CRITICAL EVALUATION OF BICYCLE SPINDLE FAILURE

MTRL 585
Case Study 5

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GROUP 7

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1.0 BACKGROUND

A used road bicycle was purchased by a former MTRL student and used for commuting to school. During a commute, the left crank arm completely fractured. The bicycle was a 1979 Raleigh Super Course and was stored inside for most of its life. For a period prior to the failure, a distinct wobble was noticed in the left pedal. Figure 1 shows general pictures of the fractured component [1]. Using the information provided and engineering theory, the cause of failure was analyzed.

![Figure 1: Fractured bicycle crack arm spindle [1]](image)

2.0 TASK 1

2.1 Material Determination

Energy dispersive x-ray (EDX) spectroscopy was used to perform an elemental analysis on the material, and results were provided for this analysis. As shown in Table 1, the results were very inconsistent. EDX elemental analysis is considered semi-quantitative, as it is dependent on topography and cleanliness of the sample, and has limited accuracy for light elements. It was not indicated where the scans were taken (outer surface, fracture surface, hardened layer, etc.), which may have had an influence on the results. The results from EDX often require a certain degree of subjectivity from the operator to prevent such misdiagnosis. Although considered a destructive test, the exact alloy should be determined according to accurate measurements of chemical composition using optical emissions spark (OES) spectrometry as per ASTM E415 [2].
Table 1: Provided EDX results

<table>
<thead>
<tr>
<th>Run #</th>
<th>C (wt.%)</th>
<th>Mn (wt.%)</th>
<th>Si (wt.%)</th>
<th>Fe (wt.%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>-</td>
<td>1.1</td>
<td>0.4</td>
<td>98.5</td>
</tr>
<tr>
<td>2</td>
<td>34.3</td>
<td>0.6</td>
<td>0.1</td>
<td>65.0</td>
</tr>
<tr>
<td>3</td>
<td>37.7</td>
<td>-</td>
<td>0.5</td>
<td>61.8</td>
</tr>
<tr>
<td>4</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>100</td>
</tr>
<tr>
<td>5</td>
<td>37.7</td>
<td>-</td>
<td>-</td>
<td>62.3</td>
</tr>
</tbody>
</table>

The results provided include some obvious error and a lot of scatter, so it must be assumed that the analysis was performed incorrectly. Therefore, the fractured sample was brought to Powertech Labs for some additional EDX analysis. The spectra were obtained at mid-range magnifications of ~200 times to avoid areas of segregation, uneven topography, or contamination. As can be seen from spectra in Figure 2 and measurements summarized in Table 2, Mn content was about 0.86 wt% and although Si was not measured, the spectra show small peaks that may have been discounted as noise. As such, it is likely that the material is plain carbon steel in the 1000 series. To determine which alloy in the series, carbon content must be measured with OES. For the purpose of this investigation, the alloy can be assumed to be 1045 since it is common in this application.

Table 2: EDX measurements

<table>
<thead>
<tr>
<th>Run #</th>
<th>Mn (wt.%)</th>
<th>Si (wt.%)</th>
<th>Fe (wt.%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.88</td>
<td>0.0</td>
<td>99.12</td>
</tr>
<tr>
<td>2</td>
<td>0.83</td>
<td>0.0</td>
<td>99.17</td>
</tr>
<tr>
<td>Average</td>
<td>0.86</td>
<td>0.0</td>
<td>99.14</td>
</tr>
</tbody>
</table>

Figure 2: EDS spectra of general fracture surface

2.2 Visual Examination

Many features become immediately apparent on first inspection of the fracture surface. Many circular concentric lines can be seen extending from the left side of the image. These features are called beach marks, which are typical of fatigue
failure. Beach marks are formed when there is a change in conditions of cyclic load, whether it is load amplitude or environment. In this case, we see many broad bands of orange product, which is assumed to be iron oxide (rust). Crack propagation in the banded regions of rust may have occurred during wet cycling seasons, which may have accelerated fatigue crack growth (via corrosion fatigue). Since beach marks are aligned with the crack front as it progresses, they can be tracked back toward the crack's origin. As indicated in Figure 3, the origin is in the top left of the image. There were no obvious signs of defects or external damage in this area.

