Where the Rubber Meets the Road ……. *Part I* The XPAG Engine

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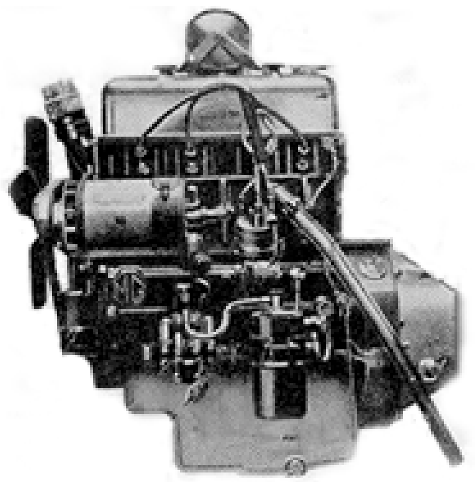
Understanding the XPAG Engines inherent design characteristics, and what effects they have on making modifications to increase Power and or Economy.



XPAG TC Engine

The objective of this article is to look into the past and gain some knowledge of William Richard Morris and the roots of the XPAG engine. How the British government influenced the engines design. And to delve into the design, and gain an understanding as to why things work. Learn how each major component effects the others. Take a systematic approach to modifying components to work better together, as a system. Areas that we will cover include:

- A brief history of William Richard Morris and his car companies
- How the British government influenced the design of all British engines
- Breaking down the engine into Subassemblies and analyzing the design characteristics
- Forced Induction

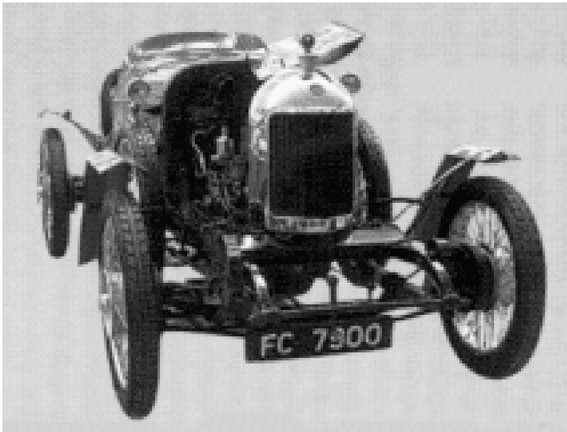
A brief history of William Richard Morris and his car companies

William Richard Morris built his first motor bicycle in1901. For the next nine years he was involved with the various partnerships and business, dealing in cycles and automobiles. In 1911 he realized that the market for automobiles would grow in the same manor in which bicycles had. In 1912 The Morris Oxford prototype was built, and in 1913 the company was named, *The Morris Garages (W.R. Morris Proprietor).* The company that we will now refer to as M.G. In 1913 the first production Morris was released. It was called the Morris Oxford, also referred to as The Bullnose.

Morris had an unusual early history as a manufacture. The engine that he used, and many of the parts were purchased from an American Company known as the Continental Motor Company. The design of this engine went on to influence in one way or another, other engines produced by M.G.

In 1921 Cecil Kimber joined the MG Car Company and worked as Sales Manager. In 1922 he was appointed General Manager of Morris Garages.

Kimber was the backbone of the company. From the beginning he adopted a sporty appearance to the cars. By 1924, Morris Garages was advertising the "MG Special four-seater Sports", and had incorporated the famous octagonal badge into the Bullnose radiator shell.



Old Number One

How the British government influenced the design of all British engines

In 1921 The British government formed the Ministry of Transportation. During that same year they imposed a new tax, the "R.A.C. hp rating", also known as the Treasury Rating. This tax was in effect a tariff against American cars. *(According to R.I. Barraclough and P.L. Jennings in the new book "Oxford to Abingdon")* The customer was charged £1 for each R.A.C. hp.

