

Development of V8 Engine for American Le Mans Series

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ABSTRACT

A V8 engine was developed to optimize it for the Prototype 2 category (LMP2) of the American Le Mans Series (ALMS) racing. This engine was introduced at the 2007 Sebring 12 hour race.

The technical character of the performance, reliability and serviceability were all changed while keeping within the ALMS regulations such as a 44 mm inlet restrictor for the 3.4L V8 engine. The Indy Racing League (IRL) engine was utilized for the comparison reference. The effects of the significant differences between the series, such as the change from ethanol fuel to an ethanol/gasoline mix and changing from a rev limited engine running mainly on oval courses to an intake-restricted engine, running on road and street courses are discussed.

New test procedures that we created to assess engine hardware changes are discussed. Novel design features of the engine such as the valve gear detail and the combustion chamber design are reviewed.

要 旨

アメリカンルマンシリーズ(ALMS)のプロトタイプカテゴリ LMP2 グループ用のエンジンの開発をおこない、2007 年セブリング 12 時間レースに初参戦をした。

エンジンは LMP2 のレギュレーションに従って V 型 8 気筒 3.4 L, リストリクタ径 ϕ 44mm を使用しており, そのレギュレーションおよび耐久レースの特徴に沿ってエンジンの出力向上, 耐久信頼性向上さらにメンテナンス性向上を中心に開発をおこなった。ディメンションに共通部分が多いインディレーシングリーグ(IRL)のエンジンを比較対象としたが, IRL と ALMS ではサーキットにおいてはオーバーメインとオードコース, 燃料仕様においてはアルコールとガソリン-アルコール混合, さらには回転数リミットと空気リストリクタ付きなどの基本的な違いがある。

ALMS のレースの特徴にマッチした動弁機構や燃焼室の仕様決定や実験方法の確立をおこなった。

1. Introduction

IRL rules give detailed limits on major engine dimensions including engine capacity, number of cylinders, cylinder bore, overall engine length, minimum engine weight, crankshaft dimensions, injector position, airbox dimensions and engine sensors⁽¹⁾. ALMS rules are set by the organization responsible for the Le Mans 24 hours race, the Automobile Club de l'Ouest (ACO) and, in contrast, allow a variety of engine configurations and capacities⁽²⁾.

Table 1 highlights the major differences in the rules between the two series and it can be seen that the ACO rules allow much more freedom of design and development over the IRL rules. While both series call for mechanical throttle control, fixed inlet lengths and fixed valve timing, they use different fuels and have different means of trying to limit engine power with the

IRL controlling engine speed and the ALMS restricting the air intake. It was in these two areas where we focused our development.



Fig. 1 General view of ALMS engine

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Table 1 Comparison of IRL and ALMS rules

	IRL	ALMS
Engine configuration	3.0 liter, 90° V8	Up to 3.4 liters, 8 cylinder, N/A or Up to 6 cylinder supercharged.
Inlet manifold	Up to 2 injectors. Placed between throttle and head face. Fixed length	8 max. Vee or in-line
Power limit	10,300 rpm limiter	Inlet restrictor. Size based on engine configuration
Development constraints	Major castings, ports, combustion chamber, pumps, valve train gears	None
Fuel	100% ethanol	92% gasoline/10% ethanol
Noise limit	IRL-supplied silencer	113 dBA

2. Engine Specification

The priorities for the engine were reliability foremost, engine cost second and engine power third. In several cases this led us to choose the most reliable solution rather than the one that would give the greatest performance.

When we made the decision to enter the ALMS, the development engine specification was 2.8 liters. It had downward-facing injectors located just downstream of the throttle bodies with supplementary injectors located at the manifold-to-head joint. This joint face was as close to the inlet valves as possible to minimize the distance from these lower injectors to the cylinder. We did not change the main engine architecture as we wanted to use our resources to concentrate on other areas which had to change.

