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Comparison Between V12 and W12 F1 Engines

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ABSTRACT

In this paper, a comparison has been carried out between two Formula 1 engine architectures: a traditional V12 and a 12 cylinder with three banks and one crankshaft, which will be referred to from here on as W12. This comparison is made in terms of geometrical features, as well as in terms of safety coefficients, torsional stiffness, state of balance and friction losses.

The W12's crankshaft is 158 mm shorter and stiffer than the V12's. Furthermore, this crankshaft is simpler and lighter. The W12 engine front section is wider. The crankshaft of the W12 has a minimum safety factor that is 30% lower than the V12's under the same operating conditions (18000 rpm, bmep=13 bar). While the V12 is perfectly self-balanced, the secondary forces are out of balance in the W12's crankshaft. This unbalance is, however, no more critical than the one occurring in a V10 or V8. Friction losses in the W12 should be slightly lower in comparison to the V12.

INTRODUCTION

The recent history of FIA Formula 1 competitions has outlined the basic importance of the engine-vehicle matching. The engine development cannot ignore implications on vehicle structure and aerodynamics and vice versa. As far as the engine is concerned, the trend of the last years has clearly pointed out as an optimum solution the 10 cylinder architecture, with a V of 72-80°.

In another paper [1], V10 and V12 engine solutions having an equal degree of sophistication have been compared in terms of pure engine performance. V12 engines show a better distribution of brake mean effective pressure vs. engine speed, mainly due to lower friction mean effective pressure. However, advantages of this solution in terms of car performance are not clear: V10 engines get less horsepower, but they are lighter and shorter, have an increased stiffness, require smaller radiators and carry less fuel. The present paper addresses the question concerning an alternative 12 cylinder architecture, allowing to maintain the benefit of the 12 cylinder while reducing or canceling its drawbacks.

The alternative 12 cylinder architecture, analyzed in this paper, is made up of three banks of cylinders and one simple crankshaft (see figure 1). Each crankpin bears three equal connecting rods, one from each bank. This kind of engine is not a new idea: for example, in 1992, Audi realized a concept car called AVUS 4, by mounting an engine with this architecture.

In this paper, the W12 architecture is compared to a V12, which presents the same geometry of the cylinder (bore, stroke, valves diameter, etc). The comparison is made, besides the geometrical features, also in terms of safety coefficients, torsional stiffness, state of balance, and friction losses. The analysis is carried out by using elementary models and under several hypotheses which, on one hand simplify the analytical problem, and on the other hand always take into account the most severe conditions.

Another aim of the paper is to demonstrate that, with the lay-out and overall dimensions indicated in figures 1,2 the two engines possess the fundamental technical requirements to be used in F1 competitions. This essentially means to show that the engines, and particularly the W12, are able to resist the thermo-mechanical stress occurring at the very high engine speed that must be reached.

Obviously, this paper presents only a preliminary study on the application of a W12 architecture to a F1 engine. More complex theoretical models should be employed to address engine design, and a substantial work is then required for the developement.

V12 AND W12 LAYOUT

Figures 1 and 2 present the lay-out of the engines, while table 1 shows the main geometrical features.

The two engines share the same geometry of the cylinder: bore, stroke, compression ratio, connecting rod length, etc. The bore to stroke ratio and the connecting rod length to crank throw ratio have been assumed to be equal to 2.22 and 4.79, respectively. The weight of the piston and connecting rod have been derived from a data base of Formula 1 engines, by using the similarity criterion. The maximum mean piston speed has been assumed to be equal to 24.1 m/s, corresponding to 18000 rpm. It should be observed that the value of the former parameter, which is notoriously the main index of the mechanical stress of the engine, is not very high for a Formula 1 engine (it is known that values up to 27 m/s have sometimes been accepted).

For the V12, the angle between the banks is 75 degrees, according to the value of the latest model of 12 cylinder engines used in F1 (1995). For the W12, the angle between the banks has been assumed to be 60 and 65°. The reason for a different angle between the banks of the W12 is the lay-out of the exhaust system. A wider angle allows a better fit of the central bank's primary exhaust manifolds, which cannot run aside the engine. With the firing orders shown in table 1, the angular distance between two consecutive combustion events is 45-75 degrees for the V12, and 55-65 degrees for the W12. Consequently, the torque output of the W12 is slightly smoother than the V12's. Furthermore, on each bank of the W12, combustion events are evenly spaced in the engine cycle with an angle of 180 degrees, instead of the 120 degrees of a V12's bank. Thanks to this wider angle, the exhaust manifolds can be joined to form a four-in-one system, without any problem of fluid-dynamic interference among the cylinders, at least at high engine speed. The four-in-one exhaust system is not only simpler to make than a 3-in 2-in 1, but it also suits, particularly well the gas exchange processes at high engine speed, allowing to achieve good volumetric efficiency.

The most important advantage of the W12, compared to the V12, is in the length of the crankshaft, intended as the distance between the middle of the first and the last bearing. The W12 is 158 mm shorter than the V12, and consequently much more stiff. If reference is made to the most widespread F1 engine architecture, i.e. the V10, the advantage of the W12 still remains consistent (about 90 mm, if the same bore-to-stroke ratio is considered, as well as 8 mm of minimum distance between the cylinders' bore). Only a V8 engine can compete with the W12 for length and stiffness (anyway, the V8 is about 20 mm longer). A further advantage of the W12 in comparison to the V engines with an equal degree of sophistication for the cylinder block casting, is the possibility of enlarging the cylinder bore by a few millimeters, with no need to increase the cylinder axle base, and subsequently the engine length.

The W12 also has a lighter (about 2 kg) and simpler-to-make crankshaft than the V12's. The W12 engine has a lower center of gravity (about 20 mm), which is an advantage for vehicle handling.

Furthermore, the W12 is much wider (160 mm) than the V12. From a structural point of view, this item is a benefit (increased torsional stiffness), but it imposes a brand-new design of the snorkel, with implications on aerodynamics not easily predictable.

One drawback of the W12 is the position of the exhaust manifolds of the central bank, which are very close to the intake telescopic tapers and can be disturbing for the snorkel design. However, the amount of heat transferred from exhaust to intake gas should be quite easy to limit.

Finally, the W12 has two more camshafts than the V12, and a more complex driving gear system. However, in the W12, it is possible to limit to 18 the number of gear (against 17 required for the V12). The W12's camshafts are shorter and stiffer.

