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SOCIETY OF AUTOMOTIVE ENGINEERS, INC.
400 Commonwealth Drive, Warrendale, Pa. 15096

PEUGEOT — RENAULT — VOLVO 90° V6 ENGINE

(6 x 88 x 73 — 2.664 cm3)

François Gastinne
Societe Anonyme des Automobiles Peugeot

M. Laliere
Regie Nationale des Usines Renault

Stephen Wallman
AB Volvo

SOCIETY OF AUTOMOTIVE ENGINEERS

Automotive Engineering Congress and Exposition
Detroit, Michigan
February 23-27, 1976

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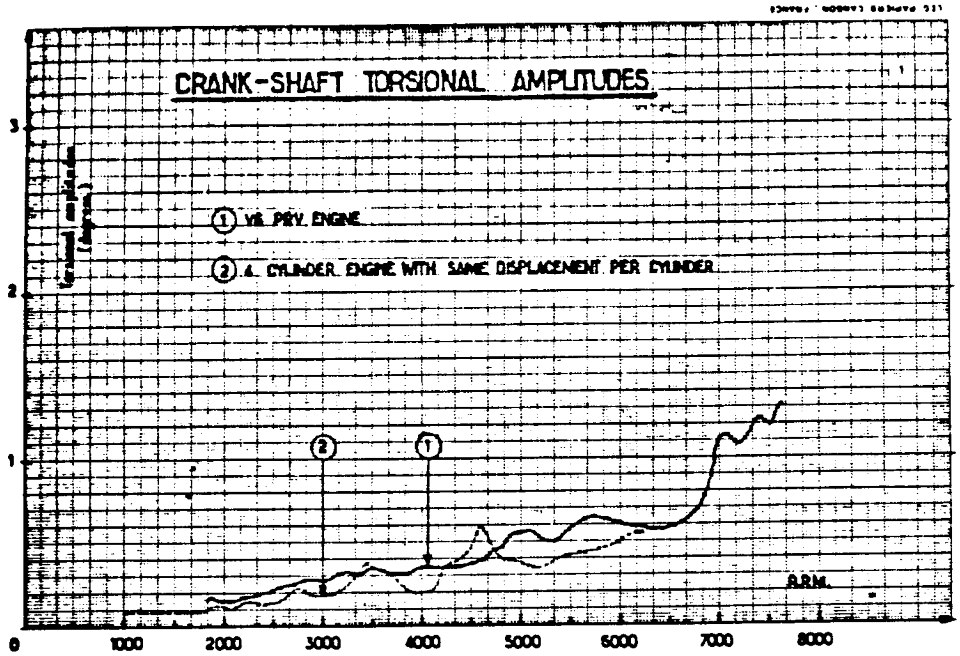


Figure 1

GENERAL CONSIDERATIONS BEFORE THE ACTUAL RESEARCH (Factors considered in the choice of the 90° V6 engine)

In 1968, the design of an engine with a comparatively large cubic displacement to be fitted in prestige European cars was commenced. Already this consideration favors the 6-cylinder solution and a rapid analysis of the problem from the standpoint of perceptible qualities confirmed this orientation. The most convenient 6-cylinder engine was to be selected.

DETERMINING ADVANTAGES OF THE 90° V6 ENGINE

Broad RPM range - The natural torsional frequency of the crankshaft of the 90° V6 engine is very high in comparison to that of other 6-cylinder engines (thanks to its three crank-pin crankshaft instead of 6 for others); thus, it permits high RPM (see Figure 1 where torsional amplitudes through speed range are figured).

As it will be seen however, the engine also has a good ability to operate properly at low speeds and full load.

This broad RPM range enables quick passing which, even within the present context of speed limitations and fuel consumption reduction, is of importance for safety.

Length and weight savings - For the same displacement, the saving in length is about 250 mm. in relation to the in-line engine.

The length reduction is at the cost of a more complicated construction, which makes the engine itself heavier if cylinder blocks are made of the same material. However, the compactness of the V6 approach has two advantages which permit weight reduction far more substantial than the weight increase resulting from the first remark.

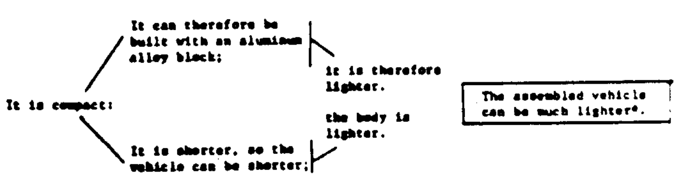
These advantages are the following:

- possible car length reduction by approximately the 250 mm., and
- possible use of an aluminum cylinder block.

In fact, the switch from the 13,000 daN/mm2 cast-iron modulus of elasticity to the 7500 daN/mm2 modulus of aluminum is consistent only with compact structures to avoid structural vibrations.

The following shows how the compactness of this engine can be used to advantage:

Chart No. 1



REVIEW OF FACTORS WHICH MAY BE CONSIDERED AS UNFAVORABLE

Overall balance imperfections - At first, the 90° V6 appears less satisfactory than the 6-cylinder in-line engine in terms of overall balance. As for us however, the comparison was made for 4-cylinder engines (the only engine type in general used by all three manufacturers concerned).

A brief report on the overall resultants of all inertia forces, which are practically impossible to balance in both above-mentioned engine types, will follow.

For the sake of simplification, calculations were made through the evaluation of "free movement"** for both engine types assuming that the weight of their mobile parts were the same, thus assuming the same displacement per cylinder.

The assumption above is obviously unfavorable to the V6 type but it is all the more significant.

A four cylinder engine is subject to a 2nd order sinusoidal force applied in the vicinity of the center bearing and parallel to the axis of cylinders, with an amplitude of:

$$\frac{4 M r^2 \omega^2}{\ell} \quad *** \qquad (1)$$

For the 90° V6, the resultant of inertia forces is a 2nd order sinusoidal torque, whose axis is perpendicular to the crankshaft and situated within the bisecting plane through both banks of the V. The amplitude is as follows:

$$= \sqrt{6}\ \frac{M}{\ell}\ r^2 \omega^2 \quad * \qquad (2)$$

On account of the overall masses for each engine (which may be calculated a priori), the "free movement" of the 90° V6, measured at the outer main bearings, is six times smaller. This clearly shows that one obtains greater freedom in choice of engine mounts.

Experience in vehicle confirmed this to a great extent and the noise transmitted to the body because of the imperfection of overall balance is practically imperceptible.

Cyclical regularity - The 3 crank-pin crankshaft requires ignition at irregular angular intervals (90° - 150° three times during a cycle).

Based on a gas pressure-angle diagram and taking into account instantaneous torques resulting from inertia, the instantaneous torque for 3 engine types having the same unit displacement (see Figure 2) was calculated:

- 4-cylinder engine
- 6-cylinder in-line engine (or 60° V)
- 6-cylinder 90° V engine

Though these calculations do not take into account internal frictional forces, these results are accurate enough for comparison purposes.

To express more clearly the differences between the three engine types above, the follow-

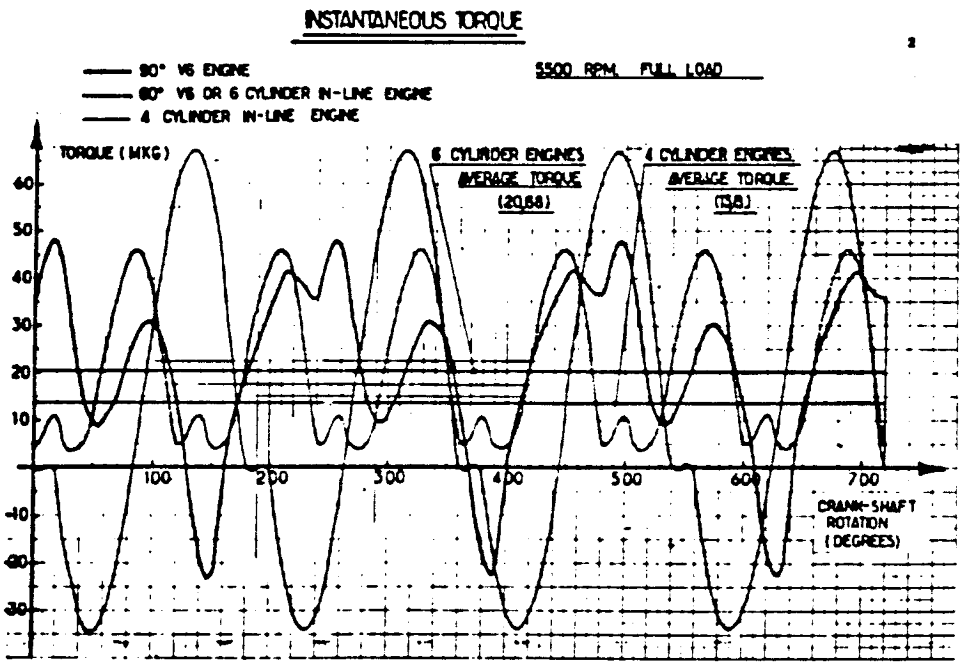


Figure 2

* In fact, this gain in weight certainly makes it possible to reduce the dimensioning of all components calculated as a function of the mass to be transported (suspension and braking components, structure, bumpers).

