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TBO EVALUATION OF NAVAL AND AIRCRAFT DIESEL ENGINES

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ABSTRACT

This paper demonstrates that the experience from Formula 1 and watercraft racing can be applied directly to assess and improve the aircraft/maritime conversion of automotive commercial engines. A direct comparison of the main parameters that characterizes modern CRDID (Common Rail Direct Injection Diesel) and Formula 1 racing engine demonstrates that the similarities are hidden inside the design criteria. In fact, CRDIDs should output high torque at low rpm (1000-3000rpm) while racing engine should have top torque at 9000-11000 rpm. This fact introduces much shorter strokes in racing engines that reduce inertia loads. Since pressures are higher for CRDIDs the combustion loads are similar. The techniques used to improve the TBO of Formula 1 spark ignition engine and racing watercraft diesel can then be directly applied to naval and aircraft engines where the low-cost requirements are not so stringent as in mass-produced automotive CRDIDs (millions of items). The same technology that prolongs the Formula 1 TBO from a single race to the whole season can then be successfully used in aircraft/naval CRDIDs. A quantitative assessment of the TBO increase is included in this paper for the various systems that compose a CRDID.

Keywords: piston engines, TBO, common rail, load factor method, friction, wear.

INTRODUCTION

The duration of a combustion engine depends on its weakest component. Paradoxically reliability is a common critical factor in maritime/aircraft engines and racing engines. In fact, in aircraft engines certification and maintenance standards require very high reliability standards. Typically, the requirements are: 1 out of 10 thousand of malfunction probability and failure probability of 1 in 100 thousand in 3000h. Naval engines are similar but the minimum life is around 10,000h. The same requirement is present in races where the fate of a team often depends on the outcome of a single race. The only important difference is the life: 10, 000 hours for the navy, 1, 500 hours for aerial vehicles, 120 hours for Formula 1 racing. For this reason the experimental racing experience, that is much shorter, is vital also for the other "similar" fields.

Traditional reliability assessment methods, based on statistics, are difficult to use in engines. In fact, the failure or damage of a single component, even if redundant, may conduct to the failure of the whole engine in very short time (seconds). For example, if the high-pressure pump cam wears out, the debris may contaminate the lubrication system and the whole engine fails. This happens even if you have a double, redundant and completely separate injection system. On the other hand, in single-injector-per-cylinder injection engines, the application of extra-throttle to the other cylinders compensates the failure of injector/power-electronics, without reducing the power output. The reliability of the single components are so high, that in many cases it is difficult to represent it with numbers. So a simplified approach, based on functional groups is introduced herein.

In a first TBO (Time between Overhaul) evaluation, the engine accessories can be excluded. In fact, they are, in many cases, redundant and electric motor driven. External accessories can be substituted without "opening the engine". The EOBD (Electronic On board Diagnostic) system can easily detect the failure well before it occurs. The maintenance of piston engines follows the “closed engine concept”. This means that ordinary maintenance is conducted without opening the engine, just by substituting lubricants, filters and accessories. These operations are performed on the field, without requiring depot maintenance levels or specialized tooling. When a major failure occurs, the engine is overhauled as a whole.

Therefore, from the point of view of reliability and TBO, the engine can be seen divided into four parts.

a) Injection and related systems. In this point you can also include the FADEC (Full Authority Digital Electronic Control)

b) Piston assembly (blow by)

c) Bearings

d) Structural parts

In the following paper, each single group will be examined separately.

Basic concepts

The wear (duration) of an engine depends on friction. As a rule of the thumb, it can be said that: the more is the friction the more is the wear. This is true with

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the same technology: lubricant, materials, surface finish and treatments. However, giant steps have been made in this field in the last 20 years. For example, lubricant viscosity is no more the most important factor. In addition, coatings have greatly improved the wear resistance.