Table 1. Main geometrical features

	V12	**W12**
Total Displacement [cc]	3000	3000
Number of Cylinder	12	12
Bore [mm]	89	89
Stroke [mm]	40.1	40.1
Compression Ratio	12:1	12:1
Connecting Rod Length [mm]	96	96
Full piston weight [g]	290	290
Connecting rod weight [g]	277	277
Max. mean piston speed [m/s]	24.1	24.1
Angle between banks [deg]	75	60/65
Firing Order	1-12-5-8-3-10-6-7-2-11-4-9	1-8-9-3-6-11-4-5-12-2-7-10
Cylinder Axle Base	97	106
Number of Bearings	7	5
Journal diameter [mm]	50	52
Crankpin diameter [mm]	36	40
Crankshaft total length [mm]	582	424
Crankshaft weight [kg]	13.4	11.3
Engine height [mm] (°)	330	325
Engine width [mm]	500	660
Center of gravity height [mm] (*)	140	120
Number of camshafts	4	6
Number of gears (**)	17	18

(*) from crankshaft axis;
(°) from the cylinder axis to the top of telescopic tapers;
(**) for driving the accessories

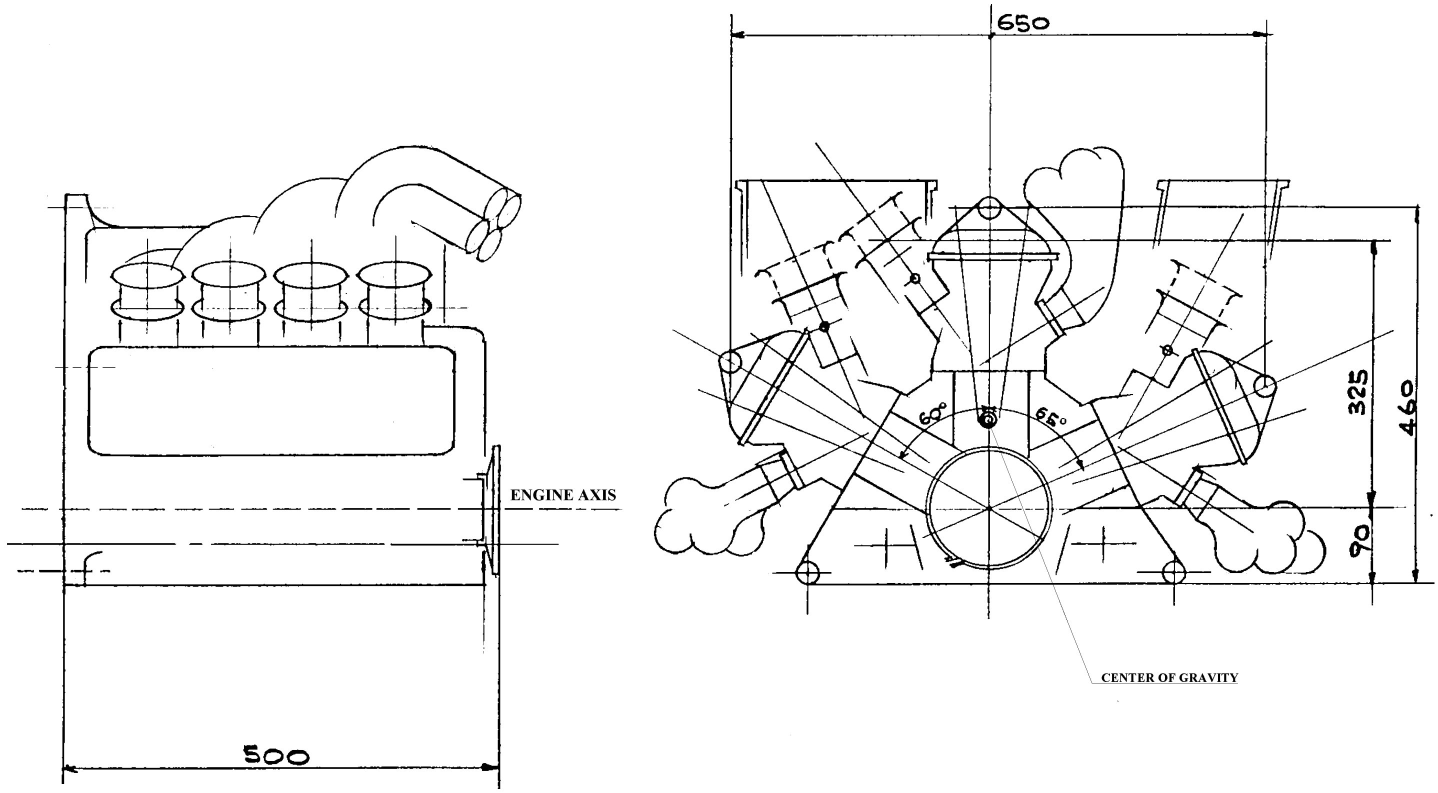

Figure 1. Lay-out of the W12 engine. It is made up of three banks of cylinders and one simple crankshaft. Each crankpin bears three equal connecting rods, one from each bank. The engine is connected to the cockpit by 5 point.