One such area was the airbox. The IRL airbox was flexible and leaked air where the air snorkel fitted to it and where it fitted to the head. The ALMS officials carry out a leak test at events in which the engine inlet has to maintain a 150 mbar vacuum for over half a second. The air box, made from carbon-fiber, reinforced plastic (CFRP), was stiffened by adding foam inserts to the floor and sides. The IRL specifies the latch to fix the airhorn to the airbox. These have a spring element in them. On the dynamometer, the combustion air supply pressure was sufficient to lift the airhorn off the airbox when the throttles closed. At higher speeds where the restrictor took effect, the joint closed up. This fatigued the fasteners in a few hundred miles' running. We changed to self-locking fasteners similar to those used to attach the car's body panels to the chassis.

Studying the ACO table of restrictor size against engine capacity showed that the 3.0 liter with its 44.7 mm diameter restrictor would allow the highest power

Table 2 Restrictors for normally-aspirated gasoline engines

	> 2 valves per cyl.	2 valves per cyl.
< 3000 cm ³	44.7	46.2
>3000 and up to 3200 cm ³	44.3	45.8
>3200 and up to 3400 cm ³	44.0	45.5
Homologated engines up to 4000 cm ³	44.7	46.2

output but we chose the more conservative route of having the maximum capacity with its corresponding smaller (44.0 mm) restrictor on the basis that this combination chokes at a lower engine speed. Reducing the engine speed would extend engine life.

The original IRL engine capacity was 3.5 liters, although this was reduced to 3.0 liters in 2004, so we knew that the crankcase would accommodate the longer crank throw to achieve the higher swept volume. Another change between the two series is the higher electrical load in ALMS to support the use of electric fuel pumps, electric power steering, lights, an on-board engine starter and, in our case, an electrically-operated gearshift. This meant we had to accommodate a larger alternator. This is driven off the back of the water pump, running at 0.48 engine speed. To protect them from vibration and heat, the alternator's electronics are remotely-mounted in the radiator inlet duct.

3. Inlet System

Changing from ethanol to 92% gasoline and 8% ethanol fuel reduced the injector flow rate requirements. We decided to maintain the two-injector-per-cylinder layout as this allows us to modify the torque curve based on the proportion of the fuel flow through the upper injector. The lower injector near the inlet valves gives good response but little charge cooling from fuel evaporation while fuel from the upper injector cools the charge but displaces air. In an effort to reduce the cost

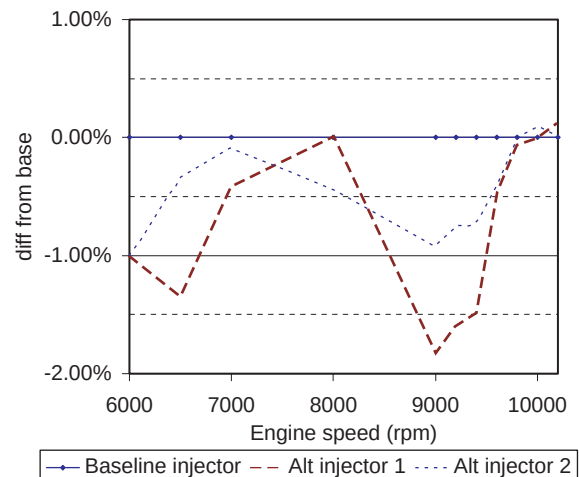


Fig. 2 Injector test results

of the engine, we tested injectors used on several Honda production cars. The 2006 IRL engine ran 10 bar fuel pressure but the production injectors would not open at such a high pressure. We reduced the fuel pressure to 5 bar to suit the injectors. Figure 2 shows power variation as injector type is changed. One can see that the alternative injectors reduced performance by almost 2% but, for this series, reliability is the most important characteristic. Rather than going for the injectors which gave the most power, we chose the Alternative injector 1 as this design gave the most reliable starting, an important consideration for this series as the engine has to be shut off in pitstops and restarted using the on-board starter. The figure also shows that, when running in the restricted region, the injector has almost no effect on power.

