Analysis of a Failed Saw Arbor

C. Kendall Clarke[1]

Authorized Reprint from
Special Technical Publication 918

C. Kendall Clarke[1]

Analysis of a Failed Saw Arbor

REFERENCE: Clarke, C. K., **"Analysis of a Failed Saw Arbor,"** *Case Histories Involving Fatigue and Fracture Mechanics, ASTM STP 918*, C. M. Hudson and T. P. Rich, Eds., American Society for Testing and Materials, Philadelphia, 1986, pp. 336–343.

ABSTRACT: Stress analyses incorporating stress concentration, fatigue, and fracture calculations should be an important aspect of failure analyses. The following description of a saw arbor failure provides a simple example of this necessity. The failure was a classical stepped shaft in bending fatigue failure. Stress concentration at the change in diameters was calculated to be a factor of five. However, stress calculations revealed normally expected stresses to be well below the endurance limit even with the stress concentration effects. Additional investigation revealed evidence for severe imbalance loads from one saw blade. It was hypothesized that some of the teeth from the carbide tipped saw had been knocked off, and the resulting imbalance loads were more than sufficient to initiate fatigue cracking.

KEY WORDS: fatigue, stress concentration, stress analysis, mutual hardness calculations

A saw arbor holding a 0.56 m (22 in.) diameter saw blade in a lumber mill trimming saw machine failed while in operation. The spinning carbide tipped blade then cut through guards and seriously injured a worker. Visual observation of the fracture revealed that fatigue cracking had initiated at and propagated from a change in diameters on the arbor. No significant radius was observed at the change in diameters.

This type of shaft failure is often written up in failure analysis literature as a stress concentration induced failure caused by a change in shaft diameters. The purpose of this paper is to show that routine calculations reveal that this particular failure, and probably many other similar failures, are not always simple. Stress calculations involving stress concentration and fatigue calculations provide a powerful tool with which to arrive at a more complete analysis of failure. The underlying principle is that results of stress, fatigue, and fracture calculations should be consistent with the proposed mode of failure.

The trimming saw consisted of eleven identical, independent saws and was used to trim lumber to finished lengths. Figure 1 shows a side view of

[1] President, Metallurgical Consulting, Inc., Mobile, AL 36609.

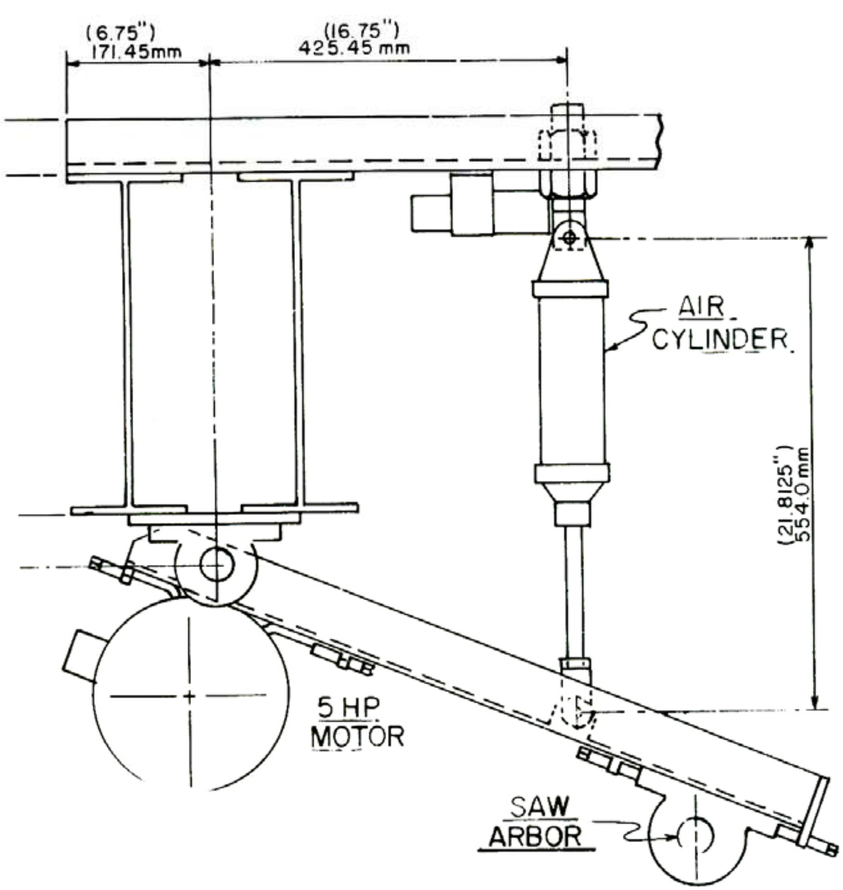

FIG. 1—*This schematic drawing shows the general arrangement of the trimming saw.*

one of the identical trimming saw systems. The arbor design and crack origin location are shown in Figure 2. This design had been in service for several years without any reported problems or failures. The specific equipment involved in the accident had been in service nearly two years prior to the accident.