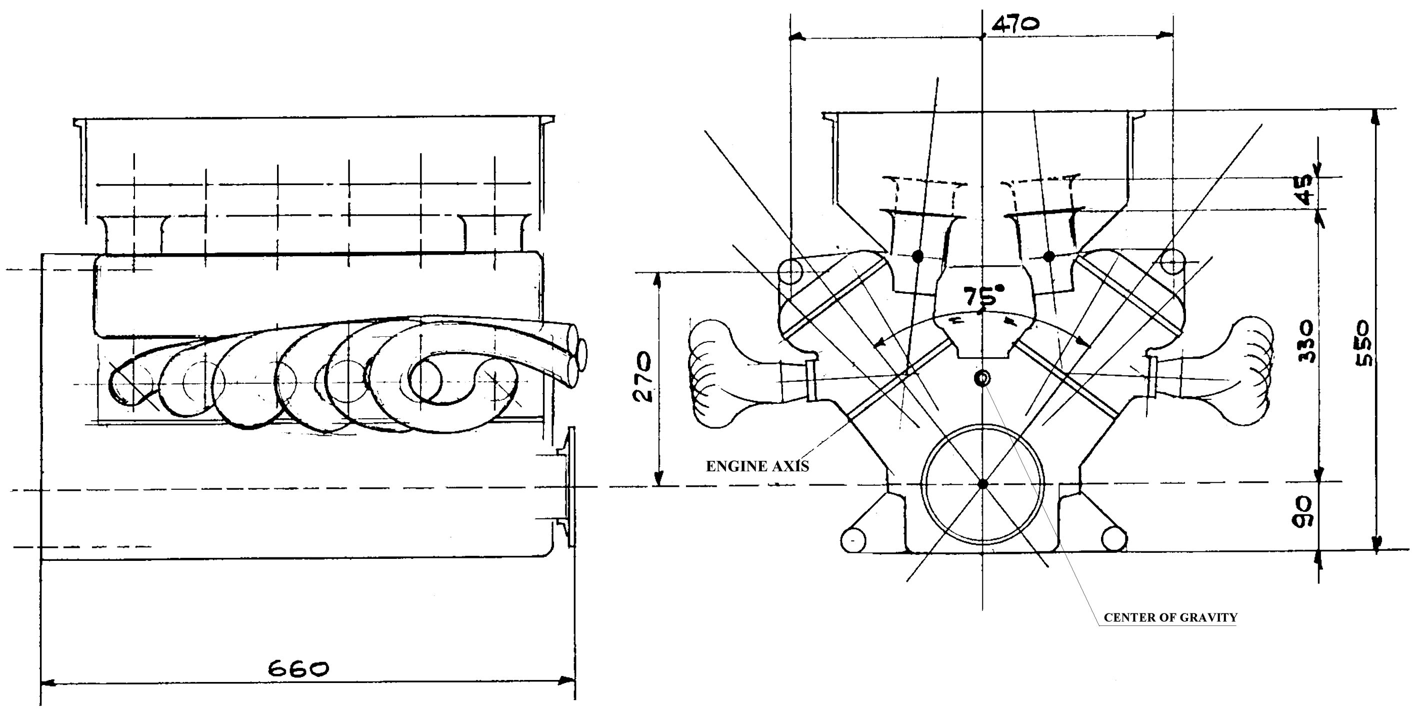

Figure 2. Lay-out of a traditional V12 engine. The engine is connected to the cockpit by 4 points.

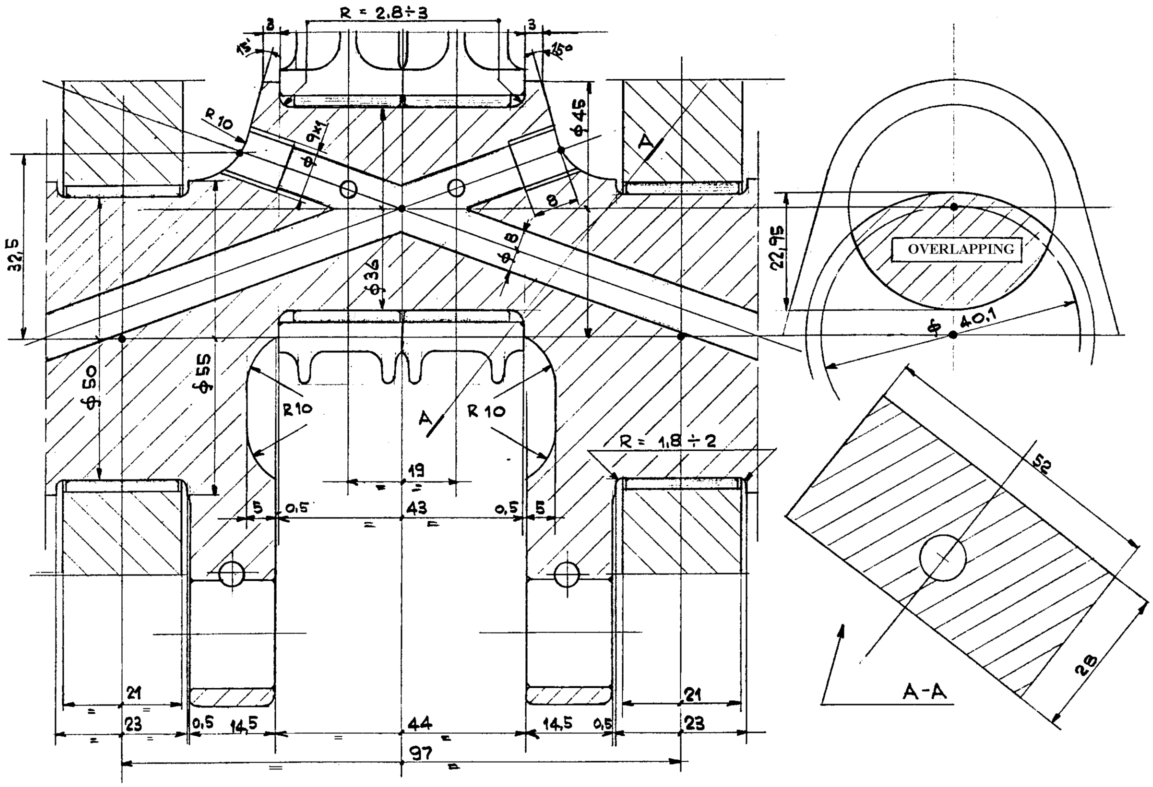

Figure 3. Working drawing of one V12 crank

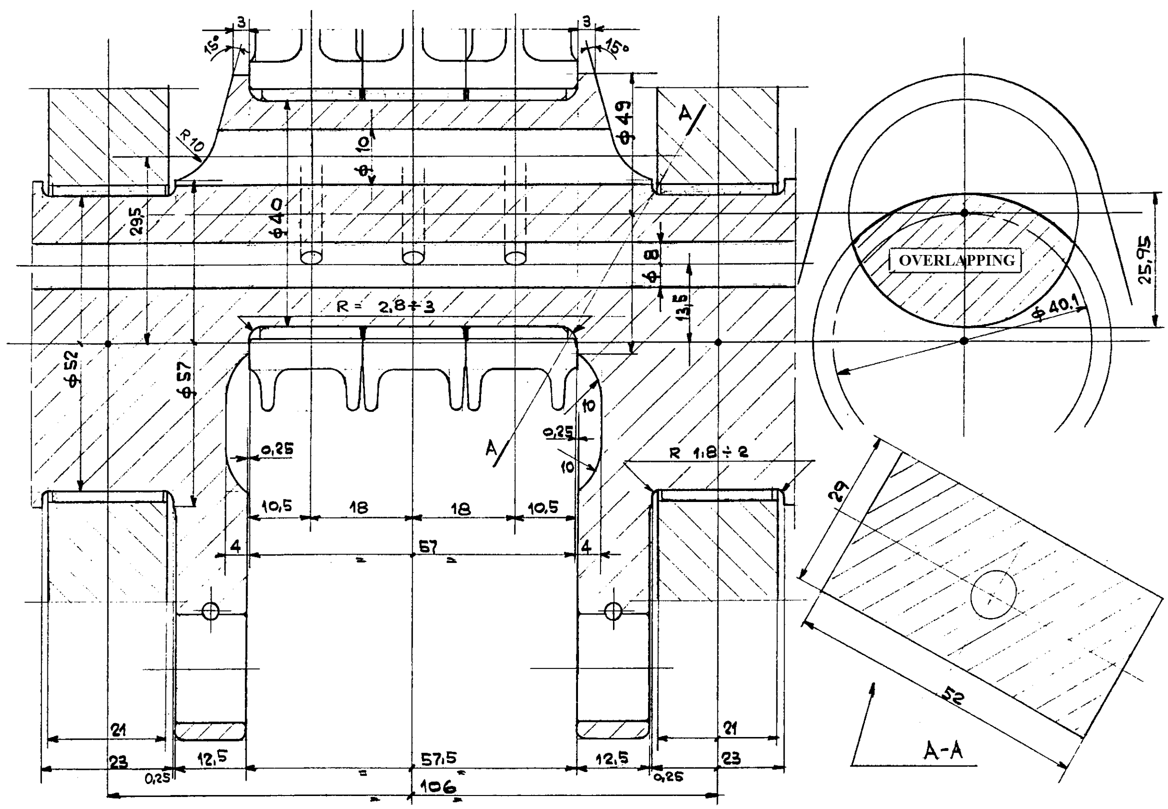

Figure 4. Working drawing of one W12 crank.