Injection and related systems

The common rail system is handled independently and can be removed from the engine from the outside in the same way as the ignition system in a petrol engine. The only component usually connected to the engine is the high pressure pump. However, in modern naval engines that drive a generator, also this system is completely decoupled from the main engine. A diagnostic system (EOBD) integrated into the FADEC (Full Authority Digital Electronic Control) enables the constant monitoring of the system with a safe determination of residual life and diagnosis of any malfunction of the sensors. The sensors, being redundant, tolerate malfunctions. Nearly all the sensors can be emulated by software. Only the throttle and the crankshaft rpm/phase sensors should be redundant to have the necessary reliability. Sensor software emulation limits the performance only in terms of specific consumption (no more optimized), without affecting engine life. The FADEC EOBD highlights the faulty sensor and it can be replaced during a stop without affecting the remaining life of the engine. In many cases, the diagnostics detects the degradation of the sensor well before its failure, giving the possibility to substitute it during a scheduled maintenance. In multi-injection technology (Patents Luca Piancastelli University of Bologna), in case of a failure of an injector, the second injector can restore cylinder power until the next stop. In multi-injection technology, even the failure of a rail or of one of the high-pressure pumps does not have appreciable operational consequences and does not reduce the TBO of the engine, nor the power delivery. In single injector-per-cylinder engines, uneven idling and the difficulty in starting the engine can detect early injection failure. Typical damage to the injectors is: power electronics reduced efficiency, tribological wear, pitting, cavitation and thermal damage. In the case, the damaged injector is not able to give a sufficiently large amount of fuel, resulting in difficult start-up and a non-uniform angular velocity of the engine. Reduction of the injected fuel results in a leaner fuel-air ratio. Increasing the amount of oxygen relative to the amount of fuel is accompanied by the increase in the speed and temperature of combustion. This process further increases the injector damage by overheating the injector tip and needle. In addition, damage of the return valve causes leaner air-fuel ratios. All these components can be replaced without "opening" the engine. The duration of the system depends on two factors: fuel and mapping. If the engine is not mapped correctly, the life of injectors and pumps can be severely impaired. In fact, if the software does not handle fuel of poor quality in terms of cetane and lubricity, life of injectors and pumps can be severely impaired. If the injection system does not handle the fuel contamination, life of injectors and pumps is also severely impaired. All these problems can and should be managed in software and hardware (both mechanical and electronic). At the software level, it is possible to use specialized libraries that manage all these eventualities.

Valves and their drive system

Valve technology and its operation, has suffered in the last 20 years a significant acceleration. Until the 2000s, the racing engines had to be revised very often; now you can, with a suitable mapping, run a complete season of Formula 1 without overhauling the engine. In terms of valves, there are significant differences between commercial spark-ignition and diesel engines. Due to special fuels, pressures and temperature of the racing spark-ignition engines are nearly aligned with those of diesel engines. However, the difference lies in the commercial diesel fuels that can be much more corrosive than racing ones. In today's F1 engines rarely you meet problems with valves, except in the event of accidental over speed. The modern Formula 1 technology allows for reliable and durable engines. The duration can be evaluated accurately. An algorithm implemented in the FADEC is capable of providing the residual life for various load profiles (full power, defined cycles and frequent start-stop). The software giving an evaluation of the residual life of the engine can also manage the thermal overload/underload. Lubrication of the valve train is in the mixed or boundary regime for most of its operating conditions. Measurements of valve-train friction-torque showed that the valve train friction torque increases as the viscosity of the oil decreases. It is also in the valve train that friction modifier additives and surface coatings as DLC (Diamond like Carbon) have the largest effect. In a valve-train lubrication model, the friction losses may be calculated with equation (1).

$$T = T_0 + c\mu\omega \tag{1}$$

Where friction torque T for the valve train depends on the boundary friction T_0. The second term represents hydrodynamic friction in the valve train system (both from the cam-follower contact and from the camshaft bearings). The boundary friction torque has been successfully calculated by estimating the oil film thickness using elasto-hydrodynamic theory. In this case, the effective friction coefficient evaluation starts from the combined surface roughness of cam and follower. Friction modifier additives, surface finishing and surface coatings are effective at reducing the boundary friction torque. Typically, an automotive 100 kW CRDID (Common Rail Direct Injection Diesel) will dissipate 600W in the valve train at maximum torque; 80% of this loss is due to T_0. At full power and load, the power loss will be nearly four