As shown in Figure 3, the fracture surface at the right of the image looks quite rough. It is presumed that this is the region of final overload failure which occurred once the stress intensity at the tip of the fatigue crack exceeded a critical value. The hardened layer surrounding the entire fracture surface also has a distinct boundary. Although it was not apparent from the provided images, visual inspection of the spindle revealed that the fracture initiated at an angle of approximately 58 degrees from the longitudinal axis, or 32 degrees from the transverse plane (Figure 4). As fatigue will typically progress across planes of maximum shear, this indicates that the spindle was under multi-axial loading [3].
2.3 SEM Analysis

To supplement the high-magnification scanning electron (SEM) images provided, additional examination was performed at Powertech Labs. As shown in the secondary electron image (SEI) in Figure 6, fracture of the case hardened layer was predominantly intergranular, as evidenced by a “rock candy” appearance. The hardened layer was approximately 750 µm thick. Figure 6 shows a higher magnification SEI of intergranular fracture surface.
The region displaying beach marks was also imaged at high magnification. As shown in Figure 7, the presence of fatigue striations was confirmed. The fatigue striations mark the advance of the crack front with every load cycle. The direction of fatigue crack propagation is indicated in the picture with a black arrow.

An SEI was taken within the region of final rupture, and is provided in Figure 8. The features of the fracture surface can be described as microvoid coalescence, which is typical of overload in ductile materials. Thus, the boundary of fatigue cracking was confirmed as that illustrated in Figure 3. The length of the fatigue crack at final
fracture was approximated as 69% through the thickness of the spindle (~10.0 mm).

As shown in Figure 9, backscattered electron imaging (BSEI) revealed frequent occurrence of unexpected features across the fatigue fracture surface (indicated by arrows). The EDX elemental color mapping shown in Figure 10 identifies the features as being high in manganese and silicon, and devoid of iron. An SEI was taken at higher magnification, as shown in Figure 11. Here it can be seen that the particles are inclusions within the steel, rather than simply deposited on the surface. It was also observed that the particles would turn white when viewed in secondary electron mode, which indicates that they are non-conductive. Therefore, it is proposed that the particles are oxide inclusions introduced during production of the material. These brittle oxide inclusions will act as stress concentrators within the material, contributing to early initiation of a fatigue crack. The material’s strength and fracture toughness will be reduced, and fatigue crack growth rate will be accelerated.
Figure 9: BSEI showing anomalies on fracture surface

Figure 10: EDX elemental color map of Mn distribution (left) and Si distribution (right)
3.0 TASK 2

3.1 Mechanical Analysis

Reaction forces and reaction moments due to the peddling force were calculated to understand the stress state within the spindle. The rider’s initial force was originally assumed to be 500N and a simple sinusoidal function of crank rotation. It was also assumed that forces were only felt on the left spindle during a 180 degree rotation of the crank arm, beginning in the vertical position. Using a coordinate system fixed to the spindle, as shown in Figure 12, the pedaling force was first broken down into normal and radial components.

\begin{align*}
F_n &= F_0 \cdot -\sin(\theta) \\
F_r &= F_0 \cdot -\cos(\theta)
\end{align*}

Figure 12: Spindle coordinate system
Next, using the spindle geometry and applied forces depicted in Figure 13, the reaction forces and moments were calculated.

\[ N_x = 0 \]
\[ N_y = -F_r \]
\[ N_z = -F_n \]
\[ M_x = -F_n * .17 \]
\[ M_y = -F_n * .055 \]
\[ M_z = -F_r * .055 \]

Each reaction is a function of theta. Plots depicting the dependence on theta are shown below.

**Figure 13: Configuration and basic dimensions of pedal/crank/spindle assembly**

- **Dimensions**
  - \( a = 55 \text{ mm} \)
  - \( b = 170 \text{ mm} \)
  - \( d = 12 \text{ mm} \)
Figure 14: Pedaling forces as a function of rotational angle

Figure 15: Reaction forces as a function of rotational angle
After the reaction forces and moments were calculated, the general stress state in Cartesian coordinates was calculated. The stress in the x direction (due to bending) was plotted in four locations: \((y, z) = (R,0), (-R,0), (0,R)\) and \((0,-R)\), where \(R\) is the radius of the spindle. Positive values represent tensile stress while negative represent compressive.
It is easily seen that the maximum bending stresses are encountered along the z axis at \( z = \pm R \). One can also quickly see that these maximum stresses occur at a theta value equal to 90 degrees. Lastly the shear stresses were calculated. The shear due to the reaction forces \( N_y \) and \( N_z \) are assumed to act uniformly over the cross sectional area of the circular spindle face. The shear torque encountered from the reaction moment about the x axis will be zero at the radius = 0 and vary linearly to a maximum at the outer radius. It should be noted that shear torque >> simple shear and simple shear will be disregarded for the remainder of the analysis.