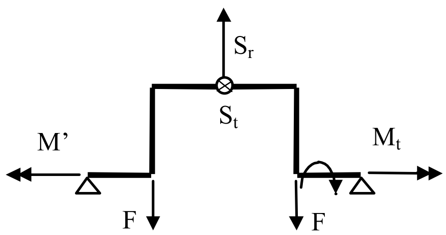

Figure 5. Sketch of the crank model and loading

CRANKSHAFT SAFETY ANALYSIS

According to the hypothesis made in the previous section, the differences between the V12 and the W12, from the point of view of structure resistance, have to be searched for in the crankshaft.

In the working drawings of figures 3 and 4, the dimensions of each crank are shown for both the V12 and the W12. These dimensions arise from the typical geometrical parameters of Formula 1 engines, and have been adjusted according to the results of the safety analysis, which is presented in the following section.

The crank has been modeled as a continuous beam, on two bearings, placed in the middle of the journals (see figure 5) [2]. For the evaluation of the constraint reactions, it is supposed that each crank is internally balanced (i.e., the reactions depend only on the load applied to the crank).The beam is loaded by the resulting force, transmitted by the connecting rods. This force, applied in the middle of the crankpin, is resolved into a radial (outward, along the crank throw direction) and a transverse component (opposite to the rotation versus), called S_r and S_t, respectively. The two equal forces applied to each crank web and indicated by F represent the centrifugal forces acting on the balancing masses. Finally, a torque is applied at each end of the crank. Since the most highly stressed crank of the shaft can be considered the last one, close to the gearbox, the couple M'_t is the sum of the couples due to the other cranks, and M_t is the engine torque.

The load acting on the crank has been evaluated under two different hypotheses [3]. In the former, the engine rotates at the maximum speed (18000 rpm), at full load (bmep=13 bar, maximum pressure within the cylinder=100 bar). In the latter, while the engine is running at 18000 rpm, the gas pressure within the cylinder is neglected (only the inertia forces are considered).

The sketch of figure 6 is now considered in order to express the components of the force transmitted by the connecting rods to the crankpin. Reference is made to the W12 crank geometry, with three concurrent connecting rods. If the central connecting rod (cylinder 2 in figure 6) is simply ignored, the sketch can also represent the V12 (or any other traditional V engine).

The components S_r and S_t are expressed in equation (1) [4]. Equation 1 takes into account the inertia forces due to the reciprocating mass (first term in curled brackets), the forces related to the rotating mass (which are always radial), and the forces arising from the gas pressure within the cylinders (last term in curled brackets). The last term has a sign opposite to that of the inertia forces.

$$S_r = m_{rec}\omega^2 r\left\{\begin{array}{l}\left(\mathrm{Cos}\theta+\frac{\mathrm{Cos}2\theta}{n}\right)\frac{\mathrm{Cos}(\alpha_2+\theta)}{\mathrm{Cos}\alpha_2}\\+\left(\mathrm{Cos}(\theta+\beta)+\frac{\mathrm{Cos}2(\theta+\beta)}{n}\right)\frac{\mathrm{Cos}(\alpha_1+\theta+\beta)}{\mathrm{Cos}\alpha_1}\\+\left(\mathrm{Cos}(\theta-\gamma)+\frac{\mathrm{Cos}2(\theta-\gamma)}{n}\right)\frac{\mathrm{Cos}(\alpha_3+\theta-\gamma)}{\mathrm{Cos}\alpha_3}\end{array}\right\}$$

$$+m_{rot}\omega^2 r - A_p\left\{\begin{array}{l}p_1\frac{\mathrm{Cos}(\alpha_1+\theta+\beta)}{\mathrm{Cos}\alpha_1}+p_2\frac{\mathrm{Cos}(\alpha_2+\theta)}{\mathrm{Cos}\alpha_2}\\p_3\frac{\mathrm{Cos}(\alpha_3+\theta-\gamma)}{\mathrm{Cos}\alpha_3}\end{array}\right\}$$

$$S_t = m_{rec}\omega^2 r\left\{\begin{array}{l}\left(\mathrm{Cos}\theta+\frac{\mathrm{Cos}2\theta}{n}\right)\frac{\mathrm{Sin}(\alpha_2+\theta)}{\mathrm{Cos}\alpha_2}\\+\left(\mathrm{Cos}(\theta+\beta)+\frac{\mathrm{Cos}2(\theta+\beta)}{n}\right)\frac{\mathrm{Sin}(\alpha_1+\theta+\beta)}{\mathrm{Cos}\alpha_1}\\+\left(\mathrm{Cos}(\theta-\gamma)+\frac{\mathrm{Cos}2(\theta-\gamma)}{n}\right)\frac{\mathrm{Sin}(\alpha_3+\theta-\gamma)}{\mathrm{Cos}\alpha_3}\end{array}\right\}$$

$$-A_p\left\{\begin{array}{l}p_1\frac{\mathrm{Sin}(\alpha_1+\theta+\beta)}{\mathrm{Cos}\alpha_1}+p_2\frac{\mathrm{Sin}(\alpha_2+\theta)}{\mathrm{Cos}\alpha_2}\\+p_3\frac{\mathrm{Sin}(\alpha_3+\theta-\gamma)}{\mathrm{Cos}\alpha_3}\end{array}\right\} \qquad \text{(Eq. 1)}$$

The reciprocating mass, m_{rec}, is evaluated as the sum of the full piston and one third of the connecting rod weight. The total rotating mass, m_{rot}, is given by the sum of two thirds of the connecting rod weight, C, multiplied by the number of cylinder concurrent on the same crank, z, and the crankpin mass (m_{pin})

$$m_{rot} = z\ .\ (0{,}67.C)+m_{pin} \qquad \text{(Eq. 2)}$$

The angles α_1, α_2, α_3 are functions of θ according to the following:

α_1= arcsin[Sin(θ+β)/n] $\qquad$ α_2= arcsin[Sin(θ)/n]

α_3= arcsin[Sin(θ-γ)/n] $\qquad$ (Eq. 3)

For precision's sake, it should be observed that, in equation (1), the forces due to each connecting rod moment of inertia have been neglected. Furthermore, under the hypothesis of no gas pressure within the cylinder, the last term between curled brackets, is null .

