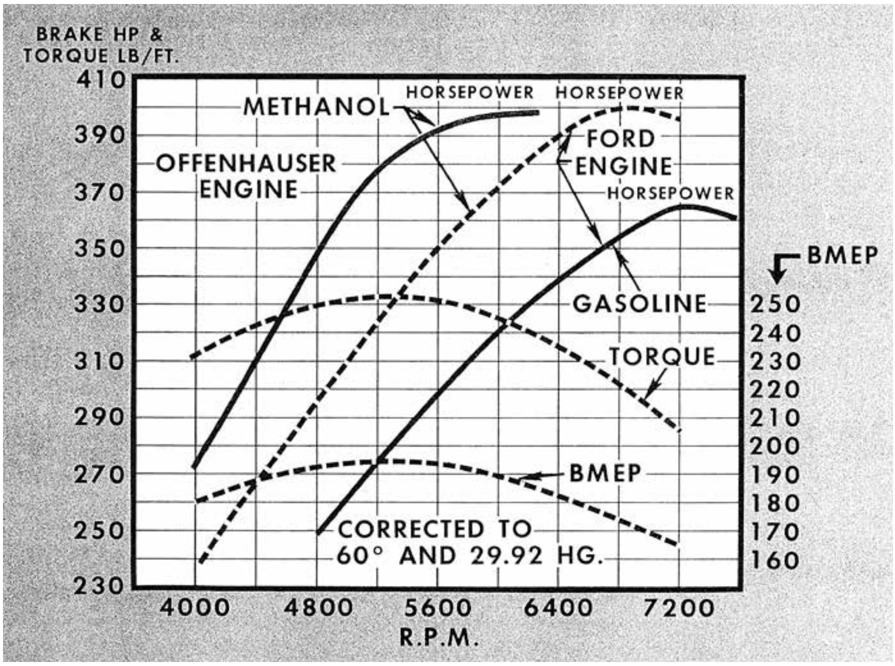

**Figure 5. Performance comparison.**

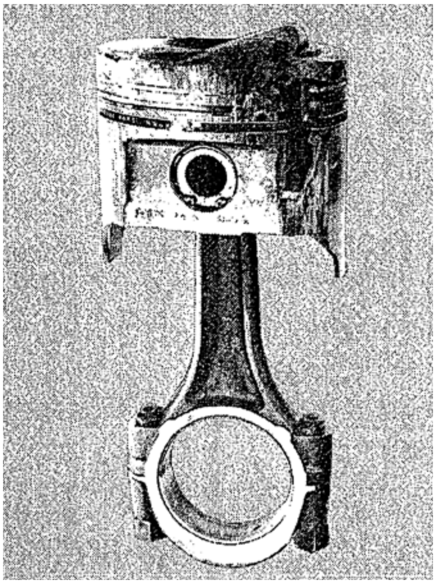

**Figure 6. Burnt piston from the use of nitro-methane in methanol.**

tailored to methanol. This led us into the controversial subject of racing fuels. It is probably safe to say that every blend of fuel capable of being ignited by a spark has been used. The blends of racing fuel that have proved best are aviation gasoline/petrol, benzol acetone, methyl alcohol, ethyl alcohol, isopropyl alcohol, and butyl alcohol. Other additive agents such as nitro-compounds, water, organic nitrates, peroxides, tetraethyl lead, ether, and explosive elements have been used. However, the air/fuel ratio and heating value in British Thermal Units (btu) per pound of fuel are the key to power output and possible advantage. The high specific gravity and the low stoichiometric ratio of exotic fuels presented a disadvantage to us because we wanted to consume only 400 gallons of fuel in the whole course of the race. We experimented with several fuels to study their pre-ignition tendencies, octane limited effect, and latent heat of vaporization against volumetric efficiency; we also mixed several nitro-parafins. By making several runs on methanol we were able to compare our engine with the Offenhauser (see Figure 5). This data spelled out the distinct advantage of the Offenhauser engine and seemed to substantiate our previous analytical study. This study had indicated that because of the short straightaways (between the corners at Indianapolis), torque wins at Indianapolis.