\[ \sigma_{xx} \] (from bending) and \( \tau_{xa} \) (shear stress from torque). Note that the subscript \( \alpha \) is used to denote the angular dimension on the spindle cross section, which is different than \( \theta \) which denotes the whole crank’s angle. The two dominant stresses are maximum at crank angle \( \theta = 90^\circ \).

Assuming a circular cross-section the failure occurs very near the point \( y = 0, z = R \). The two dominant stresses, \( \sigma_{xx} \) and \( \tau_{xa} \), are sketched into imaginary planes at this point, as shown in Figure 19.
3.2.2 Principal stresses and direction

In calculating the principal stresses, all other stresses other than the two dominant stresses mentioned above (σ_{xx} and τ_{xa}) were neglected due to their small magnitudes in comparison. Technically, since σ_{yy} and σ_{zz} are zero, only τ_{xy} can change the principal stress and direction results for the x-y plane illustrated above; however, the magnitude of τ_{xy} ≪ τ_{xa} therefore it is not included. Using the convention for the directions of the stresses illustrated in Figure 20 below, and the σ_{xx} and τ_{xa} values in Figure 21 (from Task 2 calculations), the principal stresses and corresponding angles were found using an online calculator\(^1\) and presented in Figure 22. The plane orientation at which there are principal normal stresses is turned 36.1° clockwise of the original x-y orientation, as shown in Figure 23. In other words, the principal stress σ_1 makes a (36.1 + 90)° or 126.1° angle with the positive y-axis of Figure 19. This correlates well with the observed angle of the fatigue plane (32°).
3.2.3 Comparison of calculated and measured failure planes

Technically, the failure location of the actual spindle is not exactly at point $y = 0, z = R$, rather, slightly in the counter clockwise direction. All calculations were performed assuming that the spindle had a circular cross-section. Since the actual cross-section was square with chamfered corners, the initiation point was offset to a region of slightly higher stress concentration.

Observation of this point on the failed spindle reveals that the $y$-$z$ crack face is not flush with the crank arm; it spans from recessed of the inner face of the crank arm to standing proud. As expected, however, the initiation point occurred right at the junction of the spindle and crank arm. Since the crack initiated at an angle, the
propagation direction was both inward and outward of the inner face of the crank arm.

As previously mentioned, the fatigue plane was measured as -32° relative to the spindle y-z plane. Since the initiation and growth of fatigue cracks commonly occurs perpendicular to the direction of maximum stress, this observation corresponds very well with the calculated principal angle of -36.1° relative to the spindle y-z plane.

3.2.4 Von Mises stress and yielding safety factor

The equation for calculating the von mises stress in 2 dimensions (using principal stresses) is as follows:

\[
\sigma_{VM} = \sqrt{\frac{(\sigma_1 - \sigma_2)^2 + \sigma_1^2 + \sigma_2^2}{2}}
\]

where \(\sigma_1\) and \(\sigma_2\) and the principal stresses

Inserting the principal stress values found in Figure 4 into the equation yields \(\sigma_{VM} = 464.4\) MPa

From our EDX analysis, the most likely material the spindle is made from is 1045 Steel, which, from Zhang et al. [5], has a yield strength of 1636 MPa in the case hardened portion. Thus, the safety factor with respect to yielding is:

\[
FS = \frac{S_y}{\sigma_{VM}} = \frac{1636}{464} = 3.5
\]

It is important to mention that, as evident by the equation above, the factor of safety is directly proportional to the yield strength value. Therefore, the above factor of safety result can vary greatly for different yield strengths. In the literature, it was found that there is a transition zone between the case hardened surface and the core section of hardened parts, as will be presented in subsequent parts of this report. The least conservative value for the yield strength in this case is at the border between the transition zone and the core, which is around 620 MPa. This is the same value reported for 1045 Steel after water quenched treatment in [11]. Using this value, the factor of safety significantly drops, to a value of \(620/464 = 1.3\).