I think that words of the late Cecil Cousins, that were presented at a special meeting with the New England MG 'T' Register in 1975 says it best; "…. You see, in those days we didn't sell engines by cubic capacity. We built engines on a cranky thing called RAC rating which didn't take in to consideration the cubic capacity of the engine. It was based, pure and simple, on the bore of the engine; hence, the fact that for years and donkey years, all the English motor trade suffered with great engines with tiny little bores and whacking great long stroke."

Breaking down the engine into Subassemblies and analyzing the design characteristics

The roots of the XPAG date back to 1939 and the MG "TB" Midget. For all practical purposes, the engines are virtually the same. They are both OHV, pushrod operated. They share the same capacity, 1250cc, Bore, Stroke, Connecting rod length, compression ratio, camshaft timing, brake horse power rating, and many other similarities. Although changes were made between 1939 and 1955. The inherent design characteristics of the engines were the same.


Late Model XPAG engine

The XPAG responds fairly well to supertuning. In modern day engines, it is easy to obtain between 1 to 1.5 horsepower per cubic inch, in a normally aspirated configuration. The 1250cc engine displaces 76.28 cubic inches (1250 x 0.0610239), and has a break horsepower rating of 57hp at 5500 rpm.

At 76 cubic inches, With the addition of an extractor exhaust manifold, free flowing exhaust system, and modified to meet the stage II configuration, as defined in the; "XPAG Engine Data Service Supertuning" by; W.K.F. Wood, Edited 11/98 by our own Jerry Austin. You could approach the 76hp rating. (If you do not have a copy, I suggest that you contact Jerry ASAP).

At first glance the XPAG engine seems like a rather simple device. A mixture of fuel and air is drawn into a closed chamber where it is ignited and burned to produce energy. In its simplest form the XPAG engine is nothing more than a pump. However, as we spend more time analyzing the engine, it becomes obvious that it is quite complex and offers and endless number of possibilities to increase power and efficiency.

There are three power variables that control horsepower; engine displacement (expressed as Cubic Centimeters, Liters and Cubic Inches), rpm and brake mean effective pressure or BMEP.

First, if an engine that is bored to a larger capacity, it breathes in greater amounts of air/fuel mixture consequently it produces more power. Second revving the engine faster through higher rpm allows an engine to perform its power producing cycle more frequently, which produces more power. The last variable, BMEP, is more complex than the first two because it involves the following: the amount of air/fuel mixture filling the cylinder; the amount of mixture that is burned; and the amount of power that is lost to internal friction and heat. In simple terms BMEP means average effective combustion pressure.

These three power variables lead to five methods for increasing power. The first method is to increase the displacement with larger bore. Stroke length also effect's displacement but it is one of our fixed variables. Second is to rev the engine faster to take advantage of more power strokes in a given amount of time. Third is to fill the cylinders with more air/fuel mixture. The forth method is to enhance combustion efficiency by burning the greatest possible amount of air/fuel mixture. The last method is to minimize internal friction losses by using the correct parts and proper assembly techniques.

In order to ascertain specific information, I will be using data from the following sources for the analysis:

- Interviews with David Anton of Advanced Performance Technology.
- Internal Combustion Engine Simulation Software developed by Curtis Leaverton and published by Dynomation, Inc.
- Desk Top Dyno's by; Larry Atherton and Curtis Leverton.
- Auto Fundamentals by Stockel, Stockel, and Johnson.
- Rapid Line, Inc.
- The MG Workshop Manual From "M" Type to "T.F. 1500" by W.E. Blower".