** free movement of the engine subjected only to the influence of its own inertia force or torques, its weight supported by infinitely soft mounts.

*** M = reciprocating mass per cylinder
r = half stroke
ω = angular rotational speed (radian/second)
ℓ = length of connecting rod

ing procedure was adopted:

One has calculated the areas enclosed between the straight line of the average torque and the instantaneous torque curve and related the sum of their absolute value to the energy supplied during one cycle. One obtains lower values as the engine runs more smoothly:

Chart No. 2

	2,000 RPM	5,500 RPM
4 cylinder	1.01	2.2
6 cylinder in-line (or 60° V)	0.63	0.67
6 cylinder in 90° V	0.73	0.71

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* M = reciprocating mass per cylinder
r = half stroke
ω = angular rotational speed (radian/second)
ℓ = length of connecting rod
a = centerline distance of cylinders

Looking at the balance-sheet between the advantages and the drawbacks of the 90° V6 engine, we find that the former outnumber the latter, mainly due to the wide range of engine speeds and the substantial gains in compactness and total weight.

SPECIFIC FEATURES RELATED TO ENGINE MECHANICAL COMPONENTS

FEATURES DERIVING FROM THE FUNDAMENTAL CHOICE OF A 90° V6 ALUMINUM ENGINE

Dimensions of connecting rod bearings, length off-setting between V banks - The interdependance between these two design parameters is well known to those skilled in the art; it derives from:

- the minimum diameter of the cylinder liner bore, which affects the dimensions of connecting rods, hence the diameter of crank-pins;
- the specific pressure allowed on the aluminum tin alloy bearing material, which is based on the above-cited diameter and imposes a bearing width, hence the width of connecting rod big end.

From the two criteria above, a 17.5 mm. offset in length between the two cylinder-banks was derived.

It was essential to reach the minimum value since it affects both the overall length and the efficiency of transverse structure ribs.

Size of bearings. Definition of the crankshaft. - The dimensions of the crank-pins being defined, it remains to settle:

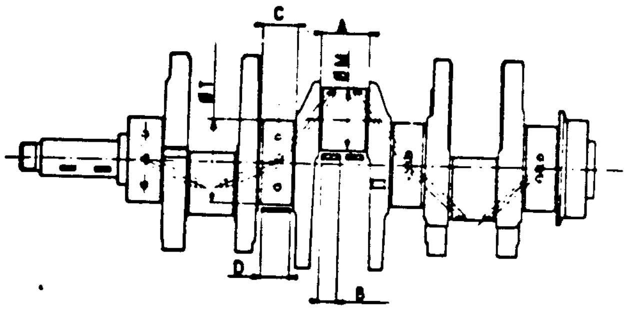
- the dimensions of journals;
- the thickness of arms (depending on the crankshaft material: nodular graphite spheroidal cast-iron).

If we take into account the choice of aluminum tin alloy for the shells, and knowing the bearing forces, all the characteristics are fixed when the diameter of the journals are determined according to the following:

the front size of the main bearing-caps has to be chosen in consideration of two imperatives:

1/ the journal diameter must be large enough to give the desired stiffness of the crankshaft
2/ the overall size of the corresponding cap must be small enough to enable the accessories to be fitted close to the crankshaft.

The dimensions finally selected for the crankshaft are shown in Figure 3.



	Distance between arms (MM)	Bushing width (MM)	Diameter (MM)	Bearing surface (CM²)	Calculated specific pressure (DAN/CM²)
CRANK-PIN BEARING	A= 40	B= 17,5	ØM=52,3	9,16	425
MAIN BEARING	C= 29,6	D= 23,4	ØT=70	16,35	141

Figure 3

A 108 mm. centerline distance between cylinders of the same bank corresponds thereto.

Aluminum cylinder block (AS9U3) - All the foregoing concerning the dimensioning of some essential components, which finally conditioned the choice of crankshaft geometry, remains valid whichever material selected for the cylinder block.

Based on our experience in die-cast cylinder blocks for 4-cylinder engines, we have deliberately built the cylinder block around the crankshaft, assuming that the reinforcement of weak points experienced in the first tests would make it possible to reach exactly the same final overall dimensions as with a cast-iron block.

The analysis of stresses on the weakest points allowed distributing them by adding or sometimes removing material; this, in the end, led to a rigid, durable cylinder block.

Separate wet cast-iron liners have been chosen for the following reasons:

- to enable use of "low silicon" aluminum alloy in the cylinder block (simplification of the die casting and reduced machining tool wear);
- to be able to use conventional pistons.

Here are some interesting specifications concerning the cylinder block:

- Weight without caps and liners 14.3 kg.
- Weight with cast-iron caps and liners 27 kg.
- Overall length 392.5 mm.
- Distance between crankshaft axis and cylinder head gasket joint face 221 mm.
- Distance between crankshaft and lower gasket joint face 60 mm.

After leaving the die-casting molds, cast parts receive successively an impregnation and a blasting for sealing respective appearance purposes.

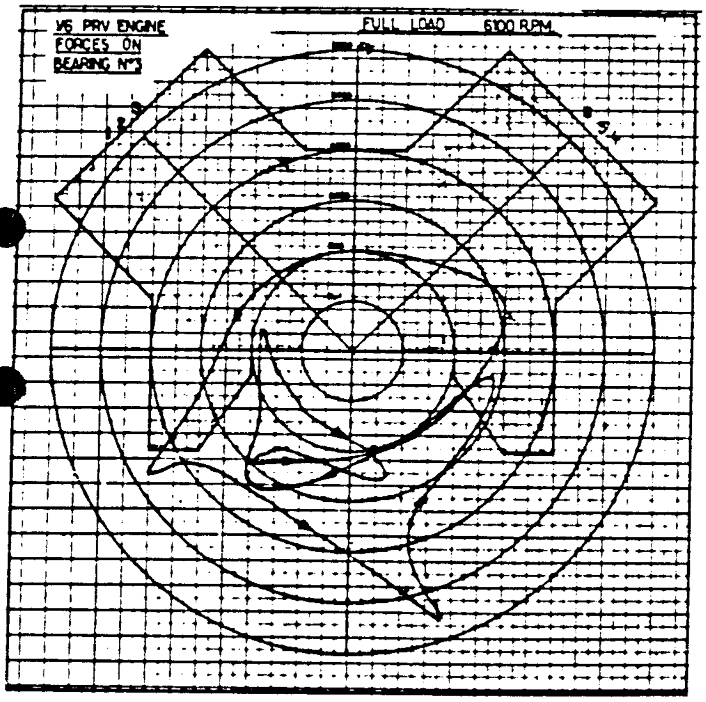


Figure 4

Attachment of bearing caps - The problem of attachment of the bearing caps is critical on a V-type engine. It is still more difficult with a die-cast light alloy cylinder block.

First of all, the main point is to determine the value and direction of forces applied to each bearing.

This is accurately calculated along the 720° cycle, by use of the gas pressure diagram with addition of the force from the reciprocating masses.

In addition, the position of counterweights that balance the 1st order rotating torque (see Balance) must be known:

By using six counterweights, the peak values of stress at the most heavily loaded bearings can be minimized.

Figure 4 shows a diagram of the instantaneous forces on bearing No. 3* at full load with an engine speed of 6100 RPM which represents the most severe bearing load.

It can be noted that the most severe condition occurs when we have a separating force of 2180 daN and a sliding force of 750 daN.

The size of these forces as well as the concern for reducing clearance between crankshaft and bearing (a reduction which was experimentally recognized as essential for controlling full-load noise) finally led to the arrangement shown in the cross-sectional view of Figures 5 and 6.