3.2.5 Initiation of Yielding

Yielding is expected to occur at the location of maximum \(\sigma_{VM}\). The \(\sigma_{VM}\) equation in 2 dimensions presented above reveals that \(\sigma_{VM}\) can be a maximum in multiple scenarios, since the principal stress terms are squared. In other words, if \(\sigma_1\) is positive and \(\sigma_2\) is negative, the same \(\sigma_{VM}\) will be found as with \(\sigma_1\) negative and \(\sigma_2\) positive (with the same magnitudes). This means that \(\sigma_{VM}\) can be maximum at two
points at crank arm $\theta = 90^\circ$, $y = 0, z = R$ or $y = 0, z = -R$. These points correspond to the maximum bending stresses in both the tensile and compressive directions, respectively. One might say that any one of these points observes both compressive and tensile stress in a full crank rotation; however, this is not true for the present case, since it is assumed that each spindle only transmits torque for one half of the crank arm rotation. So, point $y = 0, z = R$ would yield due to maximum $\sigma_{VM}$ due to tension, and $y = 0, z = -R$ would yield due to maximum $\sigma_{VM}$ due to compression. We believe the reason why $y = 0, z = R$ yields first in the present case is because the case hardened layer is more brittle than it is ductile, thus, the yield strength is not the same in tension and compression (it is less for tension).

It is useful to note that, in actuality, the failure occurs at a location counterclockwise to $y = 0, z = R$. We believe this slight difference occurs as a result of a stress concentration, most likely due to uneven torque transmission from the crank arm to the spindle. This caused stress amplification at the location slightly counterclockwise to $y = 0, z = R$ to a value above that of 464 MPa, even though the theoretical stress at that location is lower (it has the same $\tau_x$, yet it has smaller $\sigma_{xx}$ due to being closer to the neutral axis of the section). Other possible explanations for this occurrence are the presence of a defect, such as a gauge or scratch, or environmental assisted degradation at the surface of the locations slightly counterclockwise to $y = 0, z = R$.

4.0 TASK 3

4.1 Hardness Measurements

Hardness values for the spindle were provided in the Vickers scale, as summarized in Table 3. The hardness values were measured on the case hardened surface, which was reported to “resemble” flame or induction hardened steel. The results reported have a range of 183 Vickers points, which can be associated with a range in tensile strength of approximately 600 MPa, which suggests that the measurements are a poor representation of the actual hardness. Although the source of this variability cannot be confirmed, it may be a result of poor surface preparation or excessive load selection (i.e. partial deformation of core material).

Table 3: Summary of hardness measurements

<table>
<thead>
<tr>
<th>Measurement</th>
<th>Hardness, HV</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>695</td>
</tr>
<tr>
<td>2</td>
<td>800</td>
</tr>
<tr>
<td>3</td>
<td>878</td>
</tr>
<tr>
<td>4</td>
<td>712</td>
</tr>
<tr>
<td>5</td>
<td>753</td>
</tr>
<tr>
<td>Average</td>
<td>768</td>
</tr>
</tbody>
</table>
During flame or induction hardening, the surface layer is heated to the austenitization temperature and then rapidly quenched to form a layer of very hard martensite. Regardless of the source of error, hardness tests confirmed that the surface was martensitic, which would contribute to abrasion and wear resistance. The phase transformation from face centered cubic to body centered tetragonal involves a volume expansion, which will result in a residual compressive stress. Although this will not change the total stress amplitude, only tensile stress is considered for fatigue crack propagation rates. Residual compression will also help to resist fatigue crack initiation by lowering the mean cyclic stress. A popular method for accounting for this is by preparing a plot of stress amplitude versus mean stress for a given cycle life (known as a Haigh diagram). A Goodman line is drawn by connecting the value of fatigue limit to the ultimate strength, as seen in the example in Figure 24 [3]. The Goodman line defines a safe working envelope for any combination of mean and amplitude of stress. The diagram clearly illustrates that a decrease in mean stress and stress amplitude (tending to the lower left of the graph) will decrease the likelihood of fatigue failure.

![Figure 24: Sample Haigh diagram with Goodman line [3]](image)

### 4.2 SAE 1045 Mechanical Properties: Estimated Upper and Lower Limits

Sound engineering judgment has been used to identify the failed bike spindle material as SAE 1045. This determination was based on the following available information concerning:

- visible inspection of failed component and fracture surface
- the morphology of the fracture surface (SEM images)
- the chemical composition of the material on the fracture surface (EDX analysis)
- surface hardness testing (of case hardened surface)

Table 4 summarizes the typical mechanical properties (with upper and lower limits) of SAE 1045 in a hot-rolled condition. Justification and verification of these material limits is explained.