In this section we will take a "bottom- up" approach to the dissection of the engine. Although most of the horsepower is made above the block The XPAG has some unique characteristics relating to its bore and stroke and lost horsepower that may provide a better starting point. The assemblies we will study include the following:

- Block, Crank Rods and Pistons
- Camshafts, Cam Timing, and Valve Train
- Cylinder Head, Compression ratio, Intake Manifold, and Carburetors
- Ignition System
- Cooling System
- Exhaust System

Block, Crank, Rods and Pistons

The XPAG Block displaces 1250 cubic centimeters and has a stock bore diameter of 2.6181 inches. The block can be safely bored to 0.060 oversize giving a capacity of 1309 cc's. It is also possible to re-sleeve the block to 1466 cc's. The XPEG uses Siamese cylinders with no water jacket between the cylinders. The XPEG can be bored to 0.040 over size to increase the capacity to 1506 cc's.

The Crankshaft has a stroke length of 3.5433 inches. The crankshaft is one of the weak links in the chain of engine components. It has a propensity to crack and even break at the web between the front main and the first throw.

Why? I had suspected that this problem occurred because of crankshaft deflection. Each time the air/fuel mixture inside a cylinder is ignited, the combustion that results creates a torque spike (an extremely rapid rise in cylinder pressure). This pressure, applied to the top of the piston, becomes the force that is applied to the crankshaft through the connecting rod. Each torque spike is like a hammer blow. In fact, it hits with sufficient intensity that it not only causes the crankshaft to turn, it actually deflects or twists it. This twisting action, and the resulting rebound (as the crank arm snaps back in the opposite direction) is know as torsional vibration.

The crankshaft will always encounter a characteristic known as the natural frequency of vibration at the same rpm. To get a good picture of natural frequency, imagine a bell. When struck, it makes a particular sound because its mass vibrates at a particular frequency; if material is added or removed the bell's sound is altered because the change in mass causes it to vibrate at a different frequency.

A crankshaft isn't much different. Strike one with a hammer, and you'll hear it ring. Add or remove material, and it will make a different sound because it's ringing at a different frequency. Inside an engine the torque spikes transfer through the connecting rods, like the hammer striking the bell, causing the crankshaft to vibrate. At a particular rpm the frequency of the torque spike comes into phase with the natural frequency of the crankshaft, thereby creating a harmonic torsional vibration. (A harmonic is a vibration that occurs at a half or whole multiple of the original frequency).

Harmonic vibrations are especially destructive because they "excite" the natural frequency so each torque spike causes the crankshaft to vibrate with ever increasing severity and for a longer period of time. While discussing this mater with Pete Thelander, he told me that he had, in the past, seen a graph of the XPAG crankshaft that showed a peak at the 5th harmonic torque spike at 4700 rpm. His recommendation is to move quickly through this range as the engine is revved up or down. This is definitely not the rpm to cruse at.

How can we fix this problem? The XPAG crankshaft does not use a harmonic damper. For all practical purposes, it probably doesn't need one. I have been corresponding with Vibratech, located in Alden, New York, to see if a current production harmonic damper could be modified to fit the XPAG. Vibratech also provided me with the information used to write this section of the article.

If you are rebuilding your XPAG the crankshaft should be Magnafluxed to locate any cracks. If the crank passes this inspection, then a process known as ion nitriding can strengthen it.

If your crank is cracked, the going price from Moss Motors, for a forged crank in 4340 Chromemoly steel goes for a cool $1,795!

The crankshaft stroke of 3.5433 inches, and the connecting rod length of 7.0078 inches, center to center, forms the foundation of the XPAG engine design. These two design elements effect every component in the engine.

But what does this really mean? There are two equations that are often used to compare and describe the engine characteristics. The first is the Bore-to-Stroke Ratio. The second is the Connecting Rod-Stroke Ratio, also referred to as the "Rod Ratio".

An engine is sometimes referred to as being under-square, square or over-square. To determine the engines Bore-to-Stroke Ratio, the bore is divided by the stroke. The XPAG bore to stroke ratio is 0.739:1. I have also seen the equation expressed as stroke divided by the bore, which gives a ratio of 1:1.353. I will use this method to describe the Connecting Rod-Stroke Ratio since it is commonly used.