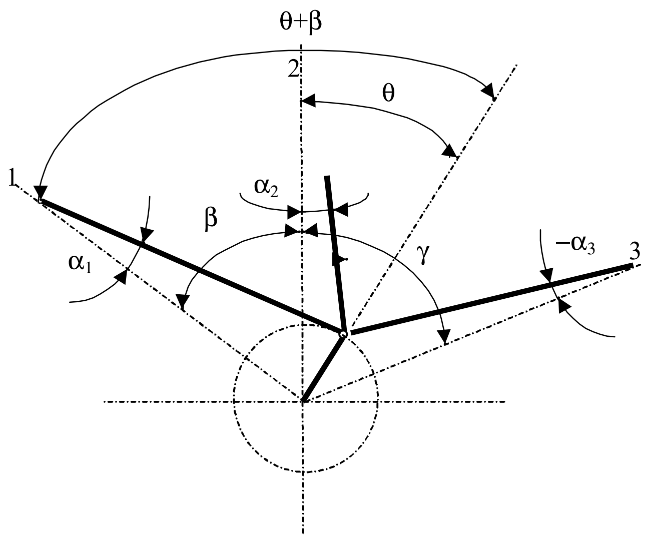

Figure 6. Sketch of the crank with the connecting rods

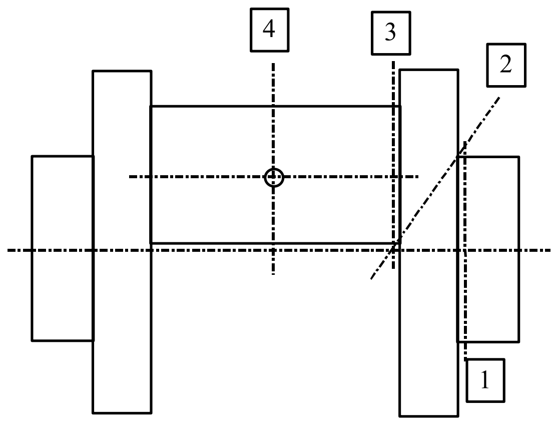

Figure 7. Critical sections for fatigue resistance

Force F is given by the following expression:

$$F = 0.5\,(m_{rot} + K.m_{rec}).\omega^2 r_b \qquad \text{(Eq. 4)}$$

where r_b is the distance of the balance weights' center of gravity from the crankshaft axis, K is a parameter which can range from 0 to 1. The entity of this parameter, which can vary from crank to crank, can influence not only the crank loading (and then the safety factors), but also the load on the bearings (then the friction losses), as well as the state of balance of the crankshaft. Obviously, high values of K lead to an increasing weight of the crank-shaft. The choice of K comes from a compromise among all these requirements and it is far beyond the purpose of this study. Just in order to perform a simple computation, it is supposed that the optimum value of K is the one that minimizes the friction losses on the journals, which are computed as discussed afterwards.

The safety analysis has been carried out for fatigue on four critical sections, for both the V12 and the W12. With reference to figure 7, these sections are: the journal in the fillet with the crank web (1); a squint section of the crank web (2); the crankpin in the fillet with the web (3); the middle of the crankpin (4), with an oil hole drilling.

On each section the nominal normal and tangential stress have been evaluated on two locations, indicated as 'a' and 'b' in figure 8.

With reference to figure 8, the normal and tangential stress have been evaluated according to the following equations. For location (a):

$$\sigma = \left|\frac{M_y}{W}\right| + \left|\frac{N_z}{A}\right|$$

$$\tau = \left|\frac{M_z}{W_0}\right| + \frac{3}{2}\left|\frac{T_y}{A}\right| \qquad \text{(Eq. 5)}$$

For location (b):

$$\sigma = \left|\frac{M_x}{W}\right| + \left|\frac{N_z}{A}\right|$$

$$\tau = \left|\frac{M_z}{W_0}\right| + \frac{3}{2}\left|\frac{T_x}{A}\right| \qquad \text{(Eq. 6)}$$

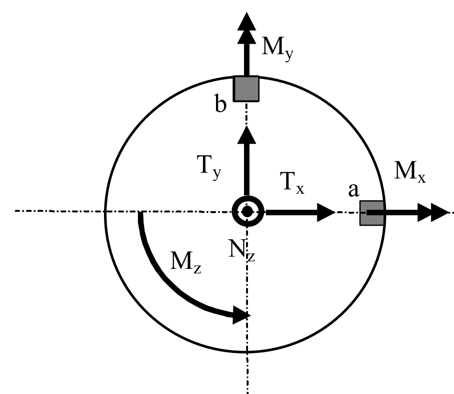

Figure 8. Load on a pin section: bending moments (M_x,M_y), torque moment (M_z), normal force (N_z), shear forces (T_x,T_y)

The values of couples and forces to be considered are the maximum ones over the cycle. W and W_0 are the bending and torque section modulus, while A is the area of the cross section of the pin.

The bending and torque fatigue limits, σ_l, and τ_l , are given by [5]:

$$\sigma_l = \sigma_m + \frac{0{,}45 \cdot R}{\beta_{k\sigma}} \cdot \left(1 - \frac{\sigma_m}{R}\right)$$

$$\tau_l = \tau_m + \frac{0{,}45 \cdot R}{\sqrt{3} \cdot \beta_{k\tau}} \cdot \left(1 - \frac{\tau_m}{\sqrt{3} \cdot R}\right) \qquad \text{(Eq. 7)}$$

σ_m and τ_m are the nominal cycle averaged normal and tangential stress, R is the ultimate strength (assumed to be equal to1000 MPa); $\beta_k\sigma$ and $\beta_k\tau$ are the effective stress concentration factors for bending and torque, given by:

$$\beta_k\sigma = 1 + \eta_k(\alpha_k\sigma - 1) \qquad \beta_k\tau = 1 + \eta_k(\alpha_k\tau - 1) \qquad \text{(Eq. 8)}$$

where η_k is the notch sensitivity (assumed to be equal to 0.9), $\alpha_k\sigma$ and $\alpha_k\tau$ are the stress concentration factors for bending and torque.

Finally, the safety factor n_s is evaluated by the following:

$$n_s = \frac{1}{\sqrt{\frac{\sigma^2}{\sigma_1^2} + \frac{\tau^2}{\tau_1^2}}} \qquad \text{(Eq. 9)}$$

In table 2, for the sections indicated in figure 7, the stress concentration factors and the calculated safety factors are shown [5],[6].

Table 2. Stress concentration and safety factors

	$\alpha_k\sigma$	$\alpha_k\tau$	n_s(*) V12	n_s(*) W12	n_s(**) V12	n_s(**) W12
1a	2.5	1.8	7.51	13.26	5.60	5.01
1b	2.5	1.8	7.61	13.53	7.04	4.27
2a	2.6	1.8	2.46	2.51	2.57	1.79
2b	2.6	1.8	3.27	4.79	3.27	1.84
3a	2.5	1.8	1.94	2.43	1.88	1.60
3b	2.5	1.8	2.66	5.56	2.16	1.41
4a	3	2.6	1.42	1.86	**1.36**	1.18
4b	1	1	1.75	1.63	1.66	**1.07**

(*) No pressure within the cylinder;
(**) bmep=13 bar, maximum pressure within the cylinder=100 bar

From table 2, it is quite clear that the most critical condition for the safety analysis occurs when the pressure cycle within the cylinder is considered. The W12 is much more sensitive to the cylinder pressure. For both the V12 and the W12, the most critical section is '4'. The W12 presents a minimum safety factor slightly greater than 1 (1.07), which is about 30% lower than the V12's (1.36).