In summary, our data indicated a performance improvement of 10-12% by substituting methanol for gasoline/petrol. Upon adding nitro-methane to methanol we obtained data which showed an improvement of 1/3 of 1% increase in indicated power for each percent of nitro-methane added. Consequently, a 20% improvement in horsepower was possible with a mixture of 30% nitro-methane in methanol. This involved a penalty of 2½ to 3 times the gasoline/petrol consumption, providing of course that the engine could tolerate this oxygen-laden fuel reliably. Figure 6 concludes our fuel studies, and reinforces our decision to use gasoline/petrol as a dependable low-consumption fuel and to shun all exotic additives.

### Analytical

It became clear that the original objective of 325bhp on gasoline/petrol would at best produce an optimum lap speed of only 146mph. 365bhp on gasoline/petrol would make possible 150.50mph. 400bhp on methanol would make possible 153mph. Finally, 425bhp on gasoline/petrol could achieve a 155mph lap speed. We now had a driver pattern and a horsepower rating that indicated we should go at least 150mph. The accuracy of this last figure would be proven later in the programme when confirmation of our calculations would be checked by car testing in March 1963 at Kingman, Arizona and the Indianapolis Speedway.

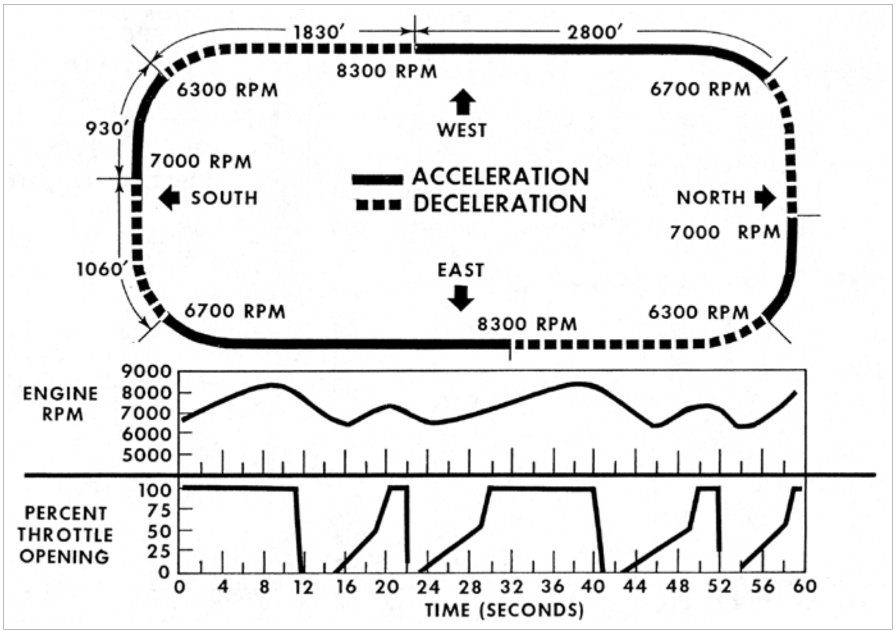

**Figure 8. Ford test vehicle, Indianapolis track, 25th March 1964: average speed 153.8mph.**

includes heat to the coolant, as well as water flow and temperature rise. It will be noted that the heat generated is in the order of 73,00 BTU/min (British Thermal Units) at 8000rpm. Brake horse power (bhp) accounts for 16,000-17,000 BTU and about 6200 BTU is delivered to the coolant. The water flow at 8000rpm is approximately 92 gallons per minute. There is an actual temperature fall of 2°F between engine operation at 4000rpm versus 8000rpm.

At this point, I should like to return to two items of data upon which we based important technical decisions at the outset of this engine development programme. The first of these is the track acceleration and deceleration characteristics. We carefully analysed the considerable amount of information obtained from our 1963 race entries. Using these as a datum in conjunction with the projected increase in engine performance to be achieved by our 1964 engine, we were able to work out the required acceleration and deceleration pattern to lap faster than before.

Figure 8 shows the engine characteristics, in one racing car during the March 1964 Indianapolis test, such as engine speed and throttle opening for a typical lap of the Indianapolis Speedway. The best average lap speed during this test period was 153.8mph (246km/h). To accomplish this, the driver reached a peak speed of 183mph (294km/h) down the straight section of the track, and completed the 2.5m (4km) lap in just over 58.5 seconds. During this particular lap the engine speed varied between 6700rpm and 8300rpm on the straight sections, decreased to 6300rpm entering the turns, and reached 7000rpm on the short, straight sections between the turns. Although this was the performance of a single racing car and driver on a single lap, we felt that it was sufficiently typical of the desired pattern to warrant using this data for our dynamometer durability cycling regime.

