

Rotary Fatigue Analysis of Forged Magnesium Road Wheels

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ABSTRACT

Fatigue analysis incorporating explicit finite element simulation was conducted on a forged magnesium wheel model where a rotating bend moment was applied to the hub to simulate rotary fatigue testing. Based on wheel fatigue design criteria and a developed fatigue post-processor, the safety factor of fatigue failure was calculated for each finite element. Fatigue failure was verified through experimental testing. Design modifications were proposed by increasing the spoke thickness. Further numerical and experimental testing indicated that the modified design passed the rotary fatigue test.

INTRODUCTION

Vehicle wheels are critical components in the vehicle suspension system, which support static and dynamic loads encountered during vehicle operation. Wheels as rotating mass and unsprung weight must be as light as possible to benefit from increased fuel savings, quicker braking, improved tire wear and better acceleration [1]. Since the wheel is subjected to repeated cyclic loads, the major concern during wheel design is the fatigue behavior of the wheel [2]. Rotary fatigue testing is one of the traditional durability tests to simulate dynamic bending on the wheel [3]. In the rotary fatigue test, the wheel is constrained at the bottom flange of the rim, and a rotating bending moment is applied to the hub by a mandrel attached to the hub mounting face. Numerical simulation of wheel rotary fatigue test is highly desirable for wheel design, which shortens design time, reduces wheel weight, and enhances engineering performance.

This paper deals with the rotary fatigue performance of a proposed forged magnesium road wheel. The SAE J328 test procedure was used to assess the wheel rotary fatigue resistance. According to wheel design criteria considering fluctuating multiaxial stresses, the safety factor of fatigue failure has been calculated, using a developed post-processor [4]. The lowest safety factor of 0.749 was identified on the lower edge of the spoke

close to the hub, which represented the location of potential fatigue failure. All tested wheels failed during the rotary fatigue testing at this location. Further numerical analysis was completed to modify the wheel design for improved fatigue performance. Specifically, an increase on the spoke thickness was proposed. Experimental testing on the wheels with the modified design indicated that the alterations significantly improved the rotary fatigue resistance and the modified design passed the test.

WHEEL ROTARY FATIGUE TESTING

The testing standard, SAE J328, provides detailed information for wheel rotary fatigue testing [5]. The new 10-spoke forged magnesium wheel, with a diameter of 16 inches, is shown in Figure 1. The experimental test set-up of wheel rotary fatigue testing illustrates in Figure 2. In the rotary fatigue test, the wheel was secured at the bottom flange, and a rotating bending moment was applied to the hub. For the wheel in consideration, the bending moment was calculated to be 3486 Nm. Before the rotary fatigue test, 100% dye penetrating inspection of the wheel was checked for surface flaws prior to the wheel being mounted to the test fixture. Wheel nuts were tightened to 122-144 Nm and retighten as necessary during the first 5000 cycles to assure adequate torque stabilization. Inspection for cracks and reduction in the torque of the wheel nuts was completed after the required cycles.



Figure 1. 10-spoke forged magnesium wheel

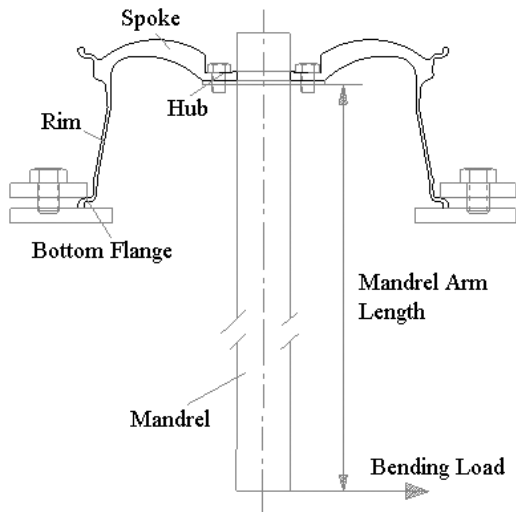


Figure 2. Wheel rotary fatigue test set-up.

MATERIAL PROPERTIES

The material for the wheel was a customized magnesium alloy MA2-1, the composition of which is close to AZ31B with the exception of a higher aluminum content. Table 1 lists the alloy designations of MA2-1 and AZ31B, and the analyzed chemical composition of the forged magnesium wheel. Sub-sized tensile and fatigue specimens were extracted from the spoke and rim of the forged magnesium wheel. Figure 3 presents the curves of engineering stress versus strain at the spoke and rim. The ultimate tensile strength of 253 MPa was considered for the entire wheel for fatigue analysis purposes. The stress/number of cycles to failure (S-N) curve of the forged magnesium wheel are plotted in Figure 4, in which fatigue data were approximated using a straight line on a logarithmic plot. The nominal fatigue strength of 90 MPa with 10^7 cycles was concluded.