The governing body caused some confusion over the restrictor size. The ACO reduced restrictor diameters for the P2 class by 5% for 2007. We set about optimizing the intake length to suit the lower choking speed this gave us. To ensure reliability, we had one-piece trumpets made of the appropriate length. At the end of the January, the International Motorsports Association (IMSA) who controls the ALMS rules enforcement announced that they would maintain the larger, 2006 restrictor sizes. This meant that our intake was too long for optimum performance but we decided to keep the one-piece trumpets for Sebring in the interests of reliability. Another late rule change was the introduction of fuel consisting of 92% 100 octane gasoline and 8% ethanol. This cost some performance and economy while needing a complete engine re-map.

4. Cylinder Head for ALMS P2

The cylinder head is a 4 valve per cylinder design with 10.4° exhaust angle and 8.5° intake angle. The poppet valves are fitted with twin coil springs, and are activated by a pair of camshafts per bank with finger followers located on a common axis.

Oil for the cylinder head is fed through the cylinder head to the camshafts. The shafts are hollow and distribute oil to the finger followers, cam bearings and spring cooling. There are two scavenge ports at either end of the spring chest core, and the core has a flat

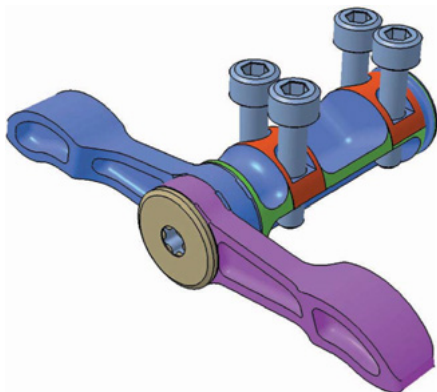


Fig. 3 Early swing arm location method

bottom to aid oil drain.

Support for the camshafts and swing arms are from twin buttresses between cylinders. This design was chosen for the space saving through the cylinder area.

The spark plug centrally located in the compact chamber passes through the spring chest core. A light weight alloy tube sandwiched between the cam cover and the head seals it from the engine oil. The ignition coils reside in these tubes and the wiring is attached to a carrier for the coils, which is covered by a CFRP cosmetic cover.

The swing arm assembly has been designed to be stiff, light weight and low cost. Originally the shaft was designed as an interference fit to the top faces colored red in Figure 3.

The shaft could be fitted by hand and then drawn upwards to lock with the bore by tightening the bolts. This led to a rather unsatisfactory condition where by the forces on the followers were great enough to overcome the system and lead to instability.

We redesigned the cylinder head to join adjacent pairs of buttresses together enabling a bolt through system that is now used in the race engines. Figure 4 shows how the pair of pivots are joined through the buttresses via an M8 stud and the finger followers are then retained by a special keeper. Lubrication is performed by splash via slots machined in the upper thrust faces of the arms.

