Note 13

MECHANICAL SPEED LIMITS

Part I : Piston Stressing

Part II : Piston-Ring Flutter as a limit to Piston Speed

Part III : Limiting Mean Valve Speed

This Note is an update and extension of theories discussed in two privately-issued papers by DST, as follows:-

Piston Engine Speed Limits	February 1987	(Ref. 748);
Optimum Bore/Stroke Ratio for Piston Engines	June 1991	(Ref. 749).

Optimum Bore/Stroke ratio

The consequence of mechanical speed limits to the "Bottom-End" of a piston engine (as discussed in Part I and Part II) and to the "Top-End" (as set out in Part III) is that there is an Optimum Bore/Stroke ratio to maximise volume-specific power at a chosen number of cylinders at a given state of Design and Material technology.

This Optimum is discussed in Note 21.

From Newton's 2nd Law of Motion:-

$$\text{Acceleration} = \text{Force/Mass}$$

$$\therefore \quad \text{Acceleration} = (\text{Stress/Density}) \times (\text{Area/Volume}) \qquad \text{Eqn. 1.}$$

In the general case, Acceleration, Area and Volume are expressed in terms of a Characteristic Length and Time, so that:-

$$\{\text{Length}/(\text{Time})^2\} \propto (\text{Stress/Density}) \times \{(\text{Length})^2/(\text{Length})^3\}$$

resolving to:-

$$\textbf{Characteristic Speed} \propto \sqrt{(\textbf{Stress/Density})} \qquad \text{Eqn 2.}$$

Therefore, in the absence of other limits, the constructional materials available at a particular date will limit the speed of machinery, via the Stress permitted for a required fatigue life.

In a particular machine, the dimensions used to determine Acceleration, Area and Volume must be based on the critical parts. Practical experience is the guide here. In the highly-loaded – but also highly-successful – Cosworth type DFV V8 3L racing 4-stroke poppet-valve piston engine, designed originally in 1966 and developed steadily until 1982, it was found in the years of overhaul experience that parts lives were as follows (59):-

- Pistons (Al-alloy) — 500 racing miles (800 km) – About 2 races plus some practice
- Valve Springs (Steel Coils) — ." ." ."
- Valves (Steel) — 4 or 5 races
- Valve Spring Retainers — 7 or 8 races
- Bearings (Copper-Lead) — Full season (16 races)
- Connecting Rods (Steel) — ⎤Apparently no life limit
- Crank (Steel) — ⎦ i.e. parts sized by stiffness required, not by stress

The *piston* and *valve-spring* lives therefore determined the routine rebuild time for the DFV engine. Engines prior to the DFV and post-WW1, particularly when supercharged, or post-DFV when turbocharged, also had the piston as the limiting component of the "Bottom-End". By the '90's, when normally-aspirated again, the competition was so fierce that pistons were being scrapped after only a few laps at Qualification power rating or one race with a fresh engine at "400km" race rating (419).

Consequently, where the "Bottom-End" mechanical limit on speed is concerned, it is the stressing of the piston which should be considered in the more detailed analysis (the mechanical limits which may be imposed alternatively or simultaneously by piston-ring flutter or the "Top-End" – the valve gear - are considered elsewhere).

The Theoretical Case

In the simplest analysis, considering the piston as the critical part:-

$$\text{Area} \propto (\text{Bore})^2 \propto B^2$$
$$\text{Volume} \propto (\text{Bore})^2 \times \text{Piston Height} \propto B^2 \,.\, PH$$

The Maximum Acceleration of the piston (MPD) is towards the crank at Top Dead Centre, as it is brought to rest and restarted in the opposite direction, and is given closely by:-

$$\text{MPD} \propto \text{Stroke} \times (\text{RPM})^2 \times \{1 + \text{Stroke}/(2 \times \text{Connecting-Rod Length-between-centres})\}$$

or

$$\text{MPD} \propto S \,.\, N^2 \,.\, \{1 + 1/(2 \,.\, CRL/S)\}$$

Entering these functions into Eqn 1. :-

$$S \,.\, N^2 \,.\, \{1 + 1/(2 \,.\, CRL/S)\} \propto (\text{Stress/Density}) \times B^2/(B^2 \,.\, PH)$$

$$(PH \,.\, S) \,.\, N^2 \,.\, \{1 + 1/(2 \,.\, CRL/S)\} \propto (\text{Stress/Density}) \text{ of the piston material}$$

The term { 1 + 1/(2 . CRL/S)} is fairly constant. An average number for CRL/S in racing engines from 1910 to 1998 is 2 and the variation of the term from CRL/S = 1.5 to 3 is +7% to –7% about the level at CRL/S = 2. This is the variation of stress at a given RPM, and the permitted variation of RPM at a given stress would be half the above percentages and of the opposite sign. Therefore, the expression may be treated as :-

$$(PH \,.\, S) \,.\, N^2 \propto (\text{Stress/Density}) \text{ of the piston material} \qquad \text{Eqn. 3.}$$

Assumptions regarding piston dimensional ratios.
(a): if the design of pistons leads to:-

$$PH \propto S$$

Then Eqn 3.. reduces to:-

$$S \,.\, N \propto \sqrt{(\text{Stress/Density})}$$

Or introducing Mean Piston Speed (MPS) = 2 . S . N

$$\mathbf{MPS \propto \sqrt{(Stress/Density)} \text{ of the piston material}} \qquad \text{Eqn. 4.}$$

(b): if the design of pistons leads to:-

$$PH \propto B$$

which could be the case if it is necessary to keep the skirt side-thrust rubbing pressure constant, or to keep the angularity of the piston rings constant, at some limiting values,

Then Eqn 3 becomes:-

$$\sqrt{(B \,.S)} \,.\, N \propto \sqrt{(\text{Stress/Density})}$$

also written as

$$\mathbf{\sqrt{(B/S)} \,.\, MPS \propto \sqrt{(Stress/Density)} \text{ of the piston material}} \qquad \text{Eqn. 5.}$$

to show more clearly how the simple "Mean Piston Speed Theory" of Eqn 4 is amended. This is known generally as the "Lanchester Theory", although it was not stated explicitly as such by that pioneer in 1906, but has been *derived* from his choice of an expression

$$\text{Power} \propto B^{1..5} \,.\, S^{0.5} \quad \text{which is given on p.164 of (369).}$$

Note that Lanchester made the condition that this would apply "given adequate port areas" (p.158 of 369) – a requirement often not met in subsequent designs (nowadays one would put "adequate valve-opening area x time").

<u>The Practical Case</u>

The Piston Ratios PH/B and PH/S have been examined over nearly 100 years of engine development on Figures 98/DST and 99/DST respectively. From 1906 to1955 it could be said, crudely, that PH/B was constant, averaging around 1.1, although with much scatter. From 1955, however, there has been a steady decline to about 0.5 in modern times. Similarly, the tendency in this latter period has been to reduce PH/S from about 1.1 to 0.9, having risen from about 0.7 in the preceding period. Again crudely, *if no other internal data are available*, it is probably better to assume for stress comparative purposes that PH/S is constant over the whole period at about 0.9, since PH/B certainly is *not* constant, i.e. Eqn 4 should be used rather than Eqn 5.

These two charts emphasise really that it is *not safe* to assume a fixed ratio of PH to either B or S for general conclusions about stressing limits – it should be included as a dimension in its own right. This is true particularly through the change since 1967 from high-compression 2-valve hemispherical heads with high piston crowns to the flat-topped pistons of lower mass permitted by 4-valve heads.

An analysis of Al-alloy racing piston weights (to use the usual term, although mass is the correct description) versus B and PH is given on Figure 97/DST. This includes slipper pistons (intended to reduce friction by cutting-down rubbing area) as well as full-skirted designs (Note I). The range of B is 50 to 110 mm. It will be seen that there is a strong correlation of:-

$$\text{Total Piston Weight (WPT)} \propto B^{1.5} \,.PH \qquad \text{Eqn. 6.}$$

where WPT is the whole purely-reciprocating assembly, including rings and gudgeon pin. The average error is 6% over 29 examples. Inability to scale-down over the range means that dimensional consistency is not followed in practice – a common situation in mechanical engineering, and one which Lanchester in 1906 had not sufficient examples to have been able to observe where internal combustion engines were concerned. <u>*Density* is implicit on the RHS of Eqn 6</u>.

Returning to Newton's 2nd Law and inserting Eqn. 6 gives:-

$$S \,.\, N^2 \,.\, \{ 1 + 1/(2 \,.\, CRL/S)\} \propto (\text{Stress x Area})/(\text{Density x } B^{1.5} \,.\, PH)$$

Adopting, as before, the simplification that $\{ 1 + 1/(2 \,.\, CRL/S)\}$ is nearly constant and that Area over which Stress applies $\propto B^2$:-

$$\mathbf{[(PH \,.\, S)/B^{0.5}] \,.\, N^2 \propto \text{Stress/Density of the piston material}} \qquad \text{Eqn. 7.}$$

or

$$\mathbf{[(PH/S)^{0.5}/B^{0.25}] \,.\, MPS \propto \sqrt{(\text{Stress/Density})} \text{ of the piston material}} \qquad \text{Eqn. 8.}$$

The LHS of Eqn. 7 has been titled "Piston Stressing Factor" (PSF). This has been calculated with PH, S and B in cm and N = NP, the Peak Power Speed, divided by 1000 to give a handy number (PSF has dimensions ($cm^{1.5}$ / $min^2 \cdot .10^6$)). PSF is plotted for racing engines against date on 100/DST.

Discussion of Piston Stressing Factor v. Date

PSF would be expected to rise over the years as materials of higher Stress/Density ratio became available (a general review of piston material development is given separately). Very broadly this is true, by a multiplication of 4 from 1914 to 2000. It seemed to take a few years, post WW1, before the change from cast-iron or steel pistons to Al-alloy, around 1914, had a significant effect on PSF, despite all the work done on the latter material in the War. There is a large amount of scatter, some of which is understood, i.e. some low points are push-rod (PROHV) or single-overhead camshaft (SOHC) units where the valve-gear or valve-area were limiting before the piston; some of the high points are believed to be "flash" readings (egs. Peugeot 1910 Voiturette, Delage 1926-1927 GP) or "Shelsley Sprint" or "Short Race" ratings (1936 Austin 750, 1954 BRM T15 Mk2: Note II). There is a group of Coventry-Climax engines (FPF, FWMV, FWMW) over 1959-1966 where Hassan deliberately used a valve timing which favoured mid-range torque at the expense of peak RPM (and power), and so limited peak piston stress (Note III).