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times (≈1800W). Therefore, for high-speed engines it is compulsory to use low friction, low viscosity lubricants to limit the power and wear losses. With DLC the static friction loss will be reduced down to 1/3-1/4. In this case, the total loss will be 250W at maximum torque and 1000W at maximum power. This is about 1/2 the original power loss figure. However, the duration of the valve train will be increased about 20 times. In fact, the DLC surface coating is extremely hard and wear resistant. In equation (1) a lower viscosity lubricant will give a higher boundary friction torque. On the contrary, the hydrodynamic friction torque will be lower. Synthetic oils are far more viscostatic than mineral one even with additives (for viscosity at high temperatures). Low viscosity lubricants with additives to reduce friction act like high viscosity ones at low speeds. Therefore, surface coatings and additives make it possible to reduce lubricant viscosity. It is then possible to reduce the wear rate and the friction losses in the whole range of engine use. Technology is then of paramount importance.

Piston assembly (responsible of the blow-by)

The blow-by is currently the limit of CRDIDs. In fact, a motor correctly designed, at the end of the working life shows growing oil consumption. This fact is mainly due to the blow-by. The power output falls, then, to unacceptable levels. This behaviour is well documented in aircraft engines of World War II, like the RR Merlin and the Pratt and Whitney radials. With current technologies TBO have increased, but the criterion of the engine substitution has not changed.

There are many differences between a conventional diesel engine and a high performance racing spark ignition engine. For common CRDID engines, the maximum rpm is given by injection system dynamics (injectors, high-pressure pump and FADEC). This maximum speed figure is typically around 6,000 rpm, although piezoelectric injectors in small unitary displacement engines can reach 10,000 rpm. Typical automotive CRDIDs have a bore- stoke ratio around 0.8-1. This ratio has increased from the introduction of the Common Rail technology up to now (Figure-1).

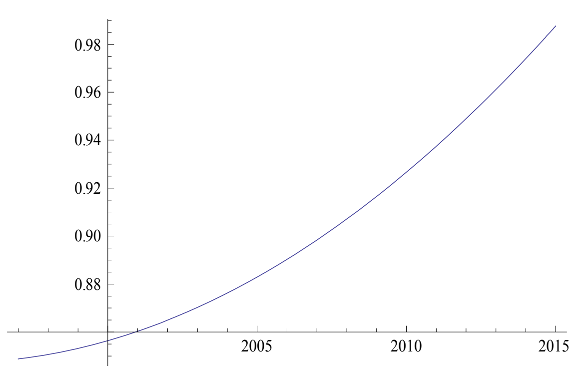

Figure-1. Bore stroke ratio vs. year.

In comparison, a V10-3000cc Formula 1 engine has a bore-stroke ratio in the range 2.0-2.5. The comparison seems to be improper, since Formula 1 engines are spark ignition engine, while diesel engines are traditionally slower in burning the fuel. However, this is not truly the case of modern CRDIDs, where ignition delays are extremely small at high loads (high pressure and temperature in combustion chamber). The reason of the long stroke in automotive CRDIDs lies in the requirement of torque at low rpm and the necessity of complete combustion with easy exhaust after treatment (to comply emission standards at relatively low loads). In case of naval and aircraft engines loads are less variables and in the upper part of the crankshaft speed range. In fact, propellers output power with third power of rpm.

From Table-1, it is possible to see that, for automotive and racing engines, the maximum piston linear speed is roughly the same.

In this case, it is strictly necessary to dispel some myths. The duration of an engine does not depend on the number of revolutions and even the number of cycles. It mainly depends on the average and maximum speed of the piston, and secondly by the maximum operating pressure. This is easily demonstrated both theoretically and empirically. In fact, given the MPS (Mean Piston Speed) we have (2):

$$MPS = 2 * Stroke * RPM / 60 \ [m / s] \tag{2}$$

where *RPM* is the number of revolutions per minute and *Stroke* it is the stroke in m. The piston travels in the time T (hours) a distance equal to

$$Length = MPS * 3600 * T \ [m] \tag{3}$$

Where T is the operating time in hours. The product of rpm and MPS outputs approximately the piston travel (5).