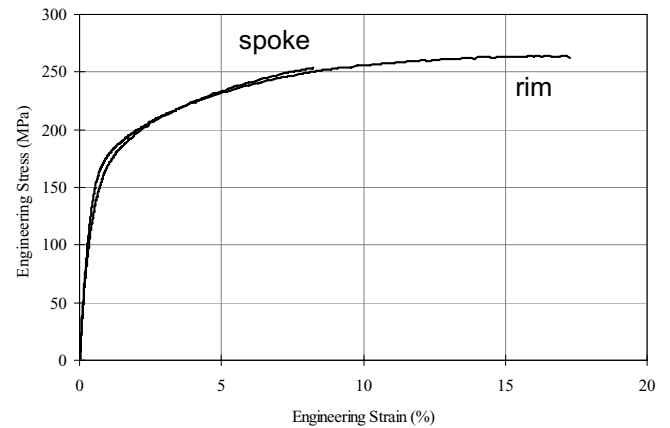


Figure 3. Tensile curve of the forged magnesium wheel.

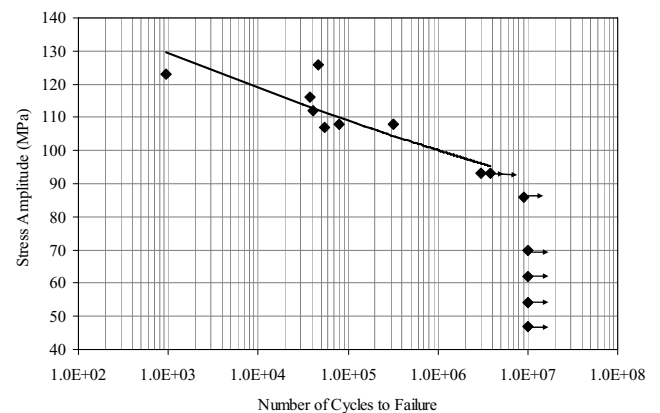


Figure 4. S-N curve of the forged magnesium wheel.

Table 1. The alloy designations of MA2-1, AZ31B, and the chemical composition of the forged magnesium wheel (wt%)

Alloy	Al	Zn	Mn	Cu	Ni	Si	Be	Fe	Others
MA2-1	3.8-5	0.8-1.5	0.3-0.7	<0.05	<0.004	<0.1	<0.002	<0.04	<0.3
AZ31B	2.5-3.5	0.6-1.4	0.2-1.0	<0.05	<0.005	<0.1	-	<0.005	<0.3
Wheel Sample	4.2	0.87	0.36	0.01	0.001	0.01	-	0.015	-

NUMERICAL MODELING

Numerical modeling of the wheel rotary fatigue test followed the experimental procedures described in SAE J328. The wheel was secured at the bottom flange to a testing fixture and a rotational bending moment was applied to the hub, which may be achieved by a rotating load on the free end of the mandrel. A rigid mounting plate and a rigid mandrel with length of 914.4 mm (3 feet) were constrained along the centerline on the hub. The total applied load on the free end of the mandrel

was 3810 N. In addition, the torque forces on the nuts were simulated by applying a 2224 N pressure around each of the bolt-hole area to represent the downward bolt load from the bolt heads. Eight locations with arc length 50 mm on the bottom flange are fully constrained to simulate the wheel being mounted on the stationary testing fixture. An elastic material model with mechanical properties corresponding to the wrought magnesium alloy ($E=45$ GPa, $\rho=1.8 \times 10^{-9}$ g/mm³, $\nu=0.3$) was assigned to the numerical model of the wheel. A rigid material was assigned to the mounting plate and

mandrel. Simulation of one complete loading cycle of the rotating bend moment was completed and used to determine the internal stress variation and distribution for each cycle for fatigue analysis purposes. The rotating bend moment was represented by applying two perpendicular loads on the free end of mandrel, which are varying with rotating time t in a sinusoidal behavior. The angular velocity ω associated with the rotating speed of the bend load was prescribed as 62.8 rad/s. Before the start of the simulation, P_x increased linearly from zero to 3810 N in 0.005 s and held constant for another 0.005 s, then varied with a sinusoidal response afterwards up to 0.135 s. Meanwhile, P_y maintained zero up to 0.01 s and followed a sinusoidal response afterwards up to 0.135 s. The combination of two perpendicular loads of P_x and P_y achieved a constant bend load P rotating around the z axis after 0.01 s with a constant angular velocity. Figure 5 illustrates the time varying dependency of P_x and P_y to simulate a constant rotating load between 0.01 to 0.135 s, represented 1¼ typical load cycles. Equations (1), (2) and (3) express the relationships of P_x , P_y and P . Figure 6 illustrates the wheel modeling and the applied loads P_x and P_y .