The points given above being acknowledged, there is something of a dip in PSF after 1957. A possible explanation for this is that it coincided with the banning of alcohol fuel in favour of petrol. The high latent heat of evaporation of alcohol, compared with petrol, had been used since 1924 not only to cool the charge but also, as inlet pressures rose, with very-high %age alcohol content and excessive fuel/air ratio, to cool the pistons (607,468,31). To a lesser extent such methods were continued in 1954-1957 with normally-aspirated engines (Note IV). It would appear that, after the reversion to petrol, it took some time to get back to high levels of PSF, and turbocharging, even with oil-gallery-cooled pistons, delayed the recovery. The levels reached by 2000 indicate the increasing use of under-piston oil-spray cooling and reduction of life down to only 400km, i.e., pistons scrapped after only a single race.

Conclusion regarding "Rating Rules"

When Lanchester produced his 1906 paper he hoped to provide a simple rule from mechanical considerations which could be used to rate engines for power. The data given above emphasise that there is *no* simple way to predict speed from a mechanical "Bottom-End" limit across a substantial time period, because of variations in (at least):- valve-gear architecture; breathing & burning capability; piston (& ring) materials, proportions, heat-loading & cooling; and especially in the life required.

A "Rating Rule" may be found for a particular maker's design philosophy over a few years (e.g. Bastow's modified Lanchester formula for Climax engines (50)), but it is then found to be inaccurate for preceding and succeeding units.

Notes

I. The results show that slipper pistons do not in themselves reduce piston weight, because substantial inner supports are still required for the gudgeon-pin bearings, which would otherwise be carried by the full skirts.

II. The 1936 Austin 750 PSF = 1696 shown is at 10,000 RPM. For long races the speed was limited to 7,500 RPM, i.e. 44% lower stress. The 1954 BRM PSF = 1655 is at 12,000 RPM, which was very rarely reached.

III. The FPF 2.5L was a unit where CRL/S = 1.44 only , because of the way it had been enlarged from 1.5L. The stress therefore would have been 7.8% higher than at CRL/S = 2.

IV. It is interesting that the 1954-1955 Mercedes M196 (PSF = 1478 and 1569 respectively) only had one type of repeated fault – a car with a piston failure in the 1st race and a car with the same thing in the last event entered.

Note 13 Part I
Sub-Note A : Some Piston Examples

Photographs

An attached Figure gives a comparison of 3 forged Al-alloy pistons, all for NA high-compression (petrol-fuelled) engines, which illustrates the great reduction in piston mass accomplished between 1961 and 1996 by using under-piston oil-spray cooling and by also accepting life reduction from perhaps a whole season down to a single race.

The pistons shown are:-

	Date	Make	Type	v/c	PH	PH/B	PH/S
		CN x B x S mm	**NP RPM**		mm	**PSF**	
		WPT g	WPT/((B)$^{1.5}$.PH)	Difference from correlation			
Top LHS:	1961	Norton	500cc Manx	2	81	0.94	0.95
		1a/c 86 85.62 = 497cc	**7,000**			**1,159**	
		592	2.90 +4.6%				
Middle RHS	1983	Honda	F2 RA263E	4	61	0.68	1.17
& Bottom RHS		**80V6 90 52.3 = 1996cc**	**12,000**			**1,531**	
		469	2.85 +2.7%				
Top, Middle	1996	Yamaha-Judd	OX11A	4	42	0.47	0.89
& Bottom		**72V10 90 47.1 = 2996cc**	**16,000**			**1,688**	
		273	2.41 -13.1%				

(The photos also show an OX11A Ti-alloy inlet valve, IVD = 37.5 mm)

These pictorial examples demonstrate how the controlling design parameter during this 35 year period was:-

$$\mathbf{PH} \propto \mathbf{S}$$

at a value around **1**, as shown by Fig. 99/DST.
(All 3 pistons are plotted on 97/DST but only the Yamaha on 98, 99, 100/DST).

Scale Drawings

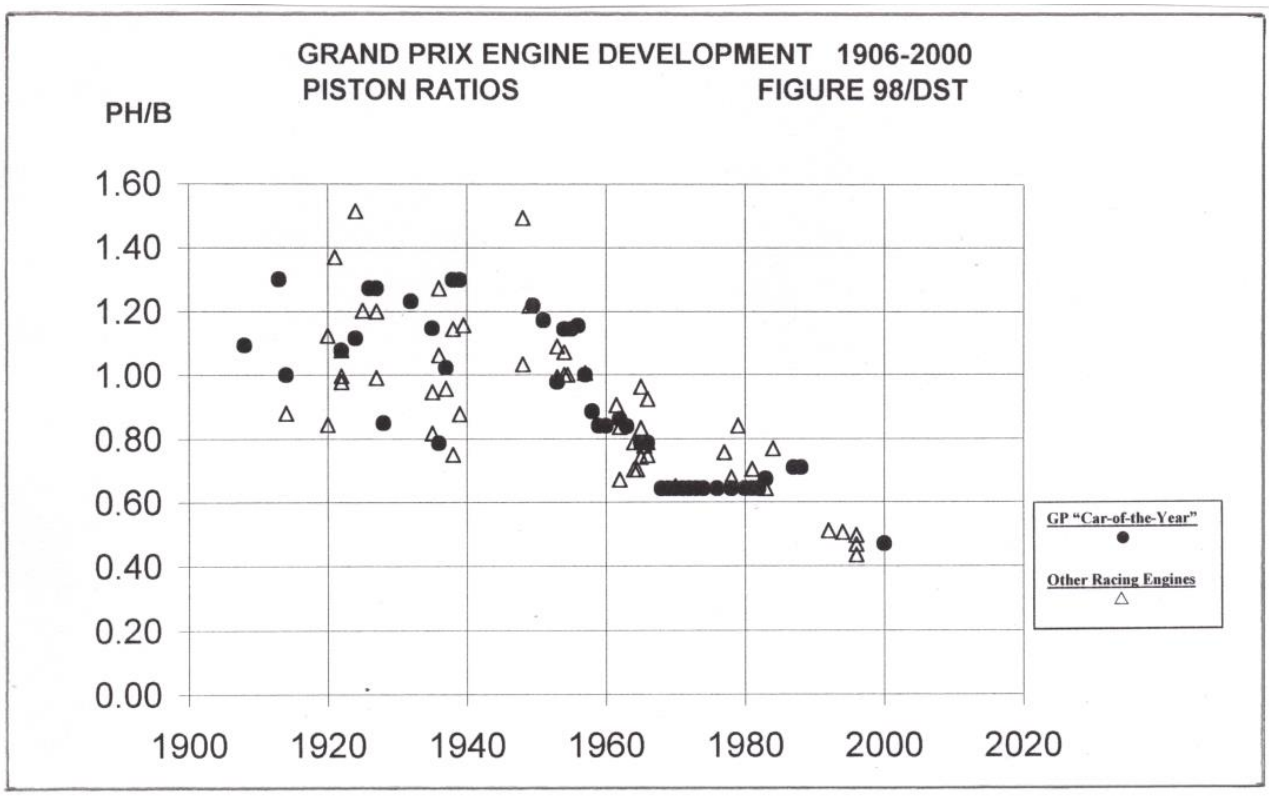
Two scale drawings are attached which give full details of:-

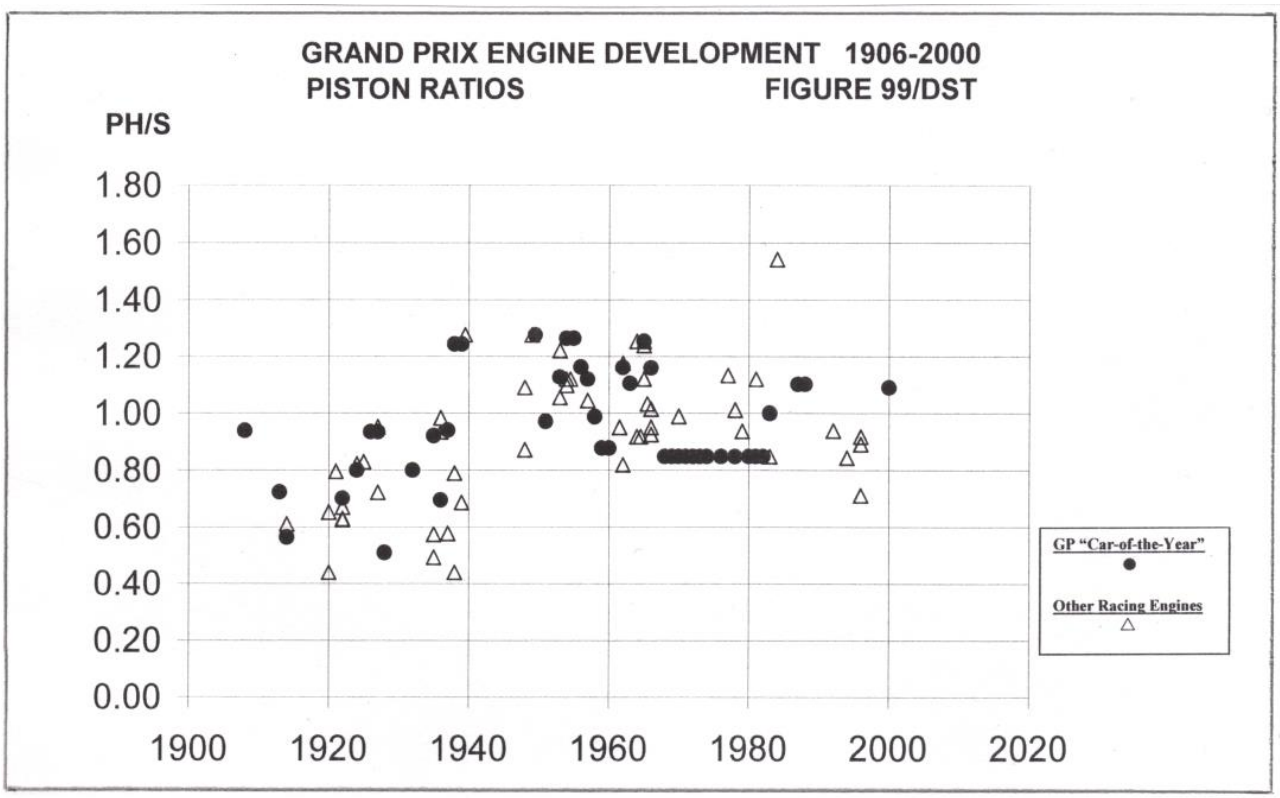
	PSF	**WPT/((B)$^{1.5}$.PH)**	**Diff. from Corrln.**
• Yamaha-Judd OX11A (DASO 674)	1,688	2.41	-13.1%
• Mugen-Honda MF301 (DASO 672)	1,318	2.82	+1.8%