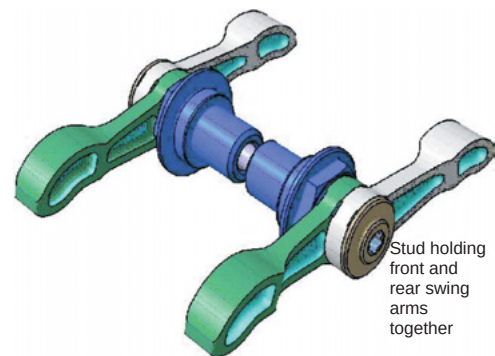


Fig. 4 Final swing arm shaft detail

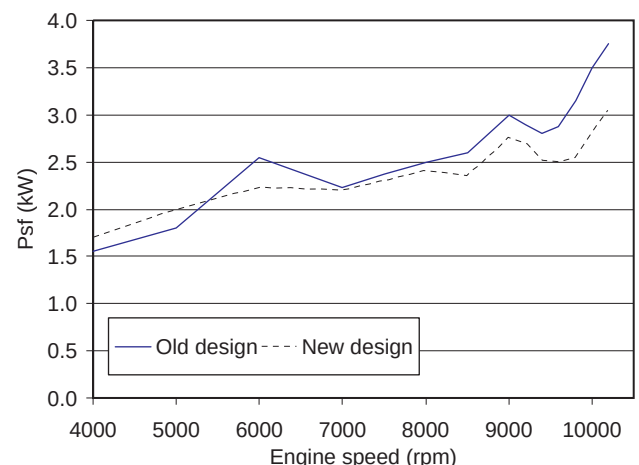


Fig. 5 Friction reduction with new design of shaft fixing

An unexpected side effect from this change was that the valve train requires less power to drive. As shown in Figure 5, the new fixing method reduced the motoring power by 20% at high engine speeds.

5. Durability Testing

The IRL engine target life was 600 miles. The initial target for the ALMS engine had to be at least 1400 to cover the Sebring 12 hour race. Moreover, the duty cycle between an IRL engine designed to operate mainly on oval circuits and the ALMS engine operating on road and street course would be different. At the time of the first durability test, we had no track data to use to set a durability cycle. Instead we used a simulation of a fast lap of Sebring with the power curve the engine had and the characteristics of the P2 chassis. The simulation had been developed by a consulting company for use in the IRL program so we were confident in its accuracy. In fact, when we did run at the track, the actual trace was close enough to the prediction that we made no change to the test cycle. After the Sebring race, where the combination of a tire with increased grip and the developed aerodynamic package allowed so much more wide-open throttle (WOT) running, we needed to revise the cycle. Figure 6 below shows the original and final traces.

Originally, we expected to have to include slow laps in the cycle but we soon learned that the races are run like sprint races. The drivers do not drive to conserve the car.

Durability testing ran on a test cell equipped with an alternating-current (AC) dynamometer to give fast response. The shift cut settings in the engine control unit (ECU) were tuned to match the dynamometer control to simulate shift-without-lift upshifts. The total cycle distance for a successful was equivalent to 1517 miles, based on a 3.7 mile lap of Sebring.

Initial tests ran standard atmospheric conditions but we were concerned that races as long as 12 hours take place under a variety of conditions and some may stress the engine more than others. Running at fixed

conditions may miss the worst case and allow the engine to pass the test but fail on the track. Therefore, later tests used weather data taken from the 2006 race to vary humidity, temperature and barometric temperature. Figure 7 shows the inlet temperature and pressure variation over the test.

We did not take into account the charging effect due to the car moving through the air. This lead to detonation damage to car engines which we had not seen on the dynamometer engines. Following the realization of the importance of this, we vary the combustion air system pressure with simulated car speed.

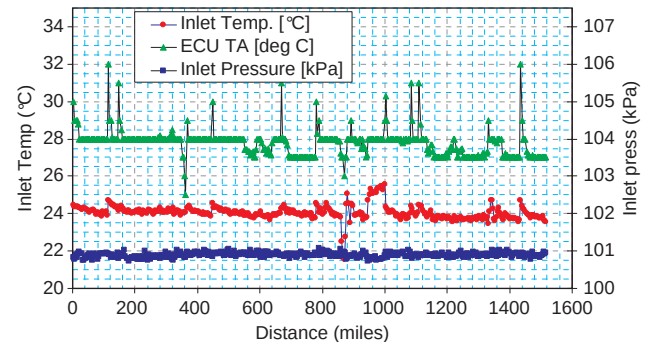


Fig. 7 Air inlet condition variation over test cycle

6. Performance Testing

We used a technique of 1-D prediction to define the flow targets, flow-bench tests of SLA models (both inflow and outflow), prediction using these results, fine tuning of the shape and flow-bench and dynamometer testing of machined parts. Early tests showed a good correlation between the SLA models and actual cylinder heads provided the valve seat was accurate in the SLA piece although we did have some instances where the aluminum part results were different from the SLA results. This always was when the port had to change from the original design to accommodate manufacturing requirements. Port refinement was already underway as part of the IRL engine development when we changed direction. During this testing we modified the NE limit calculation to divide it by a typical volumetric efficiency factor, in this case 120%. This gave good agreement with the NE limit and the peak power engine speed.