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Table-1. Comparison of F1 engine and CRDID.

	Formula 1 engine	2.0 CRDID
Engine Type	V10	Inline 4
Displacement (cc)	3000	2000
Unitary Dis. (cc)	0.3	0.5
Bore (mm)	91-100	83
Stroke (mm)	38-46	90.4
Bore/Stroke (bs)	1.97-2.61	0.91
TBO (h)	120	1500
Max Piston Speed (m/s)	36-41	29
Peak Combustion pressure (MPa)	11-14	16
Bmep (bar)	11-14	13
Max rpm	17000-18000	6000
MPS (m/s)	26.	12
MPS*3.6*TBO (km)	11000	98000
ω^2Stroke^4 bs^2 (m^4/s^2)	50-51	24
P_{peak}Stroke^{2}bs^2 (kN)	91-140	112.9

In traditionally designed modern engines, the maximum angle that the con-rod makes with the vertical should be smaller or equal to 15°. From this data, it is possible to calculate the maximum Piston Speed *MaxPS* of Table 1. This speed occurs about mid stroke, where the connecting rod is ninety degrees to the crankpin. In this condition, the hydrodynamic lubrication of the ring-to-liner assembly is prevalent. A practical approximate equation can also be used (4).

$$MaxPS = (Stroke \times \pi) \times RPM \qquad (4)$$

Note that for both types of engines (diesel and F1), the maximum linear speed is roughly the same. Therefore, MaxPS is not extremely critical. The maximum wear occurs at the topmost position of the piston near the TDC. The higher MPS and the low blow-by allowed of the F1 significantly reduce its TBO (last row, Table-1). In fact, the product of rpm and MPS outputs approximately the engine life (3). This is true up to MPS=20. Over this value, both friction and wear increase dramatically. Under MPS=20, very long stroke, very large displacement and low rpm engines may have the same TBO of small displacement, short stroke and higher rpm ones. In our case (naval, aircraft) the engine runs at constant speed, then the duration depends mainly on the tribology of the contact and the quality of the lubricant. The temperature on piston assembly is kept constant to the optimal value by cooling. In diesel piston engines, tribology is optimized by an already proven, advanced technology in the marine diesel field. The piston assembly life can be improved with surface treatments and materials. Coatings on the cylinder liners and improved piston rings may increase TBO four folds. In addition, lubricant viscosity is critical, synthetic oils with high temperature additives may improve the life two-folds. However, commercial spark ignition engines with MPS exceeding 25 m/s are present in the market. The actual champion is the Ford Mustang Shelby GT350, which has 93.0mm stroke at 8250 rpm with a MPS of 25.6 m/s. These values are obtained by surface coatings on the cylinder liner and on the piston rings as in F1 engines. In sports car, the full use of engine torque and power is very short. The LF (Load Factor) is then extremely low. Another reason for the relatively short TBO of F1 engines lies in the necessity to reduce friction, with a very limited number (usually 2) of very small rings. In a Formula V10 engine, friction loss due to piston assembly can be as high as 30 kW. Blow-by can be calculated with equation (5)

$$Q_b = \alpha\omega + \beta \frac{S_{stroke} \ln(i + \sqrt{i+1})\sqrt{D^3}}{n_r^{1/3}} + \gamma (B_{MEP})^2 \qquad (5)$$

α is the factor that takes into account the piston ring tension, while γ is related to the gas pressures. In a high performance engine, when there are often only two piston rings (the oil control ring and the top ring) a compromise must be made. If the oil control ring tension ring is reduced (to keep α low) oil consumption and blow-by Q_b may be too high (5). On the other hand, if the oil-control-ring tension is too high, oil consumption may be reduced, but there will be excessive oil starvation for the top ring, which would lead to higher friction (power loss) and high wear. This compromise is another reason of the high difference in the values of total length to TBO visible in Table-1 (See also equation 3). In any case, the friction is almost constant up to MPS=20 m/s and it has a steep increment over this value with an approximately cubic law of friction coefficient and wear.