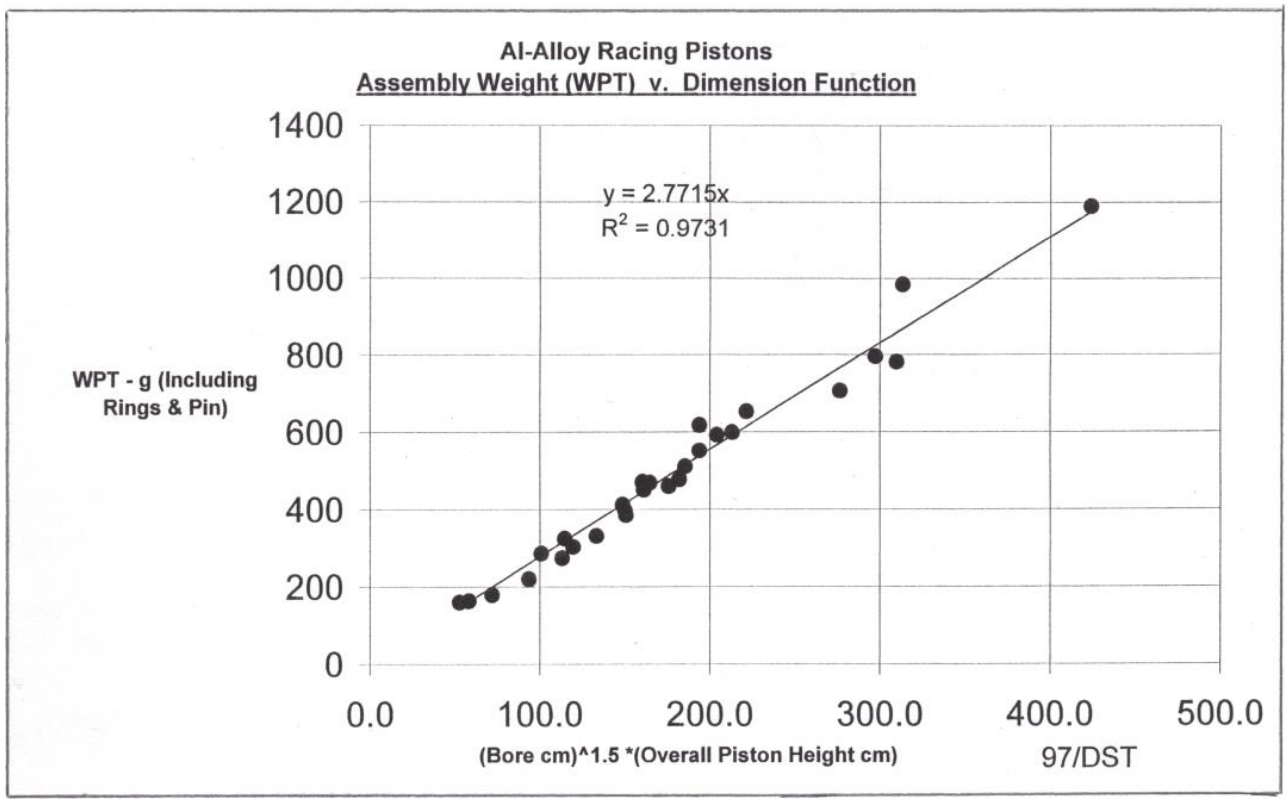
Both are 1996 pistons (and both are on 97/DST).

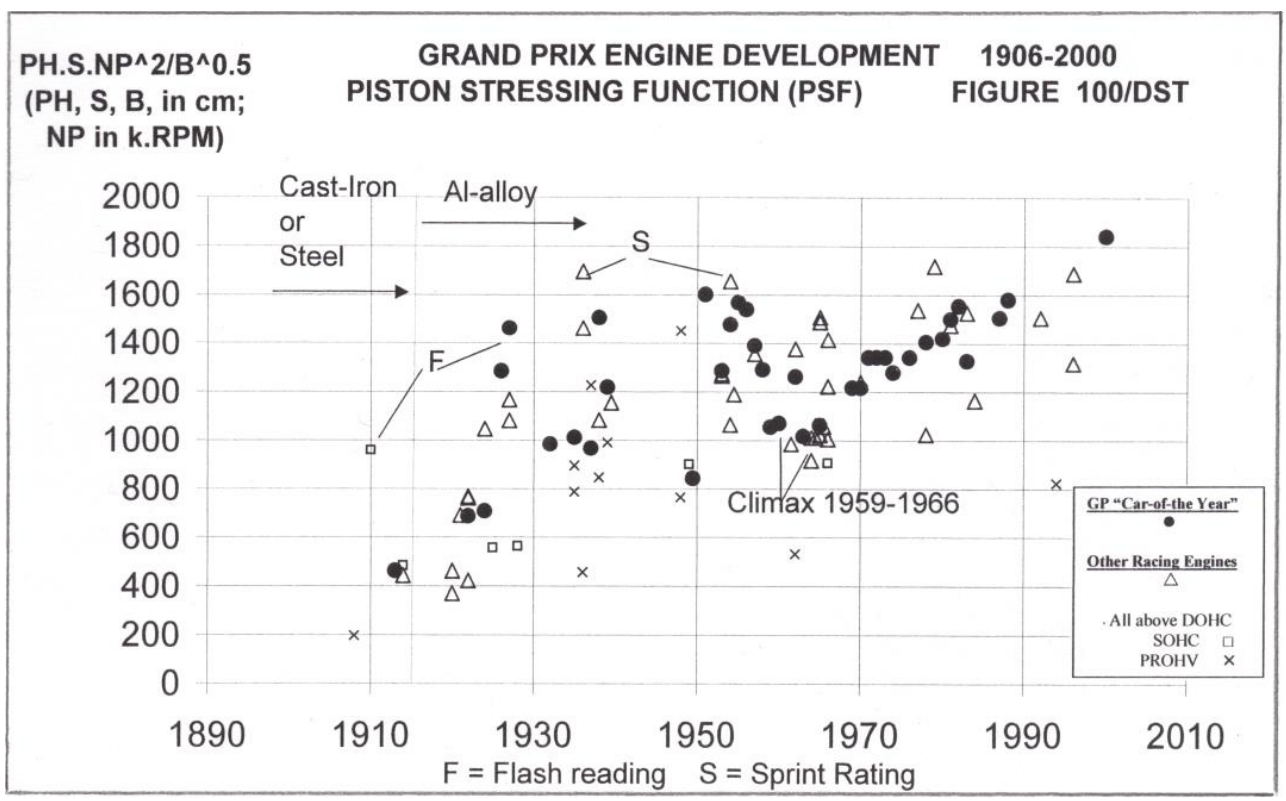
The Mugen-Honda is particularly interesting because the drawing was made of a piston taken from the winning engine at Monaco in 1996 (mounted with its Ti-alloy con-rod and given by Socheiro Honda to Mr Alfred Briggs of Derby as a mark of appreciation for many years service with Honda).

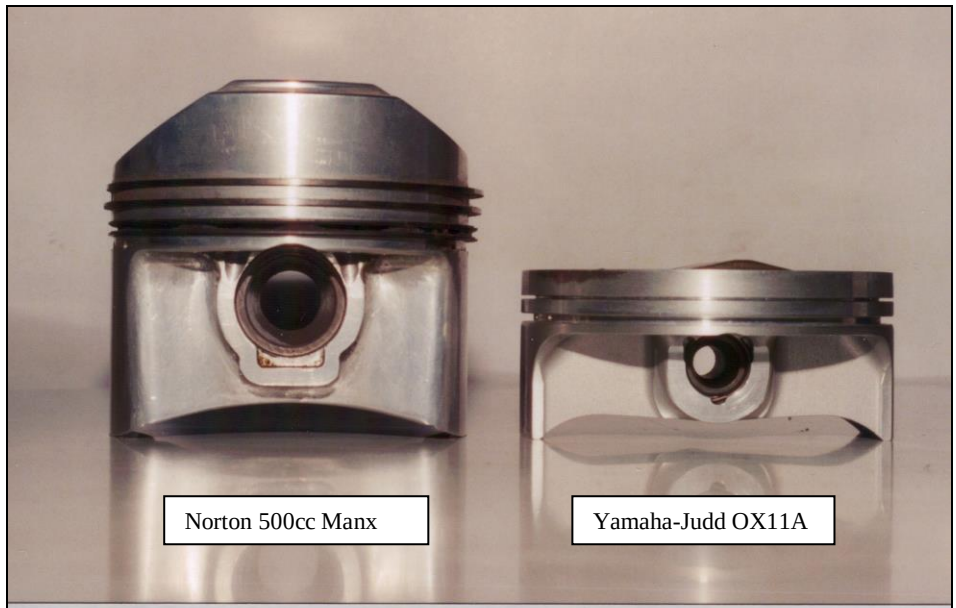
Note that the Mugen B = 93 mm and CRL/S = 2.72 were both unusually high for the period, because the engine was adapted from 3.5L to 3L in 1995 by a stroke reduction, so as to retain the existing crankcase. Compared to CRL/S = 2 the actual con.-rod dimensions would have reduced piston stress by about 5%.