To try to assess the effects of port design changes on drivability, we devised a transient power test using a rate of 2000 rpm/s based on acceleration rates from the 2005 St. Petersburg IRL race. 100 Hz data was smoothed by using a moving average of 7 data points. This testing confirmed that the IRL inlet port we had developed (IP26) would be too large for ALMS use. The first iteration, IP34, was created from the IP26 by adding material to the port floor. A further, later modification to improve lower injector nozzle clearance changed its designation to IP37. Another iteration, IP35, reduced the port volume further. Figure 8 compares the IP26 and IP35 ports.

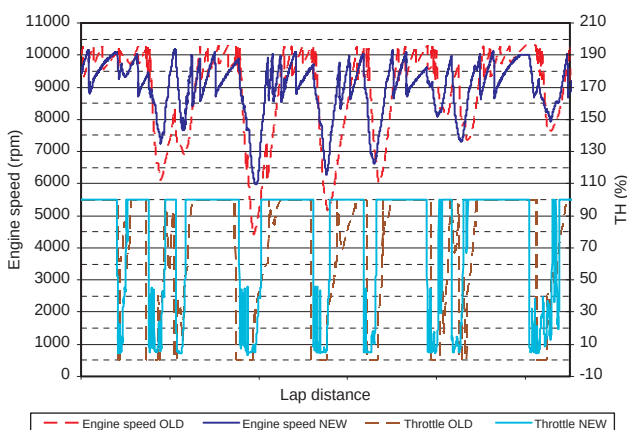


Fig. 6 Simulation speed throttle trace and final track data

Another new test, where the engine was held at WOT and 6500 rpm when the dynamometer control switched to absorb a constant 350 Nm. The time taken for the engine to accelerate to 10000 rpm was used another measure of the port's effect on engine response. Figure 9 shows an example of the results of such a response test with different sizes of inlet port.

These tests confirmed the improved response with the smaller port but also showed that we had not yet reached the lower limit. This led to the design of the smaller-diameter IP40 port. By this stage, port volume had dropped from 403 cc, for the IP26, to 320 cc for the IP40. As Figure 10 shows, this did provide a further benefit over the IP35 port at low speed and just before the restrictor went sonic but it was so small that we felt we were near the limit. Thus, rather than continuing to investigate smaller and smaller ports, we selected IP40 as the race port.

Apart from the ports being set to suit IRL racing, the camshaft design and timing also favored high-speed running. We designed several alternatives, with combinations of revised accelerations, lift version and duration as shown in Table 3.