Bearings

For low speed, conventional high compression-ratio DI (Direct Injection) diesel engines, the combustion gas pressure dominates the con-rod bearing load. Fortunately, since the output power of a propeller follows a cubic load, low rpm occurs at very low loads. However, as engine speed increases, inertial effects begin to dominate the not active cycle in four stroke engines and it also contrast the combustion-gas pressure ones. This is a very important condition since the peak pressure is usually located around 10-15 CAD (Crank Angle Degrees) after the TDC (Top Dead Centre) were inertia loads are close to the maximum ones. Since pressure loads are extremely high both in F1 and in CRDID, high inertia loads due to high rpm are favourable for con-rod and main bearings. In

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fact, bearing failures are typical of CRDID that run for long periods at low rpm and high load (high torque). This condition is typical on motorways (top gear and low speed). In this condition, for bearing loads that are dominated by gas pressure, the minimum oil film thickness in the bearing will occur at a position corresponding to the peak combustion chamber pressure. The oil film thickness will be extremely low as the cooling flow of the oil that depends on the pump rpm. Increasing local high temperatures mean low viscosity with decreasing oil thickness. For inertially dominated loads, the minimum oil film thickness position will occur elsewhere, and there may be several positions around the bearing where the oil film thickness is low. However, the bearings and bushings, if properly sized and lubricated, have no problems whatsoever.

Structural parts

Structural part reliability depends on both strength (con-rod) and stiffness (crankcase). Inertia loads are dominated by the acceleration a_p of the piston to TDC is given by equation (6).

$$a_p = \frac{\omega^2 Stroke}{2}\left(1+\frac{1}{\lambda}\right) \tag{6}$$

In motors with the bore expressed a fraction *f* of the stroke, the piston mass *m* depends on the cubic power of the bore multiplied by f^2, thus depends on the third power of the stroke. Therefore, the force of inertia F_i depends on the fourth power of the stroke and the second power of crankshaft speed (equation 7).

$$F_i = a_p m = \frac{\omega^2 Stroke}{2}\left(1+\frac{1}{\lambda}\right)m = f(\omega^2 \times Stroke^4 \times bs^2) \tag{7}$$

From Table-1 it is possible to see that inertia load of a F1 engine are approximately twofold the one of a diesel engine. However, the diesel piston is heavier than the F1 one so the inertia loads are not as different as it may appear.

The peak combustion load on piston is a function of the stroke and the peak pressure (8).

$$F_p = \frac{p_{peak} Bore^2 \pi}{4} = \frac{p_{peak}(bs \times Stroke)^2 \pi}{4} = f(p \times Stroke^2 \times bs^2) \tag{8}$$

Again, F1 engines are very close to diesel engines. Loads are then of the same order of magnitude.

The LF durability model

In the LF durability model, the rate of fuel burnt produces the TBO. The engine burns a certain mass of fuel through the engine in a short TBO when the engine output is large or you can have much longer TBO to burn the same amount with smaller power output. The Load Factor (LF) represents the relationship between fuel burned and the number of hours you are taking to burn it. In this case, the flight lasted 3.8 h. Hence, the LF can be calculated (9).

$$LF = \frac{Fuel_{Burnt}}{Fuel_{\max ratedpower}} \tag{9}$$

The typical car load factor is 0.44. This value can be used as reference LF_{ref} for european cars. It is then possible to calculate the load factor ratio LF_{ratio} (10).

$$LF_{ratio} = \frac{LF}{LF_{ref}} \tag{10}$$

A small car used for typical automotive light duty will have a TBO of 150,000 km. At the average speed 60 km/h, this means a TBO_{ref}=2,500 h. This can be the reference TBO for an automotive CRDID (TBO_{ref}). The LF to TBO law is quadratic**.** If the LF (mission profile) of the new application of the engine is known, the new TBO will be given by equation (11).