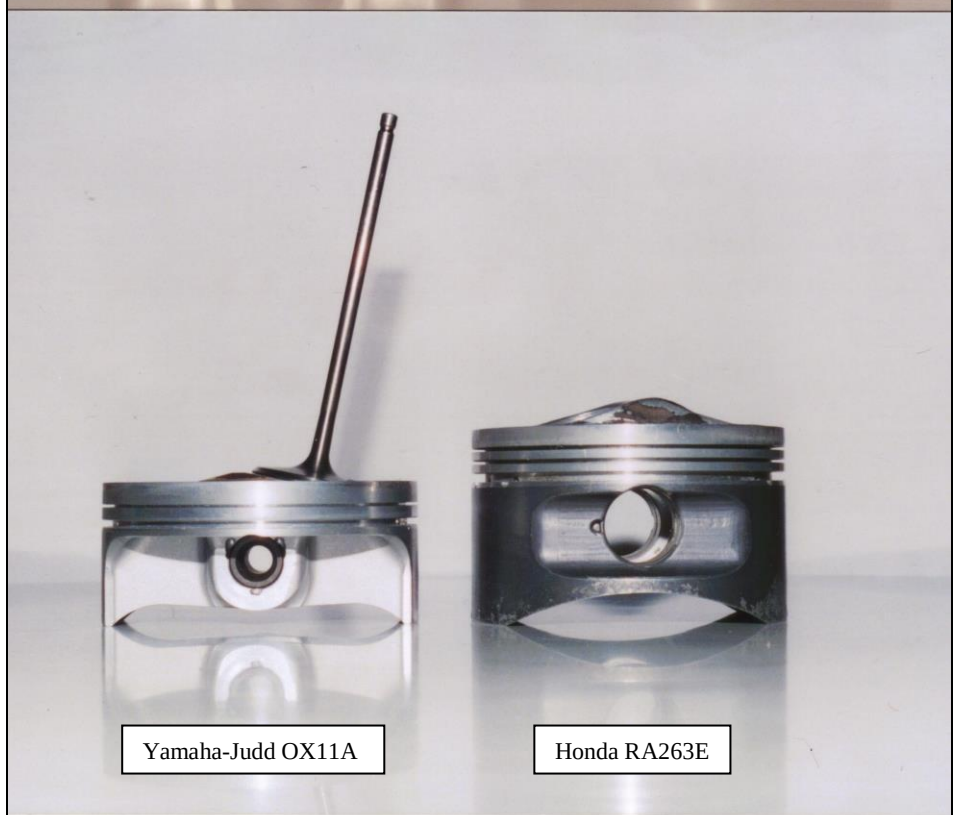

GRAND PRIX ENGINE DEVELOPMENT 1906-2000
PISTON RATIOS
FIGURE 98/DST
PH/B
1.60
1.40
1.20
1.00
0.80
0.60
0.40
0.20
0.00
1900
1920
1940
1960
1980
2000
2020
GP "Car-of-the-Year"
Other Racing Engines

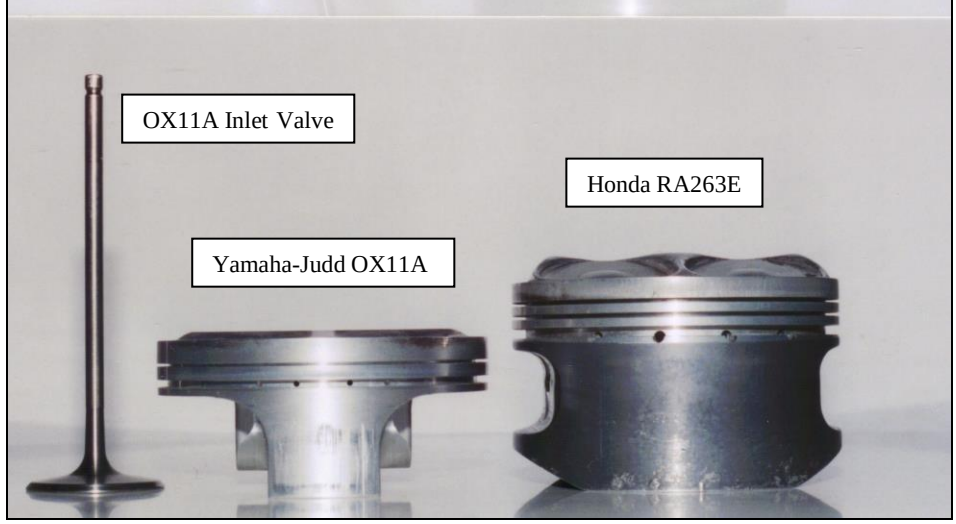

GRAND PRIX ENGINE DEVELOPMENT 1906-2000
PISTON RATIOS
FIGURE 99/DST
PH/S
1.80
1.60
1.40
1.20
1.00
0.80
0.60
0.40
0.20
0.00
1900
1920
1940
1960
1980
2000
2020
GP "Car-of-the-Year"
Other Racing Engines


Al-Alloy Racing Pistons
Assembly Weight (WPT) v. Dimension Function
y = 2.7715x
R² = 0.9731
WPT - g (Including Rings & Pin)
1400
1200
1000
800
600
400
200
0
0.0
100.0
200.0
300.0
400.0
500.0
(Bore cm)^1.5 *(Overall Piston Height cm)
97/DST


PH.S.NP^2/B^0.5
(PH, S, B, in cm;
NP in k.RPM)
GRAND PRIX ENGINE DEVELOPMENT 1906-2000
PISTON STRESSING FUNCTION (PSF) FIGURE 100/DST
Cast-Iron or Steel
Al-alloy
S
F
Climax 1959-1966
GP "Car-of-the Year"
Other Racing Engines
All above DOHC
SOHC
PROHV
2000
1800
1600
1400
1200
1000
800
600
400
200
0
1890
1910
1930
1950
1970
1990
2010
F = Flash reading S = Sprint Rating


Norton 500cc Manx
Yamaha-Judd OX11A


Yamaha-Judd OX11A
Honda RA263E


OX11A Inlet Valve
Honda RA263E
Yamaha-Judd OX11A

YAMAHA - JUDD OX11A
1996

DST 27 FEB 2001
Amended 14 APL 2005
DASO 674

PISTON LENT BY COURTESY OF MR JOHN COOPER VIA MIKE TAFT.

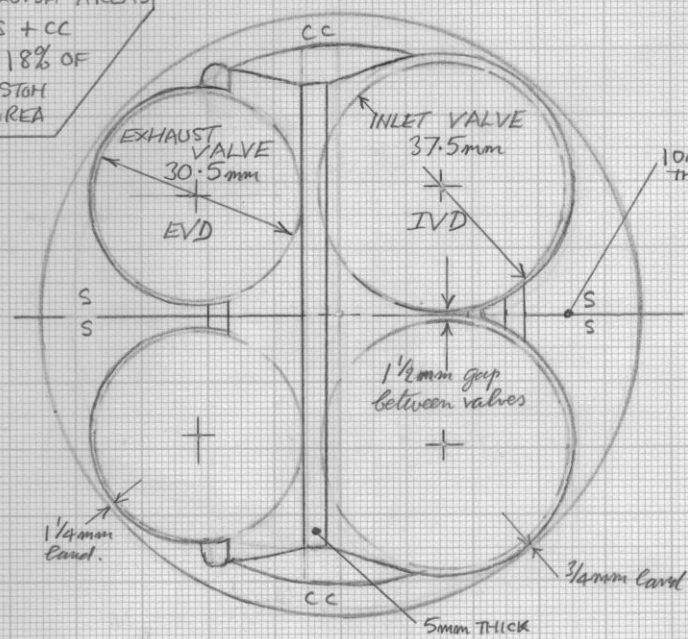
V10 3 LITRE BORE 90mm, STROKE ≃ 47.1mm } SWEPT VOLUME ≃ 2996 cc

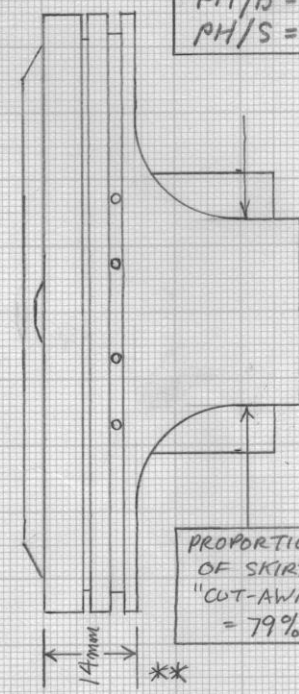
4 VALVES / CYL AT 25° INCLUDED ANGLE (VIA)

FORGED Al-ALLOY PISTON

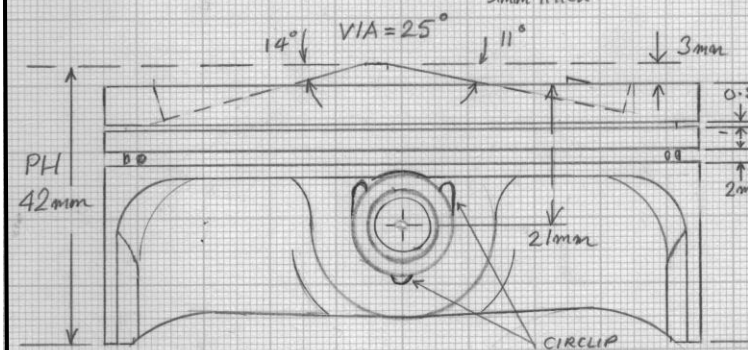
SQUISH AREAS SS + CC ≃ 18% OF PISTON AREA

PH/B = 0.47
PH/S = 0.89





PROPORTION OF SKIRT "CUT-AWAY" = 79%



0.8mm GROOVE × 3.5mm
RING ω ∴ 0.030" (0.76mm)
2mm GROOVE × 3mm

PSL * 39mm

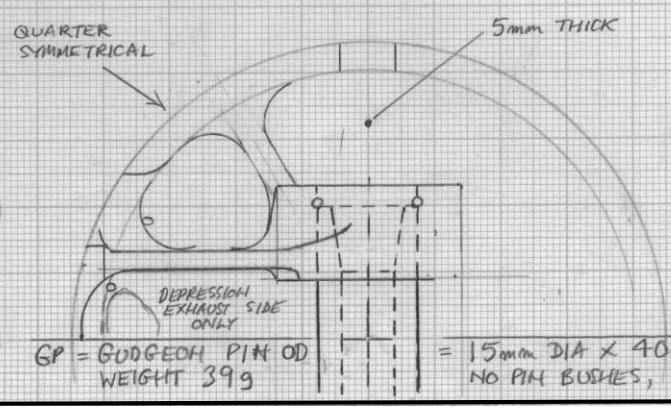
ω / STROKE = 1.6%

AREA RATIO	
2×INLETS / PISTON	= 0.347

DIA. RATIOS	
INLET VALVE / BORE	= 0.417
EXHAUST VALVE / INLET VALVE	= 0.813

WEIGHT OF PISTON + GUDGEON PIN & CLIPS = 258g.
ESTD. WEIGHT of 2 RINGS = 15g. (5+10 oil)
ESTD. WEIGHT ASSEMBLY WPT = 273g

* PSL / BORE	= 0.43
** FULL DIA. LENGTH / PSL	= 0.36



GP = GUDGEON PIN OD = 15mm DIA × 40mm LONG
WEIGHT 39g
NO PIN BUSHES, RETAINED BY 1mm OD WIRE CIRCLIPS

DST 6 JAN 2001
AMENDED 16 JAN '01
" 14 APL 2005
DASO 672

MUGEN-HONDA
ME301
MAY 1996

PISTON & CON-ROD LENT BY COURTESY OF MR ALF BRIGGS VIA MIKE TAFT.