The second example of analytical information which was carefully assessed at the beginning of Phase 1 is that concerning the air flow characteristics required for the new engine design. A Lotus racing car with the Phase 1 engine was placed in a wind tunnel and on a rolling road dynamometer, so that track conditions could be simulated as closely as possible while being in a laboratory environment.

Calculations were made of valve lift, and piston velocity as a function of crankshaft angle, on the basis of the higher engine speeds for a number of valve and port configurations. Examination of this data showed that in the four-valve configuration the vertical port would provide adequate airflow

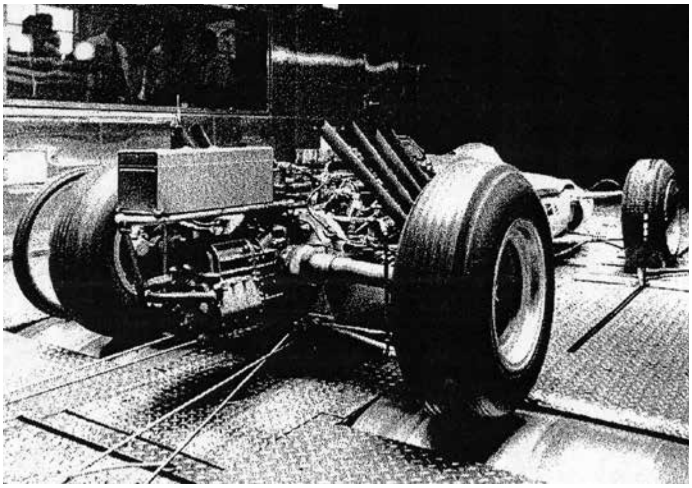

**Figure 9. Wind tunnel testing.**

### Exhaust system

It is well known that much can be done to improve engine performance by suitable tuning of the exhaust system. An exhaust system based on the theoretical tube length of 71.6in was developed on the dynamometer. It was decided to pursue a ½ and ¼ wave length study since each of these offered advantages in accommodating the system in the vehicle. Accordingly, several systems were designed using both 36in and 18in long primary pipes.

