Effect of the coefficient of friction on the fatigue life of splines

C. H. H. Ratsimba, S. B. Leen, I. R. McColl & E. J. Williams
School of Mechanical, Materials, Manufacturing Engineering and Management, University of Nottingham, UK

Abstract

The fatigue behaviour of a representative high-performance aeroengine spline coupling is investigated under test conditions designed to simulate in-service conditions. The test load cycles consist of major cycle torque and axial load, simulating maximum thrust, combined with minor cycle rotating bending moment and fluctuating torque, simulating life-limiting conditions at take-off. The objective of the study is to develop understanding of the fatigue behaviour of the coupling over a range of loading conditions, including interaction between low-cycle fatigue, fretting fatigue and fretting wear. The effect of the coefficient of friction on the fatigue life is also investigated. This information is necessary for the development of fatigue and fretting fatigue life prediction techniques. The test results are interpreted with the help of three-dimensional finite element models, which include the frictional contact between the spline teeth.

1 Introduction

Spline couplings are commonly employed on aeroengine mainshafts, such as the low-pressure and intermediate pressure compressor to turbine shafts of triple-spool, high-bypass gas turbine aeroengines. Figure 1 shows a typical schematic of a spline coupling. Depending on the application, such splines can experience a wide range of loading conditions. They are designed primarily for torque and axial load transmission, but also experience rotating bending moments, particularly during take-off and landings, and other flight manoeuvres. Those loading conditions generally lead to fatigue, fretting fatigue and/or fretting wear controlled service lives [1] [2]. Accurate stress-strain distributions are required for fatigue lifing, whereas contact variables are at least as important for fretting
behaviour. Those contact variables are contact pressure, slip amplitude, and sub-surface stress and strain. It was previously shown that the coefficient of friction (COF) can have significant effect on the slip amplitude and the sub-surface stress, and consequently on the fretting behaviour [3] [4].

![Diagram of a coupling geometry showing contact points and dimensions](image)

**Figure 1:** (a) Schematic half-section of splined coupling geometry showing z co-ordinates, and (b) schematic of spline teeth showing tooth flank contact width, $a_2$, and tooth flank co-ordinate direction, $x$.

This paper is concerned with the fatigue testing of a reduced scale aeroengine type spline coupling, under test conditions designed to simulate in-service loading and the effects of cyclic torque overload. Such data is essential for improved understanding of spline failure mechanisms and for the validation of
modelling approaches. The spline fatigue results are interpreted using finite element (FE) calculated cyclic stress-strain distributions and tension-tension uniaxial fatigue data. This paper will also investigate the extent to which the COF can also affect the plain fatigue behaviour of splines.

2 Fatigue testing and behaviour

The helical spline couplings and tension-tension fatigue specimens were manufactured from a high strength CrMoV alloy steel, based on BS 3S 132 but subjected to cleaner processing, with the composition shown in Table 1. Prior to final machining, the components were through hardened to a specific hardness $H_{\text{spec}}$. After final machining, the externally-splined shaft and tension-tension fatigue specimens were gas nitrided to a minimum metal hardness of $1.8 \times H_{\text{spec}}$ over a minimum depth of 20 $\mu$m.

The spline couplings have 18 teeth on a reduced diameter, relative to full-size aeroengine splines, to make them suitable for the shaft test facility employed. However, the spline teeth are typical of those employed on full-size couplings. Figure 1 shows the spline geometry, together with definitions of the contact axial length co-ordinate $z$, the tooth-flank contact width co-ordinate $x$ and the maximum possible contact dimensions, $a_1$ and $a_2$. The splines were not actively lubricated during testing, but did receive a coating of Mobil® Jet Oil II prior to assembly, consistent with in-service practice.

The spline test simulates the key life controlling loading regime experienced by the splines during a typical civil flight envelope, comprising the major and minor cycle loading sequence. The detail of the loading sequence was presented in previous communication [3] and the present paper will only discuss the effect of the major cycle torque.

The tension-tension fatigue specimens were of conventional hourglass form with a minimum gauge diameter of 5.00 ± 0.05 mm, and were tested in a servo-hydraulic machine at a frequency of 10 Hz under load control at an $R$ ratio of 0.1 ($R = \sigma_{\text{min}}/\sigma_{\text{max}}$).

Table 1: Composition of CrMoV steel (mass%).

<table>
<thead>
<tr>
<th>C</th>
<th>Si</th>
<th>Mn</th>
<th>P</th>
<th>S</th>
<th>Cr</th>
<th>Mo</th>
<th>Ni</th>
<th>V</th>
<th>Fe</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.35-0.43</td>
<td>0.1-0.35</td>
<td>0.4-0.7</td>
<td>&lt;0.007</td>
<td>&lt;0.002</td>
<td>3.0-3.5</td>
<td>0.8-1.10</td>
<td>&lt;0.3</td>
<td>0.15-0.25</td>
<td>Remainder</td>
</tr>
</tbody>
</table>

Table 2 summarises the spline fatigue test results, showing the applied major cycle torque, the estimated number of cycles to crack initiation, and the tooth number and axial positions of cracking. Cracking was only observed in the externally-splined shafts.