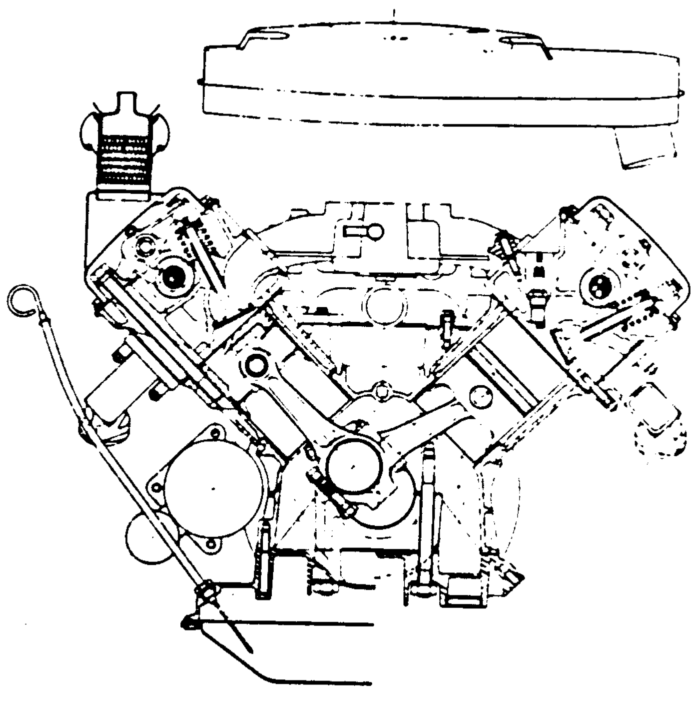


Figure 5

* bearing No. 1 being the outermost bearing on the clutch side.

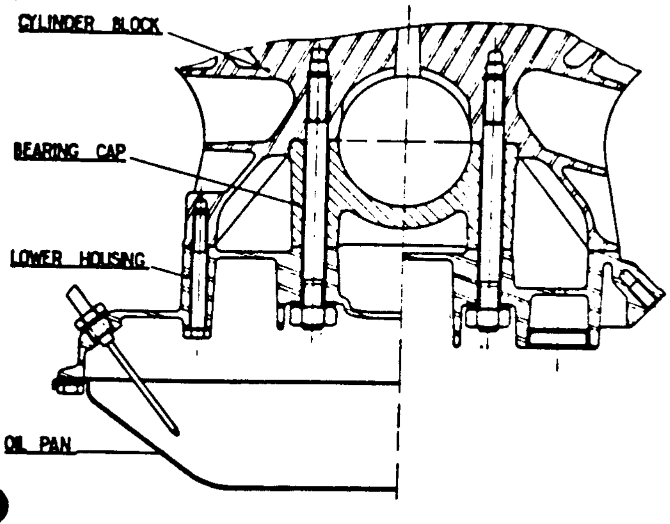


Figure 6

Main points to be noted are as follows:

The caps are made of cast-iron and separated; this requires that the bearing bore be finished with aluminum alloy (block side) and cast-iron (cap side), a process which has been used by the industry since 1965. This type of construction permits both better resistance to stresses for a given geometry and a possibility of reducing clearances. Note that this arrangement allows avoiding the matching between crankshaft, bearings and block equipped with its caps while still retaining the conventional machining accuracy in wide use for these components.

Each cap is positioned:

- laterally, by recessing its lateral faces between shoulders located on either side of conventional bearing zones;
- longitudinally, by one of the attaching studs thanks to adequate tolerances;
- vertically, each cap is enclosed and pressed by the main bearing bolts between cylinder block and lower housing (see Figure 6).

The advantage of this "enclosing" of the caps is to double the frictional force likely to resist the lateral sliding of the cap. It results in a considerable reduction of stresses in the areas where the bearing caps are tightened down and the strength of these areas in the cylinder block is quite satisfactory.

The lower housing consists of a pressure die cast aluminum piece with the surface against the cylinder block and the caps machined in one milling operation; it is bolted to the crankcase without the use of a gasket.

Balance - The overall balance imperfection already mentioned being a torque of the 2nd order frequency, it cannot be corrected unless by resorting to complicated, costly devices, which might be prejudicial to the overall mechanical efficiency.

However, it should be noted that the amplitude of this interfering torque is all the smaller as rod length is great (which is true for any 90° V-type engine; so as to avoid interference between moving components).

The 146.15 mm. center-to-center distance of a connecting rod corresponds to a connecting rod/crank ratio of 4.01 with a 73 mm. stroke.

On the other hand, as concerns the balance of the first order, it amounts to balancing a rotating torque by means of counterweights. As already mentioned, those counterweights have been determined in position, mass and geometry in order to balance the rotating torque and reduce stresses on bearings to a minimum.

As for a V8 engine, the overall balancing of the crankshaft must be done by attaching, manually or by using an appropriated cradled crankshaft holder, additional masses centered on each crank-pin and having a value of twice the rotating mass of a connecting rod (2 x 0.557 kg.) plus one time the mass in reciprocating motion for one cylinder (0.796 kg.) -- (complete piston + connecting rod small end).

This technique is very well known and the stress will be laid only on a few particular points.

There is only one crankshaft type, one connecting rod type and one piston weight so as to enable full interchangeability without matching requirements.

With reference to Figure 7, this is achieved by bringing connecting rods to the same weight on big (T) and small (P) ends through automatic milling of weights arranged at both these ends.

Likewise on pistons, the calibration of the weight is carried out by removing material from weights (M) located under the piston pin.

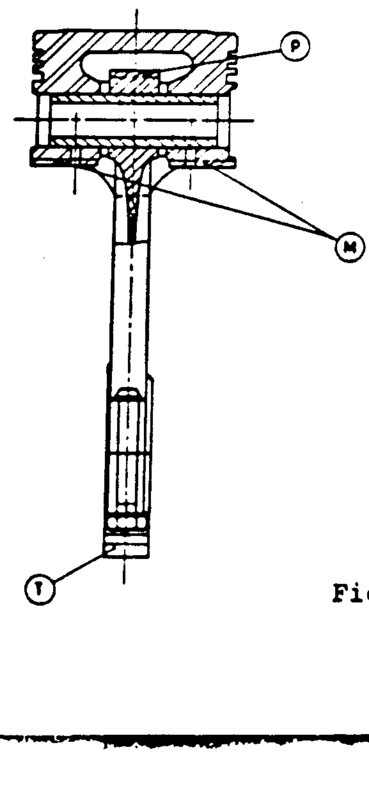


Figure 7

<u>Camshafts drive and location of oil pump</u> - In order to minimize total length, the arrangement selected includes a separate chain to drive each of the camshafts; it is the only solution which takes advantage of the relative offset between the two banks of cylinders and allows a reduction in length of 10 mm.

Each of the camshaft drive chains is kept under tension by a tensioner: on the slack side of the chain, there is a long steel blade padded with Nylatron*, hinged at a point in the vicinity of the crankshaft and pressed at the other end by an oil-pressure activated pushrod including a non-return device.

The oil pump is of the straight gear type and its housing is integrated in the left-hand front face of the cylinder block**. The pump is driven through a chain without tensioner at 18/28ths of the crankshaft speed, which provides a fair compromise between size, pressure build-up at low speed and safety against cavitation at high speed.

COMPONENTS SELECTED FOR THEIR FUNCTIONAL QUALITIES

<u>Overhead camshaft</u> - An overhead camshaft was chosen to both facilitate the design of the die-cast cylinder block and give the engine greater RPM range of operation. Generally speaking, it provides a fair compromise between emission control requirements and high speed operations thanks to the substantial valve ac-