PH/B = 0·44
PH/S = 0·92

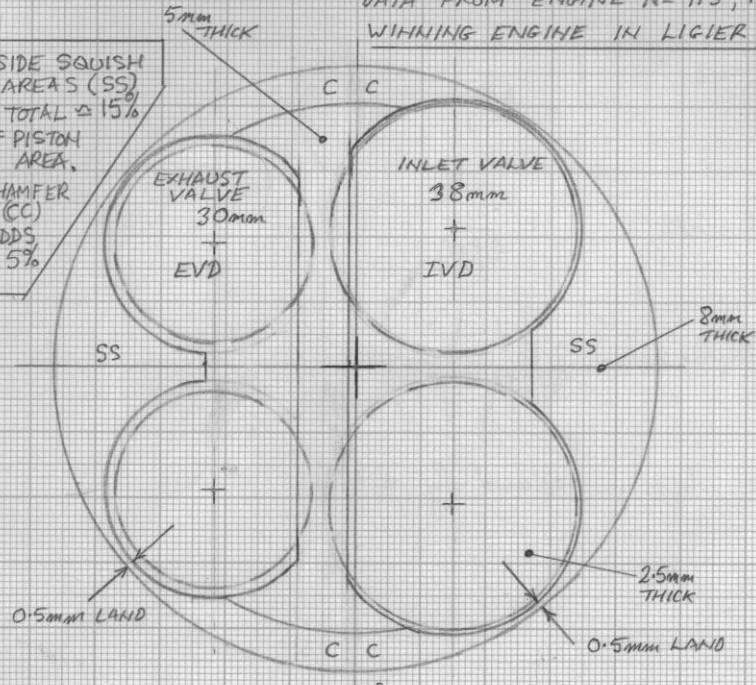
72° V10 BORE 93 mm, STROKE ≈ 44·1 } SWEPT VOLUME ≈ 2996 cc

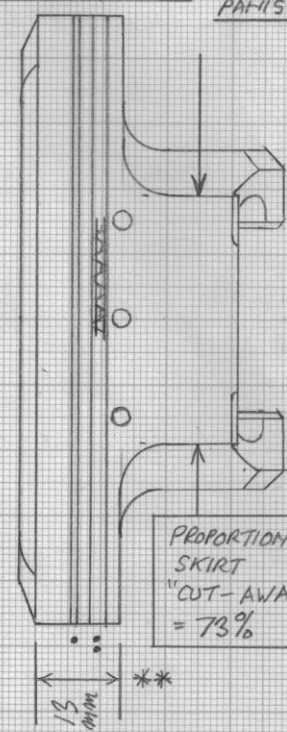
4 VALVES / CYL. AT 23° INCLUDED ANGLE

DATA FROM ENGINE Nº 115, PISTON Nº 7

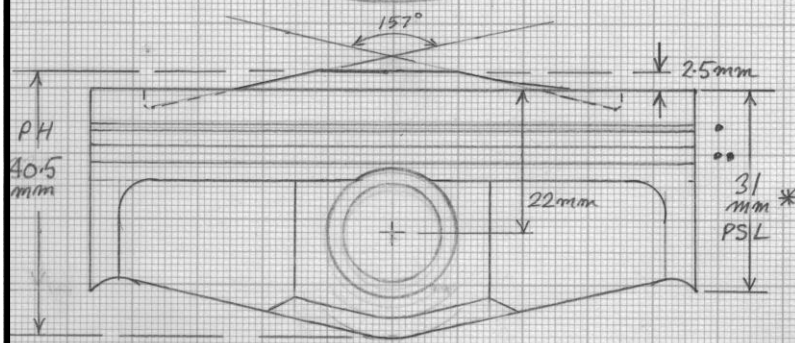
WINNING ENGINE IN LIGIER, MONACO, DRIVER OLIVIER PANIS

SIDE SQUISH AREAS (SS) TOTAL ≈ 15% OF PISTON AREA.
CHAMFER (CC) ADDS 5%





PROPORTION OF SKIRT "CUT-AWAY" = 73%



DIA. RATIOS

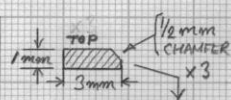
$\frac{\text{INLET VALVE}}{\text{BORE}} = 0.41$

$\frac{\text{EXHAUST VALVE}}{\text{INLET VALVE}} = 0.79$

WEIGHT OF PISTON ASSY. (INCL. RINGS, GUDGEON PIN, CIRCLIPS) BELIEVED TO BE 324g.

CON-ROD
LENGTH BETWEEN CENTRES (CRL) = 120mm

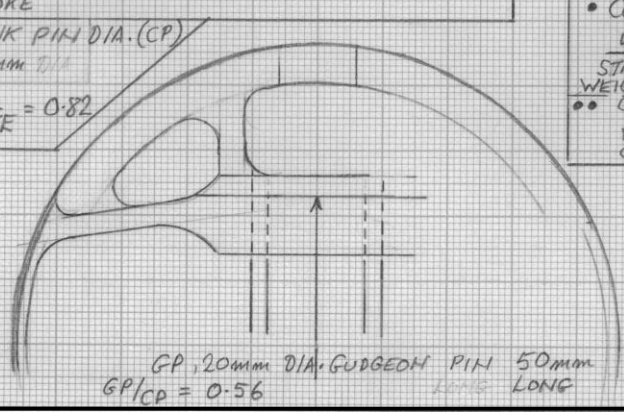
$\frac{\text{CRL}}{\text{STROKE}} = 2.72$



LESS THAN 0·002 INCH CLEARANCE
• COMPRESSION RING = 1·0mm WIDE (w)
$\frac{w}{\text{STROKE}} = 2.3\%$ { FERROUS 3·0mm RADIAL DEPTH, RECTANGULAR SECTION
WEIGHT ≈ 5·7g, DENSITY = CAST IRON
•• OIL CONTROL RING, 1/4mm LANDS WITH SPRING, 2·0mm O.A. WIDTH
0·003 INCH CLEARANCE

CRANK PIN DIA. (CP) 36mm

$\frac{\text{CP}}{\text{STROKE}} = 0.82$



$* \frac{\text{PSL}}{\text{BORE}} = 0.33$

$** \frac{\text{FULL DIA LENGTH}}{\text{PSL}} = 0.42$

GP, 20mm DIA. GUDGEON PIN 50mm LONG, NO PIN BUSHES (WIRE) RETAINED BY CIRCLIPS, EACH END PLAIN 15mm BORE

GP/CP = 0·56

Piston Ring Flutter as a limit to Piston Speed **DST 11 February 2003**

Up to the early 50s piston acceleration – or rather *de*celeration of the piston-rings at the end of the exhaust stroke – could form a limit to engine speed. This was because the ring momentum at the minimum producible rectangular-section (plain) ring axial width (and therefore minimum mass) could overcome the residual gas pressure so that the ring was thrown to the top of its groove. Gas pressure then could no longer reach the inside of the ring to compensate for that on the outside and the ring would be forced inwards. Apart from the resulting gas blow-by, losing power, degrading the oil and pushing it out of the engine, the ring radial vibration with RPM – "flutter" – would lead to very rapid fatigue failure of the cast-iron material then used. Over-revved engines would be found to have their rings broken into many pieces (see Sub-Note A and sketch on Page 5). Obviously, all the bad effects given above would then be magnified.

The basic relation

When a piston-ring is on the point of "topping-out" in its groove, then, by Newton's 2nd Law:-

(Ring Mass) x (Maximum Piston Deceleration, MPD)
= (Pressure Difference across Ring, Δp) x (Ring plan area)
and (Ring Mass) = Ring(Density,DR) x (Axial Width,w) x (plan area)
so

$$DR \,.\, w \,.\, MPD = \Delta p$$

w.MPD = Δp/DR at flutter onset

The Hepworth relation

The piston and ring manufacturer J.Hepworth must have discovered that Δp does not vary greatly between engines at the critical condition because he published detailed recommendations in June 1953 which could be translated into:-

w.MPD = Constant = 5000 inch . ft/sec^2 (say, 4000 mm.g)

for plain cast-iron rings at flutter onset (745).

The smallest width which could be made at that date was 3/64'' (1.2mm), which set a limiting value of MPD at 106,000 ft/sec^2 (32,210 m/sec^2 or 3300g). This was misunderstood sometimes as an *absolute* limit, instead of a figure which would rise when technology permitted thinner rings.

Ring geometry and piston speed

By substituting:-

$$MPD = [(Stroke,Smm) \times (RPM,N)^2 \times (1 + 1/(2 \,.\, CRL/S))]/(1.789 \,.\, 10^6) \quad g$$

where CRL = Connecting-Rod Length between bearing centres
and Mean Piston Speed, MPS = (Smm . N)/30,000 m/s
and using an average value of CRL/S = 2, then the Hepworth relation (**w.MPD = 4000 mm.g**) can be converted into:-

MPS.√(w/S) = 2.5 m/s nearly enough.

Therefore, the limits to MPS set by a satisfactory plain cast-iron piston-ring life are:-

w/S %	**4**	**3**	**2**	**1.5**	**1**
MPS m/s	**12.5**	**14.4**	**17.7**	**20.4**	**25.0**

Experimental results

In Sub-Note B are given results for 9 engines, of a wide variety, in which the plain iron rings were known to be at or over the flutter onset.

The averages are:-

w.MPD = 4060 mm.g and **MPS.√(w/S) = 2.52 m/s**

Hepworth would have known of the data on 3 of these units when formulating his relation.

If the density of iron is entered into his relation, the value of **Δp** can be calculated as **44 psi (3 Bar)**, regardless of whether the engine is Normally-Aspirated (NA) or Pressure-Charged (PC). This is above the exhaust pressure at Top Dead Centre where MPD is a maximum, but it does not approach the maximum cylinder pressure. It is probably the averaged end result of gas having to pass up and down the very-narrow annular clearance communicating from the combustion-chamber to the ring groove. The high value of **w.MPD** for the 1939 Mercedes-Benz in Sub-Note B may be a consequence of the top ring being rather low down on the piston (4) so that Δp is "trapped" at a higher-than-typical level.