The over-the-pin pressure is a fundamental parameter to be controlled in order to ensure a correct bearing lubrication. This pressure is evaluated as the ratio of the load acting on the pin and the area, resulting from the product of the diameter to the length of the pin. For the journal, and the part of the crankpin coupled with the big end of the connecting rod, the computational results at the engine speed of 18000 are presented in figure 9. It should be observed that the load acting on the journals has been computed under the hypothesis of a perfectly balanced crank (i.e., with no influence from the neighboring cranks). Furthermore, a pressure cycle within the cylinder has been considered, with a peak value of 100 bar.

Figure 9 shows that the lubrication of the big end of the connecting rod is much more critical than the journal's, both for the V12 and the W12. Combustion events can be easily picked out for the abrupt change in the plot shape. While this results as a peak for the journal, for the crankpin it reduces the entity of the load. Only slight differences can be observed between the V12 and the W12 crankpin. In any case pressures are always under control, even if a peak of about 80 MPa is quite high.

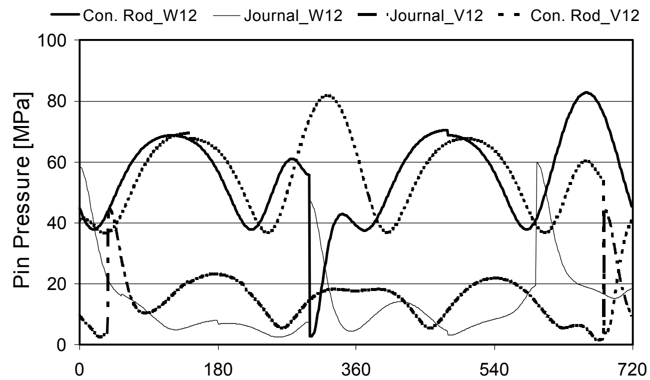

Figure 9. Pressure on the crankpin and journal against crank angle

TORSIONAL STIFFNESS OF THE CRANKSHAFT

A simple computation of the crankshaft torsional stiffness is performed in order to evaluate the difference between the two engines. The stiffness k is defined as:

$$k = \frac{G}{\sum_{i=1}^{Nc} L_i / J_{pi}} \qquad \text{(Eq. 10)}$$

G is the tangential elasticity modulus, Nc is the number of cranks, J_{pi} is the moment of inertia for the cross section of the i-journal, L_i is the equivalent length of the i-crank. This equivalent crank length has been computed by using the Carter's formula [2]:

$$L_i = 2 \cdot c + 0.8 \cdot b + 0.75 \cdot \left(\frac{D_j^4 - d_j^4}{D_p^4 - d_p^4}\right) \cdot a + 1.5 \cdot \left(\frac{D_j^4 - d_j^4}{b \cdot s^3}\right) \cdot r \qquad \text{(Eq. 11)}$$

where a, b , c, s, D_j, D_p, d_j and d_p are geometrical parameters of the crank.

The W12's crankshaft is about 64% stiffer than the V12's (56445 against 34437 N.m/rad). This result is quite obvious, since the W12 crankshaft is considerably shorter than the V12's. Thanks to this higher stiffness, as well as to the better regularity of the firing distribution, torsional vibrations in the W12 should be less critical in comparison to a V12.

THE CRANKSHAFT STATE OF BALANCE

In this section, the state of balance of the crankshaft is evaluated, through the analysis of the first and second order forces and couples transmitted to the bearings [7],[8].

In figure 10, a sketch of the V12 and W12 crankshaft is shown. It should be observed that the V12 crankshaft is equal to a 6-cylinder in-line crankshaft, with the only difference being that each crankpin supports two connecting rods instead of one. It is well known that the 6-cylinder in-line crankshaft is perfectly self-balanced for primary and secondary forces, as well as for primary and secondary couples. Consequently, if the balance weights at each crank are equal, the V12 is also perfectly balanced, being the sum of two self-balanced systems.

The W12 crankshaft is equal to that of a four-cylinder in-line, with the difference being that each crankpin bears three connecting rods instead of one. As a four-cylinder in-line, the W12 is self-balanced for first order forces and couples, although it is not balanced for the secondary ones. The resulting of the secondary forces have been evaluated as the sum of two sets of 12 vectors (one for each cylinder), which have the same modulus and rotate at twice the speed of the engine. The former set rotates in the same versus of the engine; the latter in the opposite direction. The modulus of each vector is given by:

$$H' = 0.5\, m_{rec}\, \omega^2 r/n \qquad \text{(Eq. 12)}$$

The angle formed by each vector of the first set, with its cylinder axis, is twice the angle formed by its crank throw with the cylinder axis. The correspondent vector of the second set is the symmetric one in respect to the cylinder axis.

Since the V12 can not be used as a term of comparison with the state of balance of a W12 engine, a 10 cylinder, with a V of 72°, and a 8 cylinder with a 90° V and a planar crankshaft, have also been considered. In order to carry out this comparison, all the engines have the same total displacement , the same bore-to-stroke ratio, and it is also supposed that the reciprocating mass is simply proportional to the unit displacement. In figure 11 a sketch of the V10 and V8 crankshaft is shown.