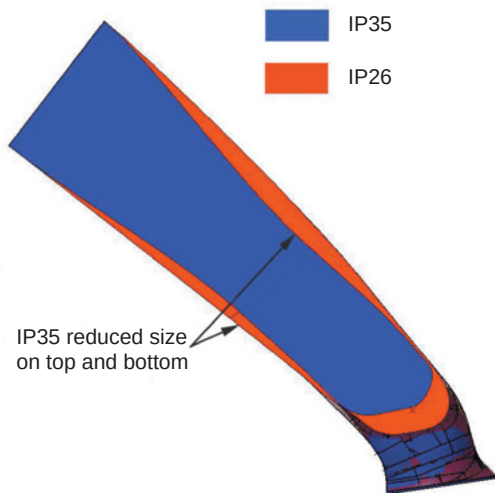


Fig. 8 Changes from IP26 to IP35 inlet port

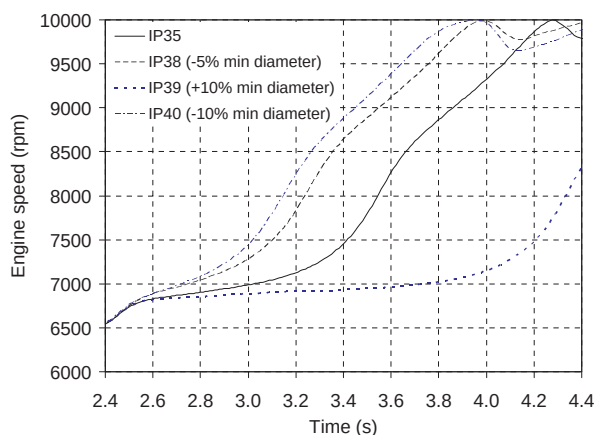


Fig. 9 Response of different inlet ports

Table 3 Development camshaft details

Camshaft spec	Intake		Exhaust	
	Lift	Duration	Lift	Duration
A400	15.34	253	12.46	236
B400	15.34	253	12.66	237
C400	14.33	254	11.65	237

The results of the camshaft changes can be seen in Figure 11.

As expected, the lower-lift intake camshafts suited the low speed nature best while the exhaust camshaft had little effect on performance.

Performance development is an on-going process in racing. We are still investigating the ideal inlet length, camshaft profile, exhaust layout, piston shape oil viscosity and pressure while trying to remove weight and cost from the engine.

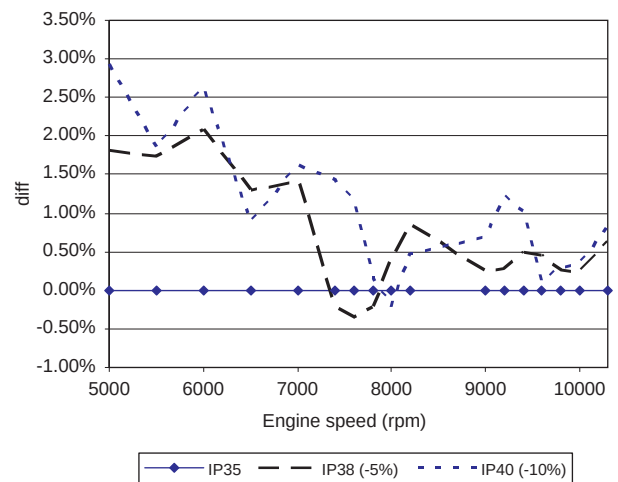


Fig. 10 Comparison of torque with different inlet ports

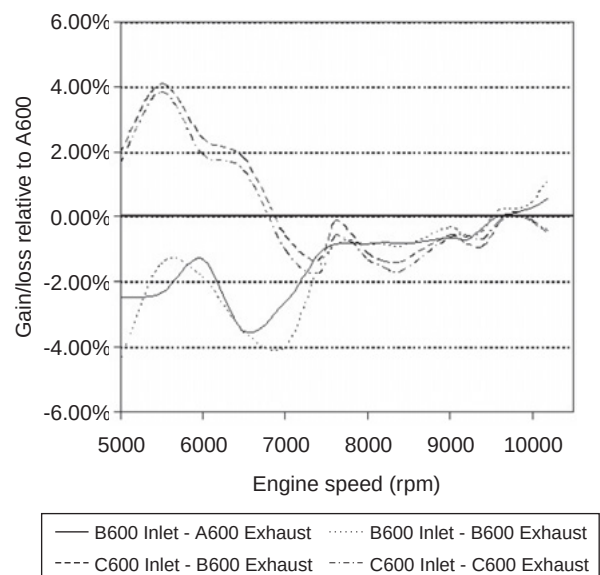


Fig. 11 Camshaft comparison

7. Conclusions

We successfully have developed an ALMS LMP2 engine. Major changes from an IRL engine were focused on the airbox, fuel injector and alternator.

Despite using a conservative approach to ensure reliability, we were able to make performance improvements.

Using a knock-control system together with piston crown geometry changes allowed us to use as high a compression ratio with 92% gasoline / 8% ethanol mix as we had been using with 100% ethanol fuel.

Reference

- (1) 2006 Indycar® Series Rule Book, p.147-155
- (2) Regelement Technique ACO – ACO Technical Regulations 2007 Prototype “LM”P1 & “LM”P2, p.25

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