For very short durations engines could be operated beyond the flutter boundary, e.g. the 1936 Austin 750cc listed in Note Ersatz 16 Part I was run up to 10,000 for Shelsley hill-climbs, a matter of only a few seconds at max. speed before the engine would be overhauled.

The Dykes' Ring

It is ironic that, by the date Hepworth published his recommendations, the invention by Prof. Dykes of the **L**-section ring in the late 40s (174) had already enabled designers to avoid the operational limits set by plain rings. The inner leg of the **L** was dimensioned to "top-out" first in a closer-clearance groove so as to permit the outer ring proper (the vertical part of the **L**) to retain its pressure backing and continue to seal the cylinder without self-destructive vibration (see sketch on Page 5). Naturally this was a very popular solution in the 50s and 60s in UK racing engines (see Sub Note C). Possibly the last car engine to use the type was the Ford-Cosworth FVA of 1966 (583) but the Dykes' ring was still fitted to some GP racing 2-stroke motor-cycle engines into the mid-80s (755)*.

In the literature of the 60s the Dykes' ring was often described as "pressure-backed", which was a misunderstanding of the basic ring operation since *all* rings are "pressure-backed" until topping-out occurs. However, because its pressure-backing was always assured, the Dykes' ring could be made with lower radial spring pressure and so reduced friction. Ref. (754) describes how this was carried to excess in the 1965 Climax and blow-by increased again so that Jimmy Clark nearly lost the British GP of that year by using up all the oil in his Lotus.

Thinner plain rings

As a less expensive alternative to the Dykes' ring, thinner plain rings were made from the early 60s onwards.An extreme example of this was used in the late 1964 Honda RC114 IL2 50cc engine. This had MPD around 7300g and, judging from a full-size photo in (75), w = 0.5mm for a product of about 3600mm.g. The Ford-Cosworth DFV 3L, introduced in 1967, never used Dykes' rings (746) and the 1983 development was able to run at 5650g at peak power with a plain top ring in stainless-steel (Mo-filled) reduced to 0.030'' (0.76mm)(746). The product **w.MPD** therefore was 4300mm.g for a material slightly less dense than iron. The value of **w/S** was 1.2% and **MPS.√(w/S)** = 2.62.

Engines of the 90s

Values of MPD continued to rise in GP engines through the 90s as B/S ratios were increased, but 'w' did not decline in inverse proportion. Two examples are known:-

Engine	Data Source	Smm	w mm	w/S	N	CRL/S	MPD	MPS
					w.MPD		**MPS.√(w/S)**	
1996 Mugen-Honda MF301 72V10 3L								
	(672)	44.1	1	2.3%	15,000RPM	2.72	6565g	22.05m/s
					6565 mm.g		**3.32 m/s**	
1996 Yamaha-Judd OX11A 72V10 3L								
	(674)	47.1	0.762	1.6%	16,000RPM	≅ 2.2	8271g	25.12m/s
					6303.mm.g		**3.20 m/s**	

While there is good agreement of the critical factors between these engines, both with plain rings, the Mugen ring known to be ferrous, they were running at **w.MPD** *60% higher* than the relationship derived from earlier units. The 1998 Ilmor-Mercedes-Benz FO110G is known to have run at 8500g (559) but 'w' is unknown. All these rings are mounted high up on the piston. Modern ring material is about 10% less dense than cast-iron (606), which adds that proportion to the limiting product **w.MPD**.

It is possible that, with improved materials, the rings are *allowed* to flutter and can still survive for a life required of only 400km and that the really-powerful sump-scavenging and de-aeration systems now used (69), with improved oil, still allow the lubricant to do a satisfactory job for the less-than-2 hours needed, despite blow-by.

Cases are known in recent years where piston-ring-flutter has been acknowledged: in the 1989 Canadian GP, run in rain, the Honda V10 3.5L engine of a McLaren failed when leading because on-off use of the throttle to keep control on the slippery track led to ring flutter and blow-by which caused excessive oil loss (727); the 1994 Peugeot V10 3.5L engine, also in a McLaren, lost all of its 15L of oil at Monaco (727), almost certainly because of ring-flutter since, not long afterwards, a McLaren-Peugeot caught fire on the British GP grid and it was admitted that ring-flutter had forced oil loss onto the exhaust (574).

There is a further possibility to explain the modern high **w.MPD** factors – that the engines now rotate in the normal running range so much faster than the natural frequency of radial vibration of the ring that it cannot resonate, i.e. flutter.In this situation, dropping the RPM, as might occur while waiting on the grid, at slow corners or in slippery conditions, could bring the ring into its flutter region. That would explain the Honda and Peugeot cases cited above.

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*

2-stroke piston-rings cannot flutter, because the outward stroke is always under compression. The use of a Dykes' ring in such engines will have been to reduce friction compared with a plain ring having high radial pressure.

The Fiat – Cappa piston-ring solution

Looking back to the standard Fiat piston-ring arrangements of 1922-1927 provides a clue to another method of avoiding ring flutter. In these engines, at the initiative originally of Guilio Cappa, the rings were dimensioned so that they bottomed in their grooves before the piston touched the wall. There were *double* plain rings in each groove, each with w = 2.5 mm, 2 double sets per piston, plus double oil scrapers. Undoubtedly the intention was to reduce friction, the cylinders being steel and the

rings almost certainly cast-iron. Since the effective mass of the top ring was equivalent to 5 mm width, the operating **w.MPD** was far into the flutter region described above, but clearly no flutter occurred since both the 1922 and 1923 engines raced successfully at high speed over 800km at Monza. The values of **w.MPD** were 8500 and 9400 mm.g respectively (66).The 1927 engine, which won a short but fast race at Monza in that year ran at no-less than **14,000 mm.g** (66).

The explanation of these very high-but-innocuous values of **w.MPD** must lie in the inability of the rings to flutter radially at destructive amplitude because of their near-filling of the groove.

There is another piece of evidence concerning this type of ring-groove relationship. When the 1931 Rolls-Royce 'R' engine was in excessive oil consumption trouble, undoubtedly due to ring flutter (see Sub Note B), the Development Engineer, Cyril Lovesey, recorded afterwards:-

"We had been told by various piston ring manufacturers that considerable reduction in oil consumption could be obtained by rings having only a few 1000ths of an inch between them and the back of the ring groove – in other words the rings practically filling the groove – but we could find no appreciable benefit from this" (615).

This shows that, although at that date ring-flutter was not understood – that came about gradually during the mid 30s (e.g. 626) – a way of avoiding it *had* been deduced for some cases and quite likely from the Fiat experience, although not found effective in the "R" engine. It is not entirely clear how that unit's problems *were* solved (there being no reference to narrower rings), although extra scrapers and a larger sump were fitted and the consumption reduced to 1/10th of the early tests (615).

However, the groove-filling-ring solution appears to have been neglected post the Fiats as far as many racing engines were concerned, hence the limiting situations described in Sub Note B. The Mercedes-Benz M196I (300SLR) sports-racing engine is shown very clearly by a good section drawing in (468) to have *2 upper rings "chock" in their grooves*, which is *not* inaccurate draughtsmanship because the 2 lower rings are shown *with* radial clearance. The data for this unit are:-

Engine	Data Source	Smm	w mm	w/S	N	CRL/S	MPD	MPS
					w.MPD		**MPS.√(w/S)**	
1955 Mercedes-Benz M196I IL8 3L								
	(468)	78	2	2.56%	7,500RPM	1.86	3112g	19.50m/s
					6224 mm.g		**3.12 m/s**	

The 1954 M196 GP engine, from which the 300SLR was derived, had similar factors. It is *possible* – the data are not complete – that the pre-WW2 Mercedes engines were designed the same way.

It is known that, in the early 80s, highly-tuned Ford V8 7L stock-car engines, revving to 6,000 RPM although standard production 5/64'' rings had to be retained by the rules in an engine with S = 3.78'' so that w/S = 2.1% and MPS = 19.2 m/s so **MPS.√(w/S) = 2.76**, had *shims fitted behind the rings* to reduce radial clearance to only 0.005'' (1/10th % of the bore) (220). Clearly these were flutter-stoppers.

It may be that the tight ring-groove design was revived in the 90s in some GP engines as another way to beat the **w.MPD** limit. However, there is no hint of this in the literature as reviewed by the author and the Mugen-Honda MF301 quoted earlier had a ring of 3 mm radial depth in a 3.5 mm groove, i.e., *not* "chock".

Sub Note A

The famous "Motor Sport" correspondent Denis Jenkinson reported on the 1953 German GP practice as follows:-

"*…the Belgian Ferrari engine"* (IL4 Type500 2L) "*was spread all over the floor of its lock-up. It had lost a lot of power and when the block was lifted it was found that all the piston rings had disintegrated into tiny pieces, a source of wonder to the owners, but a sign of over-revving to the Ferrari mechanics*" (750).

Ref. (752) gives Harry Mundy's report on a tuned-up Ford Zodiac road-car engine (IL6 2.6L) in which higher RPM had been used regularly until excessive blow-by was noticed and :-

"*…on stripping the engine it was found that rings were broken in each of the pistons*".

See also Sub Note B for the factors of these engines.

