Mean Gas Velocity for the intake valve(s) and also by this extension for the exhaust valve(s), it falls short of telling us what to do with either of them to tailor a required engine power characteristic.

Firstly, there is no information as to the valve lift profile which should accompany a Mean Gas Velocity; such as how high should the valve(s) be lifted?; such as the required duration or the angular positions of valve opening or closing or maximum lift?; or what happens if I employ a more or a less aggressive valve lift profile? Secondly, in the absence of an extension of the Mean Gas Velocity concept to dimension the exhaust valves, we would not know the required size of the exhaust valve(s) which should accompany the intake valve(s); or how high and for how long, and when, they should operate, etc., etc?

The good thing about the Mean Gas Velocity (Kl) concept is that it can be easily derived on a hand calculator but as a design tool, even with the above-proposed extension for sizing exhaust valves, it is much too simplistic to be universally useful. Technical journalists et al should consider quoting Kl data with the relevant caveats and not as Holy Grail.

It was, as I understand it, the late Brian Lovell of Weslake who conceived Mean Gas Velocity with respect to intake valves. Before I get literally savaged by some technical journalist who feels that I have demeaned the memory of a great design engineer, I should point out that Brian Lovell proposed Mean Gas Velocity as a means of comparing engines for which precious little data was available in a design era populated with slide rules and not computers; more complex calculations were definitely not on the menu.

CONCLUSIONS

If I have thoroughly bored my expert reader I can only but repeat my earlier apology; the fault is yours, you should have stopped reading back at the first page!

To those to whom this paper is a refresher course from their university days, then that is no bad thing. Although I very much doubt that your undergraduate university course ever extended to unsteady gas dynamics, pressure waves and engine tuning, not to speak of specific time areas, to be reminded of the fundamentals and see them extended into effective design techniques is, as has been said before, no bad thing.

To those who find even this level of maths somewhat daunting, but yet have a basic understanding of engine tuning, rest easy because all

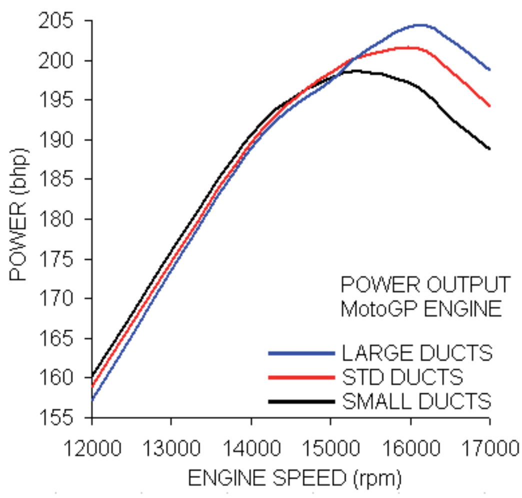

Fig.21 MotoGP engine power (bhp) with alternate ducting.

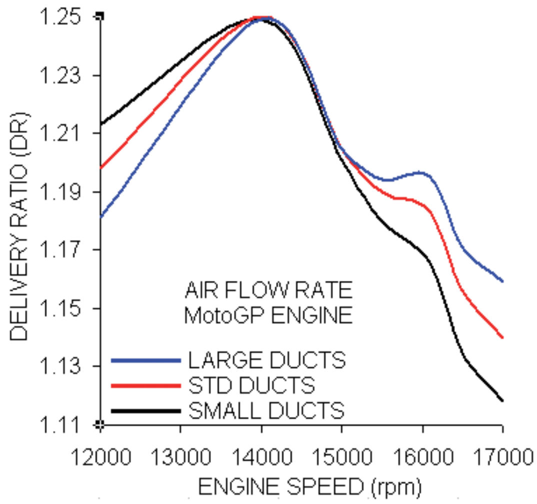

Fig.22 MotoGP engine airflow (DR) with alternate ducting.

the mathematics of unsteady gas dynamics, valve lift profile design, valvetrain dynamic analysis, cylinder pressure analysis, discharge coefficient analysis, and specific time area calculations are packaged nowadays into computer software that you can effectively use for design and thereby gain total understanding of the theoretical concepts which are discussed here.