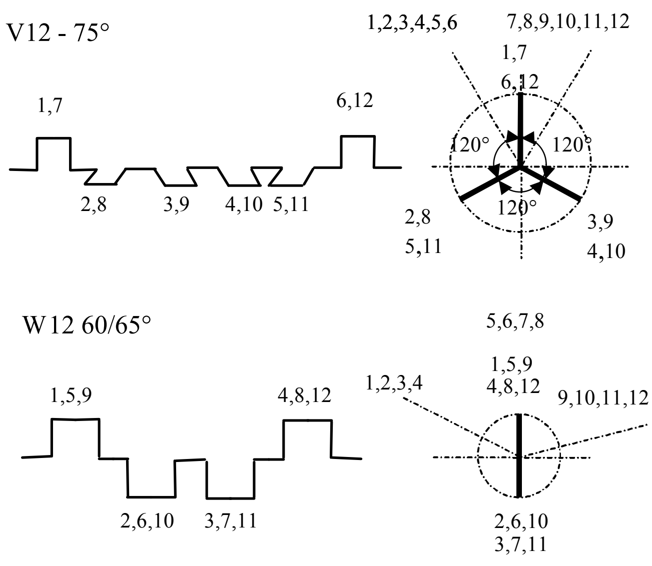

Figure 10. Sketch of the V12 and W12 crankshaft

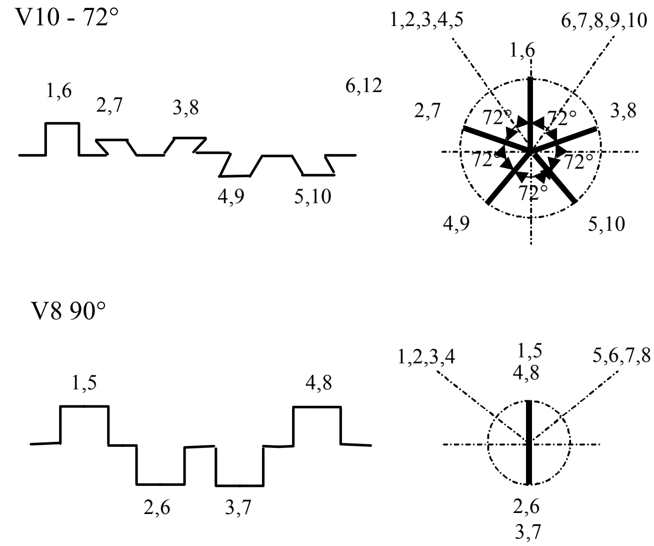

Figure 11. Sketch of the V10 and V8 crankshaft

The V10 engine is self-balanced to the first and second order forces; nonetheless, it presents a severe imbalance for the first order couples. For the evaluation of these couples, it has been supposed that the balance masses m_b are evenly distributed on each crank web, which are given by:

$$m_b = 0.5\, r\, (m_{rot} + 0.5\, m_{rec}) / r_b \qquad \text{(Eq. 13)}$$

The V8 with a planar crankshaft is the closest architecture to the W12, at least from a balance point of view. The V8 presents no unbalanced primary forces and couples. The secondary forces remain out of balance, and they can determine secondary couples.

In figure 12, the results of the horizontal and vertical forces on the bearings are shown for the W12 and the V8 (forces are null for the V12 and the V10). In figure 13, the resulting of the horizontal and vertical couples calculated regarding one end of the crankshaft are presented for the W12, the V10 and the V8 (couples are null only for the V12). The engine speed considered in both figures 12 and 13, is 18000 rpm.

From figures 12 and 13, the W12 engine proves obviously to be worse than a V12, but not considerably different from the other architectures. Regarding the V10, the W12 presents a secondary force which doesn't occur in the V10, althought the cockpit experiences much smaller couples. The maximum V10 horizontal couple (dark bold line in figure 13), with an arm of 350 mm, is equivalent to a pair of forces of 40000 N, a value higher than the maximum one transmitted by the W12. Regarding the V8, the W12 shows lower forces and couples, essentially due to the lower reciprocating mass.

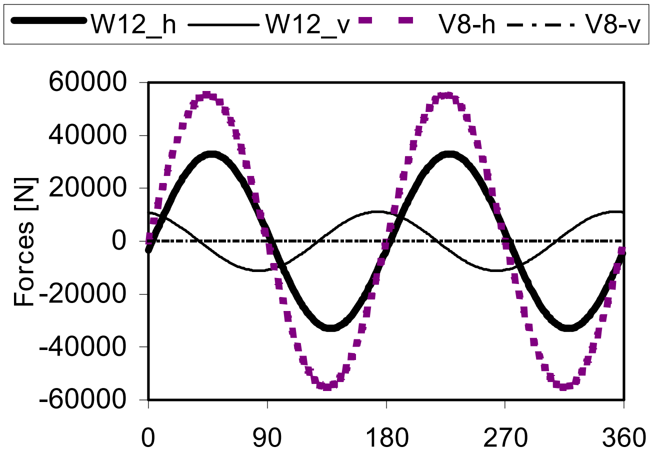

Figure 12. Unbalanced crankshaft forces.

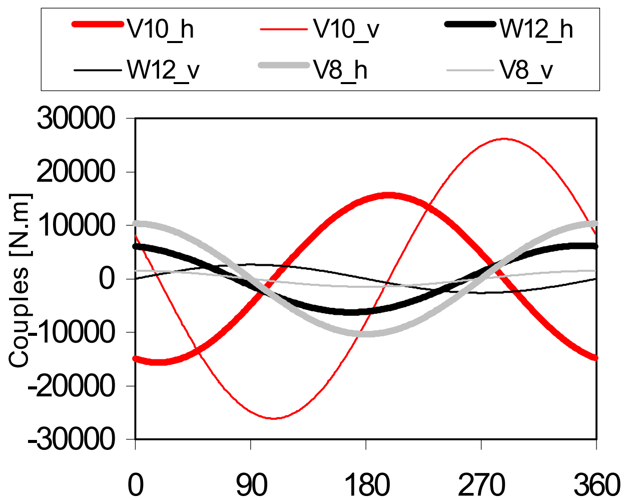

Figure 13. Unbalanced couples regarding crankshaft's left end (cockpit side)

FRICTION LOSSES

Under the hypothesis of equal cylinder geometry, the differences which can be expected between the V12 and the W12 in terms of friction losses arise only from the crankshaft, and the camshafts (including the gears necessary to drive them).

The entity of the difference related to the last item is strongly reduced if it is considered that the main source of energy loss is the actuation of each poppet valve, which is the same for the two engines. Only a slight increase of friction can be accounted to the W12 due to the presence of one additional gear.

In order to evaluate the losses on the journals, the force acting on each bearing has been computed for both engines. The hypothesis has been made that each crank is internally balanced (i.e., each crank constraint reaction is not influenced by the neighboring cranks). Furthermore, the balance weights of each crank are equal.

The work per cycle carried out by the friction force on the bearing (i) is:

$$L_{f,i} = \int_0^{2\pi} f \cdot R_i(\theta) \cdot \frac{d\varphi_i(\theta)}{d\theta} \cdot d\theta \qquad \text{(Eq. 14)}$$

where f is a friction factor, $R_i(\theta)$ is the force acting on the bearing, $\varphi_i(\theta)$ is the angle formed by the direction of this force with the vertical axis, θ is the crank angle.

For the bearings at each end of the crankshaft, the force R corresponds to the constraint reaction exerted on the journals of an isolated crank, i.e.:

$$R(\theta) = \sqrt{\left(\frac{S_r(\theta)}{2} - F\right)^2 + \left(\frac{S_t(\theta)}{2}\right)^2}$$

$$\varphi(\theta) = \mathrm{arctg}\left[\frac{S_t(\theta)}{S_r(\theta) - 2F}\right] \qquad \text{(Eq. 15)}$$

The gas pressure within the cylinder has been neglected in the expression of S_r and S_t

The value of R_i is given by the vectorial sum of the constraint reaction calculated for each crank resting on the bearing.