### Table 4: Estimated upper and lower limit material properties

<table>
<thead>
<tr>
<th></th>
<th>Yield Strength ($\sigma_y$) [MPa]</th>
<th>Tensile Strength ($\sigma_{UTS}$) [MPa]</th>
<th>Elongation [%]</th>
<th>Stress Intensity ($K_{IC}$) [MPa(\sqrt{m})]</th>
<th>Fatigue limit [MPa] (S-N curve)</th>
</tr>
</thead>
<tbody>
<tr>
<td>typical</td>
<td>310</td>
<td>565</td>
<td>16</td>
<td>60</td>
<td>310</td>
</tr>
<tr>
<td>lower</td>
<td>300</td>
<td>570</td>
<td>14</td>
<td>58</td>
<td>300</td>
</tr>
<tr>
<td>upper</td>
<td>450</td>
<td>700</td>
<td>30</td>
<td>87</td>
<td>450</td>
</tr>
</tbody>
</table>

### Justification and Verification

**Material Specification Datasheet**

Reported material limit values for SAE 1045 yield strength, tensile strength and elongation have been referenced from an Interlloy material data sheet [4].

Given SAE 1045’s suitability for hardening, these material properties can vary greatly. A study performed by Zhang et al [5] illustrates this variability in SAE 1045 round shaft specimens (with a diameter of 35mm and a nominal induction-hardened case depth of 6mm), as shown in Figure 25.

**Stress Intensity Factor**

Verification of the reported typical stress intensity factor can be carried out by, for instance, considering the spindle as a plate loaded in tension with an edge crack [6], such that:

$$K_{IC} = \sigma_y \sqrt{\pi a_c} \cdot f(g)$$

- $K_{IC}$: stress intensity factor
- $\sigma_y$: yield strength
- $a_c$: critical crack length
- $f(g)$: geometry function

If $f(g) = 1$ is assumed, then by back-calculation it can be shown that $a_c \approx 12$mm.

$$60 = \frac{310}{\pi a_c}$$

$$a_c = \frac{1}{\pi} \left( \frac{60}{310} \right)^2 = 0.012$$
Thus the reported typical stress intensity factor is considered reasonable. Lower and upper stress intensity factor limits can be obtained by scaling ratios, such that:

\[ K_{IC,LIM} = \sigma_{y,LIM} \left( \frac{K_{IC,TYP}}{\sigma_{y,TYP}} \right) = \sigma_{y,LIM} \left( \frac{60}{310} \right) \]

Hence \( K_{IC,LWR} = 58\text{MPa} \) and \( K_{IC,UPR} = 87\text{MPa} \) respectively.

Fatigue Limit

The fatigue limit is the stress amplitude that can be applied to a material that will not cause fatigue failure. The reported typical fatigue limit is equal to the yield strength is a reasonable approximation. Figure 26 shows an S-N curve for SAE1045, from a study by Altenberger et al [7]. For a conventional deep-rolled condition, the fatigue limit is approximately 270MPa after \( 1 \times 10^6 \) cycles.
4.3 Fracture Mechanics Evaluation

Linear elastic fracture mechanics was used to evaluate the final fracture of the spindle. For simplification, the geometry was assumed to be cylindrical. Loading likely occurred in multiple modes (opening and shear), which makes fracture mechanics analysis very difficult. Since the modes are likely in phase and proportional, it was assumed that only Mode I is experienced in the direction of principle stress.

The stress intensity, $K$, ahead of a crack originating at the circumference of a solid cylinder can be described using the following relationship:

$$K = Y\sigma\sqrt{\pi a}$$

$Y =$ geometry factor
$\sigma =$ far field stress
$a =$ crack length at center

For bending applications, the geometry factor is defined by Reference [3] as follows:

$$Y = A + Bx + Cx^2 + Dx^3 + Ex^4 + Fx^5 + Gx^6$$

$x = a/D$
$A = 0.666$
$B = -1.2628$
$C = 10.737$
$D = -50.539$
$E = 139.29$
$F = -183.85$
The dimensions of the fracture plane were physically measured from the fractured spindle. As such, the crack length at the point stress intensity reached the fracture toughness of the material (i.e. final overload) was approximately 10.0 mm and the diameter was approximated as 14.5 mm. Figure 27 shows the variation in stress intensity as the crack grows. The far field stress that would be required to reach a stress intensity of 60 MPa m\(^{1/2}\) at a crack length of 10 mm was calculated to be 223 MPa. This stress is the principal stress \(\sigma_1\) perpendicular to the crack plane. In order to find the load \(F_0\) which creates this \(\sigma_1\), we must first back-calculate to find the bending stress and shear stress in the imaginary 2-dimensional plane discussed in section 3.2.1. We know that the angle between the crack plane and the perfect spindle cross-section is 36.1° and that \(\sigma_y\) in the imaginary plane must be zero. The parameters we would like to solve for are \(\sigma_x\) from bending and \(\tau_{xa}\) from torque. Using the stress transformation equations (basis of Mohr’s circle), it is not possible to find \(\sigma_x\) and \(\tau_{xa}\) without knowing a \(\sigma_2\) value. This is the lateral principal stress acting in in the direction of the crack width, perpendicular to \(\sigma_1\). From Figure 22, we know it is non-zero, yet the back-calculation cannot be performed without assuming a value for it. So, we assume \(\sigma_2\) to be zero (this reduces the accuracy of the end result of \(F_0\)). Table 5 below shows the excel spreadsheet used to find \(\sigma_x\) and \(\tau_{xa}\) with the solver function, where the green parameters are known, the amber parameter is assumed, and the red parameters are what is being solved for. The two constraint equations are written on the right. Note that the principal angle is multiplied by 2, since the angle in the stress transformation equations (and Mohr circle) is double that of the actual principal angle shift.

Table 5: Setup used to back-calculate sigma_x and tao-x-theta from sigma_1

<table>
<thead>
<tr>
<th>theta</th>
<th>72.2</th>
<th>3.114635</th>
<th>3.114635</th>
<th>(\tan(2*\theta_{\text{principal}}) = \frac{(2*\tau_0)}{(\sigma_x - \sigma_y)})</th>
</tr>
</thead>
<tbody>
<tr>
<td>sigma_x</td>
<td>104.4195</td>
<td></td>
<td></td>
<td>(\sigma_1 = \frac{(\sigma_x + \sigma_y)}{2} + \sqrt{\left((\sigma_x - \sigma_y)/2\right)^2 + \tau_0^2})</td>
</tr>
<tr>
<td>sigma_y</td>
<td>0</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>sigma_1</td>
<td>223</td>
<td></td>
<td>223</td>
<td>(\Rightarrow)</td>
</tr>
<tr>
<td>sigma_2</td>
<td>0</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>tao</td>
<td>162.6144</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

The solved \(\sigma_x\) and \(\tau_{xa}\) values are then used in the mechanical analysis spreadsheets to solve for \(F_0\). The \(F_0\) value found is 322 N (72 lb), which seems like a reasonable amount of force to apply to the pedal while accelerating from a stopped position.
4.4 Sensitivity Analysis

A simplified sensitivity analysis was created to show the ranges of realistic values that may have been encountered had the initial $F_0$ force been different. Also the fracture mechanics were adjusted for the upper and lower bounds of the material properties discussed earlier. The figures below show the results for the base case of $F_0$ equal to 500N and the variations of adjusting this value by +30%, +15%, -15%, and -30%. The range for the max tensile stress seen at (0, R) is 113 to 210 Mpa. The shear torque ranges from 175 to 325 Mpa. The Von Mises Stress ranges from 325 to 602 Mpa.
Sigma xx @ (0,R)

Shear Torque
The material property boundaries were also used to estimate usable ranges for the values of fracture toughness and correspondingly the back calculation for force needed for fracture.
It should be noted that the values for pedaling force and shear all fall within a realistic range of applicable magnitudes.

5.0 TASK 4

5.1 Data Presented

As discussed in previous sections of this report, the pictures and data provided had some obvious flaws that may create confusion. Most simple to correct was that the angle of the fracture surface with respect to the spindle axis was not portrayed in the photographs. Fatigue cracking will typically propagate perpendicular to the principle tensile stress, so an angled fracture surface indicates that multi-axial loading was involved. This omission could be accounted for by including a profile view, as has been included in this report.