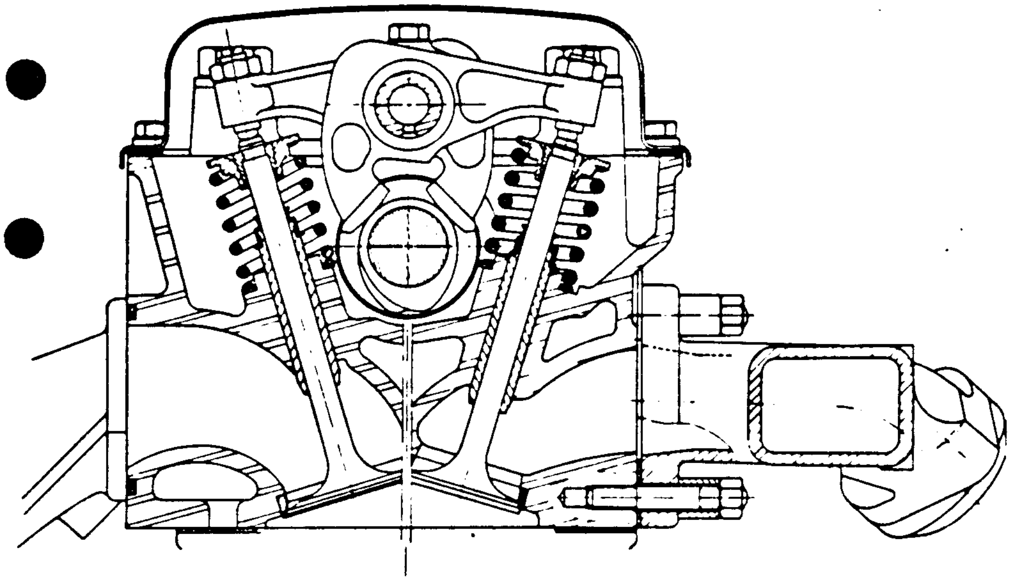


Figure 8

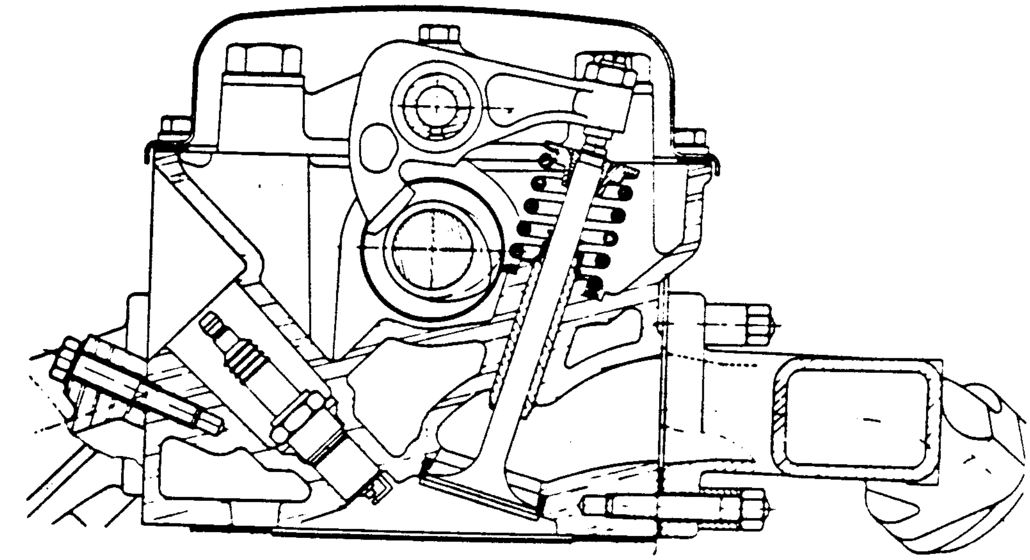


Figure 9

* 66-Nylon added with 30% Molybdenum disulfide
** The left and right sides mentioned in the descriptions are valid for the engine viewed from the clutch side.

celerations allowed.

As shown in Figures 8 and 9, this cylinder head is very compact, which is important on a V-type engine. Its height ensures a great rigidity. It must be pointed out that the non-machined combustion chamber shape gives greater freedom to optimum dimensioning of valves and the use of rocker-arms without offset. The left and right cylinder heads are designed so that they can be machined in the same transfer machine.

Lubrication - The oil pump speed and size have been chosen to obtain an in-use output over 14 liters per 1000 engine RPM as soon as 600 crankshaft RPM is reached.

The pressure regulation takes place at about 0.45 MPa (4.5 bars) by a piston discharge valve with return to the suction side of the pump.

The full-flow oil filter is a throw-away type cartridge consisting of a paper filtering element.

There is a by-pass relief valve in the engine block oil circuit to avoid deterioration of the filtering element when oil viscosity increases (cold weather).

The cartridge installed in production at Douvrin* provides a filtering threshold of 5 to 8 x 10-3 mm. to efficiently protect the engine during the break-in period, while those supplied as spare parts have a threshold of 10 to 15 x - 10-3 mm. Regarding lubrication of joints, a special feature of this engine is the location of the oil passage holes in the crankshaft, in order to continuously lubricate the connecting rod joint; this arrangement derives from the absence of oil grooves in the bearing cap inserts.

Elimination of this groove enables reinforcement of the oil film at each instance when gas pressure becomes greater than the inertia forces of the piston/connecting rod assembly.

This reinforcement of the oil film contributes to the elimination of noise under load.

Cooling - The single outlet water pump has a stationary seal; it is belt-driven at 1.10 times the speed of the crankshaft. Its pressure-output charateristics, measured with a water temperature of 185° F. (85° C.) for various engine speeds, are shown in Figure 10 with approximate indications of the pressure losses due to the flow circuits. The output of the single-outlet pump is divided and feeds the two cylinder banks through a double manifold directly attached by bolts to the cylinder block.

Coolant flow regulation is obtained with a three-way type thermostat operated by an expanding wax temperature sensor. The coolant flow to each of the 6 cylinders is controlled by calibrated water passage holes in the two cylinder head gaskets.

The center-to-center distance of the cylinder liners of each bank is 108 mm.; the thickness of the water room is 5.8 mm. at the upper flange and 9.5 mm. along the liners.

Ignition -

Conventional ignition - The top view of the engine (Figure 11) shown without the air filter showns the location of the ignition distributor.

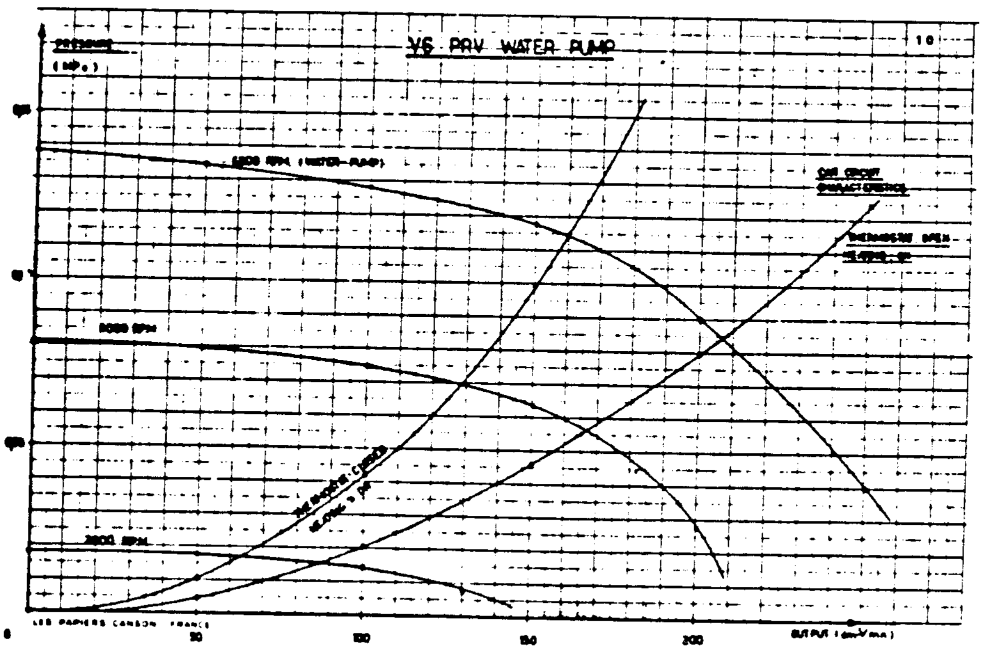


Figure 10

It is driven by the right hand side camshaft, and its rigid mounting on the cylinder head insures a particularly low vibration level.

* Franco-Swedish Engine Company (Pas-de-Calais, France)

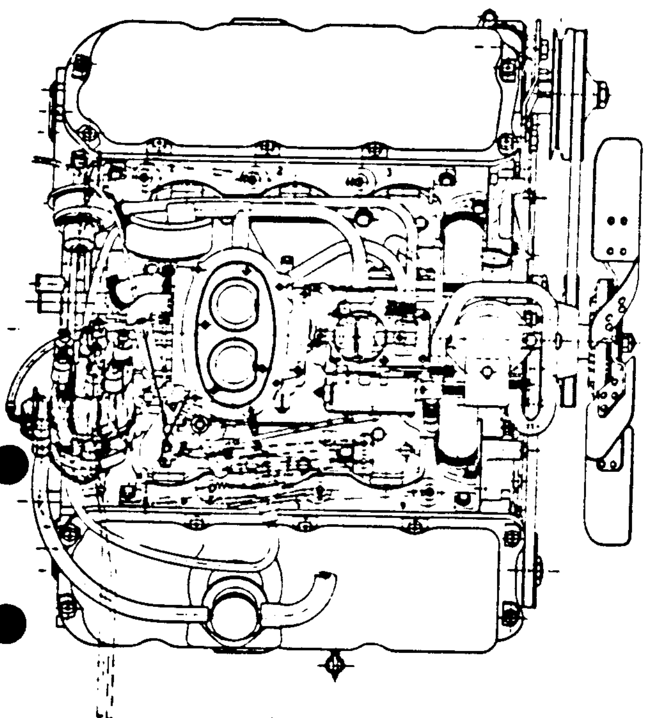


Figure 11

In the conventional ignition system, there are two breakers and a double high voltage rotor, in order to work with a separate coil for each cylinder bank. In order to adjust the correct timing for the two banks, the following procedure is used:

> Timing is adjusted with a stroboscopic light at the timing mark of the left bank only by rotation of the distributor body. Timing of the right bank can then be set by adjusting the corresponding breaker from the outside of the distributor. The dwell angle check on the right bank breaker shows if the components are correct and compatible.