Trimming saws were arranged every 0.6 m (2 ft)[2] so that lengths from 0.6 m (2 ft) to 6 m (20 ft) could be trimmed. Each saw was equipped with a 3730 J (5 HP) electric motor turning the saw blade at 2200 rpm. All of the saws were normally turned on each day of operation. However, the saw which failed was at the 5.5 m (18 ft) location and only cut wood about 2% of the time.

[2] All measurements and calculations were originally in English units.

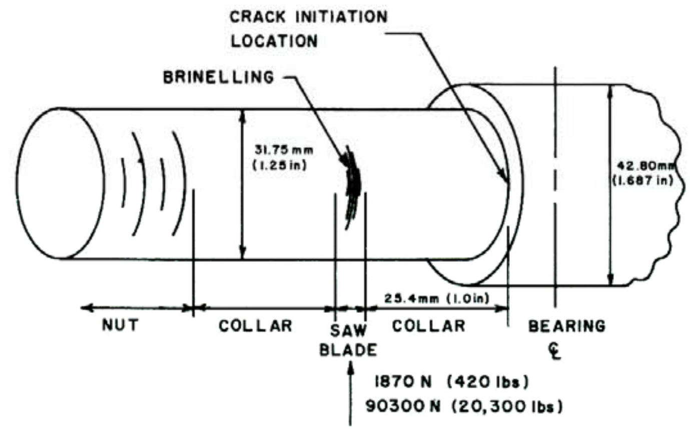

FIG. 2—*Two heavy collars and a nut were used to hold the blade in place upon the saw arbor.*

Results

Fractographic Evaluation

Figure 3 shows the fracture surface of the saw arbor. The collar which butted up against the step in the arbor is still on the shaft. Figure 4 shows the arbor surface where the saw blade rode. Fracture occurred where the 31.75 mm (1 ¼ in.) arbor shaft stepped sharply up to 42.8 mm (1 ¹¹⁄₁₆ in.). A large collar to support the saw blade was press fitted onto the arbor and up against the shoulder.

The fracture surface had the typical characteristics of a fatigue failure. A single crack origin can be observed in Fig. 3. No significant defect was observed at the origin. No evidence for fretting was observed at the crack origin. The very initial cracking propagated straight across the arbor diameter for approximately 2.5 mm, as if a bending moment were applied only at the crack origin location. Cracking then changed abruptly to a manner more typical of rotating bending. (The point of maximum bending stress changed with shaft rotation.) The final fracture region was small, thus indicating low service stresses at failure.

Stress Calculations

Stress calculations were made assuming normal operating conditions to determine if stresses capable of initiating fatigue cracking were possible at the shoulder (change in diameters where cracking initiated) in the arbor. These calculations included stress concentration effects. When these initial calculations revealed low stresses, additional calculations became necessary to justify fatigue crack initiation.

FIG. 3—*The complete arbor fracture surface is shown here. An arrow points to the origin. No obvious material defect was observed.*

A determination of the stress concentration factor was the first calculation made. The radius at the base of the shoulder on the cold-rolled steel shaft was measured to be 0.13 mm (0.005 in.) by using wire feeler gages and a low power microscope. This combination of radius and ratio of diameters produced stress concentrations of 4.7 and 5.1 using charts from Peterson [*1*]. A shear stress concentration factor was also found from the same charts to be 4.0 to 4.5. The collar on the arbor was reportedly press fitted up against the shoulder where the fracture occurred. This contribution to stress concentration was ignored for lack of data.