Sub Note B

Experimental Results

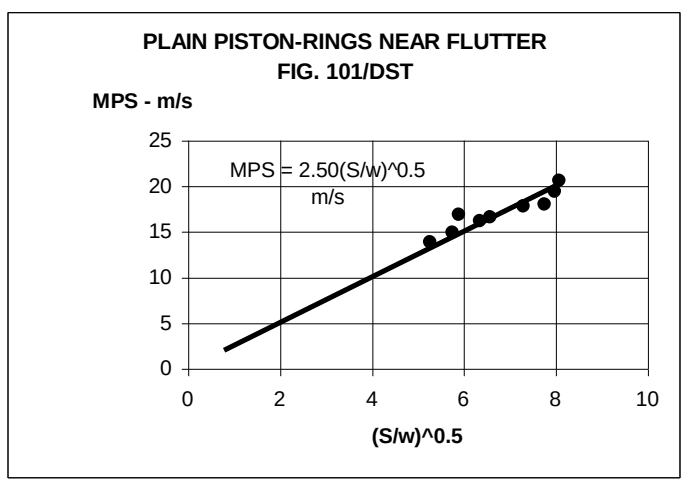
Data collected from engines in which the plain iron rings were known to be at or over the flutter limit have been tabled as follows (NA = Normally-Aspirated; PC = Pressure-Charged):-

Engine	Data Source	Smm	w mm	w/S	N RPM	CRL/S	MPD g	MPS m/s
					w.MPD mm.g	**MPS.√(w/S) m/s**		
1. 1931 Rolls-Royce 'R' 60V12 36.6L PC - Early Tests								
	615,766	167.64	2.778	1.66%	3200	1.706	1241	17.88
					3448	**2.30**		
2. 1936 Austin IL4 750cc PC - Long-Distance Rating								
	197	65.09	1.5	2.30%	7600	1.951	2639	16.49
					3959	**2.50**		
3. 1939 Mercedes-Benz M163 60V12 3L PC								
	4,30,468	70	2	2.86%	7200	2.257	2479	16.80
					4957	**2.84**		
4. 1950 BRM T15 135V16 1.5L PC – Early Tests – Before fitting Dykes' rings								
	174	48.26	1.191	2.47%	10,000	2.172	3318	16.09
					3952	**2.53**		
5. 1951 Jaguar XK120C IL6 3.4L NA Early Tests – Before fitting Dykes' rings								
	107,747	106	1.984	1.87%	5000	1.858	1880	17.67
					3729	**2.42**		
6. 1953 Ferrari Type 500 IL4 2L NA – at RPM permitted (This is the unit described in Sub Note A (750))								
* It is assumed that w = the minimum then producible in the UK, i.e. 3/64inch								
	80,749	78	1.191*	1.53%	7900	1.821	3468	20.54
					4132	**2.54**		
7. 1954 Coventry Climx FWA IL4 1100cc NA – Early Tests – Before fitting Dykes' rings								
	132,133	66.675	2.381	3.57%	6200	1.8095	1828	13.78
					4353	**2.60**		
8. 1960 Ford Zodiac modified IL6 2.6L NA (Described in Sub Note A)								
	752	79.4	2.381	3.0%	5600	1.836	1772	14.84
					4220	**2.57**		
9. 1983? US Racing Rolls-Royce Merlin 60V12 27L PC – Before fitting Dykes' rings								
	753	152.4	2.381	1.56%	3800	1.693	1593	19.30
					3791	**2.41**		

It is probable that Hepworth used Nos 2, 4, and 5, *inter alia* in developing his relation, as described in the main text. Both PC and NA units appear in this data and the averages are:-

4060 mm.g **2.52 m/s**

These experimental results are plotted as **MPS** v. **√(S/w)** below.



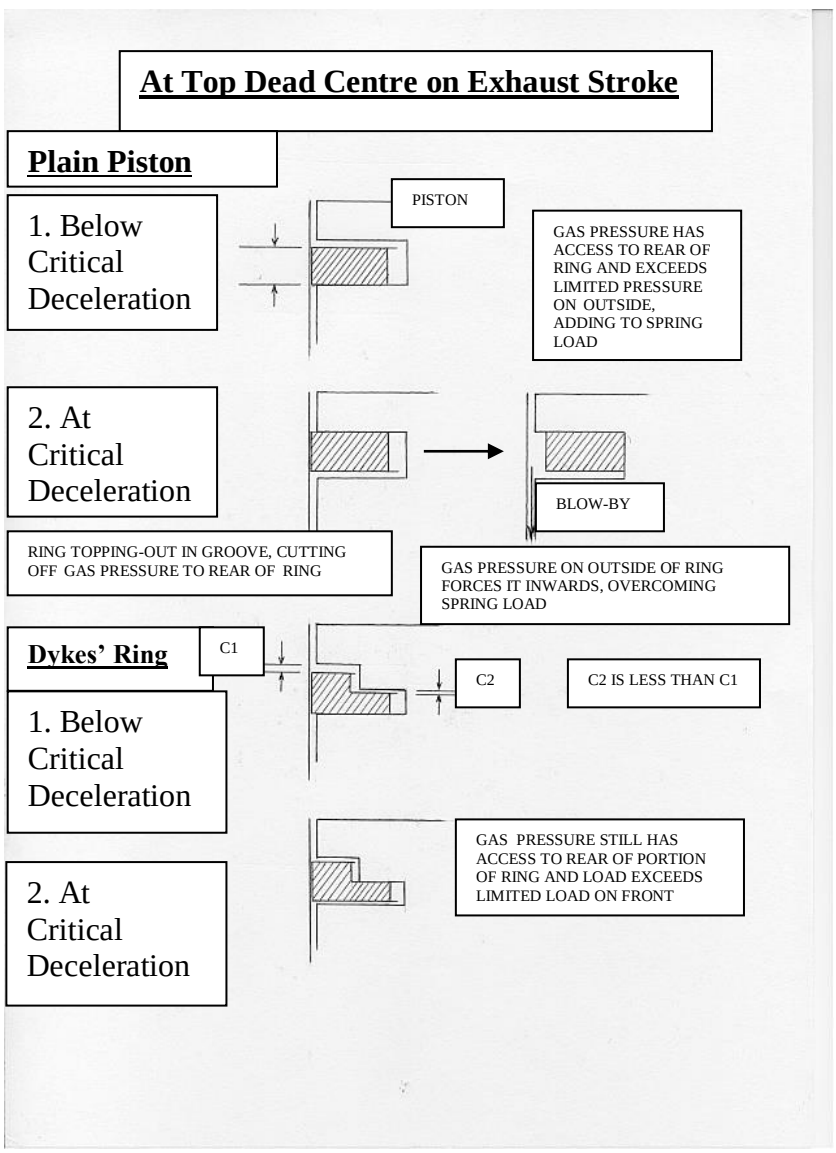
Sub Note C

Dykes' Rings

The first publication of Prof. Dykes' work was in 1947, with a final MIRA report in December 1951 (174). By that date the BRM 135V16 1.5L PC engine had already benefited from his invention. When using plain piston rings of 3/64 inch (1.191mm) width there had been serious blow-by at 10,000 RPM (limiting ring deceleration of nearly 107,000 ft/sec^2 (3318g)), **w.MPD** = 3952 mm.g (Example No 4 in Sub Note B). Fitting 2 x 3/64'' maximum width **L**- section rings (turned from plain rings) to each piston provided a cure up to the design speed of 12,000 RPM, equal to 4778g.

By 1956 the Vanwall IL4 2.5L NA was also using 2 x Dykes' compression rings per piston, and the 1959 –1960 Coventry Climax IL4 2.5L NA used them as the top ring. The same use was made of the type in the 1961 –designed 90V8 1.5L NA engines from BRM and Climax.

In the USA (for e.g.), the 1965 Ford 90V8 7L stock-car racing engine was fitted with Dykes' rings (220).



Postscript to Note 13 Part II
1954 Maserati 250F: Piston-Ring Flutter

Information on the Maserati 250F piston-ring width (1.5 mm in Moss' customer engine 2508 (882)) was received after the data in Sub-Note B was collected.

The 1954 250F engine, when run at the maximum speed of 7,200 RPM recommended to customers who did not wish to change bearings after every GP (147), was oil-tight (790). This represented **w.MPD = 4114 mm.g** and **MPS.√(w/S) =2.55 m/s**. The "works" cars were geared to use much higher RPM, beyond the power peak which was then 7,250 RPM (147), and, in equalling his 1951 Alfa Romeo 159 lap speed at Spa in June 1954 (at 120.5 MPH, 193.9 kph), Fangio used 8,100 RPM (and not his own car but the other "works" car of Marimon!). The engine threw out a great deal of its oil at this speed (790), which was 27% higher in **w.MPD** at **5207 mm.g**. In the race, presumably with the same engine, Marimon also used 8,100 RPM and had to stop for plugs after 1 lap and retired with terminal engine problems 2 laps later. Fangio won the race, with a fastest lap 1.3% slower than his practice speed i.e. not using the same RPM.

This incident does illustrate how piston engines could be run beyond the flutter boundary for short periods but with life-reducing consequences.

Maserati later improved their oil-cooling, by shifting the oil tank from the intake side of the engine compartment to the tail, connecting it with Al-alloy pipes (790)- which then caused their own race-losing fatigue failure at Monza later in 1954.

Precisely how the flutter problem was overcome is not known to this author.

Postscript 2 to Note 13 Part II
Confirmation of the hypothesis that 3NA engines at Peak Power run too fast for piston-rings to flutter

Ref (1062), published in 2004, gives the driver Martin Brundle's account of the McLaren-Peugeot fire on the grid for the 1994 British Grand Prix:
p.179 "*As the start lights came on, I held the engine revs as usual at 11,000 RPM. I looked in my mirror and saw a massive sheet of flame erupting from the rear of the car....the entire"* [engine] "*system had become pressurised and the engine had actually started to consume its own lubricant.....I was then blamed for using the wrong revs on the start line....It turned out that I was holding the revs in a zone in which **the harmonics made the piston rings leak** which ultimately allowed the pressure to go from the cylinders into the sump and send the oil on its way to create the pyrotechnics*". Peugeot had changed from 3 rings to 2. "*No-one had told me about that or **the crucial rev band***" [author's **bold**].