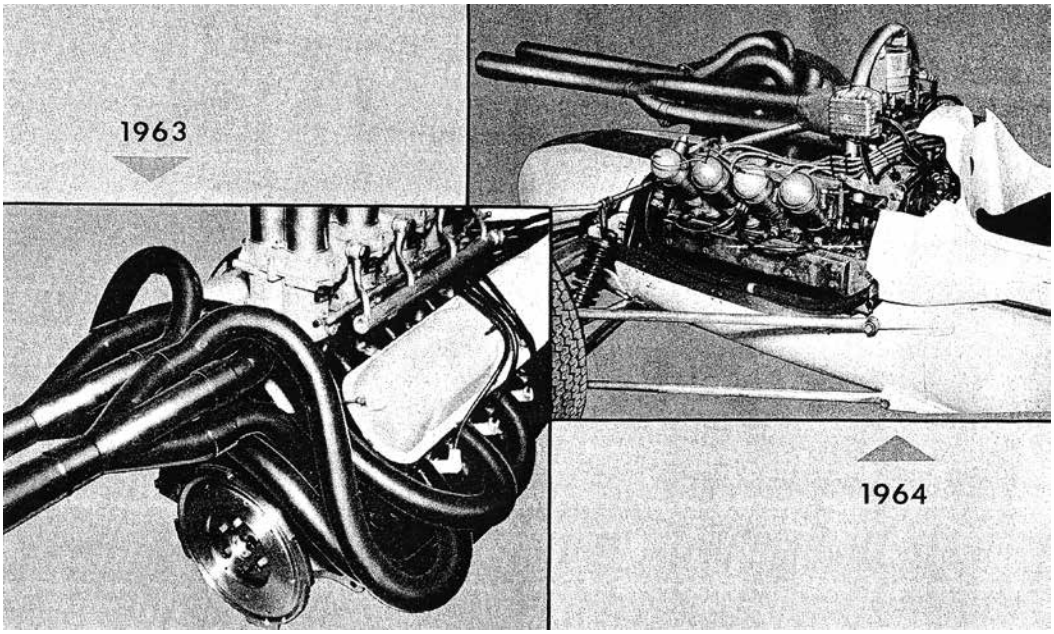

**Figure 26. Exhaust comparison. The 1963 exhaust system uses 36in long primary pipes.**

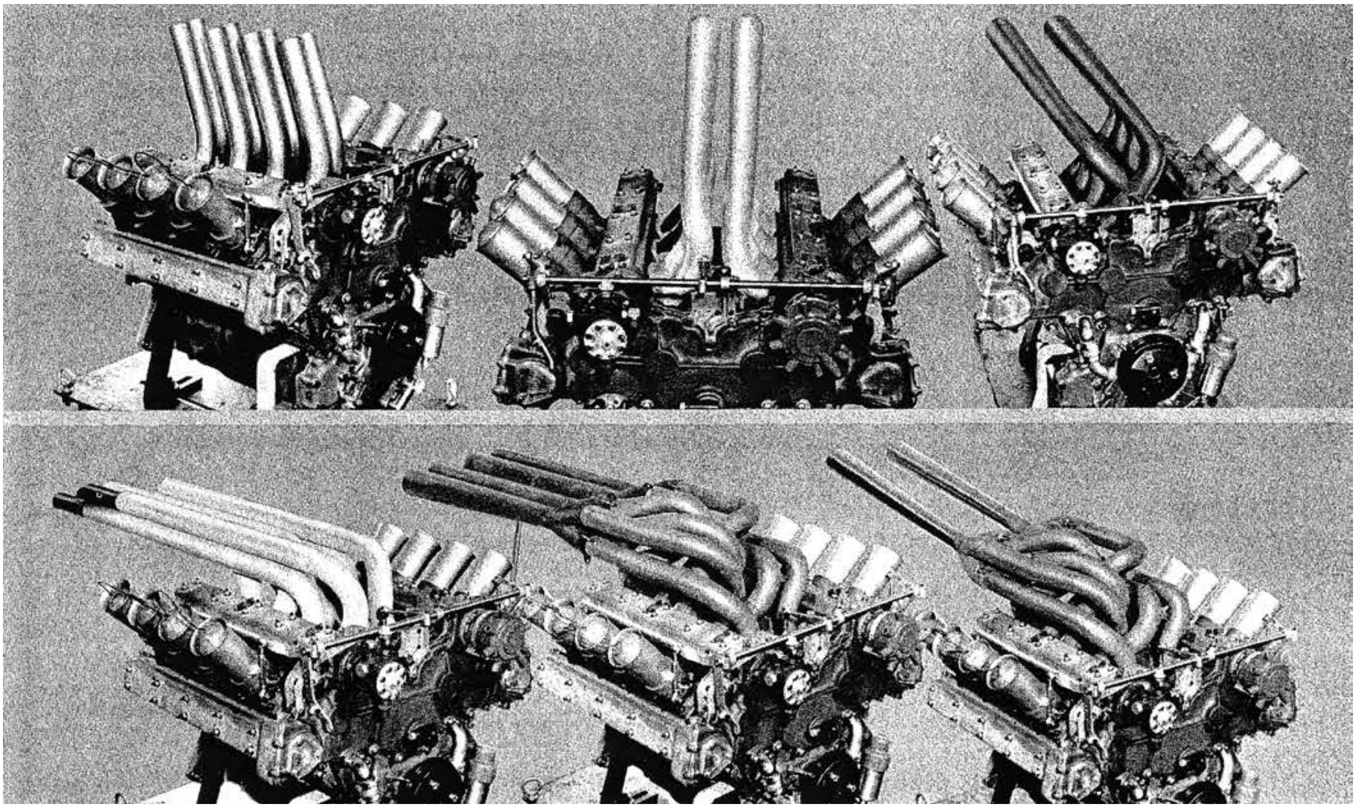

**Figure 27. The evolution of the exhaust system.**

## THE BASIC ENGINE

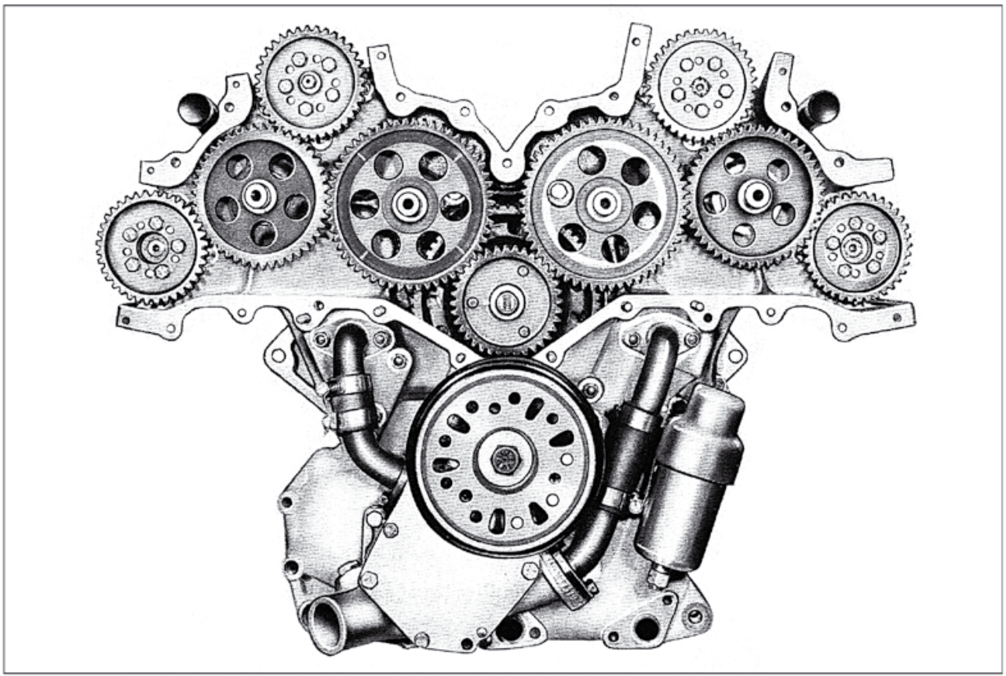

*Figure 15 – The Camshaft and Accessory Drive Gear Train for the Ford D.O.H.C. Engine*

### Camshaft and Accessory Drive

The camshafts, water and oil pumps, as well as the distributor, fuel pump, and alternator are driven from the gear train at the front of the engine. Steel spur gears are used throughout. Idler gears are straddle-mounted by ball bearings in the engine front cover and gear cover. The crankshaft gear is heated to 400° F, then pressed on the shaft and a steel key is used for positive location of the gear hub on the crank. Camshaft drive gears are bolted-on and have multiple mounting holes to provide a vernier adjustment for variable camshaft timing. The camshaft drive gears, as shown in Figure 15, would not normally be installed until the gear cover was assembled to the engine and gear backlash checked.