An engine whose bore and stroke are the same would be considered "Square" For example, An MGB 1800 bored .040 inch over, would have a bore of 3.2 inches. The stroke is 3.5 inches. The Bore-to-Stroke Ratio of 1:1.09 is very close to being "square". On the other hand the XPAG has a Bore-to-Stroke Ratio of 1:1.35. This engine would be considered under-square, and very much a long stroke unit. A British Leyland A Series engine for the Mini Cooper 1071S has a Bore-to-Stroke Ratio of 1:0.967 is considered over-square.

Under-square engines generally run at lower rpm and generate maximum torque at lower rpm than over-square engines. On the on the other hand, over-square engines have a larger bore than stroke and typically generate maximum torque at a higher rpm level.

But wait, we have a connecting rod that is 7 inches long! The Connecting Rod-Stroke Ratio varies between 1.5:1 and 2.0:1. The rod ratio, is calculated by dividing the center-to-center length of the rod, 7.0078 by the stroke, 3.5433. The rod ratio for the XPAG is 1.98:1.

A greater rod ratio decreases rod angularity, which decreases the lateral 'G', loads on the piston, gudgeon pin, rings, crank and the rod itself. This results in decreased wear on the pistons, rings and cylinder bore. Less angularity also accelerates the piston from TDC at a slower velocity. Moving the piston away from top dead center at lower velocity can minimize potential ring flutter while the piston is traveling through the critical upper half of the cylinder bore. The lower velocity also affects the combustion process because the piston dwells near TDC longer. When the piston dwells longer at TDC, combustion gases have more time to act upon the piston before the piston begins to lower in the bore. Cylinder pressure also increases because a smaller combustion volume is maintained for a longer period. And since most combustion takes place during the first 90 to 100 degrease after TDC, increased piston dwell near TDC maximizes pressure when the rod has the greatest mechanical advantage.

With a large rod ratio peak airflow demand occurs later after TDC, so the intake valves opening may be delayed and overlap reduced. A later opening intake increases valve-to-piston clearance, and reduced overlap improves low speed performance. In general, and engine with a long rod ratio needs a less radical and aggressive cam. Additionally, maximum intake airflow demand is lower, so a large rod ratio works better with the XPAG's restricted intake and exhaust systems.

Although a large rod ratio increases piston dwell at TDC, it does the opposite at BDC. At BDC the piston approaches and departs more quickly, reducing dwell time. This tends to require a slightly earlier closing intake valve and slightly earlier opening exhaust valve. This is why the XPAG tends to be more sensitive exhaust system scavenging. The power that is lost to friction and heat can be staggering.

According to David Vizard " Only about 18 to 25% of the engines power actually reaches the flywheel." For example; for every 100hp's worth of fuel burned in the cylinders, a GOOD engine will deliver about 25hp at the flywheel.

What about horsepower lost to friction? When I am cruising my MG down highway the engine runs between 3500 and 4000 rpm. (This is with a 4.30:1 gear). What is the average piston speed of the engine at 4,000 rpm? The following formula can be used to determine the average piston speed.

$$Average\ Piston\ Feet / Min = \frac{stroke\ x\ rpm}{6}$$

Using this formula with a stroke of 3.5433 inches and the rpm at 4,000 (My average cruising speed), the average Piston speed for the XPAG at this rpm is ***2362 Feet/Min***. As piston speed increases, internal friction increases by the square of piston speed. Consequently, frictional horsepower losses increase.

At this speed my car is traveling at about 70 miles per hour. This is the legal speed limit where I live and most cars are doing 85 mph! By modern standards this Average Piston Speed is not excessively high. But, we are not driving modern cars. So how do we fix this situation?

- Drive slower?
- Build an engine with more torque and use a 5th gear?
- Use a larger diameter tire?

Next month we will cover: Camshafts, Cam Timing, and Valve Train. We will compare four (4) factory cams and several after market cams using an engine simulator running on my computer.