For the V12 crankshaft, shown in figure 10, R_i is given by:

$$\begin{aligned}
\vec{R}_1(\theta) &= \vec{R}(\theta) \\
\vec{R}_2(\theta) &= \vec{R}(\theta) + \vec{R}(\theta - 120) \\
\vec{R}_3(\theta) &= \vec{R}(\theta - 120) + \vec{R}(\theta + 120) \\
\vec{R}_4(\theta) &= 2 \cdot \vec{R}(\theta + 120) \\
\vec{R}_5(\theta) &= \vec{R}_3(\theta) \\
\vec{R}_6(\theta) &= \vec{R}_2(\theta) \\
\vec{R}_7(\theta) &= \vec{R}_1(\theta)
\end{aligned} \qquad \text{(Eq. 16)}$$

For the W12 crankshaft, shown in figure 10, R_i is given by:

$$\begin{aligned}
\vec{R}_1(\theta) &= \vec{R}(\theta) \\
\vec{R}_2(\theta) &= \vec{R}(\theta) + \vec{R}(\theta + 180) \\
\vec{R}_3(\theta) &= 2 \cdot \vec{R}(\theta + 180) \\
\vec{R}_4(\theta) &= \vec{R}_2(\theta) \\
\vec{R}_5(\theta) &= \vec{R}_1(\theta)
\end{aligned} \qquad \text{(Eq. 17)}$$

The mass of the balance weight of each crank strongly affects the entity of the friction losses. As previously discussed, the mass is equal to the sum of the crank rotating mass, plus a fraction of the reciprocating mass. The value of such a fraction (indicated as K), which minimizes the losses of the whole crankshaft, has been computed for the V12 and the W12 , resulting in a 27% and in a 44%, respectively.

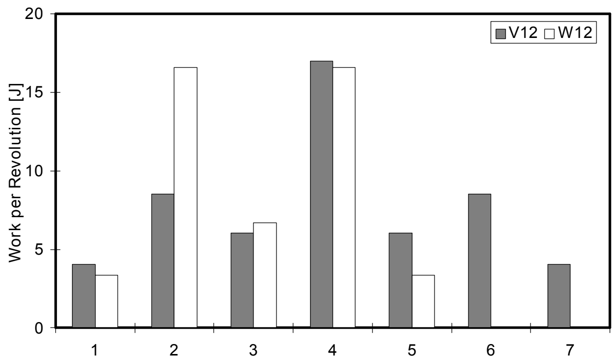

Figure 14. Comparison between friction losses in each crankshaft bearing of a W12 and V12, at 18000 rpm.

In figure 14, the work done at an engine speed of 18000 rpm, during a complete revolution by each friction force (f.Ri) is shown. The numerical values of the work depend on the value of friction coefficient, which has been assumed to be constant and equal to 0.005 [9], for the sole purpose of performing a comparison between V12 and W12.

The W12 shows a reduction of the total friction on the journals of 17.4%, regarding the V12. This result can be explained by figure 14. In the W12, the number of bearings is reduced from 7 to 5. Furthermore, the W12 loss on each bearing is very close to the correspondent loss on the V12 (with the exception of the second bearing).

CONCLUSION

In this paper, the W12's engine architecture has been compared to a V12's, presenting the same geometry of the cylinder (bore, stroke, valves diameter, etc). A comparison has been made in terms of geometrical features, as well as in terms of safety coefficients, torsional stiffness, state of balance, and friction losses.

The W12 engine crankshaft is 158 mm shorter and 64% stiffer than the V12 (and also 90 mm shorter than a V10). Furthermore, this crankshaft is simpler and lighter. In any case, the W12 engine front section is wider. This last item imposes a brand-new design of the snorkel, with implications on aerodynamics not easily predictable.

The crankshaft of the W12 presents a minimum safety factor that is 30% lower than the V12's, under the same operating conditions (18000 rpm, bmep=13 bar). The W12 should, in any case, be safe (minimum safety factor 1.07)

While the V12 is perfectly self-balanced, in the W12's crankshaft, the secondary forces are out of balance; this unbalance is however no more critical than the one occurring in a V10 or V8.

Friction losses in the W12 should be slightly lower than in the V12.

The W12 can be considered as an alternative for the Formula 1 to the V10 architecture, since it allows to maintain most of the benefits typical of a traditional 12 cylinder, turning the problem of length and stiffness into a further advantage.

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NOMENCLATURE

α_l: Angle between the connecting rod and the axis of the cylinder 'i'.
$\alpha_{k\sigma}$: Bending stress concentration factor
$\alpha_{k\tau}$: Torque stress concentration factor
β: Angle between the cylinder 1 axis and the vertical axis
$\beta_{k\sigma}$: Bending effective stress concentration factor
$\beta_{k\sigma}$: Torque effective stress concentration factor
ϕ_i: Angle formed by the direction of the friction force with the vertical axis
γ: Angle between the cylinder 3 axis and the vertical axis
η_k: Notch sensitivity
θ: Crank angle
σ: Normal stress
σ_l: Bending fatigue limit
σ_m: Nominal cycle averaged normal stress
τ: Tangential stress
τ_l: Torque fatigue limit
τ_m: Nominal cycle averaged tangential stress
ω: Angular engine speed
a: Crankpin length
A: Section area
A_p: Piston area
b: Crank web width (along the shaft axis)
bmep: Brake Mean Effective Pressure
c: Journal length
C: Connecting rod mass

D_j: Journal diameter
d_j: Diameter of the journal drilling hole
D_p: Crankpin diameter
d_p: Diameter of the crankpin drilling hole
f: Friction coefficient
F: Centrifugal force acting on the balancing mass
G: Tangential elasticity modulus
J_p: Polar moment of inertia
k: Crankshaft torsional stiffness
K: Fraction of reciprocating mass to be balanced
L_f: Work per cycle done by the friction force
L_I: Equivalent length of the i-crank
m_b: Balance mass
m_{pin}: Crankpin mass
m_{rec}: Reciprocating mass
m_{rot}: Rotating mass
M_t: Engine torque
M'_t: Resultant of couples due to the other cranks
M_x: Bending Moment along the x-axis
M_y: Bending Moment along the y-axis
M_z: Torque Moment along the z-axis
n: Ratio between the connecting rod length and the crank throw
Nc: Number of cranks
n_s: Safety factor
N_z: Normal force along the z-axis
p_i: Gas pressure within the cylinder i
r: Crank throw
R: Ultimate strength
r_b: Distance of the balance weight's center of gravity from crankshaft axis
R_I: Force acting on the i-bearing
s: Crank web width (in the plane normal to the shaft axis)
S_r: Radial component of the force transmitted by the connecting rods to the crankpin
S_r: Transverse component of the force transmitted by the connecting rods to the crankpin
T_x: Shear force along the x-axis
T_y: Shear force along the y-axis
Z: Number of cylinder concurrent on the same crank
W: Bending section modulus
W_0: Torque section modulus