### Timing Marks

Timing marks are stamped on the vibration damper on the front of the crankshaft. A pointer on the engine cover indicates TDC and other markings on the damper. Duplicate timing marks appear on the flywheel, and may be used if engine installation permits.

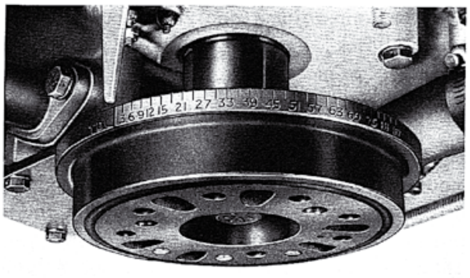

*Figure 16 – Timing Marks for the Ford D.O.H.C. Engine*

## Updating the GT-40 OHV 289 CID V-8 engine

This section is written specifically for owners of the 289ci GT-40 engine, particularly those interested in updating their current engine.

The standard engine in your Ford GT-40 is a 289ci 90 degree V-8. The original version of this powerplant developed 375bhp at 6800rpm. The bore and stroke are 4.000x2.870in, and the compression ratio is 10.5:1.

The components listed on the following pages are designed primarily to update the engine for increased durability and performance. The new engine block provides beefed-up main bearing webs and pan rails. It also incorporates four-bolt main caps in the centre three positions for improved durability.