Tests 1 and 2 were carried out under identical loading conditions, i.e. 39% torque, and demonstrate good repeatability with respect to estimated number of major cycles to crack initiation, i.e. about 450 cycles in each case. Knowledge of the expected failure time from Test 1 permitted observation of the early cracking behaviour during Test 2, when the cracks were observable using a
magnification of 3. The cracks initiated in the tooth root in the proximity of the end of engagement at the $z = a_1$.

For Tests 3 to 5, reduced overload magnitudes were applied. The resultant fatigue failures were almost identical to those of Tests 1 and 2, but at increased numbers of cycles, as shown in Table 2.

Table 2: Summary of loading conditions, estimated numbers of cycles to failure.

<table>
<thead>
<tr>
<th>Test parameter</th>
<th>Test 1</th>
<th>Test 2</th>
<th>Test 3</th>
<th>Test 4</th>
<th>Test 5</th>
<th>Test 6</th>
</tr>
</thead>
<tbody>
<tr>
<td>% major cycle axial overload</td>
<td>24%</td>
<td>24%</td>
<td>39%</td>
<td>30%</td>
<td>20%</td>
<td>0%</td>
</tr>
<tr>
<td>% major cycle torque overload</td>
<td>39%</td>
<td>39%</td>
<td>39%</td>
<td>30%</td>
<td>20%</td>
<td>0%</td>
</tr>
<tr>
<td>Estimated no of major cycles to initial crack</td>
<td>450</td>
<td>450</td>
<td>530</td>
<td>1500</td>
<td>3450</td>
<td>$&gt;3.5 \times 10^4$</td>
</tr>
<tr>
<td>Axial position ($z$) of crack initiation</td>
<td>$z = a_1$</td>
<td>$z = a_1$</td>
<td>$z = a_1$</td>
<td>$z = a_1$</td>
<td>$z = a_1$</td>
<td>N/A</td>
</tr>
</tbody>
</table>

Test 6 used the design loads. When the test was stopped at 35,000 major cycles there were no observable cracks, although there was significant wear along the tooth contact regions.

Relationship between stress range, normalised by the 0.1% proof strength, $\sigma_y$, versus number of cycles to failure for the $R = 0.1$ uniaxial fatigue tests were experimentally determined. The data were fitted using:

$$\left( \frac{\sigma_r}{2\sigma_y} \right) = \sigma_c N_f^{a}$$

where $N_f$ is the number of cycles to specimen failure, $\sigma_r$ is the applied stress range, and $\sigma_c$ and $a$ are material constants. Due to scatter in the stress-life data, upper and lower bound fits were made, using different stress axis intercepts, $\sigma_c$, with the same slope, $a$. The upper and lower bound fitted values of $\sigma_c$ are 0.646 and 0.6157, respectively, and the value of $a$ is $-0.0524$.

3 Finite element modelling

For the simulation of major cycle torque presented here, the use of cyclic symmetry boundary conditions permit one-tooth modelling of the spline which facilitates a high level of mesh refinement and thus more refined results. Details of the modelling, including implementation of the helix angle and the axial profile modification, for the spline coupling considered here, have been reported previously [4] together with mesh refinement studies which lead to the converged mesh shown in Figure 2. Eight-noded brick elements are employed, as recommended for reliable prediction of contact variables [5]. The spline root torsional stress distribution predicted by the cyclic symmetry model has been
validated [4] by comparison with photoelastic results and the boundary element (BE) results of McFarlane et al. [6].

All of the FE analyses reported here used Young’s modulus and Poisson’s ratio values of 210 GPa and 0.28, respectively, Coulomb friction applied using the penalty method with an allowable elastic slip of 0.0005a2 [5] and elastic-plastic behaviour derived from a measured stress-strain curve [4].

The present aeroengine splined coupling is not actively lubricated; they just receive a coating of lubricant before assembly. However, cylinder-on-flat fretting wear tests carried out by the authors (to be reported on elsewhere) over a range of representative contact loads and applied stroke combinations, without lubrication, have indicated that the coefficient of friction settles down, after about 18,000 cycles, to an average steady-state value of 0.5 to 0.7. On the other hand, well-lubricated contact gave rise to coefficient of friction value of 0.15. Detailed observations of the spline after testing has shown that the lubricant is to some extent expelled from the contact region. That would suggest that the friction behaviour in the spline might evolve from lubricated to dry friction during the test. Therefore, a series of analyses are carried out for friction coefficient values of 0, 0.3, 0.5 and 0.7 to cover the range of potential spline operating conditions.

![Figure 2: Cyclic symmetry, one-tooth, FE model of splined coupling.](image)

**4 Stress distribution**

A local axis system is employed here for stress transformation, with the local 1 direction in the x-direction of Figure 1, the local 3 direction normal to the tooth flank surface in the plane transverse to the spline axis, and the local 2 direction makes a right-handed set with the other two.