$$TBO = \frac{TBO_{ref}}{LF_{ratio}^2} \tag{11}$$

CRDID TBO improvement

Given the mission profile and the SFC surface (SFC/rpm/load) of the engine it is possible to calculate the new TBO form equation (11). The value obtained is approximated, as it is the concept of TBO that should be a statistic concept. However, statistics require large numbers to be significant and large numbers are not available for new application. The LF criteria are very simple to apply one you have CRDID/propeller performance data. Mission profile (duty cycle) depends on the application. For aircraft piston engines, an acceptable mission profile is summarized in Table-2.

Table-2. Typical small piston aircraft duty cycle.

Flight phase	**Power setting** (%)	**Duration** (h)
TO	100	3/60
Climb	92	0.5
Cruise	73	3
Descent	60	0.5
Total		4.05

To understand Table-3, the procedure for the "cruise" line is explained. The automotive OM651-100kW@4600rpm is, just for explanatory reasons, taken

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as it is and installed on an aircraft. The propeller absorbs the power in a cubic way. Therefore, it is necessary to run the engine at 4, 145 rpm so to achieve 73% (73kW) of the maximum output power for cruise (12).

$$rpm_{cruise} = \left(\frac{P_{cruise}}{P_{\max}}\right)^{\frac{1}{3}} \times RPM = 4145[rpm] \quad (12)$$

At rpm_{cruise} the maximum torque from the engine is about T_{rpm_cruise}=243 Nm. The power available from the engine at 4,145 rpm is then again $P_{max\_rpm\_cruise}$=100 kW. Then it is confirmed that the engine is able to output the required power P_{cruise}=73 kW, but the power output should be 73% of the maximum available (4) at rpm_{cruise}. From the specific fuel consumption surface it can be seen than the SFC (Specific Fuel Consumption) at cruise point is SFC_{cruise}=230 [gr/kWh]. The fuel consumption per hour at cruise will then be evaluated (13).

$$FC_{cruise} == P_{cruise} SFC_{cruise} = 16.8[kg/h] \quad (13)$$

The cruise part of the duty cycle fuel consumption is then (14):

$$Fuel_{cruise} == FC_{cruise} t_{cruise} = 50.4[kg] \quad (14)$$

The values calculated in Table-3 and equations (12) (13) and (14) are just for explanatory reasons; the Manufacturer should supply the true data of the engine.

Table-3. Aircraft piston engine duty cycle-fuel consumption.

	% Load	**rpm**	**T(s)**	**Fuel rate** (kg/h)	**Total fuel** (kg)
TO	100	4,600	3/60	25	1.25
Climb	92	4,473	0.5	22	11.05
Cruise	73	4,145	3	16.8	50.5
Descent	11.55	3,879	0.5	13.8	6.91
Total			4.05		69.7

From Table-3, it is possible to calculate an approximated duty-cycle fuel consumption of 69.7 kg. In the very simplified Load Factor (LF) durability model, an engine has a lifetime that is the weight of fuel burnt. You can run that mass of fuel through the engine in a short time period if you are extracting large amounts of power, or you can take much more time to burn the same amount if you only extract small amounts of power. The Load Factor (LF) represents the relationship between fuel burned and the number of hours you are taking to burn it. In this case, the flight lasted 4.05 h. At max power (load), the fuel burnt during the flight would have been 25*4.05=101 kg. This fuel consumption is an approximated by defect, a more precise value can be calculated or measured by the FADEC. Therefore, the LF=69.7/101=0.69 (9) and the LF_{ratio}=0.69/0.44≅1.6 (10). The new TBO for the aircraft engine is then $TBO_{aircraft}$=2500/(1.6^2)≅1000h (11). For a longer TBO several options are available. The simplest ones are to change the original oil to a more performant one. For example, a more viscostatic oil reduces hydrodynamic friction. Oil additives further improve TBO. Low temperature additives reduce friction at startup and in valve trains at low rpm. High temperature ones reduce friction and wear for the piston assembly. In general, additivities improve engine performance with the drawback of more frequent oil substitution. Table-4 summarizes the improvements obtainable starting from a top quality commercial automotive CRDID.