The SEM images provided were mediocre quality and did not seem complete. First, the initiation location was mostly obscured by charging of foreign debris on the surface. Sometimes tracking the initiation point can reveal a stress concentration from geometry, external damage, or metallurgical defects. It was not the case in this instance, but it was not made apparent from the picture provided. Therefore, supplemental SEM imaging was performed to obtain higher quality images. The SEM examination revealed brittle intergranular cracking of the case hardened layer, and the presence of numerous large inclusion defects (identified by EDX as manganese and silicon oxides). Both of these attributes help characterize the fracture, and may play a potential role in determining a sequence of events leading to failure.
The EDX analysis provided was variant and inaccurate. As mentioned, EDX is not ideal for chemical composition analysis so cannot be used to identify the material. Three of the five measurements reported over 30% carbon, and one measurement reported pure iron. The source of the error cannot be confirmed, but it may be due to contamination of the surface, or unfavorable surface topography.

The hardness measured fell in a very broad range (183 Vickers points). Since hardness can be used to estimate tensile strength of ferritic materials, this relates to a range of about 600 MPa. It is very unlikely that such a variation would be encountered, but it does confirm that the surface would be quite hard.

5.2 Analysis Assumptions

Key areas analysis assumptions and simplifications concern:

- loading:
  - modeling half of the crank revolution, assuming that on the back stroke (180° to 360° crank angles) that the other pedal is fully taking the load
  - consideration of only dominant stress contributions (bending and torsion)

- geometry:
  - simplifying the spindle cross-section geometry

- material properties:
  - determination of the spindle material
  - estimation of typical tensile strength and fracture properties

- calculations:
  - assuming a zero value for \( \sigma_2 \) when calculating \( F_0 \)

5.4 Failed Spindle Design and Material Choice

The failed bike spindle was a solid square tapered design that we reasonable deduce has been manufactured from SAE 1045 (a heat treated plain carbon steel).

With respect to the geometry of this spindle, this part may have been over-engineered. Considering the age of the bike (1979 Rayleigh Super Course), it may not have been a critical design driver to design low weight bottom bracket assemblies. Presumably this bike’s frame would have been a steel frame. However, the basic mechanical analysis in part 2 of this report indicates that the maximum stresses (due to bending and torsion) occur at the circumference of the spindle, thus suggesting the spindle could have been designed with a hollow cross-section. Clearly it is unlikely that the cross-section of the spindle has not contributed to the failure of the spindle.

A feature of the spindle design which may have contributed to the catastrophic failure is the transition from the square taper to the circular cross-section. If such a transition is sudden, it could be considered a stress concentration ("stress-raiser").
Given that the spindle operates in an environment which is subject to mechanical wear, it is reasonable to expect that the material is surface treated to resist abrasion. In this instance, the spindle appears to have been case hardened.

What we are less uncertain about is the history of the bike. That is, in what conditions was it stored, was the bike regularly maintained and the riding environment, rider behavior, whether it was continually ridden on flat-roads, uphill or a combination of both.

5.5  Avoiding Future Failures

Although 30 years would be considered quite a long life for a mechanical component in sports equipment, fatigue issues could have been avoided. A primary concern is the abundant content of oxide inclusions. Their presence will reduce the tensile strength of the material, and introduce stress concentrations leading to early initiation of a fatigue crack. The weak interfaces and differing deformation characteristics will also accelerate the rate of propagation under normal loading conditions. These deviations from the expected properties make it difficult to design around fatigue failure, and must be prevented during production of the material.

Material selection for low-risk components will often have a considerable influence of cost. For example, a catastrophic failure of a bike spindle would be very unlikely to cause an injury to the rider, so the manufacturer may use less expensive material that may limit its longevity. For a larger investment, the manufacturer might have selected a precipitation hardened stainless steel such as 17-4 PH. This alloy can be hardened to resist abrasion or wear of the surface, and would be corrosion resistant in rainy conditions.

