

## METALLURGICAL ANALYSIS OF ENGINE FAILURE

Lotus Elen Engine No. LP3367 LBA

### Description of Failure

The failure in this engine occurred while it was being briefly run in the speed range 5500 to 6000 rpm. After dismantling the engine the following damage was noted:

1. The engine block contained two holes, each about 1-1/2" dia., one in the no. 1 cylinder wall and the other on the left side of the block about 2" up from the sump flange. In addition the block was cracked at the front main bearing and in the vicinity of the oil pump. It was definitely beyond repair.
2. The connecting rod for the no. 1 cylinder was broken in two places. The bolts holding the bearing cap in place were broken, and the bearing cap was separated from the rest of the assembly.
3. The piston was shattered in the vicinity of the wrist pin.
4. The jackshaft (used to drive the oil pump and distributor) was broken in several places.
5. The crankshaft was severely gouged on the no. 1 connecting rod journal.
6. The exhaust and intake valves for the no. 1 cylinder were bent.
7. The vibration damper on the crankshaft was cracked.

The apparent primary cause of this damage was a fracture on the bolts holding the bearing cap on the connecting rod.



# Metallurgical analysis of connecting rod assembly

A series of hardness readings were taken on the pieces of the broken connecting rod and bolts to determine their strength. (See Figure 1). The undamaged rod from the no. 4 cylinder is included for comparison. The hardness was measured on the Rockwell C scale ( $R_C$ ); the results are reported in Table I. The tensile strength is the maximum steady stress the material can tolerate without failing. The fatigue endurance limit is the peak cyclic stress which can be applied to the material indefinitely. Cyclic stresses above the fatigue endurance limit but below the tensile strength will result in eventual failure.

TABLE I  
Connecting rod strengths

	Hardness, $R_C$	approx. tensile strength, psi.	approx. fatigue endurance limit, psi.
No. 1 cylinder:			
Con. rod, top	$20 \pm 1$	110,000	50,000
Con. rod, bottom	$20 \pm 1$	110,000	50,000
Con. rod, cap	$20 \pm 1$	110,000	50,000
Bolts	$30 \pm 1$	142,000	60,000
No. 4 cylinder:			
Con. rod	$20 \pm 1$	110,000	50,000
Con. rod, cap	$20 \pm 1$	110,000	50,000
Bolts	$29 \pm 1$	138,000	58,000



The bolts are made from a different steel than the connecting rod and are hardened to a higher strength. There is no appreciable difference in the strengths between the parts from the no. 1 cylinder and the no. 4 cylinder. The failure of the no. 1 connecting rod does not appear to result from the use of inferior material.

X-ray diffraction analysis of the region surrounding the fractures indicated severe deformation of the metal. This is obvious from a visual inspection as well. This deformation occurred immediately after the failure and before the engine finally stopped. There is no reason to believe that this deformation was the cause of the failure rather than the result of it. X-ray analysis of the metal for fatigue was precluded by the severe deformation.

#### Stress Calculations

The stress in the connecting rod and its bolts are primarily the result of the inertial forces of the piston and connecting rod assembly. These stresses are calculated here and compared with the tensile strengths of the parts to determine if the original design was adequate.

The stress in a connecting rod is approximated by the formula:

$$\sigma = 1.42 \times 10^{-5} (\text{RPM})^2 W S / A$$

- $\sigma$  = stress in psi.
- RPM = engine speed, revolutions per minute
- W = weight of piston and connecting rod, lb.
- S = stroke of engine, in.
- A = cross sectional area at narrowest part of rod, sq. in.

The stress in the bolts is calculated similarly by substituting the cross sectional area of the bolts in the above formula. The calculated stresses are given in Table II.