Torsion stresses were calculated to be very low on the shaft. The torque on the arbor with no load was calculated to be

$$T = \frac{(396000)\ (\mathrm{HP})}{2\pi\ \mathrm{rpm}} = \frac{(396000)\ (5)}{2\pi\ (2200)} = 143\ \mathrm{in.} \cdot \mathrm{lb\ or\ } 16.1\ \mathrm{N} \cdot \mathrm{m} \quad (1)$$

Or under load:

$$T = \frac{(396000)\ (5)}{2\pi\ (1100)} = 286\ \mathrm{in.} \cdot \mathrm{lb\ or\ } 32.3\ \mathrm{N} \cdot \mathrm{m}$$

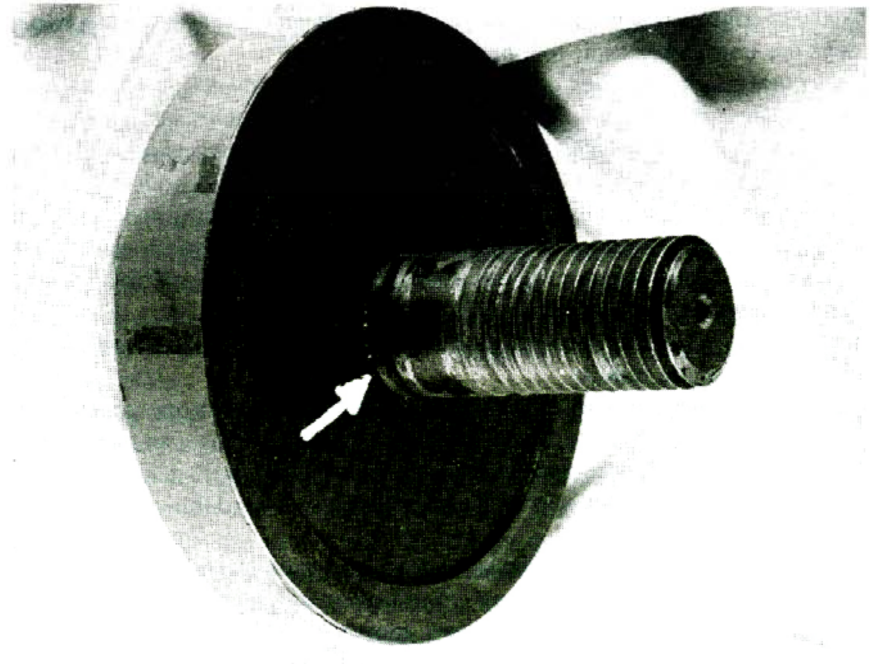

FIG. 4—*The large washer or collar was press fitted onto the shaft and up against the shoulder. A brinelled area can be seen* (arrow) *where the unbalanced saw blade deformed the shaft.*

The torsional stress on the shaft is therefore

$$\tau = \frac{Tr}{J} = \frac{(143\ \mathrm{to}\ 286)\ (0.625)}{(0.782\ \mathrm{in.}^4)} \quad (2)$$

$$= 114\ \mathrm{to}\ 228\ \mathrm{psi}\ (7.86 \times 10^5 - 1.57 \times 10^6\ \mathrm{Pa})$$

Including stress concentration effects, these stresses are too low to have initiated a failure.

The basic saw dimensions are shown in Figs. 1 and 2. The air cylinder could provide up to 3560 N (800 lb) in downward force. A simple statics analysis was made assuming the primary force on the arbor was the air cylinder. The nut clamping force was ignored because torquing requirements were low. (The wood feed force was much less.) The maximum force on the arbor was 1870 N (420 lb), and this force decreased as the saw arm dropped. This force translated into a bending moment on the 31.75 mm (1 ¼ in.) arbor of 47 500 N · mm (420 in. · lb). The maximum or outer fiber bending stress was, therefore

$$\sigma = \frac{Mc}{I} = 15.0\ \mathrm{MPa}\ (2180\ \mathrm{psi}) \quad (3)$$

A stress concentration factor of 5 would not elevate this stress to anywhere near the endurance limit for the steel.

No material properties other than hardness were available for the arbor because destructive testing had not been permitted. Therefore, a lower limit on the fatigue endurance limit was estimated from the hardness. The cold rolled steel shaft had a hardness of 143 BHN which produces a tensile strength of approximately 482 MPa (70 000 psi). The endurance limit could be conservatively estimated to be 35% of the tensile strength or 169 MPa (24 500 psi). This number is well above the calculated bending stresses in the shoulder region including effects of stress concentration. Therefore, some other factor was required to have raised stresses high enough to have initiated a fatigue failure.

Careful examination of the arbor revealed brinelling at one point on the steel shaft where the saw blade fit (Figs. 2 and 4). This was caused by a force on the saw blade high enough to deform the shaft in one location. The circumferential positions of the brinelled area and of the crack initiation site coincide.

The principle behind hardness testing was used to estimate the force required to deform the saw arbor. Hardness tests are run by pressing with a known force an indenter of known geometry and hardness into an unknown material. The hardness or plastic flow stress of the unknown material is calculated on the basis of the area of the indention and the known force. However, in this case, the hardness of the arbor and the indention area were known. Reference *2* provides a method for determining the force required to make the indention. This method provides factors which take into account the indentor geometry and hardness.