For some applications, two alternative ignition systems have been developed:

- A breakerless transistorized ignition system which allows both reducing maintenance costs and obtaining a stable, durable spark timing as required by emission regulations, with a minimum risk of misfiring due to ignition system failure.

  The ignition curve is, however, conventionally controlled by a mechanical centrifugal advance, and a retard/advance vacuum control.
- A fully electronic ignition system such that this spark is triggered by a sensor indicating the position of the left camshaft. There are no longer mechanical parts subject to wear, therefore timing does not deviate from its orignal setting. Advance in relation to RPM is obtained electronically by signals from the sensor; correction in relation to load is obtained through a diaphragm acutating a potentiometer.

SPECIAL FEATURES OF FUEL SUPPLY SYSTEMS

Currently, the P.R.V. V6 engine is available either with a carburetor fuel supply system or a fuel injection system.

Each of the main versions of both these types of fuel supply systems will now be described.

CARBURETOR FUEL SUPPLY SYSTEM

In the process of development, it soon became obvious that, as only manifold water preheating was used, it was necessary:

a) to use a double barrel carburetor with simultaneous opening, whose throttle plate axis are parallel with the crankshaft;
b) to position the carburetor right in the middle of the V at a well determined point in relation to engine length.

The fuel mixture distribution results obtained with this single double barrel carburetor were already relatively satisfactory and we noted that two options were available.

The first option consisted in separating almost completely the manifold ducts between the two cylinder banks so as to have each of them supplied by one barrel. A connection hole 20 mm. in diameter would allow correct balancing at idle and very low loads, whereby a rather high specific torque* would be achieved (about 8 m.daN/ - liter).

The second option consisted in connecting the six ducts of the manifold under the carburetor.

In this way, the specific torque was slightly lower (7.75 m.daN/liter) but the specific power was increased.

Introduction of an additional small nozzle carburetor upstream of the distribution center further improved the air/fuel preparation at low loads.

This arrangement enables good distribution at all load conditions and a slight increase in maximum power output.

Figure 12 shows the manifold installed in production. It illustrates the principle of this fuel supply system.

The corollary of this principle was to develop a "compound" fuel supply system. The double barrel carburetor is automatically opened by use of a diaphragm.

The development has been performed in close cooperation with the Solex Carburetor Company.

The following documents represent results

* torque related to engine displacement.