5.6  Modern Bike Spindle Design

The key design drivers for road bike spindle design are:

- light weight
- sufficient strength
- sufficient stiffness (an increasingly more desirable requirement)
- durability
- interoperability / compatibility (interfacing components / manufacturers)

Prior to the square tapered spindle design (as per the failed spindle in this case study) cranks were often connected to the spindle via a cotter pin. This particular design is now considered obsolete. Table 5 summarizes bike spindle concepts typical in road bike designs. Current spindle design evolution owes much from the design and materials selection evolution from BMX bike technology [8],[9].
Table 6: Summary of current road bike spindle design

<table>
<thead>
<tr>
<th>Material Selection</th>
<th>Square / Hexagonal Taper (Cotterless)</th>
<th>Spline Internal Bearing</th>
<th>Spline External Bearing</th>
</tr>
</thead>
<tbody>
<tr>
<td>Strength</td>
<td>Y</td>
<td>Y</td>
<td>Y</td>
</tr>
<tr>
<td>Stiffness</td>
<td>N</td>
<td>Y</td>
<td>Y</td>
</tr>
<tr>
<td>Durability</td>
<td>Y</td>
<td>N</td>
<td>Y</td>
</tr>
<tr>
<td>Spindle Cross Section</td>
<td>Solid</td>
<td>Hollow</td>
<td></td>
</tr>
<tr>
<td>Spindle Diameter</td>
<td>~12mm</td>
<td>~24mm</td>
<td>~30mm</td>
</tr>
<tr>
<td>BB Integration</td>
<td>N</td>
<td>N</td>
<td>Y</td>
</tr>
<tr>
<td>Example</td>
<td>Failed Spindle (Case Study)</td>
<td>Shimano Octalink V1</td>
<td>Campagnolo Ultra-torque</td>
</tr>
<tr>
<td></td>
<td></td>
<td>ISIS Open Design</td>
<td>Cannondale BB30</td>
</tr>
</tbody>
</table>

The differences in square taper versus spline in typical road bike spindle design are shown in Figure 28.

Figure 28: L-R: (i) a traditional square taper (hollow), (ii) Shimano Octalink V1, (iii) Shimano Octalink V2, (iv) ISIS bottom bracket interfaces [10]

Geometry

A significant observation is the shift from a solid cross-section to a hollow cross-section. This is not only a weight-savings win but it also indicates that some form of mechanical analysis (as per task 2 of the case study) has been considered and performed. Hence the design decision to remove material from the spindle core is sound since the maximum principal stresses occur at the circumference of the spindle. Bottom Bracket (BB) integration refers to the bottom bracket assembly (containing spindle and bearings) and crank set being considered a 2-piece assembly rather than 3-piece. Clearly a reduced part count is also considered an important cost saving measure which is being considered in current and future designs.
**Material Choice**

Typically bike parts, particularly rotating components are subject to impact damage, mechanical wear, abuse (riding behavior and maintenance) and environment (rain, mud, corrosion attack). Material selection, in this instance, is often based on weight, strength, stiffness, wear resistance, corrosion resistance and machinability.

Titanium components are often used in mid-to-high end road bike applications. Such materials offer a significant weight advantage, but are not always as strong or as stiff compared to alloy steel equivalents. Titanium is also more costly than alloy steel. Similarly, stainless steels are more costly than alloy steel and never rust. Alloy steels require appropriate heat treatments and surface coatings to ensure that they are wear resistant.

The shift towards larger spindle diameters not only satisfies stiffness requirements sought by riders, but it also enables designers to consider materials that would have otherwise been unfit-for-purpose. An example is the introduction of an Al-alloy in the Cannondale BB30 design. This is also a significant design shift as it enables manufacturers to consider and select alternate materials as they seek to lower the cost of production whilst maintaining acceptable part quality.

**6.0 Conclusion**

Through careful examination of the data provided and some independent investigation, it was determined that the bicycle crank arm spindle failed due to initiation and propagation of a fatigue crack. The fatigue crack propagated approximately 69% through the thickness before rupture of the remaining ligament.

The angle of the fracture surface suggested that multi-axial loading was responsible, which was confirmed by stress analysis. The loads encountered include shear, bending, and torsion. The most critical stress was due to bending and occurred when the crank was rotated 90 degrees forward from the vertical position. This agreed well with observations of the fracture surface, because in this position, the origin of the fatigue crack was located at the point of greatest calculated stress.

Bands of corrosion product across the fatigue fracture surface suggest that the growing crack was periodically exposed to moisture. Oxidation at the tip of a growing crack can accelerate propagation rates via corrosion fatigue. It was also observed that there was a very high content of oxide inclusions. Oxide inclusions will introduce stress concentrations, facilitating crack initiation, and have weak interfaces, facilitating crack propagation. The oxides are a result of poor quality control during production of the steel.
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