Why is this empiricism so important if all I have to do is buy a complete engine simulation, like I use here, and just keep stuffing the input data numbers of the engine and duct geometry into it until I come up with the required engine design?

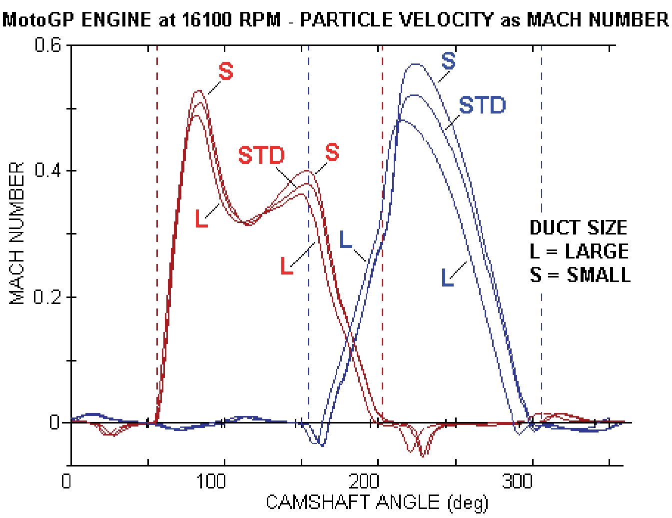

Fig.23 MotoGP engine duct particle velocities (Mach no).

$$\text{Inlet Valve(s) AreaRatio} \quad K_{iv} = \frac{A_{iv}}{A_b} = \frac{n_{iv} \times \frac{\pi}{4} \times D_{iv}^2}{\frac{\pi}{4} \times B^2} = \frac{n_{iv} \times D_{iv}^2}{B^2} \qquad (25)$$

$$\text{Mean Gas Velocity (intake valve) (m/s)} \quad K_L = \frac{C_p}{K_{iv}} \qquad (26)$$

Fig.24 The Mean Gas Velocity theory (Kiv and Kl) Eqns.25-26.

This is especially the question as some of these engine simulations come with built-in automatic performance optimisers [7]. The answer is that you can keep stuffing numbers as input data into an engine simulation, where the data involved number in the hundreds if not thousands, but you may never attain a design as well optimised as the exemplar MotoGP engine [2]. The reason is that it was initially created in the 4stHEAD software using the above empiricism to reach a 'matched' design which employed real valve lift profiles that not only provided valvetrain dynamic stability but also a satisfactory cam design and manufacture potential. It was only when all such design considerations were satisfied that it was run through the engine simulation to check that, as shown in Fig.21, (a) the design target was achieved and, (b) an effective power and torque characteristic extended over the usable speed range. All readers, be they experts or tyros, must conclude that does constitute a design process.

In short, it is through an understanding of the basics that we get the guidance to efficiently use today's sophisticated computational tools for engine design.

REFERENCES

[1] G.P. Blair, sidebar contribution on combustion in diesel engines, Race Engine Technology, Volume 5, Issue 2, May 2007 (see www.highpowermedia.com).

[2] G.P. Blair, "Steel Coils versus Gas", Race Engine Technology, Volume 5, Issue 3, June/July 2007 (see www.highpowermedia.com and download at www.profblairandassociates.com).

[3] G.P. Blair, "Design and Simulation of Four-Stroke Engines", Society of Automotive Engineers, 1998, SAE reference R-186.

[4] G.P. Blair, "Design and Simulation of Two-Stroke Engines", Society of Automotive Engineers, 1996, SAE reference R-161.

[5] 4stHEAD design software, Prof. Blair and Associates, Belfast, Northern Ireland (see www.profblairandassociates.com).

[6] G.P. Blair, W.M. Cahoon, "Life at the Limit", Race Engine Technology, Volume 1, Issue 004, Spring 2004.

[7] Virtual 4-Stroke engine simulation, Optimum Power Technology, www.optimum-power.com