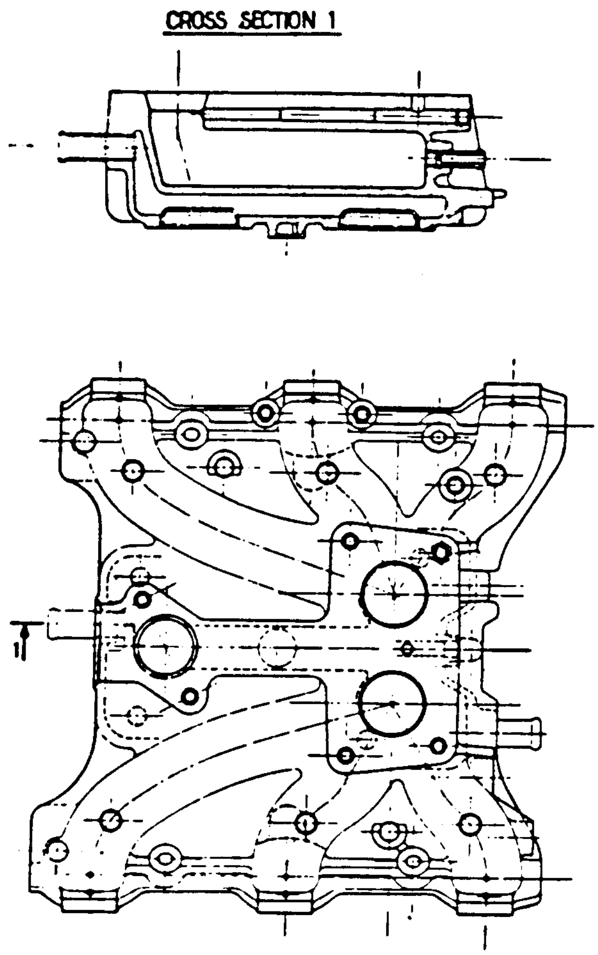


Figure 12

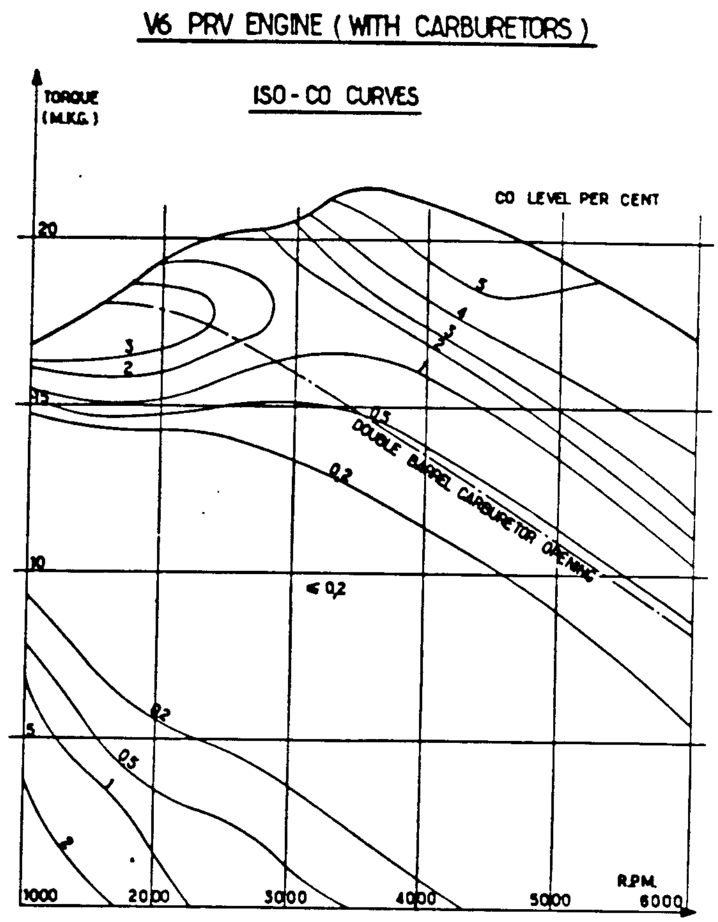


Figure 14

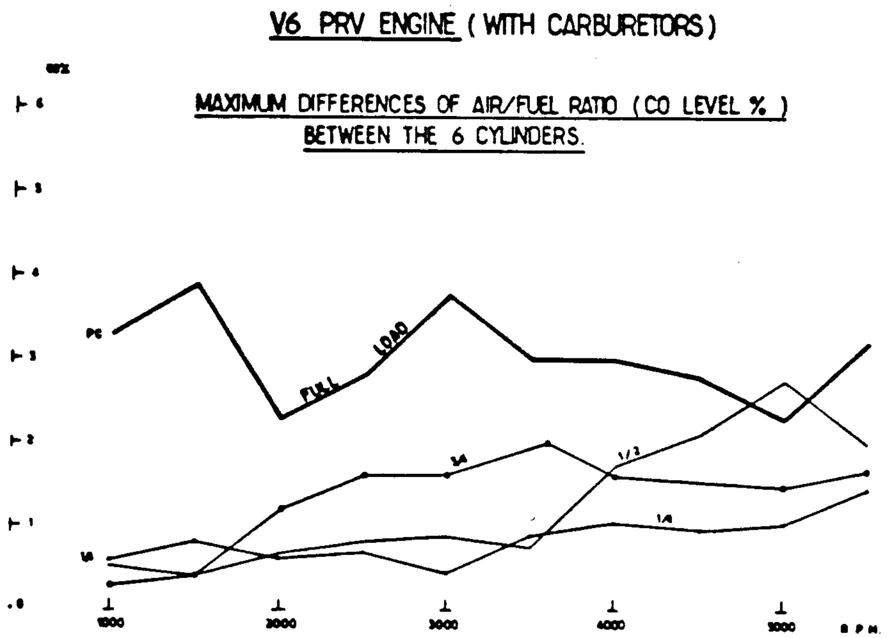


Figure 13

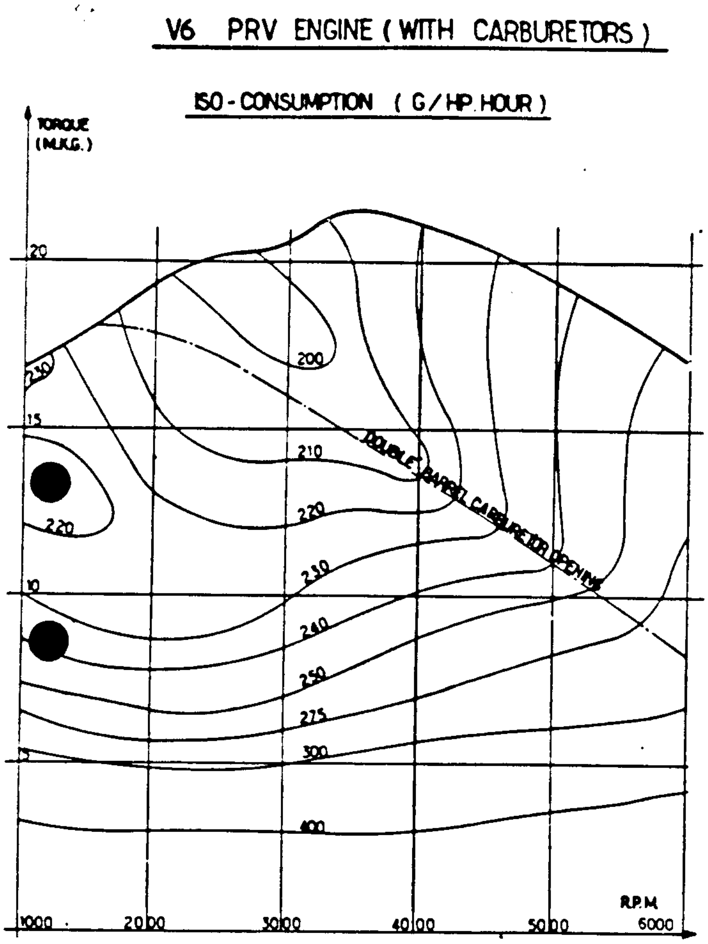


Figure 15

from this development:

- Chart showing maximum differences of air/fuel ratio (CO level %, which is a valid expression because of the high enough medium CO level) at ¼, ½, 3/4 and full load. (Figure 13)
- Chart showing actual CO levels obtained with the production carburetor described above, along the complete operation range of torque/RPM (so-called ISO-CO curves). (Figure 14)
- Chart showing ISO consumption curves in g/hp hour on the range of torque/RPM. (Figure 15)

FUEL INJECTION VERSION

Choice of system - The adoption of the continuous fuel injection system (Bosch K-Jetronic) was decided upon on the basis of the following major advantages:

1. Direct load sensing by the air metering device which continuously measures air flow through the engine.
2. Simple injector design since the fuel is continuously injected.
3. Fewer and simpler service requirements. Most components within the system can be checked by a fuel pressure gauge.
4. Partial altitude compensation by the air metering device.
5. Less influence on actual air/fuel ratio by the EGR rate since actual air flow is measured.

Fuel injection system description - The fuel injection system was invented by the Bosch Company and is now in production; its description was already widely diffused; however, we describe it in Appendix II.

Air intake system of the injection version engine - One of the major design constraints in adapting a fuel injection system to a new engine is the requirement for good cylinder-to-cylinder air distribution. The air distribution is more crucial in any feed system where the air and fuel supplies are separated, as a variation in cylinder-to-cylinder air distribution solely results in a variation in cylinder-to-cylinder richness. When the fuel and air are well mixed in advance, a variation in flow results mainly in a variation in respective charge to each cylinder.

Two different intake manifold systems have been designed.

For markets where emission requirements are moderate, top priority has been given to simplicity and serviceability. The major components of this air intake system is shown in Figure 16. The intake manifold resembles a six-legged crab which straddles the air/fuel metering unit, enabling the latter to be placed low in the V between the cylinder banks. Intake air passes upward through the metering unit and through the single throttle into the plenum chamber, which forms the body of the "crab". The air then travels downward through the six "legs" to each cylinder.

In order to permit a lean overall mixture without having one or more cylinders running so lean that the lean firing limit is reached, a dual manifold has been designed for the US fuel injection version with the objective of optimizing air distributrion (Figure 17).

The total amount of air is measured, as in the other injection version, in the air metering device, and the air flow is then divided into two and passes a dual throat throttle, each throttle feeding one of the manifolds. Each manifold supplies one of the cylinder banks and is located at the opposite side of the V, thus facilitating comparatively long intake ducts.

EMISSION CONTROL SYSTEM FOR THE VARIOUS U.S. MODELS

The achievement of good mixture distribution among cylinders as well as a good overall richness, whatever the engine operating conditions may be, has made the achievement of good emission control results much easier though,

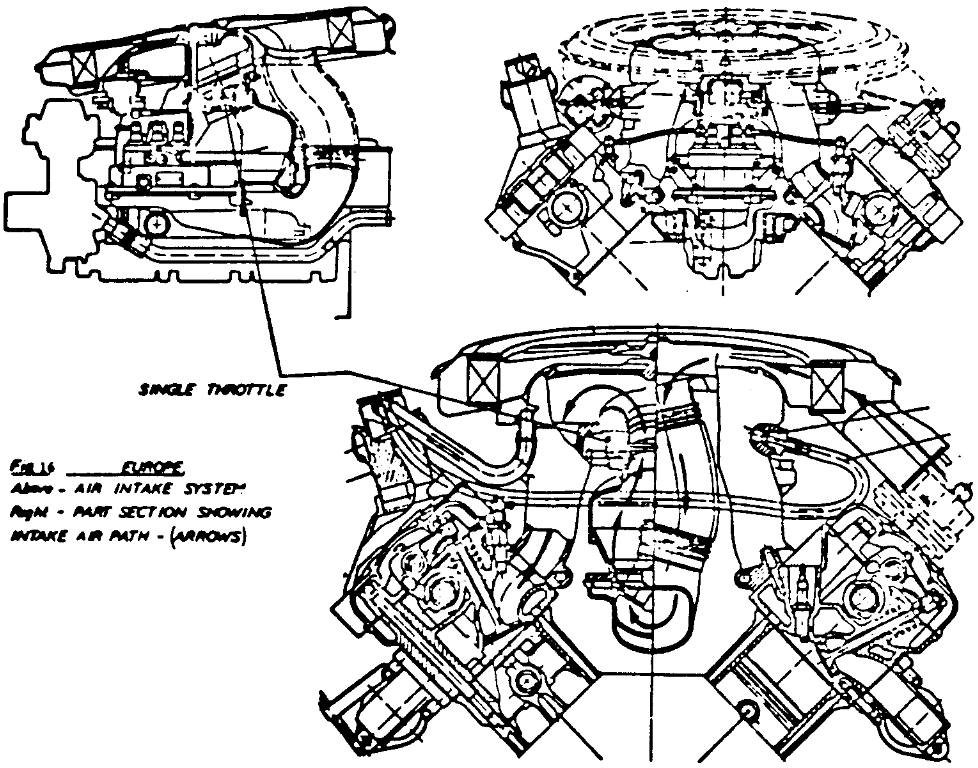


Figure 16

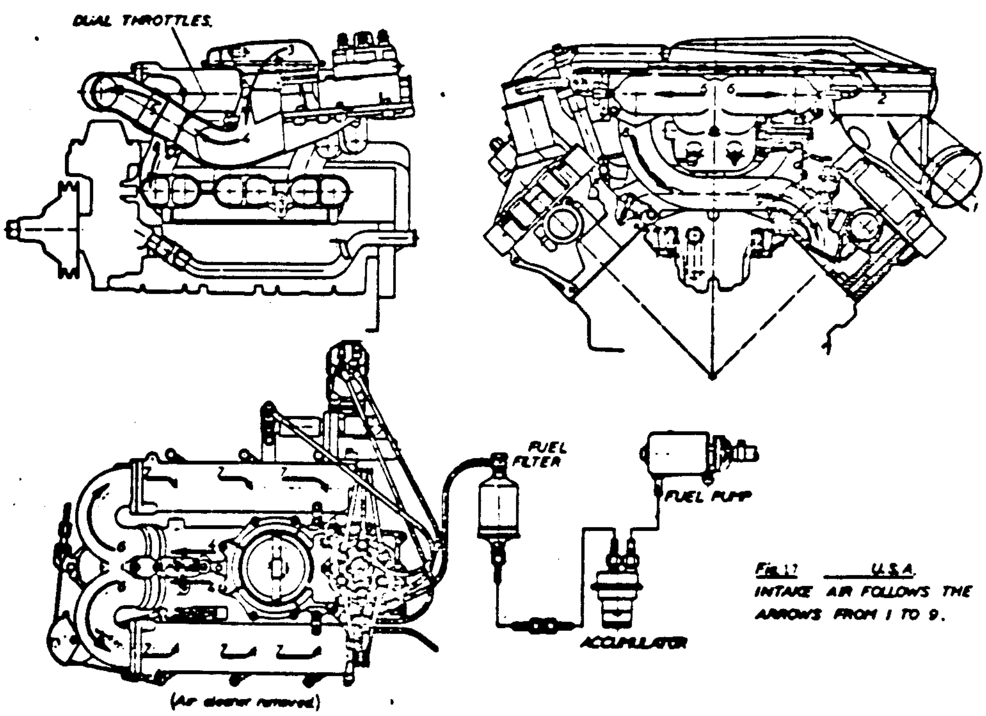


Figure 17

with a view to complying with the very low US emission levels, it has been necessary to equip the engine with several extra emission control systems, whose main features are mentioned below:

IGNITION SYSTEM

US models are equipped either with the breakerless transistorized ignition system or with the fully electronic ignition system described in Ignition (less risk of misfiring, durability of catalytic converters).

ANTI-EVAPORATION SYSTEM

The anti-evaporation system consists of a canister and a fuel tank pressurizing system.

Some models are equipped with a purge valve, which both prevents carbon canister purging at idle speed and, henceforth, improves operation under such conditions.

EXHAUST GAS RECIRCULATION

The EGR systems used are of two types: "on-off" or proportional (in the latter case, the vacuum in a venturi located in the air intake system or carburetor is used for controlling recirculation valve opening).

Extra devices can be used so as to prevent any recirculation during operation periods: at idling speed, with a cold engine, or at full load.

SECONDARY AIR INJECTION

The air injection device is of a conventional type comprising a belt-driven vane air pump, a diverter valve (which acts as a relief valve at high engine speeds and removes air injection during decelerations) and a check value (in order to protect the pump against possible exhaust back flow or backfiring).

CATALYTIC CONVERTER

The catalytic converters used are oxidation catalysts .

The active catalytic substance is comprised of a platinum/paladium mixture deposited on al monolith ceramic substrate.

CONCULSION

This paper has explained the reasons for development and the particular characteristics of the P.R.V. V6 engine; the main objectives in the design process have been: minimized weight, and minimized overall dimensions.

Thanks to its ease of application derived from its two overhead camshafts, and the sturdiness of its moving parts, Peugeot, Renault and Volvo will be able to technically meet emission control regulations and to achieve the performances required by commercial policies. These three companies will be able to accomplish this at a reasonable cost because of the advanced features and production methods of this engine.

NOTE: Basic specifications are found in Appendix I.

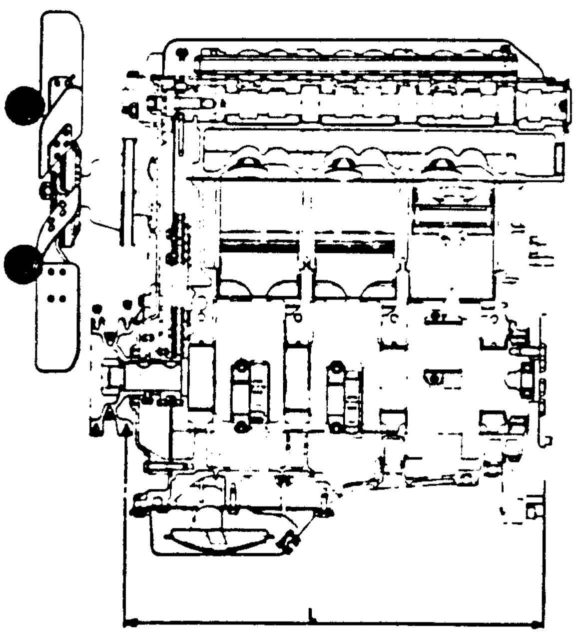


Figure 18

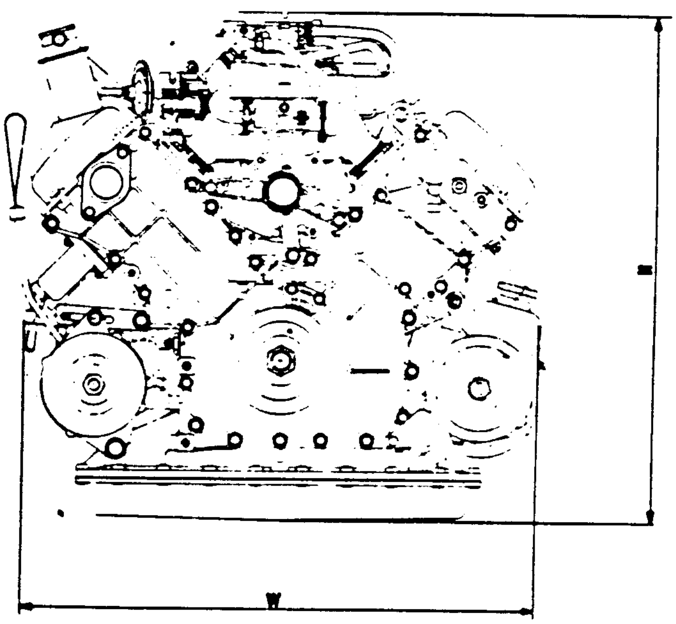


Figure 19

APPENDIX I

Basic Engine Specifications

Cylinders	90° V6	
Bore and Stroke	88 x 73 mm.	
Displacement	2664 cm3	
Cylinder Bore Spacing	108 mm.	
Compression Ratio	8.65/1 - Europe 8.2/1 - USA	
Camshaft	One overhead camshaft per bank (One separate chain drives each of the camshafts)	
Crankcase	Die-cast aluminum	
Crankshaft	Graphite spheroidal cast-iron	
Main Bearing Diameter	70 mm.	
Crankpin Bearing Diameter	52.3 mm.	
Connecting Rods	Forged	
Lubrication	Geared pump	
Capacity	6 dm3	
Oil pump output @ RPM	14 dm3 @ 1000 RPM	
Filter	Full flow	
Cooling System	Liquid	
Valve Timing	Left Bank	Right Bank
Intake opens BTDC	9°	7°
Intake closes ABDC	45°	43°
Exhaust opens BBDC	45°	43°
Exhaust closes ATDC	9°	7°
Ignition		
Firing order	1 - 6 - 3 - 5 - 2 - 4 1 2 3 4 . 5 6	
Length (L)	466,5 mm. (Distance between first belt working plane and clutch disc friction areas - See Figure 18)	
Height (H)	623 mm. (Distance between oil pan and upper air intake areas of carburetors - See Figure 19)	
Width (W)	632 mm. (Total width measured at level of the exhaust manifolds - See Figure 19)	
Weight	156 kg. (Total weight of carburetor European version - with air filter, flywheel, water pump, alternator, fluid drive type cooling fan, starter motor and oil, but without engine mounts and brackets)	

Performance Data of Existing Versions:

Gross horsepower (DIN)	125-140 HP
Gross torque (DIN)	20-21.5 daN.