Hardness measurements on the shaft yielded a hardness of 143 BHN. The hardness of the saw blade was assumed to be comparable. A local yield stress can be calculated using a factor of 2.8 to account for the Brinell ball shape and hardness

$$\frac{143\ \text{BHN}}{2.8} = 51.1\ \text{kg/mm}^2 = 500\ \text{MN/m}^2\ (72.6\ \text{ksi}) \tag{4}$$

Similar hardnesses and a cylinder inside a cylinder geometry yield a hardness factor of 1.4[2]. This factor times the calculated yield stress from Eq 4 yields a constrained flow stress of approximately 700 NM/m^2 (Eq 5). The indentation area

$$(500\ \text{MN/m}^2)\ 1.4 = 700\ \text{MN/m}^2\ (102\ \text{ksi}) \tag{5}$$

$$(700\ \text{MN/m}^2)\ (1.29 \times 10^{-4}\ \text{m}^2) = 0.0903\ \text{MN}\ (20.4\ \text{kips}) \tag{6}$$

was measured to be 1.29×10^{-4} m^2, and this area times the constrained

flow stress yields a force required to produce the indentation of approximately 90 300 N.

This calculated force is relatively high and yields a maximum bending stress at the site of crack initiation of 730 MPa (106 ksi). This is at or near the failure stress of the steel in bending. Assumptions regarding the hardness of the saw blade and the probability that the total measured indented area developed as a result of increasing blade movement as the blade hole deformed certainly would affect the magnitude of this force. (The actual instantaneous contact area could have been half or less of the total measured area of damage.) The point is that a source for higher stresses has been found, and that refinements to the calculations could be made if the data were available.

The possibility of an overloaded saw blade was confirmed when a check with local saw repair shops revealed that the particular saw mill in question had experienced problems with pieces of occasional steel in the lumber knocking carbide teeth from the saw blades. The resulting imbalance would have overloaded the saw arbor. In this particular case, half of the teeth missing would produce just about the level of force calculated in Eq 6.

A closer approximation of the actual stress on the shaft might be made by considering the fracture surface. Cracking initially propagated about 2.5 mm straight along the shaft diameter. Then the crack front changed to one more typical of rotating bending. It was known that the saw on the arbor at failure was in good condition and was about 1.5 mm wider than the one which initiated the failure. Thus the change in crack propagation can be explained by a saw blade change. When the blade was changed is not known. However, the damaged saw ran for at least several minutes before being shut down and the blade changed. Therefore, the initial crack growth on the arbor would have occurred over the period of a few thousand cycles to over a million cycles. Arbor stresses can then be considered to be bounded by the endurance limit of 170 MPa and the 1000 cycle limit. One thousand cycle life limits can be estimated by the following equation [*3*]

$$\sigma = \sigma_f'(2\,N)^b \tag{7}$$

where

σ = predicted stress for N cycle life,
σ_f' = fatigue strength coefficient (estimated = 900 MPa),
b = exponent (estimated = -0.13), and
$\sigma = 900\ (2000)^{-0.13} = 335$ MPa (48.6 ksi).

The estimated brinelling force is certainly in this order of magnitude and would probably fall within this range if more data were available.

Conclusion

The failure of the saw arbor is hypothesized to have started when a piece of steel in lumber being trimmed knocked off several carbide teeth on the saw blade. The resulting imbalance load was adequate to initiate fatigue cracking, but it was not adequate to cause blade failure before the blade was changed. (A new blade was put on the arbor and cracking continued at a much slower rate.) Calculations or the force required to cause the brinelling were on the high side. Actual forces were most probably lower. Therefore, the imbalance hypothesis was consistent with the stresses required for fatigue fracture while stress concentration alone was not.

This case provides an excellent illustration of the necessity for performing stress calculations as part of a failure analysis. Conclusions from the fractographic analysis should be consistent with the stress calculation results. Sophisticated fatigue and fracture calculations were not required for this particular case. However, these calculations are often very helpful in more complex cases such as pressure vessel failures.

References

[1] Peterson, R. E., *Stress Concentration Factors*, Wiley, New York, 1974, Figs. 78, 78*a*, 79 and 79*a*.

[2] Atkins, A. G. and Felbeck, D. K., *Metals Engineering Quarterly*, Vol. 14, No. 2, May 1974, pp. 55–61.

[3] Fuchs, H. O., and Stephens, R. I., *Metal Fatigue in Engineering*, Wiley, New York, 1980, p. 155.