Table-4. % increase in TBO.

	Low T additive	**High T additive**	**Coatings**	**Surface finish**
Valve train	100%	-	1000%	10%
Piston assembly	10%	100%	400%	5%
Bearings	100%	-	20%	10%

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For example, from Table-4, if you apply DLC in the valve train (all contact surfaces), a 1000% (ten folds) improvement in TBO is obtained. If also a very high quality Low Temperature additive is used and additional 100% is obtained with a twenty fold improvement of the TBO for the valve train. The improvement of the surface finish is limited. If the original engine is an automotive top quality CRDID produced in millions, an additional 10% can be obtained for a total of 22 folds the original TBO (for valve train only). These values are taken directly from the Formula 1 engine tests and from experience in top performance engines for racing watercrafts.

CONCLUSIONS

The load factor criteria make it possible to evaluate the TBO of an automotive engine from the duty cycle of the new application. It is then possible to evaluate if the engine has a sufficient TBO and suitable scheduled maintenance intervals. If a TBO improvement is required, this paper demonstrates that the experience from Formula 1 and watercraft racing can be applied directly to assess and improve the aircraft/maritime conversion of automotive commercial engines. A direct comparison of the main parameters that characterizes modern CRDID (Common Rail Direct Injection Diesel) and Formula 1 racing engine demonstrates that the similarities are hidden inside the design criteria. In fact, CRDIDs should output high torque at low rpm (1000-3000rpm) while racing engine should have top torque at 9000-11000 rpm. This fact introduces much shorter strokes in racing engines that reduce inertia loads. Since pressures are higher for CRDIDs the combustion loads are similar. The techniques used to improve the TBO of Formula 1 spark ignition engine and racing watercraft diesel can then be directly applied to naval and aircraft engines where the low-cost requirements are not so stringent as in mass-produced automotive CRDIDs (millions of items). The same technology that prolongs the Formula 1 TBO from a single race to the whole season can then be successfully used in aircraft/naval CRDIDs. A quantitative assessment of the TBO increase is included in this paper for the various systems that compose a CRDID.

It is then possible to evaluate the feasibility of the usage of an engine for application fields like watercrafts, yachts, aircrafts and helicopters.

Symbols

Symbol	Description	Unit
T	Friction Torque	Nm
To	Sliding friction torque	T0
C	Constant for hydrodynamic component of frictionj	$Nm/(s^2Pa)$
μ	oil viscosity	Pa s
MPS	Mean (Average) Piston Speed	m/s
Stroke	stroke	m
RPM	Max engine rpm	1/s
Length	Piston travel to TBO at RPM	m
T	time	h
MaxPS	Maximum piston speed	m/s
Q_b	Blowby volumetric flow rate	m^3/s
α	Blowby speed constant	m^3
i	Number of cylinders	-
n_r	Number of rings per piston	-
β	Blowby geometric constant	$m^{-3/2}s^{-1}$
γ	Blowby BMEP constant	m^4s/kg
BMEP	Brake Mean Effective Pressure	Pa
a_p	Piston Acceleration at TDC	m/s^2
ω	Crankshaft angular velocity	1/s
r	Cranks radius	m

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l	C_L Eye to eye con-rod length	m
λ	r/l	-
m	Piston assembly mass	kg
bs	Bore/Stroke	-
F_i	Inertia load	N
F_p	Combustion load	N
P_{peak}	Peak combustion pressure	Pa
LF	Load Factor	-
P_{cruise}	Cruise engine power	kW
P_{max}	Max engine power	kW
rpm_{cruise}	Propeller rpm@cruise	1/min
FC_{cruise}	Fuel burnt rate@cruise	kg/h
SFC_{cruise}	Specific Fuel Consump.@cruise	kg/(kWh)
F_{cruise}	Fuel burnt @cruise duty cycle	kg
T_{cruise}	Cruise time in duty cycle	h

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