'C.I.' CONTINUOUS INJECTION FUEL SYSTEM

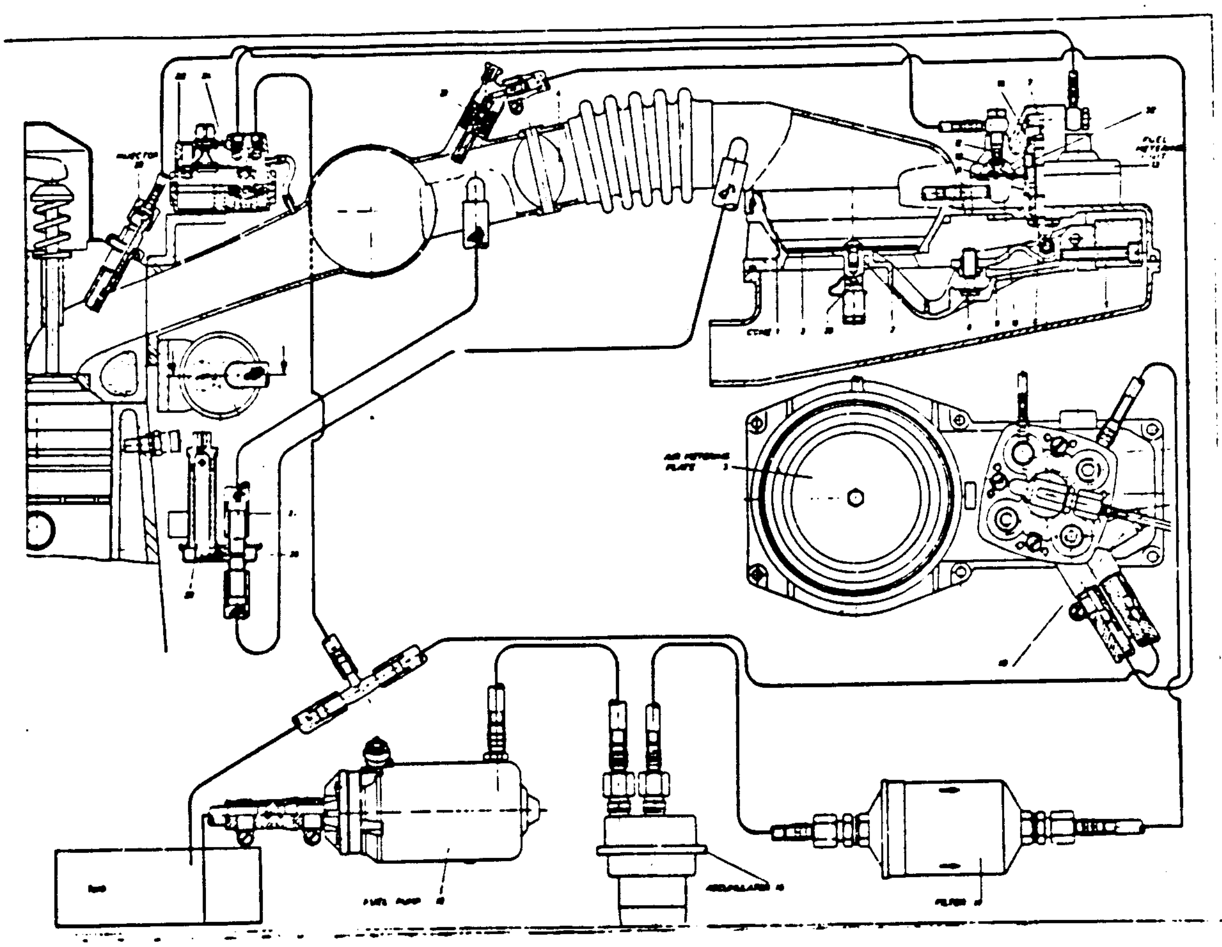


Figure 20

APPENDIX II

Fuel Injection System Description

Referring to the numbers in the diagram (Figure 20), the operating principle of the fuel injection system is as follows:

An electrically driven roller type fuel pump (15) pumps fuel at a pressure of about 5 daN/cm2 via a pressure accumulator (16) and fuel filter (17) to the fuel metering device. The objective of the pressure accumulator is to maintain fuel pressure after the engine is stopped, in order to prevent vapor lock, which otherwise could cause hot-start problems.

The air metering device consists of an air metering plate in the form of a disc (3) attached to a pivoted arm (2) which moves in the conical tube (1). (Also see Figure 21 - bottom.) The movement of the air is proportional to the air flow into the engine which is conventionally controlled by the driver via a throttle plate (4). As the air flow depresses the arm (2), the fuel control piston (6) of the fuel metering device allows fuel to flow equally to the six injectors (20) via the six rectangular fuel metering slots (32). The pivoted arm (2) is counter-balanced by a weight and also by means of hydraulic counterbalancing force on top of the fuel metering piston (6).

The ratio between the fuel and air flow, and thus air/fuel ratio, is controlled by the profile of the conical tube (1) of the air metering device.

Pressure regulator (19) maintains the fuel feed system at a pressure of 4.5 daN/cm2 and a diaphragm valve (11) controls the pressure drop over the metering slots at a constant drop of 0.1 daN/cm2.

Under cold starting conditions, a bi-metallic strip (24) within the control pressure regulator (22) reduces the control pressure working on the top side of the fuel metering piston (6), thus providing a diminished force opposing the movement of the air metering plate. A specific air flow results in a larger opening of the fuel metering slots and a richer mixture. During warm up the bi-metallic strip is heated both by engine and electrical heat and raises the counterbalancing control pressure progressively to weaken the air/fuel mixture

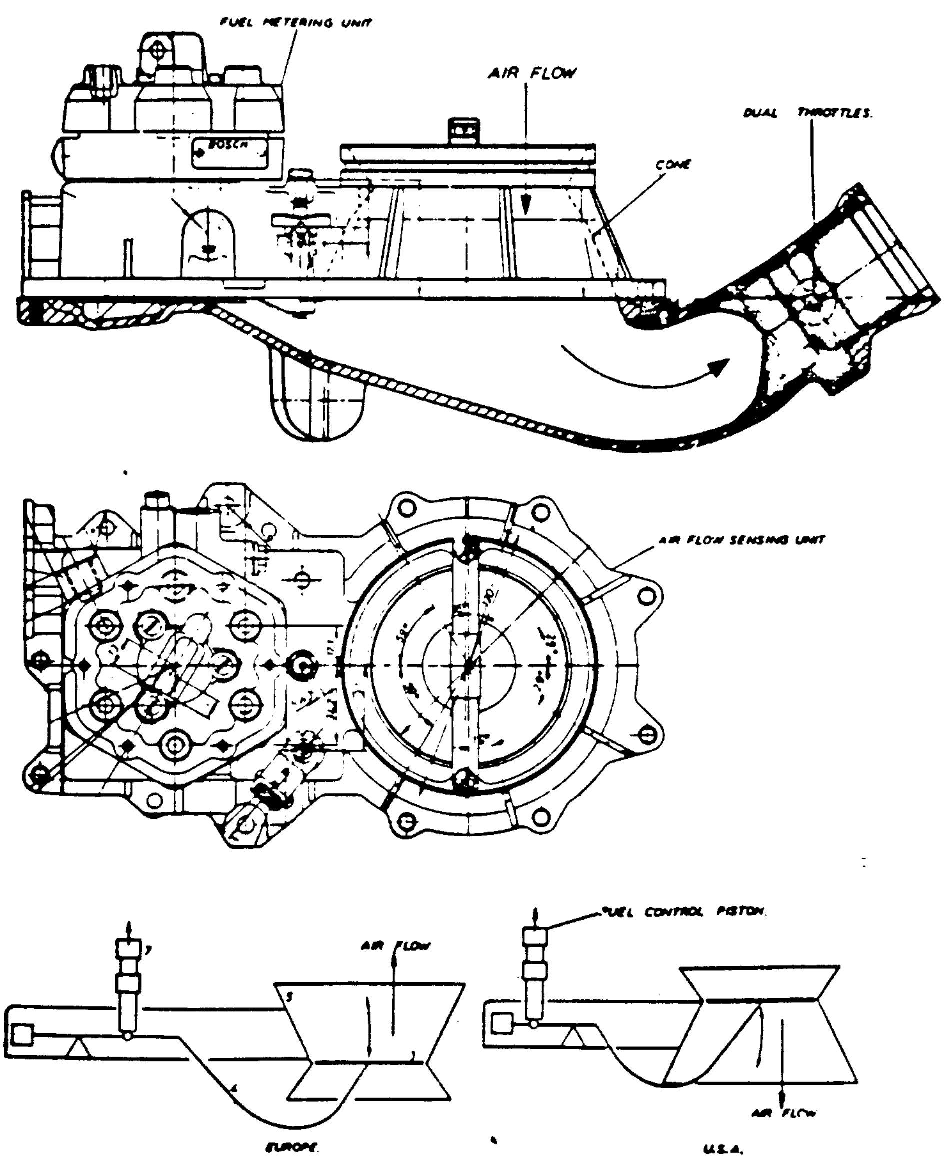
For extra enrichment during "key on" condition, an additional injector (31) is provided behind the throttle plates. This injector has a maximum injection time limit to prevent flooding.

An extra fast idle air supply is provided via a throttle by-pass circuit controlled by a disc valve (27). An electrically heated bi-metallic strip (29) opens a rotating air valve (29) at low temperature, which is closed again by a combination of electrical and engine-produced heat as the motor warms up.

Full throttle enrichment is also provided by the control pressure regulator (22), which senses the low manifold depression, and reduces the control pressure to enrich the fuel mixture (as during choke operation).

On switching off, the fuel supply is positively cut off to prevent engine run-on.

Fig.21 U.S. ENGINE AIR/FUEL METERING UNIT



PRINCIPLE OF AIR METERING UNIT

Figure 21