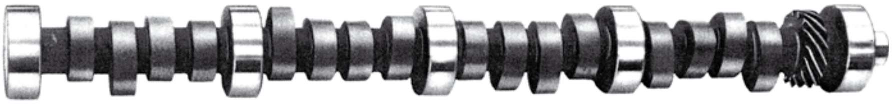

**Figure 9. The camshaft is the same as the 1966 Le Mans one, except it's a new casting to insure uniform lobe hardness and machining.**

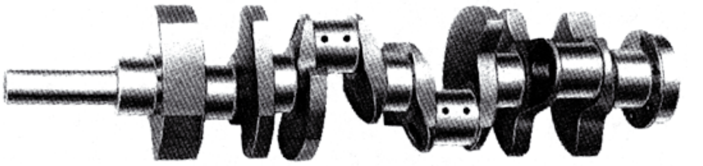

**Figure 10. The new crankshaft has revised counterweights to reduce the main bearing loads.**

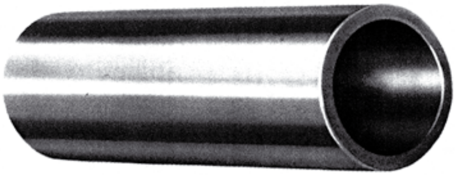

**Figure 11. The new piston pin is a cantilevered design for greater fatigue life.**

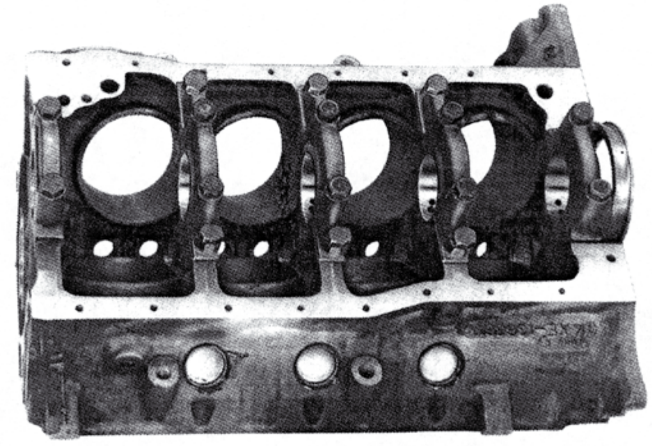

**Figure 7. New 4-bolt block.**

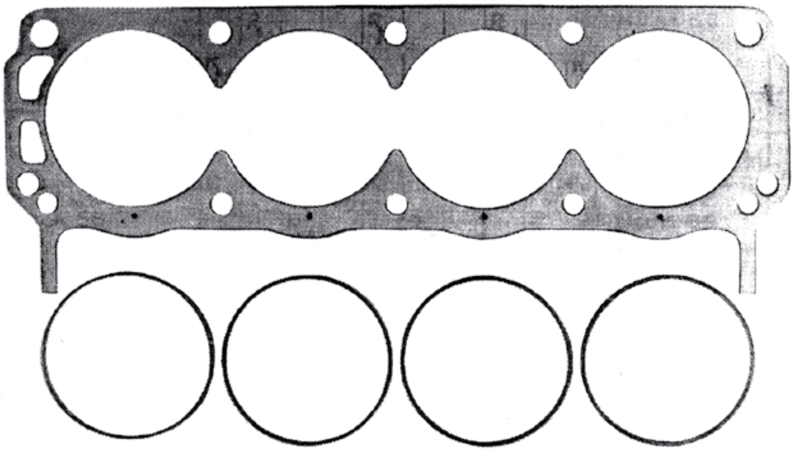

**Figure 12. This new cylinder head gasket has metallic O-rings with a periphery mattress.**

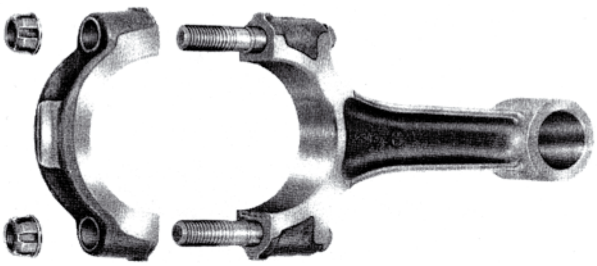

**Figure 8. The new connecting rod assembly has been fatigue-tested at 7000rpm.**

### Torque specifications

- Main cap 7⁄16in bolts x ten: 75-85ft-lb
- Main cap bolts x six: 35-40ft-lb
- Connecting rod bolt nuts: 45-50ft-lb
- Damper fit on crankshaft: 0.001in interference fit

### 1967 GT-40 update component part numbers

- Cylinder block C7FE-6010-A
- Cylinder heads C6FE-6049-A
- Piston C7FE-6110-B
- Piston pin C7FE-6135-A
- Connecting rod C7FE-6200-A
- Camshaft C7FE-6250-A
- Valve springs C7FE-6A536-A
- Forged steel crankshaft C7FE6303B