$$P = P_x + P_y \quad (1)$$

$$P_x = P \cos[\omega(t - 0.01)] \quad (2)$$

$$P_y = P \sin[\omega(t - 0.01)] \quad (3)$$

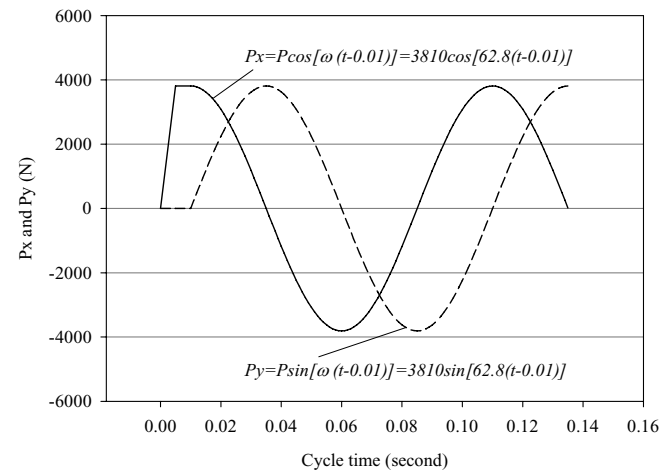


Figure 5. The time varying dependency of P_x and P_y .

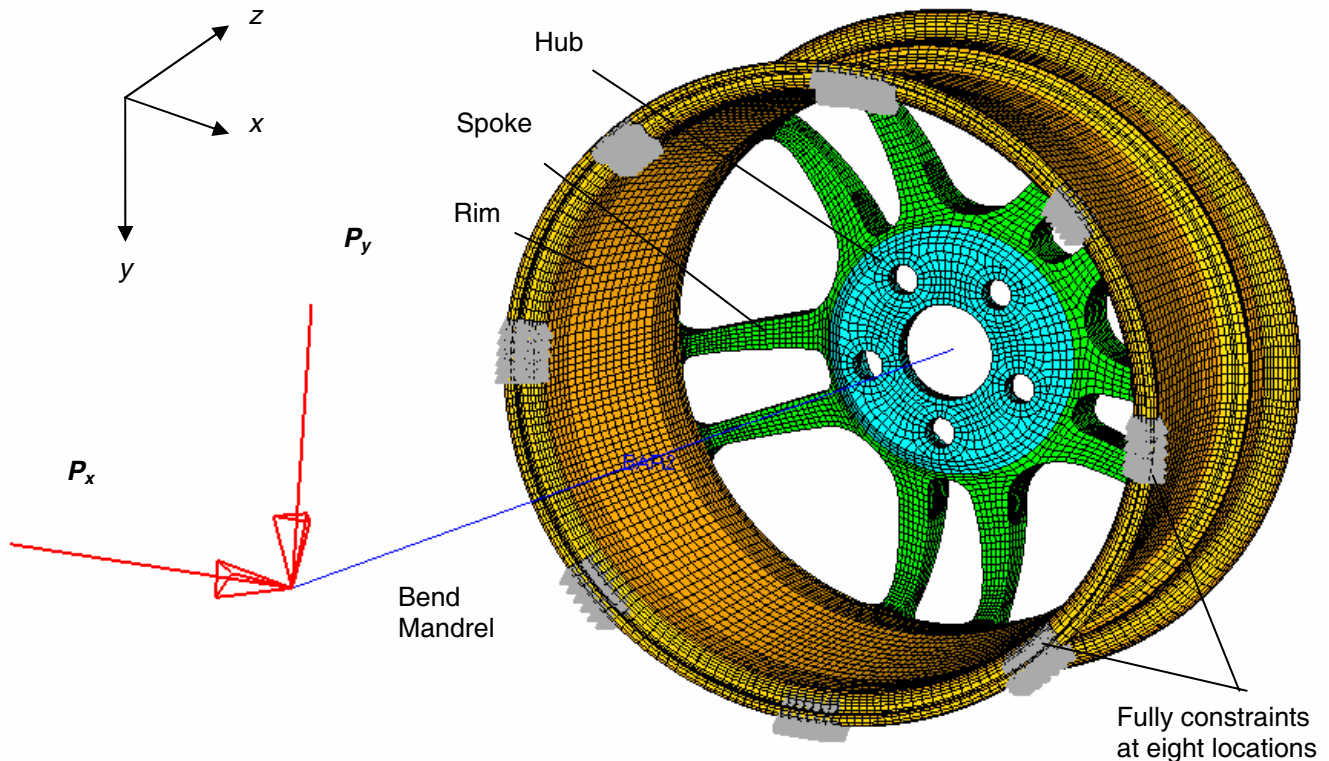


Figure 6. The wheel model and the applied loads P_x and P_y .