The trailing-edge normal stress parallel to the surface, often referred to as the tangential stress has been highlighted as particularly important to fretting damage [7]. In that region, for x = a2, this tangential stress is both influenced by the tangential friction and the tooth bending. The latter occurs in the spline fillet region and extends upwards along the tooth flank, directly affecting the aforementioned trailing-edge normal stress. Figure 3 shows the distribution of
the tangential stress $\sigma_{11}$, for $z/a_1 = 0.99$, in the contact width direction, i.e. from one tooth tip to the other, for a range of coefficient of friction covering observed experimental variation. The magnitude of loading of Test 6 is simulated. The sharp compressive to tensile transition at the trailing edge can be attributed to frictional contact as well as tooth bending. It can also be observed that the stresses are more tensile on the contact side fillet than it is compressive on the non-contact side fillet for non-zero coefficient of friction. This is due to the superimposed effect of the effect of the friction on the effect of the tooth bending. Comparing the stress distribution between frictionless condition (COF equal to 0) and the non-zero friction conditions illustrates the significant effect of the friction. It should also be noted that the stresses increase more for COF between 0.3-0.5 than 0.5-0.7. This confirms previous report [4] that the smaller increase in stress for COF between 0.5–0.7 is related to zero slip for 0.7, which limits the increase in tangential stress.

Figure 3: Distribution of local $\sigma_{11}$ stress along the tooth flank, fillet and root at axial position $z/a_1 = 0.99$.

Figure 4 shows the principal stress distribution in the contact width direction for Test 6, just outside the end of engagement for $z = 0.99 \, a_1$, for identical range of COF as for Figure 3. The effect of the COF is less significant than for the tangential stress shown previously but it could nevertheless have a significant effect on the fatigue life. There is a 6% increase in peak maximum principal stress between a frictionless and 0.7 coefficient of friction. The high peaks in the spline root are biased towards the contact side fillet which correlates with the observed location of crack initiation site mentioned in section 2. It should also be
noted that the trailing edge tangential stress peaks observed on Figure 3 is not observed here because the torsional shear stress has the major contribution to the maximum principal stress.

Figure 4: Distribution of maximum principal stress along tooth flank, fillet and root at axial position \( z/a_1 = 0.99 \).

5 Spline life prediction

In order to make spline life predictions for the load cases of Tests 1 to 6, the FE-predicted maximum principal stress \((R = 0)\) in the spline root is employed in Equation (1), together with the upper and lower bound intercept \((\sigma_i = 0.646 \text{ and } 0.616)\) and slope \((\alpha = -0.0524)\) values obtained from the load controlled uniaxial fatigue data. Figure 5 compares the upper and lower bound predictions with the experimental spline coupling fatigue data, and shows good correlation. At first sight, this may seem surprising since load controlled uniaxial fatigue data were employed. However, the spline coupling fatigue tests were also load controlled and, as noted earlier, the spline tooth root stress concentration is modest \((K_t = 1.6)\), so that the local load regime would be expected to be predominantly stress controlled. Also, FE investigation previously reported [8] showed that shakedown to linear elastic behaviour is predicted after the first cycle. This suggests that the use of the maximum principal stress in the life calculations is not unreasonable.
Figure 5: Comparison between predicted and measured spline fatigue lives in terms of log number of major cycles versus log maximum principal stress range.

A coefficient of friction of 0.3 was employed to obtain the spline root stresses for the present life prediction. Figure 4 shows that there could be an increase of up to 6% in stress due to increased COF, which could have a significant effect on the spline fatigue life. An increasing COF due to an evolution from lubricated to dry friction during the spline test is the likely friction behaviour of the spline and the implications on the fatigue life reduction needs to be investigated.

6 Effect of coefficient of friction on spline lives

It was previously shown [4] that the variation of COF, can have a pronounced effect on salient macroscopic fretting variables i.e. relative slip, contact tractions and sub-surface stresses. Figure 4 shows friction can also affect the stress magnitudes down in the spline root and consequently the torque induced fatigue failure discussed in the present paper.

Equation (1) for the upper and lower bound, determined from uniaxial test, was used to calculate the reduction in life with increasing COF. The corresponding plot is shown Figure 6. We observe a reduction in life of more than a factor 2 from a COF of 0 to 0.7, taking the loading condition of Test 6.
Figure 6: Estimated effect of the coefficient of friction on spline lives.

7 Conclusions

The present paper has shown the following:

(a) Under overload conditions the spline coupling failed by plain fatigue in the spline tooth root-fillet region at the high torque end of the externally-splined shaft, just outside the length of engagement.
(b) Good correlation is obtained between the spline and tension-tension uniaxial fatigue test results, interpreted via the finite element predicted maximum principal stress range.
(c) There is significant effect of an increasing coefficient of friction on the reduction of the spline life. The frictional contact influences the maximum principal stress down in the spline tooth root-fillet, at the end of axial engagement, which is the observed site of crack initiation.

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Fatigue Damage of Materials: Experiment and Analysis

References