Peak Power RPM of the 1994 Peugeot was around 13,500 RPM. A guesstimate of the critical 11,000 RPM situation suggests that;

w.MPD was **4,400mm.g.**

Limiting Mean Valve Speed **DST 18 February 2003.**

By analogy with the "Bottom-End" of a piston-engine it would be expected that the "Top-End" of a poppet-valve unit would be limited by "Mean Valve Speed", i.e.:-

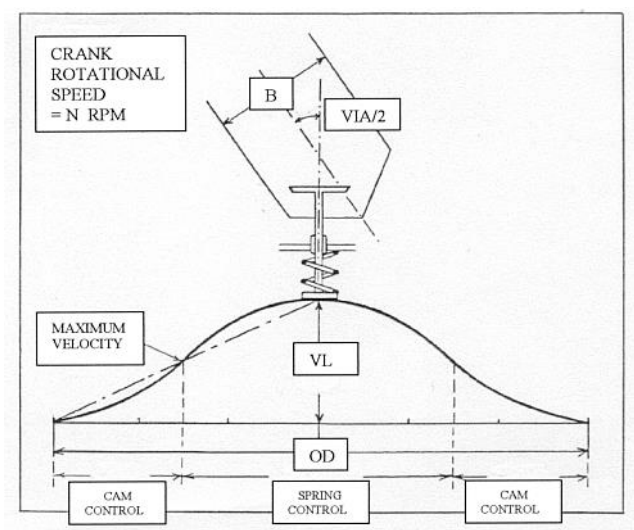
$$\left[\frac{2 \times \text{Valve Lift} \times \text{Crank RPM}}{\text{Valve Angular Opening Duration (Crank Degrees)}}\right] \propto \sqrt{\left[\frac{\text{Stress in Valve Springs}}{\text{Density of Valve Gear}}\right]}$$

With Valve Lift (VL) proportional to Valve Diameter (VD) and this proportional to Cylinder Bore (B) *for a given type of valve gear* (including in that a statement about the Included Angle between Valves (VIA), Number of Valves per Cylinder (VN) and also about Angular Opening Duration (OD), then a relation *at a given state of material technology* should be:-

B.N = a Constant depending on type of valve gear

However, despite the attraction of this broad "**Bore Speed**" analogy with Piston Speed, the details should be examined from first principles for valve gear. This follows in this Note , *for gear using return springs*.
"Desmodromic" gear, giving positive control of valves at all times (within very tight tolerances), even when over-revved beyond specified limits, *should* be the answer to a racing-designer's prayer, but in practice only the Mercedes-Benz M196 of 1954-1955 (468) and Ducati in the motor-cycle sphere since 1956 and to date (74, 95, 762) have made a real success of this technology. Maserati (711), Norton (761) and Cosworth (59) are known to have experimented with it but not carried it through to racing. It may be that expense has played its part in this*.

In spring-controlled poppet valve-gear, the spring absorbs the energy given by the cam to the valve and all associated reciprocating or rocking parts during the first part of the opening, thereby bringing these parts to rest at maximum lift while they are just in contact with the cam (at valve-bounce speed) and with the spring at maximum designed stress. The spring then returns the gear by releasing this stored energy until reaction from the cam brings the valve to rest on its seat. This process applies to any type of spring, including an air or gas spring.
The process is sketched below in terms of Valve Lift v. Angular Opening. With a constant rotational speed the Opening corresponds to time, so that the slope of the curve represents valve velocity.



The Mean Valve Speed (MVS) is given by:-

$$\text{MVS} \propto \left[\frac{2 \,.\, \text{VL} \,.\, \text{N}}{\text{OD}}\right]$$

It is assumed that the cam characteristic is such that:-

Maximum Valve Speed ∝ MVS

Continued on Page 2

*OSCA raced an IL4 1.5L sports car with desmodromic valve-gear briefly in the 50s.

The *Maximum energy given to the Valve Gear*, whose total reciprocating mass is VGM (with appropriate geometrical adjustments to include the effect of any rocking parts and to include a proportion of the spring mass) is therefore:-

$$\propto \text{VGM} \times [\text{MVS}]^2$$

This energy has to be stored by compressing, and stressing the valve spring, additional to the pre- loading when the valve is on its seat. The pre-loaded energy can be taken as proportional to the additional energy (see Sub Note A).

The further analysis assumes a coil spring, which was by far the most-used type from 1906 to 1990. Part I of this Note includes data which shows that the steel coil springs of the Ford-Cosworth DFV were co-equal with the piston in limiting the engine's overhaul life.

For a relatively close-coiled spring, the torsional energy stored is given by:-

$$\frac{(\text{Torsional Stress})^2}{\text{Modulus of Rigidity}} \times \text{Volume of Spring}$$

Ref. (760)

Therefore $$\text{VGM} \times [\text{MVS}]^2 \propto \frac{(\text{Stress})^2}{\text{Modulus}} \times \text{Volume of Spring}$$

From which:-

$$[\text{MVS}]^2 \propto \left[\frac{(\text{Stress})^2}{\text{Density} \times \text{Modulus}}\right] \times \left[\frac{\text{Volume of Spring}}{\text{Equivalent Volume of Valve Gear}}\right]$$

the expression $\left[\frac{(\text{Stress})^2}{\text{Density} \times \text{Modulus}}\right]$ is the "**Valve-Gear Material Factor**" (**VMF**) (which assumes for simplicity that the spring and other gear are the same density) and, when it is at its limit according to the current state of material technology, it sets the limit to **MVS** according to the other factor $\left[\frac{\text{Volume of Spring}}{\text{Equivalent Volume of Valve Gear}}\right]$ which is the "**Valve-Gear Volume Ratio**" (**VVR**). This factor **VVR** will have characteristic values according to broad designs; for Double Overhead camshaft (DOHC) types, operating the valves through "inverted bucket" tappets, **VVR** will be high; for push-rod-and-rocker types with a low-mounted camshaft, **VVR** will be low.

$$\therefore \quad \mathbf{MVS \propto \sqrt{[VMF]} \times \sqrt{[VVR]}}$$

MVS with known internal data

Where the data are known, MVS is therefore the guide to the "Top-End" limit of a specified type of engine at a given date, i.e. a given state of material technology. Usually the Inlet Valve is heavier than the Exhaust Valve (the Bugatti 3-Valve engines are exceptions) and so Inlet Valve Maximum Lift (**IVL**) and Angular Opening (**IOD** in crank degrees) are used to find **MVS**.

The Database uses:-

$$\mathbf{MVS} = \frac{\mathbf{IVLmm \times N\ crank\ RPM}}{\mathbf{83.333 \times IOD\ crank\ degrees}} \quad \text{m/s}$$

Obviously, ways of raising N at a given value of the product of **VMF** and **VVR** are to reduce **IVL** by increasing the number of cylinders for the required Swept Volume and/or increase the number of valves per cylinder for the required inflow. Apart from the use of valve gear with the highest value of **VVR**, which can include, for examples, the rifle-drilling of valve-stems and making the valve heads hollow, a lower-density material may be used in the valve-gear. This latter was made possible by the development of a Titanium alloy (for US military aerospace purposes originally) which could withstand the exhaust valve temperature of a Normally-Aspirated (NA) engine for a sufficiently long racing life (there is no point in only using Ti-alloy for inlet valves, as the exhausts would then become the limiting factor). Ti-alloy was applied to Grand Prix engines from 1989, after the return to NA rules, although Honda had applied it to all valves in racing motor-cycles in 1983 (97) and big US V8 engines with push-rod gear had fitted all-Ti-alloy valves as early as 1962 (220).

B.N as a surrogate for MVS

If the required internal details are *not* available, the analysis may be continued as follows:-
where engines are operating to the limits of power and speed **IOD** will be similar for all engine types, around 320 to 340 crank degrees, so that the expression for MVS can be reduced to :-

$$\mathbf{IVL \times N \propto \sqrt{[VMF]} \times \sqrt{[VVR]}}$$

IVL is proportional to Inlet Valve Diameter (**IVD**) in order to pass the flow and **IVD** is related to **B** by **VNI** and **VIA**.

With 2 opposed, inclined valves per cylinder:-

$$\mathbf{IVL} \propto \mathbf{IVD} \propto \frac{\mathbf{B}}{\mathbf{2 \times \cos(VIA/2)}}.$$

With in-line pairs of valves per cylinder (i.e. *not* radially disposed**):-**

$$\mathbf{IVL} \propto \mathbf{IVD} \propto \frac{\mathbf{B}}{\mathbf{2}} \quad \textbf{irrespective of VIA.}$$

So the overall expression can be reduced further to:-

$$\mathbf{B.N} \propto \sqrt{[\mathbf{VMF}]} \times \sqrt{[\mathbf{VVR} \times [\cos(\mathbf{VIA/2}) \text{ or } 1]}$$

depending on whether a 2 or 4- valve-per-cylinder engine is being considered. It follows that, *at a given state of material technology*, if the required valve-gear life is restricting the operating speed, then:-

$$\mathbf{B.N} = \textbf{a Constant depending on type of valve-gear.}$$

Alternative types of Valve Spring

The main text discusses, against the relevant designs, the use of Hairpin springs (which bend the wire) in place of Coil springs (which twist the wire). Also the introduction of air or gas springs (Pneumatic Valve Return System, PVRS) invented by J-P Boudy of Renault in 1984 (474), which superseded wire springs in all front-line Grand Prix engines in 1990.

Valve-Bounce Speed

The analysis above is derived from the requirement to restrict engine operating speed so as to achieve some specified life of the springs. The *engine* life, in a non-desmodromic-valve-gear high-compression unit, can always be cut to zero by over-revving until a valve leaves the cam (because the spring cannot absorb its energy within VL) and is struck by a piston. It is reasonable to assume that this “Valve-Bounce” speed is proportional to the time-limited speed.

Coil-Spring “ Surge”

With coil springs the moving mass of the wire has its own natural frequency and higher rotational speeds may resonate with this (causing “Surge”) and lead to premature fatigue failure. The analysis for this leads to an expression similar to that given above. It shows that the natural frequency may be increased by raising spring stress by increasing wire diameter and/or reducing the number of turns. This of course is limited as before by the state of spring material technology. Surge damping can also be introduced by an inner spring of different frequency having an interference fit against the outer coils, but this is of limited life before the fit wears out (e.g.,the technique was considered unsuitable for a 24-hour life for a Le Mans engine in 1993 (16)).

Hairpin and pneumatic springs do not suffer from surge.

See Sub Note A on following page

Sub Note A

Examination of valve-spring design details, in particular in Coventry Climax (33,34) and Ford-Cosworth (583) engines, over a range of valve diameters from 25 to 49 mm, has shown that the (Max. Open Load)/(Shut Load) ratio was very close to constant, at about 2.2 (see the details and chart 102/DST below). Therefore, with Energy Stored being proportional to $(\text{Load})^2$ (760), the (Additional Energy Stored)/(Pre-Loaded Energy) was around a constant value of 4 in a wide variety of engines.

VALVE-GEAR DATA No.	ENGINE	DASO	SPRING LOAD-SHUT lbf	SPRING LOAD-OPEN lbf
1	COSWORTH FVA	583	100	230
2	CLIMAX FWA2	33	60	92
3	CLIMAX FWA3	33	*72.3*	156
4	CLIMAX FPF 1.5	33	110.4	257.6
5	CLIMAX FPF 1.5 Mk2	33	136.5	290.7
6	CLIMAX FPF 2.5	33	110.4	257.6
7	CLIMAX FWM	33	27.2	55
8	CLIMAX FWMA	33	52	88.75
9	CLIMAX FWMV1 (Ass'd)	34	69	174
10	CLIMAX FWMV6 (Ass'd)	34	86	185
11	CLIMAX FWMW	34	89	181
12	DRAKE 2.8L TC	711	160	360